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Research Article

NUMERICAL ANALYSIS OF HEAT TRANSFER ENHANCING PARAMETERS ON IMPINGING AIR JETS

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ABSTRACT

Impinging air jets are widely used in industry for heating, cooling and drying. Single and multiple impinging air jets provide the best configuration for convective heat and mass transfer to a surface. In this study, the convection air jets were examined numerically. Air velocity (12, 23 and 35 m/s), geometry dimensions (H/D distance= 4, 6, 8, 10, 12), number of nozzles (single and double) and the distance between nozzles (n = 50, 75, 100) were selected as parameters. For single nozzle, the most efficient condition was seen in H/D=10 and the highest Reynolds number. It was determined that the efficiency of heat transfer after H/D=10 has decreased for single nozzle. The most efficient heat transfer for the double nozzle was obtained for the 12114 of Reynolds number and H/D=8. In this study, a numerical approach is presented to find an optimum solution for cooling problems in the electronics industry.
Keywords: Heat transfer, impinging air jet, numerical method.

1. INTRODUCTION

High-velocity air jets are widely used in heating, cooling and drying due to the high heat transfer coefficient occurring in the diameter region and can do so at the desired velocity and focused way. Industrial systems, gas turbines, paper and vegetable drying, glass production and electronic systems have a wide application area [1]. Fluids have a significant effect on heat transfer. There are many experimental, theoretical and numerical studies on this subject. Hatemi et. al, [2] solved the effect of wavy wall shape on heat transfer using finite element method. Tang et. al, [3] investigated the effect of nano fluids with natural convection heat transfer between two sinusoidal walls. They showed that the effects of sinusoidal amplitude and phase deviation between the inner and outer walls were effective on surface heat transfer coefficient. Zhou et. al, [4] studied a sinusoidal wave structure of the microchannel for cooling of electronic devices numerically. They found that the heat transfer in the wavy microchannel was 2.8 times higher than the normal flat channel.

Many literature studies have been carried out about impinging air jet. Many factors such as jet angle, wing and non-wing surfaces, wing geometry, nozzle-plate distance, jet-jet distance, jet angle plate type were taken into consideration. The experimental study of the effect of the

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thermodynamic geometry on heat transfer properties was researched by Hardisty and Can [5]. Gau and Chung [6] investigated the cooling flow and heat transfer process on the surface. Golcu et. al, [7] conducted an experimental study of air jets and cooling in the tempering of auto glass. Fregau et. al, [8] studied the numerical heat transfer correlations of hot air jets hitting a 3-dimensional concave surface. Celik and Eren [9] investigated the effect of turbulence density on heat transfer in in a striking circular jet by keeping the jet diameter large. Garimella [10] studied local heat transfer using multiple jets (5000< Re < 20000) and jet plate distance (0.5<H/D<4). It was found that a decrease in the jet –plate Range increased the coefficient of heat transfer in multiple jets and this effect was stronger in higher Reynolds numbers. Elibol and Turkoglu [11] conducted a numerical survey of the jet hitting a flat surface with pore. When the hot plate surface is covered with porous layer, studies with the help of ANSYS fluent show that the porous layer is more efficient than the porous layer if the porous material is within the range of certain porosity and thickness values. Geers et. al, [12] made heat transfer correlations for hexagon and sequential sequences in impinging air jets. Plate and nozzle output range 3D–10D and nozzle range 2D-6D are taken. As a result of experimental studies, more than one jet heat transfer was affected by jet interactions. Calisir et. al, [1] examined the effect of triangles and square wings. Reynolds number, jet-plate distance (H/D) and effect of wing geometry on flow. With the increase in velocity, it was observed that the decomposition on the surface of the square wings was earlier. When H/D = 2, he observed that the wall jets were more effective, that two adjacent wall jets were formed, and that the wall jet velocities were more effective on triangular wing surfaces. Donovan [13] examined the fluid flow and heat transfer in the impinging air jets taking into account Re (10000-30000), jet-plate distance (0.5-0.8) and the angle of the jet (30° and 90°). The maximum heat transfer at the surface occurred in the radial direction that varies with the Reynolds number and H/D. Baydar and Ozmen [14] studied air jets in high Reynolds-numbers (30000-50000) experimentally and numerically The subatmospheric district took place in the radial direction out of the stagnation point with increasing nozzle to plate. Etemoglu et. al, [15] showed that slightly better heat transfer performance was obtained using the array of holes, which is less expensive to manufacture. In order to provide data for ink dryer designers, an investigation was carried out by Turkan et. al, [16] to obtain the heat and mass transfer coefficients under impinging air jets which constitute the evaporative drying of thin ink films. The theoretical results are compared for the constant rate and falling rate periods with some experimental and theoretical results which are found to be satisfactory particularly for the drying time. Beitelmal et. al, [17] reported that Nusselt number rises as the inclination angle declines. The maximum heat transfer occurred towards from the uphill side. The experiments were conducted for jet velocity between 6.3 m/s and 18.7 m/s and for nozzle exit to plate between 4 and 12 and for inclination angle between 40° and 90°.

