

**THERMAL HYDRAULIC ANALYSIS OF MOLTEN CORE MATERIAL
CONTAINED WITHIN LOWER HEAD OF PWR PRESSURE VESSEL**

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PREFACE

In case of a severe reactor accident, leading to full or partial degradation of the fuel assemblies, molten core material will be collected in the lower head of the Reactor Pressure Vessel (RPV). This heat generating, naturally circulating fluid will thermally attack the pressure vessel wall that is the last barrier against the release of the radioactivity to the containment. In this case, most crucial action is to arrest the accident scenario and ensure the pressure vessel integrity. Propagation of the accident could only be stopped with the help of detailed knowledge of the possible events, which could occur afterwards. Thermal hydraulic analysis could develop this essential understanding of the events. In this study, a detailed analysis of the buoyant flow in RPV lower head, heat transfer from the corium to RPV wall and pool surface radiation towards the upper non-wetted wall has been carried out. Whole the analysis was accomplished with software, in FORTRAN-90, developed for this purpose which is based upon Finite Volume Method.

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During this long period of my stay here in the university, my wife and son suffered a lot due to my absence for which I am very much obliged to them.

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ABBREVIATIONS

BWR	:Boiling Water Reactor
FVM	:Finite Volume Method
LWR	:Light Water Reactor
PWR	:Pressurised Water Reactor
RPV	:Reactor Pressure Vessel
TMI-2	:Three Mile Island Unit 2



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LIST OF SYMBOLS

- A** :Solid to fluid thermal diffusivity ratio, (α_s/α_f)
A_r :Area of radiating surface
b :Thickness of the solid wall [m]
C :Convective mass flux [kg/m²]
C_p :Specific heat at constant pressure [J/kg.K]
D :Diameter of the semi-circular pool [m]
D_f :Diffusion flux
e :Prescribed error
E_b :Black body emissive power
E_r :Emissive power of a radiating gray surface [J/m².s]
F :View factor for radiative heat transfer
g :Gravitational acceleration [m/s²]
H :Dimension of the square cavity [m]
H_r :Incident radiant flux [J/m²]
J :Net radiative heat flux [J/m²]
K :Thermal conductivity ratio, $(= k_s/k_f)$
k_f :Thermal conductivity of the fluid [W/mK]
k_s :Thermal conductivity of the solid wall [W/mK]
Γ :Diffusion coefficient
NR :Total number of sub-regions for radiative heat transfer
NR_C :Conduction-radiation interaction parameter, $(= D \sigma T_w^4 / k_s \Delta T)$
Nu :Local Nusselt number
 $\overline{Nu}$:Average Nusselt number
p :Physical pressure, [N/m²]
P :Dimensionless pressure, $(= \frac{p D^2}{\rho \alpha^2 (Pr Ra)^{4/5}})$
P' :Pressure correction [N/m²]
P* :Estimated pressure field [N/m²]
Pe :Peclet number
Pr :Prandtl number, $(= \nu / \alpha)$
q'' :Heat flux [W/m²]
q''' :Uniform volumetric heat source [W/m³]
q_r :Net radiative flux [J/m².s]
r :Radial Co-ordinate [m]
r₁ :Inner radius of the cavity [m]
R :Dimensionless radial Co-ordinate, $(= r/D)$
R₁ :Dimensionless inner radius of the cavity, (r_1/D)
Ra :Rayleigh number based on the cavity diameter, $(= \frac{g \beta q''' D^5}{k \alpha \nu})$

Ra_R :Rayleigh number based on the radius of the cavity, $(= \frac{g\beta q'' R^5}{k\alpha\nu})$

S :Source term

S_i :Spatial position

T :Temperature [K]

t :Physical time [s]

T_u :Temperature of the upper structures [K]

ΔT :Reference temperature, $(= \frac{q'' D^2}{k_f (Ra Pr)^{1/5}})$

u, v :Velocity components in r, ϕ directions [m/s]

U :Dimensionless velocity components in R direction, $(= \frac{u}{(\alpha/D)(Ra Pr)^{2/5}})$

V :Dimensionless velocity component in ϕ direction, $(= \frac{v}{(\alpha/D)(Ra Pr)^{2/5}})$

$\hat{u}, \hat{v}$:Pseudovelocities along r, ϕ direction

u^{*}, v^{*} :Estimated velocities along r, ϕ direction [m/s]

x, y :Cartesian Co-ordinates [m]

X, Y :Dimensionless Cartesian Co-ordinates, $(= x, y/D)$

GREEK SYMBOLS

α_f :Thermal diffusivity of the fluid [m²/s]

α_r :Absorptivity of a radiating surface

α_s :Thermal diffusivity of the solid wall [m²/s]

β :Coefficient of volumetric thermal expansion of the fluid [K⁻¹]

δ :Kronecker delta

δ_b :Boundary layer thickness [m]

ε :Emissivity of radiating surface

Δ :Step size on the wall

γ :Stretching parameter

Γ :Diffusion coefficient [W/mK]

θ :Dimensionless temperature, $(\frac{T - T_w}{\Delta T})$

ν :Kinematic viscosity of the fluid [m²/s]

ρ :Density of the fluid [kg/m³]

ρ_r :Reflectance of a radiating surface

σ :Stefan-Boltzmann constant

τ :Dimensionless time, $(= \frac{t\alpha}{D^2} (Ra Pr)^{2/5})$

ϕ :Angular coordinate

Φ :Viscous dissipation function

SUBSCRIPTS

- bottom** :Wall enclosing the semi-circular pool from bottom
- f** :Fluid domain
- i** :Grid point index
- p** :Value for fluid in the pool
- r** :Value for radiative transfer
- s** :Solid wall
- top** :Top surface of the pool
- u** :Value at upper structures
- w** :Value at the outer surface of the wall



BİR BASINÇLI SU REAKTÖRÜNÜN ALT REAKTÖR KABINDA ERGİMİŞ KALP MALZEMESİNİN TERMO HİDROLİK ANALİZİ

ÖZET

Bu çalışmanın önemi, reaktör kalbinin kısmen veya tamamen erimesi ile sonuçlanan bir şiddetli reaktör kazasının kontrol altına alınmasından kaynaklanmaktadır. Böyle bir durumda ergimiş malzeme basınç kabının alt tarafında, bir seri ergime ve katılaşma hallerinden geçtikten sonra toplanacaktır. Reaktördeki radyoaktivitenin açığa çıkmasına karşı son engel olan reaktör basınç kabı koryum tarafından ısıtılarak etkilenir. Uranyuma dayanan reaktör yakıtı ve yanmış yakıttan gelen fisyon ürünleri koryumda mevcuttur. Yarı ömürlerine göre fisyon ürünlerinin bozunumu, ergimiş kütle içinde az veya çok homojen olan bir iç ısı kaynağı oluşturur. Seramik yakıt bileşenlerinin yüksek ergime noktası ve fisyon ürünlerinden kaynaklanan radyoaktivite nedeni ile sistem deneysel olarak yapılamaz hale gelir. Bu nedenle reaktör basınç kabında oluşan koryum havuzunun içindeki ısıtım hidrolik olayı incelemek için sadece sayısal benzeşim tek yol olarak kalır.

Problemle ilgili akış ve ısı transferi karakteristiklerinin anlaşılması kazayı kontrol etmek ve böylece radyoaktivitenin kaptan yayılması ve atmosfere çıkması tehlikesini azaltmak mümkün olabilir.

Bu çalışma düşük Rayleigh sayılarında yandan ısıtılmalı bir kare kapta doğal taşınımın incelenmesi ile başlamıştır. Bu problem hesaplamalı akışkanlar dinamiğinde bir benchmark problemi olması sebebi ile seçilmiştir. Sonuçlar daha önce yapılan çalışmalar ile karşılaştırıldığında çok iyi bir uyum elde edilmiştir.

İkinci olarak düşük Rayleigh sayıları için yazılan program içten ısıtılmalı bir kare kapta doğal taşınım için analiz edilmiştir. Sonuçlar yayınlanmış diğer sonuçlarla karşılaştırıldığında çok iyi bir uyum içindedir. Daha önce yayınlanmış çalışmalarda içinde ısı üretimi olan ve yüksek Rayleigh sayıları için ölçekleme mevcut değildir, böylece bu hal için bir ölçekleme gerçekleştirilmiş ve sonuçlar yayınlanmıştır.

Reaktör basınç kabı bir konjugate problem oluşturan kalın paslanmaz çelik bir duvara sahiptir. Konjugate problemin analizi için hayali noktalar yerleştirme yöntemi kullanılmıştır. Bu konuda elde edilen sonuçlar da yayınlanmıştır.

Koryum düşük Pr sayılı bir akışa sahiptir, böylece koryumdaki doğal akışın analizi, kalın duvarda bir kare kapta yapılmıştır ve sonuçlar bir uluslararası konferans bildirisi olarak sunulmuştur.

Yarı silindirik kapta kalın duvarın etkisini anlamak için analiz içten ısıtılmalı bir yarı silindirik kap için yapılmıştır. Bu sonuçlar, duvarları sabit sıcaklıkta olan bir yarı silindirik kap ile karşılaştırılmış ve fark incelenmiştir.

Yarı küresel basınç kabının özel durumu nedeni ile kabın sadece yarısı incelenmiştir. Ortalama Nusselt sayısı için sonuçlar karşılaştırılabilir olmasına rağmen, akışın aslında simetrik olmaması sebebi ile, nihai inceleme için yarı silindirik kap seçilmiştir.

Son incelenen geometri mümkün olduğu kadar gerçek duruma yakın seçilmiştir. Koryum havuzundaki sıcaklık 3500 °K ulaşabilir, bu nedenle havuz yüzeyinden ışınlama ısı transferi bu analize dahil edilmiştir. Üç ısı transferi mekanizması, iletim, taşınım ve ışınlama bu analizde mevcuttur.

Sonuçlar göstermiştir ki, havuzdaki sıcaklıklar havuzun alt tarafında iyi bir tabakalaşma göstermiş ve üst tarafında taşınımın hakim olduğu küçük bir bölge oluşmuştur. Havuz yüzeyinin neşretme oranı ve $\Delta T/T_w$ parametresi de ayrıntılı olarak incelenmiştir. Ra sayısı 3.2×10^7 den 3.2×10^{13} e kadar çalışılmıştır, son değer türbülanslı akışın başlama sınırındır. Aynı zamanda sonuçlar gösteriyorki, reaktör basınç kabının hasara uğramasının mümkün olduğu nokta, basınç kabının duvarı ile serbest yüzeyin üst temas noktasına yakındır. Kabın alt bölgesi koryumun ısı etkisinden hiç bir zaman zarar görmeyecektir.

Böylece sunulan problemin ayrıntılı bir incelemesi yapılmıştır. Bu analizin, söz konusu problemle ilgili ısıl hidrolik olayın anlaşılmasına önemli ölçüde katkı sağlayacağı umulmaktadır.

Aşağıdaki bölümlerde, bu doktora çalışması sırasında oluşturulan sayısal çalışmanın ayrıntıları anlatılmıştır. Bölüm 2 de, farklı haller için denklemler boyutlu ve boyutsuz olarak sunulmuştur. Yüksek Rayleigh sayıları için ölçekleme analizi de bu bölümde anlatılmıştır.

Bölüm 3, bu çalışmada kullanılan ızgara yapısını ve basınç ve basınç düzeltme denklemlerinin türetilmesini içermektedir. Bölüm 4 de sayısal algoritmalar ve bilgisayar programının düzenlenmesi tanıtılmıştır. Bu bölümde aynı zamanda bilgisayar programı ile ilgili bir akış diyagramı verilmiştir. İki ana parçadan oluşan bölüm 5 de; ilk kısımda önceki sonuçlarla karşılaştırmalar sunulmuştur, ikinci kısımda ise sonuçlar ve tartışmalar verilmiştir.

THERMAL HYDRAULIC ANALYSIS OF MOLTEN CORE MATERIAL CONTAINED WITHIN LOWER HEAD OF PWR PRESSURE VESSEL

SUMMARY

The importance of the present analysis stems from the interest to arrest a severe reactor accident, resulting in partial or full degradation/melting of the core assembly. In such a case the molten material will relocate in to the lower head of the pressure vessel, after passing through a series of molten and refrozen states. Reactor Pressure Vessel (RPV), which is the last barrier against the release of the radioactivity to the containment, is thermally attacked by the corium. Uranium based reactor fuel and fission products from the burned fuel are present in the corium. Fission products' decay with their respective half lives, introduces an internal heat source, which is more or less uniform within the molten mass. Due to high melting point of the ceramic fuel constituents and the radioactivity associated with fission products, the system becomes unmanageable experimentally. Thus leaving numerical simulation only viable tool to look for thermal hydraulic phenomenon occurring inside the corium pool formed in the RPV lower head.

The understanding of the flow and heat transfer characteristics associated with the problem can possibly lead to accident management scheme, thus reducing the risk of release of radioactivity to the containment and eventually to the atmosphere.

Present analysis was started with the study of buoyant flow in a discretely heated square cavity for low Rayleigh numbers. This problem was selected due to its importance in the Computational Fluid Dynamics literature as a benchmark solution. A very good agreement was obtained between the results from present software and those from the open literature.

Secondly natural convection flow in an internally heated square cavity was analysed using the present software for low Ra numbers. Results were compared with published results of other researchers. Both results showed a very good agreement.

As scaling for high Ra number with internal heat generation was not available in the open literature, so a scaling was developed for such a case. These scaling and obtained results were published.

Reactor pressure vessel has a thick stainless steel wall, thus introducing a conjugate problem. For the analysis of the conjugate problem ghost nodes induction method was adopted. Then the results obtained for an internally heated square cavity having thick conducting walls were presented in a paper.

Corium is a low Pr number fluid, so the analysis of the buoyant flow in corium contained in a thick walled square cavity was performed and results were presented as a conference paper.

To understand the effect of thick conducting wall enclosing a semi-circular cavity from bottom, analysis was carried out for an internally heated semi-circular cavity. These results were compared with a semi-circular cavity with isothermal walls and the difference was analysed.

Due to the peculiar layout of the hemi-spherical RPV lower head, only half of the cavity was analysed. The results indicated that although average Nusselt number was comparable but the non-symmetric nature of the flow could not be obtained. Therefore, the semi-circular cavity was selected for final analysis.

Final analysis geometry was selected to represent actual situation as closely as possible. As temperature in the corium pool will reach 3500 °K, so radiation heat transfer from the pool surface was included in the analysis. The three heat transfer mechanisms, conduction, convection and radiation, are present in the analysis. Only radiative heat transfer from the pool surface and associated radiating surfaces was included.

Results show that the temperatures in the pool are well stratified with convection dominated region confined to upper part of the pool. Effect of pool surface emissivity and parameter $\Delta T/T_w$ has also been analysed in detail. Ra number range of 3.2×10^7 to 3.2×10^{13} has been studied with last value at the margin of initiation of turbulent flow. Results also indicate that most probable point of failure of the RPV wall is close to the point of contact of the free surface with RPV wall. Also the stagnation point will never be in danger from thermal attack of the contained corium.

Thus a detail analysis of the present problem has been performed. It is hoped that this analysis will greatly increase the understanding of the thermal hydraulic phenomenon associated with the present problem.

In the following chapters details of the computational work performed during present doctoral dissertation has been presented. In chapter 2, governing equations for different cases analysed are presented in full dimensional form together with dimensionless equivalents. Also, scale analysis for high Rayleigh number buoyant flow has been outlined. Chapter 3 includes the summary of grid layout employed for the analysis and derivation of pressure and pressure correction equations. Organisation of software and numerical methodology adopted for the present analysis has been listed in chapter 4. A flow diagram of the algorithm is also included in chapter 4. Chapter 5 has two main parts, in first part benchmark results are presented while in second part results and conclusions concerning the present analysis are given.

1. INTRODUCTION

1.1 The Problem

Nuclear reactors have the highest core power production density, as compared with other power generation systems. To transport such high quantities of heat out of the reactor core, coolant (water for Light Water Reactors or LWRs) having flow rate of millions of kilograms per hour is circulated through the core “Table 1.1”. This water circulation is accomplished by using 2 to 4 coolant loops “Table 1.1”, depending upon the total heat output of the system.

Table 1.1

Typical Thermal Hydraulic Characteristics of a PWR (Todreas and Kazimi, 1989)

Net electrical power	1148 Mwe
Primary coolant flow rate	10^7 kg /h
Primary water inlet temperature	286 °C
Primary water outlet temperature	324 °C
Number of primary coolant loops	4
Core power density (core average)	105 kW / L

Partial or full blockage or leakage in any one of the coolant loops will result in reduction of water flow through the core, leading to the condition known as Loss of Coolant Accident or LOCA. In the LWRs systems the Emergency Core Cooling Systems (ECCSs) are designed to provide core cooling for different postulated accidents up to and including the design basis loss-of-coolant accident. If, however, it is assumed that the ECCS does not function properly, the core would be inadequately cooled. Due to huge total heat output of the reactor core, water available within the RPV would not be able to fulfil the cooling requirements. Within a few minutes of the start of LOCA, core assembly will start to dry up beginning from the top of the core. Temperature of the fuel rods will increase to a point where

oxidation of the cladding metal. This metal-steam reaction is highly exothermic (**Henrie, 1989**), thus fuel temperatures will rise at an increasing rate. Thus in a recursive way fuel temperature will shoot up to dangerous levels.

If water can not be inserted in the pressure vessel, this ever-increasing temperature will result in the partial or full degradation/melting of the core assembly (**Thinnes and Moore, 1989**). The exact process in which core debris would form, is dependent upon accident sequence and reactor design. After passing through a series of refrozen and melting scenarios (**Nourgaliev et al., 1997a**) the final configuration may contain two main parts of the core debris.

- (a) Part of the core debris could resolidify in the lower part of the core assembly.
- (b) Second portion of the core debris may settle in the lower plenum of the reactor pressure vessel as a mixture of various materials constituting the reactor core.

This heat generating molten mass is known as corium.

Theoretically, if no water is added into the core during the accident propagation stage, most of the core debris eventually may locate in the lower plenum (**Dosanjh, 1989**). This core debris relocated in the lower plenum will be transformed into a molten core material pool due to the fission product decay heat. This relocated and melted corium pool will have the configuration as shown in “Figure 1.1” and will have four main parts (**Theofanous et. al., 1994**).

- (a) Lower crust
- (b) Upper crust
- (c) Molten corium pool
- (d) Molten steel layer

As a result of the fission product decay in the corium, this molten mass will be heated to a temperature where natural convection flow will begin within the pool body with flow moving upward. At the surface this naturally circulating fluid will erode away the top crust (**Park and Dhir, 1992**) and thus will attack the lower crust on its way along the curved pressure vessel wall.

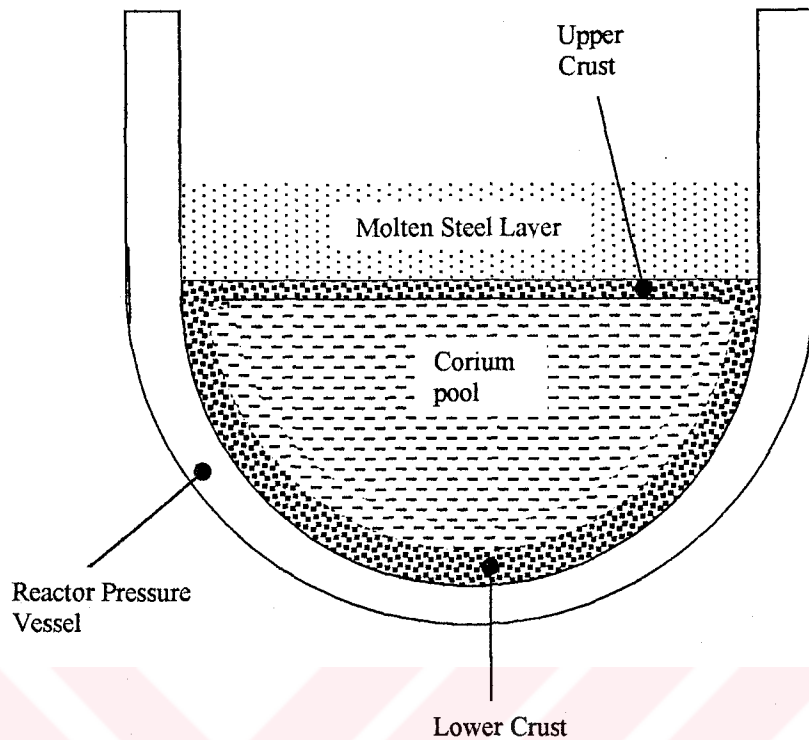


Figure 1.1 Different parts of the corium after its relocation into the lower head of RPV

Under the severe accident scenario, as discussed above, the most crucial action for the recovery from the accident-state is to cool the core debris/corium and prevent its attack on the remaining fission products barriers. One way of preventing attack on the containment structures is to retain the core debris within the pressure vessel. In present time containment designs, substantial amount of water is available or could be made available around the lower parts of the reactor pressure vessel and hot or cold legs. This water could help in removing large amount of heat from the molten pool, but if pressure vessel retains its integrity.

This scenario raises the question about the margin of safety against pressure vessel rupture (**Thinnes and Moore, 1989**). Natural convection heat transfer in internally heated liquid pools has been studied, recently, to understand the heat transfer and fluid flow phenomenon within the pool and the extent of thermal attack on the pressure vessel wall. This understanding could be used to determine the consequences of severe (meltdown) accidents in light water reactors and to devise accident management schemes (**Nourgaliev et al., 1997**). To assess the potential for

in-vessel retention of the corium, one should know the heat transfer characteristics from the melt pool to the reactor pressure vessel.

In present study, this hemispherical pool containing molten core material has been numerically analysed to obtain the fluid flow and heat transfer parameters for 2-D polar ($r-\theta$) Co-ordinate system. Due to exceptional thick pressure vessel wall “Table 1.2” a conjugate analysis has been envisaged as essential.

Table 1.2

Dimensions of RPV for a Typical PWR (Todreas and Kazimi, 1989)

Inside diameter	4.83 m
Height	13.4 m
Wall thickness	224 mm
Material	Carbon steel

Temperature of the corium inside the pool will reach upto 3500 °K “Table 1.3”. At such high temperature radiation heat transfer becomes an important factor. Thus radiation heat transfer from the pool surface has been included in the present study. A complete management of the geometry and the heat flow paths considered are shown in “Figure 1.2”. Material properties of the corium “Table 1.3” are considered to be temperature independent and flow is assumed to be laminar. To obtain Radiative Transfer Equations (RTE) enclosure theory has been utilised.

Thus present problem in complete form consists of four main parts.

- (a) Natural convection inside the corium pool initiated by internal heating.
- (b) Conjugate analysis of heat transfer from pool body to lower plenum wall.
- (c) Radiation from the corium pool surface.
- (d) Interaction of radiation and conduction for upper walls of the reactor pressure vessel.

Table 1.3

Material Properties of Corium and RPV shell (Park and Dhir, 1992)

Material	Properties
Corium	$\text{UO}_2 + \text{Zircalloy-4} + \text{SS}$ etc $\rho_f = 8200 \text{ kg / m}^3$ $C_{p,f} = 470 \text{ J / kg.K}$ $K_f = 3.6 \text{ W / m.K}$ $\alpha_f = 9.3 \times 10^{-7} \text{ m}^2 / \text{s}$ $\nu_f = 4 \times 10^{-7} \text{ m}^2 / \text{s}$ $\beta_f = 1.4 \times 10^{-4} \text{ K}^{-1}$ Melting temperature = 3153 °K (UO ₂)
RPV shell	SS clad Carbon steel $\rho_s = 8000 \text{ kg / m}^3$ $C_{p,s} = 600 \text{ J / kg.K}$ $k_s = 50 \text{ W / m.K}$ $\alpha_s = 1.04 \times 10^{-5} \text{ m}^2 / \text{s}$ Melting temperature = 1811 °K

According to best knowledge of the author, in the open literature an analysis of this much detail is not available. The methodology adopted for present analysis consists of three basic steps. First, in order to understand basic techniques, system was analysed using Cartesian coordinates. Secondly, the main results were presented as papers in various international journals and conferences, using Cartesian and polar Co-ordinates. At the end these methods were used to numerically analyse the actual system. It is hoped that the results from present analysis will greatly increase the basic understanding of the cooling of molten core material inside the LWRs pressure vessel lower head and will be helpful for devising severe accident management scenario.

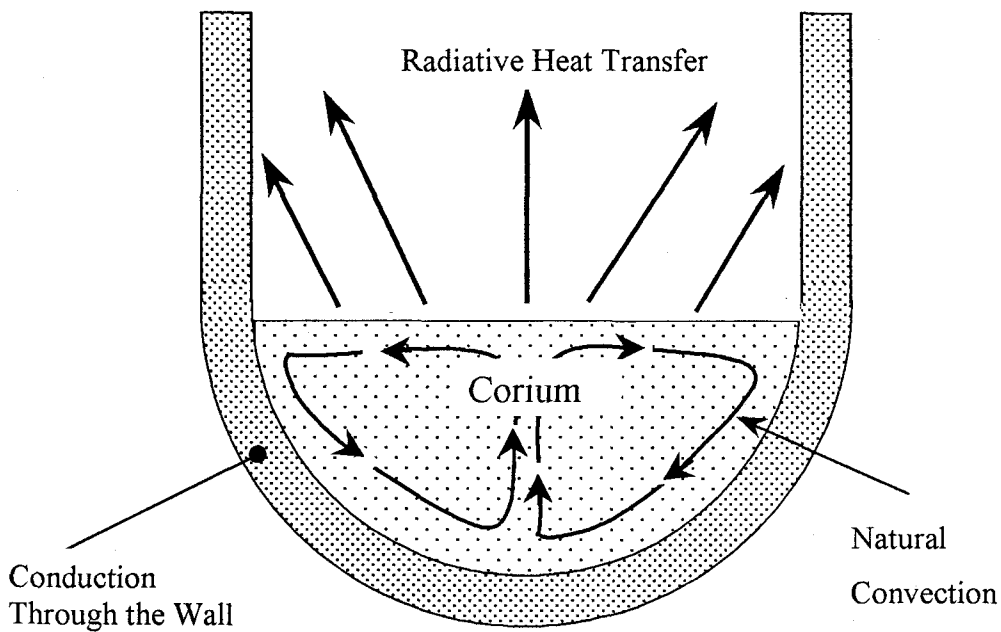


Figure 1.2 Simplified problem as simulated in the present study

1.2 Literature Survey

After TMI-2 accident, which resulted in partial degradation of the core assembly, researchers have tried to understand the sequence, effects and remedial measures for such severe reactor accidents. Some of them tried to understand the heat transfer phenomenon in the heat generating pool while others tried to analyse the accident sequence of TMI-2, based upon the instrumentation data.

Baker et al. (1976) studied the existing heat transfer correlations obtained from the experimental data. These equations were cast into a form suitable for analysis of post-accident heat removal in fast reactors. They observed that downward heat transfer rate can be accounted for by conduction alone, while the upward heat transfer rate is largely controlled by convection. They developed a new correlation to predict upward and downward heat fluxes from internally heated fluid layer with unequal boundary conditions.

In an effort to investigate the post-accident heat removal phenomenon **Faw and Baker (1976)** used microwave heating for the simulation of their experiment. Thus

they analysed heat transfer from a horizontal fluid layer of internally heated liquid to a melting solid. Experiments were designed to simulate heat transfer and melt-through process of importance in the analysis of post-accident heat removal capabilities of nuclear reactors. Results indicated that the melting and displacement process had no effect on the heat transfer process.

To describe the behaviour of the molten core, data on the properties of such core melts are required. These properties are dependent on the composition and structure of the melts. Available experimental evidence was used by **Nazare et al. (1977)** to define four types of melts that would occur in case of severe reactor accident. Theoretical and experimental results were reviewed to estimate the material properties of these four types of melts.

Thermal convection of a heat generating fluid in a rectangular enclosure, cooled at the side walls, was investigated by **Bergholz (1980)**. He performed investigations for the boundary layer regime at large internal Ra numbers. A boundary layer analysis was employed to derive a system of equations valid near the vertical walls and a corresponding system valid in the core of the enclosure. It was shown that most of the heat generated within the fluid is removed through upper half of the vertical walls.

Post dryout melt propagation in a particle bed formed by the collapse of the fuel rods in a severe reactor accident was investigated by **Dosanji (1989)**. They used a 2-D model that incorporated the presence of oxidic (UO_2 , ZrO_2) as well as metallic constituents (eg., Zirconium). He showed that the corium pool grows more rapidly if Zirconium is present in the particle bed, because Zirconium dissolves both UO_2 and ZrO_2 .

Epstein and Fauske (1989) used the data obtained from the TMI-2 reactor instrumentation to calculate the core relocation scenario in to the lower head of the pressure vessel. This analysis gives a very good insight in to the possible core debris meltdown and relocation scenario.

Henrie (1989) used forensic engineering approach to evaluate recorded data and damaged as well as survived components after the TMI-2 accident. Analysis results reported showed that the timing of the TMI-2 accident could be derived by forensic engineering method. Beginning of the fuel cladding rupture i.e. zirconium-water

reaction, could be derived from the recorded data. Total hydrogen produced during the accident and relocation of the molten core material in to the lower head of the pressure vessel was also shown to be evident from the available data.

Hobbins et al. (1989) provided details of the core debris locations inside the pressure vessel after TMI-2 severe accident. Chemical composition of core melts, approximate amounts and locations has been provided, in order to understand the behaviour of the core debris during that accident. From chemistry of the core melts it was evident that the attack of the core melt on pressure vessel wall was thermal in nature. They concluded that understanding and modelling of the core melt progression phenomena observed in TMI-2 accident are essential elements in the analysis of the consequences of severe accidents.

A two-dimensional finite element model was developed by **Moore et al. (1989)** to simulate the TMI-2 core heatup and subsequent cooling of the consolidated core region after the relocation of the core material to the lower plenum of the reactor. They used their 2-D model to calculate the heatup and subsequent failure of the corium pool formed in lower part of the core assembly. The cooldown calculations indicated that as long as the core remained covered with coolant, the core configuration would remain thermally stable.

The structural finite element creep rupture analysis was carried out by **Thinnes and Moore (1989)** to investigate the response of the reactor pressure wall towards a prescribed thermal attack. They used the temperature transient, which was judged to be a challenge to the structural capacity of the reactor pressure vessel of TMI-2 reactor. Their analysis provided an insight in to the possible creep rupture mechanisms of the pressure vessel and a method to evaluate margin to failure once maximum wall temperatures are known. It was concluded that failure of the pressure vessel lower head resulting from large temperature gradients across the vessel wall is not probable.

Park and Dhir (1992) analysed the effect of outside cooling by water on the behaviour of the PWR pressure vessel lower head. They considered two cases with different heat transfer coefficients taken from open literature, adding a radiation heat transfer from the surface of the pool. Due to the difference in heat transfer coefficients both cases could not coincide in the final conclusions.

Henry et al. (1993) reported experimental observations about the heat flux at the outer surface of the reactor pressure vessel wall and demonstrated that these fluxes are typical of nucleate boiling heat transfer. Thus they concluded that heat flux is, therefore, limited by thermal conduction through the carbon steel vessel wall.

An experimental program dedicated to the analysis of chemical interactions between core materials, that may occur with increasing temperature up to complete melting has been reported by **Hofmann et al. (1994)**. They studied high temperature material behaviour of PWR, BWR and other reactors' fuel elements in large scale integrated experiments. In many cases reaction products was reported to be liquid. This resulted in relocation of the liquefied components far below their melting positions. Separation of control rod material from fuel materials and a significant relocation of UO_2 dissolved in molten unoxidized zircalloy were reported.

