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EXPERIMENTAL STUDY OF CONVECTIVE HEAT EXCHANGE BETWEEN ZONES

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Abstract

Natural convective heat transfer between two zones through an opening is studied experimentally. Experiments are carried out in a full-scale test chamber with two zones connected to each other by an opening of 2.06mx1.25m for Grashof number in the range, $4x10^8 \leq Gr \leq 2x10^9$. The study of natural convective heat exchange is based on the energy balance of considering heat injection into the test chamber and losses through the enclosures, radiative exchange and heat conduction through the separation between the two zones in the test chamber. Experimental results are presented by empirical formulae of Nusselt number and Grashof number correlations and compared with results found in literatures. The discharge coefficient found in this study is between range, $0.42 \leq C_d \leq 0.45$, which is deduced from the relationship between the theoretical expression of discharge coefficient and the empirical relations. Measurements show that the flow through the opening between the two zones at low temperature difference range is unstable and therefore the conventional methods based on laminar flow analysis probably need to be improved.

1. Introduction:

Experimental studies has been carried out in conjunction with an international cooperation program of IEA Annex 20.2 for the study of inter-zone air flow. This subject has been studied extensively in the past few years. Previous research in this field were mainly based on one or two dimensional laminar flow analysis, similitude model experiments, numerical simulation of laminar flow N-S equation; limited experiments were in real physical configurations; viscous effect of fluid was depicted by introducing the discharge coefficient. It appears that in similitude model experiments, one has to increase much of the temperature difference in order to compensate the reduction of physical scales. As a result, the flow conditions in the scale models and real world may not be equivalent even within the same Grashof number if the turbulence effect were considered. It is also appearent that the turbulence effect may not satisfactorily be explained by adjusting the coefficient and exponent in empirical equation which was based on laminar analysis.

The purpose of this study is to investigate, through careful experiments in realistic physical scale, the effect of natural convection heat transfer between zones especially at low horizontal temperature difference. Results can be helpful in design and to have better understanding for energy consciousness in residential housing. The present study experiments are carried out in a test chamber of 5.5mx2.5mx2.5m, with two zones connected to each other by an opening of 2.06mx1.25m for Grashof number over the range $4 \times 10^8 \le Gr \le 2 \times 10^9$ or in temperature difference between two zones range 0.5°C to 2°C. Experimental results are presented in form of empirical equations providing the correlation between Nusselt and Grashof numbers. Also, results are compared to previous studies found in literatures.

2. Fundamental theory:

Figure 1 gives the concept for an one dimensional buoyancy driven flow through a door way.

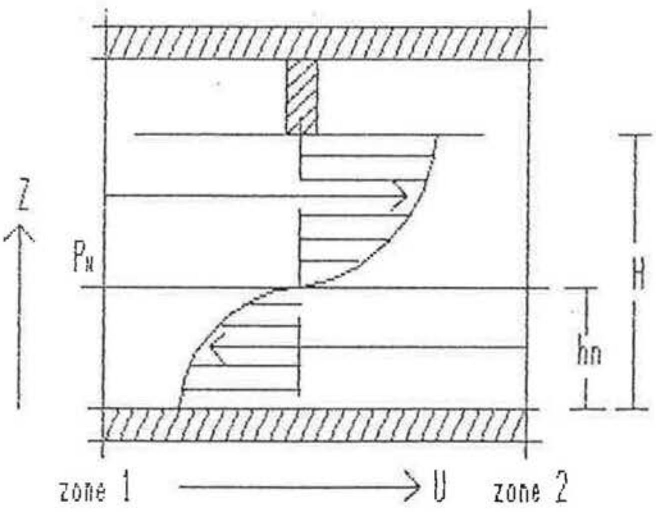

Figure 1. Conceptual pattern of 1-D inter-zone air flow

According to Figure 1, we obtain the theoretical volume flow rate passing through the opening with width of W by the following assumptions:

1) Bernoulli equation analysis;
2) air densities in the two zones satisfy $\rho_1 \approx \rho_2$;
3) discharge coefficient C_d to include the friction loss;

Let $\bar{\rho}=(\rho_1+\rho_2)/2$, $\bar{T}=(T_1+T_2)/2$ and $\beta=1/\bar{T}$, we have

$$V_{12} = V_{21} = \bar{V}_n = \frac{C_d}{3} WH(g\beta\Delta TH)^{0.5} \qquad (1)$$

Heat transport by air flow between two zones:

$$Q_c = \rho C_p \triangle T V_n = h_c \triangle T W H \quad (2)$$

and therefore,

$$h_c = (C_d/3)\rho C_p (g\beta \triangle T H)^{0.5} \quad (3)$$

Reorganise equation (3) to include the Nusselt, Prandtl and Grashof numbers, we have the following expression:

$$Nu/Pr = C\ Gr^{r} \quad (4)$$

The coefficient and exponent in equation (4) will be determined by experimental studies.

