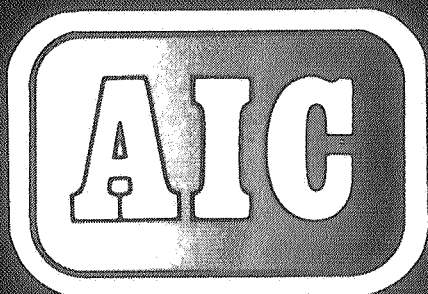


INTERNATIONAL ENERGY AGENCY
energy conservation in buildings and
community systems programme

3rd AIC Conference

Energy Efficient Domestic Ventilation Systems for Achieving Acceptable Indoor Air Quality

Proceedings



Air Infiltration Centre

Old Bracknell Lane West, Bracknell,
Berkshire, Great Britain, RG12 4AH

THE AIR INFILTRATION CENTRE was inaugurated through the International Energy Agency and is funded by ten of the member countries:

Canada, Denmark, Italy, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and United States of America.

The Air Infiltration Centre provides technical support to those engaged in the study and prediction of air leakage and the consequential losses of energy in buildings. The aim is to promote the understanding of the complex air infiltration processes and to advance the effective application of energy saving measures in both the design of new buildings and the improvement of existing building stock.

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3rd AIC Conference

**Energy Efficient Domestic
Ventilation Systems for
Achieving Acceptable
Indoor Air Quality**

(held at the Park Court Hotel
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Proceedings

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PREFACE

International Energy Agency

In order to strengthen cooperation in the vital area of energy policy, an Agreement on an International Energy Program was formulated among a number of industrialised countries in November 1974. The International Energy Agency (IEA) was established as an autonomous body within the Organisation for Economic Cooperation and Development (OECD) to administer that agreement. Twenty-one countries are currently members of the IEA, with the Commission of the European Communities participating under a special arrangement.

As one element of the International Energy Program, the Participants undertake cooperative activities in energy research, development, and demonstration. A number of new and improved energy technologies which have the potential of making significant contributions to our energy needs were identified for collaborative efforts. The IEA Committee on Energy Research and Development (CRD), assisted by a small Secretariat staff, coordinates the energy research, development, and demonstration programme.

Energy Conservation in Buildings and Community Systems

The International Energy Agency sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, etc. The difference and similarities among these comparisons have told us much about the state of the art in building analysis and have led to further IEA sponsored research.

Annex V Air Infiltration Centre

The IEA Executive Committee (Buildings and Community Systems) has highlighted areas where the level of knowledge is unsatisfactory and there was unanimous agreement that infiltration was the area about which least was known. An infiltration group was formed drawing experts from most progressive countries, their long term aim to encourage joint international research and to increase the world pool of knowledge on infiltration and ventilation. Much valuable but sporadic and uncoordinated research was already taking place and after some initial ground-work the experts group recommended to their executive the formation of an Air Infiltration Centre. This recommendation was accepted and proposals for its establishment were invited internationally.

The aims of the Centre are the standardisation of techniques, the validation of models, the catalogue and transfer of information, and the encouragement of research. It is intended to be a review body for current world research, to ensure full dissemination of this research and based on a knowledge of work already done to give direction and a firm basis for future research in the Participating Countries.

The Participants in this task are Canada, Denmark, Italy, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and the United States.

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 1 - KEYNOTE ADDRESS

THE ART OF VENTILATION

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SYNOPSIS

The relationship between ventilation needs and methods and the growth of civilisation is traced. The Industrial Revolution is a particularly significant epoch, not least because it introduced new problems and provided new solutions. The study of physiological need began late in the development of the art.

It is questioned whether currently accepted ventilation criteria are still valid; it is suggested that ventilation is only one of several means of ameliorating the internal environment.

A requirement of any future solution is to eliminate uncontrollable variables such as wind and stack effects, and to secure the greatest collaboration between architect, builder and engineer.

1. INTRODUCTION

It so happens that when I began work, at BRS, in 1937, I was asked to study air infiltration, and in particular to correlate the air flow through window cracks with the pressure difference across them arising from wind.

I remember that the instruments were of the crudest. The manometer comprised a mica diaphragm on the cut-off base of a reagent bottle; the movement of the diaphragm was detected by a pivoted needle at its centre, and this carried a mirror so that the deflections could be photographically recorded. The response of the instrument was pretty fast; but it bore no relation to the time constant of the means of measuring air change. I have to admit that I failed completely to add one iota to our knowledge; and I was greatly relieved when I was instructed to turn to heat transfer.

On my retirement 6 years ago, I became interested in the historical development of building services, and this of course includes ventilation. The development of the art of ventilation brings to light first of all why ventilation is needed. Early civilisations, all centred around the Mediterranean and the Indus valley, were in areas where no heating was normally required. Ventilation was used rather for cooling (aided by evaporation of water from sprayed floors, fountains or damp curtains (tatties)) and to create air movement. Evaporative cooling was used as long as 3000 years ago in Assyria. Bernan¹ reports that, by means of the tattie, indoor temperatures were sometimes as much as 11°C below the outdoor shade temperature. The openings to admit light were seldom glazed, and there was clearly no question of there being insufficient fresh air supply.

"In the Hall of Baths in the Alhambra at Granada the roof is perforated with ventilating openings, and it is not only of the best possible form for the purpose of ventilation, but the openings themselves are of

the best possible shape, being wider at the lower extremity than at the upper; and in order that these openings may present the least possible amount of friction to the outgoing air, they are provided with short tubes of baked earth, covered with a green vitreous glazing" (Tomlinson²)

Ingenious devices were used to increase the natural flow through the buildings - the wind towers of Yazd (Iran) - and use was made of low ground temperatures to cool air by passing it through tunnels, or over water stored in underground cisterns. The latter has been used in Iran; the former was described by Palladio, and used in Italy in the 17th century. An attempt was made to ventilate a military hospital in Cawnpore by this means in 1824; it was also used by Strutt at the Derby Infirmary some years earlier.

When civilisation spread north and west into more temperate climates, building construction changed. Windows had to be closed, both to keep out the rain and to prevent high winds causing unpleasant conditions within. The Romans probably introduced "glazing" materials - not only glass, but also hides, paper, mica and so on. We see here a change of objective - ventilation was to be restricted rather than encouraged - brought about by climatic differences. We notice, too, that there was a separation of functions, between daylighting and air supply, whereas in the earlier times, in warmer climates, the wall openings supplied both. In later periods (c 1800), after the sash and hopper windows had been invented, the window was again supposed both to light and to ventilate. This duality of function was seen by some to be wrong.

"Heating and ventilation, especially the latter, seldom enter the mind of the builder when he projects his building; he begins as if he did not know that ventilation would be necessary; he trusts to doors and windows, to neither of which belongs the business of ventilation. The doors admit the occupants to the chambers; the windows the light; and apertures ought to be introduced to admit air for ventilation as regularly as the other openings" (Birkbeck, evidence to House of Commons Committee, 1835)²

In my view, it is from this duality that many of our present ventilation problems have sprung. Our best hope for the future is to follow Birkbeck's advice, to regard windows as light sources only, and to provide ventilation by other, possibly mechanical, means.

2. VENTILATION AND HEATING

Ventilation has, almost inevitably, been associated with heating^{1,4}. The brazier used by the Greeks and the Romans, and the central hearth of Celtic, Saxon and British homes, all produced smoke

which had to be removed. The smoke turret replaced the simple hole in the roof; the later wall fireplace and its chimney alleviated the smoke problem; but all these fires needed air both for combustion and for the removal of the smoke.

The wide flue necessary when wood (the then principal fuel) was burned in open grates also produced a large flow of air, with consequent draughts across the floor from doors and windows. Sir John Winter, in 1658, was the first to provide fresh-air ducts to supply combustion air direct to the fireplace. This attempt to separate combustion air from ventilating air was sound in principle; it was re-invented several times by fireplace designers in the following centuries; it has been "perfected" in the modern room-sealed gas appliance.

Warm-air heating (first used by the Romans in one of the versions of the hypocaust, and later in the "stein-ofen" of the monasteries of the 9th and 10th centuries) used air as the heat carrier. The initial versions made use of the warm flue gases, either directly admitted to the room (after the fire has been extinguished) or indirectly via channels in the wall. And when, in the late 16th and 17th centuries, closed stoves of various kinds were used, inventors sought either to increase the efficiency of the stove by using the flue to warm fresh air from outside for ventilation, or simply to use the stove to warm ventilating air, to avoid discomfort from cold draughts. In either case, the heating "system" provided the motive power for the ventilation. Savot, in his "Louvre" fireplace of 1624, was the first to use fresh-air ducts to supply ventilating air warmed by passage through a "jacket" behind the fireback. Gauger, in 1713, and later Leutmann (1723), developed the ideas of Savot and Winter, making use of both combustion-air and a warmed fresh air supply.

The lack of ventilation afforded by a stove was well-known, and was one reason for the discouragement of stoves in Britain. Bernan wrote in 1845:

"It is essential for the economical effect of a stove that the room be nearly airtight In apartments with ill-fitting, rickety doors and windows, and a large extent of ill-glazed surface, and thin heat-absorbing walls, abounding in wind-chinks - or in general in all cases where the apartment approximates to the exposure of a field, as many English rooms do - an open fire is to be preferred."

The 1857 Commissioners, in their Report³, said:

"We are led to the conclusions that the outer surface of the stove should be only moderately heated, and that both the heat of the stove and the temperature of the hot air should be under regulation and easy power of control; that there should at all times be a flue or pipe for the passage of the smoke and gases; that humidity and ventilation should be specially

promoted, the two latter independently of the heating action of the stove."

They said, too, "It is absolutely necessary that water be present in the air to the amount of a degree of humidity about 66". Other references to humidity, often contradictory, occur in later literature⁴, but until the ASHVE Comfort Zone was established some 50 years ago, there was little detailed information (and I suspect that was largely intuitive). Humidity as a factor in the incidence of condensation is a relatively modern phenomenon - as I recollect, there was some surprise at the occurrence of condensation when flueless gas heaters were used in the 1930's, and again at its appearance in post-war dwellings. Some at least of this can be traced to ignorance of elementary psychrometry and of the thermal and moisture properties of building materials and structures. These are just as important as humidity control in the battle against condensation.

3. THE INDUSTRIAL REVOLUTION

In the late 18th century, an important epoch began - the Industrial Revolution. It led to a drift from the land and increased urbanisation, and the replacement of cottage industry by factory work⁴. The houses built in the newly expanding towns for the factory workers were, no doubt, adequate for the period; but they were cheek-by-jowl, without proper sanitation or water supply, and they were ill-lit and ill-ventilated. (The window-tax, first imposed in Britain in 1696, and not formally repealed until 1851, was a significant factor in this). The very proximity of the dwellings, back-to-back or in terraces fronting narrow streets, reduced the cleansing effect of wind.

It was a period, too, of rapidly growing population. By the early 19th century, there was a need for more schools, and by the middle of the century, for municipal and public buildings. All these, and the factories, posed new ventilation problems, partly because of their size and density of occupation, and partly because of the demands of the processes being carried out (textiles, foundry work). There grew up a body of public "sponsors" - the newly-formed municipal authorities - which, sometimes for reasons of prestige, sought to introduce innovations in heating, ventilation and lighting into their premises; and this was, to my mind, a significant factor in the development of building services during that period. Experiment was possible.

The Industrial Revolution, however, also carried the seeds of the solution of the problems it caused - namely, motive power. Now early attempts at the ventilation of mines (c 16th century) had involved a variety of devices for moving air - bellows, windsocks and rudimentary fans operated by animals or men - and these have been described by Agricola. A man-driven fan was first used in a building by Dr Desaguliers in 1736 - at the House of Commons in fact. It was far too small, and was supplemented, and then replaced, by fires at the base of upcast shafts to induce

ventilation (a return to an earlier scheme of Desaguliers in 1723). Nevertheless, the fan was here to stay and it was widely used as a part of warm-air heating and ventilating systems in the larger buildings from the middle of the 19th century onwards. Steam engines were, of course, used to drive these early fans, but were gradually superseded by electric drive by the end of the century. Some attempts were made to operate the fan from a water turbine in the early 1900's.

4. VENTILATION AND HEALTH

That there was some connection between ventilation and health was known in the middle of the 18th century. The hopper window was introduced at St Thomas' Hospital by a Mr James Whitehurst in 1784. It succeeded in reducing the mortality rate from 1-in-14 to 1-in-16. In Paris:

"The mortality has diminished in the Hôtel Dieu in remarkable proportion Are we to count for nothing the destruction of all the high houses which surrounded the Hôtel Dieu? In our opinion, the pure and dry air which circulates now in every part, the sun which penetrates there, have as much contributed to its healthiness as the suppression of the amphitheatres of anatomy in its neighbourhood" (Du Châtelet, from Chadwick)

Here we see a realisation of the importance of pure air as well as adequate ventilation and heating arrangements.

Similar experiences were found in Dublin's Lying-in Hospital, where children's deaths were cut from 35% to 3% after ventilation improvements in 1784.

Dr Neil Arnott reported on the ill-health of school children. He noted that:

"the many deplorable cases of general ill-health and mortality were attributed at first to deficiency or bad quality of food, or to any cause but the true one, want of ventilation".

In one case he examined:

"the diet was found to be unusually good, but the ventilation very imperfect. Suitable changes were then made, and now, the same space where 700 children were by illness awakening extensive sympathy, 1100 now enjoy perfect health."

"In the dame schools, and the schools for the labouring classes, defective ventilation is the most frequent and mischievous (cause of delicate health)."

Chadwick's Report⁵ (from which the above extracts are taken) discusses ventilation and its effect on the health of workpeople, and of the ordinary family. He recommends that:

"for the prevention of diseases occasioned by defective ventilation it would be good economy to appoint a district medical officer independent of private practice."

The MOH we now have, but sadly not all are properly versed in air hygiene, and make quixotic decisions totally at variance with accepted good modern practice.

5. 19TH CENTURY DEVICES

The 19th century reformers Arnott and Galton made attempts to improve the ventilation of dwellings. Galton, in 1865, introduced his ventilating grate, in which the heat of the chimney induced a flow of air to the room via an adjacent duct. Arnott realised that ventilation required not only an inlet but also an outlet; and he devised a chimney ventilator in 1849, which was used in Buckingham Palace. It was no more than an aperture leading from near the ceiling into the chimney, and covered with a mica flap, hinged to open into the chimney. Arnott⁶ also noted the possibility of heat exchange between fresh and outgoing air:

"This double-current apparatus in an ordinary dwelling-house might consist of a number of small very thin metallic tubes, leading the hot air from near the ceiling towards the external atmosphere, and which would be included within a larger tube of which the office would be to carry pure air into the room; and currents would be produced in the two directions by suitable ventilating pumps moved like a kitchen jack by a weight wound up from time to time, or by any other power."

Tredgold (c 1820) devised a syphon ventilator, also making use of the heat of the fire.

Various window devices included a small windmill in one pane (Dr Hales ?), perforated zinc screens, louvred glass plates (Baillie, 1837) and a hit-and-miss disc (Cooper). The Tobin tube (a vertical shaft, open at the top, and communicating with the outside at the base) was popular in Victorian times (c 1878). One such had a water trough at the base of the shaft, ostensibly to trap dust from the incoming air. The Tobin tube was not a success and, shortly after its introduction, Edwards stated flatly that either the free area was too small or the incoming air immediately spilled over the top onto the floor. Worse, the provision of lids meant that all too often they were permanently closed. Yet the Tobin tube remained in use for 30 years or more, and was even recommended by an early 20th century architect.

In spite of these efforts and inventions, ventilation remained a generally neglected art. In 1850, Tomlinson wrote:

"Not only are our dwelling-houses badly ventilated, but those buildings on which the architect has lavished all his art and skill are, for the most part, entirely destitute of special means for ventilating and are so constructed as to render the application of such means extremely difficult or even impossible."

In France, Professor Ser could say, two or three decades later:

"One can see that the solution is no further forward, and that one has to be content with ordinary chimneys, cracks round doors and the opening of windows. (This was about hospitals - NSB). The problem has as yet no practical solution; our dwellings are unventilated."

In 1894, Professor Jacob, a pathologist, held the architect in contempt:

"In most cases architects are content to introduce an occasional air brick or a patented device called a "ventilator"

Real ventilation is so uncommon that the architect usually thinks this object has been attained if some of the windows can be opened. Some think that the presence of "ventilators", especially if they have long names and are secured by Her Majesty's letters patent, ensures the required end. We may as well supply our house with water by making the trap door in the roof to admit rain."

Other inventors followed - ridge ventilators and cowls (1880-1900); Boyle's patent air pump; the simple air brick to meet the demands of building by-laws; the Knapen system, with porous tubes at intervals in the base of external walls, and at high level in the internal ones (1925).

The advent of the electric fan inspired inter alia Thomas (1906)⁷ to suggest their use to ventilate wc's, and that they should also be operated by the seat.

6. HUMAN NEEDS

Knowledge of human physiology was almost non-existent until the late 18th century. The first quantitative studies of metabolism were those of Crawford in 1777, who measured the heat production of a guinea pig. The experiments were repeated by Lavoisier and Laplace in 1780; they also studied the respiratory exchanges, and showed them to be related to heat production. Lavoisier noticed that the consumption of oxygen was one-twelfth less at

26°C than at 12°C. Respiration in humans was studied by Seguin (1789), Dulong (1822), Regnault, Davy, Péclét and Dumas in the first half of the century. Dumas found that a person exhaled 330 l/h, the breath containing 4% CO₂, and that he metabolised 10 g/h of carbon and hydrogen, liberating 93W. Péclét⁸ went on to deduce that the water loss by respiration and evaporation was 61 g/h (- 44W), and hence, by difference, the sensible heat loss was 49W. Péclét, Roscoe and Pettenkofer went immediately from a knowledge of respiration to make estimates of the fresh air needed to remove water vapour and/or odours: none made any statement as to physiological need.

(Some rather strange ideas were current at the time. Thus Burns¹⁵ wrote in 1850:

"It is not only through the lungs that pure air exerts a beneficial influence upon the body; the skin, pierced by its numerous pores, also exercises a function of considerable importance In a healthy person, the inhalation of pure, and the exhalation of impure air, through the medium of the skin, goes on with considerable regularity. It has been ascertained that pure air, by being in contact with the skin, becomes chiefly carbonic acid gas.")

Arnott and Péclét both knew that the minimum needed to support life was a mere 330 l/h. Box assumed 620 l/h, but to determine the total fresh air requirement, he added the quantity needed to remove water vapour and combustion products (arriving at 7 m³/h rising to 14 m³/h in crowded rooms)⁹.

Pettenkofer established a School of Hygiene at Munich c 1865. He measured the exhalation of CO₂ at varying rates of activity.

Lavoisier attributed to CO₂ "the malaise often experienced in crowded halls, the malaise generally attributed to warmth alone". Leblanc believed (1842) the culprit was depletion of the oxygen content instead. But for over 100 years after Lavoisier, CO₂ was usually held to be the cause of discomfort, although Pettenkofer (1862) thought the unpleasantness was due to organic materials exhaled or shed from the skin. (However, he was prepared to accept CO₂ as an index of vitiation, and proposed a maximum CO₂ of 0.1%, based on odour).

It was Haldane and Hill who, in 1913, showed conclusively that no physiological distress occurred even at CO₂ levels of 3 or 4%. Any discomfort was due to excessive warmth (originally proposed by Hermans in 1883 - but cf. Lavoisier).

Subjective experiments were carried out by D.B. Reid¹⁰: he concluded that 10 ft³/min fresh air were needed. He says:

"This estimate is given with much diffidence and only as an approximation. It is the result, however, of an extreme variety of experiments made on hundreds of different constitutions supplied one-by-one with

given amounts of air Those who became the subject of experiment were for a time included in an air-tight box, and provided with a long glass pipe to draw the air they inspired from any part of the box in which they were placed. A small seat is represented on one side and on the other a series of trays on which materials could be placed either for the purpose of absorbing various ingredients from the air on special occasions, or of communicating different materials to it. On other occasions a more complicated piece of apparatus was employed, consisting of a large iron frame, glazed like a window, and connected with a pump, by which air of any quality could be supplied. The air entering is dried or charged with any peculiar materials."

He recounts an amusing story of the effect of ventilation upon diet:

"Some years ago, about 50 members of one of the Royal Society clubs in Edinburgh dined in an apartment I had constructed, where, though illuminated by gas, the products of its combustion were essentially excluded, as they were all removed by a ventilating tube connected with, but concealed in, the drop of the gothic pendant in which the lights were placed. Large quantities of mild atmosphere were constantly supplied, and passed in quick succession through the partment throughout the whole evening, the effect being varied from time to time by infusing odoriferous materials, so that the air should imitate successively that of a lavender field, of an orange grove, etc. Nothing very special was noticed during the time of dinner by the members, but Mr Barry of the British Hotel, who provided the dinner and who, from the members of the club being frequently in the habit of dining at his rooms, was familiar with their constitutions, showed the committee that three times the amount of wines had been taken than was usually consumed by the party in a room lighted by gas, but not ventilated."

Roscoe, working for the 1857 Commissioners, showed that 17 m³/h was not enough to clear odours in Wellington Barracks, and that 25 or 35 m³/h ought to be provided. De Chaumont's later work (1875) showed that odours were not generally perceived if the animal CO₂ was less than 0.04%, equivalent to a fresh air supply of 42 m³/h. These figures were much higher than Péclet proposed, on an odour basis, for prisons and schools, namely 6-10 m³/h per occupant.

Throughout the 19th century, there were, then, several hypotheses as to the cause of air vitiation. But perhaps because it was

easy to measure, CO₂ was almost universally accepted as a criterion of ventilation. "Animal" CO₂ was limited to 0.1% or less, corresponding to a minimum air supply of 28 m³/ (1000 ft³/h).

Yaglou's work of 1936 has, of course, been the basis of modern practice. It purports to yield odour-free atmospheres. Its distinct features are (a) reference to social class and (b) relation to volume per person. (Picard in 1897, and Box and others before, had noted the latter).

Although Billings, in 1884, had proposed extra fresh air where smoking was allowed, and Yaglou had worked on this, the best review is probably that of Brundrett¹¹ a year or two ago. He concluded that carbon monoxide concentration is as significant as the smell, and that an acceptable atmosphere might require as much as 40 m³ per cigarette.

It is clear that from an early stage, odour was regarded as the main contaminant, and that sufficient air to make it imperceptible would be enough to remove the other contaminants (water vapour, heat) and to support life. In passing, we should note the Synthetic Air Chart devised by Dr E.V. Hill in USA (1921), and this included bacterial concentration. Wells and Wells (1936) and others later have laid particular emphasis on air hygiene, and much work has been done on bacterial sprays, irradiation and ozone. The use of ozone as a deodorant was suggested in 1909 by Lubbert, and it was used by Rietschel in 1911, though he said it could never replace ventilation. It has been reported that ozone at a concentration of 0.015ppm is sufficient to mask body odours, and reduce fresh air requirements by 50%. The use of activated charcoal as an odour absorbent was advocated in the middle of the 19th century by Professor Stenhouse of Bart's, while in the early 1900's, the Pullman Car Company experimented with its use.

One must mention, too, the ion content of the air as a possible factor in well-being. The malaise associated with the föhn, the sirocco and similar winds is well-known; there is believed to be a high positive-ion content when they are blowing. Atmospheric electricity was studied by Coulomb in 1875, and air ionisation by Elster and Geitel in 1900. It seems to have been assumed in the early years of this century that air was ionized by radio-activity in dust (and would therefore be de-ionized by filtration and washing). Napier Shaw was unable "to state anything definite as to the effect the treatment of the air in the modern systems of ventilation exercises upon its electrical activities" (c 1905). Modern research is somewhat inconclusive as regards the benefits of high negative-ion content; but it appears to be fairly well established that air which has traversed a system of ducts has suffered a loss of negative ions.

These several factors raise a number of important questions:

- (i) How far is Yagou's work relevant to present-day standards of personal hygiene?
- (ii) Can fresh-air requirements be reduced by the use of odour absorbents, bactericidal means, ion generators,

dehumidifiers or coolers? - in other words, reduce the general or special contaminants by means other than dilution of fresh air.

A reduction in fresh-air supply would yield vast savings of energy: the proportion of energy needs simply to heat the ventilation air is rising as the thermal insulation of buildings is improved.

From what I have said, you will I hope draw the conclusion that ventilation is only one of several techniques which we can call on to ameliorate our atmospheric environment, or correct deficiencies and dangerous or unhealthy conditions. Dickson, in his paper to this Conference, notes that the internal environment must be so good that the occupants do not need to open the windows - if indeed the outdoor air is itself sufficiently pure and clean and the external noise tolerable. We should, I think, use one or more of these techniques, as may be appropriate, in conjunction with ventilation (whose function is, essentially, to provide the oxygen we need to live). We do not have to assume that ventilation is either the only or even the cheapest means of removing contaminants.

7. AIR DISTRIBUTION

I have said that Arnott and others had realised that inlets and outlets are both needed to ensure a flow of air (and also a satisfactory degree of air movement). The mere provision of these openings does not of itself however ensure that all parts of the space are efficiently ventilated; their size, disposition and siting are crucial in this. There has, I suspect, been too little research on this aspect of the problem. The argument concerning "downward" or "upward" ventilation has raged, without real conclusion, for over 100 years. It was Box, I think, who recognised that with natural ventilation, the summer problem was different from the winter one, and that the "natural" direction of air flow might be reversed from one season to the other. He saw, too, that the problem was different yet again when warm-air heating and/or plenum ventilation was employed. The problem is three-fold:

- (a) how to avoid short-circuiting and consequent stagnant areas.
- (b) how to avoid draughts, and in particular cold air in winter.
- (c) how to secure optimum removal of contaminants from the breathing zone.

Even today, with air conditioning, i.e. winter heating and summer cooling, I doubt whether any one position for the entering air openings is appropriate all the year round. There is, in short, no single answer, applicable to all seasons and to all modes of heating and ventilation, to the question of siting inlets and outlets.

One might perhaps say that the problem of introducing air (either naturally or mechanically) can usually be solved in a technical or engineering sense. It may however often be defeated by architectural or aesthetic considerations - perimeter openings at floor level may not be possible for structural reasons, or they may be unacceptable since they may become depositories for dust or other debris, or the openings may unduly restrict the placing of furniture.

8. MOTIVE POWER

Ventilation requires a motive force - ça va sans dire. Three sources are available:-

- heat, either the stack effect (indoor-outdoor temperature difference, or induced by fires, gas jets, hot water pipes, etc.)
- wind pressure
- mechanical means (bellow, fans, pumps, etc.)

The natural forces are variable and ventilation uncertain. One can never ensure ventilation in still-air conditions in the spring and autumn of temperate climates, for neither wind nor stack effect then exists. This is the central problem. When the natural forces are in play, something can be done to restrict the maximum air flow. This is usually left to the occupant (who is, by and large, an inefficient control) by window opening or hit-and-miss ventilators, though attempts have been made to design automatic variable-resistance devices with the object of fixing a maximum flow.

It was this uncertainty which led inventors to turn to means of assisting the flow of air. Fires at the base of special chimneys, the heat of gas jets, the fires of a ship's galley, steam and water jets have all been used from time to time. The circulating pipes of hot water heating systems have been used in dwellings, prisons and hospitals to induce air movement in upcast shafts. All these were intended to extract air; the fresh-air was to find its own way into the building. It is not clear from their summary whether Sherman and Grimsrud have something of this sort in mind.

Fire extract had some merit. It was used in a soap works to destroy the odours (the boiling vats were enclosed by a hood, and the vapours led to the ashpit of the fire); and in a hospital at St Petersburg to sterilize the air to be discharged.

An ingenious device was described by a Dr Anderson⁹ in 1800. It was intended to ventilate, both winter and summer, by the heat of the sun. Its essential component was a chimney connected to an aperture in the ceiling of the room to be ventilated. This chimney had its south side made of glass, and the sun shining upon it warmed the whole length of the chimney, causing the air within to ascend. Anderson further supposed that if the room were constructed with double walls, the cavity being also connected to

the chimney, then in summer the inner rooms could be kept cool, for the sun's heat would be carried away by the air current in the cavity. And if the whole outer south facade of the room were of glass, the heated air in the cavity would be used to warm the room in winter. This, we see, was effectively a Trombe wall, such as is being currently investigated as a means of using solar energy.

9. ARCHITECT AND ENGINEER

Much of the success or failure of many if not all of these schemes (and indeed of later mechanical systems) rests upon the relationship between architect, builder and inventor/engineer. When the architect and ventilation designer were at odds (as were Barry and Reid over the House of Commons in 1834), the scheme failed; when they were able to work together, as Reid was able to do with Lonsdale, the architect of St George's Hall, Liverpool, in 1851, it was an outstanding success. And with extract systems, poorly fitting doors and windows give rise to draughts, however efficient the extraction may be.

We have here a clue to future success. The building must be designed with ventilation in mind; the services cannot just be tacked on as an afterthought to a pre-existing concept. To put it the other way round, we have to decide at the outset what sort of ventilation is necessary - mechanical or natural, plenum or extract - and indicate to the architect and builder the quality of construction needed. A recent Norwegian paper by Trond Ramstad has highlighted the necessity for attention to building details.

10. CALCULATION OF AIR FLOW

I believe that the first attempts to compute infiltration rates were made by Tredgold¹² in about 1820. Certainly he calculated the air flow through the cracks round doors and windows (arriving at 11 ft³/min for each unit), and went on to discuss the ventilation of greenhouses to prevent overheating in summer. His work was founded on the stack effect. It was largely ignored, and recourse was had to empirical statements (of Hoffman and Raber, 1913, who thought that natural infiltration was impossible to determine, and recommended 2 ac/h be assumed).

About the turn of the century, the study of meteorology began to be combined with that of ventilation. J.W. Thomas seems to have made some experimental determination of wind pressure on buildings in about 1900 - 1901, and he appreciated its effect on natural ventilation. Napier Shaw in 1908 developed the electrical analogy for computing air flow in networks, building on Murgues' concept of an equivalent orifice. This work, too, was neglected for half a century, until revived by Hinsley (who in about 1950 built an analogue computer for solving mine networks), by Harrison in 1961 for buildings, and by others since, with further computer models.

But when all is said and done, the prediction of infiltration or air flow depends on the quality of the input data - in particular, wind pressures, goodness of fit of components and structure, and permeability of the fabric. All these are variable. We can give calculations based on historical data; we cannot predict what will happen, at the drawing-board stage.

If this is a negative attitude, it brings me back to saying that the variability must be eliminated as far as possible in the design and that we should use the more nearly calculable mechanical ventilation.

11. HALTING PROGRESS

It is I think worth noticing how technical progress can be halted or reversed by misguided or erroneous opinions, firmly held.

Mechanical ventilation of schools was relatively common in Britain in the early years of the century. True, its design was a trifle odd to our eyes, e.g. ductwork:

"When the ducts are of such size that it is possible to walk through them, it is reasonable to expect school masters and managers to personally supervise them and their cleanliness, whereas with small ducts they are left to the tender mercies of the caretaker, with too often a fatal result as to this necessary condition."¹³

But the LCC came to the conclusion that mechanical ventilation and heating in schools was a failure, "because teachers manipulated the apparatus in such a way that the heating could not be relied on and was made useless". This seems to have been the death knell of mechanical ventilation in British schools. Later, in 1928, H.M. Vernon¹⁴ was totally convinced of the virtues of natural cross-ventilation, and preached that mechanical ventilation should be avoided. He went so far as to advocate open-air schools, since because they were good for tubercular patients, they would be equally good for healthy children. His colleague, Bedford, though, observed that the poor little devils were never warm. Not until after World War II was mechanical heating and ventilation again seen in British schools.

An older story of official obstruction relates to the efforts of Desaguliers and Sutton to improve the ventilation of warships⁹. At the beginning of the 18th century, troops embarked for an expedition against the Spaniards were taken ill and had to be relanded. In consequence, the Lords of the Admiralty asked Dr Desaguliers to demonstrate his newly developed fan. Although the trial seems to have been successful, the Surveyor to the Navy, Sir Jacob Ackworth, would have none of it. This was about 1736. Shortly afterwards, Samuel Sutton devised a method of ventilation, consisting essentially of leading pipes from the space to be ventilated to the ash-pit of the galley fire, the draught of the

fire causing air to be drawn from the spaces through the pipes. Sir Jacob said that no trial should be made if he could help it. Eventually, in 1741, the Admiralty was induced to order a trial. Sir Jacob welcomed the distinguished company (Lords of the Admiralty, Navy Commissioners and some Fellows of the Royal Society) by saying, "I am sorry that you are come to see the trial of such a foolish experiment, that I tried myself yesterday, and it would not shake a candle". In fact the trial was successful. After much delay, the Admiralty stated that:

"the apparatus does not come up to expectation, and the use thereof is dangerous and liable to accidents by fire".

But they gave Sutton an ex gratia payment of £100.

12. RESEARCH

I confess to wondering whether some current research is not being directed at the wrong questions. Studies of natural infiltration through door and window cracks, due to wind or stack effects, are, almost by definition, statistical studies: the results cannot, I suspect, be used in design because of the innate variability of building components and of wind force. We can eliminate much of this variability if we build tight structures, i.e. no opening windows, and provide ventilation by mechanical (controlled) means - or at least by designed openings such as "constant-flow" ventilators. These approaches are seen, in their separate and opposing ways, by Dickson and by Johnson and Pitts.

Computer models are valuable only for studying the effect of varying the parameters; they rely on historical data; they cannot of themselves provide a solution which can go straight to the drawing-board. And research aimed at a detailed understanding (even perhaps the minutiae) may well be irrelevant to the central problem, which is to ensure a satisfactory and healthy environment at minimum cost.

Also, why the emphasis on dwellings? True, this Conference relates specifically to them; but are there no problems in factories, schools, offices? - or have we already made up our minds that in these cases, all the decontamination methods, and mechanical ventilation, are appropriate and indeed necessary and worthwhile? Nobody has suggested that although central heating and fluorescent lighting is appropriate for non-domestic conditions, these are unsuitable or uneconomic in dwellings. In the same way, I do not think we can maintain that mechanical ventilation is necessarily out of place in the home.

13. CONCLUSION

The burden of my talk is to urge a broad approach to the problem of ventilation, not only in dwellings, but in all buildings. An occasional backward glance may bring to light a useful idea, impracticable at the time of its conception, but which has been made possible by technological advance. We do well to understand, and perhaps to query, some of the axioms of the present day, for it is likely that circumstances have changed since they were first formulated. Perhaps, above all, we should avoid the blinkered approach, the concentration on one or other aspect, without questioning its relevance to the central problem. One can often overlook the wood while one is examining the trees. Quite often the problem put to the research worker is not the crucial one, but only the Director's perception of it. The real solution may perhaps be found by going round an obstacle rather than tackling it head on. Let us not forget why we need ventilation and why we need to control it. There are many ways of killing cats. And always remember that any successful solution must involve the heating designer, perhaps the lighting and acoustic engineers, and certainly the architect, the builder and the occupant.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 2

USE OF NATURAL VENTILATION

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SYNOPSIS

Ventilation needs are outlined to show that odour dilution and moisture control are the major winter factors. Detailed studies on 24 well insulated houses show that the window opening habits are clearly linked to the outdoor temperature. More windows are opened in milder weather but even in the coldest weather some windows are open. Moisture measurements inside the house confirm this trend.

Energy input analysis shows that space heating only provides a quarter of the total heat, the remainder comes from casual sources. Analysis of energy loss suggests that a third of the losses are attributed to ventilation, the remainder through the building fabric. This ventilation loss is equally divided between infiltration and window opening. Average house seasonal ventilation rate is half an air change per hour.

1. INTRODUCTION

The five factors which influence space heating energy are:

1. Fabric insulation
2. Infiltration losses
3. Internal temperature
4. Free heat from sunshine and occupancy
5. User ventilation

The energy loss through fabric insulation is well recognised and insulation standards in Britain have been rapidly improving. Infiltration is an active research area. Ventilation rates vary according to wind and stack effects but average values for a house with closed windows and doors vary between one-third of an air change each hour to one and three-quarters. Internal temperature is assumed to be under the control of the occupant. In practice it is more usual for the living room to be controlled and for the temperature of the rest of the house to float somewhere between the living room and the outdoor temperature. The free heat comprises sunshine and internal heat. The influence of sunshine is mainly a function of the prevailing cloud cover and of the window sizes and directions. The internal heat is a function of the number and activity of the occupants themselves and of the number of electrical appliances in use, such as lighting, television and refrigeration.

The major unknown is the energy loss due to the ventilation behaviour of the occupants. This paper describes a study to estimate this energy cost.

The minimum ventilation requirement in a room is determined by the permissible carbon dioxide concentration. Carbon dioxide is exhaled during respiration. This limit is not normally reached because infiltration is more than adequate. The usual criteria are comfort and absence of smell. More generous ventilation is needed for this. A less clearly defined criterion is moisture control. The air in Britain is near saturation for the duration of winter. If the internal environment is to be controlled then more air will be needed in milder weather to dilute the moisture. The relative importance of these factors is illustrated in figure 1.

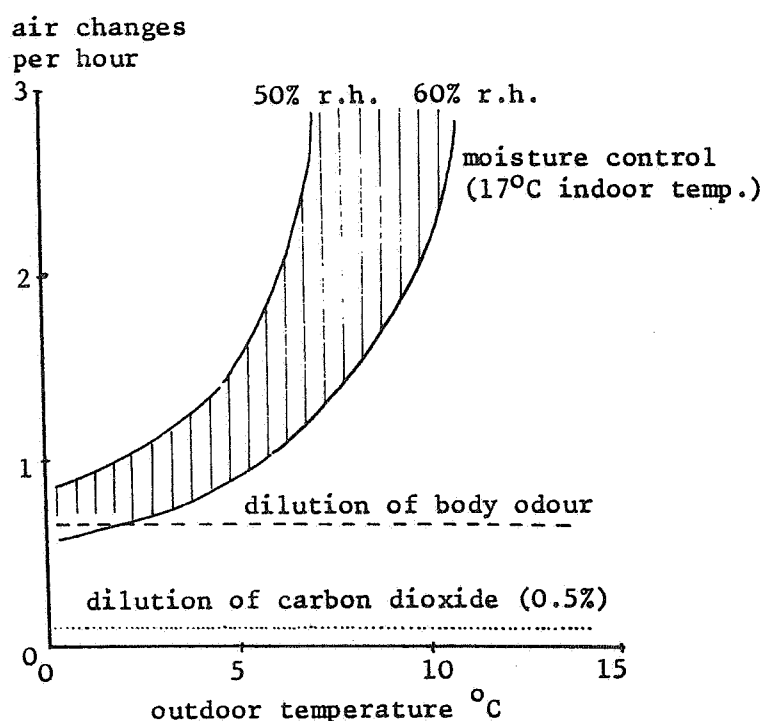


Figure 1 Ventilation needs for a living room
(2 sedentary adults, 30m³ volume room)

Behavioural studies have identified the link between window opening and outdoor temperature or moisture. Monitoring studies have resolved temperature and energy flows within low energy dwellings. The Electricity Council have made particularly comprehensive measurements on a group of twenty-four houses in Scotland and these form the basis of this analysis. The user window opening behaviour was also recorded.

3. FIELD TRIALS

3.1 The Houses

The houses are situated in an open and fairly exposed area 12 miles west of Aberdeen. Around 500 were built here by Gordon District Council to meet the urgent need for accommodation in north-east Scotland. The test houses are all prefabricated timber construction with high levels of thermal insulation and weatherstripping. Double glazing is standard throughout the houses. Each house has three bedrooms. Three house types were selected: detached, end-terrace and mid-terrace. An unusual feature is the party wall between the terraced houses. This is a double wall which is constructed and insulated as if it was an external wall to each house. It is therefore doubly insulated and house to house heat transfer will be negligible.

Electricity is the sole form of energy supplied to the houses and was charged at the normal domestic tariff. Space heating is provided by direct acting, proportional control, wall mounted heaters, supplemented in each of twenty of the houses by a radiant log-effect electric fire. This system is controlled by a two zone time clock programmer.

A central hot water storage system is provided, with the cylinder (114 litres) in a first floor cupboard. The cylinder is heated by a 3 kW immersion heater and is factory insulated with (22mm) polyurethane foam.

Each kitchen is provided with a manually operated extractor fan. Each window has a manually controlled ventilation slot.

Dimensions and thermal properties of the houses are listed in Table 1. Twenty-four houses were monitored. Results were incomplete in one house because of equipment problems.

The results given apply to the period 25th October, 1977 to 23rd May, 1978.

3.2 Monitoring

Each house had three electricity recorders. One continuously measured the space heating energy, one the water heating and one the remaining energy used by cooking, lights and appliances.

Thermographs were used to record living room, hall and bedroom temperatures. Some moisture measurements were taken in bedrooms by recording thermohygrographs. Market researchers interviewed the occupants to establish family size and the reported use of the heating system. Ventilation habits were also noted. A spot check of rooms with an open window was made each weekday for each house, between either 0009-1230 or 1400-1630 h.

Systematic recording of the weather included wind speed and direction, outdoor air temperature and humidity, rainfall, and solar irradiance on the horizontal surface.

3.3 Measurements

The average temperatures and energy consumption figures for each house type are included in Table 1. There was little difference in space heating energy between the different house types but there was a systematic difference in house temperatures. Mid-terraced houses were generally warmer than end-terraced houses, which in turn were warmer than the detached houses.

Table 1 House details: Summary 25th October 1977 - 23rd May 1978

House type		Det- ached	End- terrace	Mid- terrace
Number in sample		9	11	3
Floor area m ²		109	99	99
Fabric heat loss (18°C Δt)	kW	1.88	1.61	1.38
Ventilation heat loss (18°C Δt)	kW	1.68	1.62	1.62
Specific heat loss Watts/°C		102.4	86.0	78.8
Heating equipment (direct electric)	kW	8.4	6.6	6.6
Average winter temperature: outside	°C	4.4	4.4	4.4
Average bedroom temperature	°C	14.2	15.1	16.2
Average hall temperature	°C	13.9	15.3	15.7
Average living room temperature	°C	17.1	18.0	18.4
Average bedroom relative humidity	%	61	59.3	58.5
Space heating	kWh	2021	2194	1995
Electricity for hot water	kWh	2459	2175	2162
Electricity for cooking, etc.	kWh	2208	2253	2258
Useful electrical gains	kWh	5377	5158	4782
Useful energy from people	kWh	917	930	930
Solar gains	kWh	1355	1365	1365
Total useful energy in house	kWh	7650	7454	7077
Mean internal/external Δt°C		9.6	10.5	11.0
Average open windows over winter per house		1.66	1.14	1.03
Average total ventilation rate	ac/h	0.49	0.51	0.50
Average ventilation due to window opening	ac/h	0.24	0.26	0.25

Ventilation calibration of one empty detached house showed the ventilation rate to be mainly a function of the outdoor wind speed. This relationship is shown in figure 2. The average wind speed was 2.3 m/s over this heating season and therefore the average infiltration was taken to be 0.25 ach. Pressurisation tests on representative houses showed the mid- and end-terrace houses to have similar characteristics to the detached one, and therefore the average infiltration was taken to be 0.25 ach for each house.

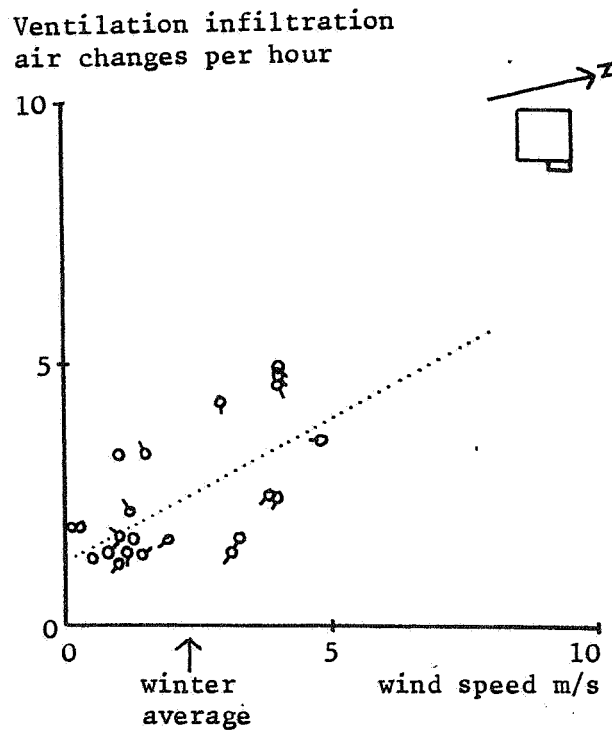


Figure 2 Ventilation characteristics of an empty detached house
(Dickson, ECRC: personal communication)

The window opening behaviour was related to the weather according to the following relationship (correlation coefficient $r = 0.75$)

$$\text{Number of rooms with an open window/house} = 1.2 + 0.1 \times T - 0.14 V \text{ m/s} + 0.14 R$$

where T = mean outdoor daily temperature $^{\circ}\text{C}$

V = wind speed m/s

R = solar energy on the horizontal kWh/day

This is illustrated in figure 3. The earlier results of Dick and Thomas 1951, and Brundrett 1977 are included for comparison.

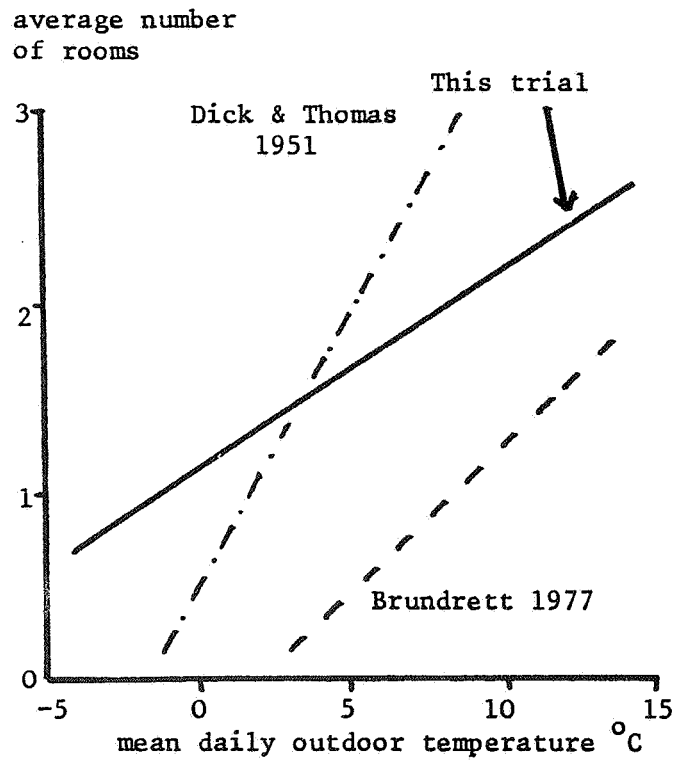


Figure 3 Relationship between open windows and outdoor air temperature

The windows which were found to be open are recorded in figure 4 as a daily average over the winter. This shows that the upstairs windows are the ones usually opened.

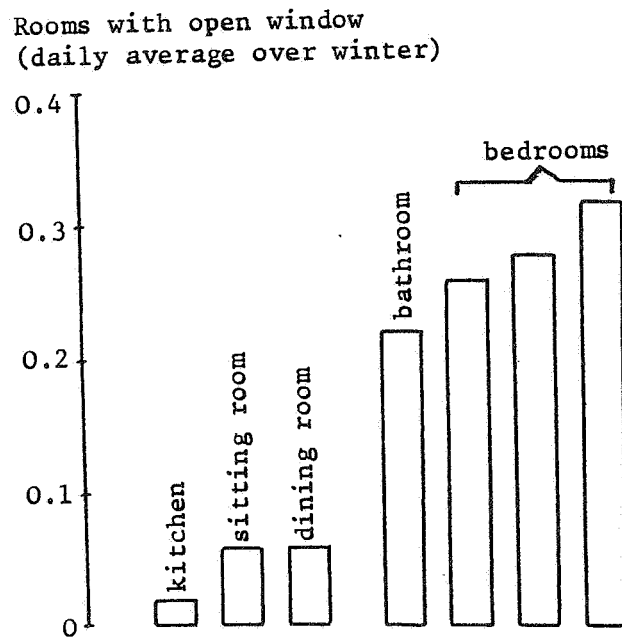


Figure 4 Distribution of windows opened

The thermohygrograph records of moisture in the main bedrooms showed a seasonal average of 60% r.h. This seasonal average varied between 54 to 67% r.h. for different houses. A particular study for part of the heating season was made for the bedrooms of the terraced houses. This is shown in figure 5. The indoor air moisture content was only influenced to a small extent by large changes in the outdoor air moisture content. This confirms the increase in ventilation rate in warmer weather as suggested by the pattern of window opening.

Moisture content of air in bedrooms
grams water/kg dry air

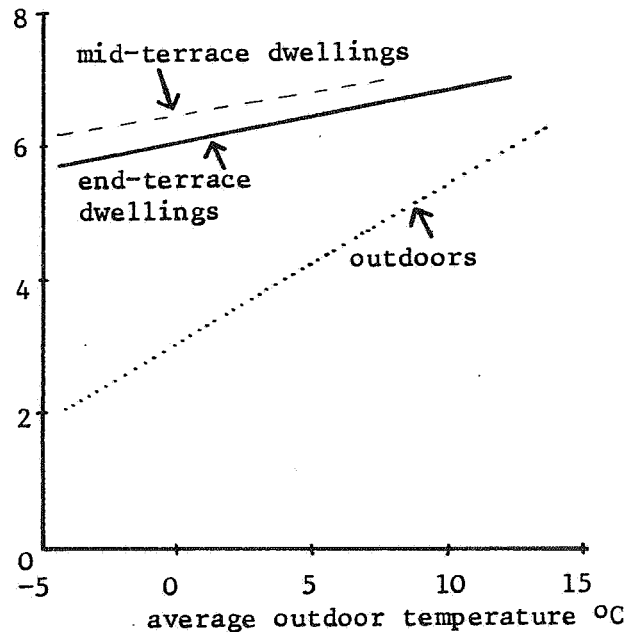


Figure 5 Moisture in bedrooms relative to outdoor air

4. ENERGY BALANCE

The four energy sources heating the homes are:

- (a) Space heating. This is mainly by panel heaters but with an occasional use of the living room radiant fire.
- (b) Other electrical gains. An approximation of the sensible heat gain from electricity is taken as 30% of the domestic hot water energy (15% through the lagging and 15% through the piping) and 87% of the rest of the electrical energy.
- (c) The occupants themselves. This is a function of the number in the family and the time they spend at home. Four people at 90W per person will add up to 2000 kWh to the house over winter, depending on their time at home.
- (d) Sunshine. The net solar gain was computed monthly for each face of three houses. The horizontal solar radiation was converted into incident radiation to vertical planes from

empirical relations derived from measurements at Kew. Corrections were made for sky radiation loss and orientation sensitivity of the glass transmission coefficients. General solar gains, based on these three houses, were applied to all the houses.

The energy input for each house type is summarised in figure 6. This shows that the space heating energy only contributes a quarter of the total released in the home. The relationship between energy and house temperature was explored for the nine detached houses and this is illustrated in figure 7.

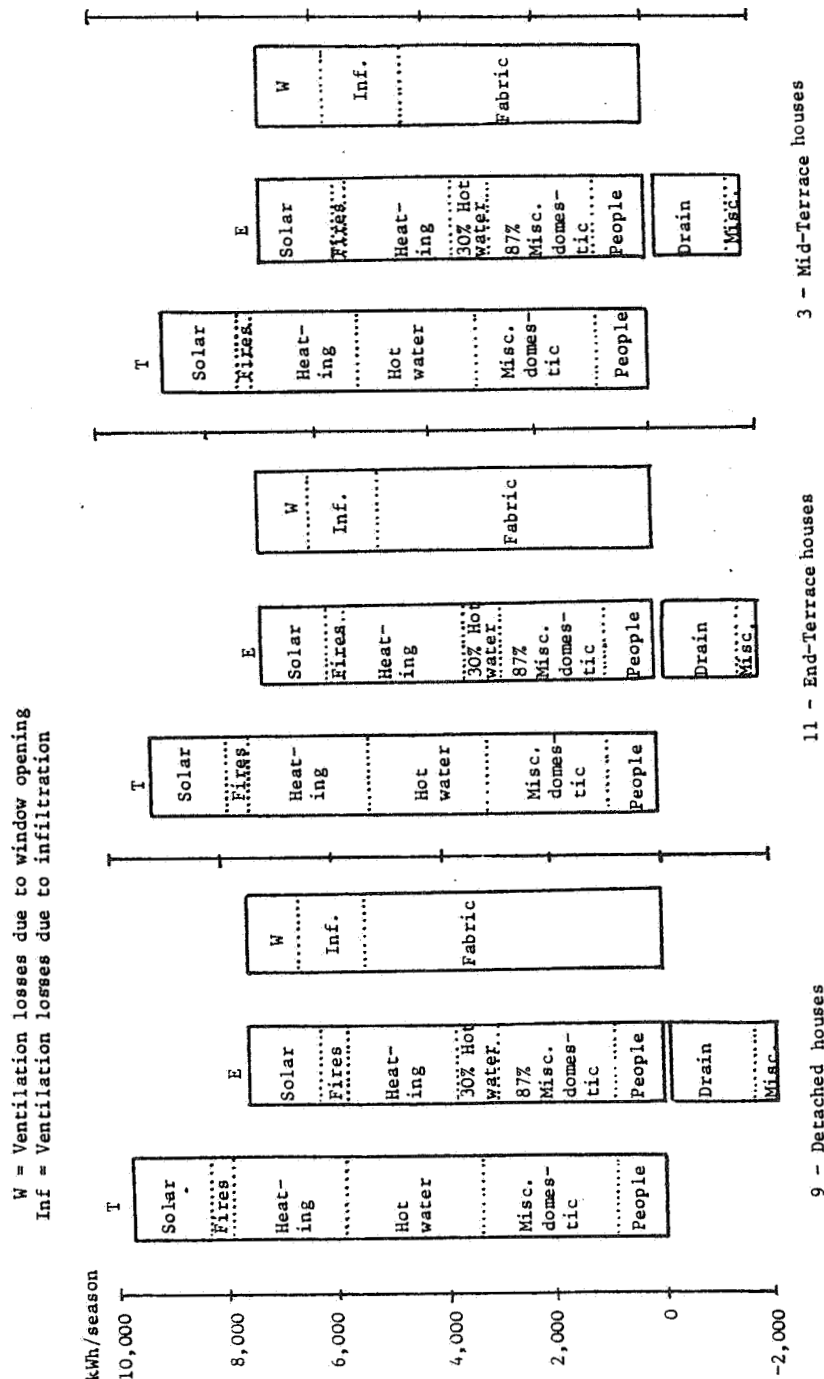


Figure 6 Energy input and losses in the houses over the heating season

Total useful energy
in the home during
the heating season

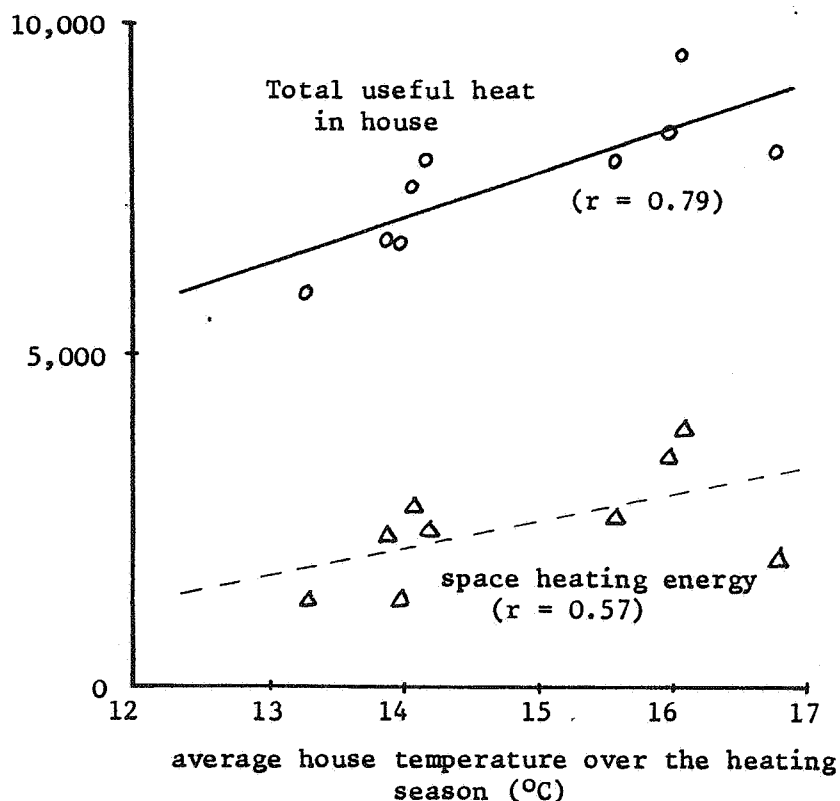


Figure 7 Energy into the homes as a function of house temperature (nine detached houses)

The four energy losses are:

- Fabric heat loss. This can be calculated from a knowledge of the internal temperature, the external temperature and the thermal transmittance of the building components.
- Ventilation heat loss. This cannot readily be calculated directly. It can be estimated as the difference between total useful energy released in the house, less that energy lost through fabric conduction. This estimate will also include all error terms.
- Hot water to drain. Since the losses from the domestic hot water systems are estimated at 30% of the total drawn, then the remaining 70% is assumed to go down the drain.
- Evaporative losses. Latent heat losses from moisture in the building fabric over the heating season are ignored. Minor losses from cooking will be lost as latent heat.

These loss routes, matched against the total energy supplied, are also shown in figure 6.

The detailed energy analysis was continued on the nine detached houses. This illustrates the loss mechanisms and shows ventilation to be a surprisingly constant energy loss, despite differences in average house temperature. The ventilation losses were calculated on the assumption that the ventilation route was into the house downstairs and from the house through the upstairs rooms. The ventilation loss was therefore taken at bedroom temperature (figure 8).

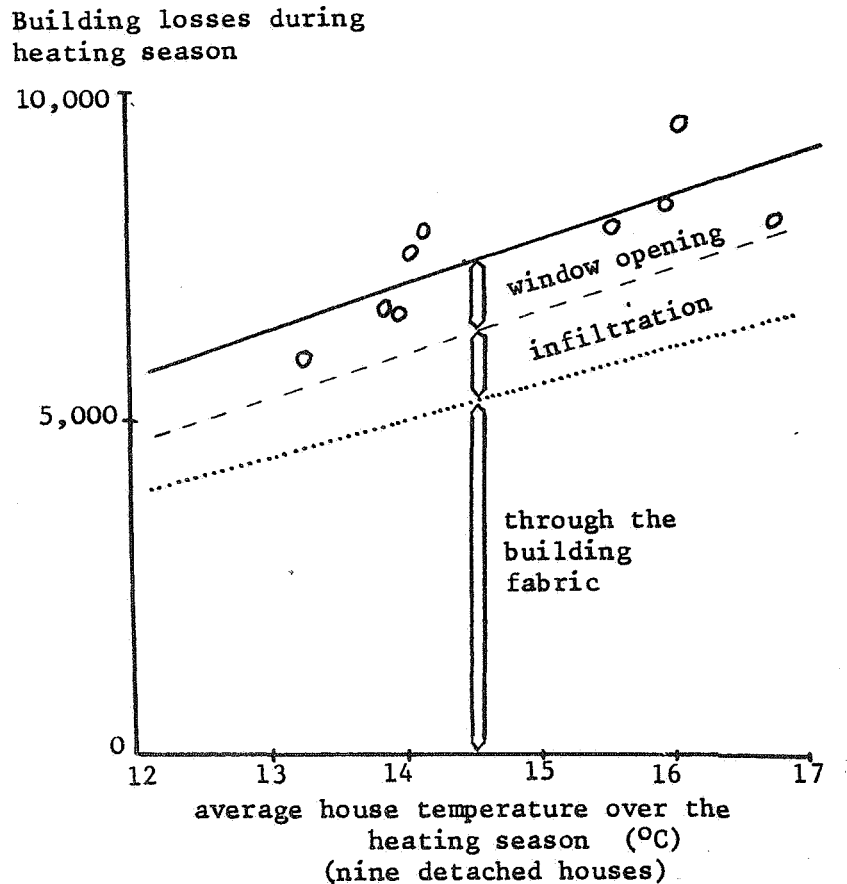


Figure 8 Allocation of energy losses in nine detached houses
(An additional 2,000 kWh is lost through drains and latent heat during cooking)

5. CONCLUSIONS

1. Studies of window opening behaviour in houses show that windows are seldom all completely closed. In average conditions houses have one or two rooms with an open window and the number reduces in colder weather and increases in milder weather.
2. Energy analyses for these houses show that only one quarter of the thermal energy in the home comes from the space heating equipment. The remaining 'free' heat comes from appliances, losses from hot water, sunshine and the occupants themselves.

3. Analysis of losses from the houses suggests that about a third is due to ventilation. The remainder is through the building fabric. Half of this ventilation loss is casual infiltration and half is attributed to window opening habits.

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ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 3

VARIATIONS IN HOUSEHOLDERS' WINDOW OPENING PATTERNS.

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SYNOPSIS.

The window opening survey was concerned with identifying the objective correlates of window opening. Preliminary analysis indicated that the number of openable windows in a dwelling was an important explanatory variable. Results showed that there were smaller differences in window opening levels between house types when the percentage of open windows was taken as the relative measure than when the absolute number of open windows was considered. It was found that the variation between households in terms of their total daily window opening was greater than that within households. It was suggested that householders adopted consistent window opening patterns, an hypothesis which received support from the finding of a strong positive relationship between window opening in different room types.

A potential model of window opening was proposed. On the basis of findings from the analysis of observed relationships between window opening in certain room types and selected weather parameters, it was hypothesised that although window opening was primarily a function of external air temperature, relative humidity and windspeed were influential at high values. It was further hypothesised that all households follow a similar curve of window opening against increases in temperature but that the threshold temperature marking the foot of the curve varies from household to another.

Analysis showed that the frequency distribution of the average percentage of open window observations in each household was skewed to the right. Thus, in order to test the model, households were divided into three groups (high, medium and low) on the basis of their window opening. Results showed that the relationship between temperature and the three groups was as predicted.

THE STUDY.

The theoretical significance of ventilation increases as insulation standards improve. It is consequently important that householders' window opening patterns should be understood. The present window opening survey was therefore concerned with identifying the objective correlates of window opening as well as the subjective motivations for the opening and closing of windows. Householders were also asked about the effects of specified weather conditions in three room types in winter and summer. However, only the results of the observed data will be discussed in this paper.

1. METHODOLOGY.

1.1 The Sample.

The sample consisted of 113 households on two local authority estates in Middlesex (at Cowley N = 78, at Mezen N = 35). The houses were of similar physical construction but were of eight design types with different elevations and varying window arrangements. They all had partial gas central heating to Parker-Morris standards.

1.2 Data Sources.

The study made use of three main data sources; a series of systematic window observations, mean hourly meteorological data and a postal questionnaire.

1.2.1 Window Observations.

All 113 households were surveyed on one day. This was repeated 100 times during the period from October 1979 to April 1980. The days and times were chosen in such a way that each house was surveyed twice at each hour between 9a.m. and 6p.m.. It is important to note however that the hours recorded as the hours of visit were not necessarily the exact times at which the observations were made. For example, an observation recorded as having been made at 3p.m. would actually have been made sometime between 3p.m. and 4p.m.. This was because on average it took twenty minutes to make a round of observations. A window was recorded as open if the bottom frame of the window was judged to be more than 1" away from the frame. A window was otherwise recorded as closed. When the 100 observations had been completed, the data were summarised to facilitate the analysis of window opening patterns in individual rooms. A count was made of (a) the total number of open windows and of (b) the number of open (i) sitting-room, (ii) kitchen and (iii) main bedroom, windows in each house on each day. The individual figures from each house were then summed to give the number of windows (total, sitting-room, kitchen or main bedroom) open on the estate for each of the 100 days.

1.2.2 Mean hourly meteorological data.

Meteorological readings for five weather parameters - external air temperature ($^{\circ}\text{C}$), relative humidity (%),

windspeed (knots), rainfall (mm), and sunshine duration 1/10 hour) - were obtained from the meteorological station at Heathrow Airport. The mean values recorded were for the sixty minutes immediately following the stated hour of observation.

2. ANALYSIS AND RESULTS OF THE OBSERVED DATA.

2.1 Basic Statistics.

2.1.1 Household consistency.

The average daily number of total open windows,

$$\frac{\sum_{\text{day } 1}^{100} (\text{TOTAL})}{100}$$

and the standard deviation about that mean, was calculated for each house. The number of total open windows for the observation period for the two estates combined has a mean of 1.27 (N = 113 households) and a standard deviation of 1.5 window observations. This figure is larger than the standard deviation of total daily window opening for most of the individual households (figure 1) indicating that the variability between households (in terms of their total daily window opening) was greater than that within households. This essentially means that while some householders did fluctuate in their window opening from day to day, they were still fairly consistent over the 100 days - some householders consistently opened only a few windows whilst others consistently opened several windows.

2.1.2 The number of windows observed to be open in relation to the number of openable windows.

Figure 2 is a scattergram of the relationship between the number of openable windows in a dwelling and the total number of windows observed to be open in each house over the 100 days. The results of a Wilson Chi Square Test show a positive relationship between the number of openable windows in a house and the total number recorded as open ($\chi^2 = 25.19$, $df = 5$, $p < .01$). This finding is taken to indicate that the number of openable windows in a house is a potentially important explanatory variable. It was therefore decided that results would henceforth be expressed as percentages. However, it must be noted that when the proportion of open windows in each house is used in a second chi square test, the results are still significant, though this time only at the 5% level.

2.1.3 Window opening in specified room types.

Table 1 gives the mean percentage of total open window observations on the two estates (N = 100 days in each case). The mean percentage of open window observations in the three room types is also given. The table shows that on both estates the windows in the main bedroom were open more frequently than those in either the sitting-room or the kitchen. The sitting-room windows were seldom observed to be open.

2.1.4 Inter-relationships between window opening in different room types.

Householders have been shown to be fairly consistent in terms of their daily window opening (figure 1). Scattergrams showing the relationships between window opening in different room types were drawn - each diagram had 100 points marking the percentage of open windows in one particular room type against the percentage observed to be open in another room type on that day (N = 100 days). Analysis of these diagrams indicates that householders' consistency extends across room types; that is, there is a strong positive relationship between window opening in different rooms. High and significant correlation coefficients were obtained for each scattergram confirming the hypothesis that householders adopt general window opening levels. However, it is important to note that these correlations do not necessarily indicate that windows in different room types are open simultaneously, though this may well be the case.

2.2 Relationship Between Window Opening and Weather Parameters.

Table 2 shows the correlation coefficients obtained between open window observations and weather parameter values at the hour of observation. The correlations between open window observations and external air temperature are highly significant and take high values for all room types on both estates (1 tailed test, df = 98, $p < .01$). The results for other weather parameters differ according to estate and room type. All the correlations for relative humidity are negative. Under a null hypothesis that relative humidity had no effect, this represents a probability of 1 in 256. Also, four of the correlations are highly significant. Similarly, the correlations for windspeed are all negative, though only one is highly significant. Seven of the sunshine duration correlations are positive, four are highly significant. However, there is no apparent relationship between rainfall and open

window observations on either state.

Third order polynomial fits of the open window observations for each room type against each of the five weather parameters were drawn. Examination of the curves for temperature (figures 3 and 4) confirm that there is a strong positive relationship between window opening and external air temperature. Relative humidity rarely fell below 60%. Thus, the left-hand extremities of figures 5 and 6 cannot be relied upon. However, the suggestion of an upturn at the right-hand extremes are supported by examination of the original scattergrams (not given). Most of the recorded windspeed values were between 3 and 15 knots. Inspection of the curves in figure 7 within that range suggest a slight negative gradient for Cowley data only, and no relationship for the Mezen data. The original scattergrams (not given) show a wide range of open window percentages with a tendency for the points to converge at high windspeeds. The curves for sunshine duration at Cowley (not given) suggest a slight increase in window opening with sunshine duration, though no such increase is apparent at Mezen. Inspection of the polynomial curves for rainfall (not given) support the hypothesis that there is no apparent relationship between rainfall and window opening.

2.3 A Tentative Model of Window Opening.

Based on the analysis discussion of the previous section (2.2), a model of window opening is proposed. It is hypothesised that window opening is primarily a function of external air temperature and that relative humidity and windspeed are influential only at high values. Also, that for any individual household the temperatures occurring at the times when windows are opened are approximately normally distributed with the same standard deviations for all households but with means varying from one household to another (figure 9). The model predicts that the percentage of open windows at given temperatures follows the cumulative normal distribution. It is a feature of the model that at extremely low temperatures all households will have no windows open, and similarly that at extremely high temperatures all households will have all windows open.

Window observations were restricted to a six month winter period. On the one hand the winter itself was mild and low air temperatures were rarely reached, with the result that some households had a few windows open on almost every observation day. On the other hand the omission of summer observations meant that few high temperatures were included in the data and window opening saturation was never reached. It is suggested that the observation period covers only the central portions of the curves as indicated in figure 10.

Figure 11 shows a frequency distribution of the average percentage of open window observations (over the 100 days) in each household ($N = 113$). The distribution is skewed and covers almost the entire percentage range. In order to test the proposed model households were divided into three groups:

- a) a "low" group who tend to open a small proportion of their windows
 - b) a "medium" group of householders who on average open a moderate proportion of their windows, and
 - c) a "high" group who tend to open most of their windows.
- These three groups together comprise "all" households ($N = 78$ and $N = 35$ for Cowley and Mezen respectively).

As previously mentioned the 113 households fall into eight house types. The total number of windows opened (over the 100 days) by householders within a particular house type, were listed in descending order and divided accordingly into three groups, i.e the top, middle and bottom third. The high window openers from each house type were then collectively referred to as the "high" group, and similarly for the "medium" and "low" groups. Thus, each house type was represented as equally as possible in the three groups.

The curves generated by third order polynomial fits (figures 12 - 35) provide a test of the model.

Figures 12 - 19 provide tentative confirmation of the model outlined in figures 9 and 10. At the lower temperatures the curves for the low groups have the smallest gradients, those for the medium groups have a small positive gradient, whilst those for the high groups have a larger positive gradient. This is as predicted from the theoretical curves in figure 9. Indeed figure 12 suggests that at the lowest observed temperatures the high group will already have a proportion of windows open, followed respectively by the medium and low groups. This feature is confirmed by figures 15 and 19. All eight diagrams provide evidence of a strong positive association between window opening and external air temperature. There is one slight exception however - the downturn of curves in figure 17. These are due to only two data points drawn for days 1 and 99. The negative portions of the curve are therefore unreliable.

Figures 20 - 27 show the relationship between window opening and relative humidity in different room types for the high, medium and low groups. The polynomial curves for the three groups are approximately parallel. The three groups are composed of different individual householders and it is therefore unlikely that important factors other than weather parameters or factors correlating with weather parameters, produce similar window opening patterns. Thus the form of the three curves is taken as evidence for the reliability of a particular trend in window opening patterns.

S - shaped relationships between window opening and relative humidity are strongly present at Cowley, and to a

lesser degree in the three Mezen groups. The agreement in terms of curve shape between the room types for each estate could be explained by other factors - such as householders adopting general window opening patterns, with the decision to open one window increasing the likelihood of further window opening. However, it is difficult to explain the similarity shown by four out of the six graphs for separate groups except in terms of common influences, namely perceived changes in the weather. No explanation can be provided for the initial maximum point in these four sets of curves, although it must be noted that this maximum occurs between 40 - 60% relative humidity, and relies on a small number of points (10 and 9 points respectively for Cowley and Mezen). Nevertheless, the right-hand extremes of these curves are based on substantial numbers of data points. The upturn between 80 - 100% relative humidity may be attributable to the emergence of an awareness of relative humidity in this range, possibly associated in some cases with rain. The convergence of data points on the polynomial curve of the original scattergrams (not shown) supports this suggestion of an increasing sensitivity to relative humidity with increases in relative humidity. The portions of these curves with a negative slope may reflect the association of relative humidity with temperature.

Figures 28 - 35 for windspeed show a peak followed by a decline in four out of six cases for the separate rooms. In all cases the curves include a substantial portion of negative gradient as windspeed increases. The one exception is for the Cowley high group where the upturn is based on one data point. No explanation can be offered for the existence of the maximum points. However, the negative gradient is consistent with the hypothesis that people close windows when windspeeds are sufficient to cause draughts or damage. In this connection a tendency for the data points to converge upon the polynomial curve of the original scattergrams (not shown) suggests that the relative importance of windspeed increases with windspeed itself.

When considering sunshine duration and rainfall, the division of householders into three groups, reveals no common pattern and adds nothing to the earlier discussion.

TABLE 1 Mean percentage of open window observations at
Cowley and Mezen

Estate	Mean percentage of open window observations			
	SIT	KIT	B1	TOTAL
Cowley	12.6	28.4	36.9	25.8
Mezen	10.0	21.7	23.8	18.1

TABLE 2 Correlation coefficients obtained between window opening
in specified room types and weather parameter values at
the hour of observation

Weather parameter	Estate	Room type			TOTAL
		SIT	KIT	B1	
Temperature	Cowley	.76**	.73**	.66**	.74**
	Mezen	.57**	.53**	.62**	.73**
Relative humidity	Cowley	-.20*	-.13	-.22*	-.23**
	Mezen	-.24**	-.07	-.31**	-.32**
Windspeed	Cowley	-.13*	-.23**	-.08	-.15
	Mezen	-.17*	-.12	-.09	-.17*
Sunshine duration	Cowley	.32**	.19*	.35**	.36**
	Mezen	.26**	-.13	.15	.12
Rainfall	Cowley	.17*	.15	.14	.14
	Mezen	-.08	.02	-.10	-.06

** p < .01

* p < .05

FIGURE 1
Standard deviations of total open window observations
for 113 households

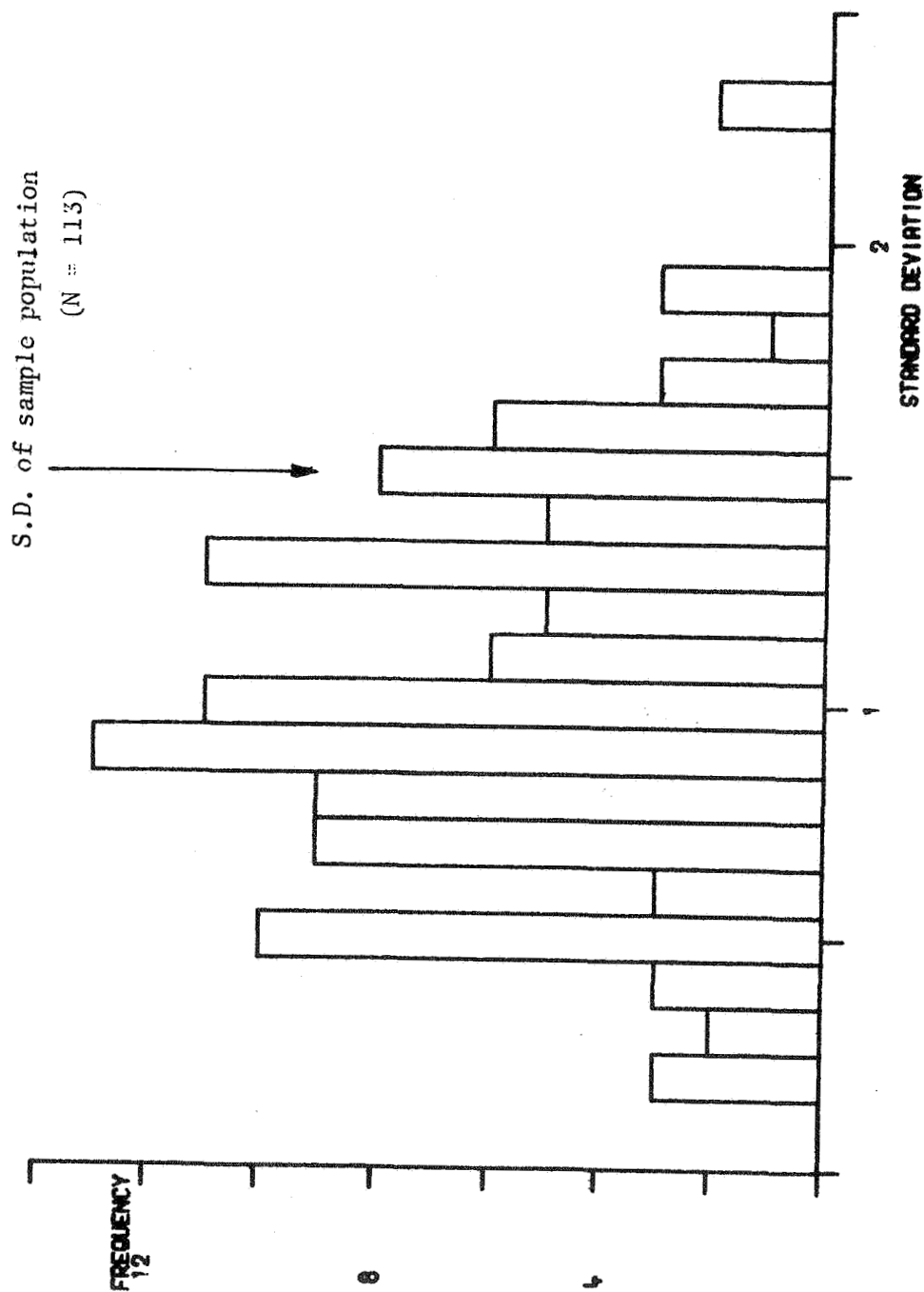
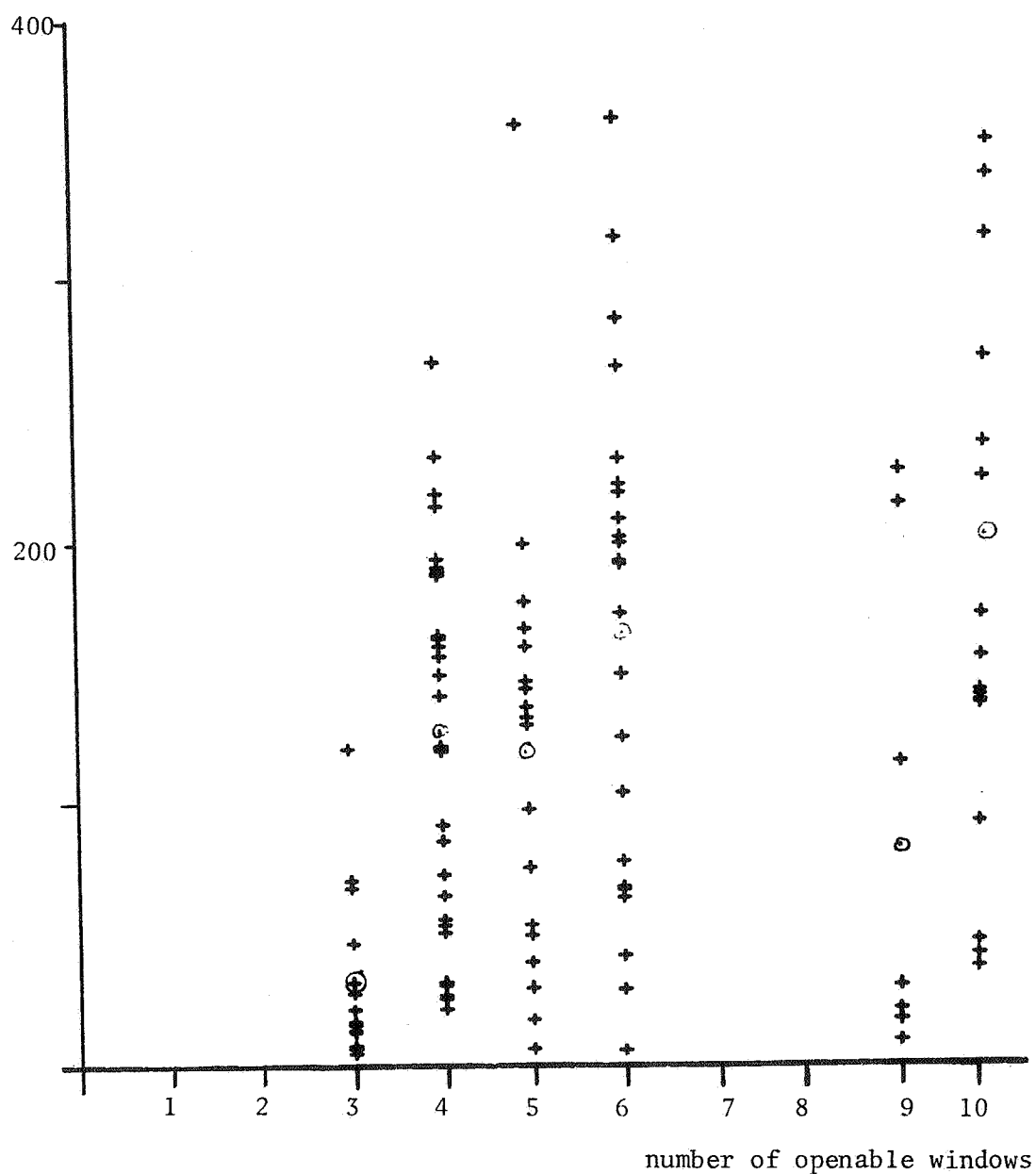


FIGURE 2 Relationship between number of openable windows and the
number of total open window observations

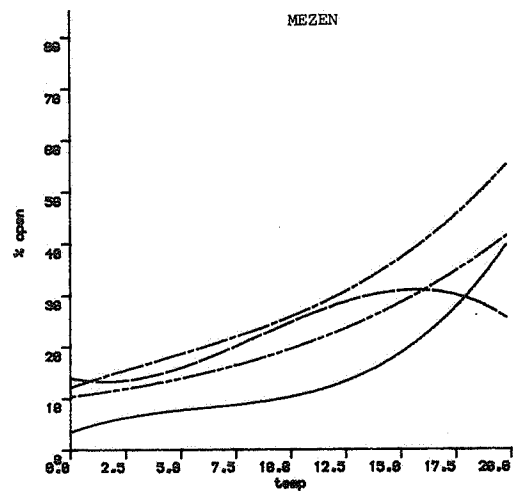
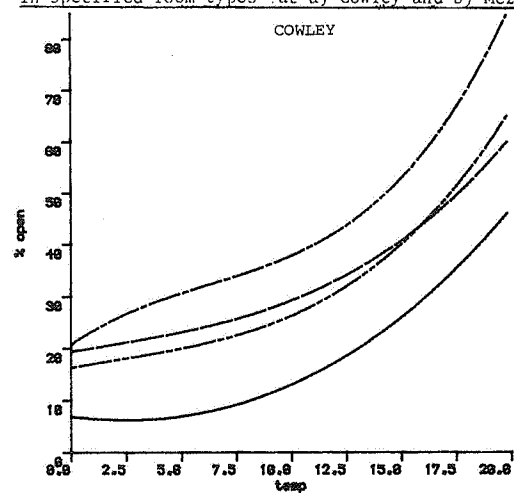
Total number of
open windows over
100 days



⊙ = mean of each group

FIGURES 3 and 4

Relationship between temperature and window opening in specified room types at a) Cowley and b) Mezen

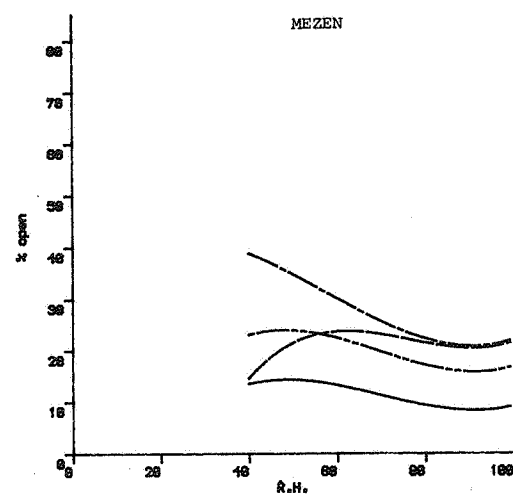
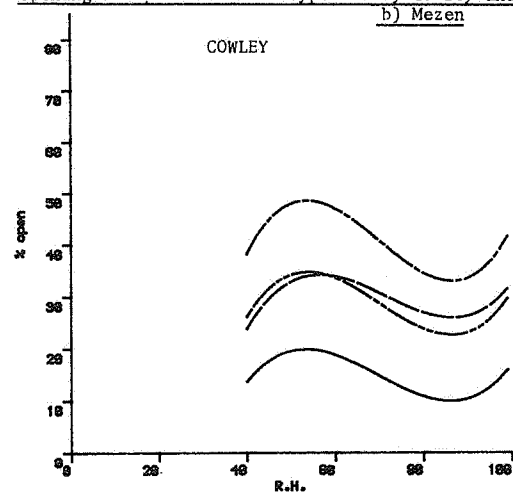


KEY:

—	SIT
- - -	KIT
- - -	BED 1
- - -	TOTAL

FIGURES 5 and 6

Relationship between relative humidity and window opening in specified room types at a) Cowley and b) Mezen

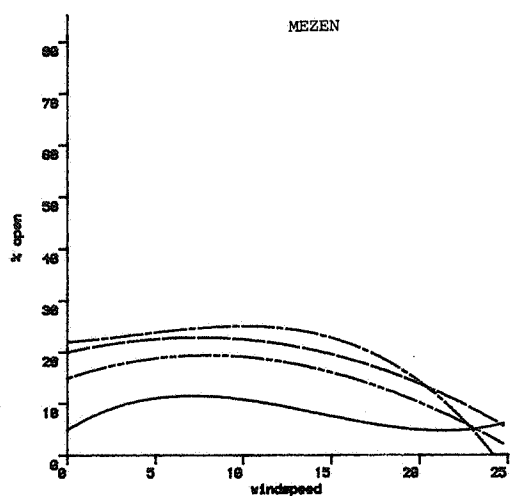
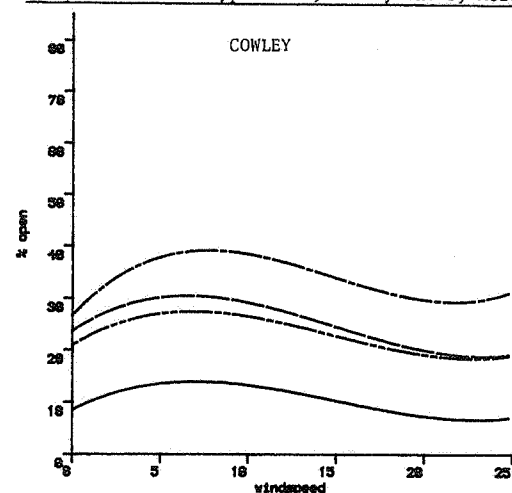


KEY:

—	SIT
- - -	KIT
- - -	BED 1
- - -	TOTAL

FIGURES 7 and 8

Relationship between windspeed and window opening in specified room types at a) Cowley and b) Mezen



KEY:

—	SIT
- - -	KIT
- - -	BED 1
- - -	TOTAL

FIGURE 9 Illustrative normal distribution curves for two households

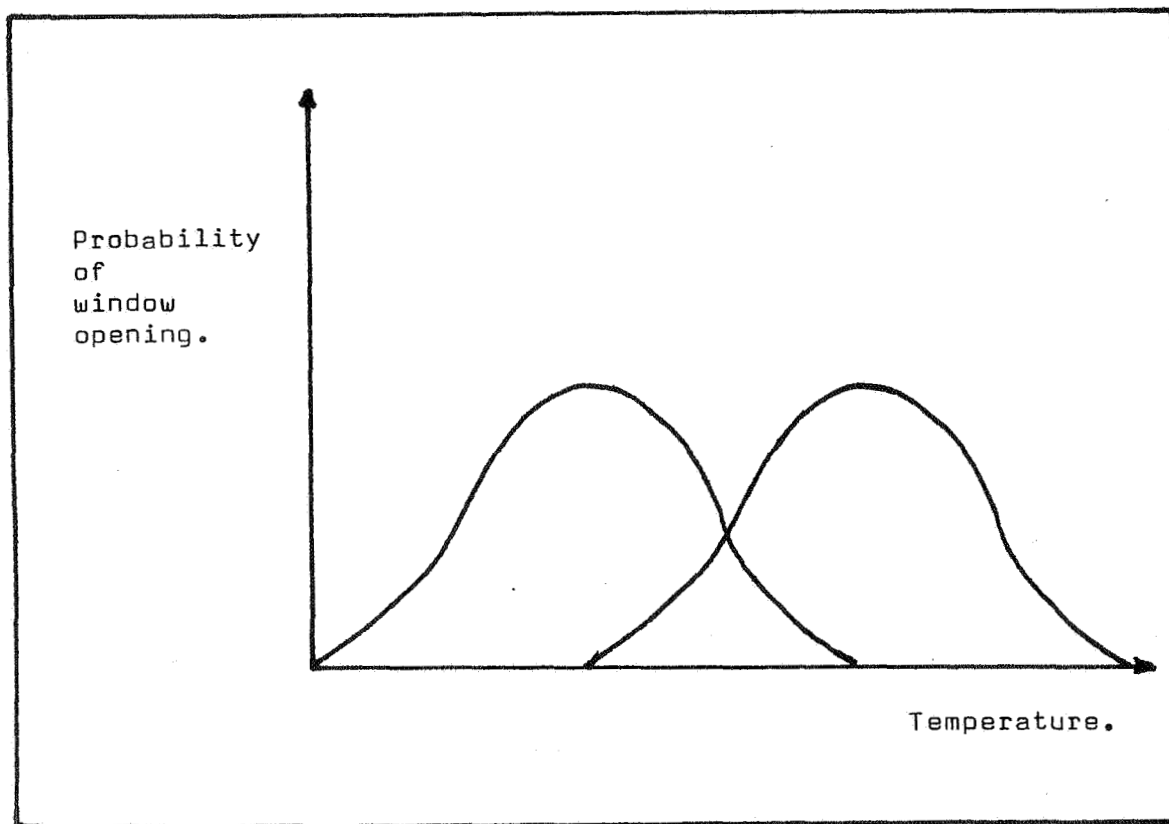


FIGURE 10 Hypothesised cumulative distribution curves for sample extremes

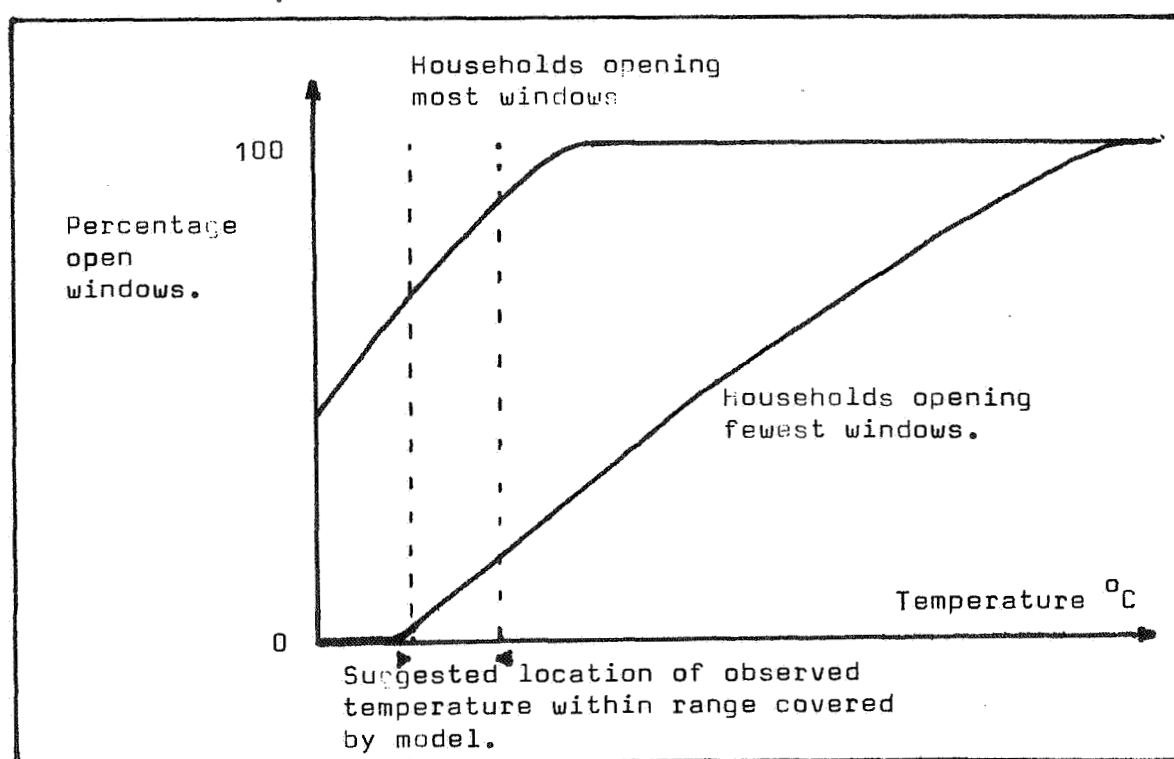
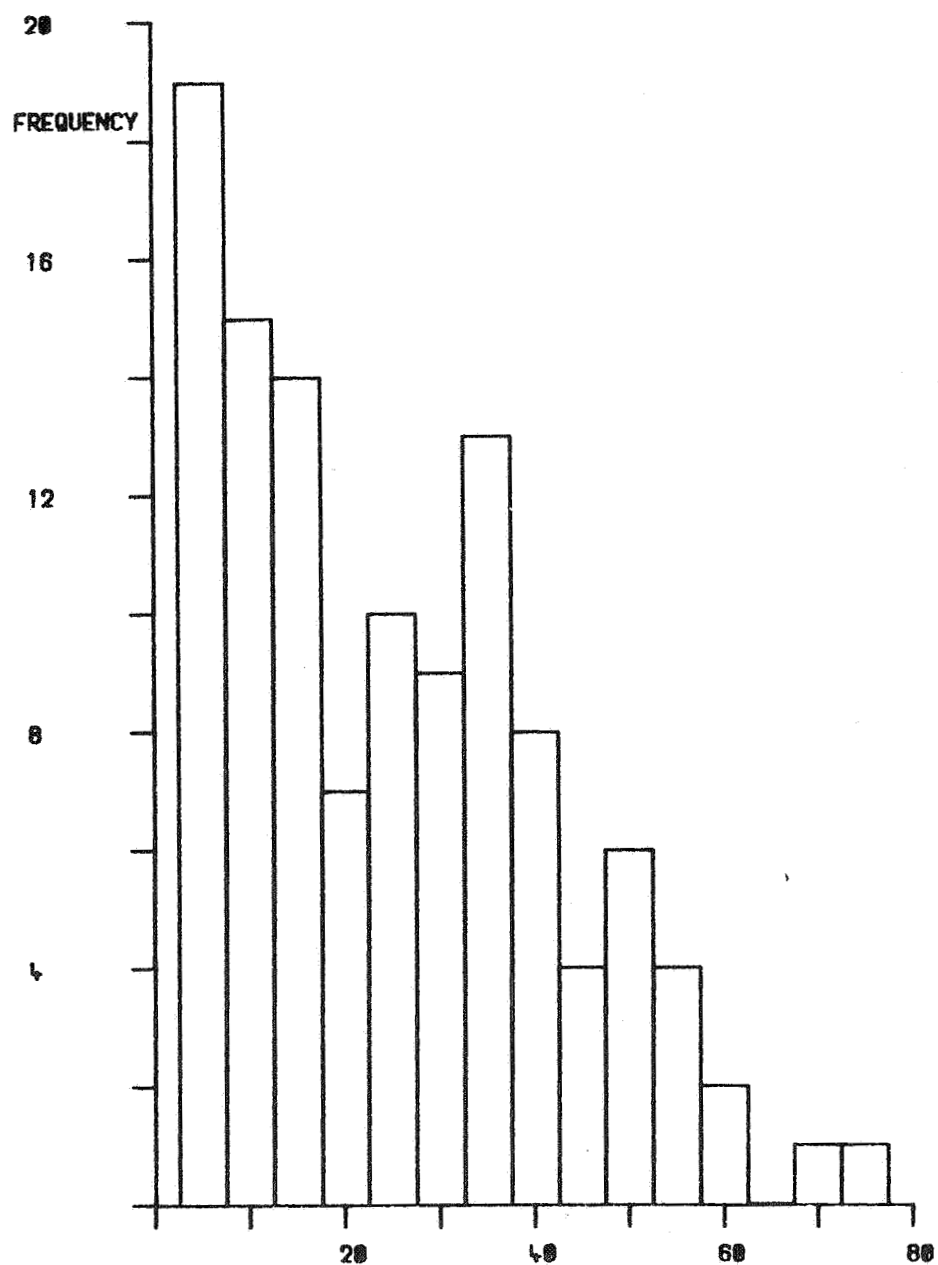


FIGURE 11

Frequency distribution of the average percentage of
total open window observations in 113 households.

