

Evaluation of Heat Pipe Heat Exchanger for Automotive Applications

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ABSTRACT

A heat pipe is a self-operating device which is capable of transferring large amounts of heat with a minimum temperature differences between the hot end (evaporator) and the cold end (condenser). However, a limited number of research work or analysis [1,2,3,4,5,6,7,8,9] has been reported in automotive industry on the applications of heat pipes in power train cooling. The advantage of a heat pipe heat exchanger is the possibility to use a more compact and lighter radiator. In addition, the proposed radiator is expected to be more robust as it is less sensitive to variations in ambient temperatures. In this paper, a proposed design for an automotive heat exchanger is investigated. The proposed design is evaluated through thermal simulation of heat pipes using various design parameters. The analysis addresses the ability of the heat exchanger to maintain engine coolant temperature at acceptable limits under different loading conditions.

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INTRODUCTION

Heat pipes are simple and passive devices that transfer heat from the heat source (evaporator) to a heat sink (condenser) over a very small temperature gradient. Heat pipes are capable of transferring large amounts of heat that can be several hundred times the amount of heat transferred by conduction through a copper rod of the same dimensions [10]. The heat pipe itself is a vacuum-tight tube that is evacuated and partially filled with a small amount of water or other working fluid. At the evaporator end, the fluid is vaporized creating a pressure gradient in the pipe. This forces the vapor to flow along the pipe to the condenser section where it condenses, giving up its latent heat of vaporization. The working fluid is then returned to the evaporator by capillary forces developed in the heat pipe's porous wick structure, or by gravity (Figure 1). In this work, heat pipes are proposed as a means of transferring heat from the engine coolant to the condenser section where heat is transferred to the cooling air. Heat is picked up by the heat pipe working fluid at the evaporator end which is immersed in the engine coolant fluid. The vapor then travels along the heat pipe and will condense at the condenser end as long as the air temperature is lower than the boiling point of the working fluid. Ambient air can be pulled to cool the condenser end through the use of a traditional cooling fan. As long as the ambient air temperature is lower than the boiling point of the working fluid, the vapor phase of the working fluid will condense, giving off its latent heat of condensation which will be picked up by the cooling air, as long as enough airflow is provided. Therefore, a minimum temperature difference between the ambient air temperature and the working fluid is required. Therefore, the heat pipe heat exchanger represents a robust system for engine cooling since its performance is insensitive to variations in the ambient air temperature. In traditional heat exchangers, the capacity of the cooling system (for a specific

heat exchanger and fan size) depends on the ambient air temperature. Therefore, for traditional cooling systems, the higher the ambient air temperature, the higher the engine coolant.

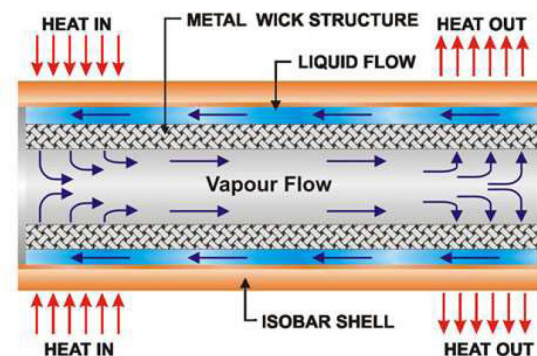


Figure 1. Schematic diagram for a typical heat pipe [11]

In hot climate regions, it is usually very challenging to maintain engine coolant within the acceptable limits unless a large size radiator and large fan are provided. Alternatively, a heat pipe heat exchanger can provide another means for engine cooling that can perform well under wider ranges of ambient temperatures. At the evaporator end, the engine coolant will transfer heat to the heat pipe walls by convection which will be transferred by conduction through the heat pipe walls. Inside the heat pipe, heat is transferred by convection between the pipe inner wall and the working fluid, at the fluid pipe wall interface. The working fluid will absorb the heat and will change from liquid phase to vapor phase. The vapor will travel to the other end (condenser) where it will condense as long as the condenser temperature is lower than the working fluid boiling point. The pressure inside the heat pipe can be adjusted so the working fluid can

boil or condense at lower temperatures. The proposed system may consist of a number of heat pipes which can be arranged vertically or horizontally. In a vertically arranged setting, advantage can be made of gravity assistance to bring the condensed fluid back to the evaporator end. Therefore, the radiator will consist of two main sections. At the lower section, the hot engine coolant enters at the highest temperature and gives off heat to the evaporator end of a number of heat pipes. The surface area at the evaporator end has to be sufficient to transfer the required amount of heat rejection. At the top section, air is being used to remove heat from the condenser end of the heat pipe arrangement. Enough surface area and air flow must be provided for the system to work successfully. The size of the heat exchanger depends on the number of heat pipes, heat pipe dimensions, and required fins. For the horizontally arranged system, a wick structure design must be used to bring the condensed fluid back to the evaporator section.

