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An Efficient Enthalpy Exchanger for Economical Ventilation.

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Synopsis

A cross-flow polymer membrane enthalpy exchanger has been designed which provides both heat recovery and moisture dissipation in the ventilation of living spaces. The exchanger is of benefit in providing fresh air during both cooling and heating seasons with minimum loss of energy. A prototype of the enthalpy exchanger has been constructed and tested.

The air leakage of the equipment has been found to be negligible; that is, the two air streams are indeed non-mixing. Testing for efficiency of the equipment involved the measurement of dry bulb and wet bulb temperatures in each of the four ports. The measured temperature values were used to calculate the efficiency of the exchanger. The results show a total efficiency of 72%, with 71% sensible heat recovery and 74% latent recovery. These efficiencies were achieved at an air flow rate of 1.7 m^3/minute (60 cfm). The measured moisture transfer rates exceed the predicted moisture transfer rates by factors of two and three.

Background

An enthalpy exhanger is a device which allows the beneficial transfer of heat and moisture between two non-mixing air streams. Such devices are useful to reduce the energy penalty (sensible and latent) for providing fresh air to building occupants. Enthalpy exchange is particularly desirable in hot, humid climates, where the difference between indoor and outdoor dry bulb temperature may be small, but the difference in moisture levels may be great. The complaint is often heard "It's not the heat, it's the humidity." Barringer and McGugan (1989) conclude that "when reduction of cooling cost is the main consideration, exchangers with moisture recovery are preferable to sensible HRVs." Enthalpy exchange is desirable in heating climates, as well, in order to reduce the potential for frost closure of the device.

There are several types of air exchange ventilators in use. Most air-to-air exchange ventilators are sensible heat ventilators that permit no moisture transport between the two air streams. There are several types of enthalpy (sensible + latent) exchangers in use. These include rotary wheel (with and without dessicant), plate, and porous plate.

Description

The apparatus described here is a prototype of a plate-type enthalpy exchanger which uses a spun-bonded polyethylene membrane to separate the two air streams: intake-supply and return-exhaust. The membrane is identical to the membrane in common use as a house-wrap and as a "breathable" envelope material. The product characteristics, derived from available product literature, are given in **Table 1**. The membrane product was selected because of its high water vapor permeance and its low air leakage characteristics.

The membrane is folded and wrapped in aa fashion to maximize the exposure of the membrane sheet to the passage of air in both directions. The wrapping is schematically illustrated in **Figure 1**. The sheet is folded in successive loops; then the edges are sealed, creating the separated chambers for air passage. In the prototype which was constructed for this research, the joints are sealed using an adhesive tape; a refinement of the design would involve the use of heat sealing of the joints. This design is a counterflow design, which has inherently higher design efficiencies than tandem flow.

It may be evident from the schematic illustration that the membrane is not maintained in a perfect flat plane but is skewed. This inevitably results in irregularly shaped passages of varying opening width, and offering varied resistance to air movement within the device. It is also evident, given the rectangular shape of the exchanger, that the membrane surface is not of optimally efficient shape. The corners at right angles should not be expected to have uniform air movement. The seams in the membrane were joined using a tape specified for use with the membrane. When taped, the folded and wrapped membrane forms a core.

The membrane core was attached to end chambers of clear rigid acrylic plastic. The apparatus is shown in the illustration of **Figure 2**. Axial single-speed fans were attached to the intake and return chambers. The effect of using fans on these two ports is to equalize the air pressures across the membrane and to keep the apparatus in positive pressure with respect to ambient pressure during operation. It was assumed that the apparatus operating in positive pressure would be less likely to cause constricted air passages.

Test methods

In the U.S., heat recovery ventilators (HRVs) are tested by the Home Ventilating Institute, using a standard test method based on Canadian Standards Association CAN/CSA-C439-88 "Standard Methods of Test for Rating the Performance of Heat-Recovery Ventilators" At present, these tests are conducted in only one laboratory. The standard test method involves measurement of:

- temperature and humidity at the four ports,
- cross leakage,
- energy consumption of fans, controllers, etc., and,
- net air flow rates and static pressures developed.