3. Description of the experimental installation:

3.1 The test chamber:

Experiments were carried out in the calorimetric chamber isolated in the laboratory hall, as shown in figure 2. The calorimetric chamber, 5.5mx2.5mx2.5m, was made of 102mm polystyrene foam clad by 1mm aluminium sheets on either sides. Two zones in the chamber were communicated to each other by an opening of 2.055mx1.25m. The partition was made of the same material as the enclosure. The geometry of the test chamber can be characterised by the commonly used parameters of aperture ratio A_p=0.88, aspect ratio A_s=0.45 and partition thickness ration t_H=0.05. The calorimetric chamber was isolated from the environment by an extra enclosure with air envelop in between. The air temperature in envelop space was controlled by an air handling unit installed next to the test chamber and connected by supply and return conduits.

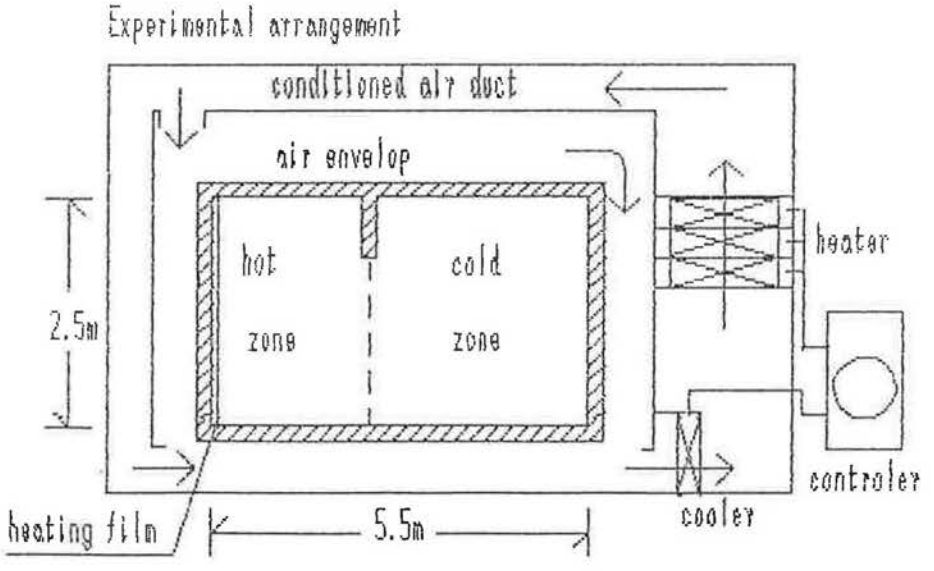

Figure 2. The test chamber and its arrangement

3.2. Measuring arrangements

Totally 93 thermocouples were used in measurement of surface and air temperatures. A movable trolley was used to carry sensors for measurements of air temperature, velocity and pressure difference on the plane of opening. These include:

- External and internal surface temperatures of each wall in both zones of the test chamber were measured by two thermocouples on each side covering the same area, respectively. 8 Thermocouples were used to measure surface temperatures on the partition.

- Air temperatures in each zone were measured by three vertical columns situated at relevant positions with 5 thermocouples on each column at different levels to cover the same volume. The centrally positioned column was used to measure the average air temperature based on five levels. The other two columns were used together to give the volume weighted average temperature based on assumption of symmetric distribution of air temperature and velocity fields.

- Air temperature in the envelop space were measured by 13 thermocouples scattered in the space 5cm away from the external surface of the test chamber.

- On the plane of opening, 17 positions were fixed for measurement of air velocity and temperature fields.

- The end wall in hot zone was heated by an uniformly distributed resistance film as heat source. Two thermocouples were used to measure the surface temperature of the heating film.

3.3. Measuring equipments and techniques

A detailed study of error estimation is given elsewhere [1], which includes the system static, dynamic error and random error in direct measurements. Critical parameters which would dominate the propagation of error were found and therefore the test system was known to be accurate provided that these parameters were under satisfactroy control.

- Copper-constantan thermocouples were used for measurement of surface and air temperatures at fixed positions. Thermocouples used for air temperature measurement all were shaded to reduce radiation error.

- An electric power transducer was used to measure instantaneously electric power injection into the test chamber.

- DISA 54N50 low air velocity analyser was used to measure air velocity and temperature fields on the plane of opening.

- A remote controllable trolley was used to carry the DISA to the 17 positions on the plane of opening. Note that the 17 positions are relevant to the top velocity sensor, the positions of temperature sensor was slightly away from this positions.

