The effect of system parameters like flow rate and Reynolds number $Re_d$ on the stagnation Nusselt number ($Nu_O$) is presented in Fig. 4.1. It can be observed from Fig. 4.1 that an increase in $Re_d$ (see curves 1 and 2) due to an increase in the velocity of air and decrease in temperature of component. In this connection, it can be observed that for $H/d < 4$ there is no significant variation in the stagnation Nusselt number. When $H/d \geq 4$, $Nu_O$ increases gradually with $H/d$ and reaches a maximum at $H/d = 6$, for curves 1 and 2. A downstream shift of the maximum $Nu_O$ position from $H/d = 6$ to $H/d = 8$ is seen. This could be due to an increase of the potential core length with increasing Reynolds number. The curves 3, 4 and 5 of Fig. 4.1 indicate that as mass flow rate of air increases from 10 LPM to 29 LPM, $Nu$ increases. It seems at $H/d = 6$, (probably an optimum value of $H/d$ ratio), the heat transfer coefficient rates are maximum. Possibly for each values of $H/d$, stagnation zone area near the component could be high. For higher $H/d$ ratio, the effective air quantity striking the component could be low.

The variation of the local Nusselt number with the dimensionless radial distance, $(r^*)$ is plotted in Fig. 4.2 for different types of nozzles at a constant $Re_d = 10500$ and $H/d$ of 6. It is observed that the distribution of the local Nusselt number along the radial distances for the square and circular nozzle are almost same and
marked by the beginning of the transition region at \( r^* = 1.5 \). The transition region extends up to an \( r^* = 2.5 \). The reason for this can be explained as follows. The mass flow rate of air is same in circular and square nozzles because of their equivalent diameters are same. For rectangular nozzle, equivalent diameter is higher than the circular and square nozzle. Hence, the Nusselt number values in the stagnation region remain same almost up to \( r^* = 0.2 \) and thereafter decreases monotonically. It can also be observed from Fig. 4.2 that in the stagnation region, the Nusselt numbers are higher for the rectangular nozzle compared to the square and the circular nozzles. This may due to the rectangular section being more effective in inducing turbulence in the flow.

The heat transfer data from the present case at radial locations is compared with the experimental data of previous researchers, available in the literature in Fig. 4.3. The results are obtained for \( T_s = 950 \text{ C} \), \( \text{Re} = 23000 \), and \( H/d = 6 \). It is observed from Fig. 4.3 that the results at the present experiments are in good agreement with the available experimental data. It is also observed from Fig. 4.3 that the local Nusselt number decreases from the surface of the electronic component with increase in radial distance. This could be due to the decrease in the air velocity with radial distance.

Fig. 4.4 shows the distribution of normalized mean velocity with the dimension less radial distance, \((r^*)\) for the circular nozzle. The plot is made for: surface temperature, \( T_s= 950\text{C} \), \( H/d \) ratio of 2, 4, 6 and 8, and ambient temperature, \( T_a = 30\text{C} \). It is observed from Fig.4.4, that
the shear flow region developed from the edge of the nozzle is expanding due to the mixing of jet flow and ambient air. It is also observed from Fig. 4.4, for H/d = 6, the effect of entrainment penetrates up to the center line when the jet flow develops to the downstream with the increase in the distance between tip of nozzle to surface of the resistor. Ambient flows are entrained up to the potential core region of the jet flow and the uniform distribution of velocity that occurred at the nozzle exit becomes a bell shape.

