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**HEAT GAIN AND LOSS ASSOCIATED WITH
WINDOW'S SURFACE CONDENSATION**

Ismail M. Budaiwi

**A Thesis
in the
Centre for Building Studies
Faculty of Engineering and Computer Science**

**Presented in Partial Fulfillment of the Requirements
for the Degree of Master of Engineering (Building)
Concordia University
Montréal, Québec, Canada**

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ABSTRACT

HEAT GAIN AND LOSS ASSOCIATED WITH WINDOW'S SURFACE CONDENSATION

Ismail M. Budaiwi

Condensation on glazed windows could be a considerable source of heat gain or loss in buildings. However, very little effort has been made to predict mass condensation rate and the corresponding heat production. In the present study, a surface condensation model has been developed using a transient unidimensional finite-difference formulation. The model predicts temperature gradient across the window, mass condensation rate, and the corresponding heat gain due to external surface condensation in hot-humid climates and heat loss due to internal surface condensation in cold climates. Mass condensation rate has been predicted by two different approaches. The first approach is based on the mass transfer theory, and the second approach is based on Nusselt's theory, which, unlike the mass transfer theory, takes into account the effect of the water film and the height of the surface upon which, condensation occurs. Yet, Nusselt's theory has been modified to account for the presence of large percentage of the noncondensable gas (air) to be applicable for atmospheric vapour condensation. Heat gain and loss due to surface condensation have been calculated, in this study, when the condensation process reaches steady state stage. To make the model more practical and easier to

use, a computer program has been developed to solve for the mass condensation rate and the corresponding heat gain or loss at any temperature and humidity conditions. The analytical results have shown that mass condensation predicted by the second approach is always less than that predicted by the first approach. To verify the applicability of one approach over the other, an experiment has been carried out. Over fifty tests have been conducted on a single glazed window simulating different climatic conditions. From these results, it has been concluded that the second approach is theoretically and practically justified over the first approach. By using the second approach, curves were constructed for fast prediction of heat gain and loss due to condensation on selected single and double glazed windows. These curves show that heat loss due to internal surface condensation on single glazed windows is the biggest concern as far as heat gain and loss due to surface condensation are concerned. The importance of heat loss due to condensation on double glazed windows is increased as the indoor relative humidity gets higher and the outdoor temperature gets lower. But, it is negligible at low indoor relative humidity. Generally, higher relative humidity in the condensing side increases the importance of heat gain or loss as compared to the conduction heat gain or loss due to temperature difference.

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I wish to dedicate this thesis to my mother, my wife and my daughter ARWA for their patience and sacrifice.

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NOMENCLATURE

A	area, m^2
AS	air space thickness, mm
C _p	specific heat at constant pressure, J/Kg-C
C _v	specific heat at constant volume, J/Kg-C
D	diffusion coefficient, m^2/s
DBT	dry bulb temperature, °C
DP	dew point temperature, °C
E	energy, J
f	dimensionless stream function
F _{S-R} , F ₁₂	configuration factor
g	gravitational constant, m/s^2
Gr	Grashof number
h	surface heat transfer coefficient, W/m^2-C
h _D	mass transfer coefficient, m/s
h _{fg}	latent heat of vaporization, J/Kg
K	thermal conductivity, $W/m-C$
KAGL	combined air and glass thermal conductivity $W/m-C$
L	length, m
M	molecular weight Kg/Kmol, glass thickness, mm
m	mass, Kg
\dot{m}	mass condensation rate, $Kg/m^2.S$
P	pressure, N/m^2
Pr	Prandtl number, dimensionless
q	local heat transfer rate per unit area W/m^2
Q	overall transfer rate per unit area, W/m^2
r	thermal resistance, m^2-C/W

R	overall thermal resistance $\text{m}^2\text{-C/W}$, universal gas constant J/Kmol-K
Ra	Rayleigh number, dimensionless
Re	Reynolds number, dimensionless
RH	relative humidity, %
SC	Schmidt number, dimensionless
SVP	saturation vapour pressure N/m^2
T	temperature, $^{\circ}\text{C}$
t	time, S
u	longitudinal velocity, m/s
U	overall heat transfer coefficient $\text{W/m}^2\text{-C}$
v	normal velocity m/s
W	humidity ratio KgV/KgA , noncondensable gas concentration, %
WAS	air space thickness, mm
WBT	wet bulb temperature, $^{\circ}\text{C}$
WV	wind velocity, m/s
x	mole fraction
x,y	normal and longitudinal coordinates
β	expansion coefficient, $1/\text{K}$
δ	condensate layer thickness, m
ρ	density, Kg/m^3
μ	dynamic viscosity, Kg/m.s
ν	Kinematic viscosity, m^2/s
ϕ	relative humidity,
ϕ	mass fraction difference
θ	inclination angle, degrees

Ψ	stream function
η	similarity variable
α	thermal diffusivity m^2/S
σ	Boltzmann constant $\text{W}/\text{m}^2\text{-K}^4$
τ	time increment

Subscripts

a	air, bulk
C	convective, critical, condensation
D	dew point
g	gas
i	indoor or internal, interfacial
L	liquid or fluid
m	mean, number of finite difference node in x-direction
O	outdoor or external
p	pressure
r	radiative
s,sat	saturation conditions
ss	steady state
t	temperature
tot	total
v	vapour
x,y	directions in cartesian coordinate system
w	water, wall
∞	bulk

CHAPTER I

INTRODUCTION

1.1 BACKGROUND

Glass windows are very essential requirement in most modern buildings. They serve a number of functions that are intimately tied to the quality of the built environment. The major benefits provided by glass windows are:

1. Serve as a means of visual communication with the outside world.
2. Provide natural ventilation to improve indoor air quality.
3. Admit daylight into interior spaces, not only to psychologically and aesthetically improve the space, but to provide quality lighting at the appropriate quantitative level for the excellence of seeing.
4. Allow the sunshine into the space, so it can be heated to the desired temperature by solar radiation in winter.
5. Provide spaciousness, which means that the size of the space appears to be larger in the presence of the window.

Although, glass windows provide daylight, ventilation and other benefits, they also can allow undesirable heat gain and loss. Moreover, surface condensation is a common problem associated with glass windows, especially single glazed ones, because they are normally the coldest part in the building. Therefore, careful assessment of the thermal and functional performances of glass windows has to be done to reach an optimal solution regarding the type, the size and the location of windows. Eventually, considerable effort has been made to develop information and strategies to reduce energy-wasting aspects of windows, while improving the energy-gaining aspects. In other words, to optimize the solar radiation and light available through windows [1,2,3,4].

The major part of heat gain and loss in buildings is experienced through exterior walls and windows. Glass windows, especially single glazed type, have the highest potential to lose and gain heat through. There are mainly three forms of gaining heat and two forms of losing heat through glass windows. Heat can be gained through glass windows by the following forms:

1. Direct solar radiation coming through the glass, which will be absorbed by the interior surfaces of the building.
2. Conduction through the glass due to temperature difference between inner and outer surfaces of the window.

3. Infiltration of hot outdoor air through openings and cracks of windows due to the presence of high pressure zone over the building exterior surface which is caused by the wind.

Heat can be lost from the building through the glass windows by the following forms:

1. Conduction through the glass due to indoor and outdoor temperature difference.
2. Infiltration of outdoor cold air due to density difference between indoor warm air and outdoor cold air.

The amount of air infiltration depends on the characteristics of the building and how tight it is. The tighter the building is the less chance for air to get in or out of the building. However, regardless of the degree of tightness air can find its way into or out of the building due to the fact that cracks do always exist in the building envelope. The process of air infiltration in buildings involves both sensible and latent heat gain or loss. In winter, cold dry air infiltrating into the building affects both the indoor temperature and the relative humidity. Accordingly, heat and humidity need to be supplied to the space in order to keep it comfortable. In summer, hot humid air infiltrating to the space causes the indoor temperature and relative humidity to rise. Consequently, heat and moisture need to be removed to keep the space comfortable.

The amount of direct and indirect heat gain by solar radiation through glass windows is a function of many variables related to the amount of solar radiation received and the characteristics of the window. The amount of solar radiation depends on the location of the building, the time of the year, the time of the day and the general climatic and environmental conditions. The amount of heat gained due to solar radiation depends on the type of glazing, the location of the window with respect to the building, the area of the window and the degree of shading. solar radiation can heat the space by direct sunshine through the glass, or indirectly through the inward flowing fraction of absorbed heat by the glass. The combined effect of solar radiation, radiant energy exchange with surroundings and the air temperature on the surface temperature is expressed in terms of sol-air temperature, which is used to evaluate heat gain by conduction.

Surface condensation on glass windows is a common problem associated with certain climatic conditions. Outdoor surface condensation is a common phenomenon in hot-humid climates. And indoor surface condensation is likely to occur in cold climates. Damage of building materials due to condensation on glass windows is not a serious problem provided the window frames are properly painted and the condensate is wiped up regularly and not allowed to soak into the wood frame and wet the wall. However, condensation on single glazed windows can remove quite a lot of water vapour from the air,

therefore, surface condensation could be an important means of losing indoor moisture when internal surface condensation occurs.

The significance of condensation heat gain or loss depends on the indoor and outdoor conditions, as well as the characteristics and properties of windows and walls. When condensation conditions prevail, a single glazed window, which has low thermal resistance compared to the rest of the building envelope, has the highest risk for condensation and removal of water vapour from air. In addition, the significance of condensation heat gain and loss depends on the glazed percentage of the exterior walls as well as the absolute glazed area. So, the greater the glazing percentage the greater the importance to consider heat loss or gain due to condensation.

1.2 FUNDAMENTAL PROPERTIES OF MOIST AIR

Atmospheric air is a mixture of many gases as well as water vapour and countless pollutants. Assuming no pollutants are present in the mixture, the composition of the dry air is relatively constant varying slightly with time, location and latitude [5]. However, the atmospheric moisture content, which is a main concern in this study, is not constant and vary according to the general climatic conditions and the air temperature.

Surface condensation and the corresponding heat transfer are dependent upon the moist air properties that describe the behavior of water vapour in air. Condensation of atmospheric water vapour is

governed by the moisture content, wet bulb temperature, dry bulb temperature, relative humidity, dew point temperature and vapour pressure. The resulting heat transfer is determined by how much heat is released due to vapour condensation, which in turn depends on the amount of the latent heat in the air-vapour mixture, and the amount of condensed water. The following is a review of the fundamental properties of moist air that governs condensation and corresponding heat transfer.

1.2.1 Relative Humidity

The relative humidity (ϕ) is the ratio of the mole fraction of the water vapour (X_v) in a mixture to the mole fraction (X_s) of the water vapour in a saturated mixture at the same temperature and pressure [5]. It is a direct index to the potential of evaporation, since it indicates how much more water vapour the air can sustain. However, condensation potential on a given surface at a particular temperature can be determined if at least two properties of the mixture are known. The relative humidity does not indicate the state of comfort, unless it is associated with the dry bulb temperature. Generally, the range of relative humidity from 30% to 65% is considered acceptable for achieving comfort [6]. Relative humidity can be expressed in terms of relative mass fraction as:

$$\phi = \left| \frac{X_v}{X_s} \right|_{t,p} \quad (1.1a)$$

or in terms of the relative vapour partial pressure as:

$$\phi = \left| \frac{P_v}{P_s} \right|_{t,p} \quad (1.1b)$$

where

P_v = partial vapour pressure in a standard mixture

P_s = saturation partial vapour pressure

1.2.2 Specific Humidity

Specific humidity (W) is defined as the ratio of the mass of the water vapour (m_v) to the mass of dry air (m_a) in a given mixture [5]. In terms of weight, it can also be defined as the weight of water vapour mixed with each kilogram of dry air in the air-water vapour mixture [7].

$$W = \frac{m_v}{m_a} \quad (1.2)$$

1.2.3 Dry Bulb Temperature

Dry bulb temperature (DBT) is a direct indication of the amount of sensible heat available in the air-water vapour mixture. Therefore, it is considered the most important direct index for comfort. However, its importance varies with the relative humidity. So, it is an important comfort indication in dry cold regions and less important at high relative humidity.

1.2.4 Wet Bulb Temperature

Wet bulb temperature (WBT) is a direct indication of the total heat content in the air-water vapour mixture. It takes into account the relative humidity and the dry bulb temperature. Therefore, it is an important index for comfort in regions where both temperature and humidity are high. The wet bulb temperature can be read from a thermometer with a wetted cotton wick covering the bulb. Its value is determined by two opposite forces. First, the sensible heat which tries to raise the reading to the dry bulb temperature. However, as long as there is a chance for evaporation from the wetted wick, there will be a second force trying to push the reading down. The net result of these two forces gives the wet bulb temperature reading. So, the higher the relative humidity, the smaller the deviation from the dry bulb temperature. But the wet bulb temperature is never below the dew point temperature, and never higher than the dry bulb temperature.

1.2.5 Dew Point Temperature

Dew point temperature (DP) is the temperature of saturated moist air at the same pressure and humidity ratio as the given mixture [5]. Therefore, it is the most important index in this study, since it defines a temperature below which condensation starts to occur. The dew point temperature is directly related to the partial vapour press-

ure in a given mixture. The following equation is suggested by ASHRAE to relate the dew point temperature with the vapour pressure for the temperature range of 0°C to 70°C.

$$DP = - 35.957 - 1.8726 \alpha + 1.1689 \alpha^2 \quad (1.3)$$

where

$$\alpha = \log_e (P_w)$$

P_w = water vapour partial pressure (Pa)

1.2.6 Vapour Pressure

Vapour pressure is an index of how much force is exerted by the water vapour in the atmosphere on a unit area. so, it is directly related to the amount of water vapour present in the atmosphere. The importance of the vapour pressure in the present study lies in being a good indicator of the potential of mass transfer in a vapour-air mixture. In other words, it gives an idea of how much vapour will condense if exposed to a cool surface at a temperature below the dew point temperature of the mixture. So, the higher the atmospheric partial vapour pressure, the greater the amount of condensate. According to ASHRAE, the saturation vapour pressure over liquid water for the temperature range of 0°C to 200°C is given by:

$$\log_e (P_{ws}) = \frac{C_8}{T} + C_9 + C_{10} \cdot T + C_{11} \cdot T^2 + C_{12} \cdot T^3 + C_{13} \cdot \ln(T) \quad (1.4)$$

where

P_{ws} = saturation pressure (Pa)

T = absolute temperature, K

C_8 = -5800.2206

C_9 = 1.3914993

C_{10} = -.048640239

C_{11} = $0.41764768 \cdot 10^{-4}$

C_{12} = $-0.14452093 \cdot 10^{-7}$

C_{13} = 6.5459673

The partial vapour pressure of a given air-water vapour mixture under saturation can be found as following:

$$P_w = \phi \cdot P_{ws}$$

where

ϕ = relative humidity (Equation 1.1a)

1.2.7 Sensible and Latent Heat

Sensible heat is that form of heat most readily perceived by the human senses. It has a direct effect on the state of comfort, therefore, the human body will respond quickly to any rapid change in the sensible heat. Most of heat gain and loss in buildings is in the form of sensible heat, therefore, direct substitution or extraction of sensible heat has to be made to maintain comfort. Sensible heat gain and loss from buildings can occur by conduction, convection and radiation. Finally, the amount of sensible heat available in a given

air-water vapour mixture can be measured by the dry bulb temperature which is directly related to the amount of sensible heat in the mixture.

Latent heat in the atmospheric air, is merely water vapour and humidity. This form of heat is always present in the atmosphere, but it is not sensed by the human body. However, it has a direct effect on human comfort, because the presence of humidity is essential in regulating the heat flow from and into the body together with the sensible heat. Latent heat can be lost from a building by exfiltration of indoor moist air or by condensation, and it can be gained by infiltration of outdoor humid air or due to moisture generation within the space. Latent heat is very important in this study, since water condensation and the associated heat release are the main issues.

1.3 RELATIONSHIP OF AIR PHYSICAL QUANTITIES

Air-water vapour mixture up to about three atmospheres pressure obeys the perfect gas law with sufficient accuracy to perform engineering calculation [7]. Therefore, the relationship between the physical quantities describing the mixture such as relative humidity, dry bulb temperature, wet bulb temperature, dew point temperature, etc. can be determined. To facilitate engineering computation, a graphical representation of the relationship between the properties of moist air has been developed and is known as psychrometric chart which is shown in Fig. 1-1. By using the psychrometric chart all physical

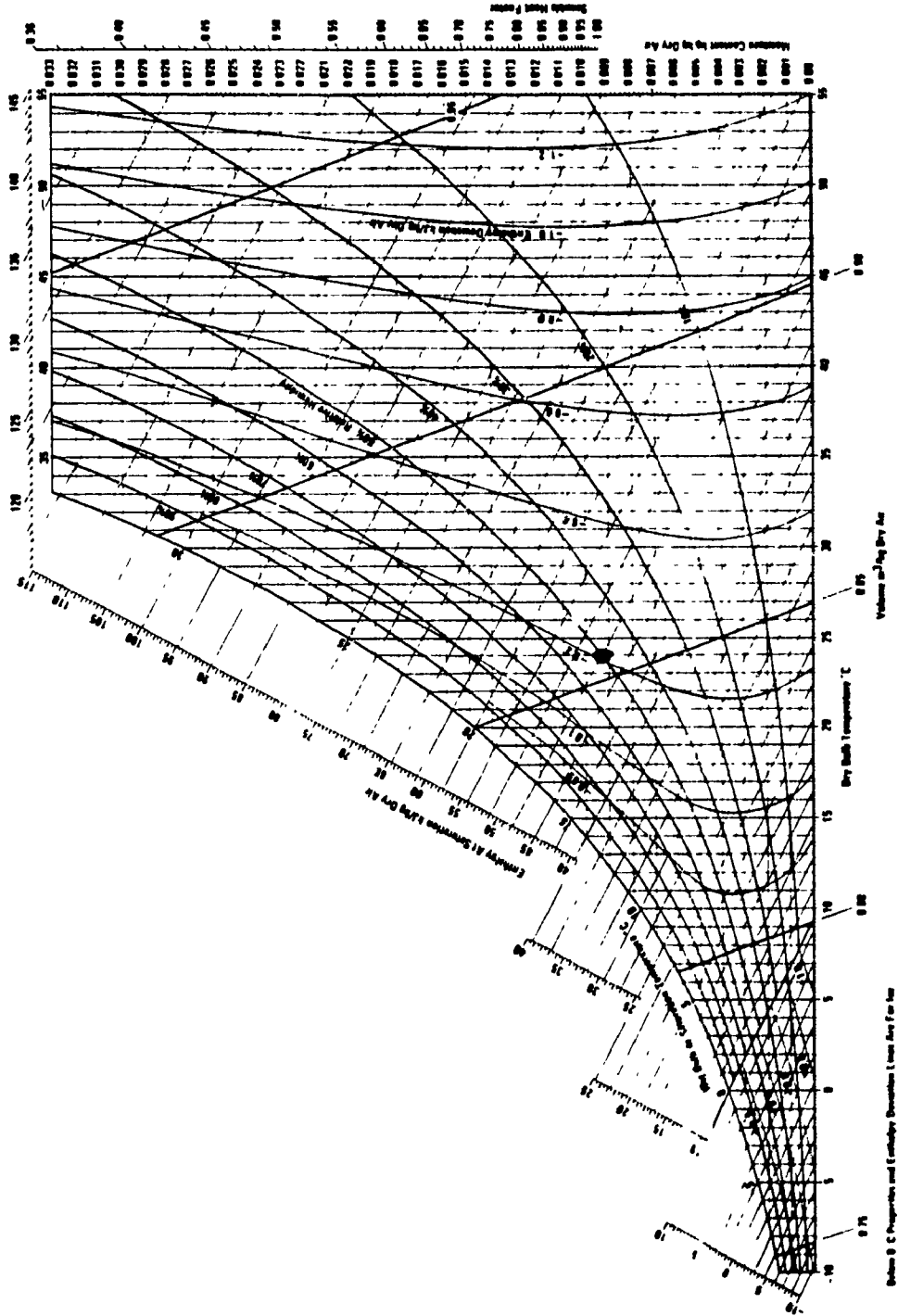


FIG. 1-1: Psychrometric chart (carrier)

properties of a given mixture can be determined if at least two of them are known.

1.4 IMPORTANCE OF THE TOPIC

Condensation in buildings mainly occurs on cold elements within the exterior envelopes. Windows, due to their thermal properties regarding conductivity, emissivity and storage capacity, tend to be the coldest surfaces in a building envelope. Windows in general and single glazed in particular have the greatest potential to remove water from the air once the window surface temperature becomes less than the dew point temperature of the air-water vapour mixture in that particular side.

Moisture accumulation and dampness in buildings due to condensation can cause major problems. The problems associated with condensation on the building surfaces or within a wall construction have been extensively discussed in literature [8-15]. However, when it comes to heat gain or loss due to surface condensation very little effort has been made [16,17]. Heat gain and loss due to surface condensation in buildings may constitute a sizable percentage of the total heat gain or loss, particularly when large glazed area, high relative humidity and large temperature difference exist. Thus, when ignored, it may lead to inaccurate estimate of heating and cooling requirements of buildings.

1.4.1 Heat Gain due to Surface Condensation

Water has the highest latent heat of vaporization compared to refrigerants used in refrigeration industry. Therefore, when outdoor surface condensation occurs on a glazed window, considerable amount of latent heat will be released, and good percentage of it will be part of the total heat gain. The transmission of a good part of the heat released on the external surface is due to two facts. First, when condensation occurs all the heat released on the external surface of the water film is transferred to the window surface by conduction through the water film. Second, the thermal resistance of the glass is relatively very low which allows much of the heat to be transferred indoor.

The significance of heat gain due to outdoor surface condensation is determined by two factors; first, the type of climate and second, the percentage and type of the glazed area. Outdoor surface condensation is very common and frequent on windows of air-conditioned buildings in hot-humid climates, when the outdoor air temperature and relative humidity are sufficiently high for condensation to occur. The occurrence of outdoor surface condensation is also dependent on the thermal resistance of the window. Single glazed windows have the highest potential for outdoor surface condensation to occur. However, it is unlikely to occur on insulated glazed windows because of their higher thermal resistance. The significance of heat gain due to outdoor surface condensation is only justified when a relatively high

percentage of exterior walls is singly glazed.

1.4.2 Heat Loss due to Surface Condensation

Indoor surface condensation on windows or walls is associated with moisture loss from interior spaces. Substituting moisture into the space requires energy, thus heat is indirectly lost in the form of moisture when indoor condensation occurs. Indoor surface condensation is very common in cold climate, especially when the indoor relative humidity is kept reasonably high. In fact, even at comfortable relative humidity level, indoor condensation can still occur on single glazed windows. However, for indoor condensation to occur on double glazed windows the ambient temperature has to be low enough while the indoor relative humidity has to be high enough for surface condensation to occur. Indoor humidity level in some buildings is maintained relatively high for functional reasons. Now, humidification is being introduced into hospitals, schools, libraries, laboratories, museums and even into apartment and office buildings. This trend toward increased winter humidification is likely to continue and a growing demand for increased humidification in existing building can be expected [18]. Although some humidified buildings are not maintained at the intended humidity level during the cold weather period, there are an increasing number of buildings, such as hospital, libraries and museums, in which the maintenance of humidity level is highly desirable at all times and is considered as one of the required functions of the building [18].

Indoor surface condensation, like outdoor surface condensation, is highly affected by the overall thermal resistance of the window. For example, to avoid surface condensation on a single glazed window, the relative humidity should be maintained at about 12% when the outdoor temperature is about -18°C , and the indoor temperature is about 21°C (assuming wind speed of 6.7 m/s). However, to avoid surface condensation on a double glazed window, the relative humidity can be as high as 41% under the same conditions, as shown in Table (2.1). Although, double glazed windows are commonly used to reduce the risk of condensation, however, single glazed windows are often selected to reduce capital costs [19]. Consequently, the risk of indoor surface condensation is increased.

Now, it can be seen that the significance of heat loss due to indoor surface condensation is determined by the indoor and outdoor conditions, as well as the thermal characteristics of the window. Therefore, heat loss by indoor surface condensation has to be considered when indoor and outdoor conditions needed for indoor condensation prevail, particularly, if single glazed windows constitute a large portion of the exterior envelope.

1.5 PREVIOUS RELATED WORK: A REVIEW

Condensation in buildings has been widely investigated in literature [8,9,10,12,19]. Corresponding problems related to building material and environmental quality, and proposed solutions have also

been widely examined [9,11,12,20,21]. However, heat gain and loss due to condensation on glazed windows have been ignored or given very little attention as part of condensation rate prediction models [16]. The main emphasis in literatures has been devoted to condensation on interior surfaces and condensation within wall constructions and roof spaces.

Condensation is known to be a very common cause for dampness in buildings. Recently, there has been a considerable increase in the moisture production in buildings because of changed methods of heating. In addition, the ventilation level which is required to remove moist air has been reduced as a result of developing more air-tight buildings. Therefore, the risk of condensation in modern buildings has been substantially increased [9]. Condensation in buildings can cause serious damage to building materials and surface finish and decoration. Insulating material, wood frames, electrical services are some examples of building materials that can be damaged by condensation. In addition, mould growth is a typical problem in buildings associated with condensation.

Heat gain and loss due to surface condensation in buildings have not been considered separately in any of the literature that have been found. However, heat production due to condensation was considered by Davies, M. [16] as part of a model for predicting condensation rate, and was found to be, in that particular situation, a sizable fraction

of the loss due to conduction through the structural element [16]. Davies, M. described a procedure to estimate the rate of superficial condensation on glass windows. This procedure is based on the following assumptions:

1. Steady state conditions.
2. The thermal resistance and the thickness of the glass are negligible.
3. The temperature of the water film is equal to the glass temperature.
4. The temperature of the glass pane is uniform across its thickness.

Calculation of the rate of condensation is based on both the heat balance and the mass transfer equations. Then by introducing the latent heat term into the heat balance equation, and solving the resulting equation with the mass transfer equation, a relation between the glass temperature (T_w) and the difference between saturation pressure at the window temperature (P_c) and the bulk partial vapour pressure (P_a) can be obtained. Using this relation and the relation between saturated vapour pressure and the window temperature, two curves can be obtained. The intersection point between these curves is then used to estimate condensation rate. And by multiplying the condensation rate by the latent heat of vaporization of water, heat production can be found.

Heat loss due to condensation can be directly evaluated by the above model, since it is equal to the amount of heat released by condensation. However, this model is based on steady state heat flow, and on the assumption that the interfacial temperature of the resulting water film is equal to the glass temperature. The more realistic conditions involve consideration of time varying heat flow as condensation starts to occur, and more accurate evaluation of the temperature of the water film surface on which condensation occurs.

Detailed analysis of filmwise condensation of a saturated pure vapour has been suggested by Nusselt as discussed in Chapter 3. However, this analysis is not directly of great help in this study, since a noncondensable gas, which is air in the case, dominates the atmospheric mixture. Consequently, condensation and the resulting heat transfer calculation will be completely different. Fortunately, several articles have been produced dealing with condensation heat transfer in the presence of noncondensable gases [24-30]. However, the solutions were based on the presence of very small amount of a noncondensable gas, which is not the case in the atmospheric mixture, where more than 90% of the mixture is air.

Heat gain due to external surface condensation cannot be directly evaluated by the procedure suggested by Davies [16], because it does not consider heat transfer through the glass by assuming that the glass temperature is uniformly distributed across its thickness and

the thermal resistance of the glass is negligible (assumption 2,4). Heat gain due to condensation cannot be directly evaluated from the condensation rate, because part of the released heat will be conducted to the inner surface of the window and becomes part of the heat gain, and another part will raise the window temperature. Therefore, to calculate heat gain through glass windows due to condensation, the thermal resistance and the thickness of the glass have to be considered to be able to estimate the rise of the inner surface temperature. Then, heat gain due to outdoor surface condensation can be found by multiplying this temperature rise by the internal surface heat transfer coefficient.

1.6 OBJECTIVES OF THE RESEARCH

The importance of heat gain and loss due to condensation on glass windows under certain conditions related to the outdoor climate and the indoor environment can be recognized at this point. However, this subject has been ignored or given little attention when dealing with surface condensation in buildings [16]. Therefore, the objective of this study is to develop a reasonably accurate procedure to predict condensation rate and the corresponding heat gain and loss under any outdoor and indoor conditions. Although, the rate of condensation has to be evaluated in order to predict heat gain or loss, however, the relation between condensation rate and heat gain is completely different from how heat loss is related to the condensation rate. Because,

the outdoor released heat has to be transferred by conduction and convection before it becomes part of the heat gain, while, heat loss is directly calculated from the mass condensation rate, since moisture loss is the main concern.

In this study two procedures will be developed; one to predict heat gain, and the other to predict heat loss, and for each procedure the mass condensation rate will be determined by two methods, which represent two different levels of analyzing the problem, so that their relative accuracy can be assessed when compared with the experimental results. The idea of first method is suggested by Davies [16], and it is simply based on mass transfer due to vapour pressure difference between the bulk and the saturation vapour pressure at the surface. The second method is based on evaluating the water film interfacial temperature by which mass condensation rate and heat transfer rate can be determined. Once the heat transfer rate or the mass condensation rate is known, the rise in window surface temperature and the rate of heat gain and loss can then be determined.

In order to determine the applicability of each method in estimating the mass condensation rate and the corresponding heat transfer, a preliminary experiment will be conducted to determine experimentally how much vapour condenses, and the corresponding rise in window surface temperature. Experimental results will then be compared with the theoretical results obtained by the two methods mentioned above to see which one is a better match with the experimental results.

CHAPTER II

CONDENSATION IN BUILDINGS

2.1 INTRODUCTION

Water vapour condensation in building can take place on external and internal surfaces, and within the building envelope. When condensation occurs on buildings external and internal surfaces, it is known as surface condensation. When it takes place within the building envelope, it is called interstitial condensation. External surface condensation is likely to occur on low thermal resistance surfaces, such as glass windows of air-conditioned buildings in hot humid climates, and it is unlikely to occur on well insulated exterior panels. This is because the temperature difference across insulated walls is not enough to cool the exterior surface below the dew point temperature of the outdoor air.

In cold climates, where high temperature gradient exists across walls and windows of heated buildings, the internal surfaces temperatures can easily be below the dew point temperature of the indoor air hence, condensation occurs. High indoor relative humidity generated either mechanically or as a result of moisture generation by indoor activities increases the risk of indoor surface condensation. Condensation also occurs within the thickness of a wall or ceiling. The phenomenon occurs in most forms of structure at some time or

another [9]. Water vapour penetrates from the warm-humid side of a wall towards the cool and less humid side. When the penetrating humid air reaches a layer within the wall which has a temperature less than its own dewpoint temperature, condensation occurs.

The effect of condensation on building materials and environmental quality is a very important subject which is not to be neglected when dealing with condensation in buildings. Although this study is intended to investigate heat gain and loss due to condensation, the process of condensation as a means for generating moisture in buildings will be covered in this chapter, due to its extreme importance as well as being a fundamental background to heat gain and loss due to condensation.

2.2 RELATION BETWEEN CONDENSATION AND CLIMATIC CONDITIONS

The occurrence of outdoor and indoor condensation on a window surface is highly dependent upon the air temperature and the relative humidity. High outdoor relative humidity must exist in order to encounter considerable condensation on the outside surfaces. Low outdoor temperature is important for condensation to occur on the internal surfaces. The degree of seriousness of condensation in buildings can not be evaluated only by the presence of condensation, but also by its frequent occurrence and its duration. Therefore, the severity of climatic conditions has to be known in order to evaluate the significance of condensation and its associated heat gain and loss in buildings.

High outdoor relative humidity and relatively high ambient air temperature are the type of climatic conditions needed to produce considerable outdoor surface condensation. High relative humidity and temperature at the same time mean more water vapour in the air, consequently higher condensation rate is expected. High relative humidity and temperature also mean higher dew point temperature, which in itself is an indication for higher condensation potential. On the other hand, higher ambient temperature increases external surface temperature, which in turns decreases the condensation potential. Indoor air temperature, which is also an important factor in determining condensation potential, acts only in one way. The lower the indoor temperature is, the greater the potential of condensation will be.

Indoor condensation is also highly dependent on the climatic conditions. In cold climate, internal surface condensation is a common phenomenon, especially on surfaces having low thermal resistances. When the outdoor temperature is low enough to maintain the indoor surface temperature below the dew point temperature of the indoor air, internal surface condensation occurs. The lower the outdoor temperature the greater the potential for indoor condensation, and vice versa. In other words, the ambient temperature acts in a one-way action in determining the indoor surface condensation potential. However, indoor temperature acts in two-way action, where higher indoor temperature means more moisture content and higher dew point

temperature hence, higher condensation potential. On the other hand, higher indoor temperature leads to higher internal surface temperature, consequently lower condensation potential. Indoor temperature has a narrow range of variation compared with the outdoor temperature. Accordingly, moisture content does not vary considerably within this range of temperature variation. Therefore, the two-way action for the indoor air temperature has less effect on the potential for internal surface condensation than the outdoor temperature effect on external surface condensation.

2.3 CONDENSATION ON THE BUILDING EXTERNAL SURFACES

Condensation on an external surface occurs whenever its temperature is below the dew point temperature of the outdoor air. This type of condensation is a common phenomenon in hot-humid climates, where air moisture content is significantly high. However, the occurrence of external surface condensation is not generated by the presence of the hot-humid climate alone. The occurrence of condensation is also dependent upon other factors related to window thermo-physical properties, and the indoor temperature. Therefore, simultaneous consideration of climatic conditions, thermo-physical properties of the window, and indoor temperature is essential in order to evaluate the risk of condensation on external surfaces.

External surface condensation occurs on exterior panels having low thermal resistance, because the indoor-outdoor temperature difference is not large enough to keep the external surface temperature of a well insulated wall below the saturation temperature of the ambient air. External surface condensation on a well insulated wall occurs when the atmospheric air is at or near saturation, which is a frequent condition in hot-humid climates. The dew point temperature of the ambient air at saturation is equal to its dry bulb temperature, which is normally higher than the wall external surface temperature in the absence of solar radiation. So, external surface condensation occurs regardless of the temperature difference across the exterior panels, as long as the air is near or at saturation. This type of condensation could result in discoloration and staining of the wall surface and may cause damage to building metal and organic material. When it occurs on window surfaces, it obstructs the view and it could be an added heat gain component as well.

2.4 INTERSTITIAL CONDENSATION

Interstitial condensation is more serious than surface condensation, because it is invisible and cannot be realized until a great damage has been done. Careful attention should be given to external construction in conjunction with the climatic conditions, so that the location within the wall at which condensation occurs can be predicted and the precautions needed to reduce the risk of condensation can be identified.

Most building materials are porous and offer little resistance to the passage of water vapour. Water vapour, under partial vapour pressure difference always tend to travel towards a less humid and colder medium. In winter, moisture content of the indoor air is higher than the outdoor air moisture content. So, water vapour is expected to diffuse through the wall towards the outside. The moist air within the wall has a defined dew point temperature depending on the mass of water vapour in the air. The mass of the water vapour of the air in the pores, and accordingly, the dew point temperature of the moist air decline as the water vapour progresses towards the outside. The temperature of the wall also declines from an inner surface temperature near the room temperature to an outer surface temperature near the outside temperature. So, the actual temperature and the dew point temperature are declining through the wall. Normally, with ordinary porous building materials such as brick, the decline in actual temperature is more rapid than the decline in dew-point temperature, so that they cross somewhere within the wall and at this point condensation occurs [8].

Interstitial condensation occurs within a wall cavity, because the still air in the cavity presents an effective resistance to movement of heat but permits easy movement of water vapour. This type of condensation is more frequent and more serious in a wall cavity filled with insulating material. Since there is a sharp drop in the temperature within the insulating material, dew point temperature may be

reached in the filling causing condensation to occur in the insulating filling and cause many problems [8]. In addition, interstitial condensation can occur near the outer face of the wall. In this case the water can easily evaporate to the outside and in warmer weather it normally escapes harmlessly.

2.5 CONDENSATION ON THE BUILDING INTERNAL SURFACES

Internal surface condensation in winter is a common problem in most buildings. In recent years, the need for energy-efficient buildings has resulted in air-tight building envelopes. Air-tight buildings, changing living habits and new methods of heating have resulted in high indoor moisture content. Consequently, the risk of indoor condensation has increased.

Internal surface condensation occurs on cold exterior walls, cold bridges and windows. Window construction often represents the poorest component of the building enclosure in a thermal sense, even when double windows are used, and hence has the lowest inside surface temperature. The window, therefore, determines the practical limit of humidity for the space in winter, and condensation may appear on the glass, frame, or sash depending on the relative thermal characteristics of these three components [19]. Condensation on finished walls surfaces may cause damage to painting and decoration. In addition, indoor condensation can also be seen in roof spaces, which may

ultimately cause a great damage to insulation, and other roofing materials.

The increase in moisture content inside buildings has increased the risk of indoor condensation. Most modern buildings were built so tight to prevent air leakage, consequently less heat loss was experienced. On the other hand, moisture produced inside has less chance to escape to the outside. At the same time, more water vapour is being produced inside buildings due to changing habits, new activities and heating methods. As a result, a significant amount of water vapour is trapped indoor, hence, the risk of indoor condensation is increased.

Indoor condensation affects both the building materials and the environmental quality of the space. In addition to building material deterioration, mould growth is a well known problem associated with the presence of condensation. Moulds are usually most severe in room corners of external uninsulated walls. This is mainly attributed to insufficient ventilation which creates pockets of stagnant air in such corners [8].

2.6 PROBLEMS ASSOCIATED WITH CONDENSATION IN BUILDINGS

Problems associated with condensation range from merely inconvenience to severe damage of building materials. Condensation on internal surfaces is a source of many problems in buildings, where it can

damange decorative finishes and wood and metal sashes. Recently, condensation has become a major problem causing about two third of the complaints of dampness in houses [8]. Wood decay is one of the most serious consequences of dampness. In severe cases the amount of water deposited may be very great causing actual pools of water on the floor, saturated clothes in wall cupboard and decay of window and door joinery. Mould growth is another consequence of condensation, which can cause serious problems related to the environmental quality.

Interstitial condensation is not considered a serious problem as long as it occurs away from the filling of the wall cavity. Condensed water in small amounts escapes freely if the cavity is ventilated or may evaporate at favourable times through the outer leaf of the wall. Interstitial condensation is more frequent and more troublesome, when the cavity is filled with insulating material. Condensed water absorbed by the cavity filling has less opportunity for evaporation in dryer weather. As a result, the filling material will be damaged, mould growth is encouraged, and the filling could be a source of water which can penetrate to the internal surfaces. Condensation within attic spaces is a common phenomenon in modern buildings. Since, the roof space may be the only place to which water vapour can escape. Water vapour will largely pass from the ground floor to upper floors and then through the ceiling to the roof space. Condensation within roof spaces can damage roofing and insulating materials and electrical services on the ceiling. Problems of

condensation in roof spaces may be especially severe during a thaw after large amounts of ice have built up on the sheeting as a result of condensation during a prolonged cold spell. The volume of water released may be large enough to give the impression of a major leak in the roof.

Condensation on or near external surfaces is not considered a problem in warm weather, since condensed water will evaporate to the outdoor harmlessly [8]. However, if deposited water interacts with chemical particels of building materials and air pollutants, harmful solvents will be formed causing discoloration and staining. Finally, condensation in buildings can now be seen as a source of trouble, damaging building materials and affecting the quality of the indoor environment. Therefore, it is important to find ways to prevent or at least reduce the risk of condensation.

2.7 CONTROL OF CONDENSATION IN BUILDINGS

The importance of preventing or at least reducing the risk of surface and interstitial condensation should be clear at this point. Building materials; walls, floors, and roofs construction; heating methods and functional activities in buildings should be carefully evaluated in order to take the proper action to reduce the risk of condensation. The risk of condensation in buildings can be substantially reduced by one or more of the following measures:

1. Water Vapour Control

Water vapour in buildings are mainly produced by occupants and the various activities within space. Cooking, bathing, dish wahsing, cloth washing and drying are typical mositure sources in dwellings. In addition, some heating methods such as gas heater, which discharge their combustion gases directly into the air inside the building, can be a major source of water vapour. Since, one liter of kerosene produces about one liter of water. Some of these water vapour sources are unavoidable, but others can be eliminated or reduced. It is helpful to know the relative importance of each source so that proper action can be taken to reduce or eliminate its contribution. For example, if the kitchen is identified to be a major source of water vapour, action like keeping the kitchen door closed or installing extractor fan may be recommended.

2. Efficient Ventilation

The increasing trend towards constructing more energy efficient buildings has led to little natural ventilation due to closed or even sealed windows and ventilation grates. As a result, in unventilated areas pockets of stagnant air occur. These areas are a potential source of localised condensation. Therefore, adequate ventilation has to be provided to the space to reduce the moisture content of indoor air by diluting it with drier air from

outside. There is no unique value of ventilation rate required to control condensation. The ventilation requirement depends on the temperature and humidity of the ventilating air, the heat input and insulation, the pattern of heat input and the nature of the internal surfaces [9].

Concealed condensation within wall cavities and roof spaces can also be prevented by ventilation. When cross-ventilation can be achieved, this is usually the cheapest and simplest method [9]. However, air passages in walls, designed to remove an unrestricted vapour supply, are unduly large and may waste considerable heat. Ventilation is most effective when each structural space has a clearly defined air passage with an inlet and outlet. In walls, a small convective effect may be utilized by locating one vent at the bottom and one at the top of each space [23].

3. Reducing the Area of Cold Surfaces

In winter, single glazed windows are usually the coldest part in buildings. They have the potential to remove a lot of moisture from the air. So, by reducing their area, the amount of condensation is proportionally reduced. Condensation is also a common phenomenon on cold areas produced by cold bridges. Cold areas are produced on relatively warm walls whenever a highly conduction element is used so that areas on internal surfaces are directly linked to the outside cold surfaces. Such cold areas may

be found in a number of places in buildings; for instance the bridging of a cavity with a dense floor slab which may result in condensation on the surface of the slab and possibly on the walls within the room.

4. Using Thermal Insulation

The risk of internal surface condensation can be reduced by raising the surface temperature so that it is higher than the dew point temperature of the indoor air. When insulating windows and walls are used, their internal surfaces are maintained at a temperature much above the external surface temperature, and in the vast majority of cases above the dewpoint of the internal air [9]. Moreover, an insulated structure will ensure for a given heat input that the air temperature will be maintained at a higher level than with a similar but uninsulated structure.

The effectiveness of thermal insulation in preventing condensation depends not only on its value but also in positioning the insulation barrier correctly in the structure. Insulation could be positioned either on the external surface or on the internal. In either case the overall thermal resistance would be the same, but the environment inside the building could be very greatly affected by the positioning of the insulation. Although

variations in heating cycles, heating methods, solar heat gain, etc., all affect this particular consideration, it is generally agreed that the insulation should be positioned as near as possible to the internal surface of the structure. This will result in a higher surface temperature than that which would be obtained with the same heat input and the same insulation positioned elsewhere. In all cases, insulation will reduce the tendency for condensation to occur and in many will completely prevent it.

5. Heating

Continuous and adequate heat levels can be the most effective single means of preventing condensation. Heating warms up the air so that it could absorb moisture and hold it as water vapour. In addition, it warms wall surfaces and helps to keep them above dew point. Wall temperature can also be raised by directly forcing hot air over the wall inner surface. For this method to be effective, low thermal capacity lining, like insulation, should be used on inner surfaces. If high thermal capacity materials are used with intermittent heating, condensation will occur as the material will not warm up quickly enough to prevent it.

6. Using Vapour Barriers (Retarders)

Thermal insulation, although keeping room surfaces warm,

may increase the risks and effects of interstitial condensation, by making the structure cooler. The only way to prevent this is to provide a vapour barrier to prevent the water vapour penetrating the wall. Most insulation materials are porous, and would allow warm moisture-laden air to penetrate to the cold structure where interstitial condensation can occur. Insulation will lose effectiveness if allowed to become saturated and to avoid this danger, it is advised to locate a vapour barrier on or as near to the inner surface as possible to prevent water vapour entering the structural material in any way. Vapour barrier is probably not worth using for most brick or masonry walls. However, it is usually considered essential for wood frame construction because condensation within the wall can cause decay of the wood [8]. If vapour barriers are used over all walls and roofs, vapour will have a much less chance to escape through them. Therefore, it is important to find a way out for the water vapour that is being produced in the space, otherwise condensation must inevitably occur on cold internal surfaces.

2.8 DEPENDENCE OF CONDENSATION ON THE RELATIVE HUMIDITY AND INDOOR AND OUTDOOR AIR TEMPERATURES

Condensation on exterior windows and walls' surfaces is very dependent on indoor and outdoor air temperatures, and outdoor relative humidity. Indoor condensation in winter depends on the indoor

relative humidity and indoor and outdoor air temperatures. To determine the surface condensation potential, the surface temperature must first be evaluated. Surface temperature of the condensation side can be evaluated when indoor and outdoor temperatures, as well as the thermo-physical properties of the window or the wall are known. The surface temperature is then compared with the dew point temperature of the air. If the surface temperature becomes less than the dew point temperature of the air on the corresponding side condensation will occur.

There are many different types of walls and windows with different thermal resistances. Therefore, for each type there is a certain indoor temperature and corresponding outdoor conditions at which condensation occurs. In the present study the dependence of surface condensation on indoor and outdoor conditions for single and double glazed windows will be discussed.

External surface condensation will only be considered for single glazed windows, since they are usually used in hot-humid climates where external surface condensation is possible. To evaluate the external surface temperature the following equation can be used.

$$T_{wo} = T_o - \frac{U}{h_o} (T_o - T_i) \quad (2.1)$$

where

T_{wo} = external surface temperature

U = overall heat transfer coefficient ($\text{W/m}^2\text{-C}$)

h_o = external surface heat transfer coefficient ($\text{W/m}^2\text{-C}$)

T_o = outdoor air temperature

T_i = indoor air temperature

Now, let $T_{wo} = T_{Do}$ = dew point temperature of outdoor air

$$T_{Do} = T_o - \frac{U}{h_o} (T_o - T_i) \quad (2.2)$$

arranging terms

$$T_{ci} = T_o \left(1 - \frac{h_o}{U}\right) + \frac{h_o}{U} T_{Do} \quad (2.3)$$

where

T_{ci} = indoor air temperature below which external surface condensation occurs.

The results of solving Eq. (2.3) at different indoor and outdoor conditions are shown graphically in Figs. 2-1 and 2-2. The external surface heat transfer coefficients in this case are evaluated using Eqs. (4.4) and (4.5) when there is no wind, and Eqs. (4.15a) to (4.15E) when the wind effect is considered. Fig. 2-1 shows indoor temperature and outdoor relative humidity combination at which outdoor condensation can be expected with different values of outdoor temperature for a single glazed window of 10 mm ($M = 10$ mm). For example, at an outdoor air temperature of 30°C , and a relative humidity of 80 percent the indoor air temperature below which external surface condensation occurs is about 22°C when there is no wind ($WV=0$ m/s). The

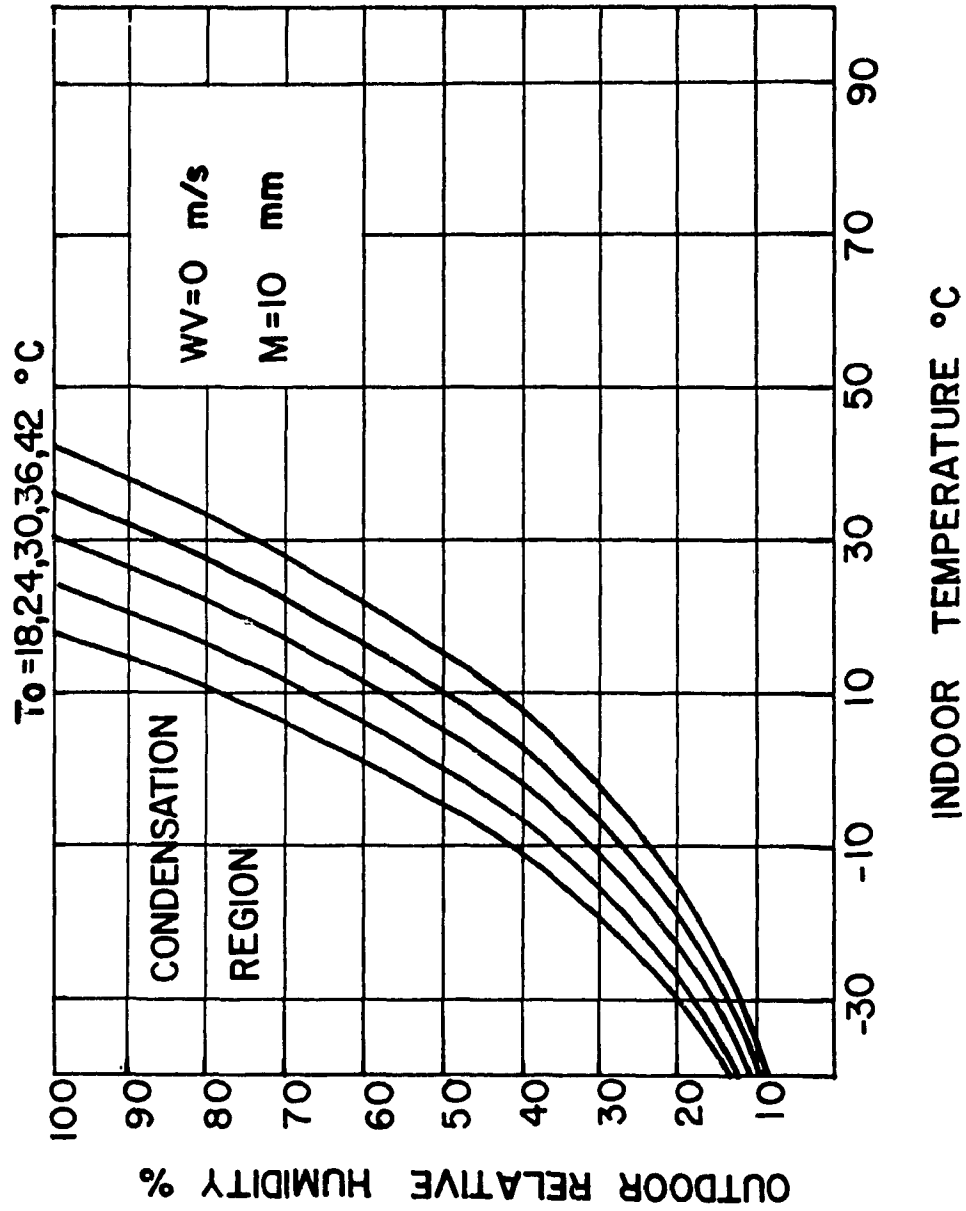


FIG. 2-1: Temperature and relative humidity combinations at which condensation on the external surface of a single glazed window is expected with a wind velocity of 0 m/s

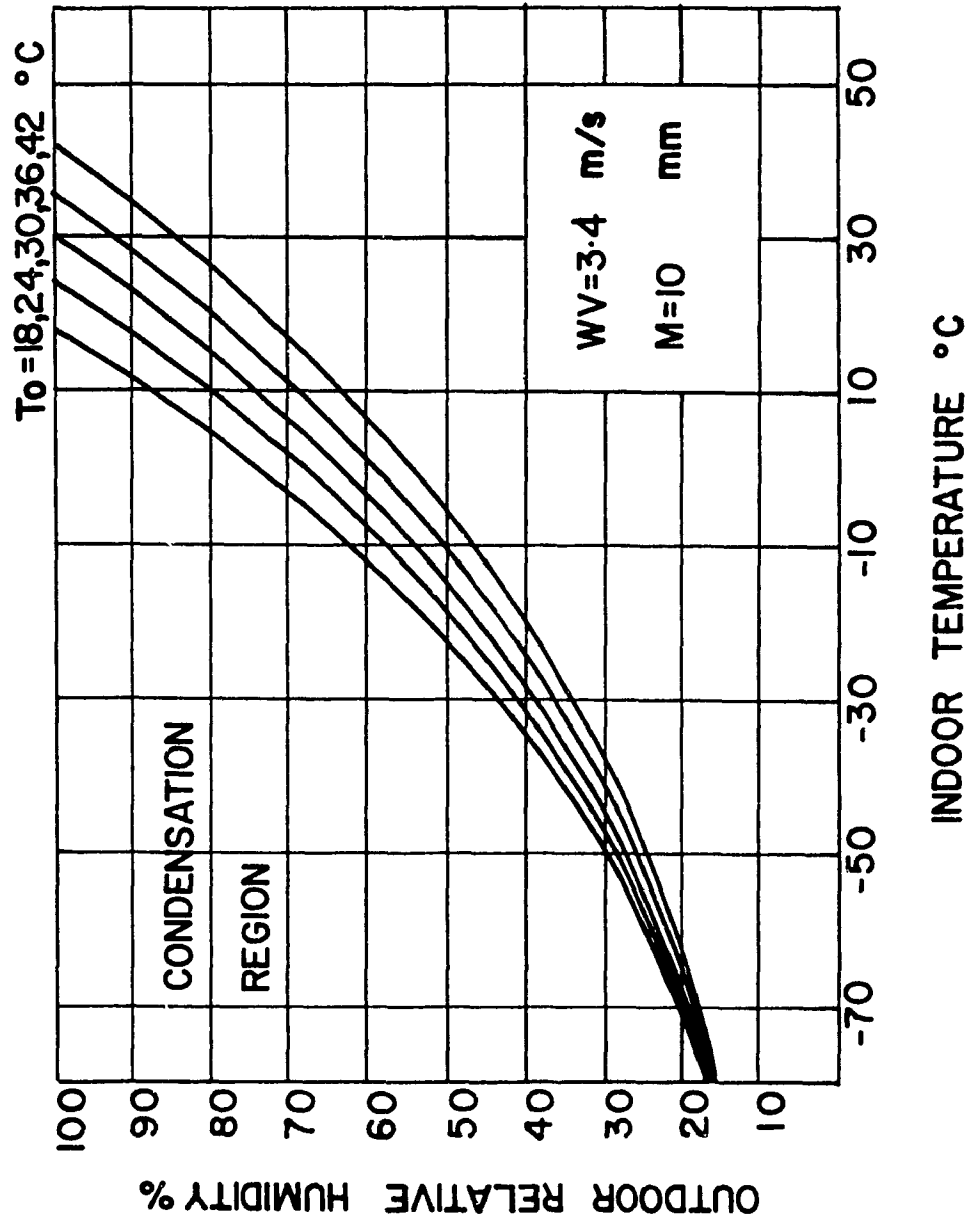


FIG. 2-2: Temperature and relative humidity combinations at which condensation on the external surface of a single glazed window is expected with a wind velocity of 3.4 m/s

effect of wind speed is shown in Fig. 2-2, where it is increased to 3.4 m/s. In this case, external surface condensation is more difficult to occur, since much lower indoor temperature is needed for condensation to occur at the same relative humidity. In addition, the curves have less slope resulting in larger condensation regions, but at lower indoor air temperatures. Each curve in both figures identifies two regions; the upper region in which condensation occurs, and the lower region in which no condensation takes place.

Internal surface condensation will be considered for both single and double glazed windows. Because both types are used in cold climates, where internal surface condensation is possible. The internal surface temperature can be evaluated by the following equation:

$$T_{wi} = T_i - \frac{U}{h_i} (T_i - T_o) \quad (2.4)$$

where

T_{wi} = internal surface temperature

h_i = internal surface heat transfer coefficient (W/m²-C)

If we let $T_{wi} = T_{Di}$ = dew point temperature of indoor air

$$T_{Di} = T_i - \frac{U}{h_i} (T_i - T_o) \quad (2.5)$$

arranging terms

$$T_{co} = T_i \left(1 - \frac{h_j}{U}\right) + \frac{h_i}{U} T_{Di} \quad (2.6)$$

where

T_{CO} = outdoor air temperature below which internal surface condensation occurs

The results of solving Eq. (2.6) at different indoor and outdoor conditions for single and double glazed windows are shown in Figs. 2-3 to 2-6. Figs. 2-3 and 2-4 show the combination of indoor and outdoor temperatures and the indoor relative humidity at which internal surface condensation occurs on single glazed windows. Each curve represents a constant indoor temperature and each group of curves is at a constant wind velocity. Wind velocity, which has been seen to have a negative effect on outdoor condensation, has a positive effect on indoor condensation. As the wind velocity increases the external surface heat transfer coefficient increases. As a result the internal surface temperature decreases which leads to more condensation. In other words, for the same indoor conditions, internal surface condensation occurs at higher indoor temperatures as the wind speed increases.

The indoor and outdoor conditions at which internal surface condensation occurs on double glazed windows can be obtained from Figs. 2-5 and 2-6. These figures were constructed for a specific double glazed window construction of 3 mm glass thickness ($M=3$ mm) and 6 mm air space ($AS=6$ mm). The outdoor temperature needed for internal surface condensation on a double glazed window is much lower than that needed for a single glazed window at the same indoor conditions. The

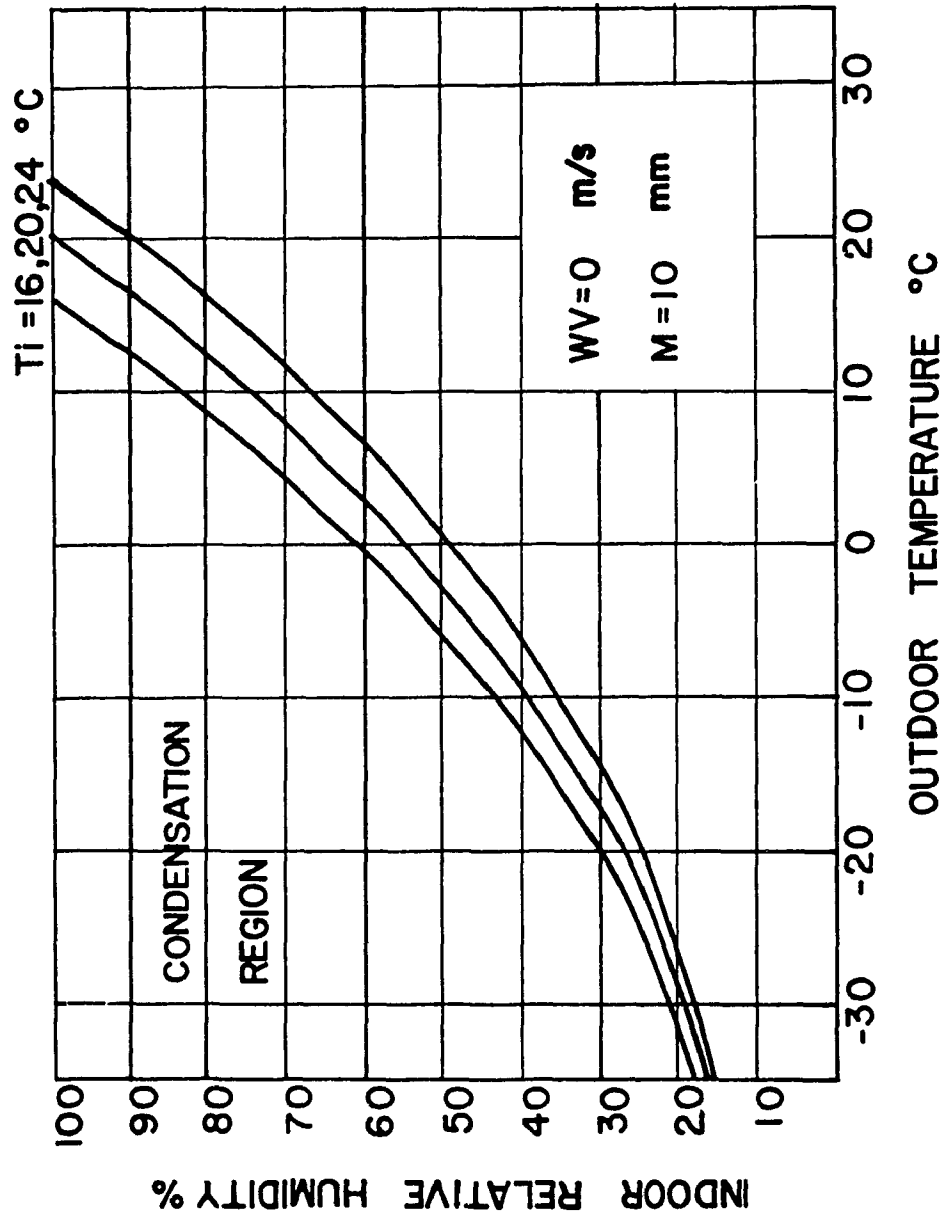


FIG. 2-3: Temperature and relative humidity combinations at which condensation on the internal surface of a single glazed window is expected with a wind velocity of 0 m/s

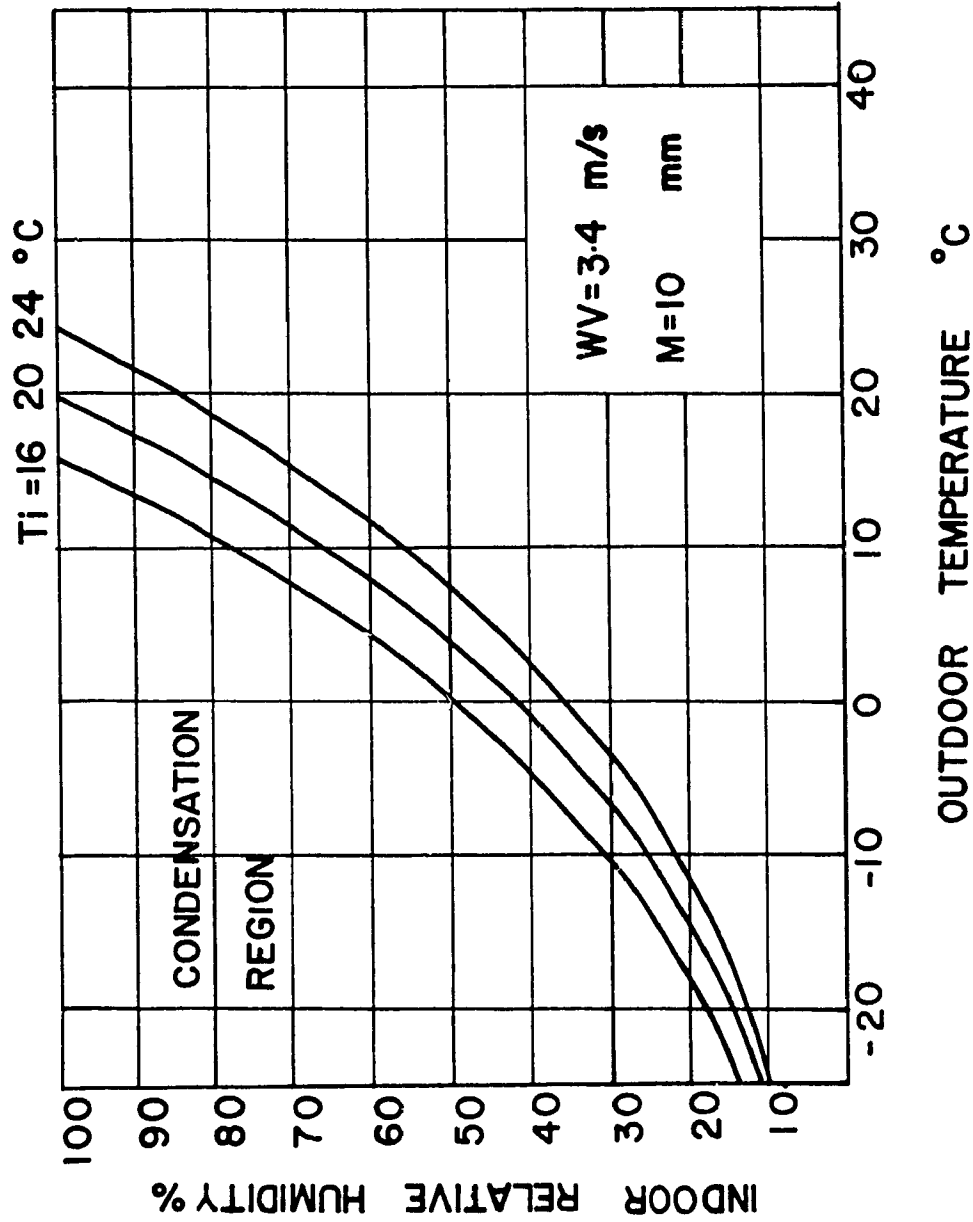


FIG. 2-4: Temperature and relative humidity combinations at which condensation on the internal surface of a single glazed window is expected with a wind velocity of 3.4 m/s

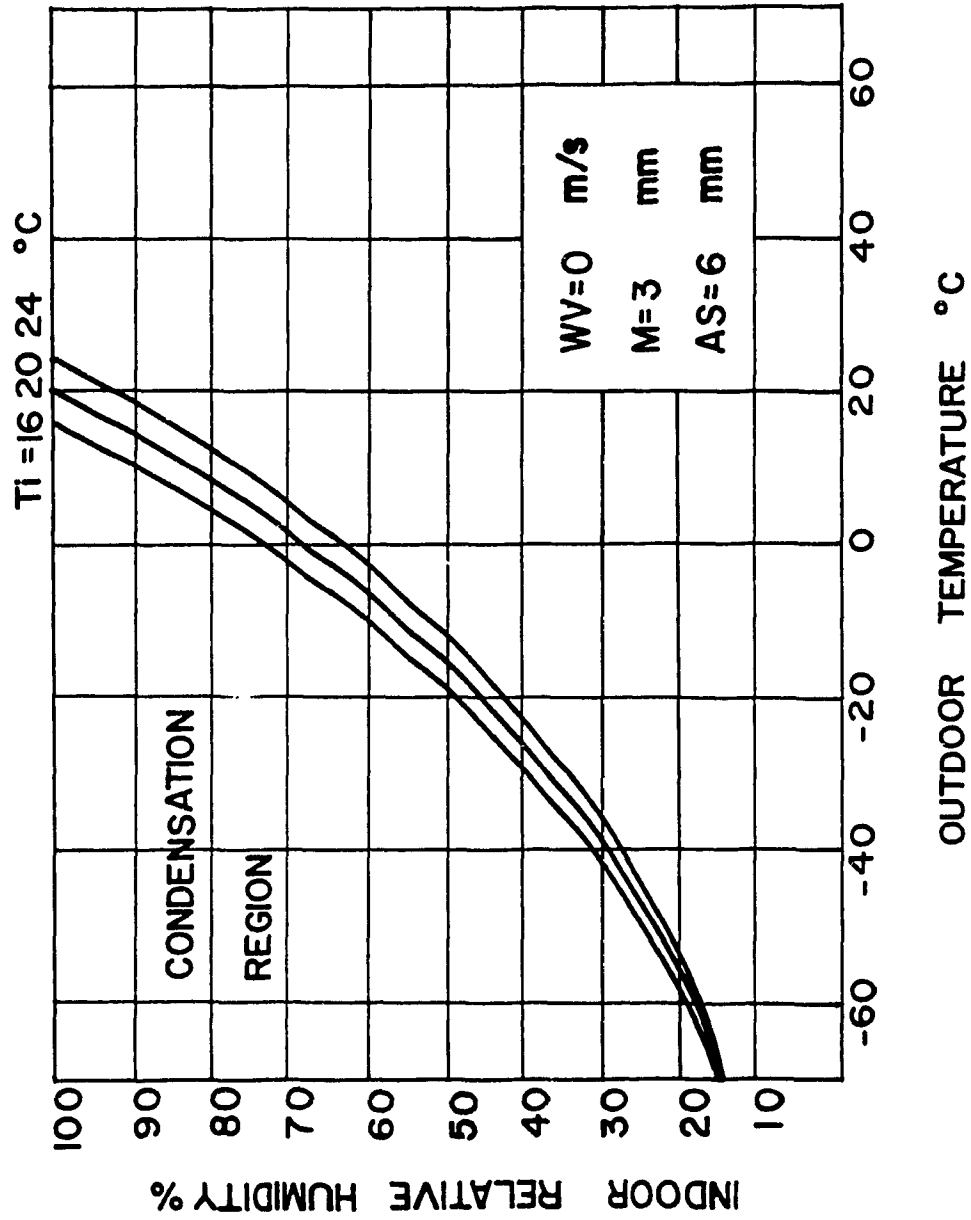


FIG. 2-5: Temperature and relative humidity combinations at which condensation on the internal surface of a double glazed window is expected with a wind velocity of 0 m/s

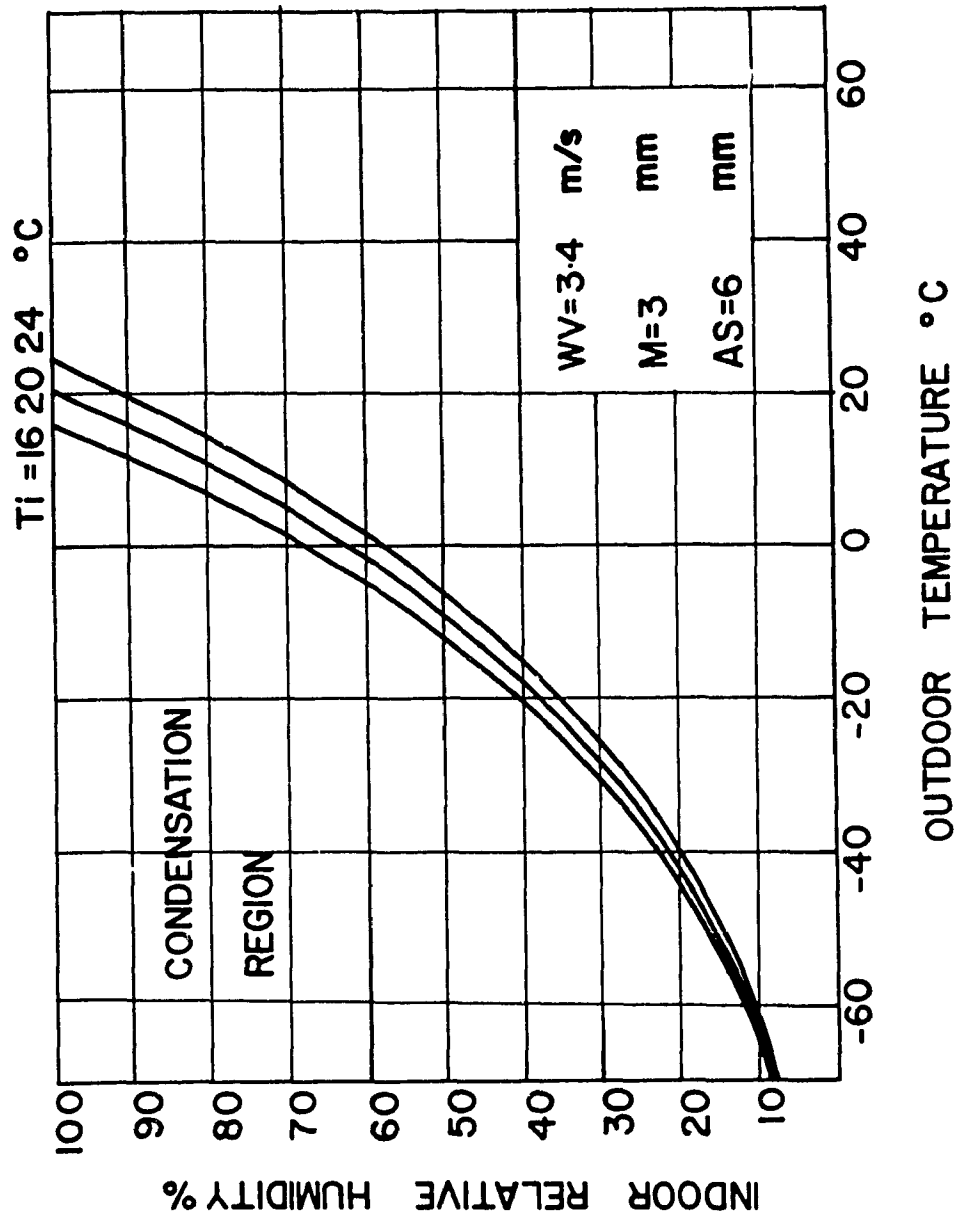


FIG. 2-6: Temperature and relative humidity combinations at which condensation on the internal surface of a double glazed window is expected with a wind velocity of 3.4 m/s

increase of the wind speed from zero to 3.4 m/s in this case has a negligible effect on internal surface condensation and the two groups of curves seem to be identical with about the same condensation regions.

