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INVESTIGATION OF TORSIONAL VIBRATION CHARACTERISTICS OF MARINE DIESEL ENGINE CRANKSHAFT SYSTEM

Won-Hyun Kim[1] and Soo-Mok Lee[1]

[1]Hyundai Maritime Research Institute, Hyundai Heavy Industries Co., Ltd.
1 Cheonha-Dong, Dong-ku, Ulsan, Korea, 682-792
whkim@hhi.co.kr

Abstract

Evaluation of the torsional vibration characteristics of a crankshaft system is indispensable to the development of marine diesel engines. It influences the selection or decision about shafting components such as the flywheel, torsional damper, generator and even modification of the crankshaft itself. In this paper, experimental and analytical investigations of torsional vibration were carried out for a newly developed marine diesel engine. The measurements were performed on various system changes e.g. engine with and without torsional damper, tuning wheel and coupling of dynamometer or generator. Based on the measurement results, analytical models were tuned to achieve better reliability. Meanwhile the sensitivity of parameters–inertia, stiffness and damping on the free vibration and torsional response-were assessed quantitatively and qualitatively. Dynamic stresses in the highly stressed area of the crankshaft were also measured under actual operating conditions with telemetry equipment. The safety factor against fatigue was evaluated with the measured stresses and compared with the analyzed ones and showed good agreement with each other.

1. INTRODUCTION

The design of crankshaft is quite important process in developing internal combustion engine. The distance between journal bearings, diameter of bearing and size of counterweight directly affect engine dimension. It is also torsionally excited by cylinder pressure and inertia forces repeatedly, the sufficient structural safety should be insured against additional stress by the torsional vibration. In early stage of engine development, analytical investigations of torsional vibration characteristics are inevitable because there is no actual object. So many studies about the modelling and analysis had been done since early 1900's and later on to increase the accuracy of prediction.[1]

To obtain the reliable analysis model, the system parameters such as inertia, stiffness and damping should be identified firstly. However it is general procedure that parameters are predetermined from a simplified method or empirical method in analysis and design stage. And then the analysis model is tuned based on the actual measurement to reduce the error and to investigate the vibration characteristics.

In this paper, the torsional vibration of the crankshaft of newly developed engine was investigated through the analysis and measurement. The test engine is 4-stroke medium speed diesel engine with bore 170 mm and stroke 280 mm. It is eight-cylindered with normal output of 920 kW at the design speed 900 rpm. According to the analysis and experiments for several kinds of system conditions, the system parameters are reasonably selected and adjusted. Also their contributions to vibration response are examined through the sensitivity analysis.

2. MODEL AND ANALYSIS

2.1 Analysis model

The torsional vibration analyses were carried out for various composition of shafting system as summarized in Table 1. The reason of each system usage is to remove the uncertain effects on inertia and stiffness of sub-component or driven machineries. That means that the fairly pure effects of inertia, stiffness and damping of crankshaft can be evaluated step by step. The experiments are also performed in same conditions and will be deeply discussed in Chapter 3.

Table 1. Analysis conditions of engine shafting system

Driven machinery condition	Additional components condition
Engine single condition	• Without torsional damper and tuning wheel • Without torsional damper and with tuning wheel
Dynamometer condition	• Without torsional damper and tuning wheel • Without torsional damper and with tuning wheel
Generator condition	• With torsional damper

The inertia and stiffness of crankshaft is not easy to estimate because of their geometrical complexity. The inertia of throw can be calculated from 3-D CAD software with considerable accuracy and stiffness is given equivalent stiffness based on B.I.C.E.R.A [2] formula and also calculated from the FE analysis. The other components such as gear, flange, flywheel and intermediate shafts are easily converted to a torsional spring and inertia.

