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ENERGY SAVINGS IN ALKYD-RESIN AND AQUEOUS-EMULSION PROCESSES USING A HEAT-INTEGRATED CYCLE

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ABSTRACT

In paint industries, synthesis of alkyd resins and aqueous emulsions consumes high energy levels required for occurrence of endothermic chemical reactions. In attempt to reduce energy consumption, the present work investigated energy integration of the heating cycles of aqueous-emulsion and alkyd-resin production lines that operate independently. The energy-integrated system between these two processes was built by removing the less efficient boiler of the aqueous-emulsion process and using the more efficient boiler of the alkyd resin for both processes. A shell-and-tube heat exchanger was applied to integrate both heating cycles. In order to maintain the production levels of the integrated processes, design calculations showed that the required surface area, based on the pipes outside diameter of the heat exchanger, must not be less than 22 m^2. Furthermore, results indicated the positive role of the heat-integrated cycle in energy saving of around 12%.

Keywords: *energy integration, energy saving, alkyd resin, aqueous emulsion, heating cycles.*

INTRODUCTION

Nowadays where energy prices are increasing rapidly, due to population growth as well as improvement of living standards, energy conservation projects have become more important than ever [1-5]. In particular, paint industries are energy intensive processes whereas the incidence of energy cost is about 20% of the total production cost [6, 7]. Therefore, the recovery of relatively small quantities of heat can accumulate to become sizeable energy savings [1]. The highly energy consuming nature of such industrial processes is the key driving force for improving its profitability and reducing the overall cost of manufacturing [7, 8]. Hence, any attempt for energy conservation in the process goes a long way in many aspects. Chemical reactions involved in the synthesis of alkyd resin and emulsions are endothermic and thus they need a continuous heat supply [8-12]. In this regard, process system engineering plays an important role in the energy-saving task [1, 4].

The objective of the present study was to minimize the energy consumption associated with the production of alkyd resin and aqueous emulsion. As a case study, energy integration concept was applied on the alkyd-resin and aqueous-emulsion production lines of a Jordanian paints factory which is known as High Quality Company. In this company, the two processes operate independently without any heat recovery network. Each production line uses a boiler to generate the utility streams required for accomplishing the endothermic chemical reactions. Fig. 1 shows the alkyd-resin process which uses one boiler to generate the utility stream of hot thermal oil which in turn enters the jackets of two reactors. This is to maintain the reactors operating temperature in the range of 200-240^{o}C. On the other hand, Fig. 2 illustrates the aqueous-emulsion process in which a second boiler is used to generate a utility stream of hot water to feed the jacket of the emulsifier. This is to maintain its operating temperature in the range of 90-95^{o}C. Metal coils are immersed inside these reactors for cooling purposes of the products, whereas liquid water comes from the cooling tower is used as a cooling medium. The processing time is about 20 and 10 hours for completion of one batch of alkyd-

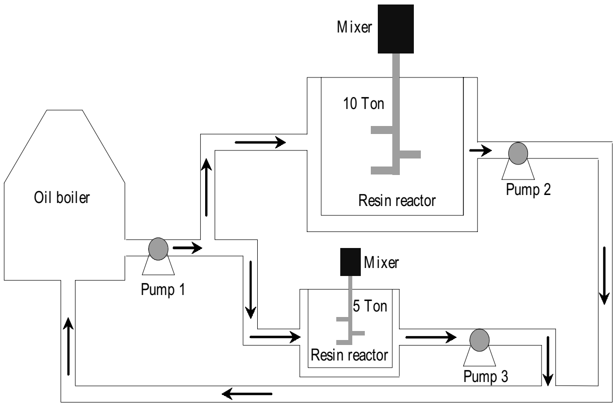

Fig.1. Schematic diagram of the heating cycle for alkyd-resin process.

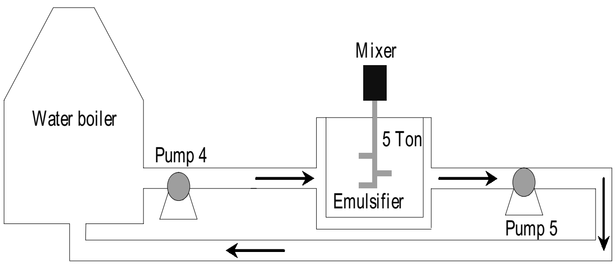

Fig. 2. Schematic diagram of the heating cycle for aqueous-emulsion process.

resin and aqueous-emulsion product, respectively. The batch is equivalent to 15 and 5 tons for alkyd resin and aqueous emulsion, respectively.