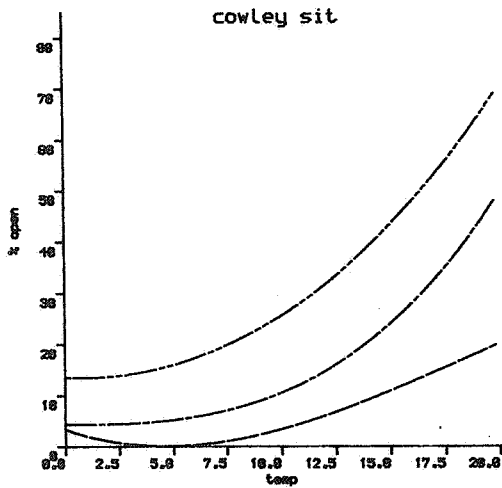
Number of households



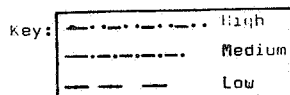
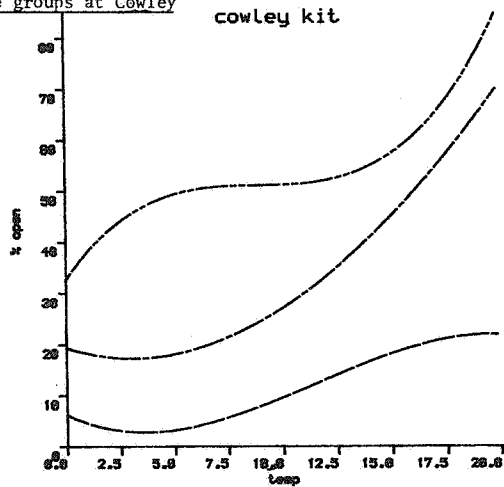
Percentage of total open window observations

FIGURES 12 and 13

Relationship between temperature and
window opening in three groups at Cowley

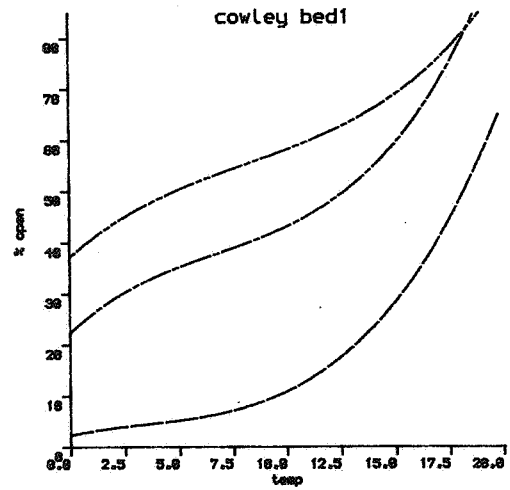


Relationship between temperature and kitchen window opening
in three groups at Cowley

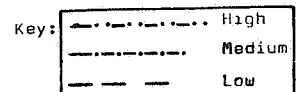
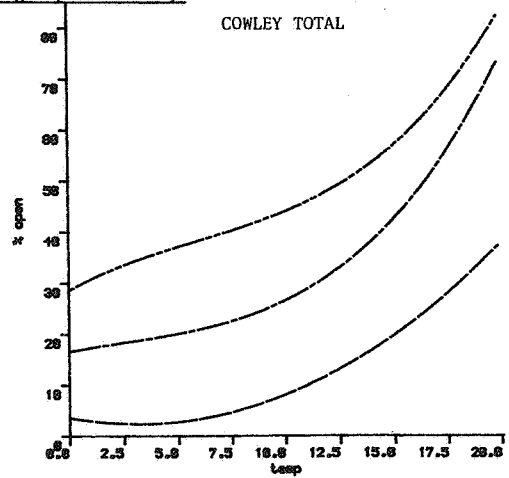


FIGURES 14 and 15

Relationship between temperature and main bedroom window
opening in three groups at Cowley

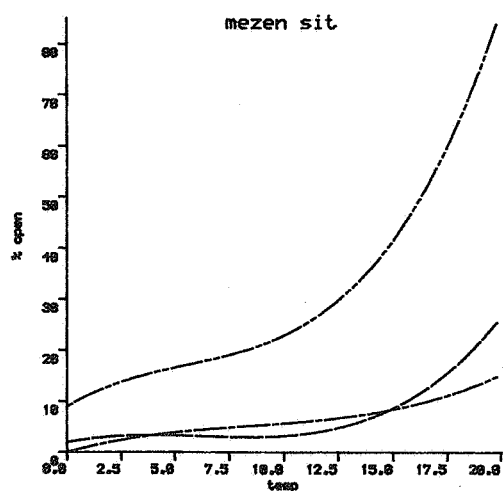


Relationship between temperature and total window opening
in three groups at Cowley

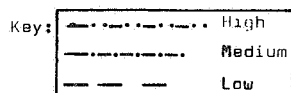
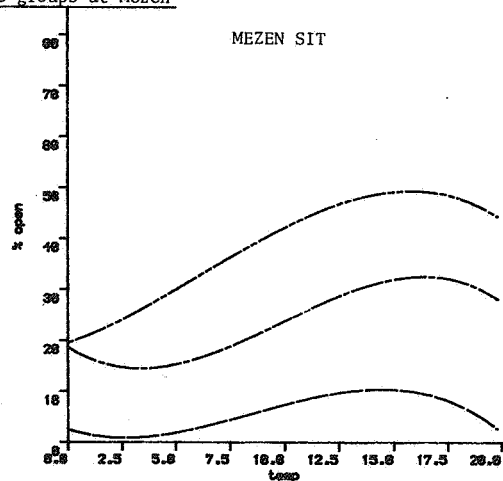


FIGURES 16 and 17

Relationship between temperature and sittingroom window opening in three groups at Mezen

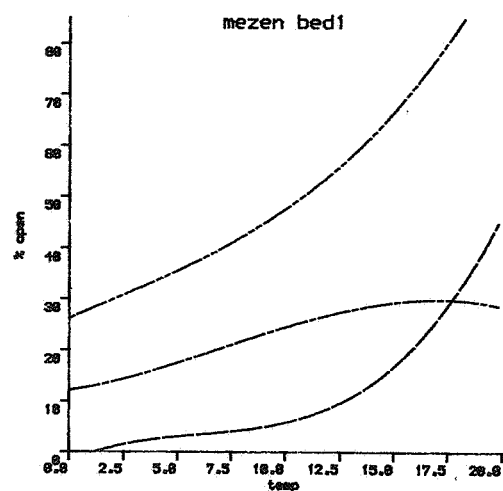


Relationship between temperature and kitchen window opening in three groups at Mezen

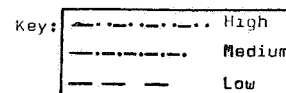
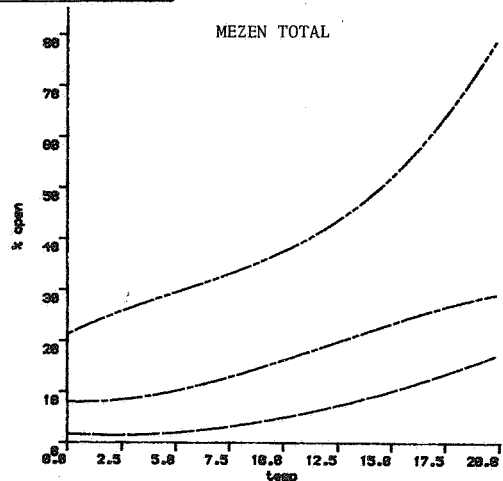


FIGURES 18 and 19

Relationship between temperature and main bedroom window opening in three groups at Mezen

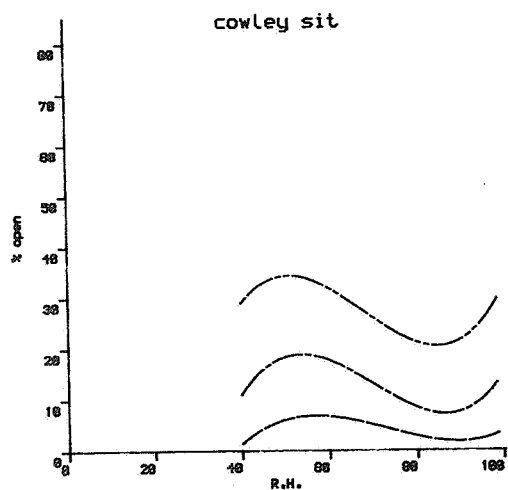


Relationship between temperature and total window opening in three groups at Mezen

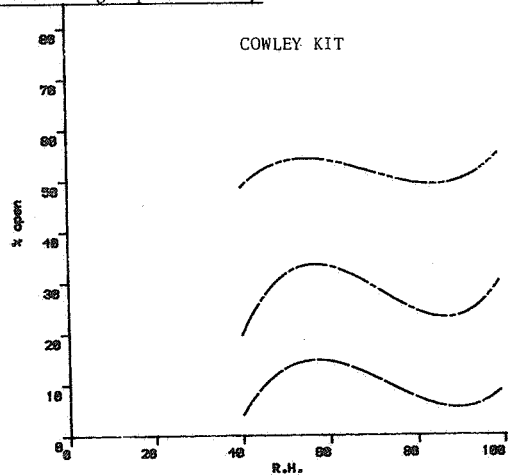


FIGURES 20 and 21

Relationship between relative humidity and sittingroom window opening in three groups at Cowley



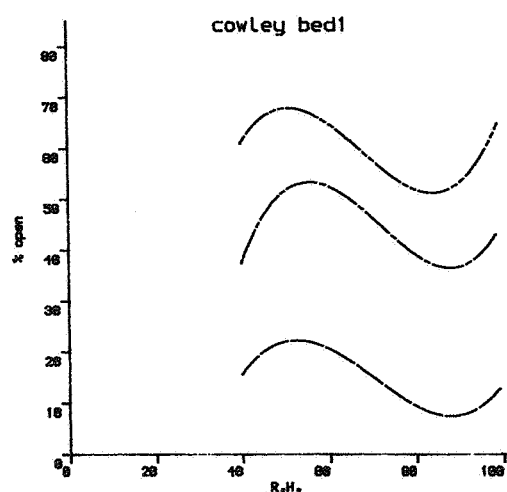
Relationship between relative humidity and kitchen window opening in three groups at Cowley



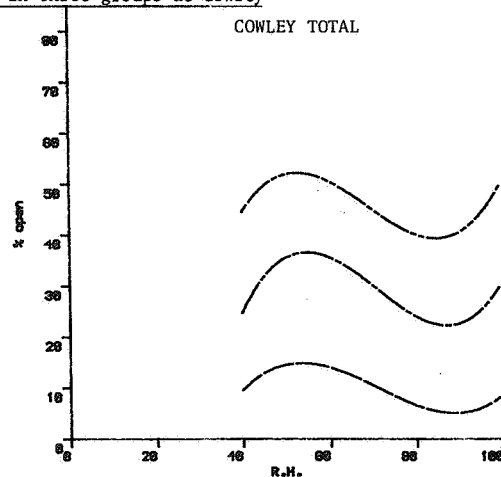
Key:
 High
 ----- Medium
 _____ Low

FIGURES 22 and 23

Relationship between relative humidity and main bedroom window opening in three groups at Cowley



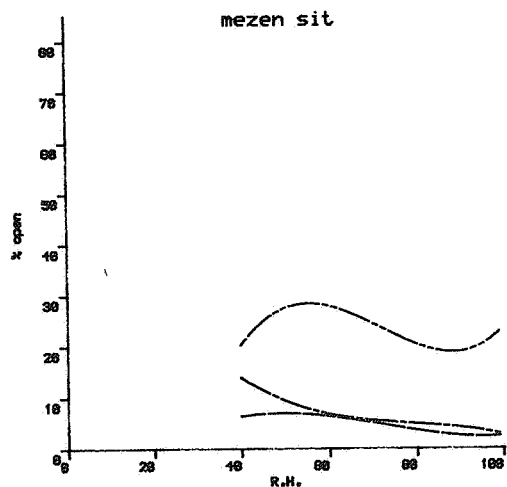
Relationship between relative humidity and total window opening in three groups at Cowley



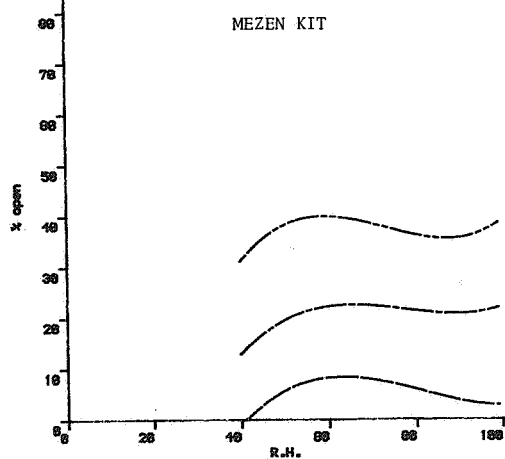
Key:
 High
 ----- Medium
 _____ Low

FIGURES 24 and 25

Relationship between relative humidity and sittingroom window opening in three groups at Mezen



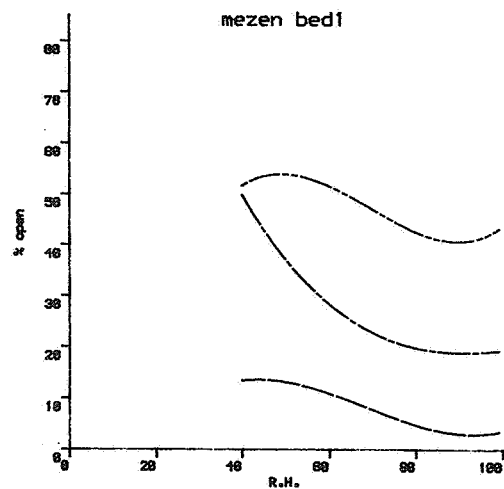
Relationship between relative humidity and kitchen window opening in three groups at Mezen



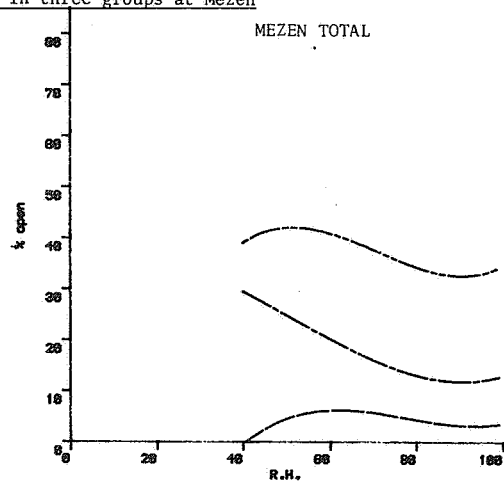
Key:
 High
 - - - - - Medium
 ——— Low

FIGURES 26 and 27

Relationship between relative humidity and main bedroom window opening in three groups at Mezen



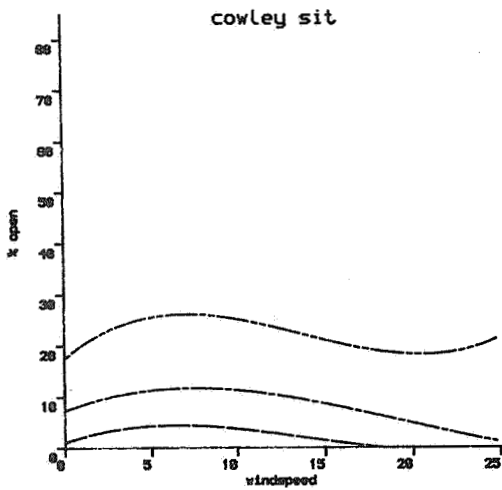
Relationship between relative humidity and total window opening in three groups at Mezen



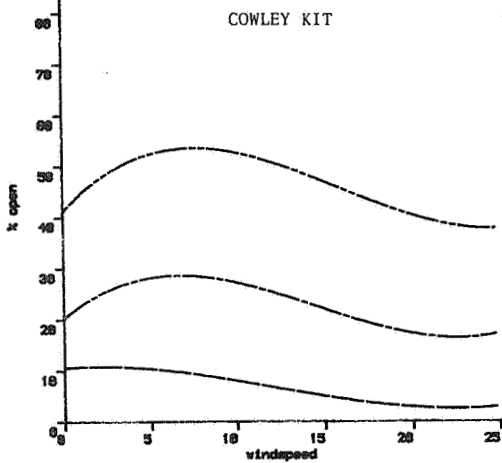
Key:
 High
 - - - - - Medium
 ——— Low

FIGURES 28 and 29

Relationship between windspeed and sittingroom window opening in three groups at Cowley



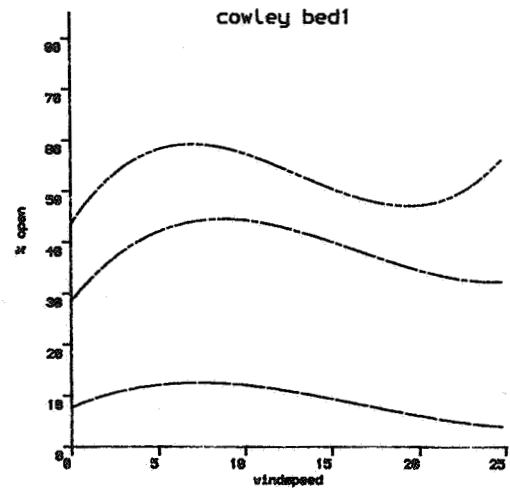
Relationship between windspeed and kitchen window opening in three groups at Cowley.



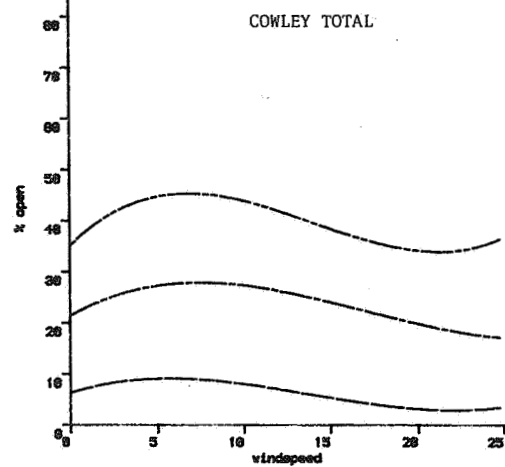
Key:
 High
 - - - - - Medium
 ——— Low

FIGURES 30 and 31

Relationship between windspeed and main bedroom window opening in three groups at Cowley



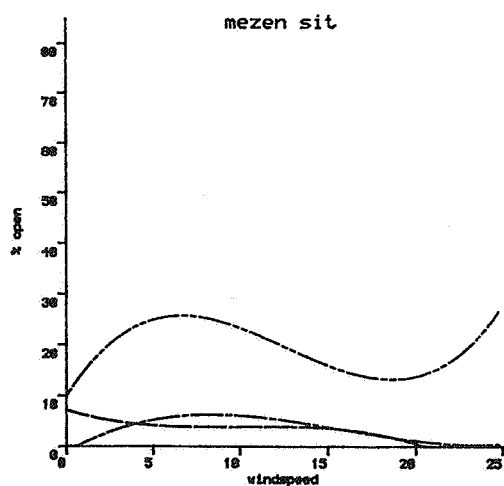
Relationship between windspeed and total window opening in three groups at Cowley



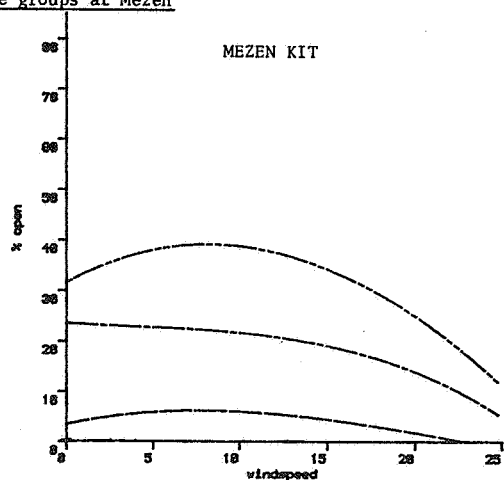
Key:
 High
 - - - - - Medium
 ——— Low

FIGURES 32 and 33

Relationship between windspeed and sittingroom window opening in three groups at Mezen



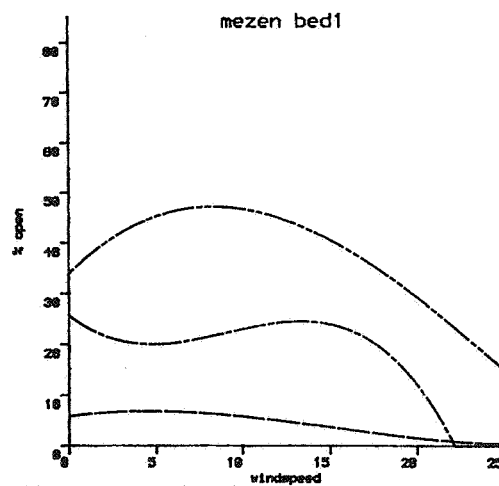
Relationship between windspeed and kitchen window opening in three groups at Mezen



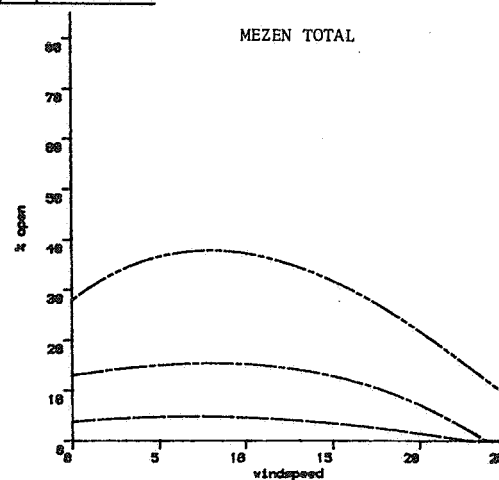
Key:
 High
 - - - - - Medium
 ——— Low

FIGURES 34 and 35

Relationship between windspeed and main bedroom window opening in three groups at Mezen



Relationship between windspeed and total window opening in three groups at Mezen



Key:
 High
 - - - - - Medium
 ——— Low

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 4

AVOIDING CONDENSATION AND MOULD GROWTH IN EXISTING HOUSING WITH
THE MINIMUM ENERGY INPUT

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SYNOPSIS

Detailed studies of public sector modernisation programmes have shown that the principal problem resulting from lack of thermal insulation and inappropriate methods of heating and ventilation is condensation and mould growth. The traditional solution in the UK has been to install permanent 'uncontrolled' ventilation openings in the external walls of the affected rooms and encourage the occupiers to increase the heat input. This is not an energy efficient solution to the problem.

Assuming typical values for moisture emission, external temperature and relative humidity, and adopting the coarse criterion of limiting the relative humidity of the room air to 70%, it is possible to show, for dwellings with particular heat loss characteristics, the optimum air change rate at which the heat input necessary to prevent the RH rising above 70% is at a minimum.

However, the average internal temperatures which result from this minimum heat input are too low to be considered comfortable and it is necessary to consider the implications of air change rates less than the optimum.

Under these circumstances, the thermal insulation characteristics of the enclosing walls etc. become important, particularly in the corners of exposed rooms where the surface temperature will be less than the remainder of the room. Given the same typical moisture emission a wall U value of $1.0 \text{ W/m}^2\text{K}$ is enough to avoid surface condensation in the corners of rooms.

LIST OF SYMBOLS

R	=	Thermal resistance of building element ($\text{m}^2\text{K/W}$)
U	=	Thermal transmittance value ($\text{W/m}^2\text{K}$)
T_R	=	Room air temperature ($^{\circ}\text{C}$)
T_S	=	Surface temperature ($^{\circ}\text{C}$)
T_E	=	External air temperature ($^{\circ}\text{C}$)
R_{si}	=	Internal surface resistance ($\text{m}^2\text{K/W}$)

1. CONDENSATION IN EXISTING HOUSING

Detailed studies of public sector modernisation programmes, particularly those for Local Authorities and Housing Associations (ie. private organisations receiving subsidy in order to sponsor housing for rent) have shown that in many cases insufficient emphasis had been put on incorporating, as part of the renovation work, recommended energy conservation measures. The standard of thermal performance of these dwellings was therefore substantially less than new dwellings offering the same accommodation.

As a result there were frequent reports of condensation and mould growth in dwellings, often within a few months of occupation.

The poor thermal characteristics of these existing dwellings can be attributed, not only to their general lack of thermal insulation, but also to their physical characteristics (shape and size).

1.1 Typical dwelling types

The types of dwelling found in this type of housing rehabilitation range from flats and maisonettes which result from the conversion of large 4 storey city mansions, to family units in small two storey houses. The variation in area and volume is considerable.

It is common for there to be an addition to the main terrace of the building. These "back additions" accommodate Kitchens, Bathrooms, and Bedrooms and are sometimes, but not always, back to back. In all cases their enclosing wall area is high in relation to their volume. In the large city conversions, the basement often accommodates Bedrooms and the roof space a small flat.

The main features of each location, insofar as they relate to thermal performance, are as follows:

Basements

- * Windows on the road side are often large but derive no benefit from solar gain.
- * There is no convective heat gain from below.
- * Internal temperatures can be raised only by means of an applied heat source. Internal heat gain is negligible.
- * Floor to ceiling heights are between 2.4 and 2.5 metres.

The main terrace

- * In large city flat conversions, ground floor rooms (usually Living Rooms, Dining Rooms or Dining/Kitchens) have high ceilings, in the order of 3 metres.
- * Where Bedrooms are above Living Rooms, they will benefit from convective heat flow.
- * Some benefit may be derived from solar gain depending on shading by adjacent buildings.
- * Due to replanning, there may be an internal Bathroom, mechanically ventilated.

Back addition

- * Back additions often accommodate moisture generating rooms (Kitchens and Bathrooms).
- * Bedrooms are often situated above Kitchens. They will receive water vapour from below but no useful convective heat from the Living Rooms in the main terrace.
- * Rooms in back additions are not usually provided with a fixed heat emitter.

It was particularly noticeable in the case of Housing Associations, that the standard of renovation varied considerably, depending on the regional cost limits in force. In many cases, only short term repairs were carried out and even the 'improvement' of amenities was kept to an absolute minimum. For example, only loft insulation was considered as "essential" and often, heating was restricted to the Living Room only. There was also no requirement even for simple draught stripping of large windows to be carried out. The result was that many tenants on low incomes were not able to maintain sufficiently high temperatures to avoid condensation and mould growth.

Generally, Housing Association staff were aware of the inadequacy of these measures, but were forced to wait until condensation and mould growth had occurred before they could claim extra funds for additional insulation and heating.

1.2 The condensation problems

Traditionally, these old dwellings were heated by burning cheap fuel on open fires, which ensured sufficient heating and ventilation. Apart from cooking, moisture generating activities such as bathing and internal clothes drying did not take place on the same scale as today. Similarly, households did not have the benefit of non-electric portable heating appliances.

The problem is that tenants are encouraged to spend their money on the amenity provided by labour saving appliances such as tumble dryers, but they are then unable to afford a level of heating throughout the dwelling, which is necessary to avoid condensation, especially with the poor level of thermal insulation provided. In other cases portable heating appliances run on calor gas or paraffin are purchased because fixed economical arrangements are not made for heating their dwellings.

Clearly, the current provision of heating and thermal insulation is not providing an adequate solution. But opinions differ on where the emphasis should be placed.

1.3 Alternative solutions

Traditionally, remedies for condensation problems were based on increasing heat input and providing permanent (uncontrolled) ventilation in the form of air bricks, this being cheaper to install than additional thermal insulation.

This solution inevitably puts a heavy financial burden on the tenant and at the same time provides no increase in comfort. It is understandable therefore, that tenants will block up the ventilation openings in an attempt to keep warm.

Alternatively, the provision of additional thermal insulation, although more costly, enables higher temperatures to be achieved for less heat input and being part of the building fabric cannot be "turned off". It is therefore guaranteed to work in the tenants' best interests in avoiding condensation problems.

2. CONDENSATION PREDICTION CRITERIA

There is general agreement on two criteria by which to predict whether condensation and mould growth is likely to persist as a problem.

The first coarse test of condensation risk is that the relative humidity of a room or dwelling should on average over a given period be no greater than 70%.

Secondly, the relative humidity of the air at the enclosing surfaces of a room should be maintained below 100% to avoid the formation of surface condensation.

In the past, the first criterion has been used as a general indicator that if the room air is kept below 70% RH, mould is unlikely to grow on organic materials, but also that the air at internal surfaces of external wall and roof elements would not be saturated. Whilst this general rule may apply for the main wall area, particularly if it is insulated, it may not be the case in the corners of rooms and behind curtains where air movement

is restricted and a greater internal surface resistance results in reduced surface temperatures. Under these circumstances surface condensation can occur when the room RH is less than 70%.

The relationships between heating, ventilation and thermal insulation in achieving the above criteria for the least heat input may be illustrated graphically.

3. WHOLE DWELLING ANALYSIS

The relative influence of the three variables is illustrated in Figures 1 and 2, the first relating to a 5 person, 3 Bedroom house (volume 230m³) and the second showing the position for a ground floor/basement maisonette (2 storey flat) of similar accommodation but much greater size (volume 326m³). The floor plans are shown in Figures 3 and 4.

The two stages of the calculation are shown together on the graph. Firstly, relating to the left hand scale are the average dwelling air temperatures which are necessary to maintain the relative humidity at 70% at a range of ventilation rates.

Curves related to the right hand scale show, for the same ventilation rates, the heat input necessary to achieve these dwelling temperatures, depending on their heat loss characteristics (in these cases influenced also by the dwelling location - middle or end of terrace).

In deriving these curves, the following assumptions were made:

- 1 - The time period is 24 hours.
- 2 - The moisture emission over 24 hours is 8.76 kg:

	<u>kg</u>
Occupants respiration	3.76
Cooking (3 hours)	3.00
Washing dishes	0.50
Washing clothes *	1.00
Bathing	0.50
	<u>8.76</u>

* It is assumed that clothes are dried outside the dwelling.

- 3 - The average external temperature is 4°C, typical of the London Area during December, January and February.

The following conclusions may be deduced from the graphs:

- (1) As the ventilation rate is increased the average internal temperature requirement decreases.

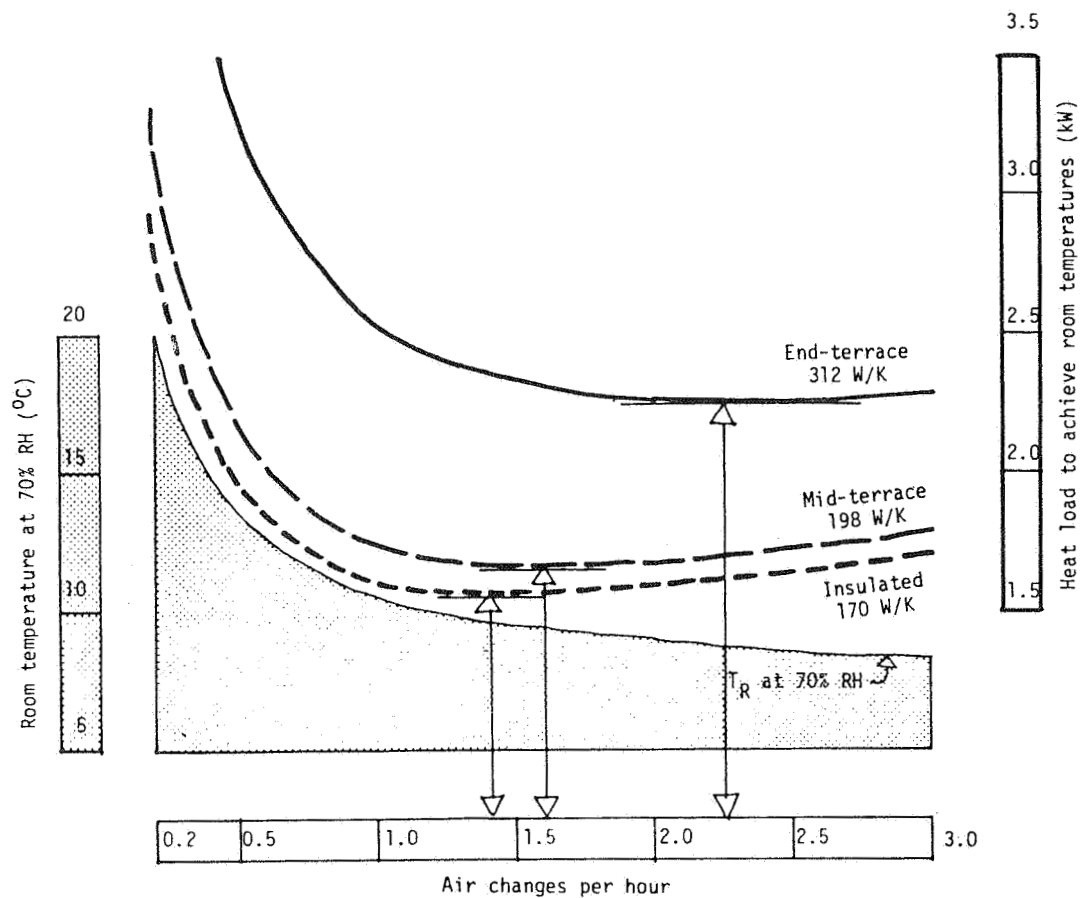


Figure 1 Room temperatures and corresponding heat loads necessary to achieve a relative humidity of 70% in a modernised HOUSE

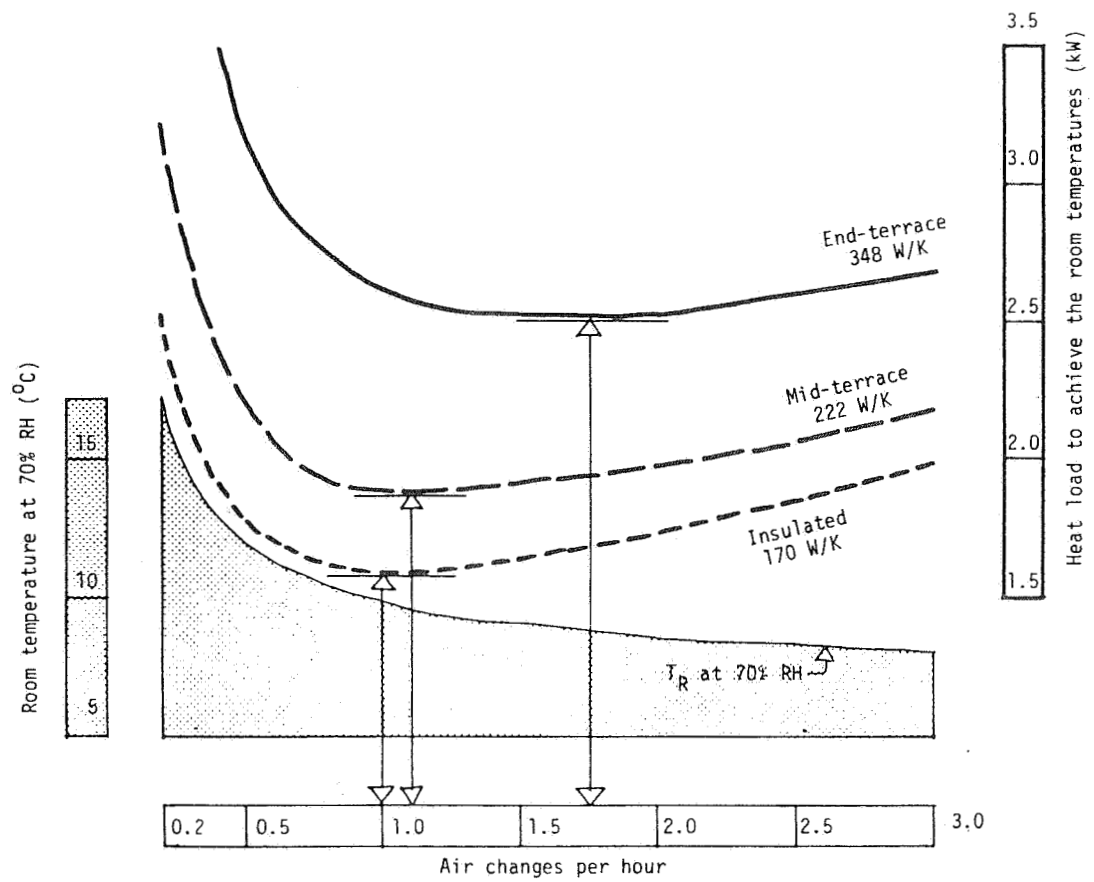


Figure 2 Room temperatures and corresponding heat loads necessary to achieve a relative humidity of 70% in a modernised MAISONETTE

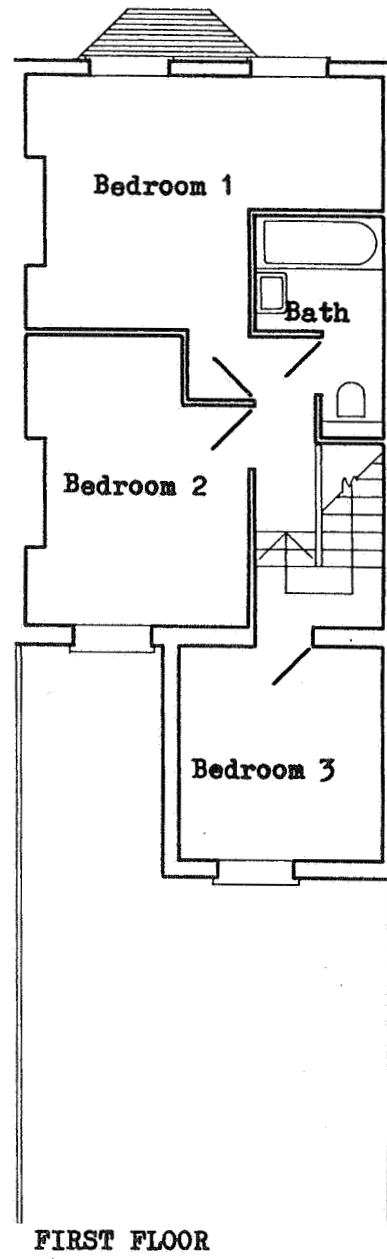
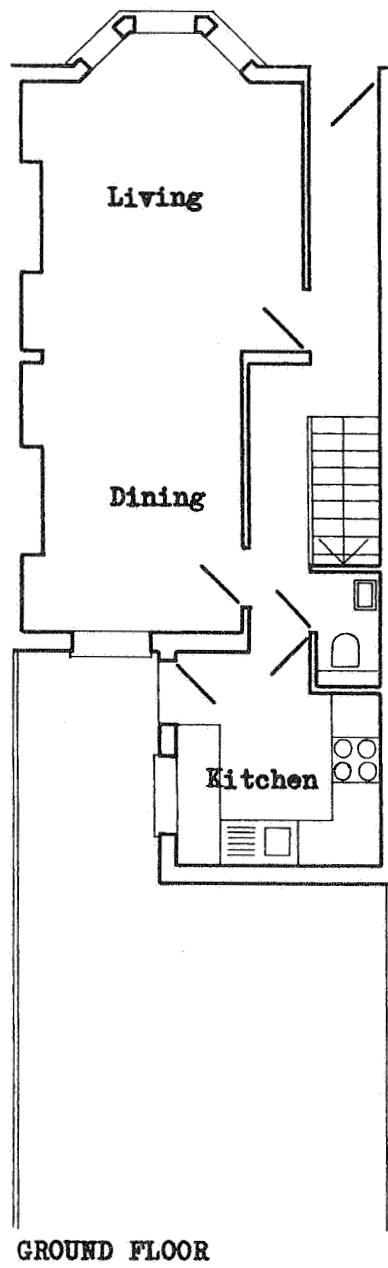


Figure 3 Floor plans of a typical modernised HOUSE

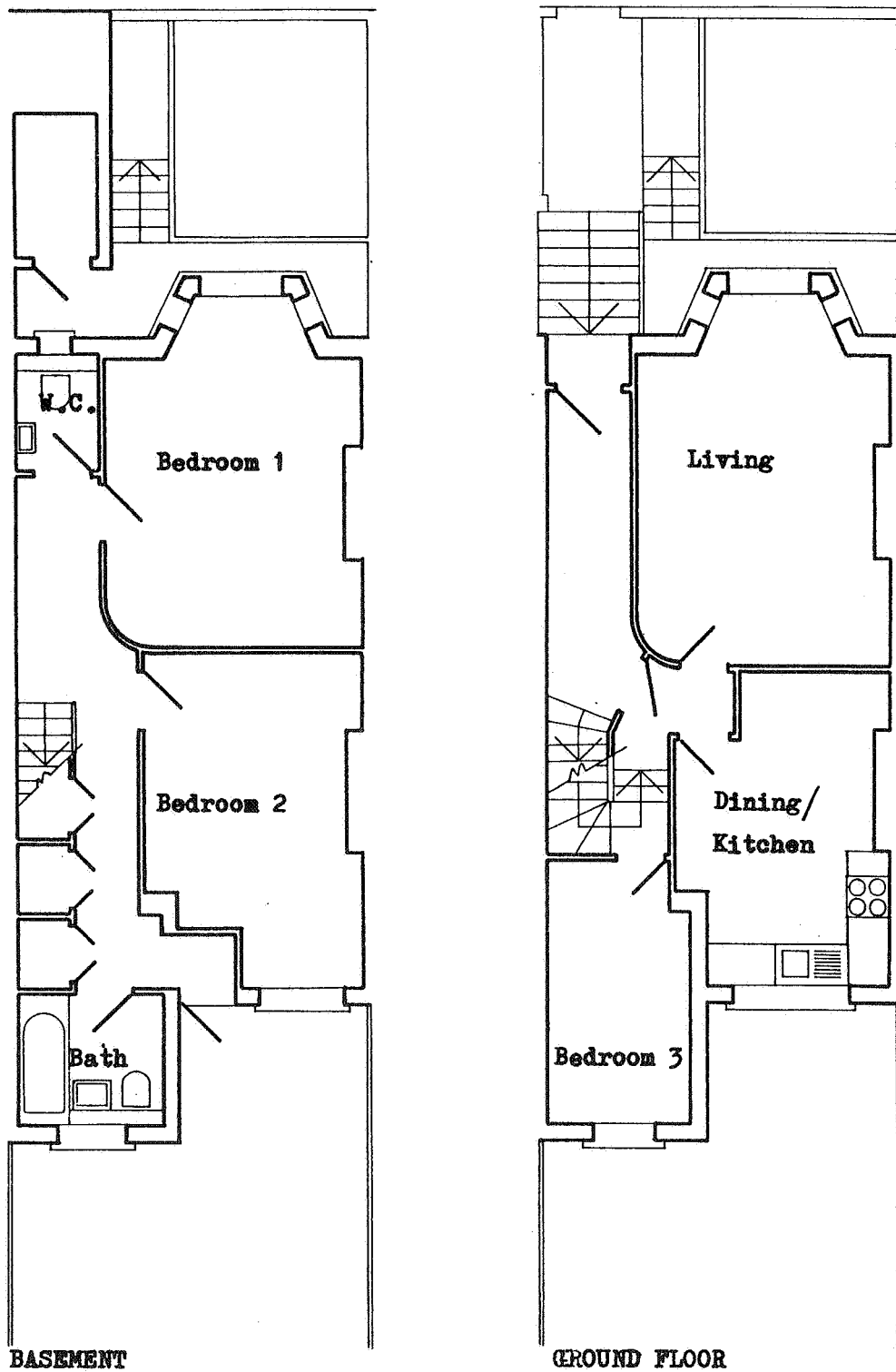


Figure 4 Floor plans of a typical modernised MAISONETTE

- (2) In all cases, there is an optimum air change rate at which the heat input necessary to achieve the required internal temperature is at a minimum.
- (3) The optimum ventilation rate becomes less as the heat loss characteristics of the dwelling are improved (ie. as the characteristic heat loss rate is reduced by additional insulation). This suggests that measures to reduce air infiltration and control ventilation may be taken without increasing potential condensation problems, provided the dwelling is insulated at the same time.
- (4) The traditional approach of increasing ventilation rates is self defeating; the required heat input is increased and is accompanied by a reduction in internal temperature, below that acceptable for thermal comfort.
- (5) If the air change rate is reduced below the optimum, it must be accompanied by an increase in heat input and/or a reduction in moisture emission to avoid condensation risk.
- (6) Acceptable anti-condensation conditions can be maintained at lower temperatures in large dwellings than in small units, because the water vapour produced is diluted in a greater volume of air.
- (7) Given the moisture emission of 8.76 kg/day, an external temperature of 4°C and an equivalent characteristic fabric heat loss rate of 170 W/K for both types of dwelling (achieved by insulating external walls in the house example and in addition insulating the Basement floor and double glazing the Living and Dining Room in the flat), we derive the following optimum air change rates and heat inputs as well as the associated average house temperatures achieved under these conditions:

	<u>Ach/hr</u>	<u>Heat input</u>	<u>Average house Temperature</u>
House	1.5	1.56 KW	9.50°C
Flat	1.0	1.60 KW	9.75°C

Figure 5 shows the heat input curves for the house and the flat when insulated to achieve a characteristic fabric heat loss rate of 170 W/K. It should be noted that the average internal temperature is less than 10°C for the whole house and may be considered unacceptably low for comfort.

4. ANALYSIS OF BEDROOMS

This technique may be extended to an analysis of the single and double Bedrooms in our typical rehabilitated house example, but in this case it is assumed that the outside temperature is 0°C and the time period 8 hours.

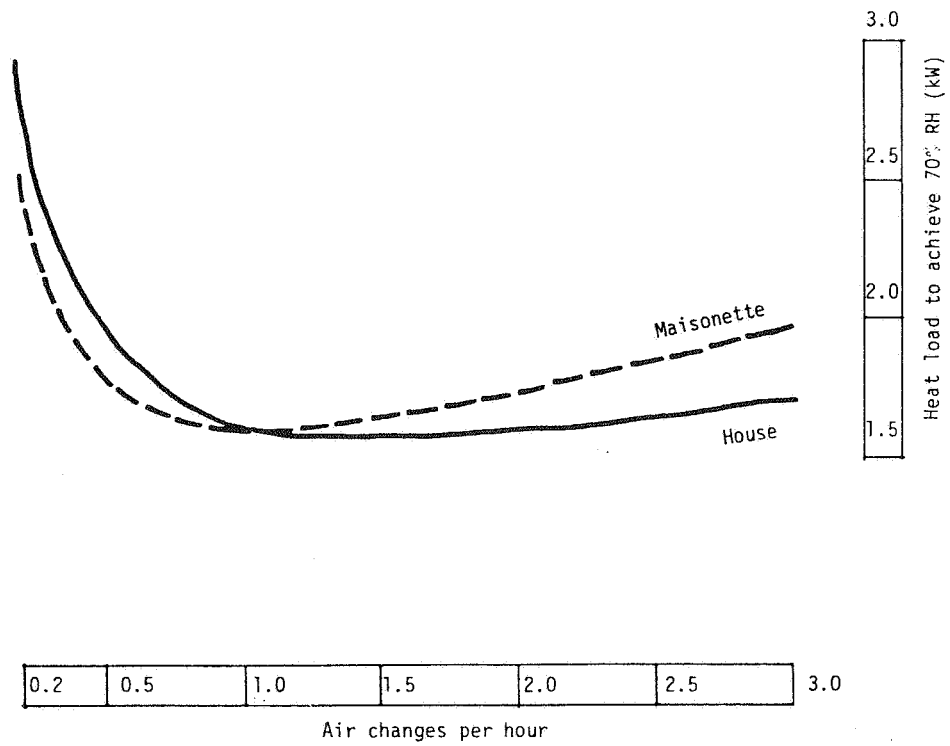


Figure 5 Comparison between the required heat inputs for the house and maisonette insulated to provide the same fabric heat loss characteristic (170 W/K).

Figures 6 and 7 show the 70% RH and dewpoint temperatures for these typical rooms.

In practice, moisture is introduced into the Bedrooms from the occupants or from the remainder of the house and ventilation air from the house or from the outside air. The change in room moisture content will not take place immediately but there will be a gradual change until an equilibrium moisture content is reached. The time taken will vary with room volume and air change rate. For these moisture contents there will be a corresponding temperature at which the relative humidity is 70%.

In the single Bedroom, for example, at a ventilation rate of one air change per hour, this equilibrium condition is reached after 3 hours of occupation. The required temperature at the start of the occupancy period may be higher or lower than the 8.8°C equilibrium temperature depending whether the moisture in the room air is assumed to be taken from the rest of the house or from the external air. The equilibrium (70% RH) temperature for the double Bedroom is much greater than the single room because of its greater occupancy to volume ratio.

The lower the air change rate, the longer it will take for the steady state to be reached. This could be all night for a double Bedroom at a ventilation rate of a half an air change per hour.

When, therefore, the wind direction is such that ventilation air is being drawn into the room from the rest of the house, the moisture content of the Bedroom air will be generally greater than when the Bedroom is on the windward side. In the latter case, the moisture content of the room air during the unoccupied period will tend towards that of the external air, thus reducing the room temperature necessary to avoid the RH rising above 70%.

The moisture content will also be reduced if the majority of the water vapour from the Kitchen and Bathroom is extracted at source. For example, the use of selective mechanical ventilation in these rooms could depress the Bedroom temperature requirement by 2°C.

The more the temperature required to maintain 70% RH can be reduced, the less heat is needed to offset the conditions likely to cause condensation.

This preliminary consideration of the dynamic condition shows that, although the temperature and therefore the heat requirements increase at low ventilation rates, the risk in practice may be reduced because of the greater time taken to reach the equilibrium condition. The time taken will of course depend on the source of the ventilation air. This approach needs further study.

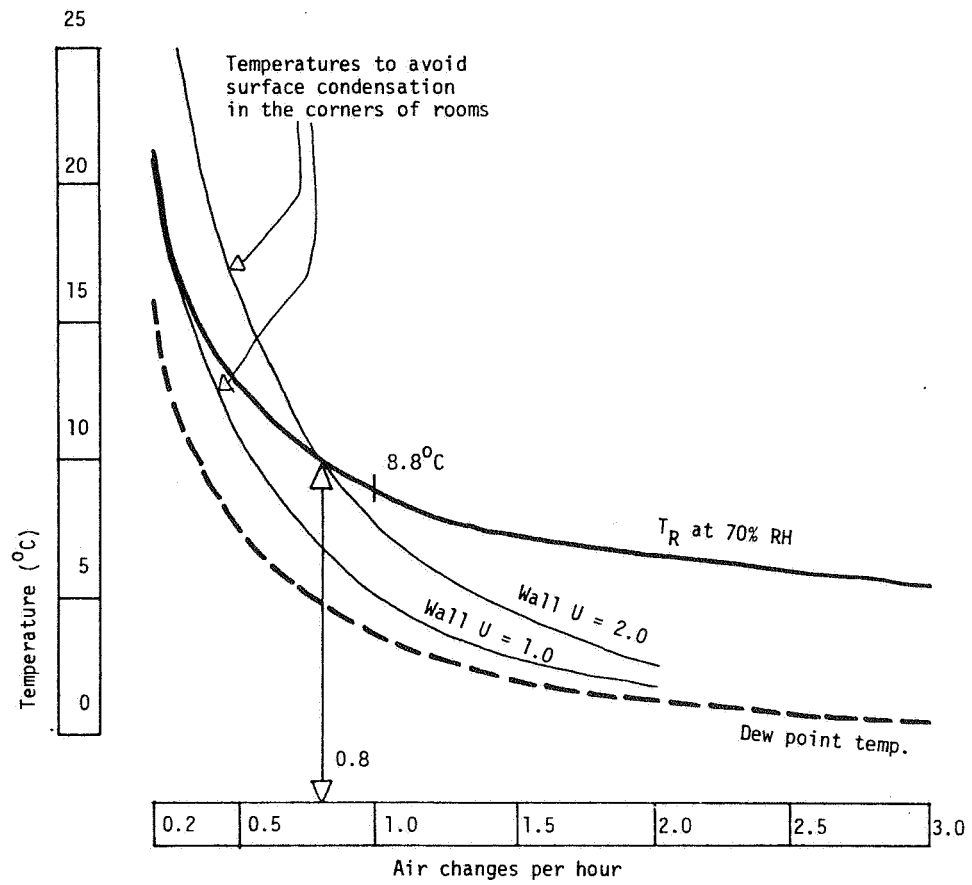


Figure 6 The relationship between airchange rate and internal temperature in a typical SINGLE BEDROOM

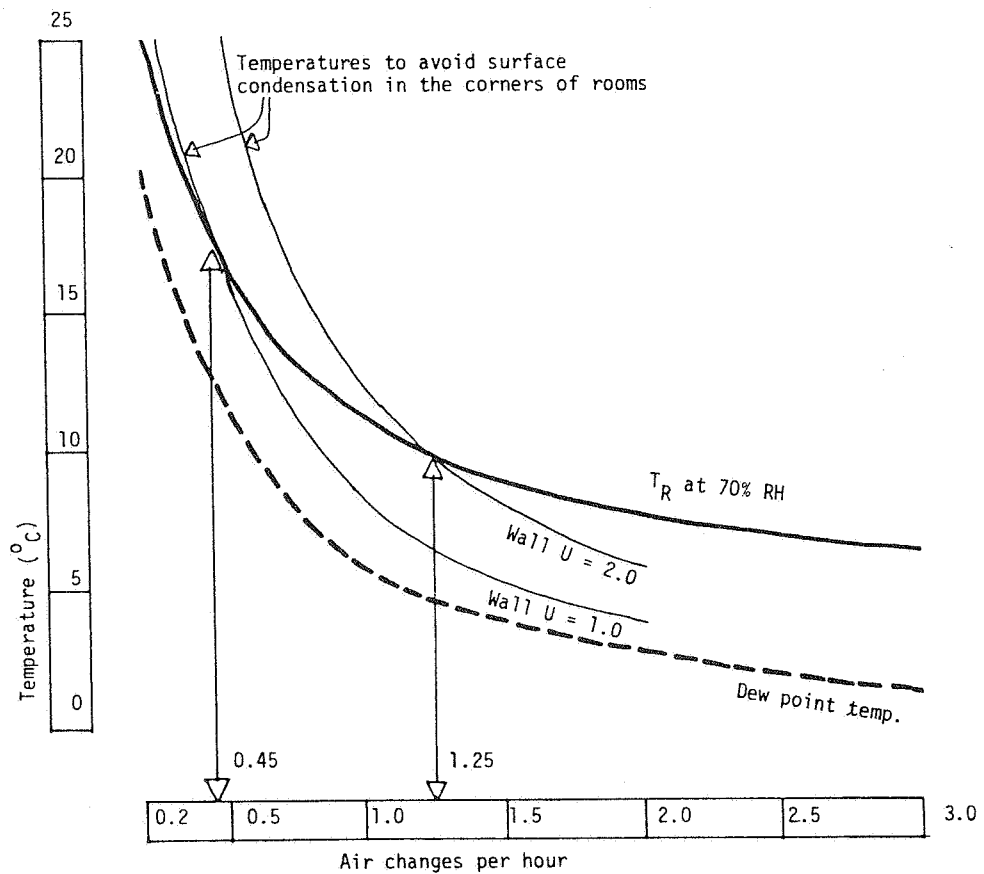


Figure 7 The relationship between air change rate and internal temperature in a typical DOUBLE BEDROOM

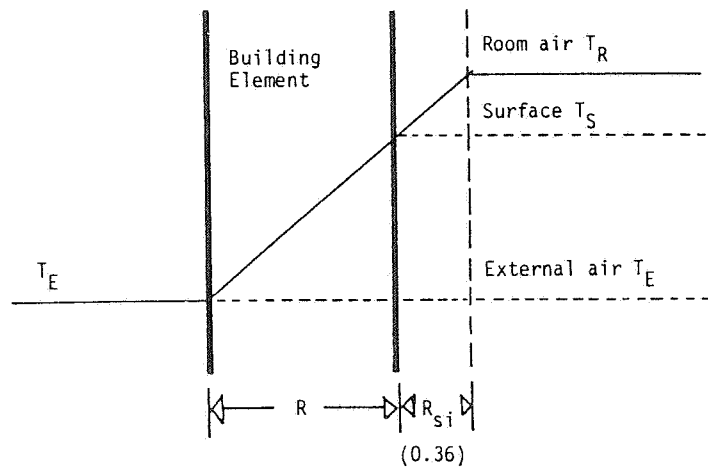


Figure 8 The temperature profile across a building element at an exposed corner of a room.

5. SURFACE CONDENSATION AT LOW AIR CHANGE RATES

The analysis so far has been based on the use of the initial coarse criterion for avoiding the risk of condensation ie. that the average temperature of the room air should be such that the relative humidity does not rise above 70%. This has identified optimum air change rates when the heat input to achieve this RH is the least.

However, the internal temperatures at these air change rates are perhaps too low for comfort. It is important, therefore, to explore the situation when air change rates are reduced to 1 or even $\frac{1}{2}$ an air change per hour.

Under these circumstances, when the enclosing surfaces of the room are not well insulated, it is possible for the air at the surface to reach saturation before the air in the room generally reaches 70% relative humidity. This is particularly true in the corners of rooms or behind curtains - indeed any location where, due to lack of air movement, the internal surface resistance to heat transfer is increased above the normal condition and where the external surface resistance tends to zero due to the high exposure at the external corners of a building.

It is German practice to quantify the situation at the internal corners of exposed rooms, by assuming that the internal surface resistance to heat transfer is three times the normal value (ie. 0.36 rather than 0.12 m² K/W) and that the external surface resistance is zero.

We can therefore derive a relationship between the U values of the enclosing elements at the corners of rooms and temperatures on their surface and in the room generally. (See Figure 8)

Given, for example, a construction element with a particular 'standardised' U value, we can determine its own specific thermal resistance (R) by subtracting the sum of the 'standard' external and internal surface resistances and then, knowing the dew point temperature of the room air, calculate the room air temperature necessary to maintain surface temperatures in the corners above the dew point.

$$R = \frac{1}{U} - 0.18 \quad \dots\dots\dots (1)$$

$$T_R + \frac{(T_S - T_E) \times 0.36}{R} + T_S \quad \dots\dots\dots (2)$$

Figure 6 shows for the single Bedroom example how, with alternative U values of walls, the room temperatures necessary for surface condensation to occur in the corners of rooms are either within those determined by the 70% RH criterion (as is the case when the wall U value is 1.0 W/m²K) or over-ride it at a certain air change rate, (in the region of 0.8 air changes per hour when the wall is uninsulated having a U value of 2.0 W/m²K).

For the double Bedroom, the ventilation rates at which the corner condition becomes the critical factor is 1.2 ach/hr for a wall with a U value of 2.0 and 0.45 ach/hr for one with a U of 1.0 W/m²K). (See Figure 7)

In both cases, the required temperatures increase dramatically below these critical air change rates. Clearly, there will be a correspondingly high heat input required to achieve these temperatures in rooms which have exposed corners, such as at the ends of terraces and "back additions".

The corresponding heat loads for our Bedroom examples have been calculated for different locations and levels of insulation and are shown in Figures 9 and 10.

Figure 9 shows that for pitched roof single Bedrooms in back additions at the end of a terrace, (BA-2U-ET), the heat load required to avoid surface condensation in the corners of rooms when the walls are not insulated increases dramatically when the ventilation rate drops below 0.8 ach/hr. The same characteristic applies to single Bedrooms at the gable end of the main terrace (T-2U-ET). This suggests that uninsulated single Bedrooms in these locations should not be ventilated below say one air change per hour, unless the daytime heat input is sufficient to evaporate the condensation formed during the night.

The addition of wall insulation to achieve U value improvement from U = 2 to U = 1 W/m²K, (BA-1U-ET) avoids this situation and makes it possible to reduce the ventilation rate, achieving more comfortable internal conditions for the same heat load. However, when the walls are insulated, a heat load of some 275 W is still necessary, even if the ventilation rate is increased. This must be provided in back additions by a heating appliance, there being insufficient heat leakage from the rest of the house.

In contrast, a typical single Bedroom within the main terrace with walls uninsulated (T-2U) needs approximately one half the heat input required by a similar uninsulated bedroom in a back addition at a gable end. In this case, the heat load remains more or less constant at 220 W at all ventilation rates above 1.5 ach/hr. To increase the ventilation rate of a mid-terrace single room only serves to lower internal temperatures unnecessarily.

In double Bedrooms, not only are higher temperatures required overnight but the heat loads to achieve these temperatures are also greater, particularly as the air change rate is reduced (see Figure 10). The heat load for an uninsulated end of terrace Bedroom (DB-2U-ET) is 400 W even at 3 ach/hr rising to 550 W at 1 ach/hr. However, if the room is insulated (DB-1U-ET) the requirement drops to just above 300 W almost irrespective of air change rate. Clearly, as the room is better insulated, increased ventilation becomes a less attractive solution. Furthermore, whilst under steady state conditions poorly insulated rooms can be shown to require less heat input to avoid

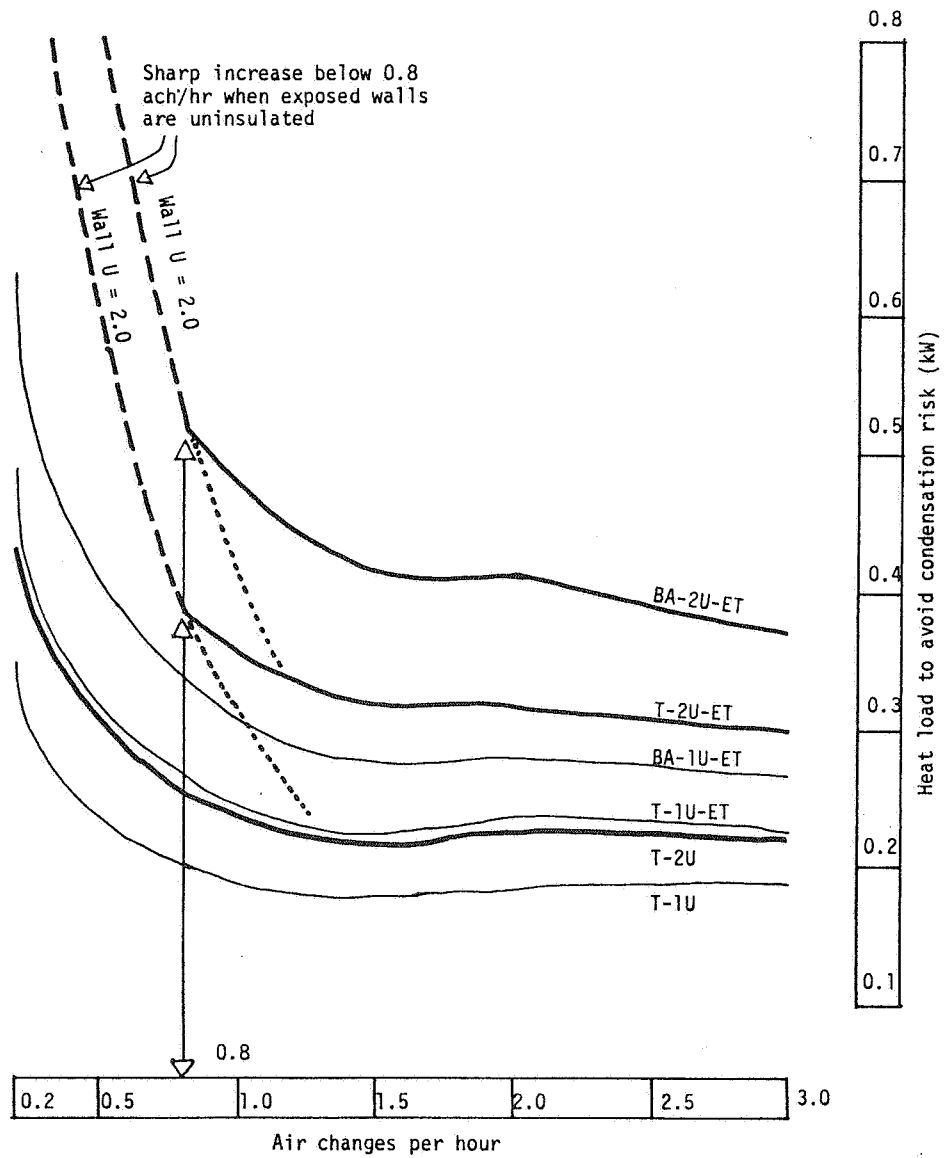


Figure 9 Heat loads necessary to avoid the risk of condensation on wall surfaces, including at corners - SINGLE BEDROOM

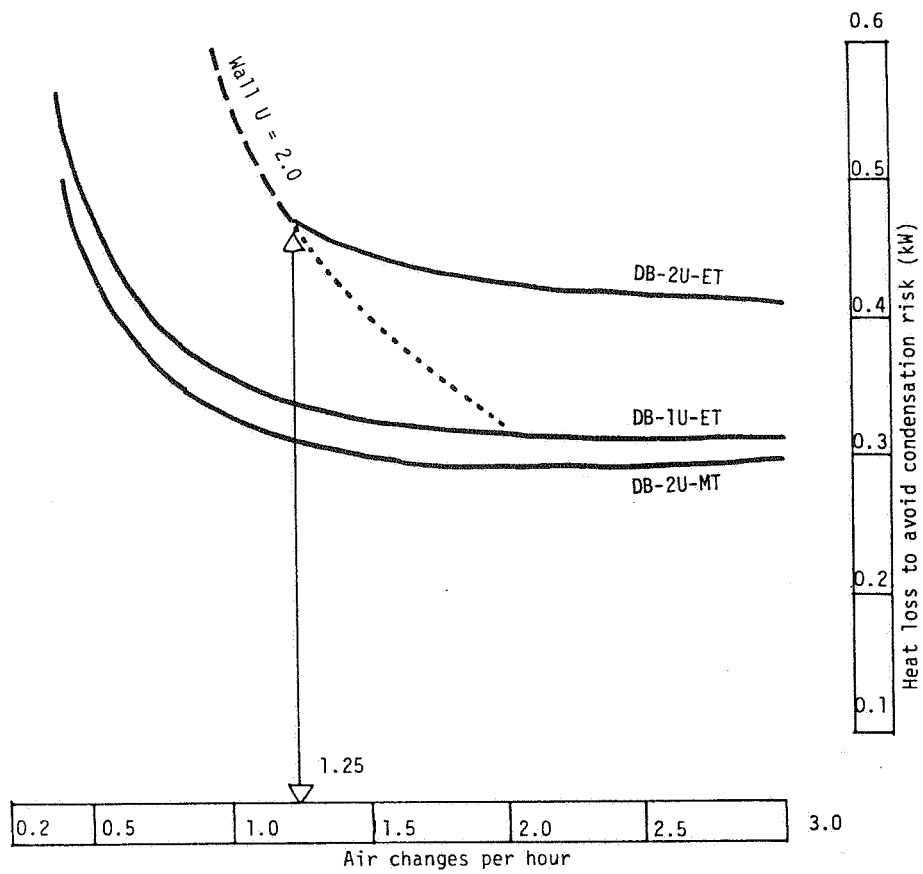


Figure 10 Heat loads necessary to avoid the risk of condensation on wall surfaces, including at corners - DOUBLE BEDROOM

condensation conditions, there is a greater danger of condensation occurring when warmer moist air is introduced to the room following a cold spell.

In the example shown the heating requirement is roughly the same for the uninsulated Bedroom in the mid-terrace position (DB-2U-MT) as for the insulated Bedroom at the end of the terrace (DB-IU-ET). These heat loads will differ depending on the configuration and window area of the room. Double Bedrooms can extend across the whole width of the frontage or be of a narrower configuration. The area of the room will vary considerably between that in a small house and that in a flat conversion. To determine appropriate heat loads, calculations would need to be carried out for a range of typical plan types.

6. CONCLUSIONS

Using accepted condensation prediction criteria, it can be shown that for a particular dwelling type, there is an optimum air change rate at which the heat input necessary to avoid condensation risk is at a minimum.

Large volume dwellings are able to be ventilated at a lower air change rate than smaller units with equivalent occupation and thermal characteristics. Similarly, the better a dwelling is insulated, the lower the optimum air change rate.

Preliminary studies of the build up of moisture in Bedrooms over night indicate that, as the ventilation rate is reduced, the time taken for the moisture content of the room air to reach equilibrium is increased. This could reduce the risk of condensation, provided the incoming ventilation air is introduced from the outside rather than from the remainder of the dwelling.

The optimum air change rates for Bedrooms are higher than those for the dwelling as a whole. But at these ventilation rates, the temperature achieved may be too low to be considered acceptable. It is necessary, therefore, to explore the situation at lower air change rates.

Under these circumstances, condensation may occur on the surface of uninsulated walls in the corners of rooms when the relative humidity of the room air is less than 70%. Calculations indicate that a wall U value of $1.0 \text{ W/m}^2\text{K}$ is adequate to ensure that the 70% RH criterion can be applied.

Measures to reduce ventilation rates in existing dwellings must be accompanied by the addition of thermal insulation and heating (particularly in Bedrooms) if condensation is to be avoided.

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 5

VENTILATION RATES IN RELATION TO EMISSION OF GASES AND VAPOURS
FROM BUILDING MATERIALS.

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1. SYNOPSIS

Mathematical models for formaldehyde concentrations of each of three normal rooms in a single family house is used to estimate ventilation rates needed to maintain the formaldehyde concentration below the Danish recommended indoor standard (0.15 mg/m^3). It appears that in an initial period after the house was finished a ventilation rate more than 10 times the recommended Scandinavian maximum value (0.5 air change per hour) was needed to keep the concentration below the indoor standard.

Neither an indoor standard nor an elaborated mathematical model exists for gases and vapours of solvent type in the indoor air. A rough estimate, however, does not eliminate the possibility that the same high ventilation rates may be needed for these compounds if complaints from occupants are to be avoided in the first period after a house is finished.

In consequence it is proposed that ventilation standards are set not only for the initial period but for the following period where emissions are in a more steady state. In the initial period the occupants should be advised to ventilate as much and as often as they feel necessary and new houses should for such purpose be constructed with extra permanent or temporary ventilation facilities.

2. INTRODUCTION

For several years both moderate and severe complaints about indoor climate have been registered among occupants of modern buildings. Malfunctioning ventilation and climate systems are in many cases found to be the main reason for the complaints. Some complaints are, however, reported from rooms where these and other technical reasons for complaints can be excluded.

The symptoms described by the occupants of these rooms are often similar to those resulting from exposure to low concentrations of irritating gases and vapours, and the symptoms are often found in new buildings built of - or older buildings renovated with modern types of construction materials with unknown effects on the indoor climate. If the complaints are related to exposure to low concentrations of gases and vapours originating from such materials their impact could increase by the general effort to decrease domestic energy consumption by reduction of ventilation rates and by the introduction into house building and house renovation of huge volumes of new types of construction and installation materials.

Excessive formaldehyde emission from e.g. particle board has been found to be the reason for some of these complaints (1). Formaldehyde concentrations are, however, in many cases so low that this compound cannot be the cause for complaints. Therefore measurements to survey the occurrence of other organic gases and vapours in indoor environments were performed. The occurrence of gases and vapours of the solvent type in indoor air was demonstrated by measurements of their concentrations in 7 new houses and 39 older dwellings (3,4). The emissions of the same compounds from 42 common building materials were later measured in our laboratory (5).

Based on our measurements mathematical models were developed for the concentrations of both formaldehyde originating from particle boards (6) and organic gases and vapours of solvent type from normal building materials (5). In this paper these models will be used to illustrate the relation between calculated equilibrium concentrations of gases and vapours in normal rooms and the ventilation rate of the room. Further the results will be discussed in relation to proposed minimum ventilation rates.

3. THE MATHEMATICAL MODELS

3.1 Formaldehyde

The mathematical model for the equilibrium concentration of formaldehyde emitted from Urea-Formaldehyde glued particle boards was first established in 1975 as a result of investigations of the performance of a set of 14 particle boards in a climate chamber (1). The general mathematical form was found to be:

$$C = \frac{(RT + S)(aH + b)}{1 + \frac{n \cdot c}{\alpha}}$$

where C (mg/m³) is the equilibrium concentration of formaldehyde in the air around a particle board, and T, H, n, and α are respectively air temperature (°C), air humidity (g H₂O/kg dry air), n ventilation rate (h⁻¹) and α amount of particle board per volume room air (m²/m³). The constants R, S, a, b and c describes the emission properties of the board under investigation.

The model was further developed for formaldehyde concentrations expected in actual rooms where different particle boards have been used (6). This model was tested in three rooms in a new single family house. The house was constructed according to normal Danish building tradition with urea formaldehyde glued particle boards coated with a formaldehyde absorbing paint. The emission of formaldehyde from all boards used for the three rooms was found according to a standard procedure (7) based on measure-

ments of the equilibrium concentration of formaldehyde around a 0.25 m² sample of the particle board under investigation in a stainless steel box (225 l). The box is ventilated 0.25 times per hour by fresh air (45% RH) and is kept at 23°C.

Each of the three rooms in the house was described with individual models for the air concentrations of formaldehyde and the calculated formaldehyde concentrations were found to agree within 15% with those measured in the rooms immediately after the house had been finished (6).

3.2 Organic Gases and Vapours

The mathematical model for the equilibrium concentration of organic gases and vapours is not so detailed as that for formaldehyde. The reason is that the influence of indoor climate parameters on the emission rates is unknown. In the establishing of the mathematical model for organic gases and vapours it was therefore assumed (5) that the emission rate is constant and independent of the indoor climate.

By this simple mathematical model the equilibrium concentration of any indoor air pollutant emitted from building materials used in a room or the total concentration of all such pollutants is estimated. According to the model the total concentration C mg/m³ in a room volume V m³ (ventilated n times per hour) can be calculated through

$$C = \frac{1}{V \cdot n} \sum_{i,j} E_{ij} \cdot A_i$$

where E_{ij} is the emission rate (mg h⁻¹m⁻²) of compound (j) from material (i) and A_i is the amount of material (i) used in the room (m²).

The validity of this model was investigated by calculation of expected total concentration of organic gases and vapours in three model rooms constructed from materials with known emission of 52 different organic compounds (5). The range of the calculated concentrations was from 1.6 to 23.5 mg/m³. Measurements in similar rooms in seven single family houses showed concentrations ranging from 0.48 to 18.7 mg/m³. In comparing these results it should be remembered that the model deals with equilibrium concentrations which very seldom occurs in homes etc., and that no corrections were made for differences in indoor climate and in room construction between investigated rooms in real houses and the model rooms used for the calculations.

4. THE INFLUENCE OF VENTILATION ON INDOOR AIR POLLUTION

The mathematical models described above for formaldehyde and for total concentrations of organic gases and vapours of solvent type have been used to illustrate the relation between ventilation rate and equilibrium concentrations of formaldehyde in the three actual rooms in the single family house described in (6), and the total concentration of organic gases and vapours in the three model rooms described in (5).

Formaldehyde: According to the model each room of the single family house was described with its own mathematical model. Each model had the same mathematical form

$$C = \frac{(R \cdot T - 0.764) \cdot (0.143 \cdot H + 0.048)}{1 + \frac{n \cdot 0.304}{\alpha}} \text{ mg/m}^3$$

where R is a constant representing the emission rate of formaldehyde in each room and α is the amount of particle board m^2/m^3 . The temperature ($^{\circ}\text{C}$), the humidity (g/kg) and the ventilation rate (h^{-1}) is designated T, H and n respectively. Values for these variables are given in table 1.

These models were used to calculate expected formaldehyde concentrations in the rooms under standard indoor climate concentrations 23.0°C . and 7.8 g water per kg dry air (45% RH).

Table 1: The mathematical model for equilibrium concentration of formaldehyde in three rooms. The general mathematical form is:

$$C = \frac{(RT - 0.764) (0.143 \cdot H + 0.048)}{1 + \frac{n \cdot 0.304}{\alpha}}$$

where C = Equilibrium concentration of formaldehyde in the room air mg/m^3 .

R = Emission constant.

T = Air temperature $^{\circ}\text{C}$.

H = Air humidity g/kg

α = m^2 particle board/ m^3 room volume.

n = Ventilation h^{-1} .

Room	TOC	H g/kg	n h^{-1}	m^2/m^3	R	C mg/m^3
I	23.4	5.7	1.4	2.30	0.0431	0.26
II	21.6	5.3	0.6	1.92	0.0494	0.27
III	23.0	5.3	2.3	2.33	0.0427	0.25

Fig. (1) shows the calculated relations between formaldehyde concentrations in the three rooms and the ventilation rate of the room. The dotted part of the figure indicates a region in which

the present model has been found invalid (1).

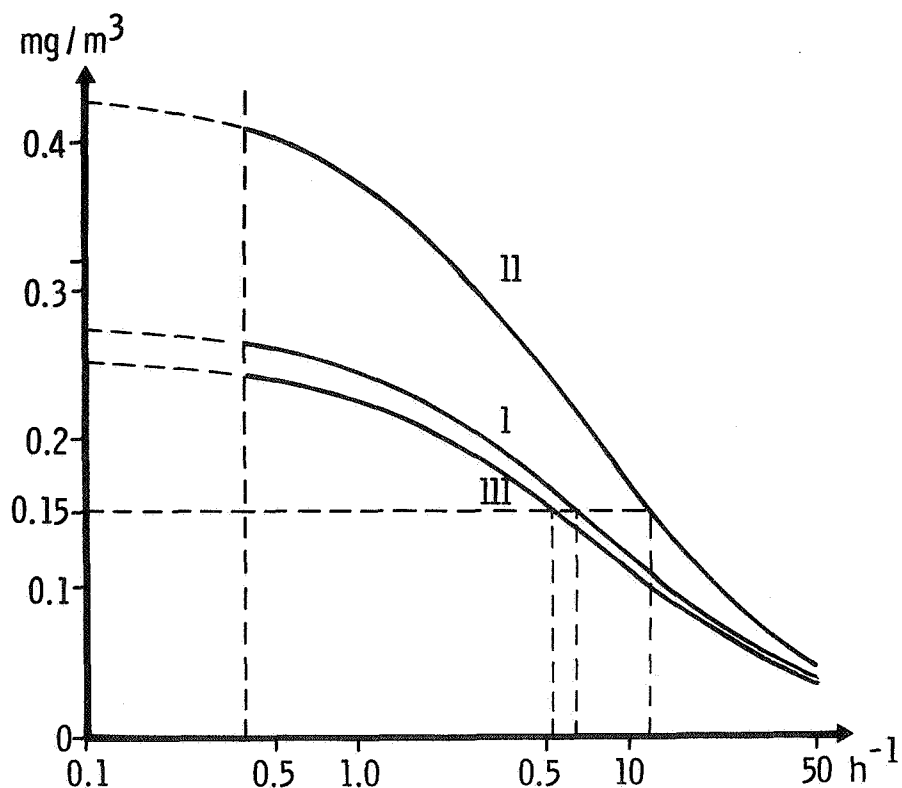


Fig. (1): The influence of ventilation rate (h^{-1}) on the equilibrium concentration of formaldehyde in three rooms containing about $2 \text{ m}^2/\text{m}^3$ particle boards per m^3 room volume.

Solvent gases and vapours: The influence of indoor climate parameters on the emission rate of organic gases from normal building materials is unknown. A detailed model as that described for formaldehyde has therefore not yet been established for other pollutants or building materials. If, however, as a first approximation the emission rate of any compound is assumed to be constant and independent of temperature, humidity and ventilation rate the equilibrium concentration of organic gases and vapours in the three model rooms described in (5) can be calculated. Fig. (2) shows for the three rooms the relation between ventilation rate and total concentration of 52 different compounds emitted from the building materials used for the rooms.

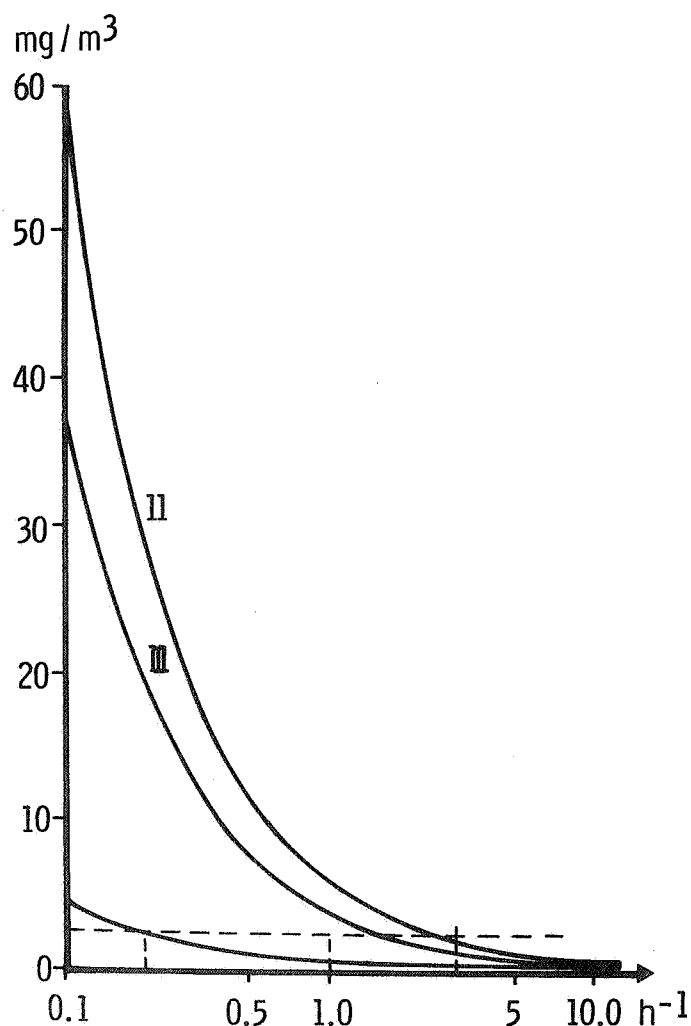


Fig. (2): The relation between total equilibrium concentration mg/m^3 of 52 organic compounds of solvent type in three model rooms.

5. DISCUSSION

For the first weeks after the finishing of a normal Danish house the model indicates that formaldehyde concentrations below the recommended Danish indoor air standard for formaldehyde (0.15 mg/m^3) is achieved only with ventilation rates higher than 6.4, 11.9 and 5.2 h^{-1} for room I, II and III respectively (see fig. 1). These figures are an order of magnitude higher than the proposed Scandinavian ventilation standard of 0.5 h^{-1} (8) for normal homes and they demonstrate further that efforts to decrease this initial formaldehyde concentration to below 0.15 mg/m^3 by increasing the ventilation are unrealistic.

Repeated measurements in the three rooms in the following 14 weeks showed a marked decrease in concentrations. This decrease occurred despite the fact that the rooms in the meantime were inhabited by occupants who through furniture etc. introduced an unknown amount of potential sources for formaldehyde in the rooms. After 14 weeks the measured formaldehyde concentrations in all rooms were below 0.15 mg/m^3 . This decrease in concentrations took place with unchanged ventilation as shown in table 1. A further decrease of the concentrations can be expected as the house grows older and in the long range a ventilation of 0.5 per hour may be sufficient to keep the concentration of formaldehyde below 0.15 mg/m^3 .

The discrepancy between recommended maximum formaldehyde concentration, proposed ventilation rate and the extended use of particle boards therefore seems to be restricted to a relatively short initial period after the house was finished.

For this initial period the model predicts the maximal acceptable amount of particle board in the rooms to be about 0.2 m^2 per m^3 room volume if the concentration is not to exceed 0.15 mg/m^3 at a ventilation rate of 0.5 per hour. This amount of particle board corresponds to that normally introduced by the occupants through furniture etc. Urea-Formaldehyde glued particle boards of the type investigated here therefore can not be used in homes at all if recommended norms for ventilation rates and formaldehyde concentrations must be maintained in the initial period of a new house. The model on the other hand indicates that ventilation rates up to 10 exchanges per hour are needed in this period if about $2 \text{ m}^2/\text{m}^3$ of normal particle boards treated with a formaldehyde stopping coating are used.

Indoor air standards only exist for formaldehyde. In a general recommendation of ASHRAE (2) maximal acceptable indoor air concentrations of any indoor pollutant is set to one tenth of its maximal allowable concentration in industrial environments. This recommendation is, however, not generally accepted.

The average MAC-value for those of the 52 compounds mentioned above for which such MAC-value exists is about 600 mg/m^3 . From fig. (2) follows that with normal occurring ventilation rates 1/10 of this value (60 mg/m^3) will not be exceeded in the 3 model rooms. The use of an average threshold value is, however, not toxicologically justified.

A more safe estimate of the low limit may be obtained by using the lowest MAC-value for the 52 compounds. This is 20 mg/m^3 and in two of the three model rooms the expected concentrations do exceed 1/10 of this value (2 mg/m^3) if the ventilation rate is below 0.5 air change per hour.

In evaluating the results it should be kept in mind that no adequate indoor standard exists for the 52 compounds nor exists an elaborated mathematical model.

The results, however, which are rough estimates do not eliminate the possibility that these compounds may reach such concentrations in the initial period that ventilation rates higher than 0.5 air-changes per hour are needed if complaints from occupants are to be avoided.

6. CONCLUSION

The investigations show that for formaldehyde and possibly for other indoor airpollutants like solvent vapours 0.5 airchanges per hour may not be sufficient to maintain airconcentrations at an acceptable level in an initial period after the completion of a house. In consequence it is proposed that ventilation standards are set not for this initial period but only for the period of normal use of the house where emissions have decreased and reached a more steady state. In the initial period occupants should be advised to ventilate as much and as often they feel necessary.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 6

VENTILATION AND INTERNAL AIR MOVEMENTS FOR SUMMER AND WINTER
CONDITIONS

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2. SUMMER AND WINTER VENTILATION

Opening lights in windows have served the purposes of ventilation for many years and it may be questioned why they cannot continue to do so. In this context it is important to distinguish between the requirements of summer time ventilation and the problems of minimum ventilation rates in winter. In non air conditioned buildings opening lights in windows serve fairly effectively to control the high rates of ventilation appropriate to summer conditions. Disturbance of papers and curtains can represent an annoyance but generally speaking the degree of controllability afforded by relatively large opening lights works well.

It does not work well in winter conditions and particularly so in circumstances where security demands that, while occupants are absent, windows should be secured. In winter time even a very narrow opening of a large opening light gives relatively high rates of ventilation and is likely to result in draughts being experienced because of the usual location of opening lights. Prolonged closure of the windows did not however represent a serious problem in the past. Habitable rooms were required to have either a flue or a special ventilator to fresh air. Quite apart from these provisions, the general level of airtightness of both the construction and the windows themselves tended to be low so that a significant rate of ventilation would take place even when the opening lights themselves were shut. Window manufacturers have, for some time, been concerned to improve the effectiveness of sealing of their windows and this, coupled with the abolition of the requirement for ventilation of habitable rooms and the powerful motivation in both new and existing buildings to reduce ventilation rates to save energy, results in potentially very low ventilation rates in the winter in many dwellings. It is therefore becoming apparent that opening windows, whilst meeting the requirements for summer ventilation do not provide a solution to the problem of winter ventilation. The situation is a particularly critical one since, to be effective, ventilation rates have to be controlled within narrow limits. From the point of view of energy conservation the ventilation rate in the heating season should be reduced to a minimum, but from the point of view of condensation risk it is vital that an adequate flow of air is maintained. Thus, although reducing ventilation is an extremely cost effective way of reducing heat loss, it is clearly very important that effective ways of control should be developed to prevent the disastrous consequences which can arise from over-enthusiastic but injudicious attempts at reducing ventilation.

3. CONTROLLED VENTILATION

Quite apart from the great differences in the living habits of occupants individual dwellings vary very greatly in their airtightness. Siting, microclimatic factors, orientation and fenestration all play significant parts in addition to the effects

of choice of materials and standards of workmanship. In a series of measurements of ventilation rates in dwellings built in the ten years prior to 1925, Warren¹ found variations between 0.45 and 1.25 air changes per hour with windows closed. The trend is clearly towards lower rates of natural ventilation (see section 3.2).

It is not possible to predict in advance what ventilation rates will be encountered in dwellings and it is difficult to do so, except by actual tests, in existing buildings. Unless a careful window opening regime is maintained, ventilation rates in the lower range are likely, giving rise to higher than normal moisture concentrations in the internal air and consequently to increase the risk of condensation.

One approach to this problem might be the careful study of materials and workmanship to determine which design features and standards of workmanship could be used to give desired standards by means of infiltration through the fabric. If possible this would certainly be a highly cost effective way of dealing with the problem. In practice, however, there seems no practical possibility that the necessary factors could be sufficiently closely specified or controlled and the only quick practicable method of testing, the pressurisation technique, will give a good measure of airtightness but there is no direct correlation between this and the mechanism of ventilation.

3.1 Full Mechanical Ventilation

A realistic approach to the problem, which has already been adopted in some countries, is to make the whole of the basic construction, including doors and windows, as airtight as possible and then to provide specific controllable features for winter ventilation.

Clearly the most predictable way to control low levels of ventilation is by mechanical means. In some countries where energy conservation is very important this method is being adopted. It is said that 95% of new Swedish dwellings are built to be as tightly sealed as possible and equipped with mechanical extract systems extracting through kitchens and bathrooms giving an overall ventilation rate of one half air change per hour. Although this may be a very effective solution in Sweden there are several reasons which prevent it being readily adopted as a solution in this country. Although temperatures in this country occasionally do drop to extremely low levels by any standard, such occasions are comparatively short lived and do not occur every year. A more consistent level of cold in Sweden means that window opening in kitchens and bathrooms is not a feasible ventilation measure in winter and the building regulations require the provision of quite expensive means of permanent natural ventilation. Thus there is a more acute situation recognised by occupants and the cost of the natural ventilation provisions can be set against a

large part of the cost of the mechanical ventilation system which requires much simpler exit provisions through the building fabric. To ensure that the controlled low rate of ventilation is maintained the background infiltration is minimised by very high standards of workmanship in sealing the building and also, if condensation with low rates of ventilation is to be avoided, in ensuring that no cold bridges will cause trouble. In addition the amount of moisture basically present in the atmosphere at very low temperatures is much less than that at higher temperatures. In terms of condensation risk, therefore, the low temperatures represent an advantage.

In the U.K. climate where typical winter temperatures hover just above freezing and the relative humidities are extremely high it is by no means certain that the ventilation rates used successfully in Sweden would be adequate. The additional expense of full mechanical ventilation in the U.K. would be considerable, designers could be less certain that they would function efficiently, the noise levels would give rise to complaint and, on the experience associated with mechanical systems installed to overcome aircraft noise it is very likely that the fans would not be adequately maintained or repaired when necessary. In this country one must, therefore, look to the possibilities of providing natural means of controllable ventilation, perhaps operating in conjunction with simple extract fans.

3.2 Natural Ventilation

The first prerequisite is to separate the winter ventilation function from the opening lights. A system of controllable ventilators separate from but perhaps included in the window opening is an obvious possibility. One factor which does appear to be quite clear is that there should be a manually operated adjustment, the position of which can be observed by the users so that repeatable settings can be made to meet differing conditions.

In a range of weather conditions the whole house ventilation rate with internal doors open was measured using the tracer gas decay method. Fig. 1 shows the results, plotted against wind speed. The internal-external temperature difference varied between 5°C and 9°C. Inevitably there is scatter but there is a clear indication that the use of the ventilator openings is likely to produce a higher rate of ventilation, especially at higher wind speeds. The superimposed lines are obtained by linear regression analyses of the results with closed and open vents respectively. The distance between the lines suggests that the ventilation rate may be expected to increase by 16 - 27% when all the ventilators are opened.

During the course of a broad investigation of ventilation rates and internal air movements in a range of dwellings results have emerged which indicate the effects of built in hit and miss slot ventilators and of extract fans. The houses so equipped are

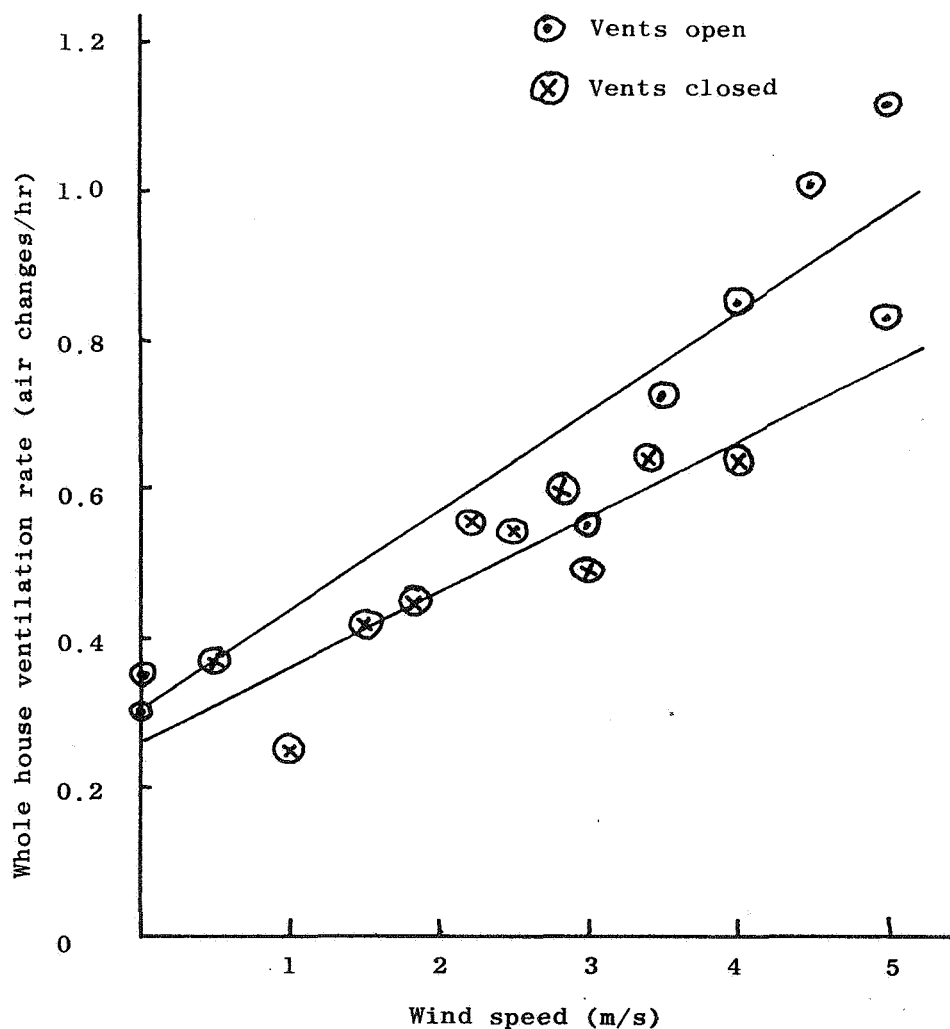


FIG. 1. Comparison of ventilation rates with ventilators open and closed.

newly constructed with double glazing and storm porches. Each window frame incorporates a slot ventilator most of which may be closed or opened to produce an open area of 7000mm^2 . The ventilator in the kitchen is permanently open. The dwelling tested was end-terraced with a volume of 200 m^3 .

The ventilation rates were measured by a tracer gas decay method which employs multiple tracer gases so that the internal air exchanges between major zones can be assessed. In addition, a pressurisation test was performed at 50 Pa .

The results show the effectiveness of the simple slots employed in one particular dwelling. It is clear that they do provide an increased rate of ventilation without the disadvantage of the relatively large opening lights.

Many devices of this type are on the market at present, some of which have a flap arrangement intended to give fixed ventilation rates irrespective of pressure differences across the unit. There is an important need to evaluate the performance of these units, both when installed and over a period of time.

A possible method of testing such devices when installed is the use of the pressurisation technique.

During the investigations described, the average of pressurisation and depressurisation tests at 50 Pa produced a throughput of $0.675 \text{ m}^3/\text{s}$ with the slot ventilators closed and $0.785 \text{ m}^3/\text{s}$ with the ventilators open - an increase of 16%. Some measure of correlation between the pressurisation results and the ventilation measurements is evident; both techniques giving a similar assessment of the effect of the ventilation openings.

It should be noted from Fig. 1 that low levels of ventilation rate are to be found in still atmospheric conditions with values as low as 0.25 - 0.30 air changes per hour.

3.3 Simple Extract Fans

The houses tested were equipped with simple extract fans discharging through the wall near to the cooking appliance. In a series of decay tests, the whole house ventilation rate varied between 1.1 and 1.9 air changes per hour with the fan on and the window vents closed. The combined effect of the fan and the open window vents was to produce higher ventilation rates of between 1.6 and 3.0 air changes per hour. An important effect of the fan is to produce a different flow pattern throughout the house. Fig. 2 demonstrates the changes of direction and flow rates. Strong evidence is also indicating that flow into roof spaces and cavities is considerably reduced when an extract fan is running in the house. Thus the primary effect is to remove moist air from the zone in which water vapour is generated but the secondary effect is to reduce the flow from the ground floor to the first floor thus minimising convection of warm moist air into remote rooms with low occupancy and a likely minimum of heating.

EXTRACT FAN

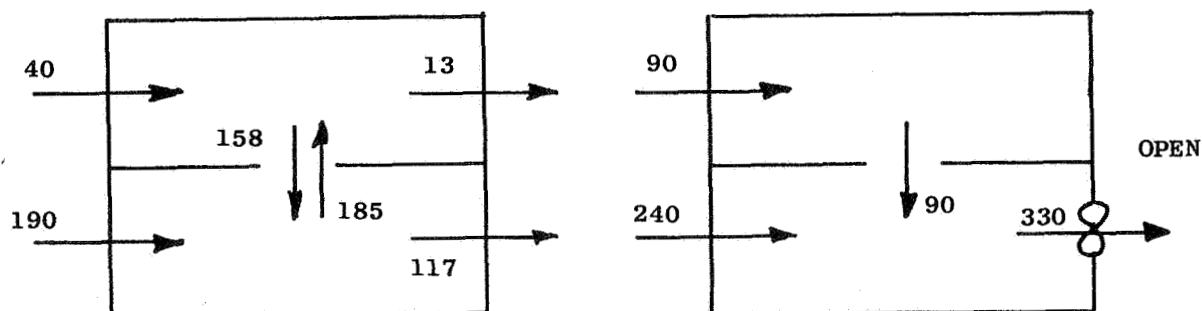


FIG. 2. Whole house air flows with open ventilators and the effect of kitchen extract fan. (Flows in m^3/hr).

4. CONCLUSIONS

1. There is a clear and conflicting distinction between the requirements for summer ventilation and for winter ventilation.
2. Built in controllable ventilation slots can provide a level of ventilation compatible with winter requirements.
3. The appropriate sizing of ventilators (and hence the extent of the increased ventilation which they provide) is not known; further research which correlates the winter ventilation requirement with ventilation size is necessary.
4. Kitchen extract fans are highly effective means of keeping moisture generated in cooking out of the main part of the house and its roof space. Reversals of stair well flows have been clearly observed.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
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PAPER 7

A COMPARISON OF ALTERNATE VENTILATION STRATEGIES

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SYNOPSIS

In order to decrease the energy costs associated with infiltration, many new residential structures are being built to more stringent air-tightness standards; additionally, many existing structures are being tightened. As structures are made tighter, the total amount of ventilation available for the removal of pollutants generated within the structure is reduced, thus causing a concern that indoor air quality may be suffering. Included among the many solutions to the problem of unacceptable air quality are cleaning the air within the structure (filtration) or replacing the contaminated air with fresh outdoor air (ventilation). Because filtration may not be practical for residential buildings, most of the effort in controlling indoor contaminants has been concentrated on finding the most effective means of increasing ventilation while incurring the smallest energy load. This report examines several ventilation strategies for both impact on the total ventilation and effect on the energy balance of the structure. The strategies we examine include natural systems such as ventilation stacks as well as mechanical systems such as air-to-air heat exchangers or exhaust fans with and without heat pumps.

1. INTRODUCTION

As architects, builders, and home-owners become more energy conscious, increasing attention is paid to the energy burden imposed by ventilation and air infiltration: the goal of decreasing total energy consumption has led to tighter houses and decreased air infiltration. However, as air infiltration is decreased in the name of energy conservation, the possibility of unacceptable indoor air quality grows. If indoor air quality is to be preserved at low infiltration rates, mechanical ventilation may have to be added to increase the total ventilation rate to an acceptable level at a minimum cost.

A comparison of mechanical ventilation systems requires calculating the effects such systems have on the total ventilation and ventilation (energy) load of the structure. In order to make this calculation we will use the single-zone infiltration model¹ developed at Lawrence Berkeley Laboratory, which uses air tightness, structural parameters, and weather to calculate air infiltration; mechanical ventilation can then be combined with the natural ventilation to yield total ventilation. From the natural ventilation, the mechanical ventilation, and the properties of the mechanical system, the total ventilation load can be calculated.

The purpose of this paper is not to make a detailed calculation for a particular building at a particular site,² but rather to make a simple calculation for a typical house (defined in the

natural ventilation section below). That calculation can then be used as a basis to compare the infiltration and load effects of different systems under different conditions (of air tightness, inside/outside temperature difference, and wind speed). Because these calculations are intended to be representative but non-specific, no economic comparison will be made among systems.

We examine three alternative mechanical ventilation strategies for increasing total ventilation. We compare a balanced (mechanical) air-to-air system, an unbalanced (mechanical) exhaust system, and a ventilation stack with completely natural ventilation. The exhaust system will be subdivided into three systems that treat heat loss through the mechanical system differently.

2. NATURAL VENTILATION

Natural ventilation (or air infiltration) is caused by the interaction of the building envelope with weather-induced pressures. The expression, as derived from detailed calculation,³ depends on inside/outside temperature difference and wind speed.

$$Q_0 = L_o \sqrt{f_s^2 \Delta T + f_w^2 v^2} \quad (1)$$

where:

Q_0 is the ventilation from air infiltration [m^3/s],
 L_o is the total leakage area [m^2],
 ΔT is the inside/outside temperature difference [K],
 v is the wind speed [m/s],
 f_s is the stack parameter [m/s $K^{1/2}$], and
 f_w is the wind parameter.

The stack and wind parameters depend on the leakage distribution, building height, and shielding and terrain factors. To calculate the infiltration, Q_0 , we use the concept of a reference house;⁴ the reference house has typical values for the structural parameters that go into the stack and wind parameters. Specifically, the house is a single-story, typically shielded house in average terrain; half of the leakage is distributed evenly in the walls and the other half to the floor and ceiling equally. With these assumptions the stack and wind parameters are*:

$$f_s = .144 [m/s-K^{1/2}] \quad (2.1)$$

* Conforming to the conventions of the model $H = 2.5m$, shielding class = terrain class = III, $R = 0.5$, $X = 0$.

$$f_w = .123 \quad (2.2)$$

We shall use these values throughout the remainder of the report.

Even though we have used a simplified reference house, there are still three independent variables (leakage area, temperature difference, and wind speed) that make it quite difficult to represent the results. In Fig. 1a, we have plotted the natural ventilation (i.e., ventilation due to air infiltration) versus inside-outside temperature difference for three envelope leakage areas. The leakage areas were chosen to represent values for a tight house (200 cm^2), a loose house (1000 cm^2), and one in between (500 cm^2). The curves have been calculated assuming a meteorological wind speed of 4 m/s, an average wind speed typical of many areas in North America and Western Europe. The pair of arrows that crosses each curve shows the effect of wind speed on ventilation; the base of each arrow was computed for zero wind speed, and the tip was calculated for a wind speed of 8 m/s -- twice the wind speed used in the curves.

