# Optimum design of a heat pipe

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## Abstract

**This research deals with the optimum design of the heat pipe components using Matlab program, version 7, at the determined operating conditions. The heat pipe is used to transfer maximum available thermal energy to heat the feed water entering steam boiler .**

**The affecting parameters on heat pipe performance were detailed as well as the transport limitations and the thermal resistances of the heat pipe parts. Satisfactory results were obtained for the heat pipe dimensions, components and design influentials. Also, the parameters affecting the heat transfer rate and the working fluid mass-flow rate were discussed. From the important results, the heat pipe heat transfer capability is directly proportional with each of the heat pipe bore, the wick c/s area and the heat pipe effective length, and inversely with each of the capillary radius and the working fluid viscosity.**

**Keywords: Heat pipe, Optimum design, Steam boiler.**

# التصميم الأمثل لأنبوب حراري

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## الخلاصة

**يتناول هذا البحث التصميم الامثل لاجزاء ومكونات انبوب حراري باستخدام برنامج Matlab الاصدار 7، عند ظروف التشغيل المحددة. يستخدم الانبوب الحراري لنقل اكبر طاقة حرارية ممكنة لتسخين الماء المغذي الداخل الى مرجل بخاري .**

**وقد تم تفصيل العوامل المؤثرة على اداء الانبوب الحراري ومحددات انتقال الحرارة والمقاومة الحرارية لاجزاء الانبوب الحراري، والحصول على نتائج مرضية لابعاد ومكونات الانبوب الحراري ومؤثرات التصميم، وكذلك مناقشة العوامل المؤثرة على كل من معدل إنتقال الحرارة ومعدل جريان كتلة مائع التشغيل.من النتائج المهمة هي ان مقدرة الانبوب الحراري على نقل الحرارة تتناسب طرديا وكل من قطر الانبوب ومساحة مقطع الفتيلة والطول المؤثر للانبوب، وعكسيا مع كل من نصف القطر الشعري ولزوجة مائع التشغيل.**

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## Nomenclature

| Symbol | Definition | Unit |
|---|---|---|
| A | Cross-sectional area | $m^2$ |
| Cp | Specific heat at constant pressure | J/kg. ºC |
| D | Diameter | m |
| $d_w$ | Mesh wire diameter | m |
| g | Gravitational acceleration | $m/s^2$ |
| HP | Heat pipe | ------- |
| $h_{fg}$ | Latent heat of vaporization | J/kg |
| K | Permeability | $m^2$ |
| k | Thermal conductivity | W/m. ºC |
| L | Pipe length | m |
| M | Merit number | $W/m^2$ |
| $m^o$ | Mass flow rate | kg/s |
| N | Mesh number | -------- |
| P | Pressure | $N/m^2$ |
| Q | Heat transfer rate | W |
| R | Thermal resistance | ºC/W |
| r | Radius | m |
| T | Operating temperature | ºC |
| WF | Working fluid | -------- |
| w | Mesh wire spacing | m |
| ρ | Density | $kg/m^3$ |
| σ | Surface tension | N/m |
| $\mu$ | Viscosity | kg /m.s |
| Δ | Difference | -------- |
| β | Angle of inclination | degree |
| θ | Contact (wet) angle | degree |
| $\varepsilon$ | Wick porosity | -------- |

## Subscripts

| Symbol | Definition |
|---|---|
| a | Adiabatic |
| am | Ambient |
| bo | Boiling |
| c | Condenser |
| cap | Capillary |
| e | Evaporator |
| eff | Effective |
| ent | Entrainment |
| i | Inner |
| in | Inlet |
| l | Liquid |
| max | Maximum |
| n | Nucleate |
| o | Outer |
| P | Pipe |
| so | Sonic |
| t | Total |
| v | Vapor |
| vis | Viscous |
| w | Wick |
| wa | Water |
| ws | Wick structure |

## 1- Introduction:

A heat pipe is a two -phase heat transfer device operats on a closed cycle and efficiently transports heat from one end to another. It utilizes the latent heat of the vaporized fluid rather than sensible heat [1][2].

Heat added to the evaporator causes the working fluid in the evaporator wicking structure to be vaporized. The high temperature and corresponding high pressure in this region result in flow of the vapor to the other cooler end of the container where the vapor condenses, giving up its latent heat of vaporization. The capillary forces existing in the wicking structure then pump the liquid back to the evaporator section [1]. This process will continue as long as there is a sufficient capillary pressure to pump the condensate back to the evaporator .

Heat pipe performance depends upon many factors concerned with its design, manufacture and operation. The structure of the wick has a strong effect on the heat pipe performance. The performance of a heat pipe is often expressed in terms of " equivalent thermal conductivity" [3].

Hung and seng[4] study the effects of geometric design on the thermal performance of star-groove micro-heat pipes.They carried out a numerical solution for one-dimensional steady state model.The heat transport rate is evaluated for different geometric designs and operating conditions.

Numerical simulation for the solution of steady incompressible flow in both vapor region and wick structure were studied by Mahjoub and Mmahtabroshan [5]. They discussed the effect of parameters variation on heat pipe operation. The results show that thermal resistance of cylindrical heat pipe grows with increasing wick porosity, and decreases with increasing of wall thermal conductivity and heat pipe radius.

Wang and Peterson [6]carried out a one-dimensional analytical model which incorporated the effects of the liquid-vapor phase interaction and the c/s area variation on the heat transfer performance.They found that the max heat transport capacity increased with increases in the wire spacing.

