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# Designing a Heat Pipe to Improve the Exhaust Emissions from Petrol Engines

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## ABSTRACT
*The national engineering Laboratory and the Shell research Laboratory have co-operated in applying the heat pipe to the problem of exhaust emission from petrol engine. In this paper, a complete design of heat pipe is carried out, taking into account the necessary criteria to decide various geometrical parameters. The design has been carried out using basic formulas in thermodynamics, heat transfer and physics. The results of this design have been checked for various practical limits.*

**KEY WORDS:** heat pip, working fluid, thermal conductance, air to fuel ratio, mixture, fuel.

## INTRODUCTION
It is known that the carbon monoxide CO, un-burnt hydrocarbons $H_xC_y$ and oxides of Nitrogen $NO_x$ content of the exhaust will vary with air to fuel ratio as shown in Figure 1, in a conventional car engine the maximum efficiency is achieved at 15:1 and maximum power is obtained at 12:1. It's known that as the air fuel ratio increases, the CO content decreases and $H_xCy$, $N_{OX}$ go through a minimum and maximum respectively. A considerable important in both CO and $NO_x$ content could be achivied by selecting a very weak mixture, but this not possible in a standard engine carburetor system due to the ignition difficulty, because the fuel is not fully vaporized, and because the fuel is not distributed equally between the cylinders and the vapor content is not as high as it should be due to the pressure of liquid fuel.

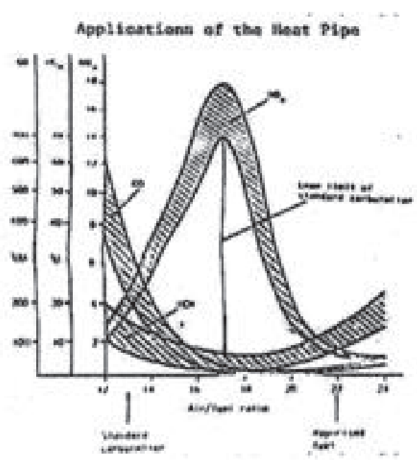


Figure 1. The Vapipe-typical variation of emissions with air/fuel ratio.

This problem could be solved by designing a heat pipe that can transfer a certain quantities of heat from the exhaust to the induction manifold at the carburetor outlet as shown in Figure2. Under this condition a mixture as lean as 22:1 will ignite without difficulty.

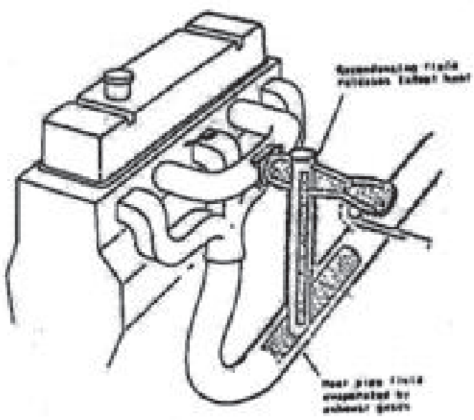

Figure 2. The Vapipe installation

The heat pipe is a device of a very high thermal conductance and capable of transferring large quantities of heat through relatively small cross section area with very small temperature difference. The heat pipe is a closed tube or chamber of different shapes, whose inner surfaces are lined with a porous capillary wick and the wick, is saturated with liquid phase of working fluid. The remaining volume of the tube contains the vapor phase. In the heat pipe the heat applied at the evaporator by an external source vaporizes the working fluid in that section. The resulting difference in pressure drives vapor from the evaporator to the condenser, which release the latent heat of evaporation to heat sink in that section of pipe. The main regions of the heat pipe as shown in Figure 3 are evaporation section, condensers section and adiabatic section.

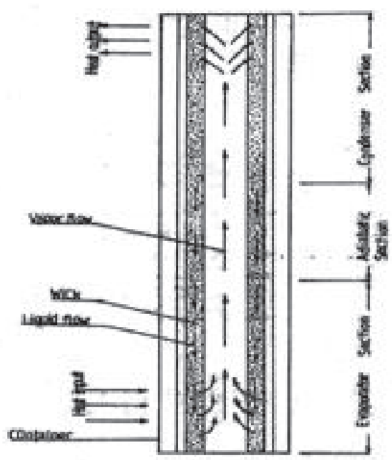

Figure 3. Components and principle of operation of heat pipe

Unlike, solid conduction, heat pipe characteristics are not dependent only upon the size, shape, and material but also upon construction, working fluid and heat transfer rate required.

