MECHANICS OF MACHINES II LABORATORY EXERCISES

I. GEAR TRAIN EFFICIENCY TEST

OBJECTIVE: To evaluate the power loss in a gear train for different values of power transmitted and the resulting efficiencies.

INTRODUCTION: Gears are compact, positive-engagement, power transmission elements that determine the speed, torque, and direction of rotation of driven machine elements. Gear types may be grouped into five main categories: Spur, Helical, Bevel, Hypoid, and Worm. Typically, shaft orientation, efficiency, and speed determine which of these types should be used for a particular application.

Gear contact is normally simultaneous across the entire width of the meshing teeth, resulting in a continuous series of shocks. These rapid shocks result in some objectionable operating noise and vibration. Moreover, tooth wear results from shock loads at high speeds. Noise and wear can be minimized with proper lubrication, which reduces tooth surface contact and engagement shock loads. The main advantage of the gear is the property of self-holding, i.e. providing the immobility of the screw with its loading only by the axial force $F(T) = 0$. The efficiency of a gear is equal to the relation between the useful (output) power and the applied (input) power.

Gear trains are used for transmitting power from a driving unit to a driven unit, with a change of speed. The output from the gear box can have a higher or a lower speed depending on the requirement. Power losses in the gear box which encloses the gear train results from viscous friction of lubricants, sliding friction, losses of energy due to vibration and noise etc. Therefore power supplied to the gear train is more than power delivered to the power absorber. This experiment demonstrates a method of determining these losses.

LITERATURE REVIEW
Evaluate how the various gear types are combined into gear drives; and consider the principle factors that affect gear drive selection.

THEORY
The efficiency η of a mechanism is determined by:

$$\eta = \frac{P_{out}}{P_{inp}},$$

Where P_{inp} is the power input in the mechanism

and P_{out} is the output power.

This main dependency in engineering is used to evaluate the theoretical and the real (experimentally determined) values of efficiency of a gear.

To measure power losses in a gearbox, the straight forward approach is to measure the power supplied to the motor and the powers absorbed by the power absorber so that the loss can be found as the difference between the two.
The value of losses is very much less than either of the two values measurable and the method described above would be very inaccurate if employed. This difficulty is overcome by feeding the power output from the gearbox back into the input. **Fig. 1** below illustrates the principle.

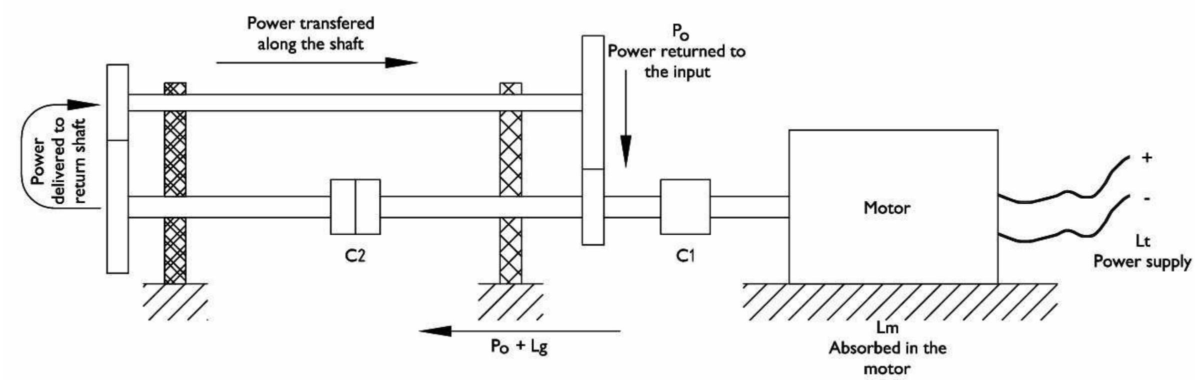


Figure 5: The gear train principle

Let power output from gear be P_O, then $P_O = T_O$
Where;

T = torque built into the train and W= angular velocity of the motor shaft

Losses = sum of losses in the motor and in the gear train

i.e. L_T (total losses) $= L_M$ (motor losses) $+ L_g$ (gear losses)

