Int. J. Heat Mass Transfer. Vol. 30, No. 8, pp. 1685–1692, 1987
Printed in Great Britain

0017–9310/87 $3.00+0.00

Influence of ventilated shrouds on the convective heat transfer to a circular cylinder

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(*Received* 7 *November* 1986 *and in final form* 6 *February* 1987)

Abstract—Convective heat transfer to shrouded cylinders in transverse flow has been studied over the Reynolds number range 2000–20 000. The influence of shroud ventilation, relative shroud diameters, and orientation of the ventilation holes was studied. In some cases, average inner cylinder Nusselt numbers were found to exceed the comparable bare cylinder values by as much as 50%. Cylinder heat convection was influenced more by the degree of ventilation and shroud diameter than by hole orientation. An equivalent inner bare cylinder diameter, based on degree of shroud ventilation and shroud diameter, was developed which can be useful in shroud design studies.

INTRODUCTION

TEMPERATURE sensors are often placed within ventilated shrouds when they are used in fluid-flow measurements. The shroud serves as a radiation shield and protects the sensor from debris and rough handling. As long as the fluid temperature variation is slow relative to the sensor response time, the influence of the shroud on the heat transfer to the sensor is of secondary concern. However, if temperature variations are rapid or the sensor is part of a control loop, the influence of the shroud on sensor thermal response can be an important consideration. The present investigation was concerned with the influence of shroud design on the steady-state heat transfer to a solid, concentrically mounted circular cylinder. The steady-state heat transfer coefficient data were required as part of an overall thermal response study [1, 2].

Shroud parameters which can influence heat transfer are: (1) degree of ventilation; (2) shroud diameter relative to inner cylinder diameter; (3) shape of ventilation holes; (4) ventilation hole pattern; (5) orientation of ventilation hole pattern; and (6) thickness and surface finish of the shroud. In addition, the external flow field, as characterized by Reynolds number, Prandtl number, and turbulence intensity, affects the heat transfer to the cylinder. A previous study reported preliminary investigations of the influence of the degree of shroud ventilation and orientation of the shroud hole pattern on heat transfer to a cylinder in transverse flow [2], but it was determined that the internal flow passage, as characterized by the ratio of the shroud (inside) diameter to the inner cylinder diameter was potentially a more important variable than hole orientation. That inference was based upon drag measurements which have been reported for shrouded cylinders [3], and also from earlier work by Warshawsky [4] who recommended the use of an internal flow impedance model to estimate heat transfer to the inner cylinder.

EXPERIMENTAL APPARATUS

An open circuit wind tunnel was used in this study. The wind tunnel test section was 300 by 430 mm and was capable of providing free stream velocities, U_∞, between 3.5 and 80 m s^{-1}. Turbulence intensities, $\sqrt{(u^2)}/U_\infty$, were measured with a hot wire anemometer and varied from 0.005 at a free stream velocity of 15 m s^{-1} down to 0.003 at 40 m s^{-1}. Kaylar [5], has shown that turbulence intensities below approximately 1% do not influence convective heat transfer to cylinders significantly. Instrumentation noise prevented measuring turbulence intensities at wind tunnel speeds below 15 m s^{-1}. However, the close agreement of bare cylinder heat transfer measurements in the present study with classical data supports the assumption that the wind tunnel turbulence intensity was within acceptable limits (below 0.01) down to 3.5 m s^{-1}.

A brass tube, 7.9 mm in diameter with a wall thickness of 1.1 mm was used as the internal test cylinder. The test cylinder was supported between two aerodynamic struts, as shown in Fig. 1, and had a spanwise

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NOMENCLATURE

- C constant in Morgan's correlation
- D inner cylinder diameter
- D_e equivalent diameter
- D_h shroud hole diameter
- D_i inside diameter of perforated shroud
- D_o outside diameter of perforated shroud
- L shroud hole spacing
- Nu Nusselt number based on inner cylinder diameter
- Nu_o Nusselt number based on outside diameter of shroud
- n constant in Morgan's correlation
- Re Reynolds number based on inner cylinder diameter
- U_∞ free stream velocity
- u velocity fluctuation.