The heat pipe consists of the following three basic components:

1. The wick structure
2. The fluid section
3. The container

## HEAT PIPE THEORY

During steady state operation of heat pipe, the working fluid in the vapor flows continuously from the evaporator section to the condenser section, and it returns to the evaporator in the liquid phase. In order for the heat pipe to operate, it must satisfy the following limitations:

1. The maximum capillary pumping head $\Delta p_{c,\max}$ must be greater than the total pressure drop in the pipe
   $$(\Delta p_{c,\max} > \Delta p_l + \Delta p_v + \Delta p_g) \tag{1}$$
2. Checking of vapor flow "sonic limitation".
   Tearing of liquid of the liquid –vapor interface by vapor flowing at high velocity "entertainment limitation".
3. Disruption of the liquid flow by nucleate boiling in the "boiling limitations".

If one of the above conditions is not fulfilled, the wick will dry out in the evaporator region and the pipe will not operate, these limits are illustrated in Figure 4.

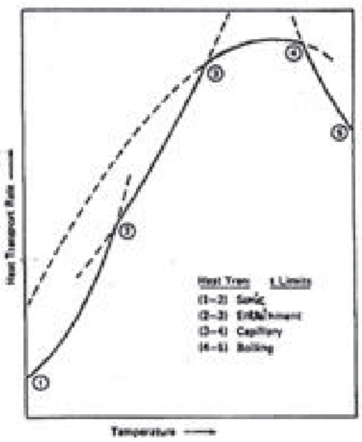


Figure 4. Heat transport limitation of a heat pipe

## PRESSURE BALANCE

For the required balance of pressure it is necessary that along the entire length of the heat pipe, the pressure at the liquid side of the liquid–vapor interface is different from that at the vapor side, except at the point where the difference is minimum. This pressure difference is called the capillary pressure.

The maximum capillary pressure can calculated using Laplace and Young equation in case of cylinder pore.

$$p_{c,\max} = \frac{2\sigma_l}{r_c} \tag{2}$$

The values of the effective capillary radius ($r_c$) depends on the wire diameter, the space between wires and the wick structure shape. The pressure drop in wick structure can be calculated by integrating the liquid pressure gradient.

$$\Delta p_l = -\int_{x_1}^{x_2} \frac{dp_l}{dx} dx \tag{3}$$

The liquid pressure drop depends on many parameters such as friction drag, Reynolds Number, hydraulic radius, axial heat flux, latent heat of vaporization, wick cross section area, inclination angle and the wick permeability. The final expression of the liquid pressure drop is:

$$\frac{dp_L}{dx} = -F_L Q \pm p_L q \sin\phi \tag{4}$$

The values of can be obtained directly using a special charts. The vapor pressure drop in heat pipe is calculated by integration the vapor pressure gradient.

$$\Delta p_v = -\int_{x_1}^{x_2} \frac{dp_v}{dx} dv \tag{5}$$

The principles of conservation of axial momentum can be applied to an elementary control volume, and the final relation is as following:

$$\frac{dp_v}{dx} = -F_v Q - D_V \frac{dQ^2}{dx} \tag{6}$$

The values of the friction coefficient $F_v$ and the dynamic pressure coefficient $D_v$ can be obtained using special charts. Hence the effective capillary pressure $P_{c,e}$ is calculated using equation:

$$P_{c,e} = \int_{X_{\min}}^{X} \left( \frac{dP_V}{dX} - \frac{dP_L}{dX} \right) dX \tag{7}$$

The maximum effective capillary pressure $P_{c,me}$ will be smaller than the maximum capillary pressure $P_{c,max}$, this difference is due to the effect of the gravitational force in direction perpendicular to the heat pipe axis that:

$$P_{c,me} = P_{c,m} - \rho_L g dv \cos\varphi$$

$$i.e \quad \frac{2\sigma_L}{r_c} - \Delta P_L = \int_0^{L_t} \left( \frac{dP_X}{dX} - \frac{dP_L}{dX} \right) dX \tag{8}$$

Substitution about from Equations (4), (5) in Equation (6), and solving for yields:

$$Q_{L_{C,MAX}} = \int_0^{L_t} Q_{c,max} dX = \frac{\frac{2\sigma_L}{r_c} - \Delta P_L - \rho_L g L_t \sin\varphi}{F_L + F_V} \tag{9}$$

