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Thermal And Cfd Analysis Of Shell And Tube Heat Exchanger Used In Solvent Extraction Plant

Jayant Kumar Ghosh[1], Satyanarayan Dubey[2], Mohammed Ali.[3], Palash Soni.[4]

[2,4]Assistant Professor, Oriental University, Indore, Madhya Pradesh, India.

[2]satyanarayandubey@orientaluniversity.in ,[4]palashsoni@orientaluniversity.in

[3]Associate Dean Research and Head Department Of Mechanical Engineering, Oriental University, Indore, Madhya Pradesh, India, Email: mohammedali@orientaluniversity.in

***Corresponding Author:** Jayant Kumar Ghosh*

**[1]M. Tech Scholar (TE), Oriental University, Indore, Madhya Pradesh, India. [1]Jayantghosh05@gmail.com*

Abstract:

A heat exchanger is a device designed to facilitate effective heat transfer between two different mediums. These mediums can either be isolated by a solid barrier to avoid mixing or come into direct contact with each other. Heat exchangers find extensive applications in various industries such as space heating, refrigeration, air conditioning, power generation, chemical processing, petrochemical refining, natural gas treatment, and sewage treatment. An illustrative instance of a heat exchanger is evident in an internal combustion engine, where a circulating fluid, known as engine coolant, passes through radiator coils. Simultaneously, air flows over these coils, resulting in the cooling of the coolant and the heating of the incoming air.

The current study aims to raise awareness regarding fouling on surfaces and ascertain the overall heat transfer coefficient in a heat exchanger. Additionally, it conducts a comprehensive energy analysis on heat exchangers and establishes a correlation for the logarithmic mean temperature difference, adapting it for various types of heat exchangers through the incorporation of a correction factor. A formula is derived for effectiveness, enabling the analysis of heat exchangers even when outlet temperatures are unknown, employing the effectiveness NTU Method. Ultimately, computational fluid dynamics (CFD) and cost analysis are performed to identify the most efficient heat exchanger.

Keywords: Heat Exchanger, Heat transfer coefficient, LMTD, NTU method, CFD.

1. INTRODUCTION

The most basic form of heat exchanger is known as the double-pipe heat exchanger. Within the double-pipe heat exchanger, two primary arrangements exist: the parallel flow heat exchanger and the counter flow heat exchanger.

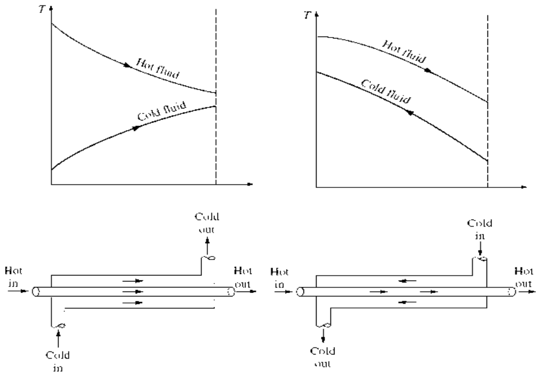

Figure 1.1: a. Parallel flow heat exchanger and

b. Counter flow heat exchanger.

Another category of heat exchanger is the Compact Heat Exchanger, characterized by a substantial heat transfer surface area per unit volume. These compact heat exchangers are frequently employed in gas-to-gas and gas-to-liquid (or liquid-to-gas) heat exchange systems. Typically, these exchangers are configured in a cross-flow arrangement where the two fluids move perpendicular to each other. The cross-flow can be further categorized as either unmixed flow or mixed flow.

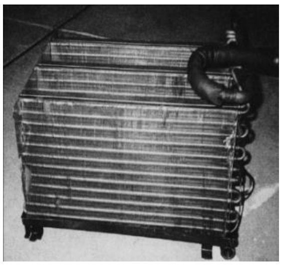

Figure 1.2: Compact (Mixed flow) Heat Exchanger

The third classification is the shell-and-tube heat exchanger, which stands as the most prevalent type in industrial settings. In this design, numerous tubes are arranged within a shell, aligning their axes parallel to that of the shell. The secondary fluid circulates outside the tubes within the shell, often with baffles strategically positioned inside.

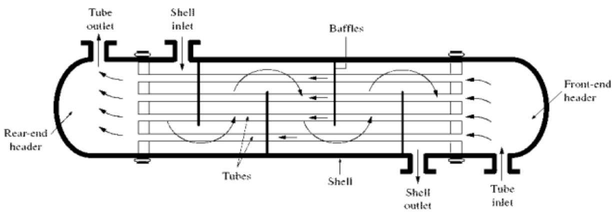

Figure 1.3: Shell and Tube Heat Exchanger

Plate and Frame Heat Exchanger: Comprising a sequence of plates featuring corrugated flat flow passages, this type of heat exchanger facilitates the flow of hot and cold fluids through alternate passages. It is particularly well-suited for liquid-to-liquid heat exchange applications, as long as the hot and cold fluid streams maintain approximately equal pressure levels.

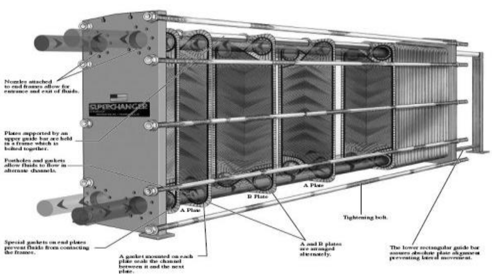

Figure 1.4: Plate and Frame type heat exchanger

The Overall Heat Transfer Coefficient: A typical heat exchanger involves the flow of two fluids, separated by a solid wall. The heat transfer process consists of transferring heat from the hot fluid to the wall through convection, through the wall via conduction, and finally from the wall to the cold fluid through convection once again. The thermal resistance network in this context encompasses two convection resistances and one conduction resistance.

