INVESTIGATION AND REPAIR OF A HEAT EXCHANGER TUBESHEET-TO-CHANNEL FLANGE

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ABSTRACT

During its fabrication hydrotest, the flanged joint between the tubesheet and the channel of a shell and tube heat exchanger leaked.

The design of the joint was confirmed as complying with the ASME Boiler and Pressure Vessel Code Section VIII Division 1[1]stress requirements and rigidity index recommendations. The joint was investigated using finite element analysis (FE). The results indicated that the flange was rotating significantly during bolt up and under pressure. The flanged joint design was judged to be unacceptable and the design was converted to a welded configuration.

This paper reports the results of the FE analysis and the ASME BPVC Section VIII Division 1 flange design calculations. The results of commonly used mechanical and code design software are also discussed. These results are compared and recommendations for the design of similar flanges are presented.

INTRODUCTION

The subject heat exchanger was fabricated from SA-240-316 materials with a removable channel design that results in a flanged joint with the tubesheet serving as one flange. The mating flange is on the channel. This joint leaked on original hydrotest. The exchanger is an 80" nominal shell diameter with 250 psi and 1000 F design conditions. The design gasket is a spiral-wound type with 316 windings and gauge rings at the OD and ID of the windings. The studs were SA-193-B8 with compression spring washers for differential thermal expansion considerations. The joint leaked before reaching 375 psi and the leak rate increased as the test pressure approached the required 461 psi. The joint gasket contact surfaces were re-machined and carefully assembled with a new gasket. The second hydrotest resulted in leaks at the joint before reaching 400 psi and, again, the leaks increased as the pressure was increased to the required test pressure of 461 psi. A third hydrotest was attempted using a carbon faced corrugated metal sheet. The compression spring washers were eliminated. The studs were tightened by use of actual stretch methodology. The joint again leaked before reaching 400 psi. The pressure had to be reduced to 375 psi to stop the leakage. At this point it was evident that delivery of the exchanger would be delayed until a successful hydrotest was achieved.

The failure of the exchanger to achieve a satisfactory leak-free hydrotest resulted in a thorough review of the design of the tubesheet-to-channel joint. The design review which included the flange rigidity investigation per ASME Boiler and Pressure Vessel Code Section VIII Division 1 Appendix S-2 (d)[1] found that the normal ASME flange design criteria had been achieved. It should be noted that it is common practice for design engineers to conclude that a typical process industry exchanger channel to tubesheet flanged joint design is satisfactory if the ASME Section VIII Division 1 [1] stress and rigidity index calculations give acceptable results.

It was apparent that further investigation of the flanged joint design was required in order to understand the redesign requirements needed to achieve a successful hydrotest.

INVESTIGATION

The tubesheet-to-channel joint design was reviewed using Finite Element Analysis (FE) to investigate the flange stresses and movements during the hydrotest. The design was reviewed for flange rotation during bolt up, cold hydrotest conditions, and for thermal expansion conditions during start up and normal operation. From previous exchanger investigations, the authors had concerns about gasket surface rotation and deflections and gasket contact surface scuffing, as reported by Martens and Porter, 1994 [2].

MODELING

An FE model (Figure 1) was constructed using axisymmetric finite elements. The tube field in the tubesheet and the flange bolting are not truly axisymmetric. However, for preliminary investigations,

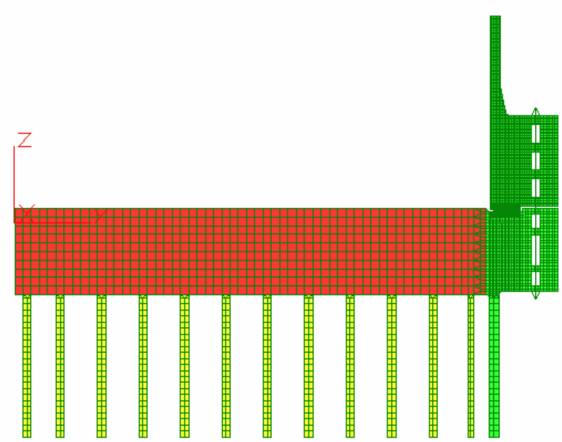

Figure 1
FE Model of Tubesheet and Flange

this lack of symmetry was not considered to be significant enough to have a major effect on the results. The tube influence on the tubesheet was approximated by use of equivalent axisymmetric rings

The gasket and flange contact surfaces were simulated with a row of elements. Elements that exhibited a tensile load at the gasket-to-flange interface were removed from the model. The model was then re-run. This process was repeated until only a compressive load remained at the interface. The thermal transient and steady state conditions were also investigated to determine their impacts on the established design.

