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EFFECTIVENESS AND PRESSURE DROP CHARACTERISTICS OF VARIOUS TYPES OF AIR-TO-AIR ENERGY RECOVERY SYSTEMS

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ABSTRACT

The performance of an HVAC system's air-to-air energy recovery exchanger is defined primarily by the exchanger's effectiveness and pressure drop. The effectiveness is dependent on several parameters such as the supply and exhaust mass flow rates and the energy transfer characteristics of the device. Because of this combination, performance data must be established for each individual type of device. This section presents the results of applying the methodology of ASHRAE Standard 84-78 in determining the performance of four types of devices: 1) coil energy recovery loop (closed run-around) system; 2) heat pipe heat exchanger; 3) fixed plate exchanger; and 4) the twin-tower enthalpy recovery loop (open run-around) system.

INTRODUCTION

The performance of an HVAC system's air-to-air energy recovery exchanger is defined primarily by the exchanger's effectiveness and pressure drop. The thermal performance of such heat exchangers is usually expressed in terms of their effectiveness in transferring: sensible heat (temperature change), latent heat (humidity change), and/or total energy (enthalpy change). The effectiveness, ε, of an energy exchanger is defined as follows:

$$\varepsilon \equiv \frac{\text{actual transfer for the device}}{\text{maximum possible transfer between the airstreams}} .$$

The effectiveness of a particular air-to-air energy recovery device is a function of several variables, including the supply and exhaust mass flow rates and the energy transfer characteristics of the device. Because of this combination, performance data must be established for each individual type of device.

Under ASHRAE RP-133, Standard 84-78, "Method of Testing Air-to-Air Heat Exchangers" (1) was developed and applied for determining the performance of the rotary air-to-air energy exchanger, commonly called a heat wheel or thermal wheel. In 1975, RP-173 was initiated to determine the applicability of the test method to other types of air-to-air energy recovery systems.

The following sections present the results of these tests conducted to determine the performance of: 1) the coil energy recovery loop (closed run-around system; 2) the heat pipe heat exchanger; 3) the fixed plate exchanger; and 4) the twin-tower enthalpy recovery loop (open run-around) system.

CLOSED RUN-AROUND SYSTEM

The first type of energy recovery system that was tested was the closed loop run-around system.

A typical closed run-around system consists of a coil (or coil bank) located in the exhaust

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air stream, and a coil (or coil bank) located in the make-up air stream. Glycol or some other sensible heat fluid is circulated through the piping connecting the two coil banks. The flow of glycol through the coils is counter-flow to the air temperature rise, or drop, across the coil. Fig. 1 is a schematic of the closed loop system.

The primary components in the circuitry are the two heat exchangers, a pump and expansion tank. Optional equipment which can be installed in this system are an air separator and/or a three-way temperature control valve. The purpose of the three-way control valve is to bypass fluid past the supply air coil when frost prevention on the exhaust air coil during winter operation is desired and also to prevent over heating of supply air during coil days when air conditioning is still required.

This system employs finned tube air-to-liquid heat exchangers, one of which is positioned in the supply air duct with the other located in the exhaust air duct. The liquid transfer medium is pumped from one exchanger to another, transferring energy between the two. The heat exchangers are 8-row, 8-circuit horizontal airflow coils, constructed of copper tubing with aluminum fins encased in a galvanized steel frame. The tested system incorporated the three-way bypass valve for frost control. Orifice plates were installed on both discharge branches of the circulating pump to measure both primary and bypass flow. The control of the three-way bypass valve was done pneumatically through a dual input controller. The sensing points were water temperature entering exhaust coil and air temperature leaving supply air exchanger. The heat transfer medium pumped through the circuit was a 35% ethylene glycol-65% water solution. The solution mix was chosen to prevent solution freeze-up during simulated winter operating conditions.

The first test set taken of the closed loop run-around system was the static pressure drop at various airflows. This test was conducted with no heat or moisture being added or removed from the system. Static pressure was recorded as airflow rates changed from 30% of rated face velocity, standard air, to 150% of rated flow, in 15% increments. Fig. 2 presents the results of the pressure drop series of runs.

The effect of airflow rate on the sensible effectiveness was determined next. With airflow rates through each coil being equal and glycol flow rate being kept at a constant 10 gpm (0.63 ℓ/s), airflow was varied from 35% to 150% of rated maximum. This test was conducted with supply air conditioners simulating both summer and winter conditions. The sensible effectiveness of the closed loop run-around heat recovery system increases as the face velocity of the heat exchanger decreases. Fig. 3 graphically portrays the variation in sensible effectivencss with changes in airflow rate through the energy recovery system.

The third test performed on this test unit was the effect of varying glycol flow rate on sensible effectiveness. The parameters that were kept constant were airflow at 2000 scfm (0.94 m^3/s), supply air temperature at 35 F (1.7°C), and exhaust air temperature and humidity at 75 F (23.9°C), 50% rh. The glycol flow rate was varied from 4 gpm (0.25 ℓ/s) to 22 gpm (1.39 ℓ/s) and sensible effectiveness was determined. Fig. 4 shows the effect of the glycol flow rate on thermal performance and substantiates the manufacturer's recommendation of 10 gpm (0.63 ℓ/s) flow. The effect of the transfer fluid flow rate or sensible effectiveness can be maximized. Initially at low flows sensible effectiveness increases with increase in fluid flow rate. As fluid flow rate continues to increase, sensible effectiveness reaches a maximum and then decreases slowly with further increase in transfer fluid flow rate.

Sensible effectiveness values were next determined for various ratios of supply air flow to exhaust air flow. With supply air flow being held constant exhaust airflow was varied from 50 to 100% of the supply airflow rate and sensible effectiveness was determined. Supply air conditions were held at 35 F (1.7°C) and ambient rh while exhaust air conditions were held constant at 75 F (23.9°C) and 30% rh. Glycol flow rate was also held constant at 10 gpm (0.63 ℓ/s). This same test was repeated at different values of constant supply airflow. Fig. 5 shows the effect of unequal airflows on the thermal performance of the energy recovery system. Data has been obtained for rated flow, one-half rated flow, and one and one-half times rated flow. Unequal airflow rates can be seen to affect sensible effectiveness by increasing effectiveness as the difference in airflow rates becomes greater. As the ratio of airflow rates, which is always greater than one by definition, increase so does the sensible effectiveness increase.

The changes in sensible effectiveness were also determined as they were affected by variances in supply air temperature. With airflow rates being held constant and equal for both supply and exhaust sides and exhaust temperature being held constant at 75 F (23.9°C) supply air temperature was varied between simulated extreme summer and winter conditions. Fig. 6

shows the effect of supply air temperature on thermal performance.

A final test was conducted to determine the effect of exhaust air humidity ratio on the sensible effectiveness of the system. While airflow rate, glycol flow rate, supply air conditions, and exhaust air temperature were held constant the moisture content of the exhaust air stream was varied. The purpose of this test was to observe the effect of condensation on sensible effectiveness. The effect of humidity on the sensible performance of the closed type of run-around energy recovery system is demonstrated by the results shown in Fig. 7.

The closed loop run-around heat recovery system is a sensible heat recovery unit. Only when the exhaust air stream is highly saturated with moisture will one see any latent transfer of energy and that will be only due to condensation or frosting. There is no means whereby water vapor can be transferred from one airstream to the other. The sensible effectiveness can be seen to increase as the dew point temperature of the airstream exceeds the surface temperature of the heat exchanger. At this point the condensation of moisture on the surface of the heat exchanger adds latent heat to the transfer fluid.

