Rotary heat exchanger model for control and energy calculations

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Abstract

Rotary heat recovery exchangers are widely used in ventilation systems, and the units are known for their high efficiency and almost maintenance-free operation. Temperature efficiencies above 80% are not uncommon. Performing dynamical analyses of rotary heat exchangers are in many situations advantageous, especially in connection to installation of such equipment in VAV systems. Efficiencies and flows are varying parameters that are crucial for energy calculations, but also for control. The dynamical analysis can effectively be carried out by addressing a dynamical model. This paper presents a simplified model for a rotary heat recovery unit that can be used both in CAV or VAV systems for control and energy analysis. The model is based on a further development of existing models, and is able to account for varying leakage, varying flows and temperatures. The model has been implemented in the Matlab Simulink environment. Simulations show that the results represent real conditions in an adequate way. This has been verified through comparison with measurements.

Keywords: Rotary heat exchanger, modeling, simulation, energy, control.

Introduction

There are many types of heat recovery units (HRU) which are in use in ventilation systems. Examples are rotary heat recovery wheels, plate heat exchangers, water/glycol coil heat exchangers and heat pumps. Rotary heat recovery exchangers are widely used, and the units are known for their high efficiency and almost maintenance-free operation. Temperature efficiencies above 80% are not uncommon. Usually, for balanced systems, this means that the recovered power can be at the same rate. Unfortunately these units are also infamous for transmission of polluted air from exhaust to supply. Due to the rotation of the device, there will always be a possibility of direct connection between exhaust and fresh supply air. Cross-contamination due to rotary heat recovery units comes from two mechanisms; (1) Leakage, (2) Carryover. The former is dependent on the differential static pressure across the wheel, and latter (carryover) occurs as a result of air being entrained within the wheel volume. The degree of carryover air is strongly dependent on whether the unit has a purge section or not. A rotating HRU in a comfort ventilation system is usually built of plane and corrugated aluminum fins mounted in parallel to the rotational axis, forming numerous parallel, axial passages (directional oriented flow). The small diameter of the passages may initiate a transition to laminar flow. Heat is transferred from warm exhaust air to cold supply air by rotating the heated part of the wheel into the cold supply air. Rotary HRUs are continuously regenerative, since heat is carried from one fluid to another by a third medium. The rotor speed is normally relatively low and in a range of 3-15 rpm.

Accurate physical description of transient heat transfer within a rotary HRU is extremely complex. In addition to heat transfer from the surface to the surrounding air, both crosswise and long side, radial heat conduction within the wheel should be addressed. Important design parameters are:

- Diameter and depth of the wheel, and the size of the duct or AHU.
- Media mass and material properties.
- Surface structure.
- Flow regime and pressure conditions.

In this work, focus has been on establishing a simple model that is sufficiently accurate for control and energy calculations, which can be used integrated in system simulation as well.

Mathematical model

Parts of the modeling approach presented below have been adopted from Børresen et al. [1], and further developed to accommodate varying flow rates.

Model assumptions:

- Balanced ventilation. Exhaust air flow rate is equal to supply air flow rate.
- Non-hygroscopic, regenerative rotor
- The heating media can be separated into two halves, one for exhaust and one for supply. The temperature in each half is uniform.

- The media temperature on the supply side is approximately equal to the supply air temperature.

Model description (also refer to Fig. 1):

Simple heat balance:

$$\frac{d}{dt}\left(\frac{m_R}{2} C_R T_R\right) = Q \cdot \rho_A C_A (1-x)\left(T_O + T_E - T_S - T_D\right) \qquad \text{(Eq. 1)}$$

m_R and C_R are the rotor mass and specific heat capacity respectively. Q is the air flow rate, and ρ_A and C_A are the density and specific heat capacity of air. x is assumed to be the share of leakage through the unit (≥ 0), and a positive value indicates that air is leaking from exhaust to supply. Combining Eq.1 with the temperature efficiency (η_T) for balanced ventilation eliminates the discharge temperature and gives the following expression:

$$\frac{m_R C_R}{2 \cdot Q \cdot \rho_A C_A (1-x)} \frac{dT_S}{dt} = \left(1-\eta_T\right) \cdot T_O + \eta_T T_E - T_S \qquad \text{(Eq. 2)}$$

