Measurement method for dynamic shaft bearing loads resulting from axial vibrations in a propulsion shaft system of a megayacht

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ABSTRACT

In this study, a measurement method concerning dynamic shaft bearing loads resulting from axial vibrations in a propulsion shaft system of a megayacht was developed. Axial shaft vibration measurement set-up was designed, materials of the experiment were selected and the steps of the measurement were defined.

The main aim of the thesis is to determine the forces coming from the propeller. A new method was developed to calculate these forces. This method is based on modal analysis and mobility function which is the back bone of the modal analysis. Therefore an experimental set-up and the steps of the experiment were developed uniquely in order to obtain the mobility function of the shaft line for megayachts in motion.

An example experiment was designed for a particular motor yacht. All the materials and the tools were selected according to particulars of this megayacht. However the steps of the experiment can be applied to any yacht and are given in a table in conclusion.

The main challenges of the thesis are applying the excitation force and finding suitable tools for the experiment. The measurement of vibration is desired to be performed on a yacht as built. Laboratory devices and conditions are not valid any longer. The spaces around the shaft, real working conditions of the yacht and the lack of convenient excitation methods are all obstacles for this experiment. There are many different types of excitation methods like pressurized air, acoustics, electromagnetic excitation and laser. They are all non-contact type excitation methods but they have different drawbacks and are not suitable to be used in this measurement.

On the other hand an impact hammer which is a contact type excitation method and also dangerous to be used on rotating shaft can be a solution. It has also disadvantages and is not a perfect solution, but it was decided to be used with special modifications and additional devices. It is a special impact hammer which can be controlled automatically.

Hitting in axial direction to the 34 ton of propeller and the shaft which is rotating with 188 rpm is the particular case considered here. Moreover there is just one suitable place to hit the shaft. It is a flange which is full of bolts. Therefore two different solutions were developed to

enable to hit the flange: covering the bolted face of the flange by additional steel plate and attaching the impact hammer to the shaft. The first one needs extra design of plates. In the second option, an automated impact hammer is attached to the shaft and can be remote controlled. These designs reduce many drawbacks of the impact hammer.

The thesis was developed as theoretical basis because of the lack of an on-going megayacht project in the shipyard during the thesis. Therefore performing the measurement and confirming of the method can be a future study.

CONTENTS

Declaration of Authorship

I declare that this thesis and the work presented in it are my own and have been generated by me as the result of my own original research.

Where I have consulted the published work of others, this is always clearly attributed.

Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely my own work.

I have acknowledged all main sources of help.

Where the thesis is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.

This thesis contains no material that has been submitted previously, in whole or in part, for the award of any other academic degree or diploma.

I cede copyright of the thesis in favour of the University of Rostock.

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1. INTRODUCTION TO AXIAL SHAFT VIBRATIONS

Axial vibration is the vibration that occurs in the same direction with the shaft centreline. Shaft axial vibrations are characterized by shafting segments oscillating in a fore-and-aft direction around neutral position.

1.1. Reasons of Axial Shaft Vibrations

The main reasons of axial shaft vibration are listed below.

1- Propeller thrust variations
2- Forces coming from propeller
3- Forces generated in the engine's crank mechanism
4- Torsional-axial coupling (in some cases)
5- Angular misalignment of shaft [1]
6- Misaligned bearing [2]
7- Bent shaft [2]
8- Resonance of some parts in the axial direction, pipes etc. [2]
9- The couple component of dynamic unbalance [2]

One of the most important reasons of axial shaft vibration is misalignment. And the possible causes of misalignment are [1]:

1- Thermal expansion
2- Directly coupled and not properly aligned machine.
3- Forces coming via piping and support members.
4- Parallel misalignment

Also bent shaft has a great role on axial vibration. Some of the causes of bent shaft are listed as shown below. [1]

1- A shaft with a high aspect ratio may develop a bend at shutdown duration due to gravity.
2- It may occur during transportation of shaft.
3- Shaft which is under high torque condition shows bending as well.

If there is a problem with bearings, it may also cause vibration. Such kinds of problems which can be occurred in bearings are:

1- Ineffective reasons
2- Contaminated lubrication
3- Heavier loading than anticipated
4- Improper handling or installation
5- Old age

1.2. Minimizing Axial Shaft Vibrations

Even if the aim of this thesis is not finding a way to minimize the axial vibration, some possible solutions are given briefly below. Each of them can be a new research topic individually.

1- Integrate axial vibration damper into the engine casing.
2- In some cases increasing the number of blades of propeller decreases the vibration level.
3- Using hydrostatic type bearing [3]

2. SHAFT LINE FORCES

During the performing of propeller, forces act on shaft line. These forces which are coming from propeller transferred to the hull by shaft line. While transmitting to the hull, they cause shaft line vibration. Shaft line forces are the most important factor for shaft vibration. Likewise pressure fluctuations are overwhelming reason for vibrations of ship structures.

Propeller forces occur because of several reasons. One of the main reasons is wake field. Propeller wake field depends on design of propeller, speed and the hull shape of the vessel. The protrusions on the aft part of the hull significantly affect the wake field as well. They destroy the wake field and create non-uniform wake. As a result non-uniform wake creates fluctuating shaft line forces. The rate of the non-linearity of the wake field shows itself in shaft line forces: more non-linearity wake field, high amplitude fluctuating shaft line forces.

Wake field affect the angle of attack at the profile section. And this change is proportional to the inflow speed variation. In every rotation of the propeller, the thrust and the tangential force on each blade behave irregularly. These forces also create fluctuating moments because they act eccentrically at about 0,7R. [4]

In the absence of high shaft inclination, the magnitude of the shaft line forces depends on:
-the characteristics of the wake field
-the geometric form of the propeller (in particular the skew and blade number)
-the ship speed
-the rotational speed of the propeller [5]

The thrust fluctuation can be as high as 10% of the mean thrust but it is usually around 2-8%. [5]

3. VIBRATION MEASUREMENT

3.1. Introduction to Measurement

FEM modelling is one of the most preferred techniques to evaluate vibration of the structures. Analytical calculations are also a solution. These techniques are vibration estimation methods and are used because of advantage of low cost in terms of both time and human resource. On the other hand vibration measurement technique gives different perspective with its realistic, confidential and easy going sides.

In parallel with development in electronics industry, experimental techniques have been progressed. Therefore measurement of vibration is getting reliable. Moreover for on-going ships the best and the realistic way are experimental methods to obtain vibration level.

In vibration experiment, it can be measured;

1- Amplitude of vibration

The amplitude of vibration usually shows the size or the magnitude of the vibratory movement. It can be amplitude of velocity, acceleration or force. It gives the severity of the vibration problem in the structure.

2- Frequency of vibration

Frequency of vibration shows the repetition of the vibratory movement occurs in a particular time period. Generally frequency is expressed in cycles per second or Hertz, or in cycles per minute. In rotational vibratory movements it is expressed in terms of shaft speed. A vibration which is at the same frequency with the shaft rotational speed is called 1X or 1 time shaft speed.

3- Waveform

It shows the changing of vibration level with time. It is a graph which shows the variation between the amplitude and time.

4- Spectrum

Spectrum shows the vibration amplitude distribution for different frequencies. It is like waveform however it shows the variation between amplitude and frequency. Spectrum and waveform measurements are usually performed for vibration analysis.

3.2. Measurement Devices

In vibration experiment acceleration, velocity, displacement, strain, force and mechanical impedance can be measured to see the vibration level. With these devices critical speeds, natural frequencies, mode shapes and bearing behaviour of the shafts can be obtained.

3.2.1. Accelerometers

An accelerometer is a sensor that measures the acceleration of the vibratory movement of the structure which it is attached. It produces electrical signal according to vibratory level. Data obtained from accelerometer can be displayed as either a velocity waveform or velocity spectrum. If it is shown as a waveform than it can be transformed to velocity spectrum by using fast fourier transform. [6]

Mainly there are two types of accelerometers: AC-response and DC-response. But they can be classified also as shown below.

Types of accelerometers

- Piezoelectric
- Tri-axial
- Laser
- Low frequency
- High gravity
- High temperature
- Magnetic induction
- Optical
- Shear mode

3.2.2. Velocity Gauges

A velocity gauge is a sensor that measures the velocity of the vibratory movement of the structure which it is attached. Velocity gauges (sensors) are convenient for low to medium frequency measurements. They have lower sensitivity to high frequency vibrations comparing to accelerometers. They are generally used on rotating machinery with the aim of vibration monitoring and balancing operations. [7]

Types of velocity gauges:

- electromagnetic (coil and magnet) sensor
- piezoelectric velocity sensor

The coil and the magnet type of the velocity transducers have been used widely so far to measure the velocity characteristics. [8] However piezoelectric velocity sensors are becoming popular in parallel with the technological developing.

3.2.3. Proximity Probes

Proximity probe is a sensor that measures distance occurs because of the vibratory movement of the structure. They are non-contact type sensors and widely used in measuring the static and dynamic distance. There are many types of proximity probes including radar, laser, capacitance and eddy current. They are widely used for shaft vibration, radial and axial shaft position as well as shaft crack detection, misalignment of the shaft and rotor imbalance. [9]

Types of proximity sensors:

- Capacitive
- Eddy-current
- Doppler effect
- Radar
- Inductive
- Laser
- Magnetic
- Optical
- Thermal infrared

- Ultrasonic sensor

3.2.4. Vibration & Thrust Transmitters

Vibration and thrust transmitters enable to measure direct vibration or target displacement of the shaft, bearing housing, or machine casing. [10]

3.2.5. Strain Gauges

Strain always occurs with vibration, therefore the strain gauge or similar devices which work according to same principle are used in the field of shock and vibration measurement. [11]

Generally strain gauges are used to measure torque and torsional vibratory movements. They are also used to measure pressure fluctuations especially propeller pressure fluctuations.

Types of strain gauges

- Static strain measurement gauges made up with constantan foil. [11]
- Dynamic strain measurement gauges made up with iso-elastic foil (iron-nickel-chrome alloy). Dynamic strain measurement gauges are used for measuring frequency of vibration and the magnitude of the cyclic stresses. [11]

3.3. Vibration Transducer Selection

The transducer is selected according to the condition of the measurement (ambient temperature, accessibility of the measurement point etc.) and the condition of the vibration (amplitude of the movement, frequency etc.). Before selecting transducer, the points mentioned below must be taken into account.

- type of measurement
- vibration level
- desired accuracy level
- frequency range
- temperature range

- environmental condition (corrosive, acidic, combustible etc.)
- sensor dimensions and weight constrain
- appropriate signal conditioning
- cost

And some practical tips to select right transducer are listed below.

- If the vibration is a signal as the relative displacement coming from a bearing clearance, then proximity probe can be used. [12]
- If the vibration is a signal as the absolute vibration coming from a gearbox or engine, then accelerometer can be used. [12]
- In turbo machinery systems proximity probes are usually used to directly measure shaft motion. [12]
- Highly stiff systems or structures which transfer the vibrations directly, accelerometers can be used. [12]
- In condition of lower frequency shaft motions, proximity probes can be used. [12]
- For high-frequency elements or structures, accelerometer can be used. [12]

Apart from above, the combination of proximity probes and accelerometers or other combinations can be used to obtain different types of information at the same time. Figure 3.1 and 3.2 show the relation between the efficiency of transducers and the frequency.

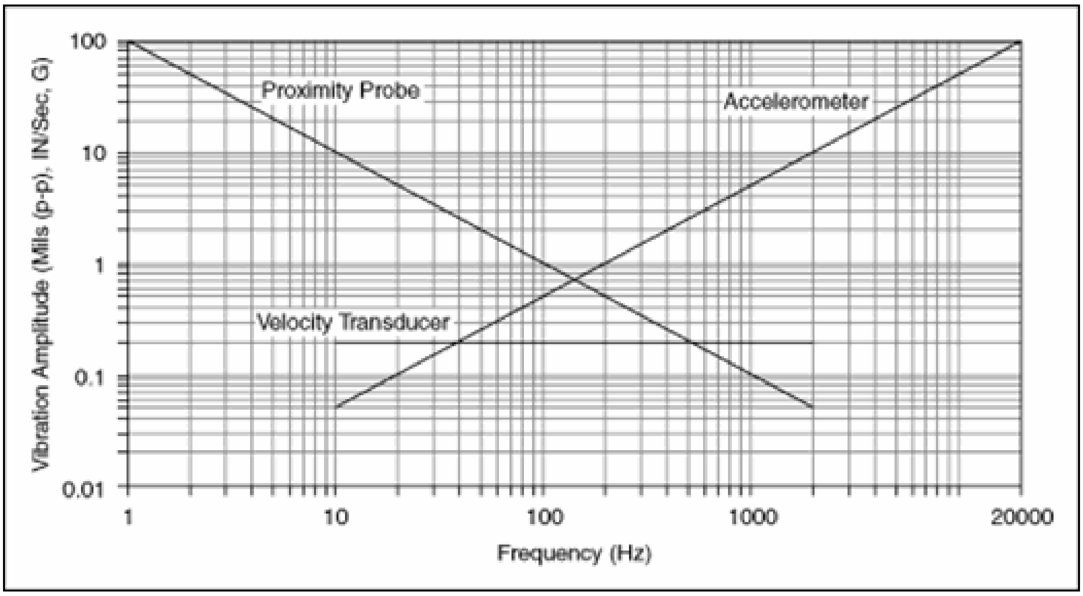

Figure 3.1 Transducers efficiency-frequency relation [12]

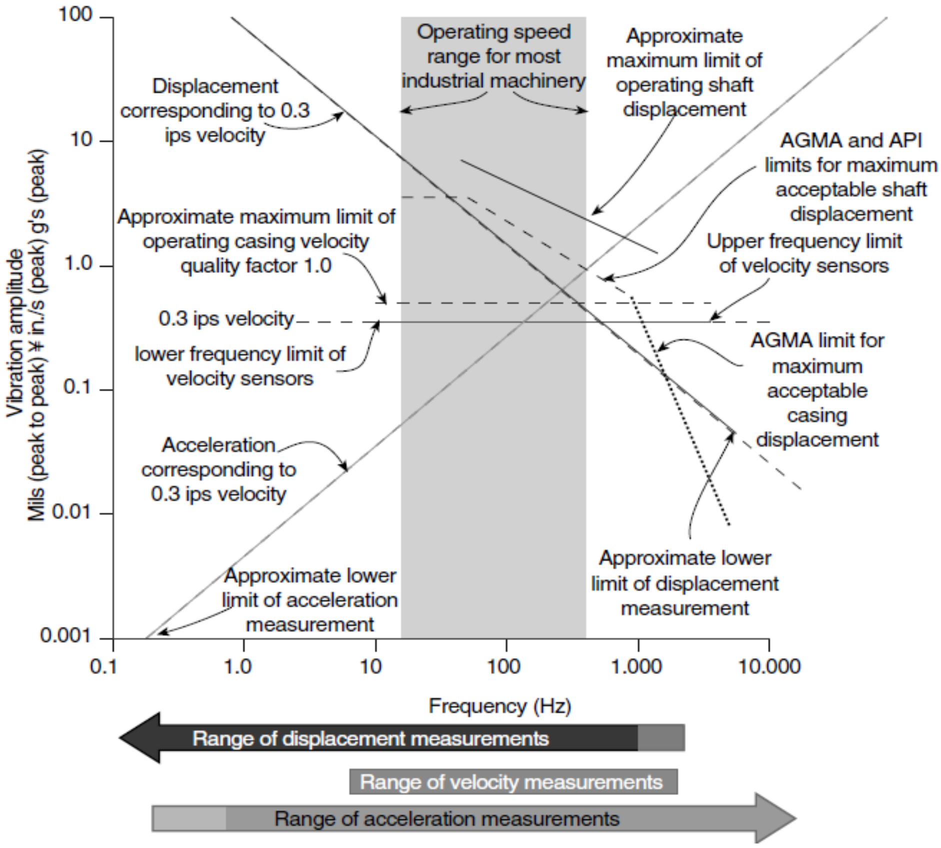

Figure 3.2 Frequency ranges of accelerometers, seismometers, and proximity probes [13]

4. SHAFT VIBRATION MEASUREMENT

4.1. Types of Transducers in Shaft Vibration Measurement

In literature there are four types of transducers for measuring shaft vibration.

