NUMERICAL ANALYSIS OF NATURAL CONVECTION IN A DOUBLE-LAYER IMMISCIBLE SYSTEM

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INTRODUCTION

Thermal convection driven by heat sources plays an important role in the post-accident heat removal problem in the event of a core meltdown in a nuclear power reactor. The success of in-vessel retention (IVR) severe accident management strategy depends among other factors on the thermal loadings imposed by the convection melt pool [20].

The motivation for this work arrises from the results of RASPLAV experiments [1] which show that the melt separation into heavy and light fluids with a jump of uranium and zirconium concentrations at the boundaries may occur for some corium compositions. The melt stratification affects the thermal convection heat transfer. Thus, its effects on local heat flux distribution and average energy transport must be evaluated.

There are a few studies reported which focus on thermal convection in volumetrically heated stratified layers. Fieg [6] investigated the natural convection characteristics of two stratified immiscible liquid layers with the lower layer heated internally. Schramm and Reineke [18] studied experimentally and numerically the natural convection in a rectangular channel filled with two immiscible fluids of different physical properties. Kulaki and Nguen [12] studied experimentally and numerically the system of superposed layers of immiscible fluids bounded in a square cavity from below by a rigid, insulated surface and from above, by an isothermal wall. The heat was generated internally in the lower layer. A semi-empirical correlation was developed in [7] which may be employed to evaluate the upward and downward heat fluxes in a horizontal double layer fluid system with internal heat generation in the lower layer. It was shown that, unlike in a uniform fluid layer, the heat transfer dependends on a number of dimensionless parameters. Gubaidullin and Sehgal [8] applied CFD analysis to study the natural convection heat transfer in the two-layered miscible (salt water and water) as well as immiscible systems at steady or quasi-steady state in a semi-circular vessel. It was found that the maximum value of the side wall heat flux can be much higher than in the corresponding uniform pool case. Average values of the Nusselt number and energy splitting, i.e. ratio of heat transferred upwards versus downwards, are significantly lower. Series of tests within the framework of SIMECO (Simulation of In-vessel Melt Coolability) experimental program [17] have been carried out for paraffin oil - water system to assess the effect of the two layers of immiscible fluids on heat transport to the boundaries of a semi-circular vessel. The results comprise of local and average heat fluxes, temperature distributions and the energy splitting at steady state. It was confirmed that interface between the layers does not deform. The energy splitting was found to be almost an order magnitude less than in the uniform pool. However, the effect of the physical properties of the simulants on heat transfer in a semi-circular pool has not been addressed so far.

In the present paper numerical analysis has been applied to study the natural convection heat transfer in a system composed of two immiscible fluids with uniform internal heat generation in the lower layer or in both layers enclosed in a rectangular or in a semi-circular vessel. The objective of the work is to perform a parametric study to assess the effect of physical properties on the heat transfer characteristics as well as to complement results obtained from experiments by means of CFD simulations for a range of lower Rayleigh number and combine the experimental data and the computational results.

CFD SIMULATIONS

In this section the model employed in computations and the results of numerical simulations are described.

Model Description

We consider the heat transport in two layers of immiscible fluids with internal heat generation. The fluids in two layers have separate velocity and temperature fields and exchange momentum and energy through the interface. The liquid-liquid interface is assumed to be horizontal and does not deform. The assumption is confirmed experimentally by [6], [18] and [10]. This model has been used for computations of natural convection in a double-layer immiscible systems, such as oil and water, previously by [19], [16],[9] and [8].

For the numerical simulation, the Navier-Stokes equations of motion for the incompressible fluid and the fluid internal energy equation are discretized and solved. The Boussinesq approximation is used to model the free convection. Individual fluid properties such as c_p, κ and μ are assumed to be constants. The governing equations are subjected to no-slip, constant temperature boundary conditions at side and upper walls. Temperature and velocity boundary conditions at the interface are obtained from the stress balance, heat flux balance and velocity and temperature continuity at the interface. The numerical calculations were performed using the general-purpose CFX-4.1 code [1]. The details of mathematical model and of numerical procedure are given in [8].

[1]Property of AEA Technology, U.K.

