Design of a Novel Centrifugal Casting Technique for the Fabrication of Metal Matrix Composites

by

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Submitted to the Department of Mechanical Engineering in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering at the MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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Abstract

Centrifugal casting has been used for hundreds of years to produce castings using centrifugal force to aid in filling the mold. In recent years, this process has been employed to produce metal-matrix composites, a new class of material that offers improved mechanical properties for high temperature applications and significant mass reduction compared to conventional metal alloys. However, despite its immense potential, this technique has not been further developed.

This thesis explores the design of a new system that aims to improve the centrifugal casting method for the infiltration of molten metal into ceramic preforms, by taking better advantage of the presence of the fluid pressure at the preform entrance to increase the infiltration pressure.

It was shown that this technique provides a fluid pressure at the mold entrance that is far greater than that which can be attained by pouring alone, or by conventional centrifugal casting, in which the metallostatic “head” is limited by the relatively small length of metal in the radial direction. Samples of tin and aluminum composites reinforced by alumina powders of 1 and 9.5 microns in diameter, respectively, were produced, and the microstructures of the composites show that the technique has great potential for commercial application. The infiltration pressure achieved in producing the tin-alumina composite was estimated to be 37 atmospheres and that for the aluminum-alumina composite was 7 atmospheres, although calculations show that it could go beyond 12 atmospheres with the 2 hp centrifuge machine used in the laboratory.

Further analysis of the centrifuge system reveals that a much large infiltration pressure of 1000 atmospheres is achievable with a larger centrifuge motor and a new centrifuge design. The motor horsepower requirement depends on the length of the runner, the speed of rotation and the time it takes for the centrifuge to accelerate to the desired speed. The “weakest link” in the design, it was found, was the runner of the centrifuge system, which must withstand the high pressures at the high operating temperatures of the centrifugal process.

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Chapter 1

Introduction

1.1 Centrifugal Casting

Centrifugal casting is defined as "the process of filling molds by (1) pouring metal into a sand or permanent mold that is revolving about either its horizontal or its vertical axis, or (2) pouring metal into a mold that is subsequently revolved before solidification of the metal is complete" [1]. The centrifugal casting process can be broadly divided into two types: horizontal centrifugal casting and vertical centrifugal casting.

The horizontal centrifugal casting technique uses the centrifugal force generated by a rotating cylindrical mold to throw the molten metal against the mold wall and form the desired shape [1]. The first patent on a centrifugal casting process was obtained in England in 1809, and the first industrial use of the process was in 1848 in Baltimore, when centrifugal casting was used to produce cast iron pipes [1]. Today, a great number of other hollow components including pipes, cylinder sleeves, piston rings and various hollow billets are produced using this method [2]. The biggest advantage in centrifugal casting is that the participation of centrifugal forces facilitates the casting of such hollow components without the use of central cores. This also allows the wall thickness to be controlled simply by controlling the weight of metal poured [3].

Vertical centrifugal casting, as the name implies, is the type of centrifugal casting that rotates about a vertical axis. As with horizontal centrifugal casting, the vertical method can also be used to make pipes and cylinder sleeves as well. However, when fully up to speed, the
liquid in the mold takes up a parabolic shape, with the result that the wall at the base of the
cylinder is thicker than that at the top [3]. Extra machining is therefore required to produce a
parallel bore.

In addition to producing hollow cylinders described above (whereby such method is known
as true centrifugal casting [1]), the range of applications of vertical centrifugal casting machines
is considerably wider. Castings that are not cylindrical, or even symmetrical, can be made using
this technique. In this method, molten metal is poured into a rotating mold. The centrifugal
force of the rotating mold forces the molten metal against the interior cavity of the mold under
constant pressure until the molten metal is solidified. This method is known as semicentrifugal
casting [1].

In recent years, both horizontal and vertical centrifugal casting methods have been employed
to produce metal-matrix composites, a new class of material that offers improved mechanical
properties for high temperature applications and significant mass reduction compared to con-
ventional metal alloys.

1.2 Fabrication of Metal-Matrix Composites

Metal-matrix composites (MMCs) are defined as “engineered combinations of two or more
materials (one of which is a metal) in which tailored properties are achieved by systematic
combinations of different constituents” [1]. Conventional monolithic materials have limitations
in terms of achievable combinations of properties, such as strength, stiffness, coefficient of
thermal expansion, etc. By combining these materials with continuous or discontinuous fibers,
whiskers or particles – usually of a ceramic material – of different sizes and in different volume
fractions, the properties of the new material that is produced from this combination may be
improved considerably, and these properties can be significantly different from the properties
of the constituents [4].

The fabrication techniques in processing MMCs are many. A basic requirement is the initial
intimate contact and the intimate bonding between the ceramic phase and the molten alloy [1].
However, because of the poor wettability of most ceramics by molten metals, intimate contact
between ceramic and metal frequently needs to be promoted by artificial means, and these can
be broadly divided into two categories: solid state and liquid state fabrication techniques [5]. Solid-state processes are generally used to obtain the highest mechanical properties in MMCs [5], and these include powder metallurgy [6], diffusion bonding [7] and high-rate consolidation [8].

However, a majority of the commercially viable applications of MMCs are now produced by liquid-state processing because of the inherent advantages of this processing technique over the solid-state techniques mentioned above. For example, liquid metal is generally less expensive and easier to handle than are powders, and the composite material can be produced in a wide variety of shapes, using methods already developed in the casting industry for unreinforced metals [5]. There are four broad classes of liquid-phase techniques: preform infiltration, mixing, spray co-deposition and in-situ fabrication [4].

The use of centrifugal casting in the fabrication of MMCs is one of the infiltration processes of the liquid phase method. The other infiltration techniques include: squeeze casting, gas pressure infiltration, Lorentz force infiltration, reactive infiltration, pressureless spontaneous infiltration, and vacuum infiltration [4, 5]. A major advantage of infiltration processes is that they allow for near-net shape production of parts fully or selectively reinforced with a variety of materials. If cold dies and reinforcements are used, or if high pressures are maintained during solidification, matrix-reinforcement chemical reactions can be minimized, and attractive, defect-free matrix microstructures can be achieved [5]. However, the limitations of these processes are the need for the reinforcement to be self-supporting, either as a bound preform or as a dense pack of particles or fibers, and that heterogeneity of the final product may result from preform deformation during infiltration or from clustering of fibers that are detrimental to the composite mechanical properties. Also, tooling may be expensive if high pressures are used [5]. The following sections provide a brief overview of the different infiltration processes mentioned.

### 1.2.1 Squeeze Casting

A strategy to overcome poor wetting between a ceramic and a metal is to supply mechanical work to force the metal into a preform that it does not wet. Although the purpose of an externally applied pressure is to overcome the capillary forces, higher pressures can provide additional benefits such as increased processing speed, control over chemical reactions, refined
matrix microstructures and better soundness of the product through feeding of solidification shrinkage [5].

Pressure, in this case, may be applied mechanically, which involves a force that is exerted on the molten metal by the piston of a hydraulic press, and subsequently maintained during solidification. The pressures involved usually range from about 10 to 100 MPa. This type of infiltration process is currently the most widely investigated for commercial applications, and has been directly adapted from established processes designed to cast unreinforced metals [9, 10, 11, 12, 13, 14, 15].

Composites produced by this method generally feature a pore-free matrix. They also show uniform reinforcement distribution, reduced fiber-to-fiber contact, and solidification of eutectic at reinforcement surface [4]. However, application of pressure may induce preform deformation or breakage during infiltration [10, 15, 16, 17, 18, 19, 20].

1.2.2 Gas Pressure Infiltration

The pressure required to force metal into the preform of reinforcing phase may also be applied by an inert gas, such as Argon, typically pressurized in the 1 – 10 MPa range [21, 22, 23, 24, 25, 26, 27]. Although the use of higher pressures (up to 17 MPa) has been investigated for aluminum matrix composites [28], the safety issues involved become a major limiting factor. With gas pressure infiltration, however, relatively inexpensive tooling (compared to squeeze casting) is usually sufficient, and a better control of atmosphere is possible [4].

1.2.3 Lorentz Force Infiltration

The use of electromagnetic body forces to drive molten metal into a preform was developed by Andrews and Mortensen [29], where they fabricated alumina fiber-reinforced aluminum composites using this method. In the Lorentz force-driven infiltration, the metal is subjected to a strong and transient high frequency electromagnetic (EM) field which interacts with the eddy currents produced within the metal [29]. This interaction results in a Lorentz body force on the metal which drives the metal into the preform when the preform is suitably oriented with respect to the direction of the force.

Residual porosity, preform deformation and fiber breakage are generally absent in such
castings. However, the frequency required for infiltration is very high, and even then, only a maximum fiber volume fraction of 25% has been achieved, and there is a limit to the infiltration length that can be attained using this method [4].

1.2.4 Reactive Infiltration

Reactive infiltration is essentially an in-situ composite fabrication process in which a compound is formed by simultaneous infiltration and reaction of a porous solid by a melt, such as the infiltration of silicon in porous carbon, and the infiltration of aluminum in titanium oxide, mullite and nickel-coated aluminum oxide [4].

Complex, near net shape components with little overall change in the part dimensions can be produced by this technique, and the infiltration conditions can be controlled to achieve the desired level of reaction and structure [4]. Unfortunately, the reaction is usually exothermic and may initiate explosively. In addition, the requirement of a reaction during infiltration clearly limits the choice of reinforcement-matrix combinations to those systems where a sufficiently large driving force exists for chemical interaction at the processing temperatures. Also, the reaction kinetics need to proceed with sufficient rapidity to form an appreciable quantity of the product phase during the time period of infiltration [4].

1.2.5 Pressureless Spontaneous Infiltration

Spontaneous infiltration of preforms by metals in the absence of an external pressure can be effected if the melt and preform compositions, temperatures and gas atmosphere are controlled such that good wetting conditions are achieved for self-permeation (wicking) of the metal in the preform [30, 31, 32, 33, 34]. Examples of such composites produced include titanium-carbide-reinforced steel and nickel-base alloys [35]. Another example is the PRIMEX$^TM$ pressureless infiltration process, developed by Lanxide Corporation, Newark, Delaware [36], whereby aluminum-magnesium alloys infiltrate ceramic preforms at temperatures between 750°C and 1050°C in a nitrogen-rich atmosphere.

As with reactive infiltration, the spontaneous infiltration technique has the ability to fabricate void-free net-shape metal- and ceramic matrix composites having high integrity. However, the shortcomings of this process are high capital cost of processing equipment, stringent control
on atmosphere and temperature, restrictions on matrix chemistry (e.g. in the case of aluminum alloys, magnesium is required), and excessive reaction of the matrix with the reinforcement due to extremely long contact times (typically 1 to 10 hours) [4]. In the PRIMEX$^TM$ pressureless infiltration process, for example, infiltrate rates can only reach a maximum of 25 cm/hour [36].

1.2.6 Vacuum Infiltration

For some matrix-reinforcement systems, creating a vacuum around the reinforcement provides a sufficiently large negative pressure differential between the preform and the surroundings, which, in countergravitational configuration, drives the liquid metal through preform interstices against the forces of surface tension, gravity and viscous drag [37].

The process is relatively simple and does not require extensive tooling [4]. Furthermore, it also produces MMCs that have less porosity and oxidation than low-pressure techniques. However, a maximum pressure of 1 atm only can be created for impregnating the fibers; this magnitude of pressure may not be adequate for melt ingress in densely packed preforms of non-wetting fibers. Therefore, vacuum infiltration is usually done in conjunction with chemical methods of wettability enhancement (e.g. fiber surface modification, addition of wetting-enhancing elements to the matrix alloy, etc.) [4].

1.3 Use of Centrifugal Casting for Fabrication of MMCs

In addition to the numerous infiltration techniques described above, centrifugal casting methods have also been used for the fabrication of MMCs, such as the production of tubular reinforced metal [38, 39, 40, 41, 42, 43, 44], using the horizontal centrifugal casting method. Subjecting metallic slurries containing ceramic reinforcement to the action of a centrifugal force – created by rotation of the mold containing the slurry – results in the formation of two distinct zones within the slurry: a particle-enriched zone and a particle depleted zone. Different densities of the particles and melt cause spatial segregation under the action of the centrifugal force in a manner similar to the segregation observed in sand castings under the influence of gravitational force [4]. If the particles are lighter than the melt (e.g. graphite or mica in aluminum or tin alloys) the particle-enriched zones form at the inner periphery of horizontally spun cylindrical
centrifugal castings [4], resulting in the formation of antifriction surfaces for bearing applications [1]. Similarly, particles heavier than the melt (e.g. zircon, silicon carbide or aluminum oxide in aluminum alloys) will segregate near the outer periphery of the casting, leaving behind a predominantly metal-rich zone which provides a tough backing material, and resulting in the formation of hard, abrasion resistant surfaces near the outer periphery.

In addition to making hollow components that consist of two distinct zones by the horizontal centrifugal casting method, the specific gravity difference between particles and melts can also be taken advantage of, through the vertical centrifugal casting method, to make solid objects that either show a “coating” of reinforced material near the casting surface, or a gradient in the amount of reinforced material within the object [45, 46, 47].

Sugishita et al. [45] produced aluminum and tin based composites containing hollow carbon microballoons obtained from volcanic ash. The microballoons were packed in an alumina tube which also served as a crucible for molten metal atop the powder bed. The infiltration and dispersion of powders was carried out at high rotational speeds of 2680 to 4000 RPM, and the rotation was continued until the solidification was completed. The carbon microballoons were concentrated near the casting surface. During rotation, the centrifugal force pressurizes the molten metal into the interstices of the particulate bed. Also, the particles, being lighter than the metal, rise up towards the axis of rotation along the crucible walls where the shear forces in the melt compress them against the walls. As the rotation is continued through solidification phase, the particles are “frozen in” at their final position in the casting [4].

From this idea, the vertical centrifugal casting method can be brought a step further by infiltrating entire ceramic preforms to fabricate fiber-reinforced metal composites, so that the reinforced material in the composite produced is uniformly distributed. Large rotational velocities can generate a centrifugal force big enough to overcome the capillary forces for melt penetration and viscous forces for metal flow in the preform [4].

1.3.1 Infiltration of Preforms

The infiltration of a ceramic preform by a molten pure metal or alloy was first attempted by Noordegraaf et al. [48]. In this process, a shaped object is cast by subjecting a mold to a centrifugal acceleration directed to the bottom of the mold. A dispersed filler (particles of a
ceramic material) is placed in the mold before adding the melt, so that upon centrifuging, the centrifugal force field that is set up will force the melt into the cavities between the particles. This process, the authors acknowledged, is a modification of that in Reference [45]. Its aim, however, is to make shaped objects in which the filler is dispersed as homogeneously as possible throughout the entire metal matrix, by preventing, as far as possible, the filler from moving during the filling of the mold with pure metal or alloy.

For some reason, however, this invention was not further pursued and developed, and the centrifugal casting process for the fabrication of MMCs was not commercialized, even though all the other infiltration methods discussed in the previous sections have already been adopted commercially. Furthermore, few papers have been published on the infiltration of molten metal into ceramic preforms by centrifugal force [49, 50, 51, 52]. Most of the articles related with centrifugal force are concerned with the segregation of particles in molten metal during solidification [50].

Jiang et al. fabricated an alumina short fiber reinforced aluminum alloy via centrifugal force infiltration [49]. Their technique was similar to the vertical centrifugal casting method of producing hollow cylinders, whereby they placed an aluminum oxide preform in a cylindrical stainless steel mold vertically fixed to the end of the driveshaft of a motor. The molten metal was poured into the mold that was rotating at high speeds (1000 – 6000 RPM), and the metal penetrated the preform and solidified under centrifugal force. A cylindrical MMC was thus obtained.

The authors found that the hardness and wear resistance of the MMCs prepared via the centrifugal force infiltration route are almost identical to those of composites obtained via squeeze casting, and that the molten infiltration process was markedly affected by the pouring temperature, preheated mold temperature, and the time of application of centrifugal force. Nishida et al. also investigated the infiltration of fibrous preform by molten aluminum in a centrifugal force field [50]. In their experiment, they rotated a preheated graphite container of uniform cross-section placed horizontally from a vertical axis of rotation, in which a preheated fibrous preform was placed at the end. When the rotation speed reached a desired value, they poured molten pure aluminum into a pouring device that was concentric with the rotation shaft. The high angular velocity of the graphite container resulted in high pressure being generated.
in the molten aluminum due to centrifugal force when the container was instantaneously filled, and infiltration of aluminum into the preform was thus initiated.

The authors found that if the pressure generated by centrifugal force exceeds the threshold pressure due to lack of wettability, infiltration progresses to the bottom of the graphite container. This was confirmed by comparing the results obtained with calculated values.

