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Longitudinal Vibration Effects on a Copper/Water Heat Pipe's Capillary Limit

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The effect of longitudinal vibration on the capillary limit of a copper/water heat pipe with a tightly wrapped screen wick was investigated. The capillary limit was measured over a range of operating temperatures under static conditions. A bench-top shaker was used to provide vibration in the longitudinal axis of the heat pipe. The capillary limit was measured at vibration frequencies of 10, 30, and 50 Hz. At each of these frequencies, tests were run at vibration amplitudes of 0.2 and 2.0 g. The pipe was maintained at a constant inclination angle and power throughput was increased until dryout occurred. The power throughput at dryout was considered the capillary limit. The measured capillary limit for each vibration test was compared to the static tests to determine the effect of the vibration. The results covered operating temperatures from 41.4 to 96.6°C. A vibration amplitude of 0.2 g caused a 2.3% decrease in the capillary limit. However, there is a 3.7% uncertainty in the data. A vibration amplitude of 2.0 g at 30 and 50 Hz caused a 9.6 and 6.1% degrade in capillary heat transport limit, respectively. At 2.0 g and 10 Hz there is a 1.8% decrease in capillary heat transport limit.

Nomenclature

c_p	=	specific heat at constant pressure, J/kg-K
g	=	acceleration due to gravity, m/s^2
I	=	current, A
$\dot{m}$	=	mass flow rate of coolant, kg/s
P	=	power, W
$Q_{c,\max}$	=	capillary heat transport limit, W
T	=	temperature, °C
T_{in}	=	manifold inlet coolant temperature, °C
T_{op}	=	operating temperature, C
T_{out}	=	manifold outlet coolant temperature, °C
$T\#\#$	=	thermocouple number location

Introduction

THERE has been very little research concerning vibration effects on heat pipes. Three previous studies were reviewed that investigate the effect of vibration on heat pipe performance. It is interesting to note that the first two date back to the late sixties/very early seventies and that the last study was done in 1992.

An experiment conducted by Deverall[1] addressed changes in operating temperature due to sinusoidal vibration. A stainless steel/water heat pipe was subjected to vibration spanning 5–2000 Hz and up to 12 *g*. The heat pipe was operated at temperatures of 60 and 90°C. Two specific tests were conducted. The first consisted of varying the frequency and noting changes in the heat pipe wall temperature. The second consisted of maintaining vibration at 60 Hz and varying the inclination angle of the heat pipe. Changes in wall temperature were again noted. This report concluded that vibration is not detrimental to heat pipe performance and can aid in the wetting of the wick.[1]

Another experiment conducted by Richardson et al.[2] addressed longitudinal vibration on heat pipe performance. A stainless steel/water heat pipe was subjected to frequencies of 60, 120, 240, and 580 Hz and inclination angles of 32, 35, and 38 deg. The time until dryout occurred was determined by sequentially increasing the power input. This study concluded that longitudinal vibration has a detrimental effect on the maximum heat transport capability and that this effect is greater at higher amplitudes and lower frequencies.[2] However, the authors did not quantify this effect and suggested further investigation.

A study conducted by Charlton[3] addressed the effect of transverse vibration on the capillary limit of a heat pipe. A wrapped screen wick copper/water heat pipe was subjected to transverse vibration at frequencies of 30, 250, and 1000 Hz and amplitudes of 1.0, 2.5, and 5.0 g. The heat pipe operating temperature varied from 50–75°C and was kept at an inclination angle of 0 deg. The test consisted of determining the heat pipe's static performance by increasing power input until dryout occurred. The vibrational performance was determined in a similar manner and the two were compared. This study concluded that there is little or no effect on the capillary heat transport limit due to transverse vibration.

The objective of this study is to determine the effect of longitudinal vibration on the capillary limit of a wrapped screen wick copper/water heat pipe. This is a direct extension of the transverse vibration experiment conducted by Charlton.[3] This study utilized the heat pipe that was designed and built by Charlton for the 1992 transverse vibration experiment. Tests to determine the static and dynamic performance of the heat pipe were conducted at 0-deg inclination and at varying amplitudes and frequencies. The operating temperature of the heat pipe was varied from 41.4 to 96.6°C. The static and dynamic capillary heat transport limits of the heat pipe were compared to identify trends occurring during vibration. Amplitudes of 0.2 and 2.0 g and frequencies of 10, 30, and 50 Hz were chosen. These amplitudes and frequencies were selected because many space-related applications are likely to encounter this vibration regime.[4] Also, Richardson[2] noted

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that lower frequencies (60 and 120 Hz) were detrimental to the performance of the heat pipe used in his experiment.

