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Investigation of Entropy Generation Through the Operation of an Unlooped Pulsating Heat Pipe

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ABSTRACT

In the present study, an unlooped pulsating heat pipe has been considered with two liquid slugs and three neighboring vapor plugs. The governing equations including momentum, energy and mass equations are solved explicitly, with the exception of the energy equation of liquid slugs. The aim of the present study is to calculate the entropy generation through the performance of a pulsating heat pipe. Additionally, the effects of different pipe diameters and evaporator temperatures have been investigated. Besides, Bejan number has been derived for each case study to investigate the share of heat transfer in entropy generation. The results show that by increasing the pipe diameter, the sensible and latent heat transferred into the pulsating heat pipe enhance and the liquid slugs oscillate at high amplitudes. On the other hand, the entropy generation value increases as the pipe diameter increases. The evaluated Bejan numbers show that the share of viscous effects in entropy generation decreases with increasing pipe diameter. Previous studies have reported that there is a threshold of pulsating heat pipe diameter to operate. However, the results of the present work demonstrate that the use of pulsating heat pipes are not feasible in small diameters. Moreover, the results show that the heat removing performance of pulsating heat pipe improves as the temperature difference of the evaporator and condenser increases. Ourresults demonstrate increments in total entropy generation in high evaporator temperatures. Moreover, the Bejan number will increase by any increment in the evaporator temperature and this phenomenon reveals the insignificant role of viscous effects in high evaporator temperatures. To validate the calculations, the results have been compared to those of the previous works. This comparison shows very good agreement.

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Nomenclature

A	Tube cross sectional area, (m^2)
C_l	Friction coefficient
C_p	Specific heat (constant pressure), (J/Kg.K)
D	Diameter, (m)
h	Enthalpy, (J/Kg)
h_{fg}	Latent heat, (J/Kg)
h_{lsen}	Local sensible heat transfer coefficient, (W/m^2.K)
K	Thermal conductivity, (W/m.K)
$\dot{m}$	Mass flow rate,(Kg/s)
Nu	Nusselt number
P	Pressure, (Pa)
Pr	Prandtl number
q'	Heat flux per unit length, (W/m)
Q	Heat transfer, (W)
R	Gas constant of vapor, (J/Kg.K)
Re	Reynolds number
s	Specific entropy, (J/Kg.K)
$\dot{S}'$	Entropy generation per unit Length, (J/m.K)
T	Temperature, (K)
t	Time, (s)

Greek Symbols

α_l	Thermal diffusivity of liquid, (m^2/s)
μ	Dynamic viscosity, (Kg/m.s)
ρ	Density, (Kg/m^3)
τ	Shear stress, (N/m^2)

Subscripts

F	Flow
gen	Generation
Lat	Latent
li	i^{th} liquid plug
Sen	Sensible
th	Thermal
T	Total
vi	i^{th}vapor plug
W	Wall

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1. INTRODUCTION

The pulsating heat pipe (PHP) is a new type of heat exchanger that was first introduced by Akachi in 1990s. Pulsating heat pipe is a two-phase heat transfer device that transfers heat from heating section to cooling section via oscillatory liquid – vapor two-phase flow. There are two types of PHPs: the unlooped and looped PHPs that have the same thermal performance, when circulation does not occur in looped PHPs.

Nowadays, research in the analysis of PHPs has been grown in both numerical and experimental branches. Sejung Kim et al. [1] examined the fluctuation effects of heating and cooling section temperatures on the oscillatory flow, temperature and pressure of the vapor plugs, as well as latent and sensible heat transfer of a pulsating heat pipe.

The results show the effect of both amplitude and frequency of periodic fluctuation, on pressure and temperature of the vapor plugs, liquid slug displacement, and latent and sensible heat transfer. In another study, Yuan et al. [2] examined the effects of gravity, liquid slug displacement, and percentage contribution of latent heat in total heat transfer in a pulsating heat pipe. The results show that if the slug flow pattern in the system remains steady, more than 80 percent of the total heat transferred will be sensible heat. Zhang et al. [3] analyzed numerically the Liquid – vapor pulsating flow in a vertical U – shaped pipe. The results indicate that the initial liquid slug displacement has insignificant effect on the amplitude and angular frequency of the oscillations.

Shafii et al. [4] studied both of looped and unlooped pulsating heat pipes with multiple liquid slugs and vapor plugs. They found that heat transfer in both unlooped and looped PHPs were mainly due to the exchange of sensible heat. Accordingly, evaporation and condensation have considerable effects on the performance of the PHPs. Also, the study indicated that by increasing the diameter of the PHP (both looped and unlooped), the average total heat transfer increases. Hao Peng et al. [5] studied the effects of mass transfer, fluid filling ratio, operating temperature, gravity and pressure drop of the bend, on the heat transfer of an unlooped PHP. P. Frank Pai et al. [6] studied a PHP with three vapor plugs. They found that oscillation frequency increases when the masses of liquid slugs decrease.