Nishida et al. later wrote a second paper [51] that models the infiltration of molten metal in a fibrous preform by centrifugal force. The authors recognized that it is difficult to obtain the appropriate infiltration conditions only by experimental work alone, and thus they developed a theoretical analysis to aid in understanding the infiltration process. Unfortunately, however, the process was not developed further, even though the authors recognized the potential of this infiltration technique.

Taha et al. [52] compared the two types of MMCs fabricated by centrifugal casting and by squeeze casting, and discussed their infiltration mechanisms. They also briefly presented the main structural features obtained in the composites prepared.

The authors found that the pressure needed for full infiltration is less in centrifugal casting than in squeeze casting. In centrifugal casting, a pressure of 120 kPa was sufficient to achieve full infiltration of the specimen for aluminum oxide powder of 47 μm in diameter. The lowest squeeze casting pressure used to achieve full infiltration of the specimen, even for a much coarser aluminum oxide grain size of 115 μm, was of the order of 65 MPa – more than 500 times the pressure needed in centrifugal casting to infiltrate a much finer powder size. This same conclusion of a lower infiltration pressure needed in centrifugal casting than in squeeze casting was reached by Walter [53] many years earlier.

The authors also discovered that full infiltration in centrifugal casting is achieved in less than 2 seconds, although more than half a minute is necessary for full densification of the specimens. In squeeze casting, however, full infiltration is achieved in 25 seconds. This is a much longer time in comparison with the few seconds required in centrifugal casting. The authors hypothesized that this might be due to the dynamic action of the centrifugal force which could shorten the infiltration time. The long infiltration time in squeeze casting was confirmed in an earlier work by Asthana et al. [54], in which the infiltration took 3 to 30 minutes.

Finally, the authors reported that in centrifugal casting, as no significant powder displace-
ment occurred during infiltration, an almost uniform distribution of aluminum oxide particulates was achieved, observed in both longitudinal and transverse sections. The inter-particle spacing was found to have an almost constant value in the specimen. On the other hand, microscopic observations on squeeze-cast specimens indicate that although the distribution of aluminum oxide in the transverse section was almost uniform, it was not uniform in the longitudinal section. A gradual increase in the inter-particle spacing was observed from the bottom to the top of the specimen [55]. Generally, the inter-particle spacing obtained in squeeze casting was relatively larger than that in centrifugal casting.

These observations all point towards the enormous potential that centrifugal casting has in the fabrication of MMCs. This work is therefore a first attempt to study the possibilities that exist in such a technique. A new centrifuge system will be designed and built, as well as studied and discussed, with samples of MMCs obtained to test the system's capabilities. The limitations of this design will then be analyzed, and with the knowledge obtained from this analysis, a second improved design will be proposed.
Chapter 2

Design of New Apparatus for Centrifugal Casting

2.1 Theory

As pointed out in the previous chapter, the potential of using centrifugal casting in the fabrication of metal-matrix composites is immense. In the centrifugal casting techniques used by Nishida et al. [50] and Taha et al. [52], it is noted that both have failed to take advantage of the presence of the fluid pressure (or metal head) at the preform entrance to increase the infiltration pressure of the molten metal into the ceramic reinforcement.

From Reference [50], it is known that the fluid pressure, $P$, acting on the preform surface is given by:

$$P = \frac{\rho_m \omega^2}{2} (r_1^2 - r_0^2)$$  \hspace{1cm} (2.1)

where $\rho_m$ is the density of the molten metal, $\omega$ the angular velocity ( = $2\pi N$, where $N$ is revolutions per second), and $r_1$ and $r_0$ denote the radii of revolution of the preform inner surface and of the molten metal column surface respectively [50], as illustrated in Figure 2-1:

In the experiment conducted by Nishida et al. [50], due to the centrifugal force field set up in the container, the stream of molten metal flowing into it through the pouring device is broken up, because the speed of pouring can never match the speed with which the molten
Figure 2-1: A runner containing molten metal and a ceramic preform rotating about a vertical axis

metal travels through the container to the preform. This results in the molten metal filling the container in jets that leads to very a small value of \((r_1^2 - r_0^2)\). The low and uncontrolled radial depth of the fluid gives rise to a low fluid pressure. This same drawback is observed in the paper by Taha et al. [52], in which the length of molten metal in the steel tube is very short, thus setting \(r_0\) at a large and finite value. Furthermore, it is clear that as infiltration of the molten metal into the preform occurs, \(r_0\) increases, thus further decreasing the fluid pressure. As a result, infiltration might not be complete, if the fluid pressure drops below the threshold pressure during the infiltration process. The threshold pressure, \(P_{th}\), is the pressure that is required to infiltrate the preform (which exists due to the lack of wettability), and is given by [56]:

\[
P_{th} = -\frac{4V_f \gamma \cos \theta}{d_f (1-V_f)} \quad \text{or} \quad P_{th} = -\frac{6V_p \gamma \cos \theta}{d_p (1-V_p)}
\]  

(2.2)

where \(V_f (V_p)\) is the fiber (powder) volume fraction, \(\gamma\) the surface energy of the molten metal, \(\theta\) the contact angle, and \(d_f (d_p)\) the diameter of the fiber (powder).

### 2.2 Proposed Design

The problem of the small and uncontrolled radial depth of the fluid can be solved by filling the entire container (or “runner”) with molten metal prior to the initiation of centrifuging. The
importance of the extended length of the molten metal in the runner is that, upon centrifuging, it provides a metal head at the preform entrance that is far greater than that which can be achieved by pouring alone. By extending the fluid length in the runner into the center of rotation by means of a reservoir filled with molten metal, \( r_0 \) is essentially reduced to zero. From Equation 2.1, it is clear then that the infiltration pressure will be a maximum at any given rotation for any given metal.

The ability to achieve very high pressures is important in the infiltration of preforms with high threshold pressures, i.e. in the making of metal matrix composites (MMCs) that have high fiber/powder volume fractions, large contact angles, or small fiber diameters (or sub-micron ceramic powders). Such high pressures can also help in infiltrating the preform sufficiently rapidly so that little or no solidification takes place before the preform is completely infiltrated (which is especially crucial in the infiltration of cold preforms), as well as in the removal of undesirable pores in the MMCs – the presence of which can significantly reduce the mechanical properties of the material, such as its fracture strength. In addition, the high pressures achieved can also be used in microcasting, where molten metal is made to flow through microchannels in order to cast micro-parts for microelectromechanical systems (MEMS) applications.

An additional advantage of having a reservoir of molten metal at the center of rotation connected to a long runner – that is also completely filled with molten metal – is that the metal remains as a continuous body during filling, so that the metal head infiltrating the ceramic preform will be constant and controlled, as well as steady and laminar. This not only ensures that the preform will be fully infiltrated, but may also result in a better and more homogenous microstructure for the MMC produced.

### 2.2.1 Reservoir and Runner System

The reservoir and runner system is made up of three Swagelok® AISI 316 stainless steel tubings (part number SS-T8-S-065-20) [57] – one for the reservoir and two for the runners. These tubings have a nominal outer diameter of 1.27 cm (0.5”) and a nominal wall thickness of 0.1651 cm (0.065”), thus giving a nominal inner diameter of 0.9398 cm (0.37”). The tubing that is used for the reservoir is cut to a length of 11.7 cm (4.6”) and the runners are each cut to a length of 20.8 cm (8.2”).
All three tubings are connected together by a Swagelok® stainless steel union tee (part number SS-810-3) [57]. The end of each runner is capped using a Swagelok® stainless steel cap (part number SS-810-C) [57], which holds the ceramic preform or powder.

In order to fit the reservoir snugly into the internal surface of a pair of semi-cylindrical heaters (which will be discussed in the following section), a cylindrical sleeve is machined out of a block of carbon steel, such that its inner diameter coincides with the outer diameter of the reservoir, and its outer diameter matches the inner diameter of the heaters. The reservoir is then inserted through the steel sleeve, and a Swagelok® stainless steel male connector (part number SS-810-1-8) [57] is attached to the top end of the reservoir to keep the sleeve in place.

The complete setup of the reservoir and runner system is shown in Figure 2-2. It has a total length of 44.45 cm (17.5") and a height of 19.05 cm (7.5").

### 2.2.2 Heater System

Three pairs of heaters are used – one for the reservoir and one for each of the two runners. These heaters are RH ceramic high temperature 180° semi-cylindrical heaters from Thermcraft, Incorporated. Each heater is a resistance heater made from iron-chrome-aluminum wire helically wound and placed in a grooved ceramic refractory holder, and has the capability of being heated to a maximum temperature of 1204°C (2200°F) [58].

The heaters have an inner diameter of 3.175 cm (1.25") and an outer diameter of 5.239 cm (2.0625") [59]. The heater that is used for the reservoir is 10.16 cm (4") in length (model...
number RH212) [59] and those used for the runners are 20.32 cm (8") (model number RH214) [59]. These heaters are placed around the reservoir and runners in such a way that they completely enclose them. Figure 2-3 shows the same reservoir and runner system in Figure 2-2, but now with the heaters attached.

2.2.3 Aluminum Casing

In order to perform centrifugal casting using the setup in Figure 2-3, a casing needs to be made which functions to not only hold the entire setup in place, but also to minimize the amount of heat loss into the surroundings. The problem of heat loss is especially important in this setup because the operating temperature in infiltrating molten aluminum into a ceramic preform is at least 645°C – the liquidus of the Al-4.5%Cu alloy used for the experiment [60].

For this purpose, an outer casing is machined out of a block of A356 aluminum alloy to house the entire reservoir, runner and heater system. This casing is made up of two sections, a vertical one for the reservoir and its heaters and a horizontal one for the runners and their corresponding heaters. This horizontal casing is further divided into two portions: an upper
Figure 2-4: Schematic of vertical portion of the aluminum casing

half and a lower half.

**Vertical Casing**

The vertical portion of the aluminum casing that houses the reservoir and its pair of 4-inch heaters is simply a vertical rectangular tube 10.795 cm (4.25") in height, with flanges coming out at the bottom so that it could be attached to the top half of the horizontal casing below it. Figure 2-4 shows a schematic of this casing. This drawing, as well as all the drawings in Figures 2-5 – 2-9, is made using SolidWorks® 3D Modeling Software, distributed by SolidWorks Corporation. All dimensions in the figure are in inches.

**Horizontal Casing – Upper and Lower Halves**

The two halves that make up the horizontal portion of the aluminum casing, which houses the runners and their corresponding pair of 8-inch heaters, are long rectangular channels that measure 48.26 cm (19") in length. The top half of the casing is 4.445 cm (1.75") in height while the bottom half is 5.842 cm (2.3"). Both halves have flanges coming out at the end so that they
Figure 2-5: Schematic of upper half of the horizontal portion of the aluminum casing could be attached to each other. The upper half of the casing differs from the lower one in that it has a square hole at the center for the reservoir to pass through it. Also, the bottom half of the casing has “legs” factored into its design so that it could be attached to a custom-made adaptor that connects the whole aluminum casing to the rotor of the centrifuge. This extra pair of “legs” on the lower half of the casing accounts for its greater height compared to its upper counterpart. Figures 2-5 and 2-6 show a schematic of each of these two halves of the aluminum casing. Again, all dimensions in the figures are in inches.

The aluminum casing, when fully assembled, looks like that in Figure 2-7. The casing measures 48.26 cm (19”) in length, 21.082 cm (8.3”) in height and 11.43 cm (4.5”) in width.
Figure 2-6: Schematic of lower half of the horizontal portion of the aluminum casing

Figure 2-7: Schematic of fully assembled aluminum casing
2.2.4 Assembly of System

Once the aluminum casing is constructed, the whole system is ready for final assembly, which consists of the aluminum casing, the reservoir and runner system, as well as the three pairs of heaters. Figure 2-8 shows the exploded view of the assembly, followed by Figure 2-9, which shows the entire assembled system. Once assembled, the whole setup rotates about a vertical axis through its center, which, because symmetry has been maintained throughout the design, is also the center of mass of the system.

Not shown in Figures 2-8 and 2-9, however, is the Fiberfrax® insulating material (distributed by Eastern Refractories Company, Inc) that is packed inside the aluminum casing to minimize heat loss from the system. This Fiberfrax® layer lies between the inner surface of the casing, and the reservoir, runners and heaters. Hence, the interior of the casing is lined with
Figure 2-9: A completely assembled system

Fiberfrax®, forming a thick nest onto which the reservoir-runner-heater assembly is placed. This insulating layer of Fiberfrax® has shown to be very effective in keeping the heat inside the system.
Chapter 3

Heat Transfer Measurements and Calculations

The entire reservoir and runner system is assembled and surrounded by heaters as described in Chapter 2, and placed in a nest of Fiberfrax® insulating material in the A356 aluminum casing, with k-type thermocouples placed at strategic locations to measure the temperature distribution of the system at high temperatures. An Omega® OM-3001 Datalogger and a DT9806 data acquisition module from Data Translation, Inc. are used for this purpose.

The first experiment conducted is to heat the reservoir and runners to an internal temperature of 720°C, and then maintained at that temperature at steady state, with only air inside the system. Temperature control of the system is achieved by controlling the amount of current flowing through the heaters, using two Omega® CN76030 Temperature Controllers and one Omega® CN8201-R1 Temperature Controller – one for each pair of heaters.

The following sections discuss the results of this experiment, as well as some calculations made to deduce the heat transfer coefficients. In addition, a simple heat transfer model is created in Microsoft® Excel to calculated the temperature distribution in the runner system at steady state. In the last section of this chapter, we will look at transient heat loss in the runner system.
3.1 Steady-State Temperature Distribution in Reservoir

Figure 3-1 shows the temperature distribution within the reservoir at steady state.

Except for the top part of the reservoir, which is exposed to ambient air above it, we see a relatively uniform temperature distribution along the bottom half, in the range 0" to 4". Therefore, assuming the case of an infinite cylinder in this region (and thus reducing this to a 1-D heat transfer problem), we can calculate the convection heat transfer coefficient, $h_c$, on the outside surface of the aluminum casing. The radiation heat transfer coefficient, $h_r$, is assumed to be negligible, given the relatively low temperature of the aluminum casing and surrounding air, as well as the low emissivity of the casing.

In 1-D, the heat flux out of the internal heating layer of the heater surface is equal to the heat flux into the Fiberfrax® insulating material, and is given by:
Figure 3-2: steadystate temperature distribution in the different materials at the reservoir region

\[ q = \frac{k}{r} \left[ \frac{T_1 - T_2}{\ln \left( \frac{r_1}{r_2} \right)} \right] \]  \hspace{1cm} (3.1)

where \( k \) and \( r \) are thermal conductivity and radius, respectively, of the material under consideration, \( T_1 \) is the known temperature at the outer radius \( r_1 \), and \( T_2 \) is the known temperature at the inner radius \( r_2 \) [61].

The heater has inner and outer diameters of 1.25” and 2.0625” respectively [59]. Therefore, the Fiberfrax® insulating material also has a corresponding inner diameter of 2.065”. Its outer diameter is taken to be 3”, equal to the inner dimension of the aluminum casing at the reservoir region. The temperatures at these regions are obtained using thermocouples. Figure 3-2 illustrates the steady-state temperature distribution in the different materials.

Taking the thermal conductivities of the heater and Fiberfrax® to be 4.03 W/mK and 0.105 W/mK respectively (from Table 3.1 on page 49), it is found, from Equation 3.1, that the heat flux out of the heater, \( q_{h-out} \), is 5.22 kW/m². The heat flux into the Fiberfrax®, \( q_{f-in} \), is found to be 4.93 kW/m², which is in excellent agreement with \( q_{h-out} \) calculated.

Next, using Equation 3.1 again, we find that the heat flux out of the Fiberfrax®, \( q_{f-out} \),
is 3.39 kW/m$^2$. Assuming that the aluminum casing is sufficiently thin such that its outside temperature is equal to its inside temperature – a reasonable assumption since aluminum has a very high thermal conductivity – we can set this temperature equal to the outside temperature of the Fiberfrax\textsuperscript{®}.

However, the heat flux out of the aluminum casing, $q_{al-out}$, cannot be set equal to $q_{f-out}$ because the actual outer surface area of the casing, being rectangular, is larger than that of a cylindrical surface assumed. This results in a smaller heat flux out of the aluminum casing. Figure 3-3 illustrates this.

A square with sides 3” in length has a perimeter that is roughly 1.27 times that of a circle with a diameter of 3”. Therefore, the heat flux is correspondingly approximately 1.27 times less. Hence, we assume that $q_{al-out}$ is 2.67 kW/m$^2$.

