Numerical Study of the Effect of Different Variables on the Cooling of a Flat Plate Using a Synthetic Jet

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1. Introduction

A synthetic jet is caused by the periodic motion of a diaphragm within a cavity. There is one or more orifices or outlets in this cavity. The main advantage of this type of jet compared to a continuous jet is that the synthetic jet is composed of transverse flow, and therefore, it does not need a continuous source of fluid, unlike the continuous jet. In recent years, synthetic jets have received a great deal of attention so that they have been used in a wide range of applications such as controlling separation and turbulence, besides, the cooling of electronic equipment and propulsion. In the present study, the jet is placed perpendicular to the flat plane with constant heat flux, thereafter, the effect of some geometric parameters were evaluated numerically such as the ratio of the distance between the jet and the impinging plate to the nozzle width, the ratio of the impinging plate length to the jet nozzle width, the ratio of cavity width of the synthetic jet to the nozzle width, the ratio of the cavity height to the nozzle width, the angle of the impinging plate, besides, the diaphragm specifications including amplitude and frequency of the jet diaphragm in heat transfer using OpenFOAM open-source software. The results show that the frequency and the length of the impinging plate are the most effective parameters, respectively, in terms of the diaphragm and geometry.

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According to studies on synthetic jet actuators, the excitation frequency and orifice dimensions are among the most important factors influencing the formation of these oscillating eddy structures.

In numerical research, Wang et al. [4] studied the velocity and temperature profiles and heat transfer coefficient of synthetic jets for thermal management using the LES turbulence model. In this research, momentum and heat transfer equations have been solved using Fluent and Gambit software, besides, working fluid is air. The boundary condition of the diaphragm (velocity boundary condition for a synthetic jet) is a sinusoidal wave that varies with time and place. They also considered the thermodynamic properties of air to be temperature-dependent, which increased the accuracy of the calculations.

Eventually, they compared the numerical simulation results with the experimental results. In this study, the boundary condition of the velocity function, which varies with time and place, was used. They proved that the use of piston boundary conditions for the velocity boundary condition leads to unacceptable results. The results of numerical simulations were in agreement with the results of experimental tests with acceptable accuracy.

In another numerical study, Jain et al. [5] investigated the effect of cavity and orifice dimensions and the velocity boundary condition on the heat transfer of a synthetic jet. In this study, they concluded that the moving mesh boundary condition provides the best results in the simulation of synthetic jets (the simulation results with the piston boundary condition are compared with velocity function and the moving mesh boundary condition). The simulation results showed that synthetic jets are more affected by changes in the orifice’s geometric parameters. The most important parameters affecting the performance of synthetic jets are the diameter of the cavity and the orifice, also, the height of the orifice.

In this research, diagrams for the optimal design of cavities and orifices are presented.

In this study, the results of mesh independence indicate that the highest sensitivity of the results is on the number of mesh points around the orifice, therefore, the maximum concentration of the points should be within this area. A resolution of 0.1 mm around the orifice is sufficient for a 3 mm diameter duct. Inside the cavity, the mesh is steady while the concentration of the points is exponential in the outer zone.

Zhang et al. [6] experimentally compared the heat transfer of synthetic and conventional jets. In this study, researchers stated that the excitation frequency has significant effects on the velocity field and vorticity. There are two resonance frequencies at which the average velocity and vorticity reach a maximum value (especially in the resonant frequency). Forced convection heat transfer in a synthetic jet with a piston diaphragm has been studied, afterward, the effects of orifice dimensions have also been investigated. In this research, three actuators with rectangular, circular, and square sections were used, and then, the results were compared.

Persoons et al. [7] comparatively analyzed the performance of synthetic jets and stable impinging jets at the inertia point. They proposed an equation for the Nusselt number at the point of inertia including the effect of all suitable dimensional parameters. In this research, a general equation has been obtained for the Nusselt number at the inertia point with respect to all important parameters such as the Reynolds number, the distance between the jet and the plate, and the course length.