Zhang et al. [7] investigated heat transfer in the evaporator and condenser sections of a pulsating heat pipe (PHP) with an open end. These results indicated that with decreasing pipe diameter and wall temperature of the heating section, the amplitude of oscillation decreases. Shao et al. [8] reported a mathematical model based on the thin film of liquid in the condenser and evaporator sections. They also highlighted the significant role of the sensible heat transfer and evaporator temperature effect on the heat transfer enhancement.

Ma et al. [9] modeled a single turn PHP considering the thermal energy as a driving force on the liquid slug. They reported the effect of filling ratio and pipe characteristics on the oscillation amplitude and frequency of a liquid slug. Mameli et al. [10] investigated the effect of tube turns on the operation of PHPs under the condition of fixed heat flux at the pipe boundary. Moreover, different working fluids and PHP orientation have been studied in the paper. Khandekar et al. [11] used empirical data and presented a correlation to obtain heat flux in PHPs. They highlighted the importance of PHP modeling by consideration of Boiling and condensation processes.

Experimental investigations of the PHPs are mainly focused on the heat transfer measurements. Flow pattern and bubble formation in closed loop pulsating heat pipes are presented in Tong et al. study [12]. They clearly showed the bubble formation or bubble collapsing mechanism along the pipe length and bends. Khandekar et al. [13] conducted CLPHPs experimentally. Results of vertical and horizontal orientations are reported in the paper with thermodynamic considerations.

Charoensawan et al. [14] used an experimental setup to study PHPs at constant evaporator and condenser wall temperature. Results are presented for different working fluids and internal diameter of pipe. It is shown that orientation of the heat pipe has a significant influence on the total heat transfer. A single turn CLPHP has been studied in Khandekar et al.'s work [15]. They showed that a single turn PHP represents the same thermos-physical characteristics of a multi-turn pulsating heat pipe. Mameli et al. [16] represented experimental results of CLPHP under different heat fluxes and system orientations. They reported an optimal filling ratio for a specified condition and orientation. The effects of working fluid physical properties on PHP flow and heat transfer are investigated experimentally in Han et al.'s work [17]. The study emphasizes on the role of viscosity and phase change parameters on the PHP heat transfer performance.

The oscillation flow of liquid and vapor slugs lead to the irreversibilities in the PHP system. Recently, the entropy generation in pipes has been the subject of study in many investigations [18-23]. Most of the reports highlight the entropy generation analysis to reach an optimum design. In the present study, the second law of thermodynamics and the entropy generation through the operation of an unlooped pulsating heat pipe have been investigated numerically.

2. PHYSICAL MODEL

An unlooped pulsating heat pipe is shown in Figure 1. The unlooped PHP has three evaporator sections with length of L_h and includes two condenser sections with length of $2L_c$. The liquid slugs with lengths of L_P are at the bottom section of the pipe and the vapor plugs are at the heating section. The temperature of evaporator and condenser sections are T_e and T_c , respectively. The linear schematic of the heating pipe is presented in Figure 1b. The liquid slugs oscillate, due to the pressure difference between the neighboring vapor plugs, to the left and right side. When the liquid slug moves toward the right evaporator section, evaporation occurs and its mass, temperature and pressure increase. On the other hand, the left vapor plug transmits to the condenser, and consequently condensation phenomenon occurs. Therefore, the pressure of the left vapor plug decreases. This pressure difference makes the liquid slug return toward the cooling section. This phenomenon continuously repeats to all liquid slugs and vapor plugs.

2.1. Governing Equations

The oscillatory phenomenon in PHPs can be predicted by solving the mass, momentum and energy equations for each liquid slug and vapor plug.

The continuity equation for i^{th}liquid slug can be obtained from the following equation:

$$\frac{dm_{li}}{dt} = \dot{m}_{in.li} - \dot{m}_{out.li} = \frac{1}{2}\left(\frac{dm_{vi}}{dt} + \frac{dm_{v(i+1)}}{dt}\right) \quad (1)$$

The momentum equation for i^{th}liquid slug is calculated by :

$$\frac{dm_{li}v_{li}}{dt} = \left[\left(P_{vi} - P_{v(i+1)}\right) - \Delta P_b\right]A - \pi d L_{li}\tau - (-1)^n m_{li} g \quad (2)$$

$\left(P_{vi} - P_{v(i+1)}\right)$ is the pressure difference between two vapor plugs (or pressure difference between the ends of the liquid slugs). ΔP_b is the pressure drop caused by the tube bend. τ is the shear stress between liquid slug and the tube wall that is determined as follows:

$$\tau = \frac{1}{2} C_{li}\rho_l v_{li}^2 \quad (3)$$

Due to the regime of flow (laminar or turbulent), friction coefficient can be determined as:[24]

$$C_{li} = \begin{cases} \frac{16}{Re} & Re \leq 1180 \\ (0.79\ ln(Re_D) - 1.64)^{-2} & Re > 1180 \end{cases} \quad (4)$$