Then the equivalent of power supplied $L_T + P_O = P_i$

$$P_i = L_M + L_g + P_O$$

$$Effiiciency = \frac{Power\,input - Power\,output}{Power\,input}$$

for the gear train only

Note that P_i is a calculated value. It is not read off the wattmeter. Once the system attains a steady speed, the only power supplied from the mains is the power required to overcome losses.

$$\eta = \frac{P_O}{P_O + L_g} \quad For\,the\,gearbox\,only$$

$$L_g \;\; = L_T \quad from\,wattmeter\,reading$$
$$L_M \quad at\,the\,same\,speed$$

APPARATUS

Fig. 2 shows the layout of the apparatus. A wattmeter is used for giving a reading of the power supplied to overcome losses. A variance enables a supply of variable voltage to the DC motor hence a variable speed can be achieved. The torque is incorporated into the system using coupling **(Fig 1).** The speed of the motor shaft is measured by a tachometer pressed lightly at the end of the shaft.

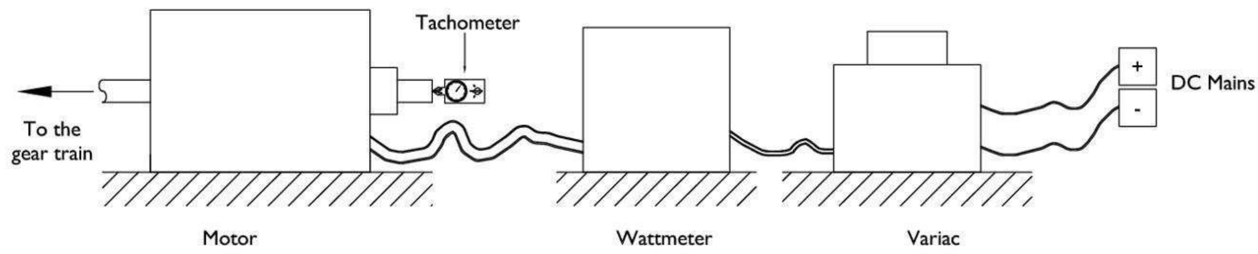


Figure 6: Experiment set up

PROCEDURE

1. Determination of losses in the motor alone.

The motor is disconnected from the gear tat coupling and set to run at a chosen speed. The power supplied to the motor to overcome losses in the winding and bearing friction is read off the wattmeter. Run the motor steadily at 800 rev/min and record the wattmeter reading as. Repeat for higher speeds increasing the speed in steps of 100 rev/min. i.e. at 900, 1000, to 1800 rev/min.

2. Determination of losses in both the motor and the gear train

Connect the motor to the gear train at and build a torque into the gear train at coupling, by holding one half of with a spanner and applying a moment on a steel rod fixed in the coupling, and tightening the two halves of the coupling together. **Fig. 3** shows how to determine the torque.

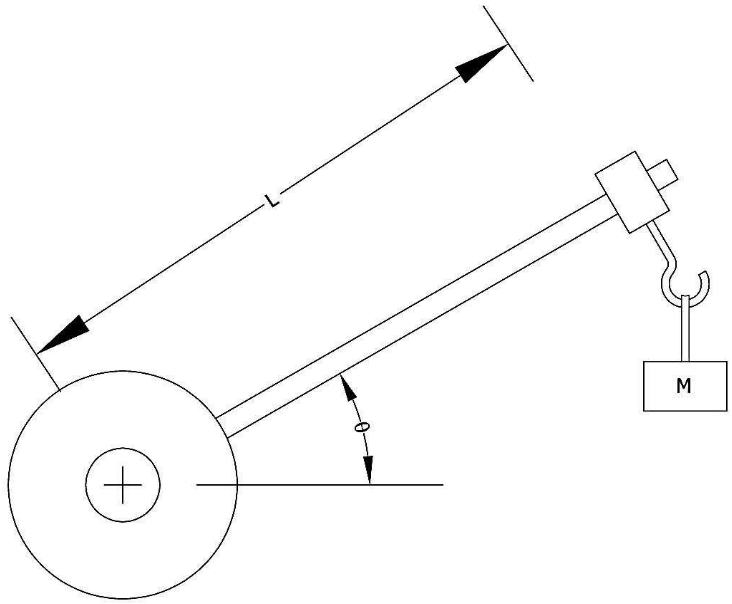


Figure 7: Determination of the torque

Measure θ using a vernier protractor.
After building the torque, run the system at the same speeds as above and record the corresponding wattmeter readings representing L_T. Take four sets of readings using 1Kg, 2Kgs, 3Kgs, and 4Kgs. masses. Tabulate all the results.

