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Natural Convection in Rectangular Pool with Volumetric Heat Sources

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Abstract

Natural convection plays an important role in determining the thermal load from debris accumulated in the reactor vessel lower head during a severe accident. Recently attentions are being paid to the feasibility of external vessel flooding as a severe accident management, and to the phenomena affecting the success path in retaining the molten core material inside the vessel. The heat transfer within the molten core material can be characterized by buoyancy-induced flows resulting from internal heating due to decay of fission products. The thermo-fluid dynamic characteristics of the molten pool depend strongly on the thermal boundary conditions. The spatial and temporal variation of heat flux on the pool wall boundaries and the pool superheat are mainly characterized by the natural convection flow inside the molten pool. In general, natural convection involving internal heat generation is delineated in terms of the modified Rayleigh number, Ra´, which quantifies the internal heat source and hence the strength of buoyancy. The test section is of rectangular cavity whose length, width, and height are 500mm, 80mm, and 250mm, respectively. A total of twenty-four T-type thermocouples were installed in the test loop to measure temperature distribution. Four T-type thermocouples were utilized to measure temperatures on the boundary. A direct heating method was adopted in this test to simulate the uniform heat generation. The experiments covered a range of Rayleigh number, Ra, between 4.87×10^{7} and 2.32×10^{14} and Prandtl number, Pr, between 0.7 and 3.98. Tests were conducted with water and air as simulant. The upper and lower boundary conditions were maintained at a uniform temperature of 10°C.

1. Introduction

Accidents like the loss-of-coolant accidents (LOCAs) that may lead to core melt may be initiated in a number of ways involving malfunction of components or safety system, or improper operator action. During a severe accident in a nuclear reactor, the core may melt and relocate to the lower plenum to form a hemispherical pool. If there is no effective cooling mechanism, the core debris may heat up and the molten pool may run into natural convection. The high temperature of the molten core material will threaten the thermal and structural integrity of the reactor vessel. The extent and urgency of this threat depend primarily upon the intensity of the internal heat sources and upon the consequent distribution of the heat fluxes on the vessel walls in contact with the molten core material. The feasibility of external vessel flooding as a severe accident management as well as the phenomena affecting the success in retaining the molten core material inside the vessel has received wide attention.

The earliest study of natural convection in volumetrically heated layers is that of Kulacki and Goldstein [1]. The fluid layer was bounded by two isothermal upper and lower plates held at a constant temperature and four insulated side walls. The Rayleigh numbers covered a range of $200 < Ra' < 10^7$. Data were obtained using interferometers and covered the laminar, transition, and turbulent regimes of natural convection. For $Ra' > 10^4$, turbulent mixing effects begin to play a dominant role in the overall energy transport process, and any periodicity or near periodicity in the mean temperature fields evident at lower Ra begins to disappear.

A numerical study of natural convection in internally heated pools contained in rectangular cavity was performed by Jahn and Reineke [2]. They found that the temperature field suggested the presence of nonuniform eddies in the upper region of the pool with a stable and calm liquid layer in the lower region. They concluded that heat was transferred more effectively in the upper region as opposed to the lower region for this geometry.

Kulacki and Emara [3] correlated upward and downward heat transfer in rectangular geometries under the constraint of relatively small temperature differences. In their study, a planar fluid layer was considered, where only the top surface was cooled and the bottom surface was insulated. For these boundary conditions, heat transfer coefficients ware obtained for Rayleigh numbers up to 2×10^{12}. They also attempted to identify an independent effect of the Prandtl number ranging from 2.75 to 6.86.

Kulacki and Nagle [4] investigated natural convection with volumetric heating in a horizontal fluid layer with a rigid, insulated lower boundary and a rigid, isothermal upper boundary for Rayleigh numbers from 114 to 1.8×10^6 times the critical value of the linear stability theory. A correlation for the mean Nusselt number was obtained for steady heat transfer, and data were presented on fluctuating temperatures at high Rayleigh numbers and on developing temperature distributions when the layer is subjected to step change in power.

