

NATURAL CONVECTION HEAT TRANSFER FROM INCLINED CYLINDERS

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(Received 8 February 1981 and in final form 13 April 1981)

Abstract—Experiments were carried out to determine the average and the local heat transfer by natural convection from the outside surface of isothermal cylinders of different diameters and lengths at different inclinations in both the laminar and the turbulent regions. The heat transfer was found to depend on both the diameter and inclination and general equations representing this are suggested.

NOMENCLATURE

- b , constant;
- c , constant;
- D , outer diameter of cylinder [mm];
- h , heat transfer coefficient [$W/m^2 \text{ } ^\circ C$];
- L , cylinder length [m];
- m , constant;
- n , constant;
- p , constant;
- t , temperature [$^\circ C$];
- x , distance from beginning of cylinder [m];
- Gr , Grashof number;
- Nu , Nusselt number;
- Pr , Prandtl number;
- β , function of Gr_D in equation 1;
- θ , angle of inclination of cylinder (from the vertical position).

Subscripts

- a , ambient;
- cr , critical;
- D , based on diameter;
- f , film;
- L , average (for a cylinder length = L);
- Lim , limit;
- s , surface;
- x , local (at a cylinder length = x).

INTRODUCTION

NATURAL convection from the outside surface of cylinders is employed for heating and cooling in many industrial processes. Data are available for horizontal and vertical cylinders. As noted by Farber and Rennat [1], however, no data were available until 1957 to enable the calculation of an inclined-cylinder cooling system. In fact the investigations carried out to study this condition are, even now, very limited. Moreover, only in a few cases (of the vertical cylinder) was the effect of diameter systematically investigated.

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The present research work was carried out in an attempt to fill a part of the existing gap by experimentally studying the effect of diameter and inclination on natural convection heat transfer from the outside surface of isothermal cylinders to air in both the laminar and the turbulent conditions.

PREVIOUS WORK

Effect of diameter

In all the experiments previously carried out to investigate the effect of diameter only vertical cylinders were employed.

Eigenson [2, 3] carried out experiments on cylinders in gases in the turbulent condition only. The following equation was suggested for the average heat transfer:

$$Nu_L = \beta (Gr_L Pr)^{1.3} \quad (1)$$

where β is a function of Gr_D .

Carne [4] carried out experiments on steam-heated cylinders in air using diameters between 5 and 76 mm and lengths from 0.076 to 3.30 m. For the same length the average heat transfer was found to decrease with diameter.

Elenbaas [5] derived the following theoretical equation for average laminar natural convection from the outside surface of vertical cylinders:

$$Nu_D \cdot \exp \left[-\frac{2}{Nu_D} \right] = 0.6 \left(\frac{D}{L} \cdot Gr_D \cdot Pr \right)^{1.4} \quad (2)$$

A difference of up to 15% was found between the theoretical and the experimental heat transfer.

Ede [6] compared the results of different observers with equation (2) and found that the data for water were some 25% higher and that the data for gases were within $\pm 20\%$.

Nagendra *et al.* [7] theoretically studied the effect of diameter on natural convection from vertical cylinders and wires. Three regions were distinguished for which the terms, wires, long cylinders and short cylinders (or flat plates) were given. The following equation represented the heat transfer:

$$Nu_D = b \left(Gr_D Pr \frac{D}{L} \right)^c \quad \dots \quad (3)$$

with b and c as shown for isothermal cylinders:

	b	c
wires	0.87	0.05
long cylinders	1.3	0.16
short cylinders	0.57	0.25

Experiments carried out employing a tube of 0.315 in (8 mm) outside diameter and 12 in (0.305 m) length in water gave results which agree with the theoretical equation (for long cylinders).

Effect of inclination

To the authors' knowledge only three investigations are available so far in this field.

Farber and Rennat [1] experimented with a 6 ft (1.829 m) long and 0.125 in (3.175 mm) outside dia. cylinder heated by passing an electric current through it to give a constant heat flux at angles of inclination from 0 to 90° (from the vertical). The heat transfer coefficient was found to increase with the angle of inclination. No general correlation of the results was made.

Oosthuizen [8] experimented with cylinders of lengths between 152.4 and 304.8 mm and outside dia. between 19.1 and 25.4 mm at angles of inclination from 0 to 90°. The heat transfer was determined by measuring the rate of cooling of the cylinders from 100 to 90°C. The average heat transfer was found to increase with inclination and the results could be correlated in terms of $Nu_D/(Gr_D \sin \theta)^{1/4}$ against $(L/D) \tan \theta$.

Al-Arabi and Salman [9] carried out experiments on a cylinder of 38 mm outside dia. and 0.95 m length in the constant heat flux condition. The angle of inclination varied between 0 and 90°. For the same heat flux the heat transfer was found to increase with inclination.

APPARATUS

The apparatus used is shown diagrammatically in Fig. 1. It consists essentially of a brass tube (a) which forms the experimental cylinder. Steam condensing at atmospheric pressure is used to heat the inside surface. Two thin brass discs (b), soldered to the cylinder, form the ends of the steam space. The upper disc is provided with a vent hole (c) to ensure atmospheric pressure inside the cylinder and the lower disc is provided with a syphon (d). The syphon, which traps the steam, enables the condensate to flow out where it is collected in a receiver (e). The receiver is provided with an aluminium cover on the lower surface of which any re-evaporated condensate is recondensed.

Two gypsum circular cylinders (f) of the same diameter as the experimental cylinder are fitted to the upper and lower end-discs so that the outside surface of the gypsum forms a continuation of the brass surface. Each gypsum piece is 120 mm long, a length sufficient to enable free flow of convection currents. To ensure proper and permanent contact between the

gypsum and the cylinder end-discs the gypsum pieces are cast in-place and reinforced with wires (g) soldered to the discs. Wire hooks (h) inserted in the gypsum enable the experimental cylinder to be carried by means of strings (i); the other ends of which are fixed to an overhead beam (j). The cylinder can be set at any angle by adjusting the lengths of the strings as required.

To determine the heat loss by conduction from the cylinder end-discs to the gypsum pieces two thermocouples are inserted centrally in each piece. From the thermal conductivity of gypsum and the distance between the thermocouples the end conduction could be calculated. This loss, as compared with the total heat transfer from the cylinder, varied between 3% for the shortest cylinder and 1% for the longest cylinder.

Eleven cylinder lengths (0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9, 1.05, 1.35, 1.65 and 2.0 m) and six diameters (12.75, 19.3, 25, 32, 38.25 and 51 mm) at five inclinations from the vertical position (0, 15, 30, 45 and 60°) were used.

