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The Effects of Heat Transfer Distribution between an Obliquely Smooth Flat Plate and Impinging Air Jet from a Circular Nozzle

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Abstract:

An experimental investigation is performed to study the effects of the plate angle, jet-to-plate spacing and Reynolds number on the local heat transfer distribution between an obliquely smooth flat plate and impinging circular air jet. In the tests, plate angles of 90°, 60°, and 45° were chosen, with 90° indicating a vertical stream. Depending on the nozzle situation jet-to-plate distance (L/D=2, 4, 6 and 8), its Reynolds number (1.2013E04,1.5E04 and 2.3016E04). Temperatures recorded from an infrared thermal imaging camera are used to assess the localized coefficients of heat transfer. The results show that the heat transfer coefficient, h is higher at the stagnation region and decreases gradually at the outlet region. The heat transfer coefficients increase with higher Reynolds numbers and a lower L/D ratio. In relation to the geometric impingement point, the displacement of the plate's minimum temperature point (region of maximal heat transfer coefficient) was measured. Results of experiments indicated that for a given position this displacement increases with increasing the plate angle (less than 90°), and the displacement occurred on the compression side of the plate.

Keywords: Heat transfer .air impingement .Obliquely flat .Nozzle shape .Thermal imaging.

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تأثير توزيع انتقال الحرارة بين سطح أملس مائل واصطدام هواء مندفع من فوهة دائرية

إبراهيم بكاري العربي[1], صلاح الدين سالم التحبو[1], محمد عبدالوهاب الطاهر[2], موهوب عبدالحميد تاج الدين[3] .

1.كلية الهندسة جامعة سبها, 2. المركز الليبي التقني العالي للتدريب والانتاج, 3.المعهد العالي للتقنيات النفطية اوباري.

الملخص:

تم إجراء تجربة لإستكشاف تأثير ميل زاوية الصفيحة بزاوية ميل 45°و60°و90°. على نافث العمودي. تم تغيير ميل الزاوية عند ارقام رينولدز1.2013E04 و 1.5E04 و 2.3016E04 واخذ في عين الاعتبار تغير فوهة خروج النافث في حين أن نسبة بعد الصفيحة الهدف عن الفوهة الى القطر النفاث.(2 و4 و 6 و 8) . تم إستخدام كاميرا تعمل بالأشعة تحت الحمراء لتسجيل معامل انتقال الحرارة . حيث ان النتائج أظهرت ان معامل إنتقال الحرارة h أعلى في منطقة الركود وينخفض تدريجياً عند منطقة المخرج. كما ان تزداد درجة الحرارة بزيادة عدد رينولدز و انخفاض نسبة L / D . .أشارت النتائج ايضا إلى أنه بالنسبة لموقع معين، تزداد الإزاحة مع زيادة زاوية الصفيحة (أقل من 90 درجة)، وتحدث الإزاحة على جانب الضغط من الصفيحة.

الكلمات المفتاحية :انتقال الحرارة .اصطدام الهواء .لوح املس مائل .شكل فوهة النفاث .التصوير الحراري.

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1. Introduction

One of the cooling techniques is impingement heat transfer, which is regarded as a potential strategy for enhancing heat transfer. Impingement cooling was found in the early 1960s and its use, especially in the high heat flux zones for example in the gas turbine blades. Due to its inherent high heat transfer rate properties, impinging flow devices have drawn a lot of interest. These devices enable short flow paths and a relatively high rate of cooling from a small amount of surface area. Jet impingement creates a very high heat transfer rate by forced convection, making it one of the most significant and effective methods of cooling hot items in modern industrial operations.

Impinging jets are used in a variety of industrial operations involving rapid heat transfer rates. not many industrial processes. Food product drying, textile, film, and paper processing, cooling of gas turbine blades and combustion chamber walls, cooling of electronic equipment, and other processes all use impinging jets. The peak, localized & mean heat transfer across various jet and surface configurations have been extensively researched in the field of impeller heat transfer. There are many applications of impingement cooling in gas turbines, including the cooling of the critical gas turbine parts, e.g., the outer wall of combustors and the inner wall in the leading-edge region of gas turbine blades.

Impinging jets have been in the interest of many researchers due to their common use in industrial heat and mass transfer applications. Using collisions to transfer heat is a process known as impingement cooling. These phenomena can occur when fluids come into touch with a cold or hot surface by impingement. Around the impact region, the boundary layer is very thin, and hence heat can be transferred easily.

