Studies on the heat/mass transfer characteristics and fluid structure in a square internal cooling channel with dimpled surfaces

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STUDIES ON THE HEAT/MASS TRANSFER CHARACTERISTICS AND FLUID STRUCTURE IN A SQUARE INTERNAL COOLING CHANNEL WITH DIMPLED SURFACES

A Thesis

Submitted to the Graduate Faculty of the Louisiana State University and Agricultural and Mechanical College
In partial fulfillment of the Requirements for the degree of Master of Science in Mechanical Engineering

In

The Department of Mechanical Engineering

By
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**NOMENCLATURE**

\[ \begin{align*} 
  d & \quad \text{dimple characteristic length} \\
  h & \quad \text{maximum dimple depth} \\
  D & \quad \text{channel hydraulic diameter} \\
  D_{n-a} & \quad \text{binary diffusion coefficient for naphthalene-air} \\
  h_m & \quad \text{local mass transfer convection coefficient} \\
  H & \quad \text{channel height} \\
  \dot{m}'' & \quad \text{local mass flux} \\
  Nu & \quad \text{Nusselt number} \\
  Nu_0 & \quad \text{Nusselt number for fully developed circular pipe flow} \\
  Pr & \quad \text{Prandtl number} \\
  p_w & \quad \text{naphthalene vapor pressure at the wall} \\
  Re & \quad \text{test-section Reynolds number based on hydraulic diameter} \left( V_{av} \frac{D}{\nu} \right) \\
  Re_H & \quad \text{test-section Reynolds number based on channel height} \left( V_{av} \frac{H}{\nu} \right) \\
  Ro & \quad \text{rotation number} \left( \Omega \frac{D}{V_{av}} \right) \\
  S & \quad \text{surface area} \\
  Sh & \quad \text{Sherwood number} \left( h_m \frac{D S c}{\nu} \right) \\
  Sh_0 & \quad \text{Sherwood number for fully developed flow} \\
\end{align*} \]
Sc  Schmidt number

$T_w$  absolute wall temperature

$V_{av}$  average streamwise component of velocity in the coolant channel

$X, Y, Z$  coordinates in the channel (X is the flow direction, Y the spanwise direction)

**Greek**

$\rho_s$  density of solid naphthalene

$\Delta t$  duration of the experiment

$\Delta x, \Delta y$  scanning steps along the x and y directions

$\delta$  local sublimation depth

$\rho_w$  Naphthalene vapor density at the wall

$\nu$  kinematic viscosity of air

$\rho_b(x)$  bulk vapor density of the naphthalene at streamwise location $x$

$\Omega$  rotational speed
ABSTRACT

Experimental and numerical studies of heat/mass transfer on the dimpled surfaces in square channels are performed. The naphthalene sublimation method is employed in the experimental study. First, in a two-passage rotating channel, circle dimple matrices are cast on the leading and trailing walls in both the inlet and outlet of the serpentine channel. The experiments are carried out with Reynolds number of 7,000 and 21,000, and with Rotation number of 0 and 0.2. The dimpled walls lead to heat/mass transfer enhancements over smooth surfaces both in the stationary case and in the rotating case. In the stationary case, the dimple enhancement factor for Re=7,000 is about 2, but is less than 2 for Re=21,000. In the rotation case at Ro=0.2, the dimple enhancement factor is also at the order of 2. Secondly, four single dimples with printed shape on the wall of square, triangle, circle, and teardrop are studied experimentally in the same test section at stationary with Reynolds number of 21,000. The four dimples are also studied numerically using FLUENT. The heat transfer distributions on the wall and flow structures inside the dimples and in the regions adjacent to the dimples are identified. Vortex rolls caused by dimples are observed. Heat transfer enhancement is found in all of the four cases, and the teardrop dimple shows the most promising heat transfer characteristics.
CHAPTER 1

INTRODUCTION

1.1 Introduction

In the development of advanced turbine systems, there has been considerable effort directed toward increasing the turbine inlet temperature to achieve maximum efficiency of a gas turbine engine. In advanced engines, the turbine inlet temperature is typically 1600-1800 K, which is beyond the melting point of the alloys from which the blades are made. This fact requires the development of more effective internal and external blade cooling strategies. Figure 1.1 shows the cooling techniques employed in a modern turbine blade.

Inside a turbine blade, single-pass or multi-pass passages are made for internal cooling. These internal passages are equipped with a variety of small devices on the walls, including pin fins, rib turbulators, or other turbulence promoters to enhance the heat transfer rate. The turbulence promoters enhance the heat transfer basically in two ways: they induce secondary flow to intensify the flow turbulence, and increase the surface area for convective heat transfer.

As shown in Figure 1.1, at first, the cool compressed air is forced to enter into the turbine blade through the blade root, and then the cool air is circulated to different passages. Inside the front portion of the blade, compressed air cools down the front wall by impingement. The cool air is also injected out of the blade through the tiny holes in the wall to form a thin film of cool air between the blade surface and the hot gas flow to diminish the heat transferred from the external gas flow. Inside the rear portion of the blade, compressed air convectively extracts the heat away from the wall while it travels...
along the internal passage roughened with pin fins; the cooling air is then discharged from the blade trailing edge. Inside the middle portion, cool compressed air passes through multi-pass serpentine passages, where ribs are mounted on the walls to enhance heat transfer, and then the air leaves out of the blade from the small holes on the top of the blade. In recent years, efforts aimed at improving internal cooling have led to the use of a new method: roughening the walls of the blade internal channels with dimples. A literature review is provided below in two parts: the first part discusses the most typical studies on the internal cooling in ribbed channels to introduce the concept of rotation effects; the second part describes the research in more detail on the internal cooling using dimpled surfaces.

Figure 1.1: Typical Cooling Arrangement in Turbine Blade (Courtesy of Han et al [1])
1.2 Internal Cooling with Ribbed channels

1.2.1 Stationary Ribbed Channels

Rib turbulators, or trip strips, are the most intensively studied turbulence promoters used in the turbine internal cooling passages. A great number of experimental and numerical studies have been carried out to investigate the effects of ribs on heat transfer in internal channels. The factors considered in the studies include the rib geometry (the cross-sections of the ribs), the rib configuration in the channel (the angle of attack of the ribs and the relative distance between the ribs), the shape and the aspect ratio of the cross-section of the channel, the rotation of the channel, the turn or bend effects of the channel (multi-pass serpentine channels), and orientations of rotating channels with respect to the axis of rotation.

The early investigations focus primarily on straight, stationary, ribbed channels with a square or rectangular cross-section. By measuring the wall temperature and flow temperature using thermocouple, Han et al. [2-4] carried out extensive experimental studies on stationary ribbed channels and identified that heat transfer rate in stationary ribbed channels primarily depends on the flow Reynolds number, the rib spacing, the rib angle, and the pitch to height ratio of ribs, and the channel aspect ratio. Heat transfer, friction factor, and thermal performance factor were presented. The experimental results showed that the heat transfer on the ribbed walls doubled or increased more compared to the smooth channel; and the skewed ribs promoted higher heat transfer than ribs normal to the main flow. The best heat transfer performance was obtained in the square channel with rib angle of attack from 30 to 45 deg.
Han and Zhang [5] further investigated the heat transfer characteristics of internal air flow in a three-pass square channel by employing the naphthalene sublimation method. The test section consisted of three straight square channels connected by two 180-deg bends. The rib height-to-hydraulic diameter ratio was 0.063; the rib pitch-to-height ratio was 10. Two angles of attack, 60 deg and 90 deg, were studied. For the 60 deg case, both the cross ribs and the parallel ribs on two opposite walls of the cooling channel were investigated. It was found that the rib angle, the rib configuration, and the bend significantly affected the local heat/mass transfer distributions; the 60 deg ribs provided higher heat/mass transfer coefficients than the 90 deg ribs; and the parallel ribs generated higher mass transfer coefficients than the cross ribs.

Liou et al. [6, 7], Taslim et al. [8, 9] and Acharya et al. [10] also studied the heat transfer and flow filed in the ribbed channels. Liou et al. found that the rib pitch-to-height ratio of 10 resulted in the best heat transfer; the heat transfer showed a periodic behavior between consecutive ribs; and both heat transfer and friction factor increased with decreasing rib spacing. Taslim et al. confirmed Han et al.’s findings that skewed (45 deg) ribs produced higher thermal performance factor than transverse (90 deg) ribs. Taslim et al. further placed ribs on all four walls of square and trapezoidal channels, and found that the heat transfer coefficients and thermal performance factors were enhanced compared to the channels with ribs only on two opposite walls. Acharya et al. experimentally investigated the flow and heat transfer in a rectangular channel with ribs attached along one wall. Their results also showed the periodic heat transfer profile induced by the ribs. The measured flow field revealed separation regions downstream the ribs followed by reattachment and redevelopment regions.
1.2.2 Ribbed Channels with Rotation

Due to the fact that the turbine blades in practical use always rotate with high speed, so the rotation effects on the ribbed passages also received many researchers’ studies [11-14]. Taslim et al. [11] studied a rotating, straight, square cross-section channel with two opposite rib-roughed walls and with radially outward flow. The ribs were made in a parallel staggered fashion with 90 deg to the main flow. Liquid crystals were used to determine heat transfer coefficients. A maximum increase in heat transfer coefficient of about 45% over that of the stationary case on the trailing surface, and a minimum decrease of 6% on the leading surface were reported for the parameters covered in the experiment. Griffith et al. [12] experimentally studied straight, smooth and 45 deg ribbed, rectangular models of $AR = 4:1$ in the parameter range of $Ro = 0.305$ at $Re = 5,000$ to $Ro = 0.038$ at $Re = 40,000$. In this study, particular care was taken to identify the variations of heat transfer along the spanwise direction of the ribbed walls. This was accomplished by installing two parallel rows of insulated copper plates on both leading and trailing surfaces along the streamwise direction. Their results showed that on one row the heat transfer was higher than that on the other row at the same streamwise location. Wagner et al. [13] and Johnson et al. [14] further studied serpentine, square cross-sectional rotating channels with ribs on the trailing and leading walls. The ribs were normal (90 deg) or skewed (45 deg) to the flow. For both rib orientations, the rib pitch-to-height ratio, $P/e$, was 10, and the rib height-to-channel hydraulic diameter ratio, $e/D_h$, was 0.1. Their results for the two rib orientations showed that heat transfer increased by more than 200% as compared with the corresponding cases on the smooth model; and the rotation effects on the ribbed passages generally augmented the heat transfer on the
trailing walls but degraded the heat transfer on the leading walls in the radially outward flow, and vice versa in the radially inward flow, however, the increase and the decrease trends in heat transfer were not monotonic.