In literature, nozzle exit to plate spacing, Reynolds number,turbulance models, air velocity, inclination angle of the air jet parameters were studied by many researcher. In this study, the effect of the multiple nozzles on heat transfer was also investigated parametrically for practical/industrial limits. The effects of single and multiple impinging air jets on heat transfer were investigated. For this purpose, the effects of different air velocities, Reynolds numbers, H/D geometrical dimension, nozzle number and nozzle distance on heat transfer were studied numerically. The results of numerical solutions are supported by theoretical solutions. Numerical solutions were made in ANSYS Fluent program.

2. MATERIAL AND METHOD

2.1. Governing equation

Fluid flow and heat transfer from the impinging air jet to the plate are expressed by momentum and energy equations. Fluid flow is defined by the conservation equations of mass (the continuity equation), momentum (Navier-Stokes equations) and energy (the temperature equation for the fluid) [18].

To obtain the extra terms arising from velocity and temperature, velocity gradient and turbulence viscosity should be calculated. The turbulence velocity scale is estimated by calculating the terms kinetic energy and dissipation rate. In the SST model, the wall approach method in Ansys program is used to define heat transfer on the surface [18].

2.2. Turbulance Model

Turbulent flow occurs when inertial forces are higher than viscous forces. The velocity behaves differently at each point. The aim in the creation of turbulence models is to assist in the solution of differential equations which are difficult to solve analytically. In fluid mechanics, this problem is that turbulence flow cannot be solved by a turbulence model by applying the oscillations to the flow type.

SST k-w (SST): It is stated that the standard k-w model is calculated with realistic velocity profiles as well as the excess shear stress. It uses the original k-w model in the boundary layer and the standard k-ε model in free slip flows [19].

k kinetic energy:

$$\rho \frac{Dk}{Dt} = \tau_{ij} \frac{\partial U_i}{\partial x_j} - \rho \beta^* f_{\beta^*} k\, \omega + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\sigma k}{\sigma x_j}\right] \tag{1}$$

ω dissipation rate:

$$\rho \frac{D\omega}{Dt} = \alpha \frac{\omega}{k} \tau_{ij} \frac{\partial U_i}{\partial x_j} - \rho \beta f_{\beta} k\, \omega^2 + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\omega}\right)\frac{\sigma \omega}{\sigma x_j}\right] \tag{2}$$

Among the turbulence models used related to impinging air jet in the literature, it was found that the most suitable results were obtained in SST k-w model [18,20]. Since the effect of turbulence models was not examined in this study, analyzes were conducted using SST turbulence model.

2.3. Solution Geometry and Boundary Conditions

In this study, the geometry, Reynolds numbers and heat flux studied by Beitelmal et. al, [17] were chosen to evaluate the results. In geometry, the surface and nozzle diameters, which were hit by air, were taken as 200 mm and 5.5 mm respectively (Figure 1).

The Reynolds number was 4100, 7960 and 12114 in the analyzes. Re = UD/v where d = 0.0055m, u = 12, 23 and 35 m/s , Re=4100, Re=7960 and Re=12114 values were reached, respectively. The H/D ratio was taken as 4, 6, 8, 10, 12. The distance between nozzles in double nozzle analysis was accepted as s = 50mm, 75mm and 100 mm.

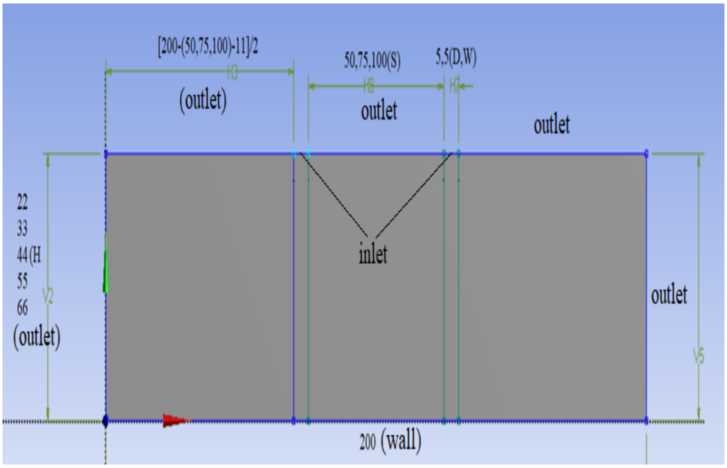


Figure 1. Geometric dimensions

2.3.1. Boundary Conditions

In the analyzes, solutions were obtained by using the following boundary conditions for free jet [17].

