PHYSIO THERMODYNAMICS OF LUBRICANT APPLICATION TO HOT DIE SURFACES

DISSERTATION

Presented in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy in the Graduate School of The Ohio State University

By

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* * *

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ABSTRACT

The hot forging process is one of the manufacturing processes where metal is pressed, pounded or squeezed under great pressure into high strength parts. Before the billet is forged, lubricant is applied to heated dies to promote the flow of metal, to reduce friction and wear, and to aid in the release of the finished part. The most commonly used lubricant in the hot forging process is water-based graphite and it is applied to dies by spraying large numbers of atomized lubricant droplets through nozzles. An understanding of film formation and heat transfer of a single droplet colliding on a heated wall will result in more precise control of spray to obtain required film patterns and heat transfer. The precision of spray control will, in turn, reduce energy consumption and pollution in the hot forging process through reduction of billet pre-heat temperature and enhancement of lubricant wetting with the die.

Interactions between the die surface and droplets of water based graphite lubricant are the subjects of this study. First, the spreading of droplets which impact on the die surface is modeled based on Navier-Stokes equations and this model is solved using a commercial CFD code, Flow-3D. Special attention is paid to the influence of lubricant dilution ratio, lubricant properties, droplet parameters, and die surface temperature on the formation of the lubricant film in this model. A non-dimension analysis of the simulation results
provides an insight into the physical phenomena and the model is validated through controlled experiments using a high speed camera.

The lubricant film is formed after all the residue liquid on the heated die surface is dried off. The evaporation time of the droplets is not only a productivity index in hot metal processing but also an important factor for product quality and life of the die. This study presents an analytical model to predict the evaporation time of a droplet impacted on a die surface, heated to temperatures beyond Leidenfrost point, beyond which are the working temperatures for hot forging dies. Calculated evaporation times from the model are compared with experimental measurements and agree well in a right magnitude.

A methodology for calculating the heat transfer coefficients in lubricant sprays is then presented. The heat transfer characteristics of the lubricants at various thermo-physical regimes on the hot surface are determined using single droplet experiments. The results of these experiments are used along with equations based on a non-dimensional and averaging approach to predict heat transfer coefficients for a complicated spray. Finally, the outcomes of dynamic interactions between lubricant droplets and heated die surface are categorized as bounce off, wetting and breakup, based on the coupling of Weber number \((We)\) and die surface temperature. These characteristic outcomes of droplets with different dilution ratio are mapped through the experimental results.
Predictions by both the fluid dynamic model and the dryoff model agree well with the results obtained in experiments. Results from single droplet studies show that both Weber number ($We$) and die surface temperature play an important role on the lubricant wetting, dryoff and heat transfer on the die. Selection of lubricant dilution ratio has also been shown to be important on interactions between sprayed lubricants and the die surface.
Dedicated to my parents and my wife.
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First and foremost I would like to thank Dr. Rajiv Shivpuri, my adviser, for his support, guidance and above all, encouragement throughout the years of my PhD program. I have learned knowledge and the way to obtain knowledge under his instructions that has prepared me to make more contributions to the future. I am also grateful in earnest for his continuous solicitude for the hard times my family experienced during my doctoral study.

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<th>Definition</th>
<th>Unit</th>
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<tr>
<td>$A$</td>
<td>areas ($m^2$) / fractional area ratio</td>
<td></td>
</tr>
<tr>
<td>$b$</td>
<td>body force acceleration ($m/s^2$)</td>
<td>pressure ($N/m^2$)</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat ($W/kg K$)</td>
<td></td>
</tr>
<tr>
<td>$d$</td>
<td>droplet diameter ($m$)</td>
<td></td>
</tr>
<tr>
<td>$\bar{D}$</td>
<td>rate of deformation tensor</td>
<td></td>
</tr>
<tr>
<td>$F$</td>
<td>fluid fraction function</td>
<td></td>
</tr>
<tr>
<td>$g$</td>
<td>gravity acceleration ($m/s^2$)</td>
<td></td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient ($W/m^2/K$)</td>
<td></td>
</tr>
<tr>
<td>$H$</td>
<td>height ($m$)</td>
<td></td>
</tr>
<tr>
<td>$H_f$</td>
<td>latent heat of fusion ($W/kg$)</td>
<td></td>
</tr>
<tr>
<td>$H_v$</td>
<td>latent heat of vaporization ($W/kg$)</td>
<td></td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity ($W/m$)</td>
<td></td>
</tr>
<tr>
<td>$L_p$</td>
<td>latent heat of pressure change</td>
<td></td>
</tr>
<tr>
<td>$\bar{L}$</td>
<td>velocity gradient tensor</td>
<td></td>
</tr>
<tr>
<td>$m$</td>
<td>friction coefficient</td>
<td></td>
</tr>
<tr>
<td>$\hat{n}$</td>
<td>unit normal vector</td>
<td></td>
</tr>
<tr>
<td>$p$</td>
<td>pressure ($N/m^2$)</td>
<td></td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
<td></td>
</tr>
<tr>
<td>$r$</td>
<td>radial displacement</td>
<td></td>
</tr>
<tr>
<td>$R$</td>
<td>radius ($m$)</td>
<td></td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
<td></td>
</tr>
<tr>
<td>$s$</td>
<td>arc length ($m$)</td>
<td></td>
</tr>
<tr>
<td>$Ste$</td>
<td>Stefan number</td>
<td></td>
</tr>
<tr>
<td>$t$</td>
<td>time ($s$)</td>
<td></td>
</tr>
<tr>
<td>$\hat{t}$</td>
<td>unit tangent vector</td>
<td></td>
</tr>
<tr>
<td>$T$</td>
<td>temperature ($K$)</td>
<td></td>
</tr>
<tr>
<td>$v$</td>
<td>velocity ($m/s$)</td>
<td></td>
</tr>
<tr>
<td>$V_f$</td>
<td>fractional volume ratio</td>
<td></td>
</tr>
<tr>
<td>$V_0$</td>
<td>initial droplet velocity ($m/s$)</td>
<td></td>
</tr>
<tr>
<td>$We$</td>
<td>Weber number</td>
<td></td>
</tr>
<tr>
<td>$z$</td>
<td>axial displacement</td>
<td></td>
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</table>
\[ \begin{align*} \alpha & \quad \text{contact angle/ thermal diffusivity} \\ & \quad (\text{m}^2/\text{s}) \\ \gamma & \quad \text{surface tension (N/m)} \\ \delta & \quad \text{vapor layer thickness (\text{\textmu m})} \\ \theta & \quad \text{angular displacement} \\ \kappa & \quad \text{curvature of the surface} \end{align*} \]

\[ \begin{align*} \mu & \quad \text{dynamic viscosity (N s/m}^2) \\ \nu & \quad \text{kinematic viscosity (m}^2/\text{s}) \\ \xi_{\text{max}} & \quad \text{spreading coefficient} \\ \rho & \quad \text{density (kg/m}^3) \end{align*} \]

\[ \begin{align*} \overline{\tau} & \quad \text{sheer stress tensor} \end{align*} \]

\begin{itemize}
\item \( \delta \) \quad \text{spacing of a distance or time}
\end{itemize}

\textbf{Subscripts}

\begin{itemize}
\item D \quad \text{die surface}
\item i \quad \text{the i-th in } r \text{ direction}
\item j \quad \text{the j-th in } z \text{ direction}
\item m \quad \text{melt}
\item n \quad \text{the n-th time level}
\item r \quad \text{r direction}
\item w \quad \text{wall}
\item z \quad \text{z direction}
\item \theta \quad \text{\( \theta \) direction}
\item \max \quad \text{maximum value}
\item 0 \quad \text{initial value}
\end{itemize}
CHAPTER 1

INTRODUCTION

The forging process is one of the manufacturing processes where metal is pressed, pounded or squeezed under great pressure into high strength parts. The forging industry in the U.S. was roughly $12 billion in 1997 (DOD National Security Assessment). Roughly 15% of the total sales value of its products is allocated to energy, an estimated 50 trillion BTUs per year. The environmental impact of the use of graphite (in hot forging) is estimated to add over 20% to the product cost. These costs include both direct costs related to cleaning and disposal and indirect costs related to immediate and long-term health costs of employees.

To improve energy efficiency and the working environment in hot forging processes, we consider lubrication as the key strategy to achieve this goal. The hot forging process is performed by preheating the metal to a desired temperature (above its recrystallization point) before it is worked. The billet is forged only after lubricant is applied to dies to promote the flow of metal, to reduce friction and wear, and to aid in the release of the finished part. The most commonly used lubricant in the hot forging process is water-based graphite and it is applied to dies by spraying large numbers of atomized lubricant droplets through nozzles. The mode of spray not only influences lubricant wetting on the
die, but also determines the thermal effects on the die surface. An understanding of film formation and heat transfer of a single droplet colliding with a heated wall will result in more precise control of spray to obtain required film patterns and heat transfer. The precision of spray control will thus reduce energy consumption and pollution in the hot forging process through reduction of billet pre-heat temperature and enhancement of lubricant wetting with the die.

1.1 Project Background

Fig. 1.1 is a flow chart of a typical hot forging process.

(1) Billet Heating
(2) Die Heating
(3) Lubricant Application
(4) Billet Transported on the Die.
(5) Forge Pressing
(6) Part Ejection
(7) Part Cooling
(8) Finishing, heat treatment and other post-forging processes
Fig. 1.1. A typical hot forging process flow.
As one can see from Fig. 1.1, energy can be spent in direct heating of forge stock to required temperatures, operation of press and other equipment, subsequent normalizing and cleaning of forgings. For every pound of steel forged, close to 2000 - 4000 BTU of energy is used (based on an average specific heat of about 1-3 BTU/lb/K). Heating is the biggest source of energy consumption in forging industry. Because of inefficiencies in heating and energy consumed in other areas of a forge shop, heating could consume over 6000-8000 BTU / lb of steel. Similar numbers can be expected in non-ferrous forging also. Typical scrap rate in forge shops that forge to conventional tolerances ranges anywhere from 5000-10000 ppm. For near net forgings the scrap rate is close to 50,000 ppm. Scrap is associated with lack of process control and degradation of tool that is not identified in time. Based on a 10,000 ppm scrap rate and a 20% improvement in scrap rate related to premature failure (which may be 20-30% of all scrap), this translates to about 35 BTU per lb of steel forged.

Obviously, most of the pollution in hot forging happens at the lubricant application stage. The following study will focus on analysis of the influence of lubrication on energy consumptions.

1.1.1 Energy Analysis of a Typical Hot Forging Process

A close examination of each of the hot forging step can reveal how energy flows, see Table 1.1. Energy flow can be sketched as seen in the flow chart in Fig. 1.2.
For spike forging, heating a billet (8620 steel) with 15mm radius and 20mm height from 25°C to 1100°C consumes about 56 kJ, and forging the billet consumes 3 kJ. The energy containing in the initial billet is about 26 kJ. When the initial billet heating temperature decreases to 900 °C, the heating process consumes 46 kJ, and the forging process consumes 6 kJ. When the scrap rate decreases from 10000 ppm to 5000 ppm, the energy saving in each part will be 13 kJ. As Fig. 1.3 and 1.4 show, billet heating consumes the largest part of energy in hot forging, almost 69%. The decreasing of scrap rate which saves the energy for steel making is another important part (almost 20%). All other processes such as material handling, spray and cleaning, etc. consume no more than 2 kJ.

1.1.2 Influence of Lubrication on Energy Consumption

As identified in Fig. 1.4, heating and scrap consume most of the energy applied to the hot forging process, not including energy wasted in the pre-forging process, i.e. steel making, because of forging scrap. If we can manage to reduce heat input to the process and scrap rate, energy consumed can be dramatically dropped. We will show that an advanced lubrication technology can be developed to fulfill this target in the following analysis.
### Steps Does Energy Change ($\Delta E$)? And How?

<table>
<thead>
<tr>
<th>Steps</th>
<th>Does Energy Change ($\Delta E$)?</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1. Billet Heating</strong></td>
<td>• Billet heated from room temperature to as high as 1100°C. Energy consumed.</td>
</tr>
<tr>
<td></td>
<td>• Scale generated. Energy loss.</td>
</tr>
<tr>
<td><strong>2. Die Heating</strong></td>
<td>• Die heated from room temperature to 200°C. Energy consumed.</td>
</tr>
<tr>
<td></td>
<td>• Scale generated. Energy loss.</td>
</tr>
<tr>
<td><strong>6. Part Ejection</strong></td>
<td>• Part is ejected from the die. Energy consumed.</td>
</tr>
<tr>
<td><strong>7. Part Cooling</strong></td>
<td>• Part is cooled down to room temperature. Energy consumed.</td>
</tr>
<tr>
<td><strong>8. Post-Forging Processes</strong></td>
<td>• Part is machined to achieve required final dimension or heat-treated to mechanical properties. Energy consumed.</td>
</tr>
</tbody>
</table>

Table 1.1. Forging process and energy flow
Fig. 1.2. Energy flow of the hot forging process
Fig. 1.3. Energy analysis for a spike forging process.
Energy Contributions in Hot Forging

- Heating: 68.66%
- Scraps rate: 19.40%
- Forging: 8.96%
- Other: 2.99%

Fig. 1.4. Energy drivers for a spike forging process.
Fig. 1.5. Schematic of the cavity filling.
During spike forging, the metal flows in two directions, one is the flash direction and the other is the direction to fill the cavity. Generally metal flows more easily into flash because of less resistance. Under-fill of the cavity produces majority of the scrap. In order to fill the cavity, the typical method in industry, is to reduce the “h” in Fig. 1.5 and make more flash. More flash means more volume of the initial billet, and hence more heating energy. Moreover, flash also increases the press load, and this may result in the die life reduction and increase in the energy consumed by the press punch.

We can find a way to reduce the flash, pre-heated temperature of the billet and press load using the tribology theory. By controlling lubricant deposition patterns to increase the metal flow resistance on some necessary positions on the die surface, while reducing resistances on other positions, more metal can be forced to flow into the cavity. From the analysis of the FEM simulation results, we can infer that this technology is feasible in the energy saving.

We selected AISI 8620 as the billet material. The diameter is 40 mm and the height is 20 mm. Only half of the billet is shown in Fig. 1.6 due to symmetrical geometry.

First, the billet is heated to 1100 °C, a typical hot forging temperature for AISI 8620. We used the flood spraying of the graphite-based lubricant, which is the conventional lubrication deposition method in industry, under which the friction factor \( m \) is assumed to be 0.3 wherever the billet metal contacts the die. After the punch moves down by 15 mm at a speed of 25 mm/sec, the cavity filling is 8.12 mm with a load of 410 kN, as shown in
Fig. 1.7. This means that if we use conventional lubricant deposition, the capacity of the cavity filling for the designed cavity is 8.12 mm under the specified forging conditions.

Controlled lubricant was used by increasing the metal flow resistance on the punch surface by setting the friction factor of 0.7, and setting the friction factor as 0.1 on the cavity surface. The simulation is then run under the same forging conditions as Fig. 1.7. The result is shown in Fig. 1.8 and the filling length increases to 15.09 mm. Comparing this result with Fig. 1.7, we can see that the filling capacity increase a lot using this innovational lubricant deposition method.

The filling length is fixed at 8.12 mm and simulations have been done at billet temperatures of 1000°C and 900°C for both flood spraying and controlled lubrication. The forging load is calculated and results are summarized in Table 1.2.

From Table 1.2, we can see that if one uses the innovational lubricant deposition method to achieve the same cavity filling as the conventional method, the billet temperature can be reduced from 1100°C to 900°C at the same load. Thus, around 15% of heating energy can be saved. This controllable lubrication technology can hence reduce by a large amount of the energy usage in the hot forging process.
Fig. 1.6. The initial configuration of the billet upsetting simulation.
Fig. 1.7. Temperature= 1100°C, m=0.3, H=8.12 mm. Flood spraying.
Fig. 1.8. Temperature = 1100°C, m1=0.7, m2=0.1; H=15.09 mm. Controlled lubrication.
The load (kN) when the filling reaches 8.18 mm

<table>
<thead>
<tr>
<th>Billet Temp. (C)</th>
<th>Conventional</th>
<th>Innovational</th>
</tr>
</thead>
<tbody>
<tr>
<td>1100</td>
<td>410</td>
<td>235</td>
</tr>
<tr>
<td>1000</td>
<td>570</td>
<td>300</td>
</tr>
<tr>
<td>900</td>
<td>810</td>
<td>440</td>
</tr>
</tbody>
</table>

Table 1.2. Simulation results (Load in kN)
1.1.3 Technological Strategy to Reduce Energy

As discussed above, technologies of innovative controllable lubrication introduce significant energy reduction in hot forging, without considering indirect saving such as die life improvement. However, two critical questions come up:

- What is the optimized lubrication pattern which produces most energy saving?
- How do we apply the lubricant on dies to obtain desired lubricant patterns?

These two questions compose our energy saving strategy as shown in Fig. 1.9. Three parts are related to each other in this strategy:

- Setup desired film patterns;
- Establish relationship between film patterns/ heat transfer and process parameters/ lubricant properties;
- Optimize lubricant process to achieve our energy saving and pollution reduction goals.

This strategy is broken down to detailed tasks on which a road map is based.

This research has been focused on the second part of the strategy, i.e., to find relationship influences of the lubricant properties and the method of lubricant application on the film formation and thermal behavior when the lubricant is applied to the die.
Fig. 1.9. Energy saving strategy for hot forging based on innovative lubrication technology.
1.2 Objective of the Research

In order to obtain required lubricant patterns and thermal management through a controlled spray process, the objective of this research is to build models that relate the formation of lubricant film and heat transfer phenomena of the die to lubricant properties and lubricant application characteristics.

A lubricant spray consists of tens of thousands of droplets which interact with the die, forming a lubricant film and taking away heat of the die. The objective of this research can be achieved by starting from droplet studies in which we focus on physical phenomena are focused. The single droplet studies reveal physical mechanisms of the lubricant spray and formed the basis of the selection of the spray parameters. After the droplet mechanism has been understood, the lubricant film formation and heat transfer of the spray can be predicted using knowledge of the spray generation. This spray modeling process is shown in Fig. 1.10.

A part of the objective of this research is also to validate the adequacy of the models. Extensive experimental work is needed to achieve this goal. The results from the experimental measurements are compared to the corresponding output from the analytical and simulation models to verify that the models adequately represent the actual lubricant droplet-die interaction process.
1.3 Approach and Methodology of the Research

In order to develop the required models and validate them, the research is divided into two parts, the analytical/simulation part and the experimental part. The analytical part is to build models through governing equations and solve them numerically or analytically. In the experimental part, measurements of the film formation and heat transfer are collected in order to validate the mathematical models and to build an empirical heat transfer coefficient model. These two parts are interwoven with each other and the final models are based on both the parts.

The first model that this study has developed is the isothermal model for lubricant spread on the die surface. This model aims at investigating the dynamics and spread of a lubricant droplet on the hot die surface. The model starts with general mass and momentum equations. The second step is to make justified assumptions pertaining to the problem. The initial and boundary conditions of the model are then identified. After the problem is defined, the general equations are simplified based on the assumptions and conditions. Simplified equations are then numerically solve.
Fig. 1.10. Spray modeling processes.
The second model developed is the dryoff model in which the die surface is heated to its working temperature (temperatures beyond so-called Leidenfrost point of the lubricants). The assumption of the model is that the lubricant droplet begins to evaporate only after the droplet deformation is stabilized and the kinetic energy has been completely consumed. The mass and momentum balance equations are simplified and analytically solved in the modeling. The dryoff times are predicted from the model for different droplet impacts and compared with experimental results.

The experimental section of the research aims to validate the modeling criteria that are used in the analytical/simulation modeling parts. Two categories of experimental data are collected and compared to the analytical/simulation results. The first category is related to the film formation. The second category of experimental data is related to heat transfer. The experiments of heat transfer are conducted to build an empirical heat transfer coefficient model and to map regions of physiothermodynamics of lubricant deposition on the hot die.

