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Research Article

Analysis of Effect of Heat Pipe Parameters in Minimising the Entropy Generation Rate

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Heat transfer and fluid flow in the heat pipe system result in thermodynamic irreversibility generating entropy. The minimum entropy generation principle can be used for optimum design of flat heat pipe. The objective of the present work is to minimise the total entropy generation rate as the objective function with different parameters of the flat heat pipe subjected to some constraints. These constraints constitute the limitations on the heat transport capacity of the heat pipe. This physical nonlinear programming problem with nonlinear constraints is solved using LINGO 15.0 software, which enables finding optimum values for the independent design variables for which entropy generation is minimum. The effect of heat load, length, and sink temperature on design variables and corresponding entropy generation is studied. The second law analysis using minimum entropy generation principle is found to be effective in designing performance enhanced heat pipe.

1. Introduction

Over the last few decades, entropy generation minimisation principle [1] has been widely used, especially in thermal systems and engineering application devices since the conventional design methods do not assure the design to be thermodynamically efficient. The extent up to which a given system deviates from an ideal one can be established with the help of entropy generation minimisation principle or thermodynamic optimisation. For thermal systems and their components to work at maximum efficiency condition, irreversibility which is a measure of entropy generation must be minimised. Second law analysis is an effective method for analyzing this generated entropy for the performance improvement of any thermal system. In the present work, a heat transfer device, namely, flat heat pipe, is analysed for which minimum entropy generation is calculated from the heat transfer and fluid flow irreversibility subjected to various constraints.

The heat pipe consists of a sealed container with a lining of porous wick structure adjacent to the wall containing liquid working fluid and a hollow space inside it having vapour medium at the operating conditions [2, 3]. The ability to transport large quantities of heat through a small cross-sectional area over considerable distances with no additional power input makes the heat pipe superior to other conventional methods.

The heat pipe is axially divided into three parts, namely, the evaporator, the condenser, and a transport (adiabatic) section as shown in Figure 1. During the operation of heat pipe, heat applied on the pipe wall of the evaporator section is conducted through the container wall to the wick structure which vaporises the liquid inside the wick. The resulting vapour, due to the pressure difference, moves along the vapour core to the condenser section where it gets condensed releasing the latent heat of vaporisation to the heat sink. The condensed liquid is forced back to the evaporator section due to capillary action of the wick material. Thus heat pipe works continuously transporting latent heat from the evaporator to the condenser as along as sufficient capillary pressure exists in the porous wick structure.

The investigations made in the area of heat pipe and entropy generation minimisation are presented in this section. A detailed overview regarding the heat pipe characteristics, its performance, and challenges has been reported by Faghri [4] and also comments about the various physical

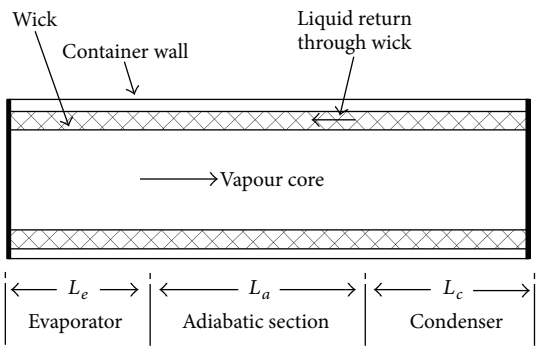

Figure 1: Schematic of flat heat pipe.

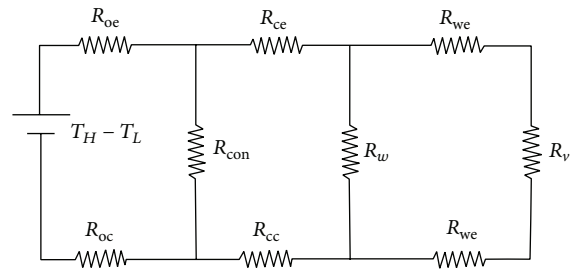

Figure 2: Thermal circuit of flat heat pipe.

