Thermal Performance of TAPS Heat Pipes with Non Condensable Gas Blockage

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Abstract

Over 124,300 heat pipes were installed in vertical support members on the Trans-Alaska Pipeline System. During infrared, IR, surveys in the early 1980's it was observed that cold topping was occurring on some of these heat pipes. The cold topping was due to an accumulation of non-condensable gases, a by-product of corrosion or chemical dissociation of anhydrous ammonia, in the problem heat pipes. Alyeska Pipeline Service Company under took an experimental program, beginning in the fall of 2000, to measure the heat transfer performance of NCG blocked heat pipes.

Results of this test program are presented and discussed showing the reduction in heat transfer performance of the heat pipes as a function of hydrogen blockage levels. Infrared images are also presented showing the increase in cold topping as the amount of hydrogen in the heat pipe is increased. These IR images are correlated with the degradation in heat transfer performance. A thermodynamic analysis is also presented showing the relationship between the mass of hydrogen within the heat pipe and apparent blockage as a function of evaporator and condenser temperatures.

Introduction

The 1,300 km Trans-Alaska Pipeline System, TAPS, transverses Alaska from Prudhoe Bay on the North Slope to its southern terminus, the Port of Valdez. Pipeline construction began in 1974 and was completed in 1977 with first oil flow occurring that summer. Along this route, the pipeline crosses permanently and seasonally frozen soils. Because warm oil flows through the pipeline, about 680 km of the pipeline was built above ground to avoid burying a warm pipeline in areas of thaw unstable permafrost. An above ground pipeline support assembly was designed to prevent thaw settlement of this ice-rich permafrost. The above ground, 1.22 m diameter pipeline, is attached to a shoe assembly that rests on crossbeam supported by two vertical support members, VSM's. Each VSM and crossbeam structure is called a bent. Bents are spaced approximately 18 m apart along the elevated portion of the pipeline.

Passive heat transfer devices, referred to as heat pipes, were included in the VSM design in those regions where warm permafrost was observed. The cooling effect of passive heat transfer devices on the soils surrounding the VSM's prevent thaw degradation of the permafrost, as well as increase the adfreeze strength of the

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pile-soil bond. North of the Brooks Mountain Range it was not generally necessary to install heat pipes in the VSM's because the permafrost is colder. However, south of the Brooks Range, where discontinuous permafrost occurs and the permafrost can be closer to its melting temperature, two heat pipes were installed in most VSM's (VSM's with heat pipes installed are referred to as thermal piles or thermal VSM's.). The total number of heat pipes installed is approximately 124,300.

Heat pipes, as used on TAPS, consist of sealed steel pipes ranging in lengths from 8.5 m to 20.1 m. The lower portions of the heat pipes, below the top of the VSM's, have an outside diameter of 51 mm with a wall thickness of 6.3 mm. The upper portions of the heat pipes, above the top of the VSM's, have an outside diameter of 76.2 mm with a 19.0 mm wall thickness. These dimensions result in a uniform inside diameter of 38.1 mm. The heat pipes are charged with anhydrous ammonia, NH_3, as the working fluid. About 2% - 3% of the heat pipe's internal volume contains liquid ammonia with the remaining volume filled with its vapor. An extruded aluminum fin assembly, of either 1.22 m or 1.83 m length, is pressed onto the top end of each heat pipe. Heat pipes 11.3 m in length or less were fitted with the 1.22 m finned section and heat pipes of greater length received a 1.83 m finned section.

Heat pipes work on the principal of continuous evaporation and condensation of a working fluid. During the wintertime, when the air is colder than the ground, heat from the ground causes the liquid in the lower portion of the heat pipe to vaporize. The vapor flows upward from the evaporator section (below ground) to the condenser section (above ground) where the vapor condenses back to the liquid phase because of cooling by ambient air. The condensate flows downward, by gravity on the internal wall surface of the heat pipe, where it absorbs heat from the ground and is re-evaporated to continue the process. The upward vapor flow is the result of a pressure gradient between the evaporator and condenser sections of the heat pipe. The vapor pressure of the ammonia increases in the evaporator section due to the addition of ground heat, and decreases in the condenser section as a result of the heat rejected through the finned section. During the warmer periods of the year, when the temperature of the air is warmer than the saturation temperature of the liquid ammonia, the heat pipe is dormant.

