FABRIC COMPOSITE RADIATORS FOR SPACE NUCLEAR POWER APPLICATIONS

FINAL REPORT March 1993

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1. Introduction.

Nuclear power systems will be required to provide much greater power levels for both civilian and defense space activities in the future than are currently needed. Limitations on the amount of usable power from radioisotope thermal generators (presently less than 1 kWe) and the limited availability of radioisotope heat source materials (specifically ^{238}Pu) lead directly to the conclusion that nuclear power reactors will be needed to enhance the exploration of the solar system as well as to provide for an adequate defense. Lunar bases and travel to the Martian surface will be greatly enhanced by the use of high levels of nuclear power. Space based radar systems requiring many kilowatts of electrical power can provide intercontinental airline traffic control and defense early warning systems. Since the figure of merit used in defining any space power system is the specific power (in electrical power output per unit power system mass), the decrease in the mass of any reactor system component will yield a tremendous benefit to the overall system performance. Also, since the heat rejection system of any power system can make up a large portion of the total system mass, any reduction in the mass of the heat rejection radiators will significantly affect the performance of the power system.

Composite materials which combine the high strength, flexibility, and low mass characteristics of SiO_2 based fibers (other materials may include silicon carbide, aluminoborosilicate glass, or graphite) with the attractive compatibility and heat transfer features of metallic foils (chosen as necessary for compatibility with the desired working fluid), have been proposed for use in a number of space radiator applications. Thus, the weave of the fabric and the high strength capability of the individual fibers are combined with the high conductivity and chemical stability of a metallic liner to provide a light weight, flexible alternative to heavy, rigid, metallic radiator structural containers.

The primary focus of this investigation revolves around two applications of the fabric composite materials, notably a fabric heat pipe radiator design and the Bubble Membrane Radiator concept. Of the two, the fabric heat pipe concept is more traditional. The basic differences between ordinary metal wall and wick heat pipes is the substitution of the fabric composite material for the structural wall and possibly even the wick of a typical heat pipe. This provides a significant savings in the mass for an individual heat pipe. The Rotating Bubble Membrane Radiator (RBMR) concept was developed specifically to take advantage of the developments in fabric materials technology which are combined with the change of phase in a working fluid to provide a significant reduction in radiator mass [Webb 1988, Webb and Coombs 1988, Webb et al 1989]. Further studies of these developments, both analytical as well as experimental, are being conducted in order to quantify the potential application of these attractive radiator concepts.

Heat pipes can be envisioned as being constructed from individual layers of wicking material, metallic liner, and high strength fabric. Layers of fabric (including graphite, E-glass, and S-glass) can be used to replace the traditional wire mesh, if the fabric is completely compatible with the working fluid. In this configuration natural arteries are formed where the tubes are separated from one another. An alternative configuration consists of individually woven cylindrical heat pipe tubes held together by additional layers of high strength fabric.

The Rotating Bubble Membrane Radiator concept is a hybrid design which incorporates the high heat fluxes and temperatures possible in heat pipes with the low mass of the liquid droplet radiator. It incorporates features designed to overcome the deficiencies of both of these radiator types, specifically by inducing an artificial gravity by rotating the radiator to eliminate the capillary pumping of the wick in heat pipes and a physical outer shell is included to eliminate the loss of any of the working fluid which can happen in the liquid droplet radiator. The RBMR maintains the advantages of using condensation phase change of the heat pipe which allows for greater heat fluxes to be transported and radiated efficiently to space.

In the RBMR, a sphere (or ellipsoid) rotates about the minor axis of the sphere, vapor is introduced at the center of the sphere, and is condensed on the inside surface of the liner. The condensate is then pumped to the equator of the device by the centrifugal force imposed by the artificial gravity where it is collected and returned to the power system for use. The driving force for the vapor transport inside the bubble is due to the density gradient which is established as the vapor condenses on the liner surface. The heat of condensation is conducted through the thin metallic liner which is only required to contain the working fluid, and structural integrity is maintained by the high strength fabric. Heat is then rejected to space by a combination of conduction into the fabric fibers and radiation by and through the fibers.

The combined heat transfer mechanism through the radiator wall is common to both the fabric composite heat pipes and the RBMR and allows a large number of variables to be examined in the optimal design of the fabric composite. The natural roughness of a woven fabric increases the effective radiation heat transfer surface area and with appropriate design of the fabric materials (characteristic absorptivity, reflectivity, and emittance) and weave pattern it should be possible to tailor the characteristics of the composite to partially radiate directly through the fabric or even to design the composite material to radiate over a restricted range of frequencies. Emissivity tests of the composite material have shown that significant improvements can be made which will greatly enhance the radiation of waste heat to space [Antoniak and Webb 1989, Antoniak et al 1988].

This report details the efforts conducted by the Oregon State University Department of Nuclear Engineering during the final year for U.S. Department of Energy Nuclear Engineering Research Grant DE-FG07-89ER12901. References will be made to the work completed during the first two budget periods of the grant, however this report only covers

the final reporting period [Klein, 1990, Klein et al 1991]. Details are given here concerning the following areas of research:

- * the fabric composite heat pipe testing facility and the construction of fabric composite heat pipes,
- * fabric composite heat transfer experiments,
- * RBMR heat transfer, condensation, and fluid flow experiments,
- * wicking rate tests as a follow on to previous dry uptake wicking tests,
- * long term materials compatibility literature survey, and

The results of a number of these studies were presented at a variety of conferences including the 8th, 9th, and 10th Symposia on Space Nuclear Power and Propulsion held in Albuquerque, New Mexico in January 1991 - 1993. All of these papers were primarily prepared and presented by the student graduate research assistants involved in the work. The paper titles and authors include:

A.C. Klein, Z. Gulshan-Ara, W.C. Kiestler, R.D. Snuggerud and T.S. Marks, "Fabric Composite Heat Pipe Technology Development," Proceedings, Tenth Symposium on Space Nuclear Power and Propulsion, p. 387-393, Albuquerque, NM, January 1993.

A.C. Klein and B.J. Webb, "Advanced Thermal Management Needs for Lunar and Mars Missions," Proceedings, Tenth Symposium on Space Nuclear Power and Propulsion, p. 847-851, Albuquerque, NM, January 1993.

W.C. Kiestler, T.S. Marks and A.C. Klein, "Design and Testing of Fabric Composite Heat Pipes for Space Nuclear Power Systems," Proceedings, Nuclear Technologies for Space Exploration, p. 812-818, Jackson Hole, WY, August 1992.

Z. Gulshan-Ara and A.C. Klein, "Fabric Composite Heat Transfer Study," Proceedings, Nuclear Technologies for Space Exploration, p. 757-764, Jackson Hole, WY, August 1992.

H. Al-Baroudi and A.C. Klein, "Experimental Simulation of a Space Radiator using Condensation on a Rotating Flat Plate," National Heat Transfer Conference, San Diego, CA, August 1992.

T.S. Marks and A.C. Klein, "An Experimental Comparison of Wicking Abilities of Fabric Materials for Heat Pipe Applications," Proceedings, Ninth Symposium on Space Nuclear Power Systems, p. 1147-1152, Albuquerque, NM, January 1992.

H. Al-Baroudi and A.C. Klein, "Rotating Flat Plate Condensation and Heat Transfer," Proceedings, Ninth Symposium on Space Nuclear Power Systems, p. 796-799, Albuquerque, NM, January 1992.

H. Al-Baroudi, A.C. Klein, and K.A. Pauley, "Experimental Simulation of the Bubble Membrane Radiator Using a Rotating Flat Plate," Proceedings, Eighth Symposium on Space Nuclear Power Systems, p. 723-727, Albuquerque, NM, January 1991.

T.S. Marks and A.C. Klein, "Material Compatibility Issues for Fabric Composite Radiators," Proceedings, Eighth Symposium on Space Nuclear Power Systems, p. 320-325, Albuquerque, NM, January 1991.

T.S. Marks and A.C. Klein, "An Experimental Approach to Compare Wicking Stabilities of Fabric Materials for Heat Pipe Applications, Proceedings, Eighth Symposium on Space Nuclear Power Systems, p. 869-874, Albuquerque, NM, January 1991.

A.C. Klein, "Microgravity Thaw Experiment (MITEX)," Proceedings of the AIAA/ASME 6th Thermophysics and Heat Transfer Conference, Seattle, WA, June 1990.

Also, during the three years of this study the following Nuclear Engineering theses have been completed:

"Experimental Simulations of a Rotating Bubble Membrane Radiator for Space Nuclear Power Systems", Homam Al-Baroudi, Ph.D. Thesis, March 1993.

"Fabric Composite Radiation Heat Transfer Study", Zubaida Gulshan-Ara, M.S. Thesis, March 1993.

"Design and Testing of Fabric Composite Heat Pipes for Space Nuclear Power Applications", William C. Kiestler, M.S. Thesis, December 1992.

"An Investigation of Fabric Composite Heat Pipe Feasibility Issues", Timothy S. Marks, M.S. Thesis, May 1992.

"Ultralite Copper Reflux Tube Life Test and Ceramic Fabric Wicking Rate Experiments", Ross D. Snuggerud, M.S. Thesis, January 1993.

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Z.I. Antoniuk and B.J. Webb 1989, "Construction and Testing of Fabric Heat Pipe Components for Space Nuclear Power," Sixth Symposium on Space Nuclear Power Systems Transactions, CONF-890103-Summs., Albuquerque, NM, January 8-12, 1989.

Z.I. Antoniak, W.J. Krotiuk, B.J. Webb, J.T. Prater, and J.M. Bates 1988, "Fabric Space Radiators," PNL-6458, Battelle Pacific Northwest Laboratory, Richland, WA, 1988.

A.C. Klein 1990, "Fabric Composite Radiators for Space Nuclear Power Applications--Technical Progress Report," OSU-NE-9007, Oregon State University, Corvallis, OR, June 1990.

A.C. Klein, H. Al-Baroudi, T.S. Marks, Z. Gulshan-Ara, and W.C. Kiestler 1991, "Fabric Composite Radiators for Space Nuclear Power Applications--Technical Progress Report," OSU-NE-9112, Oregon State University, Corvallis, OR, May 1991.

P.M. Sawko and H.K. Tran 1985, "Strength and Flexibility Properties of Advanced Ceramic Fabrics," *SAMPE Quarterly*, V17(1), p. 7, 1985.

B.J. Webb 1988, "Preliminary Design and Analysis of a Rotating Bubble Membrane for Space-based Nuclear Power Systems," M.S. Thesis, Oregon State University, Corvallis, OR, April, 1988. (Also OSU-NE-8805, June 1988.)

B.J. Webb and E.P. Coombs 1988, "Rotating Bubble Membrane Radiator," United States Patent, no. 4,789,517, December 6, 1988.

B.J. Webb, Z.I. Antoniak, A.C. Klein, K.A. Pauley 1989, "Preliminary Design and Analysis of a Rotating Bubble Membrane Radiator for Space Applications," Draft report to NASA Johnson Space Center prepared by Battelle Pacific Northwest Laboratory under USDOE Contract DE-AC06-76RL01830, Richland, WA, March, 1989.

2. Fabric Composite (FC) Heat Pipe Development and Testing.

2.1 Introduction.

Fabric composite heat pipes hold great promise to reduce the mass of heat rejection radiators for space nuclear power systems, and other space applications. Their combination of a light weight ceramic woven fabric for structural strength with a metallic liner for working fluid retention can yield significant reductions in the mass of radiator systems [Antoniak et al. 1991]. Additional advantages of using a ceramic fabric heat pipe include great flexibility and the potential for compact storage prior to use and rapid deployment.

The basic fabric composite heat pipe design is shown schematically in Figure 2.1. The heat pipes constructed and tested to date have used water as the working fluid and commercially available stainless steel piping and woven ceramic fabric tubing. In these tests, a relatively thick stainless steel end piece is used for the evaporator section, a thin walled, seamless stainless steel tube is welded to the evaporator section, and then a stainless steel end fitting is used to close off the condenser section. A ceramic fabric tube then covers the outside of the thin walled stainless steel condenser section. An additional, single layer of woven ceramic fabric tube is inserted through the end piece to act as the wick for the heat pipe.

2.2. Fabric Composite Heat Pipe Design.

Initial tests were conducted using a simple stainless steel heat pipe design. This heat pipe consists of an 81.5 cm length of 2.7 cm (1 1/16 inch) OD, 0.0032 m thick stainless steel pipe, with a threaded end piece to allow access for fitting of the wick. The threaded end piece is equipped with a 0.3175 cm (1/8 inch) tube which is coupled through the teflon end piece on the test chamber to allow evacuation of the heat pipe and loading of the working fluid. The heat pipe wick consists of a 500 μ m (20 mils) braided Nextel (trademark of the 3M Company, Saint Paul, MN) fabric tubing obtained from Battelle Pacific Northwest Laboratories. Wicking rate tests conducted at Oregon State University found such woven fabrics to have satisfactory wicking ability (Marks, 1992). The wick was loaded in the stainless steel heat pipe by first sliding it over a 0.64 cm (1/4 in) diameter dowel, and then inserting it into the full length of the heat pipe. This method allowed loading of the wick without any twisting or cramming, and the wick fit so snugly that no wire spring or screen was needed to hold it against the inside of the pipe. The threaded end piece on the condenser section of the heat pipe provided a pressure tight seal and easy access to the wick.

A first generation Fabric Composite (FC) Heat Pipe has been designed and tested. The design of the heat pipe consists of a fabric composite tube composed of two layers: a structural layer and a fluid containment layer, a wick structure, and threaded end cap. Both the structural layer and wick are comprised of the Nextel fabric tubing. Initial tests have been completed on a design that simply places Nextel tubing over the condenser section of the standard stainless steel heat pipe described above. It was possible to achieve very low

Figure 2.1. Fabric Composite Heat Pipe.

contact resistances between the fabric and the stainless steel due to the tight fit of the Nextel tubing. Expansion of the stainless steel as it is heated further enhances the contact between the two surfaces. This contact resistance has been shown to have a significant effect on the heat transfer and effective emissivity of the composite radiator (Antoniak et al. 1991).

A second generation heat pipe is currently being tested using a 2.54 cm (1 in) OD x 254 μm (10 mils) stainless steel tube surrounded by the Nextel fabric tubing. The thin-wall stainless steel is welded at one end to a 0.0032 m thick stainless steel end piece (approximately 18 cm long) to allow band heaters to be attached. The thick end piece serves as the evaporator section of the FC heat pipe. At the condenser end, the rigid condenser cap houses the heat pipe evacuation and charging line. The end pieces are also designed to facilitate suspension of the FC heat pipe in the test chamber. This heat pipe, while superior to the heat pipe described above, is still far from an optimized design. Additional heat pipes are being developed using thinner, better conducting metal foils for fluid containment. The end pieces can also be constructed of thinner metals, but must be rigid enough to allow for attachment of the fabric and foil, and facilitate heating at the evaporator end of the heat pipe. Several designs for the condenser end cap are possible, depending on the thickness of the containment tubing. For the 250 μm tubing described here, a standard Swagelock (trademark of Crawford Fitting Co, Solon, OH) fitting has been employed. However, thinner materials such as foils, will require a more complex design. These design concepts can be effectively employed to significantly reduce the weight of the fabric composite heat pipes.

2.3. Heat Pipe Test Facility.

A facility designed to allow controlled operation and testing of heat pipes has been constructed and used to test the performance of fabric composite heat pipes. The Heat Pipe Test Facility (HPTF) was designed to simulate the environmental conditions of space for radiation heat transfer, and to provide for monitoring of significant heat pipe operating parameters.

The HPTF consists of two mild steel concentric cylinders (15 and 25 cm ID) welded together to form an outer cooling jacket and an inner vacuum chamber. The inner vacuum chamber is 101 cm (40 in.) deep. The cooling jacket is insulated with 5 cm (2 in.) of highly compressed, rigid fiberglass insulation. A recirculating bath chiller is used to circulate a 50% aqueous ethylene-glycol coolant, at temperatures as low as -20°C , through the cooling jacket. The inner cylinder is smooth and painted black to allow for maximum radiation heat transfer. The test chamber is sealed by a teflon end piece with an o-ring seal. The teflon end piece also contains leads for heater power input and thermocouples, the heat pipe evacuation and charging line, and a tube for drawing a vacuum on the test chamber. The evacuation and charging line is fitted with a DC pressure transducer for monitoring the heat pipe operating pressure, and has a four-way valve that can be positioned for evacuation of the heat pipe, charging the working fluid, or shutoff. A schematic diagram of this facility is shown in Figure 2.2.

Figure 2.2. Heat Pipe Test Facility.

A mounting rack was designed to hold the heat pipe in position while testing, and to provide shielding for all forms of heat transfer other than radiation from the heat pipe condenser section. The mounting rack can be seen in Figure 2.2, and is shown in greater detail in Figure 2.3. The mounting rack consists of three teflon disks connected by stainless steel rods. The condenser end disk provides support for the heat pipe and limits the amount of radiation heat transfer in the axial direction. The other two disks help to form the thermal radiation shield around the evaporator end of the heat pipe. Ideally, all of the power input into the heat pipe would be rejected by radiation from the condenser section, however, a small amount of energy is lost by heat transfer from the evaporator section and from the heaters themselves. The purpose of the evaporator shield is to minimize the amount of heat lost through conduction and radiation to the cooling jacket from the evaporator and heaters. The evaporator shield consists of the two teflon disks and stainless steel plates for axial shielding, and four concentric stainless steel cylinders for radial shielding. The shield was shown to be fairly efficient, allowing less than 10% heat loss from the evaporator end.