phenomena and challenges in this area. A review on factors that contributes to the major transformation in various commercial applications showing great interest in heat pipes is carried out by Faghri [5]. A review of the evolution of thermodynamics during the past few decades has been illustrated by Bejan [6] to provide an insight for various engineered systems. A numerical approach for estimating entropy generation is presented by Kumar and Muraleedharan [7] for the flat heat pipe. Copper-water heat pipe was considered and the model concluded that the entropy generation due to fluid flow is negligible compared to that due to heat transfer. Khalkhali et al. [8] introduced a thermodynamic model of conventional cylindrical heat pipe and detailed parametric analysis was presented in which the effects of various heat pipe parameters on entropy generation were examined. The study showed that minimisation of entropy generation can be accomplished by adjusting the heat pipe dimensions effectively and also by controlling the external heating and cooling conditions. Maheshkumar and Muraleedharan [9] presented the second law analysis for flat heat pipe which strived to minimise the losses developed due to irreversibility using nonlinear programing software. The model was formulated with various heat transfer limits as constraints, which showed that adjustment in heat pipe dimensions will assist in reducing the entropy generation rate. The thermal performance of a miniature cylindrical heat pipe, based on second law of thermodynamics, was carried out by Ghanbarpour and Khodabandeh [10] with water based Al_2O_3 and TiO_2 nanofluids at different concentrations as working substance. A reduction in entropy generation was observed for a good range of nanofluid concentrations. The recent works on entropy generation studies performed on porous media and viscous filled fluid in buoyancy induced flows in channels and enclosures were summarised by Oztop and Al-Salem [11]. A simple procedure for minimising entropy generation in simultaneous heat and mass exchange devices has been illustrated by Narayan et al. [12] using modified heat capacity ratio. Second law analysis has been carried out analytically for a circular tube immersed in an isothermal fluid by Anand and Nelanti [13]. Entropy generation and pumping power to heat transfer ratio of a laminar flow was established to determine appropriate thermal boundary conditions for each fluid with optimum entropy generation. Various entropy generation minimisation models of heat transfer and fluid flow were formulated [14, 15] and performance was compared with different dimensionless parameters. Sahin [16] analytically investigated the entropy generation through a solid slab for both steady and transient states, bringing out the effects of thermal conductivity and internal heat generation on the total rate of entropy generation. A multiparameter constrained optimisation procedure was proposed by Jian-hui et al. [17] to design a plate finned heat sink by minimising the rates of entropy generation.

Although these published research papers over the past few decades show very encouraging results, efforts are not put in to draw a strong conclusion on entropy generation minimisation in heat pipe.

This paper proposes the scope of carrying out parametric investigations to evaluate the effect of various parameters of the heat pipe for optimising the entropy generation rate. This work is motivated by the fact that demand for high precision heat pipes has been increasing rapidly as they are more needed in cooling electronic equipment and spacecraft application. Thus the objective of the present study is to determine the active sites which cause entropy generation in flat heat pipe, estimation of entropy, and its minimisation for enhancing the performance of the system.

2. Entropy Generation Principle

The major factor which results in generation of entropy in a flat heat pipe is heat transfer through a finite temperature difference. The increase in temperature difference will increase the irreversibility associated with the system, thereby increasing the entropy generation rate.

The thermal resistance circuit of a flat heat pipe is shown in Figure 2. The thermal resistances along the vapour core, wick region, and container wall are in several orders of magnitude lower than other resistances due to the heat flow direction from heat source to heat sink and hence can be ignored, which is stated in assumptions. The rate of heat transfer in heat pipe with a nonzero temperature difference is given by

$$Q = \left(\frac{T_H - T_L}{R_{\text{eff}}}\right), \tag{1}$$

where R_{eff} is the effective thermal resistance which is given as $R_{\text{eff}} = R_{\text{oe}} + R_{\text{ce}} + R_{\text{we}} + R_{\text{wc}} + R_{\text{cc}} + R_{\text{oc}}$. The individual thermal resistance for flat heat pipe is obtained from the geometry of heat pipe and thermophysical properties:

R_{oe}, the convective resistance at the outer wall of the evaporator equals $1/h_e A_e$.

R_{ce}, the conduction resistance across container wall at the evaporator equals $t_e/A_e k_e$.

R_{we}, the conduction resistance across the wick structure at the evaporator equals $t_w/A_e k_{eff}$.

R_{wc}, the conduction resistance across the wick structure at the condenser equals $t_w/A_c k_{eff}$.

R_{cc}, the conduction resistance across the container wall at the condenser equals $t_c/A_c k_c$.