![Image: Rotation Effects on a Radially Outward Flow with Rotation]

Figure 1.2: Rotation Effects on a Radially Outward Flow with Rotation

The nature of rotation effects on the internal cooling flow is basically the effects of Coriolis force on the rotating radially flow when no heat transfer occurs. Hart [15] and Johnson et al. [16] revealed this fact in the early 1970s in rotating smooth channels by flow visualization, flow field measurement and theoretical analysis. Hart observed a formation of secondary flow in the form of paired, longitudinal vortices that drive fluid away from the leading wall toward the trailing wall. Based on the theoretical analysis on the momentum and vorticity equations for rotating channel flow, Hart concluded that the secondary flow was the result of Coriolis force. Johnson et al. observed as well the longitudinal secondary circulation within the rotating channel. They further pointed out
that Coriolis force generally acts to stabilize the leading wall boundary layer, but destabilize that on the trailing wall. As the results, the wall shear stress in the destabilized region was greater than the non-rotating value, and consequently, shear stress along the stabilized wall was reduced.

When heat transfer occurs on the walls of a rotating channel, the density gradient of the flow close to the wall coupled with the centrifugal force produces the rotational buoyancy force. This buoyancy force is another factor of rotation effects in the rotating channel. Figure 1.2 shows a pictorial view of rotation effects on the radially outward flow in a rotating channel with the Coriolis force and rotational buoyancy force.

1.3 Internal Cooling with Dimpled Surfaces

1.3.1 Early Works in the Former Soviet Union

![Typical Schematic Diagrams of Individual Dimple and Dimple Arrays Arrangement](image)

(a) Individual Dimple

Figure 1.3: Typical Schematic Diagrams of Individual Dimple and Dimple Arrays Arrangement (Courtesy of Burgess et al. [28]. Figure continued)
Dimplers are arrays of depressions or indentations on surfaces. Figure 1.3 shows typical schematic diagrams of a spherical dimple and dimple arrays. Dimpled channel is attractive from a turbine blade cooling perspective, because it is expected that, just like ribs, dimples generate strong vortices when dimples are placed in flow, the vortices then produce higher turbulence around the dimples and in the downstream area of the dimples, thus augment heat transfer. Moreover, dimples should produce lower pressure drop than ribs, because dimples do not protrude into the flow to induce any form drag.

For these reasons, several investigators have explored the use of dimpled internal coolant turbine blade passages. The first known published research on the flow structure and heat transfer characteristics of dimples are from the former Soviet Union, according to Ligrani et al. [17].

Of these early studies, Schukin et al. [18] studied the effects of channel geometry (constrictor and diffuser channels) on the heat transfer downstream of a single hemispherical dimple with sharp edge; the effect of the turbulence intensity ($T_u \infty$) in the main flow on the dimple heat transfer was also studied. The diameter of the dimple ($d$, the diameter of the cavity circle on the surface) is 0.50 mm, and the ratio of depth to diameter ($h/d$) is 0.5. For the whole range of the constrictor/diffuser angle $\theta/2 = 0$ to $8.5^\circ$, 8
the heat transfer coefficient increase to 1.2 times of that on the smooth surface without
dimple at Re=140,000. Their results on a dimple of d=150 mm and h/d = 0.5 showed that
the dimple enhanced the heat transfer when Tu_infinity <15%, but dimple made no difference in
heat transfer for when Tu_infinity ≥ 15%.

Terekhov et al. [19] experimentally studied the flow structure, pressure field, and
heat transfer in a channel with a single dimple on one wall. Two types of dimple were
tested: with a sharp edge and with a rounded off edge. The value of h/d was 0.13, 0.26
and 0.5. The velocity field was measured using LDA, and flow visualization was
achieved by use of hydrogen bubbles. Heat transfer measurement was also done within
the dimple. The authors claimed that auto oscillations of the flow arose in the dimple for
all the test conditions; pressure loss increased as the depth of dimple increased. The
experimental results showed clearly that the heat transfer was stronger in the downstream
half of dimple than in the upstream half.

1.3.2 Recent Works in the United States

Chyu et al. [20] reported an experimental study on heat transfer, using transient
liquid crystal method, on surfaces with staggered dimple arrays in rectangular channels of
aspect ratio of 2, 4 and 12. Hemispheric dimple and teardrop shaped dimple were
investigated. The depth to diameter ratio (h/d) was 0.35 for both dimples. Reynolds
numbers based on the channel hydraulic diameter and bulk mean velocity varied between
10,000 and 52,000. The heat transfer distributions showed that the heat transfer ratio
(Nu/Nu_0) everywhere on the surfaces with dimple arrays was higher than the values in the
smooth channels. The peak values of Nu/Nu_0 were more than 2.5 at the immediate
downstream areas of dimples for $\text{Re}=23,000$. Their results also indicated that the teardrop dimples generally generated higher heat transfer.

Lin et al. [21] presented flow and heat transfer computational predictions to help explain the observed heat transfer behavior. Flow streamlines and temperature distributions were provided for the dimpled surface. The authors claimed that, as the flow entered into a dimple, two vertices took place within the dimple. Their results also showed that the heat transfer on the surfaces increased when dimples were made on two opposite walls and the two walls were push closer together.

Moon et al. [22] reported an experimental investigation, by using transient thermochromic liquid crystal technique, on the heat transfer and friction in rectangular channels with dimple matrix imprinted on one wall of each channel. The effect of channel height on the heat transfer and friction of the dimpled walls was studied in rectangular channels with the ratio of the channel height to dimple imprint diameter ($H/d$) of 0.37, 0.74, 1.11 and 1.49. The ratio of depth to diameter of the dimple ($h/d$) was 0.193. The heat transfer enhancements of the order of 2.1 over smooth surfaces were reported with pressure drop penalties in the range of 1.6-2.0 over smooth surfaces. The authors claimed that the heat transfer ratio ($\text{Nu}/\text{Nu}_0$) and the friction factor in the fully developed region were invariant with Reynolds numbers for $\text{Re}$ from 12,000 to 60,000, and no detectable effect of the channel height was found for all of the $H/d$ cases covered. In a later study, by using the same experimental apparatus and test method, Moon et al. [23] investigated the heat transfer and friction on a smooth wall with an opposite dimpled wall installed with a clearance gap of $\delta/d$ between. The values of $\delta/d$ considered were 0, 0.024, and 0.055. The authors observed overall heat transfer enhancements of 1.4 to 3.08 on the
smooth wall augmented by the dimples on the opposite wall for $\delta/d$ from 0 to 0.055 and Re from 11,500 to 35,000. As expected, the heat transfer enhancement decreased as the clearance gap increased.

Another group of researchers, Moon et al. [24] at Texas A & M University, examined two types of dimple arrays in a square channel: concave dimples and cylindrical dimples. The dimple arrays were machined on an aluminum test plates. During the tests, the dimpled plate was placed as the bottom walls of the test channel, and was heated by an electric strip heater. The temperature on the selected locations on the dimpled wall was taken by thermocouples. The ratio of depth to diameter of dimple $h/d$ ranged from 0.134 to 0.250 for the nine dimple arrays considered. The local average heat transfer enhancement was found to be from 1.7 to over 3 for Reynolds number Re from 10,000 to 65,000. The authors claimed that the cylindrical dimples caused higher overall heat transfer coefficient based on the projected area and lower pressure drop than the concave dimples with the same diameters and depth.

![Diagram](https://via.placeholder.com/150)

**Figure 1.4:** Sketches of Three-Dimensional Flow Structure Generated by a Concave Dimple (Courtesy of Griffith et al. [29])
The researchers at University of Utah performed extensive studies on the flow structure and heat transfer behaviors on the dimpled surfaces (Mahmood et al. [24, 27], Ligrani et al. [25, 26] and Burgess et al. [28]). Mahmood et al. [24] made detailed flow and heat transfer measurements on a dimpled plate, and identified specific vortex structures responsible for augmenting heat transfer, especially along the downstream rim of each dimple. Heat transfer enhancements ranging from 1.8-2.4 over smooth plates were noted. Ligrani et al. [25] performed an elaborate examination on the flow structure on a dimpled wall by employing flow visualization, pressure and velocity measurement with specially designed five-hole pressure probe, and Reynolds normal stresses measurement with hot-wire probe. The dimpled wall was machined with 13 staggered rows of dimples along the streamwise direction. Each of the dimples had a print diameter (d) of 5.08 cm, and a ratio of depth to print diameter (h/d) of 0.2. The authors identified a primary vortex pair periodically shed from the central portion of each dimple, and observed two additional secondary vortex pairs formed near the spanwise edges of each dimple. Figure 1.4 shows the sketches of three-dimensional flow structure generated by a concave dimple.

Ligrani et al. [26] then added protrusions on the wall opposite to the dimpled wall, and found the protrusions produced more vertical, secondary structure and flow mixing resulting higher heat transfer enhancement and higher pressure drop compared to the channel with dimpled-smooth opposite walls. Mahmood et al. [27] further investigated the effects of the ratio of inlet stagnation temperature to the local wall temperature on the flow structure in a dimpled channel with four values of H/d (channel height to dimple diameter): 0.20, 0.25, 0.50 and 1.00. The authors found that the vortex pairs discussed
above became stronger as H/d decreased. Burgess et al. [28] studied the effect of dimple depth on the heat transfer in a rectangular channel with aspect ratio of 8. The ratio of depth to diameter of dimple (h/d) was 0.3 with d = 5.08 cm. In the channel, one wall was machined with 29 rows of dimples, and on each row, 4 or 5 dimples were located. The temperature on the dimpled wall was recorded by thermocouples and an infrared camera. Their results showed that Nu/Nu₀ was lower in the upstream halves of dimples and higher in the downstream halves, however, the value of Nu/Nu₀ everywhere on the dimpled surface including the area inside dimples was greater than 1. The maximum Nu/Nu₀ values occurred around the downstream dimple rims. The most important finding was that the local Nu/Nu₀ and average Nu/Nu₀ were higher for h/d = 0.3 than that for h/d=0.2, while other conditions held exactly the same, indicating that the deeper dimple produced higher heat transfer.

Most recently, Griffith et al [29] studied a straight dimpled rectangular channel of aspect ratio 4:1 with rotation using thermocouple to measure the local average wall temperature, and hence average heat transfer parameters were obtained. The average Nu/Nu₀ was approximately 2.0 for stationary cases with Reynolds number from 5,000 to 40,000. The authors claimed that rotation augmented the heat transfer on both dimpled trailing and leading walls.