Experiment Design

The only change to Charlton's[3] heat pipe for this study is the use of a different heater. The new heater is slightly longer than the old one, which results in a longer evaporator section; 0.1022 m instead of 0.1016 m. The condenser length does not change, remaining at 0.1111 m, therefore, the adiabatic section was shortened to 0.0915 m from 0.0921 m. All other heat pipe dimensions remain as specified in Charlton's work.

The pertinent heat pipe measurements are as follows: total working length, 0.3048 m; o.d., 2.223 cm; i.d., 1.892 cm; wall thickness, 1.651 m; wick, 100 mesh copper screen; and water volume, 5.7 g.

The sonic and entrainment heat transport limits are unaffected by the evaporator and adiabatic section length changes. At 100°C the sonic limit is 74,910 W and the entrainment limit is 8746 W. The boiling limit increased slightly to 2088 W from 2076 W. The capillary heat transport limit for this study was slightly lower than that calculated by Charlton. It ranges from 101 W at a T_{op} of 37°C to 181 W at a T_{op} of 97°C. The capillary limit is clearly the most restrictive. In fact, it is at least an order of magnitude lower than any of the other limits. Therefore, when a maximum heat transport rate is reached, it can be attributed to the capillary limit.

This experiment design is a derivative of Charlton's setup. As such, much of the same equipment is used. The primary differences include: 1) a nichrome heater wire instead of a heating tape, 2) insulation of the entire heat pipe instead of just the heater, 3) a refrigerated bath/circulator to provide coolant at a constant temperature and flow rate instead of using tap water, and 4) a longitudinal vibration fixture instead of a transverse fixture. This setup is also very similar to the one used by Buchko[5] for his low-temperature heat pipe experiment.

Figure 1 shows the main components of the experiment setup. This figure provides reference points for the following component descriptions and test procedures.

Heater

A heater capable of providing a uniform heat distribution was desired. A uniform heat distribution resulted in a constant temperature throughout the evaporator section while the heat pipe was operating in the heat pipe mode. This constant temperature in the evaporator section made the temperature increase at dryout more noticeable. This heater also had to be capable of producing dryout in the heat pipe, i.e., it had to provide enough power density to exceed the heat pipe's power throughput capability. Finally, the heater had to be capable of sustaining longitudinal vibration at 2.0 g and up to 50 Hz, which were the vibration limits of this study. Several off-the-shelf heater units were available, but none were ideal. Therefore, a rugged, yet simple, heater was produced.

The resulting heater was a single core, nichrome heater wire with an Inconel 600 sheath. The maximum predicted power throughput for the heat pipe was 185 W. Therefore, the heater was designed to deliver 500 W. This ensured that dryout conditions could be met for this heat pipe and that the heater could be used for higher power applications in the future. The heat pipe evaporator area was 95.25 mm long by 22.23 mm diameter. Thirty wraps of 3.18-mm sheathing were necessary to fully cover the 95.25 mm length. This resulted in 2.39 m of total wire length. The resistance of the wire was 5.24 Ω/m for a total resistance of 12.52 Ω. A maximum input power of 500 W was achieved by supplying the wire with 79.2 V and 6.3 A. The heater was covered with 12.7 mm of RPC-XA-AQ felt (a 1566°C ceramic fiber insulation made by Refractory Products Company). This fiber was also applied to the entire length of the heat pipe in an equal depth.

The power supplied to the heater was monitored using an ammeter and a voltmeter. The power is simply the voltage multiplied by the current, $P = I \times V$. A signal conditioning rectifier (SCR) was used to apply the power in approximately 5-W increments. The actual power input was not important to this study. The efficiency of the pipe is not under investigation, only the power throughput. The power monitoring was done to ensure that a constant increment was applied to the heat pipe for all of the tests.

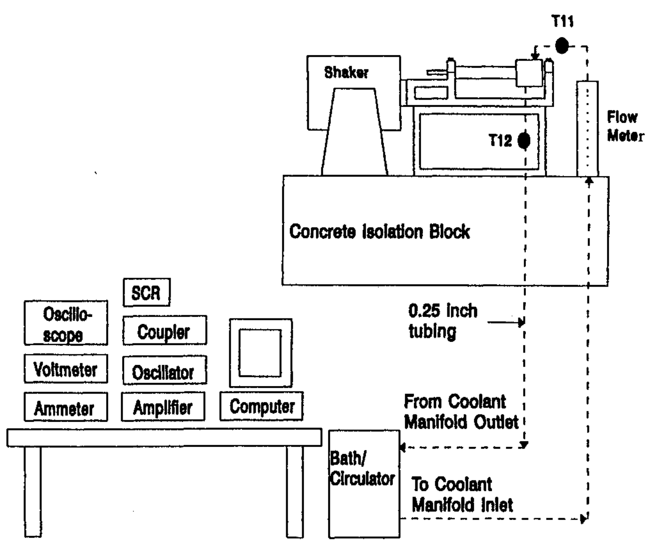

Fig. 1 Experimental setup: $T11$ = coolant inlet temperature and $T12$ = coolant outlet temperature.