Both parts of the above research are carried out using a non-dimensional concept. For example, droplet diameter and velocity, the most dominant factors, are expressed by Reynolds number and Weber number. Thus, instead of using the micron scale of droplet size, we conduct experiments and simulations based on a millimeter scale dimension, which is much more convenient to handle and accurate to measure.
After droplet models are built, they are used to predict the heat transfer of a spray. The rationality of the prediction is tested by the controlled spray experiments for lubricant.

1.4 Dissertation Outline

A brief description of each chapter in this dissertation is given below:

Chapter 1 gives a brief introduction and outlines the rationale and motivation behind this work along with the objectives and the research approach.

Chapter 2 contains background information about the general form of governing equations and their relevance to the droplet study. This chapter also includes a thorough review of previous studies which have been carried out in the past.

Chapter 3 presents the isothermal numerical model based on the VOF finite difference approach for investigation of the dynamics and spread of the lubricant droplet on the hot die surface.

Chapter 4 presents an analytical model to estimate the evaporation time of a droplet impact on a die surface heated to temperature beyond Leidenfrost point.
Chapter 5 studies experimentally the effect of lubricant properties, dilution ratio, spraying parameters and the transient nature of the hot surface on the heat transfer. The study of droplets is extended to spray applications. A case study has been done to predict heat transfer coefficient of a complicated spray.

Chapter 6 presents outcomes of the droplet-surface interaction based on experimental studies. This chapter also presents results of heat flux in spray tests with graphite suspensions in water. The experimental results are included to demonstrate an approach to study the effect of the lubricant properties, dilution ratio, spraying parameters on the pollution and energy consumption.

Chapter 7 summarizes research objectives and contributions of this work and presents the proposed scope for future work.
CHAPTER 2

BACKGROUND AND LITERATURE REVIEW

Widespread use of spray in industrial applications such as spray combustion, spray drying, spray coating, injection printing, etc., has stimulated many experimental and analytical studies associated with droplet and sprays. Generally, there are three principal areas among these studies. The first area is droplet or spray generation. The generation of droplets in most industrial applications involves the breakup of liquid, i.e., atomization. Atomization is probably the most widely used process for droplet generation [1]. In the forging industry, lubrication sprays are produced by discharging diluted lubricant through a nozzle, or pressure atomizer, through which the applied pressure is converted into kinetic energy and surface energy of the droplet. Other atomizers such as rotary atomizer [2], ultrasonic atomizer [3] and electrical droplet generator [4] are also encountered in industry. The second area is aerodynamics and evaporation of droplets suspended in air or chambers. Studies [5-6] in this area are extensively conducted related to internal combustion engines because the control of the droplet size and dispersal of the droplets in the combustion chamber is an important phenomenon. The third area concentrates on droplet deformation and heat transfer or phase changes on a surface. Fields such as spray
painting, spray coating and spray cooling are all related to this research area. Lubrication spray in hot forging is one of the third category fields that need knowledge about deformation and heat transfer of droplets impacting on a surface.

Lubrication in hot forging starts by the insertion of spray nozzles between the upper and lower dies just after a formed part is taken from the dies. Lubricant is sprayed to required thickness on the dies. The next billet can be seated on the die only after the dies are cooled to the desired temperature, i.e., 300-350°C for H13 dies, and the lubricant completely dries off. The time needed to cool the dies and deposit lubricant normally takes the majority of each forging cycle, which varies from several seconds to tens of seconds depending on the part size. Hence the efficiency of cooling and deposition greatly influences the productivity of the forging process. As schematically shown in Fig. 2.1, there are many factors which influence die cooling and lubricant deposition for a given lubricant spray duration. The factors are categorized into two groups: die surface properties and spray properties. The die surface properties include the initial temperature, roughness and die geometry. The spray properties involve droplet size, droplet velocity, dilution ratio or solid content, spray cone angle and nozzle height, just to name a few.

Therefore, we need to estimate and predict the effects of changing spray parameters such as droplet size, impaction velocity, etc., and lubricant properties on the deposition patterns and heat transfer during the sprays on the die. Analysis of single droplet impact on a hot flat die is a fundamental study to fulfill this purpose.
Hot Die (300-350°C Desired)
Lubricant Spray
Deposited Graphite
Deposition Mass (mg/m²)
Spray Time (s)
Die Surface Temperature (°C)
Spray Properties
Surface Properties
Fast Die Cooling
Fast Lubricant Deposition & Drying
Fig. 2.1 Factors influencing lubricant deposition and die cooling.
Thanks to the advances in computational power and commercial CFD codes, research of droplet in fields such as combustion engine, inkjet printing and thermal deposition, has made a lot of progress during the last few decades. However, very little research has been done in the field of hot forging and the fundamental aspects of the problem have received very little attention by the researchers in this field.

In modeling lubricant droplet in a hot forging process, many issues have to be addressed. Among these issues, the two most important are droplet flow at impaction, and thermal analysis of the cooling effect. In this chapter, we will first review the governing equations of the fluid flow and heat transfer, and then the previous studies on droplet-surface interactions. Studies on spray cooling are also reviewed.

2.1 Governing Equations

Fluid flow modeling is used to predict flow patterns when a droplet collides with a die so that we may obtain a better insight into the fluid flow behavior of the lubricant, thus improving its deposition. This has a significant impact on the metal flow in metal forming due to controlled lubrication, and therefore achievement of better quality parts and longer life dies. The conservation of mass and conservation of momentum are always applied as general governing equations for a droplet impacting on a wall.
One of the important roles of the lubricant is to chill down the temperature of the die so that the die maintains required hardness and toughness. When sprayed on the die surface, which usually reaches a temperature as high as 700°C, the lubricant takes heat from the die by conduction, convection and radiation, the three mechanisms of heat transfer. Large amount of heat loss is also caused by the phase change of the lubricant too. When the temperature of the die climbs over the so called Leidenfrost point, the area of the heat transfer surface is covered with a stable blanket of steam, preventing contact between the heat transfer surface and the spreading liquid. This is discussed in detail in chapter 3.

- Conservation of mass (Continuity Equation)

The mass can possibly be converted from one phase to another, but it is neither generated nor consumed. The conservation of mass is governed by the following equation:

For rectangular coordinates:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho v_x) + \frac{\partial}{\partial y}(\rho v_y) + \frac{\partial}{\partial z}(\rho v_z) = 0$$  \hspace{1cm} (2.1)

For cylindrical coordinates:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r}(\rho rv_r) + \frac{\partial}{\partial \theta}(\rho rv_\theta) + \frac{\partial}{\partial z}(\rho v_z) = 0$$  \hspace{1cm} (2.2)

When the density is assumed to be constant, we have:
For rectangular coordinates:

$$\frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} + \frac{\partial v_z}{\partial z} = 0$$  (2.3)

For cylindrical coordinates:

$$\frac{1}{r} \frac{\partial}{\partial r} (rv_r) + \frac{\partial v_\theta}{\partial \theta} + \frac{\partial v_z}{\partial z} = 0$$  (2.4)

- **Conservation of momentum**

All fluid flow must follow Newton’s second law of motion. Considering the surface force and body force exerted on the fluid, the general momentum balance equations in terms of Newton’s second law of motion are:

For rectangular coordinates,

$$\rho \left( \frac{\partial v_x}{\partial t} + v_x \frac{\partial v_x}{\partial x} + v_y \frac{\partial v_x}{\partial y} + v_z \frac{\partial v_x}{\partial z} \right) = -\frac{\partial p}{\partial x} + \left( \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \right) + \rho b_x$$  (2.5a)

$$\rho \left( \frac{\partial v_y}{\partial t} + v_x \frac{\partial v_y}{\partial x} + v_y \frac{\partial v_y}{\partial y} + v_z \frac{\partial v_y}{\partial z} \right) = -\frac{\partial p}{\partial y} + \left( \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} \right) + \rho b_y$$  (2.5b)

$$\rho \left( \frac{\partial v_z}{\partial t} + v_x \frac{\partial v_z}{\partial x} + v_y \frac{\partial v_z}{\partial y} + v_z \frac{\partial v_z}{\partial z} \right) = -\frac{\partial p}{\partial z} + \left( \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} \right) + \rho b_z$$  (2.5c)
For cylindrical coordinates,

\[
\rho \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + v_\theta \frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \tau_{rr} \right) + \frac{1}{r} \frac{\partial \tau_{r\theta}}{\partial \theta} - \frac{\tau_{\theta\theta}}{r} + \frac{\partial \tau_{rz}}{\partial z} \right) + \rho f_r ,
\]

(2.6a)

\[
\rho \left( \frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + v_\theta \frac{\partial v_\theta}{\partial \theta} + v_z \frac{\partial v_\theta}{\partial z} \right) = -\frac{\partial p}{\partial \theta} + \left( \frac{\partial \tau_{\theta\theta}}{\partial \theta} + \frac{\partial \tau_{\theta z}}{\partial z} + \frac{\partial \tau_{r\theta}}{\partial r} \right) + \rho f_\theta ,
\]

(2.6b)

\[
\rho \left( \frac{\partial v_z}{\partial t} + v_r \frac{\partial v_z}{\partial r} + v_\theta \frac{\partial v_z}{\partial \theta} + v_z \frac{\partial v_z}{\partial z} \right) = -\frac{\partial p}{\partial z} + \left( \frac{\partial \tau_{rz}}{\partial r} + \frac{\partial \tau_{\theta z}}{\partial \theta} + \frac{\partial \tau_{zz}}{\partial z} \right) + \rho f_z,
\]

(2.6c)

General equations for Newtonian fluid can be obtained by inserting constitutive equations which describe the linear relationship between shear stress and spatial derivatives of velocity. If further assumptions that density and viscosity are constant is made, then the momentum balance equations, often called Navier-Stokes equations, of the Newtonian fluid in rectangular coordinates are:

\[
\rho \left( \frac{\partial v_x}{\partial t} + v_x \frac{\partial v_x}{\partial x} + v_y \frac{\partial v_x}{\partial y} + v_z \frac{\partial v_x}{\partial z} \right) = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial y^2} + \frac{\partial^2 v_x}{\partial z^2} \right) + \rho f_x ,
\]

(2.7a)

\[
\rho \left( \frac{\partial v_y}{\partial t} + v_x \frac{\partial v_y}{\partial x} + v_y \frac{\partial v_y}{\partial y} + v_z \frac{\partial v_y}{\partial z} \right) = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v_y}{\partial x^2} + \frac{\partial^2 v_y}{\partial y^2} + \frac{\partial^2 v_y}{\partial z^2} \right) + \rho f_y ,
\]

(2.7b)
For cylindrical coordinates,\

\[
\rho \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_r}{\partial \theta} - \frac{v_r^2}{r} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \mu \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial v_r}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_r}{\partial \theta^2} - 2 \frac{\partial v_r}{r^2 \partial \theta} + \frac{\partial^2 v_r}{\partial z^2} \right) + \rho b_r
\]  

(2.8a)\

\[
\rho \left( \frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{v_r v_\theta}{r} + v_z \frac{\partial v_\theta}{\partial z} \right) = -\frac{1}{r} \frac{\partial p}{\partial \theta} + \mu \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial v_\theta}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_\theta}{\partial \theta^2} + 2 \frac{\partial v_r}{r^2 \partial \theta} + \frac{\partial^2 v_\theta}{\partial z^2} \right) + \rho b_\theta
\]  

(2.8b)\

\[
\rho \left( \frac{\partial v_z}{\partial t} + v_r \frac{\partial v_z}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_z}{\partial \theta} + v_z \frac{\partial v_z}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial v_z}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_z}{\partial \theta^2} + \frac{\partial^2 v_z}{\partial z^2} \right) + \rho b_z
\]  

(2.8c)\

- **Conservation of Energy**

The general form of the governing equation for heat transport is in terms of the energy conservation law, the first law of thermodynamics. By balancing the kinetic energy, internal energy, heat change and the work done by the mechanical forces, we have the energy conservation equation in material derivative form:
\[ \rho c_p \frac{DT}{Dt} + \rho L_p \frac{Dp}{Dt} = \nabla \cdot (k \cdot \nabla T) + \tau : D + \rho \dot{R} \]  \hspace{1cm} (2.9)

where the \( D/Dt \) is called the material derivative.

In Cartesian coordinates system

\[ \frac{D}{Dt} = \frac{\partial}{\partial t} + v_x \frac{\partial}{\partial x} + v_y \frac{\partial}{\partial y} + v_z \frac{\partial}{\partial z} \]  \hspace{1cm} (2.10)

When the density is constant, the term \( \rho L_p \frac{Dp}{Dt} \) in Equation (2.9) drops out because the work of volume change equals zero. If the thermal conductivity of the material is constant and isotropic, Equation (2.7) reduces to:

\[ \rho c_p \frac{DT}{Dt} = k \nabla^2 T + \tau : D + \rho \dot{R} \]  \hspace{1cm} (2.11)

If a Newtonian fluid is considered, the equation can be further reduced to:

\[ \rho c_p \frac{DT}{Dt} = k \nabla^2 T + \mu \dot{\gamma}^2 + \rho \dot{R} \]  \hspace{1cm} (2.12)

where \( \dot{\gamma} \) is the scalar strain rate.
In rectangular coordinates, Equation (2.11) is written in the form:

\[
\rho \frac{\partial T}{\partial t} + v_x \frac{\partial T}{\partial x} + v_y \frac{\partial T}{\partial y} + v_z \frac{\partial T}{\partial z} = k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + 2\mu \left[ \left( \frac{\partial v_x}{\partial x} \right)^2 + \left( \frac{\partial v_y}{\partial y} \right)^2 + \left( \frac{\partial v_z}{\partial z} \right)^2 \right] + \rho \hat{R} \tag{2.13}
\]

In Cylindrical coordinates:

\[
\rho \frac{\partial T}{\partial t} + v_r \frac{\partial T}{\partial r} + \frac{v_\theta}{r} \frac{\partial T}{\partial \theta} + v_z \frac{\partial T}{\partial z} = k \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} \right] + 2\mu \left[ \left( \frac{\partial v_r}{\partial r} \right)^2 + \left( \frac{1}{r} \left( \frac{\partial v_\theta}{\partial \theta} + v_r \right) \right)^2 + \left( \frac{\partial v_z}{\partial z} \right)^2 \right] \tag{2.14}
\]

\[
+ \mu \left[ \left( \frac{1}{r} \frac{\partial v_r}{\partial \theta} + r \frac{\partial}{\partial r} \left( \frac{v_\theta}{r} \right) \right)^2 + \left( \frac{\partial v_\theta}{\partial \theta} + \frac{1}{r} \frac{\partial v_z}{\partial z} \right)^2 + \left( \frac{\partial v_z}{\partial r} + \frac{\partial v_r}{\partial z} \right)^2 \right] + \rho \hat{R}
\]

### 2.2 Free Surface Modeling

When a droplet impacts on the die surface, the liquid surface position changes with time. The droplet-surface interaction problem involves moving boundaries and the position and shape of the liquid surface must be determined as part of the solution. The process of determining the moving boundary is often called free surface modeling. The momentum balance technique is one of the widely used computational techniques for modeling flow free surface problems. The standard approach is to numerically solve the Navier-Stokes
equations as embodied in the Marker-and-Cell (MAC), Simplified Marker and Cell (SMAC), and the Solution Algorithm - Volume of Fluid method (SOLA-VOF).

Harlow and Welch developed the MAC technique in 1965 [7]. The technique is for the numerical investigation of the time-dependent flow of an incompressible fluid, the boundary of which is partially confined and partially free. The solution technique makes use of finite difference approximations applied to the full Navier-Stokes equations. The primary dependent variables are the pressure and the velocity. The space is divided into a number of subdivisions, called cells, which are usually rectangular. A set of marker particles, which do not participate in the calculation, is introduced to the system to represent the location of the fluid at any instant. They will move according to the velocity components in their vicinities. A cell with no marker particle is considered to contain no fluid. A cell with markers, but having a neighboring cell with no markers, is defined as containing a free surface. All the other cells with particles are considered to be filled with fluid. SMAC, developed in 1970 by A.A. Amsden and F.H. Harlow [8], uses the same Eulerian mesh and marker particles, with time advancement through finite intervals to obtain the solution. It is a simplified version of MAC. This more efficient solution technique is implemented by not explicitly calculating the pressure and instead using a pseudopressure. Instead of depicting the fluid configuration with a set of marker particles, the VOF method [9] defines a volume of fluid fraction function $F$. $F$ of value one is for a cell full of fluid, while a zero value would indicate that the cell is empty, and between one and zero means a cell containing a free surface. In addition to defining which cells contain a free surface, the VOF method also defines where fluid is located in a boundary.
cell. The normal direction to the boundary lies in the direction in which the value of $F$ changes most rapidly.

### 2.3 Literature Review

This review is categorized into two groups: studies on droplet-surface interaction and those on spray-surface interaction. There are many theoretical, modeling and experimental studies during the last decade on the phenomena related to droplet deformation and thermal behavior when it hits on a wall. However, although some studies have been conducted on lubricant spray in die casting and metal rolling, limited research has been done in hot forging processes. Most of the studies concentrated on the effects of atomization methods and spray operating parameters on the cooling effects. Even fewer studies on spray cooling of the die used water-based graphite which introduces film formation and complicates the thermal behavior of the spray.

#### 2.3.1 Droplet-Surface Interaction

The methods of studies of droplet impact on a solid surface were chosen by the researchers according to the phenomena they were interested in. Phenomena with only fluid dynamics considered were studied mostly by analytical models and numerical solutions. The numerical calculations were normally validated by experimental results in most of the cases. Heat transfer without phase change was also analytically modelled by
some of the researchers. Only convection and conduction between two phases, liquid and solid, were included in these models. Experimental methods were utilized most often when phase change was involved in the problems studied.

- Numerically Modeling

Harlow and Shannon [10] were the first to use Marker-and-Cell (MAC), a finite differential method mentioned above, to solve momentum equations for droplet impact on a surface. They neglected surface tension and viscous effects to simplify the problem. However, this assumption is only applicable at the early stage of impact when inertial effects dominate. Trapaga and Szekely [11] numerically modeled deformation process of liquid-metal droplet impact on a solid plate using combined MAC and Volume of Fluid (VOF) methods to track the droplet surface. Bussmann et al. [12] developed a three-dimensional fluid flow code to simulate water droplet impact on an inclined substrate and a step, not considering heat transfer and phase change during droplet impact. Their model was based on RIPPLE, a 2D fixed-grid Eulerian code developed specifically for free surface flows with surface tension. They modeled surface tension as a volume force exerted on fluid near the free surface. The temporal variation of contact angles at the leading and trailing edges of the droplet was considered in their studies. Their simulation results agreed well with their photographic data. Fluid dynamics and solidification of the tin droplet impact on a flat stainless plate was modeled in a study by Pasandideh-Fard et al. [13]. They applied complete numerical solution of the Navier-Stokes and energy equations to model the droplet deformation and solidification and heat transfer in the
substrate. The heat transfer coefficient at the droplet-surface interaction was evaluated by matching the temperature calculated numerically from the model with measurements. They concluded that the effect of solidification on droplet impact dynamics was negligible if $\sqrt{\text{Ste}/\text{Pr}} \ll 1$, where $\text{Ste}$ is Stefan number ($\text{Ste} = C(T_m - T_w)/H_f$), and $\text{Pr}$ is Prandtl number ($\text{Pr} = \nu/\alpha$). Pasandideh-Fard et al. [14] also studied the impact of droplet on a stainless steel surface which was heated from 50°C to 120°C, a controlled temperature range low enough to prevent boiling in the droplet. VOF code was applied to calculate the shape of droplet and substrate temperature during impact. The results of experiments and numerical calculations agreed well. According to their results, they concluded that for fixed Reynolds number ($\text{Re}$) cooling effectiveness increases with Weber number ($\text{We}$).

