Design and operation of a Tesla-type valve for pulsating heat pipes

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ABSTRACT

A new Tesla-type valve is successfully designed for promoting circulation in a pulsating heat pipe (PHP) and improving the thermal resistance. Its functionality and diodicity is tested by laminar single-phase modelling and by steady two-phase flow experiments. The valve is symmetrically integrated in a single-turn PHP, which reduces variabilities to give a more thorough understanding of the behaviour in PHPs. Two transparent bottom-heated PHPs, one with and one without valves, are manufactured using steady two-phase flow experiments. The valve is symmetrically integrated in a single-turn PHP and also improve the heat transfer. This makes it interesting to investigate methods to promote circulatory motion in a PHP [9,15]. Moreover, directional promotion could increase the stability and predictability of the PHP.

Circulatory motion has been induced inside PHPs using asymmetrical heating [16], floating-ball check valves [17], a variation of channel diameters [9,18,19] and Tesla-type valves [20]. Although being of scientific interest, promoting the circulation using asymmetrical heating is not practically applicable. Similarly, using floating-ball check valves inherently contradicts the benefits of the PHP by having a moving part and being difficult to manufacture when integrated and/or miniaturised.

A more practical solution is to utilize asymmetrical flow resistance to promote directional circulation. Holley and Faghri [18] were the first to suggest this by varying the channel diameter. It was demonstrated that this could theoretically improve a directional flow in a single-turn PHP and also improve the heat transfer. These phenomena were attributed to the fact that a bubble in an

1. Introduction

The demand for faster and smaller micro-electronic systems continues to increase and consequently, the amount of produced heat per volume increases. Therefore, the need for novel efficient cooling devices is vast [1]. Heat pipes are effective passive heat spreaders that can be used to solve this problem. Due to the combined convection and phase transition, a high heat exchanging efficiency can be achieved [2]. A drawback of the traditional heat pipe is that it requires an intricate wick structure and cannot be easily miniaturised due to the capillary limit [3]. Therefore, a new heat pipe called the Pulsating (or Oscillating) Heat Pipe (PHP) was introduced by Akachi in 1990 [4]. The main benefits of the PHP are that it has a very simple shape with no wick structure, which can be easily miniaturised. This makes it a very inexpensive and easy-to-integrate heat spreader which can be beneficial for many, also non-electronic, applications [5].

A PHP generally consists of a simple meandering capillary tube or channel which alternatingly passes through evaporator (heating) and condenser (cooling) zones as schematically shown in Fig. 1. Due to the capillary size of the channel a series of liquid slugs and vapor plugs is formed by the working fluid. The constant heat exchange triggers phase-change phenomena which result in pressure variations inside the device that consequently trigger movement of the liquid slugs and plugs. The overall heat transfer is mainly determined by the sensible heat transfer and the latent heat contributes mostly to the movement of the slugs and plugs [6]. The performance of a PHP thus relies on the continuous non-equilibrium conditions throughout the system and an intricate interplay of physical phenomena. Depending on the specific conditions, the liquid slugs are stagnant, pulsating, circulating with a superimposed pulsation or purely circulating [7,8]. Furthermore, the flow pattern can vary from the normal bubble–liquid slug flow to annular flow [9]. The PHP has received large interest in the scientific community, however, due to the complexity and chaotic behaviour, no fully comprehensive theory or model and no general design tools are available [10,11].

It is known that circulation of the working fluid contributes to a better performance of the PHP [12–14]. The liquid contact in the evaporator is increased when the working fluid is circulating, which increases the heat transfer. This makes it interesting to investigate methods to promote circulatory motion in a PHP [9,15]. Moreover, directional promotion could increase the stability and predictability of the PHP.

Circulatory motion has been induced inside PHPs using asymmetrical heating [16], floating-ball check valves [17], a variation of channel diameters [9,18,19] and Tesla-type valves [20]. Although being of scientific interest, promoting the circulation using asymmetrical heating is not practically applicable. Similarly, using floating-ball check valves inherently contradicts the benefits of the PHP by having a moving part and being difficult to manufacture when integrated and/or miniaturised.

A more practical solution is to utilize asymmetrical flow resistance to promote directional circulation. Holley and Faghri [18] were the first to suggest this by varying the channel diameter. It was demonstrated that this could theoretically improve a directional flow in a single-turn PHP and also improve the heat transfer. These phenomena were attributed to the fact that a bubble in an
expanding channel will move in the diverging direction due to unbalanced capillary forces. When the tube diameter is varied in the condenser and the evaporator, i.e. having alternating channel diameters per turn, this effect is exploited to promote a circulatory flow. Kwon and Kim [19] performed experiments on several single-turn closed loop PHPs with a varied channel diameter. Besides the diameters, the heat input and inclination angle were also varied. It was concluded that a dual-diameter helps to generate a circulating flow at a lower input power and a maximum decrease in thermal resistance of 45% was found. Also it was shown that an optimum of diameter difference exists due to the fact that the smaller tube increases the flow resistance and therefore reduces the mass flow rate of the fluid. Liu et al. [9] performed experiments on three different PHPs with four turns of which one had alternating channel diameters and another had a single section in the adiabatic section which had a larger diameter. These modifications both induced the circulatory motion and improved the thermal performance compared to the standard PHP. Although these variations in channel diameter can promote circulatory flow significantly, the effect is substantially dependent on gravity. The bubble movement will cause the larger channel to contain more vapor than the smaller channel. This causes an unbalance in gravitational force which is the main driving force that promotes the circulating flow [19]. This influence is believed to be reduced when the overall size decreases and the number of turns increases [19,21].

