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# Performance Evaluation of Expansion Turbine Gas Cooler in a Steel Plant

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***Abstract: A heat exchanger is a device used to transfer heat between two or more fluids. An expansion turbine gas cooler in a steel plant is considered for this present work with hot fluid as air and cold fluid as water. The flow rate on hot side is 24200$m^3$/hr (31200 kg/hr) and cold side is 605 lpm (36300 kg/hr) with inlet temperature 90.2$^\circ$C on hot side and 28.4$^\circ$C on cold side. With the above operating conditions performance analysis is done by using Kern and Bell Delaware method. The objective of this present work is to study and analyze the heat transfer coefficient and pressure drops for different mass flow rates, inlet and outlet temperatures, using the above two methods. Based on this experimental data, various factors like Colburn-j factor, fanning friction factor and heat transfer coefficient are obtained.***

**Keywords: Gas Cooler, Colburn j factor, Friction factor, Heat transfer coefficient, Pressure drop, Effectiveness.**

## I. INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. The present work is concentrated on expansion turbine gas cooler. In this work, thermal and heat transfer analysis is carried out on expansion turbine gas cooler by using Kern and Bell Delaware method

Several researchers discussed performance of different types of heat exchangers used for different purposes and developed correlations both numerically and experimentally.

Karno and Ajib [i] in their paper developed new software for calculation, simulation and optimization of shell and tube heat exchangers. This program is able to predict the effects of baffle spacing, baffle cut, tube size, shell pass number, shell size, etc., on the average heat transfer coefficient, thermal performance and thermal efficiency of the shell and tube heat exchangers. Barman and Ghoshal [ii] in their paper considered an optimum design problem for the different constraints involved in the designing of a shell-and-tube heat exchanger consisting of longitudinally finned tubes. A Mat lab simulation has been employed using the Kern's method. Results shows that the heat transfer coefficients can change when the longitudinal and transverse tube pitches are varied and the best values of these parameters are found. Good agreement is observed between the computed values and the literature values. Y.-G. Lei et al [iii] in their paper studied the impacts of various baffle inclination angles on fluid flow and heat transfer of heat exchangers with helical baffles. They conducted numerical simulations for seven baffle inclination angles by using periodic boundaries. And also in their study they given the optimal baffle inclination angle is about $45^0$ for the shell side, with which the integrated heat transfer and pressure drop performance is the best. Shinde et al [iv] in their paper modifies the existing Bell-Delaware method used for conventional heat exchanger, taking into consideration the helical geometry of helixchanger. Thermal analysis was conducted for conventional shell and tube heat Exchanger and helixchanger for five baffle inclination angles. Analysis results indicate that continual helical baffles can reduce or even eliminate dead regions in the shell side of shell-and-tube heat exchangers. The pressure drop varies drastically with baffle inclination angle and shell-side Reynolds number. Compared to the segmental heat exchangers, the heat exchangers with continual helical baffles have higher heat transfer coefficients to the same pressure drop. Kumar and Kishore [v] in their paper experimentally studied the condensation of water vapor from a binary mixture of air and low-grade steam has been depicted. The study is based upon diffusion heat transfer in the presence of high concentration of non condensable gas. The main objective of their work is to establish an approximate value for surface area and overall heat transfer coefficient of a horizontal shell and tube condenser used in process space. Under designed working conditions, the condenser is found to work efficiently with 90% vapor condensation by mass. Patel and Mavani [vi] in their paper designed a shell and tube heat exchanger which is the majority type of liquid –to- liquid heat exchanger. General design considerations and design procedure are also illustrated in this paper. In design calculation HTRI software is used to verify manually calculated result. Kulkarni et al [vii], in their paper studied a comparative analysis of a water to water STHE and analyze the heat transfer coefficient and pressure drops for different mass flow rates and inlet and outlet temperatures, using Kern, Bell and Bell Delaware methods. And they mainly aims at studying and comparing different methods of STHE and bringing out which method is better for adopting in shell side calculations. Dahare and Basavaraj [viii], in their paper presents a recent innovation and development of a new technology, known as Twisted Tube technology, which has been able to overcome the limitations of the conventional technology, and in addition, provide superior overall heat transfer coefficients through tube side enhancement and compares the construction, performance, and economics of Twisted Tube exchangers against conventional designs for copper materials of construction including reactive metals. Shanker and Kishore [ix] in their paper carried out performance evaluation on charge air coolers with varying mass flow rates on hot side from 0.1 to 0.6 kg/s and keeping cold fluid rate constant at 0.265 kg/s and obtained Colburn-j factor, Fanning friction factor, heat transfer coefficient, overall heat transfer coefficient and effectiveness of