Greek symbols

- θ shroud orientation angle
- ϕ radius ratio
- Ψ ventilation factor.

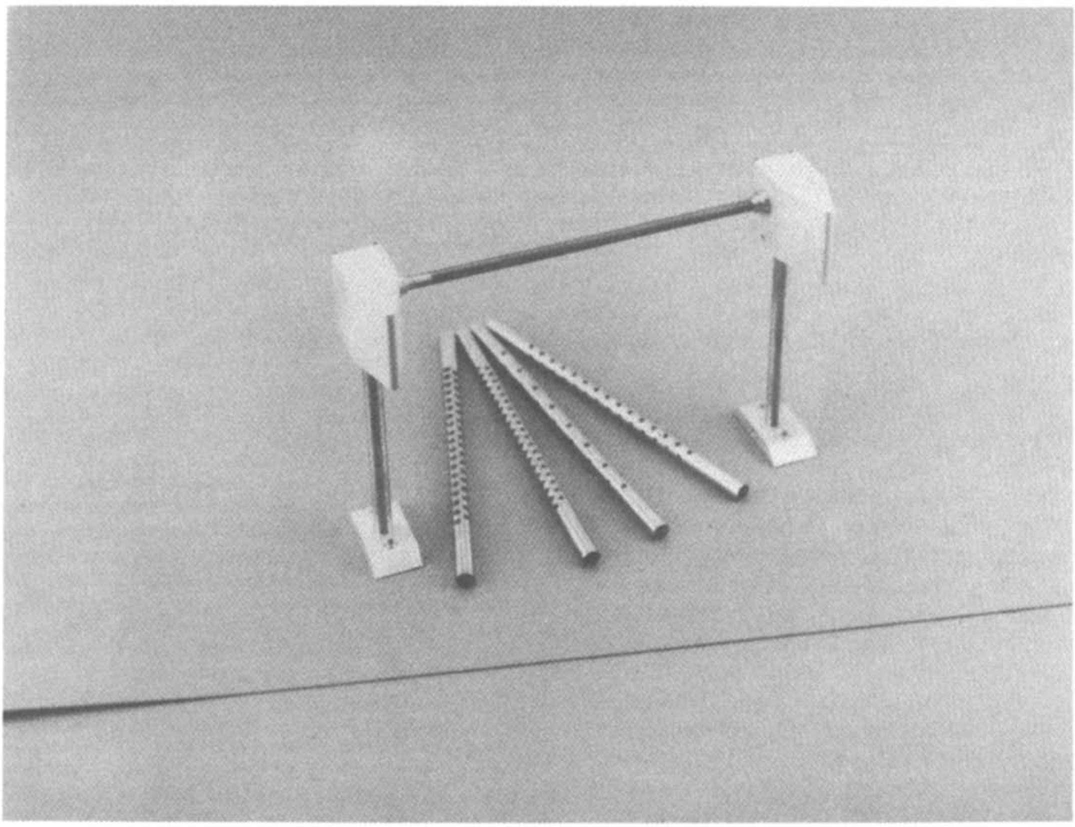


FIG. 1. Photograph of bare cylinder test assembly and four representative shrouds (not installed).

length of 200 mm. Representative shrouds are also shown in the figure. The shrouds were supported by the aerodynamic struts which blocked both ends. A 1 mm diameter chromel wire was located along the tube axis and was operated as a constant power heater. The voltage drop across the central 76.2 mm of the wire was measured and controlled to produce a heat flux of 1040 W m^{-2} on the outside surface of the tube. Five 0.25 mm diameter chromel–alumel thermocouples were pulled through the inside of the tube and then out to the tube surface through 0.64 mm diameter holes. The holes were drilled at staggered locations 12.7 mm on either side of the tube center plane at circumferential locations of 0° (forward stagnation line), 15°, 75°, 120° and 300°. The thermocouple wires were soldered to the outside surface of the brass tube and the surface was then finished. The entire heating wire, voltage pick-up, and thermocouple assembly was potted inside the tube with thermal epoxy (with a thermal conductivity of 0.52 W m^{-1} K^{-1}). Subsequently, the centrality of the heater wire was verified using X-ray techniques. The thermocouple located at the 15° circumferential location was disqualified during preliminary testing because it was not sufficiently close to the tube surface.

