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INDIRECT EVAPORATIVE COOLER PERFORMANCE

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ABSTRACT

This paper investigates three sensible air coolers: a single- and a multiple- stage indirect evaporative cooler, and a combination cooling tower/heat exchanger air cooler. Relations and performance characteristics are developed. The single-stage indirect evaporative cooler is capable of cooling inlet air to temperatures approaching the inlet wet-bulb temperature. The multiple-stage indirect evaporative cooler, the heat exchanger/evaporative cooler, and the cooling tower/heat exchanger air coolers are all capable of cooling inlet air to temperatures approaching the inlet dew-point temperature. In the region of air conditioning interest, the coefficients of performance range from 25 for the single-stage indirect evaporative cooler up to 75 for the cooling tower/heat exchanger air cooler.

INTRODUCTION

A direct evaporative cooler is an air-conditioning device commonly used to cool and humidify air [ASHRAE 1983]. Direct evaporative cooling involves the process of evaporating water into an airstream. Throughout this process, air and water are in direct contact and both heat and mass are exchanged. Direct evaporative cooling of an airstream presents an energy trade-off, since a reduction in sensible energy (temperature) is obtained with a corresponding increase in latent energy (humidity). A direct evaporative cooler never reduces the total load on an airstream; it merely exchanges one form of the load for another.

An indirect evaporative cooler is an air-conditioning device that sensibly cools process air without increasing the absolute humidity ratio of the air [ASHRAE 1983; Product literature 1980]. This process uses two separate airstreams that never mix or come in direct contact. One stream, the "wet stream," goes through an evaporative cooling process and is cooled and humidified. The second stream, the "process stream," passes through a sensible air/air heat exchanger and uses the cool, wet stream as its heat sink. The indirect evaporative cooling process requires no energy input besides that required to overcome fan and water pumping power. Coefficients of performance (air cooling benefit/energy input) for these systems, therefore, tend to be very high.

A single indirect evaporative cooler can only cool process air down to the wet bulb temperature of the wet stream. Since the wet stream is typically the ambient, the system performance decreases as the ambient wet-bulb temperature increases. Staging of evaporative coolers increases the performance. In addition, the system does not have the capability to dehumidify the inlet process stream. As in the case of a direct evaporative cooler, an indirect evaporative cooler is best suited for dry and temperate climates.

Another air-cooling device can be developed by coupling a cooling tower with a sensible air/water heat exchanger. This device was first proposed and analyzed by Maclaine-cross [1985] and Kang and Maclaine-cross [1985] as a possible air chiller for use in desiccant air-conditioning cycles. In this device, a cooling tower is used to cool a water stream to the wet-bulb temperature of the air. The cold water is then used to sensibly cool the airstream. A portion of the airstream is used in the cooling tower, which allows a low water temperature to be obtained.

In this paper, the performance relations for three evaporative cooling devices are formulated. Thermal coefficients of performance are developed and compared for these units. The range of environmental parameters over which these devices are feasible is determined. A further description of this study is given in the thesis by Crum [1986].

PERFORMANCE CHARACTERISTICS

Single-Stage Indirect Evaporative Coolers (IEC)

In the indirect evaporative cooler, the wet stream air that acts as the heat sink for the warmer process air is usually drawn from the ambient. Figure 1 illustrates an IEC with a common air source (state 1) and a fraction of the inlet air taken off before the heat exchanger for use as the wet stream. The performance of an IEC is given in terms of the effectiveness of the sensible heat exchanger and the direct evaporative cooler (DEC). These are defined as:

$$\varepsilon_{Hx} = \frac{q}{q_{max}} = \frac{C_{proc}(T_1 - T_2)}{C_{min}(T_1 \; T_3)} \tag{1}$$

$$\varepsilon_{DEC} = \frac{T_1 - T_3}{T_1 - T_{wb}} \tag{2}$$

The effectiveness of a single-stage IEC is the ratio of the actual heat transfer rate from the process flow stream to the maximum possible heat transfer rate from the process stream. The maximum heat transfer rate would be realized if the process stream were cooled all the way to the wet bulb temperature of the wet stream.

$$\varepsilon_{IEC} = \left.\frac{q}{q_{max}}\right|_{\substack{process\\stream}} = \frac{T_1 - T_2}{T_1 - T_{wb}} \tag{3}$$

In terms of Equations 1 and 2, the effectiveness of the IEC may also be written as:

$$\varepsilon_{IEC} = \varepsilon_{HX}\varepsilon_{DEC}\frac{C_{min}}{C_{proc}} \tag{4}$$

The cooling benefit per unit mass derived from the single-stage IEC may be calculated as:

$$\frac{\underline{Benefit}}{\dot{m}_{del}} = \varepsilon_{HX}\;\varepsilon_{DEC}\;\frac{\dot{m}_{min}}{\dot{m}_{del}}\;c_p\;\Delta T_{max} \tag{5}$$

where ΔTmax is the difference between the dry- and wet-bulb temperature of the inlet air and $\dot{m}_{min}$ is the minimum mass flow rate through the heat exchanger.

Assuming that there is equal pressure drop across both sides of the IEC, an energy operating cost may be calculated for the single-stage IEC from standard fan laws as:

$$\dot{w} = (\dot{m}_{delivered} + \dot{m}_{dump})\;\frac{\Delta p}{\rho} \tag{6}$$

where Δp is the pressure drop across the heat exchanger and ρ is the density of the airstreams.

A new variable, F, is introduced to better define what portion of the inlet air to the IEC is being processed and delivered to the building zone. In terms of inlet air, delivered air, and dumped air, F is defined as:

$$F \triangleq \frac{\dot{m}_{delivered}}{\dot{m}_{inlet}} = \frac{\dot{m}_{delivered}}{\dot{m}_{delivered} + \dot{m}_{dump}} \tag{7}$$

Using the definition of F and Equation 6, the cost per delivered mass flow rate may be written as:

$$\frac{Cost}{\dot{m}_{del}} = \left(\frac{\Delta p/\rho}{F}\right) \tag{8}$$

The coefficient of performance (COP) is defined as the benefit cooling derived from a system divided by the cost of obtaining that benefit. In the case of a single-stage IEC, COP can be obtained by dividing Equation 5 by Equation 8 to yield:

$$COP = F\varepsilon_{HX} \frac{\dot{m}_{min}}{\dot{m}_{del}} k \tag{9}$$

where the term k contains parameters not related to the mass flow rate.

$$k = c_p \varepsilon_{DEC} \Delta T_{max} \frac{\rho}{\Delta p} \tag{10}$$

The constant, k, is a dimensionless quantity that is a function of the airstream's specific heat, the effectiveness of the adiabatic saturation process, the maximum possible temperature difference across the heat exchanger, and the pressure drop across the heat exchanger. The maximum temperature difference term, ΔT_{max}, characterizes the state of the inlet air on the psychrometric chart. Obviously, the hotter and drier the inlet state, the larger both ΔT_{max} and the COP will be. The COP is also directly proportional to the effectiveness of the evaporative cooler and is inversely proportional to the pressure drop.

A graph of COP/k as a function of the fraction of inlet air delivered to the building is shown in Figure 2 for varying heat exchanger NTU. The COP is zero at F equalling both 0 and 1 due to infinite costs at F = 0 and zero benefits at F = 1. Further, an optimum COP is seen to exist at a fraction delivered of 0.5 for all values of NTU.

Using the optimum fraction delivered value of 0.5, a heat exchanger NTU of 3.0, and an evaporative cooler effectiveness of 0.8, lines of constant COP were mapped out on a psychrometric plane. These are shown in Figure 3 for a pressure drop of 0.5 in. water (125 Pa) across each side of the IEC. Lines of constant COP have the same shape as relative humidity lines. In the range of temperature and humidity ratios seen in air-conditioning operations, the COPs range from 15 to 50.