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Among all heat transfer enhancement strategies, jet impingement has the most potential to promote local heat transfer. With easy implementation, enhanced heat transfer ratios are obtained when a jet stream was directed from a nozzle of a certain configuration to a surface.

A majority of earlier investigations focused on jet impinging over flat, smooth surfaces. From previous work, the influence of the nozzle's form on the dispersion of localized heat transfer between a smooth, flat surface and an impinging air jet was investigated by Gulati, P., and Katti, V. [1]. There are three types of nozzles: circular, square, and rectangular. They came to the result that the performance of noncircular jets is affected by the Reynolds number like that of circle jets; as Reynolds numbers rises, the amount of heat transfer increases. There are many similarities between the heat transmission properties of square and circular jets. For a rectangular jet, there is a clear distinction between Nusselt number dispersion along the major and minor axes. It is noted that, up to L/D of 6, the rectangular jet exhibits a greater Nusselt number dispersion down the horizontal axis within a stationary area than do the circular and square jets.

Nirmalkumar, M et al [2] investigated the effect of heat transfer and flow of fluids dispersion on jet-to-plate spacing (L/D) and Reynolds number. Jet-to-plate spacing (L/D) varies between 0.5 to 12, whereas the Reynolds number based on slot width varies from 4.20E03, 5.20E03, 7.80E03, 1.04E04 and 1.20E04. They came to the finding that, for just a given L/D, an increase in Reynolds number leads to a rise of the heat transfer coefficient at all sites in the direction of flow. At smaller Reynolds numbers and larger L/Ds, the secondary peak in a slot jet is not very noticeable. Only for the maximum Reynolds number of 1.20E04 and for the (L/D=6), the

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secondary peak is visible. For a specific Reynolds number, a Nusselt number so at the stagnation point rises monotonically until L/D equals 6. The greatest stagnation point Nusselt number for greater Reynolds numbers is reached at L/D = 8.

The effect of the slope of an impinging two-dimensional slot jet on the heat transfer from a flat board was investigated by Akansua, Y. E. et al. [3]. The findings demonstrated that the angle of inclination played a major role in determining the site of maximal heat transmission. The largest heat transfer occurs on the plate's uphill side as the inclination angle rises, and for smaller jet-to-plate spacings, the maximum Nusselt number gradually rises in value. At every angle of incidence, all Nusselt number dispersion remains essentially level. The Nusselt number rises together with the rising Reynolds number. Including a rise in Reynolds number, a Nusselt number increases, especially outside the zone of maximum heat transmission.

Katti, V. V., and Prabhu, S. V. [4]. studied the result of jet-to-plate spacing L/D (0.5 - 8) and Reynolds numbers (1.2E04 – 2.8E04) on the local heat transfer distribution to typically impinging underwater circular air jet on a smooth horizontal surface. Two correlations for the local Nusselt number were advanced from the experimental data for two ranges of $L/D \leq 3$ and for $L/D \geq 4$.

Additionally, Florschuetz, L. W. et al. [5] investigated how initial crossflow affected impinge heat transfer. Their findings demonstrate that the impinging heat transfer efficiency is decreased by the initial cross flow. Convection heat transfer will be more dominated by crossflow as the initial crossflow rate rises. However, the overall cooling is better as well as Nusselt number rises with a lower cross-flow, or for crossflow velocity less than 10% of a jet flow velocity.

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Heat transfer from an obliquely impinging circular jet to a flat plate was researched by Xiaojun, Y. N. [6]. The chosen oblique angles were 90°, 75°, 60°, and 45°, with 90 ° being a perpendicular jet. For the aim of comparing the two Reynolds numbers 1.0E03 and 2.3E04 with earlier data that was previously published in the literature. This dimensionless jet-to-plate distance, or L/D, varied as another measurement parameter. In the studies, L/D levels of 2, 4, 7, and 10 were taken into consideration. According to his findings, both geometric impingement locations and the compressive face of a wall jet on the symmetrical axis are no longer where the site of maximal heat transfer occurs. With a greater jet slope and a lower jet-to-plate gap, the change is much more significant.

