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Thermal description of pseudosteady-state natural convection inside a vertical cylinder

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Abstract—The heat transfer correlation, minimum temperature, and mean temperature are presented for pseudosteady-state natural convection heat transfer to a fluid inside a vertical cylinder. The SIMPLER numerical method was used for calculation in the range $0.25 < H/D < 2$, $Ra < 10^7$, and $Pr = 7$. This range includes conduction to weak turbulence. The overall heat transfer for the convection-dominated range was found to be correlated by

$$Nu = 0.519Ra^{0.255}$$

where the temperature difference for both the Nusselt and Rayleigh numbers was the center temperature minus the inside wall temperature. Correlations using other temperature differences are also presented and provide a method for prediction of the mean temperature, minimum temperature, or center temperature by knowing any one of them.

INTRODUCTION

NATURAL convection in enclosures commonly occurs in technological applications. This phenomenon plays an important role in such diverse applications as cooling of cans of beverages, air conditioning of buildings, design of electronic components, emergency cooling of nuclear reactors, temperature stratification in cryogenic fuel tanks, and cooling (heating) chemical reactors generating (consuming) heat uniformity, to name but a few. In these cases, the fluid is driven by density variations in a body-force field, and the flow patterns depend critically on the applied heating conditions and the boundaries. These systems are governed by the Navier–Stokes equations, but, due to the complexity of the equations and the coupling of the dependent variables, general analytical solutions are still not possible. Most previous research efforts have been based on experimental work and, more recently, on numerical approaches. The purpose of this work was to investigate natural convection inside a vertical cylinder where the wall temperature was increasing at the same rate as the interior temperatures—the pseudosteady-state condition. The results also apply to the case of uniform heat generation at steady state.

Previous studies include a step change in the wall temperature of cylinders with H/D ratios from 0.75 to 2.0 by Evans and Stefany [1]. For Rayleigh numbers (based on initial temperature difference and diameter) from 6×10^5 to 6×10^9, the heat transfer was correlated by $Nu = 0.55Ra^{0.25}$. Natural convection heat transfer of a uniformly heat-generating fluid is important in reactor design. While this system is difficult to achieve experimentally, it can be numerically simulated by an uniform internal heat source or by pseudosteady state. In Murgatroyd and Watson's experiments [2], a solution of HCl ($3 < Pr < 9$) was heated by passing an alternating current between two copper electrodes, one at each end of the cylinder. Cooling water around the outside of the cylinder was used to keep the wall temperature constant and uniform. Modified Rayleigh numbers (based on rate of heat input per unit volume) from 2×10^3 to 3×10^6 were used (corresponding to laminar flow). Pseudosteady-state techniques were used by Lin and Akins [3] to study the flow pattern and overall heat transfer coefficients in cubical enclosures. Several complicated flow patterns were observed photographically for the unsteady state and the pseudosteady-state conditions.

Efforts to solve pseudosteady-state (or uniform heat generation) natural convection inside vertical cylinders with moderate height-to-diameter ratios using numerical methods have received limited attention. Seemingly there has been no previous work using primitive variables to solve this problem. The only solution available was presented by Kee *et al.* [4], who used the streamfunction–vorticity method to formulate the uniform heat generating problem. An instrumented cylinder containing radioactive tritium gas was used to demonstrate experimental and analytical agreement. Their work provides a valuable comparison for the low Rayleigh number results of this work.

Compared with experimental investigations, the proper numerical method can offer advantages of low cost, high speed, the ability to provide complete information, and ease of application to different conditions. Numerical results require the solution of the Navier–Stokes and energy equations, which are highly nonlinear and inseparably connected. For the case of natural convection, the temperature difference is the driving force for flow, therefore both fields are coupled and must be calculated simultaneously. This substantially increases the difficulty of calculation compared to forced convection problems where the flow field is usually determined prior to the temperature field. In addition to the complexity of the equations, a

NOMENCLATURE

A	area
C_p	heat capacity
D	diameter
f	H/D function, equation (10)
g	acceleration of gravity
Gr	Grashof number
Gr'	modified Grashof number
h	heat transfer coefficient
H	height
k	thermal conductivity
L	characteristic length, Df
$\mathbf{n}$	normal vector
Nu	Nusselt number based on L
p	actual pressure
p_h	hydrostatic pressure, $\pi - \rho_0 gz$
P	$p + \rho_0 gz$
Pr	Prandtl number
q	heat flux
q_v	volumetric heat effect
Q_v	$q_v/(\mu C_p \Delta T L^2)$
r	radial position
Ra	Rayleigh number based on L
t	time
T	temperature
v	velocity
z	axial position.

