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Turbulent forced convection heat transfer from a bottom heated open surface cavity

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Abstract—The results of an experimental investigation of forced convection heat transfer from a bottom heated open surface cavity are presented. The Reynolds numbers investigated, based on the cavity width, extended from 2×10^4 to 4×10^5. Four cavities with aspect ratios (height/width) of 1, 4/3, 2, and 4 were investigated. The average heat transfer coefficients on the bottom of the cavities were measured. A correlation was obtained relating the Nusselt number based on the cavity height to the Reynolds number based on the cavity height and a calculated cavity fluid velocity. This correlation fits the data with an r.m.s. error of 8%.

INTRODUCTION

CAVITIES occur in many technological and industrial applications either by design or circumstance, and are often found in heated surfaces placed in an external forced flow. Some examples are: notches in turbine flow passages and combustion chambers, cavities that are formed in the spaces between electronic components, grooves on ablating surfaces, slots on finned heat exchangers, and cavity solar central receivers.

Boundary layer separation, streamline curvature, buoyancy, turbulence production, re-attachment, and re-circulation complicate the flow phenomena around and inside the cavity and can result in substantial effects on the drag and heat transfer. Consequently, calculation of the heat transfer from a cavity is very difficult. A complete knowledge of local heat transfer coefficients are needed, however, if hot spots which can lead to component failure or preferential corrosion are to be avoided.

One of the earliest studies of heat transfer in laminar flow past a cavity was by Chapman [1], who assumed the heat transfer from a cavity was governed solely by the transfer properties across the shear layer. Charwat *et al.* [2] postulated a 'Mass Exchange Model' for predicting the heat transfer coefficient and showed that the heat transfer coefficient was proportional to $(\rho_\infty U_\infty)^{0.6}$. This result was supported by Larson [3] but not by Seban and Fox [4] who found that the heat transfer coefficient was instead proportional to $(\rho_\infty U_\infty)^{0.8}$.

Seban [5] investigated heat transfer from shallow cavities placed in a turbulent horizontal flow. The aspect ratios (height to width ratios) investigated varied from 0.21 to 0.5. It was found in ref. [5] that the shear layer contributes little to the total heat transfer resistance. Instead, the major heat transfer resistance was found to occur near the bottom surface where the eddy diffusivity is much smaller than that in the reverse flow region. These results were also found in the study by Fox [6] who investigated heat transfer from rectangular notches with aspect ratios of 0.57–4.0. Fox also showed that when the thickness of the approaching boundary layer was much smaller than the cavity height, the heat transfer results could be correlated by using the notch width as the characteristic length dimension. In the present study, the approaching boundary thicknesses were very much less than the cavity height. Consequently, the present heat transfer results were correlated with either the cavity width or height. Haugen and Dhanak [7] also measured velocity and temperature profiles, heat transfer coefficients, and pressure distributions in rectangular heated cavities with a forced flow at their openings. The cavity height was varied from 25.3 to 63.5 mm (1.0 to 2.5 in). The boundary layer thickness of the approaching flow was also varied. In contrast to Seban [5] and Fox [6], Haugen and Dhanak found that the shear layer controlled the heat transfer rate as evidence by large temperature variations in the shear layer but an almost constant temperature core region. Reference [7] also found that while the heat transfer results are only moderately affected by the boundary layer thickness of the approaching flow they are very sensitive to the cavity aspect ratio.

Yamamoto *et al.* [8] presented forced convection heat transfer results for a cavity with a heated bottom only and a horizontal force flow at its opening. Pressure coefficients, mean temperatures, and local and mean heat transfer coefficients were reported for cavity aspect ratios between 0 and 1.0. It was found in

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NOMENCLATURE

A	surface area
$\bar{h}$	average convective heat transfer coefficient
H	height of cavity
k	thermal conductivity
L	span of cavity
$\overline{Nu}_H$	average Nusselt number based on $H = \bar{h}H/k$
P	input power
Q	heat flux
Re_W	Reynolds number based on $W = U_\infty W/\nu$
Re_H^*	Reynolds number based on $H = u_c H/\nu$
T	temperature
U_∞	free-stream velocity
u_c	cavity fluid velocity
W	width of cavity
x	streamwise coordinate
y	vertical coordinate
z	spanwise coordinate.

Greek symbols

δ	oncoming boundary layer displacement thickness
ν	kinematic viscosity
ρ	density.

