DEPARTMENT OF ELECTRICAL AND ELECTRONICS ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY , COIMBATORE - 641006

CERTIFICATE

P-10

This is the Bonafide Record of the project titled " **DESIGN AND FABRICATION OF A DYNAMIC BALANCING MACHINE** *"done by Mr..................................in partial fulfilment of requirement of the Degree of Bachelor of Engineering (in Electrical and Electronics Engineering Branch) of Bharathiar University, Coimbatore - 641046, during the year 1987-88.*

HEAD OF THE DEPARTMENT **STAFF-IN-CHARGE**

Submitted for the University Examination held on 23 / 05 / 1988.

UNIVERSITY REGISTRATION NUMBER ....

INTERNAL EXAMINER **EXTERNAL EXAMINER**

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ABI

Dated: 28th March '88

Mr.Sethu Madhavan
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Coimbatoe-641 001

Dear Sir,

I am in receipt of your letter dt.23.3.88 and noted the contents.

You may use the lecture notes for your project purpose.

Wishing you all the success.

Regards,

Your s faithfully,
for ABI
(Ashok Bros.Impex P Ltd)

(S.SRIDHAR)

INDEX

Page no:

ACKNOWLEDGMENT

We wish to thank our beloved Principal Prof. R.M.LAKSHMANAN and the Head of the Department of Electrical and Electronics Engineering Prof. K.A.PALANISW. for having provided us with the neccessary facilities for the fabrication and testing of the various individual sections that has constituted our project.

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K . SETHU MADHAVAN

SAJITH KUMAR

MATHEW JOSEPH

S . R . KRISHAN

KRISHNA KUMAR

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K . SETHU MADHAVAN (Group leader)

INTRODUCTION

INTRODUCTION : THE CONCEPT OF BALANCING

DEFINITION OF UNBALANCE :

Any rotating body with an uneven distribution of mass about its axis of rotation is said to have an 'UNBALANCE'.

Whenever we talk of machines which involve parts that spin or rotate, we are always bound to have a balancing problem. A typical example is the rotor of a conventional motor.

Fig:1 shows a rotor with an unbalance caused by an extra mass 'm'.

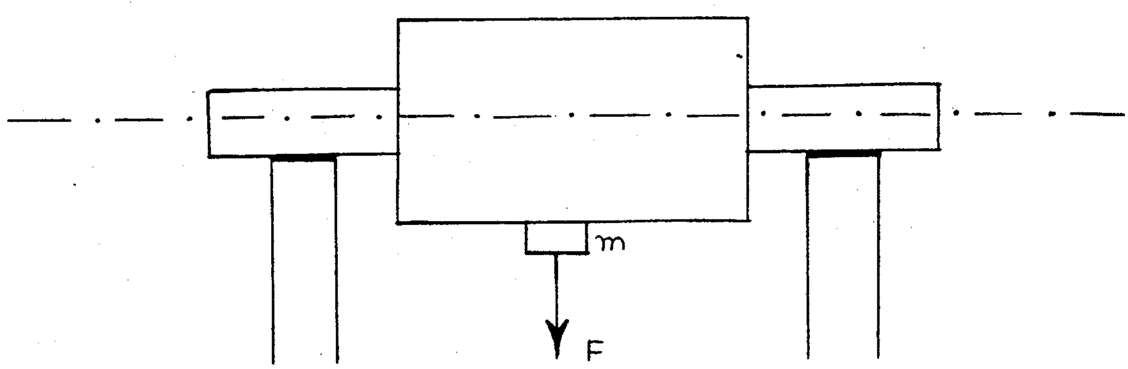


fig.1

When the rotor rotates, this extra mass 'm' exerts a centrifugal force. Obviously this centrifugal force moves around with the rotating mass and thereby causes deformation to the shaft and vibrations to the system. Moreover unbalanced bodies produce centrifugal forces that result in very high friction at the bearing ends.

Since excessive vibrations are objectionable, the location of the unbalance in the rotor is to be detected and it s magnitude reduced. Thus correct balancing leads to longevity of the parts, reliability of operation and safety. Some of the major factors that create unbalances in a system are listed below.

(1) Out - of - centre machining

(2) Non - uniform windings in armatures

(3) Blades of different sizes on rotors

(4) Internal flaws in castings

(5) Uneven density of material .