Greek symbols

α	thermal diffusivity
β	coefficient of thermal expansion
μ	dynamic viscosity
ν	kinematic viscosity
π	reference pressure
ρ	density
∇^2	two-dimensional Laplacian operator.

Subscripts

c	center
min	minimum
0	at initial temperature
r	radial-component property
w	wall
x	local variable
z	axial-component property.

Superscripts

$''$	based on rate of temperature change
$*$	based on diameter.

wide range of parameters must also be dealt with. In usual applications, the Prandtl number covers about five orders of the magnitude and the Rayleigh number may span 10 orders of the magnitude. These facts make the calculation algorithm and correlation method not only difficult, but parameter dependent. The complexity of natural convection problems has made them a challenging task for modern numerical techniques.

FORMULATION OF THE PROBLEM

The system studied consisted of a fluid completely enclosed in a vertical cylinder. Initially the entire system was motionless and at a uniform temperature. Suddenly, the temperature of the cylinder walls (top, bottom and side) underwent a step change of size ΔT. A transient period started, during which the fluid temperature and velocity changed with time due to heat conduction through walls and natural convection inside the cylinder. Following the step change, the wall temperature was dynamically adjusted to maintain a constant wall-to-center temperature difference equivalent to the step change.Calculations were carried out through the transient period until pseudosteady state was reached, at which time the relative temperature of any two points in the fluid was time invariant.

To formulate this problem, it was assumed that: (1) all variables were θ-direction independent; (2) the fluid was viscous and incompressible; (3) all dissipation terms were negligible; and (4) all physical properties were constant except for the density in buoyancy term, which was expressed as a linear function of temperature (the Bossinesq assumption). In equation form

$$\rho = \rho_0[1-\beta(T-T_0)]. \tag{1}$$

If we define pressure as

$$P = p-(p_h-\pi)$$
$$= p+\rho_0 gz \tag{2}$$

where p is the actual pressure at any point in the fluid, p_h is the hydrostatic pressure of a column of fluid at the reference temperature, π is a reference pressure at the bottom of the cylinder, then P is a pressure term without hydrostatic influence. The governing equations can be expressed in the following dimensionless form†:

$$\frac{\partial(rv_r)}{\partial r} + \frac{\partial(rv_z)}{\partial z} = 0 \tag{3}$$

$$\frac{\partial v_r}{\partial t} + v_r\frac{\partial v_r}{\partial r} + v_z\frac{\partial v_r}{\partial z} = -\frac{\partial P}{\partial r} + \nabla^2 v_r - \frac{v_r}{r^2} \tag{4}$$

$$\frac{\partial v_z}{\partial t} + v_r\frac{\partial v_z}{\partial r} + v_z\frac{\partial v_z}{\partial z} = -\frac{\partial P}{\partial z} + \nabla^2 v_z + Gr\,T \tag{5}$$

$$\frac{\partial T}{\partial t} + v_r\frac{\partial T}{\partial r} + v_z\frac{\partial T}{\partial z} = \frac{1}{Pr}\nabla^2 T. \tag{6}$$

†To make the equations dimensionless the following characteristic variables are used, D for length, D^2/ν for time, ν/D for velocity, $\rho(\nu/D)^2$ for pressure, and the dimensionless temperature is defined by $(T-T_0)/\Delta T$.

The dimensionless initial conditions which apply in this case are

$$v_r = v_z = 0 \quad \text{when} \quad t = 0, \quad T = 0 \tag{7}$$

and the dimensionless boundary conditions are

$$v_r = v_z = 0, \quad T = T_c + 1 \text{ at solid boundaries,}$$

$$v_r = \frac{\partial v_z}{\partial r} = \frac{\partial T}{\partial r} = 0 \text{ at the axial center.} \tag{8}$$

The distribution of the local heat flux may be evaluated from Fourier's law, $q = -k(\partial T/\partial \mathbf{n})$. The local and the overall Nusselt numbers (based on diameter) can be expressed as

$$Nu_x^* = \frac{h_x D}{k} = -\frac{\partial T}{\partial \mathbf{n}} \quad \text{(local)} \tag{9}$$

$$Nu^* = \frac{1}{A}\int_A Nu_x^* \, dA \quad \text{(overall)} \tag{10}$$

where $\mathbf{n}$ and A are also dimensionless.

