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The Torsional Vibration Monitoring of the Flexible Coupling between a Gas Diesel Engine and a High-Pressure Compressor

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1. Introduction

Flexible coupling is generally used to reduce the vibratory torque or excess angular acceleration in diesel power plant, reciprocating compressor and electric motor.

In this paper, the theoretical analysis and measurements for torsional vibration were performed on a flexible coupling between a gas engine type GV180TI prime mover and the high-pressure compressor type GEO-C4. The angular velocity amplitudes of flexible coupling were measured at both the engine and the compressor sides (respectively inner and outer of flexible coupling), due to the excitations caused by the same machineries.

The measured values are used directly and transformed into appropriate parameters (torque or stress) for evaluating the vibration and stability between the gas engine and the compressor.

Lastly, continuous monitoring of the torsional vibration when the system is in operation to warn plant's failure as soon as the angular velocity of flexible coupling is beyond the set limits.

2. Torsional vibration theoretical analysis

2.1 Outline for mass and elastic system

Figure 1 describes the mass elastic model for torsional vibration.

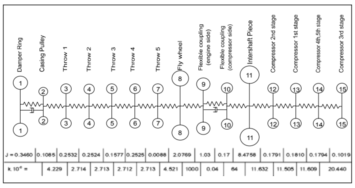

Fig. 1 Mass elastic model for torsional vibration

2.2 Dynamic characteristic of flexible coupling

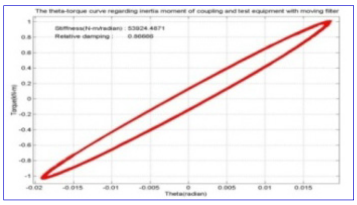

Fig. 2 Torque-angular displacement hysteresis curve by exciting frequency 10 Hz with filter

The flexibility and relative damping for flexible coupling was tested and estimated by hydraulic excitation system. Fig. 2 shows the torsional stiffness and damping testing result together with a hysteresis loop.

2.3 Excitation of torsional vibration on the compressor

This kind of excitation in the compressor is a combined force of compressed gas force and inertial force of reciprocating mass.

$$F_{CombinedForce} = F_{Comp.Gas} + F_{InertiaMass} \quad (1)$$

With this type of the compressor, the 1st, 2nd and 3rd stage is double acting cylinder, therefore the resultant gas force is sum of forces from the crank end side and head end side.

$$F_{Comp.Gas} = (A_{CE} \times P_{CE}) - (A_{HE} \times P_{HE}) \quad (2)$$

The inertia force ($F_{InertiaMass}$) is formed by the inertia mass of all reciprocating compressor parts.

The harmonics for torque variation of the 2nd stage and combination of 4th & 5th stages of the compressor are illustrated through figures 4. Normally, the 1st order will have the highest torque value (being the 4th and 5th stage), but it is dominant in 2nd order with double acting piston type (being the 1st, 2nd and 3rd stage).

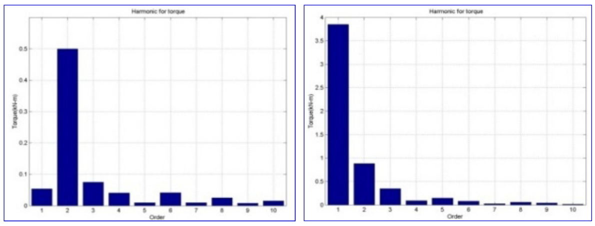

Fig. 4 Harmonics for torque variation of 2nd stage and 4th, 5th stage compressor

2.4 Torsional vibration characteristics

The figure 5 shows the torsional results with different stiffness after using Dynamics lab calculation programs.

1500.0 RPM 1.0 ORDER

MASS	AMPLITUDE MRAD	PHASE DEG.	TORQUE KN-M	PHASE DEG.	STRESS N/MM**2	EXC.TORQUE N KN-M	PHASE DEG.
MAIN BRANCH							
1	11.5084	172.8	0.11	19.4			
2	11.5084	172.8	0.16	11.2	1.11		
3	11.4728	172.8	0.75	146.7	5.24	0.94	155.5
4	11.7216	172.2	1.22	-166.3	8.54	0.94	-132.5
5	12.1420	172.9	1.22	148.3	8.50	0.94	83.5
6	12.5511	172.1	0.53	-159.3	3.70	0.94	-60.5
7	12.7227	172.5	0.50	-1.1	3.52	0.94	11.5
8	12.6121	172.5	1.15	-4.2	8.02		
9	12.6109	172.5	1.47	-4.7	10.26		
10	7.1653	161.1	1.50	-5.0	10.46		
11	7.1426	161.1	2.97	-11.5	20.77		
12	6.8893	160.8	2.98	-10.6	29.64	0.05	104.7
13	6.6334	160.5	2.99	-10.0	29.70	0.04	113.2
14	6.3799	160.1	0.06	-91.4	0.60	3.01	171.0
15	6.3790	160.1	0.00	0.0		0.07	101.9

Fig. 5 Detailed torsional calculation results with the flexible coupling of stiffness 250 kN-m

3. Measurement & Availability for torsional vibration monitoring system

3.1 Measurement results

The torsional vibration of the flexible coupling

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represented by the angular velocity amplitude was measured by the Laser Torsionmeter Polytec performed between the outer and inner side of the flexible coupling during run up and no load condition. Signal processing for total data was carried out with the EVAMOS software which was developed by the Dynamics Lab of Mokpo Maritime University. Figures 6 below, shows the schematic diagram for the vibration measurement and the equipment used.