To make sure that the steam entered the cylinder perfectly dry it was superheated before entering the cylinder in an electric superheater. The amount of superheat at entrance was kept at about 7°C. The error in the measured heat transfer caused by this does not exceed 0.3%.

The outside surface of the cylinder was nickel-electro-plated and polished to minimize radiation. An emissivity determined experimentally in the actual conditions employed was used. Two experiments were carried out employing one of the cylinders. In the first experiment the cylinder surface was polished and in the second it was blackened by carbon deposited from an acetylene flame. The emissivity was found to be 0.079. The radiation loss, as compared with the total heat transfer, varied between 8 and 11%.

Temperature measurement

As the heat transfer coefficient from the condensing steam to the inside surface of the cylinder is very high compared to that from the outside surface to the air, the resistance to heat flow on the steam side could be neglected and the inside surface temperature could be considered equal to the steam saturation temperature corresponding to the atmospheric pressure as registered by the barometer at the time of the experiment.

The temperature of the outside surface was determined by calculating the temperature drop in the metal of the cylinder from the conductivity of brass. In fact this could be neglected as it did not exceed 0.01°C.

The amount of superheat and the ambient air temperature were measured by calibrated mercury-in-glass thermometers graduated in 0.1°C. A 0.01 mV potentiometer was used to take the readings of the thermocouples.

RESULTS AND DISCUSSION

Average heat transfer

Variation of average heat transfer with cylinder diameter. Figure 2 shows the average heat transfer

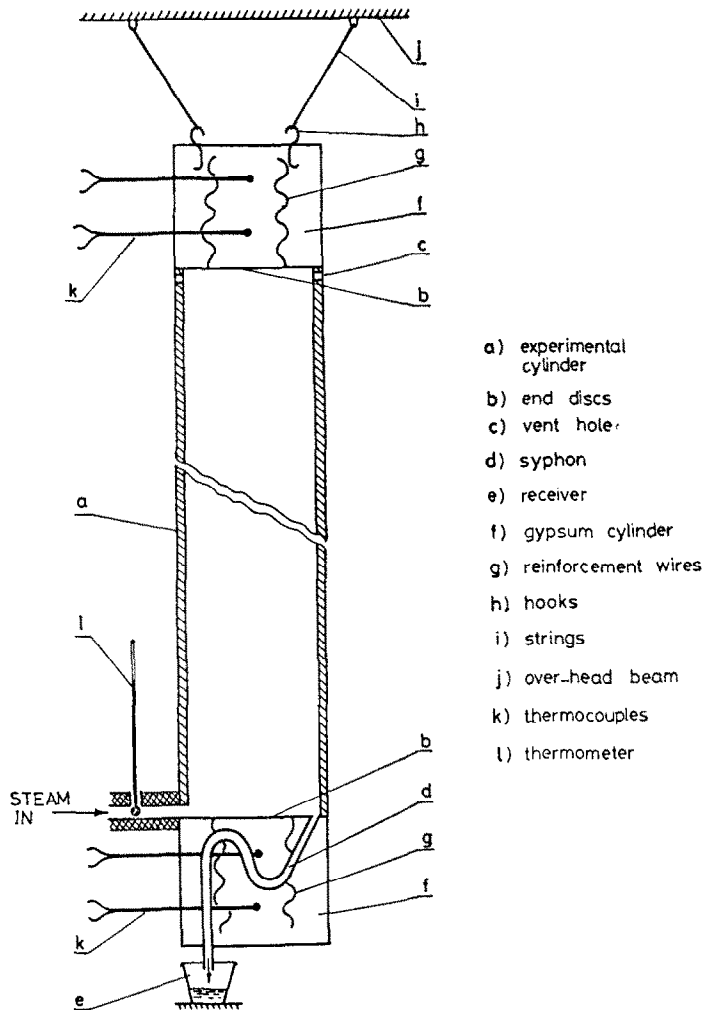


FIG. 1. Apparatus.

coefficient, h_L , plotted against cylinder diameter D for an inclination angle $\theta = 0^\circ$ (the vertical position) and different cylinder lengths. The other inclination angles give the same general characteristics. It will be seen that for the same L , h_L decreases with the increase of D . For a value of D equal to infinity the vertical cylinder should give the same h_L as a vertical plate. The heat transfer coefficients corresponding to vertical plates of the same lengths in the same conditions as calculated from the equations recommended by McAdams [10] are also shown in Fig. 2.

Variation of average heat transfer with angle of inclination. Figure 3 shows the average heat transfer coefficient, h_L , plotted against the angle of inclination θ for a cylinder dia. = 19.3 mm and different cylinder lengths. The other diameters give the same general characteristics. It will be seen that h_L for the longer lengths increases with the increase of θ . The rate of increase decreases with the decrease of L . For the shorter lengths, however, h_L decreases with the increase of θ . A certain value of L , (hereinafter called the limit length L_{lim}), at which h_L is constant

irrespective of θ , must therefore exist. The heat transfer coefficient corresponding to this length is hereinafter called the limit coefficient $h_{L,lim}$.

Extrapolated to $\theta = 90^\circ$ (the horizontal position), all the curves, as can be seen, meet at the same point A . All the cylinders (with the end losses allowed for) give, as they should, the same h_L in the horizontal position irrespective of L . The $h_L-\theta$ line corresponding to the limit length must pass through A . The heat transfer coefficient corresponding to the limit length must therefore be the same as that for the horizontal cylinder.

Variation of average heat transfer with cylinder length. The same data of Fig. 3 are plotted in Fig. 4 to show the variation of h_L with L at different values of θ .

Figure 4 clearly shows the limit length phenomena. A cylinder of this length gives the same value of h_L at any θ , as given by the point of intersection of the different curves (point A). The dashed line passing through A represents h_L for the horizontal cylinder.