To further investigate the utility of an IEC, two performance parameters are defined as follows:

$$\varepsilon_T = \frac{T_{inlet} - T_{outlet}}{T_{inlet} - T_{dew\ pt.}} \tag{11}$$

$$\varepsilon_{cc} = \frac{\dot{m}_{delivered}(T_{inlet} - T_{outlet})}{\dot{m}_{inlet}(T_{inle} - T_{dew\ pt.})} = F\varepsilon_T \tag{12}$$

The temperature effectiveness provides a measure of how close the inlet air can be cooled toward the inlet air dew-point temperature and ranges from 0 to 1. The cooling capacity effectiveness provides a measure of how much of the inlet air can be cooled and to what extent it can be cooled. The cooling capacity effectiveness is, therefore, an energy effectiveness. This effectiveness also ranges from 0 to 1, where the maximum value of 1 corresponds to cooling all of the air all of the way to its dew-point temperature. The airflow through an

IEC can only approach the wet-bulb temperature, so the above effectivenesses will always be less then 1.

The cooling capacity effectiveness for variable fraction delivered values is shown in Figures 4 for an inlet state of 86°F (30°C) and 0.009 lbm/lbm absolute humidity ratio. The evaporative cooler effectiveness is 0.8 and the heat exchanger sizes range from 1 to infinite NTU. The cooling capacity effectiveness is symmetric about the optimum operating F of 0.5. At zero fraction delivered, there is no cooling effect. At a unity fraction delivered, there is no temperature drop since there is no cold stream heat sink. An optimum cooling capacity effectiveness exists, since the relative mass flow ratio, $\dot{m}_{deliver}/\dot{m}_{inlet}$, is zero at $F = 0$ and the temperature effectiveness is zero at $F = 1$.

The performance of a single-stage IEC varies as the inlet state changes. Figure 5 shows constant air outlet temperatures for an IEC sized with a heat exchanger NTU of 3, an evaporative cooler effectiveness of 0.8, and a constant fraction delivered value of 0.5. Any combinaton of temperature and humidity ratio that intersects one of the solid lines results in an exiting air temperature as indicated. For example, air entering the single-stage cooler at 77°F (25°C) and 0.009 lbm/lbm (kg/kg) would exit at a temperature of 68° (20°C). These lines of constant outlet temperature are straight on a psychrometric chart and resemble constant wet-bulb temperature lines. All of these constant outlet temperature lines intercept the saturation curve at the value of the constant temperature line. The temperature effectiveness of a single-stage IEC increases as one moves to dryer and warmer states on the psychrometric chart.

Multi-staged Indirect Evaporative Coolers

Staging a succession of indirect evaporative coolers in a series configuration has the potential for cooling the inlet air to near its dew-point temperature. The overall performance of the multiple-stage IEC is dependent on the effectiveness of each individual evaporative cooling and heat exchange process. In the limit, for an infinite number of stages, the air introduced at the inlet to the first stage can be cooled to its dew-point temperature with no addition of moisture to the delivered process stream.

In order to operate any one of the stages at the optimum cooling capacity and COP, half of the air delivered from the last stage must be used as the wet stream cold sink. The overall fraction delivered for an n-staged IEC is therefore:

$$F_{n\text{-staged IEC}} = \left(\frac{1}{2}\right)^n \qquad (13)$$

For example, if two stages of cooling were used, only 25% of the inlet air to the first stage would be delivered as cooled process air at the exit of the second stage. Sacrificing a large portion of the inlet process stream may be expensive.

The temperature and cooling capacity effectivenesses of multi-stage units can be evaluated following the relations for a single-stage unit. The outlet temperature for a two stage unit is plotted on Figure 5 for comparison with a single stage unit. There is a significant reduction in outlet temperature over single stage units.

Figure 6 shows the capacity as a function of number of stages for a constant inlet state of 86°F (30°C) and 0.009 lbm/lbm (kg/kg) absolute humidity ratio and an evaporative cooler effectiveness of 0.8 and heat exchanger NTU values ranging from 1 to infinity. The cooling capacity effectiveness continually decreases with the number of stages despite the increasing cooling effect brought about by additional cooling stages. The drastic decrease in the delivered airflow rate dominates the cooling capacity effectiveness.

Cooling Tower/Heat Exchanger Air Cooler (CT/HX)

A system schematic and process representation of a cooling tower/heat exchanger air cooler are shown in Figure 7. In this process, the inlet air at state 1 is sensibly cooled to state 2, while at the same time, the chilled tower water is warmed from state W_1 to W_2. The air at state 2 is split into two streams. One stream is delivered as conditioned air to the building zone and the other is routed through the cooling tower. The portion of the inlet air that is channeled through the cooling tower is humidified and warmed to state 3 and then dumped to the ambient. The cooling tower receives water at state W_2 and returns it chilled to the heat exchanger at state W_1.

The model used in analyzing the coupled heat and mass transfer in the cooling tower is based on an analogy to sensible heat transfer in dry heat exchangers. This method of analyzing wet surface heat exchangers was first suggested by Maclaine-cross and Banks [1983]. A brief outline of the development of the resulting model equations follows. An energy balance on the cooling tower fluid stream in Figure 7 yields:

$$\dot{m}_f c_{p,f} \frac{dT_f}{dx} = UW\ (T_w - T_f) \tag{14}$$

An energy balance on the air/water interface yields:

$$h_{ca}\ (T_a - T_w) + h_D\ h_{fg}(\omega_a - \omega_w) + U(T_f - T_w) = 0 \tag{15}$$

Likewise, a mass and energy balance on the airstream yields:

$$\dot{m}_a \frac{\omega}{dy} = h_D W\ (\omega_w - \omega_a) \tag{16}$$

$$\dot{m}_a \frac{dh_a}{dy} = h_{ca} W\ (T_w - T_a) + h_D h_g W(\omega_w - \omega_a) \tag{17}$$

Two assumptions are now made to simplify the cooling tower model. The first is that the Lewis Number is unity, and the second is that the saturation curve is linear in the range of temperatures and humidity ratios experienced by the cooling tower. With these assumptions, it is possible to manipulate Equations 14 through 17 to solve for the wet-bulb depression of the airstream as it passes through the cooling tower.

$$T_{a,out} - T'_{a,out} = (T_{a,in} - T'_{a,in})\ \exp - \left(\frac{h_c A_a}{\dot{m}_a c_{p,da}}\right) \tag{18}$$

The expression in the exponent is the cooling tower NTU and combines the air-side heat transfer coefficient and tower surface area along with the specific heat and mass flow rate of the air.

It is possible to simplify Equation 15 to:

$$h_{wb}\ (T'_a - T_w) + U(T_f - T_w) = 0 \tag{19}$$

by defining a wet-bulb heat transfer coefficient as:

$$h_{wb} = h_c(1.0 + e h_{fg}/c_{p,da}) \tag{20}$$

where e is the slope of the assumed linear saturation line. It is also possible to simplify Equation 17 to:

$$\dot{m}_a c_{wb} \frac{dT'_a}{dy} = h_{wb} W(T_w - T'_a) \tag{21}$$

by defining a wet-bulb specific heat as:

$$c_{wb} = c_{p,da}\left(1.0 + eh_{fg}/c_{p,da}\right) \tag{22}$$

The resulting Equations 14, 19, and 21 are of the same form as the equations encountered in sensible heat exchanger theory. These have been solved previously, and the solutions result in the familiar effectiveness-NTU equations for counter and parallel flow heat exchanger configurations [Kays and London 1984]. Equation 22 may be written in terms of air mixture enthalpies and humidity ratios as:

$$c_{wb} = \frac{h'_{a,out} - h'_{a,in} - (\omega'_{out} - \omega'_{in})h_f}{(T'_{out} - T'_{in})} \tag{23}$$

Equation 23 allows for the calculation of the capacitance rates for the cooling tower air and water streams. By choosing a tower NTU, the exiting air wet-bulb temperature can be calculated with standard heat exchanger effectiveness equations. Equation 18 may then be used to solve for the wet-bulb depression of the tower air and, when used with the previously calculated exiting wet-bulb temperature, fixes the state of the air exiting the cooling tower.

