AIR-TO-AIR HEAT RECOVERY DEVICES FOR SMALL BUILDINGS

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ABSTRACT

With the escalation of fuel costs, many people are turning to tighter, better insulated buildings as a means of achieving energy conservation. This is especially true in northern climates, where heating seasons are long and severe. Installing efficient well sealed vapor barriers and weather stripping and caulking around doors and windows reduces cold air infiltration but can lead to damaging moisture buildup, as well as unpleasant and even unhealthy accumulations of odors and gases. To provide the necessary ventilation air to maintain air quality in homes while holding down energy costs, air-to-air heat exchangers have been proposed for residential and other simple structures normally not served by an active or forced ventilation system.

Four basic types of air-to-air heat exchangers are suited for small scale use: rotary, coil-loop, heat pipe, and plate. The operating principles of each of these units are presented and their individual advantages and disadvantages are discussed. A test program has been initiated to evaluate the performance of a few commercial units as well as several units designed and/or built at the University of Alaska. Preliminary results from several of these tests are presented along with a critique on their design.

INTRODUCTION

Increased concern for energy conservation due to rapidly rising energy costs has focused attention on building design and operation. In years gone by, architectural features and construction cost considerations dominated

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building design. Consequently, buildings were designed without much concern over their annual energy budget. Energy conservation in design has become a more important factor now and as a result buildings are being built with more insulation to reduce transmission heat losses and tighter construction to reduce infiltration heat losses. A need to mechanically ventilate homes and small buildings has arisen due to these improvements in insulation and building construction standards which have produced tighter structures. As heating costs have risen, it has become economic to seal a structure to the point where building air leakage may only permit 0.1 air change per hour or less. These low values of air exchange are far below the 0.5 to 1.0 air exchange rates typical of older buildings. The use of mechanical ventilation in these newer structures is a must to ensure that humidity, odor, and other contaminants to do not build up to harmful levels.

Indoor pollutants which have been found in, structures, Hollowell, et al. [1], include:

1. Formaldehyde - a bonding agent used in plywood and particle board as well as in foam insulation and foam padding in furniture.

2. Nitrogen dioxide and carbon monoxide - fumes from gas cook stoves and portable unvented heaters.

3. Soot particles and benzopyrene - cigarette, pipe and cigar smoke.

4. Radon - a radioactive gas produced from soil and rock based building materials as well as water taken directly from underground wells. This noble gas is a daughter product of radium-226 and appears as part of the uranium-238 decay chain.

In addition to these pollutants, high humidity levels often occur in tightly sealed buildings. The sources of this moisture are breathing, bathing, cooking and washing. High humidities can lead to degradation of building materials due to condensation and/or frost formation in cold climate regions. Water and ice formation on windows and window sills or frost accumulation in insulation due to a faulty vapor barrier are more common in the well sealed structures due to the higher humidities.

Several solutions to the indoor pollution, are possible, Roseme, et al. [2], which include:

1. Installation of a mechanical ventilation system incorporating an air-to-air heat exchanger to increase outdoor ventilation air in order to dilute contaminates and also transfer thermal energy from the exhaust airstream to the fresh outdoor airstream.

2. Selection of building materials in new construction which do not produce potentially harmful gases.

3. Installation of a filtering system to remove specific pollutants.

4. Installation of sealing or venting devices at the time of construction to eliminate the harmful contaminates at their source.

The focus of this paper is on the first measure mentioned. It appears on the surface that the mechanical ventilation system incorporating an air-to-air heat exchanger is also the most attractive economically.

MOISTURE CONTROL

It is well known that operating buildings at high indoor relative humidities in northern climates during winter can lead to a whole host of moisture related problems. Ice and frost accumulation on windows and their sills, on doors and their frames and jams, and electrical outlets are just a few. However, with the present trend toward very tight construction, it is ensured that these problems will occur if the structure is not dehumidified. This is usually accomplished by mechanical ventilation, i.e. bringing in cold dry outdoor air. If cold air is brought in, then warm air must be exhausted from the building. Without recovering some of the thermal energy from the warm airstream, nothing in the way of energy conservation has been accomplished by specifying and paying for a tightly constructed building.

Further insight into this problem can be gained by performing a few simple calculations. If a 2,000 sq. ft. single story structure is to be maintained at 70°F and 20% relative humidity at an outdoor temperature of 0° or colder, then 3.75 pounds of water per hour must be be added in the structure, if the building air leakage rate is equal to 1.0 air changes per hour. However, if the air changes per hour are reduced to one tenth, characteristic of a tight building, then only .38 pounds of water per hour are required. A single person produces approximately .25 pounds of water per hour breathing and perspiring while performing light work. Based on this, having more than two people in this tightly constructed building will lead to elevated humidities and the onset of moisture problems during cold weather. (An equilibrium relative humidity of about 40% would result in this building with three people. However, it will take quite a while to reach this condition due to the moisture absorpence of building materials. Furthermore, the windows or other cold surfaces may act as dehumidifiers before the relative humidity ever reaches 40%). In most situations, other necessary life functions such as cooking and washing serve as additional sources of water vapor.

HEAT RECOVERY DEVICES

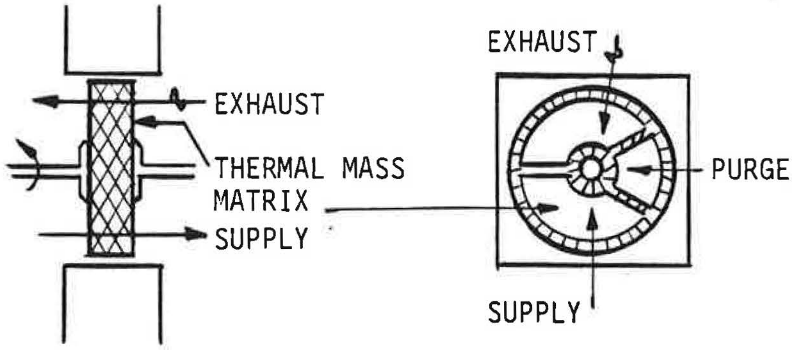
There are at least four different types of heat recovery devices that could easily be adapted to a mechanical ventilation system for small structures. These four types are classified as: rotary, coil-loop run-around, heat pipe and plate. For commercial applications, air-to-air heat exchangers of each of these types have been available for many years. However, it has been only recently that small units appropriate to the small building market have become available (a list of manufacturers and their addresses are given in Appendix I). For a small building, the mechanical ventilation system would use small fans; one to exhaust the indoor air and the other to bring in the fresh outdoor air, with both airstreams ducted through the air-to-air heat exchanger.

The four basic types of air-to-air heat exchangers, Sauer, et al. [3], are shown in Fig. 1, with a description of their individual features. The following paragraphs provide a more in-depth description of each type.

a. Rotary Heat Exchanger This is a porous wheel through which the exhaust and supply airstreams flow. As the cylinder rotates it is alternately heated and cooled by the airstreams. If latent heat transfer is also desired the wheel material is treated with a hygroscopic substance. Some cross contamination can occur between the two airstreams which can be almost entirely eliminated by including a purge section. Defrosting of the wheel can be accomplished by rotational speed control. Supply and exhaust air ducting must be brought together at the wheel.

b. Coil-Loop Run-Around Heat Exchanger This system uses a standard liquid-to-air finned-tube coil with a pump to circulate a water-glycol mixture. Sensible heat only is transferred between the airstreams by the circulated liquid. Frost build-up in the coil can be controlled by intermittent operation or a three way control valve using recirculation. No cross contamination between airstreams occur and the supply and exhaust airstreams can be widely separated. Ease of installation, simplicity, and low maintenance have made this system attractive.

SCHEMATIC	FIG. 1	FEATURES



ROTARY

SENSIBLE AND LATENT HEAT RECOVERY.

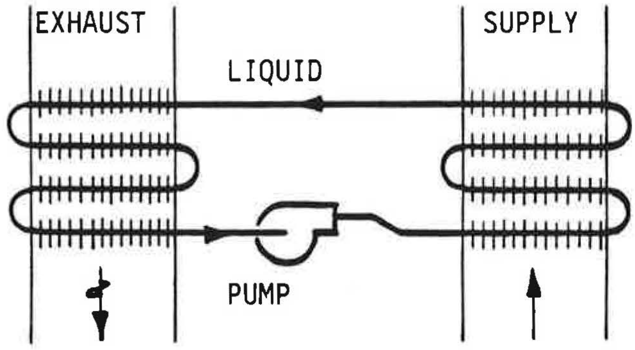
MINIMAL CROSS CONTAMINATION

PURGE SECTION REQUIRED

MECHANICAL DRIVER REQUIRED

FROSTING LIMITATIONS

CLOSE PROXIMITY OF SUPPLY AND EXHAUST.



COIL LOOP RUNAROUND

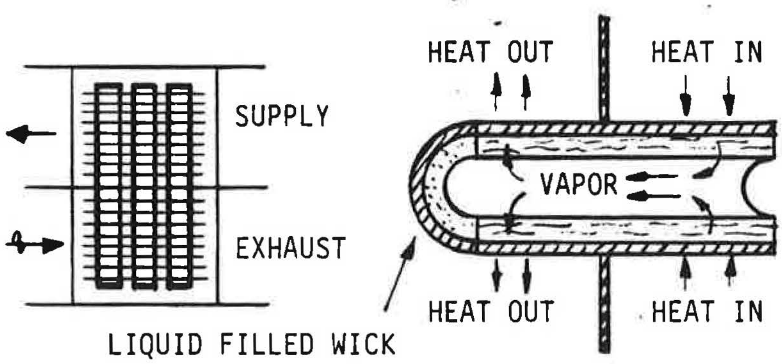
SENSIBLE HEAT RECOVERY

NO CROSS CONTAMINATION

PUMP REQUIRED

FROST CONTROL AVAILABLE

FLEXIBILITY OF SUPPLY AND EXHAUST.



HEAT PIPE

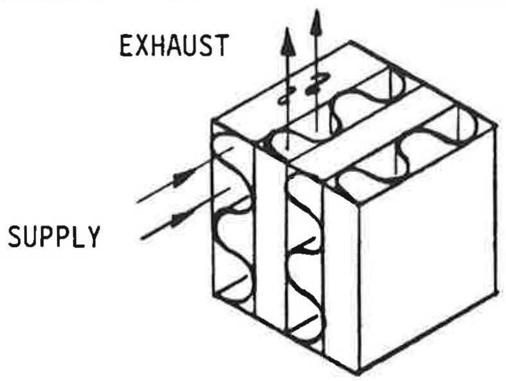
SENSIBLE HEAT RECOVERY

NO CROSS CONTAMINATION

NO PUMP OR DRIVES REQUIRED

FROST CONTROL AVAILABLE

CLOSE PROXIMITY OF SUPPLY AND EXHAUST.



PLATE

SENSIBLE HEAT RECOVERY

NO CROSS CONTAMINATION

NO PUMP OR DRIVES REQUIRED

FROST CONTROL AVAILABLE

CLOSE PROXIMITY OF SUPPLY AND EXHAUST.

c. Heat Pipe Heat Exchanger This system is composed of a set of finned tubes. The supply and exhaust airstreams are separated by a baffle and the flow is normal to the heat tubes in a counterflow fashion. The tubes, heat pipes, contain a working fluid such as freon, carbon dioxide, or ammonia which boils or vaporizes on the hot side and condenses on the cold side of the exchanger. The vapor travels from the warm to the cold end of the tube through its core because of the pressure difference created between the evaporating and condensing ends of the pipe. The liquid returns to the evaporation end by capillary flow in a wick material placed inside of the tube. The heat transfer rate through the tube can be controlled by the tilt of the tube. Placing the warm end down provides a gravity assist to returning the condensate thereby increasing the heat transfer rate. The heat pipe unit only recovers sensible heat. Frost and temperature control can be performed by tilting the tubes or using by-pass air. These techniques, however, are usually used in large commercial units. In a small structure, intermittent operation would be the most economical approach.

d. Plate Type Heat Exchanger This system contains multiple plates separating the supply and exhaust air streams which pass through the exchanger in either a cross-flow or counterflow arrangement. Metal, plastic, or paper have been used as the transfer surface. Sensible and latent heat transfers are possible depending on plate material, however, short circuiting can occur between supply and exhaust air inlets and outlets with improper design.

HEAT RECOVERY DEVICE PERFORMANCE

Air-to-air heat exchangers used in heat recovery applications have historically been rated based on a term called efficiency. However, when the mass flow rates in the exhaust and supply air sides are not equal, the efficiency measure can lead to erroneous results. This efficiency or heat recovery factor is defined as:

$$\eta = \frac{(X_1 - X_2)}{(X_1 - X_3)} \qquad (1)$$

where X represents the dry-bulb temperature, enthalpy or humidity ratio of the entering and leaving airstreams as shown in Fig. 2. The term η is then the sensible, total or latent efficiency, respectively.

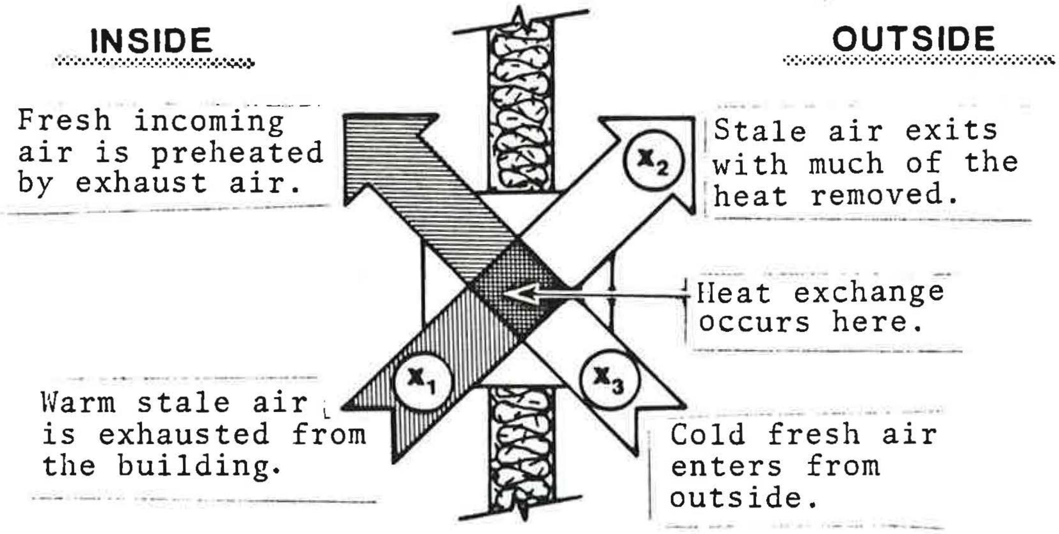


FIG. 2 Schematic Diagram and Flow Stream Nomenclature For An Air-To-Air Heat Exchanger.