ANALYSIS

1. Plot the four graphs of power lost in the gear train vs. speed on the same sheet.
2. Plot the efficiency vs. speed for the four torques on the same sheet.

DISCUSSIONS AND CONCLUSION

Discuss and draw conclusions on your results.

DEPARTMENT OF ENVIRONMENTAL & BIOSYSTEMS ENGINEERING

FEB 324 THEORY OF MACHINES – LABORATORY EXPERIMENT

FORCED VIBRATION OF A SINGLE DEGREE OF FREEDOM SYSTEM

OBJECT: To obtain curves of amplitude of the system against the frequency of excitation for various degrees of damping, and to derive the undamped natural frequency of vibration and damping ratio in each case.

INTRODUCTION

An elastic system undergoes forced vibration when it is subjected to a continuous exciting force. Usually this is brought about in machinery by unbalanced rotating parts. This vibration can cause damage to machines if its frequency of vibration coincides with the natural frequency of vibration of the system. Theoretically, at this condition, the amplitude of vibration should increase to infinity. In practice, this is not true because of resistances to the vibration from friction, and strain energy absorbed by the parts undergoing deformation such as the spring, etc., and they maintain the maximum amplitude at a certain level. However, the outcome of such vibration is to cause failure of machine parts especially by fatigue.

The designer desires to know the resonance point of this product so that he can design the running condition at a point remote from resonance. The experiment demonstrates a method of determining these parameters.

APPARATUS

Fig. 1 shows the apparatus used. A rigid bar is pinned at one end and a spring is attached towards the other end and fixed on the frame. An electric motor driving an out of balance mass is mounted on the bar. The motor has a variable speed which enables different frequencies of excitation to be used. A dashpot fitted where oil is used to get different values of damping for the system.
The out of balance mass rotates at half the motor speed and the speed of the mass is the one used in the analysis. A speed control unit gives a reading of the motor speed.

A velocity pick up mounted on the bar generates a voltage proportional to the linear velocity and hence the linear displacement of the bar the point where the velocity pickup is mounted. The voltage is small and is amplified by an electronic unit and fed into a voltmeter where readings of amplitude of vibration are taken in volts millivolts. A **CRO** is used to display the shape of the trace produced by the vibration for interest's sake. The frequency of the exciting force (and the resulting vibration) can be measured with a microscope or the CRO. However, the frequency is determined from the speed control unit. The calibration factor of the pick-up unit is 44 mV per mm/S. Using this calibration factor, the actual values of velocity and displacement can be determined if so desired.

THEORY

When the excitation forces the system to vibrate, there are inertia torques about the pivot given by $I_p \theta$

There are external damping torques opposing inertia and exciting torques given by $C\theta$ where C is damping coefficient There is a spring torque $Ka^2\theta$ opposing inertia and exciting torques.

The exciting torque = T Cos wt

The internal torque = All external torques

Then $I_P \ \theta = -C\theta - Ka^2 \ \theta + T \, Cos \, wt$

$$I_P \ \theta + C\theta + Ka^2 \ \theta = T \, Cos \, wt \qquad (1)$$

This is a linear record order differential equation, solved by employing the mathematics differential equations as follows:
It has been shown (in free vibrations laboratory sheet) that

$\frac{{}^I p}{Ka^2} = \frac{I}{W_n^2}$ where W_n is the natural frequency of vibration without damping.

$\frac{C}{Ka^2}$ Can be expressed as $\frac{2\nu}{W_n}$ since, $\nu = \frac{C}{C_C} = \frac{C}{2a\sqrt{K \, I_P}}$

Then $\frac{{}^*\theta^*}{W^2{}_n} + \frac{2\nu \, o^*}{W_n} + \theta = \frac{T}{Ka^2}$ Cos wt. (2)

If $\frac{T}{Ka^2}$ is taken as A_o and recalling that Cos wt = Real part of e^{jwt} i.e Re^{jwt}

Then $\frac{{}^*\theta^*}{W^2{}_n} + \frac{2\nu \, o^*}{W_n} + \theta = = A_o \ Re^{jwt}$ (3)

And this is $\frac{I}{W_{n^2}} \bullet \frac{d^2\theta}{dt^2} + \frac{2\nu}{W_n} \ \frac{d \ \theta}{dt} + \theta = A_0 Re^{jwt}$ (4)

It has also been established in mathematics that a differential equation of this form has a solution of the form $\theta = Ae^{jwt}$.

