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A PROJECT REPORT ON

"DESIGN, STUDY AND EVALUATION OF TORSIONAL DAMPED VIBRATION TESTER"

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DEPARTMENT OF MECHANICAL ENGINEERING

CERTIFICATE

Certified that the project work entitled **"DESIGN, STUDY AND EVALUATION OF TORSIONAL DAMPED VIBRATION TESTER"** carried out by **Mr. YASHWANTH K R (4BW14ME032), Mr. MANOJ R(4BW15ME009) and Mr. SACHHIDANANDA N G (4BW15ME049)** a bonafide students of BGS Institute of Technology, B G Nagar in partial fulfilment for the award of **Bachelor of Engineering** in **Mechanical Engineering** of the, **Visvesvaraya Technological University, Belagavi** during the year **2018-2019**. It is certified that all corrections/suggestion indicated for Internal Assessment have been incorporated in the report deposited in the department library.

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13/6/19

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ABSTRACT

A mechanical vibration creates random-magnitude cycles. These vibration cycles makes the parts to be work the parts with different velocities and noise. This effect may tend create load distribution and displacement from one part to another part. All this effect makes failure of machines, mechanical elements results with permanent breakage of machineries. Evaluation of vibration characteristics, magnitudes and nature is one of the critical phenomena. Many methods are there to reduce vibrations by means of damping. In the present study, a torsional vibration tester has been designed, fabricated and evaluated in order to identify the magnitude of vibrations and found out the influence of graded SAE oil on damping characteristics. Three shaft materials such as mild steel, bronze brass with three diameter variations i.e. 5 mm and 6 mm were selected. Three different SAE graded oils i.e. SAE 40, SAE 50 and SAE 90 oils were used as damping fluids. Experiments were conducted for 10 number of cycles and time period was recorded. Natural frequencies and theoretical frequencies were calculated. Theoretical time period and experimental time period was calculated. Logarithmic decrement graphs were obtained for all shaft materials and SAE graded oils. Damping ratio and critical damping factors were calculated and found that copper base alloy materials have good torsional stiffness than and SAE 90 grade oil have better damping characteristics.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

Vibration is a mechanical phenomenon whereby oscillation occur about an equilibrium. A motion which repeats itself after a certain interval of time may be called as vibration. Vibration is the motion of a particle or a body or a system of connected bodies displace from the position of equilibrium. Vibration occurs when a system is displacing from a position of stable equilibrium. The system keeps on moving back and forth across its position of equilibrium.

A vibratory system, in general includes a means for storing potential energy (spring or elastic), a means for storing kinetic energy (mass or inertia), and a means by which energy is gradually lost (damper).

The vibration of a system involves the transfer of its potential energy to kinetic energy and kinetic energy to potential energy, alternatively. If the system is damped, some energy is dissipated in each cycle of vibration and must be replace by an external source if the state of steady vibration is to be maintained.

Most human activities involve vibration in one form or other. For example, we hear because our eardrums vibrate and see because light waves undergoes vibration. Breathing is associated with the vibration of lungs and walking involves oscillatory motion of legs and hands. We speak due to the oscillatory motion of larynges. Early scholars in the field of vibration concentrated their efforts on understanding the natural phenomenon and developing mathematical theories to describe the vibration of physical systems. In recent times many investigation has been motivated by the engineering application of vibration, such as the design of machines, foundation, structures, engines, turbines and control system.

Most prime movers have vibrational problems due to inherent unbalance in the engines. The unbalance may be due to faulty design or poor manufacture. Imbalance in diesel engines for example, can cause ground waves sufficiently powerful to create nuisance in urban areas. The wheels of some locomotives can rise more than a centimeter off the track at highs speeds due to imbalance. In turbines, vibration cause spectacular mechanical failures. Engineers have not yet been able to prevent the failures that result from blade and disc vibrations in turbines. Naturally, the structures designed to supports heavy centrifugal machines, like motor and turbines, or reciprocating machines, like stream and gas engines and reciprocating pumps, are also subjected to vibration.

1.1 Causes of Vibration

- Unbalance forces in the machine
- External excitations
- Dry friction between to mating surfaces
- Earthquakes
- Winds

Most vibration are undesirable as they produce excessive stress, energy losses, increase bearing loads, induce fatigue, undesirable noise, partial or complete failure of parts etc.

This undesirable vibration can be eliminated or reduced by one or more of the following methods.

- Using shock absorbers.
- Dynamic vibration absorbers,
- Resting the system proper vibration isolators.
- Removing the causes of vibration.

1.2 Types of Vibration

Vibrations in a system can be classified into three categories; free, forced and self-excited.

Free vibration of a system is the vibration that occurs in the absence of any force, where damping is may or may not be present.

An external force that acts on the system causes forced vibration.

Self-excited vibrations are periodic and deterministic.

Free Vibration

When no external force acts on the body after giving it an initial displacement, then the body is said to be under free or natural vibration. The oscillation of a simple pendulum is an example of free vibration.

Forced Vibration

When the body vibrates under the influence of external force ten the body is said to be under forced vibration. Machine tools, electric bells etc. are the suitable examples of forced vibration.

Damped and Un-damped Vibrations

If the vibratory system has damper then there is a reduction in amplitude over every cycle of vibration since the energy of the system will be dissipated due to friction. This type of vibration is called as damped vibration.

If the vibratory system has no damper then the vibration is called as un-damped vibration.

Deterministic and Random Vibrations

If the magnitude of the excitation force or motion acting on a vibrating system is known then the excitation is known deterministic. The resulting vibration is known as deterministic vibration.

If the magnitude of the excitation force or motion acting on a vibrating system is unknown, but the averages and deviation are known then the excitation is known as non-deterministic. The vibration is called as random vibration.

Longitudinal, Transverse and Torsional Vibrations

When the particles of the shaft or disc moves parallel to the axis of shaft, then the vibrations are known as longitudinal vibrations and is shown in figure 1.1(a).

When the particles of the shaft or disc moves approximately perpendicular to the axis of the shaft, then the vibrations are known as transverse vibrations and is shown in figure 1.1(b).

When the particles of the shaft or disc moves in a circle about the axis of the shaft i.e., if the shaft gets alternately twisted and untwisted on account on account of vibratory motion, then the vibrations are known as torsional and is shown in figure 1.1(c).

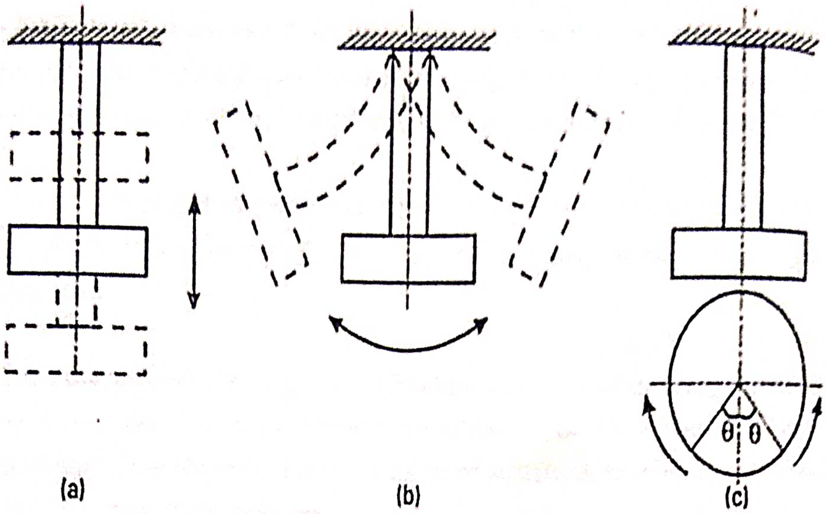


Figure 1.1 (a) Longitudinal, (b) Transverse and (c) Torsional vibration

Transient Vibration

The free vibration continue indefinitely in an ideal system as there is no damping.

There is a reduction in amplitude of vibration continuously because of damping in a real system and vanishes ultimately. The vibration in a real system is known as transient vibration.

1.3 Measurement of Vibration

Vibration measurement is necessary due to the following reasons

- Demands of higher productivity and economical design leads to higher operational speeds of machinery and efficient use of materials through light weight structures.
- To select the operating speeds, it is necessary to known the natural frequencies of a structure or machine in order to avoid resonant conditions.
- Theoretical vibration characteristics may differ from the actual values.
- To identify the system in terms of mass, stiffness and damping, the knowledge of input and the resulting output vibration characteristics are required.
- To design an effective vibration isolation system, measurement of vibration frequency and forces developed are necessary.

- In many application, the survivability of structure or machines in a specified vibration environment is to be determined. If the structure or machine can perform the expected task even after completion of testing under the specified vibration environment, it is expected to survive the specified conditions.

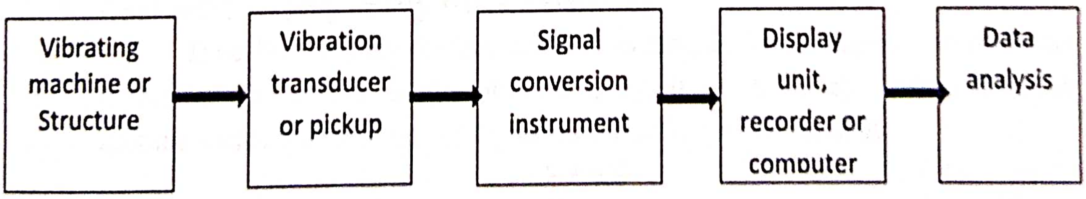
- The measurement of input and the resulting output vibration characteristics of a system helps identifying the system in terms of its mass, stiffness and damping.

- The information about ground vibration due to earthquakes, fluctuating wind velocities on structures, random variation on ocean waves and road surface roughness are important in the design of structure, machines, oil platform and vehicle suspension systems.

Vibration Measurement scheme

The below block diagram shows the basic features of a vibration measurement scheme. The motion (or dynamic force) of the vibrating body is converted into an electrical signal by the vibration transducer or pickup. In general, a transducer is device that transforms changes in mechanical quantities (such as displacement, velocity, acceleration or force) into changes in electrical quantities (such as voltage or current). Since the output signal (voltage or current) of a transducer is too small to be recorded directly, a signal conversion instrument is used to amplify the signal to the required value. The output from the signal conversion instrument can be presented on a display unit for visual inspection or recorded by a recording unit or stored in a computer for later use. The data can be analyzed to determine the desired vibration characteristics of the machine or structure.

Depending on the quantity measured, a vibration measuring instrument is called a vibrometer, a velocity meter, an accelerometer, a phase meter or a frequency meter. If the instrument is designed to record the measured quantity, then the suffix "meter" is to be replaced by "graph" is as shown in below. In some application we need to vibrate a machine or structure to find its resonance characteristics. Foe this, electro-dynamic vibrators, electrohydraulic vibrators and signal generators are used.



CHAPTER 2

LITERATURE REVIEW

Damping is a medium it tends to damp or cease the motion of the oscillator generally by frictional force. A vibration occurring in a damping medium is called damped vibration. As a result of damping a portion of energy of the vibrator is converted to heat energy and thus amplitude of the vibration gradually decreases longitudinally over time.

The damped contain fluid that get heated when compressed by the movement of the suspension (kinetic energy converted into heat energy). This heat energy then dissipated into the surrounding environment every time the fluid is compressed. Creating the damping effect and reducing the intensity of the vibration finally killing its effect.

Although the amount of energy converted an accurate prediction of the vibration response of a system. A damper is assumed to have neither mass nor elasticity, and damping force exists only if there is relative velocity between the two ends of the damper. It is difficult to determine the causes of damping in practical systems. Hence damping is modeled as one or more of the types.

2.1 Types of Damping

1. **Viscous Damping**

   Viscous damping is the most commonly used damping mechanism in vibration analysis. When mechanical systems vibrate in a fluid medium such as air, gas, water, or oil, the resistance offered by the fluid to the moving body causes energy to be dissipated. In this case, the amount of dissipated energy depends on many factors, such as the size and shape of the vibrating body, the viscosity of the fluid, the frequency of vibration, and the velocity of the vibrating body. In viscous damping, the damping force is proportional to the velocity of the vibrating body. Typical examples of viscous damping include (1) fluid film between sliding surfaces, (2) fluid flow around a piston in a cylinder, (3) fluid flow through an orifice, and (4) fluid film around a journal in a bearing.
2. **Coulomb Damping or Dry-friction Damping**

   Here the damping force is constant in magnitude but opposite in direction to that of the motion of the vibrating body. It is caused by friction between rubbing surfaces that either are dry or have insufficient lubrication.

$$F = \mu N$$

Where μ =co-efficient of friction

N =normal force

3. **Material or Solid or Hysteretic Damping.**

When a material is deformed, energy is absorbed and dissipated by the material. The effect is due to friction between the internal planes, which slip or slide as the deformations take place. When a body having material damping is subjected to vibration, the stress-strain diagram shows a hysteresis loop. The area of this loop denotes the energy lost per unit volume of the body per cycle due to damping.