A second option of creating an asymmetrical flow resistance is to utilize flow rectification structures or ‘no-moving-parts’ passive valves. Proven to function for micro-pumps, these valves have received great interest in the field of microfluidics due to the fact that they are easy to manufacture, durable and can transport fluids containing particles [22,23]. The valves are structures that have a higher pressure drop for the flow in one direction (reverse) than the other (forward). This difference in flow resistance causes a net directional flow rate in the forward direction in oscillating flows. The efficiency is often expressed in diodicity $Di$, being the ratio of pressure drops for identical flow rates [24]:

$$Di = \left( \frac{\Delta p_r}{\Delta p_f} \right)_q \tag{1}$$

**Nomenclature**

- $a$: D-valve geometry radius, mm
- $b$: D-valve geometry radius, mm
- $CW$: clockwise
- $CTR$: circulation tally ratio
- $CCW$: counter-clockwise
- $D$: channel diameter, m
- $Di$: diodicity
- $e$: D-valve geometry radius, mm
- $F$: field of view
- $g$: gravitational acceleration, 9.81 m/s$^2$
- $I$: identity matrix
- $IG$: gas inlet
- $IL$: liquid inlet
- $I/O$: inlet/outlet
- $J$: channel junction
- $l$: entrance length, m
- $L$: D-valve geometry length, mm
- $O$: outlet
- $P$: pressure evaluation line
- $p$: pressure, Pa
- $Q$: flow rate, m$^3$/s
- $q$: heat rate, W
- $R$: D-valve geometry radius, mm
- $Re$: Reynolds number
- $T$: thermocouple location
- $u$: velocity, m/s
- $V$: volume, m$^3$
- $W$: channel width, mm

**Greek symbols**

- $\alpha$: D-valve geometry angle, °
- $\beta$: D-valve geometry angle, °
- $\gamma$: Surface tension, N/m
- $\epsilon$: D-valve geometry angle, °
- $\mu$: Viscosity, Pa·s
- $\rho$: Density, kg/m$^3$
- $\phi$: Air–water volume ratio

**Subscripts**

- $av$: average
- $cond$: conduction
- $conv$: convection
- $f$: forward
- $hyd$: hydraulic
- $in$: input
- $l$: liquid
- $m$: main channel
- $r$: reverse
- $rad$: radiation
- $s$: side channel
- $transfer$: transferred
- $v$: vapor

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**Fig. 1.** Standard schematic configuration of a closed-loop PHP. At the evaporator section heat is added to the system while at the condenser section heat is removed. Due to the capillary size of the channel the working fluid naturally distributes itself into liquid slugs (grey) and vapor plugs (white). Pressure variations inside the system trigger movement of the working fluid which transfers heat from the evaporator to the condenser.
where $\Delta p_f$ is the reverse flow pressure drop and $\Delta p_p$ the forward flow pressure drop for flow rate $Q$. A large number of different rectifying structures exist [23,25], but the Tesla-type valve [26] is the most promising option to apply in a PHP. It is easy to integrate, has a low additional flow resistance in the promoted direction [27] and could enhance the thermal efficiency due to additional mixing effects [28]. The difference in flow resistance of the Tesla-type valve results from the difference in flow path for the majority of the flow through the two separated channels of the valve for both flow directions [24].

Thompson et al. [20] were the first to apply Tesla-type valves in a PHP which was designed for flow visualization using neutron radiography. Eight valves were applied in the adiabatic section of a six-turn flat-plate PHP. An experimental analysis was performed and the results show that the circulation is promoted in the desired direction, and that this promotion increases with heat input. Moreover, the PHP with the Tesla-valves had a smaller thermal resistance than the standard PHP and the difference was in the order of 15–25% depending on the heat input. An additional interesting observation is that the percent-increase in thermal performance was of the same order-of-magnitude as the percent-increase of flow-directionality between both heat pipes.

In order to optimise these Tesla-type valves, a better understanding of the behaviour of a two-phase flow in the valve is needed. To the authors’ knowledge, no comprehensive research on two-phase flow in Tesla-type valves exists. For this reason, this paper provides more insight to this subject with the focus on PHP application. A new valve design is proposed of which the functionality is tested using a single-phase model and a steady two-phase flow controlled experiment. To gain better insight in the valve performance in a PHP, it is symmetrically integrated in a single-turn flat-plate transparent PHP of which the temperature and fluid motion is measured. A single-turn is used to reduce variabilities and for the reproducible behaviour. The findings can be used as building blocks for a multi-turn PHP [29,14].

