振动摇摆复合载荷下船用柴油机附件的疲劳强度分析

Investigation on Fatigue Reliability of Ship Engine Accessories under Vibration and Swing Load

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一、研究背景及意义

Research Background and Significance

船用柴油机附件（如水泵、启动电机等）在使用过程中，不仅要受到温度、湿度等环境因素的影响，还要同时承受船体随海面波浪摆动带来的**摇摆载荷**，以及发动机本身运行产生的**振动载荷**。

Marine diesel engine accessories (such as water pump, starting motor, etc.) are not only affected by temperature, humidity and other environmental factors during use, but also bearing the **swinging load** brought by the oscillation of the hull along with the sea wave, as well as the **vibration load** generated by the engine itself.

一、研究背景及意义 Research Background and Significance

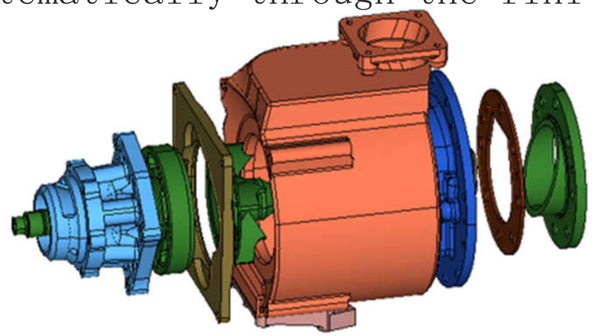
目前，已出现船用柴油机海水泵、淡水泵、压缩机等产品在**陆上台架试验**时，各项指标均满足使用要求，但在**装船使用不久便故障频发**的情况。究其原因主要是，该类船用发动机附件在陆上试验中**未综合施加振动、摇摆载荷**，导致试验过程中附件的**应力分布、失效机理**与其实际工作条件下的不一致，即陆上台架试验过程中没有充分暴露和验证附件在真实运行条件下的**薄弱环节和可靠性**。

At present, Marine diesel engine accessories such as sea water pump, fresh water pump, compressor and other products have been tested on the **land bench**, all the indicators meet the requirements of use, but **soon after use on the ship failure frequently occurred**. The main reason is that this kind of Marine engine accessories are **not comprehensively applied vibration and swinging loads** in the land test, resulting in the inconsistency between the **stress distribution or failure mechanism** of the accessories and their actual working conditions, that is, **the weak parts and reliability** of the accessories under real operating conditions are not fully exposed and verified in the land bench test.

一、研究背景及意义 Research Background and Significance

为研究振动、摇摆载荷的复合作用对附件疲劳强度的影响，以**某型号海水泵**为对象，测量其实际工况下**激励响应参数**，并通过有限元分析的方法，对该水泵在振动、摇摆载荷单独及复合作用下的**应力分布及疲劳强度**进行系统分析。

In order to study the influence of the combined action of vibration and swinging load on the fatigue strength of accessories, **a certain type of sea water pump** was taken as the object, the excitation response parameters were measured under the actual working conditions, and the **stress distribution and fatigue strength** of the pump under the separate and combined action of vibration and swinging load were analyzed systematically through the finite element analysis method.

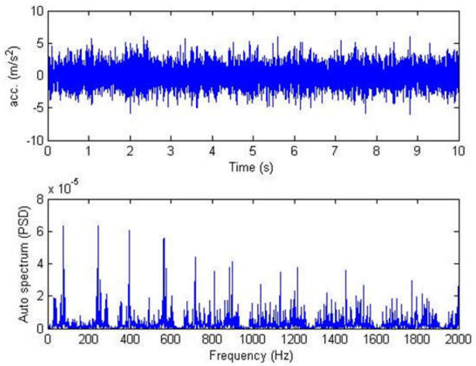


二、振动、摇摆载荷测试及分析

Vibration and Swinging Load Test and Analysis

二、振动、摇摆载荷测试及分析 Vibration and Swinging Load Test and Analysis

● 海水泵振动加速度测试 Vibration acceleration test of sea water pump



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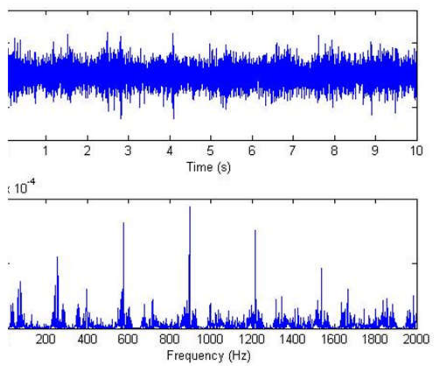

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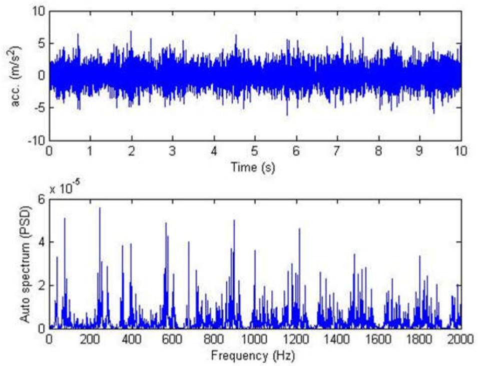

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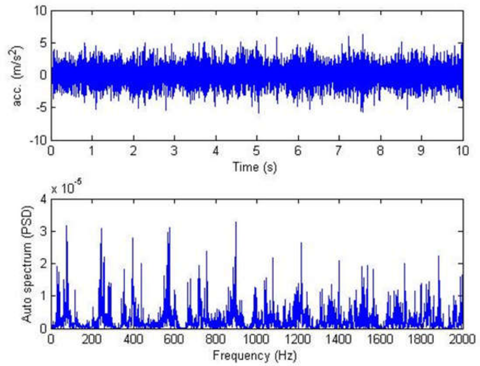

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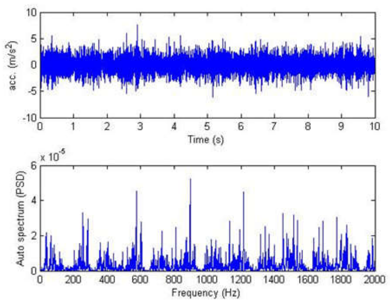

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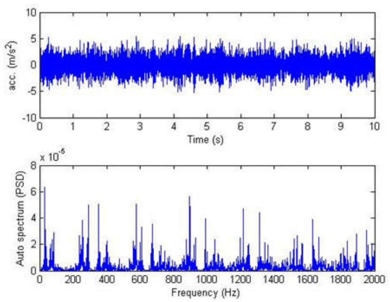

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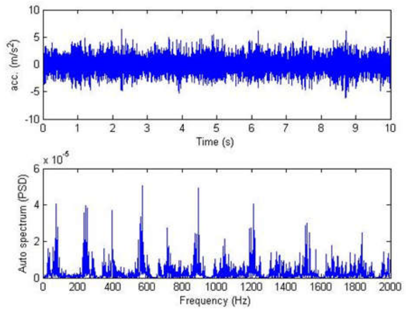

测点5Z (P 5Z)

二、振动、摇摆载荷测试及分析 Vibration and Swinging Load Test and Analysis

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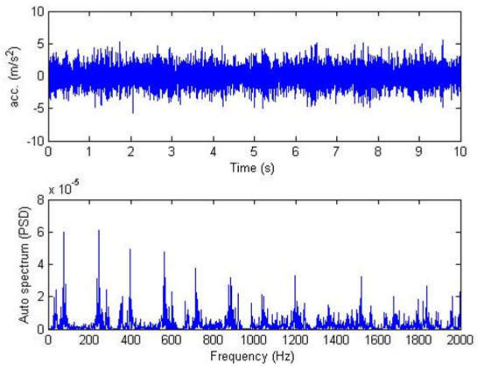

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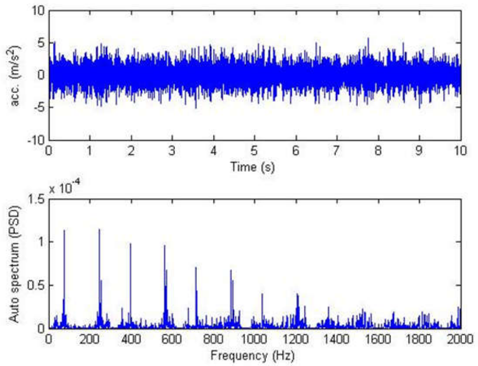

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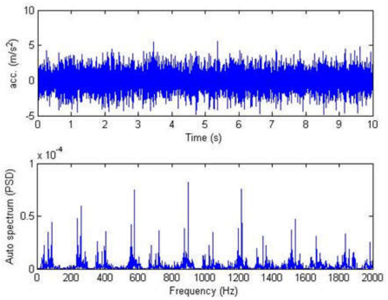

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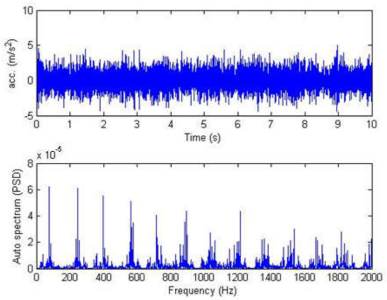