Jet output $v = u_\infty$ (velocity profile), $u = 0$, $T\infty = 300K$,
Jet walls $u = 0$, $v = 0$ (wall condition),
On the target plate $u = 0$, $v = 0$ (wall condition) $q = 3950\ W/m^2$ (constant heat flux),
$P = 0$ kPa (Output condition) to the left of the plate,
$P = 0$ kPa (Output condition) to the right of the plate,
Right and left of the jet $P = 0$ kPa (Output condition).

2.4. Calculation of Average Nusselt Number

In this study, the Nusselt number at the point of stagnation for the single nozzle can be calculated using Eq. (3) and (4) [21].

$$\frac{\overline{Nu}}{Pr^{0.42}} = \frac{3.06}{\frac{0.5}{Ar}+\frac{H}{W}+2.78} Re^{m} \tag{3}$$

$$m = 0.695 - \left[\left(\frac{1}{4Ar}\right) + \left(\frac{H}{2W}\right)^{1.33} + 3.06\right]^{-1} \tag{4}$$

Available range:

$$3000 \leq Re \leq 90.000 \quad 2 \leq H/W \leq 10 \quad 0.025 \leq A_r \leq 0.125$$

Eq. (5) and (6) equations can be used for the average Nusselt number in the case of double nozzles [21].

$$\frac{\overline{Nu}}{Pr^{0.42}} = \frac{2}{3} {A_{r,o}}^{3/4} \left(\frac{2\,Re}{\frac{A_r}{A_{r,o}}+\frac{A_{r,o}}{A_r}}\right)^{2/3} \tag{5}$$

$$A_{r,o} = \left[60 + 4\left(\frac{H}{2W} - 2\right)^2\right]^{-\frac{1}{2}} \tag{6}$$

Available range:

$$1500 \leq Re \leq 40.000 \quad 2 \leq H/D \leq 80 \quad 0.008 \leq A_r \leq 2.5\, A_{r,o}$$

Also in the case of double-nozzle, the correlation between Gardon and Cobonpue [22] was used.

$$Nu = 0.993\, Re^{0.625} * \left(\frac{H}{D}\right)^{-0.625} * \left(\frac{s}{D}\right)^{-0.375} \tag{7}$$

2.5. Mesh Independence Study

In order to get the most accurate results in the analyzes, the independence of the mesh was made by taking into consideration the solution period. In the studies, three structures were selected: rough mesh (7550), middle mesh (12800) and frequent mesh (16900). Geometry is divided into three parts for more accurate results. In mesh structure, the frequency of mesh is increased towards the impinging air jet surface (Figure 3).