## 2- The outlines of the present research :

The aim of this study is to design a heat pipe for heating feed water entering a steam boiler from (20ºc to 78ºc) at a rate of (0.0114kg/s), using three electrical heaters located around the evaporator, as illustrated in figure (1). This amount of heat is rejected from the heat pipe condenser, and transferred to the water in the water jacket around the condenser section. Assuming perfect heat pipe insulation, the rejected heat is calculated from the following equation:

$$Q= m^{o}_{wa} C_{p,wa}(T_{in}-T_{am}) \qquad \ldots\ldots(1)$$

## 3. Design considerations and procedure:

The main components of a heat pipe are the container, the working fluid and the wick or capillary structure. The HP design considerations are discussed in the following sections and illustrated in figure (3).

### 3.1: Working fluid choice:

The intended operating temperature of the heat pipe can be met only with a fluid whose saturation temperatures cover the design temperature range [7] .

The prime requirements for the working fluid selection are [3,7]:

1- High latent heat.
2- High thermal conductivity.
3- Compatibility with wick and wall materials.
4- Low liquid and vapor viscosities.
5- High surface tension .
6- Wettability of wick and wall materials .
7- Availability and cost .

Merit number is a convenient means for comparing working fluids to select the more suitable for heat pipe operation and is defined from the following equation :

$$M = \frac{\sigma_l \, h_{fg} \, \rho_l}{\mu_l} \qquad \ldots\ldots(2)$$

Figure (2) shows the superiority of water over the range (350-500K) compared with other fluids.

Working fluid selection must also be based on thermodynamic considerations which are concerned with the various limitations to heat flow occurring within the heat pipe, as detailed

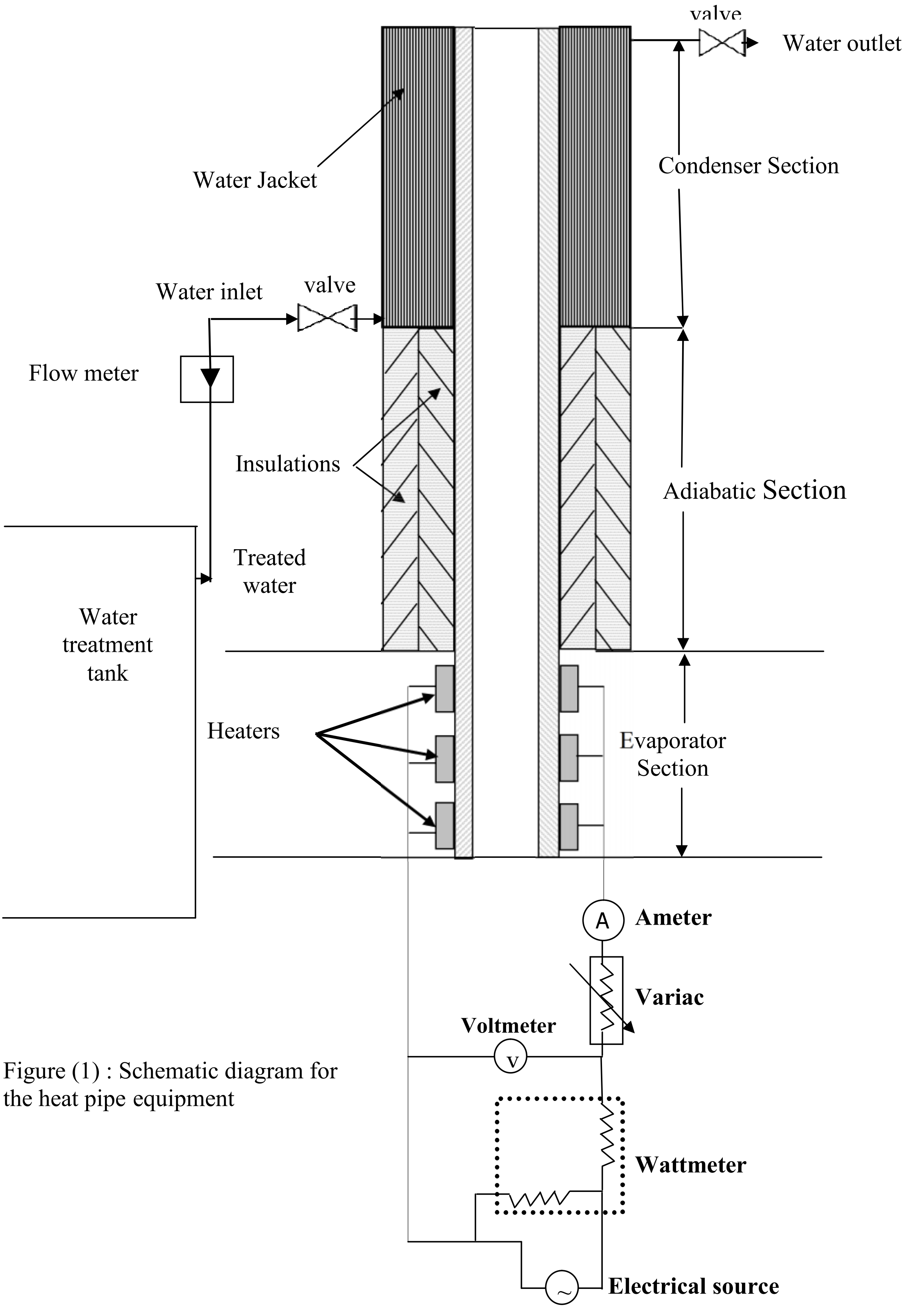
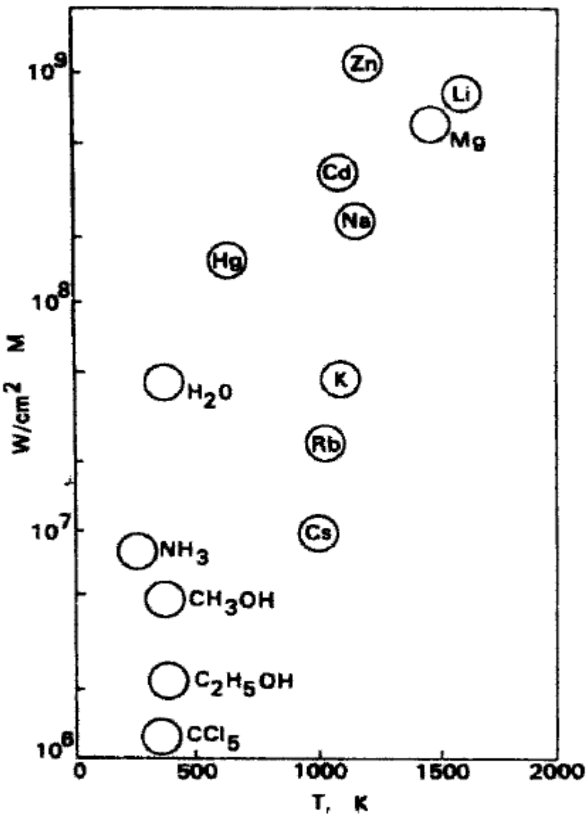