- Piezoelectric accelerometers
- Eddy current probe
- Linear laser vibrometer
- Shaft rider

4.1.1. Piezoelectric Accelerometers

They work according to absolute motion method. They produce electrical output parallel to the strain caused by the inertial force of the seismic mass in accelerometer. [14] Sections of piezoelectric transducer are shown in Figure 4.1.

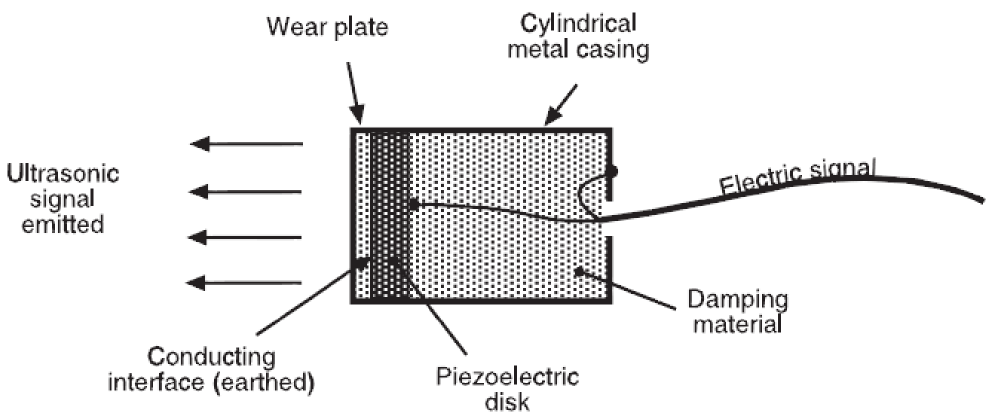

Figure 4.1 Piezoelectric transducer sections [15]

The proportion between electric output and relative displacement is represented by the charge sensitivity function $k_Q(\omega)$. This factor depends on the elastic properties and dimensions of the piezoelectric element. [14]

Advantages

- Wide frequency response (up to 3.000.000 cpm) [13]
- Lightweight, physically small and no moving parts
- Temperature range from −254 °C to above 760 °C without external cooling [11]
- Acceptable price level comparing with other transducers
- Acceleration magnitude range from 10^{-6}g to more than 10^{5}g [11]

Disadvantages

- Sensitive to stresses induced when permanently mounted
- No sensitivity to static acceleration
- Low frequency cutoff [14]

4.1.2. Eddy Current Probe

They are usually called proximity probes or sensors. Meanwhile they are also known as inductive pickup. [13] Eddy current probes measure the static and dynamic distance between the object and the probe. Eddy current probes work according to relative motion method and measure the distance directly. Proximity probes are non-contacting sensors.

They are commonly used to measure the vibrations of rotating shafts, and relative motion between a shaft and its bearings.

In detail proximity probes are used for the items below:

- Radial vibration for indicating bearing condition and measuring machine malfunctions. [16]
- To determine the shaft radial position relative to rotor attitude angle [16]
- To measure shaft rotation speed and phase angle of shaft [16]
- To monitor the axial/thrust position of a rotor [16]

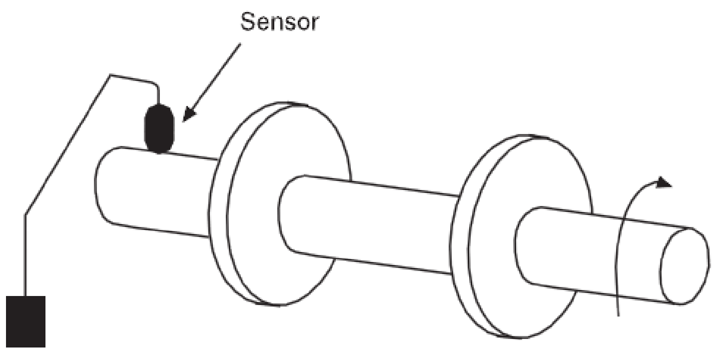

Figure 4.2 Eddy current probe [7]

The eddy current probe system includes probe, extension cable and driver system. All elements are selected according to vibration operation and the system's natural frequency. The whole system should be in harmony. Therefore every eddy current probe system is provided with fixed cable length which are usually 2, 5, 9 or 14 meter. [17] Application of eddy current probe on a shaft is illustrated in Figure 4.2.

Advantages

- Actual shaft motion can be measured [13]
- Lightweight, physically small
- Probe diameters can be made in any length. [13]
- No effect of contaminants or lubricants on the proximity probes measurements
- Temperature range up to 120 ^{o}C
- Frequency ranges from 0 to 300.000 cpm [13]
- Relatively low cost

Disadvantages

- Required special mounting of probes to bearing housings
- Effected easily by the surface imperfection
- Power supply is necessary

4.1.3. Laser Vibrometer

They are known also as a Laser Doppler vibrometer (LDV). LDV is an optical non-contacting type transducer. It determines the instantaneous velocity by the method based on Doppler Effect phenomena. [15] Working principle of LDV is shown in Figure 4.3.

With LDV, performing vibration measurement in extreme conditions (for example too hot, too cold, difficult to access) is possible. It is strictly recommended to be used for low amplitude vibrations. It is also perfect option if non-destructive measurement is necessary.

The biggest drawback of laser Doppler vibrometer is its price. They are quite expensive compared to other vibration measurement devices.

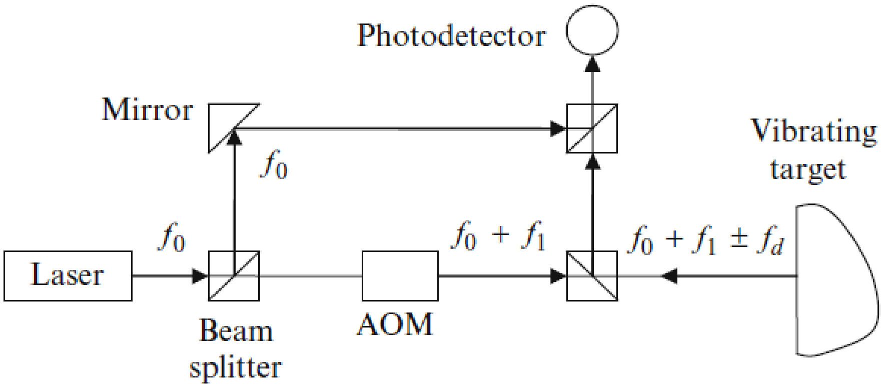

Figure 4.3 Laser vibrometer layout [18]

4.1.4. Shaft Riders

Shaft riders work according to absolute motion method and measure vibration of a rotating shaft directly from the shaft surface. The layout of shaft rider is given in Figure 4.4.

Shaft Rider can be used for large rotating machines which has moderate speeds. However they are not used widely nowadays because of safety and the application uncertainty.

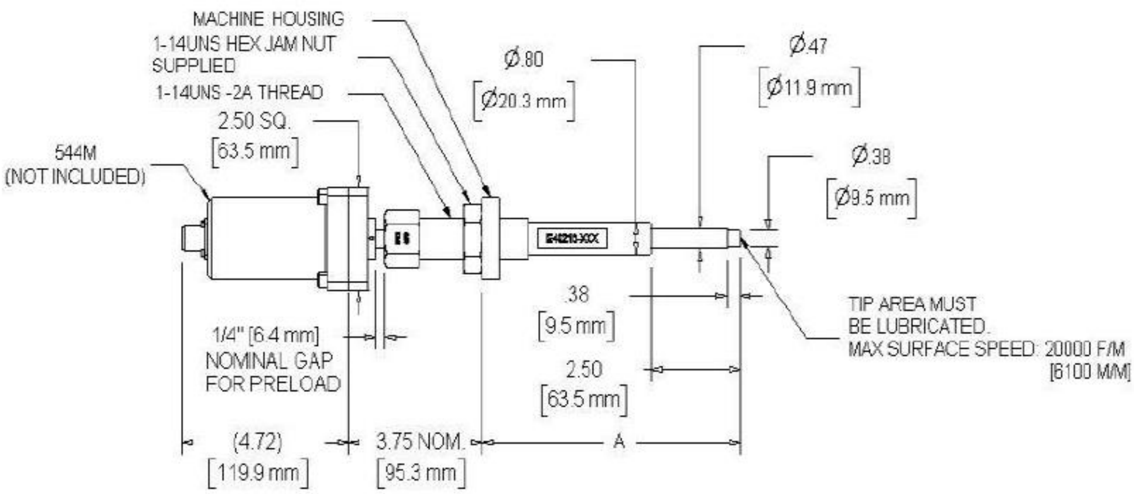

Figure 4.4 Shaft rider layout [19]

4.2. Shaft Vibration Measurement Methods

The shaft vibration can be measured either relative or absolute vibration methods.

4.2.1. Absolute Motion Method

In this method a system which includes mass, spring and damper that provides reference point for the model is attached to the moving object which is vibrating. Therefore the vibratory motion of the object can be measured with the help of this system and it is called seismic system. If absolute method is desired then the transducers must be seismic type. Absolute motion method mechanism and seismic system are shown in Figure 4.5 and 4.6 respectively.

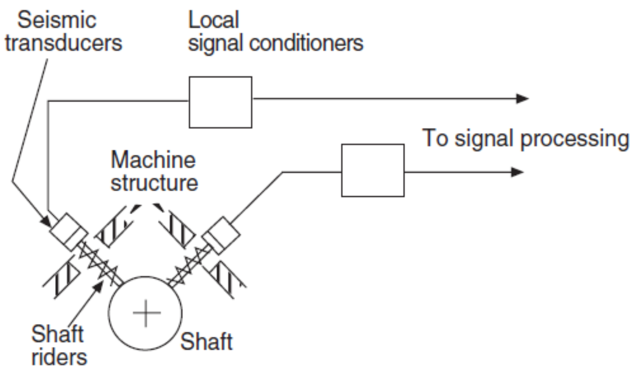

Figure 4.5 Absolute motion measurement system using shaft-rider mechanism with seismic transducers [15]

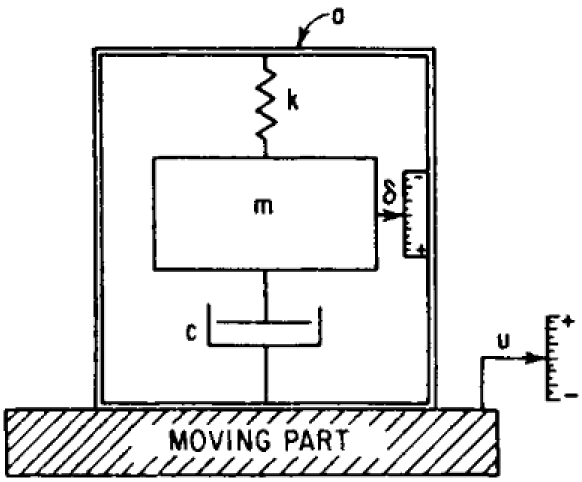

Figure 4.6 Seismic system demonstration [11]

4.2.2. Relative Motion Method

In this method, there is a transducer between the object and the reference point. One terminal of the transducer is attached to moving object which is vibrating and the other terminal is attached reference point fixed in space. Relative motion method is given in Figure 4.7.

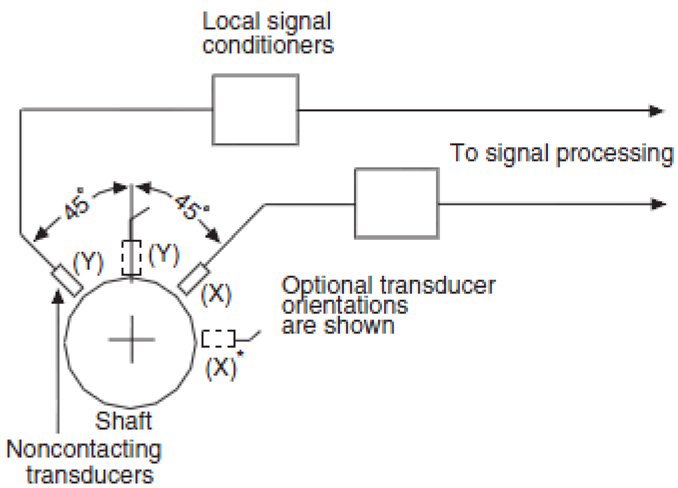

Figure 4.7 Relative motion measurement system using non-contacting transducers [15]

4.2.3. Selection of Measurement Method

As in the selection of transducers almost the same approach is used for selection of method. Actually they are related to each other. If absolute method is selected then seismic type transducers must be used. Thee appropriate type of seismic transducer can be selected according to measurement conditions. Like absolute method, if relative motion method is selected, the transducer that is suitable for this method must be selected.

Selection of method is explained in Reference [21] is as shown below.

“If the vibration level of the support structure is less than 20% of the relative vibration of the shaft, either relative or absolute shaft vibration can be measured for the purpose.

However, if the vibration level of the supporting structure is greater than 20% of the relative shaft vibration, absolute vibration should be measured or a combination shaft vibration vectorially combined (using phase angles).

If a limited kinetic load of the bearing is necessary to protect the bearing, vibration of the shaft relative the bearing structure should be monitored as the most important criterion. If clearances of the rotor relative to stationary elements are the criteria, the type of measurement used depends on the level the structure supporting the transducer used to measure relative motion.”

Finally, if the vibration level of the support structure is less than 20% of the relative vibration of the shaft, relative vibration is a measure of the clearance absorption. If the vibration level of the support structure exceeds 20% of the relative vibration of the shaft, the latter may still be used as a criterion unless the vibration of the supporting structure is not representative of the vibration of the total stator housing. In such a case special measurement of vibration of the stator housing must be made at critical clearances.

It is explained in Reference [21] in another way as well:

“If the majority of shaft motion energy is absorbed within the bearing clearance, and very little is transmitted to the bearing hosing, then shaft relative measurement is required. On the other hand, if a significant amount of shaft vibration is transmitted to the bearing housing, then shaft relative measurement alone may not be satisfactory.

Indeed, if virtually all the shaft motion is directly transferred to the bearing housing, then bearing housing measurement is adequate. There are such cases with fluid-film bearing machinery. But in most situations, there is usually a significant level of shaft activity relative to the bearing. And that vibration may or may not be transmitted to some degree to the bearing housing.

Concerning evaluation criteria, the document suggests: "when the relative motion transducer support structure vibration is 20 per cent or more of the relative shaft vibration, then the absolute shaft vibration will be measured, and if found to be larger than the relative shaft vibration, it will be used as the measure of shaft vibration.

Absolute vibration is recommended on units which have fabricated bearing housings. This absolute system includes a relative sensor to measure shaft movement in relation to bearing movement and a seismic sensor to measure bearing movement in relation to a fixed frame of reference."

Two vibration measurement examples from literature are given in Appendix from Reference 40 (bentley nevada design) and 17 (turbine shaft example).

5. Modal Analysis

5.1. Introduction

Modal analysis is a way to generate modal parameters of structure or the element. In this analysis, excitation force is applied one or several points and the responses of the structure are measured.

With modal analysis the parameters below can be obtained.

- mode shapes
- frequency
- damping

Modal analysis can be used for linear systems only. However if it is not linear, linear approximation should be applied to have acceptable results.

5.2. FRF Measurement

FRF; frequency response function is a way in modal analysis to acquire modal parameters. Actually it shows the relationship between the output and input data. And it achieves this relationship as a function of frequency.

It is also described as the ratio of the Fourier transform of an output response (X(w)) divided by the Fourier transform of the input force (F(w)) that caused the output in Reference 22.

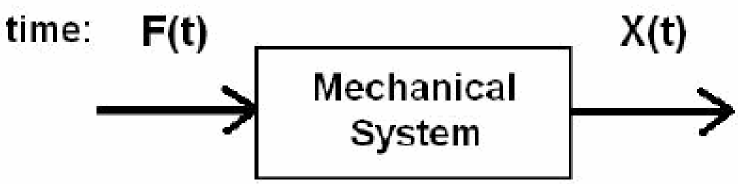

Figure 5.1 Schematic definition of FRF

So FRF can be expressed as

$$H(w) = \frac{X(w)}{F(w)} \tag{5.1}$$

In FRF, response can be displacement, velocity or acceleration. Therefore the FRF definition varies according to response.