Hollands et al. [5] reported on the experimental study on the natural convective heat transport through a horizontal layer of air, covering the Rayleigh number range from sub-critical to 4×10^6. In a fluid with a high thermal diffusivity (low Prandtl number) packets of fluid leaving the outer edge of the boundary layer will lose heat and take up a temperature equal to the surroundings within a much shorter distance from its starting point than a fluid with a low thermal diffusivity.

Correlation of the experimental data on heat transfer from non-boiling, horizontal fluid layers with internal heat generation was cast into a form suitable for analysis of post-accident heat removal in fast reactors by Baker et al. [6]. Available data on layers with equal boundary temperatures indicated that the downward heat transfer rate could be accounted for by conduction alone, while the upward heat transfer rate was largely controlled by convection.

A phenomenological model of eddy heat transfer in natural convection with volumetric energy sources at high Rayleigh numbers was developed by Cheung [7]. The model was applied to the problem of thermal convection in a horizontal heated fluid layer with an adiabatic lower boundary and an isothermal upper wall. At high Rayleigh numbers, the mean temperature was found to be essentially constant throughout the layer except in a sublayer region near the upper wall. The thickness of such a region was observed to be inversely proportional to the mean Nusselt number. Outside the sublayer region, the distribution of eddy heat flux was linear for all the Rayleigh numbers under consideration. Production of thermal variance was negligible in the lower 75-95% of the layer and was greatest near the upper boundary. Comparison of the heat transfer predictions with measurements indicated an excellent agreement in the turbulent thermal convection regime.

Steinberner and Reineke [8] investigated experimentally and numerically the buoyant convection with internal heat sources in a closed rectangular cavity for the range of $10^7 < Ra' < 10^{14}$. Their investigation of the structure and the dynamical behavior of the turbulent boundary layer at free convection with internal heat sources in rectangular cavities, revealed a number of equal properties in comparison to the turbulent boundary layer at the heated vertical plate.

Nourgaliev et al. [9] investigated turbulence characteristics of the flow and thermal fields in an internally heated horizontal fluid layer for Rayleigh numbers up to 5×10^8 using a finite-difference code for direct numerical simulations. Calculated results indicated nonequilibrium of turbulent kinetic energy and thermal variance under unstable stratification conditions. They found that important turbulence constants are remarkably nonuniformly distributed across the layer and strongly dependent upon the Rayleigh and fluid Prandtl numbers.

Previous investigations suffer from serious weakness to represent a molten core material behavior in the reactor vessel lower plenum involving specific conditions such as high Rayleigh numbers, turbulent boundary layers, a low height-to-diameter ratio, and a hemispherical geometry. Therefore any extrapolations to different geometries and convection conditions have to be considered with reservation. In addition, whereas in many previous studies only the average heat transfer coefficients from liquid to surrounding walls have been correlated, the heat flux profiles and their peak values are also needed for safety assessments concerning the external vessel cooling.

2. Experimental Apparatus

The test section is of rectangular cavity whose length, width, and height are 500mm, 80mm, and 250mm, respectively as shown in Fig. 1. Twenty thin cable-type heaters, with a diameter of 5mm and a length of 500mm, are used to simulate internal heating in the pool. They are uniformly distributed in the rectangular pool and thus they can supply a maximum of 2kW power to the pool. A total of twenty-four T-type thermocouples were installed in the test loop to measure temperature distribution. Four T-type thermocouples were used to measure the boundary temperature. Eight thermocouples were installed at mid plane 0, 3, 40, 90, 140, 190, 247, 250 respectively from the bottom plate. Other sixteen thermocouples were installed at the symmetry line, which is 150mm off from the centerline. Initially the working fluid in the test section was heated up to reach a steady state. The average temperature increase rate was 0.6 /min. The rate of temperature rise at any location in the pool did not differ by more than ±10% from the mean value. The results demonstrated feasibility of the direct heating method to simulate uniform volumetric heat generation within the pool.