CFD Model Validation

Firstly, the computed heat transfer coefficients were compared to a single, volumetrically heated, layer case. In this benchmark, the fluid layer of height H with uniform volumetric heat generation Q_v is bounded by a cooled rigid wall at the top and insulated rigid boundary at the bottom. The computed Nusselt numbers are compared to the Emara-Kulaki correlation $Nu = 0.477 \cdot Ra^{0.21} \cdot Pr^{0.041}$ valid for the Rayleigh number range $5 \cdot 10^3 \geq Ra \leq 5 \cdot 10^8$ [5]. The numerical results compare fairly well to the experimental correlation within experimental ($\pm 15\%$) accuracy and are presented in Figs.2-3.

Secondly, the computational two-layer model was validated to experimental correlation of Kulaki and Nguen [12] for a double-layer system of two immiscible fluids with internal heat generation in the lower layer.

In their study, the system of superposed layers was bounded in a square cavity from below by a rigid, insulated surface and from above, by a isothermal wall. The heat was generated internally in the lower layer. Experimental measurements of transient and steady state convection up to Rayleigh numbers of 10^{11} were presented for silicone oil - water ($\kappa_{12} \simeq 0.2$, $\mu_{12} \simeq 5$) and heptane - water ($\kappa_{12} \simeq 0.2$, $\mu_{12} \simeq 0.2$) systems. The Nusselt number based on the average heat transfer coefficient for different layer thickness ratios were obtained from the experiments and correlated. The results are summarized in Table 1.

Table 1: The Nusselt number correlations of Kulaki and Nguen

top layer	L_{12}	Nu'	range of Ra_d
silicon oil	0.035	$0.186 \cdot Ra_d^{0.1986}$	$10^7 < Ra_d < 10^{11}$
silicon oil	0.111	$0.183 \cdot Ra_d^{0.1988}$	$10^7 < Ra_d < 10^{11}$
silicon oil	0.433	$0.115 \cdot Ra_d^{0.2262}$	$10^4 < Ra_d < 10^{11}$
heptane	0.04	$0.126 \cdot Ra_d^{0.255}$	$10^7 < Ra_d < 10^{11}$
heptane	0.111	$0.112 \cdot Ra_d^{0.232}$	$10^7 < Ra_d < 10^{11}$
heptane	0.433	$0.135 \cdot Ra_d^{0.232}$	$10^5 < Ra_d < 10^{11}$

Table 2: Physical properties (taken at $T = 30°C$)

	ρ	β	c_p	κ	μ	Pr
water	995	$3 \cdot 10^{-4}$	4200	0.61	$8 \cdot 10^{-4}$	6
silicon oil	900	$1 \cdot 10^{-3}$	1900	0.12	$4 \cdot 10^{-3}$	67
heptane	670	$7 \cdot 10^{-4}$	2050	0.13	$4 \cdot 10^{-4}$	6

The present simulations are performed in 2D and with the Rayleigh number Ra_d up to 10^9 i.e. in the regimes of laminar and soft turbulence convection. The physical properties used in the calculations are given in Table 2. The computational settings and the boundary conditions are presented in Fig.1. Mesh employed is densed near the boundaries as well as near the interface. The chosen grid appeared to be satisfactory to the mesh dependence tests.

The comparison presented in Figs.2-3 shows that within the given Rayleigh number range, 2D computed results are in good agreement with the measured heat flux data. At higher Rayleigh numbers 3D simulations must be used to describe correctly turbulent mixing of the fluid [13]. The applications of different turbulence models to the thermal convection flows with internal heat generation in 2D are discussed in [3].

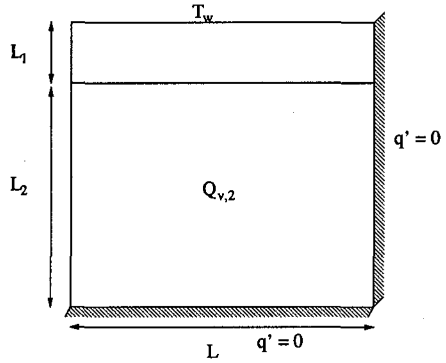


Figure 1: Schematic of the problem

Typical computed isotherms are shown in Figs.4-5. In the upper layer the isotherms are similar to those known from Bénard problem and in the lower layer the isotherms are similar to those obtained in an internally heated fluid insulated at the bottom and cooled at the top. It should be noted that the exponent of Ra_d in the correlations listed in Table1 is similar that obtained by Kulaki and Emara for a single volumetrically heated layer. The empirical constant multiplying Ra_d is smaller due to additional resistance imposed by the top fluid layer [2]. The computed steady state temperature profiles are shown in Figs.6-7.