Two region integral approach was developed by **Kune and Robert (1994)** to analyse the debris behaviour in the lower plenum under severe reactor accident conditions. Their analysis included molten corium pool, lower crust and pressure vessel wall. The influence of the external cooling of the RPV wall was modelled as nucleate boiling off the outer surface when the lower head is submerged. Data available from open literature was used for input purposes. It was concluded that RPV wall will fail if no outside cooling is present. Deficiency of the input data was especially mentioned.

Kymalainen et al. (1994) reported experimental results on the heat flux distribution at the boundaries of volumetrically heated pools at high Rayleigh numbers to be directly relevant to the problem of retention of a molten corium pool inside the lower head of a pressure vessel. They found that downward flux strongly depends upon the position along the curved wall and that the flux along the lower curved wall seems to be independent of the presence and extent of the liquid pool portion above the wall.

Park et al. (1994) analysed the effect of outside cooling by water on the reactor pressure vessel lower head of Boiling Water Reactor (BWR). They considered two cases with different heat transfer coefficients taken from open literature, adding a radiation heat transfer from the surface of the pool. Due to the difference in heat transfer coefficients both cases were too much apart in their conclusions. For certain cases outside cooling of the pressure vessel was found to be effective in cooling the

inside corium pool while for other cases it was stipulated that pressure vessel will fail.

Rempe et al. (1994) analysed the behaviour of the RPV and penetrations during severe reactor accident conditions. They used thermal attack parameters from open literature as input to analyse the RPV lower head behaviour. Their results illustrated the importance of the vessel and tube geometrical parameters and material properties for predicting the RPV failure mode. Same type of mechanical analysis was also performed by **Chavez and Rempe (1994)** using Finite Element approach. They emphasised the importance of the RPV penetrations in order to understand the response towards thermal attack by core debris.

A series of experiments are reported by **Theofanous et al. (1994)** aimed to calculate the coolability limits of the reactor pressure vessel lower head. They used a configuration more appropriate for the practical interest in case of severe reactor accidents and similarly, heat flux shapes representative of the core melt contained in the lower head. Critical heat fluxes as a function of angular position on the lower head are reported and related to the observed two-phase flow regimes.

Tuomisto and Theofanous (1994) has outlined the Severe Accident Management approach (SAM) adopted at Loviisa nuclear power plant (in Finland) that is mainly based upon the idea to retain the core debris inside the RPV vessel. This is assumed to be achieved by cooling RPV from outside by water made available through different means. Also steps has been described to make the water RPV contact as easy as possible. This has been done through reducing the thermal insulation and neutron shield assembly.

The SCDAP/RELAP/COUPLE code has been employed by **Invernali et al. (1995)** to simulate the evaluation of an accident implying core damage, fuel overheating, and molten material slumping into the vessel lower head of reactor pressure vessel. The temperature distribution in the molten material and vessel metal as well as the upward and downward heat rejection paths have been investigated. Heat rejection limitation imposed by the vessel wall is found to result in increase in molten corium pool temperature in order to reject the decay heat Also they noted that if no water is present in the vessel, lower corium superheat is expected.

Spindler et al. (1995) used TOLBIAC code for the simulation of the behaviour of a molten core within a structure (pressure vessel or core catcher), taking into account the relative position of the core components, the wall ablation and the crust formation. They simulated some experimental results from the open literature.

Heat generation in a corium pool may be time dependent. To study the effect of time varying heat source on the heat transfer and fluid flow properties, **Baytaş (1996)** analysed the effect of periodic sources on the buoyancy driven flow and heat transfer characteristics.

Pool boiling from downward facing curved surfaces was studied experimentally by **El-Genk and Cheng (1996)**. A semi-scale model was used for the experiments that were designed to provide data on extent of cooling possible by flooding of LWR reactor cavity. Effect of the radius of curvature and edge angle was studied in detail. High edge angles were reported to enhance coolability while cooling was not much sensitive to total cooling area.

An integral predictive physico-numerical model was developed by **Kune and Robert (1996)** to understand and interpret debris interactions in the reactor vessel plenum such as those which took place in the TMI-2 accident. The model represents the extent of debris jet disintegration by a jet-water entrainment model that can result in two types of the debris configurations ie. particulate debris and melted debris. Both the parts were treated as separate regions and different governing principles were developed for their behaviour. As a result they obtained the RPV failure mechanisms and locations for most possible failure.

The integral physico-numerical model for the reactor vessel lower head response has been exercised for the TMI-2 accident and possible severe accident scenarios in PWR and BWR designs (**Suh and Henry, 1996a**). Final cooling of the TMI-2 core debris has been postulated due to vessel wall creep and subsequent water ingress. This phenomenon clearly demonstrated the potential for the in-vessel cooling of the reactor vessel without a need to externally submerge the lower head should such a severe accident occur as core melting and relocation.

Kim et al. (1997) used APRIL computer code to model the core melting and in-vessel corium relocation in the severe reactor accident scenario. Such studies could help to understand the extent of core damage possible during severe reactor accident

and could be translated into RPV thermal attack parameters. This study can also help to analyse the time behaviour of the accident.

Severe reactor accident will produce a fluid with much smaller Prandtl number (Pr). This Pr number effect upon the wall Nusselt number (Nu) was investigated by **Nourgaliev et al. (1997)**. They concluded that in case of small Pr number fluids lower parts of the pool will not be well stratified owing to the deep penetration of the descending boundary layer flow. Also the wall Nu number behaviour at the bottom of the pool is different both for high and low Pr number fluids. Their results indicated a strong effect of the geometry upon the flow and wall Nu number variation.

A subscale boundary-layer boiling test facility was developed (**Cheung et al., 1999**) with the help of scaling analysis to simulate the phenomenon of pool boiling and Critical Heat Flux (CHF) on the external bottom surface of a heated hemispheric vessel. Saturated and subcooled experiments were performed in the facility to measure the spatial variation of CHF and observe the underlying mechanisms, including vapour dynamics and resulting buoyancy driven two phase boundary layer flow along the downward facing hemispheric heating surface.

To investigate the heat transfer mechanism from the outer surface of the pressure vessel **El-Genk and Cheng (1999)** performed some experiments with downward facing subscale hemispheres of Steel and Aluminium. They found remarkable difference between the behaviour of the two materials, thus indicating the material aspects of the scenario.

Loctionov et al. (1999) numerically analysed reactor pressure vessel behaviour under severe accident conditions taking into account the combined processes of the vessel creep and the molten pool natural convection. They considered different types of molten pools (homogeneous, conventionally homogeneous, conventionally stratified and stratified) to cover the bounding scenario during a hypothetical severe accident. Numerical results in this paper illustrate that the large creep deformations of the vessel lower head can lead to an appearance of the gaps between the vessel surface and the molten pool crust. This phenomenon can result in a significant change in the heat transfer to the vessel wall.

Nakamura et al. (1999) utilised high frame rate neutron radiography to observe the behaviour of a high temperature molten Pb-Bi alloy dropped into a vessel both when empty and when filled with water. They concluded that this method could be reliably used for such investigations and has provided very useful insight in to the actual phenomenon.

An experimental study was performed by **Park et al. (1999)** on natural convection heat transfer with and without crust formation in a molten metal pool. They observed that when natural convection flow is developed in the pool, the overlying coolant conditions do not effect the crust formation. On the other hand when the natural convection flow is not developed, the coolant conditions effect the crust formation. Heat transfer rate was lower with crust as compared to the cases with no crust formation. Thus they concluded that the crust formation acts as thermal barrier.

During present doctorate studies some papers were published from the important results obtained, as described below.

In order to verify the high Ra number scaling, developed during the course of this study, flow within an internally heated square cavity was analysed (**Liaqat and Baytaş, 2000**).

Wall thickness of the RPV is large as compared to the systems encountered in normal conditions. In present time analysis conduction through the walls is usually ignored by taking isothermal highly conducting bounding walls. To analyse the effect of the thick walls on natural convection inside the cavity, square cavity with finite conducting walls was analysed (**Liaqat and Baytaş, 2000a**).

Conjugate natural convection flow inside heat generating square corium pool was presented in **Liaqat and Baytaş (2000b)**.

Conjugate natural convection flow inside heat generating semi-circular corium pool was presented in **Liaqat and Baytaş (2000c)**.

Due to 2-D constraint on the flow, only half of the hemi-spherical geometry could be analysed and results were presented in **Liaqat and Baytaş (2000d)**.

Finally the most complicated problem was handled which has all three heat transfer modes (Conduction, Convection and Radiation) included in it. Final results from this analysis have been presented in **Liaqat and Baytaş (2000e)**.

In the following chapters details of the computational work performed during present doctoral dissertation has been presented. Governing equations for different cases analysed are presented in chapter 2 in full dimensional form together with dimensionless equivalents. In section 2.2, scale analysis for high Rayleigh number buoyant flow has been outlined. Chapter 3 includes the summary of grid layout employed for the analysis and derivation of pressure and pressure correction equations. Organisation of software and numerical methodology adopted for the present analysis has been listed in chapter 4. A flow diagram of the algorithm is also included in chapter 4. Chapter 5 has two main parts, in first part benchmark results are presented while in second part results and conclusions concerning the present analysis are given.



2. MATHEMATICAL MODEL

2.1 Governing Equations

General governing equations are a set of differential equations describing conservation/balance of specific properties of the fluid or dependent variables. These properties are usually expressed on unit mass basis that is why these are called specific properties. Complete governing differential equation is a collection of different statements for processes affecting the balance of a given dependent variable within a unit volume. Thus these equations may contain a large number of influences, affecting the balance of certain specific property.

2.1.1 Conservation of Mass Equation

General conservation of mass equation in tensor form could be given as:

$$\frac{D \rho}{D t} + \rho \frac{\partial u_i}{\partial x_i} = 0 \quad (2.1)$$

Where ρ is the fluid density, x_i denotes particular Co-ordinate directions, u_i is the fluid velocity and t is the time. In present case, fluid is assumed to be incompressible thus “Equation 2.1” reduces to following form:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (2.2)$$

This equation is generally given by using divergence operator as:

$$\text{div } u_i = 0 \quad (2.3)$$

2.1.2 Balance of Momentum Equations

General Navier-Stokes equations express the balance of momentum for a given control volume in the flow domain and could be written in a variety of ways

depending upon the assumptions used to formulate the problem and flow domain Coordinates. The form appropriate for present analysis could be given as:

$$\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial}{\partial x_j} \left\{ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right\} + g_i \frac{\rho_i}{\rho} \quad (2.4)$$

In this equation all material properties are taken to be constant. For most problems of natural convection the density variations are mainly caused by thermal expansion of the fluid and can be expressed as:

$$\rho_i = \rho \{1 - \beta(T - T_{ref})\} \quad (2.5)$$

Where β is the volume expansion coefficient at constant pressure, positive for most of fluids, and given by the equation below:

$$\beta \approx -\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_{p'} \quad (2.6)$$

Furthermore, for most of the fluids, value of β ranges from 10^{-3} to $10^{-4} \text{ } ^\circ\text{C}^{-1}$ (**Arpaci and Larson, 1984**) and for small variations in temperature the variations in density are at the most 1%. Thus the density could be considered as constant in all governing natural convection equations, except density in buoyancy term which is famous Boussinesq approximation (**Bejan, 1995**). Thus last term ie buoyancy component can be formulated by using Boussinesq approximation that describes the relation between temperature of the fluid and its density. After substituting “Equations 2.5 and 2.6” in “Equation 2.4” final momentum equation can be written as:

$$\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p'}{\partial x_i} + \nu \frac{\partial}{\partial x_j} \left\{ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right\} + g_i (1 - \beta(T - T_{ref})) \quad (2.7)$$

Where T_{ref} is the reference temperature, held constant for the present analysis.

Furthermore, pressure variation in natural convection flow could be given as below:

$$p' = p_0 + p + \rho g \cdot x_i \quad (2.8)$$

Where p_0 is the applied constant pressure, p is dynamic pressure and the term $\rho g \cdot x_i$ gives the static fluid pressure. After substituting “Equation 2.8” in “Equation 2.7” following form of momentum equation could be obtained:

$$\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial}{\partial x_j} \left\{ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right\} + g_i \beta (T - T_{ref}) \quad (2.9)$$

2.1.3 Balance of Energy Equation

General energy equation is a statement of the energy balance for an infinitesimal control volume and its form appropriate for the present analysis could be given as:

$$\rho C_p \left(\frac{\partial T}{\partial t} + u_j \frac{\partial T}{\partial x_j} \right) = \frac{\partial}{\partial x_i} \left(k \frac{\partial T}{\partial x_i} \right) + q''' + \mu \Phi \quad (2.10)$$

Where q''' is the volumetric rate of internal heat generation, a constant for the present case, Φ is the viscous dissipation function and all other variables are defined in the nomenclature.

2.1.4 Initial and Boundary Conditions

In order to obtain a numerical solution of the differential equations, a set of initial and boundary conditions is needed to describe the behaviour of the various variables at the boundary points. These boundary conditions must describe the variation of all flow variables at all the boundary points. In present method mass conservation equation is embedded within the momentum and energy equations so no separate boundary condition is needed for mass conservation equation.

2.1.4.1 Initial and Boundary Conditions for Momentum Equations

Initial and boundary conditions for all velocity components are needed for the momentum equations for whole the domain under consideration. For present analysis non-slip boundary condition was thought to be more appropriate because no mass flux has been considered across the solid boundaries. These boundary conditions could be summarised as given below:

Initial Condition:

$$\text{For } t = 0: \quad u_i = 0 \quad \text{for the whole domain} \quad (2.11)$$

Boundary condition:

$$\text{For } t > 0: \quad u_i = 0 \quad \text{at all the boundaries}$$

2.1.4.2 Initial and Boundary Conditions for Energy Equation

Energy equation requires initial and boundary conditions to be imposed upon the temperature variation at the boundary points. For the present analysis boundary temperature was kept fixed for whole of the interval, so Dirichlet boundary condition was applied for the temperature at the boundaries as given below:

Initial Condition:

$$\text{For } t \leq 0: \quad T = T_{\text{ref}} \quad \text{for whole domain} \quad (2.12)$$

Boundary condition:

$$\text{For } t > 0: \quad T = T_{\text{ref}} \quad \text{at all the boundaries}$$

2.1.5 Governing Equations in Cartesian and Polar Co-ordinates

Equations for Square Cavity with Internal Heat Source

Square cavity was considered during present numerical analysis to demonstrate the basic methodology adopted. This geometry was chosen due to its benefits of generality and its easiness to implement in any numerical scheme. Also, most of the analysis available in the present day published literature is for Cartesian Co-ordinate system that was necessary to perform a benchmark comparison.

Conservation of Mass Equation

Conservation of mass equation or continuity equation, as it is generally known, could be given for a square cavity by expanding divergence operator from “Equation 2.3” for 2-D Cartesian Co-ordinate system.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (2.13)$$

Momentum Equations

x-momentum equation

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (2.14)$$

y-momentum equation

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - g \beta (T - T_{ref}) \quad (2.15)$$

Energy Balance Equation

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + \frac{q'''}{\rho C_p} \quad (2.16)$$

Equations for Circular Cavity with Internal Heat Source

A 2-D cylindrical (r - ϕ) Co-ordinate system was used for the analysis of circular cavity. It was only possible with this geometry to analyse the non-symmetric behaviour of the flow using 2-D geometry. A schematic view of the coordinate system and vectorial components used for the circular cavity could be shown as in “Figure 2.1”.

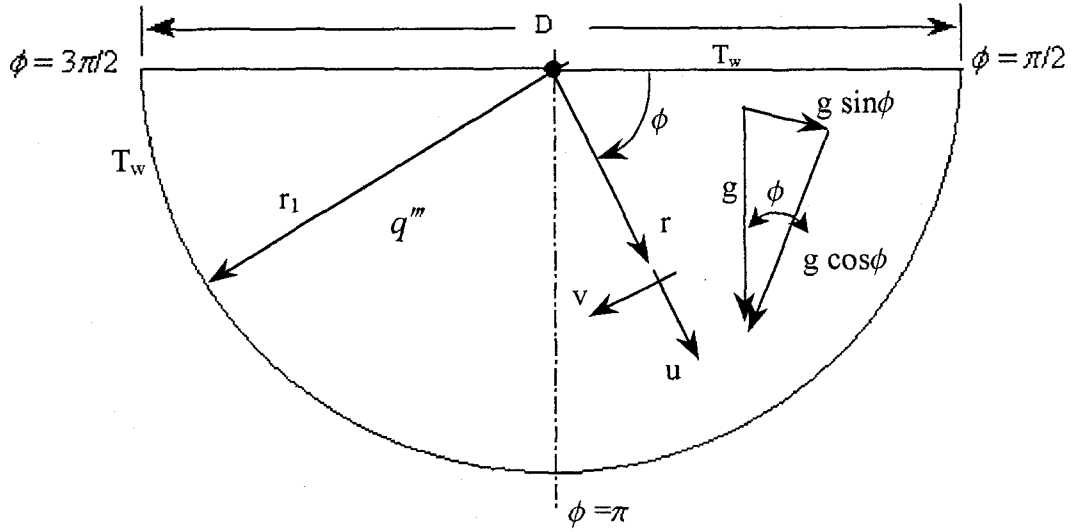


Figure 2.1 Co-ordinate system used for semi-circular cavity

Mass Conservation Equation

$$\frac{\partial u}{\partial r} + \frac{u}{r} + \frac{1}{r} \frac{\partial v}{\partial \phi} = 0 \quad (2.17)$$

The N-S Equations in Cylindrical Coordinates (r, ϕ)

r-momentum equation

$$\begin{aligned} \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial r} + \frac{v}{r} \frac{\partial u}{\partial \phi} - \frac{v^2}{r} = & -\frac{1}{\rho} \frac{\partial p}{\partial r} + \nu \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} - \frac{u}{r^2} + \frac{1}{r^2} \frac{\partial^2 u}{\partial \phi^2} - \frac{2}{r^2} \frac{\partial v}{\partial \phi} \right) \\ & - g [\beta (T - T_{ref})] \sin \phi \end{aligned} \quad (2.18)$$

ϕ -momentum equation

$$\begin{aligned} \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial r} + \frac{v}{r} \frac{\partial v}{\partial \phi} + \frac{uv}{r} = & -\frac{1}{\rho r} \frac{\partial p}{\partial \phi} + \nu \left(\frac{\partial^2 v}{\partial r^2} + \frac{1}{r} \frac{\partial v}{\partial r} - \frac{v}{r^2} + \frac{1}{r^2} \frac{\partial^2 v}{\partial \phi^2} + \frac{2}{r^2} \frac{\partial u}{\partial \phi} \right) \\ & - g [\beta (T - T_{ref})] \cos \phi \end{aligned} \quad (2.19)$$

Energy Balance Equation

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial r} + \frac{v}{r} \frac{\partial T}{\partial \phi} = \alpha \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \phi^2} \right) + \frac{q'''}{\rho C_p} \quad (2.20)$$

Equations for Spherical Segment with Internal Heat Source

A 2-D spherical (r - ϕ) Co-ordinate system was used for the analysis of spherical segment cavity. A schematic view of the Co-ordinate system and vectorial components used for the spherical cavity is shown in “Figure 2.2”.

Mass Conservation Equation

$$\frac{1}{r} \frac{\partial}{\partial r} (r^2 u) + \frac{1}{\sin \phi} \frac{\partial}{\partial \phi} (v \sin \phi) = 0 \quad (2.21)$$

Balance of Momentum Equations

r-momentum equation

$$\begin{aligned} \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial r} + \frac{v}{r} \frac{\partial u}{\partial \phi} - \frac{v^2}{r} = & -\frac{1}{\rho} \frac{\partial p}{\partial r} + \nu \left(\frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial u}{\partial r} \right) + \frac{1}{r^2 \sin \phi} \frac{\partial}{\partial \phi} \left(\sin \phi \frac{\partial u}{\partial \phi} \right) \right. \\ & \left. - \frac{2u}{r^2} - \frac{2}{r^2} \frac{\partial v}{\partial \phi} - \frac{2v \cot \phi}{r^2} \right) - g [\beta (T - T_{ref})] \sin \phi \end{aligned} \quad (2.22)$$

ϕ -momentum equation

$$\begin{aligned} \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial r} + \frac{v}{r} \frac{\partial v}{\partial \phi} + \frac{uv}{r} = & -\frac{1}{\rho} \frac{1}{r} \frac{\partial p}{\partial \phi} + \nu \left(\frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial v}{\partial r} \right) + \frac{1}{r^2 \sin \phi} \frac{\partial}{\partial \phi} \left(\sin \phi \frac{\partial v}{\partial \phi} \right) \right. \\ & \left. + \frac{2}{r^2} \frac{\partial u}{\partial \phi} - \frac{v}{r^2 \sin^2 \phi} \right) - g [\beta (T - T_{ref})] \cos \phi \end{aligned} \quad (2.23)$$

Balance of Energy Equation

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial r} + \frac{v}{r} \frac{\partial T}{\partial \phi} = \alpha \left(\frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin \phi} \frac{\partial}{\partial \phi} \left(\sin \phi \frac{\partial T}{\partial \phi} \right) \right) + \frac{q'''}{\rho C_p} \quad (2.24)$$

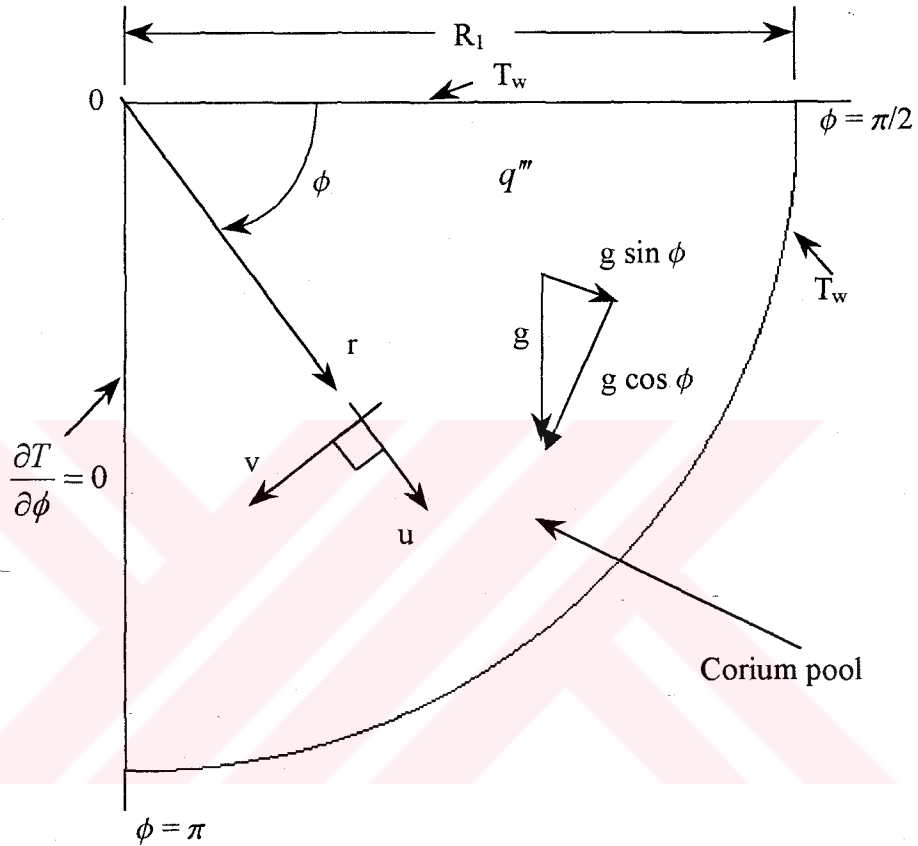


Figure 2.2 Schematic diagram and Co-ordinate system for spherical segment geometry

2.1.6 Conjugate Equations

Numerical investigations are usually carried out for isothermal containing walls for all types of enclosures. By taking isothermal bounding walls the temperature gradient through the walls is neglected by assuming negligible wall thickness and very high wall conductivities. This assumption holds well where wall thickness is small and solid to fluid conductivity ratio is high. But, reactor pressure vessel have exceptionally thick stainless steel wall while wall conductivities are not much higher than the enclosed molten core material. This scenario necessitated the analysis of interaction of convection flow within the pool body with that of conduction through

the RPV wall, giving rise to a situation that is technically known as conjugate problem. Conjugate situation was handled, for the present analysis, by insertion of the ghost nodes at the solid-fluid interface. These nodes are called ghost nodes because they do not arise during normal grid layout but are introduced artificially to handle the solid to fluid interface boundary conditions. Details of the method are given by **WanLai et al. (1997)** and investigation is given for natural convection by **Liaqat and Baytaş (2000a)**. Location of these ghost nodes is shown schematically in figure given below.

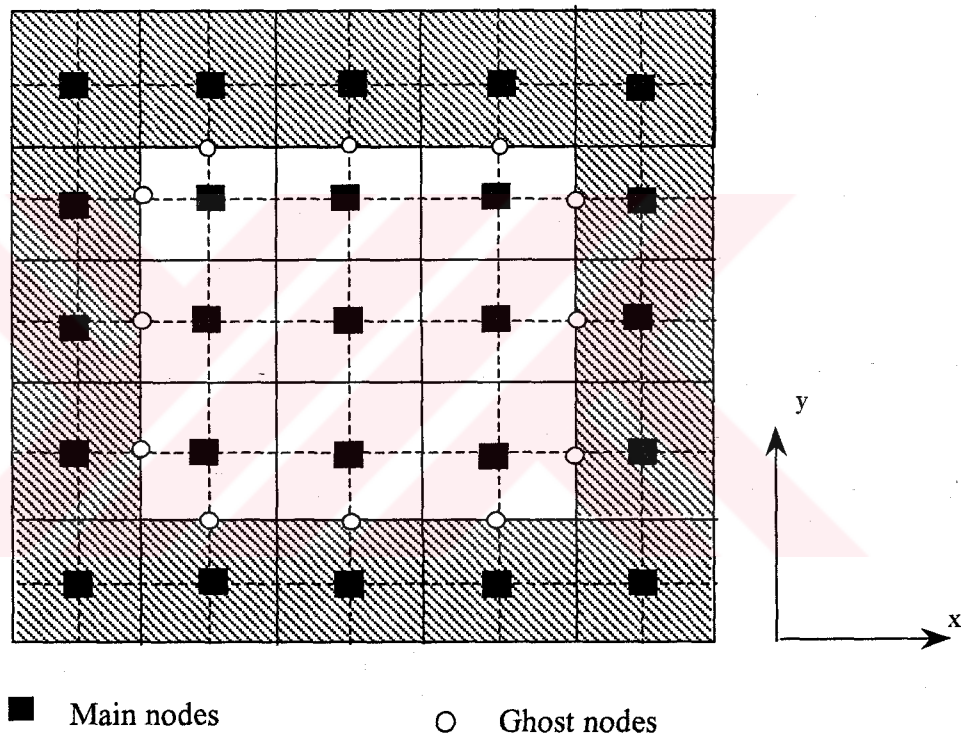


Figure 2.3 Schematic layout of main and ghost nodes

Effectiveness of the ghost nodes is realised when whole the domain is solved as one region for x or y sweeps. This method decreases the effort to analyse the both regions separately. General governing equations are valid for the fluid domain while conduction equation is solved for the solid region given below:

$$\frac{\partial T}{\partial t} = \alpha_s \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (2.25)$$

At solid-liquid interface the temperature and heat flux must be continuous. First condition is fulfilled by taking a common node point at the solid-liquid interface. The latter condition could be expressed as:

$$k_f \left(\frac{\partial T}{\partial n} \right)_{fluid} = k_s \left(\frac{\partial T}{\partial n} \right)_{wall} \quad (2.26)$$

and $T_{fluid} = T_{wall}$ at solid-fluid interface

Where, n stands for the direction normal to the wall surface. By calculating the ghost node temperatures by “Equation 2.26” in numerical scheme, continuity of flux at the interface is ensured. As is clear from the above equation, ghost node temperature (solid-liquid interface temperature) is a function of conductivities of fluid and wall material and specific grid layout adopted for the analysis.

2.1.7 Radiative Heat Transfer Equations

After the slumping of the molten core materials in the lower plenum of the reactor pressure vessel, molten mass will be super heated by the heat generated through the decay of fission products. Due to high melting point of all oxide constituents of the corium, temperature of super heated fluid will exceed 3500 °K. At such a high temperature radiative heat transfer from the pool surface could not be ignored. Thus, calculations including radiative heat transfer may put more light upon the actual heat transfer and fluid flow phenomenon in the corium pool. So, it was decided to include radiation heat transfer from the pool surface in present investigation. In order to calculate the radiative heat transfer rate from the pool surface enclosure formulation of radiative heat transfer (**Modest, 1993**) was utilised. The basic derivation of the enclosure theory could be summarised as given below.

Net radiative flux leaving a black enclosure could be described as:

$$q_r = q''_{out} - q''_{in} \quad (2.27)$$

or

$$q_r = (q''_{emission} + q''_{reflection}) - q''_{irradiation} = (E_r + \rho_r H_r) - H_r = E_r + (1 - \rho_r) H_r \quad (2.28)$$

This equation for black body could be modified for gray diffuse surface as given below

$$q_r(r) = \epsilon(r) E_b(r) - \alpha_r(r) H_r(r) \quad (2.29)$$

“Equation 2.29” could be modified if it is considered that $\alpha_r(r) = (1 - \rho_r(r))$

$$q_r(r) = \epsilon(r) E_b(r) - (1 - \rho_r(r)) H_r(r) \quad (2.30)$$

or

$$q_r(r) = \epsilon(r) E_b(r) - H_r(r) + \rho_r(r) H_r(r) \quad (2.31)$$

Which could be modified by utilising total flux leaving a gray diffuse surface ($J(r)$) as total flux is the sum of reflected and emitted parts.

$$q_r(r) = J(r) - H_r(r) \quad (2.32)$$

But the heat transfer rate leaving a segmental area dA' and intercepted by another segmental area dA is given by equation:

$$J(r') dA' dF_{dA' \rightarrow dA} \quad (2.33)$$

By integrating “Equation 2.33” for total area of the surface we can get total irradiation as given below:

$$H_r(r) = \int_A J(r') dF_{dA \rightarrow dA'} \quad (2.34)$$

After substituting “Equation 2.34” into “Equation 2.32” following equation could be obtained:

$$q_r(r) = \epsilon(r) E_b(r) - \alpha_r(r) \left[\int_A J(r') dF_{dA-dA'} \right] \quad (2.35)$$

Combining “Equations 2.31 and 2.32”, which could be written:

$$J(r) = \epsilon(r) E_b(r) + \rho_r(r) H_r(r) \quad (2.36)$$

Substituting equation “Equation 2.34” in “Equation 2.36” result is:

$$J(r) = \epsilon(r) E_b(r) + \rho_r(r) \left[\int_A J(r') dF_{dA-dA'} \right] \quad (2.37)$$

Multiplying “Equation 2.34” with $\alpha_r(r)$ and then subtracting from “Equation 2.29” and after minor modifications following equation is obtained.

$$J(r) = E_b(r) - \frac{1 - \epsilon(r)}{\epsilon(r)} q_r(r) \quad (2.38)$$

By substituting $\epsilon(r) = \alpha_r(r)$ (from definition of gray diffuse surface), “Equation 2.35” could be modified as:

$$\frac{q_r(r)}{\epsilon(r)} = E_b(r) - \left[\int_A J(r') dF_{dA-dA'} \right] \quad (2.39)$$

Combining “Equation 2.38” and “Equation 2.39” result is as below:

$$\frac{q_r(r)}{\epsilon(r)} - \left[\int_A \left(\frac{1}{\epsilon(r')} - 1 \right) q_r(r') dF_{dA-dA'} \right] = E_b(r) - \int_A E_b(r') dF_{dA-dA'} \quad (2.40)$$

Breaking the surfaces A_r and A_r' into NR sub-surfaces, then “Equation 2.40” could be written as:

$$\frac{q_{rk}}{\epsilon_k} - \sum_{j=1}^{NR} \left(\frac{1 - \epsilon_j}{\epsilon_j} \right) F_{k-j} q_{rj} = \sum_{j=1}^{NR} (\sigma T_k^4 - \sigma T_j^4) \quad \text{for } k = 1, NR \quad (2.41)$$

The view factors for each zone were calculated by utilising Hottel's crossed-string method (**Hottel and Sarofin, 1967**). In this method view factor between two surfaces is obtained by using the lengths of both diagonals, sides and sizes of the surfaces.