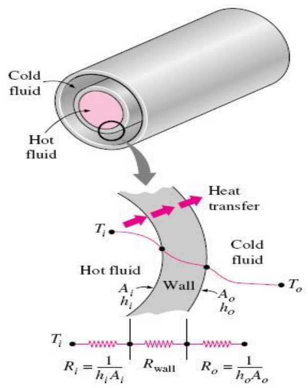

Figure 1.5: Overall heat transfer coefficient

Fouling Factor: The efficiency of heat exchangers typically degrades over time due to the accumulation of deposits on their heat transfer surfaces. The layer of deposits introduces additional resistance to heat transfer, leading to a decline in the rate of heat exchange within the heat exchanger. The fouling factor, denoted as "R_f," encapsulates the cumulative impact of these deposits on heat transfer. Two prevalent types of fouling include the precipitation of solid deposits in a fluid on the heat transfer surfaces and corrosion, along with other forms of chemical fouling.

2. LITERATURE REVIEW

Durgesh J Bhatt et al. [1] A steady-state model was formulated and simulated to predict the outlet temperatures of both the cold and hot fluids in a shell-and-tube heat exchanger. Various experiments were carried out, revealing a consistent trend. It was observed that as the flow rate of the cold fluid increased, the outlet temperature of the cold fluid decreased. Concurrently, there was an observed increase in the overall heat transfer coefficient. These findings suggest a correlation between the flow rate of the cold fluid and the thermal performance of the heat exchanger, highlighting the significance of fluid dynamics in influencing temperature outcomes. **Kirubadurai.B et. al. [2]** An analysis was conducted on orifice baffle and convergent-divergent tubes within a shell-and-tube heat exchanger through experimental investigations. The newly designed heat exchanger demonstrated superior performance, achieving a maximum heat transfer coefficient while maintaining a lower pressure drop. These results underscore the effectiveness of the innovative design, showcasing improvements in both heat transfer efficiency and fluid flow characteristics compared to traditional configurations. **Muhammad Mahmood Aslam et. al. [3]** The conclusion drawn is that the conventional methods employed in the design and development of heat exchangers are costly. Computational Fluid Dynamics (CFD) emerges as a viable alternative, offering a cost-effective and expeditious solution for heat exchanger design and optimization. This shift towards CFD signifies a more efficient and economical approach to achieving optimal heat exchanger performance.**Ender Ozden et. al. [4]** This paper presents a comprehensive thermal analysis, examining the impact of severe loading conditions on heat exchanger performance. The study concludes that insulation proves to be an effective tool in enhancing the heat transfer rate when utilized appropriately, particularly when maintained well below the critical thickness level. The findings underscore the strategic use of insulation as a means to optimize heat exchanger efficiency under challenging operating conditions. **Apu Roy and D.H. Das [5]** The Taguchi method was applied alongside CFD analysis to conduct a screening of experiments, aiming to identify the significant parameters influencing the efficiency of a shell-and-tube heat exchanger. The analysis results led to the conclusion that the optimal parameter conditions for enhancing the efficiency of the heat exchanger, with an outlet temperature of 303.38 K, include a tube diameter of 8.52 mm, a pitch length of 27 mm, and a mass flow rate of 1.9 kg/s. Among these parameters, tube diameter was identified as the most influential, followed by mass flow rate and pitch length in minimizing the outlet temperature. This insight provides valuable guidance for optimizing the performance of the shell-and-tube heat exchanger. **G. E. Kondhalkar & V. N.**

Kapatkat [6] A novel flow arrangement for hot and cold fluids was implemented in the design, with the hot fluid following an axial path and the cold fluid adopting a spiral path. The primary goal of this study is to enhance the design methodology for spiral tube heat exchangers. The proposed spiral tube heat exchanger will undergo development, and experiments will be conducted to analyze pressure drop and temperature changes in both hot and cold fluids on the shell side and tube side. The objective is to gain insights into the performance characteristics of the newly designed spiral tube heat exchanger through systematic experimentation and analysis. **P. Naphon [7]** This paper provides a comparative analysis of various correlations proposed by different researchers for helical coil heat exchangers. These equations utilize different parameters for their analyses, and the paper explores the overall impact of these parameters on Nusselt number (Nu) and heat transfer coefficient (hi). The findings indicate that, at low Reynolds numbers (Re), the Nu vs. Re and hi vs. Re graphs exhibit steeper slopes compared to those at high Re, suggesting that helical coils demonstrate enhanced efficiency at low Re. The analysis further reveals that as the outer coil diameter (D) increases with a constant inner coil diameter (d), the curvature ratio (δ) also increases. This results in a heightened intensity of secondary flows developed within the fluid, subsequently increasing Nu. Therefore, it is recommended to have a small coil diameter (D) and a large tube diameter (d) in a helical coil heat exchanger to optimize its performance.

3. METHODOLOGY

Specifications of SS tube heat exchanger:

1	No. of tubes	500
2	ID	17mm
3	OD	19mm
4	Length	6m
5	Diameter	1m
6	Thickness	2mm
7	Pitch	35mm

S_t/D_o=35/19=1.84
From data table
C=0.482, n=0.566
Velocity of Mesilla (inside the tube) = mass flow rate/A_c x ρ=0.04m/s
(ρ=.6.8kg/m^3, mass flow rate=122.4kg/hr)
Velocity of steam=mass flow rate/A_c x ρ=**0.09m/s**
(ρ=.309kg/m^3, mass flow rate=81.6kg/hr)
Reynolds no. (For inside fluid) R_{ei} =4xmass flow rate of Mesilla/π x d_i x μ=**324.55**
Reynolds no. (For outside fluid) R_{eo} =4xmass flow rate of steam/π x d_o x μ=**2406.**
Prandlt no. (For inside fluid)=μ x C_p /k=**0.02**
(C_p =1.96, μ=.331x10-3 kg-m/s^2, k=.023j/kg-k)
Prandlt no. (For outside fluid) N_{ui}= μ x C_p /k=**0.001**
(C_p =1.8, μ=.119x10-3 kg-m/s^2, k=.17j/kg-k)
Nusselt no. (For inside fluid) N_{uo} =1.13xC (Re)nx(pr)0.23=**2.76**
(C=0.482, Re=432.55, Pr=0.001)
Nusselt no. (For outside fluid)=1.13xC (Re)nx(pr)0.23= **16.80**
(C=0.482, Re=2406.22, Pr=0.02)
Convective heat transfer coefficient (For inside fluid) =h_i= Nu_i x k / d_i
=**2386.58w/m^2 ℃**
Convective heat transfer coefficient (For outside fluid) =h_o= Nu_o x k /d_o
=**12997.89w/m^2 ℃**
Now overall Heat transfer coefficient (For inside tube) =

$$U_i = \frac{1}{\frac{1}{hi} + \frac{ri}{k}\ln\left(\frac{ro}{ri}\right) + \left(\frac{ri}{ro}\right)X\frac{1}{hO} + \frac{\delta}{kA}}$$

=1792.1 w/m^{2o}C

Now overall Heat transfer coefficient (For outside tube) =

$$U_o = \frac{1}{\frac{1}{hO} + \frac{ro}{k}\ln\left(\frac{ro}{ri}\right) + \left(\frac{ro}{ri}\right)X\frac{1}{hi} + \frac{\delta}{kA}}$$

=1633.98 w/m^{2o}C

Now LMTD

θm= θ1-θ2/loge (θ1/θ2)

Where:-

θ1=th1-tc2=90°c-75°c=15°c

θ2=th2-tc1=75°-65=10°c

Now based on this value

θ m=12.34°c

Now Q= m_h x C_{ph} (th1-th2)

Then Q_i =2203.2KJ/hr (For inner side)

Q_o =5042.88 KJ/hr (For outer side)

Now correction factor:-

(For inner side) Q_i =F x U_i x A_i x θ_m

By the above formulae

F=1.72x10-4

(For outer side) Q_o =F x U_o x A_o x θ_m

By the above formulae

F=1.68x10-4

Effectiveness:-

ε = m_h C_h (th1-th2)/C_{min} (th1-tc1)

By this formula we will get

ε=0.6

Now,

The thermal capacity of water vapor=m_h x c_h

=155KJ/hr-k

Similarly,

The thermal capacity of oil stream (Mesilla) =m_c x c_h

=221.04 KJ/hr-k

Capacity ratio:-

C=C_{min} / C_{max}

C=155/221.04

C=0.70

ε=1-exp (-NTU {1-c})/1-exp (-NTU {1-c})

By this we will get:-

NTU=1.23

Now the heat transfer area

A_i=NTU x C_{min}/U_i (for inner side) =**0.1m^2**

A_o= NTU x C_{min}/U_o (for outer side) =**0.11m^2**

Considering Fouling Factor-

Rfi =1/h_i =4.19x10-4m^2 °c/w

Rf_o=1/h_O =7.69x10-5 m^2 °c/w

Now overall Heat transfer coefficient (For inside tube) =

$$U_i = \frac{1}{\frac{1}{hi} + \frac{ri}{k}\ln\left(\frac{ro}{ri}\right) + \left(\frac{ri}{ro}\right)X\frac{1}{hO} + \frac{\delta}{kA} + \mathbf{Rfi} + \left(\frac{\mathbf{ri}}{\mathbf{ro}}\right)\mathbf{Rfo}}$$

=1023.54 w/m^{2o}C

Now overall Heat transfer coefficient (For outside tube) =

$U_o =$

$$= \frac{1}{\frac{1}{h_O} + \frac{r_O}{k}\ln\left(\frac{r_O}{r_i}\right) + \left(\frac{r_O}{r_i}\right) X \frac{1}{h_i} + \frac{\delta}{kA} + \mathbf{Rfo} + \left(\frac{\mathbf{ro}}{\mathbf{ri}}\right) \mathbf{Rfi}}$$

=**872.60 w/m**^{2o}C

Now correction factor:-

(For inner side) $Q_i = F \times U_i \times A_i \times \theta_m$

By the above formulae

F=3.02x10-4

(For outer side) $Q_o = F \times U_o \times A_o \times \theta_m$

By the above formulae

F=7.27x10-4

Now the heat transfer area

$A_i = NTU \times C_{min}/U_i$ (for inner side) =**0.18m^2**

$A_o = NTU \times C_{min}/U_o$ (for outer side) =**0.21m^2**

4. RESULTS

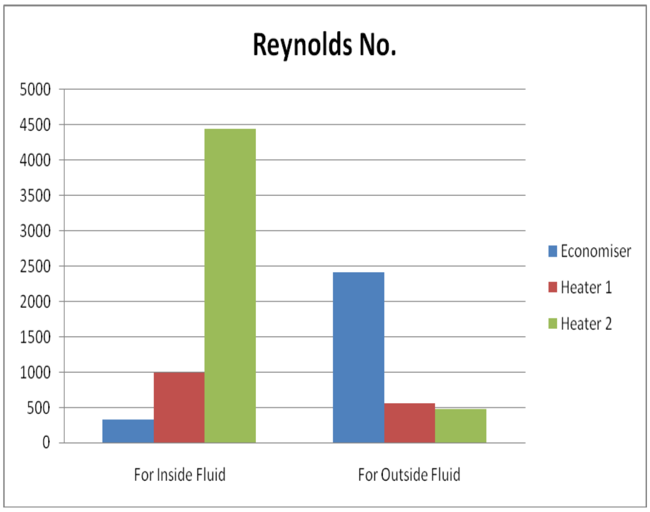

Figure 1.12: Reynolds Number for SS tube without fouling factor.