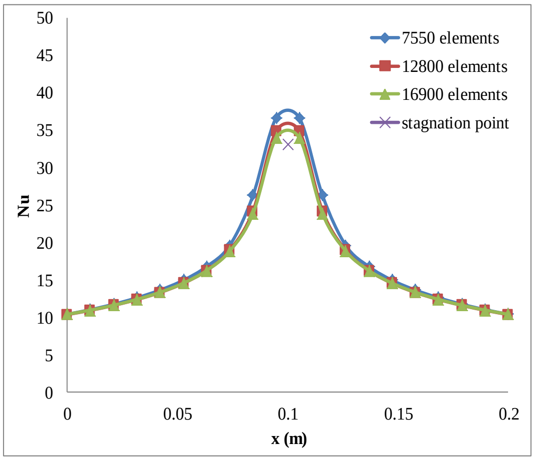
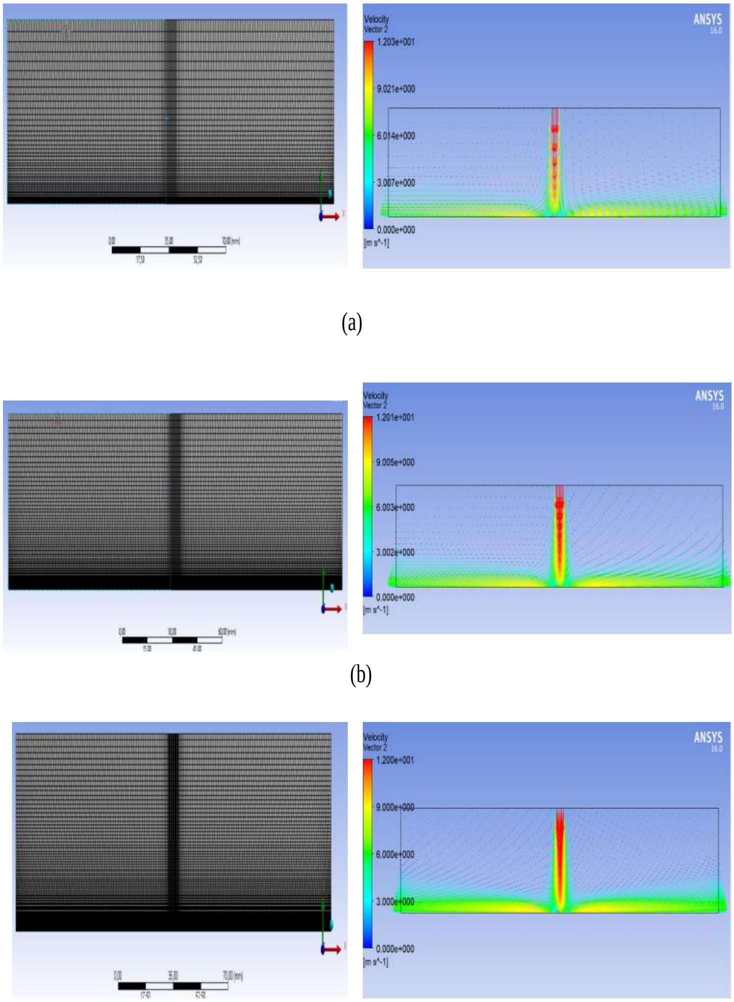


Figure 2. Comparison of the results of different mesh numbers

The comparison in Figure 2 was made in the Reynolds number of 4153 and H/D = 2. In the analyzes conducted between 7550 and 12800 mesh numbers, the average Nusselt number value converged to 2.8%. The average Nusselt number of 0.5% convergence was obtained in the analysis of 12800 to 16900 mesh numbers. As a result of the mesh independence analysis, 16900 was chosen as the most suitable mesh number considering the solution time of the analysis. Also, for the validation study, it was seen that there was a 2.7% difference between the calculated value of the stagnation point (x = 0.1 Nu = 33.01) and the calculated value of 16900 mesh (x = 0.1 Nu = 33.95).



(a)

(b)

(c)

Figure 3. Mesh and vector velocity distribution (a-7550, b-12800, c-16900 mesh number)

3. RESULTS AND DISCUSSIONS

3.1. Results for Single Nozzle

The effect of H/D, Reynolds number and nozzle number on heat transfer was investigated.

