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ABSTRACT

Active magnetic regenerative refrigeration (AMRR) systems have emerged as a possible alternative to vapor compression refrigeration cycles. The University of Wisconsin has developed a one-dimensional numerical model of an AMRR system. The thermal interaction between the fluid and the bed material is captured using a correlation for the Nusselt number as a function of the Reynolds number and Prandlt number. The model is flexible with regard to the matrix configuration. However, experimental data are available for a regenerator composed of a packed bed of spheres. The observed discrepancy between the model results and these experimental data can be explained by deficiencies in the correlations that are available for flow through a packed bed of spheres. These correlations were developed primarily for low Prandlt number heat transfer fluids (i.e., gases) whereas the heat transfer fluid used in AMRR systems are liquids with relatively high Prandlt number. The available correlations from the literature are no self-consistent at high Prandtl number. Therefore, this paper presents the results of experiments with a single blow passive packed sphere regenerator that are used to measure the heat transfer coefficient with high Prandtl number fluids. These measurements are used to develop a new correlation for the Nusselt number in a bed of packed spheres that is accurate at high Prandtl number and therefore useful for modeling AMRR systems.

1. INTRODUCTION

The concept of magnetic refrigeration has existed since 1933 when Giauque and MacDougall (1933) used an adiabatic demagnetization refrigeration (ADR) cycle to break the 1 K temperature barrier. The active magnetic regenerative refrigeration (AMRR) overcomes the limited temperature span associated with the ADR cycle and therefore enables magnetic refrigeration for near room temperature applications. A proof of concept AMRR cycle was first developed by Brown (1976). Since then, there have been several other AMRR system prototypes constructed and substantial work in this area. A review of recent developments can be found in Engelbrecht *et al.* (2007). With the increase in interest surrounding AMRR system development, there has also been in interest in developing computer models that can predict their behavior.

1.1 Magnetic Refrigeration Thermodynamics

The driving phenomenon for magnetic refrigeration is the magnetocaloric effect, which is the entropy change of a material that is induced by magnetization. The magnitude of the magnetocaloric effect is highly sensitive to temperature and for most materials a large magnetocaloric effect only occurs near the Curie temperature where the material changes from a ferromagnetic state to a paramagnetic state. Isothermal magnetization of a ferromagnetic solid aligns the magnetic moments of the material which reduces the magnetic portion of the material's entropy. Similarly, in the reverse process, isothermal demagnetization returns the material to its zero field entropy. An analogous process would be the isothermal compression and expansion of a gas. When a gas is isothermally compressed, the positional disorder, and therefore entropy, is reduced. Isothermal expansion of the gas then returns the entropy to its original uncompressed value. The mechanical work required to compress the gas is analogous to the magnetic work transferred to a magnetocaloric material, as shown in Figure 1.