The cooling of a hot flat plate by an obliquely impinging slot jet was researched by Haydar, E. C. [7]. the jet was slanted at angles ranging from 90° to 30° (90°, 60°, 45°, and 30°). The fluctuation of local temperatures with respect to dimensionless length (z/L) was studied for Reynolds numbers (5.86E03, 8.88E03, and 1.16E04). Through terms of the Reynolds number, dimensionless distance (z/L), and oblique angle (sin), new correlations for local temperatures were discovered. In accordance with the geometric impingement point, the displacement zone of maximal heat transfer (lowest temperature point) upon that pleasure is measured. The outcomes of the experiments indicated that displacement increased with an increasing inclination for a given position and occurred upon that compression of the plate.

The purpose of this study is to evaluate the effect of plate angle on the localized heat transfer coefficients distribution on a flat plate for an impinging circular air jet. Plate angles of 90°, 60°, and 45° were chosen, with 90° indicating a vertical jet. The influence of jet to target plate spacing (2,4,6, and 8) as well as Reynolds numbers

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1.2013E04, 1.50E04, and 2.30E04 on the dispersion of localized coefficients of heat transfer.

2. Experimental setup

The schematic layout of the experimental setup is shown in figure 1. Air was supplied by an air blower through pipes to the settling chamber and the pressure regulator and control valve to control the flow rate. The settling box provides uniform and low turbulence intensity at the nozzle exit. The air ejects from the nozzle into ambient air with a temperature that is kept constant with the jet temperature within ± 0.3 ºC during the measurements. This project study used one type of nozzle a circular nozzle with an inner diameter of 3.2cm. The flat plate with dimensional (40 cm × 30 cm and thickness of 0.5 cm).

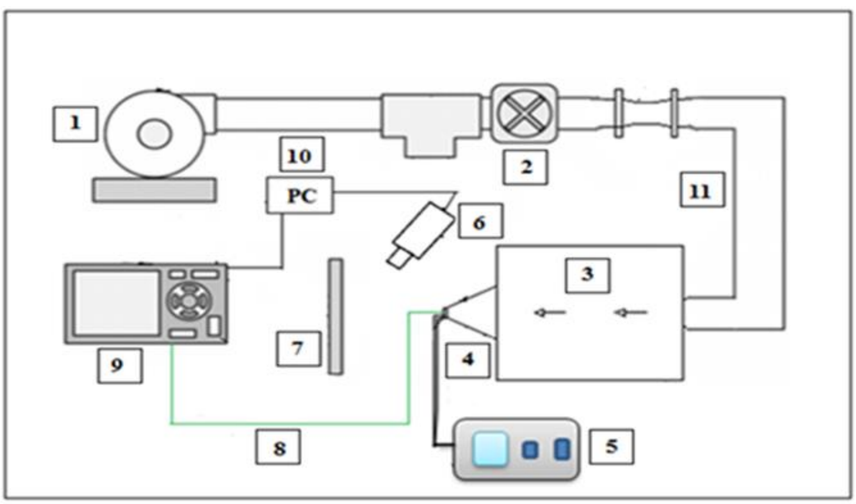

Figure 1: Layout of the experimental set-up. (1) Centrifugal blower. (2) Control valve. (3) Settling chamber. (4) Circular nozzle. (5) Air flow meter. (6) Infrared thermal camera. (7) Target plate. (8) k-type thermocouple. (9) Data logger thermocouple. (10) Computer. (11) Pipe.

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The temperatures were measured on the target plate using an infrared thermal imaging camera and which reads the temperature of the plate depending on the emissivity value of the surface of the plate. One thermocouple was installed on the center of the plate to make comparisons with the temperature that get from the thermal image and another thermocouple was installed at the nozzle exit to measure jet temperature. The measured temperature data is recorded by the data logger to the personal computer through the GPIB interface. The heat source used stainless steel as the heating element and its temperature constant at 60 °C. Diverse jet-to-plate distances are set using a traversing system.