The standard presents a method for using the measured values to calculate the efficiency of the exchanger. There are three sets of environmental conditions for the standard test of heat recovery efficiency according to the HVI standard:

- indoor air at 71.8°F (22.1°C) and 40% relative humidity, "outdoor" air at 32°F (0°C), and,
- indoor air at 71.8°F (22.1°C) and 40% relative humidity, "outdoor" air at -13°F (-25°C), and,
- indoor air at 75°F (23.9°C) and 50% relative humidity, "outdoor" air at 95°F (35°C) at 50% relative humidity (optional).

The first test is intended to measure performance during mild heating climate conditions, the second, under severe heating climate conditions, and the third under hot, humid outdoor conditions (cooling climate).

A facility at the University of Illinois was used to conduct tests on the prototype enthalpy exchanger. This facility was not equipped to maintain the standard conditions. The intended use of this equipment is primarily for hot, humid climate, so the first two climate conditions were not applied. The following tests were conducted on the prototype equipment:

- dry bulb and wet bulb temperature at the four ports,
- cross leakage, and
- net air flow rates.

The energy consumption was not metered, but rather was estimated from the electrical ratings of the two fans. A static pressure test was not conducted because it was felt that the test deviates significantly from the pressure regimen maintained by the equipment during operation, so that the results would be of little value. It is important to note that the methods used were *not* HVI standard methods, because the equipment available was not capable of achieving and maintaining the conditions of the standard test.

Dry bulb and wet bulb temperature

Type "T" thermocouples were used to measure temperature. A reference thermistor was used for all eight thermocouples in this test, and the EMF was converted to temperature using a fifth-order polynomial resident in the data acquisition unit.

Wet bulb temperature measurements were taken using aspirated psychrometry; that is, a cotton wick was placed around the junction of each wet bulb thermocouple, and the wick was maintained wet from a nearby water source. The moving air stream in the exchanger itself obviated the need for a separate air flow source. The contribution of moisture from the wet bulb thermocouple to the overall performance of the apparatus was assumed to be negligible. A schematic drawing of the thermocouple placement is shown in **Figure 3**.

Considerable difficulty was encountered in maintaining the wicks so that they neither dried out nor dripped water. Data which showed the wet bulb temperature converging toward the dry bulb temperature were not included in this study. Wet bulb psychrometry may be the source of considerable error. In the data used for calculation, it is assumed that there was sufficient water on the wick for proper aspiration. If indeed the wick lacked proper moisture, then the humidity of the air stream would be seen as erroneously high.

Leakage

Leakage was tested using methods developed in the testing for attic ventilation (Hinrichs 1962), that is, smoke testing. Though the drawbacks to this method are that it is messy and potentially destructive of membrane performance, it has the distinct advantage of allowing continuous concentration measurements at intervals as small as three seconds. A further advantage, in this case, is that the deposit of smoke particles on the membrane allows, after disassembly of the prototype, an inspection of the apparatus for efficiency. The leakage test setup is shown in **Figure 4**. Three solar spectrum pyrgeometers were used as sensitive photovoltaic cells. The test was conducted outdoors in the shade, using illumination from a clear sky. A dense smoke source (from a marine signal flare) was discharged into the return port. Concentrations of smoke in the supply and exhaust ports were measured using voltage output from the pyrgeometers. It is assumed that the reduction in voltage output from the pyrgeometer is directly proportional to the increase in concentration of smoke particles between the light source and the pyrgeometer.

HVI uses Exhaust Air Transfer Ratio (EATR) as the ratio of the leaked air to the total air supplied. In this instance, the EATR that was measured was the ratio of the exhausting air in the supply port to the gross exhausting air flow. It should be noted that the smoke leakage test was assumed to be a test that would damage the membrane, and so it was conducted only after all other tests were completed.

Illumination values were sampled at three-second intervals. Illumination values were gathered for five minutes both before and after the smoke test, and those values were averaged. The values taken during the one-minute test were averaged, and the two sets of values were compared. The opaquing at the return port, into which the smoke was injected, was found to be 42%. The opaquing at the exhaust port was found to be 40%. The opaquing at the supply port was negligible: 0.78%. There was no smoke visibly leaking into the supply port. Of course, there was no leakage into the intake port, which received its air mechanically from the outside. So, for purposes of this report, losses due to leakage into the supply port are ignored. The test did not measure leakage to the outside. Slight leakage to the outside could be seen, and the same is indicated in comparing the opaquing at the return port where the smoke was injected (42%) with the opaquing at the exhaust port (40%).

The aim of this test was to examine the equipment for gross leakage through cracks in the assembly. Porosity of the membrane to the crossover of air molecules was not tested.