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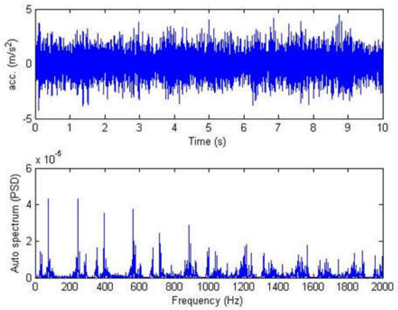
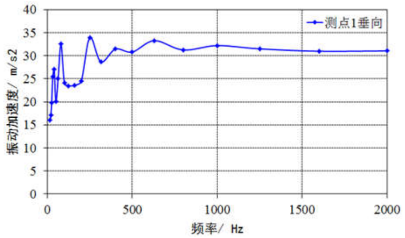


测点8Z (P 8Z)

● 海水泵振动加速度测试 Vibration acceleration test of sea water pump

海水泵各测点的振动加速度三分之一倍频程曲线图

One third octave curve of vibration acceleration of each measure point of sea water pump



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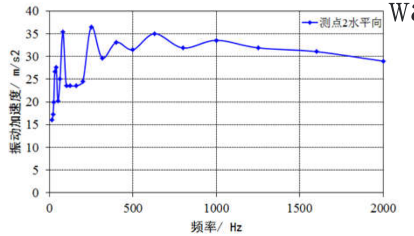

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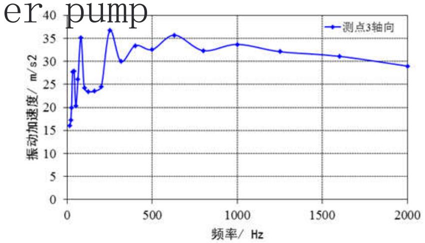

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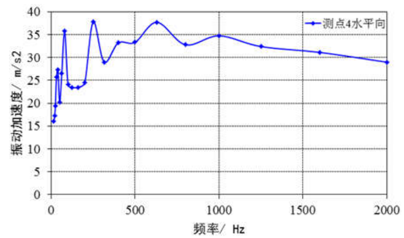

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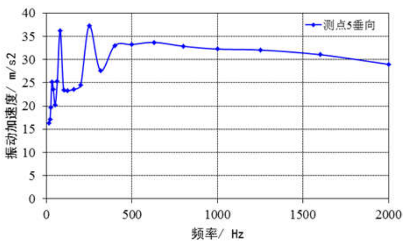
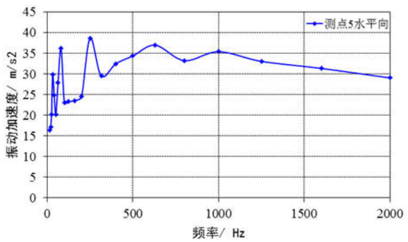


测点5X (P 5X)



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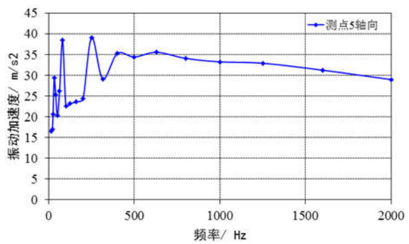

测点5Z (P 5Z)

二、振动、摇摆载荷测试及分析 Vibration and Swinging Load Test and Analysis

● 海水泵振动加速度测试 Vibration acceleration test of sea water pump

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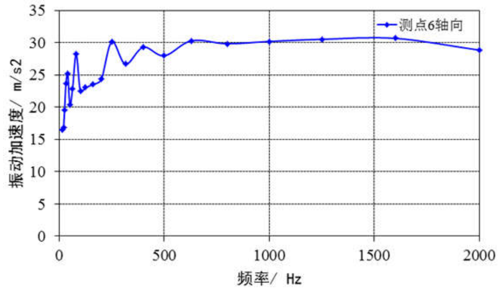

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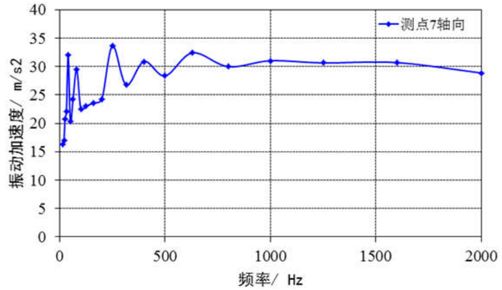

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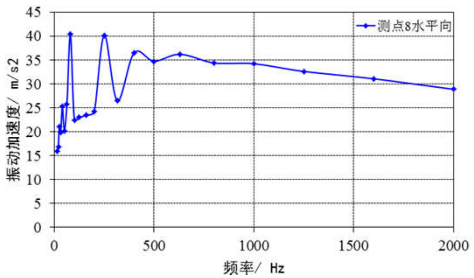
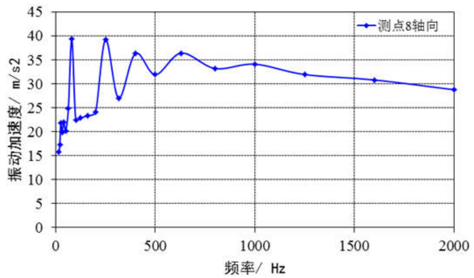


测点8X (P 8X)



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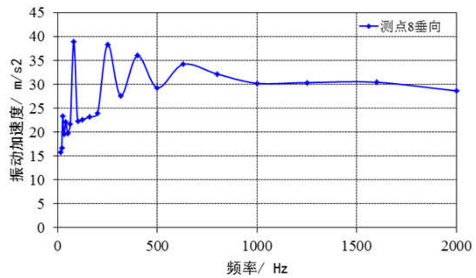


测点8Z (P 8Z)

● 海水泵振动加速度测试 Vibration acceleration test of sea water pump

根据测试结果，水泵的**振动能量主要集中在1000Hz以下**，最大振动加速度为40.2m/s²，为8号测点的水平方向。

According to the test results, the vibration energy of the pump is **mainly concentrated below 1000Hz**, and the maximum vibration acceleration is 40.2m/s², which is in the horizontal (X) direction of P8.

测点/Measure point	测试值/Test value (m/s²)		
	水平/X	轴向/Y	垂向/Z
测点1/P1	-	-	33.9
测点2/P2	36.5	-	-
测点3/P3	-	36.7	-
测点4/P4	37.8	-	-
测点5/P5	38.7	39.1	37.2
测点6/P6	-	30.2	-
测点7/P7	-	33.6	-
测点8/P8	40.2	39.2	38.3
最大值/Maximum	40.2	39.2	38.3

各测点振动加速度看，水泵振动最严重的位置出现在8号测点，X方向为**40.2 m/s²**，Y向为**39.2 m/s²**，Z向为**38.3 m/s²**。以**1.5倍**的8号测点振动加速度为激励源，得到水泵加速度激励幅值。

According to the vibration acceleration of each measuring point, the most serious vibration of the water pump occurred at P8 (X: 40.2m /s² ; Y: 39.2m /s² ;Z: 38.3m /s²). Taking 1.5 times the vibration acceleration of P8 as the excitation source, the pump acceleration amplitude was obtained.

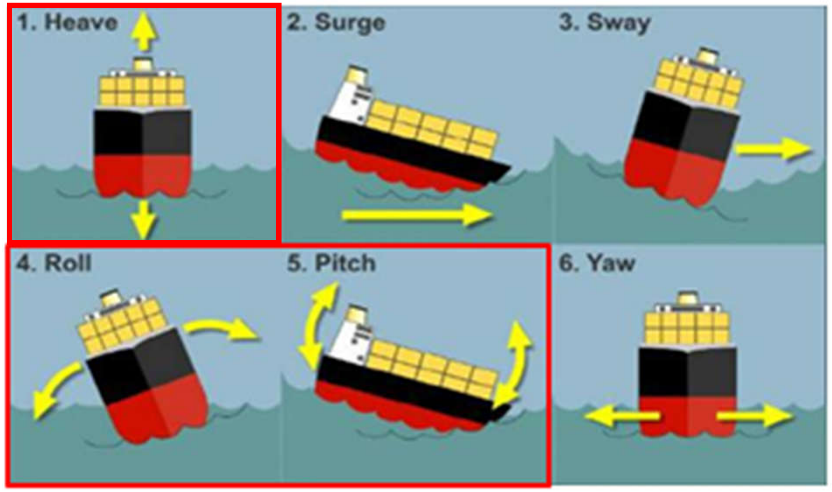
×1.5

振动激励/Vibration	测试值/Test value (m/s²)		
	水平/X	轴向/Y	垂向/Z
	60.3	58.8	57.5

● 海水泵摇摆边界条件确定 Determination of swing boundary condition of sea water pump

船舶在航行时，受到海浪的冲击时会发生船舶摇摆问题。船舶摇摆时会对安装在船舶内的部件产生周期激励，影响各零部件的工作状态。对此进行摇摆载荷的分析，确定海水泵在船舶摇摆时的受力边界。

When a ship is sailing, it will be swayed by waves. When the ship is swaying, the parts installed in the ship will be stimulated periodically, affecting the working condition of each part. Based on the analysis of swinging load, the force boundary of sea water pump is determined.

