Effect of fouling on thermal and hydraulic parameter of Shell and Tube Heat exchanger

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Abstract: Fouling is inevitable phenomenon in shell and tube heat exchanger. Even regular water has considered as working fluid in shell and tube heat exchanger, it has significant impact on thermal and hydronic performance due to fouling properties. Amount of layer deposition has direct impact on heat transfer and pressure drop, as it affect on physical dimension of shell and tubes of heat exchanger. The heat transfer surface fouls during operation, resulting in increased thermal resistance and often an increase in the pressure drop and pumping power. It is required to understand, how mass flow rate can affect on deposition rate of fouling layer inside the tube. In this present paper, the study has been conducted to evaluate optimal operating condition that lead to lower fouling deposition rate inside the heat exchanger while keeping in mind thermal and hydraulic requirements.

Keyword: Fouling; Heat Transfer coefficient; Pressure drop; Pumping Power; Effect of fouling.

1. Introduction

Heat Exchanger is device, which exchange the flow of thermal energy between two or more fluids by convection mainly at different temperatures. The main problem caused by fouling is related to the fact that the accumulation of fouling deposits has an important effect on the thermal and hydraulic performance of the affected equipment. Heat exchangers are used in a wide variety of application. This includes power engineering, waste heat recovery, manufacturing industry, air-conditioning, refrigeration and space applications. Shell and tube heat exchangers are most versatile type of heat exchanger; they are used in process industries, in conventional and nuclear power station as condenser, in steam generators in pressurized water reactor power plants, in feed water heaters and in some air conditioning refrigeration systems. Shell and tube heat exchanger provide relatively large ratio of heat transfer area to volume and weight and they can be easily cleaned. Shell and tube heat exchanger offer great flexibility to meet almost any service requirement. Shell and tube heat exchanger can be designed for high pressure relative to the environment and high pressure difference between the fluid streams.

2. Literature review

A detailed study of shell and tube heat exchanger with various mass flow rate and fouling factor has been provided in references. **Thanhtrung Dang et al.**[1] gives idea about theoretical idea of shell and tube heat exchanger heat transfer rate and fouling thickness with two cases, investigated in this study, when the thickness of substrate of the heat exchanger increases, its actual heat transfer rate decreases. However the accumulation of deposits rate describe by **Rima Harcheon et al**[2] which effect the heat transfer surface and finally leads to a reduction of the energy transfer effectiveness.

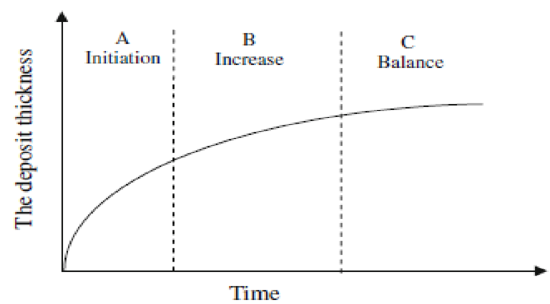

Figure 1*: the deposition thickness* [2]

The first part (A) corresponds to the initiation phase. This period depends largely on the deposit type varies few minutes at a few weeks, for example in the air-conditioning systems. The second part (B) corresponds to the increase in this deposit resulting from a competition between the mechanisms from deposit and wrenching. The fouling rate decreases gradually while wrenching increases, finally, leading that to a balance in (C) and a height fouling held