In defining the Rayleigh and Nusselt numbers, several characteristic lengths were evaluated. The following gave the best correlation:

$$L = 6 \times \frac{\text{volume}}{\text{surface area}} = D\left[\frac{3(H/D)}{1+2(H/D)}\right] = Df. \tag{11}$$

This choice of L also had the following advantages: (1) $L = D$ when $D = H$; (2) L is characteristic of both D and H–when their magnitude is close, more weight is put on the smaller of the two, which is intuitively realistic; (3) L depends only on the smaller dimension as one becomes very large compared to the other. The Rayleigh number and the Nusselt number using L and D can be related as

$$Nu = f Nu^* \tag{12}$$

$$Ra = f^3 Ra^*. \tag{13}$$

The overall Nusselt number can also be calculated by the rate of temperature change at any position (usually the center point) when the system is at pseudosteady state

$$Nu'' = \frac{Pr f^2}{6}\left(\frac{dT_c}{dt}\right). \tag{14}$$

At any time, Nu^* depends only on the temperature field and its determination involves the entire calculation scheme as well as estimation of the temperature gradients at the walls. On the other hand, Nu'' depends only on the dynamic temperature of any point. Therefore, the calculation of Nu^* and Nu'' are somewhat independent and their difference serves to check the precision of the calculations. Constancy of Nu'' indicates that pseudosteady state has been achieved.

In general, a modified Grashof number based on the heat effect

$$Gr' = \frac{q_v g \beta L^5}{k \nu^2} \tag{15}$$

is used as a parameter in the uniform heat generation systems instead of the Grashof number, Gr, based on a characteristic temperature difference. These two systems are related by

$$Gr' = Q_v Ra \tag{16}$$

where Q_v is the dimensionless heat effect.

Numerical method

The equations above, with the associated initial and boundary conditions, provide a complete mathematical description of the problem. These equations were solved by the Semi-Implicit Method for Pressure-Linked Equations, Revised (SIMPLER) algorithm [5]. The essence of this algorithm was to successively correct the pressure field so as to satisfy the boundary conditions and the continuity equation with the velocities calculated via the momentum equations. Recently, this method has become a powerful tool for solving fluid flow problems, and numerous papers based on the method have been published. However, Pollard and Thyagaraja [6] pointed out that this method appears to encounter convergence difficulties when the momentum equations are driven by body forces, even if no physical instability occurs. In the numerical calculations of this work, difficulties with convergence were also experienced. A control algorithm was set up such that the number of iterations required for each equation and rate of primitive variable change were used to modify dynamically the convergence criteria of each equation, the time step sizes, and (more importantly) the under-relaxation coefficients. For each run, the calculation proceeded by marching through transient steps until pseudosteady state was achieved, which was indicated by no change in any temperature differences, no change in velocities, and no change in mass flux for each control volume.

The calculation parameters were Rayleigh number and the height-to-diameter ratio. Cases with the Rayleigh number up to 10^7 and H/D of 0.125, 0.25, 0.5, 1 and 2 were calculated. A Prandtl number of 7 was used, corresponding to water, however, the influence of the Prandtl number was not expected to be significant since it is generally accepted that steady-state natural convection can be described by geometric parameters and the Rayleigh number alone when the Prandtl number is greater than 5. To further support this, two cases (with the Prandtl number equal to 7 and 180, $Ra = 10^5$ and $H/D = 1$) were calculated for comparison, and the resulting temperature profiles coincided exactly. Finite-difference grids of 10×10, 10×28 and 19×19 were employed. Considerably finer grids were applied to the regions close to the boundaries to reduce errors due to steep gradients in those areas, and to increase the accuracy of heat flux calculation. The results were determined to be grid independent by comparisons with solutions obtained using different grids.

