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Laminar natural convection of cold water enclosed in a horizontal annulus with mixed boundary conditions

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Abstract—A computational analysis of steady laminar natural convection of cold water within a horizontal annulus with constant heat flux on the inner wall and a fixed temperature on the outer surface has been undertaken to explore the occurrence of density inversion of water and its effects on the flow and temperature fields. Results are generated for three values of radius ratio over various ranges of the modified Rayleigh number and density inversion parameter. For the flow circumstances considered, the occurrence of density inversion is found to be mainly dependent on the density inversion parameter. With a negative inversion parameter, the increase of either the modified Rayleigh number or the radius ratio results in a more pronounced effect of density inversion on the flow and temperature fields in the annulus.

INTRODUCTION

THERE has been considerable recent interest in buoyancy-induced recirculating flow phenomena, because of the importance of such phenomena encountered in a wide range of occurrences in environment and technology such as thermal energy storage systems, aircraft cabin insulation, underground transmission cables, cooling of electronic equipment, solar collector design, reactor-spent fuel cooling, pollution control in lakes and estuaries, and many others. The annulus enclosed by two horizontal cylinders is one of the fundamental configurations for which buoyancy-driven convection problems have been extensively investigated. Comprehensive reviews of the works concerning this configuration were given by Kuehn and Goldstein [1] and more recently by Rao *et al.* [2]. In the case of buoyancy-driven convection in cold water, the recirculating flow can be considerably affected by the occurrence of density inversion associated with the temperature about 4°C. Several studies have addressed the problem of cold water natural convection within a horizontal annulus. Seki *et al.* [3] studied experimentally the effect of density inversion on the natural convection flow patterns and heat transfer of cold water between two horizontal isothermal cylinders. Nguyen *et al.* [4] and Vasseur *et al.* [5] conducted theoretical analysis on a similar problem, respectively. Recently, ref. [6] further performed a numerical study on the combined effect of density inversion and eccentricity of the inner cylinder on the natural convection heat transfer of cold water in a horizontal annulus. Results from all the previous works demonstrate that the occurrence of density inversion can cause the reversals in the direction of buoyancy force and velocity field, thus drastically influencing the heat transfer characteristics of cold water in the annulus.

However, it appears that the existing literature on the problem of cold water natural convection in the horizontal annulus are rather limited to the one subject to isothermal boundary conditions; little or no information is available for other types of thermal boundary conditions of engineering interest. Specifically, the situation of the annulus with mixed boundary conditions, the outer cylinder held at a fixed temperature, T_o, and the inner cylinder subject to a uniform heat flux, q_i, as depicted in Fig. 1, is of fundamental interest in connection with the problem concerning natural convection dominated melting of ice around a heat source of constant heat flux. This has motivated the present study. A similar mixed boundary condition problem has been considered by Vasseur *et al.* [7] for buoyant water flow in a horizontal annular porous layer.

In this paper, a computational analysis of the problem described above is presented. The details of flow and temperature fields were examined to provide additional insight into the occurrence of density inversion under the flow circumstance considered. Also, a parametric response of the problem in heat transfer and flow patterns were discussed in terms of the underlying physical processes involved.

ANALYSIS

The flow under consideration shown in Fig. 1 is assumed to be two-dimensional, steady and laminar. In addition, the following assumptions have been made.

(1) The thermophysical properties of water except density are temperature independent.

NOMENCLATURE

a	exponent in the density equation
g	gravitational acceleration
k	thermal conductivity
L	annular gap, $(r_o - r_i)$
Nu	Nusselt number
Pr	Prandtl number
q_i	uniform heat flux on the inner cylinder
r^+	radial coordinate
r	dimensionless radial coordinate, r^+/L
Ra^*	modified Rayleigh number, $g\, rsp\, L^{(3+a)} q^a/(\nu \alpha k^a)$
r_i	radius of inner cylinder
r_o	radius of outer cylinder
RR	radius ratio, r_o/r_i
rsp	coefficient in density equation
T	temperature.

Greek symbols

α	thermal diffusivity
γ	density inversion parameter
θ	dimensionless temperature, $(T - T_o)k(q_i L)$
θ_m	dimensionless temperature parameter, $(T_m - T_o)k/(q_i L)$
ν	kinematic viscosity
ρ	density
ϕ^+	angular coordinate
ϕ	dimensionless angular coordinate, ϕ^+/π
ψ^+	stream function
ψ	dimensionless stream function, ψ^+/α
ω^+	vorticity
ω	dimensionless vorticity, $\omega^+ L^2/\alpha$.

Subscripts

i	inner cylinder surface
m	physical property at $T_m = 4.029325°C$
o	outer cylinder surface.

Other symbol

∇^2	Laplace operator.