The PHP is not assumed to be a straight tube. Thus, the pressure drop due to the bends are considered in the present study:[25]

$$\Delta P_b = \sum_{k=1}^{N_t} \frac{K_k}{d}\rho_l v_l^2 \quad (5)$$

in which

$$K_k = \frac{K_{Re}}{Re_l} + K_r\left(1 + \frac{K_d}{(\frac{d}{0.0254})^{0.3}}\right) \quad (6)$$

This empirical equation depends on three main parameters that have constant values in the case of a 180^oturn (K_{Re}=1000, K_r=0.1 and K_d=4). [25, 26]

The continuity equation for i^{th}vapor plug can be obtained from the following equation:

$$\frac{dm_{vi}}{dt} = \dot{m}_{in.vi} - \dot{m}_{out.vi} \quad (7)$$

$\dot{m}_{in.vi}$ is the mass rate transferred into the vapor plugs resulting from evaporation. $\dot{m}_{out.vi}$ is the mass rate transferred out of the vapor plugs due to condensation. These parameters can be calculated using the following equations:

$$\dot{m}_{in.vi} = h_h \pi d L_{li}\frac{(T_{vi} - T_h)}{h_{fg}} \quad (8a)$$

$$\dot{m}_{out.vi} = h_c\,\pi d L_{li}\frac{(T_c - T_{vi})}{h_{fg}} \quad (8b)$$

The energy equation for a vapor plug is:

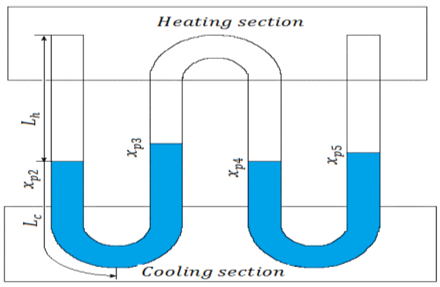
$$\frac{dm_{vi}u_{vi}}{dt} = \dot{m}_{in\,.vi}\,h_{in\,.vi} - \dot{m}_{out\,.vi}\,h_{out\,.vi} - P_{vi}\frac{dV_{vi}}{dt} \quad (9)$$

With regard to the enthalpy ($h = C_P T$) and internal energy ($u = C_V T$) relationships for an ideal gas, the energy equation of a vapor plug can be rewritten as:

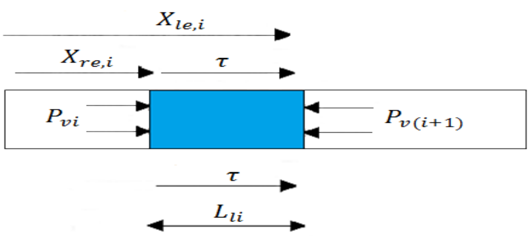
$$m_{vi}C_v\frac{dT_{vi}}{dt} = (\dot{m}_{in.vi} - \dot{m}_{out.vi})RT_{vi} - P_{vi}A\frac{dX_{vi}}{dt} \quad (10)$$

The pressure of the i^{th} vapor plug is obtained from the ideal gas equation:

$$P_{vi}\,V_{vi} = m_{vi}\,R\,T_{vi} \quad (11)$$



a



b

Figure 1. (a) Unlooped PHP and (b) control volume of the liquid slug.

It should be noticed that to operate the heat pipe in the saturated or superheated condition, when the vapor pressure from the above equation is more than the saturated pressure, the fluid phase is not vapor, and thus the saturated pressure should be replaced by the Clapeyron equation:

$$P_{vi} = P_{sat} = P_0 \, exp(\frac{h_{fg}}{R} \, \frac{T-T_0}{T-T_0}) \tag{12}$$

2. 2. Heat Transfer Total Heat transfer into/out of a PHP is due to the phase change mechanism (evaporation/condensation) in cooling/heating section and sensible heat into/out of liquid slugs. The latent heat transfer, due to evaporation and condensation, of the i^{th} vapor plugs can be determined from the following equations:

$$Q_{in.vi} = \dot{m}_{in.vi} \, h_{fg} \tag{13}$$

$$Q_{out.vi} = \dot{m}_{out.vi} \, h_{fg} \tag{14}$$

By solving the energy equation for the liquid slugs, the sensible transferred heat into/out of the PHP is determined:

$$\frac{1}{\alpha_l} \frac{dT_{li}}{dt} = \frac{d^2T_{li}}{dx^2} - \frac{h_{lsen}\pi D}{k_l A} (T_{li} - T_w) \tag{15}$$

with the boundary conditions of:

$$T_{li} = T_{v(i+1)} \qquad x = x_{le(i+1)}$$
$$T_{li} = T_{vi} \qquad x = x_{re,i} \tag{16}$$

By consideration of a thermally developing Hagen-Poiseuille flow and taking isothermal wall condition into account, in laminar flow ($Re \leq 2000$), Shah and London equation can be used to calculate the forced convection heat transfer coefficient: [27]

$$Nu = \begin{cases} 1.615\, x^{*^{-\frac{1}{3}}} - 0.70 & x^* \leq 0.005 \\ 1.615\, x^{*^{-\frac{1}{3}}} - 0.20 & 0.005 < x^* < 0.03 \\ 3.657 + \frac{0.0499}{x^*} & x^* > 0.03 \end{cases} \tag{17}$$

