

Improving Cooling System for Energy Saving in Composite Plant

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Abstract: This paper describes an improving cooling system in the composite plant. In order to save the energy, a plate-and-frame heat exchanger has been used to replace the traditional worm cooler. The composite plant in Thailand was studied as an illustrative case study. The experimental results demonstrating the saving in energy costs and some economic benefits of the proposed technique are obtained.

Keywords: energy saving, heat exchangers, cooling system, composite plant

1. INTRODUCTION

There are several characterizations for industrial consolidation and upgrading such as process technology upgrading, and automation of production processes for improved throughput and reduced contamination. The energy cost minimization is one of the most important problems to be solved [1].

The cooling system is one of the most energy consuming parts to be minimized. Accordingly, there has been much effort to reduce the dissipated energy of the cooling process [2-4]. This paper aims to present the similar energy saving to minimize the energy costs for industrial cooling process.

Heat exchangers are commonly used in a wide variety of industrial, chemicals, and electronics processes to transfer energy and provide required heating or cooling. When designing a heat exchanger, one must first determine the type of exchanger to use. The selection process must take account of a number of factors; all of which are related to the specific application. These factors include the following [5]:

- Economic factors
- Thermal and hydraulic requirements
- Compatibility with fluids and operating conditions
- Maintenance
- Availability

When two or more aspects are applicable, one usually goes with the economics first.

Traditionally, in many off-design cooling systems of Thailand's industries, the automatic choice has been the worm cooler. It consists of the pipe coils submerged into a box filled with water. Although worm cooler can be simply constructed, it is extremely expensive on a square meter basis and its rate of heat transfer is varied seasonally. Moreover, the operation time of cooling process is time-consuming. In order to save the energy costs, the new type of heat exchangers will be used to replace the off-design worm cooler.

Although heat exchangers come in every shape and size that you can imagine, the construction of most heat exchangers falls into one of two categories: shell-and-tube, or plate-and-frame. There are many advantages of plate-and-frame heat exchangers over shell-and-tube heat exchangers [6], such as higher overall heat transfer coefficient, lower space requirement, lower holdup volume and residence time, and easier maintenance.

This paper presents the improving cooling system to minimize the energy costs based on the energy saving. The proposed technique uses a plate-and-frame heat exchanger to replace the traditional worm cooler. The performances of the proposed technique were observed using a cooling system of

the composite plant at Shell Company of Thailand as an illustrative case study. The experimental results demonstrating the energy saving and some economic benefits of the proposed improving are included.

2. COOLING SYSTEM IN COMPOSITE PLANT

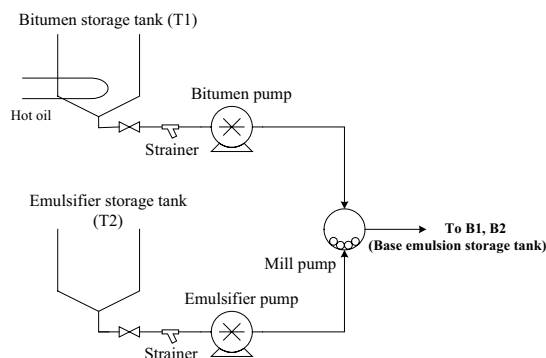


Fig. 1 Bitumen Blending Unit.

The bitumen blending unit is shown in Fig. 1. The bitumen is stored in feed tank T1 usually at temperature of 135-150°C maintained by thermostatically controlled heating coils. The bitumen feed pump draws from tank T1 through a 0.5mm mesh strainer. The dilute emulsifier (concentrated soap) stored in feed tank T2 generally at temperature of about 35°C. The bitumen pump and the dilute emulsifier pump are set to deliver the components in the accurate proportions for the required grade to the mill pump (homogenizes). The bitumen is dispersed as small particles in the soap solution to form the base emulsion. With the foregoing component temperatures, the base emulsion exit temperature should be within the range of 75-78°C. The centrifugal action of the mill pump creates sufficient pressure to deliver the base emulsion to the nearby storage tanks B1 and B2 (see Fig. 2) for cooling system.