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constant. **J.W. Suitor et al.**[3] gives idea about theoretical developments by using kern method concluded that fouling resistance increases indefinitely with time. The deposition term is a complex function of velocity, water chemistry, the wall, scale surface and bulk fluid temperature. **P.J. Frayer et al.**[4] demonstrated the design of fouling resistance in which pulsate flow system may be effective in many cases, but that antifouling exchangers should not be designed without understanding the fouling mechanism. **D. Gulley** [5] Studied that, pressure drop is a big help in analyzing performance problems. They also provide a rough check of flow rates. **Liljana Markovska et al.**[6] Studied that, increased fluid velocities result in larger heat transfer coefficients and, consequently, less heat-transfer area and exchanger cost for a given rate of heat transfer. On the other hand, the increased fluid velocities cause an increase in pressure drop and greater pumping power cost. **Kevin M. Lunsford**[7] presented some methods for increasing shell and tube heat exchanger performance. The method considers whether the exchanger is performing correctly to begin with, excess pressure drop capacity in existing exchanger, the re-evolution of fouling factor and their effect on exchanger calculations. **K.C.Leong and K.C.Toh**[8] describes user friendly computer software(HTRI and HTFS) developed for the thermal and hydraulic design of shell and tube heat exchanger based on open literature Bell Delaware method. The use of this software will bridge the gap between engineering practice and teaching of shell and tube heat exchanger design. A rating, sizing and simulation can be performed in this computer software such as HTRI & HTFS. For estimation of optimal operation condition of shell and tube heat exchanger these type of optimization software are very useful. **D.I.Wilson et al.**[9] proposed a semi-empirical approach to quantify the effect of flow velocity on tube-side fouling in crude oils at high temperatures which pilot plant studies indicated that: (i) Fouling rates increased with increasing temperature – initially interpreted as film temperature, elsewhere as wall/deposit temperature. (ii) Fouling rates decreased with increasing flow velocity (figure 1) By using the fouling model and the concept of the fouling threshold fouling rate in heat exchanger can be predicted.

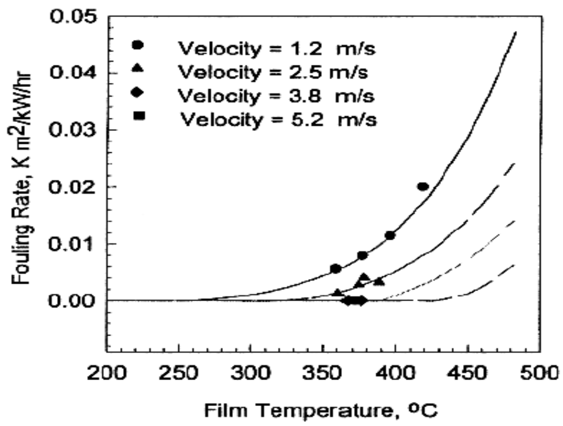

***Figure 2**: The Ebert-Panchal model* [9]

John M. Nesta and Christopher A. Bennett[10] **showed** the fouling layer is a conductive resistance to heat transfer that must be accounted for in the design heat transfer coefficient. Also, Fouling thickness and thermal conductivity both contribute to the resistance. Reduced cross-sectional flow area also increases pressure drop in the fouled region. **M.S. Abd-Elhady et al.**[11] Studied the influence of flow direction with respect to gravity on particulate fouling of heat exchangers is investigated experimentally to determine the optimal flow direction to minimize fouling. Four orientations of flow have been investigated, horizontal flow, upward flow, downward flow and a flow under an angle of 45°. **E.M.Ishiyama et al.**[12].presented an analysis of the thermal and hydraulic impacts of fouling. For fouling rate laws that incorporate the threshold fouling concept, characteristic times for fouling of an exchanger were identified, based on hydraulic or thermal limitations. **D.H. Lister and F.C. Cussac**[13] showed the formulation mechanism for the effect of the bubbles on deposition in water under boiling condition. Paper describes the main mechanisms occurring during boiling and to determine how bubbles influence the deposition of iron oxide particles. Deposition, removal and consolidation of the deposit are included in the model. The relation between heat transfer, bubbles formation near wall and fouling amount is studied. **Su Thet Mon Than et al.**[14] evaluated design data for heat transfer area and pressure drop and checking whether the assumed design satisfies all requirement or not. The primary aim of this design is to obtain a high heat transfer rate without exceeding the allowable pressure drop. **Arturo Reyes Leon et al.**[15] developed relationship between heat transferred and energy loss for turbulent flow and also identified the advantages of having the appropriate exchanger with working conditions, environmental conditions and economical aspects. In addition to thermal design, mechanical design of heat exchanger is also part of it. **Rajiv Mukherjee** [16] developed correlation for optimal condition of shell and tube heat exchanger. The optimized thermal design can be done by sophisticated computer software however a good understanding of the underlying principles of exchanger design is needed to use this software effectively.