2. Valve design

A new Tesla-type valve has been developed for implementation in a PHP based on two criteria. The inlets and outlets should be aligned while maintaining a compact geometry that can be easily integrated in a PHP and the diodicity should be maximised while maintaining a low pressure drop in the promoted direction. The second criteria is based on the fact that movement in a PHP is related to pressure differences. A diodicity should thus be produced while having minimum resistance in the promoted direction. Based on these criteria and the general valve design guidelines, proposed by Bardell [24], the design shown in Fig. 2 is drafted. The dimensions of this design, referred to as the D-valve, are indicated in Fig. 2A and are given in Table 1. The left channel of the valve is referred to as the side channel and the right as the main channel. The channel junctions, indicated by the grey areas in Fig. 2B, are referred to as junctions J1 and J2. The main guidelines to improve diodicity are that the side channel should be aligned with the main channel in junction 2, such that for reverse flow a significant amount of flow enters the side channel, and that the angle of the side channel with the main channel in junction 1 is high. The current design is a trade-off between these guidelines and the above mentioned criteria, however it is not fully optimised.

2.1. Single-phase model description

To verify the functioning of the D-valve a laminar single-phase model is made with COMSOL [30]. It must be noted that the goal of the numerical work was not to model the performance of the valve in a PHP-type flow, but purely investigate its efficiency in a single-phase flow. Due to the high computational costs of an accurate three-dimensional model, a two-dimensional model is used to verify the mesh dependence and make a comparison with other Tesla-type valve designs in literature. Various authors state that a two-dimensional model overestimates the diodicity compared to a three-dimensional model, however 2D simulations still give a good indication of the functioning of the valve therefore making it useful for comparing geometrical variations [22,31–33]. Since the valve will be implemented as a square channel geometry, also a 3D model is made using similar mesh settings to compute the actual diodicity. The results of this model are used to show the similarities and differences in diodicity between the 2D and 3D model and can be used to compare with experimental results. The working fluid used for the simulations is water at 20 °C. The flow is assumed to be incompressible and transient effects can be neglected for determining the diodicity, i.e. the efficiency [24]. Furthermore, isothermal conditions are considered, thus gravity
effects can be disregarded. Therefore, the following Navier–Stokes and continuity equations are solved by the model:

\[ p(u \cdot \nabla)u = \nabla \cdot [-pI + \mu(\nabla u + (\nabla u)^T)] \]  

\[ p \nabla \cdot (u) = 0 \]  

The geometry shown in Fig. 2 is meshed with a free-tetrahedral mesh with refinement of the mesh where large velocity gradients are expected, i.e. near channel junctions and the walls. Mesh independence was verified with the 2D model, using the same criteria as Gamboa et al. [31], i.e. by doubling the number of elements starting with a coarse mesh until the solution changed less than 4%. This resulted in a mesh of 44,000 elements for the 2D model and 1260,000 elements for the 3D model.

A forward and reverse flow case is studied for a range of Reynolds numbers. The Reynolds number is defined as \( Re \equiv \rho D_{hyd}u / \mu \) with \( \rho \) the fluid density, \( D_{hyd} \) the hydraulic channel diameter, \( u \) the characteristic fluid velocity and \( \mu \) the fluid viscosity. The \( Re \) of the liquid slugs in a PHP can be of an order of \( 10^2 \) to \( 10^3 \) [29], but Thompson et al. [32] indicate that transitional flow behaviour can exist in a 3D Tesla-type valve for \( Re \geq 300 \). The modelled \( Re \) range is therefore determined by the convergence of the model, i.e. the part where pure laminar flow exists. In forward flow the bottom boundary is set as the inlet and the top boundary as the outlet, both indicated with I/O in Fig. 2. This is the other way around for reverse flow. At the inlet, a laminar inflow condition is applied using an entrance length to have a fully developed inlet flow [34]:

\[ l = 0.056 \cdot D_{hyd} \cdot Re_{in} \]  

where \( Re_{in} \) is the Reynolds number based on the average inlet velocity and hydraulic diameter. A zero pressure condition is applied at the outlet. The other boundaries (solid black lines in Fig. 2) have a no-slip boundary condition. The symmetry of the problem is used by applying a symmetry and no-slip boundary condition at the additional symmetry plane and bottom boundary of the 3D model respectively. Thus, only a 1 mm deep geometry, corresponding to a hydraulic diameter of 2 mm, has to be solved which saves computation time. For the 2D model a direct PARDISO solver is used and for the 3D model an iterative GMRES solver. The relative tolerance is 10\(^{-4}\). The pressure drop in each direction is calculated by evaluating the average pressure on the evaluation lines/surfaces, indicated with P1 and P2 in Fig. 2B. Computing the pressure drop for each flow direction and dividing these results in the diodicity for the corresponding Reynolds number.