Early in the test schedule, two test runs were repeated specifically to check for reproducibility of results. Two tests were each repeated with the following consistency in results:

Test Condition A: $\varepsilon_s = 0.612$ (1st run) & $\varepsilon_s = 0.611$ (2nd run)

Test Condition B: $\varepsilon_s = 0.549$ (1st run) & $\varepsilon_s = 0.546$ (2nd run)

HEAT PIPE HEAT EXCHANGER SYSTEM

The heat pipe air-to-air heat exchanger used in this investigation was constructed in the form of a plate fin coil heat exchanger without any return bends or headers. The heat pipe heat exchanger consisted of a number of actual heat pipes staggered on equilaterial centers thus giving center-to-center spacing transverse to the flow direction and center-to-center spacing in the direction of flow.

The actual heat pipes are constructed with seamless aluminum alloy tubing with aluminum alloy fins. A vertical partition is installed to provide a positive barrier preventing the supply airstream from being cross-contaminated with the exhaust airstream. The refrigerant contained in the heat pipes for this application was dichorodifluiromethane (R-12). Fig. 8 is a photograph of the heat pipe exchanger tested in this program.

For this investigation a tilt control was used to regulate the heat transfer capability or capacity. The amount of heat that a heat pipe can transfer is dependent on its orientation relative to level position. Lowering the evaporator (hot) section below horizontal increases the heat transferring capability, as gravity assists in returning the condensate to the evaporator section to be reused. Conversely, raising the evaporator section above horizontal decreases the heat transferring capability, as gravity retards the return of the condensate to the evaporator section. Changing the orientation thus provides a positive means of regulating the heat pipe capacity and hence, the effectiveness of the heat pipe. This tilt feature may be desirable in three situations: 1) to change between heating and cooling seasons in those instances where the required heat transfer capacity in the winter requires other than level orientation; 2) to regulate the supply air temperature leaving the unit to prevent more recovery than is desired, and 3) to prevent frost formation on the warm face of the units' exhaust side at low outside air temperatures.

Fig. 9 shows the relationship between the pressure drop through the heat pipe heat exchanger and the airflow rate as the airflow rate was varied from 500 scfm to 2500 scfm (0.3 to 1.2 m^3/s). The supply and exhaust airflow rates were equal during each run, and those cases in which the supply air temperature was not 75 F (23.9°C) the airflow rates were corrected to a 75 F (23.9°C) air temperature. It can be seen that as flow rate increases, pressure drop through the heat pipe also increases. Also, during the variation of exhaust air humidity ratio, pressure drop increased with increases in humidity ratio at a constant airflow rate. This increase in pressure drop can be attributed to the condensation of moisture from the exhaust air on the finned surfaces of the exhaust side of the heat echanger, and therefore, restricting flow.

An increase in the airflow rate of the supply and exhaust air, where the supply and exhaust mass flow rates are equal, would be believed to have an effect on sensible effectiveness of the heat pipe heat exchanger due to the increasing air velocity accompanying this greater flow rate. Tests were performed on two different seasonal conditions at measuring station 1: summer conditions where supply inlet temperature was 95 F (35°C) and wet-bulb temperature was 72 F (22.2°C) and winter conditions where supply inlet temperature was 35 F (1.7°C) and ambient relative humidity. Fig. 10 shows that sensible thermal effectiveness decreases, as expected, with increased airflow rate.

The changes in the mass flow rate ratio, Msup/Mexh, would be expected to have a substantial effect on the sensible effectiveness since the relative energy content and transfer potential is a direct function of this ratio. The tests involving the variation of mass flow rate ratios were taken with three different constant supply airflow rates [0.940 m^3/s (2500 scfm), 0.820 m^3/s (2000 scfm), 0.470 m^3/s (1000 scfm)] while the exhaust flow rates were varied. The supply and exhaust temperatures and relative humidities were similar for all three cases. Fig. 11 shows the data obtained for all three cases. It is apparent from this data that the heat pipe heat exchanger becomes more effective as the mass flow rate ratio increases. It is also apparent that at a supply flow rate of 1000 scfm (0.470 m^3/s) this effectiveness is substantially higher than at 2000 scfm (0.820 m^3/s) and 2500 scfm (0.940 m^3/s).

The changes in density, viscosity, specific heat and thermal conductivity due to changes in temperature of the supply air may effect the sensible effectiveness of the heat pipe heat exchanger. For the normal operating temperatures encountered in air-conditioning applications, changes in the above variables are small, resulting in negligible changes in sensible effectiveness. Fig. 12 shows the sensible thermal effectiveness obtained from the heat pipe heat exchanger at a constant exhaust air temperature of 75 F (23.9°C) and relative humidity of 50% and a variety of inlet temperatures at an ambient relative humidity. The data shown in Fig. 12 indicates the negligible effect of the variation of supply inlet temperature on sensible effectiveness of the heat pipe heat exchanger; therefore, the heat pipe is equally effective in all seasons.

An increase in exhaust air humidity ratio would be expected to have an effect on the sensible effectiveness of the heat pipe heat exchanger for two reasons: 1) the increase in moisture content changes the air density; and 2) condensation releases latent heat which is available for sensible heating. A constant airflow rate of 2000 scfm (0.82 m^3/s) was maintained for both supply and exhaust. Temperatures were maintained at 35 F (1.7°C) supply and 75 F (23.9°C) exhaust while the supply air relative humidity was ambient and the exhaust air relative humidity was varied from 20% to 70%. The effects of exhaust air humidity ratio on sensible effectiveness can be seen in Fig. 13. In the range of exhaust air humidity ratios from 0.005 kg/kg to 0.012 kg/kg the sensible effectiveness remains constant; however, at a humidity ratio of 0.012 kg/kg condensation begins on the exhaust side of the heat pipe. This condensation releases more energy for transfer to the supply side of the heat pipe. Therefore, in Fig. 24 it can be seen that as the humidity ratio increases past 0.012 kg/kg the sensible effectiveness begins to increase. The higher the humidity ratio, the higher the sensible effectiveness due to larger amounts of condensation. Although the sensible effectiveness increases substantially towards 100% it would never reach 100%, since at some point the heat pipes' ability to remove condensate will reach a maximum at a point much lower than 100%. The uncertainty in sensible effectiveness remained constant for the range of humidity ratios at approx. 20%.

PLATE TYPE EXCHANGER

The plate type air-to-air heat recovery devices contain fixed surfaces for sensible heat transfer and may be made of metal, plastic or composition fiber materials. Airflow is usually crossflow or counterflow and theoretically there is no cross-contamination of the two airstreams. Counterflow provides the greatest temperature difference for maximum heat transfer, but crossflow can give more convenient air connections. The usual plate exchanger transfers sensible heat only, except when the temperature of one airstream is lower than the dew point temperature of the other, and there is direct condensation. No auxiliary pumps or drives are necessary and frost and temperature control can be accomplished with bypassing of the airstreams. Plate type devices require that the supply and exhaust airstreams be brought together. As previously mentioned the plates of the plate-type heat exchanger are commonly made of metal, plastic or fiber materials. A. A. Field (2) recently discussed the use of sheet glass and asbestos paper plates instead of previously used materials. The advantages of glass are said to be better corrosion resistance, easy cleaning and longer life. The asbestos paper plate is a completely new idea which originates from Japan. Unlike metal, plastic, or glass, asbestos paper is vapor permeable permitting mass transfer as well as simple conduction transfer; however, the integrity of the paper must limit air leakage due to pressure difference.

