Performance of Rotary Heat and Mass Exchangers

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Rotary regenerative heat and mass exchangers (enthalpy exchangers) can reduce air-conditioning costs in ventilated buildings by recovering energy from the exhaust air and transferring it to the supply air stream. In this study the adsorption isotherms of a desiccant used in a commercially available heat and mass exchanger are measured. The isotherms and other property data are incorporated into the program MOSHMX which numerically solves the governing equations for combined heat and mass transfer. The numerical results are then used to develop a computationally simple model for determining the performance of a specific enthalpy exchanger as a function of the air inlet conditions and the matrix rotation speed. The numerical results agree with the catalog information provided by the manufacturer.

The enthalpy exchanger model is used in the transient simulation program TRNSYS to estimate the annual performance. Integrated energy savings (heating and cooling) are determined for a commercial application (a 200-person office building) in three different locations in the United States, each representing a different climate. The ventilation system follows ASHRAE outdoor air guidelines with a ventilation air flow of 20 cfm/person (1.9 m^3/s total ventilation air flow).

Life cycle savings that take into account the initial cost of the exchanger as well as the heating and cooling energy savings are evaluated for both an enthalpy exchanger and a sensible heat exchanger over a system life time of 15 years. The present worth of the accumulated savings ranges from $28,000 to $38,000 for the enthalpy exchanger and from $7,000 to $24,000 for the sensible heat exchanger. The enthalpy exchanger results in greater payoffs in all locations, but its advantage over a sensible heat exchanger is most significant for cooling in warm and humid climates where the sensible heat exchanger performs poorly.

INTRODUCTION

Occupied buildings rely on outdoor air ventilation to keep the concentration of indoor-generated pollutants at a low level and maintain acceptable indoor air quality. Requirements for clean air in public buildings have resulted in recommendations given by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE 1989) to increase the minimum ventilation rates for commercial buildings. A major drawback of high ventilation, however, is the energy and associated cost required to condition (heat/cool and humidify/dehumidify) the entering outdoor air. It is possible to decrease the energy requirement by adding a rotary regenerator to the conventional space-conditioning equipment to allow energy recovery from the conditioned exhaust stream.

In a rotary regenerator, heat is transferred from the hot fluid to a solid energy carrier (the matrix) during the first period and from the solid to the cold stream during the second period. Continuous operation is permitted by rotating the matrix cyclically from one air stream to the other. Rotary regenerators may also be designed to transfer water

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between the two air streams by using a matrix containing a desiccant material. In this case, the regenerator exchanges both sensible and latent energy, and it is termed either a dehumidifier or an enthalpy exchanger. Dehumidifiers operate at relatively low rotation speeds with the objective to maximize the drying potential of the process stream. In contrast, enthalpy exchangers operate at higher rotation speeds with the objective to maximize the heat and mass transfer to or from the process stream.

There is an accepted set of equations that describe the heat and mass transfer in rotary regenerators (e.g., Maclaine-cross 1972, Kays and Crawford 1980, Jurinak 1982, Van den Bulck 1987, Farooq and Ruthven 1991, Chau and Worek 1995). The methods that have been used to solve these equations can be classified as direct numerical solution, analogy between heat and mass transfer and heat transfer, and analogy to counter flow heat exchangers. A number of investigators (Pla-Barby and Vliet 1979, Mathiprakasam and Lavan 1980, Maclaine-cross 1974) have employed finite differencing of the governing equations in primitive coordinates (temperature and humidity ratio). The MOSHMX FORTRAN program developed by Maclaine-cross (1974) solves the equations numerically and provides a convenient but computationally-intensive method of calculating the performance of heat and mass exchangers. If the Lewis number is assumed to be unity, the governing equations can be reduced to a simpler form in which enthalpy becomes the potential (e.g., Kays and Crawford 1980), however, two coupled, partial differential equations are still needed to represent the mechanisms. In the analogy method of Banks (1972), Close and Banks (1972) and Maclaine-cross (1972), a combined potential is defined that allows the two partial differential equations to be reduced to a single partial differential equation with two characteristic solutions. This latter approach provides a computationally simple solution for Lewis number of unity and it has been extensively used to represent dehumidifiers e.g, Banks (1972), Maclaine-cross (1972), Jurinak (1982) and van den Bulck (1987). The approach of Barlow (1982) that has been used in the DESSIM program solves the governing equations using an analogy to counter flow heat exchangers. This heuristic approach has been shown to provide results in agreement with direct numerical solutions (Schultz and Mitchell 1989). A wide variety of methods for solving the governing regenerator equations are available with tradeoffs between the speed of the solution and the level of detail that can be accommodated.

Most studies of rotary regenerators have focused on dehumidifiers and have not investigated the conditions relevant to enthalpy exchangers. Klein et al. (1990) established the minimum rotation speed under which maximum enthalpy exchange can be accomplished. Further, the equilibrium exchanger model they used was the basis for development of enthalpy exchanger effectiveness factors. However, their model is not applicable at the reduced speeds necessary to prevent condensation or freezing of water in the exhaust stream under conditions experienced during winter operation. For some operating conditions, sensible rotary heat exchangers that have no desiccant can also transfer small amounts of water between the two air streams by condensation/evaporation mechanisms. Holmberg (1989) used a numerical solution to predict condensation and frosting limits in rotary exchangers both with and without desiccant materials and his study establishes guidelines for prediction of condensation and frosting in regenerators.

None of the existing models is suitable for annual simulations of transient enthalpy exchanger performance in which the speed must be controlled to prevent condensation and freezing. A computationally simple model is developed in this study to provide performance estimates for commercially available rotary enthalpy and sensible heat exchangers as a function of matrix design and matrix properties. These properties include the two air inlet states, air flow rates and matrix rotation speed. The model incorporates experimental measurements of the adsorption isotherm of the matrix material together with other relevant properties to enable estimation of the annual (heating and cooling) energy savings which result from the use of rotary enthalpy and sensible heat exchangers in different climates. The annual performance estimates include the effects of operation at reduced speed in order to avoid condensation and freezing, using results from Holmberg (1989). The results of these annual performance estimates allow conclusions to be drawn regarding the applicability of enthalpy exchangers for heating and cooling in different climates.

