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Design of exhaust manifold for pulse converters considering fatigue strength due to vibration

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Abstract: The design of the exhaust manifold for the pulse converters of a 4 strokes high power medium-speed diesel engine is presented in terms of fatigue analysis. The said system undergoes thermal expansion due to high temperature of exhaust gas and is exposed to intrinsic vibration of the internal combustion engine. Moreover, the exhaust pulse generates pressure pulsating along the runner inside manifold. Under such circumstances, the design and construction of exhaust manifold must be carried out in a way to prevent early failure due to fracture. To validate the design concept, a test rig was developed to simulate the combination of thermal and vibrational movements, simultaneously. Experimental results showed that a certain sense of reliability can be achieved by considering a field factor obtained from the results of engine bench tests.

Keywords: Exhaust manifold, Heat expansion joint, Resonance, Modal analysis, Fatigue analysis, Thermal load, Vibration load.

1. Introduction

Turbo-charged, medium-speed diesel engines are used for power generation, rail traction, and marine applications **[1]**. The system of pulse convertors, as depicted in **Figure1**, is a useful design technique to separate exhaust flows of cylinders which are liable to interfere with each other and to produce beneficial wave reflections to assist scavenging of the cylinders. The passage of blow-down pulse from the cylinder has to be designed in a way to minimize wave interference, while maximizing wave reflection for scavenging. This, in turn, results in highly complicated, geometric constraints which makes the system vulnerable to thermal expansion and harsh vibration. The design of exhaust manifold is then discussed to provide a compromise between geometric constraints and fatigue behaviour.

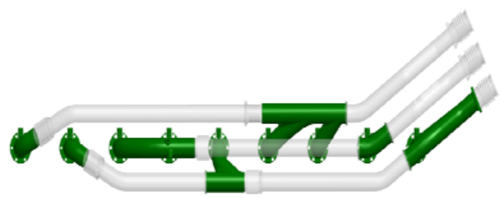

Figure 1: Example of exhaust manifold

1.1 Absorbing thermal expansion

Because of varying pressure and temperature fluctuations in the manifold of the engine, the pulse converters are subject to thermal expansion. The temperature measured at the outlet of a cylinder is varying in the range of 400 ~ 500 ℃. Considering 4 m long

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bank length, a bank is a exhaust gas runner combining a group of cylinders, the thermal expansion should be taken into account in the design of exhaust manifold. It is a common engineering practice to use a heat expansion joint to absorb thermal stress.

1.2 Vibration operating condition

The medium speed diesel engines normally operate in a speed range around 700 ~ 1,500 rev/min, and have power outputs up to 6,000 kW **[2]**. When used as a power generator, the range of 70 ~ 80 percent of full load is set to be a normal operating condition. Under this operating condition, the first natural frequency of the exhaust manifolds must be located over 100 Hz which might be considered as a threshold to signify early fracture caused by resonance. It is a rule of thumb to relocate the first natural frequency of the system at least over 100 Hz for such systems which are subject to thermal expansion and vibrational movements. However, it should be recognized that the use of separate bank to include a group of pulse converters might require some engineering compromises in terms of stiffness of the system. The complicated geometric structure open results in lower natural frequencies. Durability could be then seriously deteriorated as the operating load exceeds over the normal.

Thus, the heat expansion joint used is designed to absorb thermal expansion and not movements induced by vibration. Since the heat expansion joint is made of a thin stainless sheet metal (0.2 ~ 0.3 mm), the lower frequency resonance of the exhaust manifold could cause permanent damage to the heat expansion joints. This is because the thermal expansion compounded with pressure pulsation acts as mean stress to shorten the expected life cycle of the material. The operating load has to be regulated at a certain level not to exceed the design standard, for example, 80 percent of full load. Exceeding this might be critical in safety as shown in **Figure 2**. In other words, a properly designed exhaust manifold can help to increase the operating load over the limit.

Figure 2: Early failure due to fracture

2. Design Considerations

2.1 Configuring layout of the system

The configuration of the pulse converters must be made to minimize mounting tolerances. The best way to minimize the accumulation of mounting tolerances is not to use the welded pipes and converters. This is, especially, true for the design of exhaust manifolds for medium-speed diesel engines. The design volume of $4{,}000 \times 1{,}000 \times 500$ mm3 is too big to cover with weldings. Welding itself is not sensitive to tolerance, but layers of welding might cause the accumulation of tolerances. When designing a system which is subject to thermal expansion and vibration, welding free construction must be considered.

The converters are made by cast parts having general tolerance of 1/20 mm, and connected by heavy duty V-clamps. It allows mounting tolerance to the similar level of cast parts. The design target of the exhaust manifold is 8,000 hours of operation without significant failures. Comparing damage calculations of the cases of ± 5 mm and ± 2 mm offsets, the former results in 50 percent of failure rate (damage of 110 %) and the latter 0.05 % (damage of 11.8 %). Here, damages are calculated at the heat expansion joints by employing the Palmgren-Miner rule.