The physical configuration of the fixed plate exchanger tested is shown in Fig. 14.

Test results are shown in Fig. 15 through 18. Pressure drop performance is given in Fig. 15. The variation of sensible effectiveness with face velocity (airflow rate) is shown in Fig. 16. The effect of unbalanced airflow rate is presented in Fig. 17 for three different supply flow rates. As with other types of air-to-air energy recovery exchangers, there is a significant increase in effectiveness as the ratio of supply to exhaust increases. The effect of supply air temperature on the thermal performance of the plate exchanger is given in Fig. 18. Fig. 19 demonstrates the effect of condensation in the exhaust airstream on the energy transfer. At the higher exhaust stream humidity ratios, wet performance with some latent heat transfer improves the effectiveness of the exchanger.

OPEN RUN-AROUND ENERGY RECOVERY SYSTEM

The open run-around heat recovery system is a sensible and latent heat transfer system in which the energy transfer between airstreams is accomplished by alternately contacting the airstreams with a hygroscopic liquid, usually a halogen salt solution such as lithium chloride and water. The solution is continuously circulated between the supply air and exhaust air towers and acts as a vehicle for transporting heat and moisture from one airstream to the other. Pumps are used to circulate the solution between towers. Make-up water must be piped to the unit in order to control the solution concentration. The control of frosting and supply air temperature are accomplished much like that of the coil loop run-around; a three-way valve controls the flow of circulating fluid.

A front elevation sketch of the actual test unit for RP-173 is given as Fig. 20.

The test program for the open run-around (twin-tower enthalpy) exchanger included:

1. Pressure drop
2. Winter-variable, but equal, flow rates
3. Summer-variable, but equal, flow rates
4. Summer-unequal flow rates, 2000 scfm supply
5. Variable supply temperature
6. Variable exhaust humidity, winter
7. Variable exhaust humidity, summer
8. Latent only transfer, equal temperature

The planned test program was modified as testing proceeded. In the case of the open run-around, the original program consisted only of the first seven test sets. However, since this was the only exchanger investigated under RP-173 with the capability of mass (humidity) transfer as well as heat transfer, Test Set 8 was added. On the other hand, the range of unbalanced flows attainable was so small that these tests were abandoned.

Pressure drop characteristics of the twin-tower exchanger are shown in Fig. 21, both with and without circulation of the run-around fluid.

Fig. 22 shows the variation in thermal performance with changes in supply air temperature. Since the open run-around is an enthalpy exchanger, total effectiveness is used as the ordinate in Fig. 22.

Fig. 23 shows that the total effectiveness increases with an increase in humidity in the exhaust airstream. Also presented in Fig. 23 is the latent only performance of the exchanger. For this series of tests, both the supply inlet and exhaust inlet temperatures were maintained at 75 F (23.9°C). The potential or driving force for energy transfer in this case was the difference in humidity ratios between the two streams.

Fig. 24 gives total effectiveness vs sensible heat ratio for all runs.

CONCLUDING COMMENTS

Based on the results of the test programs, the main conclusion is that ASHRAE Standard 84-78 is essentially sound. However, there are several suggested changes to make the standard either more complete and/or accurate or more compatible with air-to-air systems other than the rotary:

1. Inaccuracies in pressure measurements have relatively little effect on the final result of this experimental investigation; however, inaccuracies in temperature measurement can have a very significant effect. It must be concluded from this that care is to be taken in both the choice of temperature measuring devices and in their use. Inaccuracies in wet- and dry-bulb temperatures are equally important since they have an effect on the latent effectiveness.

2. Consider the other corrections to ASHRAE Standard 84-78 which are the result of this test program, such as: accounting for auxiliary energy, redefining sensible effectiveness when condensing from one stream, static pressure testing prior to thermal testing, and providing procedures for hybrid units.

3. Delete from ASHRAE Standard 84-78 the references to rotary heat exchangers and system parameters that do not apply to the general field of air-to-air heat recovery systems.

REFERENCES

1. –, "Method of Testing Air-to-Air Heat Exchangers," ASHRAE Standard 84-78, ASHRAE, New York, 1978.