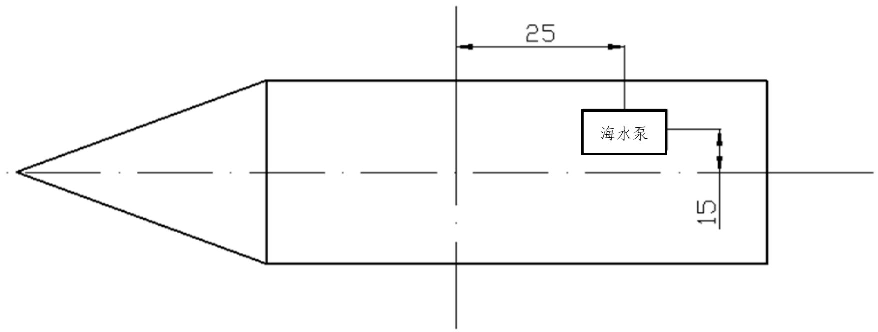


舰船摆动示意图
Ship swing diagram

● 海水泵摇摆边界条件确定 Determination of swing boundary condition of sea water pump

GJB4000-2000中规定，船舶横摇最大工况：**±45º（周期6-11S）**；船舶纵摇最大工况：**±10º（周期7-8S）**；船舶垂荡最大工况：**加速度0.1g，周期5s**。海水泵安装在法兰端面及支架螺栓孔处，在船体内部的安装位置如图所示：

In GJB4000-2000, the ship's maximum rolling condition: ±45° (period 6-11S); Maximum pitching condition: ±10° (period 7-8S); Maximum heave condition: acceleration 0.1g, period 5s. The sea water pump is installed at the face of the flange and the bolt hole of the support. The installation position in the hull is shown in the figure:



海水泵安装位置示意图
Installation position diagram of sea water pump

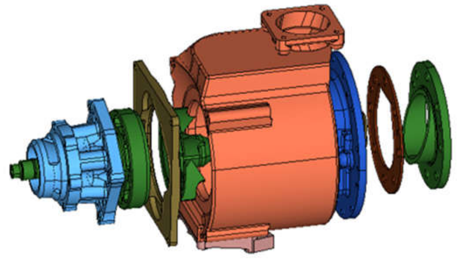
三、海水泵应力特性分析

Stress Characteristics Analysis of Sea Water Pump

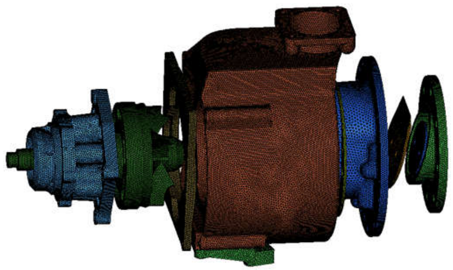
（1）海水泵模态分析 Modal analysis of sea water pump

水泵各部件是通过螺栓连接装配在一起的，各连接面需要紧密贴合，以保证水泵工作过程中不发生漏水或漏油的现象。在水泵有限元模型装配时，各接触面采用绑定约束（tie约束）完成装配关系的建模。

The components of the water pump are assembled together by bolted connection, and the connecting surface needs to be closely fitted to ensure that water leakage or oil leakage does not occur during the working process of the water pump. In the finite element model assembly of the pump, the binding constraint (tie) is used to complete the assembly modeling of each contact surface.



水泵CAD模型
CAD model of water pump

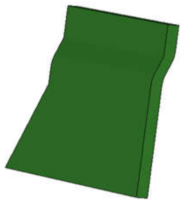


水泵有限元模型
Finite element model of water pump

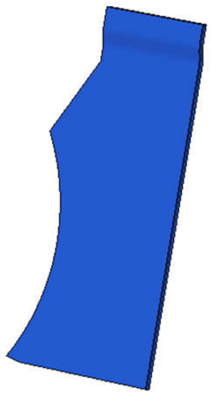
（1）海水泵模态分析 Modal analysis of sea water pump

在对水泵安装支架建模时，采用两个支架方案，用以对比支架长度对水泵动态性能的影响。支架1长度为187mm，支架2长度为450mm。

Two support schemes are used to compare the influence of the length of support on the dynamic performance of the pump. The length of support 1 is 187mm and support 2 is 450mm.



安装支架1（187mm）



安装支架2（450mm）

support 1 (187mm)

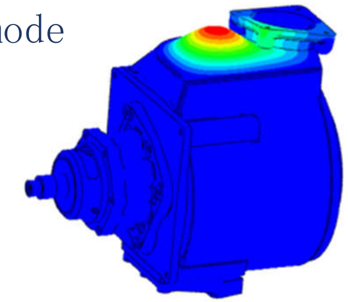
support 2 (450mm)

三、海水泵应力特性分析 Stress Characteristics Analysis of Sea Water Pump

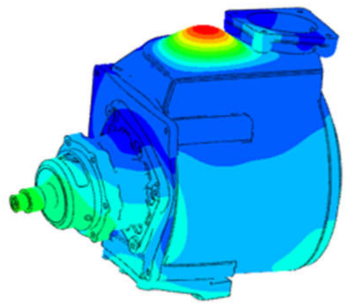
（1）海水泵模态分析结果——自由模态

Modal analysis results : Free mode

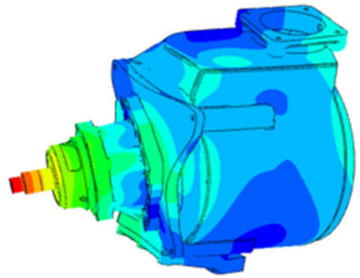
模态阶数 /Modal order	模态频率(Hz) /modal frequency
1	725.7
2	749.6
3	847.6
4	952.4
5	1011.6
6	1525.7



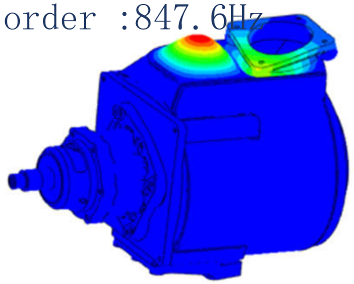
第一阶模态725.7Hz
First-order :725.7Hz



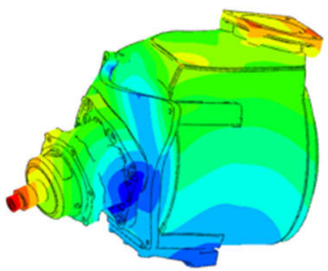
第二阶模态749.6Hz
Second-order :749.6Hz



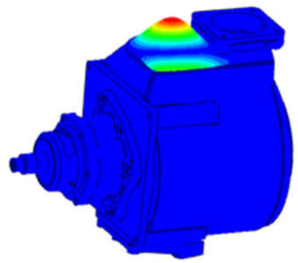
第三阶模态847.6Hz
Third-order :847.6Hz



第四阶模态952.4Hz
Fourth-order :952.4Hz



第五阶模态1011.6Hz
Fifth-order :1011.6Hz

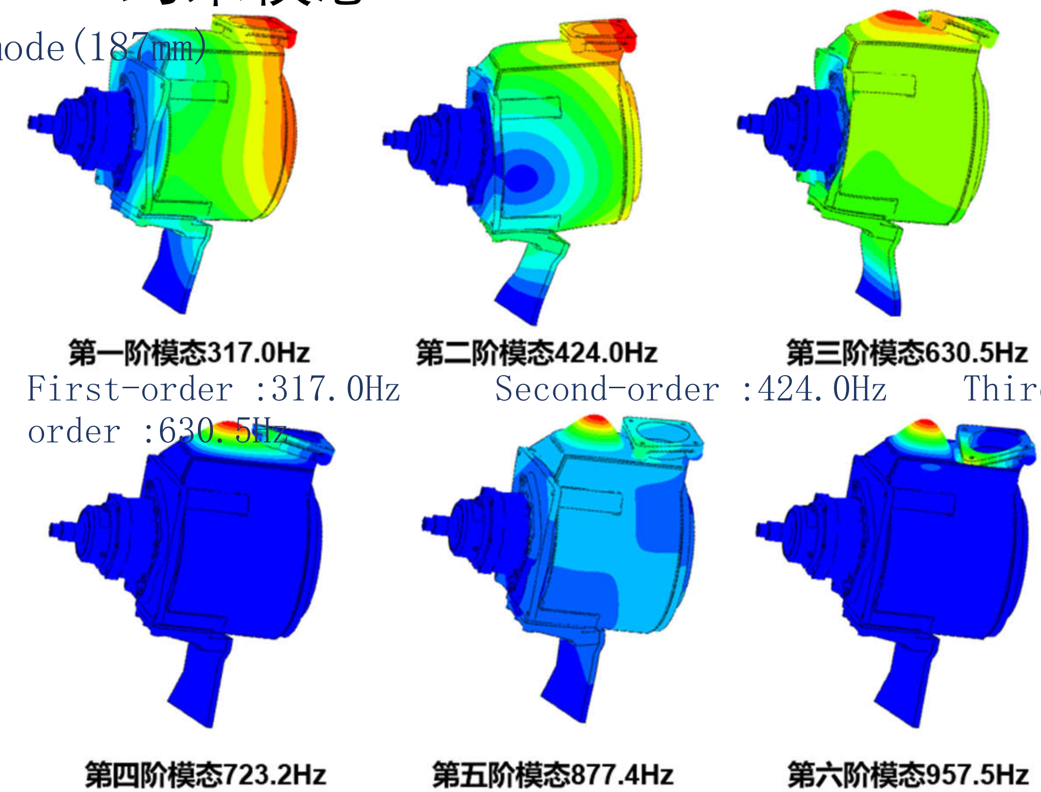


第六阶模态1525.7Hz
Sixth-order :1525.7Hz

(1) 海水泵模态分析结果——187mm约束模态

Modal analysis results : Constrained mode(187mm)