Leakage through rotary heat exchangers can in some cases represent significance for the dynamic performance and energy efficiency of the unit. Pressure conditions around the wheel (fan placements) are important, as well as carryover of air within the wheel. Earlier studies have shown that carryover is important especially for VAV systems, where the share of carryover become large at low flow rates [2], [3]. In the model, this has been compensated for simply by introducing a bypass factor. This means that a certain amount of air bypasses the wheel, and the flow rate through the wheel is reduced accordingly. Hence, after computing supply temperature, it should be corrected for the flow rate bypassing the rotor (leakage), i.e.

$$T_{S,korr} = (1 - x) \cdot T_S + x \cdot T_E \qquad \text{(Eq. 3)}$$

Temperature efficiency depends both on rotational speed and flow rate. In general, there are nonlinear relationships between efficiency and speed, and between efficiency and flow rate. At constant rotor speed, efficiencies at low air velocities are higher than efficiencies at high velocities. On the other hand, efficiencies at high rotor speeds are higher than those at low speeds, at constant flow rate. The relationship between efficiency and rotational speed, for a given flow rate, is assumed to be on the form expressed by Eq.4 and Eq.5.

$$\eta_T(n, Q) = \eta_T(n) \cdot \eta_T(Q) = \eta_n \cdot \eta_Q \qquad \text{(Eq. 4)}$$

$$\eta_n = 1 - e^{-C \cdot n} \qquad \eta_Q = A + B \cdot Q \qquad \text{(Eq. 5)}$$

n is the rotational speed of the heat recovery wheel and A, B and C are scaling/calibrating factors which must be determined from known data. Thus some (3) data sets must be available to calibrate the model correctly. Since better efficiencies are obtained for lower flow rates (assuming constant rotor speed), we know that η_Q will descend slightly as Q increases (negative B). To find A and B, we use two sets of data taken from the product catalog of the unit for the same rotor speed; (Q_1, η_1) and (Q_2, η_2), and we assume further that the derivative of η_Q is constant for all rotor speeds. To find C we use yet another set at an arbitrary flow rate (n_3, η_3), so that Eq.4 becomes:

$$\eta_T(n,Q) = \left(\eta_2 - \frac{\eta_2 - \eta_1}{Q_2 - Q_1} \cdot (Q - Q_1) \right) \cdot (1 - (1 - \eta_3)^{\frac{n}{n_3}}) \qquad \text{(Eq. 6)}$$

Together with Eq.2 and 3, Eq.6 represents the governing model for the rotary heat recovery unit. The model can also be used for variable flow rates (VAV systems).

Limiting the rate of heat transfer due to rotation of the wheel:

Consider that the exhaust temperature is altered instantly, for instance as a step. On the supply side of the wheel, the full effect of altered exhaust conditions cannot be seen instantly. That is due to the time it takes to transfer heat from the exhaust side, by the media, to the supply side. The rate at which heat is transferred is restricted by the rotational speed of the wheel. Thus, introducing a signal rate limiter on the exhaust inlet of the model seems appropriate.

Block diagram:

The block diagram of the model is shown in Fig. 2 and includes the efficiency estimation part. First the efficiency is calculated and put through a multiplexer together with other model input data. The exhaust temperature is fed through a first order filter in order to estimate ramp behavior. Finally the output (supply air temperature) is calculated.

Measurement setup

Data from laboratory measurements have been collected and used to validate the performance of the model. Measurements were performed in a laboratory on an AHU system containing an ABB EC2000 unit with a non-hygroscopic rotary wheel and no purge section. A DAQ-system was used together with Lab-view software to collect the temperature data. Logging was performed for different fan speeds (40%-100%, corresponding to 0.09- 0.25 m^3/s air flow) and rotor speeds (0-12 rpm). The measurement setup was as shown in Fig. 3, but only the measurement points T0, T4, A0 and A4 have been used for comparison to the simulated model output. Flow rates were measured using Brüel & Kjær gas analyzator and multiplexer system. Leakage over the heat recovery unit was calculated from the measured tracer gas concentrations. In this particular setup, fan placements were unfavorable with regards to leakage, causing a relatively large amount of air to flow from the exhaust to the supply side (refer to Fig. 4).