2.1 Pressure drop in STHE

From the literature review we had carried out that Pressure drop is one of the important factors which affect the performance of STHE. Two methods have been adopted for calculating the Pressure drop inside the STHE based on name of their formulators 1. Kern ; and 2. Bell Delaware. In this research we have adopted the formulation proposed by Kern.

2.2 Preliminary calculation

A selected shell and tube heat exchanger must satisfy the process requirements with the allowable pressure drops until the next scheduled cleaning of plant. The methodology to evaluate thermal parameters is explained with suitable assumptions.

The following are the major assumptions made for the pressure drop analysis;

1. Flow is steady and isothermal, and fluid properties are independents of time.
2. Fluid density is dependent on the local temperature only or is treated as constant.
3. The pressure at a point in the fluid is independent of direction.
4. Body force is caused only by gravity.
5. There are no energy sink or sources along streamline; flow stream mechanical energy dissipation is idealized as zero.
6. The friction factor is considered as constant with passage flow length.

Heat transfer or the size of heat transfer exchanger can be obtained from equation,

$$Q = U_o A_o \Delta T_m \qquad (1)$$

The overall heat transfer coefficient U_o based on the O.D. of tubes can be estimated from the estimated values of individual heat transfer coefficients, the wall and fouling resistance and the overall surface efficiency using equation (2)

$$\frac{1}{U_o} = \frac{A_o}{A_i}\left[\frac{1}{\eta_i h_i} + \frac{R_{fi}}{\eta_i}\right] + A_o R_w + \frac{R_{fo}}{\eta_o} + \frac{1}{\eta_o h_o} \qquad (2)$$

For the single tube pass, purely countercurrent heat exchanger, F= 1.00. For preliminary design shell with any even number of tube side passes, F may be estimated as 0.9

Heat load can be estimated from the heat balance as:

$$Q = (mC_p)_c (T_{c2} - T_{c1}) = (mC_p)_h (T_{h2} - T_{h1}) \qquad (3)$$

If one stream changes phases:

$$Q = mh_{fg} \qquad (4)$$

LMTD (Log Mean Temperature Difference Method) calculation:

If three temperatures are known, the fourth one can be found from the heat balance,

$$\Delta T_{lm} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} \qquad (5)$$

Heat transfer area can be calculated from equation (1). Number of tubes of diameter (d_o), shell diameter (D_s) to accommodate the number of tubes (N_t), with given tube length (L) can be estimated,

$$A_o = \pi_e N_t L \qquad (6)$$

One can find the shell diameter (D_s), which would contain the right number of tubes (N_t), of diameter (d_t).

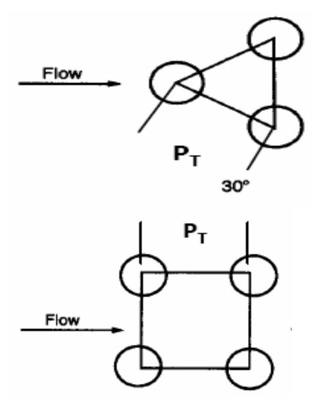

***Figure 3**: Square and Triangular Pitch Tube Layout*[17]

The total number of tubes can be predicted in fair approximation as function of the shell diameter by taking the shell circle and dividing it by the projected area of the tube layout (fig 3) pertaining to a single tube A_1.

$$N_t = (CTP)\frac{\pi D_s^{\ 2}}{4A_i} \tag{7}$$

Where CTP is the tube count calculation constant that accounts for the incomplete coverage of the shell diameter by the tubes.

