Increasing Plate Heat Exchanger Thermal Duty: An Industrial Case Study

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INTRODUCTION

The design and rating of Plate Heat Exchangers (PHE) entails the classic engineering trade-off between heat transfer and pressure drop. Countless studies have characterised the thermal and hydraulic performance of a wide variety of PHE and formulated fundamental correlations for fluid film coefficients, *h*, and friction factor, *f* [1]. However such academic results appear seldom to inform the site engineer regarding the best design and operating practices of heat exchangers in the New Zealand dairy industry. In this paper we present the thermal analysis of a site hot water preheater located at one of New Zealand's largest dairy factories. Using fundamental heat exchanger theory, the analysis also looks at how to increase the heat exchanger's duty and the coupled effect on pressure drop and pumping cost.

HEAT TRANSFER THEORY

The thermal duty (Q) of counterflow PHEs is often calculated using

$$Q = UA\Delta T_{LM} \tag{1}$$

Where U is the overall heat transfer coefficient, A is the heat transfer area and ΔT_{LM} is the log-mean temperature difference. U may also be further described by

$$\frac{1}{U} = \frac{1}{h_h} + R_{f,h} + R_w + R_{f,c} + \frac{1}{h_c} \tag{2}$$

Where h is the heat transfer film coefficient, R_w is the resistance due to the wall and R_f is the resistance due to fouling; subscripts h and c refers to the hot and cold fluid respectively. The film coefficient in PHE's is presented in literature correlations [1] as a function of Reynolds number (Re) and Prandtl number (Pr) in this form

$$h = a\,\mathrm{Re}^b\,\mathrm{Pr}^c \tag{3}$$

Where a, b and c are constants. For the purposes of this study it is useful to define a film coefficient ratio relative to Re at some reference (*ref*) condition while assuming Pr is constant due to no significant temperature change

$$\frac{h}{h_{ref}} = \left(\frac{\mathrm{Re}}{\mathrm{Re}_{ref}}\right)^b \tag{4}$$

Exponent b in the above equations typically varies between 0.5 and 0.7 depending on plate geometry [1]. In this work the median value of 0.6, as shown in Eq. 5, is applied to calculate film coefficients relative to a reference condition, which in this study is the current PHE operating condition. Since Re is proportional to velocity, the Re ratio in Eq. 4 is directly proportional to the ratio of the volumetric flow rates ($\dot{V}$). The Re ratio may be also affected by possible design changes to A and the number of fluid passes (n). As a result, Eq. 4 is rewritten with h as the subject

$$h = h_{ref}\left(\frac{A_{ref}}{A}\frac{n}{n_{ref}}\frac{\dot{V}}{\dot{V}_{ref}}\right)^{0.6} \tag{5}$$

A similar technique was successfully applied by Walmsley et al [2] to predict the performance of PHE's facing variable fluid flow rates.

DATA ACQUISITION

Temperature and flow data in and out of an industrial Site Hot Water (SHW) preheater PHE have been recorded at 10 min intervals for one and half months during peak milk production. Using the temperature and flow data, the heat exchanger duty and overall heat transfer coefficient are calculated and individual film coefficients are estimated. Hydraulic pressure drop is estimated using the *f* correlation for 45° chevron plates in Wang et al [1].

The cold fluid in the PHE is partially treated bore water that is known to foul exchanger surfaces, enters cold between 15 – 20 °C and needs to be raised to 65 °C for SHW use. The flow rate of the SHW is dependent on a variable hot water demand. The hot fluid in the PHE is warm process water pumped from a storage tank (30 – 35 °C). The warm process water has no outlet temperature constraint; although a PID feedback control loop is applied to control the flow rate of the process water for an outlet temperature set-point of 25 °C. Shortly before the monitoring period of the PHE, the plates were mechanically cleaned. Duty not supplied by the pre-heater through heat recovery is supplied using low pressure steam priced at $30/t.

The single-pass PHE has an area of 500 m^2 and was designed to achieve a duty of 3.8 MW. The PHE design was based on ΔT_{LM} of 7 °C and U of 1.09 kW/°C/m^2. The plant runs for 5000 h/y and site's electricity price is $150 /MWh.

INDUSTRIAL HEAT EXCHANGER ANALYSIS

The heat exchanger duty, SHW flow rate, maximum exchanger temperature difference (ΔT_{max}) and U are compared in Figure 1. At the start of the monitoring period, the PHE duty initially increases with increasing SHW flow rate. After a couple of weeks, the duty starts to fall from 3.0 MW (on average) and levels out at below 2.0 MW, while the SHW, in the second half of the analysis period, gradually rises. For the entire period the average duty was 2.25 MW.

The decreasing duty in Figure 1 is due to decreasing U and ΔT_{max} values. Initially U is about 0.82 kW/°C/m^2, which is lower than the design U, and by the end of the monitoring, U levels off at 0.53 kW/°C/m^2. ΔT_{max} follows a similar trend to the duty data with a gradual reduction over most of the analysis period.