Table 5.1 FRF measurement definition

FRF response	**Name**
Displacement/ Force	Compliance
Force/ Displacement	Dynamic stiffness
Velocity/Force	Mobility
Force/Velocity	Impedance
Acceleration/Force	Inertance or Receptance
Force/ Acceleration	Dynamic mass

As can be seen either in Equation 5.1 or in Table 5.1, the input is needed to have frequency response function (FRF). And this is force (F(w)) here. This force is excitation force that refers mathematical signal function and applied by using many different ways. Generally its value is known in advance.

In order to achieve modal analysis there are several methods for producing excitation forces. The main and most used method is impact hammer. However other methods have been developed basically because of inadequate use of impact hammer. These are:

- impact hammer
- shaker
- laser
- pressurized air
- acoustics

5.3. Types of Excitation Methods

5.3.1. Contact Type Excitation Methods

In these methods, the excitation force generator device and the test structure are in contact. They can be either in contact for a short period like in impact hammer or a long period like in shaker.

5.3.1.1. Impact Hammer

This method is used widely to produce excitation force and it proved its reliability. However it has some handicaps using it for rotating elements like shafts. Damaging the element and the repeatability are the firsts coming into mind. As a result, impact hammer method does not seem a good solution for shafts. However with modifications on the impact hammer and the test structure, it can be used for shaft vibration too. As it will be explained in section 6, impact hammer is used in this thesis.

5.3.1.2 Shaker

Impact hammer can not be used for structures which have fragile or delicate surfaces. In this case shakers can be used.

A shaker is a device that vibrates the object or structure by different ways. Shakers are used because of their high energy density comparing to impact hammer. On the other hand connection of a shaker to the test structure is a drawback. This application affects the structure's dynamics. Shakers connect to structures in three ways [23]:

- by the stinger which is a thin metal rod
- by the table which is located on the shaker (test structure is placed on the table)
- by the table built onto the shaker (shaker vibrates the test structure in horizontal direction)

Shakers are classified as a contact type method and there are two main shaker types:

- electromagnetic shaker
- hydraulic shaker

An electromagnetic shaker is built on the principle of magnetism like speakers. A magnet inside the shaker provides a magnetic field which produces electromagnetic force and this force is used as an excitation force in modal analysis. With electromagnetic shakers it is possible to make high frequency experiment.

A hydraulic shaker is based on hydraulic cylinders. With hydraulic shaker, it is possible to produce much higher force levels comparing to electromagnetic shaker. They are usually used for big structures.

Shakers can be used for shaft application (applying the force on bearings) however the nature of rotating elements limits the reliability of the measurement.

5.3.2. Non-contact Type Excitation Methods

For rotating elements contact type excitation does not seem so possible. Therefore new methods are coming into question. Even if they have still problems in application, they seem better alternatives. In these methods the most significant point is to calculate the excitation force accurately.

5.3.2.1. Pressurized Air

In this method, pressurized air is used as an excitation force. Any pneumatic devices that can be found in the market can be used in this method. Pressurized air method can provide up to 0,6 N at frequencies up to 1 kHz. [24] Flow chart and a schematic view of pressurized air method are shown in Figure 5.2 and 5.3.

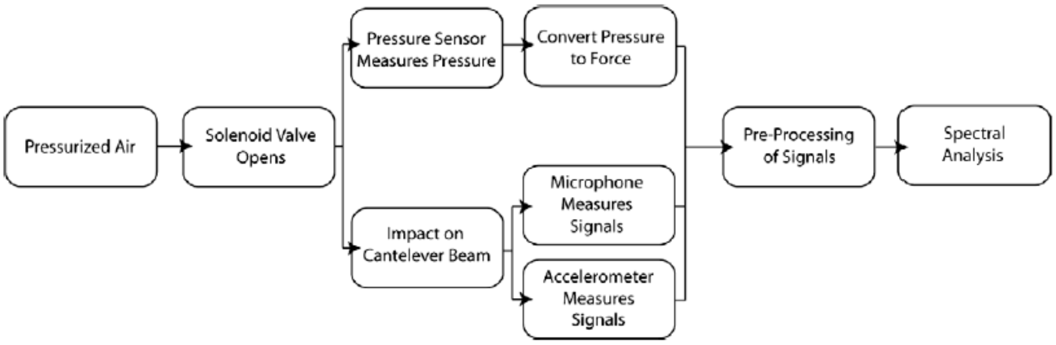

Figure 5.2 Flow chart of pressurized air method [25]

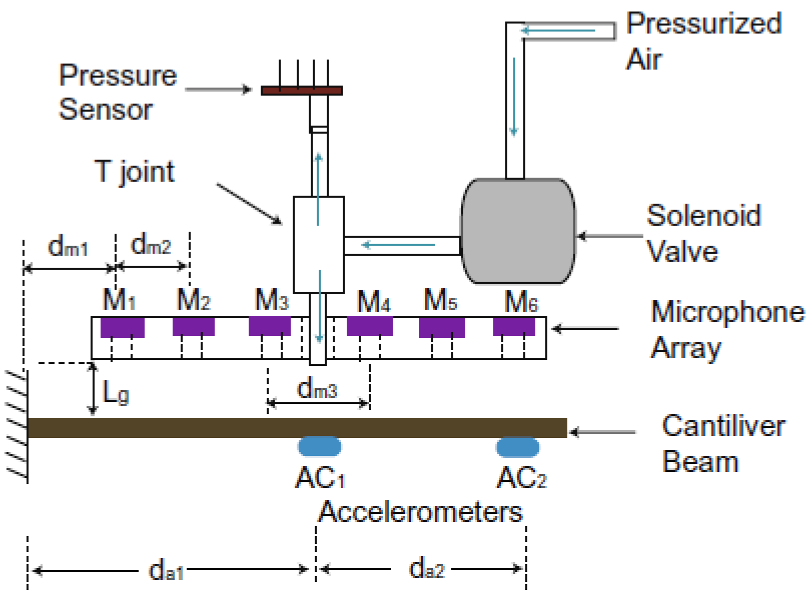

Figure 5.3 Modal analysis of a cantilever beam schematics using pressurized air method [25]

The force applied to the structure can be calculated as shown below. [24]

$$F = \dot{m} c_m \tag{5.2}$$

m: mass flow (kg/s)

C_m: gas flow velocity

Mass flow can be calculated by the Equation 5.3 or 5.4. [24]

$$\dot{m} = CP_u \rho_0 \sqrt{\frac{293}{T_u}} \sqrt{1 - \left(\frac{P_d / P_u - b}{1 - b}\right)^2} \qquad \text{if} \qquad P_d / P_u \geq b \tag{5.3}$$

P_d: downstream absolute pressure

P_u: upstream absolute pressure

T_u: upstream temperature (Kelvin)

ρ_0: the ambient air density

b: the critical pressure ratio

C: the specific mass flow (m^3/(sPa))

$$\dot{m} = CP_u \rho_0 \sqrt{\frac{293}{T_u}} \qquad \text{if} \qquad P_d / P_u < b \tag{5.4}$$

There are also other ways to calculate the force. One of them is given in Reference 25. In this method, parameters such as pressure, temperature and distance between the nozzle and the structure affects the force applied. The schematic view of creating force in this method is shown in Figure 5.4.

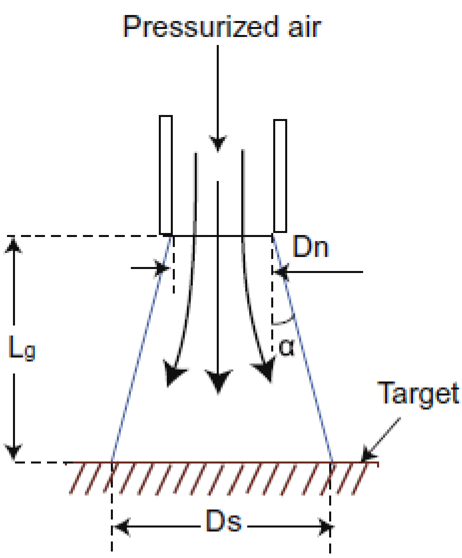

Figure 5.4 Pressurized air impulse force measurement setup [25]

This method can be used for producing excitation force on shaft. Moreover it is a cheap solution in non-contact excitation techniques. But the force amplitude that can be generated in this method is not enough to excite megayacht shaft.

5.3.2.2. Acoustics

In this method the excitation force is created by the sound energy. For creating this force, high power horn drive loudspeakers are used and the force level can be reached around $5x10^{-2}$ N. [26]

The main problem of the acoustic method is the difficulty of measuring excitation force. In Reference 26, a method is offered for estimation of force on rotating machine. When this system is used on shaft as shown below in Figure 5.5, the excitation force is calculated by integrating the pressure applied by the loudspeaker over the surface. Meanwhile the pressure can be measured by the microphone.

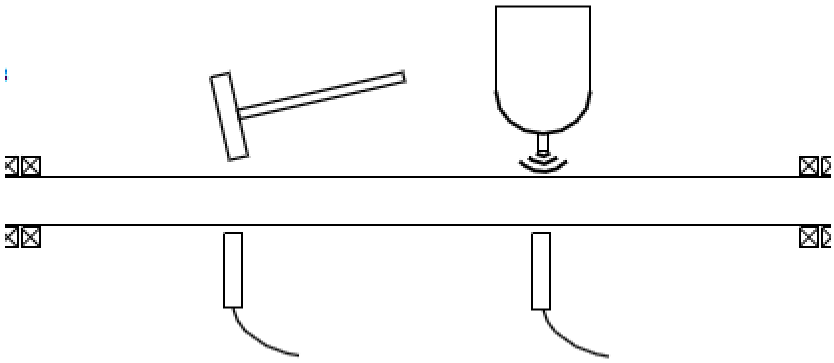

Figure 5.5 Acoustic excitation scheme [26]

In their method, it is assumed that the pressure measured at a certain distance from the shaft is equal to the pressure whole around the surface of the shaft. With the help of this simplification the acoustic force can be calculated as shown below. [26]

$$F_{acuustic} = P_{acouistic} \frac{\pi d l_{\exp osed}}{2} \tag{5.5}$$

$P_{acoustic}$: pressure measured at a certain distance from the shaft

d : shaft diameter

$L_{exposed}$: the length of the shaft part under excitation force

The advantages of this method are

- any kind of excitation signal can be used (multi -sine, random noise, pulse) [26]
- easy to repeat
- cheap
- non-destructive

However the force amplitude is too low like in the pressurized air method. Therefore it is not acceptable to excite megayacht shaft. Both methods can be used just for small structures in laboratory conditions.

5.3.2.3. Electromagnetic Excitation

There are many types of electromagnetic excitation. They can be either DC or AC system, or they can be on fixed frame or rotating etc. However all types use electromagnetic principle based on Lorentz force.

Lorentz force is the force on the charged particle due to electromagnetic fields. The direction of this force can be found by the right hand rule.

$$F=q(E + v \times B) \tag{5.6}$$

q: electric charge

v: instantaneous velocity

E: electric field

B: magnetic field

Here the method which is given in Reference 27 is explained. The force in this method is produced by the current into electric unit which consists of permanent magnets and coils.

When the current travels along the coils, the Lorenz force is created between the coil-magnet couples which are located near the test structure. By arranging the coil-magnet couples the force direction can be adjusted. Two arrangements are given below in Figure 5.6 and Figure 5.7. They call it electromagnetic acoustic method. The basics of electromagnetic acoustic and eddy current method are same. [27]

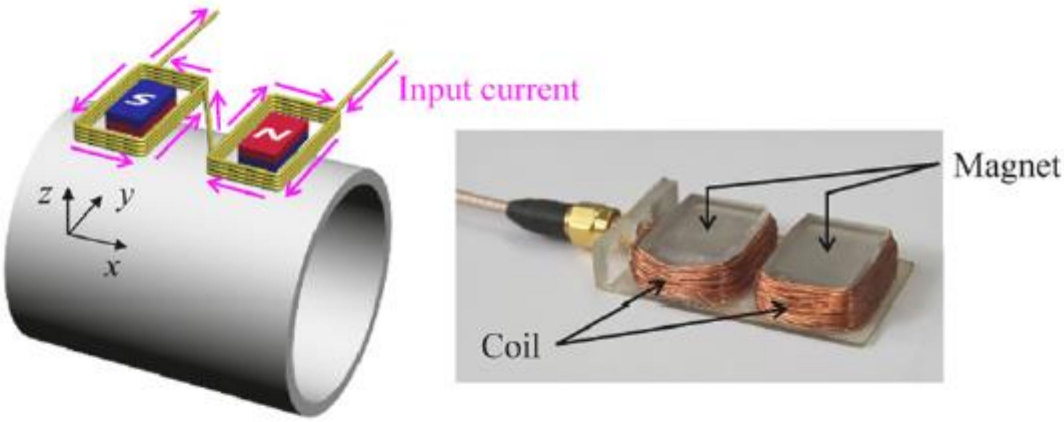

Figure 5.6 General view of electromagnetic acoustic method and the coil-magnet couples [27]

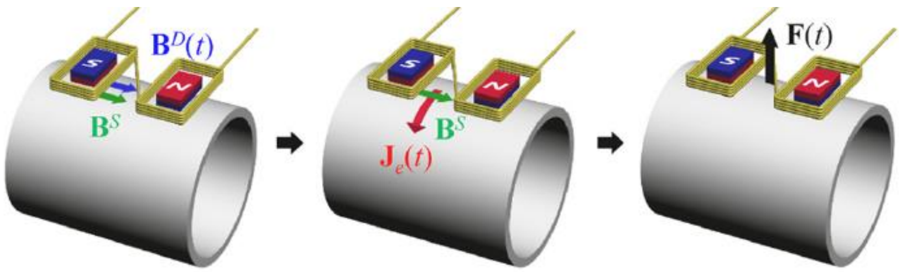

Figure 5.7 Creating force in z direction [27]

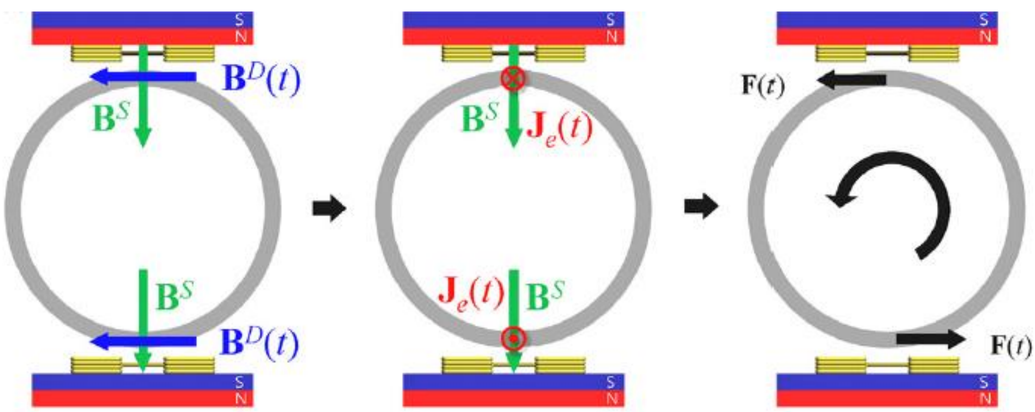

Figure 5.8 Creating force in y direction [27]

As can be seen from the Figure 5.7, when the coil-magnet couple is located side by side, then the z direction force occurs which induces bending vibration. When they are located upside down, then the y direction force occurs which induces torsional vibration as shown in Figure 5.8. Like samples above when they are located facing one another in the y axis, the x direction force can be created which means axial force. This arrangement can be used for axial shaft modal analysis.

Advantages of the method:

- the system itself can be used as a sensing unit
- suitable for low frequency application up to 100 kHz [27]
- no mechanical motion of the transducer is needed for vibration excitation

5.3.2.3. Laser

This method is a comparatively new technique. The force amplitude that can be generated in this method is quite low. Therefore it can be used if excitation force necessary for the process is small.