Assuming that $h_r$ is negligible, $h_c$ can be found from the basic heat transfer equation:

$$q = h_c(T - T_{amb})$$

Given that the ambient temperature, $T_{amb}$, is 19°C, we find that $h_c$ is therefore 11 W/m$^2$K.
3.2 Steady-State Temperature Distribution in Runner System

The temperature distribution in the runner system is measured in a similar way using thermocouples placed at strategic locations. Figure 3-4 shows the temperature distribution along the inside of the runner, as well as that inside the ceramic reinforcement (aluminum oxide) at the end of the runner, and outside the runner in the Fiberfrax® region.

Figure 3-4 allows us to calculate the heat transfer coefficients in the air gap between the stainless steel runner and the interior of the heater. The predicted uniform temperature inside the runner is estimated to be 744.5°C, and this is assumed to be equal to the temperature of the stainless steel tube as well, since the internal diameter of the tube (0.37") is too small to justify a significant temperature difference between the tube wall and the air column within the tube.

The temperature of the heating layer of the heater surface facing the runner is measured to be 756.6°C at steady state. This gives an average air temperature, $T_{avg}$, of 750.55°C between
the heater and the runner. From Table B.4 of Reference [61], at 1000 K, air has a kinematic viscosity, \( \nu \), of \( 121.9 \times 10^{-6} \) m\(^2\)/s, a thermal conductivity, \( k \), of \( 66.7 \times 10^{-3} \) W/mK, and Prandtl number, \( Pr \), of 0.726. From these values, the Grashof number, \( Gr \), can be found, using:

\[
Gr = \left( \frac{l^3 g \beta \Delta T}{\nu^2} \right)
\]

(3.3)

in which \( l \) represents some characteristic length parameter and \( \Delta T \) represents the temperature difference between the heater and runner surfaces [62]. \( g \) is the acceleration due to gravity and \( \beta \) is simply the reciprocal of \( T_{avg} \), in K\(^{-1}\). For free convection around horizontal cylinders, \( l \) is based on the cylinder diameter as the characteristic length [62], which, in this case, is the outer diameter of the stainless steel runner (0.0127 m, or 0.5").

The value of \( Gr \) calculated is 16, which gives a combined \( GrPr \) value of 11.6. Referring to Fig. 9.2 in Reference [62], the Nusselt number, \( Nu \), is found to be 1.58. The convection heat transfer coefficient, \( h_c \), is given by [61]:

\[
h_c = Nu \frac{k}{l}
\]

(3.4)

which gives us \( h_c = 8 \) W/m\(^2\)K.

Due to the very high temperatures of both surfaces of the heater and the runner, which result in the stainless steel runner being highly oxidized – thus leading to a high emissivity of the runner – the radiation heat transfer coefficient is considered significant. However, this cannot be calculated straightforwardly because, unlike the value of \( h_c \), which does not change much even though the temperature along the runner changes (as shown in Figure 3-4), \( h_r \) changes significantly with different surface temperatures.

The method of calculating \( h_r \) at different regions along the runner, corresponding to the different temperatures, shall be discussed in a later section.
3.3 Modeling of Steady-State Heat Transfer Calculations by Finite Differences Using Microsoft Excel

While obtaining accurate measurements of the temperature distributions in and around the reservoir and runners are important, this is only an isolated case given the particular design of the system. In general, we might be interested to know what the temperature distributions would be like if we were to change the design parameters or the operating parameters to conditions where no experimental measurements were possible. For example, this includes cases where the diameter of the runner is doubled, thus increasing by 4 times the volume of ceramic sample to be infiltrated; or where the thickness of the insulating layer of Fiberfrax® around the heater were changed; or where we have molten iron (instead of aluminum) in a ceramic runner. It might also be of interest to know what temperature we should heat the heaters to in order to achieve the temperature distributions desired in the runners. As it is impractical to conduct numerous experiments to find out the answer to each of the above questions, and the complexity of the design setup also precludes the use of analytical methods to predict temperature distributions, a simple heat transfer model of the runner system is therefore developed using a spreadsheet in Microsoft® Excel to calculate the temperature distributions, by the method of finite differences.

3.3.1 Finite Difference Method to Calculate Heat Transfer

The use of finite differences to solve heat transfer problems is a fairly straightforward procedure for most situations, and this topic has been extensively covered by numerous authors who have written books on heat transfer [61, 62, 63, 64, 65, 66, 67]. This section presents a brief discussion on the principles of using the finite difference method as applied to a spreadsheet in Microsoft® Excel.

The basic principle of the numerical approach to a heat conduction problem is the replacement of the differential equation for the continuous temperature distribution in a heat-conducting solid by a finite-difference equation which must be satisfied at only certain points in the solid [61]. An Excel spreadsheet, where each cell in the spreadsheet represents a “node”, or point, in the system, is therefore an ideal tool for such calculations. The temperature at that
node is determined from the equations that relate it to its neighboring nodes [68].

**Nodal Mesh**

The first step in any numerical analysis is to select the points where the temperature shall be determined. A mesh is defined that covers the region where temperature predictions are of interest. This mesh consists of small regions of orthogonal geometry. Figure 3-5 shows an example of a mesh.

In Figure 3-5, a 2-D mesh displays interior nodes and three types of boundary cells. Temperatures are assumed to be located at the center of interior cells. In order to have temperature information directly on a boundary, boundary cells are either three-quarters, one-half, or one-quarter the size of interior cells.

In order to achieve better accuracy in the calculations, a cylindrical coordinate system is used for the 2-D mesh in Figure 3-5, where the z-dimension represents the longitudinal direction in the runner system and the r-dimension represents the radial direction. Each cell denotes a ring element in the mesh. Details of this ring element shall be discussed in the following section.

Although cylindrical coordinates are used, it should be noted that the aluminum casing at the runner region is not cylindrical in shape, but rectangular, as shown in Figure 2-9. There-
fore, a cylindrical approximation may result in a small loss of accuracy in the calculations. However, given that: 1) all other components of the runner system are nonetheless cylindrical; 2) the layer of Fiberfrax insulating material provides such a good insulation to the runner system that the shape of the casing becomes less important; and 3) we are most interested in the temperature distribution in the runner system enclosed within the Fiberfrax, this small sacrifice in accuracy is not considered crucial. In addition, a cylindrical approximation is also closer to the actual rectangular shape of the casing than is a 2-D mesh of infinite length into the paper (if Cartesian coordinates are used).

With the nodal mesh thus defined, an Excel spreadsheet is used to calculate the temperature distributions along the runner system at steady state. The mesh that is set up for this purpose is shown in both Figures 3-6 and 3-7. Figure 3-6 shows the mesh as it is, with the different materials clearly defined in separate regions and uncluttered with labels. Figure 3-7 shows the same mesh, but with labels added for additional explanation.

Each cell in the spreadsheet represents a node where the temperature at that node will be computed. It should be noted that, as explained above, each node is at the center of each cell. And since the mesh is arranged such that nodes lie directly on a boundary, all boundaries therefore lie at the center of the cells. Hence, the different colors depicting the different materials in Figures 3-6 and 3-7 are not coincidental with the material boundaries.

**Interior Cells**

The governing differential equation for interior cells is obtained by solving for the temperature, \( T \), of that cell from the finite difference form of the thermal energy equation. For steady-state, 2-D conduction with uniform thermal conductivity, \( k \), the thermal energy equation is simply:

\[
0 = \frac{\partial^2 T}{\partial r^2} + \frac{\partial^2 T}{\partial z^2}
\]  

(3.5)

The area through which heat flows is very different in the radial \((r)\) direction compared to that in the longitudinal \((z)\) direction. Figure 3-8 illustrates this difference.

The shaded area in the diagram on the right in Figure 3-8, which is the cross section of a ring element, represents a typical interior cell in Figure 3-5. Each ring element has a volume equal to
Figure 3-6: Nodal mesh set up in Microsoft® Excel
Figure 3-7: Nodal mesh set up in Microsoft® Excel (with labels – dotted lines represent physical boundaries)
Figure 3-8: Radial ($r$) and longitudinal ($z$) heat transfer through a typical ring element

Figure 3-9: An interior cell surrounded by neighboring cells

$2\pi r\Delta r\Delta z$. The cross-sectional areas in the $r$-direction are $2\pi \left( r + \frac{\Delta r}{2} \right) \Delta z$ and $2\pi \left( r - \frac{\Delta r}{2} \right) \Delta z$, and that in the $z$-direction is $2\pi r\Delta r$. Symmetry is assumed in the $\theta$ direction.

Suppose each interior cell, labeled “o”, is surrounded by cells “N”, “S”, “E” and “W” as shown in Figure 3-9. (This is also represented by the 5 black cells in Figure 3-7, with cell “o” being the black cell at the center.) If cell “o” is a ring element with radius $r$ and at position $z$ from the origin, then the neighboring “N” and “S” cells are ring elements at the same $z$ position with radii $(r+\Delta r)$ and $(r-\Delta r)$ respectively. The “E” and “W” cells, on the other hand, are ring elements at the $(z+\Delta z)$ and $(z-\Delta z)$ positions respectively, but each with the same radius $r$. Diagonal neighbors do not affect a cell. This is because there is no area between the cell and its diagonal neighbors for energy transfer.

An energy balance must be applied to each cell in order to determine an expression for the temperature of that cell. Only the simple “specified temperature” boundary condition does not
require an energy balance. Energy balances on cells are derived by using simple one-dimensional heat transfer relations between a cell and its neighbors. The steady-state energy balance on a typical interior cell is:

\[ 0 = Q_N + Q_S + Q_E + Q_W \]  

(3.6)

The finite difference forms of the individual heat transfers are:

\[
Q_N = 2k\pi (r + \Delta r/2) \Delta z \left( \frac{T_N - T_o}{\Delta r} \right)
\]

\[
Q_S = 2k\pi (r - \Delta r/2) \Delta z \left( \frac{T_S - T_o}{\Delta r} \right)
\]

\[
Q_E = 2k\pi \Delta r \left( \frac{T_E - T_o}{\Delta z} \right)
\]

\[
Q_W = 2k\pi \Delta r \left( \frac{T_w - T_o}{\Delta z} \right)
\]

(3.7)

Solving for the temperature in cell “o”, we obtain:

\[
T_o = \frac{\Delta z^2 \left[ (r + \Delta r/2)T_N + (r - \Delta r/2)T_S + r\Delta r^2(T_E + T_w) \right]}{2r(\Delta z^2 + \Delta r^2)}
\]

(3.8)

Since each cell, with temperature \( T_o \), needs to refer to its neighboring cells in order to solve for its own temperature value, a circular reference is created. Excel can solve large systems of simultaneous equations like this (as well as the equations discussed in the following sections) using its iteration function.

**Boundary Cells**

The complexity in modeling the experimental setup is in establishing the energy balances for elements at the boundaries of the mesh. Several different types of boundaries are defined for this model, and these are individually discussed in the following sections.

**Left Boundary** The left boundary is assumed to be an “insulated” boundary where the temperature gradient at that point is zero. This is based on the assumption that the temperature
distribution along the runner at a certain distance from its end – taken to be 4” from the end of the heater – can be considered to be uniform. Looking at the temperature distribution in Figure 3-4, this assumption is deemed reasonable. Hence, all cells to the left of the insulated boundary are set equal to the cells directly to their right across the boundary line (see Figure 3-7).

**Top Boundary** As illustrated in Figure 3-5, cells at the top boundary of the mesh in Figures 3-6 and 3-7 are one-half the height of an interior cell. Assuming that a heat transfer coefficient, $h$, connects the boundary cell to its surroundings, and that conduction transfers energy from the other surrounding cells, the steady state energy balance on the cell becomes:

$$0 = Q_{amb} + Q_s + Q_e + Q_w$$

(3.9)

The individual heat transfers are:

$$Q_{amb} = 2h \pi r \Delta z (T_N - T_o)$$

$$Q_s = 2k \pi \left( r - \frac{\Delta r}{2} \right) \Delta z \left( T_s - T_o / \Delta r \right)$$

$$Q_e = k \pi \left( r - \frac{\Delta r}{4} \right) \Delta r \left( T_e - T_o / \Delta z \right)$$

$$Q_w = k \pi \left( r - \frac{\Delta r}{4} \right) \Delta r \left( T_w - T_o / \Delta z \right)$$

(3.10)

These relations can be solved for the boundary cell temperature, $T_o$:

$$T_o = \frac{2r \Delta r \Delta z^2 h T_N + k \Delta z^2 \left[ 2 \left( r - \frac{\Delta r}{2} \right) T_s + \Delta r^2 \left( r - \frac{\Delta r}{4} \right) (T_e + T_w) \right]}{2r \Delta r \Delta z^2 + k \left[ \Delta z^2 \left( r - \frac{\Delta r}{2} \right) + \Delta r^2 \left( r - \frac{\Delta r}{4} \right) \right]}$$

(3.11)

**Right Boundary** As with the top boundary, cells on the right boundary of the mesh are one-half the width of an interior cell. The steady state energy balance on each cell in this case is:

$$0 = Q_N + Q_s + Q_w + Q_{amb}$$

(3.12)
The individual relations for the heat transfer terms are:

\[ Q_N = k\pi \left( r + \Delta r/2 \right) \Delta z \left( T_N - T_o/\Delta r \right) \]
\[ Q_s = k\pi \left( r - \Delta r/2 \right) \Delta z \left( T_s - T_o/\Delta r \right) \]
\[ Q_w = 2k\pi \Delta r \left( T_w - T_o/\Delta z \right) \]
\[ Q_{amb} = 2h\pi \Delta r \left( T_E - T_o \right) \]  \hspace{1cm} (3.13)

As before, these relations can be solved for the cell temperature, \( T_o \):

\[ T_o = \frac{k\Delta z^2 \left[ \left( r + \Delta r/2 \right) T_N + \left( r - \Delta r/2 \right) T_s \right] + 2r\Delta r^2 \left( kT_w + h\Delta z T_E \right)}{2r\left( k\Delta z^2 + \Delta r^2 \left( k + h\Delta z \right) \right)} \]  \hspace{1cm} (3.14)

The cell at the top-right hand corner of the mesh in Figures 3-6 and 3-7 requires special attention because, as explained in a previous section, it is only one-quarter the size of an interior cell. The areas for heat transfer are reduced because of the cell size. The energy terms for the balance on this cell are simply a combination of Equations 3.10 and 3.13, and the cell temperature \( T_o \) can be easily deduced from the relations.

**Bottom Boundary** The bottom boundary is given a symmetry border, represented by the horizontal centerline in Figure 3-7. This centerline corresponds to the concentric center of the runner, heater and aluminum casing. Hence, each cell in the row directly below this centerline is actually part of the ring element represented by the cell directly above the line. The values in this row of cells are therefore set equal to those in the row of cells directly above the centerline.

At the centerline itself, the elements are no longer rings, but become solid cylinders instead. A typical element at the centerline is illustrated in Figure 3-10.

The steady-state energy balance on the centerline element is:

\[ 0 = Q_N + Q_E + Q_w \]  \hspace{1cm} (3.15)

The energy terms for the balance are:
\[ Q_N = k \pi \Delta z (T_N - T_o) \]
\[ Q_E = \frac{k \pi \Delta r^2}{4 \Delta z} (T_E - T_o) \]
\[ Q_W = \frac{k \pi \Delta r^2}{4 \Delta z} (T_W - T_o) \]  

Solving for \( T_o \) yields the relation needed for the centerline boundary cell:

\[ T_o = \frac{4 \Delta z^2 T_N + \Delta r^2 (T_W + T_E)}{2 (2 \Delta z^2 + \Delta r^2)} \]  

**Material Boundaries**  Probably the most difficult and tedious equations involve heat flow through composite materials. This heat transfer can be via conduction, convection or radiation, or a combination of the three methods. For the sake of simplicity, the following discussion will focus on steady-state heat conduction through two materials. The equations for convection and radiation through multiple materials (e.g. three different materials contained in a single cell) can be deduced in a similar fashion.

Figure 3-11 illustrates the case where two materials are separated by a horizontal boundary.

