

High temperature liquid metal loop
– Final report –

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I. Introduction

In order to provide UCSD with a high temperature liquid metal loop which would have a unique capability for high temperature compatibility testing, a literature review was conducted to consider each of the already tested possibilities, and to see where new challenges and promising technologies in this field lie. This would help determine the focus of the project to ensure maximum scientific and engineering value of the final results determined by the experiments performed with the high temperature loop. The literature study conducted here, by considering what has already been accomplished in the past, will shed some light on what we at UCSD would like to accomplish, and what the implications of our findings would mean for the future of structural materials used in liquid metal blanket designs for fusion and possibly generation IV fission reactors.

Liquid metals have surfaced as a useful tool in harnessing the energy from nuclear reactors by trapping the heat produced and transporting it to be converted into electrical energy. Unfortunately, liquid metals are often incompatible with structural materials needed to support the weight of the liquid metal blanket. Loop tests on various eutectics including Lead, Sodium, Lead-Bismuth (LBE), Lithium, and Lead Lithium have all been completed using mainly ferritic steels or stainless steel as test material, subjected to either a forced or natural convective flow of the liquid metal. Corrosion of steels is an inevitable [1] problem in most liquid flows being considered for use in liquid metal blanket design, and must be addressed to ensure a respectable reactor lifetime. Tests on silicon carbide have been performed as well in flowing liquid metals, but to a much lesser extent. Therefore, much of the literature review will

focus on SiC. Here the experiments from other loops around the world will be considered in the light of a new innovative loop to be designed and built by CER at UCSD.

a) Past Experimentation

In 2007, a study was done at the Institut fur Materialforschung in Germany using the PICOLO loop to understand the corrosive effects of flowing Pb-17Li at 550°C on EUROFER [3]. Fusion reactor temperature

requirements were recently

increased from 480°C to

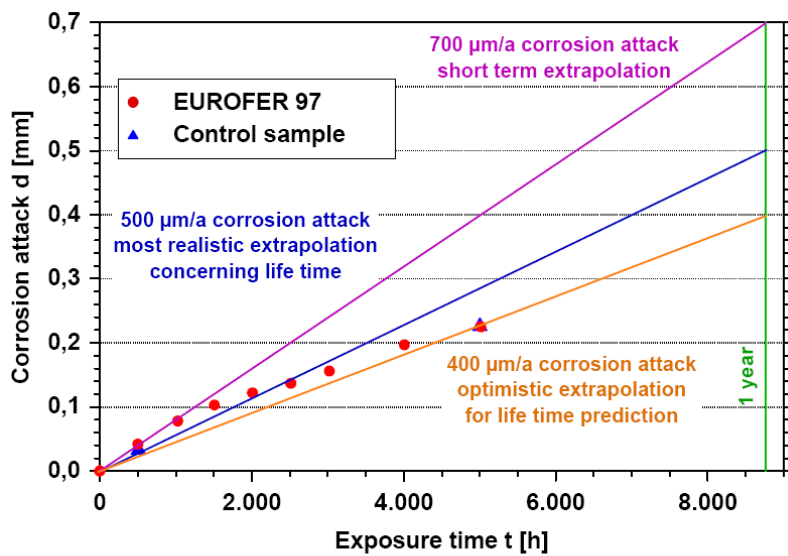
550°C. Since testing of

EUROFER material was absent

at such temperatures,

engineers in Germany

experimented with it in their



Corrosion attack of EUROFER 97 samples at 550°C in PICOLO loop|

Figure 1 from [3]

loop at a flow rate of $v = .22\text{m/s}$ at the 550°C temperature for various exposure times as

expressed in the graph from their paper in figure 1. The goal of the testing was to attain 5000

hours exposure, which was reached successfully as shown. 8mm diameter, 400mm long

cylindrical rods were tested in the test section of the loop. They discovered that over time, that

with only a 70K increase in temperature, with the same flow rate, that a 500% increase in

corrosion rates (from 90μm/a to 500μm/a) occurred. This reveals a dangerous design concern

for a reactor that may overheat only by a small margin, bringing about 5 times the normal

corrosion rates. In this case the lifetime of a reactor blanket could be drastically reduced by a

Test temperature	480 - 550°C
Loop temperatures at operation of 550°C	
T _{max} in test section	550°C
T _{low} at EM-pump	350°C
Pb-17Li volume	20 litres
Length of loop	12.2 m
Flow velocity range (design value)	0.01 to 1 m/s
Test velocity	0.22 m/s
Loop materials	
Cold legs	18 12 CrNi steel
Hot legs	10 % Cr steel
Total loop operation	
at 480°C	> 120,000 h
at 550°C new test temp	> 5,000 h

Table 1 from [3]

550°C, perhaps to 800°C to ensure the safe operation of such a reactor. Also, the fact sheet in table 1 about the PICOLO loop helps us understand what the loop was capable of and how that shows the design parameters we might expand upon, namely the T_{max}. The loop consisted of 9 major components, including the test section, the electromagnetic pump (giving forced convection flow around test coupons), and the glove box as shown in figure 2. The glove box was flooded with argon to prevent contamination of coupons with atmospheric O₂, and specimens could be studied or sealed to be transported within its confines. Similar test have been performed with other loops in Germany using different eutectics.

Another experiment was done in Germany using the CORRIDA loop in 2006 at similar working reactor temperatures of 550°C, however using a 46% lead, lead-bismuth eutectic (LBE) [4]. Before the experiment took place a prediction of the corrosion was made using a mass transfer model [1]. The model predicted that a corrosion increase with an oxygen decrease would occur because more solute is released as the oxide layers grow thinner. The CORRIDA loop at KALLA (Karlsruhe Lead Laboratory) is 35m long and holds 1000kg of eutectic, all of the

small increase in temperature, at least in the hottest section of the loop. More testing could be done to understand what happens to EUROFER at these much higher temperatures than the design

components of which are made of 17-12 Cr-Ni Steel. The mass flow rate of the loop is 5.3kg/s, which corresponds to approximately 2m/s flow velocity. Like the PICOLO loop discussed previously, the test section lies in a glove box which can be flushed with argon such that specimens are not contaminated by atmospheric oxygen and nitrogen. Here the purpose of the experiment was to control the oxygen activity and thus the previously discussed oxide layer.

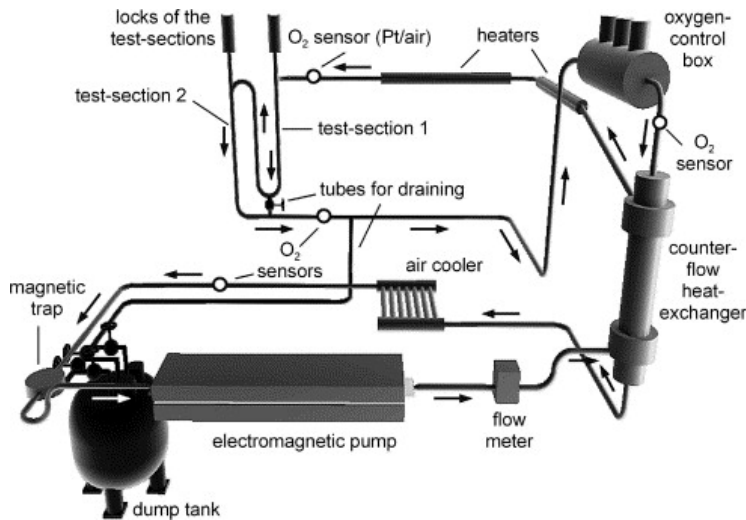


Figure 2 CORRIDA Loop in Germany
from [3]

The oxygen activity is expressed as the activity of lead oxide (PbO), in order to *control* the corrosion of the steel through the varying thickness of the oxide layer. After exposing 8mm diameter 35mm long cylinders of T91 steel, the samples were

analyzed using X-ray diffraction to determine the

corrosion product phases formed during the exposure of 1200 hours. 160°C vegetable fat was rubbed over the samples to remove excess eutectic. It was found that the only mode of material degradation was oxidation, and that 25% of the overall material consumption was Fe₃O₄ at a rate of .13mm/year. It was also noted that a lead oxide activity between 10⁻⁵ and 10⁻⁴ would be beneficial for technical applications, due to the formation of an exceptionally protective Cr rich scale that forms over the surface at that activity level. It seemed that the problem here was “solved” and a valid conclusion was reached. Other lead alloys could also be used on gen IV fission reactors.

The U.S. Department of Energy is interested in developing the gen IV nuclear reactor for use in the US by 2020. In response to this, several entities including MIT, Idaho National Engineering and Environmental Laboratory, Los Alamos National Laboratory, and quite a few others began testing lead alloy coolants, namely lead bismuth. It has been pointed out, [5] that lead alloys are now being reconsidered for use in nuclear fission reactors because of their inertness to water and air, unlike sodium, as well as its better shielding against gamma rays, higher boiling temperature than sodium, and successful tests on a smaller scale (the Alpha class Russian submarines).

What is lacking in the LBE case is testing on natural and forced convection to determine the heat transfer and natural circulation characteristics on a more full scale [5]. While it is rather difficult to fabricate a blanket structure in a laboratory setting for testing, even on a smaller scale, corrosion has been the main issue with LBE since 1950, and still today is one of the fundamental problems facing the advancement of the gen IV reactor. More specifically, dissolution of the blanket wall in hot regions and precipitation in the high hot-to-cold gradient zones causes blockage of components such as heat exchangers [5, 615-616] especially with steel. It is also pointed out that adding Zr, Tn, and Cr to the coolant itself or adding surface coatings like Al, Mo, Zr, or SiC on metal surfaces greatly reduce corrosion, however erratic adherence to the surface has been a problem, and irregularities on the surface tend to corrode more readily. Each of these solutions has not been fully explored.

Moving into the lithium lead eutectic, the following were found [6] to be compatible in a static environment: ZrC, TiN, MgO, and SiC. Lithium lead was first considered because it is able to both breed tritium and act as a coolant. In the study performed [6], the lithium lead eutectic

was made out of 99.26% Pb and .74% Li. 1"x 1" squares of SiC A were prepared using SiC cloth as the fiber and chemically vapor deposited SiC as the matrix. It yielded a composite that was 85% of the density of the theoretical value. These tests were static, and simply done in Mo containers at 1073K. After 1500 hours, the samples were reviewed using a scanning electron microscope. Each was found to qualitatively have "no reaction" as the only observation, revealing that the study was done rather simplistically. While conducted at a high temperature, the eutectic was not flowing over the specimens, so the results are questionable as far as their implications in the use of composites in a true reactor. Also, for fusion reactors, 83% lead with 17% lithium will be used, which poses more of a corrosion problem since the corrosion (especially with steels) is mainly associated with the activity of lithium.