- Data logger ORION 3530 was used for data acquisition and also as controller for regulating the air temperature in envelop space by sending digital signal to the relays to activate the ON/OFF status of the electric heater in the air handling installation. A Zenith microcomputer was used for coordinating the data logger in order to achieve better processing possibility.

- A stabilizing period of 20 hours was specified after each time of power change for the system to reach equilibrium condition. The selection of this period was based on the study of system dynamics. It has been proved that keeping such period for this test system would reduce the static error due to thermal storage of enclosure and air volumes to less than 0.9%. At each level of change of injected power, a transient record including all the surface temperatures, air temperatures in space and electric power was made over the stabilizing period of 20 hours at an interval of 15 minutes per scan.

- In equilibrium condition, measurements began after the stabilizing period when the controlled parameters are within acceptable perturbation ranges. It has been proved that the fluctuation of power injection has less enfluence to the measurement error, however, the fluctuation of air temperature in envelop space would cause significant dynamic deviation. In steady-state test, the acceptable envelop air temperature fluctuation should be less than ±0.5°C. And for this the uncertainty in heat balance would be less than 17W calculated based on maximum temperature fluctuation. During measurements in equilibrium conditions, all data were recorded for a 3 hours period by 2 minutes per scan. Finally the arithmetic mean from about 90 valus of each parameter were taken for heat balance calculation. At the same time, standard deviation of each parameter was calculated for further checking of the equilibrium condition.

- Measurements of instantaneous air velocity and temperature on the plane of opening consecutevely started after each steady-state test by speed of 2 samples per second for a period of 50 second for each position. The whole period would eventually take about 40 minutes for all the 17 positions. This measurement was based on the assumption that the air temperature and velocity fields would remain unchanged within the equilibrium period and therefore measurements in different time could still represent the time average situation.

3.4. Experimental data processing.

A package of softwares were developed on microcomputer for measurement data processing. These mainly include:

- The test system calibration. This was done by a global energy balance of taking into account the measured power injection and heat losses through enclosures according to measured temperature difference of walls during the equilibrium conditions by an iteration procedure. The global thermal conductivity of the enclosure was then calculated and represented by a linear correlation of mean temperature of enclosure.

- Calculation of mean value and standard deviation for each parameter from measurements of the equilibrium period to ascertain the equilibrium condition and for heat transfer calculation.

- Calculation of mean values, standard deviation and turbulence intensity fields from the instantaneous measurements of velocity and temperature on the plane of opening.

- Determination of convective heat transfer coefficient. This was done by firstly calculating the radiative exchange between the two zones using view factor of each pair of surfaces between the two zones, and then the heat conduction through the partition by knowing the temperature difference on either sides. Finally the pure convective heat exchange through opening was obtained by deducting the conductive heat through partition and radiative heat exchange between the two zones from the total enclosure heat loss of the cold zone.

4. Experimental results and discussion

Experiments were carried out at four power injection levels around 150W, 250W, 500W and 900W, respectively. This is equivalent to a Grashof number in range $4 \times 10^8 \leq Gr \leq 2 \times 10^9$, horizontal mean temperature difference $0.5°C \leq \Delta T \leq 2.0°C$, or net convective heat transfer rate $45W \leq Q_c \leq 285W$. Opening height remained unchanged for all tests, which was 2.055m.

4.1 Temperature and velocity measurements:

Figure 3 shows a cooling period after the 150W power injection. Figure 4 shows the transition period started from change of injected power 500W to 900W. The average system time constants are also indicated. From experiments, an average system time constant of approximately 4 hours was found, which is close to the estimation [1].

Figure 5 shows certain temperature evolutions for a 3-hour period test within the equilibrium condition. Maximum standard deviation for all the temperatures were usually less than magnitude of ±1/10°C. Fluctuation of power usually was higher. This confirms the system dynamic error would be much less than estimation based on ±0.5°C.

Figure 6, 7 present the non-dimensional vertical air temperature profiles in central positions of the hot and cold zones at various heat exchange rates. It can be found that the dependence of vertical temperature profile on zonal temperature (or in fact zonal heat exchange) is weak. Temperature gradients at top and bottom are higher. This seems that the main air motion may flow along the periphery of the test chamber.

Figure 8 shows the vertical air temperature gradients of the two central positions in hot and cold zones versus Nu/Pr. A rather linear relationship was found. This also can be explained by the fact that when horizontal temperature difference increases, vertical temperature difference is also proportionally increasing.

Figure 9 and 10 are temperature and velocity fields on the plane of the opening. Note that the left half field in figures were plotted by the assumed symmetrical pattern, only two positions were checked. For velocity, since the sensor is omnidirectional, only positive values appeared. It can be seen that the vertical temperature field is nearly linear, which seems agree with the hypothesis of the one dimensional flow. However, velocity field does not satisfy the vertical square root relation of one dimensional flow. This may be explained by the reasons that instead of flow in the main horizontal direction, there exist vertical and transversal velocity components which gave rise to measurements of higher resultant velocity. And also, frequency of air velocity fluctuation at minimum velocity level was observed higher and so that the mixing of the two air streams may be significant at this level. This again emphasizes that the real condition of interzonal air flow pattern could be very much different from one or two dimensional analysis.