R_{oc}, the convective resistance at the outer wall of the condenser equals $1/h_c A_c$.

k_{eff}, the effective thermal conductivity of the wick is

$$k_{eff} = \frac{k_l\left[(k_l + k_w) - (1-\varepsilon)(k_l - k_w)\right]}{\left[(k_l + k_w) + (1-\varepsilon)(k_l - k_w)\right]}. \quad (2)$$

From (1),

$$T_H = T_L + QR_{eff}, \quad (3)$$

$$S_{gen,\Delta T} = \frac{Q}{T_L} - \frac{Q}{T_H}. \quad (4)$$

The rate of entropy generation due to heat transfer through finite temperature difference is given as

$$S_{gen,\Delta T} = Q\left(\frac{T_H - T_L}{T_H T_L}\right) = \frac{Q^2 R_{eff}}{\left[T_L\left(T_L + QR_{eff}\right)\right]}. \quad (5)$$

As the working fluid flows through the different regions of heat pipe, some energy is depleted in overcoming the friction exerted by the wall and the wick structure on the fluid. This energy lost cannot be recovered and hence it becomes a source of irreversibility, which in turn is proportional to the entropy generation rate. Applying first and second laws of thermodynamics for a steady state flow of vapour through the vapour core, entropy generation due to vapour pressure drop is given by

$$S_{gen,\Delta p_v} = \frac{(m\Delta p_v)}{(\rho_v T)}. \quad (6)$$

The expression for vapour phase pressure drop is

$$\Delta p_v = \left(\frac{12\mu_v V l_{eff}}{t_v^2}\right). \quad (7)$$

Taking $Q = mh_{fg}$, the entropy generation due to frictional pressure drop of vapour along the vapour core is obtained as

$$S_{gen,\Delta p_v} = \left(\frac{12\mu_v V l_{eff} Q}{t_v^2 \rho_v T h_{fg}}\right). \quad (8)$$

Liquid pressure drop in the wick through which liquid flows is given by Darcy's law:

$$\Delta p_l = \left(\frac{8\mu_l m l_{eff}}{\rho_l \varepsilon t_w w r_{cap}^2}\right). \quad (9)$$

Then the entropy generation due to liquid pressure drop in the wick is

$$S_{gen,\Delta p_l} = \left(\frac{8\mu_l Q^2 l_{eff}}{\rho_l^2 \varepsilon t_w w r_{cap}^2 T h_{fg}^2}\right). \quad (10)$$

Thus the objective function is to minimise the total rate of entropy generation which is expressed as

$$S_{gen,Total} = S_{gen,\Delta T} + S_{gen,\Delta p_v} + S_{gen,\Delta p_l}. \quad (11)$$

3. Constraints

The constraints constitute the limitations on the heat transport capability.

3.1. Heat Transfer Limits. The limits of heat transfer play a significant role in the design of the heat pipe since the lowest limit defines the maximum heat transport limitation at a given temperature. The parameters limiting heat transport in the conventional heat pipe are capillary limit, sonic limit, viscous limit, entrainment limit, and boiling limit.

3.1.1. Capillary Limit. The ability of the capillary wick structure to provide continuous circulation of working fluid is limited with a constraint or limit called capillary limit or hydrodynamic limitation. When the heat transfer is increased above this limit, pumping rate of the working fluid will not be sufficient to provide enough liquid to the evaporator section. This occurs due to the fact that maximum capillary pressure that the wick can sustain is lower than the sum of liquid and vapour pressure drops in the heat pipe. The expression for the capillary limitation is given as

$$Q_{cap} = \frac{h_{fg}\left[\left(2\sigma_l/r_{cap}\right)\cos\theta + \rho_l g l \sin\alpha\right]}{\left(2\mu_v/t_v^3\rho_v + \mu_l/2Kt_w\rho_l\right)}. \quad (12)$$

The constraint for the limit is expressed as $Q - Q_{cap} \leq 0$.

3.1.2. Sonic Limit. At higher temperatures, when the vapour flow velocity becomes equal to or higher than the sonic velocity, it results in a choked flow condition. This limit is called sonic limit. There is a maximum axial heat transfer rate at this limit and subsequently it does not increase by decreasing the condenser temperature under choked flow condition. The expression for the limit can be written as

$$Q_s = 0.5 h_{fg} t_v w \left(p_v \rho_v\right)^{0.5}, \quad (13)$$

where $Q - Q_s \leq 0$.