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heat exchanger. And found that water-cooled charge air cooler is having higher effectiveness than air-cooled type

## II. DESCRIPTIONS AND WORKING OF EXPANSION TURBINE GAS COOLER

### DESCRIPTION:

The shell and tube heat exchanger is named for its two major components – round tubes mounted inside a cylindrical shell. The shell cylinder can be fabricated from rolled plate or from piping (up to 24 inch diameters). The tubes are thin walled tubing produced specifically for use in heat exchangers. Other components include: the channels (heads), tubesheets, baffles, tie rods & spacers, pass partition plates and expansion joint (when required). Shell & tube heat exchanger designs and constructions are governed by the TEMA and ASME codes.
The main components of a shell and tube exchanger are shown in below figure

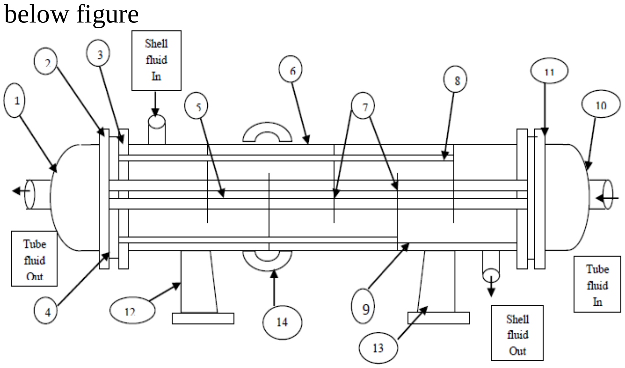


**Fig 1 Shell and tube geometric terminology**

| | |
|---|---|
| 1 Channel Cover | 2 Channel Flange |
| 3 Shell Flange | 4 Tube Sheet |
| 5 Tube | 6 Shell |
| 7 Baffles | 8 Tie Rods |
| 9 Sealing Strips | 10 Floating Head Cover |
| 11 Floating Head Flange | 12 fixed Support |
| 13 Sliding Support | 14 Expansion Joint |

**Table 1 Different parts of shell and tube heat exchanger**

**Shell:**
The shell is the enclosure and the passage of shell side fluid. It has a circular cross section and the selection of shell material depends upon the corrosiveness of the working fluid temperature and pressure. Carbon steel is a common material for the shell under moderate working conditions.

**Tubes**
The tubes provide the heat transfer area in a shell and tube heat exchanger. Tubes of 19 mm and 25 mm diameter are most commonly used and the tube wall thicknesses designated interms of BWG. Tubes are generally arranged in triangular or square pitch**.**

**Tubesheets**
Tubesheets are plates or forgings drilled to provide holes through which tubes are inserted. Tubes are appropriately secured to the tubesheet so that the fluid on the shell side is prevented from mixing with the fluid on the tube side. Holes are drilled in the tubesheet normally in either of two patterns, triangular or square. The distance between the centers of the tube hole is called the tube pitch; normally the tube pitch is 1.25 times the outside diameter of the tubes.

**Baffles**
Baffles serve functions like support the tube, maintain the tube spacing and direct the flow of fluid in the desired pattern through the shell side. A segment, called the baffle cut, is cut away to permit the fluid to flow parallel to the tube axis as it flows from one baffle space to another. Segmental cuts with the height of the segment approximately 25 percent of the shell diameter are normally the optimum.

