A Guideline for Design Pressure – Part 2

Saeid Rahimi

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Introduction

All process equipment in general should be designed for the maximum pressure which can be attained in service during normal operating, upset, startup and shutdown conditions. For example, centrifugal pumps are often provided without relief valve as they are designed for highest possible pressures corresponding to blocked outlet condition. However, this pressure in most cases is extremely high, unpredictable or not economically feasible to design for. That is why equipment are usually designed for a pressure which is calculated by adding margin to maximum operating pressure and full protection is achieved by a relief valve set at or below specified design pressure.

As shown in Figure 1, this margin should be sufficient to accommodate high pressure alarm and high pressure trip and to avoid unintentional relief valve opening.

The design pressure and temperature is the basis for mechanical design of equipment and piping. Therefore, it is vital to specify all design pressures in conjunction with the corresponding design temperatures (coincident design conditions) for calculating minimum wall thickness and related aspects of vessel and piping design.

This note presents some guidelines on design pressures of compressors, heat exchangers, storage tanks and piping.

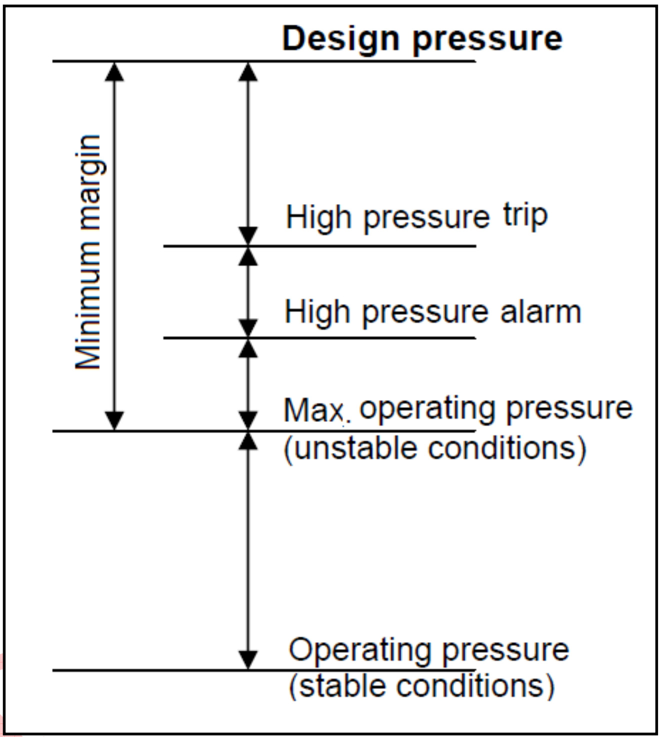

Figure 1 – Margins between operating and design pressure

Compressors

- **Centrifugal Compressor**

The maximum pressure during surge or restricted discharge conditions calculated via below formula can be considered as the design pressure of centrifugal compressor and its discharge system. In this case, no relief valve is required.

- ➢ Maximum Suction Pressure + Differential Pressure at surge point

High-high pressure trip or suction vessel relief valve set point in the most conservative conditions can be used as the maximum suction pressure.

Differential pressure at surge point should be obtained from compressor performance curve considering the highest gas molecular weight and the lowest suction temperature. If such vendor data is not available, the compressor differential pressure at surge point can be estimated:

- ➢ Surge Differential Pressure = 1.25 x Normal Differential Pressure — for constant speed driver
- ➢ Surge Differential Pressure = 1.30 x Normal Differential Pressure — for variable speed driver

But it is usually economical to set the design pressure lower than the maximum possible pressure that the compressor can develop, and to provide appropriate relief valve on the discharge side of compressor (Note 1). In this case, relief valve should be set at the higher of the followings:

- ➢ Discharge pressure + 2 bar — for discharge pressure ≤ 20 barg
- ➢ 1.1 x Discharge pressure (Note 2) — for discharge pressure > 20 barg

- **Reciprocating Compressor**

For reciprocating compressors, discharge relief valves are nearly always required. Reliance on stalling of a reciprocating compressor is generally not economically attractive because driver stalling pressure is usually very high in comparison to operating pressure.