3.1.3. Entrainment Limit. There exists shear forces at the liquid-vapour interface, since both the vapour and the liquid inside the heat pipe move in the opposite direction. When the corresponding velocities of vapour and liquid are relatively high, a limit is reached where liquid will be torn off from the wick surface and entrained to the vapour region flowing towards the condenser section. When the entrainment begins

and becomes predominant, it results in dry out of the wick at the evaporator section. The entrainment heat load for flat heat pipe is obtained as

$$Q_{\text{ent}} = wt_v h_{fg} \left(\frac{2\pi\sigma\rho_v}{r_{\text{hs}}} \right)^{0.5}. \tag{14}$$

The limit of entrainment can be represented as $Q - Q_{\text{ent}} \leq 0$.

3.1.4. Viscous Limit. At low operating temperature the vapour pressure difference between the evaporator and condenser end will be small. Viscous forces may be dominant and limit the operation of the heat pipe. This flow condition where total vapour pressure in the vapour region becomes insufficient to sustain an increased flow is called viscous limit. The expression for viscous limit is obtained as

$$Q_{\text{vis}} = \left(\frac{t_v^3 w h_{fg} p_v \rho_v}{24\mu_v l_{\text{eff}}} \right), \tag{15}$$

where the limit is written as $Q - Q_{\text{vis}} \leq 0$.

3.1.5. Boiling Limit. Heat transfer across the liquid saturated wick is accompanied by a transverse temperature gradient in the liquid. When this heat flux in the evaporator becomes high, the wall temperature becomes excessively high resulting in boiling of liquid in the evaporator. The vapour bubbles that form in the wick structure cause hot spots and obstruct the circulation of fluid which leads to dry out of the wick in the evaporator. This limitation of heat transfer due to transverse heat flux density is termed as boiling limitation. Unlike the other heat pipe limits discussed above which are due to axial heat flux limitation, the boiling limitation is a transverse heat flux limit. The expression for the boiling heat transfer limit is obtained as

$$Q_b = \frac{2wL_e k_{\text{eff}} T \left[(2\sigma/r_n) - P_c \right]}{\left(h_{fg} \rho_v t_w \right)}. \tag{16}$$

The constraint of $Q - Q_b \leq 0$ should be satisfied.

Thus the present optimization problem can be formulated to find X which minimizes the objective function $F(X)$ subjected to constraints: $g_i(X) \leq 0$, $i = 1, 2, 3, \ldots, m$ where $X = N, t_w, T_L, L_a$.

$F(X)$ is the entropy generation in a heat pipe system.

$g_i(X)$ is the functional constraints.

m is the number of constraints.

4. Methodology

In the present work a flat heat pipe is analysed with following assumptions.

(1) The operation of the heat pipe is carried out at steady state.

(2) The fluid is laminar and incompressible.

Table 1: Thermophysical properties taken as input parameters for design.

Thermophysical properties	Design value
Density of water	985 kg/m^3
Density of water vapour	0.13 kg/m^3
Viscosity of water	0.000797 N·s/m^2
Viscosity of water vapour	0.0000134 N·s/m^2
Surface tension of water	0.070 N/m
Latent heat of vapourisation	2300000 J/kg
Thermal conductivity of water	0.608 W/mK

(3) The heat transfer through the liquid wick is modelled as pure conduction with an effective thermal conductivity.

(4) The temperature difference which exists within the liquid-vapour interface between the vapour core and wick structure is small and neglected.

(5) The axial heat conduction through the container wall and wick is neglected.

Analysis has been carried out for a heat pipe with horizontal orientation. The heat input and sink temperature are taken as input parameters while surface temperature of the evaporator is switched as a function of heat input, sink temperature, and thermal resistance in the heat pipe. The selection of working fluid is important since it depends on the operating temperature of the heat pipe and the compatibility with the material of the container wall and the wick. Copper is used as the material for the container wall and wrapped wire wick (mesh number = 4000). Thermal conductivity of copper is taken as 385 W/mK. The working fluid used is water. The thermophysical properties of water at corresponding operating temperature and pressure taken for the analysis are summarised in Table 1. Heat transfer coefficient is taken as 600 W/m^{2}K, assuming evaporator and condenser section being maintained at desired temperature by circulating water in a constant temperature bath.