Results of Computations for Semi-Circular Pool

The SIMECO experiments [10]were performed for a superposed paraffin oil - water system enclosed in a semi-circular vessel. In the tests with the lower layer heated, the height of the lower pool is kept constant at $L_1 = 0.26$ m and and two different height ratios of $L_{12} = 4{:}26$ and $L_{12} = 6{:}26$ are chosen for the tests with only the lower layer heated. For the tests with the heat generation in both layers, the total height of the pool L is kept equal to 0.26 m. The ratios of upper to lower layer heights L_{12} are kept equal to 4:22 and 8:22. The ratio of upper layerto lower layer thermal conductivity and viscosity are equal to $\kappa_{12} = 1 : 3$, $\mu_{12} = 20 : 1$. The ratio of densities ρ_{12} is equal to 0.88. The interface during all the tests remained undisturbed and no mixing between the two layers was observed.

The Rayleigh number based on the lower layer properties (Ra_d) ranges from $7{\cdot}10^{12}$ to $2{\cdot}10^{13}$ in these experiments, so that turbulent convection is established in the liquid

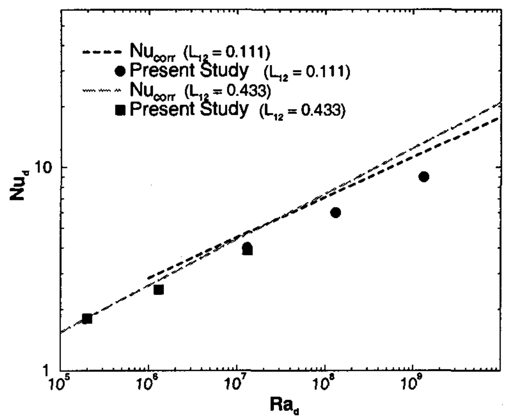


Figure 2: Computed and Experimental Nusselt numbers for Silicon Oil-Water

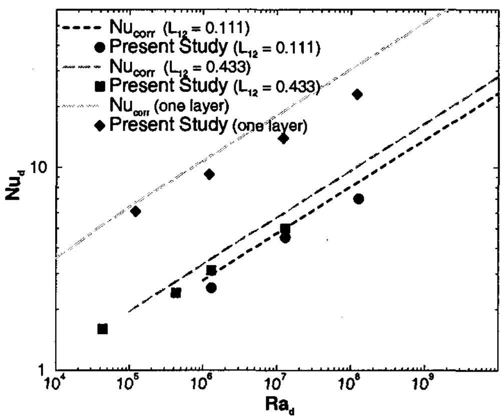


Figure 3: Computed and Experimental Nusselt numbers for Heptane-Water

pool. The choice of the Rayleigh number was determined mainly by size of the vessel and, hence could not be varied much.

Limitations of the SIMECO experimental apparatus did not permit the investigation of the phenomenon in the wide range of the Raylegh number (e.g. from $Ra_d = 10^8$ to $Ra_d = 10^{15}$) as well as the CFD model employed applies only for regimes of laminar or soft turbulence convection (upto $Ra_d = 10^{10}$).

Problem Formulation: The fluid layers of heights L_1 and L_2 are enclosed in 2D semi-circular cavity of radius L bounded by isothermal surfaces ($T = T_w$). Heat Q_v is generated internally in either both layers or in the bottom layer only. Schematic of the problem and boundary conditions employed are shown in Fig.8. The physical properties of the bottom layer fluid taken are those of water and the properties of the top layer fluid are those of paraffin oil. The Prandtl numbers are $Pr_1 = 370$, $Pr_2 = 4.$. The definition of Ra_d is based on the depth L_2 and physical properties of the lower layer. The computational mesh is constructed using a multi-block approach. 17 blocks are attached to form the full domain. A very dense mesh was used in the boundary regions. The computational grids with about $13{\cdot}10^3$ nodes each are chosen for simulations. Grid sensitivity tests were performed.