Some other researchers utilized deforming meshes instead of the fixed-grid mentioned above to numerically solve their models. Fukai et al. [15] presented a finite element technique to model the droplet spreading process. All effects in momentum conservation equations, i.e., inertial, viscous, gravitational and surface tension effects, were considered in their model. Two liquids, water and liquid tin, were utilized in the simulation based on the Lagrangian model and significant differences of the flow field were observed for droplets of these two liquids. Zhao et al. [40] extended Fukai’s study to model both fluid dynamics and heat transfer during the impact of liquid droplet on a substrate. However, only heat transfer of convection and conduction were included and phase change phenomena were ignored to facilitate the modeling of heat transfer. They concluded that the maximum splat thickness might occur at the periphery of the splat where mass
accumulation was observed. The heat transfer results showed that temperature at the spreading front is substantially lower than the temperature at the splat center in a molten metal droplet.

- **Experimental Studies**

Because most of the droplet applications in industry involve complex phase change phenomena, the analytical modeling of the droplet-surface is not enough to thoroughly understand the droplet-surface interactions. Experimental studies are often the only way to evaluate thermal behavior of a droplet impact on a super-heated solid surface [16 – 19].

Mundo et al. [20] examined three liquids with different viscosities and surface tensions to formulate an empirical model describing the deposition and splashing process. They used a two-component phase Doppler anemometer to characterize the size and velocity of the secondary droplets. The results showed that the outcome of an impact, splashing or deposition, depended on the Reynolds number (Re) and Ohnesorge number (Oh).

Most other experimental studies focused on the influence of heating of the solid surface on the droplet-collision process. Understanding the influence of surface temperature on droplet-collision dynamics for a particular fluid requires the mapping of various boiling regimes. The droplet-evaporation lifetime as a function of surface temperature can be used to delineate different heat-transfer regimes, thus providing a mechanism to better understand the influence of surface temperature on droplet-collision dynamics. Tamura
and Tanasawa [21] examined a droplet gently deposited on a surface and found that the total droplet-evaporation lifetime would decrease with increasing surface temperature until a minimum evaporation time was obtained. The Leidenfrost temperature occurred where the total evaporation lifetime of the droplet reached a local maximum. For temperatures greater than the Leidenfrost temperature (film boiling), the droplet-evaporation lifetime will monotonically decrease with further increases in surface temperature. Wachters and Westerling [22] investigated water-droplet impact on a heated surface. In their experiments, a polished gold surface maintained at 400°C was used. Three distinct regions were identified, each a function of the impact Weber number (We).

Makino & Michiyoshi [23] performed a water-droplet-impact study using four different impact surfaces, namely, copper, brass, carbon steel and stainless steel. The surface temperature was varied from 80 to 450°C and a correlation for the contact period was obtained for all surfaces. Karl and Frohn [24] experimentally examined the mechanical interaction processes of small droplet impact on hot solid surface heated to a temperature well above Leidenfrost temperature. The experiments were conducted with the methods using mono-disperse droplet streams in combination with a standard video camera. A minimum impinging angle for droplet break-off was identified for small impact angle in their study.
2.3.2 Spray-Surface Interaction

Although some studies have been conducted on lubricant spray in die casting and metal rolling, limited research has been done in hot forging processes. Most of the studies concentrated on the effects of atomization methods and spray operating parameters on the cooling effects. Even fewer studies on spray cooling of the die used water-based graphite which introduces film formation and complicates the thermal behavior of the spray.

Tseng et al. [25] examined the roll cooling system used in steel rolling mills and its effects on the roll life by combined experimental and numerical methods. The heat transfer coefficient was evaluated experimentally and then utilized for numerical simulation of the rolling processes. Their results led to the conclusion that uniform cooling minimized the cyclic stress or the resultant fatigue stress and thus extended roll life. Horsky et al. [26] conducted cooling experiments to evaluate the heat transfer coefficient on a steel plate heated up to 630°C which was cooled by water sprays with a set of nozzles. The heat transfer coefficient data showed that the heat transfer drops for pressures higher than 6 bar and the magnitude of the heat transfer coefficient for the scaled surfaces was of approximately 1000 W/m²K lower than that for the polished surfaces. Bariani et al. [27] developed an experimental and FE simulation combined method to estimate the heat transfer coefficient at the tool surface during cooling and deformation phases in hot forging processes. The heat transfer coefficient at the interface of spray- surface was inversely analyzed by minimizing the difference of the temperature values generated in FE model and those measured in the experiments. The heat transfer
coefficients were examined during cooling phases as a function of spraying parameters. Bariani et al. [28] used this inverse calculation approach to evaluate the effect of feed pressure and flow rate of the spray on the heat transfer coefficient between heated flat dies and coolant experimentally. They compared the results from two different sprays: the downward spray from the overhead nozzle and the upward spray from the beneath nozzle. They concluded that increase of the feeding pressure and the flow rate enhanced the heat transfer coefficient for the both spray configurations. Some of the other studies obtained the similar conclusion that a higher liquid flux density, defined as average liquid flow rate per unit area, provides a higher heat transfer coefficient for a given surface temperature [29 - 31].

In fact, very little work has been done on mechanisms of cooling and film formation on the die in hot forging from the droplet aspect. Interesting issues are introduced in this research for a typical droplet of hot forging lubricant comparing with those of the droplet in previous studies: 1) there are a large number of solid particles, i.e., graphite in several to tens of microns, dispersed and initially suspended inside a droplet, 2) evaporation with formation of bubbles inside the droplet is a significant process. Solid particles evolve into film deposited on the die after the liquid is vaporized. These particles not only influence fluid mechanics during droplet impact but also complicate heat transfer and evaporation, an intractable process itself even in the absence of particles.
Hence, the main purpose of this research was to establish a model to estimate effects of changing parameters such as droplet size, impaction velocity and lubricant properties on the deposition patterns during droplet hitting the die at a wide range of temperature.
CHAPTER 3

LUBRICANT SPREAD IN THE SPRAYING OF HOT DIE SURFACES

This chapter proposes an isothermal numerical model based on the VOF finite difference approach for investigating the dynamics and spread of lubricant droplet on hot die surface. The lubricant properties for this model are calculated in simple capillary experiments. This model is validated by comparing with spreading results from single droplet experiments at different lubricant dilution ratios (lubricant properties) and droplet kinetics. It is found that the isothermal assumption is valid for surface temperatures where no-steam forms, less than the Leidenfrost temperatures. It is also found that a simple linear relationship exists in the logarithmic scale between the spread factor and the Weber number for dilution ratios less than equal to unity (for dilute suspensions).

3.1 Introduction

In the hot forging processes, the dies are lubricated and cooled by spraying dilute water based lubricants on the heated die surfaces. This spray is produced by atomizing lubricants into small droplets using spray devices with pressurized air. The lubricant is
often a fine suspension of fine graphite particles in water, with surfactants and binders added to aid in the spreading and formation of adherent lubricant films. In addition, this lubricant spray is used to maintain the thermal conditions of the die surface suitable for metal flow in deep cavities. As the phenomena of spraying, spreading and film formation are poorly understood, the lubricant is often applied in large quantities resulting in excessive cooling of the surface, and excess graphite in the die cavities, and the atmosphere. While excessive chilling of die surfaces causes poor metal flow and filling of die cavities, excess graphite leads to air and ground water pollution. Consequently, understanding the ideal spraying conditions for lubricant spread and film formation has become critical to both the manufacturing and lubrication community.

The important factors governing the mechanics and thermodynamics of the spray deposition can be categorized into three groups: those related to the hot die surface, those related to the lubricant properties, and those related to the spray characteristics. The factors related to die surface include the initial surface temperature, its roughness, its geometry (cavity design) and surface treatments (such as diffusion treatments or coatings). Important lubricant properties include surface tension and viscosity. Lastly, the spray characteristics involve droplet size, droplet velocity, dilution ratio (or solid content) and, spray cone angle and nozzle height. Composition of the lubricant and the selection of the spray parameters are often determined by the lubrication and cooling needs of the intended application.
The primary objective of this study is to develop a model that captures the mechanics of lubricant spread as the spray droplets impact the hot die surface. This spread governs the thickness of the resultant film and the heat transfer from the hot die surface. Such a model would enable optimal design of the lubrication process and consequently, significant reduction in the graphite related pollution.

This phenomenon of wetting and spreading in spray lubrication in hot forging is difficult to study and model due to the following reasons:

- The spray consists of very fine droplets with random distribution of size (tens to hundreds of microns in diameter) and velocity (several to tens meters per second). A methodology must be adopted to quantify the spray characteristics.

- The lubricant properties and dilution ratios (amount of water added to dilute the liquid) affect the physics of the droplet mechanics. They vary considerably from application to application. This model must include these properties and parameters.

- The physics and thermodynamics of the spreading phenomenon are complicated by its transient and evolving nature. The lubricant droplet impacts the hot die surface at temperatures between 250 to 450°C. At these temperatures steam forms first, followed by nucleate boiling, convective cooling and dry-off (evaporation).
It may be very difficult, if not impossible, to develop a single model that can represent all these phenomena.

In order to make the problem solvable, it was decided to make several assumptions and limit the scope of the model. First, it was decided to represent the spray by the Sauder Mean Diameter (an average diameter for log-normal distribution). This assumption will permit the spray to be represented by single droplet experiments. Second, it was decided to use non-dimensional numbers, using the similarity principle, to quantify the physics of the process. These numbers will enable the simple model to represent many physical conditions (lubricant properties, droplet diameters, impact velocities etc.). Third, it was decided to limit this study to lower surface temperatures where two phase flow is not present. Finally, it was decided to develop a numerical model that will permit application of the governing equations to complicated geometries and boundary conditions.

The phenomenon of a droplet impact on hot surface has been studied by many researchers since the classical work of Leidenfrost. Most of the previous work is on water droplets and the heat transfer resulting from the evaporation and boiling of the liquid. The study of the droplet spread during impact has applications to painting, thermal spray coatings, rapid solidification, lubrication etc. Earlier studies were mainly experimental with high speed or flash photography used to capture droplet dynamics [32-37]. An interesting study was carried out by Makino and Michiyoshi [23] with water droplets on heated surfaces (350°C). They found that for temperatures below Liedenfrost (<200°C), the droplet spread factor became independent of surface temperature. In 1991,
Chandra and Avedisian [38] used flash photography to capture dynamics of n-heptane droplet (about 1.5 mm) on heated (24 to 260°C) stainless steel surface at. Weber numbers kept fixed at 43. They also found that for lower temperatures, below Leidenfrost point, the spread factor became independent of surface temperature. Recently, Manzello and Yang [39] studied the impact dynamics of water diluted sodium acetate trihydrate droplet (diameter 2.7 mm) on a stainless steel heated surface (from 20 to 340°C). They confirmed the previous findings that below 230°C (below Leidenfrost point) the droplet spread did not depend on the surface temperature. From these studies with water and other liquids it can be concluded that in the nucleate boiling and convection boiling regimes, the droplet spread is mainly governed by the droplet dynamics and independent of surface temperature and heat transfer. This may be due to small dry-off times (high heat transfer) at these temperatures with no-steam formation. This results in inertia and viscous forces to dominate the droplet spread.

Development of numerical models for droplet impact on a solid surface has also received much attention. For example, Harlow and Shannon [7] were the first to use Marker-and-Cell (MAC), a finite differential method, to solve momentum equations for droplet impact on a surface. They neglected surface tension and viscous effects for the simple of the problem. However, this assumption only applies at the early stage of impact when inertial effects dominate. Trapaga and Szekely [11] numerically modeled the deformation process of a liquid-metal droplet on impact with a solid plate using combined MAC and Volume of Fluid (VOF) methods to track the droplet surface changes. Bussmann et al. [12] developed a three-dimensional fluid flow model to simulate water droplet impact on
an inclined substrate and a geometrical step; not considering heat transfer and phase change during droplet impact. Their model was based on RIPPLE, a 2D fixed-grid Eulerian code developed specifically for free surface flows with surface tension. They modeled surface tension as a volume force exerted on the fluid near the free surface. The temporal variation of contact angles at the leading and trailing edges of the droplet was considered in their studies. Their simulation results agreed with their photographic data.

Fluid dynamic and solidification of the molten tin droplet impact on a flat stainless plate was modeled by Pasandideh-Fard et al [13]. They applied the numerical solution of Navier-Stokes and energy equations to model the droplet deformation and solidification, and heat transfer in the substrate. The heat transfer coefficient at the droplet-surface interaction was evaluated by matching the temperature calculated numerically from the model with measurements. They concluded that the effect of solidification on droplet impact dynamics was negligible if the factor $\sqrt{Ste/Pr} \ll 1$, where $Ste$ is Stefan number ($Ste = C_p(T_m - T_{w,0})/H_f$), and $Pr$ is Prandtl number ($Pr = \nu/\alpha$). Pasandideh-Fard et al. [14] also studied the impact of droplet on a stainless steel surface which was heated from 50$^\circ$C to 120$^\circ$C, a controlled temperature range low enough to prevent boiling in the droplet. VOF code was applied to calculate the shape of droplet and substrate temperature during impact. The results of experiments and numerical calculations agreed well. According to their results, they concluded that for fixed Reynolds number Re cooling effectiveness increases with Weber number $We$.

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In this chapter, an isothermal finite difference VOF based numerical model is proposed for studying droplet dynamics and spread. The isothermal assumption is based on the demonstration by previous researchers that for temperatures lower than Leidenfrost, the droplet spread is independent of surface temperature. Hence, heat transfer plays a minor role in the mechanics of spread in this temperature range where the heat transfer is very high and dry-off times short. The physical constants for this model are determined by simple single droplet experiments. The model is applied to different lubricant dilution ratios, surface temperatures and range droplet kinetics (Weber numbers). The predictions of the model are validated by comparing with results of single droplet experiments.
3.2 Theoretical Analysis

The complexity of the fluid dynamical aspects of droplet impact on a solid surface is exemplified by the extreme deformation of the droplet surface occurring within very short time scale. The model presented here simulates the impact of a liquid droplet on a solid from the moment that the lubricant droplet comes into contact with the die surface and proceeding until the droplet becomes rest after the spreading process is completed or it is terminated at any desired time. The numerical solution based on VOF method is subsequently presented.

3.2.1 Governing Equations

As shown in the Fig. 3.1, for the ease of problem description and boundary condition writing, the cylindrical coordinates are used to define the conservation equations. Any point of the droplet can be described by $r$, $z$, and $\theta$, which are the radial, axial and azimuthal coordinates.

- Conservation of Mass

As we have discussed in Chapter 2, the equation of mass balance for cylindrical coordinates can be written as:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} (\rho rv_r) + \frac{\partial}{\partial \theta} (\rho rv_\theta) + \frac{\partial}{\partial z} (\rho vz) = 0$$

(3.1)
Fig. 3.1. The Cylindrical Coordinate description of lubricant droplet on a solid surface.
For an isothermal condition, the following assumptions can be taken for a initially spherical droplet impact on a solid surface:

I. Constant liquid density \( \rho \).

II. The velocity at the azimuthal direction \( v_\theta \) is always zero.

Then Equation (3.1) reduces to:

\[
\frac{1}{r} \frac{\partial}{\partial r} (rv_r) + \frac{\partial v_z}{\partial z} = 0
\]

(3.2)

where \( v_r \) is the velocity at the radial direction, \( v_z \) the velocity at the axial direction.

- Conservation of Momentum

From Chapter 2, the general form of linear momentum equations in cylindrical coordinates is shown as follows.

\[
\rho \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + v_\theta \frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \tau_{rr} \right) + \frac{1}{r} \frac{\partial \tau_{r\theta}}{\partial \theta} + \frac{\partial \tau_{rz}}{\partial z} \right) + \rho b_r
\]

(3.3a)

\[
\rho \left( \frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + v_\theta \frac{\partial v_\theta}{\partial \theta} + v_r v_\theta + v_z \frac{\partial v_\theta}{\partial z} \right) = -\frac{1}{r} \frac{\partial p}{\partial \theta} + \left( \frac{1}{r^2} \frac{\partial}{\partial r} \left( r^2 \tau_{r\theta} \right) - \frac{1}{r} \frac{\partial \tau_{r\theta}}{\partial \theta} + \frac{\partial \tau_{\theta\theta}}{\partial z} \right) + \rho b_\theta
\]

(3.3b)
\[
\rho \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + v_\theta \frac{\partial v_r}{\partial \theta} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \left( \frac{1}{r} \frac{\partial}{\partial r} \left( \tau_{rr} \right) + \frac{\partial \tau_{r \theta}}{\partial \theta} + \frac{\partial \tau_{r z}}{\partial z} \right) + \rho g \tag{3.3c}
\]

Dropping the terms including azimuthal direction and considering only gravity as the body force, we can rewrite the momentum equations as:

\[
\rho \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \left( \frac{1}{r} \frac{\partial}{\partial r} \left( \tau_{rr} \right) + \frac{\partial \tau_{r z}}{\partial z} \right) + \rho g \tag{3.4a}
\]

\[
\rho \left( \frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + v_z \frac{\partial v_\theta}{\partial z} \right) = -\frac{\partial p}{\partial r} + \left( \frac{1}{r} \frac{\partial}{\partial r} \left( \tau_{r \theta} \right) + \frac{\partial \tau_{r z}}{\partial z} \right) + \rho g \tag{3.4b}
\]

where \( p \) is the pressure, \( \tau_{ij} \) the stress normal to \( i \) direction and pointing to \( j \) direction.

### 3.2.2 Initial and Boundary Conditions

It is very important to correctly define the initial and boundary conditions before we go further to numerically solve the equations listed above.

When \( t=0, \ v_r = 0, v_z = V_0, \ and \ p = p_0 + \frac{2\gamma}{R_0}, \) \tag{3.5}

where \( V_0 \) is initial impact velocity, \( p_0 \) the ambient air pressure, \( \gamma \) the surface tension of the lubricant, and \( R_0 \) the initial radius of the droplet.
When \( t=0 \), \( \xi - R_0 \xi + r^2 = R_0^2 \), or \( r = \sqrt{2zR_0 - z^2} \), \( 0 \leq z \leq 2R_0 \) \hspace{1cm} (3.6)

At \( r = 0 \), \( v_r = 0 \), \( \frac{\partial v_r}{\partial r} = 0 \) \hspace{1cm} (3.7)

At \( z = 0 \), \( v_r = 0 \) and \( v_z = 0 \), \hspace{1cm} (3.8)

Considering the viscosity of the ambient air is much lower than that of the lubricant liquid, the tangential stresses at the free surface can be neglected and set to zero at the free moving surface, that is:

\[ \tilde{\tau} = 0 \] \hspace{1cm} (3.9)

This means that all velocity derivatives that involve velocity components outside the surface are set to zero.

\[ \frac{\partial v_r}{\partial r} = 0 ; \quad \frac{\partial v_z}{\partial z} = 0 ; \quad \frac{\partial v_z}{\partial z} = 0 ; \quad \text{and} \quad \frac{\partial v_z}{\partial r} = 0 \] \hspace{1cm} (3.10)

Thus, the boundary condition should only represent the balance between the pressures and the surface tension stresses. The force exerted on the moving surface is sketched in the Fig. 3.2. In Fig. 3.2, \( \hat{n} \) is the unit vector normal to the droplet surface, \( p_o \) the ambient pressure, \( p \) the pressure inside the droplet surface, \( \gamma \) the surface tension, \( \hat{t} \) unit vector tangent to the droplet surface.
At the free moving surface, the boundary condition should represent the balance between the viscous and the surface tension stresses. The force exerted on the moving surface is sketched in the Fig. 3.2.

As shown in the figure, the position along the surface is measured by the arc length $s$, and the forces acted on a small increment $ds$ on the surface are considered. The depth in the direction perpendicular to the paper is taken as unit length for convenience of analysis. Then, the total force acting on this area must be zero.

