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Performance Analysis of a 1-1 Shell and Tube Intercooler for Compressed Air-Cooling Systems

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ABSTRACT. This study presents an evaluation of a 1-1 shell-and-tube heat exchanger (Intercooler-2) utilized in a multi-stage air compressor. The evaluation was conducted using plant operational data and classical heat exchanger analysis methods, including heat balance, overall heat transfer coefficient under clean (Uc) and operating (Ud) conditions, dirt factor (Rd) and pressure drop (ΔP). The calculation results show a heat transfer rate of 557,640.622 Btu/h with a true temperature difference of 113.774 °F, indicating an adequate thermal driving force. The flow on the shell side is in the turbulent regime ($Re \approx 79{,}952$), while the tube side shows laminar flow ($Re \approx 829$), which limits the optimization of convective heat transfer of water as a cooling medium. The overall heat transfer coefficient under clean conditions of 25.627 Btu/h.ft².°F decreased to 23.125 Btu/h.ft².°F due to a fouling factor (Rd) of 0.004. The pressure drop values on both sides are still well below the permitted limits, so the unit is declared operationally feasible. However, the presence of fouling and laminar flow regime on the tube side indicates the potential for performance improvement through optimization of water flow rate and periodic cleaning. This study emphasizes the importance of periodic performance evaluation to maintain the efficiency and reliability of heat exchanger operations in air compression systems.

1. INTRODUCTION

Heat exchanger systems are crucial components in industrial processes that facilitate heat transfer between two or more fluids to cool or heat the process fluid to a desired temperature. The implementation of this technology is vital to meet regulatory standards, save operational costs, and improve energy efficiency in various sectors, such as power plants, oil refineries, air conditioning, refrigeration, waste heat recovery systems and automotive cooling systems. Some heat exchangers found in the industrial sector include heater, evaporators, condensers, economizers, and cooling towers [1], [2], [3]. In practice, the heat exchanger network configuration is designed to maximize heat recovery to reduce primary energy consumption [4].

PT Y is an industrial gas manufacturing company that produces oxygen (O_2), nitrogen (N_2), and argon (Ar) in liquid phase through an Air Separation Unit (ASU). In its production cycle, ambient air is used as the main raw material. Ambient air intake is carried out through a compression system to increase air pressure before entering the cooling, purification, and separation stages of compressed air. Thermodynamically, this pressure increase process is followed by a significant increase in air temperature due to the heat of compression. To maintain energy efficiency and system performance, PT Y operates a three-stage compressor integrated with two intercooler units between each compression stage. The intercooler functions to lower the temperature of the compression results at each stage before entering the next compression stage to approach isothermal compression conditions, thereby minimizing compression work in the next stage and preventing thermal damage to engine components [5].

The intercooler used in the air compression system at PT Y is a shell and tube type heat exchanger. Its main components are the shell, tube bundle, and baffle. The use of baffles is designed to direct the fluid flow on the shell side to create turbulence that effectively expands the contact and heat transfer rate [6]. The heat transfer performance of the intercooler is highly dependent on the heat transfer surface area and the mass flow rate of the fluid flowing on both sides [7].

Several previous studies have evaluated intercooler performance using various parameters, including heat transfer rate, overall heat transfer coefficient, cooling capacity, effectiveness, and pressure drop. For example, study [8] investigated intercooler performance under laboratory-based design conditions with controlled variations in flow rate, but it did not consider fouling problems that may occur during long-term operation. Study [9] developed a numerical model to predict thermal effectiveness; however, the model was not validated using actual field data. Meanwhile, study [10] focused on pressure drop evaluation in industrial-scale heat exchangers, but the analysis was still based on simulation and did not include the effect of dirt factor accumulation during operation. These studies provide important insights into the parameters that influence intercooler performance. However, their evaluations were mostly conducted under controlled conditions, either in laboratory experiments or numerical simulations. This condition is different from actual industrial operation, where an intercooler may run continuously for months or even years. During long-term operation, fouling can gradually accumulate on the heat transfer surfaces and change the thermal performance of the equipment. Therefore, the deviation between actual intercooler performance and its original design specifications, especially in air compression systems for Air Separation Units (ASU), still needs further investigation. This issue is important because without periodic performance evaluation, fouling and performance degradation may remain undetected. As a result, heat transfer effectiveness can decrease, compressor work may increase, and the overall energy efficiency of the ASU system can be reduced.