A more accurate description of the heat exchangers performance is the effectiveness which is defined as:

$$\varepsilon = \frac{\text{actual heat transfer}}{\text{theoretical maximum heat transfer}} = \frac{m_s\ (X_1 - X_2)}{m_{min}\ (X_1 - X_3)} \qquad (2)$$

where ε is the sensible, latent, or total effectiveness, m_s and m_e the supply and exhaust air mass flow-rates, and m_{min} the minimum of m_s or m_e. Effectiveness is defined as the ratio of actual heat transfer to the thermodynamic maximum heat transfer that would occur in a counter flow heat exchanger of infinite area. The effectiveness definition reduces to equation (1) when the mass flow-rates are equal on both sides of the heat exchanger such that the effectiveness and efficiency are identical.

Fig. 3 is a psychrometric chart showing the "paths" of the supply and exhaust airstreams as they pass through the air-to-air heat exchanger. In all sensible heat recovery units (no moisture is transferred) the supply airstream will not change in moisture content as it passes through the exchanger. This process is shown as line a-b. Notice that heating of the cold supply air results in a drop in its relative humidity but no change in its moisture content. The change in energy per unit mass of this airstream is reflected by its change in enthalpy, (h_b-h_a). The exhaust airstream follows the path shown as e-f-g. Sensible cooling, without change in moisture content, occurs from point e to f. At point f the exhaust airstream has been cooled to its dew point temperature becoming saturated with water. Further cooling results in condensation or frosting on the exchanger surfaces until point g is reached. The change in energy per unit mass of the exhaust airstream is (h_e-h_g). The heat transferred to the supply airstream must be equal to the heat transferred from the exhaust airstream or:

$$Q = m_s\,(h_b-h_a) = m_e\,(h_e-h_g)$$

The thermodynamic maximum transferable heat is the change in enthalpy occurring between points a and e. Therefore, the total effectiveness of the exchanger by equation (2) is:

$$\varepsilon = \frac{m_s(h_b-h_a)}{m_{min}\,(h_e-h_a)} = \frac{m_e\,(h_e-h_g)}{m_{min}\,(h_e-h_a)} \qquad (3)$$

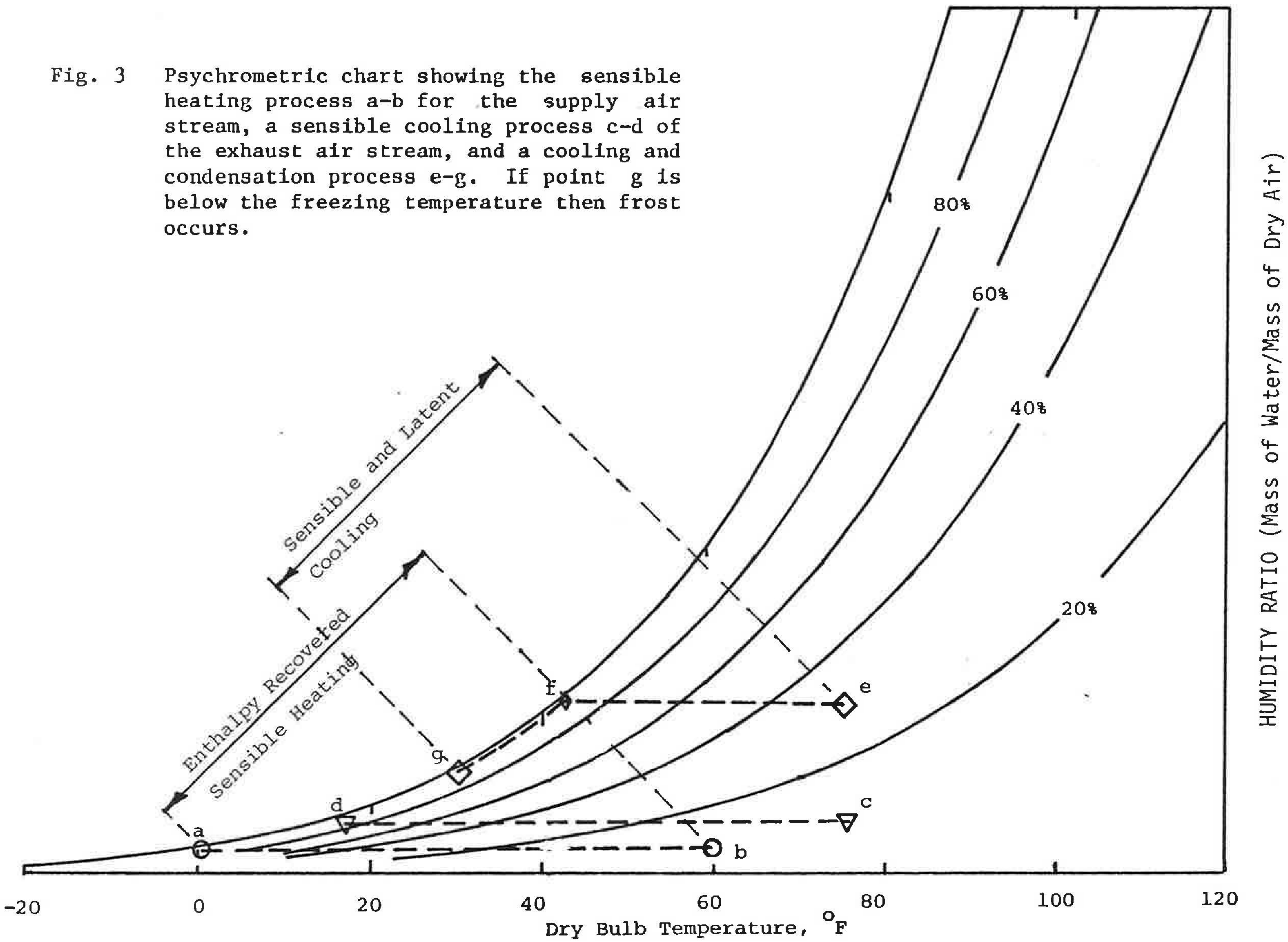


Fig. 3 Psychrometric chart showing the sensible heating process a-b for the supply air stream, a sensible cooling process c-d of the exhaust air stream, and a cooling and condensation process e-g. If point g is below the freezing temperature then frost occurs.

Sensible heat recovery devices do not permit the transfer of moisture between airstreams. Therefore, this device cannot be used to heat and humidify supply air during winter operation. If the surfaces of the exchanger drop below the dew point temperature of the exhaust airstream, then the sensible device will transfer the heat of condensation (and freezing in some cases) to the supply airstream, but no transfer of condensate or moisture will take place. This fact leads to another definition of effectiveness which is the ratio of actual heat transferred to the maximum transferable sensible heat.

$$\varepsilon = \frac{m_s \, (h_b - h_a)}{m_{min} \, (h_e - h_a)} = \frac{m_e \, (h_e - h_g)}{m_{min} \, (h_e - h_a)} \qquad (4)$$

Whenever effectiveness information is stated, care must be excerised in working with this data (high total effectivenesses do not necessarily mean high sensible effectivenesses and vice versa). The four types of heat exchangers discussed in an earlier section all exhibit similar performance characteristics. However, quantitative results will vary between types as well as between units of the same type. Fig. 4 shows the variations in effectiveness that would occur when the flow rates of the supply and exhaust airstreams are unequal. Also evident is the fact that reducing flow rates increases effectiveness.

The pressure drop and effectiveness variations as a function of coil face velocity are shown in Fig. 5. Increasing the face velocity leads to reduced effectiveness and increased pressure drop. The benefits of low velocity flow on heat recovery effectivenss and operating pressure drop are evident. However, this increased performance and reduced operating cost is a result of an increased capital investment for a larger exchanger.

POTENTIAL ENERGY SAVINGS

The potential energy savings produced by an air-to-air heat recovery device depends on numerous factors. The most obvious are the duration and severity of the heating season, length of time of operation, quantity

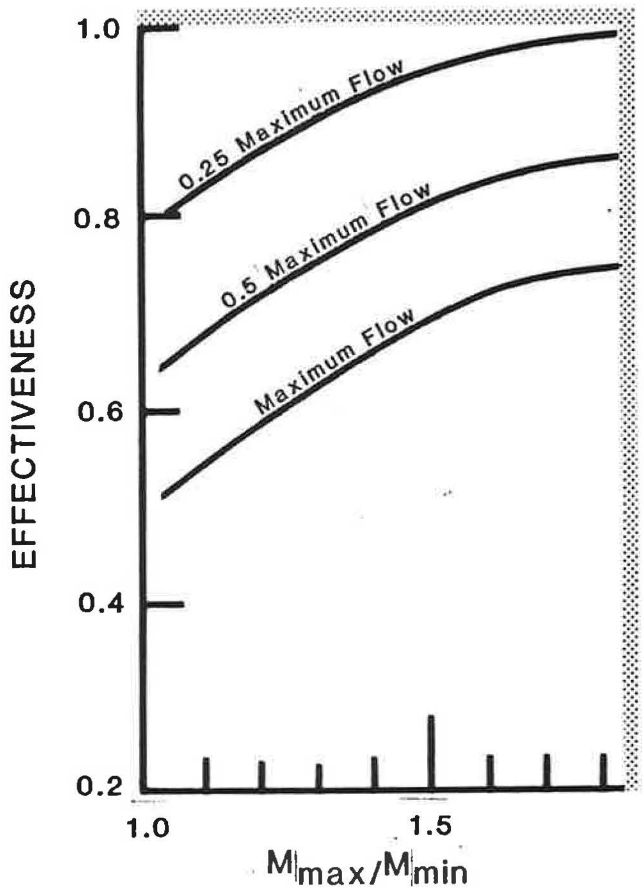


FIG. 4 Variation in Heat Exchanger Effectiveness as a Function of Flow Rate Ratio

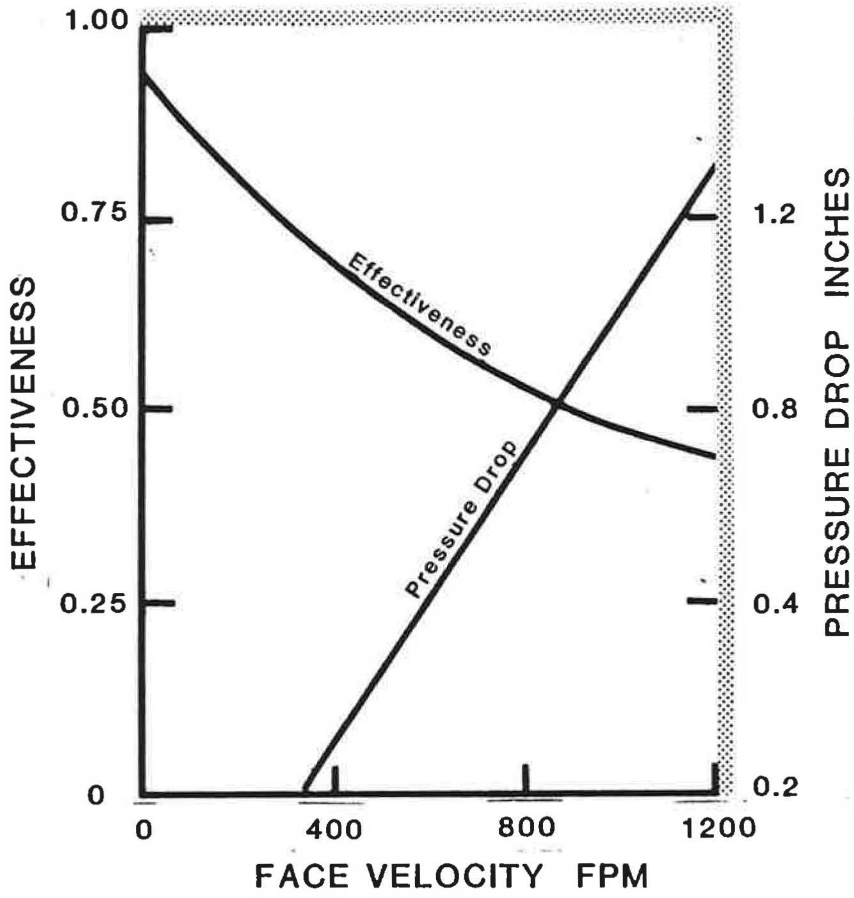


FIG. 5 Variation in Heat Exchanger Effectiveness and Pressure Drop as a Function of Face Velocity

of outdoor ventilation air required, local weather conditions, frosting within the heat exchanger, etc. It must be remembered that the use of an air-to-air heat exchanger will result in an increase in the heating/energy requirements in a well sealed structure. In small buildings that have sufficient natural ventilation due to infiltration-exfiltration, an air-to-air heat exchanger is not recommended at all.

The potential energy savings for a 2,000 sq. ft., single story structure, assuming that 0.5 air changes per hour (ACPH) are required, can be estimated. This air exchange rate is equivalent to a volumentric flow rate of 133 cfm. Further, it is assumed that 33 cfm is provided by natural infiltration. Then the required flow through the heat recovery device is 100 cfm. The maximum potential energy saving can easily be estimated by:

$$E = .26\eta\ V(H.I.) \qquad (5)$$

where E is Btu/year; η is overall thermal effeciency in %; V is volumetric flow rate, cfm; and H.I. is the total heating degree-days accumulated during the heating season, °F-day. For central interior Alaska with a 12,000 °F-day heating season (September - April), a savings of 20 x 10^6 Btu is possible assuming 24 hour day per operation, unit overall efficiency of 65% and a flow rate of 100 cfm. This energy savings is equivalent to 200 gallons of fuel oil assuming a heating system efficiency of 70%.

FROSTING PROBLEMS

Designing heat recovery systems using air-to-air heat exchangers systems for the Alaskan environment, requires knowledge of when and how often frosting will occur. This information will allow the designer to justify and incorporate proper frost prevention equipment and controls.

