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A Study on the Reduction of the Torsional Angular Acceleration on Chain Drive Wheel of Marine Diesel Engine

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Abstract : When the propulsion shafting system of marine diesel engine is designed, the vibratory stresses on shafts should be reviewed and be satisfied with limits which are laid down by classification societies. In addition, the torsional vibration aspects for crankshaft of main engine are requested to be checked by engine designers.

Especially, for the 4, 5, and 6-cylinder engines, the 2nd order moment compensator(s) may be installed to compensate the external moments of engine and not to excite the hull girder vibration. This moment compensator which is mounted on fore and/or after-end of engine is driven by the roller chain drive for some of MAN 2-stroke diesel engines. While the engine is running, the roller chain is worn down, which causes the extension of roller chain. The chain therefore should be checked and tightened by periods in order to keep its functionality.

However, when the torsional angular acceleration of chain drive exceeds the certain limit, the chain will suffer the excessive slack and transverse vibration. This may cause fatigue, wear or damage on the chain and the chain ultimately may be broken.

The research object of this thesis is to review factors which affect the angular acceleration of chain drive and to find out how to decrease the angular acceleration of driving chain by checking factors which have a major contribution to acceleration reduction using the statistical method of DOE(design of experiment), correlation analysis and regression analysis methods.

Key words : Torsional vibraton, Angular acceleration, Design of Experiment, Correlation, Regression

1. Introduction

The torsional vibration calculation for the propulsion system of diesel engine has been set up by the effort of many researchers after the paper regarding this had been issued by Bauer in 1900s[1]. When the propulsion shafting system of marine diesel engine is designed, the shaft diameter and its length is determined according to the rule of classification society. Then the natural frequencies and their corresponding amplitudes of torsional vibration is

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reviewed and confirmed in order that the calculated stress amplitude does not exceed the limit stress defined by the rule. The general methods to avoid the exceeded stress are to move the resonant point out of the service operating speed or to lower the stress level below the limit value by modifying the shaft diameter and/or the moment of inertia of flywheel or by applying the tuning wheel or the torsional vibration damper.

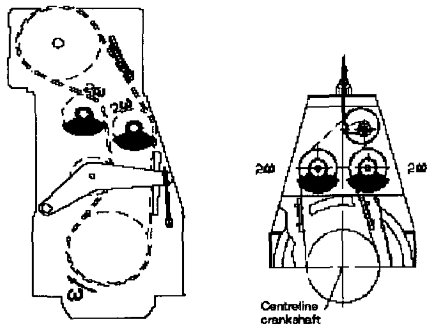
When the frequency of 2nd order moment in vertical excitation which is coming from engine coincides with one of the natural frequencies of hull structure, the resonance will occur and result in the quite high vibration level. In order to control the resulting vibratory responses of hull, the 2nd order moment compensator can be installed on fore and/or after side of MAN 2 stroke diesel engines. This compensator is driven by chain drives which rotate the camshaft by a chain connection from the crankshaft to camshaft. While the engine is running and the excessive torsional vibration of crankshaft is transferred to chain, the chain may cause the fatigue, wear or damage in the chain, chain drive, chain wheel and guide bar and lead to a break of the chain. Therefore, the MAN Diesel A/S has a limit of angular acceleration on chain drives in order for the chain to have a safe running condition

The research object of this thesis is to review factors which affect the decrease of angular acceleration of chain drive and to find out how to reduce the angular acceleration of driving chain by checking factors which have a major contribution to acceleration reduction using the statistical method of DOE(design of experiment), correlation analysis and regression analysis methods.

2. Chain drive system and the angular acceleration of chain drive wheel

2.1 2nd Moment compensator and Roller chain drive system

The company MAN Diesel A/S designs large marine diesel engines and these engines have for many years used roller chain drive to drive the camshaft by a chain connection from the crankshaft to camshaft as seen Fig.2.1. The roller chains on these engines are very large and this chain may wear and extend while the engine is running. The chain therefore has to be tightened, frequently as it gradually extends, in order for chain to be functional.



