# Asymmetry and geometry effects on the dynamic behavior of a pulsating heat pipe

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# ASYMMETRY AND GEOMETRY EFFECTS ON THE DYNAMIC BEHAVIOR OF A PULSATING HEAT PIPE

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## KEY WORDS

Pulsating heat pipe, multiphase flow, heat transfer

## ABSTRACT

*A mass-spring-damper model is developed to investigate the motion in a pulsating heat pipe (PHP). A heat transfer model is coupled to this mass-spring-damper model in order to study the effectivity of a PHP under different operating conditions. Four different configurations (one PHP with 12 turns; two parallel PHPs with 6 turns; 3 parallel PHPs with 4 turns; 6 parallel PHPs with 2 turns) are used to investigate the geometry effects. More oscillatory and translational motion and less stops are observed when increasing the number of turns of the PHP. A decrease in the number of motion stops leads to less fluctuations in the evaporator temperature and therefore less spread in thermal resistance of the system. In addition to that, the total thermal resistance of the system decreases with an increase in the number of turns of the PHP.*

## 1. INTRODUCTION

In recent years, flexible electronics are becoming more important due to their mechanical flexibility, lightweight and low price [1]. However, thermal management has always been challenging in flexible electronics. Low thermal conductivity of the polymer substrates is a big limitation on the design process and reliability of these devices. Conventional cooling approaches are not suitable to integrate in plastic foils. Also they are becoming increasingly insufficient to maintain acceptable temperature levels. This called for the development of miniature/micro heat pipes characterized by two-phase heat transfer. The pulsating heat pipes (PHPs) invented in 1990s by Akachi [2] are promising devices that can work in micro scales. In contrast with conventional heat pipes, a PHP does not have an internal wick structure [3]. Also, it does not require mechanical pumps or valves. A pulsating heat pipe is made of a serpentine tube of capillary dimensions with a certain number of turns (Fig. 1a). It is evacuated and then filled partially with a working fluid. It is heated from one end and cooled at the other end. Therefore evaporator and condenser sections can be distinguished (Fig. 1a). Generally an adiabatic section is placed between the evaporator and condenser sections. The PHP may be either closed or open loop, where the tube ends are connected to each other or sealed respectively [3]. The closed loop PHP has a better heat transfer performance [4], so most research is done with this type of PHP.

The working principle of such device is not yet fully understood. An interplay is assumed between convective heat transfer, evaporation of liquid at the hot side and condensation of the vapor at the cold side. Vapor plugs in the evaporator section heat up and pressure difference between evaporator section and condenser section occurs. Due to this pressure difference, a liquid slug is pushed to the condenser section where both vapor plugs and liquid slugs are cooled down. The hot vapor plug, that reaches the condenser section, partly condenses and the pressure decreases. This forces the liquid slug to move back toward the evaporator section. In this way, oscillatory motion can occur inside the PHP.

