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EXPERIMENTAL STUDY OF NATURAL CONVECTIVE HEAT TRANSFER FROM AN INCLINED ISOTHERMAL CYLINDER WITH AN EXPOSED TOP SURFACE MOUNTED ON A FLAT ADIABATIC BASE

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ABSTRACT

Natural convective heat transfer rates from inclined cylinders with a circular cross-section and which have an exposed top surface have been experimentally measured. The cylinder is mounted on a large flat essentially adiabatic surface with the other cylinder surfaces exposed to the surrounding air and with the cylinder, in general, inclined to the vertical at angles between vertically upwards and vertically downwards. The diameter-to-height ratio of the cylinders used in the present study was comparatively small, diameter-to-height ratios of between 1 and 0.25 being used. The main aim of the present work was to determine how the diameter-to-height ratio of the cylinder, i.e., *D/h*, influences the mean heat transfer rate from the cylinder at various angles of inclination. The heat transfer rates were determined by a transient method, which basically involving heating the model and then measuring its temperature-time variation while it cooled. Tests were carried out in air with all models at various angles of inclination to the vertical of between 0^{o} and 180^{o}. The Rayleigh number, *Ra*, based on the cylinder height, *h,* was between approximately 2E4 and 4E6. The experimental results have been compared with the results obtained in an earlier numerical study.

INTRODUCTION

The components that occur in some electrical and electronic cooling problems can be approximately modeled as involving natural convective heat transfer from an inclined cylinder with a circular cross-section mounted on a flat adiabatic base plate. The present study is concerned with the experimental measurement of natural convective heat transfer rates from such cylinders mounted on a large flat essentially adiabatic surface with the other cylinder surfaces exposed to the surrounding air. This flow situation, which is considered here is shown in Figure 1. The heat transfer rate from the surfaces of the cylinders at various angles of inclination between vertically upwards and vertically downwards, as shown in Figure 2, has been experimentally studied. One of the main aims of the present work was to determine how the cross-sectional size-to-height ratio of the cylinder, i.e., *D/h*, influences the mean heat transfer rate from the cylinder at various angles of inclination, and to compare the present experimental results with those obtained in an earlier numerical study [1].

Natural convection from vertical slender cylinders has been studied for many years. Reviews of early work in this area were provided [2-5]. Investigator [6] gives numerical results for Prandtl numbers from 0.01 to 100, extending earlier work that had been for $Pr = 1$ and 0.72. This work was further extended in [7]. Integral method results were obtained [8]. The effect of Prandtl number was further studied [9, 10]. Several more recent studies have further extended this earlier work with a typical such study being that of [11]. They found that for a constant heat flux the cylinder, diameter has a great effect on the local heat transfer coefficients of the cylinder and show marked increase in the both regions of laminar and turbulent flows with decreasing the diameter. The effects of wall temperature variations have been studied by, for example [12-14]. Typical of the experimental studies of natural convective heat transfer from vertical cylinders are those of [15-17]. Some attention has been given to the case where there is a specified heat flux at the surface of the cylinder, e.g., see [18, 19].

Most previous studies of natural convective heat transfer from inclined cylinders have only been concerned with relatively long cylinders [20-24]. Although some work on short cylinders where the end effect are negligible have been undertaken [25-27], their results indicate that the mean Nusselt number based on diameter increases with decreasing *h/D* for constant diameter and the mean Nusselt number increases with increasing angle of inclination from vertical to the horizontal position, also they provide a correlation equations. Cylinders with exposed ends have received relatively little attention, the vertical cylinder case having been considered [28-31] who obtained results for the cases of short cylinders having exposed ends that were pointing "upwards", he found that the mean Nusselt number for the heated top horizontal surface is much lower than that for the heated vertical surface and provide a correlation equation for the mean Nusselt number. In [32, 33] researcher undertook an experimental study for the case where the cylinders are inclined to the horizontal with the exposed end inclined from horizontal to vertical position where top surface was pointing "downwards", their results indicate that the heat transfer rate from cylinders with exposed ends are lower than those from cylinders with insulated ends, the difference increase with increasing length-to-diameter ratio and the heat transfer from rates from short cylinders with exposed ends pointing downward are very close to those for short cylinders with exposed ends pointing upwards. Little attention have been given to cases in which there is a possible interaction of the flow from the heated "vertical" wall with that over top heated portion of the cylinder for different angles of inclination relative to the vertical and there is still a need for a broader range of the governing parameters which can be used as a basis for prediction heat transfer rates in practical situation involving these types of flow. The situation considered here involves such a potential interaction because for different angles of inclination the flow from the side heated wall of the cylinder can interact with the flow over the heated top surface of the cylinder. The present work involves an experimental study undertaken in an effort to provide further confirmation of the previously results obtained in [1].