Evaluation of the intercooler is crucial because a decrease in heat absorption performance not only reduces production quality and quantity but also risks shortening the equipment's lifespan due to excessive operating temperatures. This study focuses on the performance of intercooler-2 in the air compressor unit (C-10) because this unit is considered one of the most critical parts of the system and is located at the early stage of the air compression process. If performance problems occur at this initial stage, they may affect the performance of subsequent equipment and reduce the overall efficiency of the compression system. Therefore, the evaluation of intercooler-2 is expected to serve as an initial assessment for further evaluation of other related units. The performance analysis is carried out through the calculation of the overall heat transfer coefficient in clean conditions (U_c), the actual operating heat transfer coefficient (U_d), dirt factor (R_d), and pressure drop (ΔP). Analysis of these parameters is expected to provide a quantitative picture of the condition of intercooler-2, so that it can be used as a basis for making technical decisions related to operational optimization and equipment maintenance strategies.

2. MATERIALS AND METHODS

This study was conducted using a case study approach on Intercooler-2 of the Air Compressor C-10 unit at PT Y, a shell-and-tube heat exchanger with a one-shell-pass and one-tube-pass (1-1) configuration, see Figure 1. This approach was selected to evaluate the actual performance of the equipment against its ideal clean condition by comparing the overall heat transfer coefficient under clean conditions (U_c) and operating conditions (U_d), as commonly applied in performance evaluation studies of industrial heat exchangers

The data used in this study consist of field operating data recorded under steady-state conditions, including the inlet and outlet temperatures of air on the shell side, the inlet and outlet temperatures of cooling water on the tube side, and the air mass flow rate. The equipment geometry specifications, including the shell inside diameter, baffle spacing, number of tubes, tube outside diameter, and tube thickness based on the Birmingham Wire Gauge (BWG) standard, were obtained from the manufacturer’s technical datasheet. A summary of the geometry specifications and operating data is presented in Table 1 and Table 2.

Table 1. Geometry Specifications of Intercooler-2

Parameter	Value
Configuration	1 shell pass – 1 tube pass
Shell inside diameter (ID)	21,5 in
Baffle spacing	5 in
Number of tubes	200
Tube outside diameter (OD)	0,675 in
Tube thickness (BWG)	18

Table 2. Operating Data of Intercooler-2

Parameter	Value
Air inlet temperature, T_1	273,20 °F
Air outlet temperature, T_2	187,40 °F

Water inlet temperature, t_1	89,60 °F
Water outlet temperature, t_2	141,80 °F
Air mass flow rate, W	27.080,45 lb/h

The performance evaluation was carried out following the calculation steps of the Kern method [11]. The energy balance, or heat balance, was calculated on both fluid sides to verify the equilibrium between the heat absorbed and the heat released (Equation 1), while the effective temperature difference (ΔT) was obtained by multiplying the logarithmic mean temperature difference (LMTD) by the correction factor FT (Equation 2).

$$Q_{hot\ fluid} = Q_{cold\ fluid} = WC(T_1 - T_2) = wc\ (t_1 - t_2) \tag{1}$$

$$\Delta T = LMTD \times F_T \tag{2}$$

The caloric temperatures on the shell side (T_c) and tube side (t_c) were calculated as the average of the inlet and outlet temperatures of each fluid (Equations 3 and 4), which were used as the basis for determining the physical properties of the fluids in the subsequent calculation stages.

$$T_c = \frac{T_1 + T_2}{2} \tag{3}$$

$$t_c = \frac{t_1 + t_2}{2} \tag{4}$$

The flow area on the shell side (a_s) was determined from the shell inside diameter (ID), baffle spacing (B), and tube pitch (Pt) (Equation 5), while the tube-side flow area (a_t) was determined from the number of tubes (Nt), tube cross-sectional area (a′t), and number of tube passes (n) (Equation 6). The mass velocities on both sides ($\mathrm{G_s}$ and $\mathrm{G_t}$) were obtained by dividing the mass flow rate by the respective flow area (Equations 7 and 8), which were then used to calculate the Reynolds numbers (Re_s and Re_t) based on the equivalent diameter or tube diameter and the fluid viscosity (Equations 9 and 10).