For reciprocating compressors, a greater differential than 10% may be applicable while setting the relief valve set point because of pressure pulsation or fluctuation. Following guideline can be used:

- Discharge pressure + 3 bar — for discharge pressure ≤ 10 barg
- 1.2 x Discharge pressure (Note 2) — for discharge pressure > 10 barg

Note:

1) In particular cases where the flow through the relief valve is the governing load of disposal system (flare), the cost of following options can be analyzed to reach an economic solution:
   - Setting the design pressure of the compressor casing and downstream equipment equal to the maximum pressure that can be generated at the surge point and avoid relief valve.
   - Increasing the design pressure of the compressor and downstream equipment to an economic level (may be the piping rating limit) which can results in smaller relief valve and reduced relieving rate. See note 10 in part 1 of this article.
   - Using modulating pilot operated relief valve which releases the required relieving rate.
2) Especially in high pressure systems, the margin on discharge pressure may be reduced due to economic reasons. Some companies recommend setting relief valve not more than 5.0 bar above the compressor discharge pressure.

Heat Exchangers

1. Heat exchangers located on discharge side of pump should be designed for shut off pressure of the rotating machinery, except where protected by pressure relief valve.
2. For heat exchangers on gas route, the method used for pressure vessel is applied for heat exchangers. However, in order to specify the right design pressure, a detailed hydraulic should be undertaken and operating pressure of all equipment should be specified. Then margin specified in part 1 of this note for pressure vessels can be added to operating pressure. Figure 2 shows a typical example where the design pressures of equipment have been specified according to pressure profile.
   In this example, the pressure drop through system is such that having different design pressure for equipment in series is practically and economically justified. Moreover, there is no valve in gas route so single relief valve sized for full flow should be sufficient to protect entire system against blocked discharge. However, the presence of manual or control valve (or any object which can block the flow) in the same system may result in totally different approach and design conditions.