Figure (1) : Schematic diagram for the heat pipe equipment in section (3-6) .

According to the high characteristics and superiority of water , because of its high latent heat, a high "figure of merit" and low cost, it was chosen as the working fluid for the heat pipe of current research.




**Figure(2):Merit number for selected Working fluids[8]**

**3.2: The container:**

To select container material, the following requirements have to be taken into consideration [8]:

1- compatibility, both with working fluid and the external environment .
2- High strength to weight ratio and ease of fabrication .
3- High thermal conductivity and wettability .

The two major results of incompatibility are corrosion and the generation of non-condensable gas .

Copper is the best one to meet the requirements mentioned recently.

**3.3: The wick or capillary structure :**

The wick material must be compatible and wettable with the working fluid.

The structure of the wick has a strong effect on the performance of the heat pipe. Of the wick forms available, meshes are the most common. These are manufactured in a range of pore sizes and materials.

The characteristics of a wick can be changed by changing the size and the number of the pores per unit volume and the continuity of the passageway [9]. The maximum capillary head generated by a wick increases with decreasing pore size, whereas the wick permeability increases with increasing pore size. Wicks in horizontal and gravity assisted heat pipes should permit maximum liquid flow rate by having a comparatively large pore size as with (100) or (150) mesh, and where pumping capability is required against gravity, small pores are needed [8].

The heat transport capability of the heat pipe is raised by increasing the wick thickness, with a small increase in thermal resistance [10]. However, the increased radial thermal resistance of the wick created by this, would work against increased capability, and would lower the allowable maximum evaporator heat flux [8].

According to the mentioned above, copper wick of the form (100) mesh and six layers was chosen .

**3.4: Tilting of heat pipe**:

Capillary action permits the heat pipe to operate in any orientation in a gravity field. However, the performance of a heat pipe will be best when the capillary and gravity forces act in the same direction (evaporator end down ) and vice versa. Gravity dose not affect the capillary force when the heat pipe is in the horizontal position [9] .

So, on this basis, gravity assisted situation for heat pipe orientation were used for this research, at a tilt angle of (90º) with horizontal.