METHODOLOGY

Fig. 3 presents the heat-integrated cycles of the alkyd-resin and aqueous-emulsion processes. As shown in the figure the boiler of the aqueous-emulsion process was removed and a shell-and-tube heat exchanger was introduced. The energy integrated system was conducted according to the following stages. First, the operating conditions of the independent production lines shown in Figs. 1 and 2 were monitored. Second, the heat required for the aqueous-emulsion process was evaluated in order to know the heat load on the heat exchanger, thereby, the additional heat load on the boiler of alkyd-resin process. Third, efficiencies of boilers were assessed. Fourth, fluid flow rates and heat-transfer calculations were incorporated in the detailed design stage of the heat exchanger. Finally, energy consumption levels of both non-integrated and integrated processes were evaluated and compared to see the benefit of the adopted energy integration system.

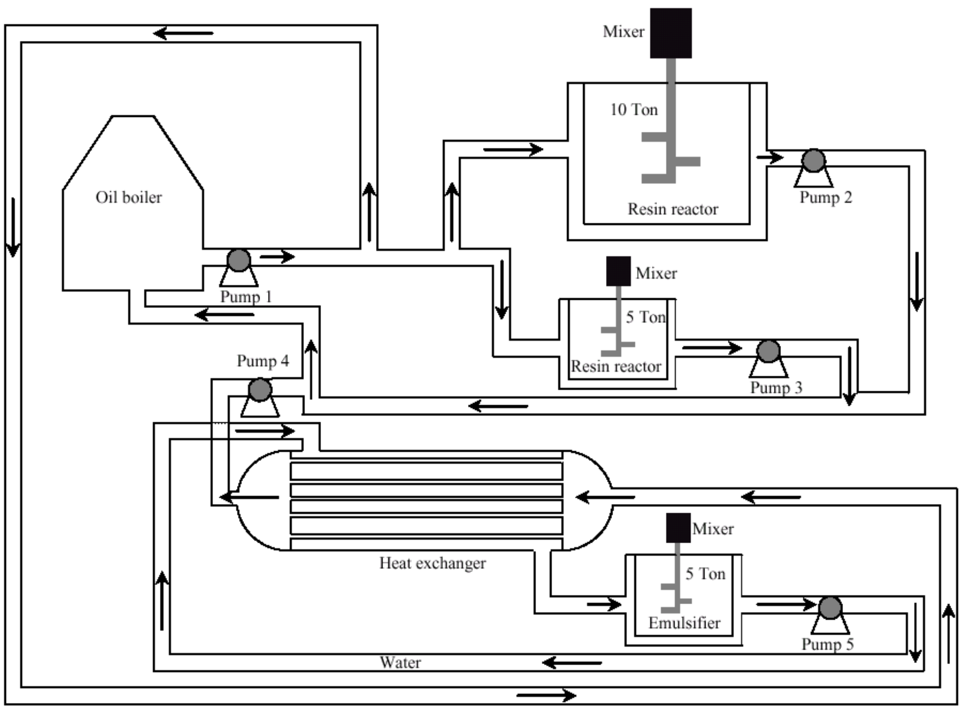

Fig. 3. Schematic diagram of the heat-integrated cycle for alkyd-resin and aqueous-emulsion processes.

