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OPERATING EXPERIENCE WITH A RESIDENTIAL MECHANICAL VENTILATION SYSTEM WITH HEAT RECOVERY

Report No 81-51-K

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ABSTRACT

The efficacy of an engineered mechanical ventilation system is assessed in controlling indoor humidity levels in one of the HUDAC Mark XI houses. A mathematical humidity model of the home is developed and used to demonstrate the interaction between outdoor humidity, infiltration and ventilation rates and indoor moisture generation, and their influence on indoor humidity levels. The operating effectiveness of the rotary heat exchanger and the energy saved through its use are discussed.

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740624-368-109	824.521	March 17, 1981	81-51-K

26573
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EXECUTIVE SUMMARY

OPERATING EXPERIENCE WITH A RESIDENTIAL MECHANICAL VENTILATION SYSTEM WITH HEAT RECOVERY

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As an attempt to prove the concept of using dilution ventilation to control the build-up of humidity and, in general, indoor generated pollutants in homes, an engineered mechanical ventilation system was operated in one of the HUDAC Mark XI homes and monitored through the 1978-79 heating season.

The recorded data of outdoor humidity and the corresponding indoor humidity levels were used to develop a mathematical model of the home from which the infiltration rate and indoor moisture production rate were evaluated.

The model was used to show that under the infiltration rate of 0.15 air changes per hour alone, the indoor humidity would have peaked to 55% rh and that the imposed increase of 0.34 ach by the ventilation system was capable of reducing the indoor humidity peak to 37 per cent rh. The latter was confirmed by the field results.

The effect of ventilation flow rate on indoor humidity and on the operation of the ventilation system was demonstrated. For instance, a 25 per cent reduction in the designed flow rate of 2.3 m^3/min caused a 40 per cent decrease in outage time of the operating system when controlled by an indoor humidity level of 35 per cent rh.

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An in-house designed and constructed rotary heat exchanger was used to recover heat from the ventilation exhaust. No operating problems of the mechanical or thermal (ie, freezing) type were encountered. If the ventilation system was operated continuously throughout the 1978-79 heating season, an energy saving of 5020 kWh would result through the use of the heat exchanger.

CONTENTS

26572
new 79-01

ontario hydro research division

To Mr. F.J. Simpson
Director of Research

OPERATING EXPERIENCE WITH A RESIDENTIAL MECHANICAL VENTILATION SYSTEM WITH HEAT RECOVERY

1.0 INTRODUCTION

A mechanical ventilation system/1/ with an in-house designed and constructed regenerative rotary heat exchanger/2/ was installed in the HUDAC Mark XI No 2 house/3/ and operated during the 1978-79 heating season.

The experiment ran from December, 1978 to April, 1979, and the house was occupied by a family of four (2 adults and 2 children) until March 13, 1979. Due to problems encountered during the commissioning of the site data acquisition system, acceptable operating data covered only the period from February 27 to March 26, 1979.

The Mark XI No 2 house had a forced-air, electrically heated furnace, and the furnace fan operated continuously. At a rate of 2.3 m^3/min, outside air was added to and indoor air was extracted from the return duct of the furnace. The extracted indoor air was exhausted to the outside of the house. A schematic drawing of the ventilation system with the rotary heat exchanger is shown in Figure 1.

The objective of the experiment was to assess 1) the capability of mechanical ventilation in controlling indoor humidity, and 2) the economic viability of heat recovery from the ventilation exhaust.

2.0 MONITORED PARAMETERS

The following parameters were monitored every hour from February 26, 1979 to March 26, 1979.

a) The outdoor dry bulb and dew point temperatures.

b) The indoor dry bulb and dew point temperatures.

c) The dry bulb temperatures of the air in the supply and exhaust ducts of the rotary heat exchanger.

d) The flow rates of indoor and outdoor air through the heat exchanger.

e) The pressure drop across the heat exchanger matrix.

f) Condensation from the heat exchanger.

All the recorded data are on file. For clarity and consistency, only continuous and acceptable data were used for analyses, and in most cases, the data are included in the report.

3.0 ANALYSIS OF RECORDED DATA

The recorded data were analysed in a way to facilitate observations and discussions on the following:

1. Predicted rates of infiltration and moisture production for the home.

2. Effectiveness of the ventilation system in controlling indoor humidity.

3. Assessment of indoor humidity ratio as the control parameter.

4. Performance of the rotary heat exchanger.

3.1.0 Predicted Rates of Infiltration and Moisture Production for the Home

3.1.1 Humidity Model Formulation

The model is derived from the dilution equation which is derived under certain assumptions described in /4/.

$$M(t_n) = \gamma_1 e^{-ct_n} + (1-e^{-ct_n})(\gamma_0+\gamma_d) \qquad 1.$$

where $M(t_n)$ is the humidity ratio in the house at time t_n

γ_1 = indoor humidity ratio at time t_n-1

γ_0 = outdoor humidity ratio at time t_n-1

γ_d = the effective elevation of outdoor humidity by the indoor moisture production (MPR) at $G \frac{kg}{min}$

or $\gamma_d = \frac{Gv}{Q}$

where v = specific volume of air (m^3/kg)

Q = sum of the infiltration rate (IR) and the ventilation rate (VR), (m^3/min)

and $c = \frac{Qk}{V}$

where V = volume of the home (m^3)

and k = mixing time factor/4/.

3.1.2 Parametric Evaluation

The parameters under consideration were infiltration rate, ventilation rate, and moisture production rate. These parameters were estimated or sought for through an optimization scheme. The computer code developed for that purpose was based on minimizing the difference between calculated and measured values as an objective function.

For the occupied house with VR = 2.3 m^3/min, and MPR assumed to be 9.1 kg/day/5/, the infiltration rate and k were quantified. Starting with this value for IR, the moisture production rate in the unoccupied house was determined. The values of IR, MPR and k were then optimized to obtain an indoor humidity level as close as possible to the one measured. Figure 2 shows the predicted indoor humidity level with the measured value. It should be noted that the k factor has a negligible effect on the magnitude of γ_1. However, it is included to account for the response or mixing time effects. Other values were assigned to MPR and the effects on the predicted value of γ_1 are shown in Figures 3 and 4. Therefore, the optimization scheme evaluated the infiltration rate to be 1.0 m^3/min (ie, 0.15 air changes per hour), the moisture production rate of the occupied home to be 11.8 kg/day and of the unoccupied home to be 3.9 kg/day. The mixing factor was evaluated to be 0.07. Further refinement of these parametric values is possible, however the values were found sufficient to demonstrate the objectives of the experiment.

3.1.3 Discussion

The natural infiltration rate was evaluated to be 1.0 m^3/min or 0.15 ach. When compared with the generally accepted rate of infiltration of 0.25 ach, below which condensation results/6/, the occupied HUDAC No 2 home was a prime example for experimentation.

With VR = 2.3 m^3/min, the total air change per hour was 0.49 ach. Residential construction methods employed to this day, produce homes (with fuel-burning furnaces) with an average infiltration rate of 0.5 ach. Electrically heated homes are of the order of 0.2 to 0.33 ach/6, page 33/. The control of other indoor generated contaminants, eg, formaldehyde, radon, etc, to within acceptable levels is possible with 0.5 ach/6, page 15/. Therefore, the total of the natural and imposed air changes per hour for the HUDAC No 2 home was within the established norm.

3.2.0 Effectiveness of Ventilation System

3.2.1 Observations

The operation of the ventilation system was controlled by an indoor dehumidistat. When the setting was kept at 25% rh from February 27 to March 6, the system operated continuously. However, at a setting of 35% rh from March 6 to March 13, two shutdown periods were noted, each lasting for approximately 4 hours. A setting of 30% rh from March 13 to March 16, caused a 27-hour shutdown period. These observations were noted on Figures 5 to 13. Also, it is very apparent in these Figures that the high and low indoor humidity levels occurred during the same period as the outdoor humidity level did, but were of a smaller magnitude.

3.2.2 Ventilation Rate

The effect of the ventilation rate of 2.3 m^3/min is demonstrated by Figure 14. The model equation was plotted for the occupied interval, with VR = 0, 1.7 and 2.3 m^3/min, respectively and maintaining IR and MPR constant at 1.0 m^3/min and 11.8 kg/day respectively.

With a forced ventilation rate of 2.3 m^3/min, the level was controlled to less than 37% rh, and with 1.7 m^3/min, the maximum level was predicted to be approximately 39% rh. If the home were to rely only on the infiltration rate of 1.0 m^3/min (ie, VR = 0) then the indoor humidity would have peaked at 55% rh at a time when the outdoor temperature and humidity ratio were dropping and reached the low of -17°C and 0.0008 respectively (see Figures 11 and 12). At the same time, a lowering trend of indoor humidity was seen with forced ventilation. In addition, from Figure 14 it can be seen that the designed ventilation rate of 2.3 m^3/min controlled the indoor humidity to less than 35% rh for 43 per cent of the total operating time compared with 17 per cent if the ventilation rate was dropped to 1.7 m^3/min.

3.2.3 Influence of Outdoor Humidity

The ventilation system controls indoor humidity levels by replacing indoor air with drier outdoor air. Obviously, the lowest indoor humidity level attainable will be influenced by the outdoor humidity level. From Equation 1, the attainable level is given by ($\gamma_0 + \gamma_d$) and is quantified in Table 1.

A comparison of the minimum attainable level with the dehumidistat setting given in Table 1, explains the continuous operation of the ventilation system during February 27 to March 6, and the "off" periods during March 6 to March 16.