The damping of shafting system was modelled as damping between the inertia (internal damping or structural damping) and external damping in each cylinder. In this paper, the external damping was calculated based on the empirical formula of Equation (1)[3], and the internal damping is applied with dynamic magnifier M_S as Equation (2).

$$C_C = K_e A R^2 \tag{1}$$

where, K_e: empirical constant, A: area of cylinder, R: crank radius

$$M_S = \frac{K_S}{C_S \omega} \quad \text{or} \quad C_S = \frac{K_S}{M_S \omega} \tag{2}$$

The analysis model was constructed by lumped mass and spring model as shown in Figure 1. One node was assigned for every cylinder of engine. Other elements were also modelled on proper nodes. As shown in Figure, dynamometer and generator were modelled with 3 nodes. Effects of cooling water and lubrication oil pumps were added to node of gear. Damper(spring type) and tuning wheel can be connected or disconnected according to their condition.

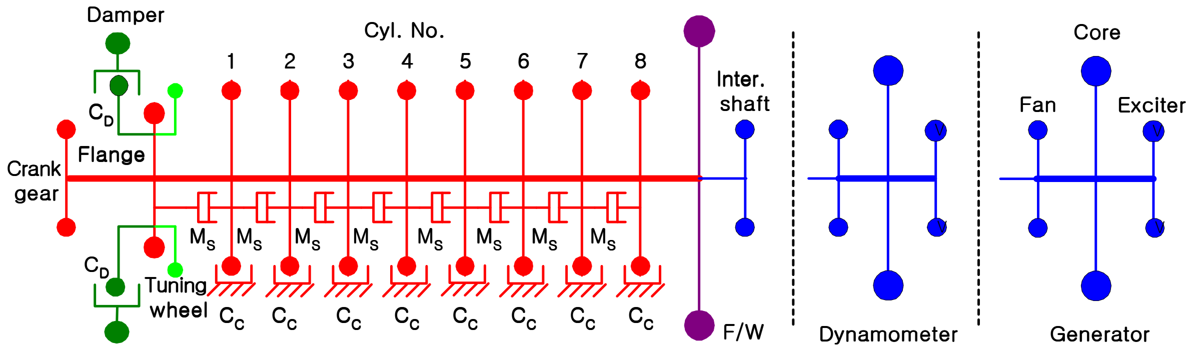


Figure 1. Models for torsional vibration analysis

The equation of motion of shafting system is expressed with well-known matrix notation as following.

$$[J]\{\ddot{\theta}\}+[C]\{\dot{\theta}\}+[K]\{\theta\}=\{T\} \tag{3}$$

From the constructed model, free vibration analysis is made with the proper numerical method and Equation (4). The natural frequencies and mode shapes of engine single model were shown in Figure 2 as reference.

$$\left[K-\omega_i^2 J\right]\{\Phi_i\}=\{0\} \tag{4}$$

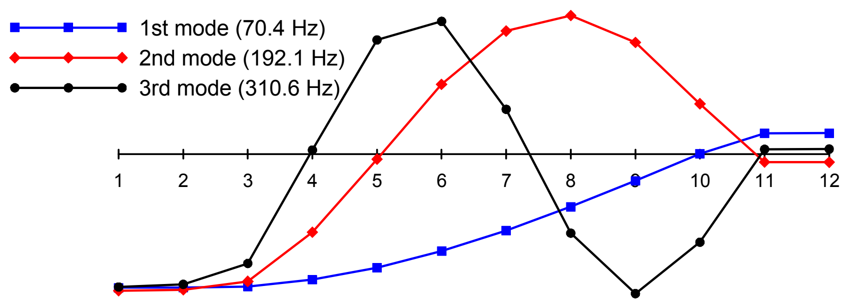


Figure 2. Natural frequencies and mode shapes of engine single model

Figure 3 and Figure 4 show the measured cylinder pressure (average of 8 cylinders) and torque variation on the single crank. In forced vibration analysis, the corresponding harmonics of torque variation are applied as excitation force.