模态阶数 /Modal order	模态频率(Hz) /modal frequency
1	317.0
2	424.0
3	630.5
4	723.2
5	877.4
6	957.5



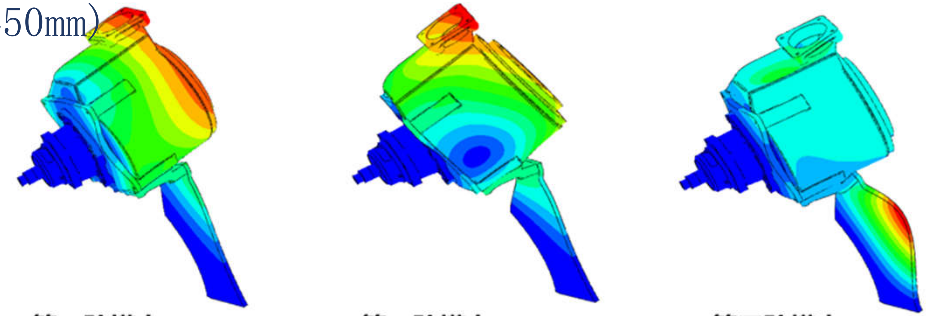
First-order :317.0Hz Second-order :424.0Hz Third-order :630.5Hz

Fourth-order :723.2Hz Fifth-order :877.4Hz Sixth-order :957.5Hz

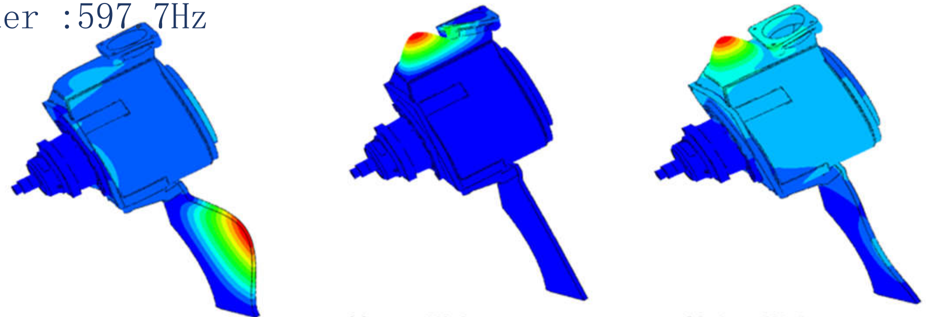
(1) 海水泵模态分析结果——450mm约束模态

Modal analysis results : Constrained mode(450mm)

模态阶数 /Modal order	模态频率(Hz) /modal frequency
1	308.1
2	420.6
3	597.7
4	667.2
5	723.1
6	858.3



第一阶模态308.1Hz 第二阶模态420.6Hz 第三阶模态597.7Hz
First-order :308.1Hz Second-order :420.6Hz Third-order :597.7Hz



第四阶模态667.2Hz 第五阶模态723.1Hz 第六阶模态858.3Hz
Fourth-order :667.2Hz Fifth-order :723.1Hz Sixth-order :858.3Hz

（1）海水泵模态分析结论

Modal analysis results

- 水泵总成的一阶**自由模态频率725Hz**；安装支架后，水泵的**一阶约束模态为308Hz**，远离发动机主要激励频率，具有较高的抗振性能；

  The first-order **free mode frequency** of the pump assembly is **725Hz**; After the bracket is installed, the **first-order constraint mode** of the water pump is **308Hz**, which is far away from the main excitation frequency of the engine, and has high vibration resistance performance;

- 水泵支架长度增大后，水泵约束模态有所降低；

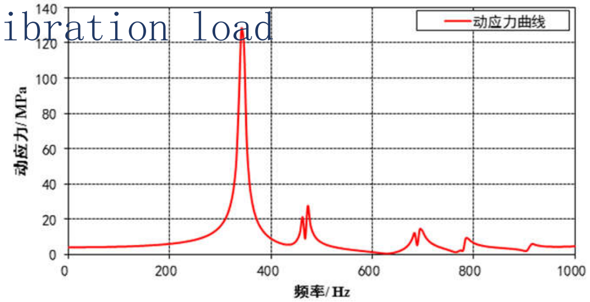
  With the increase of the length of the pump support, the constrained mode of the pump decreases;

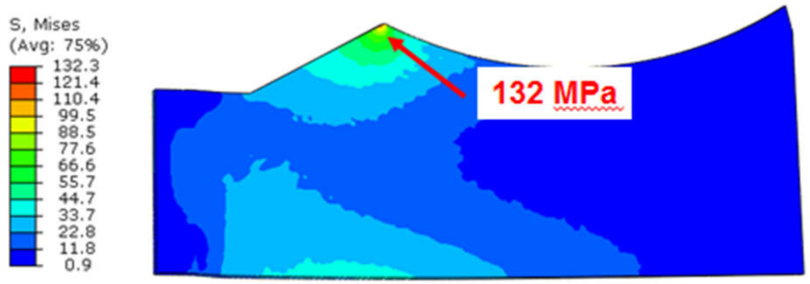
- 后续分析将采用模态频率较低的450mm支架装配方案，用以检验海水泵的可靠性设计。

  Subsequent analysis will use a 450mm bracket assembly scheme with lower modal frequency to test the reliability design of the sea water pump.

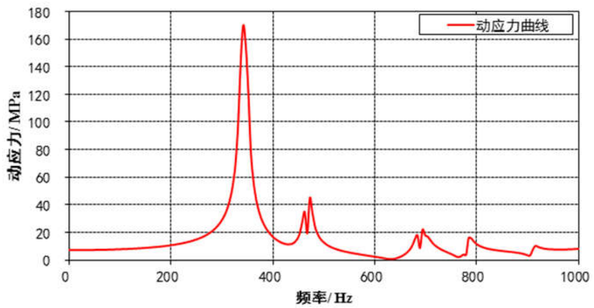
（2）海水泵振动载荷下的有限元分析

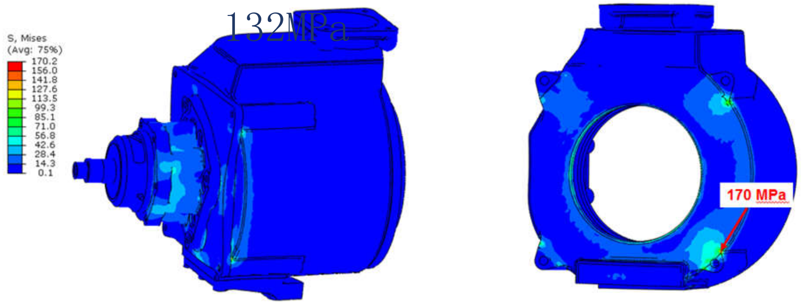
Finite element analysis of sea water pump under vibration load





水泵支架动应力计算结果：最大应力为132MPa Calculation results of dynamic stress: Maximum 132MPa





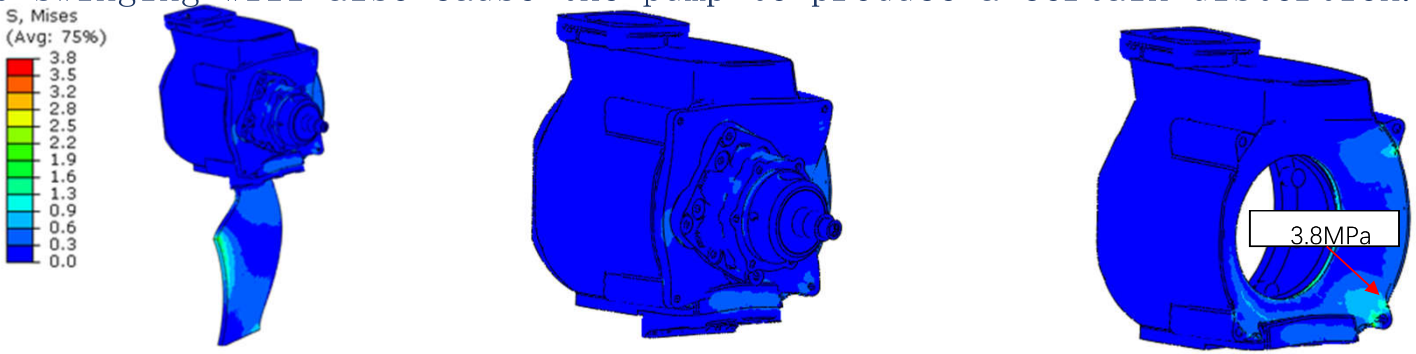
水泵壳体动应力计算结果：最大应力为170MPa Calculation results of dynamic stress: Maximum 170MPa

（3）海水泵摇摆载荷下的有限元分析（纵摇）

Finite Element Analysis of Sea Water Pump under Swing Load (Pitch)

在船舶纵摇时，摇摆轴线为X轴；根据船舶尺寸及摇摆周期，确定摇摆半径为25米，则纵摇角速度为0.04987 rad/s。此时水泵的最大应力为3.8MPa，最大应力的位置在水泵壳体的螺栓连接处。同时，船舶摇摆后，也会导致水泵产生一定的扭曲变形。

When the ship is pitching, the axis of swing is the X-axis; According to the ship's size and swing period, the swing radius is determined to be 25 meters, and the angular velocity of pitching is 0.04987 rad/s. At this time, the maximum stress of the water pump is 3.8MPa, and the maximum stress is located at the bolt connection of the water pump housing. At the same time, the ship swinging will also cause the pump to produce a certain distortion.