Fig. 2 shows the control loop diagram for the cooling system in the composite plant. The base emulsion at temperature 75-78°C from blending unit is first stored in tanks B1 and B2 for 3-5hrs to cool down the temperature to about 65-70°C before cooling with the heat exchanger. The heat of the base emulsion is transferred to the colder water. The heat taken water is then cooled down using the cooling tower. The tanks B3 and B4 are used in cooling sequences to achieve the target temperature. The cooled base emulsion at desired

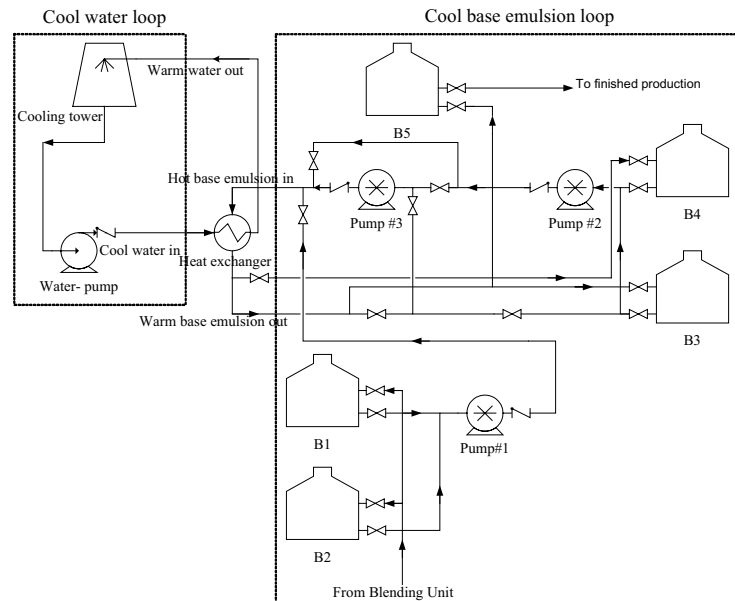


Fig. 2 Cooling system control loop.

temperature of 40-42°C is stored in tank B5 before continuing successive processes to become finished product. To reserve the right final emulsion viscosity, the maximum cooling rate should be 20°C/kg/hr and the maximum operating pressure should be 75psi. The important properties of cooling base-emulsion used for the design conditions of heat exchanger are summarized in Table 1.

Table 1 Important properties of cooling base-emulsion.

Density (kg/m ³)	992.4
Specific heat capacity(kcal/kg°C)	0.5
Thermal conductivity (kcal/m,hr,°C)	0.103
Viscosity (cP) at 68.0°C	2000
Viscosity (cP) at 42.0°C	9000
Inlet temperature of heat exchanger (°C)	68.0
Outlet temperature of heat exchanger (°C)	42.0

3. COOLING SYSTEM WITH WORM COOLER

The worm cooler (see Fig. 5a) is the one of various types of heat exchangers. At Shell Company of Thailand's composite plant, the worm cooler is used to transfer heat energy from the base emulsion to the colder water for over the past 20 years. The worm cooler consists of the stainless pipeline with 3in in dimension and 50m in length submerged into a box filled with water. The rate of heat transfer is varied seasonally. The experimental operation time of cool base emulsion loop using worm cooler is about 13hrs based on the amount of cooled base emulsion for 20,000kgs. The mass flow rate \dot{M} can be estimated by

$$\dot{M} = \frac{m}{t} = \frac{20000}{13} = 1,538.46 \text{ kg/hr} \quad (1)$$

where m and t are the mass of cooled base emulsion and cooling operation time, respectively.

The square meter of heat transfer area A can be calculated based on the dimension of the used steel pipe [6], where pipe dimension, pipe length, and schedule number are equal to 3 inches, 50 meters, and 40, respectively. Thus, the surface area A is approximately 13.97m². The total heat transfer rate \dot{Q} can be stated as

$$\begin{aligned} \dot{Q} &= \dot{M}c_p(T_h - T_c) \\ &= 1,538.46 \times 0.5 \times (68 - 42) \approx 20,000 \text{ kcal/hr} \end{aligned} \quad (2)$$

where:

c_p is the specific heat capacity of base emulsion (0.5kcal/kg,°C).