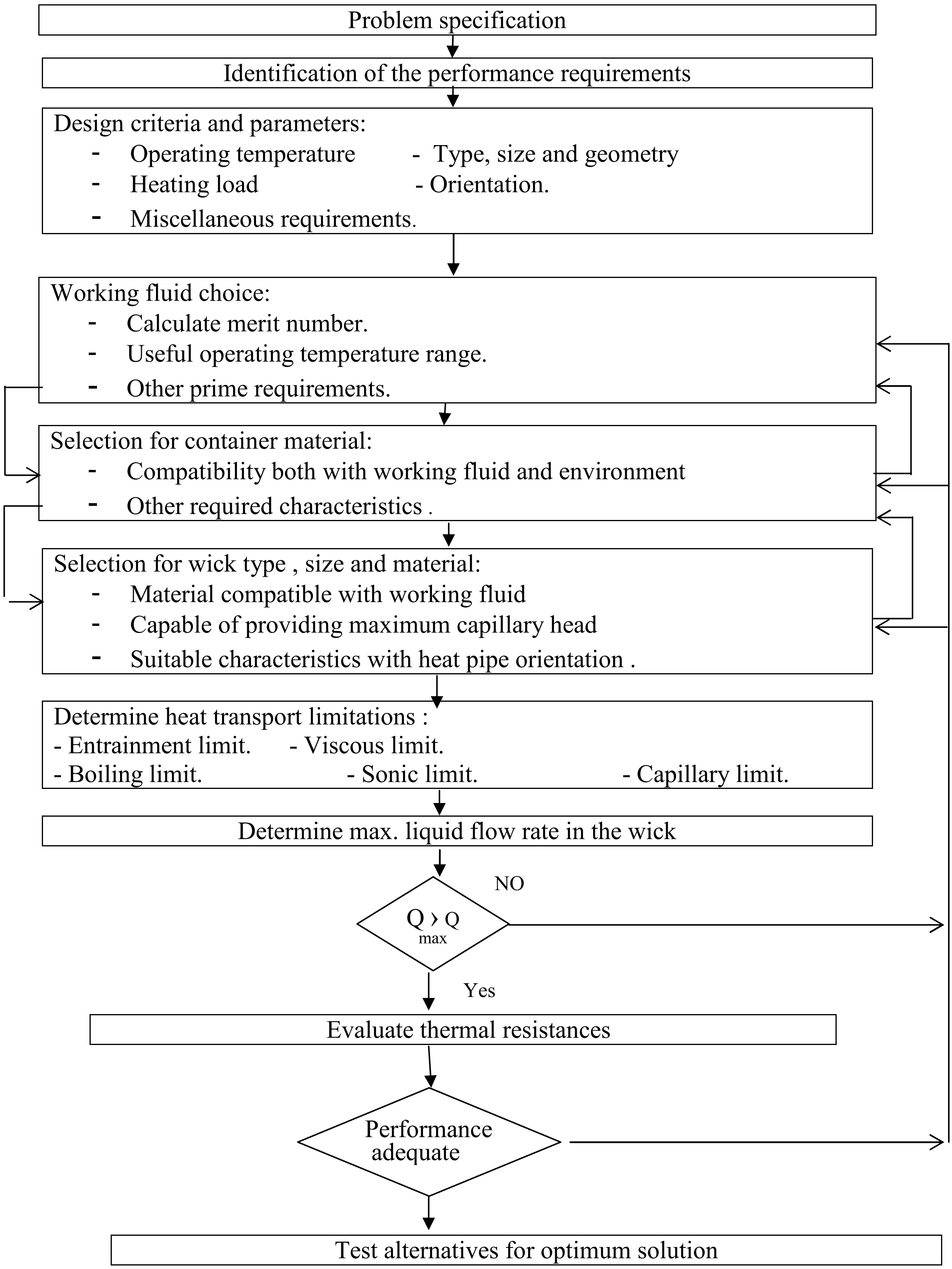



Figure (3): flowchart of heat pipe design for the current research

**3.5 :Heat transport capability** :

A meaningful definition of heat transport capability of the heat pipe requires that [11]:

1- Both liquid and vapor regimes are laminar and momentum effects are negligible.

2- All geometric properties of the wick and heat pipe and the fluid properties are constant along its length .

The driving mechanism in any heat pipe is the heat input (Q), which is related with the mass flow rate of the working fluid in the wick and its latent heat of vaporization, so the maximum heat transport at a given vapor temperature obtained from the equation [8]:

$$Q_{\max} = \overset{\circ}{m}_{\max} . h_{fg} \qquad (3)$$

Using the standard pressure balance equation :

$$\Delta p_c = \Delta p_v + \Delta p_l + \Delta p_g \qquad (4)$$

And neglecting, for the purposes of a first approximation, the vapor pressure drop, we can substitute for the pressure terms and obtain:

$$\frac{2\sigma_l \, Cos\,\theta}{r_{cap}} = \frac{\mu_l}{\rho_l \, h_{fg}} . \frac{QL_{eff}}{A_w K} + \rho_l g L_{eff} \; Sin\beta \qquad (5)$$

Rearranging and substituting for ( $\overset{\circ}{m}$ ), we obtain :

$$m^{\circ} = \frac{\rho_l K \, A_w}{\mu_l \, L_{eff}} [\frac{2\sigma_l}{r_{cap}} Cos\,\theta - \rho_l g L_{eff} \; Sin\beta] \qquad (6)$$

Where, for mesh screen wicks:

$$A_w = 2d_w * \text{no. of layers} * \pi * \text{bore} \qquad (7)$$

And permeability could be determined, for wrapped mesh screen wicks, from the following empirical relation [1]:

$$K = \frac{1}{122} d_w^2 \, \varepsilon^3 / (1-\varepsilon)^2 \qquad (8)$$

Where

$$\varepsilon = 1 - (1.05 \pi \; N d_w / 4) \qquad (9)$$

Also, for mesh screen wick:

$$r_{cap} = 0.5(w + d_w) \qquad (10)$$

**3.6: Heat transfer limitations:**

It is essential for a designer to examine all types of limitations to be sure that the heat pipe works perfectly at high level of performance.

These limits are depending on the working fluid properties, the wick structure, the dimensions and the operational temperature of the heat pipe.

Limitations can be divided into two primary categories: limits that result in heat pipe failure and limits that do not [12].

**3.6.1: Failure limitations:** which include:

**3.6.1.1: Entrainment limit:**

Since the liquid and vapor flow in opposite directions at high enough vapor velocity, liquid droplets may be picked up or entrained in the vapor flow. This entrainment results in excess liquid accumulation in the condenser, and hence dryout the evaporator wick [1] .

Using the weber number, defined as the ratio of the viscous shear force to the force resulting from the liquid surface tension, an expression for the entrainment limit can be found as [13] :

$$Q_{ent,\max} = A_v h_{fg} [\frac{\sigma p_v}{2r_{cap}}]^{1/2} \qquad (11)$$

**3.6.1.2: Capillary limit:** It occurs when the pumping rate is not sufficient to provide enough liquid to the evaporator section and causes evaporator dry out[8]. The maximum heat transfer due to this limit is [2 , 14] :

$$Q_{cap,\max} = [\frac{\rho_l \sigma_l h_{fg}}{\mu_l}][\frac{A_w K}{L_{eff}}][\frac{2}{r_{cap}} + \frac{\rho_l}{\sigma_l} g L_t \sin\beta] \qquad (12)$$

Where: $L_{eff}=0.5L_e+L_a+0.5L_c$ ………….…………………(13)

**3.6.1.3:Boiling limit:** when the vapor bubbles that form in the evaporator wick prevent the liquid from wetting the pipe wall. Severe case of this phenomenon is complete evaporator dry out. An expression for the heat flux beyond which bubble growth will occur, can be written as [14, 15] :

$$Q_{bo,\max} = [\frac{2\pi L_{eff} (k_{eff})_{ws} T_v}{h_{fg} \rho_v \ln(r_i / r_v)}][\frac{2\sigma}{r_n} - \Delta p_{cap,\max}] \qquad (14)$$