The numerical simulations on the flow filed and heat transfer inside and around dimples are very limited. Choudhury et al. [30] performed a computation for fluid flow and heat transfer in a parallel plate channel with periodically spaced dimples. The flow is assumed to be of constant property, two-dimensional and with uniform wall temperature. Local results for the periodic fully developed velocity and temperature fields, the average
heat transfer and pressure drop were presented. The authors found that decrease in the channel width and the dimple spacing was both accompanied by increase in the heat transfer and pressure drop. It must note that the flow Choudhury et al. [30] studied was laminar, which is not the practical situation in the dimpled internal cooling channel.

1.3.3 Numerical Simulations

![Vortex Pattern and Heat Transfer Distributions along Streamwise, Main Flow Direction Is from Left to Right](courtesy of Isaev et al. [33])
Isaev et al [31] then reported a simulation of flow structure on a single spherical dimple in turbulence flow. The flow was assumed to be incompressible liquid. The authors found that, in the cross-sectional view, two large-scale vortex cells were evident, and non-symmetric flow patterns were observed in the dimple, although the dimple itself and main flow were symmetric.

In 2003, Isaev et al [32, 33] presented a numerical simulation in detail of a vortex flow and heat transfer in the vicinity of a dimple in a narrow channel, where a single spherical dimple was made on one of the channel walls. The k-ω turbulence model was employed in the computation. The variation of the ratio of the depth to diameter of dimple (h/d) from 0.04 to 0.22 was examined. Special attention was given to the vortex structure induced by the dimple. Both symmetric and asymmetric vortices were observed, as seen in Figure 1.5 (a). The authors referred this kind of helical or spiral vortex as of tornado-like mechanism. The heat transfer enhancement was also provided along the longitudinal and spanwise central lines. As shown in Figure 1.5 (b), the heat transfer ratio \( \frac{Nu}{Nu_0} \) dropped below 1 on the upper half of dimple inner surface, and increased along the streamwise direction, the maximum value of \( \frac{Nu}{Nu_0} \), up to 3.5, occurred on the downstream dimple rim within a very narrow band, then \( \frac{Nu}{Nu_0} \) decreased sharply in the immediate wake area of the dimple rim, and reached to about 1.3 at a downstream distance of one dimple diameter. Detailed heat transfer distributions inside the dimple or on the wake area of the dimple were not available.

1.4 The Objectives of the Present Study

As seen in the discussions above, the flow structure and heat transfer characteristics in dimpled passages are not fully investigated. First of all, the rotation
effects have not been examined for dimpled channels as done for the ribbed ones. It must be pointed out that, the only published research on the rotation effects on the dimpled walls (Griffith et al [29]) was performed three years after the present study was finished (Zhou et al. [34]). Secondly, most studies on the dimples are focused on the spherical dimples. Limited attention has been paid on the other dimple shapes, even the other dimple shapes are expected to be more effective, for example, the teardrop shaped dimple. Thirdly, even for the most extensively studied spherical dimples, the heat transfer enhancement reported by a variety of researchers is not consistent, varying form 1.5 to almost 3. Further, the flow structure induced by the spherical dimple has not yet been fully understood.

The present study has two major objectives. First, the rotation effects on the dimpled walls are to be investigated in a U-shaped internal cooling passage. The heat transfer enhancement on both leading and trailing walls is to be examined in the radially outward inlet channel and in the radially inward outlet channel. Second, four dimple shapes are to be investigated experimentally and numerically. The four dimple shapes are square, triangle, circle and teardrop on the channel surface, respectively. The heat transfer characteristics and flow patterns of the four dimple shapes are to be explored.

The experimental setup, test section, numerical simulation procedure, data reduction procedure, results and discussions are elaborated in the chapters followed.

1.5 Outline of the Thesis

The thesis consists of six chapters. It is basically divided into four parts. The first part is Chapter 1, which presents a brief introduction of the internal cooling of turbine blade and a critical literature review on studies of the internal cooling techniques with
ribs and dimples. The objectives of the thesis are also described in this chapter. The second part includes chapter 2 and 3. In chapter 2, the experimental setup, the method and steps of the experiment, and the data reduction procedure are elaborated. Chapter 3 discusses the experimental results of heat transfer on the leading and trailing walls in a two-passage rotating square channel. The leading and trailing walls of the inlet and outlet passages are cast with circle dimple matrix, while the side walls are smooth. The third part covers chapter 4 and chapter 5 discussing the experimental and numerical results of four dimple shapes: square, triangle, circle and teardrop. Chapter 4 introduces the method and conditions of the computation. The dimensions of the four dimples are given in this chapter as well. Chapter 5 discusses the numerical results on heat transfer and flow structures of square dimple, triangle dimple, circle dimple and teardrop dimple. The last part, Chapter 6, summarizes the major conclusions on the rotating dimpled passages and on the four single dimples.
CHAPTER 2

EXPERIMENTAL APPARATUS AND DATA REDUCTION PROCEDURE

All of the experiments in the present study are done using the naphthalene sublimation method. The most important advantage of this method is that the local mass/heat transfer distributions can be obtained in detail (Goldstein et al. [36]).

2.1 Experimental Apparatus

Figure 2.1 shows a schematic diagram of the test section. Compressed air is delivered through rotating seals into a hollowed rotating shaft, and is then redirected through a rigid tube connecting the hollow shaft to the test section. The air enters the test section through a conditioning plenum, and the naphthalene laden exhaust air is discharged through a flexible tube to a fume hood. The aluminum alloy test section consists of a 69.85mm tapered settling chamber; a frame that supports eight removable, hollow test section plates; and a removable 180 degree bend. These major components are secured in a flange-like manner, using O-rings between all parts to prevent air leakage. When assembled, the test section forms 25.4mm x 25.4mm x 304.8mm long inlet and outlet sections 38.1mm apart that are connected by the 180 degree, 25.4mm x 25.4mm square cross-section bend. Each test section plate is a reinforced recessed frame to accommodate casting of a naphthalene layer for mass transfer measurements. The two-pass test section is housed inside a small pressure-vessel, and prior to rotation, pressure inside the pressure vessel is equalized with the pressure in the two-pass coolant channel. The test-section assembly is connected to the hollow rotating shaft driven by a hydraulic
Figure 2.1: Schematic of the Rotating Rig with Two-Pass Test Section
motor. For load balancing, a dummy weight is placed opposite to the test-section assembly. The entire rotating arm is contained within a large pressure vessel for safety purposes.

Detailed surface profiles of the cast surfaces are required for local mass transfer results. These profiles are obtained by moving the walls under a fixed, linear variable differential transducer (LVDT) type profilometer. A custom written program run on a personal computer is used to control the motion of the traversing table through micro-step drive motors with a 0.00127mm step size.

A thermocouple is mounted into the pipe upstream of the test section to measure the free stream temperature, which is required for calculating the mass flow rate. To measure the naphthalene surface temperature, two thermocouples are embedded into the naphthalene surface along one wall. The outputs of the thermocouples are saved in an Omega temperature data logger (OM-SP-1700). The recorded data is read out by a computer, and the average value is taken as the wall temperature of the naphthalene surfaces.

2.2 Casting of the Smooth and Dimpled Test Plates

Fresh, 99% pure naphthalene crystals are melted, and the molten naphthalene is quickly poured into the hollow cavity of the plate frame to fill completely the region between the frame walls and the stainless steel casting plate.

The pattern on the casting plates can be used to generate a specific pattern on the cast naphthalene surface. For a smooth surface the casting plate is polished to a mirror-smooth finish, and checked for flatness. For a dimpled surface, the casting plate surface has machined protrusions that are the inverse of the dimples desired. After eight hours of
casting time, each wall is then separated from the casting plate (with the protrusions leaving the desired dimpled impressions on the naphthalene surface) and placed on the mounting plate for scanning. After scanning, the plates are stored in an air-tight container, saturated with naphthalene vapor, to hinder natural sublimation until the test section is assembled. After the experiment is over, the test section is disassembled and the walls are placed in the storage container until they are scanned again.

Figure 2: (a) Schematic of the Dimpled Test Surface (b) Geometrical Dimensions of the Dimples in a Plane Parallel to the Test Surface (c) Geometrical Dimensions of the Dimples in a Plane Normal to the Test Surface. (Unit: mm. Figure continued)
(b) Staggered Dimple Matrix, Front View

(c) Dimples in Cross-Sectional View
Table 1: Comparison of Parameters Used in the Present Study with Those Reported in the Literature

<table>
<thead>
<tr>
<th></th>
<th>d (mm)</th>
<th>H/d</th>
<th>SD/d</th>
<th>h/d</th>
<th>dx/d</th>
<th>dy/d</th>
<th>Re</th>
<th>ReH</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Present Study</strong></td>
<td>5.2</td>
<td>4.88</td>
<td>1.15</td>
<td>0.29</td>
<td>1.17</td>
<td>0.82</td>
<td>7,000</td>
<td>7,000</td>
</tr>
<tr>
<td><strong>Mahmood et al. [23]</strong></td>
<td>50.8</td>
<td>0.50</td>
<td>1.45</td>
<td>0.20</td>
<td>0.81</td>
<td>0.81</td>
<td>21,000</td>
<td>21000</td>
</tr>
<tr>
<td><strong>Moon et al. [22]</strong></td>
<td>17.1</td>
<td>0.35~</td>
<td>1.46</td>
<td>1.49</td>
<td>0.19</td>
<td>1.09</td>
<td>29,202</td>
<td>1,250~61,500</td>
</tr>
<tr>
<td><strong>Chyu et al. [20]</strong></td>
<td>13.5</td>
<td>1.33,4.8</td>
<td>1.06</td>
<td>0.35</td>
<td>1.06</td>
<td>1.06</td>
<td>10,000~52,000</td>
<td></td>
</tr>
<tr>
<td><strong>Lin et al. [21]</strong></td>
<td>8.3</td>
<td>2.30,4.60</td>
<td>2.30</td>
<td>0.58</td>
<td>2.30</td>
<td>2.30</td>
<td>23,000</td>
<td>46,000</td>
</tr>
</tbody>
</table>

A schematic of a dimpled wall employed in the experiments is shown in Figure 2.2 (a). The plate is 305.3 mm long and 25.5 mm wide with 47 rows of dimples. The dimple matrix has two kinds of dimple rows cast alternately. One row has three dimples along the spanwise direction; the other row has two dimples. For each test, four such identical walls are installed in the inlet and outlet passages forming the two leading walls and the two trailing walls. The dimples are staggered as shown in Figure 2 (b).