Flow rate

The net flow rate was measured at the supply and exhaust ports using a velometer. The diameter of the port is 3 1/2 inches (89mm). The flow rate, calculated from the velometer reading (length) times the port opening area, is 60 cfm (1.7 m^3/min) in each direction. This flow rate may be considered adequate for one person in a dwelling, but would not be considered adequate for meeting the fresh-air requirements of a family of four.

Results

Data were collected during a two day period using environmental conditions limited to the capacity of the university facility. The "indoor" and "outdoor" conditions are shown in **Table 3**. Values for the dry bulb and wet bulb measurements during the test period were averaged. The averages were used to derive other psychrometric values, namely, humidity ratio and total enthalpy. Relations found in *ASHRAE Handbook of Fundamentals* (1989) were used in making these conversions. It may be noted that wet bulb psychrometry was appropriate for this analysis due to the similarity between wet bulb temperatures and enthalpy values, and due also to the uniformly moving air stream.

The output values are shown in **Table 4**. This table also contains the values used in the calculation of three efficiencies: sensible, latent and enthalpy. In each of these three cases, the efficiency is defined to be the ratio between the beneficial contribution by the equipment (Intake - Supply) compared to the range of indoor and outdoor condtions (Intake - Return). The HVI standard requires that these efficiencies be modified to become "net" efficiencies by multiplying the values by the Exhaust Air

Transfer Ratio. In the present test, that reduction was found to be negligible, so the efficiencies stand as calculated from the temperatures.

The findings are represented graphically in **Figures** 5 and 6. Calculation of the humidity ratio permits the results to be posted on a psychrometric chart, allowing the effect of the exchanger on the characteristics of the two air streams to easily illustrated. **Figure** 5 shows the process undergone by each air stream. It is apparent in viewing this chart that one cannot maintain the assumption that all heat and all mass being transferred in the apparatus is being transferred to the opposing air stream. Clearly, the moisture loss in the supply air stream is greater than the moisture gain in the exhausting air stream. This is due, most likely, to losses to the outside from the exhausting air stream. It may also be due to errors due to wet bulb psychrometry, as described above.

The same graphical method is applied in **Figure** 6, which illustrates the efficiencies derived from the calculations.

The predicted moisture transfer rate has been compared to the measured transfer rate. The calculations are shown in **Table** 5. It can be seen that the actual rate of moisture transfer exceeds the predicted rate by a factor of two in the exhausting air stream and by a factor of three in the supply air stream. This should not be surprising. Permeance testing of materials takes place under standard low rates of air flow over only one surface of a diffusing membrane. In this instance, there is a high rate of air flow over two surfaces. The difference in moisture transfer rates between the supply and the exhausting air streams is, at present, attributed to air leakage out of the apparatus from the supply air stream.

Discussion

Two concerns are often raised with regard to the performance of air-to-air exchangers with non-metallic cores: fire safety and indoor air quality. Fire safety concerns can often be addressed by separating the flammable core from possible sources of combustion, often by incorporating it in metal ductwork. Of course, fire movement through the ductwork must be addressed as well.

The core of an enthalpy exchanger should not be a harbor for irritating, allergenic or toxic microorganisms. In the development of an appropriate exchanger, it will be very important to study the conditions under which the membrane material supports the growth of harmful microbes. A long study period should predate any commercial introduction.

Further development of this apparatus should include the use of a replaceable core, which can be removed for testing.

Conclusions

A prototype of a cross-flow plate-type enthalpy exchanger has been constructed and tested. It is shown to be quite free of leakage. It is also shown to be quite efficient in both sensible and latent transfer. An apparatus with such performance may be of considerable benefit in reducing the energy penalty often associated with providing fresh air to occupants in conditioned spaces. This is particularly true in climates where cooling finds frequent use.