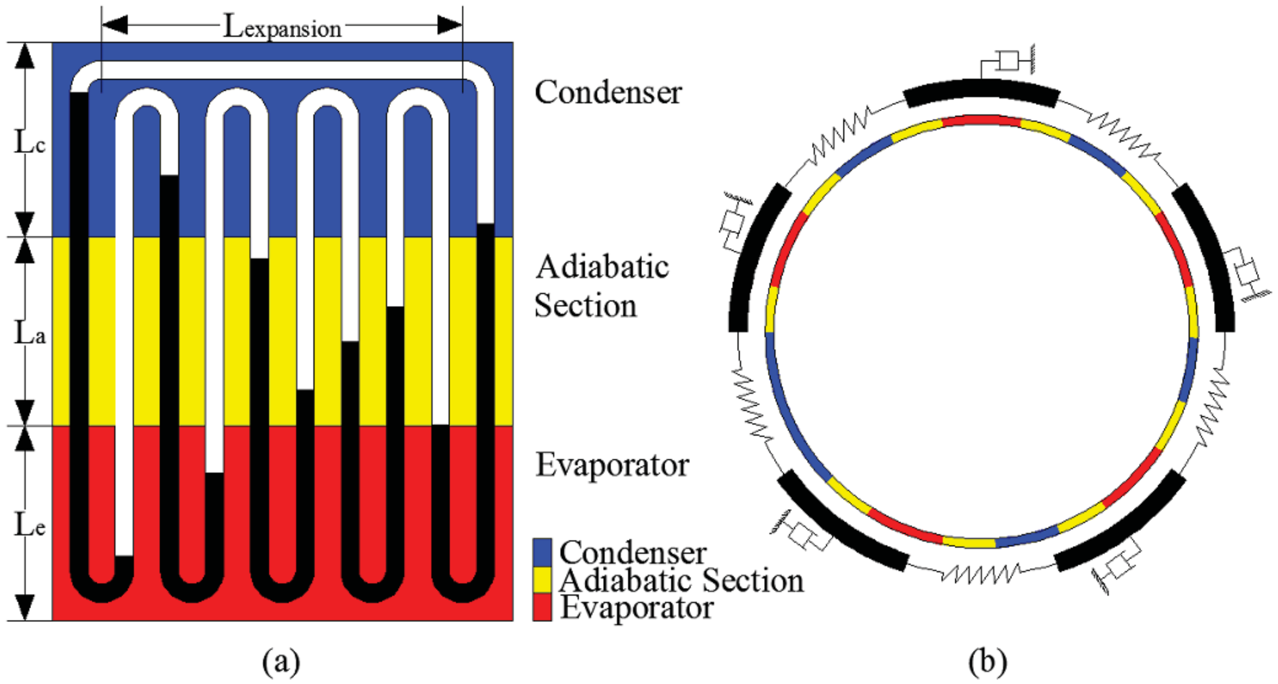


**Figure 1**: (a) Schematic view of a pulsating heat pipe with 5 turns. *Black* and *white* portions represent liquid slugs and vapor plugs, respectively. The PHP is heated in the evaporator section (*red*) and cooled in the condenser section (*blue*). (b) Mass-spring-damper representation of the PHP. Liquid slugs, vapor plugs, friction and capillary forces are represented by masses, non-linear springs and non-linear dampers, respectively. The locations of the evaporator, adiabatic and condenser sections are indicated by the colored inner ring.

Several modeling approaches can be found in the literature. Oscillations are described as a combination of several factors, including pressure difference, frictional force, gravitational force, capillary force and energy and mass transfer [5]. Dobson [6], [7] used an open loop pulsating heat pipe to investigate the influence of the geometry on the motion of one vapor plug and two liquid slugs system. He found that surface tension has a vital role on the formation of the liquid slug but friction, gravity and pressure forces were more important for the motion. Ma et al. [8] described how oscillations are affected only by thermal energy due to the temperature difference between the evaporator and condenser. Sakulchangsatjatai et al. [9] developed a mathematical model that includes coalescence of the liquid slugs to predict the heat flux of a closed end PHP at normal operating state. The effects of evaporator length, inner diameter and working fluid were also studied. They showed that the maximum heat transfer rate of the closed end PHP occurred at the highest evaporator temperature (150°C for their study).

Although a large number of mathematical and numerical models already exist in literature, all of them are based on a uniform distribution of liquid slugs and vapor plugs which is not observed in experiments. Khandekar et al. [10] observed that vapor plugs and liquid slugs are not symmetrically distributed which cause an uneven capillary pressure difference. Tong et al. [11] conducted experiments to investigate the flow characteristics of PHPs. They observed that vapor plugs and liquid slugs are unevenly distributed during initial and operating states. They stated that large amplitude oscillations from evaporator to condenser occur in the PHP during the start-up period. After that, a translational motion of the working fluid occurs in a PHP.

In this paper, the main objective is to investigate the effect of asymmetry in the initial liquid slug and vapor plug positions and the effect of an increase of the number of turns on the thermal performance and on the modes of motion (oscillatory, oscillatory-translational, translational or no motion) in a PHP. To achieve this, a mass-spring-damper model is developed to predict the motion of liquid slugs in a PHP. It can be used for both symmetric and asymmetric situations. This mathematical model is completed by adding a heat transfer module by which the thermal performance can be investigated.