In case of that heat pipe has a uniform heat flux distributions along its evaporator and condenser sections, the axial heat flux will has the following final form:

$$Q_{c,\max} = \frac{Q_{L,\max}}{\frac{L_c}{2} + L_a + \frac{l_e}{2}} \qquad (10)$$

The general procedures to evaluate the capillary limitation of heat pipe are described in the material that follows:

- Calculate the required capillary pressure and compare it with the maximum effective capillary pressure as follows:
    1. If $P_{c,r}=P_{c,max}$ , the assumed heat load is the required capillary limitation .
    2. If $P_{c,r}<P_{c,max}$ , increase the assumed heat load ,and repeat step1.
    3. If $P_{c,r}>P_{c,max}$, decrase the assumed heat load ,and repeat step1.

## LIMITATIONS TO HEAT TRANSPORT IN HEAT PIPE

## BOILING LIMITATION

The liquid pressure at the evaporator is equal to the saturation pressure at the temperature of the liquid vapor interface minus the capillary pressure at the temperature of the liquid vapor interface. Since the difference increase with increase of the radial heat flux of the heat pipe at the evaporator, vapor bubbles may be formed in the evaporator wick, and this may hot spots of the liquid and abstract the circulation of the liquid. Hence, there is a heat flux limit for the evaporation at the heat pipe. This limit is known as the boiling limit which represents the limitations of the axial heat flux.

The boiling heat transport limit is calculated using the following expression:

$$Q_{b,\max} = \frac{2\pi L_e K_e T_v}{\lambda \rho_v \ln\left(\frac{r_i}{r_v}\right)} \left(\frac{2N}{r_n} - P_c\right) \qquad (11)$$

## SONIC LIMITATION

The maximum mass flow rate relates directly to the maximum heat transfer rate. This takes place when the evaporator exit velocity reaches the local sonic velocity. Increasing the heat rejection rate decreases the condenser temperature, induces the supersonic vapor flow and creates very large axial temperature gradients along the pipe. The sonic limit is given by the following expression:

$$Q_{s,\max} = A_v \lambda \rho_v \left(\frac{1+\gamma_v}{2+\gamma_v}\right)\left(\gamma_v R T_v\right)^{0.5} \qquad (12)$$

## ENTRAINMENT LIMITATION

Since the vapor and liquid move opposite direction in a heat pipe, a shear force exists at the liquid vapor interface. If the vapor velocity is sufficiently high, a limit can be reached at which liquid will be turn from the surface of the wick and entrained in the vapor, once entertainment begins, there is a sudden substantial increase in fluid circulation until the liquid return; system cannot accommodate the increased flow. When this occurs, abrupt dry out of the wick at the evaporator results. This limit will prevent the heat pipe operation and represents one limit to the performance of the heat pipe. The entertainment heat transport limit can be calculated using the following formula:

$$Q_{e,max} = A_v \lambda \left( \frac{\sigma_L \rho_v}{2 r_{h,c}} \right)^{0.5} \tag{13}$$

## DESIGN CRITERIA

The design criteria of the heat pipe is based on the following general points:

1. Selection of the suitable working fluid.
2. Design and selection of the wick structure.
3. Structural design of the container.
4. Heat transfer limits must be checked to ensure the pipe will operate within all limits.

## SELECTING OF WORKING FLUID:

The performance and life of a heat pipe is greatly dependent o working fluid employed. Care must be exercised in selecting a suitable fluid for the operating condition. The working fluid should satisfy the following requirements:

- The working fluid must have a melting point temperature below and critical temperature above the pipe operating temperature boiling.
- Good thermal stability to prevent the fluid breaking down into different components.
- A high enthalpy of vaporization is desirable in order to transfer a large amount of heat.
- The thermal conductivity should be high in order to minimize the radial temperature gradient and to reduce the possibility of nucleate boiling at wick wall interface.
- Compatibility with wick and wall material is necessary to increase the life of the heat pipe.
- High surface tension is desirable in order to enable the heat pipe to operate against the gravity.
- Wettability of wick and wall material is necessary so that the working fluid will meet the wick and container material with very small contact angles.

## MATERIAL SELECTION

The selection of the heat pipe material and wick material is based on the following requirements:

- Compatibility with working fluid.
- High thermal conductivity.
- Ease of fabrication to decrease the cost of heat pipe manufacturing.
- High strength to weight ratio.

