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Article

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Numerical investigation on the cooling of electronics components with synthetic multi-jets and non-sinusoidal bi-periodic forcing functions

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Abstract

Recently, the cooling process for electronics components has attracted many researchers and several techniques for improving the cooling efficiency and heat transfer rate have been demonstrated. One of the best efficient techniques is the introduction of a synthetic jet and the modification of heating surface. In the present study, the form of heating surface and the signal of the diaphragm has been modified to improve the synthetic jet. These modifications are novel and have been applied for the first time with very good thermal enhancement efficiency for microchannels with synthetic jets applications. This study allowed us to make a quantitative comparison between a basic case with a periodic signal and modified case with bi-periodic signal with two cavities having a slope of 3° , 60 percent obstruction orifices and 10 μm of undulation heated wall. The unsteady flow and heat transfer for the two-dimensional synthetic jet are solved using ANSYS fluent code and $k-\omega$ (SST) model is selected to account for fluid turbulence. Obtained results showed an increase of Nusselt number by about 51% for the modified case compared with the basic case.

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Keywords: Heat transfer; Synthetic jet; (SST) $K-\omega$ turbulence model; Undulation heated wall; Nusselt number; Periodic signal; Bi-periodic signal

1. Introduction

The electronics components become more powerful and increasingly small. The components can dissipate several hundred Watts per square micro-meter, thus requires a cooling process. Several cooling techniques have been developed with the aim of improving the heat transfer rate and increasing the efficiency of the cooling systems. Recently, research studies focused on the use and the development of the cooling technique based synthetic jet. The distance between the orifice and the heated surface is one of the major factors that influences the heat transfer by synthetic jet and have been studied and proven by many researchers experimentally [1–3] and numerically using

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Nomenclature

A	Diaphragm amplitude (m)
d_c	Cavity width (m)
d_0	Orifice width (m)
D	Channel width (m)
F	Diaphragm frequency (Hz)
h_c	Convective heat transfer coefficient ($\text{W/m}^2 \text{ K}$)
H	Channel height (m)
L	Heater width (m)
K	Thermal conductivity (W/m K)
L_s	Stroke length
Nu	Nusselt number, hH/k
Re_c	Jet Reynolds number, $2\pi f x_0 H / \nu$
S	Stokes number
t	Time (s)
T	Time period, $1/f$ (s)
T_w	Wall temperature (K)
U_c	Characteristic jet velocity (m/s)
u_0	Velocity through orifice (m/s)
V_i	Micro-channel inlet velocity (m/s)
φ	Density (kg/m^3)
μ	Dynamic viscosity (kg/ms)
ω	Angular velocity (rad/s)
ν	Kinematic viscosity (m^2/s)

ANSYS Fluent code where the unsteady Navier–Stokes equations and the convection–diffusion equation were solved using a fully unsteady [4], experimental study based on the characteristic of synthetic jet impinging upon a vertical heater, and results showed that the heat transfer have the higher performance with a jet-to-surface spacing between 5–10 [5].

Another factor is the shape of the opening orifice. Tan et al. [6] investigated experimentally the efficiency of different opening hole forms including square and rectangular shapes. Results indicated that the synthetic jet with rectangular orifice is more efficient. Furthermore, the comparison between the rectangular and the elliptic shapes are experimentally tested and showed that the rectangular orifice develop stronger streamwise vortices [7]. Another experimental investigation compared a circular and chevron opening hole forms for an impinging synthetic jets and results showed that for the chevron opening hole shape the heat transfer is greater up to 20% compared to the circular opening hole shape [8]. Different alternative techniques have been investigated experimentally to increase the heat transfer such as the introduction of multi-orifice into the cavity [9] or using multi-cavities using Particle Image Velocimetry (PIV) [10–12]. Paolillo et al. [13] studied experimentally the influence of impinging quadruple synthetic jets for different configuration monopole-like, dipole-like, quadrupole-like and 90-degrees-circularly-shifted-jets for Reynolds = 4000 and Strouhal numbers = 0.2, results indicated that the 90-degrees-circularly-shifted-jets configuration give the highest heat transfer rates.

