Transient natural convection and conduction heat transfers on hot box of a coke drum in Pre-heating stage

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Abstract. In an oil refinery unit, coke drum is subjected cyclic thermal stress and mechanical loads due to cyclic heating and cooling loads. Thus, the useful life of a coke drum is much shorter than other equipment. One of the most severe locations due to thermal stress is shell to skirt junction. Here, a hot box is proposed. In this study effectiveness of a hot box will be analyzed numerically. The addition of hot box (triangular cavity) was expected to generate natural convection, which will enhance heat transfer. As for the result show that heat flux conduction and natural convection have the same trend. The peak of conduction heat flux is 122 W/m\textsuperscript{2} and for natural convection is 12 W/m\textsuperscript{2}. In the heating stage of coke drum cycle it found that the natural convection only provide approximately 10 \% of heat transfer compare to conduction heat transfer. In this study it was proved that in the heating stage, the addition of triangular enclosure is less effective to enhance the heat transfer than previously thought.

1. Introduction

The aforementioned studies reveal that the function of a hot box is debateable. Thus, further analysis is extremely needed. In this study, the transient numerical analysis of convection and conduction heat transfer in a hot box of a coke drum is carried out. Since the operation time of a coke drum is very long, only the preheating stage will be analysed.

As for the data provided by S.H.I Examination and Inspection, four identical coke drums is taken into consideration. Which can we said as coke drum A, B, C, and D. the shell and skirt of coke drums made by low alloy steel SA-387 Gr.11 CL.2. The cladding inside the coke drum is made of 3 mm thick SA 240TP. 410S material. Thickness of shells vary within the range 28.5 mm – 34 mm and the skirt thickness are 24 mm. Ratio between diameter and wall thickness of the coke drum varied from 125 to 150 mm/mm. For the coke drums aforementioned above, dimension, and thermocouple positions is shown in Fig.1. The measurements were carried out in the point which shown in the Fig.1, where the thermocouple and strain gauges is located. Temperatures and strains were taken continuously every minute for around 6 months. The cycle recorded for each coke drum A, B, C, D were 52, 53, 53, and 54 cycles respectively.

The cyclic duration of this four coke drums is around 48 hours which are preheating stage from 0 to 460 min, filling stage from 460 to 2000 min, cooling stage from 2000 to 2400 min and cutting from 2400 min to 2800 min. For preheating stage, it divided again as preheating and switching stage,
which is for coke drums consider in this paper, the switching time is around 80 minutes, start from approximately 380 min.

![Fig. 1 Dimensions of the considered coke drums and measured location.](image1)

In preheating stage of coke drum, it is heated to around 350°C, and increasing rapidly in switching temperature to desirable injection temperature around 440°C to 460°C. Because coke drum is heated from the inside, the temperature in shell part of the coke drum increase more rapid than in skirt part, thus the temperatures at points T1-T4 is larger than T5-T8. The typical stages of the coke drum shown in Fig.2 below.

![Fig. 2 Typical operational temperatures at shell-to-skirt junction area of coke drum.](image2)

Nowadays natural convection in enclosure is a topic of contemporary importance, because enclosure filled with fluid are central components in long list of engineering and geophysical system. Natural convection in an enclosure is the result of the complex interaction between a finite sized fluid system in thermal communication with all the walls that confine it. The complexity occurs for the diversity of flow that exists inside enclosures [1]. Some studies had been done in the field of natural convection inside triangular enclosure, whether with experimental method or in computational fluid dynamics to further understanding in the flow and heat transfer inside triangular enclosure.
In coke drum, the shell-to-skirt junction tends to be the most severe part [2] because of the thermal stress due to temperature difference. Hot box is a triangular cavity in the shell-to-skirt portion, which be added in hope to generate natural convection to enhance heat transfer inside shell-to-skirt junction. Thus, the main objective of this present study is to observe the transient natural convection and conduction heat transfers on hot box in pre-heating stage, and compare these two heat transfer, therefore we can concluded whether the hot box is effective or not.

2. Method
Some of the cycle in coke drum was observed, and we found that coke drum C have the most exemplary cycles compared with cycles from coke drum A, B, and D, thus we choose coke drum C cycle as the representative to be analyzed with numerical analyses, to see the effectiveness of hot box in the shell-to-skirt junction part of coke drum.