2. Field, A. A., "Heat Recovery from Air Systems," Journal of Heating, Piping and Air Conditioning, October 1975.

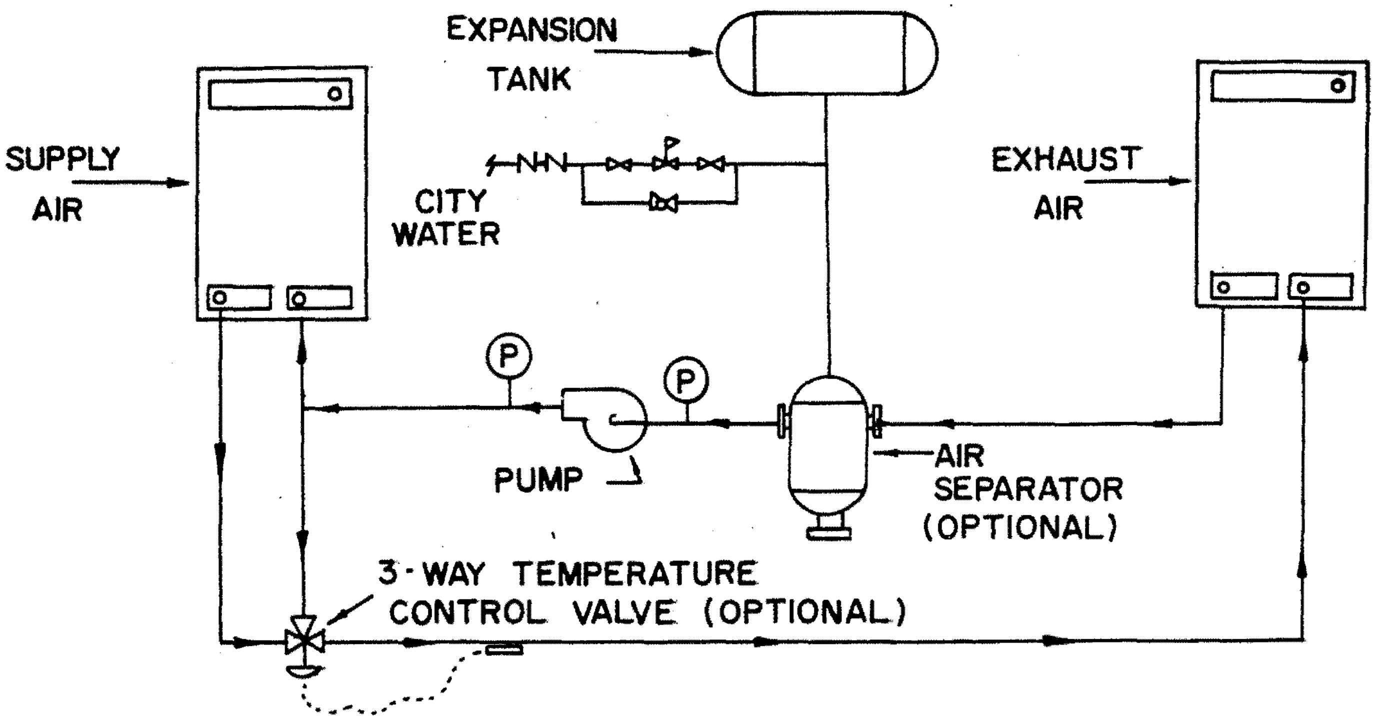

Fig. 1 Schematic of typical closed loop system

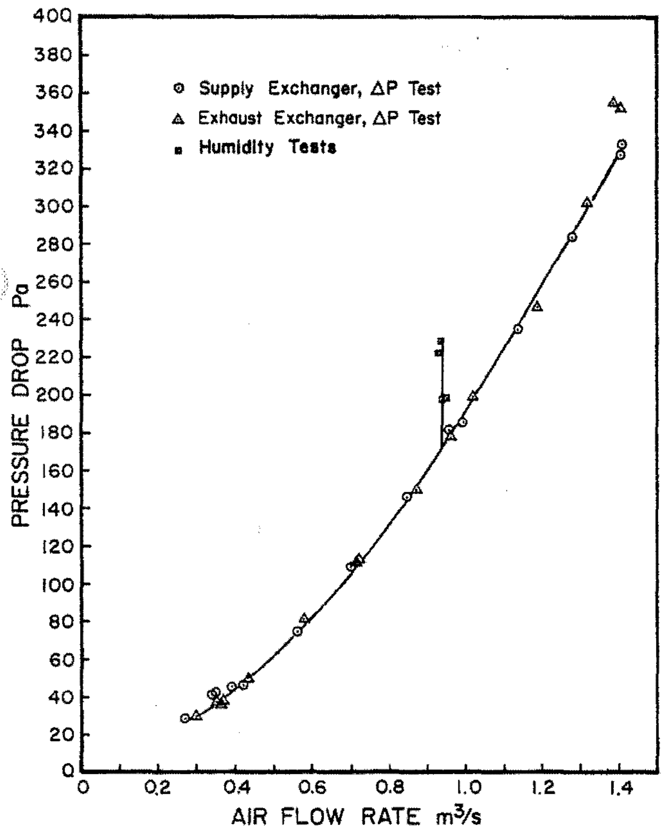

Fig. 2 Pressure drop of coil loop exchanger

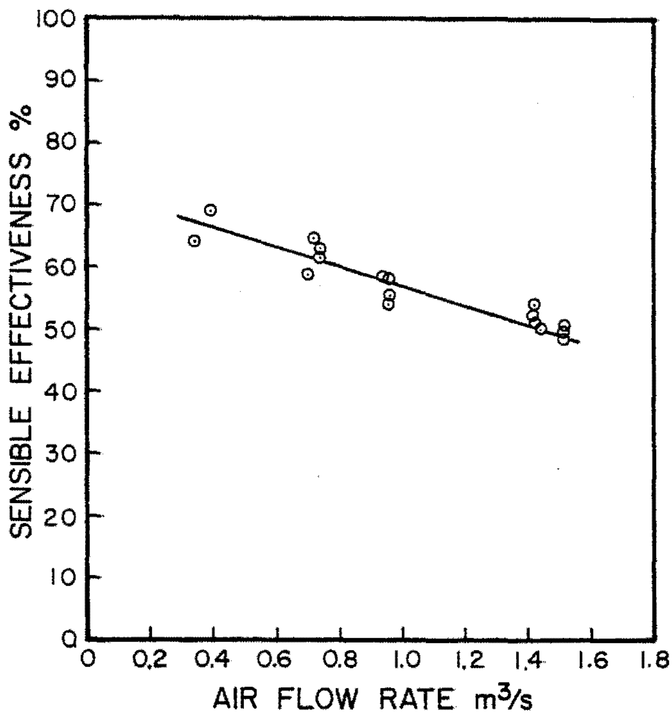

Fig. 3 Effect of airflow rate on effectiveness of coil loop system

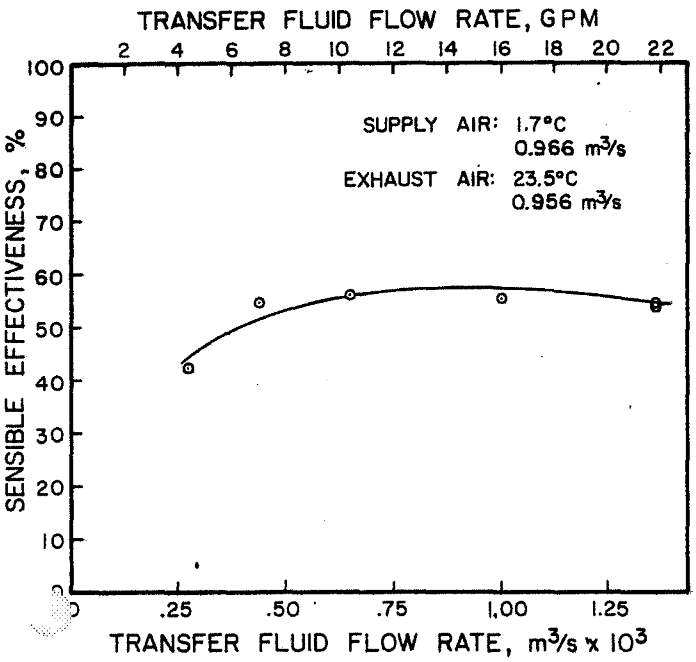

Fig. 4 Effect of transfer fluid flow rate on effectiveness of coil loop system