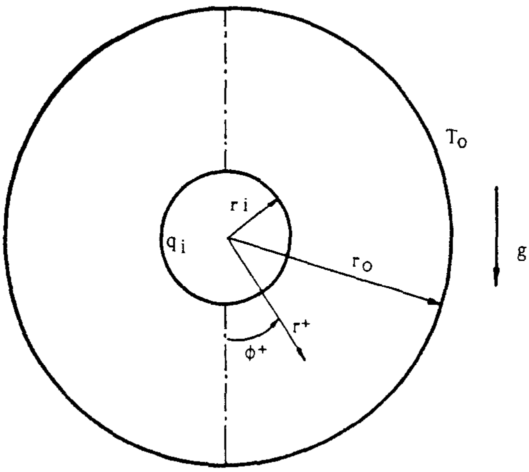

FIG. 1. Flow configuration and coordinate system.

(2) The Boussinesq approximation is employed such that the variation of density with temperature can be neglected except for the buoyancy force terms.

(3) The viscous dissipation and compressibility effects are negligible.

Moreover, symmetry with respect to the vertical mid-plane through the axis of the cylinders is assumed in order to reduce the computational effort.

As for the density–temperature relation of cold water, the following relation by Gebhart and Mollendorf [8] is used:

$$\rho(T) = \rho_m[1 - rsp|T - T_m|^a] \tag{1}$$

where $\rho_m = 999.9720$ kg m^{-3}, $rsp = 9.297173 \times 10^{-6}$ $(°C)^{-a}$, $T_m = 4.029325°C$, and $a = 1.894816$.

With the foregoing, the dimensionless governing equations of the problem can be written in the formulation of vorticity–stream function as follows:

$$\frac{1}{\pi r}\left(\frac{\partial \psi}{\partial \phi}\frac{\partial \omega}{\partial r} - \frac{\partial \psi}{\partial r}\frac{\partial \omega}{\partial \phi}\right) = Pr\,\nabla^2\omega + Pr\,Ra^* \times \left[\sin(\pi\phi)\frac{\partial}{\partial r}|\theta - \theta_m|^a + \frac{\cos(\pi\phi)}{\pi r}\frac{\partial}{\partial \phi}|\theta - \theta_m|^a\right] \tag{2}$$

$$\nabla^2\psi = -\omega \tag{3}$$

$$\frac{1}{\pi r}\left(\frac{\partial \psi}{\partial \phi}\frac{\partial \theta}{\partial r} - \frac{\partial \psi}{\partial r}\frac{\partial \theta}{\partial \phi}\right) = \nabla^2\theta \tag{4}$$

where

$$\nabla^2 \equiv \frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r} + \frac{1}{(\pi r)^2}\frac{\partial^2}{\partial \phi^2}. \tag{5}$$

The associated dimensionless boundary conditions for the problem are

$$\psi = \frac{\partial \psi}{\partial r} = 0, \quad \frac{\partial \theta}{\partial r} = -1, \quad \text{at } r = \frac{1}{(RR-1)} \tag{6a}$$

$$\psi = \frac{\partial \psi}{\partial r} = 0, \quad \theta = 0, \quad \text{at } r = \frac{RR}{(RR-1)} \tag{6b}$$

$$\psi = \omega = \frac{\partial \theta}{\partial \phi} = 0, \quad \text{at } \phi = 0 \text{ or } 1 \text{ (symmetry line).} \tag{6c}$$

The above formulation shows that the dimensionless temperature and flow fields are dependent upon the modified Rayleigh number, Ra^*, the radius ratio, RR, and the dimensionless temperature parameter, θ_m, which is in fact a parameter accounting for density inversion of water. Instead of directly using θ_m as a parameter, we adopted an inversion parameter, γ, following that used in ref. [7], defined as

$$\gamma = -2\theta_m/[\ln RR/(RR-1)]. \qquad (7)$$

The set of equations (2)–(6) was solved numerically by a finite difference method. The domain of interest was covered with a mesh of m angular by n radial uniformly spaced grid lines. The governing equations were then discretized by using centre difference approximations for spatial derivatives except the convective terms for which a quadratic upwind difference scheme [9, 10] was employed, in which the centred difference formula is corrected by some local diffusion-like terms. At points adjacent to the solid boundaries where the quadratic upwind differencing scheme is not applied, the second upwind scheme is used. The finite-difference equations were then solved in an iterative procedure for the steady-state solutions of the problem considered. The temperature and vorticity distributions were obtained from equations (4) and (2), respectively, by using the successive line relaxation method [11]. The stream function equation was then solved by the modified strongly implicit (MSI) method. This efficient solution procedure was developed and recommended by Schneider and Zedan [12]. The iterative process was repeated until the following convergence criterion was satisfied for each field variables ($\xi = \psi, \omega, \theta$):