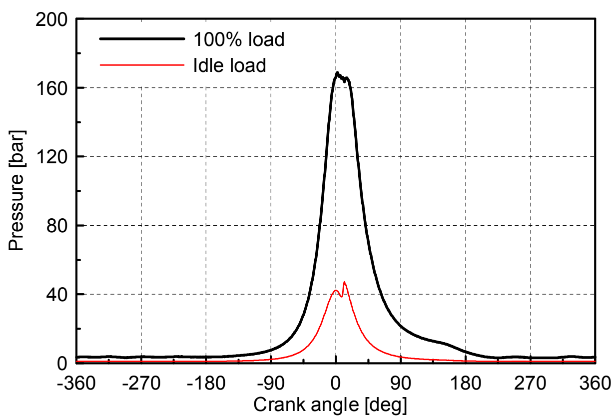


Figure 3. Cylinder pressure

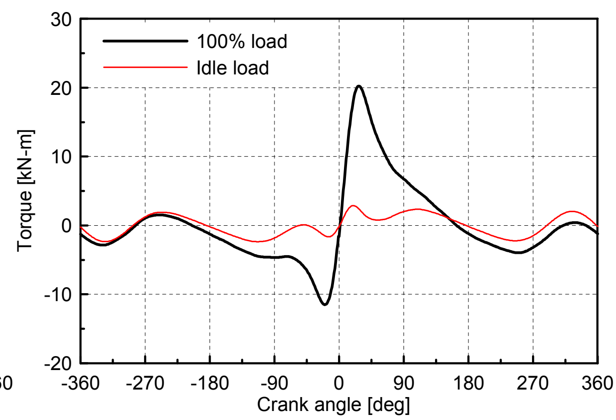


Figure 4. Torque variation

2.2 Sensitivity analysis

As mentioned above, because the stiffness and damping were determined by simplified or empirical method, the model can't help including a little inexactness. To reduce these deviations, inertia and stiffness should be tuned based on the results of measurement. At that time, adjusted parameters are reasonably selected with consideration of actual condition and effectiveness.

In this study, the sensitivity analysis is made through the calculation of the eigenvalue and eigenvector derivatives with respect to inertia and stiffness parameters[4]. This method enable the analyser to determine rapidly the most efficient modification of parameters needed to minimize the discrepancy between analysis and experiment results.

At first, premultiplication of Equation (4) by eigenvector Φ_i^T gives Equation (5). We can also get the relations of Equation (6) with the orthogonality of eigenvectors and real symmetric matrix K, J.

$$\Phi_i^T (K - \omega_i^2 J)\Phi_i = 0 \tag{5}$$

$$\Phi_i^T J\Phi_i = 1,\ \ \Phi_i^T K\Phi_i = \omega_i^2,\ \ \Phi_j^T J\Phi_i = 0\ \ (i \neq j),\ \ \Phi_j^T K\Phi_i = 0\ \ (i \neq j) \tag{6}$$

Differentiating Equation (5) with respect to the system parameters P_i results in Equation (7). Using the relations of Equation (6), desired expressions for the derivatives of the eigenvalue with respect to parameters are given as Equation (8).

$$\frac{\partial\{\Phi_i^T (K - \omega_i^2 J)\Phi_i\}}{\partial P_i} = \frac{\partial\{\Phi_i^T K\Phi_i - \omega_i^2 \Phi_i^T J\Phi_i\}}{\partial P_i} = 0 \tag{7}$$

$$\frac{\partial \omega_i}{\partial P_j} = \frac{1}{2\omega_i}\left\{\Phi_i^T \frac{\partial K}{\partial P_j}\Phi_i - \omega_i^2 \Phi_i^T \frac{\partial J}{\partial P_j}\Phi_i\right\} \tag{8}$$

From the Equation (8), eigenvalue derivatives with respect to inertia and stiffness are calculated with Equation (9) and (10) respectively.

$$\frac{\partial \omega_i}{\partial P_j} = \frac{-\omega_i}{2}\Phi_i^T \frac{\partial J}{\partial P_j}\Phi_i \ \ \text{or}\ \ \frac{-\omega_i}{2}\Phi_{ji}^2 \ \ \text{when } P_j \text{ are inertia} \tag{9}$$

$$\frac{\partial \omega_i}{\partial P_j} = \frac{1}{2\omega_i}\left(\Phi_i^T \frac{\partial K}{\partial P_j}\Phi_i\right) \text{when } P_j \text{ are stiffness} \tag{10}$$

Table 2 shows the results of sensitivity analysis for each shafting system. The 1st eigenvalue (fundamental natural frequency) derivative is only concerned because the torsional vibration responses are dominated by fundamental natural frequency in this system.