The substituting in (4) we get

$$\frac{-W^2 0}{W_n{}^2} + \frac{2^\nu jw\theta}{W_n} + \theta = A_0 \mathrm{Re}^{jwt}$$

$$= \left\{ \frac{-W^2}{W_n^2} + \frac{2\nu jw+1}{W_n} \right\} \theta = A_0 \mathrm{Re}^{jwt}$$

$$\text{and } \theta = \frac{A_o \mathrm{Re}^{jwt}}{\frac{-W^2}{W_n{}^2} + \frac{2\nu jw}{W_n} + 1}$$

Multiplying by the conjugate to eliminate the complex denominator:-

$$0 = \frac{R(1 - \frac{W^2}{W^2{}_n} - \frac{2\nu jw}{W_n} A_o e^{jwt}}{(1 - \frac{W^2}{W_n{}^2})^2 + \frac{4\nu^2 w^2}{W^2{}_n}} \qquad (5)$$

Expanding and taking the real part only

$$0 = \frac{A_o (1 - \frac{W^2}{W^2{}_n}) \cos wt + \frac{2\nu W}{W_n} \sin wt}{\text{Denominator in (5) denoted by } Z^2}$$

If an angle α is defined such that

$$\text{Cos } \alpha = \frac{1 - W^2}{z} \text{ and Sin } \alpha = \frac{2w}{z.W_n}$$

$$\text{Then } 0 = \frac{ZA_o (Cos\alpha + SinwtSin\alpha}{Z^2}$$

$$\text{Which gives } 0 = \frac{A_o Cos(wt^{Z^2} - \alpha)}{Z}$$

A_o is the deflection produced if a torque equal in magnitude to the maximum value of exciting torque were applied to the system under static conditions.

Then the maximum value of 0 which is $0_o = \frac{I}{Z}.A_o$ (7)

$\frac{I}{Z}$ is called the magnification factor = β

Note that the amplitudes read off the voltmeter are values of (A_o/Z) at the respective settings of the system. Setting A_o = the value of 0 under stating torque identified by 0_{st}

$$0_o = \frac{0_{st}}{\left\{1 - \frac{W^2 2}{W_n^2} + \frac{2W^2}{W_n}\right\}^{\frac{1}{2}}}$$

The maximum value of β occurs when W/W_n is slightly less that unit. Differentiating β with respect to W/W_n makes the determination of ν possible, if the maximum point of β on the graph of β versus W/W_n is known.

PROCEDURE

With no oil in the dashpot, switch on the CRO and check that a trace is obtained by initiating small free vibrations with your hand. Start the motor and run it at 100 rev/min i.e. the disc rotates at 50 rpm. Switch the voltmeter on and choose a suitable scale to give you a good order of accuracy for your reading. Make sure that the piston does not rub on the walls of the dashpot cylinder. The voltmeter pointer is very unsteady. Take your reading as the average of the band within which the pointer dwells most. Record the voltmeter reading and the disc speed.

Repeat the above with higher motor speeds, increasing the motor speed in steps of 100 rev.min. After each increment select a suitable range for the voltmeter to give you a reasonably accurate reading. Do not allow full scale deflection on the voltmeter since this can damage it.

When the system vibrates violently, hunt the position of resonance very carefully. If you miss it, you will not be able to get your analysis right. Record the resonance speed and amplitude.

After locating this important point, continue increasing the speed and taking readings up to a motor speed of 1500 rpm.

Repeat the above procedure with two different samples of oil having different viscosities. First, put Tellus 27 in the dashpot with the oil level about 30 mm over the piston. Remove this oil and repeat the experiment with vitrea 90. This gives you three values of damping

ANALYSIS

1. Plot amplitude of vibration against frequency, thus determine W_n for each value of damping. Plot all the curves on the same graph.

