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Katseseade vibratsiooni mõõtmiseks

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Tallinn, 2017

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TALLINNA TEHNIKAÜLIKOO
TALLINN UNIVERSITY OF TECHNOLOGY

Department of Electrical Power Engineering and
Mechatronics

Vibration Measuring Test Rig

MASTER THESIS

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Tallinn, 2017

AUTHOR’S DECLARATION

Hereby I declare, that I have written this thesis independently.

No academic degree has been applied for based on this material.

All works, major viewpoints and data of the other authors used in this thesis have been referenced.

Thesis is completed under the supervision of Trieu Minh Vu

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MASTER'S THESIS TASK

Year 2016 Autumn Semester

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Study Programme: MAHM02/13 - Mechatronics
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MASTER'S THESIS TITLE:

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Assignments to be completed and the schedule for their completion:

No.	Description of tasks	Timetable
1.	Study of the previous works related and comparing the existing apparatuses with our apparatus	First 2 weeks
2.	The designing of the concept and study of the various designs suitable for the final approach.	Next 3 weeks
3.	The design are checked and tested for the various conditions	Next 3 weeks
4.	Test and rectification of the system along with making of the report.	Next 3 weeks
5.	Thesis report completion and changed recommended by the supervisors	Next 2 weeks

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Solution for a proposed Vibration Measurements Test Rig.

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Supervisor Igor Penkov /signature/ signed digitally date

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FOREWORD

This thesis was written for my Master degree in Mechatronics at the Tallinn University of Technology. The work was executed at the Tallinn University of Technology. The work consists of electronic, mechanical, data communication and software fields integration. The purpose of this modernization work is to redesign a test shaft for investigation of vibration of materials and machine components. I would like to thank the following people, without whose help and support this thesis would not have been possible. First, I like to show my gratitude to the people of departments of mechatronics. My supervisor Trieu Minh Vu for his suggestions, encouragements, and guidance in writing the thesis and approaching the different challenges during the thesis. I would also like to thank Igor Penkov for his practical support, vision, and help. Finally, I would like to thank my parents and my friends for their constant support during the time I studied.

Tallinn, 2017,
Nagarjuna Kandarapu

EESSÕNA

Uurimustöö on kirjutatud Tallinna Tehnikaülikooli mehhatroonika magistri kursuse raames. Antud töö koosneb elektroonika, mehhaanika, andme vahetuse ja tarkvara integratsioonist. Moderniseerimise eesmärk on kujundada test investigeerimaks materjalide vibreerimist ja masinate komponente.

Ma tahaksin tänada järgmisi inimesi ilma kelle toe ja abita oleks uurimuse läbiviimine olnud võimatu. Esmalt soovin avaldada tänu mehhatroonika osakonnale. Samuti soovin tänada juhendajat Trieu Minh VU, kes andis soovitusi, julgustas ning suunas katsetama ning tegema järeldusi uurimustöö jaoks. Lisaks soovin avaldada tänu Igor Penkovi praktilise toetuse, visiooni ja muu abi eest. Ning lõpetuseks soovin tänada oma perekoda ja sõpru toetuse eest õpingute ajal.

Tallinn, 2017,
Nagarjuna Kandarapu

1. Introduction:

A vibration measurements test rig that already exists basically consists of purely mechanical components. My work comprises of modernizing and selecting of this apparatus which includes introduction modern sensors for nearly exact and precise outputs. It also includes the system being monitored and controlled by means of the computer based software. In this paper vibration measuring apparatus which is in disassembled state should be redesigned and has been discussed.

Tallinn University of Technology has a vibration measurements apparatus which has been in a dismantled state where many of the parts are missing. The apparatus needs to be redesigned and has to be modernized. The redesigned part mostly is vibration measuring system.

The figure 1.1 shows the existing system in Tallinn university of technology need to be designed and modify. The existing system consists of bearing, shaft, disc mass, and shaft supporters.



Figure 1.1 Present existing rig

Studying the vibration phenomena is an important part of machine elements course in many universities. This is the motivation to develop a modern, computer controlled adjustable test rig in the of machine elements.

Laboratory of machine elements of Tallinn University of Technology has a vibration measuring test rig, which has to be modernized. The rig work principle is based on the simple pre-existing in lab. The test rig has started rebuild few year ago but it came to half with bearing selection, shaft and machine components. The modernization task as shown in figure 1.2 includes:

- Redesign of a test mechanical system
- Selection of a measuring system
- Selection of a drive control system
- Selection of a data acquisition system
- Design of a rig user computer interface.

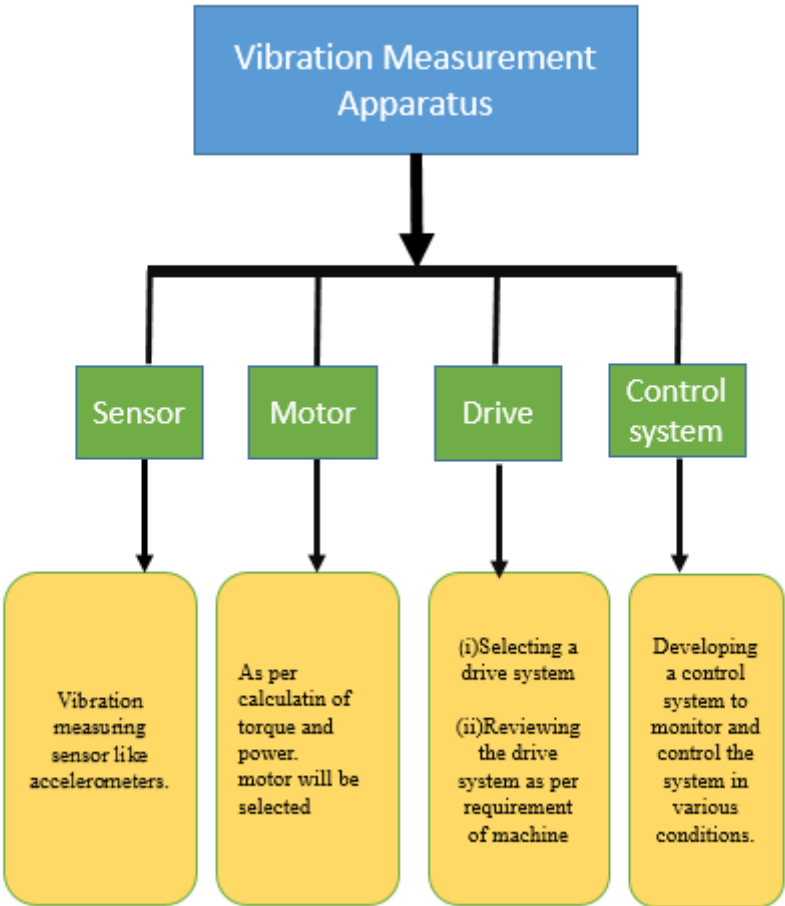


Figure 1.2 Block diagram of vibration measurement

The vibration measuring test rig must include modern technical solutions and all applicable safety rules should be assumed. While developing 3D model design must be consider all the issues and the test rig can be operated safely by anyone. The vibration measuring test rig is easily reconfigure for different types of objects.

1.1 Scope

The aim of the project is to develop a suitable solution for vibration measurement on a machine with motion system, control system and measuring system with suitable parameters. However, the pervious works and the researches were done related to the vibration measurements when it is subjected to the various faults as well the various parameters are discussed in this work.

1.2 Deliverables & methodology

In Tallinn University of Technology, laboratory study in the university the thesis work will carry out systematic research. So,The students will be able to operate the machine, study and analyze the vibration measuring in the machine using the apparatus:

- 3d design model of the Vibration measuring apparatus using the Computer Aided Designing software
- Control system of the apparatus
- Human Machine Interface
- Technical documentation of the material suggested

Chapter 2 Background and Literature review of the present existed System. Their Properties are discussed, working principle and measuring system of vibration. Chapter 3 mechanical design and cause for the vibration are discussed and explained. In Chapter 4 has overview of system. It presents the subsystems and tasks of the systems which is to be solved. In chapter 5, motion control system is presented. motion control system component connections and system set up are explained. Chapter 6, Measurement structure and components. In chapter 7, motion control software is explained. Measurement, control and monitoring software has been presented.

2. Background and LITERATURE REVIEW

2.1 What is vibration?

Vibration is a mechanical movement whereby oscillation occurs about an equilibrium point. The oscillations may be periodic like the motion of a pendulum or random like movement of a tire on a gravel road. Vibration is undesirable, wasting energy and creating unwanted sound. Vibration mainly caused due to unbalances in rotating parts, uneven friction or the meshing of gear teeth. Careful is needed while design any product to minimize unwanted vibrations [1]. If we could watch carefully a vibrating object in slow motion, you could have seen movements in different directions. As any vibration has two measurable quantities, one is how far (amplitude or intensity) and another one is how fast (frequency) the object moves help to determine its vibration characteristics. The terms used to describe the movement are frequency, amplitude and displacement as shown in Figure 2.1 [2].

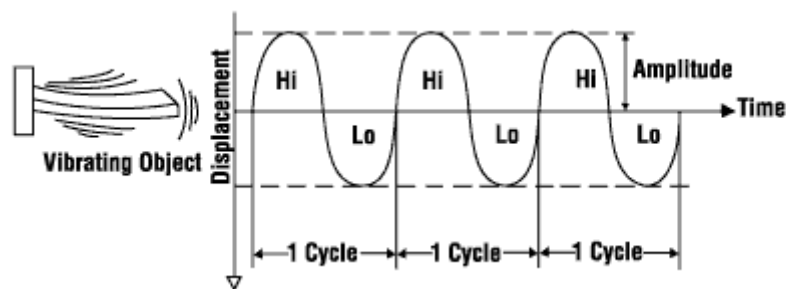


Figure 2.1 Representation of the measures of vibration exposure [2]

Vibration is not always a problem. In some cases, vibration is extremely important. Machines such as oscillating sanders and vibration tumblers use vibration to remove material and finish surfaces. Vibration feeders use vibration to move materials. In other cases, vibration is inherent in machine design. For instance, some vibration almost unavoidable in the operation of reciprocating pumps and compressors and gear boxes. In well-engineered and well-maintained machines, such vibration should be no cause of a problem. Most industrial devices are designed and engineered to operate smoothly and avoid unnecessary vibration. Vibration can indicate the problem in the equipment. If vibration problem is not corrected, the unwanted vibration itself can cause additional damage. Vibration can cause by number of conditions. Vibration in machine are cause due to auxiliary equipment, not just the main equipment. The major causes of vibration in a machine like unbalance, misalignment, wear, and etc [3].

So as to stay away from such disastrous accidents in future, different testing disciplines have created since fifties, including vibration analysis and testing. Later, when computing power turned out to be more accessible outside particular registering focuses, numerical displaying end up plainly principle device for beginning outline approval. In perfect case all of the three of these controls ought to be required in any product design in following order [3].

- Vibration analysis of a preliminary design;
- Finite element modeling of design;
- Vibration testing for expected environment with analysis results

2.2 Theory of Vibrations

Vibrations in the context of condition monitoring is defined as “a periodic motion or one that repeats itself after a certain interval of time” by Mobley R Kieth in his book ‘Vibration Fundamentals’ (Mobley, Vibration Fundamentals, 1999) [4].

Following is the equation which is called as ‘Harmonic motion’, describes a relation between the Vibration displacement in a spectrum with amplitude, frequency and time [4].

$$X = X_0 \sin(\omega t)$$

Where

X = Vibration displacement (mm)

X₀ = Maximum displacement or amplitude (mm)

ω = Circular frequency (radians per second)

t = Time (seconds)

The period of the vibration is the time interval, T. A spectrum or profile of a vibration is shown in the following figure 2.2, which shows the period, T, and the maximum displacement or amplitude, X₀. [4]

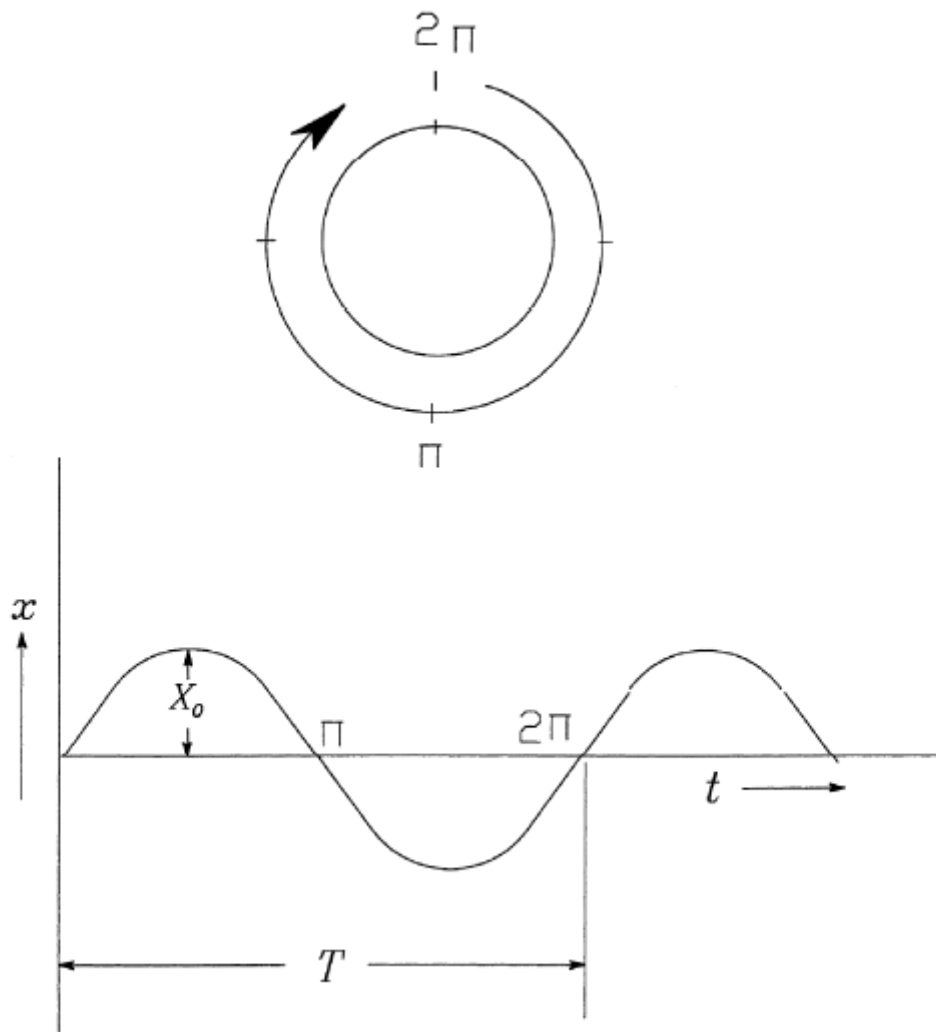


Figure 2.2 Illustrations of vibration cycles (Mobley, Vibration Fundamentals, 1999) [4]

Frequency of vibration is defined as the reciprocal of time period 'T', which can be expressed in units of rotations per minute (rpm) or cycles per second (cps) or Hertz (Hz). Such vibration oscillations can be illustrated and studied by looking at a graph of simple harmonic motion (periodic motion) of a pendulum which is illustrated in the figure 2.2. [4]

The displacement, velocity and acceleration can be obtained by first, second and third order differentiation of the equation, respectively. (Mobley, Vibration Fundamentals, 1999). [4]

2.3 Relations between acceleration, velocity and displacement

Classic kinematic equation explained one-dimensional motion in three terms specific to movement: [5]

Displacement (\vec{x}) is distance traveled by an object in given period of time (t) with average velocity (\vec{v}): [5]

$$\vec{x} = \vec{v}t \quad (2.3.1)$$

It is know as vector distance from some initial point via use of unity vectors [5]. A two dimensional example in cartesian coordinate system would be [5].

$$\vec{x} = \vec{x}i + \vec{y}j. \quad (2.3.2)$$

Velocity(\vec{v}) is rate of change of displacement. Since velocity has a value and direction. It is inherently a vector quantity consisting of initial velocity \vec{v}_0 and changing in time t at a rate of acceleration \vec{a} : [5]

$$\vec{v} = \vec{v}_0 + \vec{a}t, \quad (2.3.3)$$

Or it expressed simply as a rate of change of displacement in time: [5]

$$\vec{v} = \frac{\vec{x}_2 - \vec{x}_1}{t_2 - t_1} = \frac{\Delta \vec{x}}{\Delta t}. \quad (2.3.4)$$

Acceleration (\vec{a}) is rate of change of displacement. Since is derived from velocity, it is also a vector quantity: [5]

$$\vec{a} = \frac{\Delta \vec{v}}{\Delta t} = \frac{\vec{v}_2 - \vec{v}_1}{t_2 - t_1} \quad (2.3.5)$$

The above Equation (2.3.3) (2.3.4) (2.3.5) show the time and distance are base quantites, while velocity and acceleration and derived from these base quantites [5].

$$\vec{a}(t) = \frac{d\vec{v}}{dt} = \frac{d^2\vec{x}}{dt^2} \quad (2.3.6)$$

Or reverse the operation to get displacement or velocity from acceleration value, based on initial values \vec{x}_0 and \vec{v}_0 : [5]

$$\vec{v} = \int \vec{a} dt = \vec{a}t + \vec{v}_0 \quad (2.3.7)$$

$$\vec{x} = \int (\vec{a}t + \vec{v}_0) dt = \frac{\vec{a}t^2}{2} + \vec{v}_0t + \vec{x}_0 \quad (2.3.8)$$

These equations are used when dealing with oscillating systems and accelerometers which measure acceleration. In case the oscillation waveform and durations are known, the related quantities can be derived. [5]

2.4 Dynamics of mass-spring-damper system:

Consider an ideal mass-spring-damper system with m (in kg), spring constant k (in N/m) and damping coefficient R (in N-s/m) as show in figure 2.3. [6]

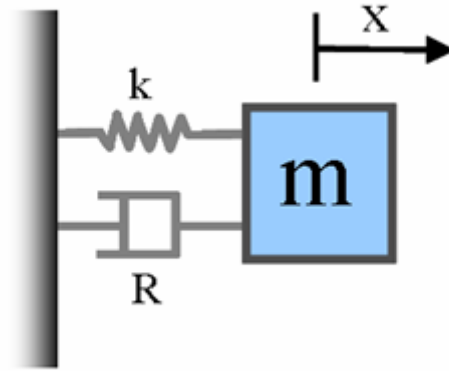


Figure 2.3 mass-spring-damper system

From above diagram the following equation is expressed: [6]

$$F(t) - kx(t) - b\dot{x}(t) = ma(t) \quad (2.4.1)$$

$$F(t) - kx(t) - b \frac{dx(t)}{dt} = m \frac{d^2x(t)}{dt^2} \quad (2.4.2)$$

$$m \frac{d^2x(t)}{dt^2} + b \frac{dx(t)}{dt} + kx(t) = F(t), \quad x_{t_i} = x_0 \quad \& \quad \left. \frac{dx(t)}{dt} \right|_{t=t_i} = v_0 \quad (2.4.3)$$

From above equation (2.4.3) we can solve the mass-spring-damper system with Euler method . we will can solve it as 2^{nd} order differential equation is split into two 1^{st} order differential equations. [6]

$$x_1 = x \quad \& \quad x_2 = \dot{x} = \frac{dx}{dt} \quad (2.4.4)$$

$$\frac{dx_1}{dt} = \dot{x} = x_2 \quad (2.4.5)$$

$$\frac{dx_2}{dt} = \ddot{x} = \frac{1}{m}(F_0 \sin \omega t - b\dot{x} - kx) = \frac{1}{m}(F_0 \sin \omega t - bx_2 - kx_1) \quad (2.4.6)$$

By solving above equation (2.4.6) we also obtain the discrete version of above differential equations. [6]

$$x(n, 1) \leftarrow x_1(n\Delta t)$$

$$x(n, 2) \leftarrow x_2(n\Delta t)$$

$$x(n + 1,2) = x(n, 2) + x(n, 2) \times \Delta t$$

$$a = \frac{1}{m} (F_0 \sin(\omega \times (n\Delta t)) - b \times x(n, 2) - k \times x(n, 1)) \quad (2.4.7)$$

Trajectory of mass-spring-damper system with different parameters and initial conditions of Runge-kutta method and Euler method plots are shown in following: [6]

```
Oe=MSDSE(1,0.5,1,1,3,0,0);
```

```
Or=MSDSRK(1,0.5,1,1,3,0,0);
```

```
plot(Oe(:,1),Oe(:,2),'r',Or(:,1),Or(:,2),'b--','LineWidth',2); axis tight; grid on;
```

```
xlabel('\bfTime (sec)'); ylabel('\bfParticle Position (m)');
```

```
title({'\bf Mass-Spring-Damper System Simulation'; 'm=1kg, b=0.5Nsec/m, k=1N/m, F_0=1m, \omega=3rad/sec, x_0=0m, & v_0=0m/sec'})
```

```
legend('\bfEuler', '\bfRunge-Kutta', 'Location', 'NorthEast')
```

As shown in the below figure 2.4. It explain the response of the sinusoidal excitation for the particle when it is stationary, and there is no spring is compressed or stretched. Comparing two results from Euler method and Runge Kutta method, they produce similar simulation results. [6]