Figure 4 also shows that h_L for any θ decreases gradually with cylinder length until it becomes practi-

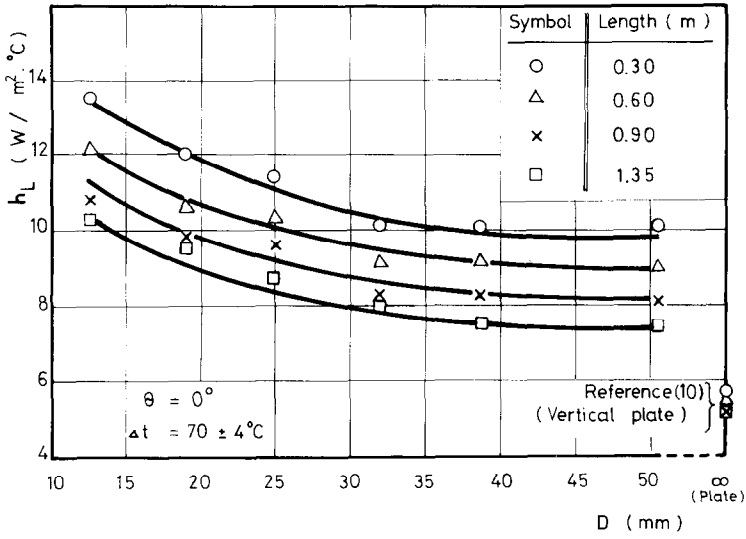


FIG. 2. Variation of average heat transfer coefficient with cylinder diameter.

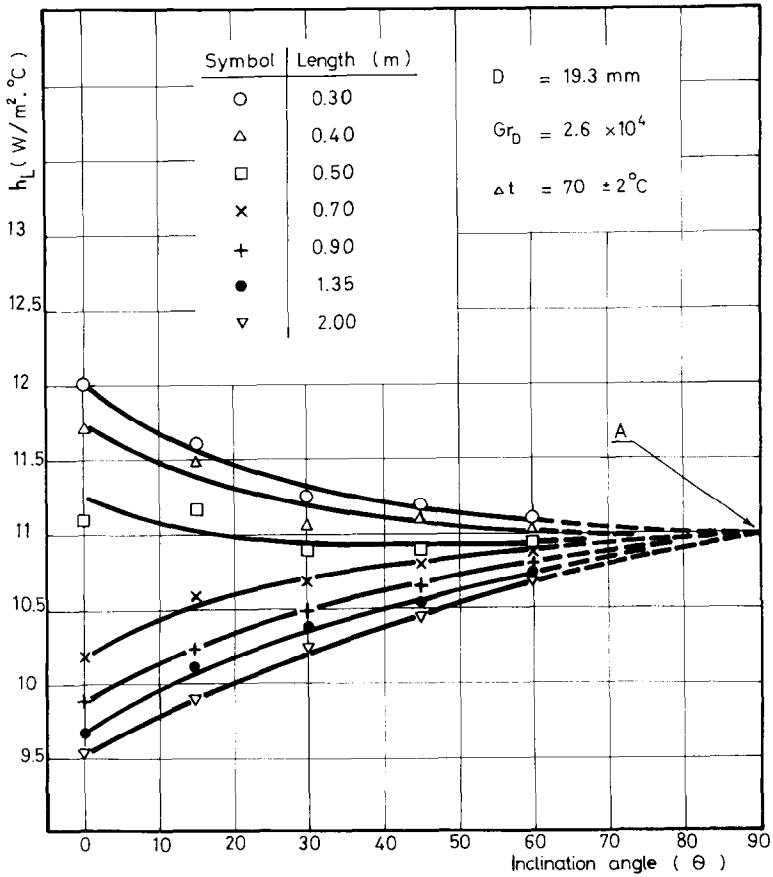


FIG. 3. Variation of average heat transfer coefficient with angle of inclination.

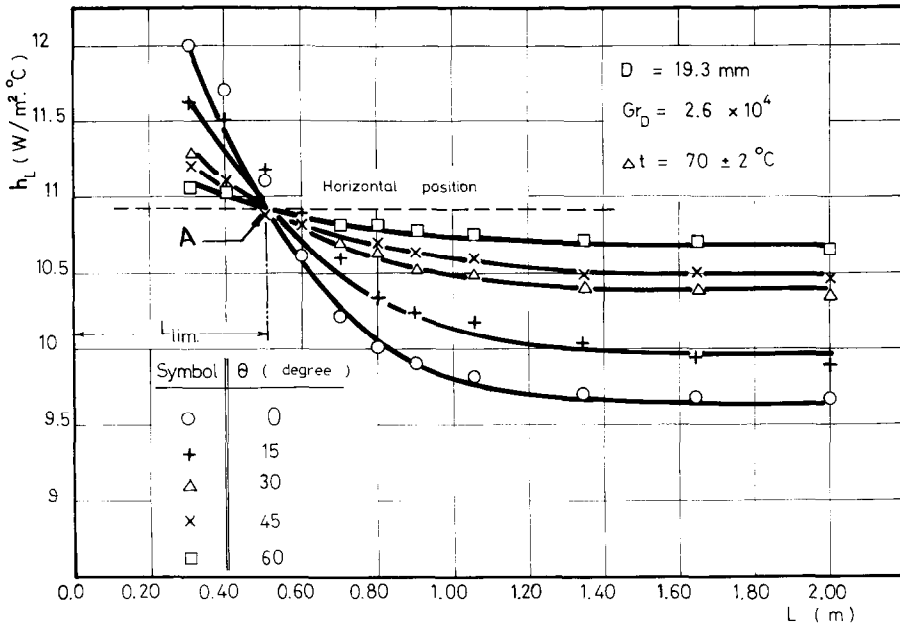


FIG. 4. Variation of average heat transfer coefficient with cylinder length.

cally constant (independent of L) indicating the beginning of turbulence.

The limit length phenomena has widened the scope of the present work. Thus, although the experiments were carried out at angles of inclination from 0 to 60°, the data for the horizontal cylinder ($\theta = 90^\circ$) are also obtainable from the experiments.

Correlation of average heat transfer data. Figure 5 shows $\log Nu_L$ plotted against $\log (Gr_L Pr)$ for a cylinder dia. = 19.3 mm. In the calculation of the dimensionless numbers, the physical properties were taken at the mean film temperature $t_f = (1/2)(t_s + t_a)$. It will be seen that for each angle of inclination θ the results are represented by two straight lines one with a slope increasing with θ followed by another with a slope equal to 1/3 irrespective of θ . The two lines indicate the laminar range in which h_L depends on cylinder length, and the turbulent range in which h_L is independent of length. All the other diameters show the same general characteristics.

The value of $(Gr_L Pr)_{Lim}$ corresponding to L_{Lim} is also shown in Fig. 5 (point A). A line HH with a slope = 1/3 passing through point A represents the horizontal cylinder results.

Critical transition point. No clear transition region appears in Fig. 5. The laminar and the turbulent lines meet at point C. The results of the different diameters show that the value of the critical $Gr_L Pr$ (corresponding to point C) depends on inclination angle only and is independent of diameter as shown in Table 1.