Akdag et al. [8] experimentally investigated the effect of a synthetic jet on transverse heat transfer on a flat plate. The experiment consists of a copper plate heater with a constant heat flux placed in the wind tunnel, and a synthetic jet actuator injecting into flow over the plate inlet. The synthetic jet was created by a cylinder-piston mechanism. In this study, the Reynolds number in the mainstream, the frequency, and the amplitude of the actuator were changed while the geometry and Prandtl number were constant for all cases, finally, the effect of these parameters was analyzed on the heat transfer. The experiments were performed with 6 and 4 different frequencies and amplitudes, respectively. The results were presented as dimensionless parameters, and it was observed that the average Nusselt number of the cycle increases with increasing the dimensional amplitude.

One of the new issues in this field, which has strongly influenced the cooling of flat plates, is the material and the use of new alloys. Recently, the use of glass materials and composites as a material with high strength and good conductivity has been considered. Mokhtari et al. Have conducted various studies on the properties of glass composites. The thermal properties of each glass composition were determined using differential thermal analysis (DTA) and hot stage microscopy (HSM), and the results are presented in Figures 4 and 5. DTA profiles for each glass shows that the glass transition temperature (Tg) increases from 351 °C (BG) to 559 °C (SC-1) and to 627 °C (SC-2) when TiO₂ was substituted with CaO and Na₂O concentration respectively [9-12].

Due to some advantageous features of composites such as high strength-to-weight ratio, being resistant to fatigue and so on, they are widely used in different engineering fields. The thermal buckling and post-buckling of geometrically perfect and imperfect hybrid laminated composite plates reinforced with shape memory alloys (SMA) subjected to a uniform thermal loading are investigated by a semi-analytical finite strip method by Rostamijavanani et al. in 2021 [13, 14].
The study process on synthetic jets and their heat transfer began two decades ago in scientific centers. However, many factors influencing this cooling method remain unknown. Due to the structure of these types of jets, there is a vital need for further investigation. Besides, inadequate research done in this field is mainly experimental so there is a need for numerical analysis due to its low expenses and availability, also, easy changes in the jet than experimental methods to determine the effect of different parameters. By studying different references, it was concluded that this article was the first research investigating the effect of different parameters of synthetic jets on heat transfer.

2. Material and methods

2.1. Modeling the flow and heat transfer of a synthetic jet

Modeling of synthetic jets is based on conservation equations, like other fluid dynamics problems. Due to the oscillating nature of these jets, the transient solution of the equations should be used. Examining the previous literature, most researchers have considered the synthetic jet flow to be turbulent, therefore, they have used various turbulence models to simulate it. As a result, turbulence in these jets (regardless of the Reynolds number of the jet) is usually considered.

Compression is also a significant issue in these jets. Some researchers consider the flow to be compressible due to rapid changes in density/pressure because of the diaphragm movement, however, most of them consider the flow to be incompressible according to Ma <0.3. In the present study, the density changes with temperature due to the heat transfer resulting in a compressible flow. Also, due to the assumption of flow compressibility (density change), natural convection heat transfer is considered. The solution field simulation is two-dimensional (2D).

In this study, according to the above assumptions, compressible and Unsteady Reynolds-averaged Navier-Stokes equations (URANS) were used [15]. Compressible RANS equations are called Favre-averaged Navier-Stokes equations (FANS). In the following, first, the governing conservation equations are stated, thereafter, Favre averaging method and its equations are presented.