**Tie Rods and Spacers**
Tie rods and spacers are used for two reasons: hold the baffle assembly together and maintain the selected baffle spacing. The tie rods are secured at one end to the tubesheet and at the other end to the last baffle. They hold the baffle assembly together. The spacers are placed over the tie rods between each baffle to maintain the selected baffle pitch.

**The Bonnet and The channel:**
The closure of heat exchanger is called bonnet or channel depending upon its shape and construction. A bonnet is an integral cover and a channel has a removable cover. The bonnet closure consists of a short cylinder section with a bonnet welded at one end and a flange welded at the other end. The bonnet type of closure is replaced by channel type of closure if a nozzle is required to be fixed.

**The pass partition plate:**
The channel is divided into two compartments by a pass partition plate. The number of tube and shell passes can be increased by using more pass partition plate on both sides of the fluid. The number of passes in shell or tube side indicates the number of times the shell side or tube side fluid traverse the length of the exchanger. For given number of tubes, the area available for flow of the tube side fluid is inversely proportional to the number of passes

**Flanges and gaskets:**
The flanges fixes the channel and bonnet closures to the tube sheets Gaskets are placed between two flanges to make the joint leak free

**Expansion joint:**
The expansion joint prevents the problem of thermal stresses which may occur when there is a substantial difference of expansion between the shell and tubes because of temperature difference between fluid streams

**Working of expansion turbine gas cooler**
Air is filtered for removal of dust, compressed in a centrifugal compressor to a pressure of 56.4 Kg/Cm$^2$ and to a temperature of 97℃ and this air is passed through cooler with water as coolant and the temperature of the gas after it passed through cooler is 47℃ with same pressure, mass flow rate. After that carbondioxide and moisture are removed from the air by the adsorbents activated alumina and molecular sieves. Air is

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separated in to products Oxygen and Nitrogen by cryogenic distillation of air in the temperature range of - 175℃ to -192℃. The cold or refrigeration required for the system is generated by the expansion of high pressure air to low pressure through a set of reducing valves. An intermediate product from the distillation column, crude argon containing 10% Argon is enriched to 90% Argon in a distillation column is called crude argon column. Oxygen in the argon is removed with the injection of hydrogen and formation of water in presence of deoxocatalyst. The water from Argon is separated through separator & adsorption system and Nitrogen from the argon separated by cryogenic distillation system at pure argon column. These products are stored in cylinders and supply to different departments.

## III. ANALYSIS OF EXPANSION TURBINE GAS COOLER:

**Calculation of Shell Side Heat Transfer Coefficient Using Kern Method:**

Cross flow area of shell is given by the equation $A_s$

$$A_s = \frac{D_s C B}{p_t} \tag{1}$$

Where $D_s$ is diameter of shell, C is clearance, B is baffle spacing, $p_t$ is tube pitch

Mass velocity of shell side fluid is given by $G_s$

$$G_s = \frac{m_s}{A_s} \tag{2}$$

Where $m_s$ is mass flow rate, $A_s$ is cross flow area

Equivalent diameter is calculated by the equation $D_e$

$$D_e = 4\left[\left(\frac{\frac{\sqrt{3}}{4}p_t^2 - \frac{\pi}{8}d_o^2}{\frac{\pi d_o}{2}}\right)\right] \tag{3}$$

Where $d_o$ is outer diameter of tube

Reynolds no. of the shell side fluid is determined by $Re_s$

$$Re_s = \frac{G_s \times D_e}{\mu_s} \tag{4}$$

Where $\mu_s$ is dynamic viscosity

Prandtl number of the shell side fluid is given by $Pr_s$

$$Pr_s = \frac{(C_{ps}\mu_s)}{k_s} \tag{5}$$

Where $C_{ps}$ is specific heat, $k_s$ thermal conductivity

Heat transfer coefficient of the shell side fluid is given by

$$\frac{h_o D_e}{k_s} = 0.36\left(\frac{(Gs \times De)}{\mu_s}\right)^{0.66}\left(\frac{(\mu_s C_{ps})}{k_s}\right)^{1/3}\left(\frac{\mu_s}{\mu_w}\right)^{0.14} \tag{6}$$