RESULTS AND DISCUSSION

In the non-integrated process shown in Fig. 2, water which comes from the jacket of emulsifier will enter the boiler at a temperature of around T_1 = 90°C and it is heated to reach a temperature of around T_2 = 95°C. It was monitored that the heated water flows at a volumetric rate of $\dot{V}$ = 51.4 $m^3 h^{-1}$ which is equivalent to a mass flow rate $\dot{m}$ = 13.76 kg s^{-1} since water average density is about ρ_w = 963.7 kg m^{-3}. The rate of heat transferred from boiler to water can be calculated from the following sensible heat equation:

$$\dot{Q} = \dot{m} C_p (T_2 - T_1) \quad (1)$$

where C_p is the mean heat capacity of water which is estimated at the average temperature of around T_{avg}= 92.5 °C to have a value of around C_p = 4203 J kg^{-1} $°C^{-1}$). Substituting all known quantities into Eq. (1) gives a heat rate of around $\dot{Q}_w$ = 289 kW. Furthermore, the efficiency of the aqueous-emulsion boiler can be calculated using:

$$\eta = \frac{\dot{Q}}{\dot{V} \times GHV} \quad (2)$$

where $\dot{Q}$ is the rate of heat transferred from boiler to the operating fluid (water) passes through the heating cycle, $\dot{V}$ is the volumetric rate of diesel consumed, and *GHV* is the gross heating value of the fuel introduced to the boiler. The two boilers considered in this work use a diesel as fuel. The gross heating value of diesel is *GHV*= 38.6 MJ L^{-1}. It was recorded that the aqueous-emulsion boiler consumes around 1320 litres of diesel per day, i.e., $\dot{V}$ = 1.53x10^{-2} L s^{-1}. Now, applying Eq. (2) gives a boiler efficiency of about η= 0.49.

On the other hand, the boiler of alkyd-resin process, shown in Fig. 1, consumes around 2617 litres of diesel per day, i.e, $\dot{V}$ = 3.03X10^{-2} L s^{-1}, to heat thermal oil from T_1 = 220°C to T_2 = 240°C. At the average temperature of T_{avg} = 230°C, the corresponding physical properties of thermal oil are approximately as follows: the oil heat capacity is C_{po} = 2769 J kg^{-1} $°C^{-1}$ and the oil density is ρ_o = 770 kg m^{-3}.The thermal oil enters the jacket of the 5 tons and 10 tons alkyd-resin reactors with a volumetric flow rate of 32 and 45 $m^3 h^{-1}$, respectively. Hence, the oil enters the alkyd-resin boiler with an overall flow rate of 77 $m^3 h^{-1}$ which is equivalent to mass flow rate of $\dot{m}_o$ = 16.47 kg s^{-1}. The rate of heat transferred from alkyd-resin boiler to the thermal oil can be estimated using Eq. (1) to have a value of $\dot{Q}_o$ = 912 kW. By using Eq. (2), the corresponding alkyd-resin boiler efficiency is η= 0.78. Since the boiler efficiency of alkyd resin is larger than the corresponding one of the aqueous-emulsion process, the integrated-heat cycle was built by removing the aqueous-emulsion boiler so that the heat duty of 289 kW will be added to the alkyd-resin boiler. This results in an overall heat rate of 1201 kW. While, the non-integrated processes consume 3937 litres of diesel per day, it can be demonstrated using Eq. (2) that the amount of diesel consumed in the integrated process is 3446 litres per day. Therefore, a diesel saving of around 12 % was achieved in the integrated energy system. Note that these processes operate 300 day per year. Hence, the amount of diesel consumed in the non-integrated processes is 1181 m^3 per year and it is 1034 m^3 per year for the integrated one. This leads to an annual diesel saving of around 147 m^3 (Fig. 4).

Design computations of the heat exchanger

In the integrated process, the heat required for emulsifier reactions is supplied via a stream of water which leaves the shell side of the heat exchanger and

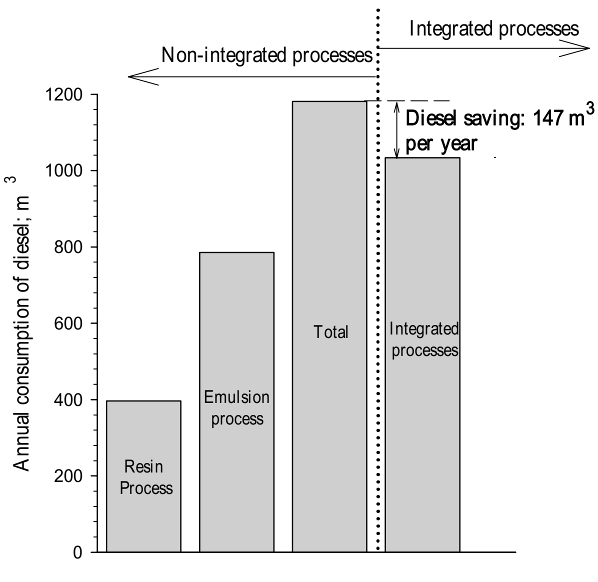

Fig. 4. Comparison between non-integrated and integrated processes of the Jordanian paint company (High Quality Company; Amman). Note that 1year = 300 operating days.

