Failure analysis of blots for diesel engine intercooler

Ping Ren 1, Zongquan Li 1, Jiangfei Wu 1, Yibin Guo 1, Wanyou Li 1
1 College of Power and Energy Engineering, Harbin Engineering University, Harbin, Heilongjiang Province, China
E-mail: guoyibin@hrbeu.edu.cn

Abstract. In diesel generating sets, it will lead to the abominable working condition if the fault couldn’t be recovered when the bolt of intercooler cracks. This paper aims at the fault of the blots of diesel generator intercooler and completes the analysis of the static strength and fatigue strength. Static intensity is checked considering blot preload and thermal stress. In order to obtain the thermal stress of the blot, thermodynamic of intercooler is calculated according to the measured temperature. Based on the measured vibration response and the finite element model, using dynamic load identification technique, equivalent excitation force of unit was solved. In order to obtain the force of bolt, the excitation force is loaded into the finite element model. By considering the thermal stress and preload as the average stress while the mechanical stress as the wave stress, fatigue strength analysis has been accomplished. Procedure of diagnosis is proposed in this paper. Finally, according to the result of intensity verification the fatigue failure is validation. Thereby, further studies are necessary to verification the result of the intensity analysis and put forward some improvement suggestion.

1. Introduction
Turbocharged diesel engine is characterized of high ventilation efficiency and strong power. Before sending the high temperature air into cylinders of diesel engine, inter-cooler is required to cool the hot air, which is heated by supercharging. If inter-cooler fails to work, the compressed high temperature air will go directly into the cylinders, which resulting in knocking or even damages to the engine.

This article takes the inter-cooler of certain diesel engine as research object, which is composed of encloser, tank body and inter-cooler bed. Encloser and tank body are connected by 48 pieces of 10.9 grade M12 high strength bolts while tank body and inter-cooler bed are connected by 46 pieces of 10.9 grade M12 high strength bolts. In respect that high temperature air is cooled by inter-cooler, temperature of inter-cooler is very high. The highest temperature of the inter-cooler which is taken as research object can be 150 degrees Celsius, hence, bolts on the inter-cooler not only bear the mechanical load but are also under large thermal load, and the working condition of blots are very bad.

Faulty parts of diesel engine are bolts on inter-cooler, after running for less than 24 hours, fracture occurred to the bolts. After replacing the faulted bolts for many times, failure still remains unsolved. Analysing the microstructure and fracture morphology by electron microscope determined that fracture was caused by fatigue, but the cause of fatigue fracture is unclear. Therefore, this article carried out research on bolts failure of inter-cooler and find out the causes of fatigue fracture.

2. Analytic train of thought
Static analysis and fatigue analysis of bolts on inter-cooler were based on accurate thermal analysis and fluctuating load calculation. For fluctuating load calculation, the force source and load path should be
determined. Reliable transfer function can be gotten through by precise modelling of main structure and appropriate simplification of complex boundary. However, the vibration excitation source of diesel engine is relatively complex and should be equivalent simplified, fluctuating stress calculation flow chart are as shown in figure 1.

![Flow chart of blots fluctuating load analysis](image)

Figure 1. Flow chart of blots fluctuating load analysis

3. Forced vibration response calculation of the inter-cooler

3.1 modal analysis of the inter-cooler

The hypothesis of diesel engine vibrostand was introduced here. Due to the complexity of diesel engine structure, in respect that diesel engine is not taken as main research object, it was equivalently simplified and taken as vibrostand.

![Diesel engine FE model](image)

Figure 2. Diesel engine FE model

The diesel engine is equivalent to a cuboid model, the mass, centre-of-gravity position and position of vibration isolator are consistent with the actual diesel engine. The accuracy of modal calculation results is directly affected by the accuracy of finite element model. Finite element model should reflect the actual status of equipment as far as possible [1], thus establishment of finite element model is correspondingly corrected. The finite element model is continuously revised by comparing results of test modal and simulation to making it close to reality [2]. Figure 2 is the finite element model of diesel engine.

Modal of inter-cooler is calculated by finite element software ANSYS. Comparing with test modal simulation result is revised. According to the differences between the results, stiffness of blots and
expansion joint is adjusted to make the calculating modal results consistent with the test modal results [3-4]. Due to the variable combinations, multiple correction maybe required to obtain the ideal effect. By means of model adjustment comparison between modal analysis and test is shown in table 1.

Table 1. Results comparison of modal analysis and modal test

| Modal Order | Modal Analysis (Hz) | Modal Test (Hz) | Error (Hz) | Percentage Error (%) |
|-------------|---------------------|-----------------|------------|----------------------|
| 1           | 76.1                | 77.7            | 1.6        | 2.1                  |
| 2           | 107.9               | 100.7           | 7.2        | 7.1                  |
| 3           | 116.0               | 116.5           | 0.5        | 0.4                  |
| 4           | 178.4               | 178.2           | 0.2        | 0.1                  |
| 5           | 195.4               | 205.4           | 10.0       | 4.9                  |
| 6           | 248.5               | 251.5           | 3.0        | 1.2                  |
| 7           | 262.2               | 265.3           | 3.1        | 1.2                  |
| 8           | 339.2               | 344.2           | 5.0        | 1.5                  |
| 9           | 349.2               | 349.4           | 0.2        | 0.1                  |

Seen from the comparison results between the natural frequencies that maximum frequency error is 11.5 Hz, maximum error percentage is 7.1%. The remaining errors are below 5%, meeting the engineering requirements.