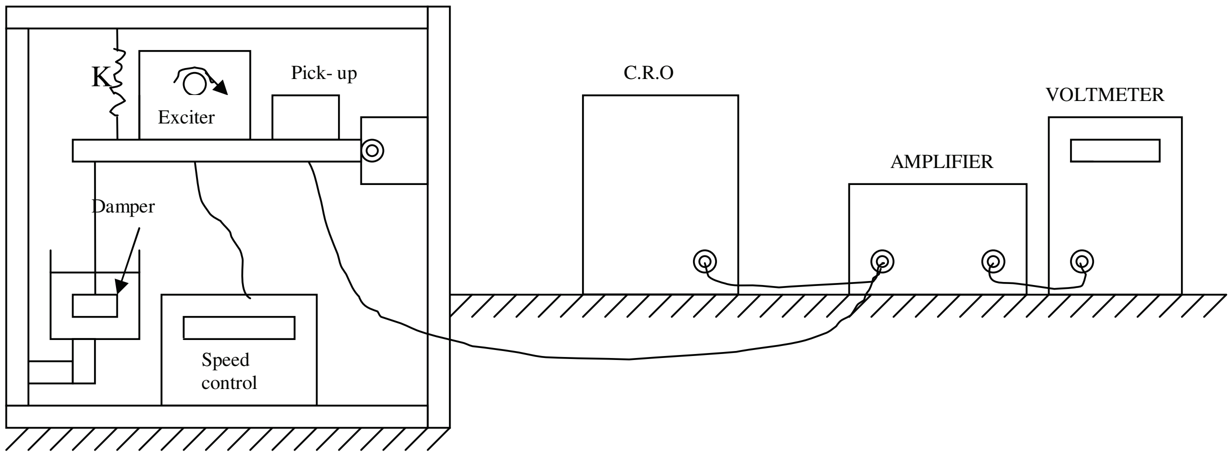
2. Plot amplitude against W/W_n for the three cases and obtain values of W_1 and W_2 for each curve being the half power point frequencies.

NOTE: The half power points are the positions on the curve for which power requirement is half that at resonance. Above this level, the system is sensitive to damping. This level is used for defining the band width W_2-W_1 corresponding to the half power points. The amplitude at this level is $1/\sqrt{2}$ of the resonance amplitude. It is important to know W_1 and W_2 because $\frac{w_2 - w_1}{w_n} = 2\nu$ when ν is small. If a stroboscope is available, values of angle α can be measured. α can be plotted against ν. If damping is too heavy α can only be estimated from the condition that $\alpha = \frac{\pi}{2}$ at W = W_n, and at this point $\theta \quad \frac{0st}{2c}$, i.e $\beta = \frac{1}{2C}$.

Derive a mathematical expression for maximum β in terms of α in terms of W_1 and W_2, hence determine ν for each curve.

DISCUSSION AND CONCLUSIONS

Comment in details on your results.



2. FREE VIBRATION OF "SINGLE DEGREE OF FREEDOM" SYSTEM

OBJECTIVE: To determine experimentally

(i) The damped natural frequency of vibration

(ii) The damping ration of a system subjected to free vibration, and

(iii) To compare the experimental values of undamped natural frequency with calculated estimates for different masses of the systems.

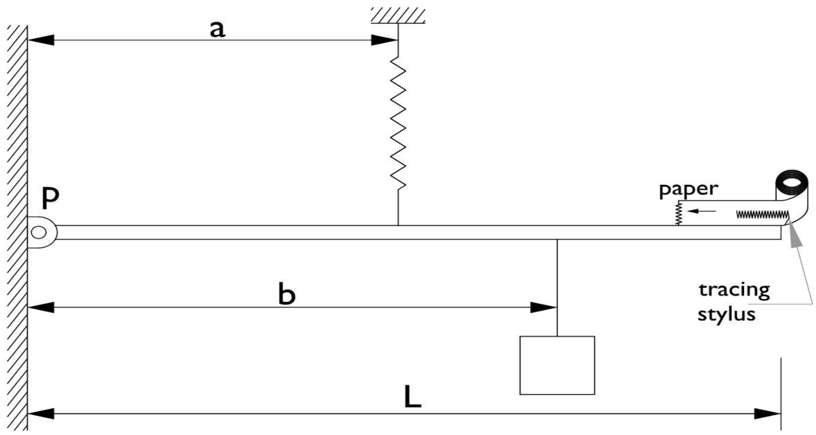
INTRODUCTION:

Free vibrations are those which occur after an elastic system in equilibrium is displaced from the position of equilibrium and let free. The system then oscillates about the equilibrium position with the amplitude of vibration reducing gradually from a maximum value to zero. The suppression of vibration called 'damping' is brought about by air resistance, frictional forces and hysteresis losses resulting from the internal friction of the strained elements such as a spring.