As part of the TAPS's thermal pile maintenance program, surveillance of the heat pipes using Infrared (IR) viewing equipment has been used to determine if heat pipes were continuing to transfer heat from the ground. During periods when the heat pipes are actively transferring heat, the finned condenser sections or "radiators" will appear "hot" or "glow" in the infrared portion of the radiation spectrum. Johnson (1983) reported that during the 1980 surveillance effort, many of the heat pipes appeared to have "cold tops" and that probably caused by non-condensable gasses, NCG. This condition was recognized as a potential problem during design, although the extent and magnitude were not predictable. A typical infrared image of a "cold top" or blocked heat pipe next to a fully functional heat pipe is shown in Figure 1. The NCG blocks the upper portion of the finned section and cause a reduction in performance of the heat pipe. IR surveys conducted in subsequent years showed that the level of blockage and number of heat pipes blocked appeared to increase over time. Following the 1990/1991 survey, Williams (1991) reported that the number of heat pipes with detectable blockage was 46% based on a grading process estimated to have an error of ±10%. At that time about 28% of those heat pipes were reported to have blockage equal to or greater than 20%. A blockage level of 30% or more was set to identify heat pipes that were candidates for repair. In 1991 the total number of heat pipes with over 30% blockage was reported at about 10% of the total population heat pipes.

Alyeska Pipeline Service Company, APSC, has conducted extensive investigations to establish cause(s) for the occurrence of the NCG's and potential solutions to the problem. Edelstein (1989) verified that the non-condensing gas was predominantly hydrogen with lesser amounts of nitrogen, naphthenate grease, water, and detergent also identified. He noted that only those heat pipes that were installed on the pipeline have experienced NCG blockage, whereas NCGs have not been found in stored surplus heat pipes.

In 1983 a "getter" repair method was initiated, which utilized a hydrogen absorbing metal halide, zirconium-dimanganese. The getter material was placed in a cavity in a tapered steel pin. These pins had an opening on their tapered end allowing the hydrogen to be absorbed by the getter material. The larger pins, with a getter capacity of 165 grams, are about 254 mm long and 32 mm in diameter with a tapered end having a nominal 16 mm diameter. These, or similar but smaller, pins were installed through the top sidewall of those heat pipes deemed in need of repair. To install the getter pins, a tapered hole was drilled almost through the thick, upper end of the heat pipe wall extending above the fin assembly. The getter pins were then pressed through the remaining heat pipe wall, using a hydraulic press assembly. The repair required accurate drilling and close tolerance fit-up for a satisfactory friction fit between the heat pipe wall and the getter pin wall. By 1990, 2,777 heat pipes were repaired with "getter" pins of varying metal halide capacity.

The getter pin method was successful in relieving hydrogen blockage in most cases. However, in-service observations of getter pin repaired heat pipes have identified a notable percentage of heat pipe failures, probably due in part to the installation method. Although the getter pins initially appeared to "absorb" hydrogen accumulation as designed, about 15% of the repaired heat pipes experienced leakage of the anhydrous ammonia to the atmosphere shortly after installation and resulting in "dead" units. The getter pin program was discontinued in 1993 with the intention of revisiting the friction fit design, and identifying a more effective repair strategy.

In this context, it was widely held that a general solution to NCG blockage of heat pipes lies in predictive repair by relieving molecular hydrogen from the system, or by simply replacing the entire ammonia volume, and not in preventive measures. Because of the large population of heat pipes and increasing number of heat pipes with blockage, it was important to identify those actually needing repair. In order to find the population of heat pipes in need of repair, it was first necessary to define performance of the heat pipes as a function of blockage level. Thus was born the subject of this paper.

Problem Statement

Heat pipe performance became a concern when IR surveys showed an increasing trend in both the number of NCG affected heat pipes and in the amount of NCG accumulation in the TAPS heat pipes. Further analysis of partially blocked heat pipe performance determined that NCG affected heat pipes still retain some capacity to transfer thermal energy from surrounding soils. A portion of the heat transfer capacity of the blocked heat pipe was thought to still occur in the finned condenser section, with additional heat rejection occurring in the section between the bottom of the finned section and the top of the pad, along the above ground portion of the VSM. Conduction along the aluminum fins, as well as some level of convection and condensation within the blocked section, are postulated as the reasons for continued heat transfer. The safety factor associated with the original design criteria coupled with carefully analyzed IR imagery provided a valid argument that blocked heat pipes were still thermally effective during partially blocked conditions.

In an effort to fully understand the performance of a blocked heat pipe with respect to the minimum requirements established in the original APSC design basis, a series of performance tests were conducted during the winters of 1999-2000 and 2000-2001. These tests on TAPS heat pipes were conducted using a full-scale apparatus. The primary initiative of the testing was to establish overall heat pipe performance with respect to varying levels of hydrogen blockage. In the process of modeling and testing heat pipes with NCG's, a series of observations were made concerning the effectiveness of IR imagery, as well as the general behavior of the ammonia-hydrogen mixture. This test program on TAPS heat pipes provided an essential component for development of an enhanced repair methodology.