Based on fixed tube sheet the following values are suggested:

One tube pass: CTP = 0.93

Two tube pass: CTP = 0.90

Three tube pass: CTP = 0.85

$A_1 = (CL)\ (P_T)^2$ (8)

Where CL is the tube layout constant:

CL = 1.0 for 90 and 45

CL = 0.87 for 30 and 60

Equation (7) can be written as:

$$N_t = 0.875\left(\frac{CTP}{CL}\right)\frac{D_s^{\ 2}}{(\mathrm{P_r})^2 d_o^{\ 2}} \tag{9}$$

Where P_R is the Tube Pitch Ratio ($P_R = P_T/d_o$).

The shell diameter in terms of main construction diameter can be obtained as from equations (6) and (9),

$$Ds = 0.637\sqrt{\frac{CL}{CTP}}\left(\frac{A_o(\mathrm{P_r})^2 d_o}{L}\right)^{1/2} \tag{10}$$

2.3 Tube side pressure drop

The tube side pressure drop can be calculated by knowing the number of tube passes (N_p) and length (L) oh heat exchanger,

The pressure drop for the tube side fluid is given by equation

$$\Delta P_t = 4f\frac{LN_p}{d_i}\rho\frac{\mu_m^{\ 2}}{2} \tag{11}$$

$$\Delta P_t = 4f\frac{LN_p}{d_i}\frac{G_i^{\ 2}}{2\rho} \tag{12}$$

The change of direction in the passes introduction in the passes introduction an additional pressure drop due to sudden expansions and contractions that the tube fluid experiences during a return that is accounted for allowing four velocity head per pass

$$\Delta P_t = 4N_p\frac{\rho\mu_m^{\ 2}}{2} \tag{13}$$

The total pressure drop of the side becomes:

$$\Delta P_t = \left(4f\frac{LN_p}{d_i} + 4N_p\right)\frac{\rho\mu_m^{\ 2}}{2} \tag{14}$$

2.4 Pumping power and pressure drop

The fluid pumping power is proportional to the pressure drop in the fluid across a heat exchanger, the equation can give by:

$$P = \frac{m\Delta P}{\rho\eta_p} \tag{15}$$

Where η_p is the pump or fan efficiency. (η_p = 0.80 to 0.85)

The cost in terms of increased fluid friction requires an input of pumping work greater than the realized benefit of increased heat transfer. For gases and low density fluids and also for very high viscosity fluids, pressure drop is always of equal importance to the heat transfer rate and it has a strong influence on the design of heat exchangers.

Pressure drop of shell and tube as well as pumping power can be calculated by these theories. Fouling factor, the effect of fouling on pressure drop and overall heat transfer coefficient estimation are explained in next chapter.

2.5 Effect of fouling on heat transfer

Heat transfer rate can be given by,

$Q = U_c A_o \Delta T_M$ (16)

Where, U is based on outside heat transfer surface area of the exchanger. Fouling coefficient (U_f) can be related to the clean surface overall heat transfer coefficient (U_c)as:

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft} \tag{17}$$

The heat transfer under fouling conditions (Q_f) can be expressed as:

$$Q_f = U_f A_f \Delta T_M \tag{18}$$

Overall heat transfer coefficient based on outer surface area under fouled conditions can be obtained by adding the inside and outside thermal resistances.

$$U_f = \frac{1}{\frac{A_o}{A_i h_i} + \frac{A_o}{A_i} R_{fi} + \frac{A_o \ln\left(\frac{d_o}{d_i}\right)}{2\pi k l} + R_{fo} + \frac{1}{h_o}} \tag{19}$$

2.6 Effect of fouling on pressure drop

There is a finite thickness of fouled surface; causes change in geometry due to fouling affects the flow field and the pressure drop. Fouling layer roughness of surface decreases the Inner Diameter and increases the Outer Diameter of the tubes.