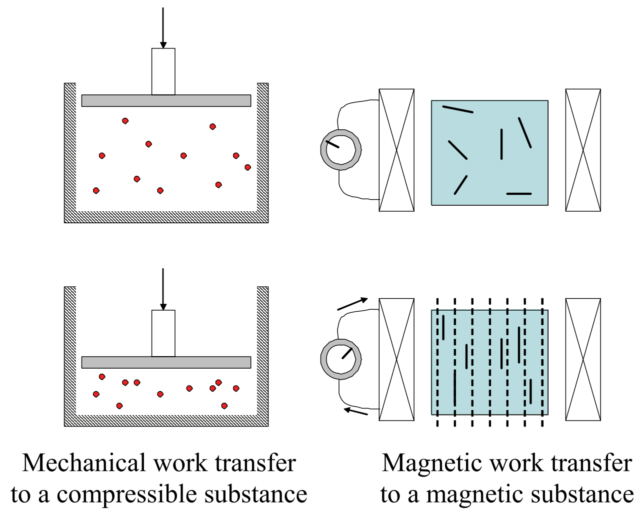

Figure 1: Analogy between magnetic work and mechanical work

1.2 Description of an AMRR Cycle

During an AMRR cycle, a porous media is exposed to a time varying magnetic field and a time varying flow of a heat transfer fluid. There are several mechanical realizations of this process, but they are all similar to the example shown in Figure 2. The process begins at state (1) where a porous bed of a magnetocaloric material initially has a temperature distribution that extends, approximately, from the cold reservoir temperature (T_C) at one end to the hot reservoir temperature (T_H) at the other. As the cycle moves from state (1) to state (2), the bed is magnetized with no fluid flow, causing the entire bed to increase in temperature due to the magnetocaloric effect. The new temperature distribution is shown qualitatively in Figure 2, notice that the highest temperature in the bed has risen above the hot reservoir temperature. Heat transfer fluid then flows from the cold reservoir to the hot reservoir during the cold-to-hot blow process. The bed temperature decreases as it comes into contact with the colder fluid. The temperature distribution at the conclusion of the cold-to-hot blow process is shown qualitatively in Figure 2 as state (3). During this process, fluid at a temperature that is greater than T_H is enters the hot reservoir and therefore a heat rejection is accomplished. Next, the bed is demagnetized which causes the temperature of the entire bed to decrease. The temperature distribution at the conclusion of the demagnetization process is shown qualitatively in Figure 2 as state (4). Notice that the temperature at the cold end is below T_C. Therefore, during the ensuing hot-to-cold blow process, the heat transfer fluid flows into the cold reservoir at a temperature that is below T_C which induces a cooling load. At the conclusion of the hot-to-cold blow process the temperature of the bed has returned to state (1) and the cycle repeats.

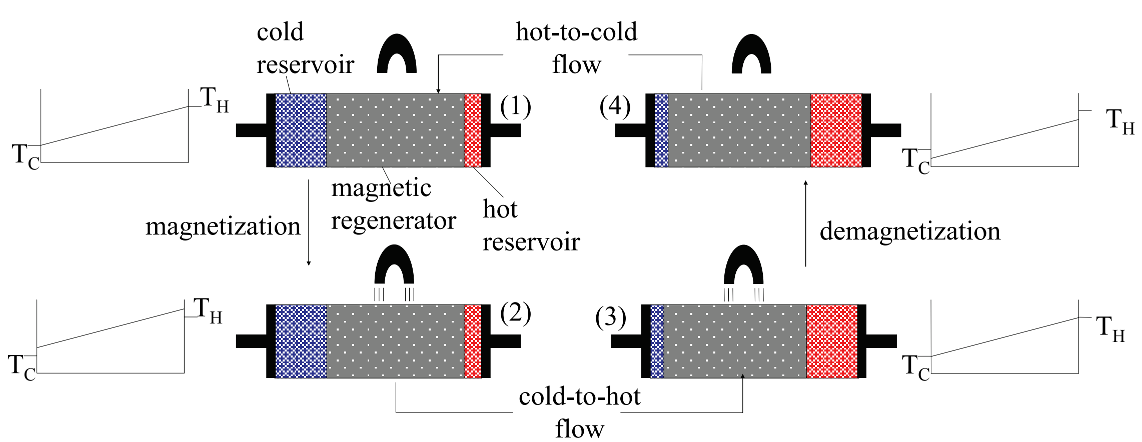

Figure 2: Active Magnetic Regenerative Refrigeration Cycle

1.3 AMRR System Configurations

There are several different AMRR system configurations, and there have been several prototypes constructed. One configuration involves using a stationary bed and varying the magnetic field by varying the current to a solenoid. This type of system configuration is most practical at cryogenic temperatures where superconductors can be used to handle the large currents that are required to supply a large magnetic field. These systems are not practical near room temperatures because the power required to maintain the superconducting magnet temperature exceeds the cooling power obtained from the AMR. AMRR systems for use in residential or other small scale applications will likely use more practical permanent magnets to generate the required magnetic field. Permanent magnet systems require some relative motion of the bed with respect to the magnet. The bed may physically move into and out of the magnetic field or the magnet may move with respect to the regenerator bed. One example of a rotating bed, stationary magnet system is the rotary AMRR experimental apparatus built by Astronautics Corporation and described by Zimm *et al.* (2006). The rotary device is composed of a regenerator wheel that rotates in the presence of a stationary permanent magnet. The regenerator bed contains six individual regenerator beds that are arranged back-to-back, i.e., the hot ends of each bed share a separating wall. These six beds are arranged on a wheel that rotates through a 1.5 Tesla $Ne_2Fe_{14}B$ permanent magnet. Each regenerator bed completes one AMRR cycle during one revolution of the regenerator wheel. A photograph of the Astronautics device is shown in Figure 3.