Despite the previous results, in 1995 the water cooled Pb-17Li (Water Cooled Lead Lithium or WCLL) DEMO breeding blanket was selected for further study [7] for a testing representation module in ITER. The proposal desired that the blanket have a lifetime of 20,000 hours and an outlet temperature of 325⁰C, using a structural material of martensitic steel (considered for design purposes only, because corrosion has not been solved). Work has begun to design a Test Blanket Module (TBM) for use in ITER starting in 1996. The R&D section of this paper still shows ongoing testing of new materials including EUROFER 97, which was previously discussed in this paper after being tested in the PICOLO loop, and showing perhaps dangerous corrosion at above 550⁰C. Perhaps other materials are better suited for the TBM.

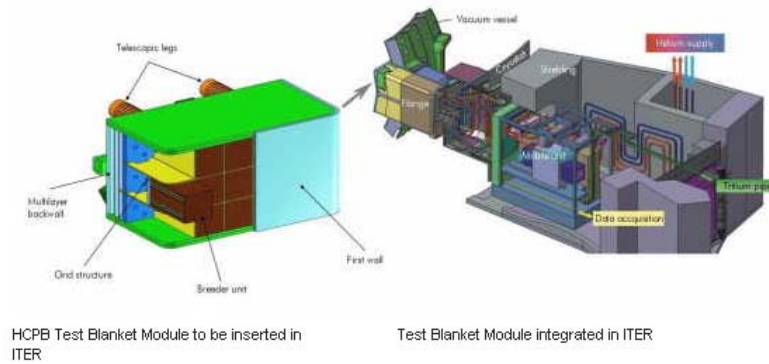


Figure 3 from Institute for Reactor Safety
 a TBM concept for ITER insertion designed by Forschungszentrum Karlsruhe, in Germany

Another proposal later, by one of the same researchers from the WCLL TBM proposal came up with a helium cooled (HCLL) proposal as well for lead lithium [8]. The paper merely showed that the “Integral TBM” one of four designs, would be implemented in the high-duty D-T phase of ITER to demonstrate a reliable and controllable mock up of a HCLL device. No design elements were disclosed in the document, unfortunately. A few researchers from Germany, however, did have some design ideas for how an advanced lead lithium blanket might be designed.

These researchers felt that SiC could be used as an insert in the lead lithium channels, utilizing the steel as a structural element for the wall, while merely using the brittle SiC as corrosion shielding. They came up with a conceptual design [2] that was planned for launch in 2001 within the EU fusion program. These inserts would allow for a high outlet temperature for the lead lithium, increasing the efficiency of a power

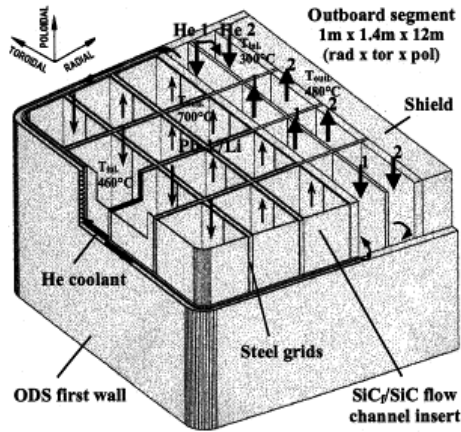


Fig. 1. Isometric view of an A-DC outboard blanket segment at torus centre.

Figure 4
Sectioned view of Helium Cooled Blanket with Flowing Lithium Lead as well as 5mm SiC Inserts, From [2]

conversion system. The reason SiC seems like such an attractive solution is due to the approximately 7 Tesla magnetic field present in the blanket region, which would require electrical insulation from the first wall structure. The proposal consists of adding 5mm thick inserts into the flow channels. The only question is compatibility of SiC with lithium lead. The study uses [6] as a reference to show that it is in fact compatible at up to 800°C. Keeping in mind that the study done in [6] was a static fluid experiment with a different % of lithium to lead, more quantitative testing is necessary to show that it is compatible in flowing lead lithium, and what modes of mass transfer come into play.

Some laboratory results have determined that thermally grown Alumina on a small piece of “commercial steel” has abnormally large grain growth after being exposed to 800°C lead lithium for 1000 hours [9]. It seems that this solution, a layer of aluminum oxide, seems to be questionable as well as untested fully.

b) Research Possibilities

From the literature review conducted, it seems that the two viable solutions for future fusion and/or fission reactors are either lead lithium or lead bismuth coolants. While steel and EUROFER have been well tested, SiC inserts, nor any other electrically thermally insulating composite has been tested. This could make composites a starting point for the liquid metal loop design at UCSD.

To test steel in a loop, the main concern is with the control of lead oxide in the liquid metal, requiring O₂ sensors, and other equipment. It seems that the loop could be later upgraded to perform such a task, but could be constructed mainly to test composite material. Also, it would seem superfluous to test steels for oxidation rates based on the activity of lead oxide in the coolant since it has been done already in [3] [1].

Of course, something that not many others have done is sample the lead lithium in the loop and chemically analyze it for concentrations of ceramic material, and in this sense gauge the ill-effects of corrosion on system components (i.e. clogging) like counter-flow heat exchangers. While corrosion of SiC would be the main focus, the analysis of the coolant itself seems a necessary task and a method for doing so will need to be developed.

SiC coupons could easily be tested in the same manner which laser struck coupons are currently examined, with an SEM. This would be necessary to see the grain size and material loss of each specimen. Also, an accurate weighing of each SiC specimen would be difficult to determine due to the porous qualities of a composite. Thus this method would also need to be devised.

Material standards would also need to be set both for lead lithium eutectic (or possibly lead bismuth) as well as for SiC. Since EUROFER has its own standards, we would merely need to verify these. A technique for making SiC test coupons would also need to be defined. Tests of at least up to 5000 hours should be conducted, but I would like the UCSD loop to at least reach 15000 hours of continuous operation, which is undocumented currently.

The loop design itself should start very simply to ensure ease of manufacture and accuracy of initial results. Once the loop is proven, further upgrades could be added to enable the additions of different types of coupons (i.e. EUROFER 97, Alumina coated steels), as well as other types of equipment (electromagnetic pumps, upgraded heaters/heat exchangers, ect...). Starting simply is the key.

Also, heavy safety measures will have to be implemented. A loop designed to run for thousands of hours unattended must have an alarm linked to security or police in case of malfunction or natural disaster. Before deciding detailed parameters and components of the loop itself, I'd like the opportunity to work on an already established loop and completely understand its operation.

Detailed CAD drawings and thermodynamic programming will be necessary to make sure that the loop is manufactured correctly and efficiently and that each component is capable of the tasks we need it to perform. Thermodynamic programming could be done by hand through MATLAB or a similar compiler, or it could be accomplished through the use of a commercially available program. Extensive hand calculations will need to be performed to ensure the accuracy of the program.

At this point labor time and costs are difficult to estimate. It depends on material cost, component (heaters/exchangers) costs, hours of labor involved, safety training, tools, software ect... These are just some of the costs associated with the project; an ongoing cost analysis before the time of construction will be necessary to remain within budget boundaries.

c) Project Scope

If these tests are successful, it could lead to a more immediate advance in fusion reactor design. The coolant liquid metal blanket is essentially the method by which we are able to attain energy from a fusion or fission reaction. Without a suitable blanket wall material, the process for developing these reactors will be hindered. Developing this unique capability for testing at UCSD will shed some light on this age old problem of corrosion with liquid metals.

This may also open the doors for UCSD to work with engineers/physicists from other universities and allow for collaboration efforts that otherwise would not have occurred. These sort of relationships would also open the doors for others students and faculty to gain new knowledge and ideas.

The results of such a project could lead to the fabrication of a Test Blanket Module (TBM) made by UCSD which would be of use in ITER. I believe such results would be innovative and if ITER tests are successful, a breakthrough in corrosion for this application.

II. Conceptual Design

To narrow the many ideas behind each of the components of the high temperature loop at UCSD, a conceptual design study was conducted. This allows the determination of which types of loop components should be focused upon and studied with greater detail and modeling.

a) High Temperature Components

The three major loop components which must be carefully reviewed given the high temperatures involved in the experiments are the heater, the heat exchanger, and the electromagnetic pump. The loop must be designed around these key components in order to keep the test section at the temperatures and flow rates desired. Design parameters such as performance, cost, and ease of manufacture were taken into consideration in the following trade studies.

i) Heat Exchanger

The heat exchanger was considered first in the design of a high temperature loop since the overall performance of the exchanger dictates the amount of heating necessary to maintain the test section temperature. Two types of exchanger configurations were considered based upon available high temperature materials including a counterflow and shell-and-tube with multiple passes. Each configuration has potential strengths and weaknesses. The following figure of merit chart describes these. The scale is from 0 to 5, 5 being the best, 0 being the worst.

Table 2, FOM for Conceptual Heat Exchanger Selection

Type of Heat Exchanger	Ease of Manufacture	Performance	Pressure Drop	Size	Proven in High Temp. Applications	Totals	Select/Reject
Concentric Tube Counterflow	4	3	3	1	5	16	Select
Multiple Pass Shell and Tube	1	4	1	4	1	11	Reject

The concentric tube heat exchanger will have many less parts to be machined and bonded than a shell and tube exchanger due to the many tubes involved. This also increases the chance for leaks and cracks. The multiple pass exchanger does have an advantage in performance due to the increase surface area by which heat can be transferred. This however does give a higher pressure drop due to frictional losses. A concentric tube design is much more unwieldy than a compact shell and tube design. However, in previous experimentation, (see the CORRIDA loop in figure 3) a counterflow design has been tested and is in use with a high temperature liquid metal flow. For now, a concentric tube exchanger is selected for further analysis, however the multiple pass shell and tube will be considered until the pressure drop for each case can be calculated more accurately in the preliminary design phase.

ii) Liquid Metal Pump

The liquid metal pump is an integral part of the high temperature loop design. It must be able to produce a pressure that will allow for the test section to be run at the desired flow rate. It must also be able to overcome the pressure drops throughout the loop associated with any bends, entrance/exit effects of other components, frictional effects and changes in height and geometry of the loop piping. It must also be able to withstand the enormously high (up to 900 degrees C or higher) temperatures involved in a liquid metal application. For this reason, a

contactless design is desirable. Traditional impeller pumps simply could not withstand the corrosion of a liquid metal and thus would chemically contaminate the loop.

Table 3, FOM for conceptual pump configurations

Type of EM Pump	Ease of Manufacture	Performance	Electrical Efficiency	Size	Proven in High Temp. Applications	Totals	Select/Reject
AC Coil Pump	2	1	1	2	4	10	Reject
Permanent Magnet Pump	3	3	5	4	2	17	Select

iii) Heater

The final component selection was the heater for the loop. It was determined last because one must know the performance of the heat exchanger before deciding how much heat energy is necessary to keep the loop running at the desired temperature. Two types of heating were considered.