Frosting in air-to-air heat exchangers is a problem that can exist whenever supply air (outdoor) temperatures fall below freezing. Condensation will occur in all air-to-air heat exchangers when any surface drops below the dew point temperature of the warm airstream. Provisions should, therefore, be made in the design of air-to-air heat exchangers for the collection and removal of condensate or melt water to avoid water from reaching the duct work. A water drip pan or piping to a drain and a heat exchange element not damaged by moisture should be included in the design.

Whenever exhaust side surfaces of an air-to air heat exchanger drop below the frost point, either frosting (sublimation of water vapor) or icing (freezing of condensate) will occur. The rate of frost or ice accumulation depends upon the temperature of the supply air, temperature and humidity ratio of exhaust air, type and effectiveness of exchanger, and duration of the frosting conditions. Frost or ice will usually first form on the discharge face of the exhaust air side and then increase in thickness and depth of penetration. Under severe conditions, complete blockage can occur stopping the exhaust air flow.

Ruth and Chant [4] have carried out a modest research program on frosting in hygoscopic rotary wheel heat exchangers. They used the naturally occurring subfreezing weather of Manitoba, Canada for the supply side air and controlled the temperature and humidity of the exhaust air. Whenever the pressure drop across the wheel exceeded 0.2 in. water gage, frosting had commenced. Their results showed that frosting occurred between supply air temperatures 15°F and -15°F, and can best be detected by the pressure-drop across the exchanger. Therefore, it is recommended that a differential pressure sensor be used on the exhaust air side of the heat exchanger to detect frost build-up. Frost can be prevented either by preheating the supply air or reducing the effectiveness of the heat exchanger. The latter can be accomplished by bypassing a portion of the cold supply air, by speed control for rotary wheel exchangers, by tilt control for heat pipes, or by a three way valve in coil-loop-run-around systems. All of these techniques result in a reduction of system efficiency, i.e. decrease in the amount of energy recovered.

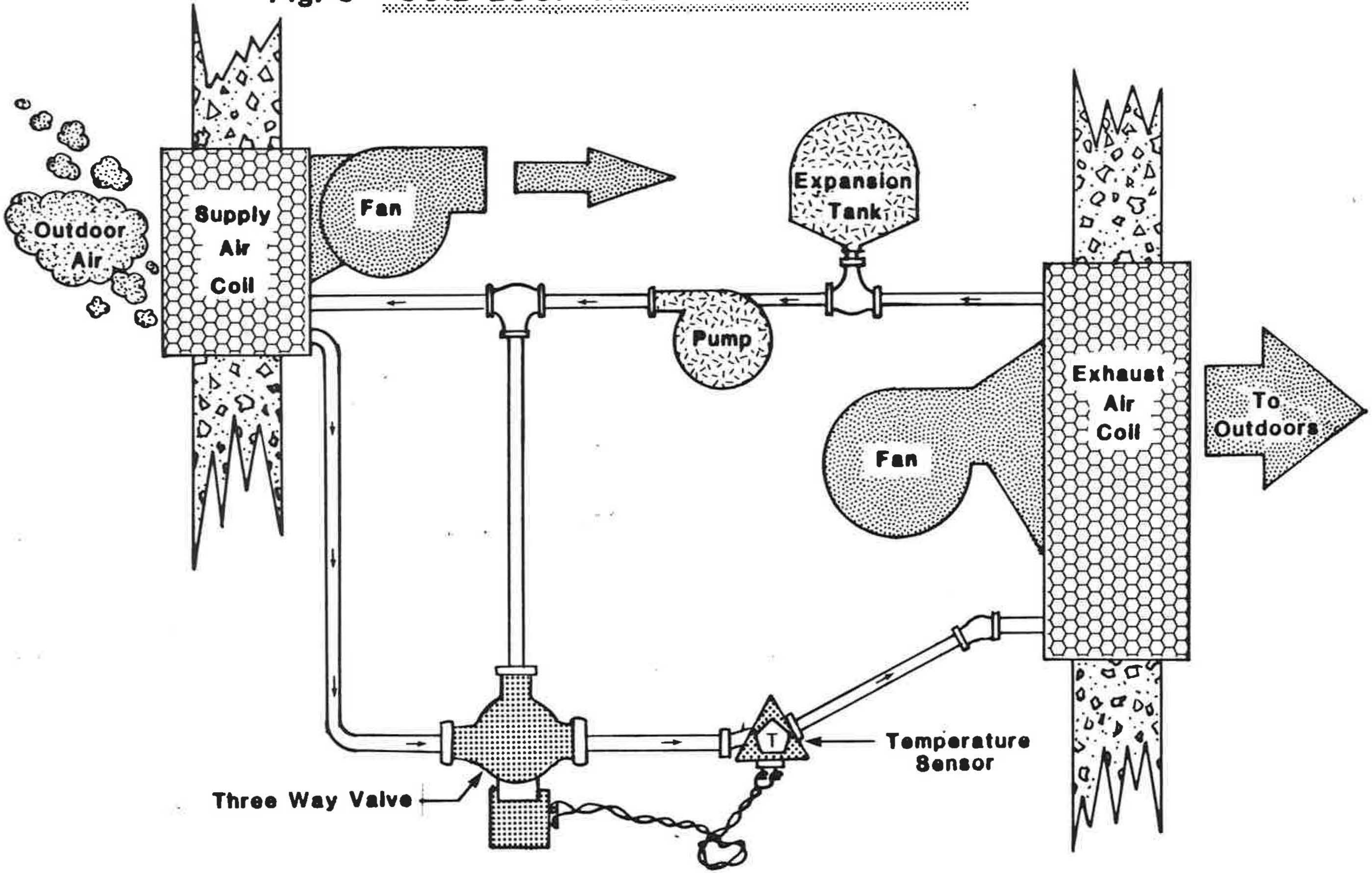
Another approach to the solution of the frost and ice problem is intermittent operation. In this case, the unit is shut-down whenever frost has accumulated to the point where the pressure drop exceeds the set point. Once the frost has melted and the water drained away, the unit is placed into operation again. The control for this system could be reduced to a simple inexpensive on-off timer which would provide intermittent operation as well. The unit could be programmed to be turned off during those hours when forced ventilation air is not needed or is at a minimum requirement.

Further details on frost prevention control systems will be discussed in the following sections.

Coil-Loop-Run-Around System A three-way thermostatically controlled flow valve can be incorporated into the recovery loop as shown in Fig. 6. The valve is usually adjusted so that the entering liquid temperature to the exhaust coil is maintained above 30°F. This is accomplished by by-passing a portion of the warmed liquid returning from the exhaust air coil. A second purpose for the valve is to ensure that a prescribed exit air temperature from the exhaust coil is not exceeded for those installations where heat recovery must be limited.

Heat Pipe System Gravity can be used to assist the return of the condensate in heat pipes to the evaporator section by operating the heat pipe on a slope with the hot end down. Increasing the slope increases the gravitational assist and the units effectiveness. Tilt adjustment can, therefore, be used as a means of frost control. In practice, tilt

Fig. 6 COIL-LOOP-RUN-AROUND SYSTEM



control is accomplished by pivoting the exchanger about the center of its base and using flexible ducting to allow for the small rotational movement. A temperature controlled actuator adjusts the tilt of the unit to maintain the exhaust air exit leaving temperature at or above the prescribed set point.

Plotted in Figs. 7-9 are frost threshold temperature curves at indoor temperatures of 60°F, 70°F, and 80°F for Q-DOT heat tube heat exchangers having fourteen fins per inch. Each figure has a family of curves covering the range of exhaust air to supply air ratio of 2.0 to 0.5. These curves have been plotted in terms of relative humidity as a function of temperature. Frosting within the exchanger should not occur when conditions are such that the unit is being operated to the right of these curves.

Plate Type System Plate type heat exchangers have no provision for the prevention of frost. Therefore, ASHRAE [5], has conducted and reported the results of an extensive test program to determine the limiting conditions that will produce frost or ice buildup in a plate unit. The results of this program are shown in Fig. 10. Four lines are shown in this figure, each one representing a different ratio of supply to exhaust air flow. Operating the exchanger so conditions are above the line should result in no frosting, and if conditions fall below the line, frost can be expected. From these results it is clear that higher humidity exhaust airstreams are less likely to experience frosting or icing. For plate heat exchangers, preheating supply air or bypassing supply air are the only frost control methods available.

CROSS CONTAMINATION IN EXCHANGER DESIGN

Cross contamination or mixing of the supply and exhaust airstreams can occur by leakage across the heat exchanger surfaces due to differences in static pressure, or short circuiting between the exhaust and supply

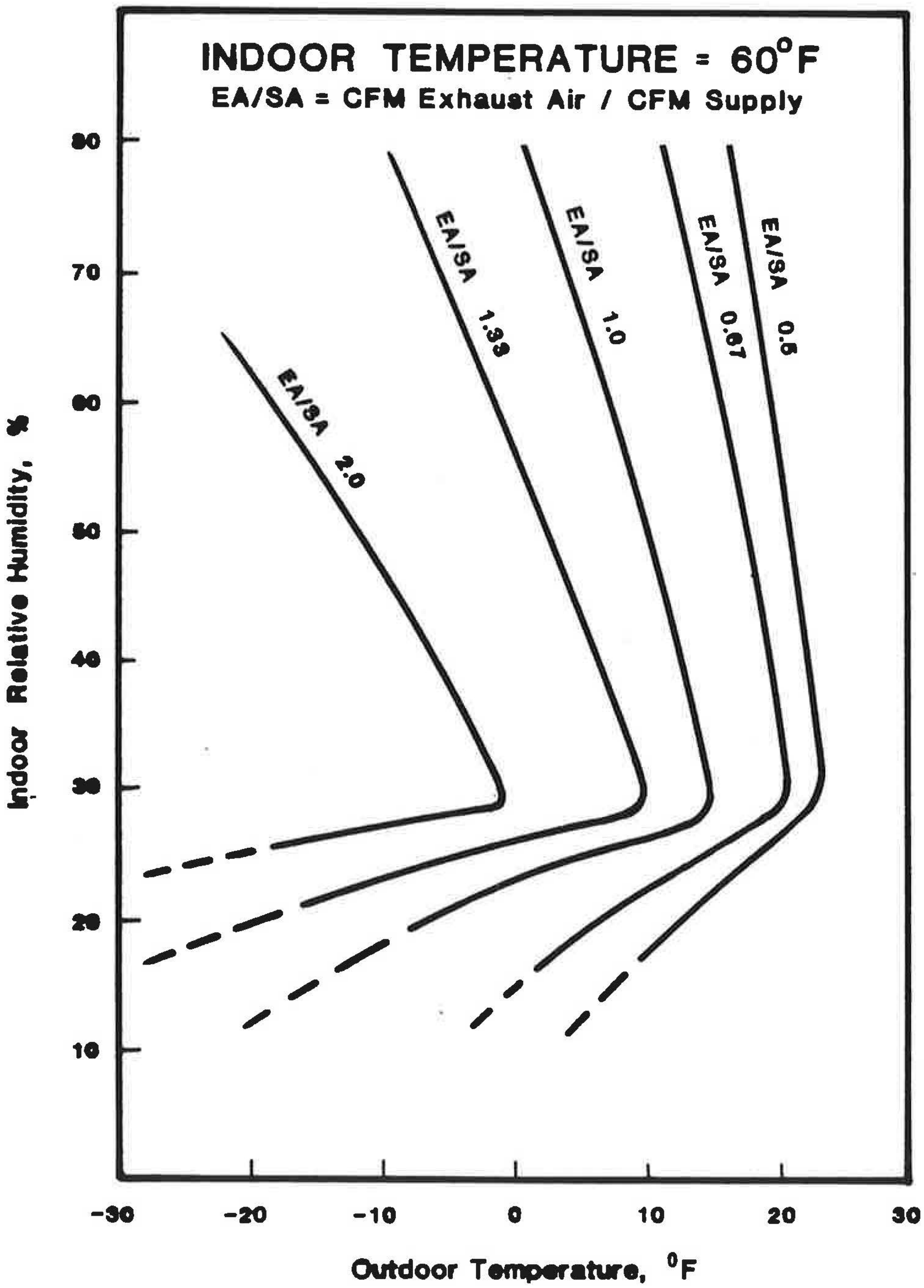


FIG. 7 Frost Threshold Curves For Q-Dot [6] Air-To-Air Heat Exchanger. Operating conditions to the right of the curve should result in no frosting on exhaust side of unit.