2.2 Free Vibration of an Un-damped Translation System

Using Newton's second law of motion, in this section, section we will consider the derivation of the equation of motion. The procedure we will use can be summarized as follows:

- Select a suitable coordinate to describe the position of te mass or rigid body in the system. Use a linear coordinate to describe the linear motion of a point mass or the centroid of a rigid body, and an angular coordinate to describe the angular motion of rigid body.
- Determine the static equilibrium configuration of the system and measure the displacement of the mass or rigid body from its static equilibrium position.
- Draw the free body diagram of the mass or rigid body when a positive displacement and velocity are given to it. Indicate all the active and reactive forces acting on the mass or rigid body.
- Apply Newton's second law of motion to the mass or rigid body shown by the free-body diagram. Newton's second law of motion can be stated as follows:

Thus, if mass m is displaced a distance x(t) when acted upon by a resultant force F(t) in the same direction. Newton's second law of motion gives

$$\vec{F}(t) = \frac{d}{dt}\left(m\frac{d\vec{x}(t)}{dt}\right)$$

If mass m is constant, this equation reduces to

$$\vec{F}(t) = m\frac{d^2\vec{x}(t)}{dt^2} = m\ddot{\vec{x}}$$

Where

$$\ddot{\vec{x}} = \frac{d^2\vec{x}(t)}{dt^2}$$

is the acceleration of the mass, equation can be stated in words as

Resultant force on the mass = mass × acceleration

For a rigid body undergoing rotational motion, newton's law gives

$$\vec{M}(t) = J\ddot{\theta}$$

Where $\vec{M}$ is the resultant moment acting on the body Θ and $\ddot{\theta} = d^2\theta(t)/dt^2$ are the resulting angular displacement and angular acceleration, respectively. The procedure is now applied to the undamped single-degree-of-freedom system shown in Fig. 2.1(a). Here the mass is supported on frictionless rollers and can have translatory motion in the horizontal direction. When the mass is displaced a distance from its static equilibrium position, the force in the spring is kx, and the free-body diagram of the mass can be represented as shown in Fig. 2.1(c). The application of above equation to mass m yields the equation of motion

$$F(t) = -kx = m\ddot{x}$$

Or

$$m\ddot{x} + kx = 0$$

2.3 Free Vibration of an Un-damped Torsional System

If a rigid body oscillates about a specific reference axis, the resulting motion is called torsional vibration. In this case, the displacement of the body is measured in terms of an angular coordinate. In a torsional vibration problem, the restoring moment may be due to the torsion of an elastic member or to the unbalanced moment of a force or couple.

Figure2.1 shows a disc, which has a polar mass moment of inertia J_o mounted at one end of a solid circular shaft, the other end of which is fixed. Let the angular rotation of the disc about the axis of the shaft be Θ; Θ also represents the shaft s angle of twist. From the theory of torsion of circular shafts, we have the relation

$$M_t = \frac{GI_o}{l}$$

Where, M_t is the torque that produces the twist Θ, G is the shear modulus, l is the length of the shaft, is the polar moment of inertia of the cross section of the shaft, given by

$$I_o = \frac{\pi d^4}{32}$$

and d is the diameter of the shaft. If the disc is displaced by Θ from its equilibrium position, the shaft provides a restoring torque of magnitude M_t. Thus the shaft acts as a torsional spring with a torsional spring constant

$$k_t = \frac{M_t}{\theta} = \frac{GI_o}{l} = \frac{\pi G d^4}{32l}$$

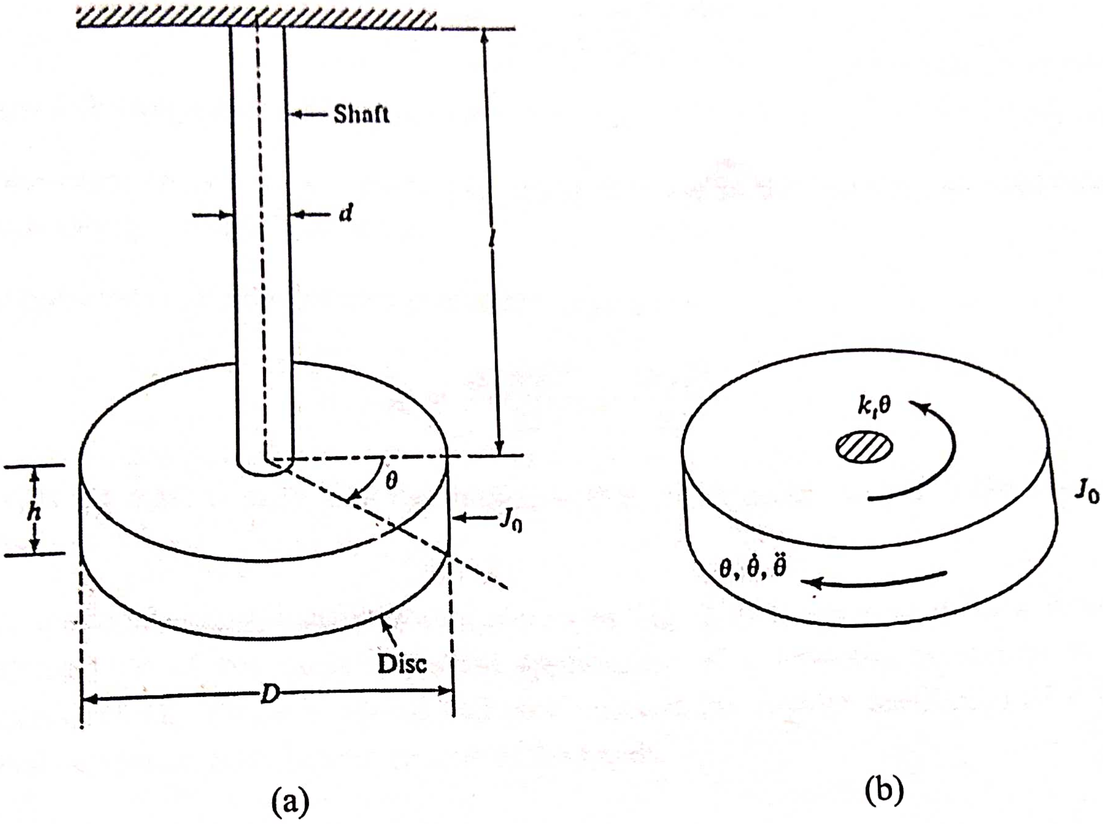


Figure 2.1 Torsional vibration of a disc.

2.3.1 Equation of Motion

The equation of the angular motion of the disc about its axis can be derived by using Newton's second law. By considering the free body diagram of the disc, we can derive the equation of motion by applying Newton s second law of motion:

$$J_0\ddot{\theta} + k_t\theta = 0$$

which can be seen to be identical to above equation if the polar mass moment of inertia J_o, the angular displacement Θ, and the torsional spring constant k_t are replaced by the mass m, the displacement x, and the linear spring constant k, respectively. Thus the natural circular frequency of the torsional system is

$$\omega_n = \left(\frac{k_t}{J_0}\right)^{1/2}$$

and the period and frequency of vibration in cycles per second are

$$\tau_n = 2\pi\left(\frac{J_0}{k_t}\right)^{1/2}$$

$$f_n = \frac{1}{2\pi}\left(\frac{k_t}{J_0}\right)^{1/2}$$

Note the following aspects of this system:

1. If the cross section of the shaft supporting the disc is not circular, an appropriate torsional spring constant is to be used.

2. The polar mass moment of inertia of a disc is given by

$$J_0 = \frac{\rho h\pi D^4}{32} = \frac{WD^2}{8g}$$

Where, is the mass density, h is the thickness, D is the diameter, and W is the weight of the disc.

3. The torsional spring-inertia system shown in Fig. 2.14 is referred to as a torsional pendulum. One of the most important applications of a torsional pendulum is in a mechanical clock, where a ratchet and pawl convert the regular oscillation of a small torsional pendulum into the movements of the hands.

2.3.2 Solution

The general solution of Eq. can be obtained, as in the case of Eq.:

$$\theta(t) = A_1 \cos \omega_n t + A_2 \sin \omega_n t$$

Where ωn is given by Eq. and A1 and A2 can be determined from the intial conditions. If

$$\theta(t = 0) = \theta_0 \quad \text{and} \quad \dot{\theta}(t = 0) = \frac{d\theta}{dt}(t = 0) = \dot{\theta}_0$$

The constants A_1 and A_2 can be found:

$$A_1 = \theta_0$$

$$A_2 = \dot{\theta}_0/\omega_n$$

2.4 Free Vibration with Viscous Damping

2.4.1 Equation of Motion

As stated in Section 1.9, the viscous damping force F is proportional to the velocity or v and can be expressed as

$$F = -cx$$

Where c is the damping constant or coefficient of viscous damping and the negative sign indicates that the damping force is opposite to the direction of velocity. A single-degree-of-freedom system with a viscous damper is shown in Figure. 2.2. If x is measured from the equilibrium position of the mass m, the application of Newton's law yields the equation of motion:

$$m\ddot{x} = -c\dot{x} - kx$$

Or

$$m\ddot{x} + c\dot{x} + kx = 0$$

2.4.2 Solution

To solve, we assume a solution in the form

$$x(t) = Ce^{st}$$

Where, C and s are undetermined constants. Inserting this function into leads to the characteristic equation

$$ms^2 + cs + k = 0$$

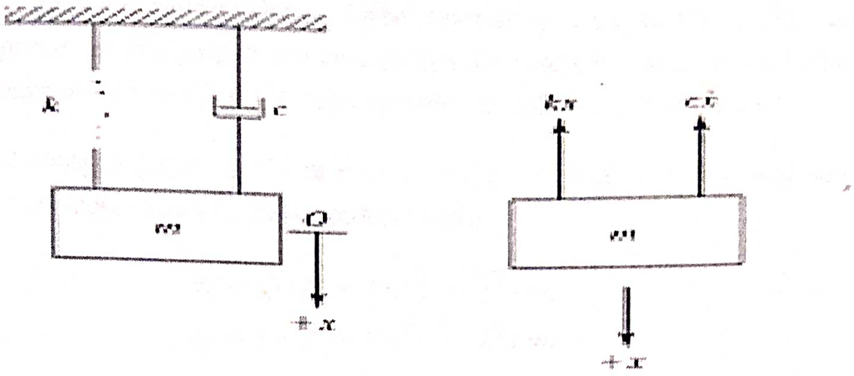
the roots of which are

$$s_{1,2} = \frac{-c \pm \sqrt{c^2 - 4mk}}{2m} = \frac{c}{2m} \pm \sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}}$$

These roots give two solution give to

$$x_1(t) = C_1 e^{s_1 t} \quad \text{and} \quad x_2(t) = C_2 e^{s_2 t}$$

Thus the general solution is given by a combination of the two solutions x_1 and x_2;



System (a)

Free-body Diagram (b)

Figure 2.2 Single degree of freedom system with viscous damper

Critical Damping Constant and the Damping Ratio. The critical damping is defined as the value of the damping constant c for which the radical in Eq. (2.62) becomes zero:

$$x(t) = C_1 e^{s_1 t} + C_2 e^{s_2 t}$$

$$= C_1 e^{\left\{-\frac{c}{2m}+\sqrt{\left(\frac{c}{2m}\right)^2-\frac{k}{m}}\right\}t} + C_2 e^{\left\{-\frac{c}{2m}-\sqrt{\left(\frac{c}{2m}\right)^2-\frac{k}{m}}\right\}t}$$

For any damped system, the damping ratio ε is defined as the ratio of the damping constant

To the critical damping constant:

$$\zeta = c/c_c$$

We can write

$$\frac{c}{2m} = \frac{c}{c_c} \cdot \frac{c_c}{2m} = \zeta\omega_n$$

And hence

$$s_{1,2} = (-\zeta \pm \sqrt{\zeta^2 - 1})\omega_n$$

Thus the solution can be written as:

$$x(t) = C_1 e^{(-\zeta+\sqrt{\zeta^2-1})\omega_n t} + C_2 e^{(-\zeta-\sqrt{\zeta^2-1})\omega_n t}$$

The nature of the roots and hence the behavior of the solution, Eq. (2.69), depends upon the magnitude of damping. It can be seen that the case $\xi = 0$ leads to the undamped vibrations. Hence we assume that $\xi \neq 0$ and consider the following three cases

Case 1. Under-damped system ($\xi < 1$ or $c < c_c$ or $c/2m < \sqrt{k/m}$. For this condition (ξ^2 - 1) is negative and roots s1 and s2 can be represented as;

$$s_1 = (-\zeta + i\sqrt{1-\zeta^2})\,\omega_n$$
$$s_2 = (-\zeta - i\sqrt{1-\zeta^2})\,\omega_n$$

