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# Heat transfer enhancement by dynamic corrugated heat exchanger wall: Numerical study

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**Abstract.** A new concept of heat exchanger at sub-millimeter scale is proposed for applications in cooling on-board electronics devices, in which the quality of the exchanges between fluid and wall is very critical. In the proposed system, the upper wall of the channel is deformed dynamically to obtain a sinusoidal wave on this surface. The lower wall is exposed to constant heat flux simulating the imprint of an electronic component. A systematic 3-D numerical study in transient regime on the different deformation parameters allowed obtaining both the pumping characteristics and the heat transfer characteristics of the system. It was observed that the dynamic deformation of the wall induces a significant pumping effect. The intensification of the heat transfer is very important even for highly degraded waveforms, although the pumping efficiency is reduced in this case.

**Keywords:** heat transfer enhancement, wall morphing, dynamic deformation, pressure gain, millimetric heat exchanger

## 1. Introduction

Increasing demands in many industrial fields regarding the thermal systems, particularly in devices that need to be miniaturized as in the case for example, cooling of embedded electronic chip, has contributed to the development of new and efficient technologies involving small diameter channels. If these systems can improve the compactness and specific exchange surfaces, a major drawback is the difficulty of disrupting the boundary layer to increase the heat and mass transfer coefficients, the flow being generally strongly laminar because of the small characteristics size of the channel. Moreover, these devices generally lead to very large pressure drop values. Indeed, to ensure adequate flow in the system and thus limiting the rise in coolant temperature, high fluid velocities need to be achieved leading to significant pressure drop and therefore high mechanical powers. It is also important to note that the flow distribution in the individual channels constituting the heat exchanger can be difficult to control, and thus generating additional pressure drop. One possible way to simultaneously disrupt the boundary layers and integrating the pumping function within the heat exchanger is to generate a dynamic deformation of one of the channel walls. Like a peristaltic pump, the fluid is thus made to flow. The advantage here is that the integration of this pumping function within the heat exchanger



allows *a priori* generating simultaneously the disturbances in boundary layers and thus an intensification of the heat transfer.

Very few studies on the effects on dynamic deformation on flow and heat transfer inside channels are available in the literature. Nakamura et al., [1] investigated numerically the influence of the wall oscillation on the heat transfer characteristics in a two-dimensional channel. Khaled and Vafai [2] studied the flow and heat transfer inside oscillatory incompressible thin films. Kumar et al. [3] studied heat transfer inside circular tubes of millimetre size with static and mobile sinusoidal corrugated walls. In their work, numerical analyses are reported to study the effect of the spatial wavelength ( $\lambda = 1/2, 2/3, 1, 2$  mm), the Reynolds number (1-120) and the amplitude (1 -20% of the diameter of the tube) on the heat transfer coefficient and pressure losses. The heat transfer coefficient for mobile corrugated walls, considering the standing wave, is enhanced up to 35-70% for all the considered frequencies, with respect to the case where the wall was static. A sharp decrease in pressure drop (by a factor of 1.2 to 5) was also observed in the case of high amplitudes. However, no general trend for such changes in heat transfer and pressure drop values in terms of operating parameters was found. Leal et al., [4] showed that the dynamic deformation of the wall of a straight channel is an effective technique for increasing heat transfer, and simultaneously creating a pumping function in the device.