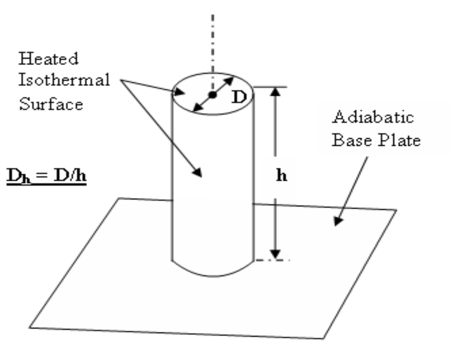

Figure 1 Flow Situation Considered

NOMENCLATURE

A	[m^2]	Dimensionless total surface area of heated cylinder
C	[KJ/KgK]	Specific heat of material from which model is made
D	[m]	Diameter of cylinder
g	[m/s^2]	Gravitational acceleration
h	[m]	Height of heated cylinder
k	[W/mK]	Thermal conductivity of fluid
m	[kg]	Mass of model
Nu	[-]	Mean Nusselt number
Ra	[-]	Rayleigh number based on height of the cylinder *h*
T	[K]	Model temperature
t	[s]	Time to go from T_i to T_e
Special characters		
σ	[-]	Stefan-Boltzman constant
ε	[-]	Emissivity of the model
α	[m^2/s]	Thermal diffusivity
β	[1/K]	Bulk coefficient
ν	[m^2/s]	Kinematic viscosity
φ	[Degree]	Angle of inclination of the cylinder relative to the vertical
Subscripts		
total		Dimensionless total surface area of heated cylinder
side		Dimensionless surface area of side portion of heated cylinder
Top		Dimensionless Surface area of flat surface of heated cylinder
h		Dimensionless diameter of cylinder (=*D* / *h*)
t	[W/m^{2}K]	Total heat transfer coefficient
c	[W/m^{2}K]	Convective heat transfer coefficient
cd	[W/m^{2}K]	Conductive heat transfer coefficient
r	[W/m^{2}K]	Radiation heat transfer coefficient
mcemp		Mean Nusselt number given by correlation equation
0		Mean Nusselt number when curvature effects are negligible
e		Final model temperature
i		Initial model temperature
∞		Ambient air temperature
F		Fluid temperature
wavg		Wall average temperature of surfaces of cylinder

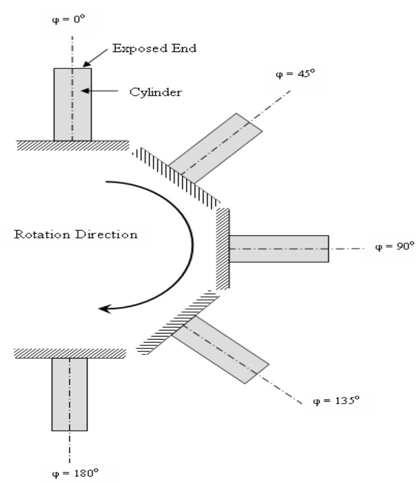

Figure 2 Definition of Inclination Angle and Faces

EXPERIMENTAL APPARATUS AND PROCEDURE

The present experimental study of natural convective heat transfer from a cylinder which is inclined at an angle to the vertical, was carried out by mounting the models on one wall of the test chamber. The test chamber was open at the top and bottom and constructed in such a way that it could be rotated around a fixed horizontal axis, in this way allowing the inclination angle of the test section to be changed. Figure 3 shows a schematic front view of the experimental apparatus. The test chamber was constructed using transparent acrylic plates and has dimensions of 120cm height × 25cm width × 30cm depth so it surrounded the experimental models with a large volume of air. The test chamber was large enough to ensure that its walls did not affect the flow over the models and the heat transfer from the models. Furthermore, the test chamber was placed in a larger box 190cm height × 135cm width × 125cm depth. This arrangement ensured that no external disturbances in the room air and short term temperature changes in the room did not interfere with the experiments.