RESULTS

Heat transfer

Figure 1 shows the relationship of the overall Nusselt number and the Rayleigh number. In defining the Nusselt number and the Rayleigh number, a characteristic temperature difference of $(T_w - T_c)$ was used since the center temperature is easy to measure practically. There are, of course, other characteristic temperature differences that could be used. For example, the volumetric mean temperature, although usually difficult to determine, is an indication of the entire temperature field and the difference between it and the wall temperature should be a good characteristic temperature difference of the whole system. For some applications, such as the design of catalytic packed bed reactors, the extreme temperature and its location is important. As a result, a characteristic temperature difference based on the minimum (or maximum) temperature might be useful. To provide information of these different temperatures, $(T_w - T_c)$, $(T_w - T_{mean})$, and $(T_w - T_{min})$ were used to define the Rayleigh number in the correlation with the Nusselt number.

The low Rayleigh number region of Fig. 1 shows horizontal lines for various H/D ratios, indicating conduction-dominated heat transfer. When conduction is the only significant mode, the Nusselt number depends only on the H/D ratio. For higher Rayleigh numbers, the results fall on straight lines with slopes of approx. 1/4. This suggests a correlation of the form $Nu = a \cdot Ra^b$ for each H/D ratio. Table 1 lists the values of a, b, and the correlation coefficients for H/D ratios of 1/4, 1/2, 1 and 2, and for Rayleigh numbers based on different characteristic temperature differences. However, only $(T_w - T_c)$ was used in defining the Nusselt number for all cases since the selection of its temperature difference is somewhat arbitrary and a reference is necessary for comparison purposes. Since the data for all H/D ratios were so nearly the same, one overall correlation was made and this is shown at the bottom of Table 1. Table 1 can be used to calculate the center temperature, volumetric mean temperature, and the minimum temperature by knowing any one of them. Note that the correlations for the three temperature difference definitions are very similar as far as the ability to fit the data is concerned. The coefficients using $(T_w - T_{mean})$ are slightly better than others in all cases of H/D ratios, and the worst correlation is observed for $H/D = 1.4$. For $H/D = 1/4$ and for Ra between 8×10^4 and 2×10^5 there were two solutions at each Rayleigh number. As H/D decreases, the importance of the heat transfer through the sidewalls is also decreased compared to that through the top and bottom, and the problem evolves into that for horizontal plates. The two solutions were considerably different in detail (i.e. flow patterns, etc.) which showed up as a 10–20% difference in the Nusselt number. The double solutions are believed to be the transition between the two heat transfer processes; therefore, extrapolation of these results to smaller H/D ratios needs to be handled with care.

Table 1. Convective heat transfer correlation coefficients for $Nu = a \cdot Ra^b$

H/D	Temperature difference in Ra	a	b	Correction coefficient R^2
1/4	wall—mean	0.923	0.215	0.913
	wall—center	0.654	0.237	0.892
	wall—minimum	0.722	0.225	0.905
1/2	wall—mean	0.706	0.235	0.999
	wall—center	0.514	0.254	0.999
	wall—minimum	0.519	0.249	0.999
1	wall—mean	0.682	0.243	0.992
	wall—center	0.517	0.259	0.990
	wall—minimum	0.489	0.258	0.994
2	wall—mean	0.564	0.254	0.995
	wall—center	0.371	0.281	0.988
	wall—minimum	0.381	0.272	0.993
Overall	wall—mean	0.727	0.234	0.971
	wall—center	0.519	0.255	0.963
	wall—minimum	0.536	0.248	0.967

It is worthwhile noting that the selection of the characteristic length is critical in making the overall correlation possible. Although the slopes will be the same, other characteristic lengths will result in different intercepts. The characteristic length used here provided the best overall correlation.

Figure 2 shows the Nusselt number for conduction cases and the minimum Rayleigh number for which the overall coefficients in Table 1 may be used. The minimum of the two curves in Fig. 2 occurs at $H/D \sim 1$ which is expected since the surface area/volume ratio is a minimum when $H/D = 1$. When we speak of 'conduction', it is not our intention to relate these results to the onset of flow, since by the SIMPLER method the velocity field is nonzero even for an infinitesimal driving force. It is better to think of this as the region where heat transfer by conduction is significantly larger than that by convection.