Assuming that all infiltrating air mixes with interior air, the infiltration load can be calculated easily from the infiltration:

$$E_0 = \rho C_p \Delta T Q_0 \quad (3)$$

where:

E_0 is the energy load due to natural ventilation [W],
 ρ is the density of air [1.2 kg/m^3], and
 C_p is the heat capacity of air [1023 J/kg-K].

Note that here and in the sections to follow we ignore the effect of the latent load on energy consumption and assume that the ventilation efficiency of all forms (i.e., natural and mechanical) is unity.

Fig. 1b, using the same conventions as Fig. 1a, shows the ventilation load due to natural infiltration as a function of temperature difference for the three leakage areas. Because the infiltration in Fig. 1a is a monotonically increasing function of temperature, the loads increase faster than would be shown by a straight line.

3. VENTILATION REQUIREMENTS

The purpose of mechanical ventilation is to assure that indoor pollutants are adequately diluted by fresh air; the exact amount of fresh air, however, is not generally agreed upon. Although the necessary ventilation rate will depend strongly upon the types and source strengths of the pollutants, for the purpose of this report

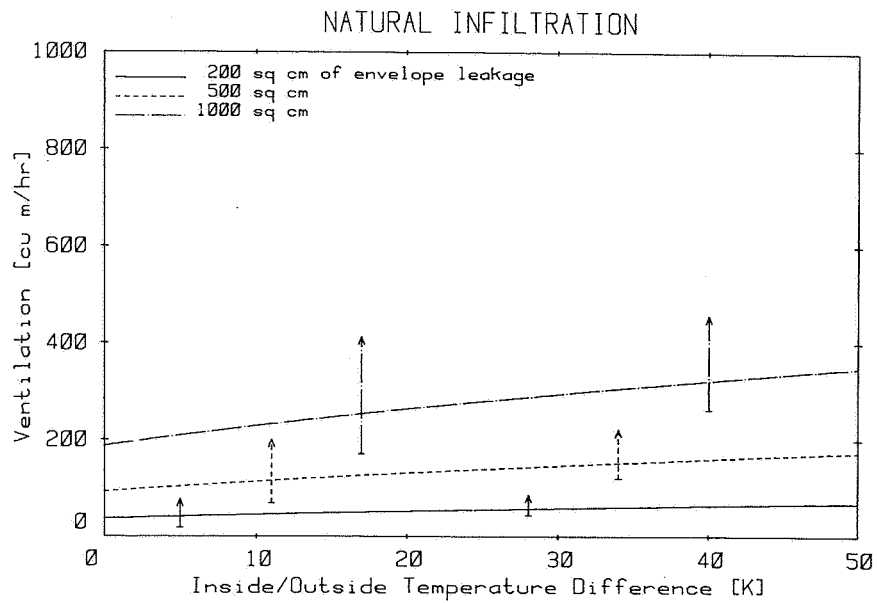


Fig. 1a. Ventilation due to air infiltration.

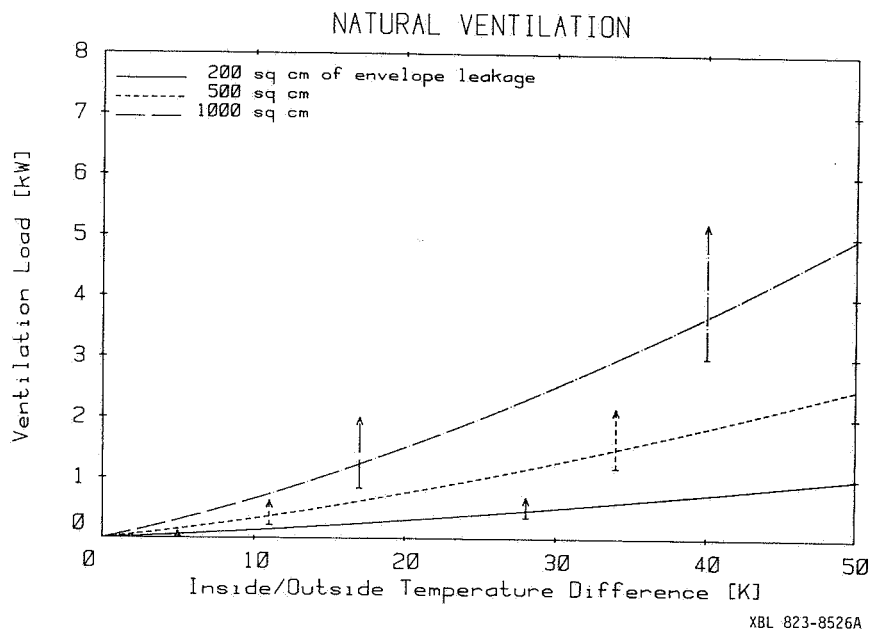


Fig. 1b. Ventilation load due to air infiltration.

we assume that $200 \text{ m}^3/\text{hr}$ is the amount of total ventilation necessary for adequate indoor air quality. This could correspond, for example, to one air change per hour in a 200 m^3 house or one-half an air change in a 400 m^3 house. Referring again to Fig. 1a, we see that under average weather conditions only the leakiest envelope meets our ventilation requirement with natural ventilation. The tighter two envelopes (and the leaky envelope under very calm conditions) require additional ventilation to meet the requirement. We must, therefore, consider mechanical ventilation systems.

4. BALANCED AIR-TO-AIR SYSTEMS

Mechanical ventilation systems employing air-to-air heat exchangers have recently received consideration as a possible systems for the control of indoor pollutant levels in residences.² In these systems there are two airstreams, one supply and one exhaust, which come into thermal contact but do not mix. Heat is transferred from one airstream to the other, reducing the increased load from additional ventilation. Because the system is assumed to be balanced (i.e., exhaust flow and supply flow are equal), the ideal heat exchanger has no effect on the internal (house) pressure. The system, however, does affect air infiltration because it contributes to the leakage area of the structure.

We can calculate the total ventilation from air infiltration, the fan flow through the exchanger, and the extra leakage area:

$$Q_1 = Q_{f1} + \left(1 + \frac{L_1}{L_0}\right) Q_0 \quad (4)$$

where:

- Q_1 is the total ventilation for the air-to-air system [m^3/s],
- Q_{f1} is the air flow from each fan [m^3/s], and
- L_1 is the additional leakage area from the exchanger [m^2].

Note that the presence of a non-zero L_1 term will effectively "unbalance" the system. This simplification ignores any non-linear interactions that may couple fan performance to external pressure. For our calculations we have sized the system as follows:

$$Q_{f1} = 150 \frac{\text{m}^3}{\text{hr}} \quad (5.1)$$

$$L_1 = 50 \text{ cm}^2 \quad (5.2)$$

The assignment of additional leakage area was based on a restricted set of calculations and, in any specific situation, could vary by a factor of three.

Figure 2a is a plot of the ventilation for this system using the same weather and leakage values as in Fig. 1a and 1b. For all temperatures and leakage areas the total ventilation rate meets the target of $200 \text{ m}^3/\text{hr}$, and may provide as much as twice that for the leaky envelope or extreme weather.

The energy load due to the combined effect of the air-to-air system and natural infiltration can be calculated from the system parameters:

$$E_1 = \rho C_p \Delta T (Q_1 - \epsilon_1 Q_{f1}) \quad (6)$$

where:

E_1 is the total ventilation load [W], and
 ϵ_1 is the effective heat-recovery efficiency.

The effective heat-recovery efficiency is the fraction of the sensible heat ($\rho C_p \Delta T Q_{f1}$) that is recovered by the heat exchanger. We have assumed an efficiency of 70%.

$$\epsilon_1 = 0.70 \quad (7)$$

When outdoor temperatures are substantially below freezing, the performance of the heat exchanger may be degraded by ice forming in the exhaust stream; freezing temperatures necessitate periodic defrosting of the heat exchanger. Although the overall efficiency of this system will be reduced in very cold climates, we have not specifically included this in our analysis.

The total ventilation load will be higher with an air-to-air system than without one. Figure 2b shows the increase in ventilation load due to the heat exchanger*. Because this system is balanced, there is no effect on the internal pressure of the envelope, and the increase in ventilation load is independent of leakage area -- hence there is one curve instead of three. Furthermore, the size of the wind arrows indicates the magnitude of the effect that additional leakage area has on the load. Because the wind arrows are quite small, our assumptions about the size and interaction of

* Conventionally, the total ventilation load is compared between two systems which have the same total ventilation (i.e., a natural system and one with a heat exchanger and a much smaller amount of infiltration), implying a different envelope leakage for the two cases. We are comparing a structure without a heat exchanger to exactly the same structure with a heat exchanger.

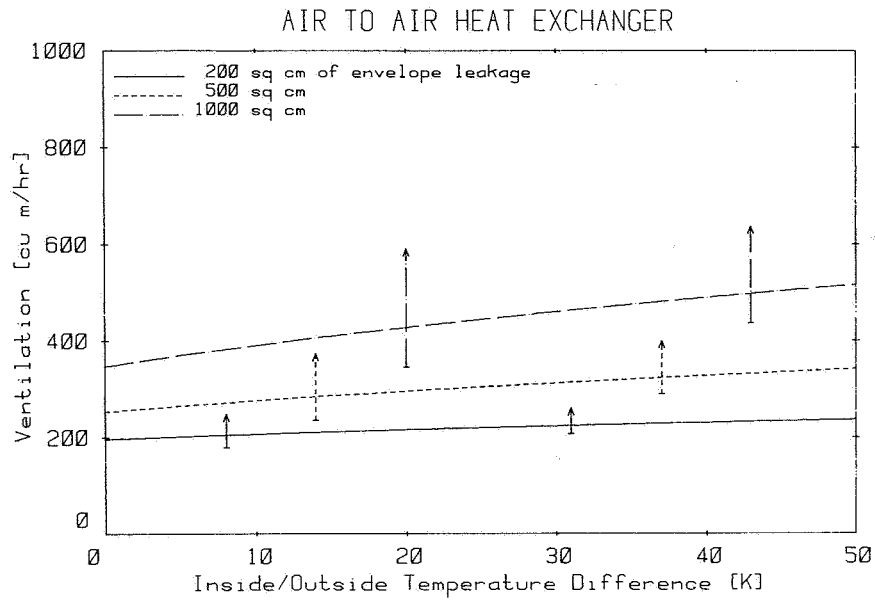


Fig. 2a. Ventilation due to a balanced heat exchanger.

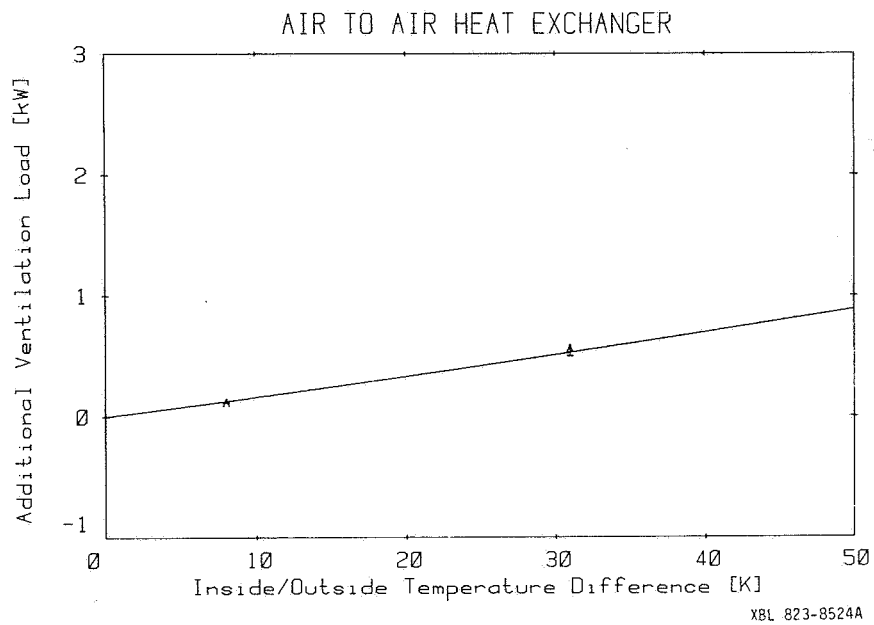


Fig. 2b. Increased load due to a balanced heat exchanger.

the additional leakage area are not critical.

The above discussion assumes that the weather-induced pressures on the heat exchanger operate in the same direction. This would be the case if the supply and exhaust ducts were near each other as, for example, in a window unit. If the weather-induced pressures act in opposite directions (e.g., a high supply and a low exhaust on opposite sides of the building), the amount of heat recovered through the heat exchanger will be increased and the total load mitigated:

$$E'_1 = \rho C_p \Delta T \left[Q_1 - \epsilon_1 \left(Q_{f1} + \frac{L_1}{L_o} Q_o \right) \right] \quad (8)$$

We have not, however, used this expression in our calculations.

5. EXHAUST SYSTEMS

In a balanced heat exchanger, supply and exhaust flows are matched and there is no effect on internal pressure. In an exhaust system there is no (mechanically provided) supply air; rather all air exhausted from the mechanical system must enter through the building envelope. Thus, the internal pressure will be significantly reduced as will the naturally driven part of the ventilation; the infiltration through the envelope will be increased and the exfiltration will be decreased. The overall effect is an increase in total infiltration, with a smaller absolute quantity of air crossing through the envelope. We can calculate the total infiltration rate using the rule of superposition in quadrature:

$$Q_3 = \sqrt{Q_o^2 + Q_{f3}^2} \quad (9)$$

where:

- Q_3 is the total ventilation for the exhaust system [m^3/s], and
- Q_{f3} is the flow through the exhaust fan [m^3/s].

For our exhaust systems we have used a flow equal to our target infiltration.

$$Q_{f3} = 200 \frac{m^3}{hr} \quad (10)$$

Exhaust systems may be especially susceptible to incomplete mixing and lower ventilation efficiencies; therefore, the value of Q_{f3} should be taken as the effective flow (i.e., the product of the ventilation efficiency and the actual flow rate) for calculating both infiltration and load.

Because exhaust systems affect the internal pressure, the total ventilation will be greater than the exhaust ventilation only when the natural air infiltration is large. Figure 3a shows the total ventilation for all exhaust systems.

5.1 No Heat Recuperation

We have elected to treat the heat loss of the exhaust systems in three different ways, corresponding to three types of exhaust ventilation systems. The first of these systems is the simplest; that is, a simple system with no heat recovery whatsoever. In this case the total ventilation load is derived from the total ventilation.

$$E_{3a} = \rho C_p \Delta T Q_3 \quad (11)$$

where:

E_{3a} is the total ventilation load without recovery [W].

Figure 3b shows the additional ventilation load imposed by an exhaust system without heat recuperation. Note that the arrows indicating wind dependence point down instead of up. This indicates that the additional load imposed by the exhaust system at high wind speed is smaller than the additional load imposed by the same system at lower wind speeds. Similarly, the curves for the leaky envelope are below those for the tight envelope. This behavior is a direct consequence of the fact that exhaust systems affect the internal pressure, and is common to all the exhaust systems.

5.2 Heat Recovery

Another possible exhaust system is one in which there is some heat recovery in the exhaust stream (e.g., a regenerative heat exchanger). Systems of this type would have a single stream of air that passes through a heat storage medium in alternating directions, first storing heat in the medium during the exhaust phase and recovering it during the supply phase. It may even be possible to use a building envelope as the heat-exchange medium.⁵ In any case the total ventilation load will be reduced from the value calculated above.

$$E_{3b} = \rho C_p \Delta T (Q_3 - \epsilon_3 Q_{f3}) \quad (12)$$

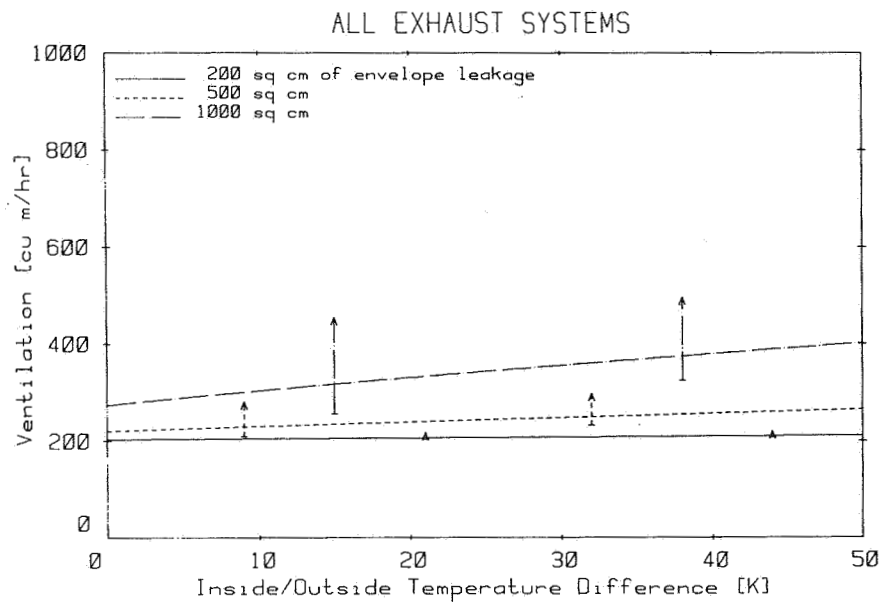


Fig. 3a. Ventilation from exhaust systems.

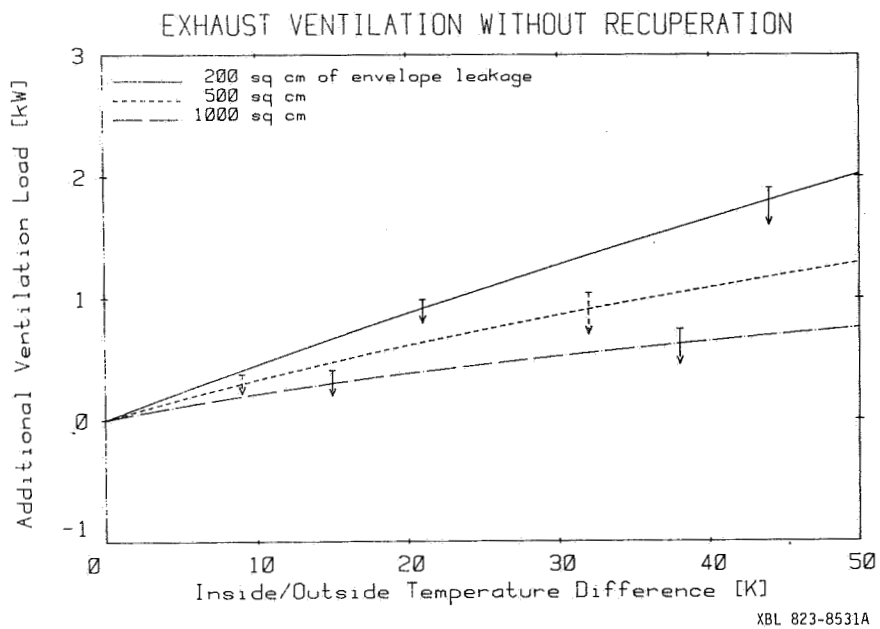


Fig. 3b. Additional ventilation load from exhaust systems--no recuperation.

where:

ϵ_3 is the effective efficiency of the heat recovery.

Because it is either storing heat (as its temperature increases) or releasing heat (as its temperature decreases), the overall efficiency of such a regenerative heat exchanger cannot be as high as for the air-to-air system in which there is a constant transfer of heat from one stream to the other. Accordingly, we have chosen a reduced value for the effective heat-transfer efficiency of this system.

$$\epsilon_3 = 0.30 \quad (13)$$

Figure 3c shows the additional load induced by an exhaust system with heat recovery. This figure demonstrates the same sort of upside-down behavior seen in the previous plot, but it is interesting to note that some parts of these curves extend below the origin; indicating that in some cases (extreme weather/leaky envelopes) an exhaust system with heat recovery can decrease the total ventilation load.

5.3 Heat Pump

Heat recovery by a heat-exchanger is limited by the difference in temperature across the heat-exchange medium; an alternate method of heat recovery is to use a heat pump to pump the heat out of the exhaust stream. In some existing systems such heat pumps are used to preheat the domestic hot water, but we will assume that the heat is pumped in the conditioned air space. Because the air in the exhaust stream will always be about the same temperature (internal), the effect of the heat pump will be to shift the ventilation load down by a fixed amount.

$$E_{3c} = E_{3a} - \rho C_p \Delta T_3 Q_{f3} \quad (14)$$

where:

ΔT_3 is the temperature drop across the exhaust stream [K].

The choice of temperature difference is essentially arbitrary and will probably depend on the systems available at the time. We have, therefore, chosen one which we believe is representative of what can be expected in consideration of latent heat and freezing issues.

$$\Delta T_3 = 20 \text{ K} \quad (15)$$

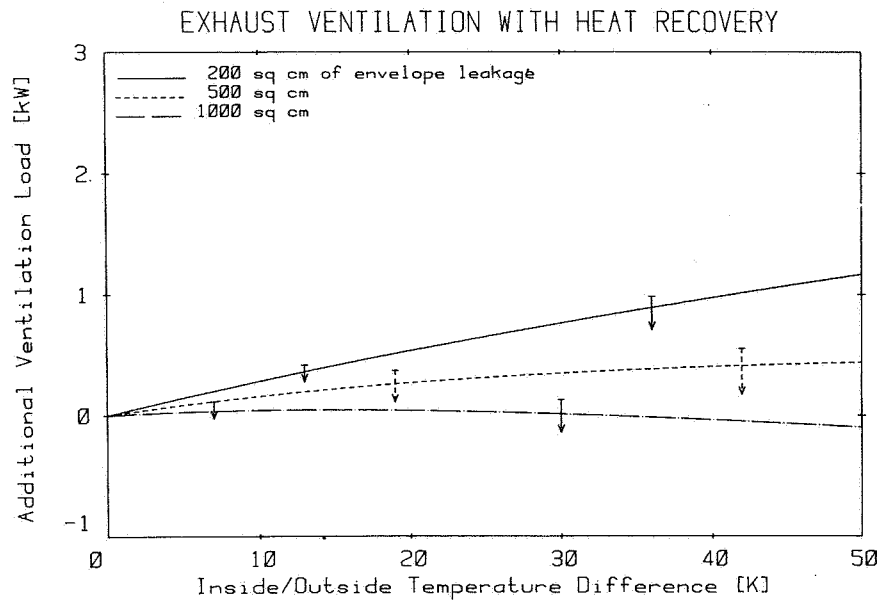


Fig. 3c. Additional ventilation load from exhaust systems--with regeneration.

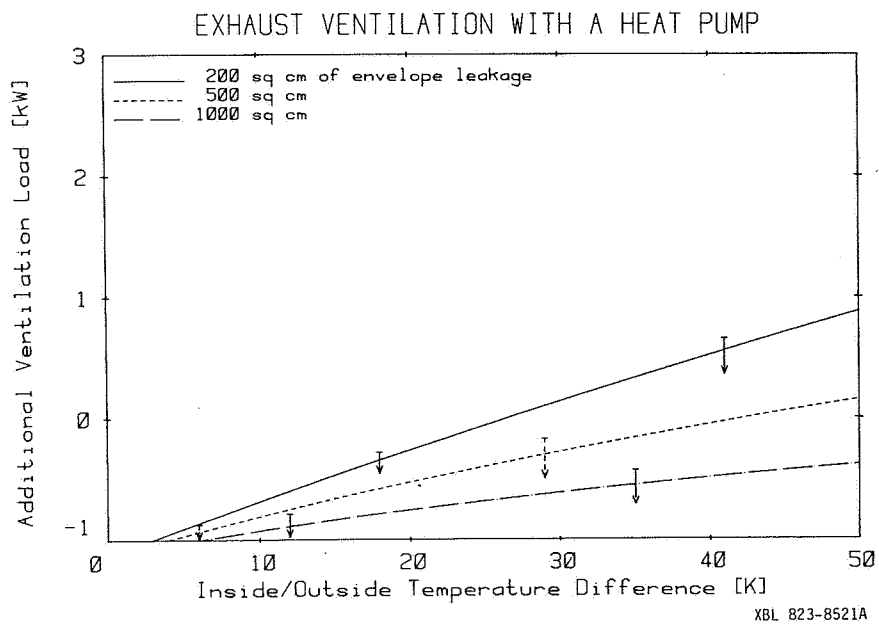


Fig. 3d. Additional ventilation load from exhaust systems--with heat pump.

Figure 3d shows the additional load induced by an exhaust system with heat-pump recuperation, which is the plot for exhaust without recuperation shifted down by 1150 watts. For most climatic conditions this system reduces the total ventilation load; and for very mild conditions the total ventilation load may become negative.

6. VENTILATION STACK

Once very simple approach to increasing total ventilation, which has been used in large structures but rarely on individual residences, is that of a ventilation stack. A ventilation stack is a chimney that projects into the free air above the structure and increases ventilation by its own stack and wind effect.⁶

$$Q_4 = \sqrt{Q_0^2 + Q_{f4}^2} \quad (16)$$

where:

Q_4 is the ventilation from a ventilation stack [m^3/s],
and

Q_{f4} is the flow induced by the ventilation stack [m^3/s].

The flow induced by the ventilation stack has an form analogous to that of air infiltration:

$$Q_{f4} = L_4 \sqrt{f_{s4}^2 \Delta T + f_{w4}^2 v^2} \quad (17)$$

where:

L_4 is the leakage area of the stack [m^2],

f_{s4} is the stack parameter of the ventilator [$m/s-\sqrt{K}$],
and

f_{w4} is the wind parameter of the stack.

If we assume the height of the stack to be six meters and the pressure coefficient at the top of the stack to be 0.95, we can calculate the rest of the parameters:

$$L_4 = 200 \text{ cm}^2 \quad (18.1)$$

$$f_{s4} = 0.45 \text{ m/s-K}^{1/2} \quad (18.2)$$

$$f_{w4} = 0.8 \quad (18.3)$$

Figure 4a shows the total ventilation with a stack ventilator of the above properties. For mild conditions and a tighter envelope the total ventilation is below our target level. Because this ventilation system is completely passive, there will be no ventilation if there is no wind or temperature difference. However, for most weather conditions of interest there will be an acceptable amount of ventilation from a stack ventilator. Because there is no heat recovery, the total ventilation load can be simply calculated:

$$E_4 = \rho C_p \Delta T Q_4 \quad (19)$$

where:

E_4 is the total ventilation load for a stack ventilator [W].

The additional load, as shown in Fig. 4b, shows some new features that previous plots did not. Like the exhaust plots, the curves for the tighter envelopes show larger increased ventilation loads, but like the balanced systems the arrows, which indicate the effect of increased wind speed, point upwards. Furthermore, some of the arrows do not cross their corresponding curves, indicating that the additional ventilation load may not be a monotonic function of wind speed. This occurs because the air infiltration through both the envelope and the ventilation stack are influenced by wind and temperature difference, but the relative weightings are much different.

7. COMPARISON

So far we have shown the effect that each system would have on ventilation and ventilation load for different leakage areas. We must now compare the different systems with each other. Although an economic criterion will ultimately be used to make a decision on which system to use, a comparison of effects on total ventilation and ventilation load is instructive.

Figure 5a compares the effects on total ventilation that these systems would have on a structure having 200 cm^2 of leakage area. The natural ventilation alone is far below the target of $200 \text{ m}^3/\text{hr}$ we are using, and would quite likely cause problems with indoor air quality. The other three systems yield adequate ventilation, with the possible exception of the stack ventilator (which does not meet the target for mild conditions). Because the stack ventilator is completely dependent on natural driving conditions, it is not surprising that it is the most weather-dependent of the mechanical systems. Conversely, because the exhaust systems have a pronounced effect on internal pressure, they are the least weather-dependent; the balanced air-to-air system is between these

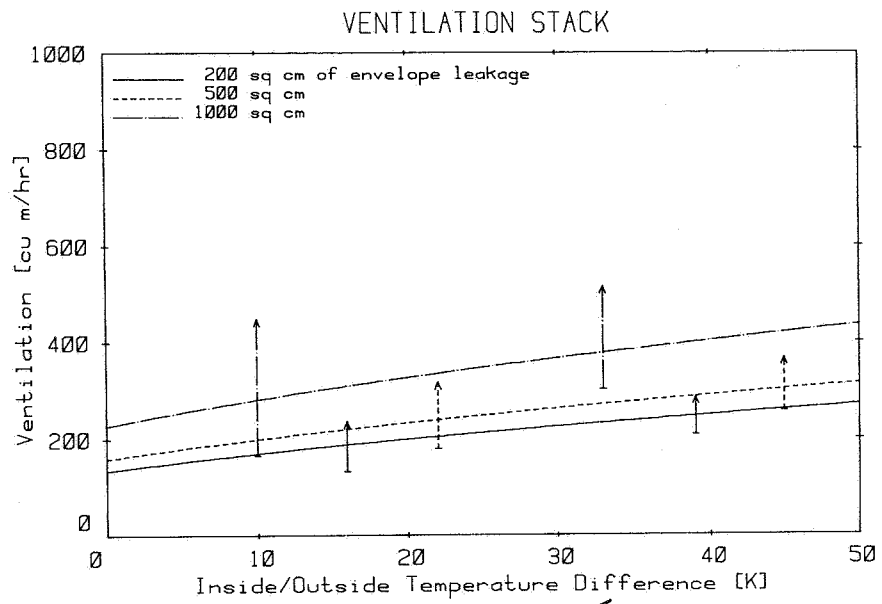


Fig. 4a. Ventilation from a ventilation stack.

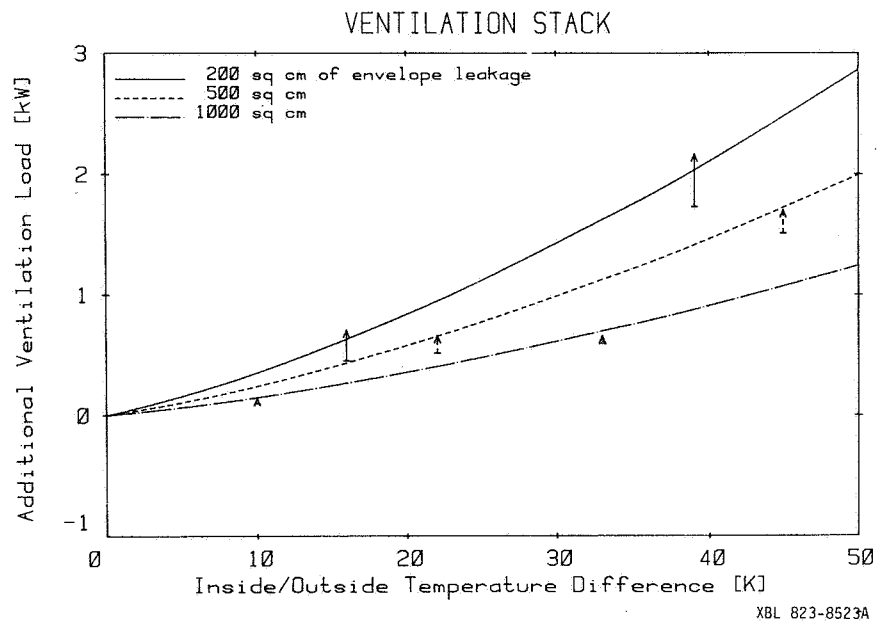


Fig. 4b. Additional load from a ventilation stack.

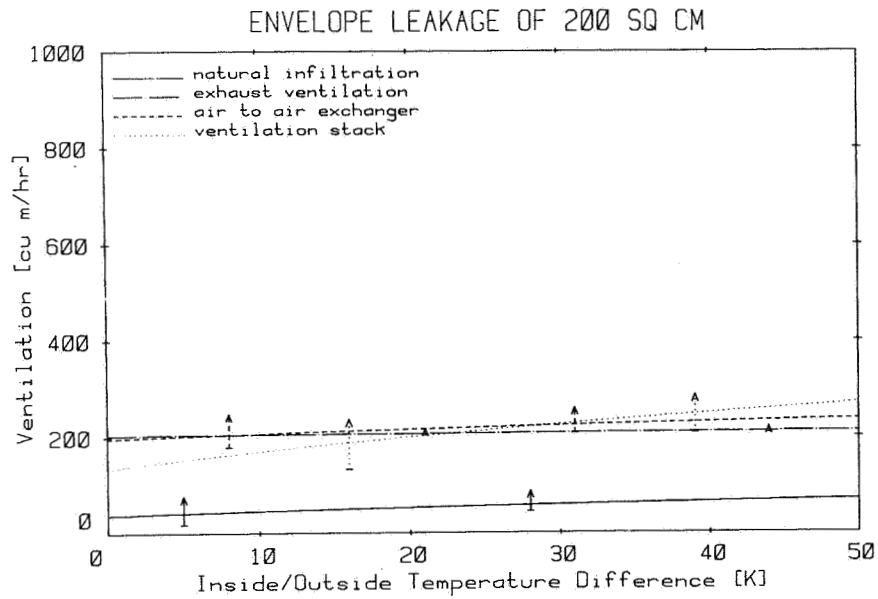


Fig. 5a. Ventilation for envelope leakage area of 200 cm^2 .

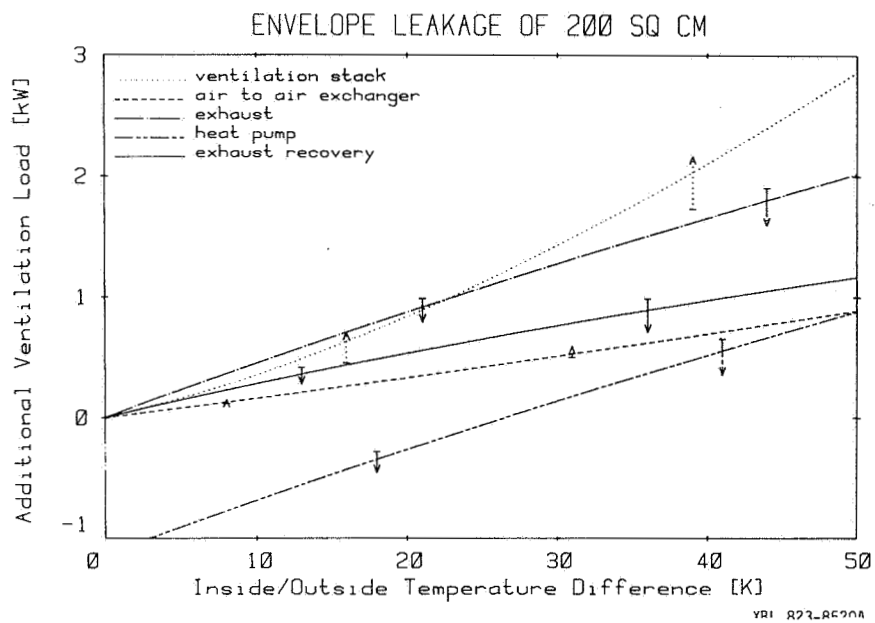


Fig. 5b. Additional ventilation load for envelope leakage area of 200 cm^2 .

two extremes.

Figure 5b again compares the effectiveness of these systems on a structure having 200 cm² of leakage area, but compares additional load rather than total ventilation. Whereas in Fig. 5a the results for all systems were bunched rather closely, the differences among them are quite noticeable in Fig. 5b. The order of decreasing load is approximately: ventilation stack, exhaust without recuperation, exhaust with heat exchange, air-to-air system, and exhaust with a heat pump. Unsurprisingly, this is also the probable order of increasing cost. Because the stack ventilator is only one of the systems to show positive curvature with temperature, it will perform relatively poorly under extreme temperature differences, making it unattractive for cold climates. The stack ventilator is the most sensitive to temperature differences; the air-to-air system is the least. Under moderate weather conditions the heat-pump system is a net producer of energy; but under more extreme conditions the net load of the heat-pump system approaches that of the balanced air-to-air system.

Because envelope leakage has an important effect on the performance of each system, we will compare the systems again -- this time at 500 cm² of leakage area. Figure 6a shows total ventilation for all the systems under these conditions. Compared to Fig. 5a, all the curves are shifted upwards somewhat, especially the one for natural ventilation, although it is still inadequate when used alone. While the exhaust systems and ventilation stack have moved little, the balanced system has moved up as much as natural ventilation, causing the air-to-air system to deliver a bit more than the target.

The additional loads in Fig. 6b have all decreased (with the exception of the balanced system, which is unchanged). There has been some change in the order, but the general trend is the same. The air-to-air system has become relatively less attractive as the envelope leakage area has increased.

Figure 7 shows the pair of ventilation plots for a loose envelope of 1000 cm² of leakage area. The target ventilation is met almost all of the time (See Fig. 7a); only for the calmest conditions will natural ventilation be insufficient. Although it is unlikely that a mechanical ventilation system would be installed in such a leaky structure, it is useful to consider the effect it would have on ventilation loads.

From Fig. 7b we see that the additional ventilation load again has decreased for all the unbalanced systems. The balanced system has become about as effective as the exhaust fan without recuperation, and the ventilation stack is only marginally worse. The exhaust system with heat recovery deviates only slightly from zero -- implying that it creates no additional ventilation load; furthermore, the heat-pump system is a net energy producer under any weather conditions. These facts suggest that putting in an exhaust system with heat recuperation may be effective even in a

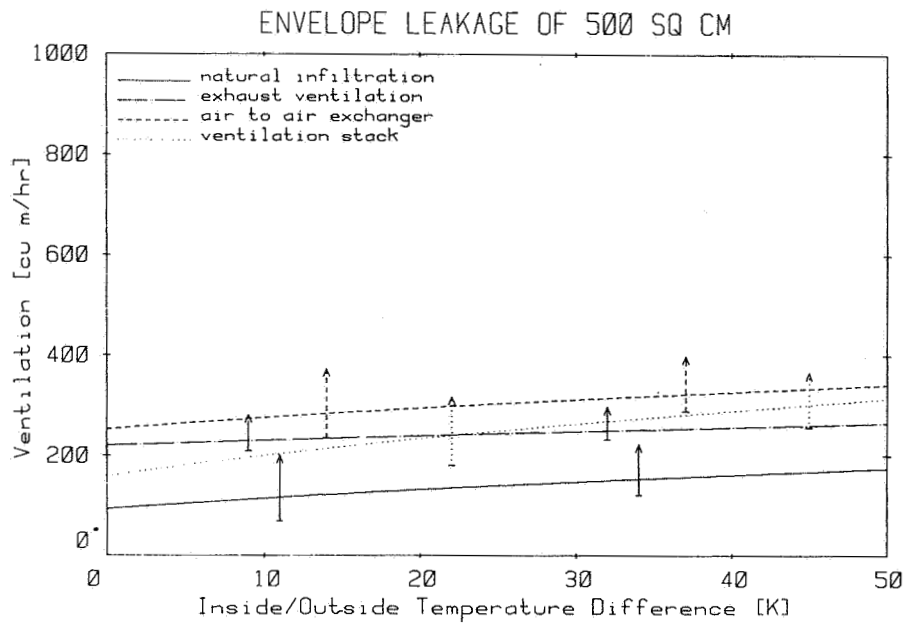


Fig. 6a. Ventilation for envelope leakage area of 500 cm².

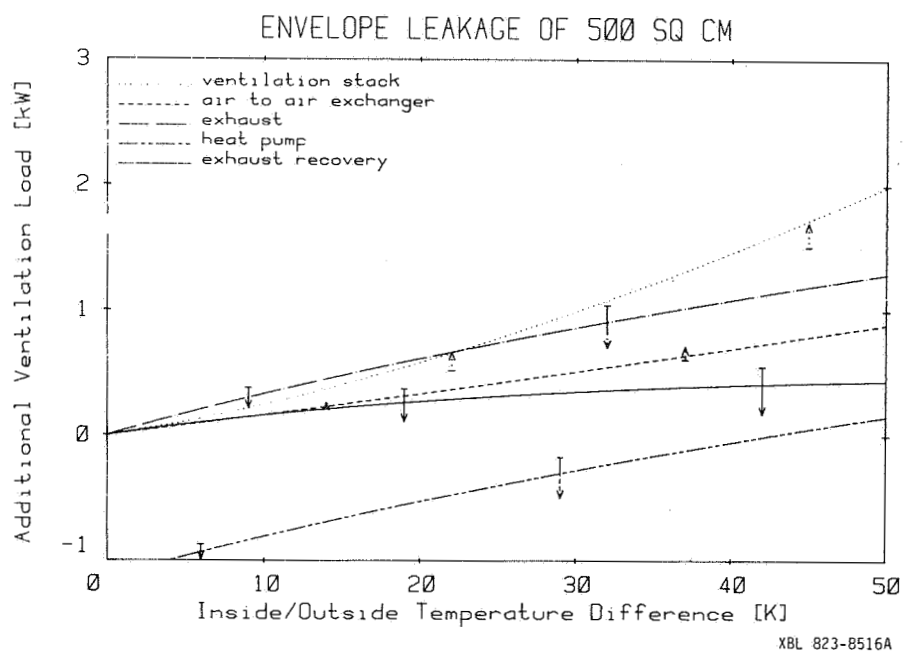


Fig. 6b. Additional ventilation load for envelope leakage area of 500 cm².

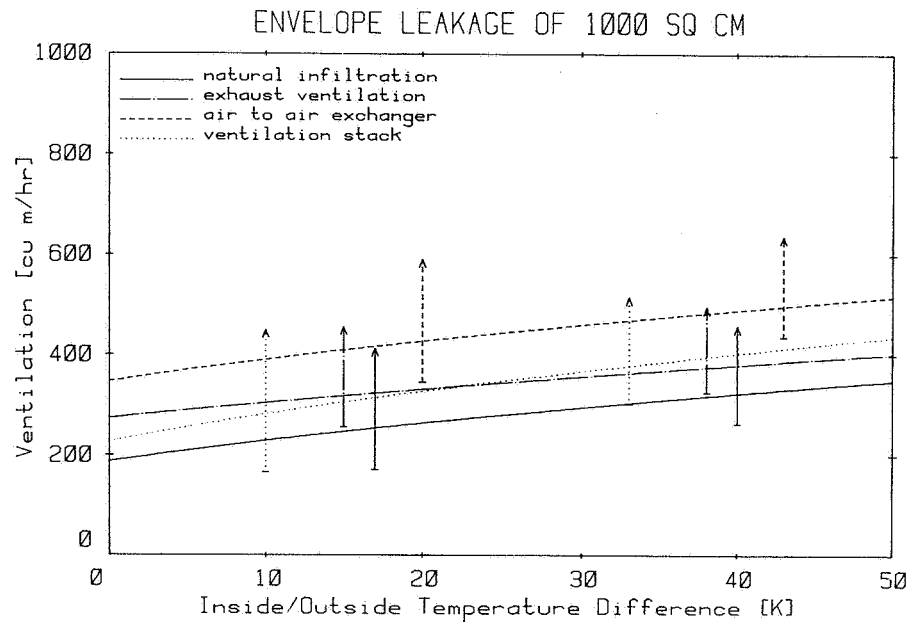


Fig. 7a. Ventilation for envelope leakage area of 1000 cm^2 .

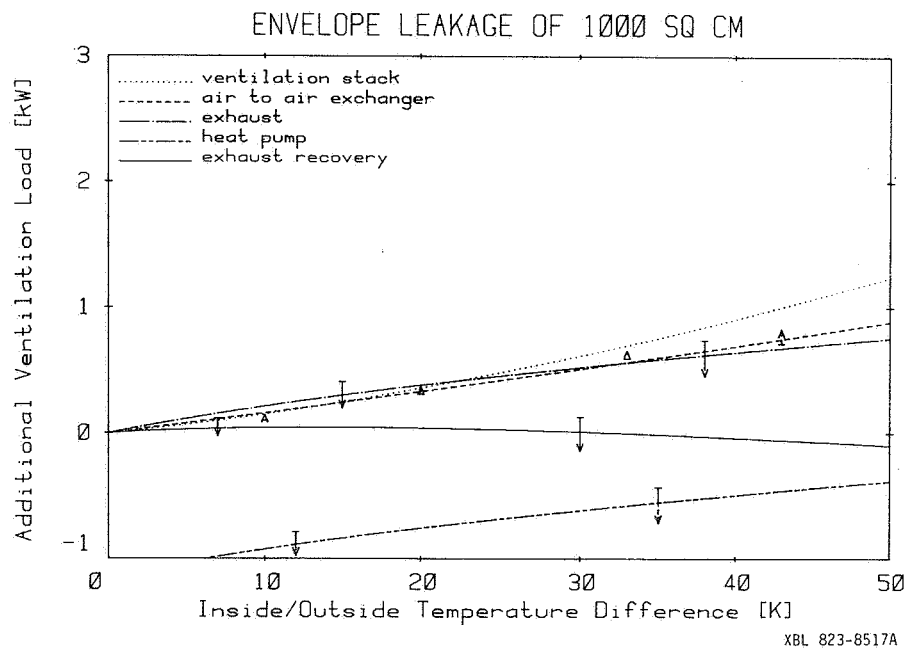


Fig. 7b. Additional ventilation load for envelope leakage area of 1000 cm^2 .

leaky structure.

Because all of these systems have different construction costs, maintenance costs, and operation characteristics it is difficult to compare them in general. For example, we can look at the power requirements of each system. Natural ventilation and a stack ventilator are completely passive and require no power; the other systems all require air-handling fans that will consume between 50 and 200 watts of power; additionally, the heat pump will use an extra 400 to 600 watts to pump the heat back into the house. The disposition of this energy will be different also. In the balanced system and the regenerative system about half the power will be recovered; in the pure exhaust system none of it will be recovered; and in the heat-pump system almost all of it will be recovered. The effect of this recovered heat on the energy costs of the house will depend on such factors as climate and type of fuel and heating system used.

8. SUMMARY

The effect of leakage area on the total ventilation provided by various systems can be quite different. The natural ventilation will be proportional to the leakage area, and the balanced air-to-air system will increase the ventilation by about the same amount; the unbalanced systems, however, are far less sensitive to air infiltration than is the balanced one. Thus, as can be seen in Fig. 8, the natural ventilation and air-to-air system are strongly influenced by an increase in leakage area, but the exhaust systems are not.

The influence of leakage area on different systems becomes more apparent when one looks at the additional loads, as shown in Fig. 9. The additional load for the balanced system is independent of leakage area, but the additional load for any of the unbalanced systems decreases with increasing leakage area, thus favoring the balanced system in tighter structures and the unbalanced systems in looser structures.

In this report we have presented a comparison of the ventilation and load aspects of a few mechanical ventilation systems as a function of weather and leakage area for a typical detached structure. The results do not purport to indicate the best ventilation system for all cases; rather, we have shown that which system is best, and of course whether one needs a ventilation system at all, is critically dependent on the envelope leakage. Furthermore, the choice of an economical optimum will depend strongly on climate, system performance, existing HVAC plant, fuel type, and costs. Previous studies⁷ for much more restricted conditions have borne this out.

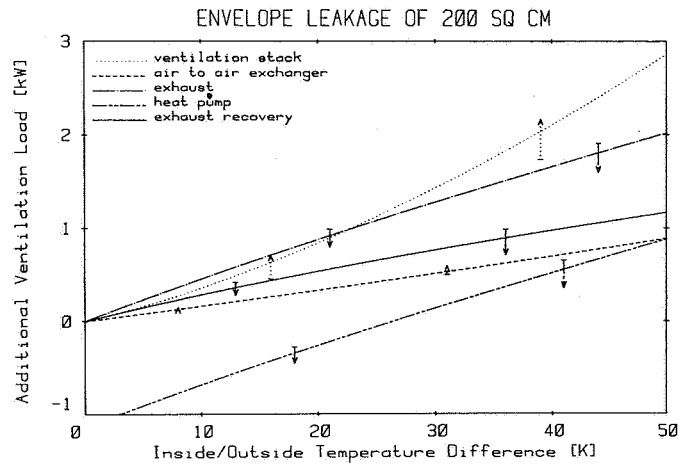


Fig. 8a. Summary of total ventilation for 200 cm² of leakage area.

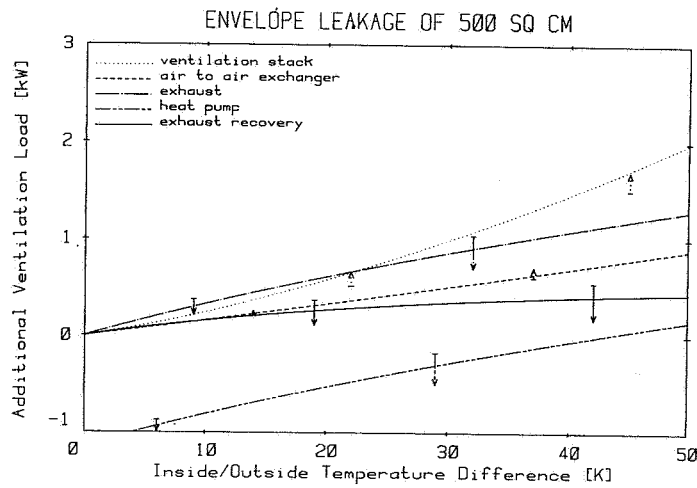


Fig. 8b. Summary of total ventilation for 500 cm² of leakage area.

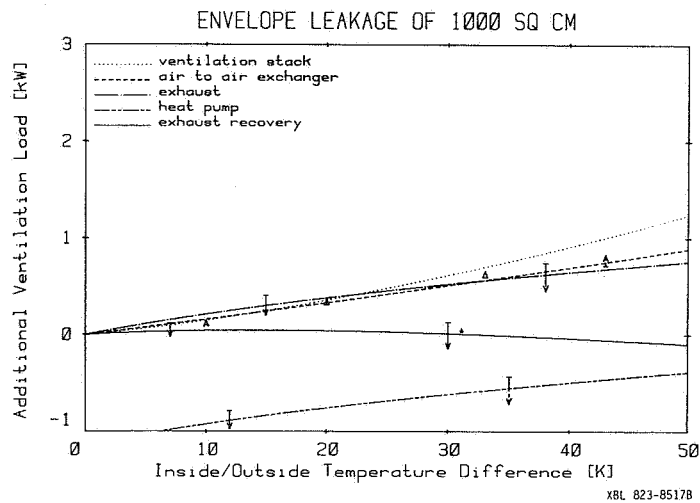


Fig. 8c. Summary of total ventilation for 1000 cm² leakage area.

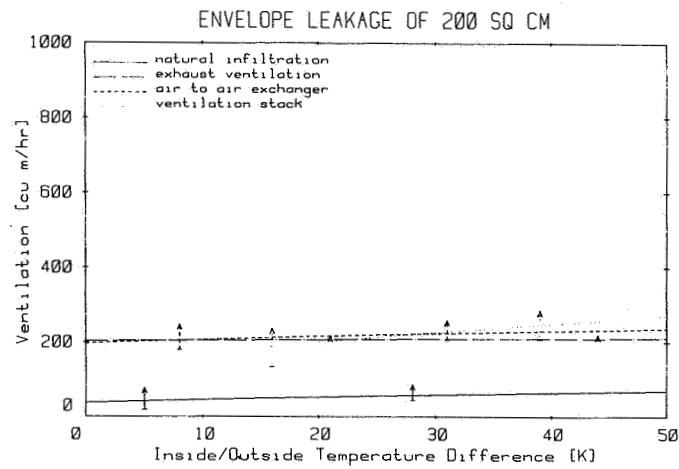


Fig. 9a. Summary of additional load for 200 cm² of leakage area.

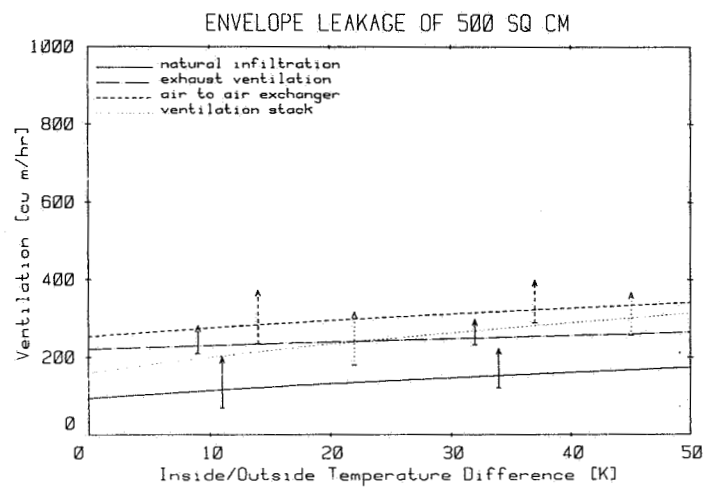


Fig. 9b. Summary of additional load for 500 cm² of leakage area.

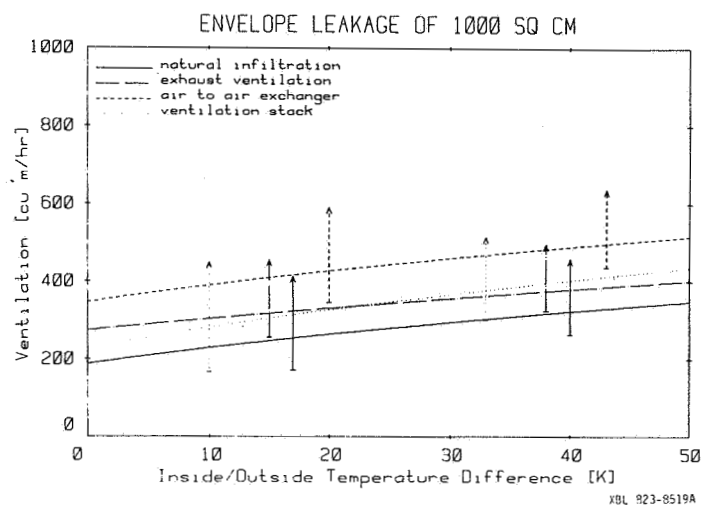


Fig. 9c. Summary of additional load for 1000 cm² of leakage area.

10. ACKNOWLEDGEMENT

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 8

VENTILATION AND ENERGY CONSUMPTION.
PRACTICAL EXPERIENCE OF PROBLEMS RELATED TO
VENTILATION IN SINGLE FAMILY HOUSES.

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1. INTRODUCTION

During the last few years, ventilation has become increasingly important in the total consumption of energy. The rules of the Swedish Building Code (SBN 75) have been made stricter regarding heat loss. Therefore, it has become necessary to build houses that are more airtight and to reduce the air infiltration through elements and the building in general.

These requirements increase the need for mechanical ventilation systems, which would lower the energy consumption during the cold part of the year and, if required, increase the ventilation during the summer.

Basic knowledge seems to be insufficient in the field of building and ventilation technology. The reason for this may be that ventilation in small houses is considered to be far too elementary. There is no total view of the balance of energy and no regard is paid to the interplay between different flows.

2. RATE OF INFILTRATION

Quantitative information about the magnitude of the ventilation losses has been given by, among others, Boman [1], Gustén and Johansson [2]. These analyses indicate that the total ventilation in small houses built between 1960 and 1980 is much lower than previously presumed and amounts to 0.2-0.4 changes per hour. The measurements have been done by the tracer gas method during "normal" climatic conditions in about fifty small prefabricated houses with natural or forced ventilation.

The new building regulations have necessitated considerable construction development aiming at tighter houses. For small houses, the code prescribes that infiltration maximally should be three changes per hour at an average of 50 Pa positive or negative pressure difference. This roughly corresponds to a natural ventilation of 0.1-0.2 changes per hour, as measured by the tracer gas method [2]. Depending on the type of design and the degree of prefabrication, the additional cost for obtaining this level of air-tightness varies between 500 and 2,000 Sw. Cr.

3. REGULATIONS FOR VENTILATION ACCORDING TO SBN 75

When determining air-flows in a building, one has to consider regulations both for certain rooms and for the total ventilation, which should be $1.26 \text{ m}^3/\text{h}/\text{m}^2$ of residential floor area. For ordinary types of houses this requirement corresponds to 0.5 changes per hour. According to SBN 75, natural ventilation is still permitted in spite of the risks of hygienic inconveniences.

If a mechanical ventilation system is chosen, the system is to be dimensioned for a total ventilation of 0.5 changes per hour. The contribution from unintentional ventilation is not taken into consideration. It must be possible to reduce the designed air flow rate and, if required, force the flow.

4. CONSUMPTION OF ENERGY

The heat balance is exemplified by a 1½-storey newly-built house in the south of Sweden. The house (figure 1) has a volume of 320 m³ and is designed in accordance with SBN 75.

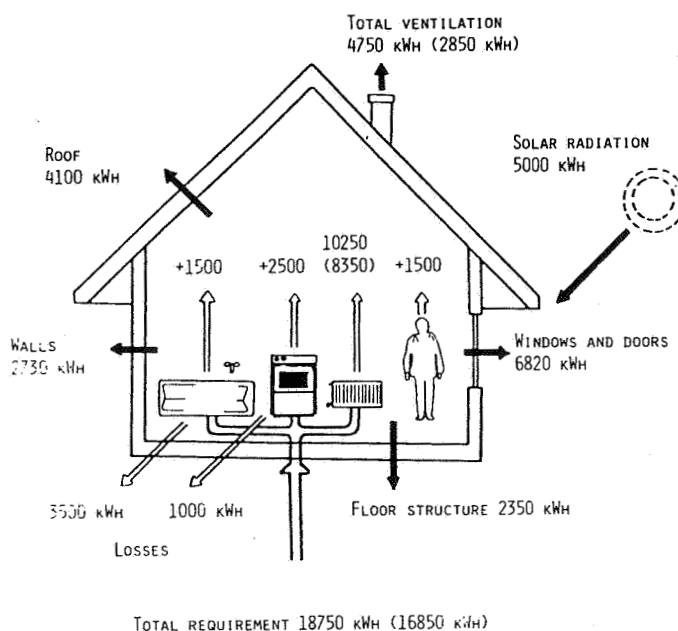


Figure 1. Heat balance at a total ventilation of 0.3 and 0.5 changes per hour. The values within parentheses correspond to 0.3 ach.

The prescribed air change rate - 0.5 changes per hour - gives energy losses of 4750 kWh. In practice, as mentioned before, the air exchange rate will amount to 0.2-0.4 changes per hour. Compared with that of standardized ventilation (0.5 changes per hour), the usual energy losses characterized by 0.3 changes per hour will be 1,900 kWh less. Cooking requires a change rate of 1.0 for one hour per day, resulting in an annual addition of 280 kWh. Smoking and laundry would also increase the necessary change rate.

5. THE NEED FOR VENTILATION

Several factors determine the necessary level of ventilation. Forced ventilation can be the cause of draught and costly losses of energy. Low ventilation can result in:

- Problems with moisture
- Lack of oxygen
- Unpleasant smell
- Problems with CO and CO₂
- Unacceptable content of poisonous gases

Special attention must be given to studies of radioactive gases from different building materials.

At the Department of Structural Design, Chalmers University of Technology, measurements have been done in about fifty timber houses without basements. These investigations indicate that an air change rate of 0.2-0.3 changes per hour is sufficient, provided that the ventilation systems are working and the distribution among different rooms is proper.

Since 1975 roughly 700 houses per year, of the same type as the houses studied, have been produced. The number of complaints is extremely low, and where legitimate, the main reasons are defective workmanship, incorrect adjustment, or insufficient information about how to use the ventilation systems.

The occurrence of radon in dwellings has for some time dominated and to some extent guided the debate on ventilation. The existence of radioactive gases does not mean that the rate of ventilation should be increased generally. The regulations should consider building materials and soil conditions, and the

requirements should be less rigid for, e.g., timber houses without basements.

A number of houses that have been built according to SBN 75 have an increase or only a slight decrease in the consumption of energy compared with similar houses built before the oil crisis. The Swedish Parliament's political decision to reduce the consumption of energy by 40% in new houses has not been fulfilled. Therefore, it seems wrong, at least from an energy-conserving point of view, to solve the problems of radon merely with more ventilation. Many years' work to save energy will be frustrated if increased ventilation and airing and in certain cases removal of draught strips are recommended. Instead, the ventilation losses must be limited by means of appropriate building materials in combination with moderate ventilation and air-tightness.

6. CONTROL OF OPERATION AND ADJUSTMENT OF EXHAUST AIR TERMINAL DEVICES

There are some important difficulties in getting even a simple mechanical ventilation system to function properly. Figure 2 shows some differences between the total ventilation for a house and the air flow through the ventilation system.

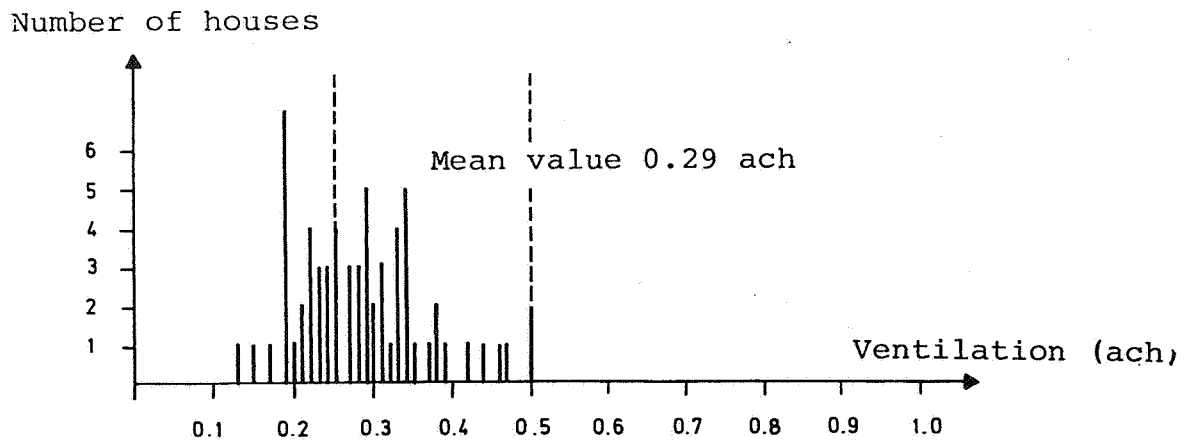


Figure 2A. Measured total ventilation (ach).

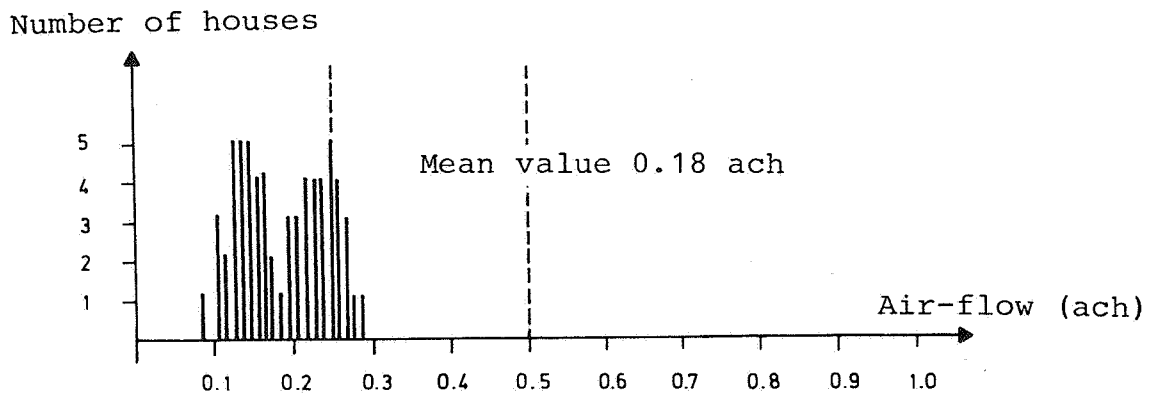


Figure 2B. Measured air-flow through the ventilation system (ach)

The reasons for such differences are careless and defective assembly and lack of knowledge of the system design; in addition, control and inspection often do not occur, figure 3. Consequently, the building industry is not able to meet advertising claims and promises made by the manufacturers of the systems. The equipment must be of good quality and must be installed and tested by competent personnel. Therefore, ventilation systems should be delivered with

a guarantee covering adjustment and operation.

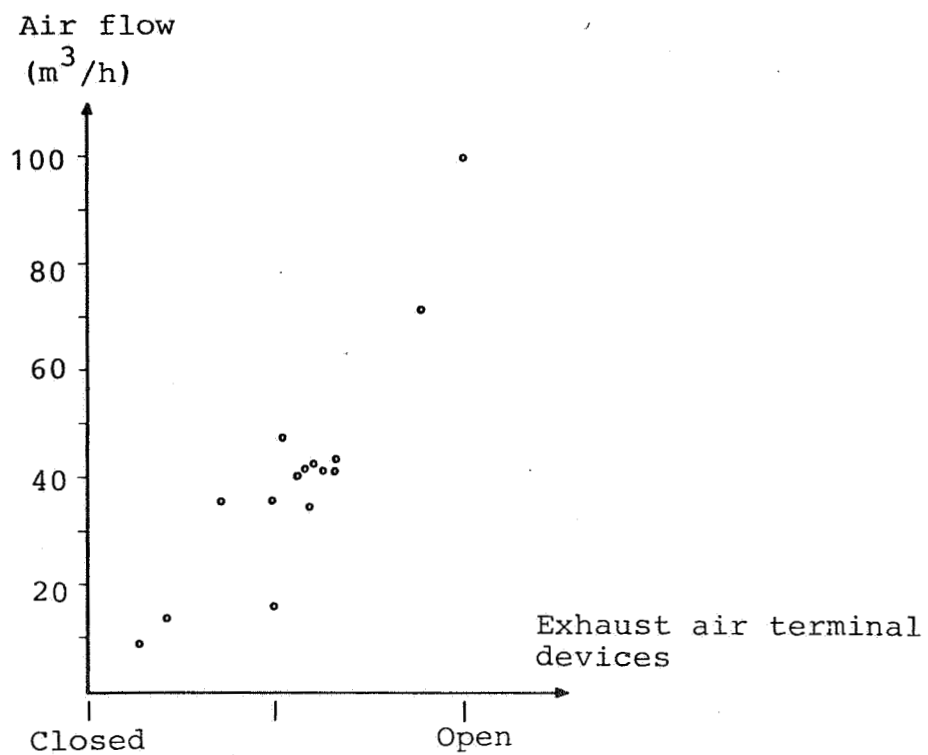


Figure 3. Recorded air-flow and adjustment of exhaust air terminal devices. The devices were presumed to have an air-flow of 50 m³/h

7. BALANCED VENTILATION WITH HEAT RECOVERY UNITS

Compared with those of mechanical exhaust air systems, the risks of incorrect air flow rate and air-leakage from ducts are in practice considerably larger with balanced ventilation. Balanced ventilation makes severe demands on production and results in a greater amount of work for the builders. It is rather difficult to adjust and measure the air flow rate through supply air terminal devices.

Advantageous loans have to a great extent promoted the sale of balanced ventilation systems with heat recovery units. These systems are expected to give better comfort and recovery of heat. Reduced ventilation losses can lead to a reduced need for heating and to smaller radiator-surfaces.

The proportion of real saving of energy with these kinds of systems has not been determined. When estimating the profitability, one must pay attention to the real airflow rates and the interplay between different flows in newly-built houses.

Heat recovery units belong to those components for saving energy which have not been subjected to any severe criticism. Reasons for this may be that:

- The recovery units make it possible to get favourable loans
- The marketing is effective
- Experience from heat recovery units in industries, offices and stores is applied to small houses, and no attention is paid to the smaller building volume and need for ventilation
- The builders' knowledge about ventilation is insufficient

8. THE EFFICIENCY OF HEAT RECOVERY UNITS

All the producers of heat recovery units show the performance of the exchangers with the so-called temperature efficiency η_t . This quotient is defined as

$$\eta_t = \frac{t_{sa} - t_{sb}}{t_e - t_{sb}}$$

where t_{sa} = supply air temperature after the exchanger
 t_{sb} = supply air temperature before the exchanger
 t_e = exhaust air temperature.

The temperature efficiency gives a comparison between different types of heat recovery units but can hardly be the basis for forming a judgement of the energy saving (figure 4). Measurements indicate that only 60-80% (i.e., 0.3-0.4 changes per hour) of the total ventilation is guided through the ventilation ducts [2].

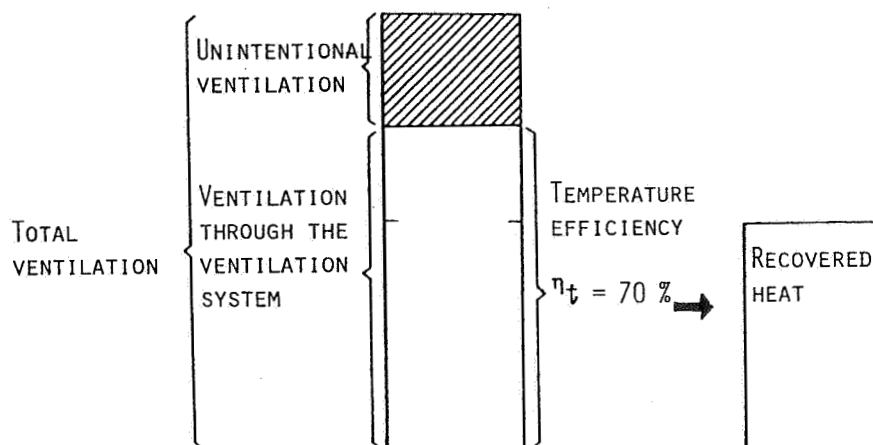


Figure 4. The limitation of the temperature efficiency

9. ARE STRICTER REQUIREMENTS ON AIRTIGHTNESS DESIRABLE?

Of course, heat recovery units become more profitable in extremely airtight houses with more air changes. Balanced ventilation with heat recovery will probably be of interest when houses are tighter than 3 changes per hour at 50 Pa. This value, according to SBN 75, has been attained at a cost of 500 to 2,000 Sw. Cr. Still stricter requirements for air tightness, to one change per hour, will result in an additional cost of between 2,000 and 5,000 Sw. Cr. [3]. Different types of adhesive tape are being used to achieve more airtight houses. The durability of such engineering must be questioned. The consequences of a decision like that must, however, be analysed, particularly since the need for ventilation to reduce radon and moisture has not been determined. Is it - from an economic and practical point of view - sensible to reduce the unintentional ventilation further? What will be the consequences if the mechanical ventilation system ceases to work due to a power failure or for any other reason?

10. RECOMMENDATIONS AND CONCLUSIONS

The purpose of this paper has been to indicate the need for development in the ventilation field. The following recommendations can be given:

- Increased knowledge.
Because of defective workmanship and lack of information about the quality and function of the ventilation systems, the expected saving of energy has failed to appear, wholly or partially.
- Differentiated requirements on ventilation.
Differentiated requirements for ventilation, well-documented from radiation research, ought to be introduced in the Swedish Building Code.
- Inspection of systems.
The authorities must make sure that mechanical ventilation systems always are adjusted.
- Comfort-criteria.
Comfort-criteria must be defined for different types of ventilation systems and supply air terminal devices.
- Changed conditions for heat recovery.
Heat recovery can be used only during the heating season. Therefore, heat pumps, which use the heat in exhaust air flows to produce hot water, are good alternatives.
- Well-documented requirements for air-tightness.
The requirements on air-tightness must be put in relation to the real possibilities of

saving energy. Considering the costs, the regulations must not be made stricter until problems of workmanship, durability and dependability have been solved.

11. ACKNOWLEDGEMENTS

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 9

EXPERIMENTS WITH A PASSIVE VENTILATION SYSTEM

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SYNOPSIS

The trend towards more tightly sealed houses is now well established and controlled ventilation is becoming a necessity; a simple method of ventilation is proposed comprising an extract system and air inlets. The extract system is effectively a flue connecting to vents in the kitchen and bathroom and it relies on thermal differences and the wind to create air flow; air enters the house via slot vents over windows. No maintenance is required other than cleaning the vents, there are no running costs and initial installation costs are low.

The proposed system has been installed in a timber framed house. Air has been extracted through the system up to the equivalent of 0.3 ac/h for the house volume, but this is less than expected.

The house needs to be tightly constructed, the extract tubes need to have very low air resistance and care is needed in siting the inlets for efficient operation. The effect of wind has not been measured, but a device may be necessary to prevent over extraction. Modifications to the system are proposed for future work.

1. INTRODUCTION

As insulation standards increase in houses, the transmission heat losses decrease and therefore the energy loss associated with the ventilation air becomes a greater proportion of the fuel bill. In order to attain further energy savings therefore, attention turns to reducing ventilation rates and a balance has to be found between a rate which represents high costs and one which keeps the air quality sufficiently high for comfortable, healthy living.

Many papers have been written on this subject and a suitable rate appears to be between 0.5 and 1 ac/h (air changes per hour). Having determined the necessary rate, the next problem is how to achieve it and again many papers have been written discussing various solutions using mechanical or natural ventilation.

Mechanical ventilation is attractive because the chosen ventilation rate can be easily controlled provided the house has a low natural air infiltration rate, but the cost of installation may be high, maintenance is needed and there may be a problem of noise from the fans. There may be other objections also such as that from one Local Authority who will not fit mechanical systems in its public sector houses because when breakdowns put houses at risk from condensation, the first complaint it would receive in many cases would be only when damage had already occurred. The repair costs would then be increased because of the need for remedial work such as dealing with mould growth or redecorating. Whatever the reasons, although the use of such systems is increasing, they are not yet commonly installed.

Natural ventilation is seen as the alternative but it suffers from variability of rate depending on the weather; this means that if no intervention is expected from the householder, there may be times of excessive or insufficient ventilation. For example, if the design is for the 'correct' rate under 'average' conditions, then high wind will cause excessive ventilation and possibly even draughts.

It is a problem which must be faced and a solution found if energy consumption is to be reduced to a minimum without harmful side effects. There are many such effects but probably the most easily recognisable is that of surface condensation which ranges from that occurring on single glazed windows to extensive mould growth, peeling wallpaper and degraded plasterwork. Clearly if ventilation rates are reduced too much in a situation where moisture production is maintained, the risk of condensation is increased.

The current trend in British housing is towards timber framed construction which uses insulation extensively in walls and roofs: any problems of increased condensation risk in these houses is therefore the concern of TRADA (Timber Research and Development Association) and of Pilkington Brothers (Fibreglass) for although there should be no condensation problems associated with the use of timber or Fibreglass as such, any problems arising for other reasons might become associated with the use of these materials.

A passive system of ventilation was proposed as an inexpensive solution to control the ventilation rate in a tightly constructed house and TRADA and Pilkington Brothers have jointly carried out a preliminary investigation to determine if it is feasible; the results and conclusions of this initial work are presented here.

2. THE PASSIVE VENTILATION SYSTEM

2.1 Principles

One of the causes of high humidity and consequent condensation problems is claimed to be the demise of the open fire¹ and the absence of its associated chimney. The Local Authority referred to previously insists on having a flue built into each of its public sector houses and claims that it receives few complaints of condensation problems because of the effects of this policy.

The chimney forms the basis of the solution proposed here: if their removal has resulted in insufficient ventilation rates and the objection to their retention is excessive ventilation and draughts, then surely one built solely for the purpose of ventilation and which could therefore be fitted with a device to limit extraction, could provide a basis for house ventilation. The system would be cheap to install, free to run, and would be virtually maintenance free.

A flue will extract air from a building by virtue of the pressure differences created by a) wind and by b) the stack effect (the result of the difference in temperature between inside and outside). In a house, the stack effect can be relied upon throughout the winter and the effect will be greatest when the outside temperatures are lowest, that is, when most needed because then particularly, windows are kept shut. Consider a 200 m³ house constructed tightly so that its natural air infiltration rate is low, say 0.25 ac/h under normal conditions. If the house is at 20°C with an external temperature of 0°C with no wind, then a single flue with a diameter of 165 mm, terminating at the roof ridge should produce an air change rate of 0.5 ac/h provided there is an equivalent diameter air inlet to the house; this air change would be in addition to the basic infiltration rate² producing 0.75 ac/h in total.

If the heating season is taken as October to April inclusive then using the warmest monthly mean temperature (for Manchester), the extraction rate produced by the flue would drop to 0.35 ac/h, giving 0.6 ac/h total for the house.

The effect of wind will be to increase these figures. Some spot checks have been made on air flows in unused house flues and these indicate that extraction rates equivalent to up to 0.6 ac/h might be expected with winds up to 6 on the Beaufort Scale, which is probably not as high a rate as is generally expected. However, this rate would not be fully additive to the stack effect rate³ and so with this wind and 'full' stack effect the total rate would be less than 1.35 ac/h.

The proposed scheme of fitting a flue into a house should thus result in a total ventilation rate of between 0.6 and 0.75 ac/h during the heating season in the absence of wind; it should drop below 0.6 only when the outside temperature exceeds about 10°C. This will be for short periods in winter and will be followed by increased air change rates when the temperature drops again, and for longer periods in autumn and spring, but will again be compensated for, on average. During autumn and spring opened windows are more likely as the outside temperatures rise and the extraction rate drops. Wind will increase the ventilation, but it should be possible to design against an excessive rate if this is found necessary; presumably the basic infiltration rate for the house will also increase under the influence of wind and so from this point of view alone it is desirable to have that rate as low as possible.

The flue should extract direct from the kitchen and the bathroom where most moisture is created. If the kitchen has twice the volume of the bathroom and the flue divides in the proportion of 2:1, then for the 200 m³ house, ventilation rates of about 3 ac/h can be expected for these rooms when the overall house rate is 0.5 ac/h.

A successful passive ventilation appears possible, particularly as with suitably positioned inlets, air movements will be encouraged into the kitchen and bathroom from the rest of the house, preventing the spread of moisture as well as removing it.

2.2 Practical Details

A system has been built into an experimental house owned by TRADA and situated at their premises near High Wycombe. It is of timber framed construction with the lower half externally clad with brick and the top with timber boarding; it faces east and there is extensive shading from the sun and wind by a building to the south and by rising ground with high trees to the west. The volume is about 200 m^3 and there are no internal doors fitted. The floor plan is given in Fig.1.

The flue or extract duct is made from components from the Marley Discharge Pipe System with 82.4 mm pipe to the bathroom and 110 mm pipe to the kitchen, which combine in the loft into a single 160 mm pipe. This is terminated at the ridge with a proprietary ridge ventilator and at the lower ends with 90° elbows to simulate fixing into casing, cupboards or the structure, Fig.2.

The lower ends were fitted with Marley self-regulating grilles rates at 40 and $30 \text{ m}^3/\text{h}$ for kitchen and bathroom which are the nearest to the required 66 and $33 \text{ m}^3/\text{h}$ for the 2:1 split of 0.5 ac/h for the whole house. The pipe in the loft is heavily insulated with Fibreglass to maintain the temperature of the air in the pipe, otherwise there would be a condensation risk and a loss of stack effect.

To allow the fresh air make-up into the building, Titon Trimvents were fitted to the main room windows, and these gave a total inlet area of 0.02 m^2 ; it was thought that with these the householder would need to close them to avoid draughts in windy weather. Later, vents of total area 0.05 m^2 were fitted to the external doors to increase the air inlet area for the experiments although it was realised that in practice these would create unacceptable draughts. A pipe through the lounge with an outlet was considered. The basic ideas are shown in Fig.3.

3. MEASUREMENTS

Air flow directions were determined using smoke tubes, and speeds using a thermistor bead air velocity meter. It was found that air flows varied significantly with time and so there was no point in attempting to measure the speeds in accordance with B.S. 1042; what apparently was needed was some form of full flow device, but it was thought than an anemometer fitted into the full diameter of the ducts might cut flow rates.

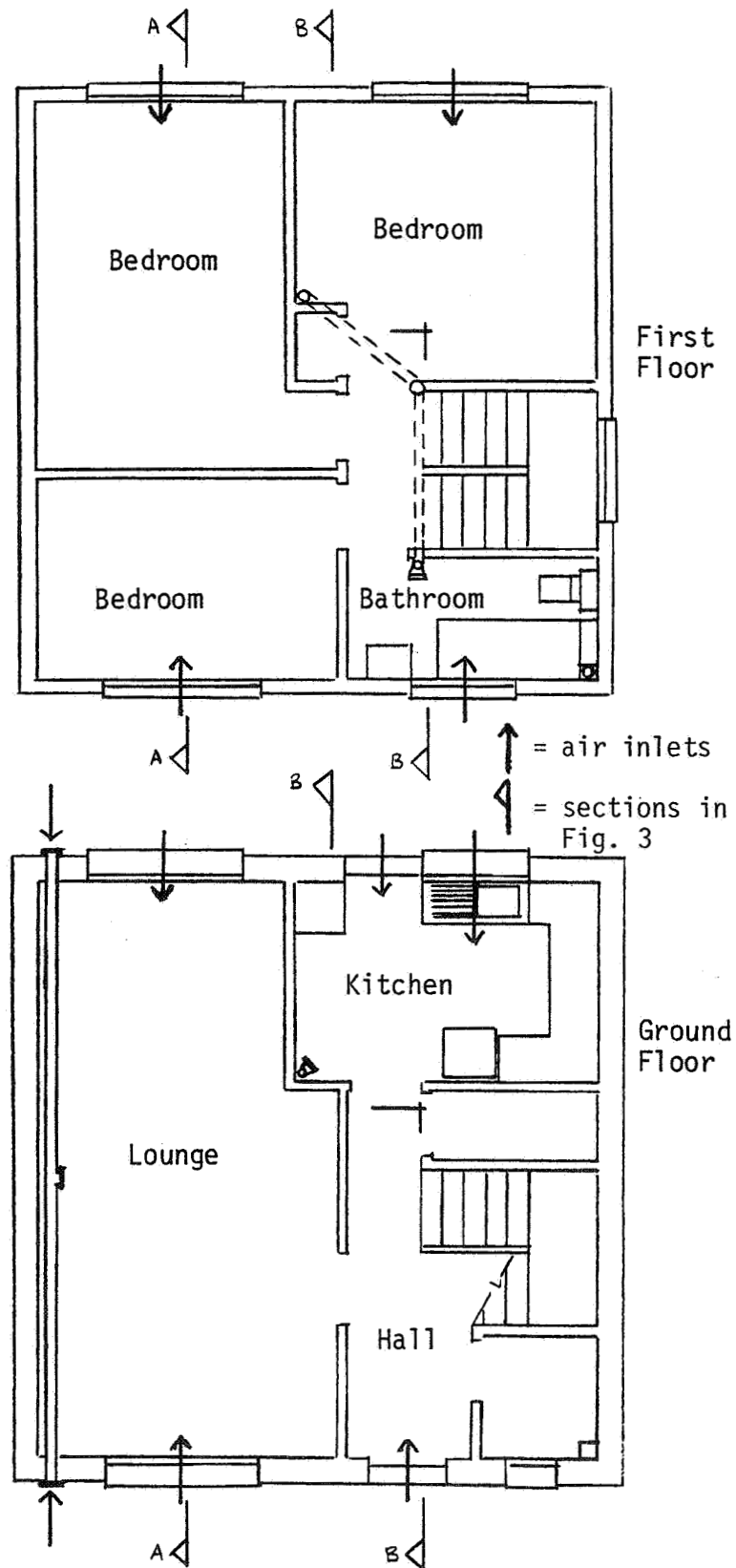


FIG.1 FLOOR PLAN OF TRADA HOUSE

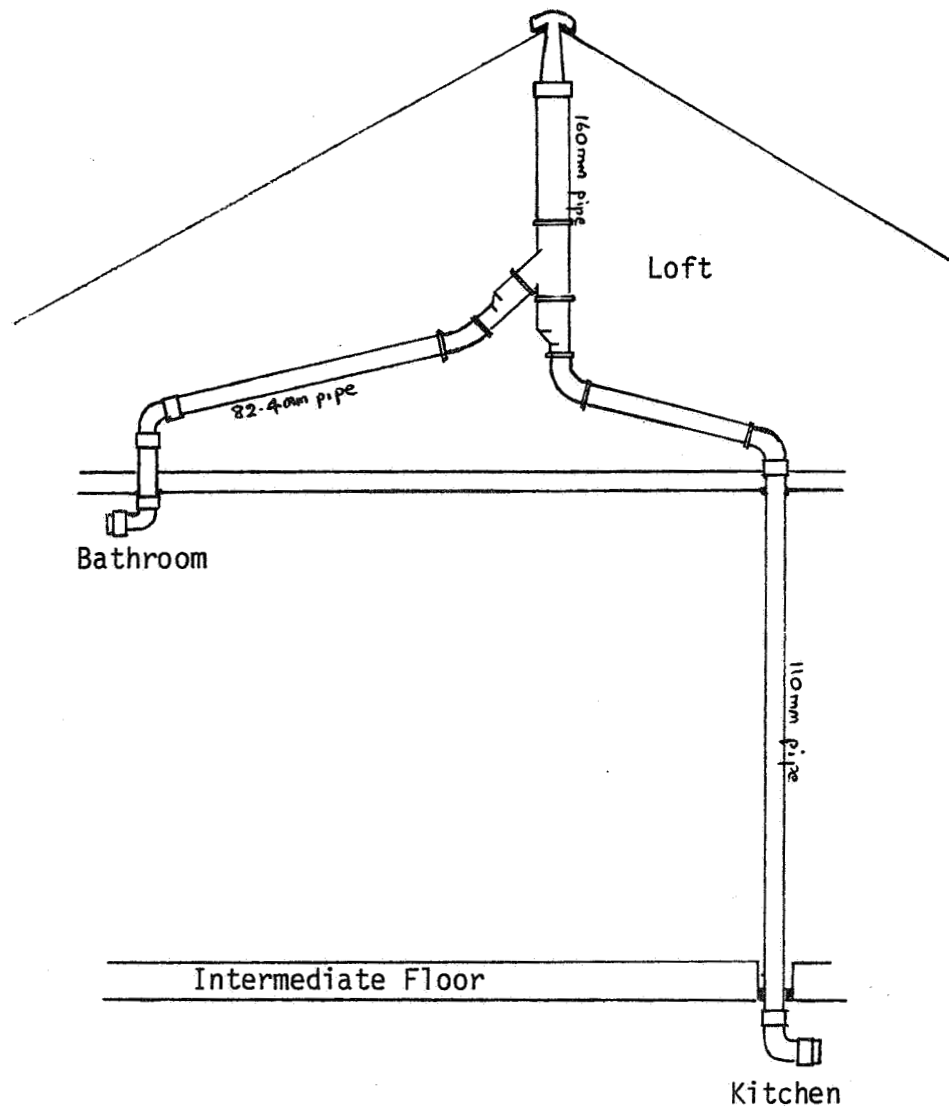


FIG.2 LAYOUT OF DUCTWORK

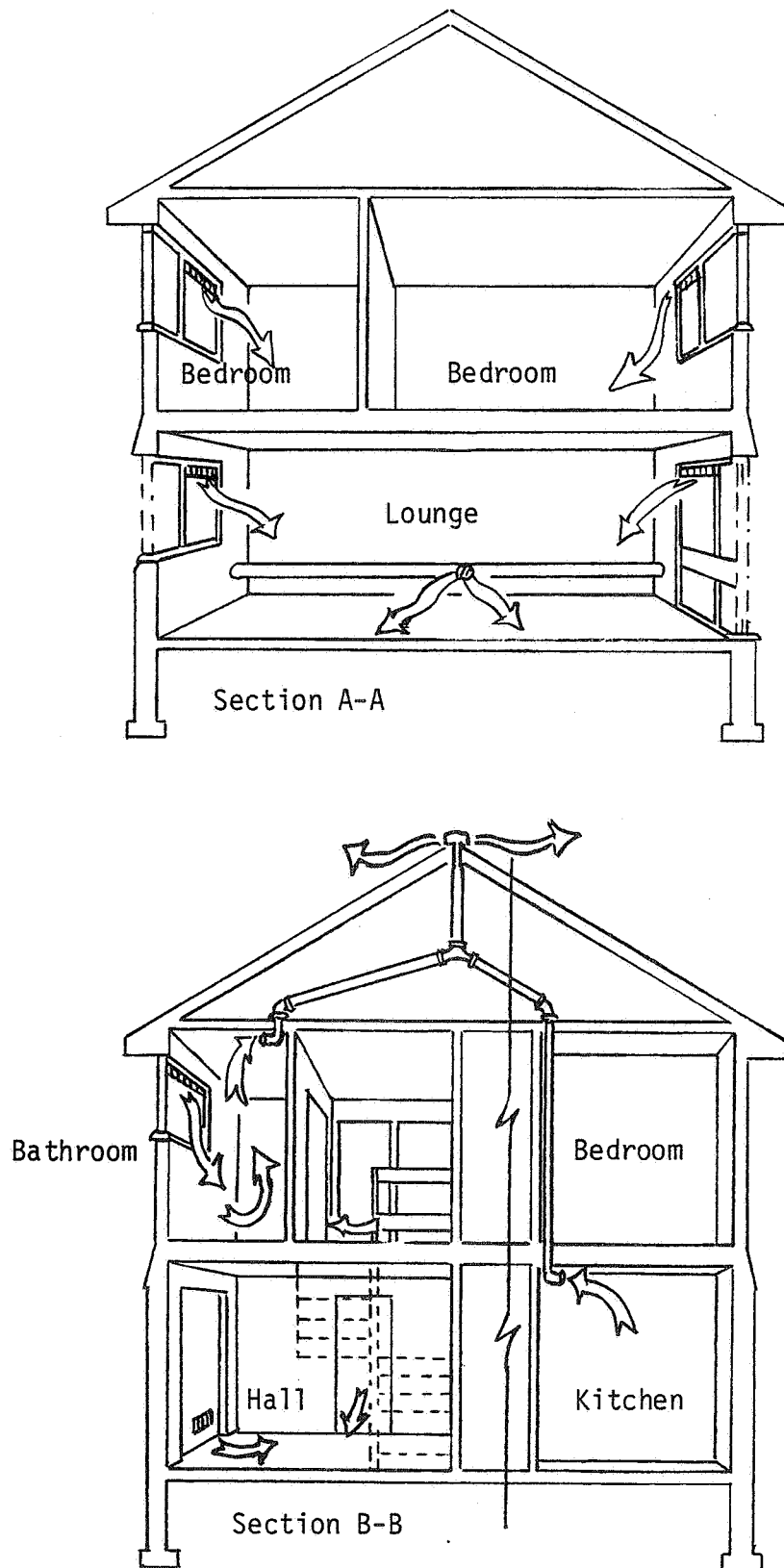


FIG.3 SHOWING POSSIBLE AIR FLOWS IN TWO
SECTIONS (SEE FIG.1) THROUGH HOUSE

A figure for the flows was determined by taking a number of spot readings across the diameter of each duct, but these can only be very approximate.

Overall house air infiltration rates were determined using the CO₂ decay method.

4. RESULTS

The site was very sheltered from the prevailing winds; during the tests, nothing greater than a light breeze was encountered in the vicinity of the house and it is therefore thought that the extract rates achieved were mainly due to temperature effects.

Initially it was found that the house had a high air infiltration rate of between 0.8 and 0.9 ac/h, measured when the inside to outside temperature differences were between 9 and 16°C; the addition of the extract ducting and the opened window vents had no significant effect on the rate although it was shown that with the window vents open, there was a definite flow of air through the house. Total air flows in the ducts varied between 11 and 20 m³/h and seemed to be independent of the status of the window vents. These flows represent up to only 0.1 ac/h for the house and the flow was split equally between the kitchen and the bathroom resulting in 0.4 and 0.8 ac/h for these rooms respectively.

With the self-regulating grilles removed, flows increased to between 27 and 55 m³/h representing up to 0.3 ac/h for the house (average 0.2) and the split was nearer to that calculated giving up to 2 ac/h for kitchen and bathroom (averages of 1.3 and 1.5 respectively).

The building was then sealed by taping component joints and it was found that the basic air infiltration rate had dropped to 0.4 ac/h with inside to outside temperature differences of between 13 and 15°C, that is, with the window vents closed and the extract ducts sealed. The measured total air exchange rate increased to 0.7 ac/h with a temperature difference of 20°C when the window vents were opened, but dropped to 0.6 with a temperature difference of 13°C when the extract ducts were also operative.

With the building sealed and the self-regulating grilles removed flows measured in the ducts ranged from 25 to 46 m³/h with the temperature differences between 4 and 15°C when the window vents were open; when they were closed flows were much less, but insufficient readings were taken to be sure of significance in view of the range of results. There was no correlation apparent between flows and the temperature differences.

With the house still in the sealed condition and no self-regulating grilles fitted, the window vents were shut and effectively replaced by the vents fitted to the front and back doors. With inside to outside temperature differences of 5 to 10°C, total flows in the ducts were between 31 to 40 m³/h; however, it was found that the unconnected soil pipe installed in the house had inadvertently been left unsealed during this part of the tests. About 9 m³/h of air was found to be flowing out of the house through this soil pipe and therefore it is likely that the figures of 31 to 40 for the ducts are lower than they would have been. No difference in air flows was noticeable whether just one door vent or both were open and again no correlation could be found between flows and the temperature differences.

The directions of the air flows in the ducts and at the window and door vents were determined in the tests. At all times, the flow in the extract ducts was out of the building, and the open window vents on the ground floor always had air entering the building through them. Upstairs, the flows through the window vents were complex, with some vents with air flowing in, some flowing out; there was no correlation with wind direction (but wind strength was between light and not measurable). There was a significant decrease in the proportion of the upstairs vents which were found to have air leaving the building after the house was sealed. The external door vents always had air entering the building.

5. DISCUSSION

The results have suffered because of the inaccuracy of the air flow measurements, but nevertheless trends are apparent, some conclusions can be drawn and suggestions for improvement made. The tests using the CO₂ decay may be useful for determining the general level of tightness of a house, but the use of the difference between two tests to determine the flow into a ventilation system which is a minor proportion of the total flow is not suitable in view of the variation from test to test and the extended time of such tests; an accurate direct measurement of flow is required.

The initial test results with the house 'unsealed' suggest that in such a house the addition of ventilation openings will not significantly affect the overall ventilation rates at least under the conditions of little wind. If one accepts the suggestion that the total leakage paths in a house may add up to half the size of a door ⁴ then this is not surprising as the window vents are only 2% of this area.

Flows were insufficient for the self-regulating grilles to be called on to control the rate, but in any case they caused a considerable reduction in flow in the ducts under the conditions used, presumably because of the restriction of cross-sectional area, which is similar in both grilles used.

The restriction explanation is strengthened because the flows in the differently sized ducts became equal with the addition of the grilles. There is no criticism of the grilles however, as they are primarily designed for use in a forced extract system and not with the (presumed) very low pressure drops associated with this system.

Without the grilles, extract rates achieved were less than expected. The calculated rate for the ducts used and temperature differences encountered was between 38 and 77 m³/h compared to 25 to 55 measured. This calculation assumes equal size of inlet and outlet and when the house was unsealed, it was thought than an inlet of at least this size would effectively exist; the window vents themselves provide this.

The sealing of the house made little difference to the flows in the ducts and thus if a house is made very tight and extra ducting installed, or the current system made more efficient, then it is likely that a high proportion of the air change will be via the duct system. The small number of results with the house sealed and the window vents shut are consistent with what might be expected: as the fortuitous openings are reduced it becomes necessary to deliberately let air in for the extract system to work.

The experiments with the door inlets are interesting. Calculations show that flows of 54 to 77 m³/h were expected with one door inlet open and of 68 to 97 m³/h with the two inlets open, compared to the 31 to 40 achieved.

It is significant that doubling the inlet area by adding a vent to the opposite side of the house had no effect on the measured flows, even though air was entering both door vents.

This result coupled with the direction of air flows at the window vents and the effect of the vent status on air flows in the ducts leads to the suggestion of a model for a house in the absence of wind, given below.

In a house the general flow of air is in near the ground and out near the top because of the temperature differences between inside and outside. If a duct system is introduced in a house which has a high natural air infiltration rate some of the existing air flow will travel via the ducts because there is less resistance to flow by that route, but the overall air change rate for the house will not be markedly affected because the increased area of flow path available through the outer structure is not significantly altered. If in addition to the ducts, extra inlets are introduced, again little change will be apparent because the increase in area is still insignificant.

If the house has a low natural air infiltration rate, the introduction of a duct system alone will again have little effect on the overall exchange rate because the existing fortuitous inlet areas are already feeding the fortuitous outlet areas, but with

the addition of downstairs window vents flow will increase in the ducts because both inlet and outlet areas are increased and the flow path resistance is decreased. However if 'inlet' window vents are fitted upstairs as well, the duct system will be partially by-passed with air passing directly between the vents.

6. FUTURE EXPERIMENTS

The efficiency of the duct system itself needs increasing and this could be brought about by a series of measures. The less the natural infiltration rate of the house, the stronger will be the routing of air flows through the ducts, so further sealing is required. In view of the large effect caused by the self-regulating grilles on the flow rates, it appears that the system is very sensitive to resistance to flow and the following modifications are suggested: removal of the bottom end elbows with the pipes then terminating flush with the ceiling with simple grilles to make them visually acceptable; removal of other bends and junctions in the pipes and removal of the sloping sections, which means two separate ducts for the kitchen and bathroom; possible change of design of the outlet terminal to ones with less restriction to flow. The fitting of separate ducts will allow an increase of total cross-section of ducting.

In an area where the house is subject to winds, it will almost certainly be necessary to fit some device in the ducts to stop over extraction.

The details of how to allow air into the building causes some concern. On the ground floor, the through tube with an outlet may not be necessary as no cross flow of air between the window vents was recorded, but some form of large area inlet is needed which is draught free: window vents may be acceptable if the householder can be relied upon to open them. These should not be in the kitchen but in the lounge and the doors between the lounge and kitchen when fitted should have a deliberate air flow path. Upstairs it appears that no inlet vents should be allowed, otherwise the air flow will be diverted from the duct system; in particular, the kitchen extract efficiency would be reduced. Although the rooms would be ventilated if the doors had vents, there would be no control over the air flows. It is possibly useful to consider fitting small ducts from the bedrooms which feed into the bathroom duct with the doors fitted with vents. These ducts would not be very efficient as they would need bends and sloping pipes if the bathroom duct is straight, but relatively low extract rates for the bedrooms may be acceptable provided the flow is continuous and guaranteed.

In a final system, it will be advisable, if not mandatory, to fit devices to shut sections of the ducts in case of fire.

7. CONCLUSIONS

It has been demonstrated that a simple duct system will at least re-route some of the ventilation air in a house; if the extract points are situated in the kitchen and bathroom and the inlet points are in, say, the lounge, this re-routing will reduce moisture levels in the house.

If the house has a low natural air infiltration rate then the system should provide an extraction rate in the region of 0.5 to 1 ac/hr throughout the heating season after suitable modifications.

ACKNOWLEDGEMENTS

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3rd AIC Conference, September 20-23 1982, London, UK

PAPER 10

THE VENTILATION PERFORMANCE OF HOUSES - A CASE STUDY

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In an energy efficient house with high insulation standards, the ventilation system must be designed. The main design criteria is that the ventilation rates be maintained close to the required minimum level throughout the heating season. Yet because so little understanding was required and achieved about the performance of natural ventilation in traditional houses, it was, in the mid-1970's, difficult to know where to start from, to design for natural ventilation in energy efficient houses. The conventional wisdom at the time was that low infiltration rates were achieved by using high performance (energy efficient) components and sealing up as much as possible of the gaps and cracks in the outer wall, and then to rely on the remaining gaps to provide the minimum ventilation level by infiltration. Not surprisingly, in many cases, this method did not work, stuffiness, stagnation and condensation occurred and the initial low energy houses did not achieve their 'ventilation' aspirations. From the results of these early houses, it was argued that designing a ventilation system for a naturally ventilated energy efficient house was not simply a matter of designing methods for reducing the whole-house ventilation rate, say from 4 ac/h to .5 ac/h, by applying a 'blanket' of sealing measures. Adequate ventilation is an essential component of the internal physical environment and a minimum standard must be maintained, not only in the house as a whole but distributed in a way appropriate to the requirements of individual spaces. The ventilation performance of a house therefore is not only determined by the distribution of cracks and the level of window opening, it is also fundamentally a product of the spatial layout. In order therefore to arrive at a design methodology for natural ventilation, air flow patterns within the house envelope must be measured and understood.

For this reason, ventilation studies were included in, and formed an important part of the Abertridwr Project^{1,2,3,4}, the main objective of which was to assess the cost benefits of additional levels of insulation. To this end two groups of houses were used, 19 insulated to Part F of the Building Regulations 1975, and 20 insulated to what is now Part F, 1982. In the design of these houses, the ventilation was achieved in the traditional manner by choosing good quality components and draught-stripping the doors and windows. The houses were therefore ideally suited for a case study on the ventilation performance of highly insulated energy efficient houses.