As shown in Figure 3-11, due to the different material properties to be considered in the heat transfer equations, the neighboring cells on the left and right sides of cell “o” will each have to be divided into two separate cells, forming cells “W₁” and “W₂”, and cells “E₁” and “E₂” respectively. These split cells each has a height that is one-half the height of a normal
The steady-state energy balance on cell “o” becomes:

\[ 0 = Q_N + Q_S + Q_{E_1} + Q_{E_2} + Q_{W_1} + Q_{W_2} \]  \hspace{1cm} (3.18)

The individual heat transfers are:

\[ Q_N = 2k_1 \pi \Delta z (r + \Delta r / 2) \left( \frac{T_N - T_o}{\Delta r} \right) \]
\[ Q_S = 2k_2 \pi \Delta z (r - \Delta r / 2) \left( \frac{T_S - T_o}{\Delta r} \right) \]
\[ Q_{E_1} = k_1 \pi \Delta r \left( r + \Delta r / 4 \right) \left( \frac{T_E - T_o}{\Delta z} \right) \]
\[ Q_{E_2} = k_2 \pi \Delta r \left( r - \Delta r / 4 \right) \left( \frac{T_E - T_o}{\Delta z} \right) \]
\[ Q_{W_1} = k_1 \pi \Delta r \left( r + \Delta r / 4 \right) \left( \frac{T_W - T_o}{\Delta z} \right) \]
\[ Q_{W_2} = k_2 \pi \Delta r \left( r - \Delta r / 4 \right) \left( \frac{T_W - T_o}{\Delta z} \right) \]  \hspace{1cm} (3.19)

These relations can be solved for the boundary cell temperature, \( T_o \):
Using the same principals discussed above, we can similarly establish the heat transfer equations for the case where two materials are separated by a vertical boundary (with material 1 on the left and material 2 on the right). In this case, the “N” and “S” cells will be split into two instead, with each half-cell having half the width of a normal interior cell. Likewise, cells “N1” and “N2”, and cells “S1” and “S2” will be formed respectively, and the steady-state energy balance on cell “o” becomes:

\[ 0 = Q_{N_1} + Q_{N_2} + Q_{S_1} + Q_{S_2} + Q_E + Q_w \]  

(3.21)

The individual relations for the heat transfer terms are:

\[
\begin{align*}
Q_{N_1} &= k_1 \pi A \Delta z \left( r + \Delta r/2 \right) \frac{T_N - T_o}{\Delta r} \\
Q_{N_2} &= k_2 \pi A \Delta z \left( r - \Delta r/2 \right) \frac{T_N - T_o}{\Delta r} \\
Q_{S_1} &= k_1 \pi A \Delta z \left( r - \Delta r/2 \right) \frac{T_S - T_o}{\Delta r} \\
Q_{S_2} &= k_2 \pi A \Delta z \left( r + \Delta r/2 \right) \frac{T_S - T_o}{\Delta r} \\
Q_E &= 2 k_2 \pi A r \left( T_E - T_o / \Delta z \right) \\
Q_w &= 2 k_1 \pi A r \left( T_w - T_o / \Delta z \right)
\end{align*}
\]  

(3.22)

\[ T_o \] then becomes:

\[
T_o = \frac{(k_1 + k_2) \Delta z^2 \left[ r + \Delta r/2 \right] T_N + \left( r - \Delta r/2 \right) T_S + 2r \Delta r^2 \left( k_1 T_w + k_2 T_E \right)}{2r (k_1 + k_2) \left( \Delta z^2 + \Delta r^2 \right)}
\]  

(3.23)

As mentioned above, the value of \[ T_o \] for a cell that is exposed to convection and radiation
Table 3.1: Thermal conductivities of the materials used in the Excel model and at the boundary of more than two materials can be deduced in a similar fashion.

**Heating Layer Boundary on Ceramic Heater**  In addition to the boundaries discussed above, one other boundary condition is given in the model – the temperature distribution in the heating layer of the heater surface (Figure 3-7). One way to obtain this distribution is to conduct an experiment to measure it. However, in the event that an experiment is not, or cannot be, conducted, the following can be done:

To set the temperature distribution, the temperature in the cell that lies on the heating layer at a distance of 3.6” away from the end of the heater (marked by a • in Figure 3-7 – referred to as the *reference cell*) is first adjusted such that the temperature in the stainless steel runner directly below this cell reaches the desired temperature (e.g. 740°C for molten aluminum). As a first approximation, the cell in the heating layer at the end of the heater surface (also marked and referred to as the *end cell* in Figure 3-7) is set to about 80% of the temperature in the *reference cell*. The temperature distribution along the heater is then set as an average between the *reference* and *end cells*. This method of setting the temperature distribution along the heating layer of the heater surface has been found to produce very good results from the model, which shall be discussed in a later section.

**Material Properties and Other Numerical Values Used in Calculations**

**Material Properties**  Because different materials have different properties at different temperatures, most of the values used in the calculations are considered approximations only. Tables 3.1 and 3.2 list the thermal conductivities and emissivities, respectively, of the materials used in the simulations.

<table>
<thead>
<tr>
<th>Material</th>
<th>$k$ (W/mK)</th>
<th>Temperature (°C)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>0.04360 – 0.12400</td>
<td>277 – 1727</td>
<td>[64]</td>
</tr>
<tr>
<td>Aluminum 356</td>
<td>189.0 – 86.9</td>
<td>50 – 700</td>
<td>[69]</td>
</tr>
<tr>
<td>Aluminum Oxide</td>
<td>11.5 – 5.08</td>
<td>260 – 1600</td>
<td>[70]</td>
</tr>
<tr>
<td>Fiberfrax®</td>
<td>0.046233 – 0.249144</td>
<td>204 – 982</td>
<td>[71]</td>
</tr>
<tr>
<td>Mullite (Heater)</td>
<td>5.43 – 3.76</td>
<td>200 – 800</td>
<td>[72]</td>
</tr>
<tr>
<td>AISI 316 Stainless Steel</td>
<td>15.2 – 24.2</td>
<td>127 – 727</td>
<td>[65]</td>
</tr>
<tr>
<td>Molten Iron</td>
<td>42.6</td>
<td>1727</td>
<td>[61]</td>
</tr>
</tbody>
</table>
## Emissivities of the materials used in the Excel model

<table>
<thead>
<tr>
<th>Material</th>
<th>Emissivity, $\varepsilon$</th>
<th>Temperature (°C)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum 356</td>
<td>0.05</td>
<td>204 – 316</td>
<td>[62]</td>
</tr>
<tr>
<td>AISI 316 Stainless Steel</td>
<td>0.9</td>
<td>727 – 927</td>
<td>[65]</td>
</tr>
</tbody>
</table>

In Table 3.1, the aluminum oxide powder used as the ceramic reinforcement in the runner is assumed to have a volume fraction of about 50%. Therefore, the value of $k$ used in the model is the average of the $k$-values of both air and aluminum oxide.

### Convection and Radiation Heat Transfer Coefficients

Three separate regions in the model require the use of convection and radiation heat transfer coefficients in the calculations, corresponding to air: 1) in the ambient surroundings, 2) between the heater and runner, and 3) within the runner (only for the case of a runner filled with air instead of molten metal). For the sake of simplicity, these heat transfer coefficients shall be referred to as $h$, which is merely the sum of both the convection and radiation coefficients ($h_c$ and $h_r$, respectively).

1. The value of $h_c$ in the ambient surroundings has been determined in a previous section, and is found to be approximately 11 W/m²K. The value of $h_r$ changes with the temperature of the aluminum casing, $T_{AI}$, and is given by:

   \[ h_r = \varepsilon \sigma (T_{AI}^2 + T_{\text{amb}}^2) (T_{AI} + T_{\text{amb}}) \]  

   (3.24)

   where $\varepsilon$ is the emissivity of the aluminum casing (Table 3.1), and $\sigma$ is the Stefan-Boltzmann Constant, which is equal to $5.67 \times 10^{-8}$ W/m²K⁴. In order to track the value of $h_r$ as it changes with $T_{AI}$, an extra row of cells is created in addition to the mesh in Figures 3-6 and 3-7 to record its values.

   In determining the value of $h_c$ in a previous section, it has been assumed that radiation is negligible, and hence the value of $h$ calculated is solely for convection. From the Excel model set up, it is found that this assumption is indeed valid. This is because even at very high temperatures of $>700^\circ$C within the runner, the value of $h_r$ (as calculated from the Excel model) is only about 0.7 W/m²K, or 6% of the value of $h$ calculated.

2. The value of $h_c$ in the heater and runner has been determined in a previous section, and
is found to be approximately 8 W/m²K. The value of $h_r$ is found in a similar way to 1) above, replacing $T_{Al}$ with the temperature of the heating layer of the heater surface ($T_{he}$) and $T_{amb}$ with the surface temperature of the stainless steel runner ($T_{stl}$) in Equation 3.24.

It is found that the value of $h_r$ between the heater and runner plays an important role in heat transfer. For example, at very high temperatures of >700°C within the runner, the range of values of $h_r$ can be 20 – 30 times that of $h_c$.

3. For the value of $h$ within the runner, it is first assumed that if the runner is filled with molten metal, and not air, then its value is effectively at infinity (giving a zero thermal resistance), since conduction plays a much bigger role in heat transfer than convection or radiation does in molten metals. If the runner is filled with air, it is then assumed that radiation is negligible. The value of $h_c$ calculated in 2) above is then used.

**Determination of $\Delta r$ and $\Delta z$** In order to retain as much as possible the actual dimensions of each component in the runner system (i.e. heater, runner, aluminum casing, etc.), three different values of $\Delta r$ and three different values of $\Delta z$ are used in different regions of the mesh. These different values of $\Delta r$ and $\Delta z$ can be clearly seen in the mesh in Figure 3-6, which is drawn to scale.

Within the heater (comprising the runner and the air gap between it and the heater), $\Delta r$ is set at 0.165 cm (0.065”). The heater and Fiberfrax® layer above it has a $\Delta r$ value of 0.206 cm (0.08125”). Finally, the outermost aluminum casing is given a $\Delta r$ value of 0.635 cm (0.25”).

From the insulated left boundary to a distance of 2” away from the end of the heater, $\Delta z$ is set at 2.03 cm (0.8”). Towards the end of the heater and runner, however, $\Delta z$ is significantly reduced to 0.25 cm (0.1”), because the temperature distribution in this region is of much interest. Finally, the outermost aluminum casing is given a $\Delta z$ value of 1.27 cm (0.5”).

Since the values of $\Delta r$ and $\Delta z$ are not uniform along the mesh, these differences need to be taken into account when determining the heat transfer terms across these regions. For example, consider the illustration in Figure 3-12, where cell “o” lies at the boundary of two different materials. The cells on its left have dimensions $\Delta z_1$ in the z-direction, while those on its right have dimensions $\Delta z_2$. Similarly, the dimensions above and below it in the r-direction
Figure 3-12: A cell “o” at the boundary of two different materials, across different $\Delta r$ and $\Delta z$ values are $\Delta r_1$ and $\Delta r_2$ respectively.

The steady-state energy balance on cell “o” becomes:

$$0 = Q_{N_1} + Q_{N_2} + Q_{S_1} + Q_{S_2} + Q_{E_1} + Q_{E_2} + Q_{W_1} + Q_{W_2}$$  (3.25)

Taking into account the different material properties as well as the different dimensions involved, the heat transfer equations become:
\[ Q_{N_1} = k_1 r A \Delta z_1 \left( r + \frac{\Delta r_1}{2} \right) \left( T_N - T_o \right) / \Delta r_1 \]
\[ Q_{N_2} = k_1 r A \Delta z_2 \left( r + \frac{\Delta r_1}{2} \right) \left( T_N - T_o \right) / \Delta r_1 \]
\[ Q_{S_1} = k_2 r A \Delta z_1 \left( r - \frac{\Delta r_2}{2} \right) \left( T_S - T_o \right) / \Delta r_2 \]
\[ Q_{S_2} = k_2 r A \Delta z_2 \left( r - \frac{\Delta r_2}{2} \right) \left( T_S - T_o \right) / \Delta r_2 \]
\[ Q_{E_1} = k_1 r A \Delta r_1 \left( r + \frac{\Delta r_1}{4} \right) \left( T_E - T_o \right) / \Delta z_2 \]
\[ Q_{E_2} = k_2 r A \Delta r_2 \left( r - \frac{\Delta r_2}{4} \right) \left( T_E - T_o \right) / \Delta z_2 \]
\[ Q_{W_1} = k_1 r A \Delta r_1 \left( r + \frac{\Delta r_1}{4} \right) \left( T_W - T_o \right) / \Delta z_1 \]
\[ Q_{W_2} = k_2 r A \Delta r_2 \left( r - \frac{\Delta r_2}{4} \right) \left( T_W - T_o \right) / \Delta z_1 \]

Solving for \( T_o \) yields:

\[ T_o = \frac{\left( \Delta z_1 + \Delta z_2 \right) \left[ k_1 \left( r + \frac{\Delta r_1}{2} \right) \left( T_N - T_o \right) / \Delta r_1 \right] + \left[ k_1 \Delta r_1 \left( r + \frac{\Delta r_1}{4} \right) + k_2 \Delta r_2 \left( r - \frac{\Delta r_2}{2} \right) \right] \left( T_E - T_o / \Delta z_2 \right)} {\left( \Delta z_1 + \Delta z_2 \right) \left[ k_1 \Delta r_1 \left( r + \frac{\Delta r_1}{2} \right) + k_2 \Delta r_2 \left( r - \frac{\Delta r_2}{2} \right) \right] + \left[ k_1 \Delta r_1 \left( r + \frac{\Delta r_1}{4} \right) + k_2 \Delta r_2 \left( r - \frac{\Delta r_2}{4} \right) \right] / \Delta z_1 + 1 / \Delta z_2} \]

The value of \( T_o \) for other types of cells can be similarly determined. Great care has to be taken to ensure that \( \Delta z_1, \Delta z_2 \) and \( \Delta r_1, \Delta r_2 \) are properly accounted for in the calculations.

### 3.3.2 Results of Modeling

The model created in Excel is run through many iterations until a steady-state temperature profile is reached. The sections below discuss the results of these runs. The results of the first
Position (Figures 3-13 and 3-14) | Measured (°C) | Calculated (°C) | % Error
--- | --- | --- | ---
1 | 228 | 223 | 2.2%
2 | 727 | 730 | 0.4%
3 | 756.6 | 756.6 | N.A.
4 | 744.5 | 741.1 | 0.5%
5 | 720.9 | 715.6 | 0.7%
6 | 640 | 637 | 0.5%
7 | 630 | 630 | 0.0%
8 | 620 | 621 | 0.1%
9 | 614.0 | 614.0 | N.A.
10 | 521 | 516 | 1.0%
11 | 214 | 219 | 2.2%

Table 3.3: Comparison of calculated results with measured results (for air at > 700°C inside the runners)

three runs are compared with actual experimental data to validate the accuracy of the model. The final four runs show the expected results without any experiment.

**Heating Setup to >700°C with Air Inside Runners**

This first experiment has already been discussed in a previous section, where the entire setup is heated to >700°C with just air inside the reservoir and runners. Using data from the experiment, we know that the temperature of the cell on the heating layer of the heater surface at a distance of 3.6” away from the end of the heater (the *reference cell*) is 756.6°C, and that of the cell at the end of the heater (the *end cell*) is 614.0°C. These measured values are then entered into the appropriate cells in the model, which is made to run through many iterations until a steady-state temperature profile is reached. After adjusting for possible errors and inaccuracies in the positioning of the thermocouples in the experiment, the results obtained from the model are compared with those obtained from the actual experiment. The results are found to agree very well with each other. These are summarized in Table 3.3 and Figures 3-13 and 3-14.

The percentage errors for positions 3 and 9 in Table 3.3 are not applicable (N.A.) because the temperatures at these positions have been manually set to be equal to the measured temperatures, giving the model a fixed boundary condition at the heating layer of the heater surface. Also, it can be seen from the table that the largest percentage errors occur in positions 1 and 11 because of the inaccuracies that result from approximating a rectangular aluminum casing.
Figure 3-13: Approximate positions of the thermocouples in the runner system to a cylindrical one.

Figure 3-14 shows that the agreement between the simulation results and the experimental data is excellent.

**Heating Setup to 300°C with Air Inside Runners**

In the second experiment, which is very similar to the first one, the entire setup is heated to only 300°C with air inside the reservoir and runners. All the thermocouples used remain in exactly the same positions as in the first experiment (refer to Figure 3-13). The main purpose of this experiment is simply to validate the accuracy of the Excel model developed. Therefore, the only things that are changed in the model are the temperature values used, as well as the thermal conductivities of some of the materials (refer to Table 3.1), thus taking into account changes in the values of $k$ with temperature.

The convection heat transfer coefficient of the air in the surroundings is assumed unchanged, and its radiation heat transfer coefficient is calculated in the same way as described before, using a row of cells outside the model to calculate the value of $h_r$ at each point by relating the temperatures of the surrounding air and the surface of the aluminum casing at that point.
Figure 3-14: A comparison of measured and calculated temperature distribution along the runner at steady state (for air at $>700^\circ$C inside the runners)
Table 3.4: Comparison of calculated results with measured results (for air at 300°C inside the runners)

This same method is used to calculate the values of $h_r$ between the inside surface of the heater and the runner. The value of $h_c$ in this region is calculated using Equations 3.3 and 3.4, and is found to be approximately 8 W/m²K as well.

Table 3.4 and Figure 3-15 summarize the results of the second experiment and simulation. They are found to agree very well with each other.

Again, the percentage errors at positions 3 and 9 are not applicable (N.A.). Also, it can be seen that the largest percentage errors occur at positions 1 and 11, which measure the temperatures between the inside surface of the aluminum casing and the outside surface of the Fiberfrax® insulating layer. The inaccuracy resulting from approximating a rectangular aluminum casing to a cylindrical one is more pronounced here.