Results and Discussion

As suggested by the comparisons between measurements and simulations shown in Fig. 5 and Fig. 6, the performance of the model is reasonable. Deviations between measurements and simulations are shown in Fig.7. The sample correlation coefficients for both datasets are close to 97%, which again suggests good fit. Dynamically the model performs adequately. From Fig.5 we realize that the step responses follow measurements quite good. For steady state conditions the model shows deviations, especially for the low rotor speeds. This indicates that the efficiency description can be improved. However, more measurement data for different rotary units are required to improve the efficiency model further. Furthermore, assuming a linear relationship for between efficiency and flow rate introduce errors in the performance. A major disturbance in the measurement data is the large amount of re-circulating air (leakage). Modeling leakage using a simplified model may not be properly done, and the comparison indicates that the model performs better when the share of leakage air is small compared to the total flow rate. When the rotor speed is zero, heat will be transported from the exhaust to the supply by conduction through the rotor material. Pure conduction can to some degree be compensated for by introducing a minimum value of the temperature efficiency. This is however not addressed by the model at this stage.

Considering the simplified nature of the model, its performance naturally cannot take into account all possible variations that may occur. Nevertheless, it reflects the major tendencies caused by the most important parameters acting on the system. Thus it will be reasonable to conclude that the model is suitable for simulating different control scenarios, and it may also be used for detecting possible errors and problems. An example of application is tuning the control loops for sequential control of ‘cooling coil-rotary heat recovery unit-heating coil’, which may represent a challenge in VAV systems. The model can also be used for estimating energy performance.

Conclusions

A dynamical model of a rotary heat exchanger has been developed and accommodated for control and energy simulations. The model can also be used for varying flows (VAV systems). The temperature responses of the model have been compared to measurement data, and performance should be adequate for control and energy simulations. Considering the simplified nature of the model, its performance naturally cannot take into account all possible variations that may occur. Nevertheless, it reflects the major tendencies caused by the most important parameters acting on the system. The model can be used alone or as part of a larger simulation system, as a tool for revealing or preventing possible problems and errors during operation of similar real equipment.

.References

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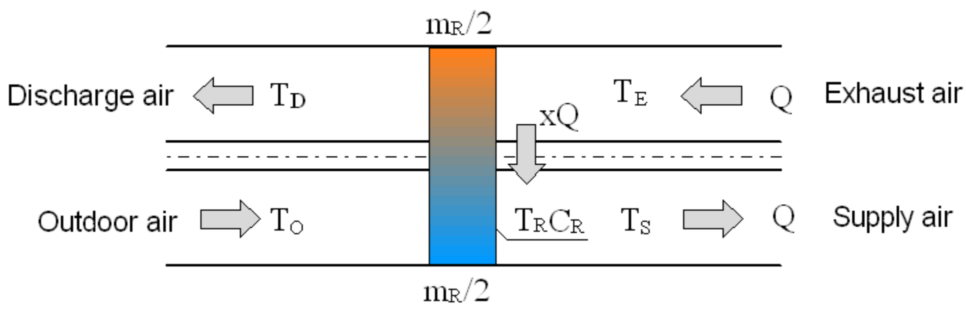