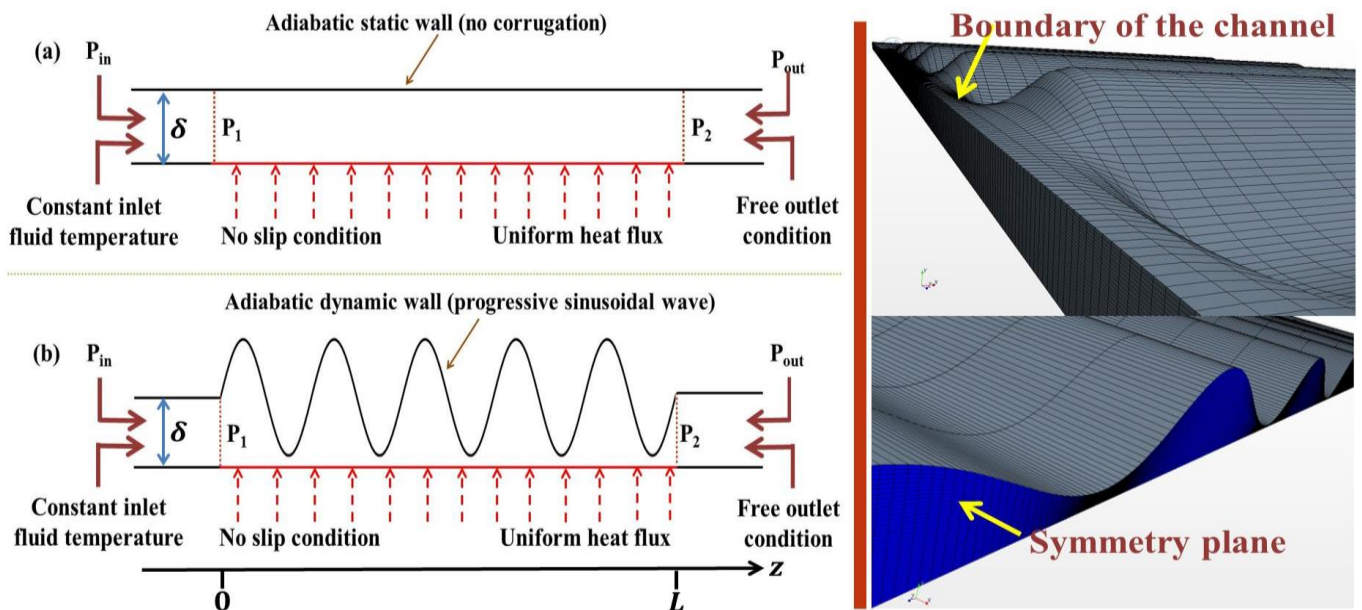
According to these studies, the dynamic deformation of a wall efficiently enhances the heat transfer. Additionally, if the amplitude of the deformation is high enough, a peristaltic pumping can be obtained. It therefore appears possible to realize an interesting multifunctional device using such a deformable system to enhance the heat transfer performance as well as eliminate the requirement of an external pump. Indeed, the configuration proposed by Leal et al., [4] was an academic one in the sense that both the geometry and the boundary conditions were idealized. For instance, it was considered that there were no boundary effects on the wave, while in practical applications the deformable wall has to be fixed on channel borders. It was also assumed that the pressure difference between the ends of the channel was very low, at a constant value, whatever the value of the imposed heat flux. With these assumptions, namely high heat fluxes and low pressure-difference, the temperature at the outlet of the channel may reach not reasonable values.

The aim of the present paper is to study the thermo-hydraulic performances of a mobile corrugated channel within a heat exchanger by taking into account realistic boundaries and geometry. As previously indicated, more realistic connection between the deformed wall and the static one at the boundaries has to be taken into account, as this transition between fixed and mobile wall zone could impact the flow behaviour. Two zones have also been added, at the inlet and at the outlet of the actuated zone, to obtain more realistic velocity profile at the ends of the heat exchanger.

## 2. Virtual prototype

In order to study the fluid flow and heat transfer characteristics in a small scale heat exchanger, a virtual prototype is developed in the form of a dynamically deformed corrugated channel initially based on flat and straight channel as presented in figure 1-left (detailed in section 3).

In the studied configuration, the lower wall is fixed and subjected to a constant and uniform heat flux over a given imprint length, facing the actuated zone on the upper wall. The latter (dynamically deformed wall) is considered adiabatic. The distance between the two walls ( $\delta$ ) is 1 mm, while the length of the heated zone (L) and width (W) of the channel are both equal to 50 mm. The lengths of the inlet and outlet zones (static) of the channel are long enough (50 mm) to ensure a hydro-dynamically established flow at the entrance of the heated zone, and to minimize boundary constraint at the exit of the actuated zone. The sinusoidal movement of the upper wall (membrane) is gradually reduced in the transverse direction to the flow in order to simulate the fact that the membrane is fixed along the side walls of the channel. In addition, a non-slip condition is imposed on the walls.



**Figure 1.** Left: 2-D schema of (a) flat channel where the upper wall is straight and static, (b) corrugated channel where the upper wall is dynamically deformed. The lower wall is uniformly heated. The upper walls (straight and deformed) were kept adiabatic. A pressure difference ( $\Delta P_s = P_{out} - P_{in}$ ) is imposed between inlet and outlet sections of the channel. Right: Different meshing views.