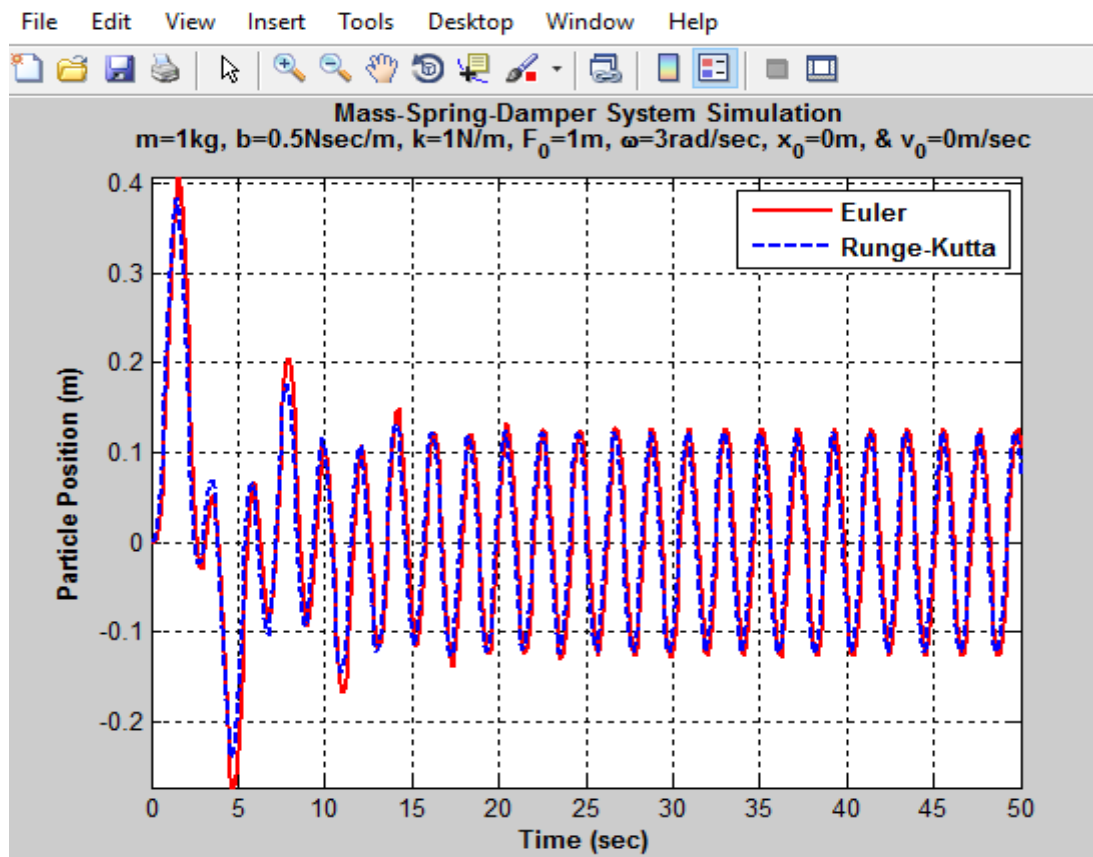


Figure 2.4 Show no spring is compressed or stretched [6]

```

Oe=MSDSE(1,0.5,1,0,0,1,0);
Or=MSDSRK(1,0.5,1,0,0,1,0);
plot(Oe(:,1),Oe(:,2),'r',Or(:,1),Or(:,2),'b--','LineWidth',2); axis tight; grid on;
xlabel('\bfTime (sec)'); ylabel('\bfParticle Position (m)');
title({'\bf Mass-Spring-Damper System Simulation'; 'm=1kg, b=0.5Nsec/m, k=1N/m,F_0=0m,
\omega=0rad/sec, x_0=1m, & v_0=0m/sec'})
legend('\bfEuler','\bfRunge-Kutta','Location','NorthEast')

```

As shown in below figure 2.5 it will explain the response with no external force and no damping when the spring is initially stretched. Comparing two results from Euler method and Runge-Kutta method as shown below we can understand Runge-Kutta method is more accurate than Euler method, based on the analytical solution of the mass-spring-damping system. [6]

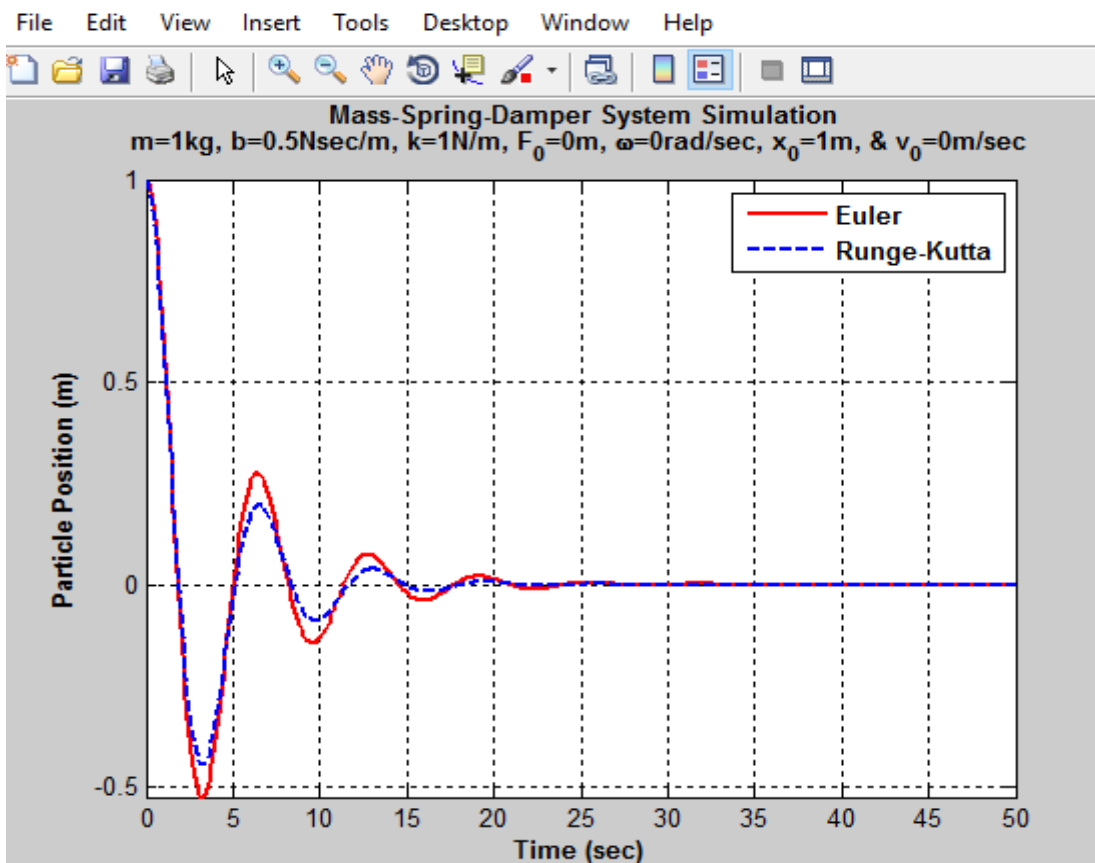


Figure 2.5 Show No extra force and damping [6]

```

Oe=MSDSE(1,0,1,1,1,0,0);
Or=MSDSRK(1,0,1,1,1,0,0);
plot(Oe(:,1),Oe(:,2),'r',Or(:,1),Or(:,2),'b--','LineWidth',2); axis tight; grid on;
xlabel('\bfTime (sec)'); ylabel('\bfParticle Position (m)');
title({'\bf Mass-Spring-Damper System Simulation'; 'm=1kg, b=0Nsec/m, k=1N/m, F_0=1m,
\omega=1rad/sec, x_0=0m, & v_0=0m/sec'})
legend('\bfEuler', '\bfRunge-Kutta', 'Location', 'NorthWest')

```

As shown in below figure 2.6 it will explain the response of external force with the system resonance frequency initially stretched. Resonance happens very quickly, which means the particle position oscillates from $-\infty$ to $+\infty$ and numerical simulation has limitation to express the infinity. So, as time goes on, oscillation for both methods become larger. [6]

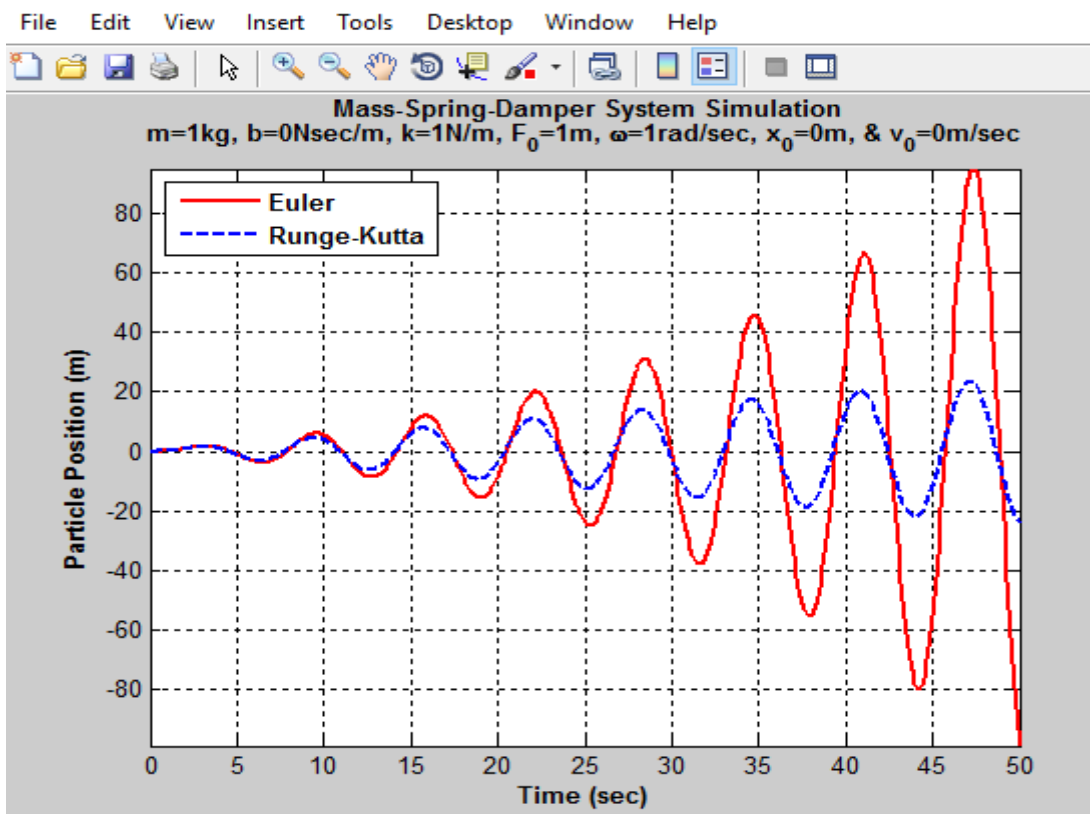


Figure 2.6 Show external force on systems [6]

2.5 Classical model:

Classical model will explain the basic role of the data acquisition part of the whole vibration system as a signal path from transducer to the display and analysis [5]. As shown in figure 2.7 single-axis measurement this path consists of transducer, cable, signal conditioner, low-pass filter, analog-to-digital conversion and run-time display. In modern Inertial-Electronic Piezo-Electric (IEPE) systems have signal conditioning microelectronic circuit built into transducer itself it helped in severely reducing of cable parameters on measurement results [5].

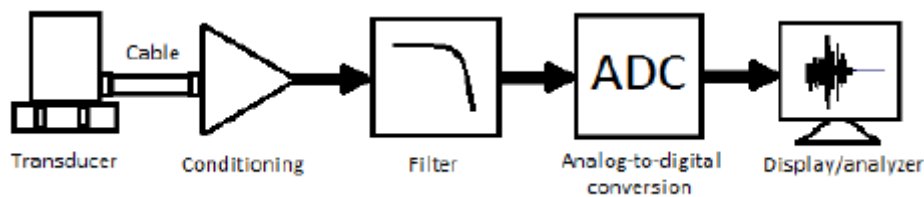


Figure 2.7 Signal path in classic modeling of vibration acquisition chain [5]

As shown in figure 2.7 each part of the data acquisition chain has its own sources of error and can be analyzed separately. But direct transfer of mechanical motion information to a computer system is not possible. Transduction into electric signals is a direct proportional equivalence between the mass-spring-damper system and a resonant circuit consisting of passive lumped elements [5].

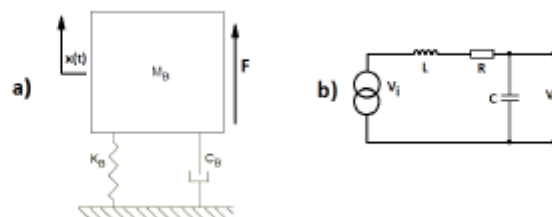


Figure 2.8 Mass-spring-damping system and b) equivalent electric circuit. [5]

As shown in figure 2.8 coefficients are indicated as:

$q(t)$ is change as a function of time,

L is inductance,

R is resistance,

C is capacitance and

V_i is input voltage.

The equation of mass-spring-damping system is represented as [5]

$$M_B \ddot{x}(t) + C_B \dot{x}(t) + K_B x(t) = \vec{F} \quad (2.5.1)$$

And its lumped-paramter circuit system

$$L \ddot{q}(t) + R \dot{q}(t) + \frac{1}{C} q(t) = V_i, \quad (2.5.2)$$

From above equation (2.5.1) and (2.5.2) we can obtain equivalence between mechanical and electrical parameters, data acquisition chain can be done in terms of electronics components. Figure 2.9 shows signal acquisition chain as a simplified equivalent electricronic circuit [5]. In IEPE devies charge amplifier is mounted in the same device as a sensor itself and it help to minimizing interface cable capacitance on high-impedance connection between sensor and charge amplifier [5].

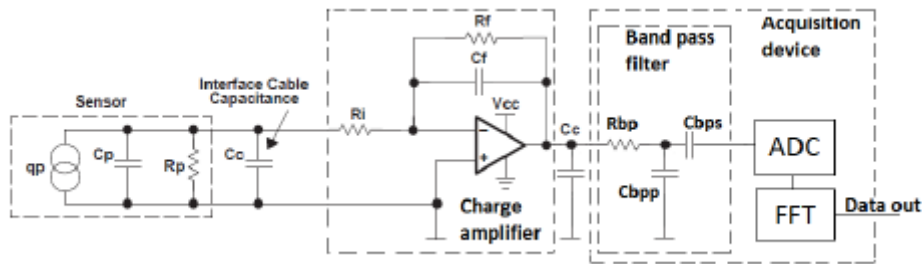


Figure 2.9 Signal acquisition chain modeled as electronic components. [5]

As show in figure 2.9, Filtering is done on the controller side along with analog-to digital conversion (ADC) and subsequent Fast Fourier Transformation (FFT). Output of ADC is digitized and signal is transfered to frequency domain [5]. The operator has control over the interfacing cable between transducer and controller [5].

2.6 LITERATURE REVIEW:

The literature review was carried out in the areas of vibration monitoring of rotating shaft machinery. The approach of fault diagnosis and analysis of resulting vibration signature have been reviewed. According to Arun Kr.Jalan and A.R.Mohanty Vibration monitoring is one the first techniques of condition monitoring of rotating machines. Vibration main cause is due to shaft misalignment and rotor unbalance in a machine. He explained a model based technique for fault diagnosis of the rotor-bearing system, using the residual generation technique. Residual vibration is generated by experimental results for the rotor bearing system due to unbalance and

misalignment. The rotating machine is very primary assets in most of the industries. The rotor-bearing system of the modern rotating machinery is complex which needs a precise and tested prediction of its dynamic characteristics. Shaft misalignment and disk unbalance are the two main cause for rotating machinery vibration. Misalignment in the rotating machinery causes a reaction force and moments in the coupling which leads to the vibration in the system [7].

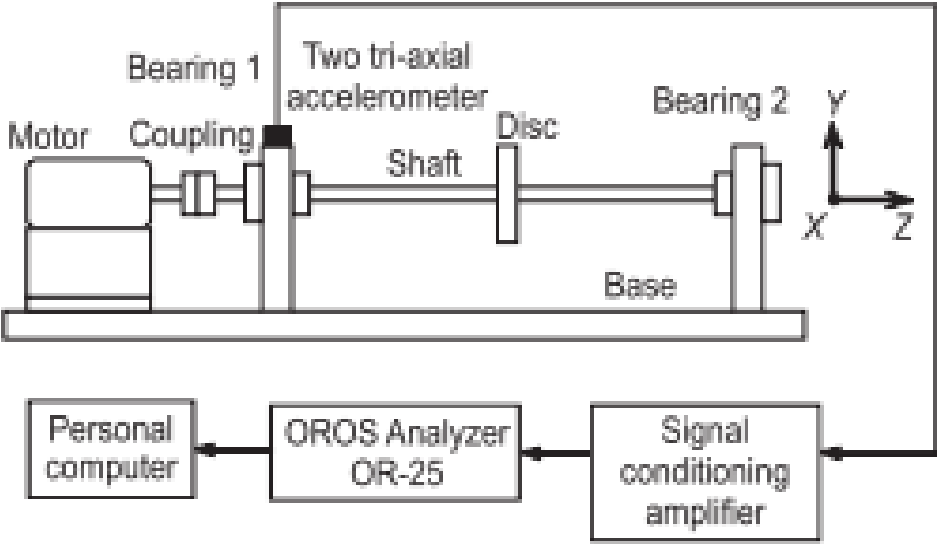


Figure 2.10 Line Diagram of Experimental rotor-coupling-bearing system [7]

The above figure 2.10 is an experimental rotor-coupling-bearing system using a fault simulator. The shaft is placed between two identical rolling bearings. A disc is placed in the mid-span of the shaft. The rotor shaft is driven by a three-phase AC motor connected to flexible coupling. The power supply to the motor can be varied through variable frequency drive (VFD) which control the speed of the motor. The motor rig was instrumented with two of tri-axial accelerometer along with signal conditioning amplifier. All the signals were analyzed on an OROS 8 channel and OR-25 FFT analyzer [7].

According to Jaswinder Singh, The vibration measured at the different surface of rotating machinery or at any other suitable place to know running condition of the machine. Due to quick development in the industry, there are increased usage of rotating machinery and demands for high speeds and greater power in various power plants and complex vibration issues have been created. Vibration monitoring is a technique which aims to know the current working condition of the machine and compare it with previously measured data or condition. Vibration analysis is became to be the process of detected vibration signal in displacement, velocity or

acceleration transducer. It is essential to consider the type and scope of transducers or pickup for capturing vibration signal. A study of vibration cause in mechanical system has been made which can give cause faults at various point in machine. Experimentation was done for monitoring and analysis of vibrations for a machine supported by the ball bearing. The displacement amplitude vibration has been utilized for the vibration signatures due to various issues like misalignment, unbalance in the bearings [8].

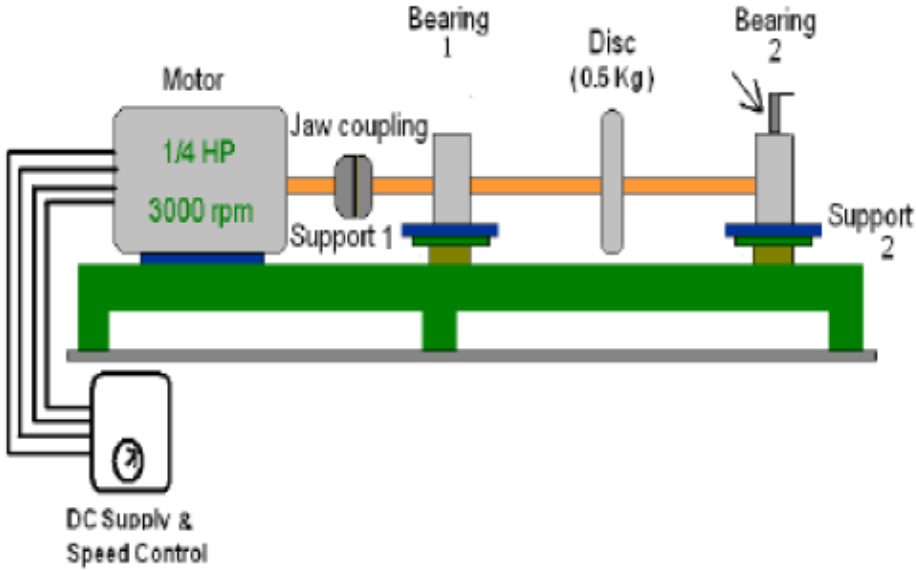


Figure 2.11 Line Diagram of Rotor Test Rig [8]

The line diagram of Rotor test rig as shown in Figure 2.11. The rig consists of 0.25 HP motor running at a speed 3000 rpm (rotations per minute). Motor is flexible coupling to the shaft of 10 mm diameter and 230 mm length fixed between two single ball bearing. The disc of 70 mm diameter and 15 mm thick is fixed at the mid-span of the shaft. Displacement vibration measurements are made by using the displacement transducers by integrating the output from a velocity transducer [8].

3. MECHANICAL DESIGN

The present rig of bench tests of rotor systems, in conditions as close as possible to the conditions that arise when working with different type of rotor systems and having different defects unbalance, defects of bearing, motor, clutch.