# 2. MATHEMATICAL MODEL

## 2.1 Mass-Spring-Damper Model

This work is based on a closed loop pulsating heat pipe [12]. A schematic view of pulsating heat pipe with five turns is given in Fig. 1a. This system can also be seen as a quasi-one-dimensional mass-spring-damper model where the liquid slugs are represented by masses, the vapor plugs by non-linear springs and the friction and the capillary forces by non-linear dampers. (Fig. 1b)

The governing equation is derived from the force balance on the liquid slugs:

$$m_{l,i}\ddot{x}_i = F_{pres,i} + F_{fric,i} + F_{cap,i} + F_{grav,i}\,, \tag{1}$$

where $m_{l,i}$, $\ddot{x}_i$, $F_{pres,i}$, $F_{fric,i}$, $F_{cap,i}$ and $F_{grav,i}$ are the mass and acceleration of the $i^{th}$ liquid slug, the effective pressure force over a liquid slug, the friction force, the effective capillary force and the gravitational force acting on the $i^{th}$ liquid slug, respectively. For calculating the pressure of vapor plugs the ideal gas law is assumed. The effective pressure force over a liquid slug is calculated by:

$$F_{pres,i} = A_c\left(P_{v,left} - P_{v,right}\right), \tag{2}$$

where $A_c$ is cross-sectional area of the capillary tube and $P_v$ is the vapor pressure.

The friction force on a liquid slug i is described by:

$$F_{fric,i} = 2\mu_l \pi L_{l,i}\dot{x}_i\,, \tag{3}$$

where $\mu_l$, $L_{l,i}$ and $\dot{x}_i$ are the dynamic viscosity of liquid, the length and the velocity of the $i^{th}$ liquid slug, respectively. Womersley number is calculated for our system and Poiseulle flow assumption is used due to the low Womersley number ($Wo < 1$)

The capillary force on the liquid slug is expressed by:

$$F_{cap,i} = \left(\sigma_{i,right}\frac{2}{D}\cos\theta_{right} - \sigma_{i,left}\frac{2}{D}\cos\theta_{left}\right)A_c\,, \tag{4}$$

where $\sigma_i$, D, $\theta_{right}$ and $\theta_{left}$ are the surface tension, the inner diameter of the tube, the advancing and the receding contact angles of the $i^{th}$ liquid slug, respectively. The modified Hoffman-Tanner's law is used to determine the dynamic contact angles [13].

Finally, the gravitational force on a liquid slug is described by:

$$F_{grav,i} = m_{l,i}\, g \sin\alpha_{inc}\,, \tag{5}$$

where g and $\alpha_{inc}$ are the gravitation constant and the inclination angle of pulsating heat pipe.

## 2.2 Heat Transfer Model

Next to the motion model, also a heat transfer model is needed. We assume that the condenser section is kept at a fixed temperature and evaporator section is heated with a constant heat flux. Furthermore we assume a uniform heating or cooling of the slugs and plugs. The temperature of a vapor plug and a liquid slug is calculated by:

$$\dot{Q}_i = m_i C_p \frac{dT_i}{dt}, \tag{6}$$

where $C_p$ and $T_i$ are the heat capacity and the temperature of the slug or plug. $\dot{Q}_i$ is the heat flux that is added to or removed from the liquid or the vapor with mass $m_i$.

Since the heat transfer coefficient for the liquid slug and vapor plug are different, the total heat input $\dot{Q}$ will be unequally distributed over the liquid and the vapor. We assume that the heat going to the liquid and the vapor is calculated by:

$$\dot{Q}_l = \frac{h_l A_l}{h_l A_l + h_v A_v}\dot{Q} \quad \text{and} \quad \dot{Q}_v = \frac{h_v A_v}{h_l A_l + h_v A_v}\dot{Q}, \tag{7}$$

where $h$, $A$ and $\dot{Q}$ are heat transfer coefficient of the liquid (l) or the vapor (v) (calculated using Nusselt relation for uniform surface heat flux for circular tubes for evaporator section and calculated using Nusselt relation for uniform wall temperature for circular tubes for condenser section), the heat exchange surface area of the liquid slug or vapor plug inside the evaporator or condenser section and total heat flux added to the system from the evaporator or taken from the system condenser, respectively. Since the evaporator section has a constant heat flux, only the heat flux of the condenser section should be calculated by the convective heat transfer equation.

When the liquid temperature reaches the saturation temperature, the liquid starts to evaporate. When the vapor temperature is below the saturation temperature, vapor starts to condense. Saturation temperature is pressure dependent and is calculated by the Antoine equation [14]. Evaporation and condensation mass flow rate are calculated by:

$$\dot{m}=\frac{\dot{Q}_{l,v}}{h_{fg}}, \tag{8}$$

where $h_{fg}$ is the latent heat of evaporation.

The thermal resistance of the system is calculated by:

$$R_{th}=\frac{T_e-T_c}{Q_{in}} \tag{9}$$

where $T_e$, $T_c$ and $Q_{in}$ are evaporator and condenser temperatures and heat input, respectively. As we will assume a constant $T_c$ and $Q_{in}$, any variation in the $R_{th}$ will correspond to a variation in $T_e$.