3.3.0 Indoor Humidity Level as the Control Parameter

A dehumidistat installed on the return duct of the electric furnace sensed the indoor humidity level and compared it with a pre-set value and accordingly controlled the operation of the ventilation system. The pre-set level was selected arbitrarily. However, in previous sections, the level was shown to depend on outdoor humidity during operation. This relationship led to continuous or intermittent operation with extended shutdown periods. As a consequence of the latter, other indoor generated contaminants /6,7/ could exceed tolerable levels of concentration. It is suggested, therefore, that the prioritized contaminant be selected as the control parameter and that the ventilation system be designed for continuous operation. As far as energy saving and indoor humidity control are concerned, the outdoor humidity level would be preferred as the control parameter.

3.4.0 Rotary Heat Exchanger

3.4.1 Effectiveness

The rotary air-to-air heat exchanger is rated by its effectiveness in recovering sensible heat. The effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible rate of heat transfer in a counter-flow arrangement. Therefore, the effectiveness

$$\eta = \frac{\dot{m}_s \; (T_4 - T_6)}{\dot{m}_{min} \; (T_4 - T_0)} \qquad 2$$

where m_s is the mass flow in the hotter duct and $\dot{m}_{min}$ is the minimum mass flow in either hot or cold duct. Figure 1 illustrates schematically the symbols of Equation 2.

The temperatures T_0, T_2 of the supply duct and T_4 and T_6 of the exhaust duct were measured near the entrance and discharge faces of the heat exchanger. Therefore, T_0 read higher than the ambient outdoor temperature, the difference being the result of 1) the heat gained by approximately 6.6 m of insulated duct, and 2) the inadvertent basement air leakage into the supply duct. The latter was calculated to be approximately 1 per cent of the outdoor air supply. The operating effectiveness was calculated and plotted against outdoor temperature as shown in Figure 15. It was found to vary between 82 to 90 per cent over an outdoor temperature range of -17°C to +12°C. The designed effectiveness was 85 per cent at a constant air flow of 2.3 m^3/min/2/. The fluctuation in operating effectiveness is attributable to the variation in supply air flow and temperature, caused by outdoor air temperature and density, wind direction and speed and the resulting basement air in-leakage.

3.4.2 Performance

The heat exchanger was in service from February 2, 1979 to May 15, 1979, amounting to approximately 1850 hours of operation. Outage intervals occurred when the mechanical ventilation system was shut off by the dehumidistat. The total number of outage hours on record amounted to 35 hours or less than 2 per cent of the total operating time. The performance is evaluated from the mechanical and thermal aspects and both indicated a remarkable performance.

3.4.2.1 Mechanical

The moving components, viz, motor, chain-drive, matrix and the static seals experienced no failures and their structural integrity was found intact at the end of the test. Some concerns were had on the plugging by house dust of the heat transfer passages in the matrix, however, a post-test inspection showed no signs of blockage or staining.

3.4.2.2 Frosting

Condensation and frosting in the heat transfer passages of the matrix was expected during operation in subzero outdoor temperatures. To determine the extent of condensation and frost formation, two indicators were employed to monitor the operation. Four differential pressure switches set at 6.25 Pa, 8.75 Pa, 12.50 Pa, and 17.50 Pa were installed across the heat exchanger and the four outputs were continuously recorded. The normal differential pressure across the heat exchanger was 4 Pa and therefore any increase in the switch output signal would have been interpreted as being caused by frost formation. The other monitoring indicator was the condensation collection device, which was examined for accumulated water on a regular basis. Both indicators showed no signs of condensation or frost formation to any detrimental extent that affected the effectiveness of the heat exchanger.

As mentioned in section 3.4.1, there was heat and mass transfer occurring from the basement air to the supply air, which caused the air temperature at the supply face of the heat exchanger to be higher than the outdoor temperature. The increase in temperature is recognized to be the main reason for the no-frost operation. The other reasons are inherent to the design and operation of the heat exchanger. Being of the rotary and regenerative type, the temperature profile is of a cyclic nature and the air flow reverses direction as the heat transfer passages complete a revolution. For these inherent reasons it is expected that the extent of frosting is less severe with the rotary design than with static plate designs of heat exchangers.

3.5.0 Waste Heat Recovery

It was shown in section 3.2.0 that the natural infiltration rate of 1.0 m^3/min was not sufficient to control indoor humidity despite extremely low outdoor humidity levels. Furthermore, it was shown that to control indoor humidity to below 40% rh, a forced ventilation rate of 1.7 m^3/min was required. Therefore, if it is assumed that the desired maximum indoor humidity level is 40% rh, then the energy lost through forced ventilation exhaust can be calculated using the "on-off" or the "continuous" operation approach.

Using a similar calculation as given in section 3.2.3 it can be shown that when the outdoor humidity ratio is less than 0.0033, the forced ventilation rate of 1.7 m^3/min will reduce the indoor humidity to below 40% rh, and consequently the system will shut down. Therefore, the accumulated "on" time or the energy lost in the 1978 to 1979 heating season can be calculated from the periods when the outdoor humidity was above 0.0033. However, the necessary outdoor humidity levels are not easily available at this time.

The "continuous" operation involves the loss of energy associated with 22°C air exhausted at 1.7 m^3/min throughout the '78-'79 heating season, ie, 6 months or 4320 hours. The energy wasted amounts to 5906 kWh.

The energy recovered by the 85% effective heat exchanger is 5020 kWh, which is equivalent to $200 at 4 cents per kWh. Using a factor of 10 for capital cost recovery, an expenditure of $2,000 for the heat exchanger alone is justifiable. The cost of the mechanical ventilation system is excluded because the HUDAC home was found to require it.

3.6.0 GENERAL REMARKS

The trend towards constructing "tight" homes will lead eventually to the mandatory use of mechanical ventilation systems. To specify a system for any tight home, the infiltration rate will have to be measured or determined through an established code. The latter is a better approach and should be based on results from tests carried out on a number of homes employing "tight" construction methods and in various regions of Canada. Admittedly, various factors, viz, wind speed and direction, size and location of leak paths in the structure, outside temperature, etc, do influence the infiltration rate. However, using boundary conditions, a viable rate can be established. Based on this assessed rate, a forced ventilation rate can be determined to attain a tolerable level of a contaminant indoors. Furthermore, the economic justification for incorporating heat recovery devices again is based on the infiltration rate. If the rate is low or incapable of controlling the contaminant to an acceptable level then additional forced ventilation is required and the associated heat loss becomes the deciding factor.

From an energy conservation point of view, high effectiveness and low operating and maintenance costs are essential criteria to be used in the selection of any type of heat exchanger. Therefore, susceptibility to freezing, the operating effectiveness, and the maintenance procedure should be assessed and compared for plate and rotary heat exchangers.

Residential construction methods and the associated air infiltration rates are being examined by HUDAC sponsored committees. Hopefully, this effort will culminate in a standard on minimum acceptable infiltration rates. Also, an in-house study has been initiated to define and quantify suitable and economic selection and operation criteria of residential type heat exchangers.

3.7.0 CONCLUSIONS

1. Through formulation of a humidity model of the HUDAC Mark XI No 2 home, the rates of infiltration and moisture production by the occupants and the structure were predicted to be 1.0 m^3/min (or 0.15 air changes per hour) and 11.8 kg/day respectively.

2. It was shown that the natural infiltration rate of 1.0 m^3/min was not sufficient to control indoor humidity. Without forced ventilation, the indoor humidity level in the occupied home would have increased to 55 per cent rh at a time when the outdoor temperature was -17°C and the humidity ratio was 0.0009.

3. The forced ventilation rate of 2.3 m^3/min, which increased the air change per hour to 0.49, controlled the humidity level in the occupied house to below 37 per cent rh. If the ventilation system was operated continuously, then the 2.3 m^3/min rate would cause the indoor level to be less than 35 per cent rh for 43 per cent of the total occupied time, compared with 17 per cent for a 1.7 m^3/min rate.

4. No mechanical failures were experienced with the operation of the rotary heat exchanger. The operating effectiveness was calculated to be in the range of 82 to 90 per cent. The operation was devoid of frost formation to an extent that influenced the operating effectiveness. Inherent design features, together with the heat and mass transfer that occurred from the basement air to the supply air were suspected to be responsible for the "no-frost" operation of the heat exchanger.

5. Based on a forced and continuous ventilation rate of 1.7 m^3/min over the entire '78-'79 heating season (ie, 6 months) the energy saved by the heat exchanger was calculated to be 5020 kWh.

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The computer analyses of data were handled by Mr. N. Mitchell, a summer student from the University of Toronto.

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5. Ventilation Requirements. Building Research Establishment Digest. No 206.

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7. Stricker, S. Ventilation state-of-the-art review. Report for the Canadian Electrical Association. June, 1980.