This experiment illustrates a method of determining parameters used in finding how much a system is dumped during its vibration and how this is related to the frequency of vibration.

APPARATUS:

The sketch below shows the elements used in this experiment.



The bar of length **L** is pin joined at the pivot **P**. A spring of stiffness **K** is attached to the bar at a distance **"a"** from the pivot and anchored to the frame of the rig. An additional mass **M** can be located at a distance **'b'** from the pivot. When the bar is displaced and let go, it vibrates; and the **electric stylus** mounted at the end of the bar moves up and down the teledeltos paper, which winds on a rotating drum. The stylus is heated electrically and burns a trace of the oscillation on the teledeltos paper. The trace produced is used to determine the dynamic parameters required.

LITERATURE REVIEW:

This dynamic system with angular oscillations has inertia torques which in this case are the products of the moment of inertia of the assembly about the pivot **P** and the angular acceleration about the pivot i.e

Inertia torque of system = $I_P \theta$ ....(1)

The inertia torque is assumed positive in the direction of positive θ. The damping torques are proportional to the angular velocity of the system.

The velocity is θ or $\frac{d\theta}{dt}$

Hence, damping torques = const. x θ
The damping torque = $cc\theta$ ....(2)
C = the coefficient of viscosity

This torque is opposite to the inertia torque, which is why it decelerates the system. Another torque of this system is the spring torque at displacement θ from the equilibrium position.

Spring torque = $aKa\theta = ka^2\theta$. (3)
(distance from pivot P perpendicular force)

The spring torque also opposes the inertia torque.

$$All\ external\ torques = all\ internal\ torques$$

$$Therefore,\ -C\theta - ka^2\theta = I_P\theta. \quad (4)$$

Negative signs are used because these torques oppose internal torques
(Further analysis of this equation will be done in class).

Note that the effect of gravity is already counter balanced at equilibrium so it does not appear in equation. (4)

Then $$I_p\theta + \frac{C}{I_P}\theta + ka^2\theta = 0 \text{......................}(5)$$

This is a linear second order differential equation solved by employing the mathematics of differential equations as follows. This is polynomial of the form

$$\frac{d^2}{dt^2}\theta + \frac{c}{I_P}\frac{d}{dt}\theta + \frac{ka^2\theta}{I_P} = 0 \text{ ....}(6)$$

The usual method of solving this type of differential equation is to use the fact that an expression of the form $\theta = C_1 e^{rt}$ has been established as a solution to this equation.

Substituting in (6)

$$r^2\theta + \frac{c}{I_P}r^2\theta + ka^2\theta = 0 \text{ ....}(7)$$

from which,

$$r^2 + \frac{c}{I_P}r + \frac{ka^2\theta}{I_p} = 0 \text{ ....}(8)$$

$$And\ r = -\lambda \pm \sqrt{\lambda^2 - \beta^2} \text{ ....}(9)$$

$$where\ \lambda = \frac{c}{2I_P} \quad and \quad \beta = a\sqrt{\frac{k}{I_P}}$$

There are three possible outcomes from equation (9) as shown below

If λ^2 is greater than β^2, it has been established that the impulse given to start vibration diminishes in one oscillation, thus the system is too stiff to vibrate. It is said to be over damped.

If λ^2 is equal to β^2 it is at the limiting condition where oscillation starts taking place. *This is the* *critically damped condition.*

$$Then \;\; \frac{C^2}{4I_P^2} = \frac{ka^2}{I_p}$$

And the critical damping coefficient

$$C_c = \sqrt{4ka^2}\, I_P \quad ....(10)$$

In this case, If λ^2 is less than β^2, damping is small and the system makes several oscillation after a disturbance before it stops vibrating. This case is what is encountered mostly in practice and is the one analyzed for this experiment.