Laser used here is a group of electromagnetic waves at the wavelength of the laser cavity. [28] When the laser pulses are applied to the surface of the structure, the temperature of the outer layer of the structure is increased and as a result thermal energy of this part is increased and

the heated part expands. Later, expanding part of the structure applies pressure on an adjacent layer. The result is compressive shock wave which affects the whole material. [28]

Laser can be used:

- for structures which are hard to reach
- for hot surfaces
- when necessary to use non-contact method
- for difficult environments like adhesive, dusty or greasy etc.

However the main drawbacks of this method are:

- the low level excitation force

- the high temperature generated by laser can cause melting or local vaporisation of the material
- can be used just for small structures

Consequently it is not suitable for megayacht shaft excitation.

6. CALCULATION METHOD OF MOBILITY FUNCTION AND FORCE

6.1. Introduction to Method and Notation

In this part of the study, calculation method of mobility function and propeller force is explained. The method is based on modal analysis and frequency response function.

Firstly mobility function is found than with the help of mobility function propeller force will be calculated. In order to obtain the mobility function of shaft propeller system experiment was designed in three steps. The details of the steps will be given in section 6.2. After it, force equations are derived depending on mobility functions and by iteration method, propeller force can be calculated.

The notation used in this section is given below.

- subscripts express the type of force, response etc. For example v_1 means response due to the hull
- superscripts express the step of calculation or the operation. For example F_1^{ii} means force from hull in operation 2.

v_1= shaft response due to ship hull itself
v_2 = shaft response due to propeller
v_3 = shaft response due to artificial excitation
v = total response of the system
v^i = total response of the system in operation 1
v^{ii} = total response of the system in operation 2
v^{iii} = total response of the system in operation 3
v_1^i = response of the hull in operation 1
v_2^i = response due to propeller in operation 1 (zero)
v_3^i = response due to impact force applied to the shaft in operation 1 (zero)
v_1^{ii} = response of the hull in operation 2

v_2^{ii} = response due to propeller in operation 2

v_3^{ii} = response due to impact force applied to the shaft in operation 2 (zero)

v_1^{iii} = response of the hull in operation 3

v_2^{iii} = response due to propeller in operation 3

v_3^{iii} = response due to impact force applied to the shaft in operation 3

M_{ss} = mobility function of the shaft

M_{hull} = mobility function of the hull

F= force

F_{impact}= impact force applied to shaft in operation 3

F_1^i = impact force in operation 1 (to the hull)

F_1^{ii} = force coming from the hull during the operation 2

F_2^{ii} = force coming from the propeller during operation 2

6.2. Calculation of Mobility Function

The mobility function is defined as shown in Equation 6.1. Velocity response to the excitation force ratio gives the mobility function.

$$v=M_{ss}*F \tag{6.1}$$

v= velocity response (m/s)

M_{ss}= mobility function (m/(sN))

F=force (N)

Mobility is a tensor which describes the effects of force acting on a structure. It can be presented in frequency domain by a matrix like in Equation 6.2. [29]

$$v(w) = M(w)F(w) \tag{6.2}$$

Here w is angular frequency which is equal to 2πf, f is frequency, F(w) is a column vector of exciting forces, v(w) is the column vector of velocity response and the M(w) is symmetric tensor of mobilities M_{ij}. [29]

Equation 6.2 can be expressed in another form as shown in Equation 6.3,

$$\begin{aligned} v_1 &= M_{11} f_1 + M_{12} f_2 + M_{13} f_3 + \dots\dots \\ v_2 &= M_{21} f_1 + M_{22} f_2 + M_{23} f_3 + \dots\dots \\ v_3 &= M_{31} f_1 + M_{32} f_2 + M_{33} f_3 + \dots\dots \end{aligned} \tag{6.3}$$

The force acting at a point j creates a velocity at point i and this velocity response is defined by $M_{ij}f_j$ in the Equation 6.3. If this velocity is noted by $\bar{v}_{ij}$ then [29]

$$v_i = \sum_j \bar{v}_{ij} \tag{6.4}$$

Equation 6.4 means that the velocity responses can be summed. And if just one force is applied to the structure, then Equation 6.3 becomes

$$\begin{aligned} \bar{v}_{12} &= M_{12} f_2 \\ \bar{v}_{22} &= M_{22} f_2 \\ \bar{v}_{32} &= M_{32} f_2 \end{aligned} \tag{6.5}$$

In situation of excitation from one point with just one force, the mobility function is expressed as shown in Equation 6.6.

$$M_{12} = \bar{v}_{12} / f_2 \tag{6.6}$$

Therefore in order to get mobility function it is needed to measure the response of the structure and force applied by impact hammer.

The response of the shaft can be expressed as shown in Equation 6.7. The total response is the sum of all responses: response coming from hull, response due to propeller and response due to impact hammer. Response always refers to velocity response in this thesis.

$$v = v_1 + v_2 + v_3 \tag{6.7}$$

After measuring responses and finding the mobility function, force coming from propeller can be used.

6.2.1. Steps of Experiment

There are three steps of experiment. The experiment starts with step one which refers to zero propeller speed. Total velocity response is measured and the components of the total response are determined in each step.

Step 1 V=0 (zero speed)

The aim of this step is to apply impact force to the hull to obtain the mobility function of the hull.

The hull can be excited by impact hammer and the response on the shaft will be measured. Therefore the mobility function of the hull can be obtained.

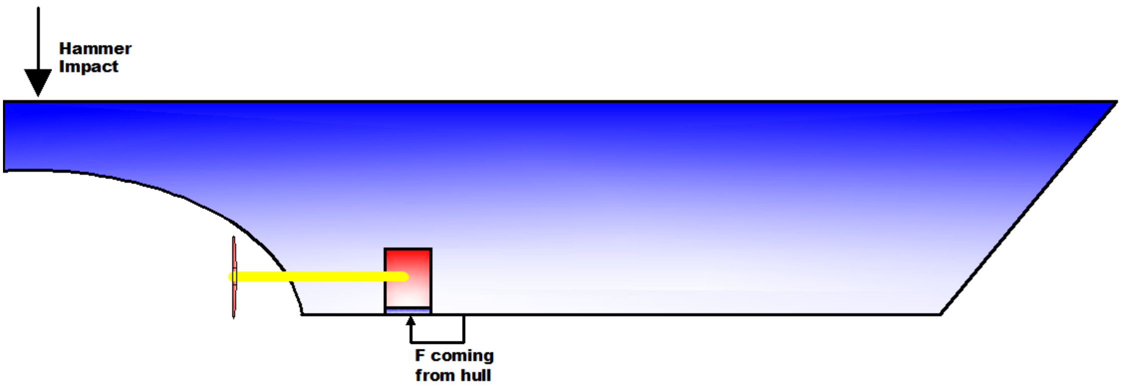

Figure 6.1 Impact hammer application to ship hull

$$v^i = v_1^i + v_2^i + v_3^i \tag{6.8}$$

The total velocity response formula for the step one is given in Equation 6.8. The vessel is moored and there is no excitation force to the shaft in this step. Therefore there is no response due to propeller and due to hammer impact to the shaft.

v_2^i =0 (no vibration from the propeller, V=0, moored)

v_3^i =0 (no impact to the shaft)

So the Equation 6.8 becomes $v^i = v_1^i$.

v^i is measured and the force applied by the hammer is known. Consequently the mobility function of the hull M_{hull} can be found.

$$v_1^i = F_1^i * M_{hull} \tag{6.9}$$

Step 2 V≠0 (velocity)

The aim of step two and three is to get the mobility function of the shaft. In step two the vessel is sailing but there is no impact either to the hull or to the shaft. The velocity response is measured on the shaft while vessel is moving. Total velocity response is the sum of response due to propeller and the response of the hull.

v_3^{ii} =0 (no impact to the shaft)

Hence, the total response becomes:

$$v^{ii} = v_1^{ii} + v_2^{ii} \tag{6.10}$$

Step 3 V≠0 (same speed with step 2)

While the vessel is at the same speed of step 2, the impact hammer is applied to the shaft. All components of the velocity response are active in this step as shown in Equation 6.11.

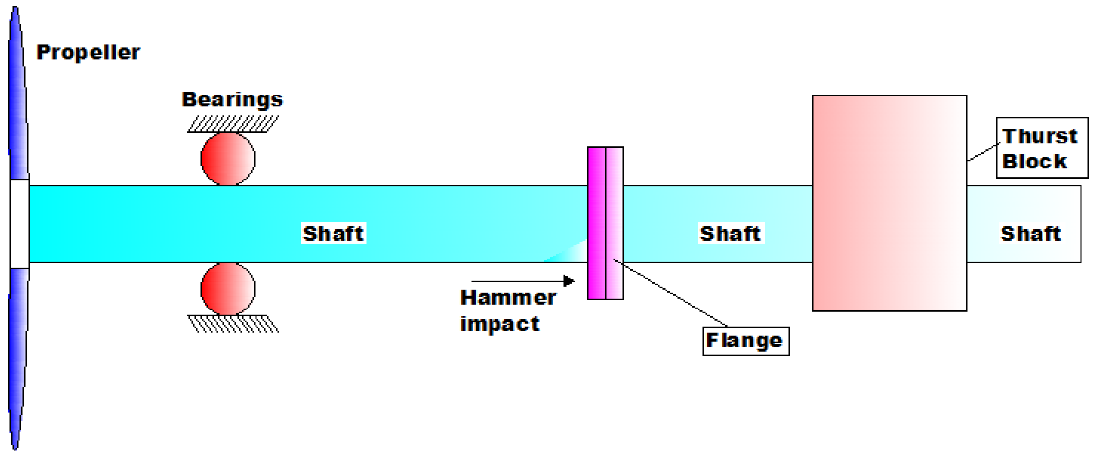

Figure 6. 2 Hammer impact to the shaft

$$v^{iii} = v_1^{iii} + v_2^{iii} + v_3^{iii} \tag{6.11}$$

v^{iii} is measured and $v_1^{iii} + v_2^{iii}$ ($v_1^{ii} = v_1^{iii}$ and $v_2^{ii} = v_2^{iii}$) is already known. Because conditions in step 2 and 3 are same, just excitation force is added in this step. The difference between operation 2 and 3 is the impact force. Therefore the response of the hull and the response of the shaft in operation 2 and 3 are equal. Therefore the velocity response of the shaft due to impact force is calculated as shown in Equation 6.12.

$$v_3^{iii} = v^{iii} - v_1^{iii} - v_2^{iii} \tag{6.12}$$

After that, the mobility function of the shaft M_{ss} can be calculated as shown in Equation 6.13.

$$Mss = \frac{v_3^{iii}}{F_{impact}} \tag{6.13}$$

Consequently the mobility function of the shaft for this fixed propeller speed is determined. Since the mobility function varies according to propeller speed, step two and three should be repeated for different propeller speeds starting from zero to top speed. Therefore mobility functions for all possible propeller speeds are obtained.

6.3. Calculation of Propeller Force

Calculation of force coming from propeller is explained in this section. And the aim of the thesis is basically to find a way to obtain this force.

The mobility functions of hull and propeller were calculated in previous section. With the help of them, the propeller force can be calculated by the iteration. In step two, v^{ii} was measured and the components of the total response was given in Equation 6.10.

$$v^{ii} = v_1^{ii} + v_2^{ii} \qquad (6.14)$$

Equation 6.14 can be expressed in another form in terms of force and mobility function. v^{ii}, M_{hull} and M_{ss} are known from the experiment steps. Therefore Equation 6.14 becomes:

$$v^{ii} = F_1^{ii} * M_{hull} + F_2^{ii} * M_{ss} \qquad (6.15)$$

The velocity response due to propeller can be expressed separately in terms of force and mobility function as shown in Equation 6.16.

$$v_2^{ii} = F_2^{ii} * M_{ss} \qquad (6.16)$$

Like Equation 6.16, the velocity response of the hull can be expressed separately as shown in Equation 6.17.

$$v_1^{ii} = F_1^{ii} * M_{hull} \qquad (6.17)$$

After writing those equations, iteration can be started. The summarized iteration steps are given in Table 6.1.

Table 6.1 Calculation steps

Step	Process
1	Guessing F_1^{ii}
2	Putting F_1^{ii} in Eq. 6.15 and finding F_2^{ii}
3	Putting F_2^{ii} in Eq. 6.16 and finding v_2^{ii}
4	Putting v_2^{ii} in Eq. 6.14 and find v_1^{ii}
5	Putting v_1^{ii} in Eq. 6.17 and find F_1^{ii}
6	Comparing F_1^{ii} found in Equation 6.17 and guessed one.
7	If F_1^{ii} 's are same, stop the iteration, otherwise continue the process…

When F_1^{ii} is obtained, than F_2^{ii} which is the propeller force can be calculated with the Equation 6.15.

7. VIBRATION EXPERIMENT DESIGN AND MATERIAL SELECTION

7.1. Introduction

In this part of the study, the selection of the accelerometer and impact hammer and the design of auxiliary components of the experiment are explained. The experiment set-up and all other designs and calculations are performed for particular motor yacht in this study.

Shaft vibration experiment has spectacular specifications in motion ship. For instance the vibration amplitude induced from shaft propeller is around 3 m/s^2 in motor yacht considered in the thesis however the vibration amplitude due to the artificial excitation in this experiment for the same yacht is around 0,03 m/s^2. It means the amplitude range of the vibration induced by the impact hammer is quite low from propeller vibration amplitude.

Since vibration amplitude due to the artificial excitation is very low, there is risk to mix the impact excitation with the electrical noise. Because of that, the noise floor of the accelerometer must be low enough.

From the frequency point of view: the shaft vibration frequency of mega yacht is around 0-500 Hz (low speed shaft propeller system, the situation studied here is max. 188 rpm) and the impact hammer excitation is around 0-10 KHz. It means the accelerometer has to have a wide frequency range to catch both vibrations.

7.1.1. The Shaft Line System of Motor Yacht Considered in the Thesis

Motor yacht considered in the thesis is a 162,5 m long, 13000 GT megaycht which is driven by diesel-electric drive. She was built by Blohm Voss Shipyard and launched in 2009. The diesel-electric drive system is based on diesel engine which is connected to the generator. The electricity which is created in generator runs both electric motors for propeller and any other engine and motors on the ship. In this system there is no clutch. The general view of diesel-electric drive and the general information about MY considered in the thesis are given in Figure 7.1 and in Table 7.1 respectively.

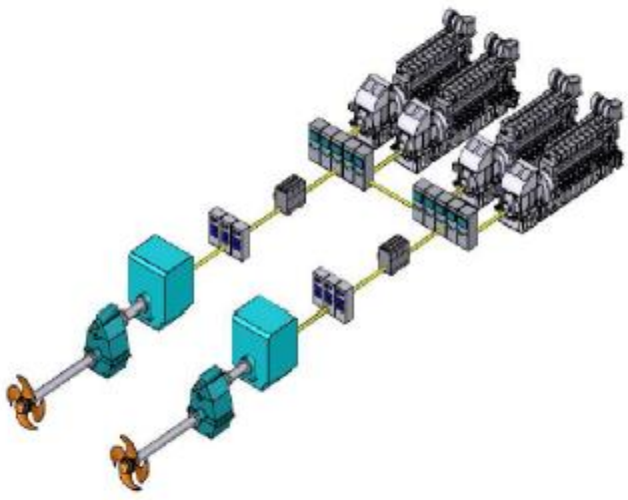

Figure 7.1 Diesel-electric drive general view [30]

Table 7.1 General propulsion properties of MY considered in the thesis

Primitive Motor Type	Synchronous Electric Motor
Primitive Motor Power Rating	6500 kW at 188 rpm
Propeller Speed	188 rpm
Propeller Diameter	3,8 m
Number of Propeller Blades	5
Axial Damper	No
Thrust	407,5 kN

7.2. Application of Vibration Measurement Experiment

At the beginning electromagnetic excitation method was planned to use in this experiment. However many obstacles were realized during the visit of mega yacht which was under construction in Blohm Voss shipyard. There is not enough space to establish a large scale experiment set up in engine room around the shaft. In addition, this difficulty comes into mind immediately when electromagnetic excitation method is wanted to use which has been not tried in a real ship before. The lack of industrial product in the market based on electromagnetic excitation prohibits trusting and using this method as well. Using not proven device in a very expensive vessel is a great risk. This is another reason to suspect of using electromagnetic excitation method.

The other methods like pressurized air or laser cannot generate enough force for the shaft vibration.