Subscripts

c	cavity
H	height
w	wall
W	width
∞	free-stream.

ref. [8] that for aspect ratios less than about 0.3, the mean Nusselt number changes significantly while for larger aspect ratios the mean Nusselt number is relatively constant. Aung [9] has reported experimental results for laminar flow past heated cavities with aspect ratios of 0.25–1.0. Heat transfer results were obtained from interferograms. Numerical predictions using the k–ε turbulence model with algebraic approximations for the turbulent fluxes were performed by Gooray *et al.* [10]. It was found in ref. [10] that the largest heat transfer rates occur on the downstream wall.

The present study is an experimental investigation of forced convection heat transfer from bottom heated open surface cavities. The cavity geometries investigated in the present study were sufficiently large that three-dimensional flow was encountered. Most of the past cavity heat transfer studies have been for two-dimensional flow conditions. The purpose of this study is to present experimental heat transfer data for flow over deep three-dimensional cavities, and to demonstrate that with the use of a cavity velocity the data for different cavity aspect ratios can be correlated.

EXPERIMENTAL APPARATUS AND PROCEDURE

Four cavity models with height to width ratios H/W of 1, 4/3, 2 and 4 were tested in the present study. The span length, L, was fixed at 609.6 mm (24 in) for all four models. Cavity dimensions are listed in Table 1. Table 1 shows that the span to width ratios varied from 2 to 8. It was found by Maull and East [11] that for the span to width ratios greater than 9 and for height and width ratios greater than 2.5, two-dimensional flow would be found in the cavities. Since these conditions were violated for all but probably the two smallest width cavities, three-dimensional flow was anticipated.

Table 1. Model dimensions

Model number	Height, H mm (in)	Width, W mm (in)	Span, L mm (in)	$\frac{H}{W}$	$\frac{L}{W}$
1	304.8 (12)	76.2 (3)	609.6 (24)	4	8
2	304.8 (12)	152.4 (6)	609.6 (24)	2	4
3	304.8 (12)	228.6 (9)	609.6 (24)	4/3	8/3
4	304.8 (12)	304.8 (12)	609.6 (24)	1	2

The cavities were tested in the UCI Mechanical Engineering subsonic wind tunnel. The test section cross-section is 609.6 × 914.4 mm (24 × 36 in), and the tunnel velocity range is about 0.3–50 m s^{-1} (1–164 ft s^{-1}). The mean velocity profiles were measured across the test section and were found uniform to within ±1%. The oncoming boundary layer displacement thickness nondimensionalized with respect to cavity height was calculated to fall in the range of $7 \times 10^{-3} < \delta/H < 9 \times 10^{-3}$. The turbulence intensity was also measured and was less than 0.1% over the velocity range tested. A slot was cut into the floor of the test section and the cavities mounted to the wind tunnel floor as shown in Fig. 1(a).

The cavities were constructed from 6 mm (0.25 in) Plexiglas sheets and fitted on the bottom with nichrome wired aluminum heaters as shown in Fig. 1(b). The downstream sidewall could be slid back to accommodate from one to four heaters laid side by side. The entire assembly was glued tight with acrylic cement and silicone seal. The cavities were backed with 76.2 mm (3 in) thick foam insulation to reduce heat losses. Twenty-four type E thermocouples were embedded in three rows of eight in the heaters as shown in Fig.

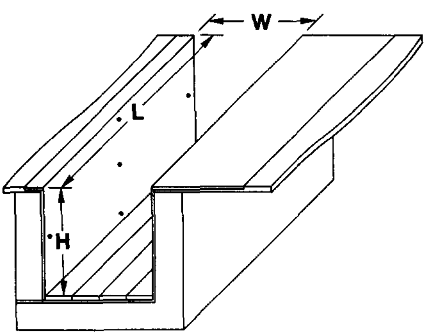

FIG. 1(a). Bottom heated cavity with end walls removed.

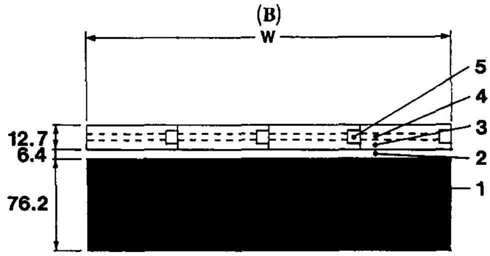

FIG. 1(b). Cavity bottom cross-section: (1) insulation; (2) Plexiglas; (3) aluminum; (4) nichrome wire; (5) ceramic cement.