Figs. 2-7, 2-8 and 2-9 compare the behaviour of double and single glazed windows at the same indoor and outdoor conditions. It can be seen that as the indoor relative humidity increases, the effectiveness of the double glazed window decreases as compared with the single glazed window. At saturation, double glazed windows have the same condensation risk as single glazed windows at the same conditions. When the indoor temperature is changed as in Figs. 2-7 and 2-8, there is no change in the slope of curves, however, lower outdoor temperature is needed for lower indoor temperature for internal surface condensation to occur. When the wind velocity is decreased from 3.4 m/s to zero m/s as shown in Figs. 2-8 and 2-9, the outdoor temperature has to be much lower for indoor condensation to occur. In addition, the shape of curves, especially for the single glazed window has changed. At this point it is important to note that as the indoor relative humidity gets higher, the risk of internal surface condensation on a double glazed window gets closer to the risk of condensation on a single glazed window at the same conditions.

The effect of doubling and tripling glass windows in the presence and the absence of wind on the occurrence of internal surface condensation is shown in Table (2.1). It can be seen that the presence of

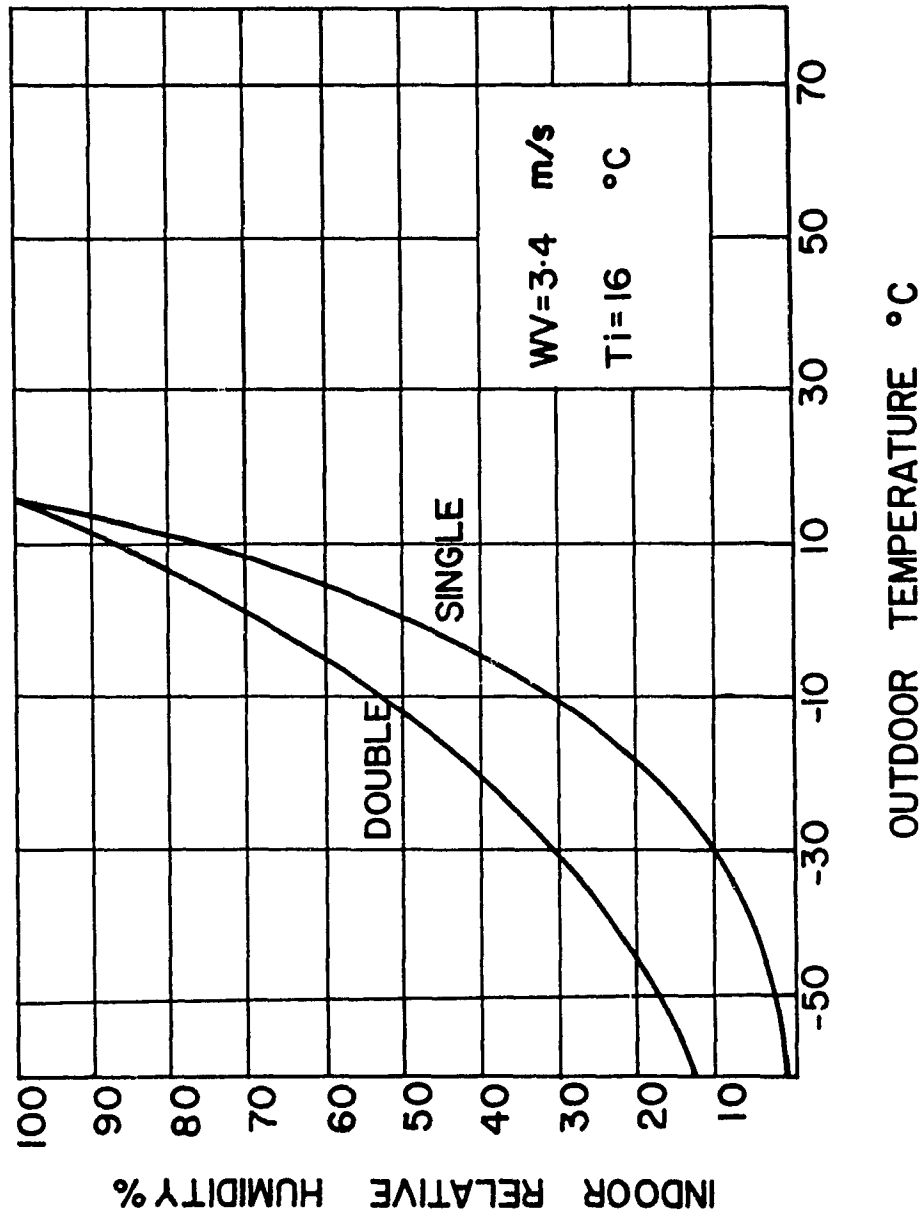


FIG. 2-7: Comparison between internal surface condensation potential of a single and a double glazed windows with an indoor temperature of 16°C; and 3.4 m/s wind velocity

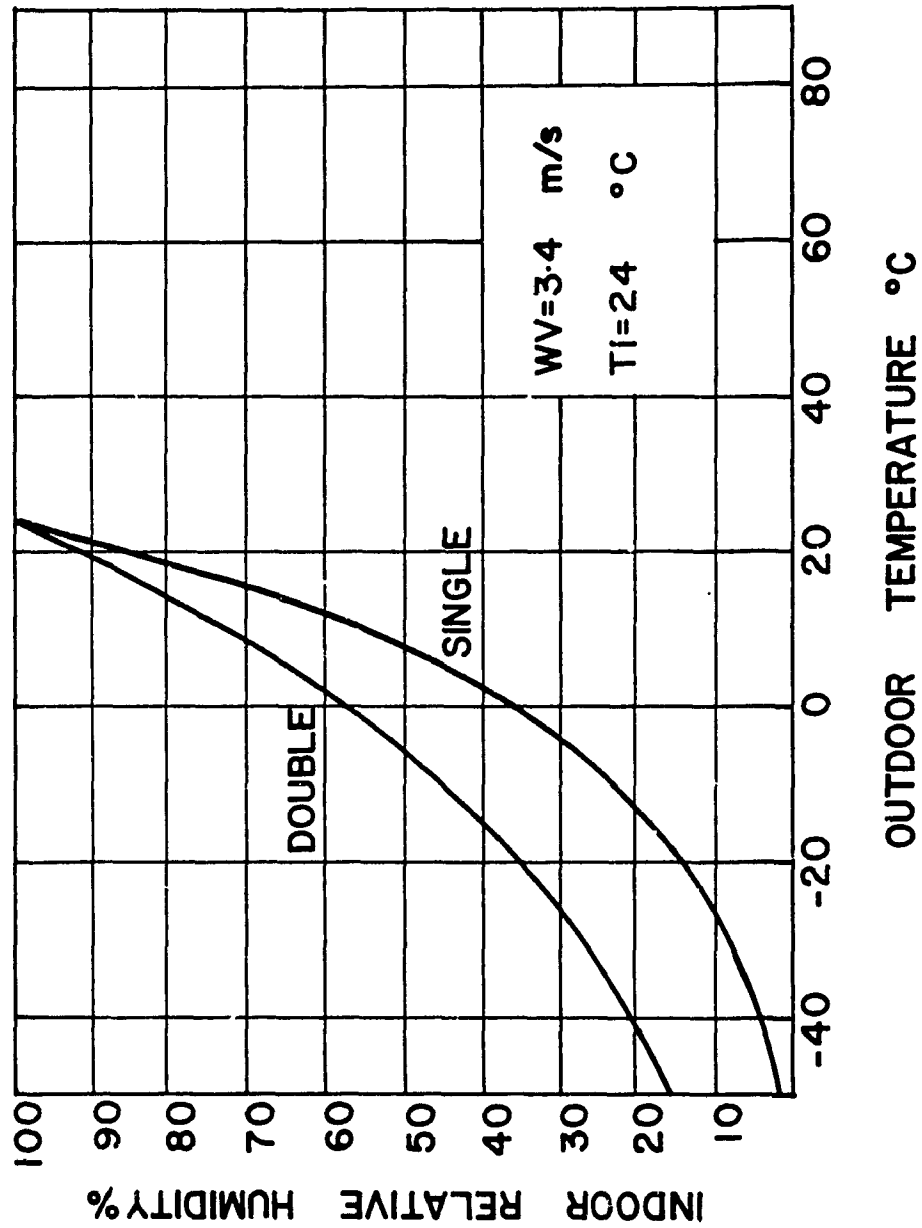


FIG. 2-8: Comparison between internal surface condensation potential of a single and double glazed window with an indoor temperature of 24°C; and 3.4 m/s wind velocity

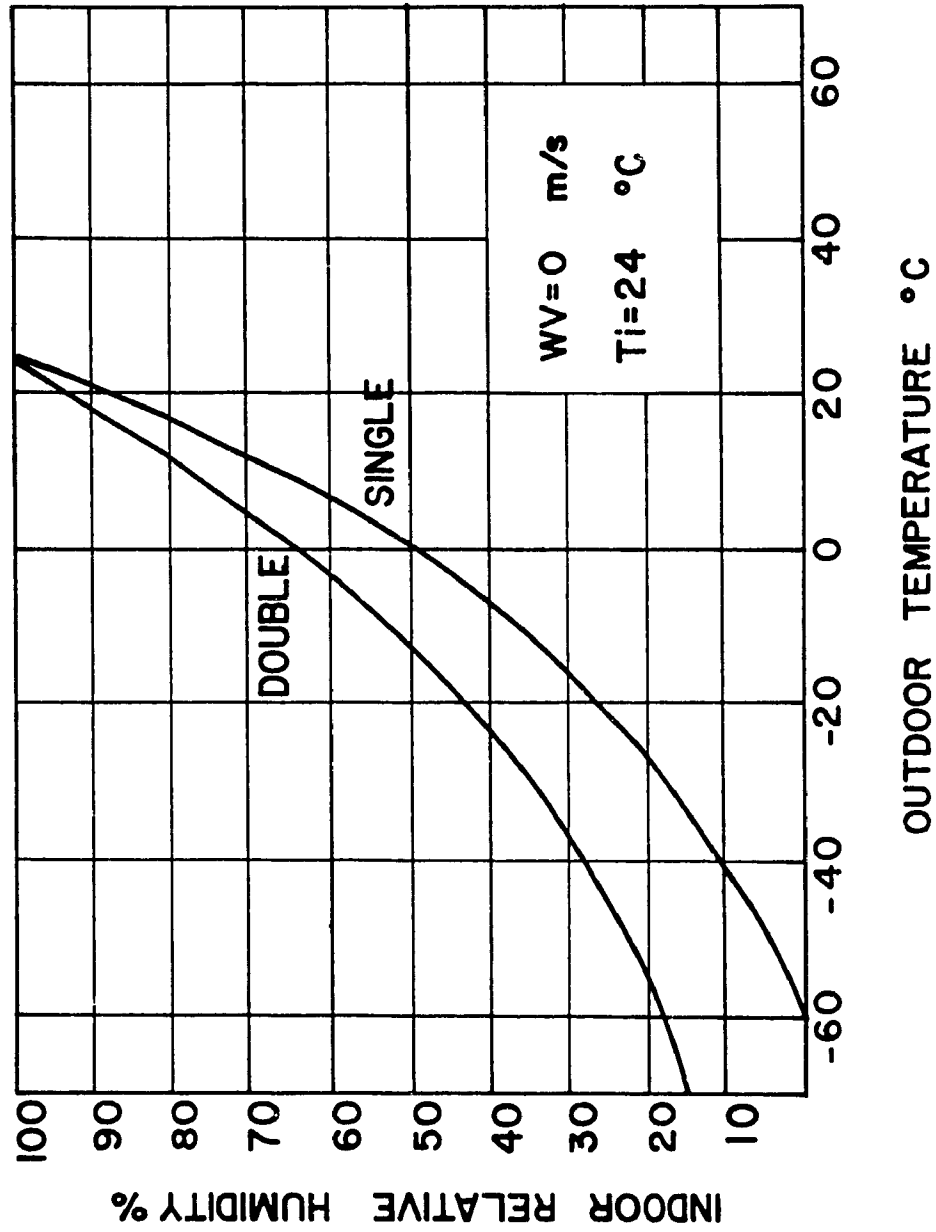


FIG. 2-9: Comparison between internal surface condensation potential of a single and a double glazed window with an indoor temperature of 24°C; and 0 m/s wind velocity

wind makes condensation possible at lower indoor relative humidity level, and much higher relative humidity level is needed for condensation to occur on double or triple glazed windows. In addition, the relative humidity needed for condensation to occur decreases more sharply for single glazed windows as outdoor temperature decreases compared to double and triple windows.

Table (2.1): Maximum Humidities for no Window Condensation and Corresponding Inside Surface Temperatures [19]

Out-door Temp. °C	Surface Temperature and Relative Humidities with indoor Temperature at 21°C											
	Single Glazed				Double Glazed				Triple Glazed			
	6.7m/s wind		No wind		6.7m/s wind		No wind		6.7m/s wind		No wind	
	Temp °C	RH	Temp °C	RH	Temp °C	RH	Temp °C	RH	Temp °C	RH	Temp °C	RH
4.4	8.3	44%	12.8	59%	15	68%	16.1	73%	17.2	78%	17.8	8%
-6.7	0	24%	7.2	41%	11.1	53%	13.3	61%	15	68%	15.6	71%
-17.8	-8.3	12%	1.7	27%	7.2	41%	10	49%	12.2	57%	13.3	61%
-28.9	-16.7	6%	-4.4	17%	3.3	32%	6.7	39%	10	49%	11.1	53%
-40	-25.6	2%	-10	10%	-0.6	23%	3.3	31%	7.2	41%	8.9	46%

CHAPTER III

CONDENSATION HEAT TRANSFER

3.1 INTRODUCTION

Phase change of any substance is always associated with either heat input or heat output. Heat is released whenever the substance changes from the gas state to the liquid state. Therefore, the process of vapour condensation is always associated with latent heat release. The amount of this heat is dependent upon the nature of the substance as well as the vapour temperature. Water, compared to all other substances, has the greatest latent heat of vaporization. So, considerable amount of heat is expected to be released as condensation of water vapour takes place.

When water vapour comes in contact with a surface that is at a temperature below its saturation temperature, it immediately starts to condense. The heat released is equal to the latent heat of vaporization times the weight of the condensed liquid. If the surface is kept at a temperature below the saturation temperature of the vapour and the condensed water is removed continuously, condensation will continue to occur and the surface will always be covered with a thin film of water.

The rate of condensation and heat transfer vary according to the type of condensation and whether the vapour is pure or contains some noncondensable gases. Condensation of pure vapour and corresponding heat transfer on a vertical surface were originally studied by Nusselt

[31], and his theory has been successful and still in use till today. However, Nusselt theory cannot be applied if the vapour contains some noncondensable gases. Noncondensable gases act as a resistance to heat flow on the condensing side. This is due to the fact that vapour has to diffuse through the noncondensable gas before it comes into contact with the cold surface of the condensate [31]. Therefore, the heat transfer mechanism in the presence of a noncondensable gas is completely different from that suggested by Nusselt which assumes direct contact between the vapour and the liquid film.

Despite the fact that the atmospheric water vapour, which is the main concern in this study, is not considered as a pure vapour, the condensation process of a pure vapour according to Nusselt will be discussed as the first step towards understanding the behaviour of condensed vapour and the mechanism of heat transfer when condensation occurs. Water vapour represents very small fraction of the atmospheric air-water vapour mixture. Therefore, it is vital to find some way to predict the behavior of atmospheric vapour and the mechanism of heat transfer during condensation. In this chapter, two approaches with different levels of analysis are discussed to predict condensation rate and heat transfer when condensation of atmospheric vapour occurs.

3.2 CLASSIFICATION OF CONDENSATION

There are several ways in which the general topic of condensation can be classified. It can be classified in terms of; i) the nature of

vapour to be condensed; ii) the geometry in which condensation occurs and iii) the mode of condensation [32]. All these factors have great impact on the behavior of the condensation process and the mechanism of heat transfer. Therefore, it is important to identify the problem in terms of the above factors, so as the condensation rate and the heat transfer rate can be evaluated properly.

i) Condensation can be classified in terms of the nature of vapour to be condensed. The vapour to be condensed may have any one of the following forms:

1. Pure one component condensable vapour
2. Pure multicomponent vapour with all of the component condensable
3. Multicomponent with a noncondensable gas
4. Multicomponent with an immiscible liquid phase

The condensation process and the corresponding heat transfer are also very much affected by the type of vapour to be condensed. However, Nusselt's theory can be used to describe the process, using the physical properties of the vapour, as long as the vapour is pure. But, in the presence of a noncondensable gas (like air) the condensation process is completely different. Therefore the effect of the noncondensable gas should be taken into account when analyzing the condensation phenomenon.

ii) Condensation can also be classified in terms of the geometry of the surface on which condensation takes place. The behavior of

the condensed liquid and heat transfer is different for each geometrical shape, because each one defines the hydrodynamic of condensate removal from the surface and the interaction of the vapour and liquid phases. For example, the mechanism of the condensation process on a vertical plane surface involves the gravitational force, since condensed water flow downward by gravity. As the surface slopes down to the horizontal position the influence of gravity decreases, until the process is completely gravity-independant when the surface is horizontal. The condensation rate and heat transfer are very much affected by the surface position, since unremoved condensed water will decrease the potential for further condensation and consequently the heat transfer.

iii) Condensation can also be classified according to the mode of condensation. Generally condensation can be classified into four different modes:

- a) Direct contact condensation
- b) homogeneous condensation
- c) dropwise condensation
- d) filmwise condensation

Direct contact condensation occurs whenever a condensable vapour is brought into direct contact with a subcooled liquid. Direct contact condensation is an attractive mode commonly used in the industrial applications because of the simplicity of the equipment required and because there is no solid heat transfer surface to be

fouled by direct or corrosive deposits [32]. However, since the condensed vapour and the coolant are intimately mixed, applications of direct contact condensation are limited to cases in which mixing is of no consequence (as in the barometric condenser) or in which the condensate and coolant can be easily separated.

Homogeneous condensation occurs when a super cooled vapour of gas-vapour mixture starts to develop nucleation centers. If such nucleation centers develop at all, they will grow very rapidly by condensation of the surrounding subcooled vapour due to warming the vapour by the release of the heat of condensation. This process will continue until the condensate droplets are in approximate thermodynamic equilibrium with the vapour. Statistically, there is a possibility that nucleation centers will develop in any subcooled vapour or gas-vapour mixture, but true homogeneous condensation as defined above is at best unlikely and more generally extremely improbable in real world situations [32]. However, this process can be simulated by introducing some pre-existing foreign nuclei such as salt or dust particles.

Direct contact condensation requires an external condensed liquid surface to occur and homogeneous condensation occurs in the absence of any external condensed phase (liquid or solid) surface. But a solid surface is needed for dropwise and filmwise to occur. Dropwise condensation occurs when a vapour is exposed to a cool solid surface and the resulting condensate does not wet the surface. And when the

condensate wets the surface and is covered by a continuous liquid phase the process is called filmwise condensation. In dropwise condensation, the droplets formed will flow down the surface leaving it unwetted and ready for further condensation. But in filmwise condensation a continuous liquid film will always be present and further condensation will occur over it. The down flow of the liquid film is governed by the usual laws of fluid dynamic, with gravity, vapour shear, and surface tension forces providing the driving force for condensate removal.

Dropwise condensation is much more effective in heat transfer than filmwise condensation, since the existence of a liquid film increases the resistance to heat transfer. Consequently, continuous efforts have been made to harness this dynamic mechanism for industrial use. Various methods have been utilized to attempt to achieve consistent dropwise condensation, such as the use of polymer surface coatings. However, dropwise condensation has only been achieved with consistency in steam under carefully controled conditions [33]. In practical situations most surfaces get wetted when exposed to a condensing vapour over an extended period of time. Consequently, film condensation can be seen as the best representation for the process of atmospheric vapour condensation on glazed windows. Therefore, the process of film condensation is discussed in this study, and is used to predict heat transfer to the surfaces of windows when condensation occurs.

3.3 ANALYSIS OF FILM CONDENSATION ON A VERTICAL SURFACE

The objectives of the present research is to study condensation on glass windows, and to determine the associated heat transfer. The process of condensation on vertical window surfaces can be well represented by film condensation analysis. The analysis of this problem is very dependent on the nature of vapour as mentioned earlier. Analysis of pure vapour condensation is completely different from that of a vapour containing a noncondensable gas.

The first fundamental analysis leading to the determination of the heat transfer coefficient during filmwise condensation of pure vapours on a flat plate and a circular tube was given by Nusselt in 1916 [33]. Since that time some improvements have been made to Nusselt's theory of film condensation. However, the basic analysis suggested by Nusselt is still being used. Nusselt's theory of film condensation of pure vapour on vertical surfaces will be utilized in this study as a basis for understanding the mechanism of heat transfer during the condensation process on windows.

3.3.1 Condensation of a Pure Vapour Using Nusselt's Theory

When a vapour is brought into contact with a cold surface, condensation occurs and a thin film of the condensate is formed on the surface. As the condensate continues to form on the surface, the film thickness increases. Consequently, the condensate starts to flow down the plate due to gravity and a thickness gradient is formed. As a

result of vapour condensation, latent heat is released at the vapour-liquid interface. Such heat is then transported through the condensate film to the solid wall by conduction when the film is in laminar flow and by turbulent eddies when the film is in turbulent flow.

Filmwise condensation and determination of the corresponding heat transfer were first given by Nusselt under the following assumptions [31]:

- i) The plate is maintained at a uniform temperature
- ii) The vapour is stationary or has low velocity
- iii) The down flow of the condensate under gravity action is laminar
- iv) The flow velocity associated with the condensate film is low
- v) Fluid properties are constant
- vi) Heat transfer across the condensate layer is by pure conduction.

The concept of the analysis of the film condensation process as suggested by Nusselt can be represented as shown in Fig. 3-1 where X is the axial coordinate, measured downward along the plate, and Y is the coordinate normal to the surface. The film thickness is represented by δ and it varies along the X axis. The shear force is shown acting upward and is denoted by τ .

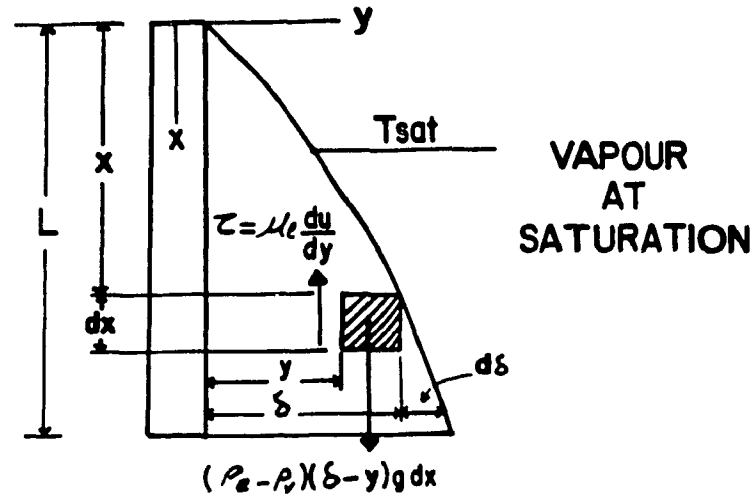


FIG. 3-1: Film condensation on a vertical Flat Plate [31]

By analyzing the fluid element of thickness dx shown in Fig. 3-1, and by using the weight balance equation and the boundary conditions, the mass flow rate (\dot{m}) through any x position is found to be [31]

$$\dot{m} = \frac{\rho_L (\rho_L - \rho_v) g \delta^3}{3\mu_L} \quad (3.1)$$

where ρ_L = liquid density (kg/m^3)

ρ_v = vapour density (kg/m^3)

μ_L = dynamic viscosity of the liquid (kg/m.s)

δ = condensate layer thickness

g = gravitational constant (m/s^2)

The heat transfer at the surface in the area dx , assuming a unit depth is found to be [31]

$$q_x = \frac{K_L}{\delta} dx (T_{sat} - T_w) \quad (3.2)$$

where

K_L = thermal conductivity of water ≈ 0.6 w/m-C

T_{sat} = saturation temperature of the vapour

T_w = surface or wall temperature

As a result of condensate flow from position x the film thickness grows from δ to $\delta + \Delta\delta$ at $x + \Delta x$. The increase in the condensate is [31]

$$\Delta \dot{m} = \frac{\rho_L (\rho_L - \rho_V) g \delta^2 d\delta}{\mu_L} \quad (3.3)$$

So the heat transfer to the wall is expressed by [31]:

$$\frac{\rho_L (\rho_L - \rho_V) g \delta^2 d\delta}{\mu_L} hfg = K_L dx \frac{T_{sat} - T_w}{\delta} \quad (3.4)$$

where hfg = latent heat of vaporization (J/kg)

Integrating equation (3.4) with the condition $\delta = 0$ for $x = 0$, yields:

$$\delta = \left[\frac{4\mu_L K_L x (T_{sat} - T_w)}{g hfg \rho_L (\rho_L - \rho_V)} \right]^{\frac{1}{4}} \quad (3.5)$$

and the resulting heat transfer coefficient can be written as

$$h_x = \frac{K_L}{\delta} = \left[\frac{\rho_L (\rho_L - \rho_v) g hfg K_L^3}{4 \mu_L x (T_{sat} - T_w)} \right]^{\frac{1}{4}} \quad (3.6)$$

By using equation (3.6), the heat transfer coefficient can be found at any distance x from the top of the vertical surface. For practical reasons, the average heat transfer coefficient needs to be determined. From Eq. (3.6), it can be seen that the local heat transfer coefficient (h_x) varies with the distance as $x^{-\frac{1}{4}}$. Then the average heat transfer coefficient (h_m) over the length $0 \leq x \leq L$ of the plate is given by [31]:

$$h_m = \frac{1}{L} \int_0^L h_x dx = \frac{4}{3} h_x|_{x=L} \quad (3.7a)$$

substituting for h_x in Eq. (3.7a), h_m can be written as:

$$h_m = 0.943 \left[\frac{g \rho_L (\rho_L - \rho_v) hfg K_L^3}{\mu_L (T_{sat} - T_w) L} \right]^{\frac{1}{4}} \quad (3.7b)$$

where

L = surface height

It should be noted that all properties in Eq. (3.7b) including hfg should be evaluated at film temperature T_f

$$T_f = \frac{T_{sat} + T_w}{2} \quad (3.8)$$

The additional energy needed to bring the film below saturation temperature is accommodated approximately by evaluating h_{fg} at the film temperature instead of the saturation temperature.

The above equations, suggested by Nusselt, have been tested under conditions closely satisfying his assumptions and have been generally confirmed [32]. However, a comparison with the experiments showed that the measured heat transfer coefficient is about 20 percent higher than Nusselt's theory would suggest [31]. Therefore, it has been recommended by McAdams that Nusselt's equation for h_m on a vertical surface be multiplied by a factor of 1.2, hence equation (3.8) should be:

$$h_m = 1.13 \left[\frac{g \rho_L (\rho_L - \rho_V) h_{fg} K_L^3}{\mu_L (T_{sat} - T_w) L} \right]^{1/4} \quad (3.9)$$

The above results for film condensation are applicable if condensate flow is laminar. And when the flow is turbulent the following empirical correlation has been recommended by Kirkbride [33] for vertical surfaces:

$$h_m = 0.076 Re_f^{0.4} \left[\frac{g \rho_L (\rho_L - \rho_V) K_L^3}{\mu_L^2} \right]^{1/3} \quad (3.10)$$

where

$$Re_f = 3.77 \left[\frac{L (T_{sat} - T_w)}{\mu_L h_{fg}} \right]^{1/4} \left[\frac{g \rho_L (\rho_L - \rho_V) K_L^3}{\mu_L^2} \right]^{1/4} \quad (3.11)$$

The determination whether the flow within the film is laminar or turbulent can be established using Reynolds number of the fluid (Ref.). The critical Reynolds number is approximately 2000, and turbulence can occur when Reynolds number is greater than this critical number. And once the type of flow is specified, heat transfer can be determined using the proper equation.

3.3.2 Analysis of Condensation of a Vapour Containing a Non-condensable Gas

It is well known that the presence of a noncondensable gas such as air in a vapour, even in very small amount, can have a significant effect on condensation heat transfer. The reason is that when a vapour containing noncondensable gas condenses, the noncondensable gas is left at the surface which means that for further condensation to occur the vapour must diffuse through the gas-vapour mixture before it reaches the cold surface. This added resistance to vapour diffusion causes a drop in the partial pressure of the condensing vapour, which in turn causes the temperature of outside surface of condensate to drop below the saturation temperature. Heat transfer calculation that accounts for the effect of noncondensable gases cannot be made by using the simple condensation theory of Nusselt. This is simply because the condensate outside surface temperature is not known. The analysis to evaluate this temperature is complicated, since heat, mass and momentum transfer in the liquid and the vapour-gas mixture must be considered and solved simultaneously.

There have been many theoretical and experimental investigations to predict heat transfer in the presence of a noncondensable gas. These studies have shown that heat transfer is very much affected by the vapour flow patterns and the magnitude of the noncondensable gas in the bulk mixture [33]. The presence of a great percentage of a noncondensable gas like air can reduce the condensation heat transfer by many folds. For example, the presence of 5 percent air by mass in the mixture can reduce the heat transfer coefficient by a factor of 5 [33]. In fact, some experimental studies have shown that heat transfer may be reduced by fifty percent or even more due to small amounts of air [29]. Therefore, heat transfer resulting from atmospheric water vapour condensation is expected to be very much less than Nusselt's theory would suggest, because the percentage of the water vapour by mass is very small compared to the noncondensable gas, which is air in this case. In fact, air could represent more than 95 percent of the atmospheric mixture. However, this percentage is variable and very dependant upon the climatic conditions. Therefore, it is clear that the presence of air is the main dominant factor affecting condensation and heat transfer to building surfaces.

Atmospheric water vapour condensation and the associated heat gain or loss is the main subject of this research. There have been limited attempts to estimate the rate of superficial condensation on building surfaces. However, there is not enough information and clear guidance in the literature to predict how much heat gain or loss will

result when condensation occurs on the building glass surfaces under given indoor and outdoor conditions. Therefore, in the present study the basic ideas of two different approaches will be utilized to predict heat transfer to glass windows and consequently heat gain or loss when condensation occurs under certain conditions. The first approach is mainly based on the equations that describe mass transfer due to partial vapour pressure difference. This approach was utilized by Davies [16], to predict mass condensation rate on glazed windows. A second approach for evaluating mass condensation rate is used in this study for the first time in order to evaluate the relative accuracy and applicability for each approach when compared with the experimental results. In the second approach more analysis is given to the condensate film which is mainly based on the conservation laws.

3.3.2.1 First Approach: Evaluation of Condensation Heat Transfer by Predicting Condensation rate using Mass Transfer Analysis

Computing the rate of superficial condensation on cold building surfaces using mass transfer analysis has been attempted by Davies [16, 17]. This approach is mainly based on the fact that mass transfer occurs as a result of the difference between the bulk vapour partial pressure and vapour partial pressure at the cool surfaces. The analysis is based on steady state conditions, but it was suggested by Davies that more realistic conditions involve considerations of time varying heat flow, and time varying moisture flow [16].

To start the analysis, consider first Fig. 3-2 which shows a single glazed window at an external surface temperature T_{wo} and an internal surface temperature T_{wi} which represent the conditions after condensation occurs

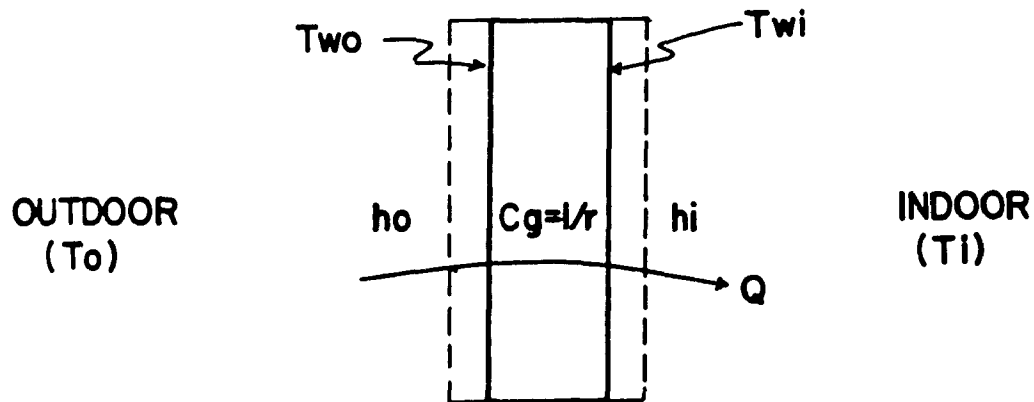


Fig. 3-2: Schematic of surface temperature of a single glazed window

If no condensation takes place, then the heat balance for a unit area requires that

$$(T_o - T'_{wo}) h_o = (T'_{wo} - T'_{wi}) C_g = U (T_o - T_i) \quad (3.12)$$

and

$$(T'_{wo} - T'_{wi}) C_g = (T'_{wi} - T_i) h_i = U (T_o - T_i) \quad (3.13)$$

where

T'_{wo} = outdoor facing surface temperature before condensation

T'_{wi} = indoor facing surface temperature before condensation

h_o, h_i = outside and inside film heat transfer coefficient (W/m^2-C)

C_g = glass thermal conductance (W/m^2-C)

U = overall heat transfer coefficient of the window (W/m^2-C)

From the above energy balance equations it can be found that

$$T'_{wo} = T_o - \frac{U}{h_o} (T_o - T_i) \quad (3.14)$$

$$T'_{wi} = T_i - \frac{U}{h_i} (T_i - T_o) \quad (3.15)$$

As condensation occurs on the inside surface, latent heat will be released on the surface. Therefore, it must be included in the heat balance equation. If \dot{m} $kg/m^2 \cdot sec$ of water is assumed to condense on the glass, the resultant heat balance equation when outdoor surface condensation occurs becomes:

$$(T_o - T_{wo}) h_o + \dot{m} h_{fg} = U (T_o - T_i) \quad (3.16)$$

and when indoor surface condensation occurs the resultant heat balance equation becomes:

$$(T_i - T_{wi}) h_i + \dot{m} h_{fg} = U (T_o - T_i) \quad (3.17)$$

To evaluate the surface temperatures in Eqs. (3.16) and (3.17) by mass transfer analysis, the bulk partial vapour pressure is first denoted by P_a (N/m^2). If there is no condensation, P_a acts directly on the window surface. As soon as condensation occurs the window is

covered by a water film, and the vapour pressure P_C at the surface of the window will be the saturated vapour pressure (SVP) at the window temperature which is denoted by

$$P_C = \text{SVP } (T_{wo}) \quad (3.18)$$

So, when the window is sufficiently cool a mass transfer will occur due to vapour pressure difference ($P_a - P_C$). This mass transfer of vapour from the bulk to the window is given by [16]:

$$\dot{m}_i = h_D \frac{M}{R T_m} (P_a - P_C) \quad (3.19)$$

where

h_D = mass transfer coefficient (m/sec)

M = the mass of one Kmol of water vapour (= 18 Kg)

R = universal gas constant = 8314 J/Kmol-K

T_m = absolute temperature which is taken as the mean of the bulk air temperature and the surface temperature

solving Eqs. (3.16) and (3.19) simultaneously the condensation rate can be found. However, the saturated temperature P_C is not linearly related to the temperature. Therefore, the following graphical solution has been suggested by Davies:

Rearranging Eq. (3.16), one can get

$$\dot{m}_i h_{fg} = U (T_o - T_i) - (T_o - T_{wo}) h_o \quad (3.20)$$

then by multiplying both sides of Eq. (3.20) by $\frac{R \cdot T_m}{h_D \cdot M \cdot h_{fg}}$ one can obtain:

$$F_1 \equiv \frac{mR T_m}{h_D M} = \frac{R T_m}{h_D M h_{fg}} [U(T_0 - T_i) - h_0 (T_0 - T_{wo})]$$

$$\equiv \left(\frac{mR T_m}{h_D M}\right)_1 = \frac{R T_m h_0}{h_D M h_{fg}} T_{wo} + \frac{R T_m}{h_D M h_{fg}} [(U - h_0)T_0 - UT_i] \quad (3.21)$$

and using Eq. (3.19) gives

$$F_2 \equiv \left(\frac{mR T_m}{h_D M}\right)_2 = P_a - P_c = P_a - SVP(T_{wo}) \quad (3.22)$$

Eqs. (3.21) and (3.22) are satisfied simultaneously at the point of intersection of F_1 and F_2 . The glass temperature and the pressure difference can be found by projection on the temperature and the pressure axes as shown in Fig. 3-3.

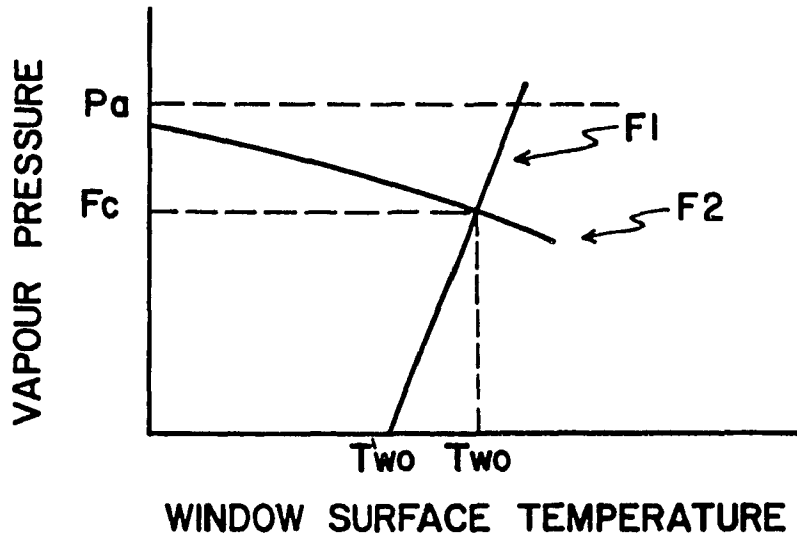


FIG. 3-3: Solution of equation (3.21) and (3.22) to find window surface temperature T_{wo} and the difference F_c of vapour pressure between ambient and window [16]

Knowing the value of F_c which represents the difference between the bulk vapour pressure and the saturated vapour pressure ($P_a - P_c$) after the window temperature reaches steady state condition, the mass condensation rate can be found using Eq. (3.19) as following [16]:

$$\dot{m} = h_D \frac{M}{R T_m} F_c \quad (3.23)$$

All other parameters in equation (3.12) are easy to evaluate, however the mass transfer coefficient h_D needs more attention and has to be evaluated properly because it dominates the estimate of the rate of condensation. And at the same time it is very sensitive to other factors like the surface heat transfer coefficient which in turn is affected by the type of boundary layer, whether it is laminar or turbulent. Therefore, clear understanding of the mass transfer coefficient is needed.

Evaluation of Mass Transfer Coefficient (h_D)

Mass transfer coefficient value depends on whether the flow is laminar or turbulent [31]. The mass transfer coefficient (h_D) relating to transport of water vapour through the boundary layer is often found using Lewis' relation [16].

$$h_D = \frac{h_c}{\rho_a C_a} \quad (3.24)$$

where

h_c = convective heat transfer coefficient $\text{w/m}^2\text{-C}$

ρ_a = air density

C_a = specific heat of air (J/kg-C)

Eq. (3.24) can be used to evaluate the mass transfer coefficient when the conditions in the boundary layer are turbulent [16]. However, for relatively still conditions like indoor conditions h_D can be found by the following relation [16]:

$$h_D = \frac{D}{K_a} \left(\frac{h_c}{\rho_a C_a D} \right)^{1/4} \quad (3.25)$$

where

D = diffusion coefficient of water vapour (m^2/sec)

K_a = thermal conductivity of air (w/m-C)

For forced flow the transition from laminar to turbulent can be known using the critical values of Reynold number. Similarly Grashof number can be used to know whether the flow is laminar or turbulent [31].

$$Gr = g \beta_a L^3 \left(\frac{T_w - T_\infty}{\nu_a^2} \right)$$

where g = gravity constant

β_a = air expansion coefficient (K^{-1})

ν_a = kinematic air viscosity (m^2/s)

T_w = wall temperature

T_∞ = bulk temperature

L = characteristic length

The critical value of Grashof number (Gr) is 2×10^9 and for laminar flow to occur Gr number should be less than the critical value.

To evaluate the convective heat transfer coefficient, the type of flow should first be determined, since it can reach high values in the turbulent conditions [16]. A typical value of h_c for convection from warm air to cool surfaces within a room, as suggested by Davies [16], is $3.0 \text{ W/m}^2\text{.C}$. However, for exterior surfaces, h_c will vary according to the wind speed and direction and the type of flow. The evaluation of the convective heat transfer coefficient (h_c) will be discussed in more details in the next chapter.

3.3.2.2 Second Approach: Evaluation of condensation heat transfer by predicting the interfacial temperature of the liquid film

In this approach the basic concept of film theory which has been originally suggested by Nusselt will be used to predict the condensation heat transfer in the presence of a noncondensable gas. However, this analysis, as suggested by Sparrow and Lin [29], will be based on the mass, momentum and energy conservation principles. It is known that the presence of a very small amount of a noncondensable gas (such as air) in a condensable vapour can cause a large build up of the

noncondensable gas at the liquid-vapour interface. As a result the partial pressure of the vapour at the interface is reduced. This, in turn, will lower the temperature at which the vapour condenses and decrease the effective thermal driving force across the liquid film. Consequently, heat transfer and mass transfer will be reduced significantly [29].

There have been several theoretical and experimental studies related to this subject [24-30]. Although each study has its own approach in analyzing the problem, the aim was always to find the interfacial temperature T_{gi} and the interfacial gas concentration w_{gi} . Once one of these parameters is known, the condensation heat transfer can be predicted using the same form of Nusselt's equation (Eq. 3.7b), but replacing the saturation temperature (T_{sat}) by T_{gi} . Fig. 3-4 shows a schematic diagram of the physical system involved in

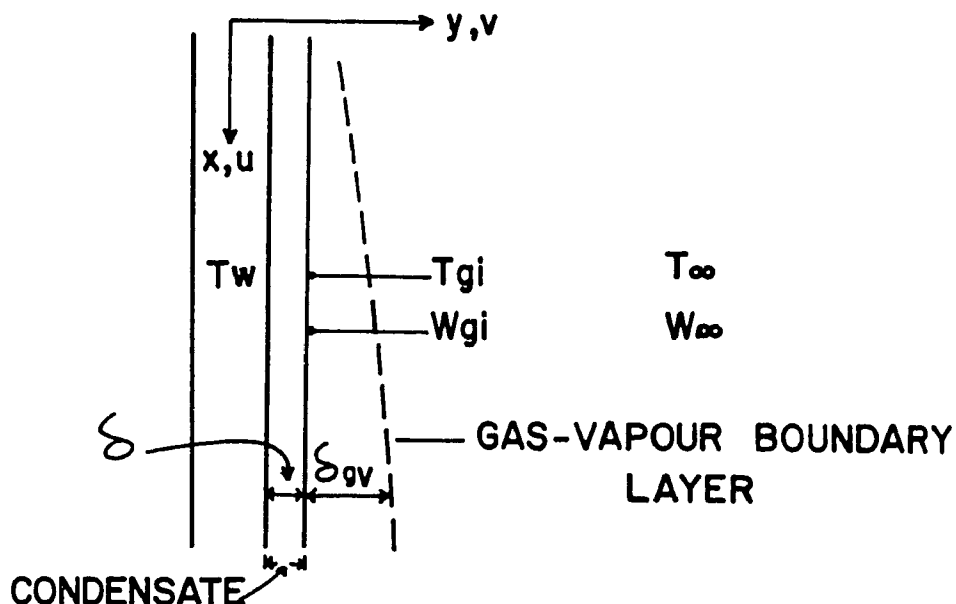


FIG. 3-4: Film condensation of a vapour containing a noncondensable gas [27]

this study. The solution of this problem is very complicated, because it consists of a pair of interacting boundary layers, an inner one (liquid film) and an outer one (gas-vapour layer). The liquid film is represented by the following momentum and energy equations that are originally suggested by Nusselt [29]:

$$g + \nu_L (\partial^2 u / \partial y^2) = 0 \quad (3.26a)$$

$$\partial^2 T / \partial y^2 = 0 \quad (3.26b)$$

Eqs. (3.26a) and (3.26b) can be solved using the following boundary conditions.

$$u = v = 0 \quad \text{at } y = 0$$

$$T = T_w \quad \text{at } y = 0$$

$$\partial u / \partial y = 0 \quad \text{at } y = \delta$$

$$T = T_{gi} \quad \text{at } y = \delta$$

where

y, x = normal and longitudinal coordinates

v, u = normal and longitudinal velocities

ν_L = liquid kinematic viscosity

T_w = wall surface temperature

T_{gi} = interfacial temperature

w_∞ = bulk noncondensable gas concentration

In the vapour-gas layer, there are three governing equations, since there may be simultaneous transport of heat, mass and momentum. The conservation equations of this layer are written as [29]:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0, \quad u \frac{\partial W_g}{\partial x} + v \frac{\partial W_g}{\partial y} = D \frac{\partial^2 W_g}{\partial y^2} \quad (3.27a)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = g \left(\frac{1 - \rho_\infty}{\rho} \right) + \nu \frac{\partial^2 u}{\partial y^2} \quad (3.27b)$$

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2} \quad (3.27c)$$

where

D = binary diffusion coefficient

ρ_∞ = bulk density

ρ = average gas-vapour mixture density

ν = average gas-vapour mixture kinematic viscosity

The boundary conditions and assumptions used to solve these equations are [29]:

- i) velocity continuity at interface, $y = \delta$
- ii) mass conservation at interface, $y = \delta$
- iii) vanishing longitudinal velocity in the bulk $y = \infty$
- iv) interface impermeable to non-condensable gas
- v) prescribed value of W_g in the bulk

By utilizing the boundary conditions. Sparrow and Lin [29], were able to formulate the above equations (Eqs. 3.27a - 3.27c) and reduce

ation:

a) similarity variable

$$\eta = c(y-\delta)/x^{\frac{1}{2}}, \quad c = \left[\frac{g (M_g - M_v) / 4 v^2}{M_g - (M_g - M_v) W_\infty} \right]^{\frac{1}{2}} \quad (3.28a)$$

where

M_g = gas molecular weight = 28.97 for air

M_v = vapour molecular weight = 18.0 for water

b) stream function and mass fraction

$$\psi = 4 v c x^{\frac{1}{2}} f(\eta), \quad W_g - W_\infty = \phi(\eta) \quad (3.28b)$$

where

ψ = stream function

f = dimensionless stream function [Eq. (3.28b)]

ϕ = mass fraction difference [Eq. (3.28b)]

The velocity component u and v can be derived to satisfy the continuity equation (3-25a).

$$u = \frac{\partial \psi}{\partial y} = 4 v c^2 x^{\frac{1}{2}} f'(\eta) \quad (3.29a)$$

$$v = - \frac{\partial \psi}{\partial x} = \frac{v c}{x^{\frac{1}{2}}} \left[\frac{c y}{x^{\frac{1}{2}}} f' - 3f \right] \quad (3.29b)$$

By transforming the diffusion and momentum equations (Eqs. 3.27a and 3.27b) it can be found that

$$f''' + 3f f'' - 2(f')^2 + \phi = 0 \quad (3.30)$$

$$\phi'' + 3 Sc f \phi' = 0 \quad (3.31)$$

in which the primes denote differentiation with respect to η and Sc represent the Schmidt number.

By utilizing the boundary conditions sparrow and Lin [29] were able to numerically solve the above differential equations (3.30) and (3.31). This numerical solution was found to be favorable when compared to other methods (29). The results for w_{gi} and P_{vi} (interfacial vapour pressure) which have been determined from the numerical solution of Eqs. (3.30) and (3.31) is shown in Figs. 3-5, 3-6 and 3-7. The curves shown in the figures are obtained for certain values of the ratio $(\rho\mu)_L/\rho\mu$ and Schmidt number (Sc). On each figure there are one family of curves representing w_{gi} . The interfacial gas concentration (w_{gi}) and the vapour pressure at the interface (P_{vi}) are related to each other through the expression:

$$P_{vi}/P_{tot} = [1 - w_{gi}]/[1 - (1 - \frac{M_v}{M_g}) w_{gi}] \quad (3.32)$$

where

P_{tot} = total pressure

$(\rho\mu)_L$ = liquid density times its dynamic viscosity

$(\rho\mu)$ = average gas-vapour mixture density times its
dynamic viscosity

The use of these figures, to obtain W_{gi} or P_{vi} , is based on trial and error. The calculation procedure can simply be described as follows: assume that the wall temperature is T_w , and it is desired to compute the heat transfer rate corresponding to a given total pressure P_{tot} and a given concentration W_∞ of the noncondensable gas in the bulk. One would start by assuming an interface temperature T_{gi} and with this in mind one could compute $C_p (T_{gi} - T_w) / h_{fg} Pr_L$. Then entering the appropriate figure with this $C_p(T_{gi} - T_w) / h_{fg} Pr_L$, one can read W_{gi} from the ordinate and find P_{vi} using Eq. (3.32). Then, for this P_{vi} , one can find the corresponding saturation temperature from tables. If this saturation temperature is equal to the assumed T_{gi} then it is taken as the value of the interfacial temperature, and the heat transfer rate can be found using Eq. (3.33). If not, then another guess should be made and the same procedure is repeated. Once T_{gi} is found, the mass condensation rate can be evaluated by the following equation [29]:

$$m' = \frac{Q}{h_{fg}} = \frac{4 \mu_L}{3} \left[\frac{C_{pL} (T_{gi} - T_w)}{h_{fg} Pr_L} \right]^{\frac{1}{4}} \left(\frac{g L^3}{4 \nu_L^2} \right)^{\frac{1}{4}} \quad (3.33)$$

where

- Q = heat released by condensation
- Pr_L = Prandtl number for the fluid
- C_{pL} = liquid specific heat

Eq. (3.33) is exactly the same as the condensation heat transfer equation suggested by Nusselt. However, the only difference is that

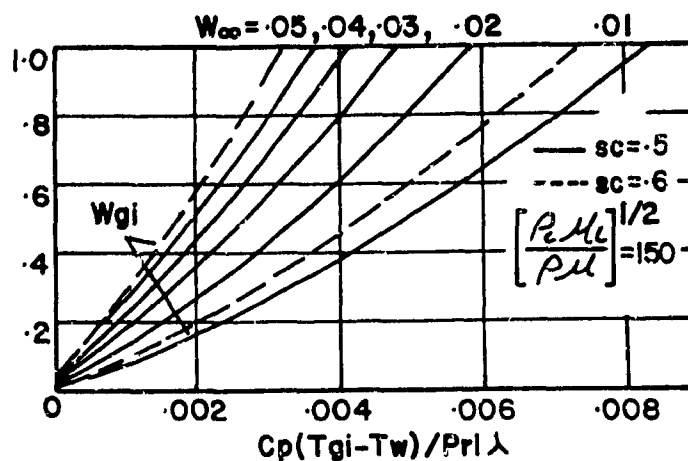


FIG. 3-5: Interfacial concentration of noncondensable gas $[(\rho\mu)_L/\rho\mu]^{1/2} = 150$ [27]

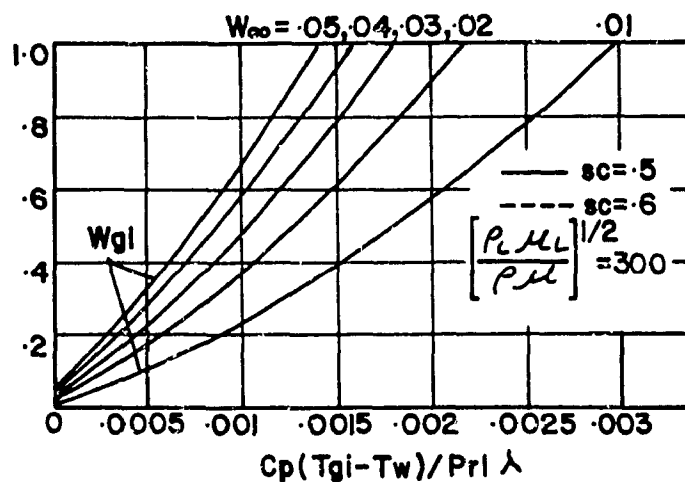


FIG. 3-6: Interfacial concentration of noncondensable gas $[(\rho\mu)_L/\rho\mu]^{1/2} = 300$ [27]

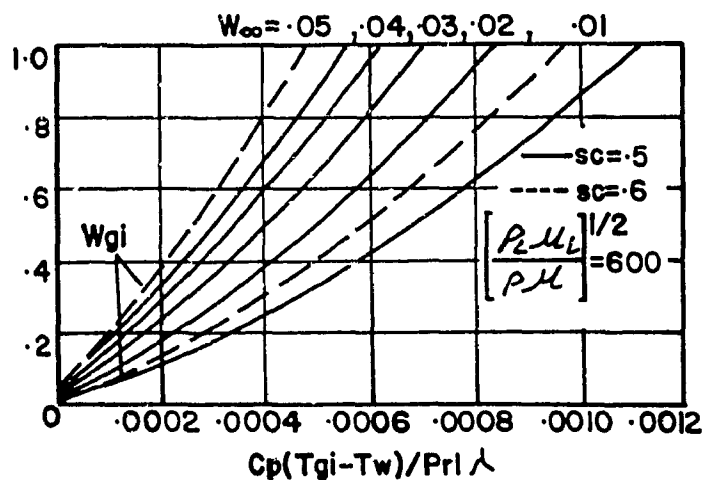


FIG. 3-7: Interfacial concentration of noncondensable gas $[(\rho\mu)_L/\rho\mu]^{1/2} = 600$ [27]

the saturation temperature (T_{sat}) has been replaced by a new interfacial temperature T_{gi} which is smaller in value.

Now, assuming that the above procedure can be used to predict the interfacial temperature when atmospheric vapour condenses on windows' surfaces the problem remains that the above curves cannot be used to predict the interfacial temperature (T_{gi}). Since these curves were produced to predict the heat transfer when the vapour contains very small percentage of a noncondensable gas, and for the atmospheric vapour-air mixture, the noncondensable gas (air) is the dominant element in the mixture (95% - 99%). Therefore, the same differential equations (Eqs. 3.30 and 3.31) should be solved for $W_{\infty} = 0.95, 0.96$, numerical solution of these equations is time consuming and needs a lot of computation especially when the surface temperature is allowed to vary. At the same time, the accuracy of this procedure cannot be guaranteed at this point when it is applied to the atmospheric mixture.

Fortunately, J.W. Rose [27], suggested an approximate solution to the problem. So, instead of solving the differential equations (Eqs. 3.30 and 3.31), an algebraic equation, which relates the heat and mass transfer parameters and the fluid properties, is to be solved. This approximate solution was found as shown in figs. 3-8 and 3-9 to be closer to the exact solution, suggested by Sparrow and Lin, when the

noncondensable gas concentration is increased. Therefore, this approximate solution is expected to give us a solution with acceptable accuracy for the problem of atmospheric vapour condensation on windows, since the noncondensable gas concentration (air) is much higher than the water vapour.

By using the same equation expressing the conservation of mass, heat, and momentum transfer (3.27), Rose suggested suitable velocity and concentration profiles in which the liquid and vapour-gas layers are assumed to have equal thicknesses. These profiles are [27]:

$$1) \quad u = u_{gi} \left(1 - \frac{y}{\delta_{gv}}\right)^2 + \tilde{u} \frac{y}{\delta_{gv}} \left(1 - \frac{y}{\delta_{gv}}\right)^2 \quad (3.34)$$

$$2) \quad \frac{w_g - w_\infty}{w_{gi} - w_\infty} = \left(1 - \frac{y}{\delta_{gv}}\right)^2 \quad (3.35)$$

where

u_{gi} = x-component of velocity at the interface

δ_{gv} = gas-vapour layer thickness

\tilde{u} = a function of x having dimensions of velocity as defined in Eq. (3.34)

w_g = noncondensable gas concentration

Then, by assuming that the interface is impermeable to the non-condensing gas, and using the boundary conditions that are satisfied by the above profiles, the integral equations are [27]:

$$\frac{d}{dx} (\delta_{gv} \int_0^1 u^2 dt) - v_{gi} u_{gi} + \frac{v}{\delta_{gv}} \left(\frac{\partial u}{\partial t}\right)_{gi} - g \times \delta_{gv} \int_0^1 \omega dt = 0 \quad (3.36)$$

$$\frac{d}{dx} (\delta_{gv} \int_0^1 u \omega dt) + \frac{D W_{\infty}}{\delta_{gv} W_{gi}} \left(\frac{\partial \omega}{\partial t} \right)_{gi} = 0 \quad (3.37)$$

where

$$t = y/\delta_{gv}$$

$$\omega = W_g - W_{\infty}$$

$$X = (M_g - M_v) / [M_g - W_{\infty}(M_g - M_v)]$$

The solution of Eqs. (3.36) and (3.37) using the assumed profiles (Eqs. (3.34) and (3.35)) gives [27]:

$$\begin{aligned} & 10 \text{ Sp SC } \left(\frac{\mu_L \rho_L}{\mu \rho} \right) \left(\frac{W_{\infty}}{\omega_0} \right)^2 \left(\frac{20}{21} + \frac{W_{gi}}{W_{\infty}} \text{ SC} \right) \\ & + \frac{8}{\text{Sp}^2 \text{ SC}} \left(\frac{\mu \rho}{\mu_L \rho_L} \right) \left(\frac{\omega_0}{W_{gi}} \right)^2 \left(\frac{5}{28} \text{ Sp} - \frac{X \omega_0}{3} \right) = \frac{100}{21} \frac{W_{\infty}}{W_{gi}} - \frac{2 \omega_0}{W_{gi}} + 8 \text{ SC} \quad (3.38) \end{aligned}$$

where

$$\text{Sp} = \frac{(T_{gi} - T_w) k_L}{h_{fg} \mu_L} \quad (3.38a)$$

$$\omega_0 = W_{gi} - W_{\infty}$$

$$\text{SC} = \nu/D$$

the values of the Kinematic viscosity (ν) and the diffusion coefficient (D) are to be evaluated for the vapour-gas mixture. However, in this study the parameters corresponding to the mixture will be taken for the air, since air dominates at atmospheric mixture. In addition

all other parameters without subscripts in Eq. (3.38) which corresponds to the mixture will also be taken for air.