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Table 1: Membrane properties

Property	standard units		I-P units		Method
Water Vapor Transmission	2755	ng/(s*m^2*Pa)	48	perms	ASTM E-96
Porosity	17.6	sec/100cc-sq.in.			Gurley-Hill

Table 2: Apparatus characteristics

Characteristic	standard units		I-P	units
width	0.60	m	23.75	in.
length	1.14	m	45	in.
slot opening	0.02	m	3/4	in.
no. of openings in height	24 (two thicknesses)			
total membrane area	37.23	m^2	400.8	sq.ft.
flow rate	1.70	m^3/min	60	cfm

Table 3: Test conditions

Condition	standard units		I-P units	
Intake ("outdoor")				
dry bulb temperature	27.1	°C	80.8	°F
wet bulb temperature	20.6	°C	69.0	°F
Return ("indoor")				
dry bulb temperature	16.3	°C	61.4	°F
wet bulb temperature	13.5	°C	56.4	°F

Table 4: Results

Condition	standard units		I-P units	
Supply				
dry bulb temperature	19.5	°C	67.1	°F
wet bulb temperature	15.7	°C	60.2	°F
Exhaust				
dry bulb temperature	22.3	°C	72.1	°F
wet bulb temperature	16.8	°C	62.3	°F
Dry bulb$_{intake}$ - dry bulb$_{return}$	10.8	°C	19.4	°F
Dry bulb$_{intake}$ - dry bulb$_{supply}$	7.6	°C	13.7	°F
sensible efficiency	70.8%			
Humidity ratio$_{intake}$ - humidity ratio$_{return}$	.006			
Humidity ratio$_{intake}$ - humidity ratio$_{supply}$	.004			
latent efficiency	74%			
Enthalpy$_{intake}$ - enthalpy$_{return}$	4.97	kg cal/kg	8.96	Btu/lb
Enthalpy$_{intake}$ - enthalpy$_{supply}$	3.60	kg cal/kg	6.49	Btu/lb
enthalpy efficiency	72.4%			

Table 5: Analysis

characteristic	standard units		I-P units	
surface area	37.23	m^2	400.8	sf
permeance	2755	ng/(m^2*s*Pa)	48	perms
average vapor pressure difference	290	Pa	.086	inHg
predicted transfer rate	.0297	g/s	1651	grains/hr
volume flow rate	1.7	m^3/min	60	cfm
mass flow rate	121	kg_{air}/hr	266.7	lb_{air}/hr
humidity ratio loss: supply air stream	.0030			
humidity ratio gain: exhausting air stream	.0012			
measured transfer rate: supply air stream	0.100	g/s	5591	grains/hr
measured transfer rate: exhausting air stream	.0415	g/s	2306	grains/hr