The solution Eq. can be written in different form

$$x(t) = C_1 e^{(-\zeta + i\sqrt{1-\zeta^2})\omega_n t} + C_2 e^{(-\zeta - i\sqrt{1-\zeta^2})\omega_n t}$$
$$= e^{-\zeta\omega_n t}\left\{C_1 e^{i\sqrt{1-\zeta^2}\omega_n t} + C_2 e^{-i\sqrt{1-\zeta^2}\omega_n t}\right\}$$
$$= e^{-\zeta\omega_n t}\left\{(C_1 + C_2)\cos\sqrt{1-\zeta^2}\omega_n t + i(C_1 - C_2)\sin\sqrt{1-\zeta^2}\omega_n t\right\}$$
$$= e^{-\zeta\omega_n t}\left\{C_1'\cos\sqrt{1-\zeta^2}\omega_n t + C_2'\sin\sqrt{1-\zeta^2}\omega_n t\right\}$$
$$= Xe^{-\zeta\omega_n t}\sin\left(\sqrt{1-\zeta^2}\omega_n t + \phi\right)$$
$$= X_0 e^{-\zeta\omega_n t}\cos\left(\sqrt{1-\zeta^2}\omega_n t - \phi_0\right)$$

Where (C'1, C'2), (X, Φ) and (X_o, Φ_o) are arbitrary constants determined from the initial condition.

$$C_1' = x_0 \quad \text{and} \quad C_2' = \frac{\dot{x}_0 + \zeta\omega_n x_0}{\sqrt{1-\zeta^2}\,\omega_n}$$

And hence the solution becomes

$$x(t) = e^{-\zeta\omega_n t}\left\{x_0\cos\sqrt{1-\zeta^2}\,\omega_n t + \frac{x_0 + \zeta\omega_n x_0}{\sqrt{1-\zeta^2}\omega_n}\sin\sqrt{1-\zeta^2}\,\omega_n t\right\}$$

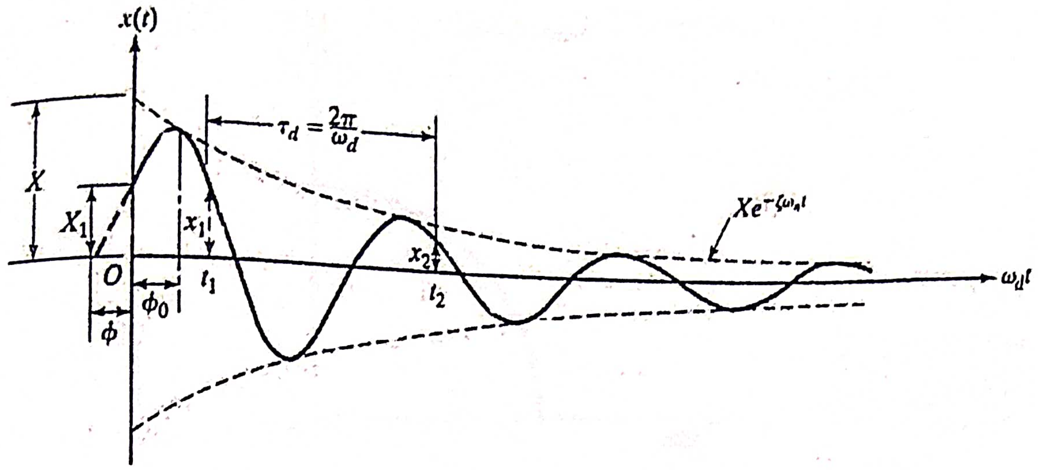


Figure 2.3 Under-damped solution

The constants (X, Φ) and (X_o, Φ_o) can be expressed as

$$X = X_0 = \sqrt{(C_1')^2 + (C_2')^2}$$
$$\phi = \tan^{-1}(C_1'/C_2')$$
$$\phi_0 = \tan^{-1}(-C_2'/C_1')$$

The motion described by above equation is a damped harmonic motion of angular frequency$\sqrt{1-\zeta^2}\omega_n$, but because of the factor $e^{-\zeta\omega_n t}$ the amplitude decreases exponentially with time, as shown in Figure 2.3. The quantity is called the frequency of damped vibration. It can be seen that the frequency of damped vibration ω_d is always less than the undamped natural frequencyω_n. The decrease in the frequency of damped vibration with increasing amount of damping, is shown graphically in Figure 2.3.

The underdamped case is very important in the study of mechanical vibrations, as it is the only case that leads to an oscillatory motion.

Case 2. Critically damped system (ξ = 1 or c = c_c or c/2m = √k/m. in this two roots s1 and s2 are equal.

$$s_1 = s_2 = -\frac{c_c}{2m} = -\omega_n$$

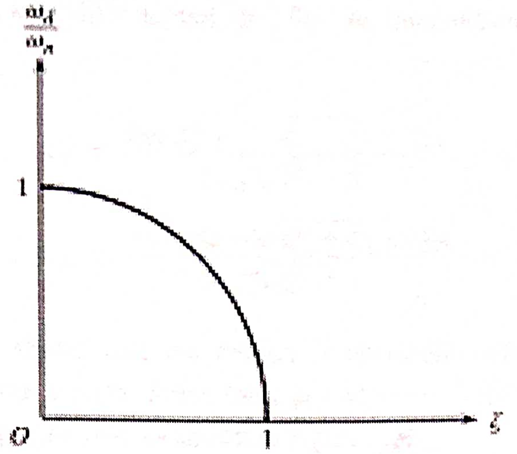


Figure 2.4 Variation of damped vibartion with damping

Because of repeated roots the Eq. is given by

$$x(t) = (C_1 + C_2 t)e^{-\omega_n t}$$

The application of the initial condition x(t = 0) = x_o and $\dot{x}(t = 0) = \dot{x}_0$ for this case gives

$$C_1 = x_0$$
$$C_2 = \dot{x}_0 + \omega_n x_0$$

And the solution becomes

$$x(t) = [x_0 + (\dot{x}_0 + \omega_n x_0)t]e^{-\omega_n t}$$

+-It can be seen that the motion represented by Eq. is aperiodic (i.e. non-periodic). Since $e^{-\omega_n t} \to 0$ as $t \to \infty$ the motion will eventually diminish to zero, as indicated in fig.

Case 3. Over-damped system (ξ > 1 or c > c_c or c/2m > √k/m. As $\sqrt{\xi^2} - 1 > 0$, equation shows the roots s1 and s2 are real and distinct and are given by

$$s_1 = (-\zeta + \sqrt{\zeta^2 - 1})\omega_n < 0$$
$$s_2 = (-\zeta - \sqrt{\zeta^2 - 1})\omega_n < 0$$

With s1 << s2. In this case, the solution, Equation can be expressed as

$$x(t) = C_1 e^{(-\zeta + \sqrt{\zeta^2 - 1})\omega_n t} + C_2 e^{(-\zeta - \sqrt{\zeta^2 - 1})\omega_n t}$$

For the initial condition x(t = 0) = x_0 and $\dot{x}(t = 0) = \dot{x}_0$, the constants C1 and C2 can be obtained:

$$C_1 = \frac{x_0\omega_n(\zeta + \sqrt{\zeta^2 - 1}) + \dot{x}_0}{2\omega_n\sqrt{\zeta^2 - 1}}$$

$$C_2 = \frac{-x_0\omega_n(\zeta - \sqrt{\zeta^2 - 1}) - \dot{x}_0}{2\omega_n\sqrt{\zeta^2 - 1}}$$

Above equation shows that the motion is aperiodic regardless of the initial conditions imposed on the system. Since roots s_1 and s_2 are both negative, the motion diminishes exponentially with time, as shown in Figure 2.5.

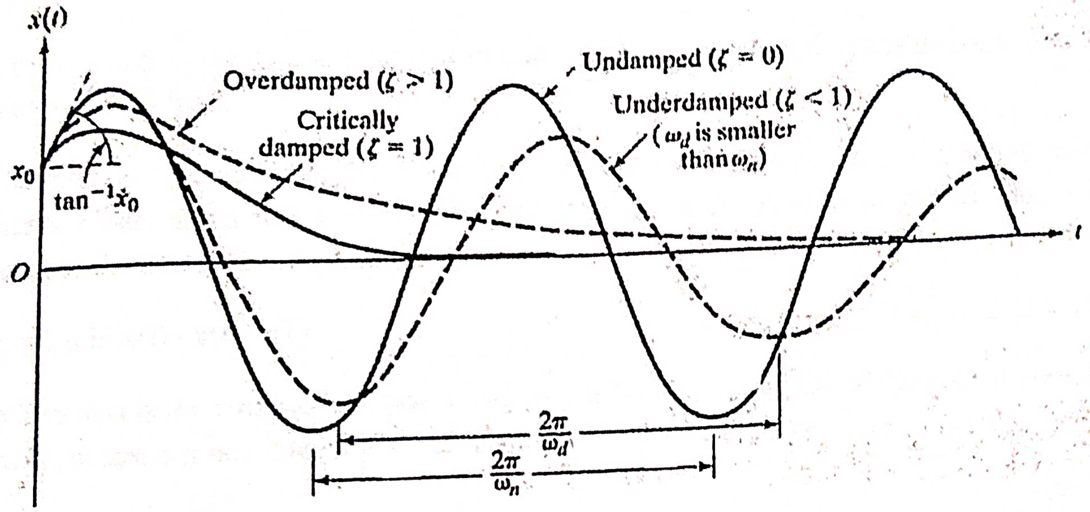


Figure 2.5 Comparison motion with different types of damping

Note the following aspects of these systems:

1. The graphical representation of different types of the characteristics roots and and the corresponding responses (solutions) of the system are presented. The representation of the roots and with varying values of the system parameters c, k and m in the complex plane (known as the root locus plots) is considered.

2. A critically damped system will have the smallest damping required for aperiodic motion; hence the mass returns to the position of rest in the shortest possible time without

overshooting. The property of critical damping is used in many practical applications. For example, large guns have dashpots with critical damping value, so that they return to their original position after recoil in the minimum time without vibrating. If the damping provided were more than the critical value, some delay would be caused before the next firing.

3. The free damped response of a single-degree-of-freedom system can be represented in phase-plane or state space.

2.5 Logarithmic decrement

The logarithmic decrement represents the rate at which the amplitude of a free damped vibration decreases. It is defined as the natural logarithm of the ratio of any two successive amplitudes. Let t_1 and t_2 denote the times corresponding to two consecutive amplitudes, measured one cycle apart for an under-damped system, it is shown in figure2.6 below.

It is used to find the damping ratio of an under-damped system in the time domain. And it is represented by δ.

The method of logarithmic decrement becomes less and less precise as the damping ratio increase past about 0.5; it does not apply at all for a damping ratio greater than 1.0 because the system is over-damped.

$\Delta = 1/n \ln x(t) / x(t + nT)$

Where x(t) is the amplitude at time t and x(t + nT) is the amplitude of the peak n periods away, where n is any integer number of successive, positive peaks.

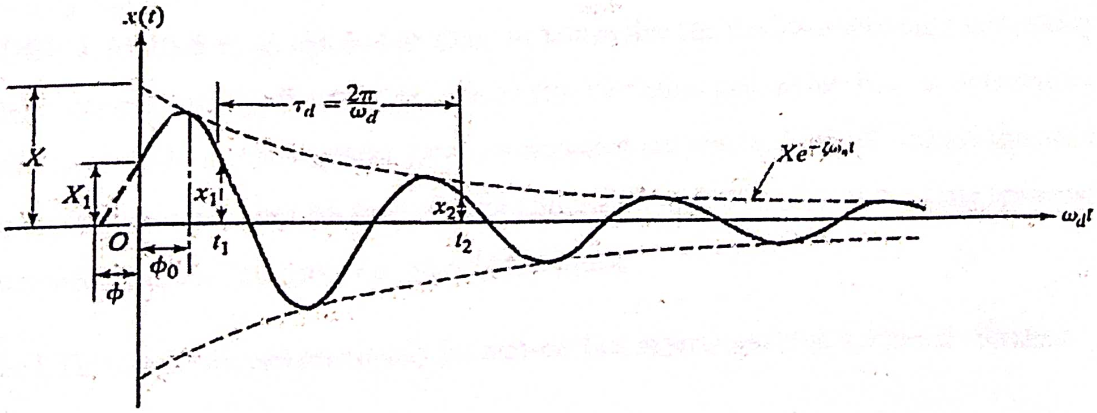


Figure 2.6 Logarithmic decrement

R.D.ADAM et al. (1) "The damping characteristics of certain steel, cast iron and other metals": In this study he determined damping capacity of a wide range of engineering metals at direct cyclic stress up to the fatigue limit. A recently developed damping apparatus was used in which the specimen, vibrated in its fundamental free longitudinal mode, driven by a magnetostrictive vibrator. The energy dissipation was determined from the rate of rise of temperature at different sections of the specimen.