This paper briefly describes the Abertridwr Project as a whole. It then describes the sequence of ventilation experiments within the project carried out over a period of four and a half years and summarizes the major results.

It describes how as a result of the first stage of ventilation experiments, modifications were made to a number of houses, in that 'trickle' ventilators were installed. The results of these modifications are next discussed.

Finally, conclusions are drawn, and suggestions made concerning a methodology for the design of natural ventilation systems in low energy houses.

2. DESCRIPTION OF ABERTRIDWR PROJECT

The project involved the physical and social monitoring of thirty nine occupied houses sited in Abertridwr, South Wales.

The houses are three-bedroomed, terraced (Fig.1), owned by the Housing Association, and are heated by gas fired, low-pressure central heating systems.^{1,2,3,4}

The project had two main objectives:

1. To assess the effects of better insulation standards in houses together with appropriately smaller heating systems in providing comfort conditions at lower energy costs.*
2. To measure the operating efficiencies of domestic gas-fired central heating systems.**

To achieve these objectives, twenty houses (Better Insulated [BI] houses= were designed with higher standards of insulation compared with the remaining nineteen houses (the control houses) in the form of:-

- (a) Dry lining wall insulation (a method considered appropriate to the exposed conditions)
- (b) Double thickness of roof insulation
- (c) A border slab of floor insulation embedded in the concrete base.

The overall steady state fabric heat loss of the test houses is reduced by approximately 30% compared to the control houses.

In addition, the heating system in the test houses is appropriately smaller, employing an 8.3 kw boiler compared with the 14.7 kw boiler of the control houses. The number of the radiators was also matched to the reduced fabric losses in the BI houses, there being originally no radiators in the bedrooms. However, the controls are the same in both BI and control houses. A programmable clock allows two 'on' periods per day. There is a room thermostat in the living room which controls the boiler and pump. All other radiators have thermostatically controlled radiator valves fitted. The heating system is also used, in both groups, to provide domestic hot water.

* Sponsored by Department of Environment, Housing Development Directorate

** Sponsored by Science and Engineering Research Council

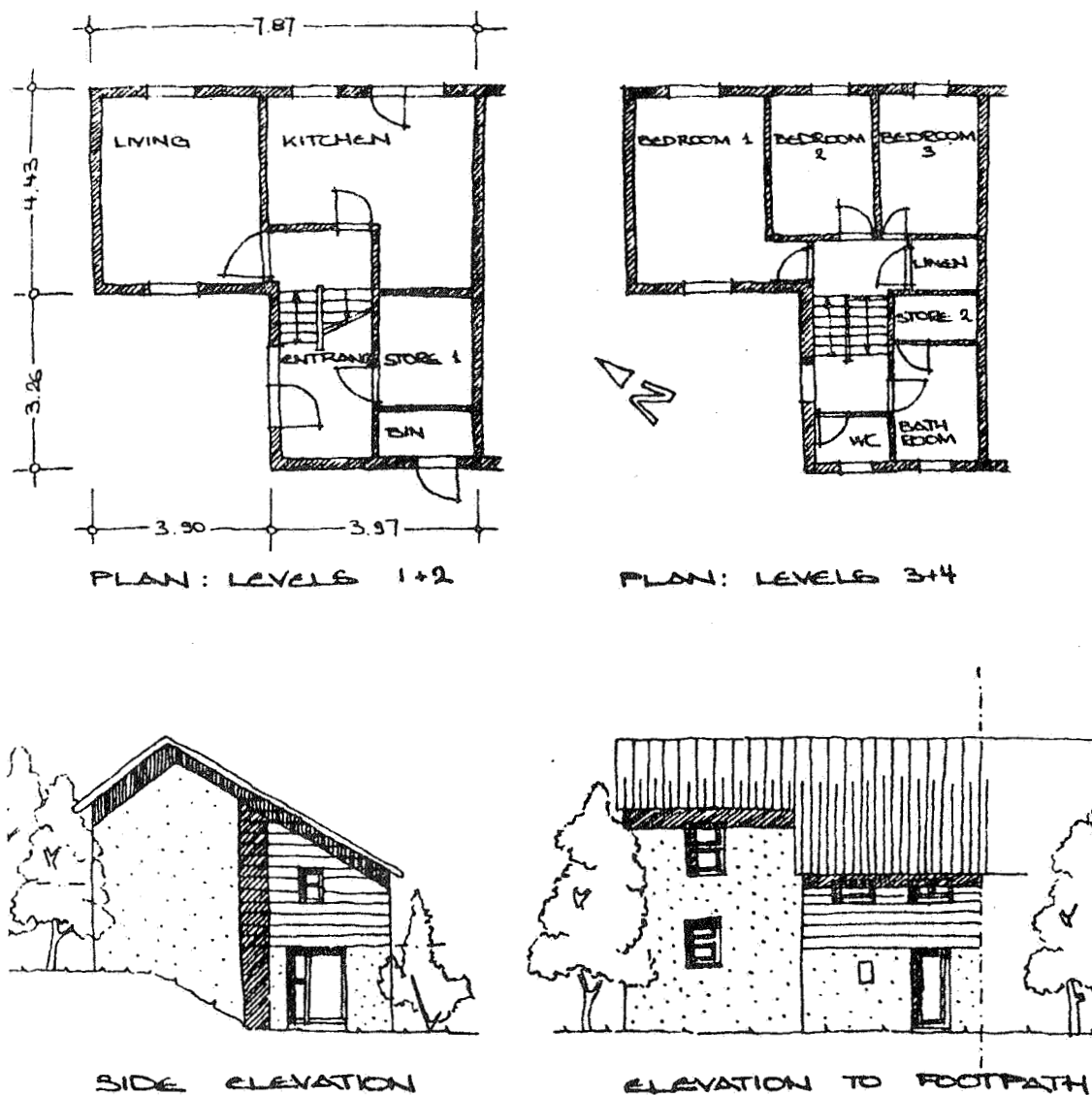


FIGURE 1 ABERTRIDWR HOUSE - PLAN AND ELEVATION

Finally, traditional draught-stripping measures have been employed in both BI and control houses to reduce ventilation losses through infiltration.

Physical data has been collected by means of the continuous monitoring of energy inputs, internal air temperatures, external climate and boiler efficiency parameters. Values are stored on a five-minute time scale though the measurement associated with

boiler efficiency are measured every twelve seconds and then averaged over the five minute period. In the order of 800 measurements are stored every five minutes.

Social data is collected by means of interviews with the occupants concerning the daily patterns of occupation and 'energy use' of the houses. In addition, information has been collected concerning comfort conditions, knowledge of the control system, previous heating experience, condensation problems and faults with the heating system.

Additional information has been provided by:

- (a) Measurements of ventilation rates
- (b) Thermography surveys*
- (c) Experiments in unoccupied houses

One BI house (the test house) has been occupied by the project team for the duration of the project and this house has been used to carry out 'controlled' experiments.

The project was in full operation from September 1980 (although monitoring and experiments had proceeded on a reduced sample since 1978) and ended in September 1981. (A more detailed description of the project and initial results are reported elsewhere^{1,2,3,4}).

Since September 1981 the Abertridwr houses, together with the monitoring system, have been used for two further projects. One involving a study of incidental gains in houses** and the other demonstrating the performance of trickle ventilators***. The later project is described in Section 5 of this paper.

3. VENTILATION EXPERIMENTS

3.1 Measurement Programme

The first objective of the ventilation measurements programme, carried out under the Abertridwr Project, was to determine the ventilation characteristics of the Abertridwr Houses.

As our understanding of the ventilation performance of the houses developed over the four and a half years and as modifications were carried out to the ventilation system, the objectives of the experiments changed in their emphasis.

-
- * Thermography surveys were carried out by British Gas
 - ** An Investigation into the Effective Use of Incidental Heat Gains in Dwellings (funded by SERC)
 - *** Monitoring the Performance of Trickle Ventilators (funded by BRECSU)

In measuring the ventilation characteristics of the houses the following experimental techniques have been used by the organisations concerned (Building Research Establishment [BRE], British Gas [BG], and UWIST).

To determine the whole-house and room infiltration rates, Tracer Decay methods were used by BRE, while BG used their Autovent technique together with a limited number of Tracer Decay Tests.

Conventional pressurization techniques have been used to determine the leakage characteristics of the houses. In addition a method developed by BG was used (by BG) for determining the area leakage distribution⁵.

The first experiments were carried out by BRE in March 1978 and their objectives were to assess the general ventilation characteristics of the houses. They performed Pressurization measurements in eight houses, four BI houses and four control, and extensive Tracer Decay measurements in one of the BI houses (house 2).

In January 1979 BRE performed further leakage tests on three of the houses that had been tested the previous year, the objectives being to obtain a measure of any increases in leakage over the first year of occupancy.

British Gas carried out ventilation experiments in February and March 1980. They included twelve houses in their programme. In these twelve houses they performed Pressurization tests and a small number of Tracer Decay tests. In one of the BI houses (the test house - house 1) they performed extensive Tracer Decay tests using the Autovent technique.⁵ The objective of this programme of experiments was to give a further insight into the ventilation characteristics of the houses covering a larger sample than had previously been used, and again to measure any change in leakage with age. In particular, the Autovent technique enables the ventilation patterns of a house to be investigated.

As a result of this first series of experiments trickle vents were installed in the test house and 2 other houses in order to try and produce an improved ventilation distribution pattern, and in an attempt to alleviate the problems of condensation.

In February 1981 British Gas carried out ventilation and leakage distribution measurements on the test house in order to determine what differences the trickle ventilators made to the house ventilation pattern.

In July 1982 UWIST carried out air leakage tests on nine of eighteen houses which had had trickle ventilators installed during February, 1982, to measure the air leakage of the houses with trickle ventilation. This last set of measurements formed part of a new project, demonstrating the performance of trickle ventilation.

Table 3.1 summarizes the programme of ventilation experiments performed in the Abertridwr houses.

TABLE 3.1 SUMMARY OF EXPERIMENTAL PROGRAMME

Date	Organization*	No. of Houses	
		Infil.	Leakage
March 78	BRE	1	8
Jan. 79	BRE		3
Feb/Mar. 80	BG	12 (1 Autovent)	12
Feb. 81	BG	1 (Autovent)	1
July 82	UWIST		9

*Measurements were performed by Building Research Establishment [BRE], British Gas [BG], and UWIST.

3.2 Results

(a) Air Leakage Measurements

Four whole-house air leakage surveys were carried out over a four and a half year period. The results are summarized in Table 3.2.

TABLE 3.2 SUMMARY OF WHOLE HOUSE AIR LEAKAGE TESTS (m^3/s)

Where whole house infiltration rates were available they have been included in brackets (ac/h). Infiltration rates for houses, other than house 1 in 1980 are each from a single Tracer Decay test.

House		Mar 78 @ 50pa	Jan 79 @ 50pa	Feb/Mar 80 @ 20pa @ 50pa		July 82 (vents closed) @ 20pa @ 50pa	
B.I.	1	0.45	0.95	0.52	0.97(0.48)	0.55	0.91
	2	0.44(0.49)		0.34	0.64(0.53)	0.48	0.79
	3	0.57	1.0	0.54	0.93(0.40)	0.52	0.99
	4	0.54	0.9	0.43	0.80(0.36)	0.44	0.79
	5					0.47	0.81
	6					0.5	0.88
	7			0.33	0.59(0.50)		
	8			0.35	0.60(0.34)		
	9			0.32	0.57(0.44)		
	10			0.36	0.65(0.44)		
	11			0.45	0.80(0.49)		
	12			0.44	0.75(0.55)		
CONTROL	13					0.38	0.78
	14					0.36	0.71
	15	0.37		0.46	0.78		
	16	0.40					
	17	0.40		0.39	0.68(0.51)		
	18	0.37					
	19					0.36	0.69
	20					0.37	0.71

The 1978 BRE survey was carried out shortly after the houses were constructed. The results show that the control houses are on average (21%) 'tighter' than the B.I. houses. This was attributed to the dry-lining construction of the B.I. houses.

Further measurements by BRE on three of the houses in 1979 showed a dramatic increase in whole-house leakage during the first year of occupation. This was probably caused by the drying out of timbers etc., and warping of doors and windows.

Leakage measurements taken over the following three years show surprisingly little change in levels.

Room leakage measurements were carried out by BG in 1980 on house 1. (The results are presented in Figure 4.1). As each room was measured the rest of the house was maintained at the same pressure, so that the results represent only the leakage to the outside. The landing, hall and bathroom exhibit the largest leakages. This is not surprising if the results of 1978 BRE component leakage measurements of house 2 are considered (Fig.3.1)

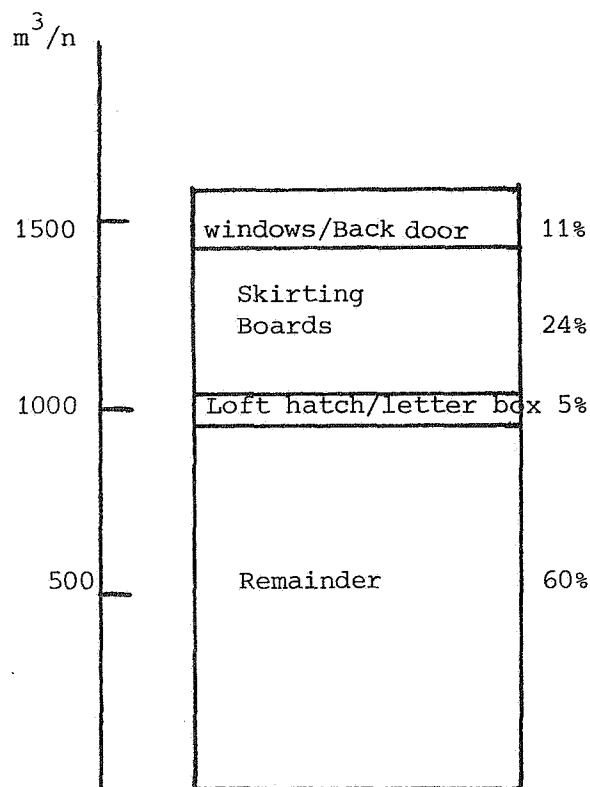


FIG. 3.1 Distribution of Air Leakage in house 2 (@ 50pa)

The majority of leakage was attributed to service entry, and general cracks. The bathroom has a large amount of pipework entering. The hall and landing have a relatively large amount of external wall area plus the front door which, from measurements made, has the largest single component crack area. Observation

of air flow patterns, using smoke detection, have identified the predominant air flow route to be in through the hall and sometimes the bathroom and out through the bathroom, landing area and bedrooms.

Although the BG leakage tests were performed on different houses it is, from experience with the house construction, believed that the major components of leakage will be the same.

(b) Air Infiltration Measurements

The results for average infiltration rates, from the 1978 and 1980 measurements are presented in Table 3.3 for two houses (houses 1 and 2), in which extensive measurements were carried out. Values of whole house infiltration rate for ten other houses have been included (bracketted) in Table 3.2, although they were calculated each from a single decay test and will therefore not be as accurate.

TABLE 3.3 AVERAGE INFILTRATION RATES OF TWO HOUSES

	House 2 BRE (1978)			House 1 BG (1980)		
	R h^{-1}	U m/s	$\sqrt{T}$ $^{\circ}K^{\frac{1}{2}}$	R h^{-1}	U m/s	$\sqrt{T}$ $^{\circ}K^{\frac{1}{2}}$
Minimum	0.33	1.2	3.04	0.17	0.3	3.0
Maximum	1.27	8.1	4.32	0.68	5.5	4.79
Mean	0.49	3.6	3.69	0.48	1.7	3.97

R = infiltration rate U = wind speed T = Inside/Outside temp.

It can be observed from the two tables that the average infiltration rate, over a range of external conditions, is in the order of $\frac{1}{2}$ ac/hr.

From the extensive measurements performed on houses 1 and 2, the dependance on external conditions have been analysed. The stack effect was found to dominate in determining the air infiltration for all wind directions except when the wind was on the front of the house. Figure 3.2 shows the relation between infiltration rate and stack effect for all wind directions except, when the wind was on the front of the houses. Figure 3.3 shows the effect of wind speed for a constant stack effect. The graphs include the data points from the 1980 BG survey, while the results from the 1978 BRE survey are represented by regression lines.

From the 1978 survey the infiltration rate was found to be highest for periods when the wind was on the front of the house. An average infiltration rate of 0.67 ac/h was calculated for an average wind speed of 4 m/s. However, most of the time the wind was either parallel to, or on the rear of, the house and for these cases the average infiltration rate was calculated to be 0.4 ac/h for an average inside/outside temperature difference of 13.6°C. The average infiltration rate for all wind directions

was calculated to be 0.49 ac/h. This was considered to be low at the time compared to other modern British houses.

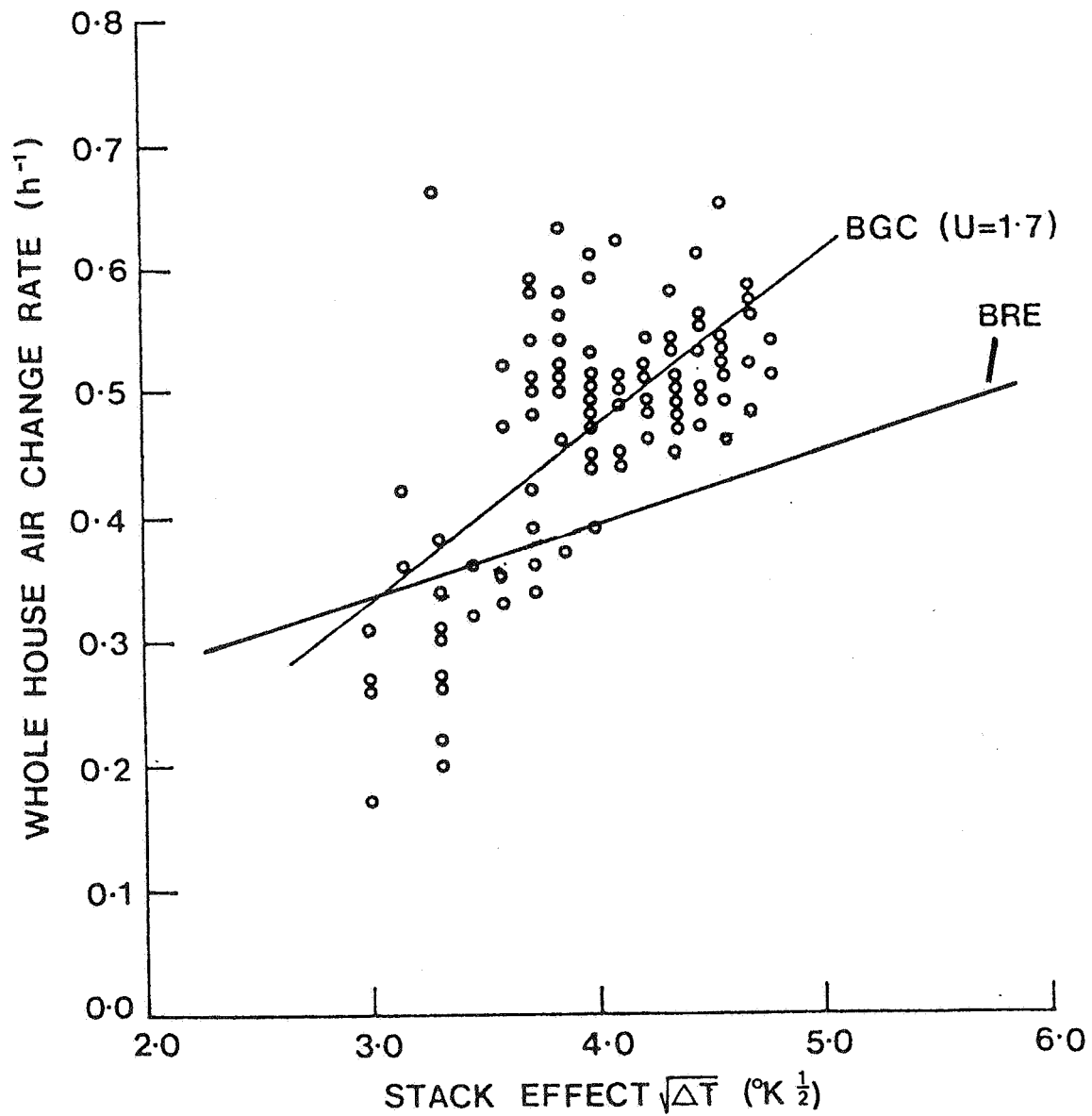


FIG. 3.2 Whole House Ventilation Dependence on Stack Effect.
Wind Direction:- All but 179-229 Degrees.
All Wind Speeds.

Wind Direction ϕ

o 91°

+ 114°

x 179° - 229°

$\sqrt{\Delta T} = 4.3$ (3.3 for 179° - 229°)

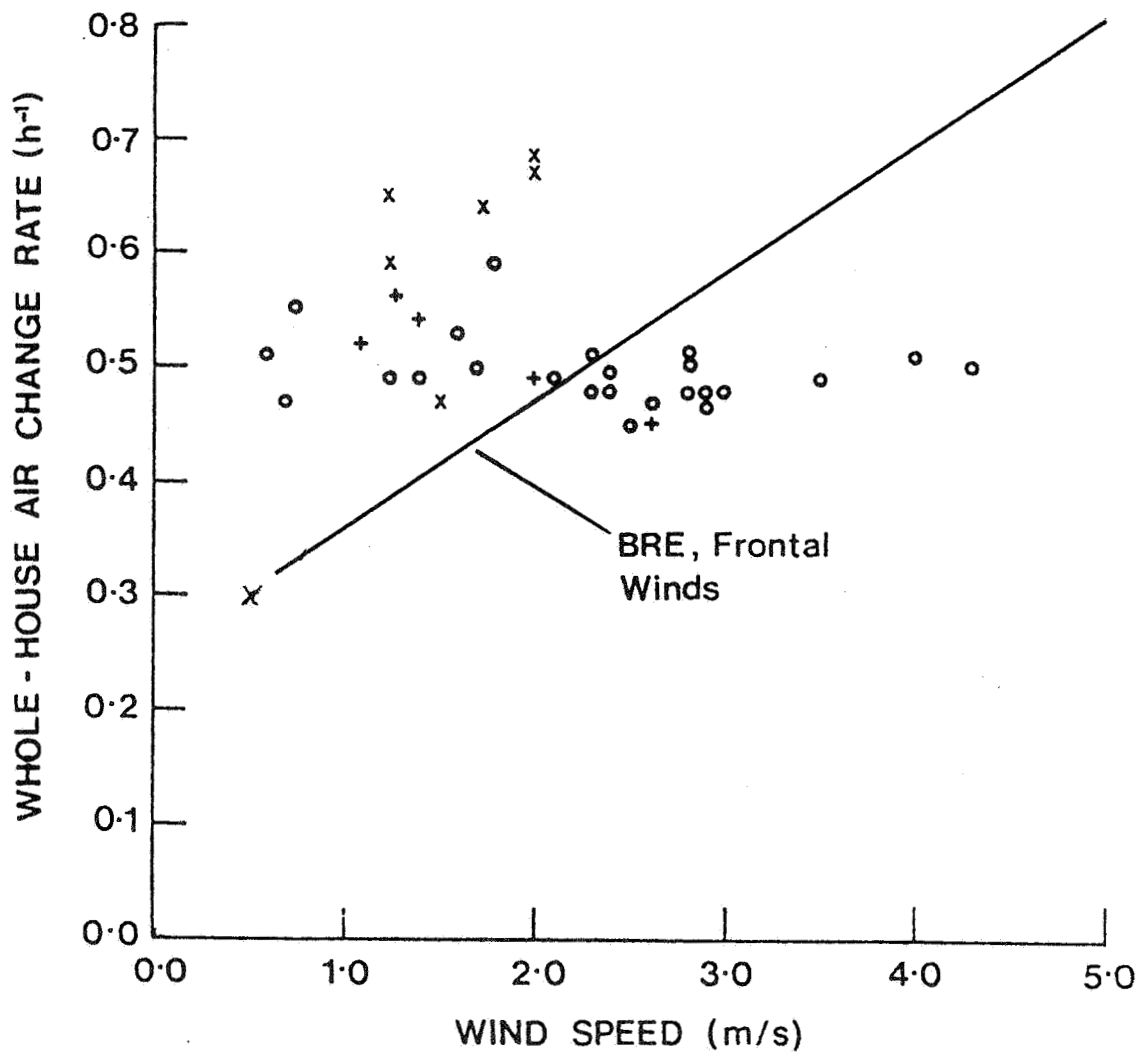


FIG. 3.3 Whole House Ventilation: Effect of Wind Speed For Constant Stack Effect.

The differences between the infiltration rates in houses 1 and 2 were not as significant as the differences in their leakage. The infiltration is dependant on the distribution of leakage with regard to the external pressure distribution. Although the major components of leakage are likely to be the same in both houses, their relationship to the pressure distribution on the building shell is likely to be different. This factor, together with the different wind conditions during each test, and bearing in mind the difference in measurement techniques was considered the major reason for the lack of correlation between the leakage and infiltration results.

Room ventilation measurements were carried out in the 1978 and 1980 surveys. The results are presented in Table 3.4.

TABLE 3.4 ROOM FLOW RATES*

Survey	Average Room Air Change Rates (internal doors closed)(House 2)								
	Living Room	Kitchen	Landing	Hall	Bath	W.C.	B1	B2	B3
BRE 1978	0.27	0.59		1.29	2.3	1.6	0.29	0.45	0.27
	Average Room Ventilation Rates (internal doors open)(House 1)								
BG 1980	0.23	0.39	0.2	2.05	2.27	0.17	0.08	0.07	0.05

The results which are again for different houses, are for two methods of measurement - the 1978 BRE survey measuring air change rates and the 1980 BG survey measuring fresh air entry. However, the methods do complement one another. The results from the 1980 survey show the hall and bathroom as having the largest fresh air entry, while the bedrooms have very low amounts of fresh air entry. The 1978 survey shows the hall and bathroom having large air change rates with the bedroom having a reasonable air change rate. The two surveys together again identify the dominance of the stack effect in determining the ventilation characteristics, with fresh air entry by the hall and bathroom (probably via the bathroom floor where there are large service gaps) and leaving, to some extent via the bedrooms, where the air change rates are noticeably higher than the ventilation rates.

During the 1980 survey the effect of window opening, and the open/closed status of internal doors, on ventilation rate, was measured. A very limited number of measurements were carried out for various combinations of window and internal door opening patterns. The results are summarised in General terms in Table 3.5.

* The ventilation rate is the amount of fresh air entering the room from outside, whereas the air change rate is the total air change of the room including air entry from other rooms.

TABLE 3.5 EFFECT ON VENTILATION OF OPENING WINDOWS
AND INTERNAL DOORS

COMBINATION	EFFECT ON VENTILATION RATE
Window opening	
At same level on one wall	Infiltration rate X 2
At different levels on one wall	" " X 3
On both walls	" " X 4
All internal doors closed	Ventilation rate X 0.6
Half internal doors closed	" " X 0.75

Because of the limited nature of the measurements there are obviously large uncertainties associated with the values quoted. However, they do contribute to the general understanding of the ventilation performance of the houses.

3.3 Discussion

The main results from the first series of ventilation measurements (1978 - 1980) can be summarized as follows:

It was observed that there was a marked increase in leakage rates during the first year of occupation. Since then there has been no comparable increases, and the leakage seems to have levelled out.

The average whole house infiltration rate of the houses was found to be about 0.5 ac/hr. This value is probably as low as can be achieved with traditional building techniques. However, though the whole house average infiltration rate can be considered satisfactory, in terms of satisfying ventilation requirements and being 'low' in energy terms, the individual room ventilation rates are often extremely low. The main air flow pattern has been identified as air coming in through the hall, going up the stairs, and out through the landing area, with the bathroom acting as an entry and/or exit according to outside conditions. The ventilation of the living room and bedrooms is often low with very little fresh air intake. This is an important observation, for three reasons. (i) Low infiltration rates could encourage occupants to open windows and thereby 'overventilate'. (ii) Two bedrooms in the BI houses depend on convective gains from the rest of the house as their major heat source. (iii) There have been a number of cases of serious condensation reported.

4. VENTILATION USING TRICKLE VENTILATORS

The initial series of measurements identified the need to improve the pattern of ventilation and at the same time reduce the risk of condensation. As a possible solution it was decided to test the performance of 'trickle ventilators' in three houses. Fine tune slot ventilators were fitted to the frames of all the windows in one BI house, the test house, (which was to undergo extensive tests), and in selected areas in two other BI houses where condensation was considered to be a serious problem, i.e. mould growth.

The main objective of this experiment was to improve the distribution of ventilation in the houses, increasing the level of ventilation to the bedrooms and living rooms and decreasing the levels in the Hall/landing and bathroom while maintaining the whole house ventilation rate the same. It was decided to use temporary sealing measures to produce the required decreases in ventilation. The measurement programme was carried out in February 1981.

(a) Leakage Measurements

Leakage measurements performed on the test houses before the application of sealing, measures showed an increase of about 10% in leakage compared to the 1980 survey. Sealing measures in the hall, bathroom and W.C. produced a 20% reduction in whole house leakage compared to the 1980 survey.

Measurements performed with the trickle ventilators open and then closed, revealed only a 3% difference in leakage.

Measurements of room leakage are presented in Figure 4.1, and compared with the measurements from the 1980 survey. The figure illustrates the redistribution of air leakage as a result of the installation of ventilation and sealing measures.

(b) Infiltration Measurements

The whole house infiltration rate was measured to be 0.54 ac/h for mean wind speeds of 1.63 m/s and a mean stack temperature difference of 18.6°C.

Measurements of individual room ventilation rates are given in Table 4.2 and compared to the measurements from the 1980 survey. They are averaged for vents open and closed and show the effect of the sealing measures. The overall patterns are different with decreases in the hall and bathroom rates. The bedrooms are not significantly changed although measurements of air change rate show that for bedroom 1 with vents open, although the ventilation

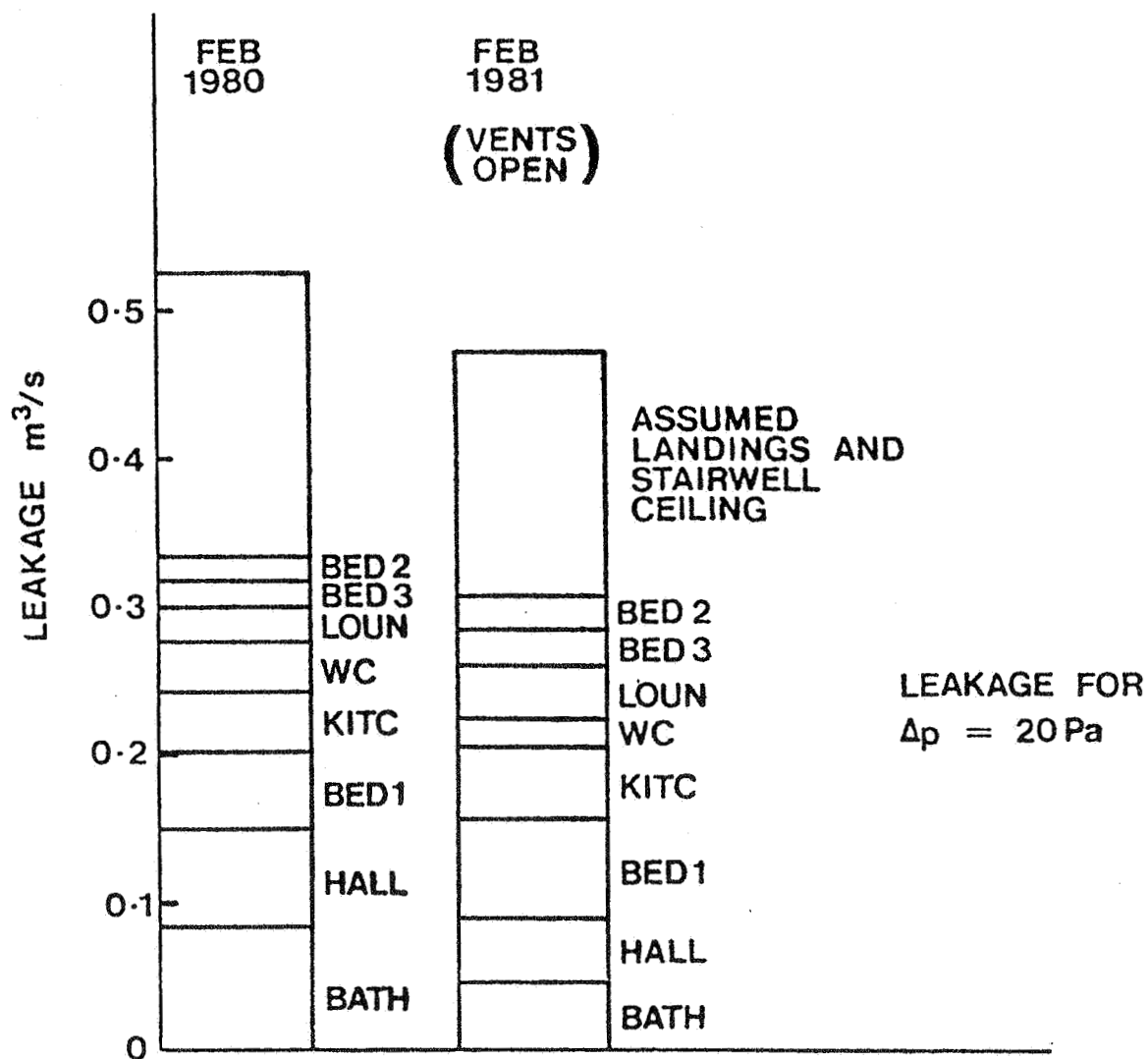


FIG. 4.1 Leakage Distributions

rate is still low the air change rate is relatively high (0.9ac/h compared with 0.29 ac/h in house 2/1978 (Table 3.4).

TABLE 4.2 ROOM VENTILATION MEASUREMENTS

	Living Room	Kitchen	Landing	Hall	Bath	W.C.	B1	B2	B3
1980	0.23	0.29	0.2	2.05	2.27	0.17	0.08	0.07	0.05
1981 (vents open & closed)	0.32	1.35	0.21	1.58	0.48	0.16	0.05	0.01	0.02
<u>Vents Open</u>									
Ventilation rate	0.04								
Air change rate	0.9								

(c) Discussion

To a certain extent the desired changes in the distribution of ventilation have been achieved, with the vents, when open, being able to draw air into the rooms that previously were poorly ventilated. The bedrooms now are able to achieve a higher air change rate. This improves the ventilation of these rooms and also improves the convective heat gains from downstairs.

The combination of ventilation and sealing measures have resulted in a redistribution of open area. There has been an upward shift in neutral plane with respect to the stack effect. The kitchen has now become a major source of air entry. This could cause problems in so far as the kitchen is also a source of moisture, and being in the path of a major ventilation route, the moisture would be more easily transferred to cooler spaces where condensation might occur. In hindsight, sealing the hall might have been inappropriate. The hall is rather suitable as a major air entry point. It is not normally occupied, has no moisture producing activities, and can act as a buffer zone and distribution path for the occupied rooms.

The remaining two houses which had had trickle ventilators installed showed a marked reduction in condensation levels.

5. DEMONSTRATING THE PERFORMANCE OF TRICKLE VENTILATORS

As a result of these initial tests the Abertridwr site was chosen to host a demonstration project on the performance of trickle ventilators.

Eighteen houses had trickle ventilators installed in all window frames during February 1982. There were three objectives to the experiment:

1. To assess the degree to which they were successful in reducing condensation levels.
2. To assess any energy 'penalty' due to any increase in infiltration rates.
3. To observe any change in window opening, and any energy savings resulting from reduced window opening.

Data on energy consumptions for each house was available for the before (October 1981 - February 1982) and after (March 1982 - May 1982), the installation of vents, periods. Window opening surveys were carried out during the before and after periods. Social data was collected, on use of windows/vents and occurrence of condensation, throughout the October 1981 - May 1982 period. Measurements of air leakage were performed on nine houses in July 1982.

5.1 Air Leakage

In order to assess the likely change in ventilation characteristics due to the trickle vents, air leakage measurements were performed on ten houses. The results are given in Table 4.1.

TABLE 4.1

House	Vents Closed		Vents Open	
	@ 20pa	@ 50pa	@20pa	@ 50pa
1	0.55	0.91	0.58(+5.5)	1.12(+21.2)
2	0.46	0.79	0.48(+4.3)	0.88(+11.4)
4	0.44	0.79	0.46(+4.5)	0.81(+2.5)
5	0.47	0.81	0.50(+6.3)	0.87(+7.4)
6	0.50	0.88	0.54(+8.0)	0.94(+6.8)
13	0.39	0.78	0.43(+10.3)	0.83(+6.4)
14	0.38	0.70	0.44(+15.8)	0.73(+4.3)
19	0.36	0.69	0.41(+13.9)	0.77(+11.6)
20	0.37	0.71	0.39(+5.4)	0.78(+9.9)
Average increase			+7.1%	+9.1%

The increase in air leakage due to opening the vents is low, an average of 7.1% @20pa and 9.1% @ 50pa. The values of leakage with the vents closed compare favourably with those of previous measurement programmes (see Table 3.5).

In view of these measurements it is unlikely that the ventilation rates would be increased by more than the order of 10% by opening all vents.

5.2 Energy Consumption

Energy consumption for all 39 houses were compared for the before installation of vents and after periods. Figure 5.1 shows the monthly normalized residuals for energy consumption for one house (house 14 which has the highest measured increased air leakage). There is no evidence of any change in energy consumption due to the ventilation. There was similarly no significant change in the energy consumption identified in any of the houses, which could be attributed to the installation of the vents.

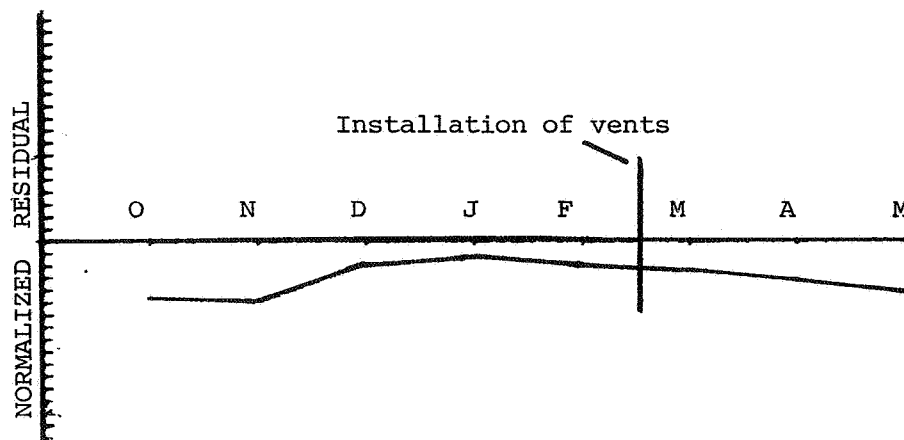


FIG. 5.1 Monthly normalized residuals of energy consumption for space heating for house 14.

5.3 Window Opening

Information on window opening was collected in two ways.

1. Occupants were interviewed about their window opening habits.
2. Window opening surveys were carried out by direct observation.

Interviews suggest that where there has been modifications to window opening because of the presence of ventilators it had mainly been during evenings and the night. Eleven out of 17 residents say they had, on occasions, reduced window opening because of the ventilators. As window opening habits are acquired through past experience one would expect change to be gradual. We found some evidence that there is a learning process as occupants, through experience, come to realise that the ventilators provide a viable alternative to window opening. This evidence is so far anecdotal. For example, residents referring to recent experience of using the ventilators as a substitute to window opening and expressing surprise at their effectiveness.

Occupants were pleased with the ventilators. They had rarely used them to 'fine tune' the ventilation. The usual pattern was to leave them permanently open. Occasional exceptions to this were the use of the living room and main bedroom (both of which are cross ventilated). The living room vents were more likely to

be adjusted and more likely to be closed.

Existing ventilation habits appear to be changing, albeit slowly, with the introduction of the trickle ventilators.

The window opening surveys carried out for 4 weeks before and 4 weeks after installation have shown no clear pattern of change. However, these surveys were only carried out during the daytime. It was considered inappropriate to conduct the survey during evenings and night time as it might have been considered an infringement on the occupants privacy.

5.4 Reduction in Condensation Levels

The most positive result to come from the experiment so far is the reduction in condensation levels.

A survey was carried out on seventeen of the houses which had had trickle vents installed in order to determine the effect of trickle ventilators on condensation levels. The occupants from twelve of the houses reported that the vents had helped reduce condensation problems. Occupants from remaining five houses, reported that they had never had condensation problems to begin with.

5.5 Conclusions

- (i) There is no detectable space heating energy penalty as a result of installing trickle vents. This implies that any increases in infiltration rates due to the vents (estimated at less than 10%) is small in energy terms.
- (ii) There is evidence that condensation levels have been reduced.
- (iii) There is no positive reduction in window opening as yet identified, however, there is some evidence of a learning process. Not surprisingly, in view of the previous findings, there is no evidence of any energy savings yet.
- (iv) The 'open area' associated with trickle ventilation although large enough to alleviate condensation problems, is sufficiently small so as not to incur a significant energy penalty.

6. DESIGN METHODOLOGY

A successful methodology for the design of a naturally ventilated house would:

- (a) Keep the whole house ventilation rate down to the minimum required.
- (b) Ensure minimum levels appropriate to each space.

Such a system would require:

- (i) Suitable air entry and exit points for each space.
- (ii) Suitable interconnection between space for the necessary air flow patterns to be established.
- (iii) That the combination of (i) and (ii) would provide a ventilation system which could operate effectively under the varying conditions of stack and wind.

The proposed design methodology is as follows:

Firstly, infiltration should be reduced by a generally 'tight' construction.

Secondly, open areas should be re-introduced to the external walls of the house by means of trickle ventilators.

Thirdly, the required air distribution should be achieved, according to the spatial layout, by either:

- (i) using the stairwell as the main distribution duct (as at Abertridwr)
- (ii) if the internal design does not allow (i), then to introduce ducting to obtain the necessary (possibly fan assisted) air distribution.

If the resulting distribution pattern includes the kitchen and/or bathroom as air entry points, extraction fans can be used to get rid of moisture at source. The risk of condensation in other spaces is alleviated by the use of trickle ventilation.

It is not envisaged that desired ventilation patterns will determine the design of the spatial layout. In energy efficient design the primary determinant of spatial layout will be the 'sensible' use of heat. Once the spatial layout has thus been determined the ventilation system can be designed around it in the way specified above.

Acknowledgements

Work carried out by the Building Research Establishment and by the British Gas Councils Research and Development section is gratefully acknowledged.

Figure 3.1 is taken from the BRE report entitled, 'Measurements

of air infiltration and leakage rates in well insulated houses, Abertridwr' by B.C. Webb and L.M. Parkins, which is contained in reference 3.

Figures 3.2 and 3.3 are taken from the British Gas Report entitled 'Ventilation measurements by British Gas in Abertridwr test houses' by D.K. Alexander, D.W. Etheridge and R. Gale, which is also contained in reference 3.

Figure 4.1 is taken from the British Gas Report entitled, 'Ventilation measurements in an Abertridwr house after modification' by D.W. Etheridge, which is contained in reference 4.

The views expressed in this paper are those of the authors alone and do not necessarily reflect the views of other contributors to the project.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 11

ENERGY IMPROVEMENT KITS:- FIELD RESULTS

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SYNOPSIS

25 different but representative dwellings from the Birmingham Local Authority housing stock have been monitored before and after energy retrofitting. All dwellings received measures aimed at reducing air infiltration.

The paper presents the findings of these field trials as a set of case studies encompassing both fabric and infiltration measures. In addition the infiltration measures are commented upon in more detail. The results are as much, if not more, pragmatic than scientific and emphasise the importance of technology transfer, i.e. implementation. The diverse nature of U.K. housing (especially local authority stock), user influences, and relatively low quality of current U.K. building practice all combine to markedly affect prediction of costs/benefits.

E.I.K. evidence suggests that simple E.C.Ms. are successful but that more sophisticated combinations tend to under-achieve. Where it is possible to identify ventilation changes in several cases this is impaired by increased use of windows and/or permanent ventilation. Nevertheless reduction of between 0.3 and 0.6 ac/hr. appear typical where such effects are not operating.

I. INTRODUCTION

The E.I.K. Project - a joint venture by The Birmingham City Housing Dept. and The Dept. of the Environment ; has recently been completed (1) (2) (3). Primarily concerned with the thermal upgrading of existing local authority housing stock (which accounts for around 10% of all U.K. energy consumption) the project was carried out in 2 phases:-

- a random 4% survey of the existing Birmingham stock
- monitored field trials (before and after) of energy retrofits to 25 representative dwellings.

The decision to include field trials in the E.I.K. project reflected the objectives and methodology of the Better Insulated Homes Programme of the Department of the Environment which commenced in 1974. Within its main objectives (4) important secondary aims were recognised, e.g. to 'explore' construction feedback, to assess real energy savings, etc.

The survey data was subjected to thermal analysis and eventually the 25 dwellings used in the field trials were selected (on criteria of associated technical and occupancy preferences).

Table I illustrates the dwellings together with the energy conservation measures. They are assigned an 'E.I.K. Number' (1 to 25) which is used throughout this paper.

Monitoring had 4 principal aspects:-

Energy use. Fortnightly reading of separate meters to heating, hot water, lighting and power

Temperature. 4 to 6 room temperatures were measured at half-hour intervals.

Behavioural aspects. The occupants have been interviewed to identify household characteristics, and use of appliances/spaces/windows.

Practicalities. For the exercise to be of direct benefit to others then operational factors also require evaluation, e.g. design, construction, management, costs, savings, etc. Useful energy (fortnightly) is compared to temperatures differences to yield Empirical Design Heat Loss (EDHL) rates. These vary with time due to changes in ventilation rates. The variations and mean values, however, are a measure of the insulation standard of the building shell and comparisons of 'before' and 'after' (retrofitting) values facilitates an assessment of the performance of energy conservation measures (ECMS). Comparisons of EDHLs with PDHLs (Predicted Design Heat Loss rates) is a further aid to performance assessment (e.g. by identifying gross under-achievement of an ECM).

This method of analysis is similar to the proposed 'Energy Signatures' of Norlen (5) and the 'Overall Coefficients of Heat Transfer' of Sonderegger (6). The principal advantage is its applicability to data from differing time periods. Additionally it can be used to model standardised annual energy consumptions and savings and hence economic viability of ECMs (normalising for changes in internal temperature if desired).

2. ENERGY CONSERVATION MEASURES (ECMs)

2.1 General

A wide range of measures were considered (within an overall budget limit of £1,000 per dwelling - 1979) ranging from simple draught exclusion, roof insulation, etc. to comprehensive improvement packages involving solid wall insulation, heating system changes, etc. Strategically, improvements were seen to fall into three categories:-

First-aid. First line of defence work such as draught exclusion, system maintenance, roof and hot water tank insulation.

Optimum. A range of measures suited to individual houses and intended to represent the maximum that could be judged as cost-effective. This may include cavity wall insulation, additional roof insulation, more thorough draught exclusion, etc.

Rehabilitation. Older properties will at some stage be due for renovation and it is more cost-effective to include energy conservation measures during such work when only the marginal or extra cost is entered onto the cost-benefit analysis.

This strategy, in response to detailed information about each dwelling e.g. condition, energy systems, defects, complaints, led to the E.C.Ms. in table 1.

2.2 Air Infiltration Reduction

A variety of measures were employed to reduce/eliminate the most obvious sources of air leakage.

- Windows were generally weatherstripped by the application of a silicone rubber sealant.
- Doors and loft hatches predominantly had nylon brush seals fixed to the jamb. Letter boxes received nylon brush closers.

(these two items were carried out by a specialist contractor - all other air infiltration measures were carried out by a general contractor).

The cost of these weatherstripping items ranged from £1.74 to £2.02 per metre run.

In general these measures proved relatively trouble free although several instances were noted of paintwork being 'pulled away' when the masking sellotape was removed. A few windows which did not have a rebated opening joint were difficult to retrofit and attempts to form a 'lip' from silicone rubber were aesthetically inadequate.

Brush weatherstripping of sash windows was troublesome and would be difficult to judge as adequate both on aesthetic or performance criteria. Indeed since it was installed in one instance after having 'freed' windows sealed by many coats of paint - the air leakage may have been increased. Occasionally brush weatherstripping of doors detached itself from the jamb due to inadequate fixing (short pins).

- Air brick - permanent ventilators in the external wall were generally 'redundant' by current standards and practice (and certainly contributed substantially to air leakage). These were replaced by conventional bricks. Where required for gas appliances adequate permanent ventilation in accordance with West Midlands Gas guidelines was either left or installed where previously omitted. The cost of this item was approx. £3 per brick and not surprisingly proved to be trouble free. The above items constituted a basic air infiltration ECM package which all dwellings received. Occasionally other items were implemented, e.g.
- unused flues were specified to be sealed. The intention was to seal it from room air whilst maintaining the void ventilated to outside. This current practice is an achievable preference with flues on external walls/but is not generally possible for fully internal flues which were simply boarded over in the room 'fire place' but with a large (approx. 0.075 m²) permanent ventilator. This, of course, would have a much reduced, if any, effect on air leakage via

the flue.

- suspended timber floors were covered with hardboard and caulk sealed to the skirting board. This measure proved relatively trouble free although in one instance differential movement caused upward bowing of the hardboard. The cost of this measure was £3 per m².
- Louvre windows were replaced (2 cases) by casement windows (at a cost of £83 and £109). The work was generally satisfactory.
- External porches were constructed in 2 cases - once where the main entrance opened directly onto the main living (and heated) space and once where an existing canopy and associated structure lent themselves to the addition of a fully enclosed porch. This work was completed at a cost of £655 and £511 respectively and generated no difficulty.

These and other ECM's are summarised in table II.

Although ϵ/α is a relative index of theoretical cost efficiency it should be modified by any knowledge of temperature differences across the ECM. For example the difference between the highest and lowest average temperatures recorded in the 'before' period are:-

	T(internal)	T(internal) - T(external)
warmest living room	23.0°C	17.4°C
coolest living room	15.0°C	9.3°C
warmest bed room	19.6°C	14.0°C
coolest bed room	10.5°C	5.0°C
warmest whole house	21.3°C	15.7°C
coolest whole house	12.6°C	7.1°C

These variations in $T_i - T_e$ introduce, approximately, a 2:1 variation on theoretical payback periods for measures to living rooms/whole houses and 3:1 for measures to bedrooms. This 'noise' can mask an otherwise 'clear' signal.

2.3 Practical Feedback

Before looking at some of the quantifiable 'benefits' arising from the ECMs it is worth noting some less measurable effects.

- A point which applied to all ECMs but in particular to air infiltration reduction measures concerned the nature of the original fabric. In simple terms it is not worth applying a

retrofit to a base component which is inappropriate in terms of design and/or condition to receive it. For example, several timber frame windows and doors were in such a condition that even the adaptable silicone rubber technique could not accommodate the large and irregular gaps around ill fitting and distorted openings.

- Several occupants were such profligate users of windows that measures to these had little effect and as is illustrated in Section 5. Of more concern than this continued user influence are those cases where it is believed that the potential benefit of reduced infiltration was mitigated by increased window use. There appear to be 2 principal reasons for this:-
 - a) an increase in condensation on window glass. This was particularly severe in 2 dwellings(6 and 10). In both cases the occupants had started to produce their domestic hot water via saucepans on the gas stove as a means of reducing their electric water heating bill. In short - an illustration of the complex interactions which can markedly influence energy consumption and conservation.
 - b) a reduced heat loss and unchanged heating system (in most dwellings room gas radiant/convector) occasionally led to high room temperatures with a related increase in window use patterns.

Both a) and b) produce the same effect - a by-passing of the energy saving from reduced infiltration.

- In one case (15) increased condensation not only promoted increased window opening but was of great concern occurring as it did on non-glass elements with consequent mould growth. There was no clear user influence. This case undoubtedly provides sustenance to those who argue that draught proofing of dwellings should not be carried out by local authorities.
- There were a few instances where ventilation rates appear to have been influenced by by-passes either remaining after or introduced by retrofitting. For example, permanent ventilators for the provision of combustion air, opening up of sealed flues).

3. IMPACT OF ECMs.

This section briefly summarises the measured effects of all the ECMs and section 4 attempts to extract information directly pertinent to the air infiltration measures. 'Benefits' are discussed under headings:-

- Envelope Heat Loss
- Temperature
- Fuel use
- Costs/Benefits

3.1 Envelope Heat Loss

Two forms of Design Heat Loss (DHL - $W/^{\circ}C$) are used:-

- P.D.H.L. Predicted D.H.L. -
(steady state assuming one airchange per hour before retrofitting and 0.75 ac/hr. afterwards).
- E.D.H.L. Empirical D.H.L.
(Derived from energy used to produce a temperature difference (one per fortnight)).

Figures 1, 2 and 3 illustrate the findings. Briefly these are:-

- since E.D.H.L. is a measure of the observed energy efficiency of a building envelope then changes in E.D.H.L. are a measure of the achieved increase in the envelopes resistance to heat loss. Since not all dwellings can be monitored most studies will be restricted to P.D.H.L. values and, consequently, it is of some interest to note the correspondence between Predicted and Empirical values. It is apparent that a consistent 'non-agreement' exists of the 'Before' and 'After' group. Taken as a sample of 50 this is not insignificant and the correlations are:-
- Before $E.D.H.L. = 0.76 P.D.H.L. + 37.3(W/^{\circ}C)$ ($r = 0.82$)
After $E.D.H.L. = 0.79 P.D.H.L. + 29.8(W/^{\circ}C)$ ($r = 0.81$)

The mismatch is not surprising in the context of the many inherent assumptions. The consistency is, however, re-assuring since it is relative rather than absolute values which are of greatest concern in a retrofit programme.

Of as much, if not more, importance than the regression equations is the observation that those cases deviating substantially from 1:1 agreement are those where defects etc. capable of explaining the errors were known to occur.

- mean values are:-

	Heat Loss $W/^{\circ}C$		
	Empirical	Predicted	E/P
Before	227	248	0.92
After	177	184	0.96
Change	50	64	0.78
(% change	22	26	0.85)

- the mean values suggest better agreement than the regression equations.
- agreement improves slightly as envelope heat loss is reduced.
- Although E/P at 0.92 and 0.96 may appear an acceptable average for design purposes (i. e. sizing of heating systems) an E/P of 0.78 for the 'savings effect' is perhaps more significant since this is the basis for estimating energy savings and hence payback periods etc. A figure of 0.78 is better than the 0.67 reported by Norlen (9).
Although 'reasonable' mean agreements and 'correlations' exist there is considerable scatter and the mean value is of little comfort to an individual case (e.g. dwelling 8).

3.2 Temperatures

Of interest is the way in which the distribution of internal temperatures is altered since this is indicative of the influence of control, use, cooling curve, etc. on the temperature condition of the house. In many cases it appears that a rise in average temperatures results primarily from increased temperatures during periods of non-heating (slower cooling). In this event such a rise cannot be prevented and it is idle to speculate as to whether it is fortuitous (lack of thermostatic control) or deliberate (occupants explicit objective).

Figure 4 indicates no significant correspondence between the rise in internal temperature and the change in envelope heat loss (E.D.H.L.) stimulating this rise. However, there are some trends, for example, there are a group of points significantly higher than the others and these can all be traced to specific causes, e.g. inadequate control on heating appliances (1 and 4), large proportion of largely uncontrolled free or casual heat gains (21, 25, 23), user preference (4), low 'before' temperatures (19) etc. Similarly although the average temperature rise for the whole group was 0.9°C it was only 0.5°C for the warmer thermostatically controlled centrally heated dwellings but 1.1°C for the rest (largely gas fires without thermostatic control).

A summary of mean temperature related data is:-

		Before	After	Change
(Te)	..	6.1	6.7	+ 0.6
(Ti) °C	..	16.5	17.4	+ 0.9
(Ti - Te) °C	..	10.4	10.7	+ 0.3
Savings offset (%)		-	-	10.4%

3.3 Fuel Use

It is difficult to summarise fuel use in such a diverse range of case studies but the following is indicative of the nature of energy usage. The figures refer to delivered energy per average fortnight in the heating season (for similar external conditions).

Purchased								
	Direct Space Heating		Indirect eg. Lighting, Cooking, DHW etc.		Free: People + Solar		Total	
	mean	range	mean	range	mean	range	mean	range
kwhrs	516	70- 998	281	104- 687	251	139- 376	1048	417- 1604
<u>Before</u>								
%	49	17- 66	27	14- 45	24	13- 45	100	
kwhrs	474	48- 871	251	97- 576	177	106- 285	903	346- 1396
<u>After</u>								
%	52	14- 66	28	15- 49	20	10- 44	100	

3.4 Costs v Benefits

Costs:- are taken as the capital cost of the ECMs based upon quotations made in 1980 but monies paid in 1981. In addition to the variability that differential inflation of costs and benefits introduces into the cost benefit ratios it is sometimes forgotten that there exists a considerable 'error' in building costs. This error is obviously self correcting in large samples but this may not be the case in these limited trials. Certain costs revealed themselves to be highly dwelling sensitive (e.g. double glazing, external wall insulation) yielding widely varying unit costs. Some items were charged on a measured basis (e.g. cavity fill) and in several instances the contractor's measurement did not exactly concur with a measurement of the area to be treated. It is the judgement of the City Building Finance Department (who administered the contracts) that the building costs charged were 'typical of the time'. (80/81 'turn of the year').

Benefits:- are expressed in terms of delivered (i.e. purchased) energy and the cash value of such energy savings. This energy is valued at 1980/81 prices namely: gas 0.72p/kwhr, electricity 4.22p/kwhr (off peak 2.33p/kwhr), solid fuel 1.77p/kwhr. Since monitoring did not proceed for a complete year both before and after improvements it has been necessary to model annual

energy savings from measured data over approximately 18 weeks before, and 30 weeks after, improvements.

Method 1. Uses measured EDHLs normalised for temperature differences to provide an estimate of annual energy savings. This is a favourable estimate since it places an 'energy value' on any temperature rises.

Method 2. Uses the measured difference in purchased energy to produce an estimate of an annual saving. Temperature rises will have an adverse effect on this estimate of the benefit. However, it is a more 'honest' estimate of cash savings.

Mean Costs and Benefits are:-

Insulation	..	£529	
Heating	...	£ 68	
Infiltration	..	£183	
<u>Capital Cost</u>	..	£780	per dwelling
	Fuel saving (kwhrs)	Fuel value (£ p. a.)	Simple Payback period (yrs)
Method 1	3605	31.50	24.8
Method 2	2532	26.20	29.8 *

(* excludes 4 cases of clearly anomalous behaviour).

Figure 5 (a) and (b) relates the estimates of energy saving by Methods 1 and 2 to capital costs. Of interest is the apparent division between a group of simple cheap retrofits for which returns are high and a group of more complex and expensive retrofits for which returns are disappointingly low. The figure implies a 'break point' at around £500, below this there is a strong relationship between capital cost and reduced EDHL but above this the slope is markedly reduced. It seems that the relatively simple (cheap) measures perform 'reliably' but then when more complex, sophisticated (expensive) measures are implemented (especially in combination with other ECMS) then under achievement severely limits the cost benefit ratio. This phenomenon has been reported elsewhere, e.g. Swedish Council for Building Research (8) who reported good returns for ECMs when implemented individually but unexplained under-performance when ECMs were combined. This may be related to the point that the expensive retrofit packages are likely to be very sensitive to the condition of the base structure and/or any defects in installation.

4. SUMMARY OF FIELD WORK

- a) The study was very much one of 25 separate case studies and as such any generalisations cannot be made with statistical confidence.
- b) The form of monitoring, although simple, has proved effective. EDHL has been useful and the development of this along the lines of 'Energy Signatures' is seen as a valuable diagnostic aid.
- c) E.C.M. selection was approached by the design team in a relatively structured manner so as to evolve retrofit packages 'suited' to the dwelling compatible with a national programme of upgrading. However, there were many instances where on-site fine details influenced these selections - primarily in one of 3 ways:-
 - to make an ECM more expensive than expected from unit costs.
 - to make an ECM virtually impossible in realistically practical terms.
 - to suggest an ECM by virtue of simplicity.

These influences injected 'noise' into an originally 'tidy' strategy such that the final set of packages appears to be more 'ad hoc'.

- d) The successful implementation of an ECM is very dependent upon its level of sophistication, the condition or suitability of the base structure, the skill/experience of operatives. If one or more of these is 'suspect' then difficulties are likely to be generated.
- e) Predicted and Empirical estimates of heat loss agree reasonably well although the former tends to be higher. The agreement between predicted and empirical estimates of the changes to heat loss is less satisfactory. The empirical estimate is, on average, 78% of the predicted but with a range of 30 to 279 %. A great majority of those with a correspondence of less than the average are associated with retrofits including external insulation and double glazing.
- f) Temperature changes induced by the retrofits were encouragingly small. Much of the temperature rise was associated with slower cooling and cannot be regarded as 'avoidable'.
- g) Ventilation appears throughout this study, as others, as an area where insufficient data was calculable in relation to the known significance to the energy balance.

Pressurisation tests on 5 houses (carried out by British Gas) proved illuminating both in terms of the value of this technique and the results it provided. (See Section 5).

- h) Empirical estimates of fuel saving agreed well with that which could have been predicted at design stages using for example the model of McNair (7). This average agreement is encouraging although within the sample there were 5 cases with savings less than 50% and 5 cases with more than 150% of the predicted. Most of these can be satisfactorily explained but the argument that on average a retrofit package will produce a stated saving will be of little consolation to those who 'under achieve' and a pleasant surprise to those who 'overachieve'. It is then important to refine expectations of 'achievement'. - the fact that almost 50% of this field study deviated so far from prediction must be regarded as unacceptable.

5. VENTILATION EFFECTS

This section reflects upon those findings capable of providing some insight into the division of reduced envelope heat loss between fabric and ventilation effects.

5.1 Empirical Estimates of Ventilation Rates

In theory it might be possible to arrive at an estimate of ventilation rates from

and
$$\begin{aligned} \text{E.D.H.L.} &= \text{P.F.H.L.} + \text{E.V.H.L.} \\ \text{F.V.H.L.} &= \text{E.V.R. (O.33 Vol.)} \end{aligned}$$

where E.D.H.L. = Empirical Design Heat Loss (W/°C)
P.F.H.L. = Predicted Fabric Heat Loss (W/°C)
(conduction)
E.V.H.L. = Empirical Ventilation Heat loss (W/°C).
E.V.R. = Empirical Ventilation Rate (ac/hr)
Vol. = Volume of dwelling (m³).

Thus, EVR is 'available' for fortnightly periods both during the 'before' and 'after' monitoring periods.

There is generally a relationship between EDHL (and hence EVR) with external temperature which substantiates theories of increased window use as this rises (IO). There are insufficient data points to confirm whether there is an external temperature below which window use is negligible (viz. stable E. D. H. L and E. V. R.).

Given an original E.D.H.L (E.V.R.) vTex relationship then after retrofitting it is expected that the greatest reduction will occur at low values of Tex (and E.D.H.L/E.V.R.) since this

relates to windows being closed and hence both insulation and air infiltration reduction measures are operating. However, at higher values of Tex (and E.D.H.L./E.V.R.) although fabric measures are still operative air infiltration reduction measures are 'by-passed' by window use. In fact fewer than 20% of the dwellings conform to this idealised behaviour, the remainder producing either an equal decrease at all temperatures or (limited window use, or increased window use at lower temperatures?) a greater decrease at low values (increased window use at higher temperatures?)

Although these and other discrepancies in the E.D.H.L./E.V.R. data can be adequately explained by reference to the detailed knowledge of use and other influences in these field trial dwellings, the reverse is not true. These difficulties are further compounded by inaccuracies in P.F.H.L. As referred to in section 3 the field trials identified under-achievement associated with external wall insulation and double glazing and this reflects itself in the derived E.V.Rs for these dwellings. The average E.V.Rs before and after improvements is 0.92 and 0.71 ac/hr respectively (14 dwellings) when the 'under-achievers' are excluded from the sample.

Figure 6 illustrates some of the effects referred to above:-

- Fig. 6a) Dwelling No. 3. No Fabric E.C.Ms. E.V.R. reduced from 1.7 to 1.0 ac/hr. The only other dwelling without fabric measures was a flat (No. 22) which showed an E.V.R. reduction of 1.3 to 1.0 ac/hr.
- Fig. 6b) Dwelling No. 15. Referred to in Section 3 where an increase in condensation led to increased window use. E.V.R. increased from 0.4 to 0.5 ac/hr. These low values are compatible with a single storey terraced dwelling.
- Fig. 6c) Dwelling No. 10. Representative of dwellings where permanent ventilation (windows or fixed units) inhibits reductions in E.V.R. due to air infiltration measures. E.V.R. is 1.3 before and 1.2 afterwards. (Users have some windows open all the time).
- Fig. 6d) Dwelling No. 17 'Over achievement' due to greater reduction in ventilation than predicted. A first floor flat with heating of two rooms. The removal of permanent vents and weatherstripping has greatly reduced the amount of adventitious infiltration. The relationship with windspeed indicates the influence of the permanent vents to heated spaces. E.V.R. was reduced by 0.6 ac/hr.

5.2 Predicted Infiltration Rates

5 dwellings (Nos. 2, 5, 10, 13 and 19) were subjected to pressurisation tests (before and after improvement) by staff of the Watson House Research and Development Division of the British Gas Corporation. Their findings (Table III) suggest that the E.C.Ms (cavity fill, vent sealing, draughtstripping) have produced leakage reductions of around 55% (flues sealed) in 4 of the 5 dwellings, with approximately 15% attributable to vent sealing. Dwelling No. 13 showed a much lower reduction commensurate with it being originally relatively 'tight'.

In addition to these measurements the leakage v pressure difference curves have been used together with climatic and envelope data to input into the predictive model of the Lawrence Berkeley Laboratory. (II).

These predicted infiltration and empirical ventilation rates (E.V.R.) are:-

Dwelling	2		5		10		13		19	
	B	A	B	A	E	A	B	A	B	A
Average infiltration rate predicted by LBL model (II)	2.5	0.5	1.2	0.4	1.1	0.7	0.8	0.8	1.6	0.6
Empirical ventilation rate.	1.1	0.8	1.2	0.9	1.8	1.6	1.3	1.3	1.5	0.9

(all ac/hr)

Figure 7 illustrates the fortnightly variations. It is expected that the predictions concur with the lower values of E.V.R. (i.e. windows closed). This is seen to be approximately the case although there are clear exceptions notably dwelling 2 where the very high leakage before improvements would seem to invalidate the use of the predictive model.

In all cases except No. 13 the reduction in E.V.R. is less than that indicated by the predictive infiltration reduction and this concurs with a knowledge of window use.

5.3 Summary

The use of empirical estimates of ventilation rates (E.V.R) although a promising technique is frustrated by major deviations between the theoretical and actual performance of some Fabric ELMs.

Where this is not the case the results suggest that reduction in ventilation heat loss actually achieved are highly sensitive to the type and condition of individual dwellings as well as the occupants pattern of window use. However, reductions of 0.3 to 0.6 ac/hr seem typical.

These findings undermine confidence in rules of thumb estimates of cost efficiency for weatherstripping as proposed by e.g. Freemans (12).

6. CONCLUSIONS

- standard retrofit packages based upon 'design' information are inadequate. ECMS perform differently on different dwellings, and also differently when in various combinations. Standardised solutions do not recognise the influence (upon costs and benefits) of fine scale details of individual dwellings.
- predictive models for energy savings which are based upon theoretical estimates of changes in heat loss (e.g. McNair(7)) are adequate for average performances.
- it seems that retrofit packages fall into two groups. Put simply these are 'achievers' and 'underachievers'. The former group tend to be simple and low cost whereas the latter are relatively more complex and expensive. The underachievers do so because of their sensitivity to factors such as condition, user influences, workmanship, etc. Those ECMs tending to underachievement seem inappropriate for specification except in cases where older properties are being substantially rehabilitated or the property in question is relatively new, i.e. the base structure is sound and already contains the 'simple' ECMS.
- a more integrated approach to the dwelling is needed than was possible in the EIK field trials. These reflected the contemporary ethos of focusing retrofits on the long life fabric. This overlooks the, now more widely appreciated, considerable potential for energy saving through attention to the dwellings heating and hot water systems.
- it is difficult (impossible?) to separate the effects of different ECMs and in particular the impact of air infiltration reduction measures. As with the whole packages a rule of thumb design estimate of a reduction in ventilation heat loss of 25% (1.0 to 0.75 ac/hr) may concur with average attainment but there are large variations between individual dwellings.
- In several cases there is strong but circumstantial evidence that either due to increased condensation on windows and/or higher room temperatures there is an increase in window use which has inhibited the full attainment of potential savings.

- Finally it cannot be overemphasised that the EIK study consisted essentially of 25 separate case studies and as such scientifically acceptable conclusions and hence generalisations cannot be made. Nevertheless it is hoped that the case law which such studies represent is not undervalued in the field of application.

7. ACKNOWLEDGEMENTS

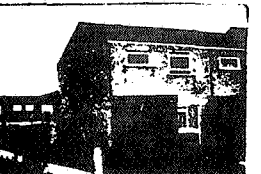
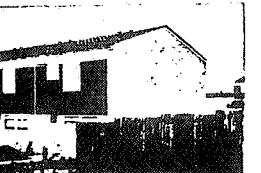
Birmingham City Housing Department and the Department of the Environment as joint initiators of these field trials deserve full recognition for entering into the difficult but essential arena of full scale testing. The assistance of West Midlands Gas and especially Watson House research station of the British Gas Research & Development Division who carried out pressurisation tests, is greatly appreciated. Finally the co-operation of dwelling occupants should not go unrecognised.

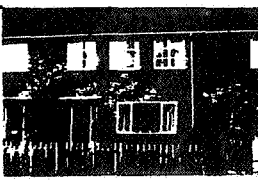
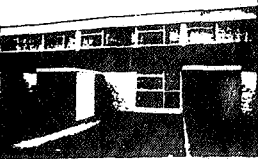
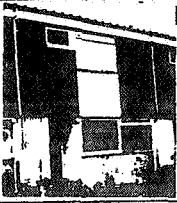
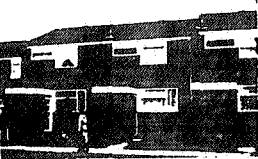
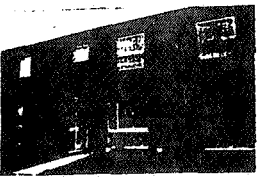
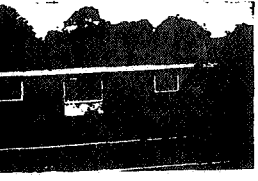
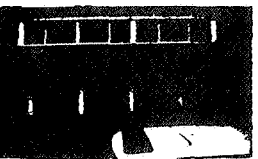
The views expressed in this paper are those of the authors alone and do not necessarily reflect those of other contributors to the EIK project.

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TABLE I

	DESCRIPTION	ECMs	Cost £
1.	 Type Semi-det. 3 bed. 1931. 9" solid brick wall, slate roof with 50 mm ins., timber suspended and solid ground floor. Services Solid fuel (liv) room heater with electric heating to landing and bed 2. Gas cooker	80mm roof ins (50- 130) Draughtstrip, fill airbricks Hardboard line liv. floor Front lobby Replace louvre windows with casement 100 mm ins. to HWC (25- 100) New room heater and 3 rads. Underfloor draught and servicing	49 132 58 114 117 21 894
2.	 Type Semi-det. 3 bed. 1948. Cavity wall, tile roof with 100 mm ins. Ground floor concrete and boarding on battens on concrete. Services Gas fire both livs. DHW electric immersion (25 mm ins). Gas cooker	75 mm cavity fill Draughtstrip, fill airbricks 75 mm ins. to HWC (25- 75) Servicing to gas appliances	129 177 7 12
3.	 Type Semi-det. 3 bed. 1953. B.I.S.F. tubular steel frame, render on expanded metal lower floor, steel trough sheeting upper. Steel roof with 100 mm ins. Concrete floor. Services Gas fire in liv, flueless gas convector then Feb 1980 paraffin in bed 2 DHW electric immersion (no ins). Gas cooker	(Rockwool to ext. walls (Feb '80) Draughtstrip Close unused flue HWC Boxed inc 50 mm ins (0- 50) Wind up timer for DHW Balanced flue gas heater to hall Servicing to gas appliances	300 157 60 37 45 103 9
4.	 Type Semi-det. 3 bed. 1928. 9" solid brick wall rendered upper floor, slate roof (no ins). Timber suspended and concrete ground floor. Services Gas fire in liv, electric heater in bed 1. DHW electric immersion (no ins). Gas cooker	100 mm ins. to roof (0- 100) 50 mm ext wall ins Draughtstrip, fill air bricks Hardboard line liv. floor Front porch 100 mm ins to HWC (0- 100) Wind up timer to DHW Servicing gas appliances	43 2136 142 52 781 7 45 9
5.	 Type End terr. 2 bed. 1951. Cavity walls, tile roof with 100 mm ins. Concrete floor. Services Gas fires in liv. and bed 1, flueless gas convector in hall. D.H.W. electric immersion (100 mm ins). Gas cooker.	50 mm cavity fill Double glazing to liv. and bed 1 Draughtstrip, fill airbricks Entrance porch 75 mm ins to H.W.C. (100 - 75) Wind up timer for D.H.W. Servicing gas appliances	129 654 174 511 7 45 19
6.	 Type End terr. 3 bed. 1970. No fines concrete walls, tile roof with 25 mm ins. concrete floor, gasket weatherstripping to windows. Services Gas warm air unit, outlets in kit, liv, hall, roomstat and timeclock. Electric heater in bed 1. D.H.W. electric immersion (25 mm ins). Inst electric shower. Gas cooker	80 mm roof ins (25 - 105) 50 mm external wall ins Draughtstrip, fill airbricks 75 mm ins to H.W.C. (25- 100) Servicing gas appliances	48 2476 84 7 12
7.	 Type End terr. 3 bed. 1969. Timber frame with 25 mm ins. Clad with plywood at ground floor, tile hanging at 1st floor, brick at gable end; tile roof with 25 mm ins. Concrete floor. Services Gas warm air unit, outlets in every room, roomstat and timeclock. D.H.W. electric immersion (25 mm ins). Gas cooker	100 mm roof ins (25- 125) 25 mm ins to bottom panel of liv. window Draughtstrip, fill airbricks 75 mm ins to H.W.C. (25- 75) Time clock for D.H.W. Servicing gas appliances	68 12 120 7 50 12
8.	 Type Mid terr. 2 bed. pre 1900. 9" solid brick walls, slate roof with 30 mm ins. Timber suspended and solid ground floor, 'tunnel' entry to rear. Services Gas fire in liv. 1, paraffin heater in liv. 2. D.H.W. electric immersion (12 mm ins). Gas cooker	80 mm ins. to main roof (30- 110) 50 mm ext. wall ins. to rear entrance 'tunnel' Flush door replacing glazed door Dryline front bed.(50 mm ins) Draughtstrip, fill airbricks Hardboard line ground floor Gas back boiler for D.H.W. 75 mm ins to H.W.C. (12- 89)	48 1460 63 149 150 85 270 7

	DESCRIPTION	ECMs	Cost £
9.	 <u>Type</u> Mid terr. 3 bed. 1928. 9" solid brick wall, slate roof (no ins). Timber suspended and concrete ground floor. <u>Services</u> Gas fire in liv, electric heater in bed 1. D.H.W. electric immersion (25 mm ins). Gas cooker	100 mm ins. to roof (0- 100) Draughtstrip, fill air bricks 100 mm ins to H.W.C. (25- 100) Wind up timer to D.H.W. Servicing gas appliances	35 145 7 45 9
10.	 <u>Type</u> Mid terr. 3 bed. 1947. Cavity wall, tile roof (75 mm ins). Boarding on battens on concrete ground floor. 'Tunnel' entry. Rear draught lobby. <u>Services</u> Gas fires both livs, electric heater in bed 1. D.H.W. gas back boiler and electric immersion (25 mm ins). Gas cooker	80 mm ins. to roof (75- 155) 50 mm cavity fill Draughtstrip, fill airbricks Servicing gas appliances	56 70 126 22
11.	 <u>Type</u> Mid terr. 3 bed. 1967. Pre-cast panel walls (25 mm ins), tile hung at 1st floor. Tile roof (25 mm ins). Concrete floor. <u>Services</u> Gas warm air unit outlets in both livs, kitchen and landing; roomstat and time clock. D.H.W. electric immersion (25 mm ins). Gas cooker	80 mm ins. to roof (25- 105) Double glazing to both livs. Draughtstrip, fill airbricks 75 mm ins. to H.W.C. (25- 75) Servicing gas appliances Return air grille above lounge door	54 417 81 7 12 6
12.	 <u>Type</u> Mid terr. 3 bed. 1968. Pre-cost walls (19 mm ins), tile roof (25 mm ins) Concrete floor (edge ins). <u>Services</u> Gas warm air unit outlets in liv, kitchen, hall, 2 beds; roomstat and time clock. D.H.W. electric immersion (25 mm ins). Gas cooker	80 mm ins. to roof (25- 105) Double glazing to liv. Draughtstrip, fill air bricks 75 mm ins to H.W.C. (25- 75) Servicing gas appliances 25mm ins. to store wall Wind up timer (+ later removal)	56 250 81 7 12 19 61
13.	 <u>Type</u> Mid terr. 3 bed. 1970. Cavity walls, tile roof with 25 mm ins. Concrete floor. Front draught lobby (installed by tenant) <u>Services</u> Gas warm air unit, outlets in liv, kit, hall, roomstat and time clock. D.H.W. electric immersion (25mm ins). Gas cooker	80 mm roof ins. (25- 105) 55 mm cavity fill Draughtstrip, fill airbricks 75 mm ins. to H.W.C. (25- 75) Servicing gas appliances	47 57 94 7 12
14.	 <u>Type</u> Mid terr. 2 bed. 1977. Cavity wall at ground floor. Tile hanging on blockwork at 1st floor. Tile roof with 25 mm ins. Concrete floor. Front draught lobby. <u>Services</u> Gas boiler with radiators in liv, kit, hall, bath. D.H.W. from boiler (25 mm ins). Gas cooker	80 mm roof ins. (25- 105) Double glazing to liv. Draughtstrip, fill air bricks 75 mm ins. to H.W.C. (25- 100) Thermostatic radiator valves Servicing gas appliances	49 154 124 7 116 12
15.	 <u>Type</u> Mid terr. bungalow, 1 bed. 1968. Cavity walls. Tile roof with 25 mm ins. Concrete floor. <u>Services</u> Gas warm air unit outlets in kit, liv, hall, bed, roomstat and time-clock. D.H.W. electric immersion (25 mm ins). Gas cooker	100 mm roof ins (25- 125) Draughtstrip, fill airbricks 75 mm ins. to H.W.C. (25- 75) Servicing gas appliances Wind up timer to DHW	59 69 7 18 45
16.	 <u>Type</u> 1st Floor flat. 2 bed. 1968. Pre cast panel walls, tile hung with 25 mm ins. Tile roof with 25 mm ins. Chipboard floor on battens on suspended concrete slab (over garages) with 25 mm ins. <u>Services</u> Gas warm air unit, outlets in liv, hall, bed 2, roomstat and time clock. D.H.W. electric immersion (25 mm ins). Gas cooker	100 mm roof ins. (25- 125) Double glazing to liv. Draughtstrip, fill airbricks 75 mm ins. to H.W.C. (25- 75) Service gas appliances	93 290 102 7 12

	DESCRIPTION	ECMs	Cost £
17.	<u>Type</u> 1st Floor flat, 1 bed, 1970 conversion, house 1930. 9" solid brick wall, slate roof (old). Cavity wall, flat timber/board/felt roof (25 mm ins)(new) <u>Services</u> Gas fire in liv, paraffin on landing and in kitchen. D.H.W. electric immersion (25 mm ins). Gas cooker	150 mm ins. to slate roof(0-150) 78 75 mm ins. to kitchen ceiling (25-100) 136 50 mm cavity fill 83 Draughtstrip, fill air bricks 63 Replace louvre windows with casements 83 75 mm ins. to H.W.C. (25- 75) 7 Wind up timer to D.H.W. 45 Servicing gas appliances 9	
18.	<u>Type</u> Ground floor flat in 2 storey block, 1 bed. 1957. Cavity wall, concrete floor. <u>Services</u> Gas fire in liv, calor gas in bed. D.H.W. electric immersion (no ins). - unused. Gas cooker	60 mm cavity fill 75 Draughtstrip, fill airbricks 65 Replace D.H.W. with gas multipoint 302 Balanced flue gas heater to bed. 130 Servicing gas appliances 9	
19.	<u>Type</u> Top floor flat in 3 storey block, 3 bed. 1954. Cavity walls. Tile roof with 100 mm ins. <u>Services</u> Gas fire in liv. D.H.W. electric immersion (10 mm ins). Gas cooker	60 mm roof ins (100- 160) 74 Drylining (35 mm ins) to bed3/stair wall, kitchen/drying room wall) 237 Draughtstrip, fill airbricks 98 75 mm ins. to H.W.C. (10- 75) 37 Balanced flue gas conv. to bed 1 224 Servicing gas appliances. 9 (Inspection of cavity walling) 21)	
20.	<u>Type</u> Mid floor flat in 3 storey block, 2 bed. 1952. No fines concrete walls. Balcony glazed in by tenant <u>Services</u> Gas fire in liv, electric heater in hall. D.H.W. electric immersion (no ins). Gas cooker	50 mm external wall ins. 1087 Draughtstrip, fill air bricks 112 Remove window vents 19 H.W.C. boxed inc. 50 mm ins (0 (0- 50) 37 Wind up timer for D.H.W. 45 Servicing gas appliances 9	
21.	<u>Type</u> Gnd flr maisonette in 4 storey deck access block. 3 bed. 1960. Cavity walls Concrete floor. Ceiling exposed above bed 1, bed 2 and bath. Suspended concrete floor in bed 1 and bath over entry. <u>Services</u> Gas fire in liv. Flueless gas conversion in hall. D.H.W. electric immersion (25 mm ins). Gas cooker	50 mm ins. to soffit of entry 235 Dryline ceilings of bed 1, 2 and bath (50 mm ins) 340 Draughtstrip, fill airbricks 102 75 mm ins. to H.W.C. (25- 100) 7 Time clock for D.H.W. 50 Servicing gas appliances 9	
22.	<u>Type</u> Gnd. flr. flat in 12 storey block, 1 bed. 1969. Pre cast panel walls with 25 mm ins. Concrete floor with 12 mm ins. Integral rubber window weather-stripping <u>Services</u> Off peak electric storage heaters in liv. D.H.W. off peak electric immersion (50 mm ins). Electric cooker	Draughtstrip, fill airbricks 81	
24.	<u>Type</u> 3rd flr. flat in 11 storey block, 1 bed. 1968. Pre cast panel walls with 12 mm ins. 25 mm ins. in communal wall to ventilated fire lobby <u>Services</u> Off peak electric underfloor heating to liv. and hall. Fixed electric fire in liv. Electric heater in bed. D.H.W. off peak electric immersion (25 mm ins). Electric cooker	Balcony opening glazed in 396 Draughtstrip, fill airbricks 51 75 mm ins. to H.W.C. (25- 100) 7	
25.	<u>Type</u> Top flr. flat in 12 storey block, 1 bed. 1964. Concrete frame with concrete infill panels with 50 mm ins. Concrete roof. Double glazing (4 mm gap) in bed, and kit. <u>Services</u> Electric heater in liv. D.H.W. off peak electric immersion (25 mm ins). Gas cooker.	Double glazing to liv. and bed. 378 Dryline liv, bed, kit (50 mm ins) 546 Draughtstrip 68 75 mm ins. to H.W.C. (25- 100) 7 Servicing gas appliance 6	
23.	<u>Type & Services</u> as 24 but no double glazing	Double glazing to liv. and bed. 378 Draughtstrip 68 75 mm ins. to HWC (25-100) 7	

TABLE II

E.C.M.s. Cost - Benefits - Pre-Installation Knowledge

ECM Type	£	Change in $U/(W/^{\circ}Cm^2)$	£/ ΔU
Air infiltration			
- Window & door weatherstripping Silicone rubber & brush	1.74 to 2.02/m		
- Airbrick removal	3 ea		
- Unused flues - 'sealed'	27 ea		
- Louvre windows replaced	83 - 109 ea		
- Suspended floors 'lined'	3/m ²		
- Porches	511 - 655		
Roof Insulation			
- Fibreglassquilt (tmm)	(40 + $\frac{1}{2}$) per m ²	1.70 to 0.17	0.89 to 11.11
- Drylining (50mm polystyrene)	12 - 15 per m ²	0.5 to 1.5	8 to 28
Wall Insulation			
- Cavity (U foam) - tmm	(122 + $\frac{15}{25}$) per m ²	1.0	1.6
- Internal (50mm polystyrene)	11.50/m ²	1.5	7.7
- External (50mm insulation)	29/m ²	1.5	19.3
Double glazing			
- Secondary Aluminium frame	51/m ²	2.8	18.2

Table III Pressurisation Test Results *

ETK No	2		5		10		13		19	
Approx vol. (m ³)	158		128		158		150		114	
Approx. surface area (m ²)	141		123		127		90		145	
Before or After ECMS	B	A	B	A	B	A	B	A	B	A
No. of air vents	6	0	6	0	5	0	1	1	6	1
No. of flues	3	3	2	2	3	3	1	1	1	1
Leakage at 50pa(m ³ /s)										
- flues sealed	1.36	0.48	0.89	0.38	0.89	0.41	0.75	0.64	0.78	0.30
- corrected for flues	1.66	0.78	1.09	0.58	1.19	0.71	0.79	0.68	0.88	0.40
Effective Orifice Area (50pa)m ²	0.247	0.087	0.161	0.069	0.161	0.075	0.136	0.116	0.142	0.055
% Reduc - All		65		57		53		15		61
- vents alone		23		6		6		-		29

(* Courtesy of British Gas - Research & Development Div.)

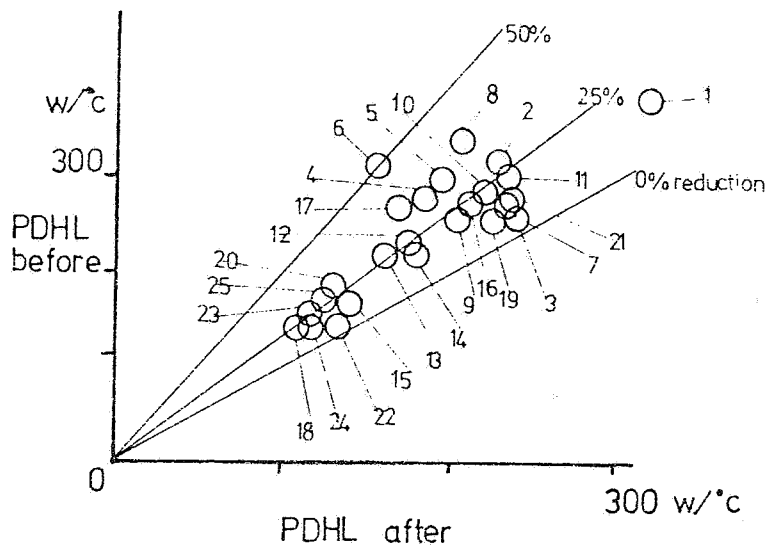


FIGURE 1

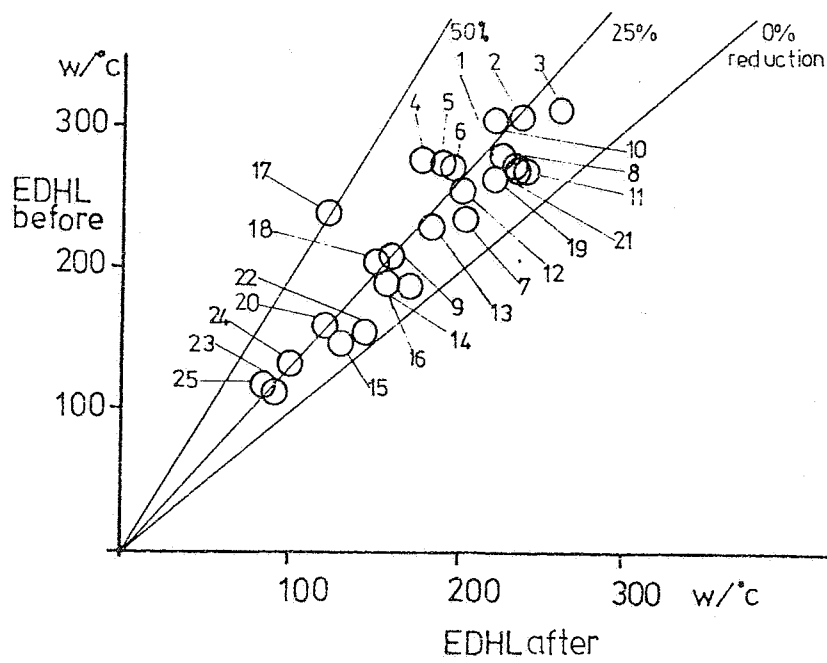


FIGURE 2

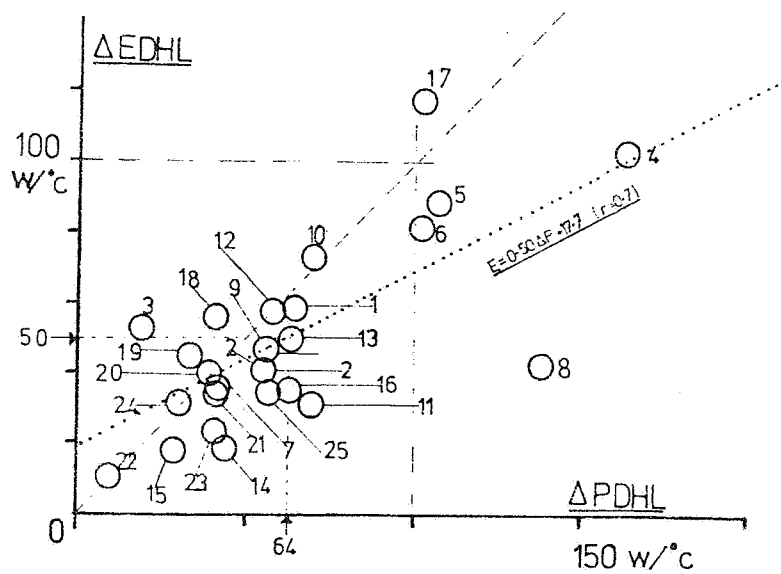


FIGURE 3

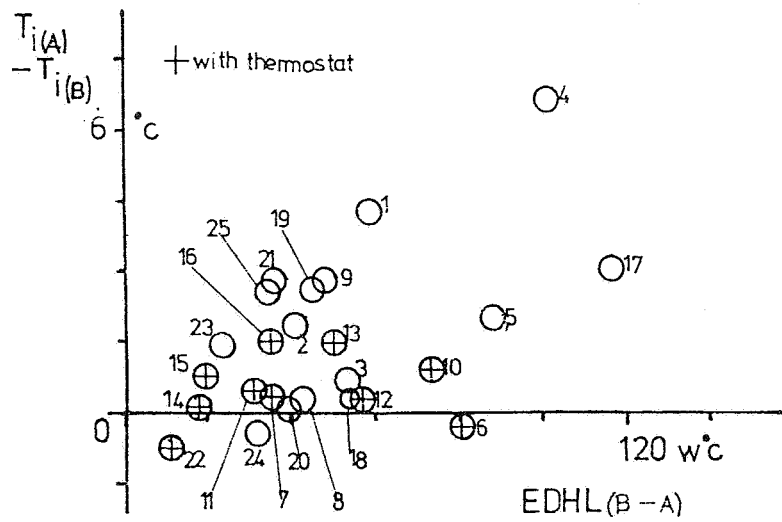
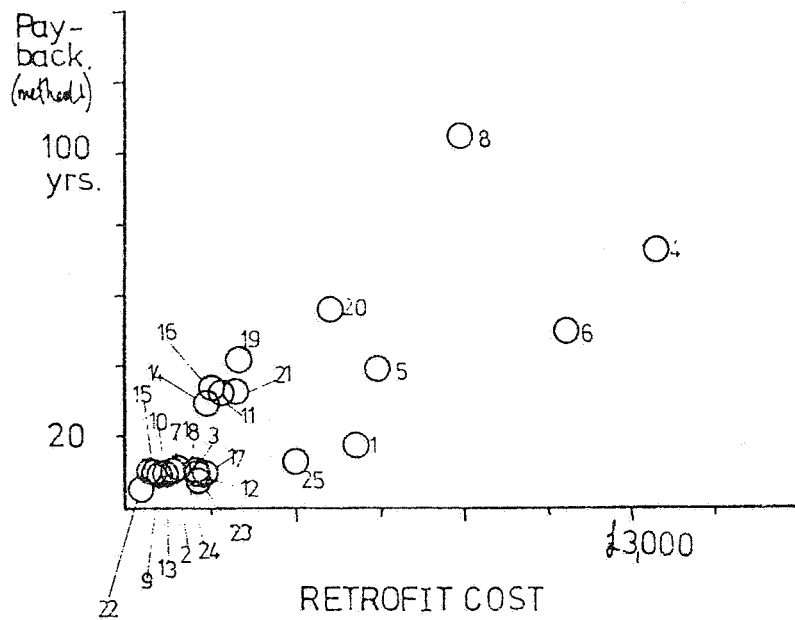


FIGURE 4



FIGURE

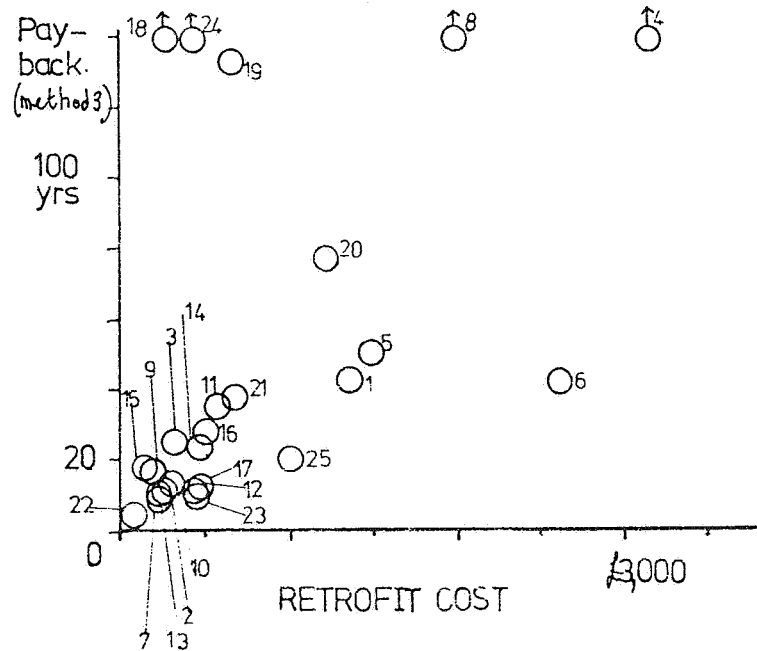
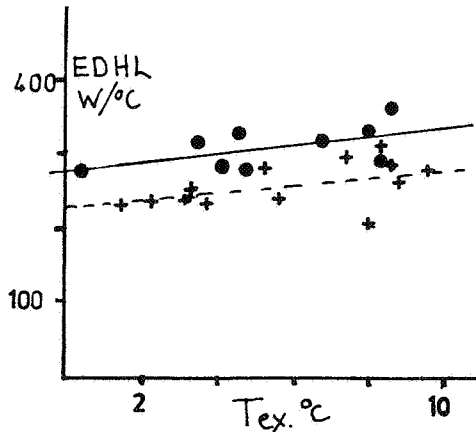


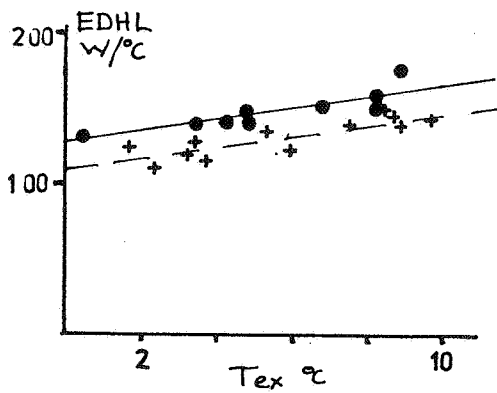
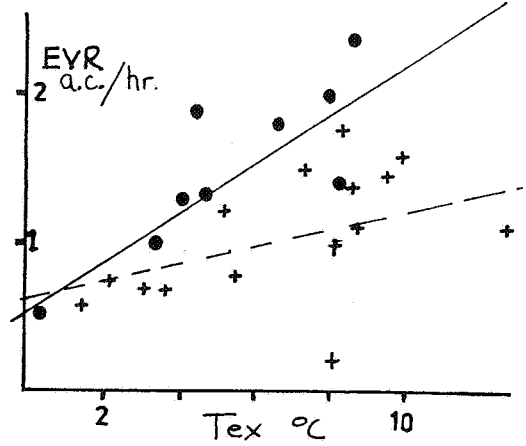
FIGURE 5(b)

FIG. 6

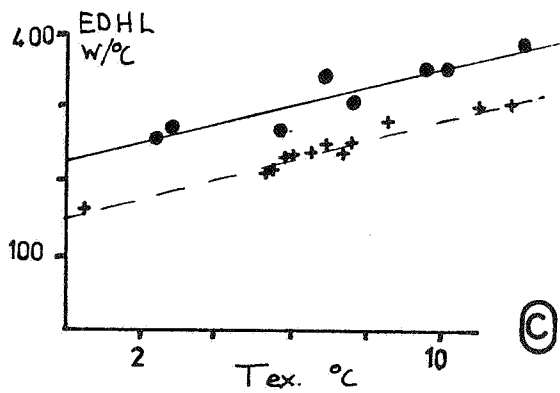
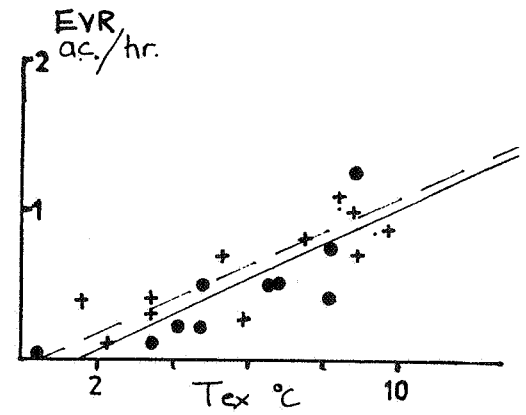
—●— Before } Improvements
 ---+--- After }



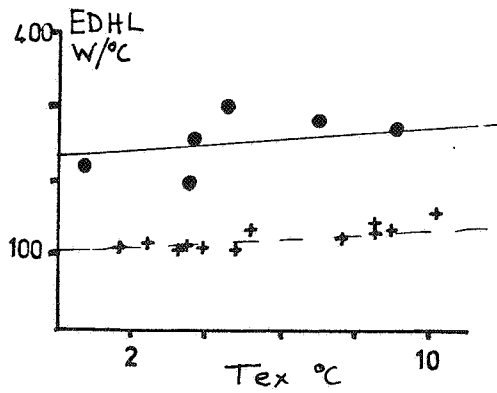
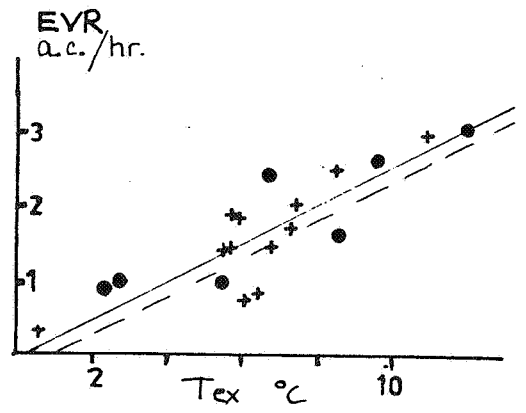
(a)



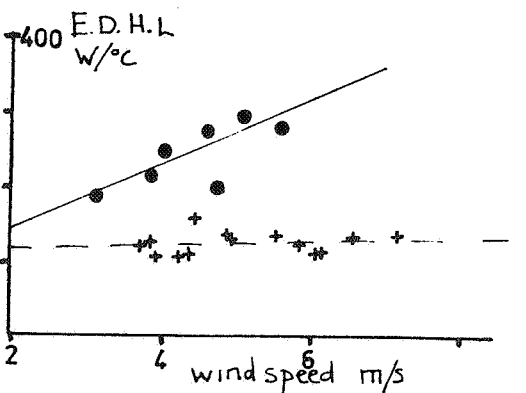
(b)



(c)



(d)



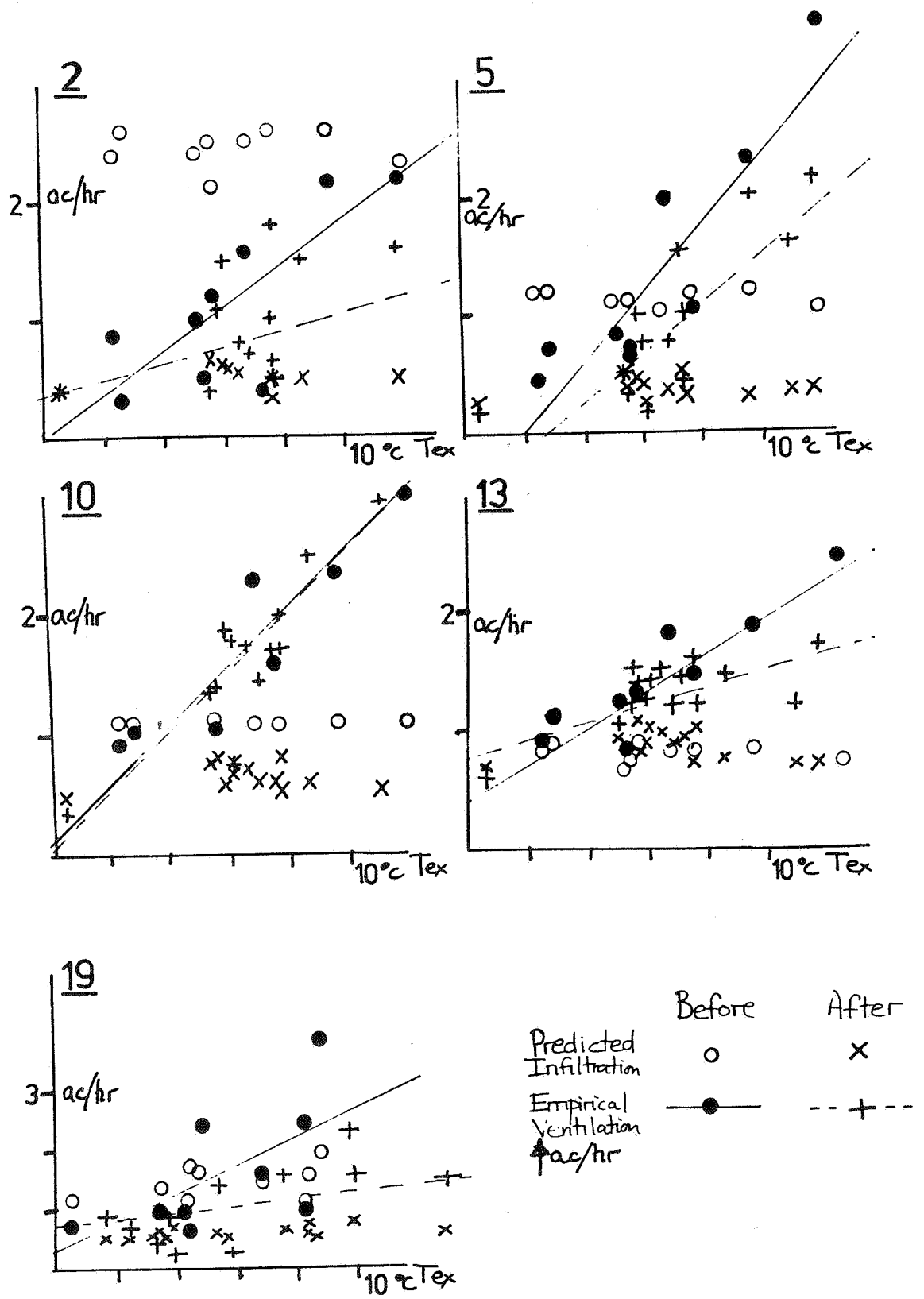


FIGURE 7

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 12

EFFECT OF A GAS FURNACE CHIMNEY ON THE AIR LEAKAGE CHARACTERISTIC
OF A TWO-STOREY DETACHED HOUSE

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SYNOPSIS

The results are reported of an experimental assessment of the effect of a chimney on the air leakage characteristic of an unoccupied, two-storey detached house heated by either gas or electric furnace. Measurements were taken of air tightness values and air infiltration rates with the chimney capped and uncapped. Infiltration rates were also measured during operation of the electric furnace.

