SUMMARY AND CONCLUSIONS.

In all chemical industries the heat transfer operation is of controlling importance and the primary objective lies in achieving the maximum heat transfer rate per unit surface area compatible with economic factors. This has led to studies on the improvement of the performance of heat transfer equipments. It is well known that the major resistance to heat transfer between solid boundaries and turbulent fluids is the laminar sub layer adjacent to the wall. The resistance of this laminar layer is proportional to its thickness and any reduction of the thickness will result in a comparable increase in the rate of heat transfer between the wall and the fluid. All the different techniques developed for augmentation of heat transfer coefficient are mostly based on this principle.

Among the different augmentation techniques namely,turbulence promoters, vibration or pulsation technique, fluid-solid suspension technique, injection and suction technique etc., the use of turbulence promoters has received considerable attention. Studies on the effects of turbulence promoters, usually in the form of twisted tapes or spiral wires have been reported by many researchers. All these results are in no way general, but they serve to indicate at least semiquantitatively the effect of turbulence promoters on heat transfer. The improvement in heat transfer operation achieved by this technique is mostly outbalanced because of the correspondingly higher power loss.

The use of jet mixing phenomena in creating high turbulence and intense mixing between the fluids is well known. The effects of this mechanism on mass and momentum transfer have been very well emphasised and the results were highly encouraging. Since the momentum of the jet coming out of the nozzle is being utilised in mixing phenomena by creating turbulence and continuous renewal of the transfer surface, it was thought that this mechanism may be effective in the efficient transfer of heat as well. So the present investigation has been undertaken to study the heat transfer in a horizontal cylindrical tube using jet mixing phenomena. Studies have been carried out under three categories namely, studies on heat transfer with jet mixing - with no secondary flow, studies on heat transfer with jet mixing - with secondary flow of water, and studies on two-phase heat transfer with jet mixing with secondary flow of air.

The thesis has been presented in the following five chapters and two appendices.

Chapter-I : Introduction.

In this chapter a comprehensive review of literature on the different techniques of augmenting the heat transfer operation has been presented. The feasibility of these techniques has been discussed from operational and economic point of view. Literature review on the effect of jet mixing phenomena on momentum transfer operations has also been summarised and presented. Lastly, the scope of the present investigation has been stressed.

Chapter-II : Mathematical approach to the problem

In this chapter an attempt has been made to analyse theoretically the temperature distribution in the system and then predict the heat transfer coefficient. For the solution of the problem the basic energy equation for fully developed turbulent flow has been utilised. The equation on simplification leads to

$$\bar{u} \frac{\partial Tr}{\partial x} = \frac{1}{r} \frac{\partial}{\partial r} \left[r(\epsilon_H + \alpha) \frac{\partial Tr}{\partial r} \right] \qquad ..(2.1)$$

and the temperature distribution satisfying equation (2.1) for the case of constant wall temperature is the solution really desired.

The solution of equation (2.1) requires that the velocity distribution for the system must be known. In the present system a jet is impinged parallely on a moving fluid and thereby imparts its momentum for intense mixing and continuous renewal of the boundary layer. The mathematical analysis of the system is highly complicated and little work has been reported in this field. Viktorin(187) carried out an analytical investigation of turbulent mixing process in a simple jet apparatus similar to the present one and has developed the following expression for velocity distribution:

$$\bar{u} = u + u''$$

$$= u + \frac{u\left(\frac{J}{\rho u^2}\right)^{1/3}}{x^{2/3}} \left[\frac{1}{3\sqrt{3}\, k_o} \left\{ \frac{r}{\left(\frac{J}{\rho u^2} \cdot x\right)^{1/3}} \right\}^{3/2} - 0.448\, k_o^{-2/5} \right]^2$$

..(2.7)

This expression for $\bar{u}$ can be applied in equation(2.1) since the present system is very much similar to that of Viktorin. Then the equation (2.1) has been solved by applying iterative technique in which the generalised temperature distribution developed by Martinelli (121) for the case of constant heat flux at the wall has been used as the first approximation.

The following expression for the temperature distribution has been finally found to converge and therefore, it gives the desired temperature distribution for the system :

$$\frac{Tw - Tr}{Tw - Tc} = 1 - \frac{\psi(\frac{r}{r_o})}{\psi(1)} \qquad ..(2.35)$$

where,

$$\psi(\frac{r}{r_o}) = \int_0^{r/r_o} \frac{\phi(\frac{r}{r_o})\, d(\frac{r}{r_o})}{(\frac{r}{r_o}) \cdot (\frac{\epsilon_H + \alpha}{\nu})} \qquad ..(2.15)$$

and $\psi(1) = \psi(\frac{r}{r_o})$ at $r = r_o$.