**3.6.2: Nonfailure limitations:** which include:

**3.6.2.1: Viscous (Vapor-pressure) limit:**

when the pressure in the vapor core reaches the same magnitude as the vapor pressure in the evaporator, the pressure drop due to flow through the vapor core creates an extremely low vapor pressure in the condenser preventing vapor from flowing in the condenser [13]. Mathematically, this limit can be expressed as [12, 15]:

$$Q_{vis,\max} = \frac{A_v \rho_v h_{fg} r_v^2 p_v}{16 \mu_v L_{eff}} \qquad (15)$$

**3.6.2.2: Sonic limit:** It occurs when the vapor velocity increases along the evaporator and reaches a maximum at the end of the evaporator section [2]. It can be determined from the following relation [12, 14] :

$$Q_{so,\max} = 0.474 A_v h_{fg} (\rho_v p_v)^{1/2} \qquad (16)$$

**3.7: Heat pipe thermal resistance:**

It can be found using an analogous electrothermal network, as illustrated in Figure(4). The overall thermal resistance is comprised of nine different resistances arranged in a series/parallel combination .

Because of the comparative magnitudes of the resistances of the vapor space and the axial resistance of the pipe wall and liquid-wick combinations, the axial resistance can treated as open circuits and neglected. Also, due to the comparative magnitudes, the liquid-vapor interface and the axial vapor resistances can be assumed to be negligible.

This leaves only the pipe wall radial resistances and the liquid – wick resistances at both the evaporator and condenser[1] .

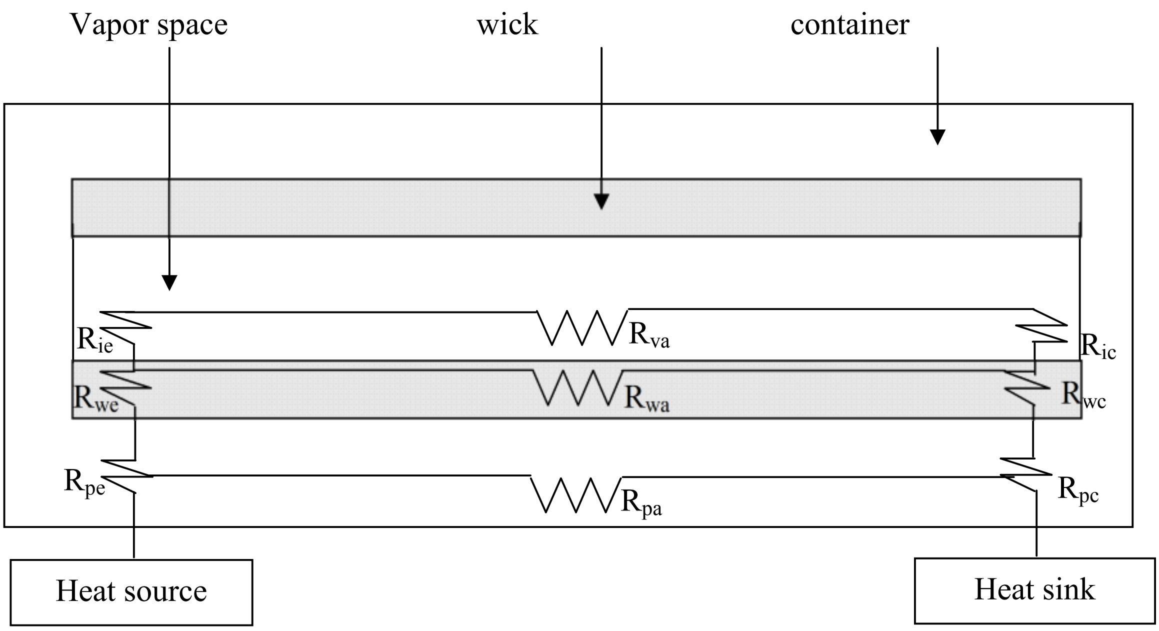



Fig (4) : Equivalent thermal resistance of a heat pipe [1]

The radial resistances at the pipe wall can be computed , for cylindrical pipe from the relation [1, 16] :

$$R_{pe} = \frac{\ln(D_o / D_i)}{2\pi L_e k_p} \qquad (17)$$

Where, $L_e$ is the evaporator length, (or is replaced by the condenser length when evaluating $R_{pc}$).

An expression for the equivalent thermal resistance of the liquid– wick combination in circular pipes is :

$$R_{we} = \frac{\ln(D_o / D_i)}{2\pi L_e (k_{eff})_{ws}} \qquad (18)$$

Where ,for wrapped mesh wick :

$$(k_{eff})_{ws} = \frac{k_w\left[(k_l + k_w) + (1-\varepsilon)(k_w - k_l)\right]}{(k_l + k_w) - (1-\varepsilon)(k_w - k_l)} \qquad (19)$$

Combining these individual resistances, the overall thermal resistance can be computed, and hence the temperature drop between heat source and heat sink can be determined, which

represented by ΔT.
The quantity (L/Ak) is equivalent to a thermal resistance of the heat pipe wall against the flow of heat by conduction, i .e :

$$R = {L}/{Ak} \quad \text{.........(20)}$$

And its reciprocal is the thermal conductance (Ak/L) [2 , 14].

$$R_t = \frac{\Delta T}{Q} \quad \text{.........(21)}$$

$$Q = \frac{kA}{L} \cdot \Delta T \quad \text{.........(22)}$$

$$(k_{eff})_{HP} = \frac{L_t}{R_t A_t} \quad \text{.........(23)}$$

## 4. Results:

The operating conditions at which the heat pipe works are:
Operating temperature =100 ºC
Quantity of heat transport = 2764W

Matlab program, version 7, was used for the optimization of heat transport parameters to reach maximum heat pipe performance at the mentioned operating conditions and for a capillary structure of six layers (100) mesh copper wire screen wick.

