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Optimizations of both reliability and thermal efficiency of a medium-speed diesel engine

Mechanics, Materials & Coatings

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ABSTRACT

One of the main methods of reducing maritime GHG and carbon footprint is to improve the thermal efficiency of the prime mover medium-speed diesel engine and lower the fuel consumption. Furthermore, a reduction in the lifecycle cost of maritime operation will improve sustainability for shipowners to invest further in new technology in the transition toward low carbon and GHG emission performance.

The optimization of the usually conflicting engine performance and emission will require shorter combustion duration and hence higher peak firing pressure (PFP). With the introduction of new and optimized equipment such as the turbocharger and the fuel injection system and the combustion to gain higher thermal efficiency, this has resulted in exceeding the current engine design limit and design changes were found necessary.

This paper provides details of predictive simulation analysis activities to ensure proper assessment to minimize any risk of failure due to increased mechanical and thermal loads associated with target gained thermal efficiency. The engine baseline (current version) already has a long reputation for reliability in the local and international marine and rail operations, which need to be maintained for the enhanced version of the engine.

Among the main activities carried out for the optimization process was the consideration of the crankshaft assessment and how to use the same space available but with design changes to pass the approval of classification societies rules. In order to improve the calculation accuracy, one-dimensional dynamics software is used to simulate the crankshaft dynamics and evaluate the engine shafting and torsional vibration to ensure that the engine shafting and torsional vibration are optimized under the increased target thermal efficiency.

The validated tools of CAE simulations of computational fluid dynamics (CFD), finite element method (FEM) and 3D multi-body dynamics are used to carry out the calculation. The calculations have also been considered with the influence of lubrication, bearing shell and cylinder block deformation, so as to evaluate the reliability of the crankshaft, static and moving parts of the engine in comprehensive manner. These assessments are used to provide a dual objective, first to ensure that engine performance and emissions are optimized, and secondly to provide precise numerical data boundary conditions for the structural analysis, fatigue and life assessment of enhanced engines with the resulting higher PFP.

1 INTRODUCTION

The crankshaft is one of the most important and expensive moving parts on the diesel engine. The quality of the crankshaft determines whether the diesel engine works reliably and whether the design life can be achieved. Applying finite element technology to crankshaft design, on one hand, can simulate and calculate the stress and strain of the crankshaft in the working process of diesel engine, find out the stress concentration, and then calculate the fatigue strength and fatigue life of the crankshaft, which is the best way to judge whether the design of the crankshaft is reasonable. On the other hand, finite element technology can intuitively simulate and calculate the influence of different size parameters on the strength and stiffness of the crankshaft, and find out the best optimal design scheme.[1] During the working process, the crankshaft of the diesel engine will always be subject to the action of periodic load force. If the design is improper, the crankshaft may resonate in the working speed, which leads to a sharp increase in stress, resulting in premature torsion and bending fatigue damage of the crankshaft. Empirical design and static design can not meet this requirement, but dynamic design can make up for the deficiency.[2-3]

With the continuous development of internal combustion engines towards high reliability, high compactness and high economy, the traditional design method, based on experience, trial and error and qualitative design can not meet the requirements. Even if the nominal stress and stress concentration coefficient are used to calculate the safety factor of the crankshaft wire position, there is considerable uncertainty. The common finite element method in the design calculation of a crankshaft is to calculate the deformation or stiffness of the crank when it is bent or twisted. It is also occasionally used in the calculation of continuous crankshaft beams to calculate support reaction, support bending moments, and nominal stresses on the crank.[4-5]

2 THE EQUIVALENT STRESS AND SAFETY FACTOR ARE DETERMINED BY ACS UR M53 STANDARD AND CCS

2.1 Calculation model

Based on the static bending model of a crankshaft, the nominal bending stres s is calculated. In stress calculation, the load is the resultant force of combusti on force and mass force, and the resul t of stress calculation is the nominal str ess value.

According to IACS UR M53 standard a nd CCS Code for Naturalization of Mari ne Steel Vessels, the stress concentrati on coefficient at the fillet radius of cran k pin and main shaft neck was calculat ed. The torsional stress is related to th e diameter of the crank pin and spindl e neck. The bending stress and compre ssion stress are related to the cross-se ctional area of the crank arm in the ov erlapping area of the crank pin. The str ess concentration coefficient of crank ar m fillet radius is calculated by finite ele ment method.[6-7]

The stress concentration coefficient of t he crankshaft fillet is calculated by finit e element method for the same cran k. Although the stress concentration coe fficient calculated by the empirical formula also meets the requirements of the standard design of oil holes, the calculation accuracy of the finite element method is higher.