The HPTF is cooled by a PolyScience Model 900 constant temperature circulating bath. This chiller circulates coolant between the HPTF cooling jacket and a 5 liter reservoir. A dual speed centrifugal pump provides flow of up to 15 l/min, depending on the coolant viscosity and system head. Under the conditions for the tests conducted, slow pump speed corresponded to a flow rate of approximately 3.8 l/min (1 gpm), and fast speed provided approximately 7.6 l/min (2 gpm). The chiller is rated at 240 W at -20°C , but the range of the constant temperature controller is -15°C to 100°C .

Pressure in the vacuum chamber is monitored using a standard bellows type gage. The gage measures pressure over a range of -100 to 200 kPa (30 in Hg vac. to 30 psig). The vacuum chamber pressure is not monitored or recorded by the data acquisition system. It must be checked visually periodically during testing to verify the vacuum in the test chamber. If the pressure rises in the test chamber, the vacuum can be easily redrawn with a vacuum pump without adverse effect on the heat pipe or heat pipe tests. This was generally not required however, as the facility held a vacuum quite well.

The volumetric flow rate of the coolant is measured by an Omega acrylic rotameter. The flow meter uses a guided stainless steel float to measure the flow rate from 0 to 5 gpm. The rotameter is accurate to within 4% of full scale, but is calibrated for fluids having a specific gravity equal to 1.0. Thus, the uncertainty in the properties of the aqueous ethylene-glycol mix contributes to an appreciable overall uncertainty in the measurement of the coolant flow rate.

An Omega PX300-500G-V 10 volt DC pressure transducer is used to continuously monitor the pressure in the heat pipe. The pressure transducer operates over a range of 0 to 500 psi and provides a linear signal from 0 to 30 mV. The output is coupled to the data acquisition system through an Omega signal amplifier, which has a maximum output voltage of ± 9 VDC. The gain of the amplifier (10x, 100x, or 1000x) is set to provide a signal within the ± 5 VDC analog input range of the data acquisition system.

Figure 2.3. Heat Pipe Mounting Rack.

Heat is provided to the heat pipe evaporator end by two 7.6 cm (3 in) band heaters. The power to the heaters is controlled using a voltage rheostat. The heater voltage is adjustable from 0 to 120 VAC. An ammeter in line with the heater wire provides a measurement of the current to the heaters. The voltage and current are used to determine the power delivered to the heat pipe.

The HPTF was designed to simulate the space environment as nearly as possible, thus it is designed to provide a cold, black surrounding for the heat pipe. It is assumed that heat transfer from the heat pipe to the vacuum chamber wall is accomplished solely by radiation, with the configuration factor for infinite concentric cylinders. These assumptions are used in evaluating the radiation heat transfer in the test chamber. The design of the heat pipe mounting rack ensures that, in a vacuum, the only mode of heat transfer is by radiation. Since there is very little contact between the heat pipe and the teflon support disks, and since teflon has very poor thermal conductivity, heat transfer by conduction is negligible. Additionally, the infinite concentric cylinder approximation allows the configuration factor between the heat pipe and the vacuum chamber wall to be unity. The electrical input power is found by direct measurement of the power input (corrected for heat loss from the evaporator section).

An 8088 processor based, personal computer and an Omega DAS-8 analog/digital interface board are used to provide continuous monitoring and logging of the heat pipe and HPTF parameters. The DAS-8 system contains eight single ended, analog input channels, and uses a 12-bit, successive approximation A/D converter with a nominal conversion time of 25 μ sec.

Two Omega Model EXP-16 Analog Input Multiplexers are used to provide up to 32 channels of monitoring on one DAS-8 channel. One EXP-16 is used to measure temperatures, and the other is used to measure the heat pipe pressure. The multiplexer boards are connected using a 37 pin D type male plug. The EXP-16 gain setting is adjustable from 0.5x to 1000x. A dip switch on the board selects gains of 0.5, 1, 2, 10, 50, 100, 200, or 1000. A higher gain setting reduces noise and fluctuation in the thermocouple signal, but limits the maximum measurable temperature. Conversely, lower gain settings increase the measurable temperature range, but result in greater fluctuation in the signal. A gain of 1000 limited the maximum measurable temperature to less than about 128°C, and a gain of 200 resulted in a fluctuation in the temperature signal of $\pm 6^\circ\text{C}$. Therefore, a 330 ohm resistor was added to the gain circuitry in order to provide a gain of 606. This gain setting allowed temperature measurements up to 225°C, while reducing the temperature signal oscillation to about $\pm 1^\circ\text{C}$. It is necessary to use separate multiplexers for temperature and pressure because the required gain settings are different. A gain of 50 was used for the heat pipe pressure measurement. A total of 32 channels are available, but only 16 were used during the heat pipe tests. Fifteen channels were used to measure the temperatures, and one channel was used to monitor the heat pipe pressure. Figure 2.4. is a schematic diagram of the complete test apparatus.

Figure 2.4. Schematic Diagram of the Heat Pipe Test Apparatus.

All temperatures were measured using Chromel - Alumel (K-type) thermocouples. These thermocouples are capable of measuring temperatures from -270°C to 1370°C . As discussed above, the upper and lower limits on the measurable temperature response, however, is a function of the multiplexer gain, since the input analog voltage is limited to ± 5 VDC. The EXP-16 provides a cold junction compensation voltage, which is used by the data acquisition software in calculating measured temperatures. A total of 15 channels of the #1 EXP-16 were used for temperature measurements. Table 2.1 summarizes the location of the thermocouples assigned to each of the EXP-16 channels. Thermocouples 3 through 9 were placed approximately 10 cm apart along the condenser section to verify the nearly isothermal conditions expected along the condenser, and to provide the temperature measurements used to calculate the emissivity of the heat pipe. For the fabric composite tests, thermocouples 3 and 9 were placed underneath the fabric.

Table 2.1. Thermocouple Locations.

EXP-16 Channel #	Location of Thermocouple
0,1	Heat pipe evaporator
2	External evaporator radiation shield
3-9	Heat pipe condenser surface
10-12	Vacuum chamber walls
13	Coolant inlet
14	Coolant outlet

The HPTF is designed to accommodate a variety of heat pipe designs, and procedures will vary somewhat for the testing of different heat pipes. Special considerations and precautions must be taken when handling and testing a lightweight fabric composite heat pipe, because the heat pipe is not rigid until pressurized. These heat pipes must be handled with great care to prevent bending or crushing the liner. Once the heat pipe is loaded into the HPTF, a vacuum must be drawn on the test chamber prior to evacuating the heat pipe to prevent collapsing the fabric composite condenser. The three heat pipes tested to date were relatively rigid, and the liners were sufficiently thick to preclude any of these problems. Nonetheless, the procedure for testing the fabric composite heat pipes assumed a lightweight design, so that one procedure could be used to test any fabric composite heat pipe. For example, the HPTF vacuum chamber was always evacuated before the heat pipe.

2.4 Heat Pipe Test Results.

The purpose of the heat pipe tests is to evaluate the radiant properties of the fabric composite and compare them with the properties of a conventional stainless steel heat pipe.

These tests were first conducted on the conventional heat pipe, and the emissivity of the stainless steel surface was empirically determined. Similar tests were then performed on the fabric composite heat pipes and the emissivity of the fabric condenser surface was calculated. This section contains an analysis of the startup and steady state operation of these heat pipes, including frozen heat pipe startup and wick dryout.

The primary objective of the heat pipe tests was to calculate the effective emissivity of the fabric composite heat pipes. The power dissipated by the heat pipe by radiation to the vacuum chamber walls is determined by measuring the power supplied to the heat pipe evaporator and correcting for the amount of heat lost from the evaporator end. The amount of heat lost at the evaporator end is determined by the radiation heat transfer between the evaporator thermal shield and the vacuum chamber wall.

In addition to the calculation of emissivity, the behavior of each heat pipe is discussed using a plot of the heat pipe pressure, evaporator temperature, and condenser temperature during operation. Values for the saturation temperature were also determined and included in the graphs in order to verify heat pipe operation in saturated conditions, and to estimate the temperature drop across the wall of the heat pipe. The saturation temperature is also of interest because the band heaters did not provide a very uniform heat input to the evaporator, and several difficulties are associated with the measurement of the evaporator temperature. The temperatures measured by the evaporator thermocouples are believed to be in error by as much as 10°C , since the heaters generally had better contact with the thermocouples than with the evaporator surface. Additionally, the very large temperature gradient between the heaters, the thermocouples, and the evaporator gives way to significant errors in the temperature measurement. The thermocouple voltage is generated at the location of the temperature gradient, and not necessarily at the thermocouple junction [Omega Engineering 1984]. The highest temperature gradient exists where the thermocouple wires pass under the band heaters, and can be expected to be very large. In other words, the evaporator thermocouples provide a better measurement of the heater temperature than the heat pipe evaporator temperature. One additional contribution to the error in this temperature measurement might be attributed to the effect of the heater coil emf on the thermocouple. All of these factors most likely contribute to the error in the evaporator temperature measurement, and help explain the excessive ΔT between the evaporator and condenser.

The startup of the conventional heat pipe was commenced from the frozen state. This differs from the majority of tests conducted in that the working fluid was loaded prior to warming up the heat pipe. It was found that if the facility was cold, excessive moisture would accumulate in the vacuum chamber, so that the heat pipe was loaded into the HPTF, and the chamber was evacuated, prior to cooling. The presence of moisture in the vacuum chamber had an adverse effect on the thermocouples and instrument wiring. The normal procedure was to load the working fluid once the entire length of the condenser section had been heated to above 0°C . However, during this test, the heat pipe had a sufficient amount of working fluid remaining in the wick from a previous test. Additionally, enough moisture was in the evacuation line to block it upon freezing, and the heat pipe pressure was not

observable until well into the startup when the evacuation line had thawed. The frozen startup is discussed more fully later in this section.

Figure 2.5 shows the behavior of the conventional stainless steel heat pipe during startup. Once the heat pipe was loaded into the HPTF, a vacuum was drawn on the test chamber and heat pipe. Power was then supplied to the evaporator end by the band heaters, and the voltage and current to the heaters was measured. An initial power of 39 W, which had produced steady state operation in previous heat pipe tests, was used to startup the heat pipe. During startup, the evaporator and condenser temperatures increase steadily, and the temperature along the length of the condenser is relatively isothermal. Once saturation is achieved, the pressure begins to rise rapidly. Both the temperature and the pressure increase linearly until an equilibrium temperature and pressure is reached for the heat pipe operation. At this temperature, the heat pipe radiates energy at a rate equal to the energy addition at the evaporator.

After 70 minutes, the heat pipe reached a steady state condition with the condenser temperature at approximately 160 °C. Then in an effort to increase the steady state temperature, the power was then increased to 47 W (50 V). Figure 2.6 shows the steady state operation. The temperature and pressure rise linearly again due to the increase in the power input. As the evaporator temperature approached 200°C, the power was returned to 39 W (45 V) to prevent drying out the wick, and to maintain steady state operation for calculations. The heat pipe was secured and the test concluded after 183 minutes of operation.

The heat pipe was evaluated at a power input of 39 W with the condenser temperature at 166°C. The heat loss was calculated to be 3.9 W, and the resulting heat pipe emissivity ϵ_{hp} was 0.32.

Startup of the heavyweight fabric composite heat pipe, shown in Figure 2.7, was performed at a slower rate than for the conventional heat pipe in Figure 2.5. The power applied during the initial heat up phase was 48 W (50 V). The power level was increased to 58 W (55 V) 90 minutes into the startup, and to 69 W (60 V) after 100 minutes. The improvement in the capacity of the fabric composite heat pipe was immediately apparent, as it could handle nearly twice the heat input observed during the previous conventional heat pipe test. The heat pipe reached steady state after approximately 165 minutes.

Figure 2.8 shows the steady state operation of the heavyweight fabric composite heat pipe. At 150 minutes, the power was reduced to 62 W (57 V) to limit the evaporator temperature to less than 200°C. This power level produced a slowly decreasing temperature profile, and when the evaporator temperature continued to decrease below 185°C 240 minutes into the test, the power was increased to 69 W (60 V) again. The heat pipe was operated at a power of 69 W until the test was secured after 336 minutes. An input of 69 W was observed to produce a relatively steady operation in the fabric composite heat pipes.

Figure 2.5. Conventional Heat Pipe, Startup.

Figure 2.6. Conventional Heat Pipe, Steady State.

Figure 2.7. Heavyweight Fabric Composite Heat Pipe, Startup.

Figure 2.8. Heavyweight Fabric Composite Heat Pipe, Steady State.

The heavyweight fabric composite heat pipe was evaluated at 68.4 W, with a condenser temperature equal to 157°C. The estimated heat loss at the evaporator end was 6.7 W, and the emissivity ϵ_{hp} was calculated to be 0.62.

Based on the experience from the previous fabric composite heat pipe test, a smooth startup and long steady operation of the lightweight FC heat pipe was achieved. The lightweight heat pipe was started up using a gradually increasing power from 18 to 69 W so as to approach the steady operating temperature and power level at a slower rate, and prevent the overshoot seen in the previous test. Figure 2.9 provides a plot of the heat pipe pressure and the evaporator and condenser temperatures during the smooth approach to power. Once a steady state condition was achieved with the input power at 69 W, the lightweight fabric composite heat pipe was evaluated for nearly four hours. Figure 2.10 shows the long, steady run at 69 W power input.

A refinement in the testing procedure is evident from the results of the steady operation of the lightweight heat pipe. During the almost four hours of steady state operation, the condenser temperature remained within $\pm 3^\circ\text{C}$ of the temperature used to evaluate the heat pipe emissivity. The lightweight fabric composite heat pipe was evaluated at 69 W and 159°C. Evaporator heat loss was estimated to be 6.1 W, and the heat pipe emissivity ϵ_{hp} was 0.69.

A heat pipe startup from a frozen state was conducted in order to observe the operation during this transient. A plot of the heat pipe parameters during this startup is provided in Figure 2.11. Two aspects of the startup are apparent from the graph. During startup of all of the heat pipes, the temperature of the condenser was seen to increase slowly, but with less of a time lag than seen for the frozen case. In the frozen heat pipe experiment, the temperature along the length of the condenser rises as the pipe is heated, but it took an appreciable amount of time to thaw out the end of the condenser. During an unfrozen startup, the axial condenser temperature is nearly isothermal due to the formation and flow of vapor in the pipe. The condenser temperature is plotted for several locations along the length of the pipe to show how the isothermal condition is reached only after the water in the wick has thawed. Once the entire length of the condenser had been thawed, at about 150 minutes, the startup proceeded normally, and the condenser temperature increased rapidly.

Note that the heat pipe pressure measurement was not available until after the condenser was completely thawed. This is due to blockage of the heat pipe evacuation and fill line where the pressure transducer is connected. A sudden dip in the pressure and temperature occurred when the evacuation line became clear, allowing the vapor in the heat pipe to expand and fill the evacuation line.

During an approach to steady state power for one of the fabric composite heat pipe tests, a "dryout" condition occurred. Dryout occurs when the capillary pumping limit of the wick is exceeded and the working fluid is evaporated faster than it can be replenished. Once dryout occurs, the heat pipe will not operate until the wick has been radiate or "quenched."

Figure 2.9. Lightweight Fabric Composite Heat Pipe, Startup.

Figure 2.10. Lightweight Fabric Composite Heat Pipe, Steady State.

Figure 2.11. Frozen Heat Pipe Startup.

Figure 2.12 shows the effects of dryout on the heat pipe operation. When dryout is achieved, the heat pipe pressure drops rapidly, and the absence of continued evaporation, results in a loss of cooling, causing the evaporator temperature to increase dramatically. The condenser temperature decreases, following the saturation temperature.

It was found that once dryout had occurred, the heaters had to be shut off and the heat pipe completely cooled down in order to re-wet the wick and begin another test. While the dryout condition was not intended during the test, it did provide data to experimentally evaluate the fabric composite heat pipe limits. A complete analysis of the dryout phenomenon for the heat pipe is not provided here, but is left for future work.

When making calculations in radiation heat transfer, a large degree of uncertainty is attributable to the value used for the emissivity of the radiating surfaces. The emissivity of a material is dependent upon the properties of the material as well as the conditions under which it is radiating. For example, the emissivity of stainless steel is found to be from 0.16 to 0.39 depending on the type of alloy, surface treatment, and the temperature of the surface. The effect of the surface condition is particularly difficult to specify with confidence. The surfaces of concern in the heat pipe tests are the vacuum chamber wall, and the evaporator shield. The emissivity of these surfaces must be estimated and used in the calculation of the heat pipe emissivity, and the uncertainty in these estimates contributes to the overall uncertainty in the calculated value for ϵ_{hp} .

The interior wall of the vacuum chamber is relatively smooth and painted with a flat black lacquer. The emissivity of the vacuum chamber wall, ϵ_{vc} , can therefore be estimated to be from 0.96 to 0.98. The value of ϵ_{vc} estimated for use in the calculations was 0.94 ± 0.04 . The value is reduced somewhat to account for scratches and bare spots on the vacuum chamber wall, and the uncertainty of ± 0.04 covers the range of reasonable values of ϵ_{vc} .

The emissivity of the stainless steel shields (ϵ_{sh}) is more difficult to estimate. The surface is not polished, but it is relatively smooth and free of oxidation. The expected range of ϵ_{sh} is from 0.2 to 0.4, so a value of 0.3 ± 0.1 was used.

Uncertainties also exist in the measured parameters. These uncertainties arise due to the accuracy and tolerance of the instrumentation. The uncertainties δx_i , or fractional uncertainties $\delta x_i/x_i$, for the values measured in the heat pipe tests are listed in Table 2.2. The uncertainties were derived from the instrument specifications, data acquisition system accuracy, and observed characteristics of the data such as noise, fluctuations, and contact resistance. The uncertainties were assumed to be random and independent. Combined and total uncertainties were calculated using the sum in quadrature method [Taylor 1982]. The resulting uncertainties for the emissivities calculated above are shown in Table 2.3.