On the other side, there have been several studies that focused on the effect of the cavity shape on the synthetic jet performance and the improvement in the heat transfer characteristics. Results showed that the basic cylindrical shape maximize momentum flux and provides better performance from an experimental work [14], Lv Yuan-wei et al. [15] carried out numerical investigation based ANSYS Fluent code on the effect of the cavity and orifice dimensions and the excitation frequency of the actuator on the synthetic jet fluidic characteristics. Obtained results showed that the orifice diameter and thickness, as well as the cavity depth and diameter, have major influence on the synthetic jet fluidic characteristics. Another experimental work tested the modification on the cavity using

silicone elastomer membranes together with either soft composite or rigid neodymium magnets, and they found good results for low-frequency range [16]. The change in function of oscillation mode in the cavity of synthetic jet proved that the sinusoidal function is better than the triangular function [17]. Numerical investigation carried out by Zhang et al. [18] in which three wave forms for synthetic jet including sinusoidal, rectangular and triangular forms for Strouhal number (St) between 0.012–2.4 have been investigated. Reported results showed that at $St < 0.06$ the rectangular jet is better than the sinusoidal and triangular jets, and for St number ranging between 0.24 to 0.48, triangular jet is better. Further, numerical investigation based Computational Fluid Dynamics (CFD) and the open-source OpenFOAM code has been carried out by King and Jagannatha [19]. Results indicated that the non-sinusoidal function gave an improvement in heat transfer than sinusoidal function.

The present study is mainly based on the modification of the heating surface form and the signal of the diaphragm for a synthetic jet. These modifications are novel and have been applied for the first time with very good thermal enhancement efficiency for microchannels with synthetic jets applications. The influence of bi-periodic non-sinusoidal pulsating motion and two cavities with a slope of 3° , 60 percent obstruction orifices and $10\ \mu\text{m}$ of undulation heated wall in microchannel are investigated. Present numerical results based two cavities are compared with the results of the basic case with one cavity to understand the phenomenon of improved heat transfer rate.

2. Mathematical model and validation

The governing standard incompressible Navier–Stokes equations were solved:

$$(\nabla \cdot \vec{v}) = 0 \quad (1)$$

$$\frac{D\vec{v}}{Dt} = -\frac{1}{\rho} \nabla P + \nu \Delta \vec{v} \quad (2)$$

$$\rho c \frac{DT}{Dt} = \phi_v + \nabla \cdot (k \Delta T) \quad (3)$$

The spatially averaged instantaneous Nusselt number is computed at every time step as:

$$Nu(t) = \frac{q''}{\overline{T_w}(t) - T_b(t)} \frac{H}{k} \quad (4)$$

where T_b is the bulk flow temperature, computed with a mass-weighted average approach and $\overline{T_w}$ is the spatially averaged wall temperature in the location of the heat source. H is the width of the cross-flow channel.

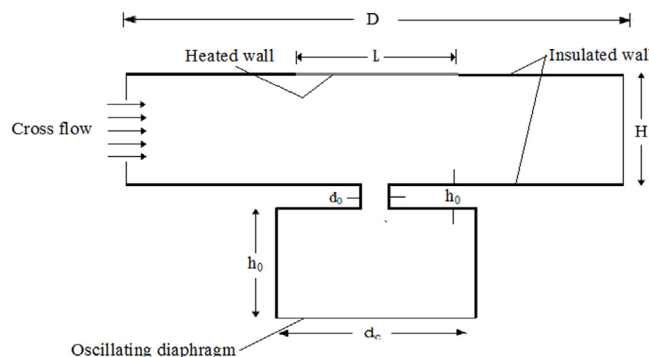


Fig. 1. Basic case with synthetic jet cross a micro-channel.

In this study two geometrical configurations are considered, the first the basic configuration without any modification as shown in Fig. 1 and presented in detail by Chandratilleke et al. [20] all geometrical parameters and applied boundary conditions are presented in Table 1. For this configuration the piston motion is sinusoidal with the function of displacement defined as [21]:

$$y = A \sin(\omega t - \varphi) \quad (5)$$

The second configuration was similar to the basic case but with two synthetic jets inclined with 60 per cent obstruction and slope of 3° and $10\ \mu\text{m}$ of undulation heated wall “modified case”. The piston motion is bi-periodic

Table 1. General parameters for the present numerical simulation.