## HEAT PIPE DESIGN PROCEDURES

The procedures of designing a heat pipe are as following:

1. Pipe diameter will first be determined so that the vapor velocity is not high, such that the maximum Mach number in the vapor flow passage does not greater than 0.2
2. Mechanical design theory will be used to determine the container details.

3. Wick details will be designed considering the capillary limit.
4. Other heat transfer limits (entertainment, sonic and boiling) should be checked to ensure that the pipe will operate within all limiting conditions.

## DESIGN OF VAPOR CONE DIAMETER

The vapor core diameter ($d_v$) at vapor Mach number Mv = 0.2 is determined by using the following equation:

$$d_V = \left( \frac{20 Q_{max}}{\pi \rho_V \lambda (\gamma_V R_V T_V)^{\frac{1}{2}}} \right)^{1/2} \tag{14}$$

## DESIGN OF HEAT PIPE CONTAINER

The most widely used design technique for heat pipe container that must withstand vapor pressure is the (ASME) code. The calculations of the maximum stress depend on the geometry of the tube and the wall thickness, for rounded tube and t/d <10 %

$$F_{max} = \frac{\Delta p\, do}{2 t_p} \tag{15}$$

## WICK DESIGN

There are many charts which are useful for quick determination of dimension of the heat pipe wick. The general procedures for designing the wick are as following:

1. Calculate the hydro-static pressure using the following formula:
$$P_{HYD} = \rho_L\, g(di \cos\psi + Lt \sin\psi) \tag{16}$$

2. Select the mesh number such that the $P_c$ is much smaller than twice hydrostatic pressure
Calculate the wick thickness $t_w = \frac{di - dv}{2}$ (17)

3. Calculate the maximum capillary heat transfer using equation
$$Ql_{c,max} = \frac{P_C - P_{hyd}}{F_L + F_V} \tag{18}$$

4. Check the maximum capillary heat transfer rate, which must be greater than that the required heat transport rate.
5. Chick the entertainment, sonic, viscous and boiling limitation that;
$Q_{req} < (Q_{c,max}, Q_{b\,max}, Q_{s,reg})$

## DESIGN PROBLEM

According to NEL/Shel report tests with piston engines, it is required to design a heat pipe for the following given data:

1. Maximum heat transfer rate $Q_{req} = 2.5$KW
2. Inlet hot gas temperature Tv = 200 C°
3. Average working pressure $Pv = 0.09 \times 10^5$ N/m$^2$

As a first design trail water is chosen as working fluid, while a copper has choose as the material of the pipe, but the primarily design calculations indicated that the boiling limitation would not be satisfied. So as a result of different trials the mercury is selected as a working fluid, the properties of mercury are shown in Table 1.

Table 1. Properties of mercury at $T_H$=200 C

| Property | Symbol | Magnitude | Unit |
|---|---|---|---|
| Latent heat | $L$ | 305.5 | KJ/Kg |
| Liquid thermal conductivity | $K_l$ | 10.01 | (W/m)C |
| Liquid density | $\rho_l$ | 13112.5 | $Kg/m^3$ |
| Liquid viscousity | $\mu_l$ | $1.025\times10^{-3}$ | Kg/m.sec |
| Liquid surface tension | $\sigma_l$ | $4.3\times10^{-5}$ | $N/m^2$ |
| Vapor density | $\rho_v$ | 0.305 | $Kg/m^3$ |
| Vapor viscousity | $\mu_v$ | $4.305\times10^{-5}$ | Kg/m.sec |
| Vapor pressure | $P_v$ | $0.095\times10^{5}$ | $N/m^2$ |
| Vapor specific heat | $\gamma_v$ | 0.0104 | (KJ/Kg)C |

**DESIGN OF CONTAINER PARAMETER:**

Assuming vapor cone diameter $d_V = 2.5\times10^{-2}$ m, using a specific charts which indicates that for $P_v = 0.095\times10^5\ N/m^2$ $\quad d_0 / d_i = 1.5$,