Figure 2.12. Wick Dryout During Heat Pipe Startup.

Table 2.2. Uncertainty in Measured Values.

MEASURED PARAMETER	UNCERTAINTY δx_i or $(\delta x_i/x_i)$
V	± 0.5 V
I	± 0.1 A
Q_{in}	± 0.059
A	≈ 0
T_{vc}	± 2 °C
T_{hp}	± 2 °C
ϵ_{vc}	± 0.04
ϵ_{sh}	± 0.1
P	± 10 kPa

Table 2.3. Overall Uncertainty in Calculated Values.

HEAT PIPE	δQ (W)	$\delta \epsilon_{hp}$
Conventional	± 2	± 0.02
Heavyweight FC	± 3	± 0.04
Lightweight FC	± 3	± 0.04

The final uncertainty in the calculated heat pipe emissivity is approximately 6% for each of the tests discussed. This is a reasonable level of uncertainty for experimental results, and supports the credibility of the procedures for testing and evaluating the performance of the heat pipes in the HPTF.

The results of the emissivity calculations are summarized in Table 2.4. The tests demonstrated a significant enhancement in the radiant heat transfer due to the properties of the fabric composite. These results correlate well with those reported by Antoniuk and Webb

in 1991 [Antoniak et al 1991], and more recent studies of fabric emissivities conducted at Oregon State University [Klein et al 1993].

Table 2.4. Summary of Experimental Results

HEAT PIPE	POWER (W)	T _{Cond} (°C)	EMISSION
Conventional	39 ± 2	166 ± 2	0.32 ± 0.02
Heavyweight FC	68 ± 3	158 ± 2	0.62 ± 0.04
Lightweight FC	69 ± 3	160 ± 2	0.68 ± 0.04

The results are significant because of the multiple improvements to radiator efficiency inherent in the fabric composite design. The increased emissivity is an important advantage of the fabric composite, but this could also have been achieved by notching the steel heat pipe, or by simply painting it black. However, the combined advantages of *higher* output capacity and *lower* mass associated with the fabric composite, and the possibility of improved survivability, should make the use of fabric composite heat pipe radiators an attractive choice in the design of future space power systems. An appreciation for the magnitude of the savings possible in system mass can be gained from Table 2.5. The overall advantage of the fabric composite design is best shown by the power to mass ratio for each heat pipe. In the simple design tested in the HPTF, for example, the heat rejection capacity nearly doubled, while the heat pipe mass was reduced by a factor of three.

Table 2.5. Summary and Comparison of Heat Pipe Designs

HEAT PIPE DESIGN	LINER THICKNESS	COMPOSITE	MASS (g)	POWER/MASS (W/kg)
Conventional	3.2 mm	—	1530	25
Heavyweight FC	3.2 mm	Stainless/Nextel	1530	45
Lightweight FC	0.25 mm	Stainless/Nextel	510	135
FC Reflux Tube	50 μm	Copper/Nextel	140	500

The tests conducted on the fabric composite heat pipes using the HPTF not only verify the feasibility of such heat pipes as potential space radiators, but show that such a design has significant advantages over many conventional design concepts.

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3. Fabric Composite Heat Transfer Studies.

3.1. Introduction.

The purpose of the Fabric Composite Radiation Heat Transfer experiment is to study the heat transfer from a hot fabric composite (FC) surface through a vacuum into a heat receiver to simulate the ultimate rejection of heat from a space based power system. The design objectives of the experimental facility are

- to determine the effective emissivities of various fabric composites materials
- to determine the effective emissivities of bare liner materials for comparison
- to determine the best combination of the fabric and liner material.

3.2. Radiation Heat Transfer Facility Fabrication Procedure.

The experimental apparatus has been constructed and a series of effective emissivity tests have been performed. In the final configuration, a vacuum vessel contains a uniform heating block at the top end and a uniform cooling block at the bottom end. The entire experimental arrangement is well insulated thermally from the environment using four stainless steel cylinders with small vacuum gaps in between to minimize undesired radiation heat loss from the inside. These small vacuum gaps act as thermal shields. Thus, it is expected that all of the heat passing through the fabric composite is radiated through the vacuum and ultimately received by the cooling system. The test facility is designed so that the heating block can be easily removed to change the fabric composite sample.

The fabric sample to be tested is placed on a flat electric heater and a metallic liner is sandwiched in between the heater and fabric as shown in Figure 3.1. This combination of the heater, liner, and fabric is then inverted and hung from the lid of the vacuum vessel in an attempt to provide uniform contact between the heated surface and the FC. The detailed design dimensions are included in Figure 3.1.

The heater unit is thermally shielded on the back side by using four stainless steel plates connected by thin stainless steel wires with small vacuum spaces in between the plates. The combined weight of the stainless steel plates and the heater itself keeps the fabric composite stretched to maintain uniform contact with the heating surface. Three K-type thermocouples are attached to the heated surface to measure the radiating surface temperature (T_1). These thermocouples are thermally insulated by Nextel ceramic fabric tubing.

A round disk shaped aluminum heat receiving system is placed opposite the heating block in the vacuum chamber. The heat receiving surface is 27.62 cm (10.875 inches) in diameter and painted with a single layer of flat black paint to absorb heat efficiently. The cooling system is designed so that it can be pushed upward or downward to vary the distance between the heating and cooling blocks. A continuous flow of a 50% mixture of ethylene glycol and water is circulated from a chiller. Three K-type thermocouples are attached to

Figure 3.1. Schematic Diagram of the Fabric Composite Heat Transfer Measurement Experiment.

the cooling surface to measure the cold surface temperature (T_2).

Appropriate thermal shields have been used to avoid any undesired heat loss from the inside of the Fabric Composite Heat Transfer facility. Four thin, smooth stainless steel (type 304) cylinders are placed inside the vacuum vessel surrounding the heating and cooling blocks with small (typically in the order of .5 - 1. cm) spaces in between, as shown in Figure 3.2. The same type of thermal shields are also used on the back side of the electric heater. Radiation heat exchange for the enclosure has been calculated by considering the side walls as well as the heat receiving surface as black bodies [Siegel and Howell 1981]. The equation used for this calculation is given below as

$$\sum_{j=1}^N \left(\frac{\delta_{kj}}{\epsilon_j} - F_{k-j} \frac{1-\epsilon_j}{\epsilon_j} \right) \frac{Q_j}{A_j} = \sum_{j=1}^N (\delta_{kj} - F_{k-j}) \sigma T_j^4 \quad 3.1$$

where, N is the number of surfaces of the enclosure, Q/A 's are the heat fluxes across the corresponding surfaces, F 's are the view factors, T 's are the surface temperatures and δ_{kj} is the Kronecker delta defined as

$$\begin{aligned} \delta_{kj} &= 1 \text{ when } k = j \\ &= 0 \text{ when } k \neq j \end{aligned}$$

Corresponding to each surface, k takes on one of the values 1,2,.....N. When the surface temperatures are specified, the right hand side of equation 3.1 is known and there are N simultaneous equations for the unknown Q 's. The simultaneous equations are solved using a small matrix solver program which uses the tridiagonal method. In general, the heat inputs to some of the surfaces may be specified and the temperatures of these surfaces are to be determined. There are still a total of N unknown Q 's and T 's, and equation 3.1 provides the necessary number of relations. Since the values of emissivities depend on temperature, it is necessary initially to guess the unknown temperatures. Then the emissivity values can be chosen and the system of equations solved. The resulting temperature values can be used to select new emissivities, and the process repeated until the temperature and emissivity values no longer change upon further iteration. It can be noted that the results of this method will be approximate because the uniform radiosity assumption is not perfectly fulfilled over each finite area.

Calculations show that by using three metallic shields the heat loss can be reduced to less than about two percent. Four shields have been used in the operation of the Radiation Heat Transfer facility as a conservative consideration, and it has been experimentally found that the heat transfer through the last shielding wall is less than two percent.

Figure 3.2. Schematic Diagram of the Metallic Radiation Shields Inside the Vacuum Vessel.

The entire experimental system is contained within a 26.67 cm (10.5 inch) high stainless steel cylindrical pressure vessel with an inner diameter of 39.4 cm (15.5 inch). Heavy duty blind flanges have been welded to each end of the vessel, and the bottom endplate is permanently fixed to the vessel. The bottom has two small penetrations for the inlet to and outlet from the cooling system, and uses O-ring seals to provide a vacuum tight enclosure. The bottom also has a 6.35 cm (2.5 inch) hole into which the instrumentation feedthrough is fitted. This hole is covered by a teflon plate with an O-ring to keep the system vacuum tight. A three way valve is included with this endplate through which it is possible to draw a vacuum. The top endplate is removable, and has one 6.35 cm (2.5 inch) hole fitted with a teflon covering and O-ring. The teflon sheet is provided with one pair of leads for the electrical power connection, and five pairs of leads for thermocouples. The upper lid also has a pressure gauge to monitor the pressure inside the vessel. The upper end of the vessel uses a gasket and an O-ring to provide the vacuum seal. The volume of this pressure vessel is small enough and so that it is not required to meet the standards published in the ASME Boiler and Pressure Vessel Codes.

A high temperature electric heater has been designed with 15.24 cm (6 inch) on each side. This special heater is called an MI Strip heater and was purchased from the Watlow Heater Company, Portland, OR. The heater can be operated at up to 760°C (1400°F). It consists of a 10.69 mm (.042 inch) thick layer of mineral insulation which has nickel chrome element wire imbedded in it. This technology provides the heater with superb thermal conductivity, and yet retains a high dielectric strength. This thin layer of insulation results in the element wire being very close to the heater sheath. The top and bottom stainless steel sheath metals are welded together at the strategic points to maintain the high compaction of the insulation and produces a rigid, solid unit. Figure 3.3 shows the schematic diagram of the heater.

3.3. Advanced Ceramic Fabrics.

Three types of advanced ceramic fabrics have been selected for this initial investigation - Nextel, Nicalon, and Astroquartz.

Nextel is a trademark of the 3M Co. and its chemical composition is Alumina Boria Silica. Nextel fabrics are woven from strong continuous Nextel fibers without the aid of organic, glass or metal inserts. It is a mullite composition of two percent added boria. It retains its strength and flexibility with little shrinkage at continuous temperatures of up to 1370°C (2500°F).

The Nicalon ceramic fabric is woven from continuous silicon carbide ceramic fiber manufactured through a polymer pyrolysis process by Nippon Carbon Co.. The fiber is homogeneously composed of ultrafine beta-SiC crystals with excess carbon and nine to eleven percent oxygen (as SiO₂). The fiber has excellent strength and modulus properties, and retains these properties at high temperatures.

Figure 3.3. Schematic Diagram of the Electric Heater.

Astroquartz is a trademark of the JPS Co. and its chemical composition is silicon dioxide. Astroquartz II ceramic fabric is woven from Astroquartz II fiber yarns with 9779 resin compatible binder. The 9779 yarn binder is compatible with many epoxy, bismaleimide, polyimide and phenolic resin systems. In addition to a proprietary aminosilane coupling agent, 9779 binder contains film formers and lubricants to facilitate weaving. It can be used at temperatures up to 1050°C.

All three types of ceramic fabrics have been heat treated to remove their sizings prior to exposing them to high temperatures. Nextel is baked at 600°C for four hours, Nicalon and Astroquartz II are baked at 450°C for six hours to be cleaned.

The heating surface temperature (T_1) and cooling surface temperature (T_2) are noted from the data logger. The effective emissivity of the fabric composite material is calculated using the radiation heat transfer equation

$$Q/A = \epsilon \sigma (T_1^4 - T_2^4) \quad 3.2$$

where,	T_1	=	liner surface temperature (K)
	T_2	=	cold surface temperature (K)
	Q	=	amount of heat transferred (watts)
	A	=	area of the heated surface (cm ²)
	σ	=	Stefan-Boltzmann constant (W/cm ² -s-K ⁴)
	ϵ	=	effective emissivity of the fabric composite based on the heater surface temperature

Thermocouples are placed on the outside walls of the cylinders to measure any undesired heat loss through the walls. Knowing the temperatures on the outside wall of the last shield (when going outward) and the inside wall of the vacuum chamber, the heat transfer across the chamber wall, is calculated using equation 3.2. Typically the heat loss is found to be less than two percent.

The design parameters for the experiment are listed in Table 3.1. A complete list of composite materials combinations of interest which could be investigated is shown in Table 3.2 and include - copper and alloys, titanium and alloys, aluminum and alloys, stainless steel, molybdenum and teflon as working fluid retention liner materials and three different ceramic fabrics - Nicalon, Nextel and Astroquartz with all weave patterns. Kevlar is also a type of strong ceramic fabric made from continuous aramid fiber, but it can not be used above 200°C. Useful parameters that can be adjusted for the experiment include the fabric composition, weave pattern, weave density, liner composition and surface temperature. The

fabric-metal combinations which have been tested as the initial investigation are given in Table 3.3. Other useful parameters of the investigated fabric and liner materials are listed in Tables 3.4 and 3.5 respectively.

Table 3.1. The design parameters for the fabric composite radiation heat transfer experiment.

Maximum temperature	800 K
Maximum power input	500 Watts
Heat flux	$\approx 3 \text{ Watts/cm}^2$
Mass flow rate of the coolant	$\approx 2 \text{ kg/min}$
Cooling liquid	A mixture of ethylene glycol (50%) and water(50%)

Table 3.2. The possible ceramic fabric and metal liner combinations.

Fabric & liner	Fabric & liner	Fabric & liner
Astroquartz & aluminum	Nextel & aluminum	Nicalon & aluminum
Astroquartz & stainless steel	Nextel and stainless steel	Nicalon & stainless steel
Astroquartz & copper	-	Nicalon copper
Astroquartz & titanium	Nextel & titanium	Nicalon & titanium
Astroquartz & molybdenum	Nextel & molybdenum	Nicalon & molybdenum
Astroquartz & teflon	Nextel & teflon	Nicalon & teflon

Table 3.3. The fabric and liner combinations tested for initial investigation.

Fabric & liner	Fabric & liner	Fabric & liner
Astroquartz II & aluminum	Nicalon & aluminum	Nextel & aluminum
Astroquartz II & stainless steel	Nicalon & stainless steel	Nextel & stainless steel
Astroquartz II & copper	Nicalon & copper	
Astroquartz II & titanium		

Table 3.4. Parameters of the investigated ceramic fabrics.

Fabric	Weave pattern	Weight (g/m ²)	Chemical composition
Nextel	5H satin	474.0	Alumina boria silica
Nicalon	plain	280.0	Silicon carbide
Astroquartz II	8H satin	284.61	Silicon dioxide

Table 3.5. Dimensions of the investigated liner materials

Material	Length (cm)	Width (cm)	Thickness (mm)
Aluminum	15.24	15.24	2.0
Stainless steel	15.24	15.24	2.0
Copper	15.24	15.24	3.0
Titanium	15.24	15.24	2.0

3.4. Results and Analysis.

The initial investigation has been performed using the following fabric composites - Nextel and aluminum, Nextel and stainless steel, Nicalon and aluminum, Nicalon and stainless steel, Nicalon and copper, Astroquartz II and aluminum, Astroquartz II and stainless steel, Astroquartz II and copper, Astroquartz II and titanium, at three different operating temperatures. A conservative estimate based on experimental measurements of ten percent radiation heat loss from the system has been considered for the calculations. Measurements of the thermal shield temperatures show that less than two percent of the heat is lost through the side walls and less than two percent is lost through the top lid of the vacuum vessel. Therefore five percent heat loss has been estimated each way as conservative consideration.

Figure 3.4 shows the graphical representation of the effective emissivities for a bare aluminum liner and for three different ceramic fabrics in combination with the aluminum liner. It can be seen from Figure 3.4 that the Astroquartz II and aluminum combination has about seventy five percent greater effective emissivity than the bare aluminum liner at 103°C (217.4°F). In general, any of the fabric composite combinations in Figure 3.4 show significantly higher effective emissivity than the bare aluminum liner at the three operating temperatures. Figure 3.5 shows the same trend for a stainless steel liner. Again, the effective emissivities of the fabric composites are significantly higher than the effective emissivity of the bare stainless steel liner. The possible reason for this enhancement in the effective emissivities, is the large increase in the effective radiating surface area of the fabric over that of a bare metal surface since emissivity is a radiating surface property of a material. This increase in the radiating surface area is most likely due to the woven nature of the fabric. All of the combinations tested are shown on one graph in Figure 3.6.

The effective emissivity obtained for the Nextel and stainless steel combination using the cylindrical geometry of the fabric composite heat pipe test facility is $0.69 \pm .04$ at 160°C (320°F) [Kiestler 1992]. This value is comparable with the flat plate configuration result tested at 103°C (217.4°F). The values reported in [Antoniak et al 1991] for the same combination is much lower than the flat plate configuration results. This discrepancy may be because of the difference in the choice of the weave pattern or fabric density, or experimental arrangement. Another reason may be the much higher radiating surface temperature (447 °C-527 °C) used by Antoniak et al.

It can be stated at this point that, there is no possibility of convective heat transfer in this experimental arrangement because of the orientation of the heated block on the top and the cold block on the bottom of the vacuum chamber which has a negative pressure of about 100 kilo Pascal. According to the kinetic theory of gases at raised temperatures [Bird et al 1960] the slightest possible remaining gas becomes stratified, which may lead to some conductive heat transfer. To estimate the conductive heat transfer, the thermal conductivity of air at the reduced pressure and relative temperature was calculated using the thermal conductivity at the critical temperature and pressure. The heat transferred by conduction was calculated using this thermal conductivity, and it was found, that conduction contributes less than one

Figure 3.4. Plot of the Effective Emissivities for a Bare Aluminum Liner and for an Aluminum Liner with Three Different Ceramic Fabrics.