3.2 Identification and verification of equivalent load

With the continuous development of dynamic load identification technology, it has been used in more fields. Application of frequency domain method in dynamic load identification field is long and relatively mature [5], so frequency domain method was applied in this article to identify the force source of the machine. Diesel engine vibration test result and finite element modal is taken as important parameter to identify and verify the excitation force.

The research object in this article is the bolts on inter-cooler, which are mainly affected by excitation force of diesel engine, thus, the force source point should be set within the simplified diesel engine.
vibrostand. In this paper, the combinations of different locations were repeatedly calculated. It found when there are three equivalent source points in diesel engine, the result is consistent with the actual situation. One force source point is in the centre-of-gravity position, the other two are at both sides of the centre-of-gravity position. The force sources location are as shown in figure 4(a), the three yellow dots are the location of equivalent force points.

![Figure 4. Location of equivalent force and vibration test](image)

3.3 Equivalent exciting force verification

The identified force source is taken as excitation force to solve the vibration response. Comparing with experimental data the error is analysed to determine whether the identified force source is accurate or not. The test point is as shown in figure 4(b), one of comparison between the actual measured responses and simulation results are shown in figure 5.

![Figure 5. Comparison of test result and response analysis](image)

3.4 forced vibration response analysis of the inter-cooler

The identified force source is load on the finite element model to analyse the bolts stress, bolts position is as shown in figure 6. Yellow dots are the bolts, totally 94 pieces.

Blots stress result of FRF-Based response analysis is shown in figure 7. Seen from the data that the maximum tensile stress of vast majority of bolts appears at 75 Hz. 75 Hz is consistent with 4.5 times shaft frequency under the rated conditions of diesel engine, at the same time, test modal results of encloser found an one-order model of inter-cooler in 77.7 Hz. Therefore, this article concluded that, under the rated conditions, 4.5 times shaft frequency excitation force aroused the resonance of inter-cooler and the resonance is the main cause of the bigger bolts dynamic force.
4. Thermodynamic analysis of the inter-cooler

High temperature compressed air can heat inter-cooler and makes its temperature higher, also, the temperature differences of each part are big, different levels of heat deformations occur to encloser, tank body and inter-cooler bed. Thus, connecting bolts on inter-cooler also suffer from different levels of heat stress. 3d model of Inter-cooler was established, initial temperature of all parts were set at 22 °C according to the environment temperature. Temperature distribution of inter-cooler took the form of decreasing from top to bottom while diesel engine was stable under the rated conditions. Thermodynamic analysis was carried out on the inter-cooler in accordance with the temperature distribution of diesel engine actually measured by factory staff. As shown in figure 8(a), the maximum temperature is 150 degrees Celsius, appeared at the connecting position between tank body and encloser, while the minimum temperature is 50 degrees Celsius in the inter-cooler bed position.
Based on the temperature distribution of inter-cooler, hot-structure coupling analysis was carried out by applying the boundary conditions of inter-cooler (namely, displacement constraints of inter-cooler bed and pre-tightening force of bolts). Displacement contour of inter-cooler are as shown in figure 8(b), seen from the displacement contour, thermal expansion amount is small where the temperature is lower, such as inter-cooler bed and lower body of inter-cooler. Thermal expansion amount is big where the temperature is high, such as encloser and upper body of inter-cooler.

5. Bolt strength verification of the inter-cooler

5.1 static force verification

In order to prevent the air leak, some certain pre-tightening force should be exerted on bolts of inter-cooler to meet the requirement of sealing conditions. According to the information provided by the manufacturer, the pre-tightening torque of bolts is \( T = 70 \text{ nm} \), calculate the pre-tightening force of bolts according to the pre-tightening force formula (1):

\[
T \approx 0.2 \times F_0 \times D
\]

\( F_0 \) is pre-tightening force of bolts (N), \( D \) are the bolts nominal diameter (m).

Therefore, pre-tightening force of M12 bolts is \( F_0 = 29167 \text{ N} \). The sum of pulling force that bolts receive in pre-tightening force and thermal stress is the actual tension. Due to the NBR (nitrile butadiene rubber) was applied in connection joint surface of inter-cooler, the relative stiffness coefficient of bolt is \( \lambda = 0.25 \) [6], and the working load of connection joint is \( F' \); then the total tensile stress can be obtained by formula (2):

\[
s = \frac{F_0 + \lambda F'}{A}
\]

Tangential working load that bolts receive is \( F_r \), then the shearing stress that bolts receive can be obtained by formula (3):

\[
\tau = \frac{4(F_r - \lambda F_0)}{3A}
\]

Thermodynamic analysis results shows that changes of bolts in axial direction is big while in horizontal direction is small. The horizontal displacement amount is less than spacing between the bolt and bolt hole, therefore, this article concludes that the tangential force that bolts receive is small and the bolts mainly bear tension.