The active components of the test apparatus were the solid aluminum cylinder models. The model dimensions are given in Table 1, the cylinder experimental models having a height of h and diameter D. The aspect ratio of the models used, D/h, ranged from 0.25 to 1. The bottom surfaces of the models were attached to a large base made of plexiglass of 29cm in length, 23.5cm in width, and 1cm thickness. The ends of the models in contact with this base were internally chamfered to a depth of 1mm in order to reduce the contact area between the model and the base sheet in order to reduce the conduction heat transfer from the model to the base. A series of small diameter holes was drilled longitudinally to various depths into the models. Six or five, depending on the model height, thermocouples type-T 0.25mm diameter at various locations inserted into these holes were used to measured models temperatures. The thermocouples were monitored using a data acquisition system supplied by TechmaTron Instrument (model USB-TC) that in turn was connected to a personal computer. This unit was self calibrating. These thermocouples with the data acquisition unit were calibrated in a digital temperature controller water bath manufactured by VWR model 1166D, using a calibrated reference thermometer manufactured by Guildline Instruments Company (model 9535/01,02,03) at temperatures between 20^{o}C to 100^{o}C and the uncertainty in the thermocouple outputs was found to be less than $\pm 0.5^o$C. As noted before the model assemblies were mounted inside the test chamber shown in Figure 3 and allowed the models to be set at any angle to the vertical.

The heat transfer rates were determined using the transient method, i.e. by heating the model being tested and then measuring its temperature-time variation while it cooled. In an actual test, the test chamber was set at the required angle. Then the test model was heated in an oven to a temperature of about 120^{o}C and inserted inside the test chamber. The model temperature variation with time was then measured while it cooled from 90^{o}C to 40^{o}C. Tests and approximate calculations indicated that, because the Biot numbers existing during the tests were very small, between 1.6×10^{-4} and 2.2×10^{-4}, the use of the lumped capacitance method was justified, this method requiring that $Bi<10^{-1}$. As a result of low Biot number, the temperature of the aluminum models remained effectively uniform at any given instant of time during the cooling process. The overall heat transfer coefficient could then be determined from the measured temperature-time variation using the usual procedure, i.e., the temperature variation of the model was given by:

$$\left(\frac{h_t A}{mC}\right)t = \ln\left(\frac{T_i - T_\infty}{T_e - T_\infty}\right) \tag{1}$$

Hence, h_t could be determined from the measured variation of $ln(T_i - T_\infty)/(T_e - T_\infty)$ with t, using the known value of (A/mc).

The value of h_t so determined is, of course, made up of the convective heat transfer to the surrounding air, the radiant heat transfer to the surroundings and the conduction heat transfer from the model to the base. The radiant heat transfer could be allowed for by calculation using the known emissivity 0.1 of the polished surface of the aluminum models. So the radiant heat transfer coefficient could be calculated as,

$$h_r = \varepsilon\,\sigma\left(T_{w_{avg}} + T_\infty\right)\left(T_{w_{avg}}^2 + T_\infty^2\right) \tag{2}$$

The conduction heat transfer to the base was determined by fully covering the models with Styrofoam insulation and using the transient method as described before which was using to find the conduction heat transfer coefficient to the base. The convective rate of the heat coefficient by convection was then calculated:

$$h_c = h_t - h_r - h_{cd} \tag{3}$$

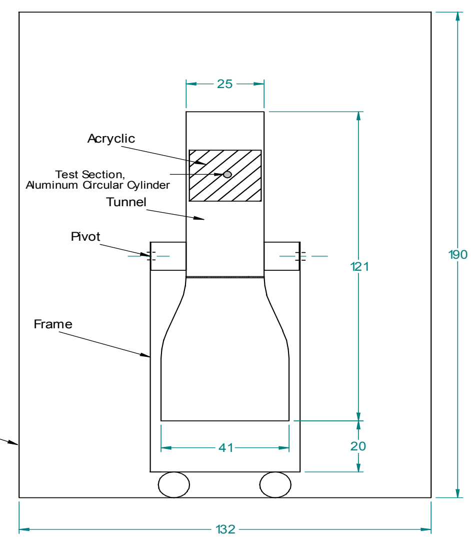

Figure 3 Experimental Apparatus, Dimensions are in cm

The estimated relative values of convection, conduction and radiation heat transfer coefficients depend on the experimental model size, orientation and Rayleigh number, e.g. when the Rayleigh number was equal to 6.54×10^4, $\varphi=0^o$ and $W=1$, $h_c\approx62.1\%$, $h_{cd}\approx35.1\%$ and $h_r<2.8\%$ of the total heat transfer while when Rayleigh number equal to 3.83×10^6, $\varphi=0^o$ and $W = 0.25$ $h_c\approx 72.4\%$, $h_{cd}\approx 23.2\%$ and $h_r< 4.4\%$ of the total heat transfer.

As discussed in [1] for air the natural convection Nusselt number, Nu, for short cylinders inclined at an angle to the vertical depends on the Rayleigh number, Ra, the ratio of the diameter to the height of the cylinder, D_h and the inclination angle to the vertical, φ, i.e.:

$$Nu = f(Ra, D_h, \varphi) \tag{4}$$

Where Ra is defined as,

$$Ra = \frac{\beta g(T_{w_{avg}} - T_F)h^3}{\nu\alpha} \tag{5}$$

The average Nusselt number is defined as usual by:

$$Nu = \frac{h_c h}{k} \tag{6}$$

In expressing the experimental results in dimensionless form, all air properties in the Nusselt and Rayleigh number were evaluated at the mean film temperature $(T_{w_{avg}} + T_F)/2$ existing during the test.

The uncertainty in the present experimental values of Nusselt number arises due to uncertainties in the temperature measurements, non-uniform heating, temperature differences in the model during cooling, and uncertainties in the corrections applied for conduction heat transfer through the base. Therefore, an uncertainty analysis was performed by applying the estimation method described in [34,35]. The uncertainty in the average Nusselt number was found to be function of temperature, the angle of inclination and the size of the model used. Overall, the uncertainty in the average Nusselt number was estimated to be less than ±13%, and the scattered in the experimental data was found to be between ±3% of the mean values.

Tests were performed with models mounted at various angles of inclination between vertically upwards and vertically downwards as shown in Figure 2. This work will focus on the effect of inclination angle and the aspect ratio of the cylinder with an exposed top surface on the mean heat transfer rate.

Table 1 Model Dimensions

Model No.	Diameter (*D*) mm	Height (*h*) mm	D_h=*D/h*
1	25.4	25.4	1
2	25.4	50.8	0.5
3	25.4	101.8	0.25

RESULTS

The solution has the following parameters:

- The Rayleigh number, *Ra*, based on the 'height' of the heated cylinder, *h*, and the overall temperature difference $T_{wavg} - T_F$,
- The dimensionless width of the cylinder, $D_h = D/h$
- The inclination angle φ of the cylinder

In order to validate the results a comparison have been done between the present experimental results and the correlation equation from a vertical cylinder with exposed top surface [1, 28], and a correlation equation for the heat transfer from the side surface of the cylinder where the heat transfer from the top surface are neglected [2, 25]. Investigators [8] give a correlation equation which includes the curvature effects. The results given by these correlation equations are shown in Figure 4. Good agreement between the correlation equations and the experimental results for vertical cylinder for $D_h = 0.25$ where the heat transfer from the top surface is small compared to the heat transfer from the side surfaces. The variation in the Nusselt number between these correlation equations comes from the fact that there is no identical solution, neglecting the effect of the top surface and some correlation covered a wide range of Prandtl number, Rayleigh number and diameter to height ratio. Also these correlation equations were determined by fitting results to an acceptance order of accuracy.

Typical variations of experimental and numerical values of the mean Nusselt number for the entire cylinder, *Nu*, with Rayleigh number, *Ra*, for various values of the dimensionless cylinder width, D_h, at different angles of inclination, φ, between vertically upward and vertically downward are shown in Figure 5, Figure 6, Figure 7, Figure 8 and Figure 9. These figures show that the mean Nusselt number increases as the Rayleigh number increases and as the dimensionless cylinder diameter decreases for all values of inclination angles considered in this study. The increase in the Nusselt number with decreasing dimensionless cylinder diameter, D_h, arises from the fact that the wall curvature and three-dimensional effect becomes more significant which change the magnitude of the flow over the cylinder surfaces.

