Engineering Consulting & Training Services

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Designing Delayed Coker Drums to ASME VIII Division 2

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Coke drums in delayed coker units are pressure vessels that are used in the oil sands and conventional petroleum refinery processing industries for the thermal cracking of reduced bitumen to recover additional saleable gas and liquid product streams. The drums are typically constructed to the requirements of ASME VIII Division 1, although they are in a low cycle, thermal-mechanical service environment. Recent practice has been to introduce design by analysis considerations from ASME VIII Division 2, even though service temperatures of the actual equipment exceed the design limits of the Division 2 Code [1, 2].

The coke drum is subjected to severe thermal cycling. It typically experiences temperature changes from ambient to 900°F [482°C] during a fill sequence extending over a time period from 12 to 24 hours. At the end of this fill sequence, a rapid steam quench is followed by a longer water quench, which cools the remnant material to a safe temperature but which mechanically strains the drum shell. Maximum Operating Pressures (MOP) extend from 35 to 50 psig [241 to 345 MPa] and are applied in a cyclic manner from 0 to the MOP over the full operational cycle. The modern coke drum has been in use since 1938 [3], and drum shell bulging and cracking have been documented since the 1950s [4]. Although the pressure and temperatures of the operating streams are known and controlled, the actual loading on the pressure envelope of the drum is not certain, especially during the water quench stage where large volumetric flow rates are impacted by the distribution, density, porosity, and internal channelling within the remnant residual coke mass.

The substantial problem facing the designer is the lack of definition of the thermal loads during the operational cycle and hence, precluding effective use of the Division 2 design-by-analysis methodology. A bounding approach to the drum's mechanical design can be considered and is illustrated in this article.

There are three significant loadings related to the imposed thermal loads. Heat up of the vessel during steam test, vapor heat, and initial oil fill results in the drum reaching a temperature of approximately 900°F. Since the drum is free to expand at locations away from the skirt support attachment, no mechanical strain would ordinarily be anticipated. However, the use of a Type 410S stainless steel clad liner imposes a loading caused by differential thermal expansion of the clad and shell base materials, typically a low alloy carbon steel.

Upcoming Training Courses held in our Rockaway, NJ office

- Course 607, *Design and Maintenance of Aboveground Atmospheric Storage Tanks,* November 1-3, 2011
- Course 1302, *Relief System Design,* November 8-10, 2011
- Course 1613A, *Turnaround Best Practices – Planning, Scheduling, and Management of Shutdowns,* November 15-17, 2011

Work Highlights

Fired Equipment/Heat Exchangers

On behalf of the refinery owner, full time engineering design audits of fired equipment and heat exchangers being provided for a major clean fuels project being designed in the US for a project located in the Middle East. This audit involves periodic visits to the prime contractor's office to review equipment design drawings and calculations, discuss the designs with contractor and supplier engineers, plus visits to equipment supplier locations to discuss critical equipment matters.

Reliability and Maintenance

Performed a Cold Eyes Review of the Reliability and Maintenance Program of a major European refinery. This was done as follow-up to previous work that we had done with them and now focused on their life cycle maintenance plans. It was concluded that significant improvements have been made in their program. However, recommendations were made to develop a comprehensive, integrated data management system rather than continue using multiple independent databases.

A useful expression to calculate this is the following:

$$\sigma_1 = \left[\frac{(\alpha_2 - \alpha_1) \cdot (T_1 - T_0) E_1}{1 + \frac{t_1}{t_2} \cdot \frac{E_1}{E_2}} \right] \cdot \frac{1}{1-\mu} \qquad \text{and} \qquad \sigma_2 = -\sigma_1 \cdot \frac{t_1}{t_2}$$

where σ_1 represents stress in the base material and σ_2 represents stress in the clad material. For a typical drum, operating with a 900°F wall temperature, this amounts to a base material stress of -4,238 psi and a clad stress of 42,384 psi, all based on typical thicknesses and physical properties used for coke drum construction.

During oil fill, vapor separates from the liquid, and a remnant carbonaceous material, or coke, is produced. This coke begins to cool, resulting in a cooling off of the drum shell once the liquid front passes a location. This cooling off results in the drum contracting while the upper levels remain at an expanded diameter from the initial temperature exposure. This results in a "vasing" effect. This can be modeled using Finite Element Analysis (FEA) and can be shown to be non-deleterious. However, a more severe vasing effect occurs during water quenching where an accelerated water fill rate causes a whipsaw curve in the stress profile as the initial cooling front passes. Rather than a single thermal cycle, two cycles are effected. Figure 1 shows the nominal quench and the "whipsaw" effect for an imposed accelerated quench.