By solving the algebraic Eq. (3.38), an approximate solution for the differential Eqs. (3.30) and (3.31), which have been suggested by Sparrow and Lin, can be found. So, the same type of curves shown in Figs. 3-5, 3-6 and 3-7 can be obtained. Consequently, the interfacial temperature (T_{gi}) of the condensate layer can be evaluated using the same procedure mentioned before. Fig 3-8 shows the comparison between the approximate solution and the exact solution, suggested by Sparrow and Lin [29]. It can be seen that the approximate solution, while giving about the right dependence on Sc , overestimates the interfacial gas concentration (W_{gi}). However, as the noncondensable gas concentration (W_{∞}) increases the deviation between the approximate and the exact solutions is diminished. In addition, Fig. 3-9 shows that as the value of W_{∞} increases, the heat transfer rate predicted by the approximate solution is more towards the exact solution. Therefore, it can be concluded that for $W_{\infty} = 0.9$ or more, which is the case in the atmospheric mixture, the predicted heat transfer rate by the approximate solution is the same as the exact solution.

The use of Eq. (3.38) has been limited to mixtures where the molecular weight of the noncondensable gas is larger than that of the vapour, which is the case for atmospheric mixture. In addition, if it

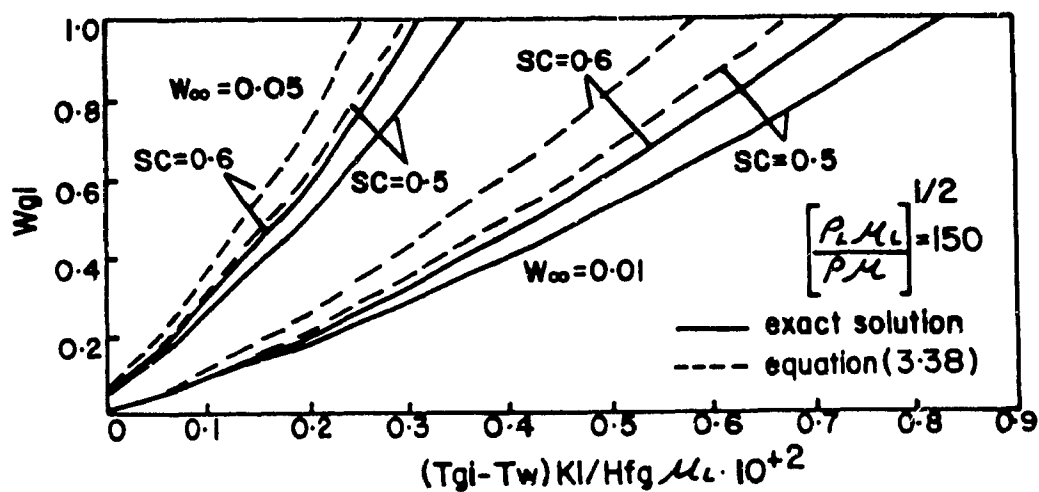


FIG. 3-8: Comparison of equation (3.38) and the exact solution [27]

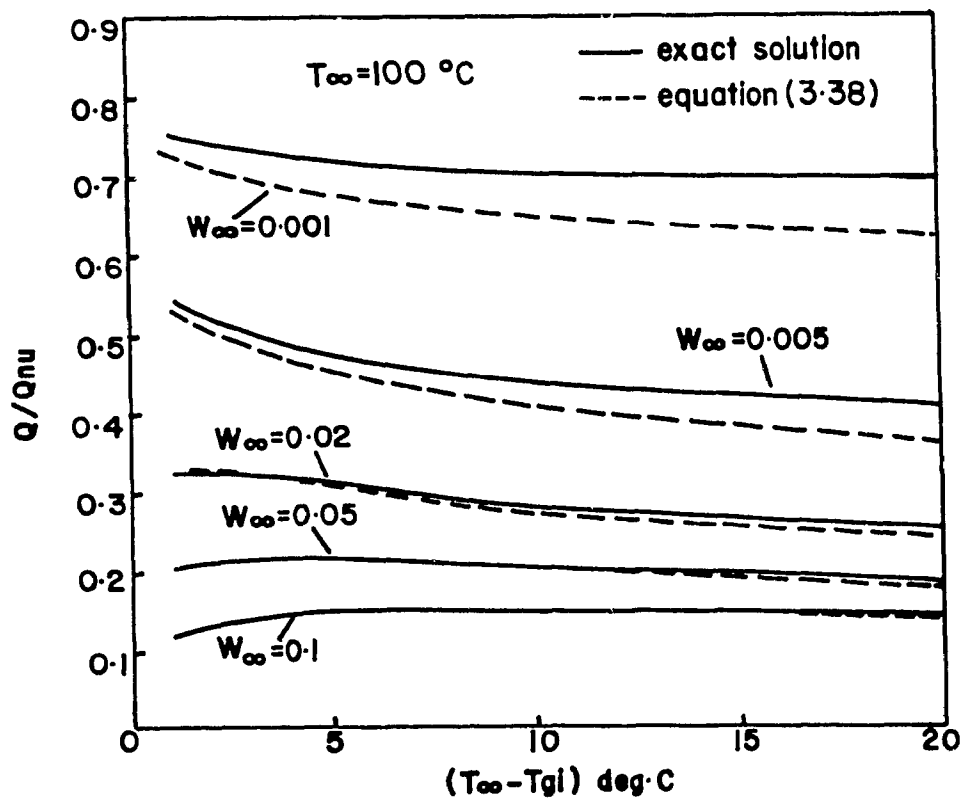


FIG. 3-9: Fractional reduction in heat transfer. Comparison of equation (3.38) with exact solution [27]

is assumed that the boundary layer approximation is valid and the flow within the boundary layer is laminar, this approximate solution suggested by Rose [27], can be used to predict heat transfer rate due to condensation of atmospheric vapour on glass windows. Figs. 3-10 and 3-11 show the relation between interfacial gas concentration against the nondimensional temperature drop across the condensate layer, $(T_{gi} - T_w) K_L / h_{fg} \mu_L$. These results were obtained by solving Eq. (3.38) for the atmospheric mixture in which the noncondensable gas concentration (W_∞) is very high. Therefore, Eq. (3.38) has been solved for values of W_∞ ranging from 0.94 to 0.99 at $Sc = 0.6$ and $[\rho_L \mu_L / \rho_\mu]^{1/2} = 195$, which reasonably describe the atmospheric mixture. So, by using the appropriate curve from Figs. 3-10 and 3-11, the interfacial temperature can be evaluated by the same procedure suggested by Sparrow and Lin. Consequently, heat transfer due to atmospheric vapour condensation can be predicted using Eq. (3.33).

The evaluation of the interfacial temperature from Eq. (3.38) requires that the parameter Sp [in Eq. (3.38a)] to be put as a function of the noncondensable gas concentration at the interface (W_{gi}). So, the terms in Eq. (3.38) are arranged to be in the following form:

$$\left(\frac{200}{21} Sc D W_\infty^2 W_{gi}^2 + 10 Sc^2 D W_\infty W_{gi}^3 \right) Sp^3 + \left[\frac{-100}{21} W_\infty W_{gi} (W_{gi} - W_\infty)^2 \right. \\ \left. + 2 W_{gi}^2 (W_{gi} - W_\infty)^2 - 2 W_\infty W_{gi} (W_{gi} - W_\infty)^2 - 8 Sc W_{gi}^2 (W_{gi} - W_\infty)^2 \right] Sp^2$$

$$+ \left[\frac{40}{28 SC} C (W_{gi} - W_{\infty})^4 \right] Sp = \frac{8X}{3Sc} C (W_{gi} - W_{\infty})^5 \quad (3.39)$$

where

$$C = \mu\rho/\mu_L\rho_L$$

$$D = \mu_L\rho_L/\mu\rho$$

the above equation can be written in the form of

$$AE Sp^3 + BE Sp^2 + CE Sp + DE = 0 \quad (3.40)$$

where

$$AE = \frac{200}{21} SC^2 D W_{\infty}^2 W_{gi}^2 + 10 SC^3 D W_{\infty} W_{gi}^3$$

$$BE = \frac{-100}{21} W_{\infty} W_{gi} SC (W_{gi} - W_{\infty})^2 + 2 SC W_{gi} (W_{gi} - W_{\infty})^3$$

$$- 8 SC^2 W_{gi}^2 (W_{gi} - W_{\infty})^2$$

$$CE = \frac{40}{28} C (W_{gi} - W_{\infty})^4$$

$$DE = \frac{-8X}{3} C (W_{gi} - W_{\infty})^5$$

Now, Eq. (3.40) has to be solved to find Sp. An since, it is a cubic equation Cardano's formula can be used to solve for Sp as a function of all other parameters. So, if Eq. (3.40) is divided by the factor (AE) the resulting equaiton is:

$$Sp^3 + BC Sp^2 + CC Sp + DC = 0 \quad (3.41)$$

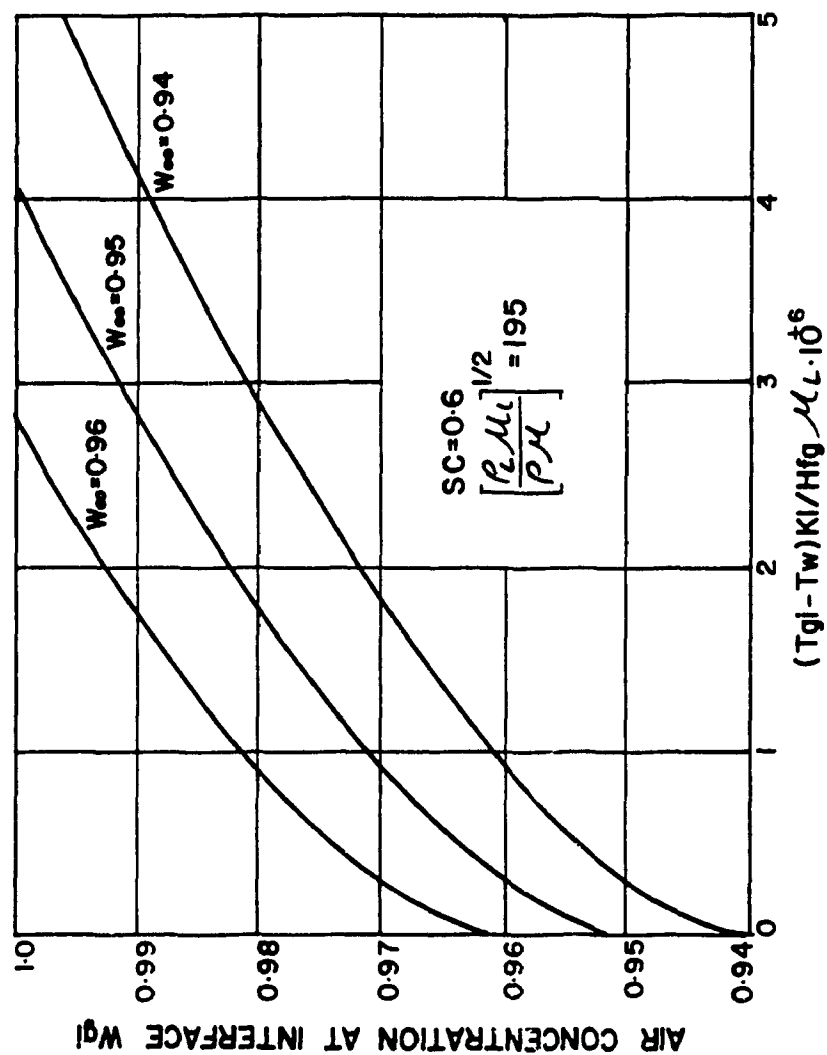


FIG. 3-10: Interfacial concentration of air $[(\rho \mu)L / \rho \mu] = 195$

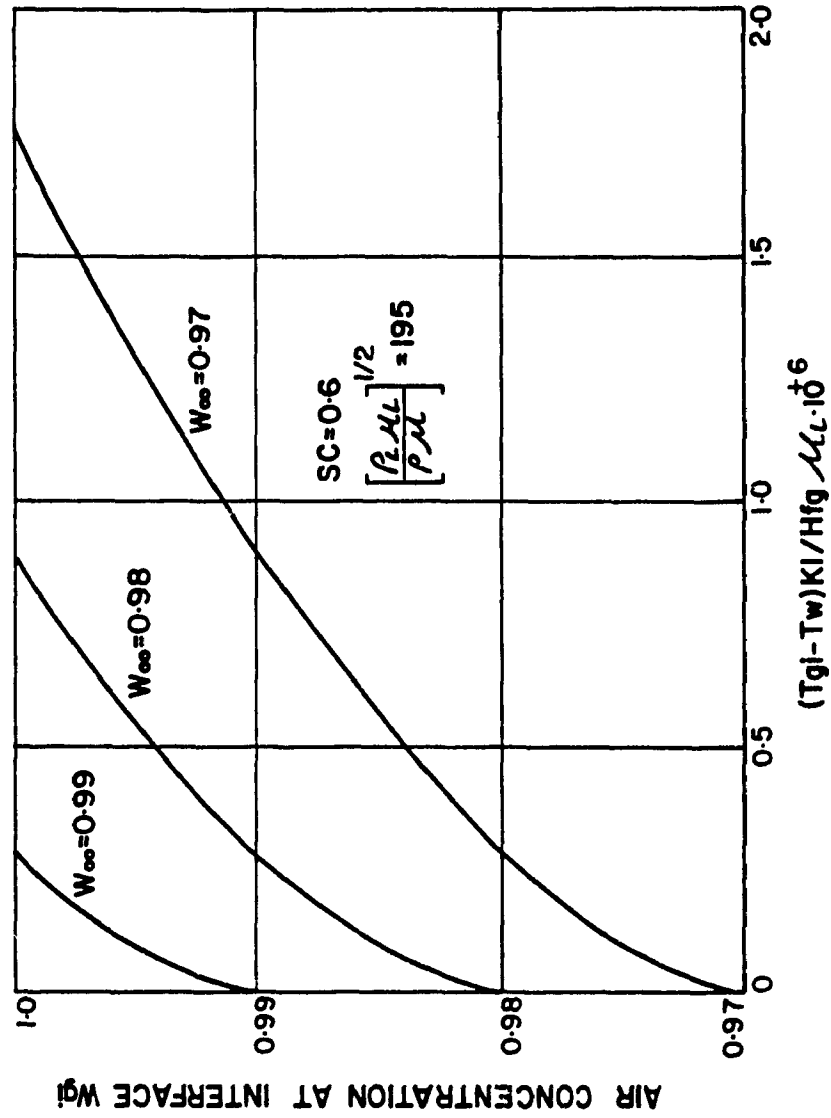


FIG. 3-11: Interfacial concentration of air $[(\rho \mu)_L / \rho \mu] = 195$

where $BC = BE/AE$

$CC = CE/AE$

$DC = DE/AE$

$$\text{if we let } Sp = YE - \frac{BC}{3} \quad (3.42)$$

and substitute (3.42) in (3.41) we get

$$YE^3 + PE * YE + QE = 0 \quad (3.43)$$

where

$$PE = CC - \frac{BC^2}{3}$$

$$QE = DC - \frac{BC*CC}{3 ZE} + 2 * \frac{B^3C}{27}$$

$$\text{let } YE = ZE - \frac{PE}{3ZE} \quad (3.44)$$

substitute (3.42) in (3.43) we get

$$ZE^6 + QE * ZE^3 - \frac{PE^3}{27} = 0$$

Now, solving for ZE^3 (the positive root)

$$ZE1 = ZE^3 = \frac{QE}{2} + \sqrt{\frac{QE^2}{4} + \frac{PE^3}{27}}$$

$$ZE = (ZE1)^{1/3}$$

$$YE = ZE - \frac{PE}{3ZE}$$

$$Sp = YE - \frac{BC}{3} \quad (3.45)$$

Once the value of Sp is known, the interfacial temperature can be obtained using Eq. (3.36a). However, the difficulty remains that the

procedure is based on trial and error. So, all the above calculations have to be repeated for each trial.

The solution start by assuming a value of W_{gi} , which has to be greater than W_{∞} , and assigning a value of Sp can be found from Eq. (3.45). Using the value of Sp the interfacial temperature can be evaluated using Eq. (3.38a).

$$T_{gi} = \frac{Sp * h_{fg} \mu_L}{K_L} + T_w \quad (3.46)$$

Then the corresponding saturation pressure is found from tables or by calculation as following:

first convert T_{gi} into Fahrenheit (F)

$$T_{giF} = \frac{9}{5} * T_{gi} + 32$$

to find the saturation pressure (P_{wsi}) we use the following equation given by ASHRAE

$$1.8893 \alpha^2 + 30.579 \alpha + 79.047 - T_{giF} = 0$$

$$\alpha = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

where

$$b = 30.579$$

$$a = 1.8893$$

$$c = 79.047 - T_{giF}$$

$$P_{ws} = e^{\alpha} \text{ in (in Hg)}$$

$$P_{wsi} = \frac{P_{ws}}{29.92} * 101325 \quad (\text{Pa})$$

The saturation pressure at the interface is then evaluated using the assumed W_{gi} value using Eq. (3.32)

$$P_{vi} = P_{tot} * [1 - W_{gi}] / [1 - (1 - \frac{M_v}{M_g}) W_{gi}]$$

where

$$P_{tot} = \text{atmospheric air pressure } (P_a) + \text{partial vapour pressure } (P_v)$$

In this study, the air pressure (P_a) is assumed to be equal to the standard atmospheric pressure ($P_a = 101.325 \text{ KPa}$). Now, if $P_{vi} = P_{wsi}$, then the assumed W_{gi} is correct and the corresponding interfacial temperature is taken to calculate the heat transfer rate. Otherwise, a new value for W_{gi} has to be considered and the whole procedure is repeated.

3.4 CONDENSATION ON INCLINED SURFACES

Although the main objective of this study is to evaluate condensation heat transfer for vertical glass windows and walls in buildings, inclined glass surfaces can also be used, especially as skylights on the roofs. Fortunately, the procedure for calculating condensation heat transfer for inclined surfaces is the same as that for vertical surfaces. Since, it also involves the evaluation of the interfacial

temperature of the condensate layer. However, there is a small modification in the heat transfer rate given by equation (3.33), since the angle that the surface makes with the horizontal θ has to be included as shown in equation (3.47).

$$Q = \frac{4}{3} \frac{h_{fg} \mu_L}{\left[\frac{C_p (T_{gi} - T_w)}{h_{fg} Pr_L} \right]^{\frac{1}{4}}} \left(\frac{g L^3}{4 \nu_L^2} \sin \theta \right)^{\frac{1}{4}} \quad (3.47)$$

and the heat transfer coefficient will be

$$h_m = 0.943 \left[\frac{g \rho_L (\rho_L - \rho_v) h_{fg} K_L^3}{\mu_L (T_{gi} - T_w) L} \sin \theta \right]^{\frac{1}{4}} \quad (3.48)$$

CHAPTER IV

HEAT GAIN AND LOSS ASSOCIATED WITH WINDOWS' CONDENSATION

4.1 INTRODUCTION

The selection of the most suitable air-conditioning system is in part dependent on how good is the estimate of the heat gain or loss of the space of concern. Heat gain or loss due to condensation on glazed windows under particular conditions may constitute a sizable fraction of heat gain or loss due to conduction through the building envelope. In the present study, the consideration is given to the estimate of the added heat gain or loss due to condensation. Condensation occurs on a window surface whenever its temperature is below the corresponding saturation temperature of the air. The latent heat released on the surface of the window by condensation tends to raise its temperature until there is a balance between heat production by condensation and heat loss from the window. Such process would require a time-dependent model that would realistically simulate what actually occurs. The surface temperature and the corresponding condensation rate as time progresses can accurately be determined through such model.

For external surface condensation, heat gain can be estimated by multiplying the increase in the internal surface temperature by the corresponding surface heat transfer coefficient. The rise in the internal surface temperature can be determined if the thermal

characteristics of the window and the amount of heat released by condensation on the external surface, which can be found by any of the two approaches discussed in the previous chapter, are known. However, the condensation rate and accordingly the amount of latent heat released decrease as the surface temperature increases. Therefore, the surface temperature and the condensation rate have to be correlated together with time so that they can be evaluated at any time interval until the process reaches steady state conditions.

For internal surface condensation, heat loss represents loss of moisture or latent heat from the indoor air. In other words, it is equal to the heat needed to vaporize the same amount of condensed water into the air. Heat loss can be estimated by multiplying the condensation rate by the latent heat of vaporization of water. However, indoor surface condensation is dependent on the internal surface temperature which varies with time as condensation continues, therefore, the internal surface temperature and condensation rate have also to be correlated with time, so that both can be predicted as time progresses. This has been achieved through the evaluation of the temperature gradient after condensation occurs at certain time intervals using finite difference formulation. The corresponding mass condensation rate is calculated at the beginning of each time interval and the effect of the corresponding released heat in raising the surface temperature is considered when evaluating the temperature gradient at the end of each time interval. These calculations are

carried out for each time increment until the steady state is reached or until there is no major change in the mass condensation rate and the corresponding temperature rise, when heat gain and loss due to condensation are calculated.

4.2 TEMPERATURE GRADIENT ACROSS WINDOWS IN THE ABSENCE OF CONDENSATION

Temperature gradient and the corresponding heat transfer across a window have to be known when dealing with surface condensation on windows. Heat transfer due to temperature difference across the window is an essential part in the heat balance equation. In addition, the surface temperature is a key factor in determining the occurrence of condensation on the surface. The temperature gradient across a window has one constant theoretical pattern at the steady state conditions for the assumed thermal characteristics of the window. The pattern of the temperature gradient is determined by the overall thermal resistance of the window, and the indoor and outdoor temperatures. The indoor and outdoor surface temperatures and the temperature at any point across the window can be determined by [10]:

$$T_x = T_o - \frac{r}{R} (T_o - T_i) \quad (4.1)$$

where

T_o = outdoor temperature

T_i = indoor temperature

r = summation of thermal resistance from a given point
outward

R = overall thermal resistance

The accuracy in evaluating thermal resistances in Eq. 4.1 and consequently, the accuracy of the calculated heat transfer across the window is very much dependent on the surface heat transfer coefficients. The surface heat transfer coefficient is dependent on the surface roughness, wind velocity and the type of flow in the boundary layer. The overall surface heat transfer coefficient consists of two components; the convective and the radiative components.

$$h = h_r + h_c \quad (4.2)$$

The value of the convective component (h_c) in equation (4.2) is dependent on the type of the convection process. The convection process could be natural (or free) in which the flow is caused by temperature induced density gradients within the fluid, or it could be forced in which the flow over the surface is caused by an external force such as wind or fans. In addition to the type of process, the convective heat transfer coefficients is dependent on whether the flow is turbulent or laminar within the boundary layer.

4.2.1 Surface Heat Transfer Coefficient for Free Convection

In the free convection process the flow in the boundary layer could be either laminar or turbulent. By using Rayleigh number, the type of flow can be determined [31].

$$Ra_L = \left[\frac{g \beta L^3 (T_o - T_w)}{\nu \alpha} \right] \quad (4.3)$$

where

g = gravitational constant

β = coefficient of expansion ($1/^{\circ}\text{K}$)

L = length of plate

ν = kinematic viscosity m^2/s

T_w = plate temperature

T_o = bulk temperature

all the above physical properties, which can be evaluated from any heat transfer text, are evaluated at the film temperature. The type of flow in the boundary layer can be determined using Rayleigh critical number $Ra_c = 10^9$.

when $Ra_L < 10^9$ the flow is laminar

and when $Ra_L > 10^9$ the flow is turbulent

once the type of flow is known, the appropriate formula can be used to find the surface heat transfer coefficient. Assuming a uniform surface temperature, the surface heat transfer coefficient can be found as following [19]:

$$\text{for laminar flow } \bar{h}_c = 0.59 \frac{K}{L} Ra_L^{\frac{1}{4}} \quad (4.4)$$

$$\text{for turbulent flow } \bar{h}_c = \frac{0.1 K}{L} Ra_L^{1/3} \quad (4.5)$$

where K = air thermal conductivity (W/m-C)

Recently, more accurate expressions were proposed by Churchill and Chu for predicting the average surface heat transfer coefficient [31]:
for laminar flow

$$\bar{h}_c = \frac{K}{L} \left[0.68 + \frac{0.67 Ra_L^{1/4}}{[1 + (0.492/Pr)^{9/16}]^{4/9}} \right] \quad (4.6)$$

for turbulent flow

$$\bar{h}_c = \frac{K}{L} \left[0.825 + \frac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right] \quad (4.7)$$

where

$$Pr = \frac{\nu}{\alpha} = \text{Prandtl number}$$

4.2.2 Surface Heat Transfer Coefficient for Forced Convection

In the forced convection process the flow within the boundary layer could also be either laminar or turbulent. The type of flow can be determined using the critical value of Reynold's number ($Re = 5 \times 10^5$) as following [31]:

when $Re < 5 \times 10^5$ the flow is laminar
and when $Re > 5 \times 10^5$ the flow is turbulent

The Reynold's number is given by [33]:

$$Re = \frac{V_\infty L}{\nu}$$

where

V_{∞} = upstream velocity (m/s)

L = characteristic length (m)

ν = kinematic viscosity (m^2/s)

The average heat transfer coefficients h_c for laminar flow along a flat plate are given by [31]:

$$\bar{h}_c = 0.664 \frac{K}{L} Pr^{1/3} Re_L^{1/2} \quad \text{for} \quad 0.6 < Pr < 10 \quad (4.8)$$

and

$$\bar{h}_c = 0.678 \frac{K}{L} Pr^{1/3} Re_L^{1/2} \quad \text{for} \quad Pr \rightarrow \infty \quad (4.9)$$

Turbulent boundary layer occurs when $Re > 5 \times 10^5$. However, over the entire plate a transition takes place from laminar to turbulent flow in the range of Reynold's number from 2×10^5 to 5×10^5 [31]. So, there will be a laminar flow over part of the plate and a turbulent flow over the rest. However, the main interest in the present study is to find the average heat transfer coefficient h_c over the entire plate length. Therefore, both regions should be considered, and the averaging should be found over the laminar and turbulent regions. By integrating over both regions the average heat transfer coefficient h_c is given by:

$$\bar{h}_c = \frac{K}{L} [0.036 \text{Pr}^{0.43} (\text{Re}_L^{0.8} - \text{Re}_c^{0.8}) + 0.664 \text{Pr}^{1/3} \text{Re}_c^{0.5}] \quad (4.10)$$

where

Re_c = critical Reynold's number at which transition occurs

So, for any assumed critical number the average heat transfer coefficient can be found. However, the above equation is valid only when the flow is parrallel to the plate surface.

The other component needed to evaluate the overall surface heat transfer coefficient [equation (4.2)] is the radiative component h_r . If we assume that the sky and all surrounding surfaces are at a uniform temperature equals to the ambient temperature (T_o), the radiative heat transfer coefficient can be given as:

$$h_r = \sigma F_{s-R} (T_o^2 + T_w^2) (T_o + T_w) \quad (4.11)$$

where

σ = Boltzmann constant = $5.673 \times 10^{-8} \text{ (W/m}^2\text{.K}^4\text{)}$

F_{s-R} = configuration factor, (= 1.0 for vertical surfaces)

T_o = Ambient temperature, °K

T_w = surface temperature, °K

The evaluation of h_r is easy once the surface temperature is known. However, the overall surface heat transfer coefficient has to be known in order to evaluate the surface temperature. Therefore, the surface temperature (T_w) has to be assumed when evaluating h_r .

4.2.3 Wind-dependent Convective Surface Heat Transfer Coefficient

There have been several attempts to relate wind speed and direction to convective surface heat transfer (h_c). However, none of them is perfect in estimating surface heat transfer coefficient in buildings. They are mostly based on a small scale model test conducted in laboratories. And even when the model is at reasonable dimensions compared to the actual building, it is still not perfect, since each building has a unique dimensions, geometry and exposure.

McAdams (1954) suggested a dimensional equation to find the heat transfer coefficient for a 0.5 m^2 plate:

$$h_c = 5.7 + 3.8 v \quad (4.12)$$

where

$$v = \text{wind speed (m/s)}$$

However, it has been suggested by Duffie and Beckman [34] that there is a possibility that the effect of free convection and radiation are included in equation (4.12). Later, Watnuff et al. (1977) suggested another formula:

$$h_c = 2.8 + 3.0 v \quad (4.13)$$

It was found that Eq. 4.13 shows some agreement with more accurate equation for a certain characteristic length. So, it can be assumed that Eq. 4.13 is not valid for other plate with different dimensions.

The wind direction is another wind related factor which affects

the surface heat transfer coefficient. It was considered experimentally by some investigators (11,35). In the experiment conducted by Rowley and Eckley (35), it has been found that the convective surface heat transfer coefficient does not vary with wind direction for quite a large range from the normal when the wind speed is low (7 m/s). And when the flow is parallel to the surface, the heat transfer coefficient (h_c) tends to increase. For wind speeds above 7 m/s, a slight reduction in the convective heat transfer coefficient was found, as the angle between the surface and the air stream was increased.

As mentioned above, the empirical formulas which relate the wind speed and direction to the heat transfer coefficients are mainly based on experimental test conducted on small models in laboratories. Therefore, when applied to predict surface heat transfer in building great inaccuracy will result. Recently, there has been a study by El-Diasty [31]. On a model which reasonably represent the actual building characteristics. He confirmed the linear relationship between surface heat transfer coefficient and wind speed at any wind direction. In addition, he experimentally evaluated the regression coefficient (a and b) used in the equation that relates wind speed with surface heat transfer coefficients. This equation takes the following form:

$$h_c = a + bv \quad (w/m^2 \text{ } ^\circ C) \quad (4.14)$$

The values of a and b vary with the wind direction in the following manner [31]:

$$\text{for normal wind} \quad h_c = 7.77 + 2.9 \ v \quad (4.15a)$$

$$45^\circ \text{ from normal} \quad h_c = 7.5 + 3.26 \ v \quad (4.15b)$$

$$90^\circ \text{ from normal} \quad h_c = 7.52 + 4.1 \ v \quad (4.15c)$$

$$135^\circ \text{ from normal} \quad h_c = 6.83 + 2.63 \ v \quad (4.15d)$$

$$180^\circ \text{ from normal} \quad h_c = 6.27 + 2.65 \ v \quad (4.15e)$$

Surface heat transfer coefficients evaluated by the above equations seem to disagree with ASHRAE, if they are used to evaluate the convective part of the surface heat transfer coefficient. However, when used to evaluate the overall surface heat transfer coefficient a good agreement is found. For example, at wind speed of 3.4 m/s. ASHRAE gives a value of 22.7 w/m²-C for the overall surface heat transfer coefficient, and Eq. 4.15C gives a value of 21.5 w/m²-C for the convective part only of the surface heat transfer coefficient, therefore, there is a possibility that the effect of radiation was included when calculating the regression coefficients. Consequently, the effect of radiation at those particular conditions should be excluded, otherwise, Eqs. 4.15a to 4.15e may be used to evaluate the overall surface heat transfer coefficients rather than the convective part only. The above Eqs. 4.15a to 4.15e will be used in this study to evaluate the overall surface heat transfer coefficient, because the regression coefficients were evaluated on more practical model in terms of geometry, and the wind direction was considered.

4.3 TEMPERATURE GRADIENT ACROSS WINDOWS IN THE PRESENCE OF CONDENSATION

The temperature gradient across a window depends on many variables related to the outdoor and indoor temperatures, and the overall thermal resistance of the window. The temperature gradient across a window can be easily evaluated under steady state conditions. However, when condensation occurs, the pattern of temperature gradient will change with time until the condensation process reaches the steady state conditions and a new pattern of temperature gradient is formed.

Numerical methods are used extensively, in practical applications, to determine the temperature gradient and heat flow in solids. A commonly used numerical scheme is the finite-difference method. In this approach, the partial differential equations of heat conduction is approximated by a set of algebraic equations for temperatures at a number of nodal points over the region. Finite difference formulation can be a unidimensional or multidimensional either in the steady state or the unsteady state of conduction heat transfer.

The problem of heat transfer in the presence of surface condensation is considered in this study to be a unidimensional problem. This seems most appropriate, since glass thickness is relatively, very small compared to the other two dimensions. This problem is regarded as a time-dependent conduction, since condensation process is a function of time until steady state stage is reached. The time-dependent

uni-dimensional heat transfer problems can be formulated by the finite difference method in terms of forward time difference (explicit), or backward time difference (implicit). In the explicit method the nodal temperature at time (t) is expressed in terms of previously known nodal temperatures at time $(t-\Delta t)$. However, in the implicit method the nodal temperature at time (t) is expressed in terms of unknown nodal temperature distribution at the same time (t) .

Energy balance equations expressed by the explicit method of the finite-difference scheme are much easier to solve than those expressed in terms of the implicit method. However, the explicit method has a restriction on the maximum size of the time step because of stability consideration. So, if the time interval needed to maintain stability is small compared to the period for which computation are to be performed, then enormous number of steps are required. Consequently, very large computer storage capacity is needed. the implicit method has the advantage of being stable for any time interval (Δt) . However, to determine the nodal temperatures, a simultaneous solution of all the equations for the nodes at each time step is required. In addition, larger truncation error is encountered at every step and accuracy suffers slightly [36]. This accuracy depends on the time interval Δt , the smaller the time interval, the more accurate the computations [36], thus, to obtain a reasonably accurate results, reasonable Δt should be considered.

Another more efficient and accurate form of the implicit scheme

has been developed by Crank and Nicolson [31]. Its accuracy is based on the use of the arithmetic mean value of the derivatives at the beginning and the end of the time interval [37] this form of the implicit scheme also consists of a number of finite-difference equations which have to be solved simultaneously. Crank-Nicolson method gives more accurate results (near the actual value), if the time interval is small. If the time interval is too large numerical oscillations will occur in the Crank-Nicolson method, but they never become unstable. These oscillations can become large enough to make the solution inaccurate [37].

The advantage of the fully implicit method over the Crank-Nicolson method is that numerical oscillation will never happen whatever the size of the time interval. This means that the fully implicit method is more accurate than the explicit method and the Crank-Nicolson method if very large time steps are being used [37]. So, a sort of compromise should be done when choosing the method to be used. Either using reasonable time intervals and sacrificing the accuracy slightly by choosing the fully implicit form, or using small time intervals and get more accurate solution by using Crank-Nicolson method. For the present study the fully implicit form will be used, since reasonable (not too large) time interval will be used, which means reasonable truncation error. In addition, there will be more flexibility in choosing large time intervals when required.

In order to apply the finite difference analysis to the present

condensation heat transfer problem the following assumptions should be made:

- 1) Outdoor and indoor temperatures and relative humidities are assumed to be constant during condensation, and sudden changes are unlikely to occur (steady-state conditions).
- 2) The surface temperature before condensation occurs is assumed to be the reference temperature at which condensation starts.
- 3) Both the length and the width of the window are infinite, so this problem can be considered a unidimensional heat transfer problem.
- 4) There is no change in the surface heat transfer coefficients when condensation occurs.
- 5) The thermal resistance of the water film has no effect on heat transfer due to temperature difference.

4.3.1 Mathematical Formulation of the Uni-dimensional Unsteady Conduction Heat Transfer Problem

The differential and the finite-difference formulations are frequently used for solving conduction heat transfer problems. Whereas the differential formulation approach provides the basis for establishing exact analytical solutions to many problems. The finite-difference formulation method lends itself to the analysis of the more complex problems often encountered in practice for which exact

solutions are difficult. For a unidimensional unsteady state conduction heat transfer, the differential formulation using the first law of thermodynamics can be written as [39]:

$$dq_x = dq_{x+\Delta x} + \frac{\partial E}{\partial t} \quad (4.16)$$

utilizing the definition of the partial derivative Eq. 4.16 becomes [39]:

$$dq_x = dq_x + \frac{\partial (dq_x)}{\partial x} dx + \frac{\partial E}{\partial t} \quad (4.17)$$

Introducing the Fourier law of conduction, we obtain

or

$$\frac{\partial E}{\partial t} = \frac{\partial}{\partial x} (K A_x \frac{\partial T}{\partial x}) dx$$
$$dv \frac{\partial}{\partial t} (\rho C_v T) = \frac{\partial}{\partial x} (K A_x \frac{\partial T}{\partial x}) dx \quad (4.18)$$

dividing by dv Eq. 4.18 becomes:

$$\frac{\partial}{\partial t} (\rho C_v T) = \frac{\partial}{\partial x} (K \frac{\partial T}{\partial x}) \quad (4.19)$$

The differential formulation is completed by writing the boundary conditions. The number of boundary conditions is equal to the highest-order derivative in an ordinary differential equation. Hence, two boundary conditions must be specified in order to solve unsteady unidimensional conduction heat transfer problems.

The finite-difference formulation for a unidimensional heat transfer problem requires that the conducting body to be subdivided so that equally spaced nodes at distance Δx apart are obtained. Once the finite-difference grid is established, the numerical finite-difference formulation is developed by merely performing energy balances of each interior and exterior node.

For the **Interior Nodes** the energy balance equation can be written as [31]:

$$\Delta q_x = \Delta q_{x+\Delta x} + \frac{\Delta E}{\Delta t} \quad (4.20)$$

using simple difference approximation for the Fourier law of conduction Eq. 4.20 becomes:

$$-K A \frac{(T_m^\tau - T_{m-1}^\tau)}{\Delta x} = -K A \frac{(T_{m+1}^\tau - T_m^\tau)}{\Delta x} + \rho A \Delta x C_v \frac{(T_m^{\tau+1} - T_m^\tau)}{\Delta t} \quad (4.21)$$

dividing by A and multiplying by Δx Eq. 4.21 becomes:

$$-\frac{K}{\Delta x} (T_m^\tau - T_{m-1}^\tau) = -K (T_{m+1}^\tau - T_m^\tau) + \rho \Delta x^2 C_v \frac{(T_m^{\tau+1} - T_m^\tau)}{\Delta t} \quad (4.22)$$

where

m = the interior node number

For the **Exterior Nodes** the heat balance equation can be generally written as [31]:

$$\Delta q_x = \Delta q_s + \frac{\Delta E}{\Delta t} \quad (4.23)$$

utilizing Fourier law of conduction. Eq. 4.23 becomes either:

$$-K A \frac{(T_m^\tau - T_{m-1}^\tau)}{\Delta x} = A q_s + \rho A \frac{\Delta x}{2} C_v \frac{(T_m^{\tau+1} - T_m^\tau)}{\Delta t} \quad (4.23a)$$

dividing by A Eq. (4.23a) becomes

$$-K \frac{(T_m^\tau - T_{m-1}^\tau)}{\Delta x} = q_s + \rho \frac{\Delta x}{2} C_v \frac{(T_m^{\tau+1} - T_m^\tau)}{\Delta t} \quad (4.24)$$

or

$$-K \frac{(T_{m+1}^\tau - T_m^\tau)}{\Delta x} = q_s + \rho \frac{\Delta x}{2} C_v \frac{(T_m^{\tau+1} - T_m^\tau)}{\Delta t} \quad (4.25)$$

where

q_s = total heat transfer to or from a unit area

τ = time increment

m = exterior node number

The use of either Eq. 4.24 or 4.25 depends on the node number, so for the exterior node numbered $m > 1$ Eq. 4.24 is used, and for the node numbered 1 Eq. 4.25 is used. Also, it should be noted that the direction of flow is taken into consideration by the negative sign associated with the first term in both equations.

4.3.2 Finite-difference Formulation for Heat Transfer Across Single Glazed Windows when Condensation Occurs

In order to efficiently utilize the finite-difference formulation, the glass pane is to be subdivided so that equally spaced nodes at a distance Δx apart are obtained as shown in Fig. 4-1.

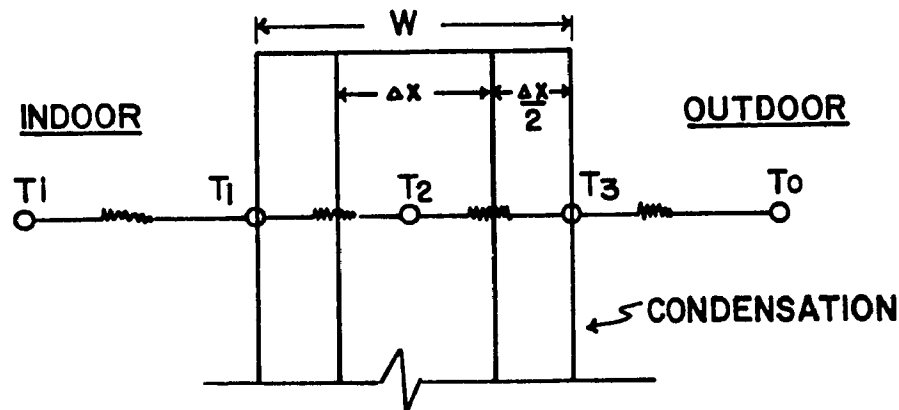


FIG. 4-1: Finite-difference grid for a single glass pane

As an approximation, the temperature of each node is taken to represent the temperature of the corresponding segment of the pane. Now, the energy balance equation for each node can be written in the implicit form.

4.3.2.1 Energy Balance Equations when Outdoor Condensation Occurs

The energy balance equations will be written and solved for constant indoor and outdoor conditions. But, in order to illustrate the effect of variable indoor and outdoor temperatures on the form of

energy balance equations, they will also be written for variable conditions when outdoor condensation occurs on single glazed windows.

I) At constant indoor and outdoor conditions

Node (1)

$$\rho \frac{C_v \Delta x}{2 \Delta t} (T_1^{\tau+1} - T_1^{\tau}) = \frac{K}{\Delta x} (T_2^{\tau+1} - T_1^{\tau+1}) - h_i (T_1^{\tau+1} - T_i)$$

arranging terms

$$(1+2F_0+2r) T_1^{\tau+1} - 2 F_0 T_2^{\tau+1} = T_1^{\tau} + 2r T_i \quad (4.26)$$

Node (2)

$$\rho \frac{C_v \Delta x}{\Delta t} (T_2^{\tau+1} - T_2^{\tau}) = \frac{K}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1}) - \frac{K}{\Delta x} (T_2^{\tau+1} - T_1^{\tau+1})$$

arranging term

$$- F_0 T_1^{\tau+1} + (1+2F_0) T_2^{\tau+1} - F_0 T_3^{\tau+1} = T_2^{\tau} \quad (4.27)$$

Node (3)

$$\rho \frac{C_v \Delta x}{2 \Delta t} (T_3^{\tau+1} - T_3^{\tau}) = \dot{m} h_{fg} - \frac{K}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1}) + h_o (T_o - T_3^{\tau+1})$$

arranging terms

$$- 2F_0 T_2^{\tau+1} + (1+2F_0+2C_0) T_3^{\tau+1} = T_3^{\tau} + 2C_0 T_o + CA * \dot{m} \quad (4.28)$$

where

$$F_0 = \frac{\Delta t K}{\rho C_v \Delta x^2}$$

$$r = \frac{\Delta t h_i}{\rho C_v \Delta x}$$

$$CO = \frac{ho \Delta t}{\rho Cv \Delta x}$$

$$CA = \frac{2 \Delta t hfg}{\rho Cv \Delta x}$$

ρ = glass density, Kg/m³

K = glass thermal conductivity, W/m-C

Cv = glass specific heat, KJ/Kg-C

Δt = time interval, sec.

τ = time step

ho = outdoor surface heat transfer coefficient, W/m²-C

hi = indoor surface heat transfer coefficient, W/m²-C

Ti = indoor temperature

To = outdoor temperature

\dot{m} = mass condensation rate per unit area

II) At variable indoor and outdoor conditions

In this case the indoor and outdoor temperatures in the heat balance equations are expressed as a function of time. The above equations can be used by modifying T_i to become $T_i^{\tau+1}$ and To to become $T_o^{\tau+1}$.

Node (1)

$$(1+2FO+2r)T_1^{\tau+1} - 2FO T_2^{\tau+1} = T_1^{\tau} + 2r T_i^{\tau+1} \quad (4.29)$$

Node (2)

$$- FO T_1^{\tau+1} + (1+2FO) T_2^{\tau+1} - FO T_3^{\tau+1} = T_2^{\tau} \quad (4.30)$$

Node (3)

$$- 2 FO T_2^{\tau+1} + (1+2FO+2CO)T_3^{\tau+1} = T_3^{\tau} + 2CO T_o^{\tau+1} + CA * \dot{m} \quad (4.31)$$

4.3.2.2 Energy Balance Equations when Indoor Condensation Occurs at Constant Indoor and Outdoor Conditions

The finite-difference grid shown in Fig. 4-2 will be used to write the heat balance equations when indoor condensation occurs.

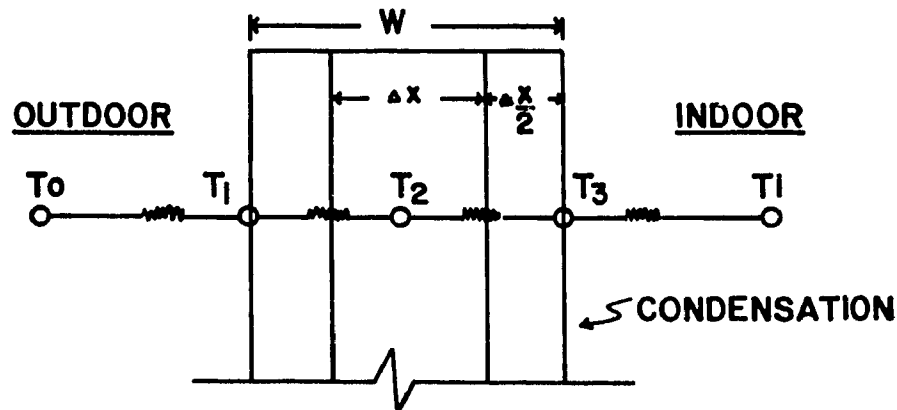


FIG. 4-2: Finite-difference grid for a single glass pane

Node (1)

$$\frac{\rho C_v \Delta x}{2 \Delta t} (T_1^{\tau+1} - T_1^{\tau}) = \frac{K}{\Delta x} (T_2^{\tau+1} - T_1^{\tau+1}) - h_o (T_1^{\tau+1} - T_o)$$

arranging terms

$$(1+2F_o+2r) T_1^{\tau+1} - 2F_o T_2^{\tau+1} = T_1^{\tau} + 2r_o T_o \quad (4.32)$$

Node (2)

$$\frac{\rho C_v \Delta x}{\Delta t} (T_2^{\tau+1} - T_2^{\tau}) = \frac{K}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1}) - \frac{K}{\Delta x} (T_2^{\tau+1} - T_1^{\tau+1})$$

arranging terms

$$- F_o T_1^{\tau+1} + (1+2F_o) T_2^{\tau+1} - F_o T_3^{\tau+1} = T_2^{\tau} \quad (4.33)$$

Node (3)

$$\frac{\rho C_v \Delta x}{2 \Delta t} (T_3^{\tau+1} + T_3^{\tau}) = \dot{m} h_{fg} - \frac{K}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1}) + h_i (T_i - T_3^{\tau+1})$$

arranging terms

$$- 2FO T_2^{\tau+1} + (1+2FO+2Ci) T_3^{\tau+1} = T_3^{\tau} + 2Ci T_i + CA * \dot{m} \quad (4.34)$$

where

$$Fo = \frac{\Delta t h_o}{\rho C_v \Delta x}$$

$$Ci = \frac{\Delta t h_i}{\rho C_v \Delta x}$$

4.3.3 Finite-difference Formulation for Heat Transfer Across Double Glazed Windows when Condensation Occurs

The risk of outdoor condensation on double glazed windows in the summer is very low, because there is usually not enough temperature difference across the window to make the outer surface temperature below the dew point temperature of the outdoor air. Even when outdoor condensation occurs (at very high outdoor relative humidity), the heat gain due to condensation through double glazed windows will be negligible, because of their relatively higher thermal resistance. In addition, in hot humid climates, single glazed windows are commonly used while double glazed windows are rarely used. Therefore, outdoor condensation on double glazed windows will not be considered in this study.

Indoor surface condensation on double glazed windows is relatively more common in winter, than outdoor condensation in summer. Because there is usually enough temperature difference across the window to make the indoor surface temperature below the dew point of the indoor air, especially if the indoor relative humidity is high. In addition, the heat released on the inner surface due to indoor condensation will be directly considered as heat loss, so the thermal resistance has no effect on heat loss once condensation occurred. Therefore, indoor surface condensation on double glazed windows will be considered as a means of heat loss from buildings.

Finite-difference formulation for double glazed windows requires special attention, because of the presence of air gap through which heat will be conducted. The air space and glass pane thicknesses should be chosen so that equally spaced nodes can be obtained, and the surface temperatures are represented by the extreme nodes as shown in Fig. 4-3. Therefore, the air space thickness will always be taken as twice the glass thickness, and Δx will be evaluated from Eq. 4.35.

$$\Delta x = \frac{WAS + 2W}{6} \quad (4.35)$$

where

WAS = air space thickness

W = glass thickness

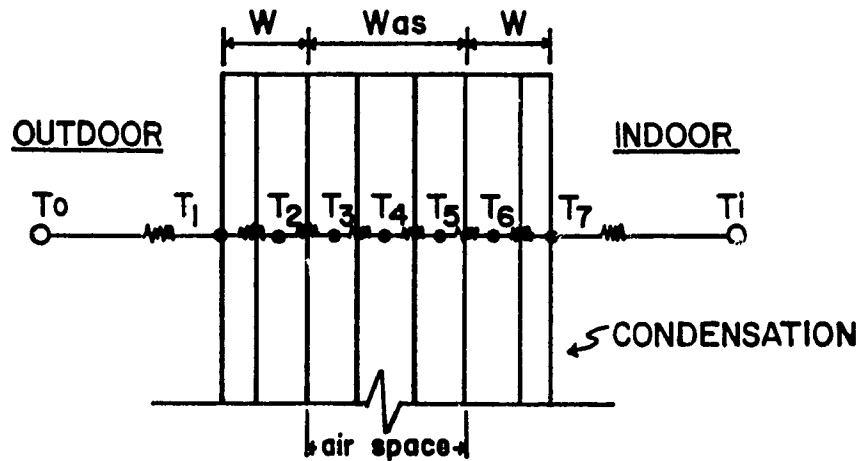


FIG. 4-3: Finite-difference grid for a double glass window

In order to be able to write the heat balance equation for each node the following assumptions will be made:

- 1) heat transfer between all nodes (including air space) is by pure conduction, so that finite-difference can be formulated.
- 2) heat transfer by radiation within the air space is substituted by assuming higher air thermal conductivity.

For the present study the following assumptions are made regarding the radiation component:

- a) The mean air temperature within the space is assumed to be equal to zero centigrade ($\bar{T} = 0^\circ\text{C} = 273^\circ\text{K}$). The radiative heat transfer coefficient can be approximated as following [34]:

$$h_r = (4\sigma \bar{T}^3) / \left(\frac{1-\epsilon_1}{\epsilon_1} + \frac{1}{F_{12}} + \frac{(1-\epsilon_2) A_1}{\epsilon_2 A_2} \right)$$

where A = Area

ϵ = surface emittance = 0.82 for glass

F_{12} = configuration factor = 1 for vertical surfaces

\bar{T} = mean air space temperature in Kelvin

$$h_r = 4 \sigma \bar{T}^3 / 1.4 = 3.3 \text{ w/m}^2\text{-C}$$

- b) The air sapce is at 0°C and has a thermal conductivity of 0.025 w/m-C , and assuming a 9 mm air space thickness, the heat transfer coefficient = $3 \text{ w/m}^2\text{-C}$. To account for $h_r \approx 3.3 \text{ w/m}^2\text{-C}$, the thermal conductivity of the air will be assumed higher to get a total heat transfer coefficient of about $\approx 6 \text{ w/m}^2\text{-C}$ within the air space. The air thermal conductivity will then be taken = 0.054 w/m-C .
3. Convective heat transfer within the air space of a sealed double glazed window is negligible, since for the air-surface temperature difference range in interest, the Rayleigh number (Eq. 4.3) is much less than 2×10^3 .
 4. No solar radiation, since condensation is unlikely to occur when the window surface is heated by solar radiation.
 5. Constant indoor and outdoor conditions.

The same form of heat balance equations, used for single glazed windows, will be applicable for double glazed windows. However, the number of nodes are more for the double glazed window, consequently, more equations have to be solved simultaneously. The energy balance equations when indoor condensation occurs on a double glazed window are:

Node (1)

$$(1+2F_0+2r_0) T_1^{\tau+1} - 2F_0 T_2^{\tau+1} = T_1^{\tau} + 2r_0 T_o \quad (4.36)$$

Node (2)

$$\frac{\rho C_v \Delta x}{\Delta t} (T_2^{\tau+1} - T_2^{\tau}) = \frac{K_{AGL}}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1}) - \frac{K}{\Delta x} (T_2^{\tau+1} - T_1^{\tau+1})$$

arranging terms

$$- F_0 T_1^{\tau+1} + (1+F_0+F_0A) T_2^{\tau+1} - F_0A T_3^{\tau+1} = T_2^{\tau} \quad (4.37)$$

Node 3

$$\frac{\rho_a C_p \Delta x}{\Delta t} (T_3^{\tau+1} - T_3^{\tau}) = \frac{K_a}{\Delta x} (T_4^{\tau+1} - T_3^{\tau+1}) - \frac{K_{AGL}}{\Delta x} (T_3^{\tau+1} - T_2^{\tau+1})$$

arranging terms

$$- FG T_2^{\tau+1} + (1+FG+FA) T_3^{\tau+1} - FA T_4^{\tau+1} = T_3^{\tau} \quad (3.38)$$

Node 4

$$\frac{\rho_a C_p \Delta x}{\Delta t} (T_4^{\tau+1} - T_4^{\tau}) = \frac{K_a}{\Delta x} (T_5^{\tau+1} - T_4^{\tau+1}) - \frac{K_a}{\Delta x} (T_4^{\tau+1} - T_3^{\tau+1})$$

arranging terms

$$- FA T_3^{\tau+1} + (1 + 2FA) T_4^{\tau+1} - FA T_5^{\tau+1} = T_4^{\tau} \quad (3.39)$$

Node (5)

$$\frac{\rho_a C_p \Delta x}{\Delta t} (T_5^{\tau+1} - T_5^{\tau}) = \frac{KAGL}{\Delta x} (T_6^{\tau+1} - T_5^{\tau+1}) - \frac{K_a}{\Delta x} (T_5^{\tau+1} - T_4^{\tau+1})$$

arranging terms

$$- FA T_4^{\tau+1} + (1+FG+FA) T_5^{\tau+1} - FG T_6^{\tau+1} = T_5^{\tau} \quad (4.40)$$

Node (6)

$$\frac{\rho C_v \Delta x}{\Delta t} (T_6^{\tau+1} - T_6^{\tau}) = \frac{K}{\Delta x} (T_7^{\tau+1} - T_6^{\tau+1}) - \frac{KAGL}{\Delta x} (T_6^{\tau+1} - T_5^{\tau+1})$$

arranging terms

$$- FOA T_5^{\tau+1} + (1+FO+FOA) T_6^{\tau+1} - FO T_7^{\tau+1} = T_6^{\tau} \quad (4.41)$$

Node (7)

$$\frac{\rho C_v \Delta x}{2 \Delta t} (T_7^{\tau+1} - T_7^{\tau}) = \dot{m} hfg - \frac{K}{\Delta x} (T_7^{\tau+1} - T_6^{\tau+1}) + h_o (T_o - T_7^{\tau+1})$$

arranging terms

$$- 2FO T_6^{\tau+1} + (1+2 FO+2CO) T_7^{\tau+1} = T_7^{\tau} + 2CO T_i + CA * \dot{m} \quad (4.42)$$

where

$$KAGL = \text{combined air and glass thermal conductivity} = \frac{(2K_a * K)}{K_a + K}$$

K_a = air thermal conductivity

$$FOA = \frac{\Delta t KAGL}{\rho C_v \Delta x^2}$$

$$FG = \frac{KAGL \Delta t}{\rho_a C_p \Delta x^2}$$

$$FA = \frac{K_a \Delta T}{\rho_a C_p \Delta x^2}$$

ρ_a = air density

C_p = air specific heat

4.3.4 Solving The Heat Balance Equations

The nodal heat balance equations, which have been written for double and single glazed windows using the finite-difference implicit form, have to be solved simultaneously to evaluate the nodal temperatures at the end for each time interval. The tridiagonal matrix algorithm has been used in this study to simultaneously solve the heat balance equations. However, to solve these equations, the mass condensation rate has to be calculated at the beginning of each time interval. The number of time intervals could reach hundreds (depending on the conditions and the chosen time interval), therefore it is impractical and almost impossible to solve these heat balance equations manually. A computer program written in Quick Basic has been developed to solve for the mass condensation rate and the nodal temperatures at any time interval, and to predict the corresponding heat gain or loss when the steady state stage is reached. Seven subprograms have been written (Appendix A) to solve for heat gain or loss due to condensation on double and single glazed windows using both approaches (discussed in Chapter 3) to predict the mass condensation rate. The application of each subprogram can be summarized as follows:

Sub-Program 1: Predicts heat gain due to outdoor condensation on single glazed windows under assumed variable indoor and outdoor conditions. The mass condensation rate is predicted by the first approach.

Sub-Program 2: Predicts heat gain due to outdoor condensation on single glazed windows under constant indoor and outdoor conditions. The mass condensation rate is predicted by the first approach.

Sub-Program 3: Predicts heat loss due to indoor condensation on single glazed windows under constant indoor and outdoor conditions. The mass condensation is predicted by the first approach.

Sub-Program 4: Predicts heat gain due to outdoor condensation on single glazed windows under constant indoor and outdoor conditions. The mass condensation rate is predicted by the second approach.

Sub-Program 5: Predicts heat loss due to indoor condensation on single glazed windows under constant indoor and outdoor conditions. The mass condensation rate is predicted by the second approach.

Sub-Program 6: Predicts heat loss due to indoor condensation on double glazed windows under constant indoor and outdoor conditions. The mass condensation rate is predicted by the first approach.

Sub-Program 7: Predicts heat loss due to indoor condensation on double glazed windows under constant indoor and outdoor conditions. the mass condensation rate is predicted by the second approach.

The procedures used in the program to predict heat gain or loss, can be well explained through the diagrammatic sequence shown in

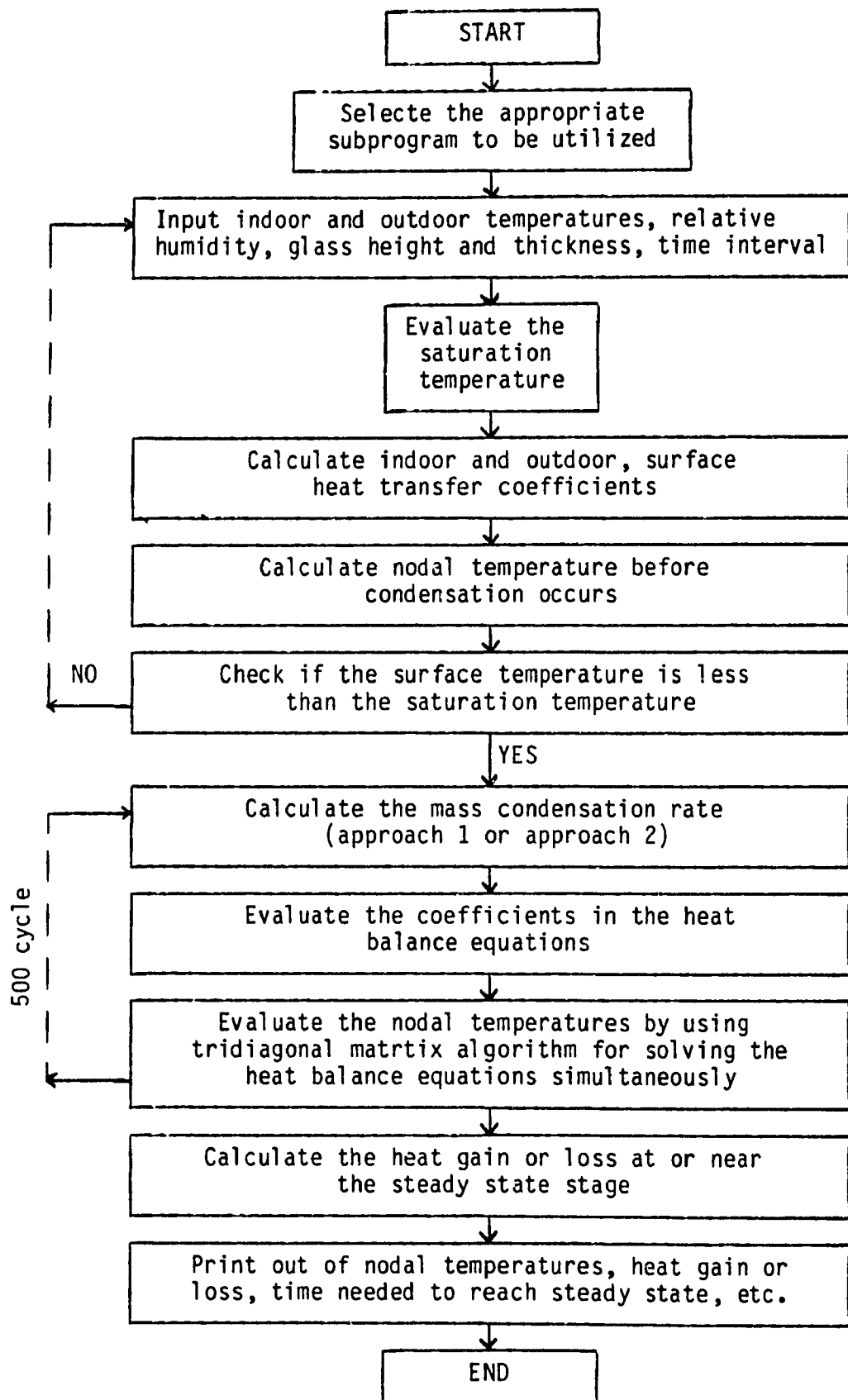


FIG. 4-4: General Flow Chart for the Program

Fig. 4-4. However, two different approaches have been used to predict the mass condensation rate (\dot{m}). The first approach was utilized by Davies [16], he assumed steady state conditions and graphically solved for the condensation rate as explained in Chapter 3. However, since the problem is considered time-dependent, a direct numerical solution for the mass condensation rate (equation 3.17) is possible by evaluating the saturation pressure (P_C) at the window surface temperature at the beginning of each time interval. The sequence of calculating the instantaneous mass condensation rates in the program is shown in Fig. 4-5. Due to computer limitation related to the BASIC language, only 500 time intervals were considered in the program. However, if the time increment (ΔT) is large enough (between 40-60 seconds), the total time will be enough for the process to reach the steady state stage.

The first step in evaluating the mass condensation rate by the second approach is to predict the interfacial temperature (t_{gi}), then Eq. 3.31 can be used to evaluate mass condensation rate. The evaluation of the interfacial temperature is done by trial and error through a subroutine as shown by the flow chart in Fig. 4-6. This process of interfacial temperature and mass condensation rate is done at the beginning of each time interval. Once the mass condensation rate is obtained by any of the two approaches, the process of heat transfer through various types of windows can be carried out normally, and the resulting heat gain or loss can be calculated.

