ADCAMAT PUMP TRAPS

UNDERSTANDING STALL CONDITION

For convenience purposes the term heat exchanger will be used to describe all types of equipment where heat is transferred from one fluid to another. This includes shell and tube heat exchangers and plate heat exchangers, but also heating coils, jacketed vessels, heating batteries, etc.

WHAT IS STALL AND WHY DOES IT OCCUR?

In a temperature control application such as the one shown in Fig. 1, when a control valve is throttling to meet the requirements of a reduced heat load, the steam pressure P1 inside the heat exchanger falls. This fall is sometimes considerable and can reduce differential pressure across the steam trap to a point where it can no longer discharge (P2 is equal to or greater than P1). Consequently, condensate accumulates inside the heat exchanger resulting in a stall condition which leads to poor heat transfer (temperature fluctuation), corrosion, water hammer, leakages and noise, amongst others.

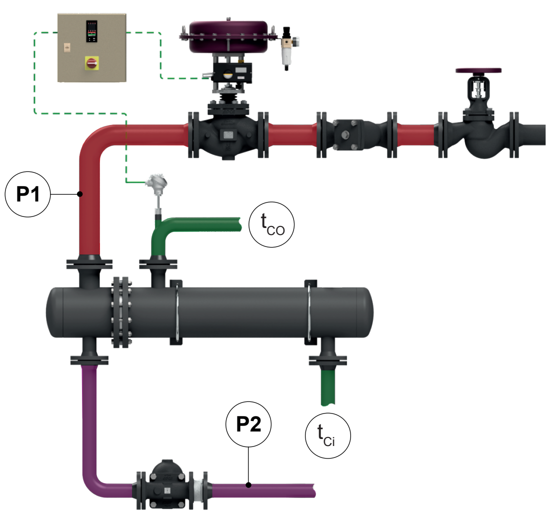

Fig. 1

OVERDESIGNED HEAT EXCHANGERS

Most heat exchangers have more heating area than required. This is because designers typically select a heat exchanger that covers the requirements from a standard range with pre-determined heat transfer areas. In addition to other safety factors normally considered, this often results in over sizing.

Overdesigned heat exchangers with capabilities above the required needs operate with lower steam pressures, and corresponding temperatures, when compared to perfectly sized units, increasing the chance of stall condition occurring.

It is thus critical that an evaluation of the load profile of a heat exchanger is taken place, determining whether or not a stall condition may occur.

HOW TO SOLVE A STALL CONDITION?

In such cases where stall condition can take place, an ADCAMat pressure operated pump and steam trap (see Fig. 2) or an ADCAMat automatic pump trap, installed in a closed loop system, is a solution.

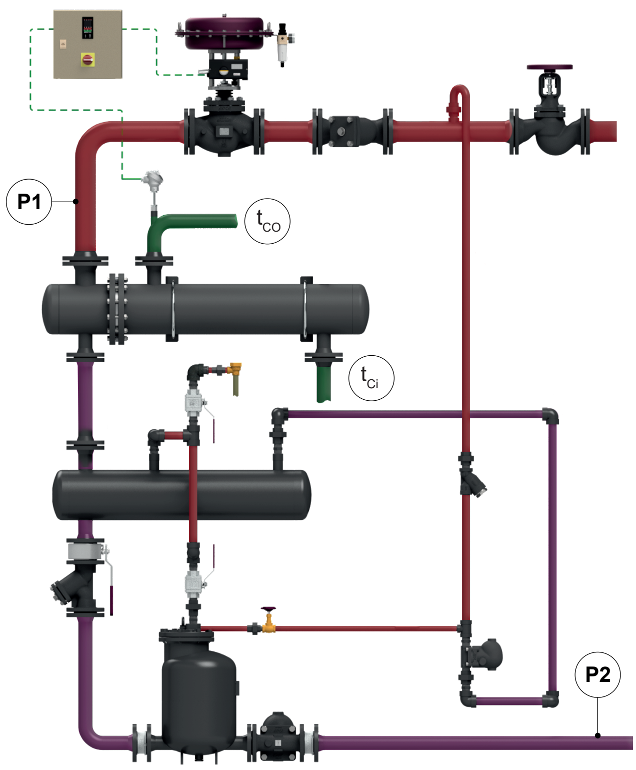

Fig. 2

Whenever the steam trap is incapable of draining condensate, the pump function is activated (using external steam pressure). The pump replaces the necessary positive pressure to lift the condensate to the return system before water logging occurs. The pump is only required during stall loads and therefore a steam trap is still required to prevent steam from discharging into the return condensate line whenever steam pressure P1 exceeds the back pressure P2.