MATRIX DESIGN AND DESICCANT PROPERTIES

This study is based on a commercially available enthalpy exchanger matrix (Carnes 1989) made of aluminum foil of thickness 0.025 mm (0.001 in) coated with a thin, uniform layer of polymer desiccant. The matrix design, shown schematically in Figure 1, is constructed by coiling smooth and corrugated aluminum sheets to produce small triangular flow passages through which the two air streams flow in opposite directions. The equivalent hydraulic diameter of the triangular passes is approximately 1.7 mm (0.07 in.) and the matrix has a length in flow direction of 0.2 m (8 in.). A typical wheel for commercial applications has a diameter of 1.23 m (4 ft) and is rotated at approximately 15 rpm.

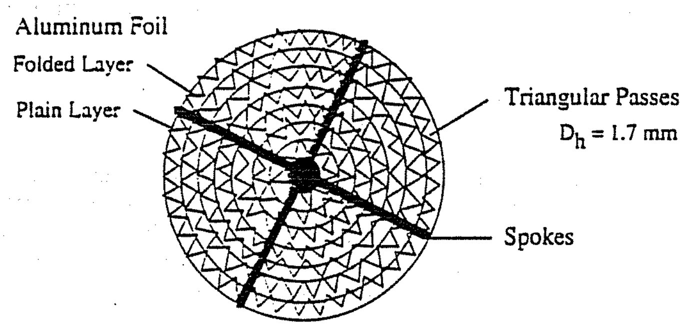

Figure 1. Matrix structure

The design parameters that govern the heat and mass-transfer mechanisms are the number of transfer units for heat transfer between one air stream and the matrix, defined as:

$$NTU_T = \frac{hA_s}{\dot{m}_j c_{p,f}} \tag{1}$$

and the number of transfer units for mass transfer:

$$NTU_w = \frac{h_w A_s}{\dot{m}_j} = \frac{NTU_T}{Le} \tag{2}$$

where the Lewis number (Le) is the dimensionless ratio of heat transfer to mass-transfer coefficients. For equal supply and exhaust air flow rates and equal wheel heat and mass-transfer areas, as experienced during normal operation, the overall number of transfer units from one air stream to the other is one-half of these values:

$$NTU_o = \frac{NTU}{2} \tag{3}$$

The wheel design and operation is such that airflow in the triangular channels is laminar at all operating conditions. The heat and mass-transfer boundary conditions are closer to constant heat flux than to constant wall temperature, and a constant value of Nusselt number of 3 is assumed (Kays and Crawford, 1980). This assumption results in a heat-transfer coefficient of $h = 46\ W/(m^2 \cdot K)$ ($8\ Btu/h \cdot ft^2 \cdot {}^\circ F$) and, for the commercial wheel, the number of transfer units for heat transfer between one air stream and the matrix becomes $NTU_T = 5$. The effect of axial conduction in the aluminum matrix is negligible for this matrix at this NTU according to the method of Shah (1988).

Kays and London(1984) provide charts relating heat exchanger effectiveness to NTU for rotary sensible heat exchangers. The information in the charts can be represented by the following empirical correlation.

$$\varepsilon = \varepsilon_{cf}\left(1 - \frac{1}{9\,(C_r^*)^{1.93}}\right) \tag{4}$$

where ε_{cf} is the effectiveness for direct counterflow heat exchangers. For equal flow rates and heat-transfer surface areas, the usual case for enthalpy exchange,

$$\varepsilon_{cf} = \frac{NTU_o}{NTU_o + 1} = \frac{NTU}{NTU + 2} \tag{5}$$

The carryover of fluid from one air stream to the other was estimated to be 3% of the total flow at typical enthalpy exchanger operating conditions (L = 0.20 m, rotation speed = 15 rpm, flow velocity = 1.5 m/s). Using the method provided by Shah (1988), this carryover rate results in approximately a 4% degradation in effectiveness. As shown below, the performance of the actual exchanger can be estimated quite closely without considering carryover.

For the desiccant, the effective Lewis number, which is the ratio of the overall mass-transfer resistance to that for heat transfer, was assumed equal to unity as suggested by Holmberg (1989). The Lewis number is essentially unity for convection only to the desiccant surface. The effective Lewis number could be significantly greater than unity if there is significant diffusion resistance within the desiccant that adds to the convective resistance. For the matrix of this study, the desiccant layer is uniform and very thin and the diffusion resistance is expected to be negligible compared to the convective resistance. The effect of Lewis number values greater than one on the long-term enthalpy exchanger performance is considered below.

In their design manual, the manufacturer of the water-based desiccant matrix (Carnes 1989) provides the effectiveness of their energy recovery wheels as a function of the face velocity of the wheel. The effectiveness values for heat and mass (humidity) transfer are equal, indicating a Lewis number of unity. Figure 2 shows the heat-transfer effectiveness for various wheelface velocity values as published in the manufacturer's catalog and a comparison with the effectiveness calculated using the effectiveness relation in Equation (4) as suggested by Klein et al. (1990). The humidity-transfer effectivenesses for Lewis numbers of two and four are also shown. The excellent agreement with the manufacturer's data indicates that the effects of carryover and axial conduction on effectiveness can be neglected and that the Lewis number can be taken as unity.

The properties that are required to determine regenerator performance are the equilibrium adsorption isotherms of the desiccant, the specific heat capacity, and the isosteric heat of adsorption. The isotherms were determined by measuring the mass of adsorbed water per mass of matrix (coated foil) for various temperature and relative humidity combinations. Samples of the foil coated with desiccant were first dried.

weighed, suspended over saturated salt-water solutions until equilibrium was reached, and then reweighed to determine the mass of water adsorbed (Stiesch 1994). The measured isotherms are shown in Figure 3. The maximum water uptake at a relative humidity of 100%, which is independent of temperature, is about 8.5% of the total matrix weight. Each symbol shown in Figure 3 represents the average of at least 10 repeated measurements. An experimental uncertainty analysis (Coleman and Steele 1989) was conducted. The maximum total experimental uncertainty is ±6% in the measured mass of adsorbed water per mass of matrix. Both bias and precision errors are included in this estimate. However the most significant part of this error resulted from replicated measurements under identical conditions. The Gaussian standard deviation was calculated for the replicates and the uncertainty was assumed to be two times as large as the standard deviation.