The influence of amplitude ( $A$ ) and imposed positive pressure difference at inlet-outlet of the exchanger ( $\Delta P_s$ ) for a given frequency ( $f$ ) and wavelength ( $\lambda$ ) on thermo-hydraulic performances is studied. Numerical simulations are performed in 3-D using the commercial software StarCCM+, considering laminar flow. The fluid used is water (at 27°C) considered non-compressible and with constant physical properties. The heat power ( $\Gamma$ ) imposed on the imprint is 125W. Parametric studies are performed for the cases where the amplitude varies from 60 to 98% of the channel height ( $\delta$ ) with a frequency of 50Hz.

### 3. Numerical simulations

The flow and heat transfer in unsteady laminar flow are solved by coupling the continuity, momentum and energy equations in 3-D. The geometry is symmetrical with respect to the vertical plane passing through the central axis of the channel. This leads to the reduction in number of mesh cells to optimize computation time by dividing the simulated configuration by two compared to its actual size. The power imposed on this half channel ( $W=25\text{mm}$ ) is thus 62.5W. Note that the fluid flow is considered in the  $z$  – direction while the width and height of the channel were considered in  $x$  – and  $y$ –directions respectively. Images of the deformed corrugated wall and mesh views are presented in Figure 1.

#### 3.1. Management of dynamic corrugated wall

Three functions have been used to impose the displacement of the mobile wall of the channel. The first describes the progressive sinusoidal wave, while the other two functions damp the movement in the vicinity of the lateral walls, and at the inlet and outlet of the actuated zone. The displacement of the upper wall (membrane) is defined by:

$$y(z, t) = \delta + A \cdot (Y_1 \cdot Y_2) \cdot \sin\{2\pi(ft + \omega z)\} \quad (1)$$

where,  $Y_1$  and  $Y_2$  are two threshold functions that takes into account the different membrane deformation effects in the vicinity of the boundaries of the actuated zone,  $f$  is the frequency,  $t$  is the time and  $\omega$  is the number of waves per unit length respectively.

The two threshold functions are defined as:  
For lateral damping:

$$\text{if } x > 0.0015m : Y_1 = 1 \quad (2a)$$

$$\text{else} : Y_1 = \left(\frac{x}{0.0015}\right)^3 \quad (2b)$$

Similarly, for longitudinal damping:

$$\text{if } z > 0.05m \text{ and if } z \leq 0.10m : Y_2 = 1 \quad (3a)$$

$$\text{if } z > 0.10m : Y_2 = \left(1 - \frac{z - 0.10}{0.05}\right)^4 \quad (3b)$$

$$\text{if } z < 0.05m : Y_2 = \left(\frac{z}{0.05}\right)^4 \quad (3c)$$

### 3.2. Thermal parameters

All thermo-hydraulic performances were calculated between  $P_1$  and  $P_2$  sections. The local heat transfer coefficient is determined from the temporal averages of the fluid and wall temperature over a period ( $\tau = 1/f$ ):

$$\bar{T}_{w,\lambda}(z) = \frac{1}{\tau W} \int_t^{t+\tau} \int_{x=0}^W T_w(x, z, t) dx dt \quad (4)$$

$$\bar{T}_{mf,\lambda}(z) = \frac{1}{\tau \delta W} \int_t^{t+\tau} \int_{y=0}^{y=\delta} \int_{x=0}^{x=W} T_{mf}(x, y, z, t) dx dy dt \quad (5)$$

and, the mean heat transfer coefficient along  $z$  –direction is given by:

$$h_\lambda(z) = \frac{q(z)}{\left(\bar{T}_{w,\lambda}(z) - \bar{T}_{mf,\lambda}(z)\right)} \quad (6)$$

In the following, the global heat transfer coefficient across the channel is then defined as:

$$\langle h \rangle = \frac{\Gamma}{S (\langle \bar{T}_w \rangle - \langle \bar{T}_{mf} \rangle)} \quad (7)$$

### 3.3. Mesh convergence criteria

The corrugation shape and fluid flow are first initialized. Then, calculations are carried out until a periodic stationary regime is reached. This latter point is checked by comparing global heat transfer and flow characteristics: calculations are performed until the values of heat transfer coefficient between two consecutive time periods differ by less than 1 %. Consequently, an additional time period is added to extract all instantaneous and time averaged values of all physical quantities.