T_h is the temperature of the hot base emulsion (68°C).

T_c is the temperature of the cold base emulsion (42°C).

4. IMPROVING COOLING SYSTEM

When improving cooling system, one must first determine the type of exchanger to replace the off-design worm cooler. This paper focuses on the effective plate-and-frame and shell-and-tube heat exchangers. Each exchanger has advantages and disadvantages [5-6]. The comparison results are as follow.

4.1 The plate-and-frame heat exchanger

The plate heat exchanger as shown in Fig. 3 consists of a stack of corrugated metal plates pressed together in a frame and sealed at their edges by a compressible gasket to form a series of interconnected narrow passages through which fluids are pumped.

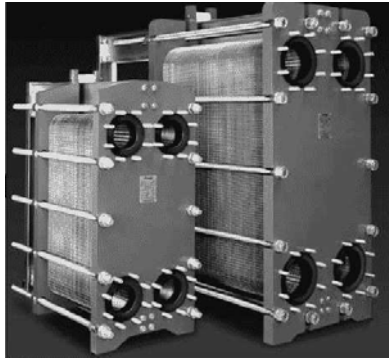


Fig. 3 Plate-and-frame heat exchangers.

Since the heat losses from the heat exchanger are negligible, the total rate of heat transfer \dot{Q} between the hot and cold fluids passing through the plate heat exchanger can be expressed as

$$\dot{Q} = \dot{M}_h c_{ph} (T_{h,in} - T_{h,out}) = \dot{M}_c c_{pc} (T_{c,out} - T_{c,in}) \quad (3)$$

$$\text{and } \dot{Q} = UA\Delta T_M \quad (4)$$

where:

- \dot{M}_h, \dot{M}_c are the mass flow rates of the hot and cold fluids, respectively (kg/hr).
- c_{ph}, c_{pc} are the specific heat capacities of the hot and cold fluids, respectively (kcal/kg, °C).
- $T_{h,in}, T_{c,in}$ are the inlet temperatures of the hot and cold fluids, respectively (°C).
- $T_{h,out}, T_{c,out}$ are the outlet temperatures of the hot and cold fluids, respectively (°C).
- U is the overall heat transfer coefficient (kcal/m²,hr,°C).
- A is the total projected plate area (m²).
- ΔT_M is the effective temperature difference (°C).

Care must be taken in defining A . It can be based on the true total effective surface area of the plate including corrugations or on the projected area. The plate heat transfer coefficients used must be based on the same definition of area as that used in calculating \dot{Q} . In this paper, A refers to the projected area which heat is being transferred as

$$A = Na = NLW \quad (5)$$

where:

- N is the number of effective plates.
- a is the projected area of a single plate.
- L is the single plate length in the direction of flow (m).
- W is the single plate width in the direction of flow (m).

Note that there are two end plates through which heat is not transferred and are not included in the total N .

The overall heat transfer coefficient (O.H.T.C) for the plate heat exchanger is given by

$$\frac{1}{U} = \frac{1}{\alpha_h} + \frac{\delta}{\lambda_p} + \frac{1}{\alpha_c} + R_f \quad (6)$$

where:

- α_h is the hot fluid heat transfer coefficient (kcal/m²,hr,°C).
- α_c is the cold fluid heat transfer coefficient (kcal/m²,hr,°C).
- δ is the plate thickness (m).
- λ_p is the plate conductivity (kcal/m,hr,°C).
- R_f is the fouling resistance for both surfaces of the plate (m²,hr,°C/kcal).

For the single-pass plate heat exchanger in counter flow, ΔT_M can be given by

$$\Delta T_M = \Delta T_{LM} = \frac{[(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})]}{\ln[(T_{h,in} - T_{c,out}) / (T_{h,out} - T_{c,in})]} \quad (7)$$

where:

ΔT_{LM} is the logarithmic mean temperature difference (LMTD).