Figures 8 shows the system performance as a function of the fraction of air delivered. The cooling unit NTU is 3 and the air inlet state is 86°F (30°C) and 0.009 lbm/lbm absolute humidity ratio. The family of curves describes the system performance. For any one given fraction of air delivered, there are many possible water flow rates. There is one optimum water flow rate, however, that maximizes the system performance. For example, at a fraction delivered of 0.7, the optimum water flow rate would be approximately 25% of the inlet air flow rate. As the fraction delivered value decreases, the optimum water flow rate increases. Figure 9 illustrates the variation in system performance with water flow rate for a cooling unit with heat exchanger and cooling tower NTUs of 3.

The maximum in performance results from heat transfer limitations. At very low water flow rates, the water circulating loop is the minimum capacitance stream for both the heat exchanger and the cooling tower. In the cooling tower, the water temperature decrease will be greatest and the temperature will approach the wet-bulb temperature of the air at state 2. However, the air passing through the heat exchanger will have a capacitance rate far greater then the water and will not be able to approach the cold water temperature offered by the cooling tower. At the other extreme, at high water flow rates, the air in the heat exchanger and the cooling tower will become the minimum capacitance streams. The water passing through the cooling tower will have a smaller temperature change then the air and exit at a temperature substantially warmer then the wet-bulb temperature of the air at state 2. In the heat exchanger, the air will have the larger temperature change and exit at a temperature close to the warm inlet water temperature.

At low water flow rates, the cooling tower is better able to cool the water loop, but the heat exchanger is less able to use the cold water to cool the process air. At high water flow rates, the heat exchanger is better able to cool the process air closer to the water inlet temperature, but the tower cannot provide the heat exchanger with water as cold. Operating the system at a flow rate between the two flow rate extremes will, therefore, optimize the system performance.

The best system performance for fractions delivered ranging from 0.6 to 1.0 occurs when the water flow rate is approximately 25% of the inlet air flow rate. Since the specific heat of water is approximately four times that of air, this flow rate ratio corresponds to nearly equal capacitance rates in the heat exchanger.

If the optimum water flow rate is chosen for all of the fractions delivered, the family of curves in Figure 8 would be enclosed by one curve that characterizes the best system performance for all possible operating points. Figures 10 and 11 show the outer envelope of the temperature and cooling capacity effectiveness curves for cooling units with NTUs ranging from 1 to infinity and an inlet state of 86°F (30°C) and 0.009 lbm/lbm (kg/kg) absolute humidity ratio. With the best possible cooling unit (infinite NTU) it is possible to chill the inlet air to the dew-point temperature and still deliver approximately 70% of the inlet air to the building zone.

Lines of constant outlet temperature for a CT/HX with NTU of 3 and a fraction delivered of 0.75 are mapped in Figure 12. The fraction of 0.75 for the process air was chosen since this is an operating point near the maximum in the cooling capacity effectiveness curves. The lines of constant outlet temperature resemble those for a single- and a multi-stage IEC, and intercept the saturation curve at a temperature equal to the value of the constant temperature line.

The variation of COP with inlet air state is also shown in Figure 12 for a fraction delivered value of 0.75, a pressure drop of 0.5" H_2O (125 Pa), and a heat exchanger and cooling tower NTU of 3.0. These lines have the same shape as the saturation curve. In the range of air conditioning practice, the COPs ranges from 10 to 75.

CONCLUSION

This paper has presented performance relations and characteristics for single- and multiple-stage indirect evaporative coolers and a combination cooling tower/heat exchanger.

The single-stage IEC can cool the inlet airstream close to the inlet wet bulb temperature. Multiple-stage IECs have increased cooling ability and are capable of cooling air close to the dew-point temperature. The cooling tower/heat exchanger air cooler can cool air close to the dew point with only one stage of cooling. Coefficients of performance for these air coolers range from 25 for the single-stage IEC up to 75 for the cooling tower/heat exchanger in the range of air states seen in air-conditioning practice.

The cooling tower/heat exchanger air cooler has the greatest thermal potential for air-conditioning applications. It can produce the lowest temperatures and highest cooling capacities for any value of fraction of inlet air delivered to the space. A possible drawback to this system, is its complexity. This cooler uses a cooling tower and requires a water circulating loop. The performance of this cooler is sensitive to the water flow rate, and a smart control system would be required to operate near the optimum conditions.

NOMENCLATURE

A = Transfer area in wet surface heat exchanger (ft^2, m^2)

c_p = Specific heat (Btu/lbm°F, J/kg°C)

c_{wb} = Wet bulb specific heat (Btu/lbm°F, J/kg°C)

C = Capacitance rate (Btu/hr°F, W/°C)

COP = Air-cooling benefit/energy input (dimensionless)

e = Slope of assumed linear saturation line ($°F^{-1}$, $°C^{-1}$)

F = Fraction of inlet air to the air cooler that is delivered to conditioned zone (dimensionless)

h = Enthalpy (Btu/lbm, J/kg)

h_c = Heat transfer coefficient (Btu/hrft^2°F, W/m^2°C)

h_D = Water vapor transfer coefficient (lbm/ft^{2}hr, kg/m^{2}s)

h_f = Enthalphy of liquid water (Btu/lbm, J/kg)

h_g = Enthalpy of water vapor (Btu/lbm, J/kg)

h_{fg} = Latent heat of vaporization (Btu/lbm, J/kg)

h_{wb} = Wet-bulb transfer coefficient (Btu/hr-ft^{2}F, W/m^{2}C)

k = Constant in Equation 10 (dimensonless)

$\dot{m}$ = Mass flow rate (lbm/hr, kg/s)

q = Heat transfer rate (Btu/hr, W)

T = Temperature (°F, °C)

U = Overall heat transfer coefficient between water film surface and fluid stream in wet surface heat exchanger (Btu/hr-ft^{2}F, W/m^2 C)

W = Width of wet surface heat exchanger (ft, m)

$\dot{w}$ = Work rate (Btu/hr, W)

ε_{DEC} = Direct evaporative cooler effectiveness (dimensionless)

ε_{HX} = Sensible heat exchanger effectiveness (dimensionless)

ε_{IEC} = Indirect evaporative cooler effectiveness (dimensionless)

p = Pressure drop (psi, Pa)

ρ = Air density (lbm/ft^3, kg/m^3)

ω = Humidity ratio (lbm/lbm, kg/kg dry air)