Table 1 lists the critical dimple parameters (dimensionless) and shows the comparison with other published studies on dimpled surfaces. As may be seen from this table, the geometrical parameters used are in the range of parameters used by other investigators.

2.3 Data Analysis and Uncertainty

Mass flow rate in the meter run is calculated from measurements of temperature, pressure, and differential pressure using standard equations for concentric bore orifice meters (Stearns et. al. [36], Miller [37]).
Naphthalene sublimation depth is calculated from the two surface profiles for each wall. Each profile is normalized with respect to a reference plane computed from three points scanned on the aluminum surface of the walls. The difference between the normalized profiles gives the local sublimation depth.

2.3.1 Data Analysis

The local mass flux $\dot{m}''$ at each location is calculated from the following expression:

$$\dot{m}'' = \rho_s \delta / \Delta t \quad \text{(2.1)}$$

where $\rho_s$ is the density of solid naphthalene, $\delta$ is the local sublimation depth, and $\Delta t$ is the duration of the experiment. Vapor pressure at the wall $p_w$ is calculated from the following equation (Sogin et al. [38]):

$$\log_{10}(p_w) = A - B / T_w \quad \text{(2.2)}$$

where $A$ and $B$ are constants(11.884 and 6713, respectively) and $T_w$ is the absolute wall temperature. Wall vapor density $\rho_w$ is then calculated using the perfect gas law. Bulk vapor density of naphthalene $\rho_b(x)$ is obtained by mass conservation balances of naphthalene from the inlet ($x = 0$) to the streamwise location ($x$).

The local mass transfer convection coefficient $h_m$ is then calculated as follows.

$$h_m = \dot{m}'' / (\rho_w - \rho_b(x)) \quad \text{(2.3)}$$

The binary diffusion coefficient $D_{n-a}$ for naphthalene sublimation in air is taken as the ratio of the kinematic viscosity of air $\nu$ to the Schmidt number for naphthalene-air ($Sc = 2.5$). The local Sherwood number $Sh$ is then calculated by:
Sh = h_w D / D_{eq} = h_w D Sc / ν  \tag{2.4}

where \( D \) is the hydraulic diameter of the test section. Comparison of heat transfer and mass transfer results is done through the use of the heat-mass transfer analogy (Sogin et al. [38]):

\[
\text{Nu} = \text{Sh}(Pr / Sc)^{0.4} \tag{2.5}
\]

where \( \text{Nu} \) is the Nusselt number and \( Pr \) is the Prandtl number of air. To facilitate comparison between mass and heat transfer results, a Sherwood number ratio, \( No \), is defined as follows:

\[
N_o = \text{Sh} / \text{Sh}_o = \text{Nu} / \text{Nu}_o \tag{2.6}
\]

where \( \text{Nu}_o \) is the correlated fully developed Nusselt number, analogous to \( \text{Sh}_o \), which is the Sherwood number of fully developed flow in smooth tube:

\[
\text{Sh}_o = 0.023 \text{Re}^{0.8} \text{Sc}^{0.4} \tag{2.7}
\]

Both local and averaged results are compared in this manner. Averaging is performed as a simple arithmetic average of the data points scanned over the region over which averaging is performed.

2.3.2 Uncertainty

Uncertainties for all computed values are estimated using Kline and McClintock method (Kline and McClintock [39]) for single sample experiments. The estimates for these experiments are comparable to previously reported values for both heat transfer and mass transfer studies, but are believed to be conservative due to the fact that each case in the present study is repeated at least twice, usually each test is run three times on the same conditions for verifying the repeatability.
Volume flow rate and duct Reynolds number (Re) uncertainties are estimated to be less than 10 percent for \( \text{Re} > 6000 \). The reported resolution of the LVDT is 0.00127 mm while the analog-to-digital (A/D) board is reported to have an accuracy of 0.002 mm in a 12 kHz acquisition rate, 16 bit resolution mode. Experimental tests of accuracy and repeatability for the entire acquisition system indicate a sublimation depth uncertainty of 0.0038 mm. Sublimation depths are maintained at about 0.152 mm by varying the duration of the experiment. This target depth is selected to minimize uncertainties in both depth measurement and changes in duct cross section area. The resulting experimental duration is about two hours for \( \text{Re} = 7000 \), and one hour for \( \text{Re} = 21000 \), respectively. Vapor density uncertainty based on measured quantities is negligible for both wall and bulk values. Overall uncertainty in Sherwood number calculation, following the same procedure as Hibbs [40], is about 10 percent for \( \text{Re} = 7000 \), and becomes smaller as Reynolds number increases.
CHAPTER 3

DIMPLED SQUARE CHANNELS WITH ROTATION

3.1 Validation of Experiment Method

To validate the experimental procedure, a series of tests was carried out with smooth walls in test section at different Reynolds numbers both at stationary and with rotation. Fig. 3.3 shows the results at Re=7,000 and Ro=0.2. The normalized Sherwood numbers Sh/Sh₀ for both the trailing and leading walls fall within the two sets of results reported by Kandis and Lau [41], which was performed in a two-passage smooth internal channel by using naphthalene sublimation method, similar to the present study. This comparison provides the requisite validation of the experimental procedure used in this study.

![Figure 3.1: Sherwood Number Sh/Sh₀ Distributions of a Verification Test at Re=7,000 and Ro = 0.2 in the Inlet Channel](image-url)
The validation of the experiment procedure can also be proved from Figure 3.2 (a) and (b), where Sh/Sh₀ on all of the smooth walls is close to 1, as expected.

3.2 Results and Discussion

Results are presented for Reynolds numbers Re= 7,000 and 21,000 and two Rotation numbers Ro=0 and Ro=0.2 in the form of either local Sh/Sh₀ distributions, or streamwise-averaged and spanwise-averaged Sh/Sh₀ distributions. For the dimpled walls, contours of Sh/Sh₀ are presented in the periodically fully-developed region. At least two experimental runs are performed for each test case.

3.2.1 Smooth Channels

![Figure 3.2: Centerline Sh/Sh₀ Distributions in the Smooth Inlet (a) and Outlet (b) (Re=21,000, Ro=0, Figure Continued)
At the beginning, tests with smooth walls are performed at Re=7,000 and 21,000 without rotation. Figure 3.2 presents the normalized Sherwood number $Sh/Sh_0$ distributions along the centerline of the leading, trailing and side walls in the smooth inlet and outlet channels at Re=21,000. The Sherwood number distributions on the all walls in the inlet (Figure 3.2 (a)) almost overlap indicating the absence of Coriolis force, and stable flow conditions and good data repeatability. The Sherwood number distributions in the outlet (Figure 3.2 (b)) show some unstable behaviors initially due to the 180-degree bend, but become stable as air flow travels downstream. It can be seen that fully developed conditions are achieved within 5-6 hydraulic diameters downstream of the inlet in the radially-outward flow passage, and downstream of the bend in the radially-inward flow passage.
Figure 3.3: Centerline $Sh/Sh_0$ Distributions in the Smooth Inlet (a) and Outlet (b) 
(Re=21,000, Ro=0.2)
Then rotation tests are carried out with smooth test section. Figure 3.3 shows the results for Re=21,000 and Ro=0.2. Rotation effects are obvious. As expected, in the fully developed region in the inlet, the Sh/Sh$_0$ on the trailing wall increases to about 1.5, but on the leading wall the Sh/Sh$_0$ decreases to 0.5, while the Sh/Sh$_0$ on the side walls almost maintains the same values as the stationary case, i.e., Sh/Sh$_0$ is about unity. In the outlet the bend effect can be seen again. And contrast to the inlet, the Sh/Sh$_0$ on the trailing wall decreases in the outlet, but the Sh/Sh$_0$ on the leading wall increases. The Coriolis force applied to the flow when the test section is rotated causes this phenomenon. In the inlet, the Coriolis force directs to the trailing wall and keeps the flow near the trailing wall stable, but drives the flow away from the leading wall, and undermines the heat transfer process on the leading wall. But in the outlet, the Coriolis force directed to the leading wall, so the rotation effects are opposite to the inlet.

3.2.2 Stationary Dimpled Channels

![Figure 3.4: Centerline Sh/Sh$_0$ Distributions in the Dimpled Inlet (Re=7,000, Ro=0)](image)

Figure 3.4: Centerline Sh/Sh$_0$ Distributions in the Dimpled Inlet (Re=7,000, Ro=0)
The experiments on dimpled walls are conducted at Re=7,000 and 21,000, mainly focusing at Re = 21000. The trailing and leading walls are dimpled, but the side walls are kept smooth. Figure 3.4 presents the Sh/Sh$_0$ distributions along the centerlines in the inlet at Re=7,000 and Ro=0. For the dimpled wall, the peak Sh/Sh$_0$ points occur just downstream of dimples, and the minimum Sh/Sh$_0$ points correspond to the locations between dimples. It is easy to see that the average Sh/Sh$_0$ increases to about 2 on the dimpled wall, while the Sh/Sh$_0$ on the side wall still maintains unity. The enhancement of the Sh/Sh$_0$ on the dimpled wall can only be credited to the dimple effects, because in the stationary case the Coriolis force does not exist.