2.2 Scale Analysis

The object of scale analysis is to use the basic heat transfer principles to obtain the order-of-magnitude estimates for the variables of interest present in the general governing equations. These estimates are then used to derive the reference magnitudes for developing the dimensionless variables. Non-dimensional governing equations are obtained by using dimensionless variables which are solved numerically. The significance of dimensional analysis could be judged from the fact that final results could be absolutely useless if dimensionless variables are not appropriate.

2.2.1 Low Rayleigh Number Scaling

Scaling laws defining low Rayleigh number natural convection flow within a cavity or pool containing heat generating fluid were available in open literature as given by **May (1991)**. Definitions for different dimensionless variables for this scaling are given in "Table 2.1".

2.2.2 High Rayleigh Number Scale Analysis

Scale analysis available in open literature was not applicable for high Rayleigh number flow dealt with in present project. This necessitated to develop new scaling to deal with fluctuating flow encountered in high Rayleigh number flows. A scaling for this purpose was proposed for high Ra number and essential results were published in **Liaqat and Baytaş (2000)**.

The governing equations of mass, momentum and energy for the square cavity, in Cartesian Co-ordinates, were given in section 2.1.

Immediately after $t = 0$, all fluid will be motionless, so "Equation 2.16" can be written as:

Table 2.1

Definitions of dimensionless variables

Variables	Reference magnitude for low Ra number	Reference magnitude for High Ra number	Low Rayleigh number scaling	High Rayleigh number scaling
U,V	$\frac{\alpha_f}{H}$	$\frac{\alpha_f}{H}(RaPr)^{2/5}$	$\frac{u,v}{(\alpha_f/H)}$	$\frac{u,v}{\frac{\alpha_f}{H}(RaPr)^{2/5}}$
X, Y	H	H	$\frac{x,y}{H}$	$\frac{x,y}{H}$
τ	$\frac{H^2}{\alpha_f}$	$\frac{H^2}{\alpha_f(RaPr)^{2/5}}$	$\frac{t}{(H^2/\alpha_f)}$	$\frac{t\alpha_f(RaPr)^{2/5}}{H^2}$
T	ΔT	ΔT	$\frac{T-T_w}{\Delta T}$	$\frac{T-T_w}{\Delta T}$
P	$\frac{\rho\alpha_f^2}{H^2}$	$\frac{\rho\alpha_f^2}{H^2}(RaPr)^{4/5}$	$\frac{p}{(\rho\alpha_f^2/H^2)}$	$\frac{pH^2}{\rho\alpha_f^2(RaPr)^{4/5}}$
ΔT	$\frac{q''H^3}{8k_f}$	$\frac{q''H^2}{k(RaPr)^{1/5}}$	-----	-----

$$\frac{\Delta T}{t} \approx \alpha_f \frac{\Delta T}{\delta^2} \rightarrow \delta \approx (\alpha_f t)^2 \quad (2.42)$$

Then cooled fluid layer along the wall will tend to descend. Velocity scale of this downward motion can be derived by eliminating the pressure term from momentum equations ie by differentiating “Equation 2.14” with respect to y and “Equation 2.15” with respect to x. The main terms are summarised as below:

Inertia	Viscous/Friction	Buoyancy
$\frac{\partial^2 v}{\partial x \partial t}$	$\nu \frac{\partial^3 v}{\partial x^3}$	$g\beta \frac{\partial T}{\partial x}$

Taking x-scale as boundary layer thickness (δ_b) along the side walls from the above terms:

$\frac{v}{\delta_b t}$	$\nu \frac{v}{\delta_b^3}$	$g\beta \frac{\Delta T}{\delta_b}$
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Using “Equation 2.42” above terms yield:

$\frac{1.0}{Pr}$	1.0	$\frac{g\beta \Delta T \delta_b^2}{\nu v}$
------------------	-----	--

By the balance of inertia and buoyancy:

$$\frac{1.0}{Pr} \approx \frac{g\beta \Delta T \delta_b^2}{\nu v} \rightarrow v \approx \frac{g\beta \Delta T \delta_b^2}{\nu} Pr \quad (2.43)$$

The reference temperature scale (ΔT) can be defined by Fourier’s conduction law as:

$$q'' = k \frac{\partial T}{\partial x} \rightarrow q'''(H) \approx k \frac{\Delta T}{\delta_b}$$

Thus:

$$\Delta T \approx \frac{q''' H \delta_b}{k} \quad (2.44)$$

If we take the balance of heat provided to the wall by convection and heat conduction through the wall for steady state, we get from energy “Equation 2.16”:

$$v \frac{\Delta T}{H} \approx \alpha_f \frac{\Delta T}{\delta_b^2} \quad \rightarrow \quad v \approx \alpha_f \frac{H}{\delta_b^2} \quad (2.45)$$

Substituting “Equations 2.44 and 2.45” in “Equation 2.43” yields scaling for boundary layer thickness as:

$$\delta_b \approx \frac{H}{(RaPr)^{1/5}} \quad (2.46)$$

Where Ra number and Pr number are defined in the nomenclature.

“Equations 2.45 and 2.46” give the scaling for the ΔT as:

$$\Delta T \approx \frac{q'' H^2}{k(RaPr)^{1/5}} \quad (2.47)$$

From “Equations 2.43, 2.46 and 2.47” scaling for velocity can be obtained as:

$$v \approx \frac{\alpha_f}{H} (RaPr)^{2/5} \quad (2.48)$$

Scaling for time (t) can be obtained from “Equations 2.42 and 2.46”:

$$t \approx \frac{H^2}{\alpha_f (RaPr)^{2/5}} \quad (2.49)$$

Pressure scaling can be obtained using dynamic pressure equation as given below:

$$p \approx \rho v^2$$

Using “Equation 2.48” in the above equation scaling for pressure can be obtained as:

$$p \approx \rho \frac{\alpha_f^2}{H^2} (RaPr)^{4/5} \quad (2.50)$$

2.3 Dimensionless Governing Equations for the Present Analysis

Present problem in complete form consists of four main parts.

- (a) Natural convection inside the corium pool initiated by internal heating.
- (b) Conjugate analysis of heat transfer from pool body to lower plenum wall.
- (c) Radiation from the corium pool surface.
- (d) Interaction of radiation and conduction for upper walls of the reactor pressure vessel.

Due to peculiar layout of the spherical geometry, only half of the hemi-spherical pool was analysed. But in this way the inherent non-symmetric character of the actual flow has to be compromised. This non-symmetric flow could only be analysed in 2-D cylindrical geometry. Thus corium pool has been assumed to be a long semi-circular geometry having thick bottom containing wall thus avoiding the three dimensional problem encountered if spherical geometry is used. This was necessary to keep the analysis manageable using Pentium class personal computer system. Thus upper part of the pressure vessel wall has to be modelled as a long Cartesian geometry covering the pool top. These assumptions made it possible to treat the present problem in two dimensions only.

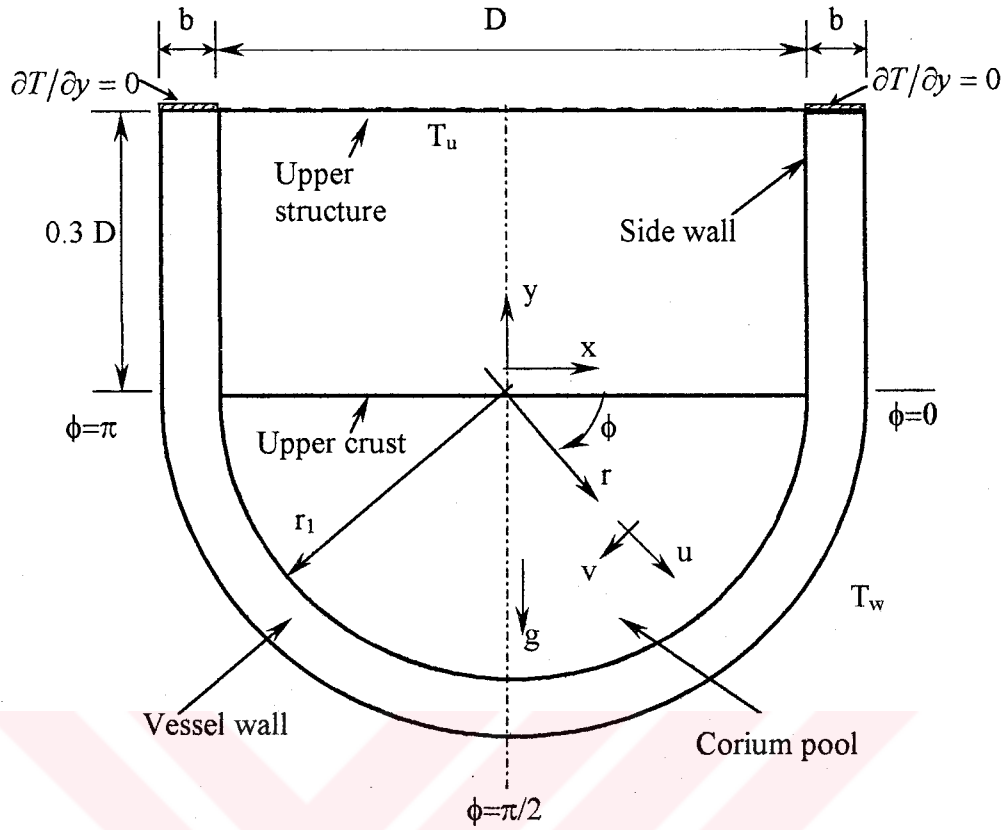


Figure 2.4 Schematic diagram of the corium inside RPV lower head and Co-ordinate system

2.3.1 Dimensionless Equation for Upper Cartesian Part of the Reactor Pressure Vessel Wall

As upper walls of the pressure vessel were modelled as Cartesian geometry, dimensionless conduction equation could be developed using non-dimensional variables given in "Table 2.1". The two-dimensional temperature distribution in the walls is governed by the heat conduction equation, given in the dimensionless form below:

$$\frac{\partial \theta}{\partial \tau} = \frac{\alpha_s}{\alpha_f} \frac{1}{(Ra Pr)^{2/5}} \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \quad (2.51)$$

2.3.2 Dimensionless Equations for Cylindrical Part of the Corium Pool

Dimensionless equations for the cylindrical pool could be given as below:

Mass conservation equation

$$\frac{\partial U}{\partial R} + \frac{U}{R} + \frac{1}{R} \frac{\partial V}{\partial \phi} = 0 \quad (2.52)$$

Momentum equations

$$\frac{\partial U}{\partial \tau} + U \frac{\partial U}{\partial R} + \frac{V}{R} \frac{\partial U}{\partial \phi} - \frac{V^2}{R} = -\frac{\partial P}{\partial R} + C \left(\nabla^2 U - \frac{U}{R^2} - \frac{2}{R^2} \frac{\partial V}{\partial \phi} \right) - \theta_f \sin \phi \quad (2.53)$$

$$\frac{\partial V}{\partial \tau} + U \frac{\partial V}{\partial R} + \frac{V}{R} \frac{\partial V}{\partial \phi} + \frac{UV}{R} = -\frac{1}{R} \frac{\partial P}{\partial \phi} + C \left(\nabla^2 V - \frac{V}{R^2} + \frac{2}{R^2} \frac{\partial U}{\partial \phi} \right) - \theta_f \cos \phi \quad (2.54)$$

Energy equation for fluid domain

$$\frac{\partial \theta_f}{\partial \tau} + U \frac{\partial \theta_f}{\partial R} + \frac{V}{R} \frac{\partial \theta_f}{\partial \phi} = \frac{1}{(RaPr)^{2/5}} \left(\nabla^2 \theta_f \right) + (RaPr)^{-1/5} \quad (2.55)$$

Energy equation for solid wall

$$\frac{\partial \theta_s}{\partial \tau} = \frac{\alpha_s}{\alpha_f} \frac{1}{(RaPr)^{2/5}} \left(\frac{\partial^2 \theta_s}{\partial R^2} + \frac{1}{R} \frac{\partial \theta_s}{\partial R} + \frac{1}{R^2} \frac{\partial^2 \theta_s}{\partial \phi^2} \right) \quad (2.56)$$

Where $C = \frac{Pr}{(RaPr)^{2/5}}$

and

$$\nabla^2 = \frac{\partial^2}{\partial R^2} + \frac{1}{R} \frac{\partial}{\partial R} + \frac{1}{R^2} \frac{\partial^2}{\partial \phi^2}$$

Where all dimensionless variables has been defined in “Table 2.1”.

Radiation Heat Transfer Equation (RTE)

Non-dimensional equivalent of “Equation 2.41”, after some re-arrangements, can be written as given by equation below (Dehghan and Behnia, 1996):

$$\sum_{j=1}^{NR} \left(\frac{\delta_{kj}}{\epsilon_j} - F_{k-j} \frac{1-\epsilon_j}{\epsilon_j} \right) Q_j = \sum_{j=1}^{NR} (\delta_{kj} - F_{k-j}) \sigma \left(\frac{T_j}{T_w} \right)^4 \quad \text{for } k = 1, NR \quad (2.57)$$

Where $Q_j = \frac{q_{rj}}{\sigma T_w^4}$ is the dimensionless net radiative heat flux from jth surface (total NR sub-divisions shown in “Figure 2.5”) and δ_{kj} is the Kronecker delta.

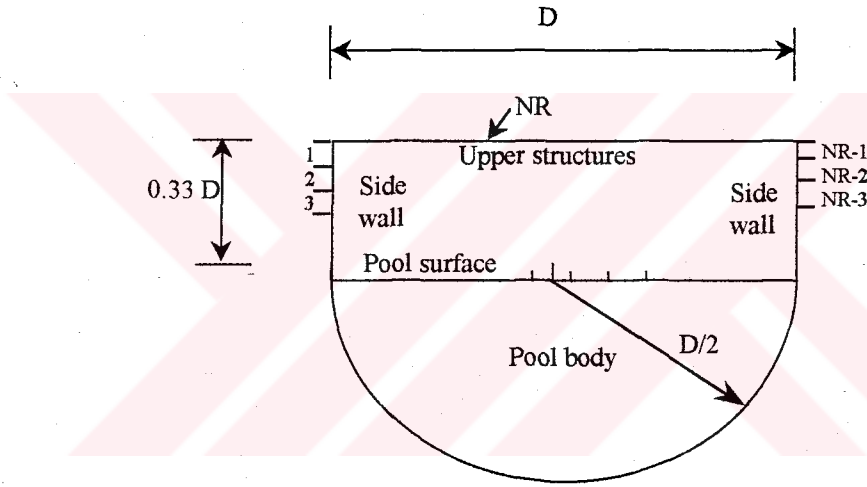


Figure 2.5 Schematic diagram showing the grid layout for radiative heat transfer

Initial and Boundary Conditions

“Equations 2.52-2.55” are subjected to following boundary conditions

For $\tau \leq 0$: $U = V = \theta_f = \theta_s = 0$ for whole domain

For $\tau > 0$: $U = V = 0$ for $R = R_1$ and $\phi = \pi/2, 3\pi/2$ and in solid domain

$$\theta = 0 \quad \text{for } R = R_1 + \frac{b}{D} \quad \text{and } \phi = \pi/2 \text{ to } 3\pi/2$$

$$\theta_f = \theta_s, \quad \frac{\partial \theta_f}{\partial R} = K \frac{\partial \theta_s}{\partial R} \quad \text{at } R = R_1 \quad (2.58)$$

$$\theta = 0 \quad \text{for } X = 0 \quad \text{and} \quad X = 2b + D \quad \text{for upper wall}$$

$$\frac{1}{R} \frac{\partial \theta_s}{\partial \phi} = 0 \quad \text{at } R = R_1 \text{ to } R_1 + \frac{b}{D} \quad \text{for } \phi = \pi/2, 3\pi/2$$

At the solid-fluid interface the temperature and the heat flux must be continuous. The latter condition could be expressed as:

$$\left(\frac{\partial \theta}{\partial R} \right)_{\text{fluid}} = \frac{k_s}{k_f} \left(\frac{\partial \theta}{\partial R} \right)_{\text{wall}} \quad (2.59)$$

All the symbols appearing in the above equations have been defined in the nomenclature. Net radiative heat flux from each sub-division, for radiative exchange, could be balanced against conduction through the boundary condition equations given below:

$$\left(\frac{\partial \theta}{\partial X} \right)_j = N_{RCW} Q_j \quad \text{For the side walls} \quad (2.60)$$

$$\left(\frac{1}{R} \frac{\partial \theta}{\partial \phi} \right)_j = N_{RCP} Q_j \quad \text{For top surface of the pool} \quad (2.61)$$

Where:

$$N_{RCW} = \frac{D \sigma T_w^4}{k_s \Delta T} \quad (2.62)$$

$$N_{RCP} = \frac{D \sigma T_w^4}{k_p \Delta T} \quad (2.63)$$

are the radiation conduction interaction parameters for the wall and the pool surface, respectively.

The view factors for each zone were calculated by utilising Hottel's crossed-string method (Hottel and Sarofin, 1967). In this method view factor between two surfaces is obtained by using the lengths of both diagonals, sides and sizes of the surfaces.

Heat Transfer

In order to evaluate the local and average heat transfer rates, the local Nusselt numbers have been defined as:

$$Nu_{top} = \frac{-(\partial T_f / r \partial \phi)}{\Delta T / D} = -\frac{1}{R} \frac{\partial \theta_f}{\partial \phi} \quad \text{for } R = 0 \text{ to } R_1 \text{ at upper wall} \quad (2.64)$$

$$Nu_{bottom} = \frac{-(\partial T_f / \partial r)}{\Delta T / D} = -\left(\frac{\partial \theta_f}{\partial R} \right) \quad \text{at } R = R_1 \text{ for bottom solid-fluid interface} \quad (2.65)$$

The average Nusselt number for the top and bottom walls is thus obtained by integrating "Equations 2.64 and 2.65", respectively, as given below:

$$\overline{Nu}_{top} = \frac{1}{R_1} \int_0^{R_1} Nu_{top} dR \quad (2.66)$$

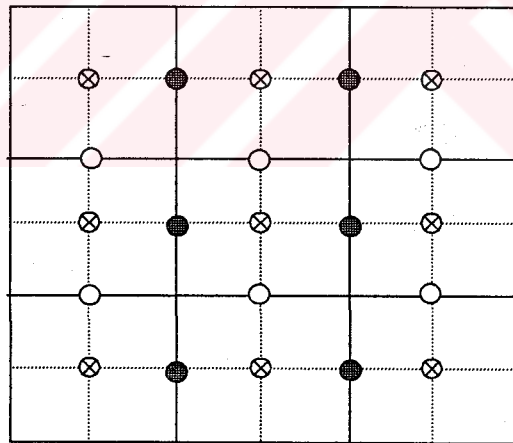
$$\overline{Nu}_{bottom} = \frac{1}{\pi} \int_{\pi/2}^{3\pi/2} Nu_{bottom} d\phi \quad (2.66)$$

3. COMPUTATIONAL METHOD

It is clear from the above given governing equations that numerical solution to the problem is the only viable choice due to complexity and non-linearity of these equations.

3.1 Grid Layout

Grid layout is a crucial part of any numerical method. It dictates the interaction between various parts of the domain under consideration. Grid description depends upon the numerical methodology adopted for the solution. Present analysis is based upon SIMPLER algorithm with a staggered grid arrangement shown schematically in “Figure 3.1”. Control volume method (**Patankar, 1980**) was adopted for the present analysis.



- Node for velocity component in x direction
- Node for velocity component in y direction
- ⊗ Node for scalar variables (pressure, temperature)

Figure 3.1 Schematic diagram of staggered grid

In staggered grid approach, variables are distinguished between vectors and scalars. Both type of variables occupy different nodes in the grid layout. Cell centred grid layout was used for scalar variables such as pressure and temperature. However, cell vertex arrangement was utilised for vectors ie for velocity components. For staggered grid arrangement grid layout for both velocity components was displaced through half of the control volume in their respective directions. Thus each velocity node occupies a position at respective face of the cell centred scalar control volume as shown “Figure 3.2” schematically.

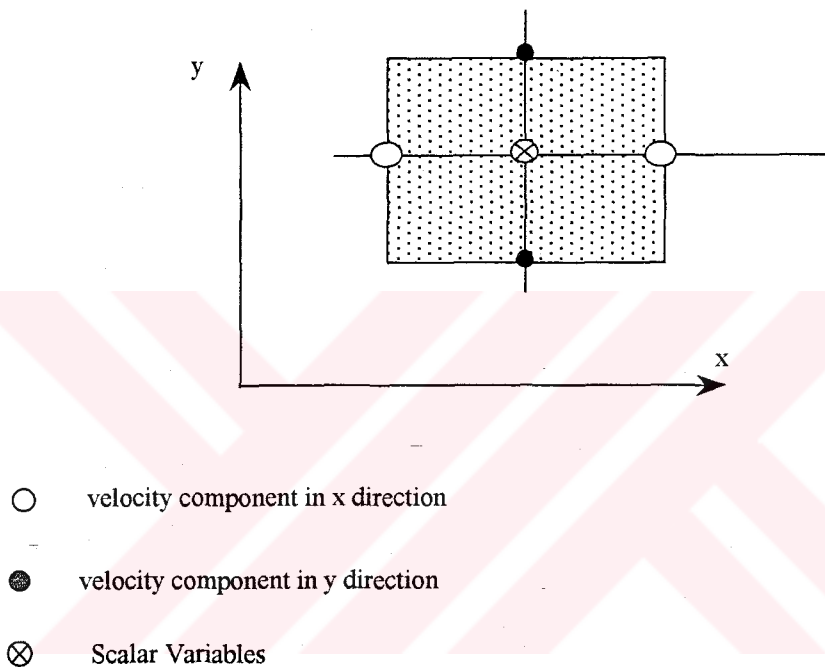


Figure 3.2 Relative positions of the scalar and vector variables in cell centred staggered grid

3.2 Discretization for Control Volume Method

Algebraic equations, which give the unknown variables at grid points described in the grid layout, are called discretization equations. These equations are obtained from differential equations governing the given situation and the process of their derivation is called discretization. Thus through this discretization process governing differential equations are replaced with simple algebraic equations. The discretization method preferred for the present investigation is called Finite Volume Method (FVM). After the introduction to governing equations, scale analysis and grid layout, second step is to discretize these equations using the control volume approach. General governing equations discussed above will be reduced to their dimensionless counterparts for respective Co-ordinate system. Those non-dimensional equations will be discretized using control volume approach. Each control volume and its adjacent control nodes will be labelled as east, west, north and south as shown in “Figure 3.3” for Cartesian and in “Figure 3.4” for Cylindrical and spherical Co-ordinate systems.

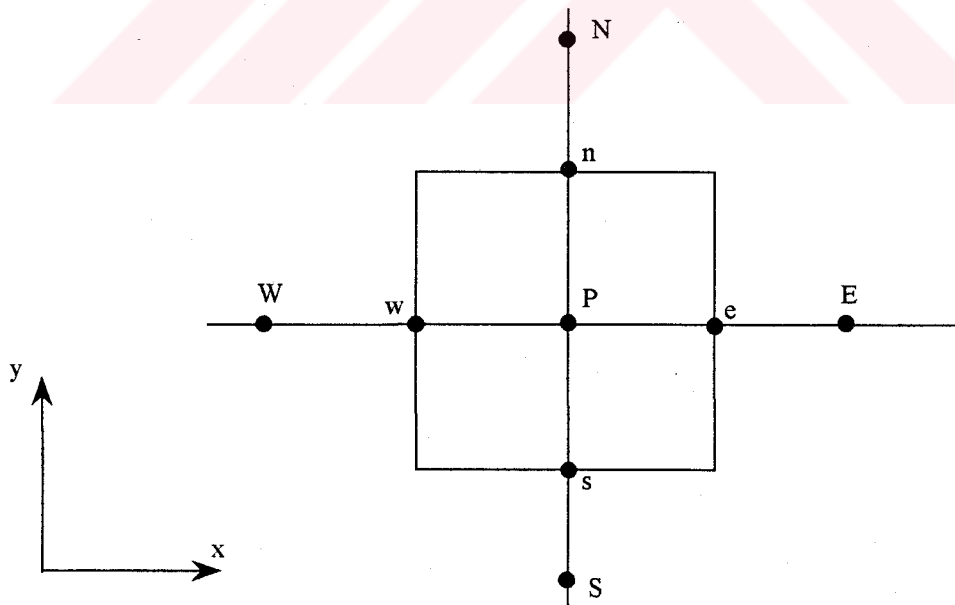


Figure 3.3 Control volume nodes nomenclature for Cartesian Co-ordinate system

Control volume cell faces are denoted by small letters as e, w, n and s while their respective positions are shown in “Figure 3.3”. Adjacent control volume cell centroids will be designated by capital letters as E, W, N and S.

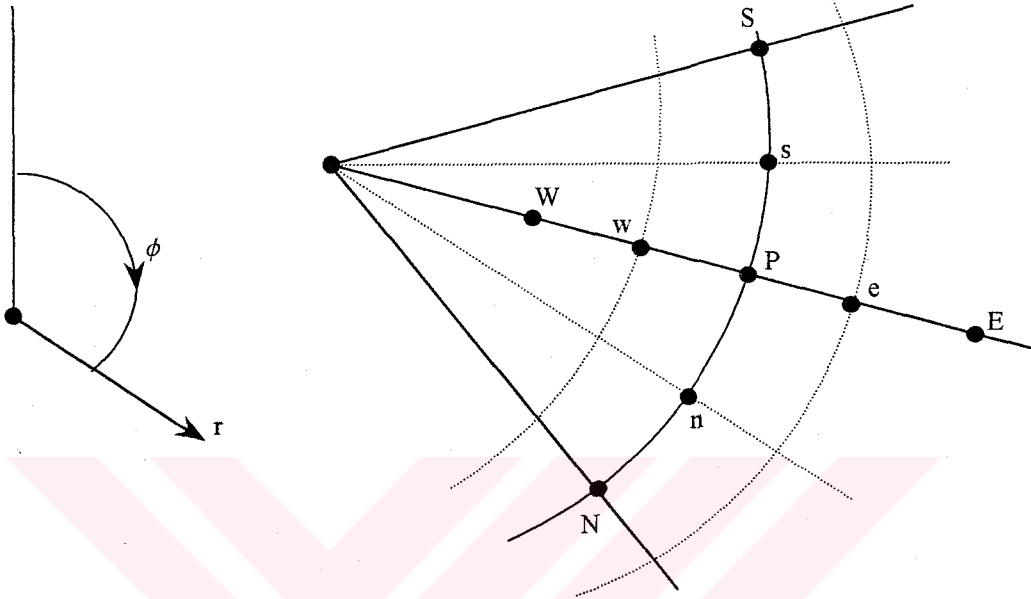


Figure 3.4 control volume nodes nomenclature for cylindrical and spherical Co-ordinate systems

All discretization process will be demonstrated with a general momentum conservation equation in Cartesian Co-ordinate system. “Equation 2.14” could be written in compact form as:

$$\frac{\partial v}{\partial t} + \text{div}(uv - \Gamma \nabla v) = -\nabla p + S \quad (3.1)$$

By integrating “Equation 3.1” over control volume area ($\Delta x \times \Delta y$) and time step interval (Δt).

$$\iiint_{\Delta V} \int_{\Delta t} \left(\frac{\partial v}{\partial t} + \text{div}\{uv - \Gamma \nabla v\} \right) dt dy dx = \iiint_{\Delta V} \int_{\Delta t} (-\nabla p + S) dt dV \quad (3.2)$$

The left hand side of “Equation 3.2” can be simplified by utilising Gauss’s theorem to convert volume integral into surface integral. Gauss’s theorem can be given as:

$$\iiint (\nabla \cdot F_i) dV = \iint (F_i \cdot \hat{n}) ds \quad (3.3)$$

By utilising “Equation 3.3”, left hand side of “Equation 3.2” could be modified as given below:

$$= (v_p - v_p^0) \frac{\Delta V}{\Delta t} + \iint (uv - \Gamma \nabla v) \cdot \hat{n} ds \quad (3.4)$$

By expansion of the dot product, in second term of “Equation 3.4”, following form can be written.

$$\begin{aligned} &= (v_p - v_p^0) \frac{\Delta V}{\Delta t} + \iint \left(uv - \Gamma \frac{\partial v}{\partial x} \right)_e dA - \iint \left(uv - \Gamma \frac{\partial v}{\partial x} \right)_w dA \\ &+ \iint \left(uv - \Gamma \frac{\partial v}{\partial y} \right)_n dA - \iint \left(uv - \Gamma \frac{\partial v}{\partial y} \right)_s dA \end{aligned} \quad (3.5)$$

After integrating above equation and keeping the sign convention in mind, following equation can be obtained for 2-D analysis being considered here.

$$\begin{aligned} &(v_p - v_p^0) \frac{\Delta V}{\Delta t} + \left[(uv)_e - \frac{\Gamma(v_e - v_p)}{\delta x} \right] A_e - \left[(uv)_w - \frac{\Gamma(v_p - v_w)}{\delta x} \right] A_w \\ &+ \left[(vv)_n - \frac{\Gamma(v_n - v_p)}{\delta y} \right] A_n - \left[(vv)_s - \frac{\Gamma(v_p - v_s)}{\delta y} \right] A_s \\ &= \iiint \left(-\frac{\partial p}{\partial x} + S \right) dv \end{aligned} \quad (3.6)$$

Where A indicates control volume face areas located at the position indicated by the subscripts. In writing “Equation 3.6” a linear piecewise profile of v between the point, P, and its appropriate neighbour has been assumed. This point will be treated separately later on. Let diffusion flux and convection flux be defined by the following equations:

$$D_f = \frac{\Gamma}{\Delta x}$$

and

$$C = u, v$$

by using these definitions of diffusion and convective fluxes “Equation 3.6” can be modified as.

$$\begin{aligned} & (v_p - v_p^0) \frac{\Delta V}{\Delta t} + [C_e v_e - D_{fe}(v_e - v_p)]A_e - [C_w v_w - D_{fw}(v_p - v_w)]A_w \\ & + [C_n v_n - D_{fn}(v_n - v_p)]A_n - [C_s v_s - D_{fs}(v_p - v_s)]A_s \\ & = \iiint \left(-\frac{\partial p}{\partial x} + S \right) dv \end{aligned} \quad (3.7)$$

In order to calculate the dependent variables at the control volume faces, some assumption for the variation of these dependent variables between the nodes is required. By using any assumption for this variation, the resultant expression will contain the values of the variable for the control volume under consideration and other for the adjacent control volume with which it shares the face. The assumption used for the calculation of these face values from the given values of neighbouring control volumes is conventionally called the differencing scheme. Hence a differencing scheme is substituted in “Equation 3.7”, the resultant expression (“Equation 3.8”) gives the variable values for the control volume under consideration, P, in terms of the variable values at the neighbouring control volumes. A differencing scheme thus provides the link between “Equations 3.7 and 3.8” and is written as an expression for a_E, a_W etc in terms of diffusion and convection fluxes.

$$a_P v_P + a_E v_E + a_W v_W + a_N v_N + a_S v_S = b_p \quad (3.8)$$

Where:

$$b_p = (p_w - p_e)A_p + v_p^0 \frac{\Delta V}{\Delta t} + S \Delta V \quad (3.9)$$

and

$$a_p = v_p \frac{\Delta V}{\Delta t} + a_E + a_W + a_N + a_S \quad (3.10)$$

“Equation 3.8”, when written for each grid point, gives a system of equations that can be solved simultaneously to obtain values of the field variable at each grid point.