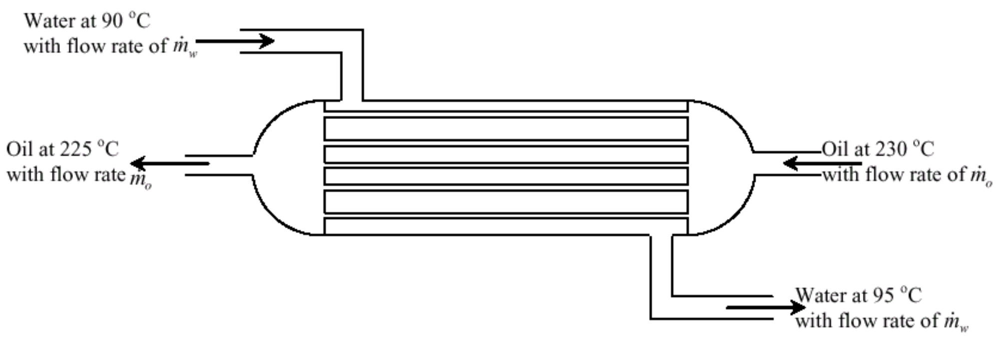

Fig. 5. Schematic diagram of the heat exchanger with its inlet and outlet streams.

enters the jacket of the emulsifier at temperature of T_{s1}=95°C (Figs. 3 and 5). This stream leaves the jacket at temperature T_{s2}=90°C. Then, it is preheated in the heat exchanger via a stream of thermal oil, coming from the heating cycle at temperature of T_{p1}=230°C, to feed the pipes of the heat exchanger. This oil stream leaves the heat exchanger at temperature of about T_{p2}=225°C and it is re-circulated to the oil boiler. In order to enhance the performance of heat exchanger, the shell-side stream is chosen to be in a counter current flow with the pipes-side stream [13].

The basic design equation used to estimate the heat-transfer surface area, A_o, based on the outside diameter of the tubes, d_o, is [14]:

$$\dot{Q} = U_o A_o \Delta T_m \tag{3}$$

where ΔT_m is the mean temperature difference defined as [14]:

$$\Delta T_m = T_{m,p} - T_{m,s} \tag{4}$$

where $T_{m,p}$ is the mean temperature in the pipe-side which is equal to $T_{m,p}$ = 227.5°C and $T_{m,s}$ is the mean temperature in the shell-side which is equal to $T_{m,s}$ = 92.5°C. This leads to a mean temperature difference of ΔT_m=135 °C. The overall heat-transfer coefficient, U_o, is based also on outside pipe diameter. It can be obtained from the following equation [9]:

$$\frac{1}{U_o A_o} = \frac{1}{h_p A_i} + \frac{1}{h_s A_o} + \frac{1}{h_{do} A_o} + + \frac{1}{h_{di} A_i} + \frac{\ln(d_o / d_i)}{2\pi k_{steel} L} \tag{5}$$

where h_p is the forced-convection heat-transfer coefficient in the pipes-side, h_s is the forced-convection heat-transfer coefficient in the shell side, A_i is the total heat-transfer area based on the total inside diameter of the pipe, d_i, L is the pipe length, h_{do} is the fouling(dirty) coefficient for water, h_{di} is the fouling(dirty) coefficient for thermal oil, and k_{steel} is the thermal conductivity of the carbon steel selected to be the material of construction of both shell and pipes (k_{steel}=43 W m^{-1} °C^{-1}).