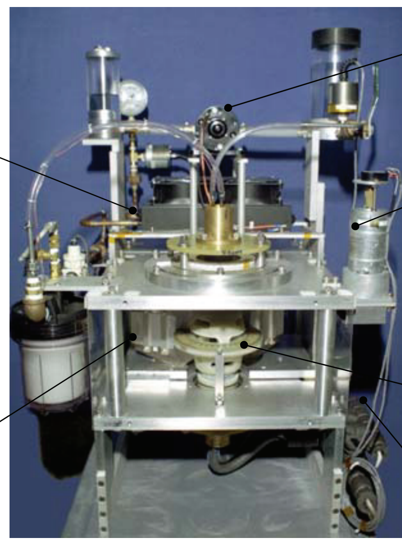

Figure 3: Rotary AMRR experiment at Astronautics

2. MODELING AMRR SYSTEMS

2.1 The University of Wisconsin AMRR Model

The AMRR model developed at the University of Wisconsin is a 1-D transient, numerical model that calculates the periodic steady state temperature of the regenerator material and the heat transfer fluid during one complete cycle (Engelbrecht, 2005 and Engelbrecht 2008). The temperatures as a function of position and time together with the prescribed mass flow rate allows the cycle performance parameters such as cooling load and coefficient of performance to be calculated. The model has been implemented in MATLAB, and is explicit with respect to time and implicit in position. A one dimensional model was used because two-dimensional effects in well-designed regenerators are small and there is a significant savings in computational time required by a one-dimensional as compared to a two-dimensional model. However, because the model is 1-D, the Navier-Stokes and 2-D energy equation for the fluid is not solved explicitly. Rather, correlations from the literature are used to calculate the axial dispersion parameter, the Nusselt number, and the local friction factor in the regenerator and these quantities are used to account for conduction and mixing, the fluid-to-solid heat transfer interaction, and pressure drop, respectively. Curve fits to experimentally determined material property data are used to calculate the specific heat, density, enthalpy, viscosity, and thermal conductivity of the fluid as functions of temperature. The heat transfer fluid is assumed to be incompressible, therefore the mss flow rate does not change spatially and the amount of fluid entrained in the matrix remains constant. Experimentally determined properties of the magnetocaloric material are either used directly by interpolation or indirectly through curve fits to the data. The magnetocaloric material is assumed to be incompressible. The thermal conductivity of the magnetocaloric material is assumed to be unaffected by magnetic field and only a function of temperature Specific heat capacity and the partial derivative of entropy with respect to magnetic field are calculated by numerically differentiation of the magnetocaloric property data. The bed geometry is assumed to be spatially uniform so that bed parameters such as hydraulic diameter and porosity are constant. However, the bed material is allowed to vary spatially in order to represent a layered bed in which the materials are selected so that the Curie temperature matches, approximately, the local operating temperature. There are several other assumptions and details which are thoroughly described by Engelbrecht (2008). The model can be downloaded from http://sel.me.wisc.edu/publications/publ.html.

2.2 Comparison with Experimental Data

Astronautics has provided experimental data from the rotary AMRR prototype described in Section 1.3. The rotary bed used to generate the experimental data was packed spherical particles of commercial grade gadolinium (Gd). Figure 4 illustrates the predicted cooling power as a function of measured cooling power for these data. Note that each data point represents a unique operating condition corresponding to an experimentally measured mass flow rate, bed temperatures, and cycle frequency.