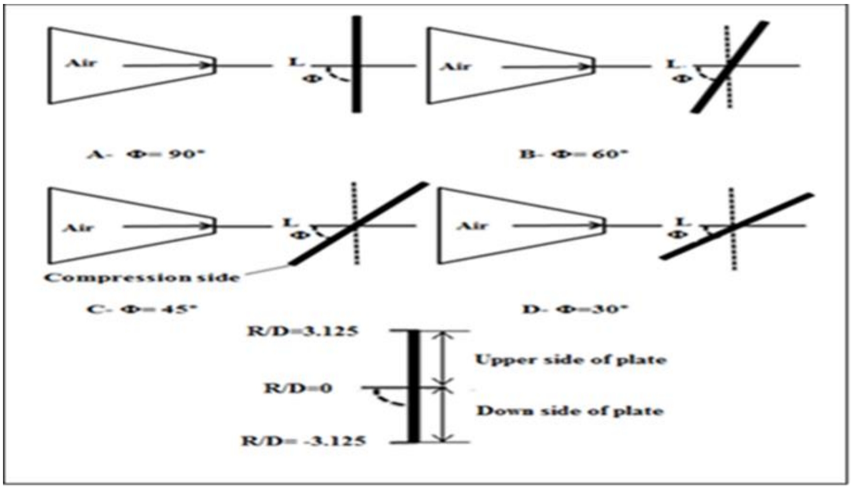

Figure 2: Positions of flat plate angles in front of jet flow

Figure 2 positions of flat plate angles in front of jet flow. The current through the heater plate is directly read from the panel of DC power source. Electricity is used to directly heat a thin stainless-steel foil that is mounted to the plate with a constant heat flux that is controlled by a slide trance and voltage regulator. On the test surface, this system may offer a constant heat flow boundary

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condition. DC power source is used to provide energy, and heat flux on the surface is controlled by varying output voltage and being measured for electrical current (I) and voltage drop (V). The data logger records the measured temperature information to the computer through the GPIB interface. On the test surface, this system is capable of supplying the boundary condition of continuous heat flux. The average jet velocity at the nozzle exit, which is utilized to define the jet Reynolds number, was found using an airflow meter instrument. In order to compute the net heat flow on the surface, the multimeter was utilized to measure the voltage and current output from the regulator voltage. Repeating this process with different flow rates, Reynolds numbers, plate angle, and jet to target plate spacing parameters. The corrections are accounted for in the local heat transfer coefficient calculation.

3. Data reduction

Impingement cooling is significantly dependent upon the jet's Reynolds number. For the target and downstream impingement surface, it is clear that area-averaged Nusselt numbers are in direct proportion to Reynolds number. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluids. The ratio is called the Reynolds number, which is a dimensionless quantity and expressed as:

$$Re_D = \frac{Inertia\ forces}{Viscous\ forces} = \frac{V\ D}{\gamma} \tag{1}$$

Where V = Free stream velocity, D = Diameter of nozzle or pipe and γ = Kinematic viscosity

At large Reynolds numbers, the inertia forces, which are inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces, and thus the viscous

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forces cannot prevent the random and rapid fluctuations of the fluid. At small Reynolds numbers, however, the viscous forces are large enough to overcome the large enough to overcome the inertia forces and to keep the fluid "in line." Thus, the flow is turbulent in the first case and laminar in the second.

For each arrangement, thermal pictures are used to determine how hot the target plate is. for a constrained circular air jet impinging on a plate with constant heat flow, the local heat transfer coefficient (h) depends on a number of factors. Together with k and D, this local heat transfer coefficient creates a local Nusselt number (Nu).

The Nusselt number for the smooth flat surface is calculated by:

$$N = \frac{h * D}{k} \tag{2}$$

$$h = \frac{q_{conv}}{(T_w - T_j)} \tag{3}$$

Radiation heat loss from the front surface, radiation heat loss from the back surface, and natural convection from the impingement plate's back surface are all examples of heat flux losses. Because total net heat flux is greater than total heat loss, total heat loss is ignored. Heat transfer rate between impinging jet and target plate is estimated as follows:

$$q_{conv} = q_{joule} - q_{loss} \tag{4}$$

$$q_{loss} = q_{rad\,(f)} + q_{rad\,(b)} + q_{\,nat} \tag{5}$$

$q_{rad\,(f)}$, $q_{rad\,(b)}$ and $q_{\,nat}$ are neglected so net heat flux from surface is equal:

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$$q_{conv} = q_{joule} = \frac{V * I}{A} \tag{6}$$

4. Results and discussions

The results of the heat transfer measurements on the impingement surface of the impinging jet at 45°, 60° and 90° degrees were collected and compared to the results presented in previous studies. Also, the results taken into consideration jet to target plate spacing L/D = 2, 4, 6 and 8. The Reynolds numbers considered are 1.2013E04, 1.50E04 and 2.3016E04. The velocities are 5.95 m/s, 7.42 m/s and 11.4 m/s corresponding to the respective Reynolds numbers, the flow air state is turbulent. The distribution of local heat transfer coefficients and Nusselt numbers along the concave surface were measured on the vertical axis. The temperature distribution on the target plate is obtained by thermal images for each configuration.