Table 4, FOM Heater Selection

Type of Heater	Ease of Manufacture	Performance	Cost Efficiency	Size	Proven in High Temp. Applications	Totals	Select/Reject
Flame Impingement	2	3	5	4	3	17	Select
Induction	3	3	1	2	3	12	Reject

Induction heaters are commercially available in a variety of sizes and power outputs. They are typically used in industrial applications where heating metals within non conducting pipes is necessary. These are typically very expensive both in terms of electrical energy(\$11k/year for 11.9kW) and purchase cost(\$26k) to keep the liquid metal loop at 900 degrees C. While a flame impingement heater (natural gas burners) would be a little more difficult to set up, maintain, and reject exhaust gasses, natural gas is much cheaper. 11.9kW are necessary to maintain 900 C which can be achieved with one natural gas burner (14kW), costing \$5k/year to run. No further design was done on heating.

III. Preliminary Design

a) High Temperature Components

Now that each design has been considered on the surface, more in depth calculations were done to determine which combination of components best suited high temperature experimental loop. Models of the concentric tube heat exchanger, an AC coil type electromagnetic pump, and a permanent magnet pump were developed to gauge more precisely their performance.

i) Heat Exchanger Configuration Selection

Each configuration was analyzed by hand calculation to formulate a quantifiable overview of what performance advantages and disadvantages existed. The goal is a high ΔT and a low overall pressure drop. The ΔT needed for each exchanger is fixed at a level of (250°C) which would enable the study of precipitation and dissolution of solutes in the liquid metal. We also have a known flow rate, 1.2kg/s, to ensure that the test section of fixed diameter has a design flow velocity of 0.2m/s, similar to [3]. For a simple concentric tube

counterflow arrangement, it was determined that an exchanger of 3m in length, with a 1.5in OD inner tube and 2.5in ID outer tube could produce this 250°C ΔT.

The head loss for the inner tube and the annulus (hydraulic diameter) due to friction for a pipe can be calculated as:

$$h_f = f \frac{L}{D} \frac{(\dot{m}/\rho A)^2}{2g} \quad (1)$$

where f is the friction factor, which is calculated from the surface roughness, L is the pipe length, D is the hydraulic diameter, \dot{m} is the mass flow rate, ρ is the fluid density (~9660 kg/m³ for 600 °C lead lithium) and g is gravity. Reaction bonded silicon carbide was assumed to have a surface roughness of 110.8nm [10]. Through the use of a common Moody chart and the inner and outer tube diameters a friction factors of .015 for the tube side and .0175 for the annulus side of the exchanger were obtained. Using these assumptions, and noting that the pressure drop for an internal flow is simply:

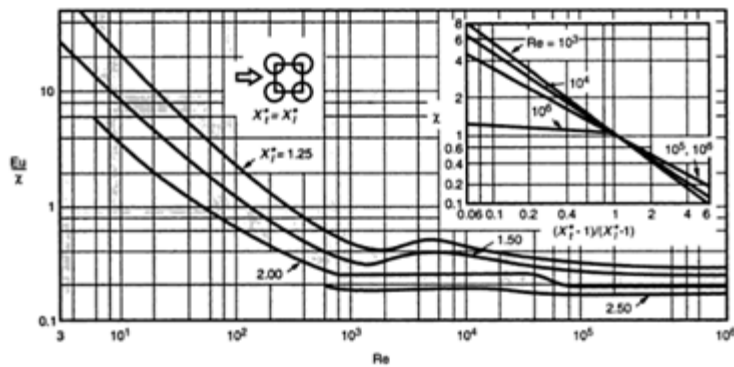
$$\Delta p = \rho g h_f \quad (2)$$

The pressure drop for this counterflow heat exchanger configuration was thus determined to be .103psi.

For a shell and tube design using commercially available 14mm OD Hexoloy tubing, the equation for pressure drop for tube bundles in cross flow becomes:

$$\Delta P = \left(\frac{Eu}{\kappa}\right) \frac{1}{2} \kappa \rho u^2 n \quad (3)$$

The $\left(\frac{E_{100}}{X}\right)$ term is determined directly from a chart in [11], shown below in figure 5. Although



The hydraulic drag coefficient for inline bundles for $n > 9$. (From Zukauskas, A. [1987] Chapter 6. In *Handbook of Convective Heat Transfer*, John Wiley & Sons, New York. With permission.)

the figure states that it is not valid if n is less than 9, this is merely being used to loosely approximate the shell side pressure drop for a tube bundle with $n = 5$. Most of the pressure drop originates on the

Figure 5
 tube side (80-90%). A MathCAD code was compiled to estimate the pressure drop by varying the number of tubes needed assuming that commercially available 14mm ID Hexoloy would be used for the tube material. A simple design with a single column of tubes is used. The shell diameter and stagger distance of the tubes are varied proportionally with the number of tubes and tube diameter. The code was iterated until the desired ΔT (250°C) was reached. This required 6 tubes, with a shell outer diameter of .196m, a length of .1m, a stagger distance of .028m, and gave a total pressure drop of 0.389 psi (.329psi tube side, .06psi shell side.) as shown in figure 6. This does not include the entrance and exit losses which would inherently occur.

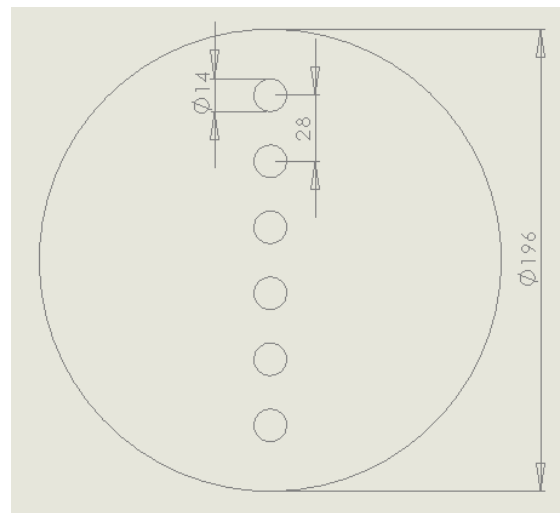


Figure 6, Shell and Tube Dimensions

However, since this pressure drop is clearly much greater than the pressure drop of the concentric tube counterflow arrangement (3 times), the shell and tube design was rejected.

Also, this decision reinforced by the difficulty of building a compact heat exchanger for a high temperature application along with the increased risk of leakage from the many bonds required to seal it as shown in table 2.

ii) Concentric Tube Heat Exchanger Model and Heater Application

A computer model of a high temperature counterflow heat exchanger was created in order to understand how the real exchanger would perform, given the properties of Pb-17Li and silicon carbide. Fluid properties vary as a function of fluid temperature and therefore have an effect on the transfer of heat radially and axially across the exchanger.

This would prove the feasibility of this design. It is a simple 1-D code (radial for concentric pipes). The pump's purpose is to reheat the cold fluid from the dump tank of a high temperature loop before it gets to a heater, and lastly the test section. This would save energy used by a heater and decrease the number of heaters necessary. For the best efficiency and performance given the flow conditions, it was decided the HX be constructed in a counter-flow arrangement with the hot fluid entering the inner tube. The study here is to determine a suitable length of the exchanger, and how much insulation will be necessary as to maximize the heat transfer rate. Also, design possibilities including a rough layout of an experimental loop will be discussed, based on the code results.

1. Abstract

A simple approach was taken to write this computer model. MATLAB 7.0 was used. Heat is transferred due to convection between the moving fluids and air as well as conduction between the fluids through the walls of the tubing as well as conduction in the fluid itself, due to the high conductivity of liquid metals. The overall result of running the code is that after the

user defines simply the mass flow rate in kg/s for each fluid, the program outputs 2 plots. These plots aided in the selection of an appropriate length for the heat exchanger. A 10m counterflow heat exchanger resulted with a 1.5" OD inner hot fluid pipe and a concentric 2" OD outer cold fluid annulus. The hot fluid enters at 900 degrees C and exits at 267 degrees C, while the cold fluid enters around 250 degrees C and is reheated to 846 degrees C. A conceptual design for the loop was also generated based on the findings of this report.

2. Assumptions

Some simple assumptions were necessary:

1. Negligible kinetic and potential energy changes
2. Fully developed conditions for Li-17Pb
3. Heat loss is in the radial direction only
4. Entering cold fluid temperature is 250 degrees C
5. Entering hot fluid temperature is 900 degrees C
6. Mass flow rate in both pipes is identical
7. Ambient air is at 25 degrees C
8. Convection heat transfer coefficient for air is 40 [W/(m*K)]

Here, a brief explanation for each assumption is provided.

Table 5, The rationale behind each assumption, carefully described.

Assumption	Why Reasonable?
1	Minimal change in height throughout the loop.
2	Overall length of exchanger is much larger than the tube diameter.
3	Dominant temperature differences occur in radial dimension.
4	Melting originates in dump tank, heat exchanger follows shortly after. Melting point of lithium lead is 235 degrees C according to [1].
5	Highest temperature necessary to perform corrosion experiments in test section. Hot fluid originates from the exit of the test section.
6	The loop as a whole will always have the same <i>mass</i> flow rate, while increasing/decreasing cross sectional pipe diameters merely affects flow velocity.
7	Temperature inside a well ventilated room.
8	A high assumption to allow for a reasonable estimate of insulation. Nominally between 25-250 for forced convection for gasses, according to [13]

3. Dimensions

Some simple dimensions had to be assumed for the purpose of getting to the task at hand, which is determining a nominal length for the exchanger based on some simple variations. To do this, the diameters of the pipes were fixed .

Pipe diameters were simply chosen: 6.35mm wall SiC tubing with 25.4mm (1”) inner pipe (OD) and 50.8mm(2”) outer pipe(OD). This was done primarily due to the availability of the pipes and a manufacturer who could conceivably construct the exchanger. 1” and 2” pipes were found readily available online at Ortech Advanced Ceramics. The dimension that is left for the program to decide is simply the length of the exchanger with concentric 1” and 2” pipes, using the above assumptions.