Carry out static strength verification of bolts on inter-cooler, the specific verification formula is as shown in formula (4) [7]:

\[
\frac{s^2}{Ft^2} + \frac{\tau^2}{Fv^2} \leq 1
\]

Figure 9. Results of static force verification
According to the ferritic steel standard [6], allowable tensile stress is $F_t = 0.58S_t$, allowable shearing stress is $F_v = 0.24S_u$. $S_t$ is tensile strength of the bolts, considering all the above-mentioned parameter tensile strength of 10.9 grade M12 bolts is 1000MPa. Verification result is shown in figure 9.

Seen from static strength verification results that the bolts of inter-cooler can satisfy the requirement of static strength, under the premise of meeting the requirement of this article, therefore, fracture failure arising from static load will not occur to bolts on inter-cooler.

5.2 Fatigue verification

According to the S-N curve provided by the manufacturer, the bolts is verified through the fatigue safety coefficient method and verification formula of bolt fatigue strength is as shown in formula (5) [7]:

$$n = \frac{2\sigma_{s1} + (K_{n} + \psi_{n})\sigma_{min}}{K_{n} + \psi_{n})(2\sigma_{s1} + \sigma_{min})} \geq [n]$$

(5)

$n$ is calculating safety coefficient, $[n]$ is allowable safety coefficient, $[n]=1.21$

$\sigma_{s1}$ is symmetrical cyclic fatigue limit, according to the formula (6):

$$\sigma_{s1} = 0.23(\sigma_{y} + \sigma_{s})$$

(6)

$\sigma_{y}$ is the material yield strength limit, $\sigma_{s}$ is the material tensile strength limit, $\psi_{n}$ is the reduction factor of stress. On the basis of mechanical design manual [6-7], $\psi_{n}=0.25$, $K_{n}$ is fatigue limit comprehensive influence coefficient.

$$K_{n} = (\frac{k_{s}^{\beta_s}}{\varepsilon_{s}} - 1) \frac{1}{\beta_{s}}$$

(7)

$k_{s}$ is effective stress concentration factor, $k_{s}=5.2$; $\varepsilon_{s}$ is size factor, $\varepsilon_{s}=1$; $\beta_{s}$ is surface quality coefficient, $\beta_{s}=0.9$; $\beta_{s}$ is strengthening factor, $\beta_{s}=2$, $\sigma_{s}$ is stress amplitude, $\sigma_{min}$ is minimum stress. Fatigue strength verification results of bolts are shown in figure 10.

Figure 10. Results of fatigue verification

Figure 10 illustrates that fatigue ratio of bolts between tank body and inter-cooler bed is not enough for safety requirements. The possibility of fracture failure arising from fatigue is big.

6. Conclusion

This article carried out related researches on fracture failure of bolts on inter-cooler, the results show that the maximum tensile stress of the vast majority of bolts appears at 75 Hz, 75 Hz is consistent with 4.5 times shaft frequency under the rated conditions of diesel engine, at the same time, actual measured modal results of enclosure found an one-order modal of inter-cooler at 77.7 Hz.

Therefore, this article concluded that, under the rated conditions of diesel engine, 4.5 times shaft frequency excitation force aroused the resonance of inter-cooler and the resonance is the main cause of the bigger bolt dynamic force. Also, bolts strength verification found that static verification of bolts reach the standard, however, fatigue ratio of bolts between body and inter-cooler bed of inter-cooler is not enough for safety requirements, which is the main reason of fracture failure of bolts.
References

[1] Toshimichi Fukuoka and Masataka Nomura 2008 Proposition of Helical Thread Modeling With Accurate Geometry and Finite Element Analysis Journal of Pressure Vessel Technology 011204-1---011204-6.

[2] Desanghere G and Snoeys R 1985 Indirect Identification of Excitation Forces by Modal Coordinate Transformation Proceedings of the 3nd MAC p685-690

[3] Bartlett F.D and Flannelly W.D 1979 Modal verification of force determination for measuring vibration loads Journal of the American Helicopter society 19(4): pp10-18.

[4] Hansen M and Starkey J M 1990 On Predicting and Fin-proving the condition of Modal -based indirect Force Measurement Algorithms Proceedings of the 8th MAC pp115-120.

[5] Kreitinger T; Luo H L 1987 Force Identification from Structural Responses Proceedings of the SEM spring conference June 1987: pp851- 855.

[6] Liang-Gui Pu and Ming-Gan Ji 2004 Mechanical Design (Seventh Edition) (China: Higher Education Press) p67

[7] Bang-Chun Wen 2010 Machineries Handbook (Fifth Edition) (China Machine Press) pp5-31