



# OPERATING CHARACTERISTICS OF NAPHTHALENE HEAT PIPES

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## ABSTRACT

Heat pipes that operate in the medium temperature range (550-700 K) are very rarely used in industry despite the potential demand of use. There is no consensus about suitable working fluids in this temperature range as research on possible working fluids is limited. One proposed working fluid is naphthalene. In this paper, a number of tests have been undertaken on both an individual naphthalene heat pipe and a naphthalene heat pipe heat exchanger. Unlike room temperature working fluids, medium temperature working fluids are solid at ambient temperature therefore they have unusual transient start up behaviour. Testing has indicated that these heat pipes start to operate when the temperature of the adiabatic section reaches approximately 200 °C. The tested heat pipes were 8 mm in diameter and 278 mm long. The container and mesh material were stainless steel. They were found to have a thermal resistance of approximately 1 °C/W and a maximum rate of heat transfer of 40 W. The orientation was found to have a large effect on the performance of the heat pipes. Compared to the bottom heat mode orientation, when in a horizontal orientation the heat exchanger effectiveness more than halved and when in the top heat mode orientation heat exchanger effectiveness was significantly further reduced.

**Keywords:** Naphthalene, heat pipe performance, transient start-up behaviour, heat exchanger

## 1. INTRODUCTION

The typical working fluid used in heat pipes is water. Unfortunately water, like all working fluids, has limitations of use. One of these limitations is the operating temperature of the heat pipe. At operating temperatures that are too low, the working fluid will solidify and stop the heat pipe from working. Therefore for water this will be at 0 °C. At operating temperatures that are too high, the pressure inside the pipe will increase to a level which may rupture the pipe. Also, the operating temperature can't exceed the critical point temperature of the working fluid because phase change is required for the heat transfer. Therefore for water this will be at 374 °C. There are many heat transfer applications which fall outside this range. In these situations, different working fluids are required. At low temperature / cryogenic temperatures (1-200 K), working fluids such as helium, argon, neon and nitrogen are used. In room temperature applications (200-550 K), working fluids such as water, methanol, ethanol, ammonia and acetone are used. In high temperature applications (>700 K), working fluids such as potassium, sodium and silver can be used. All these different working fluids have varying degrees of performance due to different fluid parameters. Working fluids for the medium temperature range (550-700 K) are not as common. Organic fluids such as biphenyl have been suggested. Naphthalene is another working fluid which is suggested (Yang *et al.*, 2012; Vasiliev, 2005).

The reason naphthalene can be used for a medium temperature range heat pipe is because of the relatively high melting and boiling points. Under atmospheric pressure, naphthalene has a melting point of 80.2 °C and a boiling point of 218 °C. Heat pipes using naphthalene will therefore not work under an operating temperature of 80.2 °C. The vapour pressure at high temperatures is relatively low as compared to water which allows the operating temperature to go beyond the maximum operating temperature of water. Experimental work has been undertaken on naphthalene heat pipes testing for performance, container compatibilities

and performance degradation over time (Kniess *et al.*, 2007; Kimura *et al.*, 1994; Mantelli *et al.*, 2010; Vasil'ev *et al.*, 1988; Anderson, 2007). Naphthalene heat pipes have been proposed for use in high temperature heat storage applications (Robak *et al.*, 2011; Khalifa *et al.*, 2015; Khalifa *et al.*, 2014; Liu *et al.*, 2015). Some Chinese companies have deployed heat exchangers using naphthalene heat pipes in conjunction with other heat pipes of a different working fluid (TianLi, 2009). Liquid metals are used in the high temperature section, naphthalene is used in the medium temperature section and water in the low temperature section. The amount of literature on naphthalene heat pipes is quite limited and very few papers have been published in recent times. Therefore the aim and novelty of this paper is to investigate the performance and characteristics of naphthalene heat pipes.

## 2. TESTING AN INDIVIDUAL NAPHTHALENE HEAT PIPE

### 2.1 Initial test set up and data

A number of tests were conducted on an individual naphthalene heat pipe. The specifications of this heat pipe can be seen in Table 1. The test set up consisted of a high temperature electrical heater to be placed on the evaporator section of the heat pipe. Fiberglass insulation was placed over the evaporator section and the adiabatic section. The condenser section was exposed to allow for heat dissipation. K type thermocouples were placed on the surface of the evaporator, adiabatic and condenser sections of the heat pipe. The power supply to the electrical heater was controlled by a variable DC power source. The schematic of the set up can be seen in Fig. 1 and the actual set up can be seen in Fig. 2.