Finally, it is decided to use impact hammer in this project even if it is not good and easy way to excite the shaft. On the other hand a technique was developed to minimize its disadvantages and to increase its acceptability. The details of the technique are given below briefly. The detail explanation will be given in next sections.

During the visit, it was also decided to hit the shaft from the flange along the x axis to generate axial vibration. Basically the biggest problem using hammer on turning shaft is security. As it can be seen from the draft drawing of shaft in Figures 7.2 and 7.3, two shafts are connected with forged flanges using several fitted connecting bolts. So hitting 100-200 rpm rotating shaft is not safe by impact hammer.

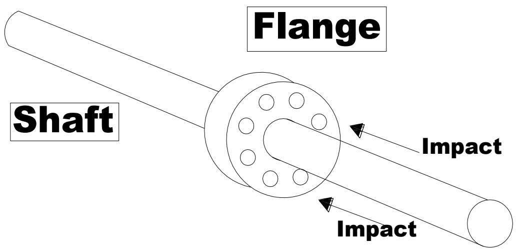

Figure 7.2 Shaft overview

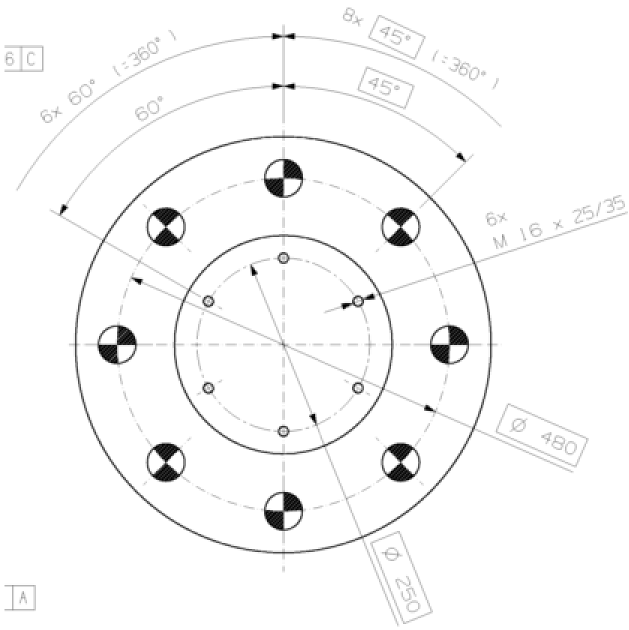

Figure 7.3 The front view of flange

Ensure safety during hitting of a rotating shaft, two half rings can be used to cover the bolted face of the flange which can be seen in Figure 7.4. There are 2 rings which cover the bolted face of the flange. This design is easy to establish and gives a safe surface for the hammer. There is no need to stop the engine for long term and the experiment equipment does not affect the utilization of engine and shaft system too much. Moreover this system (impact hammer) does not need big space around the shaft like other methods (electromagnetic excitation etc.) would.

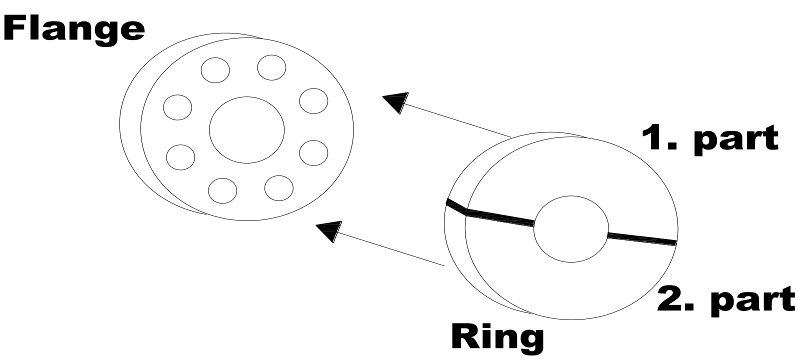

Figure 7.4 3D view of ring to cover the bolts on the flange

7.3. Experiment Set-up

The aim of the experiment is to form the mobility matrices. Therefore the experiment set-up is designed to obtain them.

In this experiment, two impact hammers, accelerometers, ring plates and electronic systems (computer, data acquisition systems) are used. The general view of experiment set-up is illustrated in Figure 7.5.

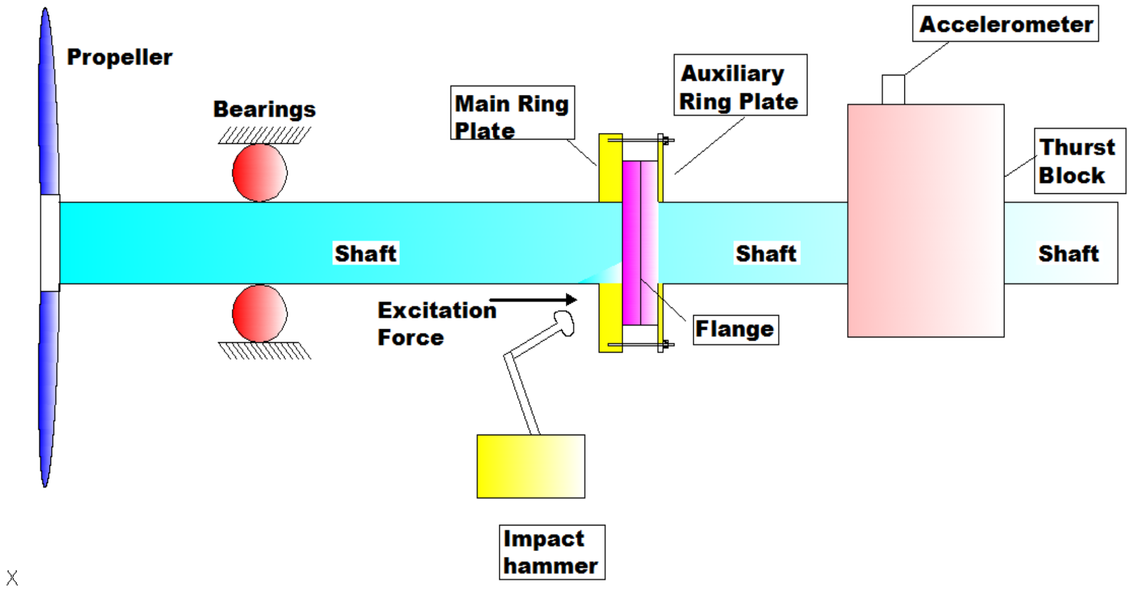

Figure 7.5 General view of experiment set-up

The first step is establishing the experiment set-up. The experiment will be carried out on-going yacht engine room. Establishing experiment set-up and performing should take as short as possible in order not to hamper the yacht daily utilization.

Accelerometers will be attached to the thrust block to measure the axial vibration. Two impact hammers should be used. The first impact hammer will be used for exciting the ship hull and the other one is for exciting the shaft propeller system. Both must be different because of the difference of the mass of structures.

Steps of the experiment which were explained in section 6.1.2 should be performed several times and than be averaged in order to eliminate random errors.

This whole experiment should be repeated for different shaft speeds to obtain the mobility functions of shaft thereby getting propeller force. Because the mobility function is not constant, it changes according to the shaft speed and as a result, the force coming from propeller changes too. Therefore the experiment should be performed for almost every possible shaft speed and than results can be plotted to show the variation of mobility function and force.

7.4 Impact Hammer Selection

In this experiment two impact hammers will be used. First impact hammer should be selected according to force that necessary for the shaft vibration excitation. Meanwhile the force amplitude must be inside the accelerometer ranges. Moreover the size of the hammer and tip specifications should meet the experiment conditions and requirements.

Second impact hammer is for exciting the hull. Blohm&Voss has one already and the specifications of this impact hammer are given in Table 7.2. This impact hammer has super soft, soft, medium and hard tips. Therefore many different force variations can be applied.

Table 7.2. Specifications of impact hammer for hull

Frequency Range	0,5 kHz
Hammer range	22.000 N
Hammer Sensitivity	0,23 mV/N
Resonant Frequency	2,7 kHz
Hammer Mass	5,4 kg
Head Diameter	7,6 cm
Tip Diameter	7,6 cm

7.4.1. Descriptions and Limitations

An impact hammer is basically a hammer with a force transducer which measures the force applied to the structure during the strike. There are many different types of tips which are used for different purposes; metal, plastic, rubber etc. Hard plastic and metal tips have higher stiffness; on the other hand rubber tips provide reduced stiffness. [18]

When the hammer hits the structure, energy transfer to the structure occurs. Transferred amount of energy is the function of mass and the velocity of the hammer. Therefore the main aspect of selecting an impact hammer is the size and the mass of the hammer. Hammer selection should be done according to both sizes and the condition of the structure from the very large variety of hammers. There are many different hammers in weights from a few grams to several kilograms. If the hammer is selected bigger than required, it may cause the structure to move non-linear behavior which is not acceptable for modal analysis.

The bandwidth of excitation depends on the mass and tip stiffness. Stiffer tips have a wider frequency range and softer tips have a lower frequency range. However while stiffer tips spread the input energy over this wider range, softer tips concentrate the energy over a lower frequency range. [18] A sample comparison is given in Figure 7.6

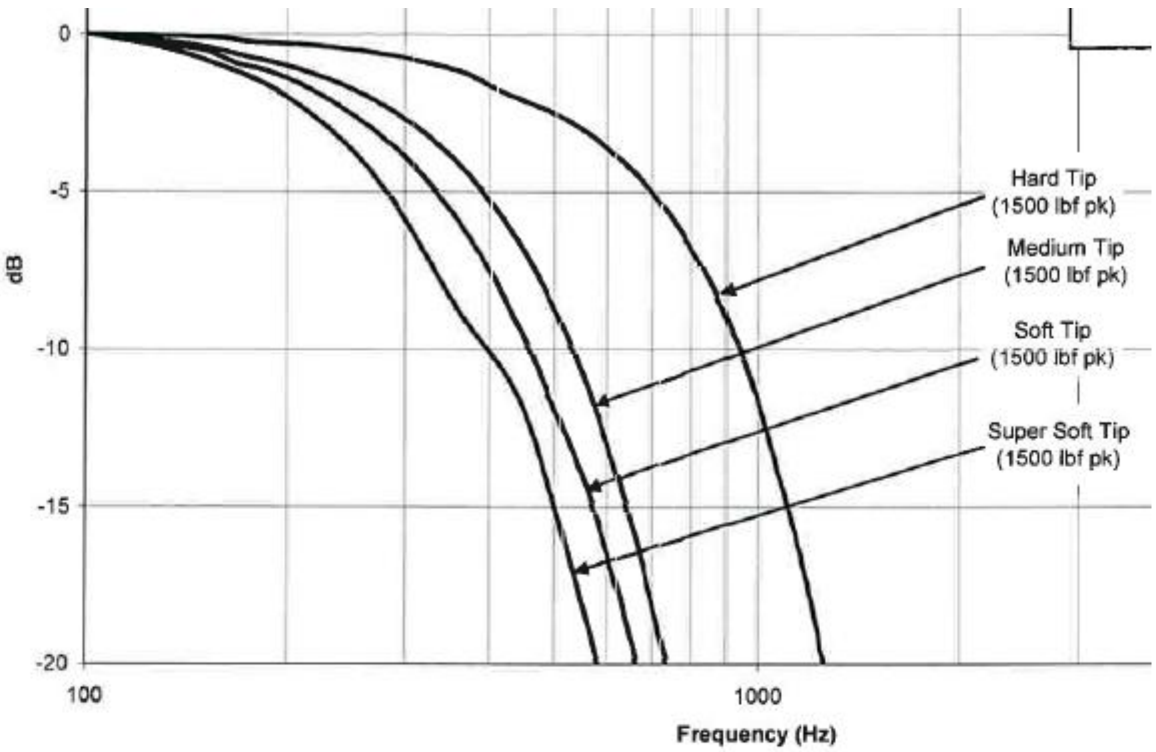

Figure 7.6 Comparison of the tips according to frequency range [31]

One of the drawbacks of the impact hammer is repetition. Obtaining same amount of energy and the same hitting location cannot be easily delivered by the impact hammer. This is one of the main obstacles of this thesis.

Because of the nature of the impact hammer excitation, it excites the structure with many frequencies at the same time. But with the help of Fourier analysis, they can be separated from each other. [18]

Excitation force from the hammer is very short comparing to the length of the experiment. Therefore the part of the signal after the force pulse is noise and should be extracted from the record. This can be done by windowing. [32]

Consequently impact hammer method requires short time and less devices, computer etc. Therefore it is an easy and cheap solution for the modal analysis.

7.4.2. Automated Impact Hammer

An electric impact hammer or automated impact hammer is basically an impact hammer which is controlled automatically to ensure impact quality. This kind of systems have a control unit, a hammer which is mounted to the control unit, an AC/DC power supply and connection cables. There are some individual researchers and companies in the market to obtain automated impact hammer. Norman et al. [33] and Suprock [34] are the researchers who have worked on automated impact hammers and one of the commercial products is TMS model 086M92ES electric impact hammer.

The advantages of using automated hammer are:

- reducing the effort of impact repetition
- obtain the same force and frequency values in every impact
- reducing the time for performing experiment

The main elements of automated impact hammer are [33]:

- actuator to create force for the impact
- an impact tip
- a control circuit to manage the actuator
- auxiliary components

The biggest drawback of impact hammer is repeatability. In every application, force range and the application location may be different. With the help of automated impact hammer, same amplitude of force can be applied several times at the same point. Different types of impact tips allow the user to control the impact’s shape and frequency as well. [35]

The TMS Model 086M92ES includes foot pedal trigger with integral cable, hammer, AC/DC power supply and sensor cables. Modal analysis, automated impact testing, FEA model verification can be done using this impact hammer. [35]

7.4.3. Selection of Automated Impact Hammer

When an impact hammer which can generate 2000N is used, than the acceleration occurred on the shaft due to impact hammer can be calculated with the help of Newton’s law as shown below. It is of course roughly calculation but gives first and sufficient estimation.

$$F=m*a \tag{7.1}$$

$2000=69069*a$

$a=0{,}029\ m/s^2$

This amount is very low relative to the acceleration coming from propeller. Therefore the accelerometer selection is getting important. Selection of accelerometer is done in section 7.5.

The impact hammer is selected TMS model 086M92ES which is electric impact hammer and generates enough amount of force for exciting the structure. The specifications of commercial product TMS model 086M92ES and general view of it are given in Table 7.3 and Figure 7.7 respectively.

Table 7.3. Electric impact hammer specifications [35]

Model	**Force Range (N)**	**Sensitivity (mV/N)**	**Max. impact rate**	**Weight (kg)**	**Overall Dimensions (LxWxH) cm**	**Producer**
086M92ES	220 -2200	1,1	2 impacts per second	7,7	40,5 x 18 x 11,5	TMS

Figure 7.7 Electric impact hammer (TMS Model 086M92ES) [35]

7.5. Accelerometer Selection

The vibration generated by the propeller is very high comparing the vibration generated by impact hammer excitation force. In other words, amplitudes of forces coming from propeller are much higher than force coming from the impact hammer. Therefore these vibration amplitudes should be estimated and accelerometers should be selected to cover this wide range of movement.

The range of the accelerometer has a crucial role in vibration measurement. If a low range accelerometer is selected, than the vibration created by the propeller will be out of range. On the other hand if a high range accelerometer is selected, the vibration from the impact hammer cannot be catched by the accelerometer. So in this experiment, it is necessary to find an accelerometer which has wide range of amplitude, high sensitivity, low noise level and high frequency level at the same time.

In Lloyd’s Register Ship Vibration and Noise Guidance Notes, the values below are given for the limit of machinery vibration. [36]

“Assessment Overall peak amplitudes for vibrations in the frequency range 2 to 100 Hz should be less than each of the following values:

• *displacement* ± *0,4 mm*

• *velocity ± 25 mm/s*

• *acceleration ± 40 m/s^2 .”*

In the axial vibration calculation of MY considered in the thesis, half range vibratory reaction force amplitude on the thrust block, maximum is 20 tones (speed at 188 rpm). At this speed permissible vibratory level is 40 tones. And the total mass of the shaft structure is given in the Table 7.4.