### 2.3 Numerical Procedure

The mass-spring-damper model and the heat transfer model are coupled and implemented in Simulink (Mathworks Inc, USA). An explicit forward time scheme is used. Time step is chosen automatically by Simulink with a tolerance of $10^{-3}$. The evaporation and the condensation condition is checked with the temperature data from the previous time step. When the temperature of the liquid from the previous time step is close to the saturation temperature, liquid temperature in the next time step can have a value slightly higher than the saturation temperature. However, it should start to evaporate when it reaches the saturation temperature. The maximum error that occurs due to this situation is calculated to be less than 0.1%. Validation of our model was made with experimental results given in literature and a good agreement was found [12].

## 3. RESULTS AND DISCUSSION

### 3.1 Thermal Resistance vs Number of Turns

Pulsating heat pipes with 2, 4, 6 and 12 turns are modelled to investigate the geometry effects (the dimensions of each PHP are given in Table 1). For a fair comparison, the evaporator area is chosen to be the same in all simulations: one PHP with 12 turns or two PHPs with 6 turns or 3 PHPs with 4 turns or 6 PHPs with 2 turns can be placed (see Fig. 2). The total thermal resistance from evaporator to condenser for each different set-up is calculated by Eq. 9. All PHPs are filled with water with a filling ratio of 60%. In the simulations liquid slugs and vapor plugs are distributed equally. The dimensions of each PHP are given in Table 1. For each number of turns, five different simulations were made. For each successive simulation as initial conditions these vapor plugs and liquid slugs are moved over $(L_v+L_l)/5$.

**Table 1:** Dimensions of the PHPs

| PHP with | $L_e$ | $L_c$ | $L_a$ | $L_{expansion}$ | $L_{PHP}$ | D |
|---|---|---|---|---|---|---|
| 2 turns | 0.03 m | 0.03 m | 0.045 m | 0.0270 m | 0.4470 m | 0.5 mm |
| 4 turns | 0.03 m | 0.03 m | 0.045 m | 0.0540 m | 0.8940 m | 0.5 mm |
| 6 turns | 0.03 m | 0.03 m | 0.045 m | 0.0810 m | 1.3410 m | 0.5 mm |
| 12 turns | 0.03 m | 0.03 m | 0.045 m | 1.1620 m | 2.6820 m | 0.5 mm |