4. Properties

There were a few properties of each material involved in the heat transfer that needed to be defined correctly for the code to function properly. Lead lithium has properties which

vary depending on temperature, and this was taken into account. For the thermal conductivity, k , values were taken from previous UCSD ARIES team research [12] in the form:

$$k = 7.3008 + 1.96 * 10^{-2} * T \left[\frac{W}{m \cdot K} \right] \quad (4)$$

Density, ρ , and specific heat, c_p , must also be considered a function of temperature where:

$$\rho = 9990 * (1 - 168 * 10^{-6} * T) \left[\frac{kg}{m^3} \right] \quad (5)$$

$$c_p = 192.45 - 8.9487 * 10^{-2} * T \left[\frac{J}{kg \cdot K} \right] \quad (6)$$

5. Theory

Now the variables in the code must be solved for. First and foremost the convection between the hot and cold fluids, the dominant driver for heat transfer, will be evaluated. We will assume here that there is no losses to the surroundings for simplicity, but will then account for these losses later. Reynolds number in both the inner and outer pipes must be calculated. For the hot fluid in the tube, with a few manipulations using hydraulic diameters:

$$Re_D = \frac{\rho U D_h}{\mu} = \frac{4 \dot{m}_h}{\pi (D_o - D_i) \mu} \quad (7)$$

where U is the flow velocity in m/s, \dot{m}_h is the mass flow rate of the hot fluid in kg/s, $(D_o - D_i)$ is simply the difference in average inner and outer pipe diameters in meters, with μ the viscosity of the fluid in kg/(m*s). It is important to note that a Re larger than 4000 would be turbulent and that for lead lithium, it is rare even in extremely small flow rates for the flow to be laminar due to the density of the fluid. For example with a flow rate of 0.1kg/s, the Re is around 9000. The Prandtl number Pr must also be calculated. This number is affected by specific heat, viscosity and thermal conductivity of a moving fluid.

$$Pr = \frac{c_p \mu}{k} \quad (8)$$

The Nusselt number must be calculated. There are many different methods of calculating this, however only one best fit the current setup of the heat exchanger. For initial calculations the Dittus-Boelter equation was used as follows with $Pr^{0.3}$ for cooling, and $Pr^{0.4}$ for heating:

$$Nu_D = 0.023 * Re_D^{4/5} * Pr^{0.3} \quad (9)$$

It was later thought that the conductive nature and low Prandtl numbers of lead lithium render the Dittus-Boelter equation assumptions incorrect. Thus, a new correlation first implemented by Lyon [14] and used in the final revision of the MATLAB code. It is specific to turbulent, liquid metal coolants flowing in circular tubes assuming a constant heat flux along the tube wall:

$$Nu_D = 7.0 + 0.025 * Pe^{0.8} \quad (10)$$

where Pe is the Peclet number, which is calculated simply by multiplying the Reynolds number by the Prandtl number.

Next the convection heat transfer coefficient is calculated with the following simple relation:

$$h_t = \frac{Nu_D k}{D_t} \left[\frac{W}{m^2 \cdot K} \right] \quad (11)$$

For the cold fluid, the same values are needed however, the Re_D is calculated differently because of the pipe geometry. The hot fluid is in an annulus and the cold fluid is simply in a pipe:

$$Re_D = \frac{4 \dot{m}_c}{\pi D_{ps} \mu} \quad (12)$$

where \dot{m}_c is the mass flow rate of the cold fluid. From here the Nusselt number for cold fluid must also be calculated using equation 6, but with $n = 0.4$ for heating the incoming cold fluid. Equations 5 and 7 are identical for the cold fluid case. After running through these equations for both cases, calculating the convection heat transfer coefficient for both hot and cold fluids, the final step in terms of convection can take place:

$$U = \frac{1}{\frac{1}{h_{c,c}} + \frac{1}{h_{c,h}}} \left[\frac{W}{m^2 \cdot ^\circ C} \right] \quad (13)$$

U is the overall heat transfer coefficient for the counter flow heat exchanger. It will allow us to calculate a heat transfer rate and thus the length, L in meters, of the heat exchanger necessary to achieve this heat transfer rate, Q in watts. Keeping in mind that assumption 6 in table 1 tells us that the mass flow rates for the hot and cold fluids must be equal:

$$Q = \dot{m} * c_{p,h} * (T_{h,i} - T_{h,o}) [W] \quad (14)$$

Where $T_{h,i}$ is known from assumption 5 and where the hot fluid temperature flow out, $T_{h,o}$, is iterated in the MATLAB code between 539 and 310 degrees C. This makes Q into a column vector, with increasing Q as the hot fluid temperature out decreases. With Q known, we can now calculate the cold fluid temperature out, giving us a log mean temperature difference, and thus the necessary length of the heat exchanger.

$$T_{c,o} = \frac{Q}{\dot{m} * c_{p,c}} + T_{c,i} [^\circ C] \quad (15)$$

Where $c_{p,c}$ is the specific heat of the cold fluid in degrees, based on temperature from equation (8).

$$\Delta T_{lm} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,e} - T_{c,i})}{\ln\left(\frac{T_{h,i} - T_{c,o}}{T_{h,e} - T_{c,i}}\right)} \text{ [}^\circ\text{C]} \quad (16)$$

Here the cold temperature in, $T_{c,i}$, as well as $T_{h,i}$ is known from assumptions 4 and 5 respectively.

$$T_{se} = \frac{Q}{\pi \cdot D_i \cdot L} + T_{c,i} \text{ [}^\circ\text{C]} \quad (17)$$

$$L = \frac{Q}{U \cdot \pi \cdot D_i \cdot \Delta T_{lm}} \text{ [m]} \quad (18)$$

The length of the heat exchanger is now known which produces the heat transfer rate Q , assuming that there are not losses. Now we must rid ourselves of the no losses assumption. To do this, we'll use this initial length we've just calculated to estimate a Q_{leak} . We need to know how long the exchanger is to estimate how much heat is being lost to the surroundings through conduction between the SiC pipes and the insulation to the surroundings. The length calculated without losses in (13) will then be increased to accommodate for this leak of heat to the surroundings.

The heat has 3 modes of escaping our ideal model into the surroundings. They have to do with the thermal resistance of the SiC inner/outer pipes, the layer of insulation, and the convection to the surrounding air. They can be characterized as follows:

$$R_{SiC} = \frac{\ln(r_2 - r_1)}{2 \cdot \pi \cdot k_{SiC} \cdot L} + \frac{\ln(r_4 - r_3)}{2 \cdot \pi \cdot k_{SiC} \cdot L} \text{ [}^\circ\text{C/W]} \quad (19)$$

$$R_{\text{insulation}} = \frac{\ln(r_5 - r_4)}{2\pi k_{\text{insulation}} L} \text{ [}^\circ\text{C/W]} \quad (20)$$

$$R_{\text{air}} = \frac{1}{2\pi h_{\text{air}} L} \text{ [}^\circ\text{C/W]} \quad (21)$$

Where the radiuses r_1 through r_5 are defined according to the following diagram:

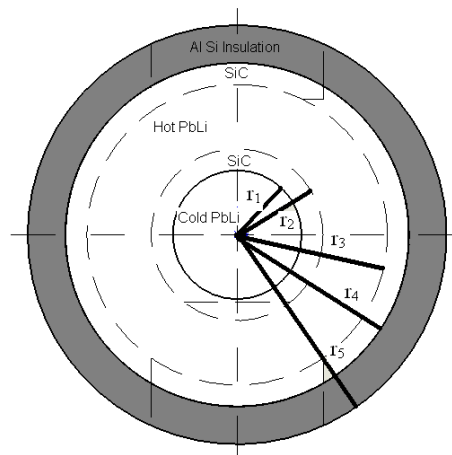


Figure 7, A cross sectional view of the counterflow heat exchanger, hot fluid annulus

Alternatively the fluids might be arranged to flow like figure 1b illustrates:

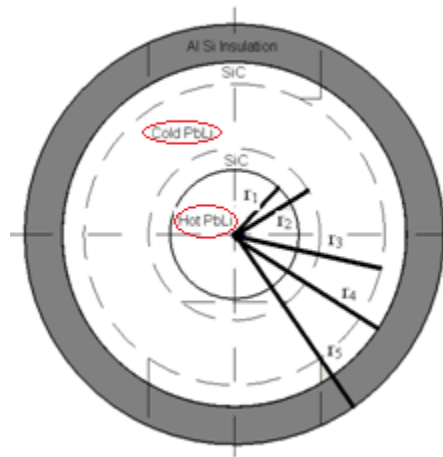


Figure 8, cross sectional view of the counterflow heat exchanger, cold fluid annulus

Both cases in Figure 7 and Figure 8 will be considered in the MATLAB coding.

Now the heat escaping to the surroundings can be characterized as:

$$Q_{\text{leak}} = \frac{T_{\text{Ht}} - T_{\text{air}}}{R_{\text{insulation}} + R_{\text{air}} + R_{\text{SiC}}} [W] \quad (22)$$

where from assumption 7, T_{air} is equal to 25 degrees C. If we subtract off the Q_{leak} from the ideal Q calculated earlier, we can run through equations 11, 12, and 13 again with a Q' which now takes into account the losses to the surroundings thereby calculating a length L based on those losses.

$$Q' = Q - Q_{\text{leak}} [W] \quad (23)$$

All the information necessary has been given except the flow rate, the thickness of the insulation, the hot and cold fluid temperatures at the outlet of the exchanger, and the length of the exchanger. They must be calculated with the given equations as described. This would require us to iterate through or assume some values. Flow rate and hot fluid temperature fluid out was thus iterated over these equations, while assuming different thicknesses of insulation, producing some interesting results.

6. Results

Since the loop at FZK in Germany tested 8mm specimens at 0.22m/s flow velocity in a 16mm test section producing a mass flow rate of about 1.2 kg/s , the results here will focus on that flow rate, while other flow rates were tested for heat transfer ability. The code in MATLAB produces three graphs as previously described in the abstract. These graphs were produced for different flow conditions with different thicknesses of an Alumina Silica insulation. It appears that from the following results, a 10 meter long heat exchanger and a layer of 1.25" AlSi insulation would be sufficient to reheat the cold fluid to 824 degrees C for a 1.2kg/s flow rate. The slower flow rate, 0.2 kg/s, the heat exchanger could reheat the cold fluid to about 817 degrees in only 2 meters, as shown in figure 3.

Now, with a flow rate of 0.2 and 1.2 kg/s, the following graphs were produced based on the solutions of the iterated values discussed previously, and a flow arrangement analogous with figure 1a.