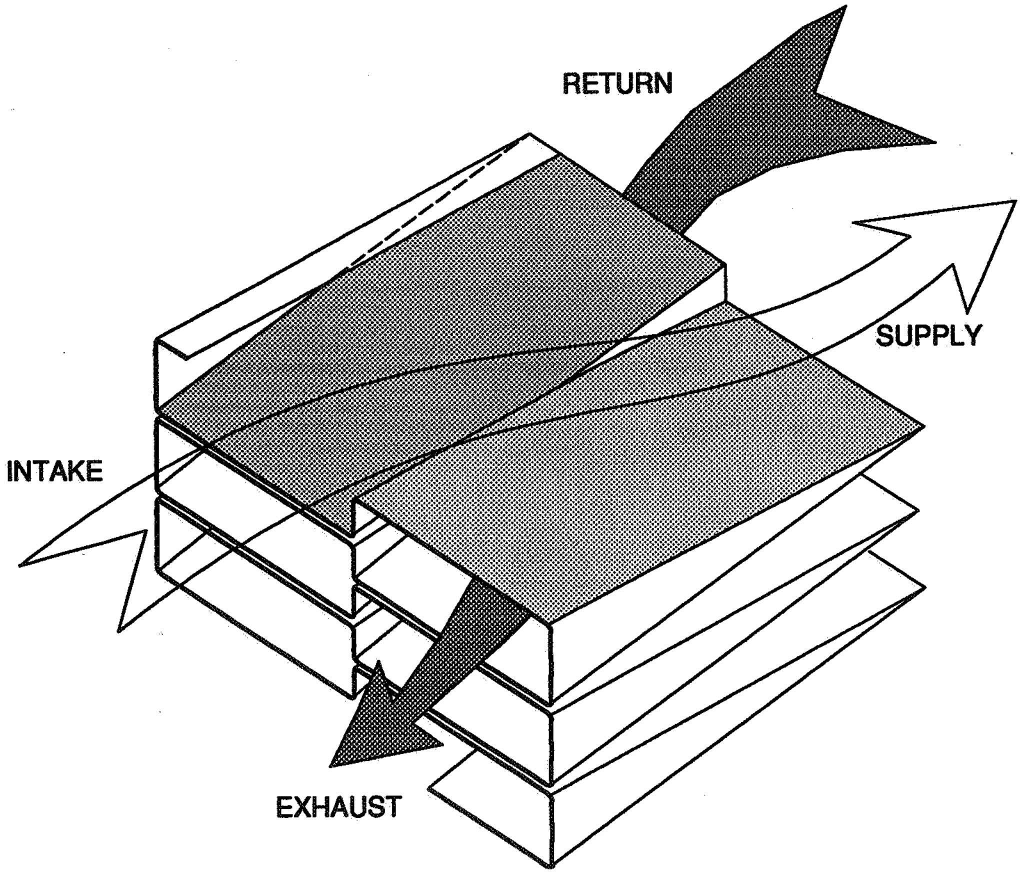

Figure 1. Schematic illustration of wrapping of membrane, showing the cross flow movement of the two non-mixing air streams.

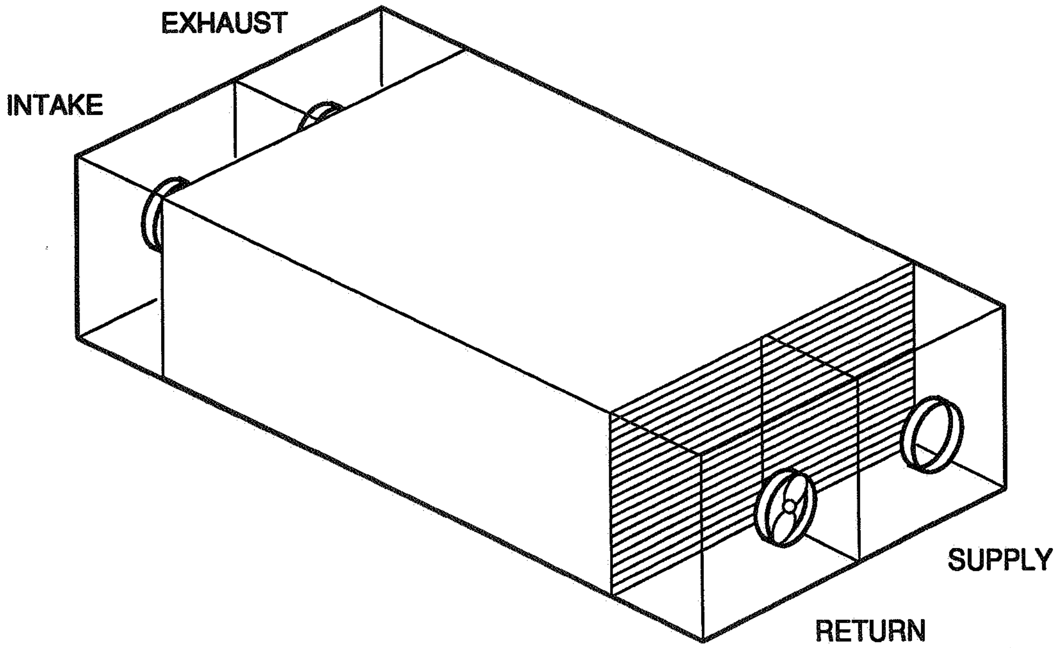

Figure 2. Drawing of enthalpy exchanger construction showing the membrane core, and acrylic plastic end ports.