The dimensionless x^* is defined as

$$x^* = \frac{L_{li}/D}{Re_D \, Pr} \tag{18}$$

In the transition and turbulent flow regimes, Nusselt number can be calculated by Gnielinski's relation: [28]

$$Nu = \frac{\left(\frac{C_{li}}{2}\right) \, (Re - 10^3) \, Pr}{1+12.7 \, (\frac{C_{li}}{2})^{\frac{1}{2}} \, (Pr^{\frac{2}{3}}-1)} \quad , Re > 2000 \tag{19}$$

Thus, total sensible heat transferred into and out of the PHP are determined from the following equations:

$$Q_{in.li} = \int_{X_{re.i}}^{X_{le.(i+1)}} \pi D h_{lsen} \, (T_{li,x} - T_w) dx \,, \quad T_{li} \geq T_w \tag{20}$$

$$Q_{out.li} = \int_{X_{re.i}}^{X_{le.(i+1)}} \pi D h_{lsen} \, (T_w - T_{li,x}) dx \,, \quad T_{li} \leq T_w \tag{21}$$

$T_{li,x}$ is the temperature of the i^{th} liquid slug at the location x. Consequently, the total heat transfer into and out the PHP (contributs both of the latent and sensible heat) can be calculated:

$$Q_{total.in} = \sum_{i=1}^{N} Q_{in.vi} + \sum_{i=1}^{N-1} Q_{in.li} \tag{22}$$

$$Q_{total.out} = \sum_{i=1}^{N} Q_{out.vi} + \sum_{i=1}^{N-1} Q_{out.li} \tag{23}$$

2. 3. Entropy Generation Generally, the entropy generation is due to irreversibilities caused by heat transfer and viscous flow. Thus, the entropy generation due to the viscous flow and heat transfer in a pulsating heat pipe is calculated as:

$$\dot{S}_{gen,T}(t) = \dot{S}_{gen,F}(t) + \dot{S}_{gen,th}(t) \tag{24}$$

The entropy generation due to heat transfer (sensible heat transfer and latent heat transfer) at any moment is defined as follows:

$$\dot{S}_{gen,th}(t) = \dot{S}_{gen,lat}(t) + \dot{S}_{gen,sen}(t) \tag{25}$$

The heat transferred into/out of the vapor plugs leads to the changes in the entropy of the vapor plugs. The entropy generation due to latent heat transfer is defined as the net of entropy changes in the evaporator and condenser sections and the changes in the entropy of the vapor plugs. Hence, the entropy generation due to the latent heat in the PHP at any moment can be determined from the following equation:

$$\dot{S}_{gen,lat}(t) = -\frac{\dot{Q}_{eva}}{T_e} + \frac{\dot{Q}_{cond}}{T_c} + \frac{\Delta S_v}{\Delta t} \tag{26}$$

The first term on the right hand side is the entropy changes in the evaporator section (transferred heat of evaporation) and the second term is the entropy changes in the condenser area.

The total entropy changes in the vapor masses, ΔS_v, at any moment is calculated by:

$$\Delta S_v = \frac{d}{dt}(m\, s)\, \Delta t = \left[\frac{dm}{dt} \, s_v + m \, \frac{ds}{dt}\right] \Delta t \tag{27}$$

In which, the rate of mass change in vapor plugs, $\frac{dm}{dt}$, is calculated by Equation (7).

By considering the ideal gas assumption, the rate of specific entropy changes in vapor pulgs can be determined by following equation:

$$\left.\frac{ds}{dt}\right|_v = c_p \, \frac{dT}{T} - R \frac{dP}{P} \tag{28}$$

Using the mentioned relations, the total entropy changes in vapor plugs in an arbitrary time will be derived. Therefore, Equation (27) can be rewritten to the following form:

$$\Delta S_v(t) = (\dot{m}_{in} - \dot{m}_{out}) \, s_v(t) \, \Delta t + m_v(t) \left[c_p ln \frac{T(t)}{T(t-\Delta t)} - Rln \frac{P(t)}{P(t-\Delta t)}\right] \Delta t \tag{29}$$

The sensnible heat transfer and fluid flow contributions of total entropy generation can be expressed in a general form [28]:

$$\dot{S}'_{gen} = \frac{q'\,\Delta T}{T^2} + \frac{\dot{m}}{\rho\,T}\left(-\frac{dP}{dx}\right) \tag{30}$$

The above equation represents the local entropy generation per unit length at any moment. The first term is the entropy generation due to the sensible heat transfer into/out of the PHP and the second term is the fluid flow contribution of the total entropy generation, in which, q' is the heat flux and is calculated from following equation

$$q'(t) = \pi\, d\, \Delta T(t)\, h_{lsen}(t) \tag{31}$$

Hence, the sensible heat transfer contribution of total entropy generation per unit length is:

$$\dot{S}'_{gen}(t) = \pi d\, \left(\frac{T_w - T_L(x,t)}{T_L(x,t)}\right)^2 h_{lsen}\, dx \tag{32}$$