The design conditions of plate-and-frame heat exchanger as shown in Table 2 use M15-MFM8 Alfa-Laval model. The estimated dimension of heat transfer process is 0.325m in width, 1.885m in length, and 0.610m in height. The estimated dimension of the installation size is 1.453m in width, 1.885m in length, and 0.610m in height. The net weight is 944kgs.

Table 2 Design conditions of plate heat exchanger.

Condition	Hot side	Cold side
Fluid	Base emulsion	Water
Density (kg/m ³)	922.40	993.60
Specific heat capacity (kcal/kg,°C)	0.50	1.00
Thermal conductivity (kcal/m,hr,°C)	0.103	0.533
Inlet viscosity (cP)	2,000	0.801
Outlet viscosity (cP)	8,910	0.705
Mass flow rate (kg/hr)	6,500	13,800
Inlet temperature (°C)	68.00	30.00
Outlet temperature (°C)	42.00	36.10
Pressure drop (mwg)	51.10	0.126
Heat exchanged (Mcal/h)	84.50	
L.M.T.D (K)	20.30	
O.H.T.C. clean conditions (kcal/m ² ,hr,°C)	112.10	
O.H.T.C. service(kcal/m ² ,hr,°C)	94.56	
Heat transfer area (m ²)	44.00	
Fouling resistance (m ² ,hr,°C/kcal)	0.17 × 10 ⁻⁴	
Duty margin (%)	18.5	
Direction of fluids	Countercurrent	
Number of plates	73	
Effective plates	71	
Number of passes	1	
Plate material/thickness (mm)	AISI 316/0.50	
Sealing material	FPMG	
Connection size (mm)	150.00	150.00
Design/Test pressure (atg)	5.10/6.60	5.10/6.60
Design temperature (°C)	70.00	40.00

The overall heat transfer coefficient is calculated based on the CAS-200 program for plate-and-frame heat exchanger design; 94.57kcal/m²,hr,°C used for sizing the total plate area. Substituting the values from Table 1 into the Eq.(3) and Eq.(7), we have

$$\dot{Q} = 6500 \times 0.5 \times 26 = 84,500 \text{ kcal/hr} \quad (8)$$

$$\text{and } \Delta T_{LM} = \frac{[(68 - 36) - (42 - 30)]}{\ln[(68 - 36)/(42 - 30)]} = 20.4 \text{ } ^\circ\text{C} \quad (9)$$

Substituting Eq.(8) and Eq.(9) into Eq.(4), the total plate area of the heat exchanger can be rewritten as

$$A = \frac{\dot{Q}}{U \Delta T_{LM}} = \frac{84500}{94.57 \times 20.4} \approx 44 \text{ m}^2 \quad (10)$$

The experimental operation time of cool base emulsion loop using plate heat exchanger is approximately 3hrs based on the amount of cooled base emulsion of 20,000kgs. The mass flow rate \dot{M} can be estimated by

$$\dot{M} = \frac{m}{t} = \frac{20000}{3} \approx 6,667 \text{ kg/hr} \quad (11)$$

The maintenance for clean and recondition the plates is required when an overall heat transfer coefficient is decreased and the pressure drop is increased. The cleaning method is very simple and easy by simply using the detergent.

4.2 Shell-and-tube heat exchanger

A host of units known as shell-and-tube heat exchangers (see Fig. 4) are built using round tubes mounted on cylindrical shells with their axes parallel to that of the shell.



Fig. 4 Shell-and-tube heat exchangers

The TEMA (Tubular Exchanger Manufacturer's Association) standards size-numbering system is straightforward and simple. It is, in general, used for process and commercial heat exchanger and also suitable when designating the size of shell-and-tube heat exchanger for our improving cooling system with the same design conditions as plate-and-frame heat exchanger's.

Given that

C^1 is the gap between tubes (m).

S_s is the cross-flow area (m²).

D_i is the inner tube diameter (m).

D_o is the outer tube diameter (m).

D_{si} is the inner shell diameter (m).