Working Fluids

A number of working fluids can be used for heat pipes. However, the selection of a working fluid for an automotive heat exchanger should consider the vehicle thermal environment. This includes the maximum engine coolant temperatures and maximum ambient air temperatures. The working fluid should evaporate at or below the engine coolant temperature. Otherwise, the heat pipe will fail to operate and remove the required heat. The vapor should condense at the condenser temperature which is subjected to cooling by ambient air. Therefore, the working fluid should condense at or below the maximum ambient air temperature. If the working fluid does not condense at the ambient air temperature, there will be no circulation of fluid through the heat pipe which can result in evaporator dry-out and failure of the heat pipe to function as required. During cold weather, the working fluid should not freeze up in order to maintain the structural durability of the heat pipe design. Table 1 lists some of the commonly used working fluids for heat pipes, their melting and boiling points at atmospheric pressure. For this study, ethanol has been selected as the working fluid based on its boiling point of 78°C and its freezing point of -112°C. This range assures that the fluid will always boil when the engine coolant temperature exceeds 78°C. For extreme ambient air temperatures such as 50°C to 55°C, the ethanol vapor will still condense as long as the ambient air temperature is lower than 78°C. The low freezing point of -112°C assures that no freezing issues will occur during extreme low ambient temperatures which may range from -20°C to -30°C for example. Therefore, the ethanol working fluid will be always available to transfer the required heat rejection under all operating conditions. However, engine coolant is allowed to flow to the radiator at relatively high temperatures which range from 90°C to 105°C for engine coolant. Ethanol has been applied as the working fluid for several heat pipe heat exchangers for various applications and the performance has been compared against various working fluids [13,14,15,16]. For traditional radiators, the under hood air temperature may exceed 100°C which can affect under-hood components durability, and will drive the use of additional heat shields or high temperature materials. Using the proposed design, the radiator out temperature will not exceed 78°C. However, additional airflow will be required to remove the same amount of heat transferred through a traditional radiator. The cost of additional airflow and its impact on overall fuel economy

should be compared to the weight reduction due to heat shield elimination and the added benefit of long term thermal durability of under hood components. The lower under hood air temperature will also benefit fuel economy as it keeps the air intake temperature at much lower temperature compared to traditional radiator.

Table 1. Heat pipe working fluid melting and boiling points [12]

Working Fluid	Melting Point, °C at 1 atm	Boiling Point, °C at 1 atm
Helium	-271	-261
Nitrogen	-210	-196
Ammonia	-78	-33
Pentane	-130	28
Acetone	-95	57
Methanol	-98	64
Flutec PP2	-50	76
Ethanol	-112	78
Heptane	-90	98
Water	0	100
Toluene	-95	110
Flutec PP9	-70	160
Thermex	12	257
Mercury	-39	361
Caesium	29	670
Potassium	62	774
Sodium	98	892
Lithium	179	1340
Silver	960	2212

The Effect of Capillary Pressure

A wick structure is usually required to bring the condensed fluid from the condenser section to the evaporator section, and also to retain the condensed working fluid within the wick structure. In case of a gravity assisted heat pipe the requirements for capillary pressure are not significant. For a horizontal heat pipe or a heat pipe which is working against gravity, the requirement for capillary pressure is much greater. Capillary pressure depends on the number of layers and wire diameter for the wicking material as well as the wetting properties between the working fluid and the wick material.

Heat Pipe Design and Operating Limits

For continuous operation of the heat pipe heat exchanger, the working fluid must be continuously circulated between the hot end (evaporator) and the cold end (condenser). Therefore, heat pipe design must meet several design and operating limits. These limits are listed as follows:

1. Capillary limit
2. Boiling limit
3. Viscous limit
4. Entrainment limit
5. Sonic limit

Capillary Limit

Capillary limit can be reached when the capillary pressure is not sufficient enough to bring the condensed liquid to the evaporator section. This will lead to dry-out and failure of the heat pipe. For a heat pipe to function properly, the capillary pressure must be greater than or equal to the sum of the pressure drops due to inertial, viscous, and hydrostatic forces, as well as, pressure gradients.