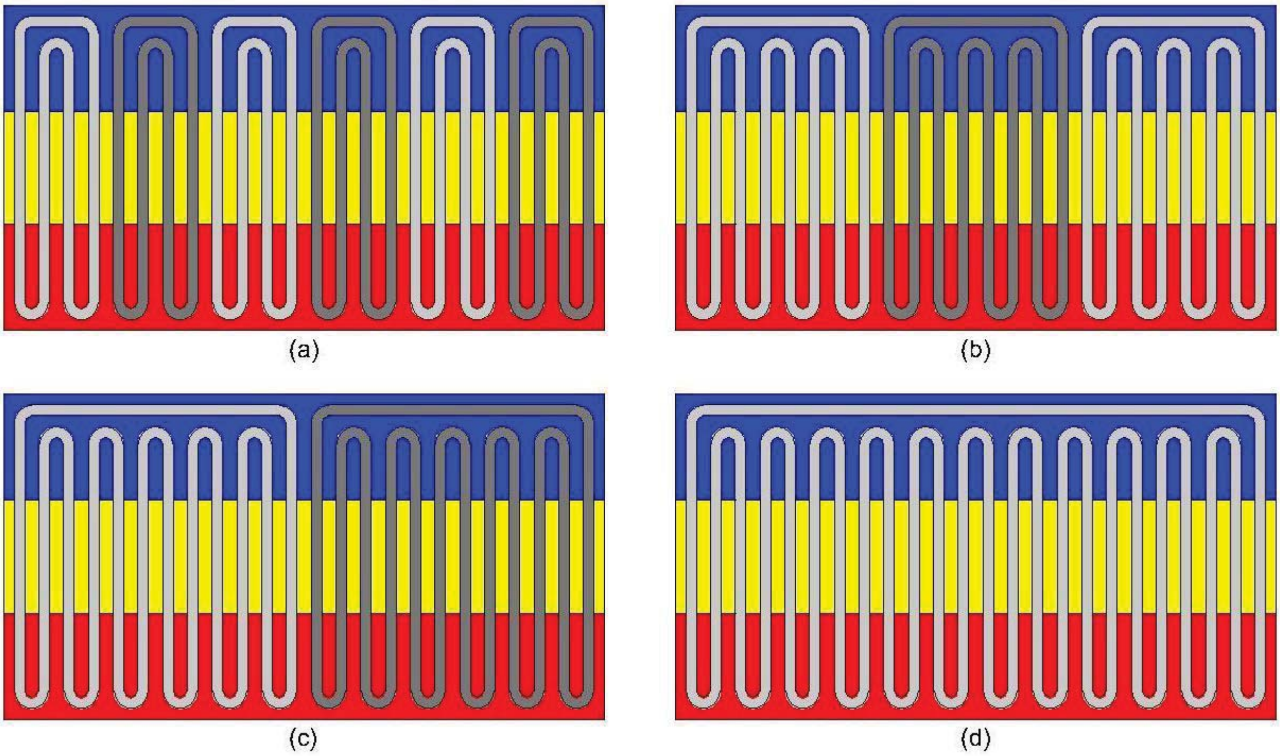


**Figure 2:** Different configurations: (a) six PHPs with two turns, (b) three PHPs with four turns, (c) two PHPs with six turns and (d) one PHP with twelve turns. Dimensions of the each configuration are given in Table 1.

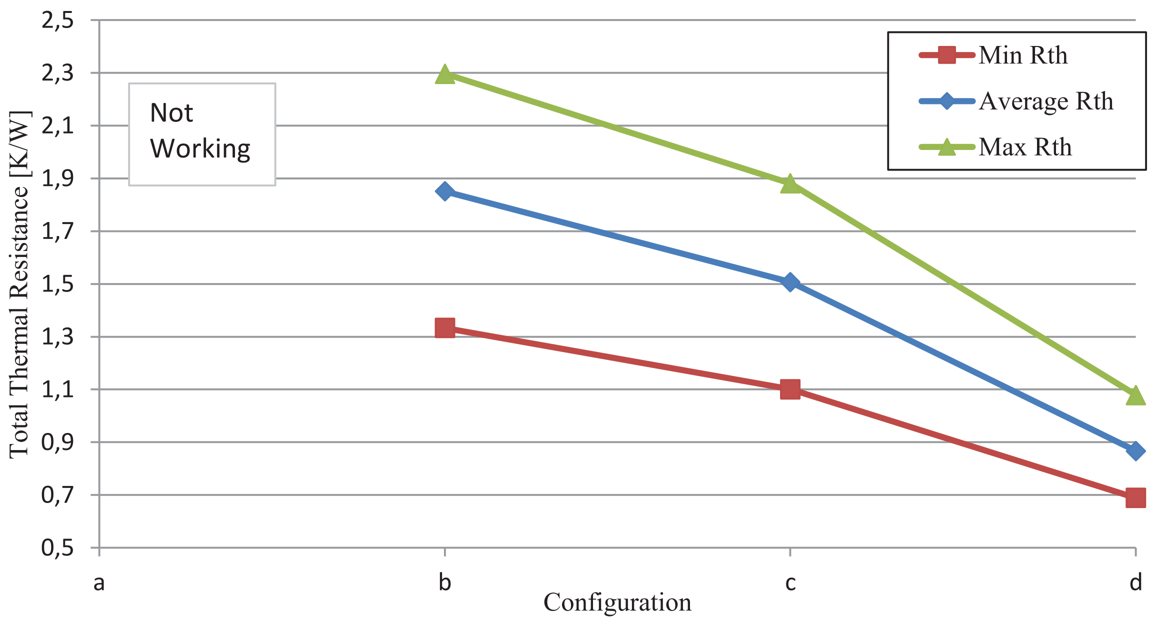


**Figure 3:** Total thermal resistance vs different configurations (See Fig. 2). The *blue line* shows the average thermal resistance of each configuration. *Green* and *red lines* show the maximum and minimum values for each configuration, respectively. The total heat input is kept the same for each configuration: $Q_{in,tot}$ = 48W.