$$\Delta P = 4f\left(\frac{L}{dl}\right)\frac{\rho u_m^2}{2} \tag{20}$$

The preceding dimensionless group involving Δp has been defined as the Fanning friction factor f:

$$f = \frac{\Delta p}{4\left(\frac{L}{d_l}\right)\frac{\rho u_m^2}{2}} \tag{21}$$

$$f = 0.046\,\mathrm{Re}^{-0.2} \quad \text{for } 3\text{x}10^4 < \mathrm{Re} < 10^6 \tag{22}$$

$$f = 0.079\,\mathrm{Re}^{-0.25} \quad \text{for } 4\text{x}10^3 < \mathrm{Re} < 10^5 \tag{23}$$

The fouling layer decreases the inner diameter and roughens the surface, thus causing an increase in the pressure drop given by equation (20). Pressure drop under fouled and clean condition can be related as:

$$\frac{\Delta P_f}{\Delta P_c} = \frac{f_f d_c}{f_c d_f}\left(\frac{u_f}{u_c}\right)^2 \tag{24}$$

By assuming that the mass flow rate ($m = u_m \rho A$) under clean and fouled conditions is the same, equation (24) can be modified as:

$$\frac{\Delta P_f}{\Delta P_c} = \frac{f_f}{f_c}\left(\frac{d_c}{d_f}\right)^2 \tag{25}$$

The fouling factor can be related to the fouling thermal conductivity k_f and the fouling thickness t_f as:

$$R_f = \frac{t_f}{k_f} \text{ (For a plane wall)} \tag{26}$$

$$R_f = \frac{d_c \ln\left(\frac{d_c}{d_f}\right)}{2\pi k_f} \text{ (For a cylindrical tube wall)} \tag{27}$$

Inner diameter under fouled conditions, d_f, can be obtained by rearranging equation (27):

$$d_f = d_c \exp\left(\frac{-2\pi k_f R_f}{d_c}\right) \tag{28}$$

And the fouling thickness, t_f, is expressed as:

$$t_f = 0.5 t_c \left[1 - \exp\left(\frac{-2\pi k_f R_f}{d_c}\right)\right] \tag{29}$$

These theories show the correlation between pressure drop clean and fouled condition. Affects of fouling on heat transfer is also discussed which will helpful in calculation of overall heat transfer coefficient.

These fundamental logics are going to use in optimization process.

3. Results and Discussion

The result in form of graph has been generated from the data produced by various iterations in HTRI exchanger suit 5.0 and formulating excel sheet to verify with manual calculations. As an application ASE TEMA type water cooling liquid-to-liquid shell and tube heat exchanger has been considered. The data has been granted by CCPL Pvt. Ltd. for the research purpose.

The first set of graph represents influence on velocity of fluid in tube under different fouled condition and relative pressure drop.