D_{so} is the outer shell diameter (m).

D_e is the equivalent diameter (m).

P_T is the pitch size (m).

L_b is the baffle spacing (m).

m_s is the mass flow rate (kg/hr).

M_T is the total mass flow rate (kg/hr).

R_e is the Reynold number.

λ is the thermal conductivity (kcal/m,hr,°C).

η is the viscosity (cP).

P_r is Prantl

α_0 is the heat coefficient (kcal/m²,hr,°C).

t_p is the plate thickness (m).

N_T is the number of tubes.

The design process of shell-and-tube [6] can be explained as follows.

1. Determine the inner and outer tube diameters
2. Determine the number of tubes, N_T , and tube arrangement. In this paper, the square-pitch type has been considered because of a rough sketch.
3. Calculate the pitch size, P_T , and approximate the shell diameter from a sketch as

$$P_T = 1.3D_o \quad (12)$$

4. Determine the length of tube, L_T , which is available for 44 m² in the heat transfer area A as

$$A = 3.1416D_oL_TN_T \quad (13)$$

5. Check whether the ratio between tube length and shell diameter is within 5 to 10.
6. Determine the baffle spacing, L_b , and the number of spacing as

$$L_b = 0.2D_s \quad (14)$$

7. Calculate heat transfer coefficients of both shell and tube sides using the following equations.

$$C^1 = P_T - D_o \quad (15)$$

$$S_s = D_s C^1 L_b / P_T \quad (16)$$

$$D_e = 50.1275(P_T)^2 - 3.1416(D_o^2)/4 \quad (17)$$

$$m_s = M_T S_s \quad (18)$$

$$R_e = m_s D_e / \eta \quad (19)$$

$$P_r = c_p \eta / \lambda \quad (20)$$

$$\alpha_0 = 0.36 \lambda R_e^{0.55} P_r^{0.333} / D_e \quad (21)$$

8. Estimate an overall heat transfer coefficient, U , of the system as

$$U = (\alpha_0 \text{ tube side}) + (\alpha_0 \text{ shell side}) \quad (22)$$

9. Repeat all calculation with other suitable parameters.

10. Select the best design with respect to the recommendation and the overall heat transfer coefficient, which is similar to the plate-and-flame heat exchanger.

11. Determine materials for each type and estimate an overall weight, that are,

$$\text{Tube volume} = 0.784D_o^2 - D_i^2 L_T N_T \quad (23)$$

$$\text{Shell volume} = 0.784D_{so}^2 - D_{si}^2 L_s + 1.568D_{so}^2 t_p \quad (24)$$

Table 3. Design conditions of shell-and-tube heat exchanger

Shell	Type	TEMA E (one pass shell)
	Inner diameter (m)	0.478
	Outer diameter (m)	0.508
	Length (m)	3.13
Tube	Number of tube	224
	Arrangement type	Square pitch
	Inner diameter (mm)	17.00
	Outer diameter (mm)	20.00
	Pitch (mm)	26.00
	Baffle spacing (mm)	92.00
	Bundle	Fixed tube sheet
Overall	Ratio (L_T/D_s)	6.54
	O.H.T.C (kcal/m ² hr°C)	95.62
	Estimated weight (kg)	1,063.96
	Material	Stainless steel AISI 150

The design conditions of shell-and-tube heat exchanger are given in Table 3. The cleaning process is very important for shell-and-tube heat exchanger. If cleaning process is not performed, the efficiency will decrease and the tube corrosion will occur. The cleaning process is required when an overall heat transfer coefficient is decreased and the pressure drop is increased. In general, the cleaning methods are high-pressure water, hydrosonic, hydroblast, burn, and sponge balls.

4.3 Comparison results

The cleaning process is the one of most important maintenance for the heat exchangers. Although the stainless steel is used, the substances and contaminants in fluids can cause the corrosion. The cleaning process of plate-and-frame heat exchanger is cheaper and easier than the shell-and-tube type's. From the design conditions shown in Table 2 and Table 3, Table 4 gives the comparison results.