TABLE 1

MINIMUM ATTAINABLE INDOOR HUMIDITY LEVELS

$$M(t_n) = \gamma_1 e^{-ct} + (1-3^{-ct})(\gamma_0 + \gamma_d) \qquad 1$$

$$\text{when } t \to \infty,\ M(t_n) = \gamma_0 + \gamma_d$$

$$= \gamma_0 + \frac{Gv}{Q}$$

The predicted values are

$$G = 11.8 \text{ kg/day} = \frac{11.8}{24x60} = 0.0082 \text{ kg/min}$$

$$v = 0.83 \text{ m}^3\text{/kg}$$

$$Q = IR + VR = 1.0 + 2.3 = 3.3 \text{ m}^3\text{/min}$$

$$\therefore \quad \gamma_d = 0.0021$$

Period	Dehumidistat Set-Point (rh)	Lowest Outdoor Humidity Ratio (γ_0)	Minimum Attainable Indoor Humidity Ratio/rh
Feb 27-Mar 6/79	25%	0.0025	0.0046/35%
Mar 6 -Mar 13/79	35%	0.0010	0.0031/24%
Mar 13-Mar 16/79	30%	0.0009	0.0030/23%

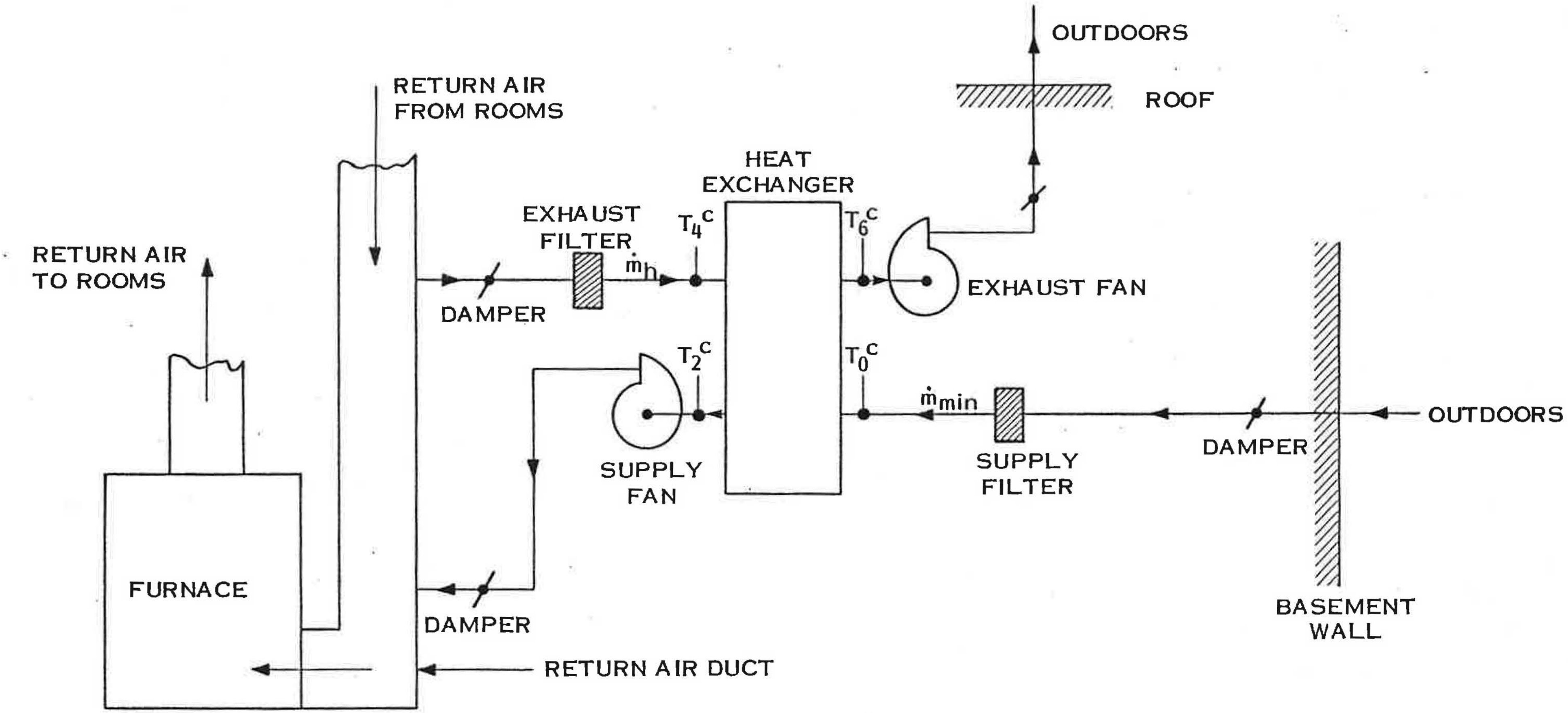

FIGURE I

A SKETCH OF THE MECHANICAL VENTILATION SYSTEM

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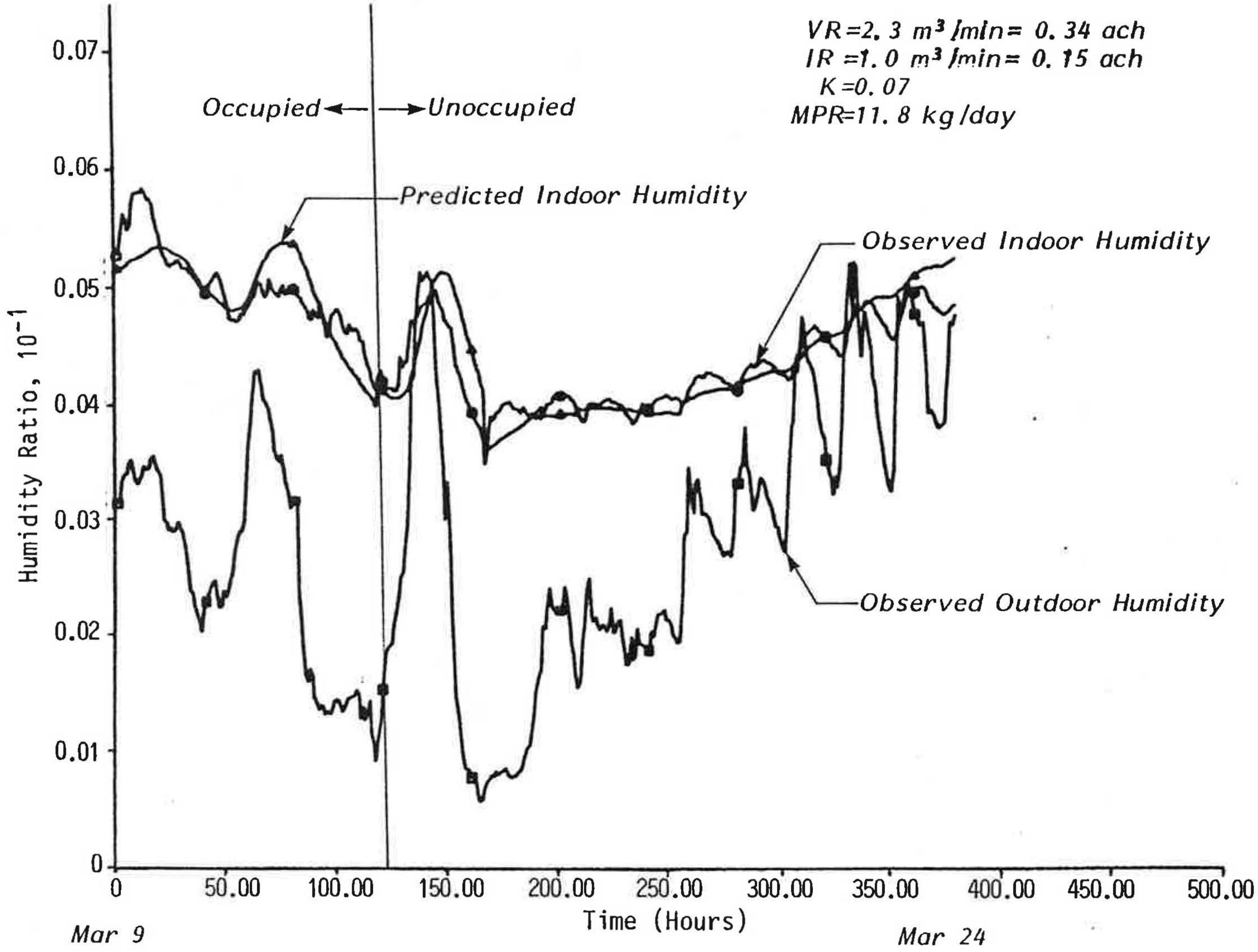