4.2 Convective heat exchange

For heat transfer study, the height of opening was taken as characteristic length. Both central column and volume weighted temperature differences were taken as characteristic temperature difference to produce two sets of result. Temperature difference based on the nodes at the central positions of the two zones was found inaccurate. The relation of the three temperatures follows:

$$\Delta T_v > \Delta T_c > \Delta T_u$$

Measured convective heat exchanges through door way Q_c, temperature difference _T and β were used to calculate Nusselt and Grashof numbers as follows:

$$Nu_H = \frac{Q_c}{w \Delta T k} \qquad (5)$$

$$Gr_H = \frac{g \beta \Delta T H^3}{\nu^2} \qquad (6)$$

Nu_H and Gr_H of different tests were then used to determine coefficient C and exponent Γ in equation (4) by logarithmic regression.

Figure 11 shows the regression equations against experimental results for two temperature differences. In our tests, the measured volume weighted average temperature always higher than the central column average and so that it stays lower in figure 11. Both lines and respective measured data present similar scattering, which seems that both temperature differences are similar acceptable. This is important, because that central column average is much easy to measure in practice.

The empirical equations related to figure 11 are given as follows with correlation coefficient of R≈0.98:

$$\frac{Nu_H}{Pr} = 1.307\ Gr_H^{0.40} \qquad (7)$$

using ΔT_c, the temperature difference between the two central column average temperatures and

$$\frac{Nu_H}{Pr} = 1.225\ Gr_H^{0.40} \qquad (8)$$

using ΔT_v, the difference between the volume weighted average temperatures.

Figure 12 shows a comparison of our experimental results with results found in literature.[2] Note that comparison would be meaningful only if made to those using same characteristic parameters.

Empirical relations of mean air velocity between two zones was found based on equation (1) by using the measured convective heat Q_c and temperatures differences, ΔT_v and ΔT_c, and by logarithmic regression. These relations are given in figure 13 along with empirical equations ($R \geq 0.98$):

$$\overline{V} = 0.117(g\beta\Delta T_c H)^{0.41} \qquad (m/s) \qquad (9)$$

using ΔT_c, the temperature difference between the two central column average temperatures and

$$\overline{V} = 0.110(g\beta\Delta T_c H)^{0.42} \qquad (m/s) \qquad (10)$$

using ΔT_v, the difference between the volume weighted average temperatures.

The coefficients C in equation (7) to (10) are equivalent to a theoretical discharge coefficient in range of

$$0.41 \leq C_d \leq 0.45.$$

which is close to 0.45 recommended by Sandberg [4] within the IEA Annex 20.2 cooperation task. The equivalence between coefficients C in empirical equation and C_d given by theoretical definition can be found by arithmetic manipulation of considering the difference between the exponents in empirical equations and theoretical value 0.5. (Appendix)

The following empirical equation was found by using experimental results (correlation coefficient $R \geq 0.98$) and can be useful for estimation of convective heat transfer coefficient between two rooms with normal size of door in residential houses. For better accuracy equation (7), (8) are recommended.

$$h_c = 357.2\left(\frac{\Delta T_c}{T_{mc}}\right)^{0.36} \qquad (W/m^2 K) \qquad (12)$$

in which T_{mc} is the mean room temperature in K.

Agreements with results obtained from some water similitude scale models are fair, which seem to have similar exponents but with difference for coefficient C. Reasons may firstly attribute to differences in geometry of the models. Brown et al [3] reported that heat transfer increases with decrease of partition thickness ratio based on their study of partition thickness ratio range $0.19 \le t_H \le 0.75$, while in reality t_H is much smaller, for example, in our test it was 0.05.

It is also believed that the similitude scale models are in general based on similarity analysis of laminar flow condition, reduction of physical scale is based on similarity analysis of dimensionless groups. However, even if the similarity transformation was performed by fully respecting the dimensionless groups, certain effects, e.g.the turbulence, still may not be well reflected because of fundamental restriction. Also, trying to increase temperature difference between two zones in order to achieve higher Grashof number may over-estimate the laminar flow condition and therefore, the effect of turbulence diffusion would then be under-estimated.

4.3 Trouble shooting:

Measurements of instantaneous temperature and velocity on the plane of opening need to be improved to include faster sampling speed, longer sampling period and visible checking of each sampling process. Also, accuracy of the sensor at low velocity range needs to be further confirmed.