The geometry of the pool is taken as in SIMECO tests [10]. The thick (23 mm) brass curvilinear walls between the pool and the coolant used in the facility created some inhomogeneity effects in the wall temperature distributions. This temperature inhomegeneity observed in the tests can be characterized by the following dimensionless number $\frac{T_{max,w}-T_c}{T_{max,p}-T_c}$ and its values were small, less than 0.15 [11]. The sidewall Nusselt number defined in the tests was based on the reference value of the averaged temperature $< T_w >$ obtained from thermocouples readings along the length of the wall. Non-constant boundary temperature distribution along the curvilinear wall may affect the thermal convection in the pool compared to constant boundary temperature case used in the present calcula-

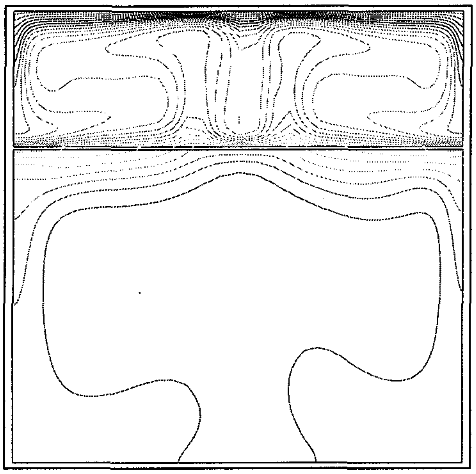


Figure 4: Computed isotherms for Silicon Oil-Water, $Ra_d = 1.3 \cdot 10^6, L_{12} = 0.433$

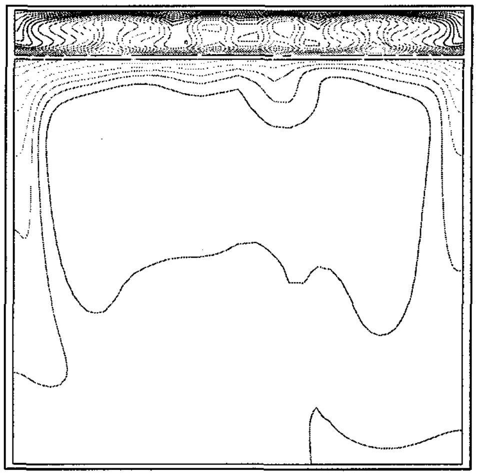


Figure 5: Computed isotherms for Heptane-Water, $Ra_d = 1.3 \cdot 10^7, L_{12} = 0.111$

tions. Therefore, this effect must be assessed. For that purpose, a conjugate problem was considered and the results are presented in Appendix 1. The results show that within the range of $\frac{T_{max,w}-T_c}{T_{max,p}-T_c}$ observed in the experiments, the non-uniformity of the temperature distribution at the sidewall boundaries affects the heat transfer very little in the pool and the employment of constant boundary temperature is justified.

Case of Heat Generation in Both Layers: The simulations are performed in the range of Rayleigh number Ra of 10^9 - 10^{11}. The computed and experimental time and space averaged Nusselt numbers are presented in Figs.9-10. It should be noted that the results of ACOPO experiments show that no sudden regime transition is expected to "hard turbulence" [20]. Thus, the comparison of the experimental and the numerical Nusselt numbers obtained for such different Rayleigh numbers can be reasonable. It can be seen on Figs.9-10 that the predicted and the experimental values of Nusselt numbers follow the same trend.

Case of Heat Generation in the Bottom Layer: The simulations are performed in the range of Rayleigh number Ra_d of 10^9 - 10^{11}. Figures 11-12 presents the average computed and measured Nusselt numbers.

