Thermal stress analysis of a damaged waste heat boiler

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Abstract. Waste heat boilers are widely used in the petroleum refining industry to improve energy efficiency. However, improper design, operation or maintenance of these boilers may result in costly failures. Cracking was found repeatedly at the same location on a circumferential weld of a certain waste heat boiler in CPC refinery. Hence a special designed fixture was used to avoid crack extension. Root cause mainly due to improper arrangement of internal tubes and external piping was then mainly found by thermal stress analysis. Recommendations was proposed to shut down the boiler immediately and to rearrange the tubes and piping without changing basic design of the waste heat boiler.

1. Introduction
A typical convective type of waste heat boiler (WHB) used as part of reformer system is shown in figure 1. The hot hydrogen gases from reactor are carried through a number of parallel small diameter tubes, while the incoming water to be heated enters a shell surrounding the tubes and passes directly over the hot tubes in the direction normal to their axes. During operation, cracking was found on a circumferential weld repeatedly at the same location after repairs or a new replacement and, a special designed fixture was used to prevent from crack propagation as shown in figure 2 [1,2]. Since the WHB and surrounding piping system are operating at high temperatures, one of two supports under the heat exchanger is fixed to the base and the other is allowed to slide horizontally to accommodate periodic temperature change. During site survey, the circumferential weld with cracks was found close to the side with sliding support. In addition, the high temperature intake manifold from reactor was parallel to the WHB and connected to the sliding side also. To find the root cause, an analysis model was formulated considering the thermal effect in the heat exchanger and as well from the intake manifold structure under the design and operating conditions as shown in table 1.

| Contents     | Design temperature (°C) | Operating temperature (°C) | Design pressure (MPa) | Operating pressure (MPa) |
|--------------|-------------------------|----------------------------|-----------------------|-------------------------|
| Tube side   | Hydrogen gas            | 470/300                    | 396.6                 | 2.45                    | 1.98                   |
| Shell side  | Boiling water           | 275                        | 223.88                | 3.92                    | 3.11                   |

Table 1. Design and operating parameters of waste heat boiler.
Figure 1. A typical convective type of waste heat boiler (WHB) used as part of reformer system.

Figure 2. (a) Damaged heat exchanger containing cracks on a circumferential weld, (b) WHB and Fixture covering the flange and welded with shell was used to prevent from crack extension.

2. Modeling and analysis
Based on the structure of WHB, the high temperature gas inlet is defined as tube side and the heat transfer section as shell side. Considering the symmetrical structure of WHB and tube arrangement, a half model to cover half of both tube and shell sides and as well half cross section of tube sheet is used as shown in figure 3. The outer surface temperature of WHB and piping are assumed constant since the whole heat exchanger and piping are fully insulated during operation. In this study, steady state heat transfer in isotropic material and Newton cooling was assumed. The governing equation [3] of thermos-elasticity of an isotropic solid is shown as in equation (1). By ANSYS, temperature distribution of the total model was first calculated. Furthermore, stresses of the structure were then analyzed considering operating pressure and temperature as internal force.

\[
(\lambda + \mu) u_{ij, ij} + \mu u_{i, ii} - \frac{E\alpha}{1-2\nu} T_i = 0
\]  

(1)

Where, \( \lambda = \frac{vE}{(1+v)(1-2v)} \) and \( E, \mu, v, \alpha, u, T \), the Young’s modulus, shear modulus, Poisson’s ratio, thermal expansion coefficient, displacement and temperature.
Figure 3. 3D brick element for numerical analysis with circumferential fillet weld in orange color.

To realize the failure mode, three different stresses of principle ($S_i$), equivalent ($S_{eqv}$) and stress intensity ($S_{int}$) are compared as indicated by equations (2), (4) and (5) respectively.

$$S_i = \text{Max}(\eta_i)$$

(2)

Where, $\eta_i$ ($i = 1, 2, 3$) is the root of equation (3) and the principle stresses of three principle axis.

$$\det(\sigma - \eta \delta) = 0$$

(3)

$$S_{eqv} = \left[ \sigma_{\pi}^2 - \frac{3}{2} (\sigma_{\pi} - \sigma_{\sigma}^2) \right]^{1/2}$$

(4)

$$S_{int} = \text{Max}[(S_1 - S_2), (S_2 - S_3), (S_3 - S_1)] = 2\tau_{\text{max}}$$

(5)

For numerical analysis, 3D brick element of 20 nodes and a total number of 513,838 elements are used to cover the shell and tubes as shown in figure 3. Material parameters and boundary conditions are listed in tables 2, 3 and figure 4. The outer surfaces of model are insulated boundaries as indicated earlier and the surface temperature ($T_{in}$) of internal gas medium sides is set to 669.6 ($K$). And, the internal surface temperature ($T_{out}$) of shell sides which are in contacted with water steam is set to 506.88 ($K$). Thermal transfer with or without convection are both considered with convection coefficients of gas side and steam side assumed as ($h_{in}$) = 20 ($W/m^2K$) and ($h_{out}$) = 30 ($W/m^2K$) respectively [4]. After the temperature field was acquired, the stress fields are computed based on the conditions as shown in figure 5. The internal pressure ($P_{in}$) of gas side is set to 1.98 (MPa) and ($P_{out}$) of steam side to 3.17 (MPa) which is the saturated steam pressure at 506.88 ($K$). In addition, a bending moment was added considering the thermal expansion difference between high temperature manifold and the heat exchanger as shown in figure 6.