Figure 3.5. Plot of the Effective Emissivities for a Bare Stainless Steel Liner and for a Stainless Steel Liner with Three Different Ceramic Fabrics.

Figure 3.6. Plot of Effective Emissivities for Five Different Ceramic Fabric and Liner Combinations.

percent to the total heat transfer in the system. Also, it was experimentally found that, at atmospheric pressure and at sufficiently elevated temperature, the heat transferred through the stratified air in the chamber by conduction is considerably lower than the heat transferred by radiation. To evaluate this, a test was performed using an Astroquartz II and aluminum composite and no vacuum was drawn. Therefore, all of the heat was considered to be conducted through air at atmospheric pressure. The heat transferred by conduction in this case was found to be $0.48\text{E-}02$ watts/cm² at 103 °C, compared to about $8.0\text{E-}02$ watts/cm² by the radiative heat transfer process, using the same fabric composite material at the same operating temperature. Therefore it can be stated that, even if there was any chance of leakage in the vacuum, the contribution of conductive heat transfer process is negligible.

It is found that the results obtained experimentally match very well with the analytical results. As an example, the heat flux passing through the bare copper liner from the heated surface is found analytically to be $4.4\text{E-}02$ W/cm² at 103 °C (217.4°F), $18.55\text{E-}02$ W/cm² at 248° C (478.4 °F), and $24.79\text{E-}02$ W/cm² at 300 °C (572°F). The experimentally determined effective emissivities have been used in computing these values. The results match reasonably well with the experimental results which are $4.33\text{E-}02$ W/cm², $18.54\text{E-}02$ W/cm² and $24.77\text{E-}02$ W/cm² respectively for the corresponding radiating surface temperatures and effective emissivities. The deviation is less than $\pm$ two percent. Similarly, for the bare aluminum liner, the results differ by about $\pm$ one percent at 521 K and 573 K and about four percent at 376 K. The calculated values are $5.74\text{E-}02$ W/cm², $20.04\text{E-}02$ W/cm² and $27.72\text{E-}02$ W/cm² at 376 °C, 521 °C and 573°C respectively. The corresponding experimental values are found to be $5.43\text{E-}02$ W/cm², $20.11\text{E-}02$ W/cm² and $27.82\text{E-}02$ W/cm² respectively. Better agreement with the experimental results could be achieved by using finer mesh grid system.

3.5. Summary and Conclusions.

It can be concluded from the experimental results that, the Advanced Ceramic Fabric Composite Materials furnish significantly higher values of effective emissivities than the bare metallic liner materials. From the results obtained so far, the Astroquartz II and aluminum combination appeared to be the best fabric composite material. The results plotted in Figures 3.4 through 3.6 (also listed in summary form in Table 3.6) indicate that the investigated fabric composite materials have very close effective emissivities and any of the combinations is recommendable for desired space application.

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Table 3.6. Summary of effective emissivity results and comparison.

FC	Temperature 376 K		Temperature 521 K		Temperature 573 K	
	ϵ	Q/A (W/m ²)	ϵ	Q/A (W/m ²)	ϵ	Q/A (W/m ²)
Al+Astroq.	.85	819.56	.68	2953.5	.61	3945.9
Al+Nextel	.83	816.1	.63	2727.3	.6	3931.26
Cu+Nicalon	.81	797.5	.64	2819.98	.59	3916.1
Cu+Astroq.	.8	785.33	.66	2833.75	.59	3893.26
Ti+Astroq.	.81	793.9	.61	2676.85	.55	3614.6
SS+Astroq.	.79	778.92	.61	2689.1	.54	3536.33
SS+Nicalon	.75	733.15	.6	2653.3	.52	3348.0
SS+Nextel	.72	719.89	.59	2590.8	.51	3338.97
Al+Nicalon	.69	688.07	.56	2492.0	.51	3353.37

4. Bubble Membrane Radiator Condensation and Fluid Flow Experiments.

4.1 Introduction.

The Rotating Bubble Membrane Radiator (RBMR) concept was developed specifically to take advantage of the developments in fabric material technology which are combined with the change of phase in a working fluid to provide a significant reduction in radiator mass. Emissivity tests of the composite material have shown recently that significant improvements can be made which will greatly enhance the radiation of waste heat to space. The RBMR, shown in Figure 4.1, is an enclosed two-phase direct contact heat exchanger.

In this system, the two-phase working fluid enters the radiator via feed lines incorporated within the central axis of rotation. The working fluid is then ejected from the central spray nozzle as a combination of liquid droplets and vapor into the radiator envelope. Within this envelope, both convection and radiation heat transfer occur between the droplets, the vapor, and their cooler surroundings, with the dominant mode of heat transfer being radiation. As the droplets move radially outward, they grow in size by the condensation of vapor upon their surface and by collision with other droplets before striking the thin liquid film on the inner surface of the radiator. Once assimilated into the thin surface film, the working fluid begins to flow from the poles of the sphere toward the equator due to the rotationally induced artificial gravity. Heat transfer between the fluid and bubble radiator then becomes a combination of conduction and convection. As the fluid reaches the equator of the sphere, it is collected in gravity wells (troughs) and pumped back to repeat the process.

In order to begin to understand the fluid and heat transfer phenomena associated with the RBMR, a condensation experiment has been developed which reduces the problem of gravity. It is obvious that it is very difficult to simulate a spherical radiator for a space application (where the gravity is negligible) on the earth (where the gravity plays a large role). Any ground based experiment needs to take into account the force exerted by gravity on the fluid droplets to the equator of the sphere. The effect of the gravitational field can be minimized by using a flat plate in which its axis of rotation is parallel to the gravity vector. As the working fluid condenses, the centrifugal force will pump the droplets towards the edge of the plate and gravitational force does not play as important a role. As a result, it is expected that the pumping capability, fluid transport phenomena, and thermal behavior which would be expected in a spherical RBMR in zero gravity can be simulated in the flat plate experiment in normal gravity. The main part of this study is devoted to investigate the thermal behavior of the condensation phenomena on a rotating flat plate experiment in normal gravity.

The construction of a full sphere simulation, i.e. an RBMR prototype, will be necessary for the purpose of comparing results of both tested configurations, flat and sphere, on the ground based tests. Since the flat plate experiment was assumed to describe the behavior of the full sphere in microgravity, this will show the effect of the gravity on the RBMR. The RBMR prototype will be, also, the first one ever built to test the RBMR's concepts

Figure 4.1. Schematic Diagram of the Rotating Bubble Membrane Radiator.

experimentally and will be ready for the proposed future testing in microgravity.

The specific objectives of this study are :

- 1) To design, construct, operate and demonstrate a heat rejection system using condensation on a rotating flat plate, this radiator may be used in space power systems as well as in ground systems.
- 2) To find the relationships between heat transfer parameters, system temperatures, and rotating speeds for the system mentioned in 1).
- 3) Recording the condensate thickness at different system's temperatures and rotating speeds, and the condensate flow patterns at different speeds; both experimentally.
- 4) To design, construct, operate and demonstrate a full RBMR prototype.

4.2. Condensation Heat Transfer on a Rotating Flat Plate.

Laminar film condensation on a vertical plate first became a subject of analytical study in 1916 with the work of Nusselt [Nusselt 1916]. Since then, studies on condensation heat transfer have been largely concerned with the situation of gravity-induced flow. Nusselt neglected the effects of both energy convection and fluid accelerations within the condensate layer. Over the years, there have been a number of improvements to Nusselt's analysis. The pioneering work of E. M. Sparrow and J. L. Gregg in 1959 on the boundary-layer treatment of laminar film condensation and the theory of rotating condensation [Sparrow and Gregg 1959a&b] opened the door for more analytical, numerical, and experimental investigations of condensation phenomena on various rotating surfaces such as cones [Sparrow and Hartner 1961], spheres [Yang 1973], drums [Roetzel 1977], curved bodies [Osei-Bonsu and Holgate 1975], tubes with suction at the wall [Faghri and Chow 1988], finned tubes [Chandan and Watson 1976]. In the absence of gravity or the generation of sufficient centrifugal force, the orientation of the plate becomes immaterial, and an identical flow condition is achieved whether the plate is horizontal, vertical, or inclined [Butuzov and Rifert 1972].

Sparrow and Gregg analytically studied film condensation on a rotating disk situated in a large body of pure saturated vapor. The centrifugal field associated with the rotation of the plate sweeps the condensate outward along the disk surface, and gravity forces were not involved. The problem was formulated as an exact solution of the complete Navier-Stokes and energy equations. Numerical solutions were obtained for Prandtl numbers between 0.003 and 100 for $C_p \Delta T / h_{fg}$ in the range 0.0001 to 1.0. Results were given for the local heat transfer, as well as for the condensate layer thickness, torque moment, and temperature and velocity profiles. In this pioneering work, the following results were obtained :

- 1) For a particular liquid and a fixed temperature difference, the local heat-transfer rate per

unit area is proportional to the square root of the disk angular velocity, or

$$q'' \propto \omega^{1/2}$$

- 2) It is found that the local heat transfer coefficient of the condensate verses temperature difference across the film condensate, both in dimensionless groups, is represented by Figure 4.2.

By inspection of this figure, it is seen that for small values of $C_p \Delta T / h_{fg}$, all the results are represented by

$$\frac{h(\frac{\nu}{\omega})^{1/2}}{K} = 0.904 \left(\frac{Pr}{\frac{C_p \Delta T}{h_{fg}}} \right)^{1/4} \quad (4.1)$$

This equation corresponds precisely to the heat transfer result for the situation where energy convection and acceleration terms are negligible. Therefore, departures from this relationship represent the effects of convection and inertia, the former tending to increase the heat transfer and the latter tending to diminish it. For high Prandtl numbers the effect of energy convection becomes relatively more important than inertia effects, while for small Prandtl numbers the argument is vice versa. The analysis shows that for thin condensate layers, the energy convection and inertia tend to become negligible.

- 3) The temperature profile across the condensate layer closely approximates a straight line [Sparrow and Gregg 1959a].

S. S. Nandapurkar and K. O. Beatty, Jr., in 1960 published their experimental results using methanol, ethanol, and Freon-113 all at atmospheric pressure. The centrifugal force has been shown to be an effective method for the continuous removal of condensate from a horizontal plate condenser [Nandapurkar and Beatty 1960].

Measured heat transfer coefficients for condensing methanol, ethanol, and Freon-113 vapors on a rotating surface have been shown to be about 25-30% below those predicted by the theory of Sparrow and Gregg. This deviation from the theory was attributed to neglecting the effect of the vapor drag on the liquid film in the theoretical analysis, which would have a force component inward along the radius of the disk, and thus would tend to retard the condensate flow and produce a thicker condensate layer on the disk. Figure 4.3 shows some of these reported results [Nandapurkar and Beatty 1960].

E. M. Sparrow and J. L. Gregg, in a technical briefing in February of the same year,

Figure 4.2. Predicted Local Heat Transfer Coefficient for Working Fluids of Different Prandtl Numbers.

Effect of rotational speed for methanol.

Effect of rotational speed for ethanol.

Effect of rotational speed for Freon-113.

Figure 4.3. Effect of the Rotational Speeds for Different Working Fluids on Mean Heat Transfer Coefficient.

added the effect of vapor drag to their previous analysis of heat transfer and showed that the effect was quite small. It was claimed that the deviation between theory and experiment must be laid to other causes [Sparrow and Gregg 1960].

One important experimental work was done by A. I. Butuzov and V. F. Rifert in 1972 [Butuzov and Rifert 1972]. The primary motivation for this study was to determine the heat transfer coefficients for steam condensing at a rotating surface which has the form of a disk for sea-water distillation. Their experimental study of the condensation of steam was carried out on a copper disk, 0.3 m in diameter, rotating in the horizontal plane. The experimental equipment was a closed circuit, consisting of a test condenser, electric boiler, steam superheated and condensate collector. The experiments were carried out at disk speeds ranging from 10 to 224 rpm and heat flux densities from 2×10^4 to 19×10^4 W/m². In all the experiments the steam was forced to condense at the downward face, i. e. the bottom surface where Sparrow and Gregg analysis assumed the condensate to occur on the upward surface of the disk where gravitational forces have much less effect.

The experiments showed that at constant disk rotational speeds the average condensation heat transfer coefficient decreased with increasing heat flux or temperature difference and the average heat transfer coefficient was found to be proportional to the average temperature difference to the -0.25 power (characteristic of laminar condensate flow). In equation form this is given as

$$h_{avg} \propto (\Delta T)_{avg}^{-0.25}$$

For less than 52 rpm, the power of the angular velocity was 0.23, this was attributed to the fact that at low rpm the condensate film flow at disk segments closer to the center of rotation is affected by gravitation forces which makes for separation of individual droplets from the film; while for high rpm the condensate film flow and the heat transfer are determined primarily by the effect of centrifugal forces and the previous relationship was applicable.

Comparing experimental data with the theoretical predictions of the heat transfer parameter (the theoretical expression that was derived in this work is close to the one developed by Sparrow and Gregg) showed a good agreement with the experimental points lying about 5-10% below the theoretical predictions. This was attributed to the fact that the theoretical expression derived in this work has neglected the circular motion of the condensate film on the rotating disk, which increases the film thickness and the thermal resistance to heat transfer.

The rotation of the surface aids in augmenting the heat transfer during the film-wise condensation of steam. All other conditions remaining equal (the same temperature difference and surface area) the average heat transfer coefficient for a rotating disk is 3-5 times greater than the heat transfer from steam condensing on a stationary vertical surface, where the condensate flows by gravity [Butuzov and Rifert 1972].

In 1972 P. M. Beckett, P. C. Hudson and G. Poots [Beckett et al 1972] employed regular perturbation theory methods to investigate the problem of laminar film condensation on a rotating disk in a large volume of quiescent vapor. Exact numerical solutions were obtained for the case of steam-water condensation and compared with those derived analytically. The flow configuration was as follows. An infinite disk is held at constant temperature with controlled known angular velocity in a large volume of quiescent pure vapor at the saturation temperature, and greater than the disk temperature. A condensate film is formed on the disk and a steady state axially symmetric flow exists in which both vapor and condensate are swept radially outward in spiral paths away from the axis of rotation. The steady renewal of condensate maintains the interface in a fixed position. The actual location being dependent on the magnitude of rotation and the rate of cooling of the disk surface. The analytical results of this work showed better agreement with experiments [Nandapurkar and Beatty 1960] than those proposed by Sparrow and Gregg [Sparrow and Gregg 1959a].

4.3. Description of the Methods and Experimental Models.

The flat plate configuration was chosen initially to reduce the effects of gravity on the simulation before proceeding to the next step of testing different configurations such as spherical or ellipsoidal shells. Analysis has shown that the tangential force, at certain location and rotational speed, on a condensate droplet of a working fluid on the inner surface of a RBMR may be simulated by the same amount of force, generated by a known flat plate rotational speed, acting on the same size of droplet of same working fluid, located at some distance from the center of a flat plate.

The primary components of the ground based Flat Plate Condensation Experiment and their uses are shown as follows :

- 1) A large glass pipe section with a length of one meter and a forty five centimeter diameter. It was constructed in England by Corning Glass Works, and was designed especially to hold zero vacuum. It was used to contain the major components of the experiment as shown in Figure 4.4. A transparent glass pipe was chosen to allow the use of laser beams to be shined through it in an attempt to measure the film condensate thickness. The glass pipe is sealed on its ends by two Aluminum plates. The bottom plate has adjustable legs to help in leveling the glass pipe, and also has a vacuum valve that connects the glass pipe to the vacuum pump, and an electrical feed through to connect the rotating motor to a power supply and controller. The upper plate has a plastic window to allow video-taping the flow pattern. It has also a steam valve, pressure gauge, and two feed through that are used while recording the flow patterns. Both plates are attached to the glass pipe by metal flanges and built-in screws.
- 2) The rotating flat plate assembly, shown in Figure 4.5, consists of two parts:
  - * Two flat plates of metal with high thermal conductivity. The upper one is a mirror type

Figure 4.4. Schematic Diagram of the Flat Plate Condensation Experiment.

Figure 4.5. Schematic Diagram of the Flat Plate Condensation Assembly.

(#8 finish) stainless steel, alloy SS304, where the condensation is designed to occur. This plate simulates the radiator's condensing surface and is 30.48 cm in diameter and 0.265 cm thickness. The mirror finish helps in smoothly sweeping the condensate to the plate edge when it is rotated, and also to, effectively, reflect the laser beam when the condensate thickness is measured. The other plate is made of Aluminum 6061-T6, with a 30.48 cm diameter and a 0.635 cm thickness. This plate is attached to the ice bucket, and is used to measure the heat flux from the stainless steel plate to the block of ice which is pressed against the lower surface of the aluminum plate. By measuring the temperature difference across the aluminum plate the heat flux can be determined.

- * An ice bucket which provides a constant temperature heat sink for the experiment. This is achieved by attaching several springs to a teflon platform tape which pushes an ice block against gravity to keep it in contact with the bottom surface of the aluminum plate at all times. The teflon platform has some free spaces (holes) to allow the melting water to pass through. After each experiment run, the ice bucket is frozen up-side-down with the teflon platform springs pulled back to allow the formation of an ice block. The ice block then simulates the cold environment of space. The ice-bucket is thermally insulated from the sides and bottom so that only a limited amount of condensation should occur on surfaces other than the upper one.