The chosen time step is small enough to allow an accurate description of the movement of the dynamic wall and all the thermo-hydraulic quantities. In the present work, 50 time steps per period are used. It can be seen in Table 1 that very small mesh size does not impact strongly on the global properties for different amplitudes. On the other hand, increasing the mesh size leads to significant

variations in the calculated quantities. Based on these observations and the % difference in the global properties and to maintain a safety margin on the accuracy of the results, mesh size has been chosen that corresponds to 281072 cells to perform the systematic studies.

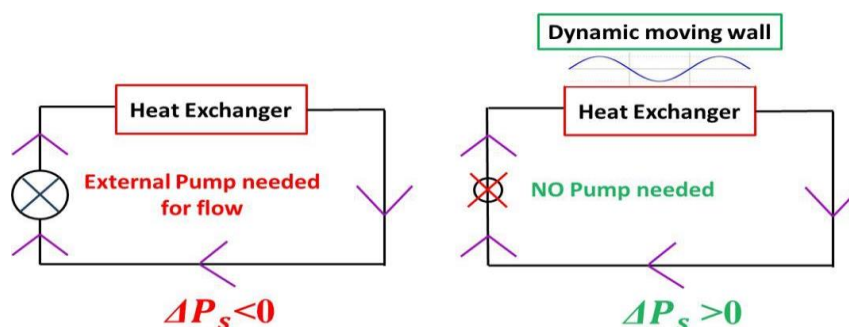
#### 4. Thermo-hydraulic results: Parametric study

Most commonly, conventional corrugated channels (heat exchangers) requires the use of an external pump to circulate the fluid in the inlet-outlet direction of the channel due to  $\Delta P_s = P_{out} - P_{in} < 0$ . In the present study, the objective is to overcome such pressure loads and obstruction to the fluid flow using dynamic corrugated channel to compensate the pressure drop due to friction within the same exchanger as well as the additional pressure drops due to the rest of the hydraulic circuit.

**Table 1.** Presentation of global flow and heat transfer properties for different number of cells and for 50 time steps per period (The grey color represents the number of cells chosen to carry out systematic studies).

Amplitude	Number of cells	Mass flow rate (kg/s)	Heat transfer coefficient ( $W \cdot m^{-2} \cdot K^{-1}$ )
60%	43560	1.11E-02	4409
	159766	1.47E-02	4382
	281072	1.43E-02	4345
80%	43560	1.63E-02	5985
	159766	2.10E-02	6190
	281072	2.04E-02	6128
95%	43560	-	-
	159766	2.66E-02	10300
	281072	2.56E-02	11413

Thus, in the studied configuration, the fluid flows in the heat exchanger from low pressure at the inlet towards high pressure at the outlet. The proposed heat exchanger with dynamic wall generates seemingly no pressure drop, as one might have with conventional heat exchangers, but pressure gain ( $\Delta P_s = P_{out} - P_{in} > 0$ ) to eliminate the need of external pump (see Figure 2).



**Figure 2.** Schema of a heat exchanger with an external pump (left) and without external pump (right) by integrating dynamic wall in the heat exchanger.

#### 4.1. Local Analysis

The result of local temperature during one period is presented (see Figure 3) to locate the local jet and the constriction zone.

##### 4.1.1 Mass flow rate

As expected, the mass flow rate depends on the amplitude of deformation imposed on the mobile wall. Increasing amplitude thus leads to a proportional increase of the mass flow rate (see Figure 4a).

However, the mass flow rate decreases with the increase of the pressure difference at a given amplitude (see figure 4b). Again, this behavior is expected. Indeed, the pressure difference between outlet-inlet of the exchanger is positive, corresponding to pressure gain that must assure the membrane actuation. An increase in this pressure gain then leads to a decrease in mass flow rate as in a "classical" pump. There is however to be noted that the mass flow rate variation is relatively small (approximately 25%) when the pressure variation varies by several orders of magnitude. The system behaves therefore from a hydraulic point of view, as a quasi-volumetric pump.