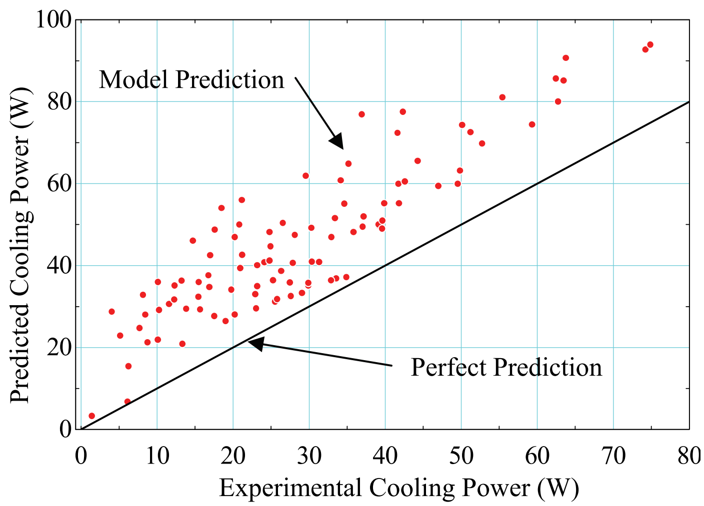

Figure 4: Predicted cooling power versus experimental cooling power

Figure 4 shows that the model consistently over predicts the cooling power by, on average, 20 W. One of the possible reasons for this discrepancy is that the Nusselt number correlation used by the model may not be accurate for the flow of the high Prandtl number working fluid used to generate the data. Several correlations can be found in the literature for heat transfer inside a packed sphere regenerator. The data used to generate these correlations have been developed in a variety of ways; however, the typical data was collected using a gaseous test fluid with a moderate Prandtl number. As a result, there is significant uncertainty in these correlations when applied to the high Prandtl number fluids used in an AMRR system. This uncertainty shows up as discrepancies between the correlations at these conditions. The AMRR model currently uses a correlation developed by Wakao and Kaguei (1982), which is given in Equation (1).

$$Nu_f = 2 + 1.1\,\mathrm{Re}_f^{0.6}\,\mathrm{Pr}_f^{1/3} \tag{1}$$

where Re_f is the Reynolds number based on particle diameter and fluid velocity through the matrix, and Pr_f is Prandtl number of the fluid. The Wakao and Kaguei correlation was developed from data measured using single blow tests with a gaseous heat transfer fluid in a variety of regenerator geometries. Another correlation derived from data is described by Kunii and Levenspiel (1969) and given in Equation (2).

$$Nu_{Kunii} = 2 + 1.8\,\mathrm{Re}_f^{1/2}\,\mathrm{Pr}_f^{1/3} \tag{2}$$

Macias – Mechin *et al.* (1991) developed a correlation that is indicated in Equation (3). The data used to develop the Macias-Mechin correlation was obtained using a hot wire that was immersed in a packed sphere regenerator operating with a liquid heat transfer fluid

$$Nu_{Macias} = 1.27 + 2.66 \, \mathrm{Re}_f^{0.56} \, \mathrm{Pr}_f^{-0.41} \left(\frac{1-\varepsilon}{\varepsilon} \right)^{0.29} \tag{3}$$

where ε is the porosity of the regenerator bed. The correlations displayed in Equations (1) to (3) all have a similar form, but they predict very different Nusselt numbers. Figure 5 shows the Nusselt numbers predicted by these three correlations, and the correlation developed at the University of Wisconsin, as a function of Reynolds number for a constant Prandtl number of 30 (which corresponds approximately to a mixture of 50% water and 50% propylene glycol) and bed porosity of 0.36. Notice that the three correlations from the literature differ by approximately a factor of 5 under these conditions. The development of the new correlation is discussed in section 3.2.