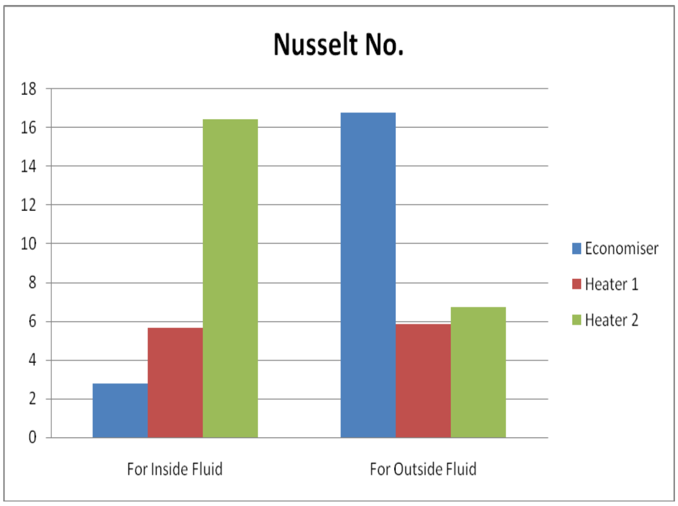

Figure 1.7: Nusselt Number for SS tube without fouling factor.

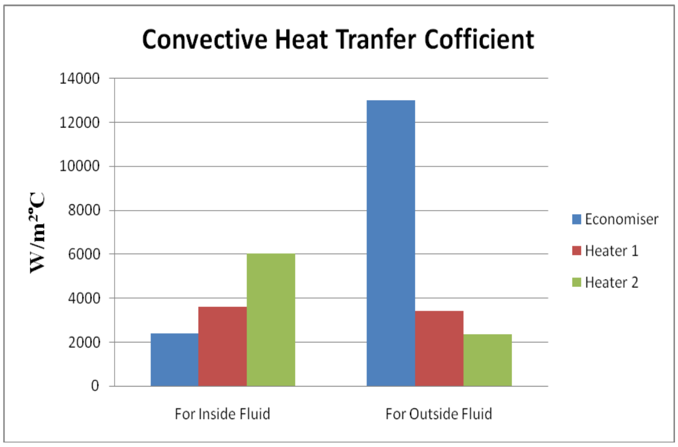

Figure 1.8: Convective heat transfer coefficient for SS tube without fouling factor.

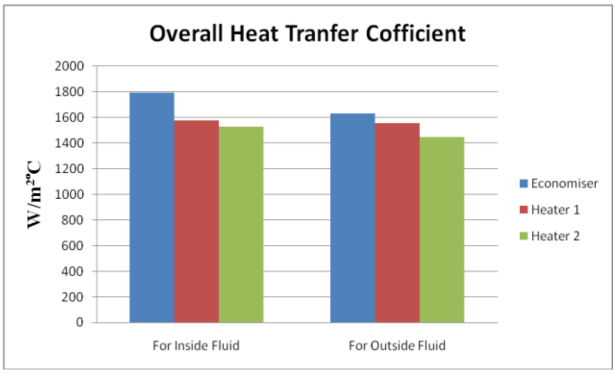

Figure 1.9: Overall heat transfer coefficient for SS tube without fouling factor.

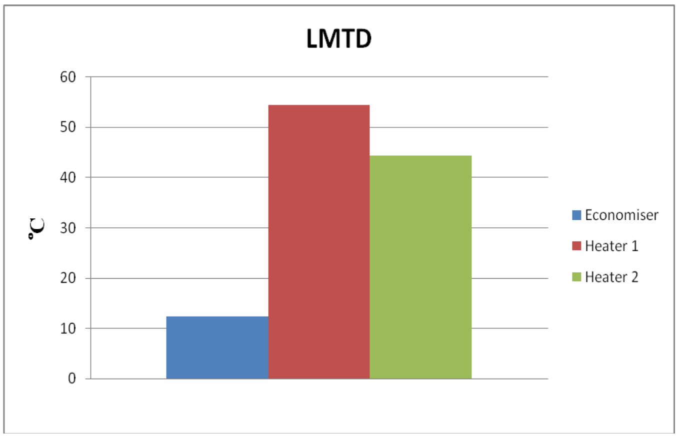

Figure 1.10: LMTD for SS tube without fouling factor.

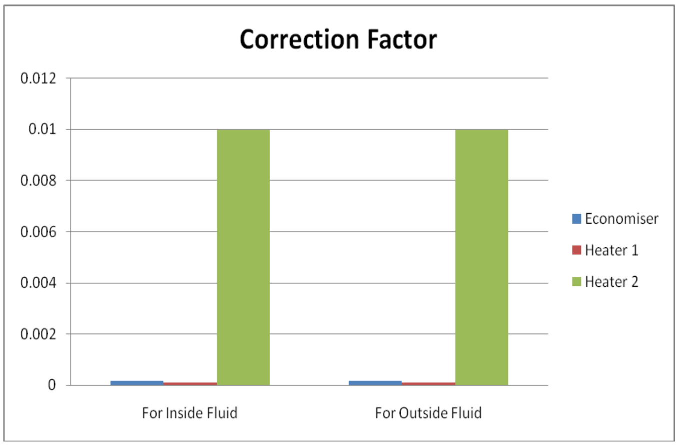

Figure 1.6: Correlation factor for SS tube without fouling factor.