The pressure in the droplet provides a force of $-p\hat{n}ds$, the negative sign occurring because the force acts opposite to the direction of $\hat{n}$. The contribution of the ambient air to the droplet surface, the interface of the liquid and air, is due to its pressure, $-p_0\hat{n}ds$.

By introducing the unit vector $\hat{t}$ that is tangent to the droplet surface, we have the surface tension on the right side of the $ds$ to be $-\gamma\hat{t}ds$, the negative sign occurring because the force acts opposite to the direction of $\hat{t}$. On the left side, the surface tension contribution is $
abla_t + \frac{d(\gamma\hat{t})}{ds} ds$.

Thus, this force balance can be written as:

$$-p\hat{n}ds - p_0\hat{n}ds - \gamma\hat{t}ds + \left(\nabla_t + \frac{d(\gamma\hat{t})}{ds} ds\right) = 0$$

(3.11)
Fig. 3.2. The boundary condition of free moving surface.
Considering:

\[
\frac{d(\hat{\gamma})}{ds} = \gamma \frac{d\hat{\gamma}}{ds} + \frac{d\gamma}{ds} \hat{\gamma} = \gamma \hat{n} + \frac{d\gamma}{ds} \hat{\gamma}
\]  

(3.12)

where \( \kappa \) is the curvature of the surface. Then Equation (3.11) reduces to:

\[
- p\hat{n} = -(p_0 + \gamma \kappa)\hat{n} + \frac{d(\gamma)}{ds} \hat{\gamma}
\]  

(3.13)

The second term includes the derivative of \( \gamma \) with the respect to position along the surface. Because here we consider only fluid dynamic of the droplet and no temperature changes along the surface, we can assume that surface tension \( \gamma \) is a constant and drop out the derivative term in Equation (3.13). Introducing the mean curvature \( \kappa_m = \frac{1}{2} \kappa_1 + \kappa_2 \), where, \( \kappa_1 \) and \( \kappa_2 \) are the largest and smallest normal curvatures respectively, we finally obtain the boundary condition at the droplet surface as:

\[
p = p_0 + 2\gamma \kappa_m
\]  

(3.14)

Because each point on the free surface can be uniquely defined by a position vector \( \tilde{s} = \tilde{s}[r(s), z(s)] \), where \( s \) is the coordinate measured along the free surface of the deforming droplet, as shown in Fig. 3.2. It has been shown by Lass [41] that
\[ \kappa_m = \frac{r^2 (r' z'' - z' r'') + [(r')^2 + (z')^2] r z'}{2r^3 [(r')^2 + (z')^2]^{3/2}} \] (3.15)

Because the position of the droplet surface is part of the solution and we do not know where the surface is, we need one more equation. Considering the normal velocity of the surface will be the velocity of the surface itself, we obtain another boundary condition:

\[ \vec{v}_{surface} = (\vec{v} \cdot \hat{n}) \hat{n} \] (3.16)

This boundary condition can be used to find the droplet surface velocity and governs its motion in time.

If the density \( \rho \) and viscosity \( \mu \) of the lubricant is assumed to be constant, and the liquid is regarded as Newtonian fluid, then the constitutive equation that describes the response of the material to applied stresses is:

\[ \tilde{\tau} = 2\mu \tilde{D} \] (3.17)

where \( \tilde{D} \) is the rate of deformation tensor: \( \tilde{D} = \frac{1}{2} \left( \tilde{L} + \tilde{L}' \right) \), in which \( \tilde{L} \) is the velocity gradient tensor and \( \tilde{L} = \nabla \vec{v} \). If Equation (3.17) is applied to Equations (3.2) and (3.3), the balance equations turn out to be Navier-Stokes equations.
3.2.3 Scaling

Before proceeding to the scaling of the equations, we make a summary of what we have obtained. The governing equations for the droplet impact on a solid surface include the continuity balance equation (3.2), and momentum balance equation 3.4(a) and 3.4(b), whereas the initial and boundary conditions are equations (3.5), (3.6), (3.7), (3.8), (3.10) and (3.14), in which equations (3.10) and (3.14) are conditions for the moving free surface of the deforming droplet surface.

In order to learn the characteristic values of all the problem variables, we use the scaling analysis by introducing the following non-dimensional parameters:

\[ r^* = \frac{r}{R_0} \]  \hspace{1cm} (3.18)

\[ z^* = \frac{z}{R_0} \]  \hspace{1cm} (3.19)

\[ v_z^* = \frac{v_z}{v_0} \]  \hspace{1cm} (3.20)

\[ v_r^* = \frac{v_r}{v_{rc}} \]  \hspace{1cm} (3.21)
\[ t^* = \frac{t}{R_0/v_0} \]  \hspace{1cm} (3.22)

\[ \kappa_m^* = \frac{k_m}{1/R_0} \]  \hspace{1cm} (3.23)

\[ p^* = \frac{p - p_0}{P_c} \]  \hspace{1cm} (3.24)

\[ \tau_{ij}^* = \frac{\tau_{ij}}{\rho v_0^2} \]  \hspace{1cm} (3.25)

where \( v_r \) and \( P_c \) characteristic values for \( v_r \) and \( p \) respectively. They are unknown at this time and will be determined in the late scaling procedure. The above non-dimensional parameters are created so that their magnitude is of the order of one.

Introducing these non-dimensional parameters into the governing equations and initial and boundary conditions, the scaled equations are then given by:

- **Scaled Mass Balance Equation**

\[ v_r \frac{1}{r^*} \frac{\partial}{\partial r^*}(r^* v_r) + v_0 \frac{\partial v_z}{\partial z} = 0 \]  \hspace{1cm} (3.26)
Dividing the equation through by \( v_{rc} \), we have

\[
\frac{1}{r^*} \frac{\partial}{\partial r^*} (r^* v_r) + \frac{v_0}{v_{rc}} \frac{\partial v_z}{\partial z} = 0
\]  

(3.27)

Because there are only two terms in this equation, both derivatives must be \( \mathcal{O}(1) \), and we must have \( v_{rc} = v_0 \). So the scaled mass balance equation is:

\[
\frac{1}{r^*} \frac{\partial}{\partial r^*} (r^* v_r) + \frac{\partial v_z}{\partial z} = 0
\]  

(3.28)

- **Scaled Momentum Balance Equations**

\[
\rho \frac{v_0^2}{R_0} \left( \frac{\partial v_r^*}{\partial t} + v_r^* \frac{\partial v_r^*}{\partial r^*} + v_z^* \frac{\partial v_z^*}{\partial z^*} \right) = -\frac{P_c}{R_0} \frac{\partial p^*}{\partial r^*} + \rho v_0^2 \left( \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( \tau_{rr}^* + \tau_{rz}^* \right) \right)
\]  

(3.29a)

\[
\rho \frac{v_0^2}{R_0} \left( \frac{\partial v_z^*}{\partial t} + v_r^* \frac{\partial v_z^*}{\partial r^*} + v_z^* \frac{\partial v_z^*}{\partial z^*} \right) = -\frac{P_c}{R_0} \frac{\partial p^*}{\partial z^*} + \rho v_0^2 \left( \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( \tau_{rz}^* + \tau_{zz}^* \right) \right) + \rho g
\]  

(3.29b)

Dividing the equations through by \( \rho \frac{v_0^2}{R_0} \), we have

\[
\frac{\partial v_r^*}{\partial t} + v_r^* \frac{\partial v_r^*}{\partial r^*} + v_z^* \frac{\partial v_z^*}{\partial z^*} = -\frac{P_c}{\rho v_0^2} \frac{\partial p^*}{\partial r^*} + \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( \tau_{rr}^* + \tau_{rz}^* \right)
\]  

(3.30a)
\[
\frac{\partial v_z^*}{\partial t^*} + v_r^* \frac{\partial v_z^*}{\partial r^*} + v_z^* \frac{\partial v_z^*}{\partial z^*} = -\frac{P_c}{\rho v_0^2} \frac{\partial p^*}{\partial z^*} + \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( r^* \tau_{rr^*} \right) - \frac{\partial \tau_{rz^*}}{\partial z^*} + \frac{gR_0}{v_0^2} \quad (3.30b)
\]

Let \( P_c = \rho v_0^2 \), the scaled momentum balance equations become:

\[
\frac{\partial v_r^*}{\partial t^*} + v_r^* \frac{\partial v_r^*}{\partial r^*} + v_z^* \frac{\partial v_r^*}{\partial z^*} = -\frac{\partial p^*}{\partial r^*} + \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( r^* \tau_{rr^*} \right) - \frac{\partial \tau_{rz^*}}{\partial z^*} \quad (3.31a)
\]

\[
\frac{\partial v_z^*}{\partial t^*} + v_r^* \frac{\partial v_z^*}{\partial r^*} + v_z^* \frac{\partial v_z^*}{\partial z^*} = -\frac{\partial p^*}{\partial z^*} + \frac{1}{r^*} \frac{\partial}{\partial r^*} \left( r^* \tau_{rz^*} \right) + \frac{1}{Fr} \quad (3.31b)
\]

where \( Fr = \frac{v_0^2}{R_0 g} \) is the Froude number. In a typical lubricant spray in hot forging with average velocity of droplets 5-20 m/s and average radium 20-200 μm, the Fr is in the range of 30 to 100,000. Thus, the gravity effect can be neglected in this case.

- Scaled Initial and Boundary Conditions

I.

When \( t^* = 0 \), \( v_r^* = 0, v_z^* = -1, \) and \( p^* = \frac{2}{We} \) \( \quad (3.32) \)

II.

When \( t^* = 0 \), \( \xi^* - 1 + r^2 = 1 \), or \( r^* = \sqrt{z^* (2 - z^*)} \), \( 0 \leq z^* \leq 2 \) \( \quad (3.33) \)

III.
At \( r^* = 0, v_r^* = 0, \frac{\partial v_r^*}{\partial r^*} = 0 \) \hspace{1cm} (3.34)

IV.

At \( z^* = 0, v_r^* = 0 \) and \( v_z^* = 0 \), \hspace{1cm} (3.35)

V. At the moving surface

\[
p^* = \frac{4\kappa_m^*}{\text{We}}
\]

(3.36)

Where \( \text{We} \) is the Weber number: 
\[
\text{We} = \frac{2\rho v_0^2 R_0}{\gamma}.
\]

\[
\tilde{v}^* = (\tilde{v}^* \cdot \hat{n})\hat{n}
\]

(3.37)

3.3 Numerical Solution – FLOW-3D Simulation

FLOW-3D is chosen to be the simulation tool for this study. FLOW-3D is a general purpose, three-dimensional, computational fluid dynamic (CFD) program that has the ability to predict flows with free surfaces. It has many capabilities. Using different input data, researchers can select various physical options to represent a wide range of fluid flow phenomena.
FLOW-3D uses the volume of fluid (VOF) method to predict free-surface fluid motions. It numerically solves the fluid flow governing equations using finite-difference approximations. The computational domain is divided into uniform/variable sized rectangular cells and the geometric features are embedded in the mesh blocks. For each cell, values are computed for the basic flow quantities (e.g., velocity, pressure, temperature).

Before the discussion of the application of FLOW-3D for numerically solving the fluid dynamic model of the lubricant droplet impact, it is important to understand how FLOW-3D works to simulate the basic fluid flow, especially the Volume of Fluid (VOF) method.

In FLOW-3D free surfaces are modeled with the Volume of Fluid (VOF), a technique first reported in Nichols and Hirt (1975), and more completely in Hirt and Nichols (1981). The VOF method consists of three ingredients: a scheme to locate the surface, an algorithm to track the surface as a sharp interface moving through a computational grid, and a means of applying boundary conditions at the surface.

### 3.3.1 Fraction Function, Fractional Volume and Fractional Area

The fluid fraction function $F$ is defined to be equal to 1.0 in the fluid and 0.0 outside fluid, i.e. in the void. Averaged over a control volume (or a cell), the value of F will be within the segment [0.0, 1.0], as shown in Fig. 3.3. The volume of fluid in a cell is defined:
\[ FV = V_f \, dx \, dy \, dz \]  \hspace{1cm} (3.38)

The kinematic equation for the VOF function \( F \) in the absence of mass sources is:

\[ \frac{\partial F}{\partial t} + \left[ \frac{\partial}{\partial x} (F v_x) + \frac{\partial}{\partial y} (F v_y) + \frac{\partial}{\partial z} (F v_z) \right] = 0 \]  \hspace{1cm} (3.39)

FLOW-3D utilizes the fractional area/volume method (FAVOR) to model complex geometric boundaries instead of using deformed grids employed by finite-element methods. The FAVOR concept can be used in connection with any type of grid including grids consisting of rectangular or distorted elements and whether or not the grid is structured or unstructured. Structured grids are best because they are easy to generate and the indices for neighboring elements are known. Rectangular grid elements make it easy to compute the fractional areas and volumes of elements used by the FAVOR method.

As shown in the Fig. 3.4, the fractional area and volume ratios \( A_f \) and \( V_f \) are defined as:

\[ A_f = \frac{\text{open area}}{\text{cell edge area}} \quad \text{and} \quad V_f = \frac{\text{open volume}}{\text{volume of the cell}} \]  \hspace{1cm} (3.40)
Fig. 3.3. Typical values of the VOF function $F$ in the droplet model.
If there are geometrical constraints of the flow, then the kinematic equation of \( F \) is:

\[
V_f \frac{\partial F}{\partial t} + \nabla \cdot (\bar{A} \bar{v} F) = 0 \tag{3.41}
\]

or

\[
\frac{\partial F}{\partial t} + \left[ \frac{\partial}{\partial x} \left( \phi A_x v_x \right) + \frac{\partial}{\partial y} \left( \phi A_y v_y \right) + \frac{\partial}{\partial z} \left( \phi A_z v_z \right) \right] = 0 \tag{3.42}
\]

where \( V_f \) and \( \bar{A} = (A_x, A_y, A_z) \) are the volume and area fractions describing the geometrical constraints of the flow, \( \bar{v} = (v_x, v_y, v_z) \) is the flow velocity.

The numerical solution of Equation (3.41) or (3.42) must prevent unphysical distortion of the interface and preserve its sharpness. The latter requirement means that the interface between fluid and void must be no more than one cell wide.

Integrating the \( A_f \) and \( V_f \) into conversion equations, we have the equation transformation as:
\[
\frac{\partial C}{\partial t} + v_x \frac{\partial C}{\partial x} + v_y \frac{\partial C}{\partial y} + v_z \frac{\partial C}{\partial z} \Rightarrow \frac{1}{VF} \frac{\partial C}{\partial t} + v_x AF_x \frac{\partial C}{\partial x} + v_y AF_y \frac{\partial C}{\partial y} + v_z AF_z \frac{\partial C}{\partial z}
\] (3.43)
3.3.2 The Governing Equations in Flow-3D Format

Introducing $F$, $V_f$ and $\bar{A} = (A_r, A_\theta, A_z)$ into equations (3.1), (3.2) and (3.3), the governing equations become:

\[
\frac{\partial}{\partial r} (v_r A_r) + \frac{\partial (\bar{A}_r \bar{A}_z)}{\partial z} + \frac{v_r A_r}{r} = 0 \tag{3.44}
\]

\[
\frac{\partial v_r}{\partial t} + \frac{1}{V_f} \left( v_r A_r \frac{\partial v_r}{\partial r} + v_z A_z \frac{\partial v_r}{\partial z} \right) = -\frac{1}{\rho} \frac{\partial p}{\partial r} + \frac{1}{\rho V_f} \left( \frac{\partial}{\partial r} (\bar{A}_r \tau_{rr}) + \frac{\partial (\bar{A}_z \tau_{rz})}{\partial z} + \frac{\bar{A}_r \tau_{rr}}{r} \right) \tag{3.45}
\]

and

\[
\frac{\partial v_z}{\partial t} + \frac{1}{V_f} \left( v_r A_r \frac{\partial v_z}{\partial r} + v_z A_z \frac{\partial v_z}{\partial z} \right) = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \frac{1}{\rho V_f} \left( \frac{\partial}{\partial r} (\bar{A}_r \tau_{rz}) + \frac{\partial (\bar{A}_z \tau_{zz})}{\partial z} + \frac{\bar{A}_z \tau_{zz}}{r} \right) - g \tag{3.46}
\]
3.3.3 Numerical Approximations

The governing equations are numerically solved by Flow-3D using Finite Differences method. To produce a simple and efficient computation, explicit scheme is applied to evaluate the current time level values of the velocities except those of pressure forces because pressures can only be calculated by implicitly coupling momentum equations and mass equation. Explicit scheme means one unknown value of a variable can be found directly from known values of the variables. However, implicit scheme means the discretization produces an equation where several unknown values are related to several known values, and thus to produce a solution with an implicit scheme a set of simultaneous equations must be solved to find the unknown values.

Discretizing equations (3.45) and (3.46), the form of explicit scheme for the finite differences approximation of velocities is:

\[
v_r^{n+1}_{i,j} = v_r^n_{i,j} + \Delta t^{n+1} \left[ \frac{p^{n+1}_{i+1,j} - p^{n+1}_{i,j}}{\rho (\hat{r}_{i,j} + \hat{r}_{i+1,j})/2} - FRR - FRZ + VISR \right]
\]

and
\[ v_z^{n+1}_{i,j} = v_z^n_{i,j} + \delta t^{n+1} \left[ -\frac{p_r^{n+1}_{i,j+1} - p_r^{n+1}_{i,j}}{\rho \delta r} - FZR - FZZ + VIS - g \right] \]  

(3.48)

where: FRR, FRZ mean the advective flux of \( v_r \) in the \( r \) and \( z \) directions respectively; Similarly, FZR, FZZ mean the advective flux of \( v_z \) in the \( r \) and \( z \) directions respectively. VISR and VISZ are the \( r \) and \( z \) components viscous acceleration respectively. And they are:

**FRR**

\[
FRR = 0.5 \left( \frac{\delta r V_f^n_{i,j} + \delta r V_f^n_{i+1,j} }{\delta r V_f^n_{i,j} + \delta r V_f^n_{i+1,j} } \right) \\
= \left[ \left( \phi^n_{i+1,j} A_r^n_{i+1,j} + v_r^n_{i,j} A_r^n_{i,j} \right) \frac{v_r^n_{i+1,j} - v_r^n_{i,j} }{\delta r} \right] \\
+ \left[ \left( \phi^n_{i,j} A_r^n_{i,j} + v_r^n_{i,j} A_r^n_{i,j} \right) \frac{v_r^n_{i,j} - v_r^n_{i-1,j} }{\delta r} \right]
\]

(3.49)

**FRZ**

\[
FRZ = 0.5 \left( \frac{\delta z V_f^n_{i,j} + \delta z V_f^n_{i+1,j} }{\delta z V_f^n_{i,j} + \delta z V_f^n_{i+1,j} } \right) \\
= \left[ \left( \phi^n_{i+1,j} A_z^n_{i+1,j} + v_r^n_{i,j} A_z^n_{i,j} \right) \frac{v_r^n_{i+1,j} - v_r^n_{i,j} }{\delta z} \right] \\
+ \left[ \left( \phi^n_{i,j} A_z^n_{i,j} + v_r^n_{i,j} A_z^n_{i,j} \right) \frac{v_r^n_{i,j} - v_r^n_{i-1,j} }{\delta z} \right]
\]

(3.50)
\[
\text{VISR} = \frac{\delta r_i + \delta r_{i+1}}{2\rho \left( \frac{\partial}{\partial t} V_j \bigg|_{i,j} + \delta r_{i+1} V_j \bigg|_{i+1,j} \right)} \left[ \left( A_r \bigg|_{i,j} - \frac{\tau_{rr} A_r \bigg|_{i-1,j}}{\delta r_i} \right) + \left( A_z \bigg|_{i,j} - \frac{\tau_{rz} A_z \bigg|_{i-1,j}}{\delta z_i} \right) \right] \left( \frac{\tau_{rr} A_r \bigg|_{i,j} + \tau_{rz} A_z \bigg|_{i-1,j}}{2r_i} \right)
\]

\[
\text{FZR} = \frac{0.5 \delta r_i + \delta r_{i+1}}{\delta r_i V_j \bigg|_{i,j} + \delta r_{i+1} V_j \bigg|_{i+1,j}} \left[ \left( A_r \bigg|_{i+1,j} + v_z \bigg|_{i,j} A_r \bigg|_{i,j} - \frac{\tau_{rz} A_z \bigg|_{i-1,j}}{\delta z_i} \right) + \left( A_z \bigg|_{i+1,j} + v_z \bigg|_{i-1,j} A_z \bigg|_{i-1,j} - \frac{\tau_{rr} A_r \bigg|_{i,j}}{\delta r_i} \right) \right]
\]

\[
\text{FZZ} = \frac{0.5 \delta z_i + \delta z_{i+1}}{\delta z_i V_j \bigg|_{i,j} + \delta z_{i+1} V_j \bigg|_{i+1,j}} \left[ \left( A_z \bigg|_{i+1,j} + v_z \bigg|_{i,j} A_z \bigg|_{i,j} - \frac{\tau_{rz} A_r \bigg|_{i-1,j}}{\delta r_i} \right) + \left( A_r \bigg|_{i+1,j} + v_z \bigg|_{i-1,j} A_r \bigg|_{i-1,j} - \frac{\tau_{rr} A_z \bigg|_{i,j}}{\delta z_i} \right) \right]
\]

And
\[ V_{ij}^{n} = \frac{\delta \mathbf{r}_{i} + \delta r_{i+1}}{2 \rho} \left[ \nabla_{i,j} \left( V_{f}^{n}_{i,j} + \delta \mathbf{r}_{i+1} V_{f}^{n}_{i+1,j} \right) \right] \]

\[
\begin{align*}
\left[ \left( \frac{n}{r} \mathbf{A}_{r}^{n}_{i,j} - \tau_{r}^{n}_{r-1,j} \mathbf{A}_{r}^{n}_{i-1,j} \right) \frac{\delta \mathbf{r}_{i}}{\delta t} \right] + \nabla_{i,j} \left( \tau_{r}^{n}_{r-1,j} \mathbf{A}_{r}^{n}_{i-1,j} \right) + \nabla_{i,j} \left( \tau_{r}^{n}_{r-1,j} \mathbf{A}_{r}^{n}_{i-1,j} \right) + \tau_{r}^{n}_{r-1,j} \mathbf{A}_{r}^{n}_{i-1,j} \right] \right]
\end{align*}
\]

Equations (3.47) and (3.48) evaluate velocities by the advective and viscous terms of the previous time level \((n)\) values. However, the pressures at time level \(n+1\) \((p^{n+1})\) are unknown at the beginning of the cycle, and \(p^{n+1}\) in Equations (3.47) and (3.48) must be placed by \(p^{n}\) to get a first guess for the new velocity.