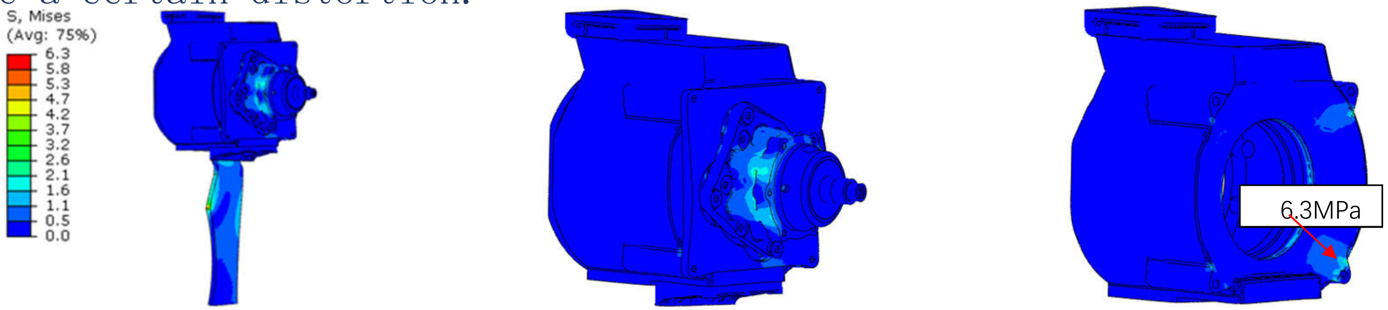


（3）海水泵摇摆载荷下的有限元分析（横摇）

Finite Element Analysis of Sea Water Pump under Swing Load (Roll)

在船舶横摇时，摇摆轴线为Z轴；根据船舶尺寸及摇摆周期，确定摇摆半径为15米，则横摇角速度为0.2618 rad/s。此时水泵的最大应力为6.3MPa；最大应力的位置与纵摇时最大应力位置一致，都在水泵壳体的螺栓连接处。同时，船舶摇摆后，也会导致水泵产生一定的扭曲变形。

When the ship rolls, the axis of swing is the Z axis; According to the size of the ship and the swing period, the swing radius is determined to be 15 meters, and the rolling angular velocity is 0.2618rad /s. At this time, the maximum stress of the pump is 6.3MPa; The position is the same as the position of the maximum stress during pitching, both at the bolt joint of the pump housing. At the same time, the ship swinging will also cause the pump to produce a certain distortion.



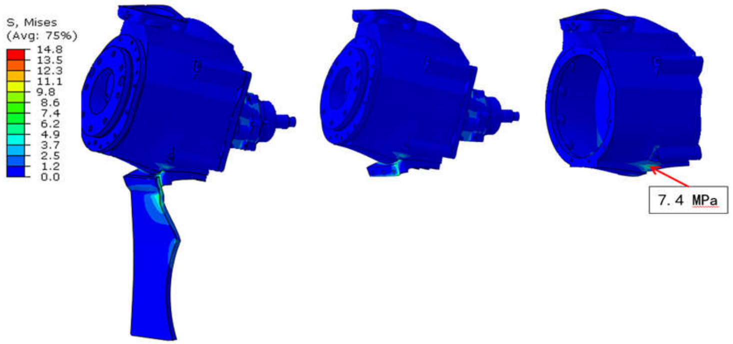
三、海水泵应力特性分析 Stress Characteristics Analysis of Sea Water Pump

（3）海水泵摇摆载荷下的有限元分析（垂荡）

Finite Element Analysis of Sea Water Pump under Swing Load (Heave)

当船舶受到垂荡载荷（竖直方向的低频振动，周期为5s，加速度0.1g）时，水泵总成所产生应力为14.8MPa，水泵壳体应力为7.4MPa，小于铸铝材料抗拉强度极限（>200MPa）；水泵支架处的最大应力为14.3MPa，小于钢的抗拉强度极限（>300MPa），无断裂风险。

When the ship is subjected to heave load (low-frequency vibration in the vertical direction, period 5s, acceleration 0.1g), the stress generated by the pump assembly is 14.8MPa, and the stress of the pump housing is 7.4MPa, which is less than the tensile strength limit of cast aluminum material (>200MPa). The maximum stress at the pump support is 14.3MPa, smaller than the tensile strength limit of steel (>300MPa), and there is no risk of fracture.



（4）海水泵振动、摇摆载荷下的有限元分析

Finite Element Analysis of Sea Water Pump under Swing and Vibration Load

力学边界:

1、船舶横摇工况：±45º（周期6—11S）；船舶纵摇工况：±10º（周期7—8S）；

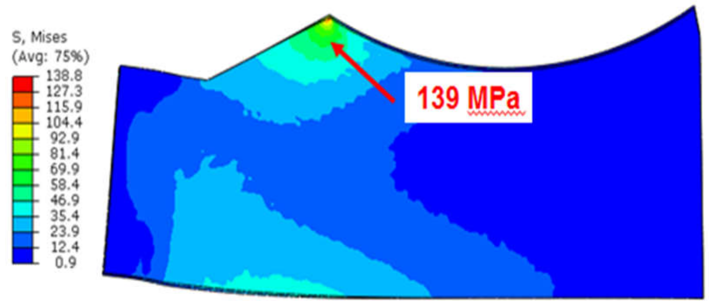
2、发动机振动激励：同时施加X方向6g、Y方向5.8g、Z方向5.7g，分析频率0-1000Hz。

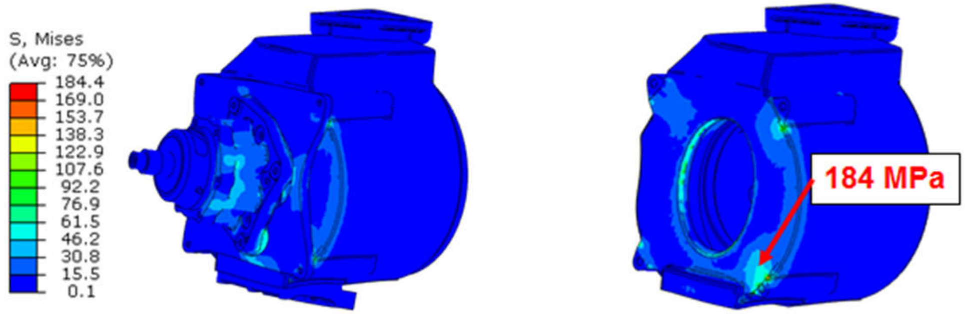
结果如下所示:

1. Ship rolling condition: ±45° (period 6-11s); Ship pitching condition: ±10° (period 7-8s);

2. Engine vibration excitation: 6g in X direction, 5.8g in Y direction and 5.7g in Z direction are applied at the same time, and the analysis frequency is 0-1000Hz.

The results are as follows:





四、海水泵疲劳寿命分析

Fatigue Life Analysis of Sea Water Pump

（1）确定疲劳计算模型：以剪切力为主的多轴疲劳计算模型

Determine the fatigue calculation model: multi-axial fatigue calculation model with shear force as the main factor

传统的应力-应变疲劳公式如下所示:

The traditional stress-strain fatigue formula is shown as follows:

$$\frac{\Delta\varepsilon}{2}=\frac{\sigma'_f}{E}(2N_f)^b+\varepsilon'_f(2N_f)^c \quad (1)$$

式中，ε为正应变，σ'_f为材料的强度极限，E为材料的弹性模量，N_f为循环次数，ε'_f为材料的延性系数，b为强度指数，c为延展性指数。

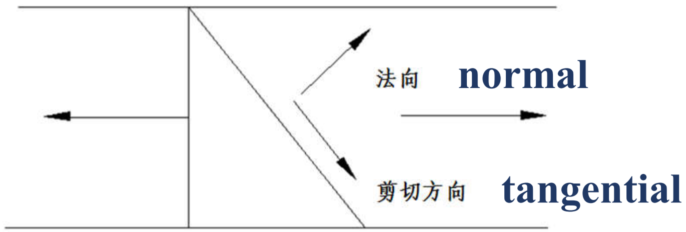
Where, ε is the positive strain, σ'_f is the strength limit of the material, E is the elastic modulus of the material, N_f is the number of cycles, ε'_f is the ductility coefficient of the material, b is the strength index, and c is the ductility index.