3.2.1 Differencing Scheme

Differencing scheme utilised throughout present calculations is termed as Power Law differencing scheme. The main criterion for the selection of a differencing scheme is relative strength of the convective and diffusion fluxes. This relative importance of the two fluxes is measured in terms of Peclet number given by equation below.

$$Pe = \frac{u \Delta x}{\Gamma} \quad (3.11)$$

Because the main focus of the present analysis is the high Rayleigh number natural convection flow, at this range of Rayleigh number convective flux is usually dominant over the diffusion flux that is why Power Law differencing scheme was chosen for present calculations (**Patankar, 1980**). Power Law Scheme is more accurate approximation to the one-dimensional exact solution and produces better results as compared to Hybrid or other schemes. In this scheme diffusion is set to zero when cell Pe exceeds 10. If $0 < Pe < 10$ the flux is evaluated by using a polynomial expression. Power Law differencing scheme is more accurate for one dimensional problems since it attempts to represent the exact solution more precisely. The expressions for a_w , a_E , a_N and a_S calculated by Power law scheme can be written as given by **Versteeg and Malalsekera (1996)**.

$$a_W = D_{fw} \max \left\{ 0, \left(1 - 0.1 |Pe_w| \right)^5 \right\} + \max [F_w, 0] \quad (3.12)$$

$$a_E = D_{fe} \max \left\{ 0, \left(1 - 0.1 |Pe_e| \right)^5 \right\} + \max [-F_e, 0] \quad (3.13)$$

$$a_N = D_{fn} \max \left\{ 0, \left(1 - 0.1 |Pe_n| \right)^5 \right\} + \max [-F_n, 0] \quad (3.14)$$

$$a_S = D_{fs} \max \left\{ 0, \left(1 - 0.1 |Pe_s| \right)^5 \right\} + \max [F_s, 0] \quad (3.15)$$

3.2.2 Source Term Linearization

The attempt is to develop linear simultaneous equations for the present analysis the source term in “Equation 3.9” must be linearized even if it is non-linear. Source could be linearized as given by the following equation:

$$S = S_p + S_u \quad (3.16)$$

Where S_u represent the constant part of the source term while S_p gives the linear term. When “Equation 3.16” is substituted in “Equation 3.8” following general linear algebraic equation results.

$$a_p v_p = \sum a_{nb} v_{nb} + b1 \quad (3.17)$$

Where nb is used to designate the neighbouring grid nodes, and a_p is given by the following equation:

$$a_p = \sum a_{nb} + \frac{\Delta V}{\Delta t} - S_p \quad (3.18)$$

and

$$b1 = (p_w - p_e) A_p + \frac{\Delta V}{\Delta t} v_p^0 + S_u \quad (3.19)$$

3.2.3 The Pressure Equation

One of the source terms (“Equation 3.19”) is the very important pressure term. There is no pressure equation included in momentum conservation equations so this term could not be calculated before hand. During present analysis pressure equation

described by **Patankar (1980)** has been used. By the method of Patankar pressure equation is obtained by utilizing the continuity equation and momentum equations. The pressure equation can be derived as given below.

General momentum “Equation 3.18” can be modified by taking out pressure term from b1 as:

$$v_n = \frac{\sum a_{nb} v_{nb} + b2}{a_n} + d_n(p_P - p_N) \quad (3.20)$$

Where:

$$d_n = \frac{A_n}{a_n} \quad (3.21)$$

$$b2 = \frac{\Delta V}{\Delta t} v_p^0 + S_u \quad (3.22)$$

“Equation 3.20” can be modified by defining pseudovelocity as given by equation below:

$$\hat{v}_n = \frac{\sum a_{nb} + v_{nb} + b2}{a_n} \quad (3.23)$$

Substituting “Equation 3.21” in “Equation 3.20” following equation is obtained:

$$v_n = \hat{v}_n + d_n(p_P - p_N) \quad (3.24)$$

Similar equation can be written for other velocity component (u) as given below:

$$u_e = \hat{u}_e + d_e(p_P - p_E) \quad (3.25)$$

Now discretised continuity (discretization grid in “Figure 3.5”) equation can be written from “Equation 2.12” as:

$$(u_e - u_w)\Delta y + (v_n - v_s)\Delta x = 0 \quad (3.26)$$

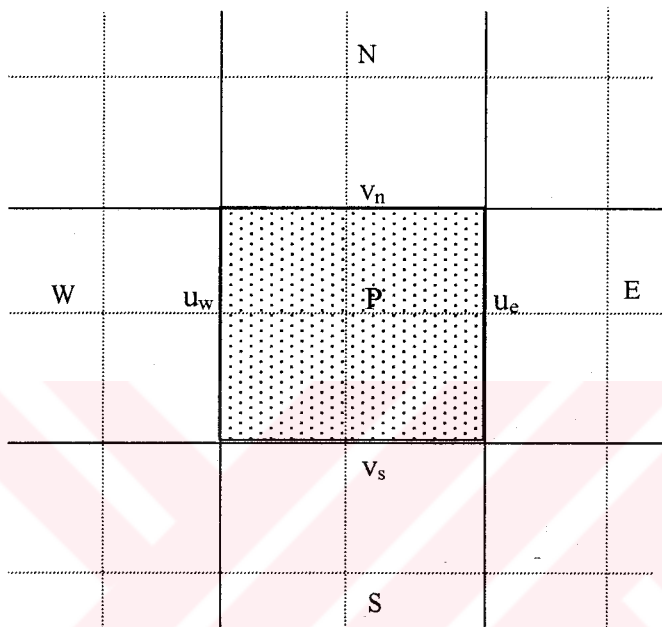


Figure 3.5 Grid layout for discretization of continuity equation

Substituting all velocity components in “Equation 3.26” from “Equations 3.24 and 3.25”, following equation for pressure can be obtained:

$$a_P p_P = a_E p_E + a_W p_W + a_N p_N + a_S p_S + b_P \quad (3.27)$$

Where:

$$b_P = (\hat{u}_w - \hat{u}_e)\Delta y + (\hat{v}_s - \hat{v}_n)\Delta x \quad (3.28)$$

$$a_E = d_e \Delta y \quad (3.29)$$

$$a_W = d_w \Delta y \quad (3.30)$$

$$a_N = d_n \Delta x \quad (3.31)$$

$$a_S = d_s \Delta x \quad (3.32)$$

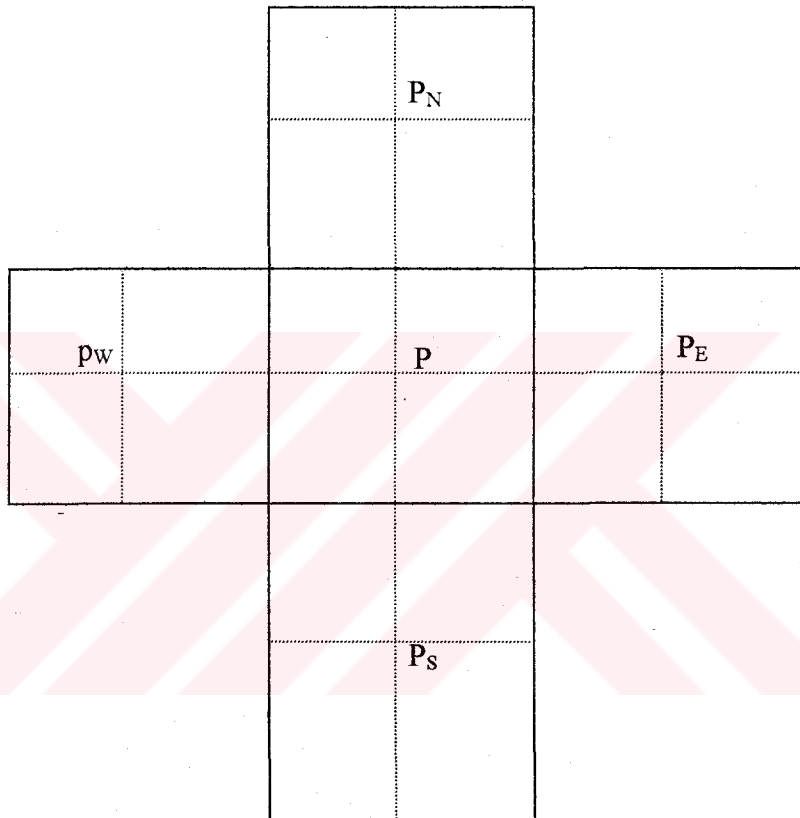


Figure 3.6 Grid layout for Pressure equation

3.2.4 The Pressure Correction Equation

The momentum equations can be solved only when the pressure field is given or is somehow estimated. Unless the correct pressure field is employed, the resulting velocity field will not satisfy the continuity equation. Such an imperfect velocity field based on a guessed pressure field p^* will be denoted by u^* and v^* . This “starred” velocity field will result from the solution of the following discretization equations, written as “Equation 3.20”.

$$a_e u_e^* = \sum a_{nb} u_{nb}^* + b_2 + A_e (p_P^* - p_E^*) \quad (3.33)$$

$$a_n u_n^* = \sum a_{nb} u_{nb}^* + b_2 + A_n (p_P^* - p_N^*) \quad (3.34)$$

Aim is to obtain correct pressure field from guessed pressure field, p^* . If correct pressure field is denoted by p and the correction introduced in guessed pressure field is denoted as p' . Then following equation could be written.

$$p = p^* + p' \quad (3.35)$$

Similarly correct velocity components can be introduced by the following equations.

$$u = u^* + u' \quad (3.36)$$

$$v = v^* + v' \quad (3.37)$$

The general discretized momentum equation for u velocity component written as “Equation 3.20”.

$$a_e u_e = \sum a_{nb} u_{nb} + b_2 + A_e (p_P - p_E) \quad (3.38)$$

If “Equation 3.33” is subtracted from “Equation 3.38” following velocity correction equation is obtained:

$$a_e u'_e = \sum a_{nb} u'_{nb} + b_2 + A_e (p'_P - p'_E) \quad (3.39)$$

Now omitting first term on right hand side of “Equation 3.39” (a detailed discussion at this point has been given by **Patankar (1980)**), following equation is obtained:

$$u'_e = d_e (p'_P - p'_E) \quad (3.40)$$

Combining “Equation 3.40” and “Equation 3.36” result is the following equation:

$$u_e = u_e^* + d_e (p'_P - p'_E) \quad (3.41)$$

If the above last few steps are repeated for v velocity component then the following equation is the result.

$$v_n = v_n^* + d_n(p'_P - p'_N) \quad (3.42)$$

Where d_n has been defined in “Equation 3.21” and d_e is defined similarly. “Equations 3.41 and 3.42” are called velocity correction equations because these equations are used to calculate correct velocity field from guessed velocity field using pressure correction. Now substituting “Equations 3.41 and 3.42” in discretized continuity equation (“Equation 3.26”), after rearrangements pressure correction equation can be written as given below:

$$a_P p'_P = a_E p'_E + a_W p'_W + a_N p'_N + a_S p'_S + b'_P \quad (3.43)$$

$$b'_P = (u_w^* - u_e^*) \Delta y + (v_s^* - v_n^*) \Delta x \quad (3.44)$$

$$a_E = d_e \Delta y \quad (3.45)$$

$$a_W = d_w \Delta y \quad (3.46)$$

$$a_N = d_n \Delta x \quad (3.47)$$

$$a_S = d_s \Delta x \quad (3.48)$$

$$a_P = a_E + a_W + a_N + a_S \quad (3.49)$$

3.2.5 Initial and Boundary Conditions for Pressure and Pressure Correction Equation

SIMPLER algorithm was adopted for present analysis that includes separate pressure and pressure correction equations. Initial and boundary conditions are same for both of the equations. Pressure is used only to correct the velocity field developed during the iterative SIMPLER algorithm. However, velocities at the boundary points are taken to be known or constant for the present numerical analysis so that no boundary condition is needed for the pressure. In other words, boundary pressure does not arise during the calculations as shown in the following figures.

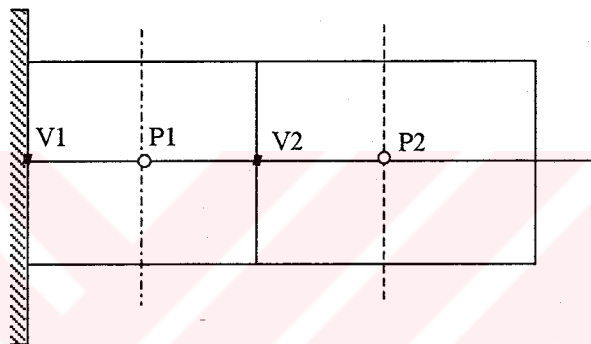


Figure 3.7 Layout for pressure boundary condition

For staggered grid arrangement adopted for the present analysis grid layout is shown in “Figure 3.7” Boundary velocity ($V1$) is always known as a given boundary condition while velocity $V2$ is corrected during each iteration. This correction needs pressure values at the left ($P1$) and at the right ($P2$) so that pressure at the boundary point does not arise in any of the equations. That is why no pressure boundary condition is necessary. Same discussion is valid for all other velocity components.

Initial pressure condition

$$\text{For } t = 0: \quad P = 0 \text{ for whole domain} \quad (3.50)$$

For $t > 0$: no boundary condition is necessary

4. SOLUTION PROCEDURE

4.1 The SIMPLER Algorithm

Various linearized equations described in previous sections are solved in a definite sequence to calculate various field variables describing the heat transfer and fluid flow within the solution domain. The specific sequence in which different governing equations are solved to calculate various field variables is called the solution scheme. For present analysis SIMPLER solution scheme (Patankar, 1980) was preferred due to its rigidity against fluctuations in temperature and velocity fields, observed during present simulations. A detailed step by step procedure SIMPLER algorithm is presented below.

1. Main program is started with guessed velocity field.
2. After calculating the coefficients for the momentum equations, $\hat{u}$ and $\hat{v}$ velocities are obtained from equations such as “Equation 3.23” by substituting the values of the neighbouring grid velocities u_{nb} .
3. Coefficients for the pressure equation “Equations 3.29-3.32” are calculated, and pressure equation is solved to obtain the pressure field.
4. Treating this pressure field as p^* , u^* and v^* are obtained from discretized momentum equations.
5. Mass source b_2 (“Equation 3.22”) is calculated which is used to solve the pressure correction equation for p' .
6. Correct velocity field is calculated by utilising “Equations 3.41 and 3.42”.
7. Discretized temperature equation is solved to obtain the temperature field.
8. Returned to step 2 and repeated until convergence.

4.2 Iterative Methods

SIMPLER algorithm does not attempt to solve all the linearized algebraic equations simultaneously instead it solves each variable separately in a sequence, which has

been designed to promote convergence. During solution of one of the variables many of the other unknown field variables exist in the discretized equations which are linearized by using their values from previous iteration. This is especially true for the momentum equations, which are essentially non-linear in nature but are linearized by using their values, in calculating coefficients, from previous iteration. Because of this non-linearity an iterative procedure is required. This has been done excellently in SIMPLER algorithm. The coefficients are calculated based on the current knowledge of the velocity and temperature fields. These linearized equations are then solved to give updated velocity and temperature fields. At end of the calculations ie convergence of the temperature field there is no change in the newly calculated field variables when compared to those in previous iteration.

4.2.1 Relaxation

The pressure equation “Equation 3.27” is susceptible to divergence unless some under-relaxation is used during the iterative process. Following equation was used to under-relax the pressure field.

$$p = p^* + \alpha_p p' \quad (4.1)$$

Where α_p is the pressure under-relaxation factor, always less than 1.

The velocities are also under-relaxed by using following equations.

$$u^{new} = \alpha_u u + (1 - \alpha_u) u^{n-1} \quad (4.2)$$

$$v^{new} = \alpha_v v + (1 - \alpha_v) v^{n-1} \quad (4.3)$$

Where α_u and α_v are the u and v velocities' under-relaxation factors, both having values between 0 and 1. Also u, v are correct velocity fields without under-relaxation and u^{n-1} and v^{n-1} represent their values obtained in the previous iteration.

4.2.2 Convergence Criterion

Convergence monitoring of temperature field was thought to be sufficient due to free convection nature of the flow that has temperature as a source term for the velocity

field. If temperature field converges, it essentially means a convergence of the velocity field. Convergence was monitored by calculating the fractional aggregate change in temperature field for each iteration using temperature fields from present iteration and previous one. Iterations are carried out again and again until this factor is driven to an acceptably low value. The convergence of computations is established by utilising the following relation for the temperature distribution:

$$\sum_{i,j} \left| T_{i,j}^{n+1} - T_{i,j}^n \right| / T_{\max} \leq e \quad (4.4)$$

Here $T_{\max}$ is the maximum temperature in the cavity for each iteration, e is the prescribed acceptable error and superscripts, $n+1$ and n , denote the present and previous iteration, respectively.

Pressure and pressure correction equations were solved using point successive over-relaxation procedure (PSOR). This technique has gone a long way in stabilising the fluctuating flow during present analysis. The convergence criteria adopted for pressure and pressure correction equations is to check the total absolute change in the pressure field during present iteration from that of previous one. This criterion could be written in equation form as given below:

$$\sum_{i,j} \left| P_{i,j}^{n+1} - P_{i,j}^n \right| \leq e \quad (4.5)$$

4.2.3 Linear Equations Solver

In the previous sections method of discretization of the governing equations has been outlined. This process results in a system of linear algebraic equations, which need to be solved during iterative solution scheme described in section 4.1. The size of linear equations developed through the iterative solution procedure depends upon the grid size utilised for the analysis. The main constraint for the solution of a set of linearized algebraic equations comes from computer resources and computing time is directly dependent upon the size of the equations set. In present analysis a transient problem was tackled and due to the fact that accuracy is an inverse function of the time step, puts further pressure upon the available computer resources. Direct methods for solution of linearized algebraic equations were not a choice due to large

computer time required. As is available in open literature number of operations required for the solution of a set of N linear equations is N^3 . Thus number of operations required for solution of linear equation system by direct method becomes troublesome with increasing grid size.

Thomas algorithm is a technique for the rapidly solving tri-diagonal systems that is now called Tri-Diagonal Matrix Algorithm (TDMA) described in detail by **Versteeg and Malalsekera (1996)**. Actually TDMA is a direct method for one-dimensional situations, but it can be applied iteratively, in a line by line fashion, to solve multi-dimensional problems. The special advantage of TDMA is minimum storage requirement and is computationally inexpensive. Due to these relative benefits TDMS was utilised during present analysis to solve linear equation developed during solution of temperature and velocity fields.

Further, pressure term appears as a main source term in momentum equations. The accuracy of the final velocity field depends mainly upon the accuracy of the input pressure field. That is why Point Successive over-relaxation (PSOR) procedure was chosen to solve the pressure equation, inspite of its expensiveness. The expensivity of the PSOR procedure was somewhat compensated by relatively large time step allowable with a combination of the TDMA and PSOR.

4.3 The Analysis Software

The numerical procedure and solution method described in previous sections was implemented in a FORTRAN-90 code ANVAR (Analysis of Natural convection during severe Accident of nuclear Reactors). Code was a new effort and no benefit was taken from any of the similar or dissimilar softwares. Thus a short explanation of its organisation, features and some other details are presented here.

4.3.1 Code Organisation

The main organisation of the ANVAR code is given in the form of a flow chart in "Figure 4.1". The software generally follows the main steps of SIMPLER algorithm given in section 4.1.

At first step program begins with reading as input the velocity, temperature and pressure fields. The input parameters, which are not to be altered during the program

execution (constants), are read by calling in a subroutine named INANVAR. After reading the input constants and initialising all the variables, program calls on the grid generation subroutine, GRIDANVAR. This subroutine generates the grid using input data from the main program. It also checks if the stretched or un-stretched grid is needed and returns the appropriate coordinates for the generated staggered grid and control volume dimensions for the flow domain. This subroutine also writes an output of the generated grid in input Co-ordinate system and another in Cartesian coordinates. These outputs can be checked for accuracy by drawing the particular grid using a grid drawing software.

After taking the grid layout from the GRIDANVR, main program calls in another subroutine, VIEWFAC, to calculate the view factors for each grid interval to be used to calculate the radiative heat transfer. For these calculations the data supplied by INANVAR and GRIDANVAR is used.

Now the executive program begins the main iteration loop. First of all linearized coefficients for the energy equation are calculated using known velocity field. Then temperature field is calculated using TDMA procedure first in radial direction and then in ϕ direction. During these calculations Three Diagonal Matrix Algorithm (TDMA) is called in to solve the linear set of algebraic equations for each direction with embedded appropriate boundary conditions.

In next step radiative heat transfer analysis is carried out. First the matrix consisting of coefficients of the radiation equations is calculated. This matrix is solved by using two subroutines naming XLUDCMP and XLUBKSB. These subroutines solve the input matrix by direct method.

At this point the previously known pressure and velocity fields are renamed as guessed pressure and velocity fields for present iteration (u^* , v^* and p^*). Then coefficients for radial direction velocity component (u) are calculated, Using these coefficients new pseudovelocity ($\hat{u}$) field for radial direction is calculated utilising TDMA subroutine. Similarly in next step Pseudovelocity field in ϕ direction is calculated by using coefficients calculated for this direction. It is worth mentioning here that both radial and ϕ direction velocity components have different coefficients due to staggered grid arrangement used for present analysis.

Next the pressure equation is solved to calculate “starred” pressure field using pseudovelocities as input and coefficients calculated using velocities from the previous iteration velocities. The pressure equation is solved by using PSOR procedure. Once new pressure field is known “starred” velocity (u^* , v^*) field is determined using “Equations 3.33 and 3.34”. For these calculations coefficients calculated during pseudovelocities calculations are utilised.

Using “starred” velocities, correct pressure field is calculated by solving pressure correction equation “Equation 3.43”. Pressure correction equation is also solved by PSOR procedure and coefficients developed for “starred” pressure calculations. Only difference is in the source term, which is calculated by using “starred” velocities.

At this point “Equations 3.41 and 3.42” are used to correct the velocity field by using correct pressure field developed in previous step. Last step of present iteration is to check for convergence as given by “Equation 4.4”. Then control is transferred to next iteration for temperature field calculations if temperature field is still not converged. When required convergence criterion for the temperature field is achieved control passes to an output subroutine, which writes all output data for velocity, pressure and temperature fields in out put files. Thus the iteration process ends with this output.

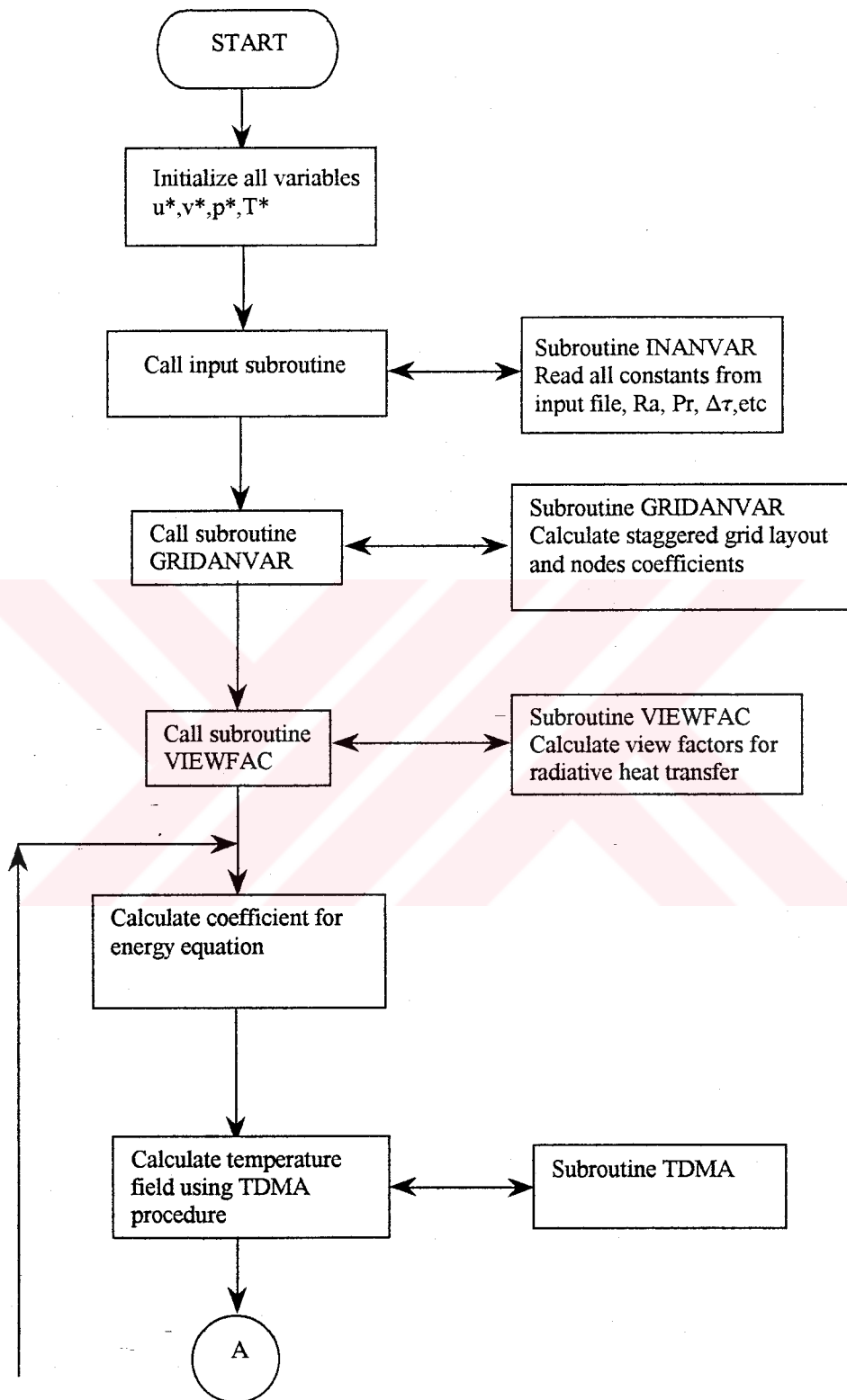
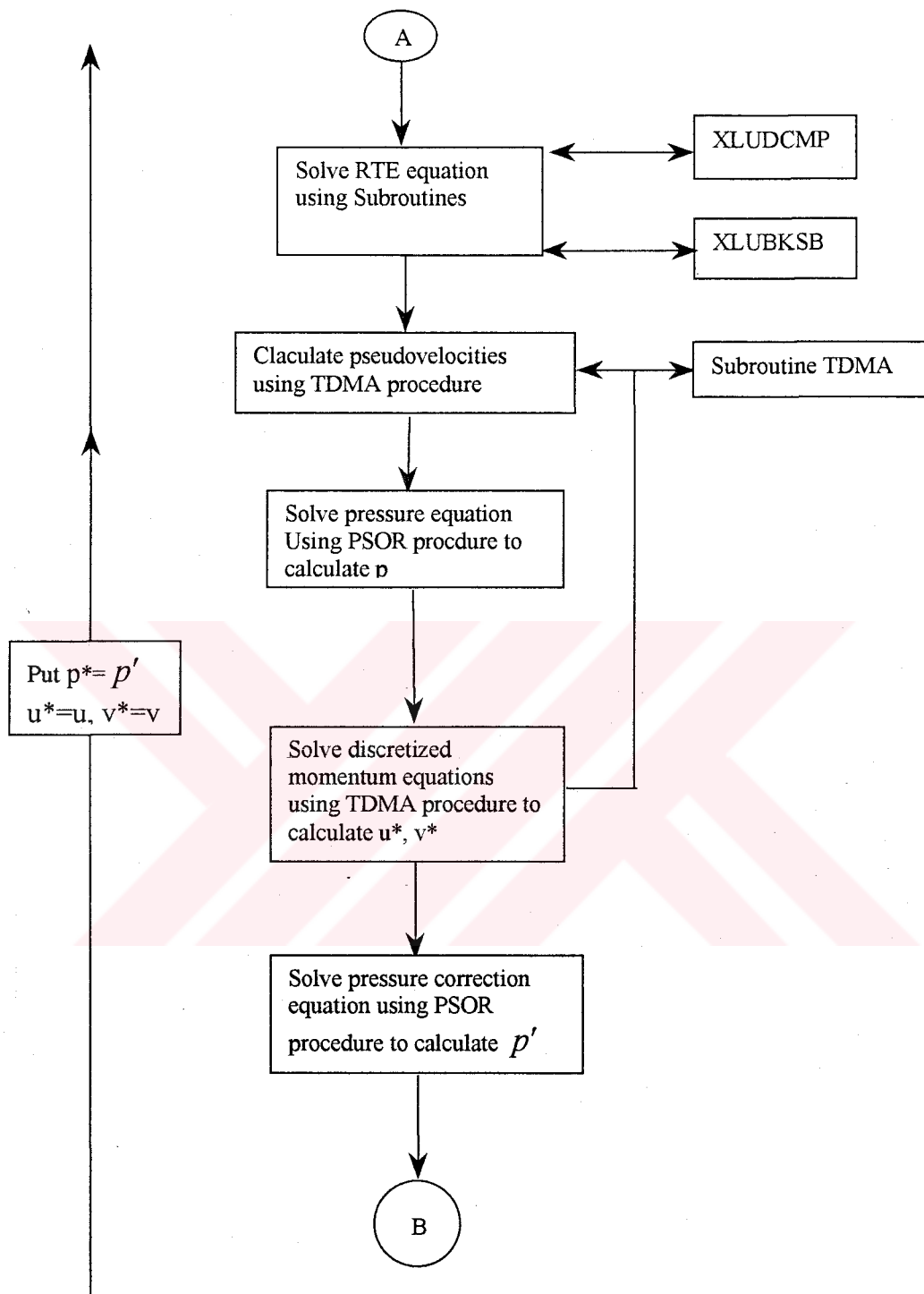


Figure 4.1 Flow diagram of the analysis software (continued)



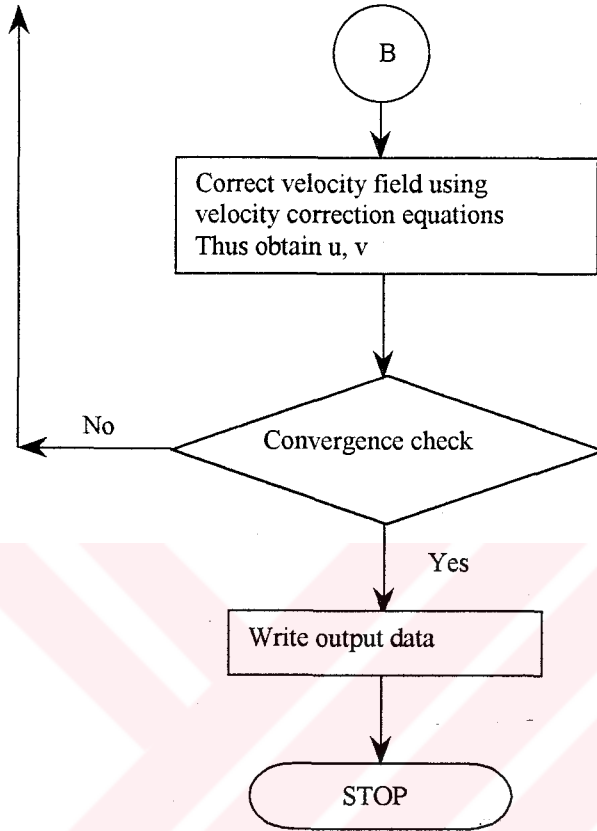


Figure 4.1 Flow diagram of the analysis software (ANVAR)

5. RESULTS AND DISCUSSION

Actual accident scenario, considered here, is very complicated both in geometry and materials properties variations due to high temperatures. Therefore, few simplifying assumptions were necessary to make the problem handleable numerically. These assumptions are listed below.

