PREDICTION OF THERMAL DISTORTION AND THERMAL FATIGUE IN SHOT SLEEVES

DISSEMINATION

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ABSTRACT

In the horizontal cold chamber die casting process, shot sleeves experience alternate heating and cooling as well as intensification pressure at the biscuit end – all of which influence the shot sleeve service life. Since the shot sleeve failure is expensive in terms of cost and downtime, it is important to understand the effects of design parameters and process conditions on shot sleeve failures. Due to the difficulties of measuring shot sleeve temperature and distortion in the actual manufacturing process, computer simulation is not only a useful method to understand and predict failures in shot sleeves, but also a tool to guide shot sleeve design.

The 2-D plane strain computer models were developed at the beginning of the work. The simulation results from the 2-D models were compared with the existed experimental data, and the best model was identified. Based on the experimental data, accurate heat transfer coefficients between the shot sleeve and molten metal were determined.

To model the shot sleeve radial distortion at biscuit end, the plane strain model was extended to 3 dimensions in order to properly apply boundary conditions. Based on the available commercial shot sleeve geometry and process data, the radial distortions of large shot sleeve with thin/thick wall thickness were studied.
Based on the computer simulated strain range results from the same 3-D model, a thermal fatigue life prediction method (Universal Slope Method) was then developed for service life prediction. Compare the predicted thermal fatigue life with commercial sleeve fatigue data, the predicted thermal fatigue life is well matched with the actual operation shot sleeve thermal fatigue life.

Combining the small ID and large ID shot sleeve simulation results, it can be concluded that control of the “out of roundness” of the ID dimensional changes is the key factor for large ID shot sleeve clearance design between its ID and plunger tip.
Dedicated to My Family –
My Parents, My Son Muchen and Wife Weidong.
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1.1 INTRODUCTION

Die casting is an extremely important and economical manufacturing process which has extensive applications in the automobile, appliance, electronics, and other industries. The die casting process has the capability of high production rates and long production runs. Die castings typically have good strength and ductility, high quality, shape complexity, with good dimensional accuracy and surface detail, and require little or no subsequent machining or finishing operations. Components such as pins, shafts, and fasteners can be cast integrally. In the fabrication of certain parts, die casting can compete favorably with other manufacturing methods, such as sheet metal stamping and forging, or other casting process [1].
The general layout for a horizontal cold chamber die casting machine is shown in Figure 1.1 [2]. In the cold chamber die casting cycle, the shot sleeve is partially filled with certain amount of molten metal through a pouring hole. A plunger is then guided through the shot sleeve forcing the molten metal into a closed die cavity. The plunger remains advanced in the shot sleeve for a certain prescribed time to maintain pressure on the metal and then moves forward to push biscuit out. Then, the die is opened, the solidified casting part is ejected and the die and plunger are coated or sprayed with coolants and lubricants. All of the four steps are shown in Figure 1.2 [3]. The shot sleeve and plunger are the main components responsible for holding and forcing molten metal into die cavity. To ensure a high quality and economic die cast product, it is very important to have a good performance and long service life of the shot sleeve.

Figure 1.1. General layout of horizontal cold chamber die casting machine [2]
The cycle time for a typical casting sequence can be anywhere from 30 to 150 seconds depending on the size of the machine and the part being produced [4].

Figure 1.2. Horizontal cold chamber die casting machine sequence of operations: (a) Shot sleeve is filled with molten; (b) Plunger or ram forces metal into die; (c) Die is opened; and (d) Casting is ejected [3]

Figure 1.3 schematically shows die casting shot sleeve, and the relevant die end, plunger tip and pouring hole position and characteristic appearance [5].
1.2 PROBLEM DESCRIPTION

Due to the cyclic thermal and mechanical loads, a variety of failure and deterioration modes occur in shot sleeves such as: washout under the pouring hole, longitudinal and radial deformation that cause plunger tip sticking and wear, gross cracking, and heat checking [5].

The distortion in shot sleeves inner diameter leads to either an excessive clearance or an insufficient clearance between shot sleeve and plunger tip. Excessive clearance may cause metal blow back past plunger tip and results in retention of aluminum in the sleeve; or aluminum solidifies in the gap between sleeve and plunger tip to create additional friction, ultimately leading to premature ware of sleeve-tip system. Insufficient clearance
may cause plunger tip sticking problem. The proper design of the clearance around the shot sleeve at biscuit end will ensure the shot sleeve is firmly located without extra load being induced.

Thermal fatigue is one of the most common problems encountered in the die casting process, which also affects the performance of shot sleeves. The commonly accepted definition of fatigue is:

The phenomenon leading to fracture under repeated or fluctuating stresses having a maximum value less than the tensile strength of the material. Fatigue fractures are progressive, beginning as minute cracks that grow under the action of fluctuating stress. For thermal fatigue, a somewhat similar type of fatigue fracture is caused not by repeated mechanical stresses but by cyclic thermal stresses [6].

Due to the cyclic, rapid, non-uniform heating and cooling in die casting shot sleeves, the phenomenon fall into the above-defined category, which can be called thermal fatigue. Thermal fatigue is one of the major causes of poor quality castings, cost of tooling, downtime, productivity and safety.

The research work initiated in early 1997 with a thin wall commercial shot sleeve used for transmission case manufacturing. At the beginning, the target cycle time is 90 seconds with the thin wall shot sleeve design. Due to the problems of washout under pouring hole and occasional longitudinal cracking of the shot sleeve, the cycle time have
to last 10 to 20 seconds longer. At that time, the lubrication used between shot sleeve and plunger tip was graphite-based lube. The plunger tip life was about 2500 cycles. The thin wall sleeve cost about $2500. In order to reduce the cycle time and increase the productivity, a newly designed H13/Copper sandwich structure shot sleeve with water cooling passage on the bottom side under pouring hole was introduced. And also the lubrication material between shot sleeve and plunger tip was changed to shot heads. Upon all the changes, the plunger tip life sharply dropped to 739 cycles in average due to the cold shot sleeve surface temperature, which couldn’t melt shot beads. There were also problems of the sleeve outer shell rotating with respect to the inner shell, causing the runner system of die to be offset, which perturbs cavity filling. Also some tip sticking problems occurred during operations. The cost of H13/Copper/H13 sandwich sleeve is about 4 times higher than the original standard H13 shot sleeves. With all the problems mentioned above, the plant decided to go back to the standard H13 sleeves with increasing wall thickness at the beginning of 1999. With the thick wall shot sleeve design, the cost of sleeves reduced to the original level, and a better sleeve life has been achieved. Also the tip sticking problem has been solved, and its life went back to the original level.

However, it is difficult to explain the exact mechanisms responsible for the failure phenomena. Furthermore, it is not easy to obtain measurements on shot sleeves to check temperature distributions and deformations while they are in service. To predict and avoid shot sleeve failures and its consequences, it is essential to understand the stress/strain field, deformation and temperature distribution of a shot sleeve during its
service and their impact on shot sleeve life and deformation. One way to better understand shot sleeve problems and failures is to develop a computer model to analyze shot sleeves during the die casting process. Once developed and compared with experimental data, a computer model could be used to show the reasons for chronic plunger tip sticking or sleeve distortion and thermal stress and fatigue problems in a shot sleeve in commercial use. Results from the model could be used to suggest potential shot sleeve design guidelines and process parameters to alleviate the problems.

1.3 RESEARCH OBJECTIVES

The objective of this research was to develop a model or method for conducting FEM computer simulations to study shot sleeve temperature distribution, stress/strain distribution and radial distortion. Based on the simulations, the goal was to predict failure mechanisms and shot sleeve service life, and also to suggest improved designs for shot sleeves.

To achieve the goal, the research tasks were divided into the following parts:

- **Fundamental model development** - With the consideration of real die casting process, develop a accurate and efficient computer model for multi-cycle shot sleeve distortion, temperature, stress and strain distribution analysis. Verify the computer model with experimental data.
- **Prediction of shot sleeve distortion** – Based on the method developed, evaluate the effects of design parameters on shot sleeve distortion at both biscuit end and middle of shot sleeve. A key factor here was the clearance between shot sleeve and die, and between shot sleeve and plunger tip.

- **Thermal fatigue life prediction** – Develop a method for predicting thermal fatigue life based on the simulation results of stress/strain data.

- **Design guideline of shot sleeves** - Based on the radial deformation analysis and thermal fatigue life prediction, develop a shot sleeve design criterion for clearance between sleeve ID and plunger tip and for desired service life.

### 1.4 OUTLINE OF THE DISSERTATION

The dissertation is structured in the following way. Chapter two provides literature review of the theoretical background necessary for understanding both the existing problems and the research efforts made in this area.

Chapter three presents the method chosen to study the existing shot sleeve problems. The Finite Element Model development and experimental verification works are presented in this chapter.

Chapter four presents the results of research performed on commercial shot sleeves used for automotive transmission case manufacturing. The shot sleeve distortion at
biscuit region and non-biscuit region, as well as the effects of runner shape, nose radii and wall thickness of shot sleeve were all been studied in details in this chapter.

Chapter five presents the method for thermal fatigue (heat checking) prediction.

Chapter six presents the design criterion and development of design guideline for shot sleeve fatigue life and OD/ID design clearances.

Chapter seven summarizes the work of the dissertation, and presents ideas for further research related thermally induced distortion and thermal fatigue.
CHAPTER 2

LITERATURE REVIEW AND THEORETICAL BACKGROUND

2.1 SHOT SLEEVE FAILURE MECHANISMS

Since most of the problems in shot sleeves are due to the thermal effects, therefore, the thermal induced deterioration and failures of shot sleeve – the leading failure mode, deserve a through and systematic analysis.

In general, due to the high temperature and heat load from the molten metal, the consistent and smooth performance of shot sleeves are significantly affected and the life is reduced. In addition to the heat, mechanical loading of the cover die, high injection pressure of molten metal and initial set up of the die casting machine also induce considerable stress in a shot sleeve. All these factors combine to cause various deterioration and failure modes, such as washout, soldering, wear, heat checking, and
even catastrophic failure, i.e., gross cracking and circumferential cracking. “Washout” refers to the gradual loss of shot sleeve material on the inside bottom under a pour hole. It results from a mechanism of over-tempering of H13 shot sleeve steel from high operating temperature [5]. Soldering is defined as when cast material (usually Aluminum) combines with the die steel to form a compound that becomes part of the die steel surface. This tendency is greatly accelerated when the die steel is at higher temperatures. Both washout and soldering failure modes are not the focus of the current research work. The remaining failure modes can basically be divided into three aspects: crack, distortion and thermal fatigue. Sometime, the shot sleeve failure appears in one mode; on the other hand, shot sleeves can show the combination of various modes of deterioration and failure [5].

2.1.1 Shot Sleeve Crack

Gross cracking of a shot sleeve is a catastrophic failure, which has often been observed in many die casting plants, such as the longitudinal cracking and circumferential cracking of shot sleeves shown in the following figures [5].
Figure 2.1. Longitudinal gross cracking on shot sleeve (bottom portion) [5]

Figure 2.2. A partial longitudinal cracking on shot sleeve [5]
Figure 2.3. Longitudinal gross cracking on shot sleeve (top portion) [5]

Figure 2.4. Circumferential gross cracking on shot sleeve [5]
Kelm [7] claimed that longitudinal cracking is due to the ramming of plunger tip into a shot sleeve inner diameter, which becomes an ‘out-of-round’ shape by uneven thermal expansion. Also any mechanical failures might be caused by a soldering buildup on the inside shot sleeve or plunger tip, or the misalignment of plunger tip with shot sleeve, or any kind of pre-existing damage on the inside surface of shot sleeve. Sobol [8] found that the cracking of shot sleeve was usually observed at the die end portion. He also noticed that cracking occurred at any changes in outer diameter of a shot sleeve along the die, which is due to the stress concentration created by abrupt outer diameter changes.

Park [5] systematically studied shot sleeve catastrophic failure modes based on the theoretical analysis work. Based on the analyzing of hoop tensile stress induced by intensification pressure, with the assumption of a small crack is oriented along the longitudinal direction on the inside surface of a shot sleeve, ‘crack opening mode’ is present to cause the longitudinal cracking. The criterion is if a stress intensity factor (K_I) exceeds plane strain fracture toughness (K_{IC}) of H13 steel, and then fracture occurs.

2.1.2 Shot Sleeve Distortion

From the deformation point of view, there are two major types of distortion – longitudinal and radial deformation. Since a shot sleeve is not filled 100% with molten metal, the bottom and lower region shot sleeve wall experiences more thermal load than the top portion of the shot sleeve and introduces uneven thermal expansion of the shot sleeve locally. On the cross section of shot sleeves, due to the uneven temperature
distribution, temporary distortion of the shot sleeve in cross section will occur. The shape distortion of cross section, the deviation from a circle to an ellipse or any other ‘out-of-roundness’ shape, is called “radial deformation”. Usually, the lower portion of shot sleeve is filled with molten metal while the upper portion has no contact with molten metal, this causes the lower portion of the shot sleeve to expand more than the upper portion, and the shot sleeve deforms longitudinally. This type of distortion is called “out-of-straightness”, “bending” or “longitudinal” deformation. Figure 2.5 shows these two distinctive types of shot sleeve distortion commonly observed in die casting industry [5] [9]. These phenomena are well known and have been a main research topics in both industry and academia [5] [7] [9] [10] [11] [12].

Figure 2.5. Two types of commonly observed shot sleeve distortion [5]
Distortions in radial and longitudinal directions due to thermal imbalance can cause very serious metal injection problems. If a proper clearance between a shot sleeve and plunger tip is not maintained, smooth movement of the plunger tip is not guaranteed. Too large a clearance allows molten metal to penetrate backward past the plunger tip. Clearances that are too small allows the distorted shot sleeve to pinch the plunger tip. Due to the radial distortion, the plunger tip can be either too tight or loose which can causes irregular injection of molten metal into the die. Irregular injection may result in poor quality casting. It is known that a change of 0.005” in diameter and 0.001” in straightness of a shot sleeve can yield a high scrap rate [8]. A clearance of 0.001” to maximum +0.006” per inch I.D of a shot sleeve between the Cu-Be plunger tip and H13 steel shot sleeve is usually recommended and used in the design [14]. At the same time, Robbins claims that ideally the clearance between the plunger tip and shot sleeve should be less than 0.004” to prevent liquid aluminum from penetrating backwards [13]. Actually, a clearance between shot sleeve and plunger tip as small as possible is the best choice unless the elastic deformation of shot sleeve induces friction. When a plunger tip and shot sleeve are exposed to hot molten metal, their magnitudes of thermal expansion would not be the same since the temperature of the plunger tip is lower than that of the shot sleeve. Meanwhile, for a given temperature difference, a typical Cu-Be plunger tip will expand at a larger rate than H13 steel shot sleeve, since the coefficient of thermal expansion is $1.7 \times 10^{-5}/\degree C$ for a typical Cu-Be alloy and $1.1 \times 10^{-5}/\degree C$ for H13 steel. Depending on the grade of plunger tip alloy, the coefficient of thermal expansion is sometimes twice as large as the coefficient of thermal expansion of H13 steel [14]. These
two competing factors (temperature and coefficient of thermal expansion) determine the change in clearance between the shot sleeve and plunger tip.

For controlling the thermal expansion, the control of temperature becomes a very important issue. In order to lower the temperature of a shot sleeve, a water jacket or cooling channel is sometimes installed in a shot sleeve. In general, any cooling device to control the temperature of a shot sleeve is believed to be effective against washout and distortion. It is reported that a water jacket under a pouring hole may curb washout tendency, however it aggravates heat checking on the shot sleeve inside surface [7]. The reason of this cracking or heat checking is because the temperature gradient near the pouring hole is too high, and it causes that area to expand and contract too much. So designing the cooling line and position should be carefully done.

Shot sleeve design starts with a choice of four basic shot sleeve styles: two-piece shot sleeve, standard one-piece shot sleeve, split collar and threaded shot sleeve [15]. To reduce the effect of thermal imbalance, new designs and materials have been invented. For example, Zecman [16] designed HI3/Copper composite shot sleeve with a circumferential copper layer inserted around the shot sleeve at a certain depth with cooling water passages close to the copper layer. It aims to readily dissipate excessive heat from the bottom toward the top of a shot sleeve by conduction and take off heat by cooling water. Also, Robbins [17] used a copper alloy heat disperser outside of the shot sleeve. However, the optimal thickness of the copper band was not specified. Some new shot sleeve products already have been used in industry, such as HI3/Copper composite
shot sleeve with cooling lines inside the shot sleeve used in GM die casting plant, and KUBOTA shot sleeves constructed of high-grade ceramics material [18].

2.1.3 Thermal Fatigue Failure

Most of the early studies on thermal fatigue were based on experimental work. There are several test methods only conducted in laboratory [19]:

- Quenching (thermal shock) test: A test bar was heated to different temperature and quenched in water to investigate the high temperature creep and resistance to thermal fatigue of various grades of iron.

- Mushroom test: A mushroom shaped test specimen was used to simulate a cylinder head. With alternative heating and cooling, the depths of crack were measured to study the effect of carbon content on fatigue resistance.

- Constrained thermal fatigue test: a specimen similar to a normal tensile bar is placed in a rigid fixture, and specimen was induction heated. With this test, the thermal stress can be measured.

- Constrained disc test: A constrained disk was used with applying heat on one side and keep constant temperature on the other side. The test can be used to determine the number of cycles required for cracking as a function of the maximum temperature for different materials. Results shown that steel was most resistance to failure, nodular iron provided intermediate life, and gray irons cracked in the shortest times [19].

- Finned disc thermal shock test: A set of finned discs mounted on a rod retainer is cycled between two fluidized beds maintained at different temperatures. Due to the
shape of finned specimen, the fin was heated and cooled more rapidly than that of hub, and thermal stress were developed during cycles. This method was used to study the cracks for different materials.

- Bridge thermal fatigue test: The specimen is a flat disk containing holes. With alternative heating and cooling, cracking occurred in the bridge between holes. The resistance to cracking was studied for different structures. It is shown that with increasing the structure strength, the resistance to cracking increases. Testing also showed that gray iron had the lowest resistance and ductile iron had the highest resistance.