Coolant Control

The coolant served several purposes. The most important was in the calculation of the capillary limit $Q_{c,\max}$. Buchko's[5] low-temperature heat pipe experiment, Charlton's[2] transverse vibration experiment, and this study all measure $Q_{c,\max}$ using

$$Q_{c,\max} \cdot \dot{m}c_p\Delta T \tag{1}$$

where ΔT is the difference between the coolant manifold inlet and outlet temperature (K).

Equation (1) was used to calculate $Q_{c,\max}$ at dryout. The ΔT was the temperature difference between the coolant inlet and outlet. The coolant also dictated the heat pipe operating temperature. The operating temperature could be varied by changing either the coolant inlet temperature or the coolant flow rate. Both had the effect of changing the mean condenser temperature, which in turn changed the heat pipe's operating temperature. A lower condenser temperature resulted in a lower operating temperature. A higher flow rate also resulted in a lower operating temperature. A constant flow rate was chosen and the coolant temperature was varied to control operating temperature.

A NESLAB Endocal RTE-100 refrigerated bath/circulator was used to supply the coolant. The bath/circulator pumps fluid at a constant mass flow and can be set to temperatures ranging from 0 to 100°C to the nearest 0.1°C. The flow rate of the coolant system was controlled with a variable flow meter placed in the coolant line. A Manostat scale/float-type flow meter was used. The flow meter was set to 100 (approximately 0.0016 kg/s) on its scale and then calibrated for each of the coolant temperatures used.

Vibration Control System

The shaker system used to provide longitudinal vibration was identical to the one used during Charlton's transverse vibration experiment. For this study, the Unholtz–Dickie shaker was bolted directly to a 1-m^3 concrete isolation block.

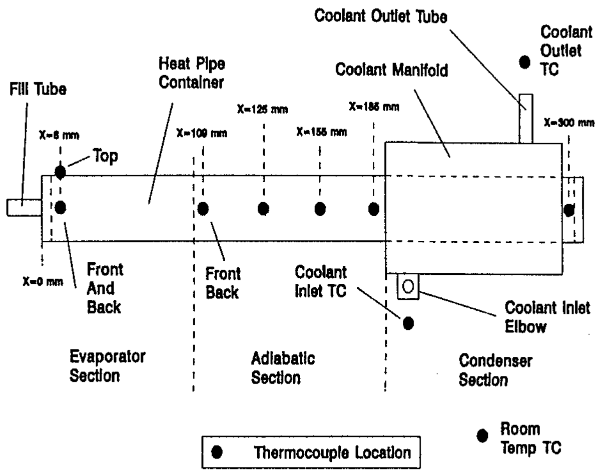

Fig. 2 TC locations.

This was done in an effort to eliminate as much vibration in the transverse and vertical axis as possible. The feedback accelerometer was recalibrated to ensure accurate vibration amplitudes.

The vibration fixture had to hold the heat pipe at a constant inclination angle and maintain mainly longitudinal motion. The fixture was designed to attach directly to the shaker actuator surface. The shaker and the support bracket were bolted directly into the concrete isolation block. The support bracket had a groove along its top in which the flange of the heat pipe mounting slides in. This reduced the amount of vibration in extraneous directions. All vibration fixture parts were made of aluminum and were manufactured in a machine shop. Phenolic insulation was used to thermally and electrically insulate the heat pipe from the vibration stand. The mounts were designed for ease of heat pipe removal if necessary.

Data Acquisition

The data acquisition system was capable of recording or displaying the following information: 1) vibration frequency, 2) vibration amplitude, 3) heat pipe wall temperature, 4) coolant inlet and outlet temperature, 5) room temperature, 6) heater power, 7) coolant flow rate, and 8) coolant temperature in the bath/circulator.

The vibration data was gathered using accelerometers. The longitudinal axis accelerometer acted as the control for the vibration level in the shaker system. The transverse and vertical axis amplitudes were fed directly to an oscilloscope for amplitude measurement. This experiment made use of the exact shaker control/accelerometer pairing used by Charlton for the transverse study. The off-longitudinal accelerometer readings were used to ensure that the vibration was occurring mainly in the longitudinal direction. The off-longitudinal vibration ranged between 2.5–40% of the longitudinal vibration.

The temperature measurements were all taken using T-type copper/constantan thermocouples. There was a total of nine thermocouples (TCs) connected directly to the heat pipe wall as shown in Fig. 2. Reference 6 provides thermocouple attachment and temperature recording details.

All temperature readings were taken at 5-s intervals to minimize the size of the data files. Time intervals of 5 s provided enough resolution to determine when dryout occurred.