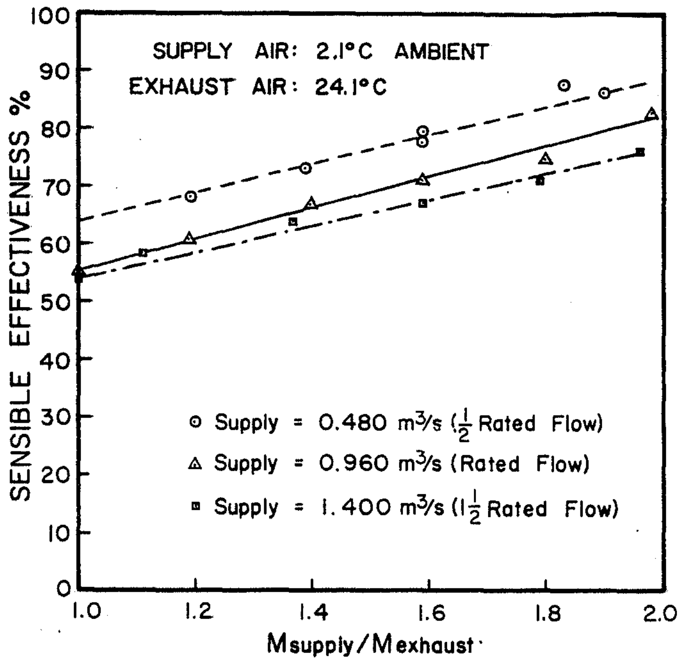

Fig. 5 Effect of unequal flow rates on effectiveness of coil loop system

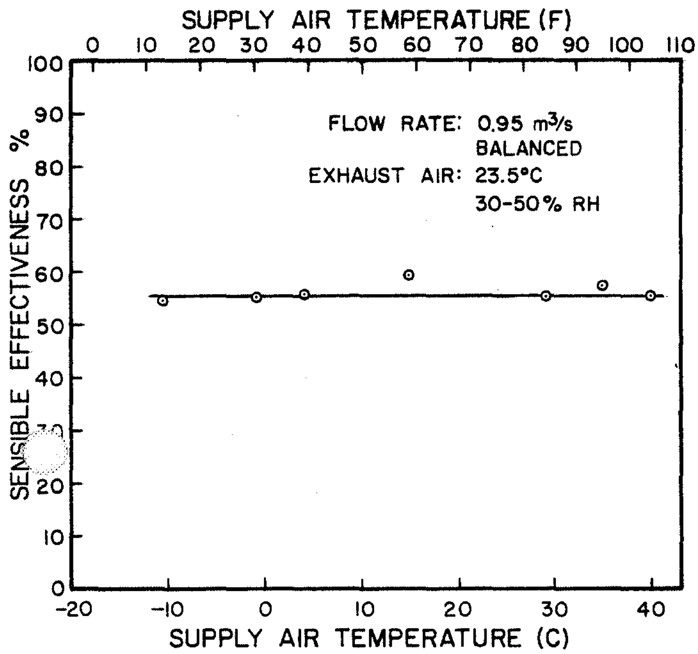

Fig. 6 Effect of supply air temperature on effectiveness of coil loop system

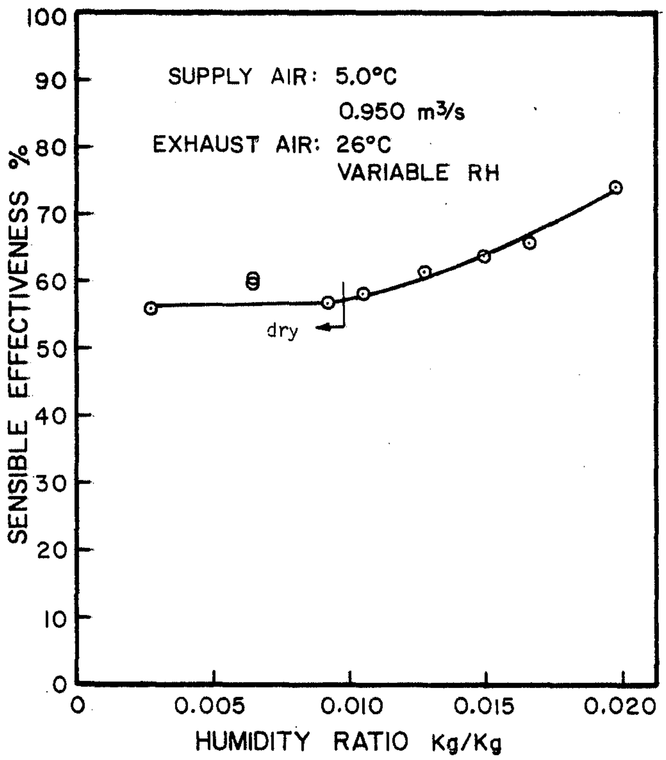

Fig. 7 Effect of exhaust air humidity ratio on effectiveness of coil loop system

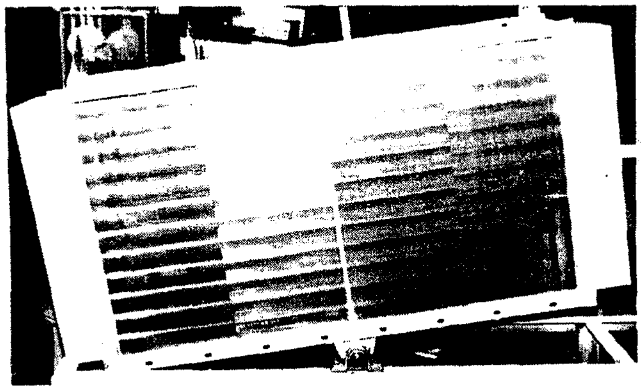

Fig. 8 Heat pipe exchanger with tilt control

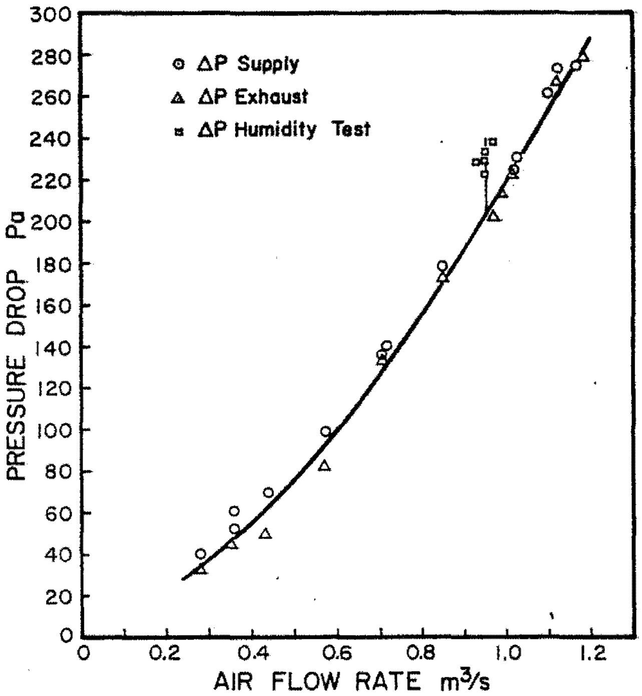