Subscripts

a = moist airstream

cc = cooling capacity

da = dry air

delivered = delivered airstream

dew pt = dew point conditions

dump = dumped airstream

f = fluid stream

inlet = inlet airstream

max = maximum

min = minimum

n = number of stages

outlet = outlet airstream

proc = process airstream

T = temperature

w = at water film surface

wb = wet-bulb

Superscripts

' at saturation conditions

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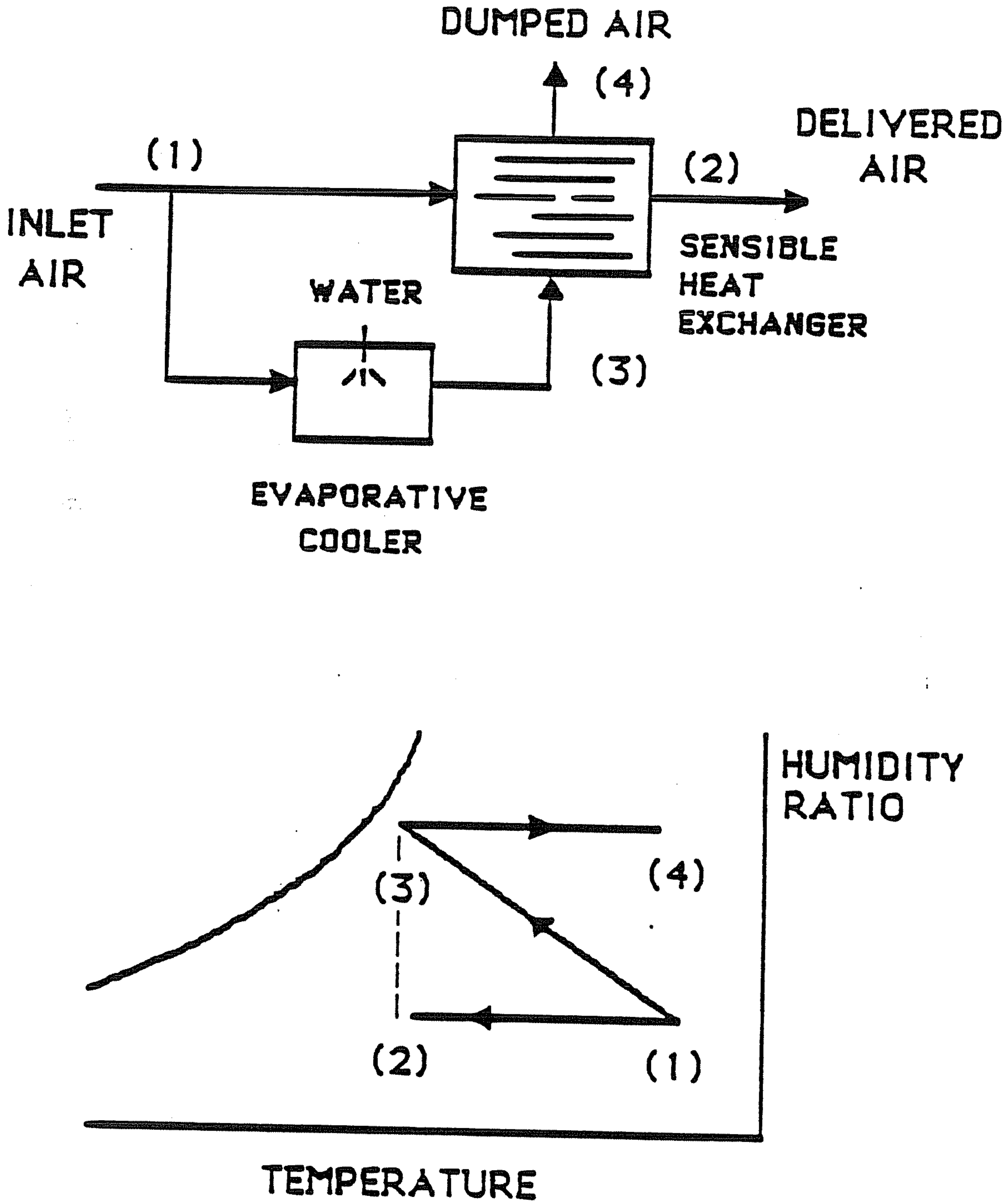
LIST OF FIGURES


DUMPED AIR
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(1)
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AIR
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AIR
SENSIBLE
HEAT
EXCHANGER
WATER
(3)
EVAPORATIVE
COOLER
HUMIDITY
RATIO
(3)
(4)
(2)
(1)
TEMPERATURE