The bare test cylinder was tested in the wind tunnel

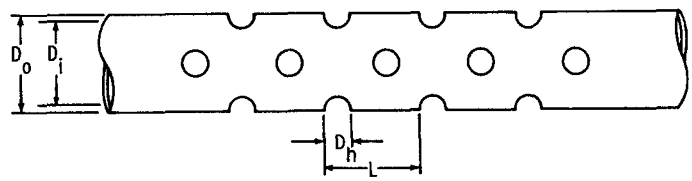


FIG. 2. Schematic diagram of shroud geometry.

to verify that classical Nusselt number data were produced over the Reynolds number range 1000–20 000. The heat transfer coefficient was developed using the average measured surface temperature and the known applied heat flux. The bare cylinder measurements were scattered on either side of Morgan's correlation [6] with a standard deviation of 6.1%.† Furthermore, the data show no resolvable signs of an influence of turbulence intensity on Nusselt number over the Reynolds number range [7].

Radiation correction, the influence of free convection and an uncertainty analysis have been described in more detail elsewhere (Daryabeigi and Ash [7]). The radiation correction was less than 5% of the steady-state heating rate and the influence of free convection was negligible in all cases. The estimated uncertainty in experimental Reynolds number ranged from 2.8% at $Re = 14000$ to 4.3% at $Re = 1800$. The uncertainty in Nusselt number ranged from 9.6% at $Re = 14000$ to 7.0% at $Re = 1800$.

Degree of ventilation (ventilation factor) was studied using a set of ten shrouds which were fabricated for these experiments. Defining the ventilation factor, Ψ, as the ratio of the open or drilled out area of the shroud tube to the total surface area of a corresponding unventilated tube of the same span, nominal ventilation factors of 0.09, 0.18 and 0.27 were studied. One shroud with a ventilation factor of 0.36 was also tested. All of the shrouds had six staggered rows of circular holes equally spaced around the tube circumference, as shown in Fig. 2. Three shroud diameters were used in the experiment in order to vary the internal flow passage. The internal flow passage was characterized by the radius ratio, ϕ, which was the ratio of the inside shroud radius to the test cylinder radius. The geometrical characteristics of the shrouds are summarized in Table 1. (Refer to Fig. 2 for shroud dimensions.)

It should be noted that heat transfer data for $\Psi = 1$ (original bare cylinder) and $\Psi = 0$ (heat conduction through an air gap to a larger bare cylinder) are also known using Morgan's correlations for bare cylinders. In the $\Psi = 0$ case, it is assumed that the only thermal resistances between a completely shrouded cylinder and the free stream are an inner conductive resistance for the annulus and an outer bare cylinder

† Morgan's correlation relates Nusselt number and Reynolds number according to $Nu = C\ Re^n$ where $C = 0.583$, $n = 0.471$ when $35 \leqslant Re \leqslant 5000$ and $C = 0.148$, $n = 0.633$ when $5000 \leqslant Re \leqslant 50000$.

Table 1. Geometric data for perforated shrouds

Shroud No.	D_o (mm)	D_i (mm)	ϕ	D_h (mm)	L (mm)	Ψ
1	9.5	9.0	1.14	3.2	18.5	0.090
2	9.5	9.0	1.14	3.2	9.0	0.180
3	9.5	9.0	1.14	3.3	6.4	0.263
4	9.5	9.0	1.14	4.0	6.9	0.359
5	12.7	10.9	1.38	4.8	30.6	0.091
6	12.7	10.9	1.38	4.8	15.3	0.173
7	12.7	10.9	1.38	4.8	10.2	0.254
8	19.05	16.6	2.09	6.3	40.2	0.083
9	19.05	16.6	2.09	6.3	17.2	0.178
10	19.05	16.6	2.09	6.3	12.1	0.250

film coefficient resistance. Hence, data for $\Psi = 0, 0.09, 0.18, 0.27$ and 1 have been used in this study.