The variation of the critical transition point with θ can be represented by equation (4):

$$(Gr_L Pr)_{cr} = 2.6 \times 10^9 + 1.1 \times 10^9 \tan \theta. \dots \quad (4)$$

In the horizontal position equation (4) gives a value of $(Gr_L Pr)_{cr}$ equal to infinity which is indeed what it should be since the heat transfer along the whole cylinder is laminar and no critical point exists. In the vertical position equation (4) gives a value = 2.7×10^9 . Other observers obtained values which vary between 1×10^9 and 3×10^9 .

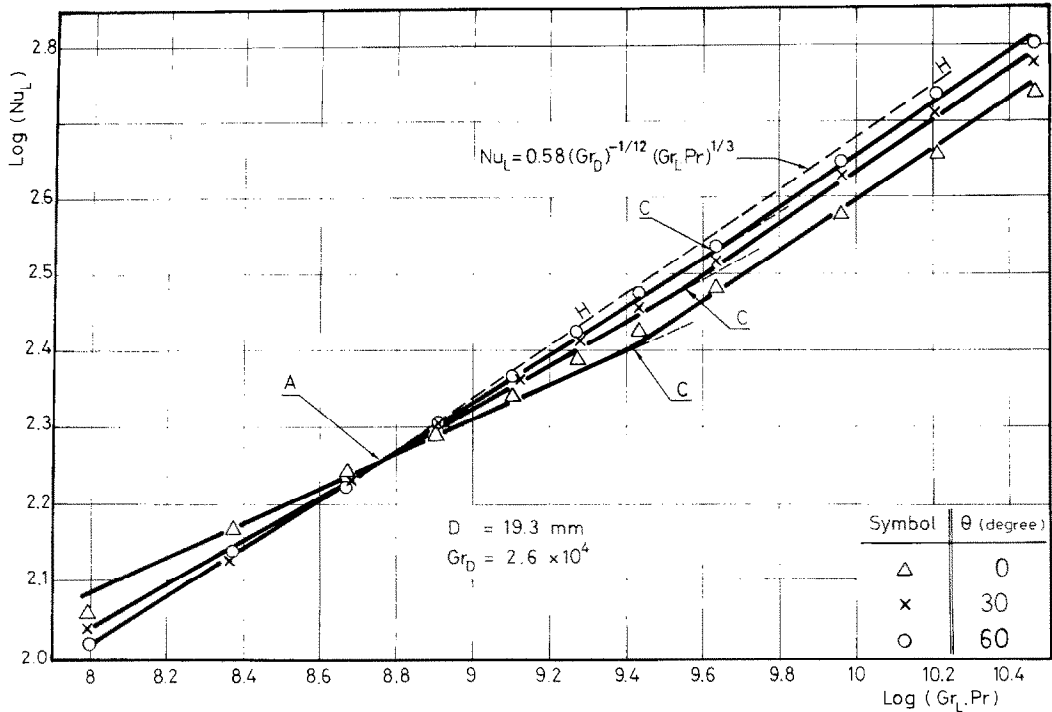
Laminar region. To obtain a correlation observers experimenting with plates suggest that a modified Grashof number equal to $(Gr_L \cos \theta)$ be used instead of Gr_L . This method has been attempted in the present work and has resulted in reasonable correlations for values of θ between 0 and 60°. However the use of $\cos \theta$ gives a modified Grashof number equal to zero in the horizontal position ($\theta = 90^\circ$). This method is unsuitable for a general correlation for all angles of inclination and has, therefore, been abandoned.

Figure 5 shows that the relation between Nu_L and $Gr_L Pr$ for any θ in the laminar region can be represented by:

$$Nu_L = m(Gr_L Pr)^n. \quad (5)$$

The calculated values of m for the different diameters and inclinations were found to vary with both diameter and inclination. The values of n , however, were independent of diameter and varied with inclination only.

In order to express the dependence of m (for the same θ) on D in a dimensionless form the ratio L/D was employed by some observers in the case of the vertical cylinder. As can be seen from Fig. 5 and equation (5), however, m is constant for the same diameter irrespective of L/D . The use of L/D cannot, therefore, represent

FIG. 5. Variation of Nu_L with $Gr_L Pr$.Table 1. Variation of $(Gr_L Pr)_{cr}$ with θ

θ	$(Gr_L Pr)_{cr}$
0° (vertical)	2.7×10^9
15°	2.8×10^9
30°	3.35×10^9
45°	4.05×10^9
60°	4.7×10^9
90° (horizontal)	infinity

the effect of diameter. In the following correlations a Grashof number based on diameter (Gr_D) will be used. For the same θ and the same D the value of Gr_D is constant irrespective of cylinder length.

The following equations were found to represent the variation of m and that of n :

$$m = [2.9 - 2.32 (\sin \theta)^{0.8}] (Gr_D)^{-1.12}, \quad (6)$$

$$n = \frac{1}{4} + \frac{1}{12} (\sin \theta)^{1.2}. \quad (7)$$

Inserting the values of m and n into equation (5), equation (8) is obtained:

$$Nu_L = [2.9 - 2.32 (\sin \theta)^{0.8}] (Gr_D)^{-1.12} \times [Gr_L Pr]^{1/4 + 1/12 (\sin \theta)^{1.2}} \quad (8)$$

Equation (8) represents the general equation for laminar natural convection in the range of the present work [$1.08 \times 10^4 \leq Gr_D \leq 6.9 \times 10^5$ and 9.88×10^7

$\leq Gr_L Pr \leq (Gr_L Pr)_{cr}$ (as taken from equation 4)]. It is valid for the whole range of θ from 0 to 90°.

The experimental points are compared with equation (8) in Fig. 6. The horizontal cylinder points ($\theta = 90^\circ$) are the limit length points as taken from the "log Nu_L - log $Gr_L Pr$ " curves for the different diameters. As can be seen equation (8) represents the results with a maximum deviation of $\pm 8\%$.

For the vertical cylinder ($\theta = 0^\circ$) equation (8) reduces to

$$Nu_L = 2.9 Gr_D^{-1.12} (Gr_L Pr)^{1.4}. \quad (9)$$

For the horizontal cylinder ($\theta = 90^\circ$) equation (8) reduces to

$$Nu_L = 0.58 Gr_D^{-1.12} (Gr_L Pr)^{1.3}, \quad (10)$$

which is represented by the dashed line HH in Fig. 5. For air with $Pr = 0.7$ equation (10) becomes

$$Nu_D = 0.566 (Gr_D Pr)^{1.4}. \quad (11)$$

Equation (11) is of the same form as that suggested by other observers for the horizontal cylinder. The constant as given by McAdams [10] is 0.53.