Much work has been done to compare the thermal fatigue behavior of cast iron. Some notable contradictions were pointed out by Rukadikar and Reddy [20]:

- In some studies, tensile strength was used to correlate the thermal fatigue resistance, while in others, tensile strength was a poor predictor for thermal fatigue. For example, bainitic gray irons, having a high tensile strength, did not appear to provide good resistance to thermal fatigue;
- High carbon contents in gray irons often were found to provide good thermal fatigue resistance, but in some studies, the high carbon contents did not appear to be helpful;
- While a number of studies indicated that alloying additions, such as Mo, provided improvement in thermal fatigue resistance, others indicated no beneficial influence of Mo.
In an effort to more closely duplicate the conditions found in permanent molds, the earliest study related thermal fatigue were conducted by Wallace [21]. A specimen was prepared from H13 material that measures 2” × 2” × 7” as shown in Figure 2.6. The specimen was then austenitized at 1900°F for one and one-half hours and then oil quenched. This was followed by tempering cycles at about 1100°F for one hour and oil quench to give the final hardness value of 46 HRC. The model was then surface ground and polished so each corner has a 0.010” radius with all corners square to within +/- 0.003” of an inch. After the specimen was well prepared, it was tested for a total of 15,000 cycles. After every 5,000 cycles the specimen was removed from the test unit, and fatigue crack initiation and growth were evaluated. Wallace’s “dunk test” as a laboratory means of simulating the temperature cycles encountered in die casting dies can be used to study the effects of several variables on the thermal-cyclic behavior of various tool steels, including H13 for casting manufacturing process. The test was calibrated so that temperature fluctuations in test pieces measured during testing were similar to those measured in production of H13 die casting dies.
From the basic theory of fatigue, it is the cyclic stresses and strains that cause fatigue. Unfortunately, due to the limitation, most of the tests mentioned above only recorded heating and cooling conditions and material properties, but no stress and strain data were collected.

To overcome this limitation of the test, Rosbrook [22] developed a finite element approach for thermal fatigue study in die casting dies based on Wallace’s test. In his work, a computer model was developed to simulate Wallace’s test part. Figure 2.7 shows the cross section of the test part. Due to the symmetry of the geometry, only 1/8 of the part was modeled and the mesh used in the computer model is shown in Figure 2.8.
Figure 2.7. Cross section of the Wallace's test part [22]

Figure 2.8. Computer modeled portion of the cross section [22]
In Rosbrook’s study, the simulated process was divided into 3 steps:

- Heating process – including injection of molten metal and solidification. In this step, heat flow into die with very high heat transfer coefficient;
- Convection process – heat goes out of die due to the die open. The heat transfer coefficient in this step is relatively low;
- Spray process – heat was taking out of the die by applying lubrication. The heat transfer coefficient is relatively high.

In Rosbrook’s study, the final heat transfer coefficient during the whole process was determined by comparing the simulated temperature data with Wallace’s temperature measurement results. Then the stress and strain range during cycles was recorded from computer simulation and universal slope methods was applied for thermal fatigue life prediction. The predicted thermal fatigue is close to the experimental results. But this work is limited only to lab test.

Later, Wei [19] used the same method to predict mold thermal fatigue life in permanent mold casting process. In his study, the predicted crack position in the mold was exactly the same as that in the experiment, and the predicted and experimental thermal fatigue lives were of the same magnitude.
2.2 HEAT TRANSFER STUDY

2.2.1 Heat transfer Coefficient in Die Casting Process

Since thermal loads play a critical role in die casting manufacturing process, it is extremely important to study the heat exchange between molten metal and tooling interface, as well as the heat transfer phenomenon in the entire process. In a die casting cycle, it includes several different heat exchange stages and cannot be simply characterized by a single heat transfer coefficient at the interface between die tooling and molten metal. Different heat transfer coefficients must be applied at different stages.

There are a variety of data collected for heat transfer coefficient in die casting process. Due to the different test processes and boundary conditions, the heat transfer values are quite different. Papai [23] conducted an extensive literature review from the available data. The heat transfer coefficients from his review ranged from 3.3 kW/m²K to 87 kW/m²K. In the study of A380 in H13 tooling steel casting process, Papai found that the surface roughness has an affect on heat transfer coefficient between the die and molten interface. For rough die surfaces, the heat transfer coefficient was about 80 kW/m²K with initial alloy temperature of 1202 °F, while with smooth die surface, the heat transfer coefficient was about half of that value.

Other researchers also studied the interface heat transfer coefficients in casting processes. Their results are shown in Table 2.1.
<table>
<thead>
<tr>
<th>Casting Material</th>
<th>Mold Material</th>
<th>Heat Transfer Coefficient (kW/m²K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A380</td>
<td>Low Carbon Steel</td>
<td>7.0 in runner [24]</td>
</tr>
<tr>
<td>A380</td>
<td>H13</td>
<td>9.0 in runner [24]</td>
</tr>
<tr>
<td>99.9 Aluminum</td>
<td>Steel</td>
<td>20 at 30 MPa [25]</td>
</tr>
<tr>
<td>A380</td>
<td>H13</td>
<td>79.87 [26]</td>
</tr>
<tr>
<td>Al-Si</td>
<td>Steel</td>
<td>50 at 38 MPa [27]</td>
</tr>
<tr>
<td>Al</td>
<td>Steel</td>
<td>3.3 [28]</td>
</tr>
<tr>
<td>Al</td>
<td>H13</td>
<td>10 [29]</td>
</tr>
<tr>
<td>Al</td>
<td>H13</td>
<td>10 [30]</td>
</tr>
<tr>
<td>Al</td>
<td>H13</td>
<td>20.8 [31]</td>
</tr>
</tbody>
</table>

Table 2.1. Experimental interface heat transfer coefficient for Aluminum casting in steel dies

Takach [32] studied the effect of metal state on the heat transfer coefficient during solidification process and reported the heat transfer coefficient values of 61.97 kW/m²K when metal at liquid state, 60.57 kW/m²K when metal at mushy state and 18.51 kW/m²K when metal at solid state, respectively. Sekhar [33] and Nishida [25] studied the effect of cavity pressure on the heat transfer coefficient. Their studies shown that heat transfer coefficients changed from 0.84 kW/m²K when there is no pressure to 34 kW/m²K when the pressure is 91 MPa and 52.5 kW/m²K when pressure is 196 MPa, respectively.

After castings are ejected from cavity, spray cooling and lubrication are applied. At this stage, a different heat transfer coefficient exists. Lee [34] investigated the effect of spray flux at different initial test plate surface temperatures on heat transfer coefficients during spray process, and the data are given in Table 2.2.
<table>
<thead>
<tr>
<th>Spray Flux (m³/s m²)</th>
<th>Heat transfer coefficient (kW/m²K) at 302 °F</th>
<th>Heat transfer coefficient (kW/m²K) at 482 °F</th>
<th>Heat transfer coefficient (kW/m²K) at 662 °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0002</td>
<td>1.2</td>
<td>2.5</td>
<td>1.5</td>
</tr>
<tr>
<td>0.0012</td>
<td>2.5</td>
<td>3.8</td>
<td>3.2</td>
</tr>
<tr>
<td>0.0024</td>
<td>4.0</td>
<td>6.0</td>
<td>5.4</td>
</tr>
<tr>
<td>0.0048</td>
<td>7.2</td>
<td>10.0</td>
<td>9.8</td>
</tr>
</tbody>
</table>

Table 2.2. Effect of heat flux on heat transfer coefficient [34]

Bishop [35] studied the effect of spray fluid temperature on spray heat transfer coefficient. From his study, the heat transfer coefficients were given with the values of 14.2 kW/m²K, 20.3 kW/m²K and 30.5 kW/m²K with respect to different spray water temperature of 75 °F, 126 °F and 176 °F, respectively.

Sirinterlikci [36] conducted extensive reviews for the previous studies of heat transfer issues in the die casting processes. From his works, the heat transfer coefficients between open die surface and surroundings were given in the range from 0.02 kW/m²K to 0.1 kW/m²K. The heat transfer coefficients between cooling lines and die casting tooling were given in the range from 4.6 kW/m²K to 10 kW/m²K. And the heat transfer coefficients between die plates were given in the range from 1.0 kW/m²K to 10 kW/m²K.
2.2.2 Shot Sleeve Temperature Measurement

In order to verify the heat transfer coefficient during the casting cycles and shot sleeve distortion, some experimental tests were conducted.

Friedrich [37] used an experimental approach to study the positional and dimensional changes in shot sleeves under normal production conditions. From his results, it was concluded that thermal imbalance along with mechanical loads, lead to the corresponding shape and dimensional changes in the shot sleeve. There were no further details available regarding the temperature and distortion information.

The earliest temperature test stand used for shot sleeve was developed by Paliani [38]. From his experimental setup, the temperature and radial distortion has been measured. The original test diagram is shown in Figure 2.9.

![Diagram of Paliani's shot sleeve test stand](image-url)

**Figure 2.9. Paliani’s shot sleeve test stand [38]**
This test stand was further modified to measure temperature data at the inner diameter at 3 different sections along the length of the shot sleeve [39]. The modified test stand consisted of a die, the shot sleeve, a plunger tip, a hydraulic cylinder and thermocouple probes. A shot sleeve of inside diameter 1.98 inches (50.3 mm) and thickness 0.5 inches (12.7 mm) of H13 tool steel material was instrumented with thermocouple probes along the longitudinal and radial direction in order to collect temperature data at the molten metal–sleeve interface. These thermocouple probes each consisted of two pairs of K-type thermocouples, fused to the probe metal at two different depths. The probes were made from H13 tool steel and the thermocouple wire was GTAW welded to the probe providing an accurate position of the thermocouple junction. The thermocouple wires were in turn connected to a data acquisition system, which recorded thermal data over extended periods of time. The die was designed with a variable cavity volume to accommodate different shot sleeve fill percentages. Figure 2.10 shows the modified experimental set-up.
Figure 2.10. Improved experimental setup for temperature measurement [39]

To better understand the location of the thermocouple in the shot sleeve, the notation used is shown in Figure 2.11. The probes were located at various longitudinal distances from the pour hole centerline (A), and radial location (B). The distance between the two-thermocouple pairs in each probe was measured (C).
Molten metal A380 alloy was poured into the sleeve and allowed to cool into a “log”. This log was pushed out using a hydraulic actuated plunger tip and the sleeve was allowed to cool for a specified period of time. Temperature data was collected during 44 consecutive such cycles during which the amounts of lubricant, type of lubricant and the fill percentage were varied. After every cycle, the outside diameter of the shot sleeve was measured using a micrometer in both vertical and horizontal directions. The temperature data collected was then used to calculate various thermal variables using the basic heat transfer equation.
Based on the above mentioned test stand, computer model were set-up to calculate the
temperature distribution and the suggested heat transfer coefficient was given from 275
W/m²K to 1400 W/m²K for different testing conditions [40].

2.2.3 Computer Model for Shot Sleeve Simulation

To study the thermally induced deformation and fatigue, due to the limitation of
experimental approaches, computer simulation is a good method for getting stress and
strain data, as well as dimensional variation data.

Most of the early simulation work to study the shot sleeve was done in the
Engineering Research Center for Net Shape Manufacturing in the Ohio State University
research [9], two simplified approaches were used for the predicting temperature and
defformation in the shot sleeve of a cold chamber die casting machine. These two
approaches were a one-dimensional approach for prediction of temperature and bending
defformation and two-dimensional approach for prediction of temperature and bending
and warping deformations. Based on the two methods, a parametric study to investigate
the effects of (a) the initial temperature of shot sleeve and molten metal; (b) the length-to-
thickness ratio and the type of shot sleeve material; (c) the filling percentage on
by using ANSYS software to study the effects of different factors and different types of
shot sleeve on shot sleeve deformation. All the above simulations were limited to only
one casting cycle.
The work of Penhollow et al. [4] is an extension and continuation of the study of Juang’s research and the model was developed by using ABAQUS finite element software, which was capable of conducting multiple casting cycle simulations. After the development of that model, Penhollow re-investigated the effects of the parameters on shot sleeve deformation: (a) filling percentage (30%, 50% and 60%); (b) the eccentricity of the shot sleeve suggested by Juang; and (c) preheating of the shot sleeve. A 2-D model was used to obtained temperature and displacement histories of the shot sleeve. A multi-cycle computer model was developed in order to incorporate the effects of casting rate on sleeve deformation. Each step was divided into two periods, metal filling/dwell and sleeve empty. The model was further enhanced by the use of a layering subroutine to simulate the gradual fill process of the shot sleeve. The deformation data was analyzed by calculating an “out of roundness” value for the sleeve as well as individual nodal displacement. The study results indicated that:

- Radial deformation decreases with increasing shot sleeve preheating temperature
- Radial deformation increases with increasing shot sleeve eccentricity
- Fill percentage did make a significant difference to radial deformation

Palani et al. [12] improved the simulation accuracy by conducting the experiment work mentioned above to get accurate heat transfer coefficient between shot sleeve and molten metal, and applied this value to his multiple cycle 2-D simulations study for direct verification of the computer model. His work can be summarized as following several steps:
• Initial simulations based on previous researcher’s work;
• Refined model based on experimental set-up;
• Improved model based on die casting machine test;
• Simulations based on the experimentally verified model for three possible machine startup procedures.

The initial simulation indicated that the temperature distribution is very sensitive to heat transfer coefficient. By performing with different heat transfer coefficients during simulated process with comparing the experimental measured temperature data, the heat transfer coefficient of 1200 W/m²K was shown to be the better value for die casting molten metal and shot sleeve interface.

2.3 SHOT SLEEVE MATERIAL PROPERTIES

H13 steel is widely used tool steel in die casting manufacturing process. For different steel suppliers, due to the different manufacturing practice, their material properties are not the same. The following tables show the related material properties from different sources.
2.3.1 Young’s Modulus

Table 2.3 and 2.4 are the temperature dependent Young’s Modulus data for H13 tool steel from three different suppliers. From different sources, the values of Young’s Modulus are little different.

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>E (MPa) × 10^5</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>2.07</td>
</tr>
<tr>
<td>200</td>
<td>2.00</td>
</tr>
<tr>
<td>400</td>
<td>1.86</td>
</tr>
<tr>
<td>600</td>
<td>1.97</td>
</tr>
<tr>
<td>800</td>
<td>1.90</td>
</tr>
<tr>
<td>1000</td>
<td>1.59</td>
</tr>
</tbody>
</table>

Table 2.3. Temperature dependent Young’s Modulus for H13 – (I) [41] [43]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>E (MPa) × 10^5</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>2.03</td>
</tr>
<tr>
<td>750</td>
<td>2.04</td>
</tr>
</tbody>
</table>

Table 2.4. Temperature dependent Young’s Modulus for H13 – (II) [42]

2.3.2 Thermal Expansion Coefficient

The following tables are the temperature dependent thermal expansion coefficient used for material suppliers. There is not much difference among the collected thermal expansion coefficient data.
<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal expansion (1/°F) × 10^6</th>
</tr>
</thead>
<tbody>
<tr>
<td>80-200</td>
<td>5.8</td>
</tr>
<tr>
<td>80-400</td>
<td>6.3</td>
</tr>
<tr>
<td>80-800</td>
<td>6.9</td>
</tr>
<tr>
<td>80-1200</td>
<td>7.3</td>
</tr>
<tr>
<td>80-1500</td>
<td>7.5</td>
</tr>
</tbody>
</table>

Table 2.5. Temperature dependent thermal expansion coefficient for H13 – (I) [41] [43]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal expansion (1/°F) × 10^6</th>
</tr>
</thead>
<tbody>
<tr>
<td>70-450</td>
<td>7.0</td>
</tr>
<tr>
<td>70-1450</td>
<td>7.7</td>
</tr>
</tbody>
</table>

Table 2.6. Temperature dependent thermal expansion coefficient for H13 – (II) [42]

2.3.3 Thermal Conductivity

The following tables are the temperature dependent thermal conductivity from the database of different material suppliers.
<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal Conductivity (W/m°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>9.78</td>
</tr>
<tr>
<td>400</td>
<td>13.00</td>
</tr>
<tr>
<td>800</td>
<td>13.94</td>
</tr>
<tr>
<td>1200</td>
<td>14.89</td>
</tr>
</tbody>
</table>

Table 2.7. Temperature dependent thermal conductivity for H13 – (I) [41]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal Conductivity (W/m°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>14.41</td>
</tr>
<tr>
<td>750</td>
<td>14.79</td>
</tr>
<tr>
<td>1450</td>
<td>15.66</td>
</tr>
</tbody>
</table>

Table 2.8. Temperature dependent thermal conductivity for H13 – (II) [42]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal Conductivity (W/m°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>9.77</td>
</tr>
<tr>
<td>400</td>
<td>12.99</td>
</tr>
<tr>
<td>800</td>
<td>13.93</td>
</tr>
<tr>
<td>1200</td>
<td>14.88</td>
</tr>
</tbody>
</table>

Table 2.9. Temperature dependent thermal conductivity for H13 –(III) [43]
2.3.4 Other Related Material Properties

Other related material properties are also available as shown in the following tables:

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Rockwell C</th>
<th>Ultimate Tensile Strength (MPa)</th>
<th>Yield Strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>500</td>
<td>48</td>
<td>1447.0</td>
<td>N/A</td>
</tr>
<tr>
<td>700</td>
<td>51</td>
<td>1820.3</td>
<td>1579.0</td>
</tr>
<tr>
<td>750</td>
<td>48</td>
<td>1358.3</td>
<td>N/A</td>
</tr>
<tr>
<td>800</td>
<td>44</td>
<td>1179.0</td>
<td>951.5</td>
</tr>
<tr>
<td>900</td>
<td>44</td>
<td>1075.6</td>
<td>882.6</td>
</tr>
<tr>
<td>1000</td>
<td>44</td>
<td>1089.4</td>
<td>N/A</td>
</tr>
<tr>
<td>1000</td>
<td>48</td>
<td>999.8</td>
<td>724.0</td>
</tr>
<tr>
<td>1100</td>
<td>44</td>
<td>758.4</td>
<td>606.8</td>
</tr>
<tr>
<td>1250</td>
<td>48</td>
<td>503.3</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 2.10. Temperature dependent hardness, ultimate tensile strength and yield strength for H13 [41]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Density (kg/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>7778.4</td>
</tr>
<tr>
<td>750</td>
<td>7667.7</td>
</tr>
<tr>
<td>1450</td>
<td>7529.3</td>
</tr>
</tbody>
</table>

Table 2.11. Temperature dependent density for H13 [42]
<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Yield Strength (MPa)</th>
<th>Ultimate Tensile Strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>758.3</td>
<td>965.2</td>
</tr>
<tr>
<td>1100</td>
<td>586.0</td>
<td>792.8</td>
</tr>
<tr>
<td>1200</td>
<td>310.2</td>
<td>482.6</td>
</tr>
<tr>
<td>1300</td>
<td>137.9</td>
<td>206.8</td>
</tr>
</tbody>
</table>

Table 2.12. Temperature dependent yield strength and ultimate tensile strength for H13 [42]

From their database, the specific heat is constant, which value is 460.2 J/(kg K) [42].