The specific damping of 26% was recorded for sunstone, a commercial high damped manganese\copper alloy. The damping of cast irons is due to shape of the graphite inclusions rather than to the quantity of free graphite in the steel matrix.

Z.A.JAESIMETAL et al. (2) "A review on the vibrational analysis for a damage occurrence of a cantilever beam": This work shows axially moving beams has several applications, including robot arms, conveyor belts, shafts and automobile engine belts, understanding the vibrations of axially moving beams are important for the design of the devices.

Recent development in research on axially moving structures has been reviewed. Natural frequencies of non-linear coupled planar vibration are investigated for axially moves beam in the supercritical transport speed ranges.

TOBY J. MILES et al. (3): In this study he notice that the torsional vibration of rotating shafts contributes significantly to machinery vibration and noise but is notoriously difficult to study experimentally. New development are reported which address the need for appropriate measurement tools, through improved understanding of the laser torsional vibrometer and the application of model techniques.

The LTU was developed previously for non-contact measurement of torsional vibration.

This experiment describe the real-time measurement of torque input to a system allows torsional vibration response function to be obtained. The aim of this work is to utilize theory and signal processing technology used excessively for convectional translational model analyze.

GIACOMO BIANCHI, SELFANO CAGNA, NICOLA CAU et al. (4) "Analysis of vibration damping in machine tools" : Today FE models provide a satisfying description of structure distribution stiffness and inertia, machine damping is usually not represented or is approximated as a uniform viscous damping with no precise reference to the actual dissipation phenomenon occurring in the structure, in order to overcome this limitation, this work aim at adding key energy dissipation mechanism into numerical structural model.

The effect of guide ways friction an axis dynamics been evaluated computing on the FEM models without position control compliance at the work piece with varying values of viscous damping carriage some vibration modes aren't influenced at all by guide ways damping because then doesn't involves significant development, if shows for different modes of vibration different viscous damping values.

C.COOLETTE, R.BASTATIS, M.IIORODINCES et al. (5) "A vibration absorber for torsional vibration of metro wheel sets": This 3 men's shows the torsional vibration of the wheel set are first studied with free boundaries condition to dynamic vibration absorber tuned to the first torsional vibration vibration of the wheel set as been designed and it's numerical efficiency is compared with experiment result. The efficiency of the dynamic torsional vibration absorber is get evaluated experimentally and compared with output of the multibody model of the roller rig.

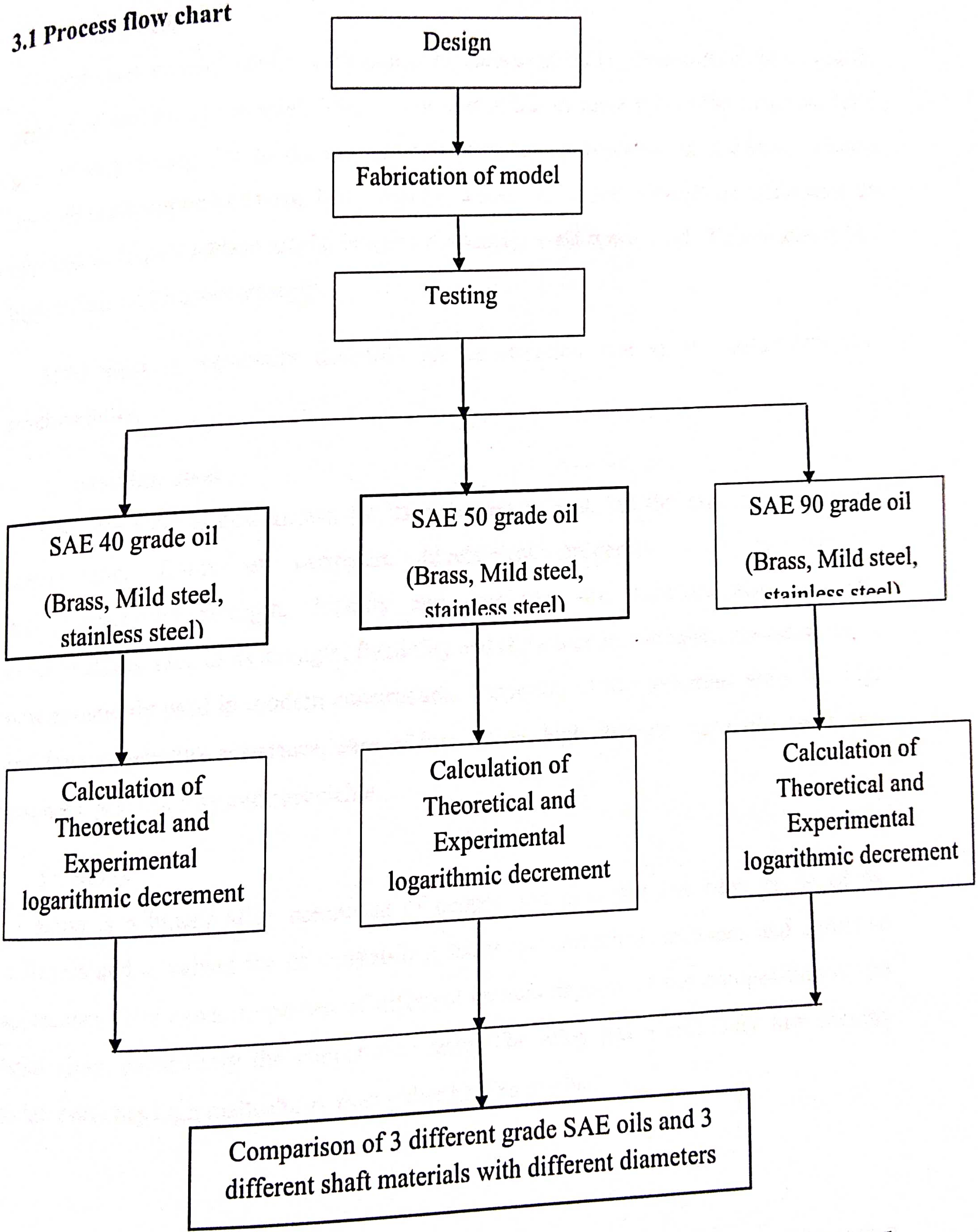
SHASHIKUMAR.C, B.AMARESHKUMAR et al.(6) "Design and development of a setup for torsional vibration system" This work shows the torsional vibration is an oscillatory angular motion causing twisting in the shaft of a system; even though the vibration cannot be detected without specific measuring instrument. Torsional vibratory motion can produce stress reverse that cause metals fatigue. Compound tolerates less reversed stress that steady stress.

By using this models can be calculating the longitudinal and torsional natural frequencies forced response analysis, to calculate the vibratory torque acting on the shaft, perform the forced response analysis to access torsional vibration to avoid torsional vibration and excessive, torsional we recommit ended changes such as but not limited to.

CHAPTER 3

METHODLOGY

3.1 Process flow chart



3.2 MATERIALS USED

1. Mild steel

Mild steel usually contains 40 points of carbon at most. One carbon point is 0.01 percent of carbon in the steel. This means that it has at most 0.4 percent carbon. Mild steel is very strong due to the low amount of carbon it contains. In materials science, strength is a complicated term. Mild steel has a high resistance to breakage. Mild steel, as opposed to higher carbon steels, is quite malleable, even when cold. This means it has high tensile and impact strength.

Mild steel is especially desirable for construction due to its weldability and machinability.

2. Stainless steel

Stainless steel is best known for its resistance to rust, but the material also resists many other forms of corrosion. Mechanical properties such as strength, high temperature strength, ductility and toughness are therefore also important considerations. Due to its strength, flexibility and resistance to corrosion, stainless steel is now commonly used in modern construction. Properties of this materials are it has high and low temperature resistance, ease of lubrication, high strength, long life cycle, low magnetic permeability and recyclable.

3. Brass

Brass is a binary alloy composed of copper and zinc that has been produced for millennia and is valued for its workability, hardness, corrosion resistance and attractive appearance. The exact properties of different brasses depend on the composition of the brass alloy, particularly the copper-zinc ratio. The alloy has a relatively low melting point; brass has high malleability than either bronze or zinc.

4. **SAE 40 Grade oil**

SAE 40 is synthetic oil derived from natural or crude oil. Here, SAE stands for Society of Automotive Engineers and 40 refer to viscosity of the oil. In layman's term we can say 40 is thickness of oil. More is the thickness, lesser is the contact between metals, which increase the life of engine.

5. **SAE 50 Grade oil**

SAE 50 has been specially developed for classic vehicles. It is extremely well suited for use with petrol and diesel engines with forced-feed. Lubrication and fine mesh filters. It is also ideal for use in engines with increased operating temperatures and also subject to high loads. The viscosity SAE 50 is designed for use in air-cooled engines and for motor bikes. It can also be used in the event of overheating problems or increased oil consumption. SAE 50 can be mixed with mineral-oil based motor oil of the same performance level. Oil and filter must be replaced in accordance with the vehicle manufacturer's specifications. Optimally formulated for classic vehicles, good wear and oxidation protection, extensive corrosion protection, good cleaning performance.

6. **SAE 90 grade oil**

SAE-90 is a lubricant used mainly as gear oil. The digit 90 indicates the viscosity of the lubricant. SAE stands for "society of automotive engineers" which was founded by American automobile industry. Viscosity means resistance of the liquid to flow. Purpose of the lubricant is to form a thin layer between the tooth surfaces of the gear box so that friction between these surfaces is reduced drastically and in turn, wear of gear teeth too gets reduced, thus enhancing the life of gears. During manufacturing of these oils, some materials are also added to it. These are called as EP additives.

CHAPTER 4

THEORATICAL ANALYSIS

4.1 For Mild steel

- Modulus of rigidity = 73Gpa
- Length of shaft = 1m
- **Diameter of shaft = 5mm**

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 5^4 \times 10^{-12}}{32}$$

$$= 6.13671 \times 10^{-11}\, m^4$$

Torsional stiffness

$$k \text{ or } q = \frac{GJ}{l}$$

$$k = \frac{73 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$$

$$= 4.4798\ Nm$$

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 5^4}{64}$$

$$= 1.2568 \times 10^{-11} m^4$$

Natural frequency

$$f_{n} = \frac{1}{2\pi}\sqrt{\frac{k}{i}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{4.4798}{3.068*10^{-11}}}$$

$$= 60805.145\text{Hz}$$

Time period

$$T = \frac{1}{f_n}$$

$$T = \frac{1}{60805.145}$$

$$= 1.6446 \times 10^{-05}\text{sec}$$

Diameter of shaft = 6mm

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 6^4 \times 10^{-12}}{32}$$

$$= 1.27251 \times 10^{-10}\, m^4$$

Torsional stiffness

$$k \text{ or } q = \frac{GJ}{l}$$

$$k = \frac{73 \times 10^9 1.27251 \times 10^{-10}}{1}$$

$$= 9.2893\ Nm$$

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 0.006^4}{64}$$

$$= 0.636255 \times 10^{-10} m^4$$

Natural frequency

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{I}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{9.2893}{0.636255\times10^{-10}}}$$

$= 60805.058\text{Hz}$

Time period

$$T = \frac{1}{f_n}$$

$$T = \frac{1}{60805.058}$$

$= 1.6446\times10^{-05}\text{sec}$

4.2 For Brass

- Modulus of rigidity = 40Gpa
- Length of shaft = 1m
- **Diameter of shaft = 5mm**

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 5^4 \times 10^{-12}}{32}$$

$= 6.13671 * 10^{-11}\, m^4$

Torsional stiffness

$$k \text{ or } q = \frac{GJ}{l}$$

$$k = \frac{40 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$$

$= 2.45468\text{Nm}$

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 0.005^4}{64}$$

$$=3.06835\times10^{-11}m^4$$

Natural frequency

$$f_{n} = \frac{1}{2\pi}\sqrt{\frac{k}{i}}$$

$$f_{n} = \frac{1}{2\pi}\sqrt{\frac{2.45468}{3.06835*10^{-11}}}$$

$$= 45043.3891\text{Hz}$$

Time period

$$T = \frac{1}{fn}$$

$$T = \frac{1}{45043.389}$$

$$=2.22008\times10^{-05}\text{sec}$$

- **Diameter of shaft = 6mm**

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 6^4 \times 10^{-12}}{32}$$

$$= 1.27251 \times 10^{-10}\, m^4$$

Torsional stiffness

$$k \text{ or } q = \frac{GJ}{l}$$

$$k = \frac{40 \times 10^9 \times 1.27251 \times 10^{-10}}{1}$$

= 5.09904 Nm

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 0.006^4}{64}$$

$$=0.636255\times10^{-10} m^4$$

Natural frequency

$$f_{n=\frac{1}{2\pi}\sqrt{\frac{k}{i}}}$$

$$f_{n=\frac{1}{2\pi}\sqrt{\frac{5.09004}{10.636255\times10^{-01}}}}$$

= 45010.17Hz

Time period

$$T=\frac{1}{fn}$$

$$T=\frac{1}{45010.17}$$

$$= 2.2217\times10^{-05}\text{sec}$$

4.3 For Stainless steel

- Modulus of rigidity = 77.2Gpa
- Length of shaft = 1m
- **Diameter of shaft = 5mm**