**Heating Setup to >700°C with Molten Al-4.5%Cu Alloy Inside Runners**

The third and final experiment is conducted to find out the temperature distribution in the runner system with molten Al-4.5%Cu alloy inside the runner. Again, all the thermocouples remain at exactly the same positions as before, and the temperatures at these locations are measured at steady state.

For the Excel model, the conditions are the same as those in the first experiment, with only a few changes, and these are discussed below:

1. The temperature distribution along the heating layer of the heater surface is changed to
Figure 3-15: A comparison of measured and calculated temperature distribution along the runner at steady state (for air at 300°C inside the runners)
reflect that obtained from actual experiment. It is found from the experiment that, in
order to heat the molten aluminum alloy inside the runner to roughly the same temper-
ature as having air in the runner, the heater needs to be heated up to a slightly higher
temperature than before.

2. The thermal conductivity of the material inside the runner is changed to indicate that
we now have molten metal instead of air in the runner. From Reference [69], the thermal
conductivity of molten A356 aluminum alloy is found to be about 80 W/mK. Assuming
that the thermal conductivity of molten Al-4.5%Cu alloy is approximately equal to that
of molten A356 aluminum alloy at the same temperature, this value is used in the Excel
model. It turns out that this approximation produces very good results that agree well
with the experiment.

3. The value of $h_c$ in the surroundings is assumed unchanged, and the values of $h_r$ in both
the surroundings and between the inside surface of the heater and outside surface of the
runner are calculated in the same way as described above. What is changed, however, is
the value of $h_c$ for the air gap in that region, which is found from Equations 3.3 and 3.4
to be approximately 10 W/m²K.

4. As mentioned before, because we now have molten metal instead of air inside the runner,
heat conduction is assumed to be the main mode of heat transfer. Therefore, both the
convection and radiation heat transfer coefficients are ignored in this simulation.

After making the appropriate changes, the model is run through many iterations until a
steady-state temperature profile is reached. Table 3.5 and Figure 3-16 summarize the findings.

As shown in both Table 3.5 and Figure 3-16, the results of the simulation agree very well
with the experimental data. Also shown in Figure 3-16 is a comparison of the simulation and
experimental data obtained from the third experiment with those from the first experiment
(with air instead of molten metal inside the runner). There is clearly an increase in the temper-
ature of the ceramic preform (points 6, 7 and 8 in Figure 3-16) with molten Al-4.5%Cu alloy in
the runner. This is expected since molten aluminum alloy is a much better thermal conductor
than air, thus conducting more heat into the ceramic preform. The larger thermal conductivity
of the molten aluminum alloy also allows the temperature distribution inside the runner to stay
Table 3.5: Comparison of calculated results with measured results (for molten Al-4.5%Cu alloy inside the runners)

<table>
<thead>
<tr>
<th>Position (Figures 3-13 and 3-16)</th>
<th>Measured (°C)</th>
<th>Calculated (°C)</th>
<th>% Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>230</td>
<td>228</td>
<td>0.7%</td>
</tr>
<tr>
<td>2</td>
<td>732</td>
<td>747</td>
<td>2.1%</td>
</tr>
<tr>
<td>3</td>
<td>774.5</td>
<td>774.5</td>
<td>N.A.</td>
</tr>
<tr>
<td>4</td>
<td>734.5</td>
<td>730.0</td>
<td>0.6%</td>
</tr>
<tr>
<td>5</td>
<td>719.7</td>
<td>714.5</td>
<td>0.7%</td>
</tr>
<tr>
<td>6</td>
<td>671</td>
<td>681</td>
<td>1.5%</td>
</tr>
<tr>
<td>7</td>
<td>661</td>
<td>662</td>
<td>0.1%</td>
</tr>
<tr>
<td>8</td>
<td>651</td>
<td>650</td>
<td>0.1%</td>
</tr>
<tr>
<td>9</td>
<td>628.5</td>
<td>628.5</td>
<td>N.A.</td>
</tr>
<tr>
<td>10</td>
<td>530</td>
<td>539</td>
<td>1.6%</td>
</tr>
<tr>
<td>11</td>
<td>222</td>
<td>224</td>
<td>0.9%</td>
</tr>
</tbody>
</table>

Figure 3-16: A comparison of measured and calculated temperature distribution along the runner at steady state (for molten Al-4.5%Cu alloy inside the runners)
somewhat more uniform than what we would get from having air inside the runner. Therefore, the slope with molten metal is less steep in Figure 3-16 than that with air in the runner.

Once the above three experiments have been performed, and their results matched by the simulation model developed in Excel – thus confirming the validity of the model – the following four cases are explored to predict the temperature distribution in each case without running actual experiments.

**Optimum Heater Temperature**

It is found, in Chapter 4, that it will take about 35 seconds for the IEC CRU-5000 centrifuge used for the experiments to reach a desired speed of 1500 RPM with the reservoir and runner system used. Assuming a safety factor of 5 seconds, it has to be ensured therefore that the Al-4.5%Cu alloy inside the runner stays molten for at least 40 seconds. An additional assumption made here is that the cooling rate of the molten aluminum alloy inside the runner remains the same whether the centrifuge is spinning or stationary. This assumption is considered acceptable given that an increase in the convection heat transfer coefficient, \( h_c \), of the surrounding air when the system is spinning will probably not significantly affect the amount of heat loss into the surroundings because of the extremely low thermal conductivity and high specific heat capacity [71] of the layer of Fiberfrax® insulating material surrounding the heaters.

To look at the cooling rate of the runner system in a stationary position, refer to Figure 3-22. From the figure, it can be seen that the temperature of the molten Al-4.5%Cu alloy at the interface between the metal and the ceramic preform (point 6 in the figure) needs to be at least 652°C if we want its temperature to reach 645°C (the liquidus of the alloy [60]) after 40 seconds, so that infiltration of the preform occurs at the liquidus of the metal.

Using the Excel model developed, it is found that the reference cell on the heating layer of the heater surface needs to be set at 743°C for the metal-ceramic interface temperature to be at 652°C. This gives the expected temperature distribution in the runner system as shown in Figure 3-17. The temperature distribution obtained from the third experiment described in the previous section (where the reference cell is set at 774.5°C) is included for comparison.
Figure 3-17: Calculated (expected) optimum temperature distribution along the runner at steady state.
Figure 3-18: Calculated (expected) temperature distribution along the runner at steady state, with runner diameter and heater inner diameter doubled

**Doubling of Runner Diameter**

The next stage in the analysis is to look at how the temperature distribution in the runner system would change if the runner diameter (and hence the corresponding inner diameter of the heater) were doubled.

Changes are made to the dimensions of the model to reflect the situation described, and the temperature value in the *reference cell* is adjusted until a steady-state temperature of 652°C is once again reached at the metal-ceramic interface.

It is found that a doubling of the runner diameter and heater inner diameter does not require a significant increase in the *reference cell* temperature of the heater. The temperature of the *reference cell* is raised only to 745°C in this case (compared to 743°C with the original dimensions). Figure 3-18 shows the expected temperature distribution in the runner system with the runner diameter and heater inner diameter doubled. The optimum temperature distribution obtained in the previous section (without doubling the diameters) is included for comparison.
Figure 3-19: Temperature distribution in the ceramic preform with the runner diameter and heater inner diameter doubled, compared with that of original setup

A very interesting observation can be made in Figure 3-18. It is apparent, from the figure, that with an increase in the runner diameter (and hence in the heater inner diameter), the corresponding increase in volumes of molten Al-4.5%Cu alloy inside the runner and the air gap between the heater inner surface and the runner has allowed a more uniform temperature distribution of molten metal along the runner, as well as a slight improvement in the uniformity of temperature along the ceramic preform. The latter observation can be seen more clearly in Figure 3-19, where the graph in Figure 3-18 is now focused only on the temperature distribution inside the ceramic preform.

The results of the simulation for the temperature distribution in Figure 3-19 shows that infiltration of a mold of more uniform temperature might be achievable by increasing the runner diameter, and the corresponding heater inner diameter.
Changes in Thickness of Fiberfrax®

In this simulation, the dimensions of the model are changed to reflect a change in the insulation layer of Fiberfrax® material. It should be noted that only the thickness in the $r$-direction is changed, while that in the $z$-direction remains the same. The thickness is first doubled, and then halved, so that a comparison could be made between different thicknesses. The temperature of the heater is fixed in all three cases (i.e. doubled, halved and original thickness). Figure 3-20 shows the results of the simulation.

From Figure 3-20, it can be observed that, because Fiberfrax® is already an excellent thermal insulator, changing its thickness does not affect the temperature distribution in the runner very significantly.
Molten Iron in Ceramic Runner

The final simulation looks at the estimated temperature distribution in the runner system if we now have molten iron (instead of molten Al-4.5%Cu alloy) inside a ceramic runner made of aluminum oxide (instead of a stainless steel one). A number of changes need to be made to the model for this simulation to be realistic, and these are discussed as follow:

1. The melting point of iron is 1538°C [73]. Therefore, the heater needs to be heated up to a very high temperature in order to keep the iron molten in the runner. Assuming that the heater is capable of being heated to such a high temperature (the actual heaters used in the experiments can only go to a maximum temperature of 1204°C), we set the reference cell in the Excel model to 2000°C to ensure that the iron in the whole length of the runner stays above its melting point.

2. The thermal conductivities of both the runner and molten metal materials need to be changed to those of aluminum oxide and molten iron respectively. The data for this is available in Table 3.1.

3. The first trial-run reveals that setting the reference cell to 2000°C will cause the aluminum casing to reach such a high temperature that it melts. To prevent this, the thickness of the insulating layer of Fiberfrax® surrounding the heater is doubled, as discussed in the preceding section. As shown in Figure 3-20, doubling the thickness of the insulation layer does not affect the temperature distribution in the runner significantly. However, the outside surface of the Fiberfrax® material is definitely reduced, thus reducing the temperature of the aluminum casing.

   After adjusting for the thermal conductivity of the aluminum casing with temperature (refer to Table 3.1), the model is run again. This time, it is found that the increased thickness of insulation has prevented the aluminum casing from melting. The simulation shows that the casing will reach an estimated temperature of 300-400°C.

4. The k-values of all other components in the model (namely, Fiberfrax®, heater, ceramic preform and air gap between the heater and runner) have to be changed accordingly to
be consistent with the increase in temperature of the whole system. The data for this is available in Table 3.1.

5. The convection heat transfer coefficient of the air in the surroundings is assumed unchanged, and the values of $h_r$ in both the surroundings and between the inside surface of the heater and outside surface of the runner are calculated automatically for each cell in the model. To find the value of $h_c$ for the air gap between the heater and runner, Equations 3.3 and 3.4 are once again used. The calculated value of $h_c$ in this case turns out to be approximately 16 W/m²K.

6. Convection and radiation in the molten iron are assumed negligible compared to heat transfer by conduction.

Figure 3-21 shows the expected temperature distribution along the ceramic runner filled with molten iron.
3.4 Cooling Curves

Having determined the steady-state temperature profiles for different cases in the previous section, an experiment is now conducted to find out the cooling profile of the system with molten Al-4.5%Cu alloy in the stainless steel runner. Figure 3-22 shows the cooling curves at different locations along the runner. These data are recorded using a DT9806 data acquisition module from Data Translation, Inc.

The “points” in Figure 3-22 correspond to the points in Figure 3-13. All these points lie along the runner system. Points 4 and 5 correspond to locations within the molten Al-4.5%Cu alloy; points 6, 7 and 8 correspond to those in the ceramic preform; and points 10 and 11 correspond to the inside and outside surfaces, respectively, of the Fiberfrax® insulating material.

As we can see in Figure 3-22, the temperature on the outside surface of the Fiberfrax® (point 11) remains steady at 210-220°C, while the temperatures in the other regions drop steadily. Of particular interest is the cooling curve of point 6, which can be taken as the interface between the molten Al-4.5%Cu alloy and the ceramic preform. It is important to ensure that
the temperature in this region remains above the liquidus of the alloy until infiltration occurs, which takes at least 35 seconds, for this is the amount of time it takes for the centrifuge system to reached the desired speed of 1500 RPM (as discussed in the next chapter). Figure 3-22 shows that we have a total time of 4 minutes before this region solidifies. The rest of the alloy in the runner, however, takes a much longer time to solidify. For example, Region A in Figure 3-22, where the cooling curves straighten out horizontally for a while, shows the solidification of the molten Al-4.5%Cu alloy a little further into the runner.

Therefore, given that it takes only about 35 seconds for the IEC CRU-5000 centrifuge to reach the desired speed of 1500 RPM for infiltration to take place, it is safe to assume that infiltration of the ceramic preform by the molten Al-4.5%Cu alloy will certainly occur. It is assumed that the cooling rate of the molten aluminum alloy inside the runner remains the same when the centrifuge is spinning compared to when it is stationary. As already discussed in a previous section, this assumption is considered acceptable since an increase in the convection heat transfer coefficient of the surrounding air when the system is spinning will probably not significantly affect the amount of heat loss into the surroundings because of the extremely low thermal conductivity and high specific heat capacity [60] of the layer of Fiberfrax® insulating material surrounding the heaters.

The information thus obtained from Figure 3-22 allows the optimum temperature (to which to heat the heaters) to be determined in the simulation, as discussed in a previous section.
Chapter 4

IEC CRU-5000 Centrifuge

Centrifugal casting using the centrifuge system described in the preceding chapters is done on a Model CRU-5000 Centrifuge manufactured in June 1975 by the IEC Division of Damon Corporation. A picture of this machine is shown in Figure 4-1.

The CRU-5000 is actually a general-purpose refrigerated centrifuge designed for use in the medical, industrial and scientific laboratory to perform separations by centrifugal force [74]. However, it was found that this machine is also ideal for use in centrifugal casting, particularly in combination with the centrifugal system discussed in the preceding chapters, because the rotor of the machine can be easily connected to the lower portion of the aluminum casing via a custom-made adaptor. Figure 4-2 shows an exploded view of this assembly, as generated by SolidWorks®.

As noted in Figure 4-2, the rotor is made of brass while the adaptor is made of steel. The entire setup is attached to the vertical shaft of the centrifuge motor and rotated at high speeds about a vertical axis through the center of the system.

4.1 Balancing and Vibration

At high rotational speeds, it is extremely important that all the rotating parts are well balanced. Unfortunately, because of the fact that many components in the system have variable weights, especially the ceramic heaters and the aluminum casing (which is machined entirely by hand), and that the same parts on each half of the assembly cannot be placed at exactly the same
Figure 4-1: The IEC CRU-5000 Centrifuge
Figure 4-2: Exploded view of rotor-adaptor-casing assembly for the centrifuge
distance away from the center, a small amount of unbalance is introduced into the system. An unbalance is defined as a discard between the system's inertia axis and the axis of rotation [75].

All rotating parts, however well balanced, need to pass through multiple critical speeds while going up to or coming down from the operational speed [75]. If a system is not perfectly balanced, some degree of vibration will be experienced as it goes through these critical speeds. The more unbalanced a system is, the worse the vibrations become.

4.2 Critical and Maximum Speeds

The centrifuge system in Figure 4-2 is rotated at high speed on the motor shaft, and its speed is monitored by a DT-2234C Digital Laser Tachometer, distributed by Kaito Electronics, Inc. It is found that the first critical speed of the system occurs at 660 RPM. After passing through this critical speed, the second one is encountered at 1500 RPM. At this speed, the vibrations become so violent that any further increase in speed is deemed unsafe.

In order to find out the maximum potential speed that the centrifuge can obtain without worrying about the vibrations, the rotor was first rotated independently on the centrifuge. This was done because the rotor was part of the centrifuge machine and therefore was made to precise dimensions for a good balance on the motor shaft when rotated at high speeds. It was found that the first, and only, critical speed of the rotor, with almost negligible vibrations, occurred at 1040 RPM, and the maximum speed obtained was 3310 RPM.

Next, half the centrifuge system (consisting of the rotor, adaptor and the lower half of the aluminum casing only) was mounted on the motor shaft and rotated to its maximum speed. The first, and only, critical speed was encountered at 710 RPM, with some vibration, and the maximum speed obtained, in this case, was 2240 RPM.

Table 4.1 summarizes the findings.