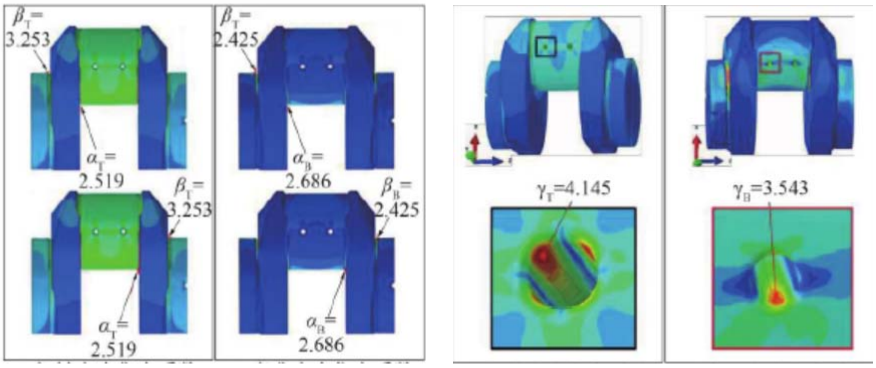

Figure1 a)Torsional stress concentration factor b)Bending stress concentration factor

The formula method usually used to calculate the stress concentration coefficient is based on the experimental measurement of the crank of various structural sizes and the empirical formula, its application range is limited by the structural size, such as the compound fillet is not specified in detail in the formula, at this time, the finite element method can be used to calculate the stress at the spindle neck and crank pin, and then get the corresponding stress concentration coefficient.

Unit load applied in all conditions.[8-10]

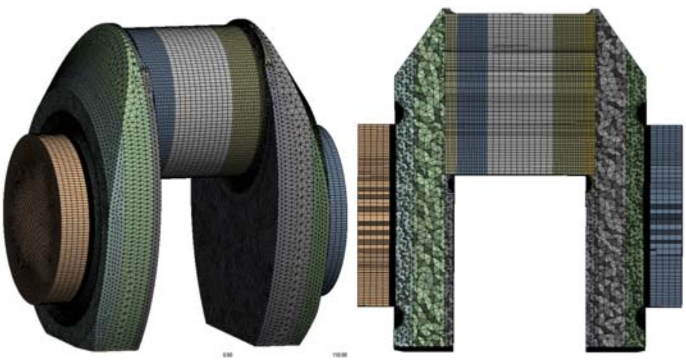

Figure 2 3D mesh

When calculating the stress concentration factor, four load conditions should be considered: torsion, pure bending, bending plus shear, and loading according to the working conditions in the code.