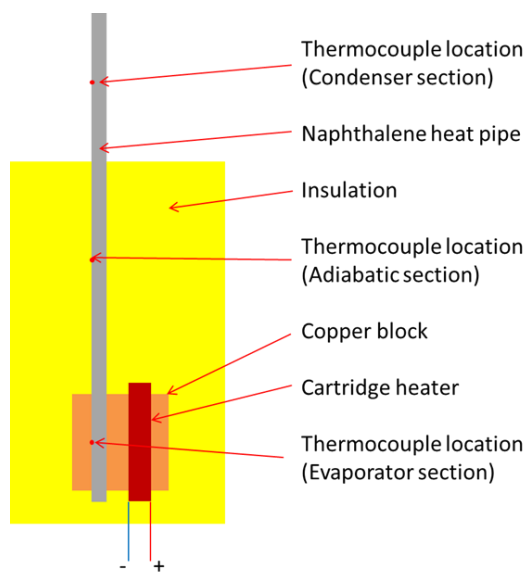
Plotted in Fig. 3 is the transient start up behaviour of the tested naphthalene heat pipe. Unlike typical water heat pipes, the naphthalene working fluid is solid when starting from room temperature. This means that the naphthalene heat pipe will not start transferring heat straight away whereas water heat pipes will. For this test, everything is initially

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at room temperature and then the electric heater is switched on to produce 50 W of heat. It can be seen that the evaporator temperature starts rising straight away. The temperature of the adiabatic section is constant for a period of time and then starts to rise. The condenser temperature starts to rise shortly after. During this initial period, the temperature difference over the three sections of the heat pipe is high, suggesting that the heat pipe is not fully operational. It was thought that when the heat is initially applied, the evaporator would rise in temperature from sensible heat transfer until it reaches the naphthalene melting point of 80.2 °C. During phase change from solid to liquid, the evaporator should remain constant temperature until it has completely melted due to latent heat transfer. After the operating temperature / adiabatic section temperature of the heat pipe surpasses 80.2 °C, the heat pipe would start operating as normal but this was not the case. It can be seen that the temperature of the adiabatic section reaches more than 400 °C before the three temperatures start to converge. It was observed that when the heat input was reduced, the temperature of the adiabatic section could drop as low as 200 °C before the three temperatures started to diverge and the heat pipe stopped operating.

**Table 1** Heat pipe specifications

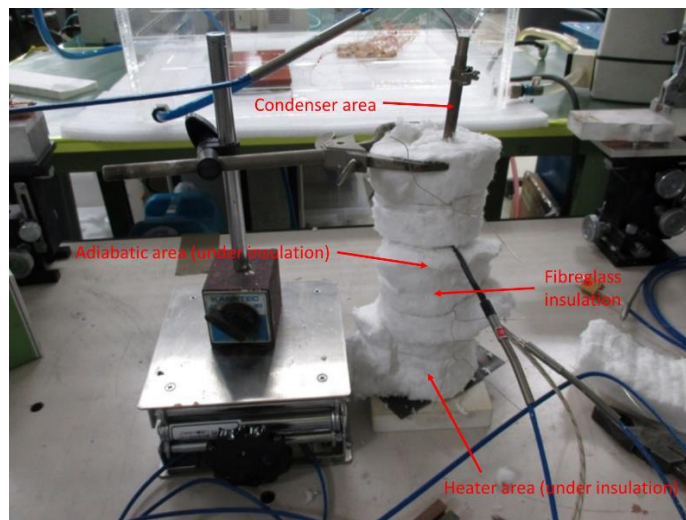
Diameter	8 mm
Length	278 mm
Working fluid	Naphthalene
Container material	Stainless steel, 0.5 mm wall thickness
Wick	Stainless steel mesh #400 X 2 layers (Note: #400 = 400 square openings across one linear inch of wick)
Fill ratio	20% (by volume)



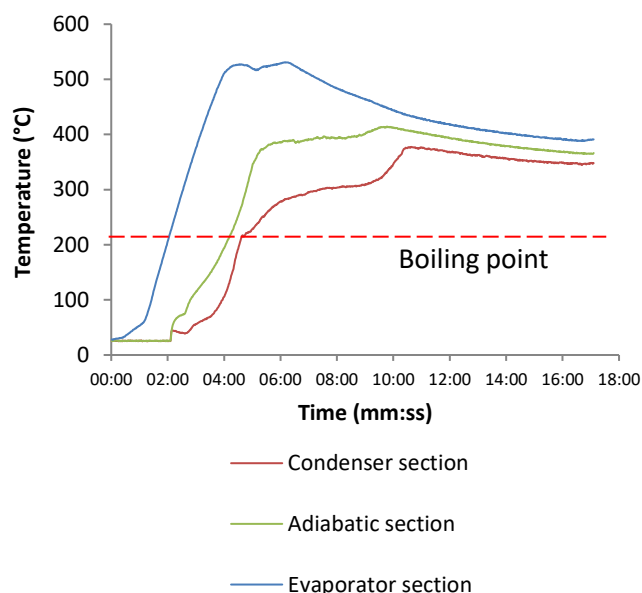
**Fig. 1** Schematic of the individual heat pipe test set up.

One possible reason for the heat pipe not working earlier is that the condenser may be blocked by solid naphthalene. There is no evidence of this because the condenser temperature surpassed the melting temperature of naphthalene and still did not work properly. A known problem is caused by the changing density of the naphthalene when changing phase from liquid to solid. After the heat pipe has stopped being used, the naphthalene will cool to room temperature and solidify. When the naphthalene solidifies, it will shrink. Density changes from 1.14 kg/m<sup>3</sup> as a solid at room temperature to 0.977 kg/m<sup>3</sup> as a liquid at the melting point. During the shrinking process the naphthalene may detach from the wall/wick. The gap between the wall/wick and the solid naphthalene will have a large thermal resistance. This may be why a large

temperature is required to initially melt the naphthalene. Fluid migration to the condenser section is another problem. When the naphthalene reaches the condenser section, it solidifies and does not return to the evaporator section resulting in dry out of the evaporator section. In this case the evaporator section temperature would keep rising.