5. Conclusions:

Natural convective heat transfer between two zones through an opening and air flow pattern were studied experimentally. This study was supported by full-scale experiments in a test chamber with two zones connected to each other by a normal sized opening. Experiments were carried out in range of $4\times10^8 \le Gr \le 2\times10^9$ or horizontal temperature difference of $0.5°C \le \Delta T \le 2.0°C$. The corresponding convective heat transfer rate was in range $45W \le Q_c \le 285W$.

The vertical temperature profiles measured showed that the main air flow was close to the peripheral surfaces. It was found that the vertical temperature gradients of the two zones were nearly linear with change of inter zone heat transfer. Measured air temperature and velocity fields on the plane of opening showed that the actual flow did not follow the one dimensional pattern of vertical square root relation. Three dimensional flow pattern may be rather different at low horizontal temperature difference range compared to one dimensional study.

Experimental results were presented by empirical formulae of Nusselt number and Grashof number correlations and compared with results found in literatures. The agreement of this study with previous studies of full scale models were excellent. Difference found in comparing to some scale models may attribute to different characteristic parameters used and ratio of partition thichness.

The discharge coefficient found in this study is between range, $0.42 \le C_d \le 0.45$, which is deduced from the relationship between the theoritical expression of discharge coefficient and the empirical relations. Measurements show that the flow through the opening between two zones at low temperature difference range is unstable and therefore the conventional methods based on laminar flow analysis probably need to be improved.

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Nomenclatures:

Gr - Grashof number
Nu - Nusselt number
Pr - Prandtl number
C_d - discharge coefficient
H - height of opening (m)
W - width of opening (m); electric power (Watt)
ρ - density of air (kg/m^3)
T - temperature of air (K)
β - coefficient of thermal expension of air (1/K)
V - air velocity (m/s)
g - gravitional acceleration (m/s^2)
ΔT - temperature difference (K)
Q - heat (W)
C_p - specific heat of air (J/kg K)
ν - kinematic viscosity of air (m^2/s)
C - coefficient
α - exponent
Γ - exponent
h_c - convective heat transfer coefficient (W/m^2K)
k - thermal conductivity of air (W/m K)
A_p - aperture ratio, (door height)/(room heigh)
A_s - aspect ratio, (room height)/(room length)
t_H - partition thickness ratio, (partition thick)/(door height)

Subscripts:

v - volume weighted average
c - central column weighted average; convective
u - single node in centre position
H - characteristic length of opening heigth
m - mean
R - regression correlation coefficient
1,2 - room number one and two

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Appendix Equivalence of discharge coefficient by theoretical expression and empirical equations

The theoritical formulae of mean velocity and heat transfer are represented by:

$$\overline{V} = C_d/3\ (g\beta\Delta TH)^{0.5} \qquad (1)$$

$$Nu_H/Pr = C_d/3\ Gr^{0.5} \qquad (2)$$

C_d is discharge coefficient, identical in equations (1) and (2).

In empirical formulae, different coefficients and exponents can be found because of statistical feature:

$$\overline{V} = C_v\ (g\beta\Delta TH)^{\alpha} \qquad (3)$$

$$Nu_H/Pr = C_h\ Gr^{\Gamma} \qquad (4)$$

In order to obtain the same air velocity by using equations (1) and (3), it is appearent that the following equivalence should be obeyed:

$$C_d = 3\ C_v(g\beta\Delta TH)^{(\alpha-0.5)} \qquad (5)$$

Since heat transfer relation is derived based on the exchange of air flow, according to equation (1) into (2), we have

$$C_h = C_v\ (H/v)^{(1-2\alpha)}Gr^{(\alpha-\Gamma)} \qquad (6)$$

Example:

From experiment, we have the following empirical equations:

$$Nu_H/Pr = 1.307\ Gr^{0.396} \qquad \bar{V} = 0.117(g\beta\Delta TH)^{0.411}$$

based on central column mean temperatures differences, and

$$Nu_H/Pr = 1.225\ Gr^{0.395} \qquad \bar{V} = 0.110(g\beta\Delta TH)^{0.415}$$

based on volume weighted mean air temperatures differences.

Recall equation (1), (2) and relation (5), (6), for average test conditions of:

$T_m = 306(K)$, $H = 2.055(m)$, $\Delta T = 1(K)$, $v = 17.3e{-}6(m^2/s)$,

we calculated:

$0.429 \geq C_{dc} \geq 0.391$ for central column average

$0.451 \geq C_{dv} \geq 0.417$ for volume weighted average

For central column average temperature difference, our calculation gives:

$C_h/C_v = 11.029$ or

$C_h = C_v*11.029 = 0.11*11.029 = 1.213$

In empirical equation, it was 1.225.