$$\frac{\max|\xi_{i,j}^{n+1}-\xi_{i,j}^{n}|}{\max|\xi_{i,j}^{n+1}|} \leqslant 5\times 10^{-5} \qquad (8)$$

where n denotes the iteration number. Furthermore, the convergence of the steady-state solution was verified by checking the balance between the overall heat transfer rate for the inner and outer cylinder within 1%. Moreover, the overall heat transfer results were used to establish an understanding of what grid fineness is necessary for accurate numerical simulations; various grids of 35 (radial) × 41 (angular), 41 × 31, 45 × 31, 61 × 81, 85 × 51, and 85 × 61 were considered for the grid size study. The results reported in the paper were mainly obtained with the grids ranging from 35 × 41 to 45 × 31, depending on the radius ratio, which were considered to represent a reasonable compromise between accuracy and computing cost. For instance, for $RR = 5.0$, $\gamma = -3$, and $Ra^* = 10^6$, the difference in overall heat transfer rate and the maximum temperature are less than 2% between using the 45 × 31 and 85 × 51 grids.

RESULTS AND DISCUSSION

Numerical calculations were carried out systematically to explore the parametric effects on the flow patterns and heat transfer characteristics of the problem under consideration. The modified Rayleigh number was varied from 5×10^3 to 10^6 with the density inversion parameter ranging over -3.0 to 2.0 for three values of the radius ratio of the annulus: $RR = 1.2$, 2.6, and 5.0.

From the abundance of results generated by the numerical calculations, only a selection can be presented here in the available space. The results to be presented will focus on the occurrence of density inversion and its effects on the flow structures and temperature fields, which provide insight into the interacting physical processes.

Flow and temperature fields

Of particular interest in this study are the recirculatory flow patterns and temperature distributions, since these provide the direct evidence concerning the occurrence of density inversion as well as its possible effects. First of all, consideration is given to the effect of the density inversion parameter γ. Numerical results obtained indicate, as can be expected from the definition of the inversion parameter equation (7), that density inversion phenomenon can only occur for the case of negative values of γ. Accordingly, Figs. 2–4 display the representative flow and temperature fields in the annulus for $\gamma \leqslant 0$.

Figure 2 shows, for $RR = 2.6$, the effect of the density inversion parameter on the temperature and flow fields. For $\gamma = 0$ (i.e. $T_o = T_m$) at $Ra^* = 10^4$, the water in the annulus as expected behaves like an ordinary fluid; a unicellular flow pattern is developed in the half annulus. With increasing modified Rayleigh number, the vortex centre of the recirculation moves toward the top of the annulus while the isotherms indicate that convection acts as the dominant heat transport mechanism. For a negative inversion parameter, $\gamma = -0.5$ at $Ra^* = 10^4$, a weak secondary circulation can be detected in the bottom region near the outer cylinder. This is apparently due to the occurrence of density inversion in the flow. Further decrease of the inversion parameter results in a more pronounced effect of the density inversion in the flow. The outer circulation developed near the outer cylinder grows drastically and moves upward while the inner circulation (corresponding to the circulation in the annulus with positive inversion parameter) diminishes and moves downward such that the inner circulation is enclosed by the growing outer circulation. The outer circulation replaces the inner circulation as the dominant flow structure in the annulus. Furthermore, the isotherm distributions for $\gamma = -1.0$ at $Ra^* = 10^4$ clearly reveal that the convection heat transfer across the annulus is reduced significantly due to the bicellular flow structure which provides more mixing of the water, thus increasing the thermal resistance. At higher Rayleigh number, $Ra^* = 10^6$, one can observe from Fig. 2 that the influence of the density inversion parameter described above tends to be further provoked. Particularly, at $\gamma = -1.0$, a convective inversion already arises that a complete reversal in the direction of the buoyancy force occurs and the outer circulation becomes the only flow structure formed in the annulus. It is also evident from the figure that