Parameter	Range
Micro-channel inlet velocity, V_i (m/s)	1
Diaphragm frequency, f (kHz)	10
Diaphragm amplitude, A (μm)	25-50-75
Jet Reynolds number, Re_c	$\approx 5-20$
Distance from orifice to heated wall, H/d_0	10
T_∞ (K)	300
q'' (W/m^2)	15 000
d_0 (μm)	50
h_0 (μm)	50
H (μm)	500
D (μm)	2250
L (μm)	750
d_c (μm)	750
h_c (μm)	500
ω	$2\pi f$
φ	$\pi/2$

non-sinusoidal with the function of displacement defined as:

$$y = A \cos(\omega_1 t - \varphi) + \frac{1}{2} A \cos(\omega_2 t) \quad (6)$$

with:

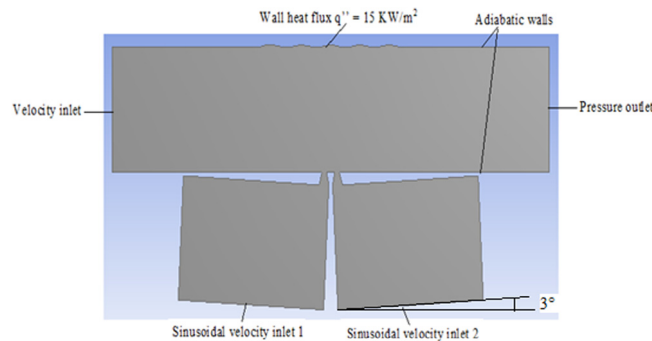
$$\omega_1 = 2 * \pi * f_0 \quad (7)$$

$$\omega_2 = \left(\frac{2}{3}\right) * \pi * f_0 \quad (8)$$

The flowing fluid (air) is supposed to be incompressible, and its properties at a temperature $T = 300$ K are presented in Table 2. Fig. 2 shows the different applied boundary condition at the borders of the considered geometry.

Table 2. Thermal properties of the working fluid.

Property	Hypothesis	Value
Density	Ideal gas	1.1614
Specific heat	Constant	1.007
Viscosity	Constant	$1.7894e-5$
Thermal conductivity	Constant	0.0263

**Fig. 2.** Boundary condition.

A structured quadratic mesh as shown in Fig. 3 is used with a variable mesh refinement in each critical area. Three different grids with total number of nodes of 40 830, 87 942 and 159 138 have been tested and results of the variation of Nusselt number versus the normalized distance are presented in Fig. 4. The maximum difference in

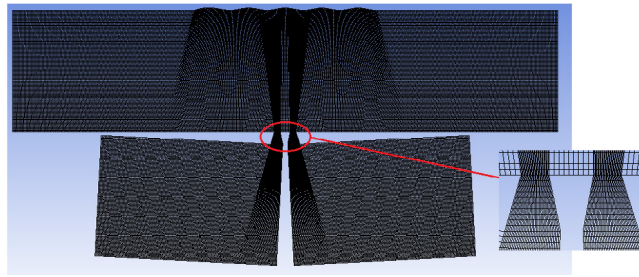


Fig. 3. Structured quadratic mesh considered.

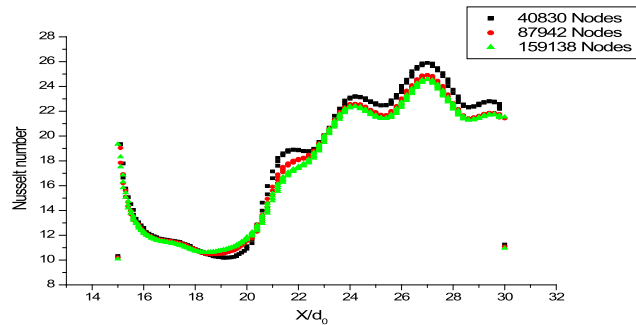


Fig. 4. Grid test.

Nusselt number between the two meshes of 87 942 and 15 138 is less than 2%. Therefore, the mesh with 87 942 total number of node has been selected for all present simulation which provides a good accuracy with less computational time.

3. Results and discussion

In the present section, numerical results are compared first with published data to validate the numerical model. Then, a comparison between the basic and the modified configurations is presented, and results are discussed in detail.

Numerical model is validated by comparing present numerical results with the results of Chandratilleke et al. [20]. The comparison for the variation of the local Nusselt number versus the normalized distance at velocity inlet of $V_i = 1$ m/s is shown in Fig. 5 and is in reasonably good agreement with Chandratilleke et al. [20] results.