FIGURE 2

PREDICTED INDOOR HUMIDITY LEVEL

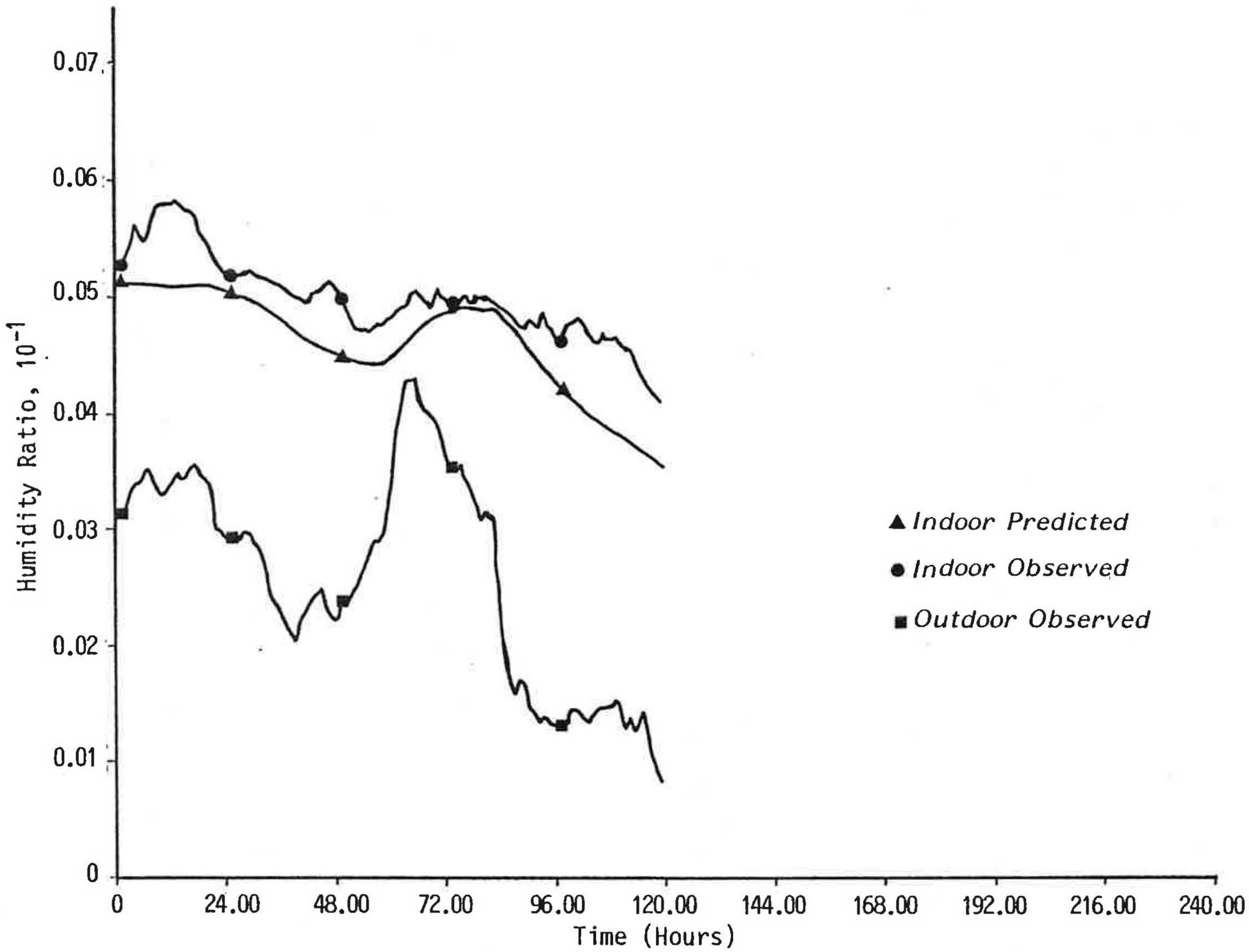

FIGURE 3

PREDICTED INDOOR HUMIDITY WITH MPR = 9.1 kg/DAY

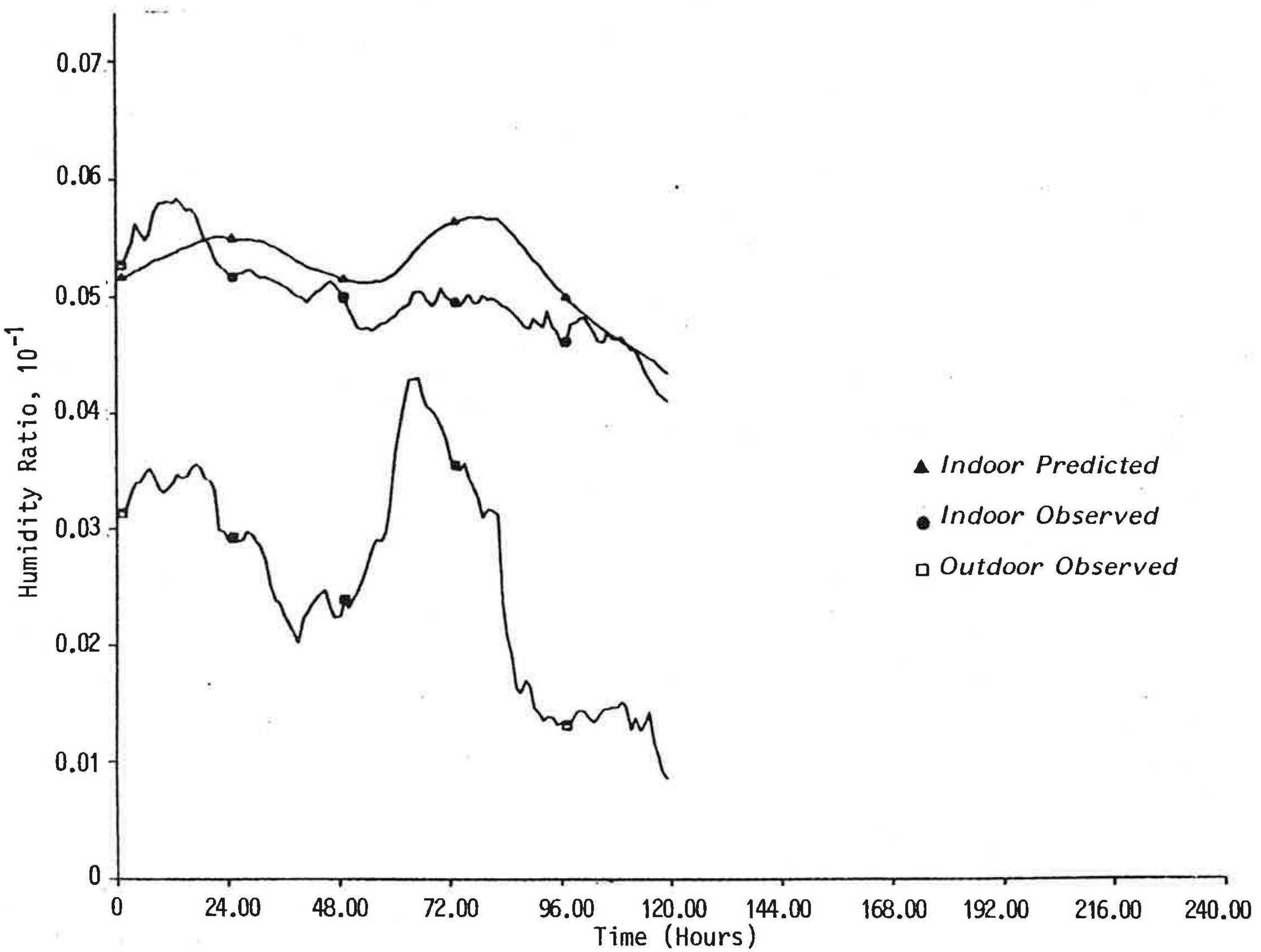

FIGURE 4

PREDICTED INDOOR HUMIDITY WITH MPR = 13.6 kg/DAY

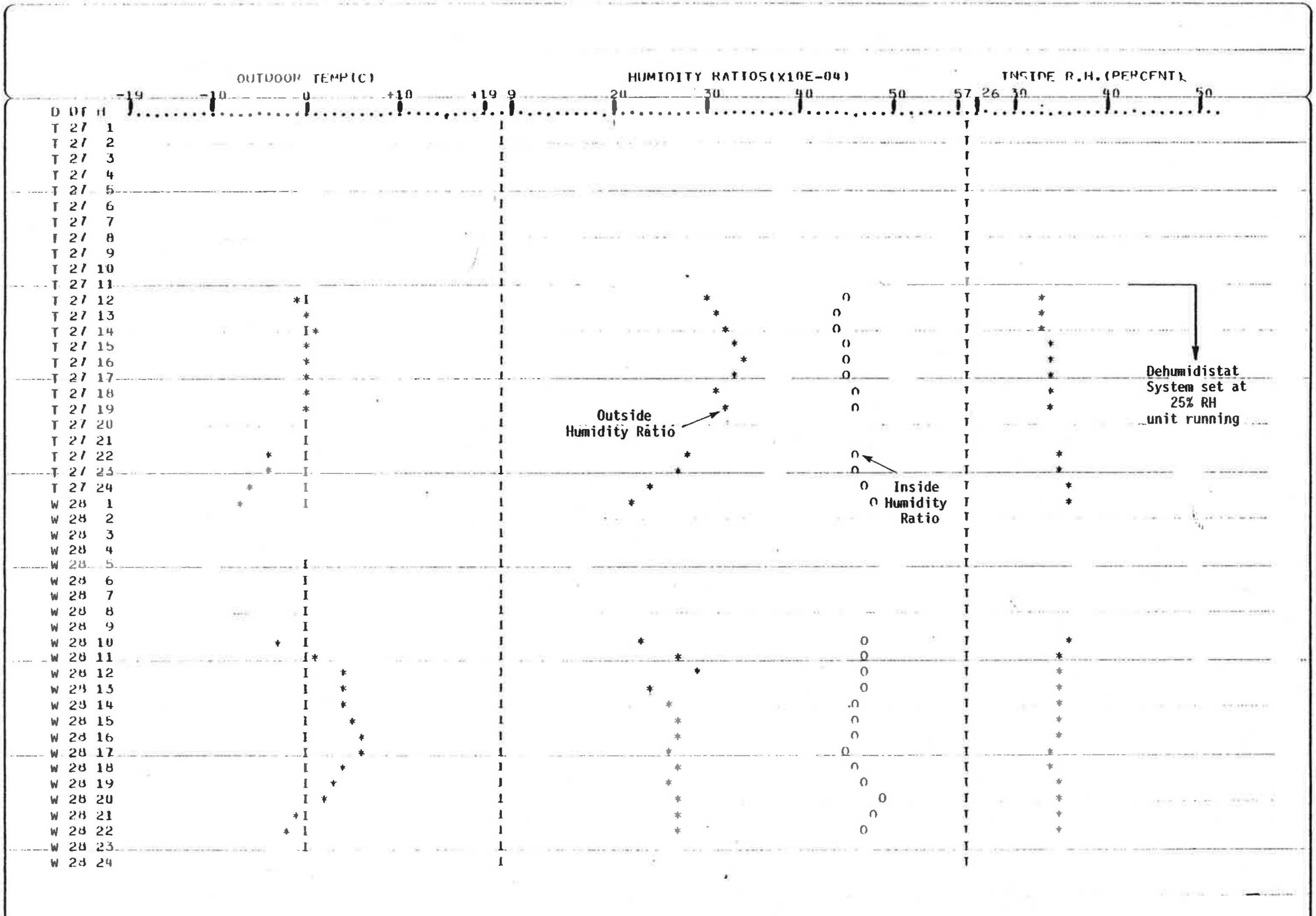

FIGURE 5