Test Program

Four heat pipes were tested during the 2000-2001 testing season, each placed in its own calorimeter bath to simulate ground heat flow and temperatures for the embedded portions of the units. The four identical calorimeter baths, fabricated by Flowline Alaska, were 5.8 m long, 200 mm diameter steel pipes. An oversize end cap was welded on the bottom and a raised face weld neck flange was welded on the top of each calorimeter pipe. Urethane rigid foam plastic insulation of 76 mm thickness and an outer sheet metal shield housed the inner pipe. Each bath pipe was filled with proproylene glycol and water with a freeze point of –32°C after the installation of a submersible circulating pump and heat pipe. Figure 2 shows a detailed section view of the calorimeters and Figure 3 shows the four calorimeters with heat pipes installed.

Dryden Instrumentation of Anchorage, Alaska installed the temperature sensing instrumentation on the four heat pipes consisting of three thermistor strings: String 1--nine thermistors installed along the root of adjacent aluminum fins on the condenser section, String 2-- five thermistors installed along the shielded section below the finned section, and String 3--three thermistors installed along the evaporator section to be submerged in the bath. A 9.5 mm thick soft foam plastic pad was placed over the thermistors so they would sense temperatures approximating condenser and evaporator surface temperatures. Along the root of adjacent fins, the foam pad consisted of a strip cut with a trapezoidal cross-section to fit at the bottom of the V. All other pads were about 76 mm by 76 mm in size. A daub of thermal paste was placed at each thermistor to further reduce contact resistance. The instrumentation was designed and installed so it could be removed and reinstalled if additional tests were required.

A "hot tap" was completed above the finned section on each of the four test heat pipes. A 19 mm Thread-O-Let was welded on the heat pipe and then the hot tapping process was performed through a ball valve attached to the heat pipe. This hot tap was then plumbed to a manifold that allowed for hydrogen injection and pressure measurement of the vapor in the heat pipe.

Thermistors to sense bath temperature were installed at the top, mid-point and bottom of each bath. A submersible circulating pump and 600 and 900-watt immersion heaters were connected to a watt transducer to measure power input to each bath. A temperature controller was used to maintain the temperature of each bath within 0.05 C° of a manually set bath temperature. An anemometer with a 0.9 m/s threshold monitored winds speed adjacent to the finned condenser sections of the heat pipes. Ambient outdoor temperature was measured with a radiation shielded themistor in a louvered enclosure. A data logger recorded the data sensed by the thermistors, wind speed transducer and power transducers.

Heat Pipe Preparation

APSC has a large quantity of "new" heat pipes stored at the Nordale Yard east of Fairbanks, Alaska. This inventory of heat pipes is maintained to serve as replacements where pipeline repairs and/or maintenance are required. Nine 9.5 m long "new" heat pipes were secured for the test program

The liquid levels of the ammonia in each of the nine heat pipes were measured. This was done by elevating the condenser end of the heat pipe to achieve an angle of about 30° with respect to the ground. An ultrasonic detector was then moved along the lower end of the heat pipe to locate the liquid-vapor interface.

The total quantity of ammonia in the heat pipes can be calculated based on the measured liquid level interface, temperature of the heat pipe and the known internal dimensions of the heat pipe. The liquid density, ρ_l, and vapor specific volume, v_v (reciprocal of density), are published for ammonia in the two-phase region as a function of temperature, ASHRAE (1993). The mass of ammonia, m, can be determined with these data in conjunction with the volumes of liquid, V_l, vapor, V_v, and total volume, V, or:

$$m = m_l + m_v = V_l \rho_l + \frac{V_v}{v_v} = V_l \rho_l + \frac{V - V_l}{v_v}$$

Using the above equations in conjunction with the measured ambient temperature and liquid level, and heat pipe dimensional data, the average total mass of ammonia was calculated for the test heat pipes at 0.199 kg. A 1.22 m long aluminum finned section was then hydraulically pressed on the condenser sections of each of the heat pipes.

The four heat pipes were installed in their respective calorimeters baths and data were gathered and IR images taken to confirm all heat pipes were performing similarly with no blockage observed. One heat pipe remained the control throughout the testing program.

Hydrogen Injection

Hydrogen injection was performed through the manifold attached to the hot tap on the top of the heat pipe. A two-stage hydrogen regulator was fitted to a 22.7 kg bottle of compressed hydrogen with the regulator set to a pressure of 550 kPa. A 737 mm long section of 12.7 mm diameter steel pipe (internal volume - 93.4×10^3 mm^3) attached to the manifold was purged with hydrogen and then filled to a pressure of 550 kPa. The fill valve to the hydrogen bottle was closed and the needle valve connecting to the heat pipe was opened until the hydrogen pressure bled down to 410 kPa. The mass of hydrogen injected through this known volume was estimated using the Ideal Gas Law. Typically, about 14×10^{-6} kg of hydrogen was injected into three test heat pipes during each charge cycle. Following injection, data were collected and IR pictures taken over a 1-2 week period before another injection cycle was performed.