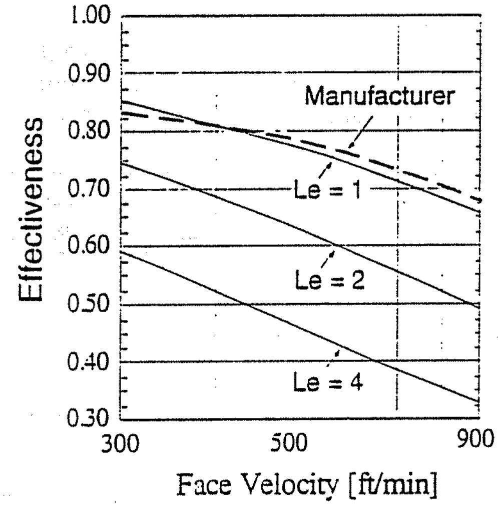

Figure 2. Comparison of manufacturer's effectiveness with calculated values for Le = 1, 2, and 4

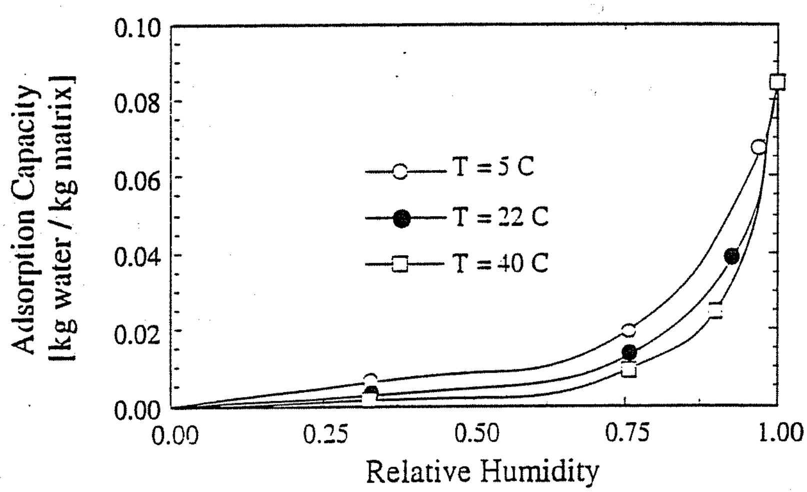

Figure 3. Adsorption Isotherms
(Symbols represent experimental measurements)

The desiccant isotherm shape in Figure 3 should be classified as a type III isotherm as defined by the International Union of Pure and Applied Chemistry (Sing 1982). The measured data were fit to the form of the Dubinin-Polstyanov equation (Dubinin 1975) for use in the computer program MOSHMX (Maclaine-cross 1974). The isotherms are represented by

$$W_m = 0.0385 \exp\left[-\left(\frac{A}{620}\right)^{0.5}\right] + 0.0460 \exp\left[-\left(\frac{A}{20}\right)^{1.5}\right] \tag{6}$$

where W_m is the adsorbed water per unit mass of dry matrix and A is the adsorption potential defined as:

$$A = RT \ln(p_s/p_v) \tag{7}$$

The adsorption potential includes the effects of both temperature and humidity and is a convenient method of representing desiccant isotherms (Dubinin 1975). Equation (6) is plotted in Figure 4 with error bars indicating the uncertainty in the experimental measurements of the adsorbed water to illustrate the accuracy of the fit.

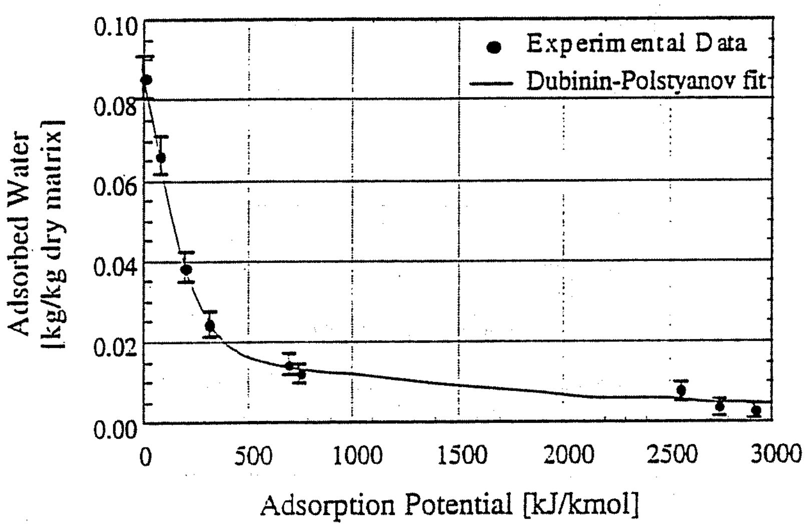

Figure 4. Adsorbed water versus adsorption potential
(Error bars indicate uncertainty in experimental measurements.)

By weighing several pairs of equally-sized, desiccant-coated and pure aluminum foil samples, it was determined that the coated foil consists of about 5% (by mass) polymer desiccant. Because the mass of desiccant is small, the specific heat capacity of the enthalpy exchanger matrix is assumed to equal that of aluminum: $c_{p,m} = 900 \text{ J/(kg·K)}$.

The isosteric heat of adsorption for the polymeric desiccant can be derived from an equation similar to the Clausius-Clapeyron equation for vaporization:

$$-\frac{i_s}{R} = \left(\frac{\partial \ln p_v}{\partial(1/T)}\right)_{W_m} \tag{8}$$

Using the isotherm measurements, the heat of sorption was found to be approximately 2530 kJ/kg (1095 Btu/lb) and essentially independent of temperature.

Contaminants such as volatile organic compounds (VOC) are often contributors to poor indoor air quality. If the desiccant adsorbs and desorbs these contaminants, they may be transferred from the contaminated exhaust air to the fresh supply air stream by the same mechanism that governs the exchange of water vapor between the two streams. Such a contaminant transfer would diminish the benefits of an exchanger in a ventilation system. The adsorption capacities of the polymer desiccant for low concentrations (10–50 ppm) of propane (C_3H_8) and toluene (C_7H_8) were measured using a gas chromatograph. Neither contaminant was adsorbed in an amount that could be detected. Similar, but less extensive, results were obtained with acetone. These results suggest that the adsorption of water on the polymer desiccant is caused by the polar character of the water molecules, rather than by van der Waal's interactions, and that the polymeric desiccant will not adsorb hydrocarbon contaminants. Contamination of supply air by sorption processes does not appear to be a problem for this polymeric desiccant.