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29076 RD

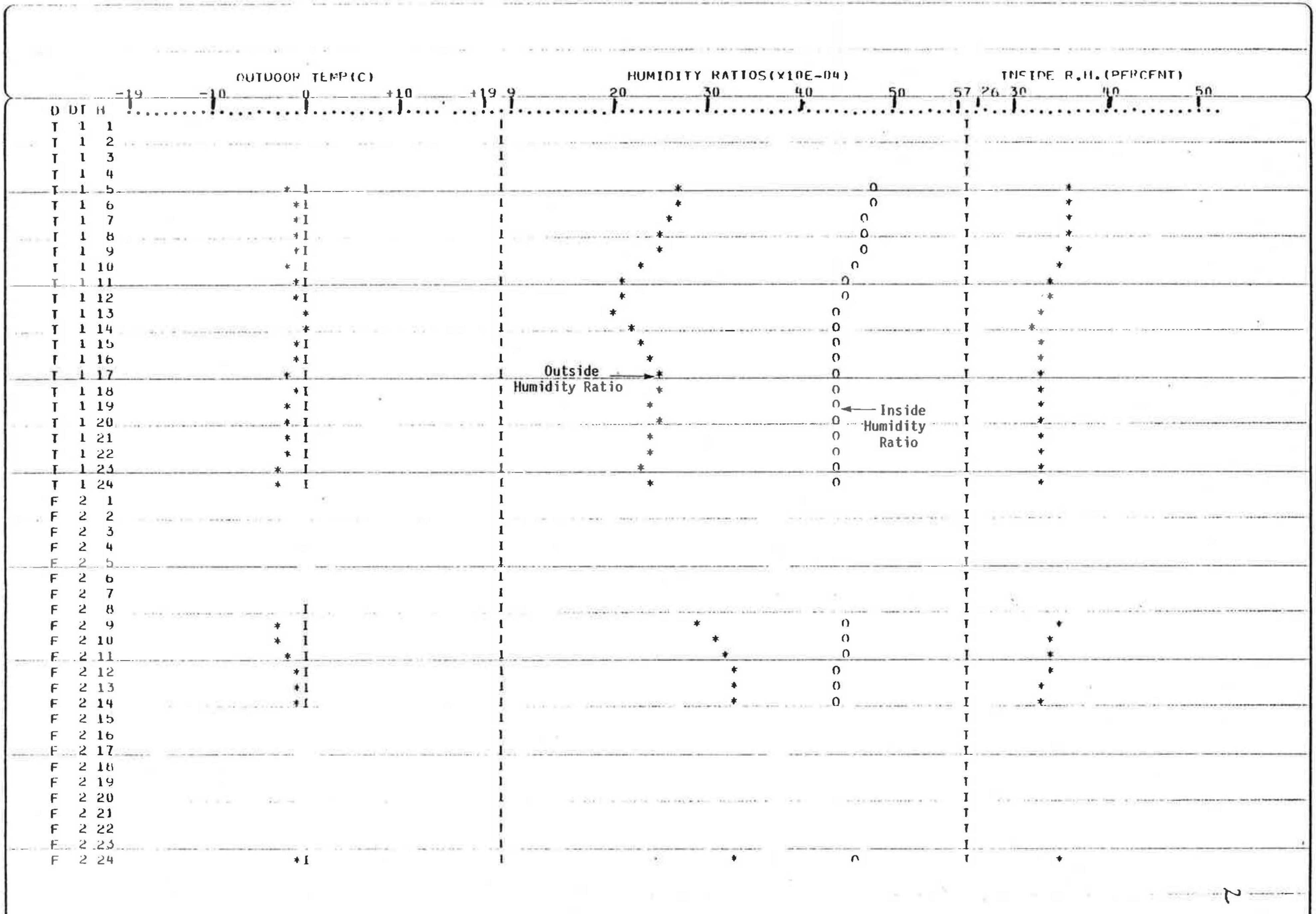

FIGURE 6

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29077 RD

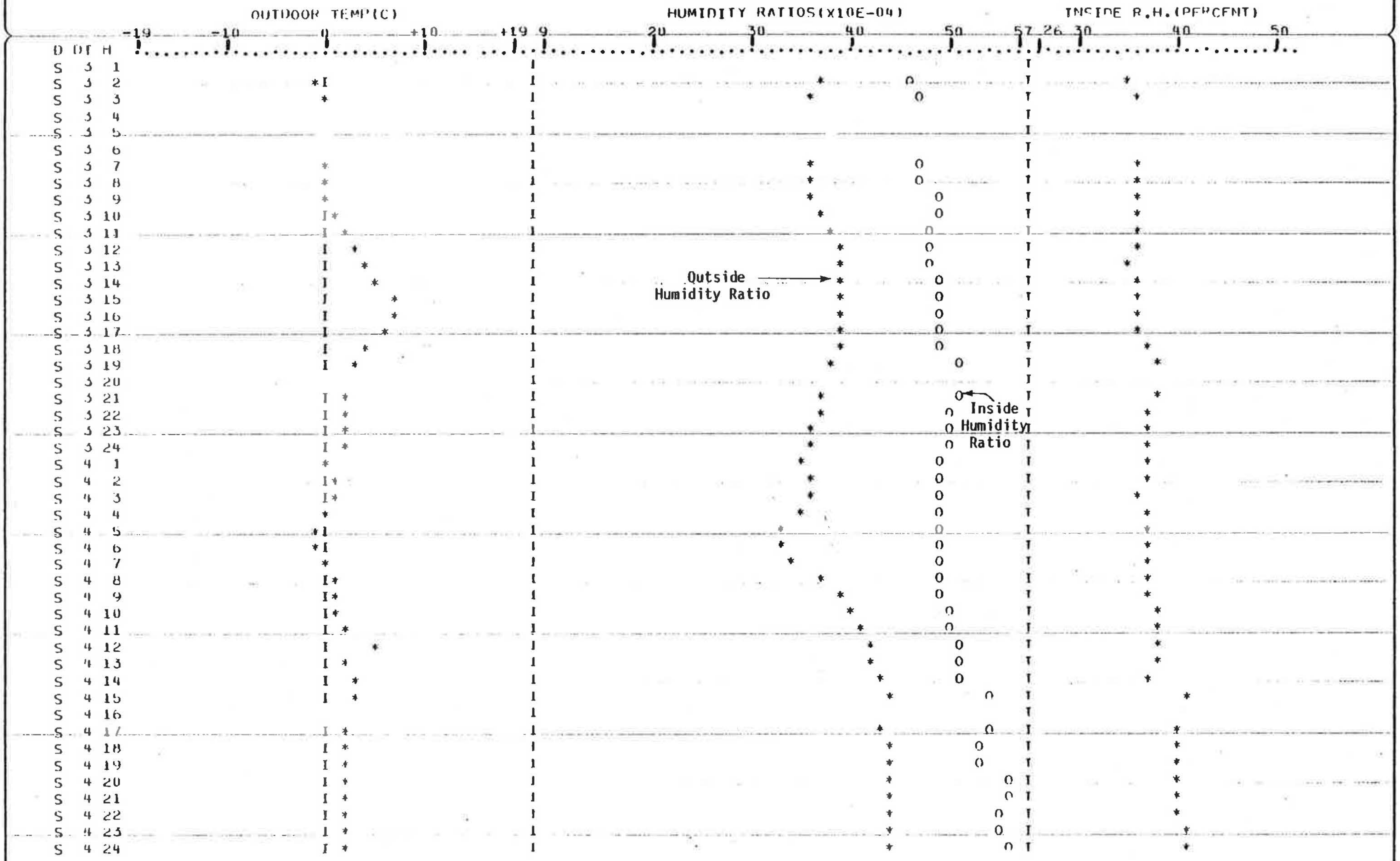

FIGURE 7

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29078 RD

