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ANCHOR BOLT FAILURES IN COKE DRUMS

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ABSTRACT

Anchor bolt failures that are commonly experienced by coke drums typically show signs of axial and/or shear overload. Temperature measurements and thermographic images have demonstrated that side-to-side temperature differences in coke drums can be more than 400 degrees Fahrenheit. In this paper, finite element models are used to examine the displacement of skirt base plates. Various combinations of circumferential thermal gradients, support deck slopes, and friction coefficients between drums and support structures are used. The study shows that observed thermal gradients can result in significant longitudinal, circumferential, and radial displacements and explains observed overload failures of anchor bolts. In addition, the study demonstrates the roles of deck slope and friction coefficient on base plate displacements.

INTRODUCTION

Coke drums are one of the most failure-prone pressure vessels in oil processing and refining. One of the common failure modes encountered in coke drums today is the failure of anchor bolts which typically starts with one or two bolts and ends up with multiple bolt failures, grout failure, and drum relocation. Most failed bolts show signs of overload and/ or fatigue damage. Unlike other failure modes that can cause unplanned outages and fires, the main consequence of bolt failures is the costly and time-consuming repairs.

In their classic 1958 paper, Weil and Rapasky examined the eight most common failures in coke drums (1). Anchor bolt failure was not among them and was not even included in the topics of the Discussion section that followed the paper. Therefore, one must conclude that either this failure mode was not as common at the time or that it was not perceived to be as serious as the other failures.

In the Discussion section of the above paper, G. Derickson, an operating company engineer, discussed the phenomenon of "jumping of coke beds" which, in one instance, reportedly caused a 13-foot diameter drum "to jump vertically approximately 18 inches during the quench cycle". Derickson reported that the drum parted off its skirt and resettled back in it after causing extensive damage to the supporting structure. He also predicted that the skirt attachment weld had failed prior to this event. In their response, Weil and Rapasky confirmed that they had observed jumping on several installations although not to the extreme extent mentioned by Derickson and predicted the cause of jumping to be the bursting of large vapor bubbles generated when processing lighter feeds (1). These observations suggest that large uplift forces are possible when processing light crudes in very small-diameter drums. Apparently, these uplift forces are insignificant in typical drums.

The 1996 API Coke Drum Survey did not include bolt failures in its list of questions to operating companies (2). It did not even discuss or mention this failure mode in the report. Again, it is unclear if these failures were not common at the time or were simply not perceived as sufficiently significant to warrant inclusion in the survey.

The most recent API Technical Report 934G on coke drums (3) contained a small section on foundation bolting. It stated that “Experience shows that foundation bolting can crack as a result of movement including rotation of the base plate on the drum support skirt” but it provided no reasons for this movement. Results of the 2013 survey, which were included in the same report, revealed that 25% of respondents noticed an increase in anchor bolt problems but it did not state the number of respondents who reported such failures.

From our experience over the last 25 years, bolt failures appear to be occurring more frequently than before. We are aware of several dozen failures that have resulted in replacements of bolts and major refurbishing of grout.

In 2011, Du Plessis reported on a history of anchor bolt failures that were determined to be a combination of shear and tension (4). Failed bolts were found to have elongated up to 0.5 inches. The paper discussed similar failures at other plants and described a retrofit that included more and larger bolts with Bellville washers.

With the apparent increase in bolt failures, at least one alternate anchoring system was developed to replace conventional anchor bolts, (5). The non-bolted restraint utilizes expansion gaps that allow the base plate to deform in response to severe thermal transients before movement is stopped. This non-conventional restraint is being used to retrofit a set of drums after they experienced significant bolt failures.

We suspect that the apparent increase in frequency is related to the following industry trends:

1. The increase in drum diameter and height,
2. The decrease in cycle time, and
3. The increase in use of side inlet nozzles.

While the increase in drum size and weight is expected to decrease uplift forces on anchor bolts that may be generated by any possible “drum jumping”, the increase in size amplifies the impact of the other two factors on increasing spatial and temporal thermal gradients in the drum in general and in the cone in particular.

Recent infrared image and measurements obtained from operating coke drums demonstrated that temperature differences in excess of 400 ^{o}F (205 ^{o}C) are common around the circumference of cones. Severe circumferential gradients are believed to be a result of uneven water flow during quenching.