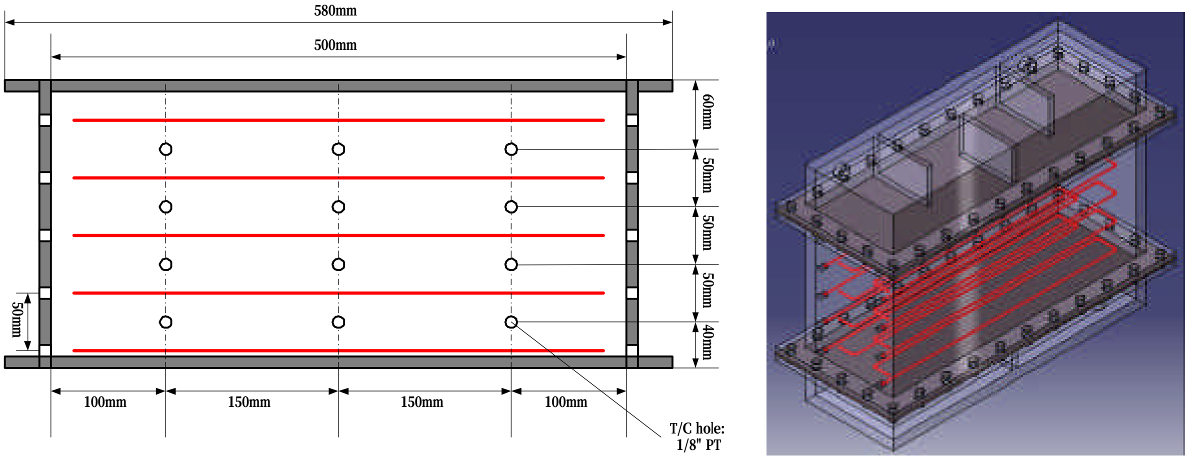

Fig. 1. Schematic of the test section

Fig. 2 presents the test loop consisting of a demineralized water system, test section, heat exchanger, and a data acquisition system (DAS). The thermocouples were submerged sequentially in a constant temperature pool. In this course we calibrated the DAS bias error so as to minimize measurement error. Once properly calibrated, thermocouples were placed at their designated locations. The water-cooling system, or heat exchanger, supplies the well-defined upper and lower boundary conditions. A performance test of this water cooling system showed the temperature difference of ±0.5 ranging from 5 to 80 . After all this procedure was checked out to be functioning properly, the pool was allowed to heat up. The modified Rayleigh number, Ra´, can be calculated for the present experiment for varying input power (Q) and characteristic pool depth (L). The experiments covered a range of Ra´ between 4.87×10^{7} and 2.32×10^{14} and the Prandtl number, Pr, between 0.7 and 3.98. Tests were performed with water and air as simulant.