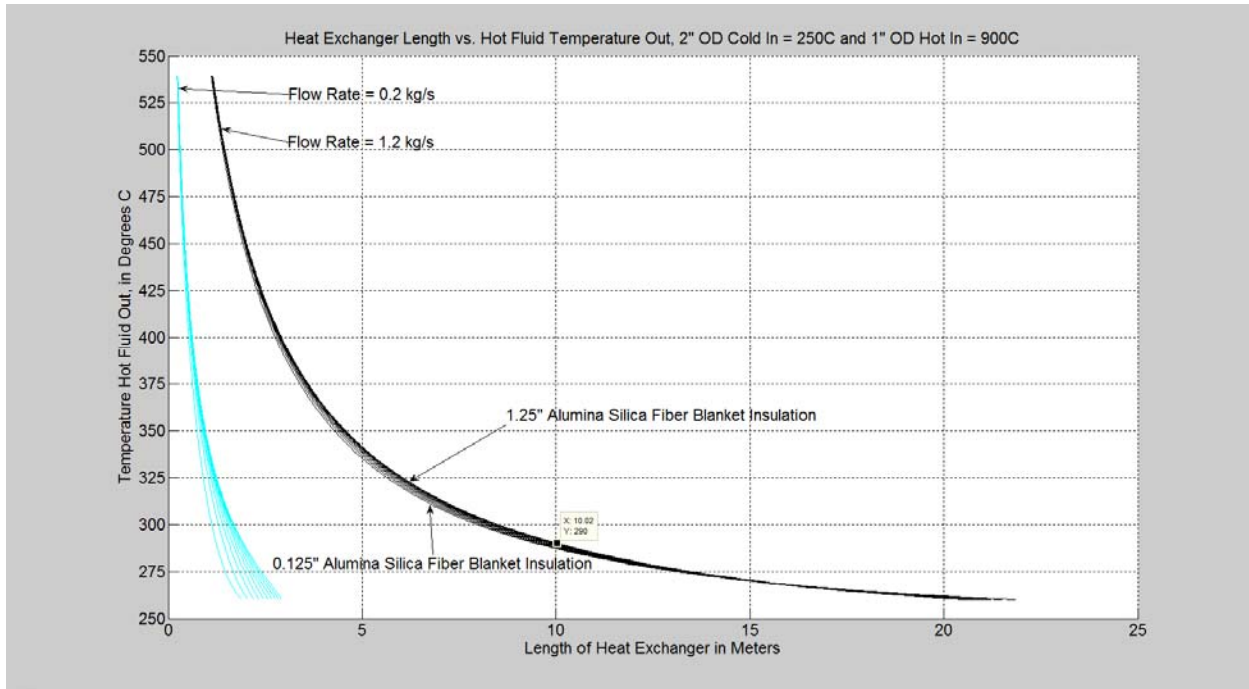


Figure 9, Hot fluid temperature out vs. length of the exchanger, flow rate = 0.2 and 1.2kg/s

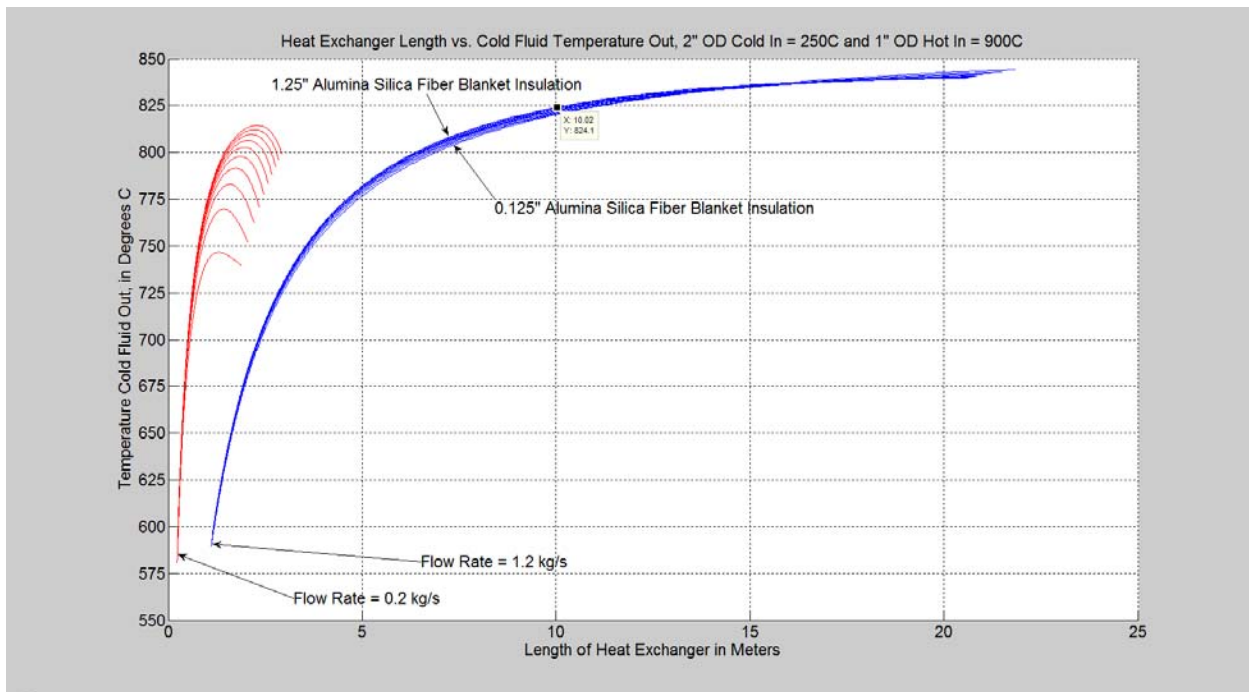


Figure 10, Cold fluid temperature out vs. length of the exchanger, flow rate = 0.2 and 1.2kg/s

By changing the hot fluid to the inner pipe and the cold fluid to the outer pipe as shown in figure 1b, and by increasing the inner pipe diameter to 1.5 inch OD from 1 inch, I was able to get an even higher performance exchanger as can be seen by the following figures in a final comparison, with the improved performance shown. Changing the pipe diameter had the most dramatic effect.

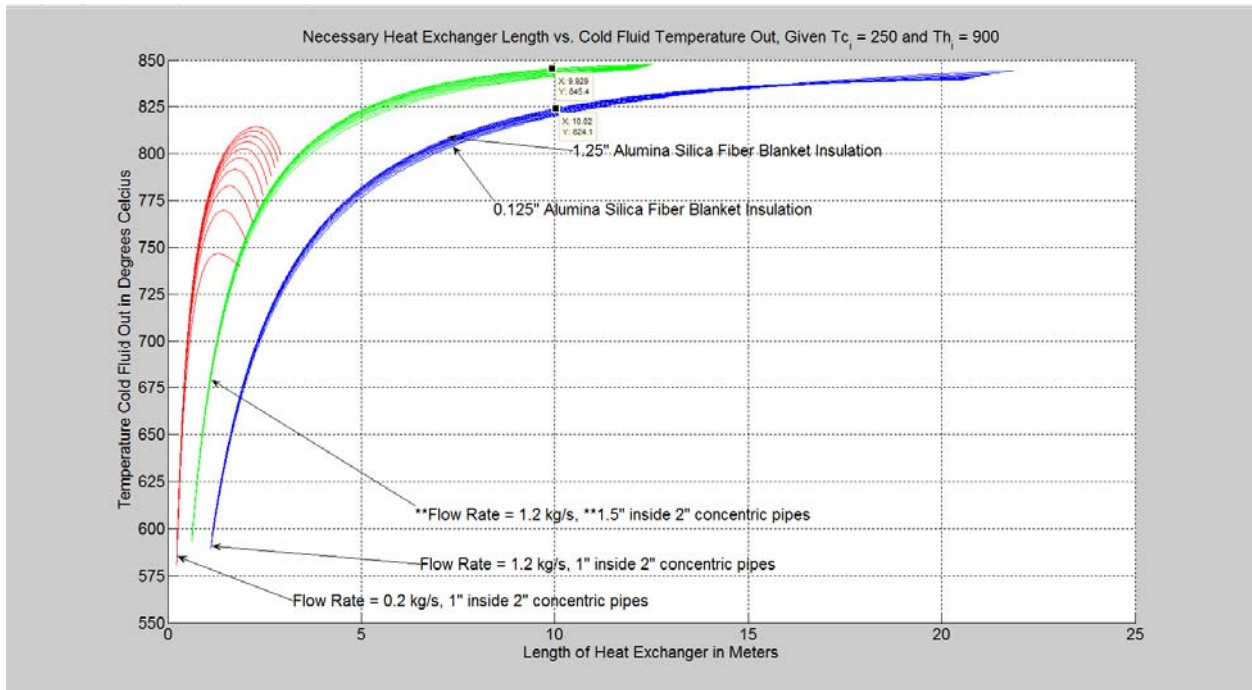


Figure 11, Figure 4 Improved Exchanger Performance Due to Pipe Diameter Change, Cold Fluid Out

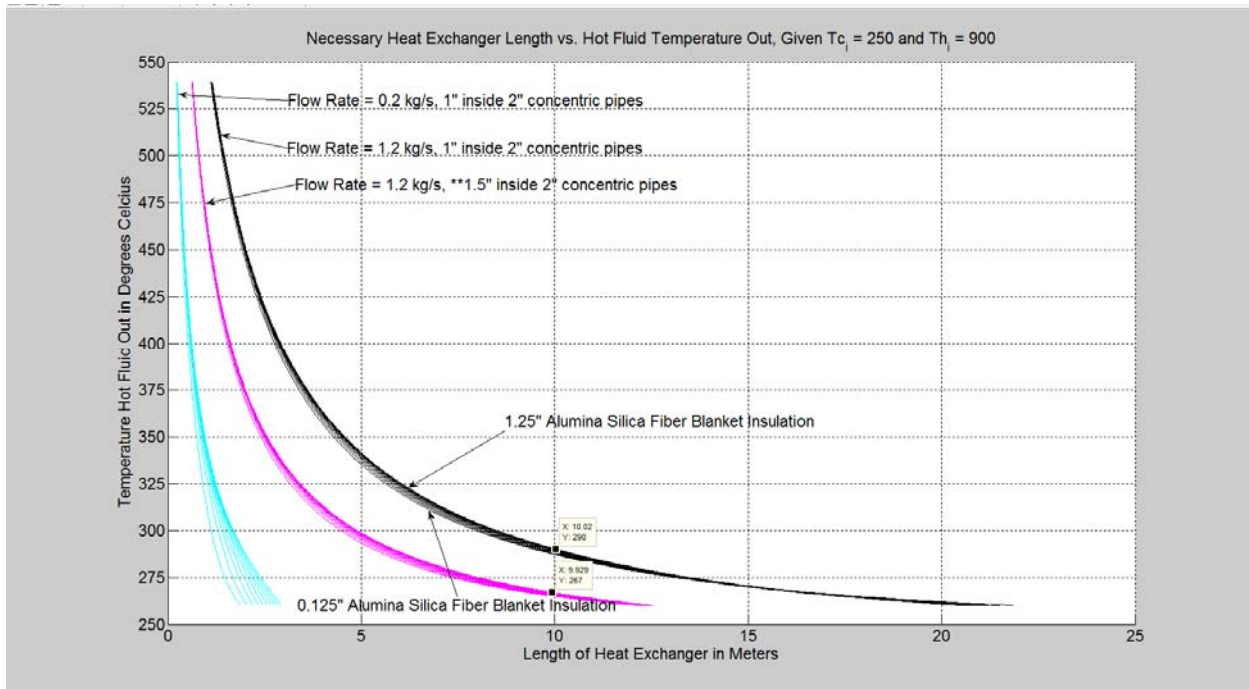


Figure 12, Improved Exchanger Performance Due to Pipe Diameter Change, Hot Fluid Out