The time averaged steady-state centerline temperature profiles are presented in Figures 13 - 16. The experimental values were obtained from time-averaged measurements with thermocouple probes located at the centerline of the pool. These values are compared to the numerical results for lower Rayleigh numbers. The agreement between the two results is reasonable even though the numerical values were produced for laminar convection or soft turbulence regime. The temperature distribution in the lower layer is almost linear in all cases and indicate that fluid is stably stratified in the region. Both computed and experimental values show a sharp gradient across the interface. For cases of heat generation in the lower layer, the profiles in the upper layer are typical for those of

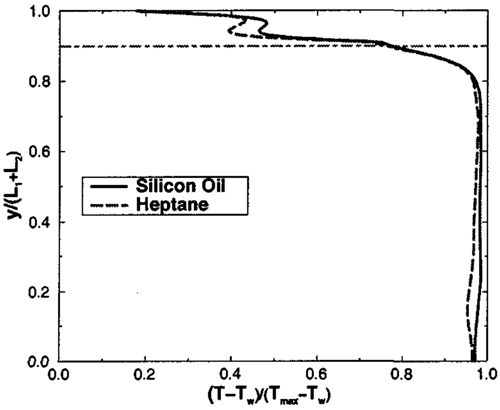


Figure 6: Centerline temperatures for L_{12} = 0.11, $Ra_d = 1.3 \cdot 10^7$

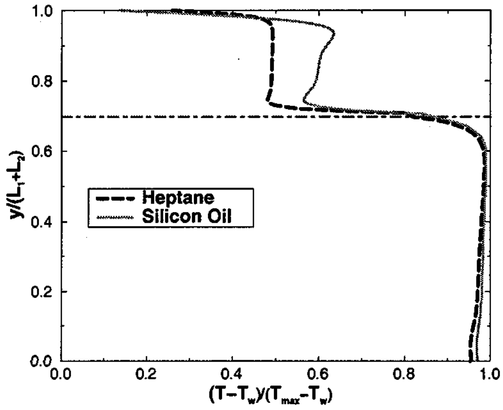


Figure 7: Centerline temperatures for L_{12} = 0.433, $Ra_d = 1.3 \cdot 10^7$

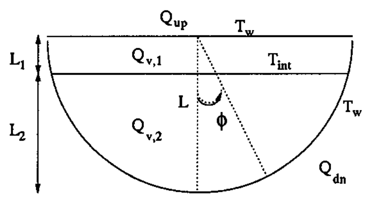


Figure 8: Problem formulation and boundary conditions employ ed in CFD simulations

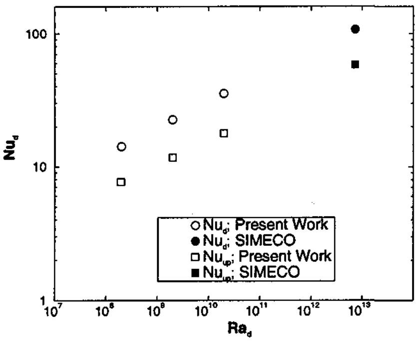
Rayleigh-Benard conv ection.

Effect of the upper la y erphysical properties

In general, the heat transfer in a multilay ered system depends on a multitude of parameters such as ratios of thermophysical properties, lay ers thickness, heat generation rates etc. The correlation between the energy split and the thermophysical parameters of interest is not known for a semi-circular geometry as it was obtained for a rectangular case [7]. Therefore the assesment of the properties effect at given lay ers thickness ratio b y means of CFD is desirable.

We would like to determine the effect of the key parameters, conductivity and viscosity ratios κ_{12}, μ_{12}, on heat transfer process. The values of κ_{12} and μ_{12} considered are [5:1], [1:1], [1:5]. The Rayleigh n umber Ra_d is chosen equal to 1.2×10^{10}. The simulations are performed for two configurations of $L_{12} = 4 : 22$ and $L_{12} = 8 : 18$ and heat generation in both lay ers.