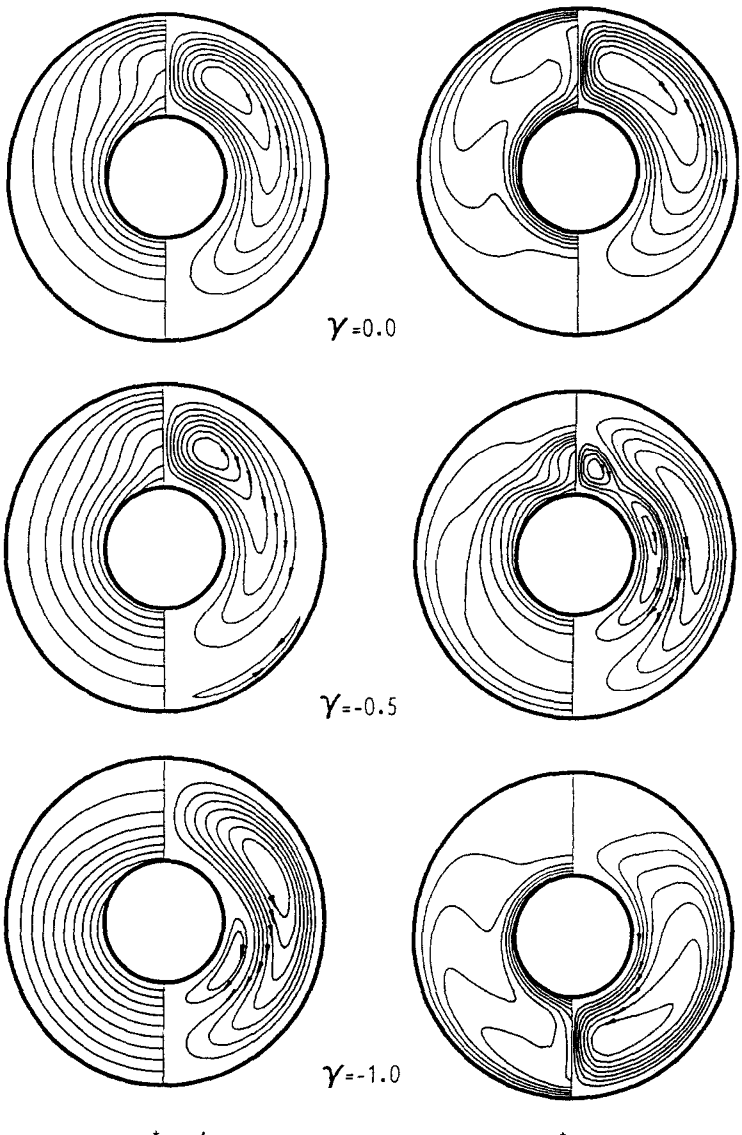

FIG. 2. Patterns of isotherms (left) and streamlines (right) for $RR = 2.6$.

besides the reverse in direction, the main features of the flow pattern and temperature distribution in the annulus with convective inversion are similar to that of $\gamma = 0$. A thermal plume activity arising from the bottom of the inner cylinder is observed while an essential stagnant isothermal zone exists in the top region of the annulus. Moreover, the isotherms demonstrate that the convection regains its role as the dominant heat transfer mode.

Figures 3 and 4 are similar to Fig. 2 but pertain to $RR = 1.2$ and 5.0, respectively. In the three figures, the streamlines and isotherms are presented for the same values of modified Rayleigh number and inversion parameter. It is interesting to note that the qualitative features of the flow structure and isotherm pattern shows little dependence on the radius ratio of the annulus. One can see from Fig. 3 that for $RR = 1.2$, a multicellular structure as commonly observed for the ordinary fluid in a narrow annulus [2] exists in the top region of the annulus. Moreover, Figs. 2–4 reveal that the relative strength and size of the two counter-rotating circulations for the negative inversion parameter appear to be dependent on the radius ratio as well as the modified Rayleigh number. The occurrence of density inversion as well as the convective inversion can be greatly enhanced with the increase of either the modified Rayleigh number or the radius ratio. Similar observations have been made in previous studies [3–5] for isothermal conditions.

For the uniform heat flux condition on the inner cylinder considered here, the surface temperature is one of the important variables in the analysis of the