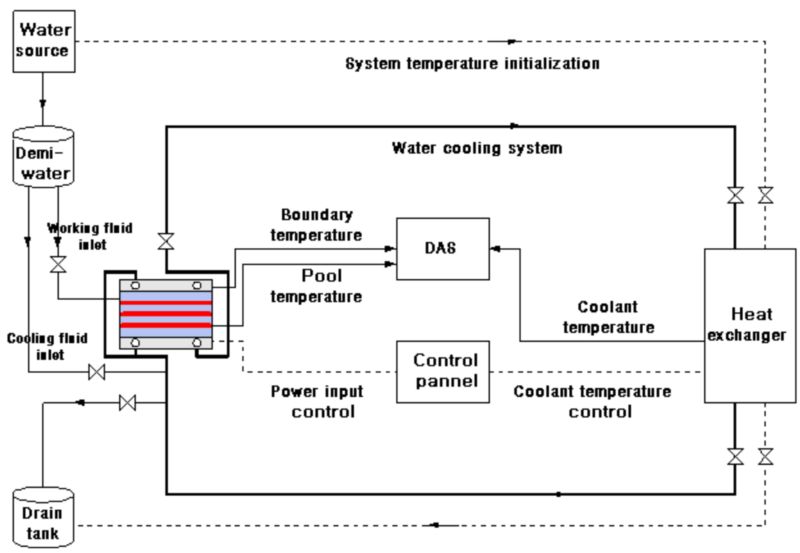

Fig. 2. Schematic diagram of the test loop

3. Experimental Procedure and Heating Method

All the thermocouples were properly calibrated to be placed in their designated locations. Uniform heat generation is one of the important conditions in the experiment. Thus, water temperatures were measured at different locations. After the uniformity of the heat generation rate was checked on, the demineralized working fluid was pumped fully into the test section through the heat exchanger. In this manner the working fluid in the test section and external cooling system temperature reached an initial steady state. After assuring that everything was properly running, the power switch was turned on, and the pool was allowed to heat up. The Hewlett-Packard DAS was adjusted to record the temperatures. Tests were conducted with water and air as listed in Table 1. The upper and lower boundary layers were maintained at a uniform temperature.

Table 1. Test matrix

Boundary condition	Working fluid	Test	Power [W]
Isothermal	Water	Case-A1	38
		Case-A2	295
		Case-A3	832
		Case-A4	1474
	Air	Case-W1	15
		Case-W2	40
		Case-W3	81
		Case-W4	149

Asfia et al. [10] conducted experiments related to the natural convection heat transfer in volumetrically heated spherical pools. Their experiments were conducted in pyrex bell-jars, using Freon-113 as the simulant liquid. Microwave heating was used in the experiment, and a surrounding water pool cooled the experimental system. They concluded that a large variation in heat transfer coefficient existed along the vessel wall.

The BALI [11] experiment was designed to study the thermal-hydraulics of core material pool for in-vessel or ex-vessel situation. The core material melt was represented by salted water. The pool was cooled from the bottom and the top and heated electrically by the Joule effect with current supplies located on the sides.

Gabor et al. [12] conducted natural convection experiment with hemispherical pool containers. The pool container served both as a heat transfer surface and as an electrode. $ZnSO_4$-H_2O was used as the heat generating liquid.

The earliest study of natural convection in volumetrically heated layers is due to Kulacki and Goldstein [1]. Energy transport was measured in a layer of dilute electrolyte bounded horizontally by two rigid planes of constant and equal temperature. Joule heating by an alternating current passing horizontally through the layer provided the volumetric energy source. Kulacki and Emara [3], Kulacki and Nagle [4] used the same heating method. Kolb et al. [13] conducted in a semicircular slice with a vertical section to study the heat transfer at the boundaries of an internally heated molten salt pool with a boundary crust. Thin cable-type heaters, with a sheath diameter of 3mm and 4m in length, provided internal heating in the pool. Kymäläinen et al. [14] investigated experimentally the heat flux distribution from a large volumetrically heated pool for the Loviisa nuclear power plant in Finland. Their experimental approach was based on using a two-dimensional slice of the Loviisa lower head, including a portion of the cylindrical vessel wall. This allowed well-controlled uniform heating using the flats as the electrodes. The pool was filled with a conduction $ZnSO_4$-H_2O solution and the current through each electrode was individually adjusted.

4. Results and Discussion

Fig. 3 shows schematic of the volumetrically heated fluid layer. There are three distinct regions: the upper boundary layer, turbulent mixing core, and lower boundary layer. In this test both boundaries are kept at the same temperature. Thus the layer is stratified only above its mid-plane under pure conduction and the onset of convection is characterized in the upper half layer. The recirculating core region moves both upward and downward as the Rayleigh number increases. The development of a thin thermal boundary layer at the upper surface is quite rapid as the Rayleigh number increases. Thus, the Nusselt number has a strong dependency on the Rayleigh number. On the other hand, the lower region of the layer is practically dominated by conduction at all Rayleigh numbers.