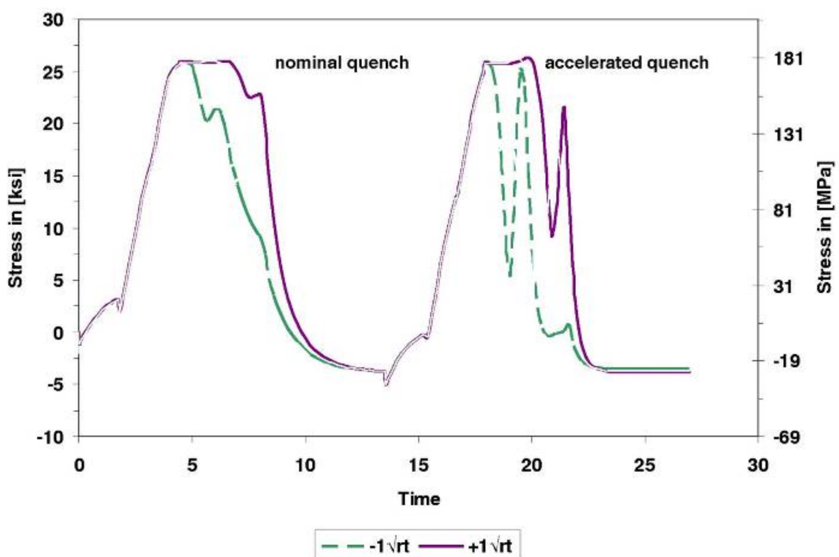

Figure 1 Axial Stresses at Shell ID due to Axial Temperature Gradients

The accelerated quench occurs because the incoming water is not able to rise uniformly within the non-homogeneous remnant coke mass and must find paths of lesser resistance. When severe restrictions impede the incoming quench water, or more advantageous paths are available, hot and cold spots form on the shell. Since these hot and cold spots occur randomly, a deterministic evaluation is made nearly impossible with the data usually available in an operating facility. A possible approach is to bound the problem.

The basic thermo-mechanical loading is described by:

$$\varepsilon = \pm \alpha \cdot \Delta T / 1 - \mu$$

which describes the strain induced by self-constraint of the shell when a hot or cold spot occurs [3]. For the typically encountered operating conditions in a coke drum, a strain of ± 7,200 με may be realized. Note that yield strength is usually defined as the stress at 0.2% offset strain or 2,000 με. Low cycle thermo-mechanical fatigue is an expected outcome. In an actual operating unit, these posed hot and cold spots do not occur at each and every operating cycle, and hence, a real operating unit would not necessarily fail at the predicted lower bound service life of 600 cycles. It is interesting to note that early failure has been described for a unit at 780 cycles [4].

This partial understanding of the thermo-mechanical loading mechanism, and impact on strains and stresses developed from these loadings, indicate that appropriate material selection and management of vessel operating controls can reduce the susceptibility to fatigue failure and improve vessel reliability. Therefore, it is important to prepare a detailed technical specification whenever new coke drums are to be purchased. This specification will typically specify drum design conditions, details of the operating cycles (temperature, pressure, duration, frequency, etc.), material specifications, and the locations of the required onstream temperature monitoring instruments.

With increased understanding of the thermo-mechanical loading mechanism, and impact on strains and stresses developed from these loadings, appropriate material selection, vessel design details, and management of vessel operating controls can reduce the susceptibility to fatigue failure and improve vessel reliability.

1. ASME, "VIII Division 1 Rules for Construction of Pressure Vessels," 2010 ASME, New York, NY

2. ASME, "VIII Division 2 Alternative Rules for Construction of Pressure Vessels," 2010 ASME, New York, NY

3. Harvey, J.F., "Theory and Design of Pressure Vessels," Van Nostrand Reinhold Company, New York, NY 1985

4. McGowin & White, "Coke Drum Fracture Experience and Analysis," Proceedings / Division of Refining, American Petroleum Institute, v. 51 1971: 778-807

About the Author

John Aumuller has over 32 years experience in the execution of refinery, oil sands, heavy oil, power, and petrochemical plant projects as a mechanical engineering specialist in piping and pressure equipment stress analysis, including use of finite element analysis, refractory systems evaluation, and as a project engineer and manager. He has extensive expertise in execution of maintenance and plant mechanical engineering activities. Expertise includes Fitness-for-Service (FFS) and remaining life assessments using ASME FFS-1/ API 579. Expert knowledge of industry relevant codes, such as ASME Boiler and Pressure Vessel Code Section VIII Div 1 and 2, Section I, B31.1, B 31.3, CSA Z662, and related standards, including API standards.

Please contact Vince Carucci (vcarucci@carmagen.com) if you'd like more information on Carmagen's expertise in this area.