$$a_s = \frac{ID \times C' \times B}{144 \times Pt} \tag{5}$$

$$a_t = \frac{N_t \times a'_t}{144 \times n} \tag{6}$$

$$\mathrm{G_s} = \frac{W}{\mathrm{a_s}} \tag{7}$$

$$\mathrm{G_t} = \frac{w}{\mathrm{a}_t} \tag{8}$$

$$Re_s = \frac{De \times \mathrm{G_s}}{\mu} \tag{9}$$

$$Re_t = \frac{D \times \mathrm{G_t}}{\mu} \tag{10}$$

The film coefficients on the shell side (h_o) and tube side (h_i) were calculated from the j_H factor, which was obtained based on the Reynolds number, thermal conductivity, and Prandtl number of the fluid, with wall viscosity corrections φ_s and φ_t (Equations 11 and 12). The h_i coefficient was then converted to an outside tube diameter basis as h_{io} (Equation 13). The tube wall temperature ($\mathrm{t_w}$) was calculated from the balance of the film coefficients on both sides (Equation 14).

$$h_o = j_H \times \frac{k}{De} \times \left(\frac{c_p\, \mu}{k}\right)^{\frac{1}{3}} \times \varphi_s \tag{11}$$

$$h_i = j_H \times \frac{k}{D} \times \left(\frac{c_p\, \mu}{k}\right)^{\frac{1}{3}} \times \varphi_t \tag{12}$$

$$h_{io} = h_i \times \frac{ID}{OD} \tag{13}$$

$$\mathrm{t_w} = t_c + \frac{\frac{h_o}{\varphi_s}}{\frac{h_{io}}{\varphi_t} + \frac{h_o}{\varphi_s}} \times (T_c - t_c) \tag{14}$$

The overall heat transfer coefficient under clean conditions (U_c) was obtained from the combination of h_{io} and h_o (Equation 15), while the overall heat transfer coefficient under actual operating conditions (U_d) was calculated from the heat duty (Q), heat transfer surface area (A), and ΔT (Equation 16). The difference between U_c and U_d represents the dirt factor (R_d), which serves as an indicator of the fouling level on the equipment surface (Equation 17).

$$U_c = \frac{h_{io} h_o}{h_{io} + h_o} \tag{15}$$

$$U_d = \frac{Q}{A\Delta t} \tag{16}$$

$$R_d = \frac{U_C - U_d}{U_C \times U_d} \tag{17}$$

The pressure drop on the shell side (ΔP_s) was calculated based on the friction factor (f), mass velocity (G_s), shell diameter (D_s), number of baffles (N), and equivalent diameter (D_e) (Equation 18). On the tube side, the pressure drop due to friction along the tube (ΔP_t) was calculated from the friction factor, mass velocity (G_t), tube length (L), and number of passes (n) (Equation 19), plus the pressure loss caused by flow reversal in the return bend (ΔP_r), which depends on the number of passes and the fluid linear velocity (V) (Equation 20). The total tube-side pressure drop (ΔP_T) is the sum of ΔP_t and ΔP_r (Equation 21), which was then compared with the allowable ΔP limit specified in the design specifications.

$$\Delta P_s = \frac{f \times {G_s}^2 \times D_s \times (N+1)}{5{,}22 \times 10^{10} \times D_e \times s \times \varphi_s} \tag{18}$$

$$\Delta P_t = \frac{f \times {G_t}^2 \times L \times n}{5.22 \times 10^{10} \times D \times \varphi_t} \tag{19}$$

$$\Delta P_r = \frac{4n}{s} \times \frac{V^2}{2g} \tag{20}$$

$$\Delta P_T = \Delta P_t + \Delta P_r \tag{21}$$

The performance of Intercooler-2 was categorized as good if: (i) the calculated R_d did not exceed the standard R_d, (ii) the Ud/Uc ratio was close to one, indicating the cleanliness percentage of the heat transfer surface, and (iii) the actual ΔP remained within the design limits. Any deviation from one of these criteria indicates the need for cleaning or further evaluation of the equipment operating conditions.