**Calculation of pressure drop using kern method:**

$$\Delta P_s = \frac{f G_s^2 (N_b + 1) D_s}{2\rho D_e \phi_s} \tag{7}$$

$$f = e^{(0.576 - 0.19 \ln (Re_s))} \tag{8}$$

**Calculation of Shell Side Heat Transfer Coefficient Using Bell Delaware Method:**

**$h = h_i\, j_c\, j_l\, j_b\, j_s\, j_r$**

Ideal heat transfer coefficient $h_i$ is given by

$$h_i = j_i c_{ps}\left(\frac{m_s}{s_m}\right)\left(\frac{k_s}{c_{ps}\mu_s}\right)^{2/3}\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{9}$$

Reynolds no. and bundle cross flow area is given by

$$Re_s = \frac{m_s}{s_m} \times \frac{d_{tube-outer}}{\mu_s} \tag{10}$$

$$s_m = B\left[l_{bb} + \frac{D_{ctl}}{p_{t,eff}}(p_t - d_o)\right] \tag{11}$$

Where $l_{bb}$ is bundle to shell diametrical clearance, $D_{ctl}$ is tube centerline limit

**Segmental baffle window correction factor $J_c$:**

$$J_c = 0.66 + 0.72 Fc \tag{12}$$

$$F_c = 1 - 2\left[\frac{\theta_{ctl}}{360} - \frac{\sin\theta_{ctl}}{2\pi}\right] \tag{13}$$

$$\theta_{ctl} = 2\cos^{-1}\left[\frac{D_s}{D_{ctl}}\left(1 - 2\left(\frac{B_c}{100}\right)\right)\right] \tag{14}$$

**Correction factor for baffle leakage effect $J_l$:**

$$J_l = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2 r_{lm}} \tag{15}$$

**Correction factor for bundle bypass effects $J_b$:**

$$J_b = exp\left(-c_{bh} F_{sbp}\left(1 - \sqrt[3]{2 r_s}\right)\right) \tag{16}$$

**Heat transfer correction for unequal baffle spacing:**

$$J_s = \frac{(N_b - 1) + L_i^{*(1-n)} + L_o^{*(1-n)}}{(N_b - 1) + L_i^* + L_o^*} \tag{17}$$

**Pressure Drop Calculation using Bell Delaware**

$$\Delta P_c = \Delta P_{bi}(N_b - 1)R_l R_b \tag{18}$$

$$\Delta P_{bi} = 4 f_i \frac{G_s^2}{2\rho_s}\left(\frac{\mu_{sw}}{\mu_s}\right)^{0.14} N_{tcc} \tag{19}$$

$$\Delta P_w = N_b\left[(2 + 0.6 N_{tcw})\frac{m_w^2}{2\rho_s}\right]R_l \tag{20}$$

$$\Delta P_e = (\Delta P_{bi})\left[1 + \frac{N_{tcw}}{N_{tcc}}\right]R_b R_s \tag{21}$$

Total pressure drop $= (\Delta p_c + \Delta p_w + \Delta p_e)$

**Calculation of tube side heat transfer co-efficient:**

$$Nu_b = \frac{\left(\frac{f}{2}\right)(Re - 1000)Pr}{1 + 12.7\left(\frac{f}{2}\right)^{1/2}(Pr^{2/3} - 1)} \tag{22}$$

**Calculation of tube side pressure drop**

$$\Delta P_t = \left(\frac{4 f l N_p}{d_i} + 4 N_p\right)\rho \frac{u_m^2}{2} \tag{23}$$

**Effectiveness of the heat exchanger**

$$\varepsilon = \frac{C_{max}(T_{hi} - T_{ho})}{C_{min}(T_{hi} - T_{ci})} \text{ or } \frac{C_{max}(T_{co} - T_{ci})}{C_{min}(T_{hi} - T_{ci})} \tag{24}$$

## IV RESULTS AND DISCUSSION

The values got from the above are shown plotted as graphs depicting their behavior as mentioned below.