Let $d_0 = 4.5\times10^{-2}$ m, hence $d_i = 3\times10^{-2}$ m

**DESIGN OF THE WICK THICKNESS**

$t_w = (d_i - d_v) / 2 = (3\times10\text{-}2 - 2.5\times10\text{-}2) / 2 = 0.25\times10^{-2}$ m

$$A_w = \frac{\pi}{4}\left(d_i^2 - d_w^2\right) = 2.1598\times10^{-4}\,m^2$$

$$A_v = \frac{\pi}{4}\left(d_v^2\right) = 4.9087\times10^{-4}\,m^2$$

**CHECK FOR HOOP STRESS**

$$f_{hoop} = \frac{\Delta P_v d_0}{2t_p}$$

$$f_{ult} = 1.379\times10^{8}\ N/m^2$$

$$f_{hoop} = \frac{0.91825\times10^5\times4.5\times10^{-2}}{2\times0.2705\times10^{-2}} = 2.7547\times10^5\,\frac{N}{m^2}$$

Hence $f_{hoop} < f_{ult}$

Assume that the number of wick layer is n = 25, hence the wire diameter ( $d_w$ ) becomes $d_w = t_w/2n = 0.25\times10^{-2} / 50 = 5\times10^{-5}$ m

**CALCULATING OF THE MAXIMUM CAPILLARY PRESSURE:**

$$P_{c,\max} = \frac{2\sigma_L}{r_c}, r_c = \frac{1}{2 \times 3937} = 1.27 \times 10^{-4}$$

$$P_{c,\max} = \frac{2 \times 4.3 \times 10^{-1}}{1.27 \times 10^{-4}} = 6771.653 \frac{N}{m^2}$$

**NORMAL HYDROSTATIC PRESSURE:**

$$\Delta P_L = \rho_L \, g \; L_t \sin\psi, \qquad \psi = 90^\circ$$

Let $L_t = 20 \times 10^{-2}$ m

$= 13112.5 \times 9.81 \times 20 \times 10^{-2} = 25726.725$ N/m$^2$

**CHECK LIMITATION OF THE HEAT PIPE**

1. **Sonic limit check**

$$Q_{c,\max} = A_v \rho_v \lambda \left( \frac{\gamma_v + 1}{\gamma_v + 2} \right) (\gamma_v \, R_v \, T_v)^{\frac{1}{2}}$$

$Q_{S,\max} = 4.9087 \times 10^{-4} \times 0.305 \times 305.5 \times 10^3 \ (2.67/3.67)\ (1.67 \times 4.1 \times 473.5)^{0.5} = 5989.248$ W

Since $Q_{s,\max} > Q_{required}$ , Thus it satisfies the required condition.

2. **Entrainment limit check**

$$Q_{e,\max} = A_v \, \lambda \left( \frac{\sigma_L \rho_v}{2\left( \frac{1}{2N} - \frac{d_w}{2} \right)} \right)$$

$$Q_{e,\max} = 4.9087 \times 10^{-4} \times 305.5 \times 10^3 \left( \frac{0.305 \times 4.35 \times 10^{-1}}{2 \times 1.02 \times 10^{-4}} \right)^{0.5} = 38 \, W$$

3. **Boiling limit check**

$$Q_{b,\max} = \frac{2 \pi L_e K_e T_v}{\rho_v \lambda \ln\left( \frac{r_i}{r_0} \right)} \left( \frac{2 \delta_L}{r_n} - P_c \right)$$

$$Q_{b,\max} = \frac{2\pi \times 8 \times 10^{-2} \times 10.61 \times 473.15}{305.5 \times 10^3 \times 0.305 \, \ln\left( \frac{1.5}{1.25} \right)} \left( \frac{2 \times 4.3 \times 10^{-1}}{2.54 \times 10^{-7}} \right) = 502.920 \, KW$$

$Q_{b,\max} > Q_{required}$

This satisfies the required condition.

**MAXIMUM HEAT RATE**

Equation (10) can generally be used to evaluate the capillary limitation for conventional heat pipe operation is the heat pipe mode. This equation has been solved numerically to calculate the maximum heat rate $Q_{c,max}$. The result of this solution indicates that:

$Q_{c,max} = 1.0493\times10^5$ W

$Q_{c,max} > Q_{req}$ which satisfies the required condition.