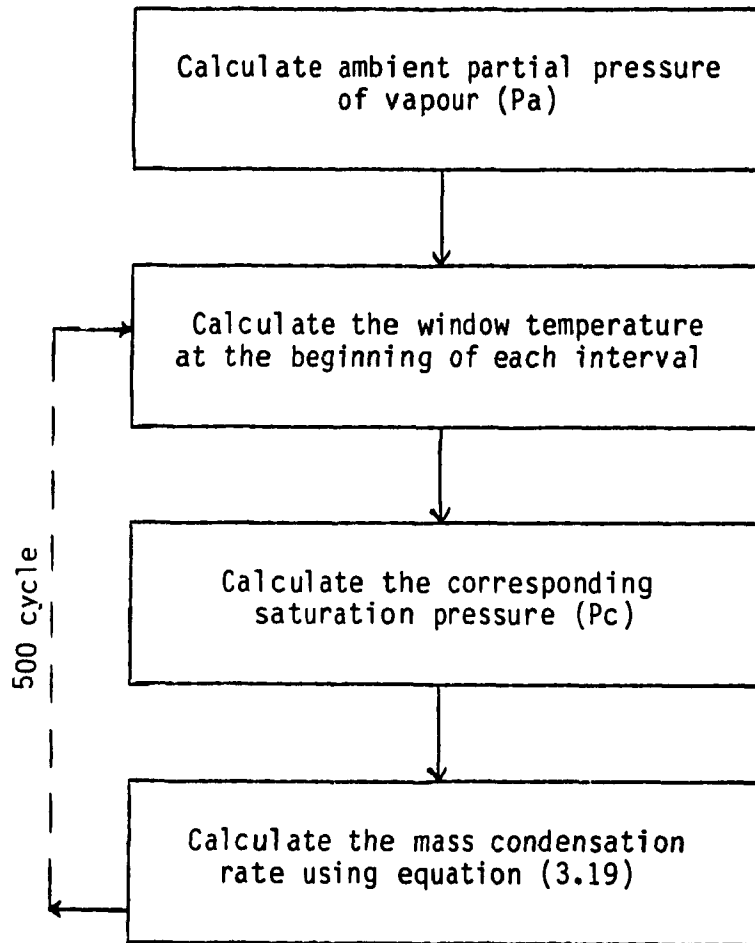


FIG. 4-5: Calculating the mass condensation rate by the first approach

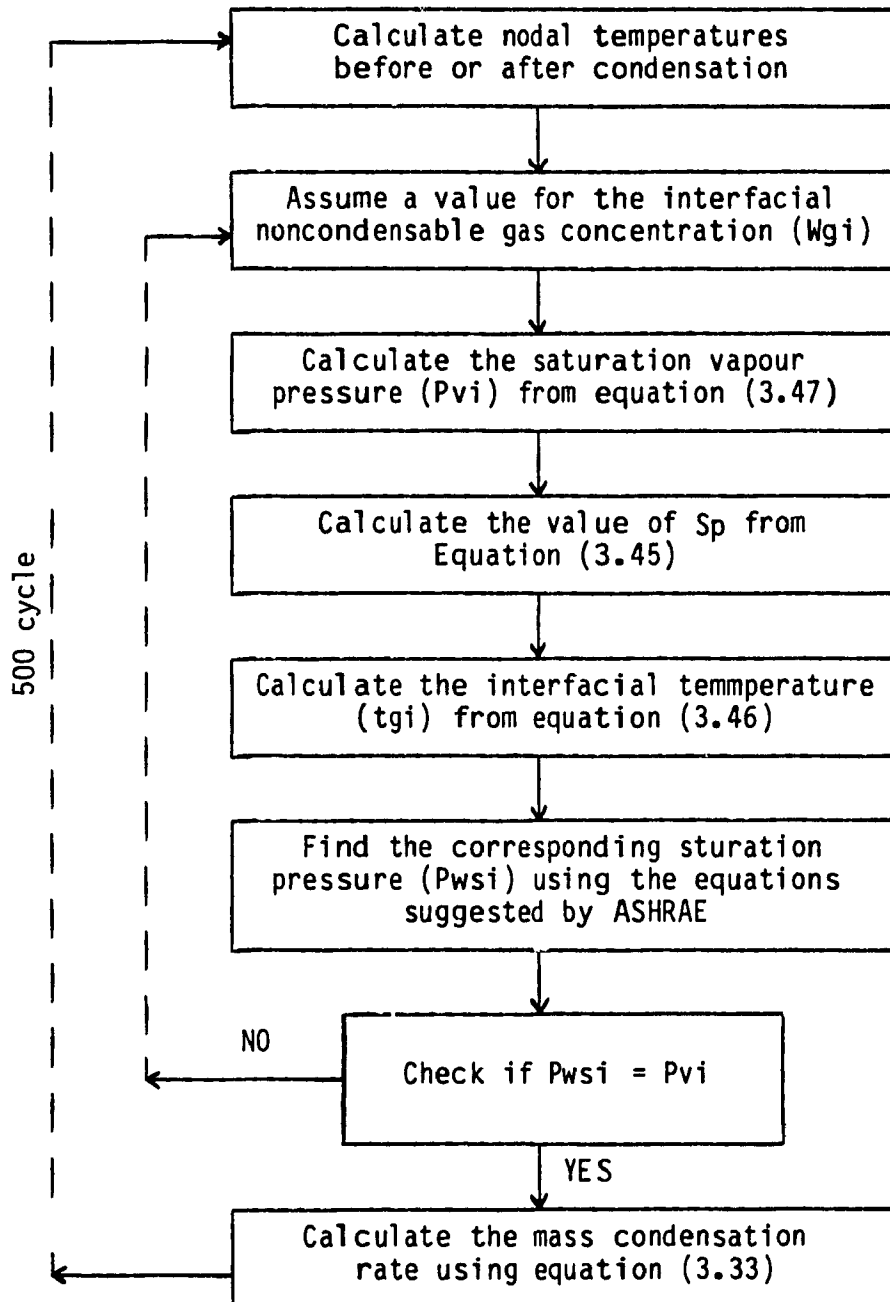


FIG. 4-6: Calculating the mass condensation rate by the second approach

4.4 HEAT GAIN ASSOCIATED WITH OUTDOOR SURFACE CONDENSATION

Heat gain due to atmospheric vapour condensation on glass windows surfaces is equal to the additional heat removed from the inner surface to the conditioned space. When outdoor condensation occurs indoor surface temperature starts to rise gradually with time until it becomes stable, when the process reaches the steady state stage. The additional heat gain can be calculated by multiplying the rise in the inner surface temperature at the steady state stage $(\Delta T)_{ss}$ by the inner surface heat transfer coefficient (h_i) as given by equation (4.43).

$$Q_{\text{gain}} = h_i (\Delta T)_{ss} \quad \text{w/m}^2 \quad (4.43)$$

outdoor surface condensation occurs when the outdoor relative humidity is reasonably high and since, high relative humidity is associated with a low wind speed, the effect of wind on the surface heat transfer coefficient will be assumed negligible, and the process is considered free convection when dealing with outdoor surface condensation. However, the effect of wind speed will be considered when dealing with indoor surface condensation.

4.4.1 Using the First Approach

Sub-program 2 can be used to predict heat gain rate due to condensation on single glazed windows at any given indoor and outdoor conditions. The nodal temperatures, the mass condensation rate at the beginning of each time interval and the other information pertaining

to the output of this subprogram are shown in output sample in Appendix B-1. As an example, the variation of the interval surface temperature (T_1) with time at different indoor and outdoor conditions is shown graphically in Figs. 4-7 and 4-8. These graphs represent the behavior of a single glazed window of 10 mm thickness ($M=10\text{mm}$), and of a height of one meter ($L = 1\text{m}$). The time interval (ΔT) used for solving the corresponding heat balance equations, is taken to be 60 seconds. It can be seen from Fig. 4-7 that the lower the indoor temperature the higher the temperature rise of the window internal surface temperature. The effect of the relative humidity on the surface temperature is shown in Fig. 4-8, where it can be seen that the higher the outdoor relative humidity the higher the temperature rise.

Knowing the temperature rise of the inner window surface at the steady state stage $(\Delta T)_{ss}$, and the surface heat transfer coefficient (h_i), which can be evaluated by either Eq. 4.4 or 4.5, the heat gain due to outdoor surface condensation can be evaluated using Eq. 4.43. Heat gain rates, estimated by the program for different outdoor conditions are shown in Fig. 4-9. These curves were obtained at different outdoor relative humidity, for a single glazed window at an indoor temperature (t_i) equals to 21°C . So, if the indoor and outdoor temperature and the outdoor relative humidity are known, the heat gain rate can be read directly from the appropriate curve for this particular window. For example, for a relative humidity of 90%, indoor temperature of 21°C , and outdoor temperature of 36°C , the heat gain

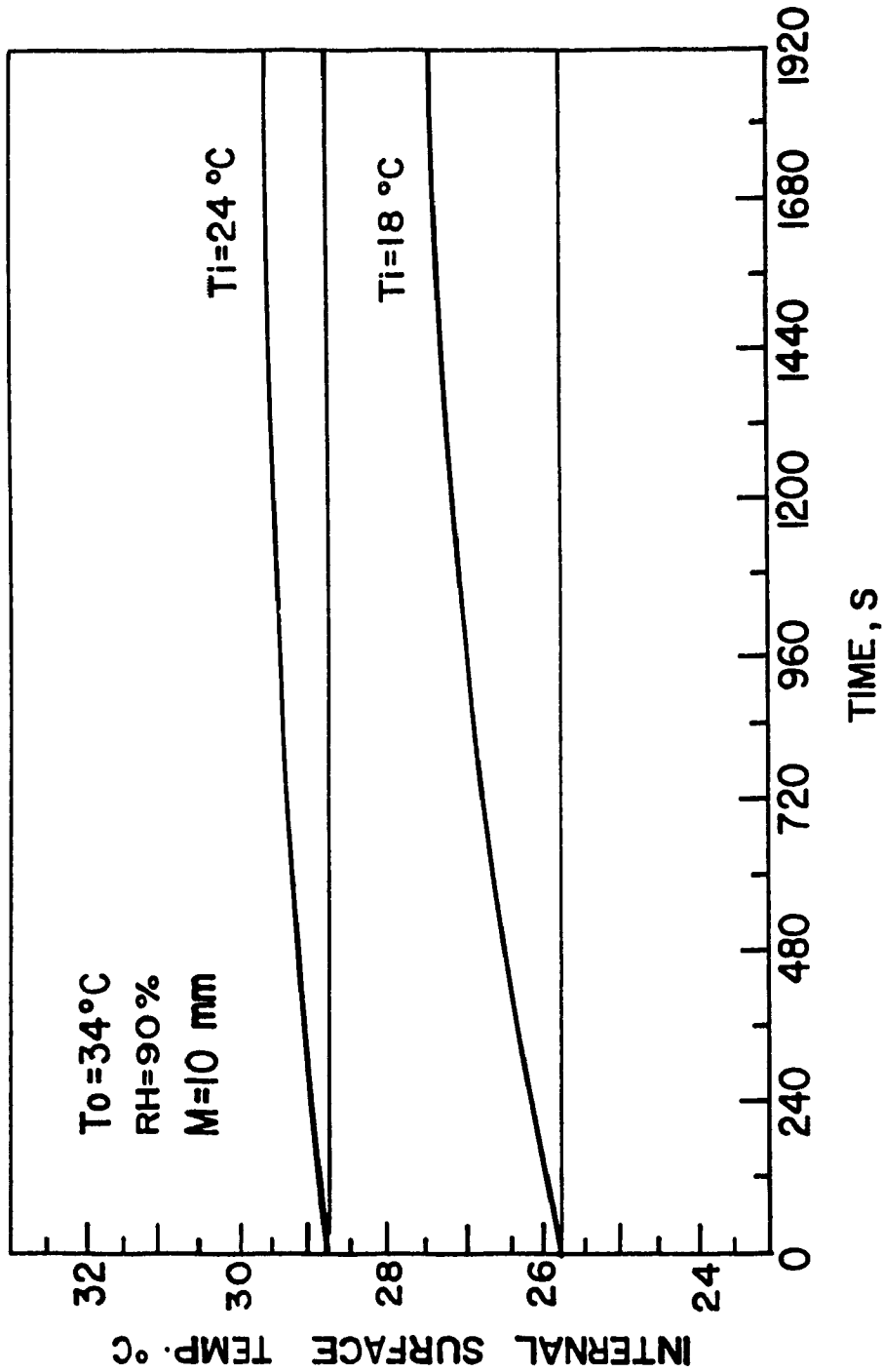


FIG. 4-7: Internal surface temperature variation with time as outdoor condensation occurs on a single glazed window at different indoor temperatures, using the first approach

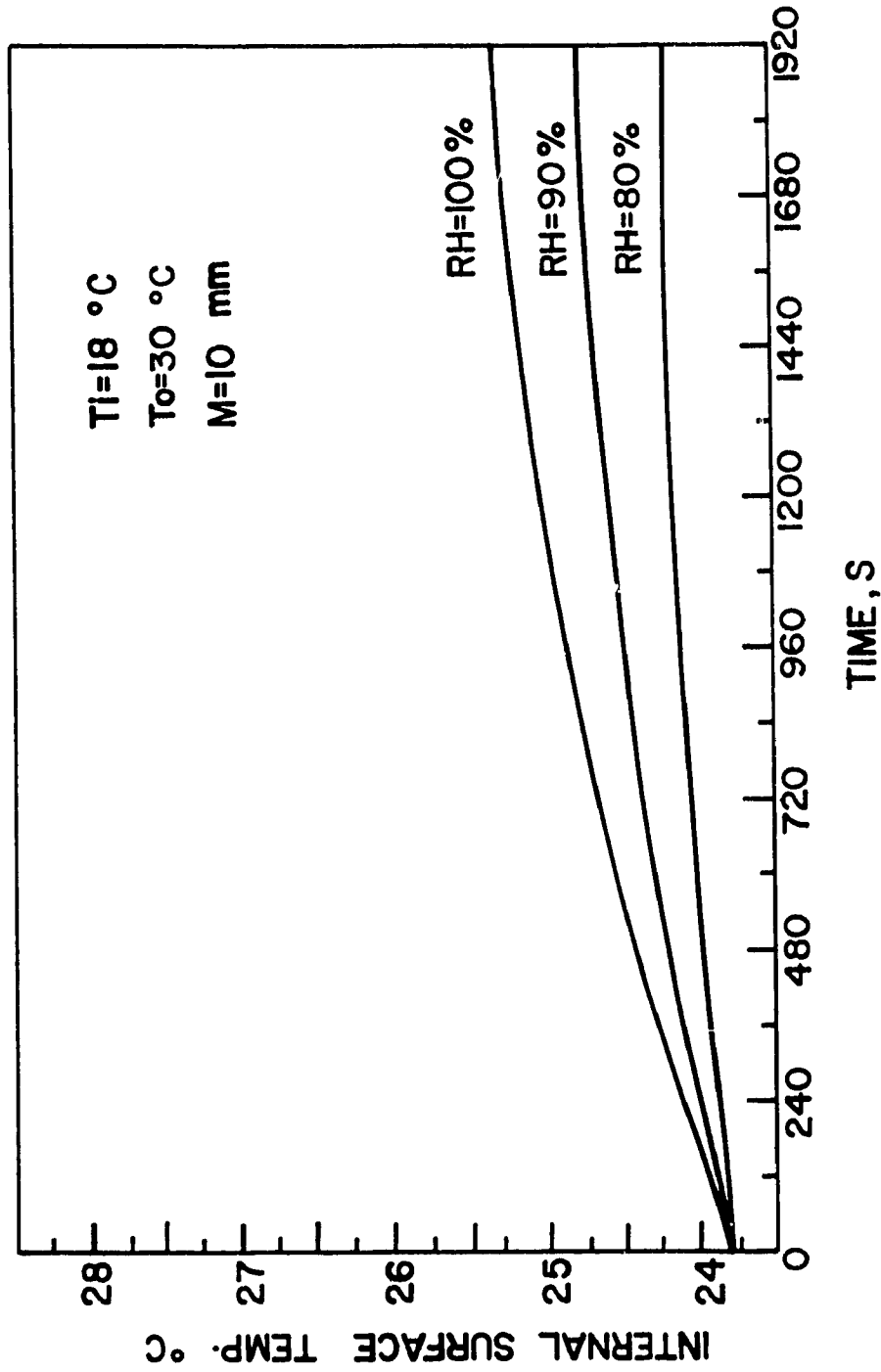


FIG. 4-8: Internal surface temperature variation with time as outdoor condensation occurs on a single glazed window at different outdoor relative humidities, using the first approach

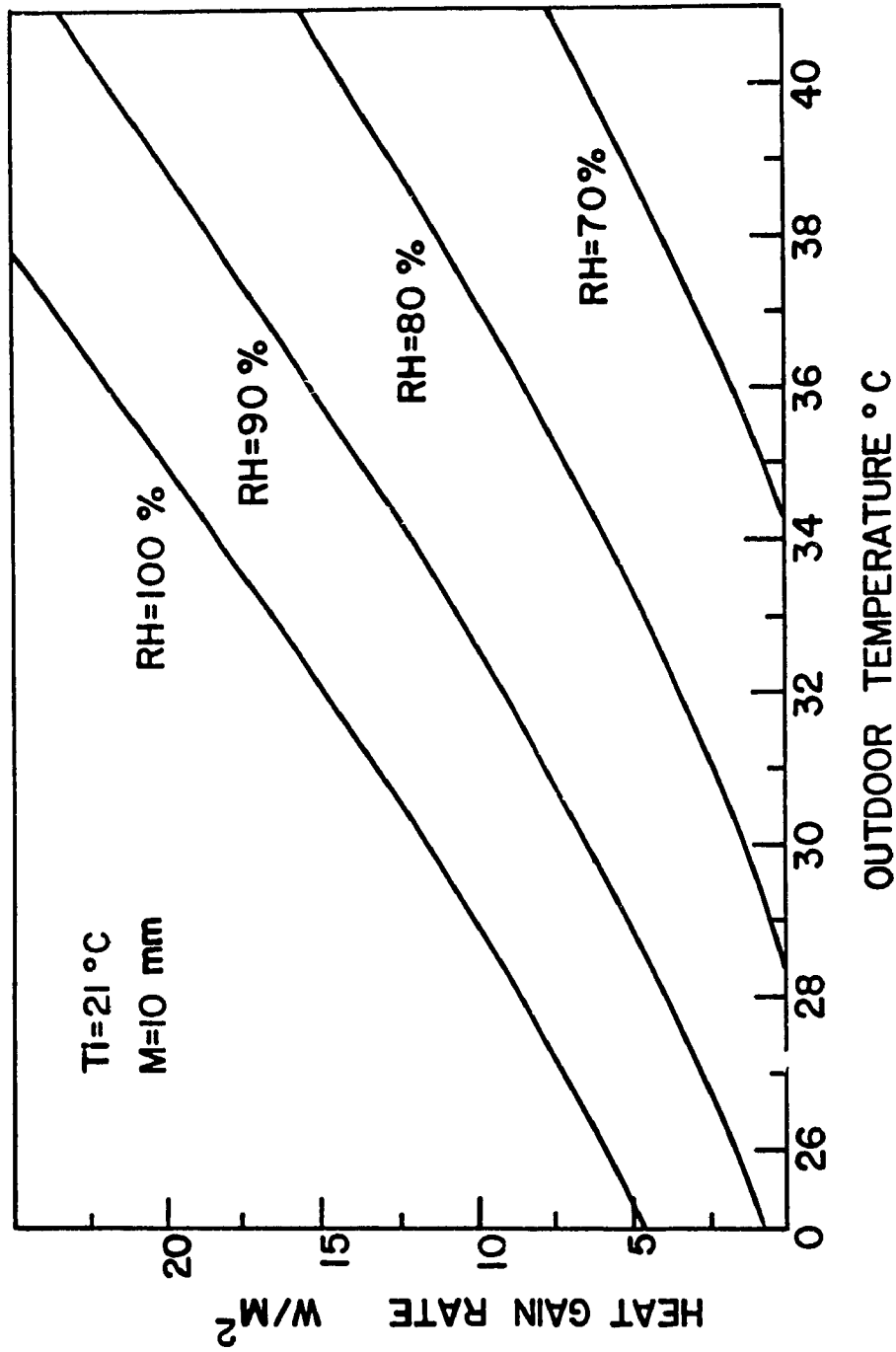


FIG. 4-9: Heat gain rate due to condensation on external surface of a single glazed window using the first approach with indoor temperature of 21°C

rate is estimated from Fig. 4-9 to be about 14 w/m^2 . The effect of the outdoor relative humidity and the outdoor temperature can also be recognized from Fig. 4-9, where it can be seen that the higher the relative humidity or the outdoor temperature the more is the heat gain.

4.4.2 Using the Second Approach

The heat gain rate due to outdoor condensation can also be calculated for the same single glazed window described above, using subprogram 4. The mass condensation rate is calculated by the second approach in this subprogram. A sample of the output of this subprogram is shown in Appendix B-2. The variation of the internal surface temperature (t_i) with time is shown in Figs. 4-10 and 4-11. Comparing these figures with figs. 4-7 and 4-8, it can be seen that the pattern of variation of the inner surface temperature at different indoor and outdoor conditions is the same. However, the rise in the surface temperature is smaller than that predicted by subprogram 2, when the first approach is used to predict mass condensation rate. Consequently, less heat gain rate is predicted by using subprogram 4. This is due to the fact that the second approach always underestimates the mass condensation rate as compared to the first approach. The heat gain rates evaluated by subprogram 4 at different outdoor conditions are shown in fig. 4-12. The curves were obtained for a constant indoor temperature (t_i) equals to 21°C . Comparing the curves in this figure with the curves in Fig. 4-9, it can be seen that the variation

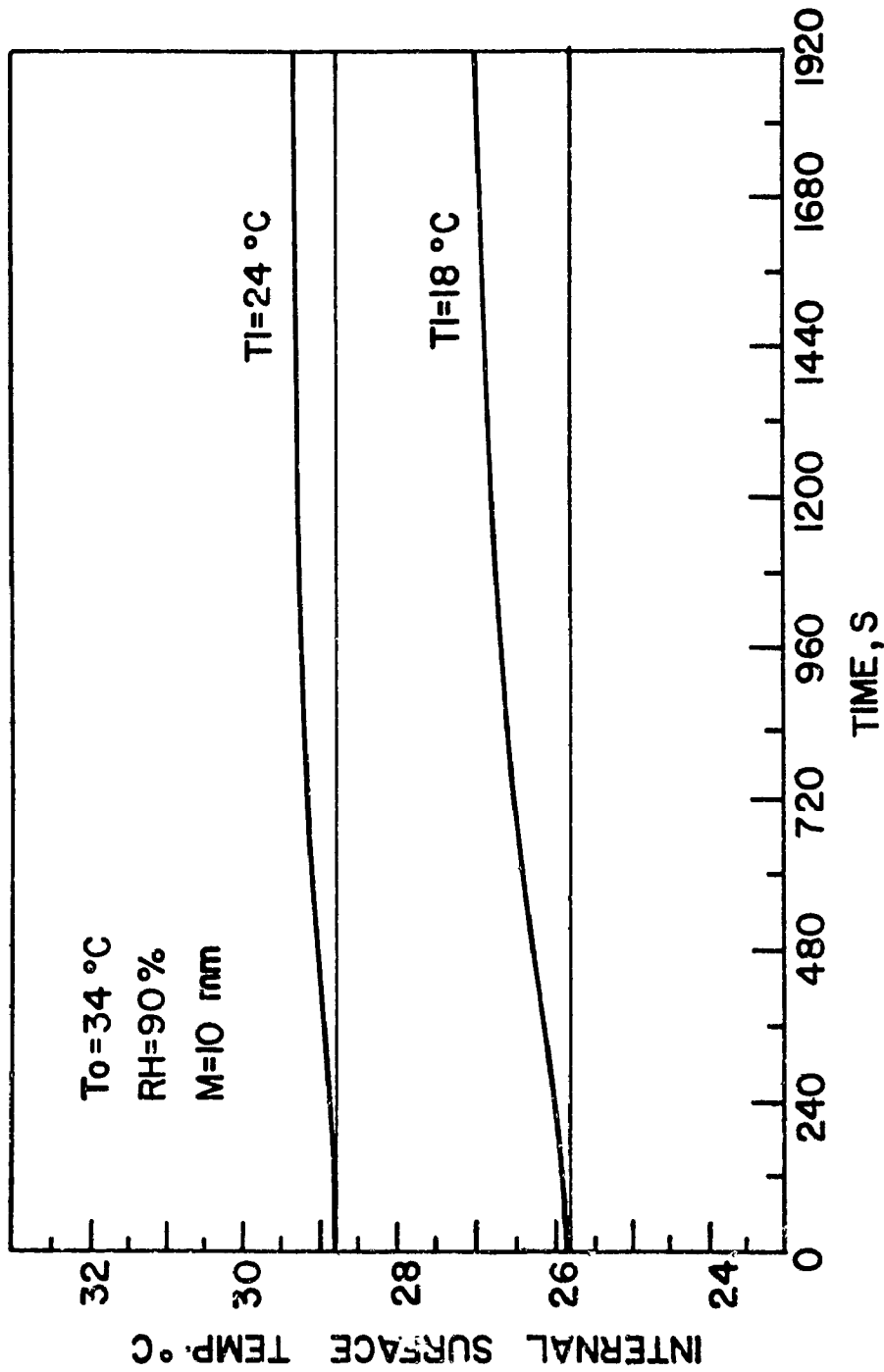


FIG. 4-10: Internal surface temperature variation with time as outdoor condensation occurs on a single glazed window at different indoor temperatures, using the second approach

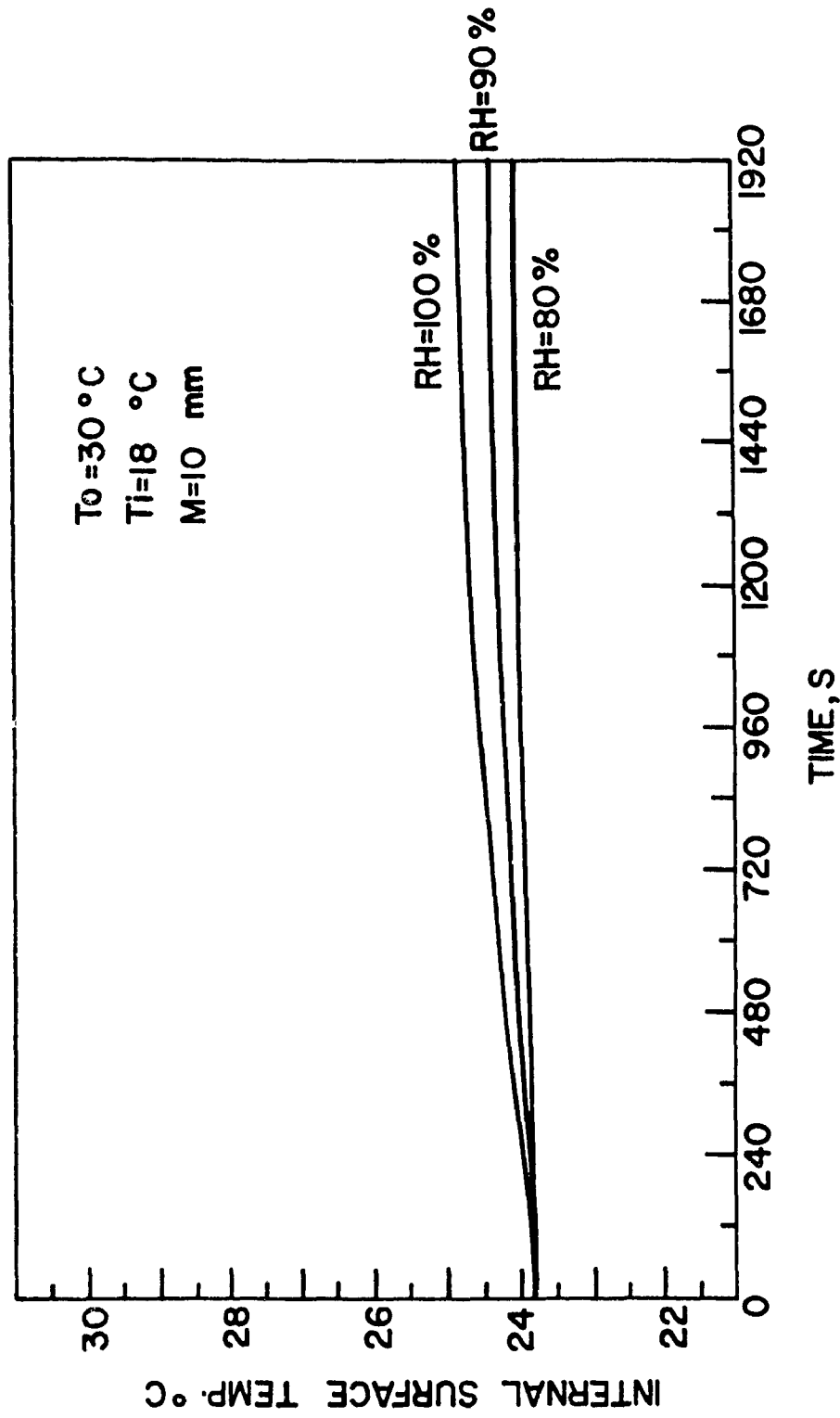


FIG. 4-11: Internal surface temperature variation with time as outdoor condensation occurs on a single glazed window at different relative humidities, using the second approach

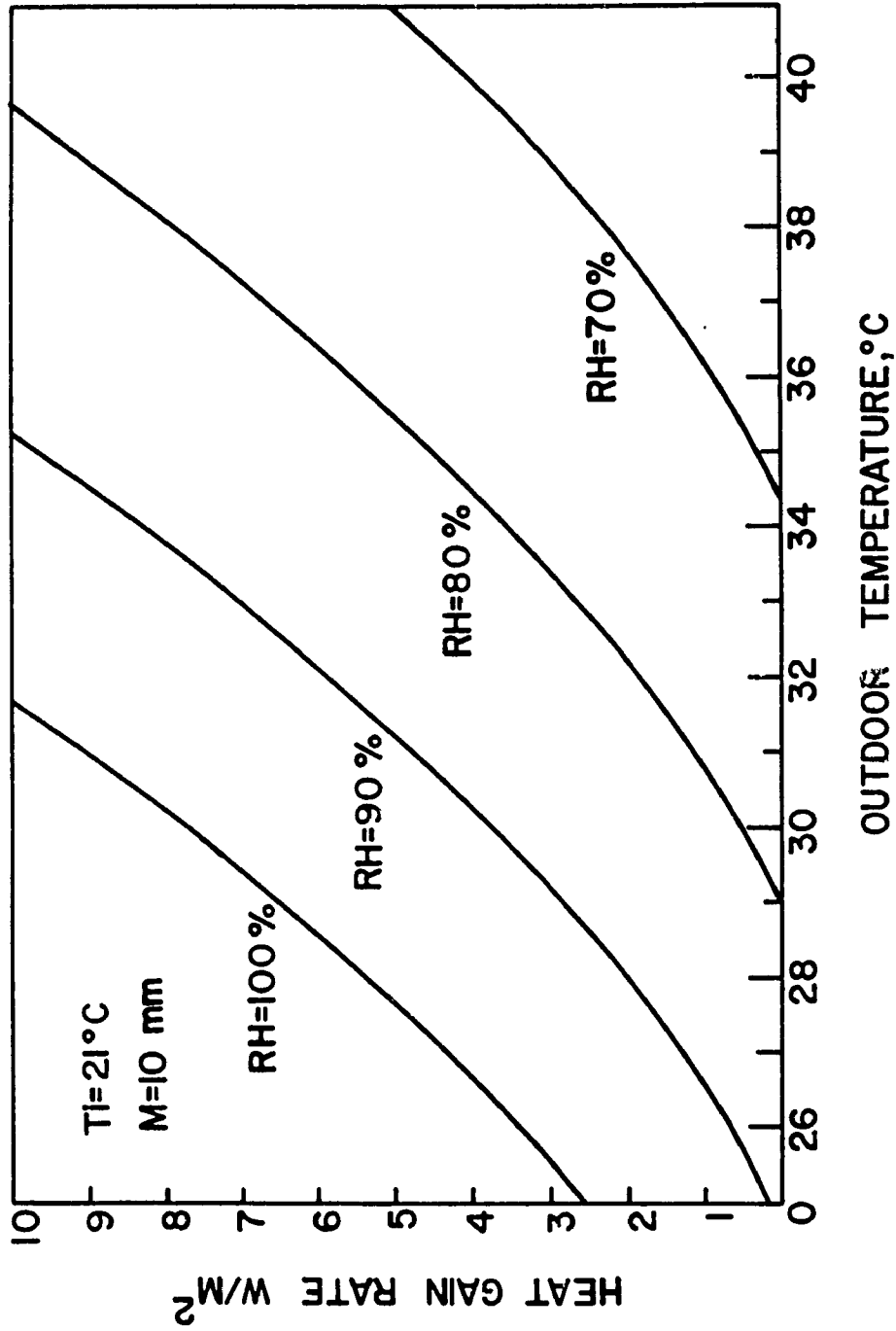


FIG. 4-12: Heat gain rate due to condensation on external surface of a single glazed window using the second approach with indoor temperature of 21°C

of the heat gain rate with outdoor relative humidity and temperature is the same. However, the heat gain predicted from Fig. 4-12 is always less than that predicted from Fig. 4-9. The deviation between the two predictions decreases as the outdoor temperature decreases until the two curves meet each other at the point where no condensation occurs as shown in Fig. 4-13.

4.5 HEAT LOSS ASSOCIATED WITH INDOOR SURFACE CONDENSATION

Indoor surface condensation on glazed windows, especially singly glazed, is a common phenomenon in winter. However, in extremely cold climate indoor surface condensation may also occur on insulated windows as well. The occurrence of the indoor surface condensation is dependent to a great extent on the outdoor temperature and the indoor relative humidity. However, the value of the indoor relative humidity is changing continuously depending on the type of activity and occupancy, even if it is controlled mechanically. Since, this variation is arbitrary, the relative humidity level which is controlled by the mechanical systems will be considered when dealing with heat loss due to condensation. Heat loss due to condensation is the loss of indoor humidity, because moisture loss through condensation process has to be substituted by the mechanical systems to maintain the desired humidity level. All air humidification processes provided by the mechanical systems require additional energy. Therefore, air dehumidification by surface condensation is a means of heat loss in buildings.

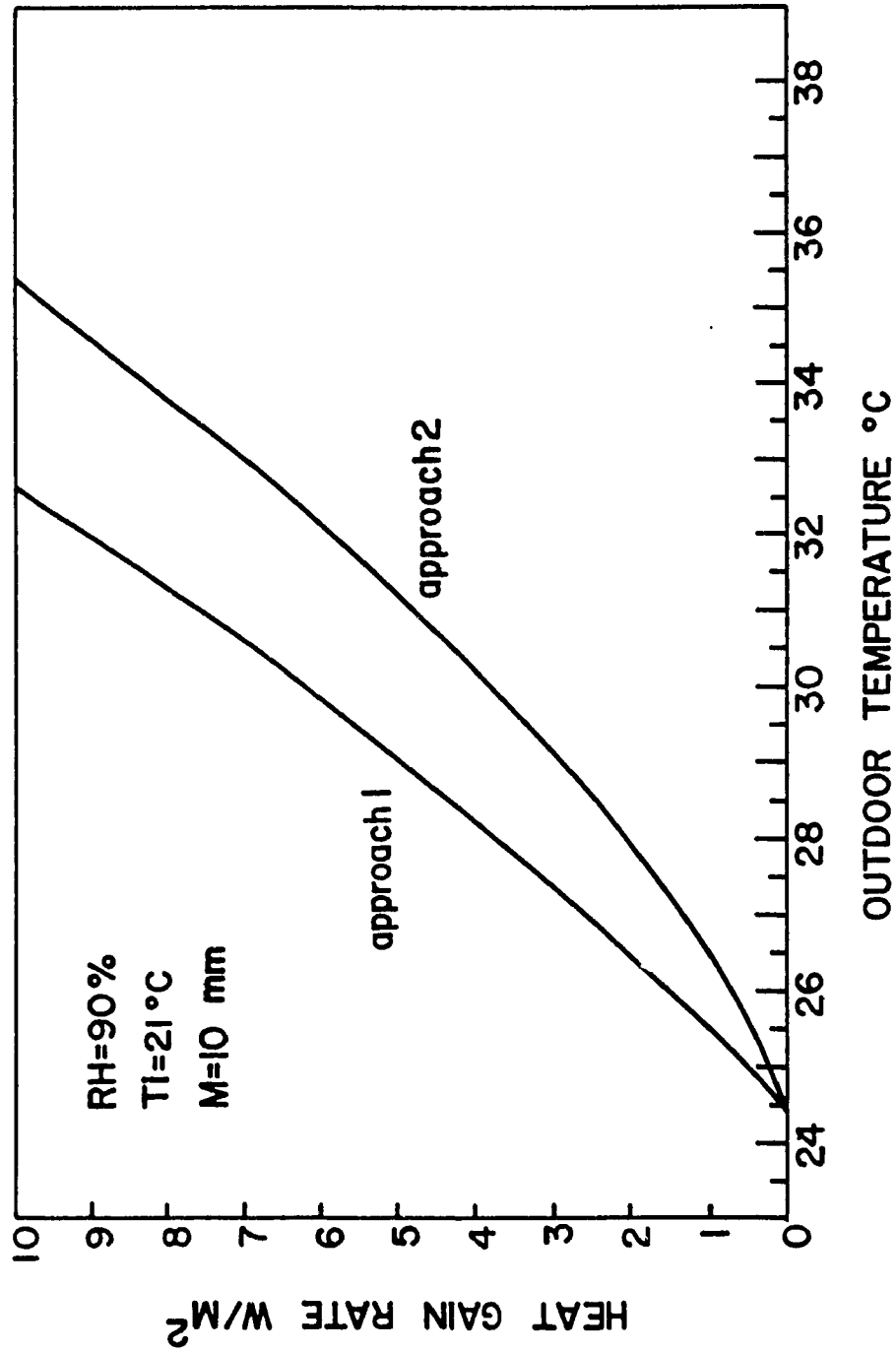


FIG. 4-13: Comparison between heat gain predicted by the first and the second approaches for a single glazed window at the same conditions

Heat loss due to indoor condensation on glazed windows can be calculated by multiplying the mass condensation rate predicted at the steady state stage $(\dot{m})_{ss}$ by the latent heat of vaporization of water (h_{fg}) as given by Eq. 4.44.

$$Q_{loss} = (\dot{m})_{ss} h_{fg} \quad \text{w/m}^2 \quad (4.44)$$

In predicting the heat loss due to condensation the two approaches will also be compared in predicting the mass condensation rate. Moreover, condensation on both single and double glazed windows are considered in the present study.

4.5.1 Using The First Approach

The heat loss due to indoor surface condensation on single and double glazed windows can be predicted by subprogram 3 and subprogram 6 respectively. A typical sample of the resulting output is shown in Appendix B-3 and B-4. These subprograms were run using input data for different indoor and outdoor winter conditions. The effect of wind speed is also considered in the calculation of the mass condensation rate. The heat loss rates due to indoor condensation on a single glazed window of 10 mm thick, and at a wind speed (WV) of 8 m/s can be predicted using Fig. 4-14, when it can be seen that the lower the outdoor temperature and the higher the indoor relative humidity, the higher the heat loss rate. The heat loss rate due to condensation on a double glazed window at particular conditions can be predicted from

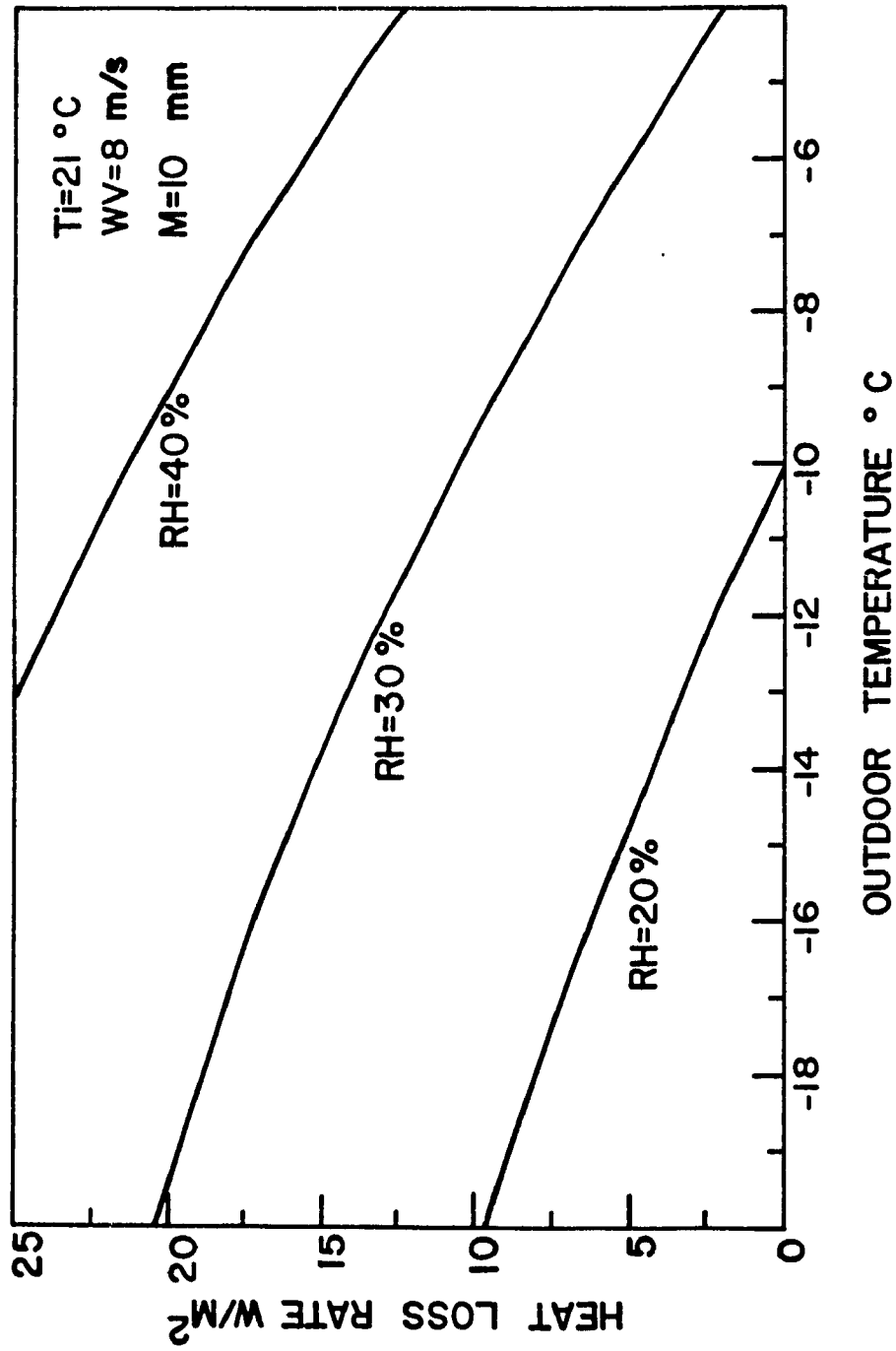


FIG. 4-14: Heat loss rate due to condensation on internal surface of a double glazed window using the first approach with indoor temperature of 21°C ; and 8 m/s wind velocity

Fig. 4-15. The glass thickness (M) for the double glazed window is 4 mm, and the air space (AS) is 8 mm. Comparing Fig. 4-15 with Fig. 4-14, it can be recognized that by using double glazed windows, condensation would only occur at much lower outdoor temperature compared to single glazed windows. Moreover, the heat loss rate is much less than that experienced when condensation occurs on single glazed windows at the same conditions. The effect of wind speed on condensation on the single glazed window is shown in Fig. 4-16. It can be seen that the higher the wind speed, the higher the heat loss rate. The wind speed has the same effect on condensation on the double glazed window as shown in Fig. 4-17. The increase in the heat loss rate, as a result of the increase in wind velocity, is almost constant over the outdoor temperature range in both case, since the two curves in both figures are parallel to each other. However, the effect of wind speed on the double glazed window is limited as compared to the single glazed one, since higher wind speed results in less increase in the heat loss rate.

4.5.2 Using the Second Approach

The heat loss rates due to indoor condensation can be evaluated by subprogram 5 for single glazed windows and by subprogram 7 for double glazed windows. A sample of the output of these subprograms is shown in Appendix B-3 and B-4. These subprograms were utilized to predict the heat loss due to condensation on the same single and double glazed windows used before. The heat loss rates predicted by

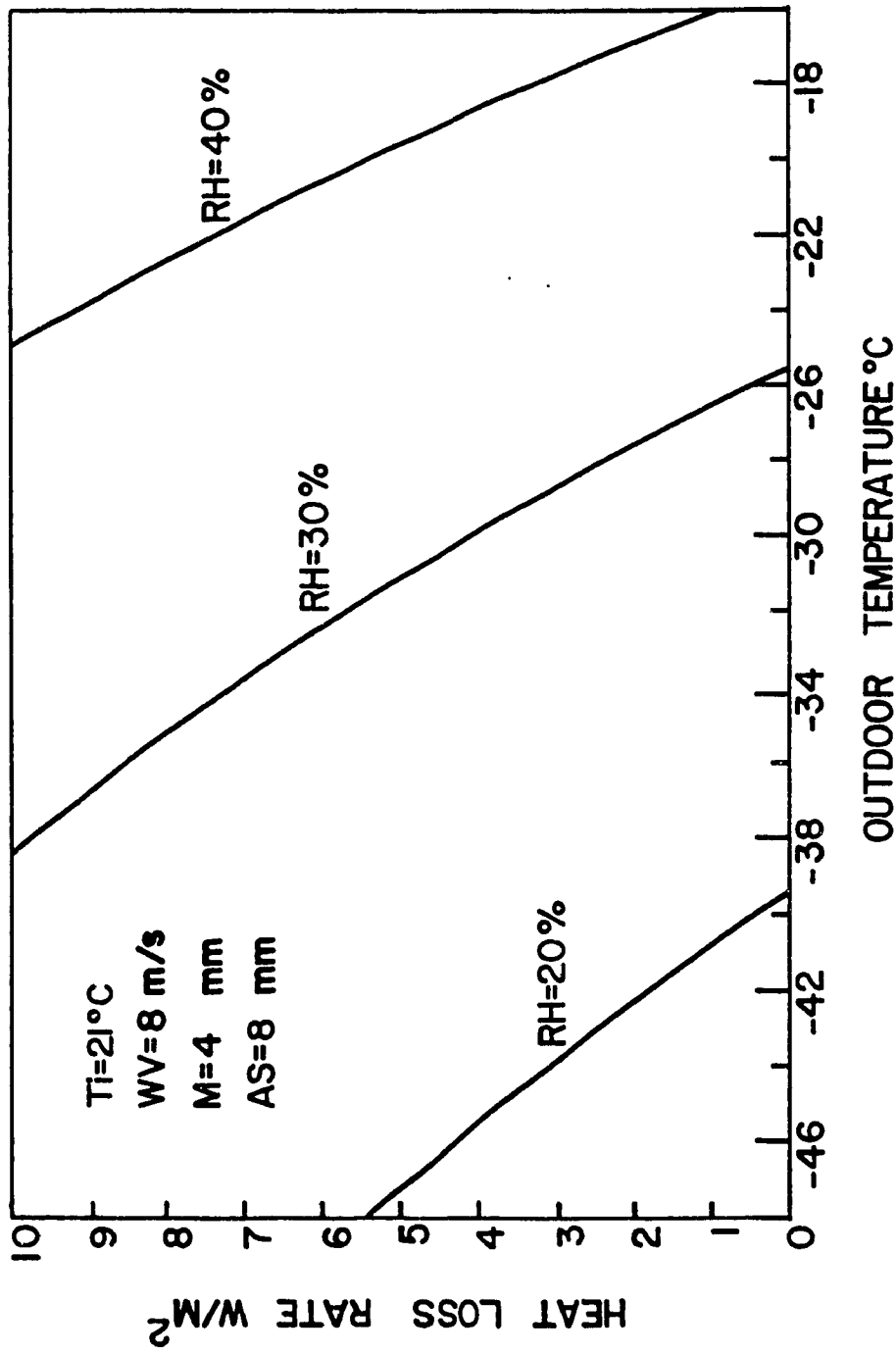


FIG. 4-15: Heat loss rate due to condensation on internal surface of a single glazed window using the first approach with indoor temperature of 21°C ; and 8 m/s wind velocity

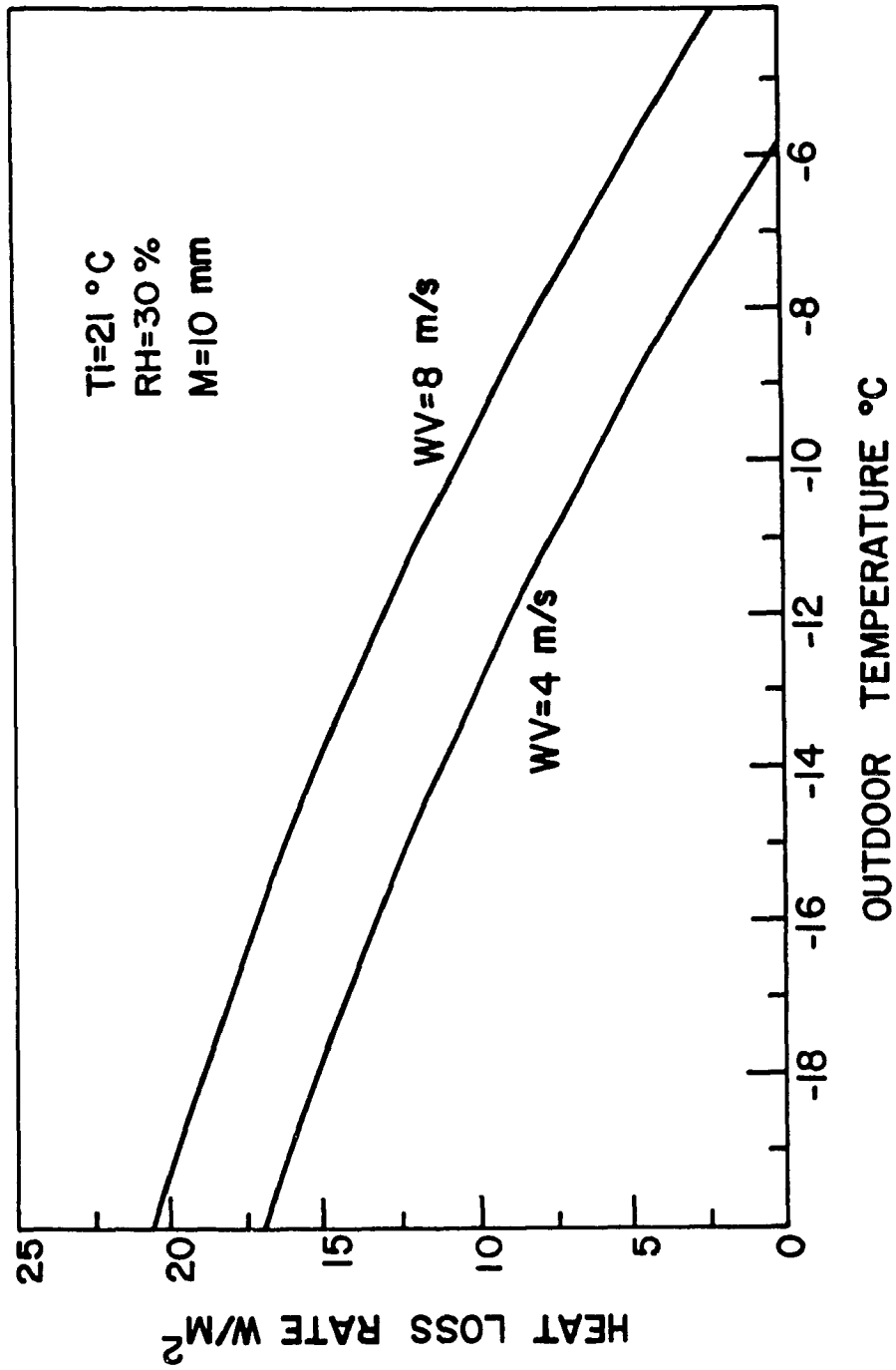


FIG. 4-16: Effect of wind speed on heat loss due to condensation on the internal surface of a single glazed window, using the first approach

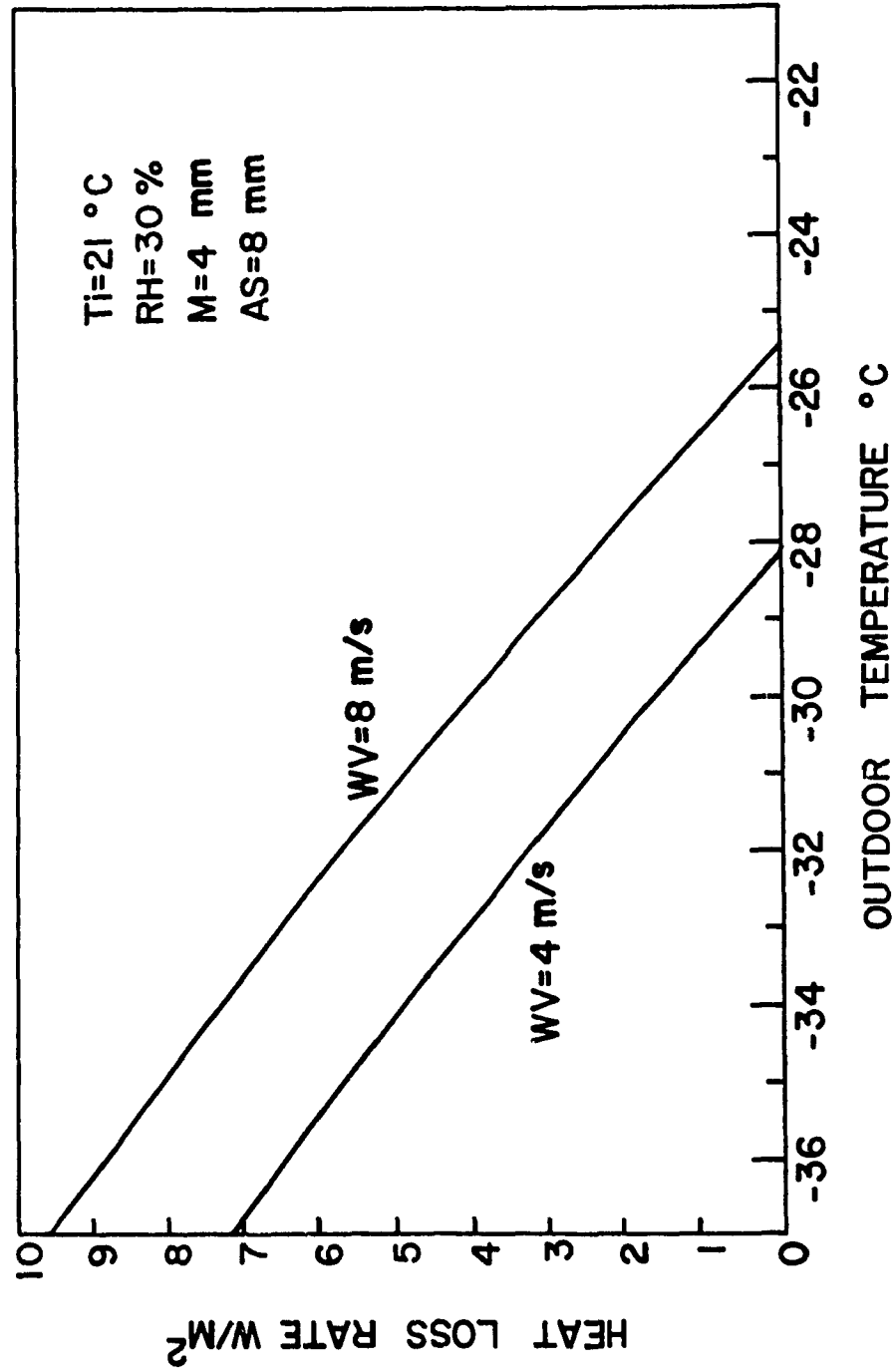


FIG. 4-17: Effect of wind speed on heat loss due to condensation on the internal surface of a double glazed window, using the first approach

subprogram 5 at different indoor relative humidities and outdoor temperatures are shown in Fig. 4-18. The heat loss rates predicted by subprogram 7 for the double glazed window at the same constant conditions are shown in Fig. 4-19. Comparing the results obtained by using the first approach, Figs. 4-14 and 4-15, and the results obtained by the second approach, Figs. 4-18 and 4-19, it can be seen that the second approach always underestimates the mass condensation rate compared to the first approach, as shown in fig. 4-20. In addition, it can be seen that the deviation between the two approaches increases as the outdoor temperature decreases. Consequently, more deviation is encountered when predicting heat loss due to condensation on double glazed windows as shown in Fig. 4-21, since much lower temperature is needed for condensation to occur on double glazed windows.

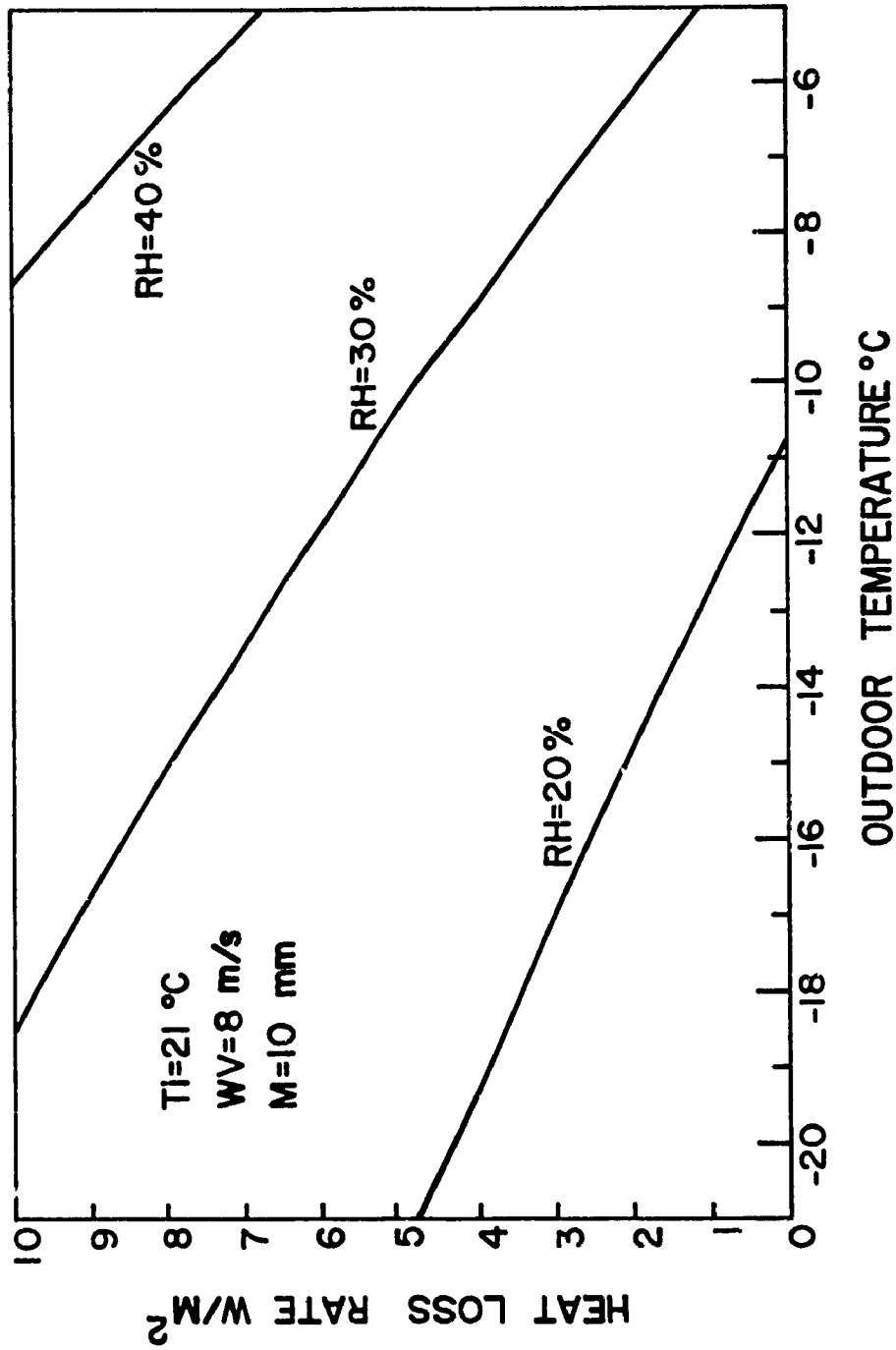


FIG. 4-18: Heat loss rate due to condensation on internal surface of a single glazed window using the second approach with indoor temperature of $21^{\circ}C$; and 8 m/s wind velocity

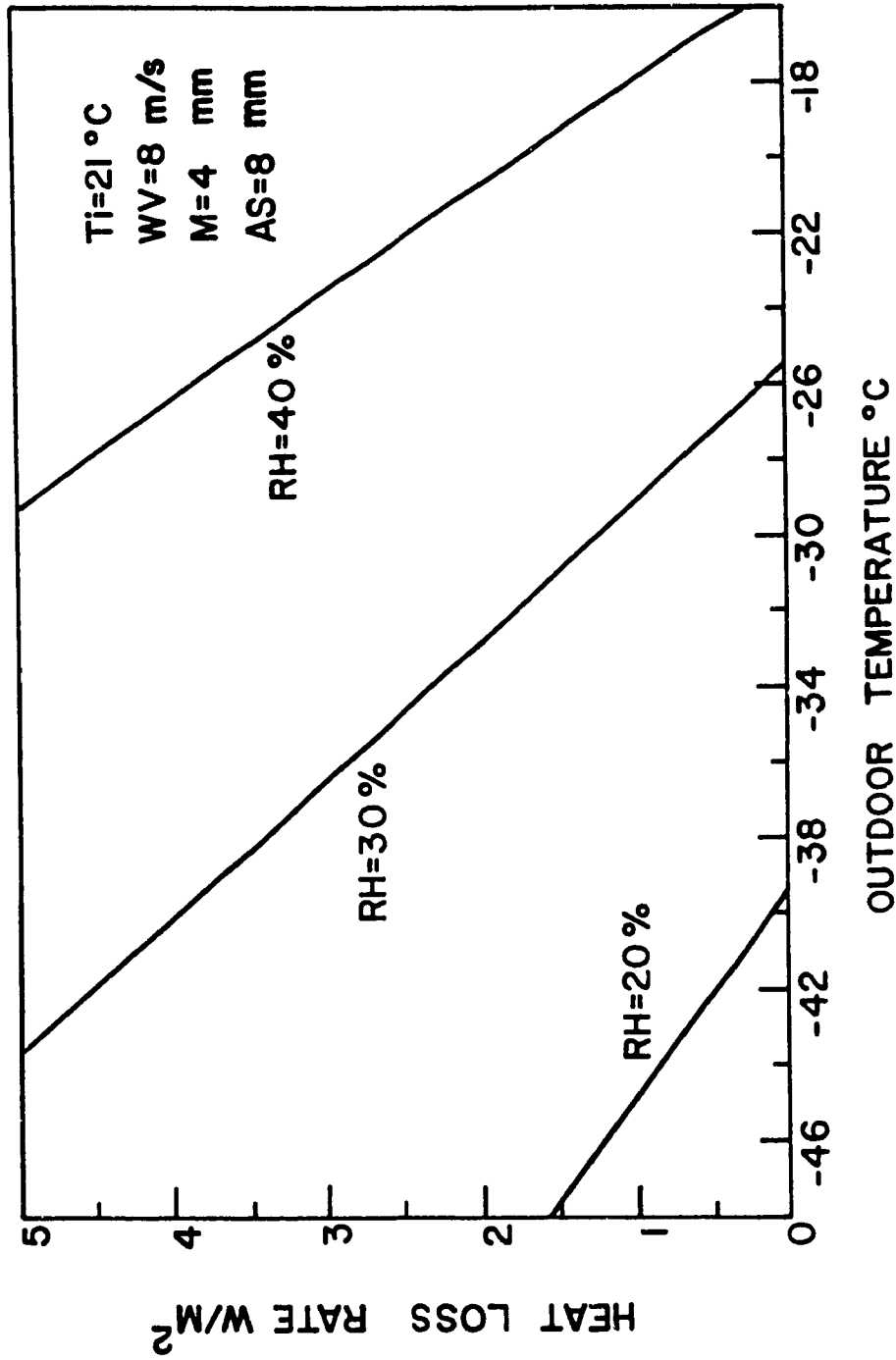


FIG. 4-19: Heat loss rate due to condensation on internal surface of a double glazed window using the second approach with indoor temperature of 21°C ; and 8 m/s wind velocity

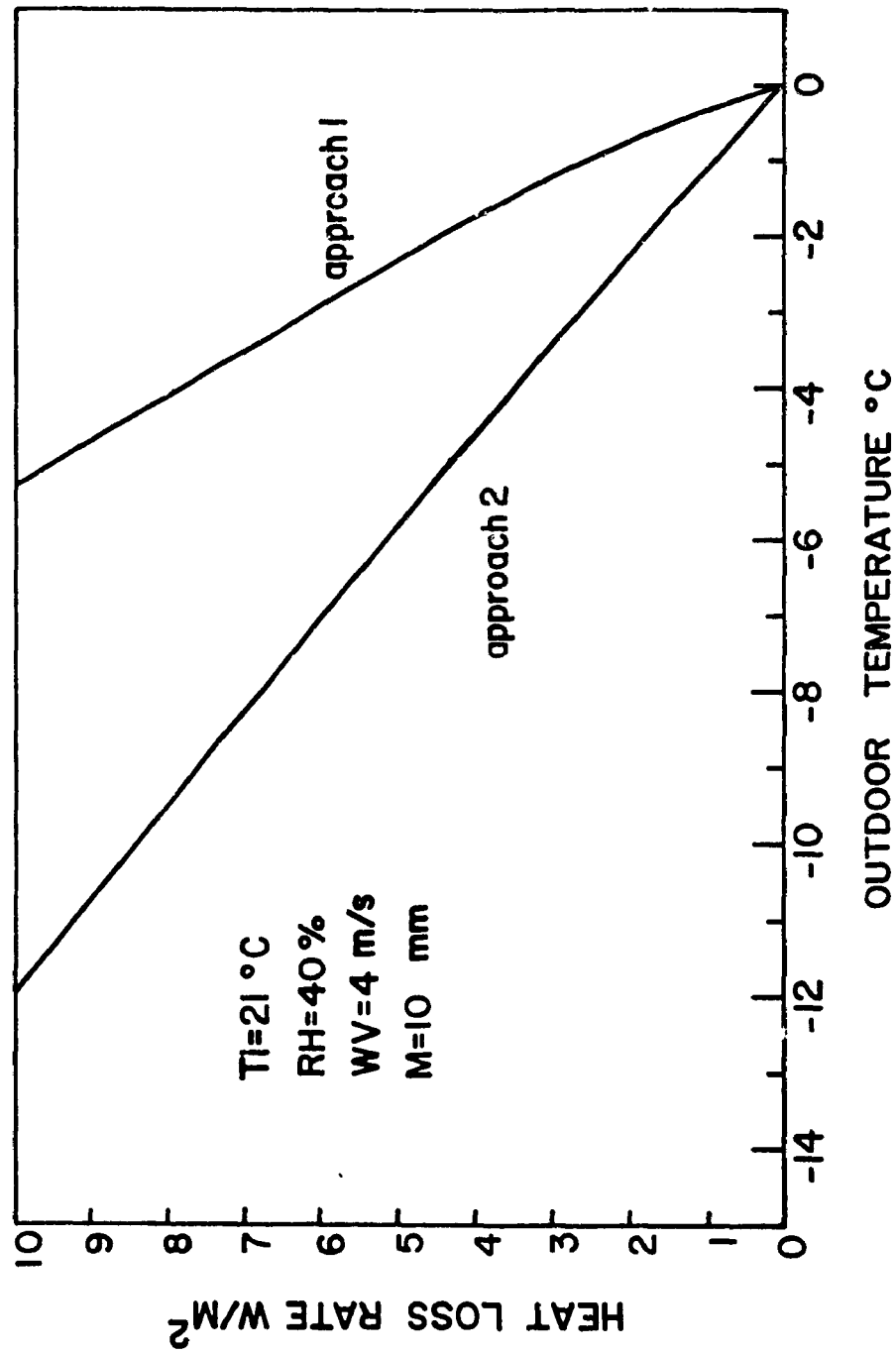


FIG. 4-20: Comparison between heat loss predicted by the first and the second approaches for a single glazed window at the same conditions

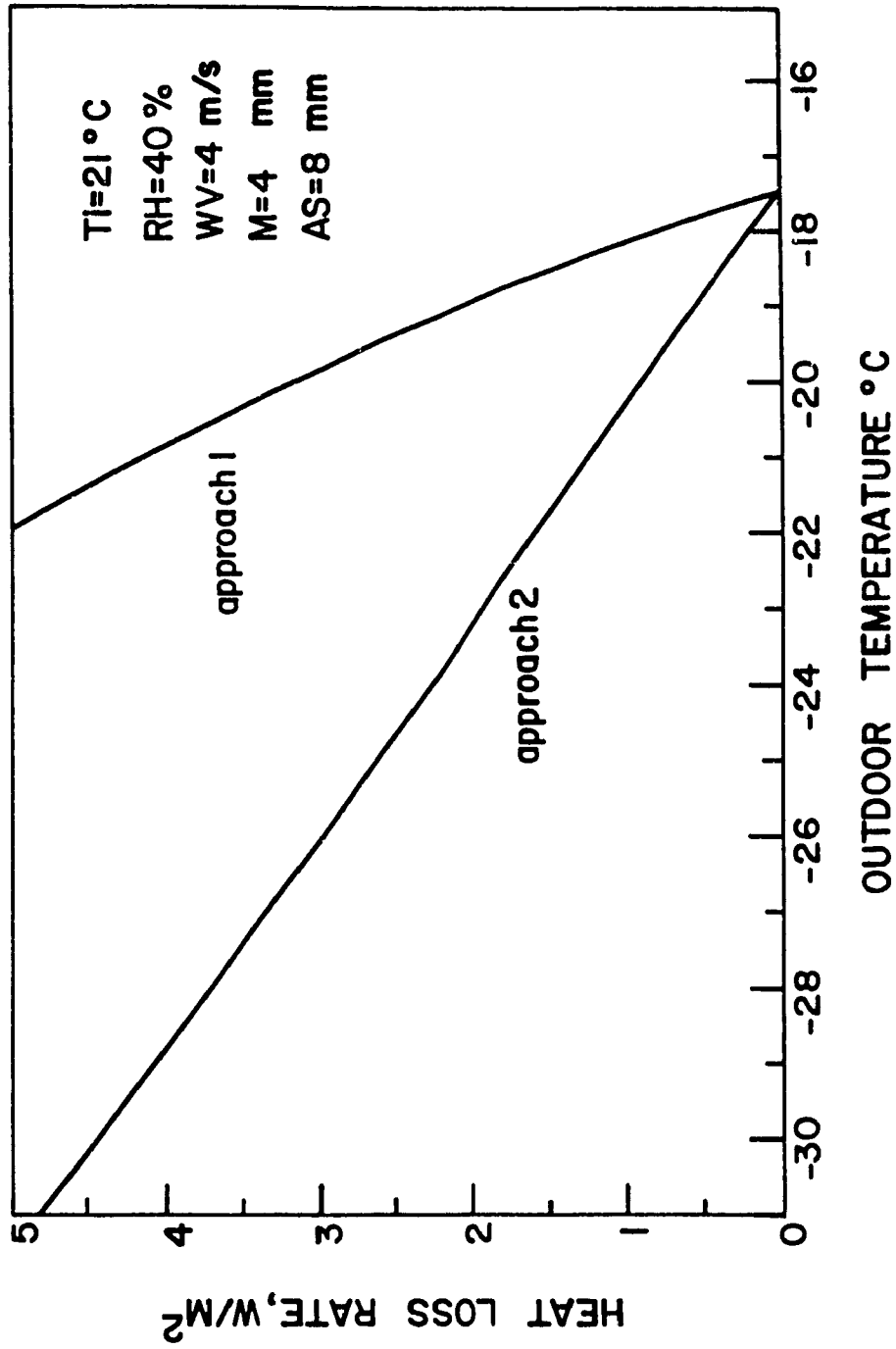


FIG. 4-21: Comparison between heat loss predicted by the first and the second approaches for a double glazed window at the same conditions

CHAPTER 5

EXPERIMENTAL EVALUATION AND RESULTS

5.1 OBJECTIVES OF EXPERIMENT

The evaluation of mass condensation rate is the key issue for evaluating heat gain or loss due to condensation on glazed windows. In this study, two different approaches have been suggested to predict mass condensation rates at different indoor and outdoor conditions. A preliminary experimental program has been conducted to evaluate the applicability of each of the two approaches in predicting mass condensation rate on windows surfaces. In addition, the finite difference analysis used to predict the glass surface temperature is to be tested by comparing the rise in the surface temperature predicted by the model to that obtained from the experimental results. In the following sections, a description of the experimental set up, test procedure and logic, and the results obtained are discussed.