2.2 Heat expansion joint (HEJ)

Thermal expansion induced by the high temperature exhaust gas is designed to be absorbed by the expansion joint having axial stroke of 18 mm (peak to peak). The heat expansion joints are installed between the cylinders

Table 1: EJMA calculation result of HEJ

Design	Pressure	p = 5bar
	Temperature	T = 600℃
	Number of Cycles	N_r = 15000
	Axial movement	x_{max} = 0 mm
		x_{min} = 15 mm
	Angular rotation	Θ_{max} = 0°
		Θ_{min} = 0°
	Lateral deflection	y_{max} = 0 mm
		y_{min} = 15 mm
Deflection	Membrane stress	σ_5 = 7.2
Stress	Bending stress	σ_6 = 765.9
N/mm^2	Total reference stress	σ_t = 845.5
Fatigue		N_c = 16295
Life		

It means the safety factor for thermal expansion is considered to be more than 10.

Table 2: 18mm of axial displacement

Sample	No. of Load Cycles	Remarks
1	158000	No failure
2	155600	No failure
3	162500	Failure at
		first inner crest

in order to decouple the converters from thermal stresses. According to the EJMA (Expansion Joint Manufacturer's Association) result (see **Table 1**), the proposed HEJ could endure at least 15,000 cycles under the given condition. The proposed HEJ in fact can go more than 10 times of the calculated value (see **Table 2**).

2.3 Measurement to make system stiffer

A HEJ is not designed to absorb any vibrational movement under 100 Hz. The displacement under this frequency possesses enough energy to kill the HEJ used to absorb thermal expansion. However, the design volume of the complete exhaust manifold is too big to relocate the first natural frequency below 100 Hz. To make the complete system stiffer, a considerable thickness increase of the cast converters would be unavoidable. It costs more and makes the system heavy. To increase the natural frequency, the banks are reinforced by U-joints as shown in **Figure 3**. As explained in the next section, the use of U-joints is simple and effective to increase stiffness of the system.

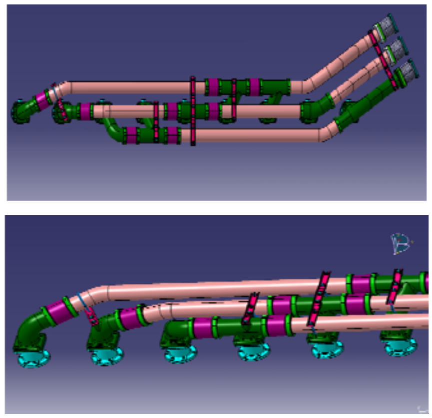

Figure 3: Reinforcement of banks

2.4 Experimental modal analysis

To measure the first natural frequency of the complete system under full load condition, a set of accelerometers are used. However, it should be noted that the ambient temperature of the exhaust manifold under full load exceeds 500 ℃. Under this condition, the measurement of acceleration is not possible unless otherwise employed specially devised sensors for such high temperature measurements. Even with those sensors, the reliable data cannot be obtained easily. To measure the vibrational behaviour of the complete system under full-load condition, water-cooled adaptors are devised to protect temperature sensitive sensors (**Figure 4**).

Accelerations were measured both at the inlet and outlet sides of each HEJ. To see the effect of reinforcements, two different cases, with and without

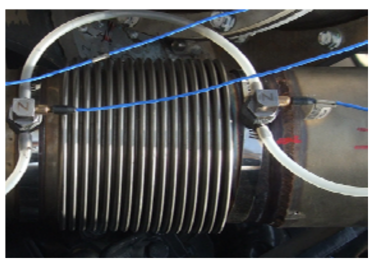

Figure 4: Water cooled accelerometer

them, were assessed in terms of their first natural frequencies. The exhaust manifold was completely bended like a bow under the full load condition. Without appropriate reinforcements, Case WO, the first natural frequency was observed at 60 Hz with the amplitude of ± 0.4mm (**Figure 5**).

With reinforcements, Case W, the first resonance was seen at 150 Hz with the amplitude of ± 0.02mm (**Figure 6**). Both were taken from the biggest signals of two different cases in the lateral direction. The magnitude of ± 0.4 mm is significant in terms of fatigue. Assuming 8,000 hours of continuous operation, the HEJ must endure $8000 \times 60 \times 60 \times 60$ cycles of axial movement. Simply, it is not working. Without appropriate reinforcements to relocate the first natural frequency over 100 Hz, the part cracks within 24 hours of continuous operation.