The roots of equation (9) are complex numbers.

$$r = -\lambda \pm i\sqrt{\beta^2 - \lambda^2}$$

$$or \quad r = -\lambda \pm i\alpha \quad whwre\ \alpha = \sqrt{\beta^2 - \lambda^2}$$

$Then\ \theta = Ae^{(-\lambda+i\alpha)t} \quad and \quad \theta = Be^{-(\lambda+i\alpha)t}$ are solutions satisfying equations (6) and (7).

A general solution equations (6) and (7) is the sum of the two.

Hence $\theta = Ae^{(-\lambda+i\alpha)t} + \theta = Be^{-(\lambda+i\alpha)t}$ ....(10)

$$= e^{\lambda t}(Ae^{i\alpha t} + Be^{-i\alpha t})$$

$$= e^{-\lambda t}\{A(Cos\,\alpha t + iSin\,\alpha t) + B(Cos\,\alpha t - iSin\,\alpha t)\}$$

$$= e^{-\lambda t}\{(A+B) + Cos\,\alpha t + i(A-B)Sin\,\alpha t\}$$

$$\theta = e^{-\lambda t}\{Cos\,\alpha t + DSin\,\alpha t\} ....(11)$$

C and D being arbitrary constants.

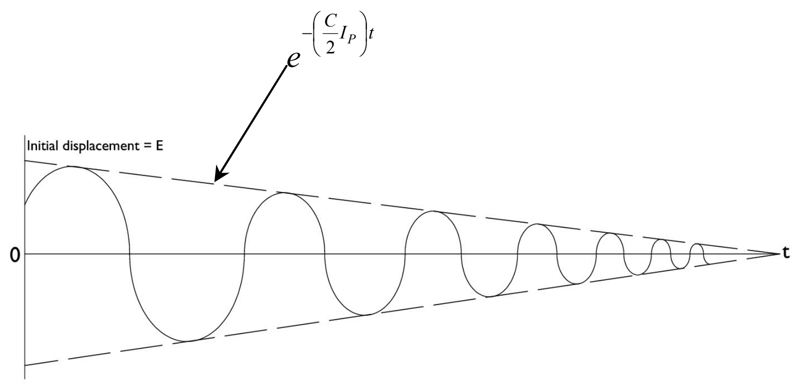
If we define ϕ such that $Sin\,\phi = \dfrac{C}{\sqrt{C^2 + D^2}}$

And $Cos\phi = \dfrac{C}{\sqrt{C^2 + D^2}}$ then equation (12) can be expressed as

$$\theta = \sqrt{C^2 + D^2}\; e^{-\lambda t}\{Cos\,\alpha t Sin\,\phi + Sin\,\alpha t Cos\,\phi\}$$

$\theta = E\,e^{-\lambda t}\,Sin(\alpha t + \phi)$ where E is an arbitrary constant......................................(12)

Equation (12) represents a vibration amplitude, the decay being governed by the $e^{-\left(\frac{C}{2}I_P\right)t}$ as shown in figure below with the initial displacement = E



The natural damped phase frequency of vibration is

$$\alpha = W_d \sqrt{\frac{ka}{I_P} - \frac{c^2}{4I_P^2}} \quad \text{..(13)}$$

From the figure above, let the displacement be θ_1 at time t_1, and θ_2 at time t_2

$$t_2 = t_1 + \frac{2\pi}{\alpha}$$

$$then \;\; \theta_1 = E\, e^{-\lambda t_1}\, Sin(\alpha t_1 + \phi)$$

$$\theta_2 = E\, e^{-\lambda t_2}\, Sin(\alpha t_2 + \phi)$$

$$= E\, e^{-\lambda\left(t_1 \frac{2\pi}{\alpha}\right)}\, Sin\left[\left(t_1 \frac{2\pi}{\alpha}\right)\alpha + \phi\right]$$

$$= E\, e^{-\lambda t_{1e}\frac{2\pi\lambda}{\alpha}}\, Sin(\alpha t_1 + \phi)$$

$$Then \;\; \frac{\theta_1}{\theta_2} = e^{2\pi\lambda/\alpha} \quad \text{..(14)}$$

$$In \; \frac{\theta_1}{\theta_2} = \frac{2\pi\lambda}{\alpha} \quad \text{..(15)}$$