MODELS FOR HEAT AND MASS TRANSFER IN ROTARY REGENERATORS

Figure 5 illustrates a rotary regenerator and the appropriate coordinate system and nomenclature for the regenerator models. The regenerator matrix rotates continuously between the supply air stream and the exhaust air stream in counterflow. The streams are physically separated by ducts in order to prevent mixing. The thermal analysis is based on the following conventional assumptions.

1. The state properties of both air streams are spatially uniform and constant with time at the inlet faces of the regenerator.
2. The thermodynamic properties of the air and the matrix are not affected by the small pressure drop, relative to the total pressure, in the axial direction of the matrix.
3. There is no mixing or carryover between the two air streams.
4. The fluid and matrix states are uniform in the radial direction at every axial position.
5. The temperature and moisture differences across the thin matrix surface are negligible.
6. Angular and axial heat conduction and vapor diffusion due to temperature and concentration gradients, respectively, are neglected.
7. The matrix is a homogeneous solid with constant matrix characteristics.
8. The convective heat and mass-transfer coefficients between the air streams and the matrix are constant with position and time throughout the system.
9. The regenerator operates adiabatically overall.

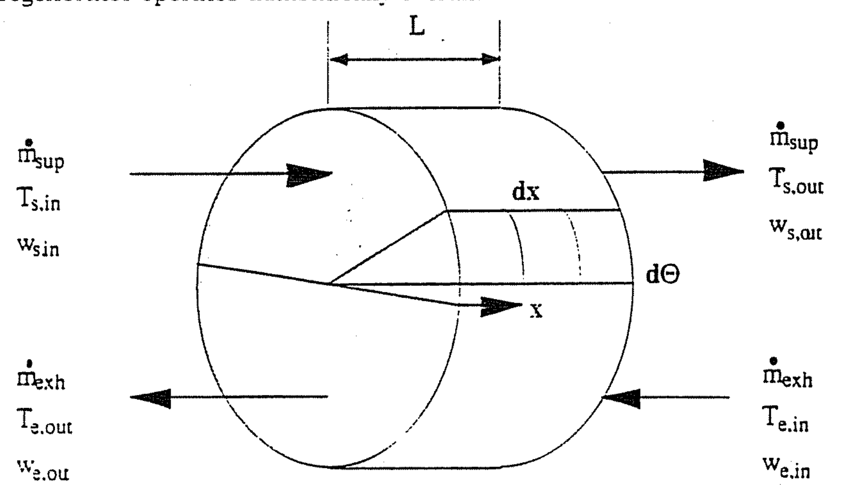

Figure 5. Regenerator Coordinate System and Nomenclature

The heat and mass-transfer processes are governed by mass and energy conservation. The driving forces for transfer are the differences in temperature and humidity ratio between the actual air state and a pseudo air state that is in equilibrium with the corresponding matrix state.

Mass:
$$\frac{\partial \omega_f}{\partial z} + \frac{M_f}{M_m}\frac{\partial \omega_f}{\partial \tau} + \frac{\partial W_m}{\partial \tau} = 0 \tag{9}$$

$$\frac{\partial W_m}{\partial \tau} = \mathrm{NTU}_\omega(\omega_f - \omega_m) \tag{10}$$

Energy:
$$\frac{\partial i_f}{\partial z} + \frac{M_f}{M_m}\frac{\partial i_f}{\partial \tau} + \frac{\partial I_m}{\partial \tau} = 0 \tag{11}$$

$$\frac{\partial I_m}{\partial \tau} = \mathrm{NTU}_T\frac{\partial i_f}{\partial T_f}(T_f - T_m) + \mathrm{NTU}_\omega(\omega_f - \omega_m)\, i_\omega \tag{12}$$

Equations (9) to (12) are coupled due to the thermodynamic state property relations, and an analytical solution does not exist. Numerical solutions to the equations may be obtained using the MOSHMX program (Maclaine-cross 1974). This program requires significant computational effort and is not suitable for use in transient regenerator simulations. However, it was employed here to obtain reference solutions that were used for developing a computationally simpler simulation model.

Klein et al. (1990) studied the performance of enthalpy exchangers and found that the maximum possible effectiveness for a counterflow direct type heat exchanger for both temperature and humidity can be expressed by a simple correlation based on the appropriate number of transfer units. The relation is the same as that for a direct-transfer counterflow exchanger with equal capacitance rates given in Equation (5).

The outlet state of the process stream of a heat and mass regenerator is a function of the regenerator rotation speed. At very slow rotation speeds, the outlet state approaches the conditions of the inlet process stream as both the thermal and mass-transfer wave fronts exit the regenerator. As the rotation speed is increased, a point is reached in which the thermal wave front reaches the regenerator outlet but the slower mass-transfer wave front does not. In this case, the regenerator operates as a dehumidifier. In order to operate at enthalpy exchanger conditions where the regenerator effectiveness is maximized, the matrix rotation speed has to be such that neither the thermal nor the mass-transfer wave front reaches the outlet of the regenerator. Klein et al. (1990) present a correlation for this minimum rotation speed as a function of the two air inlet states, the desiccant properties and the airflow rate. The minimum rotation speed generally ranges between 10 and 15 revolutions per minute for typical enthalpy wheels. Since this model predicts the maximum effectiveness only, it is useful only as long as there are no constraints that force operation at lower effectiveness. However, at very cold outdoor temperatures, condensation in the more humid exhaust stream would occur which could freeze and eventually block the flow channels in the matrix. To avoid this situation, the regenerator effectiveness has to be decreased by lowering the rotation speed to a point where the matrix temperature on the exhaust side is always above the dew point. This control strategy reduces regenerator performance and has to be taken into account when transient simulations over the heating season are performed. A new model that allows prediction of the regenerator effectiveness as a function of the matrix rotation speed is necessary.