2.1.1. Governing conservation equations

- Mass conservation equation (continuity)

\[
\frac{D\rho}{Dt} + \rho \nabla \cdot \mathbf{V} = 0 \quad \text{or} \quad \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{V}) = 0 \tag{1}
\]

- Momentum conservation equation (Navier--Stokes’s equations)

\[
\rho \frac{D\mathbf{V}}{Dt} = \rho \mathbf{g} - \nabla p + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \delta_{ij} \lambda \nabla \cdot \mathbf{V} \right] \tag{2}
\]

- Energy conservation equation

\[
\rho \frac{Dh}{Dt} \equiv \frac{D\rho}{Dt} + \nabla \cdot (\rho \mathbf{V} T) + \tau'_{ij} \frac{\partial u_i}{\partial x_j} \tag{3}
\]

\[
h = e + \frac{p}{\rho} \tag{4}
\]

In Equation (2-4), \(h\) and \(e\) are specific enthalpy and specific internal energy, respectively. Also, \(\tau'_{ij}\) (viscous stress) is as follows:

\[
\tau'_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \delta_{ij} \lambda \nabla \cdot \mathbf{V} \tag{5}
\]

2.1.2. Favre averaging

In a compressible flow, density and temperature gradients must also be considered in addition to velocity and pressure gradients. If the standard time-averaging method (Reynolds averaging method) is used, the averaged conservation equations will include additional terms for which there is no similarity in the laminar equations.

The time-averaged equations can be interestingly simplified using the weighted density averaging method proposed by Favre (1965). The averaged mass velocity, \(\bar{u}_i\), is defined as below:
\[ \tilde{u}_i = \frac{1}{\bar{\rho}} \lim_{T \to \infty} \frac{1}{T} \int_0^{T+T} \rho(x, \tau)u_i(x, \tau) d\tau \]  \hspace{1cm} (6)

Where \( \bar{\rho} \) is the normal Reynolds-averaged (time) density. In the Reynolds averaging method, the density is constant, and according to the equation (2-6), the term of density is not considered (it is used for incompressible flow). For Favre averaging, it is common to write the instantaneous velocity by two parts of mass averaged, \( \bar{u}_i \), and the oscillating, \( u''_i \):

\[ u_i = \bar{u}_i + u''_i \]  \hspace{1cm} (7)

It should be noted that:

\[ \frac{\rho u''_i}{\bar{\rho}} \neq 0 \]  \hspace{1cm} (8)

In equation (8), \( \rho' \) and \( u''_i \) are, respectively, the oscillating velocity and density in the Reynolds time averaging method. Finally, it should be noted that since Favre averaging eliminates density gradients from the averaged equations, the effect of density gradient remains on turbulence. As a result, Favre averaging is a mathematical simplification, not a physical one.

### 2.1.3. Favre-averaged conservation equations

For the Favre-averaged conservation equations method, different properties of the flow are defined as follows [16-18]:

\[
\begin{align*}
\rho &= \bar{\rho} + \rho' \\
p &= P + p' \\
h &= \bar{h} + h'' \\
e &= \bar{e} + e'' \\
T &= \bar{T} + T'' \\
q_j &= q_{lj} + q'_j
\end{align*}
\]  \hspace{1cm} (9)

Where \( q_j \) is the heat flux vector usually derived from the Fourier law. By placing the above equations in the conservation equations, besides, applying mass averaging, the Favre-averaged (mass) mean conservation equations are obtained.

- Favre-averaged mass conservation equations

\[
\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial}{\partial x_i} (\bar{\rho} \bar{u}_i) = 0
\]  \hspace{1cm} (10)

- Favre-averaged momentum conservation equations

\[
\frac{\partial}{\partial t} (\bar{\rho} \bar{u}_i) + \frac{\partial}{\partial x_j} (\bar{\rho} \bar{u}_j \bar{u}_i) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} [\bar{\tau}_{ji} - \rho u''_j u''_i]
\]  \hspace{1cm} (11)

- Favre-averaged energy conservation equations

\[
\frac{\partial}{\partial t} \left[ \bar{\rho} \left( \frac{\bar{e} + \bar{u}_i \bar{u}_i}{2} \right) + \frac{\partial u''_j u''_i}{2} \right] + \frac{\partial}{\partial x_j} \left[ \bar{\rho} u''_i \left( \frac{\bar{u}_i + \bar{u}_j}{2} \right) - \frac{\partial u''_j u''_i}{2} \right] = \frac{\partial}{\partial x_j} \left[ q_{lj} - \rho u''_j u''_i + \bar{u}_j \left( \bar{e}_{ij} - \rho u''_i u''_i \right) \right]
\]  \hspace{1cm} (12)

Here, there are several unfamiliar terms in the Favre-averaged conservation equations, for which an approximation is required to solve the system of equations, moreover, depending on the turbulence model, additional closeness approximation will be needed. The following is a summary of some of the most common closeness approximations.