2.3.5 Material Properties Used in Computer Models

In the early computer study, Paliani [12] used a constant thermal conductivity 13.88 (W/m·°F), density 7778.4 (kg/m^3) and specific heat 255.7 (J/(kg·°F)). His thermal expansion and Young’s Modulus are temperature dependent as shown below:

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Young’s Modulus (Mpa) × 10^-5</th>
<th>Thermal expansion (× 10^6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>2.14</td>
<td>N/A</td>
</tr>
<tr>
<td>200</td>
<td>2.07</td>
<td>6.6</td>
</tr>
<tr>
<td>400</td>
<td>2.00</td>
<td>6.8</td>
</tr>
<tr>
<td>600</td>
<td>1.86</td>
<td>6.8</td>
</tr>
<tr>
<td>800</td>
<td>1.79</td>
<td>7.2</td>
</tr>
<tr>
<td>1000</td>
<td>1.52</td>
<td>7.4</td>
</tr>
<tr>
<td>1200</td>
<td>1.10</td>
<td>7.8</td>
</tr>
</tbody>
</table>

Table 2.13. Temperature dependent material properties in computer model [12]
Sirinterlikci [36] improved the material property data with adding the temperature dependent thermal conductivity and specific heat in his computer simulation as shown below:

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal Conductivity (W/m °F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>122</td>
<td>13.75</td>
</tr>
<tr>
<td>212</td>
<td>13.85</td>
</tr>
<tr>
<td>302</td>
<td>13.95</td>
</tr>
<tr>
<td>392</td>
<td>14.05</td>
</tr>
<tr>
<td>482</td>
<td>14.15</td>
</tr>
<tr>
<td>572</td>
<td>14.25</td>
</tr>
<tr>
<td>662</td>
<td>14.35</td>
</tr>
<tr>
<td>752</td>
<td>14.45</td>
</tr>
<tr>
<td>842</td>
<td>14.55</td>
</tr>
<tr>
<td>932</td>
<td>14.65</td>
</tr>
<tr>
<td>1022</td>
<td>14.75</td>
</tr>
<tr>
<td>1112</td>
<td>14.85</td>
</tr>
<tr>
<td>1202</td>
<td>14.95</td>
</tr>
<tr>
<td>1292</td>
<td>15.05</td>
</tr>
<tr>
<td>1382</td>
<td>15.15</td>
</tr>
<tr>
<td>1472</td>
<td>15.25</td>
</tr>
<tr>
<td>1562</td>
<td>15.35</td>
</tr>
<tr>
<td>1652</td>
<td>15.45</td>
</tr>
</tbody>
</table>

Table 2.14. Temperature dependent thermal conductivity in computer model [36]
Temperature (°F) | Specific Heat (J/kg °F)  
---|---
122 | 261.1  
212 | 266.3  
302 | 271.6  
392 | 276.8  
482 | 282.0  
572 | 287.2  
662 | 292.4  
752 | 297.7  
842 | 302.9  
932 | 308.2  
1022 | 313.4  
1112 | 318.6  
1202 | 323.9  
1292 | 329.1  
1382 | 334.3  
1472 | 339.6  
1562 | 344.8  
1652 | 350.1

Table 2.15. Temperature dependent specific heat in computer model

Compare all the material properties from different sources, the differences are very limited. Due to the temperature changes during operation, temperature dependent material properties were used in computer simulation.
2.4 METHOD FOR THERMAL FATIGUE PREDICTION

In chapter one, the definition of thermal fatigue has been given. It is necessary to reiterate: fatigue is a form of failure that occurs in structures subjected to dynamic and fluctuating stresses. When materials are subjected to a fluctuating load, failure occurs at a stress level much lower than the fracture stress corresponding to a monotonic tension load. Due to cyclic loading, the failure occurs by crack initiation, then propagation until the cracks become unstable. For thermal fatigue, the cause is not by repeated mechanical stresses but by cyclic thermal stresses.

When material failure occurs under a relatively large number of cycles, and stresses and strains are within the elastic range of the material, the failure mechanism is called high-cycle fatigue. If the magnitude of the fluctuating stress is no longer in the elastic range of the material, significant plastic straining occurs throughout the body, especially in the highly localized areas at stress concentration sites, and the number of cycles to failure is expected to be relatively short. This failure mechanism is referred to as low-cycle fatigue. Low-cycle fatigue life is usually associated with a number of cycles to failure between 100 and 10,000 cycles (depending on material strength and ductility) and for high cycle fatigue the number is above 10,000 cycles [44].

The applied stress may be axial (tension-compression), flexural (bending, or torsional (twisting) in nature. In general, three different fluctuating stress-time modes are possible: (1) Regular and sinusoidal time dependent, wherein the amplitude is symmetrical about a mean zero stress level, termed reversed stress cycle. (2) Maximum and minimum
stresses are symmetrical relative to the zero stress level, which is termed repeated stress cycle. (3) Stress levels may vary randomly in amplitude and frequency.

The mean stress, \( \sigma_{\text{m}} \), range of stress \( \sigma_r \), stress amplitude \( \sigma_a \) and stress ratio \( R \) are the commonly used technical terms used in fatigue analysis and are defined below:

\[
\sigma_m = \frac{\sigma_{\text{max}} + \sigma_{\text{min}}}{2} \tag{2.1}
\]

\[
\sigma_r = \sigma_{\text{max}} - \sigma_{\text{min}} \tag{2.2}
\]

\[
\sigma_a = \frac{\sigma_{\text{max}} - \sigma_{\text{min}}}{2} \tag{2.3}
\]

\[
R = \frac{\sigma_{\text{min}}}{\sigma_{\text{max}}} \tag{2.4}
\]

where \( \sigma_{\text{max}} \) is maximum stress, \( \sigma_{\text{min}} \) is minimum stress.

The following paragraphs are the reviews of some existing methods used for fatigue life prediction and their theoretical background.

2.4.1 S-N (Stress-Life) Curve Approach

The S-N curve approach is a widely used tool to assess fatigue of many modern structures that are subjected to repeated loading, where the applied stress is under the elastic limit of the material and the number of cycles to failure is large.
In evaluating the number of cycles to failure for a given structure subjected to in-service fluctuating loads, fatigue test data representing the load environment must be available. The basic concept is that the life of a structure part is the same as the life of a test specimen if both have undergone the same nominal stress. The S-N curve is a plot of stress amplitude, stress range or the maximum cyclic stresses, \( S \) (selected as the controlled or independent variable), versus the number of cycles to failure, \( N \) (the dependent variable). Usually there are two methods of plotting S-N curves: (1) the actual stress, \( S \), versus the logarithmic scales of cycles, \( N \), and (2) both \( S \) and \( N \) is plotted in the form of a log-log plot of \( S \) versus \( N \).

Several types of machines and specimens are available for developing fatigue data to create S-N curves. To ensure the consistency of test data, there are standards to be used for applying loads and preparing specimens.

Figure 2.12 is the typical S-N curves for H13 tool steel [45] under room temperature with different surface finish.
From the plot in Figure 2.12, it can be seen that when applied stress is less than a certain value, the S-N curve becomes asymptotic to a horizontal line. The stress associated with this limit is called endurance limit or fatigue limit. This limit is an important parameter when designing a part with infinite life.

For most materials, the fatigue limit is not a constant and varies with all kinds of different parameters. In general, some factors influencing the shape of S-N curve are:

- Material Conditions (Heat Treatment and Cold Working)
- Types of load (Tension, Compression, Torsion and Combined Load)
- Stress Ratio
- Rate of Load Application
• Environment (Corrosive, Inert and Temperature)
• Surface Conditions \( (R_a) \)

In die casting manufacturing process, the H13 service temperature can be as high as 1250°F. Due to the reason that it is extremely difficult and costly to test fatigue data at this higher elevated temperature, there is no S-N curve data available at the mentioned temperature range. It is obvious that applying the S-N curve obtained at room temperature, such as the curve shown in Figure 2.12 for die casting tool thermal fatigue prediction, is not a wise choice.

2.4.2 Constant-Life Diagrams

Constant-life fatigue diagrams are a family of curves generated for different fatigue life, each representing the variation of either \( \sigma_{\text{max}} \) versus \( \sigma_{\text{min}} \), \( \sigma_{\text{max}} \) versus \( \sigma_{\text{m}} \), or \( \sigma_{\text{a}} \) versus \( \sigma_{\text{m}} \). These diagrams widen the application of most available fatigue data, which are obtained through testing by applying the fully reversed loading case \( (R = -1) \), where mean stress, \( \sigma_{\text{m}} \), is zero. The most common type of constant life fatigue diagram is the plot of \( \sigma_{\text{a}} \) versus \( \sigma_{\text{m}} \). Goodman [46] presented this diagram as a straight-line relationship shown by the equation below, so it also called a Goodman Diagram.

\[
\left( \frac{\sigma_{\text{a}}}{\sigma_{\text{a0}}} \right) + \left( \frac{\sigma_{\text{m}}}{\sigma_{\text{uts}}} \right) = 1 \tag{2.5}
\]

Constant life diagrams can be generated either by an empirical relationship or by actual test data. In equation (2.5), the alternating stress \( \sigma_{\text{a0}} \) represents the fully reversed
case \((R = -1)\) where \(\sigma_m = 0\). From a series of fatigue test data developed for S-N curves, the variation of \(\sigma_r\) versus \(\sigma_m\) with various combination of stress ratio, \(R\), was plotted and the general trend is illustrated in Figure 2.13.

![Figure 2.13. Constant Life Diagram [44]](image)

The data in Figure 2.13 were collected for a fixed number of cycles to failure. When stress amplitude, \(\sigma_m\), approaches zero, the data points tend to approach the ultimate tensile strength \((\sigma_{ult})\) of the material. All other stress amplitudes along the dotted line have nonzero mean stress and they are referred to as \(\sigma_r\).
Constant life diagrams come from a series of fatigue test data under room temperature. For die casting manufacturing process, H13 steel serves at elevated temperature. Since there is no fatigue test data available at this elevated temperature range for H13, and there is no application of this method for die casting tool fatigue life prediction.

2.4.3 Fracture-Mechanics Method

The application of fracture mechanics for life prediction is solely a crack-propagation life method. Life calculations are carried out from a specific initial crack size (which may be a pre-existing crack, a flaw, an intrinsic defect characteristic of material, etc.) to a final crack size at failure, determined from the material’s inherent toughness. This idea of toughness, or resistance to crack growth, is one of the basic concepts of fracture mechanics.

There are two aspects for this method: stress field around the advancing crack tip and steady-state growth rate of the advancing crack. The stress field around the crack tip is given by the stress intensity factor, \(K\).

The basic relation for the stress intensity factor is:

\[ K = F \sigma \sqrt{\pi a} \]  \hspace{1cm} (2.6)

In fatigue life calculations, it shown in a different form:

\[ \Delta K = F \Delta \sigma \sqrt{\pi a} \]  \hspace{1cm} (2.7)
where, $F$ is a factor related to the geometry of the crack and component, $\Delta \sigma$ is stress range, $\Delta K$ is stress intensity factor range and "$a$" is initial crack length. The above relations only apply for elastic deformation.

In the equations above, the stress intensity factor $K$ is only a function of load and geometry. Its limit value $K_c$, referred to as the fracture toughness of the material, is a material property, independent of loading condition. The crack length at failure can be written as:

$$a_c = \frac{1}{\pi} \left( \frac{K_c}{F \sigma_{max}} \right)$$  \hspace{1cm} (2.8)

The steady-state growth rate of the advancing crack is characterized by the empirical equation

$$\frac{da}{dN} = C(F \Delta \sigma \sqrt{\pi a})^m$$  \hspace{1cm} (2.9)

where, $C$ and $m$ are constants. Equation (2.9) is the basic form of the crack growth rate equation. It indicates that crack growth rate depends on the amplitude of the cyclic stress range and initial crack length. Both factors are related to the stress intensity factor, $K$. Thus, the crack growth rate can also be related to the stress intensity factor. The equation can also be written in the form of:

$$\frac{da}{dN} = C(\Delta K)^m$$  \hspace{1cm} (2.10)
In general, the stress ratio, $R$, has influence on the crack growth rate and the above equation can be written as:

$$
\frac{da}{dN} = C(\Delta K)^m = \frac{C_1}{(1 - R)^{m(1 - \gamma)}} \Delta K^m = f(\Delta K, R) \quad (2.11)
$$

and

$$
\frac{dN}{da} = \frac{1}{f(\Delta K, R)} \quad (2.12)
$$

For life prediction analysis, we simply integrate the equation, and we have

$$
N_f = \int_{a_0}^{a_f} \frac{dN}{da} = \int_{a_0}^{a_f} \frac{da}{f(\Delta K, R)} \quad (2.13)
$$

Combining equations (2.12) and (2.13), we have:

$$
N_f = \int_{a_0}^{a_f} \left( \frac{dN}{da} \right) da \quad (2.14)
$$

As mentioned above, the stress intensity factor, $K$, is based on the assumption of linear elastic fracture mechanics. For high-ductility materials and where the plastic strain zone ahead of the crack tip is large, the prediction based on $\Delta K$ can result in considerable error.

A standard test method exists for determining fatigue crack growth rate [47]. With the test data and initial crack and critical crack to failure, the fatigue life can be obtained.
Park [5] applied this method for his shot sleeve fatigue life prediction. The equation used for his calculation is:

\[
\frac{da}{dN} = 0.66 \times 10^{-8}(\Delta K)^{2.25}
\]  

(2.15)

and

\[
\Delta K = \frac{1.12}{\sqrt{Q}} \Delta \sigma \sqrt{\pi a}
\]

(2.16)

where \( Q \) is a geometry factor. With this method, the crack growth is a function of cyclic stress, crack size and geometry.

In Park’s study, the threshold stress intensity factor was assumed roughly between 5 and 15 (ksi\(\text{inch}^{0.5} \) for tool steel. With this value, fatigue crack growth is immeasurably slow.

With the assumption of 353 MPa cyclic hoop stress (which is induced by an internal pressure of 150 MPa in shot sleeve), and an initial size of a pre-existing 1 mm crack, the number of cycles to grow a crack to critical size (3.21 mm) were calculated for both martensitic steel and nitrided martensitic steel. The calculated results were 11,300 cycles and 32,400 cycles respectively for these two materials. Also, from this study, the average crack growth rate were obtained for martensitic steel and nitrided martensitic steel from 0.5 to 3.21 crack are \( 1.45 \times 10^{-4} \) (mm/cycle) and \( 5.26 \times 10^{-5} \) (mm/cycle), respectively.
2.4.4 Strain-Life Prediction Method (Universal Slope Method)

In dealing with high-cycle fatigue, the true fatigue strength, \( \sigma_t \), in terms of half cycle reversal to failure \( (2N_f) \), can be expressed as:

\[
\sigma_t = \sigma_j (2N_f)^b
\]  

(2.17)

where \( \sigma_j \) is the fatigue strength coefficient, and \( b \) is a constant.

In the elastic range, the quantity \( \sigma_t \) can be replaced by the elastic strain amplitude, \( \varepsilon_e \), as:

\[
\varepsilon_e = \sigma_j / E (2N_f)^b
\]  

(2.18)

where, \( E \) is Young’s Modulus.

The plastic strain amplitude, \( \varepsilon_p \), is also related to the number of half-cycle reversals to failure, \( 2N_f \) by a simple power law equation:

\[
\varepsilon_p = \varepsilon_j (2N_f)^c
\]  

(2.19)

where, \( \varepsilon_j \) is called fatigue ductility coefficient and its value equal to the true monotonic fracture strain, \( \varepsilon_f \), and \( c \) is constant. Equation (2.19) is called the Coffin and Manson law [48].

Combining the elastic and plastic components of the total strain-life amplitude described above, the total strain-life curve can be expressed as [49]:

\[
\varepsilon = \sigma_j / E (2N_f)^b + \varepsilon_j (2N_f)^c
\]  

(2.20)

51
When the cyclic material data are unavailable, Manson related the cyclic total strain range $\Delta \varepsilon$ and number of cycles to failure $N_f$ with the following equation:

$$
\Delta \varepsilon = \Delta \varepsilon^{pl} + \Delta \varepsilon^{el} 
$$

(2.21)

and plastic strain range,

$$
\Delta \varepsilon^{pl} = M (N_f)^z
$$

(2.22)

elastic strain range,

$$
\Delta \varepsilon^{el} = \frac{G}{E} (N_f)^\gamma
$$

(2.23)

and total strain range,

$$
\Delta \varepsilon = M (N_f)^z + \frac{G}{E} (N_f)^\gamma
$$

(2.24)

where $M$, $G$, $z$ and $\gamma$ are material properties.

Upon evaluation of several materials fatigue behavior, Manson [50] was able to relate to the material properties constant to:

$$
M = \varepsilon^{0.6}
$$

(2.25)

$$
G = 3.5\sigma_{uts}
$$

(2.26)

$$
Z = -0.6
$$

(2.27)

$$
\gamma = -0.12
$$

(2.28)

and equation (2.24) can be written as,

$$
\Delta \varepsilon = D^{0.6}(N_f)^{-0.6} + \frac{3.5\sigma_{uts}}{E}(N_f)^{-0.12}
$$

(2.29)
Equation (2.29) is plotted in log-log form and is shown schematically in Figure 2.14. It is clear from this figure that the total strain range is the combination of elastic and plastic strain.

![Figure 2.14. Illustration of the total strain-life curve [36]](image)

Since the material independent components $z$ and $r$ are the slopes of curves of the relations in log-log scale, the method is called universal slope method [51] [52]. The plastic strain range best approximates low cycle fatigue while elastic strain range is related to high cycle fatigue.

Several applications of this method for fatigue life prediction have been used with steel at elevated temperature. In Lee's study [53], this method was applied for thermal fatigue prediction of a steam exhaust silencer. With the same equation used in (2.29), significant low cycle fatigue failure at elevated temperature was found in the structure.
Carter [54] successfully applied the same method for failure analysis and life prediction of a large, complex fin plate heat exchanger used at elevated temperature.