The Nusselt number at a Reynolds number of 1.2013E4, Φ=90° and L/D = 6 is compared with the earlier published data as shown in figure 3.

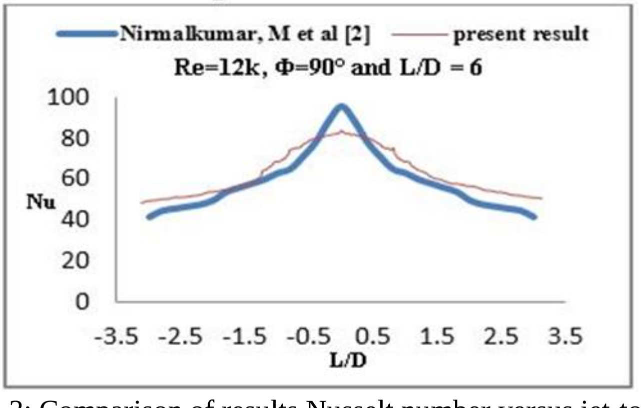

Figure 3: Comparison of results Nusselt number versus jet-to-plate distance

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It compares well with the results of Nirmalkumar, M et al [2]. The heat transfer data of Nirmalkumar, M et al is higher than the results of the present work. This difference may be attributed to the difference in the measurement techniques and thickness of the target plate.

4.1. Effect of Reynolds number on heat transfer coefficient

The heat transfer distribution for L/D=2 and L/D=6 with =90° for all Reynolds numbers is shown in Figure 4. Heat transfer has been observed to increase at all radial sites for all various Reynolds numbers as the Reynolds numbers rise. The flat surface's centreline section at R/D=0 exhibits the greatest heat transmission.

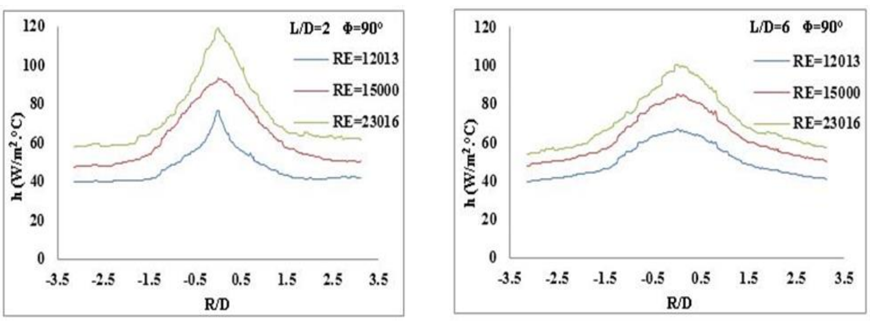

Figure 4: Variation of heat transfer coefficient for L/D=2 and L/D=6 for Φ=90° at different Re

Figure 5 depicts the distribution of heat transfer for L/D=2 and L/D=4 with =60° for all selected values of Reynolds number.

Heat transfer is seen to increase with higher Reynolds numbers at all radial positions for all different Reynolds numbers. On the primary axis, it was discovered that the location of the highest heat

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transfer had shifted from the geometrical impingement point toward the compression side.

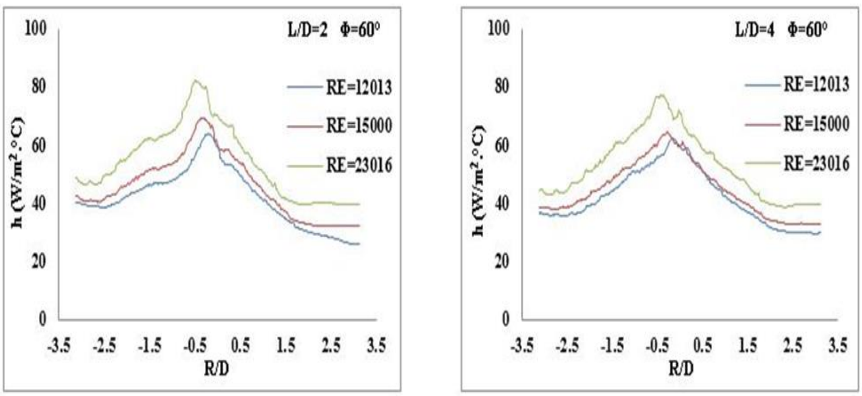

Figure 5: Variation of heat transfer coefficient for L/D=2 and L/D=4 for Φ=60° at different Re