Figure 3 shows the thermal resistance for each configuration with a different number of turns. The average thermal resistance of five simulations are shown. The upper and the lower lines show the maximum and minimum thermal resistances that are due to the change in temperature of the evaporator section. As it is seen from Fig. 3 the thermal performance of the PHP system increases with an increase in the number of turns. The PHP with 2 turns is not working. Also, the fluctuation in the thermal resistance decreases with the increase in number of turns. There are several explanations for this. Firstly, the liquid plugs make several types of motion: no motion, oscillatory motion, combined motion (oscillatory-translational motion) and translational motion [15]. The thermal resistance increases when the fluid is not moving. A PHP with less turns intends to stop more than PHP with more turns (Fig. 4). Secondly, in a PHP with less turns, the chance of the evaporator section being completely filled with vapor plugs is larger than the PHP with more turns. When the evaporator section is filled with only vapor plugs, the heat transfer to the PHP decreases dramatically which causes the increase in temperature of the evaporator section.

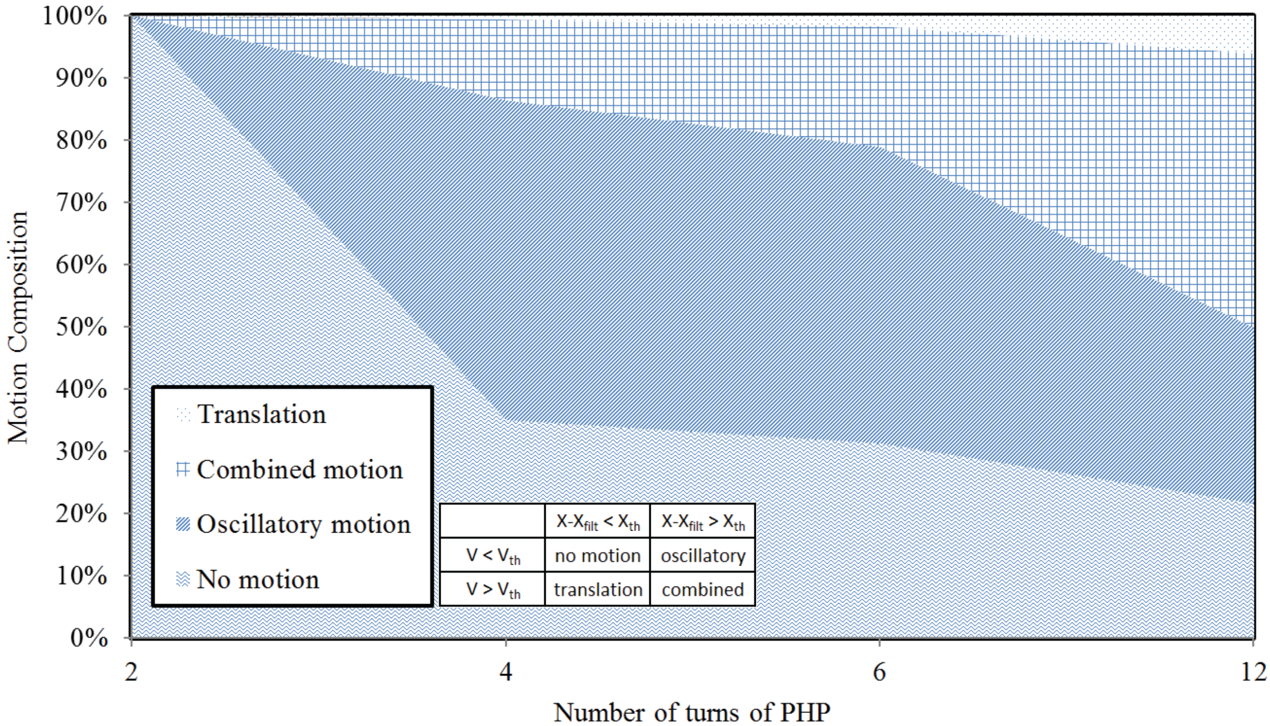


**Figure 4:** Motion composition in PHPs with different number of turns. The criteria for the different types of motion are indicated by the insert and explained in more detail in section 3.2.

