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Research Article

Experimental Study of Air Jet Heat Transfer Between Orifice and Heated Plate

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ABSTRACT

The characteristics of heat transfer from an impinging jet emerging from an orifice are examined through experimental methods. The impact of the distance-to-diameter ratio (Z/D), jet velocity (UJ), and jet diameter (D) on heat transfer is examined. The values tested include 2, 4, 6, and 8 for (Z/D); jet velocities of 3 m/s, 5.1 m/s, and 7.2 m/s; and diameters of 10 mm and 20 mm. The findings indicate that, in all instances, the peak heat transfer coefficient is observed at the midpoint of the plate, directly beneath the centre of the air jet. It was also noted that elevating the air jet velocity from 3 m/s to 7.2 m/s resulted in an improved heat transfer ranging from 87% to 112% for the evaluated Z/D. Expanding the orifice diameter from 10 mm to 20 mm leads to a 42% improvement in the heat transfer rate.

1. INTRODUCTION

Impingement heat transfer is a technique that is considered a viable possibility for improving heat transfer. It produces a significantly increased local heat transfer coefficient at the surface, the highest of all the techniques that may be used to improve convective heat transfer. It will be necessary to remove a significant amount of heat to do this. Direct application of this technology is possible via a simple design incorporating a heated plate and orifices **[1]**.

The process of impinging jets has been used widely in various systems, including drying, heating, and cooling. The cooling of gas turbines, the drying of paper or textiles, the processing of steel or glass, the cooling of high-density electrical equipment and cryogenic tissue freezing are only some of the technological applications that have used this material. There are several fields within fluid dynamics and thermal engineering that have shown interest in jet impingement cooling **[2]**. This is in addition to the applications that they have. The term "impinging jet" refers to a high-velocity stream of cooling fluid that is directed onto the surface and has to be cooled via an aperture or slot. The wall and the fluid will see a large increase in the rate of heat transfer as a result of this. As can be seen in Figure 1, heat transfer takes place as a consequence of the collision of fluid molecules travelling at high acceleration with the surface.

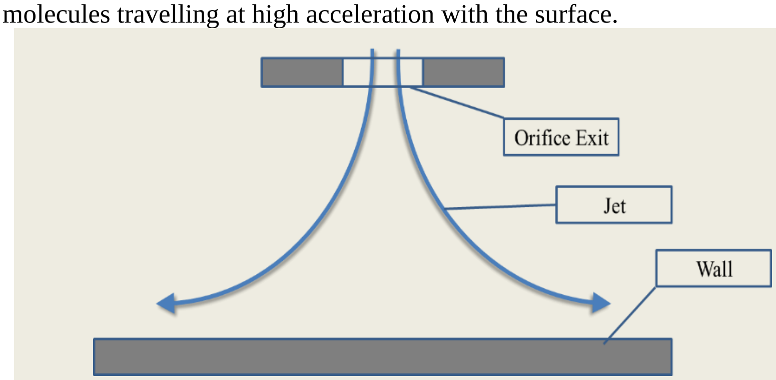

Fig. 1. An impingement jet.

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The flow field of an impinging jet may be categorised into three distinct zones, as seen in Figure 2: the free jet zone, the stagnation zone, and the wall jet zone. The jet zone is located right underneath the aperture. In this area, the ejected fluid from the opening amalgamates with the stagnant surrounding fluid, generating a flow field. Thus, in the majority of applications, the distance between the orifice and the plate is insufficient to facilitate the development of jet flow conditions. A shear layer develops surrounding the jet, with its characteristics significantly influenced by the kind of opening. The shear layer is initially relatively thin in contrast to the orifice diameter in the majority of situations, with the exception of laminar flow from a tube nozzle. As a consequence, the shear layer's dynamic behaviour is similar to that of a plane shear layer. A zone of stagnation may be found in close proximity to the point of stagnation. It is characterised by a pressure difference that halts the flow in the axial direction and redirects it in the radial direction. Additionally, the pressure gradient is responsible for restoring laminar flow once it reaches the stagnation region. As a result of the favourable pressure gradient, the boundary layer near the stagnation point is exhibiting laminar behaviour. An increase in velocity along the wall helps to keep the boundary layer thin, which in turn leads to an increase in the rate of heat transfer. As a consequence of the absence of mean pressure gradients in the wall jet zone, flow is slowed down and dispersed. A turbulent transition is brought about in the boundary layer, which was initially laminar. This change is brought about by the influence of enormous eddies that are created in the jet shear layer. It is anticipated that this chaotic transition will get more intense on a local level while the rate of heat transmission will gradually decrease [3-7].