Rotating machine are important assets in most industries. One of the mojour areas of interest in the modern day condition monitoring of the rotating machinery is that of vibration. Misalignment and unbalance are two major sources of rotating machinery vibration. In this these I am main discussing vibration caused due to unbalance. The vibration due unbalance will affects critical parts of the system such as bearings, gears, motor, seals, couplings etc.

When the center of mass of a rotating disc is not coincident with the center of rotation, it is known as unbalance. In experiment, balancing is done to balance the rotor system but due to reasons like porosity in casting non uniform density of material, non uniform density of material, manufacturing tolerances, and gain or loss of material during operation, rotors can never be perfectly balanced.

3.1 Typical machinery faults

The first step in implementing any vibration monitoring program is to know the equipment. There are many failure modes for machinery. Four basic failure modes most commonly found in any standard equipment are imbalance, misalignment, bearing faults and looseness. Each machinery fault has its own unique vibration signature that helps to identify the particular fault. Each fault has specific fault frequency that help to know the fault while the amplitude of the vibration helps to determine the fault. [9]

Imbalance:

In the case of rotating equipment, the axis (shaft) must be equally balanced. An imbalanced axis will create a centrifugal force which causes the vibrate. As the machine speed increase the effects of imbalance become greater. Imbalance can severly reduce bearing life as well as cause undue machine vibration [9].

Misalignment:

Vibration can result when machine shaft are out of line. Angular misalignment occurs when the axes of a motor are not parallel. When the axes are parallel but not exactly aligned, the condition is know as parallel misalignment. Misalignment can be caused during assembly or develop over

time, due to thermal expansion, components shifting or improper reassembly after maintenance [9].

Loose connections:

Vibration that might otherwise go unnoticed can become obvious and destructive if the component that is vibrating has loose bearing or is loosely attached to its mounts. Such looseness might or might not be caused by the underlying vibration. Whatever its cause, looseness can allow any vibration present to cause damage, such as further bearing wear, wear and other components [9].

Bearing Fault:

last but not the least, here is a major contributor for vibration. In majority of the cases every initial problem starts in the bearing and propagates to the rest of the members of the machine. A bearing devoid of lubrication tends to wear out fast and fails quickly, but before this is noticed it damages the remaining components in the machine and an initial look would seem as if something had gone wrong with the other components leading to the bearing failure. Such is the criticality of the bearings in any machinery [9].

3.2 Existing test rig

The current test rig has to modernize for vibration measurement. It has only machine components, which has needed to select the motor as per the calculation, drive to control the motor speed and sensor to measure the vibration. However the most of the parts of the apparatus are redesign as per the existing rig without changing any principle of the design, the configuration and the working of the apparatus is explained.

The current test rig has only machine components such as shaft, bearing, shaft side supporters, disc mass, extra disc mass and frame.

As shown in figure 3.1 it is the existing disc is 125 mm diameter and thickness is 47 mm is fixed at mid span of the shaft.

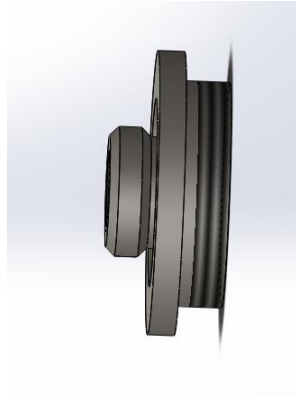


Figure 3.1 CAD model of Existing disc

The shaft as shown in figure 3.1 is to be carried by the two supporting bearings inside a frame housing and to the axis of rotation. The frame design surrounding the rotating shaft, carries and lubricates the support of bearing.

A CAD model redesigned shaft is used in this project as shown below figure 3.2. The shaft is designed to be linked from the right side by a flexible rubber coupling to the system driver (motor). As shown in figure 3.2 the existing rig has a shaft of 12 mm diameter and 330 mm length.

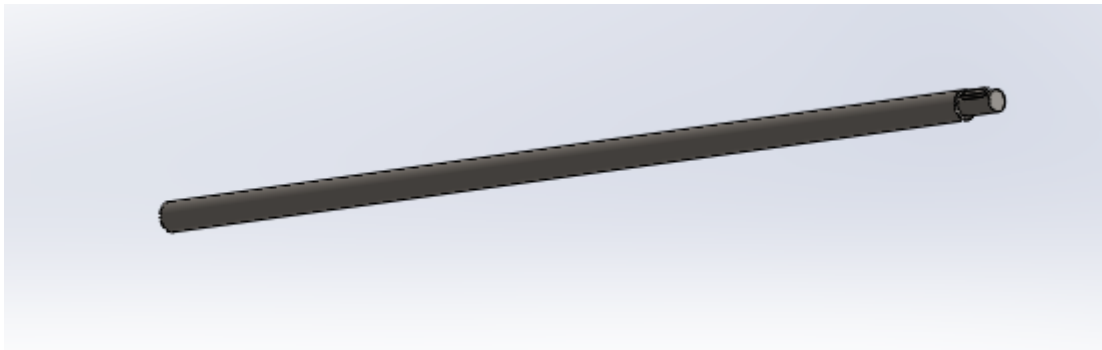


Figure 3.2 CAD model of existing shaft

By conducting FEM analysis on the shaft using von Mises Stress by SolidWorks, the maximum stress was found at the groove nearby the critical location. The yield strength is 4.100×10^4 , which is less than the maximum stress 5.849×10^3 . So, the design is safe. The result is shown in figure 3.3.

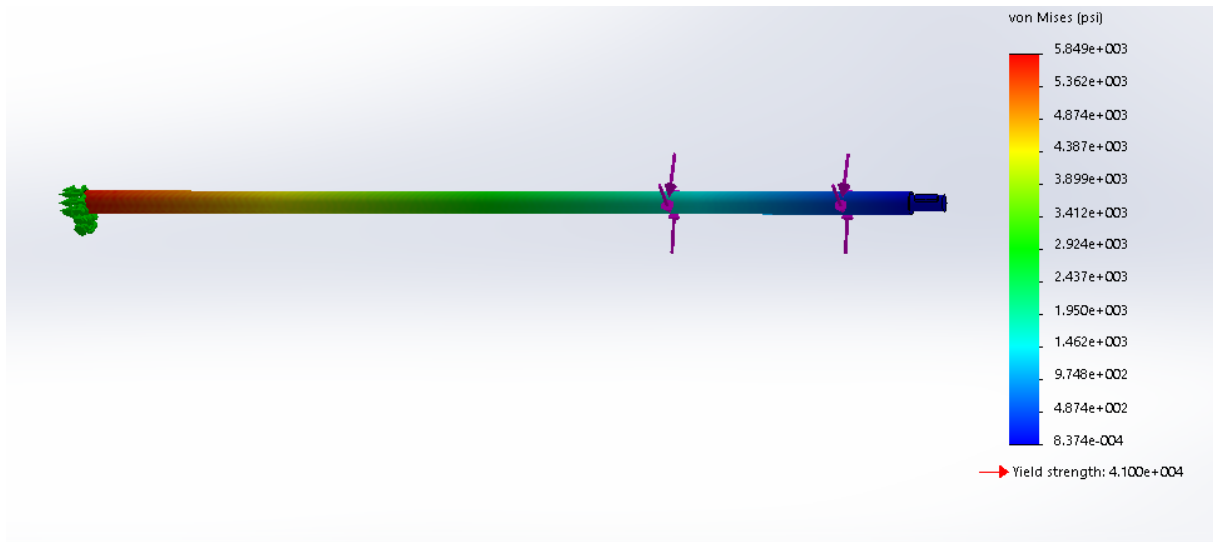


Figure 3.3 FEM analysis on shaft

Selection of Bearing for this Thesis:

The bearing has selected for this these to use UCPH201 pillow block bearings, are mounted bearing that are used to provide support for the shaft. Their mounting surface is on plane parallel and offset to the axis of the shaft. The pillow block are permit adjustment and easy to mounting. Pillow block bearing has dual set screw locking, wide inner ring and anti-rotation device as shown in figure 3.4 [10].



Figure 3.4 Pillow Block UCPH201 model [11]

3.3 Mechanical design of the apparatus:

Balancing is a technique which eliminates the inertia forces and moments. Balancing a rotor shaft is an important issue in the industry, which involves moving the centre of gravity to the centre of rotation. In every machine the balancing is related to the rotating mass, its radius and its speed.

The stand for diagnostics of rotor systems has a motor and a measuring system, consisting of vibration measuring sensors, bearing. Motor is controlled with the help of the drive which controls the speed of the motor and a PC based control system. The oscillations receiver is in the form of a shaft with discs. The receiver of vibrations is made in the form of a shaft with two discs, one of the sensors measuring is connected through a filter with DAQ and measuring connected through a computer program. Which measures the vibration and saves the data. Analysis is done to the vibration measured through vibration measuring sensor.

As you see in figure 3.5, this machine is redesigned in the concept of studying vibration causes and its effects on the machine. The machine is in a balanced state. The disc is centered on the shaft with two side supports. So, we do not have any vibration on the machine.

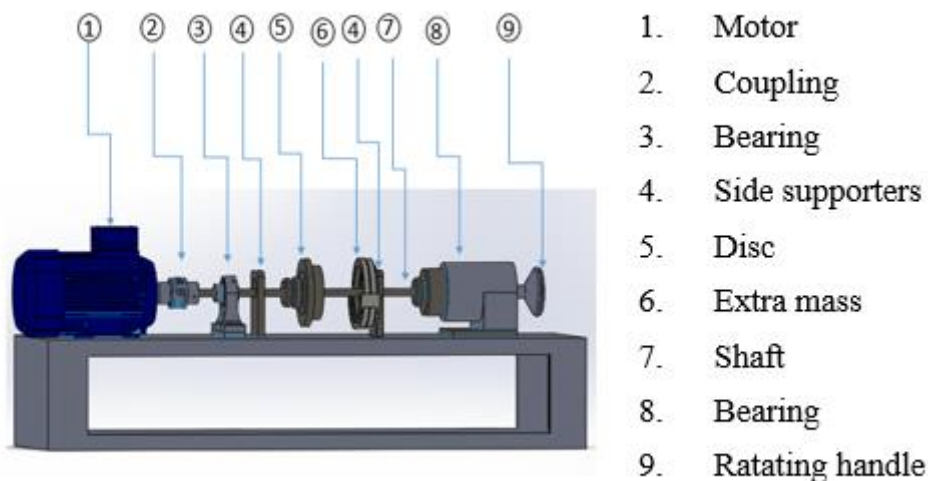


Figure 3.5 Design of vibration measuring rig

3.4 Creating disk unbalance:

Unbalance is caused due to unequal mass distribution in a rotating system which leads to centrifugal forces in the radial direction. In an unbalanced system, there are many disadvantages of an unbalanced system which will generate unwanted noise, excessive stress in machine elements and reduce the reliability of rotating parts. This will cause many failures in a short period of time.

When motor start to rotate the machine is in balance state. We don't have any kind of vibration. The motor is connected through a drive. Where drive is used to control the speed of the motor. To create imbalance we add extra disk on the shaft as shown in figure 3.6. Vibration start to vibrate from the shaft. As installed sensors at bearing point start to measure the vibration. The operator can measure the vibration at various speeds from minimum to maximum. The operator can control the vibration by applying extra force opposite to the shaft. If you rotate the rotating handle it will apply the force on shaft and vibration will decrease. It is experiment for control vibration in Laboratory at Tallinn University of Technology.

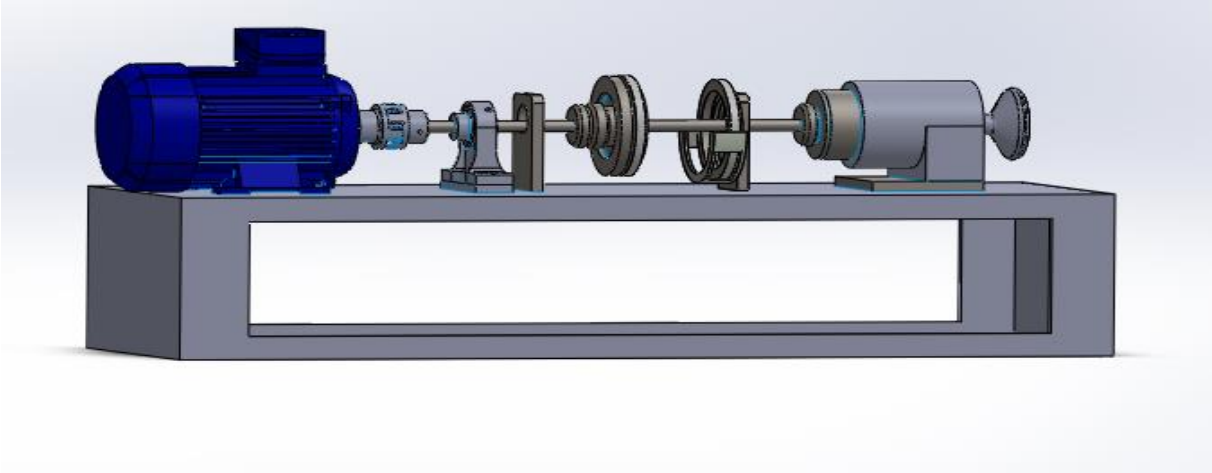


Figure 3.6 Test rig on unbalance

3.5 Safety Measurement :

Safety is very important for machine. While designing machine for safety measurement has taken to avoid injury due to some sudden or unexpected motion can inflict serious injury. So, machine is covered with side frame from all side as shown in figure 3.7.



Figure 3.7 Test rig with safety side frame

3.6 SDOF system with a Rotating unbalance:

As shown in figure 3.8 rotating unbalance. Assume that guides are frictionless. The radius e is measured from centre of the mass of mass m . The equation of the motion as we need an expression for the motion if the rotating unbalance in terms of x . If the mass rotates with a constant angular velocity ω_r . [12]

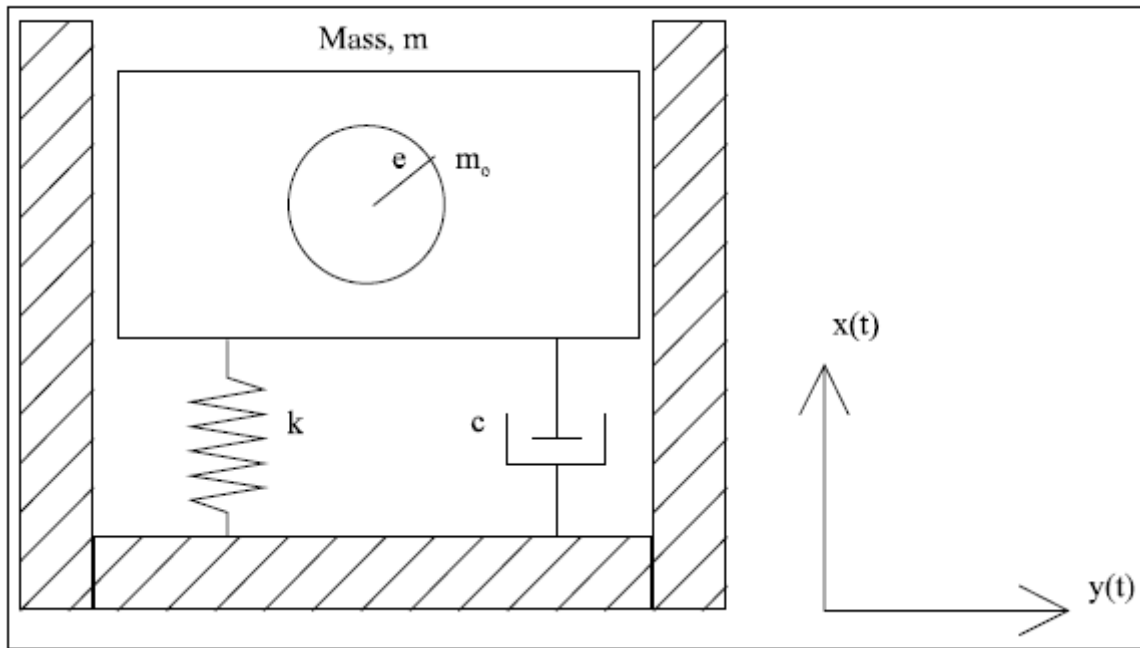


Figure 3.8 SDOF Rotating unbalance system [12]

From the figure 3.8 circle can be defines can be described parametrically as [12]

$$x(t) = e \sin \omega_r t, y(t) = e \cos \omega_r t \quad (3.6.1)$$

From above equation note that sine defines the x coordinate as it is vertical. So, we can construct the equations of the motion. The position coordinate of the rotating unbalance is $x + \sin \omega_r t$, and the acceleration is second derivative of the expression with respect to time. The acceleration of the mass without the unbalance is \ddot{x} . Adding in the effects of the stiffness and damper, we get below equation. [12]

$$(m - m_0)\ddot{x} + m_0(\ddot{x} - e\omega_r^2 \sin \omega_r t) = -kx - c\dot{x} \quad (3.6.2)$$

So, collecting x and its derivatives, moving the sine term to other side of the expression, and dividing by the system mass gives the final equation of motion [12].

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = m_0 e \omega_r^2 \sin \omega_r t \quad (3.6.3)$$

From above equation (3.6.3) this identified to the harmonic forcing function case we encountered earlier, except that now our force is in form of a sine rather than cosine [12].

$$x_p(t) = X \sin(\omega_r t - \phi), \quad (3.6.4)$$

Where, with $r = \omega_r / \omega_n$

$$X = \frac{m_0 e}{m} \frac{r^2}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}, \phi = \tan^{-1} \frac{2\zeta r}{1-r^2} \quad (3.6.5)$$

And, the homogenous solution for this expression is, [12]

$$x_h(t) = A e^{-\zeta \omega_n t} \sin(\omega_d t + \theta), \quad (3.6.6)$$

A and θ are determined from the initial conditions. Final solution is then $x(t) = x_p(t) + x_h(t)$ [12].

From below figure 3.9 we can see different natural frequency effect on rotating unbalance. and from programm natural frequency was varied while holding all other parameters constant. Where $\omega_n = 2$ for the highest natural frequency tested, the oscillation occurs along a single sinusoid. This is because the natural frequency of the system is too high to be executed by relatively slow rotation frequencies. The first two plots have natural frequencies small enough to be excited by the slow rotation of the eccentric mass [12].

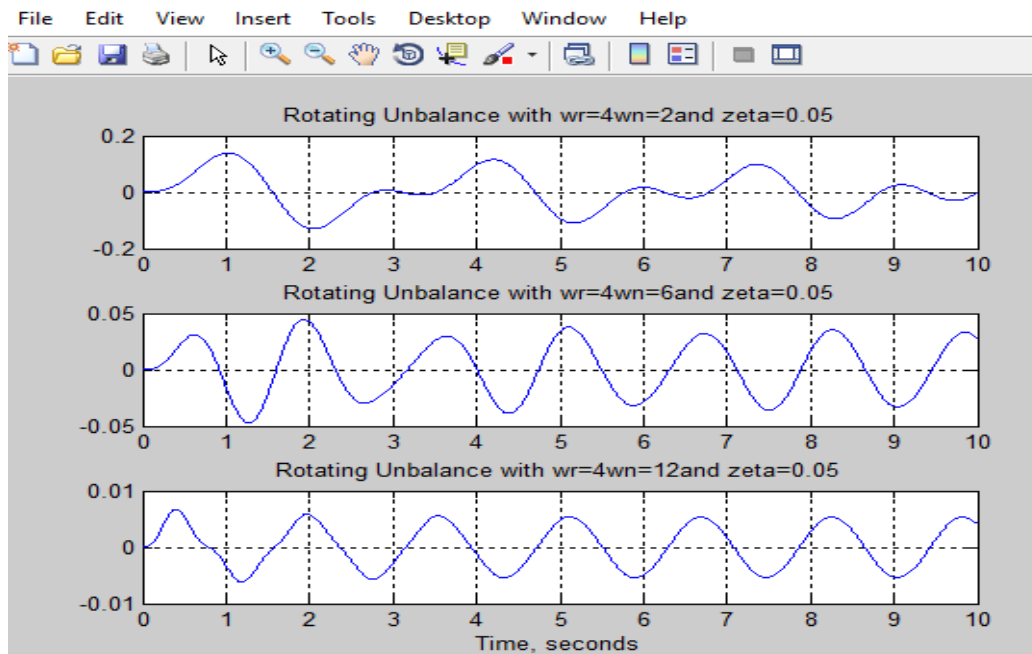


Figure 3.9 Response of system with different natural frequencies to a rotating unbalance [12]

From below figure 3.10, the system damping is varied. The result is that the transient portion becomes smaller, to the point where it disappears at $\zeta = 0.3$. A difference in the magnitude of oscillation, as would be predicted from the expression we have derived for the parameter X, is not present because the frequency ratio we are testing is in range where oscillation magnitude shows little variation with damping ratio. This consideration is important in the design of machinery; if the machine can be designed to have a much higher natural frequency than the oscillating mass, then the level of damping can be made low without increasing the amplitude past acceptable levels [12].

