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USE OF LOW TEMPERATURE AIR FOR COOLING OF BUILDINGS

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ABSTRACT

The topic of this paper is the use of low temperature air (40 F or 5 C) for room cooling. Cold air systems can offer energy and space savings relative to higher temperature cooling systems. As the supply temperature and flowrate are reduced, considerations such as adequate flowrate, jet dumping or separation, condensation on duct walls, and decreased relative humidity become increasingly important. Cold air jet separation from the ceiling can be a problem resulting in unacceptable thermal discomfort in the occupied zone. In the paper, the behavior of cold air jets , including throw and separation, is examined. A simple jet model is presented, and throw and separation point relationships are developed. The results from the ideal jet model suggest that for the same cooling capacity, the resulting room air motion, jet throw, and jet separation point will be the same for both conventional and cold air jets if the jet momentum is held constant.

INTRODUCTION

Cold air distribution systems have been proposed as a method to be used in conjunction with thermal storage systems for commercial building air conditioning. Cold air distribution systems can offer greater energy and load savings relative to competing systems due to smaller duct and fan size requirements. With a cold air distribution system, both the supply air temperature and flowrate are lower than in conventional systems for the same cooling load. The cold air is supplied to a zone at 40 F (5 C) instead of the conventional value of 55 F (13 C). Due to the lower flowrate, the fraction of outside air is increased to meet the required outside air ventilation rates, typically 15-20 cfm of outdoor air per building occupant specified by ASHRAE Standard 62-1989.

Cold air systems were originally proposed in 1947 (MacCracken, 1986) for retrofit residential air conditioning, but only have been used in commercial buildings since 1985, as an option to be used with thermal storage systems (Dorgan and Ellison, 1988). One popular method of producing cold air is to use the 34 F to 38 F chilled water produced by ice storage systems. The use of ice storage systems has increased in recent years as a cost effective means of reducing cooling peak power and associated demand charges for commercial buildings. The energy consumption of ice storage and cold air distribution systems was examined by Hittle and Bhansali (1990), who found that ice storage systems with cold air distribution were preferable from an energy basis when compared to an ice storage system with normal temperature distribution. It is not clear what the ideal supply air temperature should be in a cooling application. As the supply temperature and flowrate are reduced, considerations such as adequate flowrate, jet dumping or separation, condensation on duct walls, and decreased relative humidity become increasingly important.

In this paper we examine the behavior of cold air jets, specifically, jet throw and separation. We present a simple model of jet separation, and develop throw (velocity - distance) equations for cold air jets. Cold air jet separation from the ceiling can be a problem resulting in unacceptable thermal discomfort in the occupied zone. The thermal discomfort is due to both excessive draft and low temperatures. To mitigate possible dumping or separation of the jet, fan powered mixing boxes have been used in many commercial thermal storage applications. In these applications, mixing boxes are placed in the supply ducts to mix room air with cold primary air upstream of the diffuser outlet. However, the initial and operating cost of the mixing boxes has a negative impact on the energy and economics of the ice storage/cold air distribution system (Dunbar, 1990). Mixing boxes do not need to be used if the cold air supply can be mixed with the room air in the room without jet separation using induction diffusers.

To achieve mixing, the duct air is supplied to the room in the form of a jet. Some fraction of the room volume is required for the mixing process to occur. Thus the air jet often is directed along the ceiling or downward at a corner of the room. The resultant motion of the jet will depend on the type and location of the outlet, especially the distance between the outlet and ceiling, and the initial momentum and temperature difference in the jet. Due to the Coanda effect, jets initially will have a tendency to attach to the adjacent ceiling surface, however, they can detach from the ceiling, as shown in Figure 1, if their negative buoyancy is greater than the positive influence of the Coanda effect. Commercial application of momentum principles applied to high velocity air jets used in air conditioning applications is described further in Hillerbrant and Rudd (1984).