A flat heat pipe of one-meter length (combining length of evaporator, adiabatic section, and condenser) is considered for the analysis. The length of adiabatic section is varied from 0 to 0.5 m to study the effect of variation in entropy generation rate. The wick thickness is taken as a design variable and its value is assumed to be varying from 0.0005 to 0.0015 m. The lower and upper limit values are specified in the program as an additional constraint. Meanwhile, the length of evaporator and condenser is influenced by the length of adiabatic section, since the latter is a design variable. The linear dimension in the transverse direction (i.e., perpendicular to the flow direction) cannot be defined since it is determined by the sum of the container wall, wick, and vapour core thicknesses, which varies according to design requirement in each case. However upper bound limit to the transverse length of the heat pipe is fixed to 0.05 m as a design constraint.

The present problem is a nonlinear programming problem with nonlinear inequality constraint. The optimisation or rather minimisation is to obtain a set of design variables

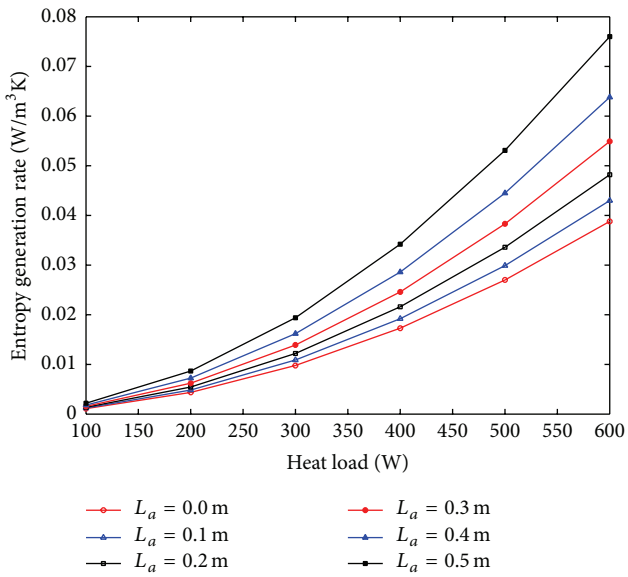

Figure 3: Variation in entropy generation rate against heat load for different lengths of adiabatic section.

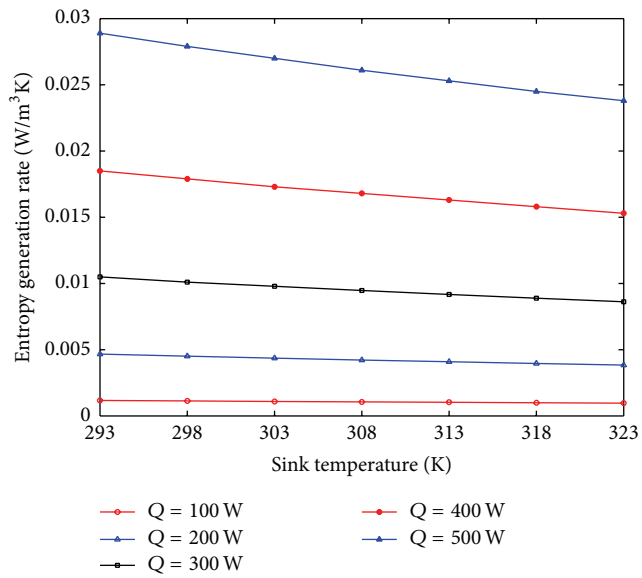

Figure 4: Variation in entropy generation rate against sink temperature for different heat loads.

which gives minimum entropy generation. The problem is modelled using optimisation modelling software "LINGO 15.0." It provides completely integrated built-in solver for solving the nonlinear optimisation model to get global optimum solution. The entropy generation rate is calculated with respect to different heat pipe parameters.