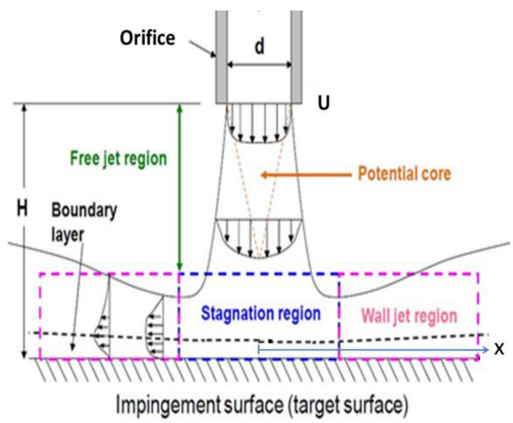

Fig. 2. Characteristic zones in impinging jets [2].

Research on the jet impingement flow structure and heat transfer has been undertaken in a great number of computational and experimental research that have been published in the literature. Heat transfer from a heated plate was measured experimentally by A. K. Mohanty and A. A. Tawfek [8]. The heat transfer was caused by the impingement of air jets from circular nozzles with diameters of 3, 5, and 7 mm. Experimental measurements were carried out. Both the Reynolds number and the ratio of nozzle to plate distance varied from 4860 to 34500. The ratio of nozzle to plate distance extended from 4 to 8. As the radial distance from the highest value at the impingement site increased, the rate of heat transfer fell at an exponential rate. An experimental investigation of the flow structure and heat transfer that occurred as a result of a single round jet impinging perpendicularly on a flat plate was carried out by Herbert Martin Hofmann et al. [9]. Using Reynolds numbers ranging from 10,000 to 30,000 and a non-dimensional surface to jet exit separation, H/D, ranging from 0.5 to 8. Tadhg S. O'Donovan and Darina B. Murray [10] investigated the heat transfer distributions that occurred when a heated flat surface was exposed to an impinging air jet. According to the findings of the research, when the surface separation between the nozzle and the impediment is minimal, denoted by H/D<2, the distribution of the mean heat transfer in the radial direction exhibits secondary peaks. It was found that fluctuations in velocity that were perpendicular to the surface had a more substantial impact on the intensity of the secondary peak in the Nusselt number distribution than these variations that were parallel to the surface. In accordance with the direction of the temperature gradient that was the highest, this was constant. A V2-f model was used by Behnia et al. [11] to predict the flow and heat transfer in circular impinging jet configurations that were either confined or unconfined. This was done in order to determine thermal transfer. There was not much of an effect that confinement had on the heat transfer coefficient, particularly at very short distances between the nozzle and the plate. On the other hand, the velocity profiles and the severity of the turbulence within the nozzle have a substantial impact on the quantitative and qualitative distribution of Nusselt number. An investigation into the numerical analysis of impinging gas jets was carried out by Eirik Martin Stuland [12]. For the purpose of determining which model is the most accurate for modelling flow and heat transfer in an impinging jet, this study makes use of a number of different computational flow models. When it comes to impinging jets, the RNG k-ε turbulence model is often recommended. For the purpose of determining the flow structure and cooling behaviour of air impingement on a heated target plate, A. Abdul Rasool and Hamad [13] conducted an investigation using both experimental and computational methods. Steeper velocity gradients may be seen in the stagnation zone of the wall jet velocity profiles when viewed from various radial locations above the surface of the target plate. This results in increased turbulence levels, which in turn enhances heat transfer in this region. Al-Dulaimi et al. [14,15] Investigated experimentally and numerically the heat transfer and flow structure of an air-sand impinging jet from a cylindrical nozzle. The results revealed that adding sand particles decreases the heat transfer rate from the plate and dampens the kinetic energy of the jet.

The primary objective of this research is to experimentally explore the impact of various parameters on the heat transfer of an air impinging jet from a heated flat surface. The influence of the distance from the jet exit and the diameter at varying jet velocities on the impingement cooling properties of a flat plate is examined. **Experimental setup**

The primary parts of the setup are the test section, measurement devices, power supply unit, air supply unit, and power supply unit. The air supply unit includes a blower, flow UPVC pipe, injection pipe and the orifice. A centrifugal blower has a port to connect the blower to the pipe. The specifications of the blower are speed (3000 rpm), power (0.44 kW), and air flow rate (0.03 m^3/min). A PVC pipe with a length (of 0.8) m and a diameter of 4 inches (10 cm) connects the blower to the orifice. It is used to emerge the air jet. Two sizes of orifices were manufactured from Teflon, which are 10 and 20 mm.