Obviously, if the back pressure always exceeds the steam pressure (full load stall) the steam trap is unnecessary.

Wheatear a pressure operated pump and steam trap set or a “two-in-one” automatic pump trap is involved each part (pump and trap) must be analysed individually.

STALL PREDICTION

Calculation of the stall load can be performed either by mathematical or graphical approach. The first uses standard thermodynamic formulas to calculate the percentage of heat load at stall, which is reached when the steam pressure P1 is equal to the back pressure P2. The second approach involves the use of a “stall chart” which yields sufficient accuracy as long as the operating steam pressure, and corresponding temperature, at full load is considered.

Example

Consider a heat exchanger operating at a nominal 6 bar g, designed to heat a constant water flow of 15 000 kg/h from 20 ºC to 80 ºC. Minimum heat load occurs at 60% of full load. The condensate lifts 10 meters into a return line at 0,5 bar g pressure.

a) Determining the equivalent saturated temperature of the total back pressure

The total back pressure is equal to the lift height equivalent pressure, plus the pressure on the return line.

Pipe friction is neglected considering a short and properly sized downstream pipe work.

10 m × 0,0981 bar + 0,5 bar = 1.481 ≈ 1,5 bar g

Therefore, the total back pressure is 1,5 bar g and, from the steam tables, the corresponding saturated temperature is t_B=127,6 ºC.

b) Calculating the full heat load

$$\dot{Q} = \dot{m} \cdot Cp \cdot \Delta T$$

$\dot{Q}$ = Heat transfer rate [kcal/h]

$\dot{m}$ = Mass flowrate of the secondary fluid [kg/h]

Cp = Specific heat capacity of the secondary fluid [kcal/kg°C]

ΔT = Temperate rise of the secondary fluid [K or °C]

$$\dot{Q} = 15\,000 \cdot 1 \cdot (80 - 20)$$

$$\dot{Q} = 900\,000 \text{ kcal/h}$$

c) Calculating the steam flow rate at full load

At 6 bar g saturated steam has a temperature of approximately 165 ºC and an enthalpy of evaporation h_{fg} = 483,8 kcal/kg.

$$\dot{m}_s = \frac{\dot{Q}}{h_{fg}}$$

$$\dot{m}_s = \frac{900\,000}{183,8} = 1860,27 \text{ kg/h}$$

d) Calculating the required heating area

Using the logarithmic mean temperature difference:

$$\Delta T_{LM} = \frac{t_{CO} - t_{CI}}{\ln\left(\frac{t_S - t_{CI}}{t_S - t_{CO}}\right)}$$

ΔT_{LM} = Logarithmic meam temperature difference [K or °C]

t_{CO}=Secondary fluid outlet temperature [°C]

t_{CI}=Secondary fluid inlet temperature [°C]

t_S=Steam temperature [°C]

$$\Delta T_{LM} = \frac{80 - 20}{\ln\left(\frac{165 - 20}{165 - 80}\right)}$$

$$\Delta T_{LM} = 112,34 \text{ °C}$$

The minimum heating area that fulfills the requirements for the full load is calculated according to the following formula:

$$\dot{Q} = A \cdot k \cdot \Delta T_{LM}$$

$$A = \frac{\dot{Q}}{k \cdot \Delta T_{LM}}$$

$\dot{Q}$ = Heat transfer rate [kcal/h]

A = Heating area [m^2]

k = Heat transfer coefficient [kcal/m^2 h°C]

ΔT_{LM} = Logarithmic meam temperature difference [K or °C]

The manufacturer considers a heat transfer coefficient of 2100 kcal/m^2 h°C for this steam to water heat exchanger which yields a heat transfer area of:

$$A = \frac{900\,000}{2100 \cdot 112{,}34}$$

$$A = 3{,}81 \text{ m}^2$$

Amongst the heat exchanger manufacturer range, a model with a heat transfer area of A = 4,15 m^2 was selected which corresponds to an overdesigning of around 9%.

e) Calculating operating steam pressure and flow rate at full load for the overdesigned heat exchanger