2.2. Valve characteristics

A surface plot of the velocity magnitude, plotted in grey-scale, obtained by the 2D model in both directions for \( Re = 200 \) is shown in Fig. 3. Reverse flow is shown on the left and forward on the right. Furthermore, uniformly positioned streamlines and the velocity field on four lines, indicated by arrows, are also plotted. As mentioned, the working principle of a Tesla-type valve is based on a difference in flow distribution between both flow directions, which results in a difference in pressure drop for the same inlet flowrate. When the pressure drop would be identical for both directions, this will result in a difference in flowrate. As can be seen in Fig. 3, 83% of the flow goes through the main channel in the forward case while 55% goes through the main channel in the reverse case. This difference in distribution is dependent on the inertia of the flow and therefore inherently increases with \( Re \). The flow through the side channel has to be redirected downstream, which requires significant pressure work. When combined with the main channel flow, it also creates a high shear region in junction 2. For the laminar range, these are the main sources of diodicity in a Tesla-type valve [24].

As shown in Fig. 3, a clear recirculation zone exists in the side channel for the forward case resulting in a small separated jet. This jet shows to be more pronounced in 3D. The 2D forward case does not converge for \( Re > 650 \), while the reverse case is still converging. With only the forward case of the 3D model not converging for \( Re > 200 \), this phenomena is believed to result in transitional flow behaviour causing the non-convergence of the laminar model. The diodicity up to \( Re = 200 \) of the 2D and the 3D model is shown in Fig. 4, indicating the difference between an infinitely deep and a square channel.

To verify the performance of the developed valve the geometric values of several other valve designs are copied from literature, scaled to the same channel size and implemented in the 2D model. Scaling the valves has no influence on the relation between the Reynolds number and the diodic performance in single-phase laminar flow [27]. The same inlet length (L1 in Fig. 2A) and pressure evaluation distance (L2 in Fig. 2A) is used for all the designs. A distinction can be made between the normal Tesla valves used for micro-pump application and the Tesla valves for PHP application. The integration of the alignment of the inlets into the valve geometry does reduce the maximum diodic potential of the valve. The computed single-phase diodicity of the different designs is shown in Fig. 5 with the valve designs indicated in the
The micro-pump valves are the traditional T45A valve (square marker) [24], the T45C valve (asterisk marker) [24] and the GMF valve (circle marker) [31]. The T45C valve and GMF valve are the result of improvements by Bardell [24] and an optimisation study performed by Gamboa et al. [31] on the traditional Tesla-valve respectively. The fourth design (triangle marker) is the design which was already applied in a PHP by Thompson et al. [20] referred to as the TMW valve. The TMW valve only converged for \( Re/C_20^2 < 250 \). However, when comparing the TMW valve with the proposed new D-valve, both designed for PHP application, it can be clearly observed that the D-valve design produces a higher diodicity than the TMW design. Due to the difference in inlet and outlet alignment and path-length between the different designs a better comparison of effectivity is based on the absolute pressure difference between reverse and forward flow, which is shown in Fig. 6. This proves that the proposed D-valve design is a properly performing Tesla-type valve, having an absolute pressure difference close to the T45C valve which is very promising especially concerning that the alignment of the inlets and outlets is incorporated in the design. The GMF valve results show the potential of a full geometrical optimisation, which could still be performed on the developed geometry.

The single-phase modelling illustrated the basic functioning of the developed valve in single-phase flow. Furthermore, the comparison with other Tesla-type valves from literature showed that the new design is a proper functioning Tesla valve in single-phase flow that can be easily integrated in a PHP. However, the final implementation of the valve will be in a two-phase flow with heat transfer. Modelling this flow is very complex and time-expensive, therefore it is chosen to be studied experimentally.

3. Experimental set-up and procedure

To identify the operational characteristics of the D-valve in PHP-type flow, two experiments are performed. First a steady liquid–gas slug-plug flow through the valve is studied with the ‘two-phase experiment’. Secondly, the valve is integrated in a single-turn PHP, where the diodic performance of the valves and the effects on the motion composition and thermal performance of the PHP are studied.

3.1. Two-phase experiment

To the authors’ knowledge, no research on the functioning of a Tesla valve in two-phase flow has been performed. To study this, a test device is developed with which a steady liquid–gas flow through the valve can be observed. The goal of this experiment is to identify the behaviour of the valve in two-phase flow and verify whether diodicity is produced. The test device is schematically shown in Fig. 7. It comprises out of two 140 × 40 mm² transparent PMMA plates. A square 2 mm deep channel, illustrated by the dashed line pattern in Fig. 7, is milled into the 4 mm thick bottom plate. The channel cross-section relates to a commonly used hydraulic diameter for water filled PHPs [11]. Related to the Bond number, which is a dimensionless number characteristic for the importance of surface tension forces to body forces, this hydraulic diameter is small enough for the water to naturally form liquid slugs and vapor plugs. This is essential for the PHP to be functional and is governed by:

\[
D_{hyd} \approx \frac{2}{\sqrt{\frac{\gamma}{g(\rho_1 - \rho_v)}}}
\]  

(5)

Fig. 4. Plot of the single-phase diodicity against the Reynolds number computed by the 2D and 3D model for the D-valve design.