Finally, by integrating the above equation through the pulsating heat pipe length, total entropy generation due to the sensible heat transfer can be calculated as:

$$\dot{S}_{gen,sen}(t) = \int_0^{L_P} \pi\, d\, \left(\frac{T_w - T_l(x.t)}{T_l(x.t)}\right)^2 h_{lsen}\, dx \tag{33}$$

The frictional share of total entropy generation in Equation (30) can be calculated by integrating through the pipe length:

$$\dot{S}_{gen,F}(t) = \int \frac{\dot{m}}{\rho\, T_l(x.t)}\ \frac{2C_{li}}{d}\ \rho V^2\, dx \tag{34}$$

That can be rearranged to the following form:

$$\dot{S}_{gen,F}(t) = \pi\, d\, C_{li}\frac{\rho\, v^3}{2}\int_0^{L_P}\frac{1}{T_l(x,t)}\, dx \tag{35}$$

3. NUMERICAL SOLUTION

All of the governing equations such as: momentum, mass and entropy generation equations are solved by using an explicit finite difference scheme. The explicit discretized form of the momentum and the energy equations of the liquid slug and the vapor plugs are as the following, respectively. These are used to obtain new values at the time of (n+1):

$$A(L_{li}\, v_{li})^{n+1} = \frac{1}{\rho_l}\Big(A\big[\big(P_{vi} - P_{v(i+1)}\big) - \Delta P_{bend}\big] - \pi d L_{li}\tau\Big)^n . \Delta t + (L_{li}\, v_{li})^n \tag{36a}$$

$$T_{vi}^{\,n+1} = \left(\frac{1}{m_{vi}c_{vi}}(\dot{m}_{in} - \dot{m}_{out})RT_{vi}\right)^n - \frac{1}{m_{vi}{}^n c_v}(P_{vi}A)^n(x_{li}{}^{n+1} - x_{li}{}^n) \tag{36b}$$

Similirly, the disctritized form of the entropy generation rate due to the sensible heat transfer and fluid flow are:

$$\dot{S}_{gen,sen}{}^{n+1} = \sum \pi d\left[h_{lsen}\left(\frac{T_W - T_l(x)}{T_l(x)}\right)^2\right]^n \Delta x \tag{37a}$$

$$\dot{S}_{gen,F}{}^{n+1} = \pi d\,\left[C_{li}\frac{\rho\, v^3}{2}\sum\frac{1}{T_l(x)}\Delta x\right]^n \tag{37b}$$

The energy equation of the liquid slug is solved implicitly to calculate the temperature distribution and the sensible heat transfer using TDMA (Tri-diagonal Matrix Method) with 10^{-4} s of time step which makes the solutions independent of time step. Mesh structure of each liquid slug is nonuniform and consists of 1200 nodes. Each end of the liquid slugs with the lentgh of 0.04 m is divided into 400 cells and 400 cells belong to the rest of the slug which has an insignificant role in the heat transfer mechanism.

4. RESULTS AND DISCUSSION

A code has been developed to solve the equations. To validate the numerical calculations, results of the present work are compared with those of Shaffi et al. [4]. The conditions are the same as those used by Shafii et al. and are: $P_{vi}(0)$=5627 (Pa), $T_{vi}(0)$=35 (^{o}C), $L_{v2}=2L_{v1}=2L_{v3}$=0.22 (m), T_c=20 (^{o}C), h_h=150 (W/m^2 ^{o}C), h_c=100 (W/m^2 ^{o}C), L_h=0.1 (m) and L_c=0.37 (m).

Figure 2 presents both fluid flow and heat transfer processes comparisons. As Figure 2(a) shows, the results of liquid slug location versus time are very close to each other that verifies the ability of the code in correct prediction of liquid flow. Comparison between the results of the heat transfer calculations of the present work with those of the previous studies is illustrated in Figure 2(b). Very good agreement between the results obvious in this figure. Hence, according to the comparison results, the code can correctly simulate the overall performance of a PHP.

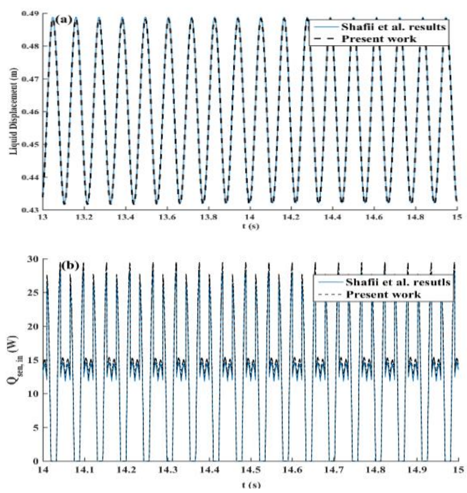


Figure 2. Comparison of predicted a) the liquid slug displacement b) the sensible heat transferred into the PHP with Shafii et al. [4].