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 5^4 \times 10^{-12}}{32}$$

$$= 6.13671 * 10^{-11}\, m^4$$

Torsional stiffness

$$k \text{ or } q = \frac{GJ}{l}$$

$$k = \frac{77.2 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$$

$$= 4.7375$$

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 0.005^4}{64}$$

$$= 3.06835 \times 10^{-11} m^4$$

Natural frequency

$$f_{n = \frac{1}{2\pi}\sqrt{\frac{k}{i}}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{4.7375}{3.06835 \times 10^{-11}}}$$

$$= 62529.648Hz$$

Time period

$$T = \frac{1}{fn}$$

$$T = \frac{1}{62529.6483}$$

$$= 1.5994 \times 10^{-05} sec$$

$$= 1.2568 \times 10^{-11} m^4$$

- **Diameter of shaft = 6mm**

Polar moment of inertia

$$J = \frac{\pi d^4}{32}$$

$$= \frac{\pi\, 6^4 \times 10^{-12}}{32}$$

$$= 1.27251 \times 10^{-10}\, m^4$$

Torsional stiffness

$$k \; or \; q = \frac{GJ}{l}$$

$$k = \frac{77.2 \times 10^9 \times 1.27251 \times 10^{-10}}{1}$$

$$= 9.82377 \text{ Nm}$$

Moment of inertia

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi \times 0.006^4}{64}$$

$$=0.636255\times10^{-10} m^4$$

Natural frequency

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{i}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{9.82377}{10.636255\times10^{-01}}}$$

$$=62529.839 Hz$$

Time period

$$T = \frac{1}{f_n}$$

$$T = \frac{1}{62529.839\times10^{-10}}$$

$$=2.3931\times10^{-05} sec$$

4.4 Calculation of logarithmic decrement

$$x_5 = 0.3x_0$$

$$\frac{{x_0}^5}{x_5} = 3.333$$

$$\frac{x_0}{x_1} = 3.333^{1/5}$$

$$\frac{x_0}{x_1} = 1.272234$$

$$\frac{x_0}{x_1} = \frac{x_1}{x_2} = \frac{x_2}{x_3} = \frac{x_3}{x_4} = \frac{x_4}{x_5} = 1.272234$$

Logarithmic decrement

$$\delta = ln\frac{x_0}{x_1}$$

$$= \ln(1.272234)$$

$$= 0.240774$$

Damping factor

$$\delta = \frac{2\pi\varepsilon}{\sqrt{1-\varepsilon^2}}$$

$$0.240774 = \frac{2\pi\varepsilon}{\sqrt{1-\varepsilon^2}}$$

$$\varepsilon = 0.0382872$$

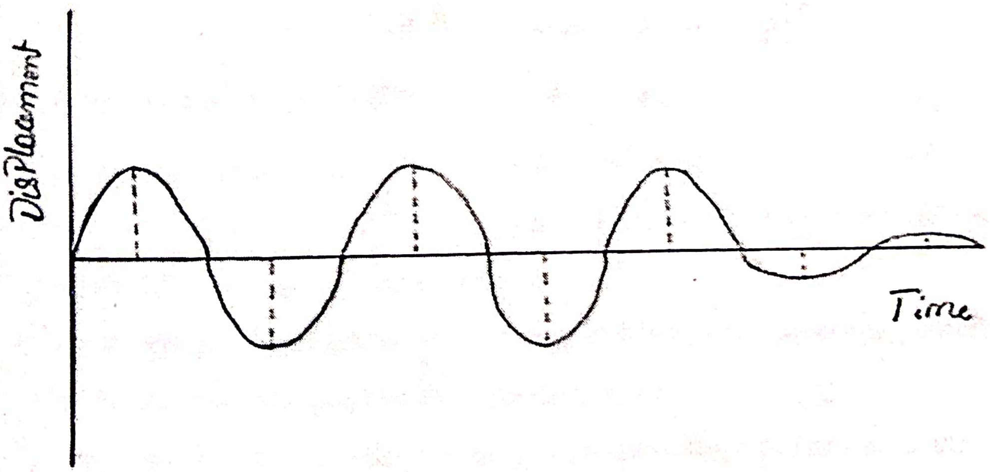


Figure 4.1 Logarithmic decrement

CHAPTER 5

EXPERIMENTAL ANALYSIS

5.1 Experimental setup

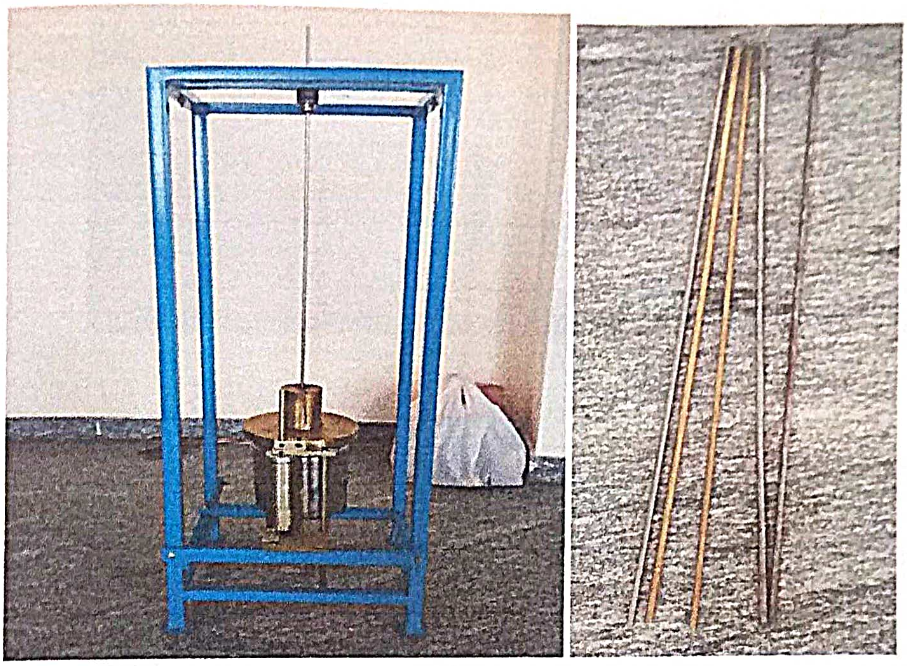


Figure 5.1 Damped torsional vibration tester and shaft materials

5.2 EXPERIMENTAL PROCEDURE

- Fit the shaft at bucket and attach the rotor damping drum.
- Warp a white paper around the rotor to note down the signature (variations) of the vibration produced by the torsional system.
- Fix the pen to the pen holder which is carried by hydraulic mover (oil), make sure that the tip of the pen should be touch the white paper.
- Then, rotate the rotor up to its twisting capacity, then release the rotor and at the same time release the pen holder also.
- As this rotating of the rotor and release it. The system vibrates take the signature of the vibration on the white paper.

Figure 5.2 Project testing

- As this experiment carried out note down the time taken for 'n' oscillation using a stopwatch and note down the value of x_1 and x_2 from the graph for calculation of logarithmic decrement.
- Repeat the experiment for all shaft materials (brass, mild steel and stainless steel) and different grade SAE oils (SAE 40, SAE 50 and SAE 90).

5.3 CALCULATION

5.3.1 For SAE 40 grade oil

For brass material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 6.12sec
- Value of x_1 =15mm
- Value of x_2= 12mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{6.12}$ = 1.63398Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{1.63398}$ = 0.6120025sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 2.45468Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367 \times 10^{-11} m^4$
4. Moment of inertia [I] = $\frac{T^2}{2\pi}$k = $\frac{0.6120025^2}{2\pi}$ 2.45468 = 0.02328m^4
5. Critical damping factor [C_c] = $2\sqrt{I \times k} = 2\sqrt{0.0232 \times 2.45468}$ = 0.4781Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{15}{12}$ = 0.22314
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.22314}{4\pi^2 + 0.22314^2}$ = 0.03548
8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 0.4781 × 0.03548 = 0.01696 Ns/m

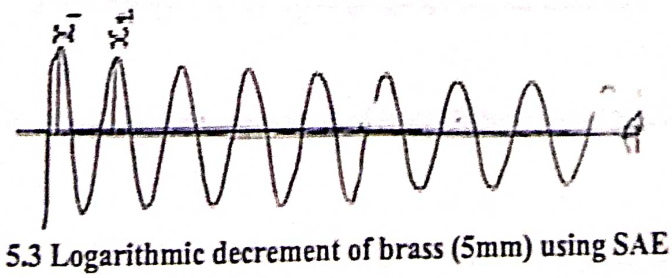


Figure 5.3 Logarithmic decrement of brass (5mm) using SAE 40 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 7.05sec
- Value of x_1 =13mm
- Value of x_2 = 11mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{7.05}$ = 1.4184Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{1.4184}$ = 0.7050sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 5.09004Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.7050}{2\pi}^2$ 5.090040.06406m^4
5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.06406 \times 5.09004}$ = 1.1421Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{13}{11}$ = 0.167054
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.167054}{4\pi^2 + 0.16705^2}$ = 0.02657
8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 1.1421 × 0.02657= 0.03032Ns/m

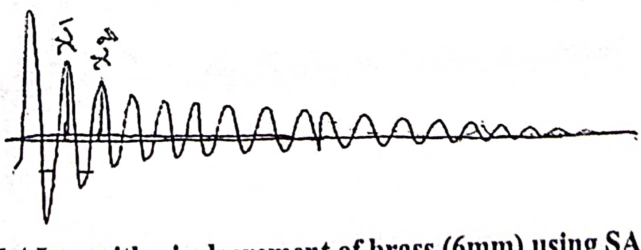


Figure 5.4 Logarithmic decrement of brass (6mm) using SAE 40 oil

For mild steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.21sec
- Value of x_1 = 8mm
- Value of x_2 = 5mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.21}$ =2.3752Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.3752}$ = 0.42101sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{73 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 4.4798Nm

   Polar moment of inertia [J]= $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367 \times 10^{-11} m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2 k = \frac{0.42101}{2\pi}^2 4.4798 = 0.0201 m^4$
5. Critical damping factor [C_c]=$2\sqrt{I \times k} = 2\sqrt{0.0201 \times 4.4798}$ = 0.6002Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{8}{5}$ = 0.47
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.47}{4\pi^2 + 0.47^2}$ = 0.07474
8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 0.6002 × 0.07474= 0.04486Ns/m

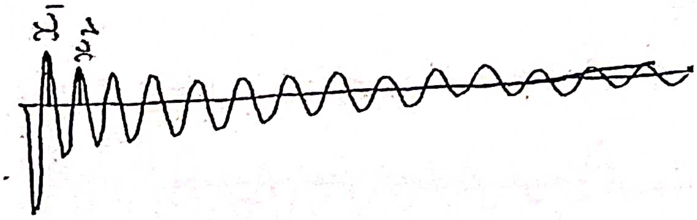


Figure 5.5 Logarithmic decrement of Mild steel (5mm) using SAE 40 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.115sec
- Value of x_1 =4mm
- Value of x_2 = 3mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.11}$ = 2.433Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.433}$ = 0.41101sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{73 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 9.2893Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2 k = \frac{0.41101}{2\pi}^2 9.2893 = 0.03973 m^4$
5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.03973 \times 9.2893}$ = 1.2151Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{4}{3}$ = 0.2876
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.2876}{4\pi^2 + 0.2876^2}$ = 0.04571
8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 1.2151 × 0.04571= 0.0555Ns/m

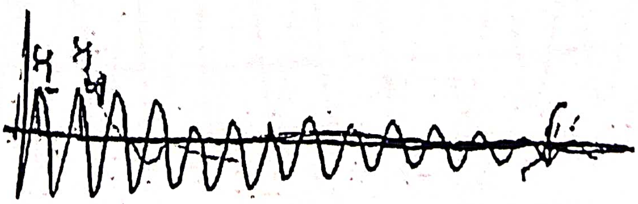


Figure 5.6 Logarithmic decrement of Mild steel (6mm) using SAE 40 oil

For stainless steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.31sec
- Value of x_1 = 8mm
- Value of x_2 = 5mm