For all selected value of Reynolds numbers, Figures 6 depict the heat transfer distribution for L/D=4 and L/D=8 =45°.

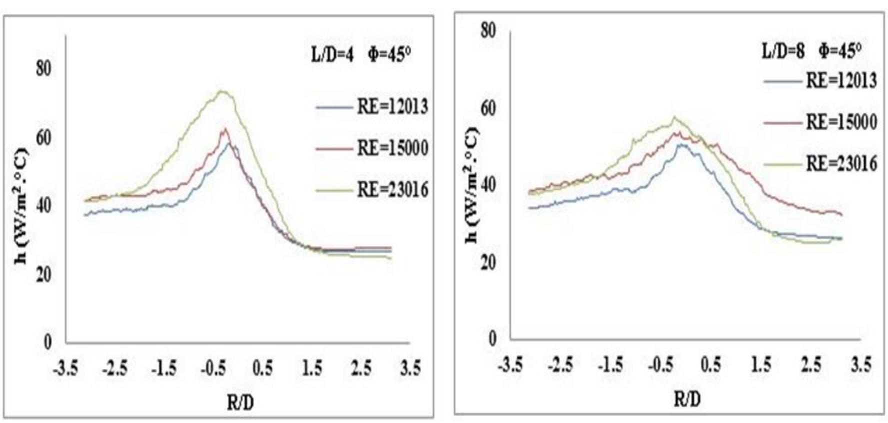

Figure 6: Variation of heat transfer coefficient for L/D=4 and L/D=8 for Φ=45°at different Re

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Heat transfer is seen to increase at all radial sites for all various Reynolds numbers as the Reynolds numbers rise. On the major axis, the highest heat transmission was discovered to go away from the geometrical impingement point and toward the compression side.

4.2. Effect of jet-to-plate distance on heat transfer coefficient

The effect of jet-to-plate distance on the local heat transfer can be seen in Figures 4 to 6. It observed as the L/D spacing increases as heat transfer coefficient decrease for all Reynolds numbers.

Figures 5 and 6 show the heat transfer results for Reynolds numbers 1.2013E04, 1.5E04, and 2.3016E04 at the shortest jet-to-plate distance, L/D=2, and at an oblique angle of 45° and 60°, respectively. It is easy to see how the point of greatest heat transfer shifts in these two cases. The heat transfer distributions on the compression side showed a monotonic decrease beyond the maximum point. On the "downhill" side, however, the distributions are those of a vertical jet for the same distance from the jet to the plate, with a second maximum heat transfer at radial distances of approximately two diameters. Actually, the distribution at 90° is typical of a corresponding vertical jet. As the jet-to-plate distance increases (see Figures 4 and 5), the drop in heat transfer on the compression side becomes less rapid. This shows that the jet flow is so evenly mixed and spread out at the greater distances (beyond the jet potential core) that the heat transfer distribution on the surface is less sensitive to the jet-to-plate distance.

For $\Phi = 60°$ and $\Phi = 45°$ the maxima of heat transfer believed to be caused by flow transition region where the local turbulence generation is strong. Because of the strong downwash of flow from the stagnation region to the "downhill" side in which flow carries more momentum as angle increases, this transition is retarded to cause the delay for the maximum point of heat transfer.