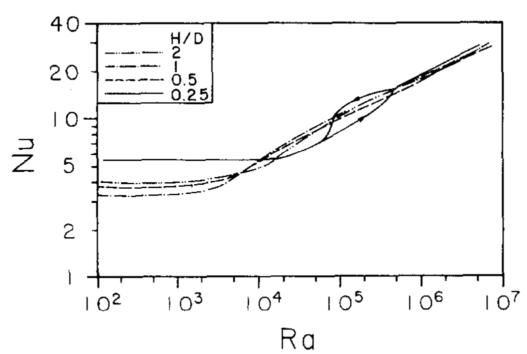

FIG. 1. Heat transfer correlation for vertical cylinders: $\Delta T = (T_w - T_c)$.

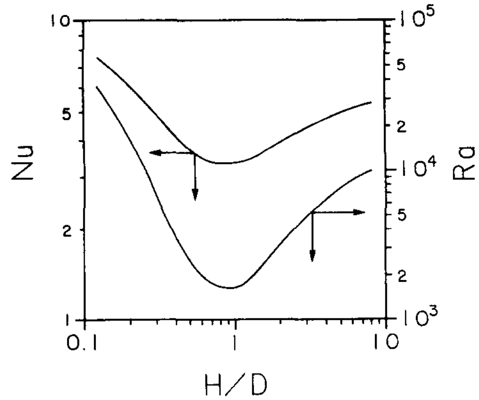

FIG. 2. Nusselt number for conduction and limiting Rayleigh number.

Minimum temperature

To make the comparisons possible, temperatures are normalized to be zero at the center and unity at the wall. Before any convection effects are significant, the minimum temperature inside the cylinder (denoted by T_{min}) is zero and is at the center of the cylinder. Figure 3 presents the locations of T_{min} as a function of Rayleigh number of H/D ratios of 1/2, 1 and 2. The ordinate is $z/(H/D)$, which is equivalent to the fraction of the total height, and the parameter is the Rayleigh number. According to Fig. 3, the minimum temperature is located in either one of two regions; on the axis or near the side toward the bottom. The shift from the first region to the second is sudden† and the points in the second region do not correspond exactly to the circulation center in that region. Figures 4 and 5 also show the T_{min} locations as a function of the Rayleigh number. As shown in these figures, the radial location is independent of the Rayleigh number below 10^4, while most of the axial changes occur in this low Rayleigh number range. As the Rayleigh number becomes

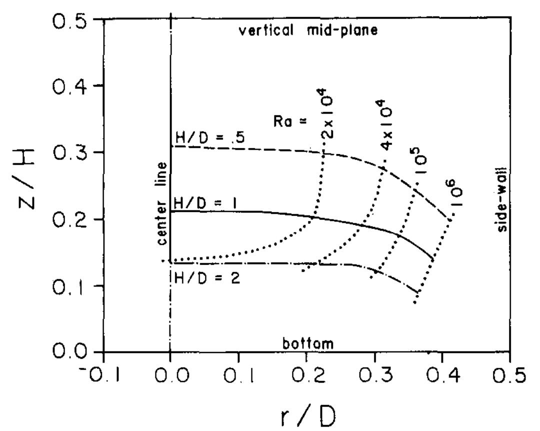

FIG. 3. Minimum temperature locus inside a vertical cylinder.

† Words such as 'sudden' and 'slow' which are usually associated with time are used here to describe the changes in the position of the minimum temperature with Rayleigh number.

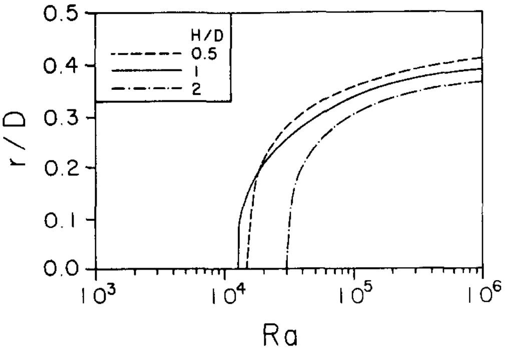

FIG. 4. Radial location of the minimum temperature.