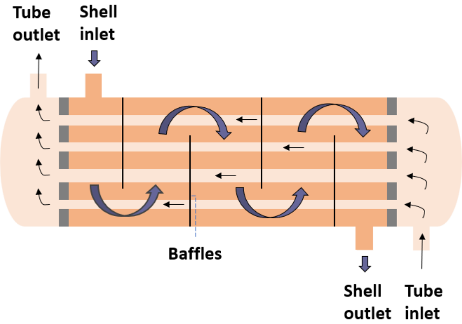

Figure 1. Schematic diagram of the 1–1 shell-and-tube Intercooler-2 showing the shell-side and tube-side flow configurations

3. RESULTS AND DISCUSSION

The thermal performance of Intercooler-2 on the C-10 Air Compressor is summarized in Table 3, which is evaluated based on the results of heat balance calculations, temperature profiles, flow characteristics, and overall heat transfer coefficients. The heat balance results show a heat transfer rate of 557,640.622 Btu/h, which means that the heat exchanger can handle the thermal load of the compressed air before it goes on to the next stage of compression. This value is in line with the true temperature difference of 113.774 °F, which indicates a strong thermodynamic driving force for heat transfer, thus ensuring energy exchange between the hot fluid on the shell side and cold fluid on the tube side. The average calorific temperature of the hot fluid is 230.3 °F and that of the cold fluid is 115.7 °F, indicating a large temperature gradient along the heat exchanger. This condition supports effective heat transfer from the hot fluid to the cold fluid. Meanwhile, the pipe wall temperature obtained is 179.029 °F, which is between the temperatures of the hot and cold fluids, confirming the occurrence of stable conductive heat transfer through the pipe material. This temperature level does not cause significant concerns regarding thermal stress and is still within the safe operating limits for conventional metal materials, thus ensuring the reliability of the structure under steady-state conditions.

Table 3. Results of Intercooler-2 in Air Compressor C-10

No	Parameters	Results
1	Heat balance	$Q = 557640.622$ BTU/h
2	True temperature difference	$\Delta t = 113.774\ °F$
3	Caloric temperature	
	a. Hot fluid: shell side	$T_c = 230.3\ °F$
	b. Cold fluid: tube side	$t_c = 115.7$ °F
4	Flow area	
	a. Hot fluid: shell side	$a_s = 0.485\ ft^2$
	b. Cold fluid: tube side	$a_t = 0.427 ft^2$
5	Mass velocity	
	a. Hot fluid: shell side	$\mathrm{G_s} = 55{,}832.527 \frac{\mathrm{lb}}{\mathrm{h}.ft^2}$
	b. Cold fluid: tube side	$\mathrm{G_t} = 25003.343\ \frac{\mathrm{lb}}{h.ft^2}$
6	Reynold number	
	a. Hot fluid: shell side	$Re_s = 79952.198$
	b. Cold fluid: tube side	$Re_t = 829.206$
7	Heat transfer coefficient	
	a. Hot fluid: shell side	$h_o = 44.312$
	b. Cold fluid: tube side	$h_i = 38.683$
8	Inside coefficient based on OD	$h_{io} = 35.875$
9	Tube wall temperature	$t_w = 179.029$
10	Clean overall coefficient	$U_c = 25.627\ \frac{Btu}{h.ft^2.°\mathrm{F}}$
11	Design overall coefficient	$U_d = 23.125\ \frac{Btu}{h.ft^2.°\mathrm{F}}$
12	Dirt factor	$R_d = 0{,}004$
13	Pressure drop	
	a. Hot fluid: shell side	$\Delta P_s = 0{,}00347\ psi$
	b. Cold fluid: tube side	$\Delta P_t = 0{,}160\ psi$
		$\Delta P_r = 0.00026$
		$\Delta P_T = 0{,}16026\ psi$