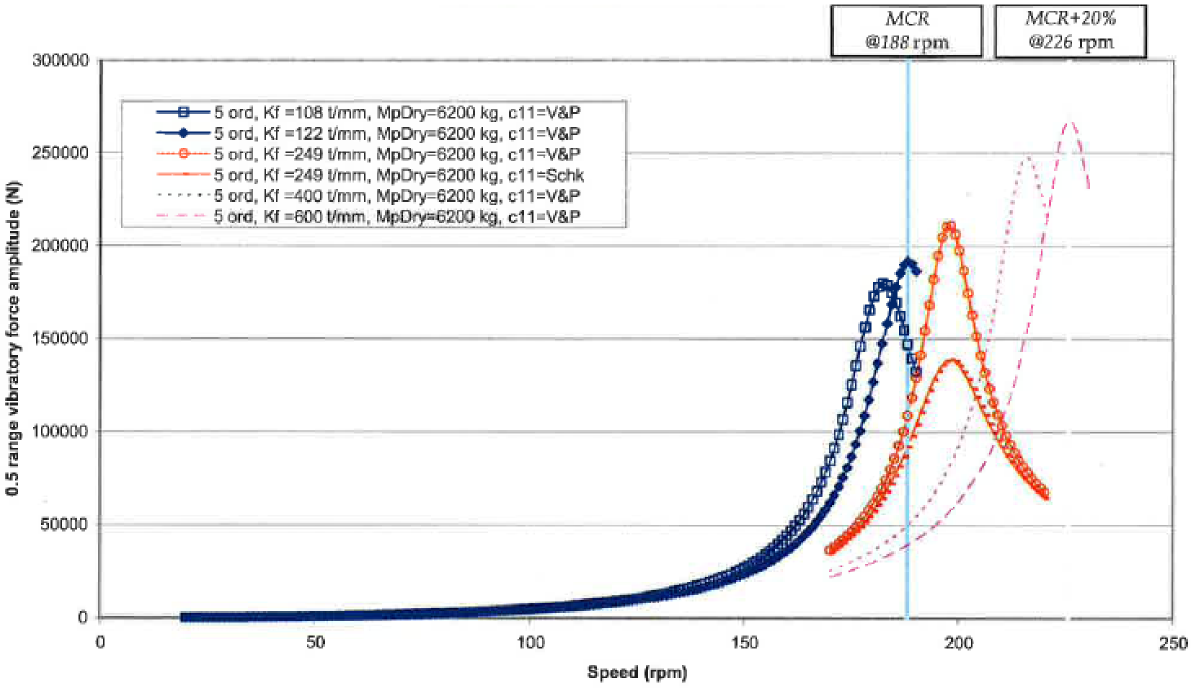

Figure 7.8 Axial vibratory force according to speed in MY considered in the thesis

Table 7.4 Weight of shaft propeller structure

Part	Weight (kg)
Shaft line + propeller	34470
Propeller added mass	3699
Thrust shaft	2100
Thrust block	2690
Fitting ring connected to the thrust shaft	110
Rotor	26800
Total	69069

The rotor is rigidly connected to the shaft in motor yacht considered in the thesis. Therefore it is also considered in weight calculation. Since the experiment will be performed during the sailing, the added mass of the propeller is taken into account as well. As a first and rough estimation the acceleration occurred in the x-axis of the shaft can be found using Equation 7.1.

*200.000 N =69.069*a*
a=2,8956 m/s^2

This is roughly but useful calculation to estimate the range of accelerometer. Consequently accelerometer which has range ±10 m/s^2 can be used in this situation. And the sensitivity must be less than 0,029 m/s^2 to avoid the interference of electrical noise and the acceleration coming from impact hammer. There are some options from different producers in Table 7.5.

Table 7.5 Specifications of some accelerometers

Model	**Range (g)**	**Noise Floor (g) (rms)**	**Rated Output (Sensitivity) (mV/g)**	**Frequency Range (Hz)**	**Producer**	**Approx. Price**
ACC103	±500	0,007	10	2-10000	Omega	365 $
ACC101	±70	0,00025	100	3-5000	Omega	265$
4508B-002	±70	0,00035	1000	0,3 -8000	Brüel Kajaer	
333B	±50	0.00007	100	2-1000	PCB	
340A16	±500	0.0006	1	1-12000	PCB	
352A56	±50	0.0006	100	0,5-10000	PCB	
352B	±5	0.00008	1000	2-10000	PCB	
3056B6T	±25	0,0003	200	1-10000	Dytran	
3056B3T	±10	0,0004	500	1-10000	Dytran	

g=9,81m/s2

The sensitivity of the accelerometer should cover lowest vibration level. It is recommended to use low sensitivity accelerometer to measure high amplitude vibrations and high sensitivity accelerometer to measure low amplitude vibrations. [37] However in this situation, the vibration of the impact hammer and the vibration induced propeller are quite far away from each other. There are high and low amplitude vibrations to be measured at the same time. Therefore high sensitivity and high amplitude accelerometers should be used.

The frequency range is another important parameter for the selecting of an accelerometer. This range should cover the frequencies occured during the experiment. Another crucial point is the resonance frequency of the accelerometer. This value should be far from both impact hammer and structures vibration frequency. The resonance frequency of the accelerometers here are quite big (around 30 kHz). So there is no danger regarding resonance for the experiment.

It is recommended not to use an accelerometer which has weight bigger than 1/10 of the test object. [38] The weight of the accelerometers chosen for this experiment is around 10 gr. (Dytran3056B3T) which are quite small comparing to shaft system. Therefore there is no need to consider the weight effect of the accelerometer to the system.

There are nine different accelerometers are listed as an example in Table 7.5. The first one (Omega ACC103) is not acceptable because of its noise floor. It is very close to the impact hammer vibration acceleration. Therefore impact hammer vibration may mix with noise. Brüel Kajaer 4508B-002 is also not acceptable because of its frequency range. It is very low for impact hammer excitation. The most suitable ones are Dytran3056B3T and PCB 352B. Their all parameters are in the desired range.

7.6. Auxiliary Components of the Experiment

This part of the study explains the practical utilisation of the impact hammer. Because of the rotating nature of the shaft, conventional impact hammer cannot be used. Moreover even electric impact hammer needs modifications and some additional devices.

To use impact hammer properly there are two acceptable approaches developed in this study. In first one, the bolted face of the flange is covered by plate to obtain flat surface for hammer impact. It is explained in section 7.6.1.

In second one, the impact hammer is attached on the shaft itself. Therefore there is no need to cover the flange. However it is necessary to use power supply and remote control devices to manage the impact hammer. It is explained in section 7.6.2.

7.6.1. Ring Plate Option

The flange which was used in motor yacht considered in the thesis is shown in Figure 7.9. The outer diameter of the flange is 600 mm and the inner diameter of the flange is 300 mm. There are 8 M52x5 bolts for connecting shafts.

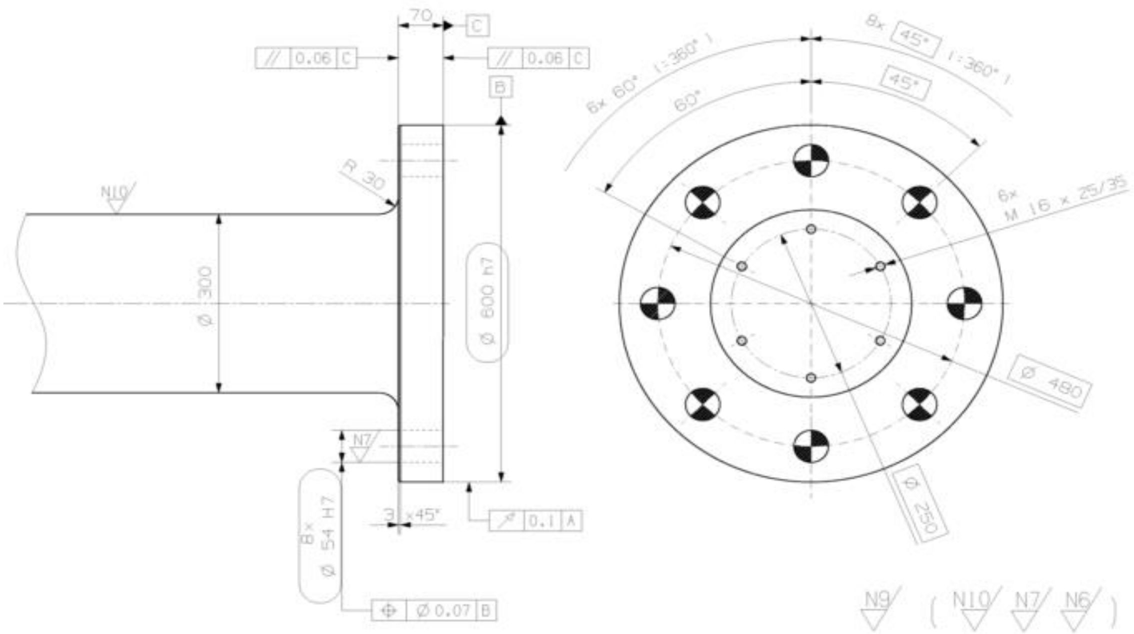

Figure 7.9 Shaft-flange drawing of MY considered in the thesis

There are 8 bolt-nut couples on the flange which inhibit hammer impact testing. It is necessary to avoid such kind of protrusions before performing the impact vibration test. And the purpose of this ring plate is obtaining smooth and flat surface for impact hammer hit. As shown in Figure 7.10, the length of the part of the bolt which is outside the flange is 98 mm (left hand side of drawing). And the diameter of the washer and nut is 104 mm.

To cover this face of the flange, there can be plate which has 8 holes for the bolt nut couples. The dimensions of hole at the ring plate must be at least 105 mm in wide and 99 mm in depth.

Thickness of the ring plate will be 105 mm which has 6 mm extra thickness after the holes for the nut-bolt couples.

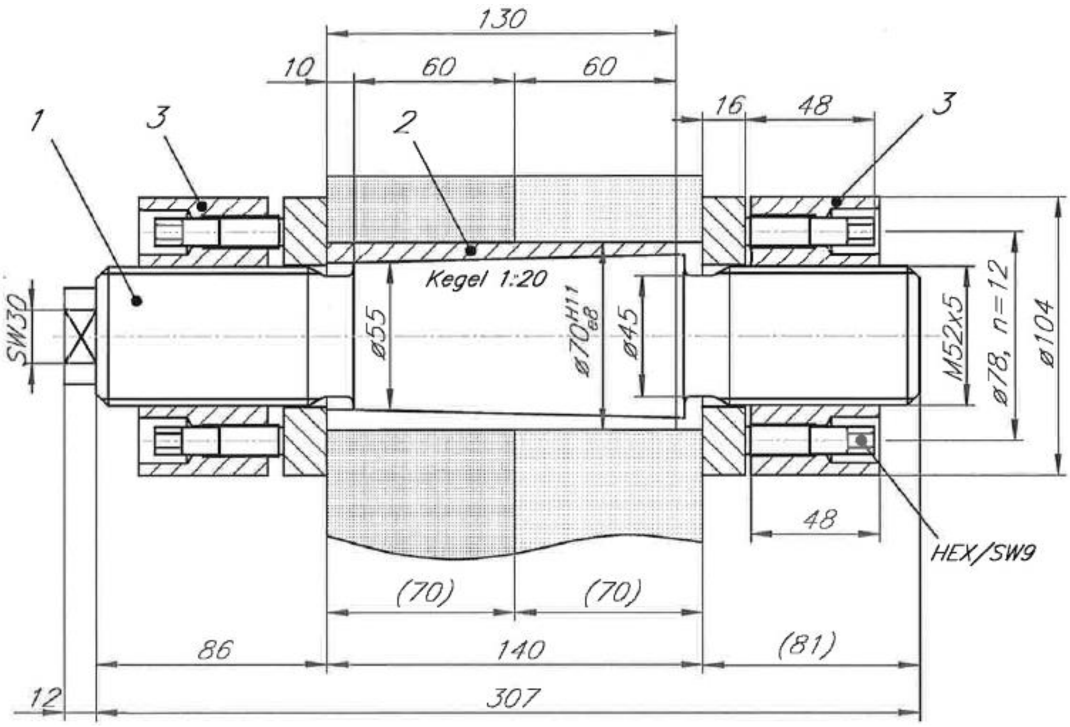

Figure 7.10 Bolt-flange connection drawing of MY considered in the thesis

To allow mounting of the ring, this will be split into two pieces which will be connected by bolts. Fixation in x-direction is realized by clamping using a second ring on the opposite flange and connecting bolts. So it becomes very tight, strong and reliable surface. The assembly drawing of the system is given in Figure 7.11.

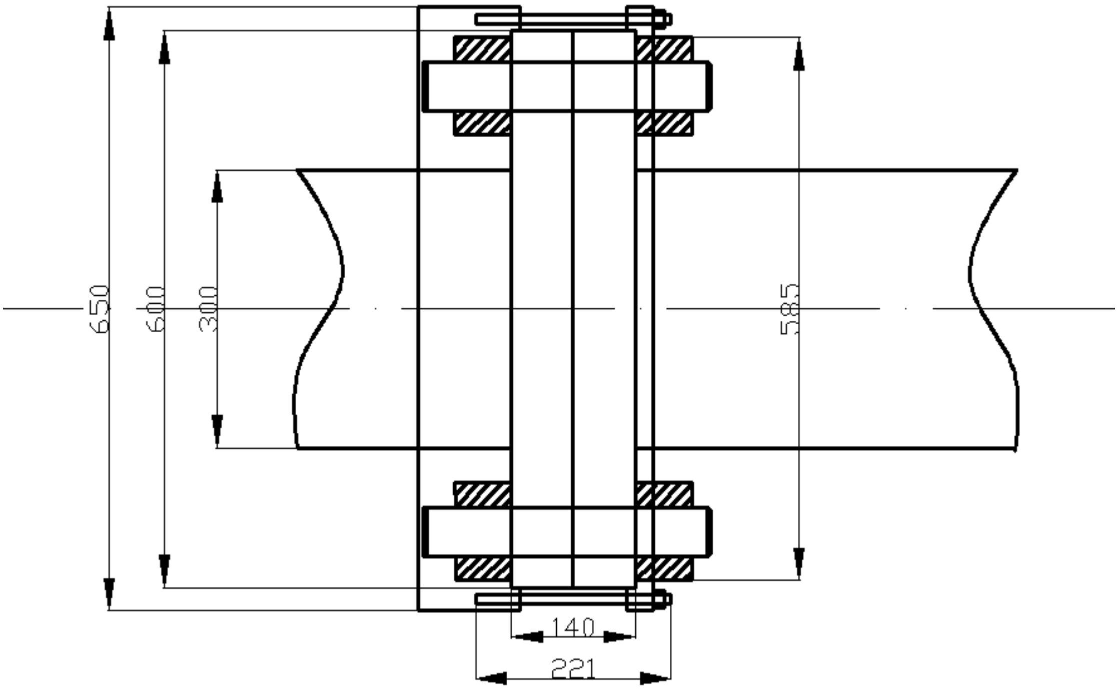

Figure 7.11 Assembly drawing of whole system

7.6.1.1. Design of Main Ring Plate

As it can be seen from Figure 7.9, the diameter of the flange is 600 mm and the diameter of the shaft is 300 mm in this example. Therefore dimensions of the ring plate that will be used here to cover the bolted face of flange must be at least same or bigger than those values. 650 mm for outer diameter and 300 mm for inner diameter were selected. The ring will be manufactured from two half rings, which will be connected with M10 bolts.