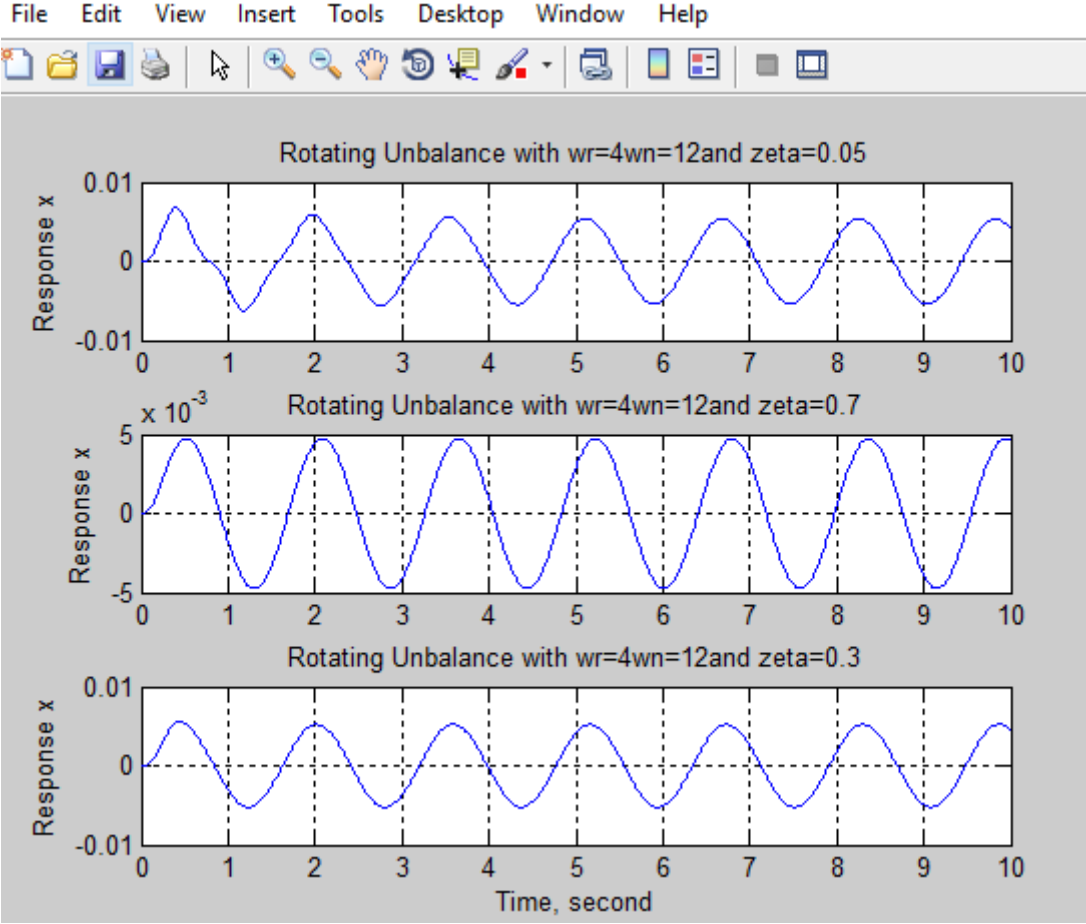


Figure 3.10 Response with varied damping ratios [12]

The below figure 3.11 show the variation of vibration with increasing system mass. Notice from figure 3.11 how the amplitude of the vibration decreases with respect to increasing of mass. This is due to the dependence of X on $m_0 = m$. [12]

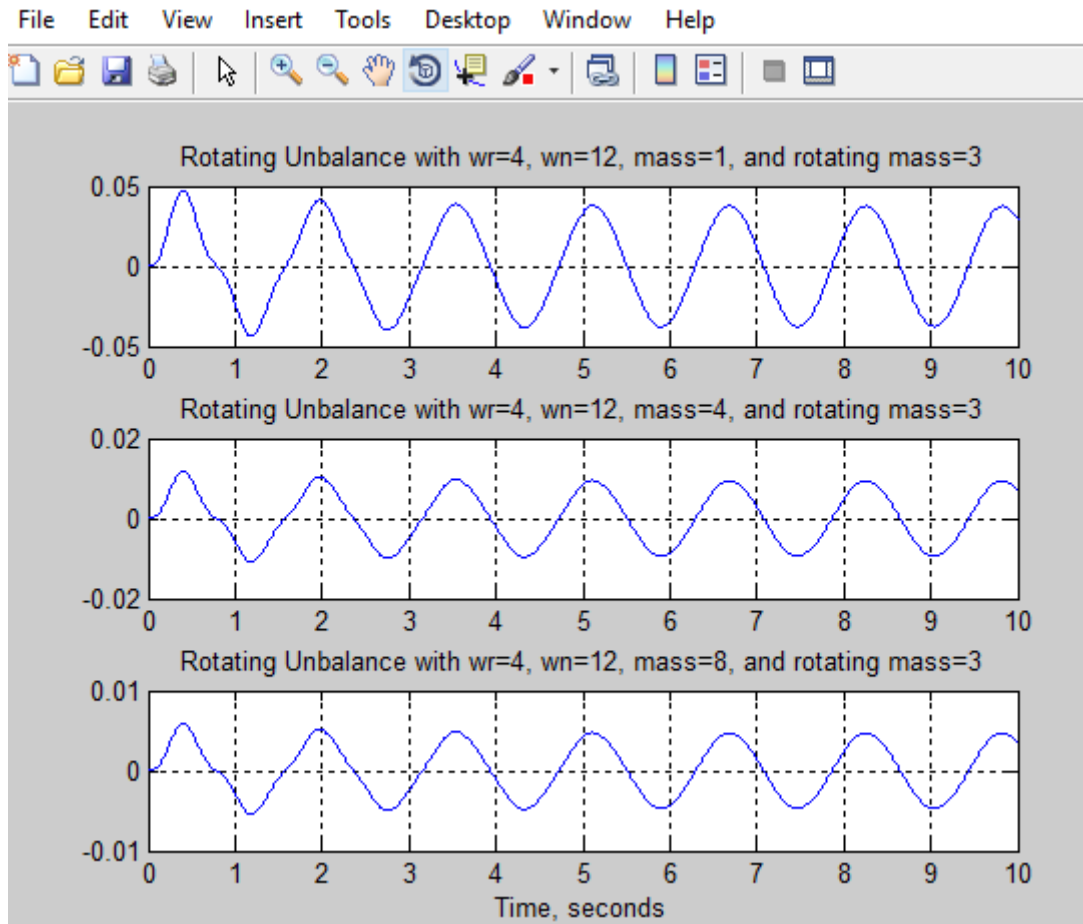


Figure 3.11 Effect of varying system mass [12]

4. Project

To start the process of redesigning the vibration measuring rig, The device is divided into the subsystems. The quality of design and selection of each module in subsystem have been identified to meet the requirements of the device [13]. Most acceptable working fundamentals for each subsystem have been investigated and combined with a number of potential concept designs, that were clearly analyzed afterward in order to propose the most suitable option.

Tabel 4.1 Vibration Measuring Subsystem Descriptions

Subsystems	Descriptions
Loading and Vibration Measuring system	Applying extra mass on the shaft to create unbalance state.
Motor Control	Contorlling motor with help of drive and testing vibration
Measuring Structure	Obtaining measurement data and process data.conversation of gathered infomation into meaning manner and registering for analyze.
Measurement Software & Real-time Monitoring	Monitoring real-time measuring process and give visualized results, charts, graphs and etc.

The Table 4.1 above show subsystems of the work. Loading and vibration measuring explain the creation of unbalance state in machine. Motion control subsystem help to find motor and drive control system for machine [13]. Measuring structure subsystem help to find out, which kind of sensor is going to be use for vibration measuring in the machine. Measuring software subsystem help to find out the software which can read the signal and operator can monitor from the software [13]. Mechanical structure involves designing of components of loading system as well as the design of vibration measuring system. The figure 4.1 below functional block diagram of the project [13].

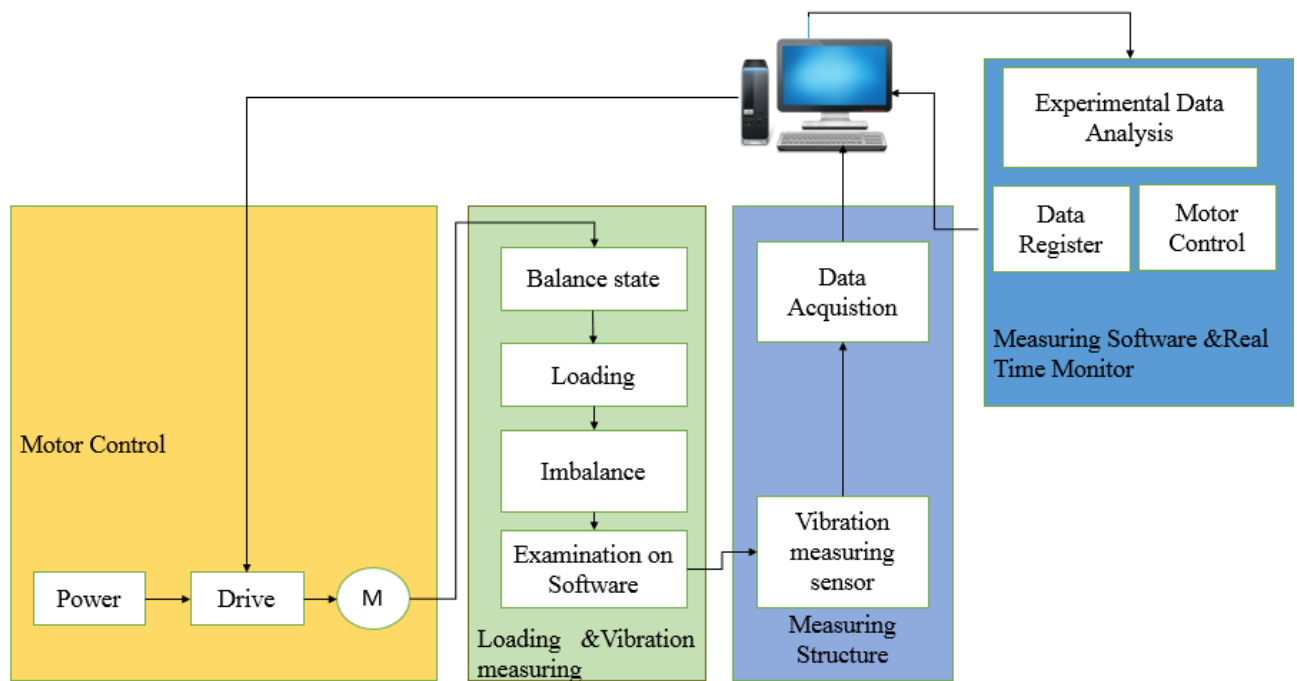


Figure 4.1. Vibration measuring subsystems configuration

4.1 Motor control:

The role of the motor control is to command system and run based on desired speed. It has the drive to receive commands from PC software that is sent based on user defined test parameters such as speed and torque [13]. Then drive creates a signal in order to run the motor.

4.2 Loading and vibration measuring system :

Loading and vibration measuring system involves a application of creating unbalance loading system. By applying the extra mass of load of shaft leads to create an unbalance state. In unbalance state, the shaft starts to create vibration in the rig. The vibration monitoring of rotating machinery is heavily depends on the spectral analysis [13].

4.3 Measuring structure:

To measure vibration we are using accelerometers. Accelerometers work on piezoelectric effect. Vibration generates analog output voltage Proportional to the acceleration of the vibration. Accelerometers create voltage signal output. Data acquisition board is used to collect voltage signal from the accelerometer and signal is transferred to the PC equipped with software [13].

4.4 Measurement software and real-time monitoring:

After all Measurement and motor control system design. In order to track measurement process and observe the result of the measurement, It is a very important platform. To achieve the excellent output, the system is divided into subsystems which involve designing of measurement, control system and monitoring software [13]. In the real-time monitoring system vibration measured signal will be processed, visualized and monitored in real time on the screen [13]. Besides, it will have an interface for defining some parameters by the user, such as vibration range, speed, and mass of bodies [13]. The vibration signal measured data will be stored and for further analysis [13].

5. Motion control system and components:

Motion control is the control of the position or velocity of machines using some type of device such as electric motor [13]. Motion control is an important part of robotics, machine tools and laboratory equipment.

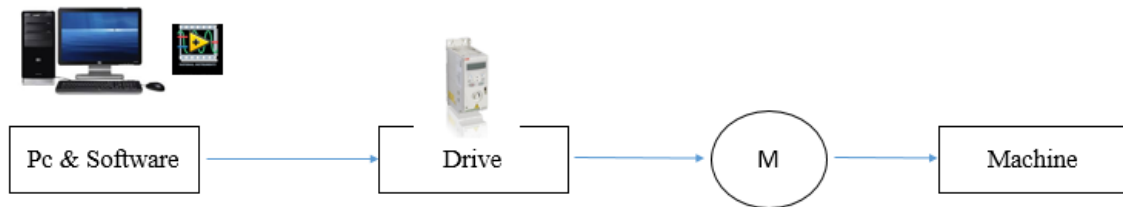


Figure 5.1 Block diagram of control system

Mechanical Apparatus- Parts such as bearing, coupling and shaft which is driven by the motor are constitute mechanical elements.

Motor- In motors, electrical energy is converted into mechanical energy and the required torque is produced to move to the desired target position. [14]

Drive- Drives is a type of motor controller that drives an electric motor by varying the frequency and voltage supplied to the electric motor. Frequency is directly related to the motor speed (RPM) [15].

Application Software- Application software can be used to measure vibration and motion control of the motor. There are many tools that is able to develop software to use for this purpose. In this work, Labview has been chosen because of its easy inegration to external hardware.

Selection of motor and drive:

- First check the initial condition:
To select the correct frequency converter and motor check the mains supply voltage level (380 V to 690V) and frequency (50Hz to 60Hz) [16].
- Check the process requirements:
Is there a need for starting torque? What is the speed range need to used? What type of load will there be? [16].

- Select the motor:

An electrical motor should be a source of torque. The motor's thermal overloadability should not be exceeded. It is also necessary to leave a margin of around 30% for the motor maximum torque when considering the maximum available torque in the dimensioning phase [16]

- Select the frequency converter:

The frequency converter is selected according to the initial conditions and the selected motor. The frequency converter's capability of producing the required current and power should have checked. [16]

5.1 Motor:

An electric motor is a machine which converts electric energy to mechanical energy. It is based on the principle that when a current carrying conductor is placed in a magnetic field, it experiences a mechanical force. An electrical signal is connected to the input of the motor, and the output of the motor delivers a characterized amount of torque related with the characteristics of the motor. To achieve rotation, there must be some interaction between attractive fluxes created by electromagnetism inside the motor. DC motor and AC motor fulfill this task in various way [17].

The main types of motor are two types

- (i) AC motors
- (ii) DC motors

The main difference between AC (Alternating current) and DC (Direct current) can be found in the name itself for, DC motors run on direct current as quite oppose to alternating current on which AC motors will run.

With direct current, energy is acquired from batteries or cells and which produces a constant voltage. This consistent voltage then stimulates the steady flow of electrons in one direction. It is this constant and exact flow of energy that makes direct current most reasonable for applications that require an exact and stable operation, such as speed and torque control applications [18].

5.2 Selection an electric motor:

Before selecting motor we have to consider the application and certain factors :

(i) The load characteristics needed:

The first and most important task one should perform is to determine the load characteristics that will drive the motor speed. These are the torque and speed versus time, for motors must produce acceptable starting torque to start the load [19].

(ii) The motor mounting:

Motors must be securely mounted to the machine, safely, adequately and efficiently. Mounts for the electric motor are available in the difference of style. The user has to select the right mounting motor as per machine purpose [19].

(iii) The motor enclosure:

Motors must be able to operate in all types of environments. The environment in which the motor is operating effects the type of enclosure needed for the motor. The motor enclosure is two type: open enclosure and totally enclosed [19].

(iv) The motor drive:

To select a drive, one must consider the speed of equipment and the size of the drive [19].

(v) The effect of power supplies to the system.

Before selecting, we must determine if needed is single phase or three phase motor. As per application and load, we should select the single phase or three phase motor [19].

(vi) The maintenance and service

It is very important to know about the service provided by the company. Maintenance is also an important factor to maintain the equipment in good working condition [19].

5.3 Calculation of Power and Torque of motor:

Rotating mass : 1.5 kg,

The diameter of rotating mass: 120mm =0.12m,

The speed of the motor:3000 rpm

Moment of mass inertia

$$I = \frac{m \cdot R^2}{2} \text{ kgm}^2 \quad (5.3.1)$$

Where m – rotating mass, Kg;

R – Radius of rotating mass,m.

Then

$$I = \frac{mR^2}{2} = \frac{1.5 * 0.06^2}{2} = 0.0027\text{kgm}^2 \quad (5.3.2)$$

The torque

$T=I \cdot \varepsilon$,

Where ε - angular acceleration, rad/s²

Let us consider achieving of maximal speed in 10 seconds. Then the angular acceleration can be as show in below

$$\varepsilon = \frac{\omega}{t} = \frac{\pi \cdot n}{30 * t} \quad (5.3.3)$$

Where ω – angular velocity, rad/sec,

t – Time,sec;

n – Fecquency of rotation,

Then

$$\varepsilon = \frac{\pi \cdot n}{30 * t} = \frac{3.14 * 3000}{30 * 10} \approx 31\text{rad/s}^2 \quad (5.3.4)$$

The torque will be equal to

$T=I \cdot \varepsilon=0.0027*31 \approx 0.0837 \text{ Nm}$



The power of motor

$$P = T \cdot \omega = T \cdot \frac{\pi \cdot n}{30} = 0.0837 * \frac{3.14 \cdot 3000}{30} \approx 26 \text{ W} \quad (5.3.4)$$

5.4 Selection of motor:

Tabel 5.1 Difference between siemens and ABB

Siemens	ABB																								
<ul style="list-style-type: none"> • The siemens industry sector is the world's leading supplier of innovation and environmentally friendly products and solutions for industrial companies [20]. • Low voltage squirrel-cage motor with the aluminum frame is suitable for a wide range of stand drive tasks in the industrial environment. • Extremely easy selection of energy-saving motors due to the efficiency classification IE2. 	<ul style="list-style-type: none"> • ABB is one of the best company in the present market to help every industry and application reach new levels of efficiency and energy saving. With the help of the best available materials with superior technology, the electric motor is designed to operate reliably [21]. • High-efficiency motors to meet IE2 regulations. • Energy efficient low noise design with suitable DOL starting or Inverter drive use [22]. 																								
<ul style="list-style-type: none"> • Specification: <table> <tr> <td>Power Rating</td> <td>0.75 kW</td> </tr> <tr> <td>Motor type</td> <td>1LE1</td> </tr> <tr> <td>Efficiency Class</td> <td>IE2</td> </tr> <tr> <td>Supply Voltage</td> <td>230 V-400 V</td> </tr> <tr> <td>Speed</td> <td>1440 rpm</td> </tr> <tr> <td>Number of poles</td> <td>4</td> </tr> </table>	Power Rating	0.75 kW	Motor type	1LE1	Efficiency Class	IE2	Supply Voltage	230 V-400 V	Speed	1440 rpm	Number of poles	4	<ul style="list-style-type: none"> • Specification: <table> <tr> <td>Power rating</td> <td>0.7 kW</td> </tr> <tr> <td>Motor type</td> <td>M2BAX 80MA 2</td> </tr> <tr> <td>Speed</td> <td>2830 rpm</td> </tr> <tr> <td>Efficiency class</td> <td>IE2</td> </tr> <tr> <td>Supply voltage</td> <td>400 V</td> </tr> <tr> <td>Number of pole</td> <td>4</td> </tr> </table>	Power rating	0.7 kW	Motor type	M2BAX 80MA 2	Speed	2830 rpm	Efficiency class	IE2	Supply voltage	400 V	Number of pole	4
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Efficiency class	IE2																								
Supply voltage	400 V																								
Number of pole	4																								

Phase	3	Phase	3
Mounting type	Foot	Mounting Type	Foot
Current Rating	1.79A	Current rate	1.7 A
Frequency	50 Hz	Frequency	50 Hz
 <p>Figure 5.2.ABB M2BAX AC motor [23]</p>		 <p>Figure 5.3.Siemens 1LE1 AC motor [24]</p>	

In my work, I selected ABB M2BAX AC motor for my equipment taking into account efficiency of the motor, safety factor, and power. For this project I have select AC motor because it is cheaper then DC. Comparing all the factor between ABB and siemens, ABB M2BAX AC motor has required speed 3000 rpm (rotating per minutes) and also ABB has the edge over siemens when it comes to planning for the future, and the being able to put those ideas into action.Siemens is a multination company and they haven` t been able to come up with innovations that ABB has managed to create and implement idea in the market. It easy enough to understand that most professionals in consider ABB to be the up and coming company of the future [25].

5.5 Drive:

When selecting a Variable frequency drive (VFD), several decisions must be made besides the obvious voltage and current selections. Even the name ought to be chosen, as it is frequently called a variable speed drive. The VFD system is used to control some action such as speed of the motor and direction of the motor, forward and reverse direction. Control system is very widely used in the industrial application wherein the speed control is a very important issue due to load changes wherein speed should be an increase or decrease accordingly [26].

The variable frequency drive operational principle is basic functionality which converts the incoming electric supply of fixed frequency and voltage into a variable frequency and variable voltage that is output to the motor.

Electric motor systems are responsible for more than 65% of the power consumption in industry today. Optimizing motor control systems by installing VFDs can reduce energy consumption in your facility by as much as 70%. Additionally, the usage of VFDs improves product quality and reduces production costs [15].