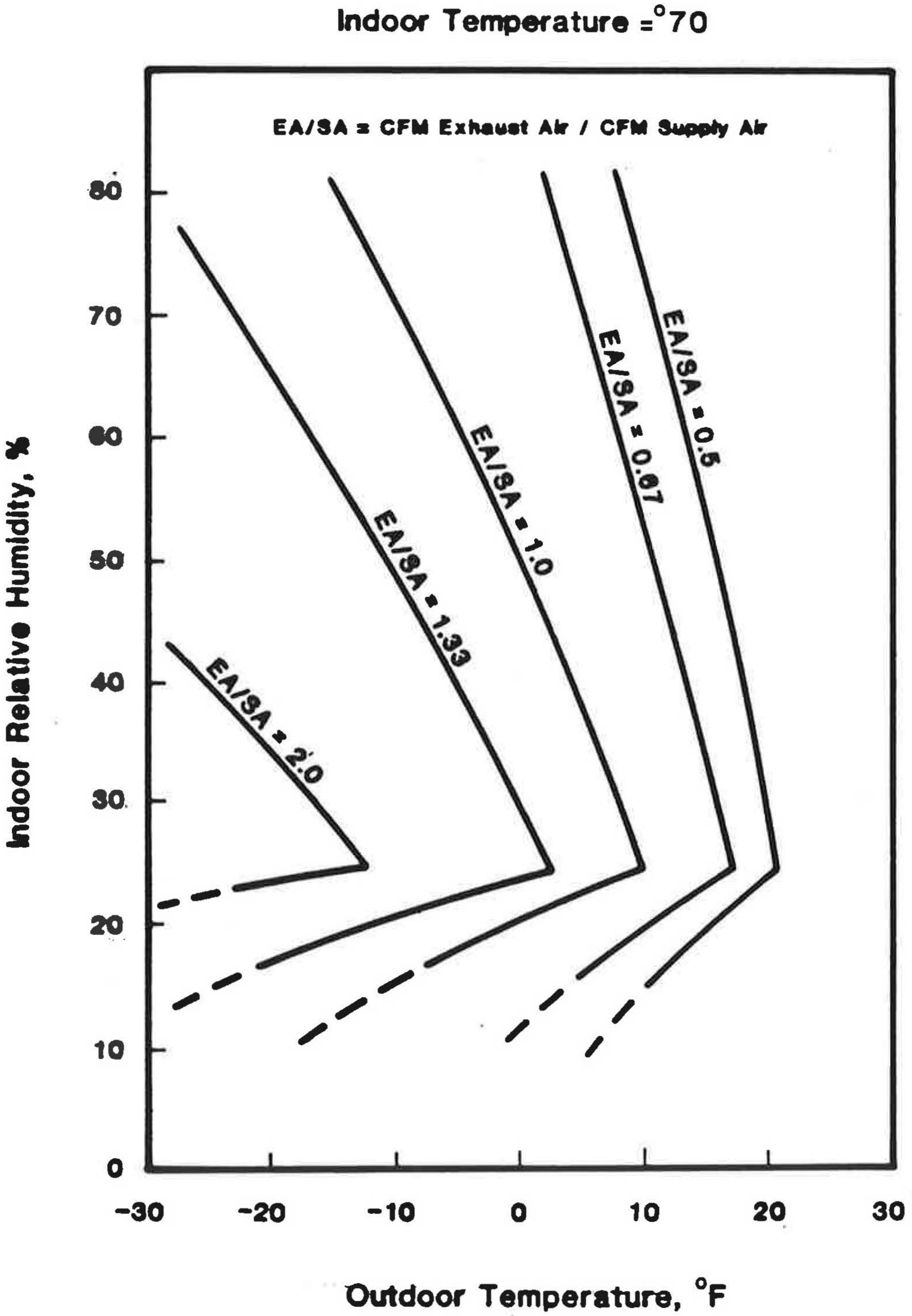


FIG. 8 Frost Threshold Curves For Q-Dot [6] Air-To-Air Heat Exchanger. Operating conditions to the right of the curve should result in no frosting on exhaust side of unit.

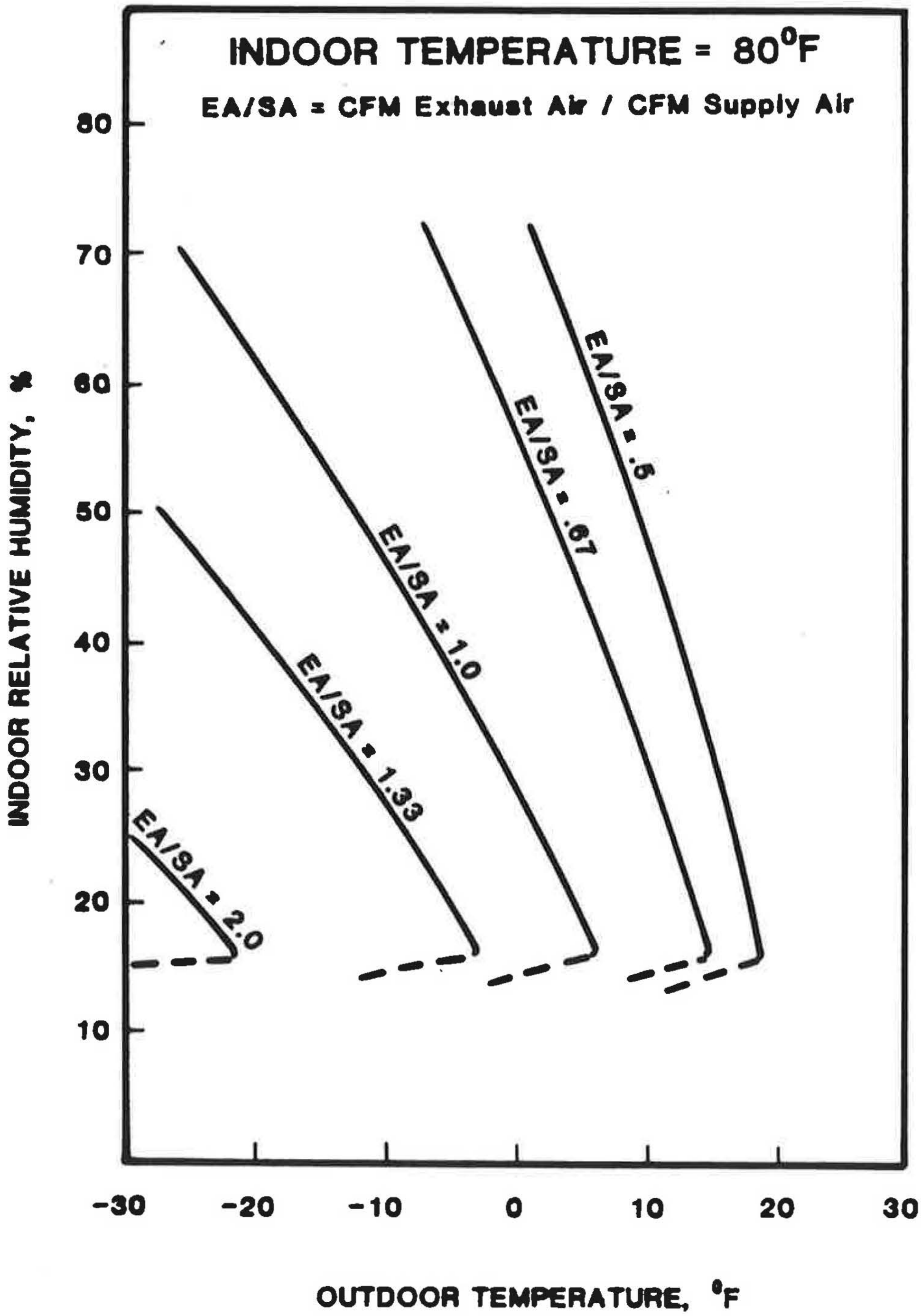
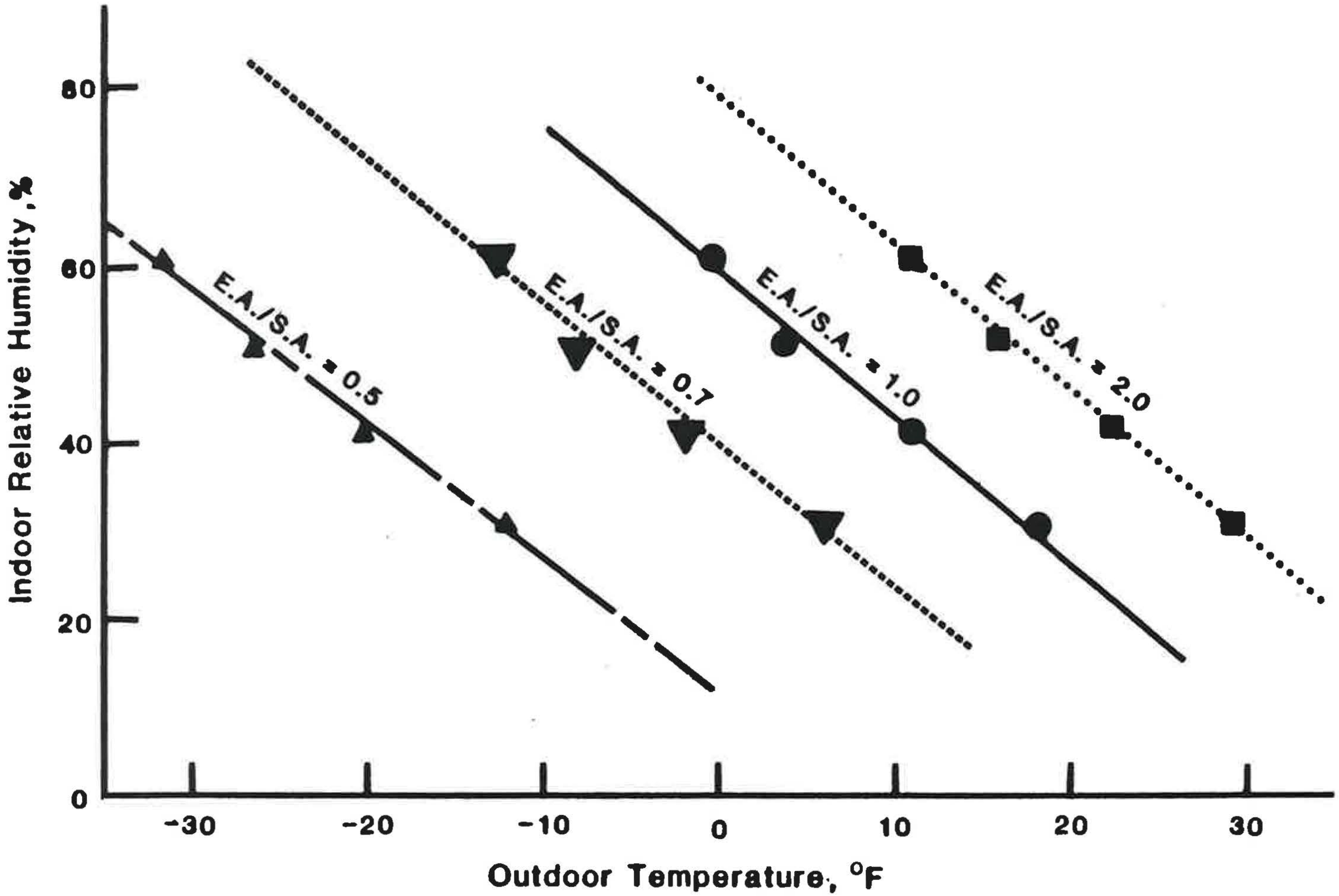


FIG. 9 Frost Threshold Curves For Q-Dot [6] Air-To-Air Heat Exchanger. Operating conditions to the right of the curve should result in no frosting on exhaust side of unit.

Fig. 10 FROST THRESHOLD TEMPERATURES FOR PLATE TYPE HEAT EXCHANGES

(If operating conditions fall below curve frosting occurs)



ASHRAE [5]

air inlets/outlets. Leakage to the supply air can be minimized in most installations by maintaining the supply air side of the heat exchanger at a slightly higher pressure than the exhaust side to ensure all leakage is into the exhaust airstream. Short circuiting can be eliminated by placing the inlet and outlet louvers and grills sufficiently far apart. If this is inconvenient, placing the supply air outlet grill below the warm exhaust air inlet grill will avoid short circuiting by taking advantage of the fact that the cool supply air will naturally fall because of its higher density. On the outside of the building the same effect holds true, i.e. warm exhaust air will rise.

SMALL SCALE HEAT EXCHANGERS

The Besant [7] exchanger is shown in Fig. 11. This is a build-it-yourself unit constructed of 1/2 inch plywood using 6 mil polyethylene sheets for the plate type heat transfer surfaces. The drawer, partly open at the bottom of the picture, serves as a condensate reservoir. It could easily be modified to accept a drain line. The view of the bottom of the unit in Fig. 12 shows 1/2 inch plywood serving as frames and spacers for the polyethylene sheets. The unit is 90 inches high, 24 inches deep and 18 inches wide. It was found during testing that it was necessary to insulate the unit to avoid condensation/frost buildup on the outside surfaces. Construction time for the Besant unit is about 30 hours if all necessary wood working equipment is available. The materials for the heat exchanger and the supply and exhaust air fans totaled $250.

A Canadian manufactured, Enercon Industries [8], commercially available air-to-air heat exchanger is shown in Fig. 13. This unit is similar to the Besant design except the housing is a fiberglass shell. A drain has been included at the bottom of the unit. The supply and exhaust fans with controls are a part of the package. The total price F.O.B. Fairbanks was approximately $800.

A second Canadian manufactured, Passive Solar Product [9], air-to-air heat exchanger is shown in Fig. 14. This unit again is a spin-off from the Besant design except in this case the housing is galvanized sheet

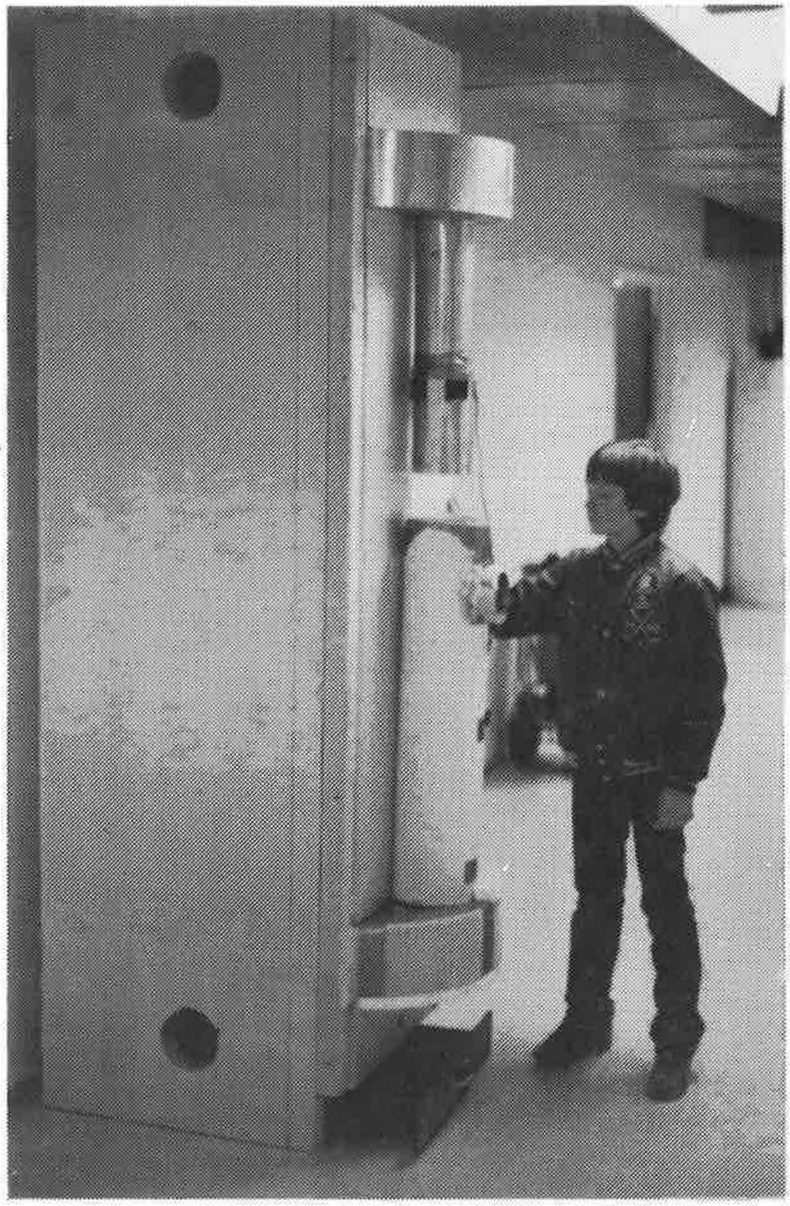


FIG. 11 Air-to-Air Heat Exchanger built from plans developed by Besant [7] at the University of Saskatchewan.