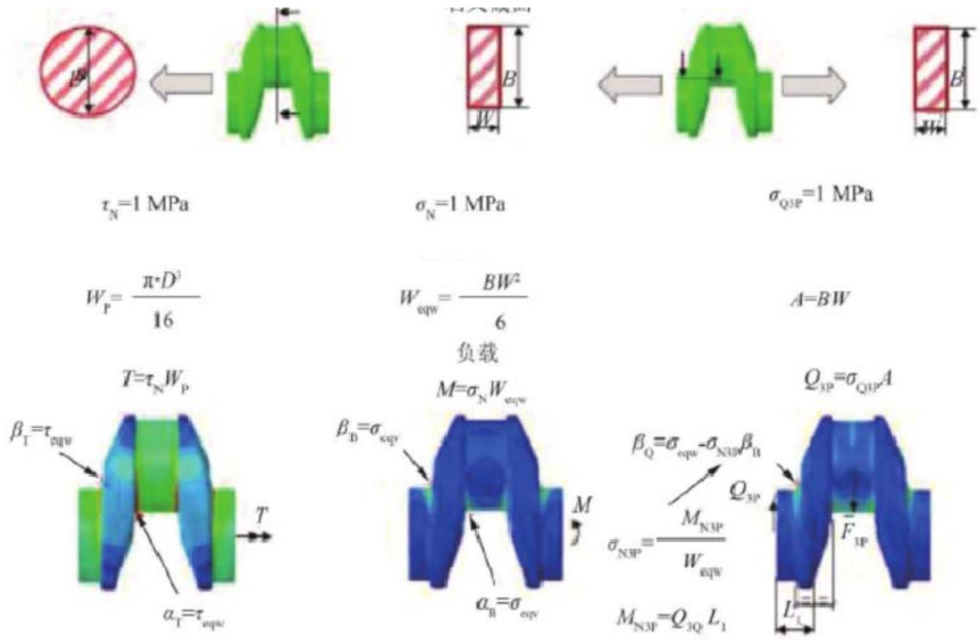

Figure 3 Conditions and formulas in the IM 53 standard

3 CALCULATION OF SHAFTING BY MULTI-BODY DYNAMICS

The analysis of crankshaft strength is mainly fatigue analysis.Through the simulation calculation of crankshaft system, the best scheme of different crankshaft schemes can be selected. In addition to the crankshaft strength to meet the requirements, but also comprehensively consider the crankshaft torsional vibration, spindle tile, connecting rod tile load.

3.1 Simulation model building

According to the design parameters, Creo software was used to carry out accurate three-dimensional design, and the completed crankshaft model was shown in the figure below.

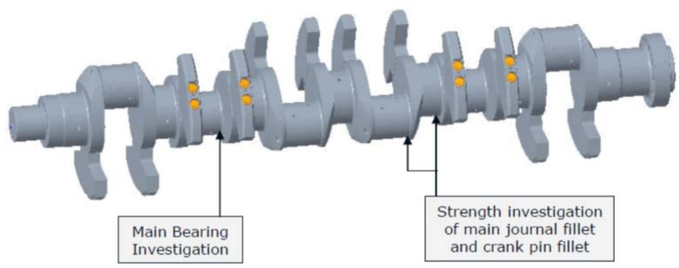

Figure 4 3D model of diesel engine crankshaft system

The data input in the software is implemented in a modular way to minimize the errors caused by human factors. The size and inertia data of the model are obtained by the modeling software, and the torsional stiffness of the unit crank is calculated by the finite element software. In the construction of the crankshaft model, the physical model is simplified appropriately.

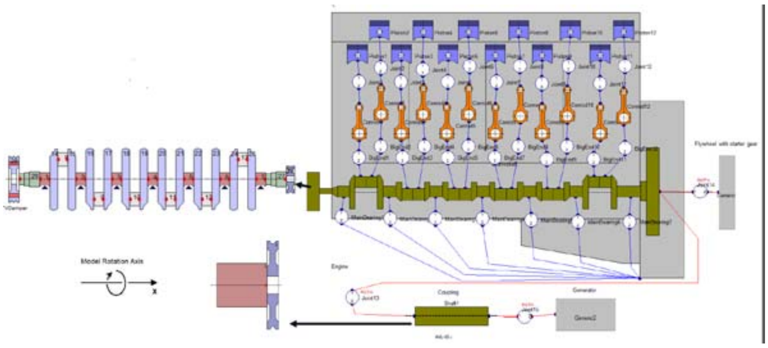

Figure 5 General crankshaft model

The nonlinear multi-body dynamics simulation model of crankshaft bearing system is created. First of all, the solid model was established in the three-dimensional model, the crankshaft and bearing seat were equivalent and simplified.The finite element software was imported for grid division, and then the structure was reduced.The simulation conditions and requirements were set for solving.

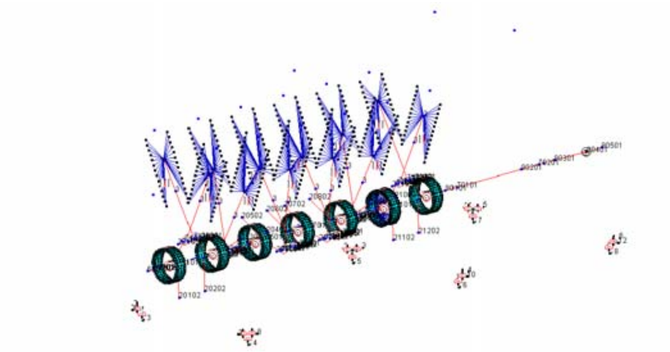

Figure 6 3D connection diagram of multi-body system

In the calculation of shafting dynamics, the main input load is the pressure curve in the cylinder, and the burst pressure change at each engine speed is input into the software.