In equations (11) and (12), \( \bar{\tau}_{ji} \) is the viscous stress tensor which is defined as follows:

\[
\bar{\tau}_{ji} = 2\mu \bar{S}_{ij} - \frac{2}{3} \mu \frac{\partial \bar{u}_k}{\partial x_k} \delta_{ij} \quad , \quad \bar{S}_{ij} = \frac{1}{2} \left( \frac{\partial \bar{u}_j}{\partial x_i} + \frac{\partial \bar{u}_i}{\partial x_j} \right)
\]  \hspace{1cm} (13)
The term \(-\overline{\rho u_i'u_i''}\) is a Favre-averaged Reynolds stress tensor, and almost all researchers use the Boussinesq approximation (buoyancy) with an appropriate generalization for compressible flow for the zero, one, and two-equation models. In particular, by determining the eddy viscosity by \(\mu_t\), the following format is generally assumed:

\[
\overline{\rho\tau_{ij}} = -\overline{\rho u_i''u_i''} = 2\mu_t \left( \left[ \frac{1}{3} \frac{\partial \overline{u_k}}{\partial x_j} \right] - \frac{2}{3} \delta_{ij} \right)
\]

In Equations (13) and (14), \(\delta_{ij}\) is a tensor of Favre mean strain rate.

The term \(\overline{\rho u_i''u_i''}\) in Equation (12), the kinetic energy is the volume unit of turbulent oscillations, so it is defined as follows:

\[
\overline{\rho k} = \frac{1}{2} \overline{\rho u_i''u_i''}
\]

The term \(\overline{\rho u_i''h''}\) in Equation (12) is the turbulent heat flux vector. The most common closeness approximation for this term obtains through the similarity between momentum and heat transfer. Therefore, this term is assumed to be proportional to the average temperature gradient. As a result:

\[
q_{ij} = \overline{\rho u_i''h''} = \frac{\mu_c p}{\text{Pr}_t} \frac{\partial \overline{T}}{\partial x_j} = -\frac{\mu_t}{\text{Pr}_t} \frac{\partial \overline{k}}{\partial x_j}
\]

Where \(\text{Pr}_t\) is the turbulent Prandtl number, which is often assumed to be constant. The most common values of \(\text{Pr}_t\) are 0.89 or 0.90 for the boundary layer and 0.5 for the free shear layers.

In Equation (12), the terms \(\overline{t_{ij}u_i''}\) and \(\overline{\rho u_i''u_i''u_i''}\) are, respectively, related to molecular diffusion and turbulent transfer of turbulence kinetic energy. The most common method for one and two-equation and stress transfer models is to generalize low-velocity closeness approximations for molecular diffusion and turbulent transfer terms. The common approximation, in this case, is as follows:

\[
\overline{t_{ij}u_i''} - \overline{\rho u_i''u_i''u_i''} = \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j}
\]

### 2.2. Turbulence model

Given the turbulent flow of synthetic jets, comparisons between experimental results and different turbulence models for a cooling jet with a diaphragm are presented in a Table 1.