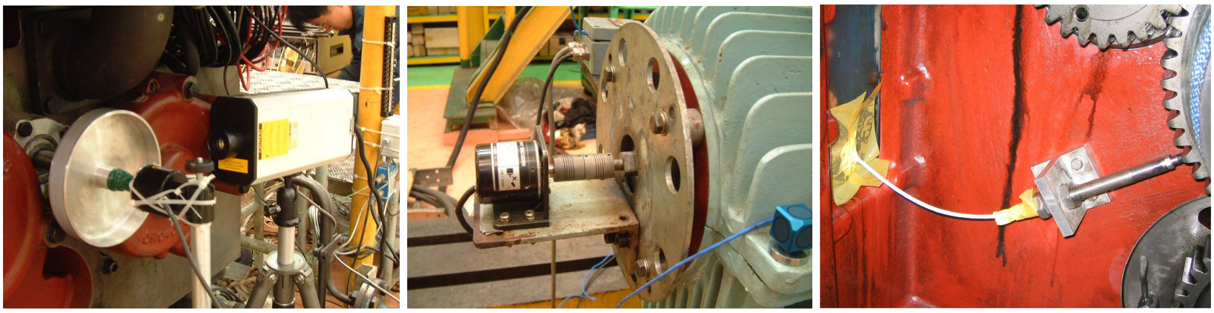
In engine single model, node 1 and 2 is gear and flange respectively. These inertias are most effective to rate of change of natural frequency. Crankshaft stiffnesses of all cylinders are also effective to that. The results of dynamometer model are almost same with that of engine single mode. Node 1 of generator model is torsional vibration damper and its inertia and stiffness have a critical influence on free vibration. The dynamometer shaft and generator rotor have minor effects on the natural frequency change. From these results, we can select the reasonable and effective parameters during the model tuning with the experiment data.

Table 2. Sensitivity analysis results (Eigenvalue derivatives w.r.t. parameters)

Inertia & stiffness nodes	Engine single model		Dynamometer model		Generator model	
	Inertia	Stiffness	Inertia	Stiffness	Inertia	Stiffness
1	-1.101	0.000	-1.120	0.000	-1.656	8.903
2	-1.097	0.007	-1.118	0.004	-0.175	0.000
3	-1.073	0.164	-1.100	0.147	-0.175	0.001
4	-0.960	0.476	-0.991	0.455	-0.168	0.012
5	-0.782	0.899	-0.813	0.881	-0.143	0.025
6	-0.567	1.363	-0.594	1.356	-0.111	0.032
7	-0.350	1.796	-0.371	1.804	-0.079	0.038
8	-0.165	2.127	-0.178	2.151	-0.050	0.044
9	-0.042	2.304	-0.047	2.342	-0.027	0.048
10	0.000	1.333	0.000	1.360	-0.010	0.051
11	-0.027	0.000	-0.025	0.000	-0.001	0.052
12	-0.028	-	-0.025	0.000	-0.001	0.029
13	-	-	-0.026	0.069	-0.007	0.000
14	-	-	-0.039	0.000	-0.007	0.001
15	-	-	-0.039	-	-0.009	0.000
16	-	-	-	-	-0.009	-