**CALCULATION OF HEAT PIPE PERFORMANCE**

The heat pipe performance is characterized by the overall coefficient of heat transfer which defined by the equation:

$Q = A\, U_{HP}\,( T_{P,e} - T_{P,c} )$

- **The thermal resistance at the evaporator ($R_{P,e}$):**

  $R_{P,e} = r_0\, t_p / ( 2\times L_e\, k_p )$

  $= 2.25\times10^{-2}\times0.75\times10^{-2} / ( 2\times8\times10^{-2} \times34 ) = 2.6768\times10^{-6}$ C°/ W

- **The thermal resistance of the saturation wick at the evaporator:**

  $R_{W,c} = r^2{}_0\, t_w / (2\, L_c\, r_i\, k_{c,c} )$

  $= ( ( 2.25\times10^{-2})^2\times0.25\times10^{-2} ) / ( 2\times8\times10^{-2}\times1.5\times10^{-2} \times10.61 ) = 497025\times10^{-5}$ $m^2.C°/W$

- **The thermal resistance of the vapor flow $R_v$**

$$R_v = \frac{F_v\, \pi r_0^2 \left(\frac{l_c}{6} + l_a + \frac{l_c}{6}\right) T_v}{\rho_v\, \lambda}$$

$$R_v = 1.294\times10^{-9}\ m^2.C^{\circ}/w$$

- **The thermal resistance of the wick at condenser ($R_{wic}$)**

$$R_{w,c} = \frac{r_0^2\, t_w}{2l_c r_i k_{ee}} = \frac{\left(2.25\times10^{-2}\right)}{2\times8\times10^{-2}} \times \frac{0.25\times10^{-2}}{2.4795\times10^{-2}\times10.61} = 4.9702\times10^{-5}\, m^2.C^{\circ}/w$$

$$U_{H,P} = \frac{1}{R_{p,e} + R_{w,v} + R_v + R_{p,c} + R_{wic}} = 9545.6378\ m^2.C^{\circ}/w$$

- **Estimation of temperature variation across the pipe wall**

  The vapor temperature differences

$$\Delta T_v = \frac{QR_v}{A_p} = \frac{2500\times1.294\times10}{\left(\frac{\pi}{4}\right)\times\left(\frac{4.5}{100}\right)^2} = 2.034\times10^{-3}\, C^{\circ}$$

  So vapor temperature of the condenser

$$T_{v,c} = T_{v,e} - \Delta T_v = 200 - 0.002034 = 199.99\, C^{\circ}$$

- **Temperature difference at wick pipe interface at condenser**

$$\Delta T_{w,c} = \frac{QR_{wic}}{A_p} = \frac{2500 \times 4.9702 \times 10^{-5}}{\left(\frac{\pi}{4}\right)\left(4.5/100\right)^2} = 78.1266\,C^{\circ}$$

$$T_{pw,c} = T_{v,c} - \Delta T_{w,c} = 199.985 - 78.1266 = 121.8525\ c^{0}$$

- **Temperature difference across the pipe wall at the condenser**

$$\Delta T_{p,c} = \frac{QR_{p,c}}{A_p} = \frac{2500 \times 2.6768 \times 10^{-6}}{\frac{\pi}{4}\left(\frac{4.5}{100}\right)^2} = 4.2076\,c^{\circ}$$

Hence the condenser surface temperature Tp,c = 121.8585 – 4.2076 = 117.65 C°

- **The temperature difference across the surface wick at the evaporator**

$$\Delta P_{w,c} = \frac{QR_{w,e}}{A_P} = \frac{2500 \times 4.970 \times 10^{-5}}{\frac{\pi}{4}\left(\frac{4.5}{100}\right)^2} = 78.127\ c^{\circ}$$

Hence the temperature of pipe wick interface at the evaporator

$$T_{P,we} = T_{v,e} + \Delta T_{w,e} = 200 + 78.127\ \ 278.127\ c^{\circ}$$

- **The temperature difference across the pipe wall at the evaporator**

$$\Delta T_{P,e} = \frac{QR_{p,e}}{A_P} = \frac{2500 \times 2.6768 \times 10^{-6}}{\frac{\pi}{4}\left(\frac{4.5}{100}\right)^2} = 4.2076\ \ c^{\circ}$$

So the pipe temperature at the evaporator is equal to

$$\mathbf{T_e = T_{pw,c} + \Delta T_{pe} = 278.127 + 4.2076 = 282.334\ c^{\circ}}$$

The final shape of the designed heat pipe is illustrated in Figure 5.