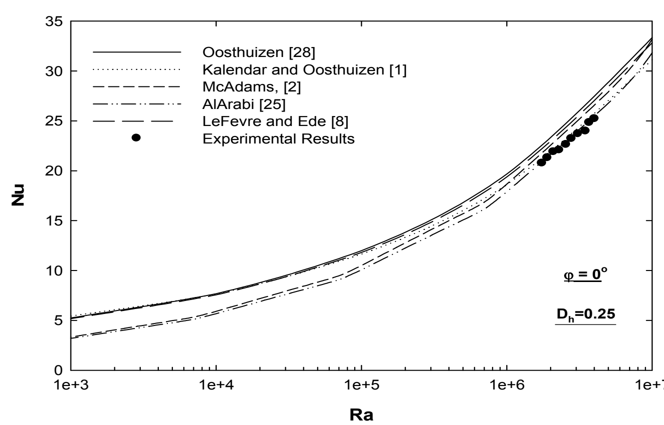

Figure 4 Comparison of mean Nusselt number values given by correlation equations and experimental results for $D_h=0.25$ and $\varphi=0^o$

The experimental results given in Figure 5 to Figure 9 will be seen to be in good agreement with the earlier numerical results [1] and show the same trends as the numerical results.

Figure 10 shows the variation of the mean Nusselt number with angle of inclination for three values of D_h. It will be seen that there is an acceptable agreement between the numerical and experimental results within the experimental uncertainty. Furthermore, the mean Nusselt number variation at the lower values of the Rayleigh number is different from that at the higher values of the Rayleigh number. At lower values of the Rayleigh number the mean Nusselt number is almost independent of the angle of inclination. However, at higher values of Rayleigh number this is not the case. This is because at higher values of the Rayleigh number the boundary layer become thinner and the interaction between the flows over the various surfaces that make up the cylinder become significant. The mean Nusselt number increases as the dimensionless diameter D_h decreases. At the lower values of the Rayleigh number the mean Nusselt number is almost independent of the angle of inclination. However, for higher values of Rayleigh number and dimensionless cylinder diameters, D_h , less than 0.5 this is not the case. At the higher values of the Rayleigh number for dimensionless diameters less than 0.5 the highest mean Nusselt number occurs when the cylinder is in a horizontal position, i.e., when φ is equal 90^o while the lowest mean Nusselt number occurs when the cylinder is in a vertical position where the "top" surface is facing up or facing down, i.e., when φ is equal 0^o or 180^o. Furthermore, the mean Nusselt number when the cylinder is in a vertical position where the "top" surface is facing up is approximately equal to the mean Nusselt number for the corresponding position when the "top" surface is facing down. These situations arise from the fact that when the angle of inclination of the cylinder is 0^o the 'top' surface is pointing upward and has the lowest heat transfer rate, whereas when the angle of inclination of the cylinder is 180^o where the top surface is pointing down all the surfaces that make up the cylinder become effective except the part of the side wall of the cylinder closer to the base which has less heat transfer rate and the interaction between the surfaces that make up the cylinder become important.

The variation of the mean Nusselt number for the top surface relative to the mean Nusselt numbers for the side surface was discussed above. The relative importance of the heat transfer rate from the top surface compared to that from the vertical side surfaces of the cylinder will, however, depend on both the mean Nusselt numbers for the surfaces and the relative surface areas. Now since:

$$A_{side} = \pi D_h, \quad A_{top} = \frac{\pi}{4} D_h^2, \quad \text{and} \quad A_{total} = \pi D_h + \frac{\pi}{4} D_h^2$$

Where A_{total}, A_{side}, and A_{Top} are the dimensionless surface areas of the entire cylinder, the 'side' of the cylinder, and of the 'top' surface of the cylinder, it follows that:

$$\frac{A_{side}}{A_{total}} = \frac{4}{4+D_h}, \quad \frac{A_{Top}}{A_{total}} = \frac{D_h}{4+D_h}, \quad \frac{A_{Top}}{A_{side}} = \frac{D_h}{4} \qquad (7)$$

This equation indicates that over the range of values of D_h considered here, i.e., 0.25 to 1, A_{Top} / A_{side} is relatively small having a maximum value of 0.25 when $D_h = 1$, i.e., the area of the top surface relative to the area of the vertical side surfaces remains comparatively small.