The effects of the system parameters Re_d, H/d ratio, and surface temperature T_S, on the dimensionless pressure distribution, C_p*(ΔP/(ρU_0^2/2)), are shown in Fig.4.5 as a function of the dimensionless radial distance, (r*). The common parameters, chosen for the plot are: T_S = 90°C, and H/d = 4. Fig.4.5 indicates that as the ratio r* increases the fluid velocity decreases, and the strength of the sub-atmospheric region reduces. Fig.4.6 shows the variation in local heat transfer coefficient along radial distance, r* for a constant Reynolds number of 15,500. Experimental results are obtained for H/d ratio of 2, 3, 4, 5, 8 and 10 with the nozzle diameter of 5mm. It is observed from Fig.4.6 shows the stagnation heat transfer coefficient increases with an increase in H/d ratio from 1 to 2.5. The reason for this can be that, at H/d = 1, the magnitude of the secondary peaks exceeds that of the stagnation point. It is also observed that increase in H/d ratio from 1 to 2, decreases the magnitude of the secondary peaks and they move radially outwards from the stagnation point.
Fig. 4.7 presents the effect of temperature on cooling time at different Re and H/d ratios. Experimental data are obtained for $T_s = 98^\circ C$, $Re_d = 7965$, and $27594$, and H/d ratios of 4 and 10. It is observed that the surface temperature of the resistors drop down rapidly in approximately 50 seconds from the time of starting of the air flow. It is also observed from Fig. 4.7 that, the temperature gradient is higher at larger values of Reynolds number and lower values of H/d ratios. The rapid decrease in temperature is also due to large temperature potential between the surface and the ambient. Fig. 4.7 also indicates that the response time is practically independent of Reynolds number and H/d with in moderate limits.

One of the important outcomes of the present work, is the influence of the system parameters on the surface heat flux ($q$), is shown in Fig. 4.8. Experimental results are obtained for values of $Re_d = 5850, 7325, 10000$ and $12200$ and H/d ratios of 2, 4, 6 and 10. It is observed from Fig.4.8, that the difference between the temperature of surface and ambient lies between $20^\circ C$ to $54^\circ C$ for mass flow rate of 8 LPM (plot 1) and varies from $28^\circ C$ to $63^\circ C$ (plot 2) at mass flow rate of 14LPM. It is also noted that the heat flux $q$, increases with an increase in ($Ts-Ta$). The plots 3 and 4 of Fig.4.9 indicate that the heat flux $q$ increases linearly from $16^\circ C$ to $60^\circ C$ for H/d ratio of 6 and from $15^\circ C$ to $62^\circ C$ for H/d ratio of 10 respectively. The importance of H/d ratio or the location of nozzle with respect to the electronic component is brought out in Fig. 4.8.
The effect of jet Reynolds number on stagnation point (i.e., \( r^* = 0 \)) Nusselt number, is presented in Fig. 4.9. The Stagnation point Nusselt number increases remarkably with jet Reynolds number. In Fig. 4.9, some of the equations for jet flow heat transfer often referred to in the literature are plotted together with the present correlation. The present equation agrees very well with that of Ma and Bergles \(^{74}\) (1990) and Lienhard et al \(^{65}\) (1992), the agreement is with in 10%.

The local Nusselt number computed on the surface of the electronic components are shown in Fig. 4.10, which shows satisfactory agreement with the theoretical data of Lytle and webb \(^{66}\) (1994) and Gao et al \(^{44}\) (2003). The agreement is within 10%. The minor discrepancy can be attributed to the difference in the experimental conditions.

The present experimental data for stagnation Nusselt number, \( \text{Nu}_0 \) as a function of \( H/d \) ratio. The nozzle to electronic device to nozzle diameter with different values of Reynolds number is compared with the numerical data of Gardon and Akfirat \(^{38}\) (1965) and Zhou and Lee \(^{105}\) (2007) available in the literature shown in Fig. 4.11. The present experimental results are obtained for \( \text{Re} = 5850 \), for a nozzle diameter of \( d = 8 \text{mm} \). It is observed from Fig. 4.11 that the present experimental results are qualitatively in good agreement with Gardon and Akfirat \(^{38}\) (1965) and Zhou and Lee \(^{105}\) (2007). The difference in the numerical values can be attributed to difference in nozzle diameters and flow velocity. It is observed from Fig. 4.11 that the decrease in the
stagnation heat transfer rate from electronic components with decrease in nozzle-to-electronic device spacing.

