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Importance of accounting for the piping system and boundary conditions in determining the maximum surge pressure following heat-exchanger tube-rupture

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ABSTRACT

Shell-tube type heat exchangers are often used to exchange heat between a high-pressure fluid and a low-pressure fluid, and the pressure difference between the two fluids could be significantly high. If the difference in the design pressure between the low-pressure (LP) and high-pressure sides is greater than that covered by American Petroleum Institute (API 520 and 521) 10/13th rule, dynamic analysis is required to ascertain that the maximum surge pressure that could be reached does not compromise the integrity of the LP side of the exchanger. API guidelines also notes that attention should also be given not only to the shell-side of the heat exchanger under evaluation, but also to the "*upstream and downstream systems*" This paper offers further insight into the importance of including the surrounding piping systems around the subject heat-exchanger where a tube-rupture scenario is considered, and also directs attention to the importance of correctly specifying the appropriate boundary conditions (B.C.) at the far ends of both the upstream and downstream piping systems. It demonstrates the effects of specifying different B.C. on the maximum pressure surge via a case study of a hot separator vapour condenser in a bitumen hydrotreating unit, where the process fluid on the tube-side is a vapour–liquid mixture at 9660 kPa(g). The vapour mass fraction of the process fluid is approximately 0.5, and is mostly hydrogen. The fluid on the LP side is cooling water connected to the plant supply and return cooling systems as well as another adjacent low pressure condenser. The design pressure for the cooling water piping system and the adjacent condenser is 1380 kPa(g).

1. Introduction

This paper is concerned with hazards related to potential tube rupture in heat exchangers in the process plants and the importance of properly designing protective systems to deal with such events. Shell-tube type heat exchangers are often used to exchange heat between a high-pressure fluid and a low-pressure fluid, and the pressure difference between the two fluids could be significantly high. If the difference in the design pressure between the low-pressure (LP) and high-pressure (HP) sides is greater than that covered by the 10/13th rule, dynamic analysis is required to ascertain that the maximum surge pressure that could be reached due to different scenarios of tube rupture does not compromise the integrity of the LP side of the exchanger. Typically the LP side is the shell-side, which contains a low-pressure liquid (e.g. cooling water in hot process vapour condensers, or heating media such as methanol or propylene in ethylene heaters). The 10/13th rule (or 2/3rd rule, as may be applied in some jurisdictions), is based on ASME Boiler and Pressure Vessel Code Section VIII, Div. 1 (2014). This code requirement eliminates the need to evaluate tube-rupture scenarios if the LP side design pressure is equal to or higher than 10/13 times the HP side's design pressure. However, API 521 6th Ed. (2014) also notes that "*Pressure relief for tube rupture is not required where the LP exchanger side (including upstream and downstream systems) does not exceed the criteria noted above. The tube-rupture scenario can be mitigated by increasing the design pressure of the LP exchanger side (including upstream and downstream systems), and/or assuring that an open flow path can pass the tube-rupture flow without exceeding the stipulated pressure and/or providing pressure relief*".

The key phrase in the API 521 clause above is "*including upstream and downstream systems*". Further explanation were provided in API 521 stating that "upstream and downstream piping

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and equipment systems must be thoroughly evaluated when this containment approach is taken to determine the influence of piping either in eliminating the need for a relieving device or in reducing relieving requirements."

The literature contains several attempts to describe the dynamic interactions between the fluid release following a tube rupture and the shell-contained fluids. Nagpal (2015) provided a summary of seven tube-rupture scenarios and associated dynamic simulations, and also offered a practical decision chart based on API 521 for evaluating the tube-rupture scenarios and for determining if steady-state or dynamic evaluation is most suitable. Fowler et al. (1968) predicted pressure transients in the shell of heat exchangers and accounted for the effects of attached piping systems via a shell-piping volume-average approach which did not simulate the dynamic behavior well. Sumaria et al. (1976) used a lumped-parameter approach to account for the dynamics in the attached piping and the pressure safety valve (PSV) riser pipe. While this is a better approach, it still does not account for the spatial and temporal aspects of the transients in the connecting piping systems. Furthermore, Sumaria et al. (1976) addressed the conditions where the fluid on the high-pressure tube-side is strictly gas, i.e. with no changes of phases across the tube break.

Often relief systems are sized to be able to handle the '*mass*' flow from the ruptured tube as in Wong (1992), as if this flow finds its way (i.e., 'short-circuit') to the relief device through the liquid-filled shell. The problem with this approach is that the relieving calculations do not account for the inertia or dynamic effects on the shell-side or along the relief riser, and the concern is that the shell-side may fail before the relief device can react. Examples of incidents of heat exchanger tube rupture causing damage to equipment and subsequent costly plant shutdown due to such a phenomenon have been reported by Simpson (1972).