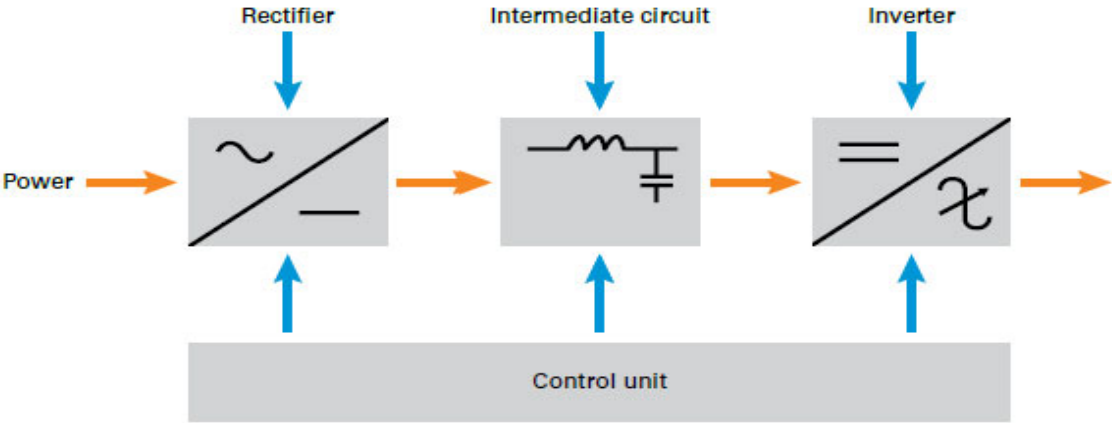


Figure 5.4 Block Diagram of Drive [15]

The above figure.5.4 shows the work flow of the drive. Rectifier the basic principle of the rectifier is changing alternating current (AC) to Direct current (DC). In the intermediate circuit normally by a combination of inductors, capacitors and resistor. Most of the VFD in the market use the fixed-voltage DC link. The inverter converts the rectified and condition DC link

into an AC supply of variable frequency and voltage. As control unit controls the whole operation of the variable frequency drive [15].

Ratings IP20/UL Open type/ NEMA 1 option			Type designation	Frame size IP20	Frame size IP66
P_N [kW]	P_N [hp]	I_{2N} [A]			
1-phase AC supply, 200 to 240 V					+B063
0.37	0.5	2.4	ACS355-01X-02A4-2	R0	-
0.75	1.0	4.7	ACS355-01X-04A7-2	R1	-
1.1	1.5	6.7	ACS355-01X-06A7-2	R1	-
1.5	2.0	7.5	ACS355-01X-07A5-2	R2	-
2.2	3.0	9.8	ACS355-01X-09A8-2	R2	-
3-phase AC supply, 200 to 240 V					+B063
0.37	0.5	2.4	ACS355-03X-02A4-2	R0	R1
0.55	0.75	3.5	ACS355-03X-03A5-2	R0	R1
0.75	1.0	4.7	ACS355-03X-04A7-2	R1	R1
1.1	1.5	6.7	ACS355-03X-06A7-2	R1	R1

Figure 5.5 Drive parameters [27]

In my work, ACS355-03X-04A7-2 AC drive by ABB has been selected for frequency control. The reason behind the selection of VFD control because it has advantage of energy savings, low motor starting current, high power factor and lower kVA(kilovolt-amps) [13]. ACS355-03X-04A7-2 AC is selected for this project based on power supply voltage, motor power and current rating. It is flexible to performance reduce the need for external PLC (programmable logic controller) components with built-in sequence programming. Quick and easy commissioning. It is compact and uniform design with remote monitoring [27].

5.6 Drive connection:

The selected drive is equipped with fieldbus interface to send and receive all its control information [13]. The drive is connected via RS-232 serial connection to the computer [13]. ABB ACS355 AC Drive can receive start, stop and directional commands and reference values from the integrated control panel [13].

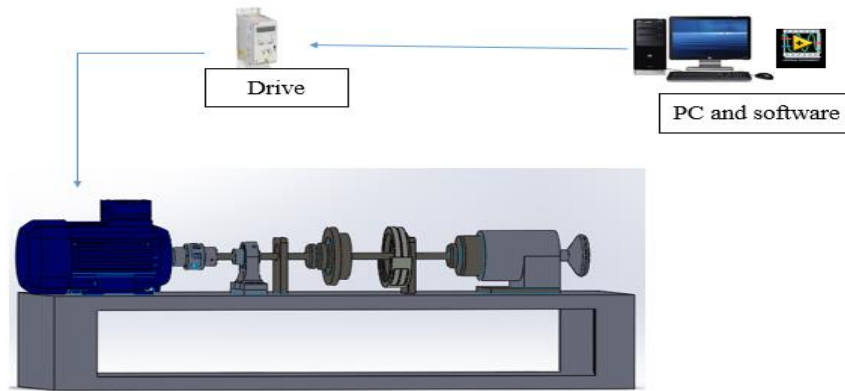


Figure 5.6 Control system of drive

The drive is connected to an external control systems through embedded fieldbus. The embedded fieldbus support Modbus RTU protocol. Modbus is a serial asynchronous protocol [13].

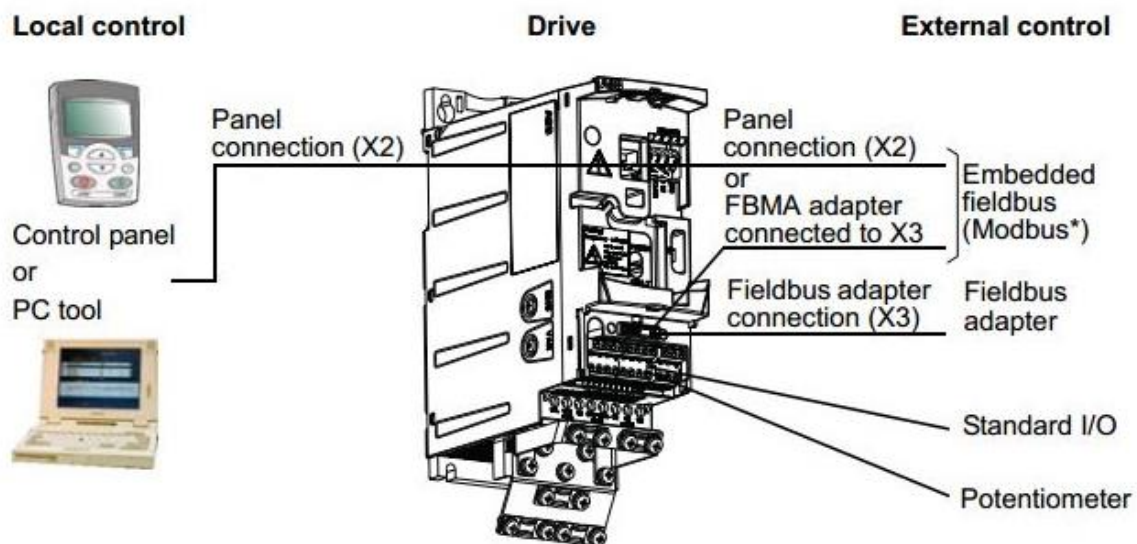


Figure 5.7 SREA-01 ethernet adapter module with modbus [28]

To connect VFD to PC, the RJ45 to serial port conversation is used [13]. It is not possible to directly connect to PC the current configuration. Pin arrangements of this conversation are illustrated in figure 5.7. Variable Frequency Drive is capable of communicating with RS 232 communication port [13]. This drive has an advanced technology that allows users to configure for their application requirements [13].

ABB SREA-01 Ethernet adapter performs all essential remote access tasks. It is designed as an optional remote interface module for ABB industrial drives, standard drives, general machinery drives, DC drives and other ABB products. It can send process or drive data and event messages as well as log data to the drive's internal memory, without the need for PLC or dedicated on site computer. The module internal web server ensures smooth access from standard PCs, mobile phone with web browser [29].



Figure 5.8 SREA-01 Ethernet adapter module [29]

6. Measuring structure and components

Vibration measuring is the major part of this project. Vibration is a time-based (periodic/cyclic) displacement of an object around a center static position. Vibration measuring is complex because of its many components. Also the measurement is in different ways as peak-to-peak, peak, average. Each of them is measured in frequency domain. Monitoring mainly the bearing part in critical application while loading on a machine, it can start to vibrate at the bearing part due to unbalance of the load. Vibration is measured by accelerometer sensor and data will be transferred to software for saving and monitoring desired information [13].

6.1. Accelerometer sensor:

Accelerometer is used as basic vibration sensor which works of the piezoelectric effect, which occurs when a voltage is generated across a certain type of crystals as they are stressed. The acceleration of the test structure is transmitted to a seismic mass inside the accelerometer that generates a proportional force on the piezoelectric crystal. This external stress on the crystal then generates a high-impedance, electrical charge proportional to the applied force and, thus, proportional to the acceleration [30].

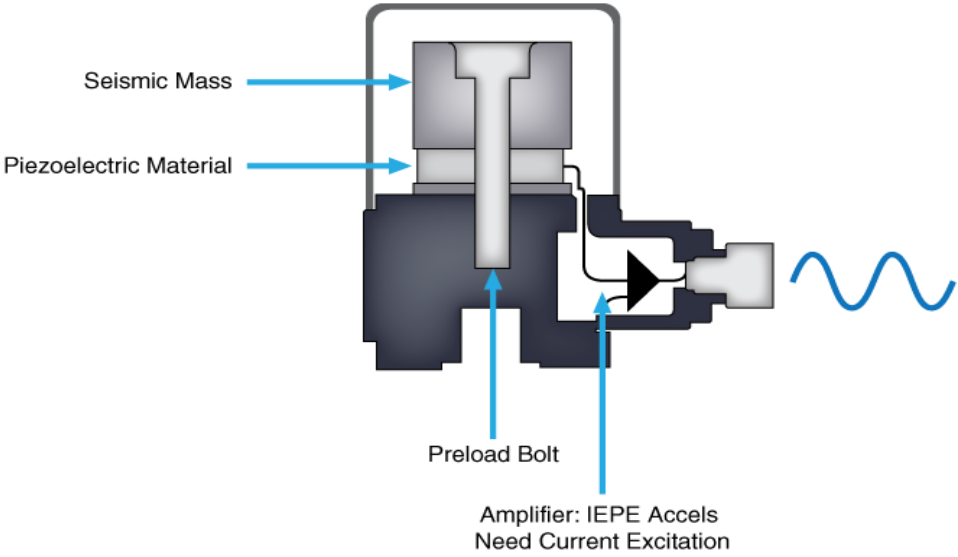


Figure 6.1 Accelerometer sensor with IEPE [30]

In Figure 6.2 it will show three difference type accelerometer sensor. Pressure type of accelerometer, which is the most prone to these parasite loads. Other two Versions of accelerometers are bending and shear types which are more resistive to them. Shear type accelerometer are which have sensitive enough and durable comparing with other two type of sensors. It is most common type of transducers used for vibration measurements. Accelerometer is easy to install on the machine, they have a wide range of frequency from (0.1 Hz to 30 kHz), Accelerometer are also Available for high operation temperatures [31].

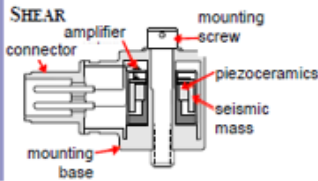
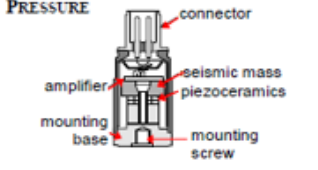
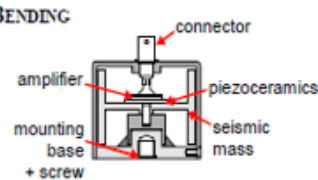
ACCELEROMETER SENSOR	ADVANTAGE	DISADVANTAGE
	<p>Wide frequency range Quite durable Low temperature influence</p>	<p>Lower sensitivity</p>
	<p>Wide frequency range durable to shocks</p>	<p>Sensitive for temperature effect Sensitive for base deformation</p>
	<p>Very low frequency measurements Very high sensitivity</p>	<p>fragile, sensitive for shocks</p>

Figure 6.2 Accelerometer Types [31]

Proper selection sensor is very important for any industrial application. The three main area required before selecting sensor are sensor design, dynamic expectations and application environment.

Specification:

Before selecting accelerometer to an intended application, we must consider the products acceleration range and frequency. A device which measures ± 2 g (gravity) is suitable for measuring the movement of robots, vehicles or aircraft. For applications involving measurements of sudden stops or violent changes in direction, a ± 5 g. [32].

Accelerometers are also limited by range of vibration frequency they can sense, and must be matched to the frequency of the intended application. The most sensitive accelerometers can detect vibration as low as 0.2 Hz, although they are typically inaccurate below frequency 10-15 Hz [32].

The graph below shows a typical frequency-sensitivity curve. The spike which occurs between 10,000 and 50000 Hz is likely due to the accelerometers natural frequency, at which point the sensor experiences a large increase in sensitivity, resulting in oversensitivity and possible saturation. Many devices feature filters which correct this oversensitivity [32].

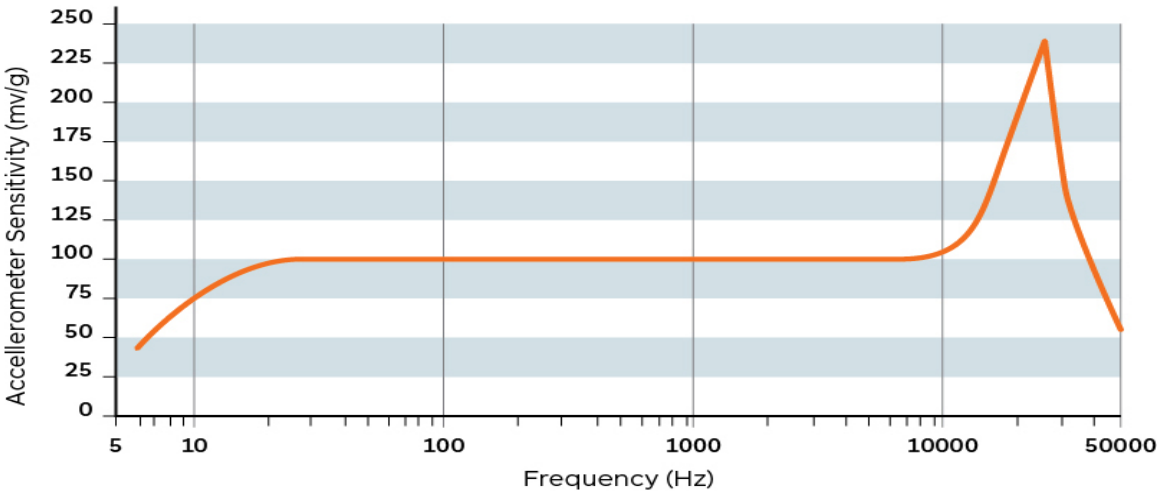


Figure 6.3 Typical frequency-sensitivity curve [32]

Piezoelectrical accelerometer is an accelerometer which work on piezoelectri effect of certain matrial to measure the dynamic behavior in mechanical variables like acceleration, vibration, and mechanical shock. Piezoelectric effect is the ability of certain materials to generate an electric charge in responce to applied mechanical stress. The force exerted on the piezoelectric material can be observed in the change in the electrostatic force or voltage generated by the piezoelectric material [32].

Accelerometer axes:

When selecting an accelerometer, it is important to determine the number of axes the device is designed for:

- Single-axis accelerometers are among the most common types. They are often used to measure simple vibration levels. An example of a single-axis device can be seen in the image under construction and operation above.
- Two-axis accelerometers are designed to measure acceleration or vibration along both x- and y-axis as it can be used for forward, backward and side-to-side.
- Three-axis device add a vertical z-axis to the two-axis accelerometer. These are capable of complex 3- dimensional positioning and measurement.

Accelerometer sensor consist of different types of shear, pressure, bending designs.



Figure 6.4 PCB model 604B32 accelerometer sensor [33].

Vibration is measured by accelerometer model 604B32 by PCB, which has triaxial ring-style, shear ICP accel 500 mV/g, 0.5 to 5 kHz, side exit 4-pin connector [33].

Specification:

- Sensitivity: ($\pm 20\%$)500 mV/g(51 mV/(m/s²))
- Frequency range : (± 3 dB) 30 to 180000 cpm (.5 to 3000 Hz)
- Measurement range : ± 10 g (± 98 m/s²)
- Electrical connector:4 – Pin MIL-C-26482

In my work I selected ICP accelerometer sensor (from PCB Piezotronics, Model No.604B32), directly fixed on a mounting base. To measure vibration due to fault of unbalance the frequencies are typically in the 2-1000 Hz range. As selected vibration range is 0.5 to 3000 Hz which more suitable for my measuring system. As piezoelectrical accelerometer is an accelerometer which work on piezoelectri effect of certain material to measure the dynamic behavior in mechanical variables like acceleration, vibration, and mechanical shock. Piezoelectric effect is the ability of certain materials to generate an electric charge in response to applied mechanical stress. The force exerted on the piezoelectric material can be observed in the change in the electrostatic force or voltage generated by the piezoelectric material.

6.2 Vibration analysis in the domain:

When analyzing vibration data in the time domain (amplitude plotted against time) we are limited to a few parameters in quantifying the strength of a vibration profile like amplitude, peak-to-peak value, and RMS. A simple sine wave is shown below diagram 6.5 [34]

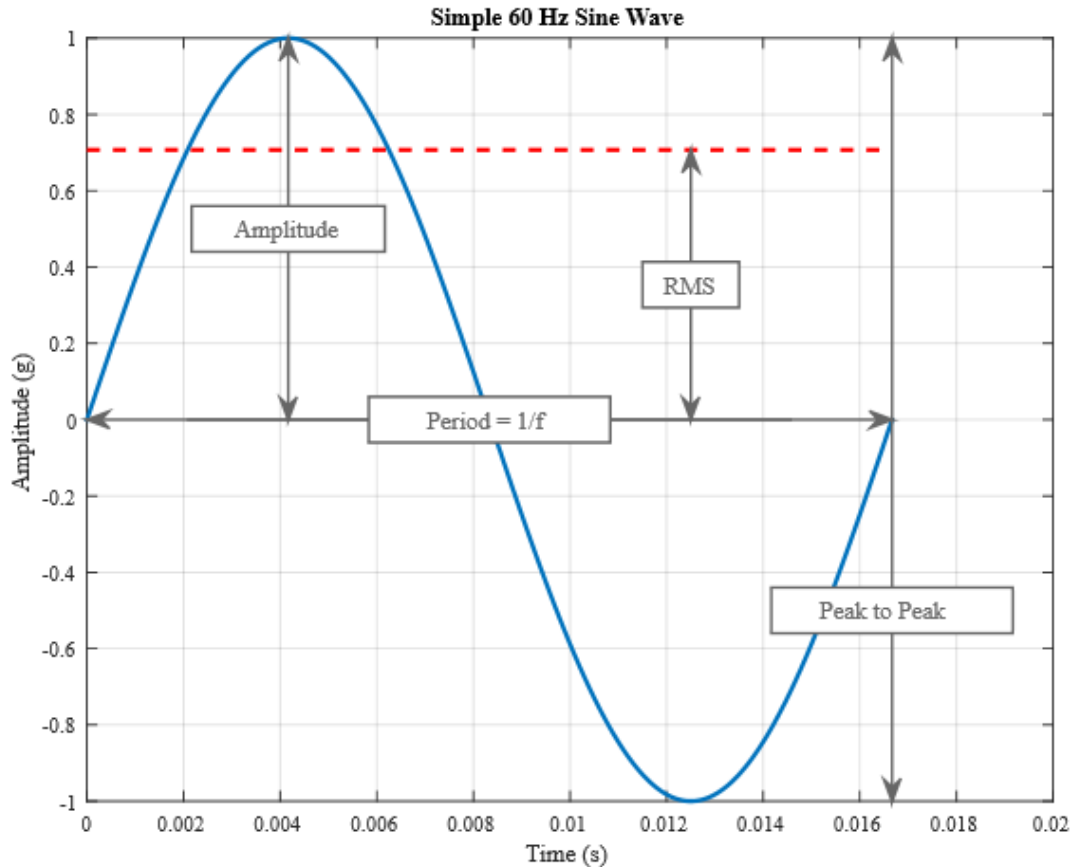


Figure 6.5 A simple 60 Hz sine wave is shown [34]

- The peak or amplitude is valuable for shock events but it doesn't take into account the time duration and thus the energy in the event [34].
- The same is true for peak-to-peak with the added benefit of providing the maximum excursion of the wave, useful when looking at displacement information, specifically clearances [34].
- The RMS (root mean square) value is generally the most useful because it is directly related to the energy content of the vibration profile and thus the destructive capability of the vibration. RMS also takes into account the time history of wave form [34].

Vibration is an oscillating motion about equilibrium so most vibration analysis looks to determine the rate of the oscillation, or the frequency which is proportional to the system's stiffness. The number of times a complete motion cycle occurs during a period of one second is the vibration's frequency and is measured in hertz (Hz). For Simple sine time domain; but as we add different frequency components and noise, we need to perform spectrum analysis to get a clearer picture of the vibration frequency [34].