The power supply unit comprises a voltage regulator and voltage-current monitors. This device provides electrical power and regulates the current and voltage supplied to the heater. It comprises a voltage regulator and a voltage-current digital monitor. The LCD digital display is used to measure the voltage and current supplied to the heater. An LCD monitor measures current with a range of AC (0-50) A, while another LCD monitor measures voltage with a range of AC (0-500) V.

The test section is composed of a target plate, an aluminium plate, an insulation layer made of polyurethane, and a wooden frame. The parts that make up the assembly are as follows: a wooden frame, a pan of polyurethane foam, thermocouples, a flat heater, and a target plate made of aluminium. A stainless-steel plate is used as a target plate with dimensions (25x25) mm2 and a thickness of (0.4) mm. Stainless steel material is used for the plate to reduce heat conducted in the lateral direction. Six thermocouples of type (K) are welded to the back face of the target plate by Epoxy resin to measure the plate temperature. Vane-meter were used to measure the and the velocity. A schematic diagram and a photo of the rig are shown in Figure 3.

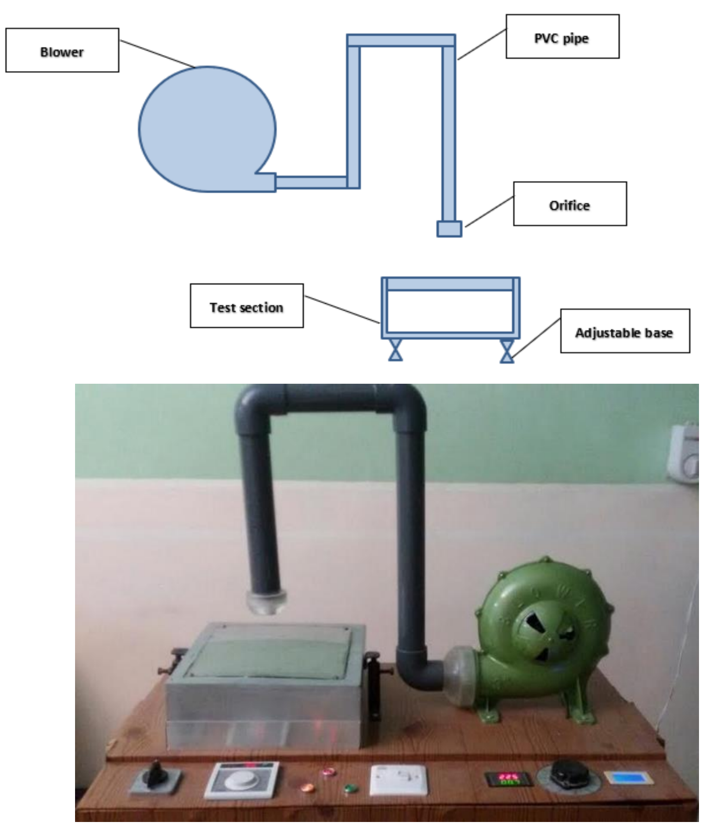

Fig. 3. Schematic and photo of the rig.

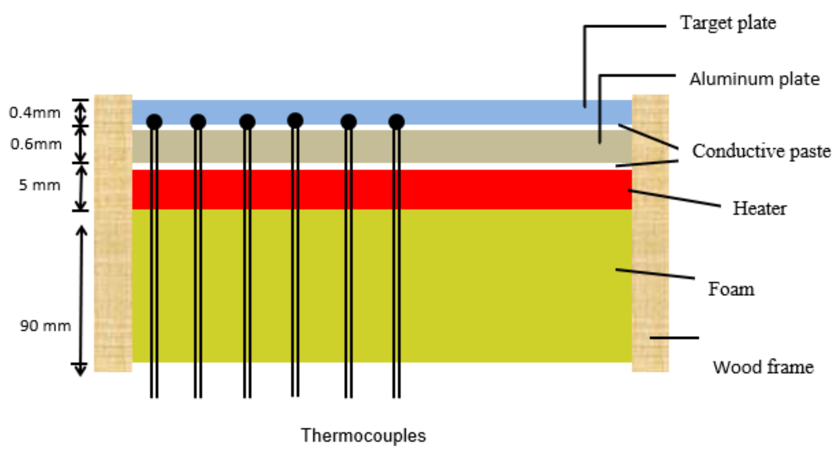

Fig. 4. Schematic of test section

2. CALCULATION PROCEDURES

The following processes have been executed to calculate the heat transfer coefficient and Nusselt number.