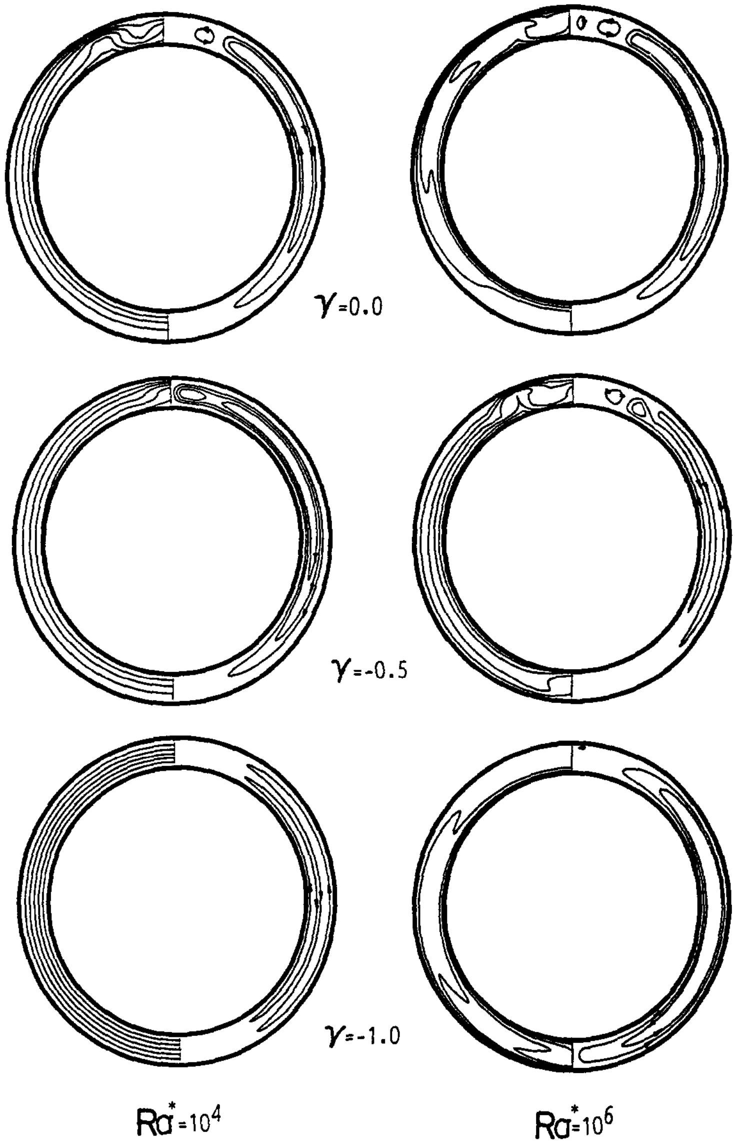

FIG. 3. Patterns of isotherms (left) and streamlines (right) for $RR = 1.2$.

problem, since it can reflect the transient phenomena that arise in the flow. Figure 5 illustrates the dimensionless surface temperature profile along the inner cylinder for three different values of inversion parameter, respectively. In general, it can be seen from the figures that the dimensionless surface temperature is higher at the lower value of either Ra^* or RR, indicative of a smaller heat transfer coefficient as a result of weaker recirculating flow. Moreover, there exists a significant variation of surface temperature along the circumference of the inner cylinder. For $\gamma = 0$ with $RR = 2.6$ and 5.0, due to the presence of a single circulation in the half annulus (Figs. 3 and 4), the local surface temperature increases gradually from the bottom to the top of the inner cylinder. On the other end, as γ is decreased to -1.0, the variation trend of the surface temperature profile appears to be reversed, signifying the occurrence of the aforementioned convective inversion in the annulus. As for the narrow annulus, $RR = 1.2$, due to the presence of the multicellular structure near the top region of the annulus for most cases considered, a significant oscillatory variation of the surface temperature is wherein observed.

Overall heat transfer results

Attention will now be turned to the results of the overall heat transfer rate across the annulus. The overall heat transfer results are cast in dimensionless form as the overall Nusselt number defined as

$$\overline{Nu} = q_i L/[k(\bar{T}_i - T_o)]. \quad (9)$$

The parametric effects on the overall Nusselt number are exemplified by Fig. 6. First of all, the variation of the overall Nusselt number passes a minimum at a certain value of the negative inversion parameter. This