This condition can be expressed as

$$\Delta P_c \geq \Delta P_l + \Delta P_v + \Delta P_g \quad (1)$$

where ΔP_c is the capillary pressure drop, ΔP_l is the liquid phase pressure drop, ΔP_v is the vapor phase pressure drop and ΔP_g is the pressure drop due to gravity effect. If this condition is not met, the wick will dry out in the evaporator region and the heat pipe will not operate. For a working heat pipe, the equation can be written as:

$$(\Delta P_c)_m = \int_{L_{eff}} \frac{\partial P_v}{\partial x} dx + \int_{L_{eff}} \frac{\partial P_l}{\partial x} dx + \Delta P_{ph,e} + \Delta P_{ph,c} + \Delta P_n + \Delta P_a \quad (2)$$

where

$(\Delta P_c)_m$ = Maximum capillary pressure difference generated within capillary wicking structure between wet and dry point

$\partial P_v / \partial x$ = Sum of inertial and viscous pressure drops occurring in vapor phase.

$\partial P_l / \partial x$ = Sum of inertial and viscous pressure drops occurring in liquid phase.

$\Delta P_{ph,e}$ = Pressure gradient across phase transition in evaporator

$\Delta P_{ph,c}$ = Pressure gradient across phase transition in condenser

ΔP_n = Normal hydrostatic pressure drop

ΔP_a = Axial hydrostatic pressure drop.

Equation (3) for maximum flow rate ($\dot{m}_{max}$) can be valid under the following assumptions:

1. the liquid properties do not vary along the pipe
2. the wick is uniform along the pipe
3. the pressure drop due to vapor flow can be neglected

$$\dot{m}_{max} = \left[\frac{\rho_l \sigma_l}{\mu_l} \right] \left[\frac{KA}{l} \right] \left[\frac{2}{r_e} - \frac{\rho_l g l}{\sigma_l} \sin(\phi) \right] \quad (3)$$

where ρ_l is the density of the liquid working fluid, σ_l is the surface tension, μ_l is the viscosity of the liquid working fluid, K is the wick permeability, A is the wick cross sectional area, l is the heat pipe

length, r_e is the effective pore radius and ϕ is the angle made by the heat pipe with the horizontal (ϕ is positive when the condenser is lower than the evaporator).

The corresponding maximum allowed heat flux can be determined from equation (4) as follows:

$$\dot{Q} = \dot{m}_{max} L = \left[\frac{\rho_l \sigma_l L}{\mu_l} \right] \left[\frac{KA}{l} \right] \left[\frac{2}{r_e} - \frac{\rho_l g l}{\sigma_l} \sin(\phi) \right] \quad (4)$$

where L is the enthalpy of vaporization or latent heat.

Boiling Limit

During conditions of high heat flux, the temperature of the fluid within the wick structure may be too high that the fluid will start to boil. Formation of bubbles or nucleates within the wick will hinder the flow of condensed liquid from the condenser to the evaporator. It is therefore desirable to reduce the chance of nucleation. The degree of superheat above which nucleate boiling may occur, was analyzed by Hsu [17].

Entrainment Limit

The liquid phase of the working fluid is retained inside the wick structure during continuous operation of the heat pipe. However, if the shear forces of the vapor are too high, the vapor may remove the liquid from the wick structure which will limit the amount of liquid reaching the evaporator. Therefore the entrainment limit depends on the surface tension between the working fluid and the wick structure. The higher the surface tension, the less likely it is that the liquid droplets will be removed by the vapor.

Sonic Limit

At the sonic limit, the vapor velocity may reach sonic values. The sonic limit is due to the fact that at low vapor densities, the corresponding mass flow rate in the heat pipe may result in very high vapor velocities, and the occurrence of choked flow in the vapor phase may be possible. It is therefore desirable to have maximum velocity below sonic. The corresponding maximum heat flux can be estimated from the following equation:

$$\dot{q}_s = \rho_v L \sqrt{\frac{\gamma_v R T_v}{2(\gamma_v + 1)}} \quad (5)$$

Viscous Limit

At the viscous limit, the vapor pressure at the evaporator is not enough to overcome the viscous forces. This condition usually occurs at the low ranges of temperature.