The pressures must be solved by coupling the Equations (3.47) and (3.48) and the following discretized mass balance equation:

\[
\begin{align*}
\frac{v_{r}^{n+1}_{i,j} \mathbf{A}_{r}^{n+1}_{i+1,j} + v_{r}^{n+1}_{i,j} \mathbf{A}_{r}^{n+1}_{i,j}}{\delta t} + \frac{v_{r}^{n+1}_{i,j} \mathbf{A}_{r}^{n+1}_{i+1,j} + v_{r}^{n+1}_{i,j} \mathbf{A}_{r}^{n+1}_{i,j}}{2 r_{i}} = 0
\end{align*}
\]

Then the updated pressures can be used in Equations (3.47) and (3.48) to get the values of velocities for the next time level.
The initial condition Equation (3.6) is used to evaluate the beginning values of $F|_{i,j}^0$, $A_r|_{i,j}^0$, $A_z|_{i,j}^0$, and $V_f|_{i,j}^0$, while Equation (3.5) is used to determine beginning values of $v_r|_{i,j}^0$, $v_z|_{i,j}^0$, and $p|_{i,j}^0$.

Then the boundary conditions are applied to get the values at the boundary points:

- Equation (3.7) sets $v_r|_{i,j}^n = 0$ and $v_z|_{i,j}^n = v_z|_{i+1,j}^n$ for all cells at $r = 0$.
- Equation (3.8) sets $v_r|_{i,j}^n = 0$ and $v_z|_{i,j}^n = 0$ for all cells at $z = 0$.
- Equation (3.10) sets $v_r|_{i,j}^n = v_r|_{i+1,j}^n$, $v_r|_{i,j}^n = v_r|_{i,j+1}^n$, $v_z|_{i,j}^n = v_z|_{i+1,j}^n$ and $v_z|_{i,j}^n = v_z|_{i+1,j}^n$ for all cells in which $0 < F < 1$ at free moving surface.
- Equation (3.14) sets $p|_{i,j}^n = p_o + 2\gamma_k|_{i,j}^n$ for all cells in which $0 < F < 1$ at free moving surface. $2\gamma_k|_{i,j}^n$ is approximated as the surface tension pressure, detailed in [42].

Additionally, for the Newtonian liquid with constant viscosity, constitutive Equation (3.17) sets:

$$
\tau_{rr}|_{i,j}^n = 2\mu \left( |v_{r+i+1,j}^n - v_r|_{i,j}^n \right) \frac{\partial f}{\partial r_{i+1}}
$$

(3.56)

$$
\tau_{zz}|_{i,j}^n = 2\mu \left( |v_{z+i+1,j}^n - v_z|_{i,j}^n \right) \frac{\partial f}{\partial z_{i+1}}
$$

(3.57)

and
Equations (3.56) to (3.58) are introduced to stress terms in Equation (3.51) and (3.54) to evaluate the viscous accelerations VISR and VISZ.

### 3.4 Experimental Setup

Before experiment, graphite-based lubricant with 54% solid content (weight %) was diluted by distilled water to 5:3, 1:1, 1:5, and 1:20, which are volume ratios of lubricant to water. Droplets of these diluted lubricants were used in the experiment. As illustrated in Fig. 3.5, the experiment was set up with a droplet generator mounted on a fixture to create droplets. A camcorder and a high speed camera took pictures of the droplet formation and its kinetics during impact on the heated surface. The images were later sent to computer for imaging processing to measure deposition dimensions.

Lubricant droplets were generated in diameters of 2mm, 3mm, and 4 mm and made to fall under their own gravity. Impact velocities were obtained between 10cm/s and 200 cm/s by changing the height “H” of the droplet generator. The flat H-13 steel surface was finished to roughness Ra 50 micro-inch (or 1.25μm).
3.5 Measurement of the Lubricant Properties

The lubricant contains many surfactants and binders that change the droplet surface energy, rheology, physics and thermodynamics. The composition of the lubricant is often a trade secret. Hence, we have to determine the properties of the lubricant before we go further to build our fluid dynamic model.
Fig. 3. Schematic experiment setup.
Two methods, pendant drop method and capillary action method, are used to measure the surface tension of the lubricants.

- **Pendant drop method**

In the experiments, lubricant is slowly discharged from a droplet generator. It is shown in Fig. 3.6.

Mechanical equilibrium between surface tension force and gravity can be expressed in the form:

\[
mg = \pi d_{\min} \gamma, \quad \text{or} \quad \rho \left(\frac{1}{6} \pi d^3\right) g = \pi d_{\min} \gamma
\]  

(3.59)

where \(d_{\min}\) is approximately the diameter of the dropper exit, \(\rho\) the density of lubricant, \(\gamma\) the surface tension. From this relation, the surface tension \(\gamma\) is:

\[
\gamma = \frac{\rho, g d^3}{6d_{\min}}
\]  

(3.60)

where \(\rho,\) is the density of lubricant and is obtained through dividing the lubricant weight measured on a balance by the volume measured using a graduated glass cylinder. As long
as we know $d_{\text{min}}$ and $d$, $\sigma$ is derivable. $d_{\text{min}}$ and $d$ are obtained through imaging processing by transporting the images taken by the speed camera.

- **Capillary action method**

Capillary action is the result of adhesion and surface tension. Adhesion of liquid to the walls of a vessel will cause an upward force on the liquid at the edges and result in a meniscus which turns upward. The surface tension acts to hold the surface intact, so instead of just the edges moving upward, the whole liquid surface is dragged upward. The method is schematically shown in Fig.3.7. The height "$h$" to which capillary action will lift liquid depends upon the weight of liquid which the surface tension will lift:

\[
\gamma \cdot 2\pi R = \rho g (hR^2)
\]  \hspace{1cm} (3.61)

So, if the height of lifted liquid $h$ is measured, surface tension can be given as:

\[
\gamma = \frac{\rho gRh}{2}
\]  \hspace{1cm} (3.62)
Fig. 3.6. Generation of single droplets by a droplet generator.
Fig. 3.7. Capillary action method.

3.5.2 Viscosity
With lubricant for which the shear stress and shear rates are proportional (Newtonian flow behavior), the kinematic viscosity can be determined by measuring the time for a volume of lubricant to flow under gravity through a calibrated glass capillary viscometer. This method is recommended by the ASTM445-01. Shown in Fig.3.8, the kinematic viscosity $\nu$ is obtained by multiplying the time of emptying a fixed volume of liquid by a container constant $C$, which is calibrated by a liquid with known viscosity. Then we have the dynamic viscosity by multiplying the kinematic viscosity, $\nu$, by the density of the lubricant.

**3.5.3 Results of the Lubricant Properties**

Table 3.1 shows the measured properties of lubricants. DP stands for pendant drop method in Table 3.1 and CA capillary action method. Liquids tested include pure water. One can observe the facts: a). higher dilution ratio companies with higher surface tension; b). for liquids with dilution ratio larger than 1:1, the surface tension approaches to that of the water. As we can see that all surface tensions estimated by drop pendant method are less than those by capillary action method for about 10 dynes/cm though they share the same trend. Considering the fact that surface tension for water is 72.8 dynes/cm, we regard that results gained from capillary action method are closer to the true values.
Fig. 3.8. Viscosity measurement based on ASTM 445-01.

\[ \nu = C \times t_{\text{flow}} \]
\[ \mu = \nu \times \rho \]
Liquids | Density $\rho_l$ (g/cm$^3$) | $\gamma$ by PD (dynes/cm) | $\gamma$ by CA (dynes/cm) | Kinematic (cSt) | Dynamic (Poise)
--- | --- | --- | --- | --- | ---
5:3 | 1.08 | 40.67 | 52.71 | 56.43 | 0.523
1:1 | 1.065 | 45.15 | 69.89 | 7.22 | 0.077
1:5 | 1.021 | 55.57 | 71.00 | 1.58 | 0.016
1:20 | 1.005 | 59.03 | 73.38 | 1.25 | 0.013
Water | 0.999 | 62.21 | 75.22 | 1.03 | 0.011

Table 3.1. Measured properties of lubricants

Since dynamic viscosity of water is 0.0114 poise, when dilution ratio is large than 1:5, the viscosity of lubricant is approaching to that of the water.
3.6 CFD Simulation

The simulations were controlled to maintain the phenomena in the spreading wetting regions (<250°C) [43]. To numerically compute fluid flow and heat transfer during a lubricant droplet impact is extremely difficult since it involves interaction between such phenomena as free surfaces undergoing large deformations, rapid vaporization of the liquid, and deposition of inside solid particles. We start the CFD model without heat transfer, i.e., only fluid dynamic is considered. This model takes no more than 0.5 second, a time which is very short compared to that of evaporation. Influence of solid particle on flow motion was neglected since the forces that particle exert on the fluid are small compared to the fluid inertia during impact, especially for high diluted lubricant droplets in which particle mass is small.

The simulations were conducted by varying droplet size (2mm, 3mm and 4mm), velocity (10cm/s, 50cm/s, 100cm/s and 200cm/s) and lubricant properties (surface tension and viscosity). Effects of contact angle were neglected in this dynamic simulation, justified by the work of Pasandideh-Fard [15].

The configuration and meshing of the simulation is shown as Fig. 3.9. Fig. 3.10 and Fig. 3.11 give examples of the simulation results of lubricant 1:1 with 4mm diameter droplet at impact velocity 10cm/s and 100cm/s respectively.
It can be observed that the flow recoils back to the center when the droplet reaches the maximum spreading. This can be explained by the balance of surface tension and inertial forces. The inertia of the droplet overcomes the viscous force and surface tension and spreads outwards from the center when the droplet hits the surface. The increasing surface consumes all inertia when the droplet reaches the maximum spreading. After that, the surface tension draws back the droplet and the flow is recoiled. This process cycles several times before the flow is stabilized.

Since the maximum contact area of the droplet with the surface is very important for the heat transfer rate (the larger this area is, the faster the heat can be transferred), we introduced the dimensionless factor, $\xi_{\text{max}} = \frac{D_{\text{max}}}{D_0}$, to describe the spreading characteristics.

### 3.7 Results and Discussions

Since the maximum contact area of the droplet with the surface is very important for the heat transfer rate (the larger this area is, the faster heat transfer can be expected) and film formation, we introduced the dimensionless factor, $\xi_{\text{max}} = \frac{d_{\text{max}}}{d_0}$, to describe the spreading characteristics.
Fig. 3.9. The configuration and meshing of the simulation.

5ms

10ms
Fig. 3.10. simulation results of lubricant 1:1 with 4mm diameter droplet at impact velocity 10cm/s.
Fig. 3.11. Simulation results of lubricant 1:1 with 4mm diameter droplet at impact velocity 100cm/s.

Fig. 3.12 shows the experimental results of $\xi_{\text{max}}$ as a function of die surface temperature. This figure presents that spreading coefficient $\xi_{\text{max}}$ are almost independent of temperature at die temperatures below Leidenfrost points (<300°C for all dilution ratios). The results
of water based graphite lubricants correspond with what observed in studies [23, 32-38] and allow the fluid dynamic model to present the droplet spreading in the region of nucleate and convection boiling.

Fig. 3.13 shows the comparison of experiment and simulation results of $\xi_{\text{max}}$ versus Reyonlds number $Re$ for lubricants with different dilution ratios, while Fig. 3.14 shows the results of $\xi_{\text{max}}$ versus Weber number $We$. Shown by both Fig. 3.13 and Fig. 3.14, results of experiment and simulation agree well except at high values of $We$, the numbers close to critical $We$ number $We_{\text{crit}}$ beyond which droplet will undergo breakup. For all lubricants at high Weber numbers, results of simulation are higher than those from experiments. This may be caused by the neglecting effects of graphite suspended in the liquid in our simulation model. It is observed in the simulations that spreading velocity in radial direction is 5 to 10 times higher than impact velocity at the high end of $We$. The velocity change is high enough to make the liquid with suspensions to be shear-thickening liquid [44-45], i.e., its viscosity increases as the sheer rate increases.
Fig. 3.12. Experimental results of $\xi_{\text{max}}$ v.s. TD. $We = 27$. 
Fig. 3.13. Experiment and simulation results of lubricants, $\zeta_{\text{max}}$ v.s. Re.
Fig. 3.14. Experiment and simulation results of lubricants, $\bar{\gamma}_{\max}$ v.s. $We$. 
The formation of shear thickening may be explained by clustering of graphite particles at high shear rate, and these flow-induced clusters results in an increased dissipation of energy and, consequently, the viscosity increases. Boersma et al. [44] stated that for a suspension to exhibit shear-thickening, two conditions must hold: firstly, the volume fraction of solids in the suspension must be high; secondly, the suspension must be nonflocculated. The lubricants studied in this paper meet both of the two criteria, and shear thickening caused by graphite is reasonable to explain the result difference of simulation and experiment shown in Fig. 3.13 and Fig. 3.14.

From Fig. 3.15 and Fig. 3.16 one can observe that there is a good linear log-log relationship between $\bar{\xi}_{max}$ and Re, We for lubricants. However, two segments are identified: one at low range of Re or We, another at high Re or We for the highly diluted lubricants. This may be caused by the effect of viscosity force coupling with the spreading velocity. At low velocity (low level of Re and We for each lubricant) viscous dissipation can be ignored for high diluted lubricants, but it can be dominant for droplet of lubricant 5:3 spreading on the die surface even at low velocity level. When impact velocity is increased, viscous dissipation becomes dominant even for highly diluted lubricants considering the shear thickening discussed above.

We observed that the higher lubricant is diluted, the larger spreading $\bar{\xi}_{max}$ is obtained for a fixed Re or We number. This is because low diluted lubricant has high viscosity and low surface tension. Viscosity of lubricant with dilution ratio of 5:3 is almost 50 times more than that of lubricant with dilution ratio of 1:20, while surface tension of the former
is more than half of the latter. This means that viscous dissipation is a dominant factor for lubricant with high dilution ratio comparing with the effect of surface tension, especially at the beginning of the impact when the change of surface area is small. Another reason may also rely on the easier shear thickening for high volume fraction of graphite suspensions in the low diluted lubricant.

Ignoring the turning of log-log curves at low Weber number $We$ in Fig. 3.16, we fit these curves by the least square method and obtain the relations shown in Table 3.2. We see that for lubricants with diluted ratio higher than 1:1, the log-log relations between spreading coefficient $\tilde{\xi}_{max}$ v.s. Weber number $We$ are pretty much close. We fit the simulation results for all these lubricants together and have:

$$\log(\tilde{\xi}_{max}) = A + B \log(We)$$

(3.63)

Where $A = 0.3906$ and $B = 0.1038$ for $\beta$ less than equal to one. Equation (3.63) in its simple expression can be used in the evaluation of droplet impact of lubricant with high dilution ratios in metal processing.
Fig. 3.15. Experiment and simulation results of lubricants, $\log(\bar{\xi}_{\text{max}})$ v.s. $\log(Re)$. 
Fig. 3.16. Experiment and simulation results of lubricants, log($\xi_{\text{max}}$) v.s. log(We).
| Dilution Ratios | Simulation | | Experiment | |
|---|---|---|---|
| | A | B | A | B |
| 5:3 | 0.1269 | 0.0679 | 0.1479 | 0.0534 |
| 1:1 | 0.3544 | 0.1093 | 0.3727 | 0.1012 |
| 1:5 | 0.3846 | 0.1154 | 0.4056 | 0.1015 |
| 1:20 | 0.3801 | 0.1257 | 0.3956 | 0.1202 |

Table 3.2. Material constants for different dilution ratios for equation

\[
\log(\bar{\xi}_{\text{max}}) = A + B \log(\text{We})
\]
3.8 Summary of the Chapter

In this chapter, an isothermal model based on momentum equations is proposed for investigating lubricant spread on hot die surfaces. The governing equations are discretized using the VOF finite difference scheme. The lubricant properties needed for this model are determined by simple capillary experiments. This model is applied to the study of droplet dynamics and spread for different lubricant properties and different impact conditions. The model predictions are compared with those of single droplet experiments at different Weber numbers and lubricant dilution ratios. It is seen that the numerical model adequately simulates the droplet behavior including its spread over the entire range of experimental conditions. Finally, a relation is derived between the spread factor and the Weber number that is valid for all dilution ratios greater than unity (dilute suspensions). This relation together with the numerical model can be used to determine the best droplet spray parameters for the desired lubricant film thickness.
CHAPTER 4

MODELING OF THE LUBRICANT DRYOFF BEYOND LEIDENFROST POINT

Lubrication plays a critical role in the efficacy of metal flow, die surface thermal softening, and quality during hot processing of metals. The lubricant is often applied to the hot die surface between 250 to 700°C by spraying a dilute graphite suspension through high pressure nozzles. The atomized droplets in the spray impact on the die surface and undergo film boiling and nucleation boiling. The lubricant film is formed after all the residue liquid on the hot surface is dried off. The evaporation time of droplets is not only a productivity index in hot metal processing but also an important factor for product quality and life of die. This chapter presents an analytical model to estimate the evaporation time of a droplet impact on a die surface heated to temperature beyond Leidenfrost point. Experiments are conducted to measure the dryoff times for lubricants interested. Calculated and experimental evaporation times are compared and agree well for highly diluted lubricants.
4.1 Introduction

In hot deformation processing, the dies are lubricated and cooled by spraying dilute water based lubricants on the heated die surfaces. The lubricant not only cools the hot surface from a temperature as high as 700°C but also deposits a lubricant film that aids in metal flow. For example in hot forging, the lubricant is often a fine suspension of graphite particles in water, with surfactants and binders added to aid in the spreading and the formation of an adherent lubricant film. When droplets of the spray approach the die surface heated to high temperatures, the fast evaporation rate can generate a “cushion” of vapor between the die surface and the droplet to repel the droplet up and support the droplet suspending above the die surface. This is so called film boiling of a droplet. Evaporation time of the droplet not only influences the cooling rate and film formation rate, but also effect of the die life and part quality of the metal processing. Residue liquid lubricant remained in the die can lead to die crack at corners due to high vapor pressure in the forging processes. Entrapped vapors of un-dried lubricant in the melt flow also result in casting porosity for die casting processes. Therefore, the evaporation time of droplets is the key factor to evaluate efficiency of metal processing.