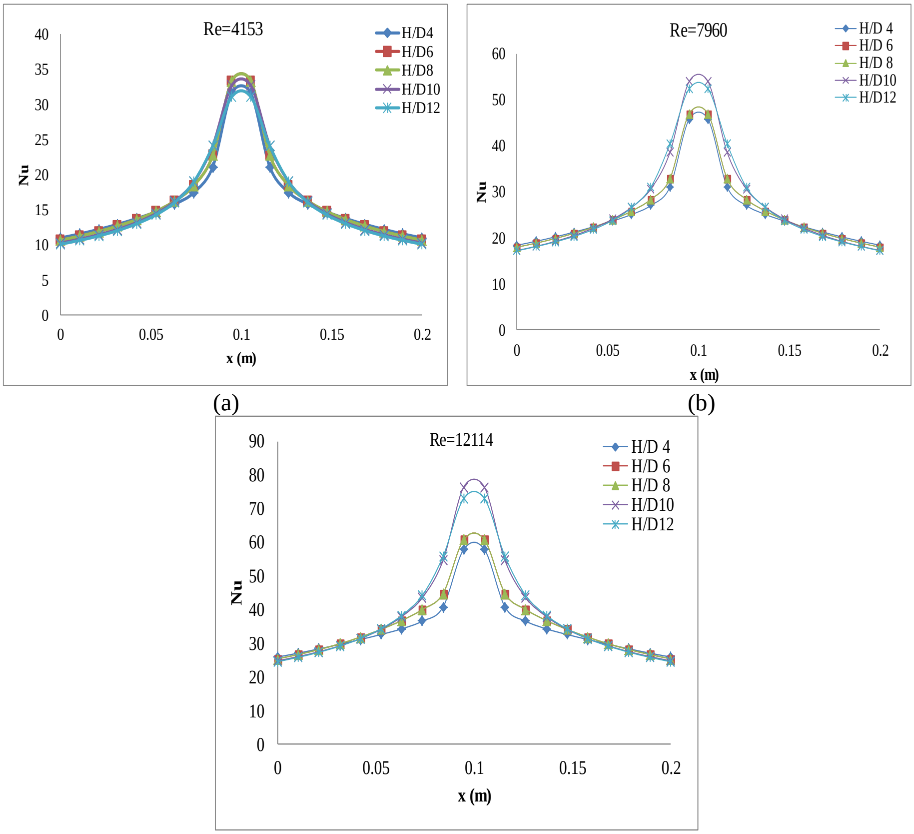


Figure 4. Variatio graphs of Nusselt number at different H/D ratios (a-Re=4153, b-Re=7960, c-Re=12114)

3.2. Results for Multiple Nozzle

In this section the Reynolds number for multiple nozzles 4153, 7960 and 12114 (velocity 12, 23, 35 m/s and d = 5.5 mm respectively) H/D = 4, 6, 8, 10, 12 and the distance between nozzles s = 50 , 75 and 100 mm were analyzed. The average Nusselt number was calculated in the equation used here [22].

For the validation study, s = 75 mm, H/D = 8 and u = 23 m/s were analyzed. The average value of the local Nusselt numbers obtained from the analysis (H/D=8) was calculated as 32.2997 and average Nusselt number using the Eq (7) was calculated as 32.4447. As a result of two different calculations, a difference of 0.44% was observed in Nusselt number (Figure 5).

Figure 4 shows that the effectivity decreased after H/D=8 or H/D=10. When the different Reynolds numbers are taken into consideration, the heat transfer on the surface rises with increasing Reynolds numbers. When the H/D=10-12 ratio is exceeded, decreases in heat transfer are observed. With the increase in the Reynolds number, a sudden and high peak occurred at the point of impact rather than the proportional spread of heat transfer. Even though the Reynolds number increased, the change in Nusselt number in low H/D ratios was not significant. However, with the increase in Reynolds number, it is seen that these changes increased up to H/D=10.

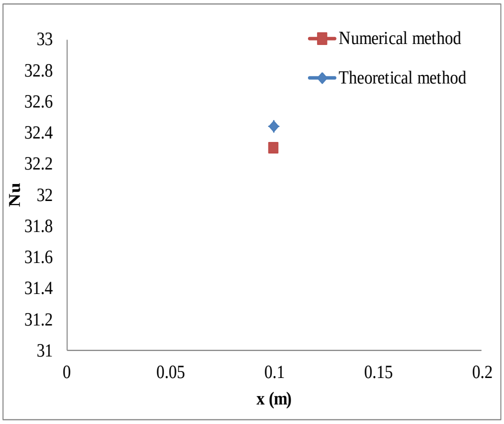


Figure 5. Comparison of numerical and theoretical methods (Re=7960 H/D=8)