a. Calculation of film Temperature

The properties of air are computed based on the mean film temperature. T_f (°C) of the plate:

$$T_f = \frac{T_S + T_J}{2} \qquad (1)$$

Where T_s and T_J are the plate and jet temperatures, respectively.

b. Calculation of Reynolds Number Re

$$Re = \frac{\rho_f U_J D}{\mu_f} \qquad (2)$$

Where ρ_f is the density of the jet air (kg/m^3), U_J is the jet velocity (m/s), D is the jet diameter (m) and μ_f is the dynamic viscosity of the jet air (Pa. s).

c. Calculation of heat flux q''

The heat flux on the target plate is calculated as :

$$q'' = \frac{VI}{A_s} \qquad (3)$$

Where V is the supplied voltage (volt), I is the supplied current (amp) and A_s is the area of the target plate (m^2)

d. Calculation of local heat transfer coefficient h_x

The local convective heat transfer coefficient h_x (W/m^2.°C) is computed as :

$$h_x = \frac{q`}{T_x - T_f} \qquad (4)$$

where T_x : temperature of the surface at a certain point.

e. Calculation of Nusselt Number Nu

The Nusselt number quantifies the augmentation of heat the transfer in a fluid layer due to convection compared to conduction within the same layer and is calculated as:

$$Nu_x = \frac{h_x D}{K} \qquad (5)$$

Where k is the thermal conductivity of air at film temperature (W/m.K)

3. RESULTS

The experimental data were collected for different parameters (U_J, Z/D, and D). The temperature distributions were measured for different jet velocities (V=3,5.1,7.2 m/s) at different (Z/D) of (2, 4, 6, and 8) and for two sizes of diameter (D=10,20 mm). Temperature distribution on the target plate is shown in figure (5) for V=7.2 m/s and Z/D=6. It can be seen that the temperature increases away from the plate centre to reach its minimum value at the far end of the plate. This is due to the fact that the jet air temperature increases as it flows over the plate from the centre to the end of the plate.

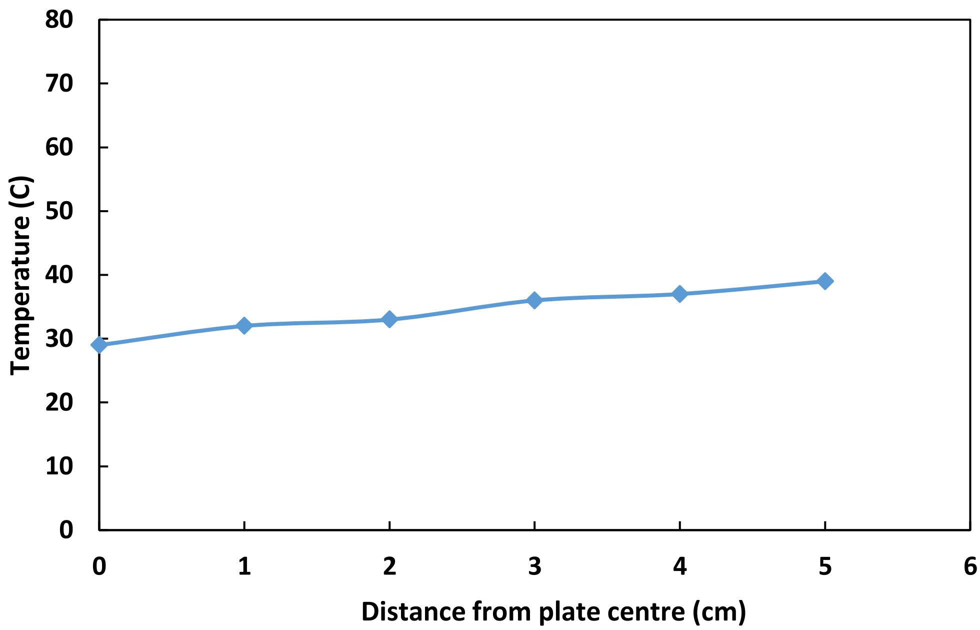

Fig. 5. Temperature distribution on the target plate (V=7.2 m/s, Z/D = 6).