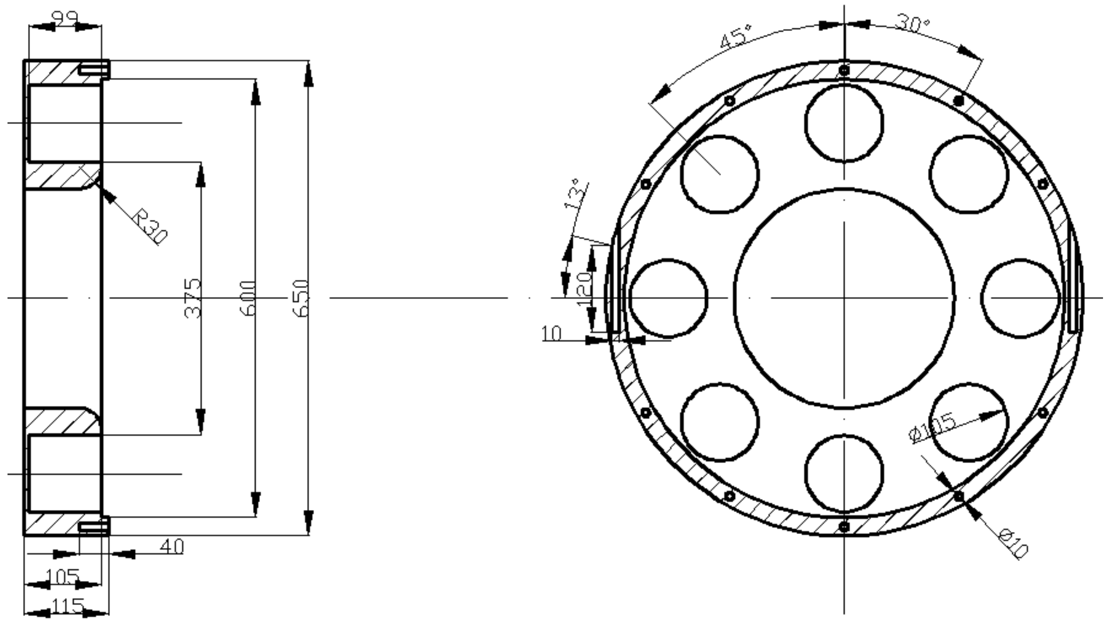

Figure 7.12 Drawing of main ring plate

Drawing of the main ring plate and the dimensions of the holes are given in Figure 7.12 and Table 7.6 respectively.

Table 7.6 Dimensions of holes on the ring plates

Holes	Location	Condition	Diameter (mm)	Depth (mm)
For the bolt-ring couples	Main ring plate	Blind hole	105	99
For the bolts to connect main and auxiliary ring plates	Main ring plate	Blind hole	10	40
For the bolts to connect main and auxiliary ring plates	Auxiliary ring plate	Open hole	10	20

There are five blind holes at outer part of each halves for the bolts to connect two ring plates (main and auxiliary rings). The dimensions of these holes are given in Table 7.6 as well.

There is no information about the head diameter of the electric impact hammer in its catalogue, but in the literature the head diameter of the 2000 N impact hammer is around 1,6 cm. And there is 17,5 cm wide place each side of the ring for impact hammer. Therefore the clearance on the flange is quite enough for the hammer hit. The general dimensions of main ring plate are given in Table 7.7.

Table 7.7. Main ring plate specifications

Plate Inner Diameter (mm)	300
Plate Outer Diameter (mm)	650
Plate Thickness (mm)	105
Material	C45
Weight of the plate ring (kg)	160

In this experiment, the impact hammer will be applied at the surface of the ring plate. When the impact hammer hits the ring plate a moment act on the shaft flange. The amount of the moment is calculated as shown in Equation 7.2.

$$M = F * x \tag{7.2}$$

*M= 2000 (N)*0,325 (m)*
M= 650 Nm

It is assumed the worst case (hitting at the edge of the ring plate). With a moment lever of 325 mm, the moment for the shaft system is negligible.

7.6.1.2. Design of Auxiliary Ring Plate

The purpose of this ring plate is fixing the main ring plate in x-direction. Therefore it is lighter and smaller than the main ring plate. It consists of two half rings similar to main ring plate without holes for the bolt-nut couples. But there are five holes at the outer part on each half ring for the connection with the main ring halves. The ring will be manufactured from two half rings, which will be connected with M10 bolts like previous design. The drawing and the specifications are given in Figure 7.13 and Table 7.8 respectively.

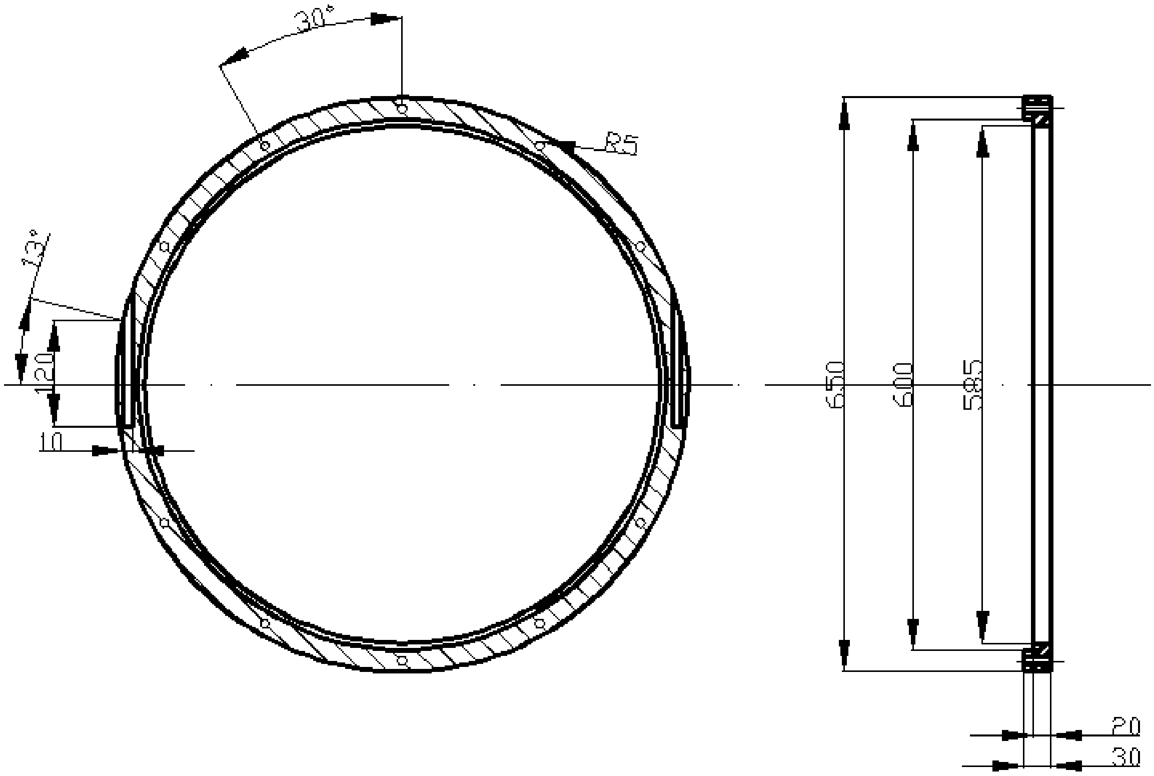

Figure 7.13 Drawing of auxiliary ring plate connection

Table 7.8 Auxiliary ring plate specifications

Plate Inner Diameter (mm)	585
Plate Outer Diameter (mm)	650
Plate Thickness (mm)	20
Material	C45
Weight of the ring plate (kg)	15

Total weight of the ring plates (main and auxiliary) is 175 kg and this weight is not considered to affect the shaft system in operation. Material list for the ring plates connection in this experiment are given in Table 7.9

Table 7.9 Material list of the ring plates

	Number of items
Ring Plate Halves Ø (650-300)	2
Ring Plate Halves Ø (650-585)	2
M10 Bolt	14

7.6.2. Remote Control Option

In this method, the electric impact hammer is attached on the shaft. The hitting point on the flange is fixed and the impact hammer will turn with shaft. Therefore impact hammer working risks on turning shaft are minimized.

When the impact hammer is attached on the shaft, it is mandatory to use power supply and remote control. Because there is no way to manage the hammer located on turning shaft by cables.

The impact hammer is controlled by remote control unit. In this study it is planned to use remote control momentary switch. There is also one battery to supply power to both impact hammer and remote control unit. The general working scheme of the remote control system is given in Figure 7.14.

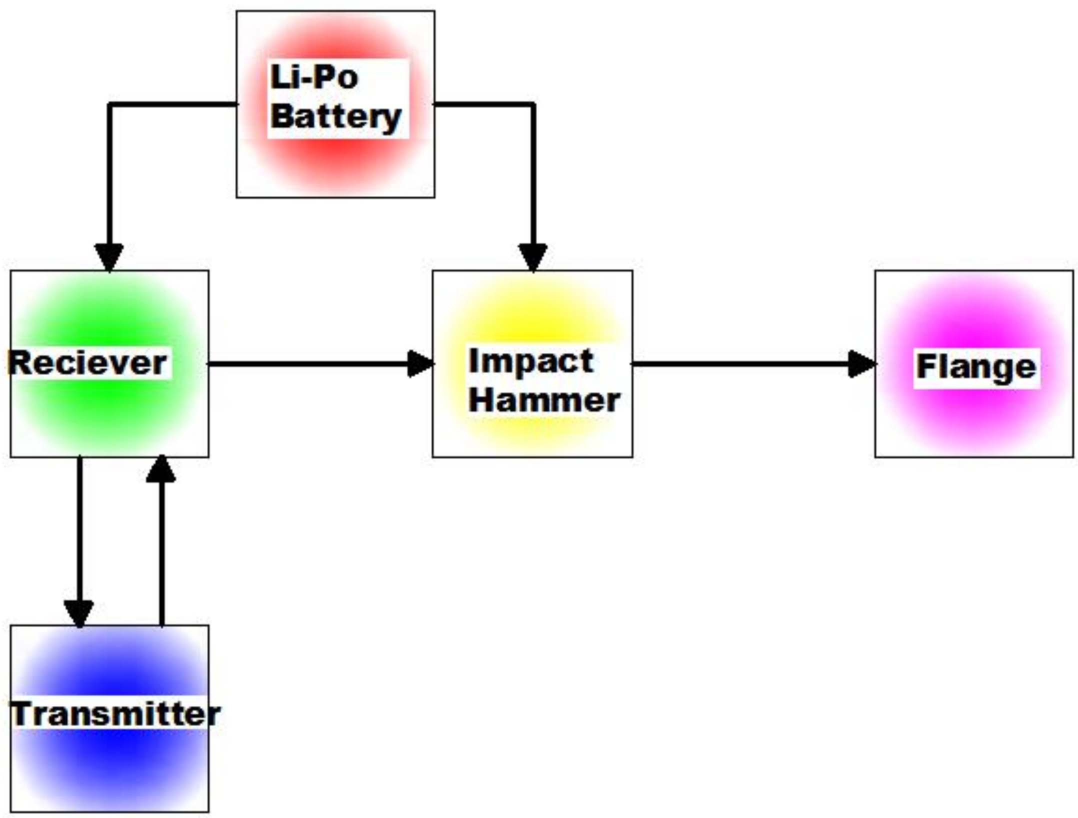

Figure 7.14 The layout of the remote control

7.6.2.1. Remote Control Momentary Switch Selection

Remote control momentary switch is used to control the impact hammer. This device will be installed instead of foot pedal trigger which manage the actuator in electrical impact hammer. It controls the impact hammer like on/off switch. As long as the operator presses the button, it keeps the circuit close position and impact hammer works.

It consists of transmitter and receiver. Operator uses the transmitter to control the receiver and the receiver manages the actuator of the impact hammer.

The theoretical influence distance of such devices vary but generally 24 V ordinary device has 100m range which is quite enough in this experiment. The size of such devices is like 75mm x 55mm x 28mm small box and the weight is not bigger than 2 kg.

The working scheme of the remote control momentary switch is presented in Figure 7.15.

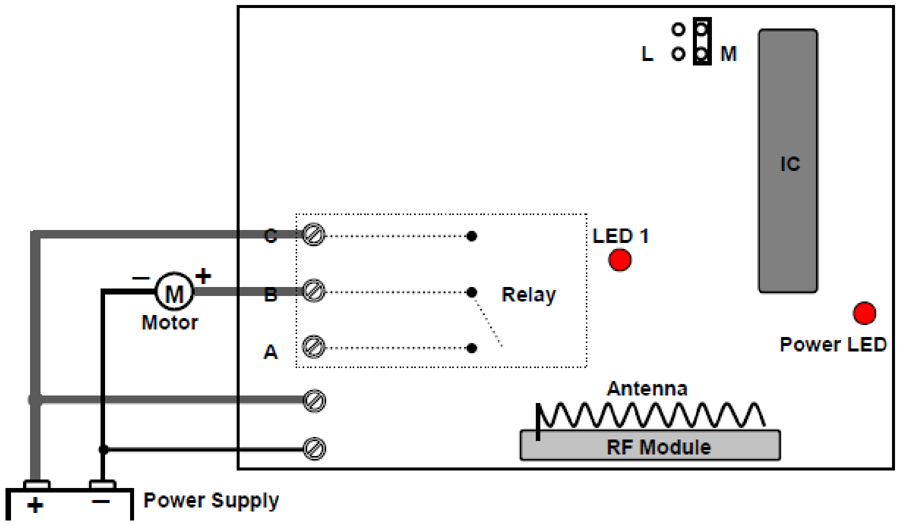

Figure 7.15 Working scheme of the remote control momentary switch [39]

7.6.2.2. Power Supply Unit Selection

Since the remote control unit and the impact hammer is on rotating shaft, they are away from any power supply or electric socket. Therefore the power supply must be on the shaft. Conventional lead-acid batteries cannot be used because of rotating shaft in this experiment.

There is always risk to loose the acid while rotating. So lithium polymer (Li-Po) batteries are decided to use. They are light comparing to lead-acid batteries and dry.

There are many different types of Li-Po batteries and brands on the market. Specifications of usual 24 V Li-Po battery are given in Table 7.10. They are rechargeable; therefore they can be used several times.

Table 7.10 General LI-Po Battery Specifications

Type	**Li-Po**
Nominal voltage	24 V
Nominal capacity	20 Ah
Operation temperature	-20-60 C
Size (length x width x height)	160mm x 250mm x 90mm (Prismatic)
Weight	Around 5 kg
Cycle life	Around 2000 times

7.6.3. Placement of the Devices

The weight of the electric impact hammer device is 7,7 kg, the remote control momentary switch is around 2 kg and the Li-Po battery is 5 kg. In order to eliminate unbalance problem due to weights on the shaft, impact hammer must be placed one side of the shaft, remote control momentary switch and Li-Po battery must be placed on the other side of the shaft. The arrangement of the devices is given in Figure 7.16.

It is planned to use two trays to carry the impact hammer, remote control unit and battery which have a same radius with the shaft at the bottom and a flat surface at the top. After that these trays are connected to the shaft by kind of hose clamps.

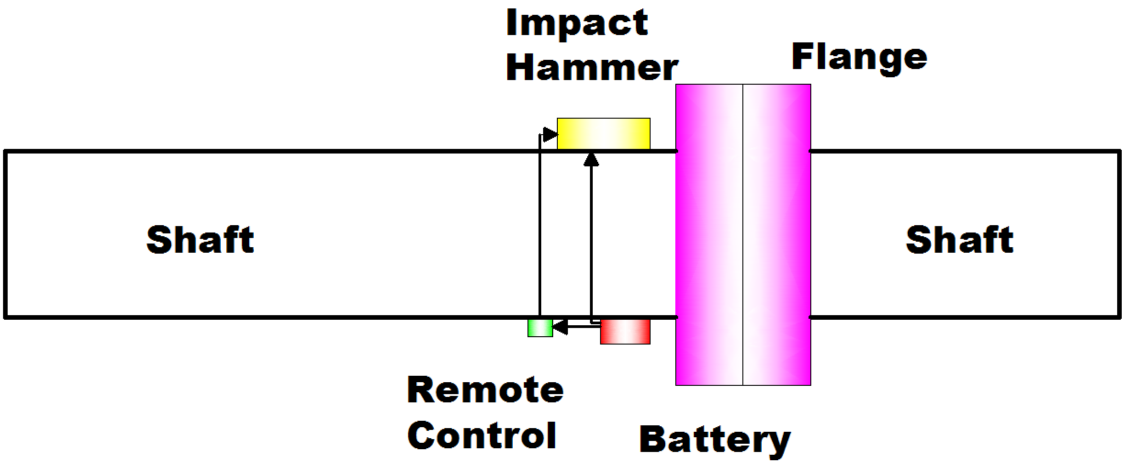

Figure 7.16 Placement of the devices on the shaft

8. CONCLUSION

In this study a new experimental method to determine forces coming from propeller was developed. Moreover a unique calculation technique to obtain mobility function and forces was studied in accordance with the experiment.