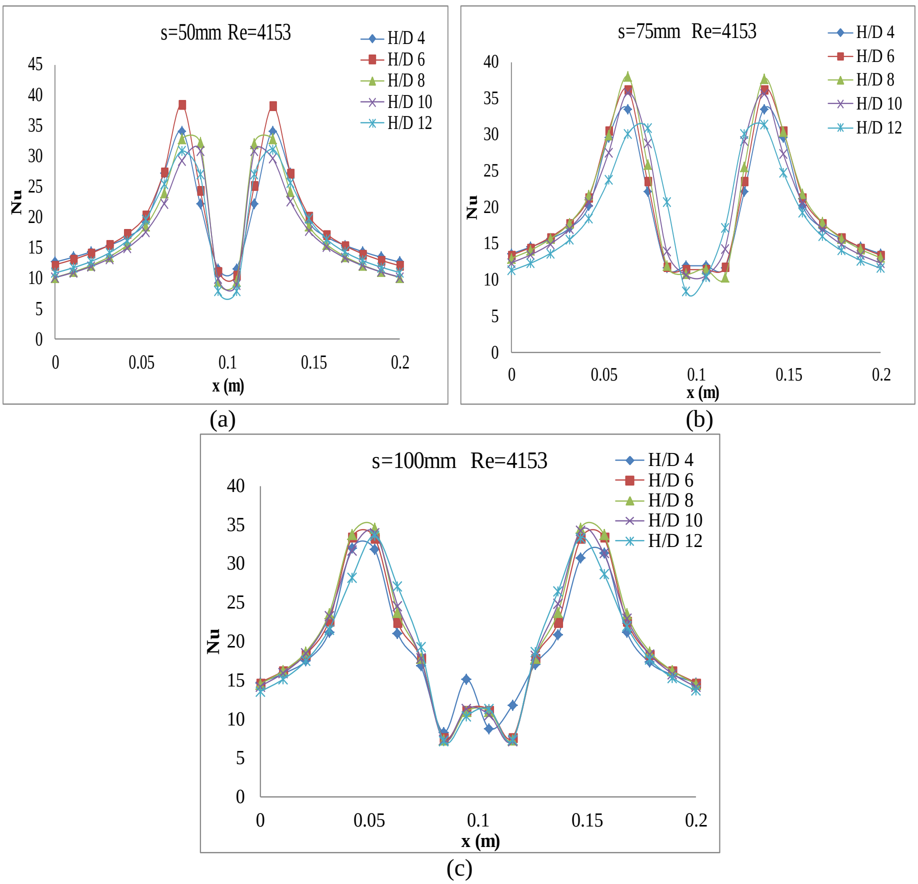


Figure 6. Variation graphs of Nusselt number obtained from different H/D ratios of 4153 Reynolds number and nozzle distances (s) (a-50mm, b-75mm, c-100mm)

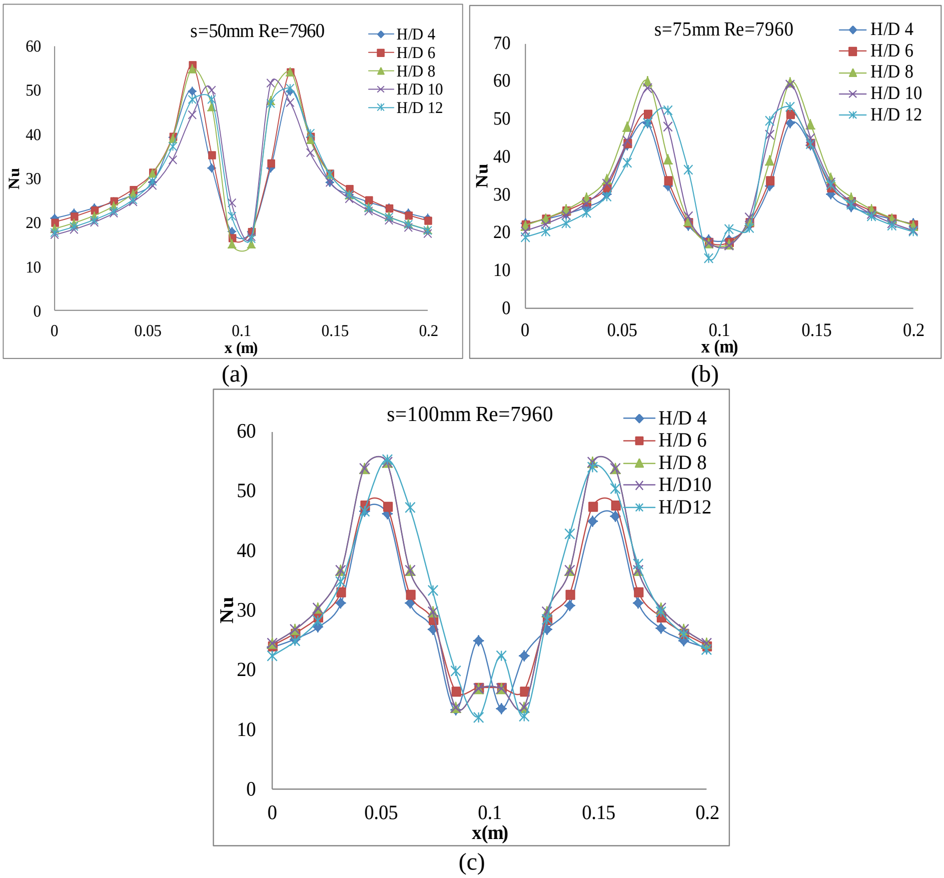


Figure 7. Variation graphs of the Nusselt number obtained from different H/D ratios of 7960 Reynolds number and nozzle distances (s) (a-50mm, b-75mm, c-100mm)