Limit length. Figure 5 shows that the value of $Gr_L Pr$ corresponding to the limit length is about 5.5×10^8 . The other diameters give almost the same value. $(Gr_L Pr)_{lim}$ is, therefore, constant irrespective of D . This in fact can be seen from equation (9) and equation (10) for the vertical and the horizontal cylinders respectively. For the vertical cylinder h_{lim} is proportional to $(L_{lim}^{-1.4} D^{-1.4})$. For the horizontal cylinder h_{lim} is proportional to $D^{-1.4}$. Since h_{lim} is the same for

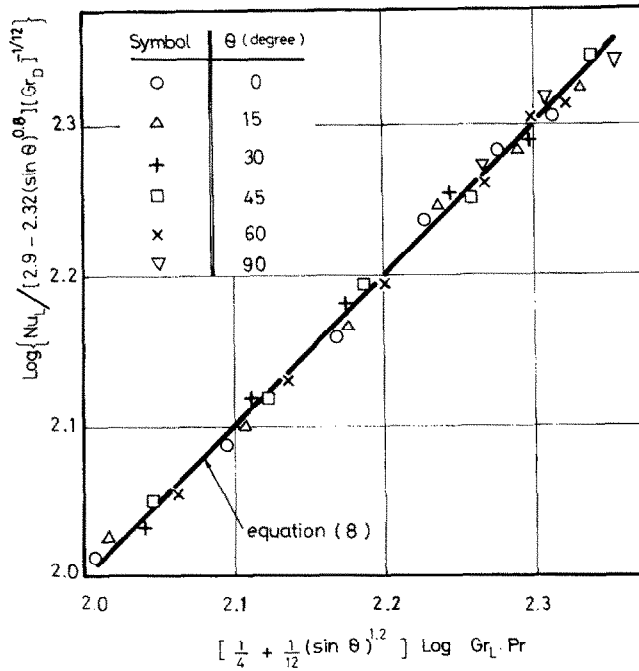


FIG. 6. General correlation of average heat transfer results in the laminar region.

both cylinders it can be seen that this can be true only if L_{lim} is constant irrespective of the value of D giving a constant $(Gr_L Pr)_{Lim}$.

Turbulent region. Figure 5 shows that the straight lines corresponding to the turbulent region are parallel with a slope equal to $1/3$ for all angles of inclination. They can be represented by

$$Nu_L = p(Gr_L Pr)^{1/3}. \quad (12)$$

The values of p , as calculated from the curves for the different diameters and inclinations, were found to vary with both D and θ . The results could be represented by

$$Nu_L = [0.47 + 0.11 (\sin \theta)^{0.8}] \times (Gr_D)^{-1/12} (Gr_L Pr)^{1/3}. \quad (13)$$

Equation (13) represents the general equation for average natural convection in the turbulent region in the range of the present work [$1.08 \times 10^4 \leq Gr_D \leq 6.9 \times 10^5$ and $(Gr_L Pr)_{cr} \leq Gr_L Pr \leq 2.95 \times 10^{10}$ where $(Gr_L Pr)_{cr}$ is to be taken from equation (4)]. It is valid for the whole range of θ from 0 to 90° .

The experimental points are compared with equation (13) in Fig. 7. As can be seen the maximum deviation is $\pm 6\%$.

For the vertical cylinder ($\theta = 0^\circ$) equation (13) reduces to

$$Nu_L = 0.47 (Gr_D)^{-1/12} (Gr_L Pr)^{1/3}. \quad (14)$$

For the horizontal cylinder ($\theta = 90^\circ$) equation (13) reduces to

$$Nu_L = 0.58 (Gr_D)^{-1/12} (Gr_L Pr)^{1/3}, \quad (15)$$

which for air with $Pr = 0.7$ can be written as

$$Nu_D = 0.566 (Gr_D Pr)^{1/4}. \quad (16)$$

Equation (16) is the same as equation (11) obtained for the horizontal cylinder from the general equation for the laminar region.

Local heat transfer

The difference between the heat transfer as obtained from two successive cylinder lengths (L_1 and L_2) of the same diameter at the same angle of inclination may be used to calculate the local heat transfer coefficient h_x at a value of $L = L_1 + \frac{1}{2}(L_2 - L_1)$ provided that $L_2 - L_1$ is sufficiently small. In the present experiments $L_2 - L_1$ was 100 mm (from $9D$ to $2D$) in the laminar region whereas it was 150–300 mm in the turbulent region in which the heat transfer is known to be constant.

Variation of local heat transfer with cylinder length. Figure 8 shows a plot of the local heat transfer coefficient h_x against x for one of the diameters at different inclinations. The curves corresponding to the other diameters have the same general characteristics. A laminar region is followed by a transition region which ends with a turbulent region. The difference between h_x at the beginning and the end of the transition region decreases with the increase of θ until, at $\theta = 90^\circ$ (the horizontal position), h_x becomes constant irrespective of x .

Correlation of local heat transfer data. Plotting $\log Nu_x$ against $\log (Gr_x Pr)$ for the same diameter and angle of inclination the results lie on two straight lines one representing the laminar region with a slope

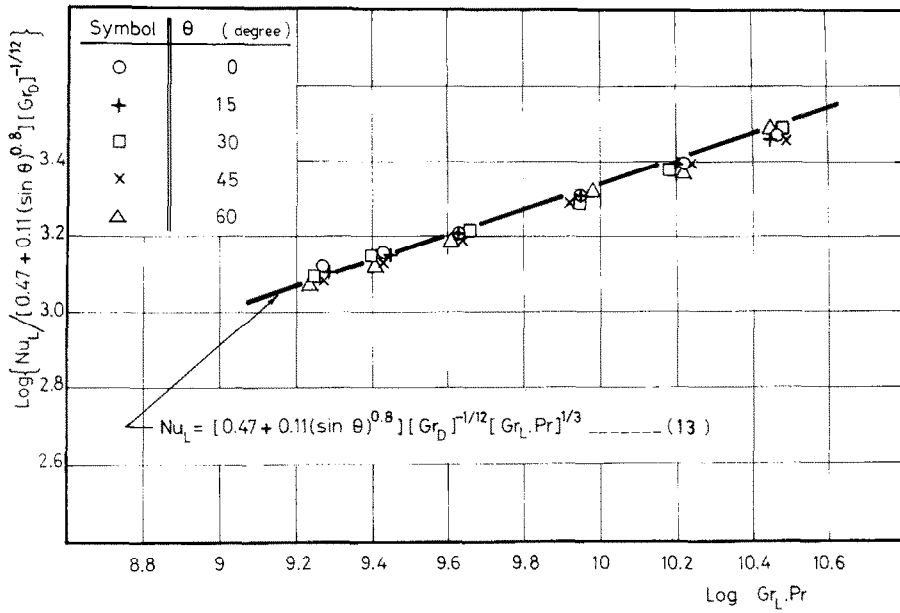


FIG. 7. General correlation of average heat transfer results in the turbulent region.