Fig. 9 Pressure drop of heat pipe exchanger

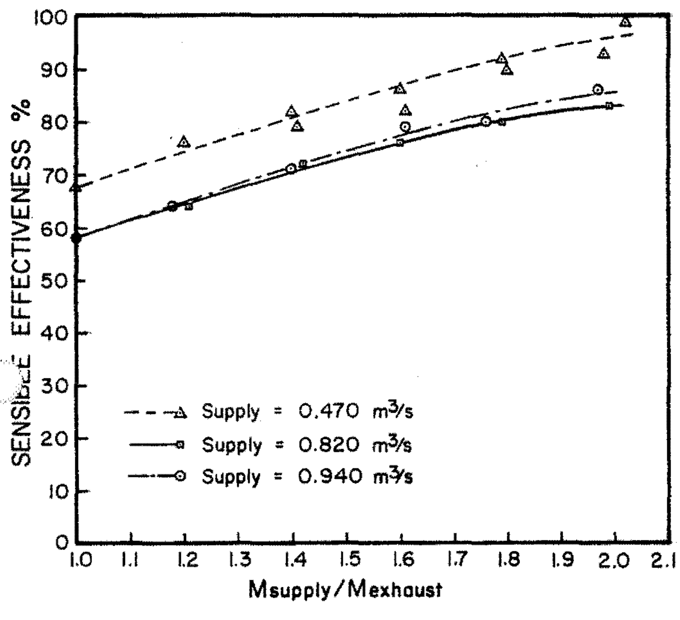

Fig. 11 Effect of unequal airflow rates on effectiveness of heat pipe exchanger

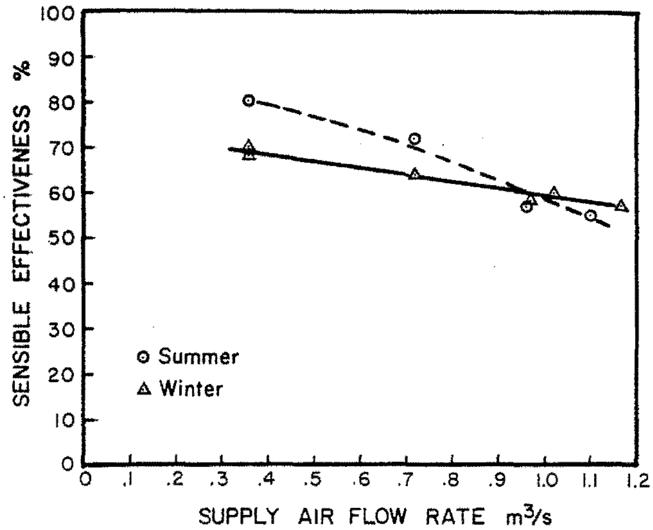

Fig. 10 Effect of airflow rate on the effectiveness of heat pipe exchanger

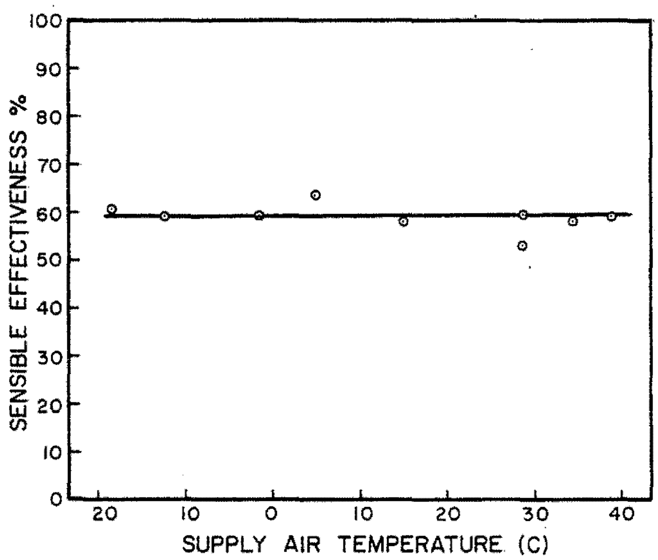

Fig. 12 Effect of supply air temperature on effectiveness of heat pipe exchanger

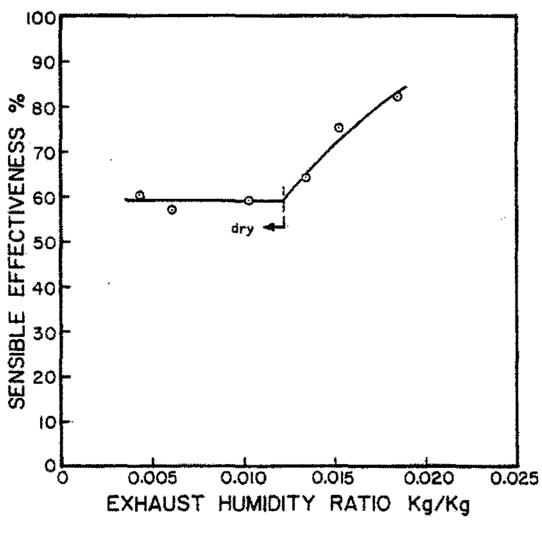

Fig. 13 Effect of exhaust air humidity ratio on effectiveness of heat pipe exchanger