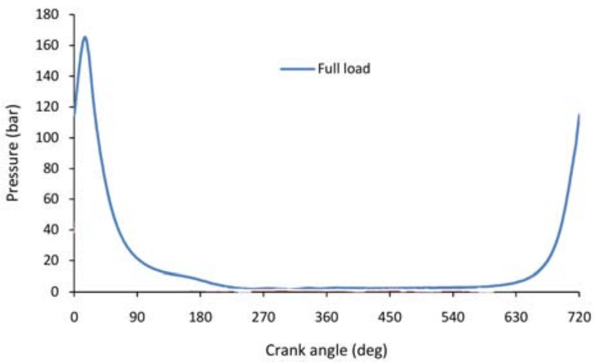

Figure7 Cylinder pressure

3.2 Calculation of crankshaft strength

The fatigue safety factor of each journal and rounded corner was calculated.

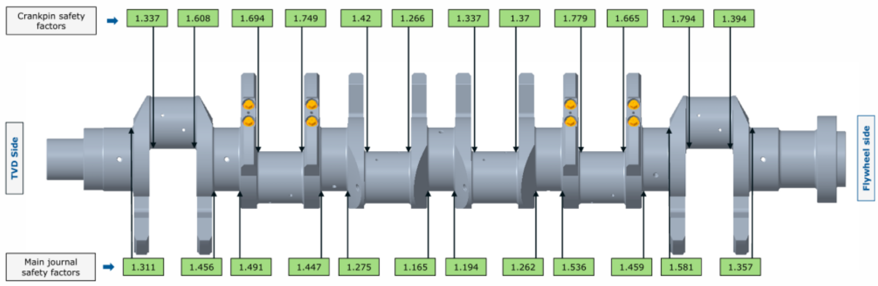

Figure 8 Minimum safety factors in crankpin and main journal fillets

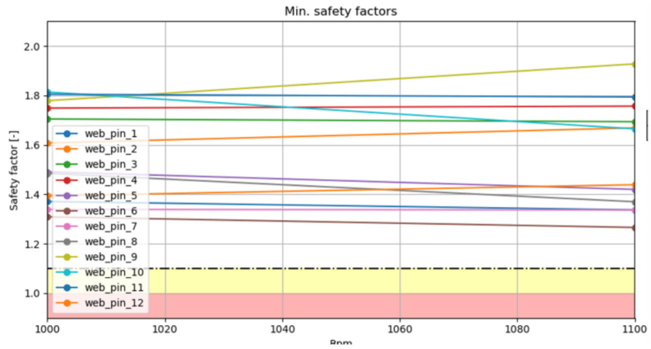

Figure9 Minimum safety factors in crankpin fillets

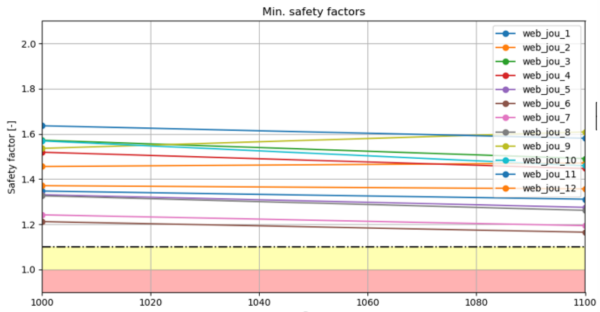

Figure10 Minimum safety factors in main journal fillets

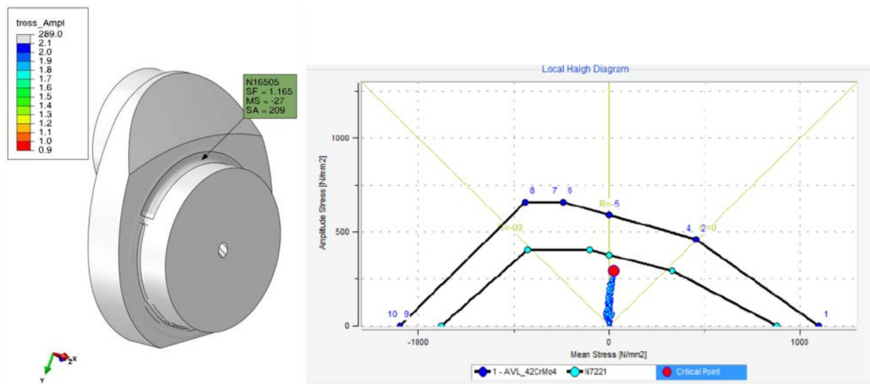

Figure11 Safety factor at the most critical point on the main journal side

Safety factor of 1.165 at the most critical point on the main journal side of web #6 is above the limit value of 1.1 and is therefore acceptable.