5.2 EXPERIMENTAL SET UP

In order to evaluate mass condensation rate and the corresponding heat transfer on a glazed window a special preliminary experimental set up has been developed. In this test arrangement the mass condensation rate on a single glass pane and the glass surface temperature can be measured. The testing system as shown in Figs. 5-1 and 5-2 comprises the following parts:

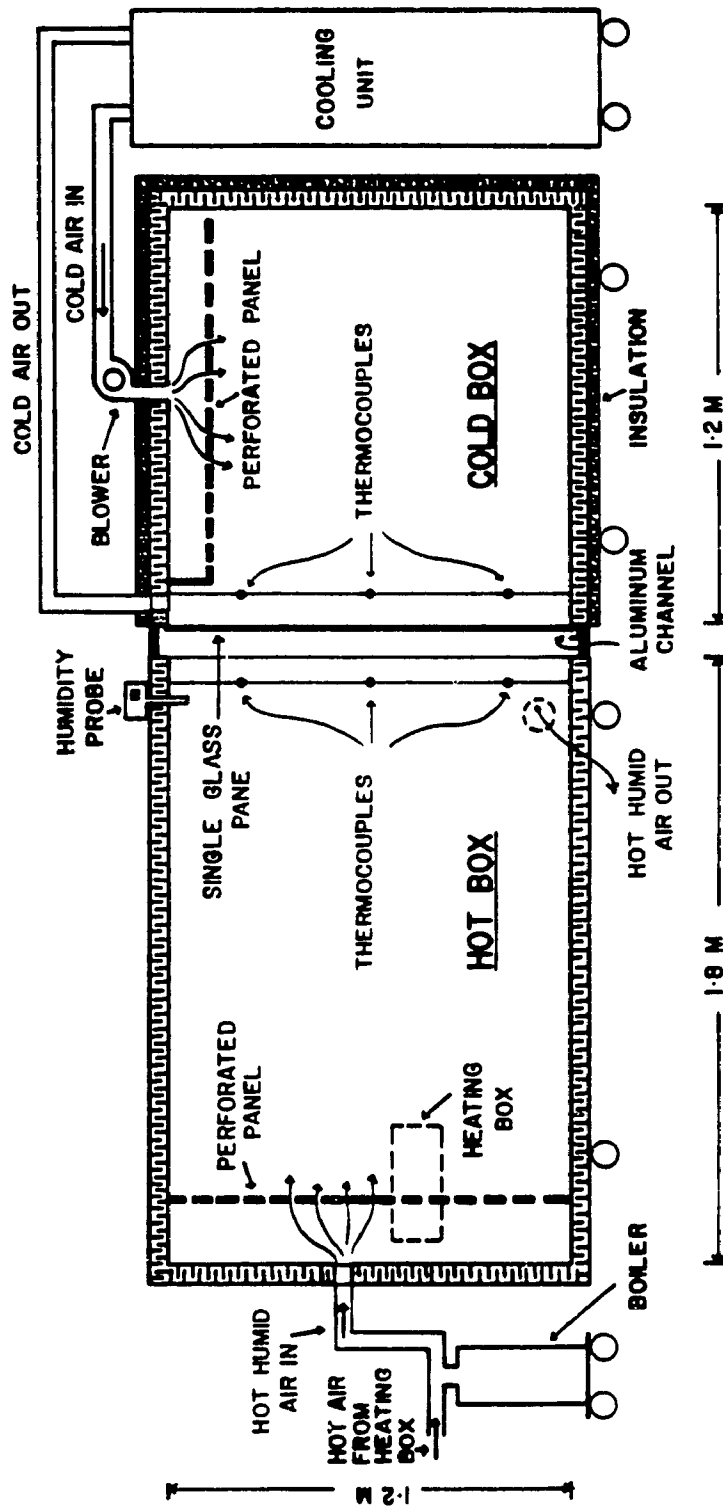


FIG. 5-1: Cross sectional view of the testing system

1. Hot box
2. Cold box
3. Single glass pane
4. Heating element and controller
5. Cooling unit and controller
6. Steam generator and controller
7. Humidity probe HMP 111Y
8. Digital multimeter
9. Copper constantan thermocouples
10. Blowers for hot and cold air supply

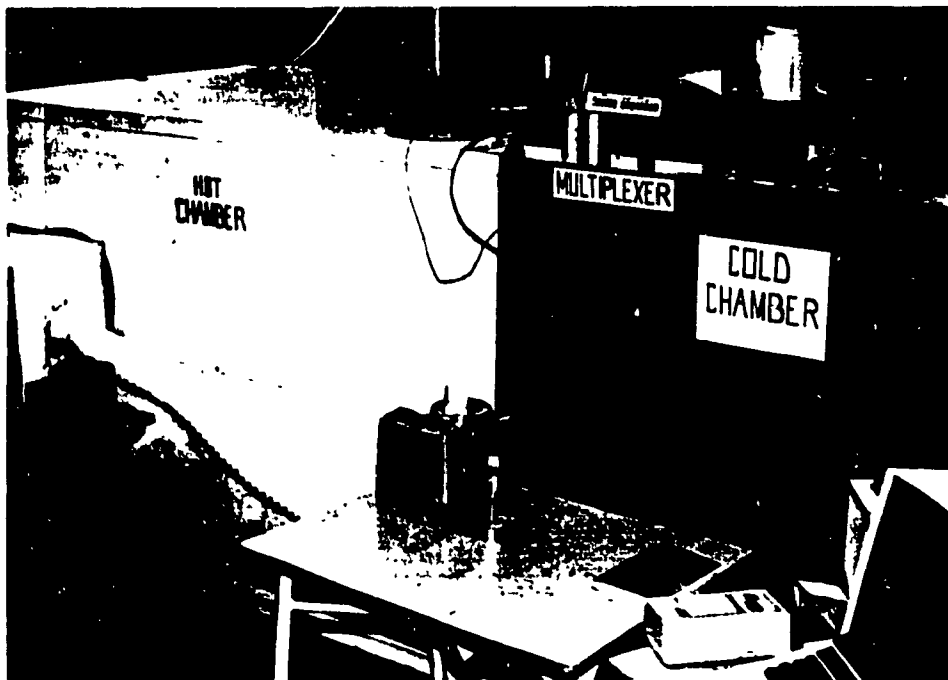


FIG. 5-2: Overall view of the testing system

5.2.1 Hot Box

The hot box is constructed with R-10 panels so as to minimize heat loss. The box is heated by an electric heater which is located in a separate small box as shown in Fig. 5-3. The hot air is extracted from the heating box by a blower which in turn forces hot air into the hot box through an opening located at the rear of the box as shown in Fig. 5-1. The inlet opening is located as far as possible



Fig. 5-3: The Heating Box Attached to the Hot Chamber

from the glass pane so that to avoid undesirable forced flow over the pane. In addition, a perforated panel as shown in Fig. 5-4 is located across the box near the inlet opening to encourage uniform distrib-

ution of the incoming air, as well as, to prevent forced flow over the glass pane by reducing the velocity of the incoming hot air. The air leaves the hot box at an outlet opening located near the bottom of the box and near the glass pane, since it is most likely that the coldest air settles at this region.



FIG. 5-4: A Perforated Panel in the Hot Box

The temperature of air inside the hot box is measured by taking the average of three thermocouples located at different heights in the box as shown in Fig. 5-1. These thermocouples were located about 20 cm from the glass pane so as to be as close as possible to the theoretical assumed temperature variation within the box, and at the same

time to be practical in terms of the actual temperature variation within a room.

5.2.2 Cold Box

The cold box is constructed from R-10 panels which are covered with additional R-3.9 insulation panels, so that the air temperature inside the box can be kept as cool as possible. The cold air is supplied through an opening located at the top of the box in order to maintain a minimum vertical temperature gradient. In addition, a perforated panel is provided near the inlet opening for the same purpose as for the hot box. The cold air is forced into the box by a blower which extracts cold air from a cooling unit as shown in Fig. 5-1. The air leaves the box at an opening located at the top of the box and near the glass pane, since the hottest air is likely to accumulate at this region. The temperature of air inside the cold box is determined by taking the average measured temperatures of three thermocouples located vertically about 50 cm apart and about 20 cm away from the glass pane.

5.2.3 A Single Glass Pane

A single glass pane 5 mm in thickness is used for testing and is placed between the two boxes. The glass pane is perfectly sealed to the cold box so that no cold air can infiltrate to the hot side and disturb the process of condensation. The condensed water from the hot side is collected by an aluminum channel which is located at the bottom of the pane as shown in Fig. 5-5.

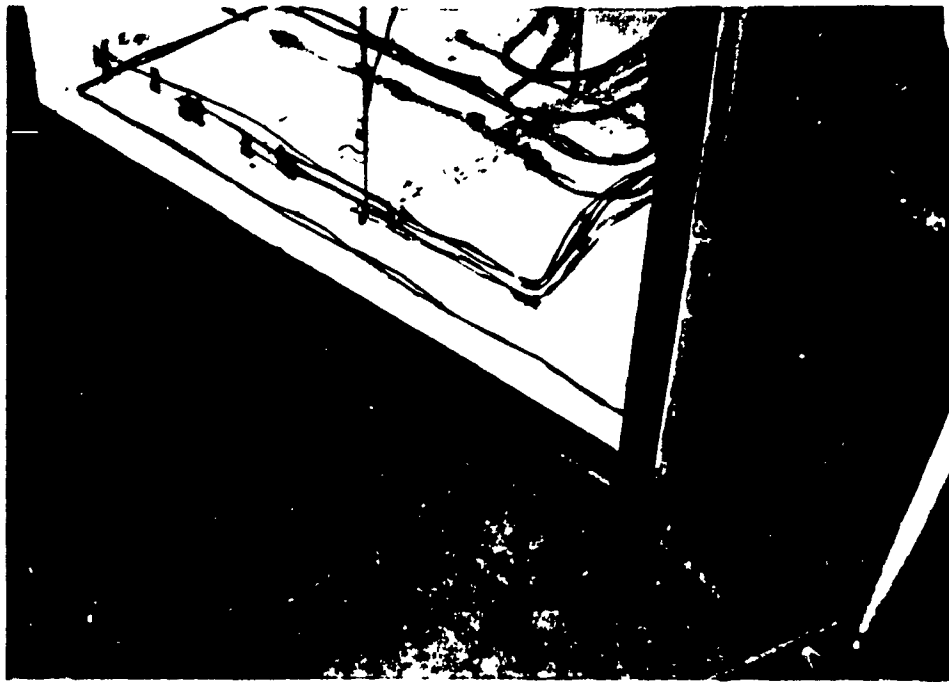


FIG. 5-5: An Aluminum Channel at the Bottom of the Pane to Collect Condensed Water

The temperature of the glass surface is measured only from the cold side by using thermocouples so as not to disturb condensation on the hot side. The glass surface temperature is taken to be the average of the readings of 24 thermocouples distributed uniformly in a grid pattern all over the glass surface as shown in Fig. 5-6. The deviation of a single thermocouple reading from the average value was found to be within the range of $\pm 1^{\circ}\text{C}$.

5.2.4 Air Humidification

Air in the condensing side (the hot box) is humidified by injecting steam into the hot air stream entering the box. Steam is generated by a steam generating system shown in Fig. 5-7 which can provide



FIG. 5-6: Glass Surface Temperature Measurement by Thermocouples



FIG. 5-7: Steam Generating System

continuous steam flow. A heating element which is controlled by a current regulator is used to boil water in a small tank of 20 litres in capacity. The water level in this tank is required to be constant, so as to minimize variation in steam output and to keep the heating element always immersed in water. Therefore an auxiliary tank with a constant water level is connected with the hot tank so that when the water level in the hot tank decreases, it is gradually substituted from the auxiliary tank. The water level in the auxiliary tank is controlled by a floater which regulates the water input from the building supply tap. The amount of steam and hence the relative humidity inside the hot box is controlled by regulating the power input to the heating element which in turn is controlled by a current regulator.

5.2.5 Humidity Measurement

Relative humidity in the hot box is measured near the glass pane by a humidity probe inserted through an opening at the top of the box as shown in fig. 5-1. A humidity probe, HMP 111Y, is used in this study to measure the relative humidity, and a data logger, Li-1000, is used for data acquisition and retrieval, and to permit a constant display for the relative humidity alone while the test is in progress. The probe covers a humidity range from 0% to 100% RH. At a relative humidity of 80% and above the error is about $\pm 3\%$. However, when there is a temperature difference of $\pm 1^{\circ}\text{C}$ between the humidity sensor and the air, an error up to $\pm 6\%$ RH is expected at 90% relative

humidity. The error mentioned is at its maximum when the sensor is colder than the surroundings, and relative humidity levels are 90% or more.

5.2.6 Temperature Measurement

Air temperature within the two boxes and surface temperature of the glass pane are measured using copper-constantan thermocouples. To ensure proper surface temperature measurement a highly conductive material (silicon heat sink compound) was used to connect the thermocouples to the glass surface. In addition, a highly adhesive tape is used to attach the thermocouples to the surface so as to avoid loose connection. The reference point for temperature measurement was taken to be 0°C, by placing one junction of the thermocouple in melting ice. An ice-box is used to maintain the ice at the melting temperature while the test is in progress.

HP-3466A digital multimeter was connected to a specially designed switching system, shown in Fig. 5-8, to manually scan the average air temperature in the two boxes, and the glass average surface temperature. According to the manufacturer, the maximum deviation by this type of voltmeter is about $\pm 0.3^{\circ}\text{C}$.

5.3 TEST LOGIC AND PROCEDURE

There are five parameters to be considered in this experiment; i) air temperature in the hot box, ii) air temperature in the cold box,

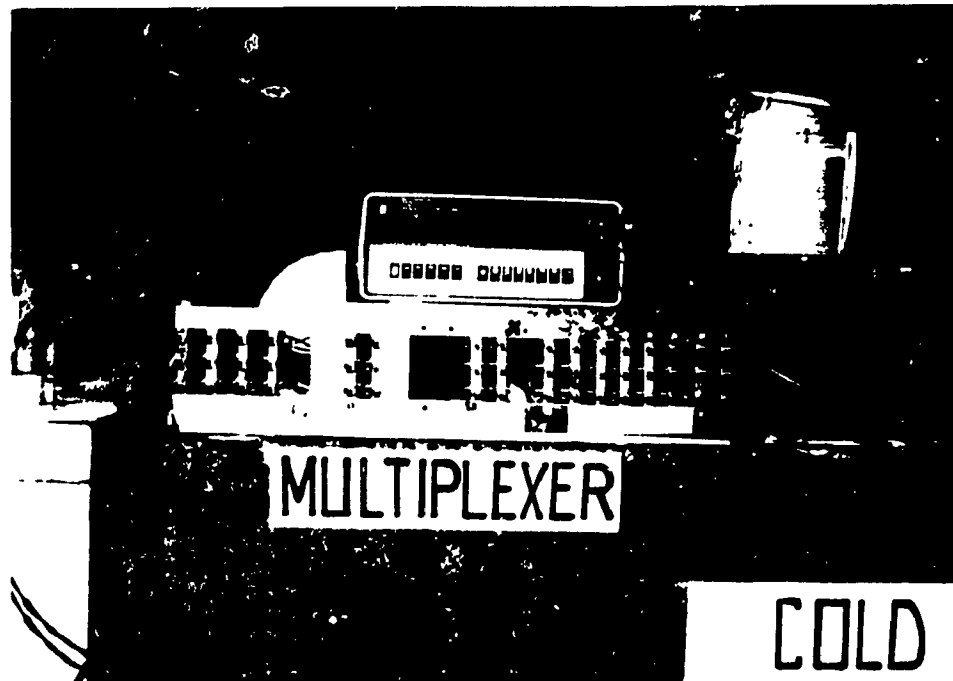


FIG. 5-8: A Switching System Connected to a Voltmeter

iii) relative humidity within the hot box, iv) glass surface temperature on the cold side, and v) mass condensation rate. The mass condensation rate and the corresponding rise in the glass surface temperature are the main findings in this experiment. In order to evaluate the rise in glass surface temperature, the surface temperature before and after condensation at the same conditions have to be known. However, it must be noted that when air in the hot box is humidified by steam injection the air temperature in the hot box

starts to rise leading to different conditions in both the hot and the cold boxes. In other words, the rise in the glass surface temperature is not only due to condensation, but also due to the increase in air temperature in both boxes. Consequently, the effect of condensation alone in raising the glass surface temperature cannot be evaluated directly. A procedure is devised in this experiment to allow the evaluation of the rise in glass surface temperature due to condensation alone.

5.3.2 Glass Surface Temperature Rise Due to Condensation

In order to evaluate glass surface temperature rise due to condensation, the effect of air temperature in both boxes on the glass temperature has to be known. Under steady state conditions, the glass surface temperature on the cold side can be expressed in terms of the air temperature in the hot box (T_h) and in the cold box (T_c) by:

$$T_{gc} = T_c - \frac{R_{sc}}{R} (T_c - T_h) \quad (5.1)$$

where

R_{sc} = cold side surface resistance (m^2-C/w)

T_{gc} = glass surface temperature in the cold side (m^2-C/w)

R = overall resistance of the glass pane (m^2-C/w)

The effect of the air temperature alone on the glass surface temperature during condensation can be calculated by Eq. 5.1 if the resistance ratio R_{sc}/R is known. In order to evaluate R_{sc}/R from Eq. 5.1,

the glass surface temperature and the air temperature in both boxes have to be known at a particular thermal setting before any condensation takes place. So, before humidifying the air in the hot box, the system was kept operating at a constant heating and cooling rates for an average of four to five hours until thermal stability is reached and no major change occurs in any of the temperature readings with time. Then these temperatures were used to evaluate R_{sc}/R using Eq. 5.1. The value of R_{sc}/R was found to differ slightly from one set of conditions to another depending on the thermal setting used (i.e. air temperature in both boxes as well as air supply rate). In this experiment it was found that the value of the resistance ratio R_{sc}/R mainly ranged between 0.405 and 0.43. These values should actually be higher than 0.43. Yet, it should be noted that they were evaluated using the measured average glass temperature and the average air temperatures in both boxes, which involves certain approximation.

Once the R_{sc}/R value is established, humidity is then introduced to the hot air stream. When condensation starts to occur, the test is kept operating at the same thermal setting for several hours until the mass condensation rate, and both humidity and all temperature readings become reasonably constant. These readings are then considered one set of data as those in Table 5-2. The measured air temperatures in both boxes and the previously found R_{sc}/R value before introducing humidity, can be used to evaluate the surface temperature due to air temperatures. Then by comparing this calculated surface temperature

with the measured surface temperature, which includes the effect of condensation, the temperature rise due to condensation can be found.

5.3.2. Determination of Mass Condensation Rate

In order to measure the mass condensation rate in this experiment, a calibrated balance to weigh the condensed water, a stop watch to measure the time needed to collect the sample, an aluminum channel to collect water condensed on the glass surface, and a small container to collect condensed water, were utilized.

After the value of R_{sc}/R is evaluated before introducing humidity as described above, the next stage starts by introducing steam to the hot air supply. As a result, the relative humidity inside the hot box starts to increase until it is sufficiently high for condensation to occur. It takes at least one hour for condensed water to drip down from the collecting channel placed immediately under the glass pane. In addition, about one more hour is needed for the relative humidity to reach a reasonably constant and stable level. The mass condensation rates were determined only when such conditions prevail. As a precaution and to ensure the accuracy of the results, the measurement process is repeated until at least three or four consecutive measurements were found approximately identical. For each measurement taken, the corresponding values of relative humidity, and air and glass surface temperatures were instantly recorded. The average time required to obtain a single measurement, that would satisfy the stability criteria, is about one hour under normal testing conditions.

There are two parameters that are essential to the measuring process of mass condensation rate. The first is the weight of the condensed water. The second is the time needed for collection. Because of the adhesion between water and the surface of the aluminum channel, the condensed water cannot be collected continuously and instantaneously as it drips into the channel from the glass surface. Condensed water drips off the channel, as shown in Fig. 5-9, at certain time intervals followed by no collection period. In order to determine the time needed to condense any collected sample a starting and an ending events have to be specified. At the instant when some condensed water drips off the channel a starting time is determined, and an ending time is determined when a new sample of condensed water, which is the one to be weighed, is collected. Thus, the collection period is specified by two collection events so that any condensation occurs during this period is collected. Knowing the mass of the collected condensed water and the collection time, the mass condensation rate (Kg/s) can be determined for any tested case.

5.4 RESULTS AND DISCUSSION

Over fifty experimental measurement of the mass condensation rate and the rise in glass surface temperature were obtained at different air temperatures and relative humidities. Temperature variation in one box immediately affects the temperature in the other box. This has made it very difficult to maintain the air temperature constant in



FIG. 5-9: Instantaneous collection of condensed water

one box while varying the temperature in the other. Therefore, the results cannot be represented in a single graph. Accordingly, the experimental results were found to be well represented by a tabular form rather than a graphical form. These experimental results were compared with the theoretical solutions of both approaches discussed previously. The measured mass condensation rate was found to be always less than what is calculated using the first approach, and always more than what is calculated using the second approach. However, the measured glass surface temperature rise is mainly lower than what is predicted by both approaches. The degree of deviation between

the measured values and the values predicted by the theoretical model does not follow a constant pattern as shown in Appendix-C. This is because of the difficulty in keeping constant testing conditions in both boxes, and the possible errors encountered when taking the readings and measuring the mass condensation rate. Therefore, these experimental results need to be evaluated so as to select the most reasonable and consistent data sets. In order to evaluate the experimental values of mass condensation rate, the pattern of variation as compared to the variation of the temperatures in both boxes and the variation in the relative humidity has to be known. However, there is some difficulty in tracing the effect of the three parameters simultaneously on the mass condensation rate. Therefore, the mass condensation rate predicted by the first approach, which represents the combined effect of the three parameters, will be used as a standard in testing the consistency of variation of the measured mass condensation rates. The mass condensation rates evaluated by the first approach are first arranged in an increasing form with the corresponding measured mass condensation rates. The degree of variation of the theoretical values and the corresponding experimental values are then compared. Those experimental values which show a variation of a reasonable degree of consistency with the variation of the theoretical values are considered acceptable. About half of all measured values of mass condensation rates, which are shown in Table (5-1), was found to be acceptable. It can be seen that the measured mass condensation

rates are about 16% less than the values predicted by the first approach and about 20% higher than the values predicted by the second approach. In order to justify the use of one approach over the other, the factors causing deviation between the measured and the calculated theoretical mass condensation rates have to be well defined. Moreover, the effect of each factor on the mass condensation rate has to be known. This effect could be positive, when more condensation is produced than what it should be, or negative when less condensation is produced than what it should be. Once all factors of deviation and their effects are known, the combined effect on mass condensation rate measurement can be determined, and it can be easily judged whether the measured mass condensation rate should be more or less under the same conditions. Consequently, one of the approaches can be justified to be used in predicting mass condensation rate. The deviation factors in this experiment and their corresponding effect can be summarized as following:

- i) Due to the effect of the blower used to supply hot-humid air, the vapour partial pressure inside the hot box is expected to be higher than the atmospheric vapour pressure assumed in the model. Consequently, more condensation on the tested glass pane than it should be is expected as indicated by Equation (3.19).

TABLE S-1:
Experimental results for mass condensation rate and the corresponding glass surface temperature rise at different conditions

Test No.	Temperature °C			Rsc/R	Relative humidity RH %	Time min.	Mass Collected gm	Mass Condensation Rate, m kg/s x10 ⁻⁶				Surface Temperature Rise °C						
	Glass	Hot Air	Cold air					Measured	1st Approach		2nd Approach		Measured	1st Approach		2nd Approach		
									Calcul- ated	Diff %	Calcul- ated	Diff %		Calcul- ated	Diff %		Calcul- ated	Diff %
1	27.2	36.6	16.2	.449	91	33	42.3	21.4	25.7	20.1	16.5	-22.9	1.84	2.8	52	2.2	19.6	
2	24	34.2	12.8	.405	94	10.8	14.8	22.8	27.3	19.7	17.3	-24.1	2.5	3.1	24	2.4	-4	
3	24.6	36.6	12.7	.405	90	5.4	8.1	25	29.4	17.6	19.1	-23.6	2.2	3.3	50	2.6	18.2	
4	24.6	34.5	13.1	.405	97	7	10.7	25.5	29.7	16.5	19.2	-24.7	2.6	3.3	17.8	2.6	-7.1	
5	31	41.2	19.6	.422	90	19	28	24.6	30.2	22.8	20.5	-16.7	2.3	3.2	39.1	2.65	15.2	
6	29.2	40	17	.43	90	11.5	18.5	26.8	31.2	16.4	21.1	-21.3	2.3	3.4	48	2.8	21.7	
7	31.4	42.4	19.2	.415	90	13	22.3	28.6	33.6	17.5	23.2	-18.9	4.6	3.6	38	3.0	15.4	
8	32.3	43.1	20.1	.42	90	21	25.3	28	33.9	21.1	23.6	-15.7	2.5	3.6	44	3.0	20.8	
9	30.9	41.3	19	.415	94	14.25	25.5	30	35	16.7	24.3	-19	2.65	3.7	39.6	3.1	17	
10	24.8	38.6	9.4	.423	90	24.75	48.3	32.5	37.8	16.3	25.8	-20.6	3.0	4.2	40	3.5	16.7	
11	28.2	42.1	12.8	.412	86	37.1	75	33.7	38	12.8	26.5	-21.4	3.3	4.1	24.2	3.5	6.1	
12	29.9	40.4	17.2	.41	100	23	50.2	36.4	40.6	11.5	28.7	1.2	3.2	4.4	37.5	3.7	15.6	
13	25.8	40.1	9.7	.423	90	31.5	67	35.4	41	15.8	28.5	-9.5	3.2	4.6	43.7	3.85	20.3	
14	27.7	41	10.3	.423	96	9.7	23.5	40.4	48	18.8	34.5	-14.6	4.4	5.3	20.5	4.6	4.5	
15	30.9	40.8	20.5	.423	86	52.78	67.6	21.3	24.4	14.6	16	-24.8	1.8	2.6	44.4	2.1	16.6	
16	33.6	44.3	22.1	.41	86	26.87	41.6	25.8	29.6	14.7	20.4	-20.9	2.3	3.1	34.8	2.6	13	
17	28	38	16.5	.44	91	27.5	39.9	24.2	28.3	16.9	18.6	-23.1	2.04	3.1	52	2.46	20.6	
18	20.3	31.9	7.1	.43	95	5.7	8.7	25.7	29.6	16.5	18.6	-26.8	2.54	3.4	33.8	2.6	2.4	

- ii) The actual glass surface temperature is less than the theoretical value assumed by the model, due to temperature gradient from the measuring point at the thermocouple towards the glass pane. Therefore, more condensation is expected than what is predicted by the theoretical model.
- iii) At high relative humidity levels condensation may occur at other surfaces rather than the glass pane. This condensed water may become part of the collected water, so the measured mass condensation rate is expected to be more than what is predicted by the theoretical model. However, as a precaution not to collect water condensed on the internal surfaces of the hot box a wooden strip was fixed and sealed to the bottom side of the box near the collecting channel to prevent water condensed on the box surfaces to drip into the collecting channel.
- iv) Difficulty to maintain a perfectly constant relative humidity level inside the box, where it could fluctuate above and below the intended humidity level. So, the measured condensation rate could be more or less than what is expected by the theoretical model. However, in this experiment, the deviation is kept as small as possible by constantly monitoring the relative humidity readings, and by regularly adjusting the amount of steam supply.

- v) Condensation may occur on the humidity sensor at high relative humidities, which leads to higher humidity reading than the actual value. Consequently, the measured mass condensation rate is expected to be less than what is predicted by the theoretical model. However, to avoid this type of error, the humidity sensor response to a different relative humidity level is tested. When the relative humidity inside the box is high, the humidity sensor is taken out of the box and exposed to the room lower humidity level. If the humidity sensor responded to the humidity variation by giving instant reading to the new humidity level, it can be recognized that no condensation has occurred on the sensor.
- vi) Air temperature near the glass surface in the hot box is less than the measured air temperature, so for the same moisture input the relative humidity near the glass is expected to be more. Consequently, there is more condensation potential than what is predicted by the theoretical model. However, to minimize the difference between the measured and the actual relative humidity near the glass surface, the humidity probe was located as near to the glass surface as practically possible.

By evaluating the effect of each factor of variation and its relative importance and contribution, it can be clearly recognized that the measured mass condensation rate has to be more than what is

predicted by any theoretical model for the same conditions. At the same time, it can be seen that the measured mass condensation rate is more than what it should be under normal atmospheric conditions. Therefore, the second approach, which gives a prediction of about 20% less than the measured values, is theoretically and practically justified over the first approach.

The evaluation of the surface temperature rise and the resulting heat gain are not directly related to the approach used. However, the surface temperature rise is directly proportional to the amount of mass condensation rate which is determined by the type of approach. Regardless of the approach used to evaluate mass condensation rate, only one approach is used in this study to evaluate the heat transfer through glass windows and the nodal temperatures as explained in Chapter IV. By comparing the measured surface temperature rise shown in Table 5-1 with the temperature rise predicted by the model, it can be seen that generally the model overestimates the temperature rise when either the first or the second approach is used to predict mass condensation rate. The deviation between the measured temperature rise and what is predicted by the model when the second approach is used is much less than what is predicted by the model when the first approach is used. In order to be consistent with the mass condensation rate deviation, the measured temperature rise has to be higher than what the model predicts when the second approach is used. However, this deviation and the lower measured surface temperature rise

can be attributed to the following factors:

- i) In reality, the glass surface temperature is not uniform, and the measured glass surface temperature is taken as the average of about 24 thermocouple readings.
- ii) The pattern of deposited condensed water on the glass surface shown in Fig. 5-10, where droplets over the water film can be seen, offers more resistance to condensation heat transfer than what is theoretically assumed. Therefore, less temperature rise is expected.
- iii) Part of the collected water may have not condensed on the glass pane. So, the released heat does not contribute to raising the surface temperature.
- iv) Higher surface heat transfer coefficient than what is theoretically assumed in the cold side due to the effect of the blower may result in lower surface temperature rise.
- v) Condensation does not occur uniformly over the glass pane, therefore, some parts of the glass pane experience higher temperature rise than others.

From the above discussion it can be recognized that the surface temperature rise associated with the mass condensation rate predicted by the second approach is practically and theoretically more acceptable than that associated with the mass condensation rate predicted by the first approach.



FIG. 5-10: Condensation Pattern on the Glass Pane

By using the second approach to predict mass condensation rate on glass windows, the resulting heat loss and heat gain can be easily evaluated at different indoor and outdoor conditions. The heat gain rate resulting from condensation on the external surfaces of single glazed windows can now be easily predicted by using the appropriate curve in Figs. 5-11 to 5-14. Similarly, the heat loss rate due to indoor condensation on single glazed windows can now be predicted by using the simple curves shown in Figs. 5-15 to 5-20.

The applicability of the second approach over the first approach in predicting mass condensation rate is also valid for double glazed windows. However, the surface temperature, which is very much

affected by the overall thermal resistance, determines how much condensation occurs. Therefore, in order to test the applicability of the whole model including the assumed temperature gradient, a double glazed window has to be tested. However, testing of a double glazed windows was not considered in this study for the following reasons.

- i) To compare the two approaches and their applicability which is the main objective of this experiment, either a single or a double glazed window may be selected to be tested. so, a single glass pane was chosen since, it is more suitable and more practical for testing.
- ii) Double glazed windows vary in their thermo-physical properties depending on the air space thickness and the type of seal, so the testing results will be applicable only for the tested window type.
- iii) Difficulty to produce sufficiently low temperature in the cold box so that condensation could occur on the other side of the window.
- iv) It is not feasible in term of time, effort and cost to experimentally test a double glazed window for the temperature gradient when only heat loss due to condensation is considered, while heat gain, which is dependent on the determination of the surface temperature rise, is negligible.

Based on the results obtained for the single glazed window, and because of the similarity between single and double glazed windows in evaluating the assumed temperature gradient, it is feasible to assume that the second approach with the assumed temperature gradient for double glazed windows is applicable to predict mass condensation rate and the resulting heat loss when indoor surface condensation occurs. Yet, this assumption is to be experimentally verified in future studies. The heat loss rate predicted by the theoretical model when indoor surface condensation occurs on a double glazed window with 8 mm air space can be easily found at certain conditions using the appropriate curve in Fig. 5-21 to 5-26.

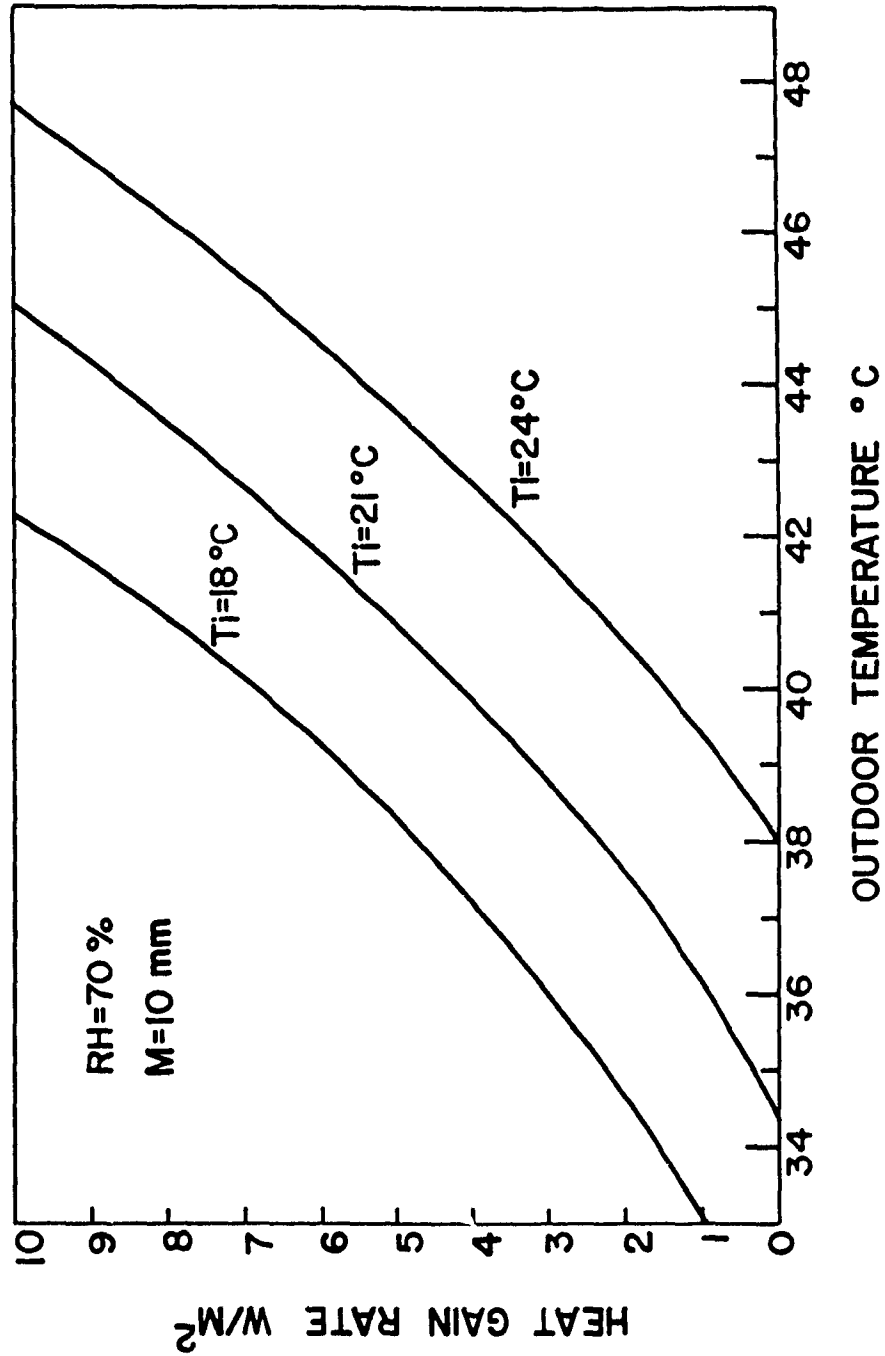


FIG. 5-11: Heat gain rate due to condensation on the external surface of a single glazed window, with outdoor relative humidity of 70%

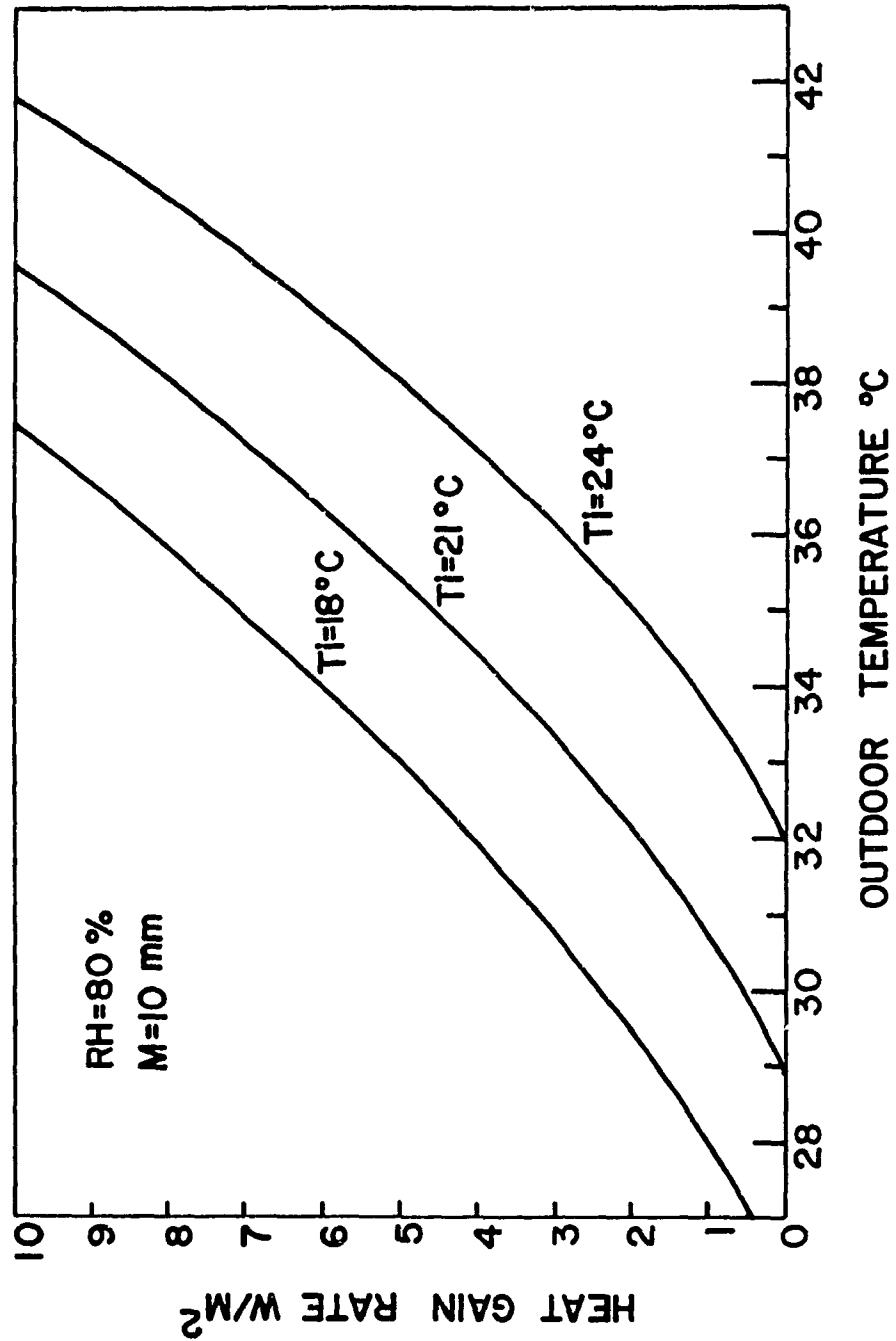


FIG. 5-12: Heat gain rate due to condensation on the external surface of a single glazed window, with outdoor relative humidity of 80%

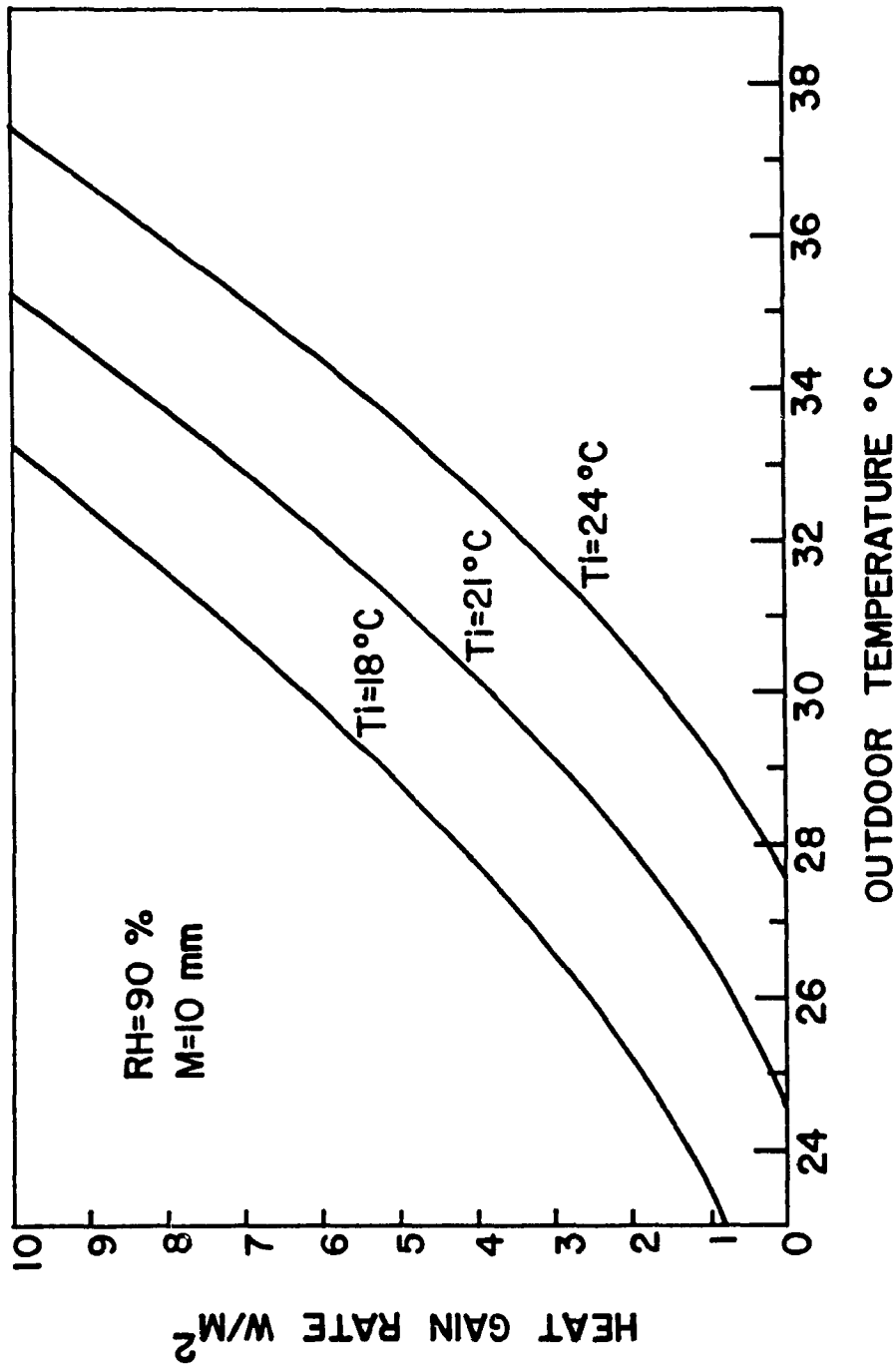


FIG. 5-13: Heat gain rate due to condensation on the external surface of a single glazed window, with outdoor relative humidity of 90%

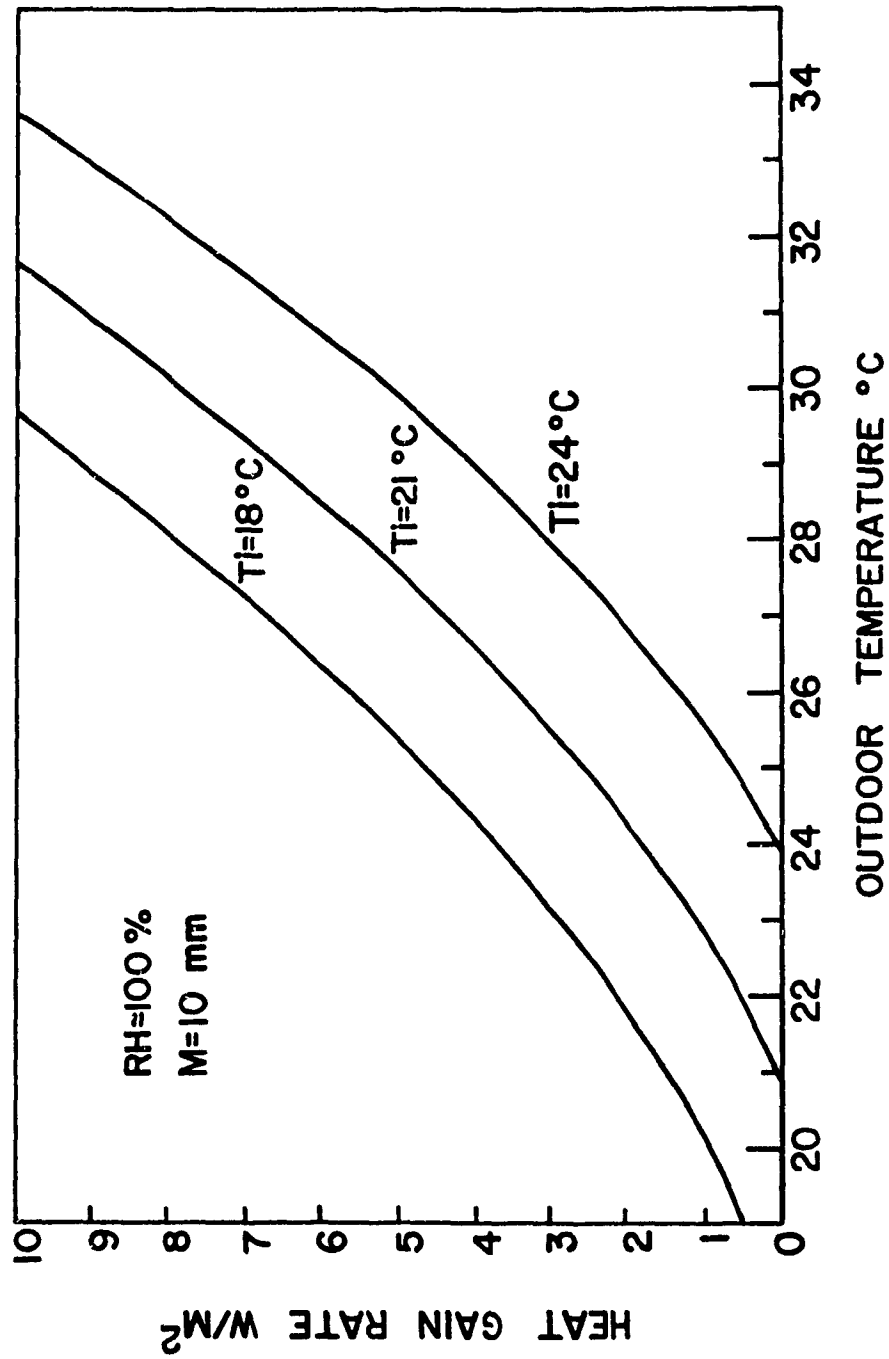


FIG. 5-14: Heat gain rate due to condensation on the external surface of a single glazed window, with outdoor relative humidity of 100%

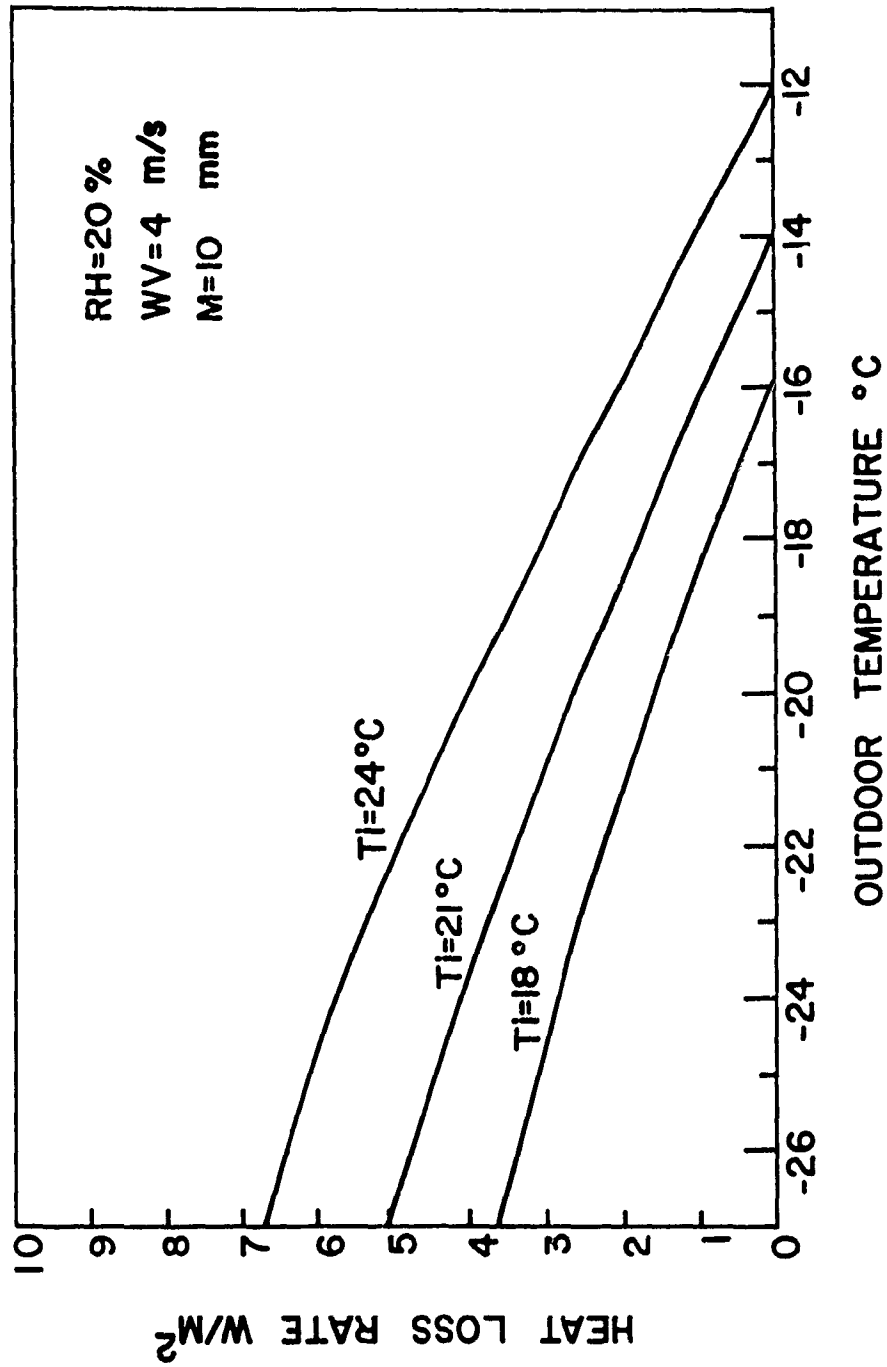


FIG. 5-15: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 20%, and 4 m/s wind speed

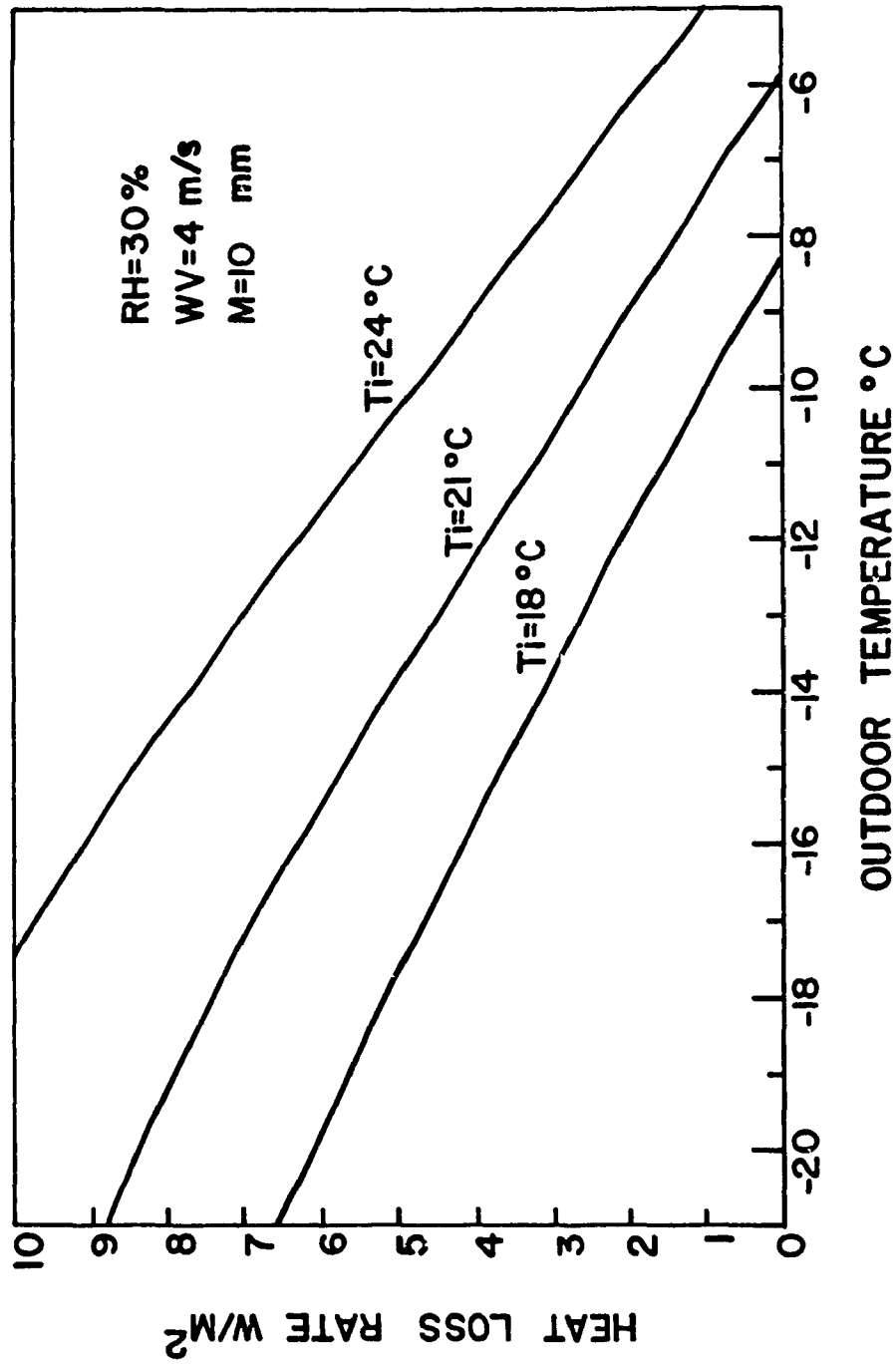


FIG. 5-16: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 30%, and 4 m/s wind speed

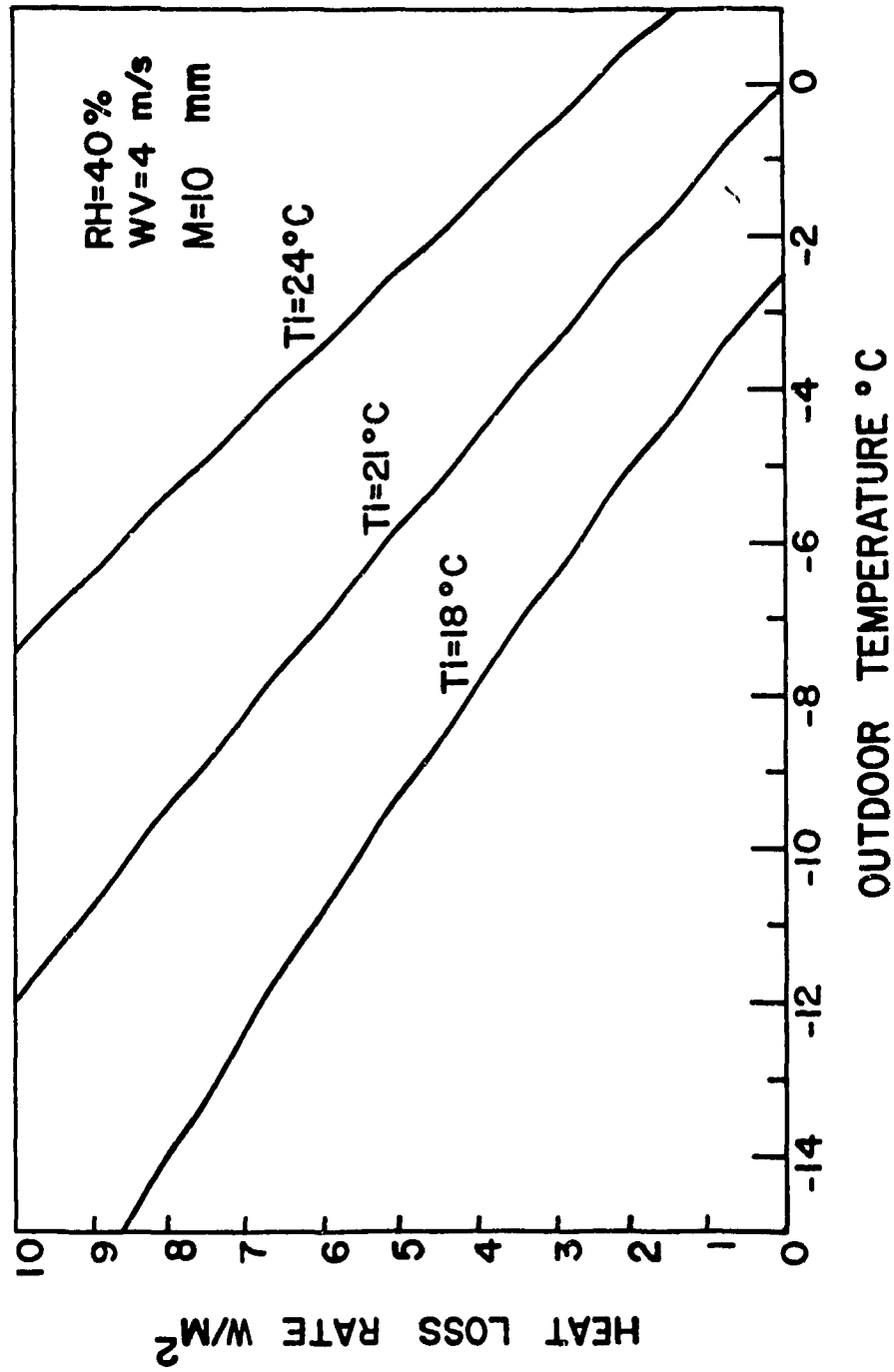


FIG. 5-17: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 40%, and 4 m/s wind speed