RESULTS AND DISCUSSION

Flange rotation was determined to be the limiting condition for the hydrotest leak problem. The flange resists the hydraulic end force from the channel and rotates in torsion as the end force is transmitted to the bolts. Table 1 provides the gasket surface rotation angle for the tubesheet and channel flange as the joint is bolted up and the pressure is increased to hydrotest pressure.

Table 1

Loads		Rotation Angle		
Bolt Preload - LBS	Pressure - PSI	Tubesheet	Flange	Combined
0	0	0.000	0.00	0.000
20,000	0	-0.016	0.038	0.055
40,000	0	-0.033	0.076	0.109
60,000	0	-0.049	0.115	0.164
60,000	250	-0.087	0.227	0.314
60,000	461	-0.051	0.645	0.696
60,000	461*	-0.051	0.367	0.448

* barrel removed from model

The rotation of the flange is strongly influenced by the hydrostatically induced expansion of the exchanger barrel. The final row in Table 1 indicates the flange rotation computed without the expansion of the barrel of the channel included in the model. This was done to determine the barrel's effect on the flange rotation. We can see that without the barrel's contribution, the rotation is considerably less. This would seem to indicate that the current Code [1] flange rigidity index calculations may not accurately represent the flange rotation because they do not address the contribution of the barrel expansion to the flange movement. In the subject case, the hydraulic pressure expanded the channel barrel such that the flange became a restraint to the barrel's expansion. This caused the flange to rotate more than it would have with only the hydraulic end load applied.

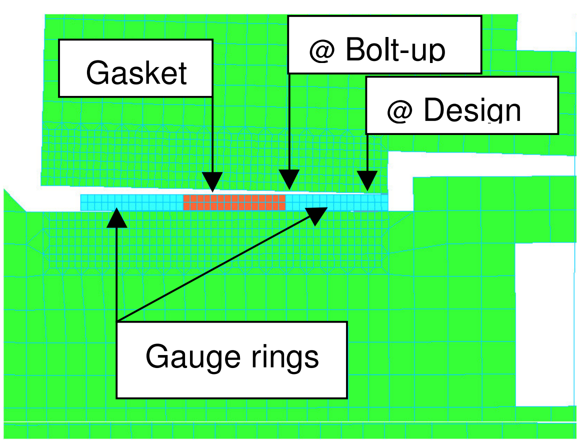

Figure 2
Deflected Shape with Contact Points Indicated

Table 2

Loads			
Bolt Preload - LBS	Pressure - PSI	Contact Radius* - IN	Required Rebound Mid Point Gasket
0	0	41.41	0
20,000	0	41.88	0004
40,000	0	41.88	.0008
60,000	0	41.88	.0012
60,000	250	42.38	.0052
60,000	461	Open	.0251

*Inside radius of outer gauge ring @ 41.78" and outside radius of inner gauge ring @ 41.03"

As the flange rotates, the gasket's contact surface lifts away from the gasket. Table 2 provides the radius at which the gasket contact surface is no longer in compression. At the beginning of the original bolt up, the gasket is in contact with the complete surface. The gasket surface contact radius begins to significantly move outward as bolt up is completed and hydraulic forces develop. Thus, the rotation of the flange reduces the gasket contact surface and the resulting contact stress. Figure 2 indicates the rotation of the flange. From inspection of Table 2 and Figure 2, it is apparent that the contact point of the flange and gasket (where the resistance to the bolt load begins) is actually outside the diameter of the spiral-wound gasket and is instead on the outer gauge ring. This is true both at bolt up and at design pressure.