A copper container with distilled water as the working fluid were used according to the discussion in sections (3.1, 3.2 and 3.3). The following results were obtained for the heat pipe design parameters and dimensions :

| Parameter | | Value | Unit |
|---|---|---|---|
| 1. | Working fluid mass-flow rate ($\overset{\circ}{m}$) | 0.001224 | kg/s |
| 2. | Max heat transport capability (Q) | 2764 | W |
| 3. | Inside diameter (bore)($D_i$) | 0.048 | m |
| 4. | Outside diameter ($D_0$) | 0.053 | m |
| 5. | Wick cross-sectional area ($A_w$) | 0.000181 | $m^2$ |
| 6. | Effective heat pipe length ($L_{eff}$) | 1.1285 | m |
| 7. | Evaporator section length ($L_e$) | 0.485 | m |
| 8. | Adiabatic section length ($L_a$) | 0.54 | m |
| 9. | Condenser section length ($L_c$) | 0.692 | m |
| 10. | Total heat pipe length ($L_t$) | 1.717 | m |
| 11. | Inclination (tilt) angle (β) | 90 | Degree |

Also, the computed values of heat transport limitations detailed in section (3.6) were listed below :

| $Q_{in}$ (W) | $Q_{ent,max}$ (W) | $Q_{cap,max}$ (W) | $Q_{bo,max}$ (W) | $Q_{vis,max}$ (W) | $Q_{so,max}$ (W) |
|---|---|---|---|---|---|
| 2764 | 23921 | 3014 | 16428 | 1171659 | 20693 |

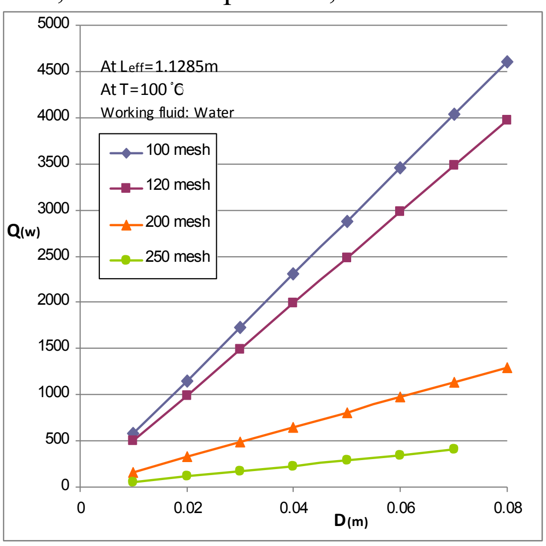
The computed values for thermal resistances, effective thermal conductivity and temperature drop between heat source and heat sink are as follows :

| $R_{pe}$ (ºC / W) | $R_{pc}$ (ºC / W) | $R_{we}$ (ºC / W) | $R_{wc}$ (ºC / W) | Overall R (ºC / W) | $(k_{eff})_{HP}$ (W/mºC) | ΔT (ºC) |
|---|---|---|---|---|---|---|
| $8.25*10^{-5}$ | $5.78*10^{-5}$ | $4.25*10^{-5}$ | $2.98*10^{-5}$ | $1.31*10^{-3}$ | 148751.4 | 3.6 |

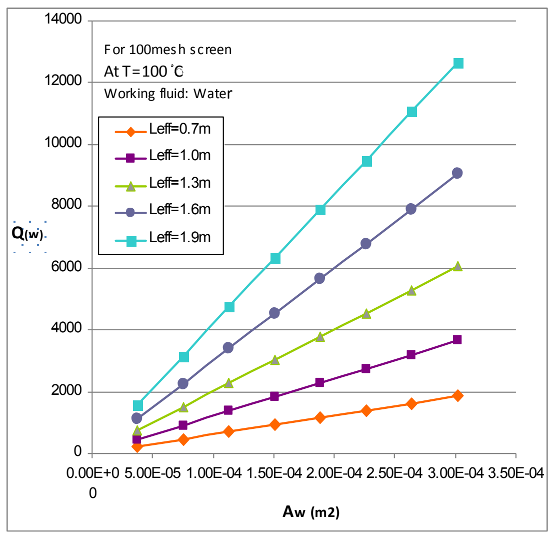
## 5. Discussion:

From the results, it is seen that the condenser section is longer than evaporator section to allow for more heat exchange surface area with water in the water jacket around the condenser to be adequate with the relatively high heat transfer rate needed.

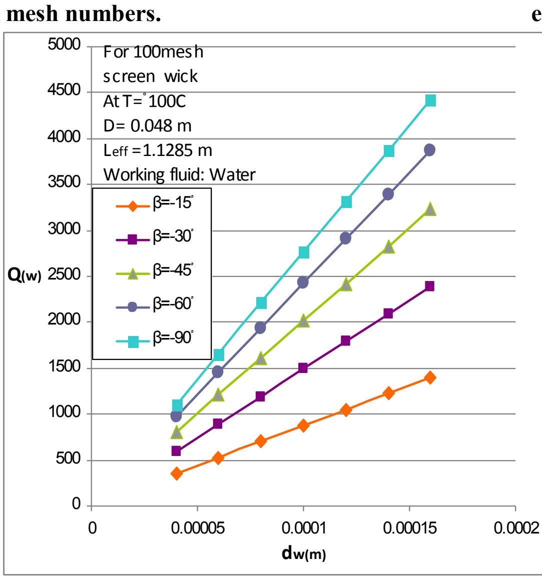
The results shows good agreements with the previous researches [5] and [6] mentioned in the introduction, especially, the relation between heat transport capability and each of the wire diameter, the WF temperature, the HP diameter and pipe cross- sectional area.