1. Natural frequency $[f_n] = \frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.31}$ =2.3201Hz
2. Time period $[T] = \frac{1}{natural\ frequency} = \frac{1}{2.3201}$ = 0.43101sec
3. Torsional stiffness $[k] = \frac{GJ}{l} = \frac{77.2 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 4.7375Nm

   Polar moment of inertia $[J] = \frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367\times 10^{-11} m^4$
4. Moment of inertia $[I] = \frac{T}{2\pi}^2 k = \frac{0.43101}{2\pi}^2 4.7375 = 0.0222m^4$
5. Critical damping factor $[C_c] = 2\sqrt{I \times k} = 2\sqrt{0.0222 \times 4.7375}$ = 0.6498Ns/m
6. Logarithmic decrement $[\delta] = ln\frac{x_1}{x_2} = ln\frac{8}{5} = 0.471$
7. Damping ratio $[\varepsilon] = \frac{\delta}{4\pi^2 + \delta^2} = \frac{0.471}{4\pi^2 + 0.471^2} = 0.07474$
8. Damping co-efficient $[C] = c_c \times \varepsilon = 0.6498 \times 0.07474$ = 0.04856Ns/m

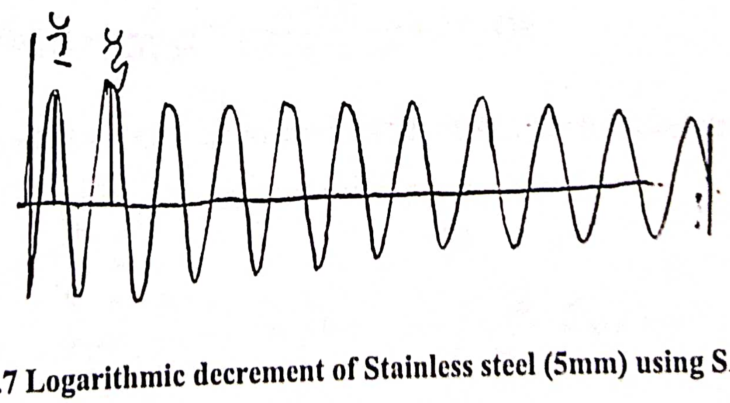


Figure 5.7 Logarithmic decrement of Stainless steel (5mm) using SAE 40 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.12sec
- Value of x_1 = 6mm
- Value of x_2 = 4mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.12}$ = 2.427Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.427}$ = 0.41203sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{77.2 \times 10^{9} \times 1.2725 \times 10^{-10}}{1}$ = 9.8237Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^{2} k = \frac{0.41203^2}{2\pi} 9.8237$ = 0.04223m^4

5. Critical damping factor [C_c] = $2\sqrt{I \times k} = 2\sqrt{0.04223 \times 9.8237}$ = 1.2882Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{6}{4}$ = 0.40546

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.40546}{4\pi^2 + 0.40546^2}$ = 0.06438

8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 1.2882 × 0.06438= 0.0829Ns/m

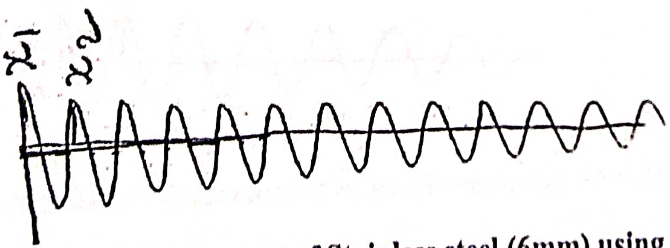


Figure 5.8 Logarithmic decrement of Stainless steel (6mm) using SAE 40 oil

5.3.2 For SAE 50 grade oil

For brass material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 6.12sec
- Value of x_1 =9mm
- Value of x_2 = 7mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{6.12}$ = 1.63398Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{1.63398}$ = 0.6120025sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9\ \times 6.1367 \times 10^{-11}}{1}$ = 2.45468Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32}$ = $6.1367 \times 10^{-11} m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.6120025^2}{2\pi}$ 2.45468 = 0.02328m^4
5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.0232 \times 2.45468}$ = 0.4781Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{9}{7}$ = 0.25131
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.25131}{4\pi^2 + 0.25131^2}$ = 0.03996
8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 0.4781 $\times$ 0.03996 = 0.01910Ns/m

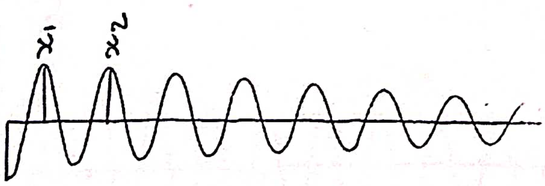


Figure 5.9 Logarithmic decrement of brass (5mm) using SAE 50 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 7.12sec
- Value of x_1 =7mm
- Value of x_2 = 6mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{7.12}$ = 1.4044Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{1.4044}$ = 0.71204sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 5.09004Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32}$ = $1.2725\times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.71204^2}{2\pi}$ 5.09004 = 0.06535m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.06535 \times 5.09004}$ = 1.1535Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{7}{6}$ = 0.154155

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.15415}{4\pi^2 + 0.15415^2}$ = 0.02452

8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 1.1535 × 0.02452= 0.02828Ns/m

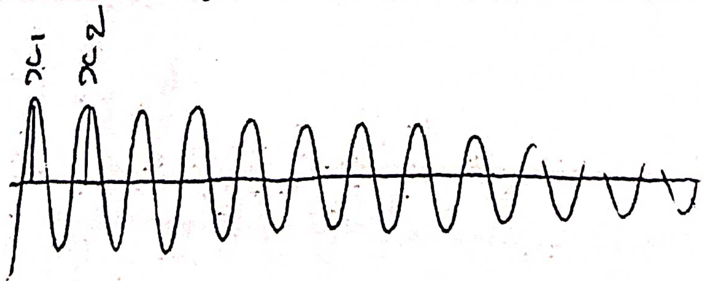


Figure 5.10 Logarithmic decrement of brass (6mm) using SAE 50 oil

For Mild steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.23sec
- Value of x_1 = 7mm
- Value of x_2 = 5mm

1. Natural frequency [f_n] $= \frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.23}$ =2.36406Hz
2. Time period [T] $= \frac{1}{natural\ frequency} = \frac{1}{2.36406}$ = 0.423sec
3. Torsional stiffness [k] $= \frac{GJ}{l} = \frac{73 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 4.4798Nm

   Polar moment of inertia [J] $= \frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367 \times 10^{-11} m^4$
4. Moment of inertia [I] $= \frac{T}{2\pi}^2 k = \frac{0.423}{2\pi}^2 4.4798 = 0.0202m^4$
5. Critical damping factor [C_c]$=2\sqrt{I \times k}=2\sqrt{0.0202 \times 4.4798} = 0.6031$Ns/m
6. Logarithmic decrement [δ] $= ln\frac{x_1}{x_2} = ln\frac{7}{5}$ = 0.0.33647
7. Damping ratio [ε] $= \frac{\delta}{4\pi^2 + \delta^2} = \frac{0.33647}{4\pi^2 + 0.33647^2}$ = 0.05346
8. Damping co-efficient [C] $= c_c \times \varepsilon = 0.6031 \times 0.05346 = 0.032246$Ns/m

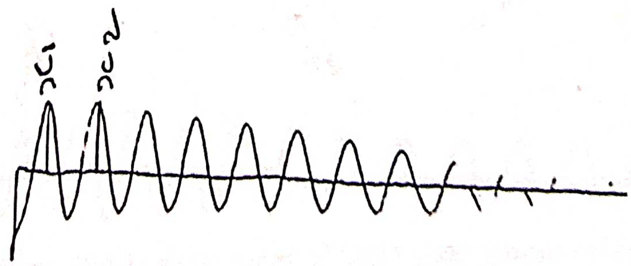


Figure 5.11 Logarithmic decrement of Mild steel (5mm) using SAE 50 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.15sec
- Value of x_1 =5mm
- Value of x_2 = 4mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.15}$ = 2.4096Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.4096}$ = 0.415sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{73 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 9.2893Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2 k = \frac{0.415}{2\pi}^2 9.2893$ = 0.04051m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k} = 2\sqrt{0.04051 \times 9.2893}$ = 1.22688Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{5}{4}$ = 0.22314

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.22314}{4\pi^2 + 0.22314^2}$ = 0.03548

8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 1.22688 × 0.03548= 0.04352Ns/m

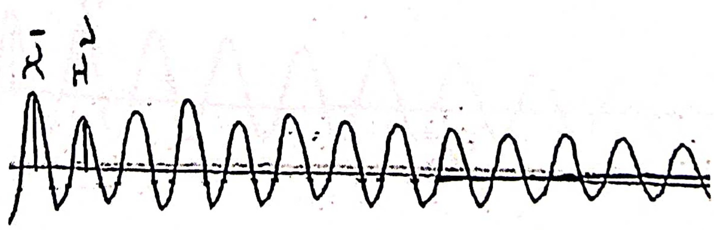


Figure 5.12 Logarithmic decrement of Mild steel (6mm) using SAE 50 oil

For stainless steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.71sec
- Value of x_1 = 8mm
- Value of x_2 = 6mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.71}$ =2.1231Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.1231}$ = 0.471sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{77.2 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 4.7375Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367 \times 10^{-11} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2 k = \frac{0.4711}{2\pi}^2 4.7375 = 0.02661m^4$

5. Critical damping factor [C_c] = $2\sqrt{I \times k} = 2\sqrt{0.02661 \times 4.7375}$ = 0.7101Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{8}{6}$ = 0.28868

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.28868}{4\pi^2 + 0.28868^2}$ = 0.04587

8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 0.7101 × 0.04587 = 0.03257Ns/m

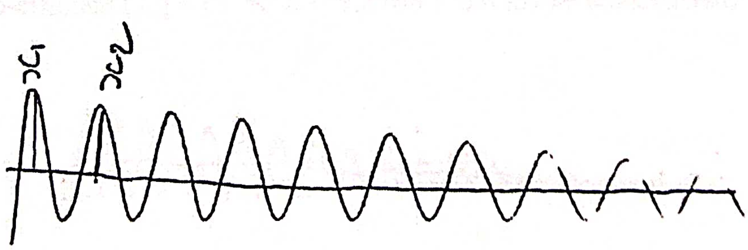


Figure 5.13 Logarithmic decrement of Stainless steel (5mm) using SAE 50 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.41sec
- Value of x_1 =4mm
- Value of x_2 = 3mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.41}$ = 2.2675Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.2675}$ = 0.44101sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{77.2 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 9.8237Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.44101}{2\pi}^2$ 9.28237 = 0.04838m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.04838 \times 9.8237}$ = 1.37885Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{4}{3}$ = 0.28768

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.28768}{4\pi^2 + 0.28768^2}$ = 0.04571

8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 1.3788 × 0.04571= 0.06302Ns/m

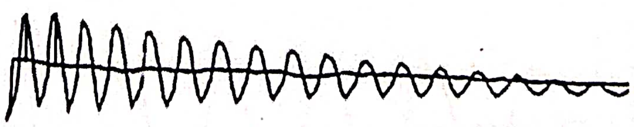


Figure 5.14 Logarithmic decrement of Stainless steel (6mm) using SAE 50 oil

5.3.3 For SAE 90 grade oil

For brass material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 6.19sec
- Value of x_1 =16mm
- Value of x_2 = 14mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{6.19}$ = 1.6155Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{1.6155}$ = 0.619sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 2.45468Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367\times 10^{-11} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2 k = \frac{0.619^2}{2\pi}$ 2.45468 = 0.02381m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.02381 \times 2.45468}$ = 0.48359Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{16}{14}$ = 0.13353

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.13353}{4\pi^2 + 0.13353^2}$ = 0.021207

8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 0.48359 × 0.021207 = 0.01281Ns/m

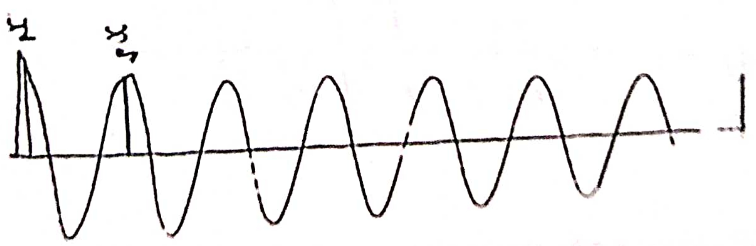


Figure 5.15 Logarithmic decrement of brass (5mm) using SAE 90 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.60sec
- Value of x_1 =13mm
- Value of x_2 = 12mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.60}$ = 2.1739Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.1739}$ = 0.46sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{40 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 5.09004Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.46^2}{2\pi}$ 5.09004 = 0.0272m^4

5. Critical damping factor [C_c] = $2\sqrt{I \times k} = 2\sqrt{0.0272 \times 5.09004}$ = 0.74518Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{13}{12}$ = 0.08004