Recent technological developments in the engineering field has necessitated higher speeds of revolutions of rotors with the smoothest possible run and minimum weight per unit power . Therefore a large variety of sophisticated and reliable industrial Balancing machines have been developed to suit all possible applications .

The increasing importance of balancing in the industry also calls for higher accuracies . As is the case in all fields of science , the advent of the computer has opened new dimensions in the art of balancing and has enhanced the speed, accuracy and reliability of the balancing operation . Depending on requirements, balancing machines , from the relatively simple to complex machines , can be equipped with a video screen and a microcomputer . A computer with a CRT readout screen and a suitable program solves the multiple plane balancing problem. Further, the most exacting requirements for balancing quality places the highest demands on all balancing machine components . Machines have been developed which balance huge turbine rotors to the extremely small and lightweight balance wheels in wristwatche

This project which has been sponsored by MEENU EQUIPMENTS LTD . COIMBATOR involves the design and fabrication of a dynamic balancing machine . The machine, which is of a soft bearing type , employs a stroboscopic electronic system .

(Classification of balancing machines have been elaborately discussed in the subsequent sections)

CLASSIFICATION OF BALANCING MACHINES

CLASSIFICATION OF BALANCING MACHINES

TYPES OF UNBALANCES

Basically , there are three types of unbalances .

(1) STATIC OR FORCE UNBALANCE

This kind of unbalance exists only on one side of the rotor and is illustrated by fig.1 .

(2) COUPLE UNBALANCE

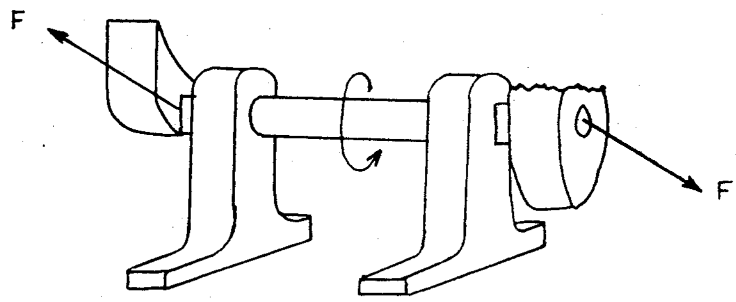


fig.2

As shown in the above figure , two equal weights are present in two different planes . The weights are displaced from each other by 180°.This type of unbalance is referred to as dynamic unbalance . When a rotor possessing a couple unbalance rotates , two equal forces are produced which constitute a couple and this couple gives rise to vibrations . Eventhough these forces are equal and opposite in direction, they do not cancel each other because they are axially displaced .

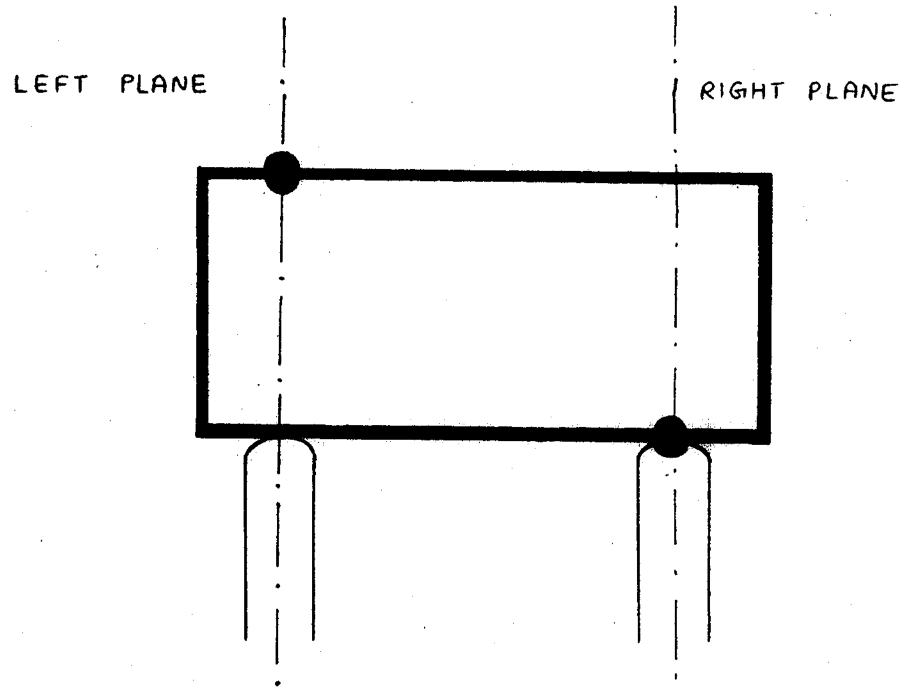


fig.3

The two types of unbalances so for discussed , exist only in theory . In actual practice , a large number of static and couple unbalances exist simultaneously in a rotor . It can be shown that all such unbalanes can be represented by two weights (unbalances) in two planes . (fig.3)