The method is based on finding the mobility function. After finding the mobility function, force coming from the propeller can be found easily. The experiment and calculation technique were developed theoretically. Therefore it is necessary to prove the method. However there was no available yacht at the shipyard during the thesis. For this reason, next step should be proving the thesis on a real yacht.

The measurement is strictly related to the shaft line layout. The weight and the dimensions of the shaft and the propeller affect the material selection. Therefore before the measurement, shaft propeller layout should be carefully examined. Impact hammers, accelerometers and auxiliary components will be selected and designed according to the layout.

Two different methods were developed to obtain flat surface on the flange for the impact hammer hit. Both have some advantages and disadvantages and most suitable one should be selected.

Lastly the outline of the whole experiment is given in Table 8.1 to briefly indicate the steps of the thesis.

Table 8.1 Outline of the Experiment

	Operation	**Notes**
Step 1	The shaft line layout is examined.	The experiment is strictly related to the shaft propeller layout.
Step 2	The weight of the shaft line and other components which belong to the test structure are calculated.	- Which components will be considered in weight calculation is decided. - The weights of the components can be found

		from the engine room drawings
Step 3	Impact hammer is selected according to the experiment, hammer contact surface etc.	There are many types of hammer in the market. The most suitable one must be selected.
Step 4	Acceleration due to the impact hammer is predicted.	This is a rough prediction but it is necessary to select accelerometers.
Step 5	Acceleration due to the propeller is predicted.	
Step 6	Accelerometers are selected according to steps 4 and 5.	The acceleration range is obtained from step 4 and 5 and parameters like sensitivity, frequency range, noise level are decided in this step.
Step 7	Selection of method for auxiliary components of the experiment	It can be either ring plate or remote control method.
Step 8	According to selection of auxiliary components, the material design should be performed.	Especially for ring plate method, the dimensions of the plates vary depending on shaft size.
Step 9	Experiment is performed for every shaft speed.	In order to minimize errors every experiment should be done 5-10 times.
Step 10	Acceleration responses are obtained.	Velocity responses are derived by integration of accelerations.
Step 11	Mobility functions are calculated.	They are different for every shaft speed.
Step 12	Mobility function – shaft speed graph is created.	This graph gives the variation of mobility function according to shaft speed visually.

Step 13	Forces coming from propeller are calculated for every shaft speed.	- This shows the magnitude of the force that creates vibration. - Making provision against vibration according to these force levels. - Obtaining of these forces is the aim of this thesis.

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APPENDIX

SHAFT VIBRATION MEASUREMENT EXAMPLES FROM LITERATURE

1. Bentley Nevada Design [40]

Design of displacement sensor manufactured by the Bently-Nevada Corporation uses electromagnetic eddy current technology to sense the distance between the probe tip and the rotating machine shaft. The sensor itself is an encapsulated coil of wire, energized with high-frequency alternating current (AC). The magnetic field produced by the coil induces eddy currents in the metal shaft of the machine, as though the metal piece were a short-circuited secondary coil of a transformer (with the probe's coil as the transformer primary winding). The closer the shaft moves toward the sensor tip, the tighter the magnetic coupling between the shaft and the sensor coil, and the stronger the eddy currents.

The high-frequency oscillator circuit providing the sensor coil's excitation signal becomes loaded by the induced eddy currents. Therefore, the oscillator's load becomes a direct indication of how close the probe tip is to the metal shaft. This is not unlike the operation of a metal detector: measuring the proximity of a wire coil to any metal object by the degree of loading caused by eddy current induction.

In the Bently-Nevada design, the oscillator circuit providing sensor coil excitation is called a proximitor. The proximitor module is powered by an external DC power source, and drives the sensor coil through a coaxial cable. Proximity to the metal shaft is represented by a DC voltage output from the proximitor module, with 200 millivolts per mil (1 mil = 1/1000 inch) of motion being the standard calibration.

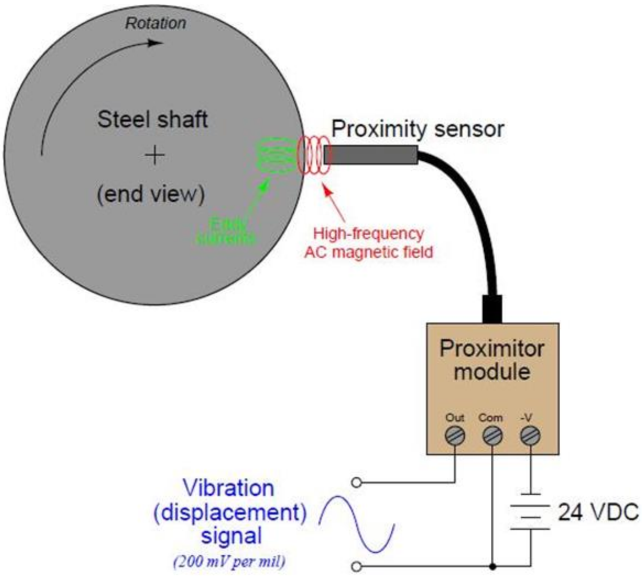

Figure A1.1 General layout of the design

Since the proximitor's output voltage is a direct representation of distance between the probe's tip and the shaft's surface, a "quiet" signal (no vibration) will be a pure DC voltage. The probe is adjusted by a technician such that this quiescent voltage will lie between the proximitor's output voltage range limits. Any vibration of the shaft will cause the proximitor's output voltage to vary in precise step. A shaft vibration of 28,67 Hz, for instance, will cause the proximitor output signal to be a 28,67 Hz waveform superimposed on the DC "bias" voltage set by the initial probe/shaft gap.

An oscilloscope connected to this output signal will show a direct representation of shaft vibration, as measured in the axis of the probe. In fact, any electronic test equipment capable of analyzing the voltage signal output by the proximitor may be used to analyze the machine's vibration: oscilloscopes, spectrum analyzers, peak-indicating voltmeters, RMS-indicating voltmeters, etc. It is customary to arrange a set of three displacement probes at the end of a machine shaft to measure vibration: two radial probes and one axial (or thrust) probe. The purpose of this triaxial probe configuration is to measure shaft vibration (and/or shaft displacement) in all three dimensions:

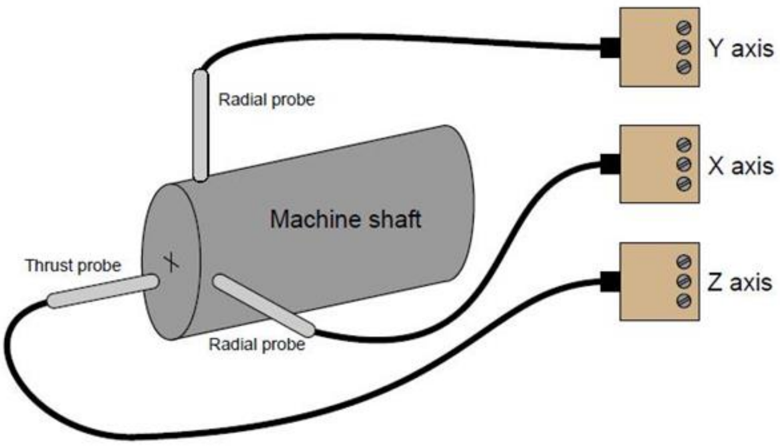

Figure A1.2 Probe orientation

It is also common to see one phase reference probe installed on the machine shaft, positioned in such a way that it detects the periodic passing of a keyway or other irregular feature on the shaft. The “keyphasor” signal will consist of one large pulse per revolution:

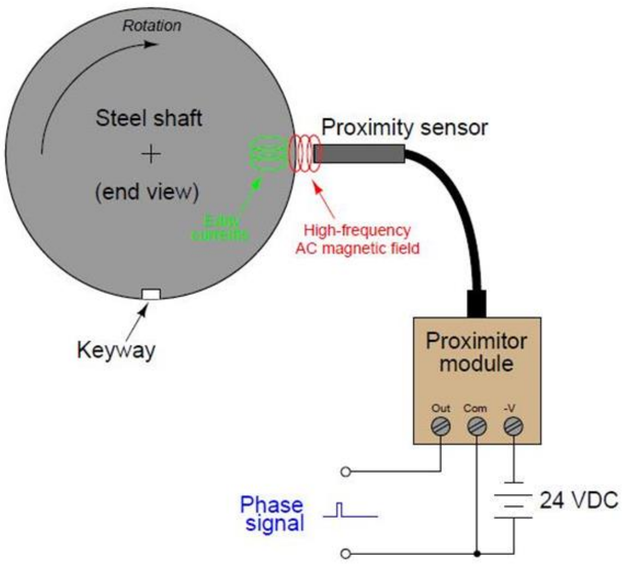

Figure A1.3 Measurement with keyway

The purpose of a keyphasor signal is two-fold: to provide a reference point in the machine’s rotation to correlate other vibration signals against, and to provide a simple means of measuring shaft speed. The location in time of the pulse represents shaft position, while the frequency of that pulse signal represents shaft speed.

For instance, if one of the radial displacement sensors indicates a high vibration at the same frequency as the shaft rotation (i.e. the shaft is bowed in one direction, like a banana spinning on its long axis), the phase shift between the vibration's sinusoidal peak and the phase reference pulse will indicate to maintenance machinists where the machine is out of balance. This is not unlike automatic tire-balancing machines designed to measure imbalance in automobile tire and wheel assemblies: the machine must have some way of indicating to the human operator where a balancing weight should be placed, not just how far out of balance the tire is. In the case of machine vibration monitoring equipment, the keyphasor signal and one of the axial displacement signals may be simultaneously plotted on a dual-trace oscilloscope for the purposes of determining the position of the imbalance on the machine shaft.

2. Turbine Shaft Example [17]

The eddy current probe, as well as providing ac vibratory information, also provides dc information of the probe to target gap. This makes it ideal for measuring rotor to casing differential expansion via a non-contact method. The eddy current probe and the measurement of differential expansion are governed by a series of empirical relationships. The linear measurement range of an eddy current probe is approximately one third of its coil diameter as shown earlier. The ideal flat target area for an eddy current probe to "observe" is twice the coil diameter. Therefore the ideal target size for an eddy current probe is six times the linear measurement range.

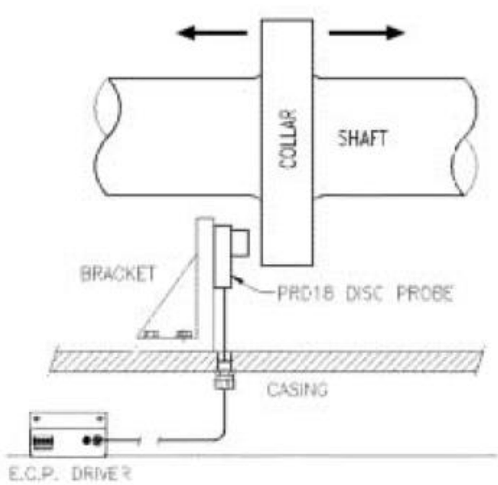

Figure A1.4 Shaft collar and the experiment set-up

For differential measurement ranges of 25mm a target of 100mm is therefore required. This large target size is often impractical to fit. It is also often the case when retrofitting differential expansion systems that the existing collar is much smaller than that ideally required. The illustrations opposite show the effect of a less than ideal target on the output of an eddy current probe. To overcome this problem and obtain a linear output the probe electronics can either be calibrated in-situ or supplied pre-calibrated with a non-linear output. This non-linear output becomes linear when the probe is fitted in-situ. Eddy probe drivers are normally pre-calibrated to give a linear output when observing an ideal target.

The diagram opposite illustrates a disc type eddy current probe measuring movement against a flat collar; the limitations in terms of target area can clearly be seen.

One option to overcome this limitation is to reduce the size of the probe and therefore obtain a more linear output against the fixed target area, in combination with measuring both sides of the collar. However, this push –pull technique does require some simple arithmetic within the signal conditioning units to generate the correct expansion measurement.

Another technique utilised in measuring differential expansion is to use tapered rather than flat collars. The use of tapered collars fitted to the turbine shaft enables longer linear ranges to be obtained. A 1 in 10 taper enables an axial expansion of 10 times the normal range of the probe to be measured.

A problem arises however if there is any radial movement eg if the shaft moves 100 micrometers within the bearings, this is incorrectly seen as 10 x 100 micrometers (ie 1mm) of differential expansion. To overcome this, two eddy probes are fitted. Thus two unknowns can be easily solved by two simultaneous equations through software manipulation.

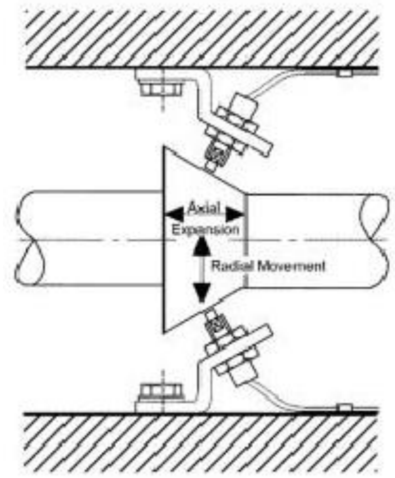

Figure A1.5 Two monitors monitoring expansion by observing a tapered collar

A further complication arises when the casing holding the eddy probes is subjected to twisting as can happen if slides start to stick (see below). A further two eddy current probes are then required to give a correct reading of differential expansion.

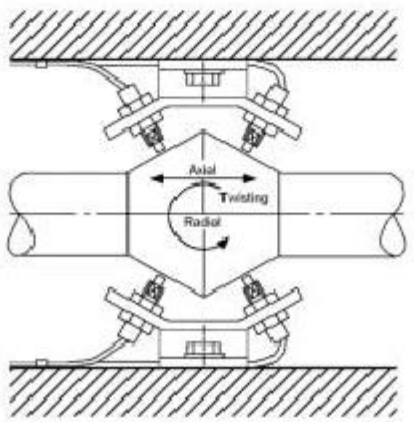

Figure A1.6 Four probes monitoring expansion by observing a tapered collar

It is also possible to measure differential expansion or axial movement with a small range probe using a mark space technique. This principle operates on detecting movement in special plates attached to the turbine shaft. The shaft target pattern consists of a number of pairs of 'teeth' and 'slots' surrounding the shaft and rotating with it. Each pair of teeth are tapered axially such that alternate teeth taper in the opposite directions, the narrow parallel slot between the teeth being at an angle to the shaft axis. There is a wider parallel slot between each pair of teeth to allow the system to identify each pair.

When the shaft rotates, the voltage pulses produced by the proximity probe and driver, have a tooth to slot pulse width ratio dependent upon the axial relationship between the shaft pattern and the probe position. The probe is mounted on a fixed part of the machine so variations in pulse width ratio are a measure of shaft axial position. The shaft pattern is illustrated below.

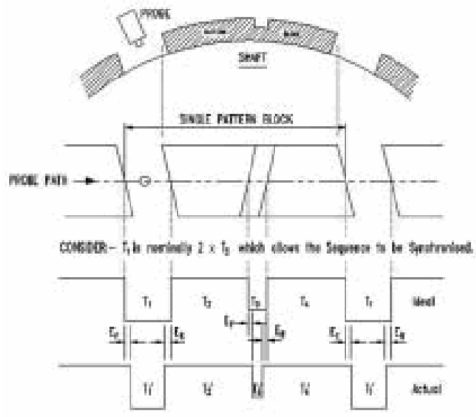

Figure A1.7 Shaft pattern

The Sensonics Sentry machine protection MO8612 module is suitable for this type of monitoring. The module exhibits a self-tracking threshold level, which ensures that the width of the signal pulses are measured at the optimum position within the pulse height. The unit is pre-programmed with specific plate patterns that can be selected to suit applications. The number of plates on the mark-space wheel is also an important parameter; when correctly set up this enables the module to minimise ‘plate wobble’ through the implementation of averaging algorithms. Customised patterns can also be entered into the module.

Since this technique measures axial movement based upon the ratio between detected pulses, it is immune to shaft movement in any other direction. This is a distinct advantage over the other techniques detailed in this section. A large expansion range can also be measured with a low cost probe through the fitting of the appropriate plate pattern, several centimetres if necessary, which would be impossible to achieve with a shaft collar.