Data Recording

Bath temperatures during the 2000–01 test program was maintained at -0.5°C yielding an evaporator surface temperature of about –1°C. Power to each of the baths and wind speed were averaged over a one hour time period and stored in the data logger. Ambient, fin, bath and evaporator temperatures were stored at the end of each hour.

Data were downloaded from the data logger to an Excel spreadsheet. Heat pipe conductance, $C=Q/\Delta T$, was calculated by subtracting the calorimeter wall heat loss from the energy input. Heat loss through the calorimeter wall was estimated based on the thermal conductivities of the steel pipe and urethane foam insulation and temperature difference between the glycol and outdoor ambient air.

Some of the data recorded for the period of November 28 - December 5, 2000 are presented in Figures 4 and 5. Because the heat removal rate by the heat pipes depends on evaporator – ambient air temperature difference and wind speed, care in the interpretation of these data is required. Figure 4 shows the energy removed from the calorimeter bath by the heat pipe and the ambient air temperature variation. During the cold spell that occurred on December 2, the heat pipe energy removal rate increased as expected because of the increased temperature difference between the evaporator and ambient air. Figure 5 shows the energy removed by the heat pipe and average wind speed. During periods of increased wind speeds, the energy removal rate by the heat pipe also increased as expected.

Ammonia and Hydrogen Mixture

The total pressure within the heat pipe is the sum of the partial pressures of ammonia and hydrogen. The partial pressure of the ammonia is dependent on the evaporator temperature, i.e. the liquid pool and liquid film in the lower portion of the heat pipe. Data from ASHRAE (1993) can be used to determine the partial pressure of the ammonia based on temperature of the liquid phase. The partial pressure of the hydrogen can be calculated based on the mass of hydrogen contained in the heat pipe using the equation of state for an ideal gas. However, when the evaporator section is heated and the condenser section is cooled, the resulting temperature difference initiates vaporization and condensation, which causes a pressure difference in the gas mixture (evaporator pressure higher than condenser). The pressure difference results in the flow of the gas mixture upward to the finned condenser. Ammonia in the mixture begins to condense resulting in a mixture enriched in hydrogen. As ammonia continues to vaporize and sweep the hydrogen upward, the mixture in the evaporator section becomes hydrogen poor. The partial pressure of the hydrogen in the evaporator section decreases and increases in the condenser section. As equilibrium is established and as long as the vapor velocity exceed the diffusion velocity of the hydrogen, most of the hydrogen is swept to the condenser section.

The partial pressures of the hydrogen and ammonia in the heat pipe can be estimated. Assuming pure ammonia in the evaporator section, the partial and total pressures are equal and can be determined from ASHRAE (1993) based on the evaporator temperature. In the condenser section, the partial pressure of the ammonia can be estimated based on the condenser wall temperature, ammonia dew-point temperature. The partial pressure of the hydrogen is equal to the total pressure minus the partial pressure of the ammonia in the condenser section. Ammonia condenses to its saturation pressure at the equilibrium wall temperature in the blocked section. Because of conduction along the condenser section, convective mixing, and diffusion of hydrogen and ammonia at the interface, there is no distinct line of separation between the "pure" ammonia and hydrogen-ammonia mixture, but a zone of transition.

The amount of hydrogen blockage can be estimated based on the mass of hydrogen and temperatures of the evaporator and condenser sections. The following equation expresses the percent volume of the condenser section blocked by the mixture of hydrogen and ammonia.

$$\frac{V_H}{V_{COND}} = \frac{m_H R_H T_{COND}}{(P_{TOTAL} - P_{AM})} \frac{1}{V_{COND}} 100\%$$

In the above equation T_{COND} is surface temperature of blocked section of the condenser, P_{TOTAL} is total pressure in heat pipe based on evaporator temperature, P_{AM} is partial (vapor) pressure of ammonia in blocked condenser section based on T_{COND}, V_H is volume of hydrogen-ammonia mixture, V_{COND} is internal volume of condenser section, m_H is mass of hydrogen and R_H is the gas constant for hydrogen.

The above equation neglects hydrostatic and dynamic pressure drops occurring between the evaporator and condenser sections. This formulation was used to develop a normalized blockage chart. Because the performance testing conducted during the winter of 2000-2001 was done at an evaporator temperature of $-1C^{o}$, that condition was chosen as a normalizing value. Normalized blockage as a function of condenser temperature and evaporator temperature are shown in Figure 6. Heat pipe testing at several evaporator temperatures is being conducted to validate these predicted trends.