Solutions for a range of rotation speeds were obtained with MOSHMX for various operating conditions and matrix sizes. Curve fits for the regenerator effectiveness were then developed for the polymer-coated aluminum matrix. There are three regenerator effectivenesses for temperature, humidity and enthalpy transfer, but only two of these parameters are independent. The effectivenesses for temperature and enthalpy were modeled since they can be approximated by simpler functions than the humidity effectiveness. The latter can have a negative value at the very low rotation speeds where dehumidification occurs. The temperature effectiveness is represented as:

$$\varepsilon_T = \frac{\mathrm{NTU}}{\mathrm{NTU}+2}\left(1-\exp\left[a_T\Gamma^2+b_T\Gamma\right]\right) \tag{13}$$

$$a_T = a_{T_1} + \frac{a_{T_2}}{\mathrm{NTU}^{a_{T_3}}}$$

$$a_{T_1} = 0.002259 - 1.376\times10^{-3}T_{amb} - 6.91\times10^{-6}T_{amb}^2$$

$$a_{T_2} = 0.09084 - 3.263\times10^{-4}T_{amb} - 7.4\times10^{-6}T_{amb}^2$$

$$a_{T_3} = 0.7388 - 0.01994T_{amb} - 3.829\times10^{-4}T_{amb}^2$$

$$b_T = b_{T_1} + \frac{b_{T_2}}{\mathrm{NTU}^{b_{T_3}}}$$

$$b_{T_1} = -1.007 + 0.0093T_{amb} + 2.778\times10^{-4}T_{amb}^2$$

$$b_{T_2} = -1.533 + 0.02287T_{amb} - 2.356\times10^{-4}T_{amb}^2$$

$$b_{T_3} = 1.111 - 2.667\times10^{-3}T_{amb} - 1.378\times10^{-4}T_{amb}^2$$

The enthalpy effectiveness is:

$$\varepsilon_i = \frac{\mathrm{NTU}}{\mathrm{NTU}+2}\left(1-\exp\left[a_i\Gamma^3+b_i\Gamma^2+c_i\Gamma\right]\right) \tag{14}$$

$$a_i = a_{i_1} + a_{i_2}\mathrm{NTU} + a_{i_3}\mathrm{NTU}^2$$

$$a_{i_1} = \begin{cases} 3.381\times10^{-3} - 9.679\times10^{-4}T_{amb} & \text{for } T_{amb}\le 0^\circ\mathrm{C} \\ 3.381\times10^{-3} - 4.127\times10^{-5}T_{amb} & \text{for } T_{amb}> 0^\circ\mathrm{C} \end{cases}$$

$$a_{i_2} = 5.088\times10^{-4} + 4.89\times10^{-6}T_{amb}$$

$$a_{i_3} = 5.298\times10^{-6} - 7.652\times10^{-7}T_{amb}$$

$$b_i = b_{i_1} + b_{i_2}\mathrm{NTU} + b_{i_3}\mathrm{NTU}^2$$

$$b_{i_1} = 6.237\times10^{-3} + 8.827\times10^{-3}T_{amb} - 6.042\times10^{-4}T_{amb}^2$$

$$b_{i_2} = -0.02123 + 1.323\times10^{-4}T_{amb}$$

$$b_{i_3} = 4.908\times10^{-4} + 6.46\times10^{-6}T_{amb}$$

$$c_i = c_{i_1} + \frac{c_{i_2}}{NTU^{0.8}}$$

$$c_{i_1} = -0.4087 + 0.00253\,T_{amb} + 3.34 \times 10^{-4}\,T_{amb}^2$$

$$c_{i_2} = -1.449 + 0.02337\,T_{amb} - 5.578 \times 10^{-4}\,T_{amb}^2$$

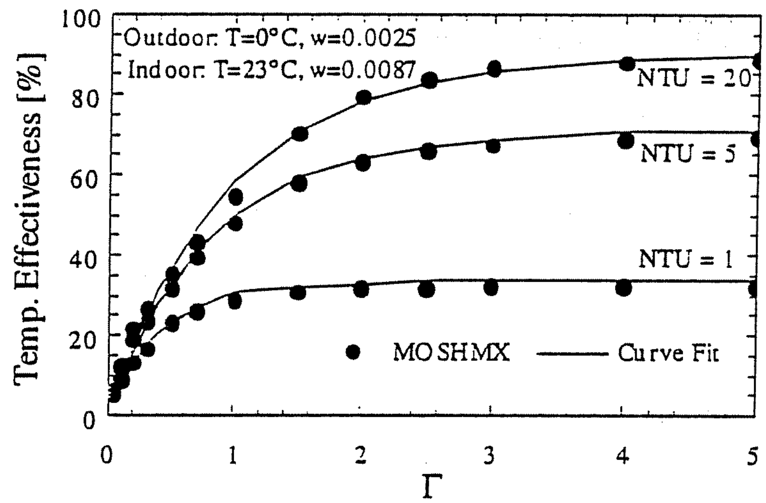

Figure 6a. Comparison of numerically-determined (MOSHMX) and curve-fit temperature effectiveness

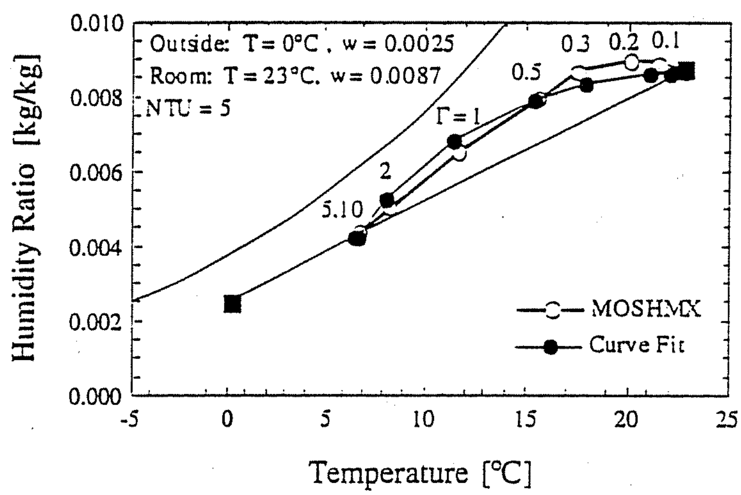

Figure 6b. Comparison of numerically-determined (MOSHMX) and curve-fit regenerator supply outlet states as a function of mass capacitance rate ratio

For high values of the dimensionless rotation speed Γ, these effectiveness equations provide the same results as provided by the model of Klein et al. (1990).

The curve fits were compared to the MOSHMX solution for operating conditions that differ from those used to generate the curve fit. Figure 6a presents the temperature effectiveness as a function of the dimensionless rotation speed Γ for various matrix sizes (number of transfer units). The regenerator supply outlet states, calculated numerically with the MOSHMX program and analytically with the curve fits, are plotted on a psychrometric chart in Figure 6b. The two plots show that the curve fits represent the actual enthalpy exchanger performance very well for the intermediate and fast rotation speeds ($\Gamma > 0.4$) typical for enthalpy exchanger operations in space-conditioning systems. However, the model is not capable of reproducing the dehumidification process that occurs at low rotation speeds ($\Gamma < 0.2$) and should not be used for annual performance simulations of rotary dehumidifiers where rotation speed is controlled below this level. The general form given in Figure 6a is expected to apply to other desiccants.