(a) After side main chain (b) Fore side chain

Fig. 2.1 Chain drive system of MAN engines[2]

In order to avoid excessive excitation when the frequency of excitation coincides with the natural frequency of the ship hull vibration it is relevant to consider outbalancing. Dependent on the engine type, some of the sprockets in the chain drive system are fitted with counterweight which can reduce the vertical moment[3].

2.2 Angular acceleration limit at chain drives

Engines with a 2nd order moment compensator have two chain wheels with counterweights incorporated in the camshaft drive and optionally a separate drive at the fore end of the crankshaft. The high moment of inertia of the counterweights exposes the chain drive to acceleration forces that have to be limited. In order to secure the safe operation of the chain drives, the angular acceleration of each plant shall be checked for each limitations.

When the synthesized angular acceleration at the chain drive of crankshaft is below LIMIT-1, the full size of counterweight on moment compensator can be applicable but the outbalancing force should be reduced according to the corresponding acceleration limit by applying the reduced counterweight in order to lower the load of chain by heavy counterweights and decrease the excessive vibration of chain when the acceleration is over LIMIT-1. When the acceleration is over LIMIT-2, the electric balance should be substituted instead of the integrated 2nd moment compensator. The LIMIT-1 and LIMIT-2 for MAN 5S70MC-C engine which is reviewed in this thesis is 17rad/s^2 and 23rad/s^2, respectively[4].

3. Mathematical equation of propulsion shafting system

3.1 Shafting system

The general particular of shafting system and engine is described on Table 3.1. The engine has the T/V(Torsional Vibration) damper on the free end of crankshaft to reduce the vibratory stress of shafting system and two 2nd order moment compensators on fore end and driving end of crankshaft to lower the external moment of engine and hull vibration by engine.

Table 3.1 General particulars of the shafting system and engine

Item		Description
Engine type		MAN 5S70MC-C
Bore/Stroke		600mm / 2800mm
Rating		21,100BHP x 91rpm
T/V damper		Geislinger D290/6
Flywheel inertia		13,000 kg·m2
Prop eller	Type	FPP
	Dia. x Blade No.	8.0m x 4 blades
	Mas in air	31,880kg
	Moment of inertia in air	84,450 kg·m^2

3.2 Mathematical equation

The mechanical impedance method is used to calculate the torsional vibration system for stead-state response. The mathematical model of engine and shafting system is described on Fig.3.1. The equation of motion of the system, in matrix form, can be written as follows.

$$[M]\{\ddot{\theta}\}+[C]\{\dot{\theta}\}+[K]\{\theta\}=\{f(t)\} \quad (1)$$

For the moment of inertia matrix[M], each mass is regarded as a discrete lumped one. The mass-elastic data of MAN is used for the lumped mass for main engine and the mass of shaft is divided into its both ends with its half mass respectively. For the damping matrix[C], the 0.85% of damping ratio as the outer damping of each cylinder is used and 5.5% for propeller. And the 1% of inner damping ratio is used for journal shafts of main engine as its hysteresis damping. For stiffness matrix[K], each shaft is regarded as a massless elastic element and is composed of the stiffness of shaft between the adjacent elements. $\{\theta\}$ is the vector of body rotations. $\{f(t)\}$ is excitation torque and consists of the gas forces and inertia forces for each cylinders.