According to the results of Table 1, the SST \(k-\omega\) model has the lowest error among other models. Also, the results of NASA Langley Research Center on CFD validation of synthetic jets and turbulent separation control (CFDVAL2004) show that the selection of the appropriate turbulence model significantly affects the simulated data, accordingly, the SST \(k-\omega\) model shows the best fit to the unsteady Reynolds-averaged Navier-Stokes model (URANS). According to the above studies, the SST \(k-\omega\) turbulence model is used in this study. In the next section, the equations of this model are presented.

| An example of a column heading | Q [L/min] | Error [%] |
|-------------------------------|-----------|-----------|
| Experimental results          | 19.0      | -         |
| SST \(k - \omega\)            | 21.6      | 13.6      |
| Spalart-Allmaras              | 22.3      | 17.1      |
| Standard \(k - \varepsilon\)   | 23.0      | 20.8      |
| RNG \(k - \varepsilon\)       | 23.2      | 21.9      |
| Realizable \(k - \varepsilon\)  | 22.3      | 17.4      |
| Standard \(k - \omega\)       | 22.9      | 20.6      |
2.3. Boundary conditions

The boundary conditions for the impinging plate are constant heat rate with a value of 1000 w/m², besides, the initial temperature of 300 K. The field output is also in the form of pressure output. The boundary condition of the wall is also non-slip. However, the most challenging boundary condition is the velocity input into the synthetic jet. For different types of synthetic jets, researchers have chosen different approaches, which are referred to as three:

- A wall-normal velocity boundary condition with an assumed profile at diaphragm or nozzle output
- Movable piston boundary condition
- Vibratory diaphragm boundary condition

In this research, the input velocity boundary condition with a specific profile in the diaphragm is used. The advantage of this method is the high speed of problem-solving along with its relatively high accuracy, which is the most appropriate method of problem-solving in this study due to the high number of assumed parameters and available computer facilities.

The output is considered at constant atmospheric pressure, and as mentioned earlier, the input velocity boundary condition is used for the diaphragm most important part of the simulation. To determine the appropriate velocity profile to simulate diaphragm motion, a model called the Xia-Luo Model (X-L) is applied to the software with a code. The X-L model is as follows:

\[
\begin{align*}
    u_x &= 2\pi f A_m \left( 1 - \left( \frac{r}{r_c} \right)^2 \right) \sin(2\pi f t + \varphi_0) \\
    u_y &= v = 0
\end{align*}
\] (18)

Where \( r \) is the distance from the center of the diaphragm, \( r_c = 23 \text{mm} \) and \( \varphi_0 = 0 \) are the radius and the initial phase of the diaphragm, respectively. Furthermore, air is used as the operating fluid in this simulation. The dimensionless time and phase of the synthetic jet diaphragm are defined as follows:

\[
\begin{align*}
    t^* &= \frac{t - nT}{T} , \quad n = \left[ \frac{t}{T} \right] , \quad T = \frac{1}{f} \\
    \varphi^° &= t^* \times 360°
\end{align*}
\] (19)

2.4. Nusselt number

The most important dimensionless number for the study of heat transfer is the Nusselt number. The longitudinal parameter of the Nusselt number in the impinging jets is the diameter of the orifice or the width of the output nozzle. The local Nusselt number on the impinging plate is defined as follows:

\[
Nu = \frac{h d_o}{k}
\] (21)

Where \( k \) is the heat transfer coefficient of the fluid at the average temperature of the impinging plate and \( h \) is the local convection heat transfer coefficient defined as follows:

\[
h = \frac{q''}{T_s(x,t) - T_j}
\] (22)

\( q'' \) is the steady heat flux applied to the impinging plate. Radiation heat transfer is neglected in this simulation while natural convection heat transfer is considered. \( T_s(x,t) \) and \( T_j \) are the local temperature of the impinging plate (hot wall) at any given moment and the jet temperature (or ambient temperature), respectively. To calculate the local Nusselt number, \( T_s(x,t) \) is estimated at the moment \( t = 2s \) when the solution reaches a quasi-steady state.