It was found that the over-all air infiltration rate was greater with the gas furnace than with the electric furnace under the same weather conditions, and that the difference was smaller than the air flow rate through the chimney. When the gas furnace was in operation, stack action was the dominant driving potential for air infiltration with wind speeds up to 7 m/s (16 mph) and with inside-to-outside temperature differences greater than 20 K (36 R).

Key Words: air-leakage, energy, electric furnace, gas-fired furnace, chimney, measurement, tracer gas.

1. INTRODUCTION

Because of the rising cost of heating fuel and the attractive subsidy from the federal government for off-oil conversion, more and more Canadian homeowners are switching to natural gas for heating. As a conventional gas-fired furnace has a chimney for venting combustion products and there is usually a draft diverter located near the exit of the combustion chamber, a vertical leakage path extending from basement to roof exists in most gas-heated houses. Consequently, inside air moves upward continuously through the chimney during the heating season. As a result, air infiltration is controlled not only by stack action and wind but also by air movement in the gas furnace chimney.

Although air leakage studies on gas-heated houses have been carried out,^{1,2} the effect of chimneys on air infiltration is still not fully understood. A project was therefore undertaken during the heating season of 1980-81 to study the air leakage characteristic of a two-storey house that could be heated by either a gas-fired furnace or an electric furnace. The main objectives were to determine

- (i) the influence of a gas furnace chimney on air tightness and air infiltration,
- (ii) the influence of various operating modes of a gas furnace on air infiltration, and

- (iii) the combined influence of weather and a gas chimney on air infiltration.

2. TEST HOUSE

The house under consideration (Figure 1) is one of four houses involved in the HUDAC/NRC Mark XI energy research project.³ It is a two-storey dwelling with a basement, located in a developed residential area in Gloucester, Ontario. It already had an electric forced-air furnace, and for this study a gas furnace was installed in parallel with it. Two diverting dampers, one in the supply and the other in the return air ducts, were added to minimize cross air flow between the two furnaces. A Class C insulated chimney, 12.7 cm (5 in.) diam, was installed for the gas furnace. The flue pipe connecting the furnace to the chimney was 10 cm (4 in.) diam. Whenever the electric furnace was in operation the flue pipe was disconnected from the chimney and a metal cap was sealed into the chimney opening. The house was unoccupied when the study was conducted.

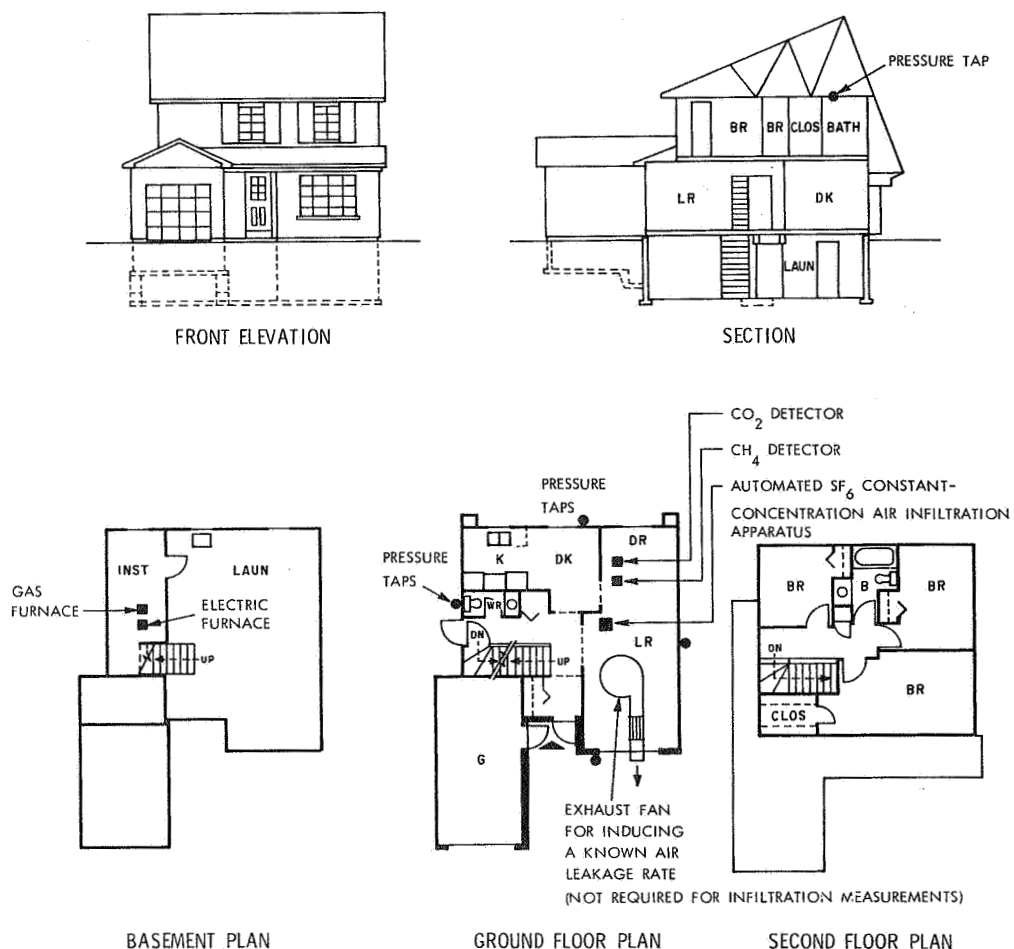


Figure 1 Test house

3. TEST METHODS

3.1 Air Tightness Values

Air tightness values with and without the chimney capped were measured using the fan pressurization method. The apparatus⁴ consisted of a centrifugal fan with a maximum capacity of 380 L/s (800 cfm) and a laminar-flow air flow meter (MERIAN LFE Element, accurate to 3% of the measured flow rate). A pair of total-pressure-averaging tubes and a thermocouple were installed in the flue pipe to measure the air flow rate through the chimney and the flue gas temperature, respectively. To facilitate comparison with house infiltration rates the measured chimney flow rates were corrected to room temperature.

3.2 Air Infiltration Rates under Controlled Gas Furnace Operation

Air infiltration rates for the house and air flow rates through the chimney were measured with the gas furnace operating under the following conditions: 1) burner on continuously; 2) burner off; and 3) burner cycling under thermostatic control.

An automated SF₆ constant-concentration tracer gas apparatus⁵ was used for infiltration rate measurements. The SF₆ detector for the apparatus was a pulsed-mode electron-capture detector. To cross-check the results of the SF₆ apparatus the infiltration rates were also measured simultaneously by the decay method,⁵ using CO₂ and CH₄ as the tracer gases. Concentrations of CO₂ and CH₄ were measured with infrared and flame-ionization gas analysers, respectively.

All three detectors were located in the living room on the ground floor of the house (Figure 1). The tracer gases were injected into the supply duct and samples collected from the return duct through individual injection and sampling tubes. Only one furnace fan was used to mix the tracer gases with the inside air.

3.3 Air Infiltration Rates under Normal Furnace Operating Modes

The air infiltration rates were measured continuously for three days with the electric furnace in operation and the chimney capped. For comparison, similar measurements were also taken with the gas furnace in operation. In both instances the furnace was under thermostatic control. The infiltration rates were measured with the automated constant-concentration tracer gas apparatus⁵ only.

3.4 Neutral Pressure Level

The neutral pressure levels were measured on a calm day with the burner both on and off. When the burner was off, the readings were taken with the chimney capped as well as uncapped. The neutral pressure level was determined by drawing a straight line through two simultaneously measured pressure differentials, one across the bottom of an entrance door and the other across the top of a second-floor window directly above the door.

4. RESULTS AND DISCUSSION

Figure 2 shows the air leakage rates of the house with the chimney both capped and uncapped. These results were fitted to Eq. (1) to obtain the flow coefficients and the exponents for both cases

$$Q = C A (\Delta P)^n \quad (1)$$

where

Q = over-all air leakage rate, L/s (cfm)

C = flow coefficient, $\text{L/s} \cdot \text{m}^2 \cdot \text{Pa}^n$ (cfm/ft² (in. of water) ^{n})

A = area of building envelope, area of exterior wall above grade and ceiling area of the top floor, 228 m² (2454 ft²)

ΔP = pressure difference across exterior wall, Pa (in. of water)

n = flow exponent.

The flow exponent is 0.71 for both cases and the corresponding flow coefficients are 0.107 $\text{L/s} \cdot \text{m}^2 \cdot \text{Pa}^{0.71}$ (1.06 cfm/ft²·in. of water^{0.71}) with the chimney uncapped and 0.098 (0.97) with it capped.

The difference between the two air leakage rates is, in theory, the air flow rate through the chimney. As shown in Figure 2, the chimney flow rate obtained by this method is smaller than that measured directly. The disagreement is attributable to the error of the indirect method, which uses the difference between two large air leakage rates to obtain the much smaller value for the chimney. Figure 2 indicates that for this house the air leakage rate through the chimney constitutes about 9% of the over-all air leakage rate.

The air infiltration rates under various modes of furnace operation were measured simultaneously, using three different tracer gases. Figure 3 compares CO₂ and CH₄ decay results with the SF₆ constant-concentration air change rates. As shown, the three results agreed closely with one another for air infiltration

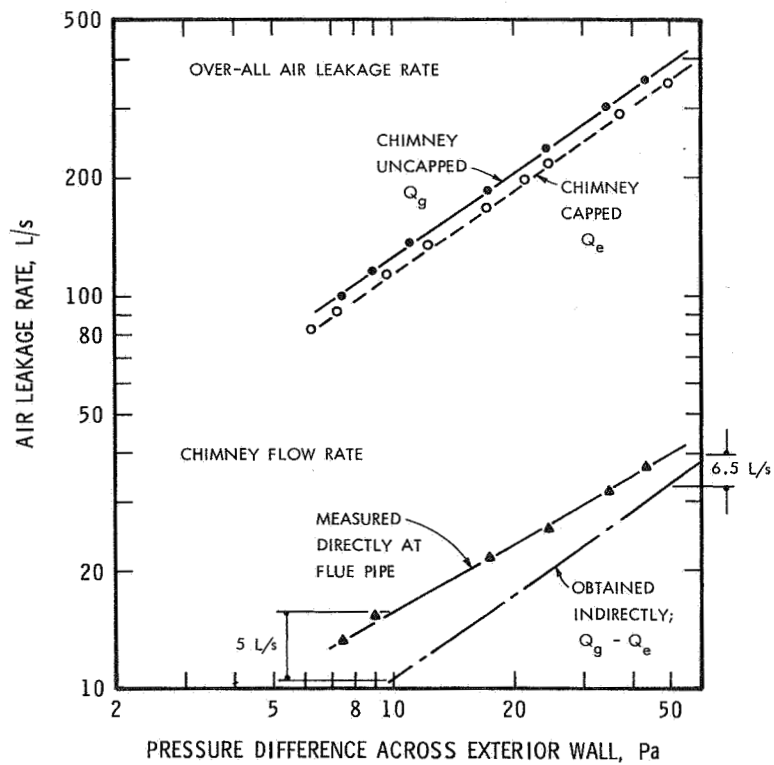


Figure 2 Over-all air leakage rates with and without chimney capped and air flow rate through chimney

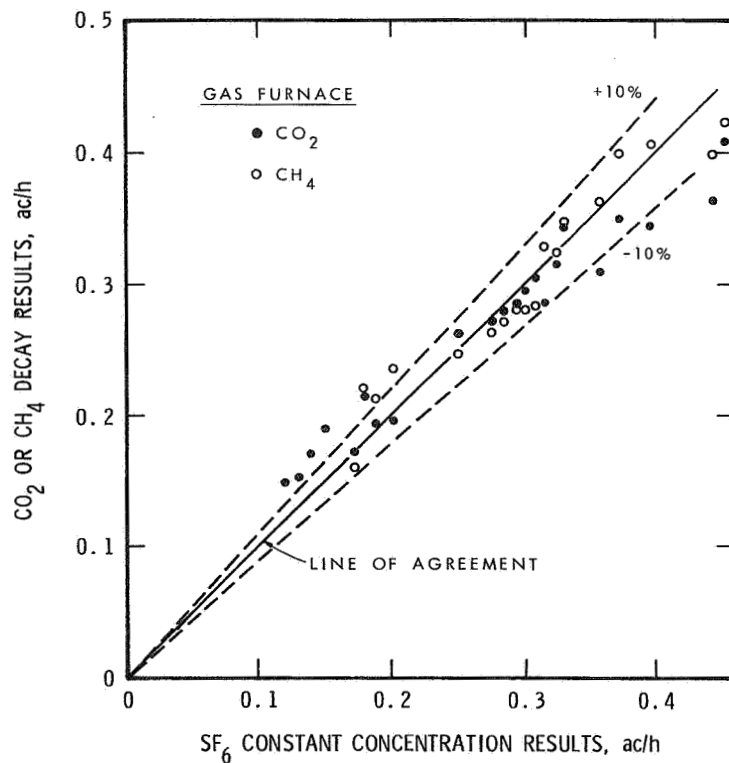


Figure 3 Comparison of air infiltration rate between SF₆ constant concentration and CO₂ or CH₄ decay methods

rates greater than 0.2 air changes per hour. Below this rate the SF₆ results appeared to be smaller than the corresponding decay results. The difference, however, was just slightly above the experimental error, which was estimated to be about $\pm 10\%$ of the measured value.⁶

Figure 4 shows the mean values for the three simultaneously obtained tracer gas results for gas burner operating continuously and not operating. The results indicate that the infiltration rates with the burner on were, on average, 10% greater than those with it off. The difference decreased as the inside-outside temperature difference increased.

These infiltration data are compared in Figure 5 with those measured separately under normal operating conditions of either the gas or the electric furnace. The results indicate that for similar weather conditions the infiltration rates with the burner cycling were almost identical to those with it shut down. When the electric furnace was in operation (chimney capped), however, there was a noticeable reduction in infiltration.

Air flow rates through the uncapped chimney were measured during these tests. As shown in Figure 6, air flow through the chimney during operation of the gas burner varied from 22 to 26 L/s (47 to 55 cfm) as inside-outside temperature difference increased from 22 to 42°K (39.6 to 75.6°R). Figure 6 also indicates that when the burner was off the chimney flow varied from 18 to 23 L/s (38 to 49 cfm). The average flow rate with the burner cycling was about 20 L/s (42 cfm). About 60% of the air exfiltration was through the chimney (Figure 5); the remaining 40% was mainly through the ceiling.

The air infiltration rates of the house with the chimney capped (Figure 5) were, on average, 11 L/s (23 cfm) smaller than those with it uncapped. The difference, which was much smaller than the average 20 L/s (42 cfm) of air flow through the chimney, was due to increased exfiltration resulting from the lowered level of the neutral pressure plane.

Figure 7 shows that with an inside-outside temperature difference of 28 K (50 R) the neutral pressure level changed from 3.6 to 4.8 m (11.8 to 15.7 ft), an increase of 1.33 times. This caused a flow reversal from exfiltration to infiltration at the exterior wall between the two neutral pressure levels, increasing the infiltration through the lower and middle walls and reducing the exfiltration through the upper portion of the house. The slope of the pressure differential profile for the electric furnace was assumed to be the same as that for the gas furnace because the slope is not dependent on the presence or absence of a chimney. As well, because of lower wind speed the scatter in the data for the gas furnace was smaller than that for the electric furnace.

Figure 8 shows that when the electric furnace operated normally, the measured air infiltration rates for wind speeds lower than 3.5 m/s agreed closely with the rates predicted by Eq. (2) (this

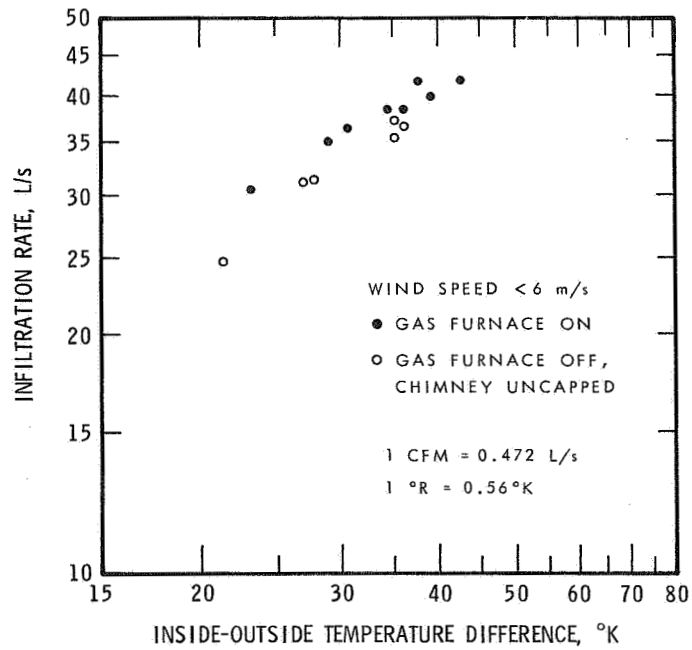


Figure 4 Air infiltration rates under various modes of furnace operation

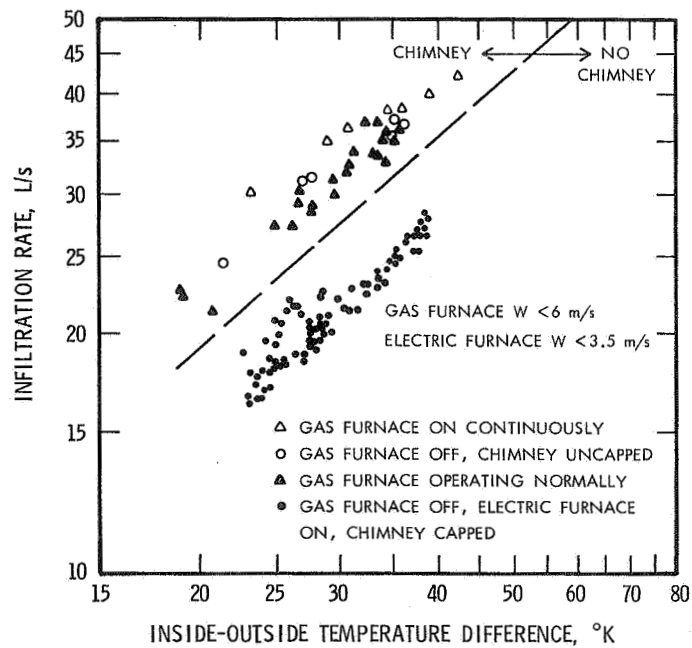


Figure 5 Infiltration rates under controlled and normal furnace operation

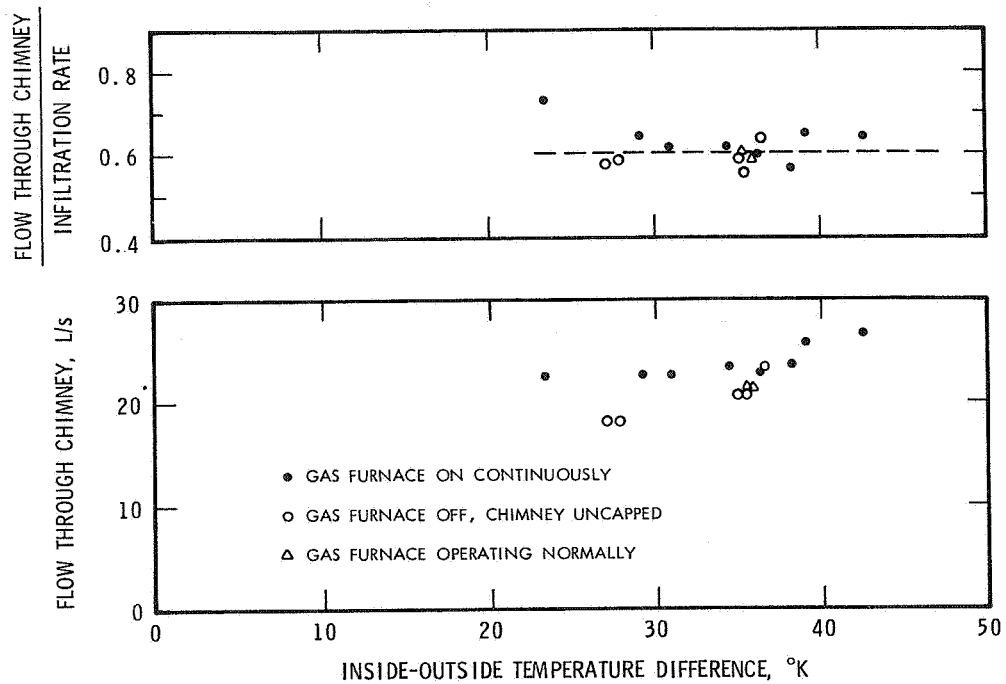


Figure 6 Air flow rate through chimney under natural conditions and ratio between chimney exhaust rate and infiltration rate of house

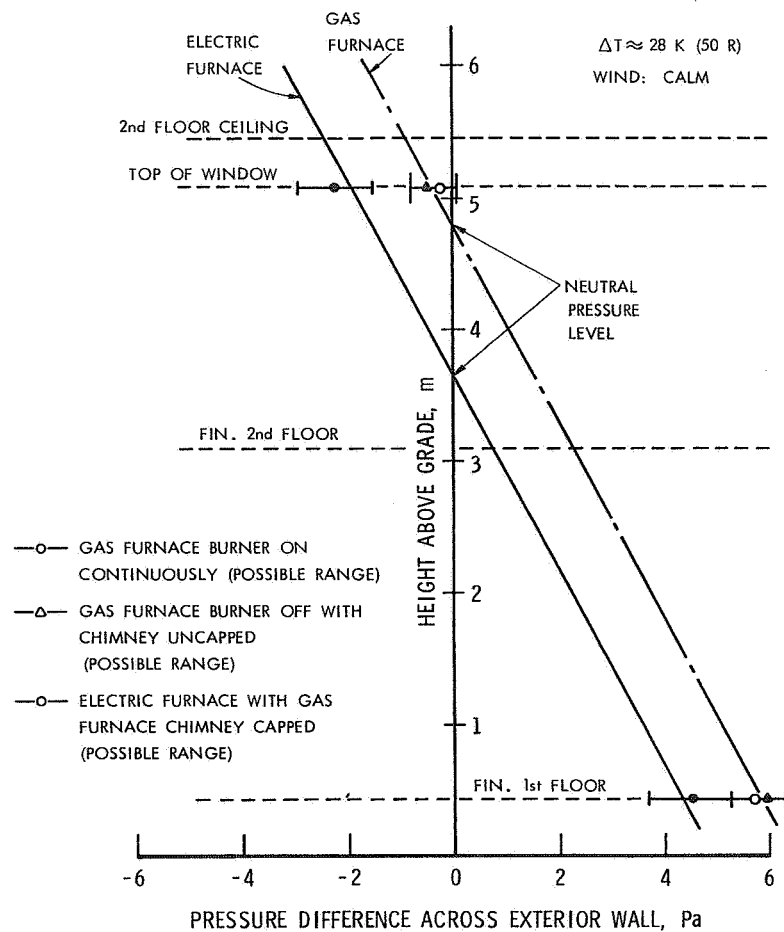


Figure 7 Neutral pressure levels under operation of gas or electric furnace

equation was derived previously for two houses without chimneys, included in the Mark XI project).⁷

$$I = 0.32(A/v) C_e (\Delta t)^n \quad \text{for } w < 3.5 \text{ m/s} \quad (2)$$

where

I = infiltration rate, air changes per hour (ac/h)

A = area of building envelope, m^2 (ft^2)

v = volume of house, m^3 (ft^3)

(386 m^3 (13634 ft^3) for this house)

C_e = flow coefficient with chimney capped, $\text{L/s} \cdot \text{m}^2 \cdot \text{Pa}^n$
($\text{cfm/ft}^2(\text{in. of water})^n$)

Δt = inside-to-outside temperature difference, K (R)

n = flow exponent (0.71 for this house).

For comparison, Figure 8 also shows the infiltration rates for the gas furnace under low wind conditions. Again, the data were fitted to an expression similar to Eq. (2).

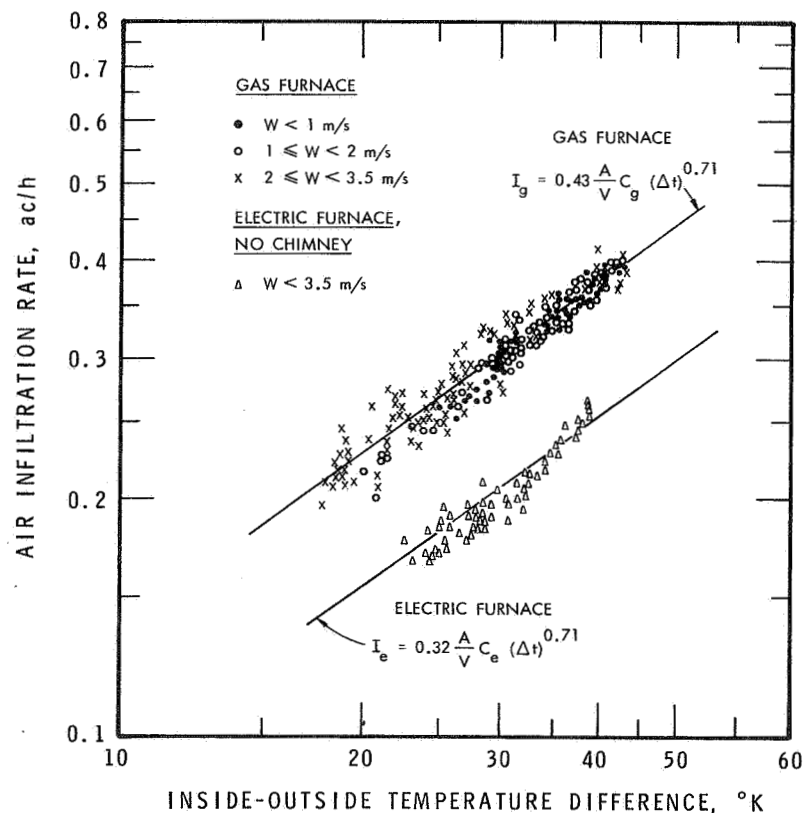


Figure 8 Air infiltration rates for electric and gas furnaces under low wind

$$I = 0.43(A/v) C_g (\Delta t)^n \quad \text{for } w < 3.5 \text{ m/s} \quad (3)$$

where C_g is the flow coefficient with the chimney uncapped and n is 0.71. Figure 8 indicates that the air infiltration for this house with gas-fired furnace operating is about 1.5 times greater than that with an electric furnace. The ratio of the constants in Eqs. (2) and (3) is approximately equal to the ratio of the two neutral pressure levels. As earlier studies also indicate that air infiltration due to stack action is directly proportional to the neutral pressure level,^{8,9} a general expression for Eqs. (2) and (3) would be

$$I = 0.32(A/v) r C (\Delta t)^n \quad \text{for } w < 3.5 \text{ m/s} \quad (4)$$

$$r = H_g/H_e$$

where H_g and H_e are the neutral pressure levels with and without a chimney, respectively. The value of r is 1 for houses without chimneys, but can vary from 1 to 2 for houses with chimneys. For this house, r is approximately 1.33. C and n in Eq. (4) are measured by means of the fan pressurization method with chimneys, if existing, uncapped.

Figure 9 shows the temperature-induced air leakage and pressure differential patterns for this house with chimney capped and uncapped. The air leakage rates were obtained from Figure 8 for an inside-outside temperature difference of 28 K (50 R). The chimney flow rate was assumed to be 60% of the over-all exfiltration rate (Figure 6). As noted previously, the infiltration rate with chimney was 50% greater than that with chimney capped (32 L/s versus 21 L/s). Although over-all exfiltration also was greater when the chimney was uncapped, the amount of inside air leaving the house through the walls and ceiling was smaller. These results suggest two reasons why houses without chimneys have more problems with condensation than those with chimneys:

- 1) for the same internal moisture generation rate, houses without chimneys have higher relative humidity than those with chimneys because there is less infiltration for dilution;
- 2) because there is no chimney venting of the inside air, chimneyless houses have more humid air flowing through the cold parts of the upper walls and ceiling; they are therefore more susceptible to condensation problems.

Unlike electrically heated houses,⁷ air infiltration rates with the gas furnace were influenced by stack action for wind speeds as high as 7 m/s (16 mph). This suggests that for tight houses the strength of stack action is reinforced by the presence of a chimney. Unless, therefore, wind strength also increases proportionately, stack action remains the dominant driving potential for infiltration. The measured air infiltration rates under this condition, as shown in Figure 10, can also be approximately expressed by Eq. (4).

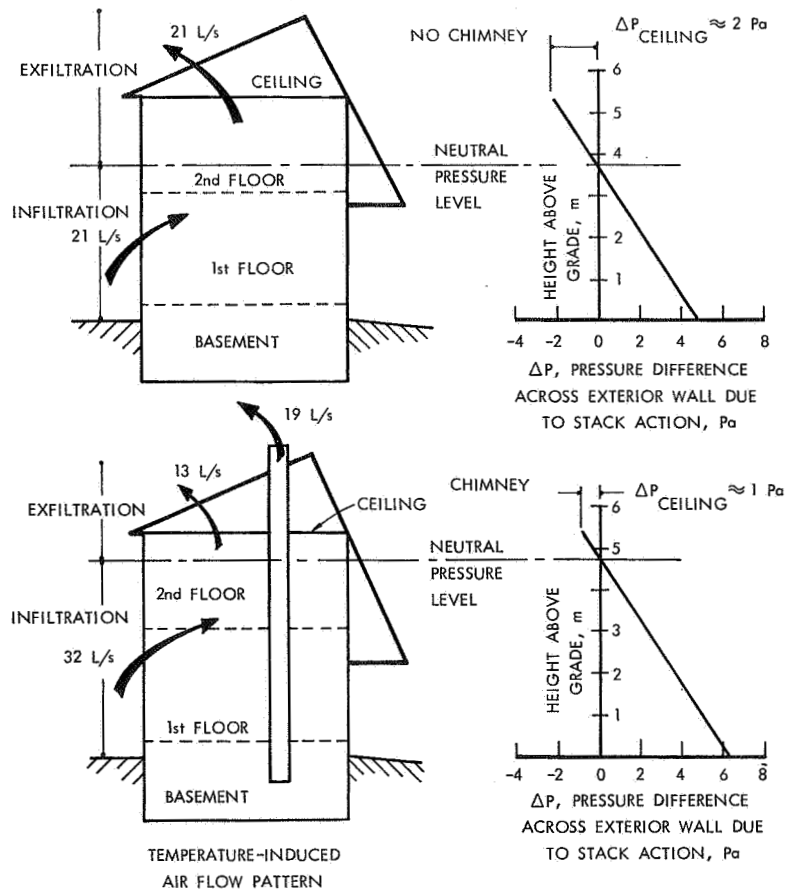


Figure 9 Temperature-induced pressure and air flow patterns under operation of electric or gas furnace for $\Delta t = 28 \text{ K}$

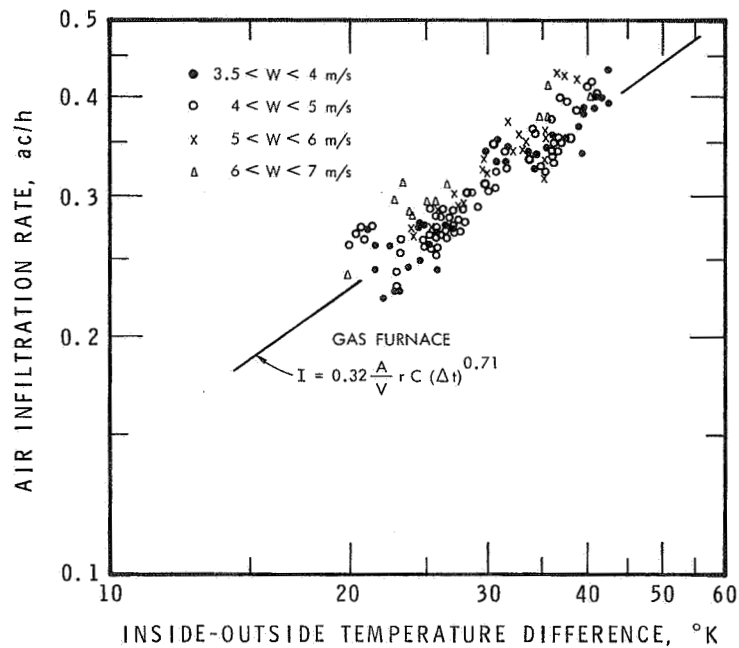


Figure 10 Infiltration rates for gas furnace under moderate wind

5. SUMMARY

The effect of a gas furnace with a Class C circular chimney 12.7 cm (5 in.) in diameter on the air leakage of a two-storey house is as follows:

- (i) The air leakage value of the house with chimney uncapped is about 9% greater than that with chimney capped.
- (ii) The air infiltration rate with the burner on continuously is 10% greater, on average, than it is with the burner cycling or off.
- (iii) About 60% of inside air leaves the house through the chimney and the remaining 40% exfiltrates through the upper portion of the house envelope.
- (iv) Switching from an electric furnace to a gas furnace results in a 50% increase in air infiltration for the test house with wind speeds lower than 3.5 m/s.
- (v) For the test house, stack action reinforced by the exhaust air through the chimney remains the dominant driving potential for infiltration with wind speeds up to 7 m/s and inside-to-outside temperature differences greater than 20 K.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 13

DEFINITION OF VENTILATION EFFICIENCY AND THE EFFICIENCY OF
MECHANICAL VENTILATION SYSTEMS

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Synopsis

Air quality and the related definitions of ventilation efficiency are discussed generally. For ventilation systems in residential buildings a definition of efficiency is suggested that takes into consideration how ventilation air spreads within a dwelling. Included in the definition is the contamination exposure to a homogeneous (fills up the whole space) and dynamically passive and chemically inert source. For other sources it is only an air quality indicator. It is shown that a room-average of the efficiency can be determined by only measuring the concentrations in the exhaust ducts.

Measurements of the efficiency for exhaust, supply and combined systems show that for combined and supply systems the highest efficiency occurs, as expected, in those parts where the air is supplied. For these systems the efficiency is sensitive to the ventilation flow rate, while for the exhaust system the relative efficiency is more or less independent of the ventilation flow rate.

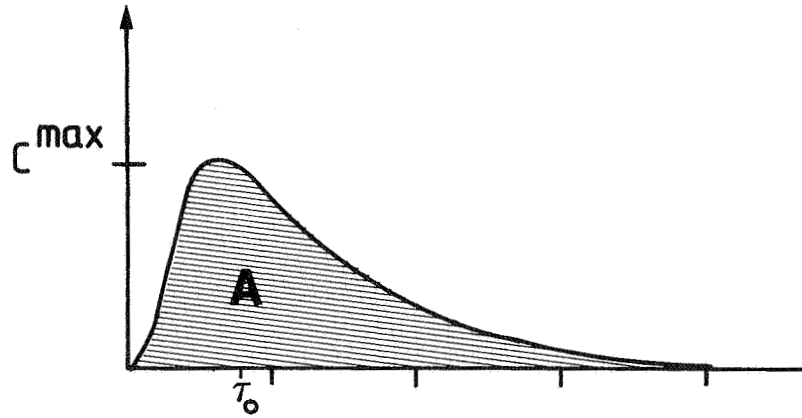
When doors between rooms are closed, the efficiency can be enhanced remarkably by suitable openings between the rooms.

Finally regarding warm-air systems it was found that the ventilation efficiency is very sensitive to the positioning of the supply and exhaust registers, the ventilation flow rate and the relative difference between supply air temperature and room air temperature. The greatest risk with warm-air systems is short-circuiting of both air and heat implying very inefficient systems.

A brief discharge has a duration much shorter than τ
prolonged discharge has a duration much longer than τ

The exposure from a brief discharge is characterized
maximum concentration and the time-integrated exposure
area under the curve, see figure 2.

CONCENTRATION



Ln CONCENTRATION

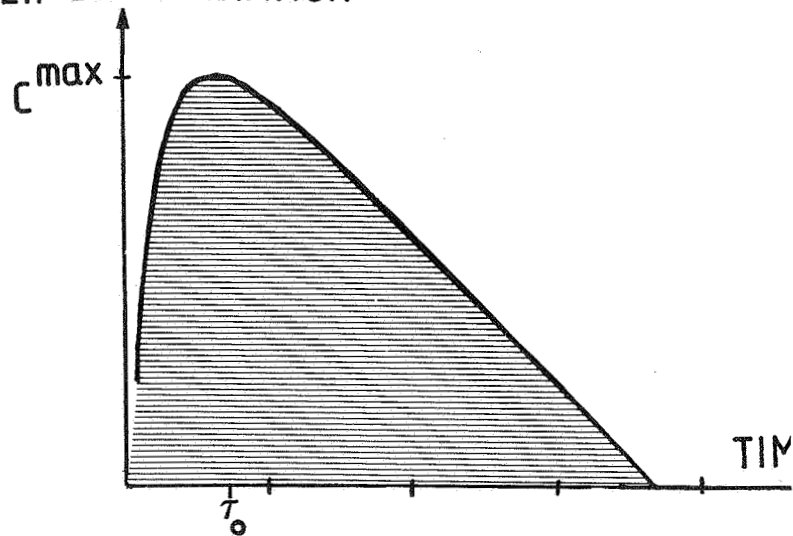


Figure 2. Brief discharge. Concentration versus time constant of the ventilation system.

The exposure from a prolonged discharge is fully characterized by its equilibrium concentration.

2.3 Absorbed dose

The absorbed dose is defined as the amount of pollutant deposited on external and in internal body tissues.

2.4 Effects

Effects on individuals are, among many factors, dependant on their activity levels, the time they spend in their dwellings and their susceptibility to the particular pollutant.

3. DEFINITION OF VENTILATION EFFICIENCY

As stated earlier the purpose of the ventilation system is to limit the contamination exposure from indoor pollutant sources. The definitions used should therefore reflect the systems performance with regard to this objective.

3.1 Stationary method

The contamination exposure from a prolonged discharge is satisfactorily described by its equilibrium concentration. An obvious, and much used, definition of ventilation efficiency is to define a stationary relative ventilation efficiency ϵ^s at an arbitrary point in the dwelling as:

$$\epsilon^s = \frac{C_e - C_s}{C_j - C_s} \times 100 \quad (\%) \quad (2)$$

where C_e is the concentration in the exhaust air duct, C_s is the concentration in the supply air duct and C_j is the concentration in the discussed point in the dwelling. The efficiency obtained by applying the definition (2) is both dependant on the properties of the ventilation system and the pollutant source. The magnitude of the stationary concentrations, and consequently the stationary efficiency, can also be obtained by other methods.

3.2 Transient method

The magnitude of the stationary concentrations at an arbitrary point can be calculated on a basis of the time-integrated exposure from a brief discharge. The following relation is proven theoretically to be valid, see /2/.

$$C_j^S = \frac{\dot{m}}{\left(\frac{m}{A_j}\right)} \quad (3)$$

$\dot{m}$ = The pollutant production rate, kg/s

m = The amount of pollutant discharged in a brief discharge, kg

A_j = The time-integrated exposure obtained in an arbitrary point from a brief discharge, kg/m³.h.

The quantity $\frac{A_j}{m}$ is in reference /1/ called **the transfer index**.

An advantage of using the transient concentration technique is that it is faster than equilibrium methods. After a while the monitored concentration decays exponentially and the remaining part of the exposure is easily calculated, see figure 2.

3.3 Distribution of the supplied air

One of the most important factors determining the air quality is the 'distribution pattern' of the supplied outdoor air within the dwelling. The air should be distributed in a fashion that removes the pollutant as quickly as possible from the dwelling.

The distribution of the supplied air can be described in different statistical/physical terms. One is the mean age of the air, in chemical engineering parlance often called the mean-residence time. The age is defined as the time that has elapsed since the air came into the room. The mean age of the air at an arbitrary point is easily measured by the tracer gas technique. One possible method is a 'purge' run. The whole space is filled with tracer gas and mixed by mixing fans to a uniform concentration. The mixing is stopped and the decay of the gas concentration is monitored at the selected point. By using the initial concentration ($C(0)$) and the area A_j under the curve, i.e. the time-integrated exposure, we calculate the mean age $\bar{\theta}_j$ at the actual point as:

$$\bar{\theta}_j = \frac{A_j}{C(0)} \quad (h) \quad (4)$$

If we invert (4) we get the local ventilation rate r_j :

$$r_j = \frac{1}{\bar{\theta}_j} \quad (1/h) \quad (5)$$

and if we finally multiply this by the volume, V , of the dwelling we get the effective flow rate:

$$Q_j^{eff} = V \times r_j \quad (m^3/s) \quad (6)$$

When the mixing is complete the effective flow rate at each point is equal to the supplied flow rate and the local ventilation rate is equal to the nominal air change rate n (supplied flow rate divided by the volume of the room). The mean value of the mean age for the whole dwelling can be obtained by monitoring the gas concentration in the extraction duct, see Appendix 1.

The area under the tracer gas curve gives us in this case, according to (3), the stationary concentration from a pollutant source that fills up the whole space. For such sources knowledge about the local ventilation rate is sufficient for determining the air quality. For other sources the local ventilation rate is only an air quality indicator.

The reference 'Air distribution pattern', though far from the best, is complete mixing. Therefore we define a transient relative ventilation efficiency as

$$\varepsilon_j = \frac{r_j}{n} \times 100 \quad (\%) \quad (7)$$

where n is the nominal air change rate.

An alternative term for the efficiency (5) could perhaps be 'the air distribution efficiency'.

When the mixing is complete the efficiency (5) is 100%, and the theoretical upper limit of the efficiency occurs for 'piston flow'. In case of 'short-circuiting' the efficiency is lower than 100%.

3.4 Characterization of pollutants in dwellings

The indoor environment normally includes a complex mixture of pollutants. Apart from cooking odours, the pollutant source in residential buildings normally has a spatially diffuse character. Examples of diffuse sources are people moving around, formaldehyde evaporating from chipboard or radon emitted from the building material.

A designer of a ventilation system for a residential building does not normally know in advance which pollutants the ventilation system is to remove. Furthermore, in national standards the demands on the ventilation are not stated in terms of limitation of contamination exposure but as a minimum ventilation rate. Therefore it is felt that the source-independent efficiency defined by (5) is a suitable basis for selecting a ventilation system.

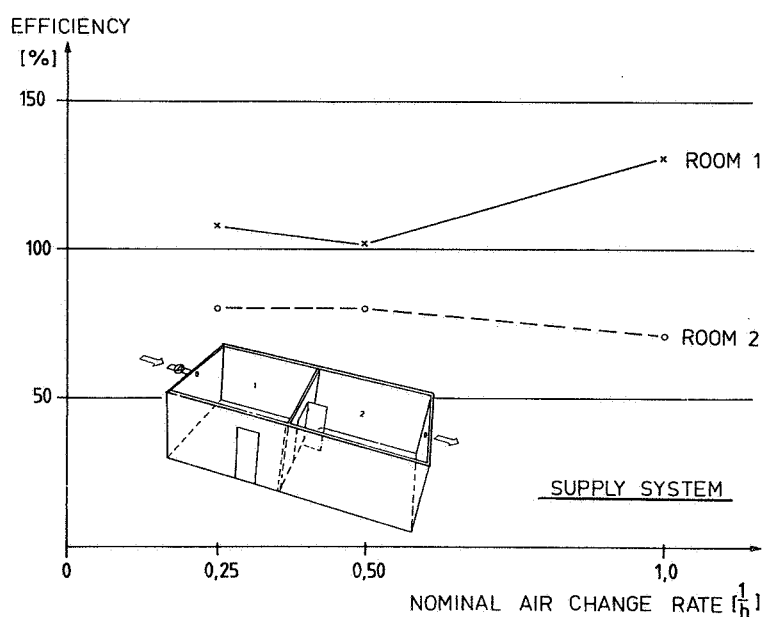
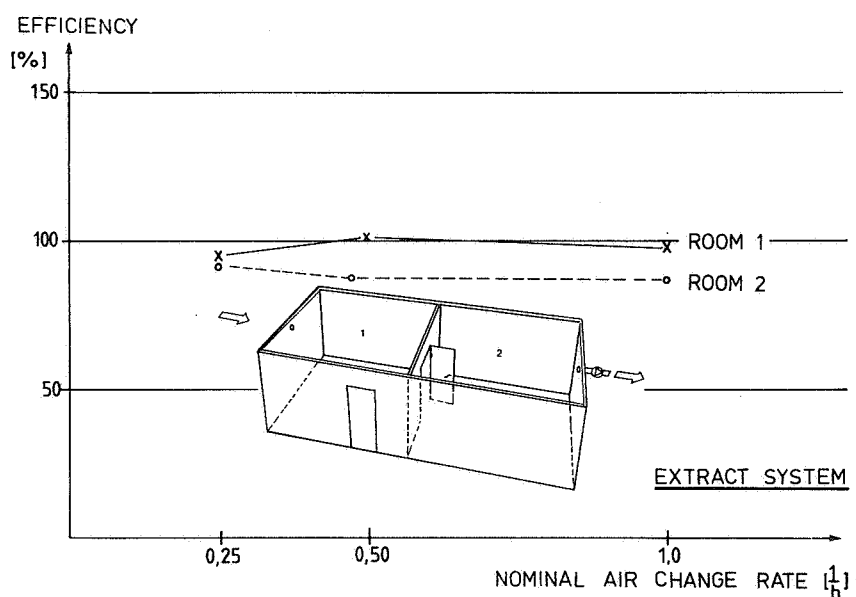
4.0 MEASUREMENTS

The ventilation efficiency and in some cases the temperature efficiency (to be defined below) of different systems have been monitored in the Laboratory for Heating and Ventilation.

For all results presented below the definition (7) of the ventilation efficiency, i.e. the transient relative efficiency, has been used. Some of the results presented below have earlier been presented in references /3, 4/.

4.1 A comparison between different whole-house ventilation systems

A small house measuring (width x length x height) 4.55 x 8.42 x 2.75 m with two rooms of equal size has successively been equipped with an extract, supply and a combined system respectively. The efficiency has been monitored under isothermal conditions in one point in the middle of each room at about 1.7 m above the floor level. The nominal ventilation rate, n , has been varied systematically between 0.25 - 1 air changes per hour. The results are shown in figure 3.



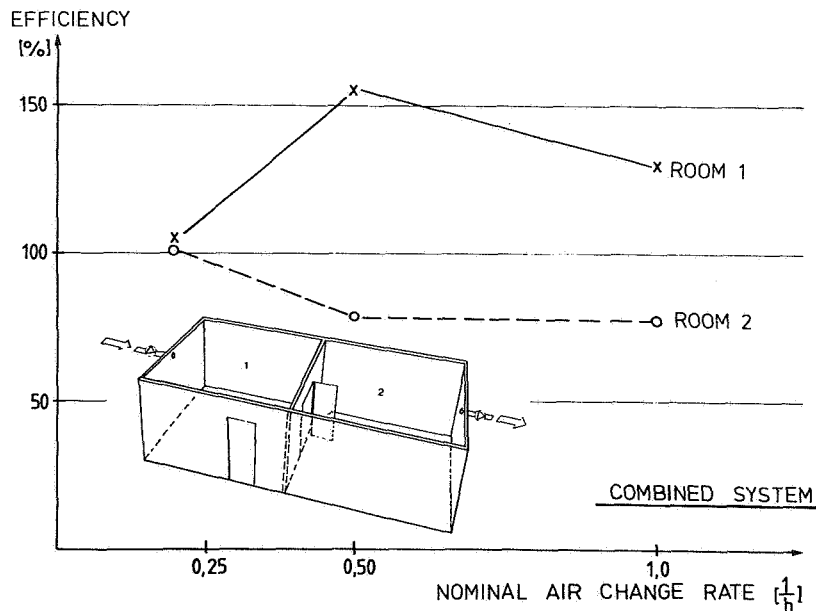


Figure 3. A comparison between extract, supply and a combined system respectively. Door between rooms open.

Figure 3 shows that the extraction system gives about the same efficiency in both rooms and the relative efficiency is rather independent of the nominal ventilation rate. This is according to what we should expect. The under-pressure created by the ventilation system gives rise to an inflow of air through the evenly distributed leakages in the building material. However the air meets the least resistance in the purpose provided opening in room 1 and therefore we get a slightly higher efficiency in this room.

The supply system gives a much larger efficiency difference between the two rooms. This is due to the fact that the supply of air is now concentrated to the supply register in room 1.

The combined (balanced) system gives, apart from the lowest ventilation rate, the highest efficiency of all systems.

4.1.1 Combined system with both supply and extraction in the same room

Figure 4 shows the monitored efficiency when the air is both supplied to and extracted from the same room.

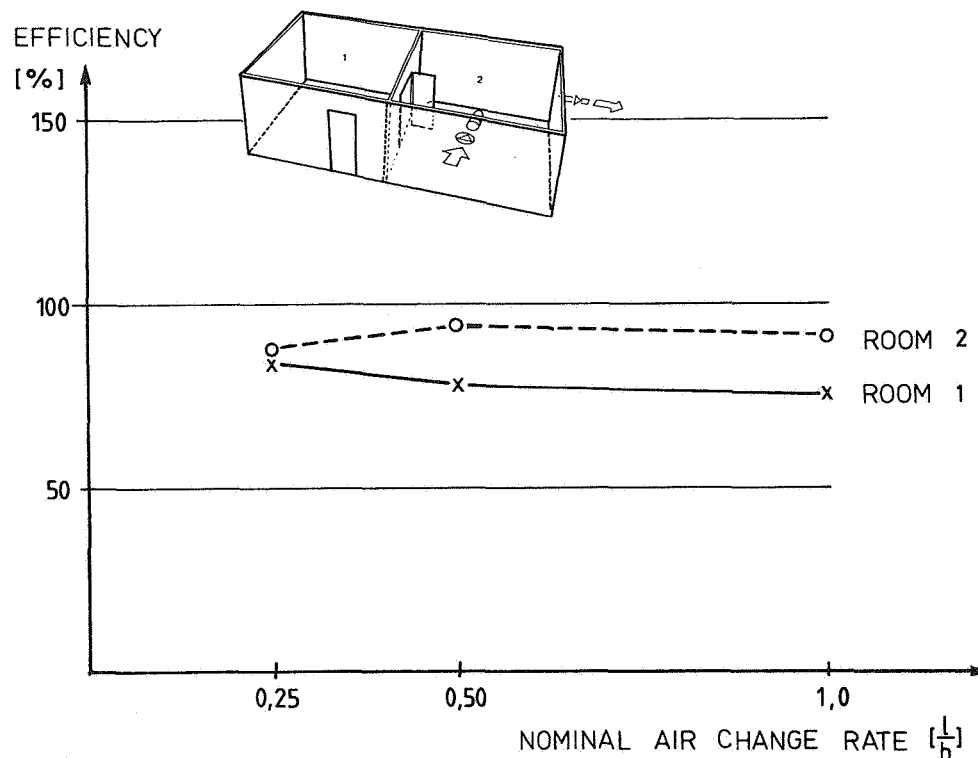


Figure 4. Combined system. Supply and extraction registers in the same room. Door between rooms open.

With both the supply and extraction registers placed close to each other the best possible performance we can expect is a uniform dispersion of the air throughout the dwelling. In other words, it is a question of minimizing short-circuiting as much as possible. The efficiency is, as expected, highest in room 2, where the air is supplied, but the efficiency in room 1 is surprisingly high.

4.1.2 Combined system with supply and extraction of air in different rooms. Door between the rooms closed

In this case the door is closed and the air is transferred between the rooms through a narrow slot under the door. The efficiencies obtained are shown in figure 5.

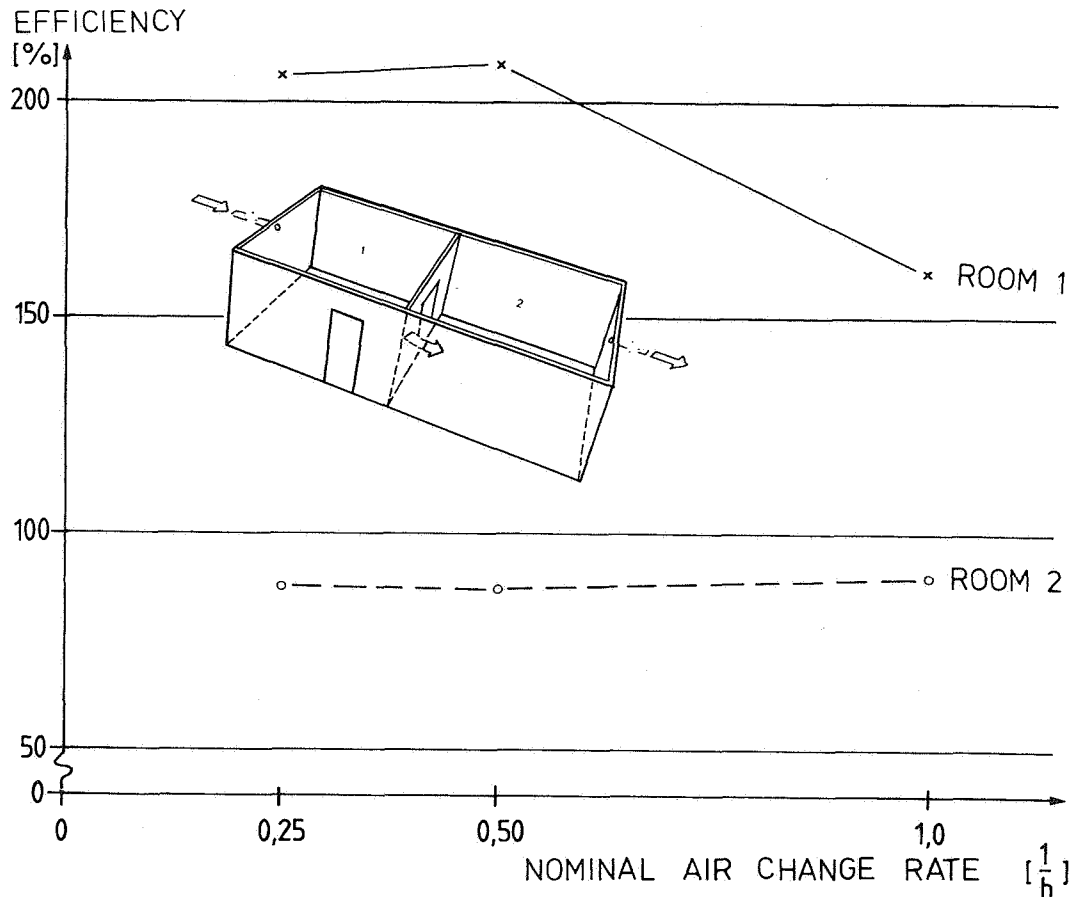


Figure 5. Combined system. Door between rooms closed. A narrow slot under the door.

Compared to the case shown before, i.e. with the door open, the efficiency is now strikingly higher in room 1. The explanation is how the slot under the door works. The area of the slot is so small, and accordingly the air velocity high, that it acts like a one-way valve for the air. The air is only transferred from room 1 to room 2 and not in the opposite direction. This implies that the dwelling is now ventilated 'in sequence' and we achieve a quick removal of the gas. This is an illustrative demonstration that a ventilation system's main purpose is to remove the pollutant as quickly as possible and not to dilute the contaminant.

4.2 Warm-air systems

In a warm-air system the ventilation system has a dual task. It must both provide heat and 'fresh' air in the occupied zone. The ability to supply air is, as before, expressed by definition (5) of the ventilation efficiency. A ventilation system's ability to supply heat may be expressed by a temperature (thermal) efficiency, ε_T , similar to the definition (2) of the stationary ventilation efficiency:

$$\epsilon_T = \frac{T_e - T_s}{T_j - T_s} \times 100 \quad (\%) \quad (8)$$

T is the stationary temperature.

A small temperature difference between the supply air and the occupied zone implies a high utilization of the energy provided. The measurements have been made in a room measuring (width x length x height) 3.6 x 4.2 x 2.5 m. The occupied zone is defined as the volume up to 1.8 m above the floor level.

4.3 System A

In this system the supply as well as exhaust air registers were placed above the occupied zone. This system is a typical short-circuiting scheme or unidirectional air flow system. Regarding the transient ventilation efficiency the best we can achieve is to disperse the supplied air uniformly in the occupied zone, i.e. complete mixing (100% efficiency) is the best feasible. The results obtained are shown in figure 6.

EFFICIENCY

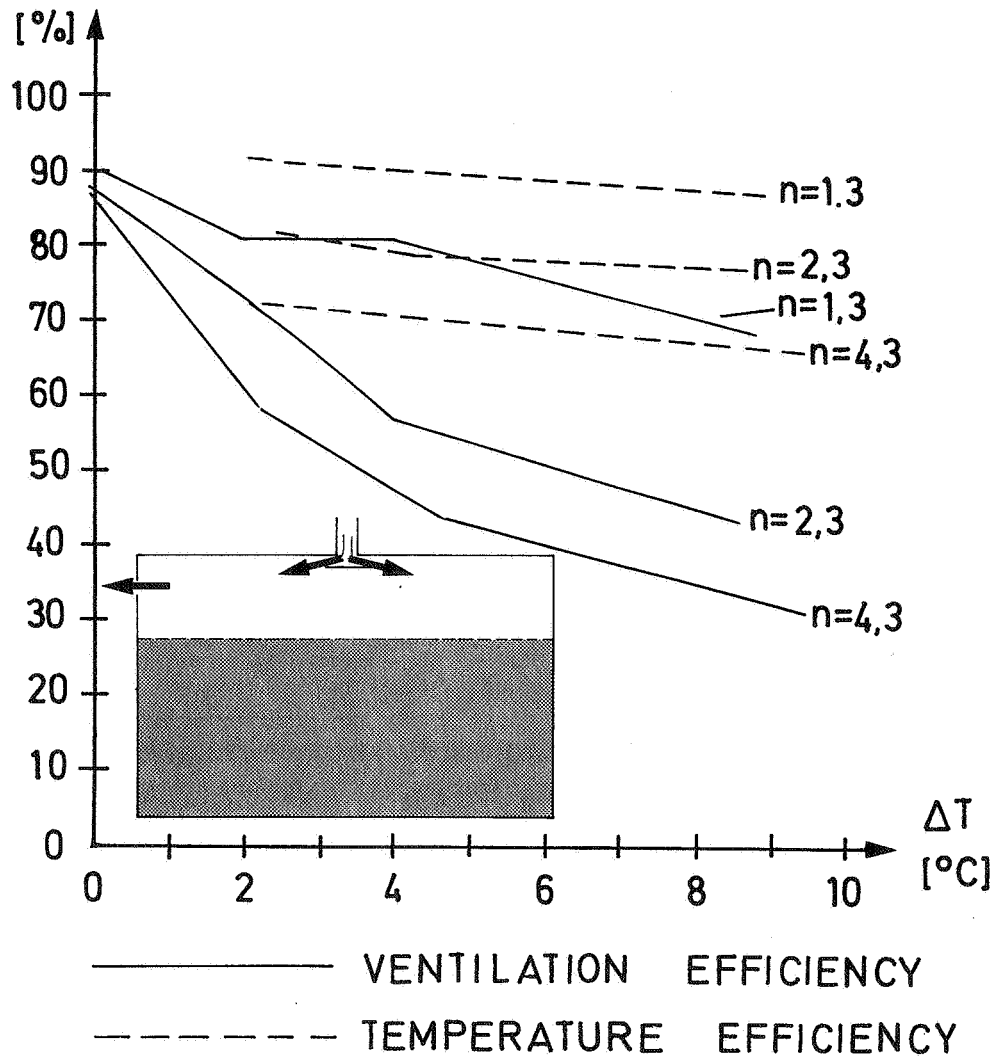


Figure 6. The transient relative ventilation efficiency and the stationary temperature efficiency in the occupied zone (shaded area).

ΔT is the temperature difference between the supply and extraction duct.

n is the nominal air-change rate.

Figure 6 shows a heavy thermal stratification effect with an increase in the supply air temperature. The increase in the nominal air change rate affects the ventilation efficiency in a perhaps somewhat unexpected manner. The ventilation efficiency decreases when the ventilation air flow is increased.

The temperature efficiencies show a similar pattern.

In this system the exhaust register has been moved to the floor level. With this placing of the supply and exhaust registers complete mixing is the poorest operational mode. The result obtained for this system is shown in figure 7.

EFFICIENCY

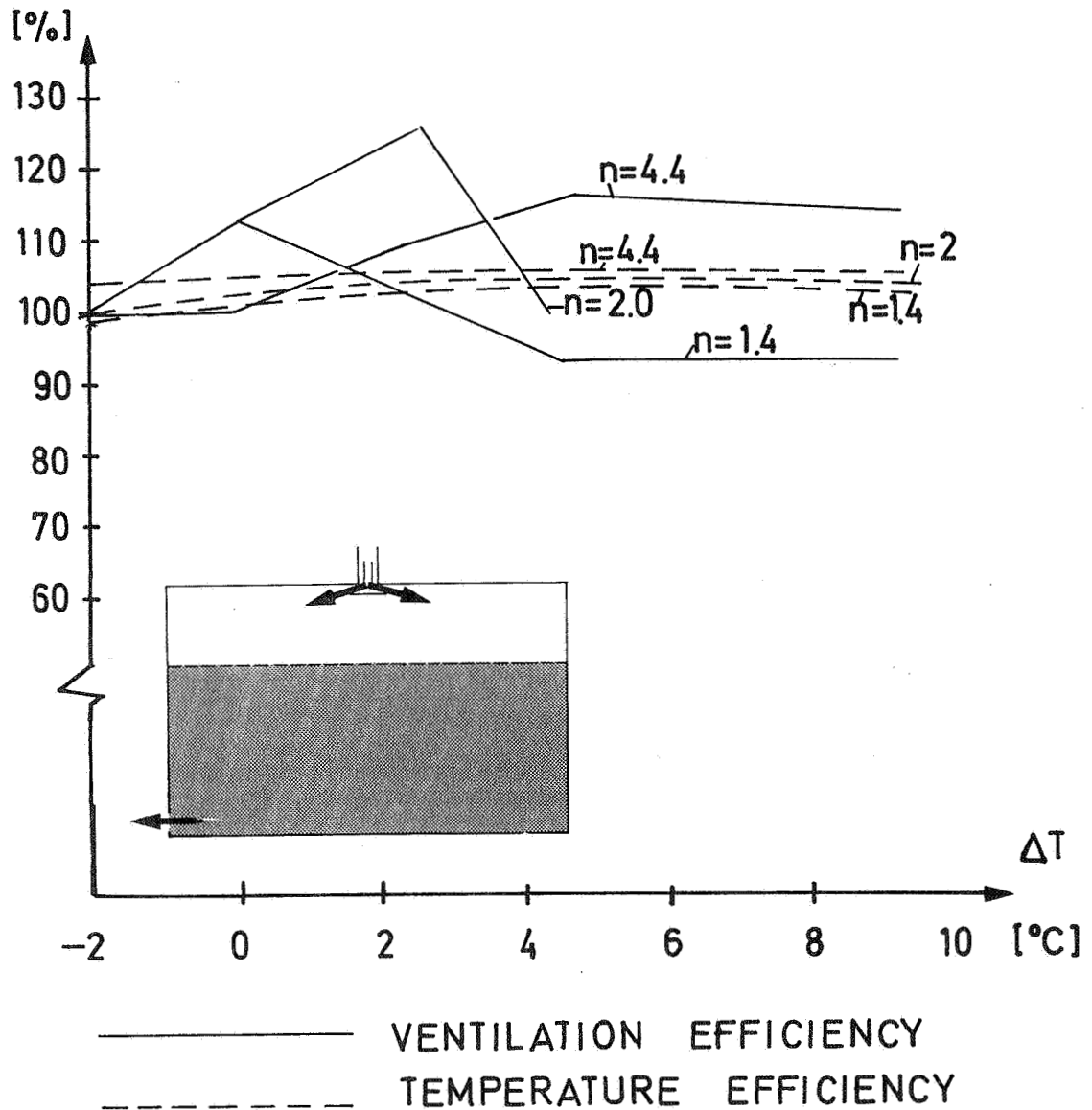


Figure 7. The transient relative ventilation efficiency and the stationary temperature efficiency in the occupied zone (shaded area).

ΔT is the temperature difference between the supply and extraction duct.

n is the nominal air-change rate.

Figure 7 shows that we almost always obtain a ventilation efficiency greater than 100%. The combined high nominal air change rate and high over-temperature give an exceptionally high ventilation efficiency. The explanation is the buoyancy effect and placing of the registers. Due to the buoyancy the supplied air sticks to the ceiling, but due to the placing of the registers the air is forced to pass through the whole room. The resulting flow is similar to a plug flow.

The temperature efficiency is just above 100%.

4.5

System C

In this system air is supplied at floor level next to the wall and is blown vertically up the wall. The exhaust air register is placed in the middle of the ceiling. This is again an undirectional air flow system and we should expect a ventilation efficiency greater than 100%. The results obtained are shown in figure 8.

EFFICIENCY

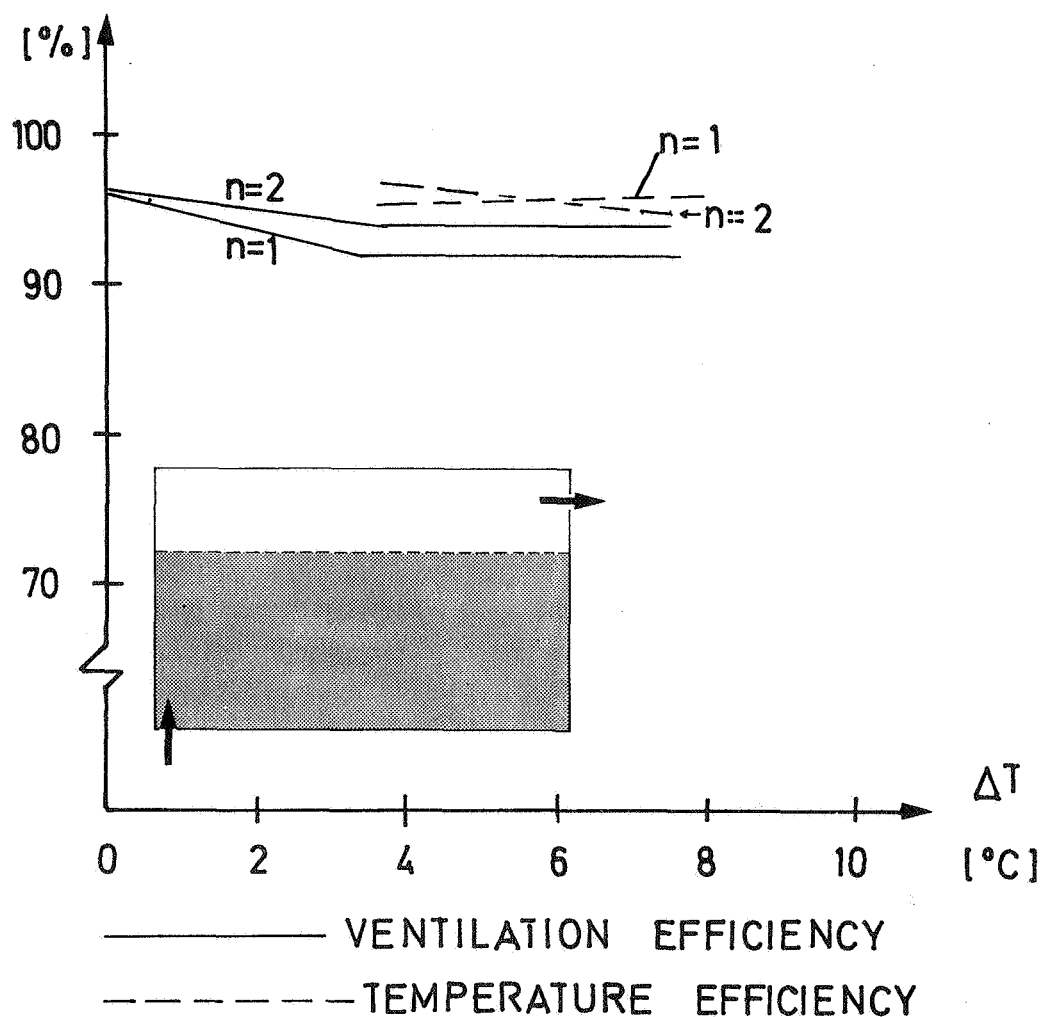


Figure 8. The transient relative ventilation efficiency and the stationary temperature efficiency in the occupied zone (shaded area).

ΔT is the temperature difference between the supply and extraction duct.

n is the nominal air-change rate.

Figure 8 shows that we get a ventilation efficiency less than 100% which is less than the expected efficiency. The explanation is probably a short-circuiting effect due to the fact that the supplied air partly goes as a wall jet directly to the extract air register. The temperature efficiency is however about 100%.

4.6 System D

The placing of the register is the same as for system D, but now the supplied air is blown horizontally along the floor. The results obtained are shown in figure 9.

EFFICIENCY

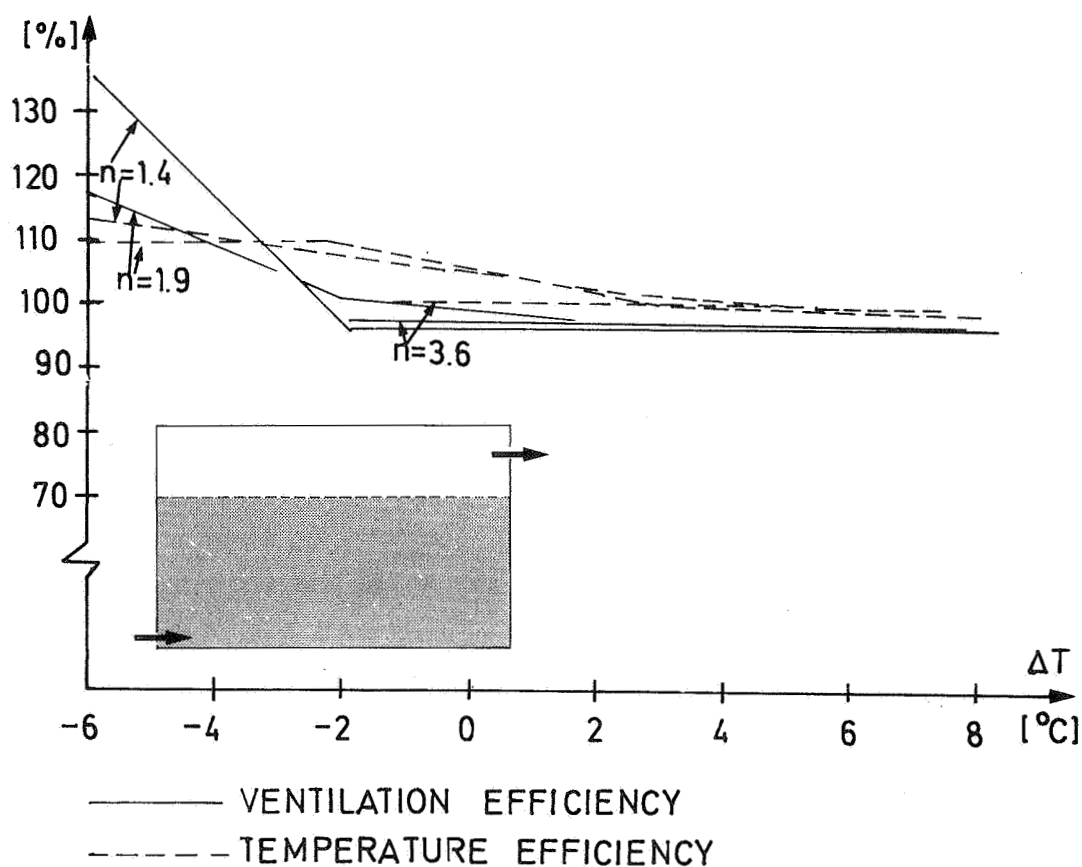


Figure 9. The transient relative ventilation efficiency and the stationary temperature efficiency in the occupied zone (shaded area).

ΔT is the temperature difference between the supply and extraction duct.

n is the nominal air-change rate.

Figure 9 shows that in the case of cooling the ventilation efficiencies are much greater than 100%. The explanation is the same as when system 2 is run as a warm air system; it can be ascribed to the buoyancy effect and the position of the registers.

Similar performances of warm-air systems have also been reported in reference /5/.

APPENDIX

Relationship between room-averaged ventilation rate and the moments of the concentration distribution in the exhaust duct

It will be supposed that the pollution (tracer gas) is transported by convection and diffusion (laminar and turbulent). Furthermore it is assumed that the velocity field is stationary. The concentration C at an arbitrary point (x, y, z) in the room obeys the conservation (transportation) equation:

$$\partial C / \partial \tau = - \nabla \cdot (\underline{U} C - D \nabla C) \quad (1)$$

where

$\underline{U}(x, y, z)$ = Local velocity vector

$D(X, Y, Z)$ = Local diffusivity

$\nabla = (\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z})$ = Nabla operator

τ = Time

The velocity and diffusivity vary with position but not with time and concentration.

The boundary conditions at the walls are (no mass flux):

$$D \frac{\partial C}{\partial n} = 0 \quad (2a)$$

where

$\partial C / \partial n$ denotes differentiation along the normal to the wall.

Furthermore we assume that the concentration in the supply duct is zero.

Generally, at an arbitrary point, we define with regard to time the moments, $\mu^{(p)}$, of the concentration distribution as:

$$\mu^{(p)}(x, y, z) = \int_0^{\infty} \tau^p \cdot C(x, y, z; \tau) d\tau \quad (3)$$

The 0:th moment is the integrated exposure or area under the curve. The condition to be imposed on C as $\tau \rightarrow \infty$ is thus that these moments should exist and be finite.

Multiplying equation (1) by τ^p and integrating from 0 to ∞ , we have

$$-p \mu^{(p-1)} + \nabla \cdot (\underline{U} \mu^{(p)} - D \nabla \mu^{(p)}) = 0 \quad p = 1, 2, 3 \quad (4)$$

and the condition (2a) becomes:

$$D \mu(p) / \partial n = 0 \quad (2b)$$

The moments in the supply duct are zero. To simplify we assume that there is only one exhaust duct. By integrating the equation (4), subject to condition (2b), over the whole room volume, V , the room average $\langle \rangle$ of equation (4) becomes

$$\langle \mu(p-1) \rangle = \frac{1}{p} \frac{Q}{V} \mu_e^{(p)} \quad (5)$$

where

$$\langle \mu(p) \rangle = \frac{1}{V} \iiint \mu(p) (x, y, z) dx dy dz$$

$$\mu_e^{(p)} = p\text{:th moment in the extraction duct.}$$

Q = Volumetric flow rate of air

The relation (5) shows that in principle we can retrieve the room average of any moment by monitoring the gas concentration in the exhaust duct.

If, for example we apply the relation (5) to a 'purge' run, i.e. the room is initially filled with tracer gas to give a uniform concentration $C(0)$, we get the room-averaged ventilation rate. To see this we put $p = 1$ and divide both sides in equation (5) by the initial concentration $C(0)$

$$\langle \bar{\theta} \rangle = \frac{\mu_e^{(1)}}{C(0)} = \frac{Q}{C(0) \cdot V} \quad (6)$$

Because all tracer gas must pass the extract duct we obtain the equation:

$$Q \cdot \int_0^{\infty} C_e(\tau) d\tau = C(0) \cdot V$$

or

$$\frac{Q}{C(0) \cdot V} = \frac{1}{\int_0^{\infty} C_e(\tau) d\tau}$$

Therefore we can write $\langle \bar{\theta} \rangle$ as

$$\langle \bar{\theta} \rangle = \frac{\mu_e^{(1)}}{\mu_e^{(0)}} = \frac{\int_0^{\infty} C_e(\tau) \cdot \tau \cdot d\tau}{\int_0^{\infty} C_e(\tau) d\tau} \quad (7)$$

and its inverse, the room-averaged ventilation rate $\langle r \rangle$ as

$$\langle r \rangle = \frac{1}{\langle \theta \rangle} \quad (8)$$

Relation (8) shows that by monitoring the gas concentration in the extraction duct and calculating the first moment and the area of the concentration distribution we can retrieve the room-averaged ventilation rate. An experimental verification of relation (8) is shown in figure 1.

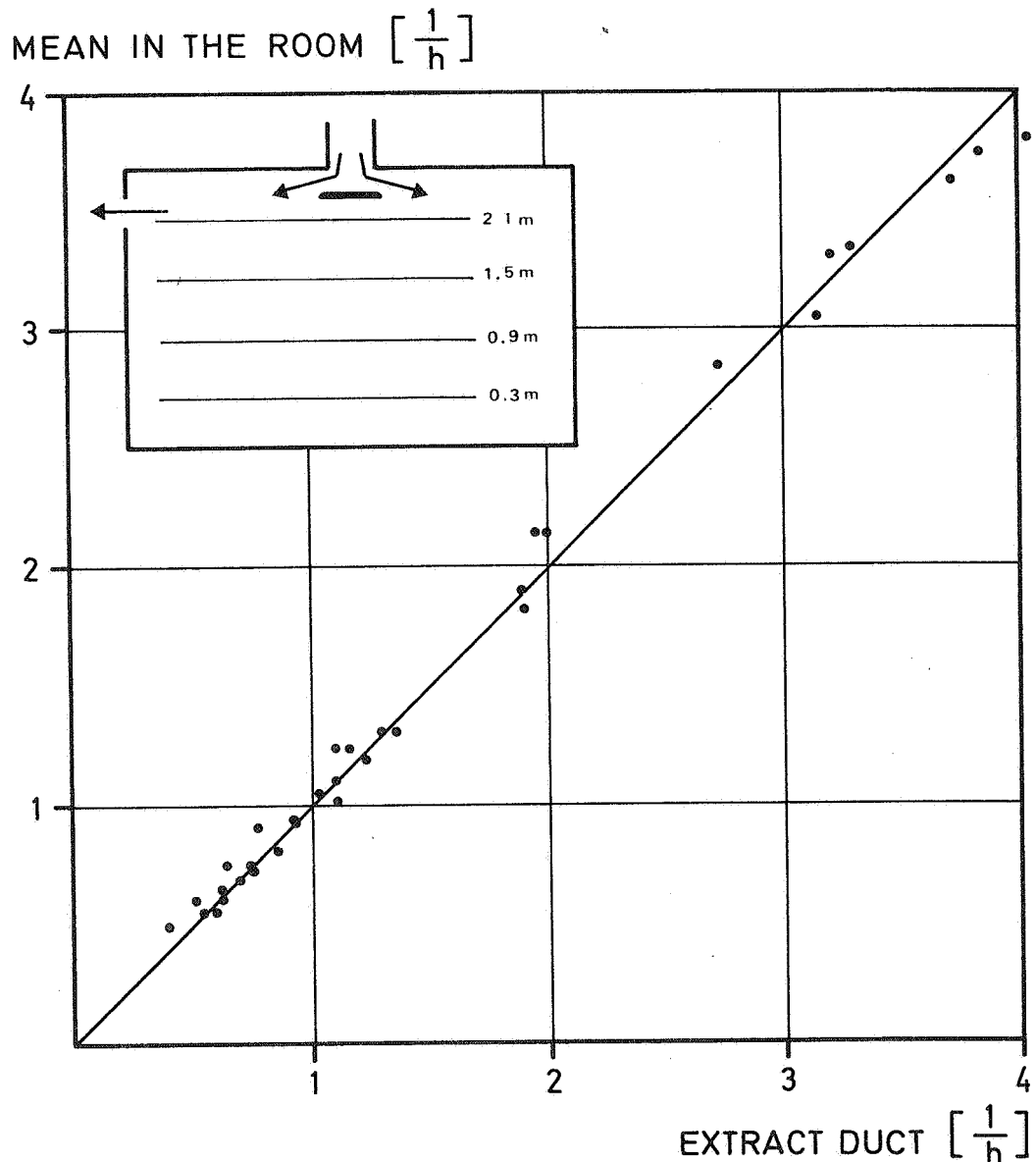


Figure 1.

Horizontal axis: Room-average ventilation rate obtained by measurements in the extraction duct and applying the relation (8) above.

Vertical axis: Room-average ventilation rate obtained by calculating the mean-values of the local ventilation rate at the four indicated levels.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 14

FIELD TRIALS OF VENTILATION EFFICIENCY IN BUILDINGS EQUIPPED
WITH MECHANICAL VENTILATION SYSTEMS

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Synopsis

By measuring the performance of ventilation systems in existing buildings we can discover systems not working properly. The gained experience from field trials gives us insight into how different systems work. This can help us avoiding to install bad systems in newly built houses.

Results from three case-studies regarding the ventilation efficiency are reported? In one of the cases it was possible to simulate three different positions of the supply register. In another case the supply air temperature was varied. The importance of both the location of the supply register and the supply air temperature are demonstrated.

INTRODUCTION

Today the designer of a ventilation system must take into consideration the cost of excessive ventilation rates. Lower ventilation rates imply that the designer also must be aware of the supplied air for the removal of pollution. Efficient utilization of the air means a high ventilation efficiency. With the today's measurement technique it is possible to study how the air is spread within a room.

The purpose of this paper is to report results from field trials of the performance of various mechanical ventilation systems. The measurements have been carried out in buildings built during the seventies. All buildings were equipped with mechanical supply and extract systems.

In one of the tests it was possible to change the position of the supply register and in another test the supply air temperature was varied. All tests were carried out with the rooms in their normal use and with the doors between the rooms closed.

2. MEASUREMENT TECHNIQUE

The tests were carried out by adoption tracer gas technique. A burst of gas (N_2O) was admitted to the room and with the aid of mixing fans the gas was mixed to a uniform initial concentration C_0 . Then the fans were turned off and the decay of the concentration was recorded at several points.

The local ventilation rate (LVRT) at an arbitrary point j was calculated as:

$$r_j = \frac{C_0}{A_j}$$

when

C_0 = Initial concentration

A_j = Total area under the curve

The total area was calculated as (see figures 1 and 2)

$$A = A_{T_0} + A_T$$

A_{T_0} = measured area

$$A_T = \frac{C_{T_0}}{\lambda}$$

C_{T_0} = Final concentration

λ = Slope of $\ln$ concentration curve in the exponential decay region, see figure 2.

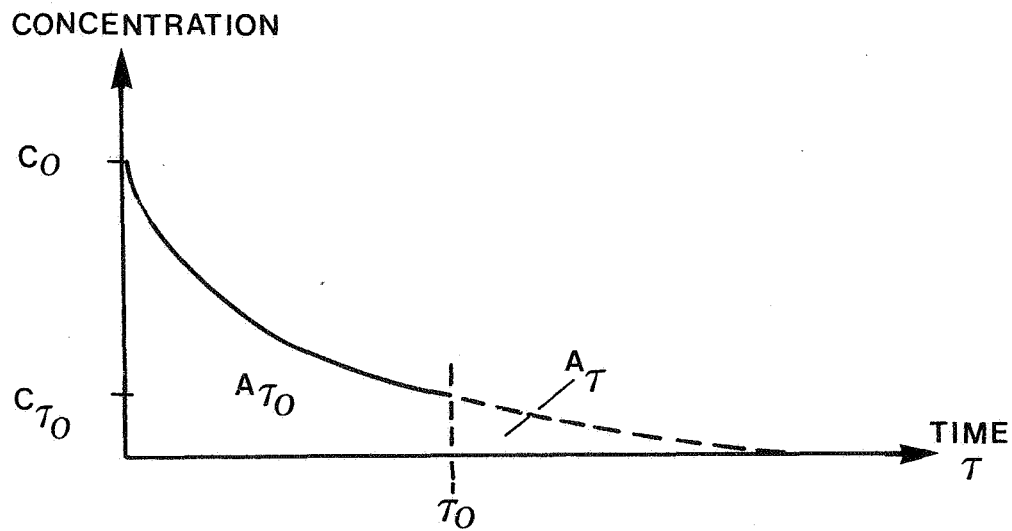


Figure 1. Concentration versus time

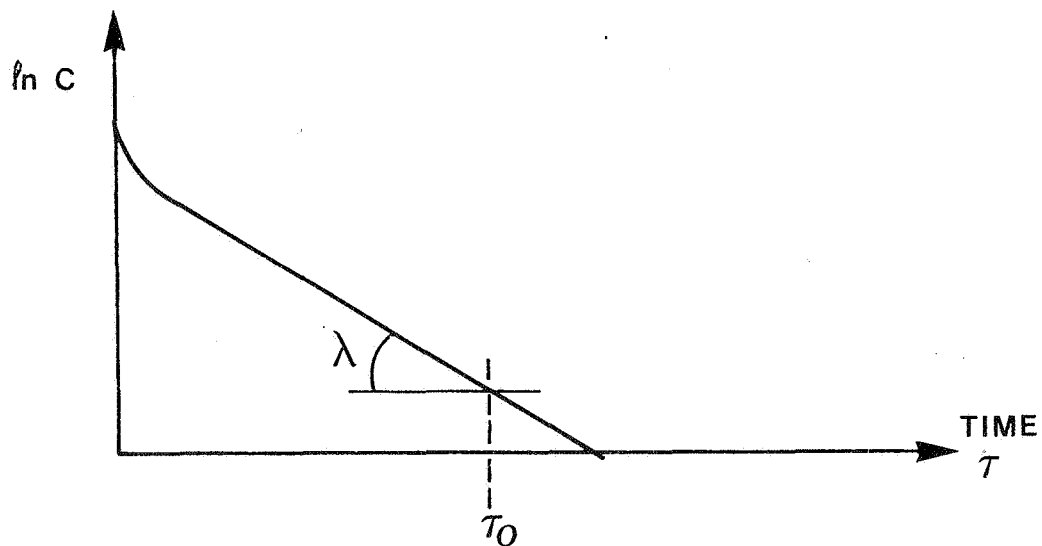


Figure 2. Concentration versus time in a linear/logarithmic diagram

The nominal ventilation rate, defined as the total outdoor air entry into the room directed by the volume of the room, was measured in exactly the same manners as described above, except that the mixing fans now were in operation the whole time.

The ventilation efficiency, ϵ_j , at air arbitrary point j is defined as

$$\epsilon_j = \frac{r_j}{n} \times 100 \quad (\%)$$

n is the nominal ventilation rate.

3. RESULTS

3.1 Ventilation efficiency as a function of the location of the supply register

The room is shown in figure 3 and the air is normally supplied into the room at two points. Above the door there is an ordinary supply register and under the door there is a slot where air also enters.

The efficiency has been measured in four points, see figure 3.

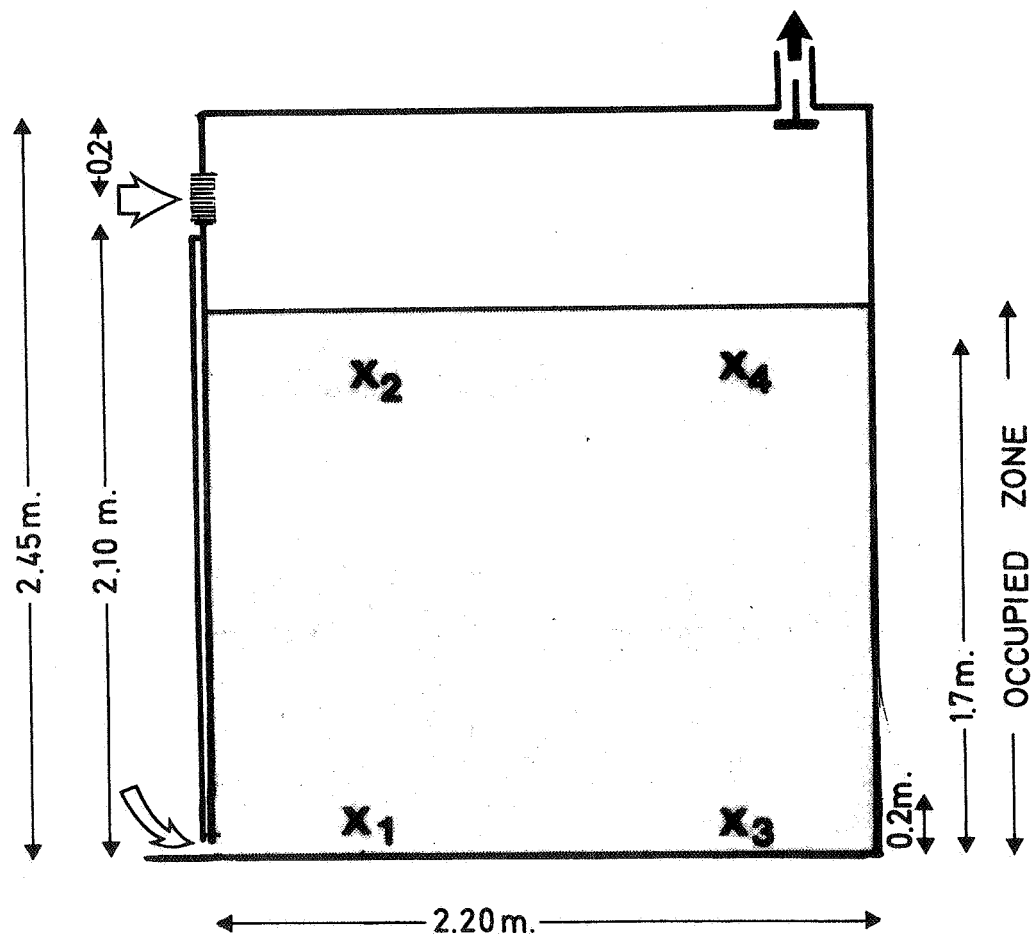


Figure 3. Test room with measurement points

Beside the normal operational mode two other cases have been simulated. The supply register above the door and the slot under the door have alternatively been taped. The cases are

Case a: Air supplied only through the register above the door.

Case b: Air supplied both through the register and the slot under the door.

Case c: Air supplied only through the slot under the door.

The measurements were carried out under isothermal conditions. Obtained results are shown in diagram 1 below.

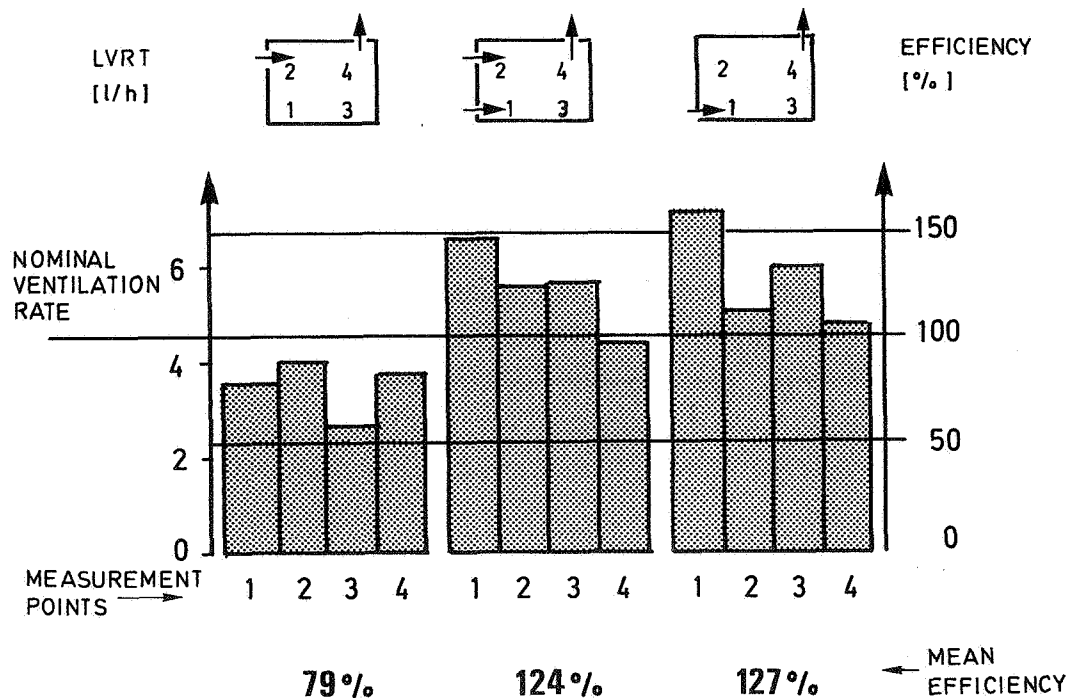


Diagram 1. Ventilation efficiency in three cases

From the results in diagram 1 we can conclude.

Case a) which is the most common placing of registers, gives a mean ventilation efficiency (79 %) which is less than the nominal ventilations efficiency (100 %). This can be ascribed to a short-circuiting effect.

Case b) gives a satisfactory mean ventilation efficiency (124 %).

Case c) gives the highest ventilation efficiency (mean 127 %) of all the cases.

By comparing the cases we can conclude that the simplest arrangement, a slot under the door, gives the highest efficiency. Therefore the cost for both registers and labour could have been saved in this particular case.

3.2 Ventilation efficiency as function of supply air temperature

Figure 4 shows the test room in this case. The room is a hospital ward room. Both supply and extract registers are located in the ceiling. The supply air is blown horizontally against the centre of the room. The room is heated with panel radiators located under the windows.

The tests were carried out with temperature difference between supply air and the mean room temperature of -1.8°C (case d) and $+3.0^{\circ}\text{C}$ (case e) respectively. In the later case the radiator heating was on.

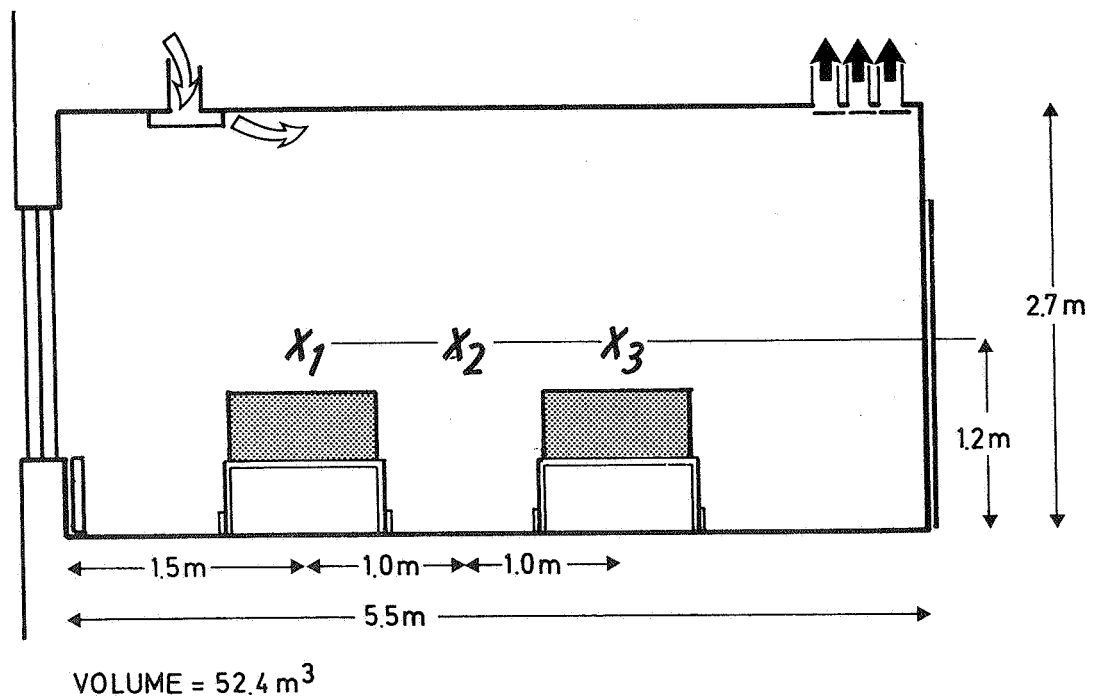


Figure 4. Test room with measurement points

The obtained results appear in diagram 2 below.