The thermal oil enters the pipes of the exchanger at temperature T_{p1} = 230°C and leaves at temperature, T_{p2}=225°C. Physical properties of the thermal oil at the average temperature, T_{mp} = 227.5°C, are : oil density is ρ_o=768.60 kg m^{-3}, oil heat capaty is C_{po}=2780 J kg^{-1} °C^{-1}, oil viscosity is μ_o=6.94x10^{-4} kg m^{-1} s^{-1} , and oil thermal conductivity is k_o=0.124 W m^{-1} °C^{-1}. The mass flow rate of the oil, calculated using the sensible heat equation, is $\dot{m}_o$ = 20.78 kg s^{-1} and the corresponding volumetric flow rate is $\dot{V}$ = 2.70x10^{-2} m^3 s^{-1}. Table 1 presents the dimensions of steel pipes and shell used to design the heat exchanger. For one-pass of tubes with a square pitch of 0.0254 m, the number of tubes in this exchanger is n = 137 [14]. The average velocity of the oil in each pipe is:

$$u_p = \frac{4\dot{V}}{n\pi d_i^2} = 1.6083 \mathrm{m^2 s^{-1}} \tag{6}$$

Table 1. Dimensions of steel pipes and shell used to design the heat exchanger.

Side	Nominal size (in)	Inside diameter (m)	Outside diameter (m)
Schedule 40 steel pipe	3/8	d_i = 0.0125	d_o = 0.0171
Schedule 40 steel shell	16	D_i = 0.381	D_o = 0.406

The pipe-side Reynolds number, Re_p is:

$$\mathrm{Re}_p = \frac{\rho_o u_p d_i}{\mu_o} = 22265 \tag{7}$$

and the Prandtl number of oil is:

$$\mathrm{Pr}_o = \frac{C_{po}\mu_o}{k_o} = 15.56 \tag{8}$$

Since the flow in pipes is turbulent (Re_p>10000), the following relation can be used to determine the heat-transfer coefficient, h_p, [13]:

$$Nu = \frac{hd}{k} = 0.027\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{9}$$

where Nu is the average Nusselt number. Hence, the forced-convection heat-transfer coefficient in the pipes is $h_p = 2414.23\ \mathrm{W\,m^{-2}\ {}^oC^{-1}}$.

On the other hand, water enters the shell of the exchanger at temperature T_{s1} = 95°C and leaves at temperature, T_{s2} = 90°C. At this average temperature, the physical properties of liquid water were approximated at the average temperature, T_{ms} = 92.5°C [13]. Water density is ρ_w= 963.7 kg m^{-3}, water heat capacity is C_{pw}= 4203 J kg^{-1} °C^{-1} , water viscosity μ_w=3.09x10^{-4} kg m^{-1} s^{-1}, water thermal conuctivity is k_w=0.677 W m^{-1} °C^{-1}. The mass flow rate of water in the shell side is $\dot{m}_w$ = 13.76 kg s^{-1} and its volumetric flow rate is of $\dot{V}$ =51.4 m^3 h^{-1}. The cross-sectional area available for this flow is:

$$A_s = \frac{\pi}{4}\left(D_i^{\,2} - nd_o^{\,2}\right) = 0.08254\,\mathrm{m}^2 \tag{10}$$

and the average velocity in shell side is:

$$u_s = \frac{\dot{V}}{A_s} = 0.1728\ \mathrm{m\,s^{-1}} \tag{11}$$

Since water flows in the region enclosed between outside-surface of the pipes and the inside surface of the shell, equivalent diameter must be calculated to be used in estimation the she-side heat-transfer coefficient, h_s, as [13]:

$$D_{eq} = \frac{4A_s}{W} \tag{12}$$

where W is the wetted perimeter defined as:

$$W = \pi\left(D_i + nd_o\right) = 8.5568\,\mathrm{m} \tag{13}$$

This gives an equivalent diameter of $D_{eq} = 0.0386\mathrm{m}$. Then, the shell-side Reynolds number, Re_s is found as:

$$\mathrm{Re}_s = \frac{\rho_w u_s D_{eq}}{\mu_w} = 20792 \tag{14}$$

and Prandtl Number of water is about $\mathrm{Pr}_w = 1.92$. Eq. (9) is now used to give a shell-side heat-transfer coefficient of about $h_s = 1750.76\,\mathrm{W\,m^{-2}\ {}^oC^{-1}}$.