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.08004}{4\pi^2 + 0.08004^2}$ = 0.01273

8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 0.74518 × 0.01273= 0.009486Ns/m

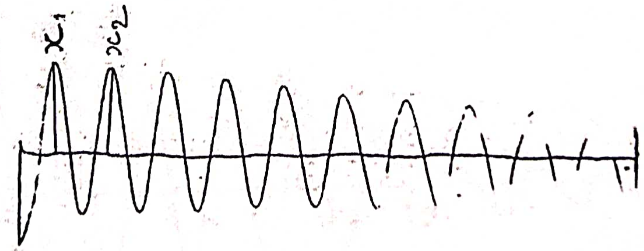


Figure 5.16 Logarithmic decrement of brass (6mm) using SAE 90 oil

For mild steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.42sec
- Value of x_1 = 11mm
- Value of x_2 = 8mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.42}$ =2.2624Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.2624}$ = 0.442sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{73\times10^9 \times 6.1367\times10^{-11}}{1}$ = 4.4798Nm

   Polar moment of inertia [J] = $\frac{\pi\times d^4}{32} = \frac{\pi\times0.005^4}{32} = 6.1367\times10^{-11}m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.442}{2\pi}^2$ 4.4798 = 0.02216m^4
5. Critical damping factor [C_c] =$2\sqrt{I\times k}$=$2\sqrt{0.02216\times4.4798}$ = 0.63015Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{11}{8}$ = 0.31845
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2+\delta^2} = \frac{0.31845}{4\pi^2+0.31845^2}$ = 0.05061
8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 0.63015 × 0.05061 = 0.03189Ns/m

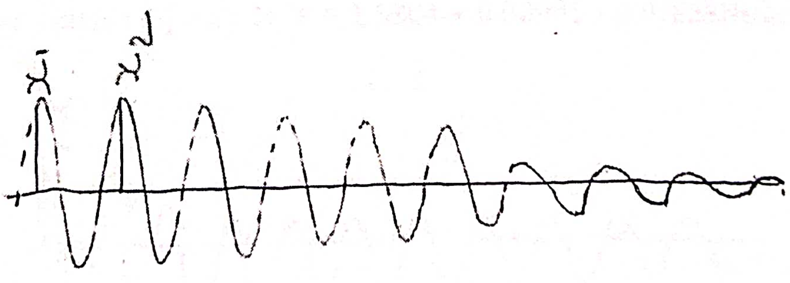


Figure 5.17 Logarithmic decrement of Mild steel (5mm) using SAE 90 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.5sec
- Value of x_1 =6mm
- Value of x_2 = 5mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.5}$ = 2.222Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.222}$ = 0.45004sec

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{73 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 9.2893Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.45004^2}{2\pi}$ 9.2893 = 0.04764m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k} = 2\sqrt{0.04764 \times 9.2893}$ = 1.3304Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{6}{5}$ = 0.18232

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.18232}{4\pi^2 + 0.18232^2}$ = 0.02901

8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 1.3304 × 0.02901 = 0.03858Ns/m

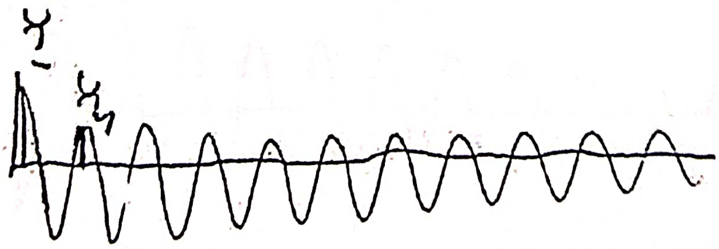


Figure 5.18 Logarithmic decrement of Mild steel (6mm) using SAE 90 oil

For stainless steel material

- Diameter of the shaft = 5mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 4.19sec
- Value of x_1 = 10mm
- Value of x_2 = 7mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{4.19}$ =2.3866Hz
2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{2.3866}$ = 0.419sec
3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{77.2 \times 10^9 \times 6.1367 \times 10^{-11}}{1}$ = 4.7375Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.005^4}{32} = 6.1367 \times 10^{-11} m^4$
4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.419^2}{2\pi}$ 4.7375 = 0.02106m^4
5. Critical damping factor [C_c] =$2\sqrt{I \times k} = 2\sqrt{0.02106 \times 4.7375}$ = 0.6317Ns/m
6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{10}{7}$ = 0.35667
7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.35667}{4\pi^2 + 0.35667^2}$ = 0.05666
8. Damping co-efficient [C] = $C_c \times \varepsilon$ = 0.6317 × 0.05666 = 0.03579Ns/m

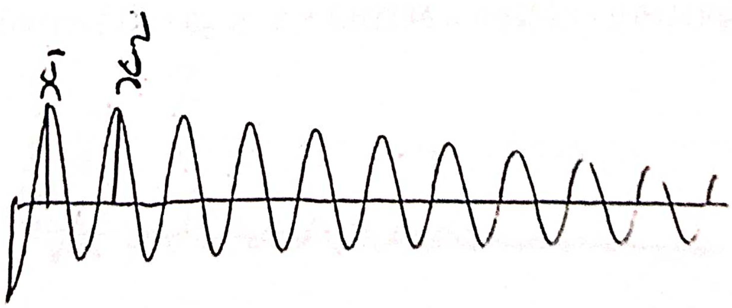


Figure 5.19 Logarithmic decrement of Stainless steel (5mm) using SAE 90 oil

- Diameter of the shaft = 6mm
- Length of the shaft = 1mm
- Number of oscillation = 10
- Time taken for 10 oscillation = 2.95sec
- Value of x_1 =4mm
- Value of x_2 = 3mm

1. Natural frequency [f_n] = $\frac{number\ of\ oscillation}{time\ taken} = \frac{10}{2.95}$ = 3.3898Hz

2. Time period [T] = $\frac{1}{natural\ frequency} = \frac{1}{3.3898}$ = 0.295sec

s

3. Torsional stiffness [k] = $\frac{GJ}{l} = \frac{77.2 \times 10^9 \times 1.2725 \times 10^{-10}}{1}$ = 9.8237Nm

   Polar moment of inertia [J] = $\frac{\pi \times d^4}{32} = \frac{\pi \times 0.006^4}{32} = 1.2725 \times 10^{-10} m^4$

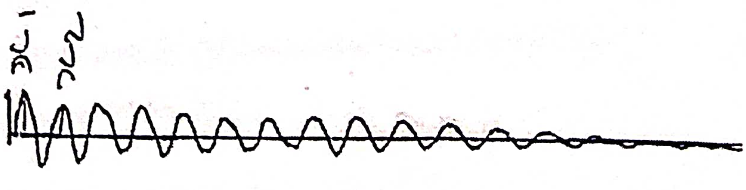
4. Moment of inertia [I] = $\frac{T}{2\pi}^2$ k = $\frac{0.295}{2\pi}^2$ 9.28237 = 0.02165m^4

5. Critical damping factor [C_c] =$2\sqrt{I \times k}$=$2\sqrt{0.02165 \times 9.8237}$ = 0.92234Ns/m

6. Logarithmic decrement [δ] = $ln\frac{x_1}{x_2} = ln\frac{4}{3}$ = 0.28768

7. Damping ratio [ε] = $\frac{\delta}{4\pi^2 + \delta^2} = \frac{0.28768}{4\pi^2 + 0.28768^2}$ = 0.04573

8. Damping co-efficient [C] = $c_c \times \varepsilon$ = 0.92234 × 0.04573 = 0.0421Ns/m



5.20 Logarithmic decrement of Stainless steel (6mm) SAE 90 oil

CHAPTER 6

RESULTS AND DISSCUSSION

6.1 For SAE 40 grade oil

Firstly, we use SAE 40 grade oil and three different shaft materials (brass, mild steel and stainless steel) with two different diameter (5mm and 6mm), SAE 40 has low viscosity, the brass has low stiffness hence in the logarithmic decrement graph x_1 and x_2 value is high compare to other two material, mild steel has medium stiffness compare to brass and SS so, logarithmic decrement is lies in between them and x_1 and x_2 values of stainless steel is very low because it has relatively high stiffness value than other two materials. As we observe upon increase in diameter of the shaft, logarithmic decrement value decreases and these logarithmic decrement graphs are shown in experimental analysis chapter 4.

Material	Diameter in mm	Logarithmic decrement
Brass	5	0.22314
	6	0.16705
Mild steel	5	0.47
	6	0.28765
Stainless steel	5	0.471
	6	0.40546

Table 6.1 Comparison of different shaft materials using SAE 40 0il

6.2 For SAE 50 grade oil

In this second trail we use SAE 40 grade oil and with same three different shaft materials (brass, mild steel and stainless steel) with two different diameter (5mm and 6mm), SAE 50 has high viscosity value compare to SAE 40 oil. The brass has low stiffness hence in the logarithmic decrement graph x_1 and x_2 value is high compare to other two material, mild steel has medium stiffness compare to brass and SS so, logarithmic decrement is lies in between them and x_1 and x_2 values of stainless steel is very low because it has relatively high stiffness value than other two materials. As we observe upon increase in diameter of the shaft, logarithmic decrement value decreases and these logarithmic decrement graphs are shown in experimental analysis chapter 4.

Material	Diameter in mm	Logarithmic decrement
Brass	5	0.25131
	6	0.15415
Mild steel	5	0.33647
	6	0.22314
Stainless steel	5	0.2886
	6	0.2878

Table 6.2 Comparison of different shaft materials using SAE 50 0il

6.3 For SAE 90 grade oil

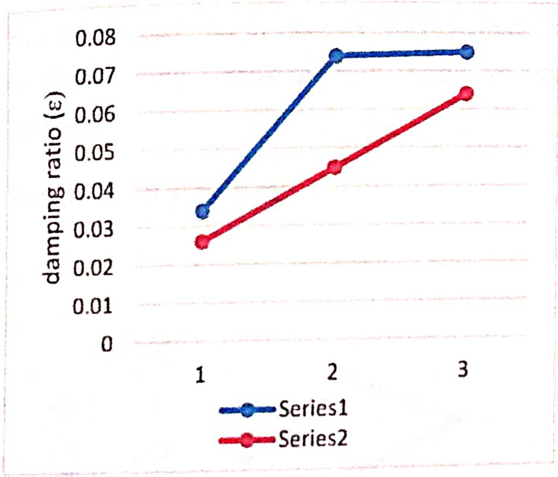
After second trail, we use SAE 90 grade oil and with same three different shaft materials (brass, mild steel and stainless steel) with two different diameter (5mm and 6mm), SAE 90 has high viscosity value relatively to both SAE 40 and SAE 50 grade oils. The brass has low stiffness hence in the logarithmic decrement graph x_1 and x_2 value is high compare to other two material, mild steel has medium stiffness compare to brass and SS so, logarithmic decrement is lies in between them and x_1 and x_2 values of stainless steel is very low because it has relatively high stiffness value than other two materials. As we observe upon increase in diameter of the shaft, logarithmic decrement value decreases and these logarithmic decrement graphs are shown in experimental analysis chapter 5.

Material	Diameter In mm	Logarithmic decrement
Brass	5	0.13353
	6	0.08004
Mild steel	5	0.31845
	6	0.18232
Stainless steel	5	0.35667
	6	0.22314

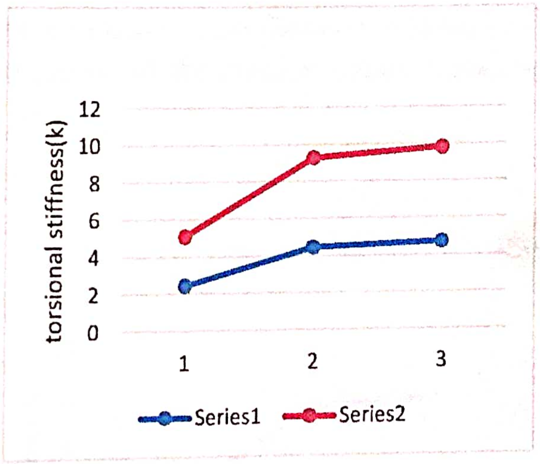
Table 6.3 Comparison of different shaft materials using SAE 90 0il

6.4 Comparison

For SAE 40 grade oil



(a)



(b)

Figure 6.1 Comparison of (a) Damping ratio and (b) Torsional stiffness for 3 materials of 5mm (series 1) and 6mm (series 2) using SAE 40 oil

The above Figure 6.1 (a) shows the variation of damping ratio (ε) of three materials i.e. brass, mild steel and stainless steel and with two different diameters i.e. 5mm and 6mm. The brass has low damping ratio than the other two materials, mild steel has damping ratio in between them and stainless steel has higher damping ratio in both 5mm and 6mm diameter in the SAE 40 grade oil. Figure 6.1 (b) shows the variation of torsional stiffness, here stainless steel has high torsional stiffness, mild steel has relatively slight low compare to stainless steel and brass has lower stiffness.