Figure 6 shows blockage as a function of evaporator temperature (pressure) and condenser temperature. As the condenser temperature is decreased at a fixed evaporator temperature, the partial and vapor pressure of the ammonia decreases resulting in an increase in the partial pressure of the hydrogen in the hydrogen-ammonia mixture and a subsequent decrease in its volume and blockage. Alternatively, as the evaporator temperature is decreased at a fixed condenser temperature, the total pressure is reduced and the volume of hydrogen-ammonia mixture in the condenser section expands resulting in an increase in blockage.

Performance as a Function of Hydrogen Blockage

As a conservative approach, the original APSC design basis for thermal VSM's used zero wind speed to establish thermal VSM performance. Therefore, the recorded data from these tests were grouped as a function of wind speed and amount of hydrogen injected. Zero wind speed data were defined as one-hour averaged anemometer readings less than 0.2 m/s.

Figure 7 shows TAPS heat pipe conductance versus blockage for both 1.22 m and 1.83 m finned sections. The curve for the 1.22 m finned section is based on data collected during the test program. Data used to generate this curve was statistically evaluated to represent the 90% confidence interval of the overall collected data. The curve developed for the 1.83 m finned section was calculated from the results of testing the 1.22 m finned section using geometric and heat transfer scaling.

Figure 8 presents heat pipe conductance versus wind speed from 0 to 1.8 m/s at various levels of NCG blockage. The heat pipe conductance increases with increasing wind speed as expected. The zero blockage conductances are greater than those reported by Haynes and Zarling (1988) for a similar heat pipe with the evaporator tilted 0 to 9 degrees from the horizontal. The higher conductances are likely due to the vertical orientation of the evaporator, larger fin radiation loss due to a lower surrounding temperature, larger evaporator-condenser temperature differences, and a 2.4 m bare pipe within a metal collar between the calorimeter bath and finned section.

Visual evaluation of blockage levels was determined using infrared cameras. Two composite IR photos and measured fin surface temperature variations along the finned condenser sections are shown in Figure 9.

Conclusions

The 2000-2001 test program obtained the data necessary to determine TAPS heat pipe thermal degradation as a function of NCG blockage. During heat pipe-testing, measured quantities of hydrogen were introduced into three of the test heat pipes. The thermal performance at various levels of NCG blockage was determined by measurements of power supplied to the evaporator section and evaporator and ambient air temperatures. These data were used to develop a relationship for heat pipe conductance as a function of observed and measured NCG blockage.

A relationship has also been presented that suggests that the observed NCG blockage level is a function of ambient air and evaporator temperatures. Further testing is being conducted to substantiate this hypothesis.

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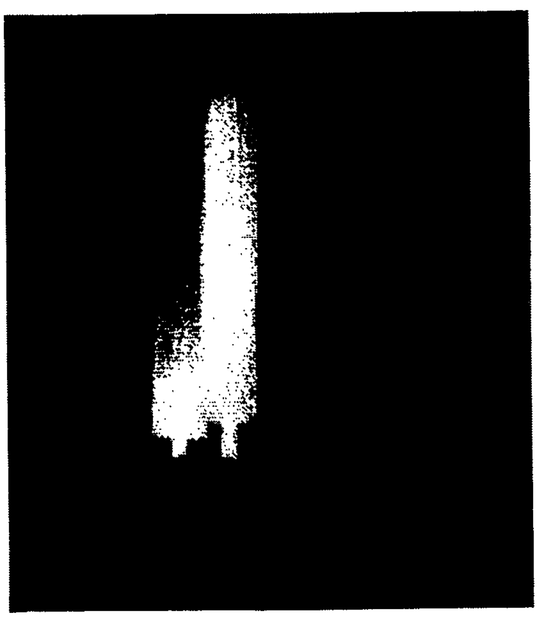