1. Phase change of Corium is not considered.
2. Fluid is Newtonian.
3. Material properties are independent of temperature.
4. Boussinesq approximation is valid for fluid density.
5. Flow is two dimensional (2-D).
6. Flow inside the cavity is laminar.
7. Internal heating source is distributed uniformly within the body of the fluid.
8. For radiation heat transfer, all participating surfaces are assumed to be gray and diffuse.
9. Heat generated in the RPV wall due to gamma rays is neglected.

For 2-D problems, it is a usual practice to cancel out the pressure term from momentum equations by cross differentiation, thus employing stream function and vorticity formulation. Thus analysis is well simplified due to disappearance of the pressure term. In present analysis methodology 'SIMPLER' based algorithm was developed using primitive variables i.e. velocity and pressure. Thus it became essential to verify the accuracy of the present method by comparing with results from the open literature. First of all a low Ra number benchmark solution for discretely heated cavity was prepared. Then high Ra number results were compared with the results from published results of other researchers. A short summary of the benchmark results is given below.

5.1 Low Rayleigh Number Benchmark Solution

5.1.1 Square Cavity with Discrete Heating

In heat transfer literature the most established benchmark problem is that of a discretely heated square cavity. In this problem horizontal walls of the cavity are adiabatic while right and left vertical walls are kept at constant dimensionless temperature of -0.5 and 0.5 , respectively.

The governing equations are unsteady Navier-Stokes equations for laminar free convection ("Equations 2.13-2.16"). These equations are non-dimensionalised by using the scaling laws defined in "Table 2.1" for low Ra number and corresponding non-dimensional governing equations are given below:

Continuity Equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (5.1)$$

X momentum equation

$$\frac{\partial U}{\partial \tau} + U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \text{Pr} \nabla^2 U \quad (5.2)$$

Y momentum equation

$$\frac{\partial V}{\partial \tau} + U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \text{Pr} \nabla^2 V + \text{Ra Pr} \theta \quad (5.3)$$

Energy equation

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \nabla^2 \theta \quad (5.4)$$

The corresponding initial and boundary conditions are:

For $\tau \leq 0$: $\theta = U = V = 0$ for whole space

For $\tau > 0$: $U = V = 0$ at all walls (5.5)

$\theta = 0.5$ at $X = 0$

$$\theta = -0.5 \qquad \text{at } X = 1.0$$

$$\frac{\partial \theta}{\partial Y} = 0 \qquad \text{at } Y= 0 \text{ and } Y = 1.0$$

The average Nusselt number is defined as follows:

$$\overline{Nu} = \int_0^1 \left| \frac{\partial \theta}{\partial X} \right|_X dX \tag{5.6}$$

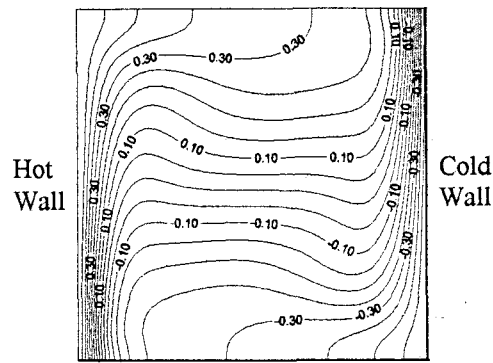
Where X is taken at 0 and 1.

Ra number range from 10^3 to 10^6 was analysed for the benchmark purpose but result presented here are for $Ra = 10^5$ in order to be brief. A comparison of the present numerical results with results from **Vahl Davis (1983)** is given in “Table 5.1”. The good agreement shown in “Table 5.1” validate the present numerical procedure.

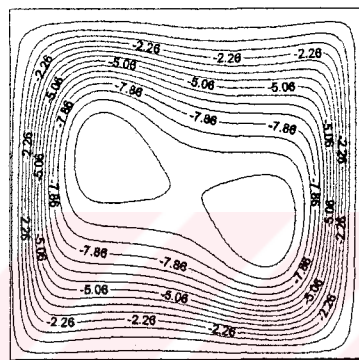
Isotherms, streamlines and pressure distribution, within the square cavity, are shown in “Figure 5.1”. “Figure 5.1(a)” clearly indicates the upward motion of the fluid along the hot wall and descending flow along cold wall. Unicellular structure of the flow is evident in “Figure 5.1(b)”, while pressure distribution in the cavity is shown in “Figure 5.1(c)”.

Table 5.1
Average Nusselt benchmark results for discretely heated square cavity for $Pr = 7$.

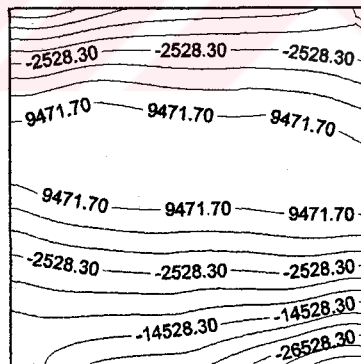
Ra	Present Numerical Results	Results From Vahl Davis (1983)
1.0×10^3	1.102	1.116
1.0×10^4	2.252	2.242
1.0×10^5	4.616	4.564
1.0×10^6	9.439	9.270



(a)



(b)



(c)

Figure 5.1 Steady state results for $Ra = 10^5$ and $Pr = 7.0$
 (a) isotherms, (b) streamlines, (c) pressure distribution

5.1.2 Square Cavity with Internal Heating Sources

May (1991) and **Baytaş (1996)** presented their results for internally heated square cavity with all walls held at constant temperature. These results were compared with the present results to check the presently developed software and methodology. The problem when subjected to related assumptions given in section 5.1, could be described by the non-dimensional governing equations as given below:

Continuity Equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (5.7)$$

X momentum equation

$$\frac{\partial U}{\partial \tau} + U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \text{Pr} \nabla^2 U \quad (5.8)$$

Y momentum equation

$$\frac{\partial V}{\partial \tau} + U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \text{Pr} \nabla^2 V + 8.0 \text{Ra} \text{Pr} \theta \quad (5.9)$$

Energy equation

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \nabla^2 \theta + 8.0 \quad (5.10)$$

The corresponding initial and boundary conditions are:

For $\tau \leq 0$: $\theta = U = V = 0$ for whole space

For $\tau > 0$: $U = V = \theta = 0$ at all walls (5.11)

The average Nusselt number is defined as follows:

$$\overline{Nu} = \frac{1}{2} \int_0^1 \left| \frac{\partial \theta}{\partial X} \right|_X dX \quad (5.12)$$

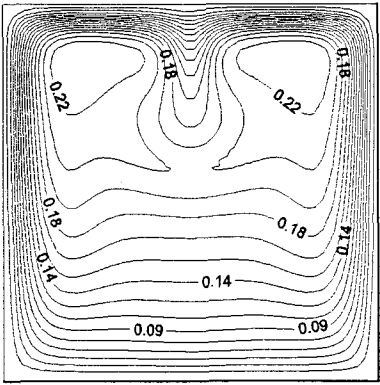
Where X is taken at 0 and 1.0.

“Table 5.2” shows at a glance the best comparison achieved through the methodology adopted for the analysis of the situation considered here. Ra number range from 10^3 to 10^6 was studied but results presented here are only for $Ra = 10^5$ in order to be brief. “Figure 5.2(a)” shows the stratified and symmetric isotherms for $Ra = 10^5$. Multi-cellular flow structure is evident from streamlines distribution presented in “Figure 5.2(b)”. Pressure distribution has been presented in “Figure 5.2(c)”, here pressure increases from top to bottom, as is for free convection.

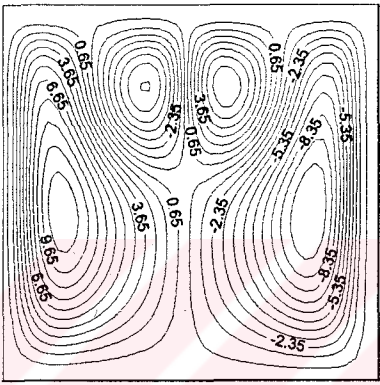
Table 5.2

Benchmark results for internally heated cavity for $\theta_{\max}$

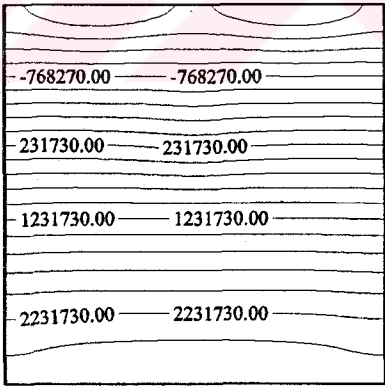
Ra	Present Numerical Results	Results from May (1991)	Results from Baytaş (1996)	Results from Lee et al. (1988)
1.0×10^4	0.374	0.382	0.378	0.386
5.0×10^4	0.269	0.270	0.280	0.293
1.0×10^5	0.2220	0.235	0.255	0.256
1.0×10^6	0.144	-----	0.153	-----



(a)



(b)



(c)

Figure 5.2 Steady state results for $Ra = 10^5$ and $Pr = 7.0$
(a) isotherms, (b) streamlines, (c) pressure distribution

5.2 High Rayleigh Number Benchmark Solution

5.2.1 Square Cavity with Discrete Heating

High Ra number buoyant flow or the boundary layer flow analysis was carried out for the square cavity with left wall at higher temperature as compared to right wall, while keeping horizontal walls adiabatic. This problem is governed by the following non-dimensional equations.

Continuity equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (5.13)$$

X momentum equation

$$\frac{\partial U}{\partial \tau} + U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{\text{Pr}}{(Ra \text{Pr})^{2/5}} \nabla^2 U \quad (5.14)$$

Y momentum equation

$$\frac{\partial V}{\partial \tau} + U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{\text{Pr}}{(Ra \text{Pr})^{2/5}} \nabla^2 V + \theta \quad (5.15)$$

Energy equation

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{1}{(Ra \text{Pr})^{2/5}} \nabla^2 \theta \quad (5.16)$$

The corresponding initial and boundary conditions are:

For $\tau \leq 0$: $\theta = U = V = 0$ for whole space

For $\tau > 0$: $U = V = 0$ at all walls (5.17)

$$\theta = 0.5 \text{ at } X = 0$$

$$\theta = -0.5 \text{ at } X = 1.0$$

$$\frac{\partial \theta}{\partial Y} = 0 \text{ at } Y = 0 \text{ and } Y = 1.0$$

The average Nusselt number is defined as follows:

$$\overline{Nu} = \int_0^1 \left| \frac{\partial \theta}{\partial X} \right|_X dX \tag{5.18}$$

Where X is taken at 0 and 1.0.

For benchmarking, present numerical results for average Nusselt number has been presented with the reference results from **Lage and Bejan (1991)** in “Table 5.3”. Ra number range from 10^6 to 10^9 has been scrutinised. Both results show a very good agreement as is clear from “Table 5.3” except for $Ra = 10^9$ which is possibly due to oscillations in the temperature field at such a high Ra number. Steady state isotherms for $Ra = 10^9$ are drawn in “Figure 5.3(a)”. A strong boundary layer flow along the hot and cold walls can be seen easily. This boundary layer flow pattern is also visible in “Figure 5.3(b)”. “Figure 5.3(c)” shows the clean horizontal contours of the pressure distribution.

Table 5.3

Average Nusselt number benchmark results for discretely heated square cavity for high Ra number

Ra	Present Numerical Results	Results From Lage and Bejan (1991)
1.0×10^6	8.973	9.20
1.0×10^7	17.05	17.90
1.0×10^8	32.81	31.80
1.0×10^9	68.38	62.70

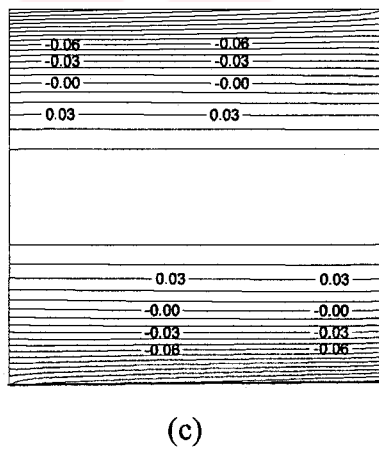
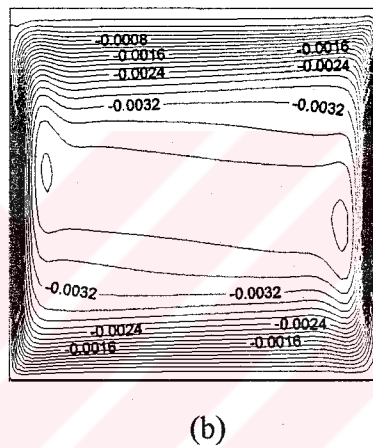
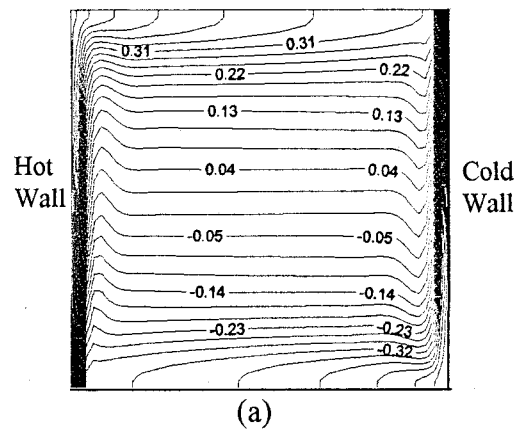


Figure 5.3 Steady state results for $Ra = 10^9$, $Pr = 7.0$
 (a) isotherms, (b) streamlines, (c) pressure distribution

5.2.2 Square Cavity with Internal Heating Sources**

Internally heated square cavity with all walls being cooled, was analysed for high Ra number buoyant flow. In this study, a new scaling for high Ra number fluctuating flow has been proposed. The non-dimensional time dependent conservation equations of mass and momentum are the same as given in “Equations 5.13-5.15” while energy equation can be written as given below:

Energy equation

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{1}{(RaPr)^{2/5}} \nabla^2 \theta + (RaPr)^{-1/5} \quad (5.19)$$

The non-dimensional variables used in above equations are defined in the Nomenclature.

The corresponding initial and boundary conditions are:

For $\tau \leq 0$: $\theta = U = V = 0$ for whole space

For $\tau > 0$: $\theta = U = V = 0$ at all the walls (5.20)

The average Nusselt number is as defined in “Equation 5.18”.

Present numerical results have been compared with experimental results from **Nourgaliev et al. (1997)** in “Figure 5.4”. A good agreement is clearly evident. Stratified, strong boundary layer flow is clearly visible in isotherms presented in “Figure 5.5(a)” for $Ra = 10^9$ and $Pr = 7$. Flow in upper part of the cavity is fluctuating with cold lobe moving to the right. Streamlines in “Figure 5.5(b)” show the multicellular structure of the flow. Standard pressure distribution for buoyant flow can be seen in “Figure 5.5(c)” where pressure increases from top to bottom in horizontal contours.

** The results presented here were also published as **Liaqat and Baytaş (2000)**

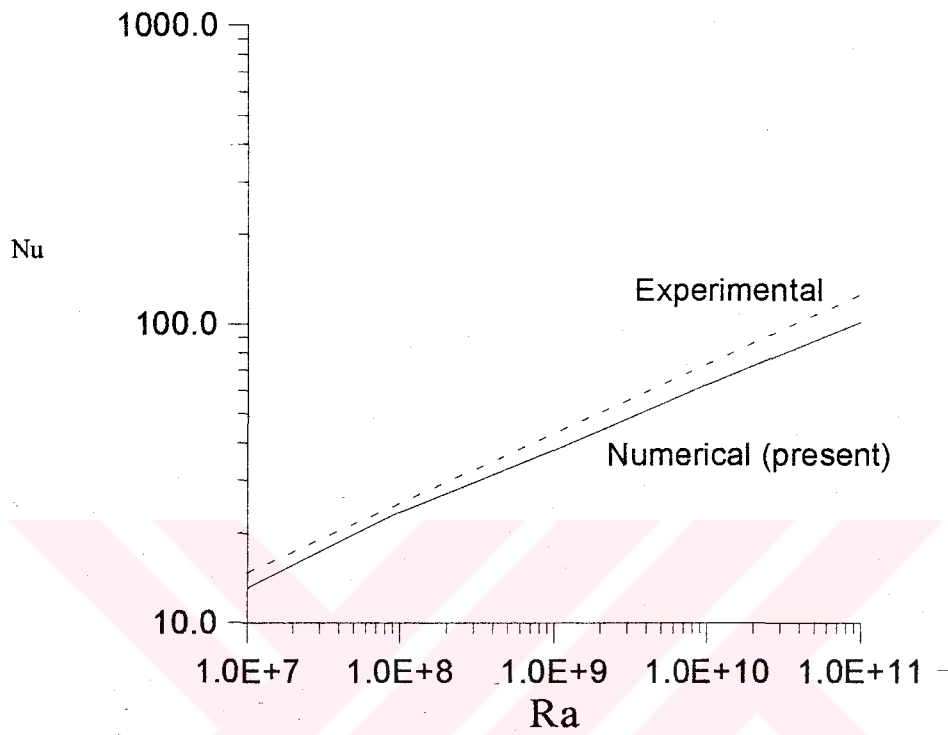
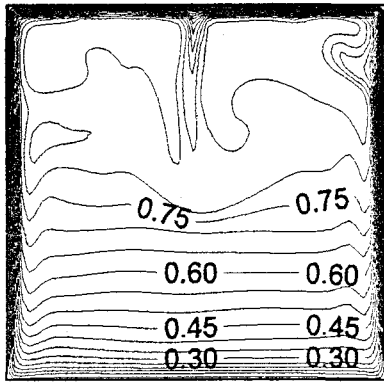
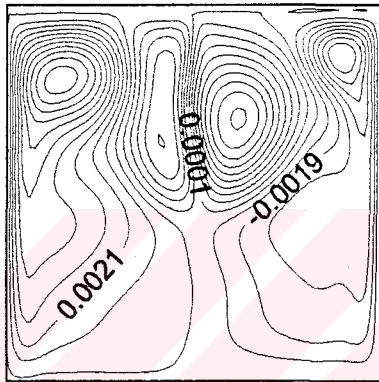


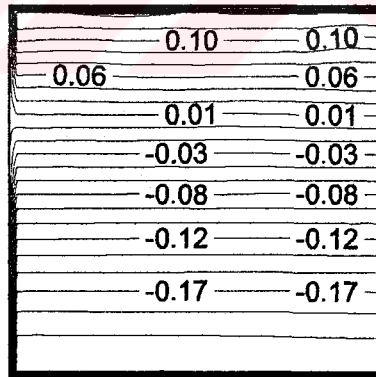
Figure 5.4 Comparison of present numerical results with experimental results from Nougaliiev et al. (1997)



(a)



(b)



(c)

Figure 5.5 Steady state results for $Ra = 10^9$, $Pr = 7.0$
 (a) isotherms, (b) streamlines, (c) pressure distribution

5.3 High Rayleigh number Benchmark Solution for Spherical Segment Geometry**

“Figure 2.2” shows the details of the physical situation analysed in the present study. A 2-D hemispherical cavity with all walls at constant temperature or isothermal ie rigid conducting walls and all velocities are zero at the walls (non-slip boundary condition). Problem was formulated using assumptions described in section 5.1.

The governing dimensionless equations of mass, momentum and energy for the hemispherical cavity could be given as below:

Momentum equations

$$\frac{DU}{D\tau} - \frac{V^2}{R} = -\frac{\partial P}{\partial R} + \frac{Pr}{(Ra Pr)^{0.4}} \left(\nabla^2 U - \frac{2U}{R^2} - \frac{2}{R^2} \frac{\partial V}{\partial \phi} - \frac{2V \cot \phi}{R^2} \right) - Ra \theta \sin \phi \quad (5.21)$$

$$\frac{DV}{D\tau} + \frac{UV}{R} = -\frac{1}{R} \frac{\partial P}{\partial \phi} + \frac{Pr}{(Ra Pr)^{0.4}} \left(\nabla^2 V + \frac{2}{R^2} \frac{\partial U}{\partial \phi} - \frac{V}{R^2 \sin^2 \phi} \right) - Ra \theta \cos \phi \quad (5.22)$$

Energy equation

$$\frac{D\theta}{D\tau} = \frac{1}{(Ra Pr)^{0.4}} \left(\frac{1}{R^2} \frac{\partial}{\partial R} \left(R^2 \frac{\partial \theta}{\partial R} \right) + \frac{1}{R^2 \sin \phi} \frac{\partial}{\partial \phi} \left(\sin \phi \frac{\partial \theta}{\partial \phi} \right) \right) + \frac{1}{(Ra Pr)^{0.2}} \quad (5.23)$$

where:

$$\nabla^2 = \frac{1}{R^2} \frac{\partial}{\partial R} \left(R^2 \frac{\partial}{\partial R} \right) + \frac{1}{R^2 \sin \phi} \frac{\partial}{\partial \phi} \left(\sin \phi \frac{\partial}{\partial \phi} \right)$$

Initial and boundary conditions for the dimensionless “Equations 5.21-5.23” can be summarised as following:

** This section is based upon the results presented in Liaqat and Baytaş (2000d)

For $\tau \leq 0$: $U = V = \theta = 0$ for the whole domain

$$\text{For } \tau > 0: \quad V=U=0 \quad \text{at } R=R_1 \quad \text{and} \quad \phi=\frac{\pi}{2}, \quad (5.24)$$

$$\frac{\partial U}{\partial \phi} = 0 \quad \text{at} \quad \phi = \pi$$

$$V = 0 \quad \text{at} \quad \phi = \pi$$

$$\theta = 0 \quad \text{at } R=R_1 \quad \text{and} \quad \phi=\frac{\pi}{2},$$

$$\frac{\partial \theta}{\partial \phi} = 0 \quad \text{at} \quad \phi = \pi$$

In order to evaluate the local and average heat transfer rates, the local Nusselt number referred to the pure conduction situation through the fluid that fills the cavity at each point of the wall could be listed as:

$$Nu = -\left(\frac{\partial \theta}{\partial R}\right) \quad \text{for bottom wall} \quad (5.25)$$

$$Nu = -\frac{1}{R} \left(\frac{\partial \theta}{\partial \phi}\right) \quad \text{for top wall}$$

The average Nusselt number for the enclosing upper and bottom boundaries is thus obtained by integrating “Equation 5.25”, as given below:

$$\overline{Nu} = \frac{2}{\pi} \int_{\frac{\pi}{2}}^{\pi} Nu \, d\phi \quad \text{For bottom wall} \quad (5.26)$$

$$\overline{Nu} = \frac{1}{R_1} \int_0^{R_1} Nu \, dR \quad \text{For top wall} \quad (5.27)$$

Total of 6 simulations for Ra number 10^7 to 10^{12} were performed for the present analysis. Every simulation was carried out for a Pr number of 7.0 and time step varying from 10^{-2} to 10^{-3} , for different Ra numbers. Average and local Nusselt numbers, flow fields and isotherms were obtained for each simulation.

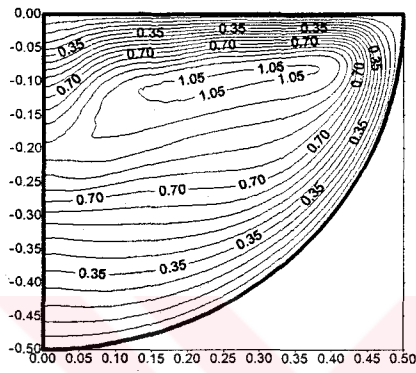
Isotherms for Ra number values of 10^7 , 10^{10} , and 10^{12} are shown in “Figure 5.6”. Temperatures are well stratified in lower part of the cavity, while upper part of the hemi-spherical cavity is dominated by time dependent oscillations in temperature and flow parameters. Also, higher temperatures are located in upper half of the cavity, as is natural for buoyant flows. Temperature boundary layer thickness along the enclosing walls decreases, indicating higher temperature gradient along the walls as Ra number increases from 10^7 to 10^{12} . As heat is removed from the upper wall, a tongue of fluid descends downward as is clearly visible in “Figure 5.6”. This phenomenon paves way for the oscillations of flow and heat transfer parameters in this investigation. Penetration depth of the fluid descending along the walls increases as Ra number is increased.

The streamlines for Ra numbers of 10^7 , 10^{10} and 10^{12} are drawn in “Figure 5.7”. For Ra of 10^7 , the flow shows a bi-cellular structure. For Ra numbers of 10^{10} and 10^{12} , the flow shows multicellular structure while two main cells dominate. The position of flow cells in “Figure 5.7(b) and 5.7(c)” changes with time and the cells gradually become stronger with the increase in Ra number. “Figure 5.8” shows $\overline{Nu}$ for upper and bottom walls as a function of dimensionless time for Ra number 10^{10} . Both walls show a non-periodic variation in average Nusselt number. Amplitude of these oscillations increases gradually with Ra number. Local wall Nusselt number variation along the curved wall is graphed in “Figure 5.9” for Ra = 10^7 . Local Nusselt number value at the lower stagnation point remains almost the same for all Ra numbers. Local Nusselt number increases along with the distance from the stagnation point for a maximum at somewhat lower point than the top most point of the cavity.

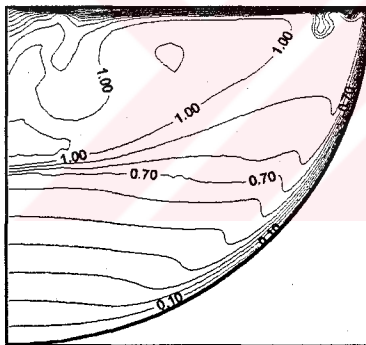
Variation of $\overline{Nu}$ for top wall with Ra number is shown in “Figure 5.10” on logarithmic scale. As expected $\overline{Nu}$ value increases with Ra number. In “Figure 5.10” experimental results for top wall average Nusselt number, taken from **Asfia and**

Dhir (1996) are also shown, present results agree very well with those experimental observations.

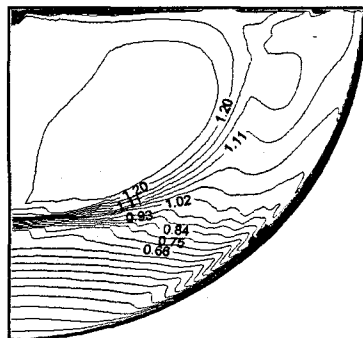
High Ra number laminar buoyant flow in a hemi-spherical cavity with uniform internal heat sources has been analysed numerically. This analysis shows that the numerical predictions of natural convection in internally heated pools, by present method, are in good agreement with the previously published experimental results. The system shows non-periodic time dependant oscillations for all Ra numbers considered for the present study.



(a)

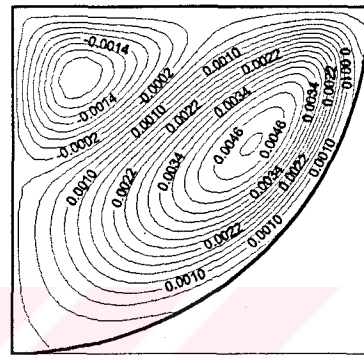


(b)

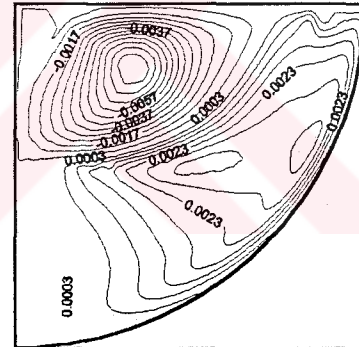


(c)

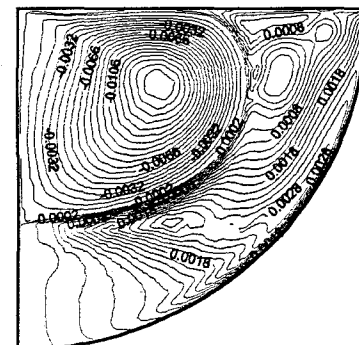
Figure. 5.6 Isotherms for (a) $Ra = 10^7$,
(b) $Ra = 10^{10}$, (c) $Ra = 10^{12}$



(a)



(b)



(c)

Figure. 5.7 Streamlines for (a) $Ra = 10^7$,
 $Ra = 10^{10}$, (c) $Ra = 10^{12}$

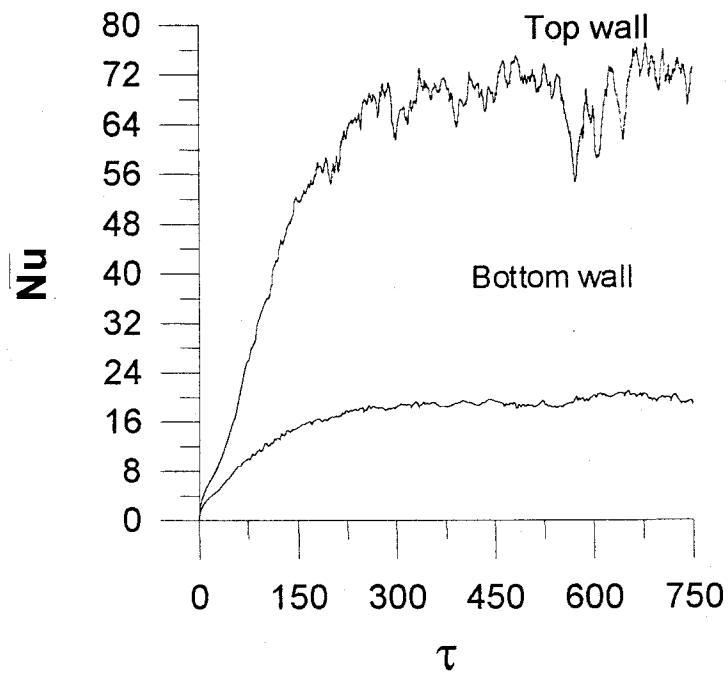


Figure 5.8 Development of average wall Nusselt number with time for $Ra = 10^{10}$

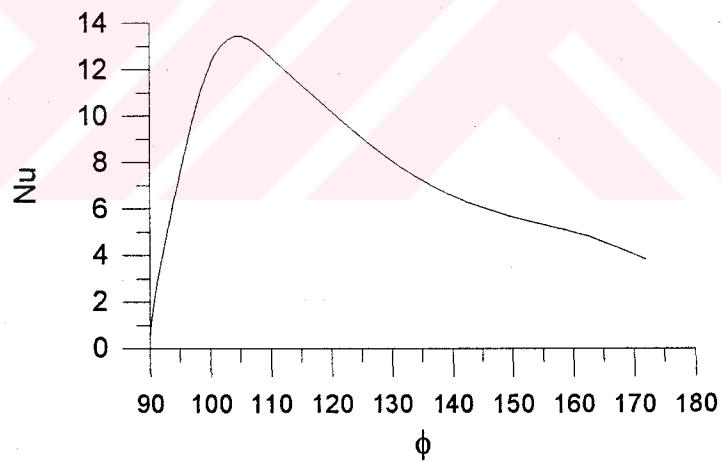


Figure 5.9 Variation of local wall Nusselt number with angular position along ϕ for $Ra = 10^7$

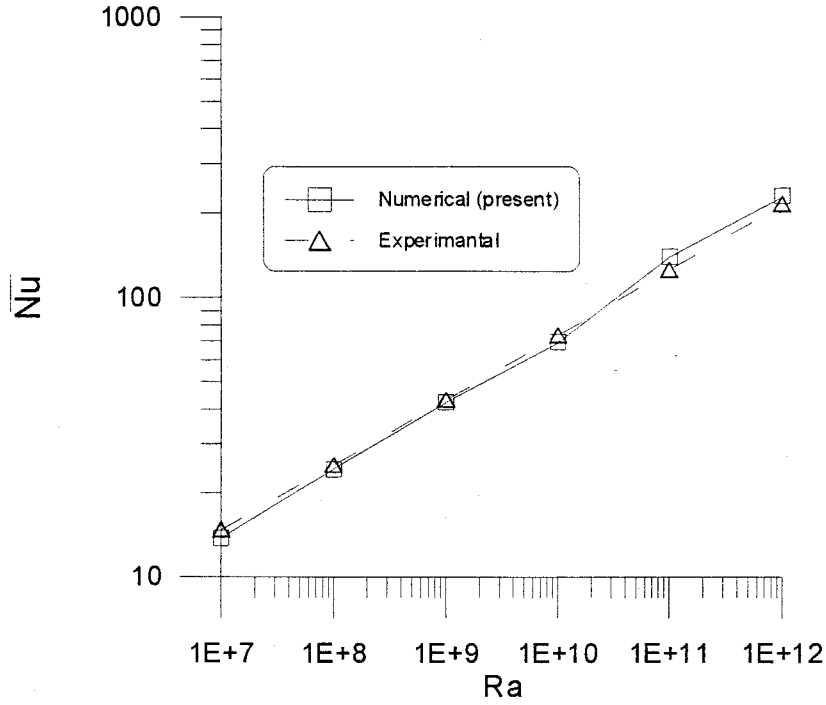


Figure 5.10 Comparison of present numerical results for $\overline{Nu}$ for pool top surface with experimental results from Asfia and Dhir (1996)

5.4 High Rayleigh Number Benchmark Solution for Semi-Circular Geometry

The domain under consideration is a two-dimensional semi-circular cavity with all isothermal walls, as shown in “Figure 2.1”. The problem is described by the following non-dimensional equations obtained from “Equations 2.17-2.20” by utilising the non-dimensional variables from “Table 2.1”.