The main aim of the present work is investigating the total entropy generation through the operation of a PHP under conditions of different evaporator temperatures and pipe diameters. Figure 3 illustrates total sensible heat transferred into the PHP for different evaporator temperatures in terms of time. It is obviously shown that by increasing the evaporator temperature the heat removing capability of the PHP improves. This conclusion is in the agreement with the previous experimental and numerical results [11,14]. Due to the phase difference between the liquid slugs displacements, since a liquid slug transmits into a condenser section, the other liquid slug moves into the evaporator section. Therefore, the total sensible heat transfer into the PHP never reaches zero.

Figure 4 shows the frictional share of total entropy generation for different evaporator temperatures in terms of time. Since the evaporator temperature increases, the masses transferred into the vapor plugs, and consequently the vapor plugs pressure increase. By increasing the pressures in the sides of any liquid slug, the liquid slug moves with high amplitudes and frequencies. Hence, the entropy generation increases due to the high Reynolds number and high viscous effects. As the figure shows, the viscous flow share of total entropy generation increases by increasing the evaporator temperature.

The entropy generation due to the sensible heat transfer in different evaporator temperatures is illustrated in Figure 5. It is shown that by increasing the evaporator temperature of both the amplitude and frequency of the entropy generation increase. As the heating temperature increases, the temperature difference between the evaporator wall and the liquid slug will increase and consequently enhances the entropy generation.

The entropy changes of the vapor plugs in terms of time are presented in Figure 6. Due to Equation (27), entropy change in any vapor plug depends on the mass change and the change of thermodynamic properties.

The masses of vapor plugs change because of the evaporation and condensation processes. On the other hand, the temperature and the pressure of the vapor plugs change due to the fluid flow and heat transfer processes. As it is shown in Figure 6, the amplitude and the frequency of the entropy change of the vapor plugs are greater in higher evaporator temperatures. This phenomenon is because of the high amplitude and frequency of the liquid slugs motions in the PHP.

Finally, by using Equation (24), total entropy generation through the operation of the PHP can be calculated. Total entropy generation is presented in Figure 7, that is sum of the sensible heat transfer, the latent heat transfer and viscous fluid flow contributions. This figure shows that the total entropy generation graphs oscillate in higher values in greater evaporator temperatures.

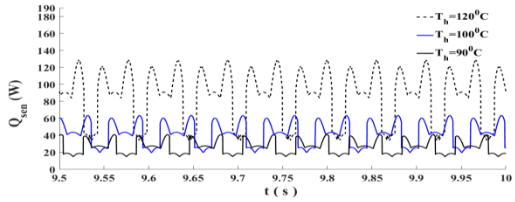


Figure 3. Total sensible heat transfer into the PHP for d=2 mm and different evaporator temperatures.

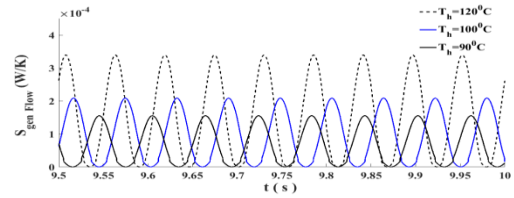


Figure 4. Variation of frictional entropy generation for d=2 mm and different evaporator temperatures.

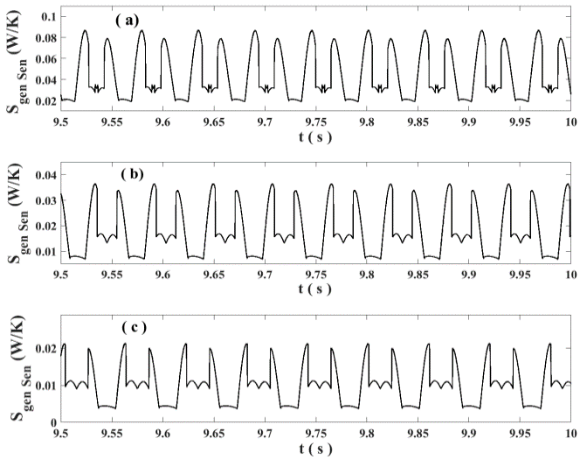


Figure 5. Sensible heat transfer contribution of entropy generation with the pipe diameter of 2 mm and the evaporator temperature of a) 120℃ (b) 100℃ (c) 90℃.

To investigate the effect of pipe diameter in the generation of entropy, d=1mm, d=2mm and d=3mm are taken into account to calculate the results. It should be noticed that a PHP can operate with pipe diameters of smaller than a threshold [13]. This pipe diameter threshold for a PHP filled with water is 3.34 mm.

Total sensible heat transferred into the pulsating heat pipe is demonstrated in Figure 8. This figure shows that sensible heat transfer into the PHP enhances as the pipe diameter increases which conforms with the previous studies [11,14]. This phenomenon is because of the large heating section area and high amplitude of the liquid slug oscillations.

Figure 9 represents the frictional share of entropy generation for different diameters in terms of time. According to the figure, the mean value of $\dot{S}_{gen,F}$ is greater in larger pipe diameters. Conversely, the frequency of the fluid flow share of entropy generation reduces as the pipe diameter increases [11]. Larger pipe area section leads to larger mass of the liquid slugs and consequently decreasing the transporting frequency. This conclusion has been reported in previous works, as well [6].