As the angle of inclination increases from 0^o, the interaction between the flows over the surfaces that make up the cylinder becomes significant and the effects of the top surface of the cylinder become more dominant and cannot be neglected and portion of the side surface becoming less important.

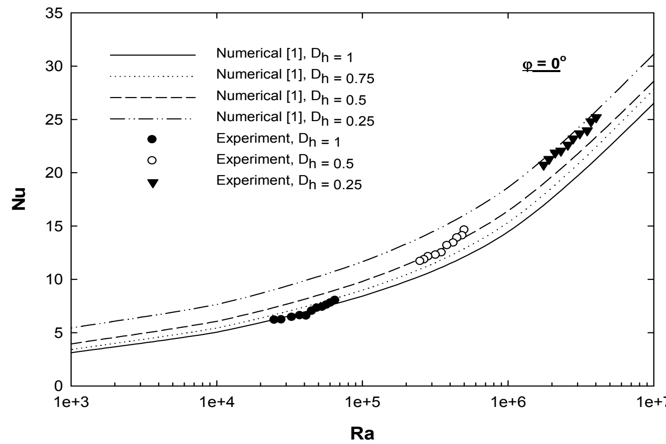

Figure 5 Variation of mean Nusselt number for the cylinder with Rayleigh number for various values of dimensionless cylinder width, D_h, when $\varphi=0^o$

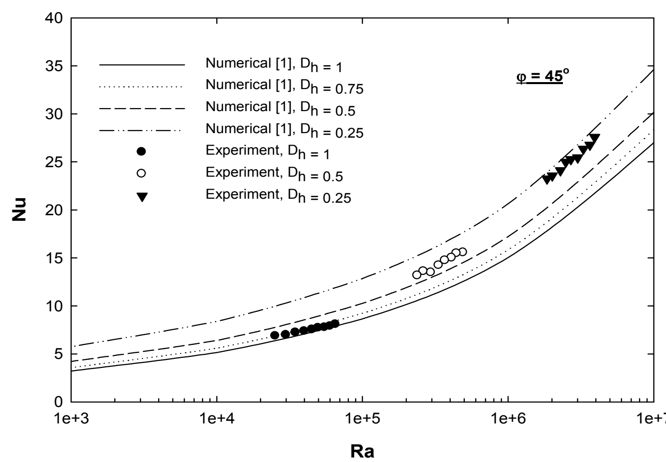

Figure 6 Variation of mean Nusselt number for the cylinder with Rayleigh number for various values of dimensionless cylinder width, D_h, when $\varphi=45^o$

Now both correlation equations for the case of isothermal vertical cylinder, i.e., for $\varphi=0^o$, when the heat transfer rate from the top surface is neglected and having isothermal surface temperature and for the case of a horizontal cylinder, i.e. for $\varphi=90^o$, for a Prandtl number of 0.7 have the form:

$$Nu_0 = \mathrm{B}\ Ra^{0.25} \qquad (8)$$

The parameter B depends only upon the Prandtl number of the fluid involved. When the diameter of the cylinder becomes relatively small and inclined at an angle to the vertical the curvature and three-dimensional effects become important, e.g. see [1, 10], and the Nusselt number depends upon the boundary layer thickness to cylinder diameter ratio, i.e. since the boundary layer thickness depends on the value of $h/DRa^{0.25}$, is dependent on:

$$\varepsilon = \frac{h}{DRa^{0.25}} = \frac{1}{D_h Ra^{0.25}} \qquad (9)$$

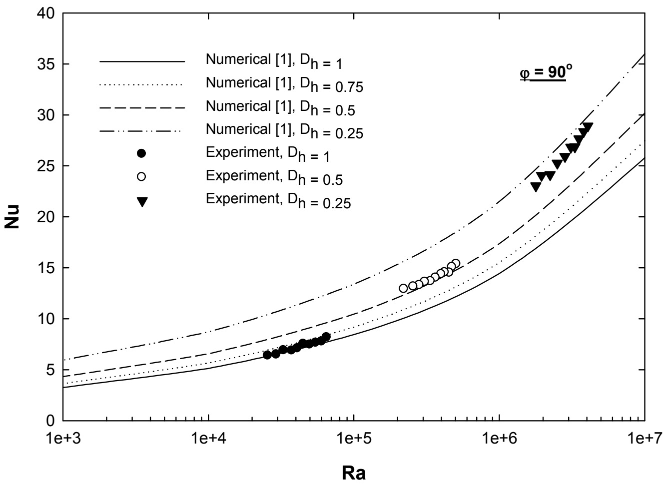

Figure 7 Variation of mean Nusselt number for the cylinder with Rayleigh number for various values of dimensionless cylinder width, D_h, when $\varphi=90^o$