Continuity Equation

$$\frac{\partial U}{\partial R} + \frac{U}{R} + \frac{1}{R} \frac{\partial V}{\partial \phi} = 0 \quad (5.28)$$

Momentum equations

$$\frac{\partial U}{\partial \tau} + U \frac{\partial U}{\partial R} + \frac{V}{R} \frac{\partial U}{\partial \phi} - \frac{V^2}{R} = -\frac{\partial P}{\partial R} + \frac{\text{Pr}}{(Ra \text{Pr})^{2/5}} \left(\nabla^2 U - \frac{U}{R^2} - \frac{2}{R^2} \frac{\partial V}{\partial \phi} \right) - \theta \sin \phi \quad (5.29)$$

$$\frac{\partial V}{\partial \tau} + U \frac{\partial V}{\partial R} + \frac{V}{R} \frac{\partial V}{\partial \phi} + \frac{UV}{R} = -\frac{1}{R} \frac{\partial P}{\partial \phi} + \frac{\text{Pr}}{(Ra\text{Pr})^{2/5}} \left(\nabla^2 V - \frac{V}{R^2} + \frac{2}{R^2} \frac{\partial U}{\partial \phi} \right) - \theta \cos \phi \quad (5.30)$$

Energy equation for fluid domain

$$\frac{\partial \theta}{\partial \tau} + U \frac{\partial \theta}{\partial R} + \frac{V}{R} \frac{\partial \theta}{\partial \phi} = \frac{1}{(Ra\text{Pr})^{2/5}} \left(\frac{\partial^2 \theta}{\partial R^2} + \frac{1}{R} \frac{\partial \theta}{\partial R} + \frac{1}{R^2} \frac{\partial^2 \theta}{\partial \phi^2} \right) + (Ra\text{Pr})^{-1/5} \quad (5.31)$$

Where all dimensionless variables has been defined in the nomenclature.

$$\nabla^2 = \frac{\partial^2}{\partial R^2} + \frac{1}{R} \frac{\partial}{\partial R} + \frac{1}{R^2} \frac{\partial^2}{\partial \phi^2} \quad (5.32)$$

“Equations 5.28-5.31” are subjected to the following initial and boundary conditions.

Initial conditions

For $\tau \leq 0$: $U = V = \theta = 0$ for the whole domain

Boundary conditions

For $\tau > 0$: $U = V = 0$ for $R = R_1$ and $\phi = \frac{\pi}{2}, \frac{3\pi}{2}$

$\theta = 0$ for $\phi = \frac{\pi}{2}, \frac{3\pi}{2}$ and at $R = R_1$

While all the symbols appearing in above equations has been defined in the nomenclature.

In order to evaluate the local and average heat transfer rates, the local Nusselt numbers has been defined as:

$$Nu_{top} = \frac{1}{R} \frac{\partial \theta}{\partial \phi} \quad \text{for } R = 0 \text{ to } R_1 \text{ at upper wall} \quad (5.34)$$

$$Nu_{bottom} = \frac{\partial \theta}{\partial R} \quad \text{at } R = R_1 \text{ for bottom solid-fluid interface} \quad (5.35)$$

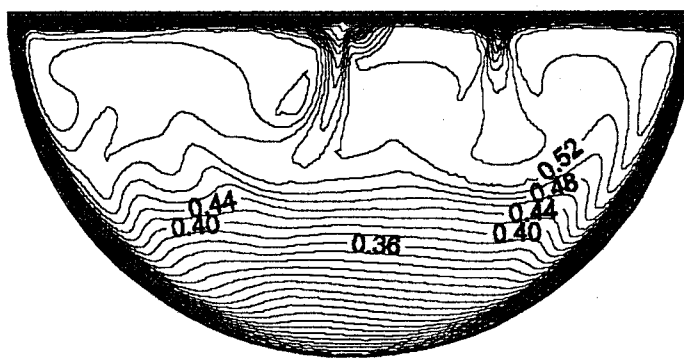
The average Nusselt number for the top and bottom walls are thus obtained by integrating “Equation 5.34 and 5.35”, respectively, as given below:

$$\overline{Nu}_{top} = \frac{1}{R_1} \int_0^{R_1} Nu_{top} dR \quad (5.36)$$

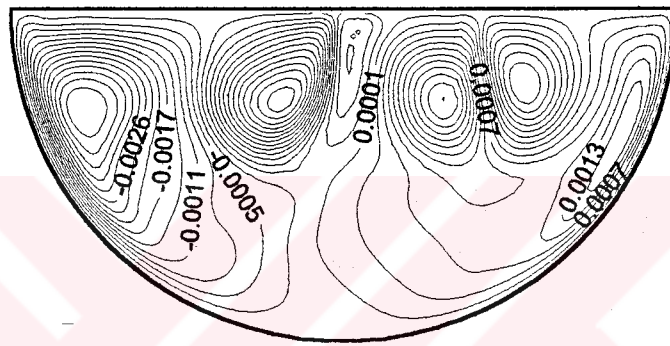
$$\overline{Nu}_{bottom} = \frac{1}{\pi} \int_0^{\pi} Nu_{bottom} d\phi \quad (5.37)$$

“Figure 5.11(a)” shows the isotherms for $Ra = 10^{10}$ and $Pr = 7.0$. Lower part of the semi-circular cavity is stratified while upper part shows a fluctuating behaviour. Two low temperature descending lobes visible in “Figure 5.11(a)” are the basic source of fluctuations in the flow field. Flow regime is multicellular in nature as is evident from “Figure 5.11(b)”, while there are very low velocities in the lower half of the cavity.

Benchmark solution prepared for the semi-circular cavity has been shown in “Figure 5.12”. A very good agreement is observed for top and bottom enclosing wall’s average Nusselt numbers.



(a)



(b)

Figure 5.11 Quasi-steady state results for $Ra = 10^{10}$, $Pr = 7.0$ (a) isotherms (b) streamlines

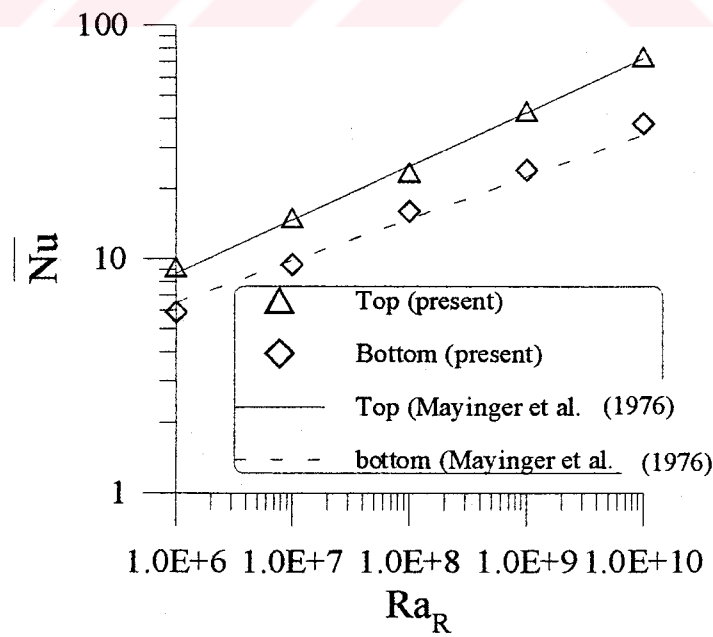


Figure. 5.12 Benchmark solution for the semi-circular cavity

5.5 Analysis of Square Cavity with Thick Walls^{**}, a Conjugate Problem

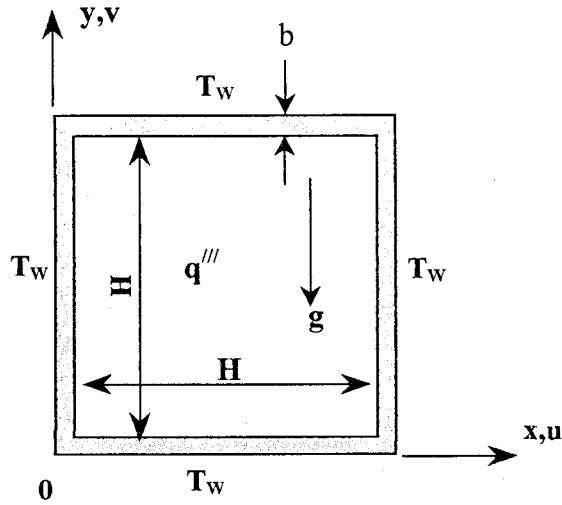


Figure 5.13 Schematic diagram of the conjugate problem

“Figure 5.13” shows the details of the physical situation to be analysed. Cavity is square and all walls are thick having constant outside temperature i.e. rigid walls with finite conductivities and all velocities are zero at the walls (non-slip boundary condition). The governing non-dimensional equations for the fluid region are same as “Equations 5.13-5.16”, while that of solid region is given below.

The two-dimensional temperature distribution in the walls is governed by the heat conduction equation, which in the dimensionless form is given below:

$$\frac{\partial \theta}{\partial \tau} = \frac{\alpha_w}{\alpha_f} \frac{1}{(Ra Pr)^{2/5}} \nabla^2 \theta \quad (5.38)$$

The equation has been non-dimensionalised by using non-dimensional variables as listed in “Table 2.1”.

The corresponding initial and boundary conditions are:

$$\text{For } \tau \leq 0: \quad \theta = U = V = 0 \quad \text{for whole space}$$

$$\text{For } \tau > 0: \quad U = V = 0 \quad \text{at all walls and in solid region} \quad (5.39)$$

^{**} The results presented here were also published as **Liaqat and Baytaş (2000a)**

For $\tau > 0$: $\theta = 0$ at $X = 0, 1.1$ and $Y = 0, 1.1$

At the interface the temperature and the heat flux must be continuous. The latter condition could be expressed as:

$$\left(\frac{\partial \theta}{\partial n}\right)_{fluid} = \frac{k_s}{k_f} \left(\frac{\partial \theta}{\partial n}\right)_{wall} \quad (5.40)$$

where n is in X or Y direction (normal to the interface).

Local Nusselt number has been defined by the following expression:

$$Nu = \left| \frac{\partial \theta}{\partial n} \right|_w \quad (5.41)$$

The average Nusselt number has been defined as follows:

$$\overline{Nu} = \int_{0.05}^{1.05} \left| \frac{\partial \theta}{\partial n} \right|_w ds \quad (5.42)$$

where n denotes the X or Y direction (normal to the surface) and ds is differential segment along the surface.

The complete conjugate problem is governed by five dimensionless variable parameters. These are the Rayleigh number Ra , the Prandtl number Pr , the dimensionless wall thickness b/D , the conductivity ratio K and thermal diffusivity ratio A . For the present analysis Pr number was taken fixed as 7.0 while Ra number was varied from 10^7 to 10^{12} . Dimensionless thickness ratio was kept constant as 0.05. For the present analysis three separate cases were studied depending upon the values of conductivity ratio and thermal diffusivity ratio. "Table 5.4" summarises these cases and respective variable values. Case 2 was considered to represent the stainless steel (SS-304) wall material and water as inside fluid. Properties for these materials were taken to be constant at 20°C .

Table 5.4

Summary of different cases analysed for the conjugate Problem

	K	A	Max. Ra No. considered	Comments
Case 1	1.0	1.0	1.0×10^9	Fluid and solid are same
Case 2	21.0	24.0	1.0×10^{11}	Solid = Stainless steel
Case 3	∞	---	1.0×10^{12}	Fluid = Water Non-conjugate
Problem				

During present analysis of the conjugate problem different cases and respective maximum Ra number considered are summarised in “Table 5.4”. Benchmark solution of present software results for a cavity having a single thick, finite conducting wall at its right are compared with the results from **Kaminski and Prakash (1986)** in “Table 5.5”. Both results show a very good agreement.

For $Ra = 10^7$ isotherms and streamlines obtained for the three cases are shown in “Figure 5.14”. A low temperature dip near the upper left corner is visible in “Figure 5.14(a)”; this dip varies in position and magnitude with time creating oscillations in the flow and temperature fields. At $Ra = 10^7$, most developed flow pattern is obtained for case 3 as is shown in “Figures 5.14(c)”. Maximum dimensionless temperature inside the enclosure is obtained for case 1 (“Figure 5.14(a)”), advocating poor heat transfer from the containing walls. Whereas the lowest value of $\theta_{\max}$ is obtained for case 2 (“Figure 5.14(b)”), indicating the efficient heat conduction from the solid walls. Streamlines in “Figure 5.14” also declare the same flow and heat transfer behaviour as discussed above.

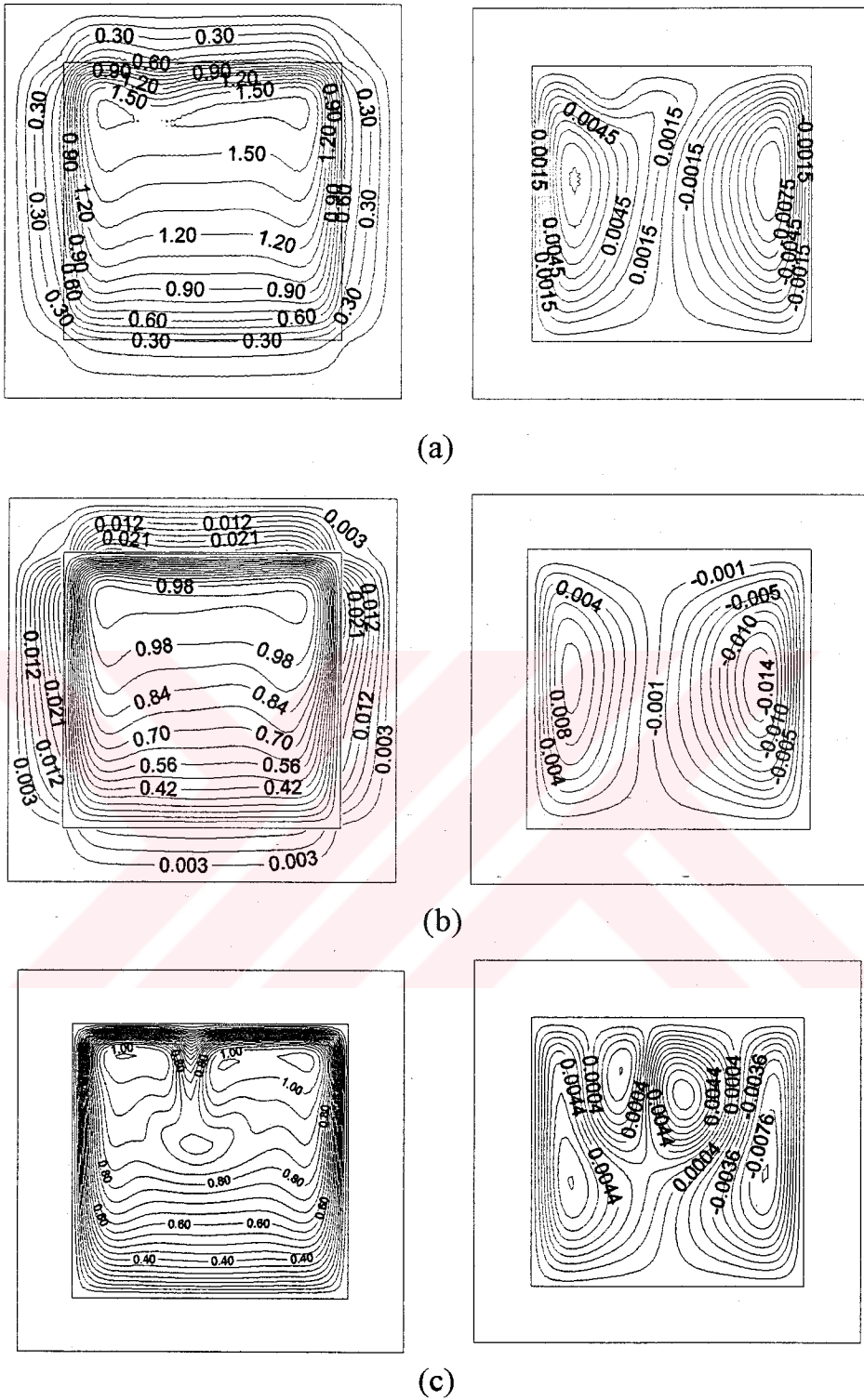


Figure 5.14 Isotherms and streamlines for $Ra = 10^7$
 (a) case 1, (b) case 2, (c) case 3

“Figure 5.15” shows the isotherms and streamlines at $Ra = 5.0 \times 10^{10}$ for the case 2 and case 3. Temperatures in “Figures 5.15(a) and 5.15(b)” are stratified in lower half of the enclosure. The flow is mainly located in the upper half of the enclosure in “Figure 5.15(b)”. Thus maintaining the temperature stratification in the lower half of the “Figure 5.15(b)”.

In “Figure 5.16”, temperature distribution across the enclosure at $Y = 0.55$ for $Ra = 10^8$ has been illustrated. Much higher temperatures are obtained for case 1, while lowest temperatures are obtained for case 3. This indicates the difference of the heat transfer across the solid walls for the two cases. There is a large temperature difference across the wall for case 1, which reduces the heat flow through the wall. Temperature variation at interface is smooth for case 1 while it shows a significant change for case 2 and 3. This is due to the difference of conductivity ratios for these cases.

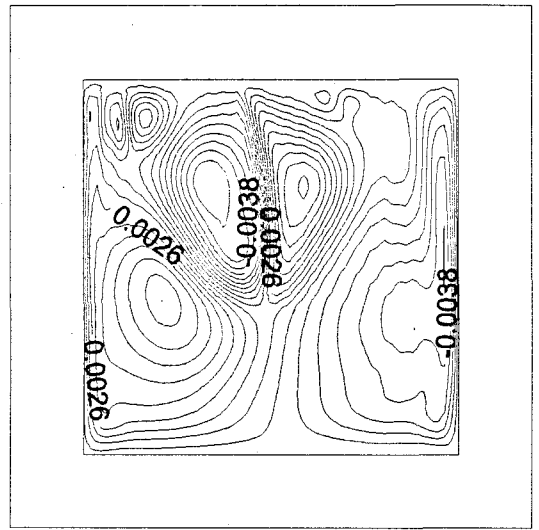
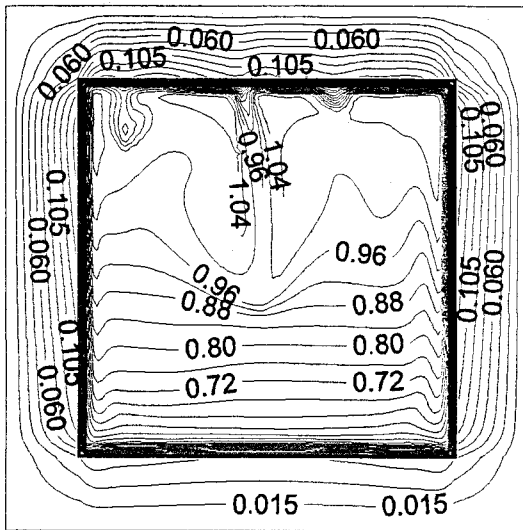
For case 1, interface temperature distribution along the four walls for $Ra = 10^8$ have been shown in “Figure 5.17”. Maximum temperature is obtained along the top wall while lowest temperature is obtained along the bottom wall. For side walls the temperature varies from that of bottom wall at lower end up to that of the top wall at the upper end. A dip in the top interface temperature indicates the position of the descending cold lobe of the fluid.

Vertical velocity profiles at $Y = 0.55$ are shown in “Figure 5.18” for all three cases, for $Ra = 10^8$. Velocity distributions clearly show the descending boundary layers along the side walls. Maximum descending velocities along the side walls are obtained for case 2. Thus indicating the efficiency of the heat extracted by the solid walls from the descending fluid.

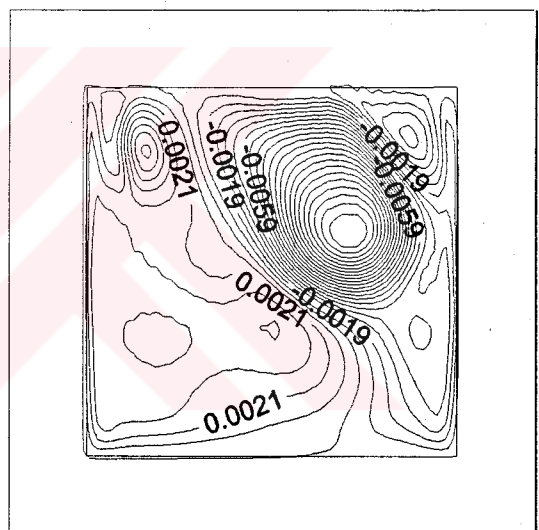
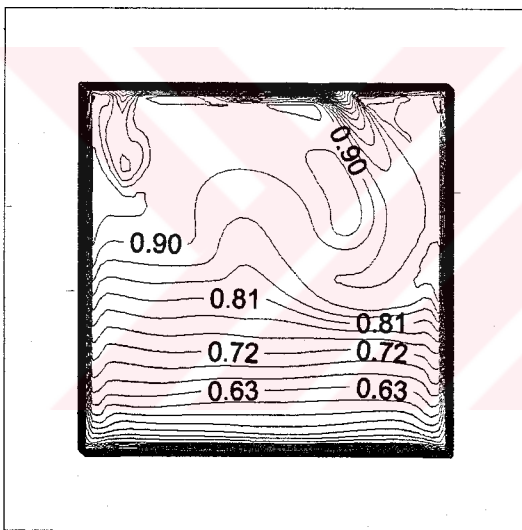
Local top wall Nusselt numbers at $Ra = 10^8$ for the 3 cases considered are shown in “Figure 5.19”. Maximum Nu is obtained for case 2, indicating sharp wall temperature gradient for this case. $\overline{Nu}$ values for the simulations performed during present analysis are summarised in “Table 5.6”. As expected, $\overline{Nu}$ increases as Ra number is increased for all cases.

Transient conjugate free convection analysis for a square enclosure having thick walls has been performed. Results indicate a strong effect of the wall conduction for thick walled enclosure. It has been shown that a higher value of diffusivity ratio

(a)



(a)



(b)

Figure 5.15 Isotherms and streamlines for $Ra = 5.0 \times 10^{10}$
(a) case 2, (b) case 3

Table 5.5

Benchmark solution for conjugate Problem

Gr	K	Kaminski et al. (1986)	Present
		$\overline{Nu}$	$\overline{Nu}$
1×10^3	1.0	0.87	0.877
	∞	1.06	1.066
1×10^5	1.0	2.08	2.082
	∞	4.08	4.122
1×10^6	1.0	2.87	2.843
	∞	7.99	8.066

plays a significant role in cooling the fluid contained in the enclosure along with the conductivity ratio. Realistic values of conductivity ratio and thermal diffusivity ratio resulted in much lower fluid temperatures even when compared with very high conductivity ratio (∞) (cases 2 and 3, respectively). Flow patterns and isotherms for conjugate analysis show a great difference from that of conventional non-conjugate solutions reported in the open literature.

The main conclusion reached in the present study is about the importance of the conjugate analysis of the thick walled problems as it may give qualitatively different results from non-conjugate analysis.

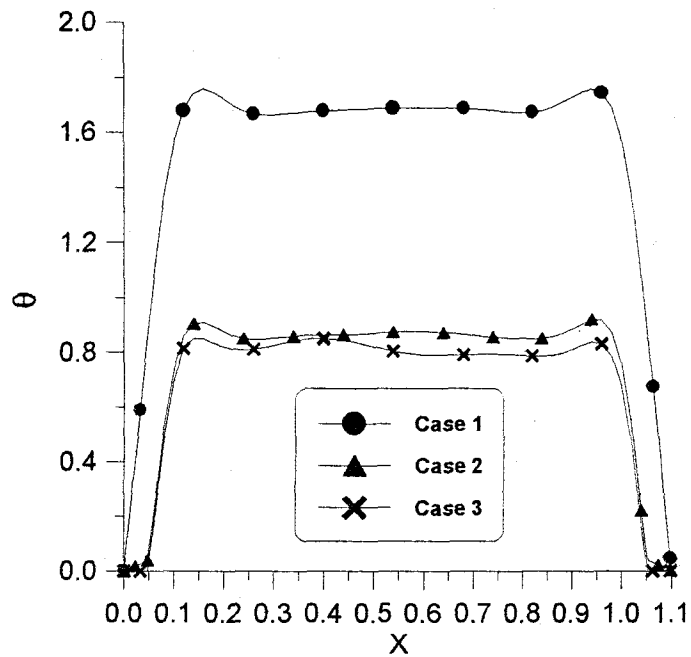


Figure 5.16 Temperature distributions across the enclosure at $Y = 0.55$ for $Ra = 10^8$

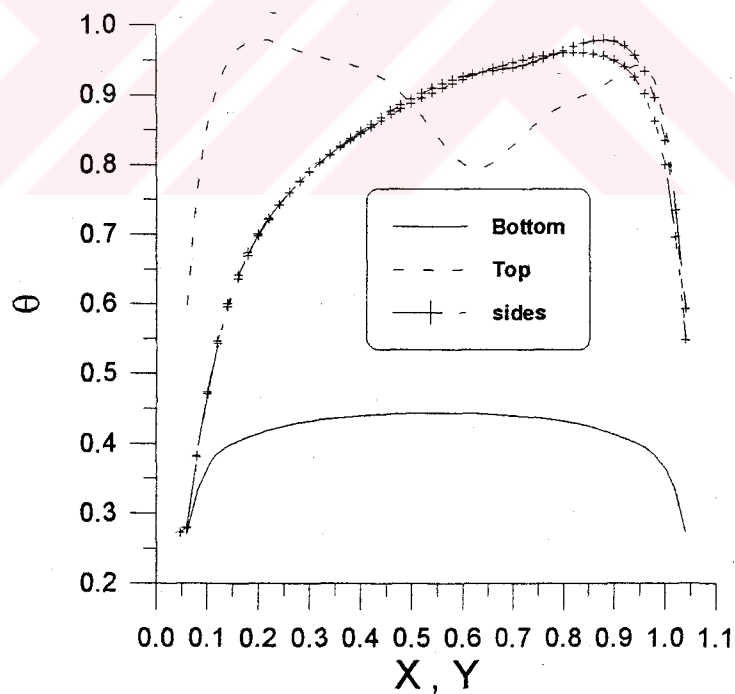


Figure 5.17 Variation of interface temperature along the enclosure walls for $Ra = 10^8$, $A = 1.0$, $K = 1.0$

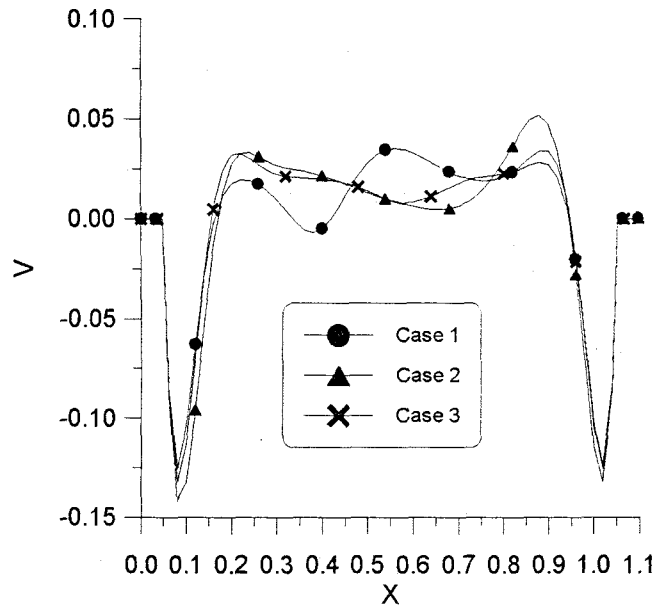


Figure 5.18 Variation of V with X Co-ordinate at $Y = 0.55$ for $Ra = 10^8$, for three cases