Previous studies on the droplet impact on surface with temperature beyond Leidenfrost have mainly focused on experimental research [46- 49]. Early theoretical approaches were limited to model evaporation of a spherical droplet of which kinetic energy is negligible, i.e., the droplet gently seats on the solid surface. Gottfried et al. [50] modeled
the film boiling of small droplets deposited on a hot flat at Leidenfrost temperature. The model assumed that the droplet is in isothermal at the saturation temperature. Both conduction and radiation were included in their model and the evaporation rate was found by solving mass, momentum and heat balances equations. Wachters et al. [22] considered the impact of droplets about 60 μm impacting a heated surface in the range of 5 m/s. Their model yielded an estimate of the height of the gap between the drop and the surface. Lifetimes of droplets agree well with experimental observations in this study.

Later studies took into account the inertia in the modeling of the impinging droplet. Buyvich et al. [51] developed a heat transfer model in which they assumed the volume of the droplet to be constant. Heat transfer in the liquid and solid was neglected and only heat transfer in vapor was considered. Rein introduced a disk model in which the inertia of both the droplet and the vapor were considered [52]. The disk model did not consider the dynamic of the droplet and took the droplet as a disk with constant radium and thickness. The vapor flow was described as an inviscid flow. Rein showed that the disk model described the main features of the dynamic Leidenfrost phenomenon. Limited droplet impact models were numerically solved by CFD methods. Harvie and Fletcher [53] modeled the behavior of the vapor and heat transfer in the solid and droplet using the computational code they named “BOUNCE”. Their simulation results were validated by experiments and showed good agreement.

This chapter presents the analytical model to estimate the dryoff time of a lubricant droplet impact on a die surface at temperature higher than Leidenfrost point. Instead of
solving the dynamic and evaporation simultaneously, we will separate the dryoff process into two steps: fluid dynamic process and quasi-steady dryoff process, Fig. 4.1. Fluid dynamic process presents the deformation phenomena of the droplet when it impacts on the surface. This model has been discussed in Chapter 3. Fig. 4.1 (a) demonstrates a droplet of diameter $d_0$ impact at the heat surface with an initial velocity $v_0$. The droplet deforms as it hits on the surface and spreads radially. Then the mass of droplet accumulates at the outer ring of the splat and all inertia energy is converted to surface energy, the droplet recoils back to the center driven by the surface tension force. The dynamic process is stabilized when the conversion cycle between inertia and surface energy ends due to viscous dissipation. This process is shown in (b) and (c) of Fig. 4.1. The time it takes to stabilize the deformation process is very short, for example, the time is less than 40 ms for a lubricant droplet of 4 mm diameter with initial impact velocity 1 m/s. The maximum radius the splat is numerically calculated from the fluid dynamic model as we developed for the highly diluted lubricants in Chapter 3. This chapter only cites the results of the fluid dynamic model. Quasi-steady dryoff process is introduced after the droplet deformation is stabilized and kinetic energy has completely converted into surface energy and consumed by viscous dissipation, (d) in Fig. 4.1. It is assumed that liquid in lubricant droplet begin to evaporate and lose mass only when quasi-steady dryoff process starts, i.e., kinematic energy of the droplet is not considered. This assumption is justified because the time scale of fluid dynamic (tens of milliseconds) is much smaller than that of the dryoff (tens of seconds).
Fig. 4.1. Dryoff process of a lubricant droplet at film boiling: (a) fluid dynamic process, (b) quasi-steady dryoff process.
4.2 Droplet Impact on a Hot Surface

In the study of droplet impact on hot surface the following has been observed. At high surface temperatures, vapor forms between the droplet and the surface causing the droplet to bounce off. In water based graphite this leads to pollution with graphite particles depositing on the surrounding surfaces and the press structure. As the surface cools due to heat loss, the droplet stays on the surface but is separated from it by a vapor blanket. The heat transfer is low in this regime of film boiling in Fig. 4.2 (regime IV). The point of least heat flux is called Leidenfrost point.

On further cooling of the surface, the droplet enters a regime of transition boiling (regime III in Fig. 4.2). The heat starts increasing as the steam can escape from the surface. The heat transfer reaches a maximum when nucleate boiling initiates with vigorous convection (regime II in Fig. 4.2). The point of maximum heat transfer is called CHF (critical heat flux). At significantly lower surface temperatures the heat transfer is through natural convection and conduction (regime I in Fig. 4.2).

This chapter concentrates the study of regime IV which represents working conditions of lubricant spray on the die surface. We will discuss the boiling curve of lubricants in the next chapter.
Fig. 4.2. Boiling curve of a droplet deposited on a hot solid surface.
4.3 Fluid Dynamic Model

The fluid dynamic model for lubricants was developed based on momentum equations. The droplet was modeled as isothermal Newtonian liquid in the model. This model simulates the impact of a liquid droplet on a solid from the moment that the lubricant droplet comes into contact with the die surface and proceeding until the spreading process is completed. The surface tension is included in the model as the linear force to calculate the free moving boundary, part of the solution of governing equation. To evaluate the influence of lubricant properties and spray parameters on the lubricant spreading on the die surface, we simulate impact of lubricants with different dilution ratios, droplet size and impact velocity on a polished die surface. Fig. 3.10 and Fig. 3.11 give the examples of the simulation result of lubricant 1:1 with 4mm diameter droplet at impact velocities of 10 cm/s and 100 cm/s.

Summarizing the simulation results, a good log-log linear relationship are found between spreading coefficient $\xi_{\text{max}} = d_{\text{max}}/d_0$ and Weber number $We = \rho_L v_0^2 d_0/\gamma$ for lubricant with dilution ratio higher than 1:5 (volume ratio). Here $d_{\text{max}}$ is the maximum diameter of the splat, $d_0$ the initial diameter of the droplet, $\rho_L$ the density of droplet liquid, $v_0$ initial impact velocity, and $\gamma$ surface tension. The relation is expressed as:

$$\log(\xi_{\text{max}}) = 0.3906 + 0.1038 \log(We) \quad (4.1)$$
Rewriting Equation (4.1), we have:

\[
d_{\text{max}} = 2.4581 d_0 \left( \frac{\rho_L v_0^2 d_0}{\gamma} \right)^{0.1038} \quad \text{or} \quad r_{\text{max}} = 2.4581 r_0 \left( \frac{\rho_L v_0^2 d_0}{\gamma} \right)^{0.1038} (4.2)
\]

where \( r_{\text{max}} \) is the maximum radius of the splat, \( r_0 \) the initial radius of the droplet.

Since the maximum diameter of the splat was observed very close to the diameter of the splat when the dynamic process becomes stabilized, we utilize Equation (4.2) as the initial condition of the quasi-steady dryoff model discussed below, shown in Fig. 4.1 (d).

4.4 Quasi-steady Dryoff Model

4.4.1 Model Description

The heat flux \( q \) due to conduction is given by:

\[
q = k_v \frac{\partial T}{\partial z} = k_v \frac{T_D - T_{\text{sat}}}{\delta} (4.3)
\]

where \( k_v \) is the thermal conductivity of the vapor, \( T_D \) the temperature of the die surface and \( T_{\text{sat}} \) the saturation temperature of the lubricant liquid.
Fig. 4.3. Modeling of the quasi-steady dryoff of a lubricant droplet.
So Equation (4.3) can be written as:

\[ v_{z0} = -\frac{k_v (T_D - T_{sw})}{L_v \rho_v} \frac{1}{\delta} \]  

(4.4)

### 4.4.2 Governing Equations

Assuming the velocity of vapor in z direction (0 < z < \( \delta \)) is negligible, then the mass balance equation for any radius \( r < r_L \) can yield:

\[
\delta \int_0^\delta 2\pi r \, dz = \pi r^2 v_{z0}
\]  

(4.5)

Introducing Equation (4.4), Equation (4.5) becomes:

\[
\delta \int_0^\delta v_{rV} \, dz = -\frac{k_v (T_D - T_{sw})}{2L_v \rho_v} \frac{r}{\delta}
\]  

(4.6)

The general form of momentum balance equations in a two dimensional cylindrical coordinates is:

\[
\rho_v \left( \frac{\partial v_{rV}}{\partial t} + v_{rV} \frac{\partial v_{rV}}{\partial r} + v_{zV} \frac{\partial v_{rV}}{\partial z} \right) = -\frac{\partial p}{\partial r} + \mu_v \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial}{\partial r} (r \mu_v) \right) + \frac{\partial^2 v_{rV}}{\partial z^2} \right)
\]  

(4.7)
\[
\rho_v \left( \frac{\partial v_{zv}}{\partial t} + v_{rV} \frac{\partial v_{zv}}{\partial r} + v_{zV} \frac{\partial v_{zv}}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu_v \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial v_{zv}}{\partial r} \right) + \frac{\partial^2 v_{zv}}{\partial z^2} \right) + \rho_v g 
\]  

(4.8)

where \( p \) is the pressure and \( \mu_v \) the viscosity of the vapor.

Because \( \delta \) is small comparing with \( r_L \), we take the assumptions that: 1). \( v_{zV} \) is negligible, 2). the flow of vapor is steady state, i.e., \( v_{rV} \) is independent of time, 3). viscous terms in \( z \) direction are dominant, i.e., the advection and viscous terms in radial direction are negligible, 4). gravity is not considered. Then the momentum equations reduce to:

\[
\frac{\partial p}{\partial r} = \mu_v \frac{\partial^2 v_{rV}}{\partial z^2} 
\]  

(4.9)

\[
\frac{\partial p}{\partial z} = 0 
\]  

(4.10)

Equation (4.10) indicates that the pressure is a function of \( r \), and Equation (4.9) can be rewritten as:

\[
\frac{\partial^2 v_{rV}(r, z)}{\partial z^2} = \frac{1}{\mu_v} \frac{dp(r)}{dr} 
\]  

(4.11)
4.4.3 Analytical Solution

To solve Equation (4.11), boundary conditions at the die surface and droplet bottom surface are needed. Assumption of no-slip at both surfaces is made and it yields:

\[ v_{rv} = 0 \text{ at } z = 0 \quad (4.12) \]

and

\[ v_{rv} = 0 \text{ at } z = \delta \quad (4.13) \]

Integrating Equation (4.11) twice over \( z \) and introducing Equation (4.12) and (4.13) to determine the integral constants, the velocity \( v_{rv} \) becomes:

\[ v_{rv}(r, z) = \frac{1}{2\mu_v} \left( \frac{\partial p}{\partial r} \right) - \frac{1}{2} \frac{\partial}{\partial r} \left( \frac{\partial v}{\partial r} \right) \quad (4.14) \]

Replacing \( v_{rv} \) in Equation (4.14) into mass balance equation (4.6), we have that:

\[ \frac{\partial p}{\partial r} = \frac{6\mu_v k_v (T_D - T_{sat})}{L_v \rho_v} \frac{r}{\delta^4} \quad (4.15) \]
Taking another boundary condition that:

\[ p = p_0 \text{ at } r = r_L \] (4.16)

and integrating Equation (4.15) yields:

\[ p = p_0 + \frac{3\mu_k k_\nu (T_D - T_{sat}) r_L^2 - r^2}{L_\nu \rho_\nu \delta^4} \] (4.17)

Let constant \( F_c = \frac{3\mu_k k_\nu (T_D - T_{sat})}{L_\nu \rho_\nu} \), which can be regarded as a characteristic force introduced by the vapor generation, then the pressure is:

\[ p = p_0 + F_c \frac{r_L^2 - r^2}{\delta^4} \] (4.18)

It is this pressure that supports the gravity of the droplet. As it is assumed that the droplet is in quasi-steady state and forces exerted on the droplet must be balanced, we have:

\[ \int_0^{\eta} \left( p_0 + F_c \frac{r_L^2 - r^2}{\delta^4} - p_0 \right) 2\pi r dr = \rho_L g V \] (4.19)

where \( \rho_L \) denotes the density of the lubricant and \( V \) the volume of the droplet.
Arranging Equation (4.19), we finally obtain the thickness of the vapor layer as:

\[
\delta = \left( \frac{\pi F_c}{2 \rho_l g V} \right)^{1/4} r_L
\]

Let \( C = \left( \frac{\pi F_c}{2 \rho_L g} \right)^{1/4} \) the coefficient of vapor layer height. Equation (4.20) represents linear relationship between the thickness of the vapor layer \( \delta \) and the radius of the impact splat \( r_L \) for a given volume of the droplet. Since volume of the droplet is a function of time as the liquid continuously escapes as the vapor from the bottom side of the droplet, the height of the vapor \( \delta \) changes with time.

To calculate the mass loss rate of the droplet, Equation (4.20) is introduced to Equation (4.4):

\[
v_{v0} = \frac{k_v (T_D - T_{sat}) V^{1/4}}{C L_v \rho_v} \frac{1}{r_L}
\]

The splat radius \( r_L \) in equation (4.21) can be approximated as the maximum radius \( r_{max} \) of the splat fitted by the fluid dynamic model, which is given by the Equation (4.2). The volume of the droplet \( V \) changes as a function of mass loss rate \( \dot{m} \) and time \( t \):

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\[ V(t) = V_0 - \frac{1}{\rho_L} \int_0^t \dot{m} dt \]  \hspace{1cm} (4.22) 

where \( V_0 = \frac{\pi l_0^3}{6} \) is the initial volume of the droplet and \( 0 \leq t < t_{dry} \), \( t_{dry} \) is the dryoff time.

Introducing Equations (4.21) and (4.22), then the mass loss rate is:

\[ \dot{m} = \int_0^{r_L} \rho_v V_c \Phi 2 \pi dr \]

\[ = \frac{\Phi}{r_L} \int_0^{r_L} \left( V_0 - \frac{1}{\rho_L} \int_0^t \dot{m} dt \right)^{1/4} dr \]

\[ \text{where } \Phi = \frac{2\pi k_v (T_D - T_{sat})}{C_L v} \text{ and } r_L = 1.229 \ln \left( \frac{\rho_L V_0^2 d_0}{\gamma} \right)^{0.1038}. \]

Since the vapor leaves evenly along the radial direction of the bottom surface of the droplet, \( \dot{m} \) is independent of \( r \). Therefore, the term \( \left( V_0 - \frac{1}{\rho_L} \int_0^t \dot{m} dt \right)^{1/4} \) can be taken out of the integral and Equation (4.23) can be rewrite as:

\[ \dot{m} \left( V_0 - \frac{1}{\rho_L} \int_0^t \dot{m} dt \right)^{-1/4} = \frac{\Phi r_L}{2} \]  \hspace{1cm} (4.24)
To find the solution of this nonlinear equation, we have to use numerical iteration techniques. This paper uses Gauss-Newton method to find the solutions of mass loss rate \( \dot{m}(t) \) as a function of time \( t \). The mass loss rate becomes unstable and unreasonably small during the computation when evaporation time \( t \rightarrow t_{dry} \) or the volume of the droplet \( V \rightarrow 0 \). To compensate this but not lose the power of the model, we applied \( \dot{m}(t_{90\%}) \), the mass loss rate at which the remaining volume of droplet is exactly 10% of the initial volume of the droplet, to be the mass lost rate for the rest process of evaporation till all droplet has been dried off.

If we assume that there is no mass loss in the dynamic process and mass loss only caused by evaporation in the quasi-steady dryoff stage, dryoff time of the lubricant droplet in boiling region \( t_{dry} \) is given by:

\[
m_0 = \int_0^{t_{dry}} \dot{m} dt
\]  

(4.25)

where \( m_0 \) is the initial mass of the droplet and \( m_0 = \frac{\rho_L \pi d_0^3}{6} \).

Again, solving Equation (4.25) numerically using the results from Equation (4.24), we can finally obtain the dryoff time \( t_{dry} \).
4.5 Experiment Setup

In this research, the fluid dynamics and thermal behaviors of a droplet impacting on a hot die were experimentally studied by water-based graphite lubricant commonly used in hot forging process. The experiment setup is similar to the one described in Chapter 3 except thermal measurement configuration is included in this experiment.

Again, graphite-based lubricant with 54% solid content (weight %) is diluted by distilled water to 1:1, 1:5, and 1:20 before experiment. Droplets of these diluted ratios are used in the experiment. Properties of droplets are measured by experiments. The results of properties are shown in Chapter 3.

As illustrated in Fig. 4.4, lubricant droplets were generated by a droplet generator with tip diameters of 1.0mm, 1.5mm, and 2.0mm, and made to fall down onto a flat H13 die under their own gravity.

Impact velocities were obtained by changing height of the droplet generator “H” and the range of velocity was controlled at levels of 10cm/s continuously to 200cm/s. A camcorder took pictures at the moment a droplet formed. The impact behaviors of the droplet were also recorded by a speed camera. The images were later sent to computer for imaging processing to measure deposition dimensions and dryoff time. Dryoff time was also measured through a stop watch with precision of 0.01s.
The flat die surface was finished to roughness Ra 50μ inch (or 1.25μm). The die was mounted around by an induction bind heater whose temperature was adjustable from room temperature to 450°C. Temperature at die surface was measured by embedded fast response thermocouples (response time≤ 2ms) which right beneath the impacting area of the die surface. The distance of thermocouple tips to the die surface is 0.5 mm, 1 mm and 1.2 mm separately. The voltage signals of thermocouples were sent to National Instrument (NI) Data Acquisition System so that the temperature could be monitored and recorded.

The thermodynamic properties of vapor, which is regarded as gas phase of water in this study, are function of temperature and pressure and are calculated using fitting equations recommended in [54-56].

**4.6 Results and Discussion**

Fig. 4.5 shows the time-averaged vapor thickness calculated by the dryoff model versus die surface temperature for droplets with different Weber numbers. Since most of the lubricants interested demonstrate Leidenfrost points a few degrees lower than 300°C [57], we start temperature at 305°C to locate the study in the film boiling regime.
Fig. 4.4. Schematic experiment setup.
From Fig. 4.5, the thickness of the vapor layer ranges from around 28 μm to 127 μm. Though there are limited experimental data on the thickness of the vapor layer, Chandra and Aziz [58] measured the thickness of the vapor layer between a deposited nitrogen droplet and a glass surface. They measured the thickness by enlarging the images taken in the experiments for different droplet sizes. The results of thickness of vapor layer beneath the evaporating nitrogen droplet range from 15 μm to 50 μm in their experiments. Since their measurements did not consider the inertia energy and thermal effects, the calculated thickness of the vapor layer by the dryoff model agree well on the magnitude of the results obtained by Chandra and Aziz at low surface temperatures and low Weber numbers.