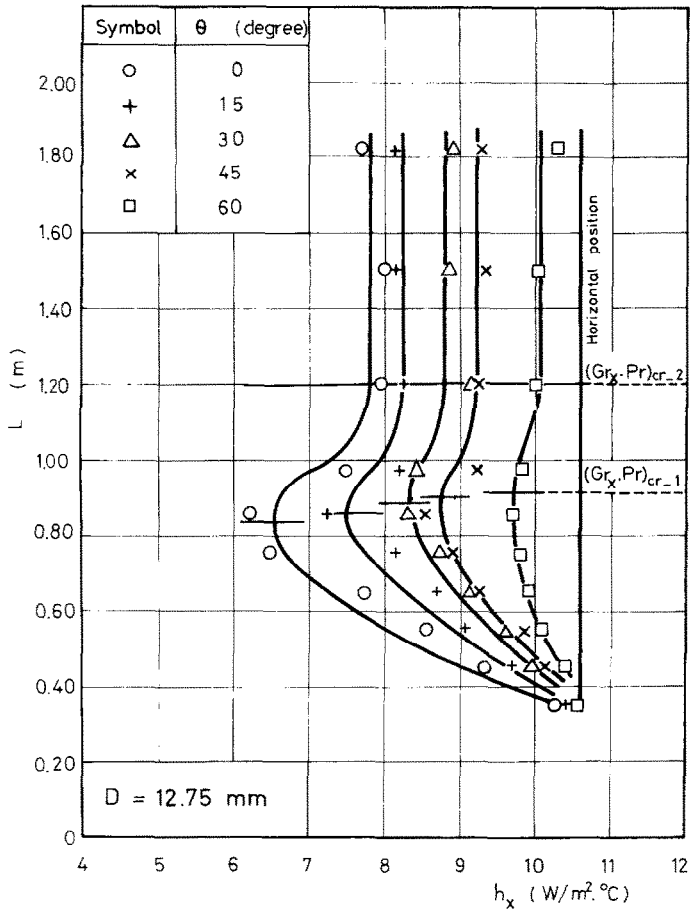


FIG. 8. Variation of local heat transfer coefficient with cylinder length.

increasing with θ , and the other representing the turbulent region with a slope = 1/3 irrespective of θ .

The experimental results could be represented by the following equations:

$$Nu_x = [2.3 - 1.72 (\sin \theta)^{0.8}] \times (Gr_D)^{-1/12} [Gr_x Pr]^{1/4 + 1/12 (\sin \theta)^2}, \quad (17)$$

$$Nu_x = [0.42 + 0.16 (\sin \theta)^{0.8}] \times (Gr_D)^{-1/12} (Gr_x Pr)^{1/3}. \quad (18)$$

Equation (17) represents the general equation for local natural convection in the laminar region in the range of the present work [$1.08 \times 10^4 \leq Gr_D \leq 6.9 \times$

10^5 and $1.63 \times 10^8 \leq Gr_x Pr \leq (Gr_x Pr)_{cr-1}$]. The equation is valid for the whole range of θ from 0 to 90°. As can be seen from the comparison shown in Fig. 9, equation (17) represents the experimental results with a maximum deviation of $\pm 8\%$.

Equation 18 represents the general equation for local natural convection in the turbulent region in the range of the present work [$1.08 \times 10^4 \leq Gr_D \leq 6.9 \times 10^5$ and $(Gr_x Pr)_{cr-2} \leq Gr_x Pr \leq 2.3 \times 10^{10}$]. It is valid for the whole range of θ from 0 to 90°. As can be seen from the comparison shown in Fig. 10 the experimental data are represented by equation (18) with a maximum deviation of $\pm 7\%$.

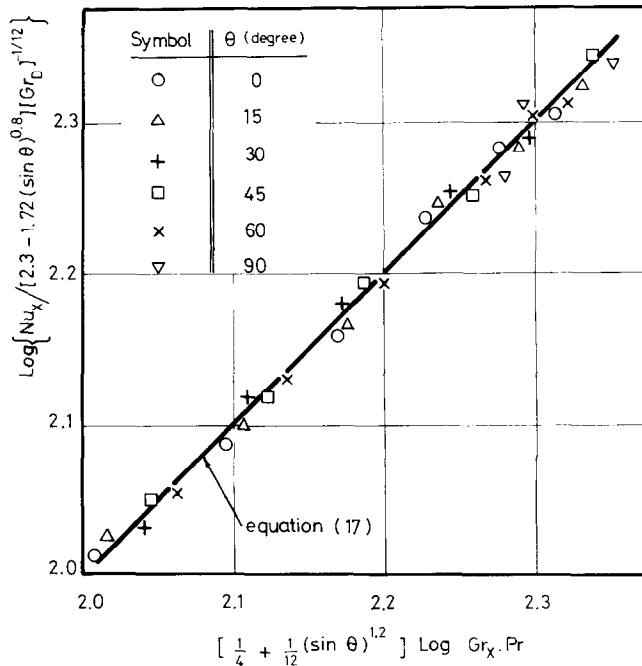


FIG. 9. General correlation of local heat transfer results in the laminar region.