Figure 1

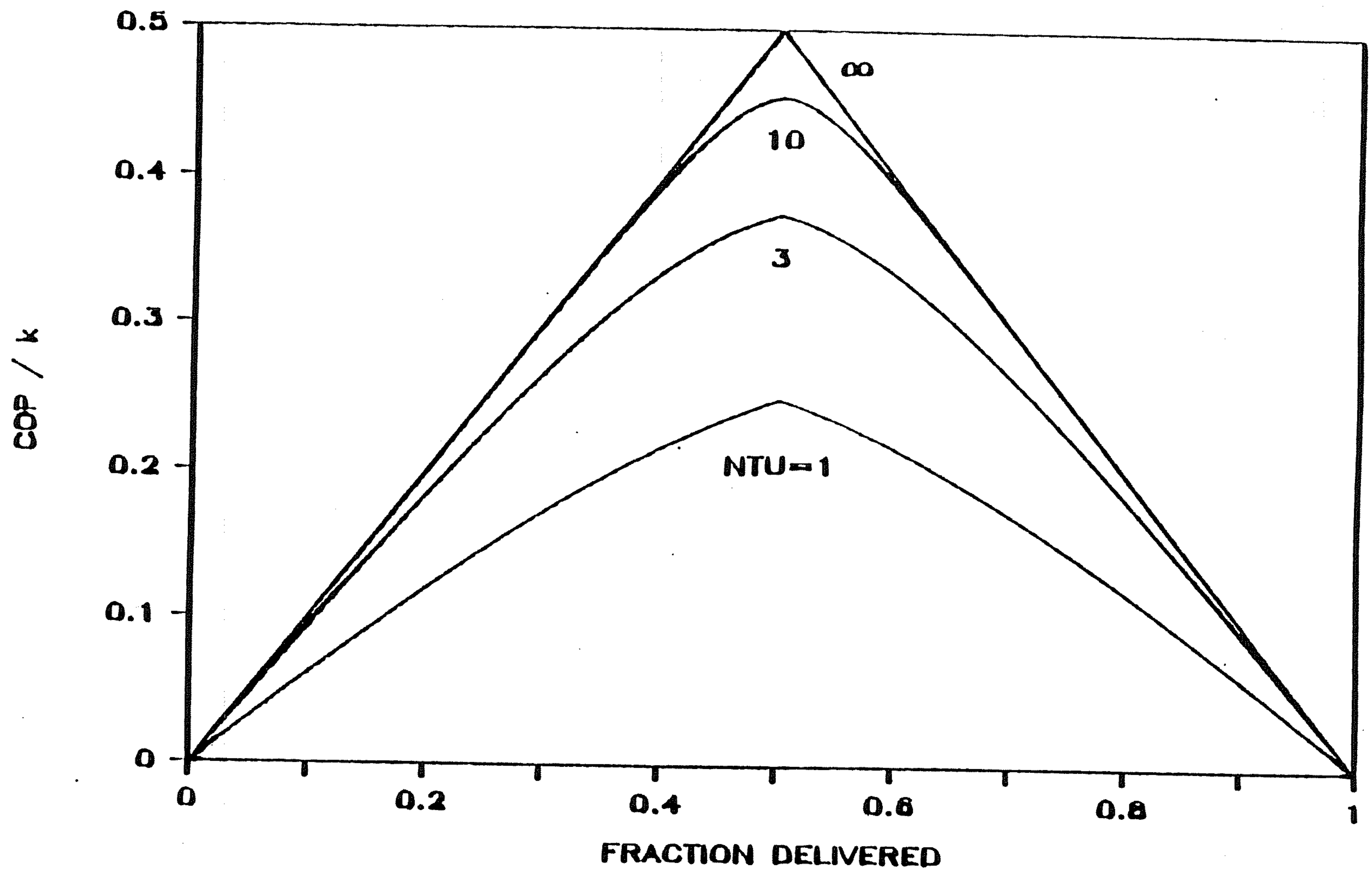


Figure 2

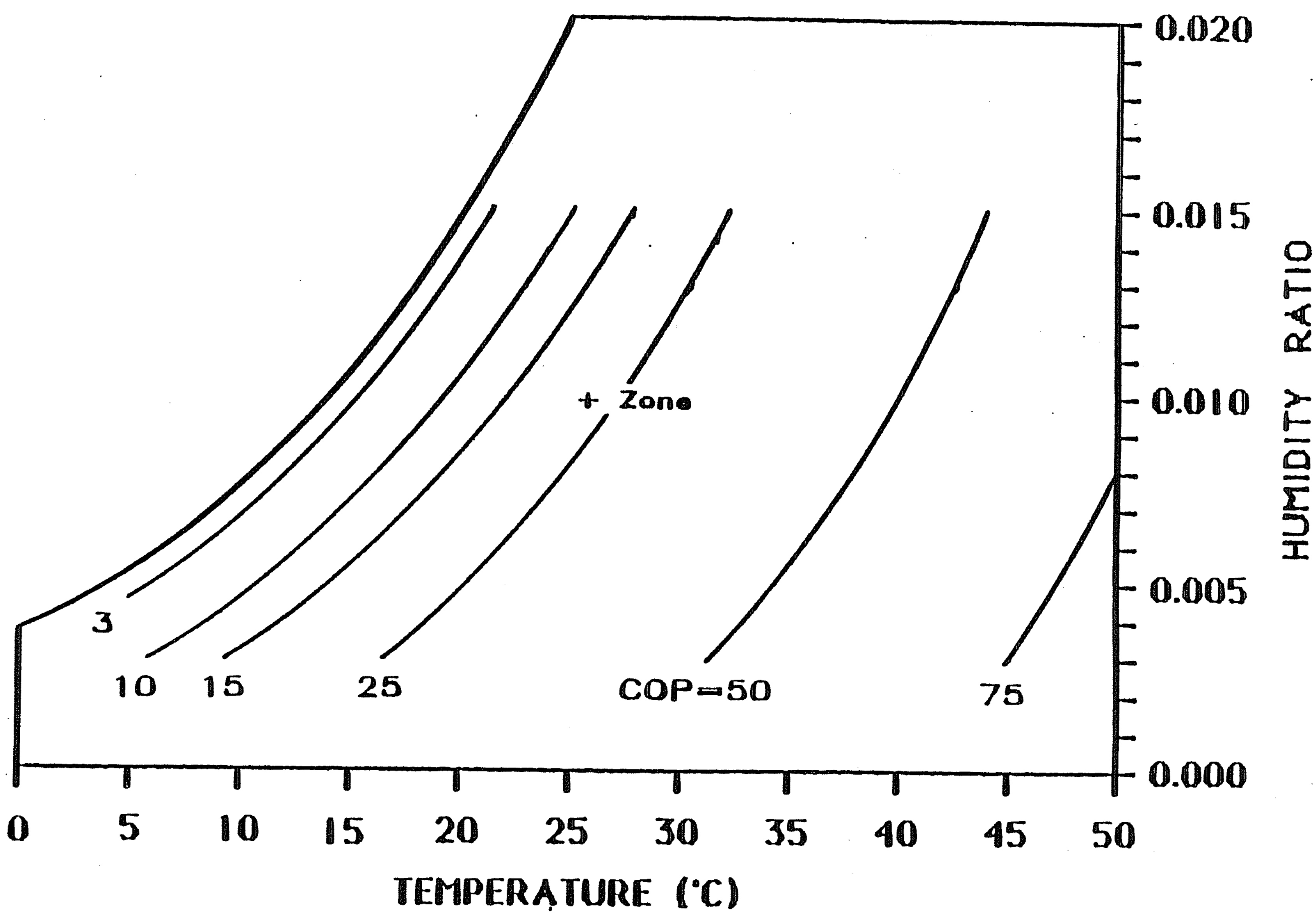
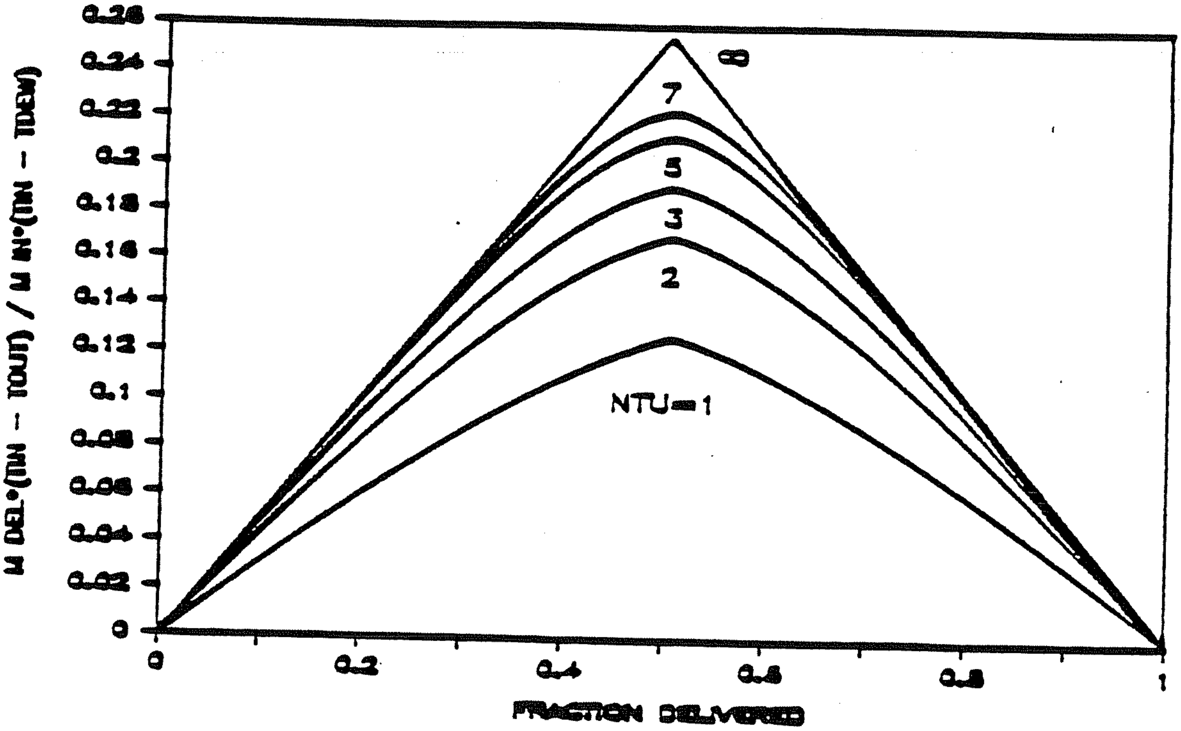
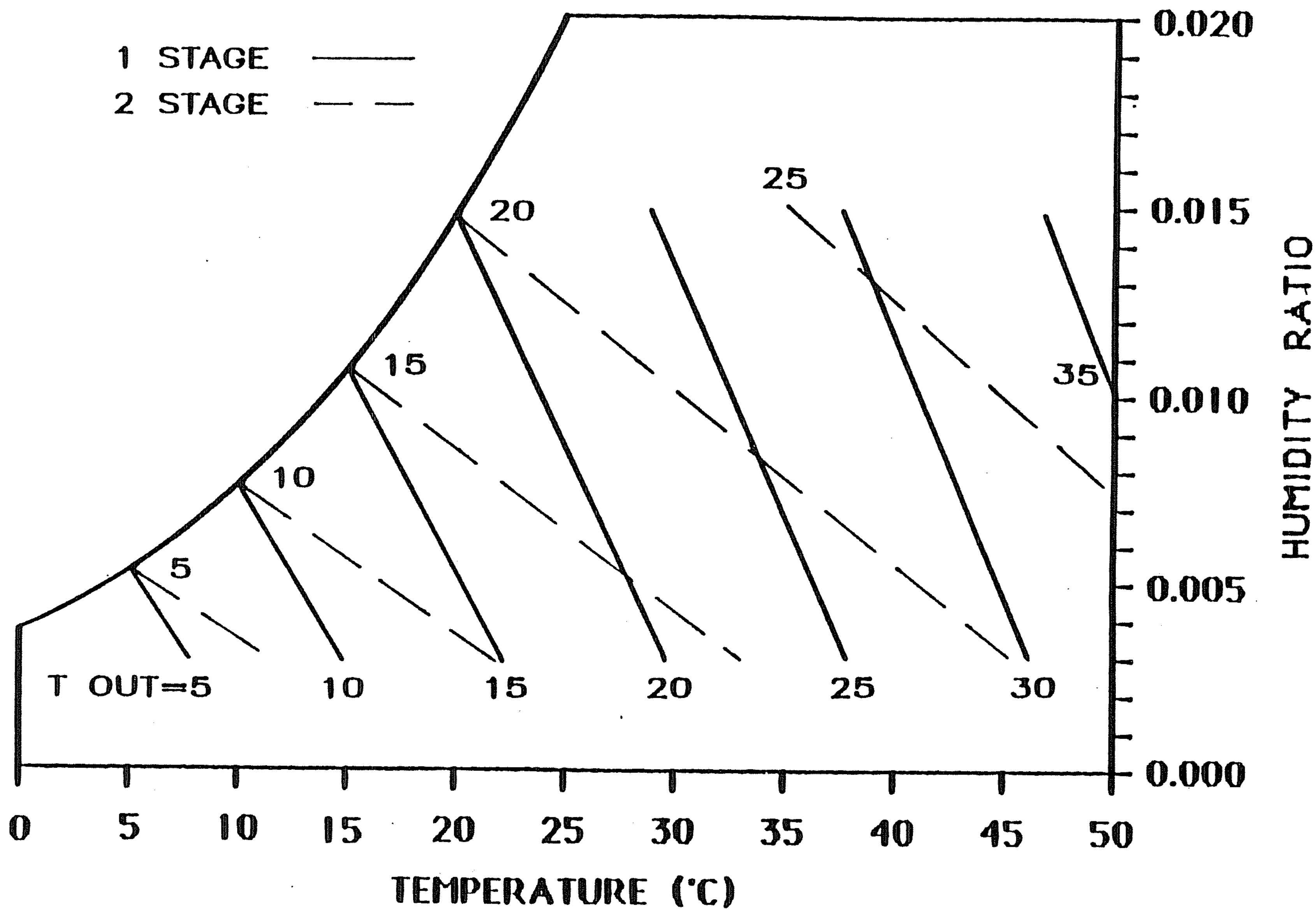