（1）确定疲劳计算模型：以剪切力为主的多轴疲劳计算模型

Determine the fatigue calculation model: multi-axial fatigue calculation model with shear force as the main factor

利用Brown和Miller的疲劳破坏模型，认为疲劳裂纹首先发生在承受最大剪应变的平面内，而疲劳破坏则是由最大剪应变和最大法向应力共同作用的结果，材料损伤示意图如下所示：

Based on the fatigue failure model of Brown and Miller, it is considered that the fatigue crack first occurs in the plane subjected to the maximum shear strain, while the fatigue failure is the result of the combined action of the maximum shear strain and the maximum normal stress. The damage diagram of the material is shown as follows:



（1）确定疲劳计算模型：以剪切力为主的多轴疲劳计算模型

Determine the fatigue calculation model: multi-axial fatigue calculation model with shear force as the main factor

法向应变与最大剪应变的公式可以如下表述：

The formula of normal strain and maximum shear strain can be expressed as follows:

$$\frac{\gamma_{max}}{2}=\frac{\varepsilon_1-\varepsilon_3}{2},\quad \varepsilon_n=\frac{\varepsilon_1+\varepsilon_3}{2} \tag{2}$$

式中，γ_{max}为最大剪应变，ε_n为法向应变，ε_1，ε_2，ε_3为三个方向上的应变分量。

Where, γ_{max} is the maximum shear strain, ε_n is the normal strain, and ε_1, ε_2, ε_3 are the strain components in three directions.

对于结构表面应力为平面应力，引入比例常数ν，则有：

If the surface stress of the structure is plane stress, the proportionality constant ν is introduced, then:

$$\gamma_{max}=(1+\nu)\varepsilon_1,\quad \varepsilon_n=\frac{(1-\nu)\varepsilon_1}{2} \tag{3}$$

（1）确定疲劳计算模型：以剪切力为主的多轴疲劳计算模型

Determine the fatigue calculation model: multi-axial fatigue calculation model with shear force as the main factor

考虑平均应力修正后，可将传统的应力应变方程变为：

After considering the average stress modification, the traditional stress-strain equation can be changed into:

$$\frac{\Delta\gamma_{\max}}{2}+\frac{\Delta\varepsilon_n}{2}=C_1\frac{\sigma_f-\sigma_m}{E}\left(2N_f\right)^b+C_2\varepsilon_f\left(2N_f\right)^c \qquad (4)$$

疲劳计算基本流程：**首先开展结构构件的动态应力计算；以动态应力计算结果为基础，提取动态应力历程；采用公式（4）计算零部件各节点处的疲劳寿命**。

The basic process of fatigue calculation: Firstly, the dynamic stress calculation of structural parts is carried out; Based on the calculation results of dynamic stress, the dynamic stress history was extracted. Formula (4) was used to calculate the fatigue life at each node of the parts.

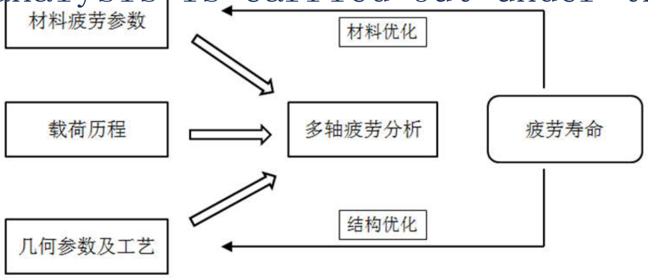
四、海水泵疲劳寿命分析 Fatigue Life Analysis of Sea Water Pump

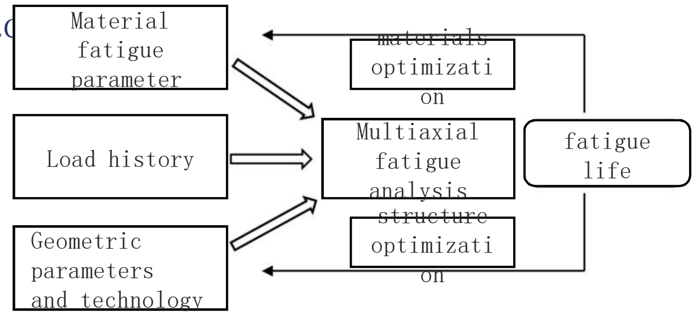
（2）船用海水泵多轴疲劳损伤累积预测方法

Cumulative fatigue damage prediction method for Marine sea water pump

多轴疲劳计算是一种能够有效预测结构件在工作状态下损伤累积的计算方法，对于疲劳仿真流程来说，除了CAE有限元模型外，还需要另外两个内容的输入：一是材料的疲劳性能参数；二是结构应力-应变的时间历程。然后**开展摇摆、振动以及两种载荷耦合作用下**的损伤累积分析。

Multi-axis fatigue calculation is a calculation method that can effectively predict the damage accumulation of structural parts under working conditions. For the fatigue simulation process, besides CAE finite element model, two other inputs are needed: first, fatigue performance parameters of materials; second, the time history of structural stress - strain. Then the damage accumulation analysis is carried out under the ac coupling load.





(3) 海水泵摇摆损伤累积预测

Swing damage accumulation prediction of sea water pump

船舶摇摆状态下，海水泵将会受到一个来自船体的摇摆激励，该激励将会影响海水泵的可靠性及耐久性；因此需要开展船舶摇摆工况下水泵的可靠性分析。首先开展海水泵应力计算，分别计算海水泵纵摇与横摇状态下水泵的应力分布状态。

When the ship is swinging, the sea water pump will receive a swing excitation from the hull, which will affect the reliability and durability of the sea water pump. Therefore, it is necessary to carry out reliability analysis of water pump under ship swinging condition. Firstly, the stress calculation of sea water pump is carried out, and the stress distribution of sea water pump in pitch and roll state is calculated respectively.

船舶纵摇状态下水泵应力计算模型
Calculation model of pump stress in ship pitching

船舶横摇状态下水泵应力计算模型
Calculation model of pump stress in ship roll

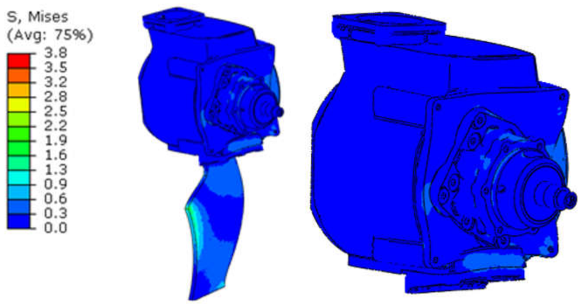
浙江大学

(3) 海水泵摇摆损伤累积预测

Swing damage accumulation prediction of sea water pump

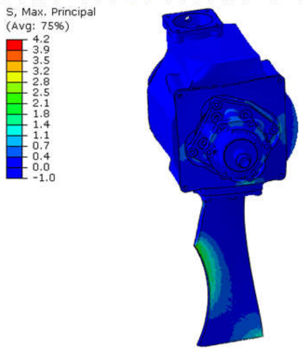
海水泵纵摇状态下，水泵总成的最大Von-Mises应力为**3.8MPa**，最大第一主应力（拉应力）为**4.2MPa**，最大第三主应力为**4.5MPa**（压应力）。

The maximum Von-Mises stress, the maximum first principal stress (tensile stress) and the maximum third principal stress (compressive stress) of the pump assembly were 3.8MPa, 4.2MPa and 4.5MPa when the sea pump was in the pitching state.



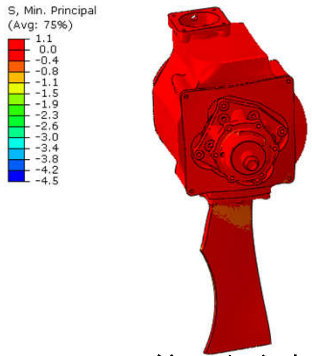
纵摇状态下水泵的应力计算结果

Stress calculation results of water pump in pitch condition



第一主应力

First principal stress



第三主应力

Third principal stress

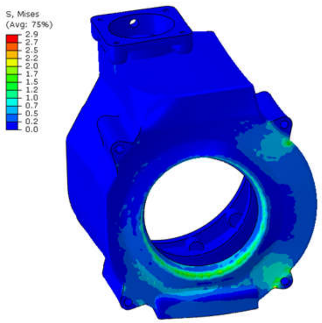
四、海水泵疲劳寿命分析 Fatigue Life Analysis of Sea Water Pump

(3) 海水泵摇摆损伤累积预测

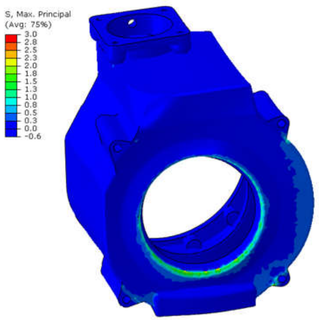
Swing damage accumulation prediction of sea water pump

海水泵纵摇状态下，海水泵壳体的最大Von-Mises应力为**2.9MPa**，最大第一主应力（拉应力）为**3MPa**，最大第三主应力为**3.1MPa**（压应力），最大剪应力为**1.99MPa**。

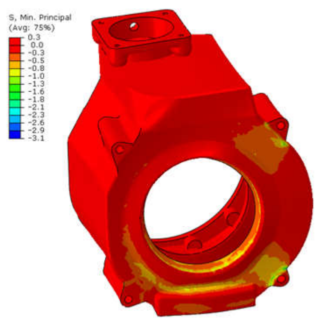
The maximum Von-Mises stress, the maximum first principal stress (tensile stress), the maximum third principal stress (compressive stress) and the maximum shear stress of the sea pump shell were 2.9MPa, 3MPa, 3.1MPa and 1.99MPa under the condition of pitching.