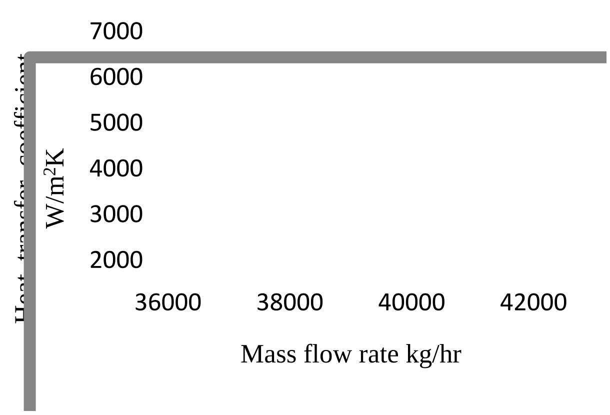


**Fig 2 Variation of heat transfer coefficient with mass flow rate of water**


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The above figure shows the effect of different mass flow rates on heat transfer coefficient of water by using kern and Bell Delaware methods. Results show that with increase in mass flow rate from 605 lpm (36300 kg/hr) to 690 lpm (41300 kg/hr) heat transfer coefficient is also increases for both the methods but the value of heat transfer coefficient given by kern method is more than Bell Delaware method. This is because kern method does not consider the effects caused due to leakage of streams in expansion turbine gas cooler as a result the value of heat transfer coefficient is more compared to Bell Delaware method

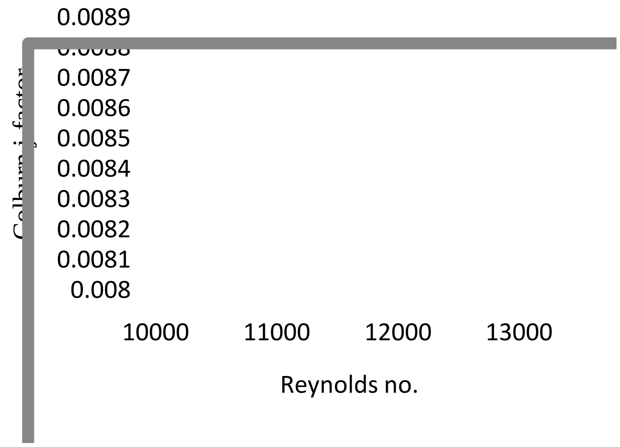


**Fig 3 Variation of Colburn j factor with Reynolds no. of water**

The above figure shows the variation of colburn j factor with Reynolds no. of water. The results show that the colburn j factor decreased with increase in Reynolds number. Colburn j factor is a function of Reynolds no. with negative power. As mass flow rate of water increases Reynolds no. is also increases there by the colburn j factor decreases.

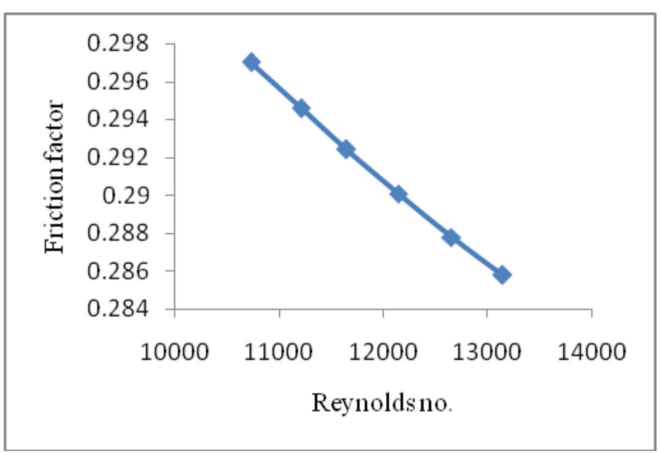


**Fig 4 variation of friction factor with Reynolds no. of water**

The above figure shows the variation of friction factor with Reynolds no. of water. The results show that friction factor f, decreases with increase in Reynolds number of water. Friction factor is an exponential function of Reynolds number, where Reynolds no. is proportional to mass flow rate. With increase in mass flow rate of water the Reynolds number increase there by the friction factor decreases.