3.3 Bearing load analysis

Bearing bearing conditions depend on the load, the selection of lubricating oil and the bearing match, etc. Bearing calculation requires parameters such as effective width of bearing bush, bearing gap, oil gap, oil type, dynamic viscosity, oil supply mode, etc. The maximum, average and minimum values are selected in the calculation.

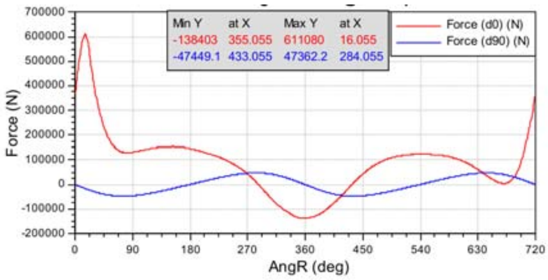

Figure12 Force applied on BE bearing at full load condition (1000 rpm)

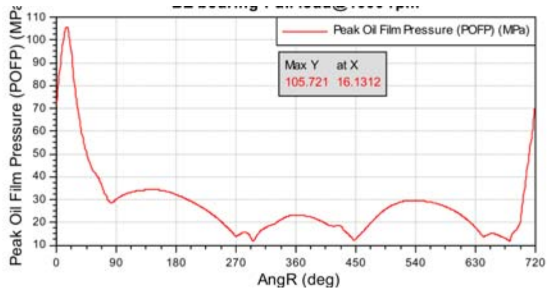

Figure13 Variation of peak oil film pressure (POFP) versus crank angle

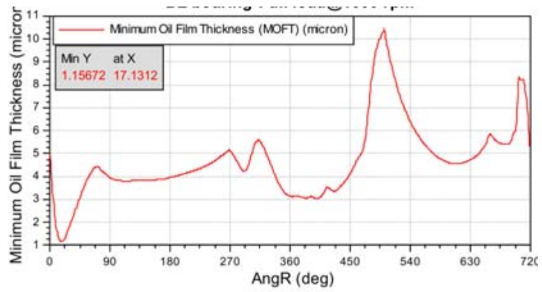

Figure14 Minimum oil film thickness (MOFT) as a function of crank Angle

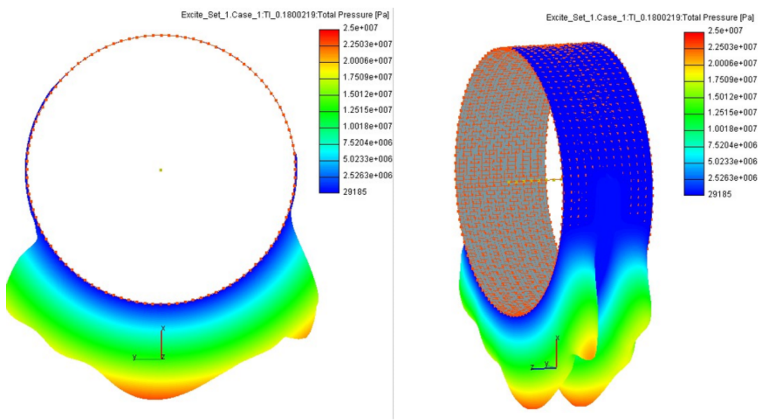

Figure15Distribution of oil film pressure due to inertia loading (exhaust TDC)

Figure 12 shows the force applied on the main bearing at full load condition (1000 rpm). Maximum compressive force is occurred at 16 degrees after firing TDC. Figure 13 plots the variation of peak oil film pressure (POFP) versus crank angle.Figure

14 shows the minimum oil film thickness (MOFT) at each crank angle. As can be seen, the overall MOFT is occurred at the edges of bearing shell. Therefore, the average MOFT across the width of shell is much higher than 1.15. Figure 15 shows the distribution of oil film pressure due to inertia loading (at exhaust TDC). Through the analysis of bearing load and force distribution, can more intuitive and in-depth understanding of bearing lubrication process, but also more helpful to analyze the force of the crankshaft.