For non-uniform or differential thermal expansion, cyclic strain is more important than cyclic stress [55]. In a die casting process, both shot sleeve and die are mainly subjected to cyclic thermal loading conditions, which are non-uniform. The Universal Slope Method should be a suitable method for thermal fatigue life prediction of die casting process. As mentioned early in this chapter, Rosbrook [22] created a finite element model to analyze Wallace’s thermal fatigue specimen [24]. In his study, the simulated temperature distribution and cyclic history were compared with Wallace’s temperature measurement as the fundamental. Then, the computer simulated stress/strain range within a two-dimensional plane-strain cross-section H13 test sample for the test specimen was recorded. Through Rosbrook’s investigation, the universal slope method was used as means for thermal fatigue life prediction. The results show that the cyclic life predicted from the calculated strain range is very close to that observed by Wallace.

The same method also was used by Sirinterlikci [36] for die casting die thermal fatigue life prediction. Due to relative low calculated strain range, this method was unsuccessful for thermal fatigue life prediction in this study.

In both of Rosbrook and Sirinterlikci’s computer model, the cyclic stress were lower than the yield stress of H13 at service temperature.
CHAPTER 3

RESEARCH METHOD

3.1 2-D COMPUTER MODEL DEVELOPMENT

3.1.1 Plane Strain Model

The initial goal of this study was to focus on the radial distortion of shot sleeves due to the potential plunger tip sticking problem. In order to develop a realistic and basic model for the simulation of shot sleeve radial distortion, a plane strain model was used for each cross section as shown in Figure 3.1 for a two-dimensional simulation study.
The models were developed using ABAQUS [56] finite element software, which is capable of conducting multiple cycle simulations of a die casting process for temperature, displacement, stress and strain distribution.

For linear elastic materials, the stress-strain relations come from the generalized Hooke’s law. For isotropic materials, Hooke’s law gives:

\[ \varepsilon_x = \frac{\sigma_x}{E} - \nu \frac{\sigma_y}{E} - \nu \frac{\sigma_z}{E} \]  (3.1)

\[ \varepsilon_y = -\nu \frac{\sigma_y}{E} + \frac{\sigma_x}{E} - \nu \frac{\sigma_z}{E} \]  (3.2)

\[ \varepsilon_z = -\nu \frac{\sigma_z}{E} - \nu \frac{\sigma_y}{E} + \frac{\sigma_x}{E} \]  (3.3)

\[ \gamma_{xy} = \frac{\tau_{xy}}{G} \]  (3.4)
In some special cases, the 3D equations can be simplified with plane stress and plane strain assumptions.

In the plane stress situation, a thin planar body subjected to in-plane loading on its edge surface is said to be in plane stress. A ring press fitted on a shaft is an example as shown in Figure 3.2 (a). Here stresses $\sigma_x, \tau_{xy}$ and $\tau_{xz}$ are set as zero. The equations from (3.1) to (3.6) reduced to:

$$\gamma_{xz} = \frac{\tau_{xy}}{G} \quad (3.5)$$

$$\gamma_{yy} = \frac{\tau_{xy}}{G} \quad (3.6)$$

On the other hand, if a long body of uniform cross section is subjected to transverse loading along its length, a small thickness in the loaded area can be treated as subjected to plane strain as shown in Figure 3.2 (b). Here $\varepsilon_z, \gamma_{xz}, \gamma_{yz}$ are taken as zero. Stress $\sigma_z$
may not be zero in this case. The stress-strain relations can be described in Equation (3.11):

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \frac{E}{(1+\nu)(1-2\nu)} \begin{bmatrix} 1-\nu & \nu & 0 \\ \nu & 1-\nu & 0 \\ 0 & 0 & \frac{1}{2}-\nu \end{bmatrix} \begin{bmatrix} \sigma_x \\ \sigma_y \\ \gamma_{xy} \end{bmatrix}$$

Figure 3.2. Diagrams of (a) Plane stress and (b) Plane strain
Usually the plane strain approach is used when the body is very thick relative to its lateral dimensions. For shot sleeves, the dimension along the axial direction is much longer than the dimension along radial directions. With the assumption of no dimensional change along the axial direction of the shot sleeve, each cross section of shot sleeve near the central regions can be simplified as 2-D, then the plane strain model can be used.

In the simulation of the cross-sectional two-dimension model, because of the symmetry, only half of the geometry in Figure 3.1 has been modeled. Because the stress and deformation of shot sleeves are based on temperature distribution and constraints, proper selection of the thermal boundary conditions is essential. The thermal boundary conditions involved in the model are symmetry boundaries, convection heat transfer boundaries, contact heat transfer, and radiation heat transfer boundaries. Convection heat transfer occurs between the shot sleeve outside surface and air, shot sleeve non-contact surface and air, and molten metal upper surface and air. Contact heat transfer occurs between the shot sleeve and molten metal in the shot sleeve. Radiation heat transfer occurs between the upper surface of the molten metal and the shot sleeve inside non-contact surface, and also between the shot sleeve outside surface and its surroundings. There is no heat transfer normal to the symmetric boundaries. For solving the thermal problem, DC2D4 element is selected, which is a 4-node linear quadrilateral element for heat transfer analysis. Based on thermal results, stress and deformation analysis is conducted by using CPE4 element, which is a 4-node bilinear plane strain quadrilateral
element. For the stress-strain analysis, the x direction movement of the symmetry line and y direction movement of the horizontal symmetry line are constrained.

The shot sleeve geometry chosen for the simulation had an inside diameter of 1.9 inches and outside diameter of 3.00 inches. These are the same dimensions of the sleeve used by Paliani [38] when he collected his experimental shot sleeve temperature and distortion data.

3.1.2 Simulation Cycle

In the computer models, the total simulation cycle can be defined corresponding to the die casting process, which includes the filling period and empty period. The filling period is the time in which molten metal is poured into shot sleeve, molten metal remains inside the shot sleeve, and the metal is pushed out of shot sleeve after solidification. The empty period is the time the sleeve has no metal in it. For the model development, the cycle time is also based on Paliani’s experiment [38].

3.1.3 Three Models for Simulation

(A). Model-1: Calculation of Molten Metal Temperature Change

In this model, all the boundaries are treated as the real situation and only radiation between molten metal surface and shot sleeve is neglected. During the filling period, convection heat transfer is applied to the shot sleeve outside surface, shot sleeve inside
non-contact surface, and molten metal upper surface. The heat transfer between the shot sleeve and molten metal interface is very complex during the filling process. This is because of the intensive convection heat transfer when pouring molten metal into the shot sleeve, the pushing of molten metal out of the shot sleeve with the plunger tip, and the heat conduction between the interface when molten metal resides in the shot sleeve. During the pouring and pushing process, the effect of convection heat transfer can be expressed as [57]:

\[ q_{\text{c}} = h_{\text{c}} (T_s - T_m) \]  

(3.12)

where, \( T_s \) and \( T_m \) are the temperatures of shot sleeve inside surface and molten metal surface, \( h_{\text{c}} \) is the convection heat transfer coefficient, which changes with the process condition. At the pouring or pushing stage, it is very difficult to know the transient heat transfer coefficient \( h_{\text{c}} \).

When the molten metal is inside the shot sleeve, the heat transfer associated with the molten metal and shot sleeve contact interface is usually treated empirically with a gap heat transfer coefficient \( h_{\text{g}} \) [58], which is defined by the heat flux across the gap. The total heat flux during filling period across the gap can be defined as:

\[ q_{\text{g}} = -k \frac{\partial T}{\partial r} = h_{\text{g}} (T_m - T_s) \]  

(3.13)

Actually \( h_{\text{g}} \) is an equivalent heat transfer coefficient during molten metal inside the shot sleeve, which includes pouring and pushing period. This coefficient can be determined based on experimental results.
Transient heat conduction within the shot sleeve and molten metal is expressed in cylindrical coordinates as follows:

\[
\frac{\partial}{\partial r} \left( k \frac{\partial T}{\partial r} \right) + \frac{k}{r} \left( \frac{\partial T}{\partial r} \right) + q_i = \rho C_p \frac{\partial T}{\partial t}
\]  

(3.14)

where, \( \rho \) is the density of material, \( C_p \) is the specific heat of material, \( k \) is conductivity of material, \( T \) is the temperature of the shot sleeve or molten metal and \( q_i \) is the latent heat of fusion. Based on this equation, the temperature distribution can be calculated.

During the empty period, the molten metal is removed and only heat convection exists at the shot sleeve surfaces. For all the three models, during the empty period, the boundary conditions are applied in the same way.

(B). Model-II: Simplified Shot Sleeve-Molten Metal Interface Boundary Condition

The difference between Model-I and Model-II is the simplification of heat transfer between shot sleeve and molten interface. In Model-II, the molten metal is simply removed and the molten metal is treated as a bulk volume film with constant temperature. The heat transfer between the interface is treated as an equivalent convection heat transfer problem. This method has been used in some metal forming process before [59] [60]. The heat transfer can be expressed as:

\[
q_0 = -k \frac{\partial T}{\partial r} = h_{eq} \left( T_s - T_{mean} \right)
\]

(3.15)

where, \( h_{eq} \) is the equivalent heat transfer coefficient and \( T_{mean} \) is the mean temperature of contact environment, which is the bulk temperature of molten metal.
(C) Model-III: Consideration of Radiation Heat Transfer in the Process

Model-III is developed based on the Model-II with bulk volume convection heat transfer assumption between shot sleeve and molten interface, with the consideration of radiation effect between molten metal surface and shot sleeve upper portion. The radiation effect between shot sleeve outside surface and environment is neglected because of the relatively low shot sleeve outside surface temperature.

A simplified approach was used in the simulations when the convection and radiation components of heat transfer were combined together in the shot sleeve upper surface to obtain a single heat transfer coefficient [61].

The total heat transfer from any surface can be expressed as:

\[ q_{\text{Total}} = q_{\text{convection}} + q_{\text{radiation}} \]  \hspace{1cm} (3.16)

\[ q_{\text{convection}} = h \left( T_s - T_\infty \right) \]  \hspace{1cm} (3.17)

\[ q_{\text{radiation}} = \varepsilon \sigma \left( T_s^4 - T_{\text{mean}}^4 \right) \]  \hspace{1cm} (3.18)

\[ q_{\text{Total}} = h \left( T_s - T_\infty \right) + \varepsilon \sigma \left( T_s^4 - T_m^4 \right) = h_{\text{eq}} \left( T_s - T_{\text{mean}} \right) \]  \hspace{1cm} (3.19)

where, \( h \) is the convection heat transfer coefficient between shot sleeve inside upper surface and molten metal, \( T_s \) is shot sleeve inside upper surface temperature, \( T_\infty \) is bulk volume molten metal temperature, \( \varepsilon \) is the emissivity of materials, and \( \sigma \) is Stefan-Boltzmann constant. Using the equivalent heat transfer coefficient \( h_{\text{eq}} \), the whole analysis of the radiation problem can be treated as a simplified bulk volume convection
heat transfer problem. This method can increase the accuracy of the simulation without increasing the computation time.

3.1.4 Structure of the Models

The geometry of a half shot sleeve was created and meshed by using I-DEAS software, then the geometry model was transferred into the ABAQUS input file. The boundary conditions can be applied either by using I-DEAS function or added into ABAQUS input file.

Figure 3.3 shows the mesh used in Model-I filling period simulation. When the inside mesh was taken off, the rest of the meshes were used for empty period simulation of Model-I, or for the whole simulation mesh of Model-II and Model-III. Between the shot sleeve and molten metal interface, very small mesh existed for contact gap elements for Model-I simulation.
3.1.5 Experiment and Simulation Cycle Time

For the model development, all the experiments and simulations were with 50% filling percentage of molten aluminum in the shot sleeve. The set up was divided into two groups; experimental set up for obtaining suitable heat transfer coefficients, and die casting machine operation set up for checking the results during shot-to-shot cycling.

The first set up consisted of 5 cycles to determine heat transfer coefficients and measure dimension changes along vertical and horizontal directions. The set up is shown in Table 3.1.
<table>
<thead>
<tr>
<th></th>
<th>Filling Time (sec)</th>
<th>Empty Time (sec)</th>
<th>Melting Temperature (°F)</th>
<th>Shot Sleeve Initial Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>55</td>
<td>42</td>
<td>1290</td>
<td>85</td>
</tr>
<tr>
<td>2</td>
<td>50</td>
<td>48</td>
<td>1295</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>49</td>
<td>46</td>
<td>1277</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>38</td>
<td>152</td>
<td>1285</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>35</td>
<td>50</td>
<td>1271</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 3.1. Experiment set up and simulation cycle

The second set up consisted of 20 cycles to simulate the operation of a die casting machine. In the operation set up, the filling time was 5 seconds (the first two cycles had longer times for heating the shot sleeve, 24 and 14 seconds, respectively). For the empty period, the first two cycles used exactly the same time as the experiment set up (39 and 45 seconds, respectively); the remaining 18 cycles used an average empty time of 43 seconds. The initial temperature of the shot sleeve is 123 °F. This procedure was used to check the selected heat transfer coefficients for all the three models.

3.1.6 Results Comparison

(A). Experiment Set-up Results

Based on the experiment set up, simulations were conducted for three models. The thermocouple measurement location and correspond simulation node was the node-I
position as shown in Figure 3.4. The experimental and simulated temperature results for this nodal point at the second cycle are shown in Figure 3.5. From this comparison, it can be seen that the tendency of experiment result and all of the three simulated temperature curves are the same and the differences among the results are limited.

Figure 3.4. Nodal points for temperature measurement & simulation, radial line location for displacement
Figure 3.5. Node-I location measured and simulated results at shot 2

The comparison of simulated displacements from side to side and top to bottom with experiment measured data for 5 cycles are given in Table 3.2. It can be seen that the deformation increases with the cycle time in both directions, and the tendencies of simulated results are well matched with the experiment measurements.

<table>
<thead>
<tr>
<th>Shot No.</th>
<th>Experiment Results (side-side/top-bottom)</th>
<th>Simulation results (unit: inch) (side-side/top-bottom)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Model (I)</td>
</tr>
<tr>
<td>1</td>
<td>0.008/0.006</td>
<td>0.008/0.008</td>
</tr>
<tr>
<td>2</td>
<td>0.012/0.007</td>
<td>0.010/0.012</td>
</tr>
<tr>
<td>3</td>
<td>0.015/0.008</td>
<td>0.012/0.015</td>
</tr>
<tr>
<td>4</td>
<td>0.015/0.011</td>
<td>0.013/0.015</td>
</tr>
<tr>
<td>5</td>
<td>0.017/0.013</td>
<td>0.015/0.016</td>
</tr>
</tbody>
</table>

Table 3.2. Experiment and simulation displacement results
After running all the three models and comparing with experimental data, the heat transfer coefficients for the three models were determined as shown in Table 3.3.

<table>
<thead>
<tr>
<th>Heat Transfer Coefficient (W/m²°F)</th>
<th>Model (I)</th>
<th>Model (II)</th>
<th>Model (III)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>766.92</td>
<td>1504.34</td>
<td>1415.85</td>
</tr>
</tbody>
</table>

Table 3.3. Heat transfer coefficient between shot sleeve and molten metal

(B) Die Casting Operation Set-up Results

Based on the operation process set up, simulations were conducted with the heat transfer coefficients listed in Table 3.3 for the three models. For the comparison between the results from experiment and verifying simulations, some thermocouple location points and the corresponding simulation nodes are shown in Figure 3.4.

The temperature measurements were recorded for shot 12 and compared with the simulation results for the same shot. The maximum difference between experiment and simulation results for the positions at node-II, node-III, node-IV, node-V, node-VI and node-VII of the three models are 10%, 20%, 9%, 18%, 7% and 10%, respectively. The errors are due to the accuracy of heat transfer coefficients and measurement procedure.

Figure 3.6 shows the temperature history for the 20 cycles of experiment and simulations results at node-V. For the first 3 cycles, the temperature difference between
the experiment and simulation is large. After 3 cycles, the difference decreased. Comparing simulation curves, it can be seen that with the increase of cycle number, the temperature differences between simulation and experiment for all three simulation data decreases. Model-III has a better result compared to the other two models. This is due to the consideration of radiation heat transfer in the process, which better reflects the real situation.

Figure 3.6. Node-VII location measured and simulated temperature results of operation set up in 20 cycles
3.1.7 Selection of Models

From the above comparison, it can be seen that Model-I best reflects the real process situation if radiation is included in the model. But the model has more nodal points and elements involved, and requires longer calculation time. Model-II simplified the boundary condition between the shot sleeve and molten metal, but results showed a lower temperature distribution in the upper portion of the shot sleeve. With the addition of the consideration of radiation between the molten metal surface and upper shot sleeve, Model-III requires the same number of nodal points and elements as in Model-II, but takes less time for the calculation. Table 3.4 is the summary of three models calculation time with an IBM RISC-6000 workstation for the two dimensional model with 20 cycles. The difference of computation times between Model-II and Model-III is due to the computer itself. It is clear that Model-III is a better one, which reflects the real shot sleeve service situation.

<table>
<thead>
<tr>
<th>Model</th>
<th>Total CPU Time (second)</th>
<th>Wall Clock Time (second)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model (I)</td>
<td>2209.7</td>
<td>2464</td>
</tr>
<tr>
<td>Model (II)</td>
<td>844.55</td>
<td>1144</td>
</tr>
<tr>
<td>Model (III)</td>
<td>721.75</td>
<td>1001</td>
</tr>
</tbody>
</table>

Table 3.4. Computer time for 20 cycles

Due to the advantage of Model III, it will be used in later 2-D plane strain model study.
3.2  3-DIMENSIONAL MODEL

The 2-D plane strain model can be used to study shot sleeve radial distortion at the central regions or near the biscuit end region without runner cutting through the biscuit end. For a commercial shot sleeve, at the biscuit end, due to the effects of runner geometry and die block/shot sleeve contact, as well as the biscuit end axial heat transfer between shot sleeve end and die block, the plane strain model was no longer valid for the detailed study. A 3-D computer model is necessary for the systematically study of the distortion and stress/strain at the biscuit end.

3.2.1 Thin/Thick Wall Shot Sleeves

The 3-D study started with two large commercial die casting company H13 steel shot sleeves: (1) a thin wall shot sleeve with an inside diameter of 6.693 inches and outside diameter of 9.974 inches, and (2) a thick wall shot sleeve with an inside diameter 6.693” and outside diameter 11.974” for making aluminum automotive components. Figure 3.7 and Figure 3.8 shows the dimensions of thin wall shot sleeve and thick wall shot sleeve, respectively.
Figure 3.7. Thin wall shot sleeve dimensions

Figure 3.8. Thick wall shot sleeve dimensions
There are also two different runners in the application. Their geometry with Design A and Design B shapes for thick wall sleeves are shown in Figure 3.9 and Figure 3.10. The runner corner and end nose radiuses are also specified in the two figures.