### 3.2 Modes of Motion vs Number of Turns

To compare different configurations, also the occurrence of different modes of motion is analyzed. As previously described there are four modes of motion: translational motion, combined translational-oscillatory motion, oscillatory motion and no motion. To determine the percentage of each type of motion liquid slug exhibits, the displacement of liquid slug is filtered using the filtfilt approach (MATLAB version 2013b). To find the numerator and denominator coefficients, first order low pass filter with a normalized cut-off frequency of 1/10 of the time step is used. This filtered data gives the displacement of the liquid slug without the noise arising from the oscillation (Fig. 5). To distinguish the translational and combined translational-oscillatory motion from oscillatory motion and no motion, the slug velocities are computed from the filtered data (Fig. 6). If the absolute velocity of the liquid slug is larger than threshold velocity $v_{th} = 0.2$ m/s, the system is considered to be in the translational or combined translational-oscillatory modes. If this value is smaller than threshold value $v_{th} = 0.2$ m/s, the liquid slug exhibits an oscillatory motion or does not move at all. To be able to differentiate between the oscillatory motion and no motion and also between combined motion and translation, the difference between the original displacement data and the filtered data $x\text{-}x_{filt}$ is calculated (Fig. 6). If the amplitude is smaller than the threshold value $x_{th} = 10^{-2}$ m, the liquid will not oscillate, but will only show a translational motion (if $|v| > v_{th}$) or will not move (if $|v| < v_{th}$). If this amplitude is larger than threshold value $x_{th}$ the liquid slugs will oscillate. It is regarded as oscillatory motion when $|v| < v_{th}$ or as a combined motion when $|v| > v_{th}$. Figure 4 shows the percentage of each mode of motion corresponding to different number of turns. It is seen that PHPs with less turns have higher tendency to stop. Furthermore, the translational motion becomes more important when the number of turns is increased.

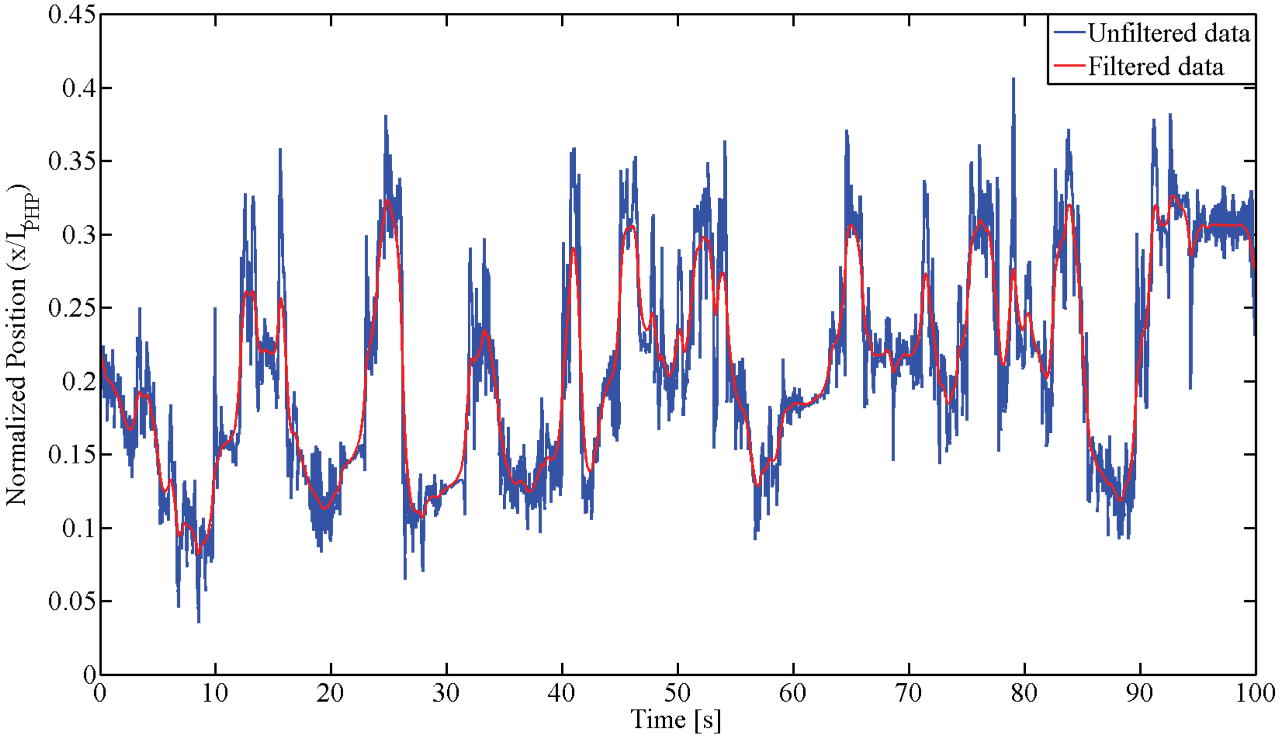


**Figure 5:** Normalized position of a liquid slug as a function of time. The *blue* and *red line* represent unfiltered and filtered displacement data, respectively.