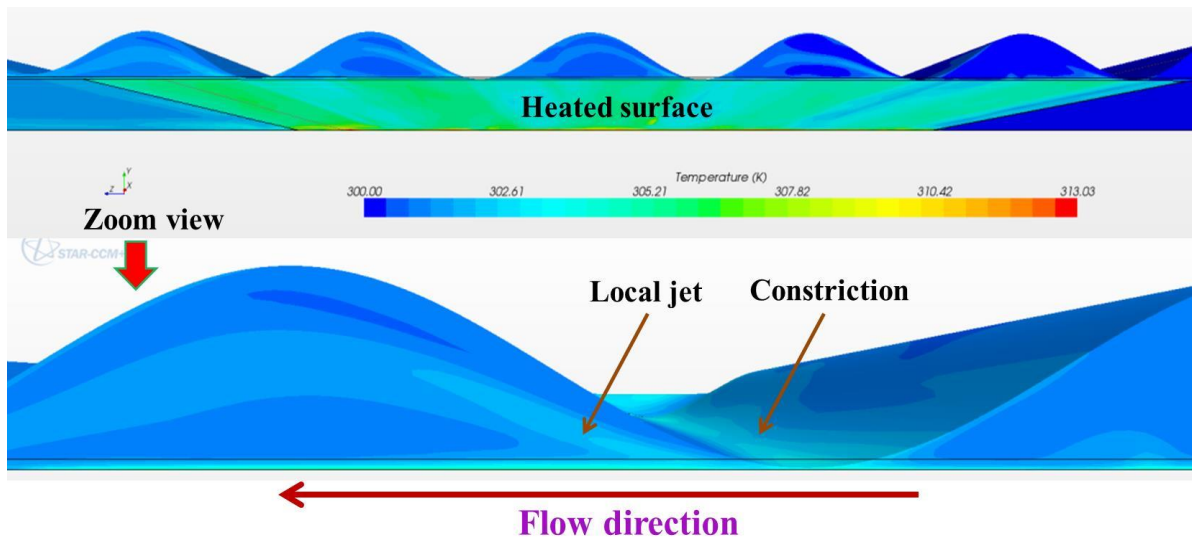


Figure 3. Instantaneous temperature field in the channel when  $A = 98\%$ .

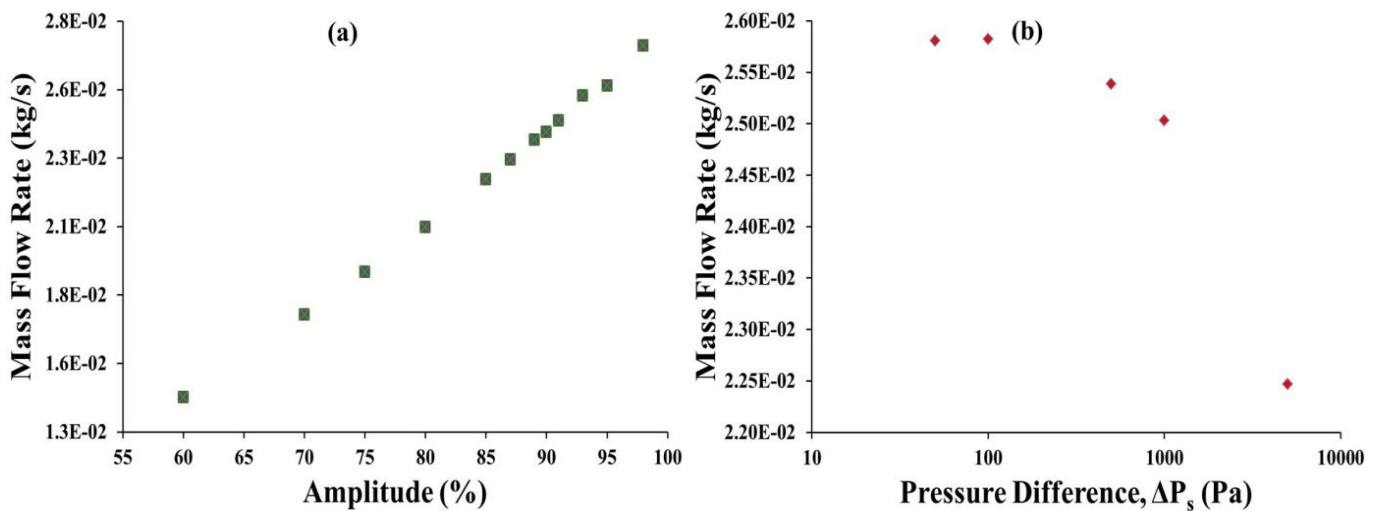
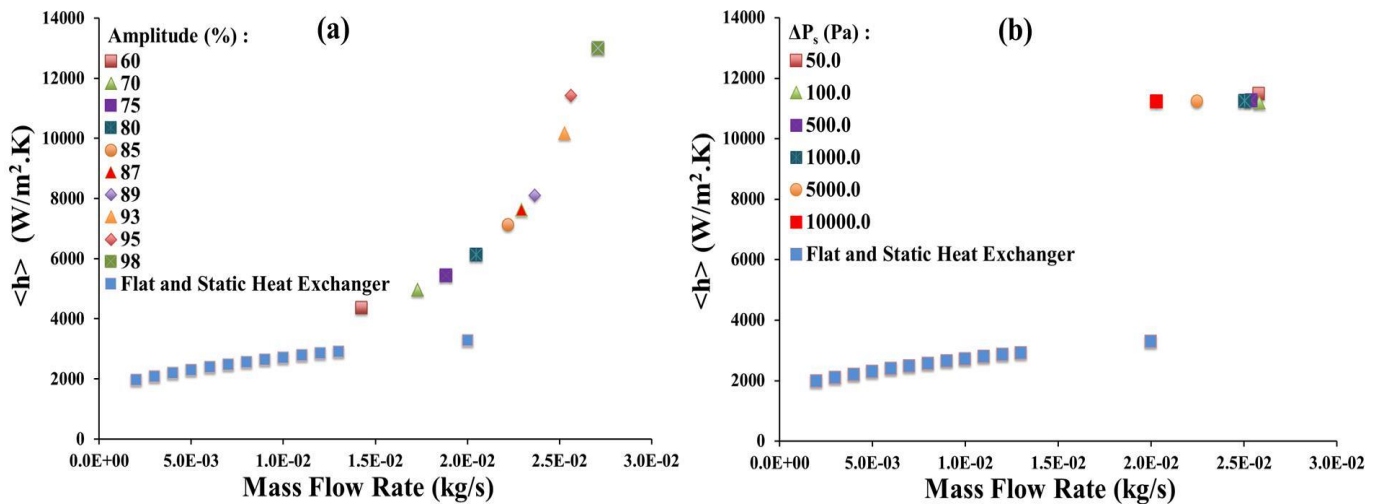