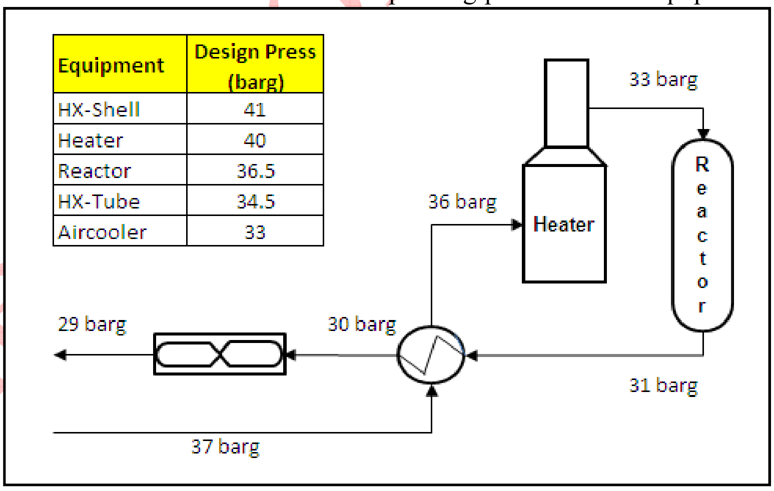

Figure 2 – Design pressure profile for heat exchangers on a gas route

   In such cases and in order to minimize number of full flow relief valves, the following guidelines can be used:
   - If there is a control valve or block valve downstream of heat exchanger, the design pressure of the exchanger should be equal to the upstream equipment design pressure.
   - If there is a control valve or block valve upstream of heat exchanger, the design pressure of the heat exchanger can be equal to the design pressure of the downstream equipment at the entry point + 1.25 times the maximum pressure drop in the system between the inlet of the exchanger and the inlet of the downstream equipment + static head (if any).
3. For design pressure of a column overhead condenser and bottom reboiler, refer to notes 13 and 14 of part 1 of this article.
4. For multi-tube and TEMA heat exchangers, any of the following methods or combination of them can be used to prevent the loss of containment of the low-pressure side to atmosphere due to tube rupture (Note 1):
   - Increasing the design pressure of the low-pressure side including upstream and downstream systems so that its corrected hydrotest pressure[1] (Note2) becomes at least equal to the design pressure (Note 3) of the high-pressure side.
   - Assuring that there is an open flow path that can pass the tube rupture flow without exceeding the stipulated pressure.
   - Providing pressure relief device.

[1] Hydrostatic test pressure multiplied by the ratio of stress value at design temperature to the stress value at test temperature

Figure 3 shows a typical system where a combination of last two methods has been utilized to protect the system against tube rupture. In this example, a low pressure gas is preheated in a shell and tube heat exchanger using 40barg steam (with design pressure of 48barg). It is not feasible to increase the design pressure of entire low pressure system to withstand steam pressure, therefore system hydraulic is studied to make sure that extra steam along with process fluid can reach downstream vessel without pressurizing the heat exchanger beyond reasonable (economic) pressure. Moreover, downstream vessel's relief valve is able to handle tube rupture flow rate. The design pressure of heat exchanger and piping system have been slightly increased (above what is needed for normal operation) and made suitable for tube rupture case.

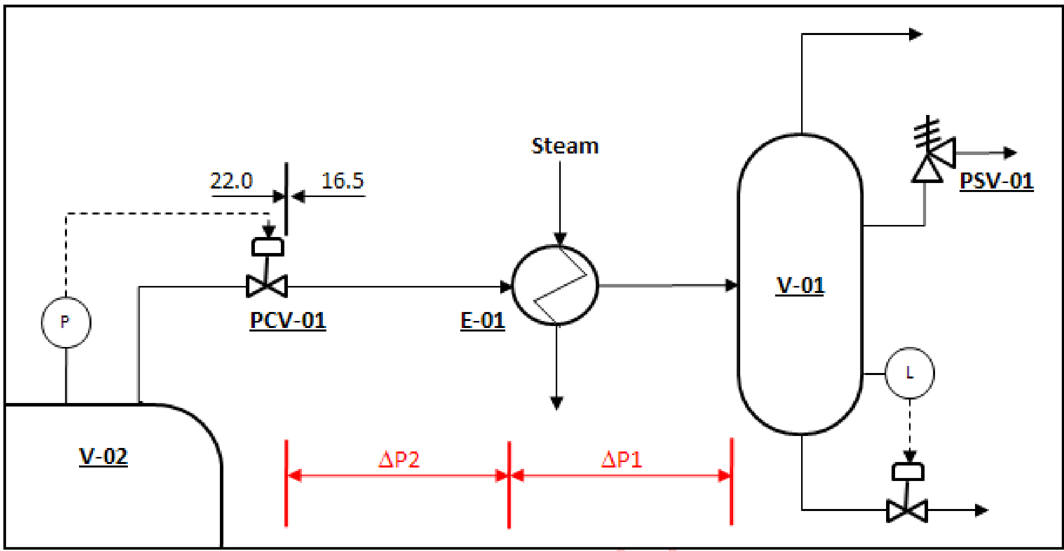

Figure 3 – A typical system where LP side of heat exchanger is designed for tube rupture

Below Table shows the pressure profile of the system before and after tube rupture. It also indicates the required design pressure to take care of tube rupture case (all pressures are in barg). According to this table, design pressure of 16.5barg can be safely used for entire system downstream of PCV-01 up to V-01 inlet nozzle.