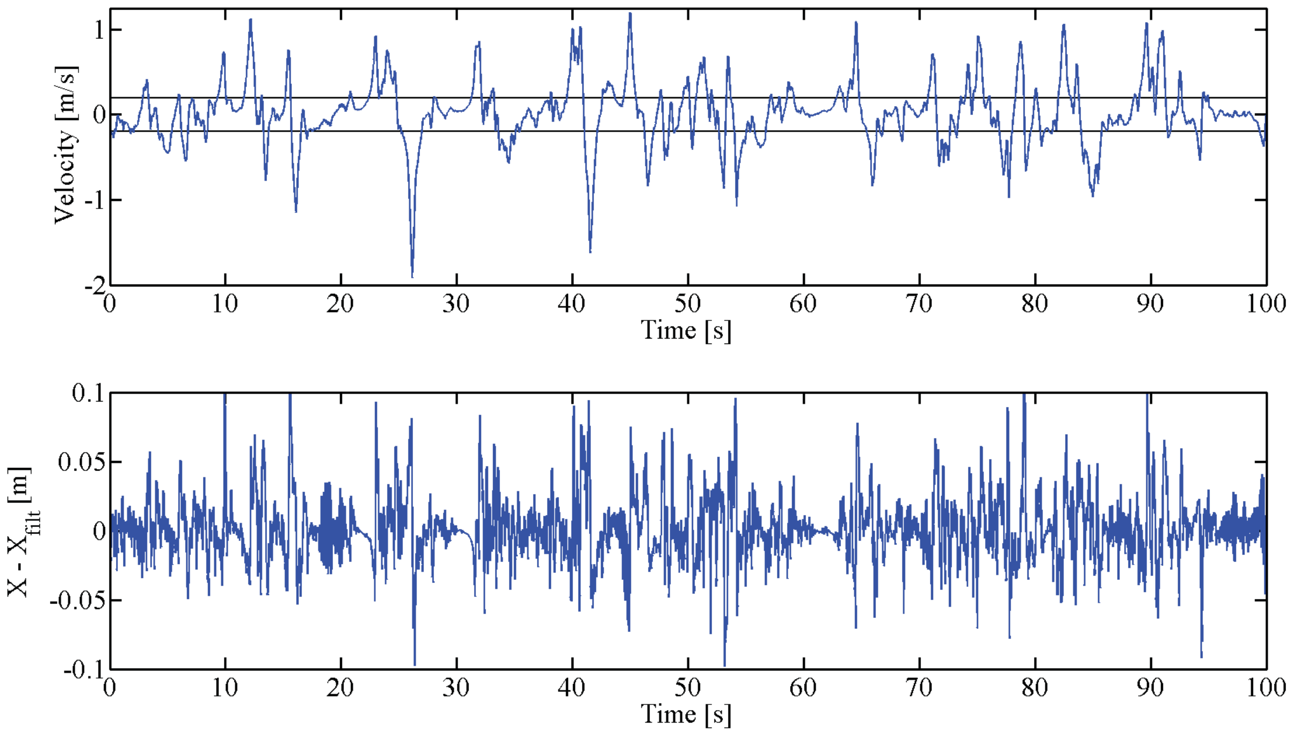


**Figure 6:** *Top*: Time derivative of the filtered displacement data as a function of time. If the velocity value is between the upper and lower thresholds, $v_{th}$ = 0.2m/s and $v_{th}$ = -0.2m/s respectively, the liquid slug is considered to be oscillating. *Bottom*: Difference between sampled data (X) and filtered data ($X_{filt}$) as a function of time. If the amplitude is smaller than threshold $x_{th} = 10^{-2}$ m, there is no oscillation in the system.

## 5. CONCLUSIONS

A heat transfer model with a mass-spring-damper model was used to investigate four different PHP configurations (one PHP with 12 turns, two parallel PHPs with 6 turns, three parallel PHPs with 4 turns and six parallel PHPs with 2 turns). It is seen that when the number of turns increases, the thermal performance of the PHP also increases. Also it is found that there is less spread on the evaporator temperature when the number of turns of the PHP increases. The probability of a stop in the liquid motion becomes smaller while the probability of an oscillatory or translational motion becomes larger when then number of turns increases. These results indicate that PHPs with a larger number of turns have better performance than PHPs with less turns.

## ACKNOWLEDGEMENTS

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