When undertemperature occurs the result is according to what one should expect. Case e shows that the air motion cauted by the radiators is not sufficient to prevent the short-circuiting effect due to the higher supply air temperature.

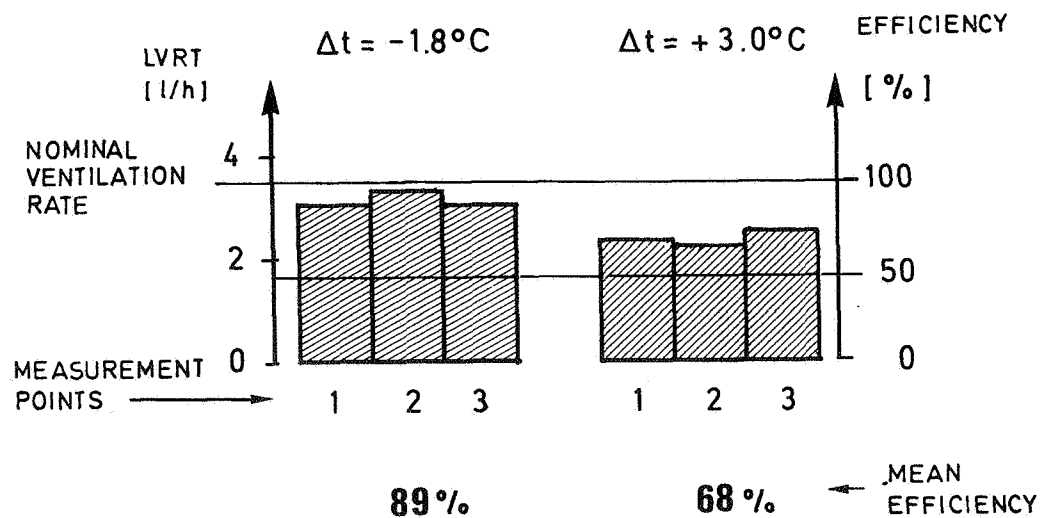


Diagram 2. Ventilation efficiency in two cases

3.3 A case with the supply and extract registers placed on the same wall

The room is shown in figure 5. The room is heated with panel radiators under the windows and the registers are placed on the opposite wall.

The measurements were made at isothermal conditions (no heating on). The results, also presented in figure 5, show that we in this case achieve a very uniform ventilation efficiency.

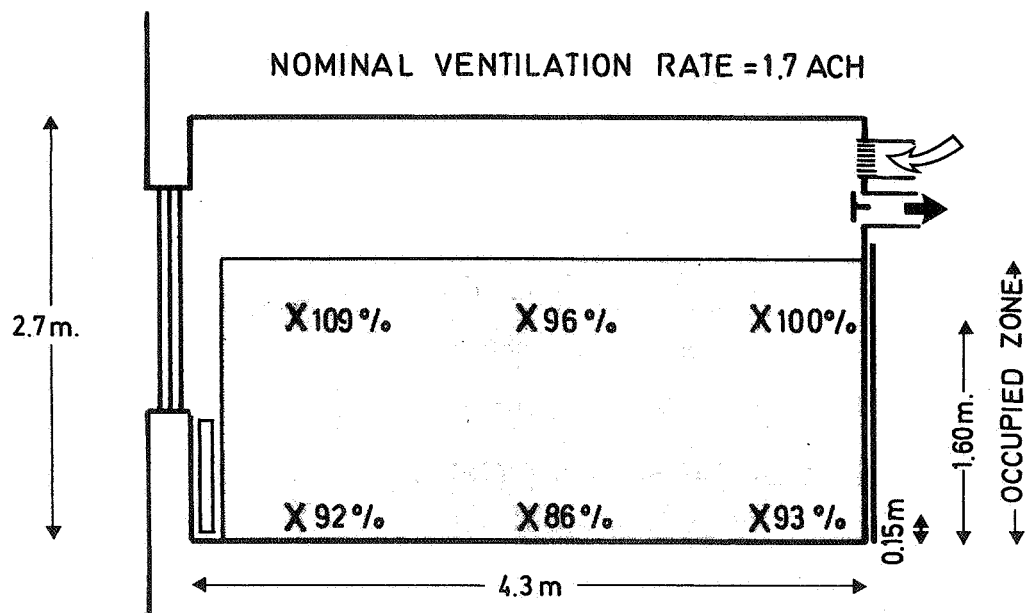


Figure 5. Test room with measurement points and ventilation efficiency results

Conclusions

The findings show that with regard to the ventilation efficiency:

- a) The location of the registers are important even at isothermal conditions.
- b) Air motions caused by radiators do not always prevent short-circuiting effects due to a higher supply air temperature.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 15

THE EFFICIENCY OF VENTILATION IN A DETACHED HOUSE

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THE EFFICIENCY OF VENTILATION IN A DETACHED HOUSE

1. Synopsis

The SEGAS "Autovent" constant concentration apparatus was used to measure the fresh air entering and the local ventilation rate in each cell of a multi-celled dwelling with both natural and mechanical extract ventilation.

Two measurements were made. The fresh air entry into each cell was measured using constant concentration techniques and then the uniform tracer gas concentration was exploited to perform a decay test. The decay tests were conducted without artificial mixing and were interpreted by computing the area under the decay curve to yield a measure of the local ventilation rate. Comparison of the two measurements gives the ventilation efficiency of each cell and an idea of air quality in each room of the house.

Further insight into the pathways of air movement around the house was gained by injecting an amount of tracer gas into one room and following the transfer to all other rooms of the house. Results of these experiments are reported.

2. Introduction

The quality of air inside dwellings is a topic of growing interest. The building industry and the general public are both becoming aware of the need to ensure that excessive ventilation rates (and consequent ventilation heat loss in winter) do not occur. As a result more houses have draught excluding measures applied to them and new houses are built with greater attention to the details of jointing between building components. These changes are leading to lower overall ventilation rates in houses and possible higher exposures to the normal domestic contaminants. The need to understand in detail the pattern of air movement within a house and the consequences of this movement in diluting or removing a contaminant is becoming more important.

It is not sufficient just to know what the overall ventilation rate is, it is now necessary to address the question of local ventilation rates at different points within a house.

This paper concerns itself principally with the methodology for measuring the efficiency of ventilation in a dwelling. In this case a detached, 2 storey, brick built house.

The experiments described here were made possible by the flexibility of the Autovent apparatus, designed initially for continuous monitoring of room ventilation rates, but easily adapted to different operating configurations.

3. Experimental Measurements

3.1. Autovent System and Its Different Configurations

The apparatus^{1,2} consists of twelve sampling and twelve injection solenoid valves. Each valve is operated independently by command from a Zilog MCZ/125 microcomputer which controls the operation of the rig. From each valve a 6mm O.D. nylon gas sampling or injection line runs to each room of the house. The injection valves are two-way solenoid valves connected to a gas manifold in which Nitrous Oxide tracer gas is held at 10 p.s.i.g. pressure. Between each solenoid valve and the injection line to the room is a fine metering valve which is used to adjust the flow in each individual line. Flow rates are measured in-situ with a bubble flow meter.

The sampling valves are 3-way with the normally open port connected to a purge pump. The normally closed ports are connected through a 12-way junction to a Leybold Herraeus Binos 1 analyser. Only one sampling valve is open at any time.

The computer controller allows a very flexible pattern of usage. The basic operating schedule is to produce a constant concentration of gas in all rooms of the house, but, since the apparatus is really only an assembly of controllable solenoid valves it can be used for other experiments. Three modes of operation are described and used in this paper, constant concentration, constant concentration followed by decay, and injection at a point followed by decay. We have also written programmes to perform constant emission experiments and step-wise injection experiments.

In the constant concentration mode the operating sequence is as follows. Each room is sampled in sequence for 8 seconds. At the end of the sampling period the computer reads the analyser and calculates the necessary injection period for that room. The injection valve is opened and the next sampling valve is selected and so on in a continuous loop. at the end of each 30 minute period the computer produces a summary of the fresh air entering every room and adds these to provide a whole house ventilation rate.

Averaged values of wind speed and direction and ambient temperature are also printed.

Constant concentration with decay is identical to constant concentration until, at the pressing of a key, the decay mode is entered. In this mode the computer samples each room ³ in sequence for 8 seconds and records and accumulates the values of gas concentrations in each room. All injection valves are closed. The decay process is allowed to continue until, at the pressing of another key, the computer produces a summary of the accumulated area under the decay curve for each room.

The averaged wind speed and direction and ambient temperature are recorded every 30 minutes.

The facility to switch from constant concentration mode to decay mode in less than 15 seconds is a great advantage and ensures that a uniform concentration of gas is present at the initiation of the decay. Uniform concentration is very hard to achieve by other means.

Injection at a point followed by decay is very similar to the previously described mode of operation. One room is selectively injected with gas until a target is reached. at this point the decay of gas in the source room and the build up and subsequent decay of gas in all other rooms is monitored by successive 8 second samplings from each room. The areas under each gas concentration curve are calculated at the termination of the experiment.

3.2. The Test House

All experiments were undertaken in a 2-storey, brick-built detached test house. The house details are shown in Figure 1 (p 4.). There are 3 rooms and an entrance hall on the ground floor and 4 bedrooms, a bathroom, a toilet and a landing on the upper floor. The total volume of the house is 240m^3 . It is equipped with a mechanical extract ventilation system with registers in the kitchen, the bathroom and the toilet and a wet central heating system with radiators in all rooms.

The house had been extensively modified to reduce its leakage area. The suspended wooden floor had been taped over at ground and first floor levels, cracks around fittings and around skirting boards had been sealed as had those in the ceiling and walls. Service pipes penetrating the outer leaf were sealed around and all external doors and windows were weatherstripped with compressible foam strips. The leakage flow rate of the house was found to be $0.46\text{ m}^3/\text{s}$ at 50 Pa. This corresponds to about 7 air changes per hour.

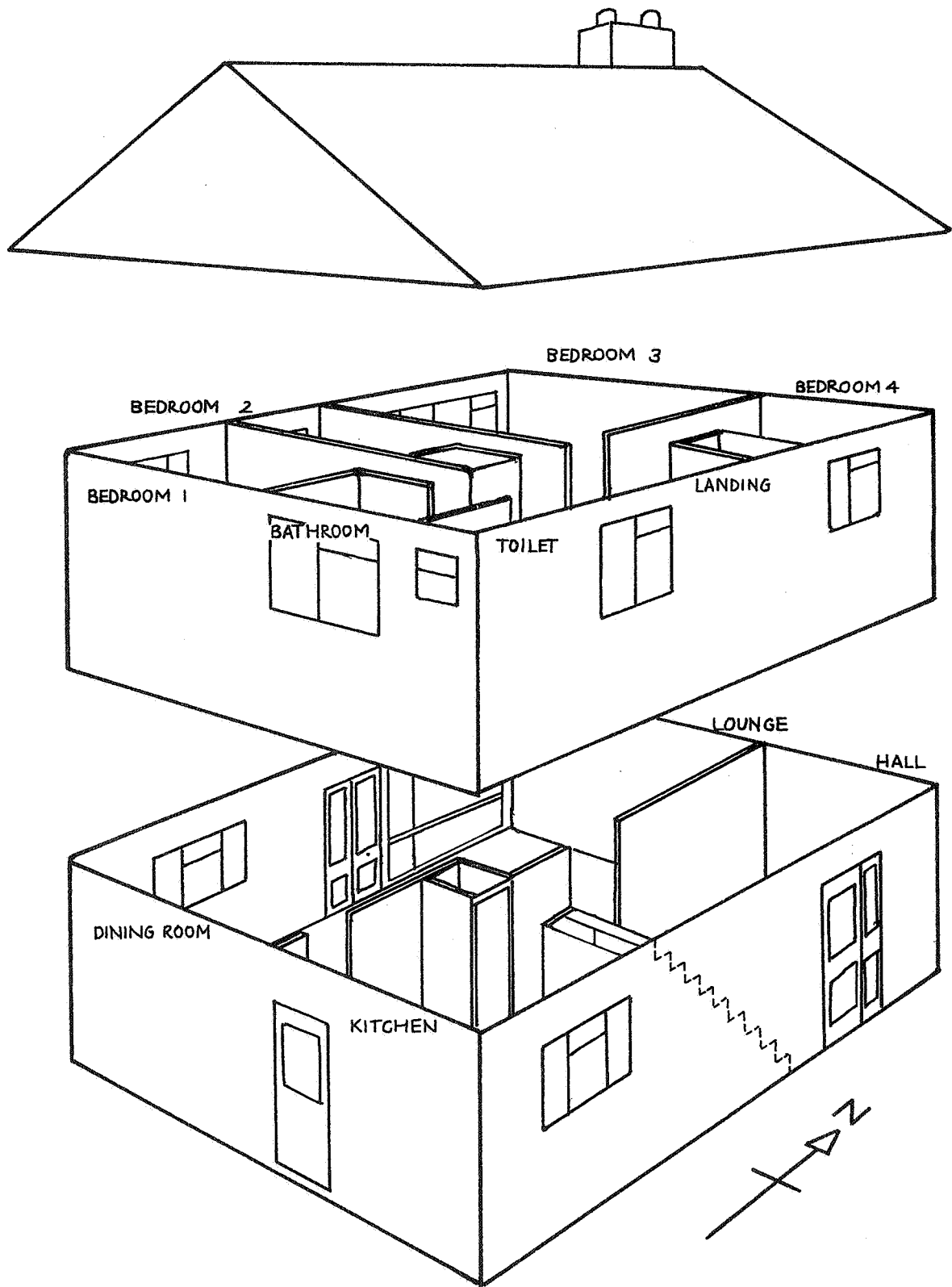


Figure 1.
The Test House Plans and Internal Details

3.3. Experimental Details

Experiments were carried out with and without heating as detailed below. All constant concentration measurements were carried out with mixing fans, all decay experiments were performed without mixing. All samples were drawn from the centres of the rooms.

In all experiments the internal doors were left open to encourage mixing between rooms. The only exceptions to this were the injection at a point followed by decay. For these tests the door of the source room was kept closed during the initial dosing and opened subsequently. All other internal doors were open.

The experiments undertaken are tabulated below:-

<u>Operating Mode</u>	<u>Ventilation System</u>	<u>Heating System</u>
1. Constant Concentration	Natural	Off
2. Constant Concentration + Decay	Natural	Off
3. Constant Concentration + Decay	Natural	On
4. Constant Concentration + Decay	Mechanical extract	Off
5. Injection in Kitchen	Natural	On
6. Injection in Kitchen	Natural	Off
7. Injection in Bedroom 3	Natural	On
8. Injection in Bedroom 3	Natural	Off

4. Theoretical Background

There are several points to note in interpreting these experimental results. Firstly, it is assumed that conditions remain constant throughout a decay run and are equivalent to those occurring during the preceding constant concentration run. This is of course extremely unlikely. Ambient weather conditions can change very rapidly and quite markedly during a slow decay process.

Secondly, it must be stressed that the decay results cannot be, interpreted to yield room ventilation rates. They provide an insight into the relative ventilation efficiency of each room and an indication of the exposure to an airborne contaminant. The constant concentration followed by decay results relate to an homogeneous pollution source and the injection at a point followed by decay relates to a point pollution source and the movement of the pollutant around the house.

Sandberg ³ has given a full account of the theory of ventilation efficiency. He has shown that, at an arbitrary point, there is a direct relationship between the integrated exposure to a short burst of released contaminant and the steady state concentration attained, at the same arbitrary point, by continuous release of contaminant.

This relationship between transient and steady state phenomena is expressed as;

$$\frac{M}{A} = \frac{\dot{M}}{C^S} \quad \text{or} \quad C^S = \frac{\dot{M}A}{M}$$

where, A = Integrated Exposure, the area under the concentration versus time curve following a short contaminant emission.

C^S = Stationary concentration attained by continuous contaminant emission.

$\dot{M}$ = The rate of contaminant emission

M = The amount of contaminant emitted in the short burst.

If we know the integrated exposure we also know the stationary concentration and vice versa.

This relationship offers us a method of obtaining data about steady state conditions from measurements of the response to a short term event. It also gives us the basis of a method of measurement which is described later.

The integrated exposure A obtained when the whole house is filled to a uniform initial concentration, $C(o)$ and then allowed to decay has an important physical interpretation.

A local ventilation rate, r , is determined by

$$r = \frac{C(o)}{A} \quad [1/h]$$

In the special case of instantaneous and complete mixing then the local ventilation rate is equal to the overall ventilation rate. This is defined as the total volume of outdoor air entering the space divided by its total volume. The larger the local ventilation rate the better the ventilation efficiency.

5. Calculation of Results

The experimental measurements were processed in slightly different ways depending on the particular experiment.

Constant concentration results are presented on the computer screen in the form of volumes of air entering each space per hour and so require no further processing.

Constant concentration followed by decay results were calculated in part by the computer which presented values of the area under the decay curve. There remained then an additional truncation error which arose from the fact that the N_2O concentration was never allowed to decay to zero. This extra term was small and was calculated by simple triangulation. The total area under the decay curve, was then divided by the initial room concentration to yield a local ventilation rate. The final slope of the decay curves (the overall ventilation rate) was compared to the whole house ventilation rate calculated from constant concentration techniques with generally very close agreement. Results appear in table I (p 8.).

The analysis of the experiments involving injection at a point again involved the computation of the area under the decay curve for each room and the estimation of an additional truncation error. These two terms gave the integrated exposure to the pollutant in each room. The crude figures were then normalised by dividing by the product of the initial gas concentration and the volume of the source room. In other words the integrated exposure was divided by the amount of pollutant released, so that the results of each experimental run could be compared directly.

In the subsequent sections we discuss each of the results, extend the interpretation and draw conclusions about the movement of air around this 2-storey house. An earlier paper in this section presented results from laboratory experiments on a 2-roomed test structure and many of the conclusions apply to this real house.

6. Discussion of Results

6.1. Constant Concentration Technique - Natural Ventilation

The results for an overnight run are depicted in Figure 2 (p 10). The constant concentration technique tells us where fresh air enters a house as a result of the wind and stack effects. Figure 2 (p 10) shows a low overall fresh air entry and quite a strong dependence on ambient conditions. Roughly speaking, the individual room ventilation rates follow the same pattern as the whole house rates, although their precise dependence on the ambient conditions is not immediately explicable.

6.2. Constant Concentration Technique - Mechanical Extract Ventilation

Figure 3 (p 11) shows an overnight run with the mechanical extract operating at approximately 0.5 ach. The first thing to notice is that the overall ventilation rate is reasonably constant but there are some wide variations in individual rooms. The extraction points are the kitchen, the bathroom and the toilet. The bathroom and kitchen seem to have widely varying fresh air entries whereas the other rooms remain constant. This may have been due to the relatively large leakage areas associated with these two rooms and their positions in the house.

6.3. Constant Concentration Followed by Undisturbed Decay

This technique was used to examine the pattern of air movement around the house under natural ventilation with and without heating and with mechanical extract ventilation.

The results are presented in Table 1 (p 8.) and discussed below. In these circumstances the final slope of the decay curve is equivalent to the overall ventilation rate.

TABLE 1. CONSTANT CONCENTRATION FOLLOWED BY DECAY
LOCAL VENTILATION RATES (h^{-1})

VENTILATION SYSTEM	NATURAL	NATURAL	MECHANICAL EXTRACT
HEATING	OFF	ON	ON
Kitchen	0.052	0.22	0.45 *
Dining Room	0.051	0.21	0.39
Lounge	0.052	0.21	0.39
Hall	0.053	0.21	0.43
Landing	0.048	0.20	0.45
Bedroom 1	0.047	0.20	0.49
Bedroom 2	0.047	0.20	0.44
Bedroom 3	0.047	0.19	0.42
Bedroom 4	0.048	0.18	0.41
Toilet	0.048	0.17	0.46 *
Bath	0.048	0.14	0.46 *
Final slope, h^{-1}	0.043	0.2	0.43
Constant concentration whole house rate, h^{-1}	0.041	0.3	0.41

* Extraction points

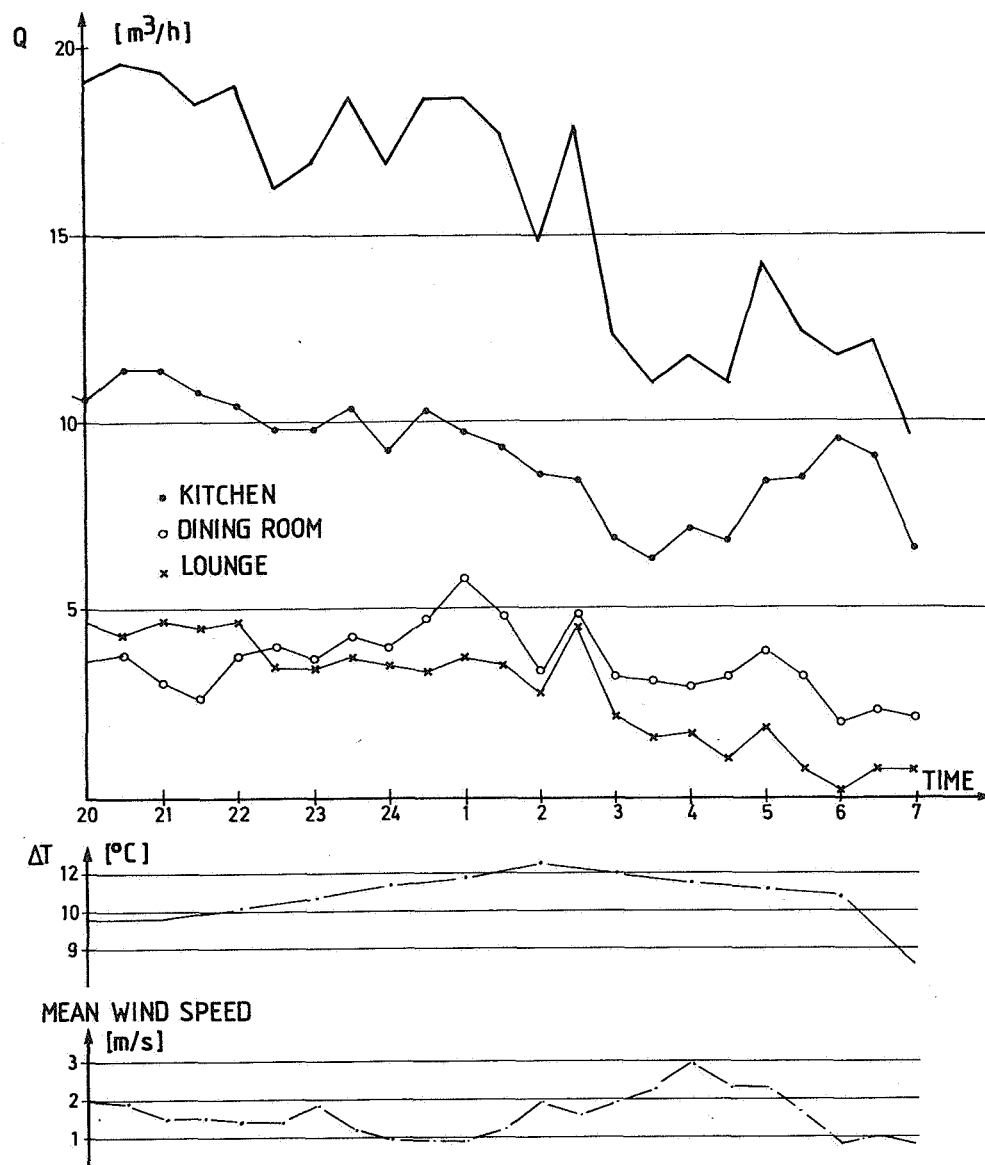


Figure 2.
Natural Ventilation Results for an overnight run showing the total air flow into the house (solid line) and the flows to individual rooms.

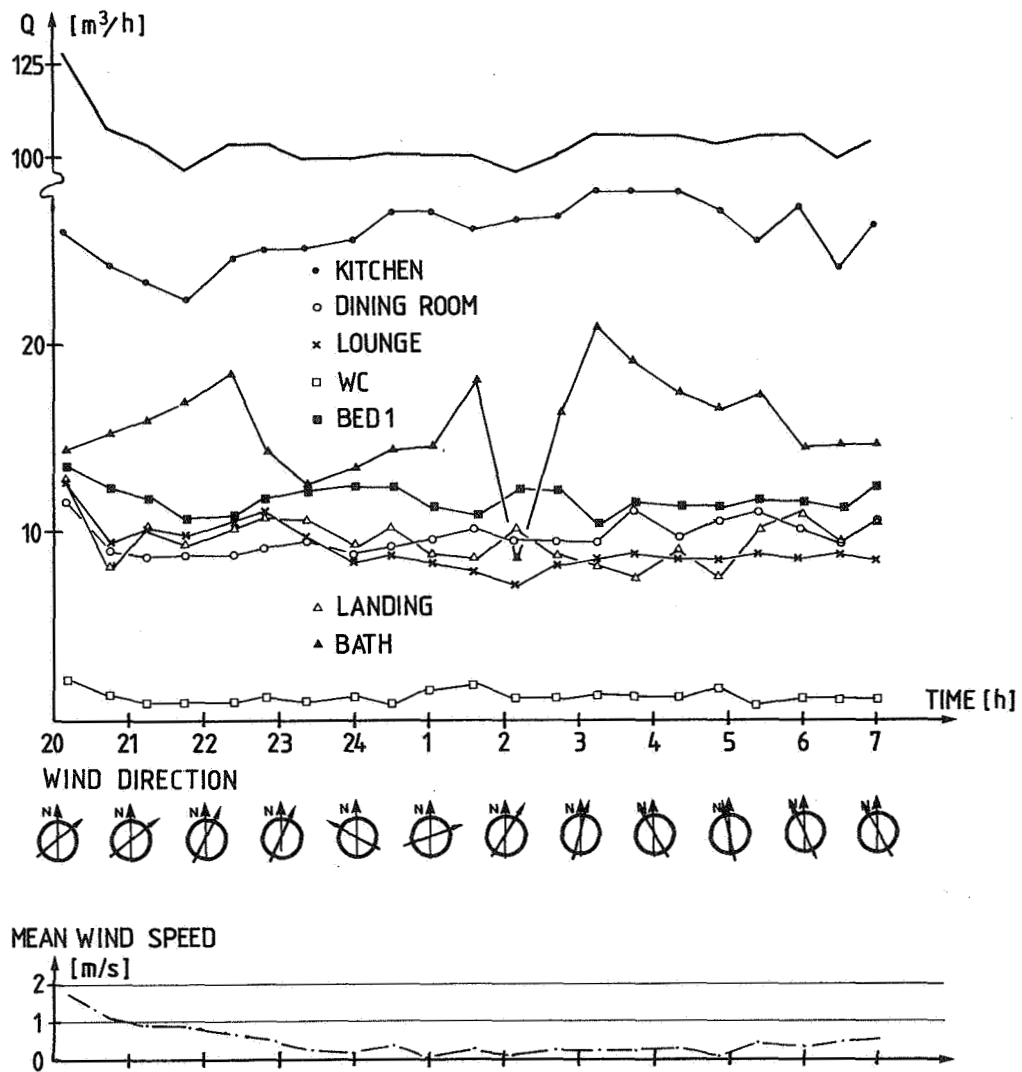


Figure 3.

Mechanical extract ventilation results for an overnight run showing the total air flow into the house (solid line) and the flows to individual rooms.

6.3.1 Natural Ventilation - No Heating

The measured ventilation efficiency is typical of that found for a low fresh air flow rate, that is, a very even ventilation efficiency implying even-mixing³.

In these circumstance the final slope of the decay curve is equivalent to the ventilation rate.

The observed value of the slope of the decay curve was -0.048. This implied a flow rate of 11.5 m³/hr into the house (volume 240m³). This result agrees well with the constant concentration results, Figure 2 (p 10).

6.3.2 Natural Ventilation - With Heating

The ventilation efficiency for this condition is higher than the previous case, Table 1 (p 8). The efficiency is almost constant again implying an even mixing through the house. The low efficiencies for the toilet and bathroom may be because these rooms were on the leeward side of the house and on the upper floor and so were subjected to a dominant flow direction from the house. The average wind speed during the decay was about 1.5 m/s and the final slope of the decay curve was -0.2. The constant concentration result immediately prior to the decay was 0.3 ach with a 2.5 m/s wind. Again, the agreement is reasonable and encouraging.

6.3.3 Mechanical Extract Ventilation - No Heating

A higher ventilation efficiency than the previous 2 results, an even distribution typical of a mechanical extract system, and the final slope, -0.433, agreed well with a constant concentration determination, Table 1 (p 8).

The rooms extracted from had the highest ventilation efficiencies. This is as expected and as found for the 2-roomed test structure.

6.4. Discussion

The two techniques so far discussed allow us to know where the fresh air gets into a house and how well that fresh air is subsequently distributed. The fresh air entry points are widely distributed since fresh air enters all rooms of the house, although not all results are depicted in figures 2 & 3 and in general, the mixing within the house is complete, although slowly achieved in some instances. An alternative interpretation of the evenness of the ventilation efficiency throughout the house is that exposure to an homogeneous pollutant would be even. The next section discusses the effect of introducing a pollutant into one room of the house.

6.5. Injection at a Point Followed by Decay

Results are presented in Table 2 (p 14) for 4 house conditions, injection into the Kitchen with and without heating and injection into Bedroom 3 with and without heating.

There are several things noticeable. Firstly the integrated exposures without heating are greater than those with, reflecting the higher ventilation rates in the latter case. Secondly, the source room always has the highest integrated exposure, not surprisingly. Thirdly, the results without heating show a greater differential exposure between floors than those with heating. The results with heating have almost equal integrated exposures outside the source room. This implies an efficient mixing of the air within the house. Fourthly, results without heating show that within a particular floor the mixing is more efficient than between floors. The heating system is designed to provide differential temperatures between living and sleeping rooms and this mechanism aids the mixing between floors.

6.5.1 Interpretation of Results

These integrated exposures can be used to predict the stationery concentration of any point pollutant source, in any room and can be used to highlight differences in exposure in different parts of the house, in a given set of weather conditions.

TABLE 2 INJECTION AT A POINT FOLLOWED BY DECAY

SOURCE ROOM HEATING	KITCHEN OFF	KITCHEN ON	BEDROOM 3 OFF	BEDROOM 3 ON
Kitchen	47.8	33.6	23.1	32
Dining Room	38.1	24.7	20.9	31.8
Lounge	38.9	24.3	19.6	36.8
Hall	44.2	25	20.4	35.8
Landing	24	23.9	35.1	40.4
Bedroom 1	23.4	23.4	40.5	35.6
Bedroom 2	23.2	23.6	40.2	35.8
Bedroom 3	23	22.6	48.1	40.8
Bedroom 4	23.8	24.1	40.9	40.4
Toilet	23.7	23.8	41.4	38.5
Bathroom	23.7	22.7	37.9	36.1
Wind speed ms^{-1}	2.0	3	1.0	2.5-3.6
Wind direction	SW-WSW	NNE	E-SE	NNW

All entries are in units of $\text{m}^3\text{h}/10^3$ at 20°C unless otherwise indicated.

Figures 4 - 9 (pp 16 - 21) show the actual tracer gas concentration in various rooms plotted against time. Figures 4,5,6 (pp 16 - 18) present the results for injection in the kitchen without and with heating. There are interesting differences between the two cases. Without heating, it takes 5 hours for the bedroom concentrations to reach their peaks, about 10 ppm. With heating, the bedroom peaks, of 18 ppm, are reached in 1.5 hours. This illustrates the movement of air between floors promoted by the differential heating of the ground and first floors and by the stack effect.

Figures 7,8,9 (pp 19 - 21) show the effect of injection into Bedroom 3 without and with heating. Without heating, it takes nearly 4 hours for the ground floor rooms to reach their peak concentrations. With heating, the peaks are reached after about 1.5 hours.

The complexity of the graphs illustrates the importance of this method of interpretation in unravelling the complex phenomena interacting.

7. Conclusions

By using the area beneath the plot of concentration against time it has been possible to draw several conclusions regarding behaviour of hypothetical homogeneous and point pollution sources within the house. We have shown that, with internal doors open, the mixing within a floor is generally very good. The mixing between floors is better when the house is heated than when not heated. These conclusions are only strictly valid under the weather conditions encountered.

The Autovent system has been shown to be very versatile and adaptable to several different experimental configurations. The experimental approaches towards the problem of measuring the quality of internal house air have proved successful and we hope that others will be encouraged to develop them further.

8. Acknowledgements

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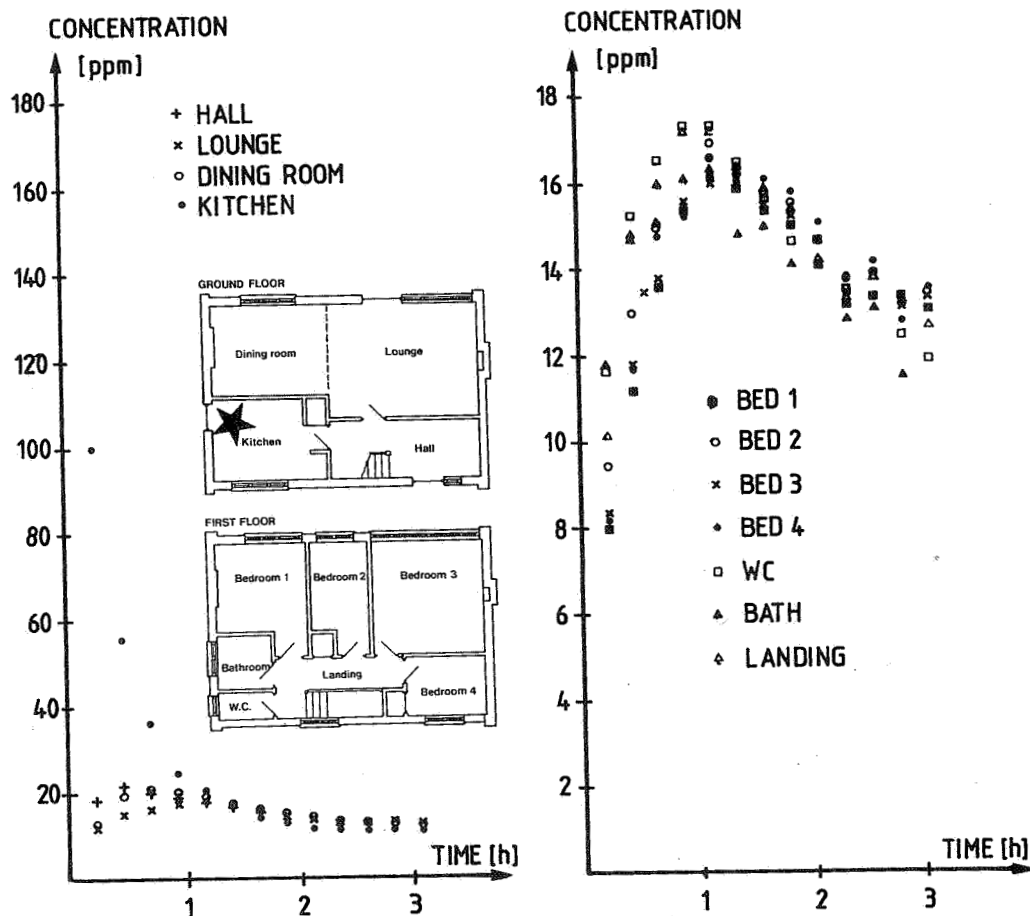


Figure 6.
Injection in the Kitchen, House Heated.
Results for Ground Floor Rooms

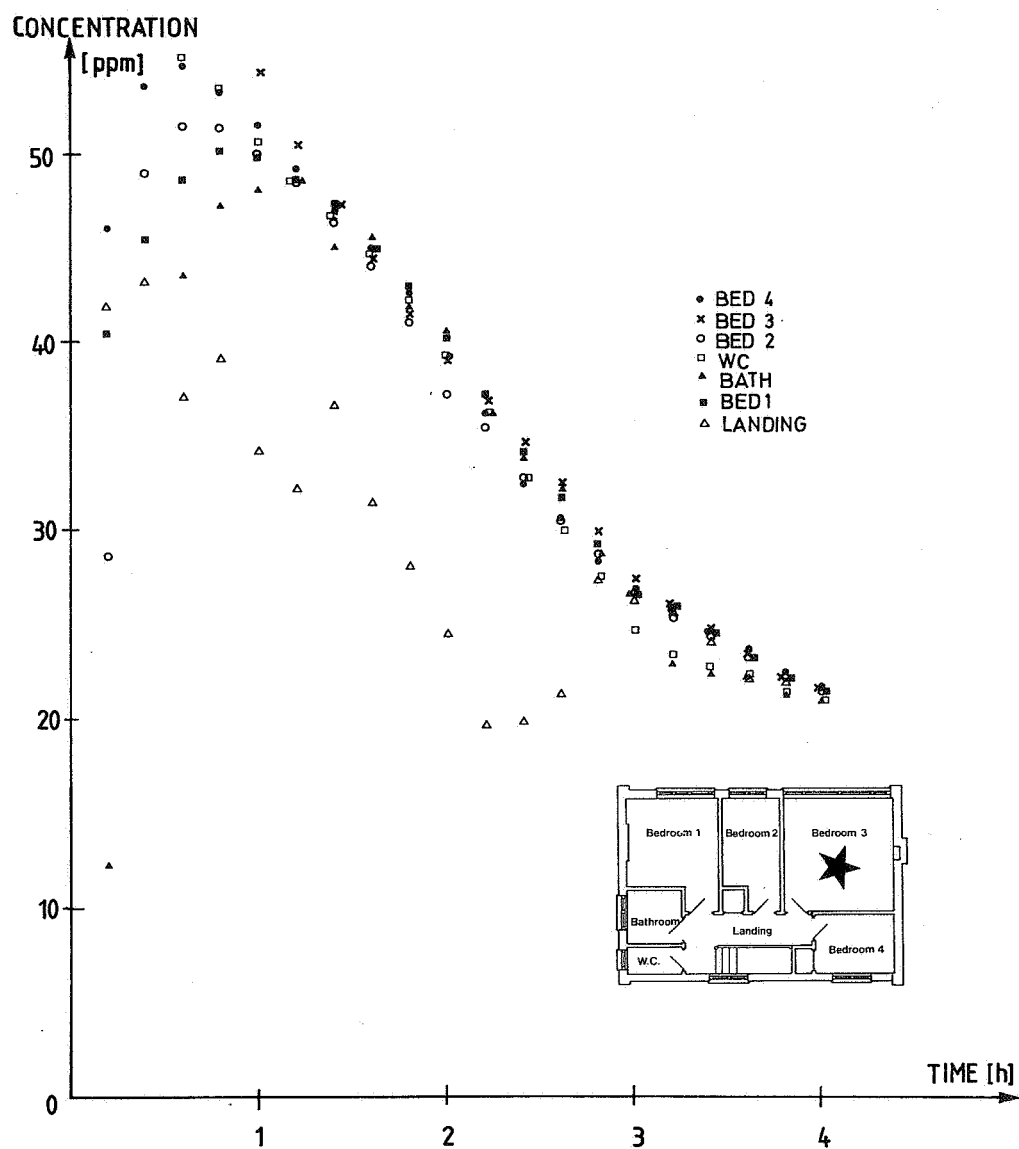


Figure 7.
Injection in Bedroom 3, House Heated
Results for First Floor Rooms

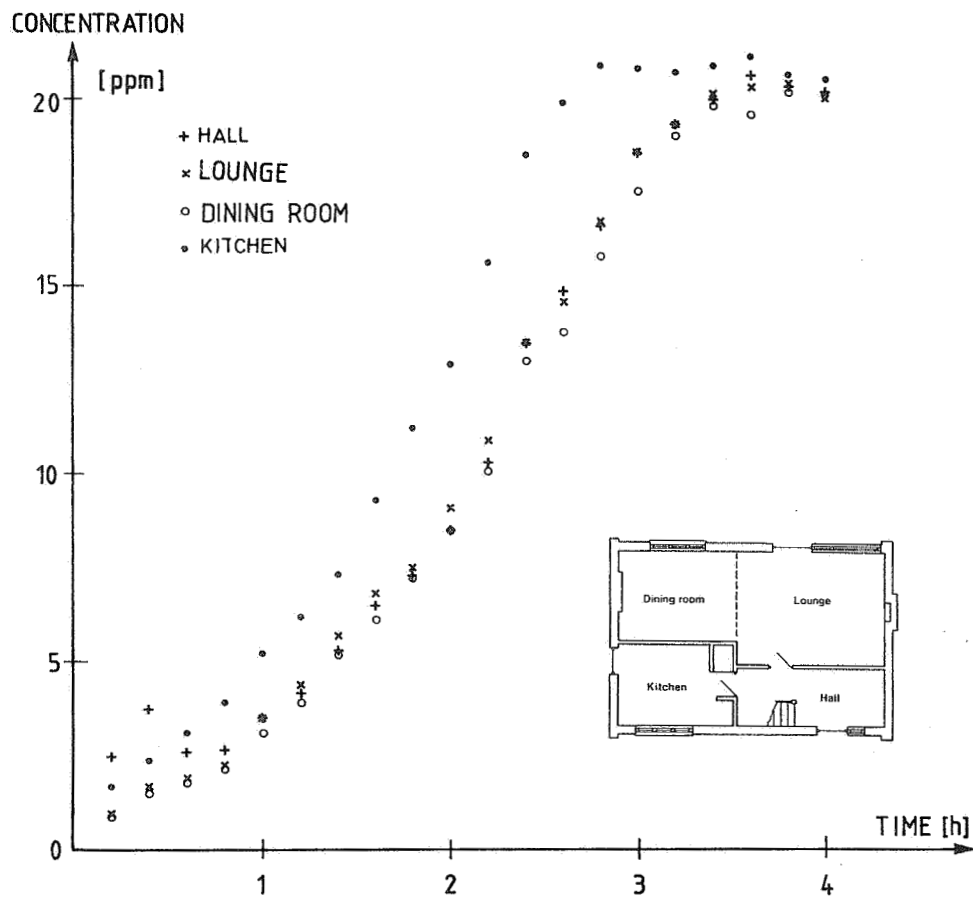


Figure 8.

Injection in Bedroom 3, House Unheated
Plot of concentration against time for Ground Floor Rooms

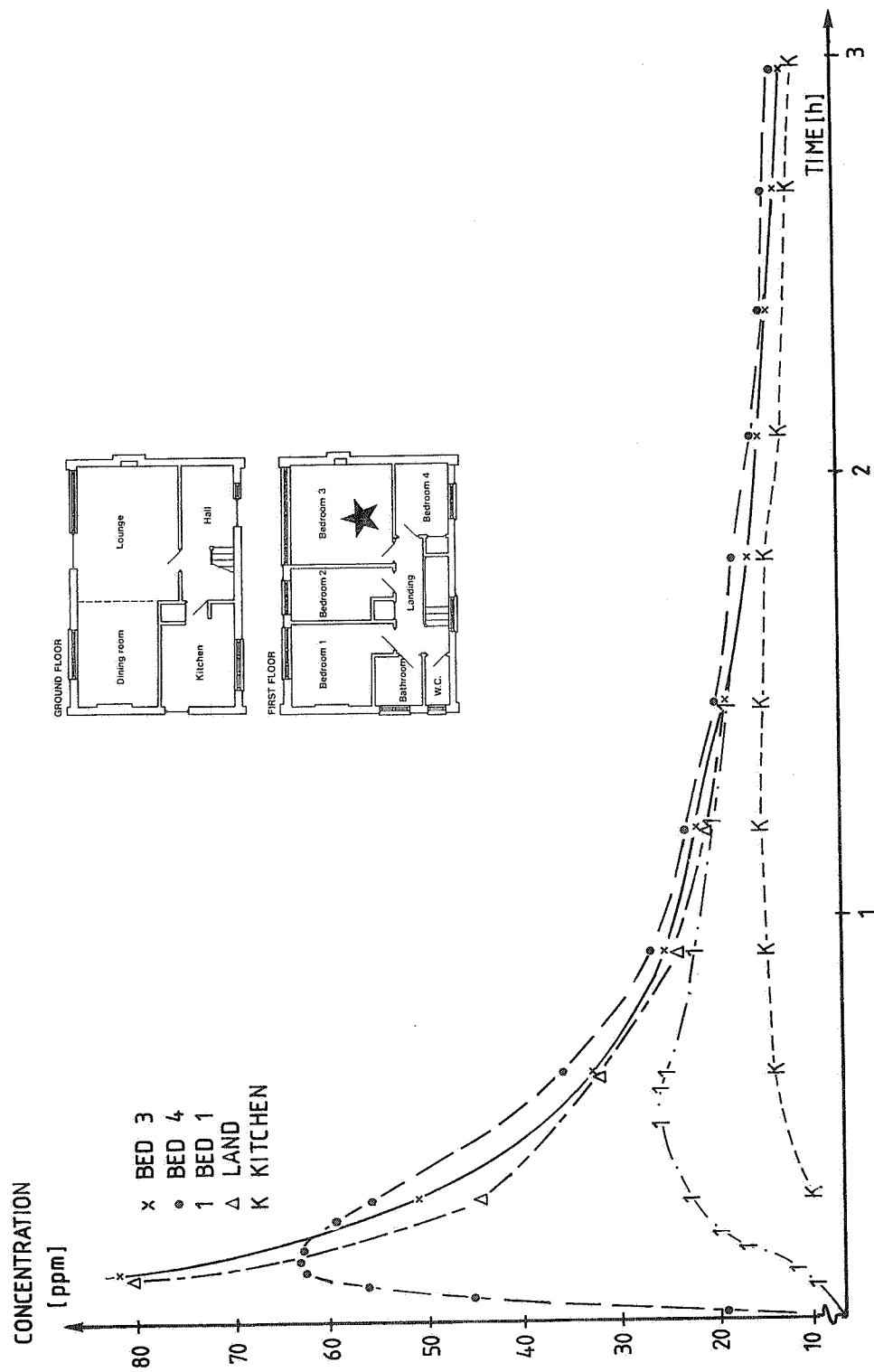


Figure 9. Injection in Bedroom 3, House Heated.
Plots of concentration against time for selected ground and First Floor Rooms

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 16

REDUCTION OF AIR INFILTRATION AND THE DEVELOPMENT OF
CONTROLLED MECHANICAL VENTILATION IN FRANCE

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SYNOPSIS

Since 1974 the french Authorities have insisted on energy being saved in all buildings. There was very strong pressure on manufacturers to obtain better sealed window frames.

In practise less than two or three meters cubed per hour at a pressure of ten pascals penetrates.

Also television campaigns have insisted on weather stripping all windows and window frames in all old buildings.

The result of these campaigns is that all buildings with no or natural ventilation systems actually have indoor condensation problems.

The use of controlled mechanical ventilation systems is well established in France and is present in 80 per cent of multi storey flats and 40 per cent of single family housing.

A new regulation is in preparation and will probably require the fitting of mechanical ventilation in all new buildings.

In the meantime we are to have a new regulation concerning the heat losses in house and flats. This means greater interest in planning of buildings and houses with centralised controlled mechanical ventilation systems with heat exchangers.

Because of this above new regulation we are sure of great developments of our systems in the coming years.

1. VENTILATION OF DWELLINGS :
CURRENT STATE-OF-THE-ART AND FUTURE PROSPECTS

Bioclimatic architecture, intensive insulation, insulation, improvement of the existing dwelling facilities, ventilation, new heating methods and equipment, regulation and programming are all part of the energy saving inventory. Among these approaches to the problem, ventilation must be highlighted as it is one of the methods most difficult to control yet is of the utmost importance as regards the health and comfort of the occupants, the preservation of dwelling and energy saving programs.

Ventilation has been subjected to numerous controversies which, should be stated, are the outcome of many conferences, but should no longer be satisfied with the approximate approach.

Considerable efforts have been devoted to the reduction of heat losses in new dwellings since 1974 but these efforts have been mainly concentrated on improving the thermal insulation characteristics of the external walls.

The new constraints of future regulations, intended to reduce energy consumption will call for higher and higher investments per equivalent tonne of petroleum saved if the solutions adopted are not to jeopardize "comfort". Until now, the improvement of thermal insulation has not been obtained at the expense of the living conditions in dwellings. On the contrary, these conditions have been improved at the same time (less cold radiation transmitted from the external walls, improved heating regulation, controlled air change with mechanical ventilation...), but do we have the resources to continue in this direction ?

The reduction of provisions contributing to comfort is always a difficult decision to take for, if excessive, the improvised means used by the occupants for maintaining the standard of comfort that they require could be contrary to the economic objectives sought after. Thus ventilation, which has become one of the priorities towards which energy saving in dwellings should be orientated, poses such a dilemma.

Is it necessary to maintain the air change rates at their present values and reduce the corresponding heat losses by adopting measures of incitement to make heat recovery systems more widespread with reasonable investment figures ?

Or should it be now recognized that the statutory requirements pertaining to ventilation are too "comfortable", with the resulting temptation of reducing the air change rate which, in theory, would enable both the investment and energy consumption levels

to be reduced ?

The choice is not so easy, it is essential that an analysis of the requirements concerning the new generation of dwellings be undertaken beforehand, for some criteria taken into consideration to justify the requirements of the Decrees dated the 22nd October 69 relative to ventilation no longer have a priority status.

2. THE NEW PRIORITY VENTILATION OBJECTIVE :
"PREVENTION OF CONDENSATION"

Probably the most sensible principle for economically ensuring the ventilation of dwellings consists in allowing fresh air to flow into the most important enclosed areas of a building (bedrooms, living rooms, etc...) and then to use this volume of air to replace that in the rooms and spaces allocated to service installations prior to being exhausted outside the building. This is one of the solutions put forward in the Decrees dated the 22nd October 1969 and it is rather a pity that it is not imposed for all dwellings for, besides the resulting energy saving features of this solution, it also provides for integral ventilation of the dwellings and thus prevents the diffusion of polluting emanations and water vapour from the spaces allocated to service installations to the main living areas. Of course, this principle results in a reciprocal state of dependence between the main rooms and those spaces allocated to services, but technological solutions can be found to meet their respective requirements.

Many studies relative to ventilation requirements in dwellings have already been undertaken, especially by the C.S.T.B. to justify the "Exemples de solutions pour faciliter l'application du règlement de construction" (Examples of solutions facilitating the application of building regulations) published in November 1971. However, the sharp rise in the cost of energy has resulted in intensive insulation of buildings and lower interior temperatures and their effect on ventilation objectives must be thoroughly examined.

The rooms and spaces set aside for service facilities are generally the starting point for very strong yet short lasting polluting emanations. The rapid increase in the pollution level in these areas is noticed by the occupants who feel that a higher air change rate is required. These service rooms and spaces temporarily justify an increased ventilation rate which must be noticeable by the occupants of the building (noise, luminous indicating devices...) or provided with a timing system so as to avoid omissions and thus restrict its use to only the most effective periods.

Besides the greater temperature rises in the well insulated kitchens with a low thermal inertia, the development in construction techniques has not changed the ambient conditions in the rooms and spaces set aside for service installations. These areas are designed to temporarily support high pollution rates without suffering from any notable damage, and a reduction in their ventilation facilities could be considered. However, any reduction in ventilation high operating rate would have a negligible influence as regards the energy consumption as its normally limited operating time (less than 10 %) would have to be increased to compensate for a loss of instantaneous efficiency.

In the main rooms of a building, the quality of the air degrades insiduously and is polluted by the direct emanations of the occupants at a slow but constant rate. The increase in the pollution level is hardly perceived for, being progressive, it leads to the human organism becoming accustomed to the environment and does not call for any change in the ventilation of the rooms. The ventilation installation must therefore provide for an idling air change rate with allowance made for the potential requirements of the main rooms, even if they are unoccupied, or be capable of automatically adapting its operating rate to the changes in the ambient conditions.

In the main rooms, the emission from the occupants mainly consists of carbon dioxide, water vapour and organic substances which encourage microbial activity, and the concentration of carbon dioxide is often selected as an indication of the coefficient of pollution to justify the ventilation objectives. The selection of such a coefficient is certainly sufficient for well heated and well ventilated rooms but the concentration of CO₂ which is hardly affected by the structural characteristics of the external walls of buildings and by the interior temperature cannot give an indication of the moisture content in the ambient air and therefore, does not permit evaluation of the risks of condensation resulting from a reduction in the number of air changes and the temperature in the new generation of dwellings.

It is an accepted fact that the moisture content in the main rooms of dwellings depends on the occupation and the air change rate, but, it also depends on the interior temperatures. In older types of dwellings, when the moisture content reaches a high level, it also depends on the condensation formed on single-glazed windows and the moisture diffusion through the walls, although quite slow often results in other more vulnerable sections of the building not being exposed to condensation. In the new buildings, these protective features are not present, owing to the use of vapour barriers to protect the insulating materials, together with the inclusion of double-glazed window units and efficient shutters and, therefore, it is the responsibility of the ventilation installation alone to oppose the damage caused by humidity.

Prior to recommending that the idling rate of air changes be reduced, its effects on the conditions of hygiene and the risk of local condensation should be established. This calls for preliminary evaluations as regards the concentration of CO₂ and the relative humidity in the main rooms normally occupied. To clearly show the influence of the different factors which determine the quality of the air, the following text is devoted to bedrooms, which are the areas of a building in which we have encountered the most damage caused by faulty ventilation during our surveys.

2.1 Development of the CO₂ concentration and relative humidity in a bedroom according to the fresh air flow rate.

- Emission of 15 liters/hour of CO₂ and 40 g/h of water vapour per person allowing for the reduced metabolism when sleeping ;
- 90 % exterior relative humidity during the heating season and the night ;
- Bedroom with a volume V (m³) occupied by two persons, doors and windows closed and provided with an air change flow rate of Q (m³/h).

The curves in Figure 1 show the development in the concentration of carbon dioxide (CO₂) according to the occupation time. These curves are established by means of the calculations given in Appendix 1. They show that, for the same ventilation flow rate, the concentration of carbon dioxide is practically independent of the volume of the bedroom when the occupation period is more than several hours and that the increase in concentration is slower as the volume of the room is increased. Thus, nothing justifies the requirements that air changes should be proportional to the volume of the bedrooms. On the contrary, the slight inertia of small bedrooms could justify higher flow rates.

Curves 1 and 2 in Figure 2 represent the development of the relative humidity in a 25 m³ bedroom for an air change flow rate of 15 m³/h, and for very different interior temperatures T_i and exterior temperatures T_e . Under these same conditions, curve 3 represents the carbon dioxide content independent of the temperatures. The curves of this figure show the relative humidity is very dependent on the exterior temperature and the heating and that its development cannot be estimated from the development of the carbon dioxide concentration values. The calculations pertaining to these curves are given in Appendix 2.

The curves in Figure 3 show the percentage of days with heating, and during which a specified moisture content is reached or exceeded in a bedroom occupied by two persons. The heating season considered is limited to all days when the exterior temperature is no higher than 13° C so as to allow for free heat inputs in well insulated dwellings. The different curves correspond to well defined

heating (T_i) and ventilation (Q) conditions and to the frequencies of the exterior temperature T_e in area B.

Thus, in the case of a bedroom occupied by two persons and heated to a temperature of 16°C during the night, if the ventilation flow rate is $15 \text{ m}^3/\text{h}$, the number of days when the relative humidity can exceed 85 % represents 60 % of the days included in the heating period. If these high moisture contents are attained too often, they would sustain conditions of hygiene having harmful consequences on the occupants. If the ventilation flow rate is $30 \text{ m}^3/\text{h}$ (value corresponding approximately to the current regulations in force), it can be seen that the days when the moisture content can exceed 85 % represent only 18 % of the total when the interior temperature is 16°C and, if this temperature is 19°C , this moisture content is never reached.

Besides the unhealthy living conditions for the occupants, an excessive relative humidity gives rise to the formation of condensation which has untold damaging effects as regards the construction.

2.2 Risks of condensation in main rooms

To thoroughly understand the conditions in which condensation is formed, it must be remembered that :

- the weight of water vapour that can be contained in the ambient air is limited. When this limit is reached, saturation occurs : the relative humidity is at 100 %. This limit depends on the air temperature ; the higher the temperature, the higher the limit. Thus, for a given water vapour density, the air cannot be cooled below the dew point temperature without the resulting formation of condensation ;
- if the distribution of the water vapour in the room is homogeneous, the same principle does not apply for the temperature and, if it is locally lower than the dew point temperature, condensation starts to occur.

The ambient temperature T_i in a room is a fictitious temperature which, when evenly distributed, causes the same heat exchanges with the occupants as the actual heterogeneous temperatures in the effective occupation area. The surface temperature θ_i of an external wall is lower than T_i ; it depends on the degree of thermal insulation that it provides and the heat exchanges of its interior lining with the other walls by radiation and with the air through the convection process. The calculation of θ_i is of the utmost importance as it enables precautions to be taken against the formation of condensation.

The surface temperature (Fig. 4a) θ_i on the typical section of a flat external wall can be calculated by the formula (1) :

$$\theta_i = T_i - \frac{K}{h_i} (T_i - T_e) \quad (1)$$

where

- T_e exterior temperature
- K coefficient of thermal transmittance in $W/m^2^\circ C$
- h_i coefficient of interior surface heat exchange in $W/m^2^\circ C$ of which the conventional values adopted in the D.T.U. (Rules Th-K77) are :
- $h_i = 9 W/m^2^\circ C$ for vertical walls
- $h_i = 11 W/m^2^\circ C$ for ceilings or roofs
- $h_i = 6 W/m^2^\circ C$ for floors.

For typical sections of double-glazed windows ($K = 3 \text{ watts}/m^2^\circ C$) :

$$\theta_i = T_i - \frac{T_i - T_e}{3}$$

condensation will be formed if the relative humidity is high ; but these surfaces can tolerate condensation without being damaged.

For the typical sections of a well insulated wall ($K = 0.55 W/m^2^\circ C$) $\theta_i = T_i - 0.06 (T_i - T_e)$, the surface temperature is very close to that of the ambient temperature and the risks of condensation are excluded. The same does not apply for certain special points where the formula (1) with conventional h_i values cannot be applied and the most vulnerable are formed by :

a) The junctions of several load-bearing walls with a floor (Fig.4b)

- The thermal insulation in this area is reduced and the transmittance of heat outside the building is increased.
- The convection air flows avoid these corners which offer to much resistance to the air, and the radiation heat exchanges of these low points are often hindered by the furniture. The local h_i coefficient can often attain a value of $4.5 \text{ watts}/m^2^\circ C$.
- The convection air flows which hug the external wall are responsible for maintaining a lower temperature at the floor level than that at the medium height of a room.

On the ground floor of a well insulated detached house, the difference can be as much as $1^\circ C$.

- b) The junctions of the window breast walls or the thresholds of French windows with the floors when there are curtains or draw curtains (Fig. 5).

The space behind the curtains forms a duct filled with a down flow of convected air which is considerably cooled upon contact with the glazed surfaces. This flow of air maintains a very low surface temperature at the junction. To this factor can be added the masking effect of the curtains which reduces the heat exchanges with the interior walls by the effect of radiation.

Condensation can then start to form on the glazing and also at the junction.

- c) The junctions of the floors with the gable walls against which are arranged ward-robes or cupboards (Fig. 6).

In this case, the volume filled with clothes forms a barrier, the thermal resistance of which can have the same magnitude as that of the wall and the air flow which circulates very slowly between the wall and the cupboard cools down and reaches a temperature low enough at the junction for the formation of condensation when the ambient moisture content is high. From these few examples detailed above, it can be shown that in recently constructed dwellings, and because of the building techniques or the normal interior appointments installed by the occupants, the surface temperature of the walls θ_i can punctually often attain :

$$\theta_i = T_i - \frac{T_i - T_e}{3} \quad (2)$$

and that the ventilation has to maintain a moisture content such that θ_i is greater than the dew point temperature so as to avoid the formation of condensation.

Thus, in case (a), if the thermal resistance at the junction corresponds to a coefficient of $K = 1.35 \text{ W/m}^2\text{°C}$ and if $h_i = 4.5 \text{ W/m}^2\text{°C}$, we obtain $\theta_i = T_i - 1.35/4.5(T_i - T_e)$

$$= T_i - \frac{T_i - T_e}{3}$$

It should be noted that during the night, the temperature (2) can often be lower than that of a double-glazed window unit provided with good quality solid shutters corresponding to $K = 2$ and $h_i = 9$.

Therefore at $\theta_i = T_i - 0.22 (T_i - T_e)$ and that the condensation can appear locally on the walls before being deposited on the

glazed surfaces.

The curves given in Figure 7 correspond to a bedroom occupied by two persons during the night. This curve shows the minimum air change flow rates needed to prevent the formation of condensation and mildew at the vulnerable points characterized by a surface temperature of :

$$\theta_i = \frac{T_i - T_e}{3}$$

(the determination method is given in Appendix 3). These flow rates are given according to the interior and exterior temperatures (T_i and T_e , respectively). Figure 7 also shows the percentage of hours in the heating season ($T_e < 13^\circ \text{C}$) during which the exterior temperature is higher than the value T_e .

Examples : If $Q = 15 \text{ m}^3/\text{h}$ and $T_i = 16^\circ \text{C}$; as soon as the average exterior temperature is higher than 2°C , the interior moisture content is excessive and local condensation can be formed. 75 % of the days during which the heating is operating are thus concerned by these conditions which are favourable to the formation of condensation :

- If $Q = 15 \text{ m}^3/\text{h}$ and $T_i = 19^\circ \text{C}$, 32 % of the days are concerned,
- If $Q = 20 \text{ m}^3/\text{h}$ and $T_i = 16^\circ \text{C}$, 34 % of the days are concerned, but if $T_i = 19^\circ \text{C}$, the risks are averted.

The curves in Figure 7 show that, contrary to certain currently accepted idea, the most favourable conditions for the formation of condensation and mildew correspond to the hottest periods of the heating season which are situated at the half-way point. They also show that the lowering of the ambient temperatures to reduce the energy consumption due to heating installations considerably increases the risks when the level of ventilation is low.

Public health specialists readily acknowledge the fact that carbon dioxide concentrations of less than 3 liters/ m^3 cause no health hazards and that the ventilation of a bedroom could be permanently reduced to a rate of a $10 \text{ m}^3/\text{h}$ without causing harmful effects, but the previous analyses indicate that such a reduction is impossible without the humidity having damaging effects on the occupants and the construction. The ventilation flow rates in the main rooms must be determined so as to prevent any condensation.

The factors contributing to the development of condensation in the main rooms are multifold (occupation, interior and exterior temperatures...) and extremely variable. Furthermore, the conditions favourable to the formation of condensation are not perceptible by

the occupants and consequently, the adaptation of the air changes to the requirements of the main rooms cannot be left to the initiative of the occupants.

Ventilation installations must therefore be designed to provide for the following requirements in the main rooms :

- either a permanent minimum air change which prevents the risk of condensation under normal occupation conditions (two persons per bedroom, for instance), for lower interior temperatures selected by the occupants so as to reduce the heating costs (16° C in the bedrooms, for instance) regardless of the exterior temperature, or
- an air change automatically adapting to the requirements. Such solutions to the problem have not yet been incorporated in dwellings but the technological progress in the fields of detection servocontrol systems should allow for their future development.

These installations must also conciliate these objectives with the current priority "energy saving".

APPENDIX 1

Development of the carbon dioxide concentration in a bedroom occupied by two persons :

$$C = C_e + \frac{q}{Q} (1 - e^{-\frac{Q}{V} t})$$

C (1/m3) : concentration in the bedroom
C_e(1/m3) : concentration of CO₂ in the inlet air 0.3 1/m3
q(1/h) : CO₂ emission by the occupants
Q(m3/h) : ventilation flow rate
V(m3) : bedroom volume
t : time in hours
Assumption : q = 30 1/h for two sleeping occupants.

APPENDIX 2

Development of the relative humidity in a bedroom occupied by two persons.

If the condensation is negligible and if the water vapour cannot be discharged by diffusion, the increase in the absolute humidity rate follows the same law as that specific to carbon dioxide but with C_e variable according to the exterior temperature.

Assumptions :

- $q = 80$ g/h for two sleeping occupants
- 90 % exterior relative humidity at night
- C and C_e are expressed in g/m³.

The relative interior humidity can be calculated from the known absolute humidity C and the interior temperature T_i .

APPENDIX 3

The absolute humidity in a bedroom occupied by two persons reaches the following level after several hours :

$$C(\text{g/m}^3) = C_e(\text{g/m}^3) + \frac{q}{Q} = C_e + \frac{80}{Q}$$

with C_e , the absolute exterior humidity corresponds to T_e and 90 % R.H.

A dew point temperature T_r corresponds to each value of C and if the surface temperature

$$\theta_i = T_i - \frac{T_i - T_e}{3}$$

is less than T_r , condensation is formed.

Each temperature couple (T_i , T_e) has therefore a corresponding ventilation flow rate Q , below which there is a risk of condensation.

The curves in Figure 7 show the limits of Q as a function of T_i and T_e below which condensation can start to form.

Example : if $T_i = 2^\circ \text{C}$, condensation can be formed if Q is less than 15 m³/h.

3. VENTILATION AND ENERGY SAVINGS IN DWELLINGS :

"BEWARE OF MIRAGES"

The new draft regulations concerning the ventilation of dwellings were presented at the Cated-C.S.T.B. one day technical meeting held the 11th June 1981. The orientation chosen to limit the heat losses is clear ; it only incites the interested parties to reduce the air changes and thus dissuades them from recovering the energy sources contained in the air.

In fact, it authorizes the provision of minimum services which will be adopted straight away as the sole objective for reducing the conventional evaluation of coefficient G to the least cost, but will the technological resources used and the behaviour of the occupants as regards these new ventilation conditions allow for gains in heating values of an importance that cannot be anticipated by the theoretical calculation of G ?

We will attempt to answer this question by presenting the heat losses due to ventilation and the theoretical energy gain resulting from a reduction in the air change rate when conventional assumptions concerning the way of life of the occupants and the influence of spurious factors are accepted. We will then attempt to assess the actual gains corresponding to concrete representative examples of current situations.

Finally, the economic aspect of ventilation cannot be dealt with, without tackling the problems specific to the extraction of burnt gases, when boilers or water heaters are installed throughout dwellings, and without drawing attention to the interest as regards the recovery of heat from the foul air or the burnt gases.

4. INFLUENCE OF VENTILATION ON THE CONSUMPTION OF ENERGY IN HEATING INSTALLATIONS

4.1 Heat losses due to air changes and heat gains due to reduction of the air change rate

For an actual air change flow rate Q in m³/h, the instantaneous heat losses are given by the following formula :

$$d_v(\text{watts-hours}) = 0.34 \times Q \times (T_i - T_e),$$

with T_i , being the estimated temperature of the extracted air and taken as the average temperature in the dwelling, and T_e being the exterior temperature.

The overall heat losses D_v during a heating season are equal to :

$$D_v(\text{watts-hours}) = 0.34 \times Q \times DH(t_r)$$

where $DH(t_r)$ represents the number of degrees-hours for a reference interior temperature t_r . $DH(T_r)$ is shown in the table of Figure 1 taken from D.T.U. (Rules Th-G77) of November 1977.

Example : 80 m² (200 m³) flat with an approximate average air change rate of one volume per hour in the main rooms, in compliance with the current regulations in force,

Q average = 140 m³/h if $T_i = t_r = 18^\circ \text{C}$

- in area A :
DH(18) = 73 088 and $D_v \simeq 3\,500 \text{ kWh}$;
- in area B :
DH(18) = 62 928 and $D_v \simeq 2\,999 \text{ kWh}$;
- in area C :
DH(18) = 43 774 and $D_v \simeq 2\,100 \text{ kWh}$.

If the average actual flow rate can be reduced by 30 % (magnitude complying with the objectives of future regulations) without causing disturbances in construction or discomfort for the occupants, the heat losses in areas A, B and C can be reduced by 1 050 kWh, 900 kWh and 630 kWh, respectively.

4.2 Influence of occupant behaviour

The strict application of impending regulations will imply overall ventilation flow rates that can almost be doubled at the initiative of the users. The occupants will in fact have the choice between a normal rate, which in most cases will correspond to the minimum overall flow rate required, i.e. : $Q(\text{min}) = 15(n + 2)$,

with n being the number of main rooms in the dwelling concerned and an accelerated ventilation rate for meeting intensive, but temporary, requirements in kitchens and possibly in bathrooms, with extraction rates higher than the statutory limit values depending on the number of main rooms. Thus, in a four-room dwelling, the accelerated ventilation rate in the kitchen should allow for an extraction of at least 120 m³/h from this area. Attention must be drawn to the authorization as regards the ventilation of bathrooms at a fixed flow rate of 15 m³/h.

The anticipated heating gain from these new arrangements obviously depends on the conventional assumptions concerning the utilization time of the different ventilation flow rates, but, the actual gains could be much lower if the accelerated ventilation rate is used more often than expected. This is not a new problem ; the examples of the C.S.T.B. solution for facilitating the application of the Decrees dated 22nd October 1969 concerning the ventilation of dwellings, already recommend air extraction vents that can be adjusted by the occupants, particularly in the kitchen, but the variations relative to the resulting air change are not as great as those considered in the future and the uncertainty as regards the actual heat losses due to ventilation will be amplified by the new arrangements. The excessive use of the accelerated extraction rate can have several reasons :

- . Forgetfulness : as we have already pointed out, the intensive emanations in the kitchen are quickly noticed and incite the

occupants to increase the ventilation rate, but the return to proper ambient conditions when the very polluting activities have ceased cause no inconvenience and, in general, nothing warns the occupants that the ventilation system is still operating at the accelerated rate. During In-situ visits, it can often be seen that the extraction vents in kitchens are always in the maximum position and many ventilation installers prematurely conclude that adjustable flow rate vents serve no purpose ;

- . The controls for the ventilation vents are often difficult to operate or inaccessible and the occupants resign themselves to permanently maintaining a maximum flow rate ;
- . The lack of information. Members of the ventilation Trade are aware of the heat losses due to air changes but, in general, they do not provide the occupants with this information and the means at their disposal are often badly exploited ;
- . An insufficient or badly distributed normal ventilation rate which is not effective against the formation of condensation and mildew. The minimum overall flow rates authorized by the draft regulation for the normal ventilation rate are sufficient for solving the problems in the main rooms, provided that the fresh air introduced is correctly distributed between all the rooms according to their respective occupation coefficient ; but, if the current design concepts, which are based on the uniform and constant distribution of inlet air between all the bedrooms (with twice the flow rate in the living rooms), are retained, the air change flow rates in the bedrooms will attain a maximum value of 18 m³/h if there is no spurious air admission into the other rooms. In bedrooms occupied by two persons and in which the temperature is lowered (16° C during the night, for instance), these flow rates are too low. As shown by the curves of Figure 7 in the first part of this article, the conditions favourable to the development of mildew will be encountered too often. Faced with these situations, the occupants will adopt solutions to the problem which are not economical from the energy saving point of view ;
- . Maintained accelerated ventilation rate which unnecessarily increases the overall ventilation values for a dwelling, or the opening of windows ;
- . Higher interior temperatures : 18 to 19° C. In fact, it should be recalled that two means can be used for the prevention of condensation ; heating and ventilation, but, the latter approach to the problem is more economic. This fact is clearly illustrated by the example given below.

Example : A bedroom designed for two persons which can be subjected to two distinct ventilation and heating conditions corresponding to equivalent risks of condensation :

a) First configuration :

$$\begin{aligned} \text{Volume} &: 30 \text{ m}^3 & G &= 1 \text{ watt/m}^3\text{C}, \\ Q &= 18 \text{ m}^3/\text{h} & T_i &= 19^\circ \text{ C} \end{aligned}$$

Heat losses in area B,

$$D = 1 \times 30 \times \frac{DH(19)}{1000} = 2.058 \text{ kWh}$$

b) Second configuration :

$$\begin{aligned} Q &= 27 \text{ m}^3/\text{h}, & T_i &= 16^\circ \text{ C} \\ D &= 1 \times 30 \times \frac{DH(16)}{1000} + 0,34 \times (27 - 18) \times \frac{DH(16)}{1000} = 1.713 \text{ kWh}. \end{aligned}$$

If, in the first configuration $T_i = 19^\circ \text{ C}$ for half of the time and $T_i = 16^\circ \text{ C}$ during the other half, the heating energy consumption is still greater than that of the second configuration (1.806 kWh).

We thus draw the attention of the interested parties to the minimum requirements concerning the bathrooms, i.e. a fixed flow rate of 15 m³/h. This tolerance should not be abused for, in many cases it does not allow for the suitable extraction of humidity resulting from the taking of a bath or shower and also from the drying of linen. The drying rate depends on the air change rate and the relative humidity of this air, and therefore, its temperature. Consequently, to solve their problems, the occupants will attempt to modify the original setting of the extraction vents when this operation is easy, or will constantly keep the heating at a temperature higher than 20° C.

4.3 Influences of technological resources used for ventilation

From the project of the future regulations, it can be thought that it is possible to considerably reduce the heat losses due to the air changes without any precautions and without investments higher than those previously encountered. This is not the case as the more the flow rates are adjusted to suit the requirements, the more it will be necessary to install efficient equipment for properly controlling the air.

The most current solutions presently adopted consist in extracting the foul air from the rooms and areas devoted to service installations by means of mechanical extraction equipment or natural draught and allowing the fresh air to flow in through air inlets which are, in general, self-adjustable and located on the facades of buildings.

If these solutions are retained, the air inlets must allow for sufficient openings to allow the air to penetrate without offering too much resistance when under accelerated ventilation conditions.

When the amplitude of the adjustment left to the initiative of the occupants is too great, the passage of the air inlets is excessive for the normal ventilation rate and the wind generates a supplementary cross ventilation which considerably attenuates the interest of major reductions in the extraction rates. For this purpose, mention should be made to the rules Th-G77 of the D.T.U. dated November 1977 relative to the calculation of the air change flow rate to be taken into consideration for heat losses from buildings ; they allow for the extraction and also for the air intake devices. These rules specify that the ventilation specific flow rate q_v to be considered is equal to the greater of the two following values :

- $\sum q_e$ sum of the typical flow rates of self-adjustable air inlets,
- $\sum q_s$ sum of the flow rates for which the specific air outlet devices are designed. In the case of an adjustable flow rate in kitchens, the calculation of q_s is made by adopting the minimum flow rate increased by 15 m³/h.

If these conventional rules are maintained, the air inlets, designed for the admission of the overall maximum flow rate corresponding to the accelerated ventilation rate, will always have a leading role and the interest of high reductions in the extraction flow rates under normal ventilation conditions will never be in evidence. These rules excessively penalize the options combining self-adjustable inlets and mechanical extraction with flow rates that can be adjusted by the occupants, but one must be aware of the fact that beyond a certain adjustment amplitude, these solutions are no longer suitable as can be seen from the example given below.

Example : Estimation (See "Calculation methods and assumptions" in the article written by P. Jardinier : "Ventilation et transparence à l'air des habitations", "Cahier techniques du Moniteur", n) 27, Feb-March 1980) of the influence of wind in a dwelling with double exposure (Fig. 2) by adopting the ideal assumption of negligible spurious leaks. From Figures 3a, 3b and 3c, it can be seen that :

- . when the wind is at a zero level, the extraction flow rate is equal to the admission flow rate and the air intake is homogeneously distributed between the inlets ;
- . when the wind load on facade n° 1 increases, the air admission is unbalanced and the extent of this imbalance depends on the extraction equipment and flow rates ;
- . beyond a specified wind speed, more air enters from facade n° 1 than is discharged through the extraction vents and grilles. The excess air volume flows out through the air inlets on the opposite facade which results in "cross ventilation".

First configuration : Mechanical extraction.

Mechanical extraction of 30 m³/h from the bathroom and toilet and an adjustable flow rate of 60 to 120 m³/h in the kitchen for a depression of 8 mm water gauge at the vents when the air admission meets a negligible resistance, i.e. when window is open.

Admission of air through five facade installed inlets each offering a resistance of 10 pascals for 30 m³/h. This first configuration corresponds to typical current installations and to the examples of solutions recommended by the C.S.T.B.

The extraction flow rates are practically independent of the wind. Average flow rates : 57 m³/h in the kitchen, 28 m³/h in the bathroom and toilet for the normal ventilation rate ; 111 m³/h in the kitchen for the accelerated ventilation rate.

The flow rates for the air inlets according to the wind speed are indicated in Figure 4. It can be seen that the flow rates of the air inlets on facades 1 and 2 are 23 m³/h under normal ventilation conditions when there is no wind. On facade 2, the flow rate reaches 15 m³/h when the wind speed $V \geq 3$ m/s. It is reduced to zero at a wind speed of 4.8 m/s and above this speed, the air flows out through the air inlets installed on facade 2 which results in cross ventilation and the air change in the dwelling exceeds the theoretical value of 120 m³/h.

2nd configuration : Mechanical extraction

With the objectives of the new regulations. Extraction of 15 m³/h from the toilet, 30 m³/h from the bathroom and 45 m³/h from the kitchen at the normal rate.

As the overall flow rate at the accelerated ventilation rate is high, the air inlets are identical to those of the first configuration. The extraction flow rates are stable. in the kitchen 44 m³/h, 15 m³/h in the toilet and 29 m³/h in the bathroom.

Figure 4 shows that the flow rates at the air inlets all have a value of 18 m³/h under normal ventilation conditions and in the absence of wind. As soon as there is a slight amount of wind, the flow rates at facade 2 become too low and the occupants have to maintain the accelerated ventilation rate or half-open a window to ventilate the corresponding rooms, When $V \leq 3$ m/s, these air inlets only allow 5 m³/h of air to enter and when V is higher than 3.5 m/s, the air circulation is reversed and cross ventilation occurs.

For wind speeds higher than 4 to 5 m/s, the admission air flow rates at facade 1 are close to those described for configuration 1. Thus, the air change rates in these two configurations is practically the same in spite of different extraction rates.

3rd configuration : Natural ventilation

Same objectives as the first configuration.
Draught assumptions

- . 5 m high vertical evacuation ducts (last levels of blocks of flats or individual houses) projecting outside at a level higher than any close obstacle ;
- . Average temperature in the dwelling 20° C ;
- . Average exterior temperature 7° C ;
(With allowance made for the low power of the natural phenomenon which generates an average draught of less than 10 pascals to overcome the resistance of the air inlets, the interior doors and the extraction devices) ;
- . The cross section of the air inlets is more than twice that of the air inlets used for mechanical extraction installations.
They oppose a pressure loss of 2 pascals for 30 m³/h ;
- . The evacuation devices (outlet grilles + ducts + outlet) oppose a pressure loss of 3 pascals for their designed flow rates ;
- . Average wind speed of 4 m/s.

In figure 5 indicating the flow rates of these air inlets according to the wind speed, it can be seen that :

- . without wind, approximately 20 m³/h flows in through each of these air inlets at facades 1 or 2 ;

- . as soon as there is a slight amount of wind, the admission is very quickly unbalances ;
- . when V (wind speed) = 2.10 m/s, there is no flow rate at facade 2 and all the air enters at facade 1 ;
- . when V is higher than 2.10 m/s, cross ventilation occurs and the air change is greater than the objectives (120 m³/h).

Figure 6 shows the development curves of the overall flow rate evacuated and the overall air change flow rate under normal ventilation conditions. Half of the overall evacuated flow rate is from the kitchen, with the remaining half being sub-divided between the toilet and bathroom. The difference between two flow rates is due to the cross ventilation. All these values correspond to well defined influence of the wind blowing against facade 1.

Under average conditions corresponding to a wind speed of 4 m/s homogenously distributed from all directions, and in winter, calculations show that the following are obtained under the normal ventilation rate :

- air change flow rate of 133 m³/h instead of 120 m³/h ;
- extraction rate in kitchen of 56 m³/h instead of 60 m³/h ;
- extraction flow rates from bathroom and toilet of 28 m³/h instead of 30 m³/h.

Under the accelerated ventilation rate, an extraction of 100 m³/h instead of 120 m³/h is obtained in the kitchen and 25 m³/h is obtained in the bathroom and toilet.

4th configuration : Natural ventilation

Same draught assumptions and same air inlets as the third configuration.

Figure 5 shows that :

- . without wind, approximately 17 m³/h flows in through each of the air inlets ;
- . the admission unbalance is caused sooner than in the third configuration and, when $V \geq 1.5$ m/s, the flow rate of the air inlets at facade 2 is already zero ;
- . when V is higher than 1.5 m/s, cross ventilation occurs.

Figure 6 shows that half of the overall evacuated flow rate is from the kitchen, with a third from the bathroom and a sixth from the

toilet. When the wind speed is higher than 2 m/s, and according to the orientation indicated in Figures 3a, 3b and 3c, the volume of air admitted at facade 1 is close to that of the third configuration and the air change rates are very similar in both configurations.

When $V = 4$ m/s and the wind is homogeneously distributed, in winter, the following average values are obtained :

- . air change flow rate of 120 m³/h instead of 90 m³/h
- . extraction rate in kitchen of 45 m³/h, 30 m³/h in bathroom and 15 m³/h in the toilet (these are the desired values)
- . under the accelerated ventilation rate, 120 m³/h in the kitchen instead of 120 and 13 m³/h and 26 m³/h in the toilet and bathroom, respectively.

It can thus be seen that the air change flow rate deviates even more from the objectives and is only 10 % less than that of the third configuration.

5. HEAT RECOVERY BY MEANS OF STATIC HEAT EXCHANGERS

Heat pump and static heat exchangers are currently available for the recovery of energy from foul air prior to discharge outside the building.

At first sight, the air-to-air dual flow pump appears to be the most attractive ; but, its operating characteristics are badly suited to heating and ventilation requirements in new and very well insulated dwellings ($G \leq 1$).

As a general rule, the control cannot be accomplished room by room when the pump, which has priority, can supply all needs. A major part of the free heat sources (internal or solar) is thus wasted because of a centralized control system. In addition, this wastage causes discomfort :

- . The modulation of the interior temperature according to the periods of occupation is limited. The operation of the installation is disturbed or impossible when the interior reference temperature, in the event of temporary unoccupation, is lower than 14° C ;
- . The occupants cannot obtain, to their liking, different temperatures in each of the rooms and thus reduce the heating in unoccupied rooms ;

- . These operating conditions are badly suited to the adjustment of extraction flow rates left to the initiative of the occupants. The ventilation flow rate must often, and permanently, correspond to the accelerated rate to be effective during peak hours and the heat losses caused by air changes are unnecessarily increased.

Heat recovery by means of theoretically less ambitious heat exchangers is rather a "rustic" but sure solution which is well suited to dwellings for the utilization and maintenance constraints are extremely limited and of low cost. This system is independent of the heating installation and thus provides the occupants with complete liberty as regards the control of their heating. It can be assimilated with an insulation improvement.

5.1 Heat recovery from foul air in double-flow ventilation installations

The expected results from certain operations accomplished during the last few years have not always been confirmed in actual fact. In these cases, the analysis of the situation shows that the factors responsible for the deceiving results are :

- . Specified performance data not corresponding to reality. Today, there is a test standard E 51-702 and the manufacturers can guarantee the quality of their equipment as a result of tests carried out by CETIAT in compliance with these standards ;
- . Incorrect balancing of the flow rates and leaks in the extraction system prior to entering the heat exchanger, resulting in preliminary cooling of the foul air by mixing with the exterior air. These faults are easy to rectify ; but, the attention of the installers must be drawn to their consequences and they will not hesitate to carefully carry out the necessary operations ;
- . Heat losses from the air between the interior of the dwelling and the heat exchanger in the extraction or blowing systems. This factor is, in general, preponderant. It is difficult to provide the air ducts with a good insulation. We are in favour of the solutions which are beginning to appear and which consist in arranging inside the heated volume, constituted by the dwelling, the heat exchanger and the interconnecting system of the rooms served by the heat exchanger. Thus, the heat losses are negligible. Figure 8 shows a double-flow collective ventilation system with individual heat recovery which resolves all the problems mentioned above and which also complies with the present preoccupation of individualizing the heating costs.

Calculation of energy recovered from foul air :

For a specified flow rate, a heat exchanger is characterized by its efficiency :

$$= \frac{T_1 - T_2}{T_3 - T_1} \quad (\text{See Figure 9})$$

This efficiency is practically independent of the temperatures in the normal operating range.

In the case of an installation comprising a heat exchanger in the dwelling, the efficiency of the installation is nearly the same as that of the heat exchanger and the energy recovered E_r from the foul air is :

$$E_r(\text{kWh}) = \frac{0,34 \times q \times \epsilon \times DH(t_i)}{1\,000}$$

where q is the ventilation flow rate, ϵ the efficiency of the heat exchanger and $DH(t_i)$ degrees-hours corresponding to an average temperature t_i .

Example : dwelling of 100 m², 250 m³ with an average value of
 $q = 165 \text{ m}^3/\text{h}$, $t_i = 18^\circ \text{C}$;

in area B, $DH(18) = 62\,928$;

$$E_i = 2\,471 \text{ kWh, if } \epsilon = 70 \text{ \%}.$$

To calculate the energy gain G , allowance must be made for the consumption of the blowing fan which can easily not be more than 40 watts per dwelling, or approximately 350 kWh per year :

$$G_a = \frac{E_r}{R} - 350$$

if R is the efficiency of the heating installation.

Thus, in the case of well regulated direct electric heating system, $R \simeq 1$ and $G \simeq 2\,100 \text{ kWh}$.

5.2 Heat recovery from foul air and burnt gases

When the gas-fired boilers are installed in the kitchens and the products of combustion are evacuated by mechanical extraction together with the foul air, the interest of the double-flow system with heat recovery is amplified.

To calculate the reduction in energy consumption, it can then be considered that the dwelling has an improved insulation because of the heat recovery from the foul air and the boiler has a better efficiency owing to the heat recovery from the burnt gases.

If we consider the previous example with a gas-fired heating sys-

tem and an initial coefficient before heat recovery,
 $G_i = 1.2 \text{ watt/m}^3 \text{ } ^\circ \text{C}.$

a) Before recovery

- . Annual requirement of the dwelling, B_i :

$$B_i(\text{kWh}) = \frac{G_i \times V}{1\,000} \times DH(18 - \frac{3}{G_i})$$

With the free interior and solar heat gains evaluated at 3 watts/m³ and the average temperature taken into consideration in the dwelling being 18° C :

$$B_i = \frac{1,2 \times 250}{1\,000} \times DH(15,5) =$$

$$\frac{1,2 \times 250 \times 49\,136}{1\,000} = 14\,740 \text{ kWh.}$$

- . Effective energy to be generated by the boiler, E_{ui} : if R_d is the efficiency of the distribution which allows for the losses in the piping, regulation system, etc...

$$E_{ui} = \frac{B_i}{R_d}$$

When $R_d = 90 \%$

$$E_{ui} = \frac{14\,740}{0,9} = 16\,380 \text{ kWh.}$$

- . Initial annual consumption, C_i :

$C_i = \frac{E_{ui}}{R_i}$, with R_i the average efficiency on P.C.I. of the boiler during a heating season.

For $R_i = 80 \%$ (efficient boiler) :

$$C_i = \frac{16\,380}{0,8} = 20\,480 \text{ kWh}$$

b) After heat recovery

- . Heat loss coefficient, G_r :

$$G_r = 1,2 - \frac{0,34 \times 165 \times 0,7}{250}$$

1,04 watts/m³ ° C.

. Annual requirements of dwelling, B_r :

$$B_r = \frac{G_r \times V}{1\ 000} \times DH(18 - \frac{3}{G_r})$$

$$B_r = \frac{1,04 \times 250}{1\ 000} \times DH(15,1)$$

$$= \frac{1,04 \times 250}{1\ 000} \times 46\ 987 \simeq 12\ 220\ \text{kWh.}$$

. Effective energy to be supplied by boiler, E_{ur} :

$$E_{ur} = \frac{12\ 220}{0,9} \simeq 13\ 580\ \text{kWh}$$

. Annual consumption, C_r

If it is accepted that the perceptible heat losses of high efficiency are only due to the evacuation of hot burnt gases, the new efficiency after heat recovery : $R_r = R_i + (1 - R_i) \epsilon$, with ϵ the efficiency of the heat recovery. In the case considered :

$$R_r = 0,8 + 0,2 \times 0,7 = 0,94\ \text{i.e. } 94\ \%$$

$$\text{and } C_r = \frac{13\ 580}{0,94} = 14\ 450\ \text{kWh.}$$

The saving in gas consumption figures can therefore attain 6 000 kWh, or approximately 30 %.

The anticipated gains from heat recovery are extremely advantageous. They can be obtained without reducing the service facilities offered to the users and the savings thus envisaged are less dependent on their behaviour and the exterior conditions.

Isn't it better to undertake this approach rather than to encourage solutions with very uncertain results ?

6. EXTRACTION FLOW RATES

In kitchens, the minimum flow rates specified by the draft regulations under normal or accelerated ventilation conditions appear to meet the objectives in these rooms.

1 room	2 R	3 R	4 R	5 R and more
20-75 m3/h	30-90 m3/h	45-105 m3/h	45-120 m3/h	45-135 m3/h

In shower rooms, we consider that the occupants should be able to obtain a flow rate of 30 m3/h regardless of the number of main rooms in the dwelling concerned and be able of reducing the rate

to 15 m³/h in small dwellings or in dwellings comprising numerous shower rooms. The tolerance of the draft regulations which permit an adjustment of the flow rate to even lower values should, in our opinion, only be taken advantage of if its use is automatically limited to short sequences so as to temporarily increase the ventilation in the kitchen during peak hours.

- . In toilets, the requirements are thoroughly perceived and the ventilation can be temporary. The specifications of the draft regulations appear adequate.

The extraction flow rates must be capable of attaining 15 m³/h or 30 m³/h according to the number of main rooms and toilets in the relevant dwelling.

7. SUPPLY FLOW RATES

- . In bedrooms, the ventilation installations should not permit actual flow rates of less than 20 m³/h, except in the case of an air transfer in favour of the living room and which is automatically limited to a few hours. During the night, the quality of the air slowly degrades in the bedrooms and the sleeping occupants are not conscious of the fact and therefore take not steps to modify the ventilation. The overall minimum ventilation rate must therefore ensure air changes in these rooms which generally prevent conditions favourable to the development of condensation or mildew during the heating season. With a flow rate of 20 m³/h in a bedroom heated to 18° C, these bad conditions will not occur more than 10 % of the nights ; but, if the bedroom is only heated to 16° C, which is a temperature often selected for energy saving reasons, the third of the nights during the heating season are affected and, during these periods, the relative humidity can be higher than 85 %.
- . In living rooms, the polluting emissions from each individual are greater than those encountered in the bedrooms when the occupants are sleeping. In addition, these rooms are intended to accept all the occupants at the same time. These considerations could therefore justify high ventilation rates in the biggest dwellings but the periods of intensive occupation are generally short and the inertia of high volume living rooms enables a fraction of the pollution to be temporarily absorbed. During these same periods, the interior doors are open and the living rooms partially benefit from the air from the unoccupied bedrooms which flows towards the service rooms through the corridors and entrances. The living rooms of small dwellings are less occupied but only slightly, if at all, benefit from the ventilation of other main rooms during the day.

On account of these observations, we feel that the occupants should be able to obtain a minimum overall flow rate of at least 30 m³/h in the living rooms of dwellings with one or two main rooms and, at least, 40 m³/h in the living rooms of other dwellings.

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 17

ENERGY CONSERVATION BY REGULATION OF THE CENTRAL MECHANICAL
VENTILATION IN HIGH-RISE BUILDINGS: REALISTIC OR NOT?

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1. INTRODUCTION

Within the framework of energy conservation in dwellings, the Steering committee, Energy and Buildings (SEG) has ordered the TNO Institute for Environmental Hygiene (IMG) to perform an investigation into the possibility of achieving energy savings by the regulation of the mechanical ventilator units in high-rise buildings [5]. The SEG was formed some years ago upon request of the Dutch Ministry of Housing and Physical Planning and the Ministry of Economic Affairs. The idea of regulating the ventilators at times when there is less requirement for ventilation, for example at night and outside the peak cooking and showering periods, has without doubt arisen from the popular conception that mechanical ventilation wasted energy.

This supposed energy waste can be divided into:

- the energy consumption of the ventilator
- higher energy losses due to the ventilation air flow.

The energy consumption of the ventilator is only a small fraction (ca. 4 %) of the energy required to warm up the ventilation air flow. It has not yet been shown that the average ventilation air flow for a heating season with mechanical ventilation is higher than with natural ventilation. Dwellings that are equipped with mechanical ventilation systems have no natural ventilation channels and are therefore more impervious to air. Above a certain windspeed, it appears that the ventilation air flow in a dwelling with natural ventilation is greater than that for a dwelling with mechanical ventilation. Dwellings that are equipped with mechanical ventilation systems have no natural ventilation channels and are therefore more impervious to air. Above a certain windspeed, it appears that the ventilation air flow in a dwelling with natural ventilation is greater than that for a dwelling with mechanical ventilation. The windspeed at which this occurs depends, among other factors, on the wind direction and the hermeticity of the dwellings (See Fig. 1). The air flow in mechanical ventilation systems is, in principle, regulated according to the values given in the Dutch standard "Ventilation in Dwellings" (NEN 1087) [1].

This standard is based upon:

- health of the occupants
- technical and hygienic considerations.

According to the ventilation standard NEN 1087, a person needs a ventilation air flow of $0.007 \text{ m}^3/\text{s}$. This is equivalent to $25 \text{ m}^3/\text{hour}$. In a recent investigation of limited extent in office buildings [4], it was found that with such a ventilation air flow, about 10 % of the occupants found the inside air unacceptable in terms of freshness.

The technical and hygienic considerations include:

- the prevention of the air in "wet" areas, such as the toilet, the kitchen, the bathroom and the shower, from becoming too damp, in which case mould growth could occur.

- the prevention of odours spreading from the above "wet" areas to other parts of the dwelling.

A specified air flow is therefore required for these "wet" areas. The ventilation standard (NEN 1087) states that the occupant of a dwelling must be able to achieve the required ventilation air flow with the use of the ventilation system provided. However, the occupant is left completely free in the manner in which he regulates the ventilation system: windows or grilles open or closed, ventilator fan(s) on or off. Mechanical ventilation is required in high-rise buildings and for open kitchens. Regulation of the ventilation is also permitted with mechanical ventilation, provided that the occupant is able to regulate the system so that it fulfills the requirements of the ventilation standard.

Recently, more and more publications have appeared on the subject of excessive radiation levels in dwellings as a result of radioactive gasses coming from building materials, and reports of excessive concentrations of other gasses, for example from chip-board.

In a number of these publications, the consequences of too low a level of ventilation are considered. In Sweden, in these circumstances, an air change rate standard of at least 0.5 per hour is used. The air change rates used in Holland are estimated to be between 0.5 and 1.0 per hour for mechanical ventilation. The air change rates for mechanical ventilation systems are thus certainly not excessive when compared to the Swedish standard.

The opening of windows will affect the energy consumption of a dwelling. In order to see how much the occupant can affect the energy consumption in this respect, a number of calculations with windows open and closed have been performed in this investigation.

2. OBJECTIVES

The object of this investigation was to see what energy saving could be achieved by the regulation of mechanical ventilation systems in high-rise buildings.

3. INVESTIGATION

The investigation can be divided into a number of parts:

- (1) measurements on a mechanical ventilation system
- (2) calculation model for this mechanical ventilation system
- (3) measurements on the air leakage on the façades of a flat
- (4) calculation model for this flat.

A calculation model (2) has been built for a mechanical ventilation system with twelve extractor points and a ventilator fan that can be used in two settings: "half" and "full speed". The system is based on the results of measurements on a flat in Delft (1). The calculation model provides the air flow in a building or part thereof, for example air flow through the:

- façades
- roof
- internal walls
- ventilation system.

For this the calculation model uses data concerning:

- wind pressures
- air temperature differences that result in pressure differences
- fan characteristics
- characteristics of seams, joints, windows, doors, ducts and extractor vents.

In this calculation procedure, the air pressure at which the incoming and outgoing air flows are balanced, for each room, was ascertained.

The results from the first calculation model (2) and the air leakage values (3) which were determined within the framework of another project [2] were used to set up a second calculation model (4) of a flat. This can take account of variations in the positions of windows and doors, the windspeed and direction and the temperature inside and outside. This model delivers a prediction of the amount of energy lost due to the ventilation. The effect of open windows is simulated from the results of another investigation [3]. At this point it should be noted that in this project [3] measurements on windows were made on buildings mostly sheltered from the wind. Many windows in high-rise buildings will not be in such a sheltered position, so that the wind will have a correspondingly greater effect on the ventilation and energy loss than the results calculated. The effect of an open window in an otherwise closed room can best be imagined by considering the window in two parts:

- one part through which air enters
- one part through which air exits.

The average air speed in such an open window lies in the range 0.1 to 0.5 m/s. In the case of a high building, with a high windspeed and an open window that catches the wind, the wind can actually blow as hard in the window opening as it does outside. An open window thus gives, per square metre of opening, an exchange of $0.5 \times (0.1 \dots 0.5) = 0.05 \dots 0.25$ cubic metres of air per second, and in certain circumstances even more. This is equivalent to 180 to 900 m³/hour per square metre of window opening. This is an important factor for heating or cooling calculations, since large heat flows can arise as a result of an open window.

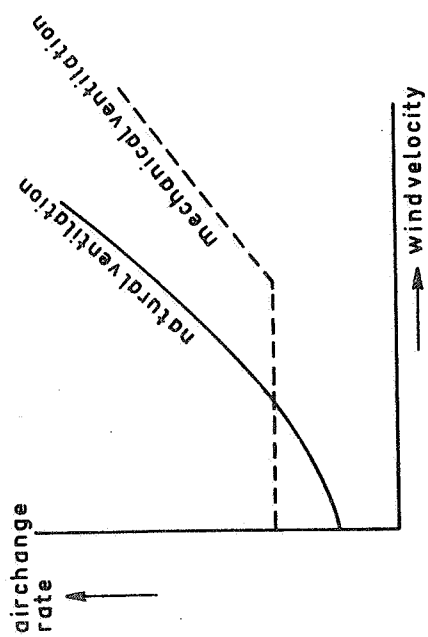


Fig. 1. The relation between the air change rate and the wind velocity.

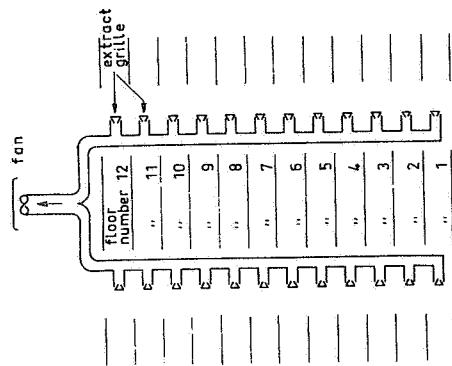


Fig. 2. Schematic diagram of the mechanical ventilation duct.

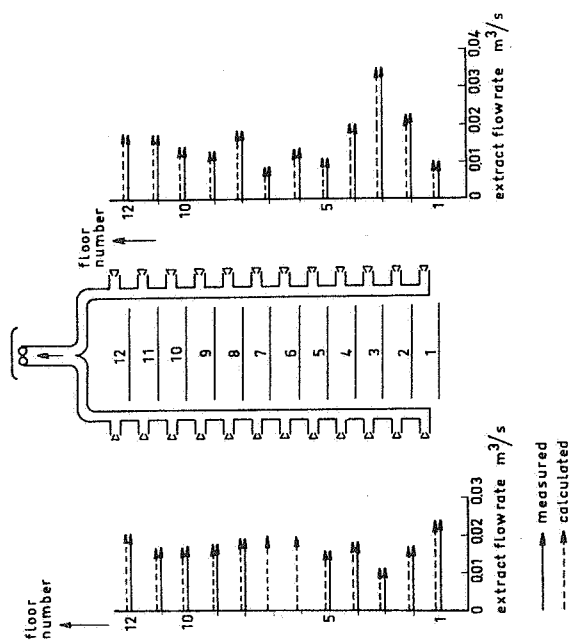


Fig. 3.1. Extract flow rates.