Fig. 1 Model outline

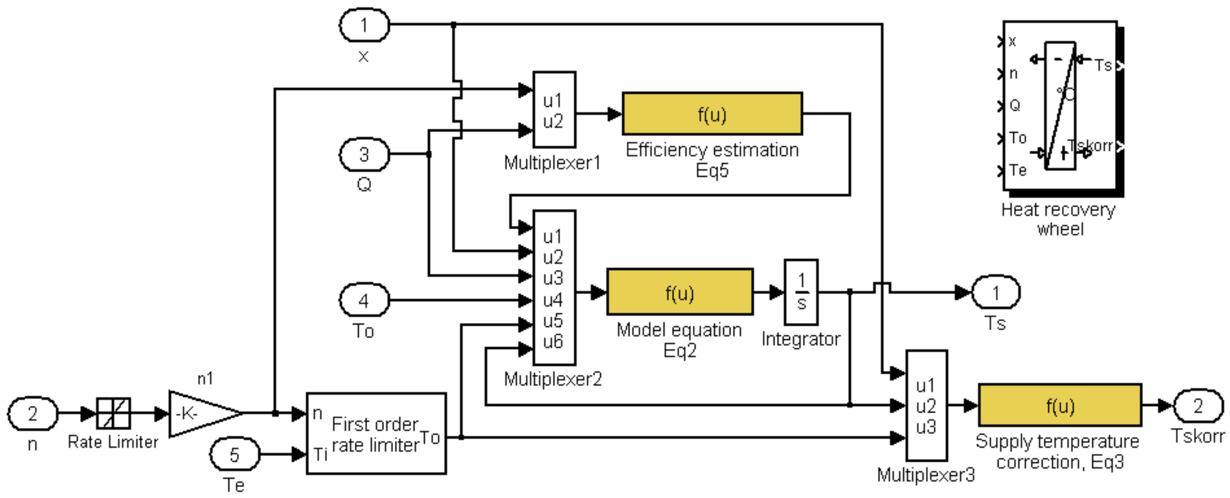

Fig. 2 Model block diagram (Matlab Simulink)

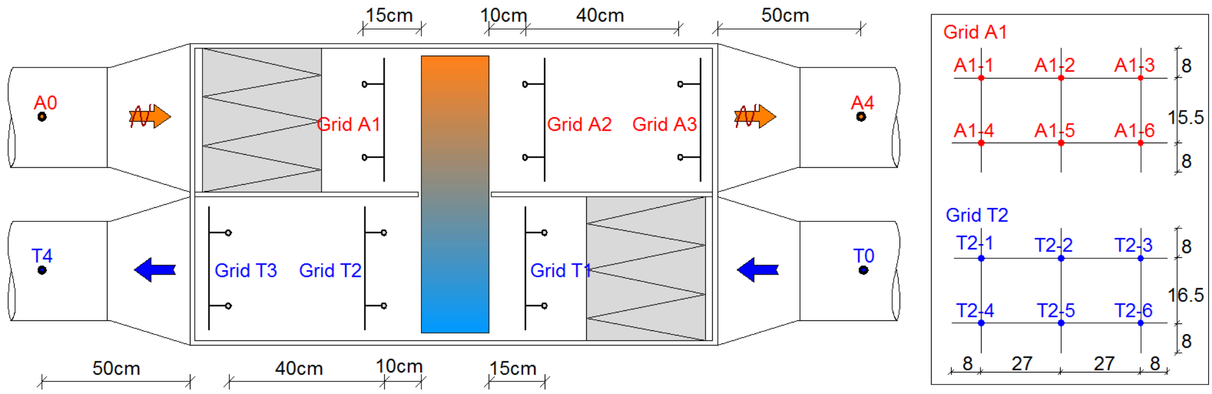

Fig. 3 Measurement setup

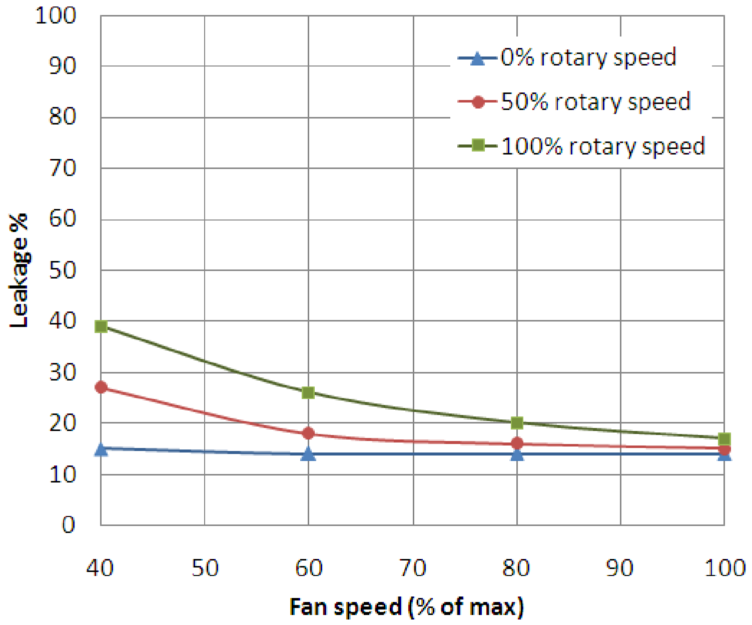

Fig. 4 Leakage [2]