RESULTS

Variation of Nusselt number with Reynolds number, based on inner cylinder diameter, for the different ventilation factors is shown in Figs. 3(a)–(d), along with Morgan's correlation for bare cylinders [6]. The $\theta = 0^0$ case corresponded to a shroud orientation with a row of holes located along the undisturbed forward (and aft) stagnation line(s). The case when shroud holes were located symmetrically on either side of the stagnation line corresponded to $\theta = 30^0$.

Data were taken in some cases for shroud orientations of $\theta = 15^0$ which was an asymmetric orientation with respect to the undisturbed stagnation line, but those data were essentially midway between the $\theta = 0^0$ and 30^0 measurements. Hence, hole patterns which were asymmetric with respect to the undisturbed external stagnation lines did not produce any unusual heat transfer effects and those measurements were not developed fully.

The influence of ventilation factor on Nusselt number is shown in Figs. 4(a)–(c). Figure 4(a) is a plot for the smallest diameter shroud set ($\phi = 1.1$) and Figs. 4(b) and (c) are for nominal radius ratios of $\phi = 1.4$ and 2.1, respectively. Reynolds number was used as a fixed parameter.

For the $\Psi = 0$ cases in Fig. 4, it was assumed that a fictitious shroud with no thickness was present and that no free convection occurred in the air gap between the imaginary solid tube and the inner cylinder. Then, the 'external surface' Nusselt number, Nu_o, could be calculated directly from Morgan's correlations [6] using the inner 'shroud' diameter, D_o

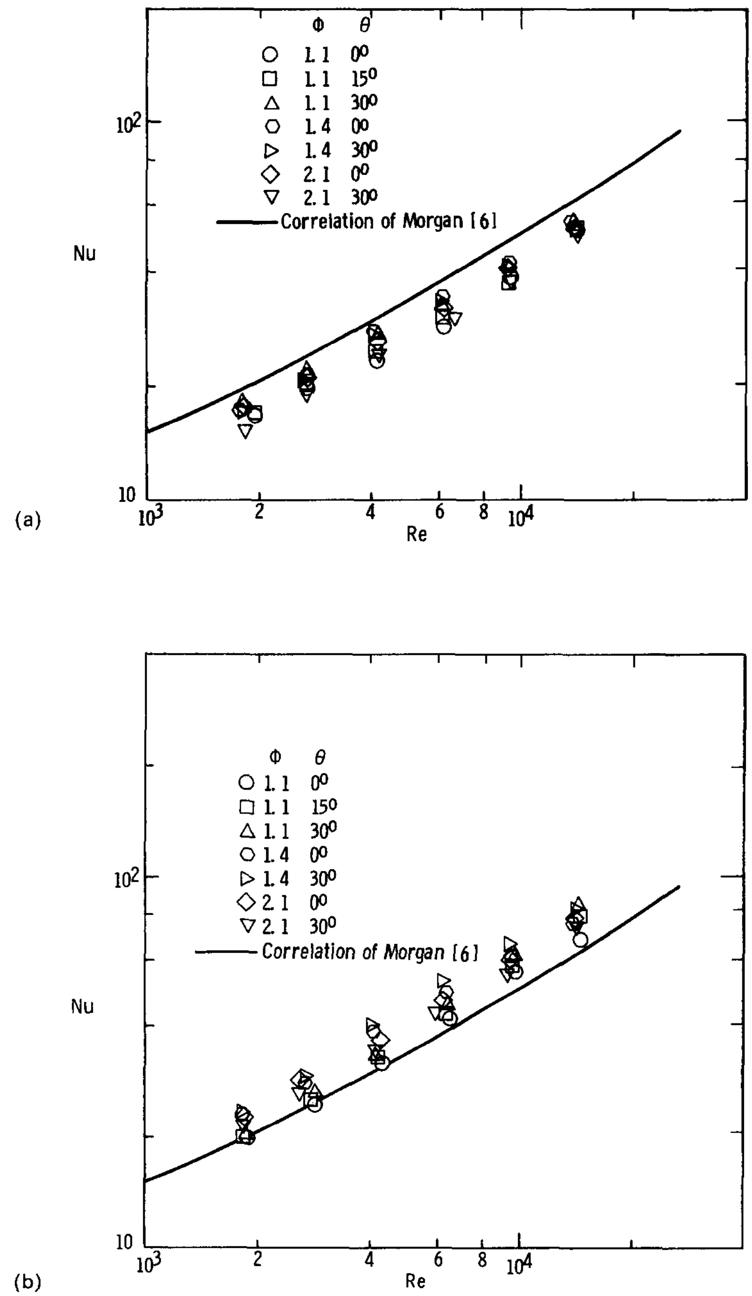
(which equals D_i), to calculate Reynolds number. The appropriate limiting Nusselt number for the completely shrouded cylinder and infinitesimal temperature differences is given by