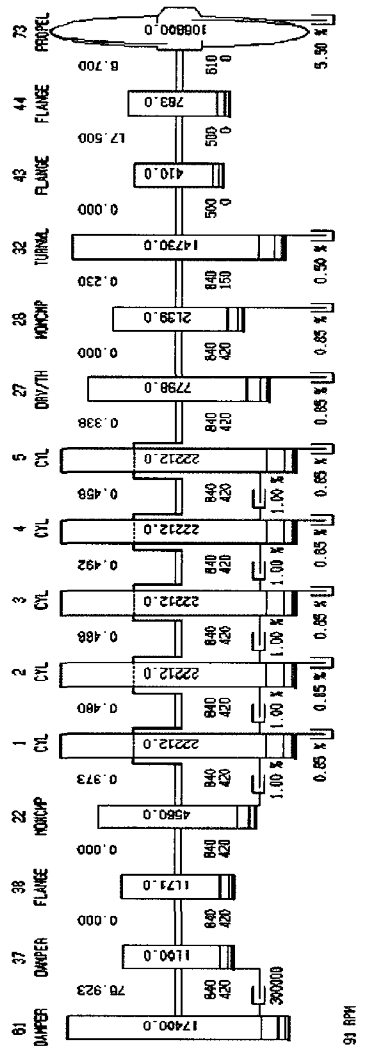


Fig. 3.1 Equivalent mass elastic system

For the equation(1),

$$\{f(t)\} = \{\bar{f}\}e^{j\omega t} \tag{2}$$

where $\{\bar{f}\}$ is complex excitation with the phase angle, ω is the excitation frequency, and $j=\sqrt{-1}$. When the steady-state is considered only, the frequency of response is equal to the one of excitation. Consequently, the amplitude can be written as follows.

$$\{\theta\} = \{\bar{\theta}\}e^{j\omega t} \tag{3}$$

where $\{\bar{\theta}\}$ is complex amplitude with the phase angle. Then the equation(1) can be rearranged using equation (2) and (3) as.

$$(-\omega^2[M] + [K] + j\omega[C])\{\bar{\theta}\} = \{\bar{f}\} \tag{4}$$

$$[Z]\{\bar{\theta}\} = \{\bar{f}\} \tag{5}$$

$$[Z] = -\omega^2[M] + [K] + j\omega[C] \tag{6}$$

where [Z] is impedance matrix. And the complex amplitude can be written as

$$\{\bar{\theta}\} = [Z]^{-1}\{\bar{f}\} \tag{7}$$

And suppose as follows for convenience.

$$[Z_R] = [K] - \omega^2[M] \tag{8}$$

$$[Z_I] = \omega[C] \tag{9}$$

$$[H] = ([Z_R] + [Z_I][Z_R]^{-1}[Z_I])^{-1} \tag{10}$$

$$[L] = ([Z_R] + [Z_I][Z_R]^{-1}[Z_I])^{-1} \cdot [Z_I][Z_R]^{-1} \tag{11}$$

Then, the complex amplitude is determined as follow using equations (8)~(11).

$$\{\bar{\theta}\} = ([H] - j[L])\{\bar{f}\} \tag{12}$$

The amplitude for each elements can be obtained using (12) by calculating $\{\bar{\theta}\}$ for all orders and running rpm[5]. The angular acceleration can be obtained by differentiate equation (3) twice after substituting the result of (12) into (3) as follow.

$$\{\ddot{\theta}\} = -\omega^2([H]-j[L])\{\bar{f}\}e^{j\omega t} \quad (13)$$

Then, the synthesized angular acceleration $\{\ddot{\theta}_{syn}\}$ can be obtained by calculating the k[th] order's angular acceleration $\{\ddot{\theta}_k\}$, summing 16 orders up and dividing the sum of the maximum and minimum value while the engine rotates one revolution.

$$\{\ddot{\theta}_k\} = -(k\omega)^2([H]-j[L])\{\bar{f}\}e^{j\omega t} \quad (k=1,2,\cdots,16) \quad (14)$$

$$\{\ddot{\theta}_{\theta,syn}\} = \{\sum_{k=1}^{16}\ddot{\theta}_k\} \quad (\theta=0^\circ\sim359) \quad (15)$$

$$\{\ddot{\theta}_{syn}\} = \frac{(\ddot{\theta}_{\theta,\max}-\ddot{\theta}_{\theta,\min})}{2.0} \quad (\theta=0^\circ\sim359) \quad (16)$$

3.3 Torsional vibration calculation and the synthesized angular acceleration

The torsional stress which is requested to confirm by classification society is reviewed as normal with barred speed range(44rpm~50rpm) for intermediate and propeller shaft due to that the stress on intermediate shaft is over the limit for continuous operation.

The calculated maximum angular acceleration on fore-moment and after-moment compensator is 20.2[rad/s^2] and 20.8[rad/s^2] at 86[rpm] as shown in Fig.3.2 and Fig.3.3 respectively. This shows that the size of counterweight should be reduced as long as this reduction will not invoke the unintended hull vibration because the acceleration exceeds LIMIT-1 except the barred range.