3.4 Shafting torsional vibration

According to the calculation method of torsional vibration each part of crankshaft system is transformed into equivalent system.

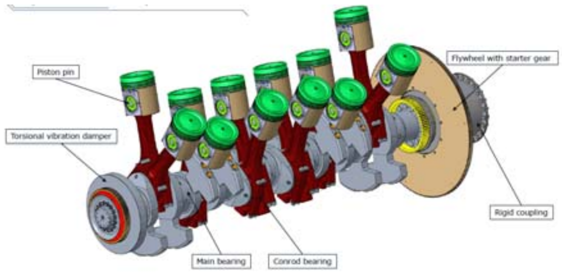

Figure16 Three-dimensional model of shafting

The finite element model verifies the simplified shafting model

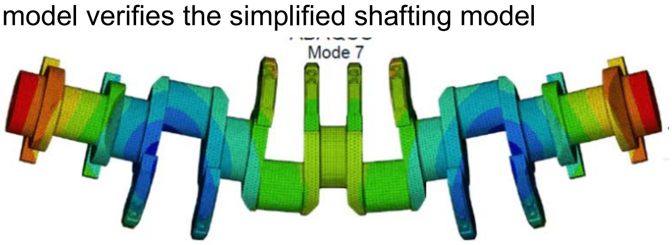

Figure17 The 7th order finite element model of shafting

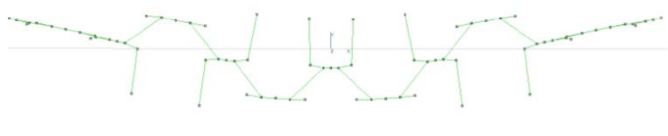

Figure18 Simplified model of 7th order shafting

The finite software was used to calculate the various modes of the crankshaft, and the results of the simplified model were compared. The following figure only shows the comparison results of the first order. The comparison is detailed in the following table, and the error is less than 5%, so the simplification is reasonable.

Table 1 Comparison of simplified model and finite element model of shafting

	Freq. [Hz]	Freq. [Hz]	Torsion [%]	Tension [%]	Bending [%]	Dominant	
7	55.60	56.20	0.1	1.5	98.5	bending	1.08%
8	62.72	63.27	0.1	0.9	99	bending	0.88%
9	132.1	132.8	7.9	2.7	89.4	bending	0.47%
10	152.0	153.6	0.3	1.6	98.2	bending	1.06%
11	159.8	159.5	31	2.9	66	bending	0.23%
12	182.4	185.0	0.2	61.8	38	tension	1.45%
13	232.0	234.8	0.5	6.6	92.9	bending	1.20%
14	249.4	254.9	0.5	59	40.6	tension	2.22%
15	272.6	268.6	53.6	12.8	33.6	torsion	1.45%
16	311.6	315.7	6.8	17.7	75.5	bending	1.32%
17	325.5	328.6	2.7	20	77.3	bending	0.96%
18	368.0	364.6	48.1	6.7	45.2	mixed	0.92%
19	394.1	394.2	16.6	7.3	76.1	bending	0.02%
20	396.1	399.7	7.3	14.9	77.7	bending	0.91%