5. Results and Discussion

The variation of entropy generation rate is studied for different heat pipe parameters such as heat load, sink temperature, and various lengths of adiabatic section using the software. From the present analysis, it is inferred that entropy generated in a heat pipe is mainly due to heat transfer, and the entropy generated due to fluid friction of the vapour and liquid flow is negligibly small. Hence the dimensionless parameter called Bejan number (ratio of entropy generated due to heat transfer to total entropy generated or sum of entropy generated by heat transfer and fluid friction) tends to 1 in this problem, which clearly indicates that heat transfer irreversibility contributes almost 100% to the total entropy generation. Figure 3 shows the variation of entropy generation against heat load for different lengths of adiabatic section with a constant sink temperature of 303 K. Entropy generation rate is found to increase with heat load, and the rate of increase is more when heat load increases. This is due to the fact that, with the increase in heat load, temperature difference between the walls of evaporator and condenser increases and moreover increase in pressure drop also results in entropy generation rate. For the same operating conditions, entropy generation rate is found to increase with increase in the length of adiabatic section. As the length of the transport section increases, effective length of the heat pipe increases; this in turn increases pressure drop in the liquid wick and vapour core along the heat pipe which results in increase of entropy generation rate. The rate of increase is found to be more at higher heat flux values. Entropy generation rate is found to increase from $1.09 * 10^{-3}$ to $3.88 * 10^{-2}$ W/m^{3}K for the heat input range of 100 to 600 W without the transport section. The entropy rate is found to be almost doubled when the length of adiabatic section is varied from 0 to 0.5 m, for the corresponding heat flux value.

The decrease in entropy generation rate with the increase in condenser temperature without the transport region is depicted in Figure 4. The increase in sink temperature results in drop of temperature difference between the heat source and sink which decreases the irreversibility associated with it, thereby decreasing the entropy generation rate. For low values of heat load, there is not much difference in the entropy generation rate for different sink temperatures. The variation is found to be $2.07 * 10^{-4}$ W/m^{3}K for a heat load of 100 W over the entire sink temperature range. But at higher heat load, the variation is found to be comparable. At heat load of 500 W the variation is found to be $5.1 * 10^{-3}$ W/m^{3}K for the entire sink temperature range. Entropy generation rate is found to increase by 25 times when the heat load is increased from 100 to 500 W. The graph shows similar variation pattern for the entire range of sink temperature. For a constant heat load the variation in entropy generation rate against the length of adiabatic section for different sink temperatures is depicted in Figure 5. The increase in entropy generation rate is due to the increase in effective length of heat pipe. Similarly the decrease in entropy generation rate with increase in sink temperature for different lengths of adiabatic section is observed with a constant heat load. The decrease is due to the fall in finite temperature difference when the sink temperature is increased. The variation of entropy generation for the entire sink temperature values with a constant heat load of 200 W is found to be almost uniform.

Figure 6 shows the entropy generation rate for different values of mesh number. The decrease in entropy generation

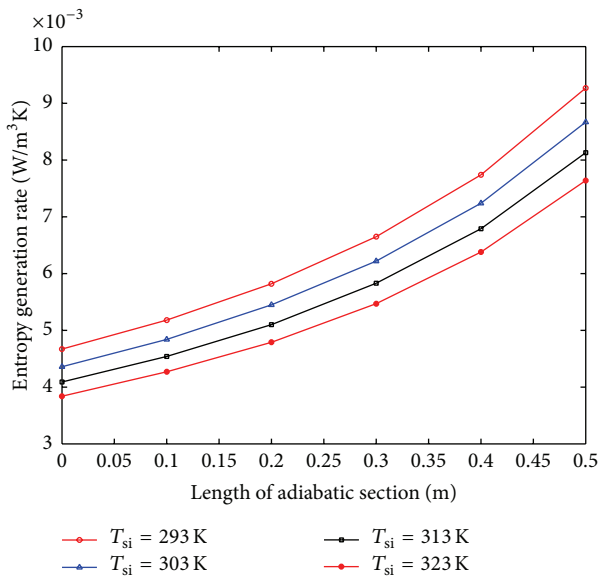

FIGURE 5: Variation in entropy generation rate against length of adiabatic section for different sink temperatures.