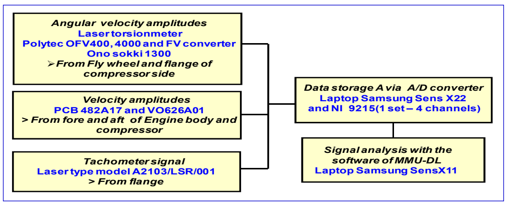

Fig. 6 Schematic diagram for global vibration monitoring system

Table 1 below shows the achieved results during full load and 1500 rpm engine mode.

Table 1 Vibration measurement result at full load and 1500rpm

Description	Amplitude (mrad or mm/s)	Phase angle (degree)
Flexible coupling (engine side)	10.22	161.0
Flexible coupling (comp. side)	9.07	165.0
Engine fore transverse	14.92	90.0
Engine aft transverse	9.88	78.3
Compressor fore transverse	2.41	82.6
Compressor aft transverse	8.017	140.0

The measurements for the angular velocity amplitude at flywheel (engine side) and flange (compressor side) during run-up with all orders are shown in figures 7.

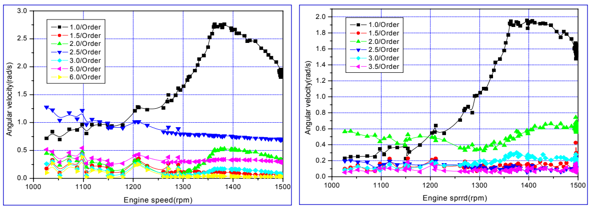

Fig. 7 The angular velocity amplitude measurement of of flywheel and flange during run-up

Figure 8 shows the 1st order angular velocity amplitude and phase angle measurements at the engine side (flywheel) and the compressor side (flange) respectively.

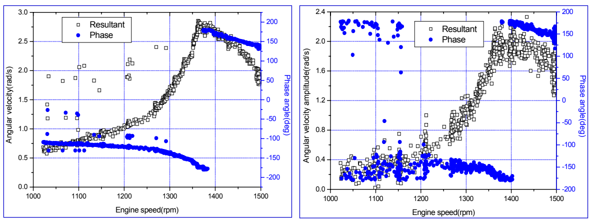

Fig. 8 The 1st order angular velocity amplitude and phase angle measurement of flywheel and flange during run-up

3.2 Availability for torsional vibration monitoring system

Even if the plant is running in normal condition, it is working at high speed of 1500±100 rpm and high pressure of nearly 300 bar. Dangerous conditions are imminent should there be a failure in the flexible coupling. Thus, the flexible coupling should be monitored constantly for safety purposes.

In this paper, the authors used the integrated vibration and condition monitoring system to measure and monitor the torsional vibration of the flexible coupling. Figure 9 shows the configuration for 4 channels of NI A/D converter. Further, high and low value alarms of angular velocitys were set directly on the software used. Whenever the measured value is out of the value limit setting, the alarm will appear simultaneously in sound and red light at status panel. The signal status panel of EVAMOS program is shown at figure 10.

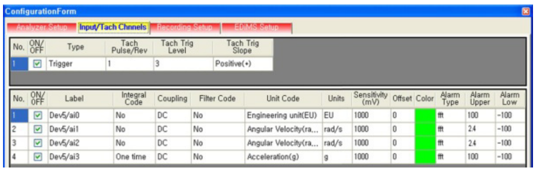

Fig. 9 The channel configuration panel of EVAMOS

Fig. 10 The signal progress status panel of EVAMOS

In this test, the measured maximum angular velocity amplitude at normal working speed (1500rpm) is 2.0 rad/s at 1st order (figure 7, 8) due to the 1st order excitation of 5th stage compressor. Hence, the high value alarm will be set at 120 percent (120%) of the maximum angular velocity amplitude corresponding to 2.4 rad/s. The alarm will be triggered once the value of the measured angular velocity is higher than the set high value alarm.

4. Conclusion

In this paper, the study has made the following results;

1) Even though the plant is working normally, torsional vibration monitoring system should be applied to monitor the torsional vibration of the flexible coupling continuously. Alerts are made as soon as the angular velocity of flexible coupling is beyond the set limits.

2) The torsional vibration of CNG shafting with a multi stage reciprocating compressor was mainly affected by the 1st order torque variation of the 4th and 5th stage compressor, due to the single acting piston at high pressure side. Consequently, this system should be controlled to avoid the resonance of the 1st node torsional vibration within the engine operational range.

3) The resonance of torsional vibration depends on the non-linear stiffness of flexible coupling. System designer should be careful in checking and designing this characteristic. Thus, to get the exact data from the test results, the inner and outer side of the flexible coupling should be tested for reliability.