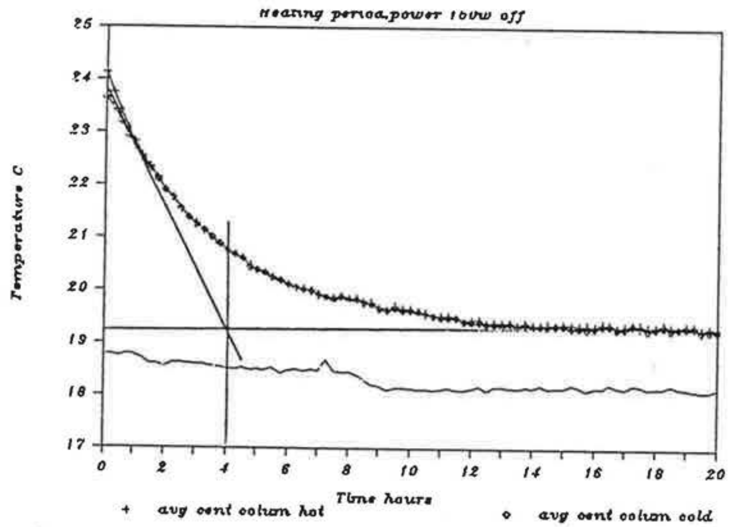
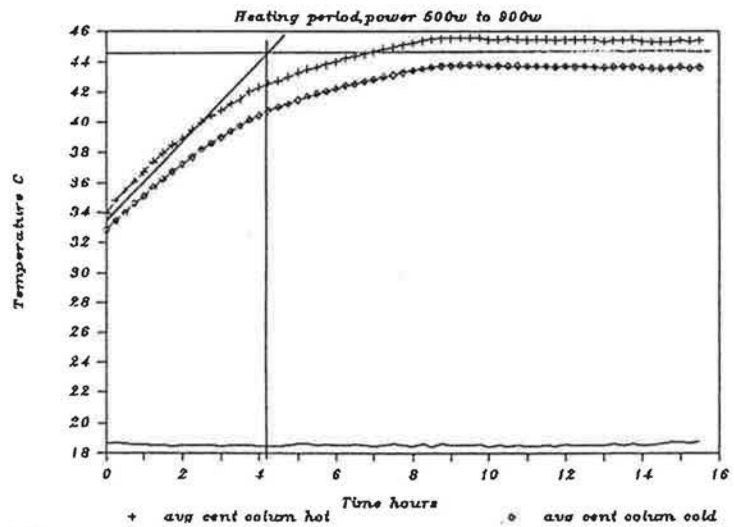


Figure 3



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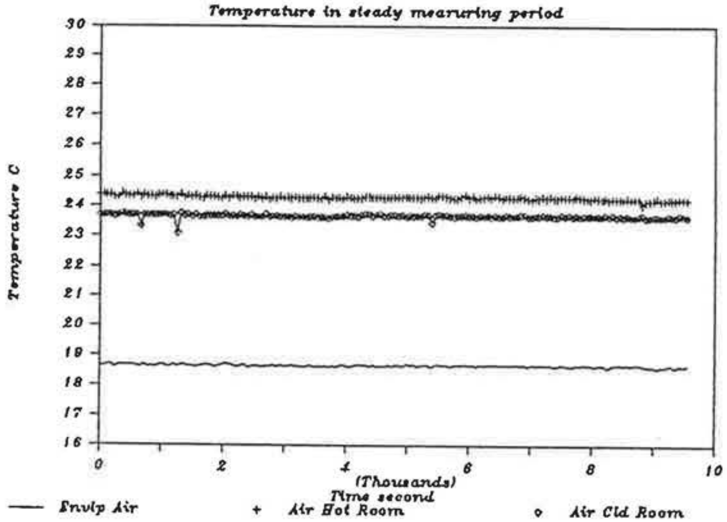

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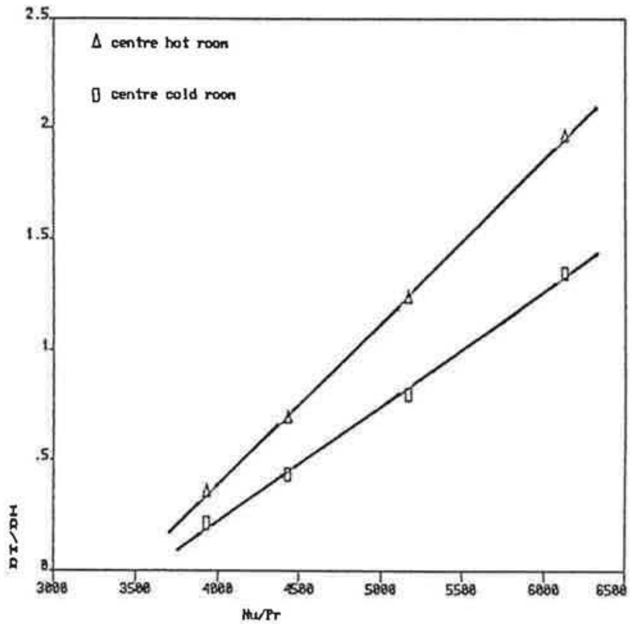