greater than 10^4, the radial location moves quickly away from the center, then moves moderately while movement in the axial direction is slow. Intuitively, one might assume that the minimum temperature of a pseudosteady-state system is about zero (center temperature) and located close to the center. Figure 6 clearly shows the discrepancy in that line of reasoning. As natural convection effects increase, the minimum temperature can be as much as 50% lower than the difference between wall and center. Since further increases in the Rayleigh number drive the T_{min} location closer to the wall and the bottom, they increase the temperature gradient in that region which, in turn, causes T_{min} to rise. This is shown in Fig. 6 as the minimum of T_{min}. In the case of cooling, these locations would be 'hot spots'. Information about the magnitude and the location of hot spots is important in the design of catalytic reactors and the arrangement of electronic components.

Volumetric mean temperature

Figure 7 presents the volumetric mean temperature as function of the Rayleigh number. Starting from the conduction-dominated region, the mean temperature decreases monotonically to approx. 0.2 as the Rayleigh number increases. The mean temperature is most sensitive to the Rayleigh number changes between 10^4 and 10^5. This is due to the creation of a large negative temperature region in this Rayleigh number range. In general, the wall–mean temperature difference is a good

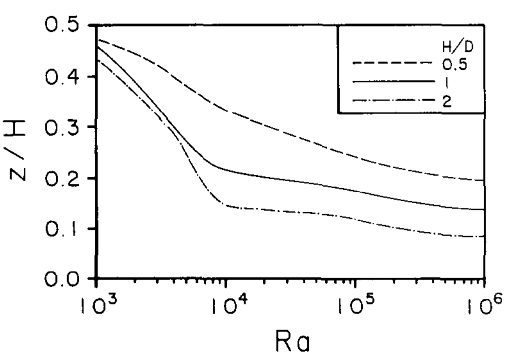

FIG. 5. Axial location of the minimum temperature.

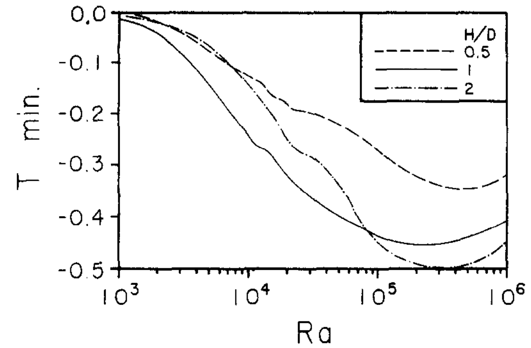

FIG. 6. Magnitude of the minimum temperature: $T_c = 0$.

characteristic temperature for heat transfer. According to this figure, the temperature difference in the convection region can easily be twice that in the conduction-dominated region. Natural convection causes both an increase in the temperature driving force and a decrease in the path length for heat transfer, i.e. the minimum temperature moves closer to the walls. Both effects increase the average gradient at the wall and, therefore, the rate of heat transfer.

DISCUSSION

During numerical calculations in this work, the low temperature range (usually between −0.2 and −0.4) became unstable as the Rayleigh number increased to about 10^7. This is the transition region from laminar flow to weak turbulence, or chaotic thermal convection [7]. The weak turbulence results in fluid mixing and an increase in the minimum temperature, this is shown in Fig. 6. In spite of the existence of a chaotic region in the body of the cylinder, the temperature distribution at the boundary region is still well stratified, therefore the temperature gradient (which is essentially the local Nusselt number) can still be calculated accurately, and the heat transfer correlation of this work can be used up to the Rayleigh number of 10^7.

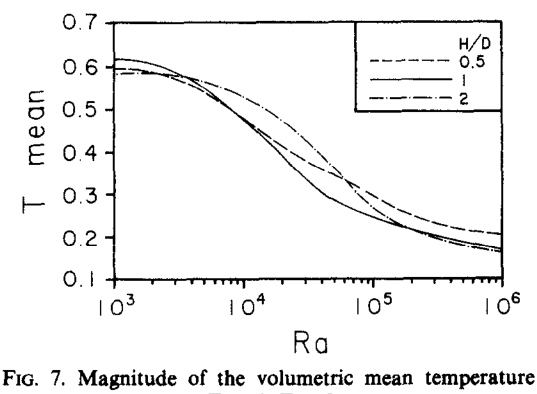

FIG. 7. Magnitude of the volumetric mean temperature: $T_w = 1$, $T_c = 0$.