Experimental Test Procedure

A very strict test procedure was developed and followed for all test runs. This ensured consistency in the data gathered. The following test procedures were used:

1) Ensure that the heat pipe is initially at room temperature.

2) Incline the heat pipe (condenser high) to approximately 45 deg to ensure that the wick has rewet (keep inclined for at least 30 min).

3) Return the heat pipe to a horizontal position.

4) Input approximately 100 W of power into the heater.

5) Allow the heat pipe to reach steady-state operation (indicated by T1, T2, and T3 changing by less than 0.5°C over 10 time steps, 50 s).

6) Begin vibration at specified amplitude and frequency.

7) Begin taking temperature data.

8) Increment the input power by 5 W and allow to reach steady operation again.

9) Repeat step 8 until dryout occurs.

10) Turn off input power and allow pipe to cool.

11) Look at the temperature data file and determine at which time index dryout occurred.

12) Record frequency, amplitude, T_{op}, T_{in}, T_{out}, ΔT, and mass flow rate.

13) Look up c_p (Ref. 7) at the average coolant manifold temperature.

14) Calculate $Q_{c,\max}$ using Eq. (1).

Experimental Analysis

Temperature Measurements

The accuracy of the temperature measurements for this study were the same as those done by Charlton.[3] In addition, because the same temperature data acquisition was used, the temperature data error analysis is very similar to Charlton's.

Great care was taken in attaching the nine heat pipe wall thermocouples. Each was wired flush to the wall and covered with a conductive grease. This, in conjunction with an added layer of insulation over the length of the heat pipe, helped to ensure that only the heat pipe wall temperature was being measured as opposed to the temperature of the surroundings. For this study, the adiabatic section's wall temperature readings were the most important. The evaporator section's wall thermocouples were used to observe temperature trends and to predict dryout. They were not used to determine an exact temperature. The adiabatic section thermocouples were used to determine the heat pipe operating temperature. The precautions taken in attaching these thermocouples result in a very small error.

Temperatures were recorded to the nearest 0.1°C; however, the temperature readings fluctuated by up to 0.4°C for both the steady and dynamic cases. It was assumed the temperature readings were accurate to the nearest 0.5°C.

Vibration Data

The accuracy of the longitudinal accelerometer was ensured by calibrating it before any test runs. This process was also identical to the one used by Charlton.[3] In fact, it was conducted on the same equipment and used the same standard calibrating accelerometer. Once calibrated, the output of this accelerometer was displayed on the oscillator to the nearest 0.01 g.

The off-longitudinal accelerometers were used only to ensure that the vibration was occurring mainly in the longitudinal direction. These accelerometers were factory calibrated and their readout was displayed on an oscilloscope. The oscilloscope was observed for all of the test runs and the worst case off-longitudinal accelerations were recorded.

Coolant Flow Measurement

The coolant flow was monitored using an adjustable flow meter. The coolant temperature ranged from 0 to 60°C. It was decided to use only one flow meter scale setting and calibrate it over the expected temperature range of the coolant. A scale setting of 100 was chosen, resulting in flow rates of 0.001446–0.001833 kg/s.

Heat Pipe Inclination

The heat pipe inclination angle has a strong effect on performance. If the condenser is higher than the evaporator, capillary pumping is aided by gravity and dryout can be delayed. If the condenser is lower than the evaporator, capillary pumping must work against gravity and dryout will occur early. This study did not investigate changes in inclination angle, but used an angle of 0 deg for all test cases.

Determination of Accumulated Experimental Error

A rss uncertainty analysis was performed to determine the error bars associated with each test run. This analysis is similar to Charlton's,[3] but varies in the mass flow error calculation.

Experimental Results

Static Data

The static performance of the heat pipe was determined by conducting 21 test runs. These static tests were conducted holding the heat pipe motionless and near to horizontal. Operating temperatures between 41.4–96.6°C were investigated. A lower limit of 41.4°C was selected because it was the lowest operating temperature feasible at the coolant temperature lower limit (0.0°C). An upper limit 96.6°C was selected because it was just within the heat pipe design operating temperature of 100°C. These limits were also chosen to closely model the limits from the transverse study.

Figure 3 presents the operating limit of the heat pipe under static conditions. The static test data were consistently above the predicted operating limit of the heat pipe. "Variations in cleanliness during assembly, the production technique, the true physical dimensions, and the tightness of the wick wrap are a few of the variables that impact the performance of the pipe."[3]

The predicted operating limit of the heat pipe assumes an inclination angle of 0 deg (horizontal). However, although the experiment kept a constant inclination angle, it was not necessarily horizontal. An inclination angle other than 0 deg allows gravity to effect the heat pipe performance:

$$Q_{c,\max} = 24.4045 \cdot (T_{\mathrm{op}})^{0.454397} \tag{2}$$

Therefore, the heat pipe inclination angle was kept constant to ensure that gravity did not effect dryout. Any of these variables could explain why the heat pipe operated above the predicted operating limit. A power law relation predicted the operating limit most accurately and was used to describe the static data: Equation (2) will be used during the statistical breakdown of the longitudinal vibration results. Specifically, Eq. (2) will represent a baseline with which to compare the vibration data.