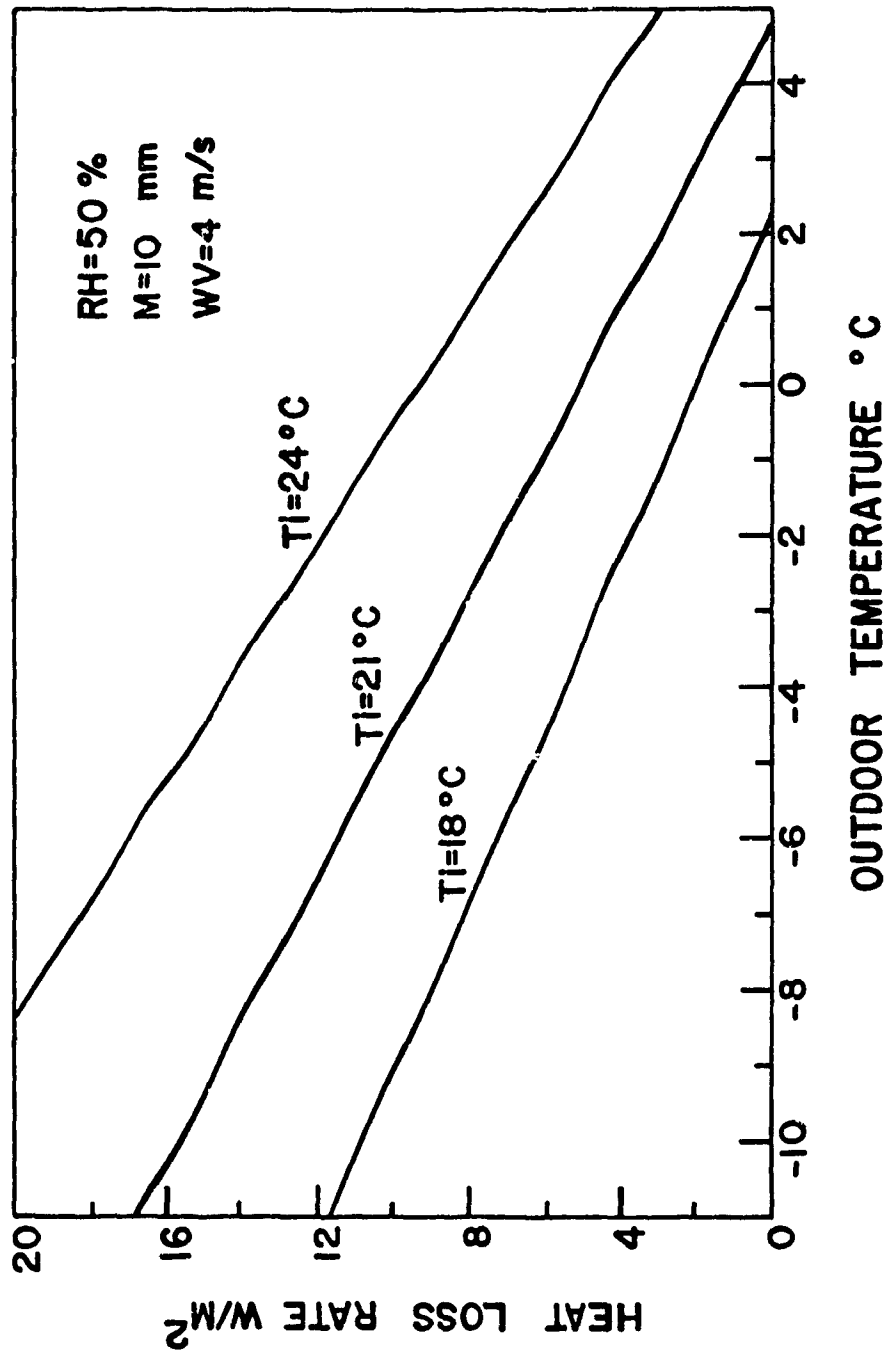


FIG. 5-18: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 50%, and 4 m/s wind speed

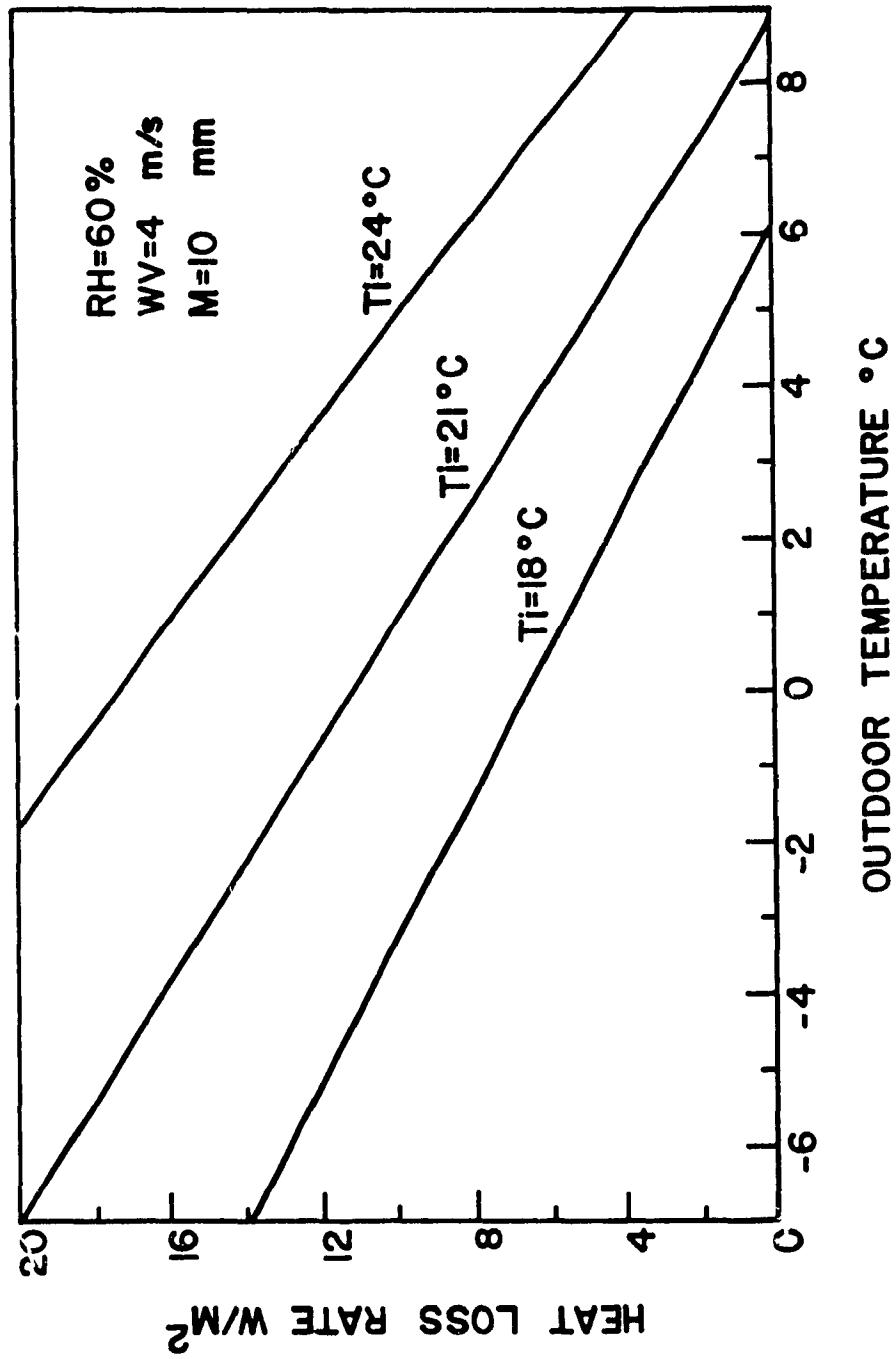


FIG. 5-19: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 60%, and 4 m/s wind speed

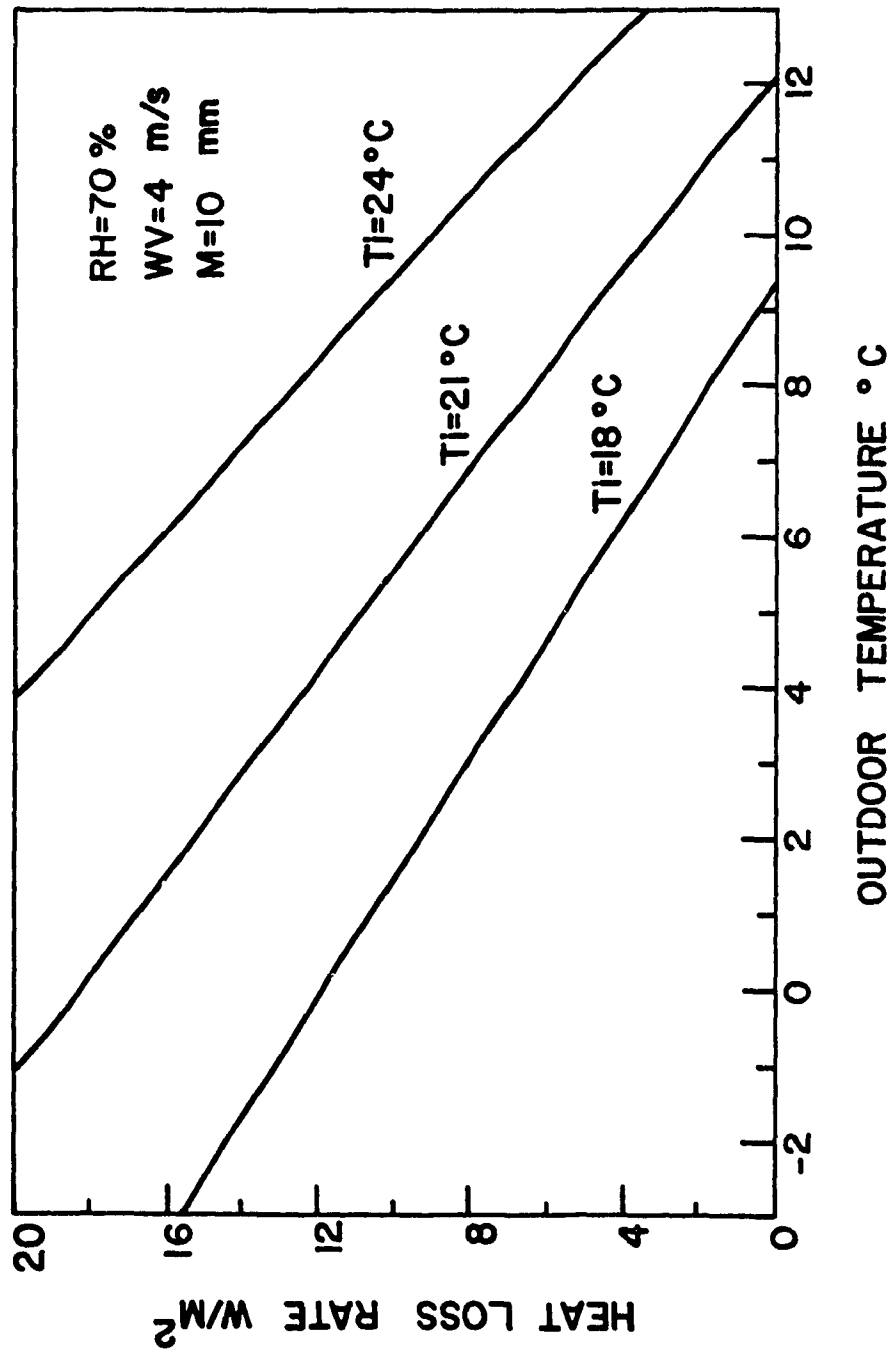


FIG. 5-20: Heat loss rate due to condensation on the internal surface of a single glazed window, with indoor relative humidity of 70%, and 4 m/s wind speed

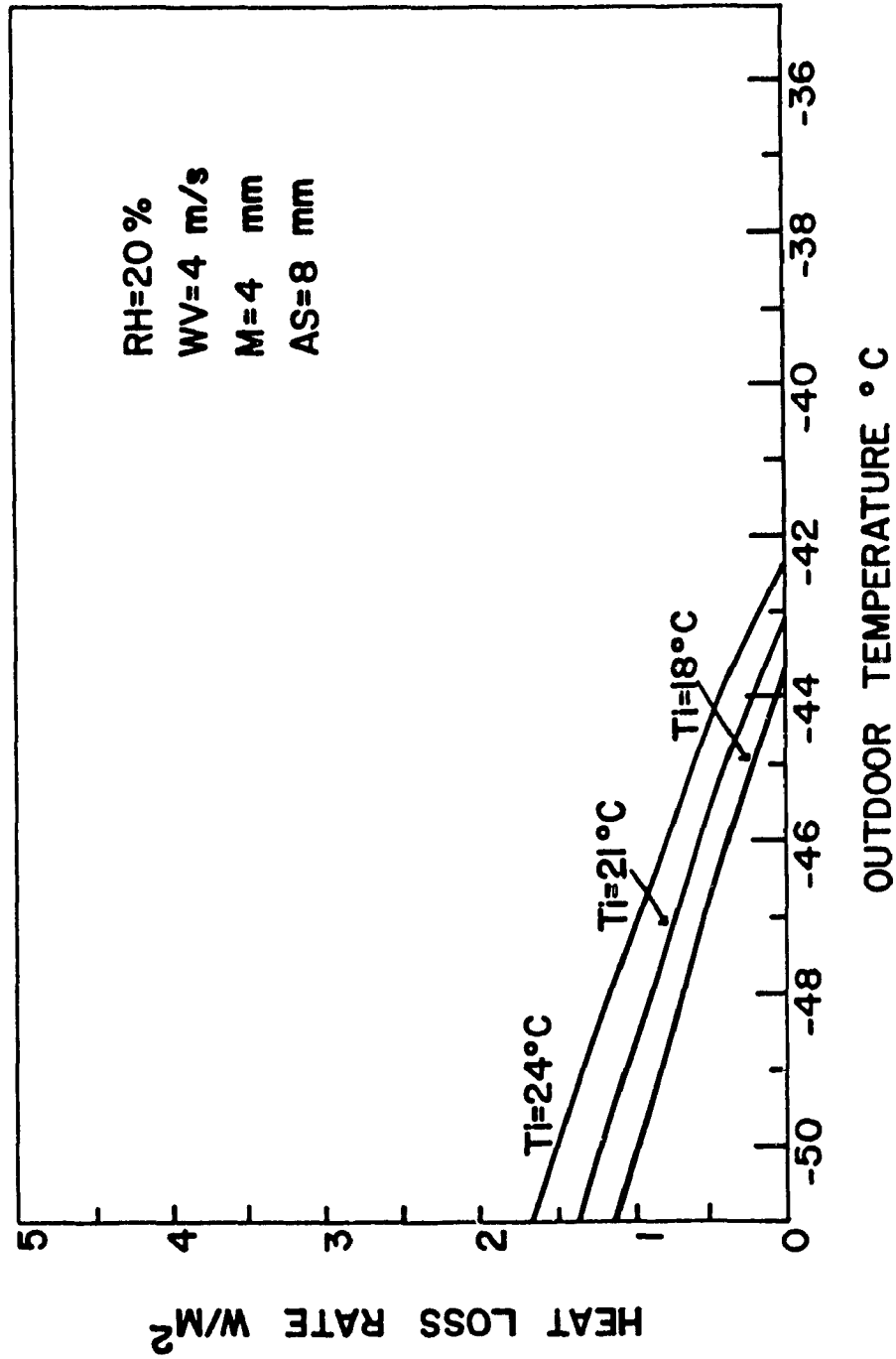


FIG. 5-21: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 20%, and 4 m/s wind speed

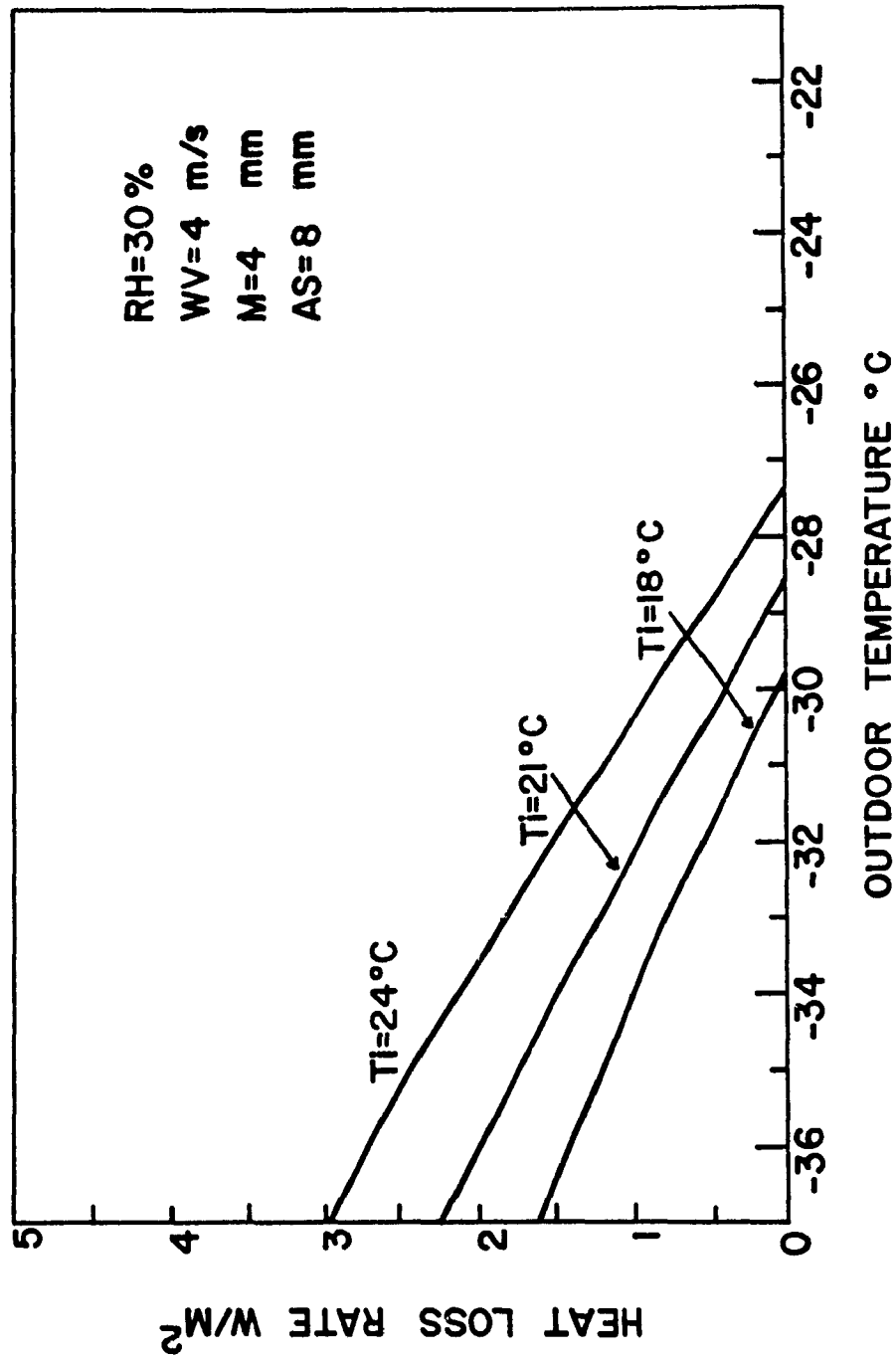


FIG. 5-22: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 30%, and 4 m/s wind speed

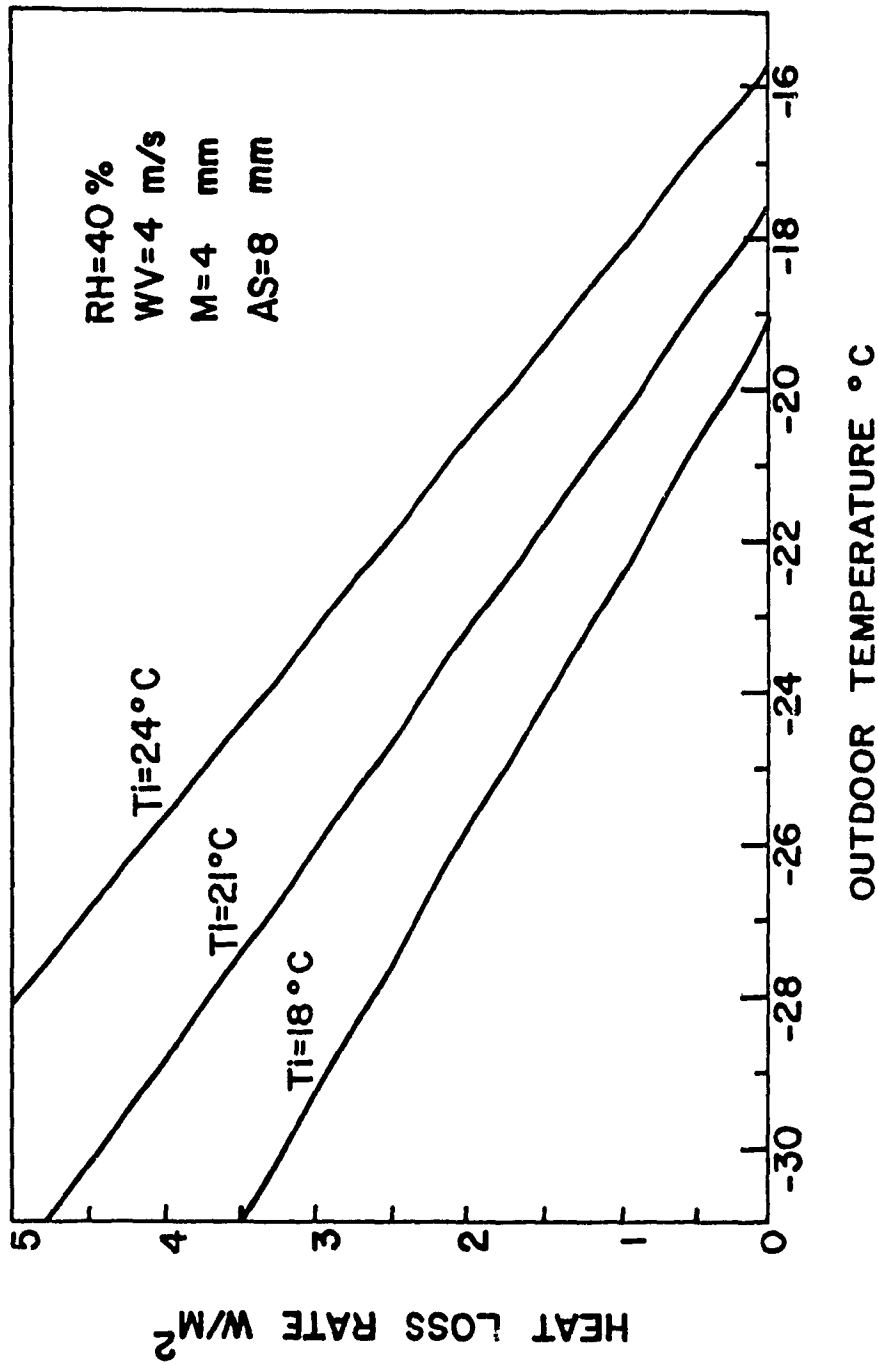


FIG. 5-23: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 40%, and 4 m/s wind speed

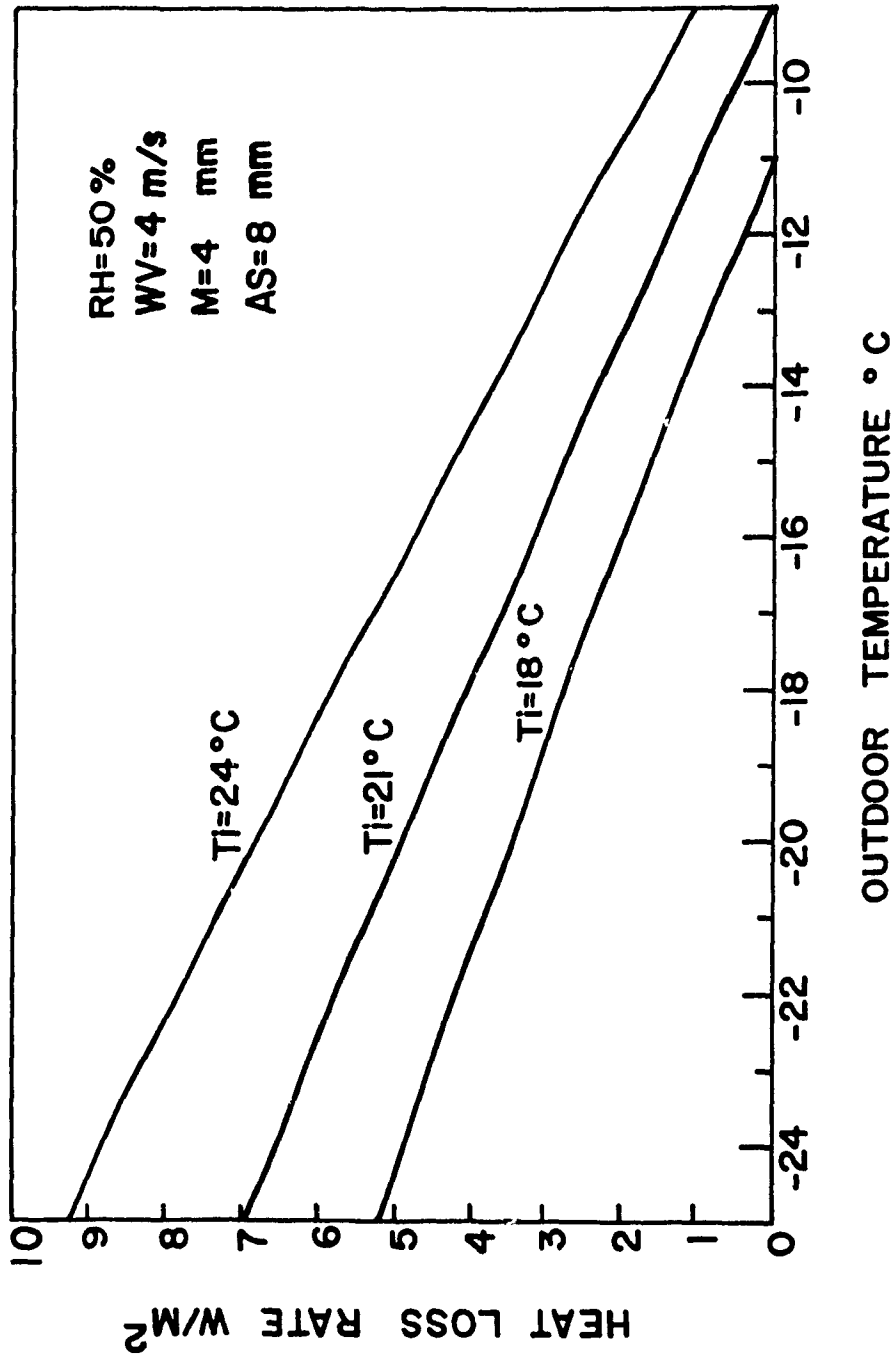


FIG. 5-24: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 50%, and 4 m/s wind speed

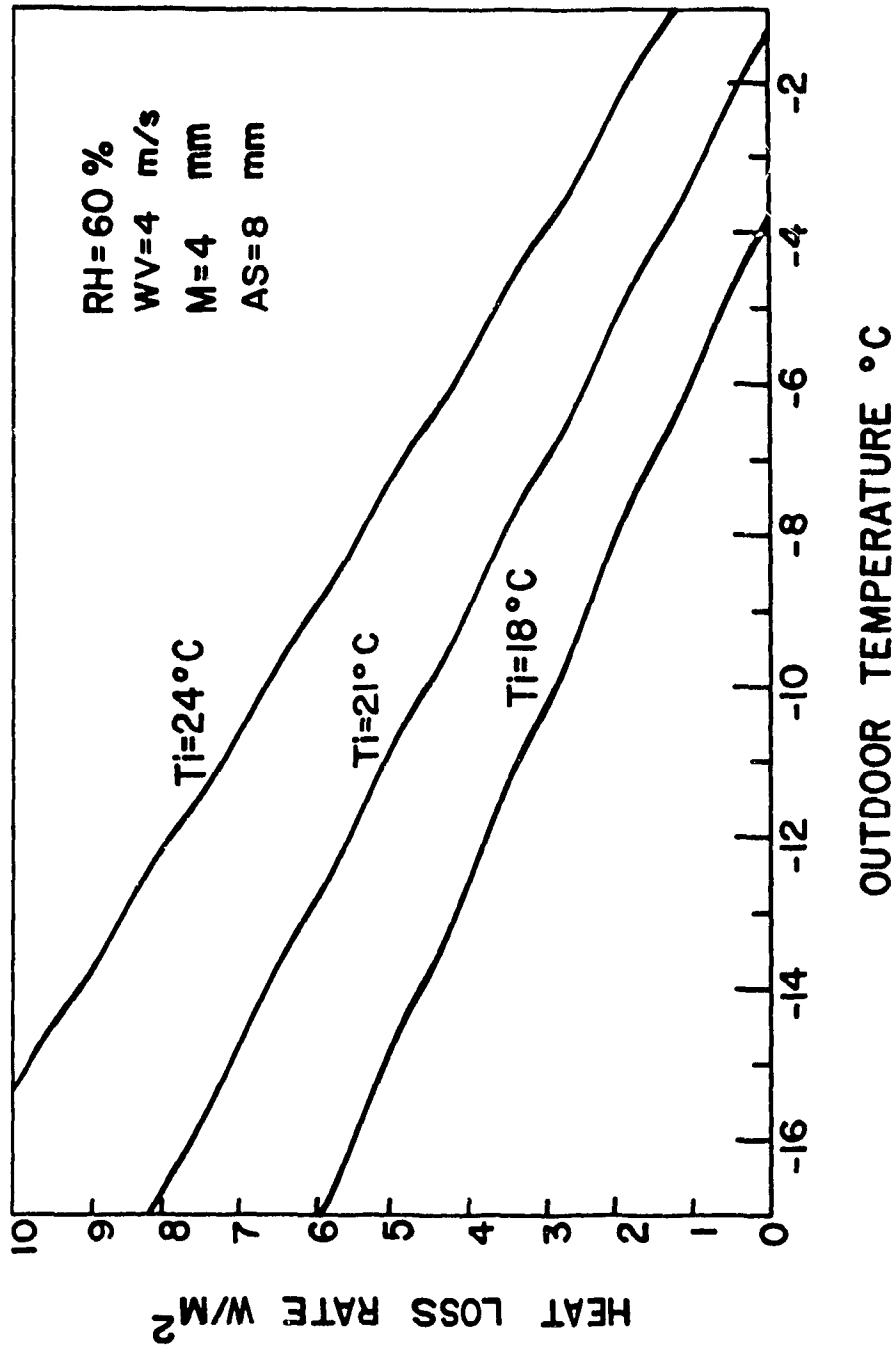


FIG. 5-25: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 60%, and 4 m/s wind speed

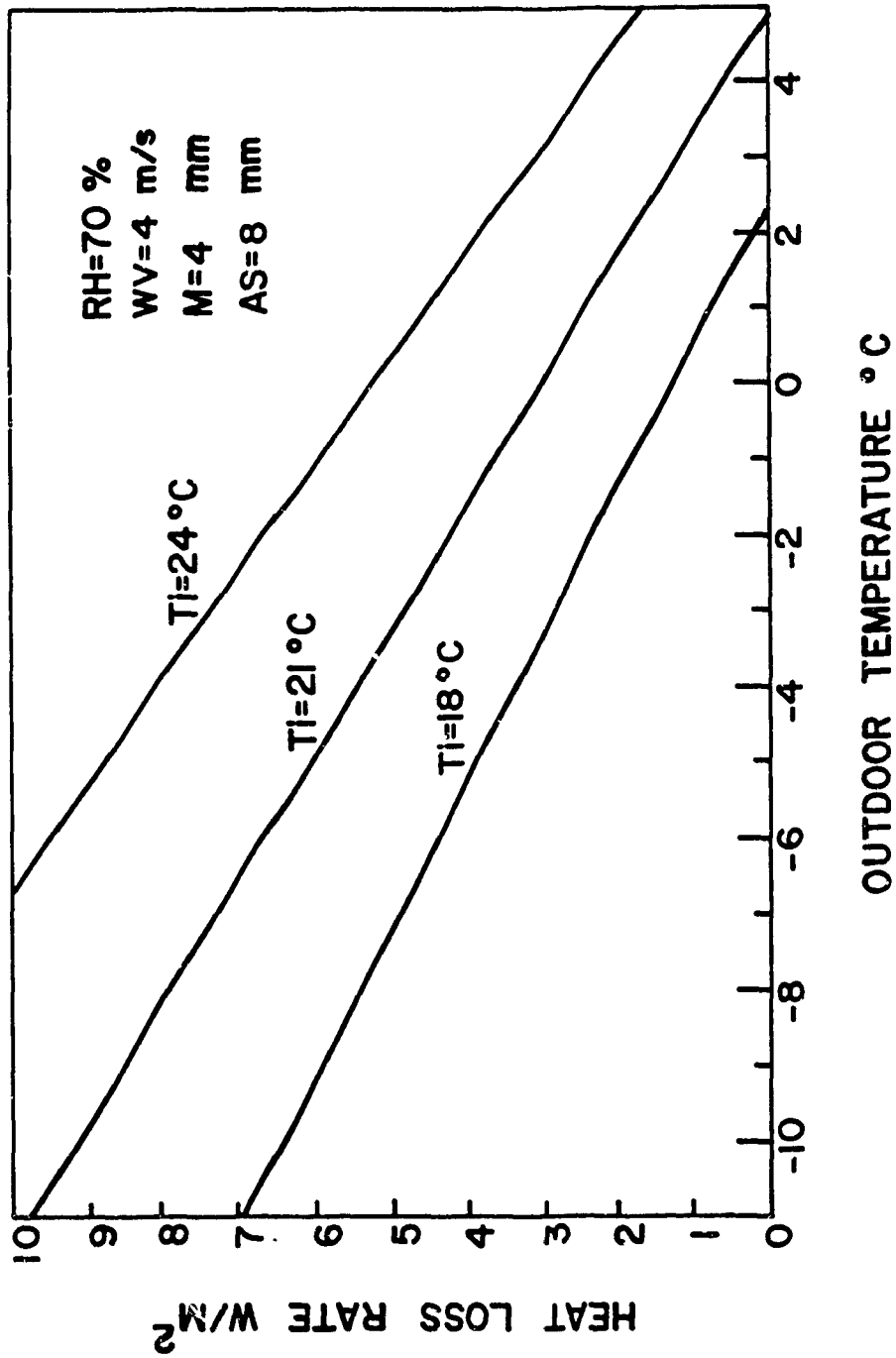


FIG. 5-26: Heat loss rate due to condensation on the internal surface of a double glazed window, with indoor relative humidity of 70%, and 4 m/s wind speed

CHAPTER 6

SUMMARY, CONCLUSIONS AND RECOMMENDATION FOR FUTURE STUDIES

Condensation in buildings has been a source of many problems ranging from merely inconvenience to severe damage of building envelope components. Glazed windows, and in particular single glazed windows, have the highest potential to remove a lot of water from the air, since they are usually the coldest surfaces in the building envelope. Water vapour condensation is associated with the release of considerable amount of latent heat. When condensation occurs on the external surface of a single glazed window, good percentage of the released heat is conducted through the window, because of its low thermal resistance, and becomes a source of additional heat gain. Condensation on internal surfaces of glazed windows is associated with moisture loss from the indoor air. Substituting the lost moisture, so as to maintain the space at a constant humidity level, requires energy. Thus heat is indirectly lost as moisture when indoor surface condensation occurs.

Heat gain due to external surface condensation on single glazed windows could be a considerable source of additional heat gain in hot-humid climates. Heat loss due to internal surface condensation on glazed windows could also be a considerable means of additional heat loss in cold climates. Nevertheless, there has been no systematic approach available in related literature to predict heat gain or loss due to surface condensation in buildings.

In this study a mathematical model has been developed to predict the temperature gradient across the window as condensation occurs, the mass condensation rate, and the corresponding heat gain or loss at any given conditions. The temperature gradient across single and double glazed windows has been evaluated using a transient unidimensional finite-difference formulation. The resulting heat balance equations have been expressed in the fully implicit form, so that large time intervals can be used to minimize the computational time. By expressing the heat balance equations in the implicit form, they have to be solved simultaneously in order to evaluate nodal temperatures at any time interval. Yet, to solve these equations, the mass condensation rate at the beginning of each time interval has to be known.

Evaluating mass condensation rate is the main and most important step in predicting heat gain or loss due to surface condensation in buildings. In this study, two different approaches have been used to evaluate mass condensation rate on glazed windows. The concept of the first approach has been derived from a procedure suggested by Davies, M. (16), which is based on the fact that mass transfer occurs as a result of difference between the bulk vapour pressure and the vapour pressure at the cool surface. By utilizing the heat balance and mass transfer equations, Davies was able to graphically evaluate the vapour pressure difference and solve for mass condensation rate. In this procedure the window thermal resistance is neglected and the window is assumed at a uniform temperature. So, the internal surface temper-

ature rise, required to evaluate the resulting heat gain, cannot be evaluated correctly. Among the drawbacks of this procedure is that a graphical solution must be obtained each time the mass condensation rate is to be calculated at a given set of conditions. In order to properly evaluate the internal surface temperature rise and the resulting heat gain, the window thermal resistance has been considered in this study. In addition, a transient unidimensional finite-difference formulation has been utilized to solve for heat transfer across the window when condensation occurs. Consequently, a graphical solution is not needed to solve for mass condensation rate. Instead, the vapour pressure difference and the corresponding mass transfer rate are evaluated at the beginning of each time interval until steady state stage is reached. In brief, the present study has expanded Davies procedure so that it becomes unnecessary to create graphical solution each time the mass condensation rate is calculated. Moreover, this study expansion allows time-dependent analysis to be performed.

In order to evaluate the relative accuracy and applicability of the first approach in predicting mass condensation rate of atmospheric vapour, a second approach has been developed in this study. In this approach the basic concept of Nusselt's theory of film condensation has been utilized. Nusselt's theory has the advantage over the mass transfer theory of being more accurate in describing the condensation process on vertical surfaces, since it accounts for the effect of the water film and the surface height on the heat transfer process. Yet,

almost all condensation analyses in buildings follow the conventional mass transfer procedure. Nusselt's theory has never been applied in building condensation, since it is limited to pure vapour condensation. In this study, Nusselt's theory has been modified to account for the presence of a large percentage of the noncondensable gas (which is air) to be applicable for atmospheric vapour condensation on building surfaces. The aim of the modified analysis is to evaluate the interfacial temperature (T_{gi}) of the condensate film to replace the saturation temperature (T_{sat}) in the equations suggested by Nusselt's. There has been several studies to evaluate the interfacial temperature in the presence of a noncondensable gas. However, the results of these studies are only applicable for mixtures containing very small amounts of the noncondensable gas ranging from 1% to 5%. The same analysis has been utilized in this study to solve for the interfacial temperature for mixtures containing high percentage of the noncondensable gas so that the results can be utilized to predict the interfacial temperatures when atmospheric vapour condensation occurs on building vertical surfaces.

Heat gain or loss due to surface condensation is evaluated when the condensation process reaches the steady state stage. Depending on the conditions and the value of time interval, the number of intervals needed to reach steady state stage could reach hundreds. At the beginning of each time interval the heat balance equations have to be solved simultaneously to evaluate the nodal temperature. In addition,

the mass condensation rate has to be evaluated by either approach at the beginning of each time interval. Therefore, it is not practical and almost impossible to manually solve for heat gain or loss due to surface condensation by this model. In order to make the mathematical model more practical and ready to be used by building designers, a computer program written in Quick Basic has been developed. This program mainly comprises six subprograms by which heat gain due to external surface condensation on single glazed windows, and heat loss due to internal surface condensation on single and double glazed windows can be evaluated using both approaches to predict mass condensation rate. The program outputs have shown that mass condensation rate predicted at the steady state stage by the second approach is always less than that predicted by the first approach at the same conditions.

In order to evaluate the applicability of each approach in predicting mass condensation rate, a special preliminary experimental set up has been developed. In this experiment, the mass condensation rate and the corresponding surface temperature rise of a single glazed window were measured at different conditions simulating indoor and outdoor conditions. The experimental set up mainly comprises; two chambers, cooling unit, heating box, humidity generator, single glazed pane, and temperature and humidity measurement instruments. By measuring the mass of the condensed water and the time needed for condensation, the mass condensation rate can be measured at different

temperatures and relative humidities. Over fifty experimental measurement of mass condensation rate and the rise in glass surface temperature were obtained at different conditions. The measured mass condensation rate was always less than what is predicted by the first approach and more than what is predicted by the second approach. However, due to the difficulty in keeping constant testing conditions in both chambers and possible errors in taking the readings, there is no consistency in the degree of deviation between all the measured and the corresponding theoretical values of mass condensation rates. By implementing a selection criterion, about half of the measured values were found to show a reasonable degree of consistency with the variation of the theoretical values. The measured mass condensation rates were mainly found to be about 16% less than what is predicted by the first approach, and about 20% more than what is predicted by the second approach. By evaluating the effect of each possible factor of deviation and its relative importance and contribution, it was found that the measured mass condensation rate has to be more than what is predicted by the theoretical model for the same conditions. Therefore, the second approach, which gives prediction of about 20% less than the measured values, is theoretically and practically justified over the first approach. In addition, the internal surface temperature rise predicted by the model, when the second approach is used, was found more acceptable when compared with the measured values.

By using the second approach to predict mass condensation rate on glazed window, the resulting heat gain or loss due to condensation on

selected single and double glazed windows can be easily estimated using the appropriate curve from Figs. 5-11 to 5-26. Figs. 5-11 to 5-14, can be used to evaluate heat gain due to external surface condensation on a single glazed window for the outdoor relative humidity range from 70% to 100%. Heat loss due to internal surface condensation on a single glazed window for the indoor relative humidity range from 20% to 70% can be predicted using Figs. 5-15 to 5-20. Finally, heat loss due to internal surface condensation on a double glazed window of 8 mm air space for the indoor relative humidity range from 20% to 70% can be predicted using Figs. 5-21 to 5-26. From these figures, it can be seen that higher relative humidity and temperature in the condensing side and lower temperature in the other side leads to higher surface condensation heat gain or loss. In addition, it can be seen that heat loss due to internal surface condensation on a single glazed window is several times more than that experienced when condensation occurs on a double glazed window at the same conditions. For example, at 60% indoor relative humidity, 21°C indoor temperature and -7°C outdoor temperature, the heat loss due to surface condensation is about 20 W/m^2 for a single glazed window and only 3 W/m^2 for a double glazed window. The importance of heat loss due to condensation on double glazed windows is increased as the indoor relative humidity gets higher and the outdoor temperature gets lower. But, it is generally negligible at low indoor relative humidity. In fact, the importance of either heat gain or loss due surface condensation can be evaluated by comparing the fabric gain or loss (due to temperature

difference) with condensation heat gain or loss at the same conditions. Higher relative humidity in the condensing side increases the importance of condensation heat gain or loss. For example, at 80% outdoor relative humidity, 18°C indoor temperature and 30°C outdoor temperature, heat gain due to condensation is about 5% the conduction heat gain, while at 100% outdoor relative humidity heat gain due to condensation is about 22% the conduction heat gain. Similarly, higher indoor relative humidity increase the importance of heat loss due to condensation on single and double glazed window as compared to the fabric heat loss.

The outdoor surface condensation and consequently the resulting heat gain can be prevented by keeping the external surface temperature above the dewpoint temperature of the ambient air. This can be achieved by either using double glazed windows or keeping the indoor air temperature as high as possible. However, in hot-humid climates, where outdoor surface condensation is more common, the average indoor-outdoor temperature difference through out the year may not be large enough to justify the use of double glazed windows. Therefore, keeping the indoor temperature as high as possible remains the only option to reduce the risk of external surface condensation and the associated heat gain.

In order to prevent or at least reduce the heat loss due to condensation on windows internal surfaces the following recommendations

are proposed:

- i) Single glazed windows should not be considered as a design option in cold climates, especially when high indoor relative humidity is a functional requirement of the space.
- ii) In cold climates, the controlled indoor relative humidity has to be maintained as low as comfort and functional requirement would permit.
- iii) Indoor air temperature has to be kept as low as comfort would permit.

As a continuation of this study, it is recommended to apply the same condensation theory to other low thermal resistance exterior panels in buildings. Moreover, it is required to experimentally verify the assumed theoretical model for double glazed windows, and to investigate the effect of wind velocity on the condensation process. The effects of surface texture, on which condensation occurs, and surface inclination on mass condensation rate and the resulting heat transfer are also required to be further investigated. Future studies should account for variation of humidity level near the window when condensation occurs, especially, when indoor surface condensation is considered. The model can be extended so that solar radiation and its effect on window temperature and consequently on the condensation/evaporation process can be taken into consideration. Finally, multi-dimensional analysis may be applied to examine mass condensation rate near edges and corners, and condensation variation along the window height.

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A P P E N D I C E S

APPENDIX A

**A COMPUTER PROGRAM TO PREDICT HEAT GAIN OR
LOSS DUE TO CONDENSATION ON GLAZED WINDOWS**

```
11 PRINT
12 PRINT
13 PRINT
14 PRINT"THIS PROGRAM CALCULATES THE MASS CONDENSATION RATE ON SINGLE AND
DOUBLE GLAZED WINDOWS USING TWO APPROACHES. IN ADDITION IT CALCULATES THE
RESULTING HEAT GAIN OR LOSS DUE TO CONDENSATION"
15 PRINT
16 INPUT"PRESS ENTER TO CONTINUE ELSE PRINT (F) ",F$
17 IF F$="F" OR F$="e" THEN 380
18 PRINT
19 INPUT"type of window double (d) or single (s)";A$
20 PRINT
21 IF A$="D" OR A$="d" THEN 120 ELSE 140
22 PRINT"note: only indoor condensation is considered for double glazed windows
-----"
23 IF A$="D" OR A$="d" THEN 210
24 INPUT"indoor condensation (i) or outdoor condensation (o)";B$
25 PRINT
26 IF A$="s" AND B$="o" THEN 180 ELSE 170
27 IF A$="S" AND B$="O" THEN 180 ELSE 210
28 INPUT"variable conditions (v) or constant conditions (c)";D$
29 IF D$="v" OR D$="V" THEN RUN "arwa0"
30 IF D$="c" OR D$="C" THEN 210
31 PRINT
32 INPUT"first approach (1) or second approach (2)";C
33 PRINT
34 IF A$="D" OR A$="d" THEN 250 ELSE 280
35 IF C=1 THEN RUN "arwa3"
36 IF C=2 THEN RUN "mona2"
37 REM
38 IF A$="S" AND B$="I" THEN 300 ELSE 290
39 IF A$="s" AND B$="i" THEN 300 ELSE 330
40 IF C=1 THEN RUN "arwa2"
41 IF C=2 THEN RUN "mona1"
42 REM
43 IF A$="S" AND B$="O" THEN 350 ELSE 340
44 IF A$="s" AND B$="o" THEN 350 ELSE 380
45 IF C=1 THEN RUN "arwa1"
46 IF C=2 THEN RUN "mona"
47 STOP
48 END
```

SUB-PROGRAM 1

```

10 DIM T(3,1010),A(3),B(3),C(3),F(3),CINAV(500),F2(8),MCR(505),TOUT(510),
    TI(510),PA(510)
20 REM
30 INPUT"this subprogram calculates the mass condensation rate and the resulting
.....heat gain for a single glazed window when outdoor condensation occurs a
.....at variable conditions if you want to use it print (y) else print (n)";B;
40 REM
50 IF B$="y" THEN GOTO 80 ELSE RUN"cond"
60 REM
70 REM *****keenan, keyes, hill, and moore formula constants*****
80 F2(1)=-741.9242
90 F2(2)=-29.721
100 F2(3)=-11.55286
110 F2(4)=-.8685635
120 F2(5)=.1094096
130 F2(6)=.439993
140 F2(7)=.2520658
150 F2(8)=.0521868
160 REM *****input of atmospneric, indoor conditions and window related data****
170 REM
180 PRINT"
190 INPUT"average indoor temperature in c=";TIAVG
200 PRINT"
210 INPUT"original ambient temperature in c=";TOUTO
220 PRINT"
230 INPUT"original outdoor relative humidity=";PHO
240 PRINT"
250 INPUT"window (OR WALL) height in meters=";L
260 PRINT"
270 INPUT"glass thickness in meters=";M
280 PRINT"
290 INPUT"time interval in seconds=";DT
300 PRINT"
310 GOSUB 1870
320 REM the average outdoor(toa), indoor(tia) and wall temperatures(twa, twi)
    will be assumed as following
330 REM
340 TCA=TOUTO
350 TIA=TIAVG
360 TWCA=TOA-5
370 TWIA=TIA+6
380 TOAK=TOA+273
390 TIAK=TIA+273
400 TWAOK=TWCA+273
410 TWIAK=TWIA+273
420 REM the air properties are evaluated at the assumed average film
    temperatures which is equal to 28 deg. c for outdoor conditions
    and 21 deg. c for indoor conditions
430 REM
440 MEUI=1.4937E-05
450 ALFAI=.0000211
460 KAI=.025764
470 MEUO=1.568E-05
480 ALFAO=2.216E-05
490 KAO=.02624
500 TFIIMI=(18+24)/2+273
510 TFIIMO=(25+30)/2+273
520 BETAI=1/TFIIMI
530 BETAO=1/TFIIMO
540 SEGMA=5.673E-08
550 FSR=1
560 PAI=9.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
570 PAO=9.8*BETAO*L^3*(TCA-TWCA)/(MEUO*ALFAO)
580 IF PAI<1E+09 THEN HCI=.59*KAI*(PAI^-.25)/L
590 IF PAI>1E+09 THEN HCI=.1*KAI*(PAI^-.333)/L

```



```

560 IF PAO<1E+09 THEN HCO=.59*KAO*(PAO^.25)/L
570 IF PAO>1E+09 THEN HCO=.1*KAO*(PAO^.333)/L
580 HPO=SEGMA*FSR*(TOAK^2+TWOAK^2)*(TOAK+TWOAK)
590 HPI=SEGMA*FSR*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
600 HI=HCO+HPI
610 HO=HCO+HPO
620 KGL=.78
630 DX=M/2
640 P1=1/HO
650 P2=(1/KGL*DX)+P1
660 P3=(1/KGL*M)+P1
670 R=P3+1/HI
680 T(3,1)=TOUTO-(P1/P)*(TOUTO-TIAVG)
690 IF T(3,1)>=TSAT-1 THEN GOTO 740 ELSE 760
700 PRINT "condensation will not occur at these conditions, start again if you
wish to try at other conditions else press ctrl-break"
710 GOTO 190
720 GOSUB 2050
730 REM water and vapour properties are evaluated at assumed outdoor average
740 REM temperatures
750 REM
760 ROWL=993
770 ROWV=.7
780 HFG=2500000
790 KL=.62
800 MEUL=.00007
810 CV=840
820 PCW=2700
830 T(2,1)=TOUTO-(P2/P)*(TOUTO-TIAVG)
840 T(1,1)=TOUTO-(P3/P)*(TOUTO-TIAVG)
850 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^-.25
860 D3=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^-(1/3)
870 D4=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^-.25
880 D5=(L*(TSAT-T(3,1))*L)^-.75
890 MW=18
900 PC=8310
910 DC=.000026
920 ROWA=1.15
930 CVA=1005
940 HD=((DC*HCO)*((KAO/(ROWA*CVA*DC))^-.25))/KAO
950 FO=(KGL*DT)/(ROW*CV*DX^2)
960 S=(DT*HI)/(ROW*CV*DX)
970 CO=(HO*DT)/(ROW*CV*DX)
980 CA=(2*DT*HFG)/(ROW*CV*DX)
990 REM
1000 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
1010 REM
1020 N=2
1030 FOR K=N TO 500
1040 GOSUB 2350
1050 TM=T(3,K-1)+273
1060 HCP(K-1)=(HD*MW*(PA(K-1)-PC))/(RG*TM)
1070 F(1)=T(1,K-1)+(2*S*TI(K-1))
1080 F(2)=T(2,K-1)
1090 F(3)=T(3,K-1)+CA*MCP(K-1)+2*CG*TOUT(K-1)
1100 B(1)=1+(2*FO)+(2*S)
1110 C(1)=-2*FO
1120 A(2)=-FO
1130 B(2)=1+(2*FO)
1140 C(2)=-FO
1150 A(3)=-2*FO
1160 B(3)=1+(2*FO)+(2*CO)
1170 GOSUB 1570
1180 NEXT K

```

```

1240 GOSUB 1680
1250 GOSUB 2470
1260 REM *****printing out the needed information*****
1270 REM
1280 INPUT "insert the time in sec. at which the first temperature is needed";
T1
1290 INPUT "insert the time in sec. at which the last temperature is needed";
T2
1300 U=(T1\DT)+1
1310 V=(T2\DT)+1
1320 PRINT "
1330 PRINT "***** the nodal temperatures at the needed time*****"

1340 REM
1350 PRINT " note: t1=indoor facing temperature and t3=outdoor facing temperature
"
1360 PRINT "time in SEC.", "t1 cel.", "t2 cel.", "t3 cel.", "mcr kg/s"
1370 PRINT "-----", "-----", "-----", "-----", "-----"
1380 FOR A=U TO V
1390 PRINT (A-1)*DT, T(1,A), T(2,A), T(3,A), MCR(A)
1400 NEXT A
1410 PRINT "
1420 PRINT "the heat gain near the steady state stages=", QINMAX "w/m^2"
1430 PRINT "the mass condensation rate near the steady state stages=",
MSS "kg/s per m^2"
1440 PRINT "the indoor facing surface temperature near steady states=", TFCO "
"
1450 IF TSS=0 THEN GOTO 1470
1460 PRINT "the time needed to reach steady state stages=", TSS "seconds"
1470 PRINT "-----"
1480 PRINT "the maximum interval heat gain rates=", QINMAX "w/m^2"
1490 PRINT "the minimum interval heat gain rates=", QINMIN "w/m^2"
1500 PRINT "the average heat gain rate =", QAVG "w/m^2"
1510 PRINT "the average mass condensation rate during condensations=",
MCAVG "kg/sec. per m^2"

1520 REM
1530 INPUT "do you want to know the temperatures at another time interval Y or N"
;A3
1540 IF A3="Y" THEN GOTO 1260 ELSE 1545
1545 PRINT "
1550 INPUT "do you want to try at other conditions y or n ";E3
1560 IF E3="Y" THEN 180 ELSE RUN "cond"
1570 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND-
ENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM
1580 REM
1590 FOR I=2 TO 3
1600 D=A(I)/B(I-1)
1610 B(I)=B(I)-(C(I-1)*D)
1620 F(I)=F(I)-(F(I-1)*D)
1630 NEXT I
1640 T(3,K)=F(3)/B(3)
1650 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1660 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1670 RETURN
1680 REM THIS SUBROUTINE CALCULATES THE AVERAGE CONVECTIVE HEAT GAIN RATE AND
FINDS THE MAXIMUM AND MINIMUM CONVECTIVE INTERVAL HEAT GAIN
1690 REM
1700 H=450
1710 QINMIN=100
1720 FOR H1=1 TO 450
1730 Q1=(T(1,H1)+T(1,H1+1))/2
1740 QINAV(H1)=H1*(Q1-T(1,1))
1750 QINTOT=QINTOT+QINAV(H1)
1760 IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
1770 IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1),

```

```

1780 IF TESS<T(1,H1) THEN TESS=T(1,H1)
1790 IF T(3,H1+1)=T(3,H1) THEN 1800 ELSE 1820
1800 IF T(2,H1+1)=T(2,H1) THEN 1810 ELSE 1820
1810 IF T(1,H1+1)=T(1,H1) THEN GOTO 1840 ELSE 1850
1820 NEXT H1
1830 GOTO 1850
1840 TSS=DT*H1
1850 QAVG=QINTOT/H1
1860 RETURN
1870 REM this subroutine calculates the saturation temperature at the
      beginning of condensation using keenan, keyes, hill, and moore formula
1880 REM
1890 F=0
1900 FOR XO=1 TO 8
1910 F1=F2(XO)*((.65-.01*TOUTO)^(XO-1))
1920 F=F+F1
1930 NEXT XO
1940 TKEL=TOUTO+273.15
1950 L1=LOG(217.99)+((374.136-TOUTO)*F*(.01/TKEL))
1960 PWS=EXP(L1)
1970 PWATH=PHO*PWS
1980 PW=29.921*PWATH
1990 FA=LOG(PW)
2000 TSATF=79.047+(30.579*FA)+(1.8893*FA^2)
2010 TSAT=(5/9)*(TSATF-32)
2020 IF TSAT>TOUTO THEN TSAT=TOUTO
2030 RETURN
2040 REM
2050 REM this subroutine calculates the outdoor, indoor temperatures as a
      function of time, also it calculates the partial vapour pressure
      at different outdoor conditions
2060 REM
2070 FOR X=1 TO 500
2080 REM ***** assumed function of outdoor temperature variation *****
2090 REM
2100 TOUT(X)=TOUTO-(DT*(X-1)/7200)
2110 IF TOUT(X)<TOUTO-10 THEN TOUT(X)=TOUTO-10
2120 REM
2130 REM ***** assumed function of indoor variation *****
2140 REM
2150 TI(X)=TIAVG-(2*SIN(3.1416*DT*(X-1)/3600*4))
2160 REM
2170 REM ***** assumed function for humidity variation *****
2180 REM
2190 RH=PHO+DT*(X-1)/(144000!)
2200 IF RH>1 THEN RH=1
2210 REM
2220 F=0
2230 FOR XO=1 TO 8
2240 F1=F2(XO)*((.65-.01*TOUT(X))^(XO-1))
2250 F=F+F1
2260 NEXT XO
2270 TKEL=TOUT(X)+273.15
2280 L1=LOG(217.99)+((374.136-TOUT(X))*F*(.01/TKEL))
2290 PWS=EXP(L1)
2300 PWATH=RH*PWS
2310 PA(X)=101325!*PWATH
2320 NEXT X
2330 RETURN
2340 REM
2350 REM this subroutine calculates the saturation vapour pressure at the
      window temperature by using keenan, keyes, hill, and moore formula
2360 REM
2370 F=0
2380 FOR XO=1 TO 8
2390 F1=F2(XO)*((.65-(.01*T(3,X-1)))^(XO-1))

```

```
2400 F=F+F1
2410 NEXT X0
2420 TKEL=T(3,K-1)+273.15
2430 L1=LOG(217.99)+((374.136-T(3,K-1))*F*(.01/TKEL))
2440 PWS=EXP(L1)
2450 PC=101325!*PWS
2460 RETURN
2470 REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2480 MSS=10
2490 FOR Z=1 TO 450
2500 IF MCP(Z)<MSS THEN MSS=MCP(Z)
2510 IF MCP(Z+1)=0 THEN GOTO 2560
2520 MINAV=(MCP(Z)+MCP(Z+1))/2
2530 MTOT=MTOT+MINAV
2540 NUMINT=NUMINT+1
2550 MCR AVG=MTOT/NUMINT
2560 NEXT Z
2570 RETURN
```

SUB-PROGRAM 2

```

10 DIM T(3,1010),A(3),B(3),C(3),F(3),CINAV(500),F2(5),C(505)
20 REM:
30 INPUT"this subprogram calculates the mass condensation rate and the resulting
..... heat gain when outdoor condensation occurs on single glazed windows at
..... constant conditions if you want to use it print (y) else print (n)";BS
40 REM
50 IF BS="y" THEN GOTO 80 ELSE RUN"cond"
60 REM
70 REM *****keenan, keyes, hill, and moore formula constants*****
80 F2(1)=-741.9242
90 F2(2)=-29.721
100 F2(3)=-11.55286
110 F2(4)=-.8685635
120 F2(5)=.1094098
130 F2(6)=.439993
140 F2(7)=.2526658
150 F2(8)=.0521866
160 REM *****input of atmospheric, indoor conditions and window related data****
170 REM
180 PRINT"
190 INPUT"the indoor temperature in c =";TI
200 PRINT"
210 INPUT"ambient temperature in c =";TOUT
220 PRINT"
230 INPUT"outdoor relative humidity =";RH
240 PRINT"
250 INPUT"window (OR WALL) height in meters=";L
260 PRINT"
270 INPUT"glass thickness in meters=";M
280 PRINT"
290 INPUT"time interval in seconds=";DT
300 PRINT"
310 GOSUB 1880
320 REM the average outdoor(toa), indoor(tia) and wall temperatures(twia,twoa)
will be assumed as following
330 REM
340 TCA=TOUT
350 TIA=TI
360 TWCA=TCA-5
370 TWIA=TIA+5
380 TCAK=TOA+273
390 TIAK=TIA+273
400 TWOAK=TWOA+273
410 TWIAK=TWIA+273
420 REM the air properties are evaluated at the assumed average film
temperatures which is equal to 28 deg. c for outdoor conditions
and 21 deg. c for indoor conditions
430 REM
440 MEUI=1.4937E-05
450 ALFAI=.0000211
460 KAI=.025764
470 MEUO=1.568E-05
480 ALFAO=2.216E-05
490 KAO=.02624
500 TFLMI=(18+24)/2+273
510 TFLMO=(25+30)/2+273
520 BETAI=1/TFLMI
530 BETAO=1/TFLMO
540 SIGMA=5.673E-08
550 FSR=1
560 PAI=9.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
570 PAO=9.8*BETAO*L^3*(TOA-TWOA)/(MEUO*ALFAO)
580 IF PAI<0 THEN HCI=.59*KAI*(PAI^-.333)/L
590 IF PAI>0 THEN HCI=.1*KAI*(PAI^-.333)/L
600 IF PAO<0 THEN HCO=.59*KAO*(PAO^-.333)/L
610 IF PAO>0 THEN HCO=.1*KAO*(PAO^-.333)/L

```

```

620 HPO=SEGMA*FSP*(TOAK^2+TWOAK^2)*(TOAK+TWOAK)
630 HPI=SEGMA*FSP*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
640 HI=HPI+HPI
650 HO=HCO+HRO
660 KGL=.78
670 DX=M/2
680 R1=1/HC
690 R2=(1/KGL*DX)+R1
700 R3=(1/KGL*M)+R1
710 R=R3+1/HI
720 T(3,1)=TOUT-(R1/R)*(TOUT-TI)
730 IF T(3,1)>=TSAT-1 THEN GOTO 740 ELSE 790
740 PRINT"condensation will not occur at these conditions,start again if you
      wish to try at other conditions else press ctrl-break"
750 GOTO 190
760 REM water and vapour properties are evaluated at assumed outdoor average
770 REM temperatures
780 REM
790 ROWL=998
800 ROWV=.7
810 HFG=25000001
820 KL=.02
830 MEUL=.00087
840 CV=840
850 ROW=2700
860 T(2,1)=TCUT-(R2/R)*(TOUT-TI)
870 T(1,1)=TOUT-(R3/R)*(TOUT-TI)
880 D0=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^-.25
890 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^(1/3)
900 D3=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^-.25
910 D4=(MEUL*(TSAT-T(3,1))*L)^.25
920 D5=(L*(TSAT-T(3,1)))/(MEUL*HFG)^.75
930 MW=18
940 PG=83101
950 DC=.000025
960 PC/A=1.15
970 CVA=1005
980 HD=((DC*HCO)*((KAG/(PC/A*CVA*DC))^-.25))/KAG
990 FO=(KGL*DT)/(ROW*CV*DX^2)
1000 S=(DT*HI)/(ROW*CV*DX)
1010 CO=(HO*DT)/(ROW*CV*DX)
1020 CA=(2*DT*HFG)/(ROW*CV*DX)
1030 REM
1040 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
1050 J=1
1060 N=J+1
1070 FOR K=N TO 502
1080 GOSUB 2060
1090 TM=T(3,K-1)+273.15
1100 MCR(K-1)=(HD*MW*(PA-PC))/(PG*TM)
1110 F(1)=T(1,K-1)+(2*S*TI)
1120 F(2)=T(2,K-1)
1130 F(3)=T(3,K-1)+CA*MCR(K-1)+2*CO*TOUT
1140 B(1)=1+(2*FO)+(2*S)
1150 C(1)=-2*FO
1160 A(2)=-FO
1170 B(2)=1+(2*FO)
1180 C(2)=-FO
1190 A(3)=-2*FO
1200 B(3)=1+(2*FO)+(2*CO)
1210 GOSUB 1590
1220 NEXT K
1230 GOSUB 1700
1240 GOSUB 2180
1250 REM *****printing out the needed information*****