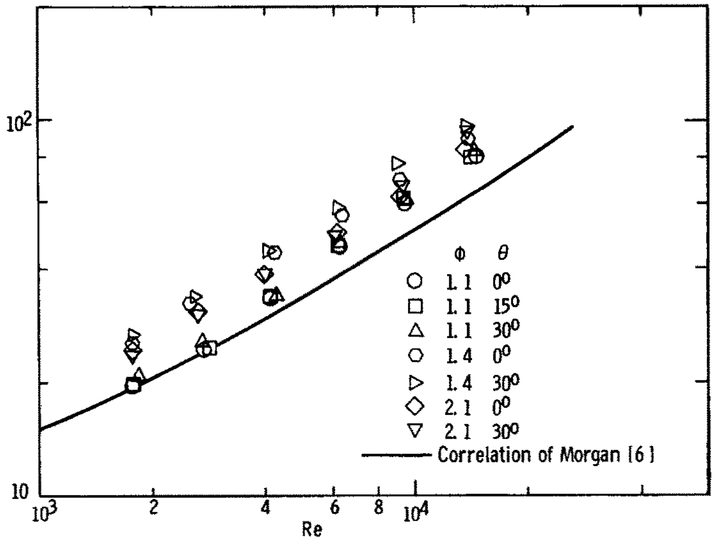
$$Nu = \frac{2Nu_o}{2+Nu_o \ln \phi} \tag{1}$$

which established the $\Psi = 0$ intercept for each case. The $\Psi = 1$ cases were obtained directly using Morgan's correlation for a cylinder with a diameter D. The $\Psi = 0$ and 1 data were assumed to be exact. Furthermore, the $\Psi = 1$ points represent the convective heat transfer to the reference bare cylinder cases so that the influence of the shroud on heat transfer is indicated by the departure from horizontal lines drawn through the $\Psi = 1$ points.

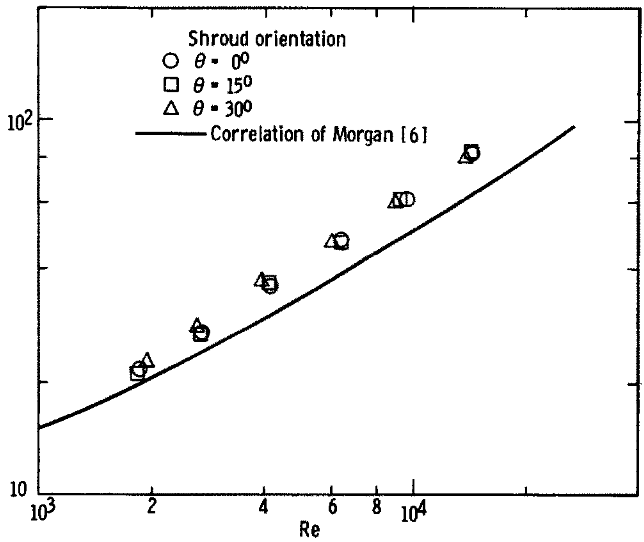
DISCUSSION

The present investigation has shown that shrouds can be designed which either reduce or enhance heat transfer to the central cylinder. The mechanism responsible for enhanced heat transfer is more complex than that for reduced heat transfer. Obviously viscous effects due to two solid boundaries can reduce thermal energy transport to the central cylinder, but two different flow mechanisms can account for