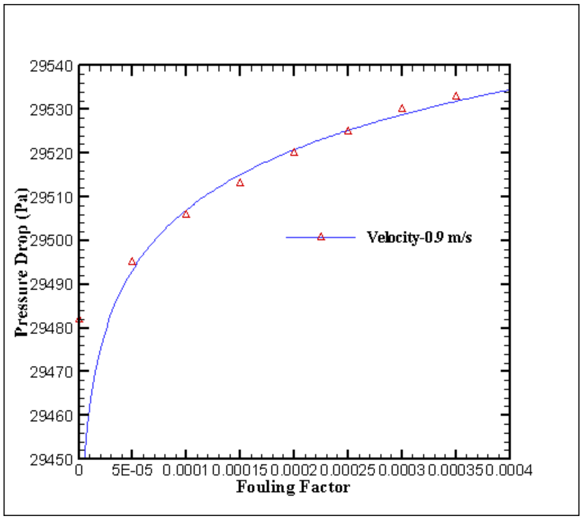

***Figure 4**: fouling factor vs pressure drop at 0.9 m/s velocity*

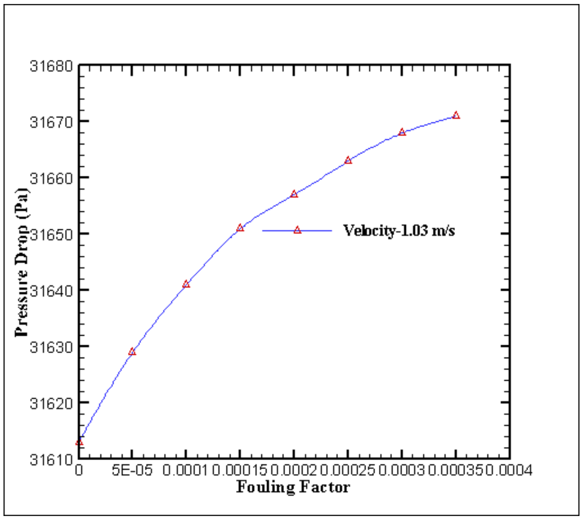

***Figure 5**: fouling factor vs pressure drop at1.03 m/s velocity*

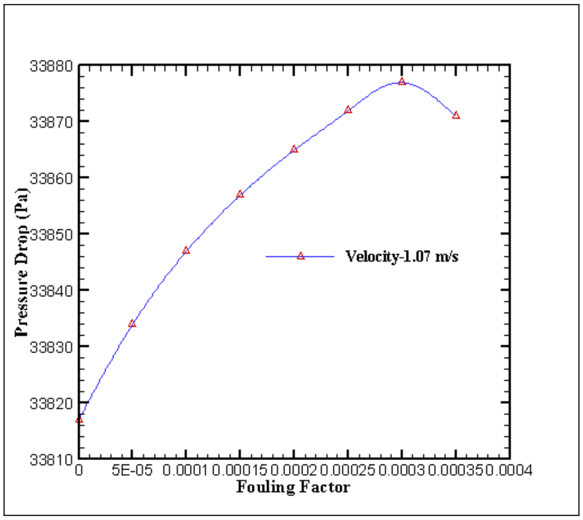

***Figure 6**: fouling factor vs pressure drop at 1.07 m/s velocity*

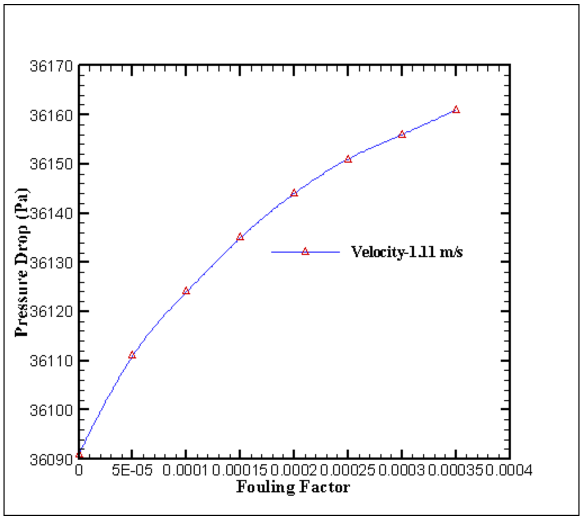

***Figure 7**: fouling factor vs pressure drop at 1.11 m/s velocity*

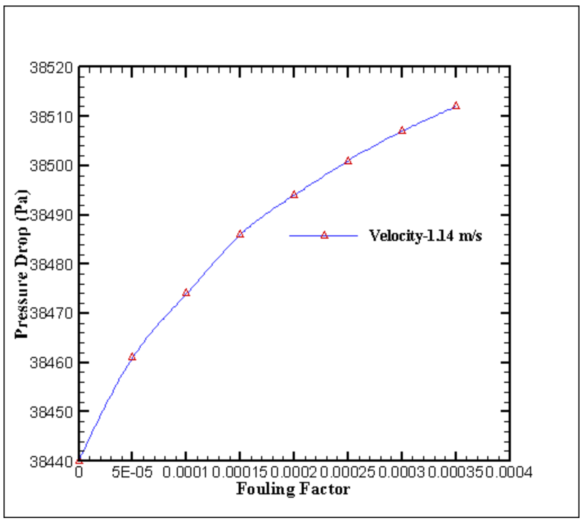

***Figure 8**: fouling factor vs pressure drop at 1.14 m/s velocity*

The graphs clearly show that, for the same fouling condition, at low velocity pressure drop has high result. This result could lead us to the concept of fouling accumulation would more significant at lower velocity. Moreover, theses results also gives the idea about the negotiating between higher velocity and respective heat transfer under certain fouled condition. Results will be helpful to adjust optimal operating condition of shell and tube heat exchanger in fouled conditions.