The total entropy generation of the PHP for different pipe diameters is presented in Figure 10. It is indicated that the total entropy generated in larger diameters is greater. This fact is because of both contributions of the sensible heat transfer and viscous fluid flow. It should be mentioned that the total entropy generation can be used as a design prameter of PHPs to obtain the optimized performance.

The dimensionless Bejan number (Be) is used to calculate the heat transfer share of the total entropy generation in the case studies. The Bejan number is defined as: [28]

$$\mathrm{Be}(t) = \dot{S}_{gen,th}(t)/\dot{S}_{gen,total} \tag{38}$$

The Bejan number has the values between 1 and 0. When Be is about 1, the main share of the total entropy generation is due to the heat transfer and in the values of about 0, the contribution of heat transfer on the total entropy generation is negligible.

Time averaged of studied parameters are briefly presented in Table 1. For specific values of the evaporator temperature and the pipe diameter, total entropy generation, total sensible heat transferred into the PHP and the Bejan number are reported in the Table.

According to the results, the Bejan numbers in all cases are close to 1. Consequently, the heat transfer process has the dominant role in the entropy generation.

Furthermore, the best performance of a PHP can be achieved when the term of $\overline{Q_{in}}/\overline{S_{gen,T}}$ is maximum. According to the results, as the pipe diameter decreases, this term deduces. So, the results express that there is a minimum pipe diameter below which a PHP does not operate efficiently and an ordinary single phase heat pipe may have better performance. So, the best performance of a PHP will be achieved in the possible largest diameter of pipe.

TABLE 1. The time averaged values of investigated parameters in different pipe diameters and evaporator temperatures

D (mm)	T_h (°C)	$\overline{Q_{total,in}}$ (W)	$\overline{S_{gen,total}}$ (W/K)	$\overline{Be}$
3	120	83.32	0.0854	0.9956
	100	40.26	0.0346	0.9930
	90	26.18	0.0201	0.9907
2	120	44.36	0.0458	0.9949
	100	21.60	0.0186	0.9916
	90	13.60	0.0106	0.9885
1	120	9.55	0.0096	0.9942
	100	6.86	0.0056	0.9908
	90	2.03	0.0016	0.9817

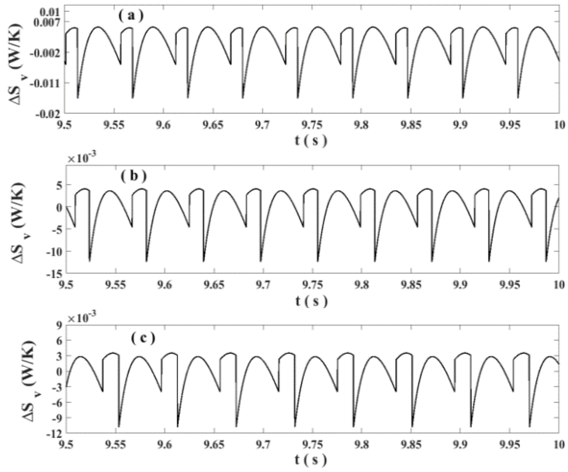


Figure 6. Entropy change of vapor plugs in PHP with pipe diameter of 2 mm and the evaporator temperature of (a) 120℃ (b) 100℃ (c) 90℃.

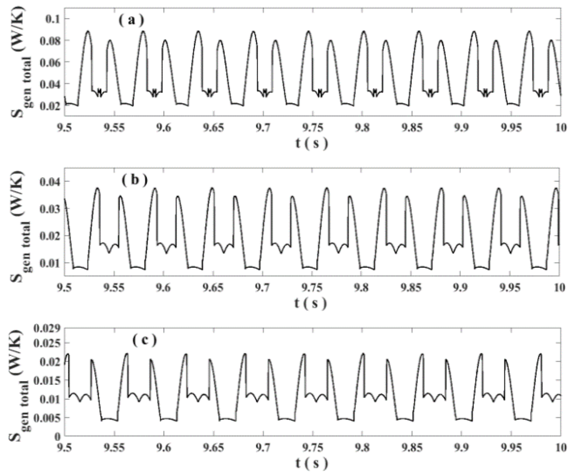


Figure 7. Total entropy generation in PHP with pipe diameter of 2 mm and the evaporator temperature of a) 120℃ (b) 100℃ (c) 90℃.

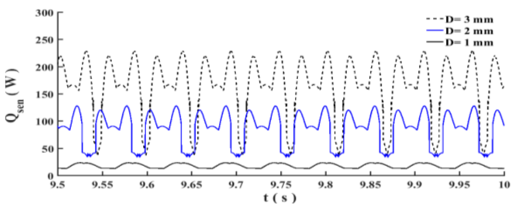


Figure 8. Total sensible heat transfer into the PHP with different pipe diameters and T_h=120℃.