Figures (6) and (7) present the heat transfer coefficient on the target plate for various Z/D ratios, with jet velocities of 7.2 m/s and 5.1 m/s, respectively. A peak is evident at the stagnation point (the plate centre) across all cases, attributed to the elevated turbulence of the flow at this location. The heat transfer coefficient exhibits a decline in the radial direction following the fluid's impact on the plate and its subsequent radial direction. The reason for this is that the fluid experiences a loss of kinetic energy, leading to an increase in the boundary layer thickness as one moves radially away from the stagnation point. The findings indicate that the distance from the orifice to the target plate significantly influences heat transfer removal. The air from the jet interacts with the surrounding air due to shear stress, resulting in the formation of small-scale vortices at the edges of the jet. The distance between the jet and the target plate influences this phenomenon.

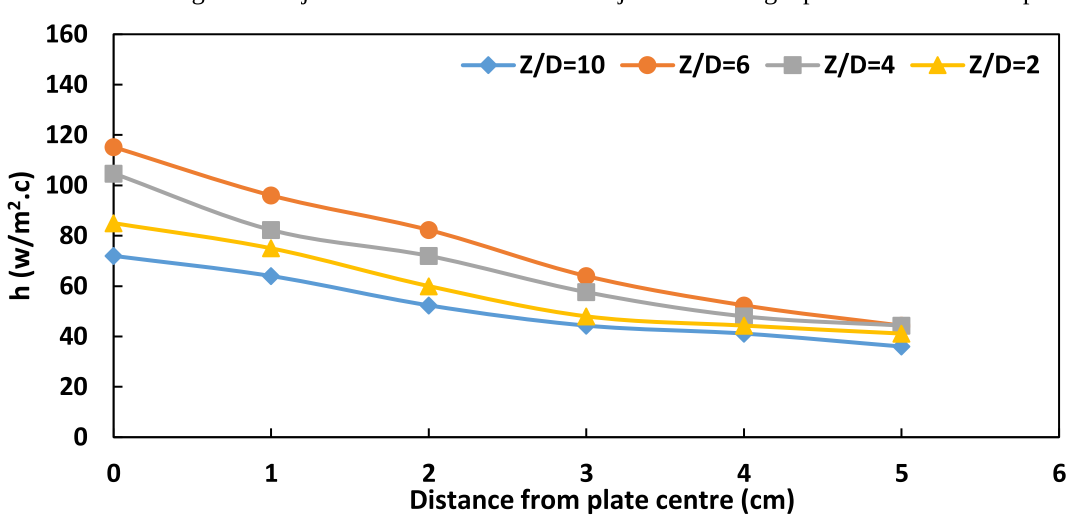

Fig. 6. Radial distribution of heat transfer coefficient at different Z/D (V= 5.1 m/s and D=10 mm).