In this paper, we use finite element analysis to examine the impact of severe cone thermal gradients on the deformation of skirt base plates. Results help us understand the role of thermal gradients in bolt failures and estimate loads to which anchor bolts are exposed during such events.

MODEL

A finite element model of a coke drum is used to examine the impact of cone thermal gradients on baseplate deformation under a range of deck slopes and friction coefficients. The drum is 30 feet in diameter, 100 feet in length, and has a 1.6 inch shell, 2.4 inch cone, 1 inch skirt, and 1.75 inch base plate. The drum is free to move on top of a rigid deck that ranges in slope from zero to 10% with a friction coefficient that ranges between 0.3 and 0.9.

The coupled thermal-stress analysis was conducted using nonlinear geometry and elastic-plastic material model. Three-dimensional linear quadrilateral shell elements were used, as shown in the mesh of Figure 1.

The analysis was conducted in four loading steps.

1. Drum weight is applied.
2. Drum tilting is simulated along the X-axis.
3. Thermal gradients are imposed
4. Thermal gradients are removed.

Typical temperature measurements point to the existence of severe thermal gradients both in axial and circumferential directions during quenching.. For simplicity, only circumferential gradients were imposed on the model. As shown in Figure 2, assigned temperature linearly varied from 800 to 200 and back to 800 degrees Fahrenheit across the diameter of the cone. The same thermal distribution was also imposed on the shell. In the skirt, temperature was distributed by conduction from assumed temperatures in the drum.

In the final step, temperature was assigned back to uniform ambient temperature (70 ^{o}F) for the whole model.

RESULTS

The following contour plots describe the condition of the drum at the end of an analysis (0.9 friction coefficient and 10% slope) when the drum is back at uniform ambient temperature:

- Figure 3: Displacement in downhill direction (X-axis)
- Figure 4: Axial displacement (Y-axis)
- Figure 5: Lateral displacement (Z-axis)
- Figures 6 and 7: Von Mises stress on inside and outside wall surfaces, respectively
- Figures 8 and 9: Equivalent plastic strain on inside and outside wall surfaces, respectively

These figures show that a single severe thermal cycle results in substantial residual plastic deformations and stresses in the base plate. More importantly, as shown in Figure 10, after completing a single cycle, the drum as a whole moves downhill a significant distance directly proportional to surface tilt and inversely proportional to friction coefficient. A walking distance between 0.05” and 0.2” is predicted under a tilt of 1%

which is sometimes intentionally introduced for correction of drum bowing and often found in cases of foundation settlement and grout failure. The unrealistically high 5-10% range of tilt is included in the study to examine extreme cases. As shown in the graph, the tilt-displacement relationship is mostly linear but friction-displacement relationship is nonlinear.

CONCLUSION

The finite element analyses described in this paper demonstrate that the severe thermal gradients commonly observed in normal operations of coke drums can result in drum walking as well as significant residual plastic deformations and stresses at skirt baseplates. Predicted baseplate displacements can impose excessive cyclic stresses on anchor bolts which may lead to failure. It is our judgement that the most effective way to address bolt failures in conventional welded skirts is to eliminate bolts and replace them with movement-activated restraints (5) that permit skirts to freely deform in response to large thermal gradients.

Above results and conclusions are based on assumed design, loading conditions, and range of parameters. This study is only intended to examine the nature of baseplate reaction to thermal gradients in coke drums. Assessment and analysis of restraints should be performed using the actual design and thermal gradients of the subject drum.

ACKNOWLEDGMENT

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(3) API Technical Report 934G - Design, Fabrication, Operational Effects, Inspection, Assessment and Repair of Coke Drums and Peripheral Components in Delayed Coking Units, pending release.
(4) Du Plessis, P., 2012, “Coker Anchor Bolts Failure Analysis”, IPEIA 2012 Meeting, Banff, International Pressure Equipment and Integrity Association.
(5) US Patent 2015/0273422, 2014, Pressure Vessel Restraint for Accommodating Thermal Cycling.