With the increase in the distance between nozzles in low Reynolds number (Re=4153), the heat transfer in the direction perpendicular to the nozzle decreased but the heat transfer towards the ends of the impinging air jet surface increased. The highest heat transfer at 50 mm nozzle distance in low Reynolds number is seen for H/D=6. After H/D=10 in single nozzle, there is a decrease in the Nusselt number. Maximum heat transfer was obtained for s = 75 mm and H/D = 8. The lowest heat transfer value for s = 50, 75 and 100 mm is determined for H/D = 12. A slight increase in the Nusselt number was observed at x = 10 cm for s = 100mm. For s= 100, it is seen that all H/D ratios of Nusselt number changes are close to each other. The highest Nusselt number for s = 50 mm is 38.4 for H/D = 6 and the lowest value is 7.8 for H/D = 12. The highest Nusselt number for s = 75mm is calculated as 37.6 for H/D = 8 and the lowest value is 8.2 for H/D = 12 (Figure 6).

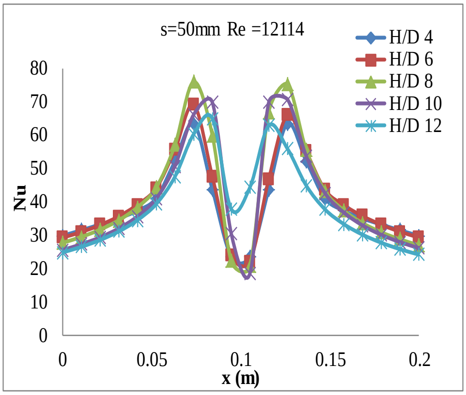
With the increase in the distance between nozzles in the Reynolds number (Re=7960), the heat transfer in the direction perpendicular to the nozzle has reached the maximum value for s = 75 mm. However, for s = 100 mm, the maximum value is reached in the regions near the outlet compared to the other nozzle distances. As the distance between the two nozzles increased, there was a fluctuation in the Nusselt number in the middle part of the impinging air jet surface.

However, no significant increase has occurred. The highest heat transfer value at 50 mm nozzle distance was obtained in H/D=6 and H/D=8 at the 50 mm nozzle distance. A significant increase was observed in the Nusselt number for H/D=10 and 12. The heat transfer in the near-outlet near the outlet yields more distant results for s=50 mm , while at s=100 mm it shows very close results with increasing heat transfer. A significant decrease was observed after H/D=12. For example, when looking for H/D=16, while it shows near-center results, a significant decrease is observed between the point of impact and the center of the geometry (Figure 7).

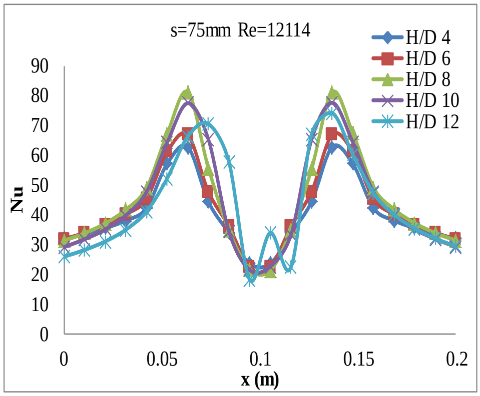
With the increase in the Reynolds number (Re=12114), the heat transfer in the direction perpendicular to nozle has increased. The results for H/D=12 at the minimum value of the distance between two nozzles in Reynolds number of 12114 were quite different compared to other analyses. In the region where the two streams collide, heat transfer remains quite high compared to the minimum value of other results. The high Reynolds number, 50 and 75 mm, maximum heat transfer for H/D=8 was observed (Figure 8).

This is especially valid for the middle (Re=7960) and high Reynolds (Re=12114) numbers in H/D=12, while the remarkable situation occurs after H/D=10 in single nozzle.

In low Reynolds numbers (Re=7960), the maximum Nusselt number for H/D=4 was seen in s = 50 mm. The difference between the highest Nusselt number and the lowest Nusselt number is 6.25%. In the low Reynolds number, the Nusselt number was observed an increase 13% for s=50 mm and H/D=4. The difference is a 15% reduction in the lowest and highest Nusselt number. However, there was no statistically significant change in the maximum and minimum Nusselt numbers for H/D=8 in the low Reynolds number, but the maximum Nusselt number was taken for s=75 mm and the minimum Nusselt number was calculated for s=50 mm.