Figure 3.5: Spanwise Average Sh/Sh$_0$ Distributions on (a) the Dimpled Inlet and (b) the Outlet Passages (Re=21,000, Ro=0. Figure Continued)
Figure 3.5 shows the spanwise-averaged Sh/Sh$_0$ distributions in the stationary dimpled coolant passages for Re=21,000 in both inlet and outlet channels. Only one of the dimpled and one of the smooth surfaces are shown. The peak Sh/Sh$_0$ are located just downstream of the rows with dimples, and the minimum Sh/Sh$_0$ corresponds to the rows with dimples. Note that the successive peak values in the Sh/Sh$_0$ distribution alternate in magnitude with higher values along the 3-dimple row, and lower values along the 2-dimple row. Comparing Figure 3.5 with Figure 3.2, it can be seen that the dimples enhance the mass/heat transfer in the fully developed region by a factor of 1.5-2 in the radially-outward flow leg, and by a factor of 1.1-1.5 in the radially-inward flow leg. Moon et al. [22] have reported enhancements of the order of 2.1 in stationary dimpled passages.
Figure 3.6: Streamwise Average Sh/Sh$_0$ Distributions in (a) the Dimpled Inlet and (b) the Outlet Passages (Re=21,000, Ro=0)
Figure 3.6 presents the streamwise-averaged $\text{Sh}/\text{Sh}_0$ distributions. The streamwise-averaging is performed by summing Sherwood number distributions along spanwise-rows that are at the same relative locations relative to the three-dimple row and dividing by the number of rows over which the summation is performed. The spanwise distributions of the averaged Sherwood numbers are shown at four streamwise locations: (1) along the row containing three dimples (the spanwise locations of the three dimples are $Y/D=-0.33, 0, \text{ and } 0.33$), (2) downstream of the three-dimple row at a location midway between the three-dimple and the two-dimple row, (3) along the row containing two dimples (the spanwise locations of the two dimples are $Y/D=-0.17, \text{ and } 0.17$), and (4) downstream of the two-dimple row at a location midway between the two-dimple and the three-dimple row. It is clear that the row downstream of the three dimples has the highest mass transfer, with the peak in the spanwise direction occurring just downstream of the dimple. Along this row, the minimum in the spanwise direction corresponds to the location midway between the dimples. The row containing the dimples has lower heat transfer, with the minimum along the row coinciding with the dimple center. The maximum along this row occurs along the spanwise edges of the dimples. These observations are consistent with the observations of Moon et al. [22] and Mahmood et al. [24] who indicated that the dimple region itself is characterized by low heat transfer and that the maximum heat transfer rates occur immediately downstream of the dimples and along the spanwise edges of the dimples. The distributions in the outlet duct (with radially-inward flow) are similar to that observed in the inlet duct, except that the Sherwood numbers are somewhat lower.
Figure 3.7: $\frac{Sh}{Sh_0}$ Contours in the Fully Developed Region on Dimpled Walls 
(Re=21,000 Ro=0)
The above-noted trends can be seen more clearly in Figure 3.7. The locations of the dimples are indicated in the contours, and regions containing both the two-dimple row and the three-dimple row are presented. The peak distributions exhibit a half-moon shape immediately downstream of the dimples. The contours of Sherwood number verify that the strongest mass/heat transfer occurs just downstream of each dimple, and along the lateral edges of the dimples.

3.2.3 Dimpled Channels with Rotation

![Graph](image)

Figure 3.8: Centerline Sh/Sh\(_0\) distributions in the dimpled inlet (Re=7,000, Ro=0.2)

The results for the dimpled walls with rotation are shown from Figure 3.8 through Figure 3.11. Figure 3.8 shows the centerline Sh/Sh\(_0\) distributions in the inlet at Re=7,000 and Ro=0.2. Comparing to Figure 3.4, it is clear that the Sh/Sh\(_0\) on the smooth side wall does not change, but the median Sh/Sh\(_0\) on the trailing wall increases from about 2 to 4, and the median Sh/Sh\(_0\) on the leading wall decreases from 2 to about 1.2.
Figure 3.9: Spanwise Averaged Sh/Sh0 Distributions in the Dimpled Inlet (left) and Outlet (right) Passages (Re=21,000, Ro=0.2)
Comparing Figure 3.9 with the data in Figure 3.5, it is observed that rotation enhances the mass/heat transfer on the dimpled trailing wall in the inlet and the dimpled leading wall in the outlet. In the inlet the $\frac{Sh}{Sh_0}$ in the fully developed region increases from 1.8 to about 3 for the trailing wall, and increases from 1.25 to 2.25 for the leading wall in the outlet. The enhancement in $\frac{Sh}{Sh_0}$ is larger in the inlet than that in the outlet. Similarly as on the smooth case, the mass/heat transfer decreases on the leading wall in the inlet and on the trailing wall in the outlet due to the Coriolis force when rotating, but the $\frac{Sh}{Sh_0}$ are still higher than unity on these walls due to the dimple effects.

![Graph showing Streamwise Average Sh/Sh0 Distributions](image-url)

Figure 3.10: Streamwise Average $\frac{Sh}{Sh0}$ Distributions in (a) the Trailing and (b) the Leading Walls in the Dimpled Inlet passage ($Re=21,000$, $Ro=0.2$)
Comparing Figure 3.9 with Figure 3.3, the dimple effects can be evaluated separately insulated from rotation effects. For example, in the fully developed region in the inlet passage, the dimple effects intensifies the $Sh/\text{Sh}_0$ from 1.5 in the smooth case to an average value of 3 in the trailing wall, and enhances $Sh/\text{Sh}_0$ on the leading wall from 0.5 to 1.5. The enhance factor is at the order of 2.

Figure 3.10 presents the streamwise average $Sh/\text{Sh}_0$ distributions for the dimpled walls in the inlet. The distributions are similar to the stationary case shown in Figure 3.6, but the peak values of $Sh/\text{Sh}_0$ has increased from 2.8 to more than 5 in the inlet-trailing wall. Figure 3.11 gives the corresponding $Sh/\text{Sh}_0$ contours in the fully developed regime.
Figure 3.11: $\text{Sh}/\text{Sh}_0$ Contours in the Fully Developed Region in Dimpled Inlet Passage ($\text{Re}=21,000$ $\text{Ro}=0.2$)
3.3 Summary

Measurements of surface mass/heat transfer rate are performed in a dimpled coolant flow passage in both the inlet and outlet. The experiments have been done at Re=7,000 and 21,000 and Rotation numbers of 0 and 0.2. The following major conclusions are observed:

1. The maximum mass/heat transfer rates are obtained downstream of the dimples. The minimum mass/heat transfer rates occur along the row containing the dimples.

2. The dimpled walls lead to enhancements over smooth surfaces both in the stationary case and in the rotating case. In the stationary case, the dimple enhancement factor for Re=7,000 is about 2, but is less than 2 for Re=21,000. For Ro=0.2, the dimple enhancement factor is at the order of 2 for both Re=7,000 and 21,000.

3. The present results show that enhancements achieved with dimples are larger in the inlet passage compared to the outlet passage for Re=21,000.

4. The Sherwood number distributions suggest the existence of three local peaks, with the strongest peak immediately downstream the dimples. These peaks appear to be related to the development of streamwise vortex structures generated from the dimples. The Sherwood number contours also support these observations.
CHAPTER 4

EXPERIMENTAL STUDIES ON FOUR SHAPES OF SINGLE DIMPLE

Four single dimples with print shapes on the wall of square, triangle, circle and teardrop, respectively, are studied. For convenience, the four dimples are referred as square, triangle, circle and teardrop dimple, respectively, in the following chapters. The goals of the present research are to explore, experimentally and numerically, the heat transfer characteristics and flow patterns of the four dimples under the given flow conditions (Re = 21,000), and to enhance the understanding of the heat transfer characteristics both inside the dimples, in the vicinity and in the wake of the dimples.

The experimental studies are carried out in the same apparatus and in the same test section as described in Chapter 2, and the test procedure and data reduction method are exactly the same. The only differences are: (1) one of the dimpled wall is replaced with a smooth wall (top wall), the other wall with dimpled matrix is replaced with the wall with one single dimple (bottom wall); (2) the dimensions of the single dimples are greater than that of the dimples in the previous dimple matrix; the single dimples are deeper, and the surfaces inside the single dimples need to be scanned separately from the scanning of the flat wall to achieve the best results; Both scanning are run with steps of $\Delta x = \Delta y = 0.838$ mm.

4.1 Dimpled Wall Dimensions and Individual Dimple Geometries

The dimensions of the dimpled wall are shown in Figure 4.1. The wall is one foot (304.8 mm) long and 1 inch (25.4 mm) wide. The dimple is placed at $X = 215.9$ mm
(X/D = 8.5) downstream from the channel entrance. Please note that only half of the wall is shown due to the nature of symmetry.

Figure 4.1: Dimensions of the Dimpled Test Wall

The geometries of the four dimples are shown in the Figure 4.2. The maximum depth for each of the four dimples is h, which is 3 mm.

The square dimple is a cavity of erecting quadrangular prism with a flat, square base. The height of the square prism (the depth of the dimple h) is 3 mm, and the length of the square edge (d) is 10 mm, as shown in Figure 4.2 (a).

The triangle dimple is a cavity of erecting triangular prism with a flat, isosceles triangle base. The height of the triangular prism (the depth of the dimple h) is 3 mm, and
the isosceles triangle has a base edge (d) of 10.5 mm long and a height (L) of 10 mm, as shown in Figure 4.2 (b).

The circle dimple, usually called spherical dimple, is in the shape of a part of sphere with a diameter of SD (not shown), as shown in Figure 4.2 (c), the print circle on the flat channel surface has a diameter (d) of 10 mm. The depth of the circle dimple is also 3 mm as mentioned above. Obviously, $(SD/2)^2 = (d/2)^2 + (SD/2 - h)^2$.

![Figure 4.2: Geometries and Dimensions of the Dimples](image-url)
The teardrop dimple is also in the shape of a part of a revolution body, as shown in Figure 4.2 (d). The symmetrical plane of the revolution body is shown in Figure 4.3. The top part chopped off along the horizontal line AB is used as the model to cast teardrop dimple. All the key parameters of the dimples are presented in Table 2. Note that the length of AB in Figure 4.3 is the value of L in Table 2 for the teardrop dimple, i.e., 13.8 mm.

![Figure 4.3: Symmetrical Plane of the Revolution Body from Which the Teardrop Shape Is Cut (Unit: mm).](image)

| Table 2: Key Parameters of the Dimples (Unit: mm) |
|---|---|---|
| Circle | d | h | L |
| Square | 10 | 3 | 10 |
| Triangle | 11.5 | 3 | 10 |
| Teardrop | 10 | 3 | 13.8 |

### 4.2 Experimental Results

The experimental Sh/Sh₀ contours for the four dimples are shown in Figure 4.4, covering a longitudinal (streamwise) distance of more than 2L for each dimple. The
upstream ends of the four dimples are located at \( X/D_h = 8.50 \). The downstream ends of square, triangle and circle dimples are located at \( X/D_h = 8.89 \), while the downstream end of teardrop dimple is located at \( X/D_h = 9.04 \) due to the larger value of \( L \). The dimple edges are marked with black lines in the figures. Due to the constraint of the head size of the LVDT profilometer, a band of dimple bottom surface adjacent to the dimple edges cannot be scanned. Only the central areas on the dimple bottom surfaces are reachable, so the \( Sh/Sh_0 \) values are available only on these areas for all of the four dimples. The vertical inner walls of square and triangle dimples cannot be scanned by the profilometer head either. Moreover, to protect the dimple edges from being cut or crashed by the profilometer head, the scanning meshes are designed one or two steps (about 1 mm) away from the dimple edges on the channel bottom surfaces. These unreachable areas are marked with white blocks in Figure 4.4.