Figure 3.9. Design A runner shape
A systematic study of the effect of design parameters on shot sleeve performance, the effects of two different runner designs with different runner round corner radius ($R = 0.25\"$ and $R = 0.5\"$) and end nose radius ($R = 0\", R = 0.25\"$ and $R = 0.50\"$) are described in the following chapters.
3.2.2 Industrial Process Time Data and Simulation Steps

For this study, the industrial process time line (Machine Cycle) was used as reference for the simulation set-up. The process data is listed in Table 3.5.

<table>
<thead>
<tr>
<th>Process</th>
<th>Time (sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pouring molten metal &amp; shot delay</td>
<td>10.5</td>
</tr>
<tr>
<td>Slow shot</td>
<td>1.3</td>
</tr>
<tr>
<td>Dwell</td>
<td>30.0</td>
</tr>
<tr>
<td>Die open &amp; core out</td>
<td>7.5</td>
</tr>
<tr>
<td>Casting ejection</td>
<td>9.5</td>
</tr>
<tr>
<td>Reciprocate lubricant in</td>
<td>2.3</td>
</tr>
<tr>
<td>Reciprocate water spray</td>
<td>6.0</td>
</tr>
<tr>
<td>Reciprocate lubricant spray</td>
<td>6.0</td>
</tr>
<tr>
<td>Air blow-off</td>
<td>13.0</td>
</tr>
<tr>
<td>Die close</td>
<td>4.0</td>
</tr>
<tr>
<td>Total</td>
<td>90.1</td>
</tr>
</tbody>
</table>

Table 3.5. Industrial process time line (machine cycle)

The industrial process data in Table 3.5 is the base for conducting computer simulation in the following 2-D plane strain and 3-D biscuit end modeling.

According to the process time line, the 3-D biscuit end simulation was conducted in 5 steps. Step 1, 31% of the shot sleeve lower portion was filled with molten metal. Step 2, the biscuit end was filled with molten metal and intensification pressure was applied; and at this step, solidification occurred. Step 3, the die was opened and the casting was ejected. In step 4, spray was applied at the biscuit end area. In step 5, the shot sleeve
was emptied. In all the steps, the heat transfer coefficients in different steps were chosen based on the 2-D plane strain verified value and as well as the data published about die and molten metal interface heat transfer [22] [62].

3.2.3 Model Setup

A shot sleeve 3-D model geometry can be created with any available CAD software based on the dimension, such as, Pro/Engineer, or IDEAS. Due to the symmetry, only half of the geometry was modeled.

The CAD geometry was imported to IDEAS for meshing; then was exported to MSC/MARC [63] format for applying all the boundary conditions and material properties. At the same step, the process cycle information was applied for simulation. In the MSC/MARC 3-D model, fully integrated 4 nodes tetrahedral element was used to construct the 3-D model. Due to the limitation of computation time and space available, only 3” thick section sleeve at biscuit end was modeled. Figure 3.11 and Figure 3.12 is the modeled section sleeve and corresponding boundaries.
Figure 3.11. End surface view modeled section shot sleeve

Figure 3.12. Cutting surface view modeled section shot sleeve
3.2.4 Boundary Conditions in the Simulation

Since the shot sleeve was constrained from moving along axial direction, both the end surface and the cutting surface were fixed in axial direction to reflect the actual situation. The bottom line at outside diameter was also constrained in vertical direction to simulate the unconstrained condition, or the outside surface of shot sleeve was rigidly constrained to model the fully constrained condition. The symmetric constrained condition was also applied on the symmetry surface.

In the process simulation, the heat transfer coefficients for different stages are very different:

- At step 1 of sleeve filling process, the experimentally determined 1415 W/m² °K heat transfer coefficient between molten metal and shot sleeve was applied. Since only 31% percent of the bottom portion of sleeve were filled, the heat exchange only applied for this portion of contact region.

- For the intensification and solidification process, the molten filled the whole biscuit end region, including the runner area. The applied heat transfer coefficient is 12,970 W/m² °K for this step.

- For the lubrication spray process, heat transfer coefficient is 4000 W/m² °K. The spray was applied to bottom and top portion, as well as the runner surface.

- At both ends (Cutting surface and End surface), contact heat transfer coefficient was applied, which is 796 W/m² °K.
• At outside and no contact surfaces, the heat transfer coefficient between shot sleeve and air is $23.5 \text{ W/m}^2 \text{ °K}$ for the whole process.

3.2.5 Material Properties Used in Simulation

Based on the material properties available from industrial and previous research, the material properties used in the computer model are shown below. Note that their properties are temperature dependent, and can reflect the real operation situation of H13.

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Young’s Modulus E (MPa) $\times 10^5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>2.14</td>
</tr>
<tr>
<td>200</td>
<td>2.07</td>
</tr>
<tr>
<td>400</td>
<td>2.00</td>
</tr>
<tr>
<td>600</td>
<td>1.86</td>
</tr>
<tr>
<td>800</td>
<td>1.79</td>
</tr>
<tr>
<td>1000</td>
<td>1.52</td>
</tr>
<tr>
<td>1200</td>
<td>1.10</td>
</tr>
</tbody>
</table>

Table 3.6. Temperature dependent Young’s Modulus
Table 3.7. Temperature dependent thermal expansion

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal Expansion (×10⁻⁶)</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>6.6</td>
</tr>
<tr>
<td>400</td>
<td>6.8</td>
</tr>
<tr>
<td>600</td>
<td>6.8</td>
</tr>
<tr>
<td>800</td>
<td>7.2</td>
</tr>
<tr>
<td>1000</td>
<td>7.4</td>
</tr>
<tr>
<td>1200</td>
<td>7.8</td>
</tr>
</tbody>
</table>

Table 3.8. Temperature dependent thermal conductivity

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Thermal conductivity (W/m °F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>122</td>
<td>13.75</td>
</tr>
<tr>
<td>212</td>
<td>13.85</td>
</tr>
<tr>
<td>302</td>
<td>13.95</td>
</tr>
<tr>
<td>392</td>
<td>14.05</td>
</tr>
<tr>
<td>482</td>
<td>14.15</td>
</tr>
<tr>
<td>572</td>
<td>14.25</td>
</tr>
<tr>
<td>662</td>
<td>14.35</td>
</tr>
<tr>
<td>752</td>
<td>14.45</td>
</tr>
<tr>
<td>842</td>
<td>14.55</td>
</tr>
<tr>
<td>932</td>
<td>14.65</td>
</tr>
<tr>
<td>1022</td>
<td>14.75</td>
</tr>
<tr>
<td>1112</td>
<td>14.85</td>
</tr>
<tr>
<td>1202</td>
<td>14.95</td>
</tr>
<tr>
<td>1292</td>
<td>15.05</td>
</tr>
<tr>
<td>1382</td>
<td>15.15</td>
</tr>
<tr>
<td>1472</td>
<td>15.25</td>
</tr>
<tr>
<td>1562</td>
<td>15.35</td>
</tr>
<tr>
<td>1652</td>
<td>15.45</td>
</tr>
<tr>
<td>Temperature (°F)</td>
<td>Specific Heat (J/kg °F)</td>
</tr>
<tr>
<td>-----------------</td>
<td>------------------------</td>
</tr>
<tr>
<td>122</td>
<td>261.1</td>
</tr>
<tr>
<td>212</td>
<td>266.3</td>
</tr>
<tr>
<td>302</td>
<td>271.6</td>
</tr>
<tr>
<td>392</td>
<td>276.8</td>
</tr>
<tr>
<td>482</td>
<td>282.0</td>
</tr>
<tr>
<td>572</td>
<td>287.2</td>
</tr>
<tr>
<td>662</td>
<td>292.4</td>
</tr>
<tr>
<td>752</td>
<td>297.7</td>
</tr>
<tr>
<td>842</td>
<td>302.9</td>
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<tr>
<td>932</td>
<td>308.2</td>
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<td>1022</td>
<td>313.4</td>
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<tr>
<td>1112</td>
<td>318.6</td>
</tr>
<tr>
<td>1202</td>
<td>323.9</td>
</tr>
<tr>
<td>1292</td>
<td>329.1</td>
</tr>
<tr>
<td>1382</td>
<td>334.3</td>
</tr>
<tr>
<td>1472</td>
<td>339.6</td>
</tr>
<tr>
<td>1562</td>
<td>344.8</td>
</tr>
<tr>
<td>1652</td>
<td>350.1</td>
</tr>
</tbody>
</table>

Table 3.9. Temperature dependent specific heat
3.3 SENSITIVITY STUDIES

Theoretically, the smaller the mesh size, the more accurate the final result will be. The higher order of integration method used for the type of element, the better result the will be for the same mesh size. Due to the constraints of computer capability, there are some constraints that should be considered at the beginning of the simulation work, which are the factors of element type and mesh size. Some sensitivity studies should be conducted to analyze the influence of these factors on final results to ensure efficient use of computer resources to obtain as much accuracy of results within the required time period.

3.3.1 Element Type Sensitivity

In the MARC software element library, there are a variety of elements that can be used for the analysis in this research. Three types of elements were studied for the element influence on final results: 4 nodes tetrahedral element, 8 nodes hex element and 10 nodes tetrahedral element [63].

The 4 nodes tetrahedral element is a linear isoparametric three-dimensional tetrahedron. As this element uses linear interpolation functions, the strains are constant throughout the element. This results in a poor representation of shear behavior. For thermal analysis, the thermal gradients are constant throughout the element. A fine mesh is required to obtain an accurate solution for both the structure and thermal analysis.
Also this element is known to give poor results for plasticity or incompressible behavior. This element should only be used for linear elastic analysis. The element is integrated numerically using one point at the centroid of the element.

The 10 nodes tetrahedral element is a second-order isoparametric three-dimensional tetrahedron. Each edge forms a parabola so that four nodes define the corners of the element and a further six nodes define the position of the "midpoint" of each edge. This allows for an accurate representation of the strain field in elastic analysis. The stiffness of this element is formed using a four-point integration. This element can be used for all constitutive relations. The second-order elements are usually preferred to a more refined mesh of first-order elements for most heat transfer problems.

The three-dimensional distorted brick element is an eight-node, isoparametric, arbitrary hexahedral. As this element uses trilinear interpolation functions, the strains tend to be constant throughout the element. This results in a poor representation of shear behavior. The shear (or bending) characteristics can be improved using alternative interpolation functions. The thermal gradients tend to be constant throughout the element. In general, a finer mesh is needed for these low-order elements than higher order elements.

All 3 types of elements were studied to compare their effects on final simulation results. Due to the geometry constraints, only a quarter shot sleeve was molded to test the element effects on maximum temperature, radial dimension changes, strain range and
stress. The process and related heat transfer coefficients applied for the element types study is the same as mentioned earlier in this charter.

<table>
<thead>
<tr>
<th>Element Type</th>
<th>Maximum temperature (°F)</th>
<th>Von Mises (MPa)</th>
<th>ID dimension change ( (\times 10^3) )</th>
<th>OD dimension change ( (\times 10^3) )</th>
<th>Max. strain range ( \Delta \varepsilon_{11} ) ( (\times 10^3) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 node Tetrahedral</td>
<td>910</td>
<td>854.9</td>
<td>1.244</td>
<td>7.303</td>
<td>3.424</td>
</tr>
<tr>
<td>8 node Brick</td>
<td>913</td>
<td>475.7</td>
<td>1.259</td>
<td>8.057</td>
<td>3.066</td>
</tr>
<tr>
<td>4 node Tetrahedral</td>
<td>925</td>
<td>654.9</td>
<td>1.640</td>
<td>8.032</td>
<td>2.792</td>
</tr>
</tbody>
</table>

Table 3.10. Element type sensitivity with rough mesh (0.5 length) results

Theoretically, compared to a 4-node and a 8-node elements, since a 10-node element uses a higher order integration method, with the same element size, the results with 10-node element should be more accurate as listed in Table 3.10. Due to the rough mesh size, there are temperature, stress and deformation result differences among the 3 different types of elements.

With reduced element size, the simulation results are convergent for all the 3 types of element. The simulation results of 3 types of element with 0.3” length mesh size are limited in less than 7% range differences for the temperature, distortion, stress and strain range, as listed in Table 3.11.
Table 3.11. Element type sensitivity with fine mesh (0.3 length) results

For 4-node and 8-node elements, the calculation times were about the same. With
the 10 node-element, the calculation time was about 20 times longer. If the same mesh
size is necessary, 4-node and 8-node elements are suggested. Using the 4-node element,
it required less space to install result files and is also easier to create the mesh. Four-node
element is the best selection for current problem.

3.3.2 Mesh Size Sensitivity

With focus on the study of mesh density to ensure accurate results, more fine mesh
have been created for 4-node element to evaluate the more efficient mesh size. Upon
reducing the original mesh length from 0.30” to 0.25” and 0.20”, the corresponding
results are shown in Table 3.12. Table 3.12 shows that between 0.20” and 0.30” length
mesh, the differences in deformation results is about 3%. For strain range results, the
difference is less 1.5%. However, for temperature and stress results, the differences among the 3 types of element are larger, and are about 13% and 10%, respectively.

<table>
<thead>
<tr>
<th>Mesh length (inch)</th>
<th>Maximum temperature (°F)</th>
<th>Von Mises (MPa)</th>
<th>ID dimension change (\times 10^3)</th>
<th>OD dimension change (\times 10^3)</th>
<th>Max. strain range (\Delta\varepsilon_{11} \times 10^3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20</td>
<td>946</td>
<td>882.4</td>
<td>7.616</td>
<td>1.614</td>
<td>4.832</td>
</tr>
<tr>
<td>0.25</td>
<td>950</td>
<td>834.2</td>
<td>7.703</td>
<td>1.654</td>
<td>4.768</td>
</tr>
<tr>
<td>0.30</td>
<td>958</td>
<td>792.8</td>
<td>7.708</td>
<td>1.619</td>
<td>4.683</td>
</tr>
</tbody>
</table>

Table 3.12. Mesh size sensitivity results

3.3.3 Cover Die Weight Influence

Due to improper alignment and assembly, sometimes the cover die will apply its weight on the shot sleeve end area. To simulate the cover die weight effect on shot sleeve deformation and stress/strain results, the 4-node element with 0.3 length mesh has been used. A large ID shot sleeve with thick wall design configuration was studied, and the cover die load was applied on the top portion of the shot sleeve biscuit end. The process data used for the simulation is the same as listed in Table 3.5. The results show that with a certain load applied at the biscuit end, there is no effect on strain range. However, it does have an effect on ID and OD dimension change. With the increase of end load on the shot sleeve, the ID and OD expansion will be reduced.
<table>
<thead>
<tr>
<th>Load condition (kg)</th>
<th>Maximum temperature (°F)</th>
<th>Von Mises (MPa)</th>
<th>ID dimension change ($\times 10^3$)</th>
<th>OD dimension change ($\times 10^3$)</th>
<th>Max. strain range $\Delta e_{22}$ ($\times 10^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No load</td>
<td>958</td>
<td>792.8</td>
<td>7.708</td>
<td>1.619</td>
<td>4.683</td>
</tr>
<tr>
<td>315</td>
<td>959</td>
<td>813.5</td>
<td>7.246</td>
<td>1.499</td>
<td>4.678</td>
</tr>
<tr>
<td>1350</td>
<td>959</td>
<td>820.4</td>
<td>5.705</td>
<td>1.265</td>
<td>4.663</td>
</tr>
<tr>
<td>4050</td>
<td>959</td>
<td>923.8</td>
<td>1.649</td>
<td>0.650</td>
<td>4.621</td>
</tr>
</tbody>
</table>

Table 3.13. Cover die weight influence results
CHAPTER 4

COMPUTER SIMULATION RESULTS

4.1 PLANE STRAIN MODEL APPLICATION

Based on the 2-D plane strain FEM model, some early work was conducted to study the effects of cooling design (internal and external cooling) and cycle time on radial distortion, as well as the effect of different design using H13/Copper composite sleeve [64] [65] [66].

In the study of the sleeve radial deformation with the developed 2-Dimensional plane strain model, the model creation and boundary conditions applied were the same as mentioned in chapter 3.
Figure 4.1 shows the model structure with cooling line location. The mesh used in the model with fine mesh near the inside surface where the temperature gradient is higher, and outside area with less mesh density. The relatively larger rectangular unmeshed areas in the figure below represent the cooling channels.

In the study, the shot sleeves have an inside diameter of 6.693 inches and outside diameter of 9.974 inches for both H13 standard sleeve and H13/Copper composite sleeve, which is the same inner and outer diameters as the previously mentioned commercial thin wall shot sleeve.
The simulation was conducted in 4 steps: 1) 31% of the shot sleeve lower portion was filled with molten metal, 2) the total cross section was filled with molten metal and intensification pressure was applied, and at this step solidification occurred, 3) die was opened and casting ejected, and 4) shot sleeve was emptied.

For the study of longer cycle time, only the time when the shot sleeve is empty was increased to let it have more time to cool down with exposure to the environment. In all the cycles, with the assumption of plane strain model at the central regions of the shot sleeve cross section, shot tip lubrication was not considered, and also there was no end effect and runner cut effect involved in the simulation. The heat transfer coefficients in different steps were chosen based on data mentioned in the model development chapter.

To study the radial deformation, two extreme boundary conditions are applied: 1) the outside diameter of shot sleeve is in a fully constrained condition for the purpose of studying the shot sleeve ID distortion as well as the possible plunger tip sticking problems, and 2) the OD is in an unconstrained condition with the goal for finding out the clearance between shot sleeve OD and cover die. The simulations were run for 10 cycles to achieve quasi-steady states. The final nodal point displacement results are recorded along the angle from 0° to 180° (top to bottom) for calculation of the final radial deformation.
4.1.1 Cooling Effect On Standard H13 Sleeve Radial Distortion

For the constrained (outside surface of shot sleeve is fully constrained) case study, the change of inside diameter due to different cooling methods is shown in Figure 4.2. Compared with the original diameter, combining the cooling methods makes the inside diameter decrease less than other methods. The average reduction of the inside diameter of both cooling method is about 0.022”, which is in the same range as reported by Zeeman (0.016” smaller at die end of the sleeve I.D. during operation) [67].