Fast Fourier Transform(FFT):

FFT Background, Any waveform is actually just the sum of series of simple sinusoids of different frequencies, amplitudes, and phases. A fourier series is that summation of sine waves; and we use fourier analysis or spectrum analysis to deconstruct a signal into its individual sine wave somponents. The results is acceleleration/vibration amplitude as a function of frequency, whcih lets us perform analysis in the frequency domain to gain a deeper understanding of our vibration profile. Most vibration analysis will perform in the frequency domain [34].

The most important thing to understand though is that the number of discrete frequencies that are tested as part of a fourier waveform. With N being the length of the signal, the number of frequency lines or bins is equal to N/2. These frequency bins occer at intervals (Δf) equal to sample rate of raw waveform (F_s) divided by the number of samples(N), which is another way of saying that the frequency resolution is equal to the inverse of the total acquisition time (T) [34].

$$\Delta f = \frac{1}{T} = \frac{F_s}{N}$$

6.3 Measuring vibration with accelerometers:

Accelerometers are small devices that are installed directly on the surface of the vibrating object. It contain a small mass which is subpended by flexible parts that operate like springs. When the accelerometer is moved, the small mass will deflect proportionally to the rate of acceleration. Different type of sensing techniques can be used to measure the amount of deflection of the mass. Because the mass and spring value are know, the amount of deflection is readily converted to an acceleration value. Accelerometers can provide acceleration information in one or more axes. Inactivity vibration measurements in which the forces acting upon the object are critical factor are well served by accelerometers, but they are sensitive to frequency. Vibrations at higher frequencies have greater accelerations than those at lower frequency. So, accelerometers will produce very low signal level for low-frequency vibration and also poor signal to noise ratios.

Accelerometers are a great choice for larger objects, vibrating at higher frequencies, in which the inertial forces acting upon the object is what needs to be measured.

6.4 Data Acquisition (DAQ):

DAQ is used to measure an electrical or physical such as voltage, current, temperature, pressure [13]. DAQ is a combination of software, hardware, and computer for getting signal from the sensor [13]. The common goal of every data acquisition system is acquiring, analysing, and presenting information. Data acquisition systems involve signals, data acquisition devices, sensors, signal conditioning, and application software [35].

Data Acquisition board contains both ADC & DAC and that allows sensors transmit a voltage or current signal directly to a computer [13]. Some of the sensors may require additional components and circuitry to properly produce a signal that can accurately and safely be ready a DAQ device [35].

A computer with programmable software we will control the operation of the DAQ device and is used for processing and visualizing and store the measuring data. We are LabVIEW software to control the application. Software such as LabVIEW controls the acquisition and processing of such data. The following properties of the input signal [35].

- Sampling rate
- Resolution
- Range
- Amplification

6.5 NI 9234

The NI 9234 bridge module contains all the signal conditioning required to power and measure. NI 9234 Measurement system measures 4 accelerometer channels at 51.2 kS/s (kilo samples per second) with 24-bit resolution, sending data to PC. NI 9234 is a 4-channel C series dynamic signal from integrated electronic piezoelectric (IEPE) and non-IEPE sensor with NI compact DAQ. It can measure up to four sensors simultaneously [36].



Figure 6.6 NI 9234 DAQ Board [36]

Properties of NI 9234 DAQ Board:

- 4 Channels, 50 kS/S per channel simultaneous AI
- ± 5 V input range
- 24-bit resolution
- 102 dB dynamic range and anti-aliasing filters
- BNC Connectivity
- -40°C to 70°C operating

6.6 System set up

The figure 6.7 below shows the connection flow diagram of measuring system components. The accelerometer sends signals to the NI9234 DAQ board through the MIL-C to BNC plug connector. The MIL connector is connected to the sensor, and the other end of the BNC plug is connected to the NI 9234 DAQ. The pinout of the MIL connector can be customized for different measurement types. It is not commonly used in every measurement type. BNC is a commonly used connector type for communication in measuring dynamic range. It has Transducer Electronic Data Sheet smart sensor compatibility [37].



Figure 6.7 Accelerometer data acquisition system

6.7 NI USB –9162 DAQ:

NI Compact DAQ USB chassis provide the plug-and-play simplicity of USB to sensor and electrical measurements on the benchtop, in the field, and on the production line [13]. The NI USB-9162 single module carrier is a low-cost and highly portable solution, which delivers high-speed data and ease of use in a flexible mixed-measurement system. [38]

The DAQ-9162 USB has advantage over other PC peripheral buses is easy to use the device detection [13]. DAQ USB chassis is directly connected to a Windows PC and it is automatically detected with no additional configuration needed [13]. Once DAQ-9162 USB is ready to run with the included LabVIEW software [13]. The setup of USB is a good choice for portable applications that may move from one PC to another [38].



Figure 6.8 DAQ-9162 USB Connector [39]

7. Measurement software and real time monitoring

Using LabVIEW, you can create exactly the type of virtual instrument you need, when you need it, at a fraction of the cost of traditional instruments. When you needs change, you can modify your virtual instrument in moments. LabVIEW tries to make your life as easy as possible. It has extensive libraries of function and subroutines to help you with most programming tasks. LabVIEW also contain application-specific libraries of code for data acquisition(DAQ). Gernerall purpose interface bus (GPIB) and serial instrument control, data presentation,data storage and communication over the internet. The analysis library contains a multitude of useful functions, including signal generation,signal processing, filters windows, statistics, linear algebra and array arithmetic.

LabVIEW offers analsis and mathematical which will make easy to natively work together with data acquisition. LabView offer point-by-point execution, this is designed to specifically to meet the needs of real-time applications. Point-by point analysis is essential when dealing with control processes. The point-by-point approach simplifies the design, implementation and test process, because the flow of the application closely matches the natural flow of the real-world processes that the application is monitoring and controlling [40].

Data gathered from measurement is often not immediately useful [13]. Most of the data is stored in a system and it will useful after processing to analysis error in the system. So, specific software is used to collect data and analysis it [13]. LabVIEW is best and useful software [13]. LabVIEW is a software environment is specifically designed for engineer and scientists analyzing data [13]. Regardless of programming experience, LabVIEW makes development fast and easy for all users. It is also known as dataflow programming as it is depending on the structure of the graphical block diagram to execute the user designed program [41].

LabVIEW has virtual instrument (VI) a front panel where you can choose to visualize data or make controls available to your users [13]. VI is easy to create with help of controllers and can displays the specific result and by providing the setting more accurate the result will take less time to get measurement data [13]. Similar to other conventional programming, LabVIEW has standard features such as looping structure,data structures, event-handling, object-oriented programming. Lab VIEW has front-end interface application that allow user to design and then use for controlling systems. In general LabVIEW has three main elements: the front panel, the block diagram and the connector panel. The front panel allows the user to build the contols and indicators. The controls are including Cons, push buttons, dials, and other input mechanism.

Indicators are graphs, LEDs and other output displays. Meanwhile, the block diagram let user to add code user VIs and structures to control the front panel objects.

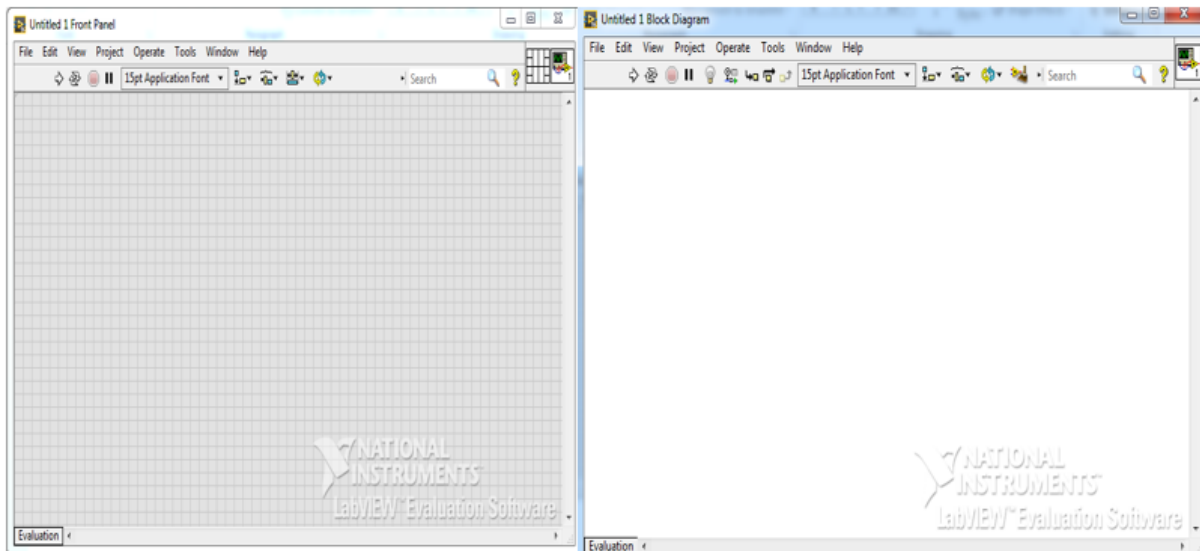


Figure 7.1 Labview front panel & block diagram

LabVIEW can be used to communicate easily between hardware. The drive used in this work is capable of communicating with RS232 communication port. For this Purpose, LabVIEW MODBUS library is needed for communicating with the drive [13].

7.1 MODBUS library for labVIEW

MODBUS library is a set of virtual instruments (VIs) that provide modbus communication from any standard Ethernet or serial port [13]. The LabVIEW library implements the Modbus protocol in software and offers both master and slave functionality [13]. Using the Modbus library, Programmable Automation Controllers (PACs) can communicate with gateway devices connectivity to a wide variety of industrial networks, such as PROFIBUS, EthernetIP and DeviceNet [42].

I selected Labview software for the control system and measurement system because as I need a software to control, collect data and analysis. For this purpose, LabVIEW is very convenient and useful software. MODBUS library in labview is really very important in this project as i am going to control the speed of the motor through MODBUS communication protocol. Instead of using PLC for control as cost will increase.

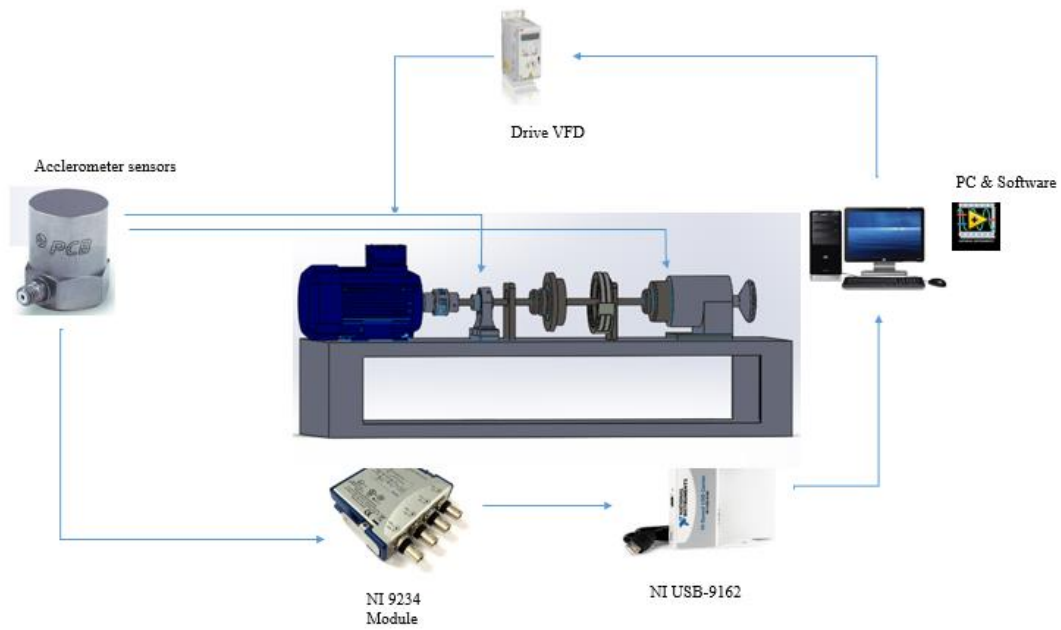


Figure 7.2 Control and vibration measuring system

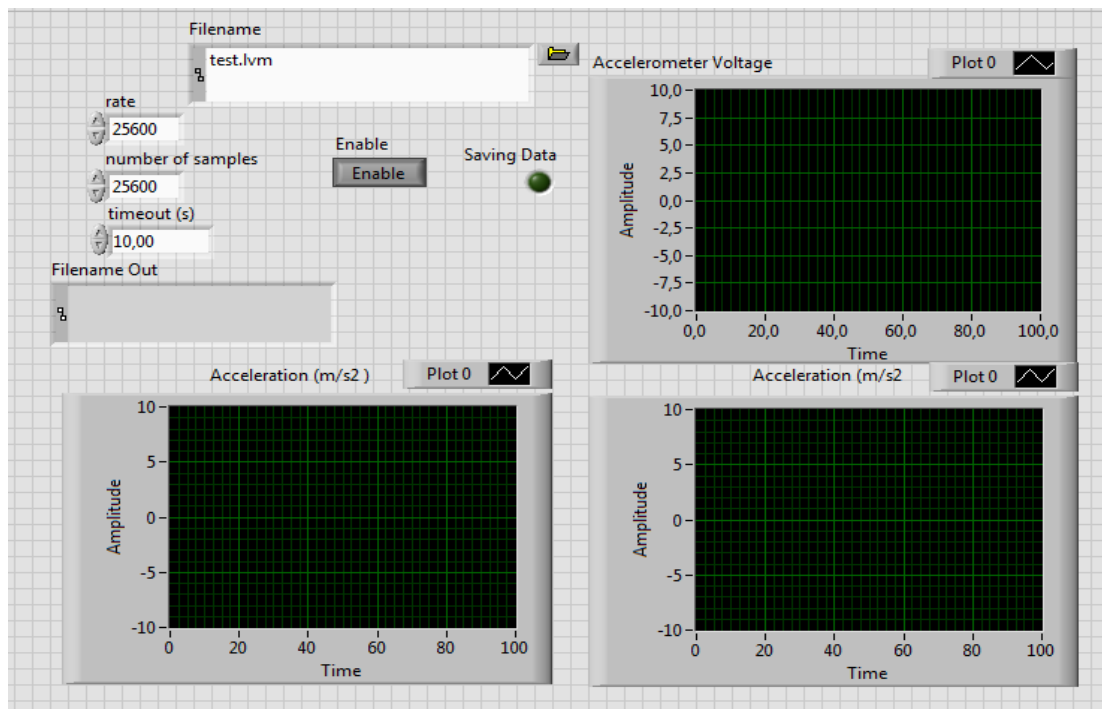


Figure 7.3 Measure control interface

In Lab VIEW, we open new VI file and selected DAQ assistant then we create rate, number of samples and timeout in seconds. one of plot diagram it is will show the raw voltage of accelerometer, Then we convert that raw voltage into acceleration meters per seconds square. In this project we are using two accelerometer sensor. So, the we are using two accelerometer to measure the acceleration of the machine. The data we have measured need to save it and we have analysis it. we can save the file with file name and it can be store in computer with date and time, we can analysis what are the cause for vibration and can take necessary precaution.

LabVIEW offers analysis and mathematical which will make easy to natively work together with data acquisition. LabView offer point-by-point execution, this is designed to specifically to meet the needs of real-time applications. Point-by point analysis is essential when dealing with control processes. The point-by-point approach simplifies the design, implementation and test process, because the flow of the application closely matches the natural flow of the real-world processes that the application is monitoring and controlling.

7.2 Motion control module

For measuring process, it is very important to observe experiment conditions [13]. Speed control of motor is an important factor. So, motion should be adjustable in terms of speed. In order to meet requirement, Software which is developed need to have following features:

- Adjustable frequency
- Adjustable Ramp
- Adjustable Acceleration
- Adjustable Deceleration

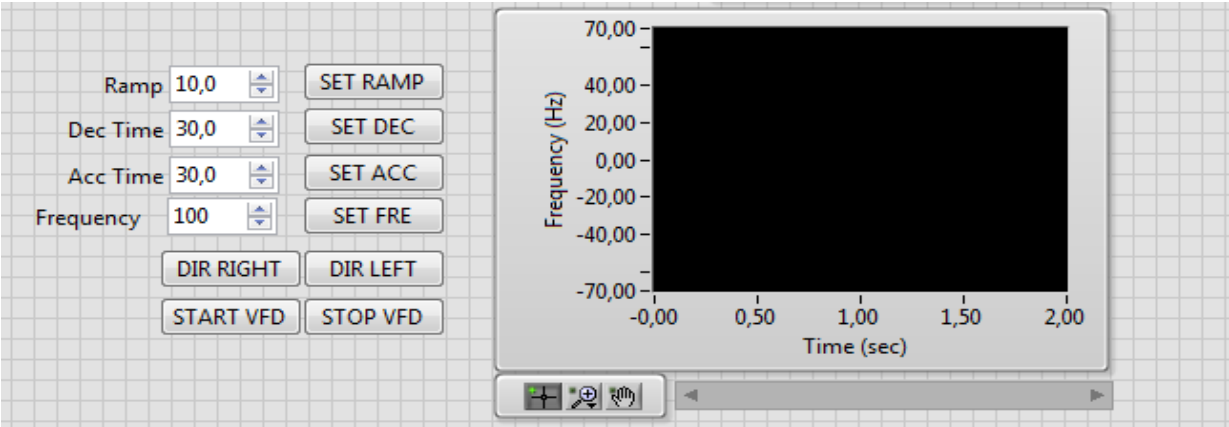


Figure 7.4 Motor control and monitoring software and user interface

Figure 7.4 above represents the user interface of motor control software. The interface has subdivisions for defining of motion profile. And it allows user to control manually and specify the first point for control the speed of the motor. As show in figure 7.4 user can control motor as it can be set forward and reverse direction, can define acceleration time and deceleration time for motor. Motor control is very important to check vibration at different speed. It is easy for user to control the motion of the system with user interface. By using Labview as the motion control, we can control a motor for multiple purposes using only one software environment. With the pushing a button in software, a motor can start, and turn in desired direction. The system can monitor the response in real-time using a waveform graph in the LabVIEW front panel, which eliminates the need for an oscilloscope.

Chapter Summary:

In this chapter measurement of vibration and monitoring software is reviewed. The concept of LabVIEW software includes a module for the configuration of the motor properties for the experiment. By this software, measured data presented on the screen in real time as well as experiment parameters for speed, acceleration and deceleration. It includes a graph which give frequency output. Measurement module is designed as well. By measurement module, LabVIEW software gets signal information through the data acquisition boards and process data.

Summary:

The purpose of this thesis was to develop a vibration measurement test rig by modernizing an existing test rig.

The aims of the thesis are:

- Redesign of a test mechanical system
- Selection of a vibration measuring system
- Selection of a drive control system
- Selection of a data acquisition system,
- Design of a rig user computer interface

At first, the current available machine has been reviewed and its working principle was studied deeply. Vibration theory and vibration on a machine are studied and its parts have been checked and redesigned. It does not have an AC motor which can be used in a motion system, VFD (Variable Frequency Drive) as a controller of motion. It is necessary to build a complete system in this thesis.

Secondly, work is divided into subsystems to set specifications and make it easier to study. The subsystems are defined as follows:

Motion control:

Motion control system is the main part of the machine systems like AC motor and VFD controller. These components were not present in the existing rig. The motor and VFD drive have been selected as per required specifications for the existing rig. A motion control system was designed to start and run based on the desired speed. To control the speed of the motor, a speed control system was developed in LabView software with the help of VFD.

Vibration measuring system:

Vibration measuring mechanism has been absent in the present test rig. According to design, two accelerometer sensors are placed on two bearings. During experiments, after completion of the physical assembly of the machine, an unbalance state is created in the machine, then vibration will develop from the rig and be measured by the accelerometer sensor. Rig material and fixing mechanisms are also considered in this system.

Measuring structure:

In order to measure vibration, accelerometer sensor are used. As a result of measurement, the accelerometer sensor output signal is not directly related to vibration. So,for that purpose, the measuring structure is designed. Data acquisition board is used to collect voltage signal from accelerometer sensor output as a result of acceleration. The signal from DAQ is transferred to PC system via software. Connections of accelerometer sensor, DAQ board and PC were explained.

Measurement software and real-time monitoring

All selection of motor, drive and sensor. It is very important which software are using to control measurement and to monitor the real-time. Measurement, control, and monitoring system in software has been created. Project has been started with a plan that can be get ready in couple of months but during the execution of plan, there are some problem has been faced. First, in motion control system, it has problem to find out how to connect drive to computer. Secondly, in vibration measuring, it is very difficult to select the range and sensitivity of sensor for designed rig.

Considering the process, work has been completed in conceptual level. Motion control system have been selected and their connections and motion control method have been explained. Mechanical system for vibration measurement has been selected. From 3D models diagram from solidworks are used to develop assembly drawings.Vibration measurement system components and the connection are presented. For the future work, these subsystems can be studied in detailed level an system assembled.

Kokkuvõte

Antud töö ülesanne on arendada vibratsiooni mõõtmise katseseadeldis moderniseerides olemasolevat katseseadeldist.