Longitudinal Vibration Data

The effect of longitudinal vibration on the heat pipe was determined by comparing test runs at specific amplitudes and frequencies against the static runs. Amplitudes of 0.2 and 2.0 g and frequencies of 10, 30, and 50 Hz were used. Higher g loads were not possible due to limitations of the bench-top shaker. Each amplitude was run at each of the three frequencies. These six combinations were then run at three different operating temperatures to create 18 different test cases. In addition, five extra cases were run to confirm repeatability, bringing the total number of test cases to 23. The work done by Charlton for transverse vibration was conducted at amplitudes of 1.0, 2.5, and 5.0 g. Each of these amplitudes were run at frequencies of 30, 250, and 1000 Hz (Ref. 3). The transverse study indicated little or no effect at these amplitudes and frequencies. However, the greatest effect was noticed at 30 and 250 Hz. This can be attributed to larger displacements corresponding to lower frequencies. In addition, space systems spend a significant amount of time exposed to vibration low frequency, low amplitude. Therefore, the longitudinal study was conducted at lower amplitudes and frequencies.

The lower frequencies caused the vibration apparatus to physically displace the heat pipe far more than at higher frequencies. A larger displacement causes more of an impact on the bulk movement (if any) of the working fluid. The lower frequencies would also allow more time for each displacement to make an impact.

Each amplitude was run at each of the frequencies resulting in six cases. These six cases were conducted at three operating temperatures resulting in a total of 18 different vibration cases. Five additional cases were also run to ensure repeatability. The three operating temperatures for the 23 desired cases were approximately 42, 63, and 87°C. The operating temperature varied slightly as a result of the uncertainties in the exact power input to the heater due to the accuracy of the power supply.

Figure 4 depicts the static, vibration, and static curve fit [Eq. (2)] data plotted on the same graph. Initial observations lead to the conclusion that the 2.0-g data have lower capillary

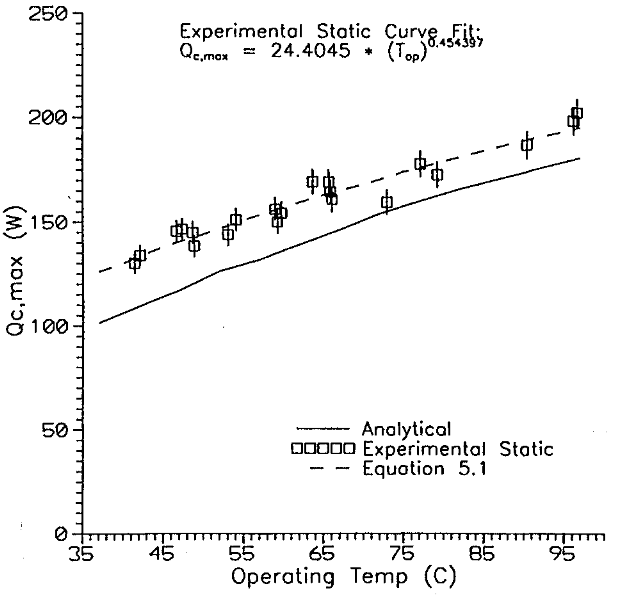

Fig. 3 $Q_{c,\max}$ vs T_{op}: static and predicted.

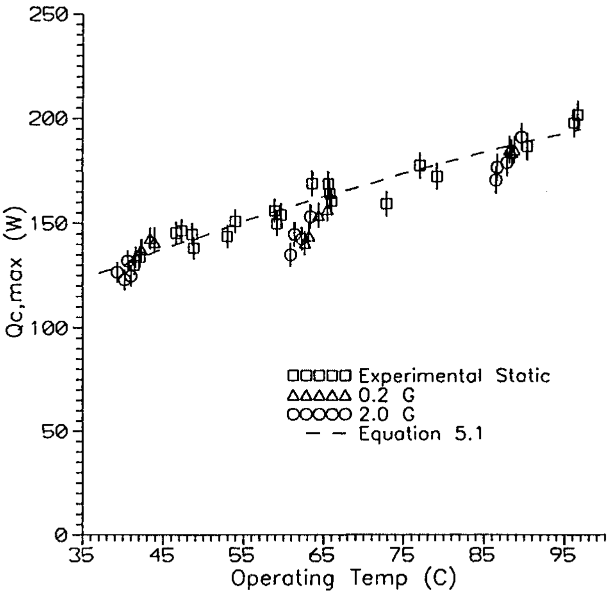

Fig. 4 $Q_{c,\max}$ vs T_{op}: static and vibration.