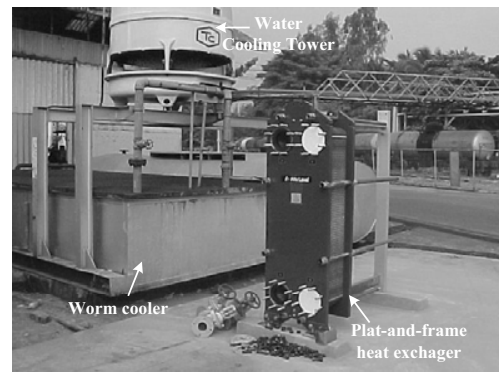
Table 4 Comparison results.

Condition	plate-and-frame heat exchanger	shell-and-tube heat exchanger
Heat transfer area (m ²)	44.00	44.00
O.H.T.C (kcal/m ² .h.C)	112.1	95.62
Dimension of heat transfer process (width × length × height)	0.325m × 1.885m × 0.610m	0.508m × 3.13m × 0.508m
Estimated weight (kg)	944	1064

The comparison results show that the plate-and-frame heat exchanger is more suitable to replace the off-design worm cooler for improving cooling system.

5. THE SAVING RESULTS

At the composite plant of Shell Company of Thailand, the plate-and-frame heat exchanger has been used to replace the worm cooler for improving cooling system as shown in Fig. 5. The costs of cooling system improving are given in Table 5.



(a) Before plate heat exchanger installation



(b) The demolished worm cooler



(c) After plate heat exchanger installation
Fig. 5 Cooling system improving.

The true production costs saving based on the amount of cooled base emulsion for one batch (20,000kgs) occurred in 2003, can be calculated as

Table 5 The costs of cooling system improving.

Plate heat exchanger	317,503Bt
Materials (piping, valve, etc)	42,300Bt
Mechanical work	29,593Bt
Civil work	12,000Bt
Total	401,396Bt

Before improving (the operation time = 13hrs):

Maintenance costs	≈ 36,000Bt/yr	
Electrical consumption	≈ 13.801kW/hr	
	≈ 13.801 × 13	≈ 179.413kW
Electrical costs	≈ 3.7Bt/kW	
	≈ 3.7 × 179.4413	≈ 663.828Bt
Manhour: 13hrs (8hrs + 5hrs Overtime)		
Labor costs	≈ (8 × 50) + (5 × 75)	≈ 775 Bt
Total costs	≈ 663.828 + 775	≈ 1438.82 Bt/batch
The amount of maximum production		≈ 120 batches/yr

After improving (the operation time = 3hrs):

Maintenance costs	≈ 10,000Bt/yr	
Electrical consumption	≈ 13.801kW/hr	
	≈ 13.801 × 3	≈ 41.403kW
Electrical costs	≈ 3.7Bt/kW	
	≈ 3.7 × 41.403	≈ 153.19Bt
Manhour: 3hrs		
Labor costs	≈ (3 × 50)	≈ 150Bt
Total costs	≈ 153.19 + 150	≈ 303.19Bt/batch
The amount of maximum production		≈ 250 batches/yr

The results from the proposed improving

Saving in energy costs	≈ 663.828 - 153.19	≈ 510.638Bt/batch
Saving in labor costs	≈ 775 - 150	≈ 625Bt/batch
Total saving in production costs	≈ 510.638 + 625	≈ 1,135.638Bt/batch
Saving in maintenance costs	≈ 36,000 - 10,000	≈ 26,000Bt/yr
Increasing in the amount of maximum production	≈ 250 - 120	≈ 130batches/yr

The improving cooling system results show that the production costs and the maintenance costs drop to 78.93% and 72.2%, respectively. In addition, the amount of production expands to 108.33%.

6. CONCLUSION

The improving of cooling system in composite plant has been presented in this paper. The proposed technique uses the plate-and-flame heat exchanger to replace the off-design worm cooler. The experimental results show that the proposed improving can reduce the energy costs and the maintenance costs. Moreover, the amount of production expanding can be achieved.

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