**Fig. 2** The test set up for testing an individual heat pipe.



**Fig. 3** Transient start up behaviour of the naphthalene heat pipe.

It is known that liquids need an excess temperature over the saturation temperature to initiate boiling. For water, an excess temperature of 5 °C has been observed (Incropera *et al.*, 2011). For naphthalene, the excess temperature may be different. It is unlikely that the excess temperature required for naphthalene could explain why the temperature of the adiabatic section needs to reach 400 °C before the heat pipe starts to operate. There have been instances where a heat pipe requires a 'shock' to get it working. This can be done either thermally or mechanically (Li *et al.*, 1991). To shock the heat pipe thermally, a heating element is used to initiate the boiling. Once the boiling has started, the heater can be turned off and the heat pipe can operate at lower temperatures and temperature differences. Mechanical shocking involves vibrating the heat pipe to initiate boiling. Once boiling has commenced the vibrating can stop. A similar process can be seen when cooling water. Water in a bottle can be cooled below 0 °C and still not freeze. As soon as the bottle is moved, freezing starts to occur even if the temperature

rises to 0 °C. For this particular naphthalene heat pipe test, thermal shock is the likely method which initiated the heat pipe operation. This theory is supported by the fact that the heat pipe still operated well below the start-up adiabatic section temperature of approximately 400 °C after the start-up occurred.

Hysteresis could be another explanation as to why the heat pipe started to work at an operating temperature of 400 °C but could work as low as 200 °C. Hysteresis is the separation of the melting and boiling temperatures. Possibly in this case the high temperature was required to initiate the boiling but due to hysteresis the naphthalene would continue boiling until the operating temperature fell below 200 °C.

Using the same set up for the start-up tests, the thermal resistance of the heat pipe was measured and plotted against the temperature of the adiabatic section. This can be seen in Fig. 4. The thermal resistance was calculated using Eq. (1). The evaporator temperature, condenser temperature and heater power are required to calculate the thermal resistance. Due to the unique transient start-up behaviour of the naphthalene heat pipe, the thermal resistance at high adiabatic section temperatures was determined first, and then the power was reduced incrementally to find the thermal resistance at the lower adiabatic section temperatures. Starting from low power to high power would get completely different results because the heat pipe would not be in operation. All tests were conducted within the heat pipe's maximum rate of heat transfer so the results would not be skewed at high power input. It can be seen that below an adiabatic section temperature of approximately 200 °C, the thermal resistance rises significantly. This graph demonstrates that naphthalene heat pipes start to operate at a working temperature / adiabatic section temperature of approximately 200 °C. This compares well with minimum operating temperatures stated in literature. For example, it has been stated that naphthalene heat pipes start to operate at 250 °C (Kniess *et al.*, 2007; Kimura *et al.*, 1994; Mantelli *et al.*, 2010). The sources of uncertainty for thermal resistance measurements are temperatures (+/- 0.5 °C) and power input (+/- 1 W). These uncertainties are used for the error analysis of all figures throughout the paper.

$$R_{th} = \frac{T_{evap} - T_{cond}}{P_{heater}} \quad (1)$$

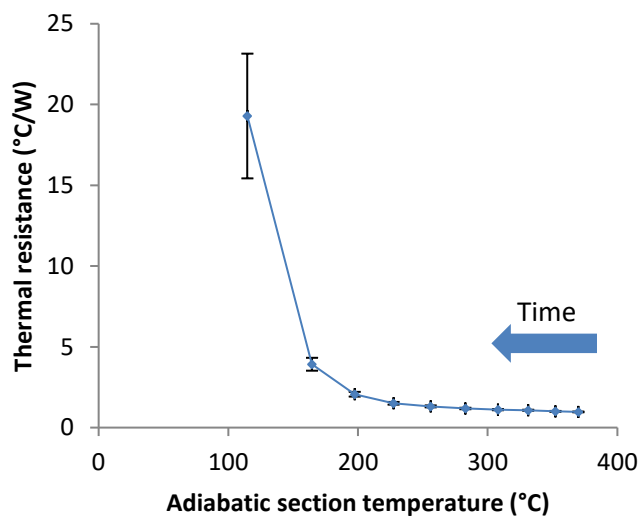


Fig. 4 Minimum operating temperature of the naphthalene heat pipe.

## 2.2 Revised test set up and data

An observation was made during testing that the naphthalene heat pipe does not operate when an aluminium heat sink is attached to the condenser section and exposed to ambient air / natural convection.

Therefore it was decided to test the heat pipe with the heat sink exposed to high temperature air. It was thought that having a higher condenser temperature could possibly increase the operating temperature of the heat pipe into its working range. To do this some hand held driers were used. The air from the driers was directed at the heat sink. The hot air supply from these driers was at a constant temperature. These driers are not to be confused with the heater. The heater is not temperature limited and will keep increasing its temperature until steady state conditions are met whereas the output air temperature of the driers is constant. The hot air from the driers was used for cooling, not heating. Two different driers were tested. The first drier tested was a domestic hair drier. The output air temperature of this drier was approximately 120 °C and maintained the condenser section at the same temperature. It was found that the heat pipe did not work under these conditions. The operating temperature of the heat pipe was still not high enough. Therefore it was decided to use a hand held industrial drier instead. The output air temperature of this drier was approximately 250 °C. It was found that the heat pipe did start working under these conditions therefore further performance tests were conducted using the industrial drier.