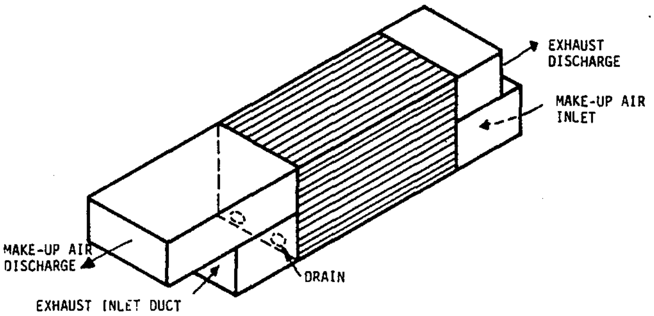

Fig. 14 Configuration of fixed plate heat exchanger

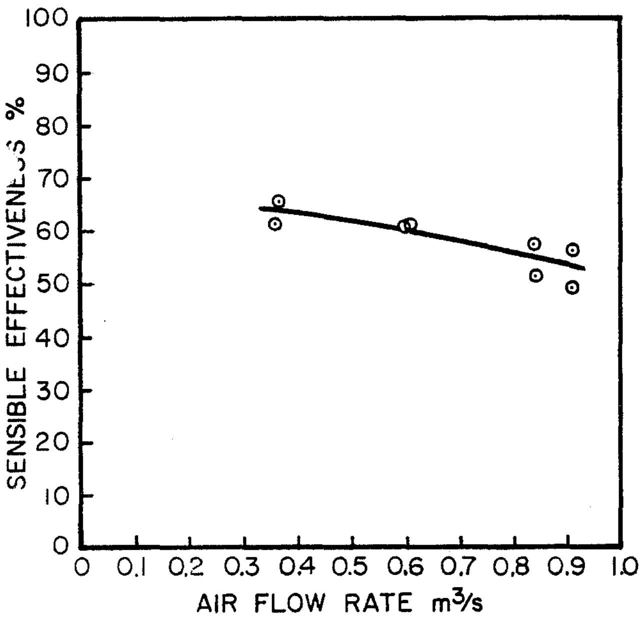

Fig. 16 Effect of airflow rate on the effectiveness of plate exchanger

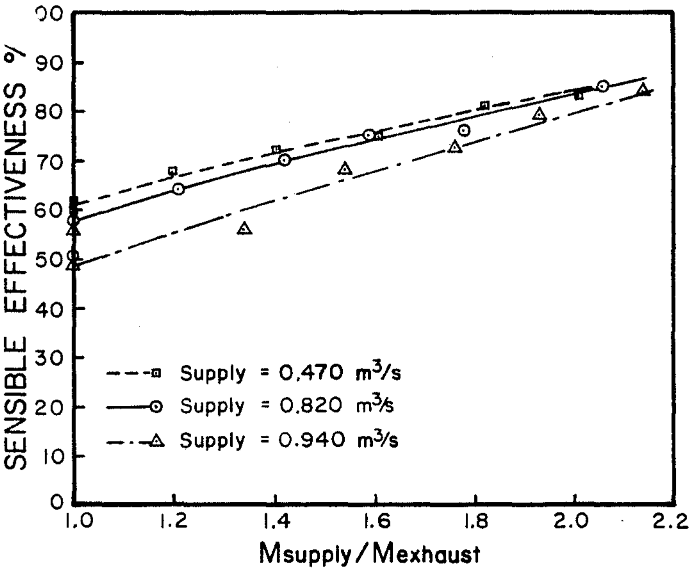

Fig. 17 Effect of unequal airflow rates on the effectiveness of plate exchanger

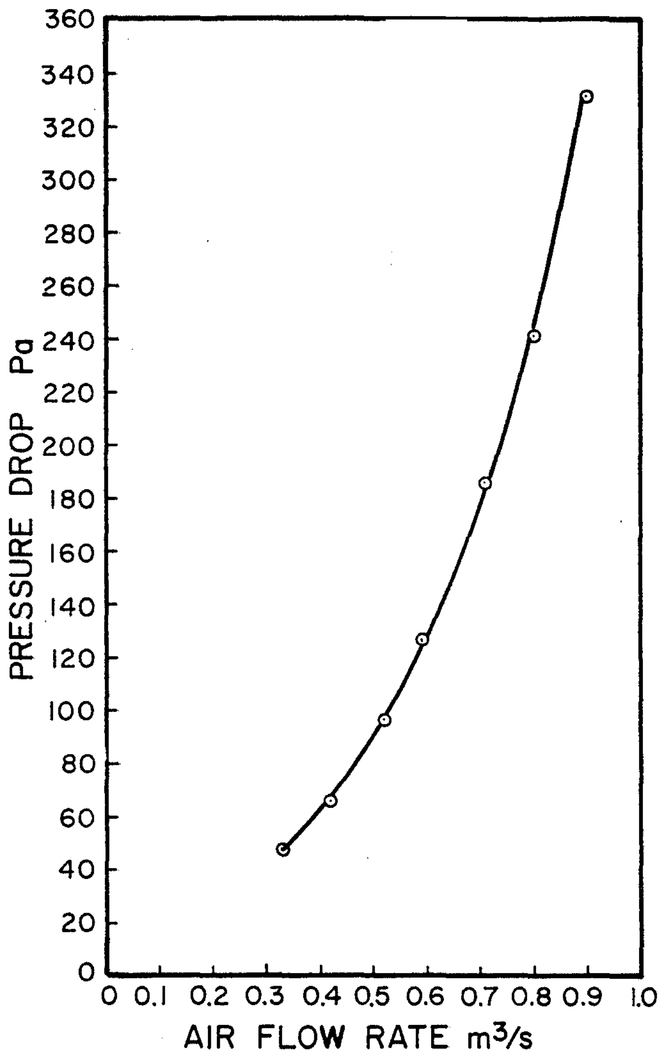

Fig. 15 Pressure drop of plate exchanger

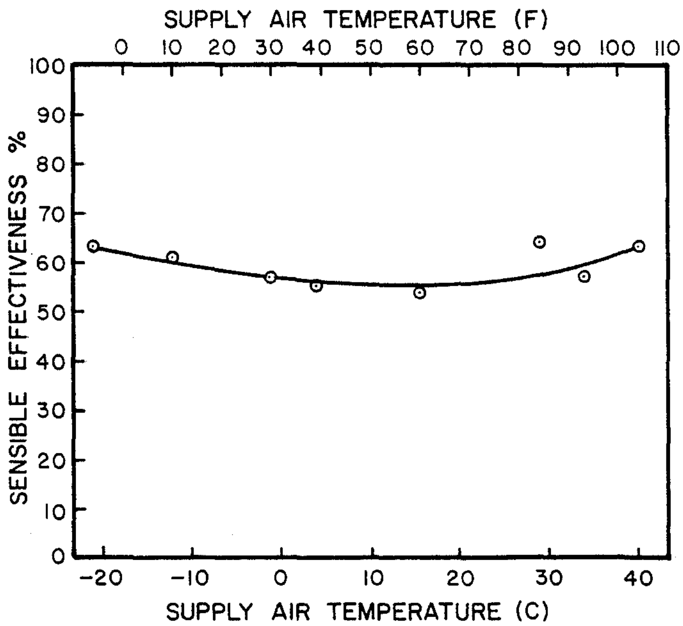

Fig. 18 Effect of supply air temperature on the effectiveness of plate exchanger

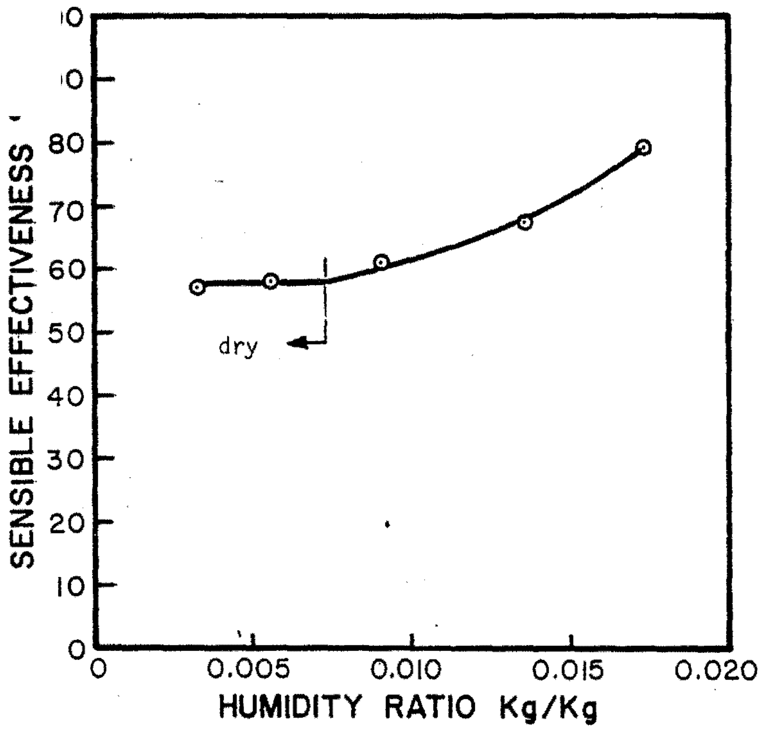

Fig. 19 Effect of exhaust air humidity ratio on the effectiveness of plate exchanger