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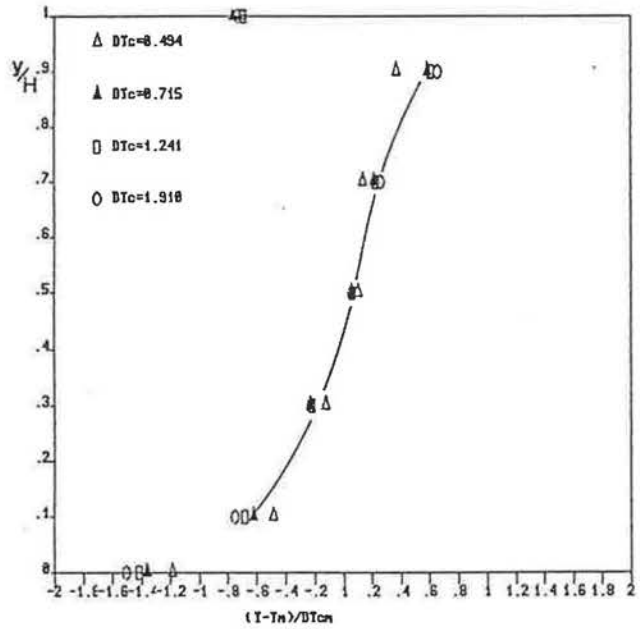

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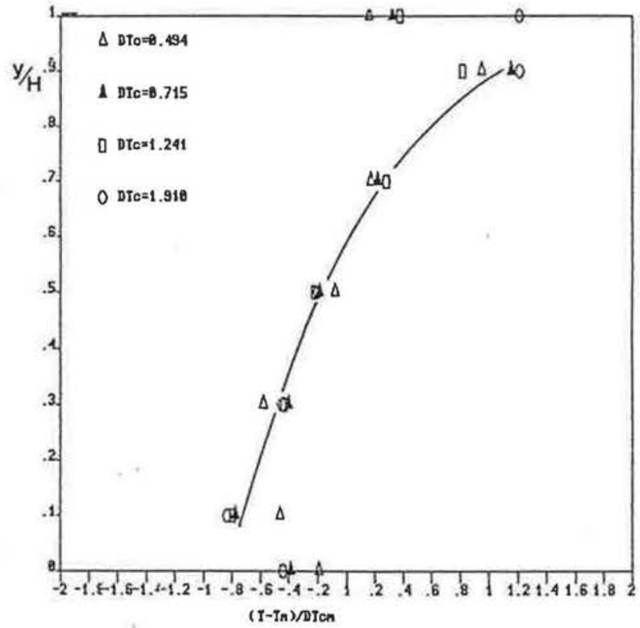

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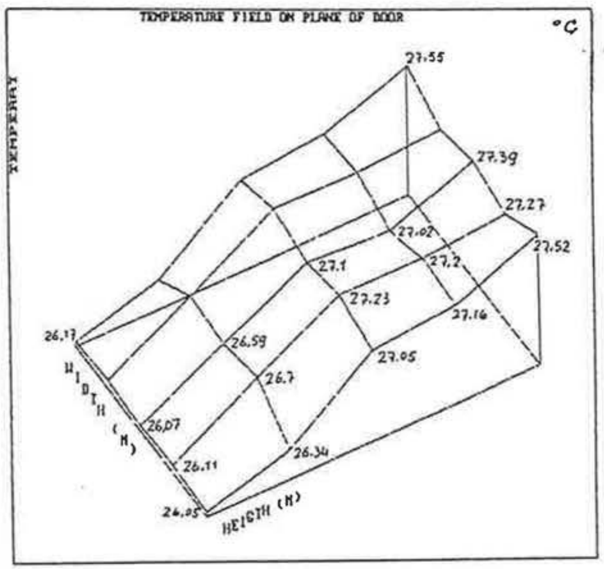