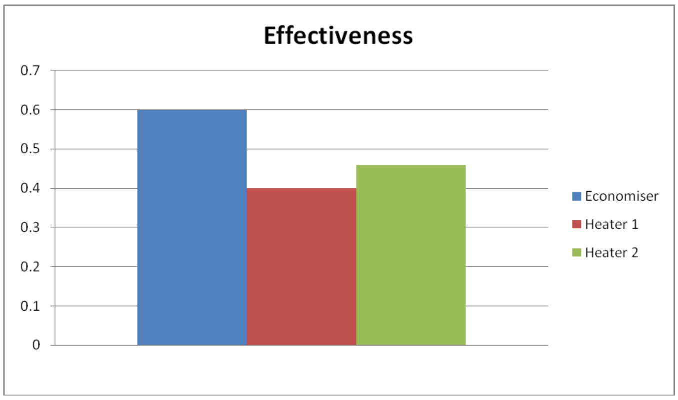

Figure 1.7: Effectiveness for SS tube without fouling factor.

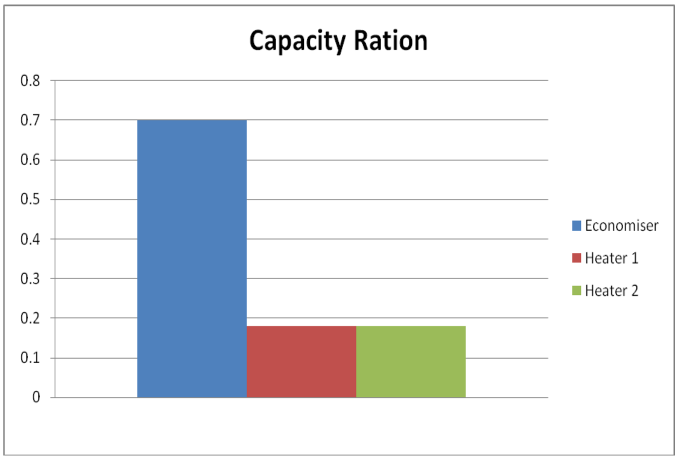

Figure 1.8: Capacity ratio for SS tube without fouling factor.

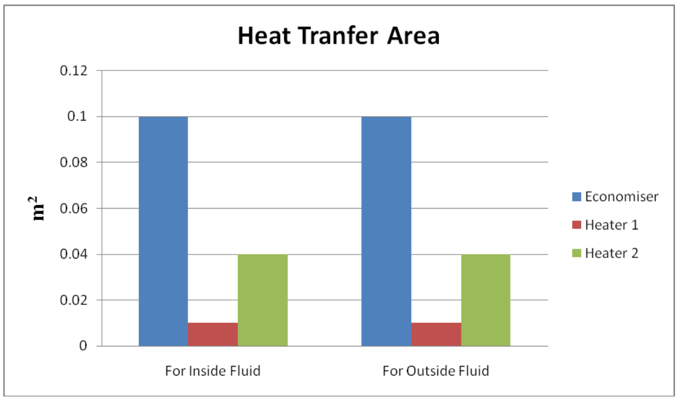

Figure 1.9: Heat transfer area for SS tube without fouling factor.

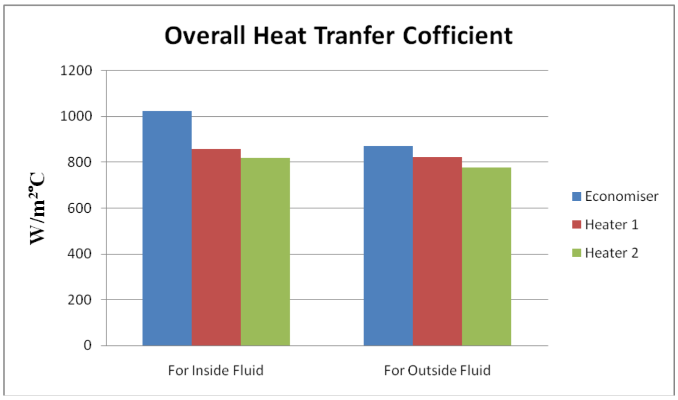

Figure 1.10: Overall heat transfer coefficient for SS tube with fouling factor.

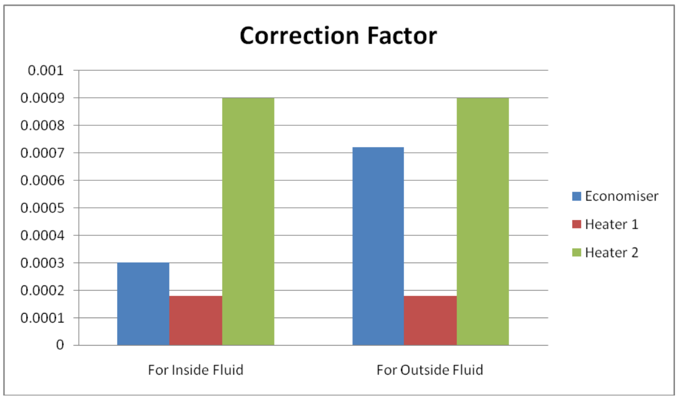

Figure 1.11: Correction factor for SS tube with fouling factor.

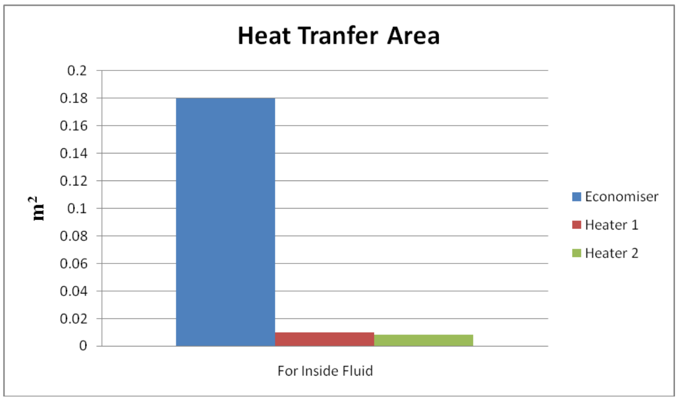

Figure 1.12: Heat transfer area for SS tube with fouling factor.