7. Discussion

The above figures are very useful in designing the exchanger by showing trends of what can be accomplished by varying a few of the parameters. These plots show that at least 1.25" of alumina silica insulation is needed. It also shows us that the longer the exchanger the better, at least in terms of the additional electrical energy required to heat the fluid back up to 900 degrees once it has left the exchanger. A slower flow rate would allow a shorter exchanger, however a slower flow rate would not allow for the flow velocities needed in the test section assuming it at least the same size as FZK's, which is quite small to begin with. 1.2kg/s is the bare minimum flow rate to conduct good compatibility experiments according to FZK [3]. From

the results of this code, the following are recommended for the design of the exchanger for maximum performance:

Table 6, Heat Exchanger Final Specs

Heat Exchanger Specifications	
Inner, Hot Fluid Pipe Diameter	1.5 in.
Outer, Cold Fluid Pipe Diameter	2.00 in.
Hot Fluid Inlet Temperature	900 C
Hot Fluid Outlet Temperature	267 C

Cold Fluid Inlet Temperature	250 C
Cold Fluid Outlet Temperature	846 C
Length	10.0 m
Insulation Thickness	1.25 in.
Flow Rate	1.2 kg/s
Total Q	134.2 kW

Table 7, Heat Exchanger Pipe Specs

Heat Exchanger Flow Specifications

Pipe Geometry	Diameter	Re	Pr	Pe	Nu	μ (Pa-s)	$T_i \left(\frac{kW}{m^2 \cdot K} \right)$
Inner, Hot Fluid Pipe	1.5 in.	233550	0.0014	323.4420	9.5455	1.8742e-004	6.8167
Cold Fluid Annulus	2.00 in.	98755	0.0029	288.5531	9.3234	1.8742e-004	8.9569

8. Costs/Manufacture

The cold fluid which has recuperated heat from the exiting hot fluid must be reheated from 846 degrees Celsius to 900 degrees before it can enter the test section where experiments

will be conducted. Using equation 10 one can calculate that the heat necessary to get the fluid from 846 to 900 degrees C is 11.9kW. This can be accomplished with a variety of RF induction heaters, which can selectively heat the lead lithium but not the silicon carbide piping. The loop will take time to “start up”, in other words, get up to temperature. The time needed will depend greatly on the amount of maximum heat the induction heater can provide.

Continuously, however, the induction heater will only need 11.9kW after the start up time. To ensure a fast start up time, I would recommend the 60kW induction heater from the Superior Induction Company as a reliable source of energy. The heater itself runs between \$26 k and \$27k. Typical electrical costs in San Diego, CA are 10 cents per kWh. If the heater ran for a year continuously, this would amount to 104.2MWh, or cost approximately \$10,424 per year to run, plus the start up time cost which the heater would typically run hotter, and plus the costs of lab

equipment collecting data.

I envision the heat exchanger to look something like Figure 13. The dark layer around the outside is the alumina silica insulation. I have

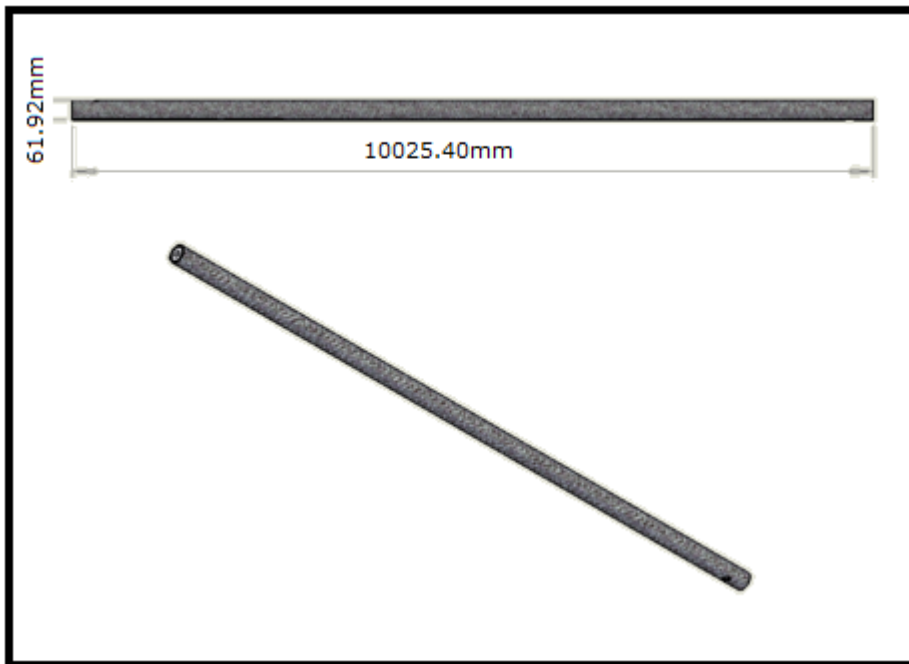


Figure 13, CAD Drawing of Heat Exchanger

spoken with Zircar Ceramics who can provide us with the “ASH” material.

I have spoken with an engineer from graphitestore.com who believes that so long as the curing process is followed closely (requiring baking), that Ceramabond 503 compound would withstand our high temperatures and be able to bond SiC tubing. Alternatively, Ortech ceramics says they are able to fire SiC tubes together, and that while the bond may not be as strong as “normal” it would hold temperature. Ceramabond 503 costs \$92.81 a pint. The engineer at graphitestore.com also recommended that for an easier curing time, but not so quite ideal bond, an alumina nitride adhesive, Ceramabond 865, would also work and survive the same temperature. It is a little more expensive at \$124.82 a pint. I recommend ordering the concentric SiC tubes, bonding caps on the ends and drilling holes in the outside of the outer tube for inlet and exit of the cold fluid.

The SiC heat exchanger is feasible after contacting some of the vendors and talking with them about these different methods of bonding and adhering. The cost of electricity was a significant factor in choosing components for this project. One last fouling possibility is the “shorting out” of the heat exchanger through the SiC down the length of the tube axially. This could pose a serious problem. I have roughly calculated that with equation 19, where A is the cross sectional area and x is along the length of the heat exchanger in meters.

$$Q = -k_{SiC} * A * \frac{dT}{dx} \text{ [W]} \quad (24)$$

With a temperature difference of 626 degrees C, the heat transfer is 12.26W. This is a trivial amount of heat lost compared to the 134kW of heat moving radially, further solidifying my previous assumption. This assumption holds in [14] for Peclet numbers greater than 35. The lowest Pe as shown in table 3 is 288.

iii) High Temperature Pumping Components

Upon initial conceptual analysis, the AC coil pump design is rejected. Its electrical efficiency was deemed poor in comparison with the permanent magnet pump. The permanent magnet pump has cheap raw materials, is easy to construct, and is much more compact than the permanent magnet pump. It also does not have coils that need to be cooled due to the ohmic dissipation of heat. However, since the FOM in Table 3 is close, and research was done on both methods, both are considered here.

1. AC Coil Pump

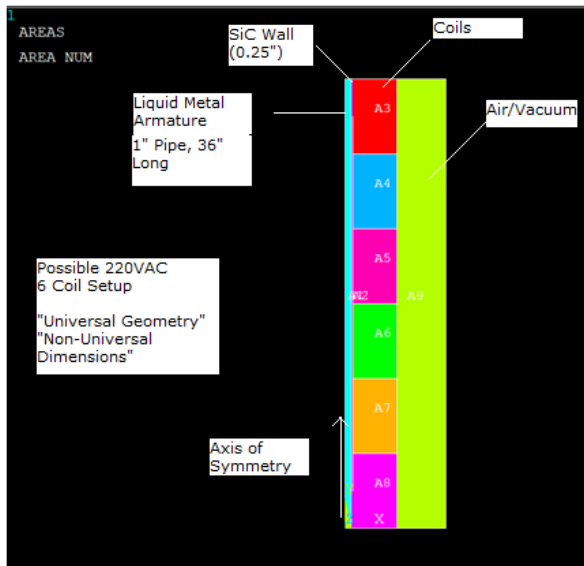


Figure 14, Ansys model of 3-phase coil pump and B' and one to C and C'. This would, in turn, induce eddy currents into the liquid metal and the Lorentz term ($J \times B$) in the magneto-hydrodynamic equation of motion

This AC Coil pump works by using 6 wound coils of wire around a pipe containing a liquid metal fluid. 3-phase 60 Hz alternating current would then generate a moving sine wave of magnetic field. In Figure 17, A and A' are wound in the opposite direction of one another as are B and B' and so on. Then one of AC three phase wires is connected to A and A' one to B

would produce a force on the liquid metal. This idea has been used in the design of electromagnetic accelerators, and would be simple to build. A 2-D model using Ansys EM

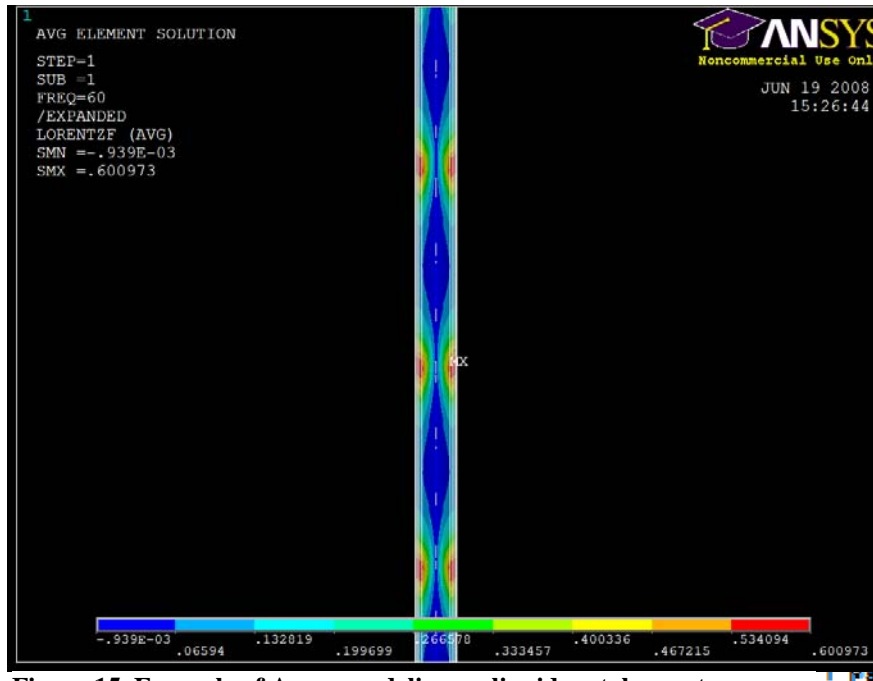


Figure 15, Example of Ansys modeling on liquid metal armature, 60Hz, JxB forces calculated

module was constructed to determine the electrical efficiency of such a design. Figure 14 shows what the model geometry looks like prior to analysis. The coils would be wound such that