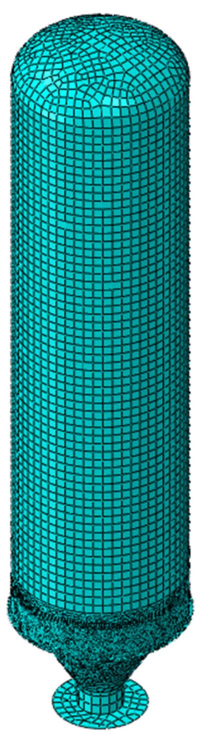

Figure 1. Finite element model

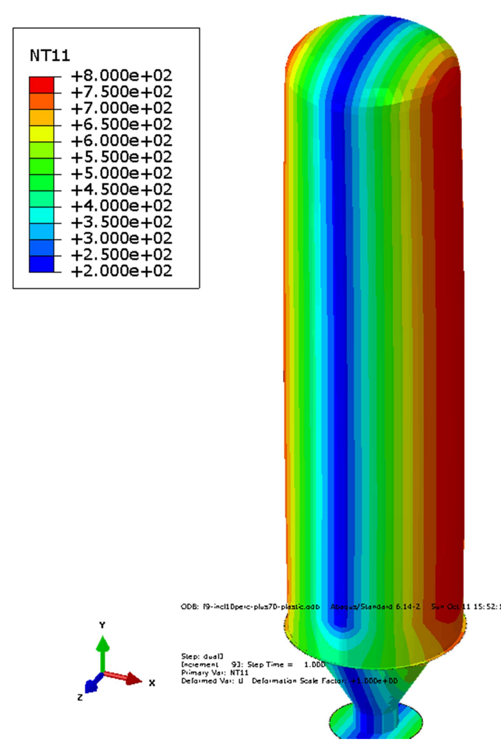

Figure 2. Temperature contour of thermal load

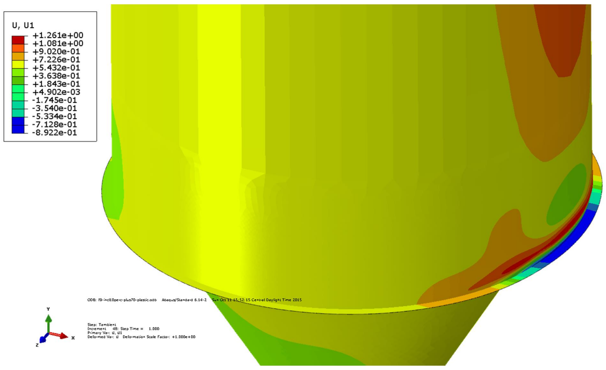

Figure 3. Displacement in downhill direction at ambient

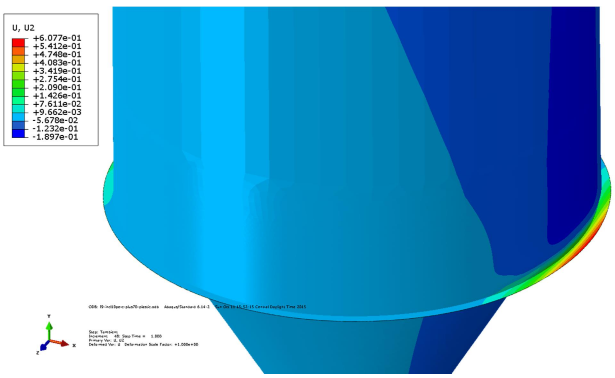

Figure 4. Axial displacement at ambient

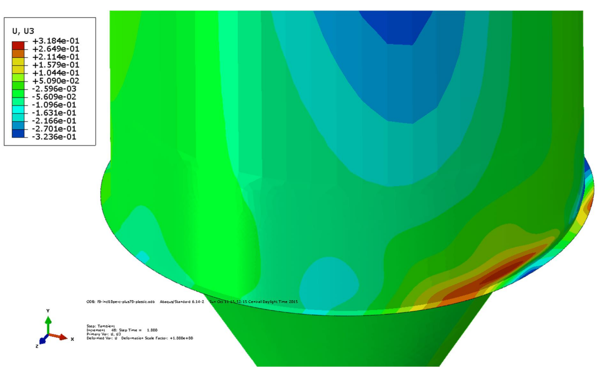

Figure 5. Lateral displacement at ambient

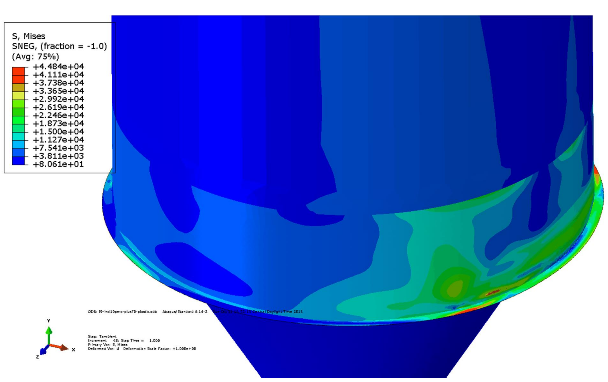

