NATURAL CONVECTION HEAT TRANSFER FROM INCLINED CYLINDERS

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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

\[ \begin{align*}
  b, & \quad \text{constant;} \\
cturn, & \quad \text{constant;} \\
D, & \quad \text{outer diameter of cylinder [mm];} \\
h, & \quad \text{heat transfer coefficient [W/m}^2{\text{C}]}; \\
L, & \quad \text{cylinder length [m];} \\
m, & \quad \text{constant;} \\
n, & \quad \text{constant;} \\
p, & \quad \text{constant;} \\
t, & \quad \text{temperature [°C];} \\
x, & \quad \text{distance from beginning of cylinder [m];} \\
Gr, & \quad \text{Grashof number;} \\
Nu, & \quad \text{Nusselt number;} \\
Pr, & \quad \text{Prandtl number;} \\
\beta, & \quad \text{function of } Gr \text{ in equation 1;} \\
\theta, & \quad \text{angle of inclination of cylinder (from the vertical position).}
\end{align*} \]

Subscripts

\[ \begin{align*}
  a, & \quad \text{ambient;} \\
cr, & \quad \text{critical;} \\
D, & \quad \text{based on diameter;} \\
f, & \quad \text{film;} \\
L, & \quad \text{average (for a cylinder length = } L\text{);} \\
Lim, & \quad \text{limit;} \\
s, & \quad \text{surface;} \\
X, & \quad \text{local (at a cylinder length = } x\text{).}
\end{align*} \]

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.

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:

\[ Nu = \beta (Gr)^{1.3} \]  

(1)

where \( \beta \) is a function of \( Gr \).

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:

\[ Nu_l = \frac{0.61}{D} \left( D_{Gr} \cdot Pr \right)^{1.4} \]  

(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 ±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:

\[ Nu = b \left( Gr \cdot Pr \cdot \frac{D}{L} \right)^{1.4} \]  

(3)
with $b$ and $c$ as shown for isothermal cylinders:

\[
\begin{array}{cc}
\text{wires} & 0.87 \\
\text{long cylinders} & 1.3 \\
\text{short cylinders} & 0.57
\end{array}
\]

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 $Nu(Gr, \sin \theta)^{0.16}$ against $(L/D) \tan \theta$. 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...
Natural convection heat transfer from inclined cylinders

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 $L$, $h_1$ decreases with the increase of $D$. For a value of $D$ equal to infinity the vertical cylinder should give the same $h_1$ 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 $\theta$ 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 $\theta$. The rate of increase decreases with the decrease of $L$. For the shorter lengths, however, $h_L$ decreases with the increase of $\theta$. A certain value of $L$, (hereinafter called the limit length $L_{lim}$), at which $h_L$ is constant irrespective of $\theta$, must therefore exist. The heat transfer coefficient corresponding to this length is hereinafter called the limit coefficient $h_{lim}$.

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 $\theta$. Figure 4 clearly shows the limit length phenomena. A cylinder of this length gives the same value of $h_L$ at any $\theta$, 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 $\theta$ 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.
Natural convection heat transfer from inclined cylinders

12
11.5
11
10.5
10
9.5
9
8.5
8
7.5
7
6.5
6
5.5
5
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
L (m)

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

Correlation of average heat transfer data. Figure 5 shows log Nu, plotted against log (Gr, 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 \( t_f \) = \( (1/2) (t_a + t_s) \).

It will be seen that for each angle of inclination \( \theta \) the results are represented by two straight lines one with a slope increasing with \( \theta \) followed by another with a slope equal to 1/3 irrespective of \( \theta \). The two lines indicate the laminar range in which \( h_1 \) depends on cylinder length, and the turbulent range in which \( h_1 \) is independent of length. All the other diameters show the same general characteristics.

The value of \( (Gr, Pr)_\text{lim} \) corresponding to \( L_{\text{lim}} \) 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 \( Gr, Pr \) (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 \( \theta \) can be represented by equation (4):

\[
(Gr, Pr)_c = 2.6 \times 10^9 + 1.1 \times 10^9 \tan \theta, \ldots \tag{4}
\]

In the horizontal position equation (4) gives a value of \( (Gr, Pr)_c \) 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 = \( 2.7 \times 10^9 \). Other observers obtained values which vary between \( 1 \times 10^9 \) and \( 3 \times 10^9 \).

Laminar region. To obtain a correlation observers experimenting with plates suggest that a modified Grashof number equal to \( (Gr, \cos \theta) \) be used instead of \( Gr \). This method has been attempted in the present work and has resulted in reasonable correlations for values of \( \theta \) between 0 and 60°. However the use of \( \cos \theta \) gives a modified Grashof number equal to zero in the horizontal position (\( \theta = 90° \)). 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 \( Nu, \) and \( Gr, Pr \) for any \( \theta \) in the laminar region can be represented by:

\[
Nu_1 = mGr_1 Pr^\beta. \tag{5}
\]

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 \( \theta \)) 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
the effect of diameter. In the following correlations a Grashof number based on diameter \((Gr_D)\) will be used. For the same \(\theta\) 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 \(n:\)

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

\[
n = \frac{1}{4} + \frac{1}{12} (\sin \theta)^{1.2}.
\]

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

\[
Nu_x = [2.9 - 2.32 (\sin \theta)^{0.8}] (Gr_D)^{-1.12} 
\times [Gr_L Pr]^{1/4 + 1/32 (sin \theta)^{1.2}}.
\]

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 \(9.88 \times 10^7 \leq Gr_L Pr \leq (Gr_L Pr)_0\) (as taken from equation 4)]. It is valid for the whole range of \(\theta\) from 0 to 90°.