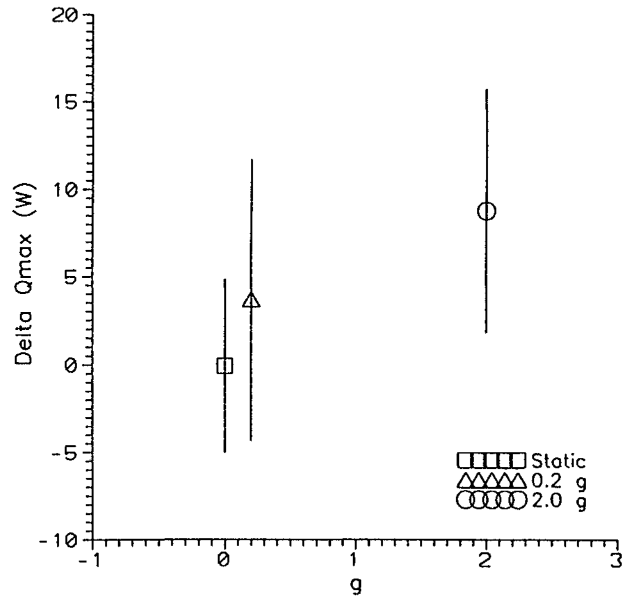

Fig. 5 Mean $\Delta Q_{c,\max}$ vs amplitude: all frequencies.

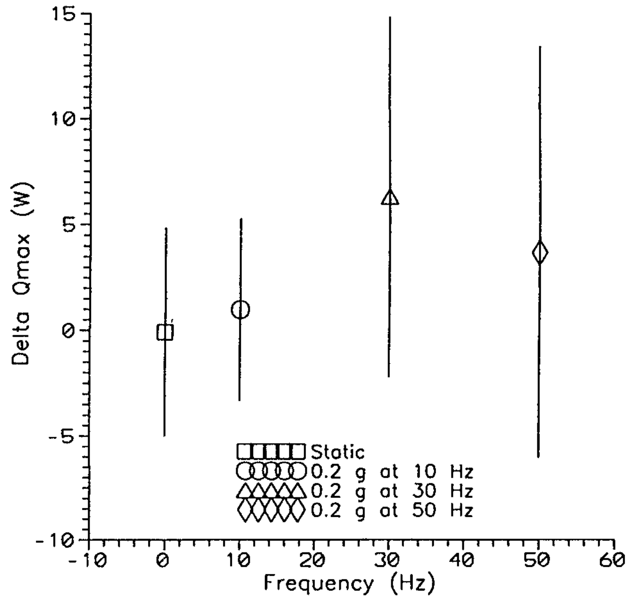

Fig. 6 Mean $\Delta Q_{c,\max}$ vs frequency: 0.2 g.

heat transport limits at similar operating temperatures compared to the static data. The only difference between the static and vibration tests was the movement of the heat pipe under vibration conditions. Therefore, the lower capillary heat transport limit for the vibration tests is attributed to the vibration. This is especially noticeable at the midrange operating temperature.

To statistically represent the capillary heat transport limit data, the change in $Q_{c,\max}$ at different amplitudes and frequencies has to be quantified. A $\Delta Q_{c,\max}$ was found by subtracting the actual $Q_{c,\max}$ from the expected static $Q_{c,\max}$ [Eq. (2)]. This effectively deleted temperature as a variable because the static and vibration capillary heat transport limits can be compared at the same operating temperature. A positive $\Delta Q_{c,\max}$ indicated that the actual $Q_{c,\max}$ was less than the expected $Q_{c,\max}$. A mean and a standard deviation for $\Delta Q_{c,\max}$ was then calculated for the static, 0.2- and 2.0-g data.

Figure 5 demonstrates that $\Delta Q_{c,\max}$ increases with increasing g level; i.e., the vibration data were below the static data. The centered symbols in Fig. 5 represent the mean $\Delta Q_{c,\max}$ and the vertical lines represent the standard deviation.

There is a 3.76-W increase in the mean value of $\Delta Q_{c,\max}$ at 0.2 g and a 8.86-W increase at 2.0 g. These increases in $\Delta Q_{c,\max}$ indicate a 2.3 and 5.4% decrease in $Q_{c,\max}$ from the static curve fit for the 0.2- and 2.0-g tests, respectively. These percentages are based on the average predicted $Q_{c,\max}$ of the heat pipe (162.6 W at 65°C). A 3.76-W increase is inside the data uncertainty of approximately 6.0 W, and so the 0.2-g data are inconclusive. However, a 8.86-W increase in $\Delta Q_{c,\max}$ is outside of the data uncertainty, so that there is a definite degradation in performance at 2.0 g. This degradation is caused by the vibration amplitude (2.0 g) being large enough to effect the heat pipe performance.