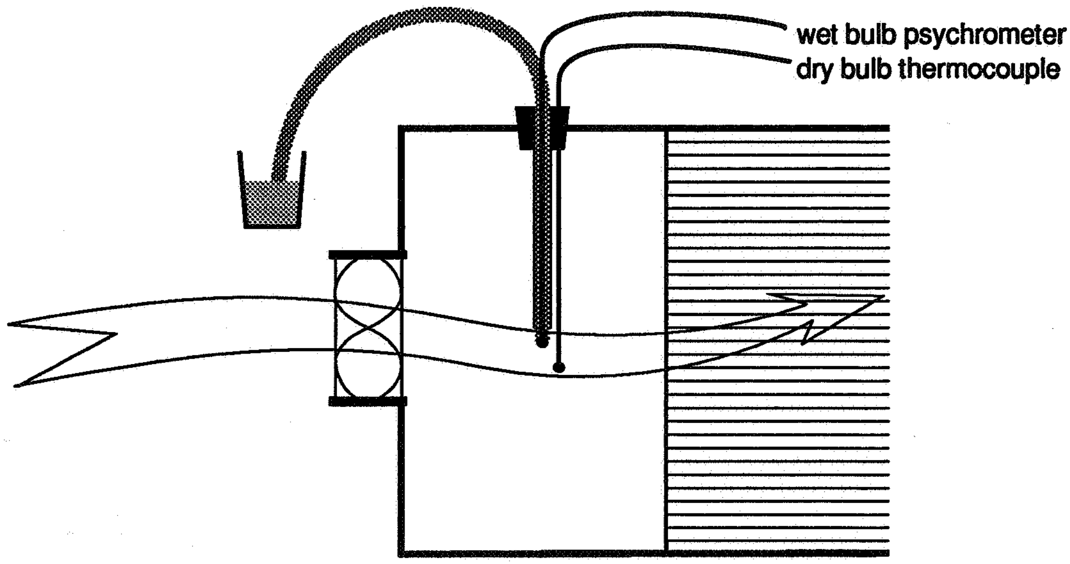

Figure 3. Test setup for measurement of wet bulb and dry bulb temperature. These measurements are used in the calculation of efficiency (sensible, latent and total enthalpy).

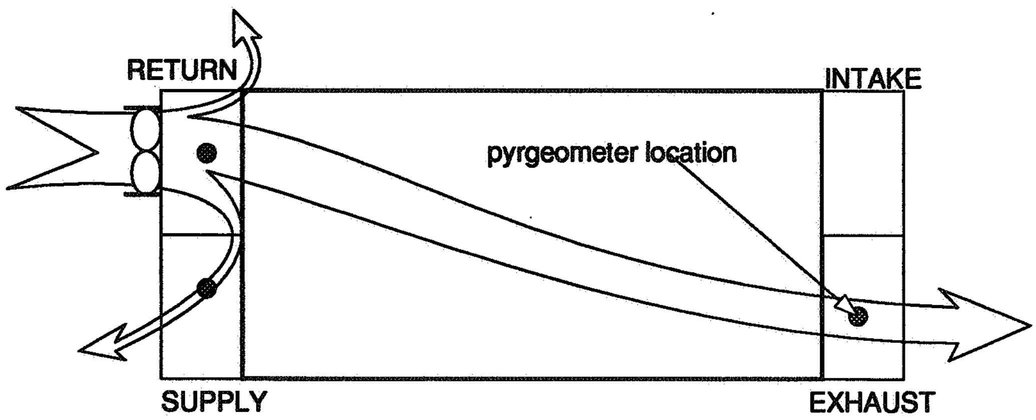

Figure 4. Schematic illustration of test setup for leakage. Smoke is introduced into the return air stream. The amount of opaquing measured by the pyrgeometers (photocells) is porportional to the concentration of smoke in that port. Leakage from the exhausting air stream into the supply stream was measured. Leakage to the exterior was not.