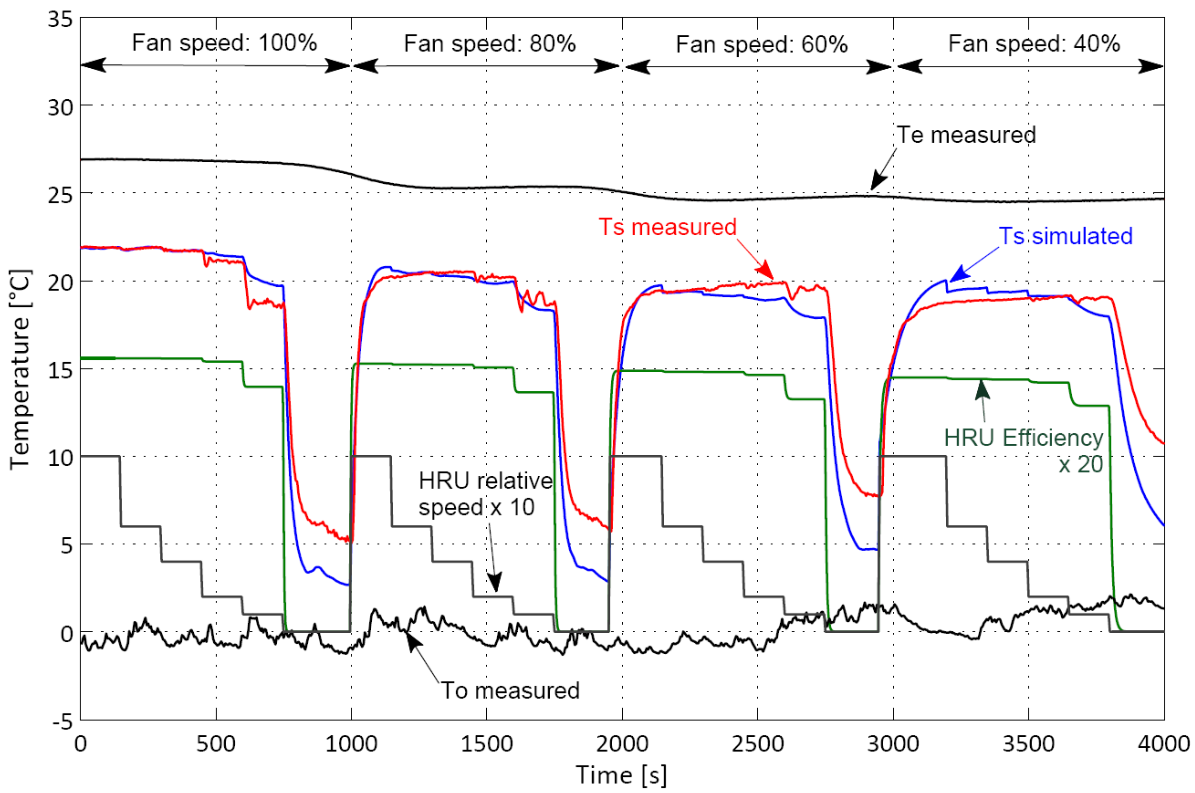

Fig. 5 Measurement and simulation data comparison (dataset 1)