This is known as the logarithmic decrement of motion. Its importance is that

$$\frac{\theta_0}{\theta_1} = \frac{\theta_1}{\theta_2} = \frac{\theta_2}{\theta_3} \text{ ....} \frac{\theta_n}{\theta_{n+1}} = e^{2\pi\lambda/\alpha}$$

$$Hence, \; \frac{\theta_0}{\theta_n} = e^{n2\pi\lambda/\alpha} \quad \text{..(16)}$$

$$Also \; \frac{\theta_0}{\theta_n} = \frac{2\pi \quad Cn}{2I_P a} = \frac{\pi \quad Cn}{I_P a} \quad \text{..(17)}$$

$$v = damping\ ratio = \frac{coefficient\ of\ viscosity}{Critical\ damping\ coefficient\ C_0}$$

$$v = \frac{C}{2a\sqrt{kI_P}} \text{ ....(18)}$$

PROCEDURE

1. Determine the spring stiffness by hanging weights and reading the displacement of the spring
2. Weigh and measure the bar
3. Vibrate the system without any additional mass and record the oscillation on teledeltos paper for each person in the group.
4. Repeat (3) with masses of 200g, 400g, 600g,800g and 1kg. Take traces for each
5. Use an alternative set of 5 masses if the spring is too weak for the selection above.

ANALYSIS

1. Estimate the undamped natural frequency for each case from the equation of motion
2. Calculate the damped natural frequency for each case using data from the traces and rotating drum.
3. Using eq 17 and 18 determine for each case.
4. From the experimentally derived parameters above, calculate the undamped natural frequency for each case, using the formula;

$$wd = \sqrt{1 - r^2}\,.\omega_n$$

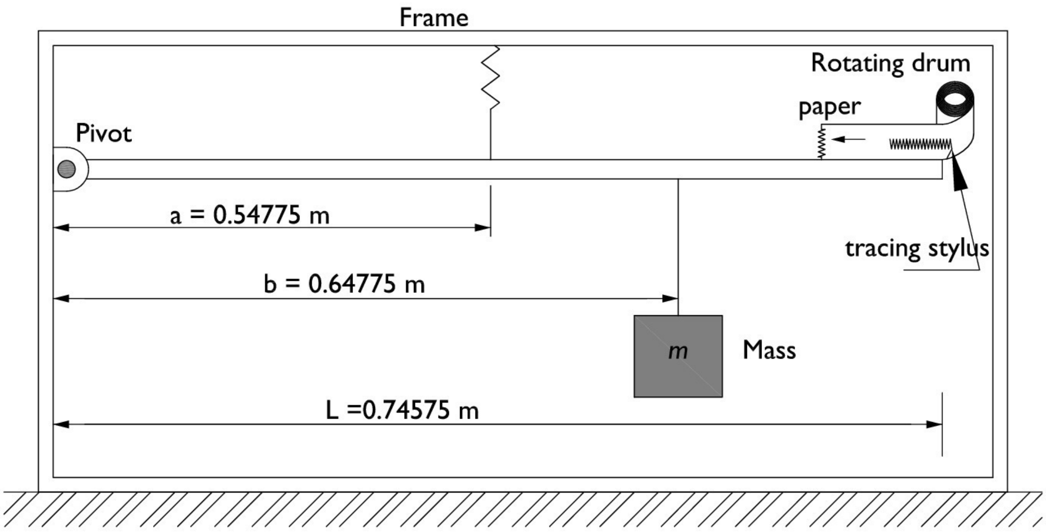
5. Tabulate all the results in the same table.

NOTE:

Results from trace are more accurate if a larger number of traces are used. K for the spring in determined graphically. All traces should be mounted on paper and included in the report under results.

DISCUSSIONS AND CONCLUSIONS

- From your results discuss and draw conclusions on the experiment, giving your opinion on the importance of this experiment in practice.
- If there are differences between the estimated and measured frequencies, suggest possible reasons.
- Check whether there is any difference in the damping ration when amplitudes are small compared with the ration when amplitudes are large.
- If these are any differences, what could be the reason?



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UNIVERSITY OF NAIROBI

DEPARTMENT OF ENVIRONMENTAL & BIOSYSTEMS ENGINEERING

3RD YEAR 2011/2012 LAB ATTENDANCE REGISTER

COURSE CODE: ………………..   TITLE: …………………………………………………………..

TIME……………………………………………………

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