The purpose of this study was to analyses thermal distribution in the model geometry at different times during cycle. In this present study only preheating and heating stages will observed. For the model that we analyzed with numerical analyses we propose shell-to-skirt junction of coke drum model which shown in Fig.3. The numerical FEM were performed by ANSYS Fluent, using a two-dimensional axisymmetric model. The model was developed based on fabricated drawing of the coke drum.

For numerical analyses, temperature boundary condition was treated as follow; the temperature of inner shell part of coke drum is modeled by UDF, imitate measured temperature from the real cycle of coke drum C cycle number 3, which start from minute 1 at 62°C and end in minute 597 at 422°C. According to Antalffy et.al [3], even though the feed is around 500°C the maximum drum skin temperatures that recorded during operation rarely exceed 440°C. Apparently, an insulating skin of coke deposits onto the cooler drum shell and insulates the drum shell from the 500°C process fluid temperature within the drum.

Because the effect of insulating layer not taken into consideration in this paper, thus for the modeled UDF, for the temperature in inner shell we only set it 20°C higher than the measured temperature from real cycle; 82°C (min 1) and 442°C (min 597) respectively. As for surrounding area we used stream temperature of 30°C, and in the under skirt part it was applied stagnant air of 90°C to enhance heat transfer in the lower part of triangular enclosure due to convection heat transfer.

![Fig. 3 Model for numerical analyses](image)

To ensure the accuracy of the numerical result, some nodes were generated. There are three nodes numbers were proposed in the numerical analyses, which are 1263, 4657, and 10321 respectively. Important parameters such as temperatures in point T9, T10, T11, T12, and T4 were compared for each nodes numbers.
All nodes number show a small difference of temperature plot in selected point of T9, T10, T11, T12, and T4. However, to make a better compatibility with measured result, the model with 10321 nodes was carried for all analyses. The coke drum material density is 7850 kg/m³ and 40.6 W/mK for thermal conductivity. For the insulator we used rockwool with thermal conductivity of 0.04 W/mK. In this analyses the wind, piping, and seismic overturning moments are excluded.

3. Result and Discussion

Fig. 4 shows temperatures plots in some points in this considered model. The numerical analyses results and thermocouple measurement results shows similar trend even though there occurred some differences from numerical and measured temperatures plot. Mostly, problem happened in peak temperature of numerical model and measured model. In numerical model the peak temperature (in T4) reached 440°C, but in the measured temperatures the peak temperature is below the numerical result. These phenomena occurred due to the insulated skin deposit in shell wall, which not taken into consideration in this paper. Thus, this problem also occurred in all temperature points. Set aside these differences between numerical and measured one, the results is still acceptable and further analyses will be taken to affirm it.

![Temperature Plot for Measured and Numerical Result in Pre-Heating Stage](image.png)

**Fig.4** Temperature Plot For Measured and Numerical Result

In the beginning of the cycle because of the stagnant air under skirt, the triangular enclosure was heated from below. Fig. 5 shows the temperature (a, c, e) and velocity (b, d, f) profiles in triangular cavity. From these figures it shown that the contour of temperature and velocity was developed first in lower part of enclosure, thus the purpose of stagnant air is to enhance the heat transfer in lower part of triangular enclosure. From fig. 5 (d) and (f), contour of velocity shows low velocity rate in the lower part of triangular enclosure, thus it concluded that stagnant air become less effective as the cycle goes by. Fig. 5 (c) and (e) shows that temperature distribution in skirt part cannot keep up the increment rate of temperature in shell part, even though the hot box (triangular enclosure) already attached. For initial conclusion it can be said that the triangular enclosure is not very effective to enhance heat transfer in shell-to-skirt part. To ensure this statement, further analyses has been done in this paper.
Temperature differences in 4 points in triangular enclosure (T9 and T10) and (T11 and T12) both for numerical results and measured thermocouples were compared. T9 and T10 have the same elevation (upper part of triangular enclosure) in Y-coordinate where T9 is located in shell part, and T10 is located in skirt part and so the temperature difference of these points will be measured, thus for T11 and T12 which is located in lower part of triangular enclosure respectively. Because the main purpose of the addition of the hot box (triangular enclosure) is to enhance the heat transfer, this four points was often be used to measure the effectiveness of hot box.
Fig 6. Temperature Differences for Numerical and Measured Result

Fig 6, the result of temperature differences show a good agreement both for numerical analyses and measured result. The temperature difference have the same trend both for numerical and measured one, even though there are some slightly difference because some parameters which occurred in real cycle not taken into consideration in the numerical analyses.