0.020
0.015
0.010
0.005
0.000
HUMIDITY RATIO
3
10
15
25
+ Zone
COP=50
75
0
5
10
15
20
25
30
35
40
45
50
TEMPERATURE (°C)

Figure 3



M DEL•(TIN - TOUT) / M IN•(TIN - TDEW)
0.26
0.24
0.22
0.2
0.18
0.16
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0
∞
7
5
3
2
NTU=1
0
0.2
0.4
0.6
0.8
1
FRACTION DELIVERED

Figure 4



1 STAGE
2 STAGE
T OUT=5
10
15
20
25
30
5
10
15
20
25
35
0
5
10
15
20
25
30
35
40
45
50
TEMPERATURE (°C)
0.000
0.005
0.010
0.015
0.020
HUMIDITY RATIO

Figure 5

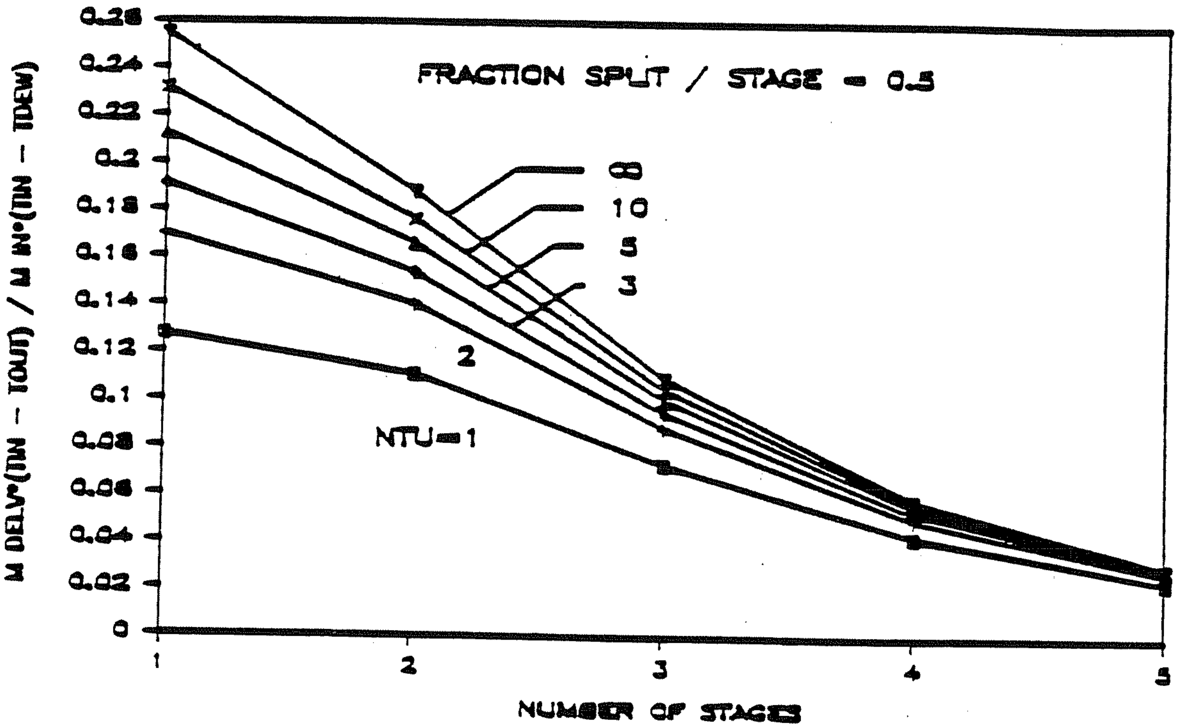

FRACTION SPLIT / STAGE = 0.5
M DELV*(TIN - TOUT) / M IN*(TIN - TDEW)
NUMBER OF STAGES
∞
10
5
3
2
NTU=1
0.26
0.24
0.22
0.2
0.18
0.16
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0
1
2
3
4
5

Figure 6

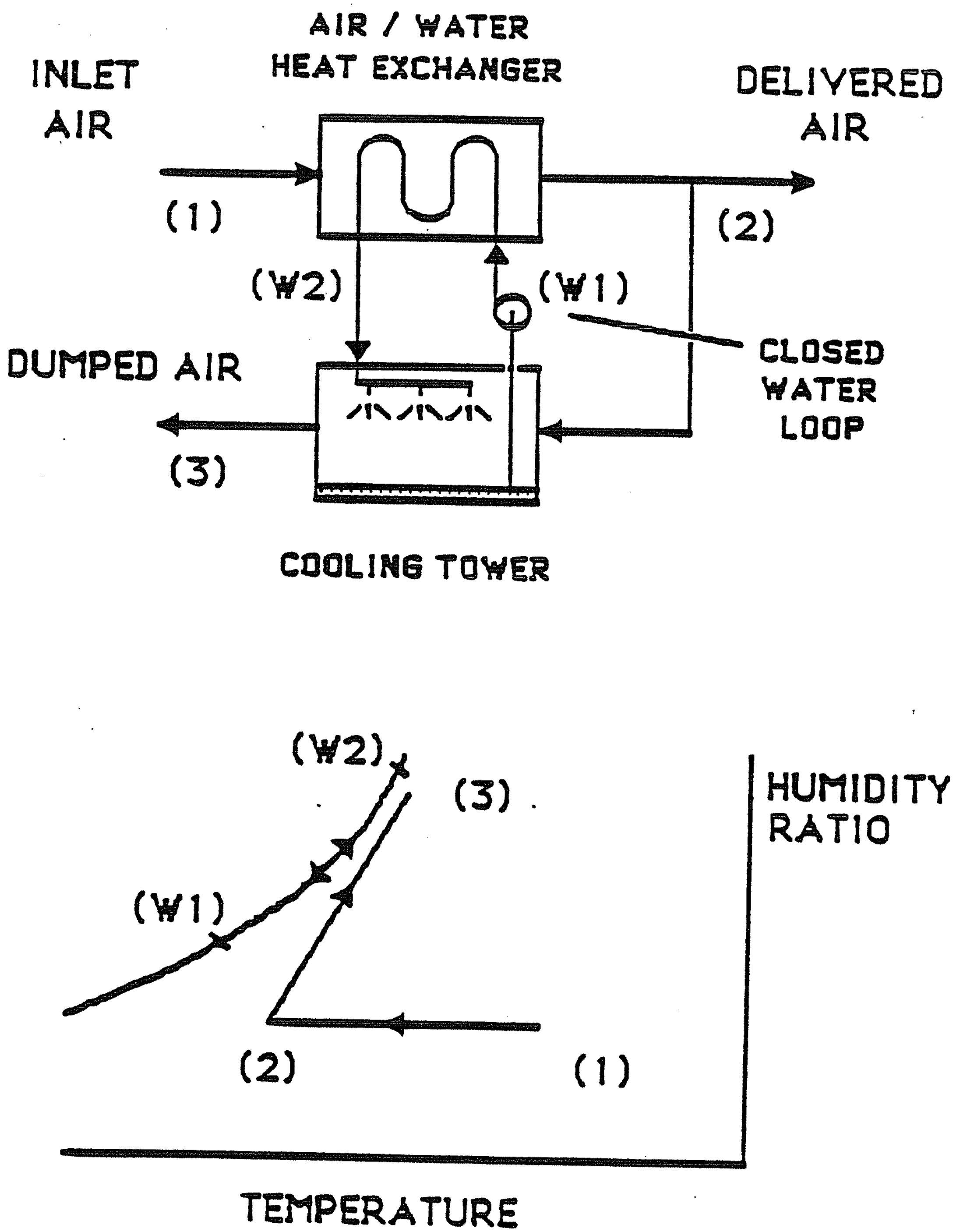

AIR / WATER
HEAT EXCHANGER
INLET
AIR
DELIVERED
AIR
(1)
(2)
(W2)
(W1)
CLOSED
WATER
LOOP
DUMPED AIR
(3)
COOLING TOWER
(W2)
(3)
HUMIDITY
RATIO
(W1)
(2)
(1)
TEMPERATURE