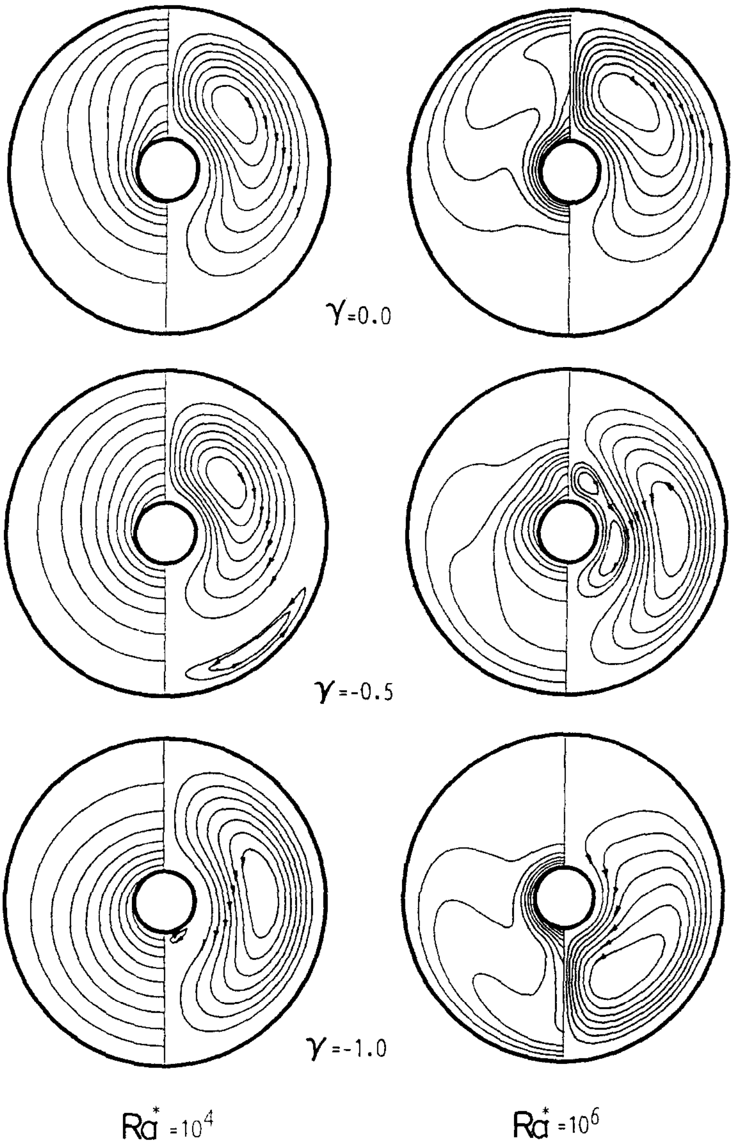

FIG. 4. Patterns of isotherms (left) and streamlines (right) for $RR = 5.0$.

can be expected from the fact that as the inversion parameter is decreased from positive to negative with fixed Ra^* and RR, a convective inversion is evoked by the complete reversal in circulation flow through a bicellular structure of approximately equal strength, which yields more mixing and hence a minimum heat transfer rate. Qualitatively, Fig. 6 clearly demonstrates that such a critical value of γ tends to increase as Ra^* is increased and seems rather insensitive to the variation of RR. Furthermore, the magnitude of the overall Nusselt number is seen to increase as either Ra^* or RR increases. These findings resemble those reported in ref. [5] for isothermal boundary conditions.

Attempts were made to correlate the Nusselt number vs Ra^* for a different radius ratio and inversion parameter using a least-square regression analysis. These correlations are obtained in the form

$$\overline{Nu} = C(Ra^*)^m \tag{10}$$

where coefficient C and exponent m are listed in Table 1 for various values of RR and γ.

CONCLUDING REMARKS

A computational analysis of steady laminar natural convection of cold water within a horizontal annulus with mixed boundary conditions has been presented. For the flow circumstances considered in the present study, the occurrence of density inversion is found to be mainly dependent upon the density inversion parameter. With a negative inversion parameter, the increase of either the modified Rayleigh number or the radius ratio results in a more pronounced effect of density inversion on the temperature and flow fields. The heat transfer characteristics of the problem