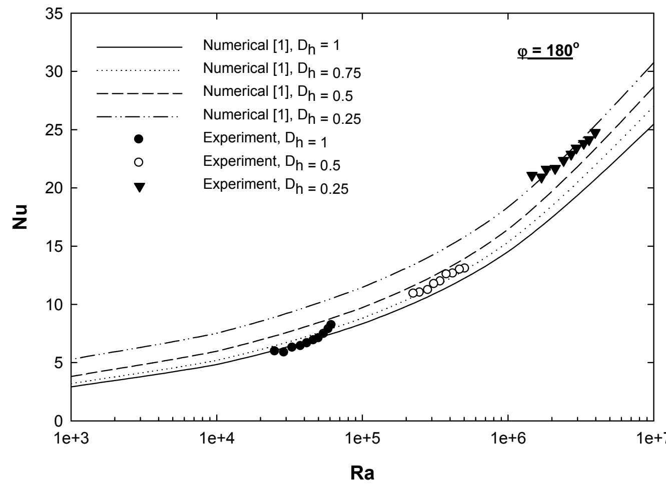

Figure 9 Variation of mean Nusselt number for the cylinder with Rayleigh number for various values of dimensionless cylinder width, D_h, when $\varphi=180^o$

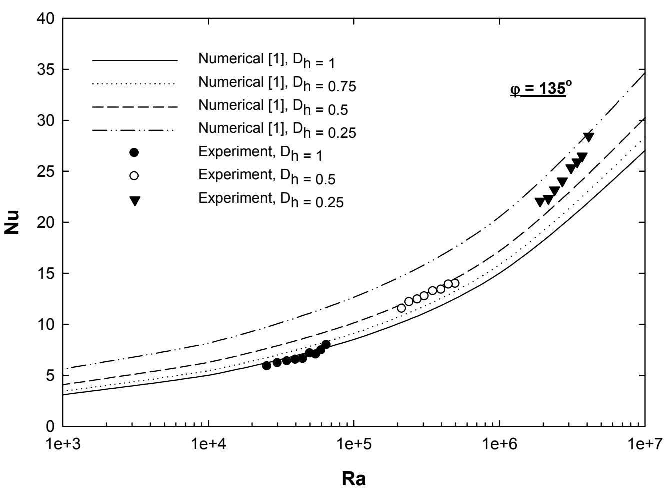

Figure 8 Variation of mean Nusselt number for the cylinder with Rayleigh number for various values of dimensionless cylinder width, D_h, when $\varphi=135^o$

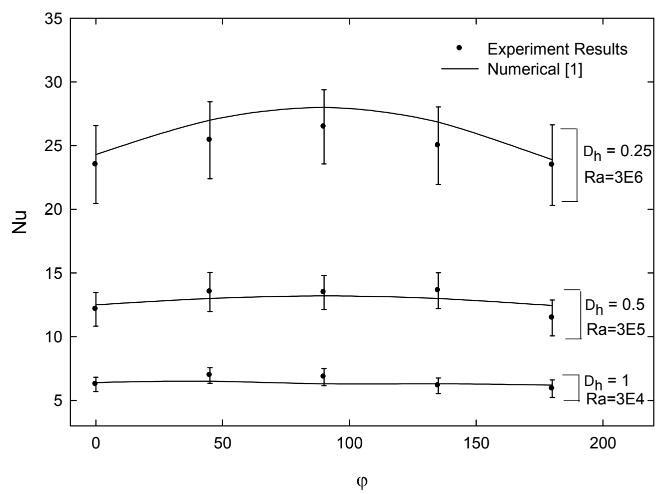

Figure 10 Variation of mean Nusselt number for the cylinder with φ for various values of Rayleigh number and dimensionless cylinder width, D_h