3) The water bucket :

It consists of two Aluminum pipe sections with their bottoms fixed to an aluminum plate as shown in Figure 4.5; the inner pipe is used to hold the rotating flat assembly and the data logger while the outer one is used to collect the condensate.

4) Variable speed motor and amplifier :

A DC servo system motor, type 0642-30-017 from Robins & Myers, environmentally sealed, with maximum continuous speed of 4500 rpm is used in these experiments. This motor is connected to a matching Electro-Craft LA-5600 linear servo amplifier. The LA-5600 amplifier is designed to drive small and medium size permanent magnet DC servo motors with DC tachometers in a velocity control mode. The motor has a 14V/Krpm tachometer. This system allows the speed of rotation to be accurately controlled.

5) Vacuum pump :

It is a mechanical high vacuum pump, model 5BA-1-G482x, GAST brand. Evacuating the system helps in getting high superheated steam at relatively low temperatures. Low steam temperatures also decrease the condensation on the other system parts, and also very few air molecules in the system for the steam to be condense on. A small amount of air in the system means less non-condensable gases which helps in getting more accurate results. It also reduces the amount of condensation on the inner surface of the glass pipe which helps in better visualizing of the condensate film.

6) LASER beam :

Helium-Neon laser beam, model 100 series, UNIPHASE brand, was used to attempt find the thickness of the condensate. It has 632.8 nm wavelength, i.e. visible beam, with 10.0mW power, 320 MHz longitudinal mode spacing, 0.68 mm diameter, and linear polarization. This type of laser beam is classified as a Class III-medium power laser.

7) Data logger :

A compact, portable battery-operated model 50 data logger from Cole-Parmer which has five thermocouple channels is used. The output data can be either printed on the 24-column thermal printer or loaded on a remote computer. Each individual channel of the data logger is programmable directly from the keyboard. The programming systems include: time, log interval, thermocouples type (J, K, T, or E), temperature unit (°F or °C), etc. T type thermocouples are used in these experiments. The data logger is used to measure the temperature of the aluminum plate at different locations.

8) Steam supply system :

A 2000 ml beaker is used as a water reservoir. It is placed on a heating plate. The beaker has two openings; one is closed by a stopper while the other is connected to the glass pipe by a hose so that when the steam is generated, it is forced to enter the glass pipe.

9) Black Light :

A black-ray ultraviolet lamp with 115 volt and 60 cycle, model B-100A, of Ultra-Violet Products, Inc., is used to illuminate a fluorescent solution that is added at the center of the condensing surface to watch the condensate flow patterns.

4.4. Methodologies.

Several important assumptions were made that have significant effects on the results. These effects caused less accuracy in the results and the effects were measured and calculated by performing statistical analysis.

The first assumption is that the heat is transferred from the condensate to the ice in a steady state mode. The fact that was noticed from the temperature measurements that the temperature increases very fast initially and then continues to increase with a very small slope where it is assumed to be in a quasi steady state condition. It takes a few minutes to reach the appropriate quasi steady state conditions.

Secondly, the heat is transferred in one dimension, i.e. the Z-direction in the cylindrical coordinates. Very little variation in R-direction, in comparison to Z-direction, was noticed.

The third significant assumption is that the aluminum plate and the stainless steel plate

are in a very close contact in such a way as to neglect the effect of the gap between them. This gap is caused by placing very thin thermocouples on the aluminum surface that is in contact with the stainless steel plate. As a matter of fact, this gap serves as a resistance that should be counted in calculating the heat transfer parameters. For reasonably high temperature difference across the gap however this assumption is quite reasonable as will be shown later.

A common way of expressing the heat transfer rate for a situation involving a composite material or combinations of heat transfer mechanisms is by determining the overall heat transfer coefficient (U). In cases where the condensation is involved, U1 is defined as

$$U1 = \frac{\frac{\dot{m} h_{fg}}{A}}{T_{wv} - T_{Ali}} \quad (4.2)$$

where

$$\frac{q}{A} = \frac{\dot{m} h_{fg}}{A} \quad (4.3)$$

An alternative way to measure the heat flux at steady state in equation (4.2), which is due to condensation, is to measure the temperatures across the condensate. For the one dimensional steady state case, the amount of energy which crosses the condensate will be same as the amount of energy which crosses the stainless steel plate coming from the condensate, and the same as that which crosses the aluminum plate and is eventually deposited in the ice block, and thus the heat flux can be measured as

$$\frac{q}{A} = \left(\frac{K_{Al}}{t_{Al}} \right) (T_{Alo} - T_{Ali}) \quad (4.4)$$

A convenient way, using $(T_{wv} - T_{Ali})$ as a temperature difference, to find U1 is then found to be

$$U1 = \frac{\left(\frac{q}{A}\right)}{(T_{wv} - T_{Ali})} \quad (4.5)$$

and, using $(T_{wv} - T_{Alo})$ as a temperature difference, U2 to be

$$U2 = \frac{\left(\frac{q}{A}\right)}{(T_{wv} - T_{Alo})} \quad (4.6)$$

Since T_{wv} , T_{Ali} , T_{Alo} , and t_{Al} are measured, and K_{Al} is known by using tables of physical properties for aluminum alloys, the only unknowns are the heat flux (q/A) , U1 and U2.

To find the average convective heat transfer coefficient, the following equation is used

$$T_{wv} - T_{Ali} = \left(\frac{q}{A}\right) \left[\frac{1}{h_{avg}} + \frac{t_{Al}}{K_{Al}} + \frac{t_{ss}}{K_{ss}} \right] \quad (4.7)$$

where t_{ss} is measured and K_{ss} is known by using tables of physical properties for stainless steel alloys, the only unknown is h_{avg} .

The average condensing surface temperature (T_c) is calculated using the following equation

$$\frac{q}{A} = h_{avg} (T_{wv} - T_c) \quad (4.8)$$

4.5. Condensate Film Thickness Measurements.

A technique to measure the condensate fluid thickness using a laser beam has been developed. A laser beam will be directed towards the disk surface (from outside the glass container), with a known angle relative to the plate surface. A part of the beam reflects from the film surface while the remainder of the beam refracts in the condensate film and then reflects from the disk surface. A laser receiver receives these two beams, and since the shift

between these beams is proportional to the thickness of the film, the film thickness can be found using the known geometry, index of refractions of the system, and basic laws of refraction and reflection. Figure 4.6 demonstrates the principles of operation of this aspect of the experiment and shows the equation that is used to find the film thickness.

4.6. Flow Pattern Observations.

To record the flow patterns of the colorless condensate water, a shiny (light) or colored (dye) materials must flow with the condensate without disturbing its flow pattern. It was found that a water solution of Sodium Fluorescein will not disturb the flow patterns since its density is similar to the condensate and it can be introduced at the center of the condensate plate. If this is the case, this shiny solution will flow with the condensate and by using matching black light, the solution will illuminate and its motion be recorded by either a video tape camera or a photo camera. When the quasi steady state case is achieved with an established condensate film, the solution is introduced into the system. The amount of solution to be entered and the location where to introduce are controlled by a controlling valve and a stick.

4.7. Results and Discussion.

In experimental heat transfer research, one of the first objectives, is to measure the temperature distribution on the tested surface. These temperature measurements form the basis for following calculations to obtain the final output from such a study which is usually heat transfer coefficients or related correlations.

After the system reaches a quasi steady state condition, the readings of the data logger temperatures were taken over a certain period of time and then averaged. The temperatures T_1 , T_2 , T_3 , and T_4 are measured in this fashion. T_{Ali} and T_{Alo} were calculated by averaging the temperature readings of the two channels measuring the inner Aluminum temperatures and the other two channels measuring the outer Aluminum surface, respectively, at the quasi steady state conditions. Standard deviations for these temperatures were found using a spreadsheet. The temperature of the interfacial area between the steam and the condensate film was taken to be the saturation temperature corresponding to the pressure measured in the system. This saturation pressure varies very little once the system reaches a quasi steady state condition. As a result one temperature reading was recorded for T_{wv} and hence no standard deviation was obtained for it. It is worth while to note that for tests 9, 12, 15, and 16 the thermocouple locations were as follows: two channels, 1 and 3, at the plate center on opposite sides, and two channels, 2 and 4, at 1.5" off the plate center, in both sides of the plate. For the rest of the tests the thermocouple locations were: two channels, 2 and 4, at 1.5" off the plate center on opposite sides, while channels 1 and 3 were located at 3.0" off the plate center in both sides.

Figure 4.7 shows a typical example of an output profile of the temperature readings from the data logger. When the flat plate assembly is taken out of the freezer, temperatures of channels 1 through 4 start to increase due to their exposure to room temperature. When the

$$A=B \quad L=\text{Film Thickness}$$

$$L = \frac{B}{\sin(b)} * \sin(c) * \sin(d)$$

Figure 4.6. Fluid Film Thickness Measurement Using Laser Beam.

Figure 4.7. A Typical Temperature Profile for Current Experiments.

flat plate assembly is inserted into the system the temperature profiles seemed to behave almost steady state and kept this profile until, all of the sudden, they started to increase rapidly. This sharp increase was when the thermocouples started to feel the effect of the introduction of steam. The thermocouple connected to channel 5 was used to watch the heat behavior in the system. Since this thermocouple can not read the right temperature of a water vapor in a system of low pressure, its readings were not used. The first part of the curve and the first steep decrease, in area (1), represent the period where the ice-bucket was not yet put in the system. The second steep decrease of the temperature profile of this thermocouple, in area (2), shows when the system was evacuated, while the increase after that was, due to the heat rejected from the motor. Then, in area (3), due to introducing the steam. By inspecting area (3), it is seen that there was a short lag period between the time the steam was introduced, which is indicated by temperature increase of thermocouple of channel 5, and the time where thermocouples 3 and 4 felt the temperature increase. Due to the high thermal conductivity of aluminum no such time lag period was noticeable between thermocouples 3 and 4, and 1 and 2. After that the temperature profile started to behave as if it was in a quasi steady state condition, area (4). The last several temperature readings of this area were used. Then the decrease in these profiles, area (5), were due to turning the motor off and allowing air of a room temperature to enter the system.

Figure 4.8 shows a magnification of the quasi steady state period, the black area in the bottom left shows the magnified area from the original whole curve. It is clear that the temperature profile throughout this minute and a half period represents a quasi steady state case. The temperature data was taken every ten seconds in this case.

4.8. Heat Flux And Overall Heat Transfer Coefficient Calculations.

It is found that the film condensate temperature is on the order of a few tens of degrees celsius. C_p varies very slowly for water in this temperature range, e. g. it goes from 4226 J/Kg. $^{\circ}$ K at 0 $^{\circ}$ C to 4175 J/Kg. $^{\circ}$ K for 40 $^{\circ}$ C [Welty et al 1984]. Also h_{fg} varies slowly, it is 2.5×10^6 J/Kg for 0.01 $^{\circ}$ C and 2.4×10^6 for 40 $^{\circ}$ C [VanWylen and Sonntag 1985]. Since it is very difficult to get the exact temperature of the film condensate, the values of the C_p and h_{fg} were used for a temperature of 20 $^{\circ}$ C, i. e. predicted mean value of the condensate, throughout the calculations, i. e. 4182 J/Kg. $^{\circ}$ K and 2.45×10^6 J/Kg respectively. The results are expected to be insensitive to this assumption.

The heat flux was plotted against two dimensionless temperature difference groups, namely $C_p (T_{wv} - T_{Ail}) / h_{fg}$ and $C_p (T_{wv} - T_{Alo}) / h_{fg}$, and for the three tested speeds individually. Figure 4.9 shows the heat flux verses $C_p (T_{wv} - T_{Ail}) / h_{fg}$ for rotational speed of 200 rpm, all data point are shown with their standard deviations in X-axis and Y-axis. Figures 4.10 and 4.11 show same things for rotational speeds 300 rpm and 400 rpm, respectively. Figure 4.13 shows the heat flux verses $C_p (T_{wv} - T_{Alo}) / h_{fg}$ for rotational speed of 200 rpm, all data point are shown with their standard deviations in X-axis and Y-axis. Figures 4.14 and 4.15 show same things for rotational speeds 300 rpm and 400 rpm, respectively. The least square fit of all data together is shown in one plot, i. e. Figure 4.12 for the heat flux verses $C_p (T_{wv} - T_{Ail}) / h_{fg}$ and Figure 4.16 for the heat flux verses $C_p (T_{wv} -$

Figure 4.8. A Typical Temperature Profile of a Quasi Steady Period of Current Experiments.

Figure 4.9. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_1 , for a Rotational Speed of 200 rpm.

Figure 4.10. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_1 , for a Rotational Speed of 300 rpm.

Figure 4.11. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_1 , for a Rotational Speed of 400 rpm.

Figure 4.12. Least Square Fit for the Three Tested Rotational Speeds of Heat Flux Verses Temperature Difference Used in U1.

Figure 4.13. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_2 , for a Rotational Speed of 200 rpm.

Figure 4.14. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_2 , for a Rotational Speed of 300 rpm.

Figure 4.15. Heat Flux Verses Dimensionless Temperature Difference, Used in Calculating U_2 , for a Rotational Speed of 400 rpm.

Figure 4.16. Least Square Fit for the Three Tested Rotational Speeds of Heat Flux Verses Temperature Difference Used in U2.

$T_{Alo}) / h_{fg}$. Since it is well known that when there is no temperature difference there is no driving force for heat transfer available, and q'' is zero when ΔT approaches zero. The least squares fits were all therefore forced to pass through the origin. The slopes (m) for these three curves for the heat flux versus $C_p (T_{WV} - T_{Ali}) / h_{fg}$ were found to be :

$$\begin{aligned} m_{200 \text{ rpm}} &= 2.05 \times 10^6 \text{ W/m}^2 \\ m_{300 \text{ rpm}} &= 2.75 \times 10^6 \text{ W/m}^2 \\ m_{400 \text{ rpm}} &= 3.21 \times 10^6 \text{ W/m}^2 \end{aligned}$$

While those slopes for heat flux versus $C_p (T_{WV} - T_{Alo}) / h_{fg}$ were found to be :

$$\begin{aligned} m_{200 \text{ rpm}} &= 2.31 \times 10^6 \text{ W/m}^2 \\ m_{300 \text{ rpm}} &= 3.27 \times 10^6 \text{ W/m}^2 \\ m_{400 \text{ rpm}} &= 3.89 \times 10^6 \text{ W/m}^2 \end{aligned}$$

All slopes of the least square curves are positive. Since increasing temperature difference will increase the heat flux, positive slopes are expected.

According to the theory of Sparrow and Gregg [Sparrow and Gregg 1959a] for a particular liquid and a fixed temperature difference, the local heat-transfer rate per unit area is proportional to the square root of the disk angular velocity and the relationship should be

$$q'' \propto \omega^{1/2}$$

This relation implies that

$$\left(\frac{q''_{400 \text{ rpm}}}{q''_{300 \text{ rpm}}} \right) = \left(\frac{\sqrt{400}}{\sqrt{300}} \right) = 1.155$$

$$\left(\frac{q''_{400 \text{ rpm}}}{q''_{200 \text{ rpm}}} \right) = \left(\frac{\sqrt{400}}{\sqrt{200}} \right) = 1.414$$

$$\left(\frac{q''_{300\text{rpm}}}{q''_{200\text{rpm}}}\right) = \left(\frac{\sqrt{300}}{\sqrt{200}}\right) = 1.225$$

Tables 4.1 and 4.2. show the comparison between the theoretical predictions and the experimental results.

Table 4.1. Comparison Between Theoretical And Experimental Results Of The Rotational Speed Effect On Heat Flux Using $(T_{wv}-T_{Air})$.

EQUATION (m_{rpm})	THEORETICAL PREDICTION	EXPERIMENTAL RESULT	ERROR %
m_{400} / m_{300}	1.155	1.168	1.105
m_{400} / m_{200}	1.414	1.565	10.661
m_{300} / m_{200}	1.225	1.340	9.384

Table 4.2. Comparison Between Theoretical And Experimental Results Of The Rotational Speed Effect On Heat Flux Using $(T_{wv}-T_{Air})$.

EQUATION (m_{rpm})	THEORETICAL PREDICTION	EXPERIMENTAL RESULT	ERROR %
m_{400} / m_{300}	1.155	1.188	2.884
m_{400} / m_{200}	1.414	1.565	18.794
m_{300} / m_{200}	1.225	1.340	15.534

The errors shown in these tables are acceptable for a system which has several sources of error. These will be discussed later. It is noticed that the lowest errors were at the higher rotational speeds where the effect of drop-wise condensation is lowest. The experimental results are in a good agreement with the theory, and the heat flux behaved with the rotational speeds as it should. This result is very important since the heat flux is the basis for the following heat transfer parameters in this research.

The overall heat transfer coefficients verses corresponding temperature difference were plotted for each speed individually, i. e. Figures 4.18 through 4.20 and 4.22 through 4.24, and then for the three speeds together, i. e. Figures 4.17 and 4.21. On the same plots,

Figure 4.17. Overall Heat Transfer Coefficient, $U1$, Correlation Verses Temperature Difference of Working Fluid and Cold Surface. Ref. S = Sparrow and Gregg 1959a. Ref. N = Nandapurkar and Beatty 1960.

Figure 4.18. Overall Heat Transfer Coefficient, U_1 , Correlation Verses Corresponding Temperature Difference for a Rotational Speed of 200 rpm. Ref. S = Sparrow and Gregg 1959a.

Figure 4.19. Overall Heat Transfer Coefficient, U1, Correlation Verses Corresponding Temperature Difference for a Rotational Speed of 300 rpm. Ref. S = Sparrow and Gregg 1959a.