(c)



(d)

FIG. 3. Variation of Nusselt number with Reynolds number for different perforated shrouds: (a) $\Psi = 0.09$; (b) $\Psi = 0.18$; (c) $\Psi = 0.27$ and (d) $\Psi = 0.36$.

increased heat transfer. First, the shroud geometry tends to prevent flow separation on the inner cylinder. The base pressure field produced by the external flow over the shroud can be relieved somewhat and provide simultaneously a pressure force which draws fluid through the confined internal passage. That flow should be attached in most cases. Secondly, the hole pattern itself promotes highly vortical, three-dimensional flow in the annular space. The three-dimensional flow field can augment convective energy transport. As long as the viscous effects are controlled through proper sizing of ventilation holes and flow passage dimensions, it is possible for the two thermal augmentation mechanisms to overcome the viscous retardation.

The shrouded cylinder data can be generalized somewhat using an equivalent diameter which is a function only of the ventilation factor and radius ratio (neglecting Reynolds number and shroud orientation effects). The equivalent diameter should predict the same Nusselt number as measured in the experiments for the same flow conditions (velocity and thermophysical properties). Morgan's correlations [6] were assumed to predict the bare cylinder heat transfer, and enabled a direct computation of equivalent diameter and diameter ratio.

Equivalent diameters were calculated for each experimental data point and are summarized in Table 2. Rather than show every combination of Reynolds number and geometry, the data have been averaged for given values of ϕ and Ψ. The data show that equivalent diameters can be used over the entire range