![Figure 4.2. Comparison of radial deformation (I.D.) at fully constrained condition with different cooling methods](image-url)
Figure 4.3. Comparison of radial deformation (O.D.) at unconstrained condition with different cooling methods

For the unconstrained case study, the change of outside diameter due to different cooling methods is shown in Figure 4.3. Without cooling, the diameter increases about 0.070”. Under the different cooling conditions, the outside diameters increase about 0.066”, 0.060” and 0.052”, respectively, for bottom cooling, middle cooling and both cooling methods. Based on the above results, it is clearly shown that when cooling lines are applied to the shot sleeve, the expansion can be reduced.

Figure 4.4 gives the result of external cooling effect on the inside diameter change. For the external cooling method, there is no constraint on the outside surface of the shot
sleeve, so the shot sleeve will freely expand and the I.D. will also increase. In this case, the I.D. increased about 0.017”. The change of the I.D. mainly depends on the cooling load.

![Graph showing comparison of I.D. radial deformation with external cooling method](image)

**Figure 4.4.** Comparison of I.D. radial deformation with external cooling method

In summary, the maximum OD expansion with unconstrained condition or ID reduction with fully constrained condition of the shot sleeve diameter occurred when no cooling was applied. In the above simulations, it is clear that the order of choice of controlling the change of shot sleeve diameter is both cooling method, middle cooling method and bottom cooling method. External cooling method can be used for controlling the expansion of shot sleeve I.D. But we should keep in mind that due to the cooling situation, the temperature gradient increase at the area where the cooling applied, and will increase thermal stress in these regions.
4.1.2 Cycle time effect on radial distortion

Figure 4.5 and Figure 4.6 show the results of the effect of the longer cycle time (100 seconds cycle time compared with the original 90 seconds cycle time) on the final outside and inside diameter dimension changes of shot sleeve. In both of the cases with longer cycle time, both cooling lines are also applied. For outside diameter comparison (unconstrained condition), near the top and bottom, the longer cycle time can reduce the expansion about 0.010". Also it seems that the longer cycle time allows the outside diameter to expand more uniformly. For the inside diameter, both the longer and the original cycle time make the diameter change with the same pattern. It seems there is not much effect with increasing the cycle time, it only reduced the I.D. expansion about 0.001".
Figure 4.5. Comparison of radial deformation (O.D.) at unconstrained condition with longer cycle time

Figure 4.6. Comparison of radial deformation (I.D.) at fully constrained condition with longer cycle time
4.1.3 H13/Copper Composite Sleeve Radial Distortion

Using the same technique, the radial distortion of H13/Copper composite shot sleeve has been studied. The composite shot sleeve is constructed by adding a layer of copper in the middle. Two designs with different copper layer thickness (copper layer thickness of 0.04” and 0.232”) were studied. The study results are shown in the following figures. As the same as last paragraph, the constrained model focus on the study of sleeve ID distortion, and unconstrained condition focus on the sleeve OD clearance study.

![Comparison of radial deformation (I.D.) at fully constrained condition with different cooling methods (Copper layer thickness 0.04”)](image)

Figure 4.7. Comparison of radial deformation (I.D.) at fully constrained condition with different cooling methods (Copper layer thickness 0.04”)

97
Figure 4.8. Comparison of radial deformation (I.D.) at fully constrained condition with different cooling methods (Copper layer thickness 0.232")

Figure 4.9. Comparison of radial deformation (O.D.) at unconstrained condition with different cooling methods (Copper layer thickness 0.04")
Based on the results, in general there is not much difference for I.D. change for thin layer copper H13/Copper composite and thick layer copper H13/Copper composite sleeves, even compared with the standard H13 sleeve. Comparing with the O.D. changes, it was found that the H13/Copper composite sleeve with thick layer copper expand more at both top and bottom radial direction and change less at middle location. The H13/Copper composite sleeve with thin layer has similar results as that of the standard H13 sleeve.
4.2 3-DIMENSIONAL MODEL FOR BISCUIT END DISTORTION

The 2-D model is a useful approach to study the radial distortion at near central regions of the shot sleeve. At the biscuit end, due to the runner cut geometry effect, contact of the shot sleeve with die block as well as heat transfer along the axial direction between the shot sleeve biscuit end and the die block, this approach is no longer valid. A 3-D model must be applied for the study. From a process point of view, the heat transfer boundary conditions are also different from the biscuit end and central regions, and lubricants spray will be applied at the biscuit end of shot sleeve. Based on the previous sensitivity analysis, 4-node element with 0.3 length size mesh gave reasonable results, so the same size mesh was used for the 3D simulation also.

4.2.1 Shot Sleeve Biscuit End OD Distortion and Constraint Condition Study

In the die casting manufacturing process, the shot sleeve biscuit end is held by a die block. A designed clearance between the shot sleeve OD and the die block is applied for assembly and operation purposes. In order to obtain the correct study results, how the shot sleeve biscuit end contacts with die block should be studied in advance.

Based on the available shot sleeve geometry and process data, different constraint conditions at the biscuit end were studied for finding reasonable boundary condition in a computer simulation.
For the current radial distortion study at the biscuit end, two different constraint conditions were involved:

- **Unconstrained** - only the bottom portion of the sleeve at the biscuit end contacts with die block as shown in Figure 4.11;
- **Fully constrained** - all of the outside surface of the sleeve at the biscuit end contact with die block as shown in Figure 4.12. This is the extremely constrained condition.

![Figure 4.11. Diagram of shot sleeve unconstrained condition](image-url)
The reason to use the mentioned fully constrained condition is due to its industrial application. Also by applying the two extreme conditions (unconstrained and fully constrained), the upper and lower bound. Results can be obtained.

(A). Unconstrained Condition

The unconstrained condition model focus on the OD change of the shot sleeve at the biscuit end to study the possible clearance between the shot sleeve OD and the die block. Several factors were involved in this study – effects of wall thickness, runner design style, runner corner radius and end nose radius.
With the applied constraint on the bottom portion of the shot sleeve at biscuit end, the OD change results are collected in the following tables.

<table>
<thead>
<tr>
<th>Design</th>
<th>R = 0.25&quot;</th>
<th>R = 0.5&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design A</td>
<td>2.184</td>
<td>1.602</td>
</tr>
<tr>
<td></td>
<td>8.072</td>
<td>5.092</td>
</tr>
<tr>
<td>Design B</td>
<td>2.178</td>
<td>1.609</td>
</tr>
<tr>
<td></td>
<td>8.154</td>
<td>5.272</td>
</tr>
</tbody>
</table>

Table 4.1. Runner shape and runner corner radius effects on shot sleeve OD dimension changes

The results from Table 4.1 are based on the unconstrained condition with the end nose radius zero. The results show that the OD dimension is more stable in thick wall design than that of a thin wall sleeve. Comparing different runner design and runner corner radius, there is not much difference on the OD dimension changes, no matter whether a thin wall or thick wall shot sleeve. The OD dimension changes in the vertical direction are more than that of in the horizontal direction.
Table 4.2. End nose radius effects on shot sleeve OD dimension (Design A runner and runner corner radius 0.5)

<table>
<thead>
<tr>
<th>Design A Runner Radius 0.5”</th>
<th>Thin wall sleeve</th>
<th>Thick wall sleeve</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Vertical OD change (× 10²)</td>
<td>Horizontal OD change (× 10²)</td>
</tr>
<tr>
<td>End Nose R = 0”</td>
<td>1.646</td>
<td>7.370</td>
</tr>
<tr>
<td>End Nose R = 0.25”</td>
<td>2.097</td>
<td>7.944</td>
</tr>
<tr>
<td>End Nose R = 0.5”</td>
<td>2.107</td>
<td>9.604</td>
</tr>
</tbody>
</table>

With focus on the effect of end nose radius on OD dimension changes, Design A runner shape with runner corner radius of 0.5” shot sleeve were studied with different end nose radius. Table 4.2 shows the results of the study. Results show that with an increase of end nose radius, the OD dimensions increase for thin wall sleeve. But for thick wall shot sleeve, the effect of shot sleeve end nose radiuses on OD dimension changes were not much different (major dimension change are based on vertical OD data).

For the given shot sleeve, the designed clearance between shot sleeve and die is 0.027 ± 0.005” in diameter at the biscuit end. The results show that regardless of the runner shape design, thick or thin shot sleeve, with different runner corner radius and end nose radius, the maximum OD dimension expansion were less than the original clearance between shot sleeve and die block. This mean that shot sleeve is supported only bottom portion at biscuit end initially, and even with the expansion, the upper portion will not make contact with die and there is no contact load applied to shot sleeve.
In summary, based on the above results, following conclusions can be obtained:

- Thin wall shot sleeve OD changed more than that of thick wall shot sleeve;
- Shot Sleeve OD Vertical direction expanded more than horizontal direction;
- For two different runner shape designs, there is not much difference for OD dimension changes;
- There is not much difference on OD dimension change due to different runner corner radius;
- With increase end nose radius, OD expanded more for thin wall shot sleeve, but not much effect of end nose radius on thick wall OD changes.

(B). Fully Constrained Condition

On the other hand, if there is not enough clearance between shot sleeve and die block, due to the expansion of shot sleeve OD, there will be contact between shot sleeve OD and die block at the biscuit end. Thus, the die block will apply contact load on the shot sleeve. The extreme situation is when there is no gap between shot sleeve and die block at biscuit end when shot sleeve is installed. To model this situation with simplicity, the shot sleeve outside surface at biscuit end was fully constrained for all the node points at all directions. All the possible movements and rotations of the shot sleeve outside surface were constrained. The over constrained condition may cause the potential higher stresses for the analysis.
The computer model shows that in this condition, the calculated stresses are much higher than the previous results, as shown in Table 4.3. These stresses are predicted to exceed the yield strength of H13 at operating temperature. The operation temperature is about 950 °F. Based on the material properties, yield strength of H13 at this temperature is about 882.4 MPa.

<table>
<thead>
<tr>
<th></th>
<th>Maximum Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Totally Constrained</td>
</tr>
<tr>
<td>Design A</td>
<td>861.8 – 930.7</td>
</tr>
<tr>
<td>Design B</td>
<td>861.8 – 930.7</td>
</tr>
</tbody>
</table>

Table 4.3. Stress comparison of unconstrained and constrained condition

The conclusion of stress range is consistent with the earlier works by Rosbrook [22] and Sirinterlikci [36]. In their study for test sample and die, the results showed that no yielding occurred for H13 steel during either test or die casting operation. On the other hand, the results also conclude that suitable clearance between shot sleeve at biscuit end and die block is critical to avoid higher stress at operation. It is clear that a properly designed clearance can achieve lower operation stress in shot sleeves.
4.2.2 Shot Sleeve Biscuit End ID Distortion

(A). Biscuit End Distortion with Design Clearance

Under the designed 0.027 ± 0.005" design clearance between shot sleeve at biscuit end and die block, the shot sleeve distortions have been studied.

In computer simulation, the unconstrained model was used. Several factors has been studied for their effect on shot sleeve ID dimension change. These factors are wall thickness, runner shape, runner corner radius and end nose radius.

Table 4.4 shows the results of ID dimension change with constant sleeve end nose radius zero, and different runner shape design, wall thickness and runner corner radius. Table 4.5 shows the results of end nose radius effect on ID dimension change.

<table>
<thead>
<tr>
<th>Design</th>
<th>R</th>
<th>Thin wall sleeve</th>
<th>Thick wall sleeve</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Vertical ID change (× 10^2)</td>
<td>Horizontal ID change (× 10^3)</td>
</tr>
<tr>
<td>A</td>
<td>0.25</td>
<td>1.3220</td>
<td>5.202</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>1.3086</td>
<td>4.614</td>
</tr>
<tr>
<td>B</td>
<td>0.25</td>
<td>1.4717</td>
<td>5.384</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>1.4572</td>
<td>5.434</td>
</tr>
</tbody>
</table>

Table 4.4. Runner shape and runner corner radius effects on shot sleeve ID dimension
Table 4.5. End nose radius effects on shot sleeve ID dimension (Design A runner and runner corner radius 0.5)

<table>
<thead>
<tr>
<th>Design A Runner</th>
<th>Thin wall sleeve</th>
<th>Thick wall sleeve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius 0.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Vertical ID change ($\times 10^3$)</td>
<td>Horizontal ID change ($\times 10^3$)</td>
</tr>
<tr>
<td>End nose radius R=0</td>
<td>1.3086</td>
<td>4.614</td>
</tr>
<tr>
<td>R=0.25</td>
<td>1.2170</td>
<td>4.436</td>
</tr>
<tr>
<td>R=0.5</td>
<td>1.1930</td>
<td>4.530</td>
</tr>
</tbody>
</table>

In general, based on the results from Table 4.8 and 4.9, the shot sleeve ID expanded during operation for all the situations with unconstrained condition. In summary:

- Thin wall shot sleeve expanded more than thick wall shot sleeve in ID;
- Vertical direction expanded more than that of horizontal direction;
- Design B runner shape structure expanded more than design A;
• There is not much difference between runner radius 0.25 and 0.5;
• With increase end nose radius, ID expanded less for both thin and thick wall sleeves.

Since the results from simulation indicate no interference between shot sleeve and plunger tip, there is no plunger tip-sticking problem exists in this situation. The above scenarios indicate a condition of excessive clearance between the sleeve and tip, and may cause molten metal blow back from the gap between sleeve and plunger tip, which might in lead to tip sticking due to the solidified metal on the tip.

From shot sleeve dimensional stability point of view, thick wall sleeve is better than thin wall sleeve, design A runner structure is more favorable than design B runner since for this design configuration, the ID dimension is stable.

(B). Biscuit End Distortion with Fully Constrained Clearance

As mentioned before, an extremely constrained condition exists – sleeve biscuit end fully constrained. Table 4.6 shows the shot sleeve ID dimension change with fully constrained condition and all the negative value indicates ID contraction.
Table 4.6. Shot sleeve ID change with end nose radius zero

<table>
<thead>
<tr>
<th>Design</th>
<th>R=0.25</th>
<th>R=0.50</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>-6.721</td>
<td>-6.750</td>
</tr>
<tr>
<td></td>
<td>-5.072</td>
<td>-5.074</td>
</tr>
<tr>
<td></td>
<td>-5.322</td>
<td>-6.595</td>
</tr>
<tr>
<td></td>
<td>-3.822</td>
<td>-4.058</td>
</tr>
<tr>
<td>B</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>-6.754</td>
<td>-6.714</td>
</tr>
<tr>
<td></td>
<td>-5.130</td>
<td>-5.046</td>
</tr>
<tr>
<td></td>
<td>-6.594</td>
<td>-6.611</td>
</tr>
<tr>
<td></td>
<td>-3.892</td>
<td>-3.764</td>
</tr>
</tbody>
</table>

Results show that, if less clearance between die and shot sleeve, in extreme situation without clearance, in both thin and thick wall design, no matter what kind of the runner shape, there definitely will exist ID reduction. The ID will reduce and pass the designed clearance between shot sleeve and plunger tip, and cause plunger tip sticking problem.

In general, without clearance between shot sleeve and die block, the ID will contract during operation:

- Thin wall contracted more than thick wall shot sleeve, but not much difference.
- Vertical direction dimension change more than horizontal direction
- Not much difference between two different runner shape designs
- There is not much difference between radius 0.25” and 0.5”
From the stability point of view, thick wall shot sleeve is more stable than thin wall. There is not much difference for two different runner shape designs and their runner corner radii. The predicted plunger tip sticking due to the excessive constrained situation is consistent with author’s earlier work [64].

4.2.3 Shot Sleeve Free End Distortion

In the central regions of shot sleeve, there is no constraint on sleeve OD, which is called the free end region. Based on the partially constrained condition at biscuit end, the shot sleeve OD and ID dimension changes were studied. Since the central region is far from end, so assume that runner shape, runner corner radius and biscuit end nose radius effect will not affect the dimension changes. For this reason, only design A runner shape with runner corner radius 0.5” was used in the simulation. Based on previous results, wall thickness has more effect on shot sleeve ID and OD dimension change, so only the thin wall and thick wall geometry will be modeled. Table 4.7 shows the simulation results with the same process data.
The results from Table 4.7 are summarized in below:

- Thin wall shot sleeve distorted more than thick wall shot sleeve;
- For OD dimension changes, both vertical and horizontal direction were expanded;
- For ID dimension changes, ID expanded in horizontal direction, and contracted in vertical direction
- Horizontal direction dimensions changed more than that in vertical direction. The final shape for both OD and ID is horizontal direction length larger than vertical direction length. This shape is different from the shape changed at biscuit end as mentioned before. The graphical display of the OD and ID dimensional changes is shown in Figure 4.13.
Figure 4.13. Free end region OD and ID shape changes
CHAPTER 5

SHOT SLEEVE THERMAL FATIGUE LIFE PREDICTION

In the operation of die casting process, another issue needing to be addressed is the tool failure due to the cyclic thermal load - the thermal fatigue. As mentioned before, several methods can be used for fatigue life predictions. But due to the unavailable of test data, the only method suitable for the current thermal fatigue application is the universal slope method.

5.1 BASIC RELATION FOR THERMAL FATIGUE PREDICTION

For the condition of enough clearance between the shot sleeve and the die block at biscuit end, there is no plastic deformation as predicted in the computer simulations. The universal slope method can be used for fatigue life prediction in this situation.
Based on the relation from chapter 2:

\[ \Delta \varepsilon = D^{0.6} (N_f)^{-0.6} + \frac{3.5 \sigma_{max}}{E} (N_f)^{-0.12} \] (5.1)

The first term is due to the plastic deformation, and in this calculation, since there is no plastic deformation, which is zero, so the equation become:

\[ \Delta \varepsilon = \frac{3.5 \sigma_{max}}{E} (N_f)^{-0.12} \] (5.2)

The equation (5.2) is the fundamental relation for fatigue life prediction. All the constants in the equation are from Manson’s early study [50].

5.2 MATERIAL PROPERTIES AT ELEVATED TEMPERATURE

Figure 5.1 and 5.2 are the temperature distribution and a typical nodal point temperature history in one cycle of a thick wall shot sleeve at the biscuit end. From the two figures, it shows that service temperature could reach as high as about 960 °F. The material properties used in equation (5.2) for fatigue life prediction should be determined at this temperature level.
Figure 5.1. Biscuit end temperature distribution (Thick wall sleeve design A runner shapes with runner corner radius 0.5)

Figure 5.2. Typical temperature history at biscuit end region (Thick wall sleeve design A runner shape with runner corner radius 0.5)
In Sirinterlikci’s work [36], a detailed summary of converting the existing mechanical properties to cyclic fatigue properties was conducted. The basic concepts for the data conversion are:

- The stress-strain data available in the literature is the outcome of uni-axial static (monotonic) tensile test. These data should be converted to cyclic stress-strain data;
- The literature included stress-strain data for H13 with greater hardness compared with that in the actual dies. Usually the hardness of H13 in operation was 41-42 HRC, while the data from the literature are much higher than this value, which is around 44-48[41];
- Tempering of the tooling surface should also be taken into account. The tool surface has softened from 41-42 HRC to 37-38 HRC during the operation due to exposure to thermal load.