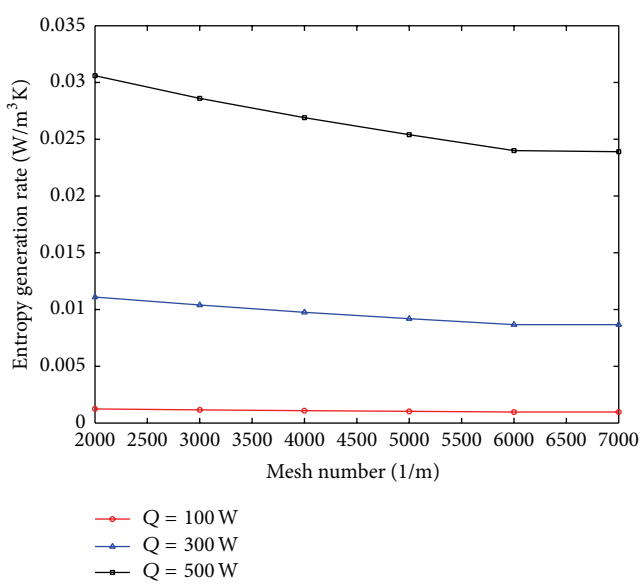

FIGURE 6: Variation in entropy generation rate against mesh number for different heat loads.

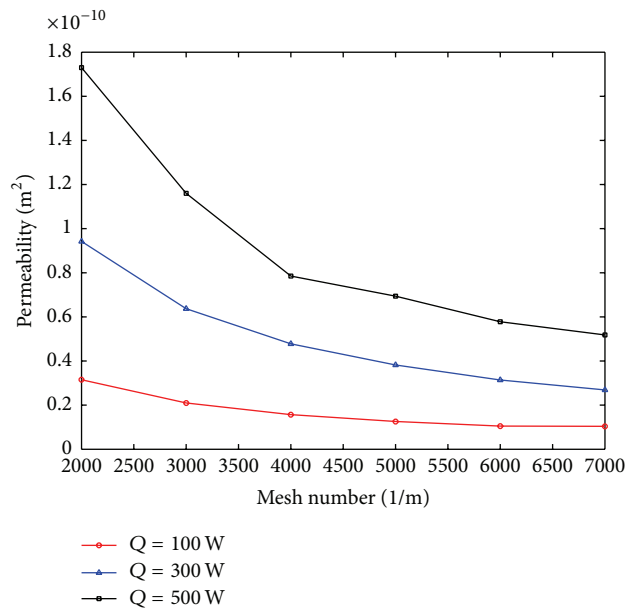

FIGURE 7: Variation in permeability against mesh number for different heat loads.

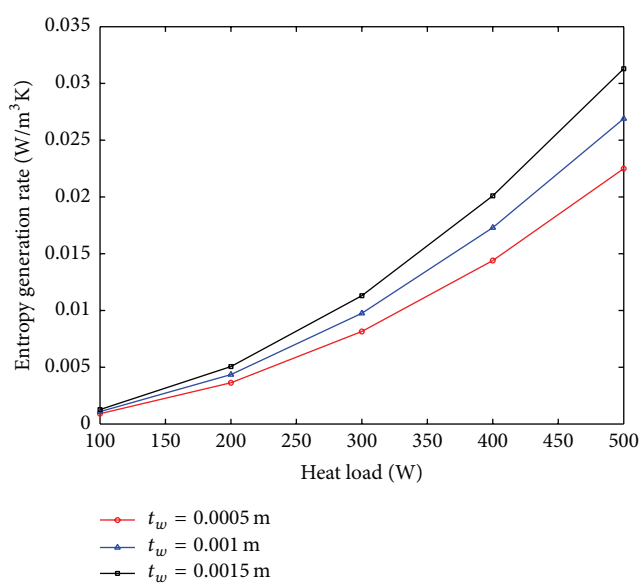

FIGURE 8: Variation in entropy generation rate against heat load for different wick thicknesses.

rate with the increase in mesh number is due to the effect of equivalent thermal conductivity, which in turn depends on effective resistance. With the increase in heat load, the pressure drop will be high which leads to increase in entropy generation rate. But with the increase in mesh number, the capillary pressure will be increased which in turn provides better circulation of working fluid; in that case, the entropy generation rate will be decreased. Permeability and corresponding values of mesh number are shown in Figure 7. Permeability is found to increase with the decrease in mesh number. The mesh number is inversely proportional to the surface pore size. The increase in pore size makes the liquid flow from the condenser to the evaporator more easily due to the increase in the flow area inside the wick thereby increasing the permeability.