Hsieh, et al (3) have reported that the spiral-wound gasket is able to recover (rebound) only about 0.005" of the initial bolt up compression deflection. As may be seen from Table 2, a larger rebound than this would be required at the design pressure. At the hydrotest pressure, nearly 5 times the available rebound would be required to maintain a seal. To expect the spiral-wound gasket surface in this situation to rebound such that it will provide any significant sealing ability is not reasonable. Thus, the linear analysis employed here is non-conservative as far as its prediction of the onset of leaking. After bolt up, when the contact point is indicated to be on the outer steel ring, a rebound of approximately 0.001" in the spiral-wound gasket is indicated as required to maintain a seal.

As we can see in Table 2, the model indicates that the outer gauge ring is no longer in contact with the flange before the hydrotest pressure of 461 psi is reached. During the actual test, the exchanger started leaking at a pressure slightly less than 400 psi.

As reported by Martens and Porter 1994 [2], the tubesheet will heat up much more rapidly than the flange, due to the effect of the tube hole-related heat transfer. Thus, a thermal expansion analysis to evaluate gasket scuffing was necessary. The analysis indicated that the gasket would be subjected to 0.150" of differential surface movement (scuffing) of the gasket sealing surface during the expected rapid heating during the startup phase of the process. There is little information available as to the acceptable scuffing of a spiral-wound gasket. However, the gasket vendor indicated that scuffing greater than 0.030" was a concern for maintaining gasket sealing ability.

Table 3		
	Design: 250 psi @ 1000º F	Hydrotest: 460 psi @ 70º F
Code Calculated Radial Stress	894	1645
Code Calculated Tang. Stress	7230	13303
Code Allowable Stress 1995 Ed.	11300	17500
Code Acceptable Rotation, Degrees (Rigidity index of 1)	0.3	0.3
ASME Rigidity Index, J< 1	0.667	1.024
Angular deflection per Roark, Section 10.9	0.253	0.391

The investigation included verification of the ASME rigidity index calculation. The ASME rigidity index from ASME Sect VIII, Div.1, S-2 (d)[1] is shown in Table 3. The index is well below the maximum allowed value of 1 for the design case, but rises slightly above 1 when the hydrotest case is evaluated. As reported above, the influence of the channel barrel is not included in the ASME rigidity index calculation. The allowable rotation is not specifically stated in ASME Section VIII Division 1 Appendix s-2 (d) [1]; however, discussions with Code committee members indicate that the K1 factor is the allowed angular deflection, in this case 0.3 degrees. When comparing the FE-reported rotation without the influence of the channel barrel of 0.37 degrees at hydrotest pressure to the rigidity index of 1.025 at hydrotest pressure (approximately 0.31 degrees), we find that the FE and rigidity index results are in reasonable agreement. The actual rotation angle of 0.645 degrees reported by the FE would exceed the Code implied 0.3 degrees maximum by more than a factor of two. However, the angular deflections calculated from the Roark formula [4] are greater than the allowed Code rotation in the hydrotest case. Neither the Code calculation nor the Roark formula includes rotational effects from the deflection of the attached shell due to pressure. This points to a problem in the use of the ASME rigidity index calculation as a qualifying investigation calculation for a flange structure, such as this exchanger channel flange.

The flange design stresses (see Table 3) are well below the ASME Section VIII Division 1 [1] allowables. The FE results were in general agreement with those listed in Table 3. At hydrotest conditions, the Table 3 radial stress was 1645 psi vs the FE value of approximately 1650. The Table 3 tangential stress was 13,303 vs the FE computed 15,200 psi, much less than the 17,500 psi allowed by the Code [1].

The channel flange design was also verified using a commercially available shell and tube exchanger mechanical design program. The program was used to do an independent design of the exchanger flange with the same design conditions as those given to the supplier. Design of channel flanges using ASME Section VIII, Division 1 [1] criteria is a standard part of the commercial exchanger design program package.