The operating steam temperature t_s for the full load condition must be determined by taking the larger heating area into consideration, however firstly the new ΔT_{LM} must be determined as follows:

$$\Delta T_{LM} = \frac{\dot{Q}}{A \cdot k}$$

$$\Delta T_{LM} = \frac{900\,000}{4{,}15 \cdot 2100}$$

$$\Delta T_{LM} = 103{,}27 \text{ ºC}$$

The steam temperature t_s can be retrieved from the following equation:

$$\Delta T_{LM} = \frac{t_{CO} - t_{CI}}{\ln\left(\frac{t_S - t_{CI}}{t_S - t_{CO}}\right)}$$

$$103{,}27 = \frac{80 - 20}{\ln\left(\frac{t_S - 20}{t_S - 80}\right)}$$

$$\ln\left(\frac{t_S - 20}{t_S - 80}\right) = 0{,}58$$

$$\frac{t_S - 20}{t_S - 80} = e^{0{,}58}$$

$$t_S - 20 = e^{0{,}58}\,(t_S - 80)$$

$$0{,}79t_S = 122{,}88$$

$$t_S = 155{,}54 \text{ ºC}$$

This temperature corresponds to a steam pressure of 4.5 bar g which means that a 9% overdesigning decreased the operating steam pressure by 25 %. The steam flowrate at the full load of 900 000 kcal/h for the heat exchanger with A = 4,15 m^2 can now be calculated. Steam tables state that the enthalpy of evaporation of saturated steam at 4.5 bar is h_{fg} = 500,76 kcal/kg.

$$\dot{m}_s = \frac{\dot{Q}}{h_{fg}}$$

$$\dot{m}_s = \frac{900\,000}{500{,}76} = 1797{,}27 \text{ kg/h}$$

f) Calculating the flow rate at stall load for the overdesigned heat exchanger

The load percentage at which stall condition occurs can be calculated according to the following formula:

$$\%\ \text{Stall load} = \frac{t_B - t_{CO}}{t_S - t_{CO}}$$

t_S = Steam temperature [°C]

t_B = Back pressure equivalent steam temperature [°C]

t_{CO} = Secondary fluid outlet temperature [°C]

$$\%\ \text{Stall load} = \frac{127{,}60 - 80}{155{,}54 - 80} \cdot 100$$

$$\%\ \text{Stall load} = 63\ \%$$

Which means that the flow rate at stall load is:

$$\dot{m}_S = 1797{,}27 \times 0{,}63 = 1132{,}28 \text{ kg/h}$$

g) Sizing the steam trap and pump or pump trap

In this case stall condition occurs above the minimum working load which means that a pump and steam trap, or automatic pump trap, must be installed.

The steam trap must be able to discharge the flow rate at full load (1797,27 kg/h with 4,5 bar g of steam pressure) and the pump must be able to handle the flow rate at stall condition of (1132,28 kg/h) against the back pressure of 1,5 bar g.

FLOW RATE CAPACITY (kg/h) OPERATING IN STEAM TRAP MODE							
MODEL	SIZE	DIFFERENTIAL PRESSURE (bar)					
		0,1	0,3	0,5	0,7	1	1,5
PPT14	11/2" x 1" DN 40 x 25	650	1100	1500	1700	2000	2600
PPT14	2" x 11/2" DN 50 x 40	1050	1750	2400	2700	3400	3900
MODEL	SIZE	DIFFERENTIAL PRESSURE (bar)					
		2	3	4	5	7	10
PPT14	11/2" x 1" DN 40 x 25	3000	3510	3990	4400	5400	6200
PPT14	2" x 11/2" DN 50 x 40	4500	5900	6600	7650	8500	10100

Fig. 3

According to Fig. 3 it can be seen that a PPT14 DN 40 x 25 or DN 50 x 40 will be able to handle the full load of 1797,27 kg/h at a differential pressure of 4,5 – 1,5 = 3 bar g.

FLOW RATE CAPACITY (kg/h) OPERATING IN PUMP MODE W/ 300 mm FILLING HEAD			
MOTIVE PRESSURE (bar)	TOTAL LIFT (bar)	11/2" x 1" DN 40 x 25	2" x 11/2" DN 50 x 40
1	0,35	1050	1220
2		1190	1490
3		1220	1530
4		1280	1600
6		1310	1640
8		1380	1730
10		1460	1830
2	1	940	1180
3		1020	1280
4		1110	1390
6		1200	1510
8		1290	1620
10		1380	1730
3	2	720	900
4		850	1070
5		940	1180
6		1010	1260
8		1130	1410
10		1200	1490

Fig. 4

The pump flow rate capacities shown in Fig. 4 confirms that with an available motive pressure of 6 bar, a PPT14 DN 50 x 40 will handle the 1132,28 kg/h of condensate at stall condition against the 1,5 bar g back pressure.

Thus, a PPT14 DN 40 x 50 is suitable for this application.

If, however, in another application the minimum working load is higher than the stall load (e.g. minimum working load of 70%) then the system will have positive differential pressure at all times and a steam trap is the appropriate solution, as long as it has enough discharge capacity at minimum and maximum system loads.