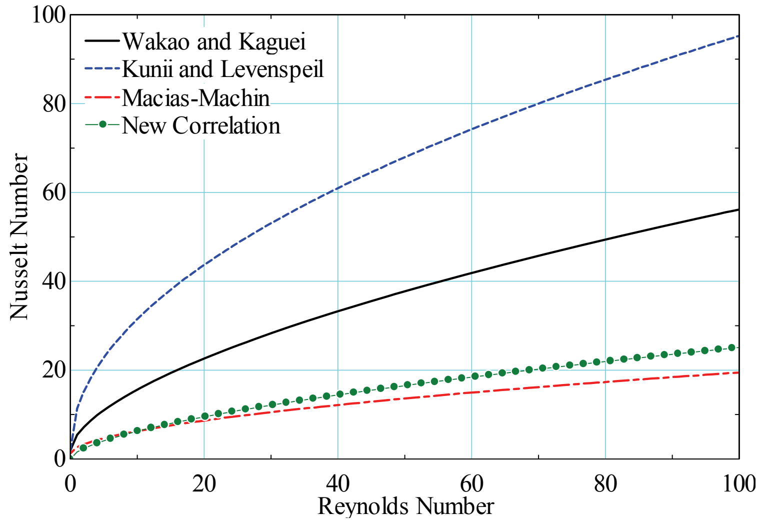

Figure 5: Nusselt number correlations from literature and the newly developed correlation from section 3.2, with a Prandtl number of 30 and porosity of 0.36

3. PASSIVE REGENERATOR TEST FACILITY

3.1 Experimental Setup

In order to determine the heat transfer characteristics of a packed sphere regenerator using a liquid heat transfer fluid, a single-blow passive regenerator has been constructed. A passive regenerator is one where no magnetization and demagnetization occurs. The single-blow experiment is carried out by bringing the regenerator to a uniform temperature during a cold-soak process and then subjecting it to a step change in inlet fluid temperature during a blow process. By measuring the fluid temperature at the outlet of the regenerator and other intermediate locations as a function of time, the heat transfer coefficient that characterizes the bed can be determined. A flow schematic and photograph of the experimental setup is shown in Figure 6.

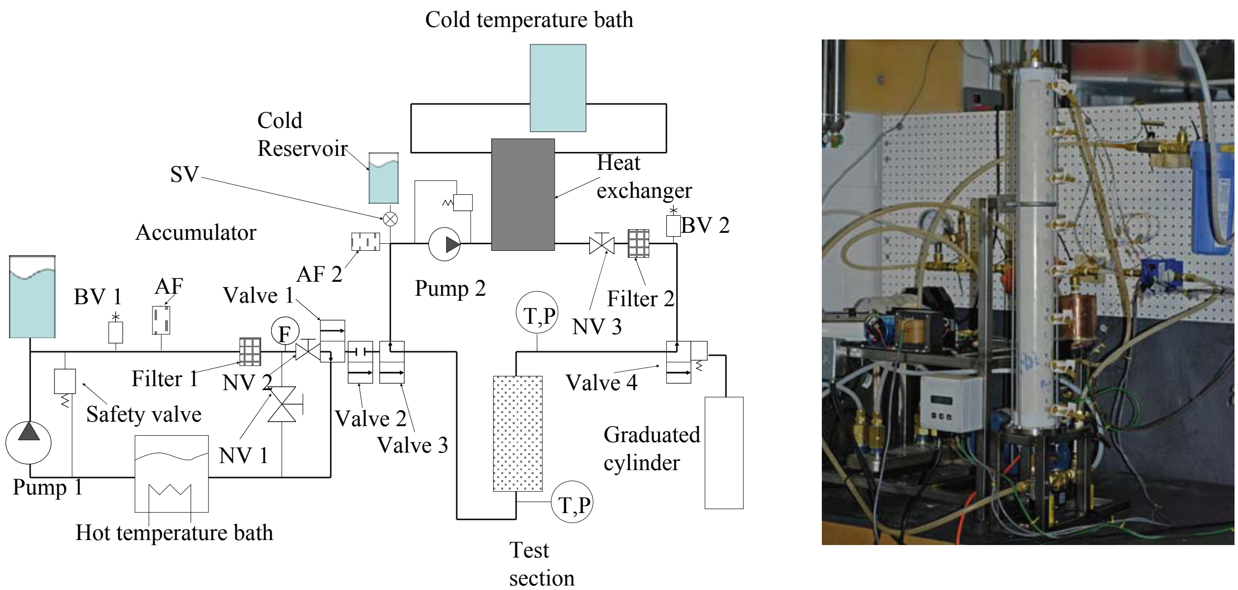

Figure 6: Flow schematic (left) and photograph (right) of the regenerator test facility