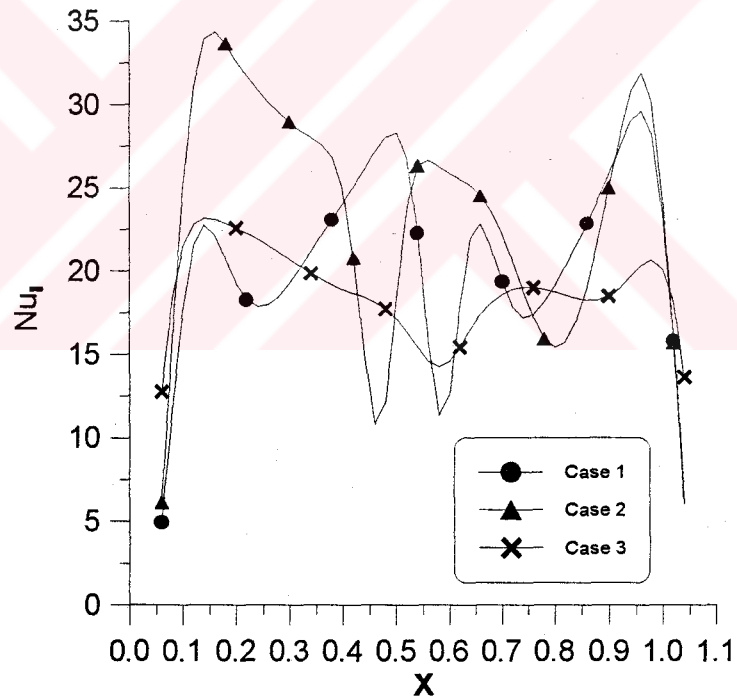


Figure 5.19 Distribution of Nu at the top wall for the three cases for $Ra = 10^8$

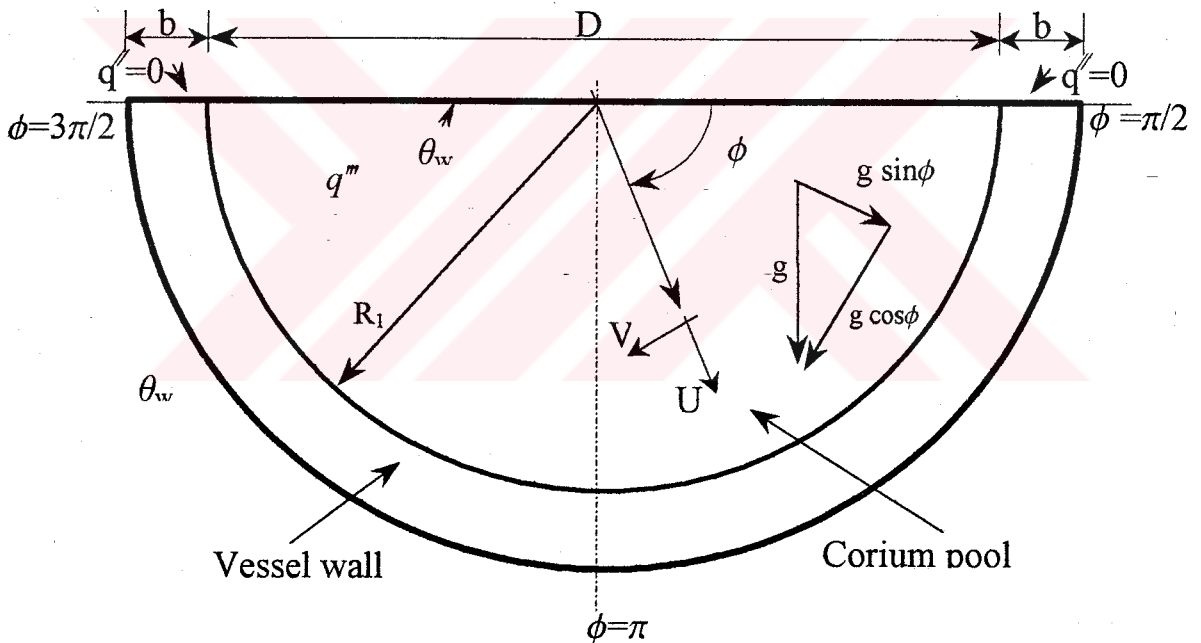
Table 5.6

Summary of final results for conjugate analysis

Ra	K	$\overline{Nu}$		
		Top	Sides	Bottom
1.0×10^7	1.0	12.37	10.43	5.67
	21.0	12.53	10.70	4.87
	∞	13.53	10.29	4.80
1.0×10^8	1.0	21.08	18.47	9.28
	21.0	22.31	18.06	6.02
	∞	23.14	17.56	5.87
1.0×10^9	1.0	35.92	35.72	23.89
	21.0	42.57	33.05	7.43
	∞	33.61	40.30	7.47
1.0×10^{10}	21.0	76.61	61.84	12.43
	∞	76.73	63.08	12.49
5.0×10^{10}	21.0	103.42	95.03	30.32
	∞	108.76	97.12	33.68
1.0×10^{11}	21.0	123.99	108.55	46.06
	∞	123.82	113.93	54.66
1.0×10^{12}	∞	187.48	183.80	146.62

5.6 Comparison of Conjugate and Non-conjugate Analysis for Semi-circular Geometry

In almost all of the studies available in the open literature, walls enclosing the pool are assumed to be isothermal, thus neglecting the conduction through the walls. In actual situation pressure vessel wall is exceptionally thick and the effect of the conduction on the natural convection flow inside the pool should be analysed. Aim of present study is to investigate the effect of conduction through the thick bottom wall of the pressure vessel on the natural convection flow in a semicircular cavity. Pr number has been taken 7.0 for benchmarking in this study since previous experimental and numerical studies are available in open literature for this Pr number. The schematic diagram of the problem has been shown in “Figures 2.1 and

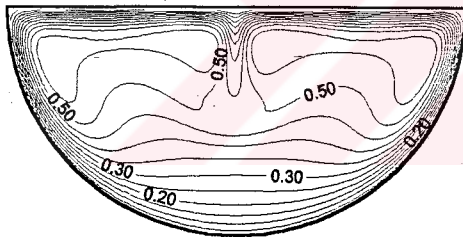


5.20” with coordinate layout.

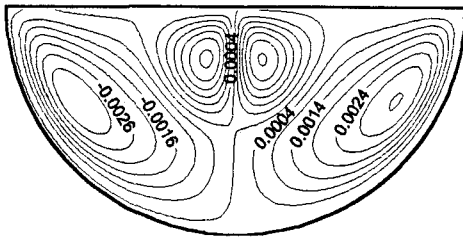
Figure 5.20 Schematic diagram of the thick walled semi-circular cavity and Co-ordinate system

Temperature distribution within the solid wall and fluid body has been obtained for Ra number from 3.2×10^7 to 3.2×10^{11} . In the following section, isotherms, streamlines, local and average Nusselt numbers for different Ra numbers will be discussed. Isotherms and streamlines for $Ra = 3.2 \times 10^8$ are presented in “Figures 5.21 and 5.22”. Both conjugate, for $K = 5$, $A = 5$, and non-conjugate systems show

balanced temperature and velocity fields while temperatures are stratified in lower half of the cavity with strong boundary layers along the top and bottom walls. But fluid temperatures for the conjugate case are higher as compared to the non-conjugate isotherms (“Figure, 5.22(a)”). These higher temperatures, in case of conjugate analysis, are translated into higher velocities as could be observed in “Figures. 5.21(b) and 5.22(b)”. Thus the thick solid wall acts as a barrier against heat transfer due to low temperature gradients across it. The phenomenon discussed above is also evident from “Figures 5.23 and 5.24” prepared for $Ra = 3.2 \times 10^{10}$ for a quasi-steady state. Here the resistance of the bottom wall to heat conduction is much more prominent. Again the cavity temperatures and the velocities are higher for conjugate case. Isotherms in “Figure 5.23(a)”, for the non-conjugate case, are stratified in lower half of the cavity but this temperature stratification has been reduced in “Figure 5.24(a)”, for the conjugate case. Thus temperature distribution inside the cavity is much more uniform for the conjugate case as compared to the cavity with isothermal walls. This is achieved through the significant mixing effect of the velocity field as is clear from the streamlines in “Figure 5.23(b)” and “Figure 5.24(b)”.

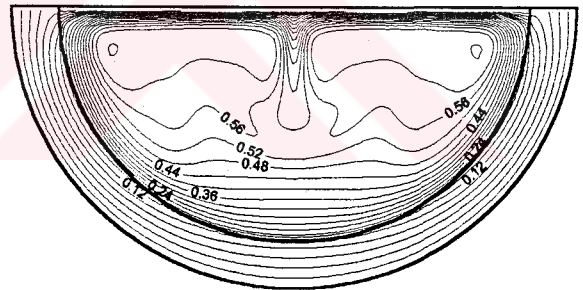


(a)

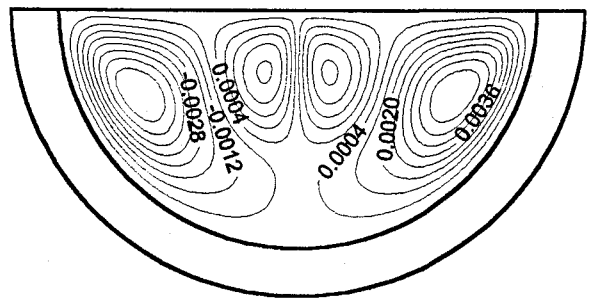


(b)

Figure 5.21 Isotherms and streamlines for non-conjugate analysis for $Ra = 3.2 \times 10^8$



(a)

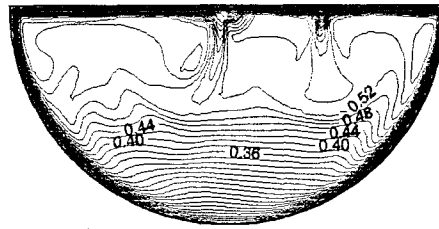


(b)

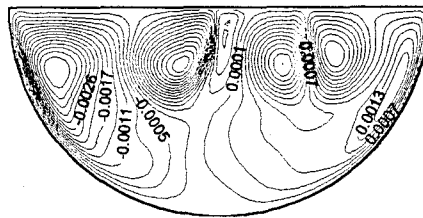
Figure 5.22 Isotherms and for conjugate analysis for $Ra = 3.2 \times 10^8$, $A = 5$, $K = 5$

Local Nusselt number distribution along the bottom wall for $Ra = 3.2 \times 10^{10}$ has been shown in “Figure 5.25” both for conjugate and non-conjugate cavities. Maximum Local Nusselt number is about 36% lower for the conjugate case while minimum value is almost same for both cases in “Figure 5.25”. Average Nusselt numbers (time averaged over quasi-steady state flow for $Ra > 10^7$) for top and bottom walls is shown graphically in “Figure 5.26”, both for conjugate and non-conjugate cases. Top wall average Nusselt number for conjugate analysis, shows an increase of about 23% from that of non-conjugate cavity (“Figure 5.26(a)”). On the other hand bottom wall average Nusselt number decreases by 11% for the conjugate case and this effect is more prominent at higher Ra numbers as could be observed in “Figure 5.26(b)”. Time Averaged $\overline{Nu}$ numbers for the top and bottom wall for a conjugate analysis with $K = 23$, $A = 23$ has also been drawn in “Figure 5.26”. It is clear from the “Figure 5.26” that at such higher value of K and A bottom wall behaves much similar to conventional isothermal wall.

Effect of thick bottom enclosing wall on the heat and fluid flow parameters in a semicircular cavity has been analysed numerically. Average fluid temperatures in case of conjugate cavity are observed to be 11% higher as compared to non-conjugate cavity with a similar increase in fluid velocities. Average top wall Nusselt number is 23% higher while average bottom Nusselt number is 11% lower for the conjugate analysis. Thus due to thick finite conducting wall enclosing the cavity from the bottom heat flow is redirected towards the top wall. This effect is more prominent for lower solid to fluid conductivity ratios.

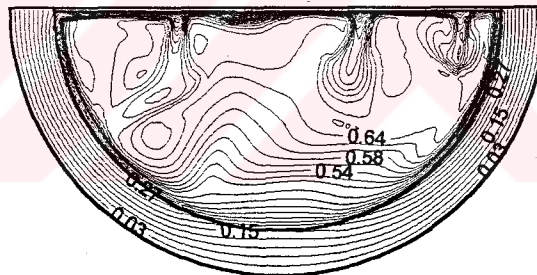


(a)

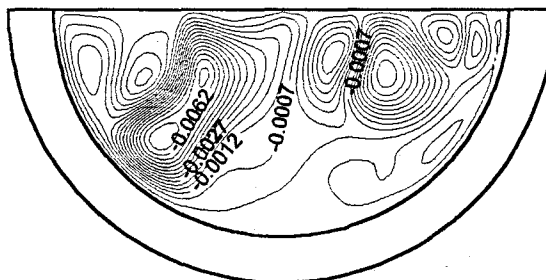


(b)

Figure 5.23 (a) Isotherms and (b) streamlines for non-conjugate analysis for $Ra = 3.2 \times 10^{10}$

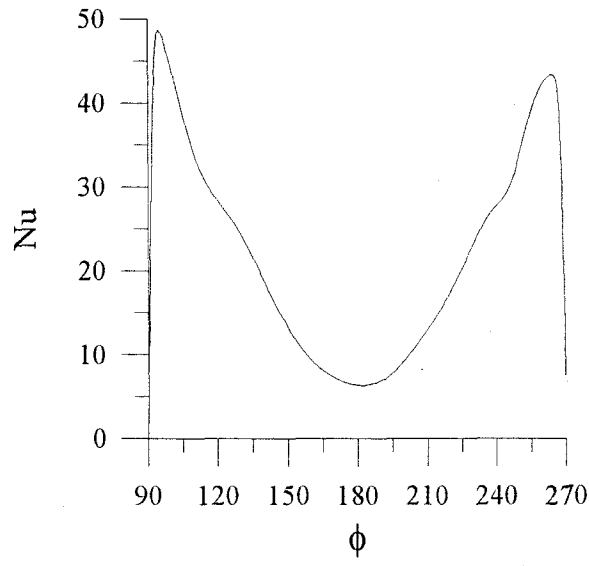


(a)

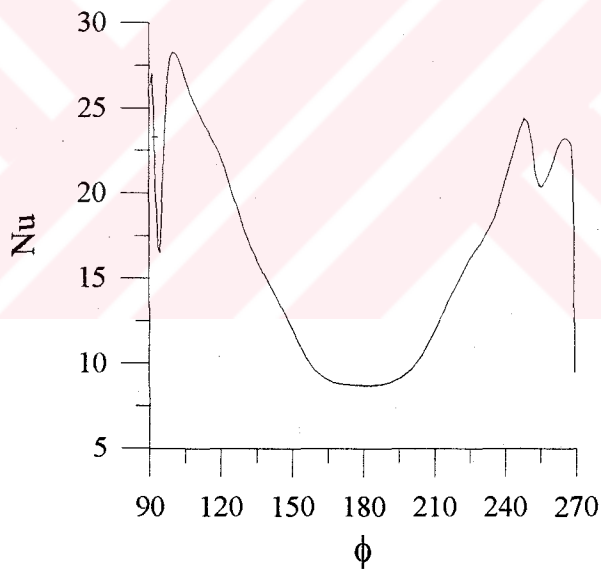


(b)

Figure 5.24 (a) Isotherms and (b) streamlines for conjugate analysis for $Ra = 3.2 \times 10^{10}$, $A = 5$, $K = 5$

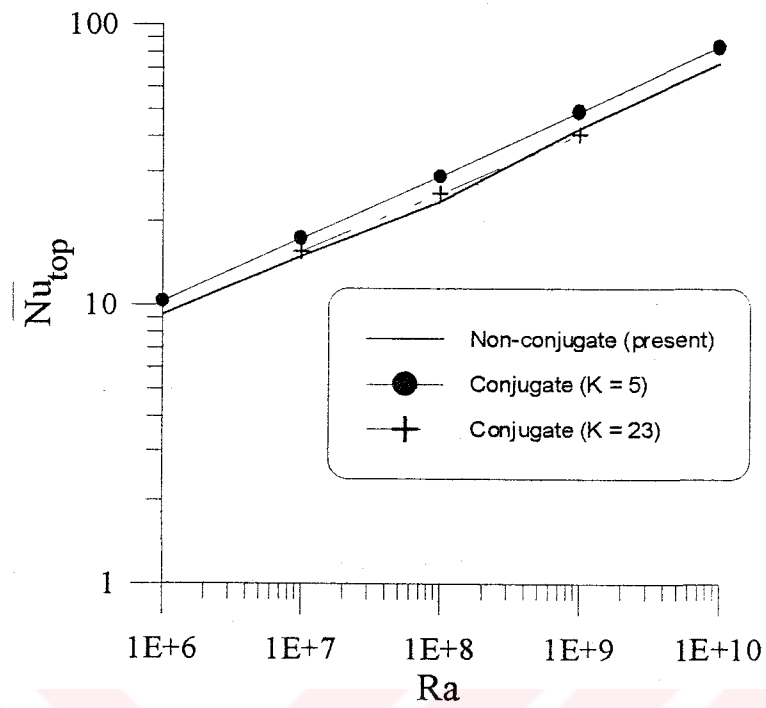


(a)

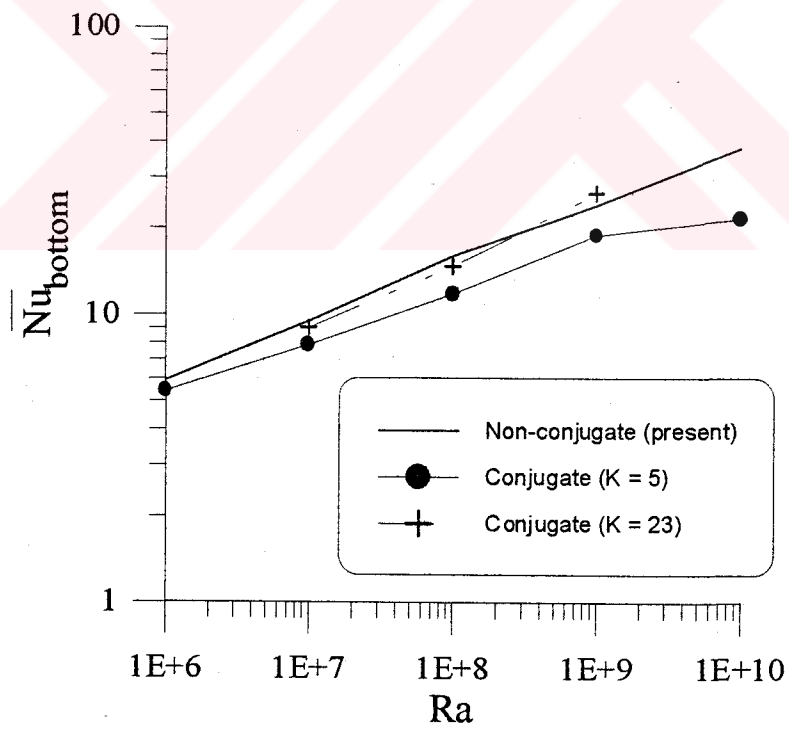


(b)

Figure 5.25 Variation of the local Nusselt number along the solid-liquid interface for $Ra = 3.2 \times 10^{10}$ (a) Non-conjugate (b) Conjugate for $A = 5$, $K = 5$



(a)



(b)

Figure 5.26 variation of $\overline{Nu}$ with Ra number for conjugate and non-conjugate cases (a) Top wall (b) Bottom wall

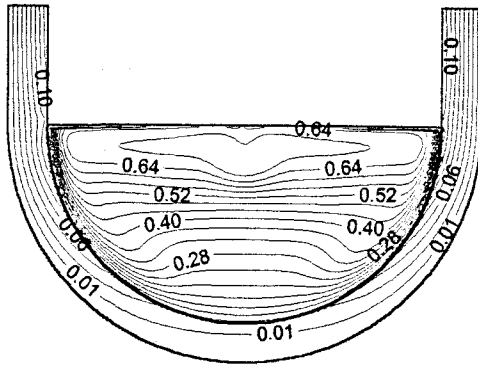
5.7 Analysis of Corium Contained in the Lower Plenum** -a Composite Study

Present study is a composite solution including all three heat transfer processes of conduction, convection and radiation. Dimensionless equations governing the flow and heat transfer for the present case have been listed in section 2.3, in detail. “Figures 2.4 and 2.5” show all the details of the cavity with containing walls, surfaces participating in the radiative heat transfer and Co-ordinate system. Main focus of present study is to analyse the effect of corium pool surface radiation upon the conjugate free convection flow within the pool body. The parameters included in the analysis and their ranges considered here, are as follows: ϵ_u and ϵ_w were kept constant as 0.5 and ϵ_p was varied from 0.2 to 0.8. N_{RC} and θ_u were 5.0×10^{-2} and 0.3, respectively. $\Delta T/T_w$ is varied from 1.0 to 7.0. Material properties for corium and RPV shell were taken from **Park and Dhir (1992)**. Fluid Pr number was 0.43 while solid to fluid conductivity was 13.90. The effect of various parameters has been discussed below.

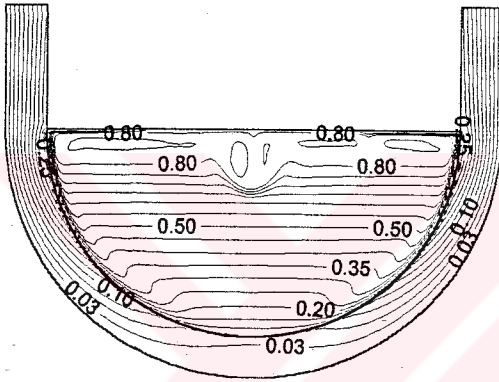
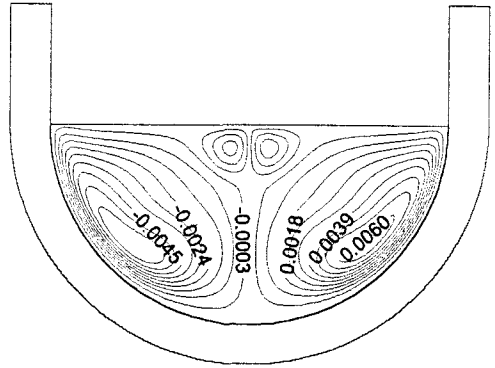
5.7.1 Rayleigh Number

Representative isotherms and streamlines, of the fluctuating flow, for $Ra = 3.2 \times 10^8$, 3.2×10^{10} and 3.2×10^{12} are shown in “Figure 5.27”. The flow development with increasing Ra number is clearly visible in “Figure 5.27(a)”, “Figure 5.27(b)” and “Figure 5.27(c)”, respectively, with non-periodic changes in flow field. For $Ra = 3.2 \times 10^8$ in “Figure 5.27(a)”, the convective flow is spread all over the pool body with boundary layer along the solid wall. The pool and side wall temperatures increase with Ra number. Convective flow becomes much stronger along top surface of the corium pool as compared to interior of the pool, as Ra number is increased. This effect gives rise to a well-stratified temperature profile within the pool as could be observed in “Figure 5.27”. The bottom of the pool in “Figure 5.27(c)” shows a pure conduction region due to the negligible flow there. This effect is only visible for $Ra = 3.2 \times 10^{12}$.

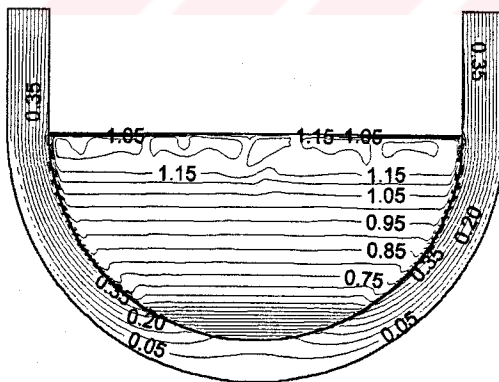
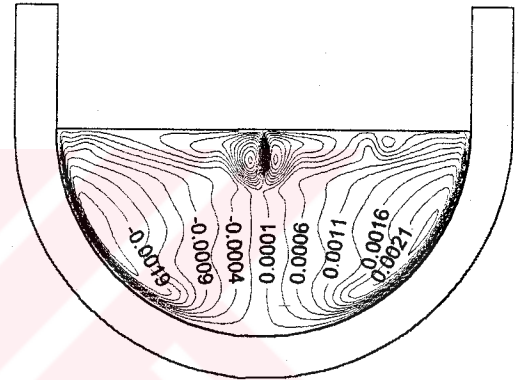
** The results presented here were also published as **Liaqat and Baytaş (2000e)**



(a)



(b)



(c)

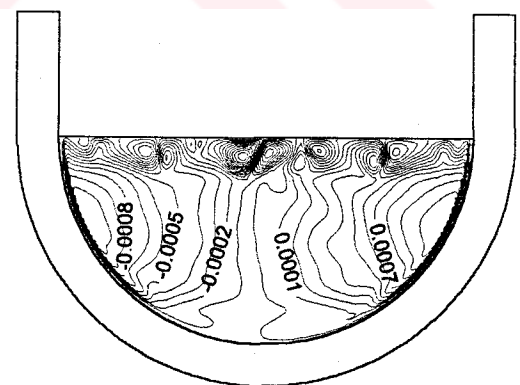


Figure 5.27 Isotherms and streamlines for $Pr = 0.43$, $N_{RC} = 5 \times 10^{-2}$, $\epsilon_P = 0.5$
 $\Delta T/T_W = 4.0$ and $\theta_u = 0.3$ (a) $Ra = 3.2 \times 10^8$, (b) $Ra = 3.2 \times 10^{10}$
(c) $Ra = 3.2 \times 10^{12}$

5.7.2 Local Nusselt Number

Local Nusselt number distribution at the solid-fluid interface for various Ra numbers is given in “Figure 5.28”. There is a drastic increase in local Nusselt number near the top edge angles while its values remain almost constant near the stagnation point. Thus ratio of maximum Nu to its minimum value increases with increase in Ra number. Local Nusselt number variation for $Ra = 3.2 \times 10^{12}$, clearly indicates the convection dominated region of the pool ie. $\phi = 0$ to 10 and $\phi = 170$ to 180 .

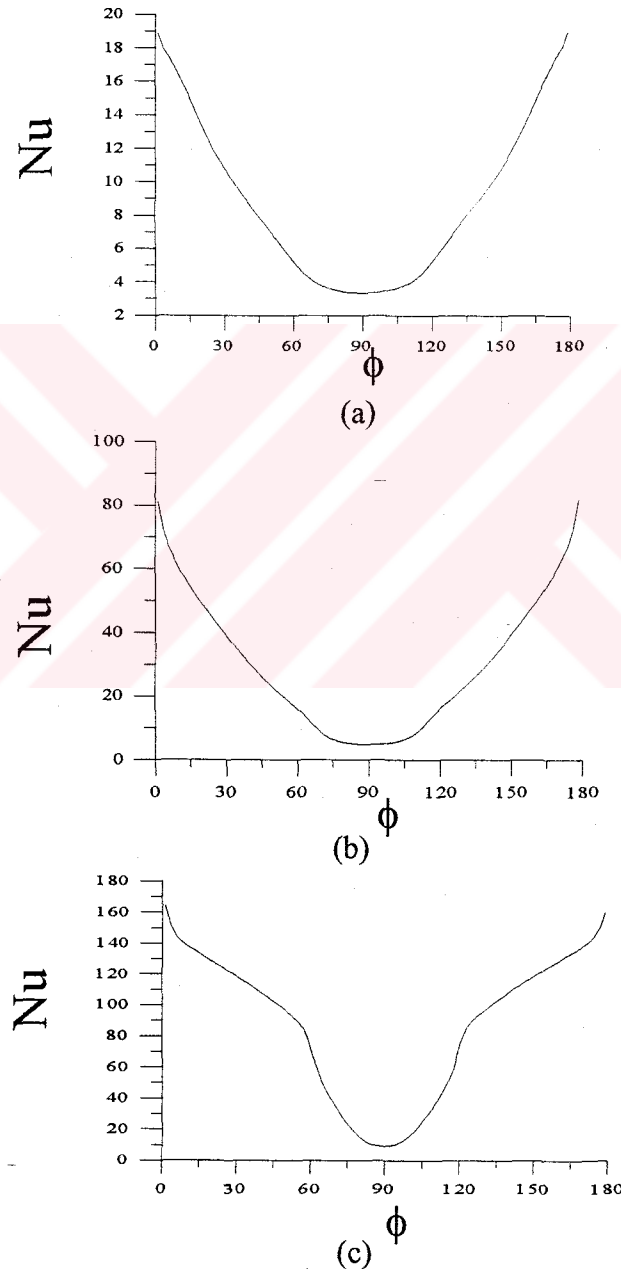


Figure 5.28 variation of local Nusselt number at the solid-fluid interface with ϕ for $N_{RC} = 5 \times 10^{-2}$, $\epsilon_P = 0.5$, $\Delta T/T_W = 4.0$ and $\theta_u = 0.3$ (a) $Ra = 3.2 \times 10^8$, (b) $Ra = 3.2 \times 10^{10}$, (c) $Ra = 3.2 \times 10^{12}$

5.7.3 Effect of $\Delta T/T_w$

The range of $\Delta T/T_w$ from 1.0 to 7.0 was analysed. Isotherms and streamlines for $\Delta T/T_w = 1.0, 4.0$ and 7.0 have been presented in “Figure 5.29” for $Ra = 3.2 \times 10^{10}$. Radiative heat transfer from the pool surface increases with $\Delta T/T_w$ which results in increased side wall temperatures. This increased cooling from the top increases the convective flow along the top surface as could be seen in streamlines given in “Figure 5.29”. Evidently, maximum pool temperature decreases as $\Delta T/T_w$ increased in “Figure 5.29”.

5.7.4 Effect of Pool Surface Emissivity

The effect of cooling from the pool surface depends upon its emissivity. This effect of variation in pool surface emissivity was studied by taking all other parameters as constant. “Figure 5.30” shows the decrease in the maximum pool temperature with increase in ϵ_p . Also solid-fluid interface average Nusselt number decreases with increase in pool emissivity as shown in “Figure 5.31”. Both $\theta_{\max}$ and time averaged $\overline{Nu}_{bottom}$ show a sharp decrease if the value of ϵ_p is increased from 0.2 to 0.5.

5.7.5 Average Nusselt Number

Finally, “Figure 5.32” illustrates the variation of the time averaged $\overline{Nu}_{bottom}$ versus Ra number from 10^8 to 10^{13} . As expected, Nusselt number increases as Ra number is increased.