3. MEASUREMENT AND MODEL TUNING

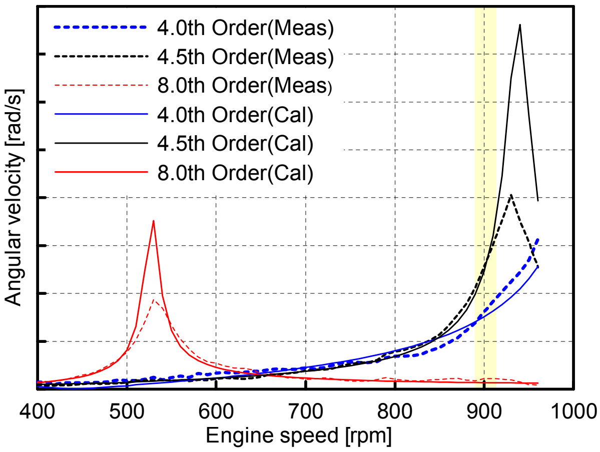
The torsional vibration measurements were carried out at each condition of Table 1. The several measurement methods which laser vibrometer, encoder and proximity probe were applied as shown in Figure 5. Torsional vibration was measured at the front end of engine crankshaft, dynamometer shaft or rotor of generator.

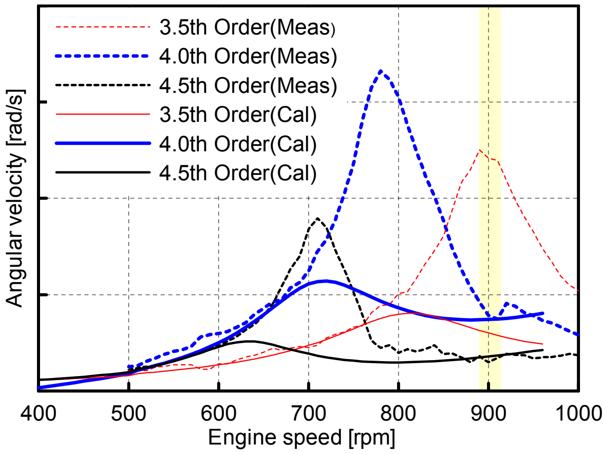


(a) Laser vibrometer (b) Encoder (c) Proximity probe

Figure 5. Measurement of torsional vibration

In each speed of operation, the torsional vibration responses are measured and analysed to order components from 0.5th order up to significantly higher orders usually 12th order. Figure 6 and 7 is the comparison of measurement and initial analysis results for engine single and generator condition. The dynamometer condition is omitted in this paper because their characteristics are very similar to that of engine single model. In case of engine single model, the natural frequency is very well predicted but response level has a large discrepancy. And generator model including torsional damper shows large difference between the initial analysis and measurement. As mentioned above, engine single model removed the uncertain elements as possible, so there is no noticeable error except for damping parameter.