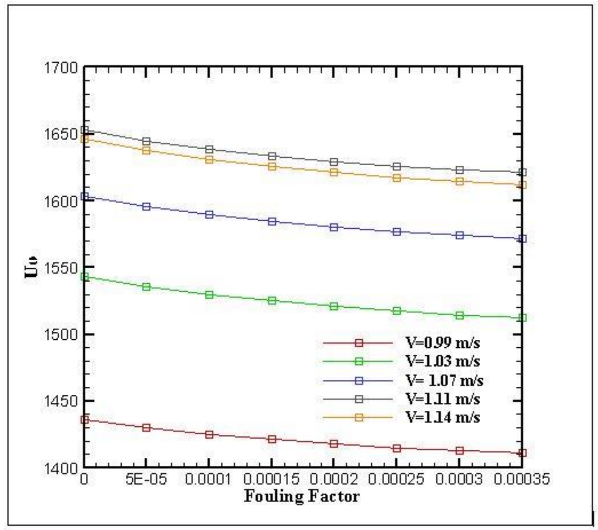

***Figure 9**: Fouling Factor vs* U_o

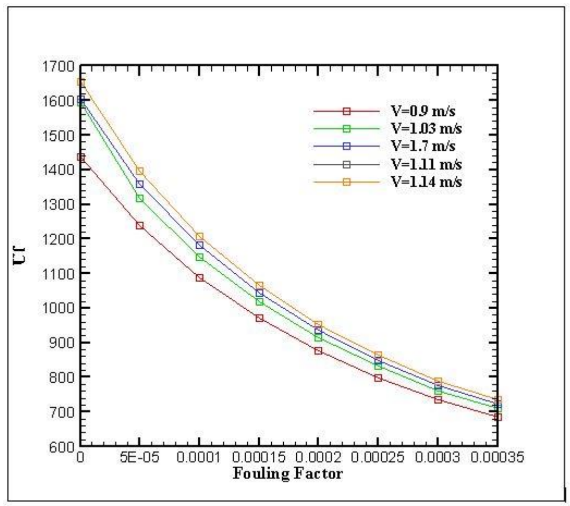

***Figure 10**: Fouling Factor vs* U_f

The graphs represent the effect on heat transfer because of fouling inside the tubes and fluid velocity; overall heat transfer condition and overall fouled heat transfer condition. The significant drop in heat transfer can be observed because of increase in fouling factor. Moreover, change in velocity is also affecting on heat transfer under fouled condition. With increase in velocity heat transfer also increases, but the difference in increase of heat transfer in very less than change in overall heat transfer.