ANNUAL PERFORMANCE SIMULATIONS AND ECONOMIC ANALYSIS

The model described above was coded for the modular transient simulation program TRNSYS (Klein et al. 1994). Annual performance simulations were performed for the enthalpy exchanger previously specified and for a comparable sensible heat exchanger operating as part of a ventilation system for a 200-person office building with a total ventilation airflow rate of 4000 cfm (1.9 m^3/s). The sensible heat exchanger matrix consists of the same, uncoated aluminum foil that is the base material in the enthalpy exchanger and has the same design shown schematically in Figure 1. Three locations in the United States, each representing a different climate, were investigated in order to determine the influence of the weather conditions.

In the performance simulations for the cold and intermediate climates, represented by Madison, WI and Washington, DC, respectively, the relative indoor humidities were fixed at 30% during the heating season and 50% during the cooling season. For the warm and humid climate in Houston, TX, a constant value of 50% was selected for both seasons. The indoor temperature was set at 23°C for all cases.

The office ventilation system is assumed to be operated during the day and turned off every night from 9 P.M. to 6 A.M., which results in an operating time of 5475 hours per year. During operation, the fans need to overcome the increased pressure drop caused by matrix. The regenerator manufacturer states that this pressure drop at the assumed flow rate is 250 Pa, and this value is used in the analysis. The extra energy required to maintain the constant ventilation rate through either regenerator or heat exchanger amounts to 5200 kWh per year.

Both exchanger types are controlled such that no excess water (condensate that cannot be evaporated back into the supply stream) accumulates on the matrix. When temperatures are below 0°C, water could freeze and block the matrix. Even above freezing conditions, excess water should be avoided because it could accumulate in the ducting system and either leak or cause undesirable mold growth. A strategy to prevent excess water from collecting could be to turn off the regenerator. However, it is preferable to reduce the regenerator speed to prevent icing and to allow continuous operation, although this involves lowering the effectiveness. This operating strategy has the added advantage of providing some heat recovery at design heating conditions which allows use of heating equipment with smaller capacity.

According to Holmberg (1989), "excess water" occurs in enthalpy exchangers with a Lewis number of one when a straight line connecting the two air inlet states on a psychrometric chart crosses the saturation curve. For the room conditions as assumed, this situation occurs when the outdoor ambient state is in the shaded region in Figure 7. For these conditions, the rotation speed must be lowered until the exhaust outlet is above the saturation curve. For example, when the ambient conditions are in the shaded region, the effectiveness must be lowered so that the exhaust outlet state is 4'. The process is as shown from state 3 to state 4'.

For sensible heat exchangers, Holmberg experimentally determined that "excess water" (i.e., condensate or ice formation) will occur when a straight line connecting the supply inlet state (1) and a state that exceeds the room exhaust dew point temperature by 4°C (state 5) intersects the saturation curve, as shown on Figure 8. To avoid condensation in sensible exchangers, the effectiveness is controlled by reducing the rotation speed such that the exhaust outlet is at the dew point of the exhaust inlet (state 4').

The economic benefits of a rotary regenerator depend on two parameters: the increase in initial cost of the complete space-conditioning system and, secondly, on the annual operating savings due to energy recovery from the exhaust. Heat recovery equipment decreases the required installed capacity of the heating and cooling equipment. For this study average incremental costs for the heating system capacity of \$30/kW (\$880/10^6 Btu/h) and of \$150/kW (\$530/ton) for the cooling system capacity were assumed. The allowable reduction in capacity was determined by evaluating the regenerator energy recovery for the design weather conditions (ASHRAE 1993). The initial costs for an enthalpy exchanger and sensible heat exchanger that were sized for the building used in this study, including variable speed drive and controls, were obtained from the manufacturer of the enthalpy exchanger and are \$8000 and \$7200, respectively.

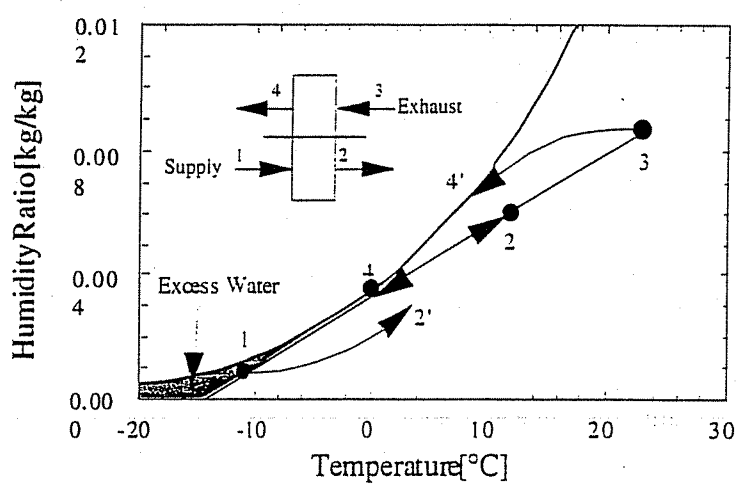

Figure 7. Excess water onset for enthalpy exchangers

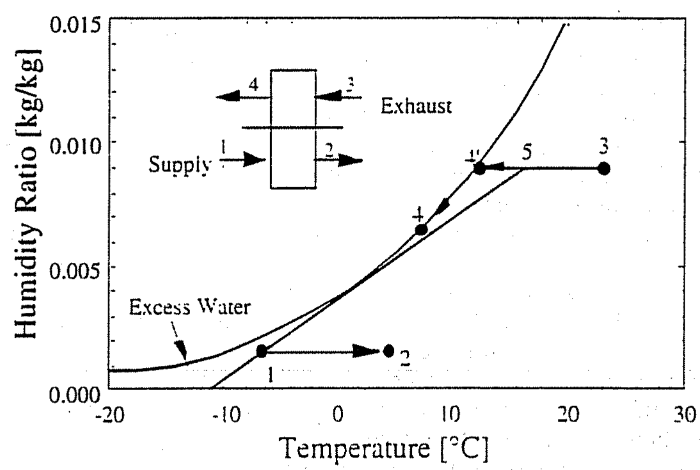

Figure 8. Excess water onset for sensible heat exchangers

A COP of 3 was assumed for the conventional cooling system, with electricity costs of \$0.08/kWh. An efficiency of the heating equipment of 0.85 was assumed, which yielded operating costs of \$0.021/kWh in the heating mode at a natural gas price of \$5.20/10^6 Btu. The added fan power costs for the exchangers is \$400/year. The net incremental installation cost and the annual operating savings were combined to estimate the regenerator life cycle savings over a period of 15 years. The inflation rate for electricity and natural gas was assumed to be 5% and all future cash flows were discounted to a present worth, with an 8% interest rate (rate of return on investment).