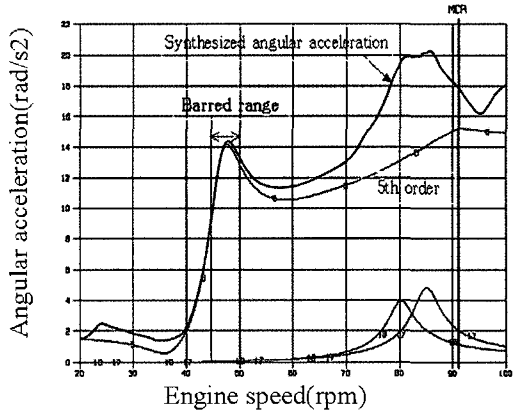


Fig. 3.2 Angular acceleration on the fore moment compensator

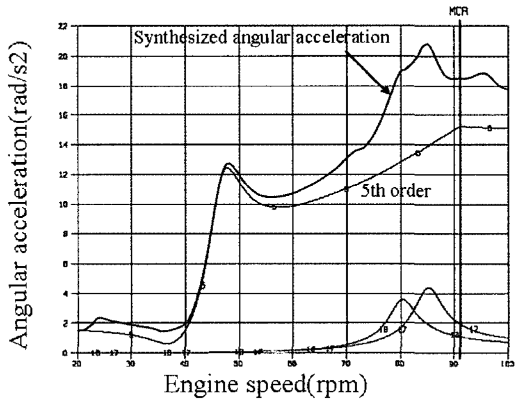


Fig. 3.3 Angular acceleration on the after moment compensator

4. Reduction of angular acceleration using Design of Experiment

4.1 Design Of Experiment(DOE)

Design of Experiment (DOE) is a technique to lay out the experimental plan of investigation, studies, survey, tests, etc in most logical, economical, and statistical way. The optimization design by DOE suggests the good way on the complex field like multi-objective optimization problems which have difficulty solving the existing optimization program[6]. We use the DOE in this study in order to find which design factors have the major contribution to reduction of the angular acceleration. And correlation analysis and regression analysis is used as tool of data processing to determine which factors have the higher effect on acceleration.

4.2 Experimental factors to angular acceleration

Although there are several design factors which affect the torsional angular acceleration of shafting system, the moment of inertia(MOI) of tuning wheel/flywheel, damping of T/V damper, stiffness of intermediate shaft and MOI of propeller which are used generally are adopted in this thesis. The experiment and analysis is divided into two cases because the tuning wheel and T/V damper cannot be installed on a engine simultaneously. The cases and its factors are as follows.

(1) Case 1: MOI of tuning wheel, MOI of flywheel, stiffness of intermediate shaft, MOI of propeller
(2) Case 2: damping of T/V damper, MOI of flywheel, stiffness of intermediate shaft, MOI of propeller

The two levels of maximum and minimum values for each design factors are used as follows. The Table 4.1 shows the cases that the maximum size of tuning wheel is installed and not installed with other various and different design factors.

Table 4.1 Factors and its level for tuning wheel

Item	T/W MOI	F/W MOI	Inter.Shaft stiffness	Prop. MOI
Min.	0	13000	57.1×10^6	85440
Max.	60000	43000	81.6×10^6	128160

The Table 4.2 shows the cases that the maximum and minimum size of T/V damper is installed with other various and different design factors.

Table 4.2 Factors and its level for T/V damper

Item	TVD damping	F/W MOI	Inter.Shaft stiffness	Prop. MOI
Min.	30000	13000	57.1×10^6	85440
Max.	700000	43000	81.6×10^6	128160

The full factorial design is used because the no. of design factors is 4 for each case. Therefore, total 32($=2^4$ x 2cases) torsional calculations are performed. For each calculation, the synthesized angular acceleration at fore and after-end chain drives are calculated and the maximum

Table 4.3 DOE design and its result for tuning Wheel(Case 1)