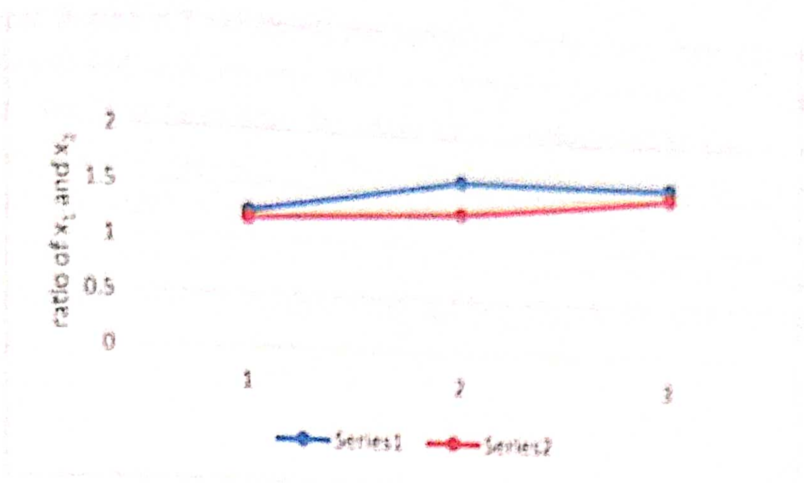
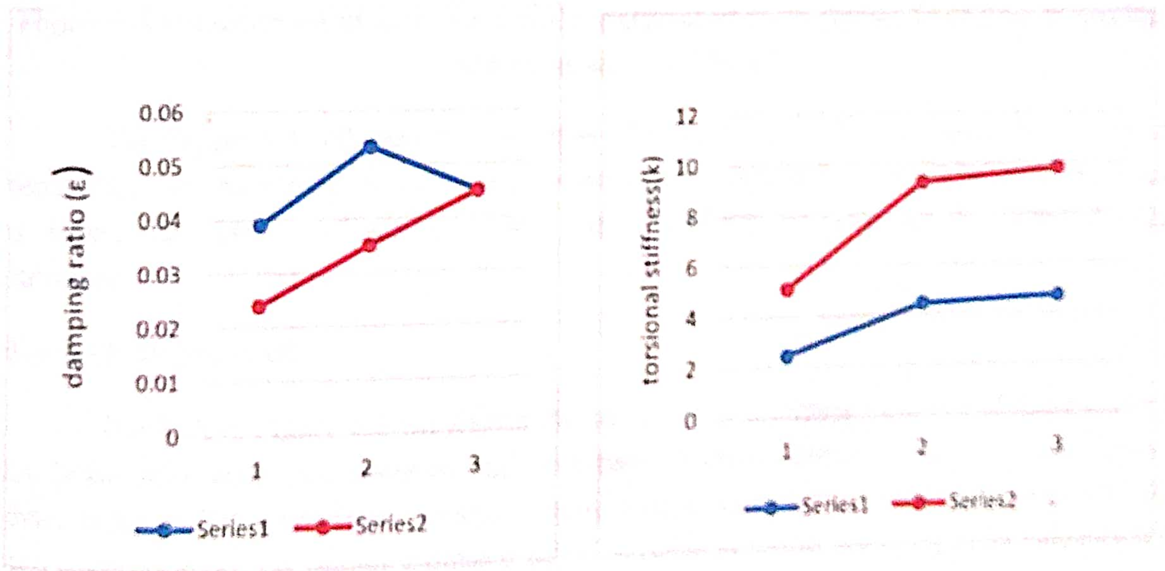


Figure 6.2 Comparison of x_1/x_2 for 3 shaft materials of 5mm (series 1) and 6mm (series 2) diameter using SAE 40 oil

The Figure 6.2 indicates the variation of x_1/x_2 value for 3 shaft materials of 5mm and 6mm diameter. As shown in the Figure 6.2 we observe that in 5mm diameter mild steel value is higher than stainless steel value, in 6mm diameter all the material values enormously increases.

For SAE 50 grade oil



(a) (b)

Figure 6.3 Comparison of (a) Damping ratio and (b) Torsional stiffness for 3 shaft materials of 5mm (series 1) and 6mm (series 2) using SAE 50 oil

The above Figure 6.3 (a) shows the variation of damping ratio (ε) of three materials i.e. brass, mild steel and stainless and with two different diameters i.e. 5mm and 6mm. The brass has lower damping ratio than the other two materials, mild steel has damping ratio higher than brass and stainless steel in 5mm diameter and stainless steel has high damping ratio in 6mm diameter but lower in the SAE 50 grade oil. Figure 6.3 (b) shows the variation of torsional stiffness, here also stainless steel has high torsional stiffness, mild steel has relatively slight low compare to stainless steel and brass has lower torsional stiffness.

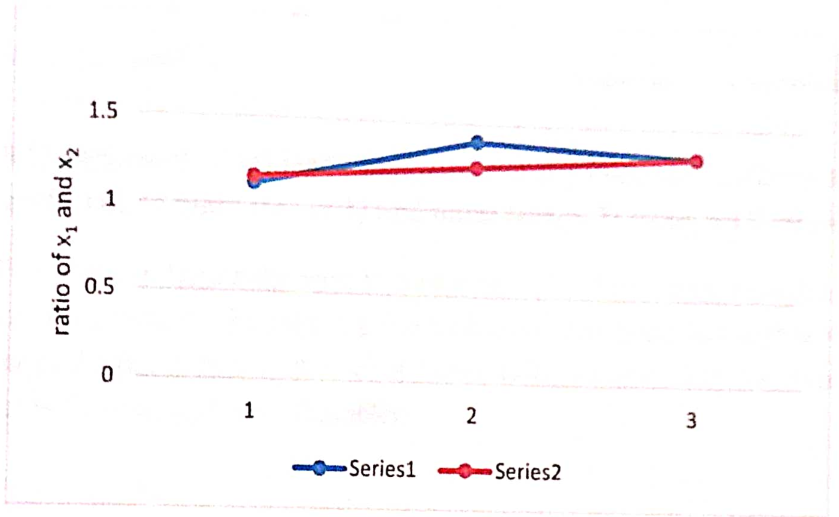
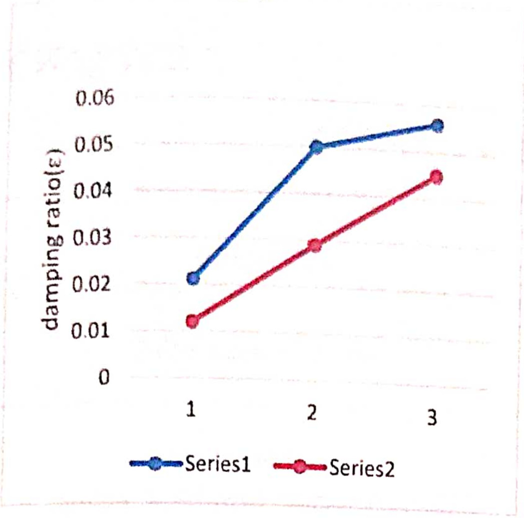


Figure 6.4 comparison of x_1/x_2 for 3 shaft material of 5mm (series 1) and 6mm (series 2) diameter use SAE 50 oil

The Figure 6.4 indicates the variation of x_1/x_2 value for 3 shaft materials of 5mm and 6mm diameter. As shown in the Figure 6.4 we observe that in 5mm diameter mild steel value is higher than brass and stainless steel value, in 6mm diameter all the material values increases.

For SAE 90 grade oil

The below Figure 6.5 (a) shows the variation of damping ratio (ε) of three materials i.e. brass, mild steel and stainless and with two different diameters i.e. 5mm and 6mm. In 5mm diameter brass has lower damping ratio, mild steel in between stainless steel and brass and stainless steel has higher damping ratio. In 6mm diameter damping ratio value of three shaft materials increase linearly. Figure 6.5 (b) shows the variation of torsional stiffness, here stainless steel has high torsional stiffness, mild steel has relatively slight low compare to stainless steel and brass has lower torsional stiffness.



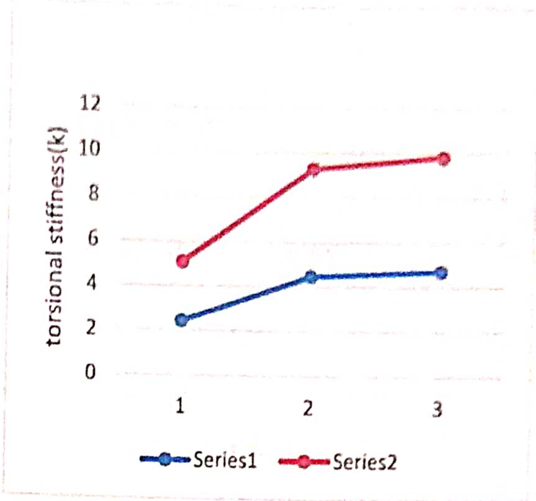


Figure 6.5 Comparison of (a) Damping ratio and (b) Torsional stiffness for 3 shaft materials of 5mm (series 1) and 6mm (series 2) using SAE 90 oil

The Figure 6.6 indicates the variation of x_1/x_2 value for 3 shaft materials of 5mm and 6mm diameter. As shown in the Figure 6.6 we observe that brass has higher value in 5mm compare to 6mm diameter, but mild steel is lower value in 5mm and stainless has alomost same value for both 5mm and 6mm diameter.

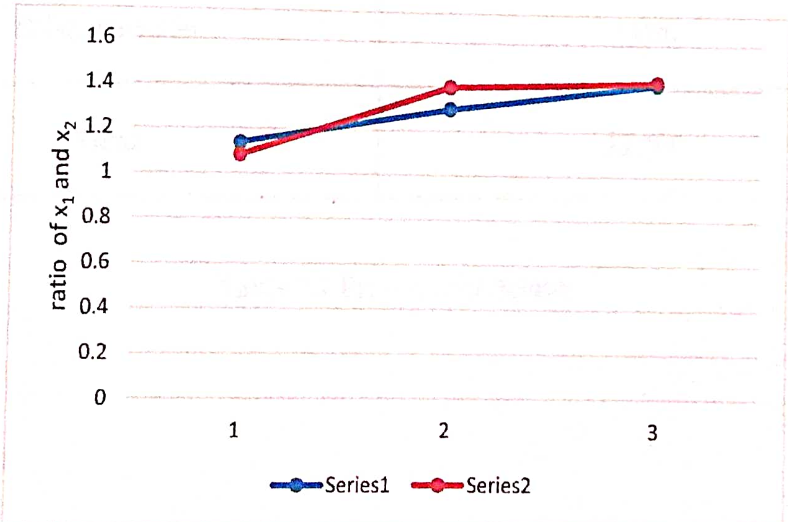


Figure 6.6 comparison of x_1/x_2 for 3 shaft material of 5mm (series 1) and 6mm (series 2) diameter use SAE 50 oil

CHAPTER 7

PROJECT COST ESTIMATION

Activities	Cost (in rupees)
Fabrication of model	20,000
Shaft materials (brass, Mild steel, Stainless steel)	1,000
SAE grade oils (40, 50, 90)	1,500
Transportation charge	2,000
Other activities	1,000
Total	**25,500**

Table 7.1 Project cost details.

CHAPTER 8

SOCPES FOR FUTURE WORK

- To create and design the vibration tester using modern software and analysis the thing.
- To design parts of the test rig that can use various shaft materials like copper, steel, aluminium etc.
- To increase the efficiency of the tester in better way.
- To know the better viscous damper and that can be used in future testing in better way to achieve the good results.

CHAPTER 9

CONCLUSIONS

In this project work, we are fabricate the damped torsional vibration tester machine and conducted test using different materials like brass, mild steel and steel with using different diameters 5mm and 6mm and using different grade of oil like SAE 40, SAE 50 and SAE 90.

Here we conducting the experiments to know the better characteristics of the damping of the different oils with using different shaft materials, using this data we calculated natural frequency (f_n), time period (T), theoretical logarithmic decrement (δ_T), experimental logarithmic decrement (δ_e), critical damping factor (C_c), damping ratio (ε), torsional stiffness (k), damping co-efficient (C).

Here we finally observe that, SAE 90 having good damping characteristics, compare to other 2 grades of oils (i.e. SAE 40 and SAE 50). Here time period increases and number of oscillation increases. Finally logarithmic decrement decreases and we are conducting the experiment on 6mm and 5mm diameter shafts.

Here we finally conclude that logarithmic decrement value affected by diameter, type of shaft materials and viscosity of the oil. As increase in diameter of the shaft, torsional stiffness value of the shaft also increases then logarithmic decrement value decreases and damping value is increases. Damping ratio and damping factor is linearly affected by the logarithmic decrement value.

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