FIG. 12 Internal View of Heat Exchanger Shown in Fig. 11. The 1/2 inch plywood spacers separating the 6 mil polyethylene sheets which serve as the heat transfer surface are shown.

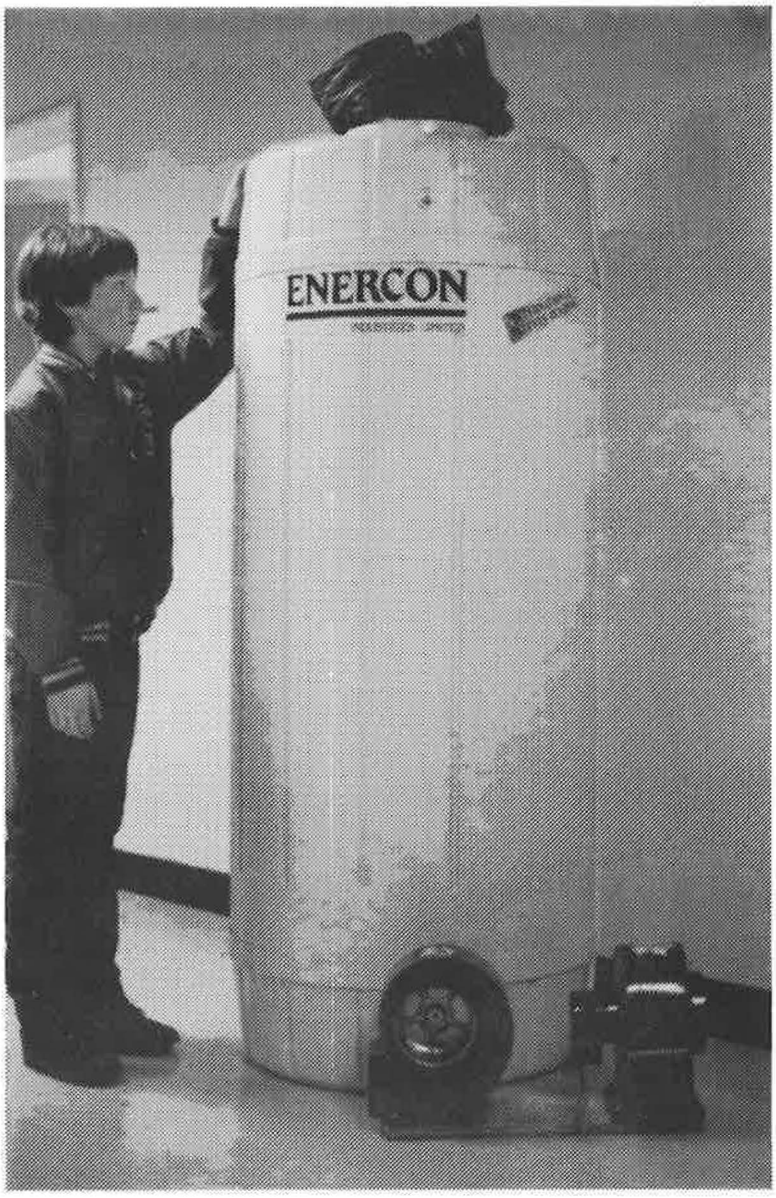


FIG. 13 A Canadian manufactured residential heat exchanger, Enercon, [8], similar in design to the Besant unit. The supply and exhaust fans included with the unit are also shown.

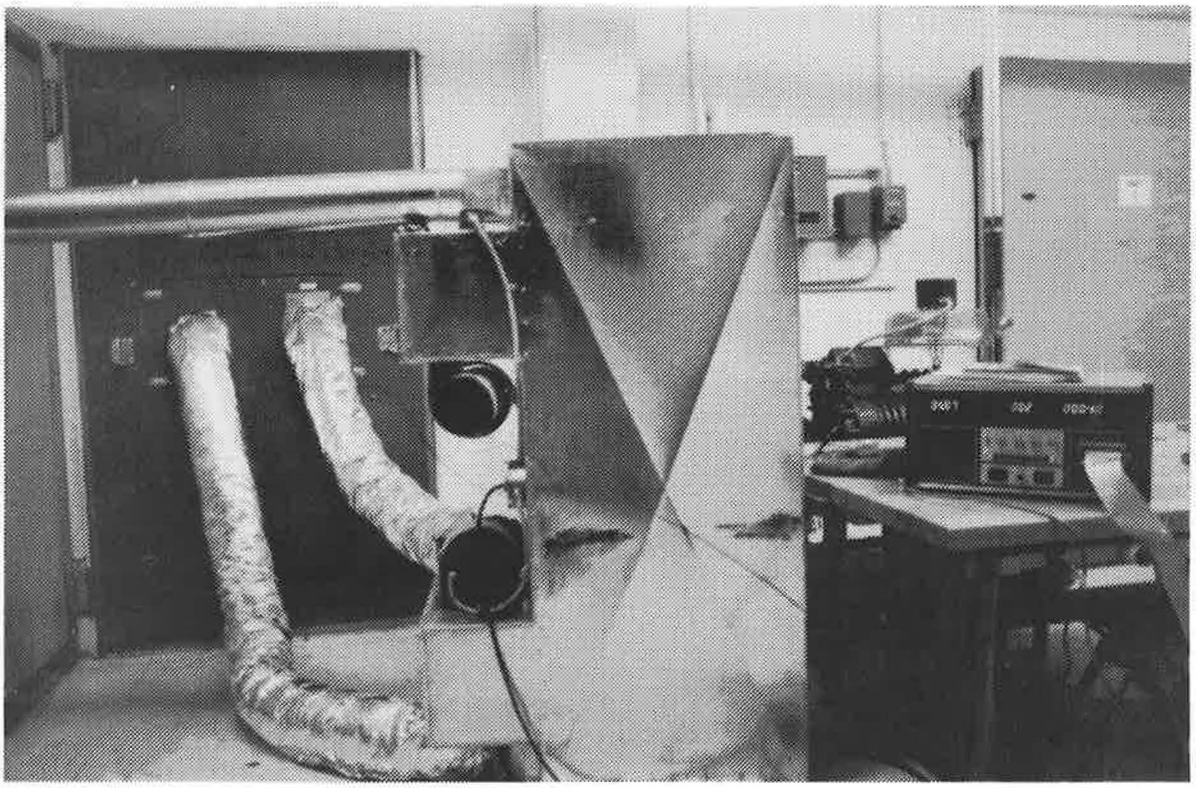


FIG. 14 A Canadian manufactured residential heat exchanger [9] with galvanized sheet metal housing also similar in design to Besant unit. The device is set up for performance testing using the cold room to simulate out-doors. Also shown is the data logger which samples temperatures every hour.

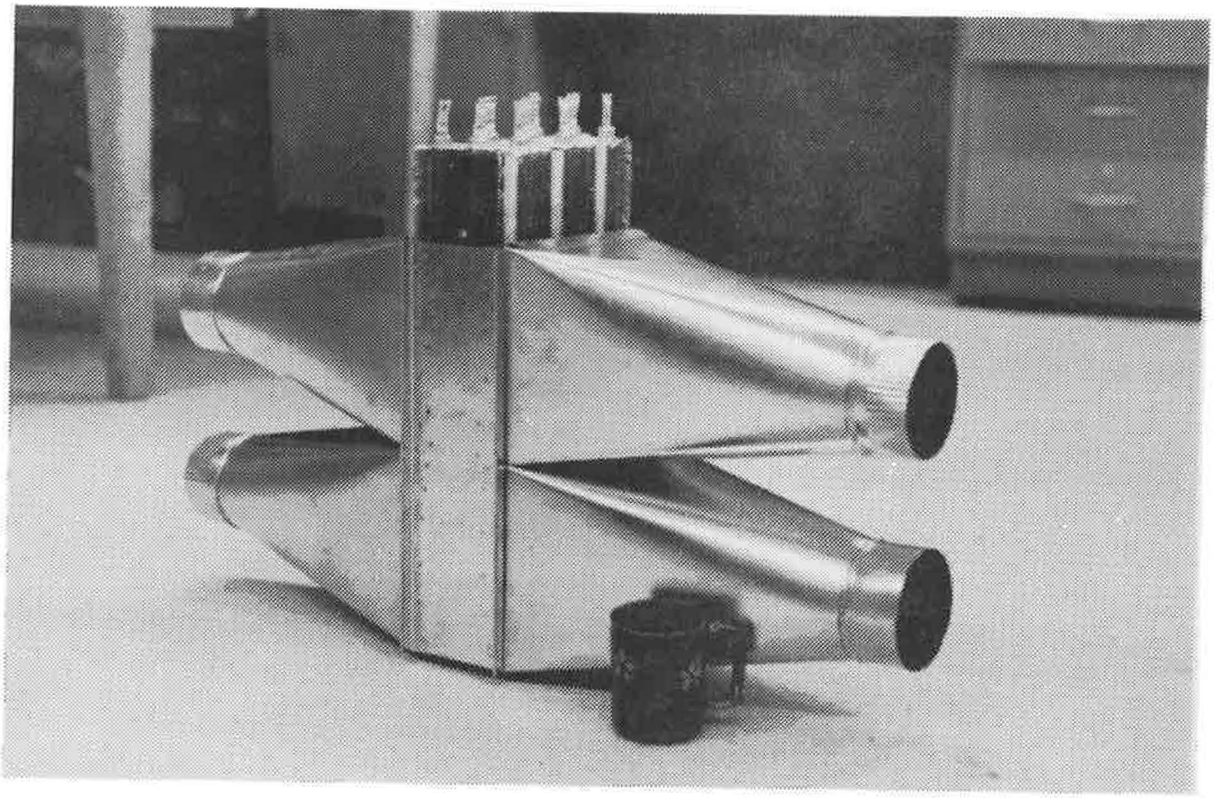


FIG. 15 A Q-Dot heat pipe coil [6] for air-to-air heat recovery partially inserted into sheet metal housing. The heat transfer rate in the unit can be controlled by tilting the entire assembly.

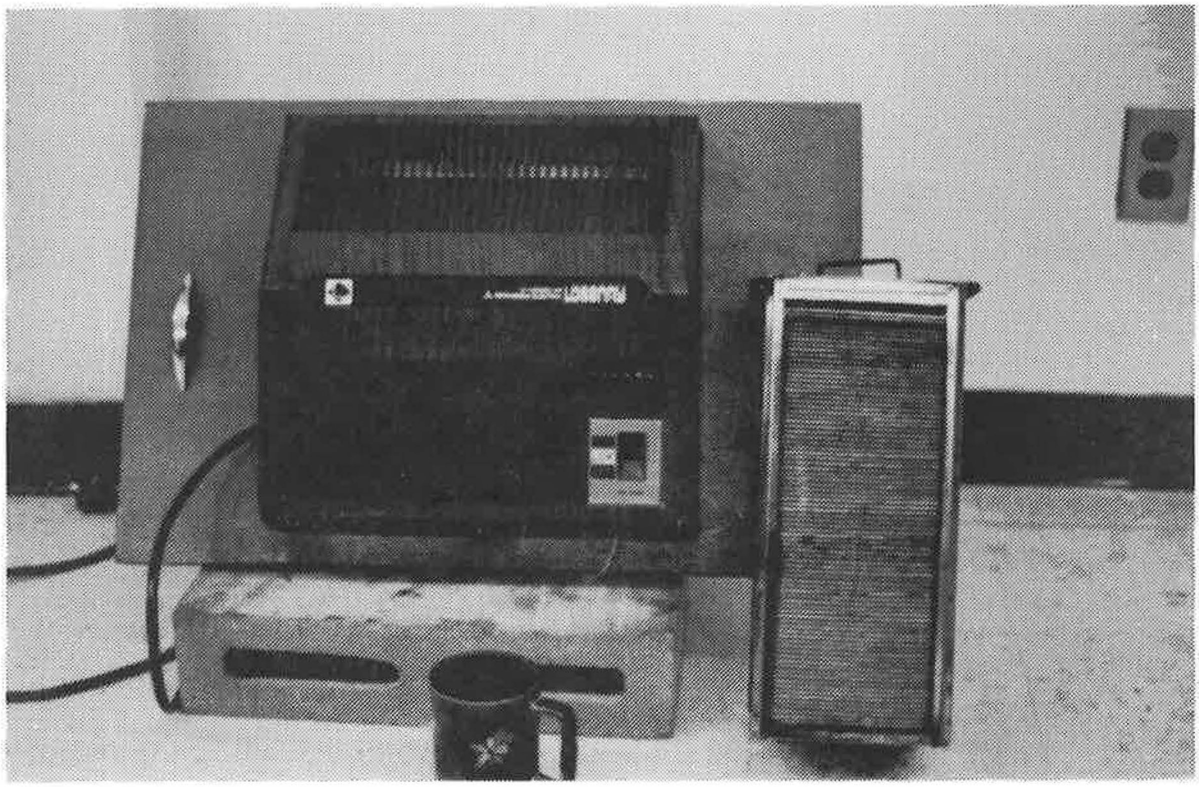


FIG. 16 The Lossnay [10] air-to-air residential heat exchanger manufactured by Mitsubishi Electric Co. in Japan. The unit is self contained including supply and exhaust fans. The cross-flow heat transfer element is made of paper which is also shown. Some shrinkage occurred when the paper element dried out after becoming plugged with frost.