Case	Input				Output		
	Tuning wheel M.O.I.	Fly-wheel M.O.I.	Inter. shaft stiffness	Propeller M.O.I.	Fore-max acceleration	Aft-max acceleration	Stress on shaft
1-1	0	13,000	43.9×10^6	85,440	**24.8**	**25.0**	N
1-2	0	13,000	43.9	128,160	**24.8**	**24.9**	N
1-3	0	13,000	81.6	85,440	**27.0**	**26.8**	N
1-4	0	13,000	81.6	128,160	**26.4**	**26.7**	N
1-5	0	42,000	43.9	85,440	22.0	20.2	N
1-6	0	42,000	43.9	128,160	22.0	20.0	N
1-7	0	42,000	81.6	85,440	20.0	19.0	N
1-8	0	42,000	81.6	128,160	20.0	18.9	N
1-9	60,000	13,000	43.9	85,440	18.6	19.0	N
1-10	60,000	13,000	43.9	128,160	18.6	18.9	N
1-11	60,000	13,000	81.6	85,440	19.5	19.5	OK
1-12	60,000	13,000	81.6	128,160	19.2	19.3	N
1-13	60,000	42,000	43.9	85,440	15.3	15.3	OK
1-14	60,000	42,000	43.9	128,160	15.2	15.2	N
1-15	60,000	42,000	81.6	85,440	16.2	15.8	OK
1-16	60,000	42,000	81.6	128,160	16.0	15.7	N

Table 4.4 DOE design and its result for T/V damper (Case 2)

Case	Input				Output		
	T/V damper damping	Fly-wheel M.O.I.	Inter. shaft stiffness	Propeller M.O.I.	Fore-max acceleration	Aft-max acceleration	Stress on shaft
2-1	30,000	13,000	43.9×10^{6}	85,440	22.6	23.0	N
2-2	30,000	13,000	43.9	128,160	22.5	22.8	N
2-3	30,000	13,000	81.6	85,440	24.8	24.6	N
2-4	30,000	13,000	81.6	128,160	24.2	24.5	N
2-5	30,000	42,000	43.9	85,440	17.0	16.3	N
2-6	30,000	42,000	43.9	128,160	16.9	16.3	N
2-7	30,000	42,000	81.6	85,440	17.8	17.2	N
2-8	30,000	42,000	81.6	128,160	17.8	17.0	N
2-9	700,000	13,000	43.9	85,440	18.3	18.7	OK
2-10	700,000	13,000	43.9	128,160	18.1	18.6	OK
2-11	700,000	13,000	81.6	85,440	20.4	20.3	OK
2-12	700,000	13,000	81.6	128,160	19.9	20.0	OK
2-13	700,000	42,000	43.9	85,440	16.3	15.5	OK
2-14	700,000	42,000	43.9	128,160	16.1	15.5	OK
2-15	700,000	42,000	81.6	85,440	17.5	16.3	OK
2-16	700,000	42,000	81.6	128,160	17.2	16.1	OK

level of acceleration between 0.2 and 1.2 of MCR speed is gotten for investigation. The maximum of synthesized angular acceleration at fore and after-end chain drives are listed for each cases like the application of tuning wheel and T/V damper at Table 4.3 and Table 4.4 respectively. These table shows that the angular acceleration can be reduced considerably by the heavy tuning wheel or large T/V damper.

5. Review of analysis and angular acceleration reduction

5.1 Angular acceleration for tuning wheel

The correlation analysis for fore-moment compensator in case of tuning wheel application shows that the factor of MOI of tuning wheel is effective to reduce the acceleration because its correlation coefficient(-0.811) is $|r|>0.7$ and the P-value(0.000) is less than 0.05. The MOI of flywheel is less effective than MOI of tuning wheel because correlation coefficient (-0.540) is $0.4 \le |r| \le 0.7$.(refer to Table 5.1)

Table 5.1 Result of correlation analysis for fore moment compensator in case of tuning wheel

Item	Tuning wheel moment M_{FTW}	Flywheel moment M_{FFW}	Intermediate shaft stiffness k_{FIS}	Propeller moment M_{FPP}
Correlation r	-0.811	-0.540	0.050	-0.020
P-Value	0.000	0.031	0.853	0.941
Result	Accept	Reject	Reject	Reject

The regression analysis for fore-moment compensator in case of tuning wheel application shows that the regression equation(17) is valid because the R^2 of equqtion(18) is over 65% and the factors to contribute to low acceleration is MOI of tuning wheel and flywheel because its P-value is lower than 0.05 and is accepted. The higher its MOI is, the lower the acceleration is. The factors like stiffness of intermediate shaft and MOI of propeller is rejected and have no contribution to low acceleration because the P-value is greater than 0.05.