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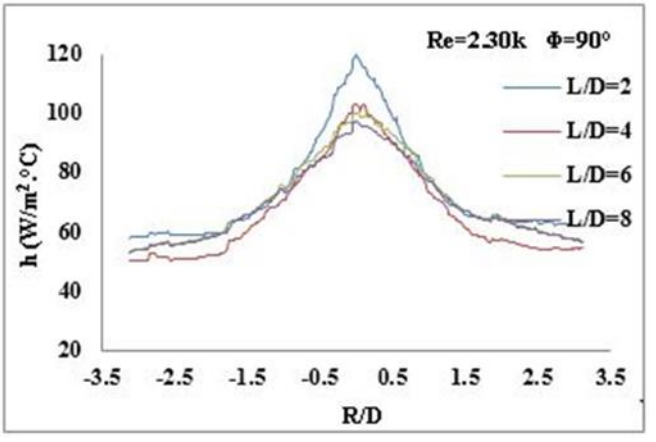

Figure 7: Variation of heat transfer coefficient for all L/D and Φ=90°at Re=23016

It observed clearly from figure 7 as the jet-to-plate distance increases the maximum heat transfer becomes near to centerline of plate and this is conformity with previous studies in this area.

4.3. Effect of plate angle on heat transfer coefficient

All of previous figures show detailed surface local heat transfer distributions for three different oblique angles (ϕ = 90°, 60°, and 45°) for various jet-to-plate distances.

Figures 8 and 9 show that for all plate angles less than 90°, the point of maximum cooling/minimum temperature drastically declines with increasing inclination degrees. The displacement of maximum heat transfer for three Reynolds numbers 1.2013E04, 1.5E04, and 2.3016E04 is shown in Figures 8 and 9. It is discovered that Reynolds numbers have a significant impact on how oblique angles affect the shift of maximum cooling. The displacement distance can be increased by raising the Reynolds number.

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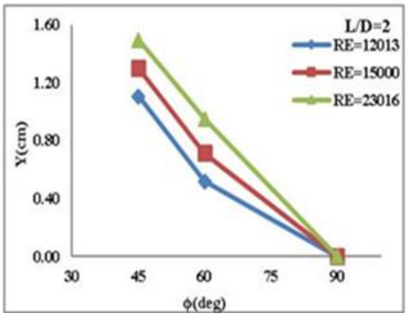

Figure 8: Variation of maximum heat transfer displacement with oblique angles for L/D = 2

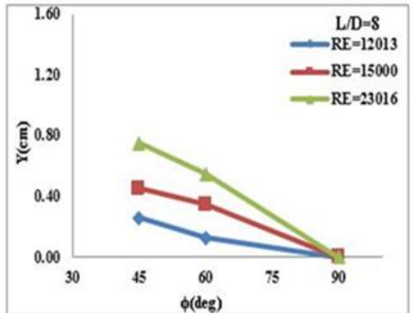

Figure 9: Variation of maximum heat transfer displacement with oblique angles for L/D = 8

For example, at L/D=2 (figure 8), the displacement of the minimum temperature point is 0.52 cm for ϕ=60° and 1.10 cm for ϕ=45° at Re=1.2013E04. In the case of Re=1.5E04, these distances are 0.71 cm for ϕ=60° and 1.30 cm for ϕ=45°. At last, for Re=2.3016E04, these values are 0.95 cm for ϕ=60°, 1.49 cm for ϕ=45°. Also figure 9 at L/D=8, the displacement of minimum temperature point is 0.12 cm for ϕ=60° and 0.25 cm for ϕ=45° at Re=1.2013E04. In the case of Re=1.5E04, these distances are 0.35 cm for ϕ=60° and 0.45 cm for ϕ=45°. At last, for Re=2.3016E04, these values are 0.55 cm for ϕ=60°, 0.75 cm for ϕ=45°.

The minimum temperature cooling points on the impingement plate with respect to oblique angles were plotted at various Reynolds numbers in Figures 10 and 11. Increasing the Reynolds number causes an increase in the cooling rate of the plate directly. Here, the best cooling is obtained at Re =2.3016E04 with Φ=90°.

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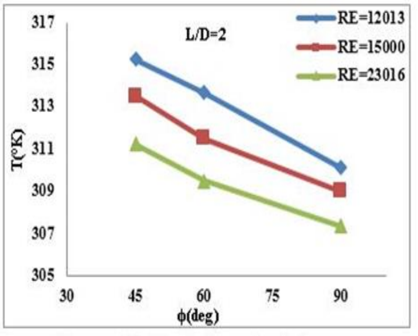