Figure 4.20. Overall Heat Transfer Coefficient, U1, Correlation Verses Corresponding Temperature Difference for Rotational Speed of 400 rpm. Ref. S = Sparrow and Gregg 1959a.

Figure 4.21. Overall Heat Transfer Coefficient, U2, Correlation Verses Temperature Difference of Working Fluid and Cold Surface. Ref. S = Sparrow and Gregg 1959a. Ref. N = Nandapurkar and Beatty 1960.

Figure 4.22. Overall Heat Transfer Coefficient, U_2 , Correlation Verses Corresponding Temperature Difference for a Rotational Speed of 200 rpm. Ref. S = Sparrow and Gregg 1959a.

Figure 4.23. Overall Heat Transfer Coefficient, U_2 , Correlation Verses Corresponding Temperature Difference for a Rotational Speed of 300 rpm. Ref. S = Sparrow and Gregg 1959a.

Figure 4.24. Overall Heat Transfer Coefficient, U2, Correlation Verses Corresponding Temperature Difference for a Rotational Speed of 400 rpm. Ref. S = Sparrow and Gregg 1959a.

theoretical correlation results found by E. M. Sparrow and J. L. Gregg, and the experimental results of S. S. Nandapurkar and K. O. Beatty, Jr., for local heat transfer coefficients were also plotted in order to compare the current results with their results. Two types of curve fitting were used in this study, the least square and a line representing the average data point. All previous studies were plotted for the same dimensionless groups in order to facilitate the comparisons. It is worthwhile to mention that for the S. S. Nandapurkar and K. O. Beatty, Jr. study the lowest rotational speed was 400 rpm, which is the maximum in this study, and they have 5 data points at this speed for methanol and ethanol cases and 4 data points for Freon-113. However their results were found to be consistently 25-30% below the predictions of the E. M. Sparrow and J. L. Gregg correlation for all speeds. Data points of this study were plotted with their standard deviation to show the probable amount of fluctuation in this particular data point. In all cases, curve fittings used in the current study were very close to the analytical curve developed by E. M. Sparrow and J. L. Gregg and also closer to it than the experimental results which were obtained by S. S. Nandapurkar and K. O. Beatty, Jr.

The average local heat transfer coefficients were plotted against the average calculated temperature difference across the film condensate, both in dimensionless groups, for the flat plate rotational speeds of 200, 300, and 400 rpm, together. The best curve fit for all data points is shown in Figure 4.25. In Figure 4.26, only values of $C_p \Delta T / h_{fg} \geq 0.105$ were considered and then same comparing curves and best fittings used in analyzing data for overall heat transfer coefficients were used too.

The high errors shown in low values of $C_p \Delta T / h_{fg}$, i.e. for values ≤ 0.015 , should be attributed to neglecting the heat resistance due to the gap effect between the Aluminum plate and the Stainless steel plate. Low ΔT tends to have higher errors, while when ΔT increases, the assumption of neglecting the gap effect on heat transport seems to be reasonable. It was not easy to precisely measure the tiny gap for each experiment.

4.9. Thickness Measurement And Flow Patterns.

Several experiments were performed attempting to find the film condensate thickness at all tested rotational speeds using expected resolution of reflected laser beam, from the condensing surface and the condensate film, on laser receivers.

A few problems were faced in measuring the condensate film thicknesses. The first major problem was the system vibration that develops wavy films in the condensate which let the laser beam to look like a fuzzy moving point at the laser receiver. A second problem, the laser beam diameter was wider than the condensate film thickness which causes overlapping points at the laser receiver. This gets worse as the rotational speed increases. Less important reasons why the measurements of the thickness were difficult were drop-wise condensation at the center of the condensing surface, and condensation on the inside surface of the glass pipe.

Figure 4.25. Best Fit of the Average Heat Transfer Coefficient Correlation Data Points.

Figure 4.26. Average Heat Transfer Coefficient Correlation Verses Temperature Difference across the Condensate Film. Ref. S = Sparrow and Gregg 1959a. Ref. N = Nandapurkar and Beatty 1960.

When the technique was tested for 0 rpm under standard pressure and temperature with water film thicknesses greater than the laser beam diameter, the technique worked nicely. As mentioned previously, no study has been found dealing experimentally with the condensate film thickness measurements on rotating geometries; and the current study fails to contribute. The results found here show pictures of the laser receivers where laser beams could not be recognized from each other as desired or too fuzzy to be noticed. None of these preliminary results were considered and additional research is needed in this area.

Few preliminary results to investigate the change of flow patterns with the rotational speeds were videotaped at all tested rotational speeds. The technique and results looked to be working nicely. Since most of the problems that were faced in measuring the condensate thickness were also faced in studying condensate flow patterns, no further experiments were performed. It was set as a proposed future work. Since reasonable efforts were spent on this study, it is worth to be mentioned. These preliminary results showed curved flow patterns which gave the expected result.

4.10. General Error Sources Discussion.

A number of specific errors were mentioned in different places in this section for specific results. The following sources of errors are common between all tests and cause some degree of uncertainty in the results.

1) One-Dimension Heat Transfer Assumption :

By inspecting the data, one can compare the temperature difference ($T_{wv} - T_{AE}$) which causes the heat to be transferred in the Z-direction, and the temperature difference of ($T_1 - T_2$) or ($T_3 - T_4$) which causes the heat to be transferred in the R-direction. T_{wv} represents the temperature of the interface area between the water vapor and condensate films, and T_{AE} represents the average temperature of the inner aluminum plate surface. The distance between these two locations in Z-direction equals the aluminum plate thickness (0.635 cm), the stainless steel plate thickness (0.265 cm), the gap between the plates, plus the film condensate thickness. The last two thicknesses were very small and difficult to measure, being conservative, it could be assumed that both of these thicknesses together equal to 0.2 cm. As a result, the total thickness in Z-direction was about 1.1 cm. The distance between the thermocouples on each side of the aluminum plate, which is in R-direction, was 3.81 cm.

A simple way to compare the amount of heat transfer in the Z-direction to that in R-direction is to look at the temperature gradient, in both directions. By inspecting the data, one can conclude that the temperature drop per centimeter in R-direction is very much lower than that of Z-direction. An example, randomly chosen, for each rotational speed is shown in Table 4.3. As a result, the one-dimension, i. e. Z-direction, heat transfer assumption seems to be reasonably accepted.

Table 4.3. Comparison Of Temperature Drop Per Centimeter In R And Z Directions.

Log # (rpm)	R-direction (°C/cm)	Z-direction (°C/cm)	(R-dir./Z-dir.)%
25 (200)	1.847	36.566	5.051
30 (300)	2.083	38.350	5.433
47 (400)	3.436	40.600	8.464

2) Steady State Condition Assumption :

Figures 4.7 and 4.8 have shown a typical sample of the output data. It is clear that a complete steady state condition in which the temperature does not change with time was not strictly achieved. But, the figures also showed that the temperature change with time was not too large. It was claimed that a quasi steady state condition was achieved where the temperature changes very slowly with time. A complete steady state problem could not be reached even for long time runs due to the fact that the ice block melts.

Assuming a fully steady state problem should add an error to the results. The amount of error depends solely on how much the temperatures change with time. Smaller change means less error and vice versa. In almost all of the adopted cases of this study, the change was very small and Figure 4.8 a typical result. Those runs that did not satisfy this condition were rejected. Thus, the assumption is reasonable and the error resulting from it should be acceptable.

3) Drop-Wise Condensation :

As was discussed earlier, drop-wise condensation is associated with much higher heat transfer rates than film-wise condensation. Since the theoretical model used for comparison assumed film-wise condensation, it is required to obtain film-wise condensation for a good comparison. Even though surface treatment was used to get film-wise condensation over the entire condensing surface, drop-wise condensation was noticed at the center area in several cases. It was noticed also that higher rotational speeds tend to decrease this area of drop-wise condensation. This seems logical since more force would be applied on the droplets to push them outward to join with other droplets forming a film.

However, it was thought that for some cases this source of error had the highest effect on the results where they tend to increase the heat transfer. Therefore, no run which had drop-wise condensation on areas other than the center area was included in the analysis.

4) Non-Condensable Gases :

In all experiments, the glass pipe was evacuated to 6-7 KPa (the maximum which could be reached using the available system) before introducing the steam. This is expected to reduce the effect of non-condensable gases dramatically. As mentioned previously, this source of error tends to reduce the heat transfer coefficient.

5) System Vibration :

In all tests it was observed that as the rotational speed increased the system vibration increased. This might affect the expected condensate thickness shape which in order affects the expected distribution of the heat transfer along the condensing surface. Even though this source of error was thought to have the least effect, it is worth mentioning.

4.11. Special Experiment Run Cases.

Several other experiments were made to see the effects of various parameters on the experimental results. The main two parameters were the effect of evacuation and the effect of zero rotational speed. Two experiments were conducted, one for standard pressure and the other for 85 KPa, each one of them used the three tested speeds. Comparing the results of these two experiments with the other ones where the starting operating pressure was 6-7 KPa, it was found that the heat flux was much lower in these two cases than in the others. This was attributed to the following: less steam could enter the system in these two experiments than in cases where the vacuum was low enough to draw in a sufficient amount of superheated water vapor to fill the system. As a result not enough steam was available to simulate the tested model where the system is full of steam to accommodate the maximum demand of the condensing surface. One other reason could be the presence of air, i.e. non-condensable gases, in the system which tends to reduce the heat transfer coefficient. The results of these two experiments, are shown in Table 4.4 and 4.5.

In addition, three experiments were conducted at 0 rpm in exactly the same way as the other experiments were conducted. As it is shown in Table 4.6, the measured heat transfer parameters are much higher than those supposed to be, for film-wise condensation, at 0 rpm. It was visually clear that drop-wise condensation was obtained in these experiments which explains why more heat was transferred. This supports the need to put more force on the condensate droplets to join each other forming a film. Creating centrifugal forces through rotating the flat plate at higher rotational speeds is the easiest way to get such a force in such a system.

Table 4.4. Heat Transfer Correlation For U1 And U2 At Standard Pressure ($T_{wv}=100^{\circ}\text{C}$).

Speed (rpm)	T_{Ali} ($^{\circ}\text{C}$)	T_{Alo} ($^{\circ}\text{C}$)	q'' (W/m^2)	$C_p \Delta T_{U1}/h_{fg}$	Corr. using U1	$C_p \Delta T_{U2}/h_{fg}$	Corr. using U2
200	8.733	12.059	87458	0.16	0.229	0.154	0.235
300	4.075	8.742	122725	0.168	0.252	0.160	0.262
400	1.184	4.983	99923	0.173	0.174	0.166	0.179

Table 4.5. Heat Transfer Correlation For U1 And U2 At 85 KPa ($T_{wv}=95^{\circ}\text{C}$).

Speed (rpm)	T_{Ali} ($^{\circ}\text{C}$)	T_{Alo} ($^{\circ}\text{C}$)	q'' (W/m^2)	$C_p \Delta T_{U1}/h_{fg}$	Corr. using U1	$C_p \Delta T_{U2}/h_{fg}$	Corr. using U2
200	8.867	12.275	89641	0.151	0.245	0.145	0.253
300	6.383	9.784	89430	0.155	0.195	0.149	0.201
400	2.45	6.275	100594	0.162	0.184	0.155	0.190

Table 4.6. Heat Transfer Measurements At 0 rpm.

T_{wv} (°C)	T_{Ali} (°C)	T_{Alo} (°C)	q'' (W/m ²)	Cp1	U1 (W/m ² /°C)	Cp2	U2 (W/m ² /°C)
47.442	3.060	11.040	197770	0.078	4456.1	0.064	5432.9
45.810	2.220	8.480	152798	0.076	3505.3	0.065	4093.2
47.442	4.260	11.925	203950	0.076	4723.0	0.062	5742.3

4.12. RBMR Prototype Design and Construction.

A full sphere RBMR test stand was designed and constructed to simulate one of the proposed configurations of the RBMR. Figure 4.27 shows a schematic drawing of major parts and equipment of this RBMR test.

An investigation was made to compare the effect of various rotational speeds on a working fluid droplet on a rotating flat plate and on the inner surface of a rotating sphere. Both droplets were assumed to have the same size, are swept on perfect frictionless surfaces, and in a micro-gravity environment, i.e. gravitational forces were neglected. Results of this study are shown in Figures 4.28 and 4.29.

Since the normal force on droplets inside the sphere will be canceled with the reaction from the sphere wall, it will not have any effect on moving the droplet. Analysis, from Figures 4.28 and 4.29, has shown that the tangential force on a droplet of the working fluid on the inner surface of a RBMR with a one meter radius located at 45° angle from the equator and rotating at 45 rpm is equal to the force acting on the same quantity of working fluid located 7.0 cm from the center of a flat plate rotating at about 330 rpm. This is also the same amount of force exerted on that droplet located on the inside of a sphere of 4.2 meter radius at about 10 rpm and 10.0 centimeter from the center of a flat plate rotating at 300 rpm. Thus, as it is shown in Figure 4.29, in order to simulate the fluid and convective heat transport of a droplet of an in-space RBMR with a 4.2 meter radius at 45° angle from the equator, to that at 10.0 centimeter from the center of the flat plate, operating with rotational speeds from 2 to 16 rpm, the flat plate must rotate at speeds from 50 to 400 rpm. These figures show examples where many similar studies may be made for comparison or simulation goals on normal gravity.

Testing a prototype of the RBMR in microgravity is a very important step to validate the concept for future considerations. The prototype that was built through the efforts of this research can be modified for use in a microgravity flight to test the operational principles of the RBMR.

The primary components of the RBMR Condensation Experiment are shown in Figure 4.27. Their purposes are described as follows.

- 1) A spherical shell made of Aluminum 6061-T6 with a 0.638 cm (0.25") thickness. It consists of two hemispheres attached to each other with screws. The bottom hemisphere has a stationary base where instrumentation and pipes may be inserted to the inside of the sphere. This stationary base is well sealed against vacuum leakage with suitable bearings.

The main function of the sphere is that the outer surface should be cooled enough to condense the working fluid vapor on the inner surface.

Figure 4.27. Schematic Diagram of the RBMR Prototype.

Figure 4.28. Sphere of One Meter Radius and Flat Plate rpm Verses Force Acting on a Droplet.

Figure 4.29. Sphere of 4.2 Meter Radius and Flat Plate rpm Verses Force Acting on a Droplet.

2) Instrumentation

Several measuring instruments are used. The data logger with different types of thermocouples is used to record the temperatures of the condensate working fluid in the condensate collecting tank. The temperature of the sphere environment during operation is taken from the saturation temperature for the pressure transducer, model 891.23.560 from WIKA, Klingenberg, Main, reading for the inside sphere.

Some additional vacuum/pressure gauges are used in various locations in the system as shown in Figure 4.27. Also relief valves are used in the steam generator tank in order that the pressure inside the tank does not exceed 30 psi for safety purposes.

A level measuring instrument, model CAP ANALOG/4100, from DELAVAN Process Instrumentation, Scottsdale, Arizona, is used in the condensate collecting tank to measure the mass flow rate which is needed in calculating the overall heat transfer coefficient. An Infra-Red (IR) thermometer of model OS52 from OMEGA Engineering, Inc., is used to measure the outer surface temperature of the sphere while it is rotating.

3) Control System

A variable speed motor, model 4Z528 DC motor from DAYTON, with controlling current input, model 5X412C from DAYTON, is used to control the rotational speed of the spherical shell. A strobe light was used to determine the rotational speeds for corresponding knob settings.

4) Steam Generator System

A cylindrical aluminum tank of 6061-T6 type is used to serve as a steam generator. It is filled with a working fluid. It is heated while the steam valve is closed to get a high saturation pressure with corresponding high saturation temperature. When the steam valve, which connects the steam generator to pipes reaching the low pressure environment inside the sphere, is opened, the steam is exposed to lower pressure resulting in high super heated steam. A 550 W heating cartridge, model TB5064TB1 from VULCAN, placed on the bottom base of the steam generator tank is used to heat the working fluid. It is connected to a thermocouple which turns the heating cartridge on/off according to pre set temperature limits.

5) Condensate Collecting System

This system consists of a cylindrical Aluminum, 6061-T6 type, tank with a conical base where the condensate can be collected and then drained. When the spherical shell rotates, the condensate is forced toward the sphere equator where stationary pipes are located with their open ends facing opposite the rotation direction of the sphere. The condensate is collected in these pipes and then transported to the collecting tank by gravitational forces.

Nearly the same methodology as explained previously for the flat plate configuration is to be used in these experiments. One main difference is using equation 4.1, rather than Equations 4.2 and 4.3, to find the overall heat transfer coefficient. This coefficient is to be found between the vapor-condensate interface temperature, which represents the working fluid temperature, and the outer surface temperature of the sphere, which represents the cold environment. The spherical shell could be cooled by putting the whole experiment in a cold room especially when a high temperature difference is desired. If this is the case, the freezing point of the working fluid must be considered in order to keep the condensate from freezing. Since the steam generator of the RBMR can generate high temperature steam, room temperature may be considered for the initial testing.

Several runs were made to assure that the system operated as it was designated. It was noticed, after to a few operational testing runs, that at relatively high speeds, i.e. 500 rpm and above, the temperatures of the outer surface of the sphere shell, as read by the IR thermometer, tends to read same every where, except at the poles, which indicates that at those speeds the effect of gravitational forces is negligible compared to centrifugal forces. Thus, a good simulation of the RBMR, using these high speeds, may be obtained in normal gravity.