The naphthalene heat pipe was then tested to find its maximum rate of heat transfer and its thermal resistance. Equation (1) was used to determine the thermal resistance. The schematic of the set up can be seen in Fig. 5 and the actual test set up can be seen in Fig. 6. Attached to the heat pipe on the evaporator section was a high temperature rated heater. The heater was attached to a variable voltage supply. The evaporator section and adiabatic section of the heat pipe was wrapped in fibreglass wool for insulation. Attached to the condenser section was the same aluminium heat sink used previously. In this case the heat sink was not exposed to ambient air but was subjected to hot air from the industrial drier. This drier kept the condenser section at a relatively constant temperature of approximately 250 °C. Using the variable voltage supply, the power inputs to the heater were made in increments of 10 W. Similarly to previous tests, the heat pipe did not start working until it was at a very high temperature (approximately 400 °C). If this test was undertaken in the traditional method of starting at a low heater power input and increasing heater power input incrementally, the results would not be reflective of the true performance of the heat pipe. Therefore testing started at 60 W heater power input with incremental reductions of 10 W. The plotted curve shown in Fig. 7 suggests that this heat pipe has a maximum rate of heat transfer of approximately 40 W and a thermal resistance of approximately 1 °C/W under these conditions. Equation (2) determines the surface heat flux of the heat pipe. With an evaporator length of 50 mm and a heat pipe diameter of 8 mm, the surface heat flux when transferring the maximum of 40 W was 3.2 W/cm<sup>2</sup>.

$$\dot{q} = \frac{\dot{Q}}{A_{evap}} \quad (2)$$

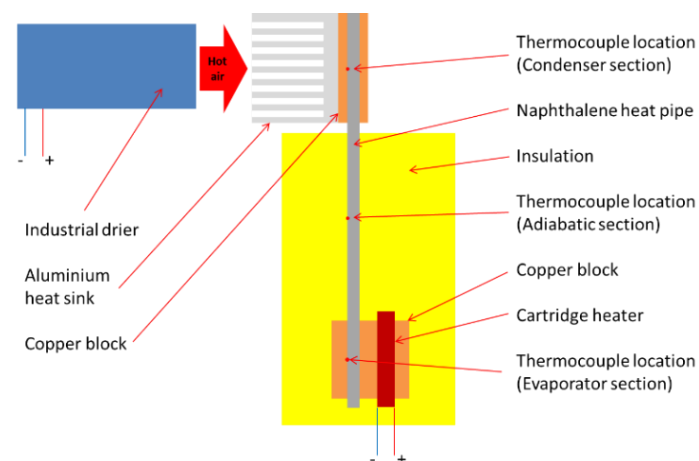


Fig. 5 Schematic of the test set up when using the industrial drier.



To put the performance of the naphthalene heat pipes into perspective, a copper/water heat pipe of the same size would have a thermal resistance approaching 0.2 °C/W and a maximum rate of heat transfer closer to 80 W (at a lower operating temperature). This is to be expected because the properties of naphthalene are not as favourable as water for use in heat pipes. One method to compare working fluids is to look at the figure of merit of the working fluids. Equation (3) is used to calculate the figure of merit. The phase change enthalpy, surface tension, liquid density and liquid viscosity are required to calculate the merit number. The merit number is used to compare the potential performance of different working fluids at different temperatures with higher numbers being better. Figure 8 shows the variation of the figure of merit with changes in temperature (Anderson, 2005). Despite being able to handle higher temperatures than water, the peak figure of merit of water is much higher than the peak figure of merit for naphthalene. The reason for this is over a wide temperature range, naphthalene has significantly lower enthalpy of latent heat, lower surface tension and higher viscosity. The density of naphthalene is higher (Liu *et al.*, 2015) but not enough to offset the other drawbacks.

$$M = \frac{h_{fg}\sigma\rho_l}{\mu_l} \quad (3)$$

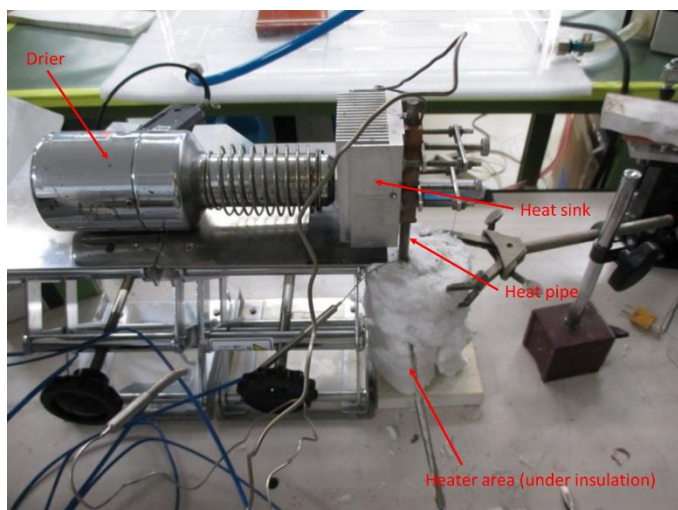


Fig. 6 Test set up when using the industrial drier.