The oscillating property of the synthetic jet causes the temperature of the impinging plane to fluctuate. Therefore, to determine the appropriate average Nusselt number of the impinging plate, it is better to consider the average temperature of the impinging plate during a certain time interval (when the plate temperature becomes a quasi-steady-state) rather than applying the average (local) temperature of the plate at the last moment so the average Nusselt number of the problem is obtained as follows:

\[
Nu_{avg} = \frac{\bar{h} d_o}{k}
\] (23)

\[
\bar{h} = \frac{q''}{\bar{T}_s - T_j}
\] (24)
\[ T_{avg} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} T_s(t) \, dt \]  
(25)

\[ T_s(t) = \frac{1}{L_0} \int_{0}^{L} T_c(x, t) \, dx \]  
(26)

In equation (24), \( T_{avg} \) and \( T_s(t) \) are, respectively, the average local and temporal temperature of the impinging plate between \( t_1 = 3s \) to \( t_2 = 4s \) moments and the average local temperature of the impinging plate at each moment.

3. Results

3.1. The effect of geometric parameters

In this section, the effect of geometric parameters on the heat transfer of synthetic jets will be investigated. Five geometric variables are examined for this research: impinging plate angle, impinging plate distance, impinging plate length, cavity diameter, and cavity height. For each parameter, diagrams of impinging plate temperature, Nusselt number, average temperature, maximum temperature, and minimum temperature will be presented.

3.1.1. The ratio of the distance between jet and the impinging plate to the width of the nozzle \((H/d_o)\)

In this section, the effect of the ratio of the distance between the jet and the impinging plate to the nozzle width \((H/d_o)\) is investigated on the Nusselt number. Moreover, a diagram of the temperature of the impinging plate for different values of \((H/d_o)\) is also provided.

3.1.2. The effect of the ratio of the width of the cavity to the width of the orifice \(D/d_o\)

In this section, the effect of the width of the synthetic jet cavity on the cooling of the impinging plate is investigated, and then the local temperature and Nusselt number diagrams are presented for different modes.

The diameter of the synthetic jet cavity is a very important variable in cooling so that the diameter of the synthetic jet cavity is directly related to the velocity of the synthetic jet. By reducing the diameter of the cavity, an improper flow hit the plate leading to a sharp rise in temperature in the middle of the impinging plate. This trend is highly evident for the diameter ratio of 10 and 15. However, by increasing this ratio, the plate temperature decreases until a uniform profile of temperature is created over the impinging plate. In general, increasing the diameter of the cavity has a significant effect on cooling while causing problems such as increased power required for vibration, higher construction costs, and problems related to the vast space for installation.

Figure 2 shows the local Nusselt number for the length of the impinging plate. It is clear that the Nusselt number has become greater with an increasing cavity diameter ratio. In general, the ratio of cavity width to groove width should be higher than 20 in synthetic jets.
According to Figure 3, the average Nusselt number in terms of the ratio of the width of the cavity to the width of the orifice has an ascending trend, therefore, the heat transfer rate can be increased by increasing this ratio.

3.1.3. The effect of the ratio of cavity height to orifice width ($\frac{h_c}{d_o}$)

In this section, the effect of the height of the synthetic jet cavity on the cooling quality is investigated and the results are presented through the charts. According to Figure 3, it can be seen that the depth of the cavity does not have a significant effect on the cooling of the plate. The optimal state for this variable is the ratio of 2 leading to the best cooling efficiency.

Figure 2. Local Nusselt number of the impinging plate at the moment ($t^* = 0$) for different values of $D/d_o$

Figure 3. The average Nusselt number in terms of the ratio of the width of the cavity to the width of the orifice

Figure 4. Impinging plate temperature at the moment ($t^* = 0$) for different values of $h_c/d_o$
3.2. The effect of diaphragm specification changes

In this section, the effect of two specifications of the synthetic jet diaphragm, introducing in the second chapter, are examined on heat transfer. These specifications include the frequency \( f^* \) of the synthetic jet diaphragm.

3.2.1. The effect of jet diaphragm frequency changes \( (f^*) \)

Here, the effect of the frequency of the synthetic jet diaphragm \( (f^*) \) is evaluated on the Nusselt number and temperature of the impinging plate.