The total outside surface area of the pipes is $A_o = \pi n d_o L = 7.36\mathrm{m}^2$ and the inside area is $A_i = \pi n d_i L = 5.38\,\mathrm{m}^2$. Note that these areas are calculated by setting L = 1 m, which will be corrected to the actual one in order to obtain the required heat load.

The fouling coefficients for both water and oil are of about 5000 W m^{-2} °C^{-1} [14]. Substitution of the above values into Eq. (5) gives an overall heat-transfer coefficient of about $U_o = 98.53\,\mathrm{W\,m^{-2}\ {}^oC^{-1}}$ and the corresponding heat rate is $\dot{Q} = U_o A_o \Delta T_m = 97.90\,\mathrm{kW}$. In order to achieve the required heat load, the one-meter length of the pipes must adjusted to be $L = 289/97.90 = 2.95\mathrm{m} \cong 3\mathrm{m}$. Therefore, the actual total outside surface area of the pipes is $A_o = 3\times 7.36 = 22.1$ m^2 and the inside area is $A_i = 5.38\times 3 = 16.1$ m^2. Table 2 summarizes the specifications of the exchanger and its corresponding operating conditions.

Pressure-drop and energy losses due to friction in the heat exchanger

The energy loss due to friction, E_f, and pressure drop, $-\Delta p$, across the pipes or the shell of the heat exchanger can be estimated using the Fanning equation [14]:

$$E_f = \frac{-\Delta p}{\rho} = 2f\frac{L}{D}u^2 \tag{15}$$

where f is the Fanning friction factor. For flow with Reynolds number Re > 4000, it can be estimated using the Colebrook equation [14]:

$$\frac{1}{\sqrt{f}} = -\log\left[\frac{\varepsilon}{3.7D} + \frac{1.256}{\mathrm{Re}\sqrt{f}}\right] \tag{16}$$

where ε is the roughness of the pipe, for carbon steel it is around $\varepsilon = 0.046\,\mathrm{mm}$. The corresponding head loss is $h_L = E_f/g$, where g is the gravitational acceleration. The power done by pump on the operating fluid is given by $Power = \dot{m}E_f$. Under the operating conditions and the geometrical specifications of the heat exchanger given in the previous section, the numerical solution of Eq. (16) and the above definitions will give the results presented in Table 3. It must be taken into consider-

Table 2. Summary of the operating conditions and the configurations of the designed heat exchanger.

Tube side	
Operating fluid	Thermal oil
Inlet temperature	230 °C
Outlet Temperature	225 °C
Volumetric flow rate	97 m^3 hr^{-1}
Tubes layout	Square pitch of 0.0254 m
Number of Tubes; n	137
Outside surface area; A_o	22.1 m^2
Inside surface area; A_i	16.1 m^2
Length of tubes, L	3 m
Shell side	
Operating fluid	Water
Inlet temperature	90 °C
Outlet temperature	95 °C
Volumetric flow rate	45 m^3 hr^{-1}
Shell inside diameter	0.381 m
Material of construction	
Carbon steel	

Table 3. Energy losses due to friction in the pipe–side and shell-side of the heat exchanger.

Parameter	Pipe-side	Shell-side
Fanning friction factor, f	0.008	0.007
Specific energy loss, E_f in J kg^{-1}	1360.77	0.0325
Pressure drop, $-\Delta P$ in atm	13	0.0003
Head loss, h_L, in m	138.71	0.0033
Power done by pump on fluid in hp	37.9	0.001

ation that the diameter used in the shell-side calculations is the equivalent one and the heat exchanger contains 137 pipes. It is found that pumps 4 and 5 of the non-integrated aqueous-emulsion process shown in Fig. 2 are able to overcome these additional energy losses.

CONCLUSIONS

To achieve a more efficient process with low energy consumption and operating cost, a systematic design tool is required in the process industries. The current work studied the synthesis strategy for minimizing energy consumption through the integration of alkyd-resin and aqueous-emulsion processes. Although the same heat load on the water boiler for aqueous-emulsion process is applied on the oil boiler in the integration process, the total amount of diesel is reduced by about 12 %; without reducing the production levels. From process design point of view, the use of one boiler instead of two boilers, to perform the same heat duty, reduces both the operating and maintenance costs.

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