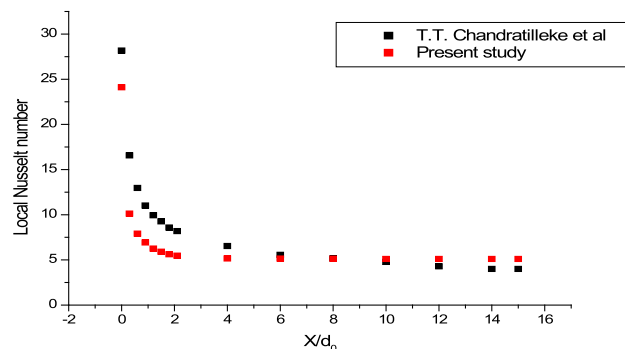


Fig. 5. Comparison of the present results with those of Chandratilleke et al. [20] for the variation of the Local Nusselt number versus normalized distance for $V_i = 1$ m/s.

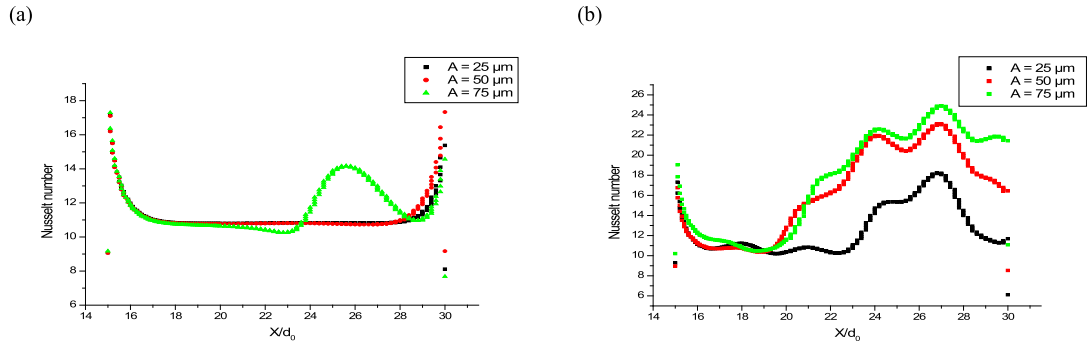


Fig. 6. Nusselt number on the heated surface for different amplitudes, (a) Basic case with periodic signal, (b) Modified case with bi-periodic signal.

Results for the basic case with periodic signal, and for the modified case with bi-periodic signal are presented in Fig. 6(a) and (b), respectively, for different oscillating amplitudes $A = 25, 50$ and $75 \mu\text{m}$ and for an inlet velocity $V_i = 1 \text{ m/s}$. Results show that Nusselt number increases with increasing the oscillation, and the maximum values could be obtained for the amplitude $A = 75 \mu\text{m}$. In addition, it is shown that Nusselt values for the modified case with bi-periodic signal are much higher than the basic case. These results confirm that the proposed modification influence the heat transfer rate and it is clearly shows in Fig. 7 with an average difference of about 51%.

To understand the evolution of the Nusselt number, Fig. 8 shows the variation of the average Nusselt number for the three amplitudes $A = 25, 50$ and $75 \mu\text{m}$, for $V = 1 \text{ m/s}$. It is shown that the case with modification gives higher values of Nusselt number.

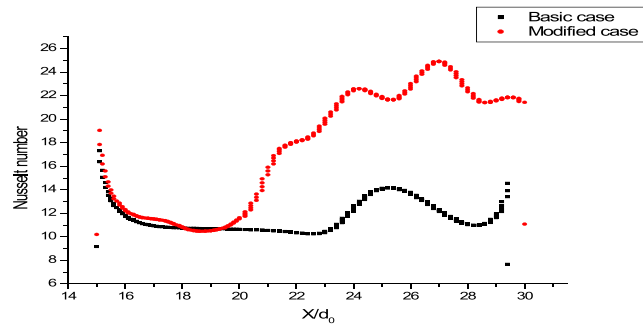


Fig. 7. Comparison of Nusselt number for the heated surface between basic case with periodic signal and modified case with bi-periodic signal at $V_i = 1 \text{ m/s}$; $A = 75 \mu\text{m}$.

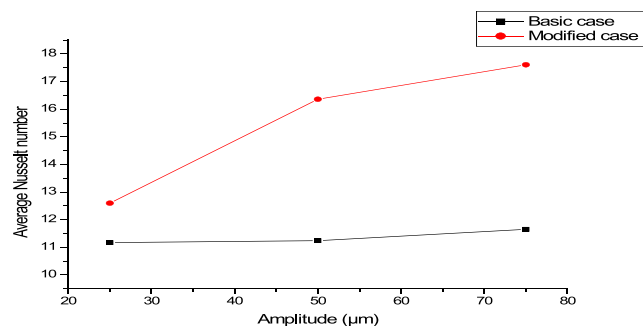


Fig. 8. Average Nusselt number for basic case with periodic signal and modified case with bi-periodic signal.