The experimental points are compared with equation (8) in Fig. 6. The horizontal cylinder points \((\theta = 90°)\) are the limit length points as taken from the \(\log Nu_x - \log Gr_L Pr\) 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 \((\theta = 0°)\) equation (8) reduces to

\[
Nu_x = 0.58 (Gr_D)^{1 / 12} (Gr_L Pr)^{1 / 3}.
\]

For air with \(Pr = 0.7\) equation (10) becomes

\[
Nu_x = 0.566 (Gr_D)^{1 / 12} (Gr_L Pr)^{1 / 3}.
\]

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 \(5.5 \times 10^8\). The other diameters give almost the same value. \((Gr_L Pr)_\text{lim}\) 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 \(h_{\text{lim}}\) is proportional to \((L_L)_{\text{lim}} D^{-1.4}\). For the horizontal cylinder \(h_{\text{lim}}\) is proportional to \(D^{-1.4}\). Since \(h_{\text{lim}}\) is the same for

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**Table 1. Variation of \((Gr_D Pr)_x\) with \(\theta\)**

<table>
<thead>
<tr>
<th>(\theta)</th>
<th>((Gr_D Pr)_x)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0° (vertical)</td>
<td>2.7 \times 10^9</td>
</tr>
<tr>
<td>15°</td>
<td>2.8 \times 10^9</td>
</tr>
<tr>
<td>30°</td>
<td>3.35 \times 10^9</td>
</tr>
<tr>
<td>45°</td>
<td>4.05 \times 10^9</td>
</tr>
<tr>
<td>60°</td>
<td>4.7 \times 10^9</td>
</tr>
<tr>
<td>90° (horizontal)</td>
<td>infinity</td>
</tr>
</tbody>
</table>
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 \((Gr, Pr)_{lim}\).

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

\[
Nu_L = a(Gr, Pr)^{1/3}. \tag{12}
\]

The values of \( a \), as calculated from the curves for the different diameters and inclinations, were found to vary with both \( D \) and \( \theta \). The results could be represented by

\[
Nu_L = [0.47 + 0.11 (\sin \theta)^{0.8}] \times (Gr, Pr)^{1/3}. \tag{13}
\]

Equation (13) represents the general equation for average natural convection in the turbulent region in the range of the present work \([1.08 \times 10^4 \leq Gr \leq 6.9 \times 10^5 \text{ and } (Gr, Pr)_c \leq Gr, Pr \leq 2.95 \times 10^{10} \text{ where } (Gr, Pr)_c \text{ is to be taken from equation (4)})\). It is valid for the whole range of \( \theta \) from 0 to 90°.

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

For the vertical cylinder (\( \theta = 0° \)) equation (13) reduces to

\[
Nu_L = 0.47 \left( Gr_D \right)^{-1/2} \left( Gr, Pr \right)^{1/3}. \tag{14}
\]

For the horizontal cylinder (\( \theta = 90° \)) equation (13) reduces to

\[
Nu_L = 0.58 \left( Gr_D \right)^{-1/2} \left( Gr, Pr \right)^{1/3}, \tag{15}
\]

which for air with \( Pr = 0.7 \) can be written as

\[
Nu_L = 0.566 \left( Gr_D \right)^{1/4}. \tag{16}
\]

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 \( h_L \) 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 \( h_L \) 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 \( h_L \) at the beginning and the end of the transition region decreases with the increase of \( \theta \) until, at \( \theta = 90° \) (the horizontal position), \( h_L \) becomes constant irrespective of \( x \).

Correlation of local heat transfer data. Plotting log \( Nu_L \) against log \( (Gr, Pr) \) 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.
Natural convection heat transfer from inclined cylinders

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

The experimental results could be represented by the following equations:

$$Nu_x = [2.3 - 1.72 (\sin \theta)^{0.8}]\times (Gr_x)^{-1.12} [Gr_x Pr]^{1/4} (1/12 \sin \theta)^{-2}.$$  \hspace{0.5cm} (17)

$$Nu_x = [0.42 + 0.16 (\sin \theta)^{0.8}]\times (Gr_x)^{-1.12} (Gr_x Pr)^{1.3}. \hspace{0.5cm} (18)$$

Equation (17) represents the general equation for local natural convection in the laminar region in the range of the present work $[1.08 \times 10^4 \leq Gr_x \leq 6.9 \times 10^5$ and $6.9 \times 10^5 < Gr_x Pr < 2.3 \times 10^{10}]$. It is valid for the whole range of $\theta$ 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 $[1.08 \times 10^4 \leq Gr_x \leq 6.9 \times 10^5$ and $(Gr_x Pr)_{r-2} \leq Gr_x Pr \leq 2.3 \times 10^{10}]$. It is valid for the whole range of $\theta$ 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, Pr)_{\delta -1} represents the end of the laminar region. As can be seen from Fig. 8 it varies with \( \delta \). No variation with diameter could be seen. The following equation was found to represent the variation with \( \delta \):

\[
(Gr, Pr)_{\delta -1} = 2.16 \times 10^9 + 0.283 \times 10^9 \tan \theta. \tag{19}
\]

(Gr, Pr)_{\delta -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 \( 4.4 \times 10^9 \).

**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 \( \beta \) in equation (1) corresponding to \( 0.47 Gr_{in}^{-1/2} \) 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 Kirpitchen [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 \( h_n \) 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.](image-url)
Fig. 12. Comparison between equations (1) and (14).

Fig. 13. Comparison with other observers' data for the vertical position.
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 $\theta$. For the shorter lengths it decreases with $\theta$. In between a "limit length" exists at which the heat transfer coefficient is constant irrespective of $\theta$. 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.

REFERENCES

Natural convection heat transfer from inclined cylinders

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ТЕПЛООТДАЧА 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.