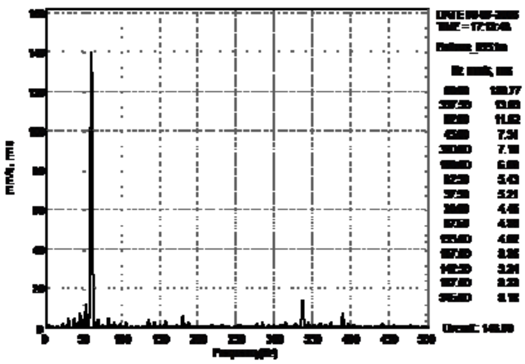

Figure 5: 1st Resonance (Case WO)

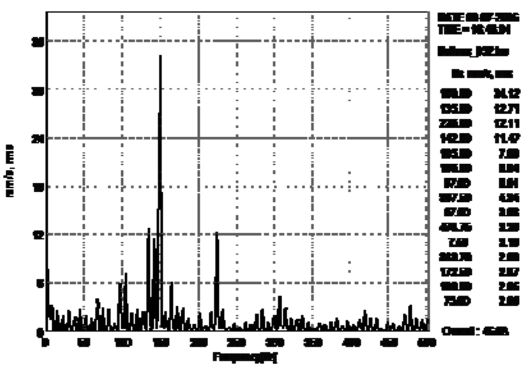

Figure 6: 1st Resonance (Case WO)

3. Fatigue Analysis

3.1 Superposition of loads

The stress in any point of the HEJ is the result from the combination of different loads are namely, 1) thermal expansion 2) internal pressure and 3) vibration. Depending on the form of load, different cycles are observed. For example, the thermal expansion can be classified as a lower cycle fatigue, while that of vibration as a higher cycle one.

For the higher cycle fatigue, the required number of load cycles must be chosen over 1 million (so called fatigue limit). For the lower cycle fatigue, the required number of load cycles is far less than 1 million. The normal safety factor of 12.5 (5: normal safety factor of $\times$ 2.5: temperature factor due to high temperature) is

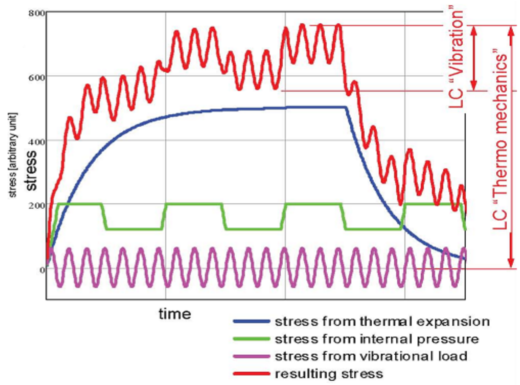

Figure 7: Different characteristics of loads

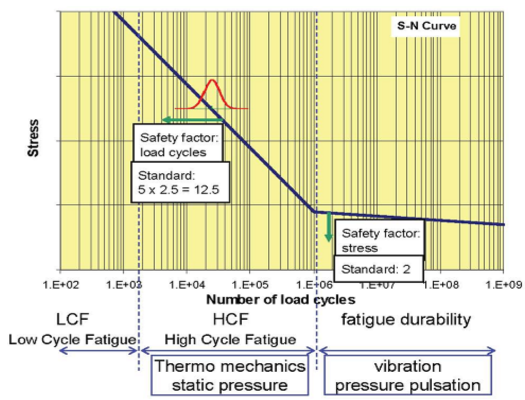

Figure 8: Stress evaluation

applied for the lower cycle load and the safety factor of 2 for the higher cycle one.

However, it should be recognized that the vibrational movements caused by the resonance which is lower than 100 Hz should be treated differently. The normal safety factor of 2 for the higher cycle fatigue is no longer valid due to the significance of its magnitude (see **Figure 8**).

3.2 Damage parameter

Unlike the usual case, the majority of fatigue comes from vibration instead of thermal expansion. Moreover, the large quasi-static load caused by the thermal expansion acts as mean stress to reduce number of cycles on the S-N curve. Then the modified Goodman line is obtained by

$$\frac{\sigma_a}{\sigma_{ar}}+\frac{\sigma_m}{\sigma_{fB}}=1 \qquad (1)$$

where σ_a denotes stress amplitude and σ_m mean stress, while σ_{fB} is obtained from the material characteristics somewhat higher than σ_u, ultimate strength. The stress amplitude under the mean stress, σ_{ar} is obtained by the Smith, Watson, and Topper (SWT) equation as shown in **Figure 9 [3]**.