The variation of entropy generation rate with respect to different heat loads for various wick thickness is shown in Figure 8. While reducing the wick thickness, the thermal resistance decreases and hence the entropy generation rate due to heat transfer also decreases. So decreasing the wick thickness is one of the methods for decreasing the entropy generation rate. For a heat load of 100 W the difference in entropy generation rate over 0.001 m thickness is found to be $3.6 * 10^{-4}$ W/m^{3}K. But the difference increases to $8.8 * 10^{-3}$ W/m^{3}K for the heat load of 500 W. The comparison of

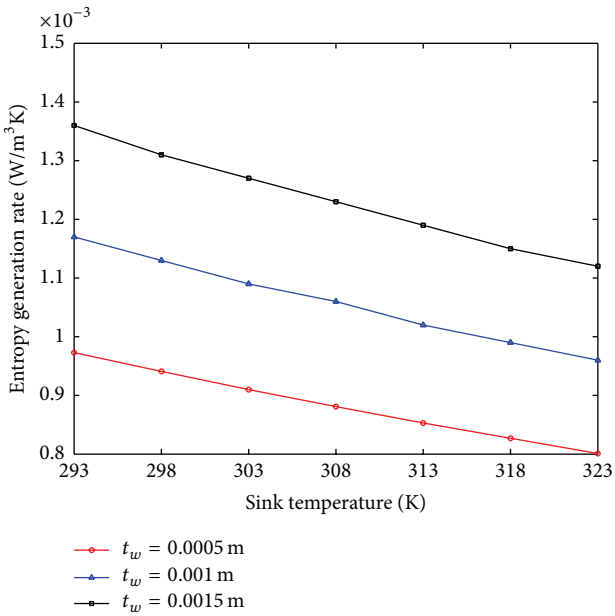

FIGURE 9: Variation in entropy generation rate against sink temperature for different wick thicknesses.

entropy generation rate at different wick thicknesses shows that the variation is more prominent for higher values of heat load.

Figure 9 depicts the variation of entropy generation rate against sink temperature for different wick thicknesses. As the sink temperature increases, there will be a corresponding decrease in the temperature difference between the heat source and the sink which results in decrease of entropy generation rate. The reduction in entropy generation rate with decrease in wick thickness is due to the same reason as mentioned in the above case.

6. Conclusions

Entropy generation rate associated with the flat heat pipe has been estimated and minimised for reducing the irreversibility. The model has been formulated and solved as a nonlinear programming problem with nonlinear functional constraints using commercial software (LINGO 15.0). The analysis shows that fluid friction and heat transfer contribute to entropy generation and has to be considered for the improvement in the performance of the device. Entropy generation is found to increase with heat load and length of the adiabatic section, while a decrease in the former is observed with increase in sink temperature. The following design modification can be recommended to improve the system performance by reducing entropy generation rate:

(i) Reducing the length of the adiabatic section as far as possible.

(ii) Increasing the condenser temperature.

(iii) Decreasing the wick thickness.

(iv) Increasing the mesh number.

The parametric study carried out employing the second law analysis facilitated a way in arriving at the optimal design of high capacity heat pipe.

Nomenclature

A: Area (m^2)
h: Heat transfer coefficient (W/m^2K)
h_{fg}: Latent heat of vaporisation (J/kg)
k: Thermal conductivity (W/mK)
K: Permeability (m^2)
l: Length (m)
m: Mass flow rate (kg/s)
N: Mesh number (1/m)
p: Pressure (Pa)
P_C: Capillary pressure (Pa)
Q: Heat transfer rate (W)
r: Radius (m)
R: Resistance (K/W)
S: Entropy (J/kgK)
t: Thickness (m)
T: Temperature (K)
V: Velocity (m/s)
w: Width (m).

Greek Symbols

μ: Viscosity (Ns/m^2)
ρ: Density (kg/m^3)
σ: Surface tension (N/m)
ε: Porosity
θ: Contact angle (degree)
α: Inclination angle (degree).

Subscripts

a: Adiabatic section
c: Condenser section
e: Evaporator section
eff: Effective
gen: Generated
hs: Hydraulic radius
H: Source
l: Liquid
L: Sink
n: Nucleation
o: Outside
v: Vapour
w: Wick
b: Boiling
cap: Capillary
con: Container
ent: Entrainment
s: Sonic
vis: Viscous.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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