Table 2. Percentage error of the Nusselt number

Ra	10^2	10^4	6×10^4
Calculation	0	1	2
Experimental	0	8	5

The vertical cylinder heat transfer results of Kee [4] were converted according to equation (16) to make them comparable. Table 2 lists the percentage difference in the Nusselt numbers at three Rayleigh numbers compared to that of Kee. Larger errors were observed in comparison to his experimental results, in which he assumed the inside wall temperature to be uniform and the same as the outside wall temperature. This assumption can cause a slightly lower Nusselt number. Actually, our results are between Kee's calculations and experiments.

The difference between *Nu* and *Nu″* was usually much less than 1% which indicates that the calculation scheme, the temperature gradient estimation, and the grid size were satisfactory.

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DESCRIPTION DE LA CONVECTION THERMIQUE NATURELLE PSEUDO-STATIONNAIRE DANS UN CYLINDRE VERTICAL

Résumé—L'expression du transfert thermique, la température minimale et la température moyenne sont présentées pour la convection naturelle pseudo-stationnaire d'un fluide dans un cylindre vertical. La méthode numérique SIMPLER est utilisée dans le domaine : $0{,}25 < H/D < 2, Ra < 10^7$ et $Pr = 7$. Ce domaine va de la conduction à la turbulence faible. Le transfert thermique global pour le domaine est exprimé par

$$Nu = 0{,}519\, Ra^{0{,}255}$$

où la différence de température pour les nombres de Rayleigh et de Nusselt est la température du centre moins la température de la paroi interne. Des formules utilisant d'autres différences de température sont présentées et on donne une méthode pour le calcul de la température moyenne, de la température minimale ou de la température du centre, en connaissant l'une quelconque d'entre elles.

QUASISTATIONÄRE NATÜRLICHE KONVEKTION IN EINEM SENKRECHTEN ZYLINDER

Zusammenfassung—Für den Fall des Wärmeübergangs bei quasistationärer natürlicher Konvektion an ein Fluid in einem senkrechten Zylinder werden die Wärmeübergangsbeziehung, die Minimal- und die Mitteltemperatur vorgestellt. Die numerische Methode nach SIMPLER wurde für Berechnungen im Bereich $0{,}25 < H/D < 2, Ra < 10^7$ und $Pr = 7$ angewendet. Dieser Bereich schließt die Wärmeleitung sowie schwache Turbulenz mit ein. Es wurde herausgefunden, daß der mittlere Wärmeübergang bei natürlicher Konvektion mit $Nu = 0{,}519 \cdot Ra^{0{,}255}$ korreliert werden kann. Dabei erstreckt sich sowohl für die Nusselt- als auch für die Rayleigh-Zahl die Temperaturdifferenz zwischen der Rohrmitte und der inneren Wand. Beziehungen, die auf anderen Temperaturdifferenzen basieren, werden ebenfalls vorgestellt und ermöglichen ein Verfahren zur Berechnung der Mittel- und Minimaltemperatur oder derjenigen in der Rohrmitte bei Kenntnis einer dieser Temperaturen.

ПСЕВДОСТАЦИОНАРНАЯ ЕСТЕСТВЕННАЯ КОНВЕКЦИЯ ВНУТРИ ВЕРТИКАЛЬНОГО ЦИЛИНДРА

Аннотация—Для псевдостационарного естественноконвективного теплопереноса к жидкости внутри вертикального цилиндра предложены соотношения для расчета теплообмена, минимальной и средней температуры. Численный метод SIMPLER использовался в диапазоне: $0{,}25 < H/D < 2$, $Ra < 10^7$ и $Pr = 7$ и от режимов чистой теплопроводности до слабой турбулентности. Найдено, что средний теплоперенос для преобладаний конвекции описывается следующим соотношением:

$$Nu = 0{,}510\, Ra^{0{,}255},$$

где как для числа Нуссельта, так и для числа Рэлея принималась разность температур в центре и на внутренней поверхности стенки. Приведены также выражения, полученные для других разностей температур, и разработана методика расчета средней, минимальной или температуры в центре при условии, что одна из них известна.