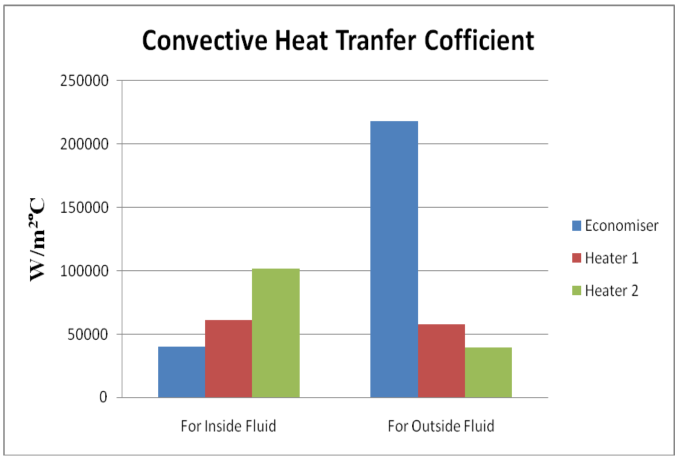

Figure 1.13: Convective heat transfer coefficient for Aluminum tube without fouling factor

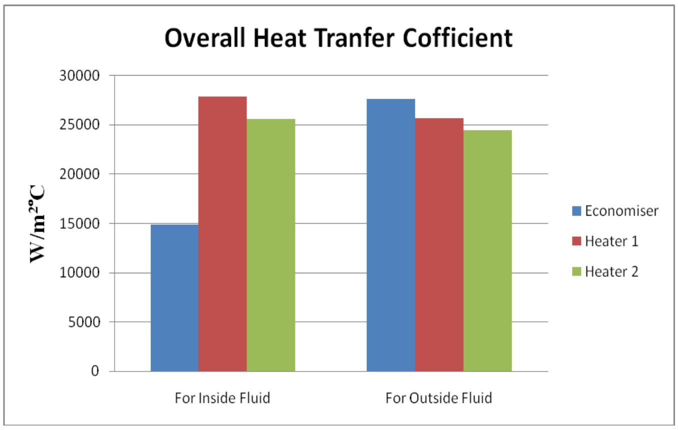

Figure 1.14: Overall heat transfer coefficient for Aluminum tube without fouling factor

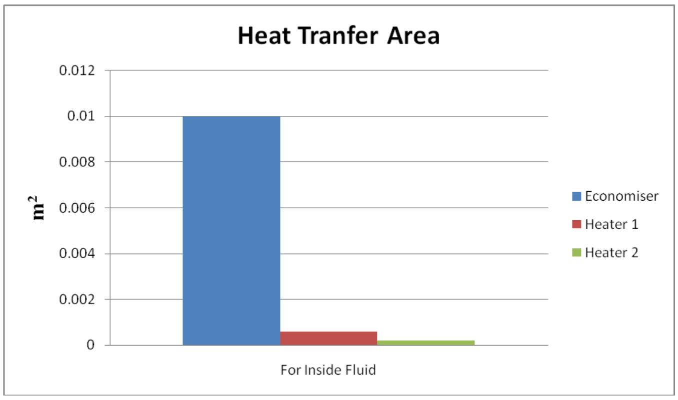

Figure 1.15: Heat transfer area for Aluminum tube without fouling factor

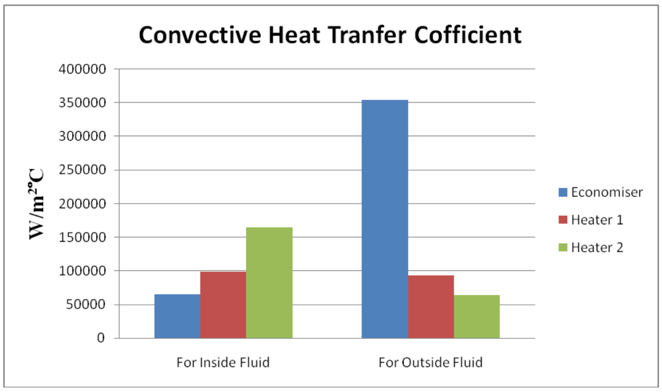

Figure 1.16: Convective heat transfer coefficient for copper tube without fouling factor

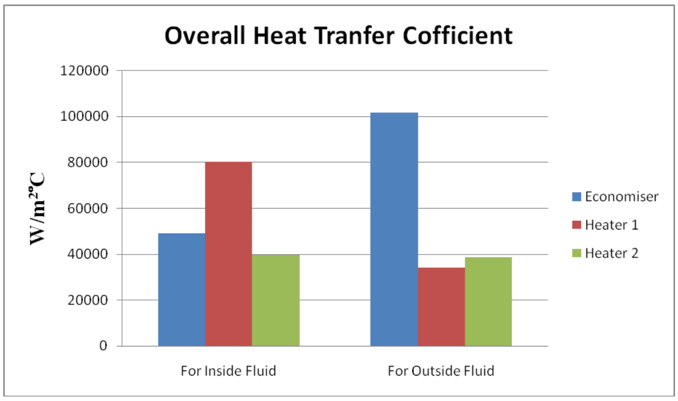

Figure 1.17: Overall heat transfer coefficient for copper tube without fouling factor