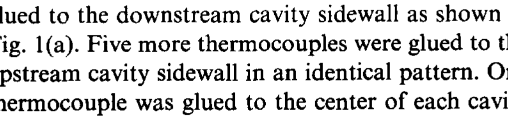

FIG. 1(c). Cavity heater with thermocouple locations.

1(c). Thermocouples were also glued to the outside of the Plexiglas cavity walls. Five thermocouples were glued to the downstream cavity sidewall as shown in Fig. 1(a). Five more thermocouples were glued to the upstream cavity sidewall in an identical pattern. One thermocouple was glued to the center of each cavity end wall.

Slots were cut in the end wall to allow a thermocouple to be traversed across the cavity. A fine wire Type E thermocouple (0.127 mm or 0.005 in) was used to measure temperature profiles within the cavity. Further details of construction are given in ref. [12].

Data reduction

The convective heat transfer coefficient can be calculated by performing an energy balance over the cavity. The wind tunnel was operated at a particular velocity setting, and the cavity was heated until steady state was obtained. Steady state was obtained within 10 h after a cold start-up. After the cavity had warmed up the time required for each data point was about 2 h. A steady state energy balance around the cavity gave

$$Q_{\text{convection}} = P - Q_{\text{radiation}} - Q_{\text{back losses}} - Q_{\text{sidewall losses}} - Q_{\text{lead losses}}. \quad (1)$$

The input electrical power, P, was measured to within $\pm 1\%$ by using voltage taps across the heater and across a precision resistor in series with the cavity heaters. The resistance of the precision resistor was calibrated for temperature and was found to be 0.5 ohms for temperatures less than 40°C. The precision resistor was kept below 40°C by attaching the resistor to a heat sink and by blowing air over it.

The heat losses from the sides, ends, and bottom of the cavity, $Q_{\text{back losses}}$, were calculated from measured temperature differences across the foam insulation using Fourier's law. The heat losses from the inside of the cavity sidewalls due to forced convection to the cavity fluid, $Q_{\text{sidewall losses}}$, were calculated considering the acrylic walls as fins, each with a base temperature equal to the heater temperature. The unknown sidewall heat transfer coefficient was approximated, setting it equal to the average cavity bottom heat transfer coefficient. Lead losses, $Q_{\text{lead losses}}$ were determined considering all power leads and thermocouple leads as pin fins with base temperatures equal to the heater temperature. Treating each lead as a separate pin fin (the leads were actually bundled together) resulted in a conservatively high estimate for total lead losses. The radiation heat losses from the cavity opening, $Q_{\text{radiation}}$, were determined using a three-node Oppenheim radiative network. The cavity walls and heater were approximated as grey surfaces with emissivities of 0.9 and 0.05, respectively. Each wall was assumed isothermal at a temperature equal to the average of the values measured by thermocouples installed in the wall. The cavity mouth was approximated as a black body at the tunnel temperature. Calculated heat losses broken down into the individual terms of the energy balance are given in Table 2 for the cavities of aspect ratios $H/W = 1$ and 2 at the highest and lowest free-stream velocities tested. Total heat losses from the cavity varied from a high of nearly 35% of the input power for the large aspect ratio cavity ($H/W = 4$) to less than 10% of the input power for the small aspect ratio cavity ($H/W = 1$). More details of the heat loss calculations are in ref. [12].

In addition to heat transfer rates, heater temperatures were also recorded and averaged. The location of heater thermocouples are shown in Fig. 1(c). Consequently, the average convective heat transfer coefficient on the cavity bottom could be calculated from

$$\bar{h} = \frac{Q_{\text{convection}}}{A(\bar{T}_w - T_\infty)}. \quad (2)$$

The average Nusselt number based on the cavity height was then calculated from

$$\bar{Nu}_H = \frac{\bar{h}H}{k}. \quad (3)$$

Table 2. Heat losses (W)

	Aspect ratio			
	$H/W = 1$		$H/W = 4$	
Heat loss mechanism	$U_\infty = 5$ m s^{-1}	$U_\infty = 25$ m s^{-1}	$U_\infty = 5$ m s^{-1}	$U_\infty = 25$ m s^{-1}
Radiation (front face)	5.3	5.6	1.7	1.8
Back loss	5.2	5.4	2.9	2.8
Sidewall loss	7.7	12.1	6.7	9.7
Lead losses	9.8	10.2	3.0	2.9
Total loss	28.0	33.2	14.3	17.3
Power in	129.9	345.2	42.0	87.2
Net convective transfer	101.9	312.0	27.7	69.9