These three issues make a significant difference to the determination of H13 material fatigue properties with cyclic thermal load. With the conversion work of Benedyk [21] and Schruff [68], Sirinterlikci [36] use of the average value of the two to create the Young’s Modulus for thermal fatigue life prediction.
<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>E (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>194,954</td>
</tr>
<tr>
<td>200</td>
<td>188,765</td>
</tr>
<tr>
<td>400</td>
<td>179,481</td>
</tr>
<tr>
<td>600</td>
<td>173,292</td>
</tr>
<tr>
<td>795</td>
<td>167,103</td>
</tr>
<tr>
<td>1000</td>
<td>151,631</td>
</tr>
<tr>
<td>1200</td>
<td>99,024</td>
</tr>
</tbody>
</table>

Table 5.1. Cyclic temperature dependent Young’s Modulus data from Benedyk [21]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>E (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>68</td>
<td>216,200</td>
</tr>
<tr>
<td>212</td>
<td>211,600</td>
</tr>
<tr>
<td>392</td>
<td>205,800</td>
</tr>
<tr>
<td>572</td>
<td>199,500</td>
</tr>
<tr>
<td>752</td>
<td>193,000</td>
</tr>
<tr>
<td>932</td>
<td>184,900</td>
</tr>
<tr>
<td>1112</td>
<td>173,600</td>
</tr>
<tr>
<td>1292</td>
<td>161,000</td>
</tr>
</tbody>
</table>

Table 5.2. Cyclic Temperature dependent Young’s Modulus data from Schruff [68]

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>E (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>213,700</td>
</tr>
<tr>
<td>200</td>
<td>206,800</td>
</tr>
<tr>
<td>400</td>
<td>199,900</td>
</tr>
<tr>
<td>600</td>
<td>186,200</td>
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<tr>
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<td>151,700</td>
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<td>110,300</td>
</tr>
</tbody>
</table>

Table 5.3. Average value of the cyclic Young’s Modulus data [36]
Schruff [68] studied the conversion factor for ultimate tensile strength at thermal cyclic load condition and the relation was given in Table 5.4.

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>( \sigma_{uts} ) at 43-44 HRc</th>
<th>( \sigma_{uts} ) at 41.5 HRc</th>
<th>( \sigma_{uts} ) at 37 HRc</th>
<th>( \sigma_{uts-cyclic} ) at 37 HRc</th>
</tr>
</thead>
<tbody>
<tr>
<td>212</td>
<td>1400.0</td>
<td>1300.0</td>
<td>1130.0</td>
<td>768.0</td>
</tr>
<tr>
<td>392</td>
<td>1360.0</td>
<td>1260.0</td>
<td>1100.0</td>
<td>764.7</td>
</tr>
<tr>
<td>572</td>
<td>1300.0</td>
<td>1210.0</td>
<td>1050.0</td>
<td>759.9</td>
</tr>
<tr>
<td>752</td>
<td>1230.0</td>
<td>1140.0</td>
<td>995.0</td>
<td>754.2</td>
</tr>
<tr>
<td>932</td>
<td>1120.0</td>
<td>1040.0</td>
<td>906.0</td>
<td>745.3</td>
</tr>
<tr>
<td>1112</td>
<td>800.0</td>
<td>792.0</td>
<td>784.0</td>
<td>689.6</td>
</tr>
<tr>
<td>1292</td>
<td>200.0</td>
<td>198.0</td>
<td>196.0</td>
<td>196.4</td>
</tr>
</tbody>
</table>

Table 5.4. Ultimate tensile strength (in MPa) for H13 at 43-44, 41.5 and 37 HRc [68]

Based on Table 5.3 and Table 5.4, at temperature range of 950 °F, Young’s Modulus value should be, \( E = 157192.4(MPa) \). And the value of ultimate tensile strength should be, \( \sigma_{uts} = 736.4(MPa) \).

5.3 STRAIN RANGE RESULTS AND ANALYSIS

Based on equation (5.2), three variables are necessary for thermal fatigue life prediction. Since Young’s Modulus \( E \) and ultimate tensile strength \( \sigma_{uts} \) are already known from the previous paragraph, the only unknown, strain range, \( \Delta \varepsilon \) can be
determined from computer simulations. From equation (5.2), at a certain temperature range, with constant $E$ and $\sigma_{m}$, thermal fatigue life is reverse proportional to strain range. Higher strain range will result in shorter fatigue life.

Based on the criteria mentioned above, several design factors will be studied for their effect on thermal fatigue life.

5.3.1  End Nose Radius Effect on Strain Range

The study was divided into two steps in the following paragraphs to evaluate the effects of the shot sleeve end nose radius on fatigue life (End nose radius refer to Figure 3.9 and Figure 3.10).

(A). End Nose Radius Effect on Bottom Region

Based on the same process data mentioned in Chapter 3, the effects of shot sleeve end nose radius on stress and strain range at the bottom region of the shot sleeve at the biscuit end were studied. Results shows that, with the change of end nose radius from R=0", R=0.25" to R=0.5", the Von Mises stress changed from 748.8 (MPa), 748.1 (MPa) to 699.8 (MPa). The corresponding strain ranges are changed from $5.052 \times 10^{-3}$, $5.052 \times 10^{-3}$ to $4.718 \times 10^{-3}$, respectively. This case study is only for thin wall shot sleeve with Design A runner shape and runner corner radius of 0.5".
The results show that with the increase of end nose radius, it will lower the stress and increase thermal fatigue life at the bottom region of the shot sleeve at biscuit end. Theoretically, with the increase of end nose radius, the stress concentration factor will be reduced, and the stress should be lower under the same operation condition. But from above analysis, there is only very small stress decrease with the end nose radius increased from 0” to 0.25”. The reason may be due to the rough mesh size that could not capture the actual geometry changes. For the stress results, cautions should be made.

(B). End Nose Radius Effect On Runner Region

Table 5.5 shows the results of strain range at the runner region of the shot sleeve with respect to the change of the shot sleeve end nose radius. The computer simulation results indicate that, with the increase of end nose radius, the calculated strain range also increases. With applying the fatigue life and strain range relation expressed in equation (5.2), it can be concluded that the predicted fatigue life around runner region will be reduced with the increase of end nose radius.

<table>
<thead>
<tr>
<th>Sleeve End Nose Radius</th>
<th>Thin Wall Strain range ($\times 10^3$)</th>
<th>Thick Wall Strain range ($\times 10^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R=0.00</td>
<td>5.941</td>
<td>5.140</td>
</tr>
<tr>
<td>R=0.25</td>
<td>6.164</td>
<td>5.120</td>
</tr>
<tr>
<td>R=0.50</td>
<td>6.776</td>
<td>5.808</td>
</tr>
</tbody>
</table>

Table 5.5. Thin & Thick wall sleeve runner region strain range vs. runner nose radius (Design A runner shape with runner corner radius 0.5)
Comparing the strain range results at bottom region of shot sleeve biscuit end with the strain range at runner region, it can be seen that the maximum strain ranges occur near the runner region. This means that a shorter thermal fatigue life will occur at the runner region. All the following studies will only focus on the runner regions for their thermal fatigue life predictions with the end nose zero degree radius design configuration.

5.3.2 Design Parameters Effect On Strain Range at Runner Region

Figure 5.3 to Figure 5.10 are the strain range results from simulations for thin/thick wall sleeves with different runner shape and runner corner radius designs. Table 5.6 shows the strain range values with different design parameters.

![Figure 5.3. Thick wall design A runner shape with runner radius 0.5 & end corner radius zero strain range](image)

Figure 5.3. Thick wall design A runner shape with runner radius 0.5 & end corner radius zero strain range

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Figure 5.4. Thin wall design A runner shape with runner radius 0.5 & end corner radius zero strain range

Figure 5.5. Thick wall design A runner shape with runner radius 0.25 & end corner radius zero strain range
Figure 5.6. Thin wall design A runner shape with runner radius 0.25 & end corner radius zero strain range

Figure 5.7. Thick wall design B runner shape with runner radius 0.5 & end corner radius zero strain range
Figure 5.8. Thin wall design B runner shape with runner radius 0.5 & end corner radius zero strain range

Figure 5.9. Thick wall design B runner shape with runner radius 0.25 & end corner radius zero strain range
Table 5.6. Strain range values due to different design parameters

<table>
<thead>
<tr>
<th>Runner Shape</th>
<th>Runner Corner Radius</th>
<th>Thin Wall $\Delta \varepsilon_{11} \times 10^3$</th>
<th>Thick Wall $\Delta \varepsilon_{11} \times 10^3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design A</td>
<td>R=0.25</td>
<td>6.004</td>
<td>5.058</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>5.941</td>
<td>5.140</td>
</tr>
<tr>
<td>Design B</td>
<td>R=0.25</td>
<td>5.343</td>
<td>5.033</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>5.592</td>
<td>4.683</td>
</tr>
</tbody>
</table>

Based on the strain data from Table 5.6, the effect of design parameters on strain range can be summarized as:
• Wall Thickness Effect on Strain Range: In general, thick wall sleeve is more stable as mentioned before. The strain range of thick wall sleeve is smaller than that of thin wall sleeve. With thick wall sleeve design, the service life will last longer than that of thin wall sleeve design.

• Runner Shape Effect on Strain Range: For thin wall design, runner shape with design B has longer fatigue life due to the small strain range generated during cycles. For thick wall sleeve, there is not much difference between two different runner shape designs.

• Runner Corner Radius Effect on Strain Range: From the results above, it can be seen that there is not much difference for strain range change with different runner corner radius.

5.3.3 Fully Constrained Condition Effect on Strain Range

A fully constrained condition was applied at biscuit end, and all other parameters were the same, the computer simulations were conducted and the strain range results are shown in Table 5.7.
<table>
<thead>
<tr>
<th>Runner Shape</th>
<th>Runner Corner Radius</th>
<th>Thin wall</th>
<th></th>
<th>Thick wall</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Von Mises Stress (MPa)</td>
<td>Strain Range ($\times 10^3$)</td>
<td>Von Mises Stress (MPa)</td>
<td>Strain Range ($\times 10^3$)</td>
</tr>
<tr>
<td>Design A</td>
<td>R=0.25</td>
<td>919.8</td>
<td>6.285</td>
<td>863.8</td>
<td>5.376</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>924.5</td>
<td>5.885</td>
<td>932.1</td>
<td>6.195</td>
</tr>
<tr>
<td>Design B</td>
<td>R=0.25</td>
<td>927.9</td>
<td>6.505</td>
<td>945.2</td>
<td>6.404</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>873.5</td>
<td>5.796</td>
<td>899.0</td>
<td>6.272</td>
</tr>
</tbody>
</table>

Table 5.7. Strain range values under fully constrained condition

It is clear that the fully constrained condition, compared with the unconstrained condition between shot sleeve and die, generates larger strain ranges. This will cause shorter service life of H13 shot sleeves under this condition.

5.4 THERMAL FATIGUE LIFE PREDICTION

Figure 5.11 to Figure 5.18 show the stress distribution results under different shot sleeve design configurations.
Figure 5.11. Thick wall design A runner shape with runner radius 0.5 & end corner radius zero Von-Mises stress distribution

Figure 5.12. Thin wall design A runner shape with runner radius 0.5 & end corner radius zero Von-Mises stress distribution
Figure 5.13. Thick wall design A runner shape with runner radius 0.25 & end corner radius zero Von-Mises stress distribution

Figure 5.14. Thin wall design A runner shape with runner radius 0.25 & end corner radius zero Von-Mises stress distribution
Figure 5.15. Thick wall design B runner shape with runner radius 0.5 & end corner radius zero Von-Mises stress distribution

Figure 5.16. Thin wall design B runner shape with runner radius 0.5 & end corner radius zero Von-Mises stress distribution
Figure 5.17. Thick wall design B runner shape with runner radius 0.25 & end corner radius zero Von-Mises stress distribution

Figure 5.18. Thin wall design B runner shape with runner radius 0.25 & end corner radius zero Von-Mises stress distribution
Comparing the stress distribution figures above, all the highest stress locations are in the same region, which are near the triangle area of the shot sleeve end, ID surface and the bottom of runner. The highest strain ranges were also observed from the same region for all the corresponding designs. The regions with highest stress and highest strain range are the possible thermal fatigue failure locations.

Based on the strain range from Table 5.7, Young’s Modulus $E = 157,192.4(MPa)$, ultimate tensile strength $\sigma_{uts} = 736.4(MPa)$ and equation (5.2), the thermal fatigue life with different design parameters were calculated and listed in Table 5.8.

<table>
<thead>
<tr>
<th>Runner Shape</th>
<th>Runner Corner Radius</th>
<th>Thin Wall Fatigue Life (cycles)</th>
<th>Thick Wall Fatigue Life (cycles)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design A</td>
<td>R=0.25</td>
<td>4324</td>
<td>18,048</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>4737</td>
<td>15,784</td>
</tr>
<tr>
<td>Design B</td>
<td>R=0.25</td>
<td>11,468</td>
<td>18,809</td>
</tr>
<tr>
<td></td>
<td>R=0.5</td>
<td>7820</td>
<td>34,294</td>
</tr>
</tbody>
</table>

Table 5.8. Thermal fatigue prediction vs. different design parameters

From the predictions in Table 5.8, it is obvious that thick wall sleeve has much longer thermal fatigue life than that of thin wall sleeve. In one study [36], the die casting dies were observed to show visible evidence of heat checking after about 15,000 cycles. Corresponding shot sleeve service life is in the range of 20,000 to 50,000 cycles. Based on the commercial sleeve failure data, the H13/Copper composite sleeves last about
37,000 cycles. But their H13 sleeves usually have less service life than that of H13/Copper composite sleeves [69]. From the actual transmission case manufacturing operation data analysis, the average fatigue life of the standard H13 shot sleeves with thick wall is about 32,477 cycles. Compare the predicted thermal fatigue life with commercial thick wall sleeve fatigue data, the predicted thermal fatigue life of 34,294 is well matched with the actual operation shot sleeve thermal fatigue life. As mentioned in Chapter 3 with the sensitivity analysis, due to the changes of mesh size and element type, the induced errors for strain range is about 3%. This strain range errors can cause the predicted thermal fatigue life vary in the range of about ± 4,000 cycles, so the predicted thermal fatigue life for the thick wall shot sleeve should be about 34,000 ± 4,000 cycle.

Figure 5.19 is a sample of the thermal fatigue failure shot sleeve of the same design configuration at the biscuit end with runner cut through the top of the sleeve end. This commercial shot sleeve is with the same manufacturing purpose as used for computer model. The thermal fatigue failure areas are well matched with the computer model prediction.
It can be noticed from Figure 5.19 that the fatigue onset also appeared near the top middle inner surface at the biscuit end. The simulated strain range for the thick wall shot sleeve in these areas are between 0.004367 – 0.004480. Based on the same method for fatigue life calculations, the predicted thermal fatigue life for the same areas are in the range 49,000 – 61,000 cycles.
In the earlier work from Park [5], it was pointed out that shot sleeve mechanical failures included design related issues such as wall thickness, fixing device type and machine alignment; and the process related issues included metal pressure and temperature. In his study, the thick wall cylinder theory was used to analyze the stress and fracture mechanics for fatigue life prediction with the assumption of considering only metal pressure, but not any thermal loads were involved. Park concluded that longitudinal cracking can be prevented by proper design, and control of metal pressure, and circumferential cracking is mainly related with fixing device in a shot sleeve with a combination of die casting machine fixturing and alignment. The wear of the shot sleeve and plunger tips from sliding friction due to the mechanical deformation during operation were also pointed out.
In Bennett’s guideline [14], checking for shot sleeve biscuit end alignment when an initial trouble occurred was suggested. Working clearances, coolant volume/temperature and shot sleeve designs have been recognized as the factors that may cause potential problems during operation.

Due to the varieties of design parameters and process change factors, shot sleeve service life will be very different under different design configurations. A good practice of design criterion must be developed. Based on this study, some criterion will be proposed in the following paragraphs.

6.1 INFLUENCE OF WALL THICKNESS ON OD/ID AND STRAIN RANGE

An additional small ID (ID diameter 1.5”, with wall thickness of 1.35” and 2.0”) shot sleeve has been used for design guideline studies. The same process data was used for the small ID shot sleeve studies. Assuming that the OD and ID dimension changes and strain range changes between its thin wall (wall thickness 1.35”) and thick wall (wall thickness 2.0”) design configurations are linear.

Based on the above assumptions, the small ID shot sleeve strain range and OD/ID dimension change results with respect to wall thickness were modeled and are plotted in Figure 6.1 and Figure 6.2.
Figure 6.1. Small ID (ID = 1.5\textquotedbl") sleeve strain range plot

Figure 6.2. Small ID (ID = 1.5\textquotedbl") sleeve OD/ID dimension change plot

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For large ID (ID = 6.693”) shot sleeve, also assume the same linear strain range and OD/ID dimension change relation between wall thickness of 1.6405” and 2.6405”. The results of these changes with respect to wall thickness can be plotted in Figure 6.3 and Figure 6.4 based on the results from Chapter 4 and Chapter 5.

Figure 6.3. Large ID (ID = 6.693”) sleeve strain range plot
The results showed that for both small and large ID shots sleeves, with increase of sleeve wall thickness, the shot sleeve ID/OD dimension changes will decrease, and the strain ranges will also decrease. It means that thick wall sleeve design will make the sleeve dimension more stable than thin wall design.

Figure 6.4. Large ID (ID = 6.693”) sleeve OD/ID dimension change plot
6.2 OD/ID DESIGN CLEARANCE GUIDELINE

6.2.1 OD Design Clearance

In Bennett's guidelines, only ID design clearance was suggested. The computer model results in Chapter 4 indicated that with different clearances between shot sleeve and cover die will cause very different stresses and ID dimension change patterns:

- From stress point of view – with enough clearance between shot sleeve OD and cover die, stresses are under yield range; but if there is no clearance between the shot sleeve OD and the cover die, stresses are much higher, even over the yield point.