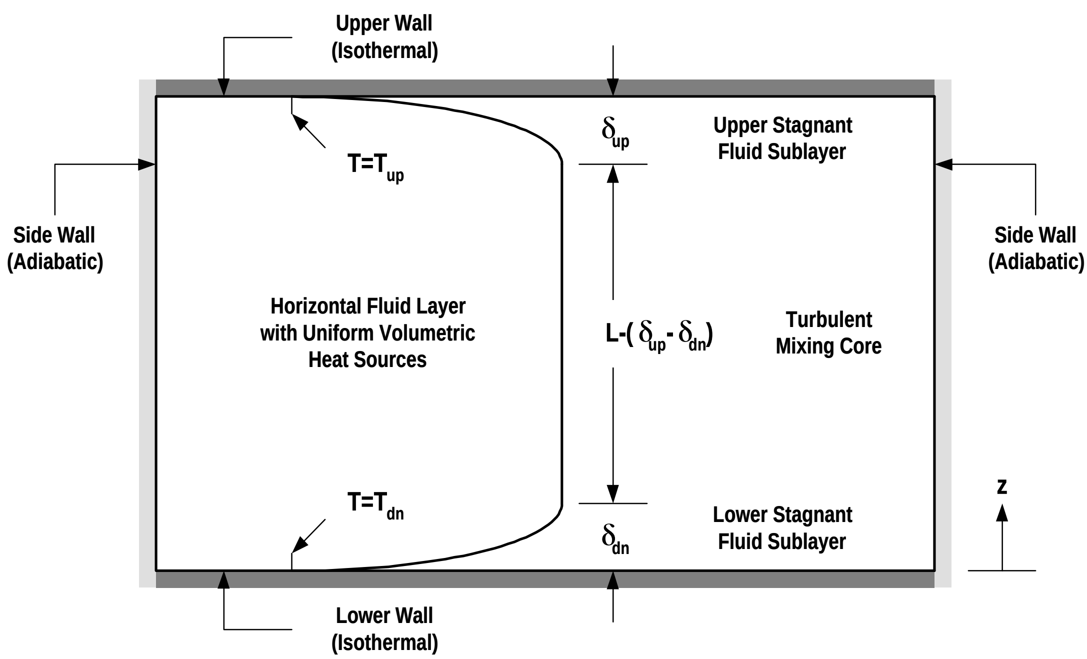

Fig. 3. Schematic of the volumetrically heated horizontal fluid layer

Downward plumes occur randomly and give the flow a locally unsteady character. At the lower surface, conduction is dominated by the downward plumes only occasionally, and convective transport is but a small fraction of the downward heat flux. The weak dependence of the lower surface Nusselt number on the Rayleigh number is the result. For low values of the modified Rayleigh number, Ra´, the turbulence intensity is small and the turbulent or eddy viscosity is negligible in comparison with the molecular viscosity. Such flows can be characterized as laminar and, furthermore, they would be steady if the internal heat source and boundary conditions vary slowly over the time scale of interest.

However, for much higher Rayleigh number, the frequency of plume occurrence increases, and the plumes become unstable as well. Plume instability results in the release of thermals or eddies from the upper surface. The flow is characterized as turbulent and unsteady for large values of the modified Rayleigh number, Ra´, at least in domains of vigorous mixing and high turbulent intensity. The molecular viscosity is small in relation to the eddy viscosity. Regions of laminar, transition and turbulent flow may coexist in the cavity according to the thermal stratification of the molten pool.

Fig. 4 demonstrates the dimensionless temperature profiles. Experimental results are in good agreement with those by Kulacki and Goldstein [1]. As the Rayleigh number is increased, mean temperature profiles are characterized by a well mixed nearly isothermal core and boundary layers on the upper and lower surfaces.