Fig. 5. Plot of the single-phase diodicity against the Reynolds number computed with the 2D model for the GMF [31], T45C [24], the proposed D, T45A [24] and TMW [20] valve as indicated in the legend from top to bottom respectively.

Fig. 6. Plot of absolute single-phase pressure difference between forward and reverse flow against the Reynolds number computed with the 2D model for the GMF [31], T45C [24], the proposed D, T45A [24] and TMW [20] valve as indicated in the legend from top to bottom respectively.
where \( \gamma \) is the surface tension, \( g \) the gravitational constant, \( \rho_l \) the liquid density and \( \rho_g \) the vapor density of the working fluid \([10,35]\).

Holes for the liquid inlet (IL), gas inlet (IG), outlet one (O1) and outlet two (O2) are made in the 1 mm thick top plate using a laser cutter. Double-sided adhesive tape (thickness \(-0.1\) mm), with the dashed line pattern cut-out, is used to bond the two plates together.

The working fluid is water dyed with blue food colourant (BPOM RI MD. 263109077128) for contrast with a ratio of 125:1. Ambient air is used as the working gas. Both the working fluid and gas had a temperature of 22 °C. The water and air are simultaneously pumped with a 1:1 volume ratio into the inlets by a syringe pump (Nexsus 3000) using 50 ml syringes. Thereby, a perfectly slug-plug alternating flow is created with slug and plug lengths between 2 and 3 mm as illustratively shown in Fig. 7. The initial meandering channel is added to reduce fluctuations in the flow by increasing the total flow resistance of the device. By closing-off one of the two outlets the direction of flow through the valve can be chosen. To quickly attain a steady-state flow and diminish the effect of initial distribution, the device is first shortly flushed with a high flow rate after which the flow rate is instantly reduced to the required value. Due to the compressibility of the air in the syringe there is a short adjustment period after which the steady-state solution for that flow rate is established. Then, using a camera (DFK 23UP031) with a 16 mm F1.6 TV lens, the flow is recorded for 9 s at a frame rate of 700 frames per second. Two repetitions were performed for each flow rate. The field whereupon the PHP is attached using two M3 bolts. Thermal paste (Electrolube – HTSP50T) is used to enhance all the thermal contacts. This block acts as a heat sink and relates to the condenser area, initially having a 22 °C room temperature. For the evaporator section an additional custom made aluminium block (40 × 40 × 67 mm) is attached to the back flange (Hositrad – ISO160B, indicated with number 6 in Fig. 9) of the vacuum chamber whereupon the PHP is attached using two M3 bolts. Thermal paste (Electrolube – HTSP50T) is used to enhance all the thermal connections. This block acts as a heat sink and relates to the condenser area, initially having a 22 °C room temperature. For the evaporator section an additional custom made aluminium block (40 × 40 × 67 mm) is attached to the back flange (Hositrad – ISO160B, indicated with number 6 in Fig. 9) of the vacuum chamber. A frame rate of 60 frames per second is used and LED-strips consisting of 66 individual LEDs (Velleman – LB12M110CWN) are placed on the inside wall of the vacuum chamber (Hositrad – HHVP-ISO160, number 8 in Fig. 9) of the vacuum chamber. A frame rate of 60 frames per second is used and LED-strips consisting of 66 individual LEDs (Velleman – LB12M110CWN) are placed on the inside wall of the vacuum chamber for illumination. Furthermore, two K-type thermocouples located at T1 and T2, as shown in Fig. 8, were used to determine the thermal resistance of the PHPs. Thermocouple data was retrieved every 0.5 s.

The PHPs are placed inside a vacuum chamber (Hositrad – Cross Reducing 6-Way – ISO-K 160/NW16KF/NW25KF) for thermal insulation, which is illustratively shown in Fig. 9. Indicated with the number 3 in Fig. 9, a custom made aluminium block (40 × 40 × 67 mm) is attached to the back flange (Hositrad – ISO160B, indicated with number 6 in Fig. 9) of the vacuum chamber whereupon the PHP is attached using two M3 bolts. Thermal paste (Electrolube – HTSP50T) is used to enhance all the thermal connections. This block acts as a heat sink and relates to the condenser area, initially having a 22 °C room temperature. For the evaporator section an additional custom made aluminium block (40 × 40 × 67 mm) is attached to the back flange (Hositrad – ISO160B, indicated with number 6 in Fig. 9) of the vacuum chamber. A frame rate of 60 frames per second is used and LED-strips consisting of 66 individual LEDs (Velleman – LB12M110CWN) are placed on the inside wall of the vacuum chamber for illumination. Furthermore, two K-type thermocouples located at T1 and T2, as shown in Fig. 8, were used to determine the thermal resistance of the PHPs. Thermocouple data was retrieved every 0.5 s.