Figure 17 presents the effect of different κ_{12} and μ_{12} on energy split Q_{12}. Strong dependency of Q_{12} on κ_{12} can be observed. F or cases of $\kappa_{12} \leq 1$, the upper layer introduces additional thermal resistance and thus, decreases heat transferred upwards. The maximum v alue of side wall heat flux can be much higher than in the corresponding uniform



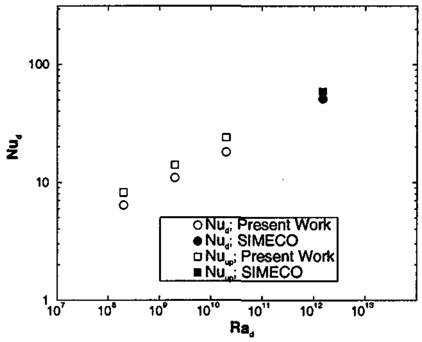
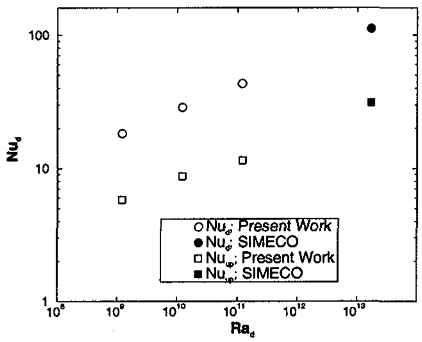


Figure 9: The average Nusselt number as a function of the Rayleigh number. $L_{12} = 4:22$, $Q_{v,1} = Q_{v,2}$

Figure 10: The average Nusselt number as a function of the Rayleigh number $L_{12} = 8:18$, $Q_{v,1} = Q_{v,2}$



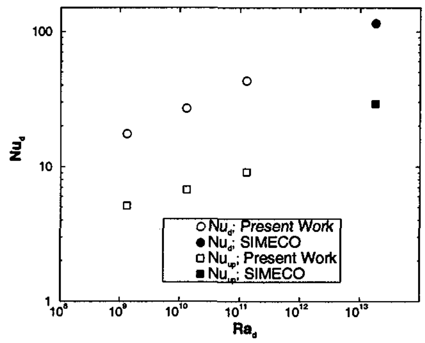


Figure 11: The average Nusselt number as a function of the Rayleigh number. $L_{12} = 4:26$, $Q_{v,1} = 0$

Figure 12: The average Nusselt number as a function of the Rayleigh number for $L_{12} = 6:26$, $Q_{v,1} = 0$

pool case and is located right below the interface. For cases of $\kappa_{12} > 1$ more heat is being tranferred uppwards and at certain values of κ_{12} for given L_{12} the Q_{12} may become higher than in the corresponding uniform pool case.

The effect of viscosity is less significant since the $Nu \sim Ra^{1/n} \sim \mu^{-1/n}$. Lower viscosity in the upper layer increases the convection intensity and the heat transfer. Different local heat distributions along the curved wall are presented in Fig.18 for various viscosity ratios. The difference in Q_{12} for μ_{12} = [5:1] and [1:5] is about 30%.

SUMMARY

In this paper, results of numerical investigation of thermal convection in a double-layer pool with uniform internal heat generation in both or lower layerare reported. In CFD computations the immiscible system is modeled by two layers of fluids separated by a fixed-in-space interface. Firstly, results of numerical simulations are presented for

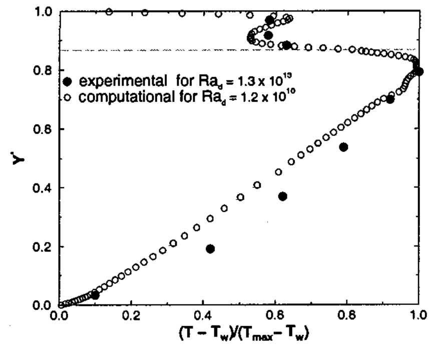


Figure 13: The time average centerline temperature profiles for $L_{12} = 4 : 26$. Heat generation in bottom layer.

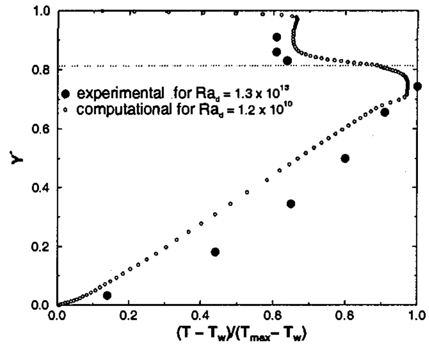


Figure 14: The time average centerline temperature profiles for $L_{12} = 6 : 26$. Heat generation in bottom layer.