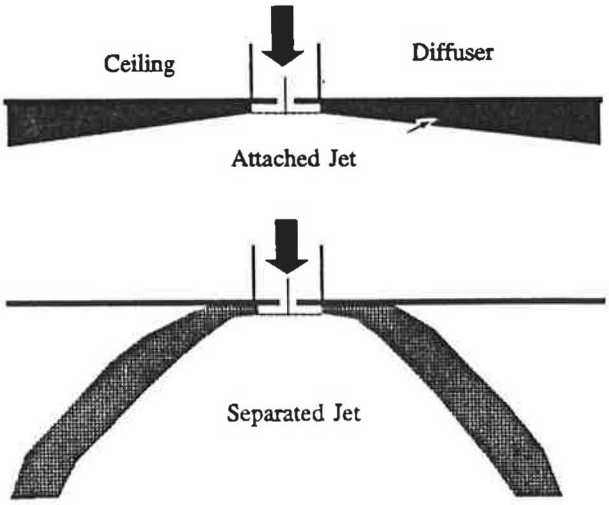

Figure 1. Attached and Separated Ceiling Jets

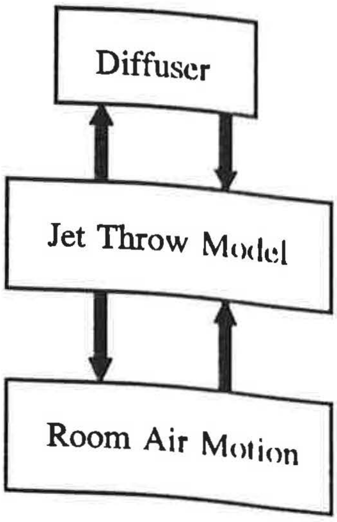

Figure 2. Jet Throw Design Tool

COLD AIR JET MODELING

In this section we describe a simplified jet model. The behavior of air jets in rooms is complex and dependent on the room geometry and thermal conditions. The location of the thermal load in a room is an important consideration, as the room air flow in the room is influenced by the existing convection currents. The point of separation of the cold air jet is influenced by the room convection currents and the radiation heat exchange. The cold air jet exchanges energy with the room through the air that is ejected into the jet and through convective heat exchange with the ceiling, which is cooled by the jet resulting in radiation from other room surfaces. As it is a difficult task to generalize the wide spectrum of room air motion patterns, design tools are used to indicate correlations between jet characteristics and room air motion.

As shown in Figure 2, a design tool used as an interface between the jet characteristics and the room air motion is the throw of the jet. The throw is usually defined as the distance from the diffuser outlet to the location where the centerline velocity has decreased to a specified terminal velocity between 150 fpm (0.75 m/s) and 50 fpm (0.25 m/s). The recommended throw depends on the type of diffuser (ASHRAE, 1989). For example, a throw of 30% of the room length is recommended for ceiling slot diffusers, a throw of 80 % of the room length is recommended for circular ceiling diffusers, while a throw of 180 % is recommended for high sidewall grilles. Jets dropping down before the throw distance may result in unacceptable draft in the occupied zone. The throw of a jet from a supply air device at a given air flow rate is calculated in a standardized way from diffuser tests. These tests are described by ASHRAE (1972), ADC (1977), and ISO Standards.

The results of the tests are evaluated by means of a simple jet model. The characteristic zones of this jet model are denoted zone two and zone three, as described in ASHRAE (1989). The ADPI Index has been developed for diffuser performance evaluation (Nevins and Miller, 1972), and also uses the throw of a jet as an interface between the diffuser and the room air motion. Zone three (radial) of the jet expansion is the characteristic zone for most diffusers. In this zone the flow behaves as if it was generated by a point source. The discussion in this section will be based on this zone, but a similar discussion is also valid for zone two (plane).

Centerline velocities have always been considered the most important jet characteristic for ventilation design, as they are the basis for the throw evaluation. The centerline velocity in zone three can for attached jets be calculated from:

$$\frac{u_x}{u_o} = K\frac{\sqrt{A_o}}{x+x_p}\sqrt{\rho_o/\rho_{xm}} \qquad (1)$$

where

u_x = centerline velocity at distance x (m/s)
u_o = effective outlet velocity (m/s)
A_o = effective outlet area (m/s)
K = a velocity decay constant
x = distance from outlet (m)
x_p = distance between outlet and virtual origin (m)
ρ_{xm} = mean density in jet at distance x (kg/m^3)
ρ_o = density of air at outlet (kg/m^3)