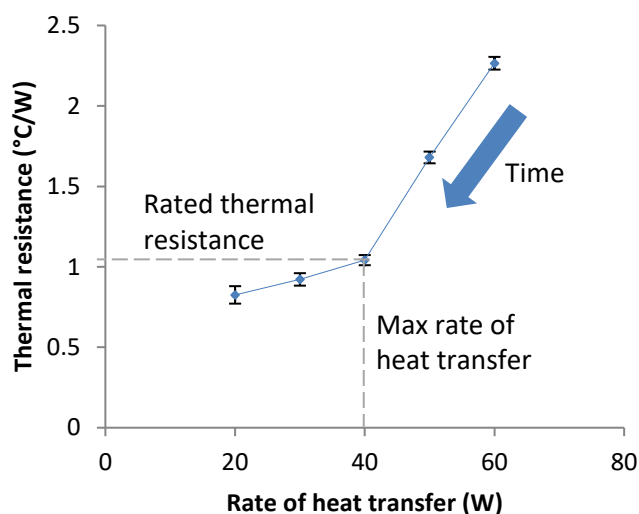


Fig. 7 Thermal resistance and maximum rate of heat transfer.

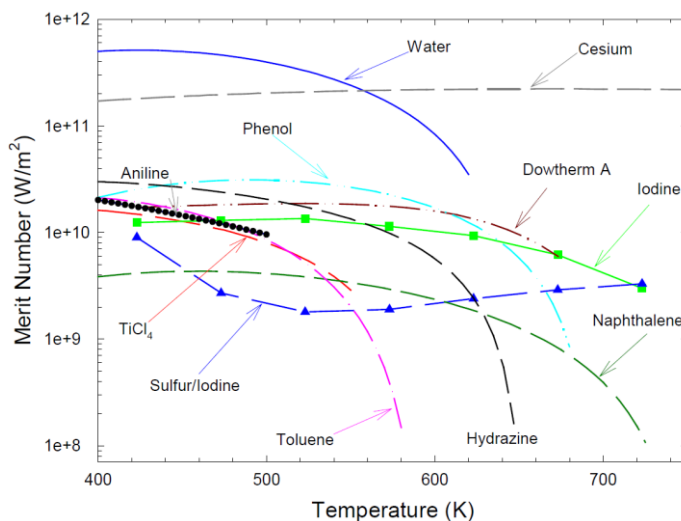


Fig. 8 Working fluid figure of merit variation (Liu *et al.*, 2015).

### 3. TESTING A NAPHTHALENE HEAT PIPE HEAT EXCHANGER

#### 3.1 Test set up and initial observations

After testing an individual heat pipe, a heat pipe heat exchanger was created using eight identical naphthalene heat pipes. A schematic of the system can be seen in Fig. 9. Four rows of two parallel heat pipes were used. The system consists of two ducts, the cool air duct and the exhaust duct. The high temperature gases were supplied from a car exhaust pipe. The car used was a 2007 Holden Commodore with a 3.0 L V6 gasoline engine. These hot exhaust gases would go through the exhaust duct with some of the heat extracted by the finned evaporator section of the heat pipes. This heat would transfer to the finned condenser section of the heat pipes inside the cool air duct. The cooler air flowing over the fins inside the cool air duct would extract the heat and this now warmer air is expelled to the atmosphere. The system is designed to be a counter flow heat exchanger. The final lab set up can be seen in Fig. 10. In this figure, the system is in a horizontal orientation. The system is entirely wrapped in fibreglass for insulation. Fibreglass is used because other insulation materials can't handle the temperatures present. Attached to the exhaust duct is a flexible metal pipe which is attached to the car exhaust pipe outlet. At the other end of the exhaust duct an extractor is attached which directs the exhaust gases out of the building. Attached to the cool air duct is an industrial drier. The previous tests conducted on the individual heat pipe show that relatively warm air was required at the condenser section of the heat pipe. The industrial drier would provide that relatively warm air. K type thermocouples were placed inside both ducts to plot the temperature profile of the ducts from inlet to outlet.

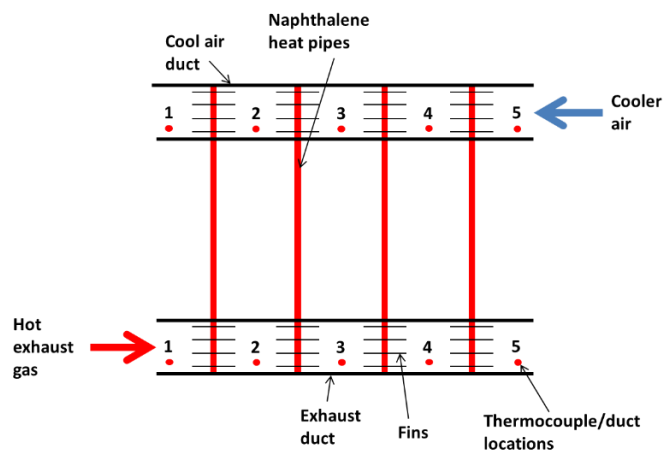
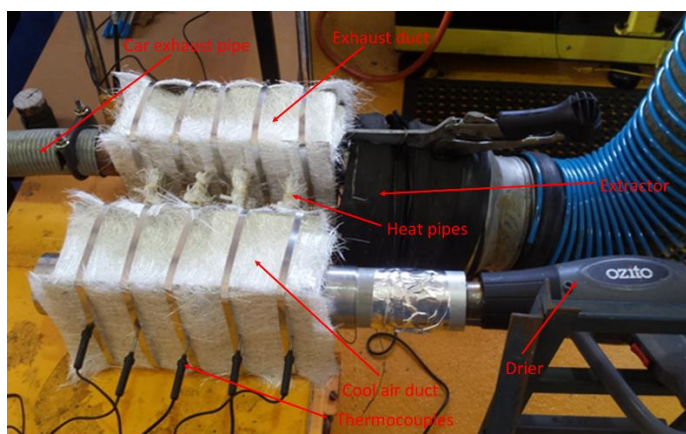