Figure 7

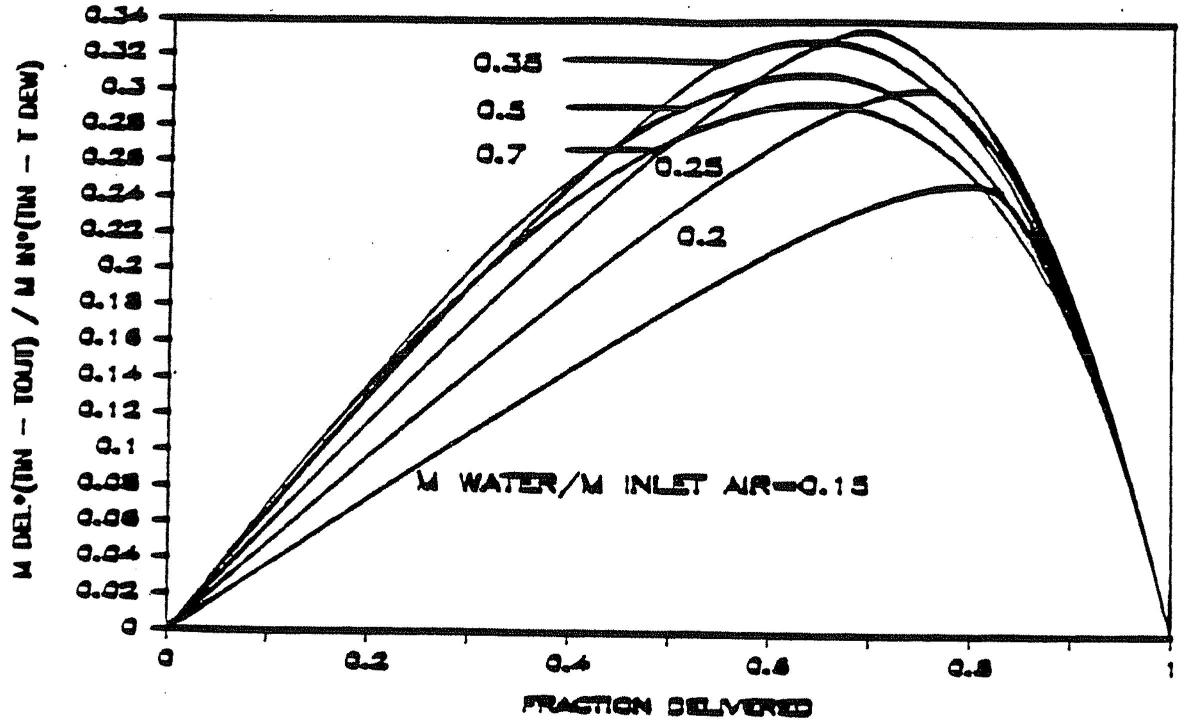


Figure 8

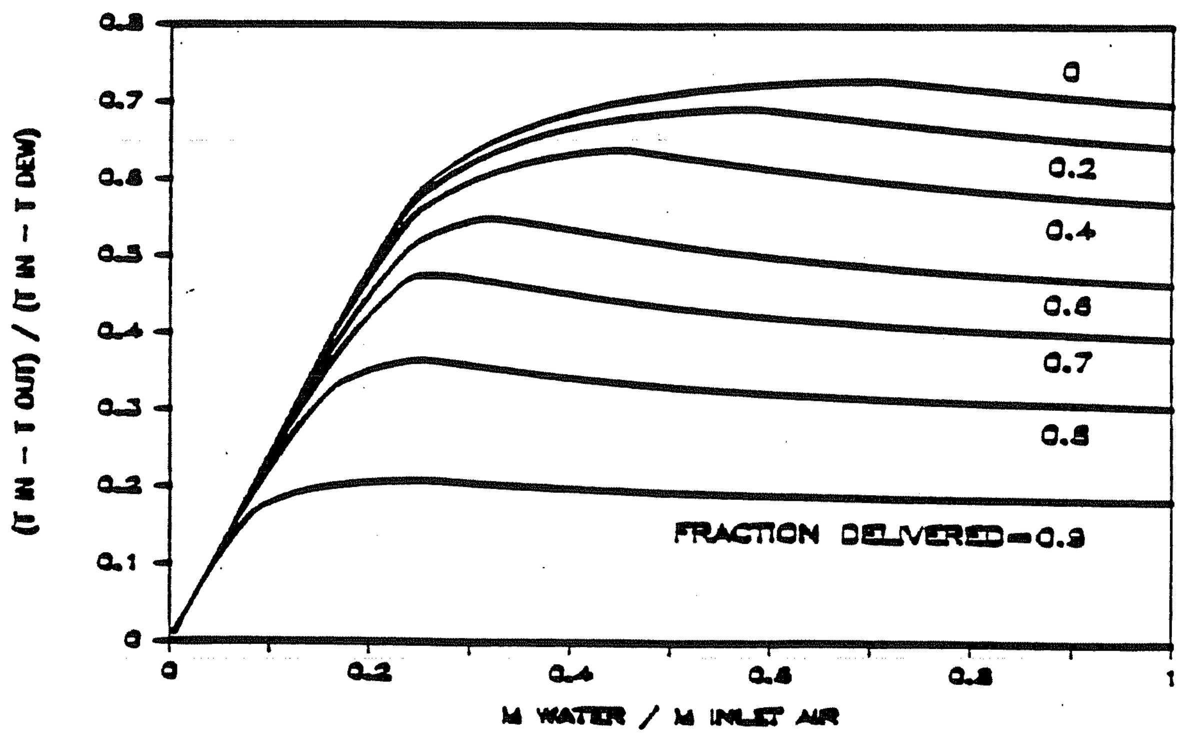


Figure 9

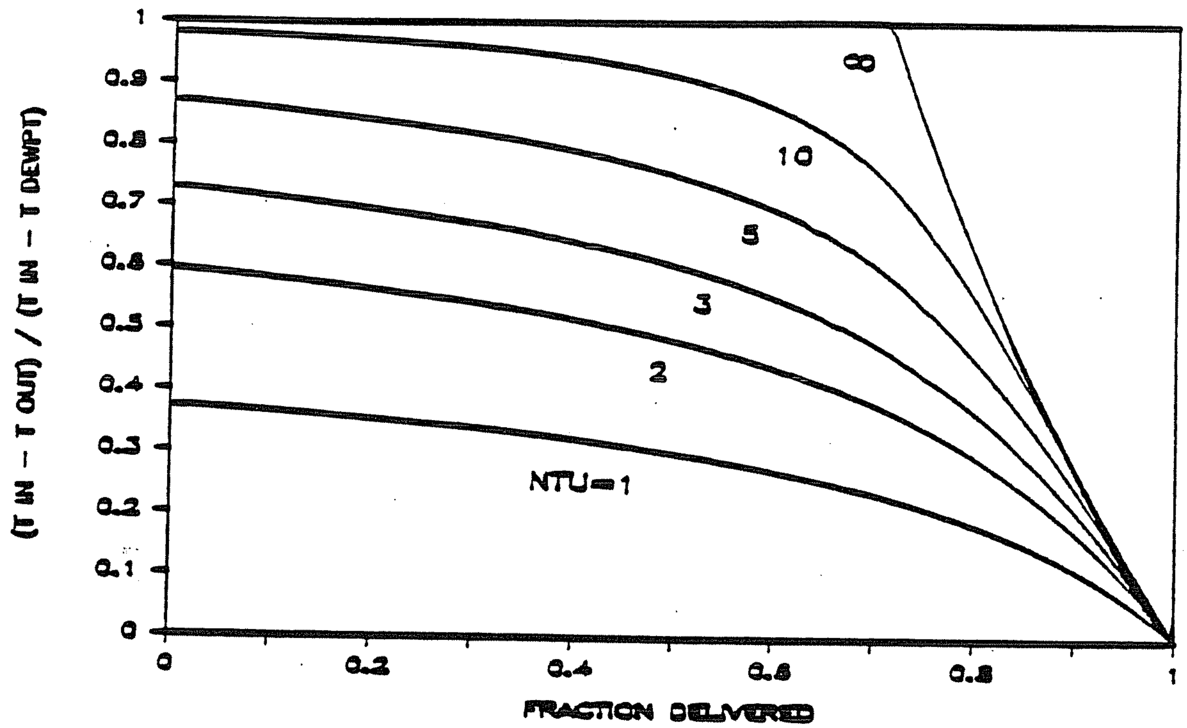


Figure 10

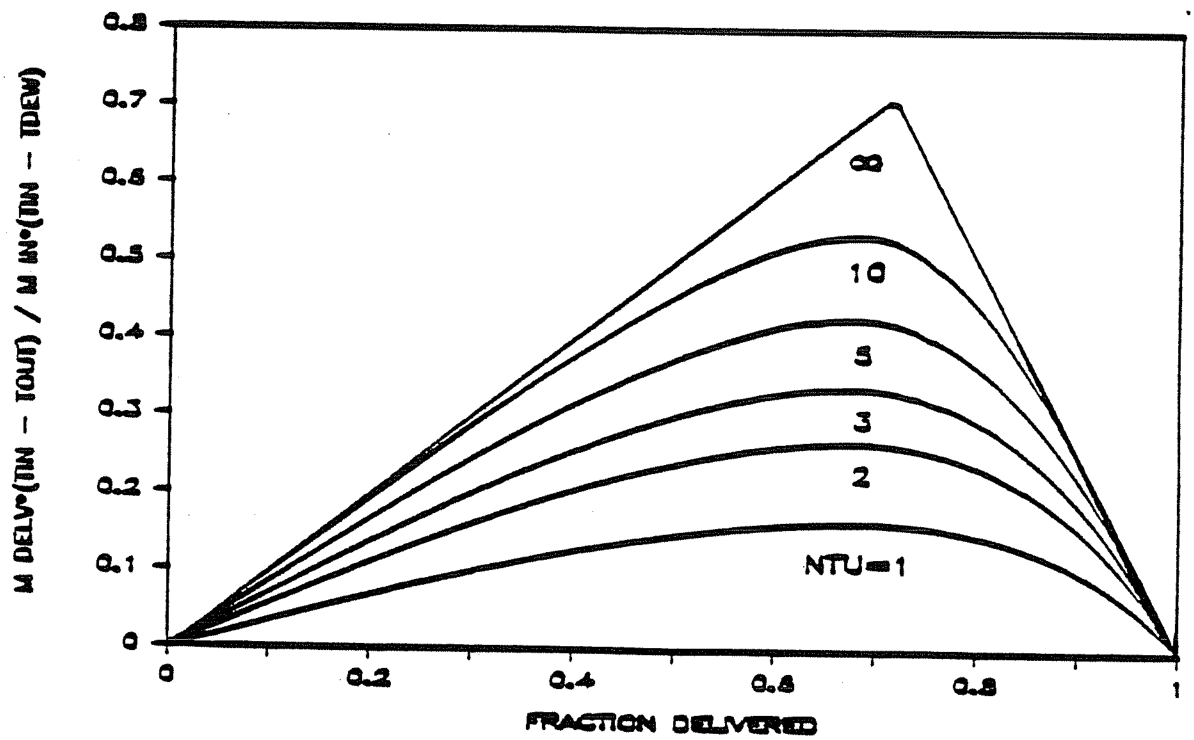