This formula has been proved to be applicable to a wide variety of jets from different type of diffusers. The formula can be derived from an assumption of a constant momentum jet. The velocity decay constant depends on the type of diffuser producing the jet. The initial jet velocity is related by definition to the jet momentum M, as follows:

$$M = \rho_o A_o u_o^2 \qquad (2)$$

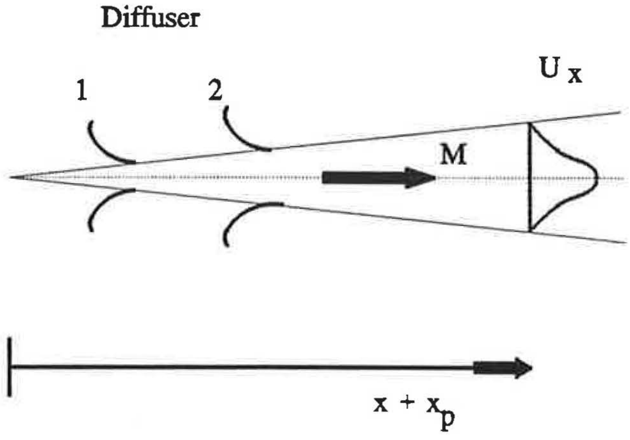

Figure 3. Equivalent Jets With Same Momentum

Reintroducing M in equation (1) gives

$$u_x = K\frac{\sqrt{M/\rho_{xm}}}{x+x_p} \qquad (3)$$

Equation (3) demonstrates that the centerline velocity in the jet at a certain distance x (m) from the outlet primarily depends on the jet momentum. K and x_p also depend on the type of diffuser and its mounting. One consequence of this is that jets from different sizes of the same type of diffuser will produce the same velocities at a given distance from the diffuser, if the jets have the same momentum, assuming K and x_p are independent of outlet velocity. This is illustrated in Figure 3 for two diffusers of different outlet area but with the same virtual origin. The same momentum for two jets also implies that their outlet Reynolds Numbers will be very similar. The value of x_p probably is the most variable. It is often small relative to x and thus is often ignored.

For the supply of cold air, the temperatures at the jet centerline must also be considered. Assuming that there is no thermal energy loss from the jet to adjacent surfaces, energy conservation implies that the centerline temperature difference between jet and room temperature) can be expressed as:

$$\frac{\Delta\theta_x}{\Delta\theta_o} = K_t\frac{\sqrt{A_o}}{x+x_p}\sqrt{\rho_o/\rho_{xm}} \qquad (4)$$

where $\Delta\theta_x$ = temperature difference at jet center at x (°C)
$\Delta\theta_o$ = temperature difference at exit (°C)
K_t = a temperature decay constant

The cooling capacity of the jet is:

$$Q = \rho_o c_p \Delta\theta_o u_o A_o \qquad (5)$$

where Q = cooling capacity (watt)
c_p = specific heat of air (J/kg, °C)

Introducing Q and M in (4) gives:

$$\Delta\theta_x = \frac{Q}{\rho_{xm}c_p}\left(\frac{M}{\rho_{xm}}\right)^{-1/2}\frac{K_t}{x+x_p} \qquad (6)$$

Equation (6) indicates that for jets from similar diffusers of different size, the temperature difference at the centerline at a given distance x could be similar for the same cooling capacity, Q, and momentum, M. Thus, there are strong indications that the similarity between jets are such that the "same" jet (as evaluated by the simple jet model) can be produced by similar diffusers of different size provided the cooling capacity and momentum of the jets are the same. This result suggests that the existing throw guidelines, obtained from conventional air supply tests, and which result in acceptable room air motion, can in such cases be used for cold air jets. It is not well established that K_t is constant when the outlet temperature difference is large.