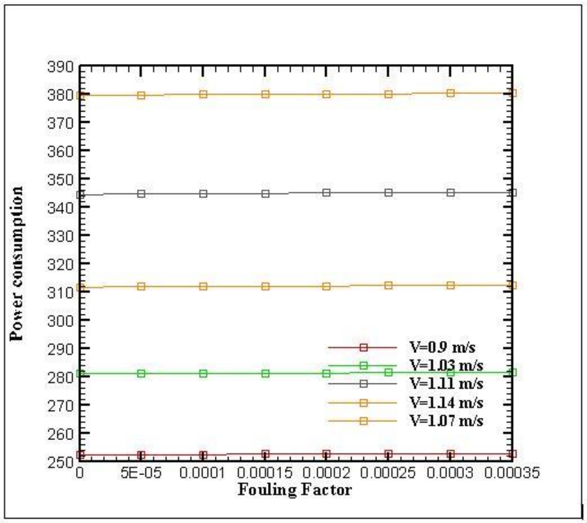

Figure 11: *Fouling factor vs power consumption*

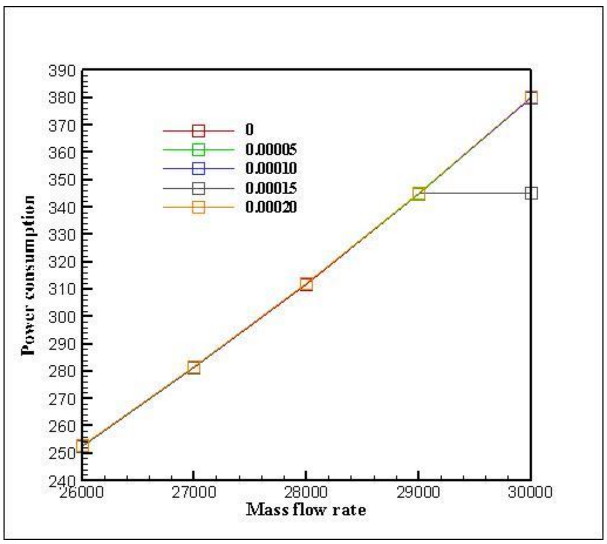

Figure 12: *Mass flow rate vs Power consumption*

The graphs shows the power consumption for tubes in different fouled and mass flow rate conditions. The minor increase in pressure drop has been noted due to fouling. This minor change could be because of fluid is water in tube side.

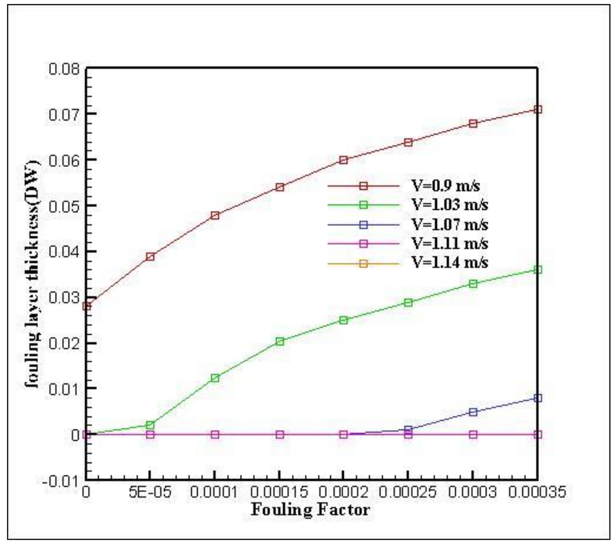

Figure 13: *Fouling factor vs Fouling layer thickness*

These two graphs show the increase in fouling thickness in tube inner side with respect mass flow rate and different fouling factor of water. The last graph clear give the idea that, at low velocity the fouling deposition is high with respect higher mass flow rate. In other words, for water, it is possible to eliminate fouling at somewhat level only with regulating the mass flow rate in tube side. Nevertheless, change in mass flow rate definitely going to affect on the heat transfer through the surface of the tubes.

Conclusion:

This research work highlights an idea about how the fluid fouling properties, although it is water, have significant impact on thermal and hydraulic parameters of the shell and tube heat exchanger tube.

The first part of research (figure 4 to 8) is focused on how mass flow rate affects on deposition rate of fouling. As the velocity increase under same fouling condition the rate of deposition decrease respectively for considered shell and tube heat exchanger. It has been also observed from the results (figure 14) that at low velocity in tube, a deposition rate is high and develop a thick layer of fouling. However, this phenomena lead to lower heat transfer and higher pressure drop in heat exchanger. The most significant point to be noted that (figure 9 & 10), minor change in fouling layer thickness is directly affecting on heat transfer compare to pressure drop for given heat exchanger. (figure 12 and 13) In nutshell, it would be very advantageous to minimize fouling phenomena just with appropriate operating conditions; this could be achieved by adjusting mass flow rate inside the tube.

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