Case	**V-02**	**PCV Outlet**	**E-01**	**V-01**	**Remark**
Normal Operation	20.0	12.0	11.0	10.0	ΔP1 =1.0bar, ΔP2 =1.0bar
Tube Rupture Case	20.0	13.5	12.5	10.0	ΔP1 =2.5bar, ΔP2 =1.0bar
Design Pressure	22.0[d]	16.2[c]	14.9[b]	11.8[a]	[a] 10 + 1.8 =11.8 (PSV-01 set point) [b] 11.8 + 1.25 x ΔP1 = 14.9 [c] 14.9 + 1.25 x ΔP2 = 16.2 [d] 20 x 1.1 = 22.0

Designing system for tube rapture can turn into a very complicated problem depending on fluid phase (vapor, liquid, two phase) in both sides of heat exchanger, the effect of LP and HP fluids on each other (vaporization, heating, reaction, surge, contamination), and the temperature and pressure difference between LP and HP sides which is out of interest of this article.

Note:

1) These guidelines were established without considering a chemical reaction in the event that the HP fluid mixes with the LP fluid. If the heat exchanger contains reactive chemicals, then a careful evaluation shall be performed to ensure that the reactive situation does not result in the pressure exceeding the low-pressure side's corrected hydrotest pressure.
2) According to the latest revision of ASME VIII, pressure vessels should be tested at 130% of design pressure. However, where the actual test pressure is different from this (some plants have been already tested at 150% of design pressure), that specific pressure should be used for this study.
3) The use of maximum possible system pressure instead of design pressure may be considered as the pressure of the HP side on a case-by-case basis where there is a substantial difference in the design and operating pressures for the HP side of the exchanger.

Storage Tanks

- **Atmospheric and Low Pressure Tanks**

The design pressure of storage tanks is not normally governed by any specific rules and can be any value below 2.5psig (0.18barg) if the design standard is API-650 and below 15psig if the tank is designed according to API-620. API STD 2000 also describes the pressure and vacuum design requirement of the atmospheric tanks in this category.

Following guidelines are generally followed when atmospheric tanks are designed:

1. For atmospheric storage tanks with open vent or without gas blanketing, if the specific gravity of stored liquid is:
   - Lower than 1.0, the design pressure shall be full of water + 50 mm water column.
   - Higher than 1.0, the design pressure shall be full of product + 50 mm product column.
2. For atmospheric storage tanks with gas blanketing, the design pressure of atmospheric storage tanks shall be designed full of water + 150 mm water column and -50 mm water column for vacuum condition.

- **Pressurized Storage Tanks (Spheroid, Bullet, etc.)**

Depending on the type of tank, higher design pressures can be specified but if storage tank operating pressure is higher than 15psig, the design pressure should be calculated according to the method described for pressure vessels.

Process Piping

The design pressure of the piping system shall be the highest of the following:

1. 110% of maximum continuous operating pressure but not less than 1 barg.
2. The vapor pressure of the fluid in the pipe at design temperature, except where protected by relief valve.
3. The set pressure of relief valve protecting a complete system, such as column overhead.
4. The shut off pressure of the rotating machinery for the pipe located at discharge side of rotating machinery, except where protected by relief valve.
5. For short period of overpressure ANSI B31 series shall be consulted. For example, according to B31.3, design pressure of piping can be calculated through dividing maximum short-time exposure pressure (Note 1) by:
   - 1.2, if overpressure lasts for maximum of 50 hours at any one time and for less than 500 hours per year.
   - 1.33, if overpressure lasts for maximum of 10 hours at any one time and for less than 100 hours per year.
   - 1.2, if it is due to thermal expansion only regardless of the number of hours.

Note:

1) For remote contingences such as heat exchanger tube rupture, the design pressure can be derived by dividing maximum short-time exposure pressure by 1.5 (or 1.3 if it has been hydro-tested at 130% of design pressure). This is just a design practice not ANSI B31.3 recommendation. The definition of remote contingency may be different from company to company.

Contact

Please visit www.linkedin.com/groups/Chemwork-3822450 should you have any comment, question or feedback or feel free to contact S.Rahimi@gmail.com.