Furthermore, the cross-sectional areas of the hot and cold fluid flows are 0.485 ft² on the shell side and 0.427 ft² on the tube side, which affects the fluid velocity. The mass velocities are 55,832.527 lb/h.ft² for the shell side and 25,003.343 lb/h.ft² for the tube side. These values result in significantly different flow regimes. The Reynolds number of 79,952.198 on the shell side shows that the flow is fully turbulent. Turbulent flow generally enhances convective heat transfer because it increases fluid mixing and disturbs or thins the thermal boundary layer near the heat transfer surface, thereby reducing thermal resistance between the bulk fluid and the tube wall [12]. Nevertheless, the shell-side heat transfer coefficient is only 44.312 Btu/h.ft².°F. This moderate value indicates that turbulence alone does not guarantee a high heat transfer coefficient. In this case, the shell-side fluid is air, which has relatively low thermal conductivity and low heat capacity compared with liquid coolants. Therefore, even though the flow is turbulent, the ability of air to absorb and transfer heat remains limited.

In contrast, the Reynolds number on the tube side is 829.206, indicating laminar flow inside the tube. Under laminar conditions, the fluid moves in relatively smooth layers, and mixing between the near-wall fluid and the core flow is limited. As a result, the thermal boundary layer remains more stable and creates additional resistance to convective heat transfer. This condition is reflected in the tube-side heat transfer coefficient of 38.683 Btu/h·ft²·°F and the inside coefficient based on the outside diameter of 35.875 Btu/h·ft²·°F. Although water has better thermal properties than air and can potentially produce a higher heat transfer coefficient, this potential is not fully utilized because the flow remains in the laminar regime. Therefore, the tube side may become one of the limiting factors in the thermal performance of the intercooler.

From a design perspective, improving the tube-side flow condition could enhance the overall heat transfer performance. Increasing the cooling-water flow rate or modifying the tube-side configuration may increase the Reynolds number and promote transition toward turbulent flow. However, this improvement must be evaluated carefully because increasing the flow rate will also increase the pressure drop and pumping power requirement. Therefore, optimization should not only aim to increase the heat transfer coefficient, but also maintain a reasonable balance between thermal performance and energy consumption .

Under clean conditions, the overall heat transfer coefficient was determined to be 25.627 Btu/h.ft².°F. This value represents the expected performance of the heat exchanger when the heat transfer surface is still clean and free from deposits. However, after considering the design dirt factor of 0.004, the overall heat transfer coefficient decreased to 23.125 Btu/h.ft².°F. This decrease shows that additional thermal resistance may occur due to fouling on the heat transfer surface Fouling is more likely to occur on the cooling-water side because cooling water may contain dissolved minerals, suspended particles, corrosion products, and microorganisms. These contaminants can form scale, particulate deposits, corrosion fouling, or biological fouling on the inner tube surface [13]. However, fouling on the air side should not be completely ignored, especially if the compressed air contains oil mist, dust, or other particulates from compressor operation.

Since Intercooler-2 has been operating for approximately four years, impurities carried by the fluid may gradually accumulate on the inner and outer surfaces of the tubes. The fouling process generally begins when impurities are transported to the heat transfer surface, adhere to the surface, and slowly form a deposit layer. As this layer becomes thicker, it increases thermal resistance, reduces the effective flow area, and weakens the heat transfer between the hot and cold fluids. Consequently, the heat transfer capacity of the intercooler decreases and its actual performance becomes lower than the clean-condition performance [13]. Therefore, the difference between the clean overall coefficient and the design overall coefficient should be interpreted as an indication that fouling resistance has a measurable effect on the intercooler performance.

The calculated pressure drops of Intercooler-2 are 0.00347 psi on the shell side and 0.160 psi on the tube side, which are far below the allowable limits of 2 psi for gas-phase flow and 10 psi for liquid-phase flow. This indicates that the intercooler is still hydraulically acceptable for operation. However, the low tube-side pressure drop is also associated with low fluid velocity and laminar flow, which limits convective heat transfer. Therefore, increasing the cooling-water flow rate may improve the Reynolds number, enhance fluid mixing, and increase the heat transfer coefficient, although this must be balanced with the possible increase in pressure drop and pumping power

The dirt factor analysis shows that Intercooler-2 exceeds the recommended permissible value, indicating the presence of fouling resistance on the heat transfer surface. Fouling can reduce the overall heat transfer coefficient by adding thermal resistance between the hot air and cooling water. A comparable study was performed by Aliyah Shahab and Anggi Wahyuningsi (2023), who assessed heat exchanger-03 at the Cepu Oil and Gas Production Plant (PPSDM Migas) by the calculation of the fouling factor [14]. The dirt factor value exceeded the recommended threshold. The value surpassed the advised limit. Consistent cleaning and adjustment of flow rate distribution can alleviate fouling effects and restore unit efficiency. Alongside regular cleaning, regulating the flow rate distribution helps mitigate fouling effects and enhance unit performance [15].