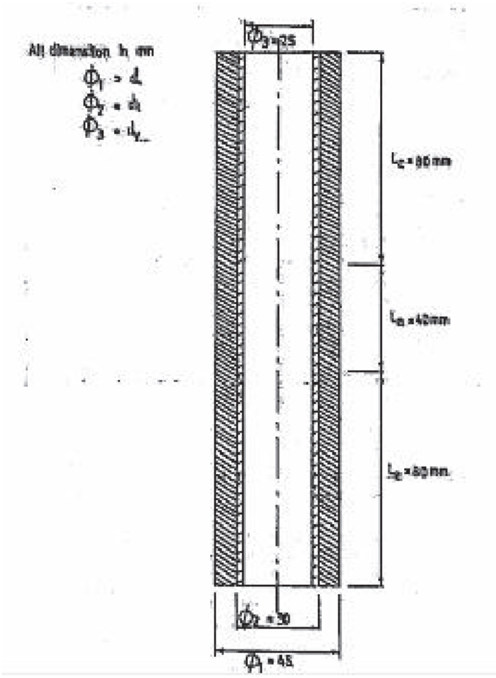

Figure 5. The designed heat pipe.

## FINAL DESIGN RESULTS

The final results of the designed heat pipe are as following

| | |
|---|---|
| Work fluid | Mercury |
| Container material | Copper |
| Wick material | Copper |
| Outside diameter | $d_0 = 4.5\times10^{-2}$ m |
| Inside diameter | $d_i = 3\times10^{-2}$ m |
| Vapor cone diameter | $d_v = 2.5\times10^{-2}$ m |
| Pipe wall thickness | $t_p = 0.75\times10^{-2}$ m |
| Wick thickness | $t_w = 0.25\times10^{-2}$ m |
| Wick area | $A_w = 2.1598\times0^{-4\ 2}$ |
| Wick diameter | $d_w = 5\times10^{-5}$ m |
| Mesh number | N = 3937 m |
| Number of wick lager | n = 25 |
| Maximum capillary pressure | $P_{c,max} = 6771.653$ N/m$^2$ |
| Axial hydrostatic pressure | $P_{h,c} = 25726.725$ N/m$^2$ |
| Pipe length | $L_t = 20\times10^{-2}$ m |
| Maximum available pumping pressure | $P_{pm}= 32498.378$ W/m$^2$ |
| Length of heat pipe adiabatic length. | $L_a = 4\times10^{-2}$ m |
| Length of heat pipe condenser section | $L_c = 8\times10^{-2}$ m |
| Length of heat pipe evaporator section | $L_e = 8\times10^{-2}$ m |
| Vapor Reynold number | $R_{ev} = 9681.1942$ |
| Vapor Mach number | $M_V = 0.3036$ |
| Sonic limit | $Q_{s,\max} = 5989.248w$ |
| Enterainmut limit | $Q_{e,\max} = 3824.3459w$ |

| | |
|---|---|
| Boiling limit | $Q_{b,\max} = 502.9297 \times 10^3\, Kw$ |
| Vapor temperature o f the condenser | $T_{V,c} = 199.998^\circ c$ |
| Pipe wick temperature of the condenser section | $T_{pw,c} = 121.8525^\circ c$ |
| Condenser surface temperature | $T_{P,C} = 117.65^\circ c$ |
| Pipe wick temperature of the evaporation section | $T_{P,w,e} = 278.127^\circ c$ |
| Surface temperature of the evaporator section | $T_{P,e} = 282.334^\circ c$ |

## CONCLUSION

1. An effective wick structure requires small surfaces pores for large capillary pressure.
2. Design of the best heat pipe type and size for any given application is however, a complex process, involving many factors such as heat pipe performance, reliability, compatibility and ease of manufacturing.
3. The performance and life of a heat pipe is greatly dependent on the compatibility between the working fluid and the material of the heat pipe.
4. Copper suited most as a material for heat pipe container, to resist the stresses developed, and being compatible with variety of working fluids.
5. Compatibility property of the material of heat with the working fluid is very importance factor.

