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. Infrared surveys in the early 1980s revealed cold topping 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 2000, Alyeska Pipeline Service Company undertook an experimental program to measure the heat transfer performance of cold-topped heat pipes. The test program results are presented and discussed showing the reduction in the heat pipes’ heat transfer performance as a function of hydrogen cold topping levels.

1 INTRODUCTION

The 1300 km Trans Alaska Pipeline System (TAPS) transverses Alaska from Prudhoe Bay on the North Slope to its southern terminus, the Port of Valdez. Construction and performance of TAPS have been documented in many articles over the years. E.R. Johnson (1983) described the design and subsequent early performance. Passive heat transfer devices, referred to as heat pipes, were included in the vertical support member (VSM) design of the warm oil pipeline in those regions where warm permafrost was observed. The cooling effect of passive heat transfer devices on the soils surrounding the VSMs prevents thaw degradation of the permafrost, as well as increases the adfreeze strength of the pile-soil bond. VSMs with heat pipes installed are referred to as thermal piles or thermal VSMs. Approximately 124,300 heat pipes were installed.

The TAPS heat pipes 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 VSMs, 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 VSMs, 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 (NH₃) as the working fluid. The heat pipe’s internal volume consists of 2–3% liquid ammonia with the remaining volume ammonia 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 through continuous evaporation and condensation of a working fluid. 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 a liquid because of cooling by ambient air. The condensate flows downward on the internal wall surface of the heat pipe where it absorbs heat from the ground and is re-evaporated to continue the process. 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 thermal pile maintenance program, surveillance of the heat pipes using infrared (IR) viewing equipment determines if heat pipes continue to transfer heat from the ground. When the heat pipes are actively transferring heat, the finned condenser sections or “radiators” will appear to glow in the infrared portion of the radiation spectrum. Johnson (1983) reported that during the 1980 surveillance, many of the heat pipes appeared to have “cold tops” and postulated that this was caused by non-condensable gasses (NCG) collecting on top. This condition was recognized as a potential problem during design, although the extent and magnitude were not predictable. A typical IR image of a “cold top” heat pipe next to a fully functional heat pipe is shown in Figure 1. The NCG collects in the upper portion of the finned section and reduces the performance of the heat pipe. Subsequent IR surveys showed that the level of cold topping and number of affected heat pipes increased over time. Following the 1990/1991 survey, Williams (1991) reported that the number of heat pipes with detectable cold topping was 46% ± 10%. At that time about 28% of those heat pipes were
reported to have cold topping equal to or greater than 20%. A cold topping level of 30% or more was set to identify heat pipes that were candidates for repair. In 1991 about 10% of heat pipes had over 30% cold topping.

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

It was widely held that a general solution to NCG cold topping 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. Since 1983 various repair methods were initiated, including use of a hydrogen-absorbing metal halide pin, heat tube replacement inserts, and refrigerant replacement. In order to find which heat pipe needed repair, it was first necessary to define performance of the heat pipes as a function of cold topping levels.

2 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 (Fig. 2). The four identical calorimeter baths 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 (Fig. 3). Urethane rigid foam plastic insulation of 76 mm thickness and an outer sheet metal shield housed the inner calorimeter bath pipe. After the installation of a submersible circulating pump and heat pipe, each bath pipe was filled with propylene glycol and water with a freeze point of $-32^\circ$C.

Temperature-sensing instrumentation consisting of three thermistor strings was installed on the four heat pipes as follows: String 1 – nine thermistors along the root of adjacent aluminum fins on the condenser section, String 2 – five thermistors along the shielded section below the finned section, and String 3 – three thermistors along the evaporator section and 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 base of the remediation “V”. All other pads were about 76 mm by 76 mm. 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 hot tapped through a ball valve. 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 maintained 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. A radiation-shielded thermistor in a louvered enclosure measured ambient outdoor temperature. A data logger continuously recorded the data sensed by the thermistors, wind speed transducer, and power transducers.

Figure 3. Four calorimeter set-up.

3 HEAT PIPE PREPARATION

APSC has a large quantity of “new” heat pipes, to serve as replacements where pipeline repairs and/or maintenance are required. Nine 9.5 m long heat pipes were used for the test program.