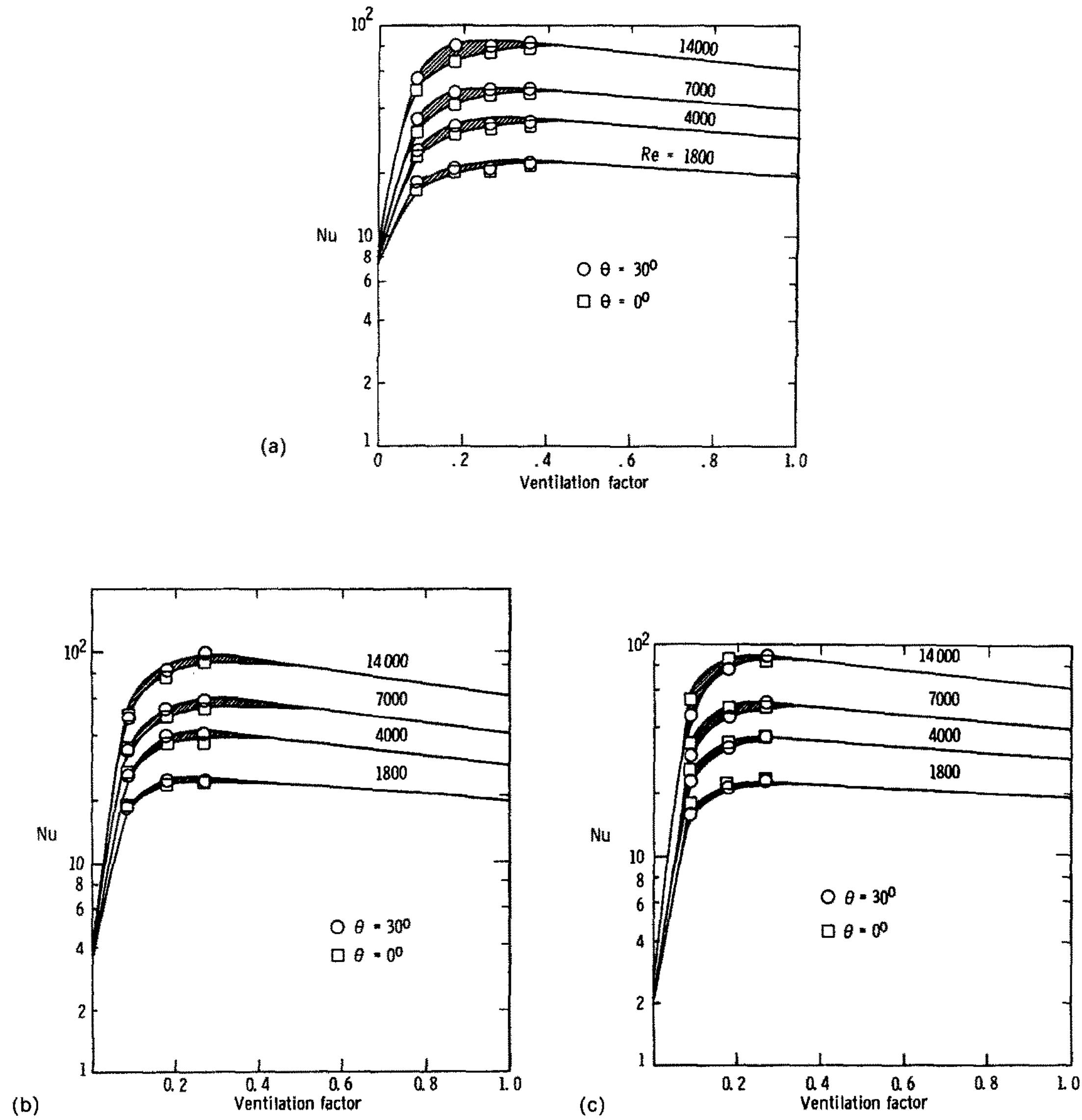


FIG. 4. Variation of extrapolated Nusselt number with ventilation factor for different perforated shrouds: (a) $\phi = 1.1$; (b) $\phi = 1.4$ and (c) $\phi = 2.1$.

Table 2. Influence of shroud geometry on effective diameter

Radius ratio ϕ	Ventilation factor Ψ	D_e/D	Standard deviation of predicted Nusselt numbers over Reynolds number range (%)
1.1	0.09	0.72	5.65
1.1	0.18	1.20	7.63
1.1	0.27	1.27	7.08
1.1	0.36	1.40	4.37
1.4	0.09	0.79	3.96
1.4	0.18	1.50	4.86
1.4	0.27	1.87	4.38
2.1	0.09	0.70	5.62
2.1	0.18	1.31	3.95
2.1	0.27	1.58	3.82

of ventilation factors and diameter ratios tested (over the Reynolds number range studied). That conclusion is based on the fact that the standard deviations of the Nusselt numbers calculated using the equivalent diameters are within typical experimental scatter. The raw data show better agreement for Reynolds numbers above 3000 and concurrently, a weak Reynolds number effect appears for lower Reynolds number—particularly at the lowest radius ratio. Hence, an equivalent diameter appears most useful for the larger radius ratios (1.4 and 2.1) and at the higher Reynolds numbers.

The Reynolds number dependent limiting case of $\Psi = 0$ can be computed directly from equation (1), using Morgan's correlation [6]. However, that limiting case, though theoretically accurate, precludes the pro-

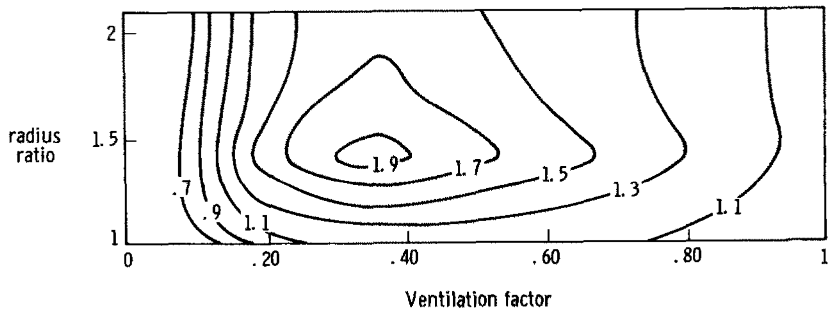


FIG. 5. Constant equivalent diameter ratio contours (D_e/D) as functions of ventilation factor, Ψ, and radius ratio ϕ.

posed augmentation mechanisms and may be misleading. Consequently, even though the equivalent diameter is strongly dependent on Reynolds number for the unventilated case ($\Psi = 0$), it does not suggest any trends for the experiments.