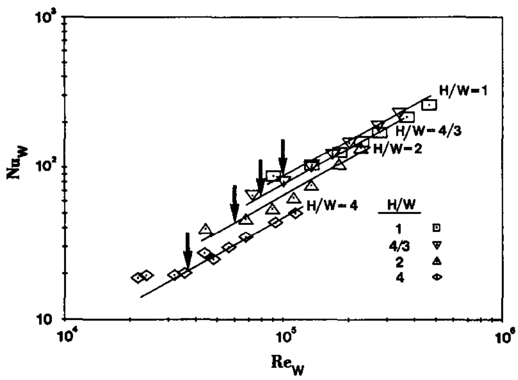

FIG. 2. Average heat transfer correlated using free-stream velocity and cavity width.

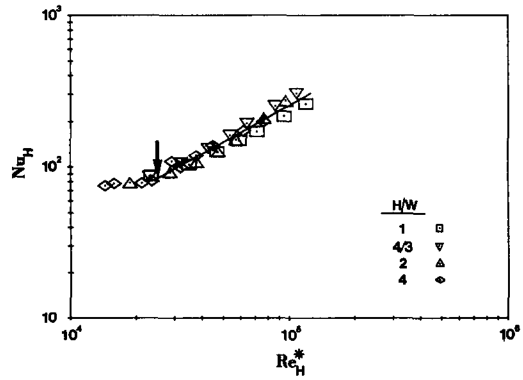

FIG. 3. Average heat transfer correlated using cavity velocity and cavity height.

RESULTS AND DISCUSSION

The average Nusselt number, $\overline{Nu}_W$, based on the cavity width is plotted in Fig. 2 vs Reynolds number, Re_W, based on cavity width and free-stream velocity for aspect ratios of 1.0, 4/3, 2, and 4. The average Nusselt number increases with Reynolds number and with decreasing cavity aspect ratio. The free-stream velocities investigated range from 5 to 25 m s^{-1}. Note that the data points for each aspect ratio fall along a different line, each line having a slope on the log–log plot of approximately 0.8. Note also evidence of perhaps a laminar–turbulent cavity flow transition indicated by arrows at the low Reynolds end of each line. There the slope abruptly changes from somewhat less than 0.5 to 0.8.

The same data is replotted in Fig. 3 as the average Nusselt number based on cavity height, $\overline{Nu}_H$, vs Reynolds number based on cavity height and cavity velocity, Re_H^*.

The cavity velocity, u_c, is determined by invoking conservation of momentum on a control volume whose boundaries coincide with the walls and floor of the cavity and the mixing layer bridging the mouth of the cavity. If the assumption is made that the recirculating fluid within the cavity is at some constant cavity velocity then a force balance gives

$$\tfrac{1}{2}\rho C_{\mathrm{ML}}(U_\infty - u_c)^2 WL = \tfrac{1}{2}\rho C_{\mathrm{BL}} u_c^2 (W+2H)L \quad (4)$$

where C_{ML} and C_{BL} are the friction coefficients for mixing layers and boundary layers, respectively. (See ref. [12] for model details.)

Solving for the cavity velocity

$$u_c = \frac{U_\infty}{1 + C\sqrt{(1+2(H/W))}} \quad (5)$$

where $C = \sqrt{(C_{\mathrm{BL}}/C_{\mathrm{ML}})}$ was set equal to 5/3, the value which gave the best data fit.

The usefulness of the new plot, Fig. 3, is seen by the way the data points for different cavity aspect ratios collapse onto a single curve. In addition, the apparent transitions from laminar to turbulent cavity flow which occurred at different values of Re_W for the various aspect ratios now occur at a single value of the Reynolds number based on cavity height and cavity velocity, Re_H^*

$$Re_H^* \sim 2.5 \times 10^4.$$

Using the plot of $\overline{Nu}_H$ vs Re_H^*, a single correlation representing all four cavity aspect ratios of 1.0, 4/3, 2.0, and 4.0 can be developed by performing a least-squares curve fit of the turbulent data. The resulting correlation is

$$\overline{Nu}_H = 0.0255 Re_H^{*0.8} \quad (6)$$

valid for

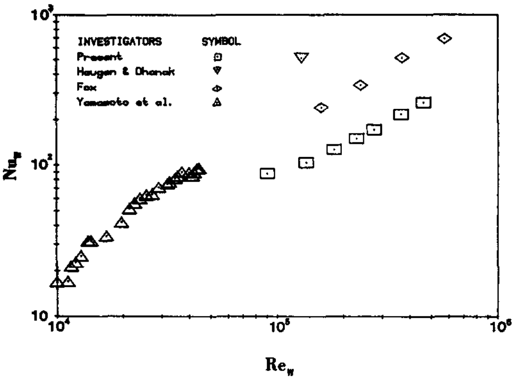

FIG. 4. Comparison of measured average heat transfer results.