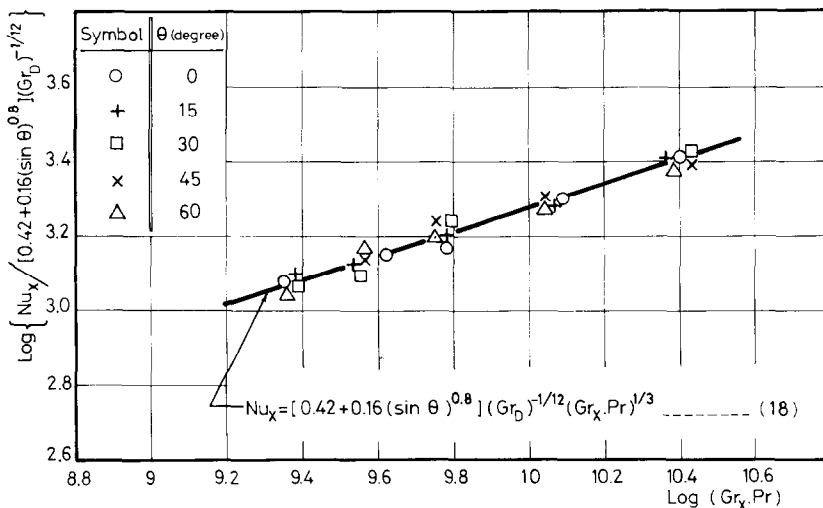


FIG. 10. General correlation of local heat transfer results in the turbulent region.

$(Gr_x Pr)_{cr-1}$ represents the end of the laminar region. As can be seen from Fig. 8 it varies with θ . No variation with diameter could be seen. The following equation was found to represent the variation with θ :

$$(Gr_x Pr)_{cr-1} = 2.16 \times 10^9 + 0.283 \times 10^9 \tan \theta. \quad (19)$$

$(Gr_x Pr)_{cr-2}$ represents the beginning of the turbulent region. Its value appears to be the same for all diameters and angles of inclination. It is equal to about 4.4×10^9 .

COMPARISON WITH PREVIOUS WORK

Effect of diameter

Figure 11, which is due to Ede [6], shows a comparison between the equation of Elenbaas for the average heat transfer for a vertical cylinder in the laminar region (equation 2) and the data of other observers with the present results added. The present results, as can be seen, are only some 10% higher than those given by equation (2) and are in good agreement with previous data.

Eigenson's equation for the average heat transfer for a vertical cylinder in the turbulent region (equation 1) and equation (14) of the present work are of the same form with β in equation (1) corresponding to $0.47 Gr_D^{-1/2}$ in equation (14). The comparison shown in Fig. 12 shows that the heat transfer given by equation (14) is only some 5% higher than that given by equation (1).

A comparison between equations (8) and (13) of the present work and the average heat transfer for vertical cylinders obtained by other observers is shown in Fig. 13. Previous theoretical studies [7] show that the difference between natural convection heat transfer for vertical cylinders in the isothermal condition and that

for cylinders in the constant heat flux condition is only about 5%. Data obtained in both conditions are therefore employed in the comparison. The data of Eigenson [2, 3], Carne [4], Griffiths and Davis [11], Koch [12] and Kirpicheff [13] are as calculated by Ede [6]. Nagendra's data represent the heat transfer calculated from his theoretical equation (equation 3) for cylinders of the same dimensions and in the same conditions as the cylinders employed in the present work.

Figure 13 shows that whereas some of the previous data are in very good agreement with the present equations the deviation of the other data is within the accepted experimental error range.

Effect of inclination

The data of Farber and Rennat [1] were obtained using a cylinder of diameter equal to 3.2 mm which is much smaller than the diameters used in the present work. The temperatures employed were from 329 to 675°C which are also much higher than the temperatures used in the present work. It appears therefore, that no comparison with their data is possible. The trend of variation of h_x with x , however, is the same as that of the present results.

All the data of Oosthuizen [8] lie in the laminar region. His data for a cylinder dia. = 19.1 mm are shown in Fig. 14 with the present results for a dia. = 19.3 added. The present results are some 20% higher.

All the data of Al-Arabi and Salman [9] lie in the laminar region. The slope of the laminar region line in Fig. 5, as calculated from equation (7) for the different inclination angles, is almost the same as that obtained by Al-Arabi and Salman. However their data are some 20% lower than those given by equations (8) and (17) of the present work.

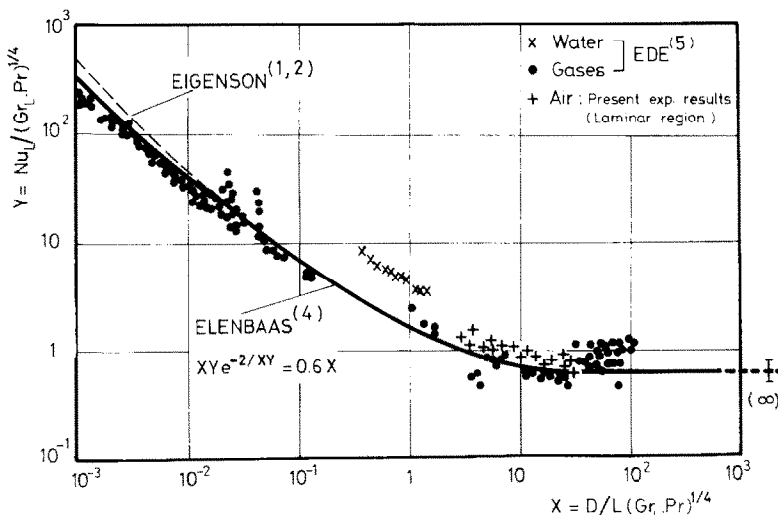


FIG. 11. Comparison between present experimental results and Elenbaas correlation.

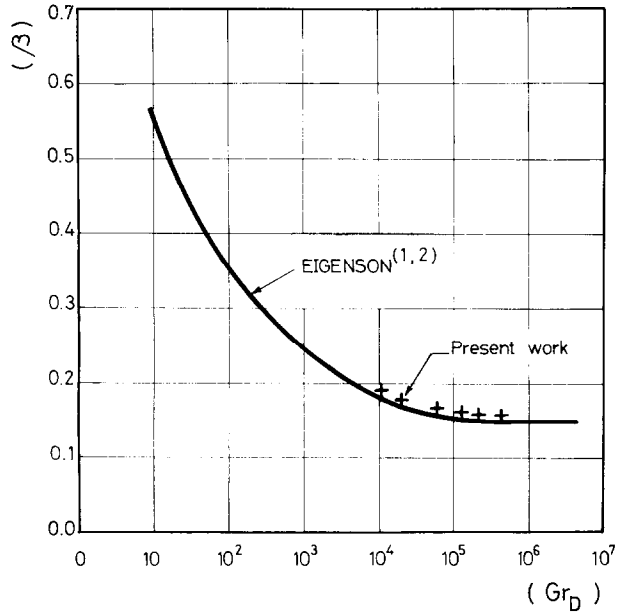


Fig. 12. Comparison between equations (1) and (14).