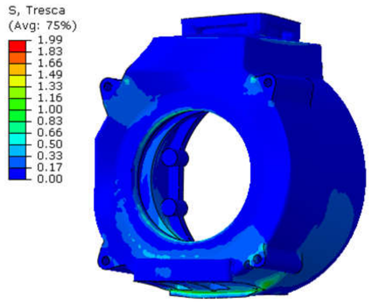
纵摇下水泵壳体应力计算



壳体第一主应力



壳体第三主应力



壳体最大剪应力

Calculation of pump shell stress under pitch stress

Shell first principal stress

Shell third principal

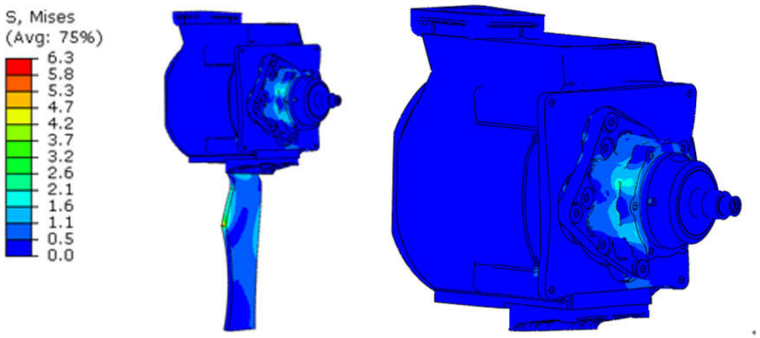
Shell maximum shear stress

（3）海水泵摇摆损伤累积预测

Swing damage accumulation prediction of sea water pump

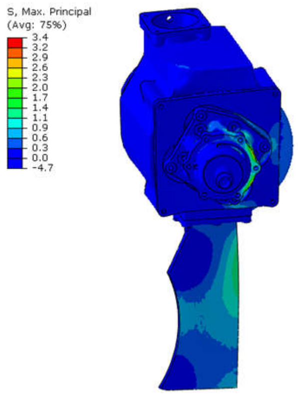
海水泵横摇状态下，海水泵的最大Von-Mises应力为**6.3MPa**，最大第一主应力（拉应力）为**3.4MPa**，最大第三主应力为**11MPa**（压应力）。

The maximum Von-Mises stress, the maximum first principal stress (tensile stress) and the maximum third principal stress (compressive stress) of the sea pump were 6.3MPa, 3.4MPa and 11MPa under the rolling state.



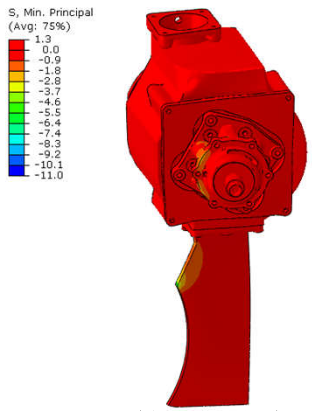
横摇状态下水泵的应力计算结果

Stress calculation results of water pump in roll condition



第一主应力

First principal stress



第三主应力

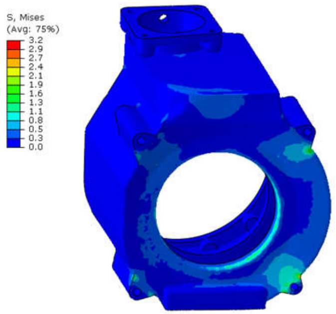
Third principal stress

(3) 海水泵摇摆损伤累积预测

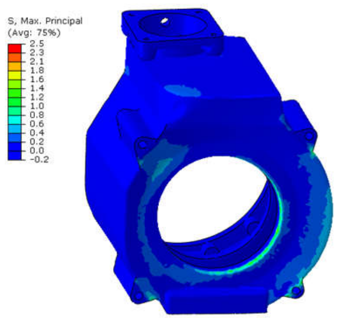
Swing damage accumulation prediction of sea water pump

海水泵横摇状态下，海水泵壳体的最大Von-Mises应力为**3.2MPa**，最大第一主应力（拉应力）为**2.5MPa**，最大第三主应力为**3.3MPa**（压应力），最大剪应力为**3.5MPa**。

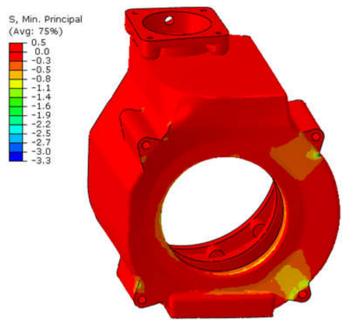
The maximum Von-Mises stress, the maximum first principal stress (tensile stress), the maximum third principal stress (compressive stress) and the maximum shear stress of the sea pump shell were 3.2MPa, 2.5MPa, 3.3MPa and 3.5MPa under the rolling state.



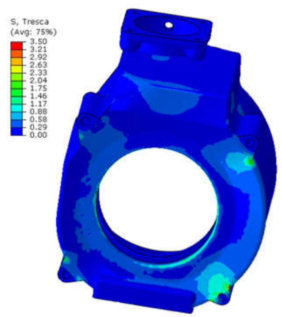
横摇下水泵壳体应力计算



壳体第一主应力



壳体第三主应力



壳体最大剪应力

Calculation of pump shell stress under roll stress Shell first principal stress Shell third principal

Shell maximum shear stress

（4）海水泵振动损伤累积预测

Vibration damage accumulation prediction of sea water pump

提取水泵振动应力计算结果，并开展水泵振动应力水平下的疲劳性能分析，首先分析海水泵各点应力状态，并求解个位置处的关键平面，用于分析疲劳计算。

The vibration stress calculation results of the pump were extracted, and the fatigue performance analysis of the pump under the vibration stress level was carried out. Firstly, the stress state of each point of the sea water pump was analyzed, and the key plane at each position was solved for fatigue calculation.

分析得出海水泵在振动状态下的最危险的平面坐标为**x=107.6mm，y=131.9mm，z=57.6mm**，坐标原点为海水泵重心（其中，X轴为海水泵水平方向，Y轴为海水泵垂直方向，Z轴为海水泵轴向），最危险平面的方向余弦为**(0.24,0.663,-0.701)**。

The analysis shows that the most dangerous plane coordinates of the sea water pump under vibration state are x=107.6mm, y=131.9mm, z=57.6mm, and the origin of coordinates is the center of gravity of the sea water pump (in which, X axis is the horizontal direction of the sea water pump, Y axis is the vertical direction of the sea water pump, and Z axis is the axial direction of the sea water pump). The direction cosine of the most dangerous plane is (0.24,0.663,-0.701).

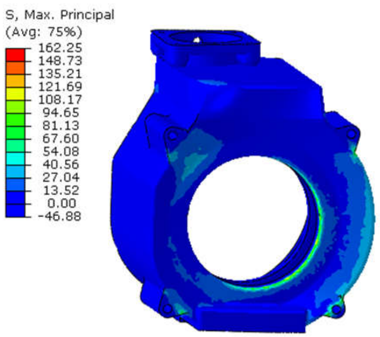
四、海水泵疲劳寿命分析 Fatigue Life Analysis of Sea Water Pump

（4）海水泵振动损伤累积预测

Vibration damage accumulation prediction of sea water pump

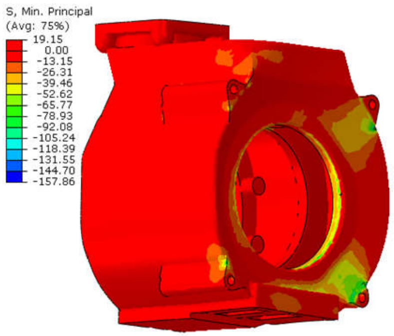
在振动工况下，水泵壳体的最大法向应力为**171.3MPa**，最大剪应力为**161.1MPa**，平均应力最大值为**57.2MPa**，水泵壳体第一主应力为**162 MPa**，第三主应力为**157MPa**。

Under the vibration condition, the maximum normal stress of the pump shell is 171.3MPa, the maximum shear stress is 161.1MPa, the maximum average stress is 57.2MPa, the first principal stress of the pump shell is 162MPa, the third principal stress is 157MPa.