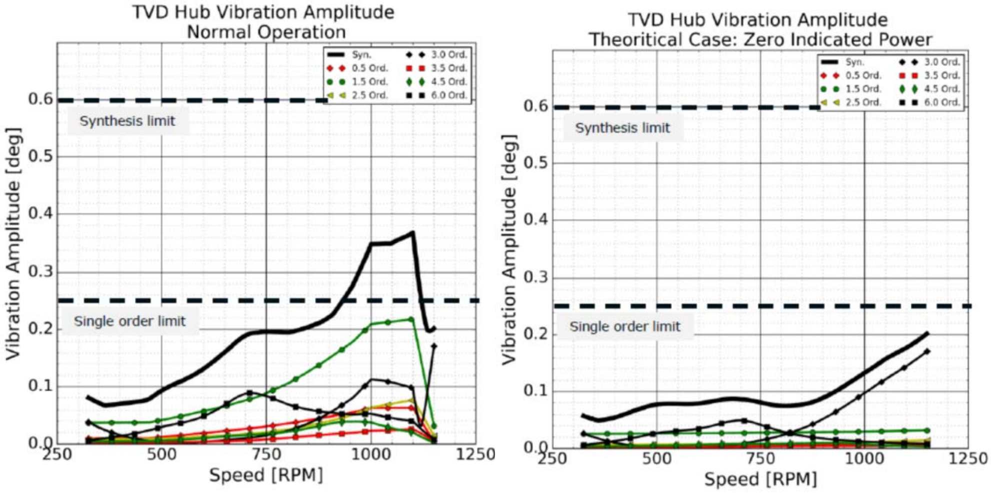

Figure19 TVD Hub Vibration Amplitude

Amplitude synthesis fulfills criteria. The synthesis limit: 0.5-0.6 deg. Single order amplitudes fulfill criteria. Single order limit :0.25 deg.

4 CONCLUSION

(1) Innovative verification of the method o f modeling of the system

By simplifying the complex axis system t o the equivalent system, and verifying th e high consistency of the natural frequen cy of the original system (as shown in ta ble 1), an efficient and reliable dynamic modeling method is proposed. This simpli fied method not only reduces the comput ational complexity, but also retains the dy namic characteristics of the original syste m, and provides the theoretical basis for the design of the follow-up diesel shaft s ystem. In the future, this method can be used to simulate and optimize the rapid simulation and optimization of the power system, such as ship and locomotive, es pecially in the multi-system integration an alysis.

(2) The combination of traditional norms and modern finite element method

Combined with the international classificat ion society (iacs) r m53 standard and the Chinese classification society (ccs) steel gauge, a method of calculating the equi valent stress and safety coefficient of finit e element method is proposed. This met hod has broken through the limitations of the traditional standard, and improves th e precision of the crankshaft strength by numerical simulation, and provides the sci entific support for the extreme design of crankshaft material. The future value is to establish the dual verification framework of "normative + simulation" for the strengt h evaluation of other power engines, and promote the evolution of industry standa rds in digital direction.

（3）The construction of multi-body dyna mics comprehensive analysis framework

For the first time, the dimensional dynami cs analysis of crankshaft strength, shaftin g, bearing load and other dimensional dy namics is integrated, and a complete set of dynamic characteristic evaluation syst em of diesel shaft system is formed. For example, through multi-body model, the nonlinear response of the crankshaft und er complex conditions makes the design more close to the material limit. Future r esearch can be based on the collaborative analysis of other key components (such as links, pistons), or combined with artificial intelligence optimization design parameters, and further improve the reliability and efficiency of diesel engines.

5 REFERENCE

[1] SHI Shao-xi , ZHANG De-qing.Diesel engine design manual .Bejing;China agricultural machinery publishing house ,1984.

[2] LONDHE A,YADAV V,MULEMANE A. multi-disciplinary approach for evaluating strength of engine cylinder head and crankcase assembly under thermo-structural loads [C]. SAE Paper 2009-01-0819,2009.

[3] Xiao Chong， Zuo Zhengxing， Qin Wenjie， et al.Coupling field analysis and application for cylinder head of diesel engine [J].Vehicle Engine ,2006 (4):26-29.(in chinese)

[4] CFD Online.Thermal phase change model :ANSYS help 13.0,CFX,theory guide,ANSYS,2009[EB/OL].[2012-02-20].

[5] XU J, YANG P, XU Y, et al. Simulation of upward subcooled boiling flow of refrigerant-113 using the two-fluid modelJ. Applied Thermal Engineering, 2009, 29(14): 2508-2517.

[6] DENG Bang-lin,LIU Jing-ping,YANG Jing.The Thermal Mechanical Fatigue Analysis on a cylinder Head [J] Journal of Hunan University (Natural Sciences),2012,39(2):30-34.

[7] XIE Mao zhao.Computational combustion science of IC engine [M].Dalian:Press of Dalian University of Technology ,2005:1 2 (In chinese).

[8] Liao Ridong， Zuo Zhengxing， Zou Wensheng.The effect of temperature field on stress distribution of cylinder head [J].Transactions of CSICE,2001,19(3):253-257.(in chinese)

[9] Zhong,H.Z., Zheng , C.S.,1994,

”Relationship between fatigue notch factor and strength”,J.Engineering Fracture Mechanics,48,127-136.

[10] Graff, WJ, "Thermal Conductance across Metal Joints, Machine Design, Vol. 32, pp. 166-172,1960.