The PHPs are evacuated with a vacuum pump (Vacuubrand – PC 8/RC 6) to a level of 10\(^{-3}\) mbar and subsequently filled for 50% with de-gassed and de-mineralised water to which blue food colourant (BPOM RI MD. 263109077128) is added with a ratio of 125:1. The PHP is filled using a 3-port switching valve (Upchurch V-100T) with a gas-tight syringe (Hamilton 1001 TILL SYR) and sealed with a shut-off valve (Upchurch P-732). Due to the permeable behaviour of PC the filling procedure is repeated before every experiment to have consistent conditions. Directly after filling the PHP the vacuum pump is connected to the chamber which is evacuated to a pressure level of 1 mbar. When this level is reached the heater is switched on and the measurement starts. For safety reasons, the maximum allowable heating block temperature is 120 °C. Due to

3.2. PHP experiment

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\[\text{Fig. 7. Schematic image of the two-phase test device. The indicated sizes are in millimetres and the dimensions of the valves are identical to those in Fig. 2. The dashed line pattern represents the 2 mm deep channel geometry which is milled into the 4 mm thick bottom plate. IL, IG, O1 and O2 indicate the liquid inlet, gas inlet, outlet one and outlet two respectively, which are holes through the 1 mm thick top plate. F1 and F2 indicate the field of view of the camera used for computing the bubble velocity in forward and reverse flow respectively. The flow-type, alternating liquid slug (dark grey) and gas plug (white) flow, created with this set-up is illustratively shown in the initial meandering channel. Gravity is directed in the negative Z-direction, with the device in the XY-plane.}\]
the relative high thermal resistance of the polycarbonate, no equilibrium temperatures could be attained before this maximum temperature heating block was reached. Therefore the transient behaviour of the PHPs is measured for 30 min. Three experiments are performed for each heat pipe. The energy balance of this system can be written as:

\[ q_{\text{in}} = q_{\text{transfer}} + q_{\text{cond}} + q_{\text{conv}} + q_{\text{rad}} + q_{\text{stored}} \]  

where \( q_{\text{in}} \) is the heat input from the cartridge heater (10 W), \( q_{\text{cond}} \) the heat that is conducted through the device and attached cables, \( q_{\text{conv}} \) the heat loss to the environment by convection, \( q_{\text{rad}} \) the radiated heat to the environment and \( q_{\text{stored}} \) the actual transferred heat by the working fluid. \( q_{\text{stored}} \) is the stored sensible heat, since the transient behaviour is considered. In order to distinguish the \( q_{\text{in}} \), three additional measurements are performed for each PHP where the temperature rise of the evacuated but unfilled PHPs is measured. The maximum contribution of the stored sensible heat in the water, based on the maximum possible temperature rise of 98 °C of the water over 30 min, is less than 1% of the total heat input and can therefore be neglected. The total power-use of the LEDs is 4 W of which a large amount is converted into heat. The strips are placed, sufficiently far away from the back flange, inside the vacuum chamber on the wall which acts as a heat sink. Therefore, due to the high thermal mass of the vacuum chamber wall and the natural convection on the outside of the chamber, the influence of this heat to the system can be neglected.

3.3. Image analysis

The Image Processing Toolbox of MATLAB [36] is used to process the acquired data. For the two-phase experiment, image registration [37] is used to compute the movement of the bubbles in the valve between two successive frames. A masking procedure is applied, as shown in Fig. 10, to separately process the inlet (1) and side channel (2) of the valve. With the translation of each section between two successive frames the bubble velocity can be determined.

The acquired movies of the total valves can be used to determine the air–water volume ratio throughout the valve between two successive frames. A masking procedure is applied, as shown in Fig. 10, to separately process the inlet (1) and side channel (2) of the valve. With the translation of each section between two successive frames the bubble velocity can be determined.

The accuracy of the method is determined by averaging the
CCW over clockwise (CW) circulatory flow, is calculated using the ratio of the number of occurrences of counter-clockwise transport. The acquired velocity is therefore representable for the fluid between two subsequent frames. Since the valves also distribute with the longitudinal difference of the centre-of-mass location weighted centre-of-mass is determined. A velocity is computed channels, the background is subtracted and the grey-level-partly blocking the view of the channel. For each frame, for both loop are discarded due to the thermocouples and shut-off valve TV-PHP, are processed separately. The top and bottom part of the condenser being moved through the evaporator and (partly)

\[ P \quad \frac{\text{CTR}}{\text{CCW}} \quad \text{CW} \]

Fig. 10. Masking procedures used for the two-phase experiment. The left image shows masking procedure used for image registration, where 1 indicates the inlet channel section and 2 the side channel section. The right image shows the applied masks for determining the air ratio in the main (A) and side (B) valve channel. An individual grey-scale frame of a reverse flow case with and average inlet velocity of 0.21 m/s is shown on top and below the corresponding binary image.

time-averaged \( \phi \) of both channels and comparing this with the prescribed 1:1 volume ratio. This showed to maximally deviate 3%.