As shown in Figure 4.4 (a) for the square dimple, the \( Sh/Sh_0 \) is less than unity (0.4 to 0.7) on the upper half of the dimple bottom surface, and is greater than 1 on the downstream half; higher values of \( Sh/Sh_0 \) (about 1.68) are observed on a small area on the dimple bottom wall close to the downstream dimple edge. In the wake area of the dimple, the highest \( Sh/Sh_0 \) of 2.10 is observed. \( Sh/Sh_0 \) value maintains 1.94 to 1.25 in a distance of one dimple length \( L \) in the wake of the dimple.

For the triangle dimple shown in Figure 4.4 (b), As same as in the square dimple, lower \( Sh/Sh_0 \) (about 0.7) and higher \( Sh/Sh_0 \) (1.60) are found in the upstream portion and downstream portion of the dimple bottom wall, respectively. In the wake of the dimple, the highest \( Sh/Sh_0 \) of 2.30 is observed; \( Sh/Sh_0 \) maintains 2.01 to 1.25 in one dimple length \( L \).
In circle dimple (Figure 4.4 (c)), lower $\text{Sh}/\text{Sh}_0$ (about 0.7) and higher $\text{Sh}/\text{Sh}_0$ (2.23) are found in the upstream half and downstream half of the dimple bottom wall, respectively. In the wake of the dimple, the highest $\text{Sh}/\text{Sh}_0$ of 2.07 is observed; and $\text{Sh}/\text{Sh}_0$ maintains 1.91 to 1.25 in one dimple length $L$. Note that the $\text{Sh}/\text{Sh}_0$ values are from 2.07 to 1.59 in the range of $L/2$ downstream the dimple edge, which are equivalent to some of the previous reported results obtained from the surfaces with hemispherical dimple rows, but less than some of higher reported values (about 2.5).

(a) Square Dimple

Figure 4.4: Experimental $\text{Sh}/\text{Sh}_0$ Contours (Figure Continued)
(b) Triangle Dimple

(c) Circle Dimple
For the teardrop dimple case (Figure 4.4 (d)), the scanned area on the dimple bottom wall is the largest among the four dimples. On the dimple bottom surface, lower $\text{Sh}/\text{Sh}_0$ (0.4 to 0.7) is found in a small area at the upstream end, and higher $\text{Sh}/\text{Sh}_0$ (3.2 to 4.2) is found at the downstream end close to the dimple edge. In the wake area of the dimple, the highest $\text{Sh}/\text{Sh}_0$ of 2.27 is observed. $\text{Sh}/\text{Sh}_0$ maintains 1.95 to 1.33 in one dimple length $L$ along the streamwise direction.

In order to compare the heat/mass transfer of the four dimples more clearly and more quantitatively, the spanwise average $\text{Sh}/\text{Sh}_0$ distributions for the four dimples are drawn together and displayed in Figure 4.5. The square, triangle and circle dimples clearly demonstrate almost the same spanwise average $\text{Sh}/\text{Sh}_0$, while the teardrop shows a noticeable higher $\text{Sh}/\text{Sh}_0$ distribution. As mentioned previously, the experimental
Sh/Sh₀ data are not available on the dimple downstream edges. This is because the measuring points (the scanning meshes) are intentionally designed to keep away from the dimple edges to prevent any damages on the dimple rims while scanning the naphthalene surfaces. Note that the data shown represent for the regions of one dimple length L along the streamwise direction in the wake of each dimple (i.e., in the upstream half of region 2 in Figure 7.1 of Chapter 7).

Figure 4.5: Comparisons of the Spanwise Average Sh/Sh₀ in the Wake of the Four Dimples

For a complete and specific assessment of the mass/heat transfer enhancement due to each single dimple, a better way is to define the dimple’s inner surface(s) as an individual region, and define the combination of the vicinity area and the wake of the dimple as another individual region, then evaluate the overall mass/heat transfer separately in the two regions. The definitions of the two regions are graphically shown in
Table 3, taking the circle dimple as an example. Note that this definition is slightly different from that of Isaev et al. [33] in Figure 7.1 (see Chapter 7). Table 3 also provides the overall area average $\frac{Sh}{Sh_0}$ in the region B (the meshed area) for the four dimples. This area average $\frac{Sh}{Sh_0}$ is calculated in the same way as did in Figure 4. That is, $\frac{Sh}{Sh_0}$ is first integrated on the region, and then divided by the area of the region. Table 3 clearly shows that the teardrop dimple demonstrates the best mass/heat transfer enhancement; its overall area average $\frac{Sh}{Sh_0}$ in the region B is 1.530, which is 25% higher than that of triangle dimple. It must also indicate that, even though the overall area average mass/heat transfer data for region A is not available in the present experimental study due to the lack of sufficient measuring points inside the dimples, the $\frac{Sh}{Sh_0}$ inside the teardrop dimple is still evidently observed higher than that inside the other three dimples. In conclusion, the experimental results attest that the teardrop dimple generates the greatest mass/heat enhancement.

Table 3: Overall Area Average $\frac{Sh}{Sh_0}$ in the Vicinity and in the Wake of the Four Dimples

<table>
<thead>
<tr>
<th>Region B</th>
<th>Square</th>
<th>Triangle</th>
<th>Circle</th>
<th>Teardrop</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area Average $\frac{Sh}{Sh_0}$</td>
<td>1.273</td>
<td>1.221</td>
<td>1.305</td>
<td>1.530</td>
</tr>
</tbody>
</table>
CHAPTER 5

NUMERICAL STUDIES ON FOUR SHAPES OF SINGLE DIMPLE

5.1 Computation Procedure Descriptions

The computation is performed with FLUENT Version 5. The channel and dimple geometries employed in the computation are exactly the same as used in the experiments. By considering the symmetry of the problem, only half of the channel and half of the dimple (cut by the x-z plane passing the centerline of the dimpled wall) are studied to save time and memory space of computer. The meshing is done by using GAMBIT. Different numbers of the grid nodes are tried to test the mesh independence. The final node numbers range from about 0.5 to 1.5 million depending on the dimple geometry. Fine meshes are generated inside the dimples, and around the rims of dimples to obtain detailed information of the flow fields and heat transfer behaviors.

The problem is modeled as a steady, 3-dimensional heat transfer process with uniform wall temperature. The Reynolds stress model (7 equations) and k-ω turbulence model are employed. During the computation, the three solid walls (the dimpled bottom wall, the smooth top wall, and the smooth side wall) are set to the same wall temperature. The two smooth side walls are assumed insulated. And the virtual wall in XZ plane containing the centerline is set as symmetrical wall.

The pressure of the airflow in the channel is selected as one atmosphere. The uniform inlet velocity $V_0$ is determined by selecting the inlet Reynolds number $Re = 21,000$, which is based on the hydraulic diameter of the channel. The temperature of the main flow is set to be 293 °K, and the wall temperature is set to be 323 °K. Because the
temperature difference is only 30 °K, and the main stream velocity $V_0$ is low (less than 10 m/s), the velocity field is assumed to be independent of temperature (no buoyancy or temperature dependent properties). Therefore velocity components and Nusselt numbers are calculated.

Note that, as shown previously in Figure 4.1, X and Y axes point to the streamwise direction and spanwise direction, respectively, and Z axis runs from the dimpled bottom wall, where dimple is located, to the smooth top wall of the channel. The origin of the (X, Y, Z) coordinates is defined as the middle point of the bottom edge of the cross section at the entrance, for both experiment and computation. So the domains of X, Y and Z are [0, 0.3048], [-0.0127, 0.0127], and [0, 0.0254] in meter, respectively. Or $X/ D_{h}$, $Y/ D_{h}$, and $Z/ D_{h}$ ($D_{h} = 1$ inch =0.0254 meter) have the ranges of [0, 12], [-0.5, 0.5] and [0, 1], respectively. All of the dimples start at $X/ D_{h} = 8.5$.

The vorticity components discussed below are normalized by $V_0/D_{h}$, where $V_0$ is the nominal velocity which is determined by $Re=21,000$. And the velocity components are normalized by $V_0$.

5.2 Experimental and Numerical Results Validations

The experimental procedure using the naphthalene sublimation method on the same apparatus has been validated by many previous researchers, such as Hibbs [40], Zhou et al [34], and Chen [41], so the present validations are focused on the numerical aspect. As mentioned above, the mesh independence is performed first for each dimple. Then, the numerical results are tried to compare with similar numerical data available in the published literature. However, as the author’s knowledge, there is no open numerical
result on square, triangle or teardrop dimples. For circle (hemispherical) dimple, only Isaev et al [32] provided heat transfer data which can be quantitatively compared with the present numerical work, as shown in Figure 5.1.

Figure 5.1: Area Average $\frac{Nu}{Nu_0}$ for Circle Dimples with Different Depth (Hollow symbol—Isaev et al. [33]; Solid symbol—the present study)

The average $\frac{Nu}{Nu_0}$ in Figure 5.1 is obtained by integrating $\frac{Nu}{Nu_0}$ separately on region 1 or region 2 for dimples with different depth, then dividing by the area of the region. The region 1 (circle symbol) is a square with edge of $d$ covering the whole dimple; the region 2 (square symbol) is a rectangle of $1.5d$ wide in spanwise and $2L$ (or $2d$, because $L=d$ for circle dimple) long in streamwise in the immediate downstream of the dimple. Both regions are shown in Figure 5.1 as meshed areas. Note that the circle dimples in Isaev et al. [33] are relatively shallow; the maximum $h/d$ is 0.22, while the $h/d$ in the present study is 0.3. The internal passage in Isaev et al. [33] “narrow” (the dimensions were not specified) channel rather than a square one, and the $d/D_h$ is smaller in Isaev et al. [33] than in the present study. However, regardless of these differences, the
average Nu/Nu₀ values are still reasonably close. For the present study, the average Nu/Nu₀ in region 1 is about 1.70, and it is 1.25 in region 2.

![Graphs showing spanwise average Nu/Nu₀ and Sh/Sh₀ in the wake of the four dimples](image)

Figure 5.2: Spanwise Average Nu/Nu₀ and Sh/Sh₀ in the Wake of the Four Dimples

The last step of validation is to compare the numerical and experimental results directly for each dimple. Figure 5.2 shows the comparisons of the numerical Nu/Nu₀ and experimental Sh/Sh₀, averaged along spanwise direction in region 2 at different streamwise locations for the four dimples. Apparently, the numerical and experimental
results match very well for all of the four dimples; the maximum error is less than 10%. Note that Figure 5.2 only shows the data in the upstream half of region 2 (1L long along streamwise direction). As a matter of fact, the data in the downstream half in the region 2 matches even better.