The function $\psi(\frac{r}{r_o})$ may be obtained from the following equation:

$$\frac{Tr - Tc}{Tw - Tc} = \frac{r_o^2 \, u \, \frac{\partial Tm}{\partial x}}{\nu \, (Tw - Tm)} \psi(\frac{r}{r_o}) \qquad ..(2.16)$$

Finally on evaluation the function $\psi(\frac{r}{r_o})$ becomes,

$$\psi(\frac{r}{r_o}) = A \left[\frac{1}{2} R^2 r_o^2 (\frac{r}{r_o})^3 - RPr_o^{1/2} (\frac{r}{r_o})^{3/2} \right]$$

$$+ A' \left[\frac{1}{10} R^4 r_o^5 (\frac{r}{r_o})^6 - \frac{2}{15} R^3 Pr_o^{7/2} (\frac{r}{r_o})^{9/2} + \frac{1}{2} R^2 P^2 r_o^2 (\frac{r}{r_o})^3 - RP^3 r_o^{1/2} (\frac{r}{r_o})^{3/2} - \frac{8}{21} RPr_o^{7/2} (\frac{r}{r_o})^{9/2} + \frac{4}{7} R^2 P^2 r_o^2 (\frac{r}{r_o})^2 \right]$$

..(2.33)

where,

$$A = \left[- \left(\frac{Q}{x^{2/3}}\right) \frac{\mu}{\tau_w \, g_c} \right], \quad A' = \left[- \left(\frac{Q}{x^{2/3}}\right)^2 \frac{1}{u} \frac{\mu}{\tau_w \, g_c} \right]$$

$$Q = u N^{1/3}, \quad N = \frac{J}{\rho u^2}, \quad R = \frac{1}{3\sqrt{3Nx}\, k_o},$$

$$P = 0.448\, k_o^{-2/5} \quad \text{and} \quad k_o = \text{constant.}$$

Theoretical prediction of heat transfer coefficient :

The rate of heat transfer per unit length of the system may be written as :

$$q = u \frac{\pi}{4} D^2 \rho\, Cp \frac{\partial Tm}{\partial x} = h \pi D (Tw - Tm) \qquad ..(2.36)$$

which may be rearranged to

$$\frac{u \frac{\partial Tm}{\partial x}}{Tw - Tm} = \frac{4h}{D \rho\, Cp} \qquad ..(2.37)$$

Combining the equations(2.16) and (2.37) and rearranging the terms one gets,

$$\frac{h D}{K} = \frac{\delta}{\alpha} \; \frac{Tr - Tc}{Tw - Tc} \; \frac{1}{\psi\left(\frac{r}{r_o}\right)} \qquad ..(2.38)$$

The function $\psi\left(\frac{r}{r_o}\right)$ when evaluated from equation (2.16) at $\frac{r}{r_o} = 1$ gives,

$$\psi(1) = \frac{\delta \, (Tw - Tm)}{r_o^2 \, u \frac{\partial Tm}{\partial x}} \qquad ..(2.19)$$

Further it may be seen from equations(2.16) and (2.19) that

$$\frac{Tr - Tc}{Tw - Tc} = \frac{\psi\left(\frac{r}{r_o}\right)}{\psi(1)} \qquad ..(2.39)$$

and therefore, the equation (2.38) finally leads to

$$\frac{h\,D}{K} = \frac{Pr}{\Psi(1)} \qquad ..(2.40)$$

Chapter-III : Studies on heat transfer with jet mixing — with no secondary flow.

This chapter describes elaborately the experimental set-up and procedure and presents discussion on the results of the investigation.

The experimental set-up consists of a nozzle chamber, a horizontal test-section and other accessories. The nozzle chamber essentially consists of a cylindrical horizontal chamber. It is provided with a secondary fluid inlet and a parallel discharge.The discharge end of the chamber is well rounded and is connected to the horizontal cylindrical test-section. The nozzle used was of straight hole type and was fixed in position by means of a spindle. Five different nozzles of diameter in the range of 0.4366 cm to 0.1588 cm have been used.