(25)

the red coil would be wrapped opposite the green, blue opposite the orange, ect..., such that each coil is 60 degrees out of phase with the one adjacent. The model is rotationally symmetric about the axis shown. The code was then run and the current density (A/m²) was varied, and noting the change in force produced on the liquid metal armature. The pressure could then be calculated from the force and the area, as well as the amount of kWh of electricity necessary to run such a device for a year at 2 psi. This pressure value was chosen nominally to overcome the pressure drops in the heat exchanger, other components and loop piping. The liquid metal armature is selected in the model, meshed, and analyzed, as shown in Figure 15. It became increasingly apparent that an exorbitant amount of electricity would be necessary to run a pump in this configuration which is discussed further below. This seemed like a reasonable finding, since this type of

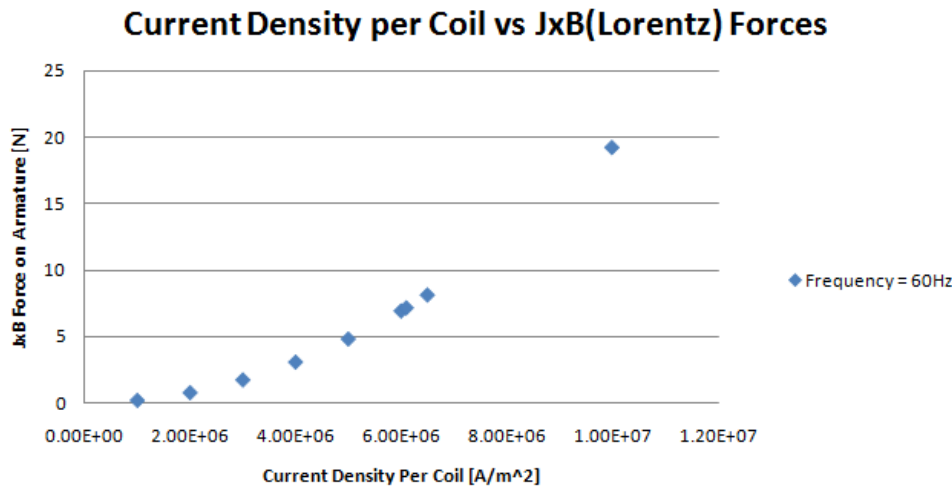
design is used in solid target accelerators, where large amounts of electricity for short amounts of time are necessary. However, for a long term application where the pump will run for a year or more, the high cost and operating conditions (i.e. temperature of coils) become problems. Iteration with a model was necessary to determine the maximum amount of current allowable in the coils before overheating.

A Mathcad model was constructed to determine the maximum current density attainable

Input Parameters		Outputs Parameters	
Magnet Wire Diameter	35AWG, 0.08mm	Ohmic Heating Loses	96.57 W
Voltage on Coil	220V	Coil Current	2.634 A
Length of 6 coils	.25ft	Coil Surface Temp	36.15 °C
ID of Coil Wrap	.5in	Chilled Water Flow Rate	3.435 $\frac{kg}{s}$
Chilled Water Pipe OD	1.00in	<u>Current Density</u>	864.111kA/m²
Maximum Wire Temperature	200C (high temp insulation)	<u>Total Number of 6 phase coils to Meet 2psi Demand</u>	7

Table 8, Mathcad model output for coil pump cooling/current requirements

considering the ohmic heating losses incurred by wrapped coils. To improve performance, a cooling water jacket of 1in diameter was used around the coils.



The pressure of the cooling water was measured in the EBU II utility closet and the cooling water flow rate in the table was calculated. With the

Figure 16, Coil Pump Ansys Iteration Results

forced convection cooling, the coil surface temperature is kept reasonably low. However, one is only able to generate 1/7th the necessary pumping head with one of these devices since the Ansys model above shows that $6 \cdot 10^6$ A/m² is necessary to generate 2psi of

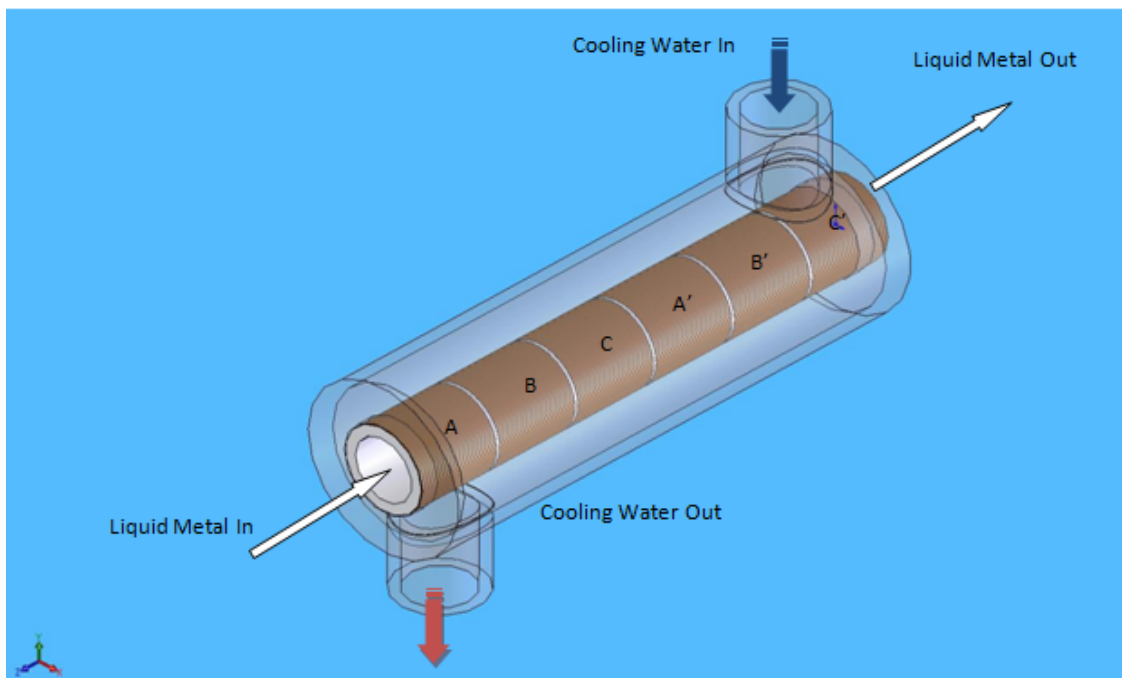


Figure 17, Conceptual Drawing of Coil Pump, 1 3-Phase Section

Number of 6-Coil Phases	Pumping Pressure (psi)	Pumping Power (W)	Yearly Cost

7	2.12	675.99	1480.36
8	2.77	772.56	1691.84
9	3.50	869.13	1903.32
10	4.32	965.7	2114.8
11	5.22	1062.27	2326.28

head for a pipe area of 0.8in² corresponding to a ½” ID diameter SiC pipe. This is shown in Figure 16.

Other frequencies were also analyzed, but 60Hz is the most commonly available. Also, higher frequencies did not have significantly higher results until reaching 400Hz. These findings show that a total of 7 of these 3” long wrapped coil structures like that of Figure 13 in series would generate the needed 2psi for the loop. This would result in 7 times the ohmic heating losses (about 700W), and thus 7 times the amount of electricity reported in the table to run the loop. If this was the case and electricity cost was \$0.25 per kWh in San Diego, CA, the cost to run for one year would be \$1,480. If the pressure needed was higher, or if higher test section flow rates were desired, the following chart shows the pumping power and yearly costs respectively.

Table 9, Cost of running coil pump

2. Permanent Magnet Pump

The coil pumping method is possible, however a more electrically efficient way of pumping high temperature liquid metals was also investigated and showed promising results. Rather than generate the moving magnetic field to produce a pressure gradient using electricity, one can generate the field from facing the opposite poles of permanent magnets together to get the field. Then, by moving the magnets in a cyclical pattern with the fluid between the pair of magnets, one can generate Lorentz ($J \times B$) forces in the liquid metal, similar to the coil pump. A 2D Ansys model was constructed using a pair of NdFeB magnets. The magnets are moved with varying velocities while the armature (liquid metal) is kept stationary. In the physical design, the magnets could be placed on a rotating disk, with pairs facing

each other, and the fluid between the pairs. Then the Lorentz forces are generated simply by rotating the disk. A conceptual drawing is shown below.

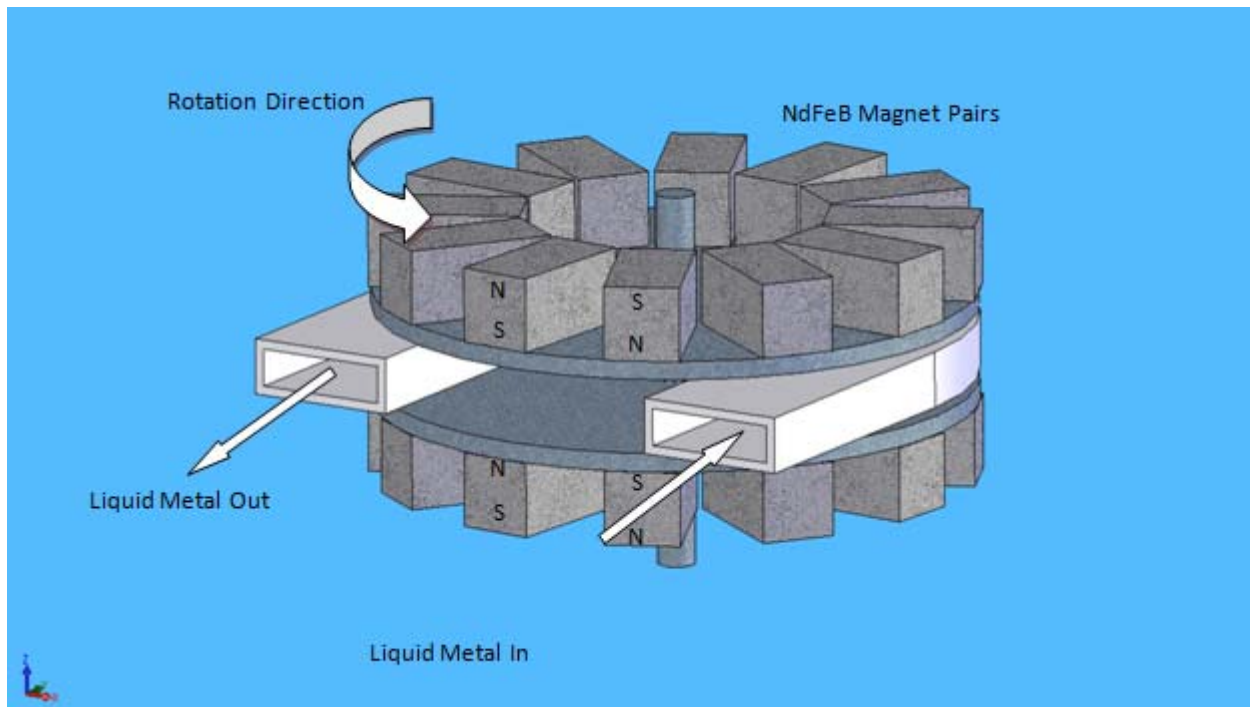


Figure 18, Conceptual Drawing of Permanent Magnet Pump

In the case of the permanent magnet pump, the power input required will simply be the power to keep the magnets rotating at the desired speed. This can be done with a simple AC electric motor commercially available, and will cost 70% less to run per year than the AC coil pump.