Töö eesmärgid olid:

- Ümber kujundada katseseadeldise mehhaaniline süsteem
- Valida vibratsiooni mõõtmise süsteem
- Valida juhtimissüsteem
- Valida andmete omandamise süsteem
- Disainida katseseadeldise arvutiliides

Esmalt vaadeldi üle vabad näidis seadeldised ning selgitati välja nende töötamise printsiibid. Uuriti vibratsiooni mõõtmise katseseadeldist ning kontrollitud said selle komponendid. Seadeldise ümberkujundamine sai võimalikuks muutmata seadeldise olemasolevaid disaini põhiprintsiipe. Seadeldisel ei olnud AC mootorit ja VFD-d. Võttes arvesse olemasolevaid komponente selgus, et on vajalik luua terviklik süsteem.

Töö on jaotatud mitmeks osaks vastavalt all süsteemide spetsifikatsioonidele, et uurimust oleks lihtsam analüüsida.

Liikumiste/toimivuste kontroll:

Liikumiskontrollide süsteem sisaldab elemente AC mootor ja VFD kui kontrollija. Kuid antud test katseseadeldisel need puuduvad. Töös on valitud mootor ja VFD kontrollimaks süsteemi liikumist ja toimivust.

Vibratsiooni mõõtmise süsteem:

Vibratsiooni mõõtmise mehhanism puudub antud katseseadeldisel. Töös on valitud sensor vastavalt sagedusele ja mõõtepiirkonnale.

Mõõtmise struktuur:

Selleks, et mõõta vibratsiooni on valitud kiirendusmõõtjaga sensor ja mõõtehõive struktuur on kujundatud vastavalt NI 9234 andmehõivesüsteemile.

Mõõtmise tarkvara ja reaajas monitoorimine:

Selleks, et jälgida ning mõõta mõõtmise tulemusi oli vaja vastavat platvormi, kust jälgida kogu protsessi – mõõta signaali reaalsajas ekraanilt.

Uurimust alustades oli eesmärk lõpetada töö mõne kuuga, kuid uuringu käigus ilmnisid kriitilised tegurid, mis pikendasid projekti.

Esmalt jälgides ja uurides liikumisi oli väga raske leida viisi kuidas on võimalik ühendada VDF arvutiga, et toimuks andmevahetus. Teiseks olid raskusi vibratsioonide mõõtmisel. Oli keeruline valida mõõtepiirkonda ning sensori tundlikkuse astet.

Võttes arvesse protsessi, on uurimus läbitud kontseptuaalsel tasemel. Liikumiskontrolli komponendid said valitud ning nende ühilduvus ja mootori kontrolli meetod sai loodud. Mehhaaniline süsteem vibratsiooni mõõtmiseks sai valitud. Detaili joonised tuleb aga teha võttes arvesse olemasolevat 3D mudelit. Mõõtehõive komponendid ja nende ühendatavused on esitletud. Edaspidi on võimalik uurida alamsüsteeme detailsemal tasemel ning luua terviklik kokkupandud süsteem.

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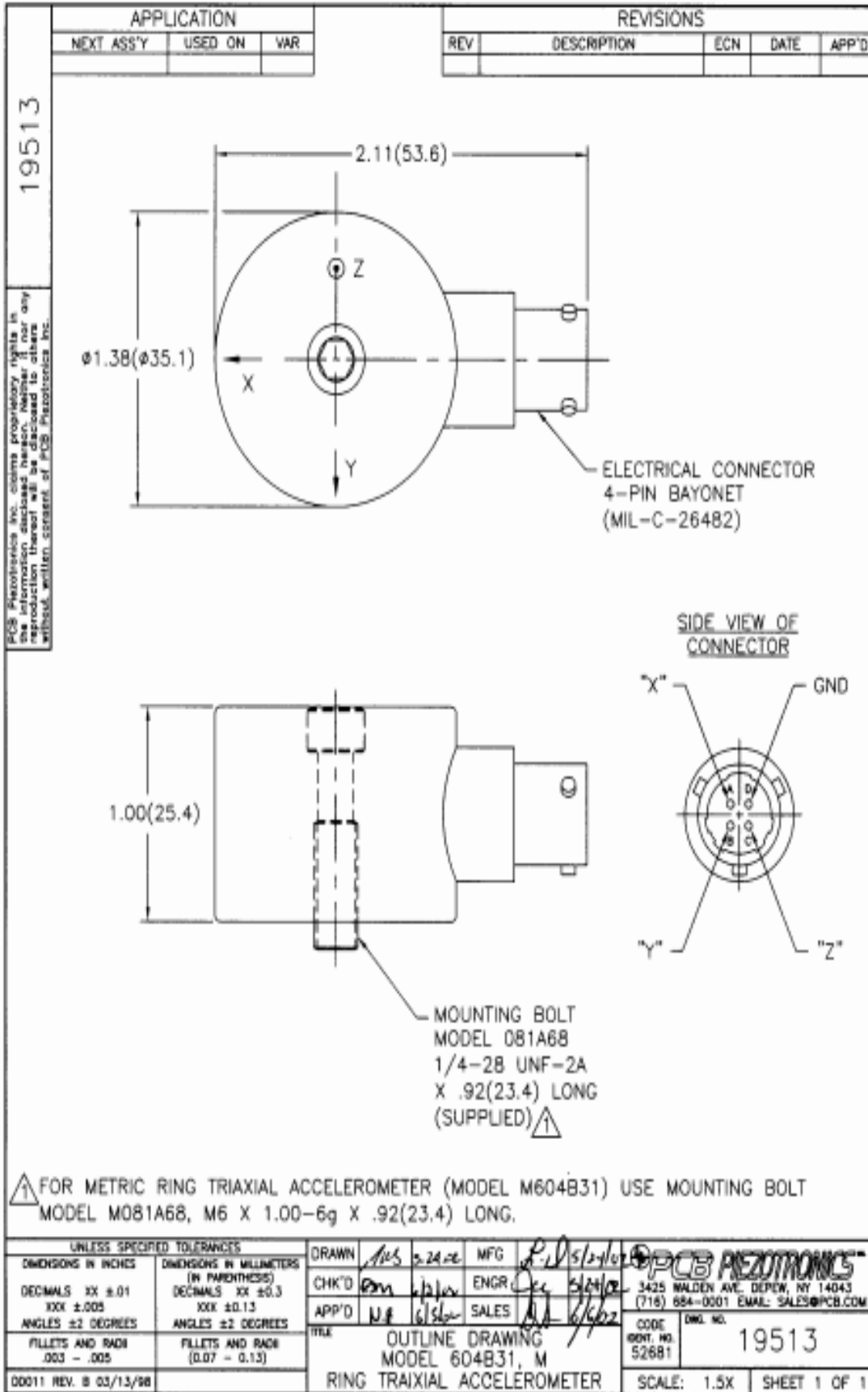
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Appendix A

Accelerometer sensor



Accelerometer specification

Model Number 604B31	INDUSTRIAL ICP® ACCELEROMETER		Revision: A ECN # 27957		
Performance	ENGLISH	SI	OPTIONAL VERSIONS		
Sensitivity(± 20 %)	100 mV/g	10.2 mV/(m/s ²)	[2] Optional versions have identical specifications and accessories as listed for the standard model except where noted below. More than one option may be used.		
Measurement Range	± 50 g	± 490 m/s ²			
Frequency Range(± 3 dB)	30 to 300,000 cpm	0.5 to 5000 Hz	[3]		
Resonant Frequency	600 kcpm	10 kHz	[1]		
Broadband Resolution(1 to 10,000 Hz)	350 µg	3434 µm/s ²	[1]		
Non-Linearity	± 1 %	± 1 %	[4]		
Transverse Sensitivity	≤ 5 %	≤ 5 %			
Environmental					
Overload Limit(Shock)	5000 g pk	49,060 m/s ² pk			
Temperature Range	-65 to +250 °F	-54 to +121 °C			
Temperature Response	See Graph	See Graph	[1]		
Enclosure Rating	IP68	IP68			
Electrical					
Settling Time(within 1% of bias)	≤ 2.0 sec	≤ 2.0 sec			
Discharge Time Constant	≥ 0.3 sec	≥ 0.3 sec			
Excitation Voltage	18 to 28 VDC	18 to 28 VDC			
Constant Current Excitation	2 to 20 mA	2 to 20 mA			
Output Impedance	<150 ohm	<150 ohm			
Output Bias Voltage	8 to 12 VDC	8 to 12 VDC			
Spectral Noise(10 Hz)	8 µg/√Hz	78.5 (µm/s ²)/√Hz	[1]		
Spectral Noise(100 Hz)	5 µg/√Hz	49.1 (µm/s ²)/√Hz	[1]		
Spectral Noise(1 kHz)	4 µg/√Hz	39.2 (µm/s ²)/√Hz	[1]		
Electrical Isolation(Case)	>10 ⁸ ohm	>10 ⁸ ohm			
Physical					
Size (Diameter x Height)	1.38 in x 1.00 in	35.1 mm x 25.4 mm			
Weight	4.4 oz	124 gm			
Mounting	Through Hole	Through Hole			
Mounting Thread	1/4-28 Male	Not Applicable	[5]		
Mounting Torque	2 to 5 ft-lb	2.7 to 6.8 N-m			
Sensing Element	Ceramic	Ceramic			
Sensing Geometry	Shear	Shear			
Housing Material	Stainless Steel	Stainless Steel			
Sealing	Welded Hermetic	Welded Hermetic			
Electrical Connector	4-Pin MIL-C-26482	4-Pin MIL-C-26482			
Electrical Connection Position	Side	Side			
Electrical Connections(Pin A)	X axis	X axis			
Electrical Connections(Pin B)	Y axis	Y axis			
Electrical Connections(Pin C)	Z axis	Z axis			
Electrical Connections(Pin D)	Ground	Ground			
	<p>Typical Sensitivity Deviation vs Temperature</p>				
	<p>All specifications are at room temperature unless otherwise specified. In the interest of constant product improvement, we reserve the right to change specifications without notice. ICP® is a registered trademark of PCB Group, Inc.</p>				
	<p>NOTES: [1] Typical. [2] Conversion Factor 1g = 9.81 m/s². [3] The high frequency tolerance is accurate within ±10% of the specified frequency. [4] Zero-based, least-squares, straight line method. [5] 1/4-28 has no equivalent in S.I. units. [6] See PCB Declaration of Conformance PS023 for details. [7] Division 1, Class I, Group C & D, Temperature Code T4</p>				
	<p>SUPPLIED ACCESSORIES: Model 081A68 Captive mounting bolt 1/4-28 x .90" (1)</p>				
	Entered: JH	Engineer: JEC	Sales: JRA	Approved: JH	Spec Number:
	Date: 1-11-08	Date: 1-10-08	Date: 1-11-08	Date: 1-11-08	21814
	<p>IMI SENSORS A PCB PIEZOTRONICS DIV. 3425 Walden Avenue, Depew, NY 14043</p>		<p>Phone: 800-959-4464 Fax: 716-684-3823 E-Mail: imi@pcb.com</p>		

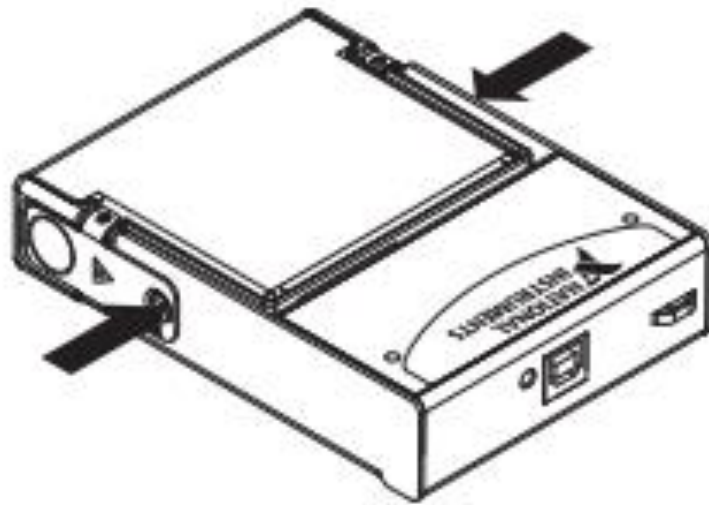
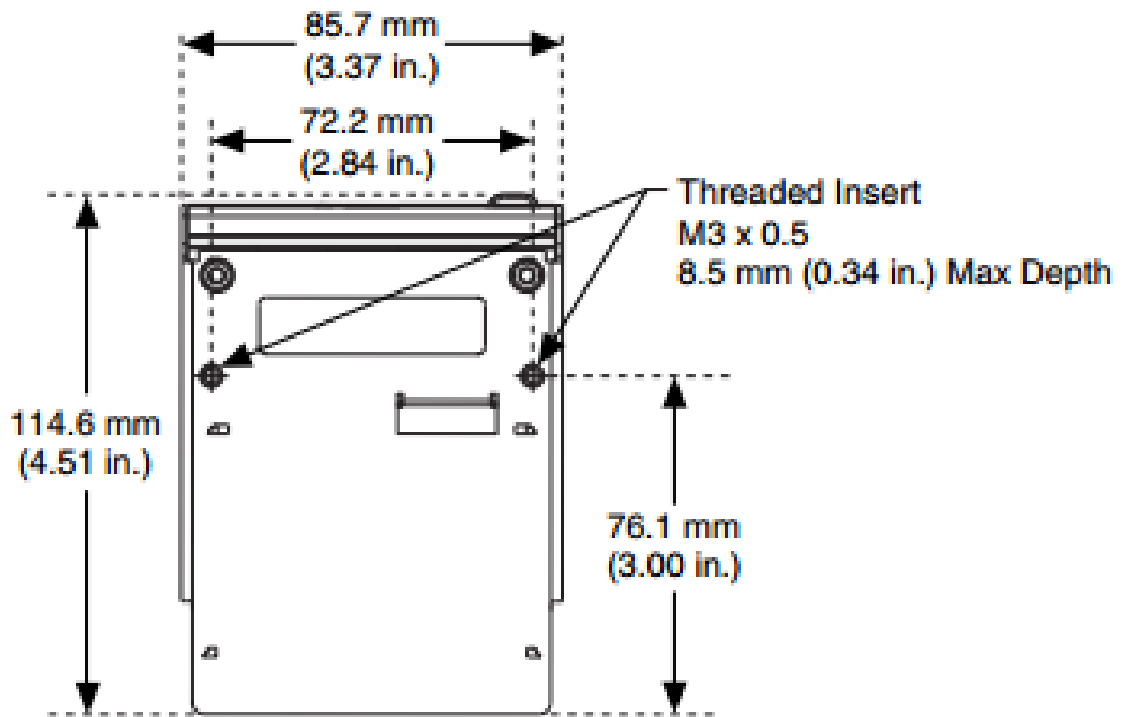
Appendix B

NI 9123

The following specifications are typical for the range –40 to 70 °C unless otherwise noted.

Input Characteristics	
Number of channels	4 analog input channels
ADC resolution	24 bits
Type of ADC	Delta-Sigma (with analog prefiltering)
Sampling mode	Simultaneous
Internal master timebase (f_M)	
Frequency	12.8 MHz
Accuracy	±100 ppm max
Data rate range (f_s) using internal master timebase	
Minimum	2.0 kS/s
Maximum	50 kS/s
Data rate range (f_s) using external master timebase	
Minimum	2.0 kS/s
Maximum	51.3 kS/s
Data rates ¹	
$f_s \leq 25.65$ kS/s	$\frac{f_M \cdot 256}{n}$, $n = 2, 3, \dots, 25$
$f_s > 25.65$ kS/s	$\frac{f_M \cdot 128}{n}$, $n = 2, 3$
Input coupling	AC
AC cutoff frequency response	
–3 dB	0.5 Hz typ
–0.1 dB	4.2 Hz max
Input range	±5 V
AC voltage full-scale range	
Minimum	±5 V _{pk}
Typical	±5.4 V _{pk}
Maximum	±5.8 V _{pk}
Common-mode voltage range (AI– to earth ground)	±2 V max
IEPE excitation current	
Minimum	2.0 mA
Typical	2.2 mA
IEPE compliance voltage	19 V max
Use the following equation to make sure that your configuration meets the IEPE compliance voltage range.	
$V_{\text{common-mode}} + V_{\text{bias}} \pm V_{\text{full-scale}}$ must be 0 to 19, where $V_{\text{common-mode}}$ is any common-mode voltage applied to the NI 9233, V_{bias} is the bias voltage of the accelerometer, and $V_{\text{full-scale}}$ is the full-scale voltage of the accelerometer.	
Overvoltage protection (with respect to chassis ground)	
For an IEPE sensor connected to AI+ and AI–	±30 V
For a low-impedance source connected to AI+ and AI–	–6 to 30 V
Input delay	
≤25.65 kS/s	$12.8/f_s + 3 \mu\text{s}$
> 25.65 kS/s	$9.8/f_s + 3 \mu\text{s}$
Accuracy (–40 to 70 °C)	

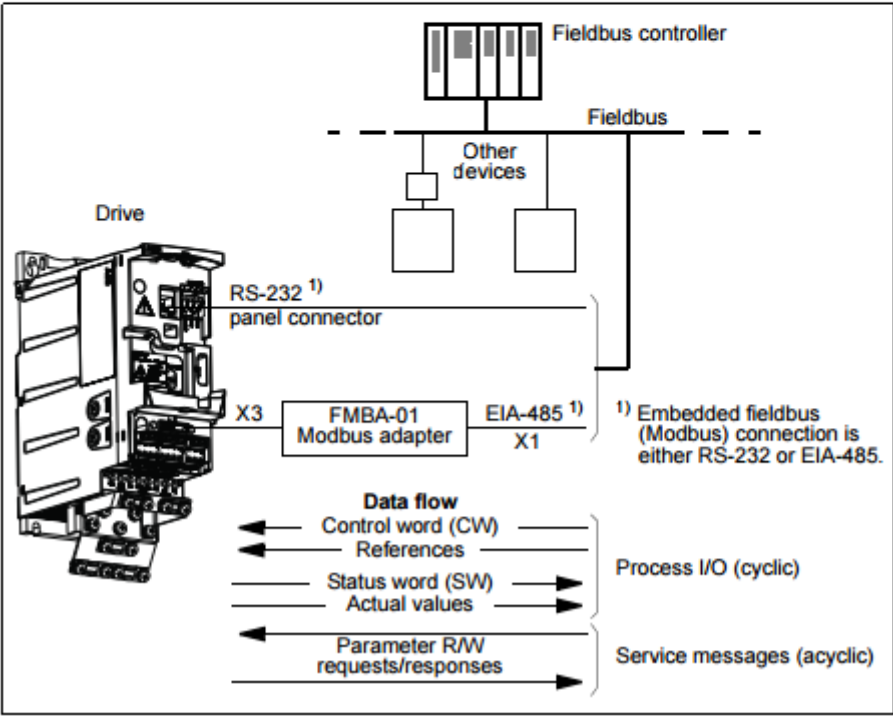
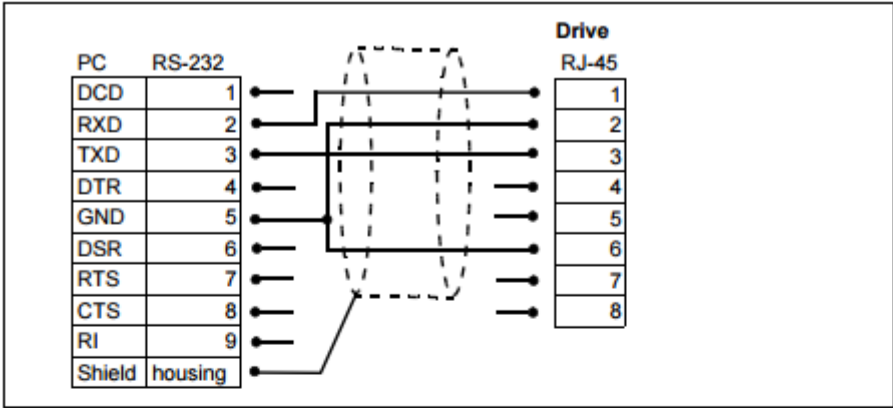
NI USB-9162



Appendix C

Drive

The pin configuration of the RS-232 connector is shown below. The maximum length of the communication cable with RS-232 is restricted to 3 meters (9.8 ft)



Setting up communication through the embedded Modbus

Before configuring the drive for fieldbus control, the FMBA-01 Modbus adapter (if used) must be mechanically and electrically installed according to the instructions given in section Attach the optional fieldbus module on page 38, and the module manual.

The communication through the fieldbus link is initialized by setting parameter 9802 COMM PROT SEL to STD MODBUS or MODBUS RS232. The communication parameters in group 53 EFB PROTOCOL must also be adjusted. See the table below.