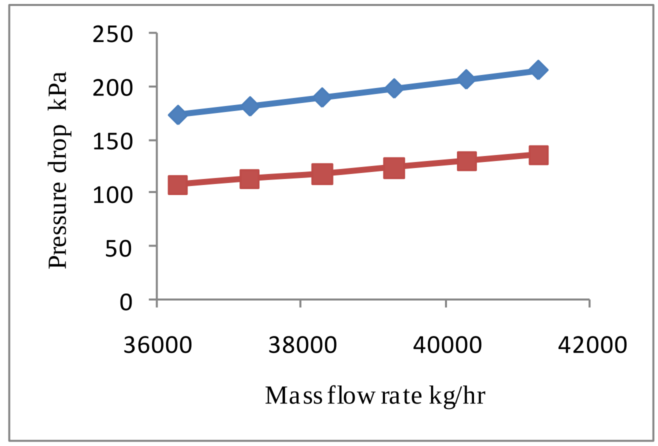


**Fig 5 Variation of pressure drop with mass flow rate of water**

The above figure shows the variation of pressure drop with mass flow rate for both Kern and Bell Delaware methods. The results show that the pressure drop increases with increase in mass flow rate. Pressure drop is a function of mass velocity and friction factor. As mass flow rate of water increases mass velocity also increases, but there is decrease in friction factor. The decrease in friction factor is negligible when compared to increase in mass velocity.

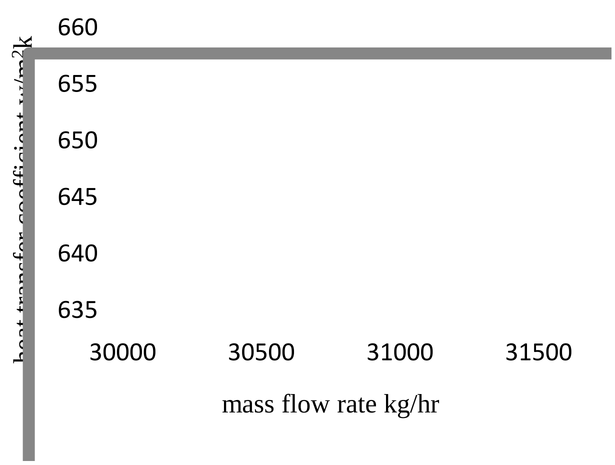


**Fig 6 Variation of heat transfer coefficient with mass flow rate of air**

The above figure shows the effect of different mass flow rates with heat transfer coefficient for air. The results show that with increase in mass flow rate from 24200$m^3$/hr (30200 kg/hr) to 25000 $m^3$/hr (31200 kg/hr) heat transfer coefficient is also increases from 639 $W/m^2K$ to 658 $W/m^2K$. Therefore from the above discussion it is clear that heat transfer coefficient is enhanced by increase in mass flow rate.

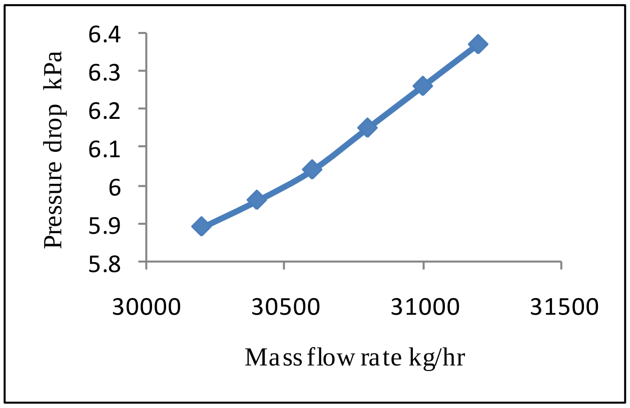


**Fig 7 Variation of pressure drop with mass flow rate of air**

The above figure shows the variation of pressure drop with mass flow rate of air. The results show that the pressure drop increases with increase in mass flow rate of air. Pressure drop is a function of mean velocity and friction factor. As mass flow rate of air increases mean velocity also increases, but there is decrease in friction factor. The decrease in friction factor is negligible when compared to increase in mean velocity.