Figure 1. Unblocked and Partially Blocked Heat Pipe

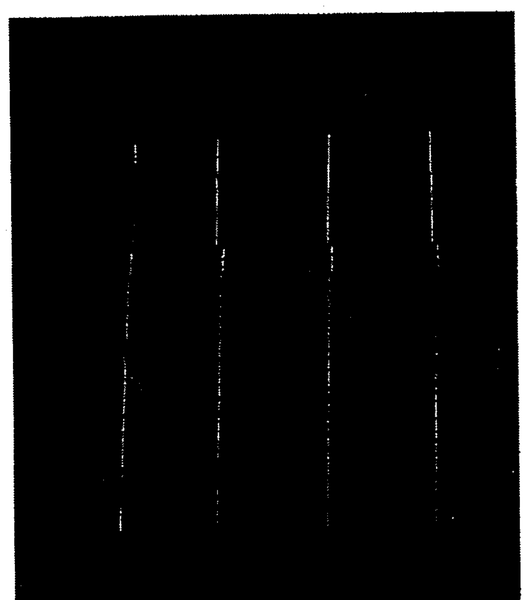


Figure 3. Four Calorimeter Set-up

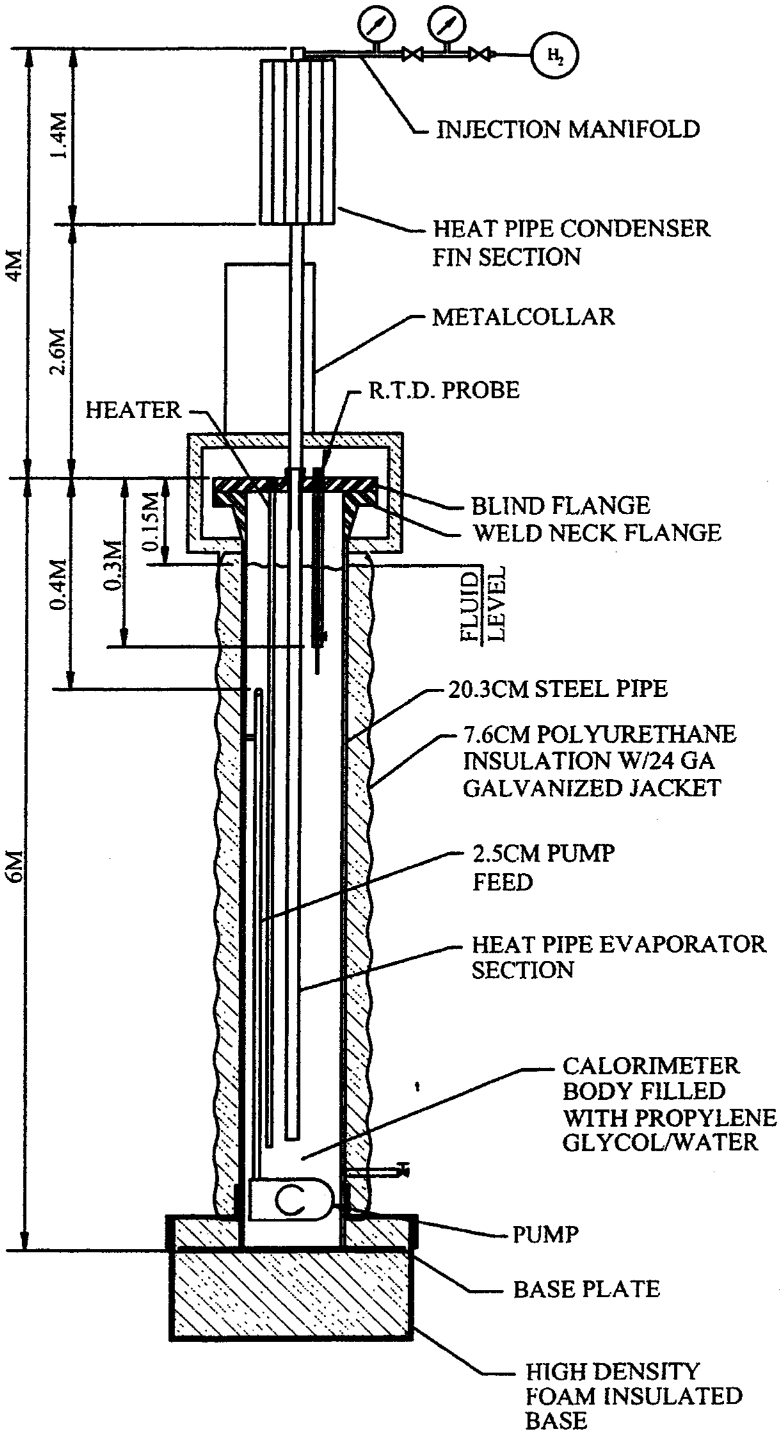


Figure 2. Blockage Test Calorimeter

Figure 4. Heat pipe unit performance and ambient temperature as a function of time.