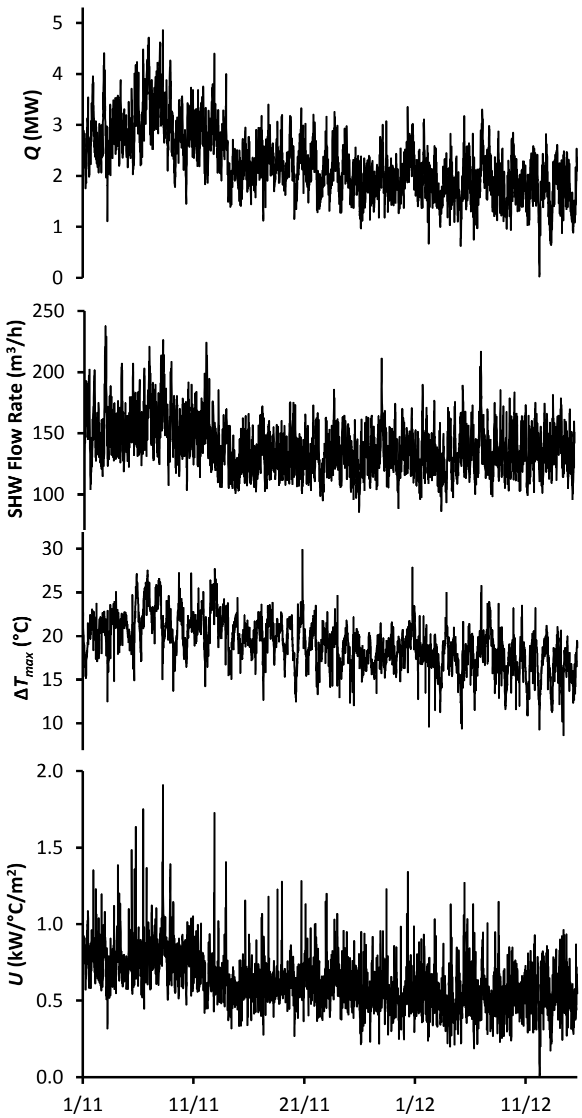

Figure 1 – Pre-heater duty, SHW flow rate, maximum exchanger temperature difference (ΔT_{max}) and U for the period analysed.

IMPROVING PHE PERFORMANCE

Three methods for increasing PHE duty (variable increased, in brackets) are:

1. Add plates (A)
2. Reduce fouling (U)
3. Increase passes (U)

Using Eq. 2 and 5, the calculated U values through h may be adjusted to estimate the outcome of the three identified methods for increasing duty for the PHE (Table 1). After adjusting the U value, the heat exchanger duty is solved for each 10 min time interval using the method of Walmsley et al [2] and the measured flow rates and inlet temperatures of the two fluids. The change in pumping cost results from a change in channel velocities and pressure drop for the two fluids in the heat exchanger compared to the status quo.

Table 1 – Methods for increasing PHE duty and effects on thermal savings and pumping costs.

Option	Duty Increase	Thermal Savings ($/y)	Pumping Cost ($/y)
Increase area by 50%	6.9%	35,400	-300
Minimise fouling*	11.7%	59,500	N/C
Increase from 1 to 2-pass	16.7%	85,400	1,400
Increase area by 50% and from 1 to 2-pass	22.0%	112,300	400
Minimise fouling* & increase from 1 to 2-pass	25.5%	130,200	1,400

* Assume constant U of 0.82 kW/°C/m^2

Often in the dairy industry adding plates (area) is looked to as the first option for increasing heat exchanger duty. In the case of increasing area by 50% on a PHE, the predicted increase in UA is less than 18% due to decreased channel velocity and U counter-acting the increase in A resulting in only 6.9% extra heat recovery. The lower velocities are estimated to reduce pumping costs by $300 per annum.

Reducing fouling can be achieved by regular cleaning, improved water treatment and/or increased fluid channel velocity. Cleaning on the product-sides of PHE's to eliminate product degradation caused by microbial growth at exchanger surfaces is essential practice in the dairy industry; however, much less action is taken to minimise fouling on the non-product (e.g. utility or waste streams) sides of heat exchangers and the associated reduction in heat exchanger duty. Maintaining U at its initial average of 0.82 kW/°C/m^2 returns an increase in heat recovery of 11.7% and an annual thermal savings of $59,500, which must be used to pay for the cleaning method.

Increasing from one to two fluid passes gives a proportionate increase in channel velocity and Re number. As a result doubling the fluid passes is expected to return the greatest increase in duty and thermal savings, which is two orders of magnitude larger than the expected increase in pumping costs. This method is also relatively inexpensive one-off capital cost. The final two options in Table 1 are combinations of the first three options.

In conclusion, it is important for engineers in industry to apply basic heat exchanger theory to analyse heat exchanger performance. In many cases, an inexpensive solution such as increasing the number of passes in a PHE is a key ingredient to yielding the greatest profit.

REFERENCES

[1] Wang et al, *Plate Heat Exchnagers: Design, Applications and Performance*, CRC Publishing, [2] Walmsley MRW et al *Energy* 55, 15-22 (2013).