If the cooling load Q is to be met by introducing colder air ($\Delta\theta_1$) than usual ($\Delta\theta_2$), the air flow rate q (m^3/s) will decrease in inverse proportion,

$$\frac{q_1}{q_2} = \frac{\Delta\theta_2}{\Delta\theta_1}\frac{\rho_2}{\rho_1} \qquad (7)$$

and the outlet velocity u_o must be increased to keep the momentum flow, M, constant:

$$M = \rho_1 q_1 u_{o_1} = \rho_2 q_2 u_{o_2} \qquad (8)$$

$$\frac{u_{o_1}}{u_{o_2}} = \frac{\Delta\theta_1}{\Delta\theta_2} \qquad (9)$$

This implies that in cold air distribution, the jet outlet velocity will need to be increased by the same proportion that the temperature difference is increased as indicated by Equation (9). The velocity increase can be accomplished by a decrease in the diffuser area. Some consequences of the increased outlet velocity are increased energy demand for the jet, in proportion to the increased velocity. However, to achieve control over the air distribution in the ductwork, a pressure drop at the outlet, or close to it, is necessary. If this pressure loss can be used to obtain the higher outlet velocity, it will result in no additional energy demand for the diffuser. In addition, there will be an increase in sound generation which will could provide one of the limits regarding suitable diffuser types for supply of cold air.

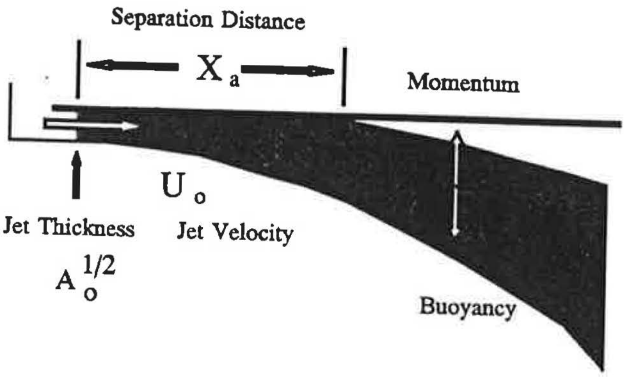

Figure 4. Schematic of a Separated Ceiling Jet

SEPARATION DISTANCE MODELING

Using a force balance applied to a simple jet model, the separation distance can be related to the jet momentum, density, and room cooling load. Separation occurs when the negative buoyancy force, due to the temperature difference in the jet is larger than the attachment force. A schematic of a separated jet is shown in Figure 4. It can be shown from the ideal jet throw equations that the upward attachment force, F_{Coanda}, on a segment of the jet, is proportional to M/x, so the Coanda force decreases with increasing x. From Archimedes principle, the downward gravitational force on a segment of the jet is proportional to the volume and temperature difference of the jet. This implies that the gravitational force goes as $Q\,x/\,M^{1/2}$, increasing linearly with distance. At any distance x, the ratio between the Coanda force and the gravitational force is proportional to momentum flow, and inversely proportional to cooling capacity, and distance squared as follows:

$$\frac{F_{Coanda}}{F_{gravity}} \sim \frac{M^{3/2}}{Qx^2} \qquad (10)$$

The jet separates from the ceiling at some critical ratio of these forces. The above force ratio can be written as an equation by including a constant C_a and the physical properties of the jet, for the separation distance x_a:

$$x_a^2 = C_a^2 \frac{c_p}{g\beta\sqrt{\rho_{xm}}} \frac{M_o^{3/2}}{Q} \qquad (11)$$

where β = volumetric expansion coefficient
C_a = a force ratio constant

This can be expressed in nondimensional form, using the outlet Archimedes number, as:

$$\frac{x_a}{\sqrt{A_o}} = \frac{C_a}{\sqrt{Ar_o}} \qquad (12)$$

where

$$Ar_o = \frac{g\beta\Delta\theta_o\sqrt{A_o}}{u_o^2} \qquad (13)$$

Grimitlin (1970) and Rødahl (1977) report values of C_a in the range 1.4 to 2.3 for an axial ceiling jet, depending on the location of the heat sources in the test room. Anderson et al. (1991) determined C_a to be about 1.6 for their experimental measurements of slot jet separation. They also predicted that C_a is a function of the square root of the velocity decay constant K for a particular diffuser.

If the separation distance x_a is to be at the same location relative to the room length when introducing cold air, the above model indicates that the ratio of the effective outlet area to the Archimedes number needs to remain constant, assuming that C_a remains constant. As the velocities in a separated jet may be too high for acceptable thermal comfort, the separation distance should be at least equal to the throw of the cold air jet.