The present experimental work is validated with correlation for stagnation point Nusselt number of Lytle and Webb\textsuperscript{66}(1994) in Fig. 4.12. They conducted experiments with very low nozzle-to-plate spacing to find the local heat transfer distribution on a flat plate impinged by a circular air jet from a long pipe nozzle. The system allowed for fully developed flow at the nozzle exit. They obtained data for stagnation Nusselt number at two different nozzle-to-plate spacings, viz., $H/d = 1$ and $H/d = 0.25$ which are shown in Fig. 4.12. Experimental results are picked up from the present study for stagnation Nusselt number, $\text{Nu}_0$, for the same conditions of Reynolds number (Re) and nozzle-to-plate spacing. The stagnation Nusselt numbers calculated at the middle of test section and plotted in Fig. 4.12, which show satisfactory agreement with correlation of Lytle and Webb\textsuperscript{66}(1994).

The effects of the system parameters $H/d$, diameter of the different nozzles and Reynolds number on the local Nusselt number, Nu are presented in Fig. 4.13 as a function of the dimensionless radial distance ($r^*$). The experimental results are obtained for the chosen common parameters of Reynolds numbers of 6000 and 12500, and nozzle-to-resistor spacing, $H/d = 2, 4, 8$ and 10. From the experimental results the local Nusselt number are estimated for different types of nozzle, viz., circular, square and rectangular nozzles. It is observed from Fig. 4.13 that the local Nusselt number of the jet
array increases with an increase in nozzle-to-resistor spacing (H/d) and decrease in dimensionless radial distance (r*). It is also observed that the local Nusselt numbers for rectangular nozzle are higher than those for circular and the square nozzles. Another observation from Fig. 4.13 is that the heat transfer rates start reducing drastically after H/d = 8. For all types of the nozzles, the local Nusselt number is significantly less after H/d = 10. This indicates that the importance of maintaining proper distance between the component and the nozzle for best performances.

The effects of the dimensionless distance (H/d) on Nusselt number for the different Reynolds numbers with different nozzles, viz., circular, square and rectangular are presented in Fig. 4.14. The results shown in these figures correspond to d = 8, 11.28, 13.3 mm and Ts = 95°C. Fig. 4.14 indicates that stagnation point Nusselt numbers increase with decrease in dimensionless distance between nozzle-to-resistor spacing (H/d). It also shows that heat transfer coefficient increases with lower nozzle-to-resistor spacing, due to the acceleration of the fluid through the gap between target plate and nozzle exit. There exits an ideal spacing H/d for each type of nozzle for best effectiveness. It is advantages to determine this value of H/d before the cooling system is designed. Ideal value of H/d depends on the type of nozzle and diameter.

Experimental results for three jet Reynolds numbers ranging from 8500 to 23000 and variation of stream wise Nusselt number divided by Re^n (n = 0.32 and 0.52) are presented in Figs. 4.15 and
The results shown in these figures correspond to H/d ratios of 4 and 6, and nozzle diameter of 8mm. Fig.4.15 shows that there are two distinct flow regimes concerning the Reynolds number dependence. For H/d = 4, the Nusselt number variation in the region corresponding to $r^* < 0.5$ are well correlated with a Reynolds number power of $n = 0.32$. It is observed that the flow in the stagnation point region is laminar. The Nusselt number variations in the wall jet region corresponding to $r^* \geq 1$ are well correlated with a Reynolds number power of $n = 0.52$, is shown in Fig. 4.16. It can be observed that the wall jet flow is in turbulent boundary layer. As expected, increase of the entrainment of the surrounding air to the jet flow has affected the heat transfer rate at the stagnation region.

The effects of the system parameters Re, H/d ratio and different diameters of circular nozzle are shown in Fig.4.17 as a function of dimensionless radial distance($r^*$). The common parameters for the plot are $T_s = 95^\circ C$, H/d = 2. Fig.4.17 indicates that the heat transfer coefficient decrease with increase of nozzle diameter. It is also observed that the heat transfer coefficient value is more in smaller diameter of nozzle as compared to the higher diameter of nozzles.