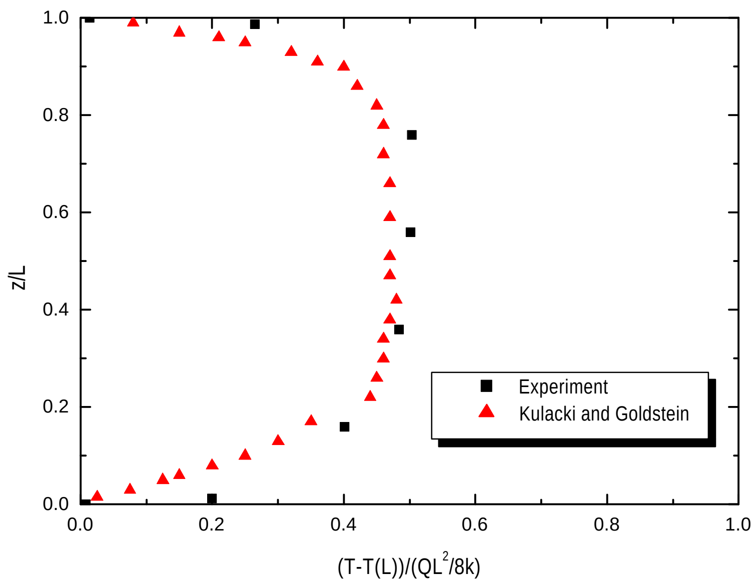

Fig. 4. Horizontally averaged temperature profiles

The natural convection heat transfer involving internal heat generation is represented by the modified Rayleigh number, Ra´, which quantifies the internal heat source and hence the strength of the buoyancy force. Natural or free convection phenomena can be scaled in terms of the Grashof number, Gr, the Prandtl number, Pr, and additionally, the Dammkohler number, Da, in the presence of volumetric heat sources. The dimensionless numbers are defined as

$$Gr = \frac{g\beta\Delta T L^3}{\nu^2};\; Pr = \frac{\nu}{\alpha};\; Da = \frac{QL^2}{k\Delta T} \tag{1}$$

The Rayleigh number, Ra, can be used to characterize the heat transfer in natural or free convection problems, including those involving external heat sources or external heating such as heating from below. This dimensionless number is defined as

$$\mathrm{Ra} = \mathrm{Gr}\,\mathrm{Pr} = \frac{g\beta\Delta T L^3}{\alpha\nu}\,;\; a = \frac{k}{r c_p}\,;\; \nu = \frac{\mu}{\rho} \qquad (2)$$

The preceding equation relates the buoyancy and viscous forces, which are linearly related via the factor, Gr/Re, but other dependencies are also present. The modified Rayleigh number, Ra´, is germane to free or natural convection problems with internal heat sources, which is defined as

$$\mathrm{Ra'} = \mathrm{RaDa} = \mathrm{Gr}\,\mathrm{Pr}\,\mathrm{Da} = \frac{g\beta Q L^5}{\alpha\nu k} \qquad (3)$$

Heat transfer data in terms of the Nusselt number versus Rayleigh number are represented in the familiar form as

$$\mathrm{Nu} = C\left(\mathrm{Ra}'\right)^m \qquad (4)$$

When the Prandtl number effects are taken into account, the modified form is

$$\mathrm{Nu} = C\left(\mathrm{Ra}'\right)^m \left(\mathrm{Pr}\right)^n \qquad (5)$$

In this work, Equation (5) was not obtained because of insufficient experiment data. Fig. 5 indicates that the Nusselt number,is low at low Pr relative to some correlations with high Pr. Theofanous et al. [15] concluded that the fluid Pr has negligible effect on high Ra natural convection heat transfer in the range of 2.5 to 11. On the other hand, Nourgaliev et al. [9] reported that the effect of Pr on Nu at the bottom surface of the enclosures is significant and amplifying with increasing Ra. Thus, the low Pr of the core melt could be an additional reason for excessive heat transfer in the downward region.