Fig. 9 Schematic of the naphthalene heat pipe heat exchanger.



**Fig. 10** Test set up of the naphthalene heat pipe heat exchanger.

During testing, the engine was run at 4,000 RPM under no load. This resulted in the input temperature of the exhaust duct being approximately 350 °C. Initially, to confirm the findings from the individual heat pipe testing, the industrial drier was not used straight away but a 12 V fan was used instead. This fan provided a flow of ambient temperature air through the cool air duct. In this situation the system was in a horizontal orientation. Similarly to the individual tests, it was found that the heat pipes did not operate in this situation because the inlet and outlet temperature of the cool air duct were the same. To see if the heat pipes worked with no air flow in the cool air duct, the fan was turned off. The air temperature at the centre of the duct was monitored to see if the heat pipes are operating. It was found that the heat pipes did not operate under these conditions as the cool air duct remained at ambient temperature. The orientation was then changed so the heat pipes are in bottom heat mode and the same two tests were conducted, with a fan and without the fan. Bottom heat mode means the heat pipes are in a vertical orientation with the evaporator below the condenser. Again, when the fan was used, the temperature of the inlet and outlet were the same so the heat pipes were not working. It was observed that when the fan was turned off, the temperature at the centre of the duct increased to approximately 200 °C. This indicates that the heat pipes were working in this situation unlike the similar test conducted for the individual heat pipe.

### 3.2 Testing with an attached drier

Leaving the system in a bottom heat mode orientation, the industrial drier was attached to the cool air duct. The output air temperature of the industrial drier was approximately 79.5 °C. During this test it was found that the outlet temperature of the cool air duct reached approximately 259.5 °C which indicates that the heat pipes were working. Similarly to the individual testing, the heat pipes did not operate straight away as the cool air duct outlet did not reach 259.5 °C quickly. At least 15 minutes was required for this temperature to be reached.

It was observed that the cool air duct input temperature was relatively low but the heat pipes still worked. During the individual testing, when a hair drier with an output temperature of 120 °C was used, the heat pipe did not work. The industrial drier needed to reach 250 °C for the heat pipes to start operating but in this case only 79.5 °C was required. One possible reason could be the very low air velocity of the industrial drier. The hair drier used had a much higher air velocity therefore the heat transfer co-efficient between the fins/air would be higher, resulting in a lower fin/air thermal resistance. Consequently temperature difference between the air and the condenser section would be quite low. This means the condenser temperature would be very close to the air temperature. When the industrial drier is used in this case, the air velocity is very small therefore the thermal resistance and consequently temperature difference between the air and the condenser section would be quite high. This means the condenser temperature would be higher than the air temperature and possibly high enough to operate. Evidence supporting this theory would be the fact that the heat pipes did work when

no fan was used and the duct was exposed to even cooler ambient air. Another possible reason for the heat pipes working with a lower than expected cool air duct input temperature would be the fact that vibration was present during testing. As stated earlier, it is known that some heat pipes need to be triggered into operation and one of these triggers was mechanical shock. The vibrations coming from the car exhaust could have helped initiate the boiling inside the heat pipes and acted as the mechanical shock trigger.

Repeat tests were conducted under these conditions to check the consistency of the results. The average values and measured variation of the duct inlet and outlet temperature can be seen in Table 2. It can be seen that the parameters are relatively consistent. Unfortunately, due to the much higher mass flow rate in the exhaust duct, the temperature difference over the exhaust duct is quite small and the variation is a significant percentage of the temperature difference.

**Table 2** Repeatability of results

	Average value	Variation
Cool air duct inlet temperature (°C)	79.5	+/- 0.5
Cool air duct outlet temperature (°C)	259.5	+/- 9.5
Exhaust duct inlet temperature (°C)	346.5	+/- 9.5
Exhaust duct outlet temperature (°C)	326	+/- 12