To establish the validity of the present correlation, in Fig. 4.18 some of the correlations often referred to in the literature are picked up and shown plotted for H/d = 4 for air (Prandtl number = 0.71). The present equation agrees very well with that of McMurry et al\textsuperscript{71}(1966) and that of Kendoush\textsuperscript{54}(1994). On the same plot, the correlation such as Zumbrunnen and Aziz\textsuperscript{104}(1993) is also shown. Their values
substantially differ from the present correlation. This is possibly due to the large difference in the diameter of the nozzle (d=20mm) used by them. However the qualitatively agreement is good.

The variation of stagnation Nusselt number with heated surface area –to-jet diameter ratio, W/d are shown in Fig.4.19(a) and 4.19 (b). Experimental results are obtained for H/d = 3 and 5, d = 8mm, Re = 5500, 10000 and 23000, and q = 5W/m². It is observed from Fig. 4.19a and 4.19(b), that the stagnation Nusselt number decreases with increase in heated surface area –to-jet diameter ratio. It can also be observed that the effect of recirculation of the fluid at higher H/d ratio, resulting in degradation of the heat transfer rate at the stagnation point.

Fig. 4.20 shows the experimental data along with the correlation for circular nozzle with diameter 8mm for different Reynolds numbers. The figure shows that the data points cluster along the solid line with uniform scatter on either side of the thick line. Nevertheless, the average deviation is ±8% with a standard deviation of ±10%.

The stagnation Nusselt number, calculated using the correlation presented by Kendoush\textsuperscript{54}(1998), and the Nusselt numbers from the present experimental investigations is plotted in Fig.4.21. It is observed that the present values are about 10% more then the values from the correlation. In Fig.4.21, The coefficient C, and exponent’s m and n are calculated from experimental data, i.e C=1.38, m=0.46 and n= 0.35 This can be attributed to the differences in the
geometry of the nozzle used in his investigation. Another reason can be that his correlation is more versatile.

In Fig. 4.22, the present experimental data subjected to the regression eq. (3.88) are plotted, valid in range \(2 < H/d < 10\), and \(6500 < Re_d < 23000\). The average deviation is \(\pm 9.8\%\) and standard deviation is \(\pm 12\%\). However from Fig. 4.22, it can be seen that the data of stagnation Nusselt number fairly agrees well with the experimental stagnation Nusselt number.

In Fig. 4.23, the correlation of Vader et al\(^{96}(1991)\) is compared with experimental data. The heat transfer coefficient from the correlation is plotted against the present experimental values. The whole range of data is found to lie above the benchmark line. It can be seen that the scatter of the data is relatively less. However, in the correlation \(H/d\) is not a parameter. In Fig. 4.23, The coefficient \(C\), and exponent’s \(m\) and \(n\) are calculated from experimental data, i.e \(C=1.296\), \(m=0.4\) and \(n=0.4\). The correlation looks to be inadequate to fully represent heat transfer phenomenon.

The experimental values of the recovery factor \((r_f)\) are plotted as a function of the dimensionless radial distance \(r^*\) with \(H/d\) of 2, 4, 6 and 8 in Fig. 4.24. The Reynolds number of the jet is chosen as 23000, diameter of the nozzle is 8mm and surface temperature of the electronic component is taken as 90°C. The apparently strange variation of the recovery factor with \(H/d\) can be attributed to the influence of atmospheric air on the jet. It is also observed that the abrupt change in the recovery factor is occurring at \(4 < H/d < 6\).
Apparently, this is occurring at the optimum H/d ratio where it was found that the heat transfer is maximum.

The heat transfer rates calculated from the experimental data for different nozzles, viz., circular, square and rectangular are compared with theoretical results as shown in Fig.4.25. Experimental results are obtained for Re_d = 6800, H/d = 4, T_S = 950°C. It is observed from Fig.4.25 that the heat transfer rate of a rectangular nozzle is more, as compared with the circular and square nozzle. It is observed from Fig.4.25 that the agreement between the experimental and theoretical results is found to be good.