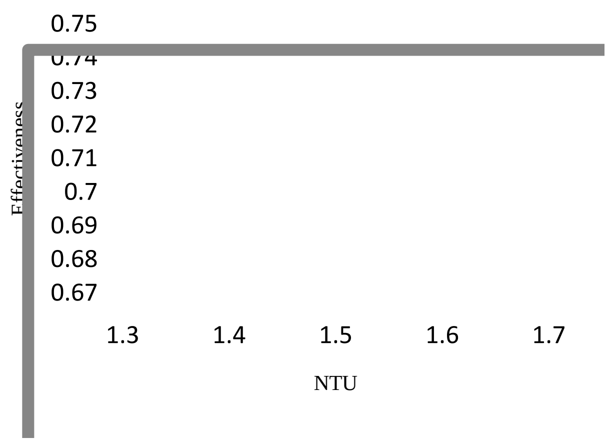


**Fig 8 Variation of effectiveness with number of transfer units**

The above figure shows the variation of effectiveness with number of transfer units. The results show that with increase in number of transfer units from 1.34 to 1.59 effectiveness is also increases from 0.67 to 0.73. Therefore from the above discussion it is clear that effectiveness is enhanced by increase in number of transfer units.

## V. CONCLUSION

Based on experimental results, the following conclusions are being made

1. Mass flow rate of water increases from 605 lpm (36300 kg/hr) to 690 lpm (41300 kg/hr) then heat transfer coefficient and pressure drop of water increases by 10.66% and 20.83 % respectively.
2. Mass flow rate of air increases from 24200m$^3$/hr (30200 kg/hr) to 25000m$^3$/hr (31200 kg/hr) then heat transfer coefficient and pressure drop of air increases by 2.97% and 7.96% respectively.
3. Reynolds number of water increases from 10733 to 13147. Then Colburn j factor and Friction factor of air decreases by 7.95% and 4.51% respectively.
4. Kern method does not take into consideration the obstructions due to baffles in calculating the pressure drop and heat transfer coefficient and hence the values given by Kern method are approximate. Whereas the values given by Bell Delaware method are more accurate then Kern method since it considers the leakage streams caused by baffles.
5. The variations of heat transfer coefficient and pressure drop values from Kern to Bell Delaware method are 45.66% and 37.48% respectively.

## NOMENCLATURE

| | |
|---|---|
| A | Cross flow area, m$^2$ |
| B | Baffle spacing, m |
| C | Clearance between tubes, m |
| $C_p$ | heat capacity, J/kg K |
| Ds | Shell inside diameter, m |
| $d_o$ | Tube outside diameter, m |
| $d_i$ | Tube inside diameter, m |
| $F_c$ | Fraction of total number of tubes in a cross flow section |
| $F_{bp}$ | Fraction of the cross flow area available for bypass flow |
| f | Friction factor. |
| G | mass velocity, kg/ m$^2$-s |
| h | heat transfer coefficient, W/ m$^2$K |
| $h_i$ | ideal heat transfer coefficient, W/ m$^2$K |
| $J_c$ | Segmental baffle window correction factor |
| $J_l$ | Correction factor for baffle leakage effect |
| $J_b$ | Correction factor for bundle bypass effects |
| $J_s$ | Correction factor for unequal baffle spacing |
| k | thermal conductivity, W/m K |
| L | Length of shell, m |
| m | Mass flow rate of the fluid, kg/s |
| N | Number of tubes |
| $N_b$ | Number of baffles |
| $N_c$ | Number of tube rows crossed in one cross flow section. |
| $N_{cw}$ | Effective number cross flow rows in window zone |
| $N_p$ | Number of tube passes |
| $N_{ss}$ | No. of sealing strips |
| $N_c$ | No. cross flow rows |
| Pr | Prandtl number |
| $P_t$ | Tube pitch, m |
| Re | Reynolds number |
| $S_m$ | Area of the shell side cross flow section, m$^2$ |
| $S_{sb}$ | Shell to baffle leakage area, m$^2$ |
| $S_{tb}$ | Tube to baffle leakage area, m$^2$ |
| $S_w$ | Window flow area, m$^2$ |
| $t_b$ | Tube thickness, m |
| U | linear velocity m/s |