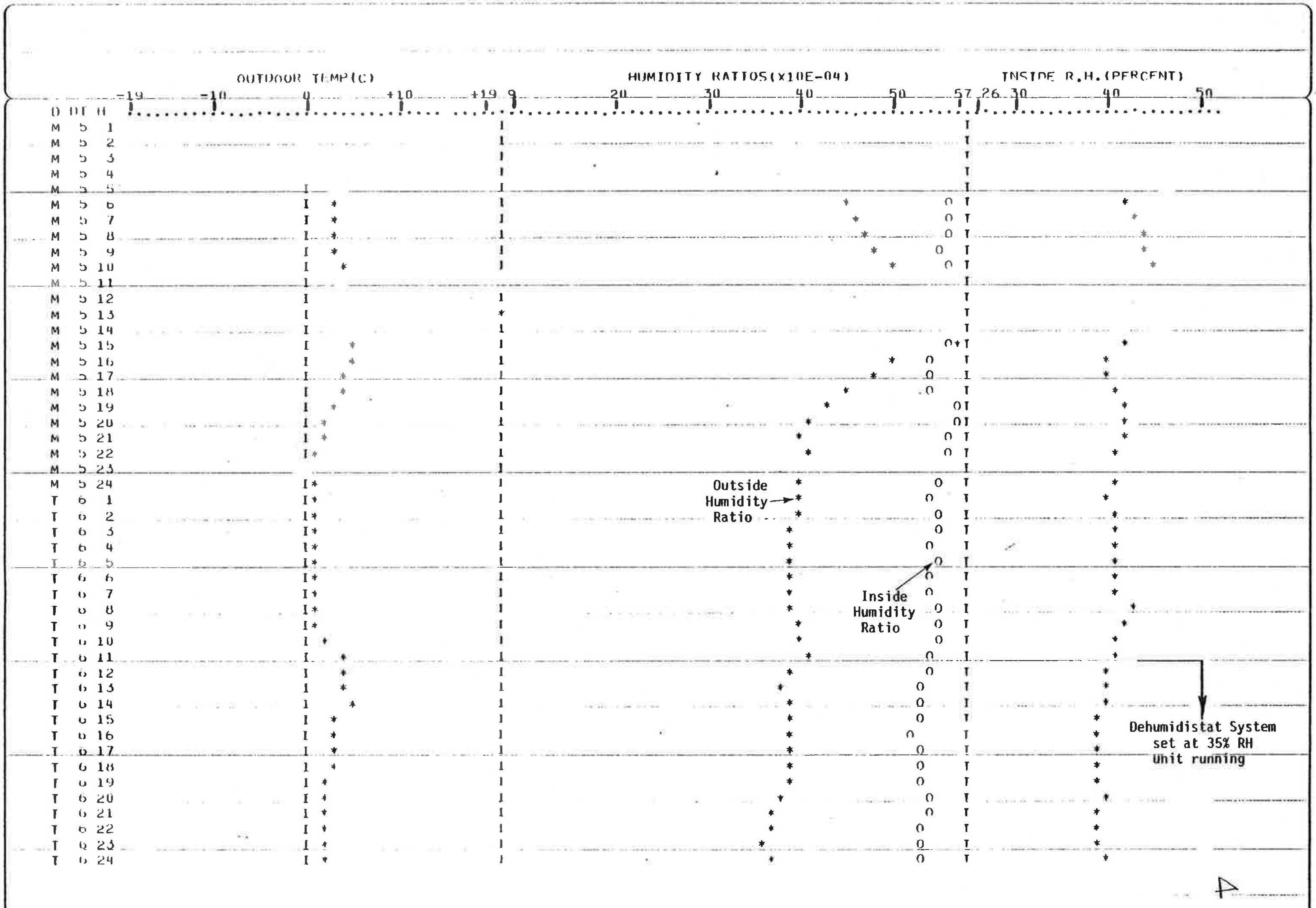

FIGURE 8

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29079 RD

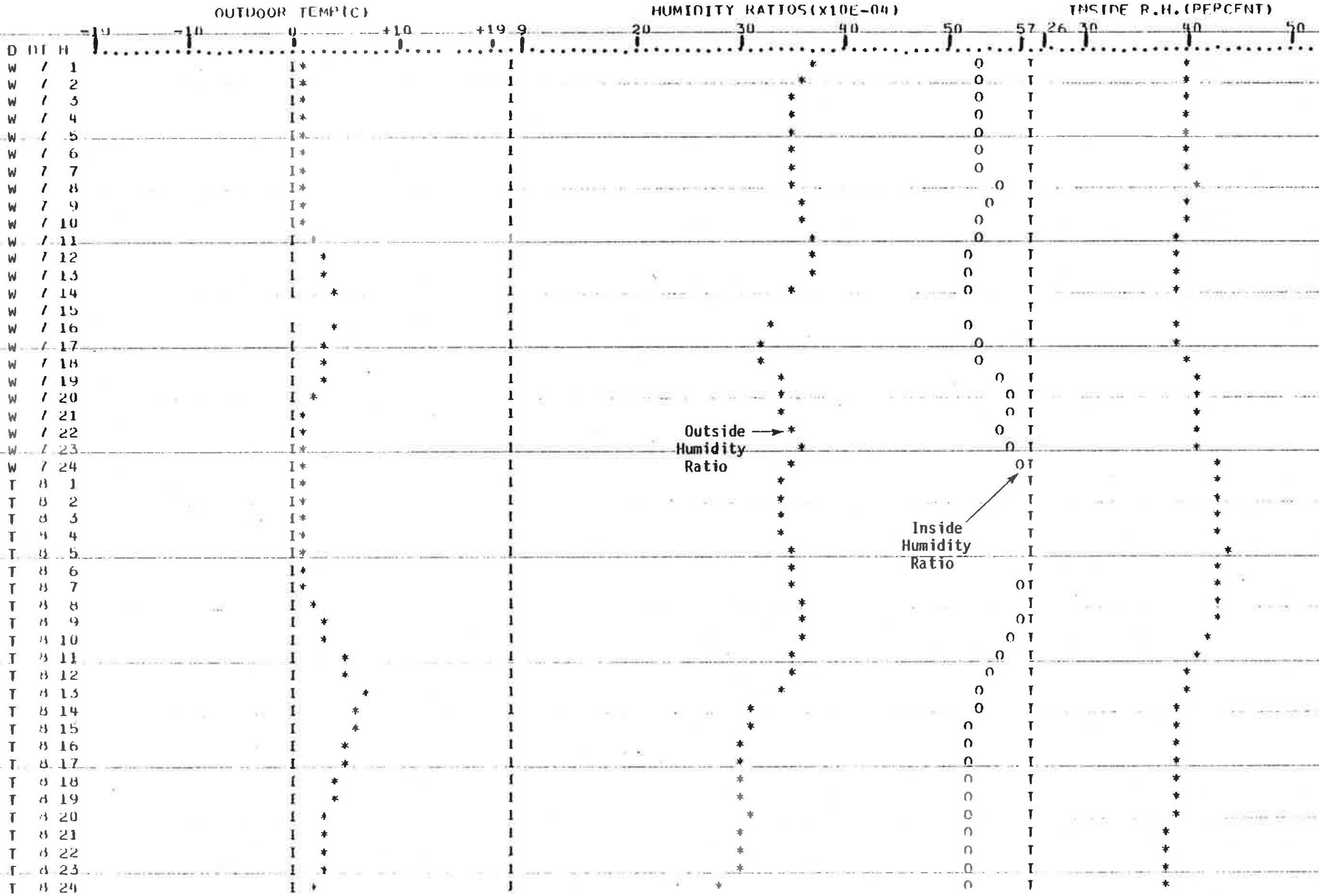

FIGURE 9

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29080 RD

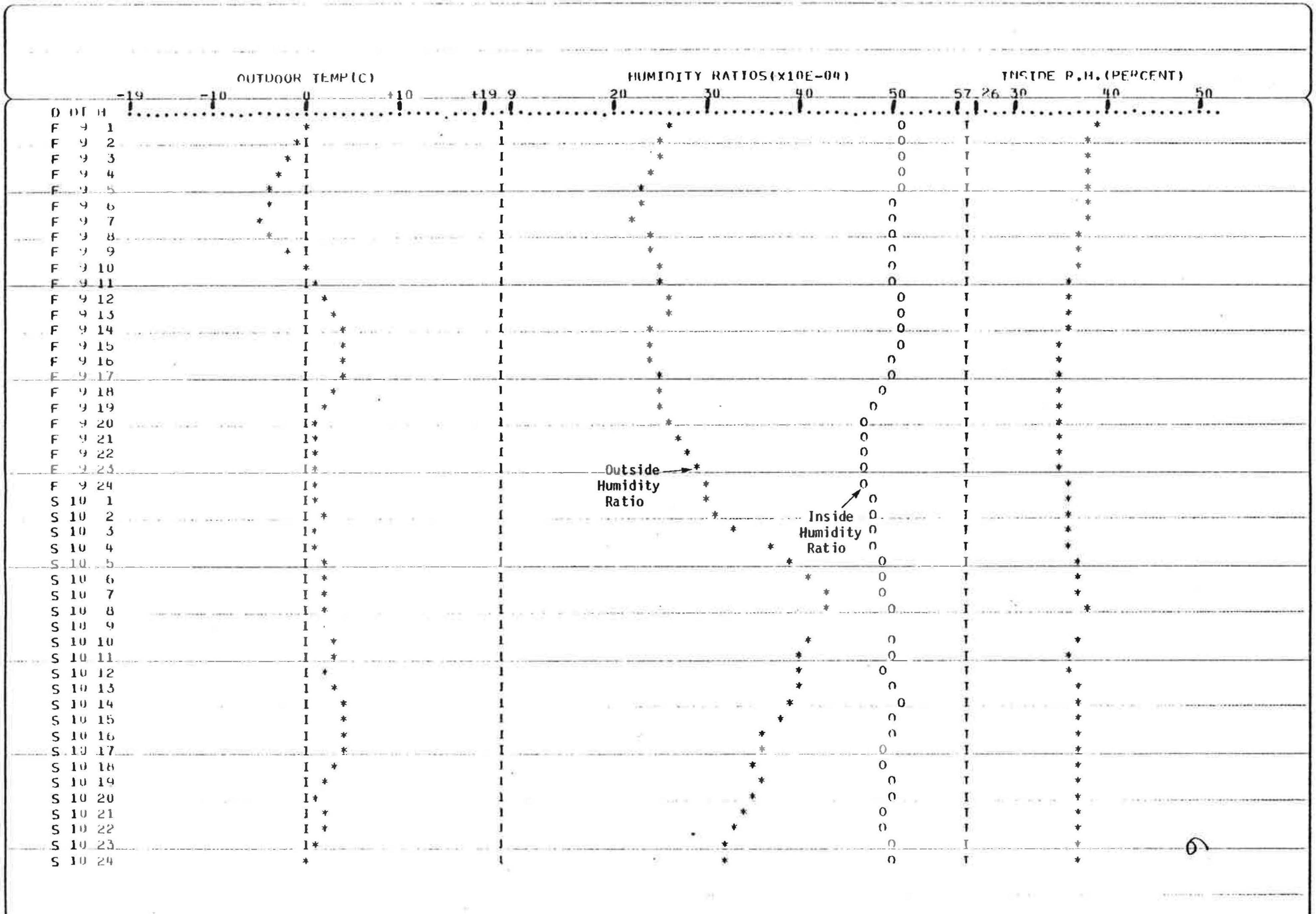

FIGURE 10

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