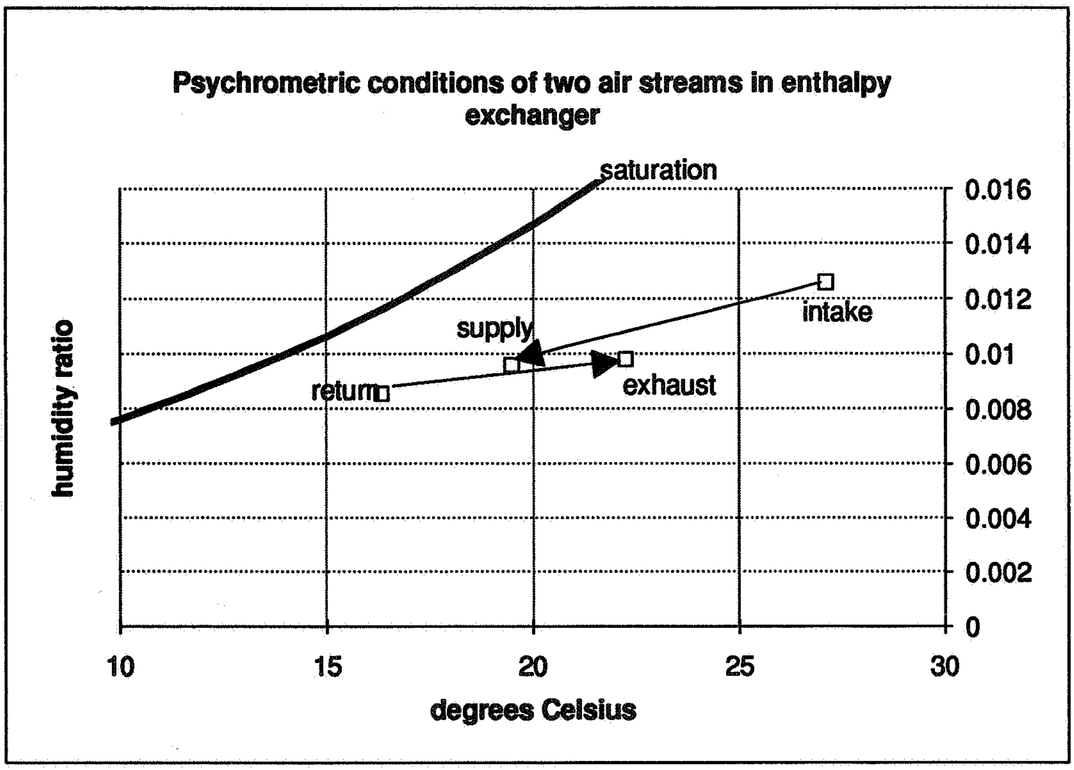

Figure 5. Chart showing psychrometric conditions of the two air streams.

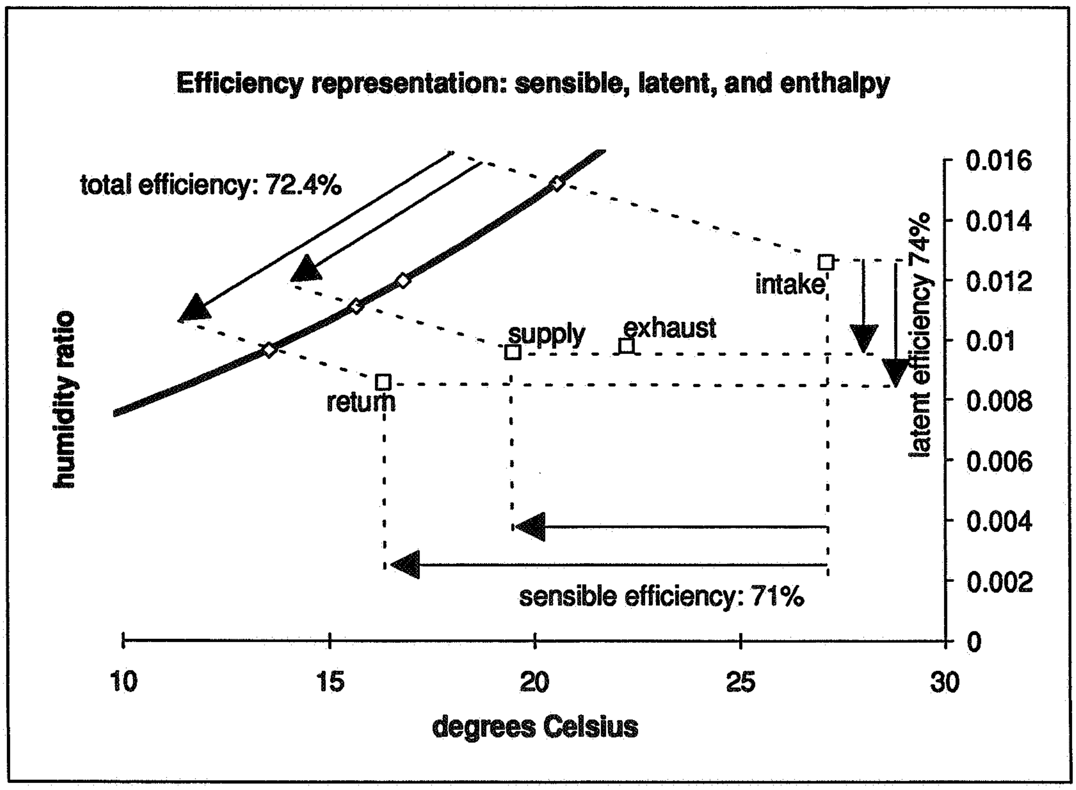

Figure 6. Chart showing the sensible, latent and total efficiencies of the enthalpy exchanger.