The Nusselt number will of course also be dependent on the inclination angle. Therefore when the cylinder diameter is relatively small and the cylinder is inclined at an angle to the vertical, the mean Nusselt number from the cylinder is given by an equation of the form:

$$\frac{Nu}{Ra^{0.25}} = \text{constant} + \text{function}\left(Ra, D_h, \varphi\right) \tag{10}$$

Using an equation of this form it has been found that the present experimental and previous numerical [1] results for the case of isothermal heated cylinder with an exposed top surfaces inclined at an angle to the vertical, φ, of between $\varphi=0^o$ and $\varphi=180^o$, with different dimensionless diameters can be approximately described by:

$$\frac{Nu_{mcemp}}{Ra^{(0.284+0.005\sin\varphi)}} = 0.2 + \frac{0.63}{\left(D_h Ra^{0.25}\right)^{0.59}} \tag{11}$$

A comparison of the results given by the above correlation equation for a wide range of Rayleigh number with the experimental results is shown in Figure 11 from which it will be seen that this correlation equation describes the numerical results to an accuracy of better than 95% as mentioned in [1]. Figure 12 shows the comparison of the experimental results with the above correlation equation for Rayleigh number close to those used in the experimental, the agreement being to an accuracy of better than 90%.

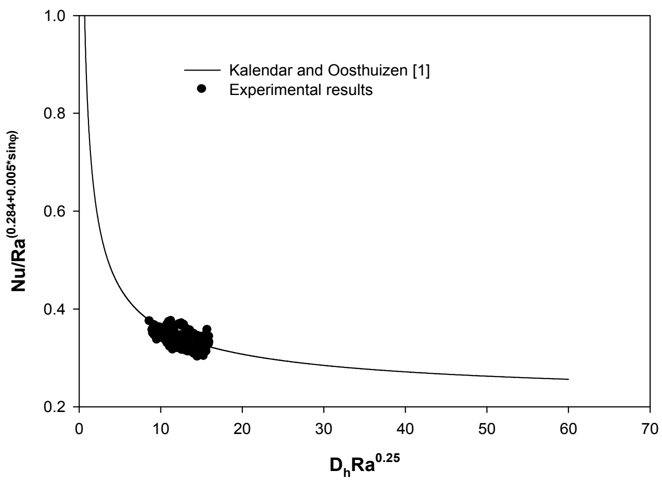

Figure 11 Comparison of correlation equation with the numerical and experimental results for inclined Cylinder

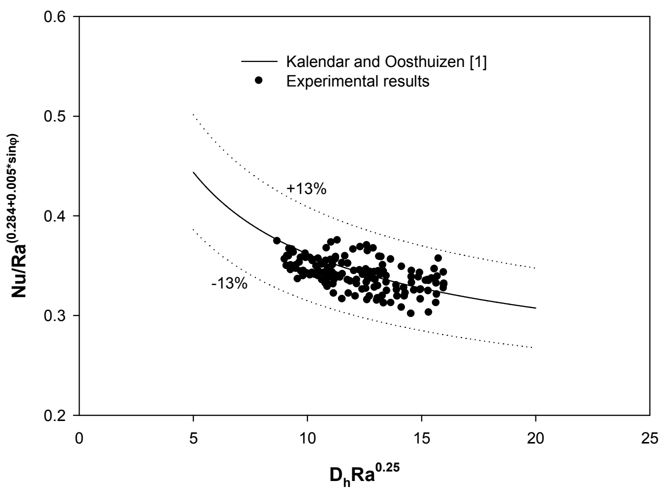

Figure 12 Comparison of correlation equation with the experimental results for inclined cylinder with exposed top surface

CONCLUSION

The results of the present study indicate that:

1. The mean Nusselt number for the cylinder increases with decreasing D_h under all conditions considered.
2. At lower values of Ra (approximately less than 10^4) for all dimensionless cylinder diameter, D_h, considered the mean Nusselt number is independent of angle of inclination φ. At larger values of Ra the dependence of the mean Nusselt number on the angle of inclination φ becomes significant.
3. Experimental and numerical results for an isothermal cylinder with exposed top surface and inclined at an angle to the vertical are overall in a good agreement and the results are well correlated by equation (11).

ACKNOWLEDGMENTS

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