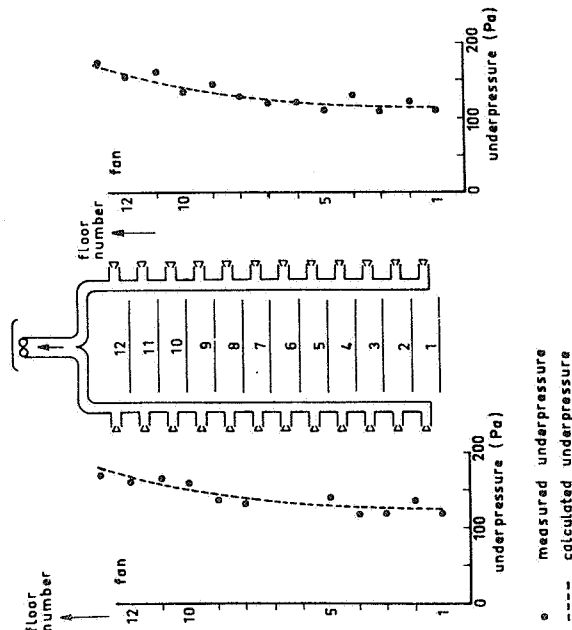


Fig. 3.2. Under pressure in the mechanical ventilation duct.

4. METHODS OF MEASUREMENT AND RESULTS

4.1 Measurements on a mechanical ventilation duct

Figure 2 shows a schematic diagram of the layout of the ventilation ducts from the kitchens of the block of flats in which the measurements were made. The toilets and the bathrooms in these flats were also mechanically ventilated.

The measurements could not be performed simultaneously at all points. However, it appeared that the wind and the positions of windows and doors do not have great influence on the air flow and the pressures in the ventilation duct. Thus, the overall result of the measurements can be considered a good average. In Fig. 3, the arrows indicate the air flows and the dotted lines indicate the measured negative pressures in the duct. The air flows were measured with a measuring orifice. The pressure loss due to this orifice was compensated by an additional ventilator.

The extractor vents often had to be cleaned before measurements could be made. With a dirty extractor vent, the air flow was often less than 5 % of the nominal flow. Many occupants had not previously realized that it is possible to clean extractor vents.

Most occupants were actually not satisfied with the air extraction in the kitchens. They often thought that the cooking vapours should disappear directly into the extractor vents. Even an extractor hood is not always capable of doing this. The kitchen window was therefore often opened during cooking. Moreover, some occupants had changed the original setting of the vent to achieve better air extraction. The kitchen door had been removed in many flats. This allows considerable spreading of vapours from the kitchen through the flat and, via the staircase, to other flats. The occupants complained about this and the mechanical ventilation system for these odours, while the cause must mainly be attributed to the missing kitchen doors and the, generally inevitable, chimney effect of the staircase.

4.2 Calculation model of the mechanical ventilation system

The calculation model was used to perform a simulation of the mechanical ventilation system in the block of flats in Delft. In Fig. 3, the calculated air flows and negative pressures are indicated with broken lines. In this simulation, the air resistance of the different parts of the system and the capacity of the ventilator were chosen so that calculated air flows agreed with the measured values. The calculated negative pressures were at times at variance with the measured negative pressures.

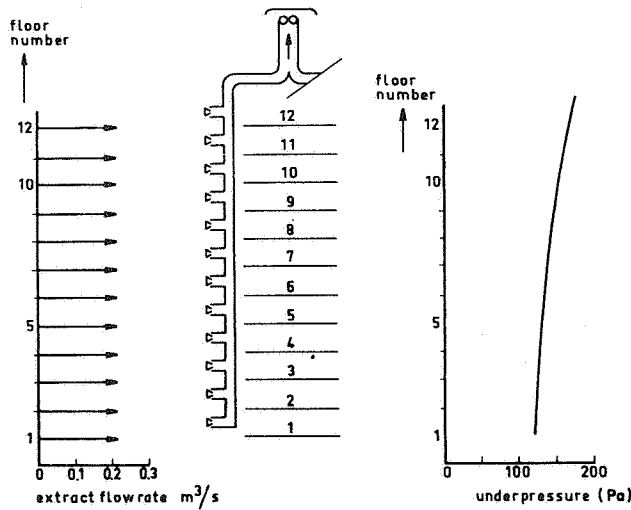


Fig. 4. The model used in further calculations. A flow rate of $0.021 \text{ m}^3/\text{s}$ is extracted at every grille.

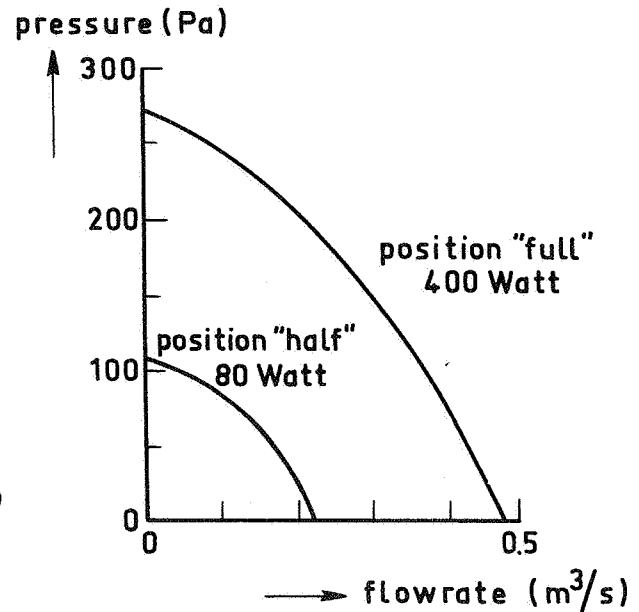


Fig. 5. Fan characteristics.

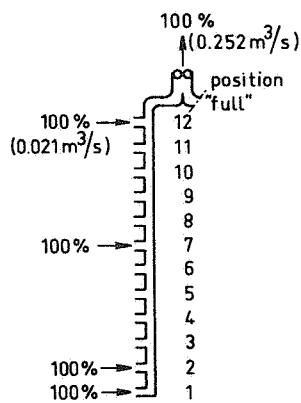


Fig. 6.1. Reference situation. Fan switched on "full".

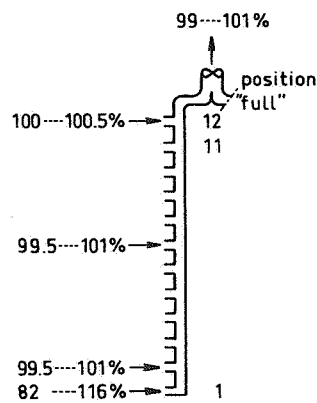


Fig. 6.2. Open window on the first floor. (leeward...windward side)

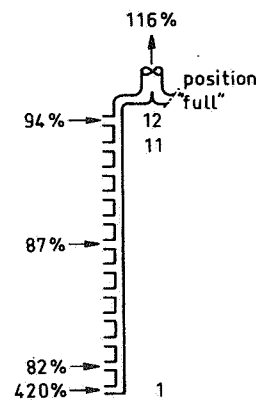


Fig. 6.3. Grille on the first floor removed.

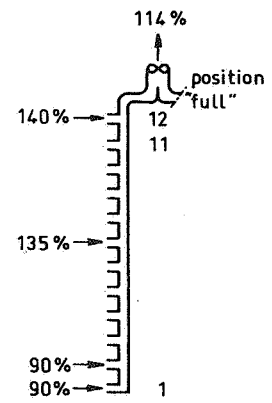


Fig. 6.4. Upper six grilles opened 1.5 x more wide.

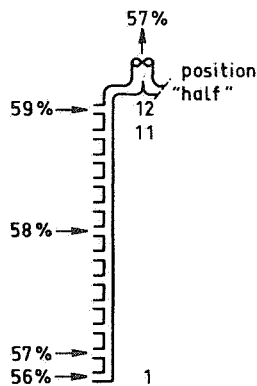


Fig. 6.5. Reference situation. Fan switched on "half".

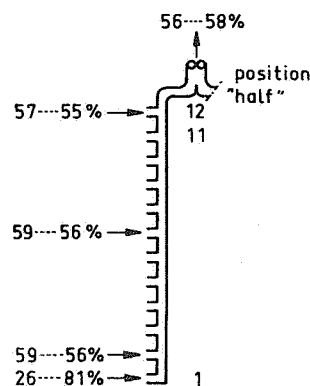


Fig. 6.6. Open window on the first floor. (leeward...windward side)

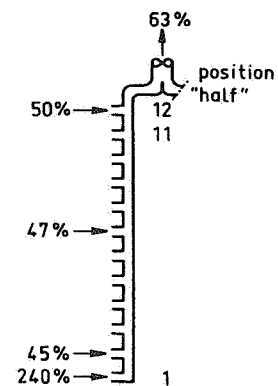


Fig. 6.7. Grille on the first floor removed.

This is partly caused by the fact that all measurements could not be made simultaneously.

To limit the number of situations to be calculated, a model of the ventilation system for a column of twelve flats was made, in which nominally $0.021 \text{ m}^3/\text{s}$ air was extracted per kitchen (See Fig. 4). Calculations were performed for about sixty situations with this model, which included combinations of the following parameters:

- ventilator setting "full" or "half"
- several windows open
- the setting of several extractor vents changed, or even one removed for cleaning purposes
- regulation to hold the negative pressure in the duct, at the ventilator, constant
- outside temperature 15°C or -5°C .

Figure 5 shows the ventilator characteristics for the settings "full" and "half". It is assumed that the pressure change in the kitchen due to opening a window is $+40 \text{ Pa}$ on the windward side and -40 Pa on the lee side. These pressure changes can be expected for a windspeed of 10 m/s . Figure 6 shows a summary of several situations that were calculated. In figures 6.1. to 6.4 the ventilator was set to "full", while in Figures 6.5 to 6.7 it was set to "half". The values adjoining the figures give the ratio of several air extraction flows compared to a reference situation ($0.021 \text{ m}^3/\text{s}$ in Fig. 6.1 and $0.012 \text{ m}^3/\text{s}$ in Fig. 6.5). The effect of opened windows on the air flows on other floors appears to be a maximum of 1 % in Fig. 6.2 and a maximum of 3 % in Fig. 6.6. The removal of an extractor vent reduces the air flow on other floors by a maximum of 8 % in Fig. 6.3 and by a maximum of 24 % in Fig. 6.7.

With the ventilator at "full" setting, the effect of changing the setting of the extractor vents was examined: the vents on floors 7 to 12 were opened an extra 50 % (See Fig. 6.4). On the other floors (1 to 6) the air flow appeared to be reduced by about 10 %. The negative pressures with the ventilator at "half" setting were about 50 Pa (170 Pa at "full" ventilator setting). With the wind pressures of $+40 \text{ Pa}$ and -40 Pa that are assumed, no flow reversal will occur in the ventilation duct. A further reduction of the negative pressure could well lead to flow reversal (See Fig. 7).

4.3 Air leakage

Figure 8 shows the plan of the flat for which, in the scope of another project [2], air leakage measurements were made on the façades. Equation [1] gives the air flow through a façade as a

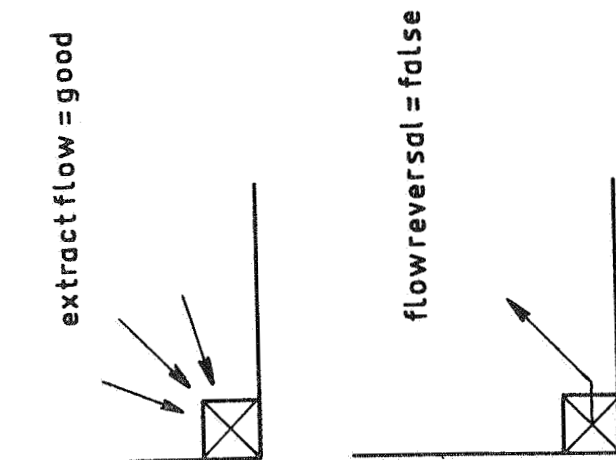


Fig. 7. Extraction and flow reversal in ventilation ducts.

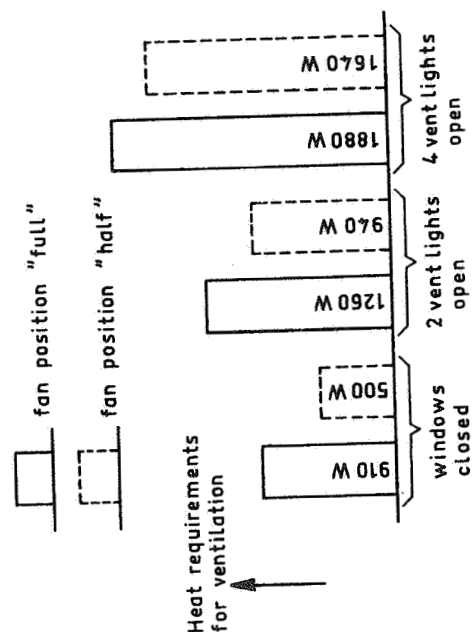


Fig. 9. Mean heat required for warming up the ventilation air per flat.

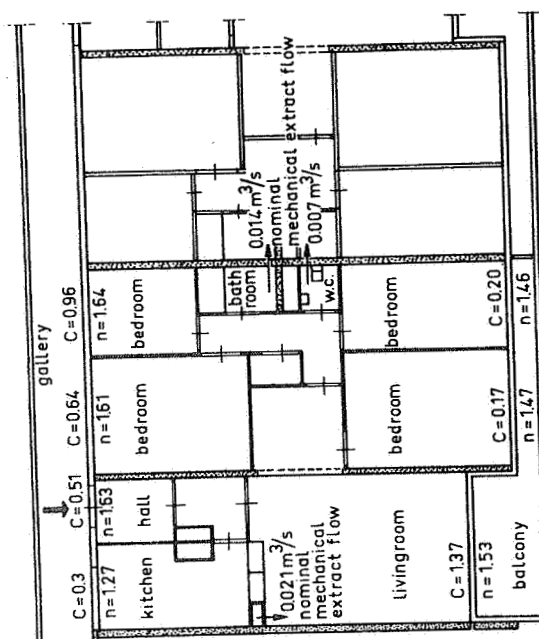


Fig. 8. Floorplan of the flat in Delft. Air leakage coefficients and exponents are indicated.

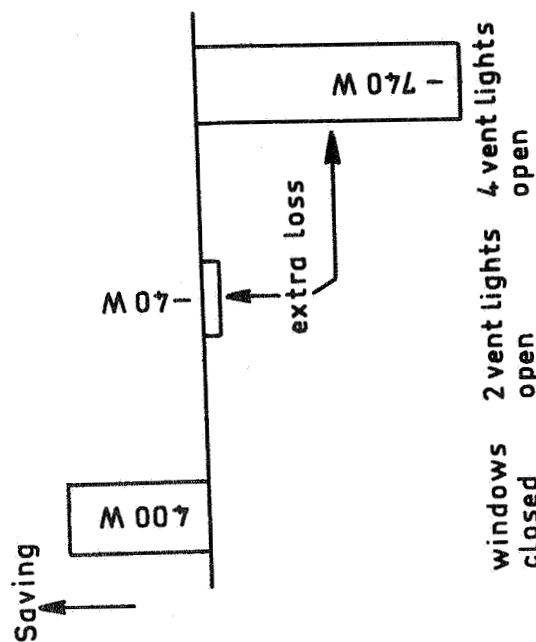


Fig. 10. "Energy savings" per apartment due to switching the fan from "full" to "half".

function of the air pressure difference Δp across the element, the air leakage C and the exponent n that describes the cause of the flow.

$$q = C.(\Delta p)^{1/n} \dots \dots \dots (1)$$

4.4 Calculation model for the flat

A calculation model for one of the twelve flats was set up based on the measured air leakage of the flat. Using this model, the air leakage through the façades, internal walls and the extractor vents of the kitchen, toilet and bathroom and the heat flow due to ventilation were calculated. This was performed for about 95 situations in which combinations including the following parameters were chosen:

- setting of the ventilator "full" or "half"
- several windows open
- air leakage of the flat
- windspeed and direction
- situation and surroundings of the block of flats from a wind protection point of view.

Several results of this calculation model are presented in Fig. 9. The calculations were made for an air temperature difference of 20 °C between inside and outside. The reduction in the energy consumption as a result of the lower power of the ventilator at low speed is not included in Fig. 9. This ventilator, for 24 flats, has a power of 400 W at "full" setting and 80 W at "half" setting. This means a saving of 13 W (electrical) per flat. These figures are extremely dependent on the type of ventilator used.

From Fig. 9, it can be seen that with the ventilator at "half" setting, the heat loss due to ventilation for a flat is 200 W to 400 W lower than with the ventilator on "full" setting. Comparing the heat flow due to ventilation for the ventilator at "half" setting and various windows open with that for the ventilator at "full" setting and all windows closed, we see from Fig. 10 that:

- by opening two windows the heat flow is already higher
- with four windows open, the heat flow is 700 W higher.

For a heating season of 200 days and a boiler efficiency of 60 %, this value of 700 W is equivalent to 630 m³ natural gas (35 MJ/m³) per year. Despite the reduced mechanical ventilation, the energy consumption is thus increased by the extra opening of windows, see Fig. 11.

The calculation model was also used to see if, with the ventilator on "half" setting, odours could spread into the flat, for instance from the toilet, see Fig. 12. For these calculations, a temperature difference of 4 °C was assumed between the toilet and the

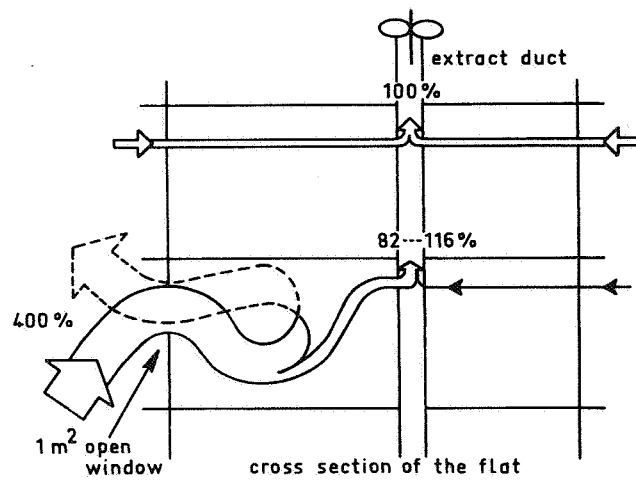


Fig. 11. Ratio of air flows through:

- mechanical ventilation system
- closed windows
- open windows.

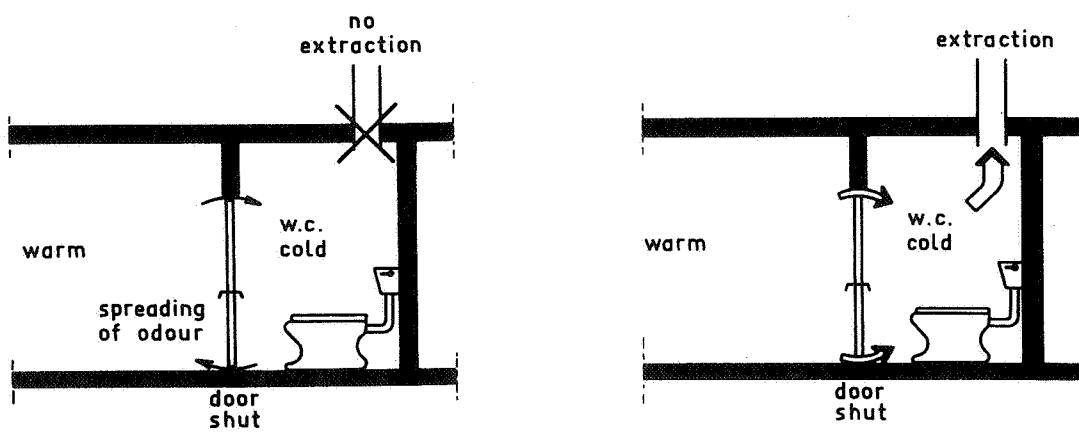


Fig. 12. Spread of odours through a shut door.

rest of the flat. Even with the ventilator on "half" setting, this temperature difference had virtually no effect on the air flow through the door joints of the toilet. No spreading of odours occurred. With the door open, however, there is certainly a spreading of odours, but this is almost independent of the size of the extraction air flow and therefore will really not increase due to reduction of the ventilator speed. For internal spaces with all doors closed, the ventilation air flow will be lower than required by the ventilation standards. In these cases, the chance of damp problems arising will therefore be greater. We have known cases in which the regular switching off of the mechanical ventilation system has lead to condensation and mould growth in bathrooms.

5. CONCLUSIONS

The reduction of ventilator speed in a communal mechanical extractor system has no disadvantageous consequences for the operation of the system, provided that the negative pressure in the duct remains about 50 Pa.

From this investigation, it appears that the extracted air flow is affected more by incorrect adjustment and fouling of extractor vents than by opening of windows, regardless of whether the ventilator is at the "full" or "half" setting.

The extra energy loss due to several open windows appears to be greater than the energy saving possible by reducing the ventilator speed. The energy saving possible is then also completely dependent on the reactions of the occupants: if they open a window for a couple of hours a day, the overall energy consumption can be increased instead of being decreased, regardless of the lower speed of ventilation.

When the mechanical ventilation is reduced, the occupants are still able, in general, to set a sufficient ventilation air flow in rooms that have either a ventilation window or grille. Further, with reduced mechanical ventilation and windows and internal doors closed, the spreading of odours from the "wet" rooms is not to be expected. The chance of damp problems arising in these "wet" rooms will, however, be greater. The saving in electricity due to reduction of the ventilator speed is only a fraction (about 4 %) of the energy losses that can be associated with the ventilation.

6. ACKNOWLEDGEMENT

The author gratefully acknowledges the financial support provided by the Netherlands Ministry of Housing and Physical Planning.

7. LIST OF SYMBOLS

C	air leakage	(m ³ /s at 1 Pa)
n	exponent in equation (1) (1 < n < 2)	(-)
p	air pressure compared to external reference pressure	(Pa)
Δp	difference in air pressure, for example across a façade	(Pa)
q	air flow	(m ³ /s)

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 18

USER EXPERIENCE OF MECHANICAL VENTILATION IN HOUSES

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SYNOPSIS

Four 'low energy' electrically heated houses incorporating extra thermal insulation and heat recovery mechanical ventilation systems, were built for sale by two national builders.

Their standard house designs were modified to accommodate the extra features; air tightness was made an objective so that the ventilation system would provide as much of the required fresh air as possible.

Instrumentation was installed during construction to monitor the performance of the houses, with the co-operation of the eventual purchasers.

The houses were leak tested and sealed where necessary. Ventilation rate calibrations were carried out to provide data for estimating the ventilation rate when the houses were occupied.

The mechanical ventilation systems were found to provide good internal conditions without opening windows and freedom from condensation. The total ventilation heat loss was less than one quarter of the total heat loss from the house.

1. INTRODUCTION

When the U-value of the walls of a house has been reduced to $0.3 \text{ Wm}^{-2}\text{K}^{-1}$ and the ventilation rate is one air change per hour, the transmission and ventilation heat losses are approximately equal, each of the order of 4000 kWh per heating season for a 200m^3 house at a mean internal temperature of 18°C . It is therefore necessary, for minimum heating costs that this ventilation rate should not be exceeded without good cause; if it can be reduced, significant savings in heating energy will result.

Provided moisture from clothes drying and moisture and smells from cooking can be prevented from entering the house air circulation, then 0.5 to 1 air change per hour (whole house) satisfies accepted ventilation criteria¹ for the average family in the average house. This low ventilation rate would minimise heat losses and could be improved upon only by dehumidification or heat recovery.

An essential component of the Electricity Council Research Centre (ECRC) low energy house programme was to ascertain how far this ideal could be realised by existing builders constructing speculative electrically heated housing for the owner-occupier market.

Two national builders modified one of their popular designs of three bedroom, brick and tile detached houses according to guidelines worked out at ECRC. The philosophy was to achieve the best that conventional building techniques could achieve without using revolutionary or untried procedures which would result in confusion on the building site, and also might not work. The houses,

two per builder, were sold on the open market to private individuals who agreed to allow ECRC to monitor the thermal performance of the houses for a limited period (i.e. one or two heating seasons).

2. VENTILATION STRATEGY

Methods of improving the thermal performance of a house envelope are relatively straightforward, whereas the best way to achieve a low and predictable ventilation heat loss is less clear.

While openable windows are virtually essential for summer cooling, they can result in very high heat losses, or low internal temperatures, if used much in winter.² On the other hand, inadequate ventilation will result in stuffiness, odours and condensation.

Local extract fans in kitchens and bathrooms offer a marked improvement in internal environment but there is still a high probability that living rooms and bedrooms will be poorly ventilated. Provision of a ducted fresh air distribution system ensures that all rooms are adequately ventilated, and while this more than doubles the cost and complexity of the ventilation system, the available option of recovering heat from the exhaust air to pre-heat the fresh air is an extra bonus.

3. VENTILATION SYSTEM

Since these houses were for sale in the market place, a commercially available heat recovery ventilation system was chosen. A cooker hood type unit was chosen since it seemed desirable to have those parts requiring maintenance or cleaning to be readily accessible in the kitchen rather than forgotten in the loft. The kitchen unit includes two fans, heat exchanger and controls. A pair of 125mm diameter ducts pass from this unit through the wall to the outside 'balanced flue' type terminal. These are the fresh air supply and exhaust air ducts. A similar pair of ducts on the other side of the kitchen unit convey fresh air to the living room and bedrooms and collect air extracted from the bathroom and W.C. The whole system is shown schematically in Figure 1.

There are three fan speeds designated Low, Normal and Boost, together with a damper on the kitchen hood to increase the extraction rate above the cooker at the expense of the other points. The system is designed for continuous running.

All inlet and outlet registers were located at or near the ceiling following the manufacturer's recommendation. Ducting to the first floor rooms passes through the airing cupboard and into the loft. Access for ducting to the ground floor ceiling depends on the floor joist direction being suitable for laying the ducts in between them. There was an extract point in the bathroom and downstairs W.C. (one of the house designs only). Fresh air was supplied to all the

bedrooms and to both ends of the combined lounge/dining room. The internal doors were considered sufficiently leaky to permit return air flow without any modifications.

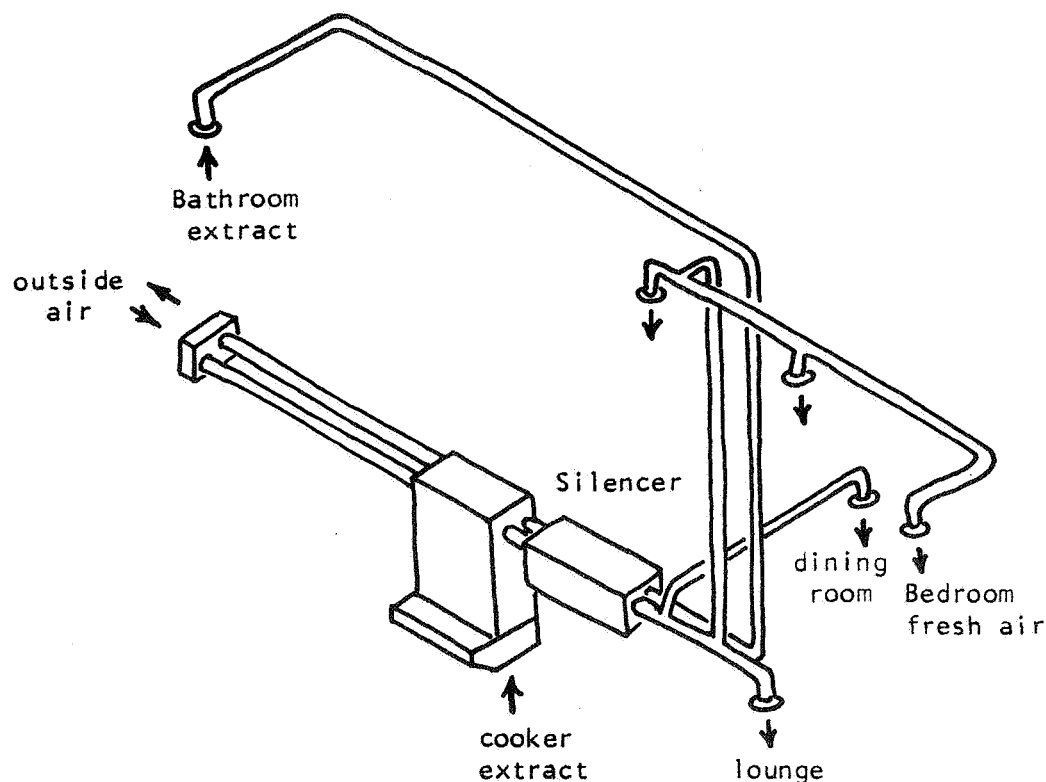


Fig. 1 Schematic view of complete ventilation system

4. VENTILATION RATE

Previous tests have shown that for a balanced ventilation system the mechanical and natural ventilation rates are virtually additive. The mechanical rate was set at 0.5 air changes per hour ($100 \text{ m}^3\text{h}^{-1}$) at the Normal setting on the assumption that the background natural leakage could probably be reduced to about 0.3 air changes per hour, giving a total rate of about 0.8 ach^{-1} . The flow was divided equally among the various registers.

5. HOUSE STRUCTURE

The houses were built with sufficient extra thermal insulation in the walls, roof and floor to give a transmission heat loss of about 120 WK^{-1} , corresponding to a wall U-value of $0.35 \text{ Wm}^{-2}\text{K}^{-1}$. To minimise uncontrolled ventilation the structure had to be less leaky than traditional housing in the U.K. Most of the leakage in conventional brick houses occurs where masonry and timber abut because the timber shrinks on drying out and gaps appear at joints and corners. Air tightness has never been a design criterion before.

A first step was to specify 'high performance' door and window frames with integral weatherstripping. Ideally the frames should be sealed into the brick wall with a flexible gasket but no practical solution could be found so, reluctantly, mastic was used after construction in the usual way.

Wet plaster finish rather than dry lining was specified because it seals the myriad of small leakages in the wall. Dry lined walls are prone to unpredictable and unsealable leakage unless very carefully installed with air tightness in mind.

Even a wet plaster finish does not seal the wall surface between the ground floor ceiling and the first floor. Joists set into holes in the inner leaf provide leakage holes through the wall which can be avoided by using joist hangers.

The other main source of leakage is where services penetrate the envelope. Unfortunately the mechanical ventilation system itself adds to the number of penetrations. The boxing-in of services can provide leaky 'chimneys' from floor to floor and usually into the loft space unless the holes are made good at each floor penetration.

Incorporation of the ventilation system also influences house design. To achieve low pressure drops and low noise levels the main distribution ducting has to be 125mm diameter and the branch ducts to individual rooms must be 80mm diameter. With ducting of this size it is not possible to notch floor joists and so the main problem in system design is how to travel at right angles to the joist direction. Modification of joist layout may be necessary.

It was hoped at first to avoid taking the ducting into the loft because, although this is very convenient for distribution, the ducting must then be thermally insulated and also the penetrations of the ceiling are likely sources of leakage unless carefully sealed. However, the use of ceiling mounted air registers makes ducting in the loft inevitable.

The type of ventilation unit chosen influences, and may severely limit, the kitchen layout because:

- (i) there must be a suitable route through an outside wall for the main supply and extract ducting.
- (ii) a large silencer on the house side of the kitchen unit determines the minimum distance between the unit and the kitchen wall on that side.

Difficulties also occur on site because the ventilation unit has to be in place at an early stage in the construction so that the ducting can be installed prior to ceiling and floor boards.

6. WORKMANSHIP

A low-leakage house is a new concept which is not understood on

British building sites. Although potential leaks should be anticipated and 'designed out' this is almost impossible. The experience of these two projects showed that a satisfactory house is unlikely to be built by subcontract labour because the supervision, motivation and commitment are lacking. It is essential that the men on the job understand that any hole through the envelope is undesirable whether it can be seen or not. For example, square holes around circular ducting going up into the loft will not be made good if the part inside the house is boxed in and the hole in the loft is concealed under a thick blanket of glass fibre insulation.

7. LEAKAGE TESTING

The houses were tested by a conventional pressurisation test using a fan installed in an external doorway. The outside terminal of the ventilation system was blocked off during tests. A smoke puffer was used for locating leaks while the door fan was running at constant maximum speed.

However, sealing after leak testing is no substitute for tight construction as it is very laborious to seal leaks at this stage and not very effective because only the surfaces of the cracks are accessible. Identified cracks and openings were sealed with mastic and polyurethane foam with insulating board spanning large openings. Additional draughtstripping was necessary on some of the external doors. This whole operation was surprisingly difficult and time-consuming, and showed that sealing a house 'after the event' is not a practical proposition even if a pressurisation fan is available to identify the leaks.

The leakage values measured were in the range 5 to 7 air changes per hour at a pressure differential of 50 Pa. The houses, even after careful construction, leaked at all the 'usual' places such as corners (especially at wood/masonry joints), outside doors and window frames, at service penetrations, loft hatches. Two of the houses had particularly bad leaks around the 'balanced flue' ventilation terminal where no attempt had been made to minimise the size of the hole in the wall or to fill in the hole after fixing. A first floor bay window proved to be a leaky type of structure. Loft hatches tended to leak even when draughtstripped, showing the need for positive closure rather than just relying on the weight of the hatch.

Leakage increased by about 20% during drying out but was reduced again by sealing up visible shrinkage cracks.

8. VENTILATION MEASUREMENTS

So that future ventilation rates could be predicted from weather records, ventilation system usage and open windows, the house was calibrated by tracer measurements using both decay and continuous

(equilibrium concentration) methods. In winds up to 7 ms^{-1} and temperature difference of 10°C to 20°C rates of 0.25 to 0.5 air changes per hour were measured (figure 2). With open windows rates up to 10 h^{-1} were measured (figure 3). Measurements with the ventilation system running, and windows closed, confirmed that natural and mechanical ventilation was additive.

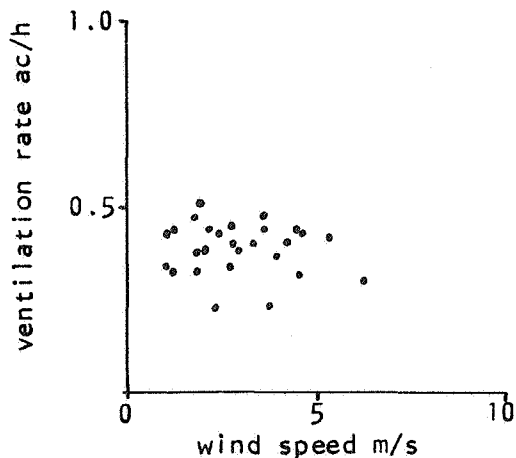


Fig. 2 Measured ventilation rates (one house): ventilation system off and windows closed

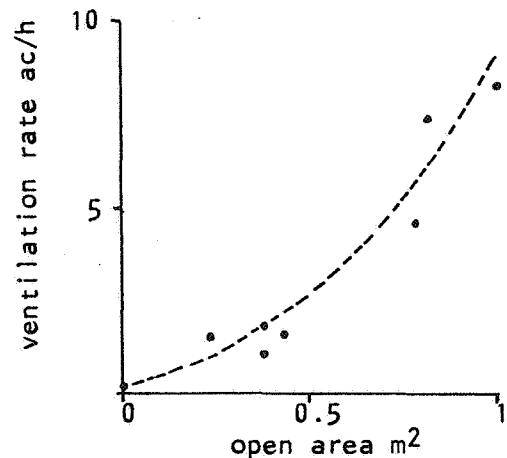


Fig. 3 Measured ventilation rates with open windows

9. INSTRUMENTATION

The house was provided with thermocouples for monitoring room temperatures and recording kWh meters; also the ventilation system was instrumented to measure inlet and outlet air temperatures and dewpoints. The fan speed and damper settings were also monitored on event recorders. An intruder alarm indicated whether a window or door was open. The data-logger recorded the actual duration of window and door opening and whether a particular switch setting on the ventilation system had been used within the previous half hour.

10. USER EXPERIENCE

10.1 Use of controls

In all four houses the occupants chose to use the Low setting of the ventilation system for more than 80% of the time. This corresponds to a mechanical ventilation rate of $65 \text{ m}^3\text{h}^{-1}$. They switch up to Normal ($105 \text{ m}^3\text{h}^{-1}$) or more commonly to Boost ($150 \text{ m}^3\text{h}^{-1}$) for periods of half to two hours around meal times. At these times the kitchen damper would also be open; it was seldom opened at other times.

Apart from a preference for the Low setting, the ventilation systems were used as intended. They were operated continuously and there was no significant window opening until the end of April, and very little until mid-May. Up to mid-April the total (i.e. mechanical + natural) fresh air ventilation was in the range 0.6 to 0.8 air changes per hour (figure 4) for all four houses, with 50 to 55% of this being mechanical.

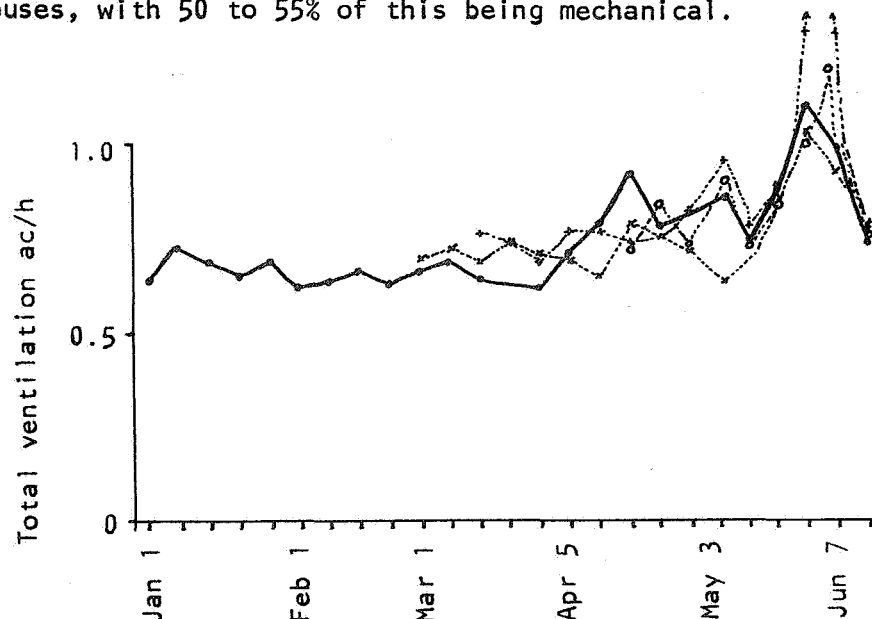


Fig. 4 Weekly mean ventilation rates Jan-June (all 4 houses)

A typical usage pattern for a week in November is shown in figure 5.

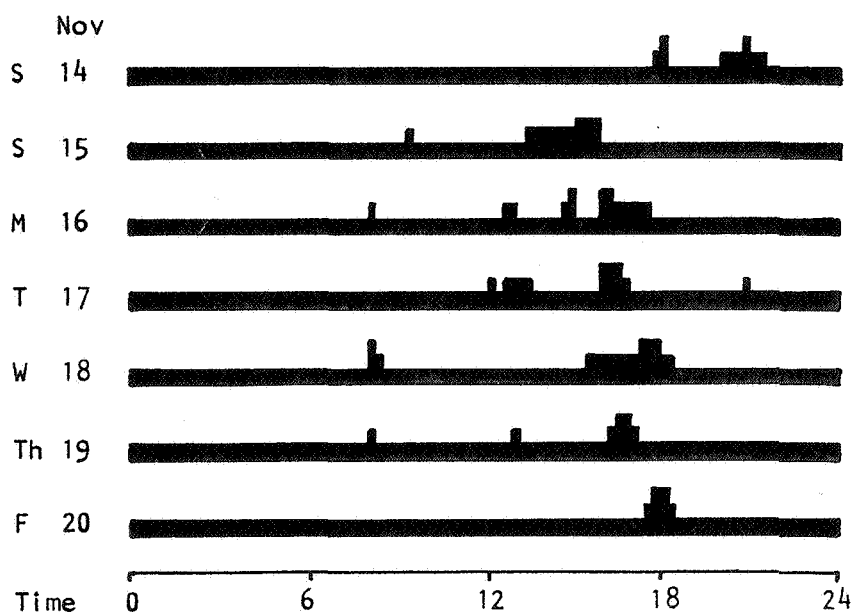


Fig. 5 Ventilation system usage for 1 week in November

Key: Boost
Normal
Low

10.2 Subjective assessment

Only one of four houses was occupied in time for monitoring a full heating season. Records for the other three commence in March. However, operation through choice on the Low setting and the lack of window opening until the beginning of May (when the mean outside temperature reached 12°C) suggests that the ventilation obtained has been satisfactory in all the houses.

There has been no condensation and this is seen as a major advantage of the systems. It also seems that window opening is not indulged in when internal conditions are acceptable.

10.3 Internal environment

Twenty-four hour dew point profiles for one of the houses averaged over the period November to March are shown in figure 6. The internal dew point is about 4.5 degrees above external dew point corresponding to an extra moisture load of 0.0015 kg water per kg air. At a ventilation rate of 160 m³h⁻¹ this corresponds to a moisture production rate of $1.2 \times 160 \times 0.0015 \times 24 = 7$ kg per day.

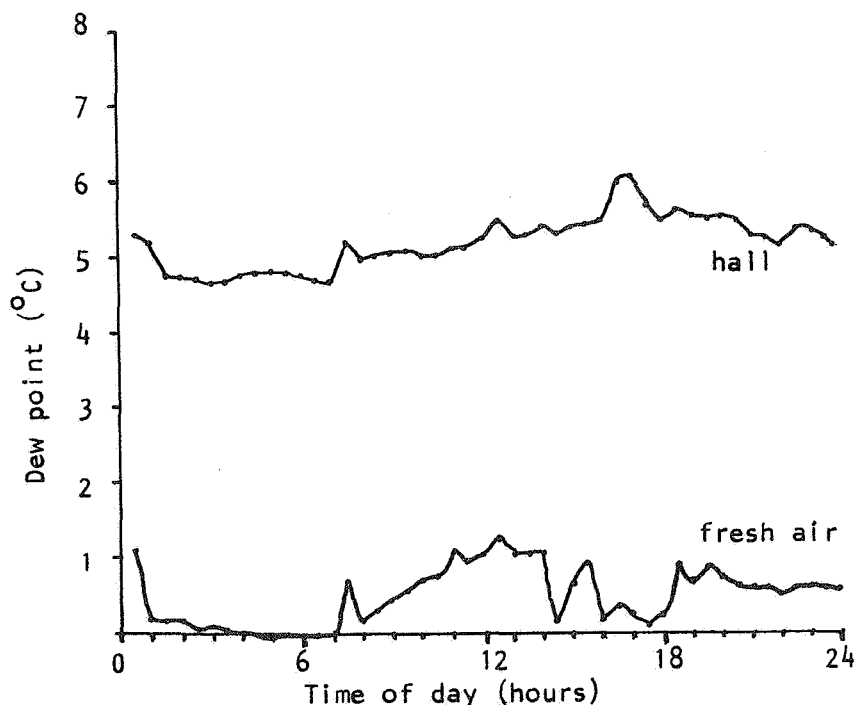


Fig. 6 24 hour dew point profile (November to March)

The dew point sensor in the exhaust duct of the ventilation system was not working until April. Dew point profiles for April are given in figure 7. It is seen that the dew points in the ventilation duct are very similar to those in the hall with moderate increases at cooking times as would be expected.

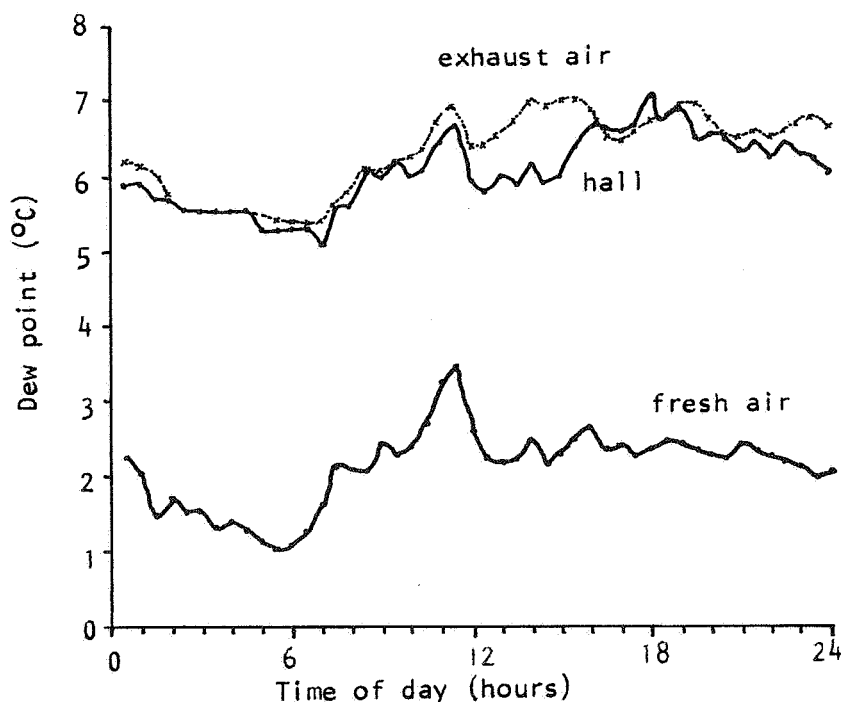


Fig. 7 Dew point profiles (April) including ventilation system exhaust

Individual records show intermittent dew point peaks to 10°C or 12°C but at internal temperatures of 18 to 20°C condensation would not be expected under these conditions, even on the window surfaces (double glazed), which will be at a temperature of approximately inside temperature minus 40% of the inside to outside temperature difference.

10.4 Energy implications

The reason for installing a mechanical ventilation system was to give a low, predictable, ventilation heat loss and an environment which was sufficiently 'fresh' to make window opening unnecessary and undesired during the heating season.

The heat loss data for November to March is given in figure 8. It is seen that the ventilation heat loss is always between a third and a quarter of the transmission heat loss.

The air leaving the house has a greater specific enthalpy than the incoming fresh air because of its higher moisture content. Some of this is recovered in the heat exchanger since the exhaust air is cooled to nearly the outside air temperature as dew point. This gives the heat exchanger an apparently higher efficiency than it would have in dry air.

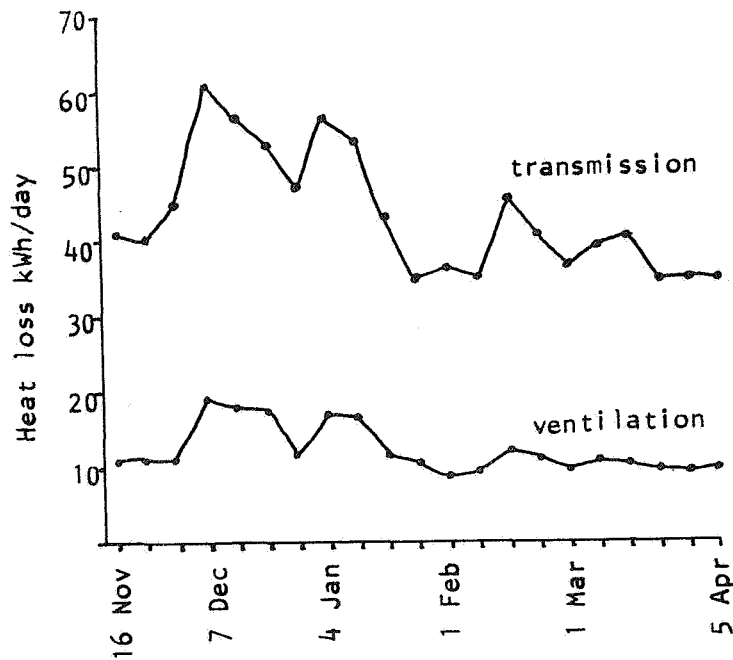


Fig. 8 Transmission and ventilation heat losses
- daily average for each week 16 Nov to
5 April

If fresh air enters the heat exchanger at the outside temperature T_o , is supplied to the rooms at a temperature T_1 , and exhaust air leaves the rooms at the mean internal temperature T_h the apparent efficiency of the heat exchanger is $(T_1 - T_o) / (T_h - T_o)$ which has values of about 0.8. This means that the energy cost of the extra mechanically provided ventilation is only 20% of what it would be if provided by the usual routes, apart from the net fan power loss which is less than 2 kWh/day.

11. CONCLUSIONS

Heat recovery mechanical ventilation systems can be successfully built into conventional electrically heated British housing with only minor modifications. However, to obtain the full benefit much more care than is customary must be taken to build a house which is as leak-tight as possible.

Air tightness is a new criterion for houses both on the drawing board and on site. A learning period is to be expected at the end of which it should be possible to construct a house which does not have to be sealed as a separate exercise after completion.

However, even with half of the ventilation supplied by accidental leakage, it is possible to provide a comfortable environment with no condensation and low running costs.

User reaction is favourable. Systems are used as they were intended to be and there has been no window opening during the heating season.

12. ACKNOWLEDGEMENTS

The co-operation of the Building Site Agents and of the owner-occupiers in these four houses has been essential to the success of the project and is gratefully acknowledged.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 19

EFFICIENCY OF AIR-TO-AIR HEAT EXCHANGERS IN OCCUPIED HOUSES

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Synopsis

The purpose of the project has been to determine the saving in energy obtained in the practical operation of an FTX-system -that is, a fan-controlled supply and exhaust ventilation system with heat recovery - compared to an F-system, which is solely a fan-controlled exhaust system.

The investigation, carried out in a terrace-house district in Skellefteå, showed the following savings for the FTX-systems in comparison with the F-systems:

in 1-storey houses (81 m^2 , airtightness approx. 1):
appr. 1000 kWh/year

in 2-storey houses (99.5 m^2 , airtightness approx. 3):
appr. 1250 kWh/year.

This saving is only about 40% of the theoretical saving at the same air exchange in the different experimental groups.

This low saving was discovered to be due to the differences in air exchanges in houses during the warming-up period. The cause of this is partly the airtightness of the houses, which influenced infiltration, and partly the draught problem met in the houses with the F-system. Because of the draught problem the windowframe ventilators were closed and thus a lower air exchange ventilation was obtained.

Special attention must therefore be given to these draught problems if F ventilation is planned in new construction.

A fairer comparison of the saving in energy is to make the comparison at the same air exchanges (0.5 exchanges per hour), but with the corrections for infiltration that occur during operation.

In this case the saving in energy becomes:

for a 1-storey house: appr. 1900 kWh/year,

for a 2-storey house: appr. 2700 - 2800 kWh/year.

EFFICIENCY OF AIR-TO-AIR HEAT EXCHANGERS IN OCCUPIED HOUSES

In new construction the requirement for airtightness of the house as laid down in Swedish Building Standards 1980 applies. This requirement usually means that fan-controlled supply and exhaust air must be installed. In this way is created a possibility of controlling the airflow that ventilates the building. Buildings put up earlier often have a relatively large involuntary ventilation because of air-leaks. In such cases the total flow of ventilation air is difficult to control.

With balanced ventilation it is possible to install equipment for the recovery of heat from the exhaust air to the supply air. Both the energy thus saved as well as the total energy requirement to maintain a certain climate in the building can be calculated. The involuntary ventilation, which in the more airtight building is relatively small, and the associated energy requirement can however only be calculated approximately. Knowledge about the extent to which degree calculations on energy saving are relevant when the plant has been in operation under actual working conditions over a longer period is however almost completely lacking. Nor has the actual energy consumption in buildings utilizing heat recovery been compared to any great extent which the actual energy consumption in buildings without heat recovery. In order to increase knowledge within these areas it seemed useful to compare energy consumption in a number of similar buildings where some are fitted out with fan controlled supply and exhaust ventilation without heat recovery and some with fan controlled supply and exhaust ventilation plus heat recovery.

Occurrence of FTX-systems

Balanced ventilation - fan controlled supply and exhaust ventilation - plus heat recovery (known in Sweden as the FTX-system) is nowadays just as common as fan controlled exhaust ventilation (known in Sweden as the F-system) in newly produced one-family houses. An investigation carried out by the Swedish Association of Heating and Ventilation Engineers shows that approximately 20 000 FTX-systems and some 19 000 F-systems have been installed in Swedish one-family houses during 1980. Of the approximately 35 000 one-family houses completed in 1980 almost 50% have an FTX-system and just as many an F-system, including an F-system with an exhaust ventilation heat pump.

Requirements by authorities

For FTX-systems in one-family houses it is necessary, among other things, that the heat exchanger packet is easy to get at so that it can be cleaned by the user. The filter in the kitchen range ventilation hood shall be able to separate fat to such a degree

that the least possible quantity is deposited in the duct system and the heat exchanger. The filter shall be cleaned at regular intervals.

In the Swedish Building Standards for 1980 the regulations state that the ventilation shall be so constructed that cooking odours and air pollutants are not spread from the kitchen and rooms for personal hygiene. These requirements mean that the kitchen range ventilation hood may not be connected to a regenerative (cyclic) heat exchanger, for this also recycles water-soluble substances in the exhaust ventilation to the supply ventilation. In certain types of regenerative heat exchangers part of the exhaust ventilation is recycled because the direction of the flow through the heat exchanger packet is altered.

The project packet

The Swedish Council for Building Research (BFR) has initiated a large project concerning heat recovery from ventilation air using heat exchangers, partly for one-family houses and partly for blocks of flats. Implementation of the project has been carried out in two stages:

Stage 1: Companies were invited to submit their technical solutions.

Stage 2: The implementation of a selection of the solutions submitted in Stage 1.

It was suggested by BFR that for one-family houses the project should comprise about 30 like buildings and for the blocks of flats four similar buildings within one and the same district.

In Stage 1 BFR received six applications for participation in the one-family house project and six for the project involving blocks of flats. Because of the difficulties met in discovering suitable housing areas there was a large falling off for Stage 2. Altogether the following projects were obtained:

One-family houses:

- o at block Anderstorp 20, Skellefteå, with heat recovery unit type Bahco Minimaster ACC. Comprises 30 houses. The project was completed in 1981.
 - o at block Malstenen, Bomhus, Gävle, with heat recovery unit type:
 - a) Husqvarna Reginair FTX in 15 houses,
 - b) Elektrostandard's exhaust ventilation heat pumps type Aquarex in 15 houses.
- The total number of houses is 44. Measurements commence in the autumn of 1982.

Blocks of flats

- o at Elinsborgsbacken, Tensta, with Luftkonditionering AB Heat-pipe system. The houses concerned are gallery-access houses erected at the beginning of the 1970s. Each house contains 25 flats, there being five flats on each of five floors.
Measurements will be completed in the spring of 1982.
- o at Stenbacksvägen, Söderhamn, with plate heat exchangers from Air Frölich. The houses were built in 1970 and each contains 30 flats divided between three floors.
Measurements will be completed in the spring of 1982.

The present paper deals only with the Skellefteå project.

Planning of the experiment

There are a number of factors of varying importance that may influence the results of such a study as this. It is important to be as clear as possible about which factors are involved and whether it is possible to give consideration to them or whether consideration can be given. It is already known from earlier investigations that the consumption of energy can vary greatly between houses of the same type. This variation depends to a certain extent on differences in the houses' construction and position and also on the behavioural differences of the occupiers.

If the variations in energy consumption between houses are large in relation to the expected effects of the action taken it may in practice be difficult to decide whether it has had any effect at all. This problem can be resolved by measuring all those factors that influence consumption. One disadvantage of such a procedure however is that the measurement program becomes extremely comprehensive and complicated. A simpler method is to use statistical theory and by means of suitable planning of the project arrange to transfer the variation concerned to a random one subjected to statistical control. The measurements can then be concentrated on the factors that are most important, and the unexplained variations can remain unexplained. Here the effect of a certain step taken means the average effect. The second and simpler method has been utilized in the Skellefteå project.

We have proceed from 15 + 15 houses, that is to say 15 houses without heat recovery and 15 houses with it, as there is then a 55% probability of obtaining a statistically confirmed difference between the experimental and control groups if the saving is 3 000 kWh and the spread is 4 000 kWh. Greater probability may be achieved by increasing the number of houses. Alternatively the ratio between the expected saving and the spread in the material can be increased, which was done in the present project as factors were measured that were of importance for the unexplained variation.

Anderstorp 20, Skellefteå

Anderstorp 20 is a terrace-house district - National Association of Tenants' Savings and Building Societies (HSB) - where initially 31 houses were selected for inclusion in the investigation. All of these houses had type Bahco Minimaster ACC heat recovery units installed. In 16 of the houses the air supply fan was shut off and the inflow duct was plugged. Thus the ventilation system in them has functioned as a fan-controlled exhaust ventilation system (an F-system). The remaining 15 houses remained unaltered and so functioned as a conventional balanced ventilation system plus heat recovery (FTX-system). One house in the second group was omitted as the tenants refused to take part in the investigation. Two types of houses were included in the investigation; one-storey houses of 81 m² (Figure 1), and two-storey houses of 99.5 m² (Figure 2).

Measurement program

The following variables have been measured for all houses involved using weekly readings of the instruments:

- o Inside temperature (average weekly value)
- o Total consumption of electricity
- o Consumption of electricity for heating
- o Consumption of electricity for hot-water heating.

Additionally in FTX-houses the following weekly measurements were carried out:

- o The temperature of supply air after the post-heater of the heat recovery unit (average weekly value)
- o Length of time the post-heater was in operation.

The outside temperature was measured at a central spot in the district. In addition to these measurements the airtightness of the houses was checked by the so-called pressure method as well as in some cases with the assistance of a thermal camera. The total air exchange of the houses in normal operation was checked using tracer-gas measurements during various different external conditions. Fan-controlled supply and exhaust ventilation was measured by means of measuring devices built into the ventilation system.

Instrument readings have mainly been done by the tenants themselves as well as by HSB's caretakers in the district. Sometimes the personnel of the National Swedish Institute for Building Research (SIB) have carried out supplementary readings. Every week forms with reply paid envelopes were sent from the Institute. After completing the forms with the various meter readings the tenants and caretakers returned the forms. Measurements were carried out from September 1980 to August 1981.

Results

The airtightness of the houses is of great importance for the saving in energy that may be anticipated. Therefore careful measurements of this were carried out in all 30 houses. The results of these measurements are shown in Table 1. The variation in any particular group is small, but between both groups of one-storey and two-storey houses the difference is comparatively large.

House type/ Vent. system	Tightness in airchanges per hour when $p = 50 \text{ Pa}$
1-storey/F	3.1 ± 0.4
1-storey/FTX	3.25 ± 0.25
2-storey/F	0.94 ± 0.18
2-storey/FTX	0.97 ± 0.19

Table 1. Airtightness of the houses included in the investigation.

In both F as well as FTX ventilation an undesired air leakage is found that depends on the weather, so-called involuntary ventilation. The involuntary ventilation must be small if the advantages of heat recovery are to be made available. The tighter the house the greater the saving of energy, and the better the advantages of the FTX-system are utilized.

It is generally thought that F-ventilation produces a low pressure in the house that to a certain degree is able to resist the pressure differences caused by the weather. With the FTX-system this counterbalancing low pressure is not present, which causes this system to be more dependent upon the weather.

In order to obtain some idea of the low pressure in the F houses the pressure was measured in the more airtight group (2-storey houses). When the windowframe ventilators were open the low pressure was approximately 2 Pa; when closed the figure was approximately 6 Pa. This small difference in pressure of 2 Pa with open ventilators has in practice very little stabilizing effect. It therefore seems reasonable to state that if the F-system is to have a more stabilizing effect than the FTX-system then the air supply ventilators should be closed. However, this should not be the normal setting of the devices in an F-system.

In houses with F ventilation there were general complaints about draught. But in the FTX houses there were only occasional complaints about draught, which could be dealt with by turning up the thermostat in the air supply duct. Because of the draught problem in the F houses the residents closed their windowframe ventilators and thus obtained a somewhat lower air exchange in their homes whilst concurrently the low pressure producing a higher stabilization was obtained so that the air leakage was less dependent on the weather.

The size of the reduction in the air exchange was measured. It was discovered that the exchange dropped by 0.06 air exchanges per hour in the 1-storey houses and 0.12 per hour in the 2-storey houses. The reason why the decrease is greater in the latter is that these houses are more airtight.

It should be noted that before the measurements were started the airflows in all houses were adjusted to values corresponding to 0.5 exchanges per hour, which is the value laid down by the building standards. During the measurement period deviations from this value were obtained, because of the problems described above.

The energy savings measured are for:

- 1-storey houses 0.155 kWh/degree and day
- 2-storey houses 0.17 kWh/degree and day.

These figures have been obtained by calculating the total energy consumption used for heating, that is to say the total of:

- o energy for heating via radiators,
- o energy for the post-heating of the air supply (FTX-system),
- o 50% of the household's electricity consumption and
- o energy consumed by the air supply fan (FTX-system).

These measured values signify that in Skellefteå with 155 500 degree-hours we get the following savings of energy with the FTX-system:

- a 1-storey house $0.155 \times 155\,500 \frac{1}{24}$ 1000 kWh/year
- a 2-storey house $0.17 \times 155\,500 \frac{1}{24}$ 1100 kWh/year

As the energy consumption of the air-supply fan during the periods when the outside temperature exceeds +11°C cannot be utilized the values above are decreased by 87 kWh. If a small error in the adjustment of the airflows is also taken into account the following corrected values are obtained:

- a 1-storey house approximately 1000 kWh/year,
- a 2-storey house approximately 1250 kWh/year.

These differences thus constitute the savings obtained in actual operation with the FTX-systems in comparison with the F-system in these two types of houses. The savings will be compared with those obtained from a theoretical calculation. According to the manufacturer of the heat-recovery units the temperature efficiency in dry heat exchangers at the airflows concerned is 64%. The investigation has resulted in an average value during the year of 69%. The higher figure may be explained by the formation of condensation in the heat exchangers during a large part of the year.

When it is known how the airflows in the system fluctuate with external temperatures, then the proportion of maximally recovered energy η_e (η_e = efficiency of energy) can then be calculated. This proportion, η_e , is defined as:

$$\eta_e = \frac{\text{energy obtained}}{\text{available energy}}$$

On the basis of these the degree of energy efficiency has been calculated to 51% on average during the heating-up period. In doing so consideration has also been given to the defrosting period of the unit.

Recovered energy when $\eta_e = 51\%$ and the number of degree-days 155 000 becomes at the actual airflows (0.028 and 0.033 m³/s respectively):

per 1-storey-house:

$$0.51 \times 155\,500 \times 0.028 \times 1.2 \times 1 = 2665 \text{ kWh/year}$$

per 2-storey house:

$$0.51 \times 155\,500 \times 0.033 \times 1.2 \times 1 = 3140 \text{ kWh/year.}$$

Taking into consideration, as earlier, that the energy consumption of the air-supply fan cannot be utilized during the periods when the outside temperature exceeds +11°C the above values are decreased by 87 kWh. This gives the following theoretical savings:

per 1-storey house: 2578 kWh/year,

per 2-storey house: 3053 kWh/year.

In the one-storey houses it has been determined that there was a heat loss in the air-supply ducts of approximately 180 kWh during the heating-up period.

The differences in saving that can be attributed to the fact that the air exchange was different between the two groups of houses can now be calculated:

a 1-storey house: 2578 - 180 - 1100 $\approx$ 1400 kWh/year

a 2-storey house: 3053 - 1250 $\approx$ 1800 kWh/year.

If the whole difference in energy consumption is explained by lower air exchange in the F houses this lower air exchange becomes Δn_1 and Δn_2 respectively as follows:

$$\text{per 1-storey house } \Delta n_1 = \frac{1400 \times 3600}{81 \times 2.4 \times 1.2 \times 1 \times 155500} = 0.14 \text{ h}^{-1}$$

$$\text{per 2-storey house } \Delta n_2 = \frac{1800 \times 3600}{99.5 \times 2.4 \times 1.2 \times 1 \times 155500} = 0.14 \text{ h}^{-1}$$

The conclusions that can now be drawn with regard to the average air exchanges during the year in the different groups of houses and from the measurements made is that the air exchanges had the values shown in Table 2:

Type of house/ Vent. system	Average air exchange Exchanges/hour
1-storey/F	0.40 - 0.45
1-storey/FTX	0.55 - 0.60
2-storey/F	0.35 - 0.40
2-storey/FTX	0.50 - 0.55

Table 2. Average air exchanges in the different groups of houses.

Assessment of profitability of the heat recovery units

The installation of a heat recovery unit is of course undertaken in order to save energy. A profitability calculation can be carried out that takes into consideration the efficiency of the recovery system, the capital, operational and maintenance costs as well as the temperature conditions of the district concerned.

In assessment of profitability it seems reasonable to carry out the comparison at an air exchange of 0.5 air exchanges per hour for both groups, that is to say for both the F- and FTX-systems, but with the corrections for infiltration that occur during operation.

On the basis of the houses in Skellefteå the saving with the same air exchange should be for a:

- 1-storey house (81 m²) approximately 2400 kWh,
- 2-storey house (99.5 m²) approximately 3050 kWh.

Presuming that the correction for increased infiltration in the one-storey houses with the FTX-system is approximately 0.05 exchanges per hour greater than in the F houses and half of this value in the more airtight two-storey houses, the following savings are obtained with the FTX-system compared to the F-system:

- for a 1-storey house approximately 1900 kWh,
- for a 2-storey house approximately 2800 kWh.

With an energy cost of 0.20 Skr/kWh the savings are:

- for a 1-storey house: 380 Skr
- for a 2-storey house: 550 Skr.

The additional cost for an FTX-system as an alternative to an F-system can be estimated at about 8 000 Skr for a smaller one-family house. So the saving in this case corresponds only to an annual interest of 5 and 7% respectively. Therefore if only the financial result is considered the installation cannot be regarded as profitable.

However, the great difference between the F- and FTX-systems has proved to be the climate in the houses. The draught problem found in the F houses has been eliminated in the FTX houses. It is not easy to evaluate this in terms of cash but it is undeniably a factor clearly advantageous to the FTX-system. In view of this it can perhaps still be said that if the airtightness factor is better than 3, then heat recovery shows a profit.

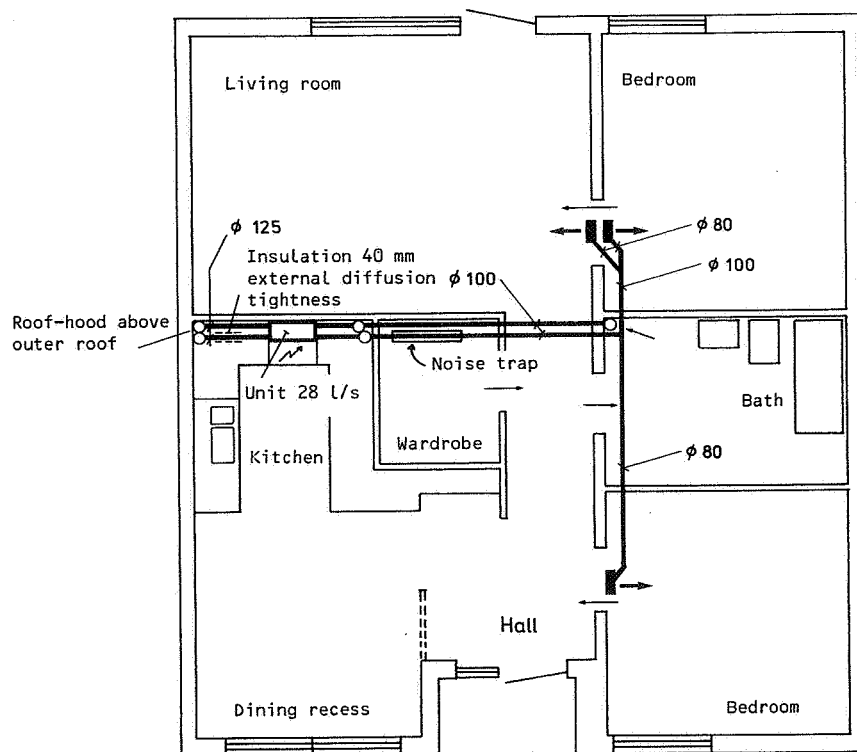


Figure 1. One-storey house

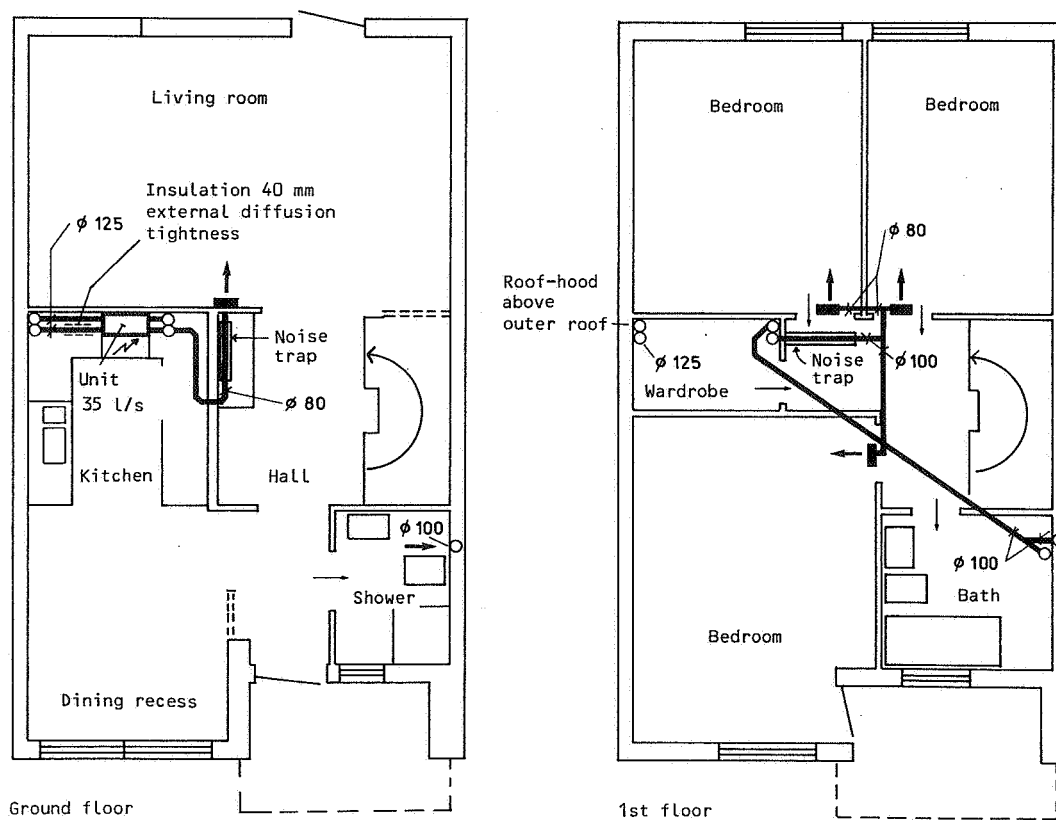


Figure 2. Two-storey house

ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

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PAPER 20

THE ROLE OF MATHEMATICAL MODELLING IN THE DESIGN OF ENERGY
EFFICIENT VENTILATION SYSTEMS

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SYNOPSIS

With the need to conserve energy, there has been a growing tendency towards the design and construction of tight dwellings. The airtightness of older houses is also being increased in efforts to reduce energy consumption. As this process continues, it is becoming increasingly more important to pay particular attention to the ventilation needs of a dwelling.

In this paper, the various roles that mathematical models can play in the design of energy-efficient ventilation systems are explored. An example is used to illustrate how models may be used to investigate the energy implications and air distribution patterns of a range of mechanical and natural ventilation options. Particular attention is focussed on predicting the performance of these systems for a wide variety of conditions.

It is concluded that models of proven validity are an invaluable aid in ventilation design studies. They enable the performance and cost benefit of ventilation strategies to be quickly assessed and have important applications in building energy analyses.

1. INTRODUCTION

As a result of uncertainties in the future supply of energy, coupled with substantial increases in its cost, it has become increasingly important for many countries to focus particular attention on methods to conserve energy. In the United Kingdom, it is estimated that 51% of primary energy is used in buildings of which approximately 60% is used in the domestic sector^{1,2}. This compares with the results of a recent analysis of energy statistics based on data from 15 industrialized countries in which prime energy consumption in buildings ranged from between 23% to more than 50%, with the domestic sector accounting for between 56 to 68% of this proportion³. Thus the conservation of energy in dwellings has become an important consideration in many national energy strategies.

Hitherto, methods to reduce energy consumption in dwellings have largely concentrated on limiting heat loss through the building fabric. It is now thought the cost effective level of thermal insulation has been attained and that future emphasis will be directed at controlling ventilation and air infiltration losses³. A variety of studies have shown that such losses can account for between 25% of the space heating demand in a poorly insulated dwelling to 50% for a well insulated building^{4,5,6}. Hence these losses represent a substantial proportion of the domestic energy demand. However, a building cannot simply be made more airtight in an effort to reduce ventilation heat loss. An inadequate supply of fresh air and poor air distribution will result in high levels of indoor air pollution, odours and condensation. Thus proper provision for ventilation must be made if energy

reductions are to be successfully achieved without detriment to the indoor environment.

In temperate climates, the normal method of ventilation has often been to rely on the natural movement of air through adventitious openings in the building fabric, supplemented by the opening of windows. On the otherhand, in countries with severe climates and limited energy resources, the need to incorporate energy efficient ventilation systems as an integral part of building services design has long been recognised. Clearly, to achieve the objective of minimising energy consumption, a general move towards the latter approach is required.

Mathematical models have an important role to play in this design process because they enable the many ventilation options to be assessed quickly and cheaply. They can also be used in cost benefit studies to identify the optimum design solution for a particular building and environment.

In this paper, some of the models which have been used in design work are reviewed and the problems associated with the use of models are discussed. An example to illustrate the role of models in determining the energy implications and airflow patterns of a range of ventilation systems is described.

2. MODELS IN DESIGN

One of the earliest numerical models used in the calculation of air infiltration in residential buildings is described by Gabrielsson and Porra⁷. A "multi-cell" technique was developed in which the building was divided into nodes interconnected by flow paths. The nodes represented individual rooms or regions of differing pressure and the flow paths represented the leakage between them. A series of simultaneous equations describing the flow in each path was developed and solved iteratively by determining an internal distribution such that the volume of incoming air was balanced by the outflow. The model had applications in air movement studies and proved to be a valuable tool in ventilation calculations and design work.

A similar approach was used by Jackman to analyse air movement in tall, naturally ventilated office buildings and to study various ventilation options in a two-storey dwelling^{8,9}. Other models following this approach are described by Bilsborrow¹⁰, Sander¹¹, de Gids¹² and Etheridge¹³.

Because "multi-cell" models can be used to predict internal air movement, they are an invaluable aid to indoor air quality studies. However, they require substantial data to describe the internal flow network and often demand a significant amount of computational effort. In some instances, it is therefore advantageous to reduce the network by assuming that the interior of the building is at a single internal pressure. This generally

enables the flow equations to be solved directly, thereby greatly simplifying the calculations.

This approximation is valid provided there is no significant resistance to internal air movement. The method enables overall air change rates to be determined and hence is useful in calculating ventilation heat loss. Methods based on this approach are described by Larsen¹⁴, Nylund¹⁵, Cole¹⁶ and Handa¹⁷.

A recent advance on the "single-cell" technique is described by Sherman and Grimsrud¹⁸ and Warren and Webb¹⁹. In this development, air infiltration is correlated with the results of building pressurization test data, by linking the leakage characteristics of the flow paths to the pressurization results.

With the majority of models, it is possible to determine the effects on ventilation of purpose provided openings, chimneys and mechanical ventilation systems. Thus a wide range of models are available to assist in all aspects of ventilation design.

3. MODEL VALIDATION

A problem with many air infiltration models is that, because the availability of air infiltration data is limited, the scale of validation has been restricted. Hence the reliability of models in design applications is uncertain. One of the tasks of the Air Infiltration Centre has been to prepare reliable datasets from experimental data covering as wide a range of building design and climatic conditions as possible. This is being used to determine the full range of applicability of models and to identify the key parameters which must be defined for reliable estimates of air infiltration. The outcome of this study will be published shortly but present results are encouraging and show that good estimates of air infiltration are possible, provided that leakage and pressure data are accurately specified.

4. KEY PARAMETERS

4.1 Leakage Characteristics

The flow through an opening is generally approximated by an exponential equation of the form

$$Q = k(\Delta P)^n \quad 1$$

where k and n are constants describing the leakage characteristics of the opening and ΔP is the pressure difference acting across the component.

In existing dwellings, the determination of air leakage characteristics are fairly straightforward and basic data may be

obtained by performing a building pressurization test. However, with a projected building such an exercise is not possible. For models to be successful in this application, it is necessary to specify the fabric or background leakage characteristics of the building at the design stage and to select the appropriate building technique. High quality control during building construction is also essential.

4.2 Pressure Data

The pressure acting on the building is maintained by the action of wind and stack effect. Relative to the static pressure of the free wind, the pressure resulting from wind impinging on the surface of the building is given by

$$P_w = \frac{\rho}{2} C_p v^2 \quad 2$$

where ρ = air density
 C_p = pressure coefficient
 v^2 = wind speed and building height

The pressure coefficient is related to the pattern of flow around the building and is the key parameter to be defined in determining wind pressure. Coefficients based on the results of wind tunnel tests performed on simple building shapes are widely available but may not be applicable to buildings shielded by local obstructions. Much work is still required to develop simple rules to cope with this problem and in the meantime it may be necessary to perform wind tunnel tests on scale models of the building in situ.

The stack effect is described by the equation

$$P_s = 3462 (h-n) \left(\frac{1}{T_{ext}} - \frac{1}{T_{int}} \right) \quad 3$$

where h = level of opening
 n = level of neutral plane
 T_{ext} = external air temperature
 T_{int} = internal temperature

In practice, the level of the neutral plane is rarely known and stack pressure is normally expressed relative to the lowest opening. This calculation should present few problems.

5. EXAMPLE

To illustrate the role of models in ventilation design, an example is described below. In a previous study⁹, a similar model was used to assess the energy implications of providing

adequate natural ventilation in a dwelling, at a design internal/external temperature difference of 4°C and calm wind conditions. The effects on energy consumption of airtightness measures and alternative natural and mechanical ventilation systems were also investigated. In the present example, an existing building is taken as the starting point.

5.1 Description of Dwelling

The floor plan and principal flow network of the dwelling is illustrated in Figure 1. It is a two storey, end terrace house of brick and block cavity wall construction, situated in relatively unobstructed countryside. The occupied volume of this building is approximately 227m^3 . The windows are single glazed and weather-stripped. Good mixing of air between individual rooms was assumed, equivalent to the internal doors being open.

Measurements over a range of pressures have been made to determine the leakage characteristics of the dwelling and of individual components (Figure 2). These results show that the leakage is dominated by unspecified "background" losses, which account for 71% of the total at a pressure difference of 10 Pa reducing to 65% at 50 Pa. The effectiveness of weatherstripping is also clearly illustrated, with the total window leakage being lower than that through either of the doors. By UK standards, the house is particularly airtight and has an air change rate of approximately 9.8 h^{-1} at a pressure difference of 50 Pa. This compares with a mean value for 19 modern dwellings tested by the Building Research Establishment, of 13.9 ach^{20} . The tightness of the building has been confirmed by a number of air infiltration tracer gas measurements which range in value from between 0.21 and 0.51 ach.

The model was used to investigate the variation in hourly air infiltration rate throughout a heating season for the building situated in its present location and for two other classes of terrain. The mean infiltration rate, energy consumption and pattern of air flow was also determined. The model was then used to study the effects on energy consumption and air change rate of alternative ventilation systems and airtightness measures.

5.2 Input Data

The flow parameters used in the model were based on the results of the component and building pressurization tests given in Figure 2. The component leakage characteristics were applied directly and the remaining background leakage was distributed uniformly about the envelope of the building. It was assumed that the background leakage resulted from cracks in joints in the inner leaf of the construction and were therefore shielded from the direct action of wind by the outer wall. Hence only air flow through component cracks was assumed to be influenced by wind pressure.

The climatic data is given in Table 1 and is based on statistical relationships between hourly mean wind and temperature throughout the heating season. The data was obtained from the Meteorological Office and is available for several localities in the UK. The wind data refers to the "meteorological" wind speed measured at a level of 10m above the surface in open country and may be corrected to give the wind speed for other terrains and levels by using the equation

$$\frac{U}{U_{10}} = Kz^a \quad 4$$

where U = wind speed at desired level
(building height)
 U_{10} = meteorological wind speed
 z = building height
and K and a = constants dependent on terrain²¹

5.3 Calculation Method

By using each combination of wind and temperature as input data to the model, it is possible to determine the air infiltration rate for each set of conditions. An alternative method is to calculate infiltration for a representative range of data and to construct a set of infiltration curves as illustrated in Figure 3. The infiltration for any set of climatic data can then simply be determined from the graph. Provided that the wind pressure on the building is not significantly influenced by local obstructions, these infiltration curves will be valid for any climate or terrain. Thus the designer is quickly able to visualise the relative importance of the various parameters influencing air infiltration.

It can be seen from Figure 3 that, in this example, there is some dependancy on wind direction, with a reduction in infiltration rate of approximately 40% for wind along the length of the building. This reduction was verified by measurement and it was estimated to apply to variations in wind direction of no more than $\pm 10^\circ$. Thus, for the purposes of energy calculations and general design studies, the higher rates of infiltration were assumed. However, each design must be considered on its merits and, if necessary, statistical data relating wind direction with wind speed and air temperature should be used.

5.4 Verification

Because air infiltration measurements were available, it was possible to confirm the performance of the model and the validity of the various assumptions. On-site wind and temperature data were used in conjunction with the infiltration curves in Figure 3, to provide estimates of air infiltration. These are compared with

measured values for the corresponding period in Figure 4.

Allowing for possible errors in measurement, the agreement was very good with the predicted and observed values being at least within 25% of each other and in more than half the instances within 10%. This result therefore gave some confidence in the remaining analysis.

Unfortunately, in the majority of design applications, air infiltration data will be unavailable and therefore the need for model validation is reinforced. Only by validating the model against as much test data as possible, will it be possible to rely on the model in design.

6. RESULTS

6.1 Variation in air infiltration rates

The variation in infiltration rates was determined by multiplying the infiltration for each combination of wind and temperature by the duration of occurrence. The results are illustrated in Figure 5 for three classes of terrain. These are

- countryside with scattered wind breaks
- urban
- city

As the terrain provides more shelter, the effect of wind on air infiltration reduces and there is a corresponding reduction in air change rate.

This type of analysis allows the designer to consider the tightness of a dwelling in relation to its environment and is therefore particularly important if designing for natural ventilation. This analysis also shows why it is not possible to correlate the results of building pressurization test data with infiltration, unless there is some knowledge of climate and location.

The existing dwelling is situated away from urban and city environments and, from the results given in Figure 5, has a mean air infiltration rate of 0.52 ach. This is very close to the value of 0.5 ach which is often regarded as a minimum desirable rate of ventilation. For the other terrains, the average air change rate actually falls below this figure. Depending on the weather conditions, the hourly air change rate varies between 0.15 and 1.45 and only exceeds 0.5 ach on 34% of occasions. The problem is therefore one of ensuring an adequate supply and distribution of fresh air at all times, without incurring an energy penalty. This is a problem which is taking on increasing importance as designers strive for energy-efficient buildings.

6.2 Ventilation Strategies

Details of air change rates and estimates of the energy required to maintain an internal temperature of 19°C against the effects of ventilation losses are summarised for the various strategies in Table 2 and Figure 7.

6.2.1 Existing System

The wide variation in infiltration rate of the existing system has already been described. This results in a combination of periods of unacceptably low ventilation and poor air quality, with others of high ventilation and excessive energy wastage. Another problem is the movement of air within the dwelling. The preferred flow pattern is for fresh air to enter the bedrooms and living areas and for stale air to leave via the kitchen and bathroom, where most of the moisture is generated. In this way, problems associated with high levels of internal humidity are minimised.

The model was used to analyse airflow patterns and these results are summarised in Figure 6. Figures 6a - c show that, for wind dominated infiltration, there is little prospect of achieving a consistent airflow pattern, as air movement is dependent on wind direction. The greatest potential for achieving satisfactory air distribution by natural means is by exploiting the stack effect. This is illustrated in Figure 6d for a temperature difference of 20°C and wind speed of 2ms^{-1} . Air movement in both sides of the house is separated and therefore moist air from the kitchen and bathroom is kept apart from the rest of the building. The difficulty is to maintain the dominance of stack action against that of wind. In Figures 6e and f, it is shown that wind begins to take over from stack effect at 5ms^{-1} , and at 10ms^{-1} the temperature effect is negligible.

The estimated annual energy demand to satisfy ventilation heat loss is 10.3 GJ and this was taken as a target value for the alternative ventilation strategies.

6.2.2 Mechanical Exhaust

The performance of two exhaust systems were analysed. The first system represented a purpose provided unit with a 0.5 ach capacity. The second represented the use of extractor fans located in the kitchen and bathroom, having a combined capacity of 0.2 ach. The assumption was made that these systems had no effect on the overall airtightness of the dwelling.

The predicted annual energy demand for the first system was found to be 14.8 GJ, thus representing an increase of 44% in energy consumption over the existing system. The mean air change rate

was 0.75 h^{-1} and varied according to weather conditions between 0.5 and 1.64 ach . The air flow patterns are illustrated in Figures 6g to i, and show that air movement is dominated by the exhaust system throughout the wind range. Apart from the ventilation system itself, the remaining outflow is into the roof space. Hence the system can be used to provide a satisfactory direction of airflow but an energy penalty is incurred.

Approximately two thirds of the outflow is through the exhaust system and therefore there is potential for heat recovery. Such a technique would need to be exploited for this system to be viable in this particular dwelling.

The individual extractor system was found to provide some measure of control over air flow with winds up to 5 ms^{-1} (see Figures 6j to l). The minimum air change rate was found to be 0.3 h^{-1} and an air change rate of 0.5 was exceeded for 67% of the time. The estimated annual energy consumption was 12.1 GJ but may be reduced by intermittent use of the fans. Therefore, this system could provide a satisfactory alternative without a substantial increase in energy consumption.

6.2.3 Existing Dwellings with 0.5 ach Balanced Ventilation

The theory describing the influence of mechanical ventilation systems on overall air change rates is explained with the aid of hydraulic models by Nylund²². This shows that for balanced systems the air change rate is the sum of the natural infiltration and the capacity of the system. When applied to this dwelling, the mean air change rate is effectively doubled from 0.52 to 1.02 h^{-1} . The airflow pattern can be adjusted to suit requirements but energy demand rises to 20.5 GJ. Fortunately, efficient air-to-air heat recovery systems are available with these systems and therefore this energy demand can be reduced. For a 50% efficiency in heat recovery, the energy demand reduces to 15.4 GJ and for a 70% efficiency it is 13.4 GJ. To match the performance of the exhaust system an efficiency of 57% is required (see Figure 7).

6.2.4 Building "Retrofit" plus Mechanical Ventilation

The mechanical systems described above provide adequate ventilation but do not meet the target energy demand of 10.3 GJ. To achieve this, building "retrofit" is required. The assumption was made that the leakage of the dwelling could be reduced to just below a third of its current level to conform to current Swedish airtightness regulations for single family dwellings of 3 ach at 50 Pa. Such a retrofit would necessitate considerable attention being given to background leakage but may be possible using the "House Doctor" approach²³. The energy calculations were repeated and show that for the 0.5 ach exhaust system the demand is 11.5 GJ and for the balanced system it is 15.6 GJ. With

air-to-air heat recovery, the energy demand reduces to 10.5 GJ at 50 % efficiency and 8.5 GJ at 70% efficiency. Thus energy demand has been reduced to below the target figure. Adventitious ventilation and the use of extractor fans provide insufficient air change rates in the "retrofitted" dwelling.

Having used the model to assess the performance of each system, it is necessary to select a cost effective method of achieving adequate ventilation. The results of the model provide sufficient information for this exercise and therefore the model has fulfilled a role in both design and cost benefit analysis.

7. DISCUSSION AND CONCLUSIONS

The example has been used to demonstrate the wide range of applications that models have in the design and analysis of ventilation strategies. They enable the performance of systems to be assessed quickly and cheaply and may be readily used in the design of energy-efficient methods to suit individual requirements.

A wide range of models of varying degrees of complexity are available to meet all design needs. These range in complexity from "single-cell" approaches suitable for the estimation of overall infiltration heat loss, to "multi-cell" methods with applications in air movement and indoor air quality studies.

Model validation is particularly important because this enables the full range of applicability of models to be determined. Work in this area is progressing well and current results are very encouraging. The Air Infiltration Centre anticipate reporting further progress on this work at a later date.

The key parameters which must be specified for accurate results have also been identified. Problems associated with determining the leakage characteristics of flow paths and the pressures acting across openings are beginning to be overcome and this is further assisting in the developing role of mathematical models in design.

As the demand for energy efficiency in buildings increases, so will the need for reliable mathematical models in ventilation design continue to grow. Only by good ventilation design, which fully satisfies the needs of the building's occupants, will it be possible to realise the additional potential energy savings in buildings.