The liquid level of ammonia in each of the nine heat pipes was measured. This was done by elevating the condenser end of the heat pipe to a 30° angle 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 ammonia in the heat pipes was calculated based on the measured liquid-vapor interface, temperature of the heat pipe, and the known internal dimensions of the heat pipe. The liquid density, \( \rho_l \), 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 this data in conjunction with the volumes of liquid, \( V_l \), vapor, \( V_v \), and total volume, \( V \):

\[
m = m_l + m_v = V_l \rho_l + \frac{V}{v_v} = V_l \rho_l + \frac{V - V_l}{v_v}
\]

Using Equation 1 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 calorimeter baths and data was gathered and IR images taken to confirm that all heat pipes were performing similarly with no cold topping observed. One heat pipe remained the control throughout the testing program.

4 HYDROGEN INJECTION

Hydrogen was injected 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 550 kPa. A 737 mm long section of 12.7 mm diameter steel pipe (internal volume \(93.4 \times 10^3 \text{mm}^3\)) attached to the manifold was purged with hydrogen and then filled to 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 \(14 \times 10^{-6}\) kg of hydrogen was injected into three test heat pipes during
each charge cycle. Following injection, data was collected and IR pictures taken over one to two weeks before another injection cycle was performed.

5 DATA RECORDING

Bath temperatures during the 2000–01 test program were 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 period and stored in the data logger. Ambient, fin, bath, and evaporator temperatures were stored at the end of each hour.

Data was downloaded from the data logger to a spreadsheet. Heat pipe conductance, \( C = \frac{Q}{\Delta T} \), 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 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 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.

6 AMMONIA AND HYDROGEN MIXTURE

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 vapor moves 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 exceeds the diffusion velocity of the hydrogen, most of the hydrogen ascends to the condenser section.

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

\[
P_H = P_{\text{TOTAL}} - P_{\text{AM}} \tag{2}
\]

\[
P_H V_H = m_H R_H T_{\text{COND}} \tag{3}
\]

The percentage of hydrogen cold topping was estimated based on the mass of hydrogen and temperatures of the evaporator and condenser sections.

\[
\text{Cold Top\%} = \frac{V_H}{V_{\text{COND}}} = \frac{m_H R_H T_{\text{COND}}}{(P_{\text{TOTAL}} - P_{\text{AM}}) V_{\text{COND}}} \times 100\% \tag{4}
\]
In Equation 4, $T_{COND}$ is the surface temperature of the cold-topped section of the condenser. $P_{TOTAL}$ is the total pressure in the heat pipe based on evaporator temperature. $P_{AM}$ is the partial (vapor) pressure of ammonia in the cold-topped condenser section based on $T_{COND}$. $V_H$ is the volume of hydrogen-ammonia mixture. $V_{COND}$ is the internal volume of the condenser section, $m_H$ is the mass of hydrogen, and $R_H$ is the gas constant for hydrogen.

Equation 4 neglects hydrostatic and dynamic pressure drops occurring between the evaporator and condenser sections. This formulation was used to develop a normalized cold topping chart. Because the performance testing conducted during the winter of 2000–01 was done at an evaporator temperature of $-1^\circ C$ that condition was chosen as a normalizing value. It also represented the warmest allowable temperature of the ground at the end of summer or beginning of winter around the VSM. Normalized cold topping as a function of condenser temperature and evaporator temperature is shown in Figure 6. Heat pipe testing at several evaporator temperatures is being conducted to validate these predicted trends.

Figure 6 shows cold topping as a function of evaporator temperature and condenser temperature. As the condenser temperature is decreased at a fixed evaporator temperature, the partial 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 cold topping level. 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 cold topping level.

7 PERFORMANCE AS A FUNCTION OF HYDROGEN BLOCKAGE

The original APSC design basis for thermal VSMs used zero wind speed to establish thermal VSM performance. Therefore, the recorded data from these tests was grouped as a function of wind speed and amount of hydrogen injected. Zero wind speed data was 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. The trend curve represents 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 cold topping. As expected, the heat pipe conductance increases with increasing wind speed. The zero cold topping conductances are greater than those reported by Haynes & 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, greater heat rejection due to a lower surrounding air temperature, larger evaporator-condenser temperature differences, and a 2.4 m bare pipe within a metal collar between the calorimeter bath and finned section (part of the design).

Visual evaluation of cold topping 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.

8 CONCLUSIONS

The 2000–01 test program obtained the data necessary to determine TAPS heat pipe thermal degradation as a function of NCG cold topping. 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 cold topping was determined by measurements of power supplied to the evaporator section and evaporator and ambient air temperatures. This data was used to develop a relationship for heat pipe conductance as a function of observed and measured NCG cold topping.

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

REFERENCES