The test-section is essentially a smooth copper tube of 2.54 cm i.d. and 0.158 cm wall thickness and 228.60 cm in length. It was aligned horizontally with the nozzle chamber and arrangements were made for uniform and controlled electrical heating of the test-section. Heat transfer studies have been carried out at constant and uniform wall temperatures varying in the range of 27°C to 50°C.

The heat transfer coefficient, hw_s for the present system has been calculated on the basis of heat balance over a control volume in the test-section and finally the following expression for hw_s has been obtained

$$hw_s = \frac{K_L \; Ret_s \; Pr_s}{4x} \quad \ln \frac{Tw - Ti_L}{Tw - T_s} \qquad ..(3.9)$$

Experimental observation on the variation of fluid temperature along the length of the test-section shows that the rise of temperature is very rapid within a small x/D and then it becomes gradual with increase in x/D. The hw_s values calculated from equation(3.9) have been found to decrease monotonically with increase in x/D.

The heat transfer coefficient, hw_s was also calculated from the theoretical equation(2.40) derived in Chapter-II and was found to agree satisfactorily with the experimental values. The deviation being within $\pm$ 18 percent.

In order to find the augmentation in heat transfer coefficient by jet mixing, the hf_s values of heat transfer coefficient for ordinary tube flow without jet mixing for the same fluid flow rate and wall to bulk temperature difference have been calculated from the Sieder-Tate relation :

$$J_H = \frac{hf_s \; D}{K_L} \left(\frac{Cp_L \; \mu_L}{K_L} \right)^{-1/3} \left(\frac{\mu_L}{\mu_W} \right)^{-0.14} \qquad .. (3.12)$$

The hw_s values have been found to be much higher than the corresponding hf_s values.

The augmentation ratio, $\frac{hw_s}{hf_s}$ was found to decrease with increase in x/D and it varied from a maximum of 11.11 at x/D=18 to a minimum of 1.13 at x/D = 90 in the experimental range covered.

In view of the complexity of the system an attempt has been made to find out the effects of physical and dynamic variables as well as the geometry of the contactor on the augmentation ratio. For this purpose dimensional analysis has been made and the following generalised equation has been obtained

$$\frac{h_s}{hf_s} = C_1 \, (d_N)^{-b-c+d+e-f+2g} \, (D)^b \, (d_e)^c \, (u_n)^d \, (u')^e \, (x)^f \, (\Delta P)^g$$
$$\times \, (\rho_L)^{d+e+g-i} \, (\rho')^i \, (\mu_L)^{-d-e-2g+\ell} \, (Cp_L)^{\ell} \, (K_L)^{-\ell} \, (g_c)^g \qquad ..(3.27)$$

For the present system the general equation (3.27) on rearrangement and simplification leads to

$$\frac{hw_s}{hf_s} = C_3 \, (Re_n)^{m_6} \, (Pr_s)^{m_2} \, \left(\frac{x}{D}\right)^{m_3} \, (A_R)^{m_4} \qquad ..(3.31)$$

The exponents of the different groups in the above equation has been determined graphically and finally the following generalised correlation for the system is obtained

$$\frac{hw_s}{hf_s} = C_3 \, (Re_n)^{-0.64} \, (Pr_s)^{7.33} \, (A_R)^{0.21} \, \left(\frac{x}{D}\right)^{-0.36} \qquad ..(3.33)$$

The value of C_3 obtained, is 0.1862 and the correlation coefficient for the above equation is of the order of 0.83.

Chapter-IV : Studies on heat transfer with jet mixing — with secondary flow of water.

When the jet issues from the nozzle, suction is created due to Bernoulli effect. In the previous chapter studies have been reported without secondary flow but in the present case suction has been utilised to suck a secondary fluid in the system. This will ensure utilisation of unused suction pressure as well as improvement in heat transfer coefficient for the same jet momentum.

The chapter describes the experimental procedure along with the different parameters measured. The jet Reynolds number ranged between 2.5×10^4 to 9.3×10^4 during the experiemnt.

Heat transfer coefficients, hw'_s have been calculated from equation(3.8) and the hf_s values for ordinary tube flow at the same Reynolds number have been calculated from equation(3.12).

The theoretical equation(2.40) derived in Chapter-II has been used to calculate the heat transfer coefficients, hw'_s and these have been found to be in agreement with the experimental values. The deviation being within ± 20 percent.