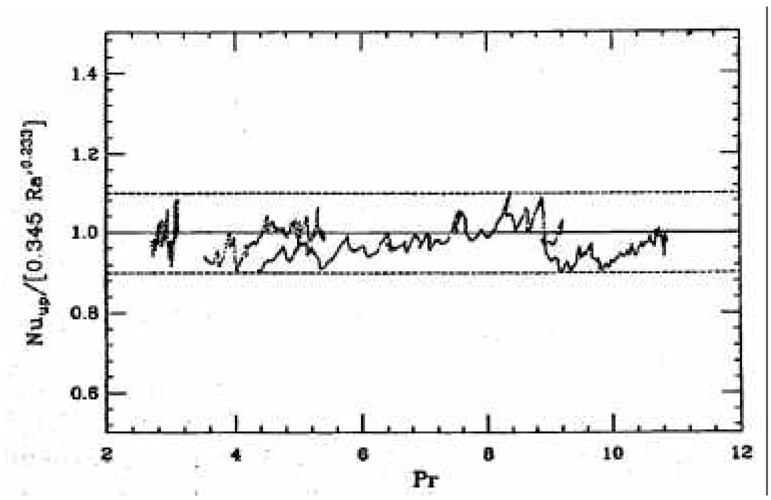

Fig. 5. Mini-ACOPO results showing the Prandtl number effect (Taken from Ref. 15)

Fig. 6 presents the Nusselt number and Rayleigh number relation. Experimental results with moderate Pr (~6.5) fluid are in fair agreement with other correlations. However, experimental results with low Pr (~0.7) fluid underestimate both upward and downward Nu.

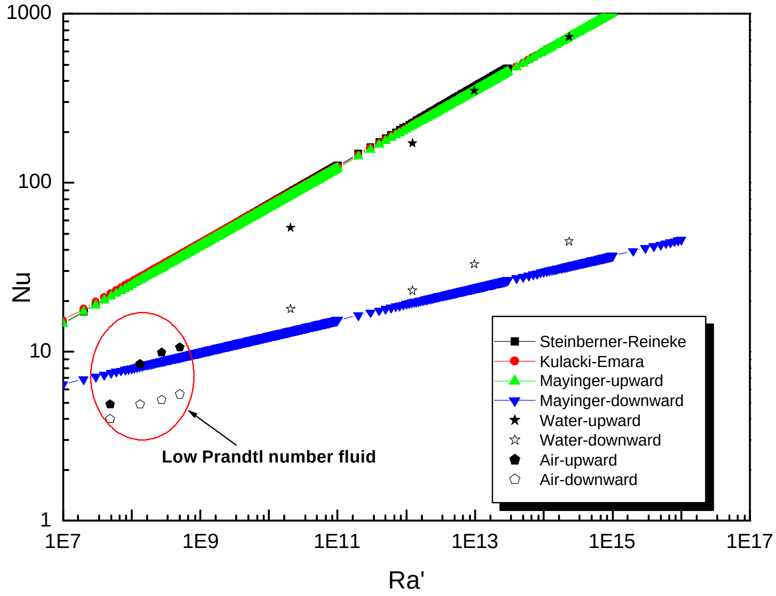

Fig. 6. Natural convection relation in pool

5. Conclusions

Comparison of the present results against the other correlations reported in the literature revealed that the Nusselt number is obviously underestimated. It was shown that dependence of heat transfer characteristics on Pr is not minor in a rectangular pool. Although the low and high Prandtl number data do not overlap in the current experimental results, the same range of Pr data will be obtained changing the test conditions in another set of experiments. Relatively few literatures exist for the Prandtl number effect. More experimental and analytical efforts are needed to quantify the effect of the Prandtl number on the Nusselt number.

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Nomenclature

C	coefficient in Equations (4) and (5)	
c_p	specific heat	[J/kg· K]
Da	Dammkohler number	
g	gravitational acceleration	[m/s^2]
Gr	Grashof number	
k	thermal conductivity	[W/m· K]
L	pool depth	[m]
Pr	Prandtl number	
Q	heat source	[W/m^3]
Ra	Rayleigh number	
Ra´	modified Rayleigh number	
DT	temperature difference	[K]

Greek Letters

a	thermal diffusivity	[m^2/s]
b	thermal expansion coefficient	[K^{-1}]
d	boundary layer thickness	[m]
m	dynamic viscosity	[N· s/m^2]
n	kinematic viscosity	[m^2/s]
r	density	[kg/m^3]

Subscript

dn	downward
up	upward

Superscript

m	exponent in Equation (4)
n	exponent in Equations (4) and (5)

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