```

```

1260 REM
1270 INPUT"insert the time in sec. at which the first temperature is needed
";T1
1280 INPUT"insert the time in sec. at which the last temperature is needed"
";T2
1290 IF T2>500*DT THEN T2=500*DT
1300 U=(T1\DT)+1
1310 V=(T2\DT)+1
1320 PRINT "
1330 PRINT"***** the nodal temperatures at the needed time*****"

1340 REM
1350 PRINT" note: t1=indoor facing temperature and t3=outdoor facing temperature
-----"
1360 PRINT "time in SEC.,""t1 cel.,""t2 cel.,""t3 cel.,""mcr kg/s"
1370 PRINT "-----","-----","-----","-----","-----"
1380 FOR A=U TO V
1390 PRINT (A-1)*DT, T(1,A),T(2,A),T(3,A),MCP(A)
1400 NEXT A
1410 PRINT"
1420 PRINT"the heat gain near the steady state stage="QINMAX"w/m^2"
1430 PRINT"the mass condensation rate near the steady state stage="MSS"kg/s
per m^2"
1440 PRINT"the indoor facing surface temperature near steady state="TESS"
1450 PRINT"temp. rise="TESS-T(1,1)"r,"
1460 IF TSS=0 THEN GOTO 1480
1470 PRINT"the time needed to reach steady state stage="TSS"seconds"
1480 PRINT"
1490 PRINT"the maximum interval heat gain rate="QINMAX"w/m^2"
1500 PRINT"the minimum interval heat gain rate="QINMIN"w/m^2"
1510 PRINT "the average heat gain rate ="QAVG"w/m^2"
1520 PRINT "the average mass condensation rate during condensations="
4CRAVG"kg/sec.per m^2"

1530 REM
1540 INPUT"do you want to know the temperatures at another time interval Y or N"
;A$
1550 IF A$="y" THEN GOTO 1270 ELSE 1560
1560 PRINT"
1570 INPUT"do you want to try at other conditions y or n";E$
1580 IF E$="y" THEN 180 ELSE RUN"cond"
1590 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND
ENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM

1600 REM
1610 FOR I=2 TO 3
1620 D=A(I)/B(I-1)
1630 B(I)=B(I)-(C(I-1)*D)
1640 F(I)=F(I)-(F(I-1)*D)
1650 NEXT I
1660 T(3,K)=F(3)/B(3)
1670 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1680 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1690 RETURN
1700 REM THIS SUBROUTINE CALCULATES THE AVERAGE CONVECTIVE HEAT GAIN RATE AND
FINDS THE MAXIMUM AND MINIMUM CONVECTIVE INTERVAL HEAT GAIN

1710 REM
1720 QINMIN=100
1730 FOR H1=1 TO 500
1740 Q1=(T(1,H1)+T(1,H1+1))/2
1750 QINAV(H1)=HI*(Q1-T(1,1))
1760 QINTOT=QINTOT+QINAV(H1)
1770 IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
1780 IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
1790 IF TESS<T(1,H1) THEN TESS=T(1,H1)
1800 IF T(3,H1+1)=T(3,H1) THEN 1810 ELSE 1830

```

```

1810     IF T(2,H1+1)=T(2,H1) THEN 1820 ELSE 1830
1820     IF T(1,H1+1)=T(1,H1) THEN GOTO 1850 ELSE 1860
1830     NEXT H1
1840     GOTO 1860
1850     TSS=DT*H1
1860     QAVG=QINTOT/H1
1870     RETURN
1880 REM this subroutine calculates the saturation temperature at the
      beginning of condensation using keenan,keyes,hill,and moore formula
1890 REM
1900     F=0
1910     FOR XO=1 TO 8
1920     F1=F2(XO)*((.65-.01*TOUT)^(XO-1))
1930     F=F+F1
1940     NEXT XO
1950     TKEL=TOUT+273.15
1960     L1=LOG(217.99)+((374.136-TOUT)*F*(.01/TKEL))
1970     PWS=EXP(L1)
1980     PWATM=RH*PWS
1990     PA=101325!*PWATM
2000     PW=29.921*PWATM
2010     FA=LOG(PW)
2020     TSATF=79.047+(30.579*FA)+(1.0093*FA^2)
2030     TSAT=(5/9)*(TSATF-32)
2040     IF TSAT>TOUT THEN TSAT=TOUT
2050     RETURN
2060 REM this subroutine calculates the saturation vapour pressure at the
      window temperature by using keenan,keyes,hill and moore formula
2070 REM
2080     F=0
2090     FOR XO=1 TO 8
2100     F1=F2(XO)*((.65-(.01*T(3,K-1)))^(XO-1))
2110     F=F+F1
2120     NEXT XO
2130     TKEL=T(3,K-1)+273.15
2140     L1=LOG(217.99)+((374.136-T(3,K-1))*F*(.01/TKEL))
2150     PWS=EXP(L1)
2160     PC=101325!*PWS
2170     RETURN
2180 REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2190     MSS=10
2200     FOR Z=1 TO 500
2210     IF MCR(Z)<MSS THEN MSS=MCR(Z)
2220     IF MCR(Z+1)=0 THEN GOTO 2270
2230     MINAV=(MCR(Z)+MCR(Z+1))/2
2240     MTOT=MTOT+MINAV
2250     NUMINT=NUMINT+1
2260     MCR AVG=MTOT/NUMINT
2270     NEXT Z
2280     RETURN

```


SUB-PROGRAM 3

```

10 DIM T(3,1010),A(3),B(3),C(3),F(3),QINAV(500),F2(8),MCF(505)
20 REM
30 INPUT "this subprogram calculates the mass condensation rate and the resulting
    heat loss when indoor condensation occurs on single glazed windows at
    constant conditions if you want to use it print (y) else print (n)";B$
40 REM
50 IF B$="y" THEN GOTO 80 ELSE RUN"cond"
60 REM
70 REM *****keenan,keyes,hill,and moore formula constants*****
80 F2(1)=-741.9242
90 F2(2)=-29.721
100 F2(3)=-11.55286
110 F2(4)=-.8685635
120 F2(5)=.1094098
130 F2(6)=.439993
140 F2(7)=.2520658
150 F2(8)=.0521666
160 REM *****input of atmospheric,indoor conditions and window related data****
170 REM
180 PRINT
190 INPUT "the indoor temperature in c =";TI
200 PRINT
210 INPUT "ambient temperature in c =";TOUT
220 PRINT
230 INPUT "indoor relative humidity =";RH
240 PRINT
250 INPUT "the wind speed in meters per second=";WV
260 PRINT
270 INPUT "the wind angle of attack from normal in degrees=";ANG
280 PRINT
290 INPUT "window (OR WALL) height in meters=";L
300 PRINT
310 INPUT "glass thickness in meters=";M
320 PRINT
330 INPUT "time interval in seconds=";DT
340 GOSUB 1820
350 REM the average outdoor(tou),indoor(tia) and wall temperatures(twia,twoa)
    will be assumed as following
360 REM
370 TIA=TI
380 TWIA=TIA+6
390 TIAK=TIA+273
400 TWIAK=TWIA+273
410 REM the air properties are evaluated at the assumed average film
    temperatures which is equal to 28 deg. c for outdoor conditions
    and 21 deg. c for indoor conditions
420 REM
430 MEUI=1.4937E-05
440 ALFAI=.0060211
450 KAI=.025764
460 TFIIMI=(18+24)/2+273
470 BFTAI=1/TFIIMI
480 SEGMA=5.673E-08
490 FSR=1
500 RAI=9.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
510 IF RAI<1E+09 THEN HCI=.59*KAI*(RAI^.25)/L
520 IF RAI>1E+09 THEN HCI=.1*KAI*(RAI^.333)/L
530 HPI=SEGMA*FSR*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
540 HI=HCI+HPI
550 IF ANG=>0 AND ANG<30 THEN HO=7.77+2.9*WV
560 IF ANG=>30 AND ANG<60 THEN HO=7.5+3.26*WV
570 IF ANG=>60 AND ANG<100 THEN HO=7.52+4.1*WV
580 IF ANG=>100 AND ANG<140 THEN HO=6.83+2.63*WV
590 IF ANG=>140 AND ANG<180 THEN HO=6.27+2.65*WV
600 KCI=.78
610 DX=M/2

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```

620 R1=1/HI
630 R2=(1/KGL*DX)+R1
640 R3=(1/KGL*M)+R1
650 R=R3+1/HO
660 T(3,1)=TI-(R1/R)*(TI-TOUT)
670 IF T(3,1)>=TSAT-1 THEN GOTO 680 ELSE 730
680 PRINT"condensation will not occur at these conditions,start again if you wis
h to try at other conditions else press ctrl-break"
690 GOTO 190
700 REM water and vapour properties are evaluated at assumed indoor average
710 REM temperatures
720 REM
730 ROWL=1000
740 ROWV=.016
750 HFG=2480000!
760 KL=.5
770 MEUL=.001
780 CV=840
790 ROW=2700
800 CVA=1005
810 PCWA=1.2
820 MW=18
830 RG=8310
840 DC=.000026
850 HD=((DC*HCI)*((KAI/(PCWA*CVA*DC))^.25))/KAI
860 T(2,1)=TI-(R2/R)*(TI-TOUT)
870 T(1,1)=TI-(R3/R)*(TI-TOUT)
880 D0=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^.25
890 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^(1/3)
900 D3=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^.25
910 D4=(MEUL*(TSAT-T(3,1))*L)^.25
920 D5=(L*(TSAT-T(3,1))/(MEUL*HFG))^.75
930 FO=(KGL*DT)/(ROW*CV*DX^2)
940 S=(DT*HO)/(ROW*CV*DX)
950 CG=(DT*HI)/(ROW*CV*DX)
960 CA=(2*DT*HFG)/(ROW*CV*DX)
970 REM
980 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
990 J=1
1000 N=J+1
1010 FOR K=N TO 502
1020 GOSUB 2000
1030 TM=T(3,K)+273
1040 MCP(K-1)=(HD*MW*(PA-PC))/(RG*TM)
1050 F(1)=T(1,K-1)+(2*S*TOUT)
1060 F(2)=T(2,K-1)
1070 F(3)=T(3,K-1)+(CA*MCP(K-1))+(2*CG*TI)
1080 B(1)=1+(2*FO)+(2*S)
1090 C(1)=-2*FO
1100 A(2)=-FO
1110 B(2)=1+(2*FO)
1120 C(2)=-FO
1130 A(3)=-2*FO
1140 B(3)=1+(2*FO)+(2*CG)
1150 GOSUB 1500
1160 NEXT K
1170 GOSUB 1610
1180 GOSUB 2120
1190 REM *****printing out the needed information*****
1200 REM
1210 INPUT"insert the time in sec. at which the first temperature is needed:
";T1
1220 INPUT"insert the time in sec. at which the last temperature is needed:"
";T2
1230 IF T2>DT*500 THEN T2=DT*500

```

```

1240      U=(T1\DT)+1
1250      V=(T2\DT)+1
1260 PRINT "
1270 PRINT"***** the nodal temperatures at the needed time*****"

1280 REM
1290 PRINT" note: t3=indoor facing temperature and t1=outdoor facing temperature
      -----
1300 PRINT "time in sec.", "t1 cel.", "t2 cel.", "t3 cel.", "mcr kg/s"
1310 PRINT "-----", "-----", "-----", "-----", "-----"
1320 FOR A=U TO V
1330 PRINT (A-1)*DT, T(1,A), T(2,A), T(3,A), MCR(A)
1340 NEXT A
1350 PRINT"
1360 PRINT"the total heat loss near the steady state stage="MSS*HFG"w/m^2"
1370 PRINT"the convective heat loss rate near the steady state stage="
      QINMAX"w/m^2"
1380 PRINT"the maximum interval convective heat loss rate="QINMAX"w/m^2"
1390 PRINT"the minimum interval convective heat loss rate="QINMIN"w/m^2"
1400 PRINT "the average convective heat loss rate="QAVG"w/m^2"
1410 PRINT "the mass condensation rate near the the steady state states="
      MSS"kg/sec per m^2"
1420 PRINT"the average mass condensation rate during condensation="
      MRAVG"kg/sec. per m^2"
1430 IF TSS=0 THEN GOTO 1450
1440 PRINT"the time needed to reach steady state stage="TSS"seconds"
1450 REM
1460 INPUT"do you want to know the temperatures at another time interval Y or N"
1470 IF A$="Y" THEN GOTO 1210 ELSE 1475
1475 PRINT"
1480 INPUT"do you want to try at other conditions y or n";E$
1490 IF E$="Y" THEN 180 ELSE RUN"cond"
1500 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND
      ENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM
1510 REM
1520 FOR I=2 TO 3
1530 D=A(I)/B(I-1)
1540 B(I)=B(I)-(C(I-1)*D)
1550 F(I)=F(I)-(F(I-1)*D)
1560 NEXT I
1570 T(3,K)=F(3)/B(3)
1580 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1590 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1600 RETURN
1610 REM THIS SUBROUTINE CALCULATES THE AVERAGE CONVECTIVE HEAT LOSS RATE AND
      FINDS THE MAXIMUM AND MINIMUM INTERVAL CONVECTIVE HEAT LOSS
1620 REM
1630 TTOTAVG=0
1640 QINMAX=0
1650 QINMIN=100
1660 H=500
1670 FOR H1=1 TO H
1680 Q1=(T(1,H1)+T(1,H1+1))/2
1690 QINAV(H1)=HJ*(Q1-T(1,1))
1700 IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
1710 IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
1720 QINTOT=QINTOT+QINAV(H1)
1730 IF TESS<T(3,H1) THEN TESS=T(3,H1)
1740 IF T(3,H1+1)=T(3,H1) THEN 1750 ELSE 1770
1750 IF T(2,H1+1)=T(2,H1) THEN 1760 ELSE 1770
1760 IF T(1,H1+1)=T(1,H1) THEN GOTO 1790 ELSE 1800
1770 NEXT H1
1780 GOTO 1800
1790 TSS=DT*H1

```

```

1800     QAVG=QINTOT/H1
1810     RETURN
1820 REM this subroutine calculates the saturation temperature at the
      beginning of condensation using keenan,keyes,hill,and moore formula
1830 REM
1840     F=0
1850     FOR XO=1 TO 8
1860     F1=F2(XO)*((.65-.01*TI)^(XO-1))
1870     F=F+F1
1880     NEXT XO
1890     TKEL=TI+273.15
1900     L1=LOG(217.99)+((374.136-TI)*F*(.01/TKEL))
1910     PWS=EXP(L1)
1920     PWATM=PH*PWS
1930     PA=101325!*PWATM
1940     PW=29.921*PWATM
1950     FA=LOG(PW)
1960     TSATF=79.047+(30.579*FA)+(1.6893*FA^2)
1970     TSAT=(5/9)*(TSATF-32)
1980     IF TSAT>TI THEN TSAT=TI
1990     RETURN
2000 REM this subroutine calculates the saturation pressure at the window
      temperature using keenan,keyes,hill,and moore formula
2010 REM
2020     F=0
2030     FOR XO=1 TO 8
2040     F1=F2(XO)*((.65-.01*T(3,K-1))^(XO-1))
2050     F=F+F1
2060     NEXT XO
2070     TKEL=T(3,K-1)+273.15
2080     L1=LOG(217.99)+((374.136-T(3,K-1))*F*(.01/TKEL))
2090     PWS=EXP(L1)
2100     PC=101325!*PWS
2110     RETURN
2120 REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2130     MSS=10
2140     FOR Z=1 TO 495
2150     IF MCR(Z)<MCR THEN MSS=MCR(Z)
2160     IF MCR(Z+1) THEN GOTO 2210
2170     MINAV=(MCR(Z)+MCR(Z+1))/2
2180     MTOT=MTOT+M*MINAV
2190     NUMINT=NUMINT+1
2200     MCRAVG=MTOT/NUMINT
2210     NEXT Z
2220     RETURN

```

SUB-PROGRAM 4

```

10 DIM T(3,505),A(3),B(3),C(3),F(3),QINAV(505),F2(8),MCR(505)
20 REM
30 INPUT"this subprogram calculates the mass condensation rate and the resulting
..... heat gain when outdoor condensation occurs on single glazed windows at
..... constant conditions if you want to use it print (y) else print (n)";B1
40 REM
50 IF B1="y" THEN GOTO 80 ELSE RUN"end"
60 REM
70 REM *****keenan, keyes, hill, and moore formula constants*****
80 F2(1)=-741.9242
90 F2(2)=-29.721
100 F2(3)=-11.55286
110 F2(4)=-.8655635
120 F2(5)=.1094098
130 F2(6)=.439993
140 F2(7)=.2520658
150 F2(8)=.0521600
160 REM *****input of atmospheric, indoor conditions and window related data****
170 REM
180 PRINT
190 INPUT"the indoor temperature in c=";TI
200 PRINT
210 INPUT"ambient temperature in c=";TOUT
220 PRINT
230 INPUT"outdoor relative humidity=";RH
240 PRINT
250 INPUT"window (or wall) height in meters=";L
260 PRINT
270 INPUT"glass thickness in meters=";t
280 PRINT
290 INPUT"time interval in seconds=";DT
300 PRINT
310 GOSUB 2040
320 REM the average outdoor(toa), indoor(tia) and wall temperatures(twia,twoa)
will be assumed as following
330 REM
340 TOA=TOUT
350 TIA=TI
360 TAOA=TOA-5
370 TWIA=TIA+5
380 TOAK=TOA+273
390 TIAK=TIA+273
400 TWOAK=TWOA+273
410 TWIAK=TWIA+273
420 REM the air properties are evaluated at the assumed average film
temperatures which is equal to 28 deg. c for outdoor conditions
and 21 deg. c for indoor conditions
430 REM
440 MEUI=1.4937E-05
450 ALFAI=.0000211
460 KAI=.025764
470 MEUD=1.568E-05
480 ALFAD=.216E-05
490 KAD=.02624
500 TFIEMI=(18+24)/2+273
510 TFIEMD=(25+30)/2+273
520 BETA1=1/TFIEMI
530 BETA0=1/TFIEMD
540 SIGMA=5.673E-08
550 FSR=1
560 RA1=9.8*BETA1*L^3*(TWIA-TIA)/(MEUI*ALFAI)
570 RA0=9.8*BETA0*L^3*(TOA-TWOA)/(MEUD*ALFAD)
580 IF RA1<1E+09 THEN HCFI=.59*KAI*(RA1^-.25)/L
590 IF RA1>1E+09 THEN HCFI=.1*KAI*(RA1^-.333)/L
600 IF RA0<1E+09 THEN HCD=.59*KAD*(RA0^-.25)/L
610 IF RA0>1E+09 THEN HCD=.1*KAD*(RA0^-.333)/L

```

```

620 HPO=SEGMA*FSP*(TOAK^2+TWOAK^2)*(TOAK+TWOAK)
630 HPI=SEGMA*FSP*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
640 HI=HCI+HPI
650 HO=HCO+HPO
660 KGL=.78
670 DX=M/2
680 R1=1/HO
690 R2=(1/KGL*DX)+R1
700 R3=(1/KGL*M)+R1
710 R=R3+1/HI
720 T(3,1)=TOUT-(R1/R)*(TOUT-TI)
730 IF T(3,1)>=TSAT-1 THEN GOTO 740 ELSE 790
740 PRINT"condensation will not occur at these conditions,start again if you
wish to try at other conditions else press ctrl-break"
750 GOTO 190
760 REM water and vapour properties are evaluated at assumed outdoor average
770 REM temperatures
780 REM
790 POWL=998
800 POWV=.7
810 HFG=2500000
820 KL=.62
830 MEUL=.00087
840 CV=840
850 ROW=2700
860 T(2,1)=TOUT-(R2/R)*(TOUT-TI)
870 T(1,1)=TOUT-(R3/R)*(TOUT-TI)
880 D0=((9.8*POWL*(POWL-POWV)*KL^3)/(MEUL^2))^-.25
890 D2=((9.8*POWL*(POWL-POWV)*KL^3)/(MEUL^2))^(1/3)
900 D3=((9.8*POWL*(POWL-POWV)*HFG*KL^3)^-.25
910 D4=(MEUL*(TSAT-T(3,1))*L)^-.25
920 D5=(L*(TSAT-T(3,1))/(MEUL*HFG))^-.75
930 MW=18
940 RG=8310
950 DC=.000026
960 PCWA=1.17
970 CVA=1005
980 MEUA=1.983E-05
990 MU=1.664E-05
1000 MUL=8.717E-07
1010 SC=MU/DC
1020 MV=MW
1030 MG=28.97
1040 WGI=1
1050 SD=.00001
1060 LIM=5
1070 DO=MEUL*POWL/(MEUA*PCWA)
1080 CO=1/DO
1090 CPL=4180
1100 PRL=CPL*MEUL/KL
1110 X=(MG-MV)/(MG-WB*(MG-MV))
1120 FO=(KGL*DT)/(PCW*CV*DX^2)
1130 S=(DT*HI)/(PCW*CV*DX)
1140 CO1=(HO*DT)/(PCW*CV*DX)
1150 CA=(2*DT*HFG)/(ROW*CV*DX)
1160 C1=((9.8*(MG-MV)/(4*MU^2))/(MG-(MG-MV)*WB))^-.25
1170 REM
1180 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
1190 J=1
1200 N=502
1210 FOR K=2 TO N
1220 GOSUB 2250
1230 IF PWGI-PVI<0 THEN GOTO 1260
1240 FO1=((9.8*L^3)/(4*MU^2))^-.25
1250 FO2=(CPL*(TGI-T(3,K-1)))/(PRL*HFG))^-.75

```

```

1260 F03=4*F01*F02*HFG*HEUL/3
1270 MCP(K-1)=F03/(HFG*L)
1280 F(1)=T(1,K-1)+(2*S*TI)
1290 F(2)=T(2,K-1)
1300 F(3)=T(3,K-1)+CA*MCP(K-1)+2*CO1*TCUT
1310 B(1)=1+(2*F0)+(2*S)
1320 C(1)=-2*F0
1330 A(2)=-F0
1340 B(2)=1+(2*F0)
1350 C(2)=-F0
1360 A(3)=-2*F0
1370 B(3)=1+(2*F0)+(2*CO1)
1380 GOSUB 1740
1390 NEXT K
1400 GOSUB 1850
1410 GOSUB 2660
1420 REM *****printing out the needed information*****
1430 REM
1440 INPUT"insert the time in sec. at which the first temperature is needed
";T1
1450 INPUT"insert the time in sec. at which the last temperature is needed"
";T2
1460 IF T2>DT*500 THEN T2=500*DT
1470 U=(T1\DT)+1
1480 V=(T2\DT)+1
1490 PRINT "
1500 PRINT"***** the nodal temperatures at the needed time*****"
1510 REM
1520 PRINT" note: t1=indoor facing temperature and t3=outdoor facing temperature
"
1530 PRINT "time in SEC.,"t1 cel.,"t2 cel.,"t3 cel.,"mcr kg/s"
1540 PRINT "-----","-----","-----","-----","-----"
1550 FOR A=J TO V
1560 PRINT (A-1)*DT, T(1,A),T(2,A),T(3,A),MCP(A)
1570 NEXT A
1580 PRINT"
1590 PRINT"the heat gain near the steady state stage="QINMAX"w/m^2"
1600 PRINT"the mass condensation rate near the steady state stage="MSS"kg/s
per m^2"
1610 PRINT"the indoor facing surface temperature near steady state="TFSS"c.
"
1620 IF TSS=0 THEN GOTO 1640
1630 PRINT"the time needed to reach steady state stage="TSS"seconds"
1640 PRINT"-----"
1650 PRINT"the maximum interval heat gain rate="QINMAX"w/m^2"
1660 PRINT"the minimum interval heat gain rate="QINMIN"w/m^2"
1670 PRINT "the average heat gain rate ="QAVG"w/m^2"
1680 PRINT "the average mass condensation rate during condensation="MCRAVG
"kg/sec.per m^2"
1690 REM
1700 INPUT"do you want to know the temperatures at another time interval Y or N"
;A$
1710 IF A$="Y" THEN GOTO 1440 ELSE 1715
1715 PRINT"
1720 INPUT"do you want to try at other conditions y or n ",FS
1730 IF FS="Y" THEN 180 ELSE RUN"cond"
1740 REM THIS subroutine SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS
CONDENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM
1750 REM
1760 FOR I=2 TO 3
1770 D=A(I)/B(I-1)
1780 B(I)=B(I)-(C(I-1)*D)
1790 F(I)=F(I)-(F(I-1)*D)
1800 NEXT I

```

```

1810 T(3,K)=F(3)/B(3)
1820 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1830 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1840 RETURN
1850 REM THIS SUBROUTINE CALCULATES THE AVERAGE CONVECTIVE HEAT GAIN RATE AND
      FINDS THE MAXIMUM AND MINIMUM CONVECTIVE INTERVAL HEAT GAIN
1860 REM
1870 H=500
1880 QINMIN=100
1890 FOR H1=1 TO H-2
1900 Q1=(T(1,H1)+T(1,H1+1))/2
1910 QINAV(H1)=HI*(Q1-T(1,1))
1920 QINTOT=QINTOT+QINAV(H1)
1930 IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
1940 IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
1950 IF TESS<T(1,H1) THEN TESS=T(1,H1)
1960 IF T(3,H1+1)=T(3,H1) THEN 1970 ELSE 1990
1970 IF T(2,H1+1)=T(2,H1) THEN 1980 ELSE 1990
1980 IF T(1,H1+1)=T(1,H1) THEN GOTO 2010 ELSE 2020
1990 NEXT H1
2000 GOTO 2020
2010 TSS=DT*H1
2020 QAVG=QINTOT/H1
2030 RETURN
2040 REM this subroutine calculates the saturation temperature at the
      begining of condensation using keenan,keyes,hill,and moore formula
2050 REM
2060 F=0
2070 FOR XO=1 TO 8
2080 F1=F2(XO)*((.65-.01*TOUT)^(XO-1))
2090 F=F+F1
2100 NEXT XO
2110 TKEL=TOUT+273.15
2120 L1=LOG(217.99)+((374.136-TOUT)*F*(.01/TKEL))
2130 PWS=EXP(L1)
2140 PWATM=PH*PWS
2150 HUR=.62198*PWATM
2160 WB=1/(HUR+1)
2170 PA=101325! *PWATM
2180 PTOT=PA+101325!
2190 PW=29.921*PWATM
2200 FA=LOG(PW)
2210 TSATF=79.047+(30.579*FA)+(1.8893*FA^2)
2220 TSAT=(5/9)*(TSATF-32)
2230 IF TSAT>TOUT THEN TSAT=TOUT
2240 RETURN
2250 REM this subroutine calculates the interfacial film temperature
      at a given window temperature and outdoor conditions
2260 REM
2270 REM WGI=WB
2280 IF PWSI-PVI<0 THEN GOTO 2310
2290 FOR J=1 TO 10000
2300 WGI=WGI-SD
2310 PVI=PTOT*(1-WGI)/(1-(1-(HV/MG))*WGI)
2320 AE=(200*SC^2*DO*WB^2*WGI^2/21)+(10*SC^3*DO*WB*WGI^3)
2330 BE1=(-100*WB*SC*WGI*(WGI-WB)^2/21)+(2*SC*WGI*(WGI-WB)^3)
2340 BE2=(-8*SC^2*WGI^2*(WGI-WB)^2)
2350 BE=BE1+BE2
2360 CE=40*CO*(WGI-WB)^4/28
2370 DE=-6*X*CO*(WGI-WB)^5/3
2380 BC=BE/AE
2390 CC=CE/AE
2400 DC=DE/AE
2410 PE=CC-(CC^2/2)
2420 QE=CC-(DC*CC/3)+(2*CC^3/27)
2430 ZE=(-QE/2)+(-QE/2)

```



```

2440      ZE=ZF1^(1/3)
2450      YF=ZF-(PE/(3*ZE))
2460      SP=YE-(BC/3)
2470      TGI=(SP*HFG*MEUL/KL)+T(3,K-1)
2480      TGIF=(9*TGI/5)+32
2490      REM
2500      A2=1.8893
2510      B2=-30.579
2520      C2=79.047-TGIF
2530      FA1=B2^2
2540      FA0=(FA1-4*A2*C2)
2550      FA=(B2+FA0^.5)/(2*A2)
2560      PWS=(2.718283)^FA
2570      PWSI=PWS*101325!/29.921
2580      IF PVI=PWSI THEN TGIC=TGI
2590      IF (PWSI-PVI)<LIM THEN TGIC=TGI ELSE 2610
2600      IF TGIC=TGI THEN 2620
2610      NEXT J
2620      IF PWSI-PVI<0 THEN MCR(K-1)=MCR(K-2)
2630      SD=.000001
2640      LIM=1
2650      RETURN
2660      REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2670      MSS=10
2680      FOR Z=1 TO 500-2
2690          IF MCR(Z)<MSS THEN MSS=MCR(Z)
2700          IF MCR(Z+1)=0 THEN GOTO 2750
2710          MINAV=(MCR(Z)+MCR(Z+1))/2
2720          MTOT=MTOT+MINAV
2730          NUMINT=NUMINT+1
2740          MCRAVG=MTOT/NUMINT
2750      NEXT Z
2760      RETURN

```

SUB-PROGRAM 5

```

10 DIM T(3,1010),A(3),B(3),C(3),F(3),QINAV(500),F2(3),MCR(505)
20 REM
30 INPUT"this subprogram calculates the mass condensation rate and the resulting
.....heat loss when indoor condensation occurs on single glazed windows at
.....constant conditions if you want to use it print (y) else print (n)";B$
40 REM
50 IF B$="y" THEN GOTO 80 ELSE RUN"cond"
60 REM
70 REM *****keenan,keyes,hill,and moore formula constants*****
80 F2(1)=-741.9242
90 F2(2)=-29.721
100 F2(3)=-11.55286
110 F2(4)=-.8685635
120 F2(5)=.1094098
130 F2(6)=.439993
140 F2(7)=.2520658
150 F2(8)=.0521866
160 REM *****input of atmospheric,indoor conditions and window related data****
170 REM
180 PRINT
190 INPUT"the indoor temperature in c=";TI
200 PRINT
210 INPUT"ambient temperature in c=";TOU
220 PRINT
230 INPUT"indoor relative humidity=";RH
240 PRINT
250 INPUT"the wind speed in meters per second=";WV
260 PRINT
270 INPUT"the wind angle of attack from normal in degrees=";ANG
280 PRINT
290 INPUT"window (or wall) height in meters=";L
300 PRINT
310 INPUT"glass thickness in meters=";M
320 PRINT
330 INPUT"time interval in seconds=";DT
340 GOSUB 1980
350 REM the average outdoor(tou),indoor(tia) and wall temperatures(twiu,twak)
will be assumed as following
360 REM
370 TIA=TI
380 TWIA=TIA+6
390 TIAK=TIA+273
400 TWIAK=TWIA+273
410 REM the air properties are evaluated at the assumed average film
temperatures which is equal to 23 deg. c for outdoor conditions
and 21 deg. c for indoor conditions
420 REM
430 MEUI=.4937E-05
440 ALFAI=.0000211
450 KAI=.025764
460 TFIIMI=(18+24)/2+273
470 BETAI=1/TFIIMI
480 SEGMA=5.673E-08
490 FSR=1
500 PAI=9.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
510 IF PAI<1E+09 THEN HCI=.59*KAI*(PAI^.25)/L
520 IF PAI>1E+09 THEN HCI=.1*KAI*(PAI^.333)/L
530 HPI=SEGMA*FSR*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
540 HI=HCI+HPI
550 IF ANG>0 AND ANG<30 THEN HO=7.77+2.9*WV
560 IF ANG>30 AND ANG<60 THEN HO=7.5+3.26*WV
570 IF ANG>60 AND ANG<100 THEN HO=7.52+4.1*WV
580 IF ANG>100 AND ANG<140 THEN HO=6.83+2.63*WV
590 IF ANG>140 AND ANG<180 THEN HO=6.27+2.65*WV
600 KGL=.78
610 DA=4/3

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```

620 R1=1/HI
630 R2=(1/KGL*DX)+R1
640 R3=(1/KGL*M)+R1
650 R=R3+1/HO
660 T(3,1)=TI-(R1/R)*(TI-TOUT)
670 IF T(3,1)>=TSAT-1 THEN GOTO 680 ELSE 730
680 PRINT "condensation will not occur at these conditions, start again if you wis
h to try at other conditions else press: ctrl-break"
690 GOTO 190
700 REM water and vapour properties are evaluated at assumed indoor average
710 REM temperatures
720 REM
730 ROWL=1000
740 ROWV=.016
750 HFG=2480000
760 KL=.6
770 MEUL=.0018
780 CV=840
790 PCW=2700
800 CVA=1005
810 PCWA=1.2
820 MW=18
830 PG=8310
840 DC=.000026
850 MEUA=.0000172
860 MU=.0000123
870 MUL=1.776E-06
880 SC=MU/DC
890 MV=MW
900 MG=28.37
910 WBI=1
920 SD=.00001
930 LIM=5
940 DC=MEUL*ROWL/(MEUA*PCWA)
950 CO=1/DO
960 CPL=4180
970 PRL=CPL*MEUL/KL
980 X=(MG-MV)/(MG-WB*(MG-MV))
990 C1=((9.8*(MG-MV)/(4*MU^2)))/(MG-(MG-MV)*WB)^.25
1000 T(2,1)=TI-(R2/R)*(TI-TOUT)
1010 T(1,1)=TI-(R3/R)*(TI-TOUT)
1020 D0=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^-.25
1030 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^(1/3)
1040 D3=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^-.25
1050 D4=(MEUL*(TSAT-T(3,1))*L)^-.25
1060 D5=(L*(TSAT-T(3,1))/(MEUL*HFG))^-.75
1070 FO=(KGL*DT)/(ROW*CV*DX^2)
1080 S=(DT*HO)/(PCW*CV*DX)
1090 CO1=(DT*HI)/(ROW*CV*DX)
1100 CA=(2*DT*HFG)/(ROW*CV*DX)
1110 REM
1120 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
1130 REM
1140 FOR K=3 TO 502
1150 GOSUB 2190
1160 IF PWSI-PVI<0 THEN GOTO 1210
1170 FO1=((9.8*L^3)/(4*MUL^2))^-.25
1180 FO2=(CPL*(TGI-T(3,K-1)))/(PRL*HFG))^-.75
1190 FO3=4*FO1*FO2*HFG*MEUL/3
1200 MCF(K-1)=FO3/(HFG*L)
1210 F(1)=T(1,K-1)+(2*S*TOUT)
1220 F(2)=T(2,K-1)
1230 F(3)=T(3,K-1)+(CA*MCF(K-1))+(2*CO1*TI)
1240 B(1)=1+(C*FO)+(2*S)
1250 C(1)=-C*F(1)

```

```

1260 A(2)=-FO
1270 B(2)=1+(2*FO)
1280 C(2)=-FO
1290 A(3)=-2*FO
1300 B(3)=1+(2*FO)+(2*CO1)
1310 GOSUB 1660
1320 NEXT K
1330 GOSUB 1770
1340 GOSUB 2590
1350 REM *****printing out the needed information*****
1360 REM
1370 INPUT"insert the time in sec. at which the first temperature is needed";T1
1380 INPUT"insert the time in sec. at which the last temperature is needed";T2
1390 IF T2>DT*500 THEN T2=DT*500
1400 U=(T1\DT)+1
1410 V=(T2\DT)+1
1420 PRINT "
1430 PRINT"***** the nodal temperatures at the needed time*****"
1440 REM
1450 PRINT" note: t3=indoor facing temperature and t1=outdoor facing temperature"
-----
1460 PRINT "time in sec.", "t1 cel.", "t2 cel.", "t3 cel.", "mcr kg/s"
1470 PRINT "-----", "-----", "-----", "-----", "-----"
1480 FOR A=U TO V
1490 PRINT (A-1)*DT, T(1,A), T(2,A), T(3,A), MCR(A)
1500 NEXT A
1510 PRINT"
1520 PRINT"the total heat loss near the steady state stages="MCC*HFG"w/m^2"
1530 PRINT"the convective heat loss rate near the steady state stages="
QINMAX"w/m^2"
1540 PRINT"the maximum interval convective heat loss rates="QINMAX"w/m^2"
1550 PRINT "the average convective heat loss rates="QAVG"w/m^2"
1560 PRINT "the average convective heat loss rates="QAVG"w/m^2"
1570 PRINT "the mass condensation rate near the the steady state stages="
MSC"kg/sec per m^2"
1580 PRINT"the average mass condensation rate during condensation="
MCRAVG"kg/sec.per m^2"
1590 IF TSS=0 THEN GOTO 1610
1600 PRINT"the time needed to reach steady state stages="TSS"seconds"
1610 REM
1620 INPUT"do you want to know the temperatures at another time interval Y or N";A$
1630 IF A$="y" THEN GOTO 1370 ELSE 1635
1635 PRINT"
1640 INPUT"do you want to try at other conditions y or n";E$
1650 IF E$="y" THEN 180 ELSE RUN"cond"
1660 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND
ENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM
1670 REM
1680 FOR I=2 TO 3
1690 D=A(I)/B(I-1)
1700 B(I)=B(I)-(C(I-1)*D)
1710 F(I)=F(I)-(C(I-1)*D)
1720 NEXT I
1730 T(3,K)=F(3)/B(3)
1740 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1750 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1760 RETURN
1770 REM THIS SUBROUTINE CALCULATES THE AVERAGE CONVECTIVE HEAT LOSS RATE AND
FINDS THE MAXIMUM AND MINIMUM INTERVAL CONVECTIVE HEAT LOSS
1780 REM
1790 TTTAVG=0

```

```

1800     QINMAX=0
1810     QINMIN=100
1820     H=500
1830     FOR H1=1 TO H-2
1840         Q1=(T(1,H1)+T(1,H1+1))/2
1850         QINAV(H1)=H0*(Q1-T(1,1))
1860         IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
1870         IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
1880         QINTOT=QINTOT+QINAV(H1)
1890         IF TESS<T(3,H1) THEN TESS=T(3,H1)
1900         IF T(3,H1+1)=T(3,H1) THEN 1910 ELSE 1930
1910         IF T(2,H1+1)=T(2,H1) THEN 1920 ELSE 1930
1920         IF T(1,H1+1)=T(1,H1) THEN GOTO 1950 ELSE 1960
1930     NEXT H1
1940     GOTO 1960
1950     TSS=DT*H1
1960     QAVG=QINTOT/H1
1970     RETURN
1980 REM this subroutine calculates the saturation temperature at the
      beginning of condensation using keenan, keyes, hill, and moore formula
1990 REM
2000     F=0
2010     FOR XO=1 TO 8
2020         F1=F2(XO)*((.65-.01*TI)^(XO-1))
2030         F=F+F1
2040     NEXT XO
2050     TKEL=TI+273.15
2060     L1=LOG(217.99)+((374.136-TI)*F*(.01/TKEL))
2070     PWS=EXP(L1)
2080     PWATM=PH*PWS
2090     HUR=.62198*PWATM
2100     WB=1/(HUR+1)
2110     PA=1013251*PWATM
2120     PTOT=PA+1013251
2130     PW=29.921*PWATM
2140     FA=LOG(PW)
2150     TSATF=79.047+(30.579*FA)+(1.8893*FA^2)
2160     TSAT=(5/9)*(TSATF-32)
2170     IF TSAT>TI THEN TSAT=TI
2180     RETURN
2190 REM this subroutine calculates the interfacial temperature at constant
      conditions, using (E. ROSE) simplified equation
2200 REM
2210     FOR J=1 TO 10000
2220         IF PWS1-PVI<0 THEN GOTO 2240
2230         WGI=WGI-SD
2240         PVI=PTOT*(1-WGI)/(1-(1-(MV/MG))*WGI)
2250         AE=(200*SC^2*DO*WB^2*WGI^2/21)+(10*SC^3*DO*WB*WGI^3)
2260         BE1=(-100*WB*SC*WGI*(WGI-WB)^2/21)+(2*SC*WGI*(WGI-WB)^3)
2270         BE2=(-8*SC^2*WGI^2*(WGI-WB)^2)
2280         BE=BE1+BE2
2290         CE=40*CO*(WGI-WB)^4/28
2300         DE=-8*X*CO*(WGI-WB)^5/3
2310         BC=BE/AE
2320         CC=CE/AE
2330         DC=DE/AE
2340         PE=CC-(BC^2/3)
2350         QE=DC-(BC*CC/3)+(2*BC^3/27)
2360         ZE1=(-QE/2)+(-QE/2)
2370         ZE=ZE1^(1/3)
2380         YE=ZE-(PE/(3*ZE))
2390         SP=YE-(BC/3)
2400         TGI=(SP*HFO*HFUL/KL)+T(3,K-1)
2410         TGI1=(TGI/5)+30
2420 REM
2430     A2=1.0000

```

```
2440 B2=-30.579
2450 C2=79.647-TGIF
2460 FA1=B2^2
2470 FA0=(FA1-4*A2*C2)
2480 FA=(B2+FA0*.5)/(2*A2)
2490 PWS=(2.718282)^FA
2500 PWSI=PWS*1013251/29.921
2510 IF PVI=PWSI THEN TGIC=TGI
2520 IF (PWSI-PVI)<LIM THEN TGIC=TGI ELSE 2540
2530 IF TGIC=TGI THEN GOTO 2550
2540 NEXT J
2550 IF PWSI-PVI<0 THEN MCP(K-1)=MCP(K-2)
2560 SD=.000001
2570 LIM=1
2580 RETURN
2590 REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2600 MSS=10
2610 FOR Z=1 TO 500-2
2620 IF MCP(Z)<MSS THEN MSS=MCP(Z)
2630 IF MCP(Z+1)=0 THEN GOTO 2650
2640 MINAV=(MCP(Z)+MCP(Z+1))/2
2650 MTOT=MTOT+MINAV
2660 NUMINT=NUMINT+1
2670 MCPAVG=MTOT/NUMINT
2680 NEXT Z
2690 RETURN
```

SUB-PROGRAM 6

```

10 DIM T(7,510),A(7),B(7),C(7),F(7),QINAV(500),F2(3),XCF(5*5)
20 REM
30 INPUT "this subprogram calculates the mass condensation rate and the resulting
    heat loss when indoor condensation occurs on double glazed windows at
    constant conditions if you want to use it print (y) else print (n)";Bs
40 IF Bs="y" THEN GOTO 70 ELSE RUN"cond"
50 REM
60 REM *****keenan,keys,nill,and moore formula constants*****
70 F2(1)=-741.9242
80 F2(2)=-29.721
90 F2(3)=-11.55286
100 F2(4)=-.8685635
110 F2(5)=.1094098
120 F2(6)=.439993
130 F2(7)=.2520658
140 F2(8)=.0521868
150 REM *****input of atmospheric,indoor conditions and window related data****
160 REM
170 PRINT
180 INPUT "the indoor temperature in c=";TI
190 PRINT
200 INPUT "ambient temperature in c=";TOUT
210 PRINT
220 INPUT "indoor relative humidity=";RH
230 PRINT
240 INPUT "the wind speed in meters per seconds=";WV
250 PRINT
260 INPUT "the wind angle of attack from normal in degrees=";ANG
270 PRINT
280 INPUT "window (OR WALL) height in meters=";L
290 PRINT
300 INPUT "glass thickness in meters(air space is taken twice that)=";M
310 PRINT
320 INPUT "time interval in seconds=";DT
330 PRINT
340 GOSUB 2150
350 REM the average outdoor(toa),indoor(tia) and wall temperatures(twia,twoa)
    will be assumed as following
360 REM
370 TIA=TI
380 TWIA=TIA+6
390 TIAK=TIA+273
400 TWIAK=TWIA+273
410 REM the air properties are evaluated at the assumed average film
    temperatures which is equal to 28 deg. c for outdoor conditions
    and 21 deg. c for indoor conditions
420 REM
430 MEUI=1.4937E-05
440 ALFAI=.0000211
450 KAI=.025764
460 TFI1M1=(18+24)/2+273
470 BETAI=1/TFI1M1
480 SEGMA=.673E-08
490 FSP=1
500 PAI=.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
510 IF PAI<1E+09 THEN HCI=.59*KAI*(PAI^-.25)/L
520 IF PAI>1E+09 THEN HCI=.1*KAI*(PAI^-.333)/L
530 HRI=SEGMA*FSP*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
540 HI=HCI+HRI
550 IF ANG>0 AND ANG<30 THEN HO=7.77+2.9*WV
560 IF ANG>30 AND ANG<60 THEN HO=7.5+3.26*WV
570 IF ANG>60 AND ANG<100 THEN HO=7.52+4.1*WV
580 IF ANG>100 AND ANG<140 THEN HO=6.93+2.63*WV
590 IF ANG>140 AND ANG<180 THEN HO=6.27+2.65*WV
600 KCL=.75

```

```

610 DX=(2*M+2*M)/6
620 KAS=.054
630 KAGL=(2*KAS*KGL)/(KAS+KGL)
640 R1=1/HI
650 P2=(1/KGL*DX)+R1
660 P3=(1/KAGL*DX)+P2
670 P4=(1/KAS*DX)+P3
680 P5=(1/KAS*DX)+P4
690 P6=(1/KAGL*DX)+P5
700 P7=(1/KGL*DX)+P6
710 R=P7+1/HO
720 T(7,1)=TI-(R1/R)*(TI-TOUT)
730 T(6,1)=TI-(P2/R)*(TI-TOUT)
740 T(5,1)=TI-(P3/R)*(TI-TOUT)
750 T(4,1)=TI-(P4/R)*(TI-TOUT)
760 T(3,1)=TI-(P5/R)*(TI-TOUT)
770 T(2,1)=TI-(P6/R)*(TI-TOUT)
780 T(1,1)=TI-(P7/R)*(TI-TOUT)
790 IF T(7,1)>TSAT-1 THEN GOTO 800 ELSE 850
800 PRINT"condensation will not occur at these conditions,start again if you wis
h to try at other conditions else press ctrl-break"
810 GOTO 180
820 REM water and vapour properties are evaluated at assumed indoor average
710 REM temperatures
830 REM
840 REM
850 ROWL=1000
860 ROWV=.016
870 HFG=2400000!
880 KL=.6
890 MEUL=.001
900 CV=240
910 POW=2700
920 D0=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^-.25
930 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^(1/3)
940 D3=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^-.25
950 D4=(MEUL*(TSAT-T(7,1))*L)^-.25
960 D5=(L*(TSAT-T(7,1))/(MEUL*HFG))^-.75
970 CVA=1005
980 PCWA=1.2
990 MW=18
1000 PG=8310
1010 DC=.000026
1020 HL=((DC*HI)*((KAI/(POWA*CVA*DC))^-.25))/KAI
1030 FO=(KGL*DT)/(POW*CV*DX^2)
1040 S=(DT*HO)/(POW*CV*DX)
1050 POWA=1.3
1060 CVA=1005
1070 FGA=(DT*KAGL)/(POWA*CV*DX^2)
1080 FGA=(KAGL*DT)/(POWA*CV*DX^2)
1090 FA=(KAS*DT)/(POWA*CV*DX^2)
1100 CO=(DT*HI)/(POW*CV*DX)
1110 CA=(2*DT*HFG)/(POW*CV*DX)
1120 REM
1130 REM *****CALCULATING NODEL TEMPERATURES AS CONDENSATION OCCURS*****
1140 J=1
1150 N=J+1
1160 FOR K=N TO 502
1170 TM=T(3,K-1)+273
1180 GOSUB 2360
1190 MFR(K-1)=(HD*MW*(PA-PC))/(PG*TM)
1200 F(1)=T(1,K-1)+(2*S*TOUT)
1210 F(2)=T(2,K-1)
1220 F(3)=T(3,K-1)
1230 F(4)=T(4,K-1)
1240 F(5)=T(5,K-1)

```



```

1250 F(6)=T(6,K-1)
1260 F(7)=T(7,K-1)+(CA*MCR(K-1))+(2*CG*TI)
1270 B(1)=1+(2*FO)+(2*S)
1280 C(1)=-2*FO
1290 A(2)=-FO
1300 B(2)=1+FG+FCA
1310 C(2)=-FOA
1320 A(3)=-FG
1330 B(3)=1+FG+FA
1340 C(3)=-FA
1350 A(4)=-FA
1360 B(4)=1+(2*FA)
1370 C(4)=-FA
1380 A(5)=-FA
1390 B(5)=1+FG+FA
1400 C(5)=-FG
1410 A(6)=-FOA
1420 B(6)=1+FG+FCA
1430 C(6)=-FO
1440 A(7)=-2*FO
1450 B(7)=1+(2*FO)+(2*CO)
1460 GOSUB 1820
1470 NEXT K
1480 GOSUB 1970
1490 GOSUB 2480
1500 REM ***** printing out the needed information *****
1510 REM
1520 INPUT "insert the time in sec. at which the first temperature is needed"
1530 " ; T1
1540 INPUT "insert the time in sec. at which the last temperature is needed"
1550 " ; T2
1560 IF T2>500*DT THEN T2=500*DT
1570 U=(T1/DT)+1
1580 V=(T2/DT)+1
1590 PRINT "
1600 PRINT "***** the nodal temperatures at the needed time*****"
1610 REM
1620 PRINT " note: t7=indoor facing temperature and t1=outdoor facing temperature"
1630 " -----
1640 PRINT "time in sec.", "t1 cel.", "t4 cel.", "t7 cel.", "mcr kg/s"
1650 PRINT "-----", "-----", "-----", "-----", "-----"
1660 FOR A=U TO V
1670 PRINT (A-1)*DT, T(1,A), T(4,A), T(7,A), MCR(A)
1680 NEXT A
1690 PRINT "
1700 PRINT "the total heat loss rate by condensation near the steady
1710 state stage="MSS*HFG"w/m^2"
1720 PRINT "mass condensation rate near steady state stage="MSS"
1730 kg/sec.per m^2"
1740 PRINT "the indoor facing temperature near steady state stage="TFSS"C."
1750 IF TSS=0 THEN GOTO 1720
1760 PRINT "the time needed to reach steady state stage="TSS"seconds"
1770 PRINT "
1780 PRINT "the maximum interval convective heat loss rate="QINMAX"w/m^2"
1790 PRINT "the minimum interval convective heat loss rate="QINMIN"w/m^2"
1800 PRINT "the average convective heat loss rate="QAVG"w/m^2"
1810 PRINT "the average mass condensation rate during condensation="MCAVG"
1820 kg/sec.per m^2"
1830 REM
1840 INPUT "do you want to know the temperatures at another time interval Y or N"
1850 ; A$
1860 IF A$="Y" THEN GOTO 1520 ELSE 1795
1870 PRINT "
1880 INPUT "do you want to try at other conditions Y or N"; B$

```

```

1810 IF E$="y" THEN 170 ELSE RUN"cond"
1820 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND
      ENSATION OCCUR BY TRIDIGONAL MATRIX ALGORITHM
1830 REM
1840 FOR I=2 TO 7
1850   D=A(I)/B(I-1)
1860   B(I)=B(I)-(C(I-1)*D)
1870   F(I)=F(I)-(F(I-1)*D)
1880 NEXT I
1890   T(7,K)=F(7)/B(7)
1900   T(6,K)=(F(6)-(C(6)*T(7,K)))/B(6)
1910   T(5,K)=(F(5)-(C(5)*T(6,K)))/B(5)
1920   T(4,K)=(F(4)-(C(4)*T(5,K)))/B(4)
1930   T(3,K)=(F(3)-(C(3)*T(4,K)))/B(3)
1940   T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
1950   T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
1950   RETURN
1970 REM THIS SUBROUTINE CALCULATES THE INTERVAL AVERAGE HEAT GAIN RATE AND THE
      AVERAGE HEAT GAIN RATE OVER A PERIOD BETWEEN TWO CONDENSATION EVENTS
1980 REM
1990   QINMAX=0
2000   QINMIN=100
2010   H=500
2020   TESS=-50
2030   FOR H1=1 TO H
2040     Q1=(T(1,H1)+T(1,H1+1))/2
2050     QINAV(H1)=HO*(Q1-T(1,1))
2060     IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
2070     IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
2080     QINTOT=QINTOT+QINAV(H1)
2090     IF TESS<T(7,H1) THEN TESS=T(7,H1)
2100     IF T(7,H1+1)=T(7,H1) THEN 2110 ELSE 2130
2110     IF T(3,H1+1)=T(3,H1) THEN 2120 ELSE 2130
2120     IF T(1,H1+1)=T(1,H1) THEN GOTO 2150 ELSE 2100
2130   NEXT H1
2140   GOTO 2160
2150   TSS=DT*H1
2160   QAVG=QINTOT/H1
2170   RETURN
2180 REM this subroutine calculates the saturation temperature at the
      begining of condensation using keenan,keyes,hill,and moore formula
2190 REM
2200   F=0
2210   FOR XO=1 TO 8
2220     F1=F2(XO)*((.65-.01*TI)^(XO-1))
2230     F=F+F1
2240   NEXT XO
2250   TKEL=TI+273.15
2260   L1=LOG(217.99)+((374.136-TI)*F*(.01/TKEL))
2270   PWS=EXP(L1)
2280   PWATM=PH*PWS
2290   PA=101325!*PWATM
2300   PW=29.921*PWATM
2310   FA1=LOG(PW)
2320   TSATF=79.047+(30.579*FA1)+(1.8893*FA1^2)
2330   TSAT=(5/9)*(TSATF-32)
2340   IF TSAT>TI THEN TSAT=TI
2350   RETURN
2360 REM this subroutine calculates the saturation pressure at the window
      temperature using keenan,keyes,hill,and moore formula
2370 REM
2380   F=0
2390   FOR XO=1 TO 8
2400     F1=F2(XO)*((.65-.01*T/7,X-1)^(XO-1))
2410     F=F+F1
2420   NEXT XO

```

```
2430      TKEL=T(7,K-1)+273.15
2440      L1=LOG(217.99)+((374.136-T(7,K-1))*F*(.01/TKEL))
2450      PWS=EXP(L1)
2460      PC=101325!*PWS
2470      RETURN
2480 REM this subroutine calculates the average mass condensation rate
      during condensation"mcravg"
2490      MSS=10
2500      FOR Z=1 TO 500
2510          IF MCR(Z)<MSS THEN MSS=MCR(Z)
2520          IF MCR(Z+1)=0 THEN GOTO 2570
2530          MINAV=(MCR(Z)+MCR(Z+1))/2
2540          MTOT=MTOT+MINAV
2550          NUMINT=NUMINT+1
2560          MCR AVG=MTOT/NUMINT
2570      NEXT Z
2580      RETURN
```

SUB-PROGRAM 7

```

10 DIM T(7,510),A(7),B(7),C(7),F(7),QINAV(500),F2(8),MCP(505)
20 REM
30 INPUT"this subprogram calculates the mass condensation rate and the
   resulting heat loss when indoor condensation occurs on double glazed windows
   at constant conditions if you want to use it print (y) else print (n)";B$
40 IF B$="y" THEN GOTO 70 ELSE RUN"cond"
50 REM
60 REM *****keenan,keyes,hill,and moore formula constants*****
70 F2(1)=-741.9242
80 F2(2)=-29.721
90 F2(3)=-11.55286
100 F2(4)=-.8685635
110 F2(5)=.1094098
120 F2(6)=.439993
130 F2(7)=.2520658
140 F2(8)=.0521868
150 REM *****input of atmospheric,indoor conditions and window related data****
160 REM
170 PRINT"
180 INPUT"the indoor temperature in c=";TI
190 PRINT"
200 INPUT"ambient temperature in c=";TCUT
210 PRINT"
220 INPUT"indoor relative humidity=";RH
230 PRINT"
240 INPUT"the wind speed in meters per second=";WV
250 PRINT"
260 INPUT"the wind angle of attack from normal in degrees=";ANG
270 PRINT"
280 INPUT"window (OR WALL) height in meters=";L
290 PRINT"
300 INPUT"glass thickness in meters(air space is taken twice that)=";M
310 PRINT"
320 INPUT"time interval in seconds=";DT
330 PRINT"
340 GOSUB 2370
350 REM the average outdoor(toa),indoor(tia) and wall temperatures(twia,twoa)
   will be assumed as following
360 REM
370 TIA=TI
380 TWIA=TIA+6
390 TIAK=TIA+273
400 TWIAK=TWIA+273
410 REM the air properties are evaluated at the assumed average film
   temperatures which is equal to 28 deg. c for outdoor conditions
   and 21 deg. c for indoor conditions
420 REM
430 MEUI=1.4937E-05
440 ALFAI=.0000211
450 KAI=.025764
460 TFIIMI=(18+24)/2+273
470 BETAI=1/TFIIMI
480 SEGMA=5.673E-08
490 FSP=1
500 PAI=9.8*BETAI*L^3*(TWIA-TIA)/(MEUI*ALFAI)
510 IF PAI<1E+09 THEN HCI=.59*KAI*(PAI^-.25)/L
520 IF PAI>1E+09 THEN HCI=.1*KAI*(PAI^-.343)/L
530 HPI=SEGMA*FSP*(TIAK^2+TWIAK^2)*(TIAK+TWIAK)
540 HI=HCI+HPI
550 IF ANG=>0 AND ANG<30 THEN HO=7.77+2.9*WV
560 IF ANG=>30 AND ANG<60 THEN HO=7.5+3.25*WV
570 IF ANG=>60 AND ANG<100 THEN HO=7.52+4.1*WV
580 IF ANG=>100 AND ANG<140 THEN HO=6.23+2.63*WV
590 IF ANG=>140 AND ANG<180 THEN HO=5.27+2.65*WV
600 KGL=.78
610 LA=12*WV+L*HO/L

```

```

620 KAS=.054
630 KAGL=(2*KAS*KGL)/(KAS+KGL)
640 R1=1/HI
650 R2=(1/KGL*DX)+R1
660 R3=(1/KAGL*DX)+R2
670 R4=(1/KAS*DX)+R3
680 R5=(1/KAS*DX)+R4
690 R6=(1/KAGL*DX)+R5
700 R7=(1/KGL*DX)+R6
710 R=R7+1/HO
720 T(7,1)=TI-(R1/R)*(TI-TOUT)
730 T(6,1)=TI-(R2/R)*(TI-TOUT)
740 T(5,1)=TI-(R3/R)*(TI-TOUT)
750 T(4,1)=TI-(R4/R)*(TI-TOUT)
760 T(3,1)=TI-(R5/R)*(TI-TOUT)
770 T(2,1)=TI-(R6/R)*(TI-TOUT)
780 T(1,1)=TI-(R7/R)*(TI-TOUT)
790 IF T(7,1)>TSAT-1 THEN GOTO 800 ELSE 850
800 PRINT "Condensation will not occur at these conditions, start again if you wis
      h to try at other conditions else press ctrl-break"
810 GOTO 160
820 REM water and vapour properties are evaluated at assumed indoor average
830 REM temperatures
840 REM
850 ROWL=1000
860 ROWV=.016
870 HFG=2486000
880 KL=.5
890 MEUL=.001
900 CV=840
910 ROW=2700
920 D0=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^25
930 D2=((9.8*ROWL*(ROWL-ROWV)*KL^3)/(MEUL^2))^(1/3)
940 D3=(9.8*ROWL*(ROWL-ROWV)*HFG*KL^3)^25
950 D4=(MEUL*(TSAT-T(7,1))*L)^25
960 D5=(L*(TSAT-T(7,1))/(MEUL*HFG))^75
970 CVA=1005
980 ROWA=1.2
990 MW=18
1000 RG=8310
1010 DC=.000026
1020 HD=((DC*HFI)*((KAI/(ROWA*CVA*DC))^25))/KAI
1030 FO=(KGL*DT)/(ROW*CV*DX^2)
1040 S=(DT*HO)/(ROW*CV*DX)
1050 ROWA=1.3
1060 CVA=1005
1070 MEUA=.0000172
1080 MU=.0000123
1090 MUL=1.776E-06
1100 SC=MU/DC
1110 MV=MW
1120 MG=28.97
1130 WGI=1
1140 SD=.00001
1150 LIM=5
1160 DO=MEUL*ROWL/(MEUA*ROWA)
1170 CO=1/DO
1180 CPL=4180
1190 PRL=CPL*MEUL/KL
1200 X=(MG-MV)/(MG-ME*(MG-MV))
1210 FOA=(DT*KAGL)/(ROW*CV*DX^2)
1220 FGA=(KAGL*DT)/(ROWA*CV*DX^2)
1230 FA=(KAGL*DT)/(ROWA*CV*DX^2)
1240 CVA=(DT*HFG)/(ROW*CV*DX)
1250 CA=(C*DT*HFG)/(ROW*CV*DX)

```

```

1260 REM
1270 REM *****CALCULATING NODAL TEMPERATURES AS CONDENSATION OCCURS*****
1280 FOR K=2 TO 502
1290 GOSUB 2580
1300 IF PWSI-PVI<0 THEN GOTO 1350
1310 F01=((9.8*L^3)/(4*MUL^2))^.25
1320 F02=(CPL*(TGIC-T(7,K-1))/(PPL*HFG))^.75
1330 F03=4*F01*F02*HFG*MEUL/3
1340 MCF(K-1)=F03/(HFG*L)
1350 F(1)=T(1,K-1)+(2*S*TOUT)
1360 F(2)=T(2,K-1)
1370 F(3)=T(3,K-1)
1380 F(4)=T(4,K-1)
1390 F(5)=T(5,K-1)
1400 F(6)=T(6,K-1)
1410 F(7)=T(7,K-1)+(CA*MCF(K-1))+(2*CO1*TI)
1420 Z(1)=1+(2*F0)+(2*S)
1430 C(1)=-2*FC
1440 A(2)=-FO
1450 B(2)=1+FO+FOA
1460 C(2)=-FOA
1470 A(3)=-FG
1480 B(3)=1+FG+FA
1490 C(3)=-FA
1500 A(4)=-FA
1510 B(4)=1+(2*FA)
1520 C(4)=-FA
1530 A(5)=-FA
1540 B(5)=1+FG+FA
1550 C(5)=-FG
1560 A(6)=-FOA
1570 B(6)=1+FO+FOA
1580 C(6)=-FO
1590 A(7)=-2*FC
1600 B(7)=1+(2*FO)+(2*CO1)
1610 GOSUB 2020
1620 NEXT K
1630 GOSUB 2170
1640 GOSUB 2980
1650 REM ***** printing out the needed information *****
1660 REM
1670 INPUT "insert the time in sec. at which the first temperature is needed
";T1
1680 INPUT "insert the time in sec. at which the last temperature is needed"
;T2
1690 IF T2>DT*500 THEN T2=DT*500
1700 U=(T1/DT)+1
1710 V=(T2/DT)+1
1720 PRINT "
1730 PRINT "***** the nodal temperatures at the needed time*****"
1740 PRINT "
1750 REM
1760 PRINT " note: t7=indoor facing temperature and t1=outdoor facing temperature
-----
1770 PRINT "
1780 PRINT "TO="TOUT"C.", "TI="TI"C.", "PH="PH*100"%", "wV="wV"m/s"
1790 PRINT "M="THI"m", "AS="2*THI"m", "DT="DT"SEC.", "TSAT="TSAT"C."
1800 PRINT "
1810 PRINT "time in ser.", "t1 cel.", "t4 cel.", "t7 cel.", "mcr kg/s"
1820 PRINT "-----", "-----", "-----", "-----", "-----"
1830 FOR A=U TO V
1840 PRINT (A-1)*DT, T(1,A), T(4,A), T(7,A), MCF(A)
1850 NEXT A
1860 PRINT "