4. CONCLUSION

The performance evaluation of Intercooler-2 on the C-10 Air Compressor shows that the unit is still able to transfer heat from compressed air to cooling water, with a heat transfer rate of 557,640.622 Btu/h and a true temperature difference of 113.774 °F. The shell-side flow is turbulent, as shown by the Reynolds number of

79,952.198, while the tube-side flow is laminar with a Reynolds number of 829.206. Although the intercooler remains functional, the laminar flow on the tube side limits convective heat transfer and becomes one of the factors affecting the overall thermal performance.

The overall heat transfer coefficient decreases from 25.627 Btu/h.ft².°F under clean conditions to 23.125 Btu/h.ft².°F after considering fouling resistance. This indicates that fouling has affected the heat transfer surface, supported by the dirt factor value that exceeds the recommended permissible limit. Meanwhile, the pressure drops on both shell and tube sides remain below the allowable limits, meaning the intercooler is still hydraulically safe for operation. Therefore, regular cleaning, dirt factor monitoring, and optimization of the cooling-water flow rate are recommended to maintain the thermal performance and reliability of Intercooler-2.

List of Notations

A	Heat transfer surface, ft^2
a_s	Flow area of shell, ft^2
a_t	Flow area of tube, ft^2
a'_t	Flow area per tube, ft^2
B	Baffle spacing, in
c_p	Specific heat of fluid, Btu/lb.°F
C'	Clearence between tubes, in
D	Inside diameter of tube, ft
De	Equivalent diameter of shell
D_s	Inside diameter of shell, ft
f	Friction factor, dimensionless
F_T	Temperature differences factor, dimensionless
G_s	Mass velocity of shell, lb/h.ft^2
G_t	Mass velocity of tube, lb/.ft^2
h_i, h_o	Heat transfer coefficient in general, for inside fluid, and for outside fluid, respectively, Btu/h.ft^2.°F
h_{io}	Value of h_i when referred to the tube outside diameter, Btu/h.ft^2. °F
ID	Inside diameter, in
j_H	Factor for heat transfer, dimensionless
k	Thermal conductivity, Btu/h.ft^2.°F
L	Tube length, ft
$LMTD$	Log mean temperature differences, °F
N	Number of shell-side baffles
N_t	Number of tubes
n	Number of tube passes
OD	Outside diameter, in
Pt	Tube pitch, in
Q	Heat flow, Btu/h
R_d	Dirt factor, h.ft^2.°F/Btu
Re_s	Reynold number of shell, dimensionless
Re_t	Reynold number of tube, dimensionless
s	Specific gravity
T_c	Caloric temperature of hot fluid, °F
T_1	Hot fluid temperature in, °F
T_2	Hot fluid temperature out, °F
t_c	Hot fluid temperature in, °F
t_w	Tube wall temperature, °F
t_1	Cold fluid temperature in, °F
t_2	Cold fluid temperature out, °F
U_c	Clean coefficient of heat transfer, Btu/h.ft^2.°F
U_D	Design coefficient of heat transfer, Btu/h.ft^2.°F
V	Velocity, fps
W	Weight flow of hot fluid, lb/h
w	Weight flow of cold fluid, lb/h
$\Delta P_T, \Delta P_t, \Delta P_r$	Total, tube side and return pressure drop, respectively, psi
ΔT	True temperature difference, °F

μ	Viscosity, centipoises × 2.42 = lb/ft.h
φ_s	The viscosity ratio of shell, $(\mu/\mu_w)^{0.14}$
φ_t	The viscosity ratio of tube, $(\mu/\mu_w)^{0.14}$

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