Frequency is one of the most important variables in synthetic jet design. The higher the frequency is, the cooling efficiency of the cooling plate improves. This trend can be clearly seen in Figure 6. Increasing the frequency creates a uniform temperature over the impinging plate, which is desirable. However, increasing the frequency increases the power consumption but reduces the working life of the diaphragm.

Figure 7 shows the trend of local Nusselt number changes over the dimensionless plate in terms of dimensionless frequency. Increasing the frequency increases the Nusselt number, moreover, increases the flux of heat transfer.
Figure 7. The local Nusselt number at the moment ($t^* = 0$) for different values of ($f^*$)

Figure 8. Average, minimum, and maximum temperature in terms of dimensionless frequency

Figure 9. Mean Nusselt number changes versus dimensionless frequency
Figure 8 demonstrates that higher frequency decreases the minimum, average, and maximum temperature of the impinging plate, besides, these temperatures approach a single value indicating the uniformity of the plate temperature at higher frequencies and thus reducing the thermal stress of the impinging plate to the lowest value.

The changes of the mean Nusselt number versus the synthetic jet frequency ($f^*$) are plotted in Figure 9. It is possible to approximate the changes of the mean Nusselt number versus frequency linearly. It should be noted that with increasing frequency, the power consumption of synthetic jets increases, which is an undesirable issue.

3.3. Comparison and validation of the results of this research with previous studies

To compare and validate the results of this study, the Nusselt number diagram of the synthetic jet in different phases using the simulation method of the present study is compared with the results of the work of Chandra Tilleke et al. [19]. According to Figures 10 and 11, the results obtained by the present simulation method are in accordance with the results of Chandra Tilleke study [19].
4. Conclusions

1) Cooling efficiency improves with increasing \((H/d_o)\) ratio (within the research range). Although the effect of this variable on the Nusselt number is not significant, its excessive reduction impairs the cooling process.

2) The synthetic jet can cool the plate with limited dimensions in steady-state conditions. By increasing the ratio of the length of the plate to the width of the orifice, the temperature on both sides of the plate increases leading to thermal stresses. This problem can be overcome by increasing the number of synthetic jets. The temperature profile becomes uniform by reducing this ratio.

3) Determining the diameter of a synthetic jet is very influential in its performance so that if this diameter is not adequate, the cooling process will not be done properly. As the diameter of the cavity increases, the temperature of the impinging plate decreases, and the temperature curve becomes more uniform.

4) The effect of height of the cavity is not significant on the performance of the synthetic jet.

5) Synthetic jets have very poor efficiencies for cooling vertical plates while they can be very effective for angled plates. Moreover, synthetic jets have a greater impact on angled plates than horizontal plates.

6) Oscillation frequency and amplitude are the most effective parameters on the performance of synthetic jets. In general, with increasing frequency and amplitude, both the mean Nusselt number and the Reynolds number increase linearly.

References

[1] A. Glezer, M. Amitay. Synthetic jets. Annual review of fluid mechanics. 34 (2002) 503-29.

[2] P. Gil, J. Wilk, R. Smusz, R. Galek. Centerline heat transfer coefficient distributions of synthetic jets impingement cooling. International Journal of Heat and Mass Transfer. 160 (2020) 120147.

[3] Y. Zhang, P. Li, Y. Xie. Numerical investigation of heat transfer characteristics of impinging synthetic jets with different waveforms. International Journal of Heat and Mass Transfer. 125 (2018) 1017-27.

[4] Y. Wang, G. Yuan, Y.-K. Yoon, M.G. Allen, S.A. Bidstrup. Large eddy simulation (LES) for synthetic jet thermal management. International Journal of Heat and Mass Transfer. 49 (2006) 2173-9.

[5] M. Jain, B. Puranik, A. Agrawal. A numerical investigation of effects of cavity and orifice parameters on the characteristics of a synthetic jet flow. Sensors and actuators A: Physical. 165 (2011) 351-66.

[6] J. Zhang, X. Tan. Experimental study on flow and heat transfer characteristics of synthetic jet driven by piezoelectric actuator. Science in China Series E: Technological Sciences. 50 (2007) 221-9.