<table>
<thead>
<tr>
<th>System Tested</th>
<th>1st C.S. (RPM)</th>
<th>2nd C.S. (RPM)</th>
<th>Max Speed (RPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor only</td>
<td>1040</td>
<td>—</td>
<td>3310</td>
</tr>
<tr>
<td>Half centrifuge system</td>
<td>710</td>
<td>—</td>
<td>2240</td>
</tr>
<tr>
<td>Full centrifuge system</td>
<td>660</td>
<td>1500</td>
<td>?</td>
</tr>
</tbody>
</table>

Table 4.1: Critical and maximum speeds for parts of the centrifuge system (C.S. denotes critical speed)
From Table 4.1, although the maximum speed of the full centrifuge system as shown in Figure 4-2 cannot be measured due to the violent vibrations that it is subjected to at the second critical speed, it is reasonable to assume that the maximum speed cannot be much larger than 1500 RPM. Hence, a maximum speed of 1500 RPM is decided upon for the centrifuge system using the IEC CRU-5000 centrifuge. A more detailed analysis of the maximum speed of the system will be discussed in the next chapter.

4.3 Time Taken to Reach Maximum Speed

Figure 4-3 shows the speed of rotation of the system as a function of time, for as far as the speed is measurable by the tachometer, since violent vibrations will be felt when we get too close to the second critical speed.

The slight “S” curve shown in Figure 4-3 demonstrates the typical acceleration-deceleration pattern of the motor as it starts from 0 RPM to reach its maximum speed of 1500 RPM.
Approximating a linear relationship between speed and time to the figure, the time it takes for the centrifuge system to reach its maximum speed of 1500 RPM can be estimated to be about 35 seconds.
Chapter 5

Samples Obtained

With the capabilities and limits of the IEC CRU-5000 centrifuge determined in Chapter 4, and the heat transfer characteristics and cooling pattern of the centrifuge system established in Chapter 3, two samples are produced from this setup to test the potential of the system.

5.1 Tin-Alumina Sample

The first sample produced is the infiltration of Sn-15\%Pb alloy into aluminum oxide powder of 1 $\mu$m in diameter. The volume fraction of this ceramic reinforcement is assumed to be 50\%, and the surface energy of the Sn-15\%Pb alloy is taken to 0.55 Pa-m [76]. It is assumed that the metal-ceramic interface is perfectly non-wetting for the present analysis. Therefore, the contact angle is taken to be 180°. From Equation 2.2, the expected threshold pressure for the particulate compact in this experiment would be 33 atm.

Molten Sn-15\%Pb alloy is then poured into the runners of the centrifuge system until it completely fills the runners and the reservoir. Giving the metal enough superheat to ensure that it stays molten until infiltration occurs, the entire system is rotated to a targeted speed of 1500 RPM in the IEC CRU-5000 centrifuge for 35 seconds, until the desired infiltration pressure is attained. Using Equation 2.1, the infiltration pressure achieved in this experiment is calculated to be 36.83 atm, indicating that infiltration is expected to occur. Indeed, infiltration is achieved. The microstructure of the tin matrix composite obtained is shown in Figure 5-1.

Few defects are observed in the microstructure in Figure 5-1. The micrograph shows that
the centrifuge system designed has a high potential to be used for the fabrication of metal matrix composites relatively cheaply and easily.

5.2 Aluminum-Alumina Sample

Obtaining an aluminum-alumina sample is slightly more difficult because of the much higher temperatures required and the lower infiltration pressure achieved (at the same speed of rotation) due to the lower density of the Al-4.5\%Cu alloy used. Using the simulation results obtained from the Excel model developed and discussed in Chapter 3, it is estimated that the interior of the heater (the reference cell in the Excel model) needs to be heated up to a temperature of 743°C in order for the metal-ceramic interface temperature to be at 652°C, so that when infiltration occurs, the temperature at this region will be at the liquidus of the Al-4.5\%Cu alloy.

In this experiment, however, the heater is heated to 800°C, to ensure that the alloy will not solidify before complete infiltration is obtained. Results of the Excel simulation show that the
stainless steel tubing used for the runner will reach a temperature of as high as 760°C.

Since it is known that the infiltration pressure achieved at 1500 RPM in this experiment will be lower than that achieved with the Sn-15%Pb alloy in the first experiment (because of the significantly lower density of the Al-4.5%Cu alloy), the aluminum powder used as the ceramic reinforcement will now have to be of a different size, so that its threshold pressure is reduced. A powder size of 9.5 μm in diameter is decided upon. The surface energy of the Al-4.5%Cu alloy is taken to be 0.893 Pa-m [76]. Once again, assuming a 50% volume fraction, and that the metal-ceramic interface is perfectly non-wetting (i.e. the contact angle is 180°), the threshold pressure needed to induce infiltration is calculated to be 5.64 atm. At 1500 RPM, the infiltration pressure achieved is expected to be 12.35 atm.

Unfortunately, however, the stainless steel tubing used for the runner is apparently not meant to withstand a pressure of 12.35 atm at 760°C. As it turns out, both runners failed and broke into two parts during the centrifugal process. As a result, only partial infiltration is achieved in the aluminum oxide powder. The microstructure of the infiltrated portion of the ceramic reinforcement is shown in Figure 5-2.

A comparison of Figure 5-2 with Figure 5-1 shows that there is much room for improvement in the centrifuge system designed to infiltrate aluminum (and its alloys) into a ceramic reinforcement. Three major areas need to be looked into:

1. It is evident, from the failure of the runners, that the AISI 316 stainless steel tubing used is not suitable for use under high pressure at high temperature. This will be further discussed in the following chapter, in which the design of a new centrifuge system will be considered.

2. As a result of the runners breaking into two, \( r_0 \) in Equation 2.1 is no longer at zero. The amount of Al-4.5%Cu alloy left in the tubing is measured and \( r_0 \) is found to be about 5". Substituting \( r_0 = 5" \) into Equation 2.1 gives us an infiltration pressure of 7.09 atm, which, fortunately, is larger than the threshold pressure of 5.64 atm required, but not much larger. More importantly, the significantly reduced infiltration pressure results in the presence of many air “bubbles” as shown in Figure 5-2. However, it is expected that if the infiltration pressure is high enough, porosity in the composite produced will be
Figure 5-2: Microstructure of aluminum oxide reinforced Al-4.5%Cu composite
sharply reduced, as demonstrated in the first experiment with Sn-15%Pb alloy, where the infiltration pressure obtained is more than 5 times that obtained in this second experiment with Al-4.5%Cu alloy. Hence, the higher pressure is needed to “squash” the bubbles to acceptable dimensions so that porosity is reduced during freezing.

3. The problem of partial infiltration can be explained by the formation of a thin oxide film by contact with air on the infiltration front of the molten Al-4.5%Cu alloy. The oxidation that occurred at this front greatly increased the surface energy of the alloy, and thus acted as a more effective barrier to wetting than the presumably much thinner film formed in vacuum. The experiment has been designed to be conducted in ambient air, which therefore may not be a suitable environment for the present centrifugal casting technique used to fabricate aluminum matrix composites. Hence, the feasibility of operating a new design of the centrifuge system either in vacuum or in an inert atmosphere filled with nitrogen or argon gas will have to be examined in future.
Chapter 6

New Machine Design

6.1 Capabilities of Current Machine

In an ideal system, where non-linear friction and windage losses in the motor are considered negligible, the torque, $T_a$ (lb-ft), that is required to accelerate the system to a certain speed, $\Omega$ (RPM), in a certain amount of time, $t$ (s), is given by:

$$T_a = \frac{Wk^2\Omega}{308t}$$

(6.1)

where $Wk^2$ (lb-ft$^2$) represents the moment of inertia of the system [77].

In order to determine the inertia of the rotor-adaptor-casing assembly in Figure 4-2, which is too complex to be calculated by hand (for example, the aluminum casing, with its contents, consists of more than 10 separate parts, as shown in Figure 2-8), SolidWorks® 3D Modeling Software is used for this computation. The density for each component in the assembly (refer to Table 6.1) is entered into the software, and the moment of inertia about a vertical axis through the center is then automatically calculated by SolidWorks®. The inertia of the system is found to be 6.15 lb-ft$^2$.

The inertia of the centrifuge motor itself may need to be included in the total inertia of the system if it is substantially large. From Reference [74], it is known that the IEC CRU-5000 centrifuge is equipped with a $\frac{3}{4}$ hp centrifuge motor. The typical dimensions of a $\frac{3}{4}$ hp motor can be approximately represented by a solid vertical steel cylinder that is 10 cm (0.328 ft) in
<table>
<thead>
<tr>
<th>Material</th>
<th>( \rho ) (kg/m(^3))</th>
<th>( \rho ) (lb/in(^3))</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al-4.5%Cu Alloy</td>
<td>2640</td>
<td>0.095</td>
<td>[61]</td>
</tr>
<tr>
<td>Aluminum 356</td>
<td>2587</td>
<td>0.094</td>
<td>[69]</td>
</tr>
<tr>
<td>Alumina (50% volume fraction)</td>
<td>1921.25</td>
<td>0.069</td>
<td>[78]</td>
</tr>
<tr>
<td>Mullite (Heater)</td>
<td>2249.25</td>
<td>0.081</td>
<td>actual measurements</td>
</tr>
<tr>
<td>AISI 316 Stainless Steel</td>
<td>8238</td>
<td>0.298</td>
<td>[65]</td>
</tr>
<tr>
<td>Carbon Steel</td>
<td>7986.4</td>
<td>0.289</td>
<td>actual measurements</td>
</tr>
<tr>
<td>Brass</td>
<td>8530</td>
<td>0.308</td>
<td>[65]</td>
</tr>
</tbody>
</table>

Table 6.1: Densities of the materials used in SolidWorks®

height, \( H \), and 10 cm (0.328 ft) in diameter, \( d \). If \( M \) is the mass of the motor, then [79]:

\[
Wk^2 = \frac{1}{2} M \left( \frac{d}{2} \right)^2 = \frac{1}{32} \pi \rho H d^4
\]  

(6.2)

Equation 6.2 gives the inertia of the centrifuge motor as 0.00784 kg-m\(^2\), or 0.186 lb-ft\(^2\). Since this value is only about 3% of the inertia of the centrifuge system, the inertia of the centrifuge motor can therefore be safely neglected.

It is known, from the previous chapter, that \( \Omega = 1500 \) RPM and \( t = 35 \) seconds for the particular centrifuge system under discussion. Thus, using Equation 6.1, the accelerating torque required in an ideal system with no losses is 0.856 lbf-ft.

In reality, however, losses – in the form of non-linear friction and windage – are always present. The presence of these losses requires the torque supplied by the motor to be more than the value calculated above, so as to compensate for the extra energy that is dissipated in overcoming these losses. In this analysis, friction of the motor is neglected because it is assumed that the motor has been designed to have negligible friction. More importantly, however, we know that frictional losses will be insignificant when compared to those caused by windage, because the centrifuge system designed (i.e. the aluminum casing) is far from being aerodynamic.

The torque that is required to accelerate a body with inertia \( Wk^2 \) is simply the product of its inertia and the acceleration. Conversely, the windage torque (or load torque, \( T_l \)) that is produced by the body during deceleration is also the product of its inertia and the deceleration. This is given by Equation 6.3:
From this equation, it is clear that the load torque of a system not only depends on the system's inertia, but it is also different at different speeds, since the deceleration of a freely rotating body in air is never constant, but instead decreases as the speed decreases. Furthermore, the relationship between deceleration and speed might not necessarily be linear. Therefore, an experiment has to be conducted to find out the value of $T_l$ as a function of $\Omega$, as the system decelerates freely from full speed to 0 RPM. The value of this deceleration is found by measuring the speed of the system at regular intervals using the DT-2234C Digital Laser Tachometer. The difference between two measurements is then taken and divided by the time interval between them to give the deceleration of the system at a certain time $t$, where $t$ is taken as the average between the two timings under consideration.

Figure 6-1 shows the relationship between $T_l$ and $\Omega$ for the centrifuge system discussed.
The load torque along the y-axis in Figure 6-1 is calculated from Equation 6.3, taking as input the value of the deceleration of the system measured at each speed along the x-axis. A quadratic equation fitted through the data points shows a very good match between the points and the curve, as shown in Figure 6-1. Therefore, for $\Omega = 1500$ RPM, $T_l$ is estimated to be approximately 0.628 lb*ft. This shows that the load torque is in fact comparable to the value of $T_a$ already calculated (which is 0.856 lb*ft). The total torque, $T$, that the centrifuge motor needs to produce in order to accelerate the centrifuge system to the desired speed of 1500 RPM in 35 seconds, is the sum of $T_a$ and $T_l$:

$$T = T_a + T_l$$

(6.4)

which gives 1.484 lb*ft.

The relationship between the motor horsepower output, $HP$, the motor torque output, $T$, at the motor shaft, and the motor operating speed, $\Omega$, under equilibrium condition is [80]:

$$HP = \frac{T\Omega}{5252}$$

(6.5)

This simple equation shows that the horsepower output of the centrifuge motor, rotating at 1500 RPM and carrying a total torque of 1.484 lb*ft is 0.42 hp. This is safely within the horsepower rating of $\frac{3}{4}$ hp of the motor.

On a further note, an estimate of the maximum speed, $\Omega_{max}$, of the machine can also be found for the centrifuge system under discussion. This requires a combination of Equations 6.1, 6.4 and 6.5, as well as Figures 4-3 and 6-1, giving Equation 6.6 below:

$$\Omega_{\max} = \frac{5252HP}{43.303Wk^2/308 + 0.0000495\Omega_{\max}^2 + 0.0003684648\Omega_{\max} - 0.0357200322}$$

(6.6)

It is not difficult to solve the above equation for $\Omega_{\max}$, which gives us a theoretical value of about 2100 RPM for the maximum speed of the centrifuge machine, for the centrifuge system designed. However, because of the violent vibrations experienced by the machine at its second
critical speed of 1500 RPM, we are prevented from going beyond 1500 RPM for this system.

6.2  Factors to be Considered in New Design

The main purpose of designing a new setup is to achieve the highest possible pressure in the shortest possible time at the lowest possible cost. The variables to be considered are discussed in the following sections.

6.2.1  Length of Runner, $l$

As shown in Figure 2-1, increasing the length of the runner increases the value of $r_1$, which, according to Equation 2.1, increases the infiltration pressure, $P$.

Unfortunately, $l$ is also directly related to the inertia of the system – the longer the runner, the larger the inertia. A horizontal rod of length $L$, mass $M$, radius $r$ and density $\rho$, rotating about a vertical axis through its center, has a moment of inertia given by [79]:

$$Wk^2 = \frac{1}{12} ML^2 = \frac{1}{12} \pi r^2 \rho L^3$$  \hspace{1cm} (6.7)

Equation 6.7 shows that the inertia of a rod is proportional to the cube of its length. Approximating a pair of runners – one on each side of the reservoir, as shown in Figure 2-2 – that is filled with molten metal as a horizontal rod described above (since the diameter of the runner is small compared to its length), we can then assume that the inertia of the system will roughly increase with the cube of the runner length. And increasing the inertia of the system will in turn result in the requirement of a more powerful centrifuge motor for the centrifuge machine, as indicated by Equations 6.1, 6.3, 6.4 and 6.5.

In addition, increasing the runner length also results in an increase in the cost of production as longer runners and heaters (as well as the corresponding increase in the amount of insulation required) will cost more.
6.2.2 Speed of Rotation, $\Omega$

Changing the speed of rotation of the system will have a direct effect on both the infiltration pressure, as seen in Equation 2.1, and the horsepower requirement of the motor, as seen in Equation 6.5. Increasing $\Omega$ is desirable because $P$ is proportional to the square of the angular velocity. However, this directly increases the power output of the motor, which translates to additional costs. The safety issue in dealing with high angular velocities (especially if the system is a large one) might also be a concern.

6.2.3 Time Required to Reach Desired Speed, $t$

The amount of time it takes for the centrifuge to reach the desired speed of rotation, so that the required pressure is obtained for infiltration of the ceramic preform to occur, determines the production rate of metal matrix composites (MMCs), as well as the amount of superheat that is required by the molten metal in the runner, so that it stays molten sufficiently long enough to infiltrate the preform once the desired pressure is reached. The temperature to which to heat the metal in the runner will then determine the amount of power drawn by the heaters, which in turn translates to costs. As such, it is desirable to make $t$ as small as possible, which makes for a faster production rate of MMC and a lower amount of superheat required.

Unfortunately, a lower value of $t$ also increases the acceleration torque required (Equation 6.1), which increases the total torque that the centrifuge needs to produce (Equation 6.4), which in turn increases the horsepower requirement of the centrifuge motor (Equation 6.5), which ultimately translates to additional costs.

Therefore, in conclusion, the above discussions show that any attempt to increase the pressure or speed, or decrease the time, will require a tradeoff in some other aspect of the system. The following sections will look at the above three factors collectively and establish some relationships among them.
6.3 Motor Horsepower Requirement as a Function of Infiltration Pressure for Current System

6.3.1 Changes in Runner Length, $l$, with Time, $t$, Fixed

We will first look at how the horsepower requirement of the centrifuge motor is expected to change with respect to an increase in the infiltration pressure due to changes in the runner length, keeping $t$ constant.