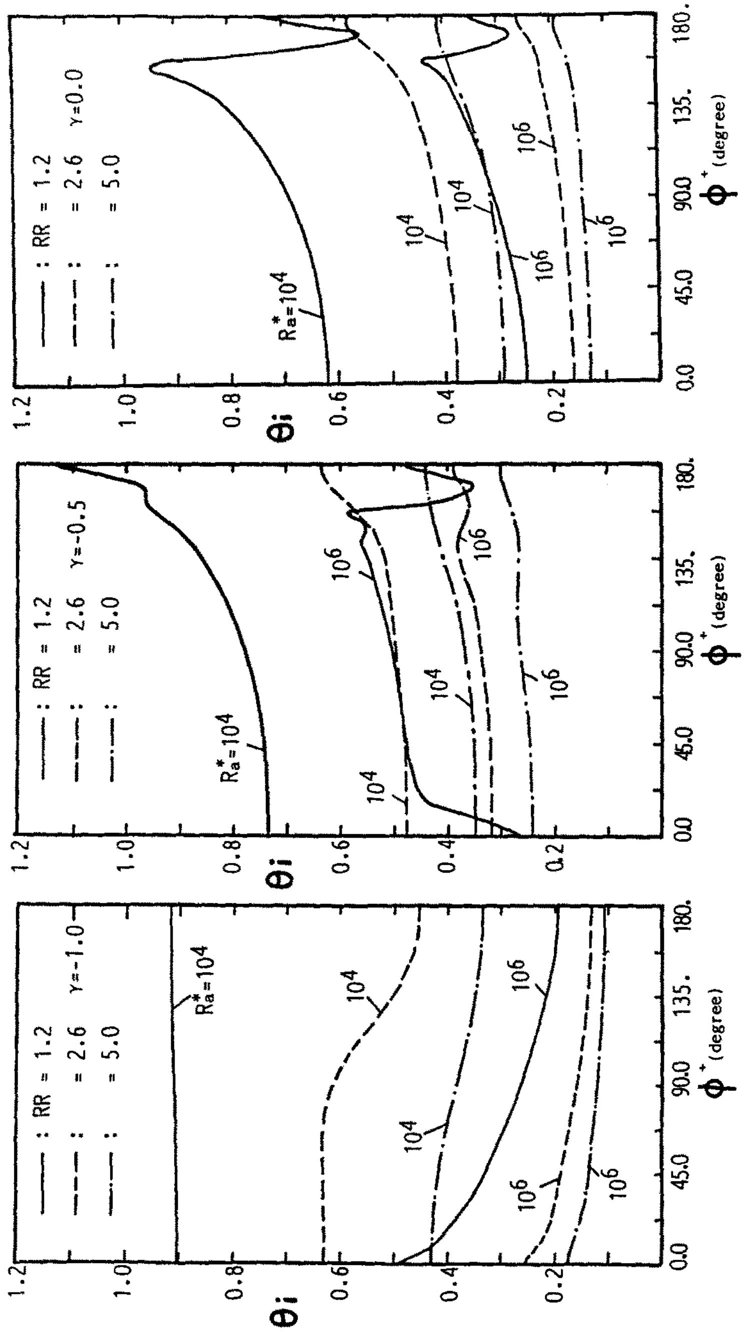

FIG. 5. The surface temperature profile of the inner cylinder.

Table 1. Constants for equation (10) for various inversion parameters and radius ratios

RR	γ	c	m	Ra^*	Mean deviation (%)	Correlation coefficient
1.2	−3.00	0.34511	0.18760	5×10^4–10^6	0.287	0.99982
	−2.50	0.32796	0.18870	5×10^4–10^6	0.264	0.99985
	−2.00	0.28524	0.19621	5×10^4–10^6	0.072	1.00000
	−1.50	0.20335	0.21564	5×10^4–10^6	1.396	0.99852
	−1.00	0.44824	0.10212	5×10^3–10^5	2.922	0.97153
	−0.75	0.59458	0.07305	5×10^3–10^5	0.964	0.99323
	−0.50	0.47324	0.10383	5×10^3–10^6	0.333	0.99981
	−0.25	0.34157	0.14570	5×10^3–10^6	0.529	0.99984
	0.00	0.26156	0.18060	5×10^3–10^6	0.689	0.99982
	0.50	0.23970	0.19900	5×10^3–10^6	0.746	0.99971
	1.00	0.24173	0.20483	5×10^3–10^6	1.199	0.99930
	1.50	0.24906	0.20677	5×10^3–10^6	1.317	0.99922
	2.00	0.25731	0.20767	5×10^3–10^6	1.455	0.99908
2.6	−3.00	0.43647	0.21018	5×10^3–10^6	1.128	0.99959
	−2.50	0.40197	0.21384	5×10^3–10^6	1.485	0.99924
	−2.00	0.35231	0.22059	5×10^3–10^6	2.033	0.99856
	−1.50	0.28635	0.23062	5×10^3–10^6	3.249	0.99422
	−1.00	0.24437	0.23258	5×10^4–10^6	2.361	0.99658
	−0.75	1.02288	0.05809	5×10^3–10^5	1.356	0.97468
	−0.50	0.84737	0.08919	5×10^3–10^6	1.271	0.99705
	−0.25	0.60628	0.13713	5×10^3–10^6	1.277	0.99852
	0.00	0.44659	0.17854	5×10^3–10^6	0.525	0.99981
	0.50	0.43199	0.19318	5×10^3–10^6	0.862	0.99965
	1.00	0.44651	0.19759	5×10^3–10^6	0.830	0.99968
	1.50	0.46538	0.19849	5×10^3–10^6	0.749	0.99978
	2.00	0.48427	0.19921	5×10^3–10^6	0.713	0.99982
5.0	−3.00	0.57574	0.20754	5×10^3–10^6	1.543	0.99921
	−2.50	0.53077	0.21105	5×10^3–10^6	1.808	0.99883
	−2.00	0.47604	0.21696	5×10^3–10^6	2.075	0.99821
	−1.50	0.40845	0.22411	5×10^3–10^6	2.420	0.99773
	−1.00	0.29309	0.24056	10^4–10^6	2.263	0.99757
	−0.75	1.80863	0.03571	5×10^3–10^5	2.426	0.93928
	−0.50	1.28099	0.07915	5×10^3–10^6	0.951	0.99727
	−0.25	0.86460	0.13052	5×10^3–10^6	1.507	0.99718
	0.00	0.62296	0.17288	5×10^3–10^6	0.277	0.99994
	0.50	0.58039	0.19006	5×10^3–10^6	1.026	0.99946
	1.00	0.59763	0.19413	5×10^3–10^6	1.128	0.99939
	1.50	0.62199	0.19540	5×10^3–10^6	1.108	0.99949
	2.00	0.64603	0.19576	5×10^3–10^6	1.066	0.99958