The regenerator is made from a 70 cm long section of schedule 40 polyvinylchloride (PVC) pipe. The test facility is broken down into seven separate regenerator pucks, each 10 cm in length. Each regenerator puck is made from schedule 5, type 304 stainless steel, and packed with type 304 stainless steel spheres that are nominally 4.0 mm in diameter. The volumetric flow rate, total pressure drop, and temperature of the fluid at the inlet and outlet of each puck are measured as functions of time. A detailed design of the regenerator is described by Marconnet (2007). Before an experiment is run, cold fluid is pumped through the regenerator until a uniform cold temperature has been reached. By activating a sequence of solenoid valves, fluid at an elevated temperature is diverted through the regenerator to accomplish the blow process. Once the entire regenerator has achieved at least 90% breakthrough (i.e., the outlet temperature has changed by 90%), the test is terminated and the regenerator is cold soaked to prepare for the next test. The breakthrough time of the experiment is defined as the time required for the temperature at the outlet of the regenerator to increase from 20% to 80% of the total temperature difference between the cold soak and hot blow. Complete operating instructions are described by Engelbrecht (2008).

3.2 Data Reduction

Heggs and Burns (1988) report that the best method for determining the heat transfer coefficient from the data collected during a single blow experiment is to match the predicted outlet fluid temperature from a passive regenerator model to the experimental data. The AMRR model developed by Engelbrecht (2008) is used for this purpose; the model is modified so that temperature is predicted continuously rather than for a single cycle and the material properties are adjusted to match the packing that is present in the experiment. The Nusselt number used by the model is taken to be a constant. The value of this constant Nusselt is varied in order to minimize the root mean square error between the experimentally measured outlet temperature and the predicted temperature at each data point, as shown by Equation (4). A more detailed description of the shape fitting process is described by Engelbrecht (2008).

$$RMSE = \frac{1}{N_{exp}} \sqrt{\sum_{i=1}^{N_{exp}} \left(T_{exp,i} - T_{pre,i} \right)^2} \tag{4}$$

Figure 7 shows the experimentally measured fluid temperature at the inlet and outlet of the regenerator, as well as the outlet fluid temperature predicted by the model using the best fit Nusselt number.

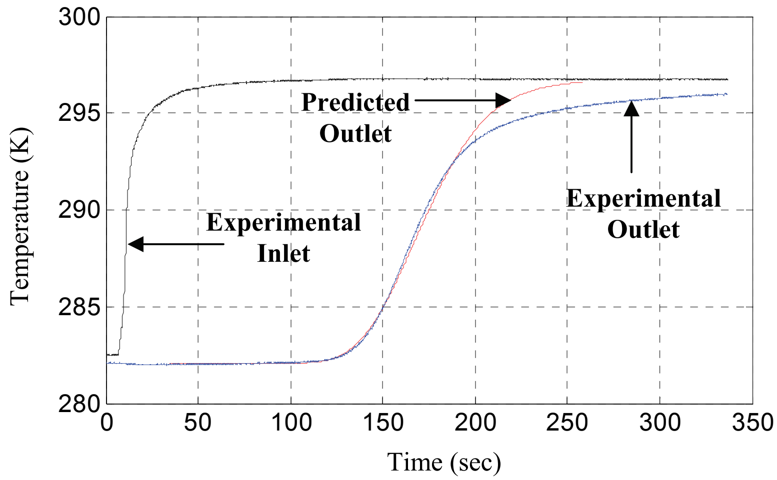

Figure 7: Experimental inlet, outlet, and model predicted temperatures as a function of time

Examination of Figure 7 shows that it is possible to match the measured outlet temperature until approximately 200 s. After that point, the experimental data rises at a slower rate than the predicted outlet temperature. This discrepancy suggests that there is a heat sink somewhere in the experiment that is not being captured by the model. The most likely source of this heat leak is the relatively slow process associated with energy storage in the portions of the PVC housing that participate in the process. The diffusion of energy into the PVC occurs on a longer time scale than diffusion in the stainless steel within the regenerator. There is a small space between adjacent regenerator pucks that is required to mount the thermocouples. The fluid encounters the PVC in this region and therefore some of the PVC housing participates in the heat transfer process. To estimate the volume of the housing that participates in the experiment, the thermal penetration depth into the material was calculated using Equation (5)