$$2.5 \times 10^4 < Re_H^* < 1.2 \times 10^5$$

$$1 \leqslant H/W \leqslant 4$$

$$Pr \sim 0.71$$

$$\delta/W < 10^{-2}.$$

Equation (6) can be re-written in terms of the cavity width and free-stream velocity as

$$\overline{Nu}_w = 0.0255 Re_w^{0.8} (H/W)^{-0.2} \times (1 + C\sqrt{(1+2(H/W))})^{-0.8}. \quad (7)$$

This correlation represents the turbulent data to within an r.m.s. error of 8%. Since the correlation fits the data for both small and large aspect ratio cavities, three-dimensional effects do not appear to significantly affect the overall heat transfer correlation.

Equation (7) can be compared to a correlation developed by Yamamoto *et al.* [8] for turbulent flow over a bottom heated only cavity. Their correlation expressed in the present nomenclature is

$$\overline{Nu}_W = 0.390 Re_W^{0.5} (H/W)^{-0.27}. \quad (8)$$

Equation (8) with $H/W = 1$ is plotted in Fig. 4 along side the data from the present study for the same aspect ratio. Equations (7) and (8) differ mainly on the magnitude of the Reynolds number exponent and to a lesser extent on the aspect ratio dependence. Equation (8) is stated to be valid for turbulent flow (ref. [8]) although the Reynolds number exponent is more representative of laminar flow conditions. It should be noted that the aspect ratios investigated in ref. [8] were smaller than those investigated in the present study. In addition, the flow was three-dimensional in the present study.

Seban and Fox [4–6] using cavities with heated sidewalls and bottom in a turbulent cross-flow found the average heat transfer to scale as

$$\overline{Nu}_W \sim Re_W^{0.8}. \quad (9)$$

The ranges of free-stream Reynolds numbers and cavity aspect ratios investigated were similar to the present study although the oncoming boundary layer thickness was three times smaller. Data from Fox [6] for a cavity of aspect ratio $H/W = 1$ are plotted on Fig. 4. The data from Fox [6] follows the same slope as the present data but are significantly different in magnitude possibly because in the Fox study all sides of the cavity were heated and the flow was two dimensional. For equation (9) to resemble equation (7), the last two terms in equation (7) would have to equal a constant, and the data do not appear to support such an interpretation.

Haugen and Dhanak [7] also tested bottom and side heated cavities. They reported the correlation

$$\overline{Nu}_W = 0.365 Re_W^{0.75} \times Pr[3(1+(W/H))]^{-0.5} (\delta/W)^{-0.14} \quad (10)$$

where δ was the thickness of the oncoming boundary layer just prior to separation. Here too, the ranges of Reynolds number and aspect ratio were similar to the present study. Data from ref. [7] for an aspect ratio of unity is also shown in Fig. 4. A value of $\delta/W = 10^{-2}$ was used in Fig. 4. The results of ref. [7] are higher than the present results because of sidewall heating and because of the difference in cavity flow conditions (δ is on the order of the cavity height).

Local air temperatures in the largest cavity tested, width of 304.8 mm (12 in), were also measured. Centerline temperature profiles are shown in Fig. 5. In Fig. 5, temperature profiles are shown for a station near the upstream wall, in the middle of the cavity, and near the downstream wall. Air temperatures were found to be highest near the upstream wall where lowest heat transfer coefficients were also expected. In the center of the cavity, the temperature profile is nearly uniform except very near the bottom heater surface. As expected, air temperatures were found to be lowest near the downstream wall where local heat transfer coefficients were thought to be highest due to reattachment (the same result was obtained by Gooray *et al.* [10]).