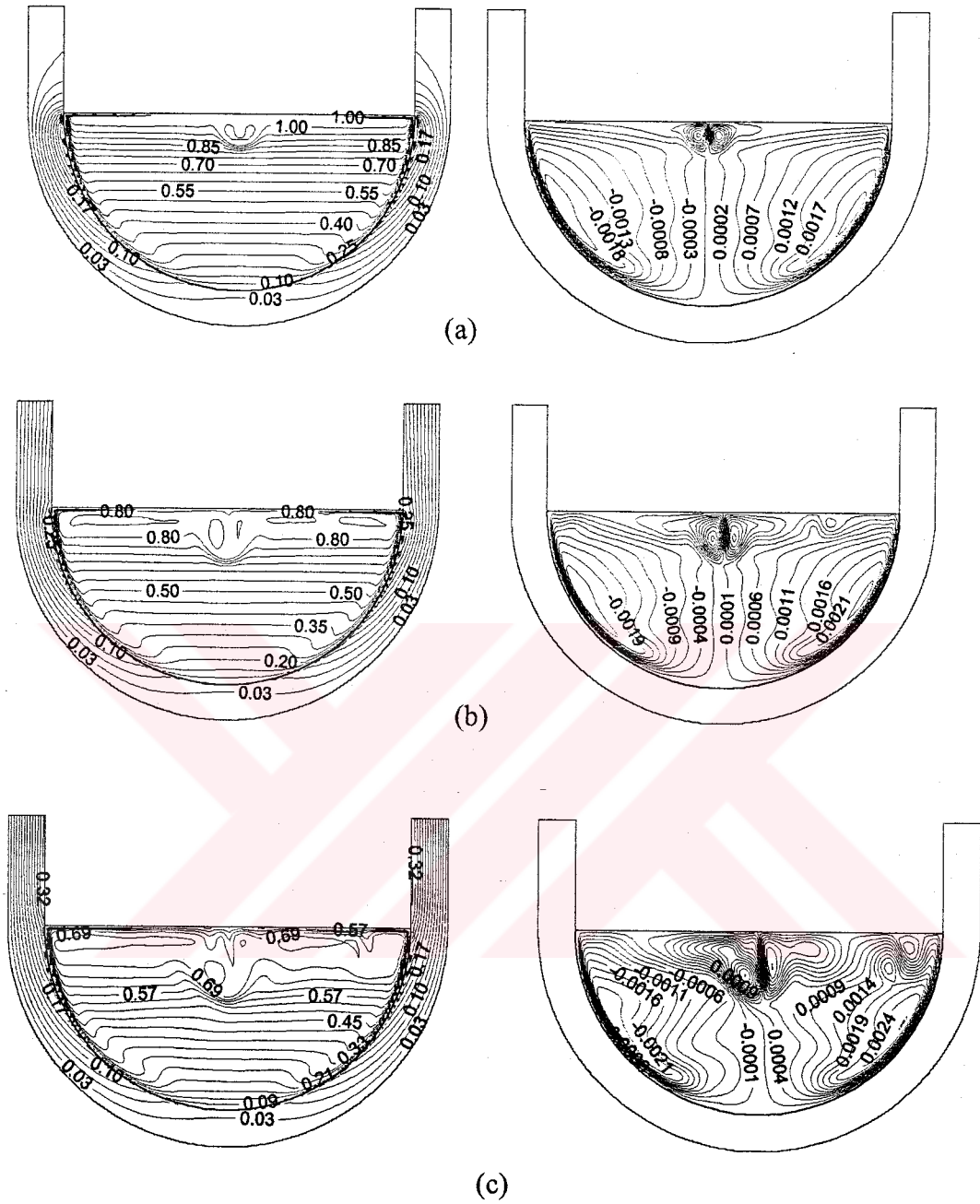


Figure 5.29 Development of isotherms and streamlines with increase in $\Delta T/T_w$ for $Pr = 0.43$, $N_{RC} = 5 \times 10^{-2}$, $\epsilon_p = 0.5$, $\theta_u = 0.3$ and $Ra = 3.2 \times 10^{10}$ (a) $\Delta T/T_w = 1.0$, (b) $\Delta T/T_w = 4.0$ (c) $\Delta T/T_w = 7.0$.

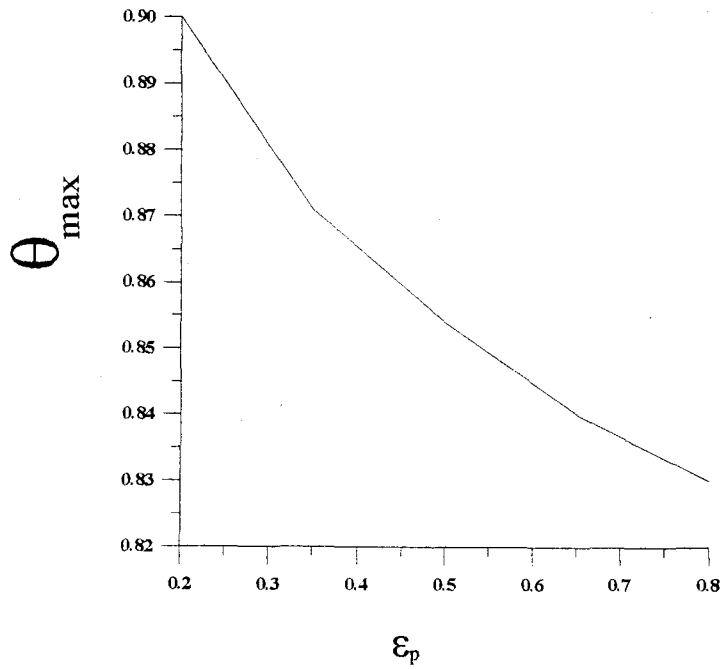


Figure 5.30 Variation of maximum pool temperature with increase in pool surface emissivity for $N_{RC} = 5 \times 10^{-2}$, $\Delta T/T_w = 4.0$, $\theta_u = 0.3$ and $Ra = 3.2 \times 10^{10}$

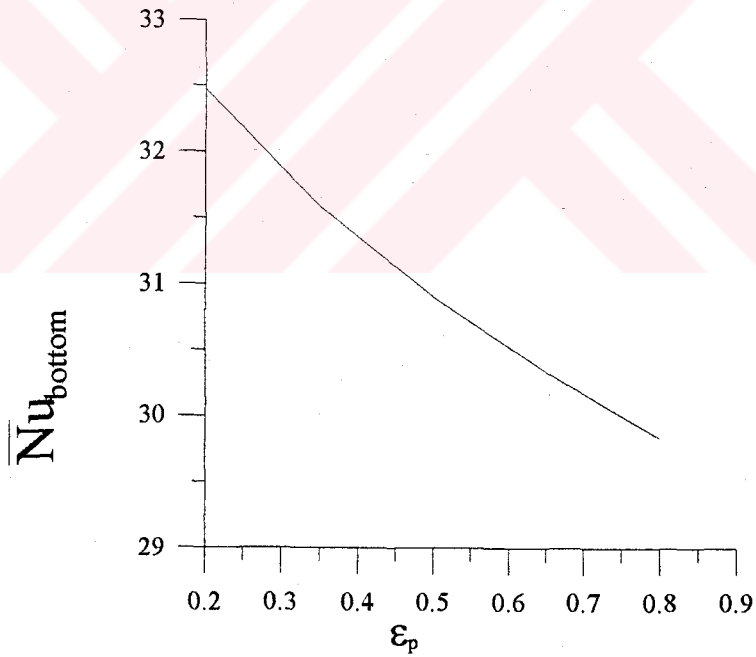


Figure 5.31 Solid-fluid interface average Nusselt number versus pool surface emissivity for $N_{RC} = 5 \times 10^{-2}$, $\Delta T/T_w = 4.0$, $\theta_u = 0.3$ and $Ra = 3.2 \times 10^{10}$

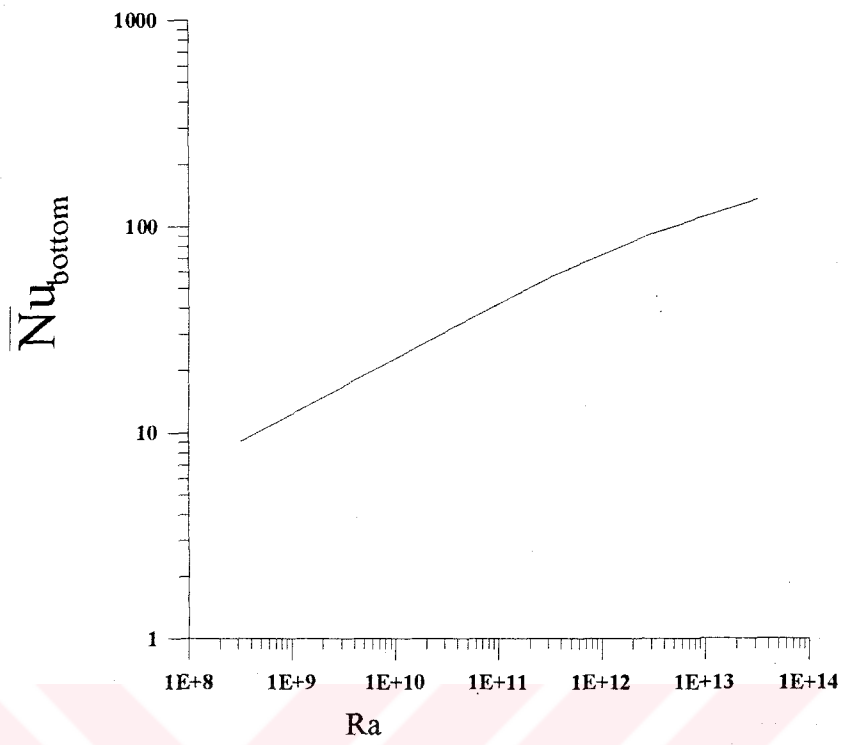


Figure 5.32 Solid-fluid interface average Nusselt number versus Ra number for $N_{RC} = 5 \times 10^{-2}$, $\epsilon_p = 0.5$, $\theta_u = 0.3$ and $\Delta T/T_w = 4.0$

6. CONCLUSIONS

In present doctoral study the thermal hydraulic analysis of the corium contained inside lower head of the PWR pressure vessel lower head was carried out. In the course, it became essential to analyse the buoyant flow of heat generating fluid in square, hemi-spherical and semi-circular Co-ordinates. As a first step, the numerical methodology adopted here was cross-checked using previously published results from open literature in Cartesian and polar Co-ordinates both for discrete heating and with internal sources. Main conclusions reached during this study can be summarised as given below.

Natural convection flow inside discretely heated square cavity was analysed to compare the numerical results obtained by using the software developed presently. This geometry was chosen due to the fact that this problem is standard benchmark problem for buoyant flow and has been tackled by many well-known researchers in this field. Present results compare very well with the published results, thus verifying the accuracy of the present methodology.

Flow field results for the internally heated square cavity were also compared with previously published results. It was obtained that both results for flow field and heat transfer compared very well.

Lower head of RPV has a hemi-spherical shape. 2-D assumption for the flow field only allowed half of the hemi-spherical bowl to be analysed due to symmetric geometry. Thus the flow field was subjected to an additional undue constraint. Although final results for the average Nusselt number were comparable to experimental data from open literature but the flow field obtained could not exhibit the non-symmetric behaviour anticipated for the actual situation.

Only option to look at the non-symmetric behaviour of the fluctuating flow was offered by the 2-D semi-circular geometry as used by many researchers in the past. Semi-circular geometry with isothermal walls was used for benchmarking purpose.

Final results for the average Nusselt number, streamlines and pressure distribution were in very good agreement with the results obtained from open literature.

Conjugate free convection analysis for a square cavity, having thick enclosing walls with finite conductivities, was performed. It was shown that higher value of diffusivity ratio plays a significant role in cooling the fluid contained in the enclosure along with conductivity ratio. Flow patterns and isotherms for conjugate analysis showed a great difference from that of conventional non-conjugate solutions reported in open literature. Main conclusion reached in this study was about the importance of the conjugate analysis of the thick walled problems as it may give qualitatively different results from non-conjugate analysis.

Effect of thick bottom enclosing wall on the heat and fluid flow parameters in a semi-circular cavity was analysed. Average fluid temperature in case of conjugate cavity are observed to be higher as compared to non-conjugate cavity with a similar increase in fluid velocities inside the cavity. Average top wall Nu number was higher while average bottom Nu number was lower for the conjugate analysis. Thus it was concluded that due to thick finite conducting wall, enclosing from the bottom, heat flow is redirected towards the top wall. This effect is more prominent for lower solid to fluid conductivity ratio.

A transient analysis of a semi-circular pool, enclosed by a thick conducting wall at the bottom, containing molten core material was carried out for Ra number range of 3.2×10^7 to 3.2×10^{12} . System reached steady state for $Ra = 3.2 \times 10^7$. However isotherms and streamlines for all higher Ra number showed a non-periodic behaviour. Solid-fluid interface average Nusselt number was always less than the top wall average Nusselt number.

Main aim of the present study was to analyse the hemi-spherical pool, containing molten core material numerically, to obtain the fluid flow and heat transfer parameters for 2-D cylindrical (r- ϕ) Co-ordinate system. Due to exceptional thick pressure vessel wall a conjugate analysis has been envisaged as essential. Temperature of the corium inside the pool will reach to such a high value that radiation heat transfer becomes an important factor. Thus radiation heat transfer from the pool surface has been included in the present study. This author could not find a study in open literature that has dealt with convection, radiation and conjugate heat

transfer, simultaneously, to be encountered in actual situation. The nearest study to the present simulation, was reported by **Park et al. (1992)**. They investigated the effect of outside cooling on the thermal behaviour of the pressurised water reactor vessel's lower head. Their study included the radiation heat transfer from the pool surface but they did not solve the flow equations within the pool body. Thus, interaction of the surface radiation with free convection flow inside the pool remains to be investigated. A detailed analysis of interaction of surface radiation and wall conduction with natural convection inside the semi-circular cavity has been performed. Special emphasis has been upon the parameters, which govern the radiative heat transfer from the pool surface. The following is a brief summary of the main findings of this study.

- (1) For high Ra number, temperatures in most of the pool are well stratified with convection dominated region confined to upper most of the pool.
- (2) Because of temperature stratification in the lower part of the cavity, the highest Nu for solid-fluid interface occurs just below the corners of the semi-cylindrical cavity.
- (3) Radiative heat transfer from the pool surface increases with increase in $\Delta T/T_w$ with corresponding decrease in the maximum pool temperature. If $\Delta T/T_w$ is larger than 6, radiation heat transfer from the pool surface is more efficient as compared to convection towards the bottom wall. That is why the side walls are heated to higher temperature as compared to bottom circular wall. This effect is also evident from the decrease in $\overline{Nu}_{bottom}$ for higher values of $\Delta T/T_w$.
- (4) The increase in pool surface emissivity (ϵ_p) is translated into decrease in the maximum pool temperature and $\overline{Nu}_{bottom}$.
- (5) Stagnation point (at $\phi = 180$) and the points close to it will never be in danger from thermal attack of the melted core material, as these points have the lowest temperatures in the whole geometry.
- (6) If rupture of the RPV wall occurs in any circumstances it will occur from the points just below the point of contact of the free surface of the pool and the RPV wall, as this section has the highest temperatures in whole of the wall.

- (7) For higher Ra numbers, the attack on the RPV wall increases simultaneously within 15 degrees angle from the point of contact of the free surface and the RPV wall.
- (8) The distinguished finding of the present analysis is the appearance of conduction only region close to stagnation point for Ra number higher than 3.2×10^{12} .
- (9) The non-periodic fluctuating, convection dominated regime is confined to upper 10% of the cavity.
- (10) For low Ra number the flow is symmetric along the symmetry line but this symmetry is lost for higher Ra numbers.
- (11) Depth of the convection dominated region increases with increase in the parameter $\Delta T/T_w$. Thus radiative cooling effect from the pool surface increases with $\Delta T/T_w$.
- (12) Ratio of the maximum local Nu number along the solid-fluid interface to its minimum value increases with increase in Ra number.
- (13) Solid-fluid interface local Nusselt number at and close to the stagnation point remains almost same for any Ra number.

Thus a detail analysis of buoyant flow within a corium contained in RPV lower head has been completed with a special emphasis upon the parameters which control the radiative heat transfer from the pool surface. It is hoped that this study will greatly increase the understanding of the thermal hydraulic phenomenon occurring within the corium pool.

REFERENCES

- Arpaci, S. V. and Larson, S. P., 1984. Convection heat transfer, Prentice Hall International, London.
- Asfia, F. J. and Dhir, V. K., 1996. An experimental study of natural convection in a volumetrically heated spherical pool bounded on top with a rigid wall, *Nucl. Eng. Des.*, **163**, 333-348.
- Baker, L. Jr., Faw, R. E. and Kulacki, F. A., 1976. Post accident heat removal-Part I: Heat transfer within an internally heated, non-boiling liquid layer, *Nucl. Sci. Eng.*, **6**, 222-230.
- Baytas, A. C., 1996. Buoyancy driven flow in an enclosure containing time periodic internal sources, *Heat and Mass Transfer*, **31**, 113-119.
- Bejan, A., 1995. Conduction heat transfer, New York, Wiley.
- Bergholz, R. F., May 1980. Natural convection of a heat generating fluid in a closed cavity, *Trans. ASME*, **102**, 242-247.
- Chavez, S. A. and Rempe, J. L., 1994. Finite element analysis of BWR vessel and penetrations under severe accident conditions, *Nucl. Eng. Des.* **148**, 413-435.
- Chawla, T. C. and Chan, S. H., 1982. Heat transfer from vertical/inclined boundaries of heat-generating boiling pools, *J. Heat Transfer*, **104**, 465-473.
- Cheung, F. B., Haddad, K. H. and Liu, Y. C., 1999. Boundary layer boiling and critical heat flux phenomena on a downward facing hemi-spherical surface, *Nuclear Technology*, **126**, 243-264.
- Cheng, G., and El-Genk, M. S., 1997. Pool boiling from downward-facing hemispherical stainless steel surface, *Transactions of American Nuclear Society*, **77**, 427-429.
- Chu, T. T., Pilch, M. M. and Bentz, J. H., 1997. Experimental Investigation of Creep Rupture of Reactor Vessel Lower Head, *Transactions of American Nuclear Society*, **77**, 271-273.
- Dehghan, A. A. and Behnia, M., Feb. 1996. Combined natural convection-conduction and radiation heat transfer in a discretely heated open cavity, *J. Heat Transfer*, **118**, 56-64.
- Dosanjh, S. S. and Pilch, M., 1991. Lower head creep-rupture sensitivity studies, *Nucl. Sci. Eng.*, **108**, 172-183.

- Dosanjh, S. S.**, 1989. Melt propagation in dry core debris beds, *Nuclear Technology*, **88**, 30-46.
- El-Genk, M. S. and Cheng, G.**, 1999. Experiments on pool boiling of water from downward facing hemispheres, *Nuclear Technology*, **125**, 52-69.
- El-Genk, M. S. and Cheng, G.**, 1996. Pool boiling from downward-facing curved surfaces: effects of radius of curvature and edge angle, *Nuclear Technology*, **114**, 351-363.
- Emara, A. A. and Kulacki, F. A.**, August 1980. Numerical investigation of thermal convection in a heat-generating fluid layer, *J. Heat transfer*, **102**, 531-537.
- Epstein, M. and Fauske, H. K.**, 1989. The Three Mile Island unit 2 core relocation-heat transfer and mechanism, *Nuclear Technology*, **87**, 1021-1035.
- Faw, R. E. and Baker, L. Jr.**, 1976. Post accident heat removal-part II: Heat transfer from internally heated liquid to a melting solid, *Nucl. Sci. Eng.*, **61**, 231-238.
- Gabor, J. D., Ellison, P. G. and Cassulo, J. C.**, 1980, Heat transfer from internally heated hemispherical pools, Proc. 19th Nat. Heat Transfer Con., Orlando, Florida, July 27-30.
- Haraldsson, H. O., Li, H. X., Dinh, T. N., Green, J. A. and Sehgal, B R.**, 1997. Hydrodynamic fragmentation of a molten-metal jet in water: effect of melt solidification and coolant voiding, *Transactions of American Nuclear Society*, **77**, 432-433.
- Henrie, J. O.**, 1989. Timing of the Three Mile Island unit 2 core degradation as determined by forensic engineering, *Nukleer Technology*, **87**, 857-864.
- Henry, R. E., Burelbach, J. P., Hammersley, R. J., Henry, C. E. and Klopp, G. T.**, 1993. Cooling of core debris within the reactor vessel lower head, *Nuclear Technology*, **106**, 385-399.
- Hobbins, R. R., Russel, M. L., Olsen, C. S. and McCardell, R. K.**, 1989. Molten material behaviour in Three Mile Island unit 2 accident, *Nuclear Technology*, **87**, 1005-1012.
- Hofmann, S. H., Nack, G. S., Shumacher, G. and Sepold, L.**, 1994. Essential experimental results of the CORA test program on severe core damage phenomenon, *Kerntechnik*, **59**, 197-202.
- Hottel, H. C. and Sarofin, A. F.**, 1967. Radiative heat transfer, McGraw-Hill, New York.

- Invernali, G., Monti, R. and Proto, G., 1995.** Assessment of the SCDAP/RELAP5 in vessel corium coolability modelling for a medium size Pressurised Water Reactor, *Transactions of the American Nuclear Society*, **73**, 499-501.
- Jeong, K. J., Lim, D. C. and Hwang, I. S., 1997.** Creep rupture behaviour of a PWR head in a core-melt accident, *Transactions of American Nuclear Society*, **77**, 273-274.
- Kaminski, D. A. and Prakash, C., 1986.** Conjugate natural convection in a square enclosure: effect of conduction in one of the vertical walls, *Int. J. Heat Mass Transfer*, **29**, 1979-1988.
- Kim, S. W., Podowski, M. Z., Lahey, Jr. R. T. and Kurul, N., 1997.** The modelling of core melting and in-vessel corium relocation in the APRIL code, *Nucl. Eng. Des.*, **177**, 265-283.
- Kulacki, F. A. and Nagle, M. E., May 1975.** Natural convection in a horizontal fluid layer with volumetric energy sources, *Trans. ASME*, 98-103.
- Kune, Y. S. and Robert, E. H., 1996.** Debris interactions in reactor vessel lower plena during a severe accident II. Integral analysis, *Nucl. Eng. Des.*, **166**, 165-178.
- Kune, Y. S. and Robert, E. H., 1994.** Integral analysis of debris material and heat transport in reactor vessel lower plenum, *Nucl. Eng. Des.*, **151**, 203-221.
- Kymalainen, O., Tuomisto, H., Hongisto, O. and Theofanous, T. G., 1994.** Heat flux distribution from a volumetrically heated pool with high Rayleigh number, *Nucl. Eng. Des.* **149**, 401-408.
- Lage, J. L. and Bejan, A., 1991.** The Ra-Pr Domain of Laminar Natural Convection in an Enclosure Heated from the side, *Numerical Heat Transfer*, **19 (part A)**, 21-41.
- Lee, J. H. and Goldstein, R. H., May 1988.** An experimental study on natural convection heat transfer in an inclined square enclosure containing internal energy sources, *Heat Transfer* **110**, 345-349.
- Liaqat, A. and Baytaş, A. C., 2000.** Heat transfer characteristics of internally heated liquid pools at high Rayleigh numbers, *Heat and Mass Transfer*, **36(5)**, 401-405.
- Liaqat, A. and Baytaş, A. C., 2000a.** Conjugate natural convection in a square enclosure Containing volumetric sources, *Int. J. Heat Mass Transfer*, (in press).
- Liaqat, A. and Baytaş, A. C., 2000b.** Conjugate heat transfer in a 2-D pool of molten core material of a Light Water Reactor, *Proc. European Congress on Computational Methods in Applied Sciences and Engineering (ECCOMAS 2000)*, World Trade Centre, Barcelona, Spain, September 11-14.

- Liaqat, A. and Baytaş, A. C., 2000c.** Characteristics of conjugate free convection in a corium pool formed during severe reactor accident, *Int. J. of Engineering Sciences*, (in press).
- Liaqat, A. and Baytaş, A. C., 2000d.** Numerical analysis of buoyant flow in a hemispherical enclosure at high Rayleigh numbers- a benchmark solution, Proc. 12th International Symposium on Transport Phenomenon, Istanbul, Turkey, July 16-20.
- Liaqat, A. and Baytaş, A. C., 2000e.** Cooling of molten core material within a Pressurised Water Reactor vessel lower head: interaction of surface radiation and wall conduction with free convection, *Int. J. of Engineering Sciences*, (in press).
- Loktionov, V. D., Mukhtarov, E. S., Yaroshenko, N. I. and Orlov, V. E., 1999.** Numerical investigation of the reactor pressure vessel behaviour under severe accident conditions taking into account the combined processes of the vessel creep and the molten pool natural convection, *Nucl. Eng. Des.*, **191**, 31-52.
- Magallon, D. and Hohmann, H., 1997.** Experimental investigation of 150-kg-scale corium melt jet quenching in water, *Nucl. Eng. Des.*, **177**, 321-337.
- May, H. O., 1991.** A numerical study of natural convection in an inclined square enclosure containing internal sources, *Int. J. Heat Mass Transfer*, **34 no. 8**, 919-928.
- Mayinger, F., Jahn, M., Reineke, H. and Steinbrenner, V., 1976.** Examination of thermal hydraulic processes and heat transfer in a core melt, BMFT RS 48/1, Institut für Verfahrenstechnik der T.U. Hanover, Germany.
- Modest, M. F., 1993.** Radiative heat transfer, McGRAW-HILL, international Edition, New York.
- Moore, R. L., Golden, D. W. and Tolman, E. L., 1989.** The Three Mile Island unit 2 degraded core heatup and cooldown analysis, *Nuclear Technology*, **87**, 990-1004.
- Nakamura, H., Sibamoto, Y. and Anoda, Y., 1999.** Visualisation of simulated molten fuel behaviour in a pressure vessel lower head using high-frame-rate neutron radiography, *Nuclear Technology*, **125**, 213-224.
- Nazare, S., Ondracek, G. and Schulz, B., 1977.** Properties of light water reactor core melts, *Nuclear Technology*, **32**, 239-246.
- Nourgaliev, R. R., Dinh, T. N. and Sehgal, B. R., 1997.** Effect of fluid Prandtl number on heat transfer characteristics in internally heated liquid pools with Rayleigh number up to 10^{12} , *Nucl. Eng. Des.*, **169**, 165-184.

- Nourgaliev, R. R., Dinh, T. N. and Sehgal, B. R.**, 1997a. Simulation and analysis of transient cooldown natural convection experiments, *Nucl. Eng. Des.*, **178**, 13-27.
- Park, H., Dhir, V. K. and Kastenberg, W.E.**, 1994. Effect of outside cooling on the thermal behaviour of a boiling water reactor vessel lower head, *Nuclear Technology*, **108**, 266-282.
- Park, H. and Dhir, V. K.**, 1992. Effect of outside cooling on the thermal behaviour of a pressurised water reactor vessel lower head, *Nuclear Technology*, **100**, 331-345.
- Park, R. J., Kim, S. B. and Kim, H. D.**, 1999. Natural convection heat transfer with crust formation in the molten metal pool, *Nuclear Technology*, **127**, 66-80.
- Patankar, S. V.**, 1980. Numerical heat transfer and fluid flow, McGraw-Hill, New York.
- Plys, M. G.**, 1993. Hydrogen production and combustion in severe reactor accidents: An integral assessment perspective, *Nuclear Technology*, **101**, 400-410.
- Powers, D. A., Copus, E. R. and Bradley, D. R.**, 1993. Results of recent investigations at sandia national laboratories on core debris interactions with concrete, *Nuclear Technology*, **101**, 255-261.
- Press, W. H., Flannery, B.P., Teukolsky, S. A. and Vetterling, W.T.**, 1986. Numerical recipes, Cambridge University Press, Cambridge, Ch.2.
- Rempe, J. L., Thines, G. L. and Chavez, S. A.**, 1994. A simpler approach for predicting LWR vessel failure during severe accident conditions, *Nucl. Eng. Des.* **148**, 365-383.
- Spindler, B., Moreau, G. M. and Pigny, S.**, 1995. Simulation of the thermal hydraulic behaviour of a molten core within a structure, with the three dimensions three components TABLIAC Code, *Proc. 7th Int. Meeting on Nuclear Reactor Thermal Hydraulics (NURETH-7), Saratoga Springs, NY USA*, September 10-15, 1881-1888.
- Steinberner, U. and Reineke, H. H.**, 1978. Turbulent buoyancy convection heat transfer with internal heat sources, *Proc. 6th Int. Heat Transfer Conf., Toronto*, **2**, 305-310.
- Suh, K. Y.**, 1994. Modelling of heat transfer to nuclear steam supply system heat sinks and application to severe accident sequences, *Nuclear Technology*, **106**, 274-291.
- Suh, K. Y.**, 1994a. Rod bundle thermal hydraulic and melt progression analysis of CORA severe fuel damage experiments, *Nucl. Sci. Eng.*, **116**, 245-268.

- Suh, K. Y. and Henry, R. E.**, 1996. Debris interactions in reactor vessel lower plena during a severe accident: I. Predictive model, *Nucl. Eng. Des.*, **166**, 147-163.
- Suh, K. Y. and Henry, R. E.**, 1996a. Debris interactions in reactor vessel lower plena during a severe accident: II. Integral analysis, *Nucl. Eng. Des.*, **166**, 165-178.
- Suh, K. Y. and Henry, R. E.**, 1994. Integral analysis of debris material and heat transport in reactor vessel lower plenum, *Nucl. Eng. Des.* **151**, 203-221.
- Suh, K. Y. and Henry, R. E.**, 1993. Integral Modelling of Debris bed Heat and Mass Transfer in the PWR lower plenum for severe accident analysis, *Transactions of the American Nuclear Society*, **69**, 506-509.
- Theofanous, T. G., Syri, S., Salmassi, T., Kymalainen, O. and Tuomisto, H.**, 1994. Critical heat flux through curved, downward facing, thick walls, *Nucl. Eng. Des.*, **151**, 247-258.
- Thinnes, G. L. and Moore, R. L.**, 1989. Comparison of thermal and mechanical responses of the Three Mile Island unit 2 reactor vessel, *Nuclear Technology*, **87**, 1036-1049.
- Todreas, G. L. and Kazimi, M. S.**, 1989. Nuclear Systems, Hemisphere Pub. Corp., New York.
- Tuomisto, H. and Theofanous, T. G.**, 1994. A consistent approach to severe accident management, *Nucl. Eng. Des.*, **148**, 171-183
- Tzanos, C. P. and Cho, D. H.**, 1993. Numerical prediction of natural convection in a uniformly heated pool, *Transactions of the American Nuclear Society*, **68** (part A), 496-498.
- Vahl Davis, G. D.**, 1983. Natural convection of air in a square cavity: A bench mark numerical solution, *Int. J. Num. Methods in Fluids*, **3**, 249-264.
- Versteeg, H. K. and Malalsekera, W.**, 1996. An introduction to computational fluid dynamics-The finite volume method, Longman, London.
- WanLai, L., Carlson, K. D. and Cing-Jen, C.**, 1997. Pressure boundary conditions of incompressible flows with conjugate heat transfer on nonstaggered grids, part 1: methods, *Numerical Heat Transfer*, **32**(part A), 459-479.
- Witt, R. J.**, 1994. Local creep rupture failure modes on a corium-loaded lower head, *Nucl. Eng. Des.*, **148**, 385-411.
- Wong, S. K., Yang, Z. and Walton, A.**, 1997. Heat transfer of molten metal layers in severe accidents," *Transactions of American Nuclear Society*, **77**, 427.

Zeinsberger, A., Horner, P. and Mayinger, F., 1997. Cooling mechanisms at the bottom of relocated porous media, *Transactions of American Nuclear Society*, **77**, 270-271.



RESUME

I was born in a village of Province Punjab, PAKISTAN in the year 1964. After completing my basic education in the village and near by cities of Renala Khurd and Sahiwal, I obtained bachelor degree in Agricultural Engineering (Operational Mechanical) from University of Agriculture, Faisalabad, PAKISTAN in the year 1988. In the same year I was selected in a national level examination, for Master's degree in Nuclear Engineering and completed my M. Sc. Nuclear Engineering in 1990. After that I joined Pakistan Atomic Energy Commission and served there as a research engineer until 1995.

I got married in 1995 and have a son who was born in 1996. In the year 1995, I obtained a scholarship for doctorate studies in TURKEY, which is now being completed.

From academic point of view, I have 7 journal papers and 4 conference papers.