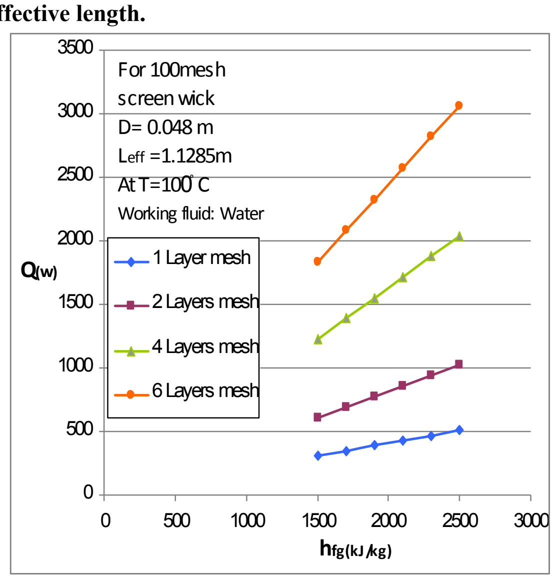

**Figure(5): Variation of heat transport capability with respect to variation of HP bore for different wick mesh numbers.**




**Figure(6): Variation of heat transport capability with the variation of wick c/s area for different HP effective length.**




**Figure(7): Variation of heat transport capability with respect to variation of mesh wire diameter for different tilt angles.**




**Figure(8): The heat transport capability with the variation of WF latent heat of vaporization for different mesh layers**

The results show significant and linear increment of the HP heat transport capability with the raising values for each of the HP bore, wick cross-sectional area, mesh wire diameter, mesh screen number of layers and the WF latent heat of vaporization, which is obvious from the figures (5), (6), (7) and (8) respectively. The reasons were that the heat transfer area increases with the growth of the bore and the wick c/s area, and a larger vapor and liquid flow areas reducing the total thermal resistance and so, raising the heat transfer rate. Also, with the increased values of the HP tilt angle, the gravity force would act with the capillary force in the same direction,increasing the amount of liquid return to the evaporator, reaching maximum value at β equal to 90º(condenser end up) and so, maximize the amount of heat transferred, as shown from figure(7). The same manner for the relation between $d_w$ and Q is obvious from the figure itself.

For higher latent heat of vaporization , more amount of heat would be taken by the working fluid and rejected in the condenser, as in figure (8).

From figure (9) , it is clear that the heat transfer capacity increased in an ascending manner with the increasing values of the HP effective length, because of more surface area for heat exchange, especially for the $L_{eff}$ values more than 1m.

When the values of the WF temperature raises, the heat transfer rate would increase in a descending manner, because the difference of the values for WF properties become smaller as increasing the WF temperature, which is shown in figure (10).

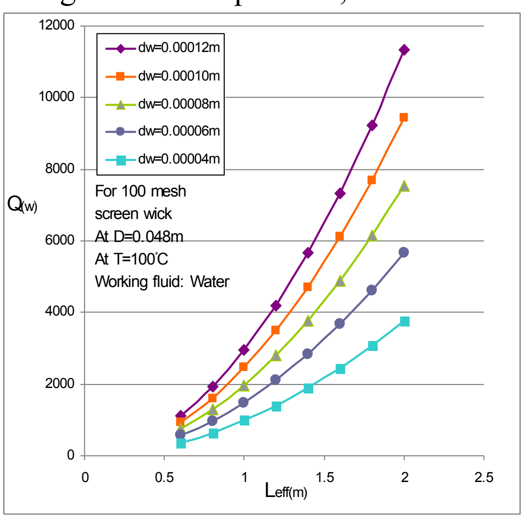
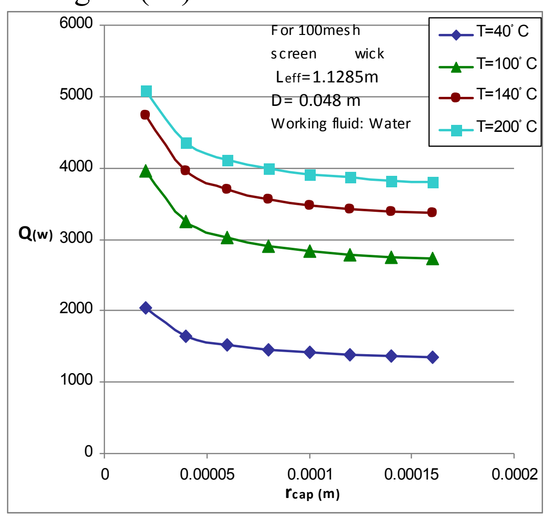



**Figure (9): Variation of heat transport capability with the variation of HP effective length for different mesh wire diameter**




**Figure(10): Variation of heat transport capability with the variation of capillary radius for different WF temperatures**