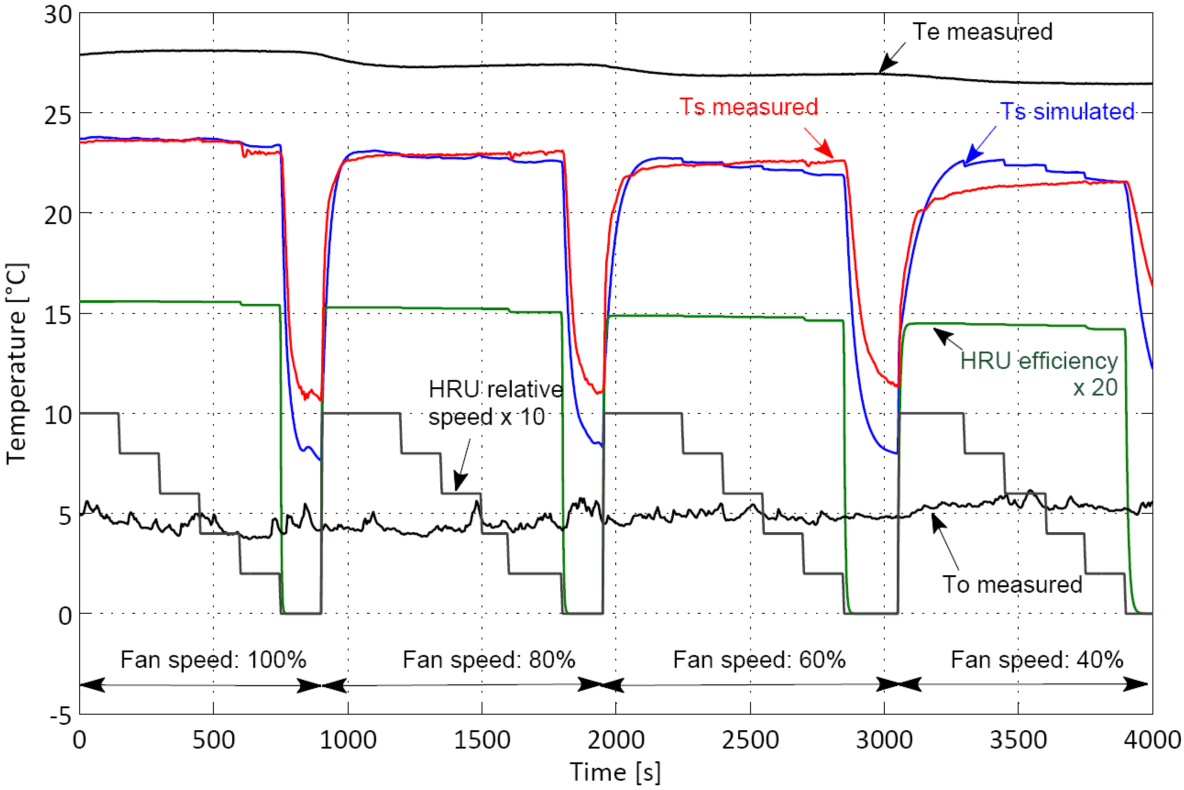

Fig. 6 Measurement and simulation data comparison (dataset 2)

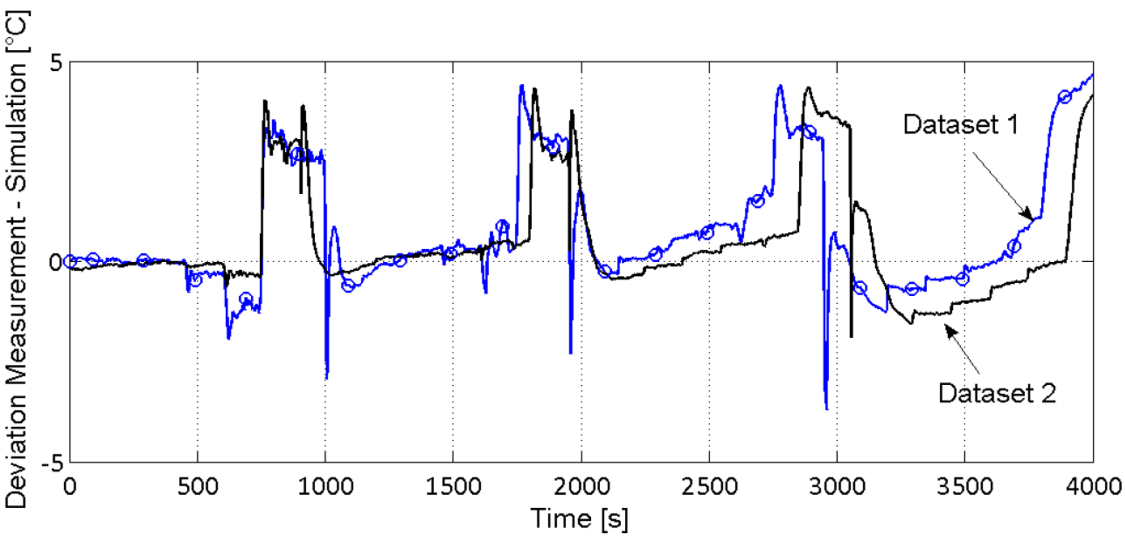

Fig. 7 Deviation between measurement and simulation data