The Variation of effectiveness with dimensionless radial distance from a circular nozzle is shown in Fig.4.26. Experimental results are obtained for this purpose for H/d ratio of 2, 4, 6 and 10 and Reynolds numbers, Re = 23000. It is observed from Fig.4.26 that the effectiveness is higher for lower H/d ratio. This is due to the better exposure of the component to air when the component is near to the nozzle. It is also observed from Fig.4.26 that the effectiveness is maximum at r* = 0.

Fig.4.27 shows that the variations of heat transfer rate with temperature difference for different Reynolds number. Experimental data are obtained for Re = 5850, Re = 7325 and Re = 10000. As expected from Fig.4.27 that the heat transfer rate increases with increase in Reynolds number. It is also observed from Fig.4.27 that there is a linear relation between the temperature difference and heat
transfer rates. This is due to the fact that the property variations for air for moderate temperature differences are negligible.

The jet Reynolds number Re, verses normalized velocity, \( \frac{U_c}{U_e} \) is plotted in Fig.4.28 as a function of the dimensionless distance, \( \frac{H}{d} \) ratio. It is observed from Fig.4.28 that the normalized velocity decreases gradually with increase in nozzle-to-resistor spacing. It is seen that, qualitatively, normalized velocity decrease is independent of the jet Reynolds number. It is also observed from Fig.4.28 that the normalized velocity is higher for the jet Reynolds number=13000 in comparison to Re = 7000 till the nozzle-to-resistor spacing is about 6. Afterwards, the trend changes due to mixing with the surrounding air. It is inferred that \( \frac{H}{d} \) of 5 to 6 is an optimum distance for heat transfer, as the influences of the surrounding air is minimum till that point. The value of normalized velocity at this point is about 0.9.

The variation of heat transfer for different types of nozzles viz., circular and rectangular is shown in Fig.4.29 with temperature difference for Re = 6500 and \( \frac{H}{d} \) ratio of 2. It is observed from Fig.4.29 that the rectangular nozzle is more effective from heat transfer point of view. It is also observed that for the same Re and \( \frac{H}{d} \) ratios of circular and rectangular nozzles, the rectangular has more effective than the circular nozzles, since equivalent diameter of rectangular is higher than the circular there by fluid flow is anticipated over the surface of the component shown in Fig 4.29. Hence, in the design of thermal components like electronic
components better cooling is achieved by rectangular nozzles. This is due to the better exposure of the electronic component to air.

Heat transfer rates for circular nozzles with different H/d ratios, viz., 2 and 6 for Re = 5850 is plotted in Fig.4.30. It is observed that the heat transfer rates are independent of the surface temperature difference till (T_s-T_a) of 40^\circ C. Afterwards the heat transfer rates are more. This is due to H/d ratio of 6 is the optimum for heat transfer rates. It is also observed that the heat transfer rates increases linearly with the temperature difference for all H/d ratios.

The experimental results of the present experimental investigations deviate from the existing literature to an extent of maximum of ±10%. The uniqueness of the present work is that the experiments are conducted on the actual components rather than simulated in-service condition. Therefore it is expected that the results of the present investigation are more closer to real world problems.

From the review of several correlations, it is observed that there is a lack of efficacy to predict heat transfer close to the experimental data in relation with different parameters. This investigation is taken up to rationalize the often contradicting experimental results in this area and fill in the technical gaps left out by previous investigators. It is expected that this work will pave the way for effectively using the heat transfer rate, heat flux and heat transfer coefficient results for the thermal design of electronic cooling devices. Moreover, it is also found that the effect of different types of nozzles impinging on the electronic component directly has not been studied by any
investigator. The results and correlations from the present investigation could possibly result in development of more efficient impingement cooling systems for electronic components.