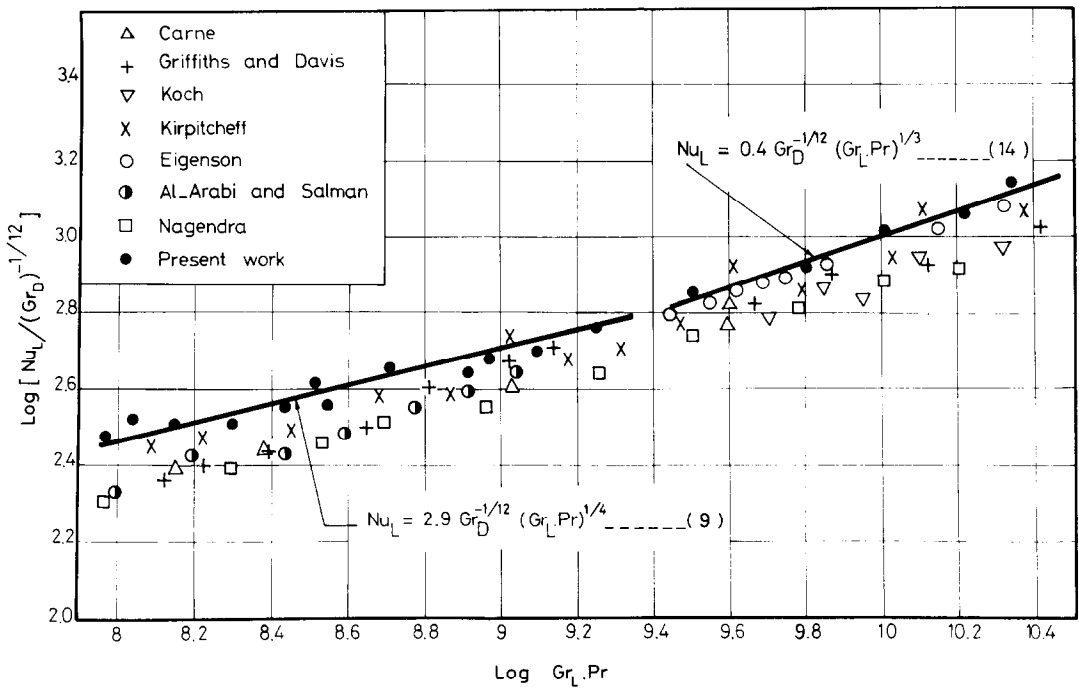


Fig. 13. Comparison with other observers' data for the vertical position.

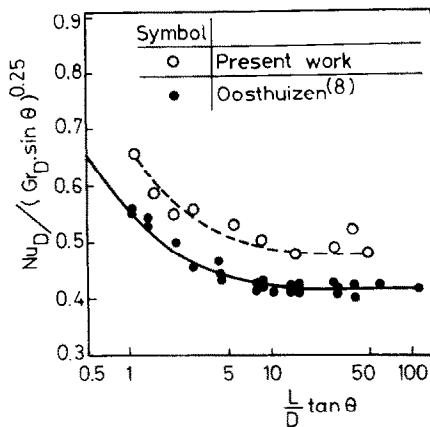


FIG. 14. Comparison with Oosthuizen data for $D \approx 19$ mm.

CONCLUSIONS

Experiments carried out to investigate natural convection heat transfer from the outside surface of inclined isothermal cylinders showed that:

(1) For the same cylinder length and inclination, the average heat transfer coefficient decreases with the increase of diameter.

(2) For the same cylinder length and diameter, the average heat transfer coefficient varies with angle of inclination. For the longer lengths, h_L increases with θ . For the shorter lengths it decreases with θ . In between a "limit length" exists at which the heat transfer coefficient is constant irrespective of θ . This is equal to the heat transfer coefficient corresponding to the horizontal cylinder.

(3) For the same cylinder diameter and inclination, the average heat transfer coefficient decreases gradually with the increase of cylinder length until it becomes constant indicating the beginning of turbulence.

(4) Based on average heat transfer the critical transition point from the laminar region to the turbulent region is independent of cylinder diameter. It depends on inclination angle only and increases with the increase of this angle.

(5) Based on local heat transfer the critical transition point representing the end of the laminar region is independent of cylinder diameter and depends on inclination only. The critical point representing the beginning of the turbulent region, however, appears to be constant.

(6) The experimental data could be correlated by equation (8) and equation (13) for the average laminar heat transfer and the average turbulent heat transfer respectively, and by equation (17) and equation (18) for the local laminar heat transfer and the local turbulent heat transfer respectively.

(7) The experimental data are generally in good agreement with previous work.

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CONVECTION THERMIQUE NATURELLE AUTOUR DE CYLINDRES INCLINES

Résumé—Des expériences permettent de déterminer les transferts thermiques locaux et moyens par convection naturelle à la surface externe de cylindres isothermes de diamètre et de longueur différents pour plusieurs inclinaisons, aussi bien en laminaire qu'en turbulent. On trouve que le transfert thermique dépend à la fois du diamètre et de l'inclinaison et on suggère les équations générales représentatives.

WÄRMEÜBERGANG BEI FREIER KONVEKTION AN SCHRÄG ANGESTELLTEN ZYLINDERN

Zusammenfassung—Es wurden Experimente zur Bestimmung des mittleren und örtlichen Wärmeübergangs bei freier Konvektion an der äußeren Oberfläche von isothermen Zylindern verschiedener Durchmesser und Längen bei unterschiedlichen Neigungswinkeln im laminaren und turbulenten Bereich durchgeführt. Es wurde gefunden, daß der Wärmeübergang sowohl vom Durchmesser als auch vom Neigungswinkel abhängt. Allgemeingültige Gleichungen zur Wiedergabe dieser Abhängigkeiten werden vorgeschlagen.

ТЕПЛОТДАЧА НАКЛОННЫХ ЦИЛИНДРОВ ПРИ ЕСТЕСТВЕННОЙ КОНВЕКЦИИ

Аннотация — Проведено экспериментальное исследование средней и локальной теплоотдачи при естественной конвекции от внешней поверхности изотермических цилиндров, имеющих разный диаметр и длину и расположенных под разными углами. Исследовались ламинарный и турбулентный режимы конвекции. Найдено, что теплоотдача зависит как от диаметра, так и угла наклона цилиндров. Предложены обобщенные зависимости, учитывающие эти параметры.