- From deformation point of view – with enough clearance between sleeve OD and cover die, shot sleeve ID will expand; but if there is no clearance between sleeve OD and cover die, shot sleeve ID will be reduced.

Figure 6.2 and Figure 6.4 showed the OD dimensional changes for small ID sleeve (ID=1.5") and large ID sleeve (ID = 6.693") with thin wall and thick wall shot sleeve designs. In general, no matter of the capacity of sleeve (ID size), with the increase of wall thickness, the shot sleeve OD dimension change will decrease, which means that the design of OD clearance for thick wall sleeve should be less than that of thin wall sleeves. Since the temperature changes of cover die and shot sleeve contact region are very
limited compared with the sleeve and tip temperature changes, assume there is no thermal expansion for cover die at shot sleeve OD contact area. With this assumption, the calculated OD expansion can be used as the design clearance between the sleeve OD and cover die as shown in Figure 6.2 and Figure 6.4. For example, for a large ID shot sleeve with the thick wall design, the OD clearance from Figure 6.4 is about 0.016”. For small ID shot sleeve with thick wall design, the OD clearance from Figure 6.2 is about 0.005”.

6.2.2 ID Design Clearance – Unconstrained Condition

Shot sleeve ID design clearances depend on the OD constrained conditions. With current OD unconstrained condition, as mentioned before, the ID will expand no matter what kind of design style and wall thickness.

(A). Bennett’s Existing Design Clearance Rule

Based on many years of design experience, Bennett [14] gave a general rule of the design clearance between shot sleeve ID and plunger tip for shot sleeve ID up to 3”. In his rule, for the tips up to 3” diameter, a good starting rule is to allow for 0.001” per inch of tip diameter, then plus an additional 0.001” to the sum. According the rule with the hypothetical extension of up to 8” of tip diameter, the design clearance can be obtained as shown in Figure 6.5.
Figure 6.5. Bennet’s design clearance rule

This rule usually only works for Aluminum 380 casting with beryllium copper tip materials up to tip diameter 3”. Above 3” diameter tip, Bennett stated that the working clearance are usually “developed” by a customer according to the variables that apply to their process. In this plot, the range with sleeve ID large than 3” stands for hypothetical design clearance.

(B). Minimum and Maximum Clearance

Before we study the minimum and maximum clearances, the first assumption is that the shot sleeve and the plunger tip is located on the same axis during operation. If the
defined clearance equals the difference between the shot sleeve ID and the plunger tip diameter, the gap between the shot sleeve ID and the plunger tip should be half of the clearance.

The minimum clearance should consider the machining tolerance on the shot sleeve ID and plunger tip, the film thickness of the applied lube, as well as the sleeve distortion and tip expansion during operation. If the clearance is too small, friction between shot sleeve ID and plunger tip will occur and cause plunger tip sticking problems. Based on Bennet’s experiences for small ID sleeve, the 0.002” minimum clearance seems is a good start point.

If the clearance is too big, the molten metal will flow/penetrate the opening at die casting pressures. Usually 0.003” – 0.004” machined channels are maintained for die casting venting applications to ensure there is no metal penetrating into the vent [70] [71]. To keep the same size of gap between sleeve ID and plunger tip, the maximum clearance should be 0.006” – 0.008”.

(C). Plunger Tip Expansion Assumption

To be able to derive the ID clearance design guide between the shot sleeve and the plunger tip, the plunger tip expansion information must be available. As mentioned in chapter 2, the plunger tip material is Cu-Be alloy with thermal expansion rate about 1.5 times higher than that of H13 tool steel [13], and for some other plunger tip materials, the expansion rate can be twice that of the steel shot sleeve [14]. With the same operation
condition, the plunger tip expansion rate is about 1.5 – 2.0 times than that of shot sleeve ID at biscuit end. With applying cooling during its operation, the operation temperature of plunger tip should be lower than that of sleeve, and this will reduce the tip expansion to a certain level.

Since the thermal conductivity of Cu-Be alloy is about 10 times higher than that of H13 steel, this will ensure the quick heat transfer and keep temperature evenly distributed inside the tip. Then the assumption of uniform expansion of plunger tip is applied. In die casting operations, the reasonable maximum operation temperature of a plunger tip is about 392 °F and its initial temperature is about 70 °F [13]. Based on this assumption, the tip expansion rate under different operation temperatures (without and with water cooling inside the tip) can be calculated based on fundamental thermal expansion equation and plotted in Figure 6.6.
To study the shot sleeve ID expansion, theoretically assume its uniform expansion at average operation temperature of 392 °F without affected by any geometrical factors and process variables. With this assumption, the shot sleeve ID expansion rate can be calculated with the same thermal expansion equation as before and plotted in Figure 6.7. In reality, due to different operation processes and shot sleeve design configurations, the temperature distribution and wall thickness will be different and affect the final sleeve ID expansion.
Figure 6.7. Theoretical shot sleeve ID expansion

(E). Operation Clearance

The term design clearance is the initial design clearance between the shot sleeve ID and the plunger tip. The operation clearance is defined as the clearance between shot sleeve ID and plunger tip during operation process.

Based on Bennett’s design clearance rule, combined with the theoretical shot sleeve ID and plunger tip expansions, the theoretical operation clearance can be plotted in Figure 6.8. If assuming tip no expansion (in the maximum cooling condition), it will generate the maximum operation clearance. And if no cooling applied inside tip (the tip...
maximum expansion condition), it will generate the minimum operation clearance. These two cases are the upper and lower bound of the operation clearances. For convenience to analysis, the maximum 0.008” and minimum 0.002” clearance limits are also plotted in the same plot.

![Theoretical operation clearances with different operation conditions](image)

**Figure 6.8.** Theoretical operation clearances with different operation conditions

Theoretically, from Figure 6.8, Bennett’s rule apply to the sleeve with ID diameter of 7” if adjusting the cooling rate inside the plunger tip so that both the tip and the shot sleeve ID have the same expansion rate to keep the clearance constant. The clearance in this whole process is 0.008”. And this clearance meets the maximum and minimum
clearance requirements. If there is no cooling for the plunger tip, no matter what kind of sleeve ID diameter, according Bennett’s rule, the operation clearance will be less than the minimum allowed clearance, and plunger tip sticking problems will occur during operation. On the other hand, if maximum cooling rate was applied to a tip to the extreme situation of no tip expansion, the Bennett’s rule is still valid for sleeve ID diameter up to 2”. Between the upper and lower bound of the tip expansion with different tip cooling rate (tip operation temperature), the operation clearance can be calculated and plotted in the same figure.

(F). Actual Shot Sleeve ID and Plunger Tip Clearance Design

Due to the wall thickness and biscuit end effect, as well as the process variable changes, it will induce uneven expansion of sleeve ID. In the actually shot sleeve operation, the sleeve ID expansions are different from the assumed expansion as shown in Figure 6.7. It can be noticed from Figure 6.2 and 6.4 that thin wall sleeve ID expanded more than thick wall sleeve for currently OD unconstrained condition. For large ID (ID = 6.693”) shot sleeve design configuration, its ID will expand about 0.012” during operation with thin wall design and 0.0077” with thick wall design, respectively. For small ID (ID = 1.5) shot sleeve design configuration, its ID will expand about 0.0012” with thin wall design and 0.0008” with thick wall design, respectively. These 4 expansion results are plotted in Figure 6.9 to compare with the theoretical sleeve ID expansion.
Figure 6.9. Actual shot sleeve ID maximum expansion (vertical direction) data

The above data are the sleeve vertical expansions, which are the maximum sleeve ID expansions. During operation, due to the different horizontal direction expansion rate, it created a “out of roundness” sleeve ID shape, which is defined as the difference of the vertical expansion and horizontal expansion. Based on the computer modeling results from previous chapters, the “out of roundness” of 1.5” ID sleeve with thin and thick wall designs are 0.0004” and 0.0002”, respectively. The “out of roundness” of 6.693” ID sleeve for thin and thick wall designs are 0.009” and 0.005”, respectively. These “out of roundness” makes the operation clearance more complex.
According to Bennett’s rule, the design clearance for 1.5” ID sleeve is 0.0025” and for 6.693” ID sleeve is 0.0077”. The actual operation clearance depends on the tip and sleeve ID expansion rate and the “out of roundness” during operation. In vertical direction, the actual operation clearance is:

\[ OC_v = \text{Design Clearance} + \text{Sleeve ID Expansion} - \text{Tip Expansion} \]

and in horizontal direction, the actual operation clearance is:

\[ HC_v = OC_v - \text{“out of roundness”} \]

With Bennett’s design clearance of 0.0025” for small ID sleeve, if adjusting tip cooling rate to ensure the same expansion rate for both tip and sleeve ID, the final operation clearance in vertical and horizontal directions are 0.0025”/0.0025” (thin/thick wall design) and 0.0021”/0.0023” (thin/thick wall design), respectively. The final operation clearances for both thin and thick wall designs meet the maximum and minimum clearance requirements in both vertical and horizontal directions. Even if with the maximum tip cooling rate applied (tip no expansion), the final operation clearance in vertical and horizontal directions will be 0.0037”/0.0033” (thin/thick wall design) and 0.0033”/0.0031” (thin/thick wall design), respectively. These vertical and horizontal direction clearances still meet the design requirements.

If there is no cooling for 1.5” tip, the final operation clearance in vertical and horizontal directions are -0.0012”/-0.0016” (thin/thick wall design) and -0.0016”/-0.0018” (thin/thick wall design), respectively. It is clear that sticking problems will occur due to the predicted minus operation clearances in both directions.
For large ID of 6.693” sleeve, design clearance is 0.0077” based on Bennett’s rule. Under this design clearance, even with control of both tip and sleeve ID with the same expansion rate, if with consideration of the “out of roundness” of 0.009” for thin wall and 0.005” for thick wall designs, the final operation clearances will exceed the maximum clearance 0.008”. Bennett’s rule is no longer valid for this large ID size sleeve. For this sleeve, if using design clearance of 0.002”, and control the same expansion rate for both sleeve and tip, the maximum operation clearance will be 0.002” in vertical direction and 0.007” in horizontal direction for thick wall design, respectively. The thick wall design meets the maximum and minimum clearance requirements during operation. On the other hand, if the thin wall design was used with the same design clearance of 0.002”, due to the large “out of roundness” value generated during operation, the final clearance in vertical direction and horizontal direction will be 0.002” and 0.011”, respectively. The final operation clearance in the horizontal direction will exceed the maximum clearance requirement.

(G). Summary

Based on the above analysis, the clearance between the shot sleeve ID and plunger tips for different shot sleeve ID with different wall thickness design configurations are very different. The general clearance guideline can be summarized as following:

- The minimum and maximum clearance between shot sleeve ID and plunger tip in the range of 0.002” – 0.008” regardless of the size of sleeve ID is suggested.
- The Bennett’s rule is valid for small ID shot sleeve clearance design with applied cooling inside plunger tip. If no cooling applied, plunger tip sticking problems may occur.

- Using the upper bound for large ID sleeve clearance design based on Bennett’s rule is not valid due to the “out of roundness” of sleeve ID expansion during operation.

- Due to the large ID sleeve “out of roundness”, use the possible minimum design clearance to compensate the different sleeve ID expansion in different directions under minimum wall thickness requirement. When the wall thickness less than minimum requirement, the “out of roundness” can not be compensated and will exceed the maximum clearance, the problem of molten metal flow/penetrate into the opening will occur.

6.2.3 ID Design Clearance – Fully Constrained Condition

Figure 6.10 shows the ID dimension change with OD fully constrained condition for large ID shot sleeve. With this constrained condition, no matter what kind of design configurations and wall thickness, the ID always will be reduced. Based on the ideal maximum design clearance of 0.004”, during operation the ID will be reduced to about 0.006” – 0.007” during operation, and the final operation clearance will be –0.002 – 0.003”. For this situation, plunger tip sticking problem will occur.
Figure 6.10. Shot sleeve ID dimension change with OD fully constrained condition

To avoid plunger tip sticking, design enough shot sleeve OD clearance between shot sleeve OD and cover die is necessary, and this will also lower the service stress during operation.
6.3 FATIGUE LIFE DESIGN GUIDELINE

6.3.1 Fatigue Design Guideline

As mentioned in chapter 2 and chapter 5, fatigue life can be predicted with Universal Slope Method based on the strain range of the shot sleeve during operation. The higher of strain range is, the lower of fatigue life will be.

Based on the strain range data from Figure 6.1 and 6.3 for small and large ID sleeve, and applying equation (5.2) for fatigue life prediction, the 34,000 and 50,000 cycles fatigue life curves can be generated and plotted in Figure 6.11. With adding the wall thickness design criterion from Park [5] in the same figure, the design guideline of fatigue life with respect to wall thickness for different shot sleeve ID can be obtained from the same Figure 6.11. It is clear that current fatigue life design require larger wall thickness than Park’s prediction, since Park’s work only considered the mechanical load in his calculation.
Figure 6.11. Shot sleeve wall thickness design requirement for thermal fatigue life

Since the small ID shot sleeve predicted results are based on the assumption of with the same operation process as that of large ID sleeve, which may not reflect the real operation process for small ID shot sleeve, and the fatigue life guideline for small sleeve is only for reference. Fatigue life data collected from plant operations for large sleeves were well match the prediction result, and it can be used to guide the same kind of ID sleeve design with the similar process information.
6.3.2 Fatigue Life Prediction Sensitivity Analysis

The possible errors for the fatigue life predictions are came from 3 different sources: process parameter changes, computer model and material property input. The studies are based on the current designed constrained condition with enough clearance between shot sleeve OD and cover die for large ID thick wall shot sleeve.

(A). Process Parameter Introduced Errors

Based on the possible process data range, the strain range change has been studied accordingly.

The original spray temperature is set at 100 °F. If changing the spray temperature range from 50 °F to 150 °F, the strain range will fluctuate at the range of around ± 5%. With the ± 5% strain range fluctuation for the current fatigue life of 34,294 cycles, the life will be changed to the range from 22,800 to 52,500 cycles.

The heat transfer between shot sleeve and molten metal is set at the upper bound for current simulation. If change the interface heat transfer to the lowest possible value, the strain range will reduce 14%. The current fatigue life of 34,294 cycles will increase to 120,500 cycles.

The heat transfer coefficient between spray coolant and sleeve surface was chosen at highest level for current simulation. If applying the moderate level heat transfer value,
the strain range will also reduce 10%. The current fatigue life of 34,294 cycles will increase to 82,500 cycle.

(B). Computer Model Effect on Fatigue Life Prediction

As mentioned in chapter 3, the element type/size can lower the strain range to about 7%. And it also has effect on the temperature range – can increase the temperature by about 15 °F. With the assumption of the same material property used previously for the fatigue life prediction, the 7% strain range difference will reduce the calculated fatigue life to 19,508 from original 34,294 cycles. With the 15 °F temperature difference, the calculated fatigue life will reduced to 32,173 cycles.

(C). Material Property Input Effect

The material property played a very important role in fatigue life prediction. The previously predicted fatigue life of 34,294 cycles is with the average Young’s Modulus from Benedyk and Schruff and the converted cyclic ultimate tensile strength from Schruff. If only Benedyk’s Young’s Modulus at the same temperature level was used for calculation, the fatigue life will increase to 37,724 cycles. On the other hand, if only Schruff’s Young’s Modulus value was used for the calculation, the fatigue life will reduce to 9329 cycles. If there are ± 5% of measured material property data errors involved in the Young’s Modulus and cyclic tensile strength, the calculated fatigue life will be in the range of 14,897 to 78,949 cycles.
In general, for the current calculated fatigue life of 34,294 cycles, the process parameters will introduce about from −11,500 cycles to + 86,000 cycles predicted errors. The computer model (element type/size) will introduce about 15,000 cycles predicted errors. While with the material property errors (assume ± 5% measured material property data errors or with different sources), the calculated fatigue life will be ranged from 9329 to 78,949 cycles compared with the original predicted life of 34,294 cycles.

From above, it is clear that to have the suitable process information with accurate interface heat transfer coefficient is extremely important for thermal fatigue life prediction. The correct material properties at elevated temperature used in the equation (5.2) are as well critical for the fatigue life prediction. Otherwise, the predicted fatigue lives are far more from the reality.
CHAPTER 7

CONCLUSIONS

7.1 CONCLUSIONS

A computer model has been successfully developed for shot sleeve thermal and structural analysis with the verification of experimental test and commercial operation data. With the developed computer model and suitable interface heat transfer boundary conditions, the shot sleeve radial distortion of different design configurations has been studied, and OD/ID design clearance has been established. A method for thermal fatigue life prediction was well developed.
The following conclusions can be obtained from the work:

- A method, combined with 2-D plane strain and 3-D computer models and process data, has been developed that allows for shot sleeve thermal and structural analysis. The method has been verified based on the experimental data and the optimized model with accurate heat transfer coefficients have been used in the computer simulation for shot sleeves.

- For a given ID configuration shot sleeve, to achieve a certain service life for a given process, the shot sleeve wall thickness must meet a minimum requirement. The minimum wall thickness can be obtained based on computer simulation method established from this study.

- The effects of shot sleeve design parameters (runner shape, runner corner radius and runner end radius) can be evaluated by the current computer model.

- Applying the developed computer simulation method, the shot sleeve OD/ID radial dimension changes can be studied, and the design clearance between shot sleeve OD and cover die and the clearance between shot sleeve ID and plunger tip can be derived from the simulation data analysis.
• The Universal Slope Method has been successfully applied for shot sleeve thermal fatigue life prediction, and the result has been verified by commercial shot sleeve information.

• The same procedure has been used for small ID shot sleeve radial distortion and thermal fatigue life prediction. Combine the small and large ID shot sleeve simulation results, it is realized that control the "out of roundness" of the ID dimensional changes is the key factor for large ID shot sleeve clearance design between its ID and plunger tip.

• The research itself proved that with the combination of computer model and theoretical analysis provided a economic and suitable way for solving the current existing problems under higher operation temperature situation, which are extremely difficult to be handled by test. This suggests that integration of initial shot sleeve design and process development with computer simulations should be the ultimate goal. By this way, it can not only reduce cost, but also make the design more efficient.

7.2 FUTURE WORK

Using the computer tool for systematically study shot sleeve thermal distortion and thermal fatigue was first tried. To establish a more reliable correlation between computer simulation result and the real industrial practice, more works for different shot sleeve
configurations and process changes should be involved for both simulation results as well as commercial shot sleeve operation data. After certain correlation verification and confidence has been established, it is the time for fully use of the computer model for shot sleeve design.
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