One can observe from Fig. 4.5 that the height of the vapor layer increases with temperature, evidently at temperatures beyond 370 °C, for all droplets with different Weber number. This is because that the thermal conductivity \( k \) of vapor changes dramatically from 0.1815 W/m/K at 370°C to 0.3238 W/m/K at 380°C [57]. The enhanced conduction generates more vapor than it can be promptly transported to the outside of the shaded area of the droplet and increases the pressure of the vapor layer \( p \). The increased pressure difference \( p - p_0 \) lifts the droplet, and in return, the widened gap decreases the vapor pressure \( p \) until the pressure difference \( p - p_0 \) drops to a level just right to support the weight of droplet, see Equation (4.15).
Fig. 4.5. Averaged thickness of the vapor layer as a function of temperature and Weber number.
Moreover, the thickness of vapor layer increases with the Weber number $We$, illustrated in Fig. 4.5. This is mainly because that, as it can be read from Equation (4.2), high Weber number brings about large spreading radius $r_L$ which enlarges the heat contact area and improves the evaporation. Another direct explanation is that enlarged contact area needs lower pressure to support the weight of the droplet than it does for a smaller contact area, and thereby, a larger vapor layer height.

As Weber number of the droplet continues to increase, the thickness of the vapor layer does not increase accordingly for all range of Weber number. Instead, a different physical phenomenon is introduced for droplet-surface interactions of high Weber number. Studies have shown that droplets of the lubricant break up on die surface at room temperature normally at Weber number $We$ greater than 180, a critical Weber number which can be lower for a heated die surface [43]. Fig. 4.5 indicates that the vapor generated by severe evaporation at high temperature may introduce high transient pressure and help to blow the droplet apart.

Fig. 4.6 illustrates the time-weighted averaged mass loss rate $\dot{m}$ versus die surface temperature for different Weber numbers. The increase of mass loss rate $\dot{m}$ with die temperature correlates with the trend of the thickness of vapor layer shown in Fig. 4.5. The mass loss rate is also observed to increase with Weber number. This is also caused by enhanced heat transfer due to large diameters of splat for a high Weber number impact, demonstrated by our fluid dynamic model. It should be noted that the density of the vapor increases with temperature [54-55]. Therefore, even if the velocity of the vapor
leaving from the droplet keeps the same, the higher temperature will result in more mass
to be taken out by the vapor. This explains the observation that the mass loss accelerates
at the high temperature illustrated in Fig. 4.6.

The dryoff times calculated for a lubricant droplet with initial diameter $d_0 = 3$ mm are
shown in Fig. 4.7 comparing with the dryoff times obtained through experiments. The
droplets impact on the die surface at two impact velocities: 0.1 m/s and 1 m/s. The
calculated dryoff times for droplets with impact velocity $v_0=0.1$ m/s are in the range of
70-90 s, while the range for those measured in experiments is 63-75 s. The comparison
for those droplets with impact velocity $v_0=1$ m/s also shows that dryoff times calculated
by the model are of the correct order comparing with those measured in the experiment
(27-40 s from the model and 34-64 s from experiments).

The interesting finding is that the model predicts longer dryoff times than the times
observed in the experiments for droplets at low impact velocity, but it is opposite for
those with a relatively high velocity.
Die Surface Temperature (°C)
300 320 340 360 380 400 420 440 460
2.0e-7
4.0e-7
6.0e-7
8.0e-7
1.0e-6
1.2e-6
We=0.6
We=14
We=56
We=127
We=182

Fig 4.6. Averaged mass loss rate as a function of temperature and Weber number.
The presence of lower prediction for high impact velocity droplets can be explained by the assumption made in the model that the spreading radius, calculated from fluid dynamic model, does not change with time afterwards at the quasi-steady dryoff stage. This may not reflect the droplet dryoff process very well especially at the late life of the droplet. In fact, the droplets were observed to draw back to a spheroidal shape after impact because of high surface tension at vapor-liquid interface. This behavior reduces the contact area of droplet with die surface and depresses the heat transfer. As the vapor takes more and more mass from the lubricant droplet, the Bond number $Bo = \rho gd_0^2 / \gamma$, a ratio of the gravity to capillary effects, becomes smaller and smaller and capillarity solely determines the shape of the droplet. At this end stage of dryoff, the shape of the droplet is almost spheroidal, which is the poorest geometry condition for heat conduction.

The higher calculated dryoff times for droplets with a low impact velocity may be caused by the underestimate of the spreading radius by the fluid dynamic model. A further study on the effect of graphite suspensions on dry off should also be conducted because it is observed in the experiments that graphite accumulate at the bottom of the droplet when liquid leaves from the bottom of droplet as a form of vapor. The graphite forms a porous phases through which the liquid passed and evaporates to compensate the mass loss in the vapor layer. This phenomenon is not considered in the analytical mode discussed above and may contribute the discrepancy of the prediction and experimental results.
Fig. 4.7. Dryoff time of a droplet with initial diameter $d_0=3\text{mm}$. 
4.7 Summary of the Chapter

A model has been developed to predict the dryoff time of a lubricant droplet impact on a die surface heated to temperatures beyond Leidenfrost point. This model couples two components: fluid dynamic model presented at a separate paper, and quasi-steady dryoff model. Thickness of vapor layer and mass loss rate have been calculated from the model. It is shown that thickness of vapor and mass loss rate are correlated with each other and both increase with die surface temperature and Weber number $We$. Validation of the model is accomplished by comparing dryoff times computed by the model and those observed in the experiments. The calculated results agree with the experimental results in the right magnitude.
CHAPTER 5

PHYSIOTHERMODYNAMICS OF LUBRICANT DEPOSITION

This chapter presents a methodology for studying lubricant behaviour during spraying on hot die surfaces. It consists of a single droplet test to study the thermo-physical phenomena and a “similitude” based approach to extend the results to geometrically complex nozzle sprays. This methodology is applied to the published results on the cooling of hot forging die surfaces and is found to predict both the magnitude and trends reasonably well. It is observed that graphite in water lubricants with high dilutions provide heat transfer coefficients similar to that of pure water while low dilutions provide lower coefficients and longer dryoff times.

5.1 Introduction

In hot deformation processing, the dies are lubricated and cooled by spraying dilute water based lubricants on the heated die surfaces. The lubricant not only cools the hot surface from a temperature as high as 700°C and but also deposits a lubricant film that aids in metal flow. For example in hot forging, the lubricant is often a fine suspension of
graphite particles in water, with surfactants and binders added to aid in the spreading and the formation of an adherent lubricant film. Composition of the lubricant and the selection of the spray parameters are often determined by the cooling and lubrication needs of the intended application. This determination is often based on the domain knowledge of lubricant and equipment suppliers and an empirical trial-and-error procedure during set up. Fundamental understanding of lubricant behavior is lacking.

The work by Bariani et al. [27, 28] on water sprays in hot forging was the first systematic attempt to study the effect of feed pressure and flow rate on the heat transfer coefficient between heated flat dies and the coolant. Their experimental work includes results from two different spray configurations: downwards towards the bottom die and upwards towards the top die. They conclude that an increase in pressure and flow rate enhanced the heat transfer coefficient for both the spray configurations. Studies reported on lubricant sprays in die casting and rolling report similar conclusions that a higher liquid flux density, defined as average liquid flow rate per unit area, provides a higher heat transfer coefficient for a given surface temperature [24, 26, 29, 59-60]. However, the results reported in literature are confusing as the transfer coefficients in water spraying of hot surfaces vary in the range 300 to 60,000W/m2.K; the results being very sensitive to the spraying conditions.

Increasingly, researchers are focusing on understanding the fundamental physics underlying the impact of liquid droplets on hot surfaces [13, 23, 61]. This knowledge not only relevant to industrial processing but also to fuel injectors, ink jet printers, painting
and mist formation. In addition to experimental studies numerical simulations models have also been developed to characterize the physics and thermodynamics of the process. Many of these studies could be relevant to the forging lubrication field.

There are several challenges unique to spray lubrication in hot forging. The lubricant contains many surfactants and binders that change the droplet surface energy, rheology, physics and thermodynamics. The lubricant contains fine particles of graphite (submicron to several microns in diameter) in colloidal suspension. Consequently, the flow is actually multi-phase. The composition of the lubricant is often a trade secret. Therefore, an inverse method must be developed to determine the properties of the lubricant relevant to lubrication deposition and film formation.

The die surface temperatures, at the time of lubricant application, are normally between 300 to 450°C [27, 28, 62, 63]. The heat transfer phenomenon is transient and cyclic. The spray consists of extremely fine droplets of varying diameters distributed spatially. It is almost impossible to measure individual droplet size and their distribution. The die surfaces are deep cavities with complex geometries, inclined surfaces and varying surface temperatures.

This problem is too difficult to solve directly. Hence, this paper presents an approximate methodology to capture the physics and thermodynamics of the process and use it for predicting lubricant behavior. It consists of a simple single droplet experiment to characterize the lubricant properties and study its behavior and a “similitude” based
approach that extends single droplet results to predicting physics and heat transfer in complex liquid sprays. This methodology is applied to the thermal spray experiments from Ref [27] to check its validity.

5.2 Droplet Impact on a Hot Surface

In the study of droplet impact on hot surface the following has been observed. At high surface temperatures, steam forms between the droplet and the surface causing the droplet to bounce off. In water based graphite this leads to pollution with graphite particles depositing on the surrounding surfaces and the press structure. As the surface cools due to heat loss, the droplet stays on the surface but is separated from it by a steam blanket. The heat transfer is low in this regime of film boiling in Fig. 5.1 (regime IV). The point of least heat flux is called Leidenfrost point.

On further cooling of the surface, the droplet enters a regime of transition boiling (regime III in Fig. 5.1). The heat starts increasing as the steam can escape from the surface. The heat transfer reaches a maximum when nucleate boiling initiates with vigorous convection (regime II in Fig. 5.1). The point of maximum heat transfer is called CHF (critical heat flux). At significantly lower surface temperatures the heat transfer is through natural convection and vapor release (regime I in Fig. 5.1).
Fig. 5.1. Boiling curve of a droplet deposited on a hot solid surface.
To study the effect of these phenomena a simple experiment was set up with a syringe mounted on a fixture to create droplets, a hot die steel surface heated by band heaters and a temperature measurement system using thermocouples, Fig. 5.2, the same as Fig. 4.4. A camcorder and a high speed camera took pictures of the droplet formation and its kinetics during impact on the heated surface. The images were later sent to computer for imaging processing to measure deposition dimensions.

5.3 Experimental Results

In the preliminary experiment, 3mm diameter droplets were put on the die surface with zero velocity so that dynamical effects on the boiling behavior were neglected, i.e., quasi-steady state. The flat H-13 steel surface was finished to roughness Ra 50 micro-inch (or 1.25 μm) and heated by a band heater whose temperature was adjustable from room temperature to 450°C. Temperatures during droplet impact were measured by fast response thermocouples (response time ≤ 2ms) embedded 0.5 mm, 1 mm and 1.2 mm beneath the surface of the die. The voltage signals of thermocouples were recorded on a data acquisition system.
Fig. 5.2 Schematic experiment setup.
5.3.1 Heat flux

Fig. 5.3 shows that lubricants with 1:20 and 1:5 dilution ratios have the best overall cooling performance, highest average heat flux, as compared with water in the convective boiling stage. As the dilution ratios are reduced, the behavior shifts from lubricants-in-water to water-in-lubricant and the heat flux reverses. The reason behind this phenomenon can be explained as the balance of the two opposite effects of suspended graphite particles: helping to nucleating bubbles at low dilution ratios (almost as contaminants) and thwarting of the bubble movement (due to high surface tension) at low dilution ratios.

5.3.2 Critical Heat Flux (CHF) and Leidenfrost

With a low dilution ratio, such as 1:20 and 1:5, CHF temperature is around 140°C which seems a few degrees lower than that for water. However, CHF temperature increases up to almost 200°C for both 1:1 and 5:3 lubricants (higher solid content).

One possible explanation of the phenomena is that the fine graphite particles in the liquid facilitate the generation of bubbles in the regime of nucleate boiling and enhance the vaporization rate, which lowers the CHF temperature below that of the water. However, when the dilution ratio goes up the suspended particles become so dense that they start to retard the movement of the bubbles which bring heat and mass out of the liquid surface, and this effect overcomes the enhanced nucleating process.
Fig. 5.3. Measured average heat flux for water and lubricant with different dilution ratio at quasisteady state.
Fig. 5.4. Heat transfer coefficient for lubricant mixtures with different dilution ratios.
Leidenfrost temperature increases as the dilution ratio increases from 1:20 to 5:3 while the Leidenfrost temperature of the water, 285°C in the experiment, is lower than those of the all diluted lubricants. The Leidenfrost temperature is defined as the temperature where the droplet becomes completely separated from the heated die surface by an intervening vapor layer. A droplet with higher dilution ratio requires higher surface temperature to produce the critical vaporization rate to overcome the weight of the droplet and lift itself. This means a higher Leidenfrost temperature.

5.3.3 Heat Transfer and Dryoff

Fig. 5.4 shows the heat transfer coefficient (HTC) between the heated droplet and the heated die surface. Lubricants with 1:20 and 1:5 dilution ratio present the largest heat transfer coefficient values when the die surface is below 200°C with highly dense lubricants showing relatively low heat transfer. The coefficient in the film boiling region, e.g. 0.34 W/m²/K for lubricant with 5:3 dilution ratio compared with that of the water (9.45 W/m²/K). As the die surface is heated to over Leidenfrost temperature and steam formation dominates, HTC becomes independent to the wall temperature as well as almost independent of dilution ratios and keep almost constant.
Fig. 5.5. Measured dryoff time curves for water and lubricant with different dilution ratios.
The dryoff time, defined as the duration from placing a droplet on the surface to complete solid remained, is averaged from the measurements using stop watch and video image combined. As expected, the dryoff times are higher for lubricants with higher solid content because the overall heat flux is lower for less diluted lubricant than that for high diluted ones, Fig. 5.5. It is interesting to note that between 150 to 250°C dryoff time is almost independent of dilution ratio and results based on pure water are sufficient.

5.3.4 Effect of Drop Dynamics on Heat Transfer

Experiments were carried out to quantify the effect of droplet and hot surface parameters such as droplet size, impact velocity and surface temperature were controlled variables in these experiments. Lubricant droplets with the diameters of 2mm, 3mm, and 4 mm were made to fall under their own gravity. Impact velocities were obtained between 10cm/s and 200 cm/s by changing the height “h” of the droplet generator. Non dimensional Weber number was used as it would enable the single droplet experiments to be applicable to lubricant sprays.

The following equations were derived from the regression fit of experimental results for the temperature range of 280 to 450°C and for 100 to 280°C respectively.

\[
h_{w1}(T) = 0.5279 We_{n}^{0.694} e^{-0.079 \cdot 10^{-4} We_{n}} + h_{\text{quasi}}(T) \quad (5.1)
\]
\[ h_{w2}(T) = 1.5347We_n^{0.35} + h_{\text{quasi}}(T) \]  

(5.2)

where \( h_{w1} \) and \( h_{w2} \) are heat transfer coefficients between droplet and die surface for each of the temperature range, \( We_n \) the Weber number of the normal velocity element, and \( h_{\text{quasi}} \) the heat transfer coefficient at quasi-steady state shown in Fig. 5.4. The first terms in Equation (5.1) and (5.2) define the contribution of droplet characteristics, while the second terms introduce the effects of surface temperature on the heat transfer. The Weber number is calculated as follows:

\[ We_n = \frac{\rho v_n^2 D}{\gamma} \]  

(5.3)

where \( \rho \) is the density of the water, \( \gamma \) the surface tension of the liquid, \( v_n \) the normal velocity of the drop.

5.4 Application to Sprays

5.4.1 Methodology for Plain-orifice Atomizers

To extend the single droplet approach to more complex spray systems where the droplet sizes and the impact velocities vary in magnitude and also spatially, the following assumptions are made: (a) The spray system is a plain orifice atomizer, most common in
liquid spraying, in which the jet break up into droplets is promoted by an increase in flow velocity or air pressure. (b) The results of the large single droplet - low velocity experiments can be applied to the small droplet-high velocity application using the “similarity approach”. In this case Weber numbers (We= momentum force/ surface tension force) will be used for this extrapolation. (c) The range of droplet sizes present in the spray can be represented by the Sauter Mean Diameter (SMD).

With the above assumptions, the average diameter of the droplet in a spray, or SMD, can be calculated using relation derived by Elktob [14]:

\[ D = \text{SMD} = 3.08 \nu_L^{0.385} (\gamma \rho_L)^{0.737} \rho_A^{0.06} \Delta P_L^{-0.54} \]  
(5.4)

where \( \nu_L \) is the kinematic viscosity of the liquid, \( \rho_L \) the density of the liquid, \( \rho_A \) the density of the air, and \( \Delta P_L \) the pressure drop across the nozzle. This relation applies to the plain orifice atomizer in which only pressure drop across the nozzle is used to atomize the liquid jet.

The pressure drop \( \Delta P_L \) in Equation (5.4) can be reversely calculated by:

\[ \Delta P_L = \left( \frac{Q}{C} \right)^2 \]  
(5.5)

where \( Q \) is the flow rate and \( C \) a constant representing the nozzle geometry.
The average of normal velocity of the droplets in the spray can be estimated as:

\[ \bar{v}_n = \frac{Q}{A} \]  \hspace{1cm} (5.6)

where \( A \) is the discharge orifice area.

Introducing Equations (5.4), (5.5) and (5.6) into (5.3), the average Weber number \( W_{e_n} \) is calculated for the spray and this is used in Equations (5.1) and (5.2) to calculate the heat transfer coefficient and flow rate of a spray at a specific surface temperature.

**5.4.2 Case Study: Hot Forging Lubrication**

The above relations are used to predict the heat transfer coefficients for the lubricant spray and the die surface temperatures used in the pure water experiments carried out by Bariani et al. [27].

In their study, Bariani et al. generated sprays by atomizing water from a plain-orifice atomizer. Three types of nozzles were used in their experiments [28] and their constant \( C \), defined in Equation (5.5), for these are determined as: 3.7378×10^8 m^3 s^{-1} Pa^{-1/2}, 6.7040×10^8 m^3 s^{-1} Pa^{-1/2} and 1.1991×10^7 m^3 s^{-1} Pa^{-1/2}.
Fig. 5.6. Predicted heat transfer coefficient for lubricant sprays with different dilution ratios.

Assuming that the droplets do not interact with each other and the impacting area of each droplet on the die surface does not overlap, the transient heat transfer coefficient of the
spray can be calculated using Equations (5.1) and (5.2). If it is further assumed the heat transfer coefficient can be averaged over time as the temperature of the die drops from 400°C to 100°C, the overall spray heat transfer coefficient can be predicted. This is plotted for different dilution ratios and flow rates in Fig. 5.6. Since the results in [27] do not distinguish the nozzles types, one can only compare the magnitude and the trend of the calculated heat transfer coefficient. Bariani et al. [27] report heat transfer coefficients for Zone 2 to 4 in the range of 1800 to 1107 W/m2K for pure water. This agrees well with results in Fig. 5.6 for water spray. Zone 1 directly under the spray in [27] has high water accumulation and hence cannot be modeled by the proposed single droplet methodology.

5.5 Summary of the Chapter

This chapter presents a methodology for calculating the heat transfer coefficients in lubricant sprays in hot forging. It consists of two steps. First, the heat transfer characteristics of the lubricants at various thermo-physical regimes on the hot surface are determined using single droplet experiments. Then, the results of these experiments are used along with equations based on a non-dimensional and averaging approach to predict heat transfer coefficients for a complicated spray.