The flange design from the commercial exchanger design program was compared to the supplier's flange design. The commercial design program used the ASME Code [1] flange design approach for verification of stresses and reported the ASME Code [1] rigidity index for reference. This is a typical methodology for determining if an acceptable flange design has been achieved. There was a 7% increase in flange thickness between the commercial design and the supplier's design. The commercial software design used a shorter hub length causing the flange to be thicker than the supplier's flange. This difference is designer preference since both designs meet ASME Section VIII, Division 1 [1] criteria. The operating load rigidity indexes were 0.715 for the commercial design and 0.729 for the supplier's design.

Both flanges have an operating load rigidity index factor well below the Code recommended limit of 1.0 and yet the exchanger leaked. This suggests that there is a problem with applying the ASME rigidity index calculation and the Code acceptable limit of 1.0 to a flange welded to a channel barrel. The program used the TEMA (5) tubesheet design approach. The thickness reported was essentially the same as the original design.

Designers who use commercial shell and tube mechanical design programs should use these programs with caution. The software solution may be using ASME Code [1] allowable stresses and rigidity index to establish and qualify that the flange design as acceptable. If the designer uses a program where the rigidity index is used as a reporting tool (as was the case with the program used for this study), then the flange thickness should be reviewed by the designer and adjusted, as appropriate, to include the effect of the channel barrel.

CONCLUSIONS

For this particular flanged joint, the Code [1] rigidity index and stress calculations did not comprise a satisfactory design acceptability criterion to confirm satisfactory sealing ability. Additional analysis of the contact stress movement of the flange gasket surface is necessary. Factors such as the joint's ability to maintain sufficient compressive contact stress on the gasket sealing surface, the rebound ability of the gasket, and scuffing of the gasket must be fully understood as these factors govern the suitability of the joint design.

The ASME Section VIII Division 1 [1] design-by-rule calculations did confirm satisfactory stress and rigidity for the flange. The commercial mechanical design software used these same design-by-rule calculations as the basis for the flange design. However, the FE investigations indicated that the flanged joint rotated excessively and actually used the outer gauge ring to provide the sealing surface. The FE models indicated that the spiral-wound gasket would require rebound with as little as 0.1° rotation, which occurred during the original bolt up. At the design conditions, the flange rotation was less than the Code allowable 0.3 degrees. Nonetheless, more gasket rebound was required to maintain a seal than the manufacture recommended as a maximum. The gasket scuffing conditions created by the differential thermal expansion of the flange and tubesheet under various operating conditions also indicated that it would be difficult to maintain a leak tight-joint without ongoing maintenance activities.

The only practical solution to significantly reduce the rotation was to replace the flange with a much thicker structure and reduce the rotation to acceptable limitations. However, replacing the flange with a more rigid structure would not have addressed the thermal-induced gasket scuffing movement.

The operating service conditions coupled with the physical size and location of the exchanger did not lend themselves to the possibility of future field re-tubing. Therefore, the resulting recommendation was to eliminate the flanged-joint by welding the channel directly to the tubesheet and to provide a man-way in the channel for inspection and maintenance activities. This design change would require the removal of the channel head for tube replacement, which was considered acceptable.

RECOMMENDATIONS

Due to all loads on the flange (including hydraulic end forces) and the effect of the channel barrel attached to the flange, the design engineer must address the flange rotation. The deflections and thermal movements of the flange structure are very complex. A thorough investigation of the ability of the gasket to maintain a satisfactory seal needs to be undertaken for critical service, large diameter stainless steel exchanger joints. The gasket rebound necessary to maintain a tight joint and the gasket contact conditions rather than an arbitrary structure rotation limit must govern the design. The scuffing displacement of the gasket during startup and operation should also be a consideration for confirming the acceptability of the final design.

It is recommended that ASME consider including all the effects of the element attached to the flange including any expansion of the element due to pressure loading in the rigidity index calculation.

There is little in the ASME Code [1] or TEMA [5] in the way of guidance for establishing the tubesheet thickness when the tubesheet is incorporated as part of a flanged joint. However, as shown in this case, a TEMA tubesheet design which results tubesheet the same thickness or thicker than the mating flange is usually satisfactory from a bolted joint perspective. If a detailed rotation investigation is undertaken for the channel flange, it is recommended that the tubesheet portion of the joint also be included in the investigation, as the effect of both elements on the gasket must be addressed.

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