A steady state 2-D Ansys model was constructed to analyze the potential of such a device. Two 1in by .5in NdFeB were placed opposite one another separated by a SiC wall of .125in on each side and a fluid gap of 1in as shown in Figure 19. Only the right (symmetrical) half of the model is shown for simplicity. Each of the listed parameters can be changed, T_{fe} is the iron back element thickness, T_{mag} is the magnet thickness, T_{ins} is the insulation thickness, T_{cw} is the channel wall (SiC) thickness, and T_c is the liquid metal channel thickness. Each was changed independently to find the best combination yielding the most Lorentz force output. The liquid metal (red) was selected and set with various

velocities and the Lorentz force was analyzed. In actuality, the magnets would be moving relative to the liquid metal, however the same force would be generated on the liquid metal in either case.

The liquid metal elements are simply conducting elements with the same electrical conductivity

as Pb-17Li, which varies with temperature.

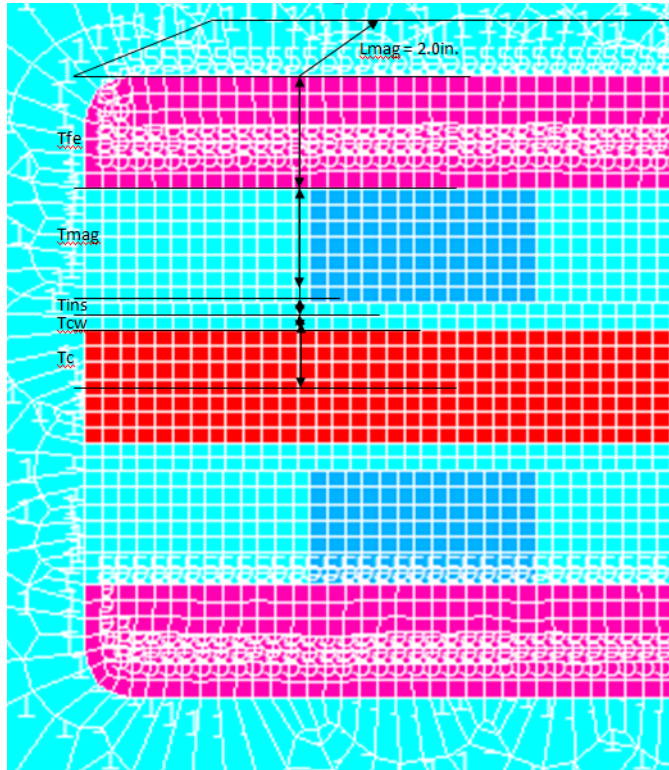


Figure 19, Variables in ANSYS Code for Permanent Magnet Pump

This test was conducted at the temperature where the pump would be placed in the loop,

(around 250 C). This is also advantageous

because the magnetic breakdown

temperature is very low [16] (working

temperature 120C, Curie Temperature 300C).

The B-H curve for NdFeB was procured from

[16]. Because of this air cooling may be

necessary if the loop is to be run at

temperatures higher than 250 C at the pump

location. Other magnet cooling methods

could easily be conceived if necessary.

The magnetic field that the liquid metal passes through and the induced currents in the transverse direction are both shown below. The cross product of these two ($J \times B$) produces the Lorentz forces necessary for pumping.

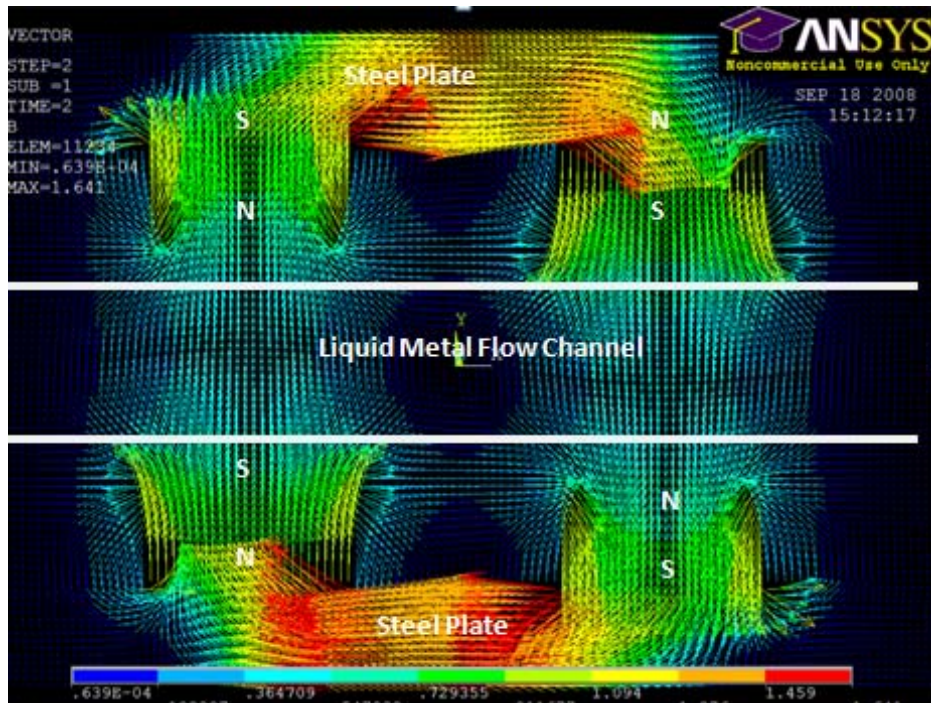


Figure 20, Model Vector Plot of Magnetic Field Between Pairs of Magnets with Iron Back Elements

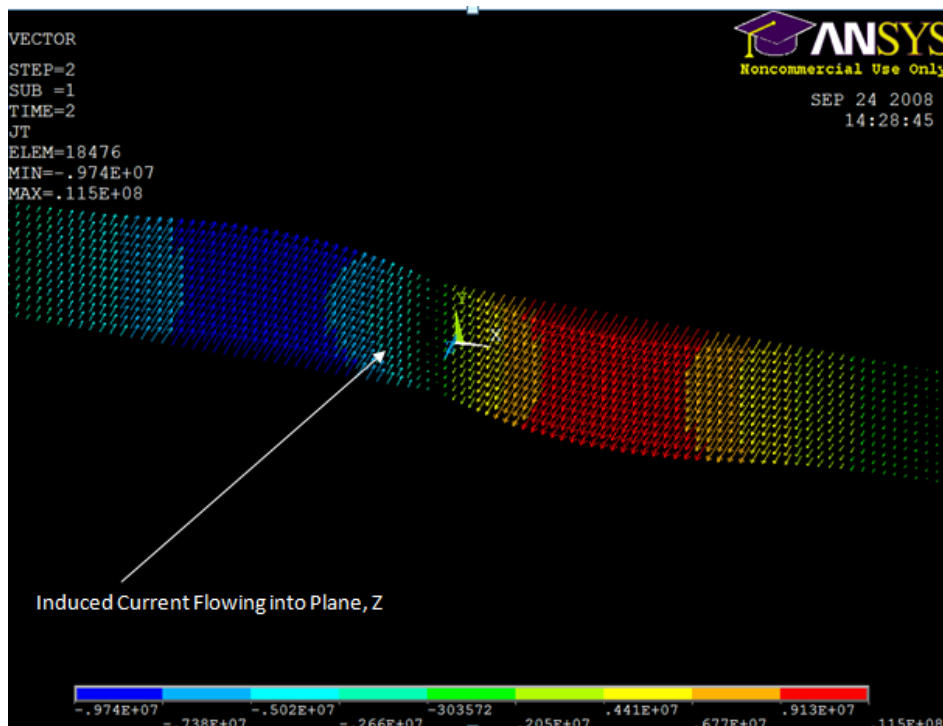


Figure 21, Current Induced by moving fluid in relation to magnetic field, velocity = 5m/s ≈469 rpms

After running through the simulation a few hundred times, it was found that having thin channel walls (see Figure 19), higher magnet velocities and a thinner flow channel all produce trends of higher pumping pressure. However, one must be careful not to make the flow channel so thin that the frictional effects and thus pressure drops become large. The following final configuration was able to theoretically produce 235.2psi at 469rpms. This angular speed assumes a design like Figure 19, where 12 magnet pairs are used.

NdFeB Magnet Dimension	.5in x 1in x 2in
Tc (Figure 19)	.5in
Tins(Figure 19)	.3175in
Magnet Velocity	5m/s or 469rpms for
Number of Magnet Pairs	12
Force Produced	1045.42 N
Subsequent Pressure Head	235.2psi

Table 10, ANSYS Iteration Results for Permanent Magnet Pump

This does not take into account the pressure head losses for entrance/exit effects, changes in channel geometry (i.e. bends), nor frictional losses, although the results are substantial: the permanent magnet pump is an efficient method of pumping a high temperature liquid metal and produces a cheaper pressure increase than with an AC coil method, so long as the Curie temperature of the magnets is not reached. These losses would need to be estimated, but the cost savings is substantial if the pump can merely produce 2psi

after all losses accounted. To get the magnets moving, A 0.5hp (373W) AC motor from McMaster-Carr costs around 100 dollars to purchase new, and assuming that after the magnets reached a velocity of 5m/s (or the motor reached 469rpm) that the aerodynamic drag on the

Magnet Velocity(m/s)	Pumping Pressure (psi)	Pumping Power (W)	Yearly Cost
5	235.2	372.8	\$816

Table 11, Specs for a Permanent Magnet Pump, as per above dimensions

magnets spinning and the friction on the bearings of the motor would take the full .5hp to overcome

(they would not), the total cost would be \$816 per year to run. Comparing this sort of cost to Table 9,

the pressure to cost ratio is better for the permanent magnet pump by over 200 times.

IV. Conclusions

The loop components have now all been successfully analyzed and decided upon. A loop with a flame impingement heater, concentric tube heat exchanger, and permanent magnet pump could potentially be realized at UCSD. Due to the high temperatures involved, a room off campus seems the most suitable place for such a device. Also, a remote monitoring system will need to be installed such that the system can be stopped in case of an emergency, as tests could go as long as a year or more. This could be done using labview in combination with PC Anywhere, a program allowing you to log into a computer with remote access. This research will provide very useful data to the designers of future fusion power plants.

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