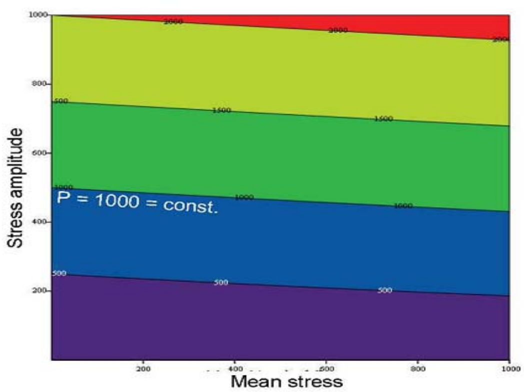

Figure 9: Amplitude-Mean Diagram

4. Experimental Validation

4.1 Thermal load

To find extra space and time for the experimental verification of the whole system is extremely difficult because of the 8,000 hours of operating condition. To run a diesel engine such a long time would be very costly. Considering that most of early failures occurred at the first 24 hours of engine-bench tests, the test rig is developed to validate the durability of the HEJ under the combination of thermal and vibrational loads. To simulate high temperature of exhaust gas, a set of LPG gas torch are used. The amount of air flow can then be controlled to provide gas temperature of 600 ℃ with the temperature variation of maximum 50 ℃ between the inlet and outlet sides of the HEJ. The temperature difference of 50 ℃ is chosen according to the result of actual temperature measurement

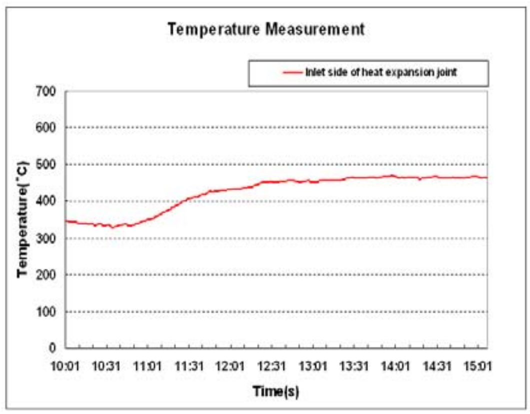

Figure 10: Temperature measurement of HEJ

4.2 Vibration load

The measurement result of the exhaust manifolds is obtained from the biggest signal among 3 directional movements. The field factor must be considered to assure a reliable, yet efficient simulation. This factor was obtained by trial and error. Considering the HEJ was cracked within the first 24 hours of full load condition without reinforcements, The test rig was used to see

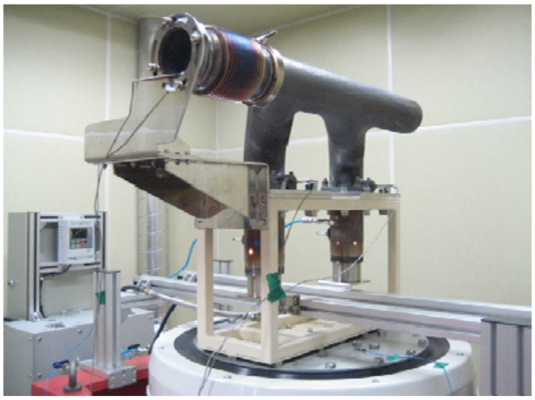

Figure 11: Test rig to simulate thermal and vibration movements

Figure 12: Field factor of 3.6

which amplitude cracks the part. It came to 22 G at 60 Hz. Comparing with the result of actual measurement, we got the field factor of 3.6. With this field factor, the HEJ was cracked within the first 24 hours on the test rig, as shown in **Figure 11**.

The field factor of 3.6 is used to simulate the vibraitonal load in case of the system with bank reinforcements. The HEJ was survived more than 100 hours with the vibration level of 64 G at 150 Hz on the test rig. 100 hours of operation might not guarantee durability of 8,000 hours operating condition, but it gives a sense of similarity to ensure durability of the system in many applications.

The test rig was devised to simulate the combination of two different fatigues, namely, thermal expansion and vibrational movements. Using a set of LPG gas torch, the exhaust gas temperature of 600 ℃ was achieved with the temperature variation of 50 ℃.

Instead of using the costly engine test bench, an electromagnetic shaker was used to generate different vibrational movements. The gas torches were installed right under the cylinder outlets of the pulse converter. In this way, we were able to simulate the converter with the HEJ under thermal expansion and vibrational movements. This is meaningful because the heat expansion joint is the most vulnerable to thermal and vibrational loads.

Field factor of 3.6 was used to ensure the reliable simulation. The value was experimentally chosen by taking into account the results of engine bench tests. With the result of the rig test, we are able to make sure that the durability of the complete exhaust manifold is partly assessed without costly engine bench tests. It is not a perfect validation in any sense, but engineering judgment could be made using the result of the rig test.

5. Conclusions

The design of the exhaust manifold for high power medium-speed diesel engine was presented. The combination of thermal expansion due to vibrational movements exhaust gas and high temperature caused by internal combustion were considered to prevent early failure. The characteristics of different loads were analyzed to allocate appropriate safety factors for damage calculation. It turned out that the normal safety factor of 2 for higher cycle load such as vibration would be dangerous if the first natural frequency of the exhaust manifold is less than 100 Hz.

6. Acknowledgments

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