The 0.2- and 2.0-g data were then reduced as a function of the three frequencies investigated. The mean and standard deviations for $\Delta Q_{c,\max}$ were calculated for the 10-, 30-, and 50-Hz data at each of the amplitudes. Figures 6 and 7 depict this analysis.

Figure 6 again shows a very small increase in $\Delta Q_{c,\max}$ for the 0.2-g data, regardless of the frequency, because 0.2 g is not enough amplitude to effect the heat pipe performance. The largest $\Delta Q_{c,\max}$ in Fig. 6 is 6.34 W at a frequency of 30 Hz. This is right at the data uncertainty, and so the 0.2-g data as a function of frequency are inconclusive.

Figure 7 shows much larger increases in the mean $\Delta Q_{c,\max}$ for the 2.0-g data. This is especially noticeable at the 30- and

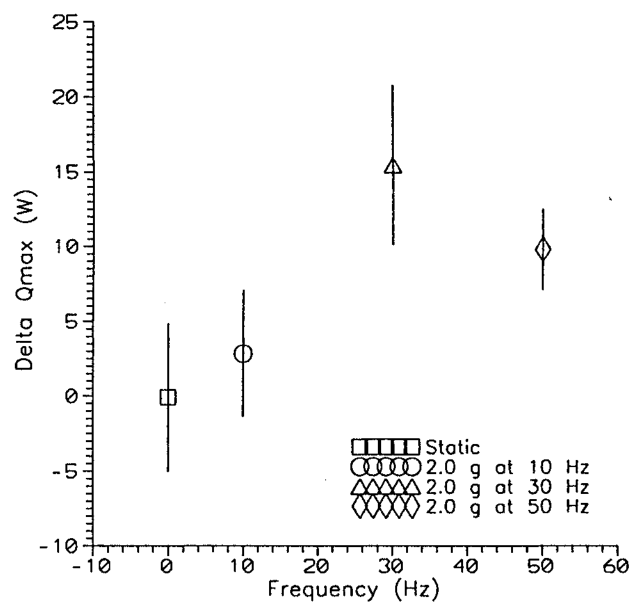

Fig. 7 Mean $\Delta Q_{c,\max}$ vs frequency: 2.0 g.

50-Hz frequencies. At 30 and 50 Hz the increase in $\Delta Q_{c,\max}$ is 15.53 and 9.88 W, respectively. These $\Delta Q_{c,\max}$ increases correspond to decreases of 9.6 and 6.1% in $Q_{c,\max}$. These are both much greater than the uncertainty in the data. Therefore, the heat pipe's capillary heat transport limit is conclusively degraded at 30 and 50 Hz.

In Figs. 5–7, the standard deviation bars for the 0.2-g data overlap those of the static to a great extent. When this happens, any change in $\Delta Q_{c,\max}$ is inconclusive. The standard deviation bars for the 2.0-g data do not overlap the static standard deviation bars as much. Figure 7 shows no error bar overlap at 30 and 50 Hz. Therefore, $Q_{c,\max}$ is conclusively changed because the mean $\Delta Q_{c,\max}$ is larger at 30 and 50 Hz than the statistical uncertainty. This is the same trend that was observed in the transverse study. In that study, the only data that had a noticeably reduced $Q_{c,\max}$ were at 30 and 250 Hz.[3] The fundamental frequencies of the heat pipe were calculated to determine if these frequencies corresponded to the peak in $\Delta Q_{c,\max}$. The first fundamental frequency was on the order of 12,000 Hz and the higher fundamentals were upwards of 30,000 Hz. No relationship appears to exist.

Table 1 Medium T_{op} temperature variation test (125.1 W)

Time	$T1$	$T2$	$T4$	$T5$	$T6$	$T7$	$T8$	$T9$	$T11$	$T12$
0	67.8	68.6	58.9	58.6	57.2	56.6	53.8	45.3	28.9	44.7
300	67.6	68.4	58.9	58.6	57.2	56.3	53.8	45.3	28.9	44.7
600	67.3	68.4	59.1	58.9	57.2	56.6	54.1	45.6	28.9	44.7
900	67.1	68.2	59.0	58.4	57.3	56.4	53.6	45.4	29.0	44.8
1200	66.5	67.9	58.7	58.1	57.0	55.9	53.1	46.3	28.7	44.8
1600	68.4	68.9	59.4	58.9	57.5	56.6	53.8	46.8	28.9	44.7

During the course of the vibration testing, temperature variations along the length of the heat pipe were noticed. Temperature measurements taken along the heat pipe during static conditions changed when vibration was induced. To investigate this phenomenon, the heat pipe was allowed to achieve static, steady conditions (operating in heat pipe mode) at a point well within its operating regime. The heat pipe was operated for 1 h to ensure that the temperature readings along its length were no longer changing. The heat pipe was then vibrated at 2.0 g at various frequencies.