$$\alpha_F = 26.9 - 0.101\times10^{-3} M_{FTW} - 0.139\times10^{-3} \cdot M_{FFW} + 0.100 k_{FIS} - 3.51\times10^{-6} M_{FPP} \quad (17)$$

Table 5.2 Result of regression analysis for fore moment compensator in case of tuning wheel

No	Predictor	Coefficient	SE Coefficient	t	P
(a)	Constant	26.943	1.584	17.01	0.000
(b)	M_{FTW}	-0.10083×10^{-3}	8.18×10^{-6}	-12.33	0.000
(c)	M_{FFW}	-0.13879×10^{-3}	16.92×10^{-6}	-8.20	0.000
(d)	k_{FIS}	0.0995	0.1302	0.76	0.461
(e)	M_{FPP}	-3.51×10^{-6}	11.49×10^{-6}	-0.31	0.766

$$S=0.982, R2=95.2\%, R2(adj)=93.5\% \quad (18)$$

In addition, the regression equation(17) can be used to calculate the angular acceleration using the described design factors in a quick and simple say. Also we get the similar results for the after-end compensator when the tuning wheel is applied.

5.2 Angular acceleration for T/V damper

The correlation analysis for fore-moment compensator in case of T/V damper application shows that the factor of MOI of flywheel is effective to reduce the acceleration because its correlation coefficient(-0.775) is |r|>0.7.(refer to Table 5.3)

Table 5.3 Result of correlation analysis for fore moment compensator in case of T/V damper

Item	T/V damper damping d_{FTV}	Flywheel moment M_{FFW}	Intermediate shaft stiffness k_{FIS}	Propeller moment M_{FPP}
Correlation r	-0.449	-0.775	0.267	-0.045
P-Value	0.081	0.000	0.317	0.868
Result	Reject	Accept	Reject	Reject

The regression analysis for fore-moment compensator in case of T/V damper application shows that the regression equation(19) is valid and the factors to contribute to low acceleration is MOI of tuning wheel and flywheel. The higher its MOI is, the lower the acceleration is. The factors like stiffness of intermediate shaft and MOI of propeller is rejected and has no contribution to low acceleration.

$$\alpha_F = 22.8 - 3.6x10^{-6} d_{FTV} - 0.147x10^{-3} \cdot M_{FFW} + 0.391k_{FIS} - 5.9x10^{-6} M_{FPP} \quad (19)$$

$$S=1.174, R2=87.5\%, R2(adj)=83.0\% \quad (20)$$

Table 5.4 Result of regression analysis for fore moment compensator in case of T/V damper

No	Predictor	Coefficient	SE Coefficient	t	P
(a)	Constant	22.785	1.899	12.00	0.000
(b)	d_{FTV}	-3.69×10^{-6}	0.88×10^{-6}	-4.22	0.000
(c)	M_{FFW}	-0.14741×10^{-3}	20.24×10^{-6}	-7.28	0.000
(d)	k_{FIS}	0.3912	0.1557	2.51	0.029
(e)	M_{FPP}	-5.85×10^{-6}	13.74×10^{-6}	-0.43	0.678

We get the similar results also for the after-end compensator. when the T/V damper is applied.

6. Conclusion

The DOE method has been proposed to find out the most effective way that the torsional angular acceleration at chain drive can be reduced in order to secure safe operation of chain drive system when the propulsion shafting system of marine diesel engine which has the 2nd order moment compensation to compensate the external moment of engine is designed.

The two levels of design factors like MOI, stiffness and damping to contribute to the acceleration are adopted and reviewed using correlation analysis and regression analysis. Its conclusion can be summarized as follows.

(1) The bigger MOI of tuning wheel and flywheel is effective to reduce the angular acceleration when the tuning wheel is installed on engine.

(2) The bigger damping of T/V damper and MOI of flywheel is effective to reduce the angular acceleration when the T/V damper is installed on engine.

(3) For the application of tuning wheel or T/V damper, the tuning wheel is proved as the outstanding solution on the

point of technical and economical view.

Also further investigation on the relation between chain local vibration, weight of counterweight, pretension of chain should be done in order to get the exact effects on chain vibration.

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