Figure 11

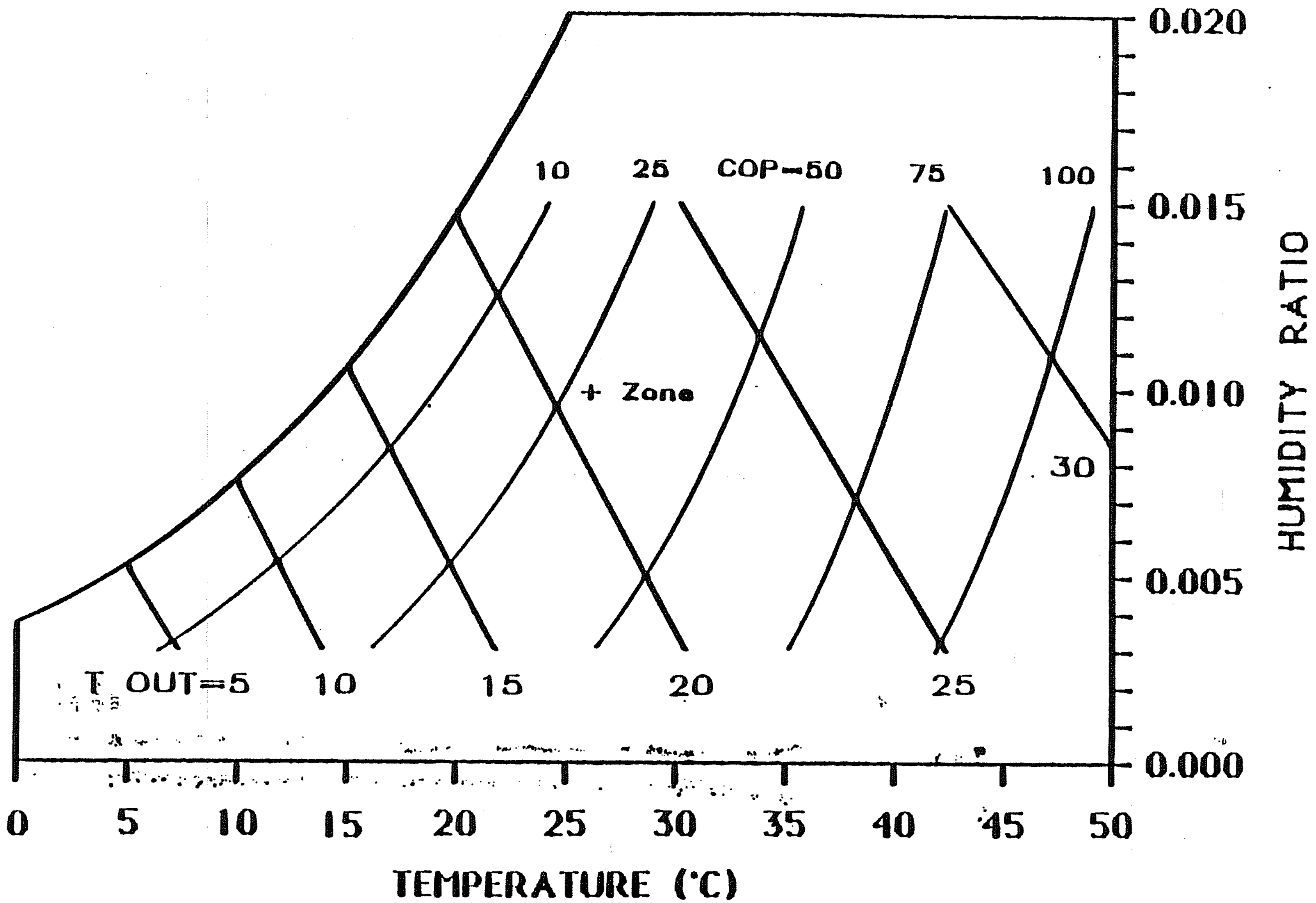

10
25
COP=50
75
100
+ Zone
30
T_OUT=5
10
15
20
25
0.020
0.015
0.010
0.005
0.000
HUMIDITY RATIO
0
5
10
15
20
25
30
35
40
45
50
TEMPERATURE (°C)

Figure 12