| Table 2. Material parameters. |
|-------------------------------|
| $k$ ($W/mK$) | $E$ (GPa) | $\nu$ | $\alpha$ ($10^{-5}/K$) | $\rho$ (kg/m$^3$) |
| --- | --- | --- | --- | --- |
| Tube side and shell side | 40 | 200 | 0.28 | 1.3 | 7860 |
| Weld | 32 | 160 | 0.3 | 1.0 | 7860 |

| Table 3. Boundary conditions. |
|-------------------------------|
| $h$ ($W/m^2K$) | $T$ ($K$) | $P$ (MPa) |
| --- | --- | --- |
| Inner surface of tube side and heat exchanger tubes | 20 | 669.6 | 1.98 |
| Inner surface of shell side, Outer surface of heat exchanger tubes within the shell side | 30 | 506.88 | 3.11 |
Figure 4. Boundary conditions of thermal analysis of heat exchanger.

Figure 5. Boundary conditions of stress analysis of heat exchanger.

Figure 6. (a) Bending moment assumed to act on heat exchanger due to thermal expansion of gas intake manifold, (b) Temperature (℃) distribution and deformation of heat exchanger and manifold.

3. Results and discussions
The temperature distribution of heat exchanger based on the assumption with thermal convection
effect and (h_in) of gas and steam equal to 20 (W/m²K) and 30 (W/m²K) respectively as shown in figure 7. The highest temperature of 382°C (655 K) is found at the inlet of tube side and the lowest temperature of 224°C (497 K) at the outlet of shell side. As a whole, the temperature distribution at the upper part of heat exchanger is higher than that at the lower part and, such a difference is well related to the tube arrangement of heat exchanger which upper tubes are closer to the shell as in figure 8.

![Temperature distribution of heat exchanger](image1.png)

**Figure 7.** (a) Temperature distribution of heat exchanger and flange portion (b) with convection and (c) without convection effect.

![Temperature distribution at the junction of weld and shell side](image2.png)

**Figure 8.** Temperature distribution at the junction of weld and shell side shows (a) The temperature increases with from the lower region (θ = -90°) to upper region (θ = 90°) of heat exchanger and is well related to, (b) the tube arrangement of heat exchanger which upper tubes are closer to the shell.

While the thermal effects of internal convection and external expansion moment by gas intake manifold are both considered, the results of maximum principal stress $S_1$ show that most area is in
tension except the corner of tube sheet on tube side and internal tubes on shell side as in figure 9. Wherein, the tension stress is the highest at the weld region connecting flange and shell. In this case along the circumferential weld of concerned, all three stresses increase gradually with orientation and reach the highest at $\theta = 90^\circ$ which located at the upper point and outer surface of the shell as in figure 10. In contrast, the stress on the inner surface of weld varied in oscillation along the circumferential direction. Definitely, the stress oscillation is closely related with the temperature variation on the inner surface of weld.

![Figure 9](image9.png)

**Figure 9.** Distribution of maximum principle stress $S_t$ of heat exchanger in which most area is in tension.

![Figure 10](image10.png)

**Figure 10.** Stress distribution at the junction of weld and shell at (a) outer surface, (b) inner surface. ($\theta = -90^\circ$ and $\theta = 90^\circ$ are the bottom and top of heat exchanger respectively).

Further referring to the crack configuration appeared on outer surface of weld, it is believed that circumferential crack preceded those two short branch cracks based on the failure principle of crack branching as described in figure 11. Accordingly, it is assumed that circumferential crack initiated from inner surface mainly due to relative high thermal stress. Axial branch cracks followed as circumferential crack propagated further in depth. To verify the crack initiation, distributions of
temperature and stress at five different depths (0t, 1/4t, 1/2t, 3/4t and 1t) of weld was investigated as shown in figure 12. The results show that all stresses at inner surface are of the highest in which \( S_1 \) is of most concern in terms of crack orientations. Actually, the stress due to thermal effect in this case is much greater than that by internal pressure solely.

![Figure 11. Failure principle of crack branching.](image)

**Figure 11.** Failure principle of crack branching.

![Figure 12.](image)

**Figure 12.** (a) Temperature distribution of weld (a) across the thickness, (b) along the weld in circumferential direction, (c) \( S_1 \) stresses at different depth, (d) \( S_{eqv} \) stresses at different depth.
4. Conclusions
Repeated cracking at the weld concerned is confirmed by the analysis of temperature, stress and failure principle of crack branching. It is found that internal tubes close to the inner surface at upper part of shell results in higher temperature and stress oscillation. Further study of the temperature and stress at different depths confirm that stress of weld is apparently of the highest at the inner surface which disclosed the evidence of crack initiation from circumferential direction and crack branching to axial ones. Based on these results, it is found that the existing fixture is not proper to avoid the propagation of circumferential crack which could be the major concern of heat exchanger rupture. Hence, proposal was given to shut down WHB immediately and to review the design of internal tubes and the external arrangement of high temperature gas intake manifold before replacing with a new one.

References
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