Several considerations were taken into account in designing and constructing the RBMR prototype. Among these, the system should be self sustained, it should have a reasonably small size so that it can fit into several micro-gravity carriers, and that it should be easy to modify. Several additional modifications are possible in this design such as a full coolant loop, inserting a fiber optic camera in the spherical shell to watch the condensation process and document the flow patterns, and it is possible, with little effort, to replace the spherical shell with other geometries for the RBMR while all other systems, equipments, and instruments remain the same.

4.13. Conclusions and Recommendations.

The first, second, and fourth objectives of the current study were completely accomplished; these are designing, constructing, operating and demonstrating a heat rejection system using condensation on a rotating flat plate and on a full sphere, and also the development of an empirical correlation to relate the overall heat transfer parameter of the rotating flat plate to the angular rotational speeds. Three rotational speeds were used in these experiments, namely 200, 300, and 400 rpm.

The third objective was partially achieved. The condensate film thickness was very difficult to precisely measure, primary because of the vibration in the entire system, the requirement of precise normality between the flat and the axis of rotation, and also the wide laser beam diameter comparing to the condensate film thickness. Condensate flow patterns for several experiments were recorded at the three rotational speeds by videotape but no more study seemed to be useful in the design of such heat rejection systems where the main concern is usually finding correlations for heat transfer coefficients. However, preliminary results obtained from experiments have proved both techniques, i.e. the concept of using

laser beams in finding the thickness of transparent consistent objects and fluorescent fluid in recording fluid motions, are working well and worthy to be considered in further experiments if these problems can be overcome.

Dimensionless groups developed by Sparrow and Gregg were used in this study by replacing their local heat transfer coefficients with the overall heat transfer coefficients, and their temperature difference across the film condensate with the temperature difference corresponding to the overall heat transfer coefficients used. Two different overall heat transfer coefficients were inspected, namely U_1 and U_2 . Two different kind of fittings were used in analyzing the experimental data, least square and average point line. One can conclude that using the same curve of Sparrow and Gregg for the overall heat transfer coefficient and its corresponding temperature difference will give satisfactory results. This conclusion was proved experimentally for intervals of $C_p \Delta T / h_{fg}$ from about 0.04 to about 0.105.

The heat flux measurements for the three tested rotational speeds verses dimensionless temperature difference were in a good agreement with the theoretical predictions.

Errors of the correlation including h_{avg} were high for low temperature difference, while they were reasonable for higher ones. This was attributed to neglecting the gap effect between the two plates where its effect could be minimized by higher temperature differences.

Possible sources of errors in these experiments were mentioned and discussed in detail. It was found that the effects of these sources of errors on experiment results were minimal. The scattered nature of the data points should be attributed to these sources of errors. Statistical analysis was performed on all studied parameters to show the probable amount of fluctuation in the experimental data.

Several significant conclusions were made from these current investigations, among which :

- * The empirical correlation found in this study, which was found to be similar to the one developed by Sparrow and Gregg, is useful for choosing the optimum rotational speed for the flat plate radiator given a desired heat rejection load. Since the working fluid properties, system temperatures, and the temperature of the environment are usually known, this correlation will give the relationship between the rotational speed and the overall heat transfer coefficient. Then the designer can choose the optimum rotational speed to reject a certain amount of heat from the system.
- * Since the work that the current study is compared to (the analytical work of Sparrow and Gregg) is insensitive to working fluid Prandtl numbers for $C_p \Delta T / h_{fg}$ less than or equal 0.1, the results of this study are not necessarily restricted to materials with Prandtl numbers similar to that of water.

- * This study also provides the basis for designing new heat rejection systems utilizing centrifugal forces and condensation phenomena in both space and ground applications.
- * The first RBMR ever built was designed, constructed, operated, and is ready for future investigations.

Using the same available apparatus and instruments, and pretty much similar procedures and methodology of these experiments, more studies can be performed and investigated. The present ground based investigation is considered as a first step in a series of experiments using different working fluids and surface configurations, such as the available RBMR prototype, and concludes with experimental tests in microgravity. Testing different working fluids with various Prandtl numbers will help in generalizing the results and hence broadening the applications of the correlations to other fluids. Investigating different geometries is a very important step in choosing the optimum configuration of the RBMR which allows the best heat transport and is suitable for other design parameters.

Further study is also needed to extend the testing interval beyond $C_p \Delta T / h_{fg}$ equal to 0.105. According to the analytical work of Sparrow and Gregg, the straight correlation line of 0.904 for the flat plate condensation experiments starts to curve below and above its original value depending on the Prandtl numbers. Increasing this interval experimentally could be achieved by increasing the temperature difference between the cooled surfaces and the operating temperatures. The existing apparatus of the steam generator system of the RBMR prototype is suitable for this purpose because it was made of aluminum which can tolerate reasonably high pressures, and hence high temperatures. The glass beaker used in the flat plate experiments must be replaced by another high pressure chamber to be suitable in generating high temperatures.

Using higher rotational speeds is highly recommended for future work. This will help in applying more force on condensate droplets to join each other and hence to get film-wise condensation, and also in assuring the conclusion of generalizing the correlation to other rotational speeds. These same experiments could be performed along with the flow pattern study, with no change in major procedures, to investigate and document the change of flow patterns with the rotational speeds.

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5. Ceramic Fabric Wicking Rate Tests.

5.1. Introduction.

Several ceramic fabrics have previously been tested for wicking velocity in a dry fabric at Oregon State University [Marks 1992 and Marks and Klein 1992]. Follow up studies were conducted to get additional information on how well these materials will transport working fluids in heat pipes. These new tests are being run to determine the fabrics wicking capacity, or the rate at which the fabric will wick a working fluid from one reservoir to another. These tests, also known as drop tests, are being used to determine the wicking capacity of these fabrics. The amount of liquid that a particular fabric is able to return to the evaporator end of a heat pipe is important in determining the maximum operating conditions [Dunn and Reay 1982].

5.2. Experimental Setup and Procedures.

The original attempts at collecting this data provided significant insight into the development of the final test set-up used to collect the data that is presented. The first experiment involved hanging a piece of s-glass 5 cm wide and 0.09 cm thick, out of a container filled with water. The fabric was allowed to hang down approximately 20 cm. Below the end of the fabric was a dish for collecting any liquid that dripped off the end of the fabric. This fabric was allowed to sit for 12 hours.

At the end of the test only the top half of the fabric was moist and the rest was dry. This test set up was dropped and a new test was developed in which the fabric was draped from one container to another and supported by a metal bridge. The bridge was intended to aid in transporting the liquid and simulate actual working conditions. The original bridge was constructed of aluminum and created a drop of 15 cm. To start these experiments the fabric was soaked in a water bath, then draped over the bridge from one pan to the other. These tests were allowed to run for several hours, at the end of which the transferred water was collected and measured. The results of these tests were disappointing. The small amounts of liquid that were actually transferred were inconsistent and probably distorted by the amount of liquid that evaporated during the tests.

To obtain values that were easier to measure, two changes were made. First, thicker fabric samples were used. This increased the cross sectional area that the liquid had to flow through. Second the material of the bridge was changed from aluminum to copper. This was done because previous work had reported that a short aluminum bridge had to be scored to increase the wettability of the metal [Marks 1992].

With these changes the amount of liquid transferred became much easier to work with. Tests were run in which the individual fabrics were allowed to transport liquid from one container to another for 30 minutes to an hour. These tests were run for a couple weeks until it became apparent that the rate at which the liquid was being transported changed from the start of the test to the end of the test. This change was caused by the lowering of the

water level in the upper dish. When this liquid level dropped it forced the fabric to wick the water up a longer grade before dropping down the bridge and into the collection dish.

In an effort to determine the change in rate, a better test format was developed. The first step was to build larger trays for the top liquid container. This made the liquid level in the top tray more constant over the duration of the test. The other improvement involved acquiring a low intensity vacuum extractor (turkey baster). This allowed the transported liquid to be removed from the collection dish, measured, and returned to the top tray, allowing continuous monitoring of the flow rate. All results presented were gathered in this way. Figure 5.1 shows a schematic diagram of the final, as tested arrangement.

Fabric strips were prepared especially for the final test procedure. The ceramic materials were heat treated to remove any sizing. These heat treatments varied for the different materials tested. The s-glass and silicon carbide were heated to 400° C for 8 hours. The Nextel fabrics 312 and 440 were baked at 550° C for 12 hours. Kevlar was not heat treated because of its low melting point. Instead the material was soaked in deionized water and detergent for 2 weeks. The Kevlar was treated in this way so that it would be consistent with the methods used in the dry uptake rate experiments. After the materials were treated they were cut into strips 6.5 cm wide. The outer 0.25 cm of fabric was impregnated with a silicon gel used for sealing aquariums. This gel was used to make sure that the fabric did not unravel.

To run a test the upper container was partially filled with deionized water. The fabric that was to be tested was moistened and laid out across the bridge connecting the two trays. The upper container was then filled to the top and the material was given 10 to 15 minutes to establish a consistent and complete flow pattern. When time was not given to the material to settle down into a consistent flow pattern the first two or three data points did not match those of the rest of the run. At the end of this set-up time the liquid in the lower reservoir was removed and returned to the upper reservoir and the test began. When approximately 100 to 150 ml of water had been transferred from the top to the bottom reservoir the water was removed from the lower reservoir, measured and returned to the upper reservoir. These tests lasted anywhere from 30 minutes to 4 hours.

The fabrics chosen were all plain weaves of approximately the same thickness. However the weave densities of the individual weave patterns differs more than would have been preferred. The differences come from the fact that the fabrics were not intended to be used as wicks and their weave densities reflect their established uses. For the fabrics used Kevlar was the most dense and the silicon dioxide was the least dense. The Nextel fabrics and the silicon carbide had similar weave densities.

5.3. Results.

Each fabric was tested in four different set-ups. They were each tested as a single layer of fabric and as two layers together. They were also tested over a drop of 14 cm and a drop of 7 cm. For each one of these four situations the set-up was run four times. The Nextel

Figure 5.1. Schematic Diagram of the Wicking Rate Test Equipment.

440 was also tested on a tall aluminum bridge for comparison with the copper bridge data. The flow rates within each individual test were very consistent, however the flow rates between separate tests with the same set-up were different by as much as a factor of 5.

The average of each individual run along with the average of all the runs put together is shown in Tables 5.1 through 5.6. This is done so that the large difference between individual tests of the same set-up can be seen clearly. The data has been normalized by the cross sectional area of the fabric to put the fabrics on more even terms. It is believed that some of this inconsistency is due to the way in which the data was acquired, however the nature of the physical phenomenon investigated does not lend itself to a consistent average. Figure 5.2 shows a bar graph of the normalized flow rates for the five fabrics tested in the four different set-ups.

5.4. Discussion.

There are several possible explanations for the differences between the separate test runs with the same set-up. One important explanation is that the test apparatus had to be reset before each run. This leads to small inconsistencies in the angle of descent, height of the test, contact between the fabric and the metal bridge, and the height of the water in the upper reservoir. A possible contributing factor to the discrepancies may have been that water traveled between the fabric and the metal, or between the fabric layers. Because of the way the fabric is made it is possible for the flow pattern from the top to the bottom of the bridge to be quite different in each case. If, because of the set-up, several fairly straight and clear arteries formed from the top to the bottom, then the flow rate would be high, if these arteries were short and narrow then the flow rate would be lower.

The addition of a second layer does not significantly change the normalized flow rate in s-glass, Nextel 312, and Kevlar in both the tall and the short set-up. It is also true of the Nextel 440 for the tall case on aluminum and the short case on copper. If it is assumed that the amount of liquid transported within the mesh is minimal compared to the flow between the fabric and the bridge, then it is possible to say that for these four fabrics the flow between fabric layers is on the order of that between the fabric and the bridge. The data for Nextel 440 on the tall set-up and silicon carbide in both set-ups, shows an increase in flow rate by a factor of about 2. For this to be true the flow rate between fabric layers would have to be approximately 3 times the flow between the fabric and the bridge.

The data for every material shows a decrease in flow rate for the single short test to the single tall test. This implies that the added pressure head gained by the larger drop is not greater than the added resistance of the increased flow length. This is also true when two layers are used, for s-glass, Nextel 312, and Kevlar. The data shows a different behavior for silicon carbide and Nextel 440. Both of these fabrics exhibit increased flow rates when they were changed from double short to double tall. This implies that the added pressure head gained with the increased height is larger than the increased resistance gained with the extended flow length. One reason for the difference may be that both Nextel 440 and silicon carbide are much stiffer than the s-glass and Kevlar. This allows them to hold their shape

Fabric Wicking Rates

Short Test 7 cm. Tall Test 14 cm.

Figure 5.2. Normalized Flow Rates for the Five Fabrics Tested in the Four Different Set-ups.

better so that they do not collapse down and block the flow paths. This was evident at the end of a test when the fabric was removed from the bridge. Both Kevlar and s-glass seemed to have a small hold on the bridge and this did not appear to be the case for the Nextel 440 or the silicon carbide.

Nextel 440 was tested on both a copper and an aluminum bridge. This data showed that the flow rates on copper were larger than the flow rates on aluminum. This was the expected result and verifies assumptions made in the test design.

If the experiment were to be run again, a different procedure might lead to more consistent results. Two improvements are strongly suggested. First, the top tray, bridge, and collection tray should be permanently fastened together or constructed in such a way that they are always in the same position relative to each other when put together. The second improvement would be to set the top tray up with a continuous feeder, so that the water level stays constant throughout the test and between separate tests. In this system a container of water is filled and placed upside down in the top tray. Holes in the water container would be positioned so that they allowed air to enter and water to come out when the level dipped below these holes. These two improvements would eliminate much of the inconsistency generated by the procedure used in this investigation.

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T.S. Marks, T.S. and A.C. Klein (1992) "An Experimental Approach to Compare Wicking Abilities of Fabric Materials for Heat Pipe Applications," in *Proc. Ninth Symposium on Space Nuclear Power Systems*, CONF-920104, M.S. El-Genk and M.D. Hoover, eds., American Institute of Physics, New York, 1992, pp. 1147-1152.

Table 5.1. Wicking flow rates for s-glass at room temperature and pressure. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.014 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 7 cm.	1190	5393	2024	3560	3042
Single 14 cm.	1536	3738	3083	2512	2717
Double 7 cm.	2173	48810	3857	3131	3510
Double 14 cm.	786	1446	952	6018	2301

Table 5.2. Wicking flow rates for Kevlar at room temperature and pressure. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.0254 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 7 cm.	4403	1942	2106	12093	5136
Single 14 cm.	3150	1818	1293	5945	3051
Double 7 cm.	3087	5174	4419	7710	5098
Double 14 cm.	5984	2854	2218	3465	3630

Table 5.3. Wicking flow rates for silicon carbide at room temperature and pressure. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.028 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 7 cm.	5179	4214	4869	4881	4786
Single 14 cm.	3929	4232	4881	4833	4469
Double 7 cm.	10030	6661	5750	7705	7536
Double 14 cm.	4649	6414	21991	5327	9595

Table 5.4. Wicking flow rates for Nextel 312 at room temperature and pressure. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.036 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 7 cm.	1868	1384	1574	1892	1680
Single 14 cm.	951	1330	1005	2032	1330
Double 7 cm.	2847	2095	2523	2074	2384
Double 14 cm.	2523	2431	1927	2760	2411

Table 5.5. Wicking flow rates for Nextel 440 at room temperature and pressure. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.043 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 7 cm.	4256	1781	2035	1876	2487
Single 14 cm.	1953	1942	2695	2531	2280
Double 7 cm.	2432	2374	2355	2742	2476
Double 14 cm.	4116	4326	3760	3992	4048

Table 5.6. Wicking flow rates for Nextel 440 at room temperature and pressure on Aluminum. Flow rates normalized by the fabric cross sectional area, mL/cm²-hr. Each layer is 0.043 cm thick.

Layers	Trial 1	Trial 2	Trial 3	Trial 4	Average
Single 14 cm.	2502	1954	1591	1906	1988
Double 14 cm.	3845	2302	2202	1688	2510

6. Materials Compatibility Literature Survey.

6.1. Introduction.

This section reviews information pertaining to the compatibility of several space radiator materials and working fluids. The materials and working fluids were chosen in a brief review of current radiator design concepts. In general specific alloys of chosen materials were not researched. Cost, ease of manufacturing, and mechanical properties were not considered for simplicity and on the assumption that these were considered by the original designers. This section consists of discussions about the individual working fluids and their compatibility with the chosen materials, followed by tables listing attack rates of working fluids on individual materials. There are also a few sections discussing some general considerations with respect to the compatibility of different systems. For additional information on the compatibility of materials or working fluids not discussed in this report, the following sources are a good starting point for any investigation:

B.D. Craig, *Handbook of Corrosion Data*, ASM International, Metals Park, OH, 1989.

D.J. DeRenzo and I. Mellan, *Corrosion Resistant Materials Handbook 3rd Ed.*, Noyes Data Corporation, Park Ridge, NJ, 1985.

J.R. DiStefano, "Review of Alkali Metal and Refractory Alloy Compatibility for Rankine Cycle Applications" in *Space Nuclear Power Systems, Vol. 9*, M.S. El-Genk and M.D. Hoover, Eds., Orbit Book Company, Inc., Malabar, FL, 1989.

P.A. Schweitzer, *Corrosion Resistance Tables, 2nd Ed.*, Marcel Dekker, Inc., New York, NY, 1986.

The bibliography listed at the end of this section includes additional sources of information found during this literature survey.

6.2. Working Fluids.

Acetone is a common working fluid considered for use in low temperature radiator systems. Acetone has a boiling point at atmospheric pressure of 56°C. Of the materials researched aluminum, copper, teflon, and titanium are all very resistant to corrosion by acetone. The resistance of both aluminum and teflon extend over a wide temperature range. No information could be obtained for molybdenum, niobium, and tantalum. Table 6.1 shows this information along with temperatures, attack rates, and references.