The global temperature field in the channel at the end of an oscillating cycle, is presented in Fig. 9(a) for the basic case with a periodic signal, and in Fig. 9(b) for the modified case with the bi-periodic signal, at an oscillating

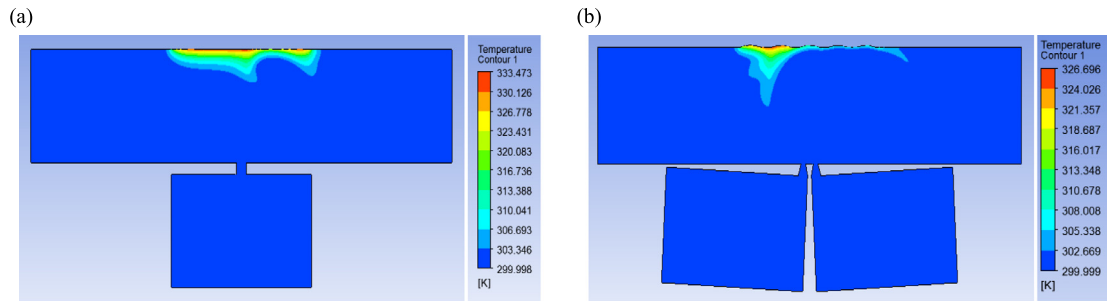


Fig. 9. Contour plots of the static temperature after one cycle for $V_i = 1$ m/s and $A = 75$ μm , (a) Basic case with periodic signal, (b) Modified case with bi-periodic signal.

amplitude of $A = 75$ μm and an inlet velocity $V_i = 1$ m/s. The contour clearly shows that the temperature decreases in the vicinity of the heated surface, for amplitude $A = 75$ μm the modified case with bi-periodic signal is much colder than the basic case because the effect of the synthetic jet is high with the presence of two jet as well as the Bi-periodic signal extremely that a periodic signal.

Fig. 10 shows the contours of the magnitude velocity for the basic case with a periodic signal, and in the modified case with the bi-periodic signal. Results correspond to a transverse flow velocity $V_i = 1$ m/s and a jet amplitude $A = 75$ μm . Results show clearly that the magnitude velocity for the modified case is higher because of the acting force of bi-periodic signal and the generation of the jet vortex due to the small inclined flow coming from the modified orifice inclination. These results are presented in more detail in **Fig. 11** for one cycle.

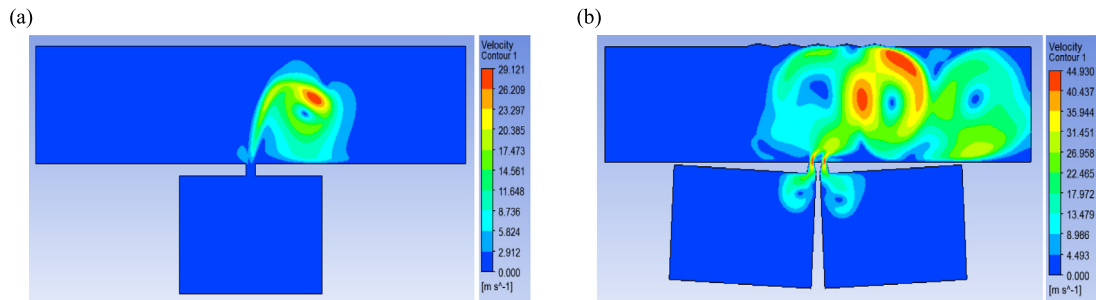


Fig. 10. Contour plots of the magnitude velocity after one cycle for $V_i = 1$ m/s and $A = 75$ μm , (a) Basic case with periodic signal, (b) Modified case with bi-periodic signal.

4. Conclusion

In the present study, the form of heating surface and the signal of the diaphragm for a synthetic jet has been modified to improve the synthetic jet performance. These modifications are novel and have been applied for the first time with very good thermal enhancement efficiency for microchannels with synthetic jets applications. The comparison between the basic and the modified configurations have been numerically investigated using ANSYS fluent code.

Temperature contours showed that the form of the undulant heated wall helps the emigration of the temperature and has a large influence on the heat transfer rate. The presence of obstruction orifices and undulant heated wall with a slope of 3° increased Nusselt number by about 51% compared with the basic case due to the increased vortices in the flow domain.