The augmentation ratio, $\frac{hw'_s}{hf_s}$ varied from a maximum of 8.4 at x/D = 18 to a minimum of 1.01 at x/D = 90 within the range of experiment.

Attempt has been made to propose generalised correlation for $\frac{hw'_s}{hf_s}$ using equation (3.27). For the present system the generalised equation (3.27) on rearrangement and simplification gives,

$$\frac{hw'_s}{hf_s} = C_5 \, (Eu'_L)^{b_{10}} \, (Pr_s)^{b_2} \, (A_R)^{b_9} \left(\frac{x}{D}\right)^{b_5} \qquad ..(4.5)$$

Using experimental data, equation (4.5) has been solved graphically and is represented by

$$\frac{hw'_s}{hf_s} = C_5 \, (Eu'_L)^{-0.058} \, (Pr_s)^{7.43} \, (A_R)^{0.526} \left(\frac{x}{D}\right)^{-0.53} \qquad ..(4.7)$$

The constant, C_5 is equal to 1.10×10^{-5} and the correlation coefficient of the equation (4.7) was found to be of the order of 0.89.

Chapter-V : Studies on two-phase heat transfer with jet mixing with secondary flow of air.

Research investigations on non-isothermal two-phase flow of air and water in cylindrical pipes have been reported in literature and improved heat transfer coefficient with respect to single phase flow of liquid have been claimed. In the present case attempt has been made to study the effect of jet mixing on two-phase heat transfer operation.

A brief literature survey on two-phase heat transfer has been presented in this chapter.

In the previous chapter studies have been reported with secondary flow of water but the present chapter deals with two-phase heat transfer studies with jet mixing while there is a secondary flow of air. Minor modifications made over the set-up and the experimental technique followed to measure the different quantities, have been thoroughly reported. Experimental data have been collected for air and water flow rates within the range of 0.218 to 1.842 cft/min and 648 to 1958 lbs/hr. respectively.

The two-phase heat transfer coefficient has been calculated from the equation (5.9) developed from macroscopic heat balance over a control volume.

$$hTP_s = \frac{K_L \, Ret_s \, Pr_s}{4x} \ln \frac{Tw - Ti_L}{Tw - Ts} + \frac{M_G' \, Cp_G}{\pi \, Dx} \ln \frac{Tw - Ti_G}{Tw - T_s} \quad .. (5.9)$$

In order to find the augmentation of the two-phase heat transfer coefficient by jet mixing, the hf_s values of heat transfer coefficient for ordinary tube flow without jet mixing for the same liquid flow rate and wall to bulk temperature difference have been calculated from Dittus-Boelter equation.

As in previous two cases it is seen that the temperature of the air-water mixture rises rapidly over an x/D of 54. Therefore, all the heat transfer calculations have been made over this length of the test section.

The augmentation in heat transfer coefficient has been found to be greater than that in previous cases. Again the hTP_s is found to increase with air flow rate upto a certain limit and then decreases as air flow rate increases. Similar observations have been reported by different researchers.

Further an attempt has been made to correlate the augmentation ratio, $\frac{hTP_s}{hf_s}$ with the different system variables. The general equation (3.27) on rearrangement gives the following equation for the present system

$$\frac{hTP_s}{hf_s} = C_6 \, (Re_n)^{a_1} \, (Pr_s)^{a_2} \, (Eu_G')^{a_3} \, (A_R)^{a_9} \, \left(\frac{u_G'}{u_n}\right)^{a_8} \quad .. (5.15)$$

Experimental data were used to solve the equation(5.15) and the different exponents were determined graphically. The final correlation obtained for the present system is presented as

$$\frac{hTP_s}{hf_s} = C_7 \, (Eu_G')^{-0.3} \, (Pr_s)^{7.3} \, (A_R)^{-0.57} \qquad ..(5.18)$$

The value of C_7 obtained, is 0.124 and the correlation coefficient for the equation (5.18) was found to be of the order of 0.81.

Appendix-I : Theoretical evaluation of heat transfer coefficient with jet mixing - with no secondary flow.

The heat transfer coefficient, hw_s can be evaluated theoretically from equation (2.40) derived in Chapter-II. This appendix gives a complete calculation procedure and presents the sample calculation for a typical experimental run.

Appendix-II : Theoretical evaluation of heat transfer coefficient with jet mixing - with secondary flow of water.

The heat transfer coefficient, hw_s' may be theoretically calculated as before, from the equation (2.40). Sample calculation for a typical experimental run has been shown in this appendix.