Smooth curves were fared through the equivalent diameter data and curves of constant equivalent diameter ratio are shown as functions of shroud geometry parameters in Fig. 5.

CONCLUSIONS

This study has shown that internal flow passage size (radius ratio) and degree of ventilation both influence the convective heat transfer to shrouded cylinders to a significant extent. Convection to the cylinder can be enhanced for ventilation factors greater than 10% and the maximum enhancement occurs when the shroud passage is characterized by a radius ratio between 1.1 and 2. Maximum heat transfer occurs for a given shroud size (radius ratio) as the ventilation factor tends toward 50%, but the actual maximum was not determined, since very high ventilation factors are impractical to build (without using screens) and lose their effectiveness as radiation shields.

An equivalent diameter was found to represent the experimental Nusselt number data reasonably well over the range of flow and geometry conditions tested, but a weak Reynolds number effect could be detected for bare cylinder Reynolds numbers of the order of 3000 and for radius ratios that approached unity.

Acknowledgement—The work was supported by NASA Langley Research Center. The authors would like to thank Mr Edward F. Germain for many helpful discussions.

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INFLUENCE D'ETUIS VENTILES SUR LA CONVECTION THERMIQUE A UN CYLINDRE CIRCULAIRE

Résumé—La convection thermique à des cylindres capotés, avec écoulement transversal a été étudiée pour un domaine de nombre de Reynolds allant de 2000 jusqu'à 20 000. L'influence de la ventilation de l'étui est étudiée en considérant les diamètres relatifs des évents et l'orientation des évents. Le nombre de Nusselt moyen du cylindre intérieur est parfois supérieur de 50% aux valeurs correspondant au cylindre seul. La convection sur le cylindre est plus influencée par le degré de ventilation et le diamètre de l'étui que par l'orientation des trous. On définit pour les applications pratiques un diamètre équivalent du cylindre intérieur basé sur le degré de ventilation de l'étui et le diamètre de l'étui.

EINFLUSS VON BELÜFTETEN UMMANTELUNGEN AUF DEN KONVEKTIVEN WÄRMEÜBERGANG AN EINEM KREISZYLINDER

Zusammenfassung—Der Wärmeübergang an quer angeströmten ummantelten Zylindern wurde für Reynoldszahlen von 2000 bis 20 000 untersucht. Der Einfluß der Belüftung der Ummantelung, des relativen Ummantelungsdurchmessers und der Ausrichtung der Belüftungsöffnungen wurde untersucht. In einigen Fällen übertraf die mittlere Nusseltzahl am inneren Zylinder die vergleichbaren Werte am entblößten Zylinder um ca. 50%. Die Konvektion am Zylinder wurde stärker durch den Grad der Belüftung und den Ummantelungsdurchmesser als durch die Ausrichtung der Öffnungen beeinflußt. Basierend auf dem Grad der Belüftung und dem Ummantelungsdurchmesser wurde ein äquivalenter Durchmesser des entblößten Zylinders entwickelt, der für Untersuchungen der Ummantelungsgestaltung nützlich sein kann.

ВЛИЯНИЕ ВЕНТИЛИРУЕМЫХ ЗАЩИТНЫХ ЭКРАНОВ НА КОНВЕКТИВНЫЙ ПЕРЕНОС ТЕПЛА К КРУГЛОМУ ЦИЛИНДРУ

Аннотация—Конвективный перенос тепла к экранированным цилиндрам в поперечном потоке изучался в диапазоне чисел Рейнольдса от $2 \,.\, 10^3$ до $2 \,.\, 10^4$. Исследовалось влияние экранов, их относительных диаметров и ориентации вентиляционных отверстий. В некоторых случаях установлено, что средние числа Нуссельта внутри цилиндра превышают сопоставимые значения для неэкранированного цилиндра на 50%. Более сильное влияние на тепловую конвекцию у цилиндра оказывает степень вентиляции и диаметр защитного экрана, а не ориентация отверстий. По данным о степени вентиляции экрана и диаметре получен эквивалентный внутренний диаметр неэкранированного цилиндра, который можно использовать при проектировании.