SUMMARY

In the present study, average heat transfer coefficients for a bottom heated only open surface cavity were measured for a range of Reynolds numbers and cavity aspect ratios. A Nusselt number correlation based on a calculated cavity velocity and cavity height as the characteristic dimension was developed, and found to represent the turbulent data with an r.m.s. error of 8%

$$\overline{Nu}_H = 0.0255 Re_H^{*0.8}$$

$$2.5 \times 10^4 < Re_H^* < 1.2 \times 10^5$$

$$1 \leqslant H/W \leqslant 4$$

$$Pr \sim 0.71$$

$$\delta/W < 10^{-2}.$$

An apparent transition from laminar to turbulent flow

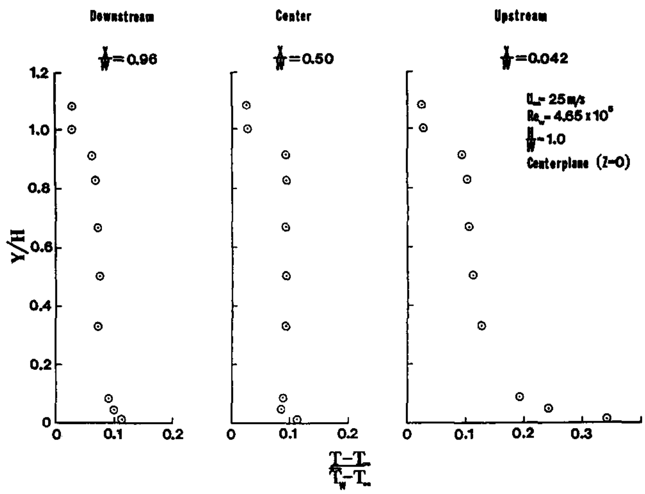

FIG. 5. Temperature profiles for the cavity of aspect ratio $H/W = 1$.

was found to occur at approximately

$$Re_H^* \sim 2.5\times 10^4.$$

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CONVECTION THERMIQUE FORCEE TURBULENTE A PARTIR DE LA BASE CHAUFFEE D'UNE CAVITE OUVERTE

Résumé—On présente les résultats d'une recherche expérimentale sur la convection thermique forcée à partir de la base chauffée d'une cavité ouverte. Le nombre de Reynolds, basé sur la largeur de la cavité, varie depuis 2.10^4 jusqu'à 4.10^5. On considère quatre cavités avec des rapports de forme (hauteur/largeur) de 1, 4/3, 2 et 4. Les coefficients moyens de transfert sur la base sont mesurés pour les cavités. On obtient une formule reliant le nombre de Nusselt, basé sur la largeur de la cavité, au nombre de Reynolds basé aussi sur la largeur et sur une vitesse de fluide calculée. Cette formule s'accorde avec les données dans une marge dont l'écart-type est 8%.

TURBULENTE ERZWUNGENE KONVEKTION IN EINEM BODENBEHEIZTEN OFFENEN HOHLRAUM

Zusammenfassung—Die Ergebnisse einer experimentellen Untersuchung der erzwungenen Konvektion in einem bodenbeheizten offenen Hohlraum werden vorgestellt. Die auf die Hohlraumbreite bezogene Reynolds-Zahl liegt zwischen 2 10^4 und 4 10^5. Vier unterschiedliche Höhen/Breiten-Verhaltnisse werden untersucht: 1, 4/3, 2 und 4. Die mittleren Wärmeübergangskoeffizienten am Boden des Hohlraums werden gemessen. Eine Korrelation wird ermittelt zwischen der auf die Höhe des Hohlraums bezogenen Nusselt-Zahl und der Reynolds-Zahl, welche ebenfalls auf die Höhe und auf eine im Hohlraum berechnete Fluidgeschwindigkeit bezogen ist. Diese Korrelation beschreibt die Daten mit einer Standardabweichung von 8%.

ТУРБУЛЕНТНЫЙ ПЕРЕНОС ТЕПЛА ИЗ ОТКРЫТОЙ НАГРЕВАЕМОЙ СНИЗУ ПОЛОСТИ ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ

Аннотация—Приводятся результаты экспериментального исследования переноса тепла из нагреваемой снизу открытой полости при вынужденной конвекции. Числа Рейнольдса, рассчитанные по ширине полости, изменялись в диапазоне от 2×10^4 до 4×10^5. Исследовались четыре полости с отношением сторон (высота/ширина), равным 1, 4/3, 2 и 4. Измерены средние значения коэффициента теплообмена на дне полостей. Получено соотношение между числом Нуссельта, определенным по высоте полости и числом Рейнольдса, построенном по высоте полости и расчетной скорости жидкости в полости. Соотношение согласуется с данными эксперимента в пределах среднеквадратичной ошибки в 8%.