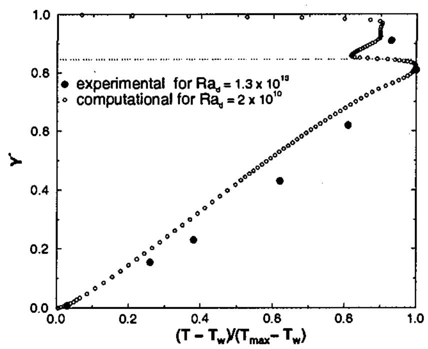


Figure 15: The time average centerline temperature profiles for $L_{12} = 4 : 22$. Heat Generation in both layers.

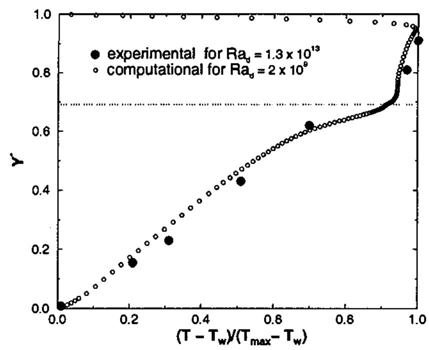


Figure 16: The time average centerline temperature profiles for $L_{12} = 8 : 18$. Heat Generation in both layers.

a rectangular cavity for Ra_d, in the range of $10^5 - 10^9$. Comparison with experimental data of Kulaki and Nguen [12] provides the basic validation for the modeling schemes employed.

Secondly, the CFD model is applied to investigate the natural convection in a two-dimensional semi-circular pool for Rayleigh numbers up to 10^{11}. The fluid properties of the upper layer are varied parametrically. A strong dependency of the energy splitting on a conductivity ratio is reported. The dependency on viscosity ratio is weak. The simulations are performed for Paraffin Oil-Water system and results are compared to data obtained in the SIMECO experiments [10]. The major trends in the vessel wall heaflux and temperatures can be predicted by the CFD model. The results obtained from CFD computations are consistent with and complementary to experimental observations and there is a potential to use them in interpretations of RASPLAV data. Further experimental work on convection in a double-layer system with varied conductivity ratio may be advisable.

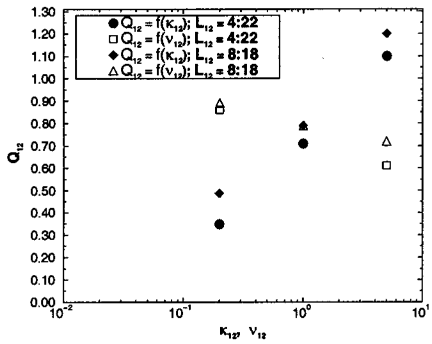


Figure 17: Effect of physical properties on the energy split. $Q_{v,1} = Q_{v,2}$.

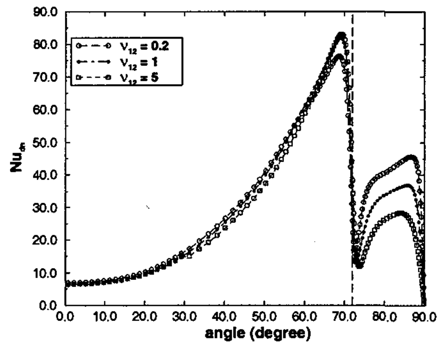


Figure 18: Effect of viscosity ratio on the sidewall Nusselt number distribution for $\kappa_1 = \kappa_2$, $Q_{v,1} = Q_{v,2}$.

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APPENDIX 1

To address the effect of 2D wall effects on heat transfer in the pool, we consider the conjugate problem of thermal convection in a semi-circular uniform pool surrounded by a solid wall of thickness δ. As a boundary condition, temperature outside the pool T_c is kept constant. The problem schematic is shown in Fig.19. To achieve the inhomogeneous temperature distribution in the wall for a given Ra number, the conductivity of the wall was varied. Thus, different T^* cases were considered. The computed isotherms are presented in Fig.20. The temperature distribution at the fluid-solid boundary along the brass wall is shown in Fig.21. The results are presented in Table 3. It can be observed that the flow in the pool is affected in the following way:

It can be seen that for $T^* < 0.15 - 0.2$, the effects of wall diffusion can be neglected and the temperature boundary condition taken as constant.