Vibration does cause the temperature along the heat pipe to change. In general, the temperatures decreased approximately 1°C during the vibration phase and then rose rapidly (within 15 s) when the vibration ceased. This effect happens predominantly at the evaporator end (T1 and T2). The condenser, coolant inlet, and coolant outlet temperatures (T9, T11, and T12) are not effected by the vibration. Therefore, even though there is some variation in the wall temperature of the heat pipe, vibration does not appear to effect the power throughput capability in this type of test. This temperature variation trend appears to be more severe for a heat pipe at a low T_{op} than a heat pipe at a high T_{op}. At a low T_{op} the temperature decrease is 1°C, and at a high T_{op} the temperature decrease is 0.5°C. These observations were also noted briefly during the vibration test done by Deverall[1] of a stainless steel/water heat pipe. In that test, the temperature drop along the heat pipe decreased from 2 to 1°C, indicating an actual performance improvement. A temperature drop along the heat pipe indicates a performance increase because the power throughput is maintained with a smaller temperature drop along the heat pipe. Table 1 shows an example of the temperature fluctuations due to vibration.

Conclusions

This study incorporated all of the suggested experiment improvements from the transverse study.[3] These included the following:

1) Maintain a constant heat pipe angle from test to test.

2) Insulate the entire length of the heat pipe to reduce convection and radiation heat transfer.

3) Insulate the inlet and outlet coolant lines to eliminate convection and conduction heat transfer losses before the temperature measurement is taken.

4) Implement a constant flow/constant temperature coolant supply.

These improvements drastically improved the quality of the data obtained during this study. The constant flow/constant temperature alone provided much greater consistency and repeatability.

Longitudinal vibration causes a decrease in the capillary heat transport limit of a wrapped screen wick copper/water heat pipe. The decrease is most noticeable at specific vibration amplitudes and frequencies. The results of this study do agree with the subtle trends indicated by the transverse vibration study.[3]

A vibration amplitude of 0.2 g has a combined 2.3% decrease in $Q_{c,max}$ over the entire frequency band investigated. The uncertainty in the actual heat transport measurement is approximately 3.7% and is greater than the difference between the static and the 0.2-g vibrating cases. Test runs involving repeated operating temperatures resulted in capillary heat transport limit decreases less than the 3.7% uncertainty. Therefore, no quantified conclusion can be drawn about the 0.2-g vibration tests other than that there appears to be a trend towards decreased heat transport.

A vibration amplitude of 2.0 g has a much greater influence on the capillary heat transport limit. At frequencies of 30 and 50 Hz there is a 9.6 and a 6.1% decrease from the measured static heat transport limit, respectively. The data taken at 10 Hz indicates a 1.8% decrease, but is not conclusive due to its uncertainty (3.7%). Once again, test runs involving repeated operating temperatures resulted in capillary heat transport limit decreases less than the 3.7% uncertainty.

Changes in longitudinal vibration do cause temperature variations to occur along the heat pipe. This effect takes place predominantly at the evaporator. The temperature at the evaporator fell approximately 1°C when a static heat pipe was vibrated and increased rapidly (within 15 s) to at least the original temperature when the vibration stopped. There was no temperature or power throughput change at the condenser end. The decrease in temperature at the evaporator is more pronounced for a low operating temperature than for a high operating temperature. At a low T_{op} the temperature decrease was 1°C, and at a high T_{op} the temperature decrease was 0.5°C.

Research in the area of capillary heat transport limit should be continued to better quantify the results of vibration on heat pipe performance. The data taken during the transverse and the longitudinal vibration study should be expanded to include more amplitudes and frequencies. The vibration wave form should also be varied during the frequency and amplitude range extensions. Also, heat pipes of other designs should be submitted to the same types of experiments. In particular, axially grooved wick heat pipes should be investigated because they do not contain the screening that could be overcoming the effect of vibration in other experiments.

In addition to the capillary limit, the entrainment limit of heat pipes under vibration conditions should be investigated. Vibration may cause fluid to be torn from the wick structure and increase the amount of entrainment occurring. Sintered metal heat pipes also need to be investigated to see if vibration causes wick breakdown.

The temperature variations should also be further investigated. This study only touched on what was noticed as an aside to the actual capillary heat transport limit investigation. Temperature variations tests should be conducted for a greater range of amplitude, frequency, and operating temperature. Other heat pipe designs should also be investigated for temperature variations during vibration.

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