## NOMENCLATURE

| Symbol | Definition | Unit |
|---|---|---|
| $d_0$ | Pipe outside diameter | m |
| $d_i$ | Pipe inside diameter | m |
| $d_v$ | Vapor core diameter | m |
| $d_w$ | Screen wire diameter | m |
| $A_p$ | Cross-section area based up on pipe outside diameter | $m^2$ |
| $A_v$ | Vapor core cross-section area | $m^2$ |
| $D_v$ | Dynamic pressure coefficient | m.sec/kg |
| $\sigma_L$ | Surface tension force at liquid -wick interface | N |
| $F_v$ | Frictional coefficient for vapor flow | $(N/m^2)/W.m$ |
| $f_L$ | Drag coefficient for liquid flow | [ ] |
| $f_v$ | Drag coefficient for vapor flow | [ ] |
| G | Gravitational acceleration | $m/sec^2$ |
| $K_{e,c}$ | Effective thermal conductivity of liquid saturated wick at condenser | $W/m.C^\circ$ |
| $K_l$ | Thermal conductivity of liquid | $W/m.C^\circ$ |
| $K_w$ | Thermal conductivity of wick material | $W/m.C^\circ$ |
| $K_p$ | Thermal conductivity of pipe material | $W/m.C^\circ$ |
| $\Lambda$ | Latent heat of vaporization | KJ/Ka |
| $L_a$ | Length of heat pipe adiabatic section | M |
| $L_c$ | Length of pipe condenser | M |
| $L_e$ | Length of pipe evaporator | M |
| $P_c$ | Capillary pressure | Pas |
| $P_{c,r}$ | Required capillary pressure | Pas |
| $P_{c,max}$ | Maximum capillary pressure | Pas |
| $P_{c,me}$ | Effective capillary pressure | Pas |

| | | |
|---|---|---|
| $P_L$ | Liquid pressure | Pas |
| $P_v$ | Vapor pressure | Pas |
| $\Delta P_g$ | Pressure drop due to the gravity | Pas |
| $\Delta P_l$ | Liquid pressure drop | Pas |
| $\Delta P_v$ | Vapor pressure drop | Pas |
| $\Delta P_L$ | Normal hydrostatic pressure drop | Pas |
| Q | Heat folw rate | W |
| $Q_{b,max}$ | Boiling limit on heat transfer rate | W |
| $Q_{c,max}$ | Capillary limit on heat transfer rate | W |
| $Q_{e,max}$ | Entraiment limit on heat transfer rate | W |
| $Q_{s,max}$ | sonic limit on heat transfer rate | W |
| $R_v$ | Thermal resistance for vapor flow from evaporator to condenser | $m^2.C^{\circ}/W$ |
| R | Raduis of cylinder | M |
| $r_c$ | Effective capillary radius | M |
| $r_i$ | Inside radius of pipe | M |
| $r_v$ | Vapor core radius | M |
| $T_v$ | Vapor temperature | $C^{\circ}$ |
| $t_p$ | Pipe thickness | M |
| $t_w$ | Wick thickness | M |
| $\gamma_v$ | Vapor specific heat ratio | [ ] |
| $\rho_L$ | Liquid density | $Kg/m^3$ |
| $V_L$ | Liquid velocity | m/sec |
| $F_L$ | Friction coefficient for liquid flow | $(N/m^2)/W.m$ |
| $f_{max}$ | Maximum tensile stress | $N/m^2$ |
| $f_{ult}$ | The ultimate stress | $N/m^2$ |
| $f_{ult,d}$ | The ultimate design stress | $N/m^2$ |
| $K_p$ | Thermal conductivity of pipe material | $W/m.C^{\circ}$ |
| N | Screen mesh number | Meshes/m |
| $r_{h,l}$ | Hydraulic radius for liquid flow | m |
| $r_{h,c}$ | Hydraulic radius of wick at vapor-wick interface | m |
| $r_{h,v}$ | Hydraulic radius for vapor flow | m |
| $r_i$ | Inside raduis of pipe | m |
| $r_v$ | Vapor core radius | m |
| $_L$ | Liquid velocity | m/sec |
| $V_v$ | Vapor velocity | m/sec |
| W | Wire spacing | m |
| $\rho_v$ | Vapor density | $Kg/m^3$ |
| $\Phi$ | Heat pipe inclination measured from the horizontal position | degree |

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# تصميم أنبوب حراري لتحسين انبعاثات العادم من المحركات العاملة بالنفط

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**الملخص**
قام كل من المختبر الهندسي الوطني (المملكة المتحدة)، ومختبر شِلْ للأبحاث، بالاشتراك في تطوير تطبيقات الأنبوب الحراري لحل مشكلة الانبعاثات العادمة من المحرك النفطي.

في هذه الورقة تم إعداد تصميم كامل لأنبوب حراري، آخذاً في الحسبان المعايير الضرورية لتحديد متغيرات الأبعاد المختلفة. وقد أعد هذا التصميم باستخدام الصيغ الأساسية في الديناميكا الحرارية، وانتقال الحرارة، والفيزياء. وقد تم أيضاً التحقق من نتائج هذا التصميم عند مختلف حدود ظروف التشغيل العملية.