It needs to point out that, Figure 5.2 (a) and 5.2 (b) show that the maximum average $\frac{Nu}{Nu_0}$ are between 3 and 4 for square and triangle dimples, while the maximum average $\frac{Nu}{Nu_0}$ are between 2 and 3 for circle and teardrop dimples as shown in 5.2 (c) and 5.2 (d). This fact does not necessarily mean square and triangle dimples generate higher local or total heat transfer than circle and teardrop dimples. This observation will be discussed in the following sections. Also note that the experimental $\frac{Sh}{Sh_0}$ results in Figure 5.2 are not available on the downstream ends of the dimples. This observation was already explained in Chapter 4.

Figure 5.3: Vorticity and Streamlines in XY, XZ and YZ Planes with Square Dimple

(a) $Z/D_h = -0.04$

Figure 5.3: Vorticity and Streamlines in XY, XZ and YZ Planes with Square Dimple (Figure Continued)
(b) $Y/D_h = -0.122$

(c) $X/D_h = 8.834$
5.3 Numerical Results

5.3.1 Flow Structure

Figure 5.3 shows the flow patterns caused by square dimple, which is located within the volume of $X/D_h = [8.500, 8.894]$, $Y/D_h = [-0.197, 0.197]$, and $Z/D_h = [0, -0.118]$. As the flow goes down deeper into the dimple, i.e., below the channel bottom wall, second flow patterns are formed on $XY$ planes. At $Z/D_h = -0.04$ (1 mm below the channel bottom, and 2 mm above the dimple bottom) inside the dimple, a small flow circulation occurs at the corner between the inner vertical downstream wall and the sidewall, as shown in Figure 5.3 (a). The reversed flow pattern can also be seen on $XZ$ planes. A large flow circulation exists on the $XZ$ planes from $Y/D_h = 0.000$ to $Y/D_h = -0.165$ (close to the inner sidewall of the dimple). That means inside the dimple, a large part of fluid volume experiences flow circulation. The main flow impinges on the upper portion of the vertical downstream wall of the dimple, and then part of the flow is forced
to turn downwards and then to move backwards. Figure 5.3 (b) shows the flow circulations at $Y/D_h = -0.122$. Figure 5.3 (c) shows the vorticity and YZ streamlines at $X/D_h = 8.834$, where is inside the dimple, but very close to the downstream inner wall of the dimple. Flow is observed shedding off out of the dimple over the side edge and move to the channel side wall. And a circulating flow pattern is identified clearly in the corner between the dimple bottom surface and dimple side wall. This is the same vortex shown in Figure 8(a). In the wake of the square dimple at $X/D_h = 8.898$, where is almost right at the downstream transverse rim of the dimple, it is clearly shown in Figure 5.3 (c) that the flow rises up out from the dimple. The flow is then found to move downwards back to the channel bottom and reattach on the wall in the downstream at $X/D_h = 9.052$ and $X/D_h = 9.160$ (not shown here). No significant vortex rolls are observed in the wake of the square dimple.

Figure 5.4: Vorticity and Streamlines in XY, XZ and YZ Planes with Triangle Dimple (Figure Continued)
(b) $Y/D_h = -0.122$

(c) $X/D_h = 8.843$
Figure 5.4 shows the flow patterns caused by triangle dimple, which is located within the volume of $X/D_h = [8.500, 8.894]$, $Y/D_h = [-0.226, 0.226]$, and $Z/D_h = [0, -0.118]$. On the XY plane, a vortex is observed at $Z/D_h = -0.04$ in the corner between the inner sidewall and the inner downstream wall, as seen in Figure 5.4 (a). On the XZ planes, as $Y/D_h$ changes from 0 (symmetrical plane) to -0.122, the flow circulation always exists as shown in Figure 5.4 (b) for $Y/D_h = -0.122$, and the center of the circulation flow keeps rising up. The vortex can be seen even more clearly on YZ planes. The vortex becomes noticeable at $X/D_h = 8.726$; then the vortex keeps growing up. As it reaches to $X/D_h = 8.834$ (Figure 5.4 (c)), the vortex fill up the whole cross-section inside the triangle dimple. And the vortex also moves transversely toward to the channel side as it travels downstream along the X direction. Then the vortex rises up above the channel bottom surface and sheds off over the downstream rim from the corner of the dimple.
Figure 5.4 (d) shows the picture of the vortex flow in the wake of the dimple at $X/D_h = 9.160$). The vortex generated from within the dimple is clearly identified in YZ plane. Moreover, it induces another vortex at its left side. The induced vortex rotates at opposite direction. So for a triangle dimple, recalling the symmetrical nature of the problem, a pair of vortex rolls with opposite rotating directions emerges from each of the two downstream corners (only one is shown in the figures), and then the vortex rolls rise up and travel to the downstream.

Figure 5.5 shows the flow patterns caused by circle dimple, which is located within the volume of $X/D_h = [8.500, 8.894]$, $Y/D_h = [-0.197, 0.197]$, and $Z/D_h = [0, -0.118]$. A vortex is observed on XY plane at $Z/D_h = -0.04$, where is 1 mm below the channel flat bottom, as shown in Figure 5.5 (a). Then the vortex rises up, travels away for the symmetric plane at $Y/D_h = 0$, and moves towards the dimple edge. On XZ-plane, flow circulation is observed at $Y/D_h = 0$. This vortex is stronger at $Y/D_h = -0.039$ and $Y/D_h = -0.083$ (Figure 5.5 (b)), but as it moves away from the symmetric plane at $Y/D_h = 0.000$, the dimple becomes shallower; the circulation flow turns into weaker. On the YZ planes, flow circulations are also observed inside the circle dimple. But the most noticeable vortex patterns are found in the wake of the dimple. A full vortex clearly appears above the channel bottom surface in the wake of dimple at $X/D_h = 8.943$ (Figure 5.5 (c)). This vortex sustains at $X/D_h = 9.052$ (Figure 5.5 (d)) and in even further downstream. There are actually two vortices generated by the circle dimple, considering the dimple’s symmetry.
Figure 5.5: Vorticity and Streamlines in XY, XZ and YZ Planes with Circle Dimple
(Figure Continued)
(c) $X/D_h = 8.943$

(d) $X/D_h = 9.052$
Figure 5.6: Vorticity and Streamlines in XY, XZ and YZ Planes with Teardrop Dimple
(Figure Continued)
Figure 5.6 shows the flow patterns caused by teardrop dimple, which is located within the volume of $X/D_h = [8.500, 9.043]$, $Y/D_h = [-0.197, 0.197]$, and $Z/D_h = [0, -0.118]$. On XY and XZ planes, the majority of the flow inside the teardrop dimple is not
reversed. Only small region of flow circulation is observed in the area adjacent to the upstream portion of the dimple surface. The incoming flow inside the teardrop dimple, which enters into the dimple from upstream, directly impinges on a large part of the dimple surface, as shown in Figure 5.6 (a) and 5.6 (b). The gentle slope of the teardrop dimple’s upstream surface is the key behind the phenomenon. The gentle slope of the teardrop upstream surface facilitates the incoming flow to reach into the dimple deeper, to directly impinge on more dimple surface, and to reduce the flow reversed region. All of these factors will enhance the heat transfer on the dimple surface. This fact indicates the most significant difference between the teardrop dimple and the other three dimples with steeper surface slope. At $X/D_h = 8.933$, where are located within the downstream half of the dimple, a flow circulation occurs on YZ plane (Figure 5.6 (c)). As the flow travels downstream and leaves from the dimple, a full vortex clearly appears above the channel bottom surface in the wake of dimple, as shown in Figure 5.6 (d) for $X/D_h = 9.369$. There are actually two vortices induced by the teardrop dimple, considering the symmetry of the dimple. The vortex pair found in the teardrop dimple is similar to that in the circle dimple.

5.3.2 Heat Transfer

The $Nu/Nu_0$ distributions on the inner walls of square dimple and on the channel bottom wall are illustrated in Figure 5.7. First of all, check at the inner vertical walls in Figure 5.7 (b). The upper portion of the downstream inside wall shows $Nu/Nu_0$ value of 3.20 or higher. This higher $Nu/Nu_0$ is caused by the flow impingement. On the inner sidewall of the dimple, the downstream end shows a $Nu/Nu_0$ value of 1.54 due to the small flow circulation (Figure 5.3 (a)). And on the upstream wall, $Nu/Nu_0$ is lower than
unity due to the flow separation on the top of this wall. Note that, in the area close to the upstream angles between the upstream transverse wall and the inner sidewalls, $\frac{Nu}{Nu_0}$ drops to 0.4 or less, this fact may cause cooling failure. Figure 5.7 (a) shows the $\frac{Nu}{Nu_0}$ distributions on the dimple bottom wall and on the channel bottom wall. The peak value of $\frac{Nu}{Nu_0}$ occurs right at the rim of the downstream dimple edge. The peak $\frac{Nu}{Nu_0}$ value reaches to more than 4.80, but decreases quickly to about 2 within a range of $\Delta X$ less than 1 mm. Then $\frac{Nu}{Nu_0}$ maintains 1.68 to 1.18 in a distance of about one dimple length $L$ in the wake of the dimple. In a narrow band around the dimple perimeter, higher $\frac{Nu}{Nu_0}$ (about 1.35) is observed. On the dimple bottom wall, $\frac{Nu}{Nu_0}$ is less than unity on upstream half of the dimple bottom surface, as low as 0.4, but $\frac{Nu}{Nu_0}$ increases as $X/D_h$ increases. $\frac{Nu}{Nu_0}$ is greater than 1 on the downstream half of the dimple bottom surface, up to 1.33.

![Figure 5.7: Nu/Nu$_0$ Distributions of the Square Dimple (Figure Continued)](image-url)
The contours of numerical $\frac{Nu}{Nu_0}$ of triangle dimple are shown in Figure 5.8. The dimple bottom wall in Figure 5.8 (a) shows higher $\frac{Nu}{Nu_0}$ (about 2.30) in the downstream portion and lower $\frac{Nu}{Nu_0}$ (0.4 to 0.7) in the upstream portion. The flow circulation within the dimple causes such $\frac{Nu}{Nu_0}$ distributions. In Figure 5.8 (b), $\frac{Nu}{Nu_0}$ as high as 2.10 is observed on the inner side wall, the vortex roll shown in Figure 5.4 (a)
and 5.4 (b) attributes this local higher Nu/Nu0. The most part of the downstream inside wall shows Nu/Nu0 higher than 1.94 which is the results of the flow impingement. It is quite evident that the triangle dimple demonstrates better heat transfer than the square dimple on all of the inner walls. Note that, in the area close to the upstream angles between the two inner vertical walls, Nu/Nu0 drops to 0.4 or less. This lower heat transfer coefficient may cause cooling failure. Peak Nu/Nu0 values (higher than 4.80) are found again right on the downstream rim of the triangle dimple. But as same as in the square dimple, the peak values decreases to about 2 within a range of ΔX less than 1 mm. Then Nu/Nu0 maintains 1.74 to 1.25 in a distance of about one dimple length L in the wake of the dimple. In a narrow band around the dimple perimeter higher Nu/Nu0 (about 1.45) is also observed.