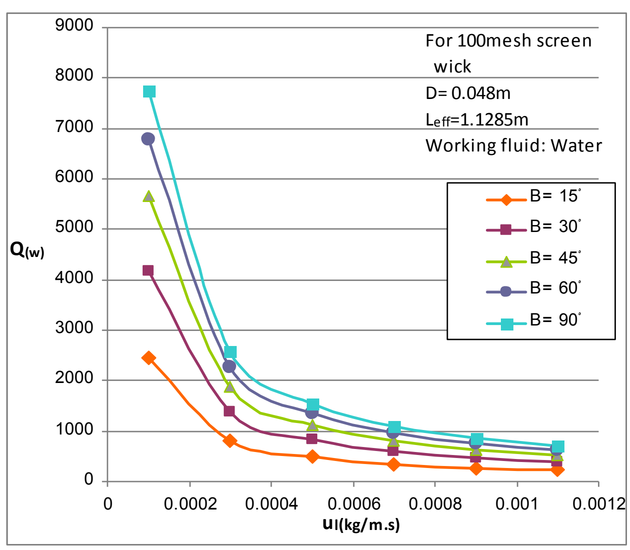
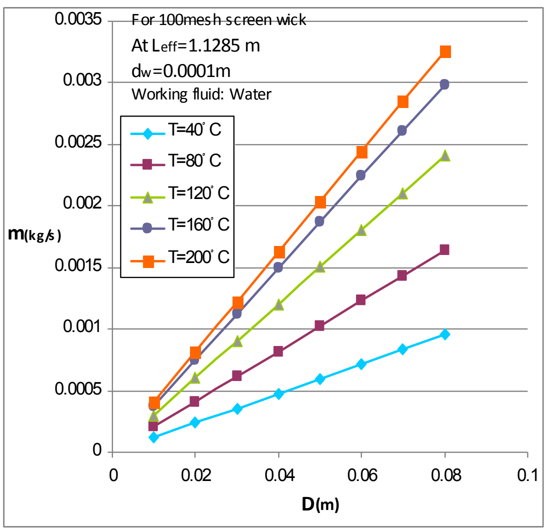


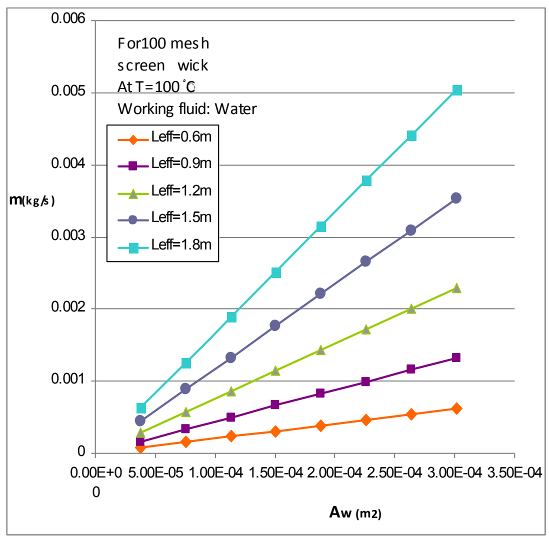

**Figure (11): Variation of heat transport capability with respect to variation of WF viscosity for different tilt angles**

On the contrary, there is an inverse relations between HP heat transport capability and each of the wick mesh number, the capillary radius and the fluid viscosity, and that is shown in the figures (5), (10) and (11) respectively. The reason is that as $r_{cap}$ increased, which comprises half the summation of w and $d_w$ (eq. 10), the wick permeability would decrease, resulting in less amount of liquid pass through the wick back to the evaporator and less amount of heat had been transferred. The same situation with N, which equal 2 $r_{cap}$ for wire screen wick structure.

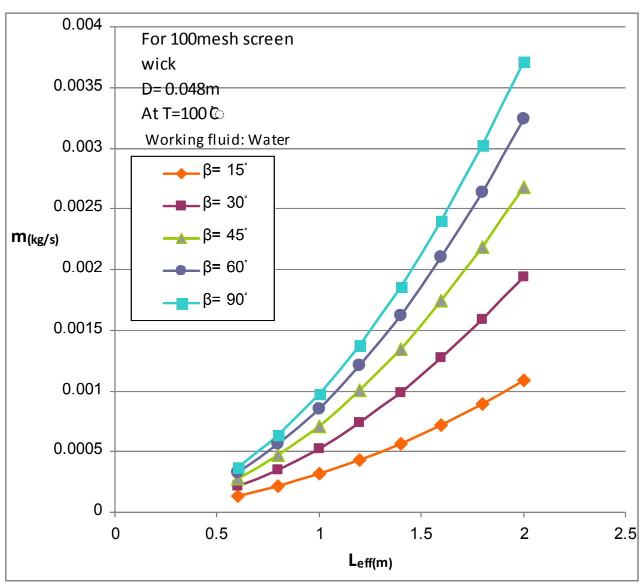
Working fluid mass-flow rate also increased linearly with the increment for each of the HP bore and wick c/s area, whereas with HP effective length in an ascending way and with each of the tilt angle and WF temperature in a descending manner, as shown in figures (12),(13) and (14)respectively, for the same reasons detailed in the first, third and fourth paragraphs of the previous page for their relations with heat transfer rate.




Figure(12): variation of WF mass-flow rate with the variation of HP bore for different WF temperatures




Figure(13): variation of WF mass-flow rate with the variation of wick c/s area for different HP effective length




Figure(14): variation of WF mass-flow rate with the variation of HP effective length for different tilt angles

## 6. Conclusions :

From the present research, the following conclusions can be extracted:

1. The HP heat transfer capability increases linearly and steadily with the increment for each of the HP bore, the wick c/s area, mesh wire diameter, mesh screen number of layers, the WF latent heat of vaporization, and increases ascendingly with the HP effective length, and descendingly with the inclination angle value (up to 90°), and the WF temperature .
2. A contrast relation between the heat transport capacity and each of the capillary radius , the mesh number and the fluid viscosity.
3. The WF mass–flow rate increases with the raising values for each of the bore, wick c/s area, effective length, HP tilt angle(up to 90°) and the WF temperature.

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