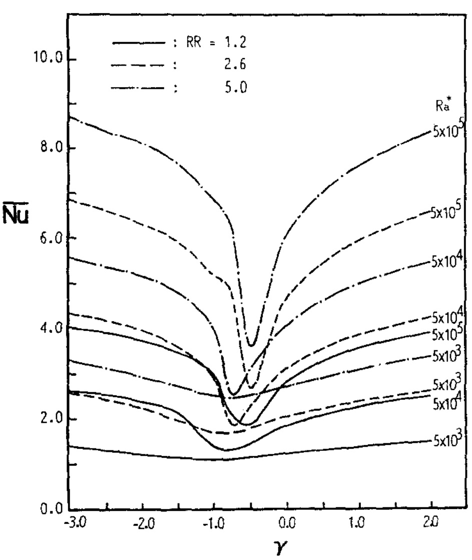

FIG. 6. Typical parametric effects on the overall Nusselt number.

investigated appear to be qualitatively similar to those for isothermal boundary conditions [4–6]. There exists a value of the negative density inversion parameter which yields minimum heat transfer. Such a critical value of the inversion parameter is found to depend primarily on the modified Rayleigh number.

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CONVECTION NATURELLE LAMINAIRE D'EAU FROIDE CONTENUE DANS UN ESPACE ANNULAIRE HORIZONTAL AVEC CONDITIONS AUX LIMITES MIXTES

Résumé—Une analyse numérique de la convection naturelle laminaire d'eau froide dans un espace annulaire horizontal avec flux uniforme sur la paroi interne et température uniforme sur la paroi externe est conduite pour explorer l'éventualité de l'inversion de densité et son effet sur les champs de vitesse et de température. Des résultats sont obtenus pour trois valeurs des rapports des rayons et pour des domaines variés du nombre de Rayleigh modifié et du paramètre d'inversion de densité. Dans les circonstances considérées, l'inversion de densité dépend principalement du paramètre d'inversion de densité. Avec un paramètre négatif, l'accroissement soit du nombre de Rayleigh modifié, soit du rapport des rayons conduit à un effet plus prononcé de l'inversion de densité sur les champs de vitesse ou de température dans l'espace annulaire.

LAMINARE NATÜRLICHE KONVEKTION IN KALTEM WASSER IN EINEM WAAGERECHTEN RINGRAUM MIT GEMISCHTEN RANDBEDINGUNGEN

Zusammenfassung—Eine rechnerische Untersuchung der stationären laminaren natürlichen Konvektion bei kaltem Wasser in einem waagerechten Ringraum wurde durchgeführt, um das Aufreten der Dichteinversion und ihren Einfluß auf Strömungs- und Temperaturfeld zu ermitteln. Am inneren Zylinder wurde ein konstanter Wärmestrom, am äußeren konstante Temperatur als Randbedingung gewählt. Es wurden Ergebnisse für drei verschiedene Durchmesserverhältnisse über verschiedene Bereiche der modifizierten Rayleigh-Zahl und des Dichteinversionsparameters ermittelt. Bei den untersuchten Strömungen hängt das Auftreten der Dichteinversion vor allem vom Dichteinversionsparameter ab. Bei negativem Inversionsparameter führt die Zunahme der modifizierten Rayleigh-Zahl oder des Durchmesserverhältnisses zu einem stärker ausgeprägten Einfluß der Dichteinversion auf das Strömungs- und Temperaturfeld im Ringraum.

ЛАМИНАРНАЯ ЕСТЕСТВЕННАЯ КОНВЕКЦИЯ ХОЛОДНОЙ ВОДЫ В ГОРИЗОНТАЛЬНОМ КОЛЬЦЕВОМ КАНАЛЕ ПРИ СМЕШАННЫХ ГРАНИЧНЫХ УСЛОВИЯХ

Аннотация—Для выяснения влияния инверсии плотности воды на поля скоростей и температуры проведен численный анализ стационарной ламинарной естественной конвекции холодной воды внутри горизонтального кольцевого канала при постоянном тепловом потоке на внутренней поверхности и при фиксированной температуре внешней цилиндрической поверхности. Результаты получены для трех значений отношения радиусов в различных диапазонах изменения модифицированного числа Рэлея и параметра инверсии плотности. Найдено, что для рассматриваемых условий течения возникновение инверсии плотности зависит в основном от указанного параметра. В случае отрицательного его значения увеличение модифицированного числа Рэлея или отношения радиусов приводит к более существенному влиянию инверсии плотности на поля скоростей и температуры в кольцевом канале.