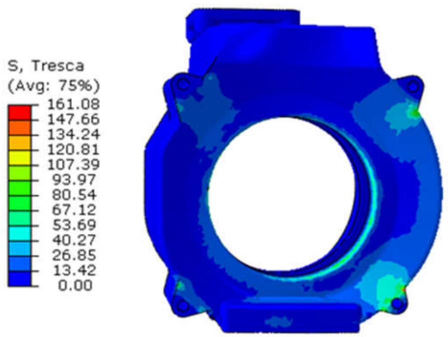
振动下水泵壳体第一主应力

First principal stress



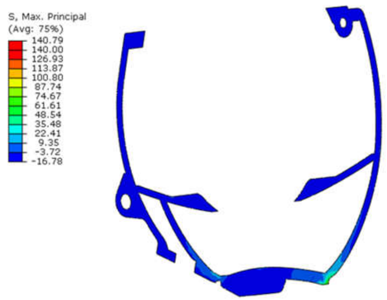
第三主应力

Third principal stress



最大剪应力

Maximum shear stress



危险截面应力分布云图

The stress distribution nephogram of the dangerous section

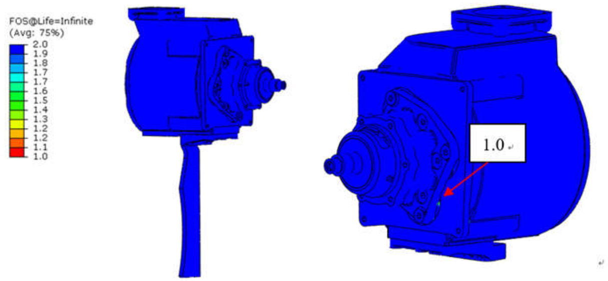
四、海水泵疲劳寿命分析 Fatigue Life Analysis of Sea Water Pump

(4) 海水泵振动损伤累积预测

Vibration damage accumulation prediction of sea water pump

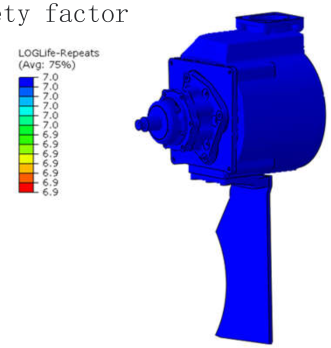
疲劳安全系数的计算结果和海水泵疲劳寿命计算结果如图所示。疲劳安全系数计算结果显示，水泵振动疲劳安全系数最小值为**1.0**，出现在水泵壳体螺栓安装面处，水泵在动态应力作用下仍处于安全范围内，无疲劳破坏的风险。根据对数寿命计算结果，疲劳计算显示最小疲劳寿命为**7943528次**，无疲劳失效风险。

The calculation results of fatigue safety factor and fatigue life of sea water pump are shown in the figure. The calculation results of fatigue safety coefficient show that the minimum value of vibration fatigue safety coefficient of the pump is 1.0, which appears at the bolt mounting surface of the pump housing. The pump is still in the safe range under the action of dynamic stress, and there is no risk of fatigue failure. According to the logarithmic life calculation results, the fatigue calculation shows that the minimum fatigue life is 7943528 times, without fatigue failure risk.



水泵疲劳安全系数计算结果

Calculation result of fatigue safety factor



水泵疲劳寿命计算结果（对数寿命）

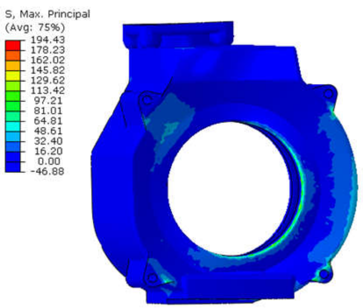
Fatigue life calculation results (logarithmic life)

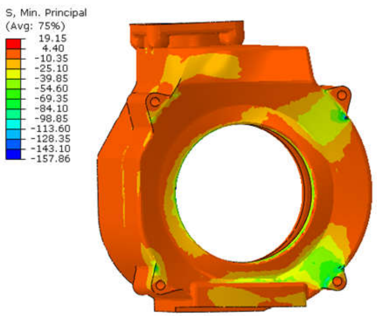
（5）振动、摇摆综合载荷下海水泵损伤累积预测

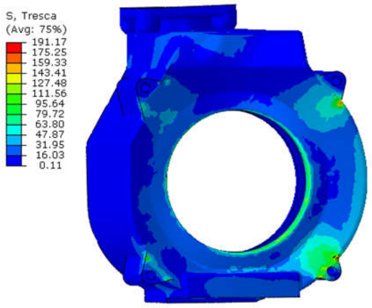
Damage accumulation prediction of sea water pump under vibration and swinging combined load

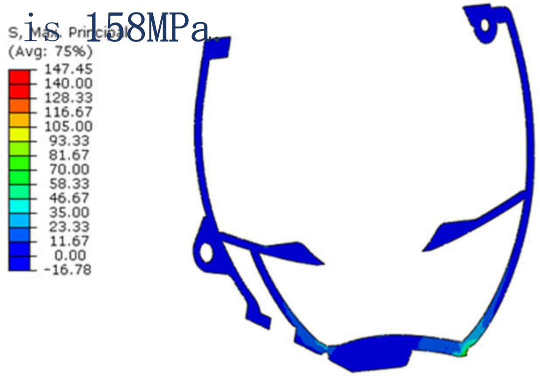
在摇摆及振动综合作用下，水泵壳体的最大法向应力为**184.6MPa**，最大剪应力为**191MPa**，最大平均应力为**70.7MPa**，第一主应力为**194.4MPa**，第三主应力为**158MPa**。

Under the combined action of swing and vibration, the maximum normal stress of the pump shell is 184.6MPa, the maximum shear stress is 191MPa, the maximum average stress is 70.7MPa, the first principal stress is 194.4MPa, the third principal stress is 158MPa.









综合载荷下水泵壳体第一主应力　　第三主应力　　最大剪应力　　危险截面应力分布云图

First principal stress　　Third principal stress　　Maximum shear stress

The stress distribution nephogram of the dangerous section

（6）不同载荷下海水泵损伤累积计算结果对比

Comparison of damage accumulation calculation results of sea water pumps under different loads

1、海水泵只受摇摆载荷时，水泵的应力水平对水泵的可靠性影响较小，疲劳安全系数计算结果看，水泵整体的安全系数均大于2；

When the sea water pump is only subjected to swinging load, the stress level of the pump has little influence on the reliability of the pump. The calculation results of fatigue safety coefficient show that the overall safety coefficient of the pump is greater than 2;

2、海水泵只受振动载荷时，水泵工作时的振动应力对水泵的可靠性影响较大，从计算结果看，动态应力作用下水泵的疲劳安全系数为1.0，疲劳寿命7.9×10^{6}次，处于失效临界区域；

When the sea water pump is only subjected to vibration load, the vibration stress during the pump operation has a greater influence on the reliability of the pump. From the calculation results, the fatigue safety factor of the pump under dynamic stress is 1.0, and the fatigue life is 7.9×10^{6} times, which is in the failure critical region;

（6）不同载荷下海水泵损伤累积计算结果对比

Comparison of damage accumulation calculation results of sea water pumps under different loads

3、在**船舶摇摆及水泵振动**同时作用下，水泵的**疲劳安全系数计算结果为0.9**，**疲劳寿命1.6×10^6次**，**存在失效的风险**。

Under the action of ship swinging and pump vibration at the same time, the calculated fatigue safety factor of the pump is 0.9, the fatigue life is 1.6×10^6 times, and there is a risk of failure.

疲劳计算结果 /Fatigue calculation result	摇摆工况 /Swing condition	振动工况 /Vibration condition	振动摇摆复合工况 /Combined condition
安全系数/safety factor	>2.0	1.0	0.9
疲劳寿命/fatigue life	$>10^7$	**7.9×10^6**	**1.6×10^6**

五、结论

Conclusion

五、结论 Conclusion

本课题通过**有限元分析**的方法，系统分析了船用柴油机水泵在振动、摇摆载荷单独及复合作用下的**应力分布**及**疲劳强度**。结果表明：单独摇摆、振动载荷下，水泵的疲劳安全系数分别为（大于）2.0和1.0，而在振动、摇摆复合载荷下，其疲劳安全全系数只有**0.9**，说明船用柴油机附件的疲劳强度评估时应考虑**振动、摇摆载荷的复合作用。**

In this study, the **stress distribution** and **fatigue strength** of Marine diesel pump under the single and combined action of vibration and swinging load are systematically analyzed by **finite element method**. The results show that the fatigue safety coefficients of the pump are (greater than) 2.0 and 1.0 respectively under swinging and vibration loads, but **only 0.9 under the combined vibration and swinging loads**, which indicates that the **combined effects of vibration and swinging loads** should be considered in the fatigue strength evaluation of Marine diesel engine accessories.

汇报完毕！

谢　　谢！

Thanks for Listening!

本研究工作中张威、颜睿东两位博士生做出了重要的贡献，

在此对他们的工作表示感谢!

Thanks to Zhang Wei and Yan Ruidong for their important contributions in this study!

2023/02