Parameter	Setting for fieldbus control	Function/Information	Modbus register address	
CONTROL COMMAND SOURCE SELECTION			ABB DRV	DCU
1001 EXT1 COMMAND S	COMM	Enables 0301 FB CMD WORD 1 bits 0...1 (STOP/START) when EXT1 is selected as the active control location.		40031 bits 0...1
1002 EXT2 COMMAND S	COMM	Enables 0301 FB CMD WORD 1 bits 0...1 (STOP/START) when EXT2 is selected as the active control location.		40031 bits 0...1
1003 DIRECTION	FORWARD REVERSE REQUEST	Enables the rotation direction control as defined by parameters 1001 and 1002. The direction control is explained in section Reference handling on page 323.		40031 bit 2
1010 JOGGING SEL	COMM	Enables jogging 1 or 2 activation through 0302 FB CMD WORD 2 bits 20...21 (JOGGING 1 / JOGGING 2).		40032 bits 20...21
1102 EXT1/EXT2 SEL	COMM	Enables EXT1/EXT2 selection through 0301 FB CMD WORD 1 bit 5 (EXT2); with the ABB drives profile 5319 EFB PAR 19 bit 11 (EXT CTRL LOC).	40001 bit 11	40031 bit 5
1103 REF1 SELECT	COMM COMM+AI 1 COMM*AI1	Fieldbus reference REF1 is used when EXT1 is selected as the active control location. See section Fieldbus references on page 320 for information on the alternative settings.	40002 for REF1	

Parameter	Setting for field bus control	Function/Information	Modbus register address
1106 REF2 SELECT	COMM COMM*AI 1 COMM*AI1	Fieldbus reference REF2 is used when EXT2 is selected as the active control location. See section <i>Fieldbus references</i> on page 320 for information on the alternative settings.	40003 for REF2

OUTPUT SIGNAL SOURCE SELECTION			ABB DRV	DCU
1401 RELAY OUTPUT 1	COMM COMM(-1)	Enables relay output RO control by signal 0134 COMM RO WORD.	40134 for signal 0134	
1501 AO1 CONTENT SEL	135	Directs the contents of the fieldbus reference 0135 COMM VALUE 1 to analog output AO.	40135 for signal 0135	

SYSTEM CONTROL INPUTS			ABB DRV	DCU
1601 RUN ENABLE	COMM	Enables the control of the inverted Run enable signal (Run disable) through 0301 FB CMD WORD 1 bit 6 (RUN_DISABLE); with the ABB drives profile 5319 EFB PAR 19 bit 3 (INHIBIT OPERATION).	40001 bit 3	40031 bit 6
1604 FAULT RESET SEL	COMM	Enables fault reset through the fieldbus 0301 FB CMD WORD 1 bit 4 (RESET); with the ABB drives profile 5319 EFB PAR 19 bit 7 (RESET).	40001 bit 7	40031 bit 4
1606 LOCAL LOCK	COMM	Local control mode lock signal through 0301 FB CMD WORD 1 bit 14 (REQ_LOCALLOC)	-	40031 bit 14
1607 PARAM SAVE	DONE SAVE...	Saves parameter value changes (including those made through fieldbus control) to permanent memory.	41607	
1608 START ENABLE 1	COMM	Inverted Start enable 1 (Start disable) through 0302 FB CMD WORD 2 bit 18 (START_DISABLE1)	-	40032 bit 18
1609 START ENABLE 2	COMM	Inverted Start enable 2 (Start disable) through 0302 FB CMD WORD 2 bit 19 (START_DISABLE2)	-	40032 bit 19

Parameter	Setting for fieldbus control	Function/Information	Modbus register address	
LIMITS			ABB DRV	DCU
2013 <i>MIN TORQUE SEL</i>	COMM	Minimum torque limit 1/2 selection through <i>0301 FB CMD WORD 1</i> bit 15 (<i>TORQLIM2</i>)	-	40031 bit 15
2014 <i>MAX TORQUE SEL</i>	COMM	Maximum torque limit 1/2 selection through <i>0301 FB CMD WORD 1</i> bit 15 (<i>TORQLIM2</i>)	-	40031 bit 15
2201 <i>ACC/DEC 1/2 SEL</i>	COMM	Acceleration/deceleration ramp pair selection through <i>0301 FB CMD WORD 1</i> bit 10 (<i>RAMP_2</i>)	-	40031 bit 10
2209 <i>RAMP INPUT 0</i>	COMM	Ramp input to zero through <i>0301 FB CMD WORD 1</i> bit 13 (<i>RAMP_IN_0</i>); with the ABB drives profile <i>5319 EFB PAR 19</i> bit 6 (<i>RAMP_IN_ZERO</i>)	40001 bit 6	40031 bit 13
COMMUNICATION FAULT FUNCTIONS			ABB DRV	DCU
3018 <i>COMM FAULT FUNC</i>	NOT SEL FAULT CONST SP 7 LAST SPEED	Determines the drive action in case the fieldbus communication is lost.	43018	
3019 <i>COMM FAULT TIME</i>	0.1... 600.0 s	Defines the time between the communication loss detection and the action selected with parameter <i>3018 COMM FAULT FUNC</i> .	43019	
PID CONTROLLER REFERENCE SIGNAL SOURCE SELECTION			ABB DRV	DCU
4010/ 4110/ 4210 <i>SET POINT SEL</i>	COMM COMM*AI 1 COMM*AI1	PID control reference (REF2)	40003 for REF2	

The new settings will take effect when the drive is next powered up, or when parameter *5302 EFB STATION ID* setting is cleared and reset.

Drive control parameters

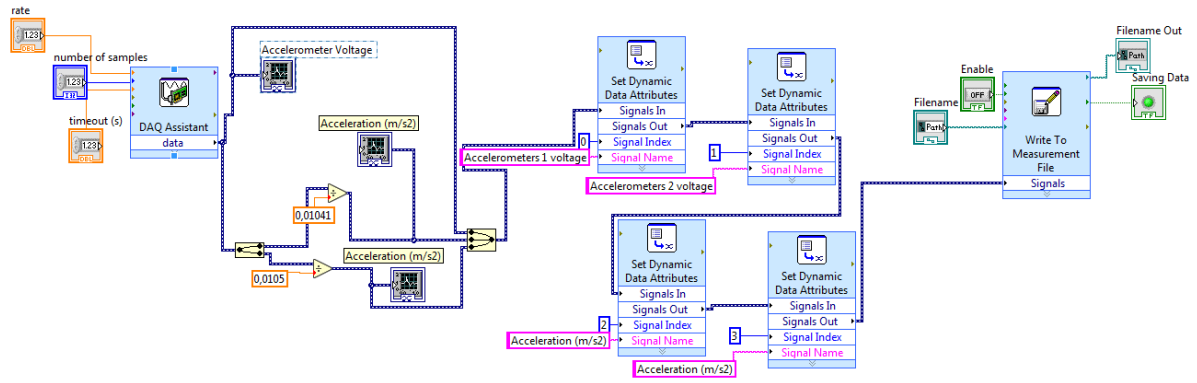
After the Modbus communication has been set up, the drive control parameters listed in the table below should be checked and adjusted when necessary.

The **Setting for fieldbus control** column gives the value to use when the Modbus interface is the desired source or destination for that particular signal. The **Function/Information** column gives a description of the parameter.

Parameter	Setting for fieldbus control	Function/Information	Modbus register address	
CONTROL COMMAND SOURCE SELECTION			ABB DRV	DCU
<i>1001 EXT1 COMMAND S</i>	<i>COMM</i>	Enables <i>0301 FB CMD WORD 1</i> bits 0...1 (<i>STOP/START</i>) when EXT1 is selected as the active control location.		40031 bits 0...1
<i>1002 EXT2 COMMAND S</i>	<i>COMM</i>	Enables <i>0301 FB CMD WORD 1</i> bits 0...1 (<i>STOP/START</i>) when EXT2 is selected as the active control location.		40031 bits 0...1
<i>1003 DIRECTION</i>	<i>FORWARD REVERSE REQUEST</i>	Enables the rotation direction control as defined by parameters <i>1001</i> and <i>1002</i> . The direction control is explained in section <i>Reference handling</i> on page 323.		40031 bit 2
<i>1010 JOGGING SEL</i>	<i>COMM</i>	Enables jogging 1 or 2 activation through <i>0302 FB CMD WORD 2</i> bits 20...21 (<i>JOGGING 1 / JOGGING 2</i>).		40032 bits 20...21
<i>1102 EXT1/EXT2 SEL</i>	<i>COMM</i>	Enables EXT1/EXT2 selection through <i>0301 FB CMD WORD 1</i> bit 5 (<i>EXT2</i>); with the ABB drives profile <i>5319 EFB PAR 19</i> bit 11 (<i>EXT CTRL LOC</i>).	40001 bit 11	40031 bit 5
<i>1103 REF1 SELECT</i>	<i>COMM COMM*AI1 COMM*AI1</i>	Fieldbus reference REF1 is used when EXT1 is selected as the active control location. See section <i>Fieldbus references</i> on page 320 for information on the alternative settings.	40002 for REF1	

Appendix D

Accelerometer measurement system labview block diagram.



Appendix D

MATLAB Programs:

The mass-spring-damper system is solved with Euler method is implemented as below matlab code: [6]

```
function Oe=MSDSE(m,b,k,F0,w,x0,v0)
% Solver for Mass-Spring-Damper System with Euler Method
% ----- Input argument -----
% m: mass for particle
% b: damping coefficient
% k: spring constant
% F0: amplitude of external force
% w: angular frequency of external force
% x0: initial condition for the position x(0)
% v0: initial condition for the velocity v(0)
% ----- Output argument -----
% t: time series with given time step from ti to tf.
% x: state variable matrix for corresponding time t matrix
% Define time step in Euler method
dt=0.1;
% Make time series with given time step
t=[0:dt:50]';
% Initialize output matrix
% the 1st column: the position of the particle
% the 2nd column: the velocity of the particle
x=zeros(length(t),2);
% the 1st row has initial conditions
x(1,:)=[x0 v0];
% Start the simulation with Euler method
for i=1:length(t)-1
% Apply Euler method
%  $x(n+1)=x(n)+v(n)*dt$ 
x(i+1,1)=x(i,1)+x(i,2)*dt;
%  $v(n+1)=v(n)+a(n)*dt$ 
% where  $a(n)=1/m*(F0\sin(wt)-bv(n)-kx(n))*dt$ 
```

```

x(i+1,2)=x(i,2)+(1/m)*(F0*sin(w*t(i))-b*x(i,2)-k*x(i,1))*dt;
end
% Extract only particle position trajectory
Oe=[t,x(:,1)];
end. [6]

```

Mass-spring-spring damper system with Runge-Kutta method as solved with following code [6]

```

function Or=MSDSRK(m,b,k,F0,w,x0,v0)
% Solver for Mass-Spring-Damper System with Runge-Kutta Method
% ----- Input argument -----
% m: mass for particle
% b: damping coefficient
% k: spring constant
% F0: amplitude of external force
% w: angular frequency of external force
% x0: initial condition for the position x(0)
% v0: initial condition for the velocity v(0)
% ----- Output argument -----
% t: time series with given time step from ti to tf.
% x: state variable matrix for corresponding time t matrix
% define time steps for solver and display
dt=0.1;
% set both initial time step size and maximum step size
% for Runge-Kutta solver
options=odeset('InitialStep',dt,'MaxStep',dt);
% set time span to be generated from Runge-Kutta solver
% from 0 sec to 50 sec with 0.1 sec time step
td=[0:dt:50];
% Solve differential equation with Runge-Kutta solver
[t,x]=ode45(@(t,X)MSD(t,X,m,b,k,F0,w),td,[x0;v0],options);
% Extract only particle position trajectory
Or=[t x(:,1)];

```

```

end
function dX=MSD(t,X,m,b,k,F0,w)
%
% With two 1st order diffeential equations,
% obtain the derivative of each variables
% (position and velocity of the particle)
%
% t: current time
% X: matrix for state variables
% The first column : the particle position
% The second column : the particle velocity
% m,b,k,F0,w: parameters for the system
%
% Apply two 1st order differential equations
%  $dx(n+1)/dt = -v(n)$ 
dX(1,1)=X(2,1);
%  $dv(n+1)/dt = 1/m*(F0\sin(wt)-bv(n)-kx(n))$ 
dX(2,1)=(1/m)*(F0*\sin(w*t)-b*X(2,1)-k*X(1,1));
end [6]

```

MATLAB program of rotor unbalance system:

For simulating rotating unbalance created a program in MATLAB, the initial conditions were assumed to be zero [12].

Response of the systems with different natural frequencies to a rotating unbalance [12]

```
%This program solves for the response of a single
%degree of freedom system having a rotating unbalance.
%The equations of motion were derived using MAPLE and
%are valid only for zero initial conditions.
%
mo=3;
m=7;
e=0.1;
wr=4;
zeta=0.05;
tf=10;
t=0:tf/1000:tf;
for i=1:3
wn(i)=input('Enter a natural frequency. ');
wd(i)=wn(i)*sqrt(1-zeta^2);
end
for j=1:3
r=wr/wn(j);
X=mo*e/m*(r^2/sqrt((1-r^2)^2+(2*zeta*r)^2));
phi=atan2(2*zeta*r,(1-r^2));
Z1=(-zeta*wn(j)+wr*cot(phi))/wd(j);
Z2=sqrt((zeta*wn(j))^2-2*zeta*wn(j)*wr*cot(phi)+ ...
(wr*cot(phi))^2+wd(j)^2)/wd(j);
Z=Z1+Z2;
theta=2*atan(Z);
Anum=X*(wd(j)*sin(phi)-Z*zeta*wn(j)*sin(phi)+Z*wr*cos(phi));
Aden=Z*wd(j);
A=Anum/Aden;
xh(j,:)=A*exp(-zeta*wn(j)*t).*sin(wd(j)*t+theta);
xp(j,:)=X*sin(wr*t-phi);
end
x=xp+xh;
for k=1:3
subplot(3,1,k)
plot(t,x(k,:))
title(['Rotating Unbalance with wr=',num2str(wr),'wn=', ...
num2str(wn(k)),'and zeta=',num2str(zeta)]);
grid
```

```
end
xlabel('Time, seconds')
```

The below MATLAB program show Response with varied damping ratios [12]

```
%This program solves for the response of a single
%degree of freedom system having a rotating unbalance.
%The equations of motion were derived using MAPLE and
%are valid only for zero initial conditions.
%
mo=3;
m=7;
e=0.1;
wr=4;
wn=12;
tf=10;
t=0:tf/1000:tf;
for i=1:3
zeta(i)=input('Enter a damping ratio (zeta). ');
end
for j=1:3
wd=wn*sqrt(1-zeta(j)^2);
r=wr/wn;
X=mo*e/m*(r^2/sqrt((1-r^2)^2+(2*zeta(j)*r)^2));
phi=atan2(2*zeta(j)*r,(1-r^2));
Z1=(-zeta(j)*wn+wr*cot(phi))/wd;
Z2=sqrt((zeta(j)*wn)^2-2*zeta(j)*wn*wr*cot(phi)+ ...
(wr*cot(phi))^2+wd^2)/wd;
Z=Z1+Z2;
theta=2*atan(Z);
Anum=X*(wd*sin(phi)-Z*zeta(j)*wn*sin(phi)+Z*wr*cos(phi));
Aden=Z*wd;
A=Anum/Aden;
xh(j,:)=A*exp(-zeta(j)*wn*t).*sin(wd*t+theta);
xp(j,:)=X*sin(wr*t-phi);
end
x=xp+xh;
for k=1:3
subplot(3,1,k)
plot(t,x(k,:))
title(['Rotating Unbalance with wr=',num2str(wr),'wn=', ...
num2str(wn),'and zeta=',num2str(zeta(k))]);
ylabel('Response x')
grid
```



```
end
xlabel('Time, second')
```

The below MATLAB program Effects of varying system mass

```
%degree of freedom system having a rotating unbalance.
%The equations of motion were derived using MAPLE and
%are valid only for zero initial conditions.
%
mo=3;
zeta=0.05;
e=0.1;
wr=4;
wn=12;
tf=10;
t=0:tf/1000:tf;
for i=1:3
m(i)=input('Enter a system mass. ');
end
for j=1:3
wd=wn*sqrt(1-zeta^2);
r=wr/wn;
X=mo*e/m(j)*(r^2/sqrt((1-r^2)^2+(2*zeta*r)^2));
phi=atan2(2*zeta*r,(1-r^2));
Z1=(-zeta*wn+wr*cot(phi))/wd;
Z2=sqrt((zeta*wn)^2-2*zeta*wn*wr*cot(phi)+ ...
(wr*cot(phi))^2+wd^2)/wd;
Z=Z1+Z2;
theta=2*atan(Z);
Anum=X*(wd*sin(phi)-Z*zeta*wn*sin(phi)+Z*wr*cos(phi));
Aden=Z*wd;
A=Anum/Aden;
xh(j,:)=A*exp(-zeta*wn*t).*sin(wd*t+theta);
xp(j,:)=X*sin(wr*t-phi);
end
x=xp+xh;
for k=1:3
subplot(3,1,k)
plot(t,x(k,:))
title(['Rotating Unbalance with wr=',num2str(wr),', wn=', ...
num2str(wn),', mass=',num2str(m(k)),', and rotating mass=', ...
num2str(mo)]);
grid
end
```

```
xlabel('Time, seconds')
```

Appendix E

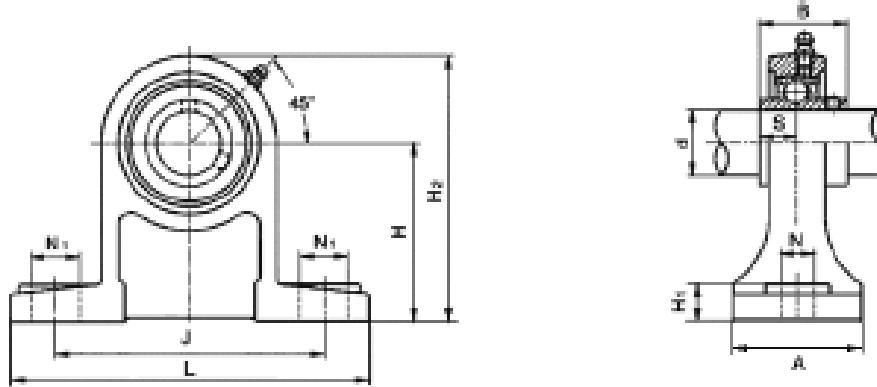
Pillow block mounted ball-bearing

BEARINGS DIRECT.COM

PILLOW BLOCK MOUNTED UNIT BEARING
& SET SCREW LOCKING SET
UCPH 200 SERIES METRIC/INCH

SPECS SHEET

HOUSING DIMENSIONS



Two bolt pillow block unit. Lubrication feature is standard.

UC type ball bearing has dual set screw locking, wide inner ring and anti-rotation device.

P Designate as Pillow Block type Housing.

NOTE: All other material made Housings or/and Ball Bearings are equally matches with the following dimensions.

Unit Number	Shaft Dia.		Dimension (mm) (inch)										Bolt Used (mm) (inch)
	d		H	L	J	A	N	N ₁	H ₁	H ₂	S	B	
	(in)	(mm)											
UCPH201 UCPH201-8	1/2	12	70 2-3/4	127 5	95 3-3/4	40 1-9/16	13 1/2	19 3/4	15 19/32	101 3-31/32	12.7 1/2	31 1.2205	M10 3/8
UCPH202 UCPH202-10	5/8	15	70 2-3/4	127 5	95 3-3/4	40 1-9/16	13 1/2	19 3/4	15 19/32	101 3-31/32	12.7 1/2	31 1.2205	M10 3/8
UCPH203 UCPH203-11	11/16	17	70 2-3/4	127 5	95 3-3/4	40 1-9/16	13 1/2	19 3/4	15 19/32	101 3-31/32	12.7 1/2	31 1.2205	M10 3/8
UCPH204 UCPH204-12	3/4	20	70 2-3/4	127 5	95 3-3/4	40 1-9/16	13 1/2	19 3/4	15 19/32	101 3-31/32	12.7 1/2	31 1.2205	M10 3/8
UCPH205 UCPH205-14 UCPH205-15 UCPH205-16	7/8 15/16 1	25	80 3-5/32	140 5-1/2	105 4-1/8	50 1-31/32	13 1/2	19 3/4	16 5/8	114 4-1/2	14.30 0.5630	34 1.3386	M10 3/8
UCPH206 UCPH206-17 UCPH206-18 UCPH206-19 UCPH206-20	1 1/16 1 1/8 1 3/16 1 1/4	30	90 3-5/8	165 6-1/2	121 4-3/4	50 1-31/32	17 21/32	21 13/16	18 23/32	130 5-1/8	15.90 0.626	38.1 1-1/2	M14 9/16
UCPH207 UCPH207-20 UCPH207-21 UCPH207-22 UCPH207-23	1 1/4 1 5/16 1 3/8 1 7/16	35	95 3.7402	167 6-9/16	127 5.0	60 2-3/8	17 21/32	21 13/16	18 0.7087	140 5-1/2	17.5 0.689	12.9 1.689	M14 9/16
UCPH208 UCPH208-24 UCPH208-25	1 1/2 1 9/16	40	100 3-15/16	184 7-1/4	137 5.3937	70 2.7559	17 21/32	21 13/16	20 0.7874	130 5-29/32	19 0.748	49.2 1-15/16	M14 9/16
UCPH209 UCPH209-27 UCPH209-28 UCPH209-29	1 5/8 1 11/16 1 3/4	45	105 4-9/64	190 7.4803	146 5-3/4	70 2.7559	17 21/32	21 13/16	20 0.7874	138 6-7/32	19 0.748	49.2 1-15/16	M14 9/16
UCPH210 UCPH210-30 UCPH210-31 UCPH210-32	1 7/8 1 15/16 2	50	110 4.337	206 8-1/8	159 6-1/4	70 2.7559	20 0.7874	22 7/8	22 7/8	165 6-1/2	19 0.748	51.6 2.0315	M16 5/8