This methodology is applied to calculate the heat transfer coefficients for water spray experiments carried out in References [27] and [28]. Validity of the methodology is confirmed by comparing with average coefficients inverse calculated by Bariani et al.
using FE models [27]. It is found that both the magnitude and trends agree fairly well. It is interesting to note that the heat transfer coefficients for highly dilute sprays are similar to that for pure water spray, while those with high solid content (lightly diluted) are significantly lower indicating less effect of steam.
CHAPTER 6

COMPREHENSIVE APPROACH TO FILM FORMATION, POLLUTION AND HEAT TRANSFER IN HOT DIE LUBRICATION

Lubrication plays a critical role in the efficacy of metal flow, die surface thermal softening, and pollution during hot processing of metals. The lubricant is often applied to the hot die surface between 250 to 600°C by spraying a dilute graphite suspension through high pressure nozzles. This results in the formation of steam, followed by nucleate boiling of the residual liquid, and in the end lubricant spreading and film formation on the hot surface that is continuously loosing heat. This chapter presents a comprehensive approach to study and model the effect of the lubricant properties, dilution ratio, spraying parameters and the transient nature of the thermal and physical phenomena. Results of single the droplet and the spray tests with graphite suspensions in water have been included to demonstrate these effects.
6.1 Introduction

In the hot forging process, lubricant is applied by spraying a dilute water-based graphite suspension on the heated dies during the forging cycle that consists of placing the billet on the preheated dies, forging, ejection and post ejection lubricant spraying. The lubricant spray chills the die surface which may reach a temperature as high as 700°C during forging and about 450°C post ejection spraying. Best operating temperatures for hot working die steels such as H-13 are in the range of 200 to 300°C. Therefore, the spray process effects the deposition of the film and the heat transfer from the die which in turn affect the thermal softening and thermal fatigue of the die surface. A large spray time and slow flow rate increases the forging cycle time, affects productivity and increases thermal softening, while a shorter time (large flow rate, large heat flux) increases thermal fatigue. Current problems like poor lubrication performance, excessive lubricant consumption, and lack of temperature control result from poor design of the spray system or poor selection of spray parameters. Another major problem is air and water pollution during due to the presence of graphite particles in the environment, Fig. 6.1. Overexposure to graphite can cause coughing, dyspnea, black sputum, or impairment of the pulmonary function. This is so important that OSHA has set up a TWA permissible exposure limit (PEL) for graphite that is 2.5 mg/ m³. The goal is to completely eliminate aerosol emissions within the forging plants.
Fig. 6.1. Excessive consumption of lubricant and significant pollution from lubrication.
The work by Bariani et al. [27, 28] on water sprays in hot forging is amongst the first systematic attempt to study the effect of feed pressure and flow rate on the heat transfer coefficient between heated flat dies and the lubricant. Their experimental work includes results from two different spray configurations: downwards towards the bottom die and upwards towards the top die. They conclude that the increase of the feed pressure and the flow rate increases the heat transfer coefficient for both the spray configurations. Studies reported on lubricant sprays in die casting and rolling processes also conclude that a higher liquid flux density, defined as average liquid flow rate per unit area, provides a higher heat transfer coefficient for a given surface temperature [25, 26, 29, 59, 62]. However, the heat transfer coefficients and the heat flux reported in literature vary by factors of magnitude. Therefore, the recent studies have focused on understanding the fundamental physics underlying the impact of liquid droplets on hot surfaces [33, 61]. This knowledge will help explain some of the conflicting results not only in liquid sprays in industrial processing but also those in fuel injectors, ink jet printers, painting and mist formation. In addition to experimental studies, numerical simulations models have also been developed to characterize the physics and thermodynamics of the spray process [13]. Many of these studies could be relevant to the forging lubrication field. Consequently, this paper takes a comprehensive look into the modeling of fundamentals phenomena and their applications to hot forging lubrication.
6.2 Factors Affecting Lubrication in Hot Forging

There are several factors unique to spray lubrication in hot forging: The lubricant contains fine particles of graphite (submicron to several microns in diameter) in colloidal suspension in liquids (surfactants, binders, oil or water) that change the droplet surface energy, rheology, physics and thermodynamics. The composition of the lubricant is often a trade secret and cannot be directly determined. Hence, an inverse method must be used to determine the properties of the lubricants relevant to lubricant deposition and the film formation. The lubricant spray consists of extremely fine droplets of varying diameters distributed spatially. It is almost impossible to measure individual droplet size and distribution. A statistical procedure must be used to capture the spray dynamics. The die surface temperature looses heat continuously due to steam formation, nucleate boiling, convention and dryoff. This heat transfer phenomenon is transient and cyclic. The die surfaces have deep cavities with complex geometries, inclined surfaces and varying surface temperatures. Therefore, a strong coupling exists between the geometrical and thermophysical attributes. This relationship is shown schematically in Fig. 6.2, the same as Fig. 2.1 in Chapter 2. The die surface properties, the lubricant dilution ratios (and the thermophysical properties), and the spray parameters determine the “spraying system”. The parameters of this system affect the deposition mass of the lubricant on the die surface (and its thickness distribution) and the heat flux (and the die surface temperature) as functions of the lubrication time. Arrows in the plots indicate increasing solid content in the lubricant (reducing dilution ratios).
Fig. 6.2. Factors influencing lubricant deposition and die cooling.
The interaction between the sprayed lubricant and the heated die surface is so complicated that it is very difficult to optimize the die cooling, with the lubricant deposition and droplet bouncing off simultaneously. Sometimes one can improve the rate of lubricant deposition but may adversely affect die cooling. Therefore, it is essential to tackle this problem in variety of ways including experiments, mathematical models (deterministic as well as stochastic) and numerical simulations. A comprehensive approach has been adopted at the Center for Excellence in Forging Technology at the Ohio State University in studies that combines continuum mechanics, stochastic modeling and simulation with observations and controlled experiments including:

- Single droplet experiments to investigate the thermo-physics of different phenomena such as steam formation (bounce off), first formation of bubbles on the heated surface (Liedenfrost point, \( T_{\text{Leid}} \)), nucleate boiling (bubbling and wetting), spreading of the lubricant film and lastly the break up of the film at high impact velocities. These phenomena are best represented through non-dimensional Weber number \( W_e \) (ratio of kinetic energy to surface energy) that defines the stability of bubbles in an isothermal system.

- Single droplet numerical modeling and simulations to study the mechanics of different phenomena and to inverse calculate the properties of the lubricants by comparing the numerical predictions to actual observations.

- Spray experiments using industrial lubricant spraying equipment and hot plates to
study the thermo-physical phenomena of multiple droplets with spatially and temporarily distributed parameters.

- Stochastic models of the spraying process (multiple-droplet sprays) that capture the mechanics of the spray process and help determine the optimal spraying conditions for desired deposition and heat transfer.

6.3 Bouncing, Wetting and Breaking up

The experimental set up for single droplet experiments consisted of a droplet generator mounted on a fixture to create droplets, a hot die steel surface heated by band heaters and a temperature measurement system using thermocouples. The setup is shown in Fig. 5.2. A camcorder and a high speed camera took pictures of the droplet formation and its kinetics during impact on the heated surface. The images were later sent to computer for imaging processing to measure deposited film dimensions. Before the experiment, graphite-based lubricant with 54% solid content (weight %) was diluted by distilled water to 5:3, 1:1, 1:5, and 1:20 dilution ratios, which are volume ratio of lubricant to water. Lubricant droplets were generated in diameters of 2 to 4 mm, and made to fall under their own gravity. Impact velocities were obtained between 10 and 200 cm/s by changing the height of the droplet generator. The flat H-13 steel surface was finished to roughness Ra 50 micro-inch (or 1.25 μm) and heated by a band heater whose temperature was adjustable from room temperature to 450°C. Temperatures during droplet impact were
measured by fast response thermocouples (response times ≤ 2 ms) embedded 0.5, 1.0 and 1.2 mm beneath the surface of the die.

When a droplet impinging on a heated die surface with an initial velocity, it can spread on the surface, break up to smaller droplets, or be rebounded from the surface [64 – 68], similarly as in droplet impacts occurring under conditions without phase-change. Not only dynamical parameters, particularly nondimensional Weber number (\(We = \frac{\rho v^2 d_0}{\gamma}\)), but also thermal characteristics determine which result of the impact will be. Weber number \(We\) is considered as the ratio of inertia force to the surface force, where \(\rho\) is the density of the droplet liquid, \(v\) the normal component of impact velocity, \(d_0\) the initial diameter of the droplet before impact, and \(\gamma\) the surface tension of the droplet liquid. Without phase-change taking place, droplets will spread, and sometimes rebound when the Weber number is small. Above a critical Weber number an impact results in a disintegration of the drop, i.e., in a splash. The critical Weber number of splashing depends on the roughness of the surface and also on the Reynolds number. Actually, limiting conditions of splashing without phase-change are often expressed in terms of the Ohnesorge number instead of the Weber number. However, in studies on the interaction of droplets with hot walls normally a dependence on the Weber number is considered. We will therefore stay with the Weber number in the following.

When drops impinge on a wall having a temperature that is above the saturation temperature of the liquid, the wall temperature becomes another important variable - in
addition to the Weber number. The importance of further thermal parameters or of other nondimensional numbers relating to thermal aspects has rarely been considered.

Therefore, in our experimental study of droplet impact on hot die surfaces both the Weber number and the wall temperature have been changed with the other variable at a fixed value. Waters and also lubricant with different dilution ratios were used in the experiments. The experimental setup remained the same illustrated in Fig. 5.2.

In this study, regimes characteristic of droplets with different dilution ratio were mapped in a Weber number ($We$) versus die surface temperature plots. As shown in Fig. 6.3 the safe lubricant zone (constrained by bubbling, bounce off, Leidenfrost temperature and spreading) increases as the solid content (less dilution) increases.

The experimental results are summarized in Fig. 6.3 (a), (b), (c) and (d) for lubricants with four different dilution ratios. These characteristics are compared with those of water. When Weber number is larger than a critical number, the droplet will always breakup to smaller droplets. It is observed that all critical Weber numbers of the lubricants are larger than that of the water. The more diluted the lubricant is, the closer the critical Weber number to that of the water. This may be caused by the high viscosity of the lubricant which decreases another important nondimensional parameter, Reynolds number $Re = \rho Vd_o / \mu$, where $\mu$ is the dynamic viscosity of the droplet liquid. For 5:3 lubricant, the Reynolds number drops below the order of 10, which means the viscous dissipation effect cannot be neglected. Part of the kinetic energy of the impacting droplet has to be
used to overcome the viscous shear force. Therefore, larger Weber number is expected to contribute to the increased surface energy due to breakup and viscous dissipation than that of the water droplet.

It is observed in Fig. 6.3 that the Leidenfrost temperature relating to a droplet with an impacting velocity is greater than the quasi-state Leidenfrost point shown in Fig. 5.3 and Fig. 5.4, i.e., the Leidenfrost point rises with the increasing of the velocity for a given droplet. This is because the bounce-off is caused by the high pressure of the vapor generated between the droplet and die surface. Now, in addition to balancing the gravity of the droplet, the direction of its momentum also needs to be changed, and this requires high surface temperature to generate faster evaporation rate. This phenomenon is the dynamic analogue to film boiling and has therefore been called a dynamic Leidenfrost phenomenon.

Fig. 6.3 also shows that the breakup is enhanced with decreased critical Weber number when the die surface is heated above boiling temperature. The reason behind is obvious that the fast boiling initiated by the contact of the droplet and die surface generates dynamic pressure which helps to blow apart the droplet. However, this effect is not evident for the lubricants with high dilution ratio. The stable evaporation rate of the lubricant which is independent of temperature may contribute to this phenomenon.
Die Surface Temperature $T_D$ ($\degree$C)

- Bounce-off
- Bubbling Wetting
- Breakup

1:1 Lubricant
Water

$T_{Leid}$

$T'_{Leid}$

$W_{ecrit}$ $W_{ecrit}'$
We observe the following:

- **Spreading Wetting**
- **Bubbling Wetting**
- **Bounce-off**
- **Breakup**

The diagram shows the relationship between the Die Surface Temperature \( T_D \) (°C) and \( \text{We} \), with the critical values indicated as \( \text{We}_{\text{crit}} \) and \( \text{We}_{\text{crit}}' \).

---

(c)
Fig. 6.3. Regimes of different phenomena of a lubricant droplet impinging on heat die surface comparing with water. (a). 1:20 Lubricant, (b). 1:5 Lubricant, (c). 1:1 Lubricant, (d). 5:3 Lubricant.
6.4 Heat Flux During Lubricant Spraying

The spray cooling test was carried on a 20” diameter, 2” thick hot plate of H13 steel with the aid of Acheson fix nozzle 682 spray system. A straight nozzle pipe with the inner diameter 0.375” is chosen for spray. A 7100W ceramic radiant heater supported by paralleled frame was used to heat up this steel plate to temperatures as high as 350°C. Fifteen thermocouples were installed 0.02” beneath the hot plate surface at different horizontal locations from the center of the hot plate. Temperature was measured at various spray heights H (14, 17 and 20”), liquid pressures $P_l$ (45, 60 and 75psi), and dilution ratios (water and 1:20 dilution). Temperatures were measured at selected positions (P1=0.0”, P2=1.5”, P3=2.5” and P4= 3.5”).

Temperature history data was continuously recorded and logged into computer through data acquisition system. Selected results from these experiments are included in Fig. 6.4. From the plots in this figure, the following is observed:

- The instant heat transfer coefficient (HTC) peak happens after spray starts (0.5s to 1.0s), reaches around 30000 W/m²K and drops to 10000 W/m²K gradually for the location closest to the center of the spray. This drop off may be due to the heat conduction from the neighboring locations at longer times. The temperature drop at this location is also the highest. However as the spraying continues to 2s the heat transfer at farther locations catch up with an average HTC around 10 to 15 KW/m²K
for most points for 1:20 dilution ratio and 60psi air pressure, Fig. 6.4(a).

(a). Different thermocouple locations: 1:20 dilution ratio, Pl = 60 psi, H = 14".
(b). Different liquid pressures: 1:20 dilution ratio, at Point P0 (H = 14”).
(c). Different heights: 1:20 dilution ratio, at Point P0 (P1 = 45psi)

Fig. 6.4. Temperature history and heat transfer coefficients in the first 2 seconds.
For liquid pressure at 45psi, 60psi, and 75psi, the mass flow rates of water are rated at 0.618GPM, 0.736GPM, and 0.830GPM respectively, when air pressure is fixed at 75psi. At 45 and 60 psi HTC increases with water flow rate but at higher 75 psi the water atomizes and lower heat transfer is observed, Fig. 6.4(b). The reasons behind this anomaly needs further study.

Plots of temperature history and HTCs for different spray height imply the opposite effects of droplets velocity on HTC, Fig. 6.4(c). At higher heights the impact velocities are high and the droplets quickly fly away hot plate when they hit the hot plate. This bouncing off phenomenon is also observed in the single droplet experiments. After 0.5 seconds the HTC for 20” height catch up as a liquid layer is now present and the droplets are splashing into this.

6.5 Summary of the Chapter

This chapter first outlines the problem of forging spray lubrication and the relations between the spray parameters and the forging process parameters. Then it describes the comprehensive research approach adopted at CEFT to model, predict and control the phenomena of pollution, heat transfer and lubricant deposition. This approach includes
single droplet experiments to investigate fundamentals of the deposition process, and the spray experiments to investigate the relationship between spray parameters and the heat transfer coefficients. Based on the preliminary results the following conclusions can be reached:

- Results from single droplet test show that both Weber number $We$ and die surface temperature play important role on the lubricant wetting on the die.

- Low dilution ratios (with higher solid contents) provide for greater lubricant process window. However, it is possible that lower dilution may also provide lower graphite air pollution due to higher Leidenfrost temperature (die surfaces below 430°C) may not experience bounce off. Also the lower dilution ratios delay the break-up of the lubricant film to higher Weber numbers ($We > 325$) due to higher viscous energy that stabilizes the bubbles.

- The spray test results show that both liquid pressure and spray height increase kinetic energy, cause droplets bouncing off from the hot plate, and hinder spray cooling. This is a reverse of the role of flow flux density (flow rate) in increasing the heat flux. This indicates that for lubricant sprays, the nozzle design and atomization will play a major role in controlling the different physical and thermal phenomena. This is the goal of the ongoing study.
CHAPTER 7

CONCLUSIONS AND FUTURE WORK

7.1 Conclusions

The main objective of this research is to build models that relate formation of lubricant film and heat transfer phenomena of the die to lubricant properties and lubricant application characteristics. Studies focused on physical phenomena of single droplet of water based graphite lubricant were conducted to fulfill the objective. Mathematical models based on balance equations were developed to predict the interaction behaviors between lubricant droplets and the hot die surface. Experiments were conducted to validate the models and also to build an empirical heat transfer model.

The work completed can be summarized as follows:

1. An isothermal model for lubricant spread on the die surface is presented to study the dynamics and spread of lubricant droplet on the hot die surface. The predictions were validated by experimental results at different Weber numbers and lubricant dilution
ratios. A relation was derived between the spread factor and the Weber number that is valid for all dilution ratios greater than unity (dilute suspensions).

2. The dryoff model for die temperatures beyond the so-called Leidenfrost point was developed to predict dryoff time of the lubricant with high dilution ratios. The model was solved under the assumption that the lubricant droplet begins to evaporate only after the droplet deformation has stabilized and the kinetic energy has been completely consumed. The dryoff times were predicted from the model for different droplet impacts and compared well with experimental results.

3. Heat transfer experiments were conducted to build an empirical heat transfer coefficient model and to map regions of physiothermodynamics of lubricant deposition on the hot die. This model was applied to calculate the heat transfer coefficients for water spray experiments. It was found that both the magnitude and trends agree fairly well. It is interesting to note that the heat transfer coefficients for highly diluted sprays are similar to that for pure water spray, while those with high solid content (lightly diluted) are significantly lower indicating less effect of steam.

4. Single droplet experiments were conducted to investigate fundamentals of the deposition, bounce-off, and breakup of the droplet as outcomes of droplet-surface interactions. Results showed that both Weber number \( We \) and die surface temperature play an important role on the lubricant wetting on the die. Low dilution ratios (with higher solid contents) provide for greater lubricant process window. However, it is
possible that lower dilution may also provide lower graphite air pollution since higher Leidenfrost temperature (die surfaces below 430°C) may not experience bounce off.

5. The spray test results showed that both liquid pressure and spray height increase kinetic energy, causing droplets to bounce off from the hot plate, and hinder spray cooling. This is a reversal of the role of flow flux density (flow rate) in increasing the heat flux. This indicates that for lubricant sprays, the nozzle design and atomization will play a major role in controlling the different physical and thermal phenomena.

7.2 Research Significance and Contribution

Through this project we get a fundamental understanding of the formation of lubricant film and heat transfer phenomena of the die relating to lubricant properties and lubricant application characteristics during the lubricant spray in hot forging. This is expected to yield the following results of industrial relevance:

1. The dynamic/ thermal models of lubricant spray can help in optimizing lubricant performance in hot forging processes or other hot metal working processes. Engineers can use these models to optimize spray parameters if film patterns and heat transfer of the spray are given. The objective of the optimization is energy saving and pollution elimination.
2. The developed models can also assist engineers in lubricant selection. Given lubrication and thermal requirements, the models can inversely calculate the lubricant properties, based on which the lubricant is selected.

3. The knowledge of the relationship between spray performance (film deposition and thermal cooling) and spray parameters / lubricant properties is very helpful for engineers to design spray applicators.

7.3 Suggested Future Work

To have a full understanding of spray lubrication in hot metalworking processes, the following tasks considering various aspects should be further investigated in the future:

1. Extensions to the research on computing spread and dryoff include the improvement of the current method to take into account the effects of graphite suspensions numerically and experimentally. Models to address the effects of colloidal suspension inside the lubricant droplet would improve the accuracy of the prediction of spray performance in lubricant application.

2. Analytical and experimental studies on the interaction between lubricant droplets and more complex surface properties, such as roughness, temperature distribution, oblique surface and arbitrary geometries should be carried out. These models can be used to
predict lubricant spray performance for industrial die cavities, which are composed of several basic geometries.

3. The statistical model which will relate the droplet models to the spray models through stochastic processes should be developed. This model would directly relate the spray phenomena to the physical mechanisms obtained in the single droplet studies.
LIST OF REFERENCES


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