### 3.3 Orientation testing

Testing was undertaken to determine the effect of orientation on the performance of the heat pipe heat exchanger. With the industrial drier attached, the heat exchanger was tested in a top heat mode, horizontal heat mode and bottom heat mode orientation. Horizontal and bottom heat mode are the same as previously described. Top heat mode is when the heat pipes are in a vertical orientation with the evaporator above the condenser. The temperature throughout the ducts was measured and compared. The plots for these three orientations can be seen in Fig. 11. The duct locations in Fig. 11 refer to the thermocouple/duct locations in Fig. 9. It can be seen that the temperature profiles follow what is expected for a counter flow heat exchanger. In all three cases the exhaust duct temperature does not drop significantly because the mass flow rate in the exhaust duct is much higher than the cool air duct. There is an obvious difference in the cool air duct temperature profiles from the three different orientations. The temperature difference over the cool air duct in top heat mode is 16°C, in horizontal heat mode is 83 °C and in bottom heat mode is 189 °C. This indicates that there is a large change in performance depending on the orientation. In this case, bottom heat mode is the best, followed by the horizontal heat mode and top heat mode. This is to be expected because the capillary limit of the heat pipes change with orientation. The capillary limit occurs when the capillary pressure of the wick equals the total pressure drop inside the heat pipe. The total pressure drop is the sum of the liquid pressure drop, the vapour pressure drop and the gravitational pressure drop. The gravitational pressure drop is a function of the cosine of orientation angle ( $\phi$ ) therefore can reduce the total pressure drop in bottom heat mode or increase the total pressure drop in top heat mode. Reducing the total pressure drop increases the capillary limit. The higher rate of heat transfer results in a higher outlet temperature of the cool air duct in this case. Assuming the vapour pressure loss is negligible, the capillary limit can be derived from the pressure drop equations as shown in Eqs. (4a-4f) where  $\phi=0^\circ$  for top heat mode,  $\phi=90^\circ$  for horizontal heat mode and  $\phi=180^\circ$  for bottom heat mode. The capillary pressure, liquid pressure drop and gravitational pressure drop/gain are considered. To calculate the capillary limit, the merit number, wick cross sectional area, wick permeability, heat pipe effective length, pore size, liquid density, surface tension and orientation angle must be known.

$$\Delta P_c \geq \Delta P_l + \Delta P_g \quad (4a)$$

$$\frac{2\sigma}{r_p} \geq \frac{l_{eff}\dot{Q}\mu_i}{A_w\kappa h_{fg}\rho_l} + \rho_l g l_{eff} \cos(\phi) \quad (4b)$$

$$\frac{2\sigma}{r_p} - \rho_l g l_{eff} \cos(\phi) \geq \frac{l_{eff}\dot{Q}\mu_i}{A_w\kappa h_{fg}\rho_l} \quad (4c)$$

$$\dot{Q} \leq \frac{A_w\kappa h_{fg}\rho_l}{l_{eff}\mu_i} \left( \frac{2\sigma}{r_p} - \rho_l g l_{eff} \cos(\phi) \right) \quad (4d)$$

$$\dot{Q} \leq \frac{A_w\kappa h_{fg}\rho_l\sigma}{l_{eff}\mu_i} \left( \frac{2}{r_p} - \frac{\rho_l g l_{eff} \cos(\phi)}{\sigma} \right) \quad (4e)$$

$$\dot{Q}_{cap} \leq M \frac{A_w\kappa}{l_{eff}} \left( \frac{2}{r_p} - \frac{\rho_l g l_{eff} \cos(\phi)}{\sigma} \right) \quad (4f)$$

The almost zero heat transfer in top heat mode can be explained by comparing the capillary pressure in Eqs. (5a-5c) and the gravitational pressure loss as shown in Eqs. (6a-6c). In this case the gravitational pressure loss was higher than the capillary pressure therefore the heat pipe will not work.

$$\Delta P_c = \frac{2\sigma}{r_p} \quad (5a)$$

$$\Delta P_c = \frac{2 \times 0.0187}{19 \times 10^{-6}} \quad (5b)$$

$$\Delta P_c = 1968 \text{ Pa} \quad (5c)$$

$$\Delta P_g = \rho_l g l_{eff} \cos(\phi) \quad (6a)$$

$$\Delta P_g = 1175 \times 9.8 \times (198 \times 10^{-3}) \cos(0) \quad (6b)$$

$$\Delta P_g = 2280 \text{ Pa} \quad (6c)$$

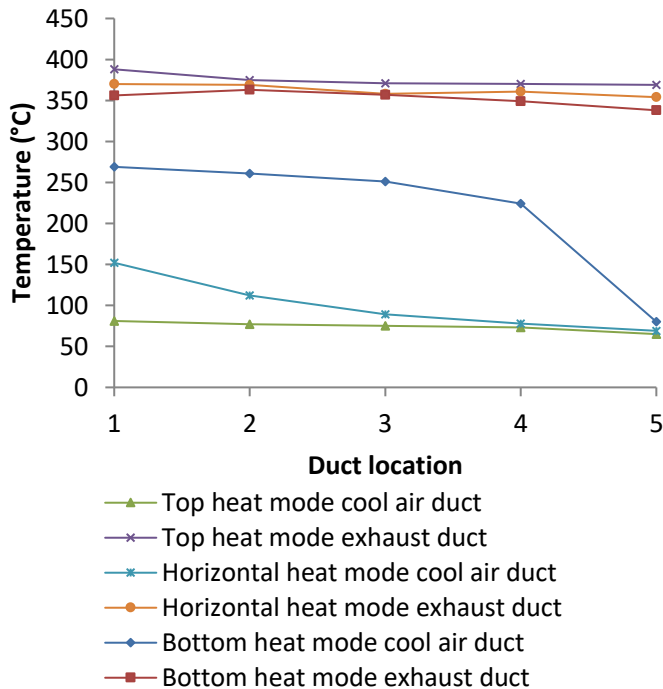


Fig. 11 Exhaust and cool air duct temperatures in 3 orientations.