Figure 4. (a) Mass flow rate as a function of amplitude when  $\Delta P_s = 50\text{Pa}$ , (b) Variation in mass flow rate with imposed positive pressure difference for  $A = 95\%$ .

#### 4.1.2 Heat transfer coefficient

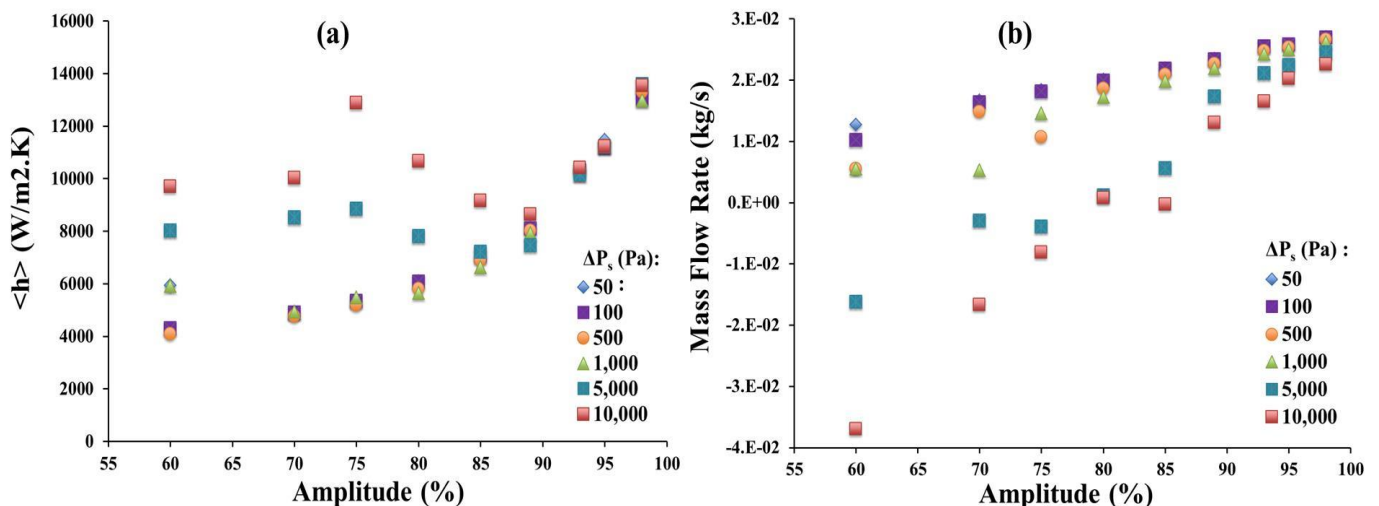
The heat transfer coefficient also depends on the amplitude applied. High amplitude leads to a significant intensification of the heat transfer compared to cases without deformation (straight channel) as presented in Figure 5a. Moreover, at very high amplitude, the heat transfer coefficient does not vary, neither as a function of pressure difference nor as a function of the mass flow rate as shown in Figure 5b. This behaviour is explained by the two following considerations:

- The dynamic corrugation of the upper wall of the heat exchanger causes a mixing intensification and therefore a reduction in the establishment lengths. The flow being laminar and very quickly established, the heat exchange becomes almost independent of the mass flow rate.
- Majority of the heat transfer occurs where the thermal resistances are the lowest, i.e. at the strongest constrictions in the channel at very high amplitude where the flow area is lowest. The value of the heat transfer coefficient is thus no longer controlled by the average size of the channel, but by the smallest dimension, leading to significant values of the heat transfer coefficient.



**Figure 5.** Comparison of global heat transfer coefficient ( $\langle h \rangle$ ) of mobile corrugated channel with flat and static channel as a function of mass flow rate for (a) different amplitudes and a  $\Delta P_s$  of 50 Pa, (b) different  $\Delta P_s$  and an amplitude of 95%.

4.2. Global Analysis



**Figure 6.** (a) Global heat transfer coefficient ( $\langle h \rangle$ ) as a function of amplitude for different  $\Delta P_s > 0$ . (b) Mass flow rate as a function of amplitude for different  $\Delta P_s > 0$ .

The use of dynamic wall at high amplitudes ( $A > 90\%$ ) allows obtaining significant heat transfer coefficient for any imposed positive pressure difference (see figure 6a) No influence of this pressure difference is found on heat transfer in that case. On the other hand, it can be observed that the

variations in heat transfer coefficient at lower amplitudes are more difficult to predict when the pressure difference between the ends of the exchanger becomes important; the evolution of the heat transfer coefficient is no longer monotonous.

However, when the amplitude is low, a high pressure difference ( $\Delta P_s=5000$  and  $10000$  Pa) leads to a high value of the heat transfer coefficient compared to the case of small values of pressure difference. This is particularly visible for an amplitude of 75% and a  $\Delta P_s=10000$ Pa. Indeed, the mass flow rate under the same conditions ( $A=75\%$  and  $\Delta P_s=10000$ Pa) is close to 0 kg/s, involving that the fluid moves back and forth in the exchanger due to high exit pressure. Similarly, mass flow rate depends on amplitude as well as on imposed pressure difference (see figure 6b). It is observed that, for high pressures at the exit, the fluid goes in the opposite direction to that of the desired direction (negative mass flow rate) at low amplitude but remains well controlled for the high amplitudes. There are significant variations in mass flow rate (negative and positive) at high value of the pressure difference and low amplitude, while at high amplitude; change in pressure difference does not impact strongly the mass flow rate.

## 5. Conclusion

A new concept of heat exchanger involving millimeter scale is proposed that consists in dynamically deforming one of the walls. The enhancement in heat transfer and mass flow rate were investigated by applying positive pressure differences in single-phase flow. It can be observed that heat transfer coefficient increases with the increase of the amplitude. The current results show that the requirement of external pump can be completely eliminated. Thus, increase in the overall performance of the system and substantial gain in terms of integration can be achieved.

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## Nomenclature

$A$	Amplitude, mm	$\langle h \rangle$	Heat transfer coefficient, $W.m^{-2}.K^{-1}$
$L$	Imprint length, mm	$\langle \bar{T}_w \rangle$	Mean wall temperature in actuated zone, K
$f$	frequency, Hz	$\langle \bar{T}_{fm} \rangle$	Mean fluid temperature in the channel, K
$q$	Heat flux, $W.m^{-2}$	$\Delta P_s$	Pressure difference between outlet – inlet, Pa
$S$	Exchange surface, $m^2$	$C_p$	Volumetric thermal capacity, $kJ.m^{-3}.K^{-1}$
$W$	Width of channel, mm	Greek symbols	
$\dot{m}$	Mass flow rate, $kg.s^{-1}$	$\delta$	Channel height, mm
$V$	Channel volume, $m^3$	$\Gamma$	Power, W
$t$	Time, s	$\tau$	Period, $s^{-1}$