In calculating the value of $P$ using Equation 2.1, it is assumed that $r_1$ in Figure 2-1 is approximately equal to 85% of the value of $l$ (leaving 15% for the length of the ceramic preform at the end of the runner). Calculations for $P$ are then made with different values of angular velocities, $\omega$.

There is, however, a limit to the maximum pressure that we can safely achieve from the runner system designed. From Reference [57], it is found that the maximum allowable pressure at 648°C for the AISI 316 stainless steel tubing used is 0.37 times that at room temperature, which is 5100 psi. This gives us a maximum allowable pressure at 648°C of 1887 psi, which is 128.4 atm. Swagelok® connectors are designed in such a way that they will not fail until after the tubings have failed [57]. Hence, it can be assumed that the stainless steel tubing is the limiting factor in the runner system designed. This gives us a maximum allowable infiltration pressure of 128.4 atm at 648°C for the centrifuge system under discussion.

It is noted here that there is no data available for the strength of the stainless steel tubing for temperatures beyond 648°C. However, given that the tubing failed at 12.35 atm at 760°C (as described in Chapter 5), we can assume that the factor for the maximum allowable pressure is much less than 0.37 at temperatures beyond 648°C. Since there is no way to estimate the value of this factor at temperatures beyond 648°C unless specific experiments are conducted to find it, the present analysis therefore assumes that the temperature of the tubing is at 648°C when infiltration occurs, i.e. when the maximum pressure is reached, so that the factor of 0.37 may be used.

Determination of the value of $HP$ is a little more tedious. As discussed above, we will assume that a change in $l$ results in a cubic change in the inertia of the system, which in turns affects Equations 6.1, 6.3, 6.4 and 6.5.
The accelerating torque can be easily calculated from Equation 6.1. However, the load torque cannot be easily determined because it depends on the deceleration of the system, which is different at different speeds for systems with different moments of inertia. Figure 6-1 obtained is only for a particular system (in this case, the full setup) with a particular moment of inertia. The graph would look differently when a different system — with a different inertia — is considered. To solve this problem, the load torque for different parts of the centrifuge system is measured with respect to speed so as to obtain a pattern for different moments of inertia, and the results are presented in Figure 6-2. The different cases are defined thus:

- Case 1 – Rotor only
- Case 2 – Rotor and bottom half of aluminum casing
- Case 3 – Rotor and full aluminum casing (without runner and reservoir system)
- Case 4 – Full assembly (Figure 6-1)

It is noted that the rotor in Cases 1 – 3 refer to the brass rotor of the IEC CRU-500 centrifuge, and not the rotor of the centrifuge motor. The moments of inertia for the above four cases are found using SolidWorks® 3D Modeling Software.

From Figure 6-2, it is evident that $T_1$ increases steadily with both the speed and inertia of the system. This provides us with a good estimation of the value of $T_1$ for different systems with different moments of inertia at different speeds. First, an approximate linear relationship is obtained for the value of $T_1$ as a function of the system inertia (i.e. Cases 1 – 4) at each speed, e.g. at 500 RPM, 1000 RPM, 1500 RPM, etc. This allows us to make a rough estimate for the value of $T_1$ for a particular inertia at each of these speeds. The data points thus obtained will further allow us to make another rough estimate of $T_1$ vs. RPM for a particular inertia, therefore extending the information obtained from Figure 6-2 to other systems with different moments of inertia, and not just to Cases 1 – 4 alone. It is, however, noted that this is an extremely rough approximation for the value of $T_1$ with respect to the speed of rotation. Accurate values cannot be obtained without conducting actual experiments to measure the load torque as a function of speed for a particular system, and separate experiments have to be done for each different system under consideration.
Figure 6-2: Load Torque as a function of speed for different systems with different moments of inertia
Figure 6-3: Estimated motor horsepower requirement as a function of infiltration pressure, which varies as a result of changes in the runner length (with time \( t \) fixed)

Nevertheless, the estimated value of \( T_l \) obtained by the method described above is used in Equation 6.4 to calculate the total torque that the centrifuge motor needs to produce in order to accelerate the centrifuge system to a desired speed. Equation 6.5 is finally used to calculate the horsepower required.

Figure 6-3 shows the results obtained from this analysis. The \( x \)-axis represents the infiltration pressure achieved with varying runner lengths at six different speeds of rotation. Since the runner can only hold a maximum pressure of 128.4 atm at the elevated temperature of 648°C, calculations are only made for \( P < 128.4 \) atm.

To find the value of \( l \) at a certain pressure at a particular speed, Equation 2.1 is used. The values of \( P \) and \( \Omega \) can be known from Figure 6-3. The molten metal used is Al-4.5%Cu alloy, whose density is given in Table 6.1.

Finally, it is noted that, in deriving this figure, the time taken to reach the desired speed is assumed to be fixed at 30 seconds.
It can be seen from Figure 6-3 that the estimated horsepower of the motor required increases with the infiltration pressure achieved. Figure 6-3 also shows that the higher the speed of rotation, the smaller is the motor required to obtain the same infiltration pressure. This shows that the inertia of the system plays a large part in the power requirement of the motor. This is because, when the speed is increased, the runner length needed to produce the same infiltration pressure is decreased. The increase in speed balanced by a corresponding decrease in runner length results a decrease in the horsepower requirement of the centrifuge motor. This can only be explained by the fact that a change in runner length, and hence the system inertia, is more significant in changing the horsepower requirement than is the speed of rotation.

Another observation that can be made from Figure 6-3 is that, even with a low rotational speed, the estimated horsepower requirement of the centrifuge motor for the present setup should not exceed 25 hp even for an infiltration pressure of >100 atm.

6.3.2 Changes in Speed of Rotation, $\Omega$, with Time, $t$, Fixed

Figure 6-4 shows the same analysis as the previous section, but from a different perspective, where $P$ varies as a function of $\Omega$ instead of $l$. The values of $HP$ are calculated in exactly the same way as described in the previous section.

In Figure 6-4, the numbers beside each graph represents the length of the runner as the speed of rotation varies. The results of these graphs agree with what has been predicted in the previous section. In order to achieve the same infiltration pressure, the speed of rotation decreases with an increase in the runner length. However, because of the larger influence that the corresponding increase in the system’s inertia has on the horsepower requirement compared to what can be offset by the decrease in speed, we therefore see a trend in Figure 6-4, whereby the graph shifts upwards as the runner length increases. In fact, the steepness of the graph is more pronounced the longer the runner, thus showing the significant effect that the inertia of the system has on the horsepower requirement of the motor.

6.3.3 Changes in Values of $t$

In the previous two sections, the time it takes for the centrifuge to reach the desired speed is assumed to be constant at 30 seconds. Increasing or decreasing the value of $t$ will have
Figure 6-4: Estimated motor horsepower requirement as a function of infiltration pressure, which varies as a result of changes in the speed of rotation (with time $t$ fixed)
different effects on the horsepower requirement of the centrifuge motor. In general, the faster we want the centrifuge to attain the desired speed, the more powerful the motor must be. This is because a change in the value of $t$ affects the acceleration torque required (Equation 6.1), which in turn affects the horsepower requirement (Equations 6.4 and 6.5). Figures 6-5 and 6-6 show the horsepower as a function of the infiltration pressure with different values of $t$, by first changing the runner length but keeping the speed of rotation constant at 3000 RPM, and then by changing the speed of rotation but fixing the runner length at 16" (41 cm).

Both Figures 6-5 and 6-6 show that the horsepower requirement of the centrifuge motor increases sharply as the value of $t$ decreases, especially in the region where $t < 10$ s. This is expected as Equation 6.1 tells us that $T_a$ is inversely proportional to $t$, which means that $T_a$ will become very large if $t$ becomes too small.

In the present setup, the analysis seems to suggest that even if we want to decrease $t$ from the present value of 30 seconds to as low as 5 seconds, we would only require a 35 – 40 hp
Figure 6-6: Estimated motor horsepower requirement as a function of infiltration pressure, which varies as a result of changes in the speed of rotation (with different values of $t$)
motor for the pressure we want to achieve, which is, unfortunately, limited by the maximum allowable pressure load of 128.4 atm for the AISI 316 stainless steel tubing used for the runner. Therefore, even though a tradeoff needs to be sought between increasing production rate and saving on the investment of a more powerful motor, it is not a difficult decision to make, as the 40 hp motor required will not be very expensive. This, however, will be a very different matter if a new design were to be made for the centrifuge system to obtain infiltration pressures of as high as 1000 atm (100 MPa). This will be discussed in the sections that follow.

6.4 New Runner Design for Achieving Infiltration Pressures of 100 MPa

The unique characteristic of the new centrifugal casting technique designed for the fabrication of metal matrix composites, is in its ability to achieve very high infiltration pressures easily, quickly and at relatively low costs. We therefore look at the case whereby it is desirable to obtain an infiltration pressure of 1000 atm (100 MPa).

6.4.1 Limitations of AISI 316 Stainless Steel Runner

As aforementioned, the maximum allowable pressure load for the AISI 316 stainless steel tubing used for the runner is 128.4 atm at 648°C. At room temperature, the maximum allowable pressure load is 5100 psi, or 35.2 MPa. This internal pressure, \( P_i \), acts on the wall of the tubing, which, for the present setup, has a nominal outer radius, \( r_o \), of 0.635 cm (0.25") and a nominal wall thickness, \( \delta \), of 0.165 cm (0.065"), thus giving a nominal inner radius, \( r_i \), of 0.470 cm (0.185").

The stresses that are developed in the tubing as a result of this internal pressure are the radial stress, \( \sigma_r \), the tangential stress, \( \sigma_t \), and, when the end reactions to the internal pressure are taken by the tubing itself, the longitudinal stress, \( \sigma_l \). Assuming that the longitudinal elongation is constant around the circumference of the cylinder, the stresses are given by:
\[ \sigma_r = \frac{P r_i^2 - P_o r_o^2 - r_i^2 r_o^2 (P_o - P_i)}{r_o^2 - r_i^2} \]
\[ \sigma_t = \frac{P r_i^2 - P_o r_o^2 + r_i^2 r_o^2 (P_o - P_i)}{r_o^2 - r_i^2} \]  
\[ \sigma_i = \frac{P r_i^2}{r_o^2 - r_i^2} \]

where \( P_o \) represents the external pressure and \( r \) the radius under consideration [79].

\( P_o \) can be assumed to be negligible compared to the very high pressure of \( P_i = 100 \) MPa that we are considering for the analysis. Also, among the 3 stresses, \( \sigma_i \) is considered the largest \( - \sigma_r \) is negative, which indicates compression, and \( \sigma_t \) is not nearly as large as \( \sigma_i \). The maximum tangential stress will occur at the inside radius, so \( r = r_i \). As a result, the equation for \( \sigma_t \) in Equation 6.8 can be simplified to:

\[ \sigma_t = P_i \left( \frac{r_o^2 + r_i^2}{r_o^2 - r_i^2} \right) \]  

(6.9)

This gives a tangential stress of 120 MPa acting on the wall of the tubing at room temperature. Given that the ultimate tensile strength of AISI 316 stainless steel is 516.7 MPa [57], a safety factor of about 4 should therefore be used when estimating the limits of the new runner design.

### 6.4.2 New Runner Design Considerations

**Runner Wall Thickness**

It has to be ensured that the new runner must be able to withstand the much higher pressure of 100 MPa at the elevated temperature of 648°C. Using a factor of 0.37 from Reference [57], this means that the runner has to hold an enormous internal pressure of 270 MPa at room temperature, which is highly unlikely, given that the maximum tangential stress that the runner can withstand at room temperature is only 120 MPa. This is confirmed by a rearrangement of
Equation 6.9 to give:

\[ r_i = r_o \sqrt{\frac{\sigma_t - P_i}{\sigma_t + P_i}} \]  

(6.10)

Since \( \sigma_t \) is significantly larger than \( P_i \), there is no solution for the inner diameter, no matter how large the outer diameter is. Therefore, it is proven that AISI 316 stainless steel is not a practical material for use as the new runner design to achieve the infiltration pressure of 100 MPa desired. Such a runner will definitely fail before the pressure can be achieved.

Equation 6.9 can again be rearranged to give:

\[ P_i = \sigma_t \left( \frac{r_o^2 - r_i^2}{r_o^2 + r_i^2} \right) \]  

(6.11)

Considering the fact that it would be impractical to have a runner whose inner diameter is less than half of its outer diameter, we can make \( r_i = \frac{1}{2} r_o \) at the minimum. Knowing that \( \sigma_t = 120 \) MPa, \( P_i \) is found to be 72 MPa at room temperature. This translates to about 26.3 MPa (263 atm) at 648°C. Therefore, the maximum infiltration pressure that can be achieved with the AISI 316 stainless steel runner is 263 atm, if its inner diameter is reduced to half of its outer diameter.

**Alternative Material**

A different material certainly has to be found if we want to achieve the infiltration pressure of 100 MPa desired. Using Equation 6.9, the minimum tangential stress that the new runner needs to withstand at room temperature, if it were to be capable of attaining an infiltration pressure of 100 MPa at 648°C without failing, is 450 MPa (if its inner diameter is at a minimum of half of its outer diameter). Using a safety factor of 4, this means that its ultimate tensile strength needs to be at least 1800 MPa at room temperature.

A good candidate for the new runner material is AISI 420 stainless steel, which is very similar to the AISI 316 stainless steel used originally, but has a much higher ultimate tensile strength of 2025 MPa at room temperature, a higher thermal conductivity and a slightly lower
density compared to AISI 316 [81]. Its high ultimate tensile strength ensures that the runner will not fail when an infiltration pressure of 100 MPa is reached at the elevated temperature of 648°C.

Using AISI 420 stainless steel, the new runner will be able to safely withstand a maximum tangential stress of 506.25 MPa at room temperature. Assuming that the factor of 0.37 is applicable to AISI 420 stainless steel as it does to AISI 316 at 648°C, the internal pressure that the runner is required to hold at room temperature is 270 MPa. Using Equation 6.10, the inner diameter of the runner is thus found to be about 55.2% of its outer diameter.

**Total Inertia of New Design – a Rough Estimate**

The runner diameter for the new design will be increased because it is understood that the new centrifuge system will be used to infiltrate larger preforms than what is already achieved in the laboratory. Therefore, it is practical to increase the runner diameter so that not only can it now carry a larger volume of molten metal for full infiltration to be achieved, the infiltration time will also be shortened. Also, in order to obtain a very high infiltration pressure of 100 MPa, the runner thickness also needs to be increased, as discussed in the previous section. However, there is a limit to the extent to which these increments can be made, as the inertia of the system will be increased drastically as a result.

**Increase in Inertia as a Result of Increase in Runner Diameter**

Although it is known that an increase in the runner diameter (and thickness) will inevitably lead to an increase in the moment of inertia of the system, it is impossible to find out the total inertia of this new system unless its exact geometry and size can be fully determined. What we have done thus far is only to look at the runner dimensions. The dimensions for the heaters, reservoir and outer casing have yet to be touched upon. A change in any of these components of the centrifuge system can affect the total inertia of the system significantly.

Therefore, a very rough estimation of the change in inertia as a result of a change in the runner diameter is attempted here. It has to be emphasized that the point of this estimation serves only as an “order-of-magnitude” approximation of what the actual inertia might be for the new system.
In order to simplify the analysis as much as possible, we first reduce the whole centrifuge system into a horizontal solid cylinder with length \( L \), mass \( M \), diameter \( d \) and density \( \rho \). For the original system, \( L \) is taken to be 18” and \( d \) to be 0.5”. The moment of inertia of this “virtual” cylinder about a vertical axis through its center is given by [79]

\[
Wk^2 = \frac{1}{48} M \left( 3d^2 + 4L^2 \right) = \frac{1}{192} \pi d^2 L \rho \left( 3d^2 + 4L^2 \right) \tag{6.12}
\]

Since we know that the actual inertia of the original system is 6.15 lb-ft², we can use Equation 6.12 to find out the “effective \( \rho \)” of the system as approximated by a horizontal solid cylinder. Knowing the effective \( \rho \) allows us to substitute its value into Equation 6.12 again to estimate the inertia of the new system with the same \( L \), but with \( d \) changed now to a new, increased runner diameter.

Although it is recognized that the centrifuge system is far from being a horizontal solid cylinder, it is assumed that the inaccuracy resulting from using this approximation will be offset, to some degree, by changes in the heater, reservoir or outer casing dimensions of the new design. Furthermore, such an approximation will likely result in an overestimation of the actual inertia of the system, which is desirable. It is better to overestimate the motor horsepower requirement than to underestimate it, and to discover later that the motor purchased has insufficient power to achieve the infiltration pressure of 100 MPa desired.