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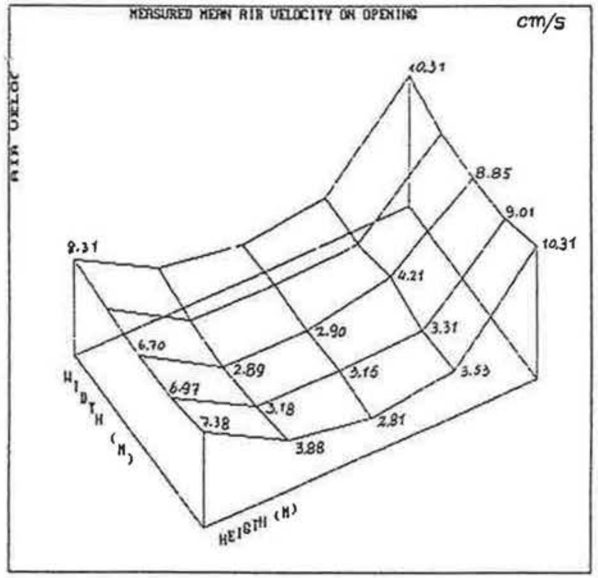

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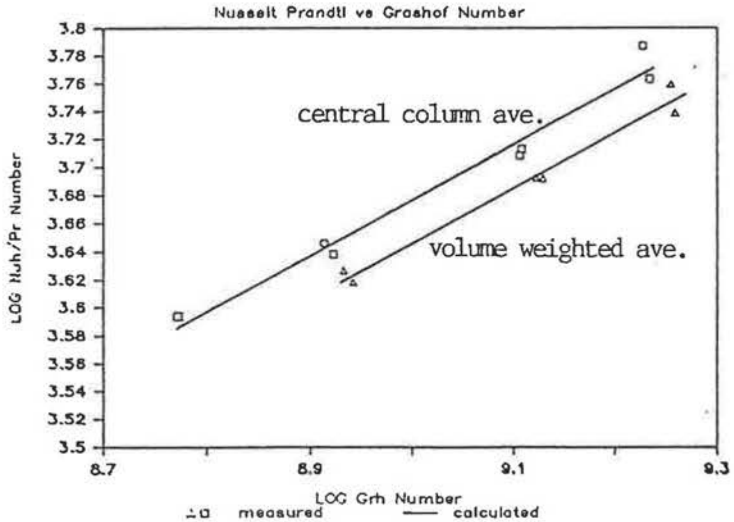

Figure 11

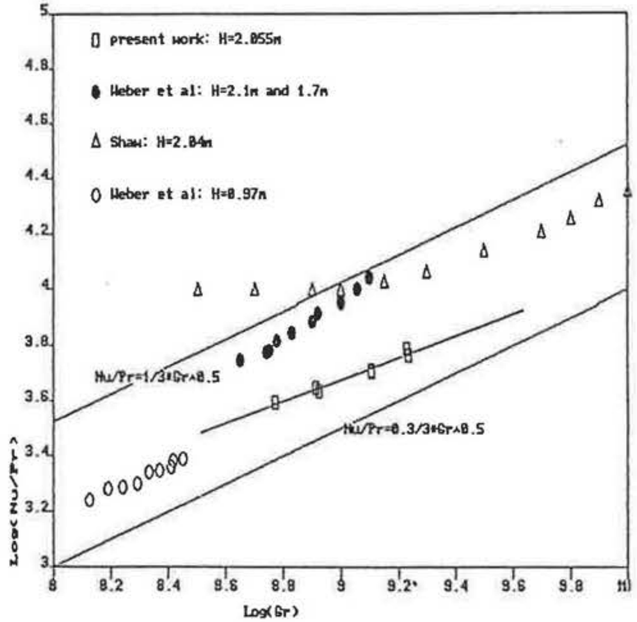

Figure 12

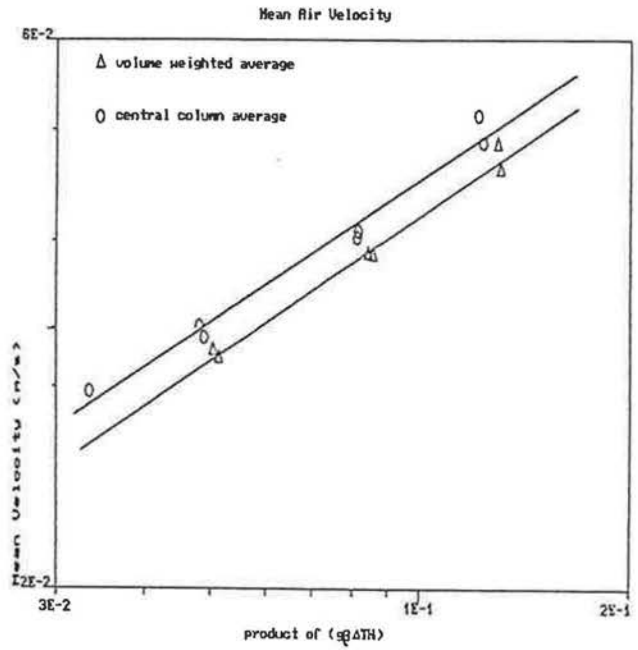

Figure 13