(a)



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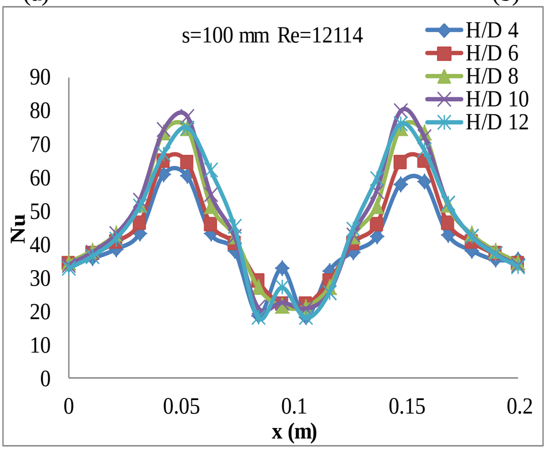

(c)

Figure 8. Variation graphs of the Nusselt number obtained from different H/D ratios of 12114 Reynolds number and nozzle distances (s) (a-50mm, b-75mm, c-100mm)

For 50 mm nozzle distance, the Nusselt number on the side exits has changed significantly. The highest Nusselt number for H/D=10 was obtained in s= 75 mm in low Reynolds numbers, but the difference with the next highest Nusselt number s = 100 mm decreased to 5.2%. The situation on surfaces near the side outlet applies to H/D=8 is also valid here. In low Reynolds number, the Nusselt number for maximum H/D has increased at s=100 mm.

The heat transfer amount for the 50 and 75 mm nozzle distance was approximately equal. In the average Reynolds number, Nusselt numbers were obtained almost equal to s = 50 and s = 75 mm for H/D=4. The maximum Nusselt number increased by 47% compared to the low Reynolds number. The difference in the output surface is approximately 33%. In the average Reynolds number, the maximum Nusselt number for H/D = 6 was obtained for s = 50 mm. The maximum Nusselt number for H/D=8 in the average Reynolds number was obtained at s=75 mm. In addition, Nusselt numbers are approximately equal to 50 and 75 mm nozzle distances. The maximum Nusselt number for H/D=10 in the average Reynolds number was calculated at s=75 mm.

4. CONCLUSIONS

In this study, the convective heat transfer with impinging air jets is investigated numerically. Air velocity (12, 23, 35 m/s), geometric dimensions (H/D=4, 6, 8, 10, 12), number of nozzles and distance between nozzles (n =50, 75, 100) were the parameters studied in this research. As expected, the highest effects were observed in the Reynolds number changes.

For single nozzle, the most effective heat transfer was seen for H/D=10 and the highest Reynolds number. For single nozzle, it has been observed that heat transfer efficiency is lost after H/D=10.

The highest heat transfer rate for multiple nozzles was obtained at H/D=8 and Reynolds number of 12114. After H/D=8 a decrease in heat transfer occurred.

The effect of impinging jets is to reduce the thickness of the boundary layer and thus augment the convection. Heat transfer coefficients in the impingement region are significantly influenced by the structure of the turbulence. Because of turbulent mixing, heat transfer is enhanced at the secondary stagnation point where two jets meet. Therefore, compared to a single nozzle, the highest heat transfer conditions are achieved with multiple nozzles.

The parameters of the single or multiple nozzle systems that affect the effective cooling or heat transfer were investigated numerically to obtain optimum operating conditions. Moreover, the numerical results of this research also provide the necessary information for the economic evaluation of the system. So, the proper use of the results of this research by the designer should lead to either a more effective heat transfer process or to a reduction in costs.

To present the necessary physical quantities of direct use for the designers, a solid numerical model has been deduced for estimating the heat transfer coefficients which appear to be reasonably accurate and reliable. Finally, it is considered that the data given in this research, provides particularly a rational basis for the air jet design of industrial equipment.

NOMENCLATURE

B	Slot width [m]
C_D	Shrinkage coefficient
D	Nozzle diameter, slot width [m]
h	Heat transfer coefficient [W/m^{2}K]
H/D	Nozzle-surface distance [m]
k	Thermal conductivity [W/mK]
L	Length of impact surface [m]
Nu	Nusselt number
P_r	Prandtl number
Re	Reynolds number
s	Distance between two nozzles [m]
S_t	Stanton number
T	Temperature [°C]
u	Nozzle output velocity [m/s]
u_{imp}	Surface striking velocity [m/s]
$q_{taş}$	Heat transfer rate [W/m^2]
μ	Dynamic viscosity [Ns/m^2]
υ	Kinematic viscosity [m^2/s]
ρ	Density [kg/m^3]

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