**Greek symbols**

| | |
|---|---|
| Δb | Bundle to shell diametrical clearance |
| Δsb | Shell to baffle diametrical clearance |
| Δtb | Tube to bundle diametrical clearance |
| ε | heat exchanger effectiveness |
| μ | fluid dynamic viscosity, Pa-s |
| ν | kinematic viscosity, $m^2/s$ |
| ρ | fluid density, kg/ $m^3$ |

**Subscripts**

| | |
|---|---|
| c | cold fluid side |
| h | hot fluid side |
| i | inlet |
| o | outlet |
| m | mean bulk temperature |
| w | wall |
| s | shell side |
| t | tube side |

## REFERENCES

*i. Karno A. and Ajib S, Effect of Tube Pitch on Heat Transfer in Shell-and-Tube Heat Exchangers New Simulation Software', Journal of Heat Mass Transfer, Vol. 42, pp. 263-270, 2006.*

*ii. Joy deep Barman and Ghoshal A.K, Performance Analysis of Finned Tube and Unbaffled Shell-and-Tube Heat Exchangers', International Journal of Thermal Sciences, Vol. 46, pp. 1311-1317, 2007.*

*iii. Yong-Gang Lei and Ya-Ling, Effects of Baffle Inclination Angle on Flow and Heat Transfer of a Heat Exchanger with Helical Baffles , Journal of Chemical Engineering and Processing, Vol. 47, pp. 2336–2345, 2008.*

*iv. S. S. Shinde and P. V. Hadgekar, Comparative Thermal Analysis of Helixchanger with Segmental Heat Exchanger using Bell Delaware Method, International Journal of Advances in Engineering & Technology, Vol. 3, Issue 2, pp. 235-242, may 2012.*

*v. Rajnish kumar and P S Kishore, Experimental Study of Condensation in a Shell and Tube Heat Exchanger in the Presence of a Noncondensable Gas, Journal of Heat transfer Asian research, May 2016.*

*vi. Sandeep K. Patel and Alkesh M. Mavani, Shell & Tube Heat Exchanger Thermal Design with Optimization of Mass Flow Rate and Baffle Spacing, International Journal of Advanced Engineering Research and Studies, Vol. 2, Issue 1, pp. 130-135, Dec-2012.*

*vii. Shweta Y Kulkarni And Jagadish S B ,Analysis Comparing Performance of A Conventional Shell and Tube Heat Exchanger using Kern, Bell and Bell Delaware Method, International Journal of Research in Engineering and Technology, Vol. 3, Issue 3, pp. 486-496, May-2014.*

*viii. Nitesh B. Dahare and M. Basavaraj, Experimental Analysis and Performance Characteristic of Heat Transfer in Shell and Twisted Tube Heat Exchanger ,International Research Journal of Engineering and Technology , Vol. 2 Issue 7, pp. 1142-1147, Oct-2015.*

*ix. Devi Shanker and P S Kishore, Thermal Analysis of Water Cooled Charge Air Cooler in Turbo Charged Diesel Engine, International Journal of Research in Engineering and Technology, Vol. 5, Issue 2, pp. 193-197, 2016.*

***x.** Eswar, Thermal Analysis of Expansion Turbine Gas Cooler, M.E Thesis, 2016.*