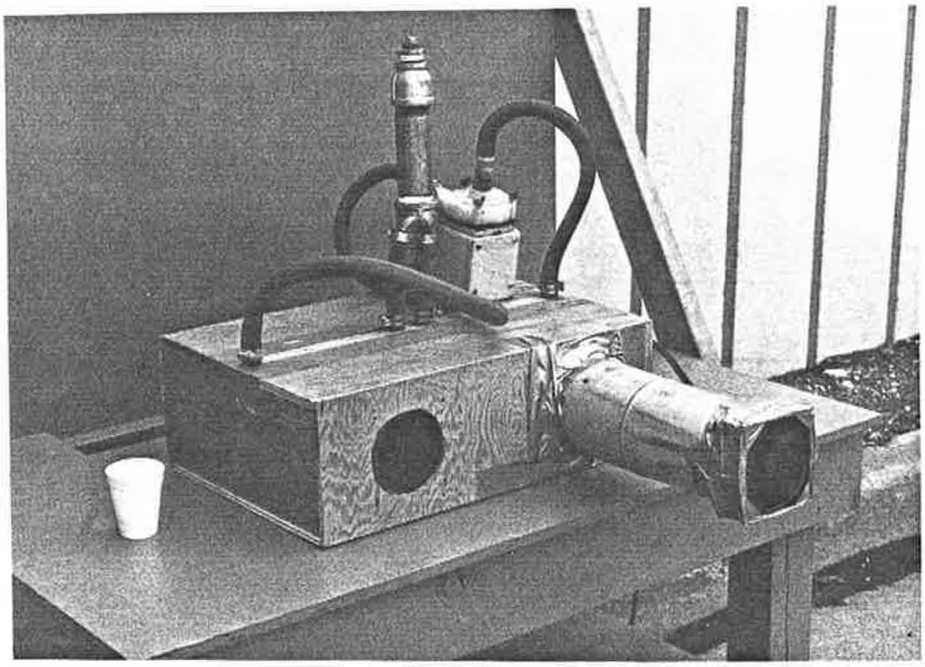


FIG. 17 Coil-Loop Run-Around Exchanger assembled, using a plywood housing with automotive heater cores as the heat transfer element. A GRUNDFOS UPS 20-42S pump was used to circulate an ethylene glycol and water antifreeze through the system.

metal. A drain is included at the bottom of the unit and the supply and exhaust fans have been premounted. Thermostat and humidistat controllers have also been premounted on the unit to provide for automatic defrosting. The F.O.B. Fairbanks price was about $750.

A small heat pipe coil manufactured by the Q-Dot Corporation [6] is shown partially out of the duct housing in Fig. 15. The galvanized sheet metal housing was designed and fabricated in Fairbanks. In contrast to the three previous described units, this exchanger is much smaller in size. Also, the coil being made of aluminum, lends itself to cleaning, should the surfaces become fouled with dust, lint, dirt, grease, etc. It is estimated that the coil, housing and fans would cost $500 as a package purchase.

Fig. 16 shows the Lossnay heat exchanger manufactured by Mitsubishi Electric Company [10] in Japan. The cross-flow plate type heat transfer element is also shown in this picture. This element is made of paper and allows both sensible and latent transfers of energy. The unit is designed for through the wall mounting and is totally self-contained having multi-speed supply and exhaust fans installed in the unit. No drain provisions have been included and the paper element experienced shrinkage when it dried out after being saturated with moisture due to condensation and frost buildup in the paper element. Short circuiting of the supply and exhaust air occurs since the cool supply air is delivered to the room at the top of the unit. A smaller Lossnay heat exchanger rated at 24 cfm has a plastic cross-flow heat exchange element which should solve the shrinkage problem. The cost of the unit shown F.O.B. Fairbanks was about $280.

A coil-loop run-around heat exchanger, Fig. 17, was constructed using two automotive heater cores, a small circulating pump, two muffin fans, heater hose for connecting pumps and heater cores, an accumulator (expansion pipe) and plywood housing. The total cost of components was $200 plus approximately ten hours of labor to assemble the unit. Used heater cores were purchased from a local salvage dealer which greatly reduced the cost of those components.

TEST PROGRAM

A test program to evaluate heat exchanger efficiency of several types of heat recovery devices has been conducted. The University of Alaska-School of Engineering cold room facilities, capable of temperatures down to -25°F, were used as a source of cold air. Shown in Fig. 14 is a picture of the Humid-Fire No. 11 under test. Insulated air conditioning flex duct of five inch diameter is used to connect the supply and return air sides of the heat exchanger to the cold room. A Digetec Model 2000 Datalogger recorded supply and exhaust air temperatures before and after the heat exchange process, as well as the room temperature. The air speed in the supply and exhaust air ducts was measured with an Alnor Model 8500 hot wire annemometer.

Each exchanger system was tested for a period of several hours and in some cases two flow rates. The reason for the long term test was to determine the influence of frost formation on the performance of the unit. The relative humidity of the room air during the test programs averaged 20% and the cold room temperature was maintained at about -15°F. Long term test results are shown in Fig. 18 for the Besant unit at volumetric flow rates of 32 cfm and 64 cfm. As seen, the efficiency of the unit decreased as a frost layer built up on the surfaces of the polyethylene sheets. At the completion of the 24 hour test, 2.2 pounds of condensate was drained from the unit. The efficiency of the heat exchanger tested was less than the results reported by Besant. A possible explanation might be lack of adequate sealing between adjacent flow passages. The data for this test as well as in the following tests, was fitted using a linear least squares best fit curve.

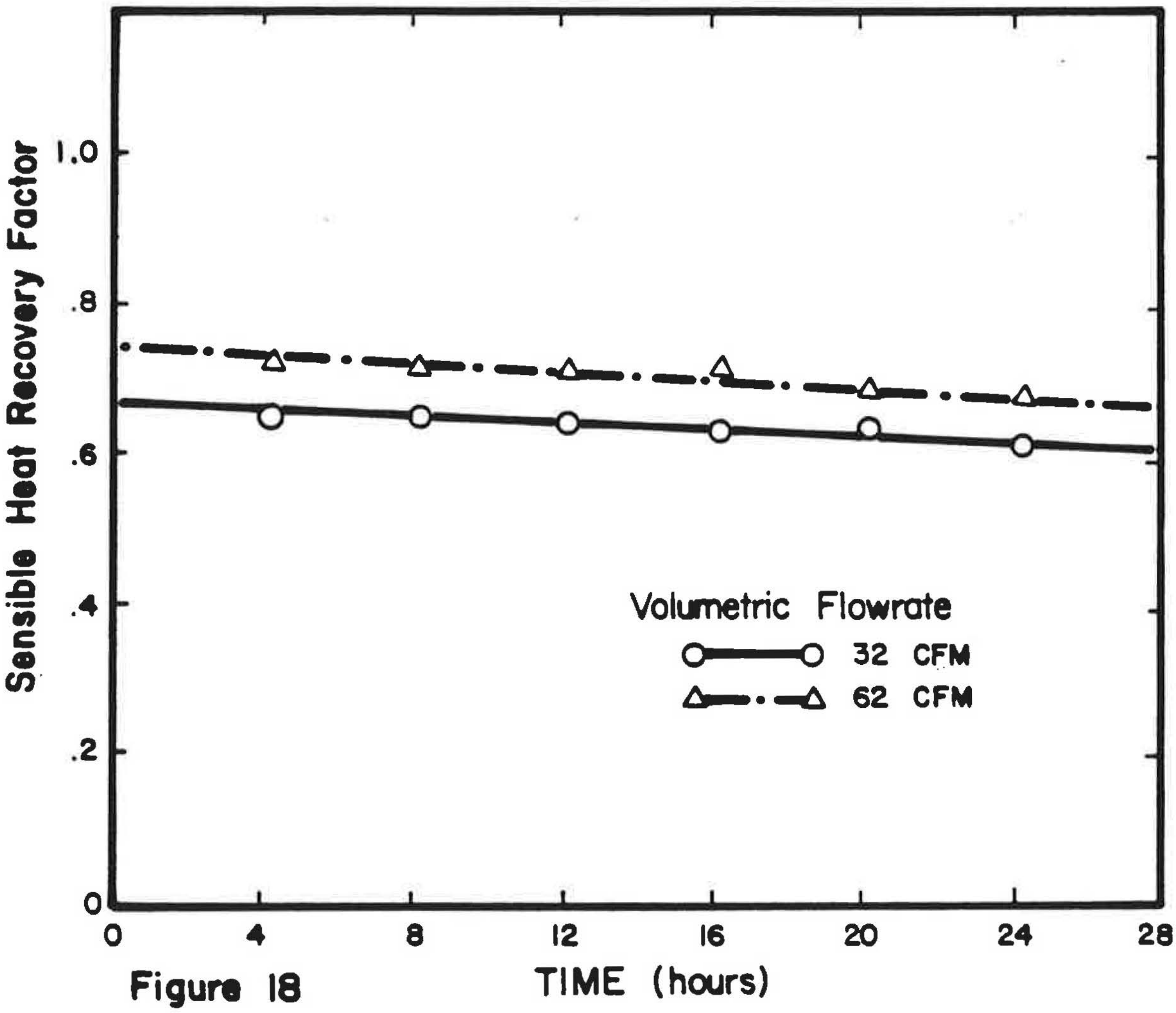


Figure 18

Time varying performance of Besant type air-to-air heat exchanger

The heat recovery factor variation of the Humid-Fire Model 11 air-to-air heat exchanger operated at a flow rate of 78 cfm is shown in Fig. 19. This unit was later installed in a Fairbanks residence. The homeowners reported no problems occurred while using the unit, Ritchie [13]. Heat recovery factor variation for the Enercon Model B unit is plotted in Fig. 20. This unit had the highest efficiency of all the units tested. Similar results for the coil-loop run-around system are shown in Fig. 21.

Figure 22 shows the heat recovery factor variation of the Mitsubishi Lossnay Model 1200 M. This unit was installed in an exterior door of a local residence, Leonard [14]. At ambient temperatures of 20°F or above, the unit seemed to function adequately. However, at temperatures below 20°F, frost build-up was observed on the out-of-doors portion of the unit whether or not the fan was operating. The air leakage taking place when the fan is not running is probably due to stack effect in the building allowing exfiltration of warm moist air through the unit. When water vapor laden warm air contacted surfaces below its dew point temperature, condensation/frosting occurred. Following a cold spell, a puddle of water would be found on the floor beneath the unit as the warmer outdoor temperatures allowed the frost and ice to melt within the unit. At temperatures colder than -20°F, the fan motor had difficulty starting; at -40°F or colder, the fan motor would not run at all. At temperatures above -20°F but colder than -10°F, severe icing of the heat exchanger core occurred. This resulted in a drop in temperature of the incoming air stream. Air temperature measurements indicated an initial sensible efficiency of 67%, which degraded to 47% after two hours of operation. Having installed the Lossnay unit in a door allowed a large portion of the sheet metal housing to protrude on the outside. A through-the-wall installation would have reduced the amount of surface area exposed to outdoor ambient temperatures and thereby possibly reducing the icing problem within the unit.

Shown in Fig. 23 is the heat recovery factor variation of the Q-Dot TRU-120M-6B heat pipe heat exchanger. The position of the heat pipes during

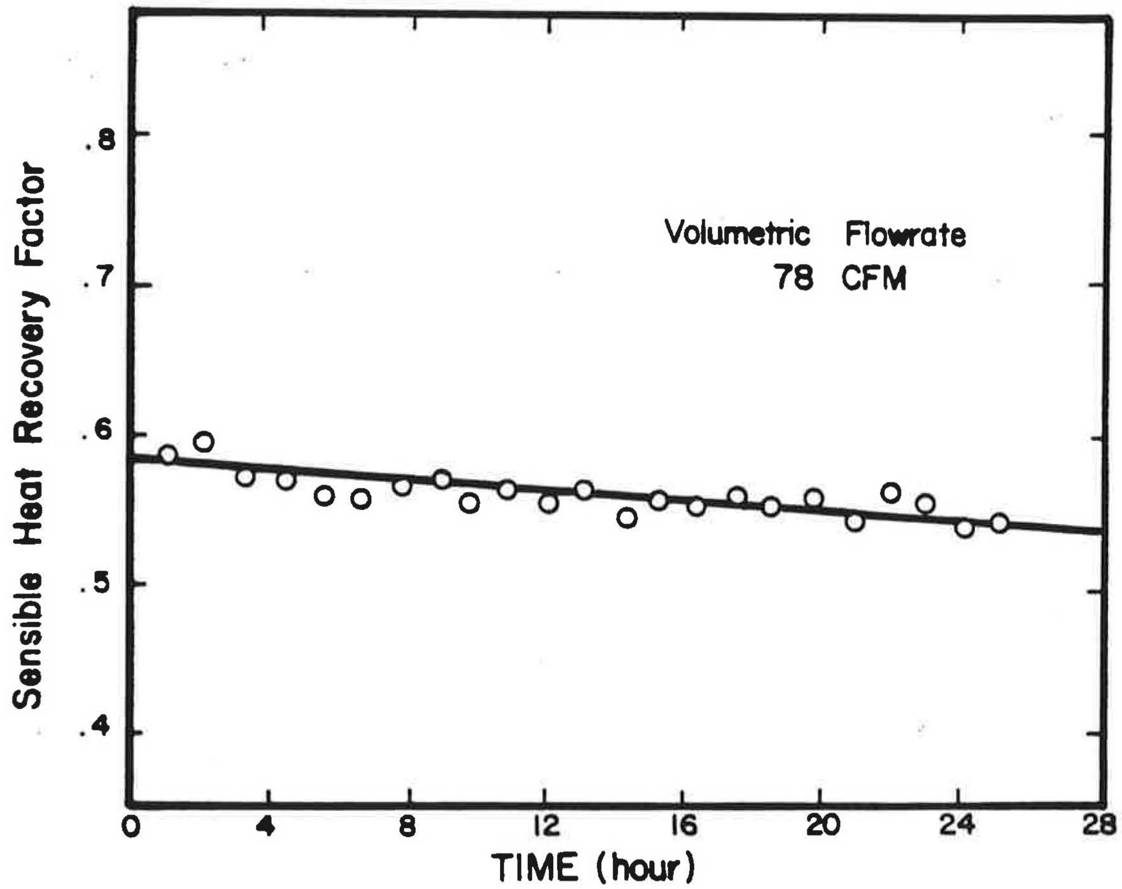


Figure 19: Time Varying Performance of Humid Fire No. 11 Air-to-Air Heat Exchanger