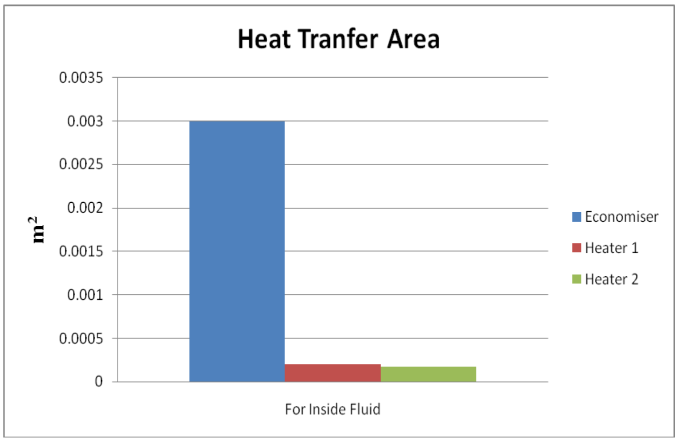

Figure 1.18: Heat transfer area for copper tube without fouling factor Analysis by software (ANSYS)

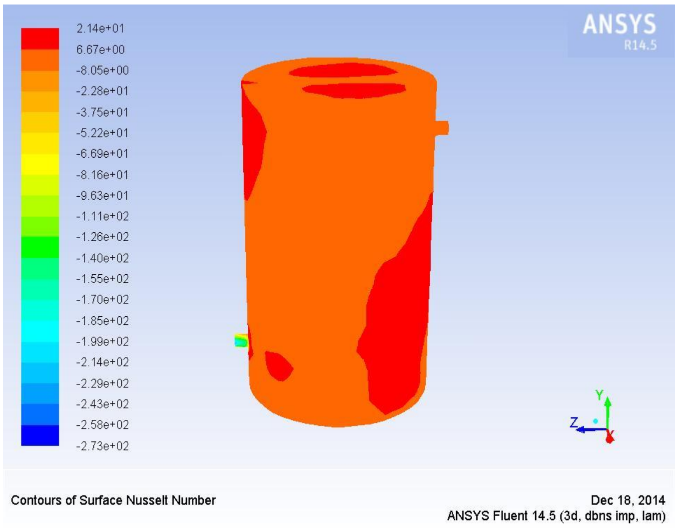

Figure 1.19: Contour of surface Nusselt number for economizer

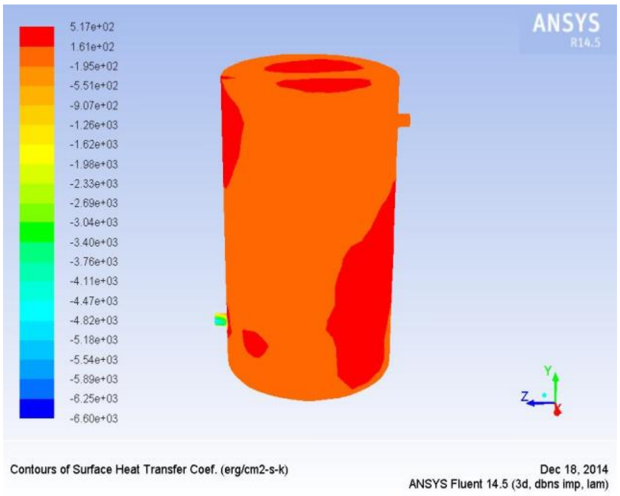

Figure 1.20: Contour for surface heat transfer coefficient for economizer

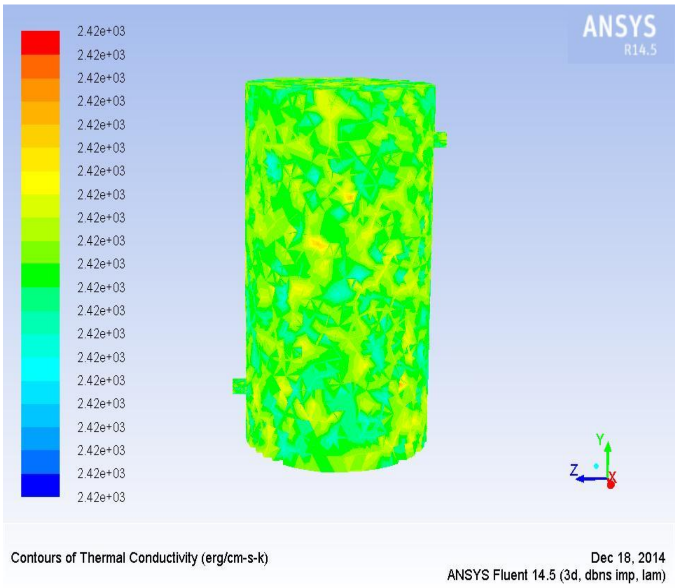

Figure 1.21: Contour for thermal conductivity for economizer

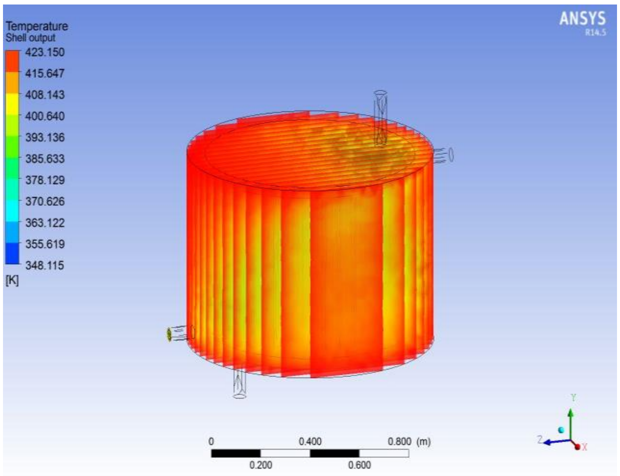

Figure 1.22: Contour for temperature shell output for economizer

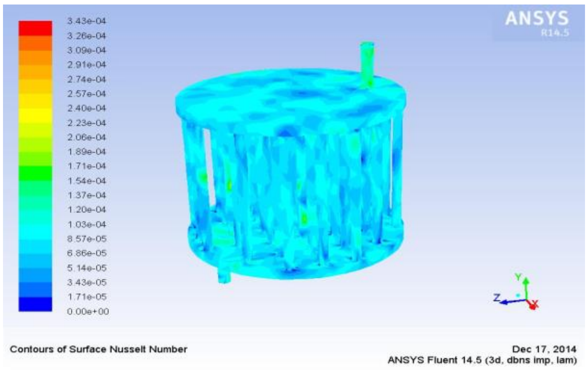

Figure 1.23: Contour of surface Nusselt number for heater 2

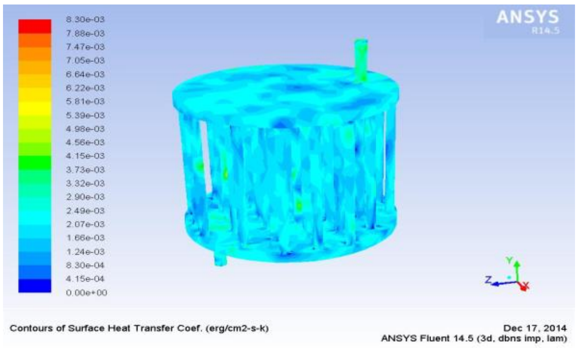

Figure 1.24: Contour of surface heat transfer coefficient for heater 2

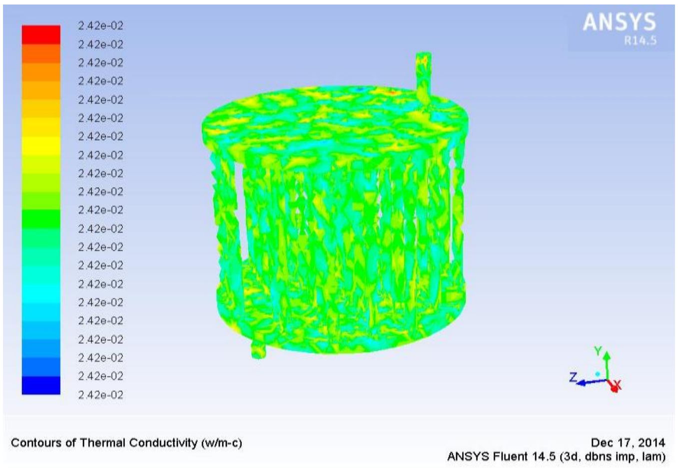

Figure 1.25: Contour for thermal conductivity for heater 2.

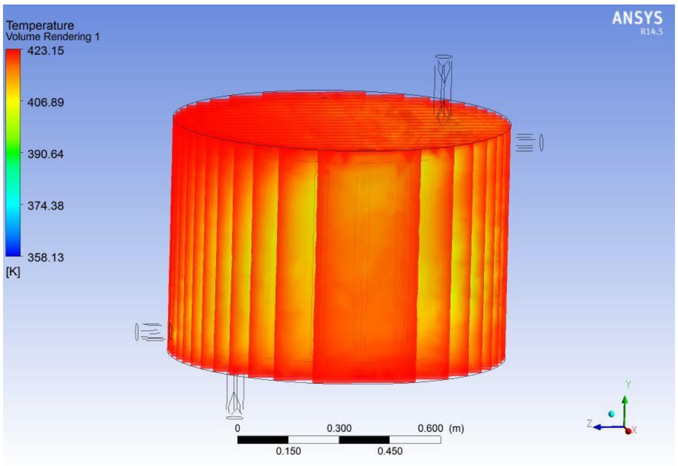

Figure 1.26: Contour for temperature shell output for heater 2.

5. CONCLUSION

1. After analyzing the shell & tube type of heat exchanger (used in Solvent Extraction Plant), it is found that, due to fouling factor, heat transfer is reduced. It shows that the Mud/Dust Particles are deposited inside the tube of heat exchanger or the system is too old. From the calculation based on the Existing data's we found that the Heat transfer Area required for transferring heat is increasing when fouling is taken into consideration.

Now the heat transfer area in case of Economizer:-

Without Fouling:-
A_i=NTU x C_{min} /Ui (for inner side) =**0.1m^2**
A_o= NTU x C_{min} /U_o (for outer side) =**0.11m^2**