**Increase in Inertia as a Result of Increase in Runner Length** As discussed in the previous section, because of the increase in the runner diameter, it is no longer possible to approximate the runners as a horizontal rod – where the calculation for the moment of inertia ignores the contribution made by the diameter to the total inertia, as shown in Equation 6.7 – but as a horizontal solid cylinder instead, where the diameter is important in determining the inertia of the system. Hence, the inertia is no longer simply proportional to the cube of its length, but is related to both its length and diameter as shown in Equation 6.12.

Using the estimation method described in the previous section, the same “effective \( \rho \)” can be used to approximate the inertia of the system at different runner lengths.
Increase in Inertia as a Result of Increase in Preform Size

As aforementioned, the new centrifuge system is designed to infiltrate larger preforms than what has already been accomplished in the laboratory. As such, the inertia of the preform can no longer be considered negligible, especially now that these bigger masses are at a larger distance away from the center of rotation.

In the present analysis, we shall consider a cylindrical aluminum oxide preform that will be attached to the end of each runner. The moment of inertia of a horizontal cylinder with length $L$, mass $M$, diameter $d$ and density $\rho$ about an axis through its center of mass is given by Equation 6.12.

The aluminum oxide preform is assumed to have a volume fraction of about 50%. Given that the density of aluminum oxide is 240 lb/ft$^3$ [78], the density of the preform is therefore 120 lb/ft$^3$. The actual contribution of each preform to the total inertia of the system has to be found using the parallel-axis theorem, which states that the moment of inertia of a body of mass $M$ about an axis parallel to the axis through its center of mass is given by:

$$Wk^2 = Wk_{CM}^2 + Mx^2$$  \hspace{1cm} (6.13)

where $Wk_{CM}^2$ is the moment of inertia about the axis through the body's center of mass, and $x$ is the distance between the two axes [82].

In addition to looking at the contribution made by the aluminum oxide preform to the total inertia of the system, the extended portion of the runner that is used to house the preform should also be considered. This part of the runner shall be assumed to be a hollow cylinder with an inner diameter, $d_i$, equal to the diameter of the preform, and which is 55.2% of its outer diameter, $d_o$. Its length, $L$, is taken to be the same as that of the preform, and its density, $\rho$, is the density of AISI 420 stainless steel, which is 487 lb/ft$^3$ [81]. The moment of inertia of a hollow cylinder is given by [79]:

$$Wk^2 = \frac{1}{48} M \left(3d_o^2 + 3d_i^2 + 4L^2\right) = \frac{1}{192} \pi \left(d_o^2 - d_i^2\right)L\rho \left(3d_o^2 + 3d_i^2 + 4L^2\right)$$  \hspace{1cm} (6.14)

Its contribution to the total inertia of the system can be likewise found using the parallel-axis
Effective Inertia Although the total inertia of the system is the sum of the moments of inertia of all the parts mentioned above, there is a possibility that the effective inertia of the entire system, $Wk_e^2$, may in fact be reduced from the values predicted if a gearbox or some other form of transmission is used on the motor. Assuming that a motor has a gear reduction ratio $i$, and that the inertia of the motor itself is negligible compared to the inertia of the load, the following equation gives the effective inertia of the system [83]:

$$Wk_e^2 = \frac{Wk^2}{i^2}$$  \hspace{1cm} (6.15)

It is known that the gear ratio in any gear motor can go up to as high as 1800:1 [77]. In the analyses that follow, the gear reduction ratio of the motor used shall be taken to be a very conservative value of 5, thus reducing the effective inertia of the system by a factor of 25.

6.4.3 Motor Horsepower Requirement as a Function of Runner Length and Speed of Rotation for New Centrifuge System

The goal of the new centrifuge system is:

*To achieve an infiltration pressure of 100 MPa with molten Al-4.5%Cu alloy in an AISI 420 stainless steel runner, infiltrating a cylindrical aluminum oxide preform with a 50% volume fraction attached to the end of the runner.*

The calculations performed for this analysis are similar to those in Section 6.3.1. The only differences are:

1. Instead of finding the estimated horsepower requirement of the centrifuge motor as a function of infiltration pressure, this pressure is now fixed at 100 MPa. Therefore, graphs are now obtained for the horsepower requirement as functions of runner length and speed of rotation for different values of $t$ instead.

2. Changes are made to the calculation of the total inertia of the system, as discussed in the sections above.
Two cases shall be looked at in this analysis:

- The first is to investigate the motor horsepower requirement as a function of both the runner length and the speed of rotation, with different values of \( t \), for an AISI 420 stainless steel runner with \( d_0 = 6 \text{ cm (2.36’’)} \) and \( d_i = 3.31 \text{ cm (1.30’’)} \), infiltrating a cylindrical aluminum oxide preform with \( L = d = 7.62 \text{ cm (3’’)} \).

- In the second case, a larger runner with \( d_0 = 10 \text{ cm (3.94’’)} \) and \( d_i = 5.52 \text{ cm (2.17’’)} \) will be used to infiltrate a larger preform, with \( L = d = 20.32 \text{ cm (8’’)} \).

In both cases, it is assumed that the reservoir of the centrifuge system has sufficient capacity to not only ensure complete infiltration of molten Al-4.5%Cu alloy into the aluminum oxide preform, but also to supply enough molten metal into the runner such that \( r_0 \) (Figure 2-1) remains at 0 even after infiltration has occurred.

**Case 1 (Small Runner and Preform)**

The new centrifuge design is now equipped with two runners made of AISI 420 stainless steel, with an outer diameter of 6 cm (2.36’’) and an inner diameter of 3.31 cm (1.30’’). An aluminum oxide preform with a 50% volume fraction and measuring 7.62 cm (3’’) in both length and diameter is attached to the end of each runner. The whole system is rotated about a vertical axis though the center of the reservoir to obtain an infiltration pressure of 100 MPa. Figures 6-7 and 6-8 show how the motor horsepower requirement for such a system is expected to vary with changes in runner length and speed of rotation respectively. It should be noted that the \( x \)-axes in both figures are actually related to each other by Equation 2.1, with the density taken as the density of molten Al-4.5%Cu alloy and \( r_0 = 0 \).

Both Figures 6-7 and 6-8 show that the motor horsepower requirement increases with the runner length and decreases with the speed of rotation, even though the infiltration pressure is kept constant at 100 MPa. This once again confirms that the increase in the inertia of the system as a result of the increase in runner length plays a significant role in the horsepower requirement of the centrifuge motor. Therefore, it needs to be emphasized again that a gear reduction ratio of 5 has been assumed in this analysis. A change in the gear ratio will change
Figure 6-7: Estimated motor horsepower requirement as a function of the runner length, at different values of $t$ (small runner and preform)
Figure 6-8: Estimated motor horsepower requirement as a function of the speed of rotation, at different values of $t$ (small runner and preform)
the effective inertia of the system significantly, which results in considerable variations in the horsepower requirement.

As shown in both Figures 6-7 and 6-8, the centrifuge motor under consideration – with a gear reduction ratio of 5 – for the new larger centrifuge system is a much larger one than that required in the original centrifuge system.

Case 2 (Large Runner and Preform)

In the second analysis, the runner is now widened to an outer diameter of 10 cm (3.94") and an inner diameter of 5.52 cm (2.17") to hold more molten Al-4.5%Cu alloy (thus increasing its inertia), and is made to infiltrate a larger cylindrical aluminum oxide preform measuring 20.32 cm (8") in both length and diameter (thus further increasing the total inertia of the system). It should be noted that, because of the much larger volume of the ceramic preform to be infiltrated now, the reservoir needs to be a large one if the runner is not long enough to hold sufficient molten metal for full infiltration to be achieved. Calculations are made as in Case 1 above to predict the motor horsepower requirement as a function of both runner length and speed of rotation. Figures 6-9 and 6-10 present the results of these calculations.

Figures 6-9 and 6-10 show that a larger runner system, infiltrating a larger preform, is achievable. With a gear reduction ratio of 5, the horsepower required of the centrifuge motor is expected to be anywhere between 100 and 1000 hp, or 100 – 700 hp if we allow the value of $t$ to go beyond 1 minute.

6.4.4 Conclusions Drawn from Present Analysis

It is acknowledged that the above discussions provide a very rough analysis on the motor horsepower requirement of the system, for it is impossible to determine the exact size of the motor that is needed unless the actual inertia of the entire system can be established. Despite this uncertainty, a few conclusions can be drawn from the above analyses.

First, it is apparent that if we try to increase the runner length, width and thickness, as well as increase the speed of rotation so as to achieve an infiltration pressure of 100 MPa, the horsepower requirement of the new centrifuge motor will have to be much larger than the one required for the original design. The factors that influence this are the length of the runner,
Figure 6-9: Estimated motor horsepower requirement as a function of the runner length, at different values of $t$ (large runner and preform)
Figure 6-10: Estimated motor horsepower requirement as a function of the speed of rotation, at different values of t (large runner and preform)
the speed of rotation and the time it takes for the centrifuge to accelerate to the desired speed. A change in any of these factors will result in an enhancement in one aspect of the system, but may also lead to a sacrifice in another. From the above analyses, it is clear that the increase in the inertia of the system as a result of an increase in the runner length will raise the horsepower requirement significantly, even though the speed of rotation can be reduced as a result. This necessarily results in the need for a larger and more powerful motor, which translates to greater costs.

Ultimately, it is recognized that the decision to purchase a new centrifuge motor and install the system for a future industrial setup will have to be made with one who is experienced in the field of motor and drive system design and selection, as there are simply too many variables involved. What is done here is only a ballpark analysis to test the feasibility of building a new centrifuge system that can achieve an infiltration pressure of 100 MPa, and to predict the general trend in motor horsepower requirement with respect to both runner length and speed of rotation. The analysis seems to suggest that as long as an appropriate runner material can be found to withstand the high pressures in the runner at high temperatures, acquiring a centrifuge motor for a much larger centrifuge system to produce infiltration pressures beyond 100 MPa will not be an issue, as long as a motor of high gear reduction ratio is used, so that the effective inertia of the system can be reduced.
Chapter 7

Conclusions

It has been shown that the centrifugal casting method of fabricating metal-matrix composites (MMC) is not only possible, but can be achieved relatively easily. Although it is acknowledged that much more detailed analysis is still required in this technique for commercialization of the process to be realized, the fact that small samples are reproducible in the laboratory is proof of its potential.

The simple centrifuge system designed has been shown to have the capability to fabricate tin MMCs easily, with few defects in the microstructure, even when the ceramic reinforcement is a very fine powder of 1 μm in diameter. The infiltration of aluminum into aluminum oxide, although not completely successful, has nevertheless identified a number of factors in which the design of a new centrifuge system should be focused on, and these are summarized briefly as follows:

- It is shown that the “weakest link” in the system, operated at a high temperature of >645°C, is the runner that holds the molten metal during centrifuging. Therefore, the new centrifuge design needs to have a runner that can withstand the high pressures within the runner at high temperatures. A new material, AISI 420 stainless steel, has been identified as a potential candidate, and a brief analysis shows that it should be capable of serving its function as a runner without breaking at the high operating temperatures that it will be subjected to.

- High pressures are definitely desirable to reduce porosity in the samples produced, by
“squashing” the air bubbles to acceptable dimensions during freezing. Therefore, the use of a runner and reservoir system, which increases the infiltration pressure significantly, is crucial to the operation of the centrifuge system discussed.

- Oxidation of the aluminum melt in air is a problem that needs to be examined in detail for successful infiltration of ceramic preforms, as the oxide film formed by contact with air acted as an effective barrier to wetting, thus greatly increasing the effective threshold pressure required for infiltration.

In addition, a heat transfer model of the runner system was successfully developed using Microsoft® Excel, which effectively predicts the temperature distribution in the system at steady state. This allows us to estimate the temperature distributions in the runner system if the design or operating parameters were changed to conditions where no experimental measurements were possible. A number of scenarios were considered, some of which were confirmed by actual experiments to verify the validity of the model. It was shown that an optimum temperature distribution in the runner system can be predicted such that the temperature at the metal-ceramic boundary would be just at the liquidus of the molten metal when infiltration occurs. More importantly, the model showed that changing the diameter of the runner or the thickness of Fiberfrax® insulating material would not significantly affect the temperature distribution within the runner. Using the Excel model developed, we are also able to estimate the temperature requirements and distributions in the runner system if other materials were used.

A rough analysis was made on the motor horsepower requirement of a new centrifuge design. It was shown that the factors affecting this requirement are the length of the runner, the speed of rotation and the time it takes for the centrifuge to accelerate to the desired speed. Changes in the inertia of the system, brought about by changes in the runner length, width and thickness, seems to be the major factor in determining the horsepower required for the centrifuge motor. However, analysis suggests that as long as an appropriate runner material can be found to withstand the high pressures in the runner at high temperatures, acquiring a centrifuge motor for a much larger centrifuge system to produce infiltration pressures beyond 100 MPa will not be an issue, as long as a motor of high gear reduction ratio is used.
Chapter 8

Recommendations for Future Work

The new centrifuge system designed has been shown to be successful in producing tin MMCs from very fine powder sizes of $1 \mu m$ in diameter, and has been partially successfully in fabricating aluminum MMCs. It is acknowledged that much more work needs to be done to bring this process to commercial application. A few of the issues to be looked into are discussed below:

1. The working environment of the centrifuge system is an important factor in the new design. Oxidation of the aluminum melt, which greatly increases the surface energy of the metal and thus acts as an effective barrier to wetting, is a huge concern during infiltration. Although the best solution would be to carry out the infiltration process in vacuum, so that oxidation of the aluminum melt could be completely prevented, it might be more practical to run the process in an inert atmosphere of nitrogen or argon gas. An oxygen-free atmosphere will considerably reduce the formation of the oxide film on the molten aluminum.

2. Temperature control of the centrifuge system can be further explored by developing another Excel model to predict the transient heat transfer in the runner system, especially in the few seconds (or minutes, for a large industrial-sized system) it takes for the system to reach its desired speed of rotation so that the desired pressure is obtained for infiltration to occur. The same finite difference method discussed in Chapter 3 may be used in this additional analysis, with the only requirements being the addition of a time step, $\delta t$, and knowing the density and specific heat capacity of each material in the model. The ability
to establish such a transient analysis of the system will greatly enhance our knowledge and ability to control the temperature distributions in the runner system.

3. The design of a runner system that can incorporate a “cold mold” should next be looked into. In the present system, the ceramic reinforcement is heated to the same high temperature as the molten metal in the runner. For metals with higher melting points, such as steel and titanium, the temperatures in the runner, and therefore of the ceramic preform, will have to be even higher. However, heating a preform to too high a temperature may result in chemical changes in the ceramic fibers, or in chemical reactions of the ceramic preform with the molten metal, both of which are deemed undesirable. Therefore, it is suggested that a new design be considered whereby the ceramic reinforcement is placed outside the heaters, while both maintaining the metal in the runner at its molten state and ensuring that heat loss from the runner system is not too significant as a result of extending the runner beyond the entire length of the heater.

Only when the above factors have been considered and analyzed, should a new centrifuge system be designed to achieve a very large infiltration pressure of 100 MPa. Then the full inertia of the new system needs to be found so that the horsepower requirement of a new centrifuge motor can be properly established.

When the complete centrifuge system is set up, with a larger centrifuge motor installed, the following development stages in the new centrifugal casting process for the fabrication of MMCs are suggested (in no particular order of importance):

1. Development of centrifugal casting technique/design setup in terms of:

   - Melt preparation
   - Holding and handling (transfer of molten metal into machine)
   - Temperature control in machine
     - Temperature uniformity
     - Cooling rate
   - Machine stability at high-speed rotations
• Attachment and removal of mold
• Gating system

2. Solving problems arising from:

• Porosity and particle agglomeration
• Particle settling or flotation
• Extensive interfacial reaction
• Segregation of a secondary phase in matrix

3. Making of preforms

• Uniform particle distribution
• Specific inter-particle distance

4. Testing and categorization

• Mechanical and physical properties obtained (compared with MMCs produced by other methods, such as squeeze casting, gas pressure infiltration, etc.)

5. Production of MMCs of different shapes and sizes – mold design considerations

6. Post-processing considerations

• Heat treatment (aging)
• Forming
• Machining

7. Mass production considerations

• Modifications (and possible redesign) of lab setup

8. Look into the potential of producing MMCs of different matrix-reinforcement combinations (particularly those that require extremely high pressure for infiltration)
• Mechanical properties
  – High fatigue strength
  – High stiffness
  – Wear resistance
  – Elevated temperature properties
  – Coefficient of thermal expansion

• Physical properties
  – Low density
  – Thermal expansion

• Good matrix-particle bonding via:
  – Surface modification of reinforcement
    * Heat treatment
    * Surface oxidation
    * Surface coating
  – Melt treatment
    * Alloying
    * Temperature
  – Processing routes

• Limited thermal expansion mismatch

• Thermal stability

• Seek applications in markets such as:
  – Automotive
  – Aerospace
  – Electronic packaging
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