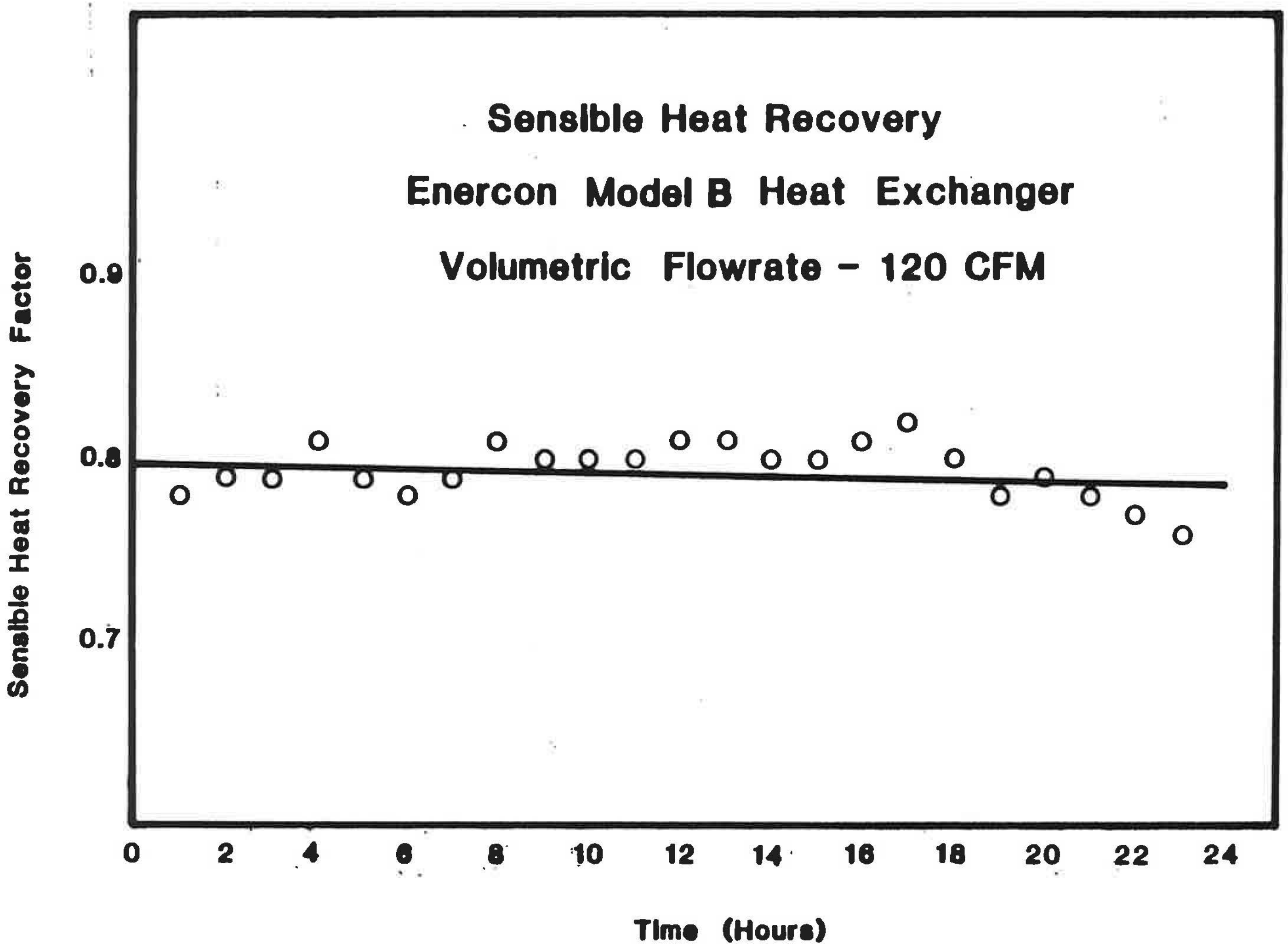


Figure 20: Time Varying Performance of Enercon Model B Air-to-Air Heat Exchanger

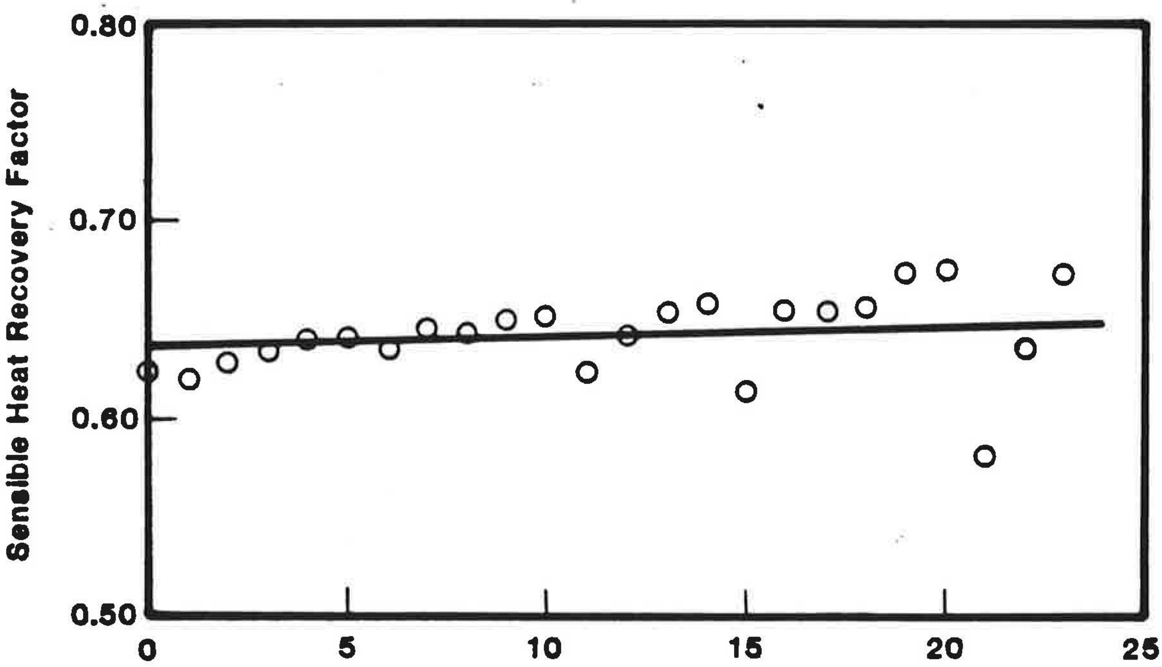


Figure 21: Time Varying Performance of Coil-Loop-Run-Around Air-to-Air Heat Exchanger

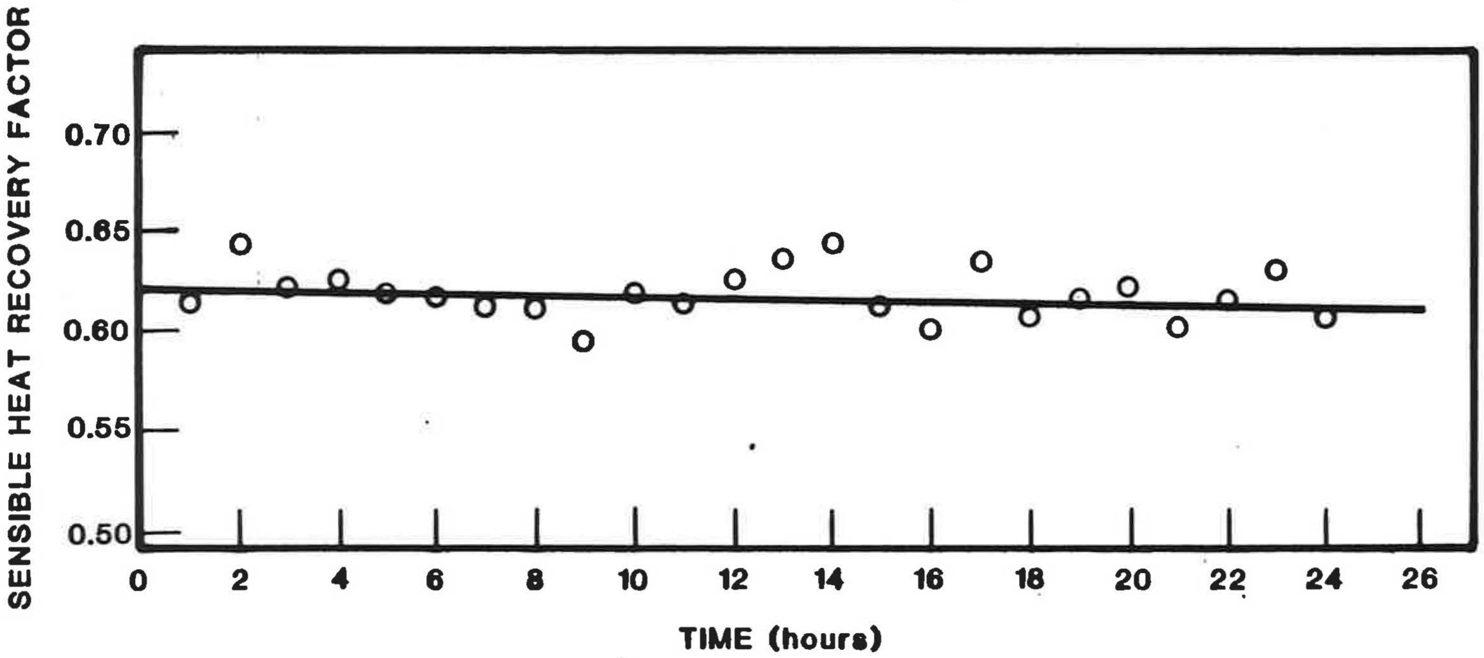


Figure 22: Time Varying Performance of Lossnay Model 1200M Air-to-Air Heat Exchanger

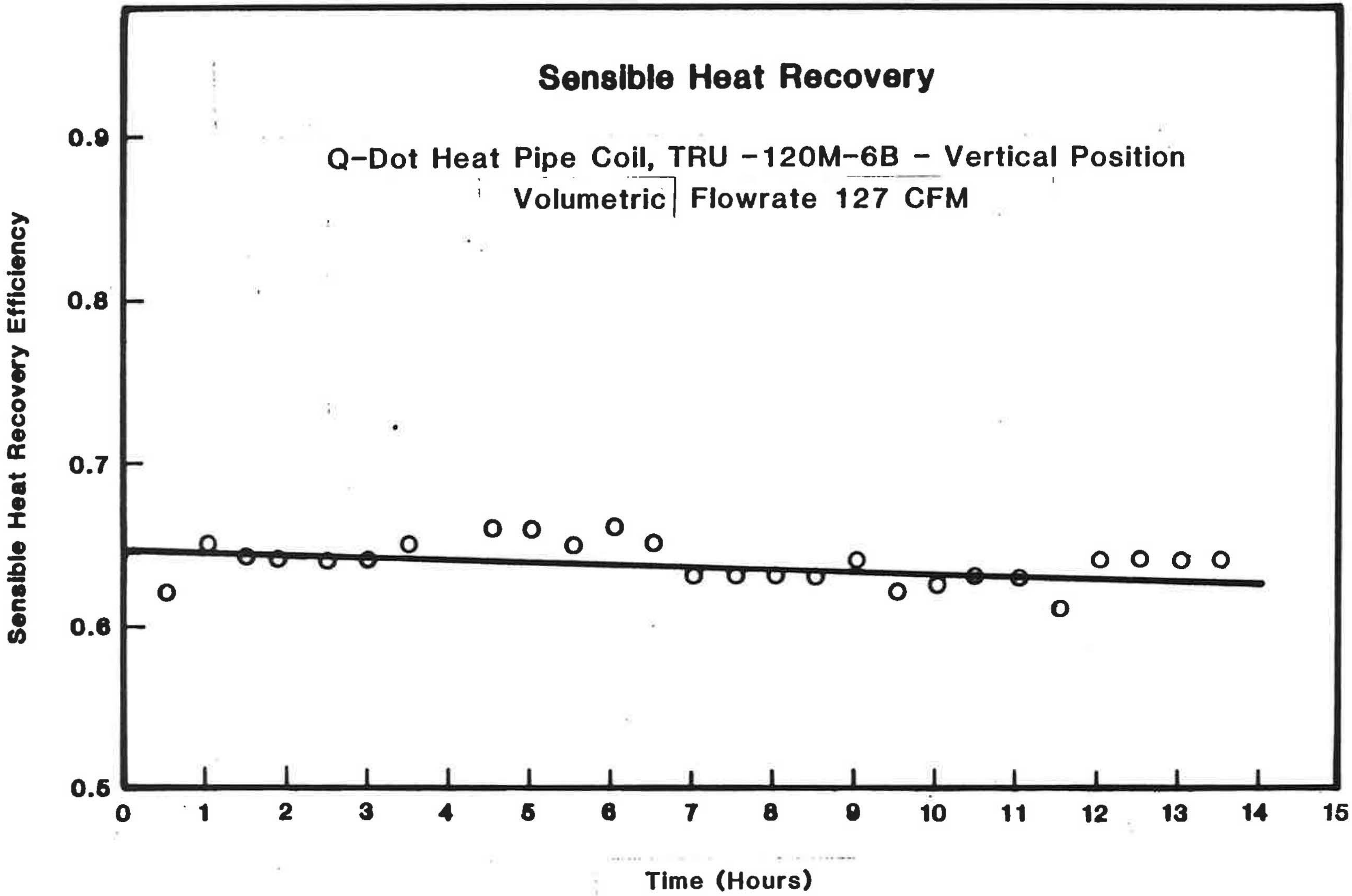


Figure 23: Time Varying Performance of Q-Dot Heat Pipe Air-to-Air Heat Exchanger

this test were vertical with the warm exhaust air ducted through the bottom section of the unit to maximize performance. The unit contained six rows of 5/8 inch O.D. heat tubes, having aluminum fins spaced fourteen per inch. The manufacturer's performance data is presented in Figs. 24 and 25. Static pressure drop as a function of actual face velocity is plotted in Fig. 24 for this unit. Fan selection can be made by combining pressure drop data for the coil and ducting with the desired CFM for the system. The heat recovery factor as a function average face velocity is given in Fig. 25 for flow ratios ranging from 1.0 to 3.0.