$$\delta_{thermal} = 2\sqrt{\alpha_{PVC}\, t_{exp}} \tag{5}$$

where $\delta_{thermal}$ is the penetration depth into the housing, α_{PVC} is the thermal diffusivity of PVC, and t_{exp} is the time associated with the experiment. If an experiment is run for 200 s then the thermal penetration depth will be approximately 9 mm. Assuming that the entire length of the regenerator is affected, the heat capacity of the participating PVC can be up to 40% of the heat capacity of the stainless steel contained the regenerator. This is an overly conservative (high) estimate of this effect, but it does show that the potential impact of energy storage in the housing is important. In order to minimize this effect, the temperatures at the inlet and outlet of each of each of the seven pucks were separately analyzed, instead of using the inlet and outlet temperature for the entire regenerator. The shape matching technique and Equation (4) was then applied to each puck and a best fit Nusselt number was obtained. The process was repeated for each puck for each test that was conducted and the measured Nusselt number was taken to be the average of the values for the 7 pucks. Tests were conducted for fluid flow rates ranging from 0.25 L/min to 4.0 L/min for water, and 20%, 30%, and 40% propylene glycol and water solutions. The best fit Nusselt numbers were then used to develop a new correlation, defined in Equation (6).

$$Nu = 0.70\,\mathrm{Re}^{0.60}\,\mathrm{Pr}^{0.23} \tag{6}$$

The Nusselt number predicted by Equation (6) is approximately half of the value predicted by the Wakao and Kageui correlation, Equation (1), and similar to the predictions of the Macias-Machin correlation, Equation (3), which was derived for liquid heat transfer fluids, reference Figure 5.Equation (6) was then implemented into the model in place of the Wakao and Kageui correlation. In addition, a slight correction was used to adjust for the demagnetization of the regenerator spheres. Demagnetization loss is an observed reduction in the effective magnetic field inside a magnetically permeable body due to its shape (Rowe and Tura, 2007). For the experiment data obtained from Astronautics, the demagnetization loss reduces the effective magnetic field from 1.5 Tesla to 1.2 Tesla. A detailed discussion of the demagnetization effect is provided by Engelbrecht (2008). The model was once again compared to the experimental data obtained from Astronautics, and the results are displayed in Figure 8.

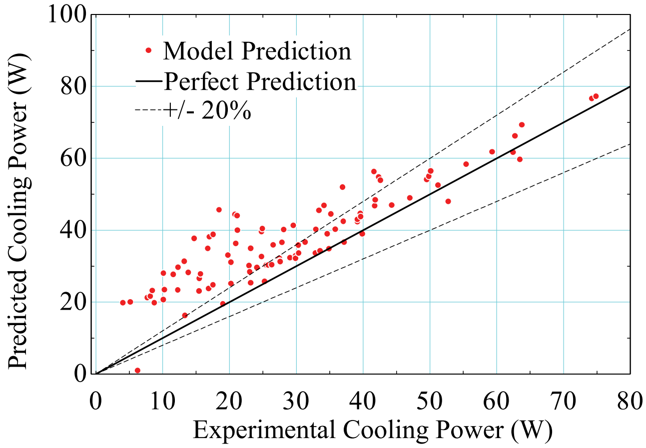

Figure 8: Experimental cooling power and model predicted cooling power with all corrections

5. CONCLUSIONS

A Nusselt number correlation for high Prandtl number fluid flow through a packed bed of spheres was developed using a single blow test facility. The Nusselt number correlation significantly improves the accuracy of the AMRR model predictions as compared to experimentally obtained data. Figure 8 shows that a large number of the lower cooling power predictions are still outside of an acceptable range. It is likely that one reason for this discrepancy is that the effect of the heat capacity of the regenerator wall in the test facility is affecting the Nusselt number. This is likely to be a larger problem for low Reynolds number and low cooling power test conditions. In order to address this shortcoming, a new regenerator bed has been designed in which the heat capacity of the regenerator wall has been reduced from the current value of 40% of the regenerator packing to more reasonable value of 12% of the regenerator packing. Data from this experiment will be used to verify and/or modify the Nusselt number correlation. Future work will be directed at taking data at low Reynolds number, ranging from 3 to 10, and for a bed of non-uniform particles.

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