The peak of temperature difference (T11 and T12) in this cycle occurred not in the switching temperature, but in the beginning of pre-heating stage which was appeared in minutes 147. From the numerical (87°C) and measured thermocouples (85°C) we found the slightly difference of 2°C, which is acceptable. The problem occurred in switching stage, that the difference of temperature peak is 11°C, which exists in minutes 544.

Because of the peak of temperature difference (T11 and T12) in the numerical results are lower than the measured one, for heat flux from numerical result we will multiply by scale-up factor, because heat flux is the function of temperature. The scale-up factor is defined as comparison of maximum temperature difference from measurement and numerical one. For cycle C3 the maximum temperature difference in switching stage from measurement and numerical (shown in fig 6) are 78°C and 67°C respectively. Thus the scale up factor will be 1.17. Thus the calculated heat flux both for conduction and natural convection will be calibrated multiplying by 1.17 (fig 7).

From fig. 7 for heat flux result of transient conduction and natural convection in triangular cavity we can see that both have the same trend. In the beginning stage of preheating, heat flux conduction gradually increased up to 70 W/m² and 5 W/m² for heat flux in triangular enclosure. This high rate heat flux increment happened due to the hot steam which started to enter in the beginning of pre-heating stage. Heat flux tends to rise until hot steam spread evenly inside of the coke drum. Started from minute 147 heat flux fell to around 53W/m² and became stagnant until the beginning of switching stage. This statement was validated due to the temperature plot from fig 4 which shown significant temperature increasing in the beginning of preheating stage and decreases toward the end of preheating stage.

In switching stage, because of the rapid temperature increasing in a short time, the heat flux conduction increasing rapidly up to 122 W/m², and for heat flux in triangular cavity was increased up to 12 W/m². Heat flux increasing rapidly from the beginning of switching stage and decreases toward the end of switching stage, the same reason as the preheating stage that in the temperature rising
significantly in the beginning of switching stage and the end of the switching stage temperature gradient becomes low.

![Comparison of Heat Flux Conduction and Natural Convection in Pre-heating and Switching Stages](image)

**Fig. 7 Heat Flux Conduction and Natural Convection**

The result shown, hot box merely have a little effect to enhance heat transfer in shell-to-skirt junction. The natural convection inside the triangular cavity of shell-to-skirt junction approximately only took apart for 10% of the heat transfer. This result has a good agreement with the previous research for the same coke drum. According to M. Oka et al. [4], a significant temperature difference exists between T9 and T10; 120°C (preheating stage) and 150°C for T11 and T12 (preheating stage) respectively. These temperatures indicate very significant differences, thus it was concluded that hot box (triangular enclosure) is less effective to enhance heat transfer in shell-to-skirt junction part unlike previously thought. Therefore the new method to enhance the heat transfer in the shell-to-skirt junction should be considered in the future research, to reduce the thermal stress and increase the life span of coke drum.

4. **Conclusion**

From this research it can be concluded that the numerical analysis can be used as an affordable method to predicted transient heat transfer in shell-to-skirt junction. Although some slightly difference occurred between measured and numerical result is undeniable due to some boundary conditions which occurred in measured system such as complex insulating layer in inner shell which was not taken into consideration in this paper, but it still acceptable with using scale-up factor which offered.

The addition of triangular cavity (hot box) to generated natural convection in shell-to-skirt junction merely took a part to enhance the heat transfer unlike previously thought. The natural convection inside the triangular cavity of shell-to-skirt junction approximately only took apart for 10% of the heat transfer. This result has a good agreement with the previous research about the same coke drum.

For the development of the coke drum fatigue life can be done further research in any ways that can increase shelf life, especially in the shell-to-skirt junction portion, which is the most severe part of coke drum, for example the provision of oil or nano-fluid instead of air to enhance the heat transfer (natural convection) inside of the triangular cavity of the shell-to-skirt-junction.
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