Figure 6. Von Mises stress on inside wall surface at ambient

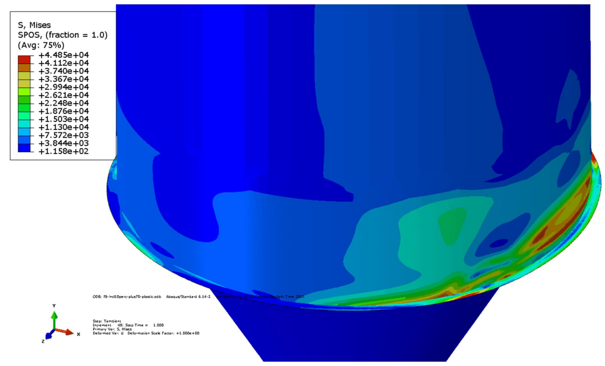

Figure 7. Von Mises stress on outside wall surface at ambient

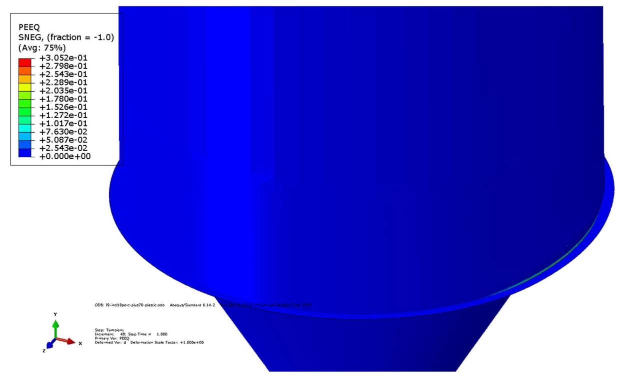

Figure 8. Equivalent plastic strain on inside wall surface at ambient

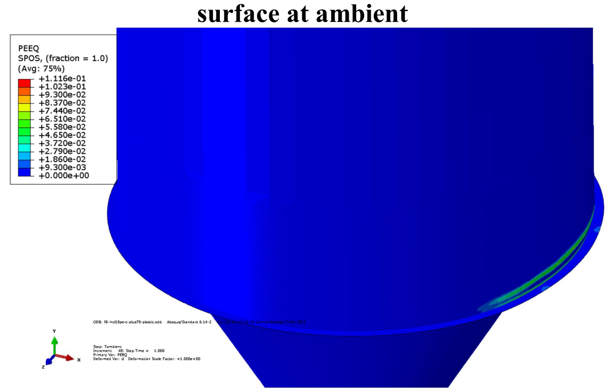

Figure 9. Equivalent plastic strain on outside wall surface at ambient

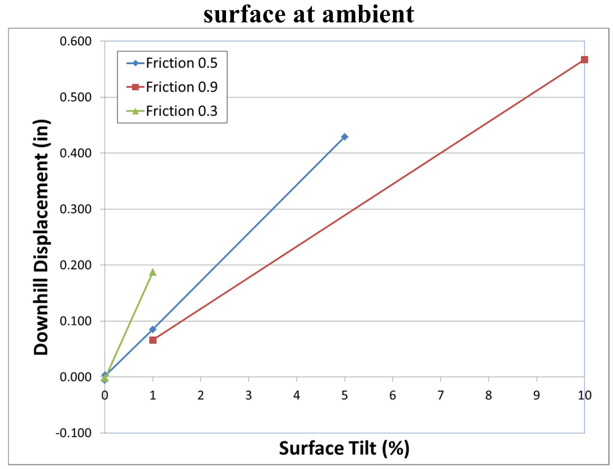

Figure 10. Downhill displacement at ambient as a function of surface tilt and friction coefficient