[7] T. Persoons, A. McGuinn, D.B. Murray. A general correlation for the stagnation point Nusselt number of an axisymmetric impinging synthetic jet. International Journal of Heat and Mass Transfer. 54 (2011) 3900-8.

[8] U. Akdag, O. Cetin, D. Demiral, I. Ozkul. Experimental investigation of convective heat transfer on a flat plate subjected to a transversely synthetic jet. International communications in heat and mass transfer. 49 (2013) 96-103.

[9] S. Mokhtari, K. Skelly, E. Krull, A. Coughlan, N. Mellott, Y. Gong, et al. Copper-containing glass polyalkenoate cements based on SiO\(_2\)-ZnO-CaO-SrO-P\(_2\)O\(_5\) glasses: glass characterization, physical and antibacterial properties. Journal of Materials Science. 52 (2017) 8886-903.

[10] S. Mokhtari, A. Wren. Investigating the effect of copper addition on SiO\(_2\)-ZnO-CaO-SrO-P\(_2\)O\(_5\) glass polyalkenoate cements: physical, mechanical and biological behavior. Biomedical Glasses. 5 (2019) 13-33.

[11] S. Chon, L. Piraino, S. Mokhtari, E. Krull, A. Coughlan, Y. Gong, et al. Synthesis, characterization and solubility analysis of amorphous SiO\(_2\)-CaO-Na\(_2\)O-P\(_2\)O\(_5\) scaffolds for hard tissue repair. Journal of Non-Crystalline Solids. 490 (2018) 1-12.

[12] S. Mokhtari, E. Krull, L. Sanders, A. Coughlan, N. Mellott, Y. Gong, et al. Investigating the effect of germanium on the structure of SiO\(_2\)-ZnO-CaO-SrO-P\(_2\)O\(_5\) glasses and the subsequent influence on glass polyalkenoate cement formation, solubility and bioactivity. Materials Science and Engineering: C. 103 (2019) 109843.

[13] A. Rostamijavanani, M. Ebrahimi, S. Jahedi. Thermal Post-buckling Analysis of Laminated Composite Plates Embedded with Shape Memory Alloy Fibers Using Semi-analytical Finite Strip Method. Journal of Failure Analysis and Prevention. (2020) 1-12.
[14] A. Rostamijavanani. Dynamic Buckling of Cylindrical Composite Panels Under Axial Compressions and Lateral External Pressures. Journal of Failure Analysis and Prevention. (2020) 1-10.

[15] M. Nabavi, M. Elveny, S.D. Danshina, I. Behroyan, M. Babanezhad. Velocity prediction of Cu/water nanofluid convective flow in a circular tube: Learning CFD data by differential evolution algorithm based fuzzy inference system (DEFIS). International Communications in Heat and Mass Transfer. 126 (2021) 105373.

[16] M. Nabavi, V. Nazarpour, A.H. Alibak, A. Bagherzadeh, S.M. Alizadeh. Smart tracking of the influence of alumina nanoparticles on the thermal coefficient of nanosuspensions: application of LS-SVM methodology. Applied Nanoscience. (2021).

[17] S.G. Holagh, M.A. Abdous, M. Shafiee, M.A. Rosen. Performance evaluation of helical coils as a passive heat transfer enhancement technique under flow condensation by use of entropy generation analysis. Thermal Science and Engineering Progress. 23 (2021) 100914.

[18] F. Bahramian, A. Akbari, M. Nabavi, S. Esfandi, E. Naeiji, A. Issakhov. Design and tri-objective optimization of an energy plant integrated with near-zero energy building including energy storage: An application of dynamic simulation. Sustainable Energy Technologies and Assessments. 47 (2021) 101419.

[19] T.T. Chandratilleke, D. Jagannatha, R. Narayanaswamy. Synthetic jet-based hybrid heat sink for electronic cooling. Heat Transfer-Mathematical Modelling, Numerical Methods and Information Technology. InTech2011. pp. 435-54.