Figure 10: Variation of minimum temperature point with oblique angles for L/D=2

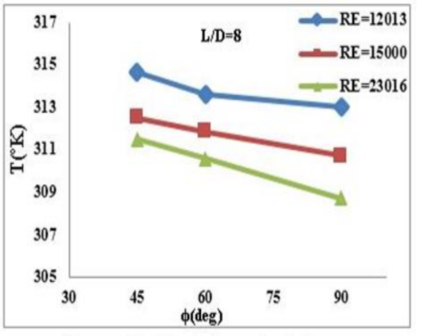

Figure 11: Variation of minimum temperature point with oblique angles for L/D=8

The impact of an oblique angle on the heat transfer coefficient is also shown in Figures 12 and 13. A standard normal jet cools more slowly than one with an angle of 45° instead of 90°, which increases the slope from the jet outlet. The full local heat transfer coefficient distributions for three different oblique angles 90°, 60°, and 45° at three different Reynolds numbers are also shown in figures 12 and 13.

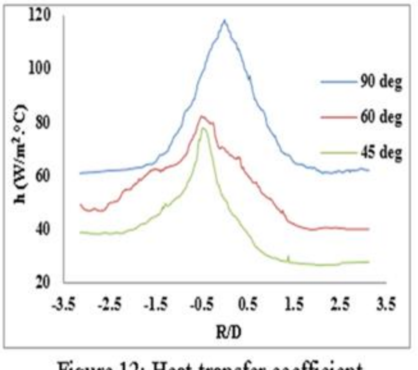

Figure 12: Heat transfer coefficient distribution with L/D=2 at different plate angles for Re=2.3016E04

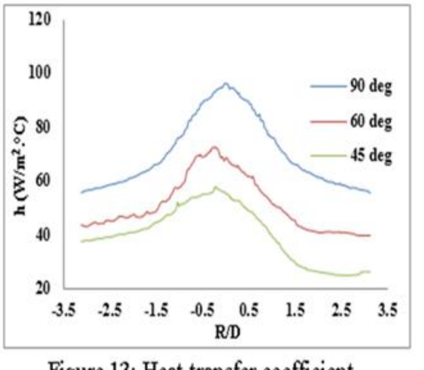

Figure 13: Heat transfer coefficient distribution with L/D=8 at different plate angles for Re=2.3016E04

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The inclination angle of 90° is well known conventional jet position. For all Re number values, the geometric impingement point has a minimum temperature of =90°. The point of minimum temperature and maximum cooling then changes away from the geometric impingement point and toward the compression side on the primary axis at =60° and 45°, respectively. The down side of plate is compression side. When an inclination angle is given to plate (90° – 45°), the upper side of plate goes far away from jet exit, so this side is not cooled as good as down side.

At the oblique angle ϕ=90°, the inclination parameter does not affect the temperature distribution. The temperature at a given location rises with decreasing angle (90°–45°), which falls with decreasing dimensionless distance (L/D). It can be also said that; at ϕ=90°; Re number is more effective than dimensionless distance. However, in the inclined positions of plate the dominant parameter on temperature variations is the dimensionless distance.

5. Conclusions

After the experimental analysis and discussion that have been done, heat transfer coefficients from an obliquely impinging air jet to a flat plate were obtained experimentally. The effect of the oblique angle (ϕ) on the local heat transfer coefficient was investigated for different Reynolds numbers of 1.2013E04, 1.5E04, and 2.3016E04. The plate angles are ϕ=90°, 60°, and 45°, and the jet-to-plate distances are 2, 4, 6, and 8. The general conclusions of this study accomplished are outlined below:

1. As increase the Reynolds number, heat transfer coefficient increases at all radial locations.
2. As decrease jet to plate distance, heat transfer coefficient increases at all radial locations.

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3. The stagnation region has a faster rate of heat transfer than the outflow region along flat surfaces. The distribution of heat transfer is more asymmetric as the distance from the jet to the plate decreases.
4. The heat transfer coefficient distributions for three different plate angles, ϕ=90°, 60° and 45°at three different Reynolds numbers. The maximum heat transfer coefficient is measured at ϕ=90°, on the geometrical impingement point for all Re number values. Then at ϕ=60° and 45° the point of the maximum heat transfer coefficient shifts away from the geometrical impingement point toward the compression side. The down side of plate is compression side and the upper side of plate goes far away from jet exit for 60° and 45°, so this side is not cooled as good as down side.
5. The scaled shift of maximum heat transfer coefficient increases with a decreasing plate angle (less than 90°). The shift is found to be clearly for smaller jet-to-plate distance

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