```

```

1870 PPINT "the total heat loss rate by condensation near the steady
      state stage="MSS*HFG"w/m^2"
1880 PPINT"mass condensation rate near steady state stage="MSS"
      kg/sec.per m^2
1890 PPINT"the indoor facing temperature near steady state stage="TESS"C."
1900 IF TSS=0 THEN GOTO 1920
1910 PPINT "the time needed to reach steady state stage="TSS"seconds"
1920 PPINT"
1930 PPINT"the maximum interval convective heat loss rate="QINMAX"w/m^2"
1940 PPINT"the minimum interval convective heat loss rate="QINMIN"w/m^2"
1950 PPINT"the average convective heat loss rate="QAVG"w/m^2"
1960 PPINT"the average mass condensation rate during condensation="MCPAVG"
      kg/sec.per m^2"

1970 REM
1980 INPUT"do you want to know the temperatures at another time interval Y or N"
;AS
1990 IF At="y" THEN GOTO 1670 ELSE 1995
1995 PPINT"
2000 INPUT"do you want to try at other conditions y or n";ES
2010 IF Es="y" THEN 170 ELSE RUN"cond"
2020 REM THIS SUBROUTINE SOLVES THE THREE NODAL EQUATIONS SIMULTANEOUSLY AS COND
      ENSATION OCCUR BY TRIDIGONAL MATPIX ALGORITHM
2030 REM
2040 FOR I=2 TO 7
2050 D=A(I)/B(I-1)
2060 B(I)=B(I)-(C(I-1)*D)
2070 F(I)=F(I)-(F(I-1)*D)
2080 NEXT I
2090 T(7,K)=F(7)/B(7)
2100 T(6,K)=(F(6)-(C(6)*T(7,K)))/B(6)
2110 T(5,K)=(F(5)-(C(5)*T(6,K)))/B(5)
2120 T(4,K)=(F(4)-(C(4)*T(5,K)))/B(4)
2130 T(3,K)=(F(3)-(C(3)*T(4,K)))/B(3)
2140 T(2,K)=(F(2)-(C(2)*T(3,K)))/B(2)
2150 T(1,K)=(F(1)-(C(1)*T(2,K)))/B(1)
2160 RETURN
2170 REM THIS subroutine CALCULATES THE INTERVAL AVERAGE HEAT GAIN RATE AND THE
      AVERAGE HEAT GAIN RATE OVER A PERIOD BETWEEN TWO CONDENSATION EVENTS
2180 REM
2190 QINMAX=0
2200 QINMIN=100
2210 H=500
2220 FOR H1=1 TO H
2230 Q1=(T(1,H1)+T(1,H1+1))/2
2240 QINAV(H1)=HO*(Q1-T(1,1))
2250 IF QINAV(H1)<QINMIN THEN QINMIN=QINAV(H1)
2260 IF QINAV(H1)>QINMAX THEN QINMAX=QINAV(H1)
2270 QINTOT=QINTOT+QINAV(H1)
2280 IF TESS<T(7,H1) THEN TESS=T(7,H1)
2290 IF T(7,H1+1)=T(7,H1) THEN 2300 ELSE 2320
2300 IF T(3,H1+1)=T(3,H1) THEN 2310 ELSE 2320
2310 IF T(1,H1+1)=T(1,H1) THEN GOTO 2340 ELSE 2350
2320 NEXT H1
2330 GOTO 2350
2340 TSS=DT*H1
2350 QAVG=QINTOT/H1
2360 RETURN
2370 REM this subroutine calculates the saturation temperature at the
      begining of condensation using keenan,keyes,hill,and moore formula
2380 REM
2390 F=0
2400 FOR XO=1 TO 8
2410 F1=F2(XO)*((.65-.01*T1)^(XO-1))
2420 F=F+F1
2430 NEXT XO
2440 *KFL=T1+272.15

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2450 L1=LOG(217.99)+((374.136-TI)*F*(.01/TKEL))
2460 PWS=EXP(L1)
2470 PWATM=RH*PWS
2480 HUP=.62198*PWATM
2490 WB=1/(1+HUP)
2500 PA=1013251*PWATM
2510 PTOT=1013251+PA
2520 PW=29.921*PWATM
2530 FA2=LOG(PW)
2540 TSATF=79.047+(30.579*FA2)+(1.8893*FA2^2)
2550 TSAT=(5/9)*(TSATF-32)
2560 IF TSAT>TI THEN TSAT=TI
2570 RETURN
2580 REM this subroutine calculates the saturation pressure at the window
      temperature using keenan, keyes, hill, and moore formula
2590 REM
2600 FOR J=1 TO 10000
2610 IF PWSI-PVI<0 THEN GOTO 2630
2620 WGI=WGI-SD
2630 PVI=PTOT*(1-WGI)/(1-(1-(MV/MG))*WGI)
2640 AE=(200*SC^2*DO*WB^2*WGI^2/21)+(10*SC^3*DO*WB*WGI^3)
2650 BE1=(-100*WB*SC*WGI*(WGI-WB)^2/21)+(2*SC*WGI*(WGI-WB)^3)
2660 BE2=(-8*SC^2*WGI^2*(WGI-WB)^2)
2670 BE=BE1+BE2
2680 CE=40*CO*(WGI-WB)^4/28
2690 DE=-8*X*CO*(WGI-WB)^5/3
2700 BC=BE/AE
2710 CC=CE/AE
2720 DC=DE/AE
2730 PE=CC-(BC^2/3)
2740 CE=DC-(BC*CC/3)+(2*BC^3/27)
2750 ZF1=(-QE/2)+ABS(QE/2)
2760 ZE=ZF1^(1/3)
2770 YE=ZE-(PE/(3*ZF))
2780 SP=YE-(BC/3)
2790 TGI=(SP*HFG*MFUL/KL)+T(7,K-1)
2800 TGIF=(9*TGI/5)+32
2810 REM
2820 A2=1.8893
2830 B2=-30.579
2840 C2=79.047-TGIF
2850 FA1=B2^2
2860 FA01=(FA1-4*A2*C2)
2870 FAA=(B2+FA01*.5)/(2*A2)
2880 PWS=(2.718282)^FAA
2890 PWSI=PWS*1013251/29.921
2900 IF PVI=PWSI THEN TGIC=TGI
2910 IF PWSI-PVI<LIM THEN TGIC=TGI ELSE 2930
2920 IF TGIC=TGI THEN GOTO 2940
2930 NEXT J
2940 IF PWSI-PVI<0 THEN MCF(K-1)=MCF(K-2)
2950 SD=.000001
2960 LIM=1
2970 RETURN
2980 REM this subroutine calculates the average mass condensation rate
      during condensation "mcravg"
2990 MSS=10
3000 FOR Z=1 TO 500
3010 IF MCF(Z)<MSS THEN MSS=MCF(Z)
3020 IF MCF(Z+1)=0 THEN GOTO 3070
3030 MINAV=(MCF(Z)+MCF(Z+1))/2
3040 MTOT=MTOT+MINAV
3050 NUMINT=NUMINT+1
3060 MCFAVG=MTOT/NUMINT
3070 NEXT Z
3080 RETURN

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APPENDIX B
PROGRAM OUTPUT SAMPLES

APPENDIX B-1

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 30 C. TI= 18 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 28.2047 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcg kg/s
0	23.78474	24.08806	24.39138	7.046893E-06
60	23.80519	24.12197	24.47984	6.854508E-06
120	23.83641	24.16135	24.53399	6.738223E-06
180	23.87119	24.20011	24.57599	6.660327E-06
240	23.90621	24.23711	24.61285	6.582122E-06
300	23.94015	24.27217	24.64668	6.503606E-06
360	23.97252	24.3053	24.67822	6.424769E-06
420	24.00321	24.33657	24.70781	6.385241E-06
480	24.0323	24.36623	24.7361	6.305913E-06
540	24.0593	24.39421	24.76251	6.26613E-06
600	24.08584	24.42072	24.78774	6.226261E-06
660	24.11053	24.44588	24.81172	6.146266E-06
720	24.13382	24.46955	24.83396	6.106142E-06
780	24.15583	24.49194	24.85518	6.065928E-06
840	24.17665	24.51313	24.8753	6.025611E-06
900	24.19636	24.5332	24.89434	5.985214E-06
960	24.215	24.55217	24.91233	5.944735E-06
1020	24.23262	24.5701	24.92931	5.904155E-06
1080	24.24928	24.58704	24.94532	5.904155E-06
1140	24.26511	24.60321	24.9609	5.863487E-06
1200	24.28012	24.6185	24.97541	5.822716E-06
1260	24.29429	24.63291	24.98899	5.781859E-06
1320	24.30764	24.64647	25.00172	5.781859E-06
1380	24.32032	24.6594	25.01416	5.740896E-06
1440	24.3323	24.67159	25.02565	5.740896E-06
1500	24.34369	24.68322	25.03685	5.699845E-06
1560	24.35445	24.69416	25.04713	5.658691E-06
1620	24.36456	24.7044	25.05665	5.658691E-06
1680	24.37413	24.71416	25.06601	5.658691E-06
1740	24.38326	24.7235	25.07505	5.617435E-06
1800	24.39188	24.73225	25.08322	5.617435E-06
1860	24.40005	24.74059	25.09123	5.57608E-06
1920	24.40773	24.74837	25.09844	5.57608E-06
1980	24.41501	24.75578	25.10554	5.57608E-06
2040	24.42194	24.76287	25.1124	5.53462E-06
2100	24.42845	24.76947	25.11848	5.53462E-06
2160	24.43461	24.77574	25.12447	5.53462E-06
2220	24.44049	24.78174	25.13028	5.493062E-06
2280	24.44598	24.7873	25.13534	5.493062E-06
2340	24.45117	24.79257	25.14037	5.493062E-06
2400	24.4561	24.79761	25.14524	5.493062E-06

the heat gain near the steady state stage= 6.174884 w/m^2
the mass condensation rate near the steady state stage= 5.283648E-06
kg/s per m^2
the indoor facing surface temperature near steady state= 24.53964 c.
the time needed to reach steady state stage= 8040 seconds

the maximum interval heat gain rate= 6.174884 w/m^2
the minimum interval heat gain rate= 8.364024E-02 w/m^2
the average heat gain rate = 5.32365 w/m^2
the average mass condensation rate during condensation= 5.34539E-06
kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 30 C. TI= 18 C. RH= 100 %
M= .01 m DT= 60 SEC. TSAT= 30 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcg kg/s
0	23.78474	24.08806	24.39138	1.217202E-05
60	23.82007	24.14664	24.54419	1.178694E-05
120	23.87384	24.2144	24.63707	1.156036E-05
180	23.9336	24.28093	24.70885	1.139761E-05
240	23.99363	24.34426	24.77164	1.126695E-05
300	24.05173	24.40424	24.82942	1.113559E-05
360	24.11714	24.46093	24.88339	1.100381E-05
420	24.15965	24.51444	24.93406	1.08715E-05
480	24.20926	24.5649	24.98167	1.077191E-05
540	24.25615	24.61259	25.02682	1.067202E-05
600	24.30047	24.65767	25.06949	1.057181E-05
660	24.34233	24.70024	25.10975	1.047128E-05
720	24.38185	24.74042	25.14767	1.040409E-05
780	24.41923	24.77846	25.18378	1.030302E-05
840	24.45451	24.81433	25.21759	1.023546E-05
900	24.48785	24.84824	25.24971	1.016774E-05
960	24.51937	24.8803	25.28009	1.006589E-05
1020	24.54907	24.91045	25.30836	1.003187E-05
1080	24.57716	24.93905	25.33558	9.963703E-06
1140	24.60371	24.96605	25.36113	9.895389E-06
1200	24.62877	24.99152	25.38513	9.826905E-06
1260	24.65238	25.0155	25.40768	9.7926E-06
1320	24.67471	25.03822	25.42927	9.723889E-06
1380	24.69577	25.05961	25.44936	9.683459E-06
1440	24.71566	25.07985	25.46856	9.654996E-06
1500	24.7345	25.09902	25.48679	9.585959E-06
1560	24.75223	25.11702	25.50361	9.551372E-06
1620	24.76896	25.13401	25.51966	9.516747E-06
1680	24.78475	25.15007	25.53485	9.482073E-06
1740	24.79967	25.16524	25.54921	9.44736E-06
1800	24.81376	25.17957	25.56274	9.412602E-06
1860	24.82706	25.19308	25.57548	9.412602E-06
1920	24.8397	25.20598	25.58791	9.377811E-06
1980	24.85166	25.21817	25.59945	9.342974E-06
2040	24.86295	25.22964	25.61023	9.308084E-06
2100	24.87358	25.24042	25.62033	9.308084E-06
2160	24.88365	25.2507	25.63021	9.273152E-06
2220	24.89317	25.26037	25.6393	9.238191E-06
2280	24.90211	25.26944	25.64774	9.238191E-06
2340	24.91059	25.27807	25.65602	9.203168E-06
2400	24.91857	25.28617	25.66359	9.203168E-06

the heat gain near the steady state stages= 10.38212 w/m²
the mass condensation rate near the steady state stages= 8.886002E-06
kg/s per m²
the indoor facing surface temperature near steady states= 25.05399 c.
the time needed to reach steady state stages= 12600 seconds

the maximum interval heat gain rates= 10.38212 w/m²
the minimum interval heat gain rates= .1444865 w/m²
the average heat gain rate= 9.475371 w/m²
the average mass condensation rate during condensation= 8.991214E-06
kg/sec.per m²

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 34 C. TI= 24 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 32.15934 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcr kg/s
0	28.77849	29.04008	29.30168	6.542907E-06
60	28.79742	29.07152	29.38371	6.305913E-06
120	28.82611	29.10769	29.4331	6.186309E-06
180	28.85792	29.14313	29.47129	6.065928E-06
240	28.88972	29.17666	29.50423	5.935214E-06
300	28.92035	29.20827	29.53449	5.804155E-06
360	28.94944	29.23802	29.56265	5.622716E-06
420	28.9769	29.26599	29.58896	5.781859E-06
480	29.00284	29.29243	29.61408	5.699845E-06
540	29.02727	29.31725	29.6374	5.658691E-06
600	29.05031	29.34075	29.65963	5.57608E-06
660	29.07196	29.36274	29.68018	5.53462E-06
720	29.09235	29.38349	29.69975	5.493062E-06
780	29.11159	29.40306	29.71827	5.451396E-06
840	29.12973	29.42154	29.73574	5.40962E-06
900	29.14684	29.43896	29.7522	5.325748E-06
960	29.16284	29.45517	29.76714	5.325748E-06
1020	29.17796	29.47055	29.78186	5.283648E-06
1080	29.19221	29.48509	29.79556	5.241437E-06
1140	29.20562	29.49873	29.80836	5.19911E-06
1200	29.21822	29.51151	29.82031	5.156661E-06
1260	29.23002	29.52349	29.83146	5.156661E-06
1320	29.24119	29.53489	29.84239	5.114106E-06
1380	29.25171	29.54557	29.8524	5.071432E-06
1440	29.26155	29.55555	29.86164	5.071432E-06
1500	29.27085	29.56503	29.87072	5.028636E-06
1560	29.27958	29.57389	29.87895	5.028636E-06
1620	29.28784	29.58232	29.88702	4.985713E-06
1680	29.29558	29.59017	29.89427	4.985713E-06
1740	29.3029	29.59763	29.90141	4.942673E-06
1800	29.30973	29.60455	29.90776	4.942673E-06
1860	29.31619	29.61112	29.91403	4.942673E-06
1920	29.32232	29.6174	29.9201	4.899499E-06
1980	29.32805	29.6232	29.92538	4.899499E-06
2040	29.33345	29.6287	29.93062	4.899499E-06
2100	29.33859	29.63395	29.93569	4.856208E-06
2160	29.34336	29.63877	29.94002	4.856208E-06
2220	29.34785	29.64333	29.94435	4.856208E-06
2280	29.3521	29.64768	29.94854	4.856208E-06
2340	29.35617	29.65165	29.95258	4.812784E-06
2400	29.35992	29.65562	29.9559	4.812784E-06

the heat gain near the steady state stage= 5.498047 w/m^2

the mass condensation rate near the steady state stage= 4.68172E-06 kg/s per m^2

the indoor facing surface temperature near steady state= 29.42228 c.

the time needed to reach steady state stage= 11520 seconds

the maximum interval heat gain rate= 5.498047 w/m^2

the minimum interval heat gain rate= 8.082537E-02 w/m^2

the average heat gain rate = 4.991277 w/m^2

the average mass condensation rate during condensation= 4.735633E-06 kg/sec per m^2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 34 C. TI= 18 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 32.15934 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcg kg/s
0	25.81781	26.22773	26.63765	1.361261E-05
60	25.85727	26.29317	26.80834	1.311756E-05
120	25.91711	26.36849	26.91113	1.283636E-05
180	25.98342	26.44219	26.99026	1.264774E-05
240	26.04988	26.51223	27.05944	1.245817E-05
300	26.11399	26.57828	27.12259	1.229947E-05
360	26.17496	26.64055	27.18156	1.214009E-05
420	26.2326	26.69919	27.23662	1.197999E-05
480	26.28693	26.75436	27.28864	1.185141E-05
540	26.33814	26.80636	27.33758	1.172234E-05
600	26.38639	26.85535	27.38367	1.159282E-05
660	26.43183	26.90147	27.42698	1.147735E-05
720	26.47467	26.94498	27.46803	1.136496E-05
780	26.51499	26.98589	27.50635	1.126685E-05
840	26.55295	27.02442	27.54257	1.116845E-05
900	26.58871	27.06071	27.57667	1.110269E-05
960	26.62245	27.095	27.60911	1.100381E-05
1020	26.65424	27.12727	27.6394	1.090462E-05
1080	26.68412	27.15757	27.66773	1.083834E-05
1140	26.71226	27.18613	27.69461	1.077191E-05
1200	26.73879	27.21307	27.71972	1.070535E-05
1260	26.76378	27.23845	27.74387	1.063865E-05
1320	26.78732	27.26234	27.76631	1.057181E-05
1380	26.80947	27.28481	27.78738	1.053833E-05
1440	26.83039	27.30608	27.80754	1.047128E-05
1500	26.85009	27.32607	27.82627	1.043771E-05
1560	26.86868	27.34497	27.84415	1.037043E-05
1620	26.88616	27.36250	27.86068	1.033675E-05
1680	26.90263	27.37941	27.87645	1.030302E-05
1740	26.91817	27.39521	27.89139	1.023546E-05
1800	26.93275	27.40996	27.90506	1.020163E-05
1860	26.94645	27.42385	27.91808	1.016774E-05
1920	26.95935	27.43693	27.93037	1.013383E-05
1980	26.97148	27.44925	27.94194	1.013383E-05
2040	26.983	27.46099	27.95323	1.009988E-05
2100	26.99389	27.47207	27.96368	1.006599E-05
2160	27.00415	27.48248	27.97343	1.003187E-05
2220	27.01378	27.49224	27.98253	9.997812E-06
2280	27.0228	27.50138	27.99102	9.997812E-06
2340	27.03135	27.51008	27.99936	9.963703E-06
2400	27.03939	27.51824	28.00697	9.929566E-06

the heat gain near the steady state stages= 11.05878 w/m²
the mass condensation rate near the steady state stages= 9.585959E-06
kg/s per m²
the indoor facing surface temperature near steady state= 27.16981 c.
the time needed to reach steady state stages= 14340 seconds

the maximum interval heat gain rates= 11.05878 w/m²
the minimum interval heat gain rates= 9.613675 w/m²
the average heat gain rate = 10.23536 w/m²
the average mass condensation rate during condensation= 9.715464E-06
kg/sec.per m²

APPENDIX B-2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 30 C. TI= 18 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 28.2047 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcg kg/s
0	23.78474	24.08806	24.39138	1.216943E-05
60	23.82006	24.14663	24.54416	1.171711E-05
120	23.87362	24.21405	24.63617	1.14433E-05
180	23.93293	24.27995	24.7068	1.123217E-05
240	23.99225	24.34238	24.76816	1.10483E-05
300	24.04937	24.40117	24.82419	1.088002E-05
360	24.10358	24.45646	24.87633	1.0723E-05
420	24.15474	24.50845	24.92516	1.057551E-05
480	24.20292	24.55735	24.97102	1.043685E-05
540	24.24825	24.60333	25.01412	1.030609E-05
600	24.29088	24.64656	25.05463	1.018307E-05
660	24.33097	24.68722	25.09272	1.00672E-05
720	24.36867	24.72544	25.12853	9.956023E-06
780	24.40412	24.76139	25.1622	9.8553E-06
840	24.43745	24.79518	25.19386	9.758614E-06
900	24.46879	24.82696	25.22362	9.667434E-06
960	24.49825	24.85683	25.2516	9.581643E-06
1020	24.52595	24.88491	25.27791	9.500996E-06
1080	24.55199	24.91132	25.30263	9.425018E-06
1140	24.57648	24.93614	25.32588	9.353627E-06
1200	24.59949	24.95947	25.34773	9.286295E-06
1260	24.62113	24.98141	25.36827	9.223048E-06
1320	24.64147	25.00203	25.38758	9.163494E-06
1380	24.66059	25.02142	25.40573	9.107374E-06
1440	24.67856	25.03964	25.42279	9.054699E-06
1500	24.69546	25.05677	25.43883	9.005154E-06
1560	24.71134	25.07287	25.4539	8.958529E-06
1620	24.72627	25.088	25.46807	8.914572E-06
1680	24.7403	25.10222	25.48139	8.873407E-06
1740	24.75349	25.11559	25.49391	8.834592E-06
1800	24.76589	25.12816	25.50567	8.798103E-06
1860	24.77754	25.13998	25.51673	8.763784E-06
1920	24.7885	25.15108	25.52713	8.731535E-06
1980	24.7988	25.16152	25.5369	8.70122E-06
2040	24.80847	25.17133	25.54608	8.672673E-06
2100	24.81757	25.18055	25.55471	8.645925E-06
2160	24.82612	25.18922	25.56282	8.620666E-06
2220	24.83415	25.19736	25.57045	8.596928E-06
2280	24.84171	25.20502	25.57761	8.574637E-06
2340	24.8488	25.21222	25.58435	8.553635E-06
2400	24.85548	25.21898	25.59068	8.533999E-06

the heat gain near the steady state stage= 9.610453 w/m^2

the mass condensation rate near the steady state stage= 8.225573E-06 kg/s per m^2

the indoor facing surface temperature near steady state= 24.95965 C

the time needed to reach steady state stage= 10140 seconds

the maximum interval heat gain rate= 9.610453 w/m^2

the minimum interval heat gain rate= .1444553 w/m^2

the average heat gain rate = 3.517711 w/m^2

the average mass condensation rate during condensation= 8.356922E-06 kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 30 C. TI= 18 C. RH= 100 %
M= .01 m DT= 60 SEC. TSAT= 30 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcr kg/s
0	23.78474	24.08806	24.39138	1.891894E-05
60	23.83965	24.17911	24.62889	1.820898E-05
120	23.92291	24.2839	24.77185	1.777801E-05
180	24.01506	24.38628	24.88155	1.744554E-05
240	24.10721	24.48326	24.97679	1.715551E-05
300	24.19592	24.57454	25.06375	1.688957E-05
360	24.28008	24.66036	25.14462	1.664139E-05
420	24.35949	24.74104	25.22034	1.640818E-05
480	24.43423	24.81688	25.29142	1.618858E-05
540	24.50453	24.88817	25.35819	1.598168E-05
600	24.57062	24.95518	25.42093	1.578663E-05
660	24.63274	25.01816	25.47989	1.560287E-05
720	24.69114	25.07735	25.53529	1.542985E-05
780	24.74602	25.13299	25.58736	1.526677E-05
840	24.7976	25.18527	25.63628	1.511315E-05
900	24.84607	25.2344	25.68225	1.496855E-05
960	24.89162	25.28057	25.72545	1.483243E-05
1020	24.93442	25.32395	25.76604	1.470429E-05
1080	24.97464	25.36471	25.80417	1.458367E-05
1140	25.01242	25.40301	25.83999	1.447028E-05
1200	25.04793	25.43899	25.87365	1.436332E-05
1260	25.08129	25.4728	25.90527	1.426288E-05
1320	25.11263	25.50456	25.93497	1.416842E-05
1380	25.14207	25.5344	25.96288	1.407954E-05
1440	25.16974	25.56243	25.9891	1.399582E-05
1500	25.19572	25.58877	26.01372	1.391718E-05
1560	25.22014	25.61351	26.03686	1.384329E-05
1620	25.24307	25.63675	26.05859	1.377375E-05
1680	25.26462	25.65858	26.079	1.370837E-05
1740	25.28485	25.67909	26.09817	1.364696E-05
1800	25.30386	25.69835	26.11618	1.358921E-05
1860	25.32172	25.71645	26.1331	1.353498E-05
1920	25.33849	25.73344	26.14899	1.348381E-05
1980	25.35425	25.74941	26.16391	1.343583E-05
2040	25.36905	25.7644	26.17793	1.339076E-05
2100	25.38295	25.77848	26.19109	1.334832E-05
2160	25.396	25.79171	26.20346	1.33085E-05
2220	25.40826	25.80414	26.21507	1.327114E-05
2280	25.41978	25.81581	26.22598	1.323591E-05
2340	25.4306	25.82677	26.23622	1.320299E-05
2400	25.44076	25.83706	26.24584	1.317179E-05

the heat gain near the steady state stages= 14.82821 w/m^2

the mass condensation rate near the steady state stages= 1.249109E-05 kg/s per m^2

the indoor facing surface temperature near steady state= 25.59754 C

the time needed to reach steady state stages= 10500 seconds

the maximum interval heat gain rate= 14.82821 w/m^2

the minimum interval heat gain rate= .2245852 w/m^2

the average heat gain rate = 13.40087 w/m^2

the average mass condensation rate during condensation= 1.289794E-05 kg/sec. per m^2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

T0= 34 C. TI= 24 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 32.15934 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcr kg/s
0	28.77849	29.04008	29.30168	1.11481E-05
60	28.81074	29.09365	29.44144	1.062688E-05
120	28.85929	29.15471	29.52413	1.031693E-05
180	28.91265	29.21392	29.58693	1.008109E-05
240	28.96567	29.26963	29.64108	9.876863E-06
300	29.01639	29.32174	29.69021	9.691263E-06
360	29.06422	29.37043	29.73565	9.51932E-06
420	29.10908	29.41594	29.77794	9.358945E-06
480	29.15105	29.45847	29.81742	9.209145E-06
540	29.1903	29.49821	29.85429	9.068884E-06
600	29.22699	29.53536	29.88873	8.937613E-06
660	29.26127	29.57007	29.92092	8.814683E-06
720	29.29331	29.6025	29.95098	8.699856E-06
780	29.32325	29.6328	29.97908	8.592431E-06
840	29.35122	29.66112	30.00533	8.491779E-06
900	29.37736	29.68758	30.02986	8.397891E-06
960	29.40179	29.71231	30.05278	8.309834E-06
1020	29.42461	29.73541	30.07419	8.22765E-06
1080	29.44593	29.75698	30.09419	8.150697E-06
1140	29.46586	29.77715	30.11268	8.078729E-06
1200	29.48447	29.79599	30.13033	8.011417E-06
1260	29.50186	29.81359	30.14664	7.948684E-06
1320	29.5181	29.83003	30.16188	7.889981E-06
1380	29.53328	29.84539	30.17611	7.834919E-06
1440	29.54745	29.85973	30.18941	7.783656E-06
1500	29.5607	29.87314	30.20183	7.735655E-06
1560	29.57307	29.88566	30.21343	7.69076E-06
1620	29.58463	29.89736	30.22427	7.64886E-06
1680	29.59543	29.90829	30.23439	7.609658E-06
1740	29.60552	29.9185	30.24385	7.573065E-06
1800	29.61494	29.92804	30.25269	7.538849E-06
1860	29.62374	29.93695	30.26094	7.506871E-06
1920	29.63197	29.94526	30.26865	7.47701E-06
1980	29.63964	29.95304	30.27585	7.449142E-06
2040	29.64682	29.9603	30.28258	7.422923E-06
2100	29.65352	29.96708	30.28886	7.39878E-06
2160	29.65978	29.97342	30.29473	7.375856E-06
2220	29.66563	29.97934	30.30021	7.354655E-06
2280	29.67109	29.98486	30.30533	7.334698E-06
2340	29.6762	29.99003	30.31012	7.31615E-06
2400	29.68097	29.99485	30.31459	7.298812E-06

the heat gain near the steady state stages 8.28315 w/m^2

the mass condensation rate near the steady state stages 7.052922E-06
kg/s per m^2

the indoor facing surface temperature near steady states 29.74841

the time needed to reach steady state stages 8940 seconds

the maximum interval heat gain rates 8.28315 w/m^2

the minimum interval heat gain rates .1377387 w/m^2

the average heat gain rate = 7.420462 w/m^2

the average mass condensation rate during condensation 7.175244E-06
kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t1=indoor facing temperature and t3=outdoor facing temperature

TO= 34 C. TI= 18 C. RH= 90 %
M= .01 m DT= 60 SEC. TSAT= 32.15934 C.

time in SEC.	t1 cel.	t2 cel.	t3 cel.	mcg kg/s
0	25.81781	26.22773	26.63765	2.050385E-05
60	25.87725	26.32629	26.89474	1.964738E-05
120	25.96706	26.43922	27.04819	1.913167E-05
180	26.06617	26.54915	27.16533	1.873553E-05
240	26.16498	26.65296	27.2667	1.839127E-05
300	26.25983	26.75038	27.35897	1.807641E-05
360	26.34956	26.84172	27.44456	1.778332E-05
420	26.43399	26.92735	27.52443	1.750866E-05
480	26.51324	27.00763	27.59926	1.725067E-05
540	26.58753	27.08267	27.66936	1.700829E-05
600	26.65726	27.15341	27.73502	1.678035E-05
660	26.72256	27.21952	27.79655	1.656633E-05
720	26.78382	27.28148	27.85422	1.636512E-05
780	26.8412	27.33955	27.90825	1.617608E-05
840	26.89499	27.39397	27.95888	1.599869E-05
900	26.94539	27.44496	28.00633	1.583212E-05
960	26.99262	27.49275	28.05078	1.567569E-05
1020	27.03688	27.53753	28.09243	1.552879E-05
1080	27.07835	27.57948	28.13145	1.539108E-05
1140	27.1172	27.61870	28.16801	1.526174E-05
1200	27.1536	27.65562	28.20225	1.514034E-05
1260	27.18771	27.69012	28.23434	1.502655E-05
1320	27.21967	27.72244	28.26439	1.491973E-05
1380	27.2496	27.75273	28.29255	1.481968E-05
1440	27.27764	27.7811	28.31892	1.47257E-05
1500	27.30391	27.80767	28.34363	1.463756E-05
1560	27.32852	27.83256	28.36677	1.455499E-05
1620	27.35158	27.85588	28.38844	1.447754E-05
1680	27.37317	27.87773	28.40875	1.440489E-05
1740	27.39339	27.89818	28.42776	1.433687E-05
1800	27.41234	27.91735	28.44557	1.427303E-05
1860	27.43008	27.93529	28.46225	1.421328E-05
1920	27.44671	27.9521	28.47787	1.415721E-05
1980	27.46227	27.96785	28.49251	1.410469E-05
2040	27.47685	27.9826	28.50621	1.405549E-05
2100	27.49051	27.99641	28.51905	1.400938E-05
2160	27.5033	28.00935	28.53107	1.396609E-05
2220	27.51528	28.02146	28.54233	1.392567E-05
2280	27.5265	28.03281	28.55287	1.388764E-05
2340	27.53701	28.04344	28.56275	1.385214E-05
2400	27.54685	28.05339	28.572	1.381887E-05

the heat gain near the steady state stages= 15.32941 w/m²
the mass condensation rate near the steady state stages= 1.332641E-05
kg/s per m²
the indoor facing surface temperature near steady state= 27.69188
C.

the maximum interval heat gain rate= 15.32941 w/m²
the minimum interval heat gain rate= .2430887 w/m²
the average heat gain rate = 13.873 w/m²
the average mass condensation rate during condensation= 1.355346E-05
kg/sec per m²

APPENDIX B-3

***** the nodal temperatures at the needed time*****

note: t3=indoor facing temperature and t1=outdoor facing temperature

TO=-10 C. TI= 24 C. RH= 30 % WV= 4 m/s
H= .01 m DT= 60 SEC. TSAT= 5.396284 C.

time in sec.	t1 C.	t2 C.	t3 C.	mcr kg/s
0	-1.72263	-.4534321	.8157654	4.550223E-06
60	-1.710495	-.4320864	.872183	4.5016E-06
120	-1.692345	-.4075102	.9070994	4.471423E-06
180	-1.672593	-.3837585	.9338521	4.44825E-06
240	-1.653224	-.3616519	.956703	4.428436E-06
300	-1.634957	-.3412685	.9771249	4.410718E-06
360	-1.618022	-.32252	.9956961	4.394538E-06
420	-1.60239	-.3052869	1.012694	4.379735E-06
480	-1.588004	-.2894499	1.028291	4.366167E-06
540	-1.574777	-.2748966	1.042616	4.353672E-06
600	-1.562619	-.2615233	1.055775	4.342159E-06
660	-1.551447	-.2492344	1.067866	4.331608E-06
720	-1.54118	-.2379423	1.078977	4.321893E-06
780	-1.531747	-.227566	1.089186	4.312968E-06
840	-1.523077	-.2180313	1.098567	4.304725E-06
900	-1.515112	-.20927	1.107186	4.29718E-06
960	-1.507792	-.2012195	1.115107	4.290227E-06
1020	-1.501066	-.193822	1.122384	4.283862E-06
1080	-1.494886	-.1870245	1.129072	4.278013E-06
1140	-1.489207	-.1807785	1.135217	4.272596E-06
1200	-1.483989	-.1750392	1.140864	4.26764E-06
1260	-1.479194	-.1697655	1.146052	4.263078E-06
1320	-1.474788	-.1649197	1.150819	4.258899E-06
1380	-1.470739	-.160467	1.1552	4.255021E-06
1440	-1.467019	-.1563757	1.159225	4.251523E-06
1500	-1.463601	-.1526162	1.162924	4.248253E-06
1560	-1.46046	-.1491618	1.166322	4.245243E-06
1620	-1.457574	-.1459377	1.169445	4.242515E-06
1680	-1.454923	-.1430711	1.172314	4.239985E-06
1740	-1.452486	-.1403912	1.17495	4.237665E-06
1800	-1.450247	-.1379287	1.177373	4.235562E-06
1860	-1.44819	-.1356659	1.179599	4.233593E-06
1920	-1.446299	-.1335867	1.181645	4.231781E-06
1980	-1.444562	-.1316764	1.183524	4.230102E-06
2040	-1.442966	-.1299212	1.185251	4.228597E-06
2100	-1.4415	-.1283083	1.186837	4.227193E-06
2160	-1.440153	-.1268264	1.188295	4.22591E-06
2220	-1.438915	-.1254647	1.189635	4.224734E-06
2280	-1.437777	-.1242136	1.190866	4.223658E-06
2340	-1.436732	-.1230638	1.191997	4.222629E-06
2400	-1.435771	-.1220075	1.193036	4.221754E-06

the total heat loss near the steady state stages 10.44416 w/m^2
the convective heat loss rate near the steady state stages 7.121775 w/m^2
the time needed to reach steady state stages 9240 seconds

the maximum interval convective heat loss rates 7.121775
w/m^2
the minimum interval convective heat loss rates .1451321
w/m^2
the average convective heat loss rates 6.536383 w/m^2
the mass condensation rate near the the steady state stages 4.211356E-06
kg/sec per m^2
the average mass condensation rate during condensation 4.218703E-06
kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t3=indoor facing temperature and t1=outdoor facing temperature

TO=-10 C. TI= 24 C. RH= 80 % WV= 4 m/s
M=.01 m DT= 60 SEC. TSAT= 20.3298 C.

time in sec.	t1 C.	t2 C.	t3 C.	mcg kg/s
0	-1.72263	-.4534321	.8157654	3.199822E-05
60	-1.637298	-.303325	1.212508	3.165258E-05
120	-1.509677	-.1305182	1.458002	3.143424E-05
180	-1.370806	3.646082E-02	1.646029	3.126474E-05
240	-1.234658	.191835	1.80657	3.111839E-05
300	-1.106357	.335056	1.949987	3.098537E-05
360	-.9873138	.4667444	2.080348	3.086535E-05
420	-.8775358	.5877423	2.199514	3.075372E-05
480	-.7765472	.6988915	2.308991	3.065059E-05
540	-.6837345	.8009827	2.409367	3.055534E-05
600	-.5984699	.8947489	2.501569	3.046736E-05
660	-.5201532	.9806652	2.586217	3.038607E-05
720	-.4482251	1.059952	2.663947	3.031108E-05
780	-.382168	1.132582	2.735325	3.024186E-05
840	-.3215054	1.199278	2.800867	3.017806E-05
900	-.2657983	1.260525	2.861051	3.011921E-05
960	-.2146438	1.316766	2.916313	3.006501E-05
1020	-.1676706	1.368405	2.967054	3.001509E-05
1080	-.1245383	1.415828	3.013644	2.996906E-05
1140	-3.493343E-02	1.459368	3.056422	2.992671E-05
1200	-4.856845E-02	1.499347	3.095697	2.988773E-05
1260	-1.517902E-02	1.536053	3.131758	2.985184E-05
1320	1.547804E-02	1.569756	3.164866	2.981883E-05
1380	4.362579E-02	1.6007	3.195263	2.978846E-05
1440	6.946927E-02	1.62911	3.223171	2.97605E-05
1500	9.319655E-02	1.655194	3.248792	2.973484E-05
1560	.114981	1.679141	3.272315	2.971121E-05
1620	.1349813	1.701128	3.293911	2.968948E-05
1680	.1533432	1.721313	3.313737	2.966949E-05
1740	.1702011	1.739844	3.331939	2.965115E-05
1800	.1856779	1.756857	3.348649	2.963428E-05
1860	.1998866	1.772477	3.36399	2.961879E-05
1920	.212931	1.786816	3.378074	2.960454E-05
1980	.2249065	1.79998	3.391003	2.959148E-05
2040	.2359006	1.812066	3.402872	2.957945E-05
2100	.2459937	1.82316	3.413769	2.956841E-05
2160	.2552596	1.833346	3.423773	2.955825E-05
2220	.2637561	1.842697	3.432956	2.954891E-05
2280	.2715751	1.851281	3.441386	2.95404E-05
2340	.2787442	1.859161	3.449126	2.95325E-05
2400	.2853256	1.866396	3.45623	2.952531E-05

the total heat loss near the steady state stage= 73.02165 w/m²
the convective heat loss rate near the steady state stage= 49.79265 w/m²
the time needed to reach steady state stage= 9900 seconds

the maximum interval convective heat loss rate= 49.79265 w/m²

the minimum interval convective heat loss rate= 1.02057 w/m²

the average convective heat loss rate= 46.00228 w/m²

the mass condensation rate near the steady state stage= 2.944422E-05 kg/sec per m²

the average mass condensation rate during condensations= 2.950107E-05 kg/sec per m²

***** the nodal temperatures at the needed time*****

note: t3=indoor facing temperature and t1=outdoor facing temperature

TO=-12 C. TI= 21 C. RH= 30 % WV= 4 m/s
M=.01 m DT= 60 SEC. TSAT= 2.842636 C.

time in sec.	t1 C.	t2 C.	t3 C.	mcg kg/s
0	-4.083612	-2.869754	-1.65592	3.770539E-06
60	-4.073547	-2.852061	-1.609126	3.736205E-06
120	-4.058472	-2.83164	-1.580066	3.714817E-06
180	-4.042043	-2.811871	-1.557748	3.698359E-06
240	-4.025911	-2.793445	-1.538654	3.684276E-06
300	-4.010683	-2.77643	-1.521564	3.67165E-06
360	-3.996531	-2.76076	-1.506003	3.660118E-06
420	-3.983458	-2.746336	-1.49174	3.649548E-06
480	-3.971411	-2.733064	-1.478636	3.639838E-06
540	-3.96032	-2.720851	-1.466585	3.630901E-06
600	-3.950113	-2.709613	-1.455499	3.622668E-06
660	-3.940719	-2.699273	-1.445299	3.615103E-06
720	-3.932076	-2.689758	-1.435914	3.608121E-06
780	-3.924122	-2.681004	-1.427279	3.601683E-06
840	-3.916805	-2.672949	-1.419334	3.595763E-06
900	-3.910071	-2.665537	-1.412023	3.590336E-06
960	-3.903876	-2.658718	-1.405297	3.585307E-06
1020	-3.898176	-2.652443	-1.399108	3.580705E-06
1080	-3.89293	-2.646669	-1.393413	3.576457E-06
1140	-3.888104	-2.641357	-1.388174	3.572549E-06
1200	-3.883663	-2.636469	-1.383352	3.568965E-06
1260	-3.879577	-2.631971	-1.378916	3.565642E-06
1320	-3.875818	-2.627833	-1.374835	3.5626E-06
1380	-3.872359	-2.624026	-1.37108	3.559808E-06
1440	-3.869177	-2.620523	-1.367624	3.557201E-06
1500	-3.866249	-2.617299	-1.364445	3.554835E-06
1560	-3.863554	-2.614333	-1.36152	3.552643E-06
1620	-3.861075	-2.611605	-1.358828	3.550636E-06
1680	-3.858793	-2.609093	-1.356352	3.548799E-06
1740	-3.856695	-2.606783	-1.354073	3.547086E-06
1800	-3.854763	-2.604658	-1.351977	3.545531E-06
1860	-3.852987	-2.602702	-1.350047	3.544074E-06
1920	-3.851352	-2.600902	-1.348273	3.542766E-06
1980	-3.849847	-2.599246	-1.346639	3.54154E-06
2040	-3.848463	-2.597723	-1.345137	3.540396E-06
2100	-3.84719	-2.596321	-1.343754	3.539356E-06
2160	-3.846019	-2.595032	-1.342483	3.538424E-06
2220	-3.844941	-2.593845	-1.341312	3.537548E-06
2280	-3.843949	-2.592753	-1.340236	3.536744E-06
2340	-3.843036	-2.591749	-1.339245	3.535997E-06
2400	-3.842197	-2.590825	-1.338333	3.535326E-06

the total heat loss near the steady state stage= 8.7481 w/m^2
the convective heat loss rate near the steady state stage= 6.005914 w/m^2
the time needed to reach steady state stage= 8580 seconds

the maximum interval convective heat loss rate= 6.005914 w/m^2
the minimum interval convective heat loss rate= .1203783 w/m^2
the average convective heat loss rate= 5.466202 w/m^2
the mass condensation rate near the the steady state stage= 3.52746E-06 kg/sec per m^2
the average mass condensation rate during condensation= 3.532834E-06 kg/sec per m^2

***** the nodal temperatures at the needed time*****

note: t3=indoor facing temperature and t1=outdoor facing temperature

TD=-12 C. TI= 21 C. RH= 30 % WV= 8 m/s
M= .01 m DT= 60 SEC. TSAT= 2.842636 C.

time in sec.	t1 C.	t2 C.	t3 C.	mcr kg/s
0	-6.795775	-5.450678	-4.105587	5.4289E-06
60	-6.782311	-5.425583	-4.038451	5.386955E-06
120	-6.762694	-5.397169	-3.99722	5.361099E-06
180	-6.741928	-5.37034	-3.966248	5.341626E-06
240	-6.722142	-5.346035	-3.94048	5.325415E-06
300	-6.704033	-5.32426	-3.918105	5.311274E-06
360	-6.687718	-5.304815	-3.898355	5.298797E-06
420	-6.673111	-5.287468	-3.88061	5.28771E-06
480	-6.660068	-5.271999	-3.865188	5.277818E-06
540	-6.648429	-5.258205	-3.851267	5.268995E-06
600	-6.638049	-5.245904	-3.838857	5.261127E-06
660	-6.628794	-5.234936	-3.827792	5.254104E-06
720	-6.62054	-5.225157	-3.817926	5.247834E-06
780	-6.61318	-5.216437	-3.809129	5.242254E-06
840	-6.606618	-5.208662	-3.801286	5.237271E-06
900	-6.600767	-5.201729	-3.794293	5.232793E-06
960	-6.595551	-5.195548	-3.788057	5.228833E-06
1020	-6.590899	-5.190036	-3.782497	5.225308E-06
1080	-6.586752	-5.185122	-3.77754	5.222139E-06
1140	-6.583053	-5.18074	-3.773119	5.219332E-06
1200	-6.579755	-5.176833	-3.769178	5.216802E-06
1260	-6.576815	-5.173349	-3.765664	5.214555E-06
1320	-6.574194	-5.170244	-3.762531	5.212575E-06
1380	-6.571856	-5.167474	-3.759737	5.210789E-06
1440	-6.569772	-5.165005	-3.757246	5.209181E-06
1500	-6.567914	-5.162804	-3.755026	5.207782E-06
1560	-6.566258	-5.160841	-3.753046	5.206524E-06
1620	-6.56473	-5.159091	-3.75128	5.205387E-06
1680	-6.563463	-5.15753	-3.749706	5.204379E-06
1740	-6.562289	-5.156138	-3.748302	5.203486E-06
1800	-6.561242	-5.154898	-3.747051	5.202681E-06
1860	-6.560308	-5.153791	-3.745935	5.201974E-06
1920	-6.559475	-5.152805	-3.74494	5.201332E-06
1980	-6.558734	-5.151926	-3.744053	5.200763E-06
2040	-6.558072	-5.151142	-3.743262	5.200267E-06
2100	-6.557482	-5.150443	-3.742557	5.199813E-06
2160	-6.556955	-5.149819	-3.741928	5.199422E-06
2220	-6.556486	-5.149263	-3.741367	5.199041E-06
2280	-6.556068	-5.148767	-3.740867	5.19874E-06
2340	-6.555694	-5.148325	-3.740421	5.198447E-06
2400	-6.555361	-5.147931	-3.740023	5.198187E-06

the total heat loss near the steady state stage= 12.88634 w/m^2
the convective heat loss rate near the steady state stage= 9.803663 w/m^2
the time needed to reach steady state stage= 6300 seconds

the maximum interval convective heat loss rate= 9.803663
w/m^2
the minimum interval convective heat loss rate= .2714337
w/m^2
the average convective heat loss rate= 8.913393 w/m^2
the mass condensation rate near the the steady state stage= 5.196105E-06
kg/sec per m^2
the average mass condensation rate during condensation= 5.199748E-06
kg/sec.per m^2

APPENDIX B-4

***** the nodal temperatures at the needed time*****

note: t7=indoor facing temperature and t1=outdoor facing temperature

TO= -32 C.	TI= 18 C.	RH= 40 %	WV= 4 m/s	
M= .004 m	AS= .008 m	DT= 60 SEC.	TSAT= 4.249453 C.	
time in sec.	t1 C.	t4 C.	t7 C.	mc kg/s
0	-25.51772	-13.23692	-9.9561234	5.095201E-06
60	-25.5156	-13.20198	-9.8539067	5.018054E-06
120	-25.51149	-13.16583	-9.7949589	4.954974E-06
180	-25.50538	-13.13185	-9.7341436	4.908513E-06
240	-25.49921	-13.10028	-9.6793394	4.861954E-06
300	-25.49186	-13.07102	-9.6247873	4.814668E-06
360	-25.4841	-13.04395	-9.5849291	4.781255E-06
420	-25.47615	-13.01894	-9.5442827	4.746383E-06
480	-25.46819	-12.99585	-9.5074213	4.71462E-06
540	-25.46033	-12.97457	-9.4739691	4.685756E-06
600	-25.45267	-12.95496	-9.4435859	4.659468E-06
660	-25.44527	-12.93691	-9.4159761	4.635554E-06
720	-25.43819	-12.92032	-9.3908706	4.613729E-06
780	-25.43145	-12.90506	-9.3680309	4.593895E-06
840	-25.42507	-12.89104	-9.3472419	4.575721E-06
900	-25.41906	-12.87817	-9.3283123	4.55923E-06
960	-25.41342	-12.86636	-9.310639	4.544168E-06
1020	-25.40814	-12.85553	-9.2953552	4.530412E-06
1080	-25.40322	-12.84559	-9.2810315	4.517879E-06
1140	-25.39863	-12.83648	-9.2679707	4.506452E-06
1200	-25.39437	-12.82813	-9.2560581	4.495996E-06
1260	-25.39042	-12.82047	-9.2451908	4.486428E-06
1320	-25.38677	-12.81347	-9.2352753	4.47773E-06
1380	-25.38339	-12.80705	-9.2262255	4.469783E-06
1440	-25.38027	-12.80117	-9.2179648	4.462509E-06
1500	-25.37739	-12.79578	-9.2104232	4.455857E-06
1560	-25.37474	-12.79086	-9.2035374	4.449824E-06
1620	-25.3723	-12.78635	-9.197249	4.444274E-06
1680	-25.37005	-12.78222	-9.191505	4.439215E-06
1740	-25.36799	-12.77844	-9.1862603	4.434594E-06
1800	-25.36609	-12.77493	-9.1814685	4.430365E-06
1860	-25.36435	-12.77182	-9.1770908	4.426502E-06
1920	-25.36275	-12.76893	-9.1730913	4.422986E-06
1980	-25.36128	-12.76628	-9.169437	4.41976E-06
2040	-25.35993	-12.76386	-9.1660977	4.416808E-06
2100	-25.35869	-12.76164	-9.1630463	4.414133E-06
2160	-25.35756	-12.75962	-9.1602574	4.411621E-06
2220	-25.35653	-12.75777	-9.1577093	4.409335E-06
2280	-25.35558	-12.75607	-9.1553811	4.407326E-06
2340	-25.3547	-12.75452	-9.1532527	4.40546E-06
2400	-25.35391	-12.7531	-9.1513071	4.403722E-06

the total heat loss rate by condensation near the steady state stage= 10.87536 w/m^2

the time needed to reach steady state stage= 8700 seconds

the maximum interval convective heat loss rate= 4.123112 w/m^2

the minimum interval convective heat loss rate= 2.536632E-02 w/m^2

the average convective heat loss rate= 3.56783 w/m^2

the average mass condensation rate during condensation= 4.462228E-06 kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t7=indoor facing temperature and t1=outdoor facing temperature

TO=-32 C.	TI= 18 C.	RH= 80 %	WV= 4 m/s	
M=.004 m	AS=.008 m	DT= 60 SEC.	TSAT= 14.47533 C.	
time in sec.	t1 C.	t4 C.	t7 C.	mcg kg/s
0	-25.51772	-13.23692	-.9561234	2.153681E-05
60	-25.50876	-13.08923	-.5663336	2.120384E-05
120	-25.49138	-12.93647	-.2750252	2.094779E-05
180	-25.46768	-12.79298	-1.825282E-02	2.071753E-05
240	-25.43954	-12.65974	.2129366	2.050658E-05
300	-25.40549	-12.53637	.4217469	2.031313E-05
360	-25.37575	-12.42235	.6105461	2.013583E-05
420	-25.34224	-12.31711	.7813857	1.997335E-05
480	-25.30866	-12.2201	.9360818	1.98246E-05
540	-25.27556	-12.13079	1.076249	1.968846E-05
600	-25.24333	-12.04863	1.203327	1.956391E-05
660	-25.21224	-11.97314	1.318599	1.945002E-05
720	-25.1825	-11.90381	1.423213	1.934584E-05
780	-25.15423	-11.84019	1.518198	1.925068E-05
840	-25.12751	-11.78185	1.604475	1.91637E-05
900	-25.10236	-11.72838	1.682872	1.908418E-05
960	-25.07879	-11.6794	1.754132	1.901161E-05
1020	-25.05675	-11.63454	1.818925	1.894527E-05
1080	-25.03621	-11.59349	1.877854	1.888473E-05
1140	-25.01713	-11.55592	1.931465	1.882945E-05
1200	-24.99943	-11.52157	1.980248	1.877893E-05
1260	-24.98305	-11.49015	2.024646	1.873284E-05
1320	-24.9679	-11.46143	2.065062	1.86908E-05
1380	-24.95392	-11.43518	2.10185	1.865236E-05
1440	-24.94104	-11.41119	2.135368	1.861734E-05
1500	-24.92917	-11.38923	2.165886	1.858535E-05
1560	-24.91825	-11.36927	2.193682	1.855616E-05
1620	-24.90823	-11.35099	2.219034	1.852954E-05
1680	-24.89903	-11.3343	2.242074	1.850522E-05
1740	-24.89058	-11.31905	2.263093	1.848304E-05
1800	-24.88283	-11.30514	2.282246	1.846278E-05
1860	-24.87573	-11.29244	2.2997	1.844433E-05
1920	-24.86922	-11.28085	2.315607	1.84275E-05
1980	-24.86326	-11.27027	2.330104	1.841214E-05
2040	-24.85781	-11.26061	2.343319	1.839812E-05
2100	-24.85282	-11.2518	2.355363	1.838531E-05
2160	-24.84825	-11.24376	2.366343	1.837369E-05
2220	-24.84406	-11.23642	2.376352	1.836305E-05
2280	-24.84026	-11.22973	2.385477	1.835334E-05
2340	-24.83673	-11.22362	2.393795	1.834448E-05
2400	-24.83359	-11.21805	2.401379	1.833643E-05

the total heat loss rate by condensation near the steady state stage= 45.26657 w/m^2

the time needed to reach steady state stage= 9180 seconds

the maximum interval convective heat loss rate= 17.16148 w/m^2

the minimum interval convective heat loss rate= .1071703 w/m^2

the average convective heat loss rate= 15.39198 w/m^2

the average mass condensation rate during condensation= 1.833191E-05 kg/sec.per m^2

***** the nodal temperatures at the needed time*****

note: t7=indoor facing temperature and t1=outdoor facing temperature

TO=-32 C.	T1= 21 C.	RH= 30 %	WV= 4 m/s	
M= .004 m	AS= .008 m	DT= 60 SEC.	TSAT= 2.842636 C.	
time in sec.	t1 C.	t4 C.	t7 C.	mcr kg/s
0	-25.07275	-11.94893	1.174879	1.61146E-06
60	-25.07207	-11.9379	1.204006	1.583635E-06
120	-25.07078	-11.92651	1.225594	1.562995E-06
180	-25.06901	-11.91584	1.244767	1.544785E-06
240	-25.06692	-11.90595	1.261907	1.528396E-06
300	-25.06461	-11.89581	1.277362	1.513572E-06
360	-25.06213	-11.88937	1.291315	1.500217E-06
420	-25.05963	-11.8806	1.303924	1.488311E-06
480	-25.0572	-11.87343	1.31533	1.477153E-06
540	-25.05475	-11.86684	1.325654	1.467243E-06
600	-25.05236	-11.86079	1.335007	1.458232E-06
660	-25.05006	-11.85523	1.343486	1.450109E-06
720	-25.04786	-11.85012	1.351177	1.442672E-06
780	-25.04578	-11.84544	1.358156	1.435944E-06
840	-25.04381	-11.84115	1.364493	1.42983E-06
900	-25.04195	-11.83722	1.37025	1.424285E-06
960	-25.04021	-11.83362	1.375481	1.41923E-06
1020	-25.03859	-11.83032	1.380238	1.41463E-06
1080	-25.03708	-11.82731	1.384563	1.410476E-06
1140	-25.03567	-11.82455	1.388499	1.406697E-06
1200	-25.03437	-11.82203	1.392081	1.403195E-06
1260	-25.03316	-11.81972	1.39534	1.400051E-06
1320	-25.03205	-11.81761	1.398308	1.397202E-06
1380	-25.03102	-11.81568	1.401011	1.394612E-06
1440	-25.03007	-11.81392	1.403473	1.392185E-06
1500	-25.0292	-11.81231	1.405715	1.390053E-06
1560	-25.0284	-11.81083	1.407758	1.388064E-06
1620	-25.02766	-11.80949	1.40962	1.386253E-06
1680	-25.02698	-11.80826	1.411316	1.384638E-06
1740	-25.02636	-11.80714	1.412863	1.383133E-06
1800	-25.02579	-11.80612	1.414273	1.381757E-06
1860	-25.02527	-11.80518	1.415558	1.380524E-06
1920	-25.02479	-11.80433	1.416729	1.379385E-06
1980	-25.02435	-11.80355	1.417798	1.378356E-06
2040	-25.02395	-11.80284	1.418772	1.377416E-06
2100	-25.02358	-11.80219	1.41966	1.376556E-06
2160	-25.02325	-11.8016	1.42047	1.375747E-06
2220	-25.02294	-11.80105	1.421209	1.375038E-06
2280	-25.02266	-11.80056	1.421883	1.374396E-06
2340	-25.0224	-11.80011	1.422497	1.373803E-06
2400	-25.02217	-11.7997	1.423058	1.373237E-06

the total heat loss rate by condensation near the steady state stages = 3.39165, w/m²

the time needed to reach steady state stages = 7380 seconds

the maximum interval convective heat loss rate= 1.26d65 w/m²

the minimum interval convective heat loss rate= 8.029735E-03 w/m²

the average convective heat loss rate= 1.105369 w/m²

the average mass condensation rate during condensations= 1.373163E-06 kg/sec.per m²

***** the nodal temperatures at the needed time*****

note: t7=indoor facing temperature and t1=outdoor facing temperature

TO= -32 C.	TI= 21 C.	RH= 30 %	WV= 8 m/s	
M= .004 m	AG= .008 m	DT= 60 SEC.	TSAT= 2.842636 C.	

time in sec.	t1 C.	t4 C.	t7 C.	mcg kg/s

0	-27.65963	-13.79895	6.174088E-02	2.656673E-06
60	-27.65864	-13.78079	.1097584	2.613663E-06
120	-27.65677	-13.76212	.1455531	2.581496E-06
180	-27.65432	-13.74471	.1770465	2.553119E-06
240	-27.6515	-13.72867	.2053511	2.527547E-06
300	-27.6485	-13.71395	.2308664	2.504476E-06
360	-27.64544	-13.70047	.2539389	2.483584E-06
420	-27.6424	-13.68816	.274673	2.464733E-06
480	-27.63945	-13.67642	.2934463	2.447637E-06
540	-27.63662	-13.66667	.3104093	2.432212E-06
600	-27.63394	-13.65734	.3257425	2.418275E-06
660	-27.63141	-13.64886	.3396063	2.405524E-06
720	-27.62906	-13.64115	.3521448	2.394192E-06
780	-27.62689	-13.63414	.3634366	2.383913E-06
840	-27.62487	-13.62778	.3737472	2.374446E-06
900	-27.62301	-13.622	.3830316	2.365953E-06
960	-27.62131	-13.61676	.3914338	2.358258E-06
1020	-27.61974	-13.61201	.3990381	2.351315E-06
1080	-27.61831	-13.6077	.4059214	2.344995E-06
1140	-27.61701	-13.60379	.4121517	2.339311E-06
1200	-27.61582	-13.60025	.4177921	2.334119E-06
1260	-27.61474	-13.59704	.422898	2.32941E-06
1320	-27.61375	-13.59413	.4275201	2.325206E-06
1380	-27.61285	-13.59149	.4317057	2.321361E-06
1440	-27.61203	-13.5891	.4354953	2.317893E-06
1500	-27.6113	-13.58694	.4389236	2.314712E-06
1560	-27.61062	-13.58498	.4420329	2.311883E-06
1620	-27.61001	-13.5832	.4448459	2.309296E-06
1680	-27.60946	-13.58159	.447393	2.306975E-06
1740	-27.60896	-13.58014	.4496997	2.304843E-06
1800	-27.60851	-13.57982	.4517881	2.302931E-06
1860	-27.60809	-13.57762	.4536793	2.301186E-06
1920	-27.60772	-13.57654	.4553915	2.299587E-06
1980	-27.60738	-13.57556	.4569416	2.298185E-06
2040	-27.60707	-13.57467	.4583456	2.296902E-06
2100	-27.60679	-13.57387	.4596173	2.295739E-06
2160	-27.60654	-13.57314	.4607687	2.294698E-06
2220	-27.60632	-13.57248	.4618117	2.29369E-06
2280	-27.60611	-13.57188	.4627553	2.29283E-06
2340	-27.60592	-13.57134	.46361	2.292032E-06
2400	-27.60575	-13.57085	.4643835	2.291332E-06

the total heat loss rate by condensation near the steady state stage= 5.665578 w/m^2

the time needed to reach steady state stage= 7620 seconds

the maximum interval convective heat loss rate= 2.237915 w/m^2

the minimum interval convective heat loss rate= 2.014893E-02 w/m^2

the average convective heat loss rate= 1.993917 w/m^2

the average mass condensation rate during condensation= 2.292635E-06 kg/sec.per m^2

APPENDIX C

EXPERIMENTAL DATA FOR MASS CONDENSATION RATE AND
CORRESPONDING SURFACE TEMPERATURE RISE AT DIFFERENT CONDITIONS

Test No.	Temperature °C			Rsc/R	Relative humidity RH %	Time min.	Mass Collected gm	Mass Condensation Rate, m Kg/s _{x10⁻⁶}				Surface Temperature Rise °C						
	Glass	Hot Air	Cold air					1st Approach		Measured	2nd Approach		Measured	1st Approach		Measured	2nd Approach	
								Calcul- ated	Diff %		Calcul- ated	Diff %		Calcul- ated	Diff %		Calcul- ated	Diff %
1	35.6	49.3	19.5	.41	91	26.7	61.5	43	52.4	21.8	39.6	- 7.9	3.9	5.4	38.5	4.9	25.6	
2	27.2	36.6	16.2	.449	91	33	42.3	21.4	25.7	20.1	16.5	-22.9	1.84	2.8	52	2.2	19.6	
3	28	38	16.5	.44	91	27.5	39.9	24.2	28.3	16.9	18.6	-23.1	2.04	3.1	52	2.46	20.6	
4	29.2	40	17	.43	90	11.5	18.5	26.8	31.2	16.4	21.1	-21.3	2.3	3.4	48	2.8	21.7	
5	32.3	43.1	20.1	.42	90	21	25.3	28	33.9	21.1	23.6	-15.7	2.5	3.6	44	3.02	20.8	
6	31.4	42.4	19.2	.415	90	13	22.3	28.6	33.6	17.5	23.2	-18.9	2.6	3.6	38	3.0	15.4	
7	33.8	45.9	19.5	.442	93	11.5	23.7	34	43.7	28.5	31.9	- 6.2	2.8	4.6	64	4.0	42.8	
8	35.2	47	21.2	.412	94	5.7	14.3	42	46.4	10.5	34.4	-18.1	3.4	4.8	41	4.3	26.5	
9	36.8	48.2	20.8	.421	98	9.1	28.2	51.6	55	6.6	41.8	-19	4.5	5.7	26.7	5.2	15.6	
10	31	41.2	19.6	.422	90	19	28	24.6	30.2	22.8	20.5	-16.7	2.3	3.2	39.1	2.65	15.2	
11	20.3	31.9	7.1	.415	95	57	8.7	25.4	29.6	16.5	18.6	-26.8	2.9	3.4	17.2	2.6	-10.3	
12	30.9	41.3	19	.415	94	14.25	25.5	30	35	16.7	24.3	-19	2.65	3.7	39.6	3.1	17.0	
13	29.9	40.4	17.2	.41	100	23	50.2	36.4	40.6	11.5	28.7	-21.2	3.2	4.4	37.5	3.7	15.6	

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Test No.	Temperature °C			Rsc/R	Relative humidity RH %	Time min.	Mass Collected gm	Mass Condensation Rate, m Kg/s _{x10-6}						Surface Temperature Rise °C					
	Glass	Hot Air	Cold air					Measured	1st Approach		2nd Approach		Measured	1st Approach		2nd Approach			
									Calcul- ated	Diff %	Calcul- ated	Diff %		Calcul- ated	Diff %	Calcul- ated	Diff %		
27	24	34.2	12.8	.405	94	10.8	14.8	22.8	27.3	19.7	17.3	-24.1	2.5	3.1	24	2.4	- 4		
28	24.2	34.2	13	.405	95	15.8	23.9	25.2	27.7	9.9	17.7	-29.8	2.6	3.1	19.2	2.4	- 7.7		
29	23.1	34.1	12.3	.405	90	13.28	14.1	17.7	24.8	40.1	15.5	-12.4	1.97	2.8	42	2.1	6.6		
30	26.5	39.5	11.9	.412	88	42.8	81.6	31.8	35.1	10.4	23.8	-25.1	3.2	3.9	21.9	3.2	0		
31	28.2	42.1	12.8	.412	86	37.1	75	33.7	38	12.8	26.5	-21.4	3.3	4.1	24.2	3.5	6.1		
32	25.4	37.3	11.6	.412	92	31.7	60.7	31.9	33.9	6.3	22.6	-28.1	3.2	3.8	18.8	3.1	- 3.1		
33	24.2	36.2	11	.412	91	12.8	22.6	29.2	31.4	7.5	20.6	-29.4	2.82	3.5	24.1	2.8	- 0.7		
34	26.5	39.2	12.6	.412	90	16.3	31	31.7	35.2	11.0	24	-24.3	2.94	3.9	32.6	3.2	8.8		
35	29.9	43.4	13.7	.412	92	19.25	49.4	42.8	45.9	7.2	33.2	-22.4	3.96	5	26.3	4.4	11.1		
36	30.4	46	14.4	.412	78	21.8	37.6	28.7	36.4	26.8	25.6	-10.8	3.0	3.9	30	3.3	10		
37	31.5	46.5	15.3	.412	83	24.17	48.1	31.9	42	31.7	30.3	- 5	3.34	4.5	34.7	3.9	16.8		
38	32.3	46.8	15.5	.412	89	12.6	30.6	40.5	49.2	21.5	3684	-10.1	3.9	5.2	33.3	4.7	20.5		
39	31.7	46.7	17.7	.418	77	10.13	12.9	21.2	32.1	51.4	22.3	- 5.2	1.9	3.4	79	2.84	49.5		

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Test No.	Temperature °C			Rsc/R	Relative humidity RH %	Time min.	Mass Collected gm	Mass Condensation Rate, m Kg/s x10 ⁻⁶						Surface Temperature Rise °C					
	Glass	Hot Air	Cold air					Measured	1st Approach		2nd Approach		Measured	1st Approach		2nd Approach			
									Calcul- ated	Diff %	Calcul- ated	Diff %		Calcul- ated	Diff %	Calcul- ated	Diff %		
40	32.2	47.2	17.9	.418	79	31.4	42.7	22.7	35.2	55.1	24.9	- 9.7	2.1	3.7	76.2	3.16	50.5		
41	33.4	47.9	19.9	.418	84	16.5	28.1	28.4	41.8	47.2	30.5	- 7.4	2.6	4.4	69.2	3.85	48.1		
42	32.1	43.7	19.9	.412	90	11.2	17	25	35.7	40	25.1	- 0.4	2.4	3.8	58.3	3.2	33.3		
43	33.8	45.9	19.6	.422	93	11.5	23.7	34	44.9	32.1	32.9	- 3.2	3.1	4.7	51.6	4.18	34.8		
44	27.6	38.2	17	.418	90	21.5	22.5	17.4	27.2	56.3	17.8	- 2.3	1.74	3.0	72.4	2.35	35.1		
45	29.3	40.4	17.1	.412	90	15.5	21.1	22.7	31.9	40.5	21.7	4.4	2.6	3.5	34.6	2.84	9.2		
46	34.5	46.7	20.9	.412	91	9.75	17.1	30.3	42.8	41.3	31.3	- 3.3	3.0	4.4	46.7	3.94	31.3		
47	29.7	40.3	17.1	.405	98	31.3	5.5	29.3	38.8	32.4	27.2	- 7.2	3.2	4.2	31.3	3.56	11.3		
48	28.2	39.4	16.8	.405	94	17.1	21.7	21.2	33.6	58.5	22.9	- 8.0	2.25	3.65	62.2	3.0	33.3		
49	27.3	41	10.3	.423	96	14.8	43.8	49.3	48	2.6	34.5	-30	4.0	5.3	32.5	4.6	15		
50	30.9	40.8	20.5	.41	86	52.8	67.6	21.3	24.4	14.6	16.0	-24.8	2.1	2.6	23.8	2.1	0		
51	31.8	42.6	21.1	.41	85	13	18.0	24.1	24.4	1.2	16.2	-32.8	1.9	2.6	36.8	2.1	10.5		
52	33.5	44.3	22.1	.41	86	26.9	41.6	25.8	29.6	14.1	20.4	-20.9	2.3	3.1	34.8	2.6	13.0		