The effectiveness of the naphthalene heat pipe heat exchanger can be compared for the three different orientations. The effectiveness gives an indication of the actual rate of heat transfer as a proportion of the maximum theoretical rate of heat transfer for the heat exchanger under the current conditions. Due to the much smaller mass flow rate in the cool air duct, the effectiveness in this case is a function of the temperature change in the cool air duct. The effectiveness can be calculated using Eq. (7). The cold duct outlet, cold duct inlet and hot duct inlet temperatures are required to calculate heat exchanger effectiveness. This information is displayed in Fig. 12. It can be seen that when in bottom heat mode, the heat exchanger works at its best with an effectiveness of 68%. The effectiveness is more than halved to 28% when changed to a horizontal heat mode. The top heat mode has a very low effectiveness of 5%. This information suggests that naphthalene heat pipe performance is sensitive to orientation as all heat pipes are.

$$E = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} \quad (7)$$

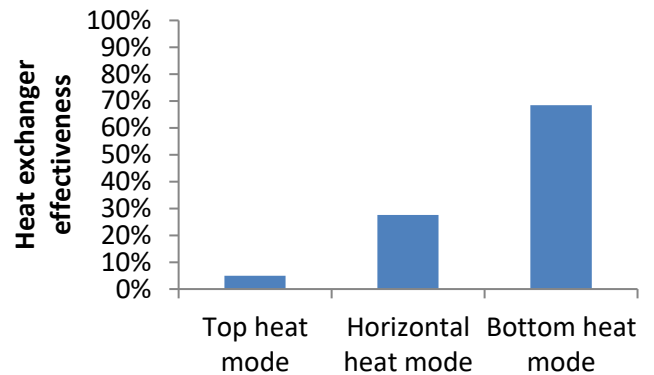


Fig. 12 Heat exchanger effectiveness for the 3 different orientations.

#### 4. CONCLUSIONS

Testing of an individual naphthalene heat pipe and a naphthalene heat pipe heat exchanger were undertaken to determine the characteristics and performance of this type of heat pipe. During the individual heat pipe testing, it was found that these heat pipes have an unusual transient start up behaviour as they need to be heated up to approximately 400 °C to initiate operation but once operating they will continue to do so at lower temperatures. It was determined that the minimum operating temperature of these heat pipes is approximately 200 °C. If ambient temperature air is used on the heat sink for the condenser, the operating temperature of the heat pipe will drop below its minimum operating temperature and will fail to operate. Relatively warm air (approximately 250 °C in this case) must be used for cooling of the heat sink. The 8 mm diameter, 278 mm long naphthalene heat pipe tested was found to have a thermal resistance of approximately 1 °C/W and a maximum rate of heat transfer of 40 W. Tests of the naphthalene heat pipe heat exchanger found that the performance of the heat pipes is very sensitive to their orientation. The heat exchanger effectiveness for the bottom, horizontal and top heat modes was 68%, 28% and 5% respectively. Results from the heat exchanger test also show that a flow of ambient temperature air can't be used for cooling naphthalene heat pipes. Potential future work would be to investigate the viability of using naphthalene heat pipes as a temperature regulator by making use of the fact that these heat pipes don't work below 200 °C but do above this temperature.



## NOMENCLATURE

$A_w$	Wick cross section area (m <sup>2</sup> )
$A_{evap}$	Heat pipe evaporator section surface area (cm <sup>2</sup> )
$E$	Heat exchanger effectiveness (%)
$g$	Acceleration due to gravity (m/s <sup>2</sup> )
$h_{fg}$	Enthalpy of latent heat (J/kg)
$L_{eff}$	Heat pipe effective length (m)
$M$	Working fluid figure of merit (W/m <sup>2</sup> )
$\Delta P_c$	Capillary pressure (Pa)
$\Delta P_g$	Gravitational pressure drop (Pa)
$\Delta P_l$	Liquid pressure drop (Pa)
$P_{heater}$	Heater power (W)
$\dot{q}$	Surface heat flux (W/cm <sup>2</sup> )
$\dot{Q}$	Rate of heat transfer (W)
$\dot{Q}_{cap}$	Capillary limit (W)
$r_p$	Wick pore radius (m)
$R_{th}$	Heat pipe thermal resistance (°C/W)
$T_{cond}$	Heat pipe condenser wall temperature (°C)
$T_{c,i}$	Heat exchanger cold gas inlet temperature (°C)
$T_{c,o}$	Heat exchanger cold gas outlet temperature (°C)
$T_{evap}$	Heat pipe evaporator wall temperature (°C)
$T_{h,i}$	Heat exchanger hot gas inlet temperature (°C)
$T_{h,o}$	Heat exchanger hot gas outlet temperature (°C)
<i>Greek symbols</i>	
$\kappa$	Wick permeability (m <sup>2</sup> )
$\mu_l$	Liquid viscosity (Pa.s)
$\rho_l$	Liquid density (kg/m <sup>3</sup> )
$\sigma$	Surface tension (N/m)
$\phi$	Orientation angle (°)

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