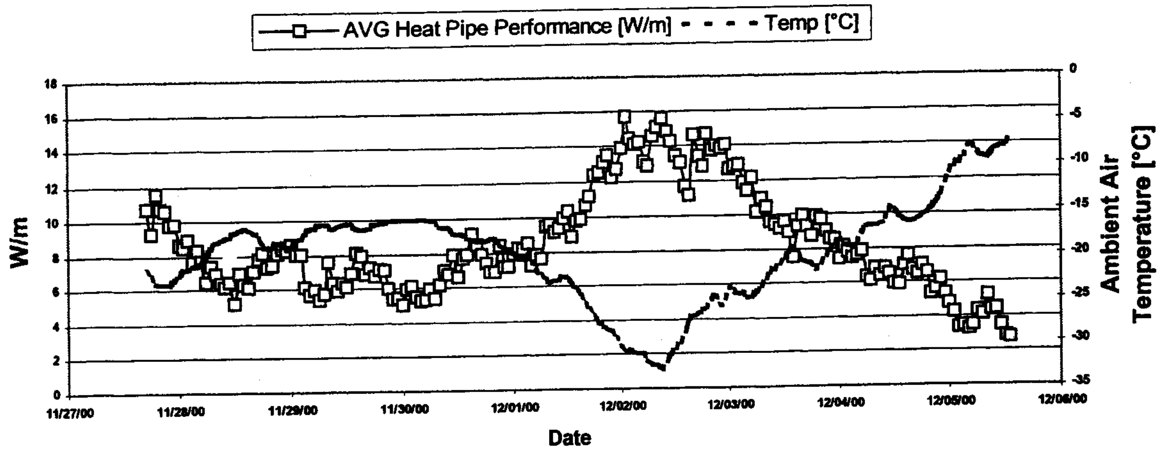


Figure 5. Heat pipe perfromance and wind speed variation

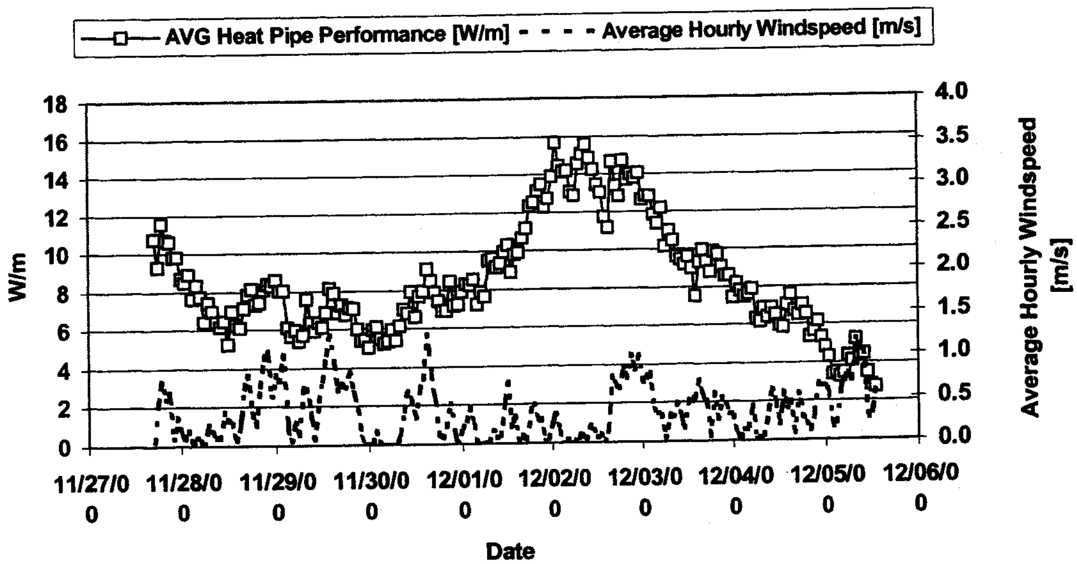


Figure 6. Normalized blockage factors

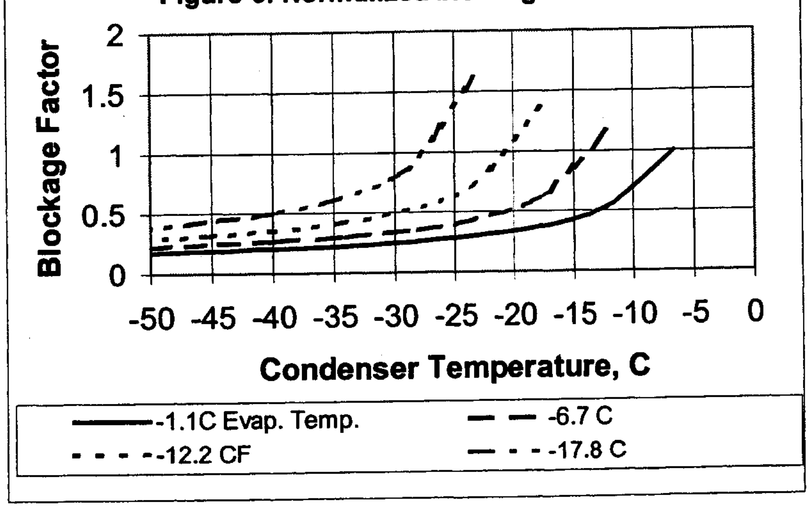


Figure 7. Conductance as a function of blockage at zero wind speed.