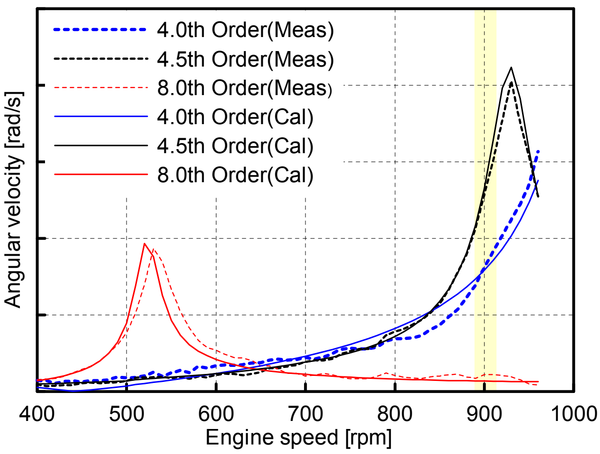


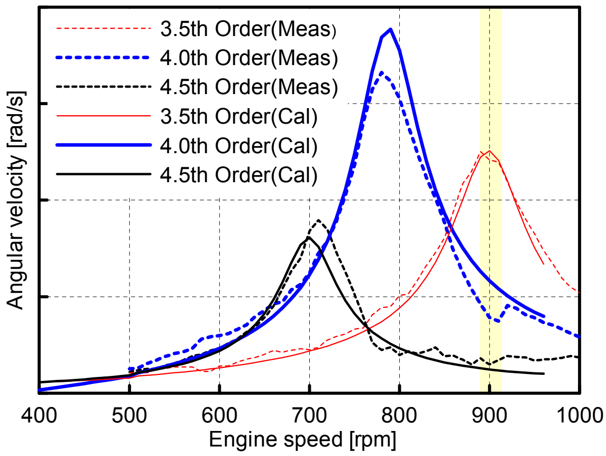


(a) Engine single condition (b) Generator condition

Figure 6. Comparison of measurement and initial analysis

Based on the sensitivity analysis, two analysis models were tuned for better accuracy. In engine single model, inertia of node 1 which include the several pumps was adjusted for more accurate natural frequency because they are relatively inaccurate compared with the crankshaft. And four times of initial value of external damping are used to fit the response level. The result was shown in Figure 7 (a) and it is clear that improvement of fitting was achieved compared to Figure 6 (a). It is also concluded that the used inertia and stiffness of crank throw are fairly reasonable. In generator model, torsional damper parameter were controlled because the crank shaft is already confirmed in engine single model and rotor of generator has a little effects on free vibration characteristic as shown in Table 2. Therefore stiffness of node 1 (damper) is adjusted almost two times of initial value proposed by maker and the result of Figure 7 (b) was obtained.





(a) Engine single condition (b) Generator condition

Figure 7. Comparison of measurement and tuned analysis

Above procedures were applied to model tuning for the various system condition step by step. Controlled parameters of inertia and stiffness were summarized in Table 3 and 4. Engine single and dynamometer model show the same results of tuning. If the tuning wheel of 20 $kg \cdot m^2$ to change the natural frequency by inertia is attached to flange, the external damping of cylinders are decreased to two times of initial damping. Meanwhile, the internal damping of crankshaft was not controlled. In previous study[1], optimal values of internal and external damping on different engine type were proposed and 50 of dynamic magnifier was used in this analysis. In case of generator model, the initial external damping was used as it is, because its variations does not effect to vibration response unlike to others. Instead, the damping of torsional vibration damper was adjusted. The stiffness of torsional damper critically influenced natural frequencies and vibration responses as shown in Figure 7 (b). These results coincide with those of sensitivity analysis.

Table 3. Tuning results of engine single and dynamometer model

Parameters	Without damper		With tuning wheel	
	Initial	Tuned	Initial	Tuned
Inertia of pumps ($kg \cdot m^2$)	0.39	0.78	0.39	0.78
External damping (Nms/rad)	18.34	73.36	18.34	36.68

Table 4. Tuning results of generator model

Parameters	Initial	Tuned
Inertia of pumps ($kg \cdot m^2$)	0.39	0.78
External damping (Nms/rad)	18.34	18.34
Stiffness of damper (MNm/rad)	8.5	16
Damping of damper (Nms/rad)	1100	1400

4. DYNAMIC STRESS MEASUREMENT

The fatigue strength of a marine engine crankshaft is evaluated based on the IACS M53 rule[5] or FE analysis in early design stage. In that time, the radial and tangential forces which acting on crank pin and alternating torque are applied as excitations. The alternating torque includes the dynamic characteristics of shafting system and should be calculated through the torsional vibration analysis.

In this paper, dynamic stress measurement was briefly introduced and the results were presented with the analysis ones. The concentrated stress of crank pin fillet was measured by rosette gage and telemetry system under realistic operating conditions. The position and direction of the gage was shown in Figure 8.