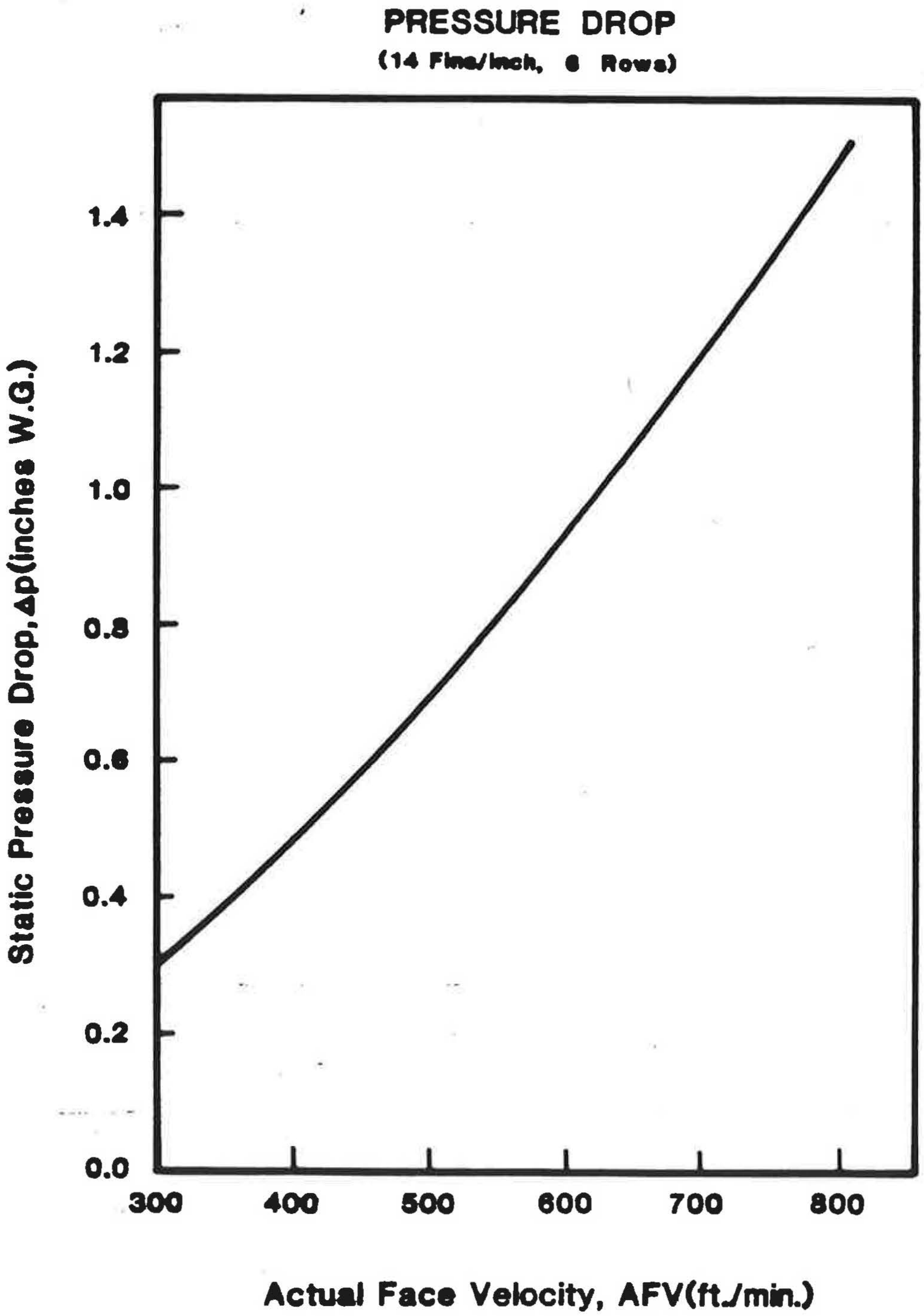


FIG. 24 Static Pressure Drop as a Function of Average Face Velocity for a TRU-120M-6B Q-DOT [6] Heat Tube Heat Recovery Device.

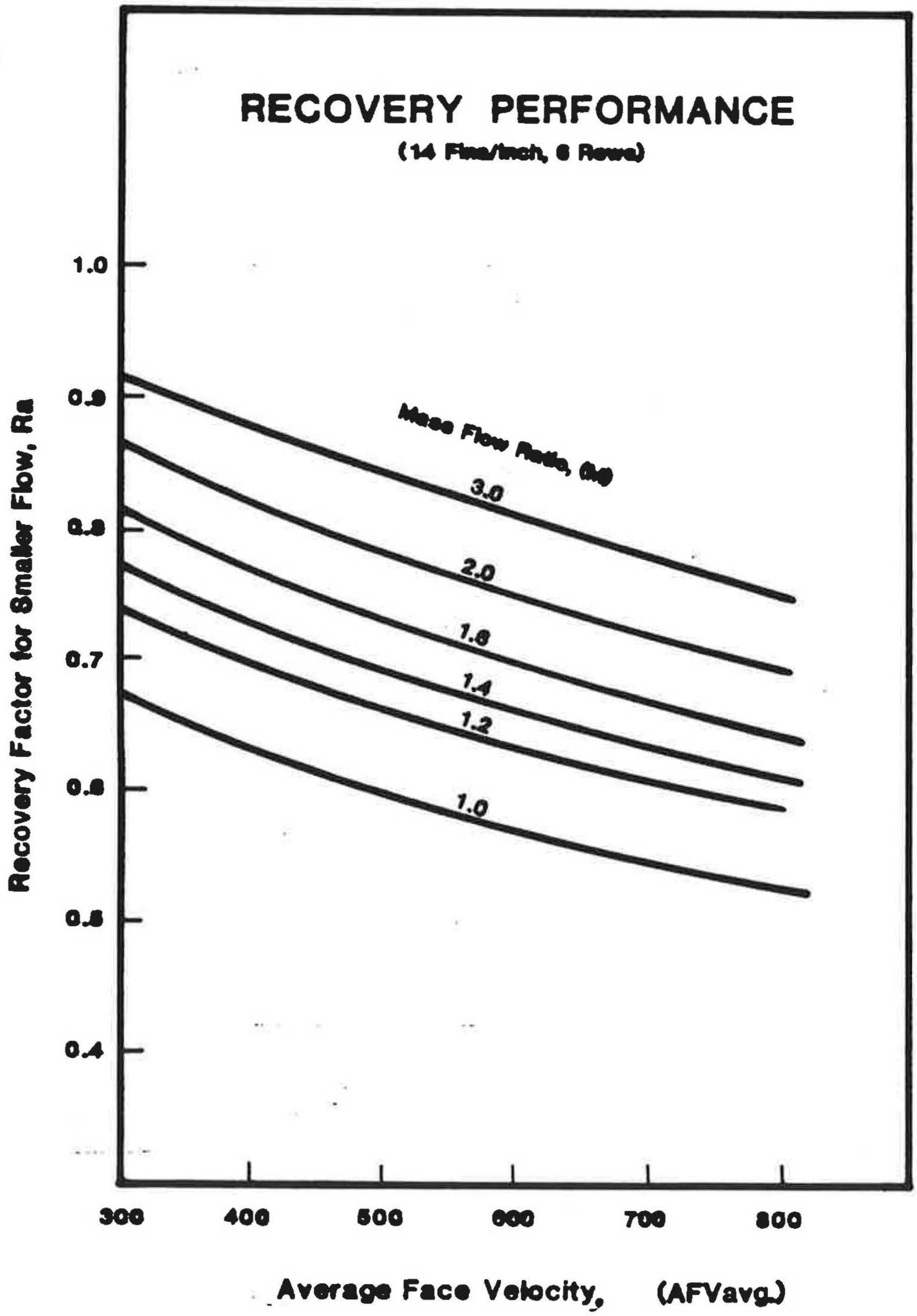


FIG. 25 Heat Recovery Factor as a Function of Average Face Velocity For a TRU-120M-6B Q-DOT [6] Heat Tube Heat Recovery Device.

ECONOMICS

In addition to the purchase cost of the unit, installation costs would also become part of the initial capital cost. With the exception of the Lossnay unit, the cost of ducting from kitchens, baths, and laundry rooms, filters, and back draft dampers; supply and exhaust ducting to the building outside wall; and inlet and outlet grills must be added to the capital cost of the unit. The operating cost for the unit would be the cost of electricity to power the supply and exhaust fans plus any required maintenance.

At a market discount rate of 10% the capital recovery factors are 40% at a three year life and 20% at an eight year life. If fuel oil costs $1.00 per gallon and electricity costs $.08 per KWH, then from the previous example, the annual savings in fuel costs would be $200 and the annual cost of operation during the heating season, assuming two 30w fans, would be $28. This yields an annual savings of $172 which allows a maximum initial investment of $430 for a three year pay back period or $860 for an eight year pay back period. Of course, higher efficiencies or rapidly inflating energy costs would make the investment more attractive.

CONCLUSIONS

Installing well sealed vapor barriers to make structures "air-tight" for energy conservation and to prevent moisture migration into building materials will become an even more common practice in the north. As a building becomes tighter, it will be necessary to provide mechanical ventilation to maintain air quality. Intergrating an air-to-air heat exchanger into the ventilation system is a means of reclaiming a portion of the heat energy that otherwise would be wasted in the exhaust air stream. The initial and operating costs of these units must be evaluated and compared to the heating cost of ventilation air. The potential savings in northern climates is greater due to the longer and more severe winter. Four types of heat exchangers have been described that would serve this function. Several units are already on the market and design drawings are available for a build-it-yourself project.

Generally, the units tested in this study had similar efficiencies. However, each unit has several advantages and disadvantages, such as, size, frost problems, cleanability, cost, etc.

The following recommendations are made based on the experience and results of this research project:

1. Ventilation of buildings is necessary to provide sufficient outdoor air to maintain air quality for a healthy environment. For tight structures, the minimum required outdoor air may not be provided by natural means (air infiltration by wind and stack effect) and a forced (mechanical) ventilation system will be needed. At present, the minimum safe air exchange rate in residential structures has not been determined, therefore it is recommended that .5 ACPH is desirable to prevent the build-up of contaminants as well as alleviate moisture related problems.

2. If a forced ventilation system is necessary, then the economics of installing and operating an air-to-air heat recovery device must be evaluated before specifying the unit. Generally, a heat recovery device should reclaim at least 60% to 70% of the thermal energy in the exhaust airstream. If a building does not require forced ventilation, installing and operating an air-to-air heat exchanger will only result in increased energy usage.

3. Frosting and/or condensation in air-to-air heat recovery devices can be a problem in units operated in cold regions. Frost threshold temperatures for several of the units tested during this project have been presented. Schemes for alleviating or eliminating the frosting problem have also been discussed. However, all these techniques result in a decrease in the heat recovery effectiveness.

4. Fans, dampers, wiring and controls must be designed to withstand the rigors of cold weather operation or be located within the heated space to avoid exposure to extreme low temperatures. Unit installation design must include cold weather considerations.

5. If an airto-air heat recovery device is being considered for a building, then savings can be realized by designing the mechanical systems to accept the unit and architectural systems to house the unit, either at the time of construction or as a later retrofit.

A recently published book, Air-to-Air Heat Exchangers for Houses, by W.A. Shurcliff, covers the principles, performance, cost, and suppliers of residential sized units. The book is available from the author, $14 prepaid, 19 Appleton Street, Cambridge, MA 02138.

Acknowledgement

The Department of Transportation/Public Facilities, State of Alaska, has provided the funding for this engineering research project. Scott Bell, a mechanical engineering student at the University of Alaska, performed the fabrication of the locally built units as well as some of the initial testing.

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7. Besant, R.W., R.S. Dumont, and D. Van Ee, "An Air-toAir Heat Exchanger for Residences." Engr. Bulletin No. 387 ($ 2), Extension Division, University of Saskatchewan, Saskatoon, June, 1980.

8. Enercon - Fresh Air Heat Exchanger, Enercon Industries Limited, Regina, Saskatchewan, S4P 2K6, 1980.

9. Humid - Fire Model No. 11, Passive Solar Products Ltd., Saskatoon Saskatchewan, S7L 6A4, 1980.

10. Lossnay Air-to-Air Enthalpy Heat Exchanger. Mitsubishi Electric Industrial Products, Melco Sales, Inc., Compton, Calif. 1980.

11. Amode J.O. and K.T. Feldman, "Preliminary Analysis of Heat Pipe Heat Exchangers for Heat Recovery," ASME 75-WA/HT-36, ASME Winter Annual Meeting, Houston, TX 1975.

12. Shoukri, A., "The Use of a Regenerative Air-to-Air Rotary Heat Exchangers for Heat Recovery in Residential Ventilation Systems," ASME 79-WA/HT-32, ASME Winter Annual Meeting, New York, 1979.

13. Ritchie, B., Personal Communication, Fairbanks, AK., April, 1982.

14. Leonard, L.E., Personal Communication, Fairbanks, AK., Feb., 1982.

APPENDIX I

LIST OF COMPANIES MANUFACTURING/SELLING SMALL SCALE AIR-TO-AIR HEAT EXCHANGERS

1. Rotary Exchanger

   Berner International Corp.
   12 Sixth Road
   Woburn, MA 01801
   (617)933-2180

2. Cross Flow Exchanger

   Mitsubishi Electric Sales America, Inc.
   (Melco Sales)
   3030 E. Victoria St.
   Compton, CA 90221
   (213)537-7132

3. Heat Pipe-Plate

   Q-Dot Corp.
   726 Regal Row
   Dallas, TX 75247
   (214)630-1224

4. Counter Flow Plate

   Des Champs Laboratories Inc.
   P. O. Box 348
   East Hanover, NJ 07936
   (201)884-1460

5. Enercon Industries Ltd.
   2073 Cornwall St.
   Regina, Sask.
   Canada S4P 2K6
   (306)585-0022

6. Conservation Engergy Systems Inc.
   Box 8280
   Saskatoon, Sask.
   Canada, S7K 6C6
   (306)655-6030

7. Temovex AB
   Box 111, S-265-01
   Astorp, Sweden
   042 550 20

8. Air Changer Co., Ltd.
334 King St. East
Toronto, Ont.
Canada, M5A 1K8
(416)863-1792

9. Besant-Dumont-Van Ee
Dept. of Mechanical Engineering
University of Saskatchewan
Saskatoon, Sask.
Canada, 57N 0W0
(Instructions for a Do-It-Yourself unit)

10. Temp-X-Changer
Div. of United Air Specialists
4440 Creek Road
Cincinnati, OH 45242
(513)891-0400