29081 RD

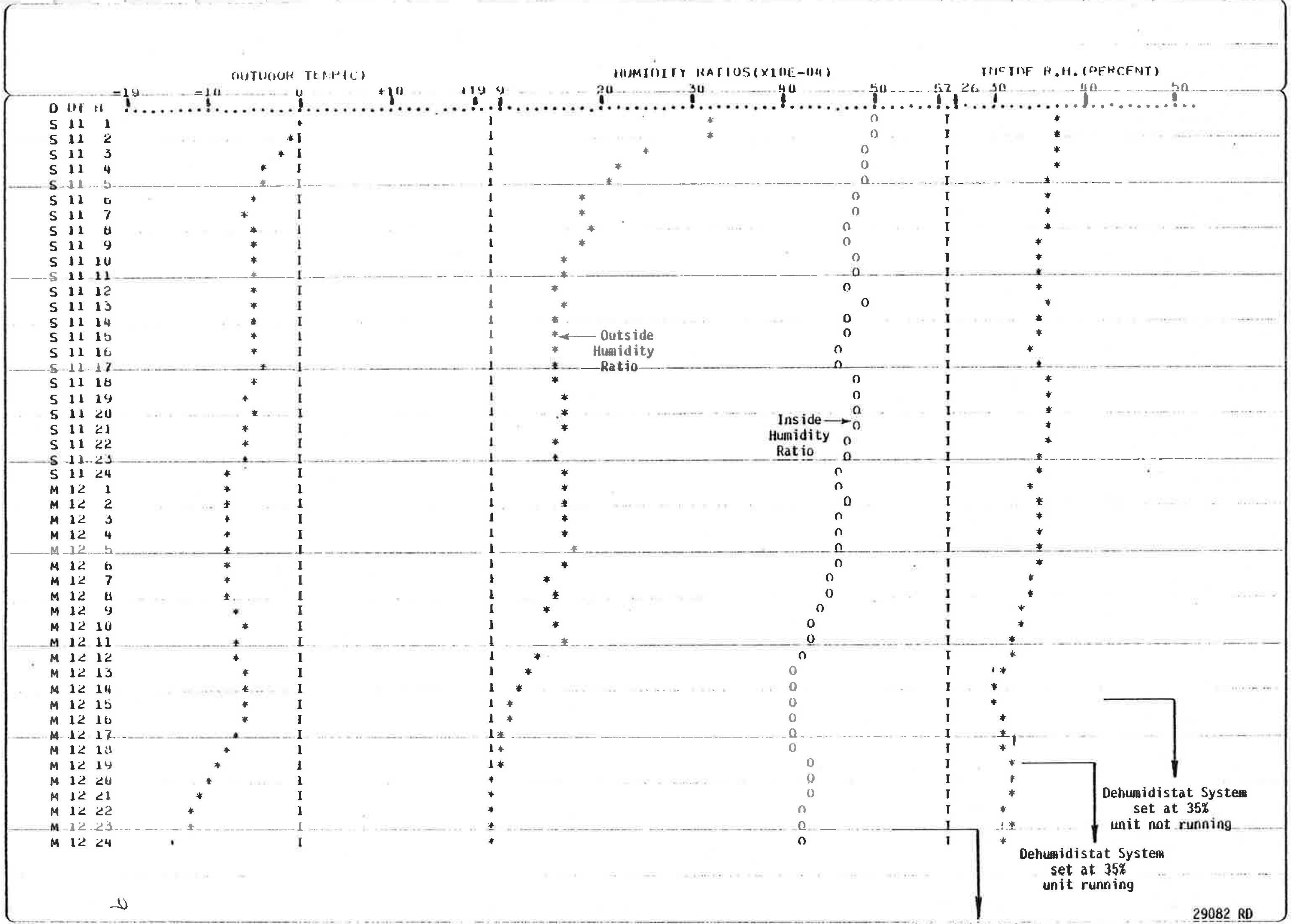

FIGURE 11

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

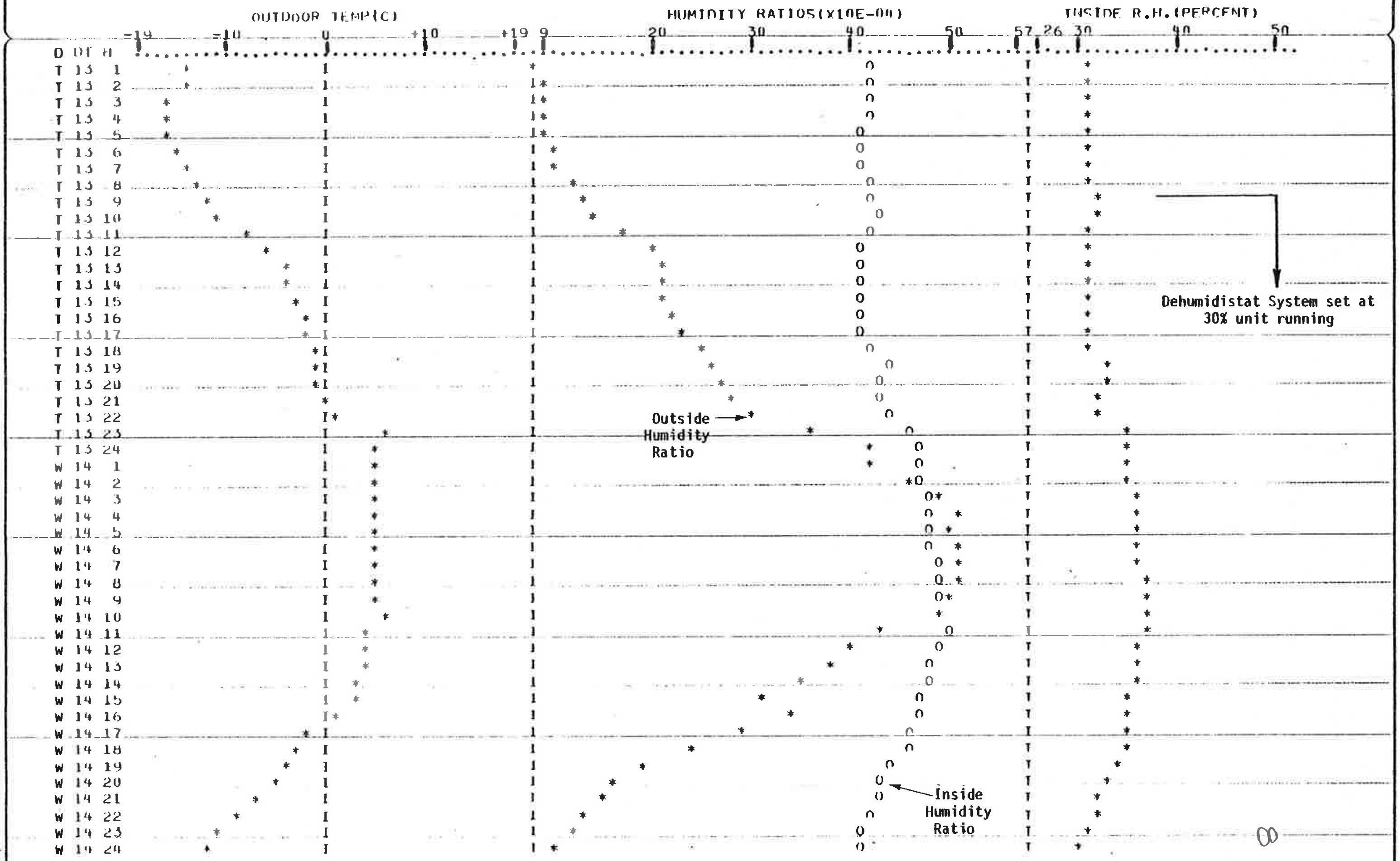

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FIGURE 12

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