The above model does not include effects such as the distortion of the jet cross section by the entrained air, and the ceiling friction. The distortion results in a larger spread of the jet parallel with the ceiling than perpendicular to the ceiling (Launder and Rodi 1983). The larger parallel spread could possibly increase the tendency for the jet to stay attached to the ceiling, compared with the slope indicated by the above equation. It should be noted that the separation point is influenced both by the temperature of the jet and the momentum of the air current. Norwegian experience from test rooms, for example, Rødahl (1977), is that the method of heating the room can influence the distance from the diffuser to the separation point by as much as a factor of two. This points out the need to start with accepted tools, such as the throw concept, when trying to relate diffuser characteristics to room air motion.

SUMMARY AND CONCLUSIONS

To maintain a given level of thermal comfort, separation of the jet from the ceiling should be avoided as long as temperature difference and velocities are too large. Analysis of ideal jets predicts that for the same cooling capacity, the room air motion, jet throw, and jet separation point will be the same for both normal and cold air jets if the jet momentum is held constant.

This is a very simple and useful result, as it suggests that existing throw data for diffusers can be used for incorporation of cold air diffusion technologies. For constant momentum, the jet outlet velocity should increase in the same proportion as the increase in temperature difference. Both computational and physical experiments are needed to examine and verify this hypothesis, and determine the limits of applicability.

ACKNOWLEDGEMENTS

Support from the Electric Power Research Institute and the Department of Energy is gratefully acknowledged. The EPRI contract monitor is Mr. Ronald Wendland.

REFERENCES

Air Diffusion Council. 1977. "Certification, rating, and test manual-1062R4." Chicago, IL.

Anderson, R., V. Hassani, A. Kirkpatrick, K. Knappmiller, D. Hittle, 1991. "Experimental and Computational Visualization of Cold Air Ceiling Jets." *ASHRAE Journal*, Vol. 33, No. 5 (May).

ASHRAE. 1989. Handbook-Fundamentals. Ch. 31.

ASHRAE. 1972. Standard 70-72. "Method of testing for rating the performance of outlets and inlets."

Dorgan, C., and J. Ellison. 1988. "Cold air distribution design guide." *EPRI EM-5730. March.*

Dunbar, M. 1990. "Thermal Storage." *Energy User News*. (October):25-26.

Grimitlin, M. 1970. "Zuluftverteilung in raumen (Supply air distribution in rooms)." *Luft und Kaltetechnik*. Vol. 5, 247-256.

Hillerbrant, B., and C. Rudd. 1984. "Air Jets: A presentation of the Flakt jet technique." *Australian Refrigeration, Air Conditioning and Heating*. (August):52-54.

Hittle, D., and A. Bhansali. 1990. "Expected Energy Use of Ice storage and Cold Air Distribution Systems in Large Commercial Buildings". *EPRI CU-6643*. February.

Launder, B., and W. Rodi. 1983. "The Turbulent Wall Jet--Measurements and Modeling."
Annual Review of Fluid Mechanics. Vol. 15, 429-459.

Miller P. 1990. "Diffuser selection for cold air distribution systems." *EPRI 2732-20*. May.

MacCracken, C. 1986. "Cold Air Systems", *Proc. Air Movement and Distribution Conference*, Purdue, Univ., West Lafayette, Indiana.

Malmstrom, T. 1975. Tekniska Meddelanden (Technical Notes)# 76. Dept. of Heating and Ventilating, KTH, Stockholm.

Nevins, R., P. Miller, 1972. "Analysis, evaluation, and comparison of room air distribution performance - a summary." *ASHRAE Transactions*. Vol. 78, p. 235.

Rødahl E. 1977. "The point of separation for cold jets flowing along the ceiling." *Clima 2000*. Belgrade, Yugoslavia.

BIOGRAPHY

Allan Kirkpatrick is an associate professor of mechanical engineering at Colorado State University in Fort Collins, CO. He received his B.S. and Ph.D. degrees in mechanical engineering from the Massachusetts Institute of Technology in Cambridge. He is a member of ASHRAE, ASME, and ISES. He has performed research in room air motion, modeling of solar building performance, and natural convection.