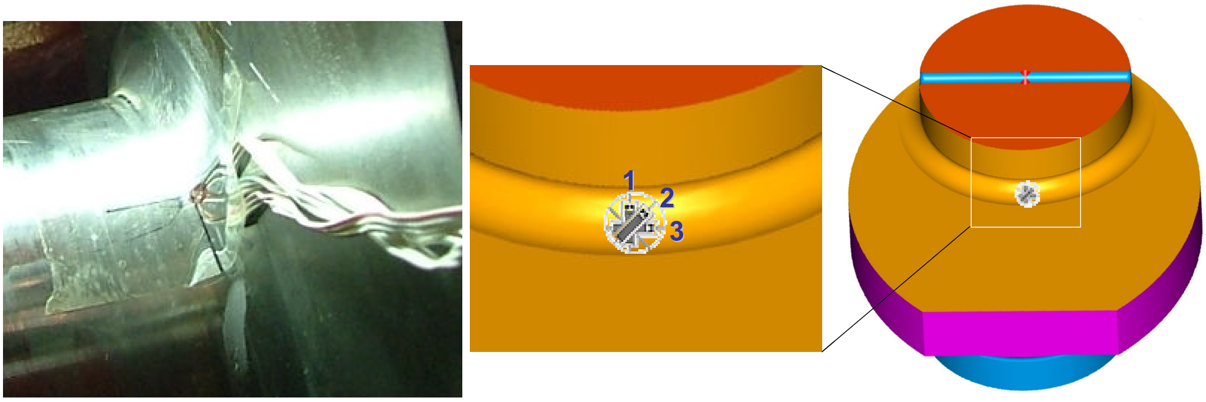
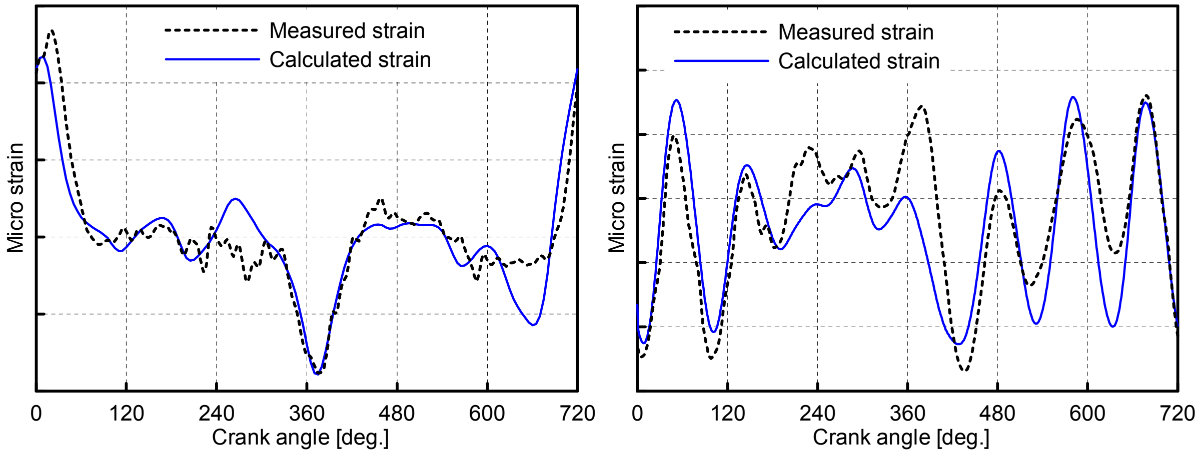


Figure 8. Installation of rosette gage in pin fillet

In Figure 9, the measured results were compared with FE analysis results. Alternating torque as excitation for FE analysis was calculated from the final tuned dynamometer model in Chapter 3. Strain of No.1 gage is dominated by radial force and No.2 is affected to torque due to the torsional vibration. As shown in Figure, the good agreement between the experiment and analysis was verified. Using these results, the fatigue strength of crankshaft was evaluated and it is concluded that the crankshaft has sufficient safety against dynamic loads of engine.



(a) No. 1 gage (b) No. 2 gage

Figure 9. Comparison of measured and analyzed strain

5. CONCLUSIONS

The crankshaft torsional vibration of the newly developed medium speed diesel engine was investigated with the analytical and experimental method. Characteristics of torsional vibration were observed for the various conditions of shafting system through the sensitivity analysis and model tuning. Model tuning to obtain the reliable analysis model was made based on the experiment and the results of sensitivity analysis were successfully adopted in process of adjustment of system parameters.

Dynamic stresses in the highly stressed area of the crankshaft were measured and briefly introduced. The measured and analyzed results show the considerable agreement and were used for evaluation of fatigue safety.

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