(a) Nu/Nu0 on the Bottom Walls

Figure 5.8: Nu/Nu0 Distributions the Triangle Dimple (Figure Continued)
The Nu/Nu<sub>0</sub> contours of the circle dimple are shown in Figure 5.9. The downstream half of the dimple bottom shows higher Nu/Nu<sub>0</sub> (about 2.07), and lower Nu/Nu<sub>0</sub> (about 0.7) is found in a small area in the upstream half. Nu/Nu<sub>0</sub> is about 1 to 1.05 in the most area on the dimple surface. Peak Nu/Nu<sub>0</sub> values (higher than 4.80) are found again right on the downstream rim of the dimple. The peak Nu/Nu<sub>0</sub> is found at the location on the edge with an angle of 58° to the X-axis. As same as in the square and triangle dimples, the peak values decreases to about 2 within a range of ∆X less than 1 mm. Then Nu/Nu<sub>0</sub> maintains 1.70 to 1.25 in a distance of one dimple length L in the wake. In a narrow band along the dimple perimeter higher Nu/Nu<sub>0</sub> (about 1.7) is also observed.
The Nu/Nu\(_0\) contours of the teardrop dimple are shown in Figure 5.10. The dimple bottom shows higher Nu/Nu\(_0\) (about 2.30) in the downstream half of the dimple, and shows lower Nu/Nu\(_0\) (about 0.7) in a small area in the upstream end. Nu/Nu\(_0\) is about 1 to 1.93 in the most area inside the dimple. Peak Nu/Nu\(_0\) values (higher than 4.80) are found again right on the downstream rim of the dimple. The peak Nu/Nu\(_0\) is found at the location on the edge with an angle of 33.5° to the X-axis. As same as in the square, triangle and circle dimples, the peak values decreases to about 2 within a range of ΔX less than 1 mm. Then Nu/Nu\(_0\) maintains 1.62 to 1.25 in a distance of one dimple length L in the wake. In a narrow band around the dimple perimeter higher Nu/Nu\(_0\) (about 1.62 or higher) is also observed.

In order to compare the numerical heat transfer results of the four dimples more clearly and more quantitatively, the spanwise average Nu/Nu\(_0\) distributions for the four dimples shown separately in Figure 5.1 are drawn together and displayed in Figure 5.11.
The square, triangle and circle dimples clearly demonstrate almost the same spanwise average $\text{Nu}/\text{Nu}_0$ except on the downstream dimple edges, while the teardrop shows a noticeable higher $\text{Nu}/\text{Nu}_0$ distribution. As mentioned previously, the higher numerical $\text{Nu}/\text{Nu}_0$ data for square and triangle dimples shown in Figure 5.11 doesn’t imply that square and triangle dimples generate higher local or overall heat transfer rates than circle and teardrop dimples. This is easy to understand when looking at the geometries of the four dimples. The square and triangle dimples have straight transverse downstream edges, all peak $\text{Nu}/\text{Nu}_0$ values occur along the transverse edges, so the spanwise averaging procedures at the downstream ends ($X/D_h=8.89$) have collected all of the peak $\text{Nu}/\text{Nu}_0$ values for square and triangle dimples, thus yielded higher average $\text{Nu}/\text{Nu}_0$. While the circle and teardrop dimples have curved downstream edges, and the peak $\text{Nu}/\text{Nu}_0$ values occur along the curved edges, not occur along transverse straight lines, so the spanwise averaging procedures on the downstream ends ($X/D_h=8.89$ for the circle dimple, and $X/D_h=9.04$ for the teardrop dimple) have not collected all of the peak $\text{Nu}/\text{Nu}_0$.
values, so the resulted average $\text{Nu}/\text{Nu}_0$ values are lower than that of square and triangle dimples.

![Figure 5.11: Comparisons of the Spanwise Average Nu/Nu₀ in the Wake of the Four Dimples](image)

Table 4 provides overall area average $\text{Nu}/\text{Nu}_0$ for the four dimples. The overall area average $\text{Nu}/\text{Nu}_0$ are separately evaluated in region A and region B, as defined in Table 3. Table 4 (a) shows the area average $\text{Nu}/\text{Nu}_0$ in region A. It is apparent that teardrop dimple has the greatest heat transfer enhancement (2.22) within its borders, in comparison with square dimple (0.824), triangle dimple (1.150) and circle dimple (1.642). Table 3 (b) further proves that teardrop dimple has the greatest heat transfer enhancement (1.416) in region B as well. Recalling the experimental results in Table 3 (shown again in Table 4 (b)), teardrop dimple also demonstrates the highest area average
Sh/\textit{Sh}_0\) values in region B. In conclusion, experimental and numerical studies both attest that teardrop dimple is of the best heat transfer enhancement among the four dimples. Table 4 (b) also shows the relative errors between the experimental and numerical results in region B. The maximum error is 7.4\%. The data further validates the reliability and accuracy of the present numerical study.

Table 4: Area Average \(\text{Nu/Nu}_0\) of the Four Dimples

(a) Area Average \(\text{Nu/Nu}_0\) on the Inner Surfaces

<table>
<thead>
<tr>
<th>Region A</th>
<th>Square</th>
<th>Triangle</th>
<th>Circle</th>
<th>Teardrop</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area Average (\text{Nu/Nu}_0)</td>
<td>0.824</td>
<td>1.150</td>
<td>1.642</td>
<td>2.22</td>
</tr>
</tbody>
</table>

Area Average \(\text{Nu/Nu}_0\) and \(\text{Sh/Sh}_0\) in the Vicinity and Wake

<table>
<thead>
<tr>
<th>Region B</th>
<th>Square</th>
<th>Triangle</th>
<th>Circle</th>
<th>Teardrop</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area Average (\text{Nu/Nu}_0)</td>
<td>1.325</td>
<td>1.277</td>
<td>1.287</td>
<td>1.416</td>
</tr>
<tr>
<td>Area Average (\text{Sh/Sh}_0)</td>
<td>1.273</td>
<td>1.221</td>
<td>1.305</td>
<td>1.53</td>
</tr>
<tr>
<td>Error %</td>
<td>-4.10</td>
<td>-4.59</td>
<td>1.36</td>
<td>7.40</td>
</tr>
</tbody>
</table>
CHAPTER 6

CONCLUSIONS

Experimental and numerical studies of heat/mass transfer on the dimpled surfaces in square channels are performed. The naphthalene sublimation method is employed in the experimental study. First, in a two-passage rotating channel, circle dimple matrices are cast on the leading and trailing walls in both the inlet and outlet of the serpentine channel. The experiments are carried out with Reynolds number of 7,000 and 21,000, and with Rotation number of 0 and 0.2. Secondly, four single dimples with printed shape on the wall of square, triangle, circle, and teardrop are studied experimentally in the same test section at stationary with Reynolds number of 21,000. The four dimples are also studied numerically using FLUENT. The major results are summarized as following:

6.1 Heat Transfer in the Rotating Internal Cooling Channel with Dimple Walls

The maximum mass/heat transfer rates are obtained downstream of the dimples. The minimum mass/heat transfer rates occur along the row containing the dimples.

1. The dimpled walls lead to enhancements over smooth surfaces both in the stationary case and in the rotating case. In the stationary case, the dimple enhancement factor for Re=7,000 is about 2, but is slightly less than 2 for Re=21,000. For Ro=0.2, the dimple enhancement factor is at the order of 2 for both Re=7,000 and 21,000.

2. The present results show that enhancements achieved with dimples are larger in the inlet passage compared to the outlet passage for Re=21,000.

3. The Sherwood number distributions suggest the existence of three local peaks, with the strongest peak immediately downstream the dimples. These peaks appear to be related to the development
of streamwise vortex structures generated from the dimples. The Sherwood number contours also support these observations.

6.2 Heat Transfer and Fluid Structure on the Walls with Single Dimples in the Stationary Internal Cooling Channel

1. Flow patterns are identified by numerical studies for the four dimples. Two pairs of vortex rolls are found in the wake of triangle dimple. One pair of vortex rolls is observed in the wake of circle dimple and teardrop dimple. No noticeable vortex roll is observed in the wake of square dimple.

2. High Nu/Nu<sub>0</sub> values are found within narrow bands on the downstream edges of the four dimples. The peak values are more than 4.80.

3. Inside the dimples, lower Nu/Nu<sub>0</sub> and higher Nu/Nu<sub>0</sub> are found on the upstream portion and on the downstream portion of the dimple bottom, respectively. The experimental Sh/Sh<sub>0</sub> shows the same behaviors. In the area close to the upstream angles between the vertical inner walls inside the square and triangle dimples, Nu/Nu<sub>0</sub> drops to 0.4 or less, this fact may cause cooling failure.

4. Around the dimple perimeters, Nu/Nu<sub>0</sub> is in a range of 1.35 to 1.7. In a distance of one dimple length in the wake of the dimples, Nu/Nu<sub>0</sub> can maintain from about 2 to 1.2.

5. The maximum error between the area average experimental Sh/Sh<sub>0</sub> and numerical Nu/Nu<sub>0</sub> is 7.4%.

6. The experimental and numerical results both prove that the teardrop dimple is of the best heat/mass transfer among the four dimples covered in this study.
REFERENCES


VITA

Fuguo Zhou was born in Hunan Province in China. He received the Bachelor of Science in Aerospace Engineering (Jet Propulsion) from Beijing University of Aeronautics and Astronautics, Beijing, China, in 1985; and he received the Master of Science in Aerospace Engineering (Rocket Propulsion) from Beijing University of Aeronautics and Astronautics in 1988. Then he worked as an instructor in a university in Wuhan, Hubei Province, and studied one year in Purdue University, West Lafayette, Indiana. He then joined the graduate program at Louisiana State University, Baton Rouge in fall, 1999. He worked as a research assistant in Turbine Innovation and Energy Research (TIER) Center on experimental and numerical studies on internal cooling in turbine blades. He is a candidate for the degree of Master of Science in Mechanical Engineering to be awarded at the commencement of summer 2007. Simultaneously, he is a doctoral candidate working in the same field and to be graduated by the end of 2007.