4. Temperature control of plate heat exchangers

Heat exchanger design

Compared to other types of heat exchanger, plate heat exchangers have a very high heat transfer to size ratio. The heat transfer coefficient attained by a plate heat exchanger is considerably greater than that attained by a traditional shell and tube heat exchanger.

As a result of this high performance characteristic, the physical heat transfer area required for a given duty is relatively small.

The actual heating area necessary to provide a given duty can be calculated from Formula 9.

Formula 9 Calculate heat exchange area

$$A = \frac{Q \times 1{,}000}{k \times T_{lmtd}}$$

where:

A = heat transfer area, m²

Q = heat requirement, kW

k = heat transfer coefficient, W/m² °C

T_{lmtd} = log mean temperature difference, °C

NB: With plate type heat exchangers the 'k' value will be in the range of 5,000 - 10,000 W/m² °C.

The logarithmic mean temperature difference (T_{lmtd}), is the logarithmic mean, of the temperature differences between heat exchanger primary, and secondary, inlet and outlet temperatures. For most practical purposes, the arithmetic mean temperature difference (T_{amtd}), can be used in its place. This calculation is shown in Formula 10.

Formula 10
Calculate arithmetic mean temperature difference

$$T_{amtd} = \frac{\left(TH_{(out)} + TH_{(in)}\right)}{2} - \frac{\left(TC_{(out)} + TC_{(in)}\right)}{2}$$

where:

T_{amtd} = Arithmetic mean temperature difference, °C

$TH_{(in)}$ = Primary side inlet temperature, °C

$TH_{(out)}$ = Primary side exit temperature, °C

$TC_{(in)}$ = Secondary side inlet temperature, °C

$TC_{(out)}$ = Secondary side exit temperature, °C

When steam is used as the primary heating medium, it gives up its specific enthalpy of evaporation at constant temperature, therefore, $TH_{(in)}$ and $TH_{(out)}$ will have the same value.
A simplified calculation can be used to arrive at the arithmetic mean temperature as shown in Formula 11.

When a plate heat exchanger is selected and the most common form of temperature

Formula 11
Simplified calculation for arithmetic mean temperature difference

$$T_{amtd} = TH_{(in)} - \frac{TC_{(out)} + TC_{(in)}}{2}$$

control, i.e. a temperature control valve fitted in the primary steam inlet to the exchanger, the pressure drop on the secondary side is a major consideration. Often, when steam is the primary heating medium, a greater heat transfer area than necessary is selected in order to keep this secondary side pressure drop within reasonable limits.

It can be seen from Formula 9., that if the heat transfer area (A) is increased from that needed to give output (Q), the only variable that can change, to give the required output with this increased heat transfer area, is the mean temperature difference. This must decrease and the only way this can decrease, without the secondary conditions changing, is if the primary side mean temperature decreases.

Put another way, the steam temperature, therefore its pressure, must decrease if the desired output is to be achieved with an increased heat transfer area.

This over sizing of the heat transfer area in plate heat exchangers that are not specifically designed to operate with a steam primary, to facilitate reasonable pressure drops on the secondary side is quite common in steam applications. This over sizing is often in the region of 100% to 200% and can have a considerable effect on the actual steam pressure within the exchanger, even under full load conditions. When the design of the plate heat exchanger is specifically optimised for use with steam, over sizing is not a problem and rarely exceeds 15%.

The pressure at the inlet of the steam trap may be less than the back pressure imposed on the outlet of the steam trap. Where this is the case, condensate will not be discharged and will back up into the heat exchanger. This will certainly cause temperature control problems and may also cause structural damage to the exchanger through waterhammer and thermal stress.

Stall condition

If the back pressure imposed upon a steam trap is greater than the pressure available at its inlet, the heat exchanger is described as being in a 'stall' condition. Condensate cannot pass through the trap and will back up, possibly resulting in the loss of effective control over the heat exchanger.

The point at which this condition will occur in a particular application can be predicted, either by calculation, or by plotting the known operating conditions onto a chart and reading off the stall points. Both of these methods can be illustrated by example.

Calculation

A plate heat exchanger is required to heat 4 kg/s of water from 30°C to 90°C. Steam is available at 4 bar g and it is initially assumed there will be a 25% pressure drop through the temperature control valve, giving 3 bar g at the heat exchanger. Condensate discharges to a gravity return system at 0 bar g. To give the required output and provide a reasonable pressure drop of approximately 35 kPa on the secondary side, a heat exchanger has been selected with a heat exchange area of 2.6 m^2 and a heat transfer coefficient (k) of 7,450 $W/m^2\ °C$.

First the full design heat load is calculated, using Formula 12.

Formula 12
Calculate heat load (full load)

$$Q = M \times Cp \times \left(TC_{(out)} - TC_{(in)}\right)$$

$$Q = 4 \times 4.186 \times (90 - 30)$$

$$Q = 1{,}004.6 \text{ kW}$$

where:

Q	**=**	**heat load, kW**
M	**=**	**secondary flow rate, kg/s**
Cp	**=**	**specific heat capacity, kJ/kg °C (for water - 4.186 kJ/kg °C)**
$TC_{(in)}$	**=**	**secondary inlet temperature, °C**
$TC_{(out)}$	**=**	**secondary exit temperature, °C**