Ammonia is another common low temperature working fluid, with a boiling point of -33.4°C. Aluminum, teflon, and titanium all reported resistance to ammonia. Copper and its alloys are resistant to ammonia, but very susceptible to attack by hydrogenous ammonia. Molybdenum was the only other material on which information about corrosion resistance could be found. Molybdenum is resistant to aqueous ammonia, no temperature range was

given for this resistance. Table 6.2 shows this information along with temperatures, attack rates, and references.

Dowtherm is a name given to a group of organic fluids. Aluminum, copper and titanium all showed poor resistance to Dowtherm, in most cases when individual Dowtherms were discussed (eg. Dowtherm-E) the attack rates were larger. Teflon was the only material only material of interest on which information showing a good resistance to Dowtherm attack could be found. Table 6.3 has this information tabulated.

Freon is the generic name given to a group of halogenated hydrocarbons. In the presents of moisture, freon breaks down and can form hydrofluoric and hydrochloric acids. In aluminum these acids are able to break down aluminum's protective oxide film, causing increased corrosion. Because of this aluminum is only useful for dry freons. Copper had mixed reviews, one source gave an attack rate of 2 to 20 mils/yr, while another source stated that copper was resistant to all freons. Several sources reported good resistance of teflon to freon. The information on freon is tabulated in Tables 6.4 and 6.5.

Methanol or methyl alcohol was another low temperature working fluid investigated by this paper. Methanol has a boiling point of 64.6°C. Of the materials of interest only Teflon has a good resistance to methanol. Copper is said to be suitable for use with methanol. Contradicting information on both titanium's and aluminum's resistance to methanol was found. Titanium and aluminum both seem to be more resistant to commercially pure methanol, than to methanol of lower concentrations. No information was available on the resistance of molybdenum, niobium or tantalum. The information on freon resistance is tabulated as Table 6.6.

Water is the last low temperature working fluid investigated by this paper. Titanium, copper and teflon are all suitable for use with water. Tantalum may also be used with water if the pH is controlled. Niobium is said to have a fair resistance to steam at 750°F. Aluminum is resistant to high purity water. No information on molybdenum was available. This information is summarized in Table 6.7.

Mercury is an intermediate temperature working fluid. Its melting point is -38.8°C and its boiling point is 356.6°C. Both aluminum and copper are incompatible with mercury. Molybdenum, niobium, tantalum, and titanium are all compatible. Teflon is compatible with mercury up to the boiling point of the metal. No information was found pertaining to teflon's compatibility above this point. Information about compatibility with mercury as a working fluid is shown in Table 6.8.

Lithium is an alkali liquid metal. It has a melting point of 180.5°C and a boiling point of 1342°C. Molybdenum, niobium, tantalum, and titanium all showed high resistance to lithium as a liquid metal. Copper is incompatible with lithium. The temperature of application for lithium and the other liquid metal working fluids studied is too high for Teflon to be considered as a containment material. The information gathered on lithium is shown as Table 6.9.

Potassium was one of the liquid metals considered as a possible working fluid. Its melting point is 63.3°C and its boiling point is 760°C. Molybdenum, niobium, and tantalum, are all very resistant to potassium. Copper is incompatible with potassium but, one source listed Cu-2 Be and Aluminum Bronze as having some resistance up to temperatures of 600°F. No information on the compatibility of titanium and potassium was found. However titanium is resistant to attack by NaK, a alloy of sodium and potassium, containing 77.2% potassium. It is therefore possible that titanium is also resistant to potassium along. The information on interactions between potassium and the materials of interest is shown in Table 6.10.

Sodium is a common liquid metal discussed as a possible working fluid for space radiators. Sodium's melting point is 97.8°C and its boiling point is 882.9°C. Molybdenum, niobium, tantalum, and titanium are resistant to attack by sodium. The highest temperature given for titanium resistance was 593°C. The same copper alloys mentioned in the discussion about potassium, have limited resistance to sodium. There was no information available on the resistance of aluminum to sodium. Table 6.11 shows the information that was gathered on sodium compatibility.

NaK is the name given to the sodium-potassium alloy. This combination is considered useful because the eutectic (77.2 % K) has a melting point of -12.3°C. Information for the resistance of niobium, tantalum and titanium all showed high resistance to NaK fluids. No information could be found to illustrate the resistance of molybdenum. However molybdenum's resistance to both sodium and potassium and the fact that none of the other metals that were resistant to sodium and potassium were attacked by NaK, leads to the assumption that it is also resistant to NaK. Again no information on aluminum was available. The available information is assembled in Table 6.12.

The information is also re-tabulated as a function of the potential working fluids in Tables 6.13 through 6.19 for aluminum, copper, molybdenum, niobium, tantalum, teflon, and titanium respectively.

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Table 6.1. Material Compatibility Data for Acetone.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum		↑ 500°F	↑ 2 mils/yr	Schweitzer
Copper		↑ 150°F	↑ 2 mils/yr	Schweitzer
Molybdenum				
Niobium				
Tantalum				
Teflon	FEP TFE	↑ 420°F ↑ 470°F	Resistant Resistant	Schweitzer Schweitzer
Titanium		↑ 200°F ↑ 150°F	↑ 2 mils/yr ↑ 5 mils/yr	Schweitzer DeRenzo

↑ Up to

Table 6.2. Material Compatibility Data for Ammonia.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum		↑ 180°F	↑ 2 mils/yr	Schweitzer
Copper	Gas	↑ 80°F Any	↑ 2 mils/yr 20 → 50 mils/yr	Schweitzer Schweitzer
Molybdenum	Aqueous	Not Given	No Attack	Mellan
Niobium				
Tantalum				
Teflon	FEP TFE	↑ 420°F ↑ 470°F	Resistant Resistant	Schweitzer Schweitzer
Titanium		↑ 280°F 104°F	↑ 2 mils/yr ↑ 1 mil/yr	Schweitzer Mellan

↑ Up to

Table 6.3. Material Compatibility Data for Dowtherm.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum		↑ 160°F Not Given	2 → 20 mils/yr ↑ 1 mil/yr	Schweitzer Mellan
Copper		↑ 160°F	2 → 20 mils/yr	Schweitzer
Molybdenum				
Niobium				
Tantalum				
Teflon	FEP TFE	↑ 420°F ↑ 470°F	Resistant Resistant	Schweitzer Schweitzer
Titanium	.	↑ 200°F	2 → 20 mils/yr	Schweitzer

↑ Up to

Table 6.4. Material Compatibility Data for Freon-11.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum	Dry Gas	↑80°F Not Given	2 → 20 mils/yr ↑ 1 mil/yr	Schweitzer Mellan
Copper	'Freons'	↑80°F Not Given	2 → 20 mils/yr Excellent	Schweitzer Mellan
Molybdenum				
Niobium				
Tantalum				
Teflon	FEP TFE	↑420°F ↑470°F Not Given	Resistant Resistant Acceptable	Schweitzer Schweitzer DeRenzo
Titanium				

↑ Up to

Table 6.5. Material Compatibility Data for Freon-21.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum	Dry Gas	Not Given	↑ 1 mil/yr	Mellan
Copper	'Freons'	Not Given	Excellent	Mellan
Molybdenum				
Niobium				
Tantalum				
Teflon		Not Given	Acceptable	DeRenzo
Titanium				

↑ Up to

Table 6.6. Material Compatibility Data for Methanol.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum		↑ 150°F Not Given	2 → 20 mils/yr 2 → 20 mils/yr	Schweitzer Mellan
Copper	Many Alloys	Not Given	Suitable	Mellan
Molybdenum				
Niobium				
Tantalum				
Teflon	FEP TFE	↑ 420°F ↑ 470°F	Resistant Resistant	Schweitzer Schweitzer
Titanium	95%	100°C ↑ 180°F Not Given	↑ 0.4 mils/yr 2 → 20 mils/yr Excellent	Craig Schweitzer Mellan

↑ Up to

Table 6.7. Material Compatibility Data for Water.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum	Highly Purified	Above B. P.	Resistant	Mellan
Copper	Many Alloys	Not Given	Suitable	Mellan
Molybdenum				
Niobium	Steam	750°F	Fair Resistance	DeRenzo
Tantalum	pH 8 → 9	↑ Boiling Point Boiling Point	No Corrosion Good Use	Craig Craig
Teflon	FEP TFE	↑ 420°F ↑ 470°F	Resistant Resistant	Schweitzer Schweitzer
Titanium		↑ 315°F ↑ 316°C	Slight Weight Gain Nil	Craig DeRenzo

↑ Up to

Table 6.8. Material Compatibility Data for Mercury.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum		↑ 560°F	20 → 50 mils/yr	Schweitzer
Copper		↑ 560°F	Incompatible 20 → 50 mils/yr	Craig Schweitzer
Molybdenum		Molten	Resistant	Mellan
Niobium		↑ 600°C	Resistant	Craig
Tantalum	100 hours	↑ 600°C ↑ 600°C 1088°K	No Attack Resistant No Attack	Craig Mellan DiStefano
Teflon		↑ Boiling Point	Resistant	Mellan
Titanium		150°C 150 → 300°C ↑ 560°C	↑ 0.1 mm /yr 0.1 → 1 mm/yr ↑ 2 mils/yr	Craig Craig Schweitzer

↑ Up to

Table 6.9. Material Compatibility Data for Lithium.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum				
Copper			Incompatible	Craig
Molybdenum	100 hours	1088°K Molten	No Attack Resistant	DiStefano Mellan
Niobium	100 hours 100 hours	↑ 1000°C 1088°K 1223°K	Resistant No Attack No Attack	Craig DiStefano DiStefano
Tantalum		↑ 1000°C ↑ 1000°C	No Attack Resistant	Craig Mellan
Teflon				
Titanium		600 → 900°F 316 → 482°C 850°C	Nil No Attack 0.1 → 1 mm/yr	DeRenzo Craig Craig

↑ Up to

Table 6.10. Material Compatibility Data for Potassium.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum				
Copper	Some Alloys	↑ 600°F	Incompatible Limited Resistance	Craig Leidheiser
Molybdenum	Mo-.05 % Ti 500 hours	1003 → 1253°K Molten	No Attack Resistant	DiStefano Mellan
Niobium	300 hours 100 hours	↑ 1000°C 1363°K 1423°K	Resistant No Attack No Attack	Craig DiStefano DiStefano
Tantalum		↑ 900°C ↑ 1000°C	No Attack Resistant	Craig Mellan
Teflon				
Titanium				

↑ Up to

Table 6.11. Material Compatibility Data for Sodium.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum				
Copper	Some Alloys	↑ 600°F	Limited Resistance	Leidheiser
Molybdenum	100 hours	1773°K Molten	0.026 mm Resistant	DiStefano Mellan
Niobium		↑ 1000°C	Resistant	Craig
Tantalum		↑ 900°C ↑ 1200°C	No Attack Resistant	Craig Mellan
Teflon				
Titanium		593°C	Resistant	DeRenzo

↑ Up to

Table 6.12. Material Compatibility Data for NaK.

Material	Conditions	Temperature	Attack Rate	Reference
Aluminum				
Copper	Some Alloys	↑ 600°F	Incompatible Limited Resistance	Craig Leidheiser
Molybdenum				
Niobium		↑ 1000°C	Resistant	Craig
Tantalum		↑ 900°C	No Attack	Craig
Teflon				
Titanium		600°C 800°C	↑ 0.1 mm/yr 0.1 → 1 mm/yr	Craig Craig

↑ Up to

Table 6.13. Material Compatibility Data for Aluminum.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone		↑ 500°F	↑ 2 mils/yr	Schweitzer
Ammonia		↑ 180°F	↑ 2 mils/yr	Schweitzer
Dowtherm		↑ 160°F Not Given	2 → 20 mils/yr ↑ 1 mil/yr	Schweitzer Mellan
Freon-11	Dry Gas	↑ 80°F Not Given	2 → 20 mils/yr ↑ 1 mil/yr	Schweitzer Mellan
Freon-21	Dry Gas	Not Given	↑ 1 mil/yr	Mellan
Methanol		↑ 150°F Not Given	2 → 20 mils/yr 2 → 20 mils/yr	Schweitzer Mellan
Water				
Potassium				
Sodium				
Nak				
Lithium				
Mercury		↑ 560°F	20 → 50 mils/yr	Schweitzer

↑ Up to

Table 6.14. Material Compatibility Data for Copper.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone		↑ 150°F	↑ 2 mils/yr	Schweitzer
Ammonia	Gas	↑ 80°F any	↑ 2 mils/yr 20 → 50 mils/yr	Schweitzer Schweitzer
Dowtherm		↑ 160°F	2 → 20 mils/yr	Schweitzer
Freon-11		↑ 80°F	2 → 20 mils/yr	Schweitzer
Freon-21	'Freons'	Not Given	Excellent	Mellan
Methanol	Many Alloys	Not Given	Suitable	Mellan
Water	Many Alloys	Not Given	Suitable	Mellan
Potassium	Some Alloys	↑ 600°F	Incompatible Limited Resistance	Craig Leidheiser
Sodium	Some Alloys	↑ 600°F	Limited Resistance	Leidheiser
Nak	Some Alloys	↑ 600°F	Incompatible Limited Resistance	Craig Leidheiser
Lithium			Incompatible	Craig
Mercury		↑ 560°F	Incompatible 20 → 50 mils/yr	Craig Schweitzer

↑ Up to

Table 6.15. Material Compatibility Data for Molybdenum.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone				
Ammonia	aqueous	Not Given	No Attack	Mellan
Dowtherm				
Freon-11				
Freon-21				
Methanol				
Water				
Potassium	Mo-.05% Ti 500 hrs	1003 → 1253°K Molten	No Attack Resistant	DiStefano Mellan
Sodium	100 hrs	1773°K Molten	0.026 mm Resistant	DiStefano Mellan
Nak				
Lithium	100 hrs	1088°K Molten	No Attack Resistant	DiStefano Mellan
Mercury		Molten	Resistant	Mellan
Cesium	100 hrs	1273°K Molten	No Attack Resistant	DiStefano Mellan

↑ Up to

Table 6.16. Material Compatibility Data for Niobium.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone				
Ammonia				
Dowtherm				
Freon-11				
Freon-21				
Methanol				
Water				
Potassium	300 hrs 100 hrs	↑ 1000°C 1363°K 1423°K	Resistant No Attack No Attack	Craig DiStefano DiStefano
Sodium		↑ 1000°C	Resistant	Craig
Nak		↑ 1000°C	Resistant	Craig
Lithium	100 hrs 100 hrs	↑ 1000°C 1088°K 1223°K	Resistant No Attack No Attack	Craig DiStefano DiStefano
Mercury		↑ 600°C	Resistant	Craig

↑ Up to

Table 6.17. Material Compatibility Data for Tantalum.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone				
Ammonia				
Dowtherm				
Freon-11				
Freon-21				
Methanol				
Water	Sub-boiling At B.P. pH 8 → 9		No Corrosion Good Use	Craig Craig
Potassium		↑900°C ↑1000°C	No Attack Resistant	Craig Mellan
Sodium		↑900°C ↑1200°C	No Attack Resistant	Craig Mellan
Nak		↑900°C	No Attack	Craig
Lithium		↑1000°C ↑1000°C	No Attack Resistant	Craig Mellan
Mercury	100 hrs	↑600°C ↑600°C 1088°K	No Attack Resistant No Attack	Craig Mellan DiStefano

↑ Up to

Table 6.18. Material Compatibility Data for Teflon.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone	FEP TFE	↑420°F ↑470°F	Resistant Resistant	Schweitzer Schweitzer
Ammonia	FEP TFE	↑420°F ↑470°F	Resistant Resistant	Schweitzer Schweitzer
Dowtherm	FEP TFE	↑420°F ↑470°F	Resistant Resistant	Schweitzer Schweitzer
Freon-11	FEP TFE	↑420°F ↑470°F	Resistant Resistant Acceptable	Schweitzer Schweitzer DeRenzo
Freon-21			Acceptable	DeRenzo
Methanol	FEP TFE	↑420°F ↑470°F	Resistant Resistant	Schweitzer Schweitzer
Water	FEP TFE	↑420°F ↑470°F	Resistant Resistant	Schweitzer Schweitzer
Potassium				
Sodium				
Nak				
Lithium				
Mercury		↑ Boiling Point	Resistant	Mellan

↑ Up to

Table 6.19. Material Compatibility Data for Titanium.

Fluid	Conditions	Temp.	Attack Rate	Reference
Acetone		↑ 200°F	↑ 2 mils/yr	Schweitzer
Ammonia		↑ 280°F 104°F	↑ 2 mils/yr ↑ 1 mil/yr	Schweitzer Mellan
Dowtherm		↑ 200°F	2 → 20 mils/yr	Schweitzer
Freon-11				
Freon-21				
Methanol	95%	100°C ↑ 180°F Not Given	↑ 0.4 mil/yr 2 → 20 mils/yr Excellent	Craig Schweitzer Mellan
Water		↑ 315°F	Slight Weight Gain	Craig
Potassium				
Sodium		593°C	Resistant	DeRenzo
Nak		600°C 800°C	↑ 0.1 mm/yr 0.1 → 1 mm/yr	Craig Craig
Lithium		600 → 900°F 316 → 482°C 850°C	Nil No Attack 0.1 → 1 mm/yr	DeRenzo Craig Craig
Mercury		150°C 150 → 300°C ↑ 560°C	↑ 0.1 mm/yr 0.1 → 1 mm/yr ↑ 2 mils/yr	Craig Craig Schweitzer

↑ Up to

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