The 2D simulation gives a general view to understand the phenomenon of cooling but it is necessary to use the 3D simulation for a global study.

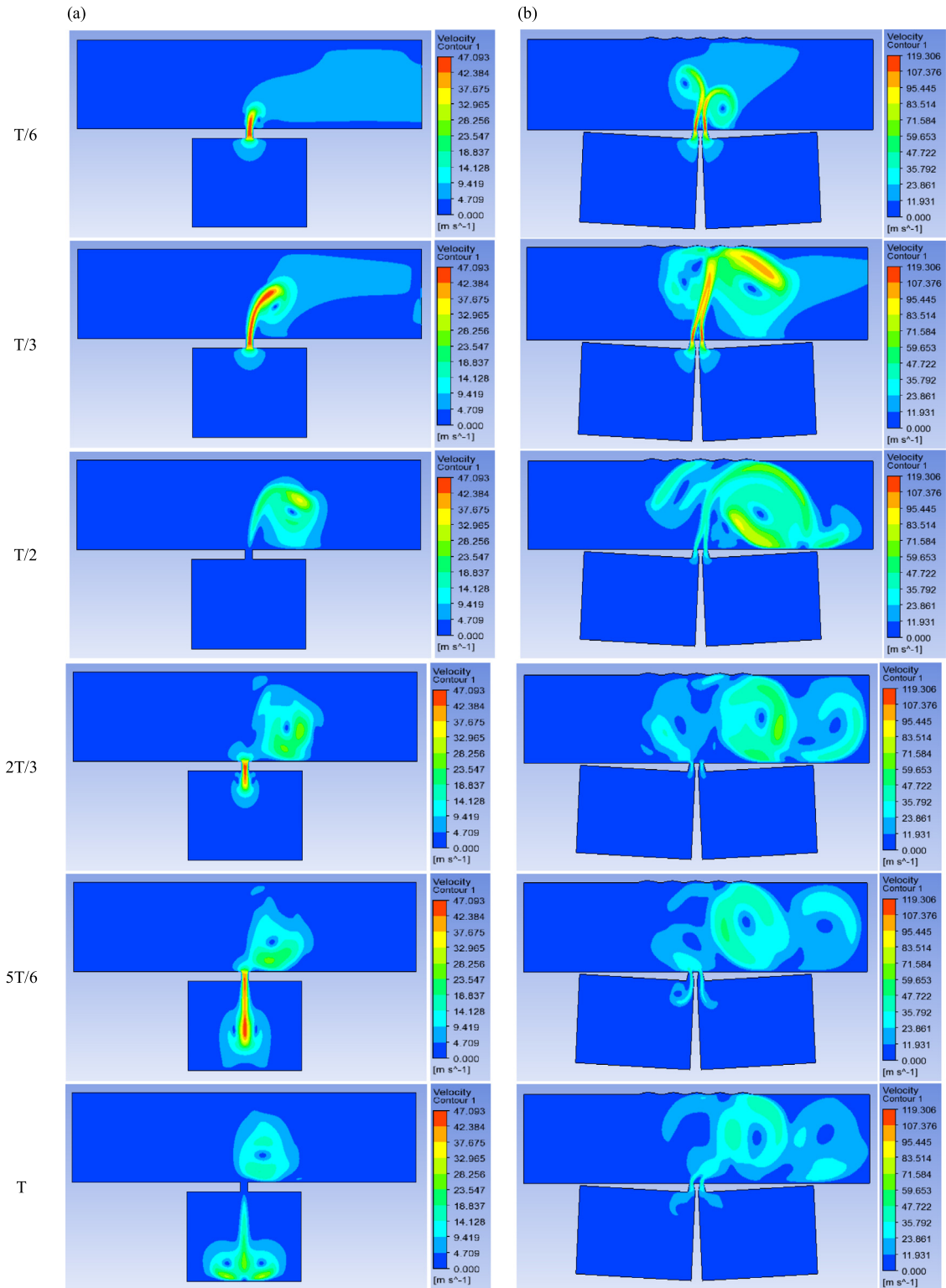


Fig. 11. Contour plots of the magnitude velocity for one cycle for $V_i = 1 \text{ m/s}$ and $A = 75 \text{ }\mu\text{m}$, (a) Basic case with periodic signal, (b) Modified case with bi-periodic signal.

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