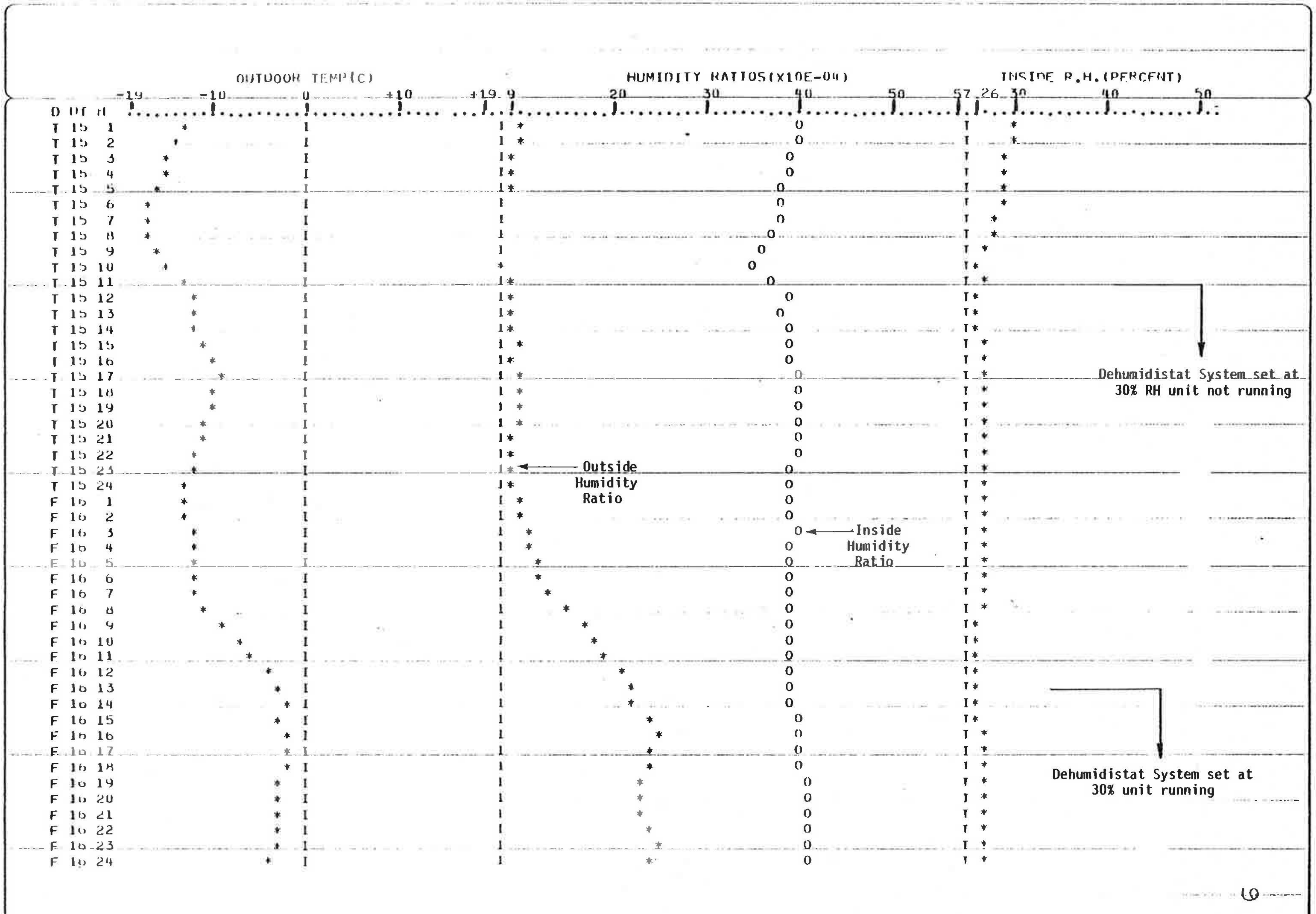

FIGURE 13

29084 RD

HOURLY INDOOR AND OUTDOOR TEMPERATURE AND HUMIDITY

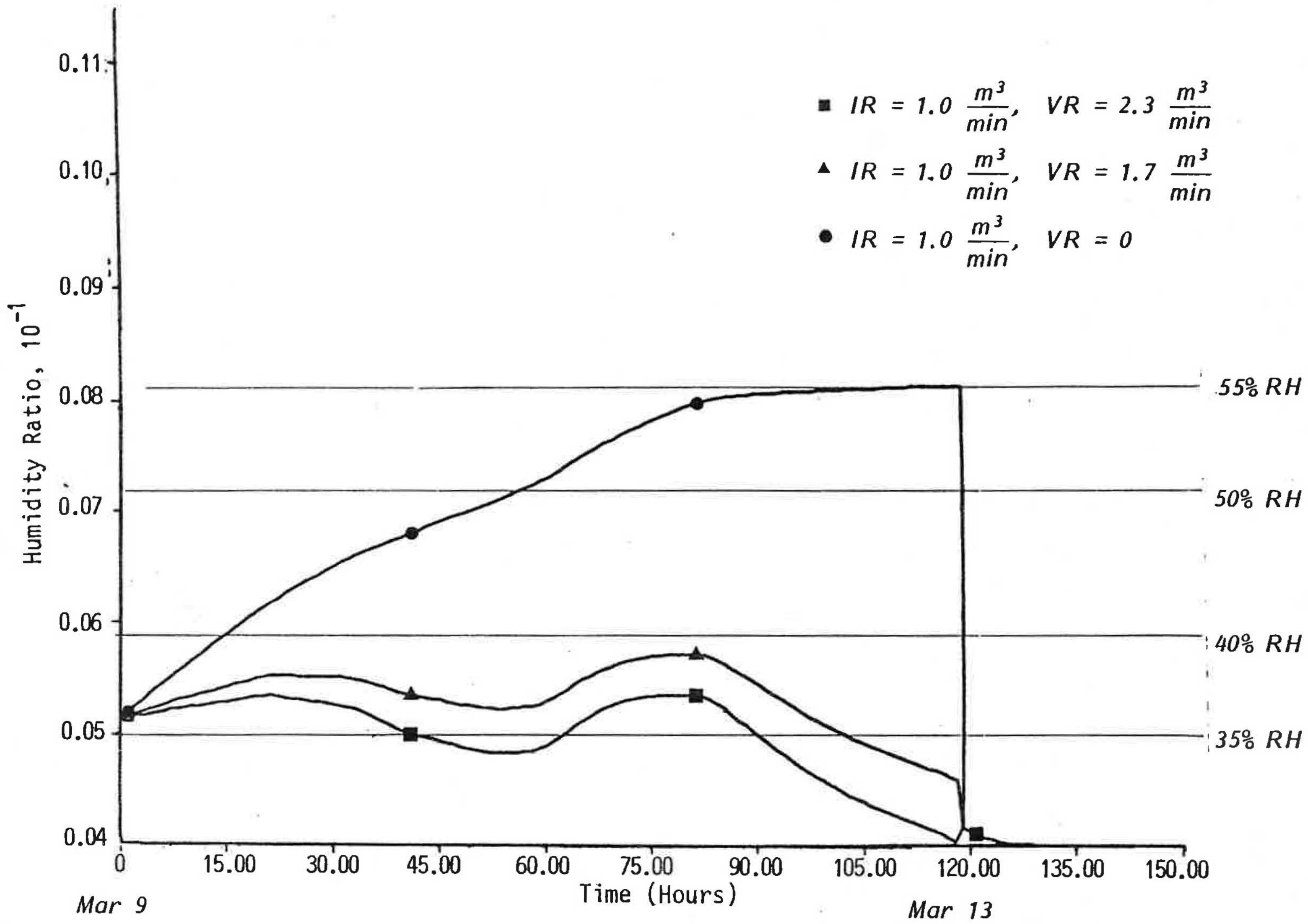

FIGURE 14

EFFECT OF VENTILATION RATE ON INDOOR HUMIDITY

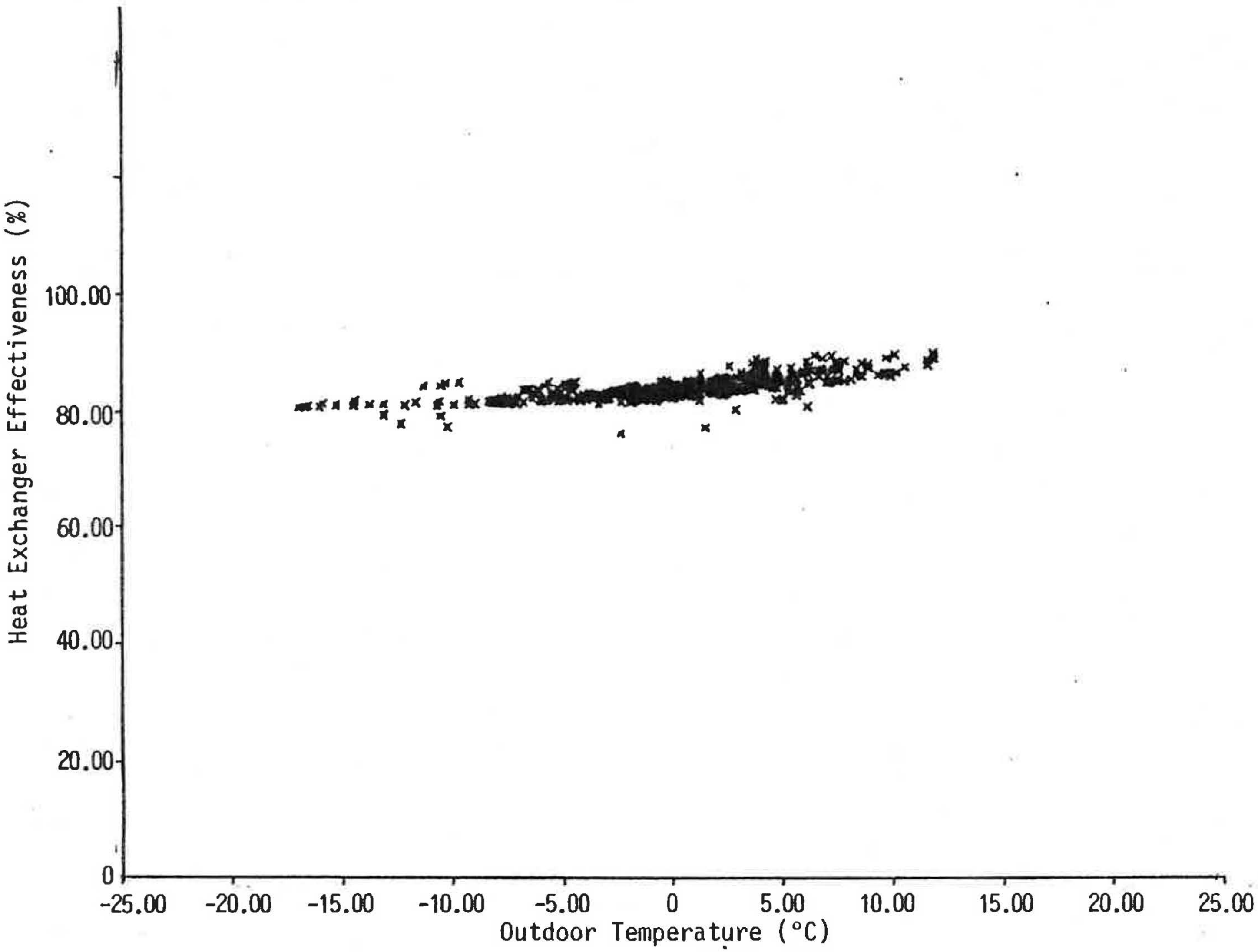

FIGURE 15

EFFECTIVENESS OF THE ROTARY HEAT EXCHANGER