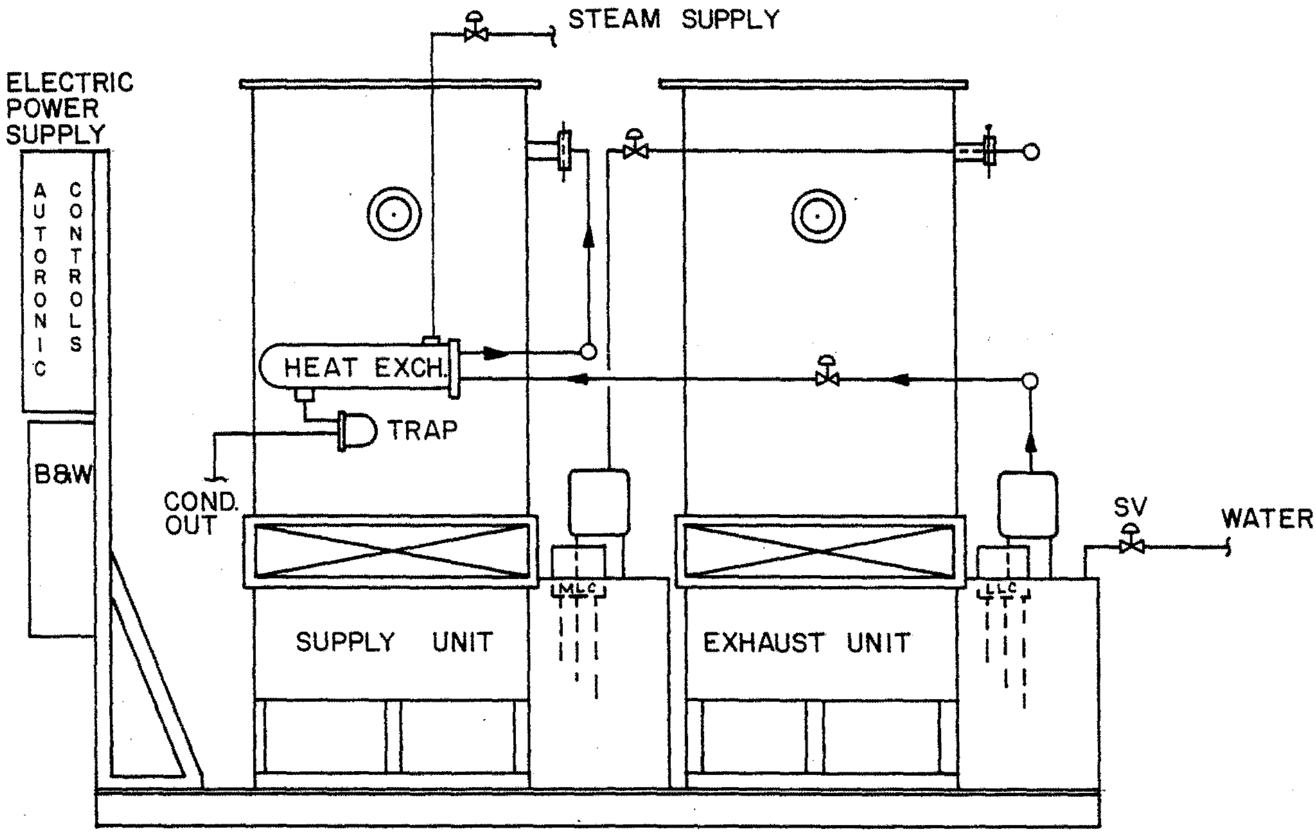

Fig. 20 Sketch of ASHRAE RP-173 twin-cell unit

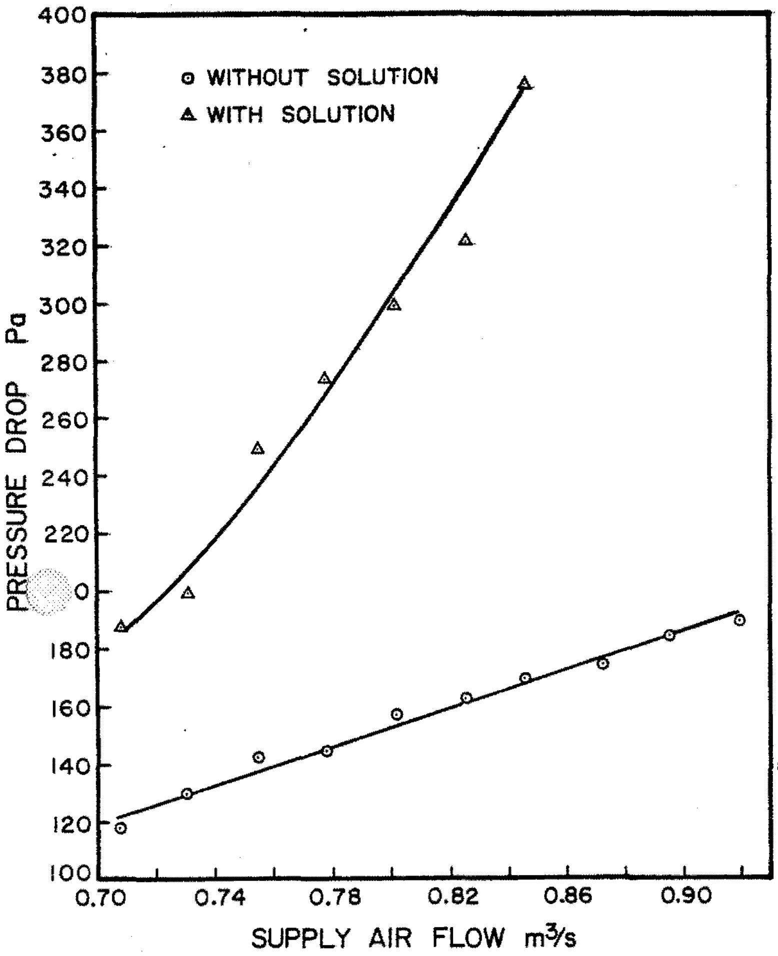

Fig. 21 Pressure drop of open run-around exchanger

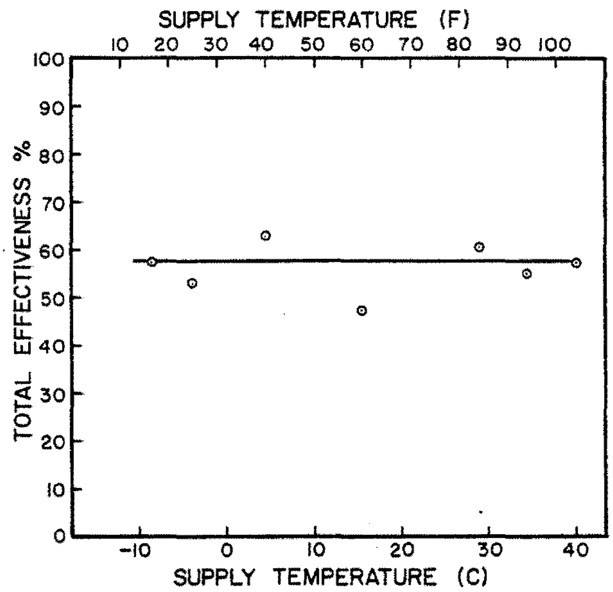

Fig. 22 Effect of supply air temperature on total effectiveness of open run-around exchanger

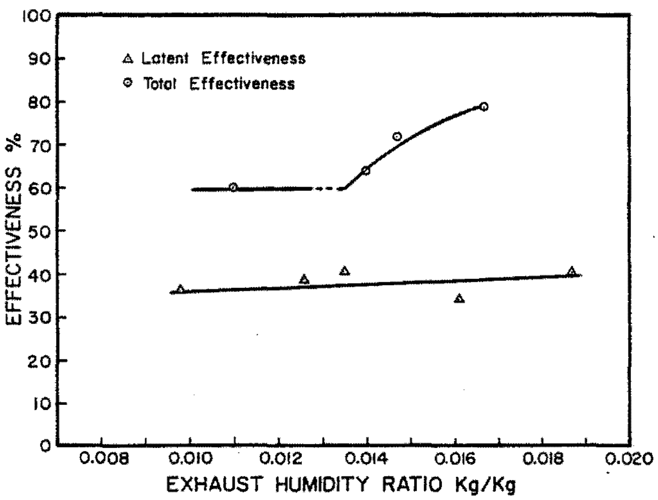

Fig. 23 Effect of exhaust air humidity ratio on thermal performance of open run-around exchanger

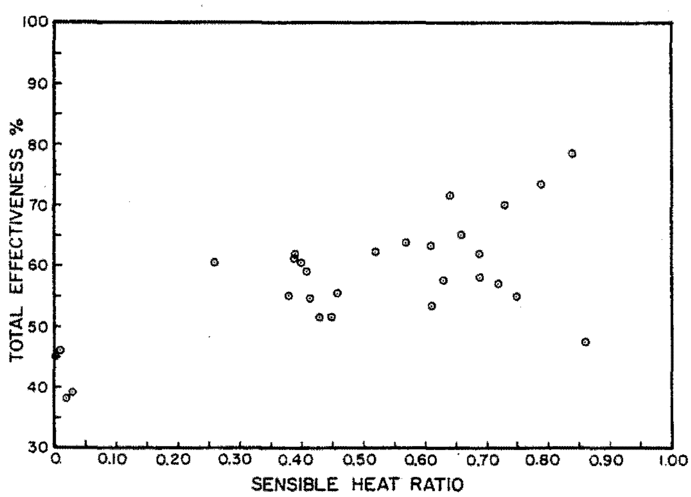

Fig. 24 Variation of effectiveness with sensible heat ratio for open run-around exchanger