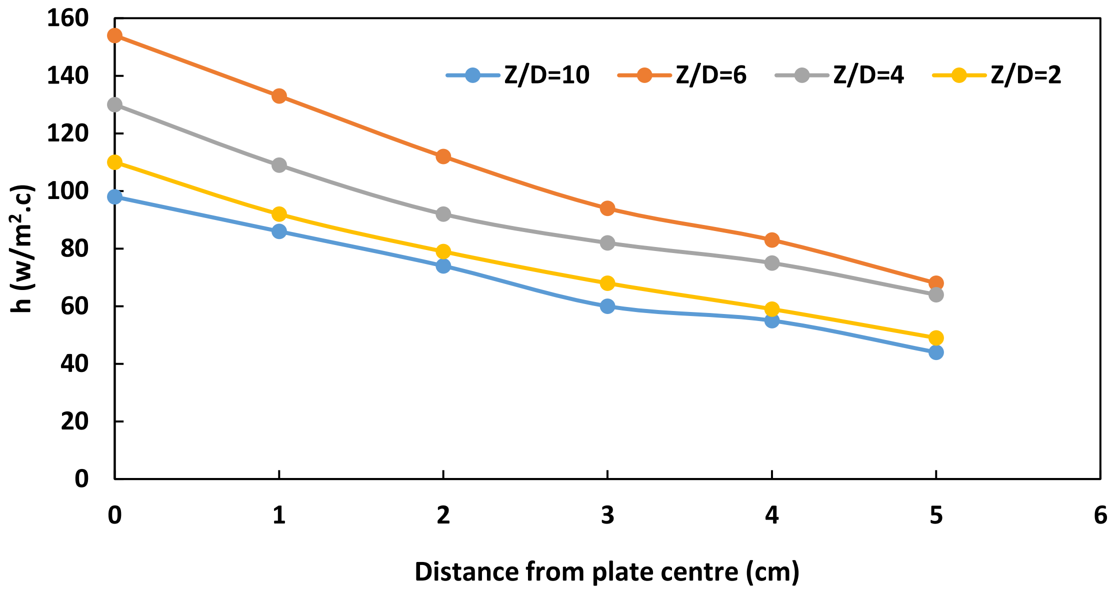

Fig. 7. Radial distribution of heat transfer coefficient at different Z/D (V = 7.2 m/s and D=10 mm).

The influence of orifice diameter for the same velocity is shown in Figure (8). It reveals that the maximum heat transfer occurs with the larger diameter. This can be attributed to the larger flow rate and the larger area subjected to direct jet in case of larger diameter.

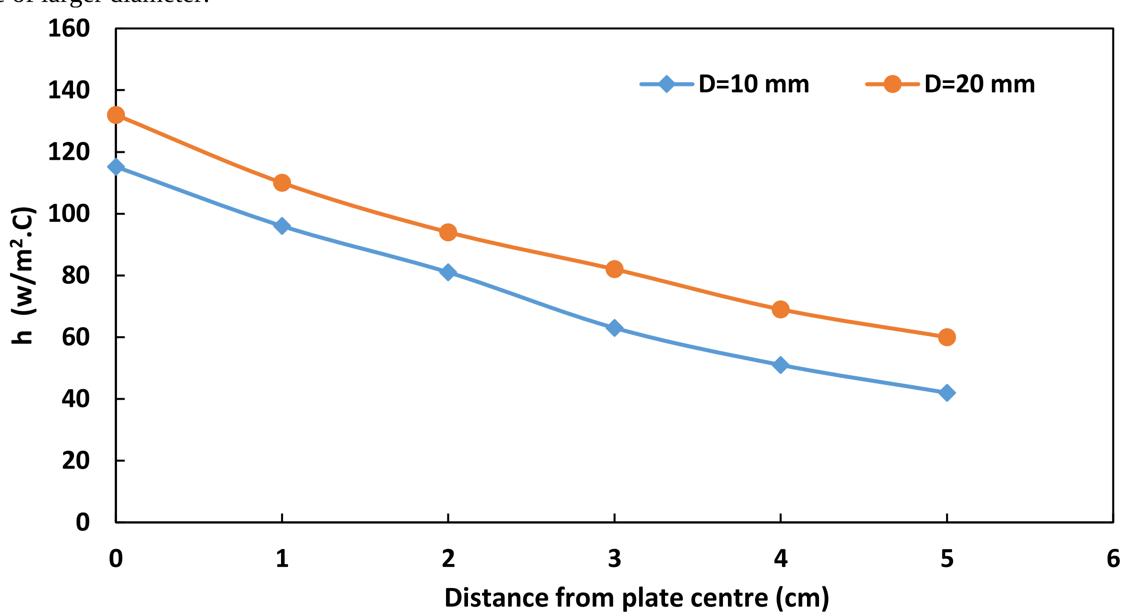

Fig. 8. Radial distribution of heat transfer coefficient for different Diameters (V=5.1 m/s and Z/D=6).

Figure (9) shows the optimum value of Z/D that gives the maximum heat transfer rate. It is obvious that the maximum heat transfer occurs at Z/D =6 for different Reynolds numbers. Variation of Reynolds number on average Nusselt number is shown in figure (10). It can be seen that average Nusselt number increases with increasing of Reynolds number, the increase in air flow over the surface resulting in more heat to remove from the heated plate.