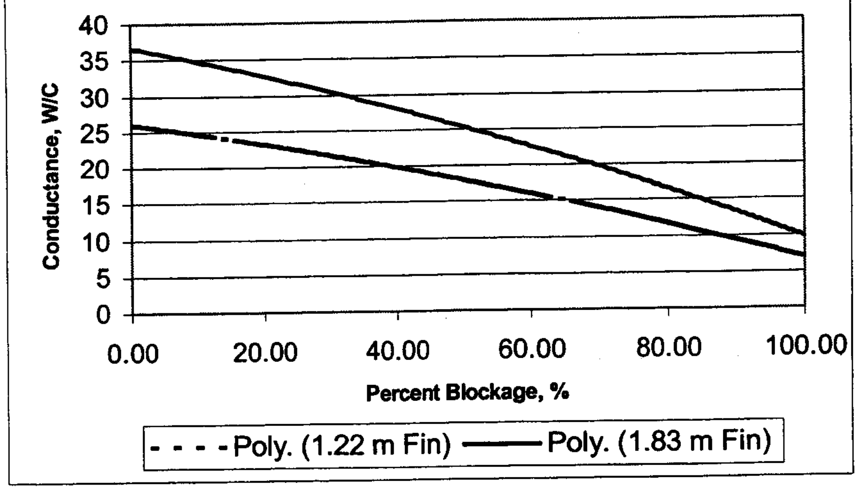
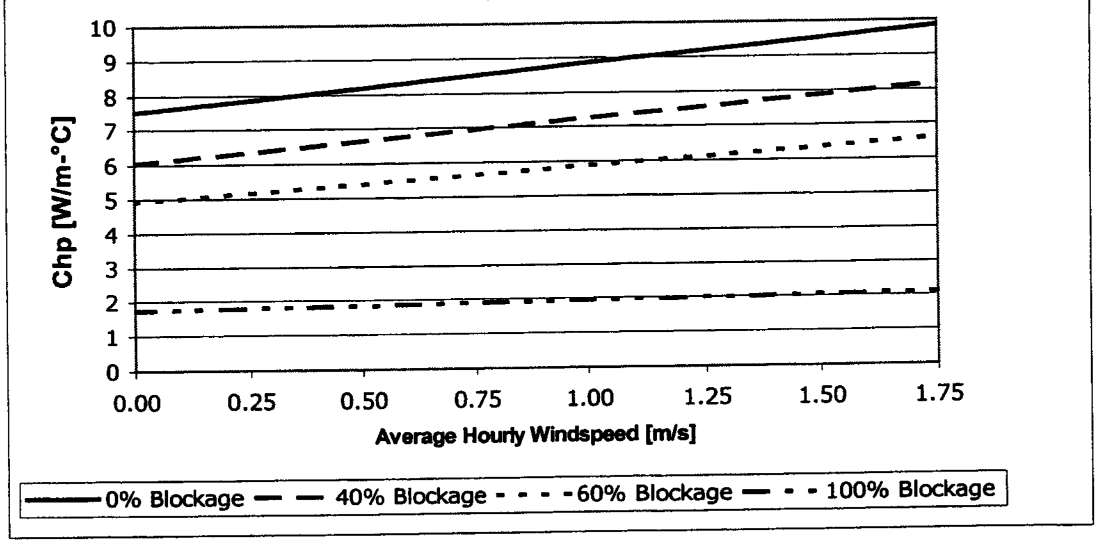


Figure 8. Unit conductance as a function of wind speed and blockage. 1.22 m finned section, 5.18 m Embeddment Length



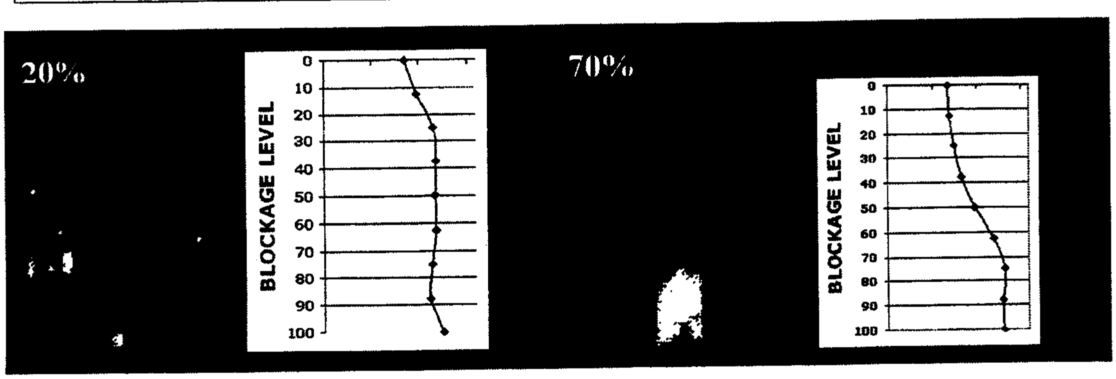


Figure 9. IR and fin temperature profiles for blocked heat pipes