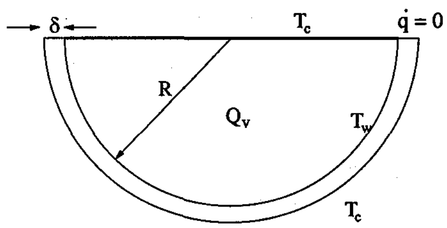

Figure 19: Schematic of the problem

Table 3: Results of the computations

	$\frac{T_{max,w}-T_c}{T_{max,p}-T_c}$	η
case 0	0	0.51
case 1	0.08	0.53
case 2	0.17	0.54
case 3	0.26	0.57

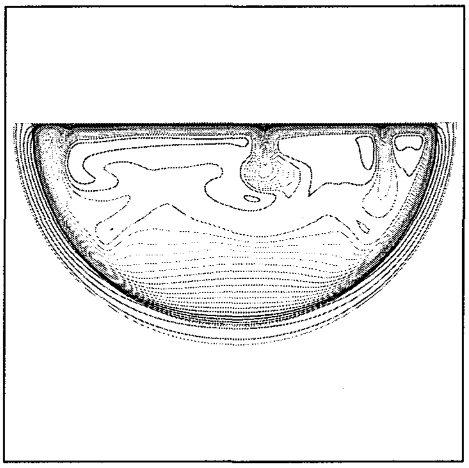

Figure 20: Temperature field for Ra = 10^9, case 3

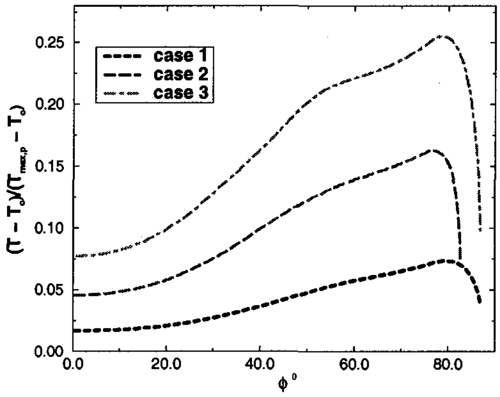

Figure 21: Temperature distribution at the inner surface of the trough

NOMENCLATURE

c_p	specific heat, J/kg·K
g	gravity, m/s^2
L	enclosure height, m
L_{ij}	ratio of i-layer height to j-layer height
Nu	Nusselt number, $Nu = \frac{q \cdot L}{\kappa(T_w - T_{max})}$
Nu_{up}	Nusselt number at the upper surface
Nu_{dn}	Nusselt number at the curved surface
Pr	Prandtl number, $Pr = \frac{\nu}{\alpha}$
Q_v	volumetric heat generation rate, W/m^3
Q_{12}	ratio of heat transferred upwards to heat transferred downwards
q	heat flux, W/m^2
Ra	Rayleigh number, $Ra = \frac{g\beta Q_v L^5}{\nu\alpha k}$
Ra_d	Rayleigh number based on the height of the lower layer
Ra_{RB}	Rayleigh number, $Ra_{RB} = \frac{g\beta\Delta T L^3}{\nu\alpha}$
T	temperature, K

Greek

α	thermal diffusivity, m^2/s
β	coefficient of thermal expansion,$1/K$
η	fraction of heat generated within the layer that is transferred downward
κ	heat conductivity, $W/m{\cdot}K$
κ_{ij}	ratio of i-layer to j-layer conductivity
μ_{ij}	ratio of i-layer to j-layer dynamic viscosity
μ	dynamic viscosity, $Pa{\cdot}s$
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
ϕ	angle, degree

Subscripts

1	top layer
2	lower layer
av	average
exp	experimental
c	coolant
$comp$	computational
$corr$	correlation
int	interface
max	maximum
o	reference value
w	wall
$<>$	averaged over surface

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