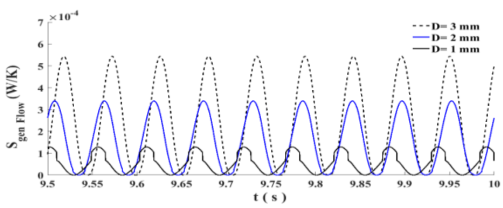


Figure 9. Variation of frictional entropy generation for T_h=120℃ and different pipe diameters.

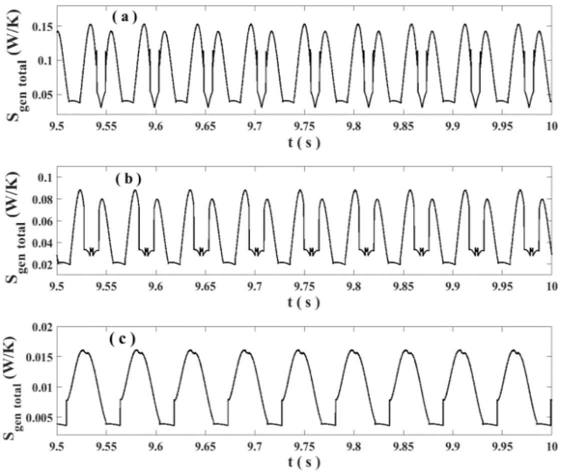


Figure 10. Total entropy generation in PHP with the evaporator temperature of 120℃ and the pipe diameter of a) 3 mm (b) 2 mm (c) 1 mm.

5. CONCLUSIONS

The operation of a pulsating heat pipe is a combination of heat transfer and fluid flow phenomena. In this study, fluid flow, heat transfer and entropy generation in a PHP are modeled numerically. Also, the frictional and the heat transfer contributions of the entropy generation have been investigated.

Our results indicate that by increasing the evaporator temperature, the sensible heat transfer into the PHP, the liquid slug flow and the entropy generation oscillate in higher frequencies and amplitudes in terms of time.

Additionally, by investigating the effect of the pipe diameter, it is concluded that the sensible heat transfer into the PHP and the total entropy generation reach higher values in the larger pipe diameters. It is shown that the best performance of a pulsating heat pipe is in the situation of highest evaporator temperature and the largest diameter of pipe. However, in small pipe diameters ordinary single phase heat pipes may operate more efficient.

The results demonstrate that the Bejan number reduces in smaller pipe diameters. This phenomenon expresses that the frictional entropy generation can be more dominant in smaller pipe diameters. According to the Bejan number results, the role of sensible heat transfer in the generation of entropy is dominant and the frictional effects can be eliminated.

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Investigation of Entropy Generation Through the Operation of an Unlooped Pulsating Heat Pipe

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چکیده

در مطالعه حاضر، یک لوله حرارتی نوسانی باز با دو اسلاگ مایع و سه پلاگ بخار در مجاورت آنها در نظر گرفته شده است. به جز معادله انرژی برای اسلاگ مایع، معادلات حاکم از قبیل مومنتوم، انرژی و بقاء جرم به صورت صریح حل شده‌اند. هدف از این مطالعه محاسبه تولید آنتروپی طی عملکرد لوله حرارت نوسانی است. علاوه بر این، قطرهای مختلف لوله و اثر دماهای متفاوت اواپراتور برروی تولید آنتروپی بررسی شده است. همین طور, عدد بجان در هر مورد برای بررسی سهم انتقال حرارت در تولید آنتروپی به دست آمده است . با توجه به نتایج، با افزایش قطر لوله انتقال گرمای محسوس و نهان به لوله حرارت نوسانی افزایش یافته و اسلاگ های مایع در دامنه بالاتری نوسان می‌کنند. از طرف دیگر, با افزایش قطر لوله مقدار تولید انتروپی افزایش می‌یابد. ارزیابی اعداد بجان نشان می‌دهد که سهم اثرات ویسکوز (گران‌رو) در تولید آنتروپی با افزایش قطر لوله، کاهش می‌یابد. نتایج مطالعات قبلی نشان داده‌اند که برای عملکرد لوله های حرارتی نوسانی، آستانه‌ای برای قطر لوله وجود دارد. با این حال، نتایج مطالعه حاضر نشان می‌دهد که استفاده از لوله‌های حرارتی نوسانی با قطر کم قابل توجیه نیست. علاوه براین، نتایج نشان می‌دهد که با افزایش اختلاف دمای کندانسور و اواپراتور، عمکرد حرارتی لوله حرارتی نوسانی بهبود می‌یابد. نتایج محاسبه شده افزایش تولید آنتروپی کل را در دماهای بالای اواپراتور نشان می‌دهد. عدد بجان با هر افزایشی در دمای اواپراتور افزایش خواهد یافت و این پدیده در دماهای بالای اواپراتور نقش ناچیز اثرات ویسکوز را نشان می‌دهد. برای ارزیابی صحت محاسبات , نتایج به دست آمده با نتایج ارائه شده‌ی قبلی مقایسه شده است. این مقایسه مطابقت خیلی خوبی را بین آنها نشان می‌دهد.

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