In the PHP experiment the total PHPs are imaged by the camera to examine the motion when this is consistent, i.e. in a quasi-steady state as defined by Khandekar et al. [29]. The velocity of the flow is determined with the centre-of-mass method [38]. Both the left and the right part of the loop, thus including valves for the TV-PHP, are processed separately. The top and bottom part of the loop are discarded due to the thermocouples and shut-off valve partly blocking the view of the channel. For each frame, for both channels, the background is subtracted and the grey-level-weighted centre-of-mass is determined. A velocity is computed with the longitudinal difference of the centre-of-mass location between two subsequent frames. Since the valves also distribute fluid in the transverse direction this centre-of-mass method is used. The acquired velocity is therefore representable for the fluid transport.

A circulation-tally-ratio (CTR), defined by Thompson et al. [20] as the ratio of the number of occurrences of counter-clockwise (CCW) over clockwise (CW) circulatory flow, is calculated using the same criteria as Thompson et al. [20].

\[
\text{CTR} = \frac{\sum \text{CCW}}{\sum \text{CW}}
\]

Circulatory flow relates to almost all the fluid located in the condenser being moved through the evaporator and (partly) moved back to the condenser again. The CTR indicates in which direction circulation is promoted.

4. Results

4.1. Two-phase experiment

The measured side channel bubble velocity \( u_s \) plotted against the mean inlet velocity \( u_{in} \), based on the prescribed total flow rate, is shown in Fig. 11 for both flow directions. The measured inlet channel bubble velocity showed to have a maximum deviation of 5% from \( u_{in} \), at the highest flow rate. As a reference, the mean side channel velocity computed with the single-phase 3D model for water at 22 °C is plotted versus the mean inlet velocity. The range of the forward case is limited by the non-convergence of the model for \( Re > 200 \). Blockage of the side channel by air bubbles occurred for lower flow rates for both flow directions. Meaning that after flushing, the flow in the side channel would stagnate and the total flow would go through the main channel. In that case the applied pressure on the side channel is not large enough for the bubbles to remain dynamic [39]. The side channel was blocked up to \( u_{in} \approx 0.24 \) m/s for forward and \( u_{in} \approx 0.16 \) m/s for reverse flow. Moreover, as can be observed in Fig. 11, a significant difference exists between the forward and reverse side channel velocity \( u_s \).

The air–water distribution was not identical for the main and side channel of the valve for reverse flow, but a factor of about 2.3 higher in the main channel than in the side channel as is shown in Table 2. For forward flow at \( u_{in} \approx 0.24 \) m/s, the air–water ratios for the main and side channel were similar. Considering the accuracy of the method, this indicates that the air–water distribution does not change throughout the valve in forward flow. In addition, the bubbles were almost never split-up in forward flow. In the contrary, for higher inlet velocities, the sharp point of the middle section in junction 2 often splits up bubbles in reverse flow resulting into two smaller bubbles moving through the main and side channel separately. This phenomena can also be seen in Fig. 10.

A clear difference between the modelled single-phase and measured side channel velocity is observed. This could be caused by principal differences between two-phase and single-phase flow like surface tension effects, gas compressibility or momentum differences between gas and liquid. Yet, since the bubbles do not always completely block the side channel in reverse flow, it cannot be ruled out that this difference can also be related to the liquid flowing past the bubbles in the side channel resulting in a difference of fluid velocity and measured bubble velocity.

The difference in blockage, redistribution of working fluid and difference in velocity indicate that there is a large difference of mass flow distribution between the different directions. Since dierictivity is produced by a difference in flow distribution, this can be related to a difference in flow resistance based on the structure of the valve. Unfortunately, the absolute value of this dierictivity in two-phase flow could not be measured. Measuring this would require very accurate (\( \Delta p \sim 10^2 - 10^3 \) Pa), properly applied and in two-phase flow applicable pressure sensors. Nonetheless, the

Fig. 11. Plot of the measured bubble velocity in the side channel \( (u_s) \) of the valve for both flow directions against the prescribed mean inlet velocity \( (u_{in}) \) with the standard deviation indicated as error bars. The mean side channel velocity versus the mean inlet velocity computed with the 3D single-phase model for water at 22 °C is also shown for comparison. The modelled forward flow is only plotted up to \( Re \sim 200 \) due to non-convergence of the model for higher Re.
side channel velocities could be measured and showed similar behaviour for the single-phase model and two-phase experiments. Therefore, the main goal of this experiment, which was to find the two-phase flow characteristics of the valve and verify whether diodicity is produced, is achieved.

4.2. PHP experiment

4.2.1. Thermal resistance

Fig. 12 shows the measured thermal resistance of the empty and filled PHPs. For the first 400s the set-up is heating up and the data is identical. No clear difference is observed in the start-up behaviour of the N-PHP and the TV-PHP.

The average of both empty PHPs deviated maximally 0.7% from each other, therefore these results are combined and plotted together for every 50s with the standard deviation of the total shown as error bars. The empty PHP measurements are prematurely terminated because the heating block exceeded 120 °C. Therefore, the data is fitted and extrapolated with an exponential function, illustrated by the dotted line, to give an indication of the full scope.