The present paper offers further insight into the importance of including the surrounding piping systems around the subject heat-exchanger where a tube-rupture scenario is evaluated. The paper also directs attention to the importance of correctly specifying the appropriate boundary conditions (B.C.) at the far ends of both the upstream and downstream piping systems. A case study of a hot vapour condenser (E-10) in a bitumen hydrotreating unit is presented, where the process fluid on the tube-side is a vapour–liquid mixture at 9660 kPa(g), vapour mass fraction of approximately 0.5, which is mostly hydrogen. The fluid on the LP side is cooling water connected to the plant supply and return cooling water system. The design pressure for this cooling water piping system is 1380 kPa(g). However, the design pressure of the shell-side of the subject condenser is 7440 kPa(g) which meets the 10/13$^{\text{th}}$ rule.

What makes the present case study interesting is that there is an adjacent condenser (E-11) to the one that is under evaluation (E-10) for tube-rupture scenario, where its shell design pressure is only 1380 kPa(g). Clearly, both the cooling water piping system as well as the adjacent condenser (E-11) do not meet the 10/13$^{\text{th}}$ rule, despite the compliance of E-10 shell design pressure to this rule. The paper shows dynamic effects of cooling water system via spatial and temporal aspects of the flow transients and associated B.C. by solving the one-dimensional hyperbolic partial differential equations of the conservation of mass and momentum as applied to the cooling water network connected to both exchangers.

2. Dynamics phenomenon following tube rupture and modeling equations

Ennis et al. (2011a and 2011b) recognized four distinct phases of a tube rupture scenario where a high-pressure sub-cooled flashing liquid is released into a liquid-filled low-pressure heat exchanger shell. These four phases are associated with systems where relief valves and/or rupture discs are employed. In cases where none of these are employed, only the first phase is applied, which is related to the pressure buildup (surge) due to the accumulation of the effluent fluid in the exchanger shell. This is the most important phase and is the subject of this paper. Upon a full-bore tube rupture, the process fluid in the tube (which could be a mixture of vapour and liquid) will expand from the high-pressure tube-side and will cause the pressure on the shell-side to increase. The rate of pressure increase as well as the maximum pressure reached depends on the following parameters:

i) Rupture scenario and rupture opening area.
ii) Thermodynamics and mass flux of the fluid released through the rupture opening.
iii) Adiabatic flashing characteristics of the high-pressure mixture into the low-pressure shell.
iv) Initial pressure of the liquid-filled shell in relation to the set pressure of the relief device (if exists).
v) Shell volume, compliance, and the characteristics of the attached piping system.
vi) Additional equipment in the vicinity of the subject heat exchanger, e.g. other exchangers, accumulators, valves, flow restrictions, etc.
vii) Boundary conditions at the far ends of the attached piping system.

The above parameters are accounted for by the pertinent governing equations which will be described in the next sub-Sections. When the pressure on the shell-side is relieved via a relief device, typically the pressure drops quickly depending on the combination of fluids in the shell at the time of relief and on the bulk modulus of the liquid on the shell-side. The main assumption in the treatment to follow is that the effluent fluid from the ruptured tube(s) into the shell side of the heat exchanger undergoes adiabatic expansion (with increase in entropy) from the condition in the tube (high pressure) to the prevailing condition in the shell (generally lower pressure) at any given time during the rupture event. This adiabatic expansion may entail flashing of the effluent fluid into vapour and multiple liquid phases, where the contribution of each ought to be considered in the formulation of the physical dynamic equations.

One additional fundamental assumption which characterize the present problem is that the time scale of the pressure rise (surge) in the heat exchange shell during a tube rupture event is much longer than the time scale of any incremental small perturbation to settle out inside the shell. This is primarily due to the fact that the speed of sound of the cooling (or heating) fluid on the shell side of the heat exchanger is relatively high such that any perturbation resulting from the incremental effluent fluid being admitted into the shell from the tube side during an incremental time step of the process will be felt everywhere in the shell fluid almost instantaneously relative to the time scale or the duration of the pressure surge till final steady state. That is, the physics of the process during tube rupture allows for developing a model whereas the pressure surge in the shell can be treated in a quasi-steady manner. Stated differently, the pressure response of the fluid in the shell side can be dealt with via the bulk modulus property of the shell fluid, provided of course that there is no overhead vapour to start with, like e.g. the case of a kettle-type heat exchanger.

2.1. Dynamics of the process on the tube-side

As mentioned above, the fluid on the high-pressure tube-side