The results for all simulations and the economic calculations are summarized in Tables 1 to 3 for the three locations. It can be seen that for both regenerator types, more exhaust energy is recovered for both heating and cooling applications as the difference between indoor and outdoor conditions increases. The advantage of the enthalpy exchanger over the sensible heat exchanger during the heating season is greater for colder climates due to the ability of the enthalpy exchanger to operate in colder temperatures at high efficiency without excess water accumulation. The integrated annual energy recovery of an enthalpy exchanger exceeds the recovery of a sensible heat exchanger by 10 to 20% in the heating mode. For cooling, the relative advantage of the enthalpy exchanger becomes much greater since the humidity ratio differences between outdoors and indoors are greater than in the winter. In the cooling season, the enthalpy exchanger performance is 3.5 to 4 times better than the sensible heat exchanger performance.

Table 1. Annual Operating Savings

Location	Regenerator	Recovery in Cooling Season (kWh)	Recovery in Heating Season (kWh)	Annual Savings (\$)
Madison, WI	HX	4,093	126,500	2,363
	EX	13,430	148,700	3,072
Washington, DC	HX	7,751	91,300	1,719
	EX	27,350	99,900	2,409
Houston, TX	HX	25,070	31,400	911
	EX	108,800	37,900	3,225

Table 2. Installation Costs

Location	Regenerator	Regenerator Cost (\$)	Cooling Cap. Reduction (kW)	Heating Cap. Reduction (kW)	Net Installation Costs (\$)
Madison, WI	HX	\$7,200	17	42	3,390
	EX	\$8,000	37	70	350
Washington, DC	HX	\$7,200	18	42	3,240
	EX	\$8,000	43	69	−520
Houston, TX	HX	\$7,200	22	26	3,120
	EX	\$8,000	50	64	−1,420

Table 3. Life Cycle Savings

Location	Regenerator	Net Installation Costs (\$)	Present Worth Oper. Savings (\$)	Present Worth LCS (\$)
Madison, WI	HX	3,390	27,146	23,756
	EX	350	35,291	34,941
Washington, DC	HX	3,240	19,748	16,508
	EX	−520	27,674	28,194
Houston, TX	HX	3,120	10,465	7,345
	EX	−1,420	37,048	38,468

The present worth of the life cycle savings are greater than zero for both regenerator types in all locations. However, the sensible heat exchanger in the warm and humid Houston climate results in savings that are significantly smaller than all other values, indicating that a sensible exchanger would probably be a relatively poor investment under similar conditions. Moreover, the use of a regenerator requires that the exhaust air be ducted to pass through the matrix. This might cause significant expenses for some building designs that are not included in the foregoing calculations. In these situations, the additional duct and installation costs have to be weighed against the life cycle savings. However, there can be a net decrease in equipment costs when using an enthalpy exchanger in intermediate and warm climates, since smaller mechanical cooling equipment with less capacity is adequate.

EFFECT OF VARIOUS ISOTHERM SHAPES AND LEWIS NUMBERS

Desiccant isotherm shape has a potential influence on the enthalpy exchanger performance. The effect of shape was determined by comparing the performance of an aluminum matrix coated with the type III polymer desiccant to that of a hypothetical silica gel (type I isotherm) matrix of the same design. Lewis number was assumed to be unity. The calculations were performed with MOSHMX. The maximum possible temperature and humidity effectiveness is a function of the matrix design (NTU) only and does not depend on the desiccant type. However, the minimum rotation speed required to achieve optimum effectiveness varies for the different matrix materials. Calculations using Klein's empirical correlations for the minimum rotation speed show that the difference for the two desiccants at all operating conditions is within a few revolutions per minute. There is no significant advantage of one material over the other as long as the regenerator can be operated at maximum effectiveness.

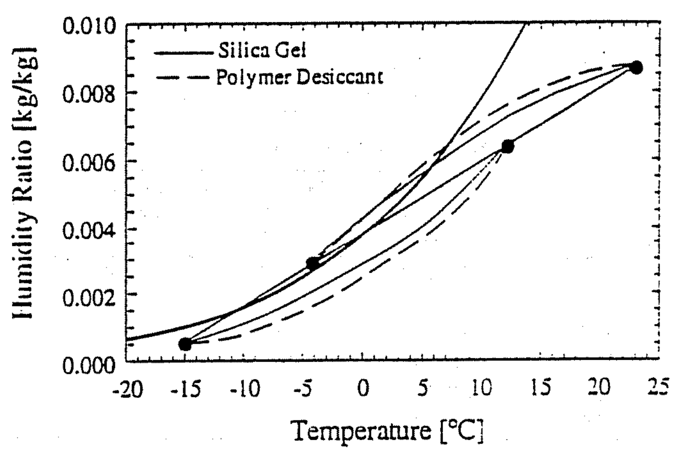

Figure 9. Comparison between polymer (type III) and silica gel (type I) matrices on a psychrometric chart

The air outlet states for effectiveness less than the maximum, as obtained by decreasing rotation speed, depend somewhat on the isotherm shape. Thus, if there is a risk of excess water and the rotation speed has to be decreased in order to avoid condensation, there might be a difference between the desiccant materials. The closer the locus of outlet states to the straight line connecting the two inlet states on psychrometric coordinates, the better the desiccant material since operation at higher effectiveness is possible without condensate forming on the matrix. Figure 9 shows the locus of the exhaust outlet curves for the polymer and the silica gel matrices for an outdoor temperature of −15°C, typical for a heating application in a cold winter climate. The silica gel matrix intersects the saturation curve at a slightly lower temperature under these conditions, and it would provide more energy recovery, but the outlet curves are very close over the entire range of rotation speeds and the advantage of the silica gel type I isotherm is small.