The mean of the three experiments of both filled PHPs is plotted together with the standard deviation illustrated by error bars at every 50s. Both reached a quasi-steady state after 15 min of heating up. This quasi-steady state mostly consists of (alternating) circulatory flow alternated with an occasional stagnant situation with minor perturbations where almost all the liquid is situated at the condenser. Fig. 12 clearly shows that adding the valves to the PHP improves the thermal resistance. The improvement is around 14% during the end of the experiment.

4.2.2. Fluid motion

When circulation occurs, the liquid slugs located at the condenser move to the evaporator. Subsequently, fast boiling results in annular flow in the opposite channel which transports the liquid at the front part of the slug back to the condenser. This continues until a significant bubble expansion clears the evaporator section of most liquid. When averaged over the full experiment, the vapor/liquid ratio will be larger in the evaporator than in the condenser section. This is similar as is observed in normal PHPs. The most significant shift in centre-of-mass occurs when the condenser is flushed and the slugs are moved to the evaporator. That is due to the fact that the upwards flow is mostly annular flow which does not give a good representation of the actual flow velocity using this method. Therefore, the most characteristic velocity for clockwise and counter-clockwise flow is the negative shift in centre-of-mass in the right and the left channel respectively. For this reason these velocities are compared to verify whether there is a difference in resistance for the different flow directions. To eliminate minor movement, a threshold of 0.025 m/s is used and the remaining points are averaged for each experiment. The resulting velocities, averaged over all the experiments, are shown for both PHPs in Table 3. Also the CTR is given, which is calculated with the number of times that a circulation occurred in counter-clockwise over clockwise direction.

First of all, the TV-PHP shows to have a 25% higher velocity in the counter-clockwise (forward) direction than the clockwise (reverse) direction. On the contrary, a difference of 1.8% between the velocities indicates that there is practically no difference in flow resistance in the N-PHP. This shows, that integrating the valve in this PHP produces a diodicity, resulting in a difference in fluid transport resistance between both flow directions. It can also be seen that the addition of the valves, in a PHP with this flow behaviour, slightly lowers the average absolute velocity.

The N-PHP showed to have a CTR close to 1 which indicates that there is no preferential flow direction. But the TV-PHP CTR shows that circulation occurred almost twice as often in the clockwise direction. Considering that clockwise is the more resistive direction (reverse flow direction in the valve), this is the opposite of what one might expect. It is believed to be related to gravitational forces. This hypothesis is in agreement with the flow behavior inside a normal single-turn PHP of similar size, as mentioned by Khandekar and Groll [14]. During the start-up phase and during motion stops, accumulation of liquid occurs mostly on the (top-) right side of the TV-PHP (Fig. 13), since the flow resistance is the least in the counterclockwise direction. It is observed experimentally that if the liquid/vapor ratio in that right channel is on average around 4, a gravitational driven motion will occur in order to redistribute the liquid in both channels such that the gravitational forces are balanced again. This results in a clockwise rotation. Once in a while also a liquid accumulation in the top-left part of the TV-

### Table 2

| Flow direction | Air–water ratio in main channel | Air–water ratio in side channel |
|----------------|---------------------------------|---------------------------------|
| Reverse flow   | $\Phi_m = 1.46$                 | $\Phi = 0.64$                  |
| ($u_{in} = 0.24$ m/s) |                                 |                                 |
| Forward flow   | $\Phi_m = 0.98$                 | $\Phi = 1.01$                  |
| ($u_{in} = 0.24$ m/s) |                                 |                                 |

### Table 3

| Velocity   | N-PHP    | TV-PHP   |
|------------|----------|----------|
| CW velocity [m/s] | 0.109    | 0.071    |
| CCW velocity [m/s] | 0.107    | 0.089    |
| Circulation occurrences CTR [-] | 0.99      | 0.51      |
The valve is implemented in a single-turn transparent bottom-heated PHP to study its effect on the heat transfer and flow behaviour. The valve produced diodicity, which resulted into a difference of 25% in velocity for the different flow directions. Due to the gravitational influence on the flow in a PHP of these dimensions, the diodicity of the symmetrically placed valves resulted in a promotion of circulation in the more resistive direction by a factor of almost 2. The addition of the valves resulted in a decrease of thermal resistance of around 14%, when compared to an identical PHP without valves.

The developed experimental set-ups and procedures are shown to be effective and low-cost tools to study two-phase flow with heat transfer phenomena in a Tesla-type valve. The proposed valve showed to improve the thermal performance of the PHP, produce diodicity and create preferential motion. However, when applied in a gravity dependent PHP, the valve placement should be such that the valves produce gravitational differences which promote circulation in the desired direction. Decreasing the channel dimension of the PHP increases the relative importance of the diodic-induced pressure differences over gravitational pressure differences. This study shows that features can be incorporated which can improve the thermal performance of pulsating heat pipes and that Tesla valves should be considered as a viable option for the promotion of circulation in pulsating heat pipes.

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