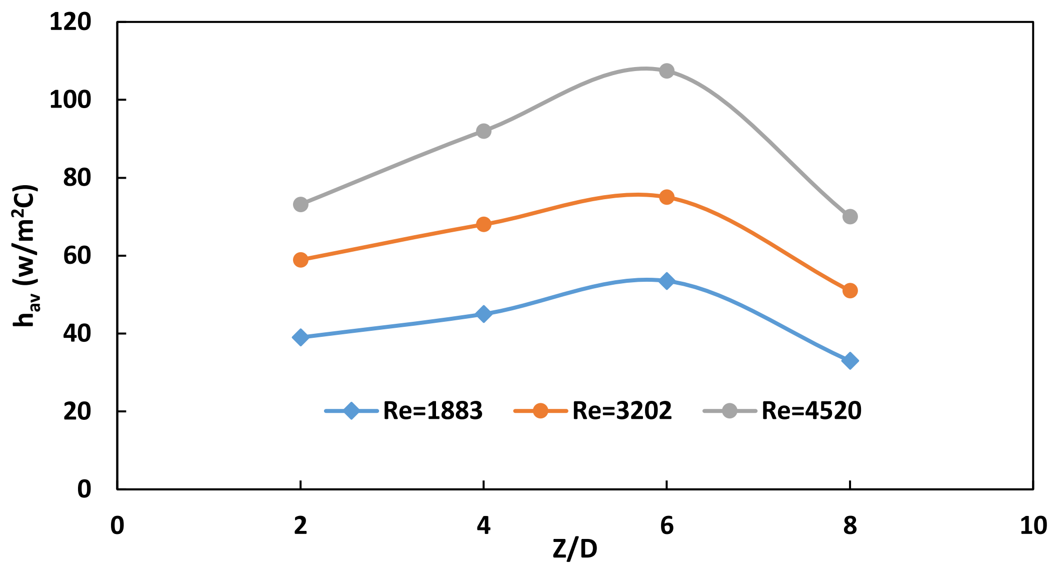

Fig. 9. Average heat transfer coefficient variation with different Reynolds numbers at Z/D values (D=10 mm).

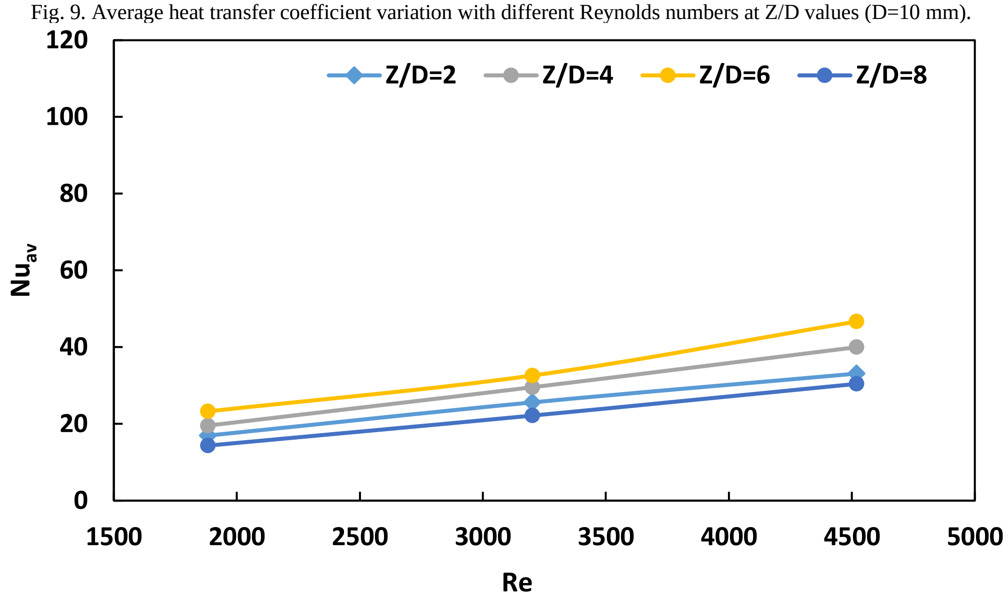

Fig. 10. Variation of average Nusselt number with Reynolds number.

4. CONCLUSIONS

Experimental research was performed to examine the heat transfer between an air impinging jet from a circular orifice and a flat plate. The Reynolds number at the orifice exit condition ranges from 1912 to 4590, while the jet distance to diameter ratio varies from 2 to 8 times the orifice diameter. The main conclusions may be summarised as follows:

1- Increasing Reynolds number augments the heat transfer from the target plate.
2- A higher heat transfer rate is achieved with a larger diameter.
3- The results from all studied cases indicate that the stagnation point heat transfer coefficient exhibits the highest values compared to other points.
4- Z/D = 6 is found to be the optimum ratio of heat transfer enhancement.

Conflicts Of Interest

The author's paper explicitly states that there are no conflicts of interest to be disclosed.

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