The influence of the desiccant Lewis number on the energy recovery of an enthalpy exchanger was estimated for three different outdoor conditions. The instantaneous energy recovery for enthalpy exchangers with Lewis numbers of 1, 2, and 4 and also for a sensible heat exchanger (Lewis number = ∞) were calculated using MOSHMX for one cooling application (T = 31°C, w = 0.0145) and two heating applications (T = −15°C, w = 0.0008 and T = 0°C, w = 0.0030). Lewis numbers greater than unity represent the effect of mass-transfer diffusional resistance in the desiccant. The two heating conditions were chosen such that one represents a case where all regenerator types can safely be operated at maximum effectiveness while the other one requires a decreased effectiveness for the sensible heat exchanger and for the enthalpy exchanger with a Lewis number of 4. The results for the total energy recovery (sensible plus latent energy) are summarized in Figure 10. The results show that the absolute differences between the energy recoveries differ significantly for the three cases. However, a general trend can be observed: the recovery of the enthalpy exchanger with a Lewis number of 4 is always about halfway between the recoveries of a sensible heat exchanger and an enthalpy exchanger with an ideal Lewis number of unity. Thus there is an effect of Lewis number, but exchangers with large values of Lewis number still produce significant savings over sensible exchangers.

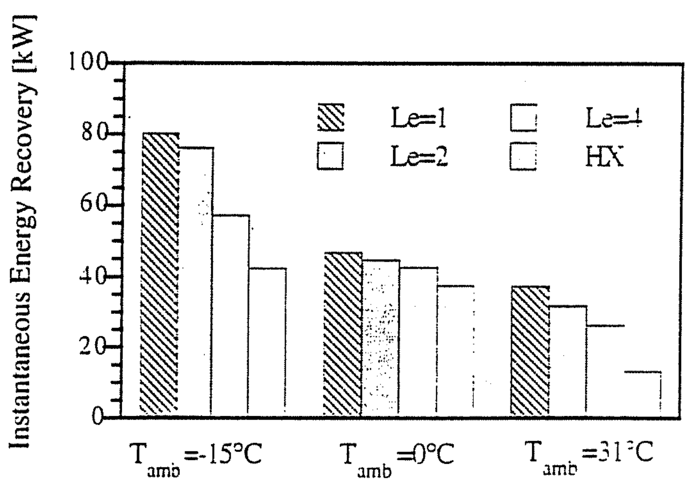

Figure 10. Energy Recovery as a Function of the Lewis Number

CONCLUSIONS

The performance of rotary enthalpy exchangers relies on the presence of a desiccant to adsorb and transfer moisture, as well as heat, between two air streams. The desiccant is characterized by the isotherm, which gives the water content as a function of relative humidity. Measurements on a commercially available polymeric desiccant showed that the isotherm is of Type III, which is termed unfavorable for dehumidification operation. However, for enthalpy exchangers, the performance does not significantly depend on the isotherm shape. Further, the polymeric desiccant used in this study did not adsorb measurable quantities of three common contaminants.

Operation as an enthalpy exchanger in winter conditions produces the possibility of condensation and freezing in the matrix. Lowered effectiveness, which is achieved by slower rotational speed, prevents condensation. Sensible heat exchangers must have significantly lower effectiveness than enthalpy exchangers at the same conditions to prevent condensation. The lower effectiveness appreciably reduces the energy recovery possible by comparison with enthalpy exchangers.

Curve fits for the moisture and temperature effectiveness were obtained to represent the performance over a wide range of operating conditions. Simulations of HVAC systems in commercial buildings with rotary enthalpy and heat exchangers showed that significant energy savings were possible. In addition, enthalpy exchangers reduced the required maximum capacity of heating and cooling equipment that may result in a reduction of first costs. Life cycle savings based on a 15 year equipment life were also positive in all climates. In general, enthalpy exchangers recover significantly more energy than sensible exchangers during cooling due to the moisture transfer. During the heating season, the difference in energy recovery between the two exchangers is not as significant, except in very cold locations in which sensible exchangers must operate at reduced effectiveness to prevent ice accumulation. Overall, enthalpy exchangers have comparable and largely positive life cycle savings in all three climates investigated.

NOMENCLATURE

A	absorption potential, kJ/kmol, [Equation (7)]
A_s	surface area, m^2
$c_{p,f}$	specific heat of air, kJ/(kg·K)
$c_{p,m}$	specific heat of the matrix, kJ/(kg·K)
C_r^*	ratio of matrix capacitance rate to the minimum fluid capacitance rate
h	heat-transfer coefficient between the matrix surface and air, W/(m^2·K)
h_w	mass-transfer coefficient between the matrix surface and air, kg/(m^2·s)
i_f	specific enthalpy of fluid stream, kJ/kg
i_m	specific enthalpy of matrix, kJ/kg
i_s	isosteric heat of adsorption, kJ/kg
i_w	specific enthalpy of water vapor, kJ/kg
L	length of matrix in flow direction, m
Le	Lewis number = NTU_T/NTU_w
$\dot{m}_j$	mass flow rate of air in period j ($\dot{m}_1 = \dot{m}_2$ in this study), kg/s
M_f	fluid mass entrained in flow passages, kg
M_m	mass of dry matrix, kg
NTU_o	overall number of transfer units
NTU_T	number of transfer units for heat-transfer
NTU_w	number of transfer units for mass transfer
p_s	saturation vapor pressure of water at a specified temperature, kPa
p_v	actual vapor pressure of water, kPa
R	universal gas constant = 8.314 kJ/(kmol·K)
t	time, s
t_j	rotation time for period j, s
T	temperature, K
T_{amb}	ambient temperature, K
T_f	temperature of air stream, K
T_m	temperature of matrix surface, K
w_f	humidity ratio of air stream
w_m	humidity ratio of air in equilibrium with the matrix
W_m	absorbed water per unit mass of dry matrix, kg/kg
x	position in matrix in flow direction, m
z	dimensionless flow coordinate = x/L
ε_i	heat exchanger enthalpy effectiveness
ε_{cf}	effectiveness for direct counterflow heat exchangers
ε_T	heat exchanger temperature effectiveness
Γ_j	mass capacitance rate ratio in period $j = M_m/(t_j\dot{m}_j)$; $\Gamma_1 = \Gamma_2 = \Gamma$ in this study
τ	dimensionless time = $t/(t_j\Gamma_j)$

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