With Fouling:-
A_i=NTU x C_{min}/Ui (for inner side) =**0.18m^2**
A_o= NTU xC_{min}/U_o (for outer side) =**0.21m^2**

2. By changing the tube material, heat transfer rate can be increased, which ultimately safe the cost. From the calculation it is found that the total cost required for economizer for Stainless steel is high as compared to the copper, But the life of the copper tube for same existing condition is **3.5 years** whereas for stainless steel is **7 years**. Thus the total cost of tube material for copper is
289124×2=578248 Rs, Which is less than the cost required for Stainless steel tube.

3. Traditional methods employed in the design and development of heat exchangers are known for their costliness. Computational Fluid Dynamics (CFD) emerges as a practical and cost-effective alternative to conventional designs, offering swift solutions for heat exchanger design and optimization. CFD results have become an integral part of the design process, rendering the need for prototypes obsolete. The development of CFD models has democratized the use of CFD, making it accessible not only to specialists but also to process engineers, plant operators, and managers.

While CFD has evolved into a widely utilized tool, it still faces challenges in accurately predicting erosion and corrosion. The lack of suitable mathematical models to represent these physical processes remains an ongoing hurdle in the field of CFD. Despite this, CFD continues to progress and remains a valuable asset in enhancing the efficiency and cost-effectiveness of heat exchanger design processes.

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