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TABLE 1 Frequency distribution of hourly mean wind and temperature data

Temperature range °C	Wind speed range (ms ⁻¹ at 10m)						
	0.0	0.5 to 1.5	2.0 to 3.1	3.6 to 5.1	5.7 to 8.2	8.4 to 10.8	11.3 to 13.9
- 7.9 to - 6.0	0.1	0.1	0.0	0.0	0.0	0.0	0.0
- 5.9 to - 4.0	0.2	0.2	0.1	0.0	0.0	0.0	0.0
- 3.9 to - 2.0	0.4	0.5	0.3	0.3	0.1	0.0	0.0
- 1.9 to 0.0	0.7	0.9	1.0	1.1	0.5	0.1	0.0
0.1 to 2.0	0.8	1.5	2.0	2.6	1.6	0.2	0.0
2.1 to 4.0	0.7	1.6	2.3	3.6	2.5	0.5	0.1
4.1 to 6.0	0.7	1.5	2.6	4.9	3.6	0.7	0.3
6.1 to 8.0	0.6	1.5	2.6	5.3	4.8	1.1	0.4
8.1 to 10.0	0.4	1.2	2.3	4.7	4.7	1.3	0.3
10.1 to 12.0	0.3	1.0	1.9	3.5	3.4	1.1	0.3
12.1 to 14.0	0.1	0.6	1.4	2.4	2.1	0.6	0.2
14.1 to 16.0	0.1	0.3	0.8	1.7	1.3	0.3	0.1

TABLE 2 Summary of results

Ventilation option	Mean air change rate (h^{-1})	Minimum air change rate (h^{-1})	Maximum air change rate (h^{-1})	Energy demand GJ	% time ≥ 0.3 ach	% time ≥ 0.5 ach
1. Existing dwelling with natural "adventitious" ventilation	0.52	0.15	1.45	10.3	86	34
2. Existing dwelling with exhaust mech. ventilation						
(i) 0.5 ach	0.75	0.5	1.64	14.8	100	100
(ii) 0.2 ach	0.62	0.3	1.50	12.1	100	67
3. Existing dwelling with balanced mech. ventilation (0.5 ach)						
(i) without heat recovery	1.02	0.65	1.95	20.5	100	100
(ii) with 50% heat recovery				15.4		
(iii) with 70% heat recovery				13.4		
*4. Dwelling retro-fitted to Swedish airtightness standards (3 ach at 50 Pa)						
(i) .5ach exhaust	0.56	0.50	1.08	11.5	100	100
(ii) .5ach balanced	0.83	0.55	1.46	15.6	100	100
without heat recovery				10.5		
50% heat recovery				8.5		
70% heat recovery						

* see also Reference⁹

FIGURE 1 Floor plan of dwelling and principal flow paths

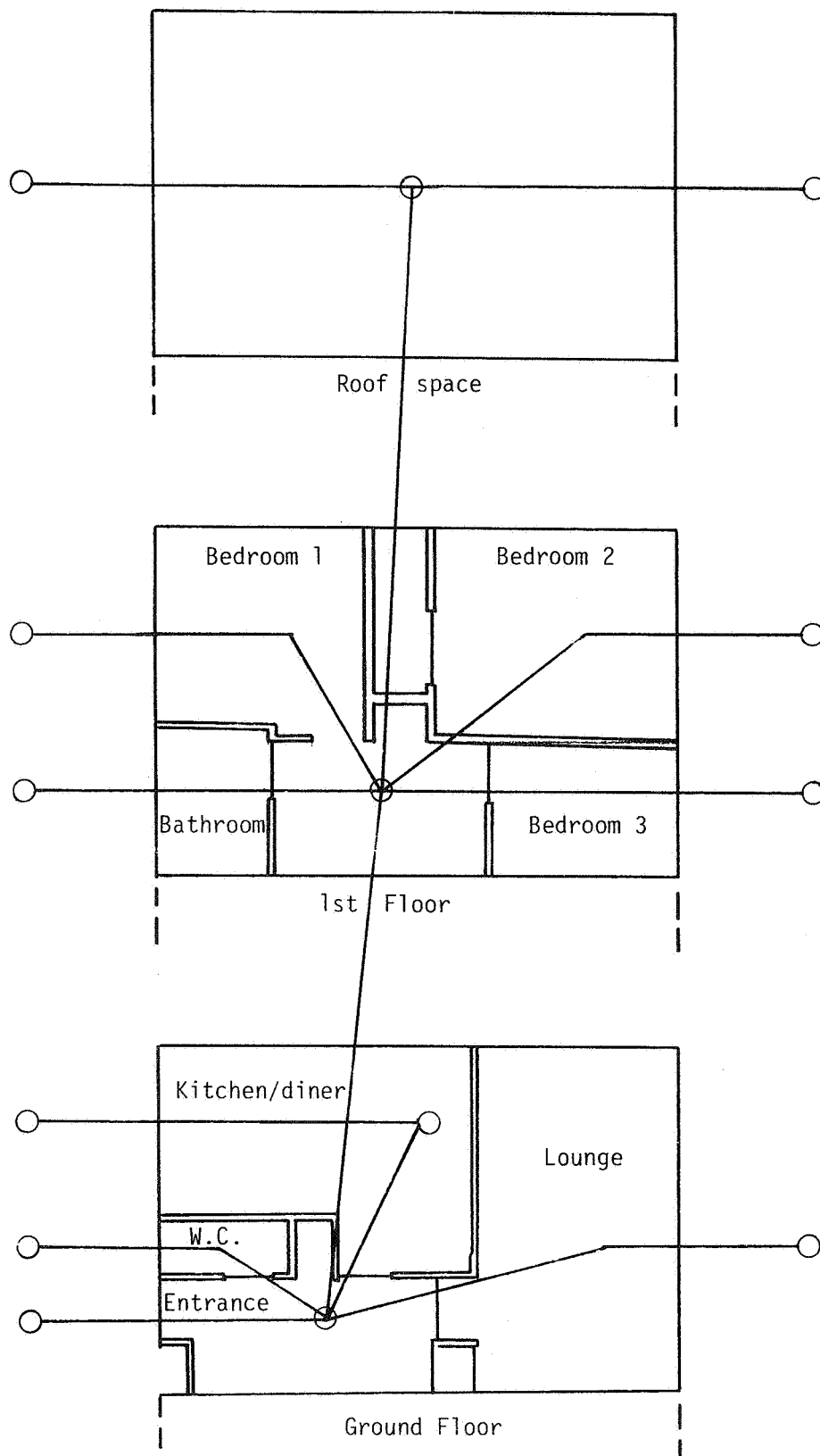


FIGURE 2 Air leakage characteristics of dwelling

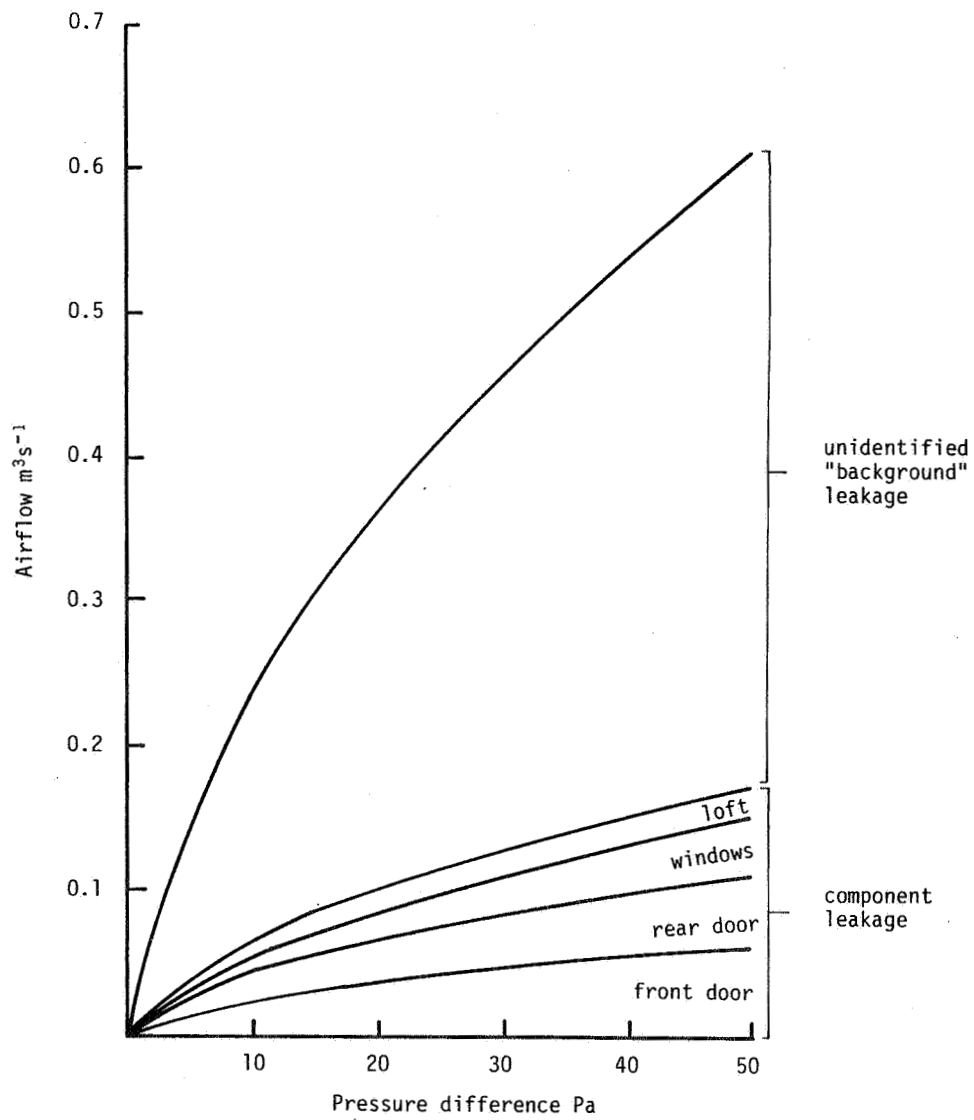


FIGURE 3 Air infiltration characteristics of dwelling

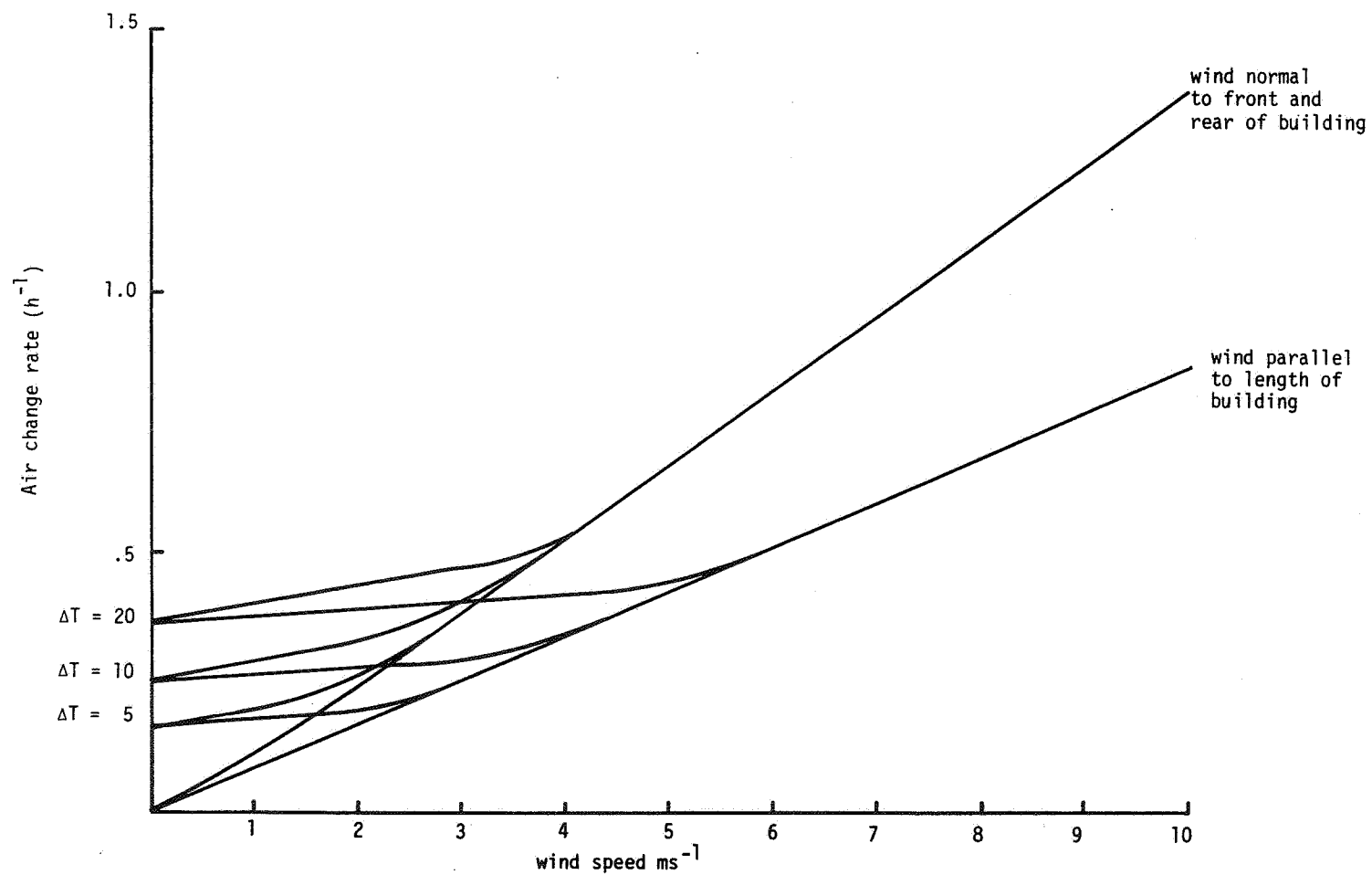


FIGURE 4 Model validation using measured infiltration data

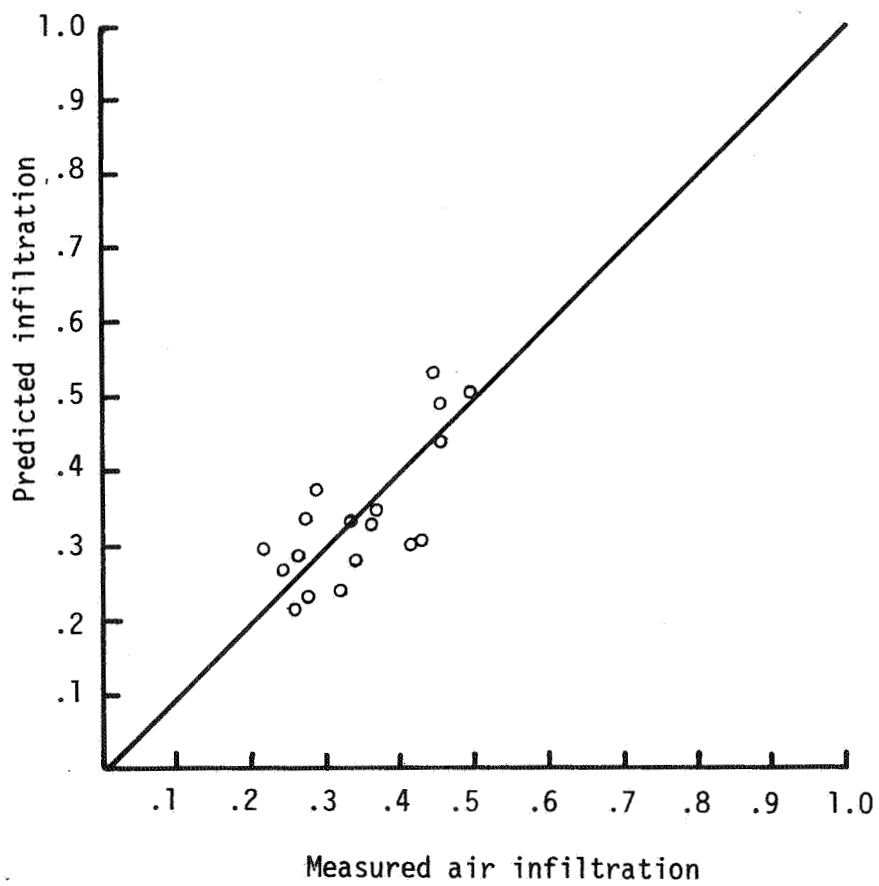


FIGURE 5 Frequency of air infiltration rates
throughout heating season

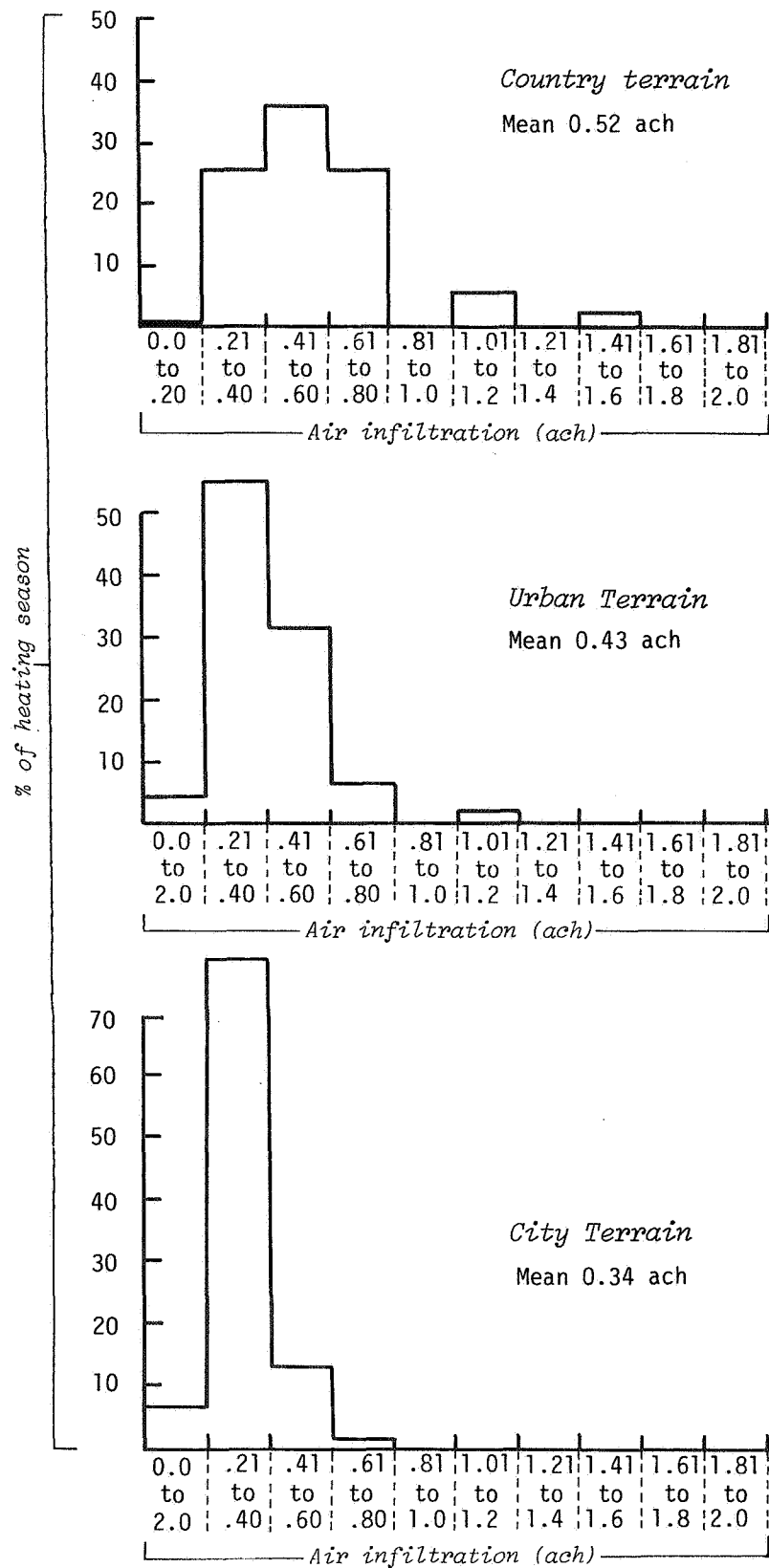
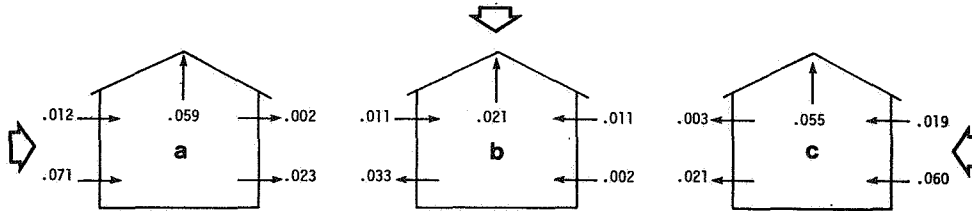


FIGURE 6 Air flow patterns (all flows in m^3s^{-1})

(i) Natural ventilation, wind dominated (10ms^{-1} $\Delta T = 0^\circ\text{C}$)

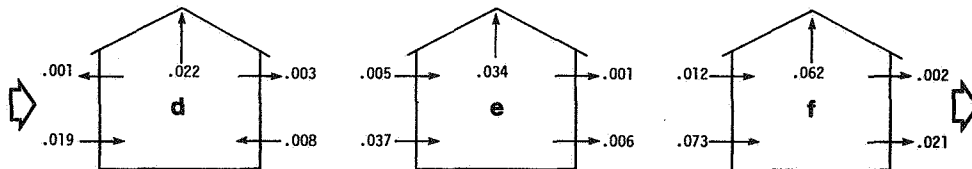


(ii) Natural ventilation, wind and stack

wind 2ms^{-1} $\Delta T = 20^\circ\text{C}$

wind 5ms^{-1} $\Delta T = 20^\circ\text{C}$

wind 10ms^{-1} $\Delta T = 20^\circ\text{C}$

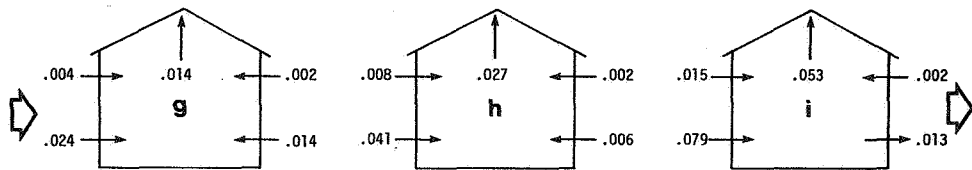


(iii) Mechanical exhaust ventilation 0.5 ach

wind 2ms^{-1} $\Delta T = 20^\circ\text{C}$

wind 5ms^{-1} $\Delta T = 20^\circ\text{C}$

wind 10ms^{-1} $\Delta T = 20^\circ\text{C}$

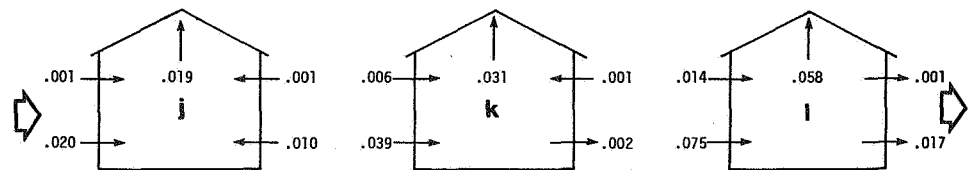


(iv) Mechanical exhaust ventilation 0.2 ach

wind 2ms^{-1} $\Delta T = 20^\circ\text{C}$

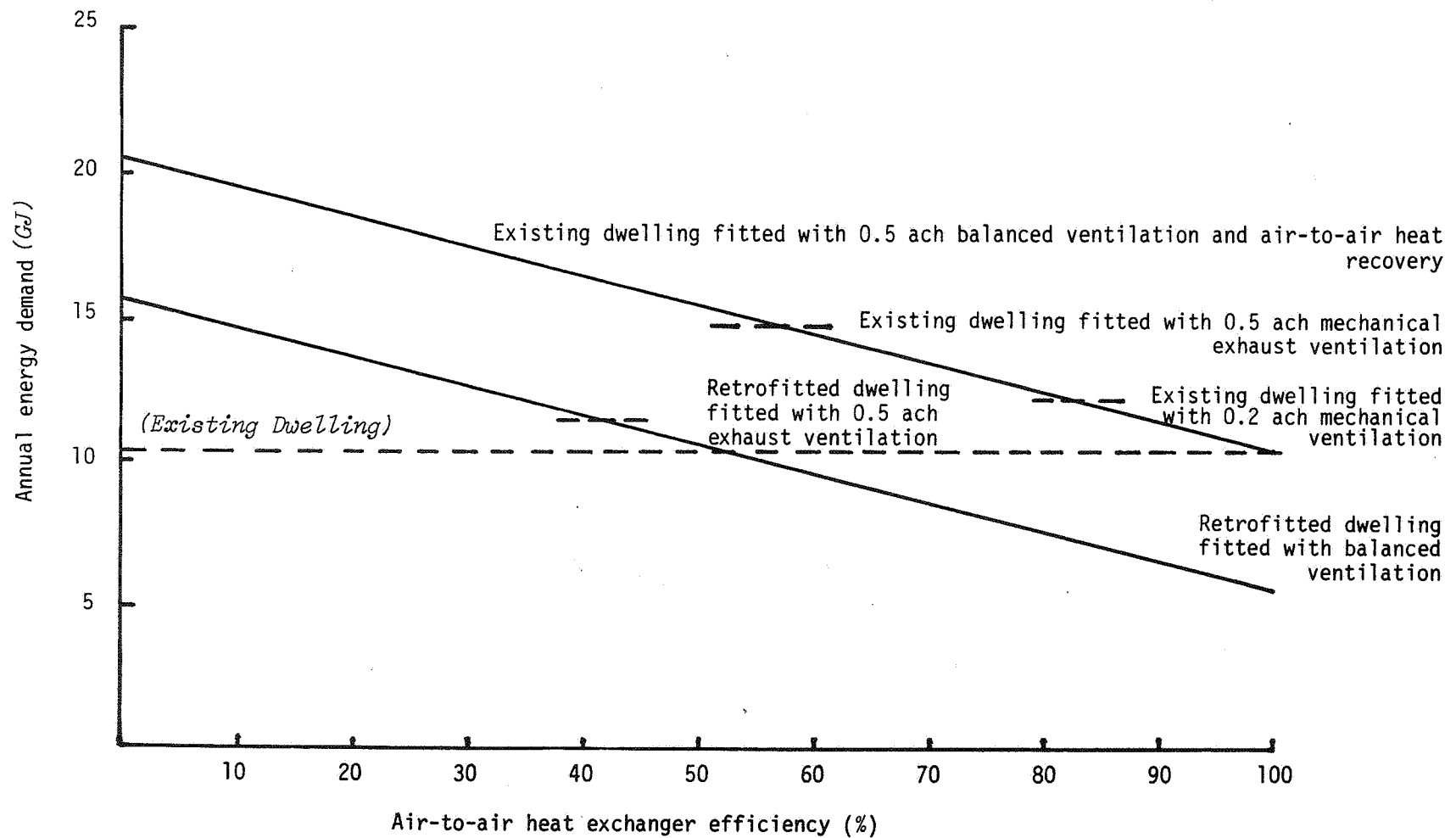
wind 5ms^{-1} $\Delta T = 20^\circ\text{C}$

wind 10ms^{-1} $\Delta T = 20^\circ\text{C}$



Key: wind direction direction of air movement (m^3s^{-1})

FIGURE 7 Energy analysis of ventilation systems



ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 21

ESTIMATION OF RATE OF AIR INFILTRATION BASED ON FULL-
SCALE WIND PRESSURE MEASUREMENTS

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SUMMARY

Natural and forced ventilation are directly and indirectly influenced by the pressure distribution around a building. The paper presents the results of full-scale pressure measurements on a typical Swedish timber house.

The rate of air infiltration has been calculated by employing the values obtained from full-scale pressure distribution, air leakage characteristics and temperature differences. The results are compared with the actual ventilation obtained from tracer gas measurements.

1. INTRODUCTION

Wind and temperature induced pressure differential and air leakage characteristics are some of the essential parameters needed in the estimation of air flow through a building.

The present day approach of employing pressure coefficients or shape factors given in the code of practice on wind loads is not strictly correct for application in the air flow studies. The main objection is that these shape factors are for calculating maximum wind loads based on extreme wind speeds. Their relevance in the design of heating and ventilating systems is doubtful as the pressure distribution at low wind speeds may not be the same as for high winds.

In theory the pressure coefficients are nondimensionalised factors independent of the wind velocity and are derived by assuming that Bernouilli's law for laminar flows holds. In reality the pressure distribution is governed by the wind speed and direction which vary in time and space. The separation of flow from the building alters the flow field and the pressure distribution on the sides and the rear of the building is influenced by the size of the structure and the surrounding environment.

Generally the pressure coefficients are derived from wind tunnel studies and calibration with full-scale measurements is lacking. The lack of full-scale data on the pressure distribution and the air tightness characteristics of a building poses one of the major problems in assigning numerical values to various parameters needed in the theoretical models.

In order to fill this gap, full-scale wind pressure measurements have been carried out on typical Swedish detached houses located in different types of surface roughness categories. Results of measurements on two test houses have previously been published in references 1 and 2. This paper presents some of the latest series of pressure measurements on the 3rd test house located in a mixed type of terrain.

2. DESCRIPTION OF THE TEST HOUSE

The test house is a 1½-storey prefabricated timber house and is designed according to Swedish standard SBN 80. Ventilation is introduced by means of a mechanical exhaust system complemented with fresh air supply intakes located in the windows.

The structure is located in an area with a few scattered houses on one side and low hills on the other. It is representative of "open" and "semi-urban" site conditions. Figures 1 and 2 show the test house together with the surroundings.



Figure 1 The test house

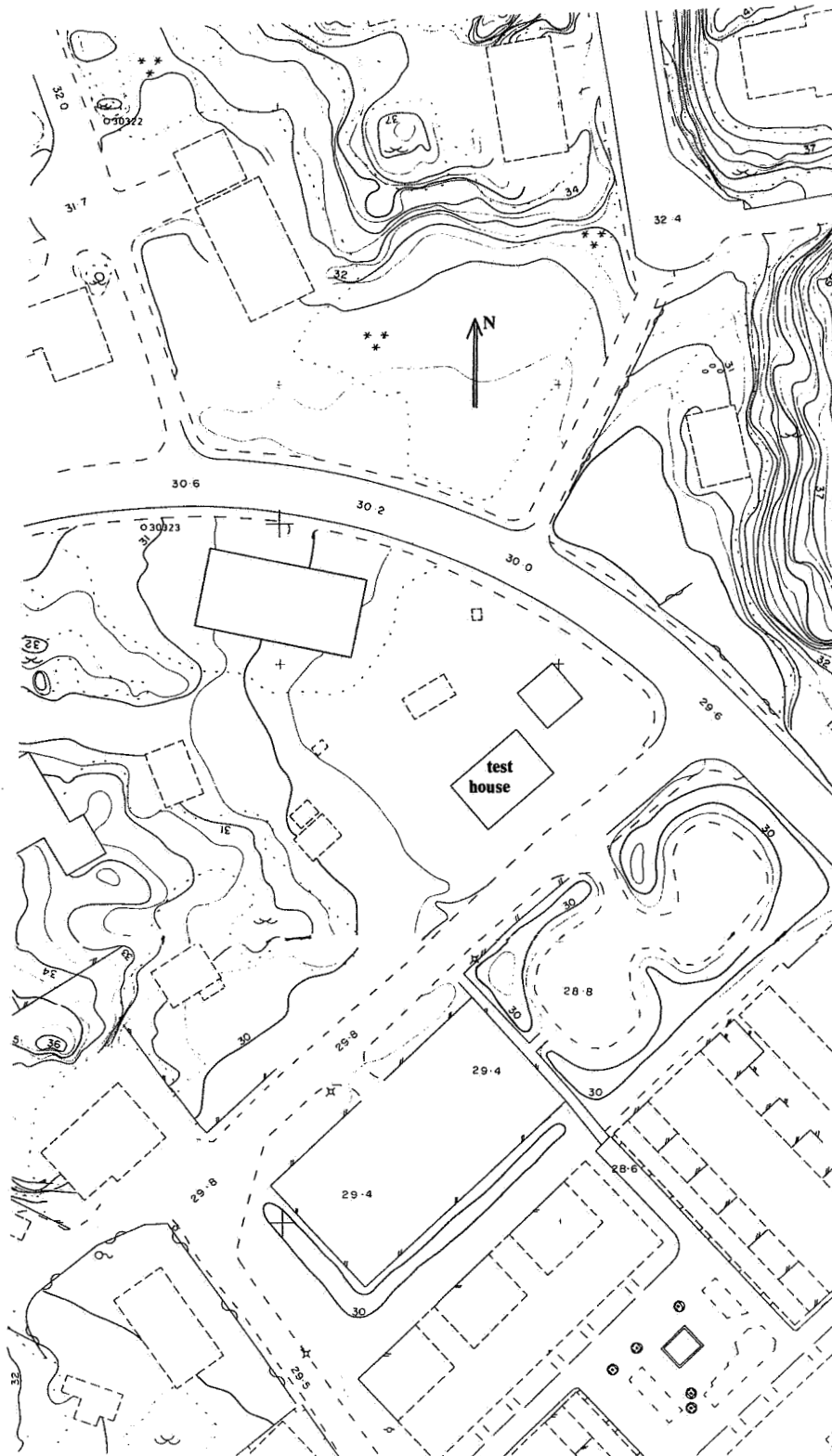


Figure 2 The surroundings around the test house

3. DESCRIPTION OF INSTRUMENTATION

The pressure measurements have been conducted with the help of a open type fluid manometer, which directly registers the difference between external and internal pressures. Plastic tubes, of same lengths, are used in connecting the manometer to the holes in the wooden panels, which are mounted flush with external walls.

The pressure distribution on the structure is registered by photographing the multimanometer every 10th second. Each series of measurements contains approximately 30 photographs. Data is obtained by scanning these photographs with the help of an electronic pen and storing it on a computer for further analysis. The advantage of this system is that the pressure distribution at all the points on the house can be obtained simultaneously. The walls and the roof of the house are divided into 100 segments as shown in figure 3 and the pressure is measured approximately at the centre of each element.

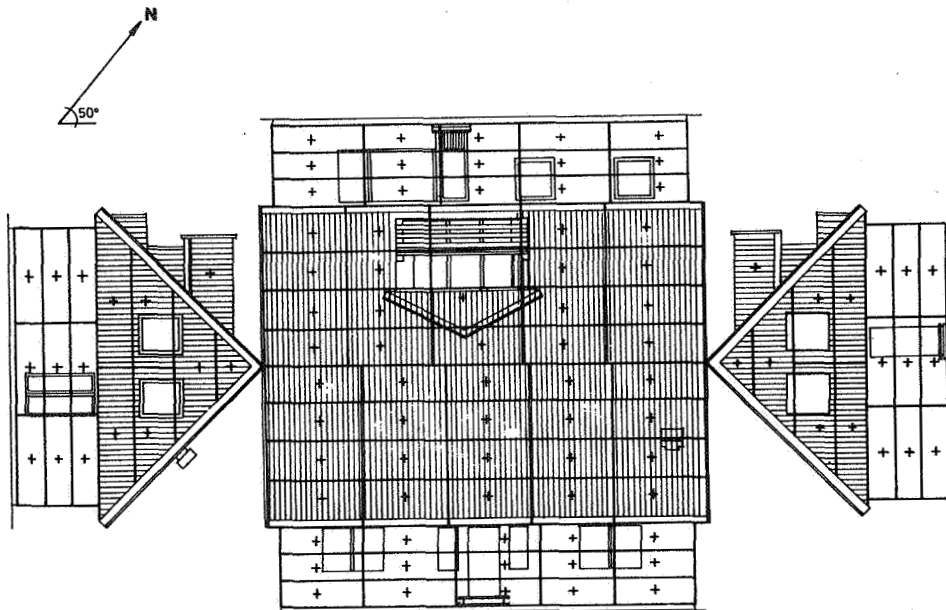


Figure 3 Elements and measuring points

4. EVALUATION OF PRESSURE COEFFICIENTS

The pressure coefficients are obtained for each segment by relating the measured values to the dynamic pressure at the roof level and are defined as

$$C_p = \frac{(p_{\text{ext}} - p_{\text{int}}) - (p_{\text{st,ref}} - p_{\text{int}})}{\frac{1}{2} \cdot \rho \cdot V^2} \quad (1)$$

where

C_p = pressure coefficient
 p_{ext} = external pressure
 p_{int} = internal pressure
 $p_{\text{st,ref}}$ = reference static pressure
 V = free stream wind velocity
 ρ = Air density

The pressure coefficients recommended in the Swedish Standards, SBN 80 are shown in figure 4. Figure 5 gives an example of the mean pressure distribution on the test house obtained by averaging over 27 recordings. It can be noted that there is a considerable difference between the measured and the recommended values.

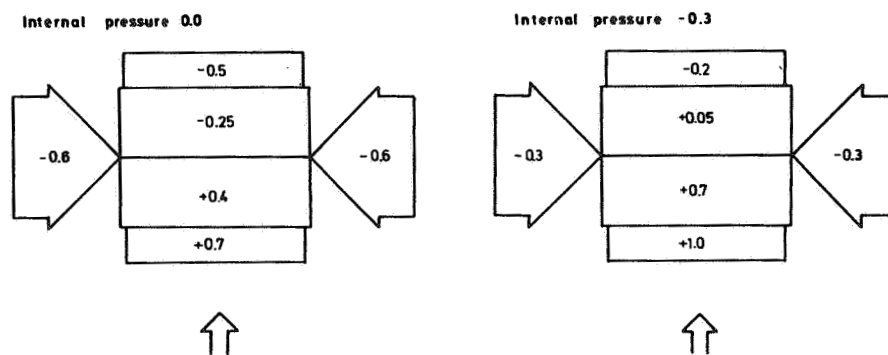


Figure 4 Pressure coefficients from the code of practice, SBN 80

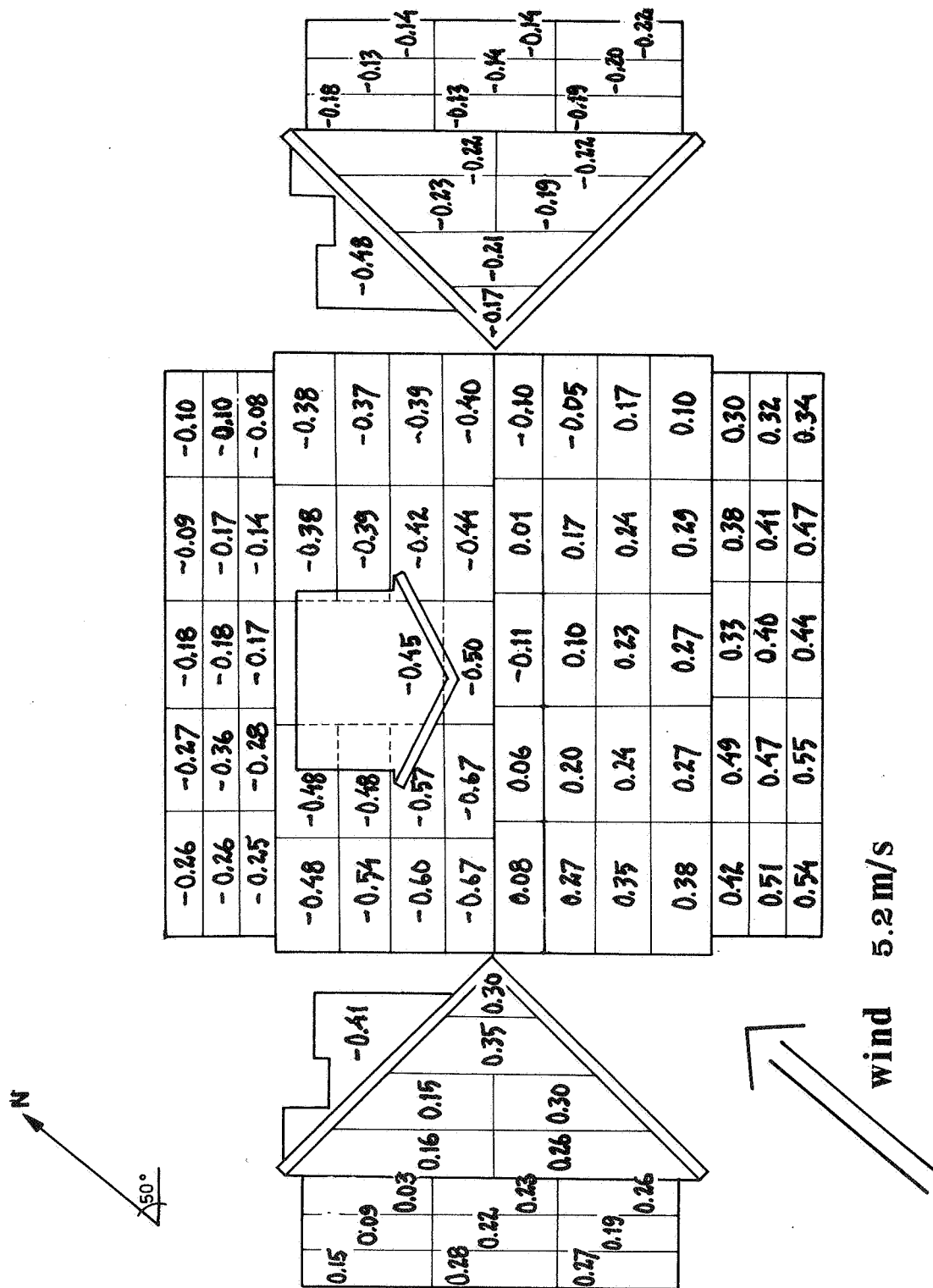


Figure 5 Pressure coefficients based on full-scale measurements

5. CALCULATION OF AIR INFILTRATION THROUGH THE BUILDING

In this section calculations of air infiltration through the test house are carried out by employing the measured pressures or pressure coefficients. A simple model describing the pressure distribution ΔP_i due to wind and temperatures over an element i can be written as:

$$\Delta P_i = P_{wi} - \rho \cdot g \cdot 273 \cdot \left(\frac{1}{T_o} - \frac{1}{T_i} \right) \cdot Z_i \quad (2)$$

wind temperature effects

where

ΔP_i = total pressure distribution over element i
(Pa)

ρ = air density (kg/m^3)

g = acceleration (m/s^2)

T_o = outside temperature (K)

T_i = inside temperature (K)

Z_i = height of element i from a reference level
(m)

Expression (2) can also be written in terms of the pressure coefficients C_{pi} and the free stream velocity V at the top of the structure as:

$$\Delta P_i = \frac{1}{2} \cdot \rho \cdot V^2 \cdot C_{pi} - \rho \cdot g \cdot 273 \cdot \left(\frac{1}{T_o} - \frac{1}{T_i} \right) \cdot Z_i + \Delta P_r \quad (3)$$

ΔP_r is difference between the static reference and the internal pressure (Pa).

Air flow through an area A_i of element i can approximately be written as

$$q_i = A_i \cdot K_i \cdot \Delta P_i \cdot |\Delta P_i|^{\beta-1} \quad (4)$$

Where K_i represents the discharge factor per unit area for element i . The units for various quantities in expression are q_i (m^3/s), A_i (m^2), K_i (m/s Pa^β) and ΔP_i (Pa). β varies between 0.5 (turbulent flow) to 1.0 (laminar flow) depending on the flow characteristics.

The total infiltration through the house is obtained by adding up positive and negative components of the pressure distribution separately together with forced airflow due to mechanical ventilation.

The total inflow, q_T and the total outflow, q_F from the building can be written as

$$\begin{aligned} q_T &= q_{TM} + \sum_{i=1}^N q_i \quad \text{for positive } \Delta P_i \\ q_F &= q_{FM} + \sum_{i=1}^M q_i \quad \text{for negative } \Delta P_i \end{aligned} \quad (5)$$

where

q_{TM} = forced inflow due to mechanical ventilation
 q_{FM} = forced outflow due to mechanical ventilation
 N = total number of segments subjected to positive pressures
 M = total number of segments subjected to negative pressures.

As the in and outflow must be balanced, sometimes it may become essential to introduce a correction term as given below in equation (6). Need for this parameter arises from the fact that the actual permeability characteristics and the reference static pressure cannot be measured accurately.

$$q_T \cdot T_i = q_F \cdot T_u + \varepsilon \quad (6)$$

6. CALCULATION OF DISCHARGE FACTOR

The discharge factor K_i in equation (4) represents the permeability characteristics of element i . In reality, it is not possible to measure the leakage properties of every segment and therefore a total discharge factor, K_B is established for the whole area, A_B of the structure. The standard procedure for establishing the value of K is to measure the rate of air flow through the building at a specified pressure. The expression for the evaluation of factor K is then given by

$$\mu = (\sum K_i \cdot A_i) \cdot \Delta^\beta = K_B \cdot A_B \cdot \Delta^\beta \quad (7)$$

where μ is the rate of air flow and Δ is the applied pressure of 50 Pa. The effect of wind and temperature is neglected compared with the forced pressure difference.

Figure 6 shows a relationship between the rate of air flow μ and the forced pressure difference Δ . The two curves indicate the influence of open and closed ventilation system on the air flow throughout the house. For low-rise buildings, the Swedish code prescribes that maximum rate of infiltration should be three changes per hour at a mean pressure of 50 Pa.

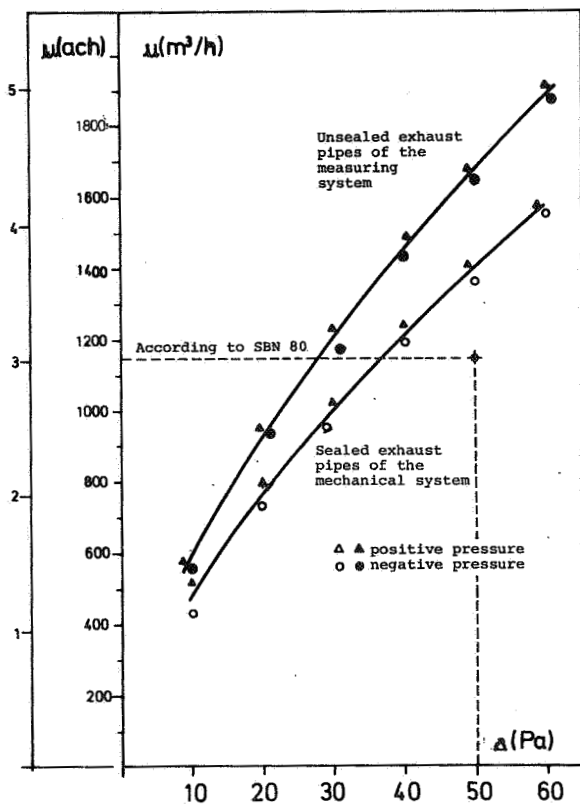


Figure 6 Pressurization tests

It should be observed that in these measurements, the mechanical exhaust system is not operating and that the pressurization tests have been performed by considering both the sealed and unsealed exhaust pipes of the mechanical system as shown in figure 6.

7. RESULTS AND DISCUSSIONS

The pressure distribution and the amplitude and direction of wind velocity are measured under different operating conditions of the mechanical exhaust system and the natural ventilation. The rate of air flow is also measured with the help of tracer gas during the same period.

Figure 7 shows the influence of mechanical exhaust ventilator on the pressure distribution. The full lines in the figure denote the pressure for non-operating ventilator, while the dashed lines indicate the case of normally operating ventilator. It is clear from the figure that the mechanical ventilation system hardly influences the pressure distribution around the house. This may be attributed to the fact that the mechanical exhaust system is working under the designed capacity of 0.5 changes per hour, which does not fulfil the requirements specified by the Swedish standards. The actual exhaust air, measured with a hot-wire anemometer, is approximately 0.3 changes per hour.

Figure 8 illustrates the influence of mechanical and natural ventilation on the rate of air flow measured by tracer gas and calculated air change rate based on the pressure distribution on the walls and the roof. Four different cases of natural and forced ventilation are studied as shown in table 1.

Table 1 Different combinations of natural and mechanical ventilation studied during the measurements

Case NO	Position of mechanical ventilator	Position of openings under the windows
1	closed and sealed	closed
2	closed and sealed	open
3	open	closed
4	open	open

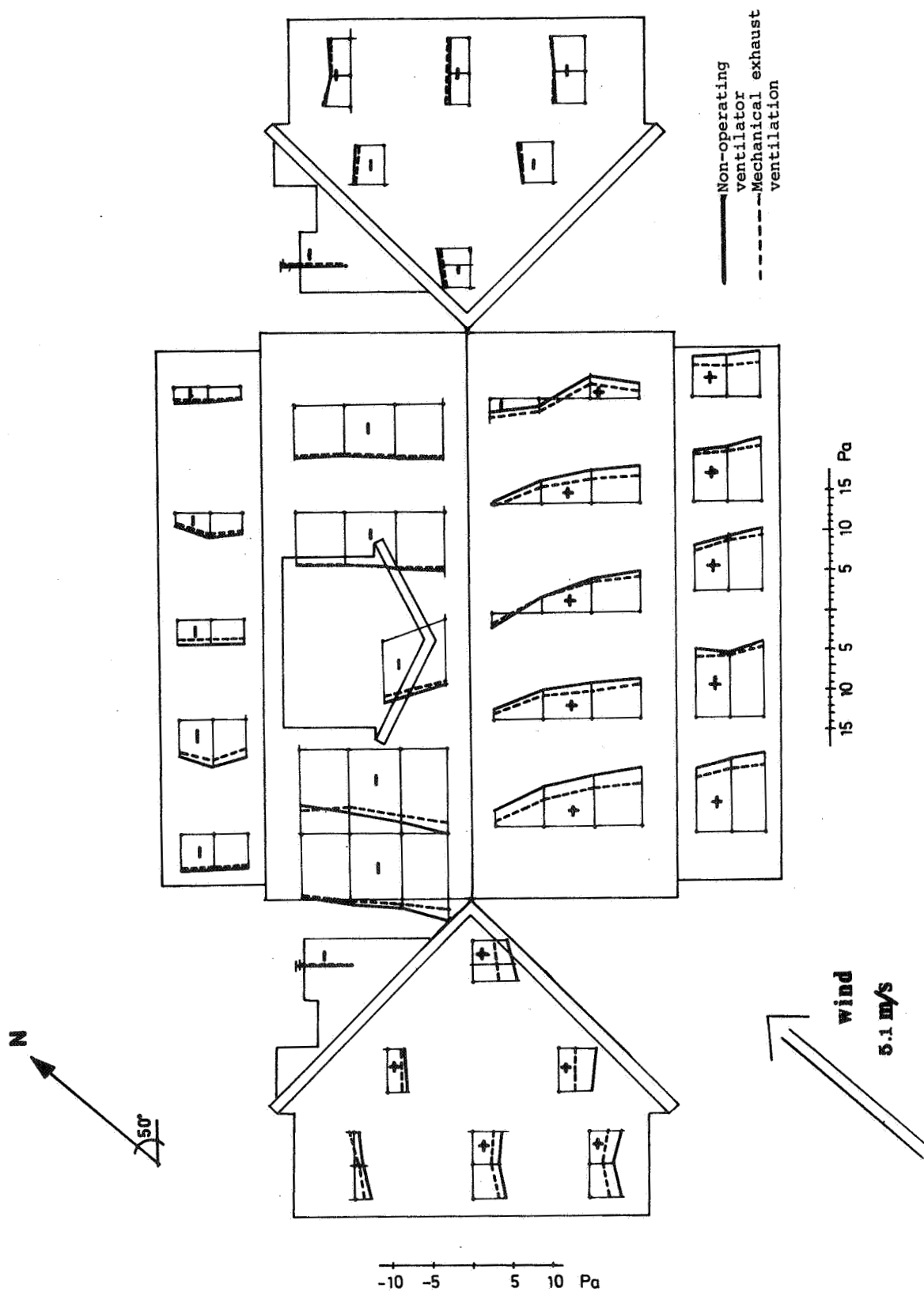


Figure 7 Pressure distribution on the walls and the roof

Examination of figure 8 shows that the rate of air flow measured by the tracer gas method is less than that obtained by the use of expressions (4) and (5). The difference in the two types of measurements indicates that the model for calculating the air flow based on pressure measurements is not adequate. It seems that the calculation model in its present form fails to reflect the behaviour of the system and needs some further modification. Of course, this tentative statement is true only if the tracer gas measurements are carried out in a systematic and professional manner.

The calculations based on the application of the code of practice are far in excess of those given by tracer gas and the wind pressure method. This is, as earlier mentioned, due to the fact that pressure coefficients from the code are really meant for calculation of wind loads and not for calculating the rate of air infiltration.

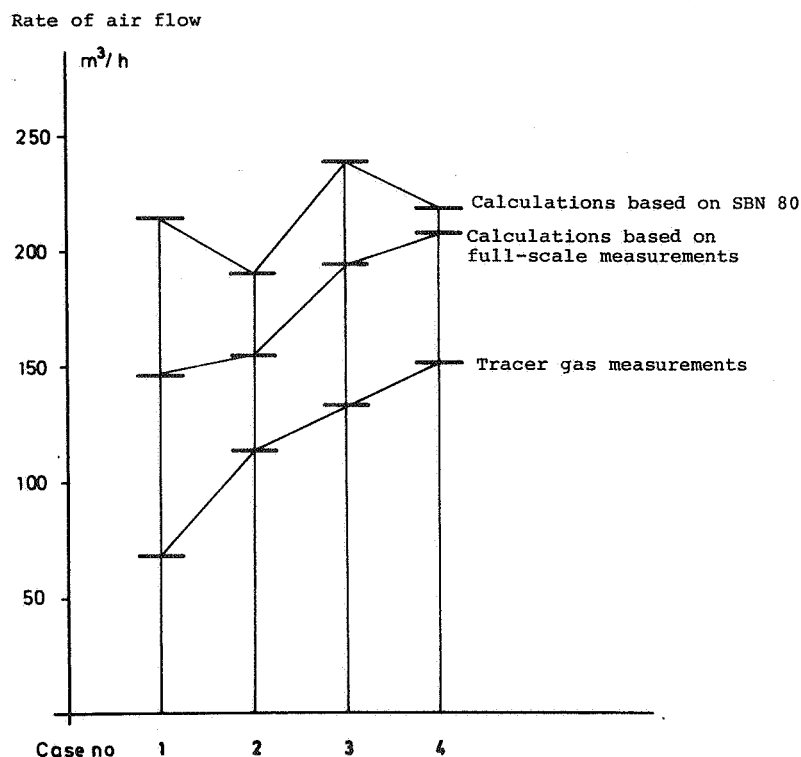


Figure 8 Measured and calculated rate of air flow

It can be of interest to study a relationship between the air flow and the actual wind velocity. This has been done by plotting a nondimensionalised factor γ versus wind velocity V in figure 9. γ is the ratio of the calculated flow Q derived from pressure distribution and air flow μ at 50 Pa based on pressurization test. Results are shown for the three houses situated in different types of surface roughness conditions. It can be noted that the values of γ are related to V in a rather distinct manner. It seems that it may be possible to estimate Q for a given velocity if an empirical relationship between γ and V can be established. This entails further full-scale measurements over a wide range of wind velocities.

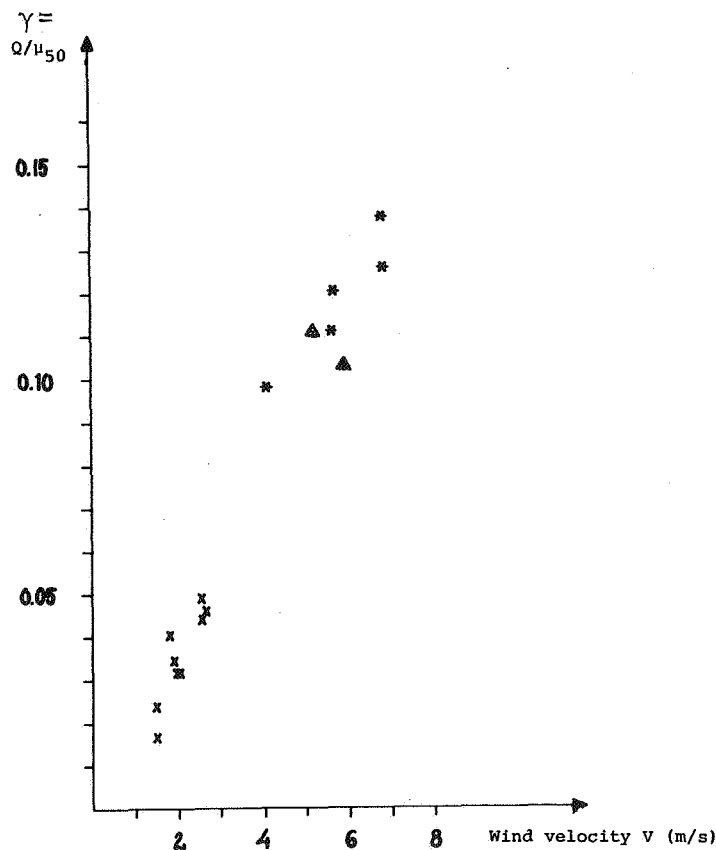


Figure 9 Relationship between ventilation losses and wind velocity

The study has shown the inadequacy of the existing calculating model when used in conjunction with full-scale measured data. It seems that further efforts are needed to correlate the experimental and theoretical models.

8. ACKNOWLEDGEMENTS

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 22

HEAT LOSSES FROM SMALL HOUSES DUE TO WIND INFLUENCE.

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SYNOPSIS

The heat losses from small houses, due to transmission and ventilation, are estimated. The estimation is based up on the house owners daily readings of electricity and water meters, and their notes on behaviour influencing the energy use. Consideration is taken to heat supply from insolation and from people. Hot water losses are calculated from use of water and use of household machinery. Besides the estimation of the heat losses, Q , wind and temperature in the area is registered. The three parameters, local wind velocity, v_l , temperature difference between inside and outside, $\Delta\theta$, and specific heat loss, $Q/\Delta\theta$, are adapted to following relation

$$Q/\Delta\theta = A + B \cdot \Delta\theta^{1/2} + C \cdot v_l + D \cdot v_l^{4/3} \quad (1)$$

According to this model, the heat losses consist of transmission heat loss, wind and temperature influenced ventilation (through ducts, window and doors) and wind and temperature influenced infiltration (through cracks).

When the constants A, B, C and D have been solved for the different houses, the relation (1) is used for estimating possible energy saving by windreduction. It is found that a windreduction to 0,5 - 2,0 m/s will give an energy saving between 3,3 and 9,4 % during the year. Some of this saving may also be achieved by tightening the houses.

SYMBOLS

v	= wind velocity, m/s
$\Delta\theta$	= temperature difference, K
U	= coefficient of thermal transmittance, W/m^2K
h	= surface transfer coefficient, W/m^2K
δ	= thickness, m
λ	= coefficient of thermal conductivity, W/m^2K
V	= air flow, m^3/s
Δp	= pressure difference, Pa
Q	= energy per day, Wh/day
A	= area, m^2
q	= gravitation constant, m/s^2
z	= distance, m
$\Delta\rho$	= density difference, kg/m^3
qs	= energy ratio
τ	= transmission factor
ψ	= shadowing factor
ϕ	= reduction factor
ρ	= density, kg/m^3
c	= heat capacity, $Ws/kg, K$
t^p	= time, h

Index

c	= outside
i	= inside
c	= convection
t	= transmission
l	= local
v	= ventilation
w	= wind
inf	= infiltration
V	= vertical
H	= horizontal
s	= solar

HEAT LOSSES FROM SMALL HOUSES DUE TO WIND INFLUENCE

A lot of research work has been carried out, concerning the wind effects on air infiltration into buildings. The air infiltration is equivalent to the ventilation rate in natural ventilated houses and has usually been stated as the number of air-changes per hour. A common way to measure this ventilation rate has been to use tracergas techniques.

An assumed relation between wind velocity and ventilation rate is that the ventilation rate is linear to the wind velocity. So did Bahnfleth¹ among others adapt his measuring data to the relation

$$q = A_1 \cdot v + B_1 \cdot \Delta\theta \quad (1)$$

But other relations are also assumed. Peterson² for instance used the relation

$$q = A_2 \cdot v^{0,8} + B_2 \cdot \Delta\theta \quad (2)$$

In both these assumptions consideration has been taken to the thermal draft, caused by the density difference between outside and inside air. In the two examples above, the wind and temperature induced ventilation rates are added up. An addition of the wind and temperature induced pressures is also often done.

Investigations of the wind influence on total heat loss of the buildings are more rare. A work of Frank³ is, however, dealing with the relation between wind velocity and energy consumption for a number of buildings. Frank used old measurement datas on the weekly consumption of energy and mean wind velocity during the actual-week. The results he found were of the type shown in figure 1.

One of the buildings had an energy consumption that increased with approximately 90 % as the wind velocity increased from calm condition to the mean wind velocity during the measuring period, 3,2 m/s. The other buildings got procentual increases of the same order, between 38 - 95 %. In Frank's investigation, that dealt with multifloor buildings, no consideration was taken to the thermal draft.

Used energy

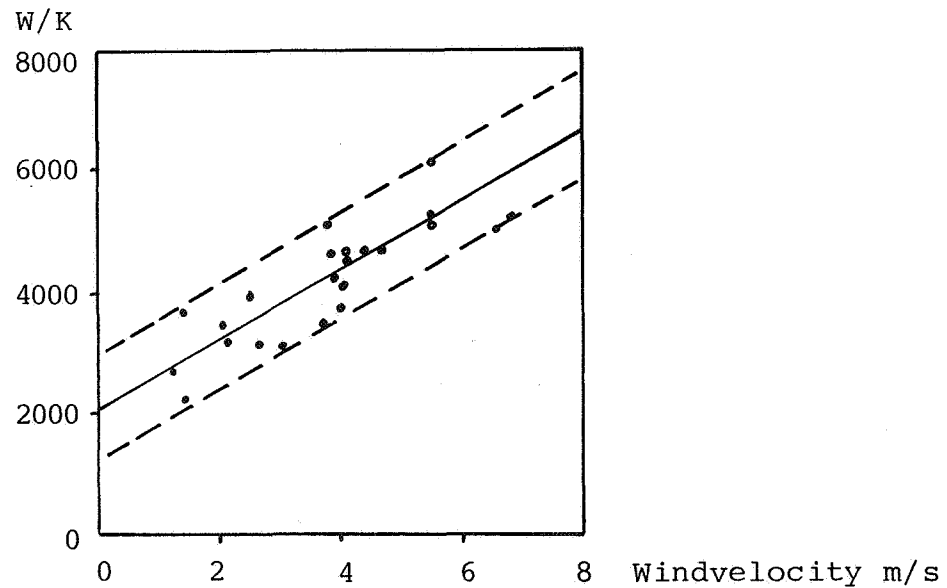


Figure 1.

1. Wind influence

The wind influences the heat losses from a building not only by increased air-infiltration. One can briefly divide the wind influence into four parts,

influence on the U-values

influence on the air-infiltration

influence on the ventilation

influence on the mechanical ventilation

1.1 Influence on the U-values

The transmission heat losses through a building's walls and windows depend on the U-values of the building's different construction materials. The U-value of a building material can be expressed as

$$\frac{1}{U} = \frac{1}{h_o} + \sum_k \frac{\delta_k}{\lambda_k} + \frac{1}{h_i} \quad (3)$$

where h_o and h_i are the transfer coefficients for the outer and inner surfaces respectively

δ_k is the thickness of k:th material layer with the conductivity λ_k

The transfer coefficients consist of a radiation and a convection part. The convection part of h_o can be approximated as

$$h_{o,c} = 10 \cdot v^{0,6} \quad (4)$$

where v is the air velocity along the surface.

A rised wind velocity will thus increase the convective part of the outer transfer coefficient. The heat loss from the building will then get higher. This is, however, only true if the surface temperature is higher than that of the air. During clear nights well insulated building parts may have a surface temperature that's lower than the outdoor airtemperature. In this case, the heat losses will decrease with rising wind velocity. In figure 2 calculated percentage changes in the heat loss from windows and walls are presented. The figure is valid for windows and walls with the nominal U-values 2,0 and 0,25 W/m²K respectively. The different curves are for two cloudiness-conditions.

1.2 Influence on the air-infiltration

The wind induced pressures on a building's surfaces are, together with the temperature dependent pressures, the main forces governing the natural ventilation of a building. A higher wind velocity will increase the over- and underpressures on the building, and with that the ventilation rate. In a building with mechanical ventilation system, the wind induced natural ventilation will intend to give a ventilation rate that diverges from the desired. The wind influence on the air infiltration depends on both the induced pressure, the number of openings and cracks in the building, their distribution and their flow characteristics. The pressures induced are highly dependent on wind direction, surrounding terrain and buildings and the turbulence of the wind.

Air flow through an opening will follow the relation

$$V = k_1 \cdot \Delta p^n \quad (5)$$

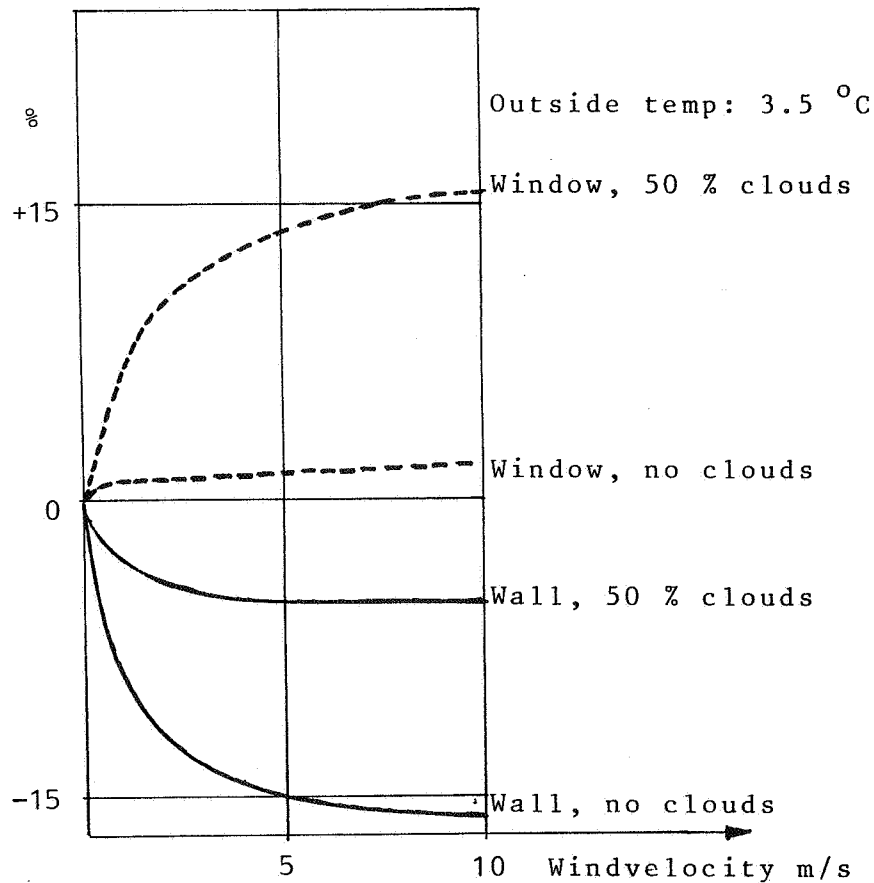


Figure 2.

where k_1 is a constant depending on the size of the opening

n is an exponent depending both on the shape of the opening and the acting pressure Δp

Since the acting pressure, Δp , is proportional to the dynamic pressure of the free wind, $\frac{1}{2} \rho v^2$, the air flow through the openings in the building can be expressed as

$$V = k_2 \cdot v^{2 \cdot n} \quad (6)$$

where k_2 is a constant depending on both the constant

k_1 and the transfer between dynamic pressure and acting pressures on the building.

1.3 Influence on the insulation

Insulation in walls and roof has to be well installed. Otherwise there may be a possible air-movement through the insulation space. This air movement will increase the heat transmission through the wall or roof. Nominal values on insulation materials conductivity are determined in laboratory tests, where the air movement through the insulation is minimized. Air movement, through the insulation space, can be caused either by the temperature difference between the insulations both sides or by wind. To reduce the wind influence it's important to have a windtight layer on the outside of the insulation.

1.4 Influence on the mechanical ventilation

The pressures on the building's surfaces, induced by wind, may also affect the mechanical ventilation. With an exhaust-air system, the exhaust air-flow is influenced by the outside pressure at the place of the air-intake. Supply and exhaust-air systems are disturbed if the inlet and outlet openings are placed in zones with different outside pressure.

2. A model for heat losses from a small house

A model for the heat losses has to contain the two parts governing the heat loss, i.e. transmission and ventilation. In this model, the ventilation part has been divided into two parts. One called ventilation, with which is meant the air-flow through ventilation ducts, windows, doors and other bigger openings. The other called infiltration and consisting of the air-flow through cracks.

2.1 Transmission

If the insulation is assumed to be unaffected by wind, and the outer transfer coefficients to have a neglectable importance, the transmission heat loss from the building can be expressed as

$$Q_t = \sum_k U_k \cdot A_k \cdot \Delta \theta \quad (7)$$

where U_k is the U-value of the k:th building part with the area A_k

Since U_k and A_k can be considered as constant, the transmission heat loss, Q_t , can be expressed as

$$Q_t = A \cdot \Delta\theta \quad (8)$$

where A is a constant.

2.2 Ventilation

When there's an air flow through ducts and similar, the exponent n in equation (6) can be taken as $1/2$. The pressure Δp , causing the air-flow, may be caused by either temperature difference between inside and outside, $\Delta\theta$, or wind-velocity in the vicinity of the building, v_l . The temperature difference gives cause to a pressure difference between inside and outside that may be written as

$$\Delta p = g \cdot z \cdot \Delta\rho \quad (9)$$

Where g is the gravitation constant

z is the distance from the neutral level (the level where inside and outside pressures are equal, see figure 3)

$\Delta\rho$ is the density difference between outdoor and indoor air

Since the density difference $\Delta\rho$ is approximately proportional to the temperature difference, $\Delta\theta$, the ventilation air flow will be proportional to $\Delta\theta^{1/2}$. Thus the ventilation heat loss due to thermal forces can be expressed as

$$Q_{v,\theta} = B \cdot \Delta\theta^{1/2} \cdot \Delta\theta = B \cdot \Delta\theta^{3/2} \quad (10)$$

where B is a constant.

The ventilation air-flow caused by wind may be expressed in accordance with equation (6) with the exponent

$n = 1/2$. i.e.

$$V = k_2 \cdot v_L^{2 \cdot 1/2} = k_2 \cdot v_L \quad (11)$$

The ventilation heat loss due to wind influence can then be expressed as

$$Q_{v,w} = C \cdot v_L \cdot \Delta\theta \quad (12)$$

where C is a constant.

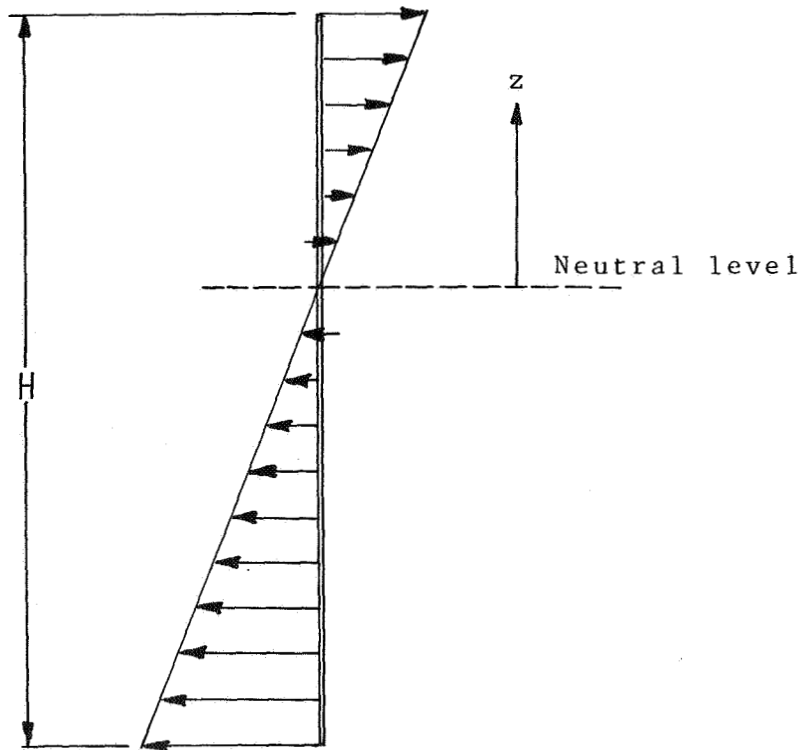


Figure 3.

2.3 Infiltration

The variation in the temperature-difference between inside and outside, $\Delta\theta$ is relatively small between different days. The exponent n , in equation (5), is therefore of less importance. The temperature-dependent part of the infiltration through cracks can be included in equation (10). The wind influenced infiltration flow will, in accordance with equation (4) and with the exponent $n = 2/3$, be

$$V = k_2 \cdot v_L^{2 \cdot 2/3} = k_2 \cdot v_L^{4/3} \quad (13)$$

The exponent 2/3 is a good approximation for characterizing the air flow through a building's all cracks, see Peterson⁴. The heat loss due to infiltration will now be

$$Q_{inf,w} = D \cdot v_L^{4/3} \cdot \Delta\theta \quad (14)$$

where D is a constant.

Summing up the four parts above will give the buildings total heat loss, due to transmission and ventilation, as

$$Q = Q_t + Q_{v,\theta} + Q_{v,w} + Q_{inf,w} = A \cdot \Delta\theta + B \cdot \Delta\theta^{3/2} + C \cdot v_L \cdot \Delta\theta + D \cdot v_L^{4/3} \cdot \Delta\theta \quad (15)$$

3. Measuring method

The heat loss due to transmission and ventilation, Q , in relation to wind velocity, v_L , and temperature difference, $\Delta\theta$, has been investigated. The investigation was made in an area of 79 small houses, in the south of Sweden. Twenty electrically heated houses, of two different types, were involved, ten of each type. The houses, which had two floors and a living area of 150 m² and 190 m² respectively, were spread on a hill, unprotected from the surrounding flat country in W-, SW and S-directions, see figure 4.

The families in the houses made daily notes on their total use of electricity and water, use of household machinery (such as washing machines) and other things important for the energy use. Figure 5 shows one family's notes for three days. (The total time of the investigation was between 23 - 27 days, depending on house). Out of the notes, a simple estimation of the houses heat loss, due to transmission and ventilation, was made.

3.1 Heat supply

Three sources contribute to the heat supply of a building. Besides the input of electricity (where also electricity to household equipment as stoves and ovens can be considered as heat supply to the building) it's

T = termograph W = cup-anemometer

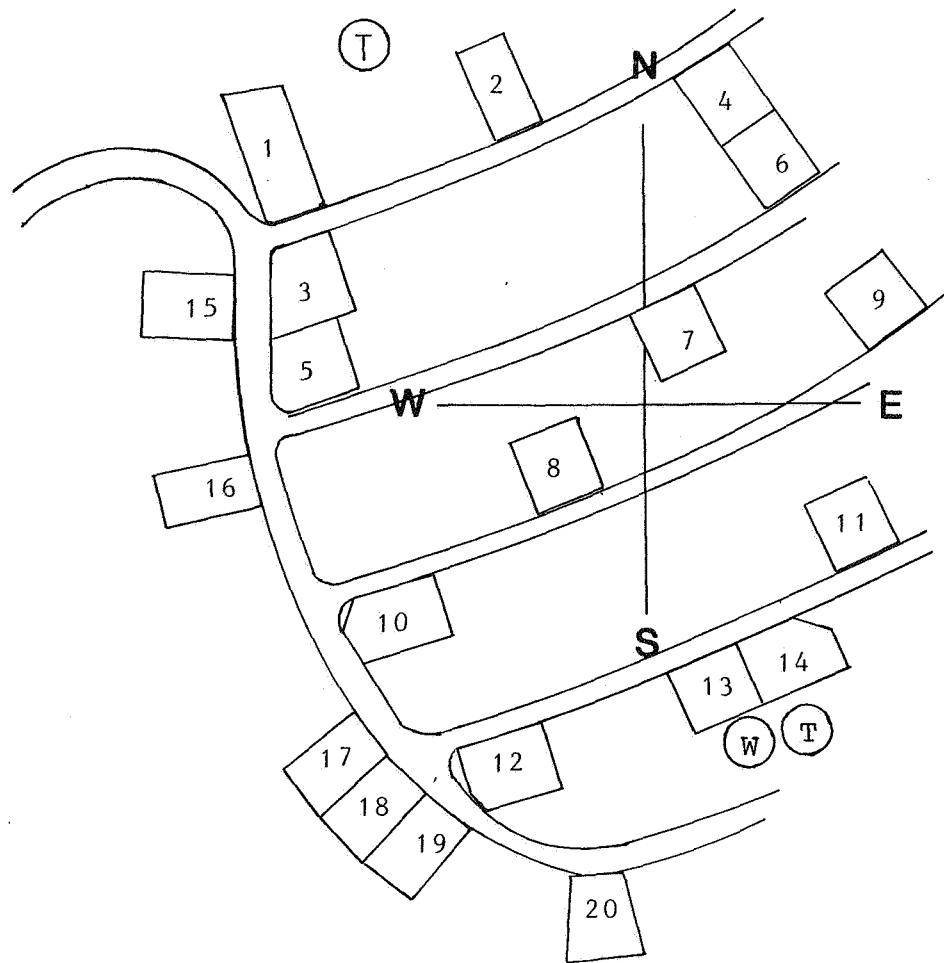


Figure 4.

the insolation through the windows and the heat emitted from persons. The electric heat supply was calculated due to the familie's notes on the used electricity. As supply for one day, the difference between the used electricity 7 am the following day and 7 am the actual day was used. The insolation through the windows was calculated from datas on house design (window-sizes and direction) and insolation from a nearby situated airport (10,5 km away). At the airport the solar radiation to a horizontal surface was registrated as energy amount per hour and per day. Since the insolation to vertical and horizontal surfaces may be assumed as linear to the cloudiness, M , it's possible to find a relation between the ratio Q_V/Q_H and the cloudiness for difference vertical directions and months of the year, see figure 6. Q_V and Q_H is the insolation to a vertical and to a horizontal surface respectively.

Date	Used amount of electricity 7 am kWh	Used amount of water m ³	Wash. mach. Dish. mach. Number of runs	
26/1	84192.3	0.580		1
27/1	84310.3	0.902	3	1
28/1	84434.7	0.350		1
	Kitchen fan Drying cabinet Time of use min	Window opening Number of windows Time min		Temperature °C
26/1	30	3	30	20
27/1	30 180			20
28/1	45			20
	Number of people at home Time: 0-2-4-6-8-10-12-14-16-18-20-22-24			
26/1	4	3	2	4
27/1	4	3	2 0 3 4	3 4
28/1	4	3	2	4

Figure 5.

Consideration to shadowing was taken by the use of a shadowing factor ψ . This factor was set as either 1, for unshadowed facade, or 0,5 for partly shadowed facades. Use of venetian blinds and curtains was assumed to give a reduction of the insolation through southeast-south- and southwest orientated windows during days with more than 50 % clouds, i.e. $M = 0,5 - 1,0$. This reduction was expressed with the factor ϕ , set to 0,5 at above mentioned conditions and 1,0 otherwise. Transmission through the window-panes was taken in consideration by the use of a transmission factor τ . This was set to be 0,75 for 2-pane windows and 0,65 for 3-pane windows. Now the insolation through a window during the day can be calculated as

$$Q_s = q_s \cdot \tau \cdot \psi \cdot \phi \cdot Q_H \quad \text{Wh/day} \quad (16)$$

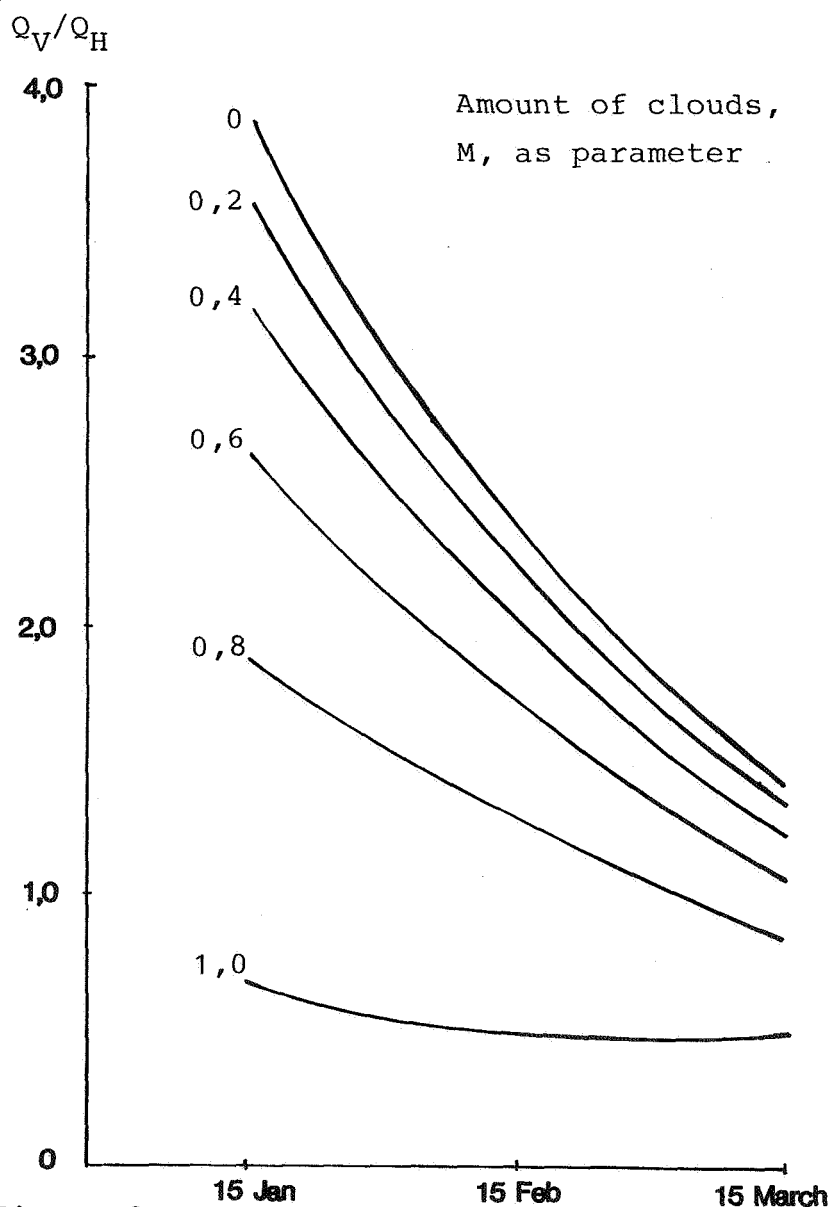


Figure 6.

Where q_s is the ratio Q_V/Q_H see figure 6

Q_H is the registered insolation to a horizontal surface at the airport

With known direction and size of the building's windows this procedure made it possible to estimate the total heat supply during the day due to insolation. The person heat, i.e. the heat emitted from people in the house, was calculated from the familie's notes on number of people in the house during the day. Different people emit varying amounts of heat, depending on both the size and the age. As an average 90 W per person

was used when estimating the persons' contribution to the heat supply.

3.2 Heat losses

The main heat losses from the building, and also the heat losses we are interested to estimate, are the transmission and ventilation heat losses. Since these are difficult to measure directly, we have calculated them as the difference between heat supply and heat losses due to other reasons than transmission and ventilation. These heat losses are hot water losses to the drain, when using washing- and dishing machines and during normal water consumption, warm-air losses, when using kitchen fan and drying cabinet, and heat losses due to window opening. The last two losses, warm air losses and heat losses due to window opening, are of course ventilation losses. But they are ventilation losses, not mainly dependent on the building and the weather conditions, but on human activities. They are therefore included among the heat losses above.

Hot water losses, from use of washing- and dishing machines, were calculated from producers datas on their products. Datas giving water consumption and used temperatures. Machinery used in the households were known trough an inquiry to the families at the start of the investigation. The water, not used in machines, was assumed to consist of 30 % hot water. 90 % of its energy was taken as heat loss to the drain. Heat losses, due to use of kitchen fan and drying cabinet, were also calculated out of producers datas, giving air flows and temperatures. Finally, heat losses due to window opening, were calculated with use of results from an investigation by Widegren-Dafgård⁵. According to these results, the air flow through a window opening, with height 1 m and with 0,3 m, is approximately 0,05 m³/s. Thus the heat loss during day can be calculated as

$$Q_w = 0,05 \cdot \rho \cdot c_p \cdot \Delta\theta \cdot t \quad \text{Wh/day} \quad (17)$$

where ρ and c_p are density and heat capacity for air respectively

$\Delta\theta$ is temperature difference between inside and outside

t is the window opening time, according to family notes, see figure 5

When the heat supply and heat losses are calculated the heat losses due to transmission and ventilation, Q , can be determined for every house and day, see table 1.

Table 1

Date	Heat supply			Heat losses (kWh/day)					Heat loss kWh/day	Specific heat loss (kWh/K, day)
	El. Q_{el}	Pers. Q_p	Sol. Q_s	Water Q_v	Wash. Q_w	Dish. Q_d	Fan Q_a	Dry. Q_{dr}		
Jan 12	105.0	6.3	0.6	8.1			0.2	0.3	103.3	4.87
13	69.7	5.3	7.4	2.1				0.4	79.9	3.43
14	118.1	6.1	0.8	3.6		1.3	0.3	0.3	119.5	5.69
15	92.3	4.9	3.7	2.5				0.3	98.1	4.81
16	89.6	5.5	7.2	6.3			0.6	0.3	95.1	4.55
17	78.7	4.9	2.1	2.9				0.9	81.9	3.44
18	83.3	6.5	8.0	5.6		1.3		0.4	90.5	3.30
19	83.2	6.0	2.2	5.8			0.4	0.4	84.8	3.50
20	97.0	6.2	6.1	5.5			0.4	0.4	103.0	4.09
21	90.5	7.6	7.7	2.7			0.5	0.4	102.2	3.80
22	75.9	6.7	5.6	3.3				0.9	84.0	3.80
23	70.2	6.2	1.2	4.0				0.3	73.3	3.72
24	80.6	8.4	0.9	6.0				0.3	83.6	4.38
25	93.1	7.9	1.0	11.5		1.3	0.4	0.4	88.4	4.68
26	64.8	7.5	7.4	5.4			0.4	0.7	73.2	3.18
27	70.3	6.4	8.3	2.8			0.2	0.7	81.3	3.73
28	75.3	5.8	0.7	2.9	1.9	1.3	0.2	0.3	75.2	4.25
29	69.8	6.2	0.7	2.5			0.2	0.6	73.4	4.01
30	73.9	6.4	0.8	3.9				0.3	76.9	4.27
31	76.6	6.5	0.7	3.0			0.3	0.6	79.9	4.37

In table 1 giving results for one of the 20 houses, the specific heat loss, $Q/\Delta\theta$, is also calculated. Wind direction and velocity, as well as the outdoor temperature, was registered at a place in the south part of the area, see figure 4. A cup-anemometer on 5 m height, was used for wind registrations. Supplementary outdoor temperatures were also registered at a place north of the area. Wind velocities at street level were measured with a handhold anemometer. These velocities were related to simultaneous cup-anemometer recordings, giving velocity ratios shown in figure 7.

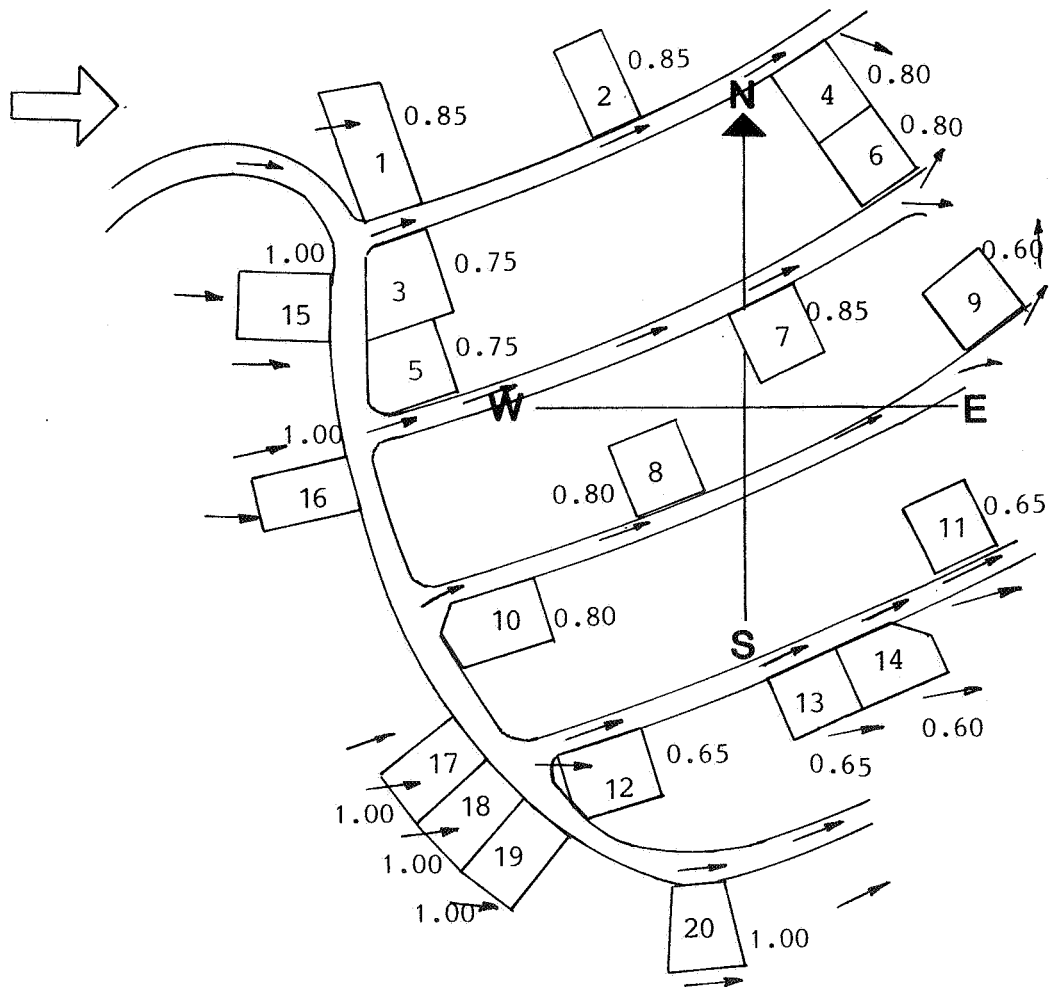


Figure 7.

These velocity ratios were determined at westwind, which was the dominating wind direction during the investigation period. They were later used to calculate local wind velocities, v_1 , from the cup-anemometer recordings. For every house and day, we now had three

parameters, the specific heat loss during the day, $Q/\Delta\theta$, the mean local wind velocity during the day, v_L , and the mean outdoor-temperature during the day, $\Delta\theta$. These parameters are given for some days for one of the houses in table 2.

Table 2

Date	Temperature difference $\Delta\theta$ K	Wind- velocity v_L m/s	Specific heat loss $Q/\Delta\theta$ kWh/K, day
Jan 12	21.2	4.6	4.87
13	23.3	0.2	3.43
14	21.0	5.3	5.69
15	20.4	3.2	4.81
16	20.9	2.7	4.55
17	23.8	1.1	3.44
18	27.4	0.9	3.30
19	24.2	2.1	3.50
20	25.2	3.6	4.09
21	26.9	0.2	3.80
22	22.1	2.6	3.80
23	19.7	0.7	3.72
24	19.1	2.6	4.38
25	18.9	3.7	4.68
26	23.0	1.9	3.18
27	21.8	1.4	3.73
28	17.7	1.0	4.25
29	18.3	1.4	4.01
30	18.0	1.7	4.27
31	18.3	1.7	4.37

4. Results

The calculated parameters above have been analyzed using a relation according to equation (15). This can be circumscribed as

$$Q/\Delta\theta = A + B \cdot \Delta\theta^{1/2} + C \cdot v_L + D \cdot v_L^{4/3} \quad (18)$$

Putting the calculated parameters for one house into equation (18) gives an equation system with as many equations as measuring days. Changing this equation system into a matrix equation will make it possible to solve the unknown matrix x , containing the constants A, B, C and D . With these constants known, we have a relation giving the specific heat loss for different wind velocities and temperature differences. Figure 8 shows this relation at three temperature differences for one of the houses.

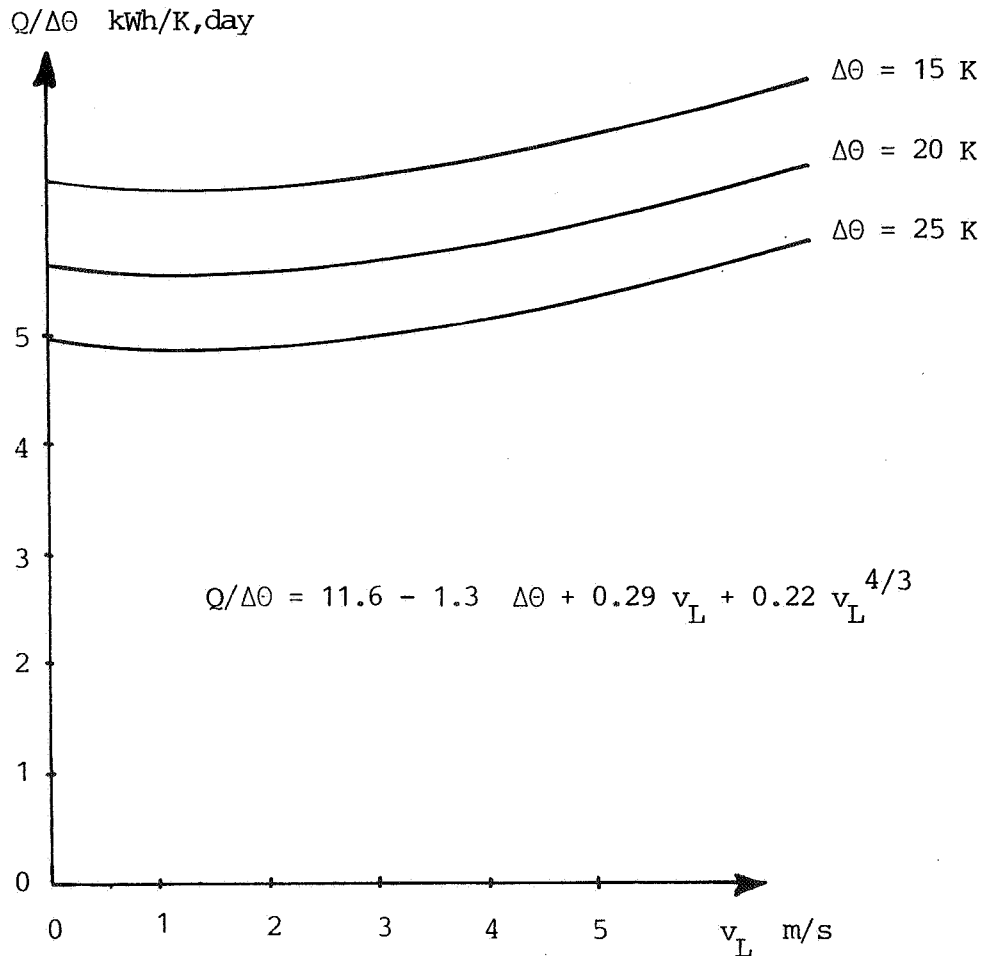


Figure 8.

The solution of the constants A, B, C and D , gives curves for the wind depending specific heat loss, that, for most of the houses, has a minimum. This minimum lies at wind velocities between 0,5 - 2,0 m/s. One can partly explain this minimum by the reduction of transmission heat loss with rised wind velocity that's possible during clear nights. The minimum may also be a result of the mathematical adjustment of the relation to the measuring datas.

The most surprising result is that the constant B is negative for all the 20 houses. Since the thermal forces increase with temperature difference, one would expect the specific heat loss due to this ventilation to increase with increased temperature difference. The constant B would then be positive. The only plausible explanation to the negative value of B is that the habits of the families change with outside temperature. With low outside temperature, and consequently high temperature difference $\Delta\theta$, people seem to be more inclined to keep their house tight.

Despite consideration has been taken to window opening, there is still a variation in opening width and opening time that's unknown. And above all there's an unknown variation in door-opening. Malik⁶ found in an investigation similar tendencie. He found that an occupied house acted as if it had two different porosities. In cold weather the house was tighter than in mild. He thought that in mild weather, windows and doors are opened rather frequently and are closed carelessly, where as in winter, people make sure their windows and doors are closed tight.

The percentage change of the specific heat loss, $Q/\Delta\theta$, is shown in figure 9 for all the 20 houses. The temperature difference is here 20K. Equations of the type (18) may be used to determine the possible energy saving from reducing wind influence. It can be calculated as the procentual difference between average heat loss during the investigation period and the heat loss at the wind velocity giving minimum specific heat loss, se figure 8 and 9. Since the average wind- and temperature conditions, during the investigation period, January and February, were approximately the same as the average conditions in this part of Sweden during heating season, the calculated possible energy-saving is a good estimation for the whole year. Table 3 gives the calculated possible energy saving in percents.

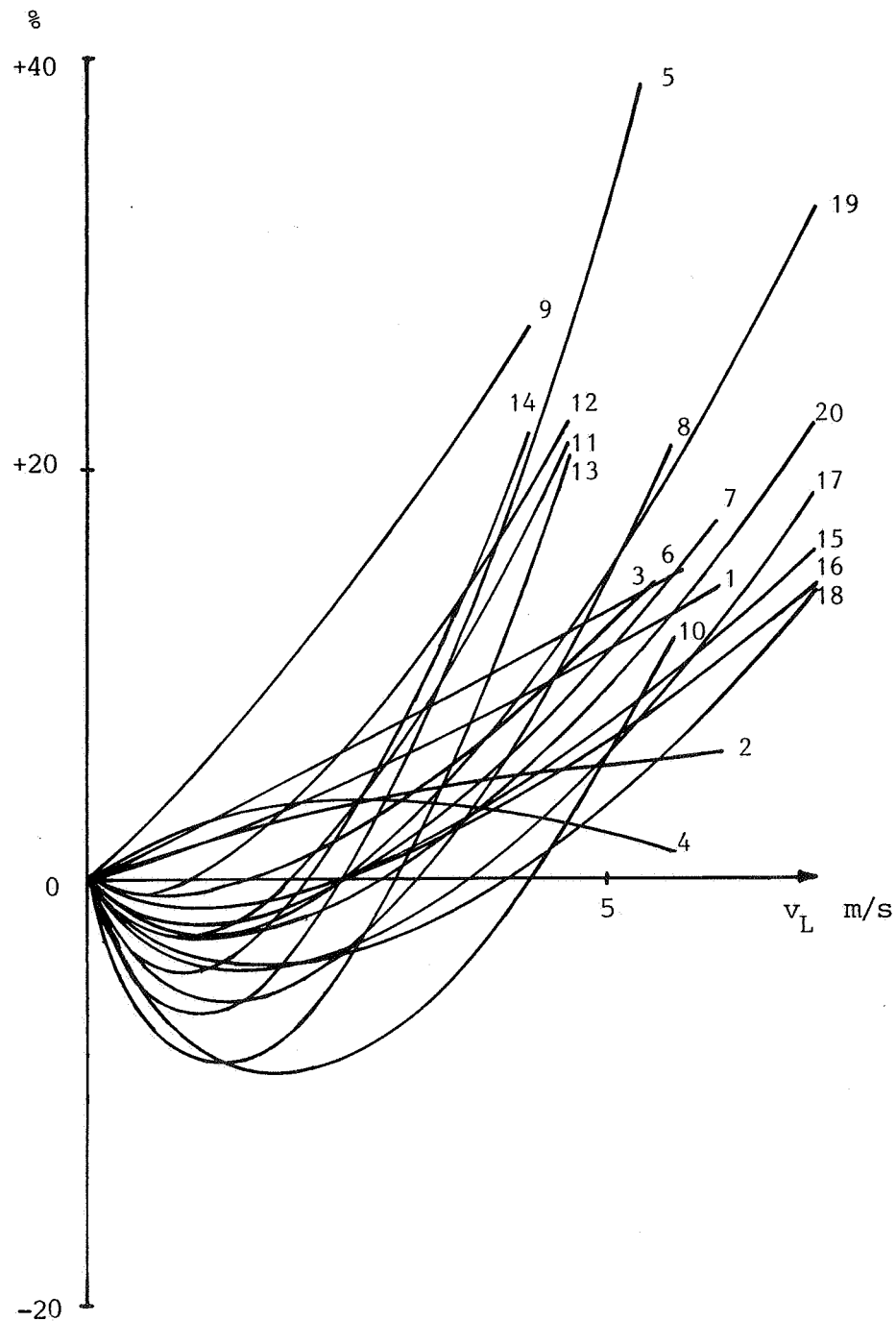


Figure 9.

The possible energy-saving is between 3,3-9,4 %. This energy saving may either be achieved by reducing the wind velocity to 0,5-2,0 m/s, or by tightening the house.

Table 3

House nr	Possible energysaving %
1	5.0
2	3.3
3	3.3
4	3.1
5	8.8
6	5.7
7	4.0
8	4.6
9	9.4
10	8.3
11	4.9
12	5.3
13	5.3
14	4.8
15	3.8
16	3.2
17	4.5
18	4.1
19	7.6
20	5.3

Figure 10.

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ENERGY EFFICIENT DOMESTIC VENTILATION SYSTEMS FOR ACHIEVING
ACCEPTABLE INDOOR AIR QUALITY

3rd AIC Conference, September 20-23 1982, London, UK

PAPER 23

ON FLOW IN NARROW SLOTS APPLIED TO INFILTRATION

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SYNOPSIS

The normally used equation for calculation of infiltration flow rates into a house, is

$$q' = \text{const} \cdot \Delta p^n$$

where Δp is the pressure drop over the walls and n a constant. The constant n is normally assumed to $2/3$ but sometimes also values as $0,5$ or even 1 can be seen in the litterature.

In the paper is the constant n calculated assuming a *non fully developed* infiltration flow. The constant n will for this asumption take values beetween $0,67$ and $0,77$ if the slots where the flow take place are long enough to get a flow close to a developed one.

ON FLOW IN NARROW SLOTS APPLIED TO INFILTRATION

1. Introduction

Flow through cracks around windows, doors etc are to a large extent the cause for the leakage into a house. Normally the flowrate, q , through such cracks estimated by

$$q' = c \Delta p^n \quad (1)$$

where Δp is the pressure difference in Pa over the crack

c a coefficient with a physical meaning (the flowrate at $\Delta p = 1$ Pa) and

n an exponent with a numerical value near $2/3$

The coefficient c in equation (1) is proportional to the length, L , of the crack.

Equation (1) has been discussed in a lot of papers. Nylund (1979) gives n to 1 for laminar flow and $1/2$ for turbulent flow. Normally - according to Nylund - there are some cracks with laminar flow and some with turbulent. Therefore, for practical cases

$$1/2 \leq n \leq 1 \quad (2)$$

Etheridge (1977) compared experimental results for q' , with a discharge coefficient.

$$c = \frac{q'}{A} \sqrt{\frac{8}{2\Delta p}} \quad (3)$$

where A is the cross section area of the air flow ρ in the density of the air.

He found that the discharge coefficient could be written as

$$c = \sqrt{\frac{1}{c_1 + B \frac{z}{d_n} Re}} \quad (4)$$

Where c_1 and B are constants

$$R_e = \text{Reynolds number} (= \frac{(q/A) \cdot d_n}{\nu})$$

ν = the kinematic viscosity

d_n = the hydraulic diameter, i.e. for a long slot = $2s$ where s is the width of the slot, and

z = the distance the air flows within the slot

Etheridge also finds that this form for the discharge coefficient corresponds to that for a laminar flow with a singular loss (at the end of the crack).

Honma (1975) suggested that the exponent n in equation (1) is dependent on the pressure loss Δp .

2. Flow in a crack

The flow in a crack is seldom a so called *developed flow*. The crack is normally short and the flow changes direction. In neither of those cases the flows will be developed. The length of undisturbed flow in a channel to get a fully developed flow is

$$L_L > \phi d_n Re \quad (5)$$

where ϕ depends on the aspect ration for the crack, see picture 1, after Han (1960).

For a tall slot $L \gg s$ the value $\phi = 0,075$ will be used for estimating the entrance length.

In a "normal crack" an old window ($s = 0,0015$ m, $L = 1$ m, $q \approx 1$ m³/hm) we will then have

$$L_L = 0,075 \cdot 2 \cdot 0,0015 \frac{1 \cdot 2}{3600 \cdot 15 \cdot 10^{-6}} = 8 \cdot 10^{-3} \text{ m}$$

The flow is therefore normally not developed. In some constructions the crack is so long so that a developed flow will occur just at the end of the crack.

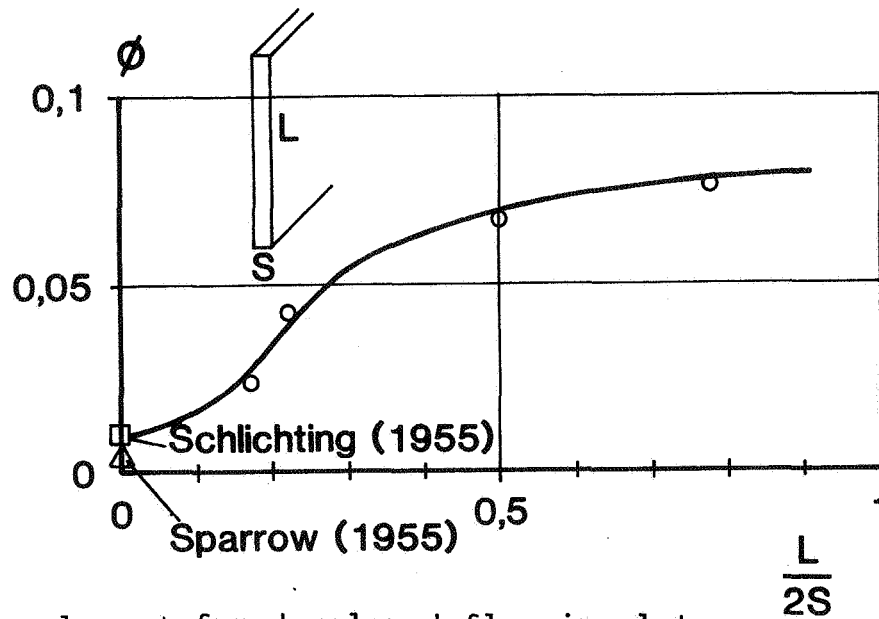


Figure 1. ϕ for developed flow in slots.

The pressure drop in such a flow has been calculated by a lot of researcher. Schiller (1922) showed that the pressure drop could be regarded as composed by two parts:

- o one part depending on the (increased) value for the friction coefficient λ in the slot
- o one part caused by the change in kinematic energy

To these is of course to add the losses at the end of the slot.

The pressure drop can be calculated as

$$\Delta p = \frac{1}{2} \rho \left(\frac{q}{A} \right)^2 \left(\frac{\psi}{Re} \right) \left(\frac{L_s}{d_h} \right) + K \quad (6)$$

(for the case of $L_s = L_e$). Here are ψ and K coefficients as in table 1.

Equation (6) can also be written

$$\Delta p = \frac{1}{2} \rho \left(\frac{q}{A} \right)^2 \left(\frac{\psi_1}{Re} \right) \quad (7)$$

where ψ_1 is another coefficient, see picture 2. Equa-

tions similar to (6) and (7) have also been given by Kreith (1957) but for tubes.

Table 1. Values for ψ and K according to Han (1960)

$\frac{L}{2sRe}$	ψ	K
1,00	56,9	3,02
0,75	57,8	3,00
0,50	62,1	2,80
0,25	72,8	2,56
0,125	82,3	2,16
0	96,0	1,85

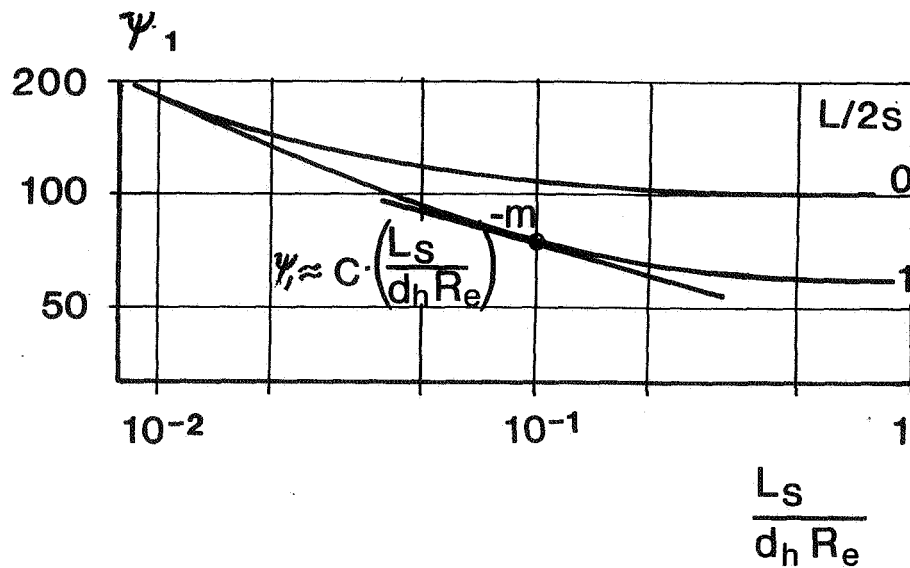


Figure 2. The coefficient ψ_1 in eq (7).

Using the values in the example above we will get

$$\frac{L_s}{d_h Re} = \frac{8 \cdot 10^{-3}}{2 \cdot 10^{-3} \cdot \left(\frac{1 \cdot 2}{3600 \cdot 15 \cdot 10^{-6}} \right)} = 0,1$$

For this value we can estimate ψ_1 from picture 2 as

$$\psi_1 \approx \text{const.} \cdot R_e^m \quad (8)$$

see also Kreith (1957) which introduced in (7) will give

$$\Delta p = \text{const } q^{1+m} \quad (7a)$$

for a given configuration. As can be found from picture 2

$$m \approx 0,3 \quad (9)$$

a value that will be slightly higher for $\frac{L_s}{d_h R_e} < 0,1$.

For a "tight" window (or similar) the value of d_h will be smaller than in the example above and so the flow rate q .

This means that $\frac{L_s}{d_h R_e}$ will decrease and so m will increase.

For $\frac{L_s}{d_h R_e} = 0,01$ a value of

$$m \approx 0,5 \quad (10)$$

is reasonable. It can be used for most cases concerning "tight" constructions. The pressure drop as given by equation (7) will therefore (again for a given geometry) give the (flow-) equation (1) and the value $n = 2/3$. For houses with larger leakage the value will be

$$q = \text{const} \cdot \Delta p^{0,77}$$

if $m = 0,3$ is used.

For the case of more komplex configuration of the slot similar equations will be useful.

3. Conclusions

The pressure drop in a crack - for instance such as those around doors, windows etc - can be calculated using an eq. $\Delta p = \text{const} \cdot q^n$, where q is the flowrate

and $n \approx 1,3 - 1,5$. The reason why n differs from 1 or 2, as for laminar and turbulent flow is that the latter applies for developed flow. Developed flow occurs very seldom in the slots under consideration.

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