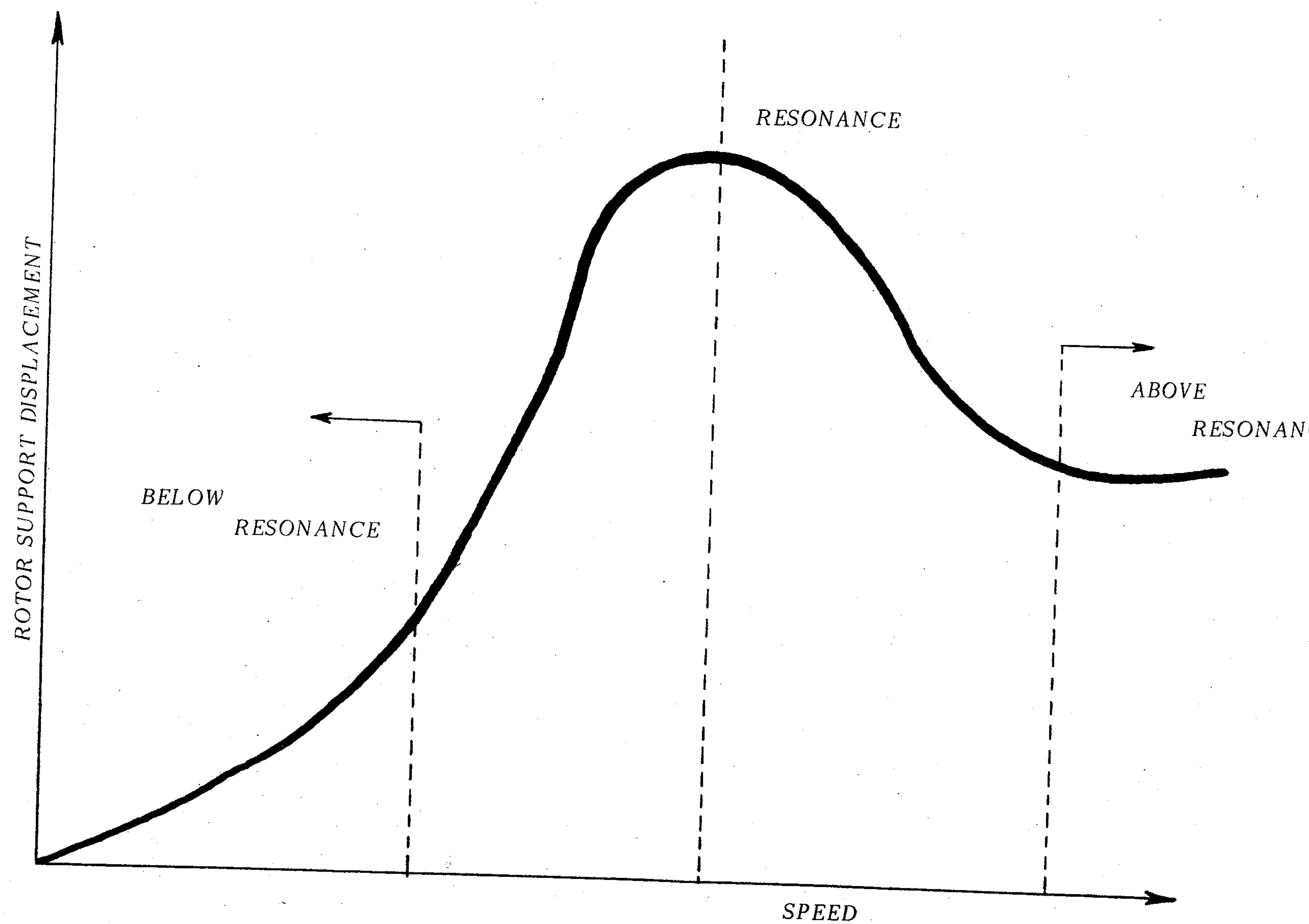
THE DYNAMIC BALANCING MACHINE :

A dynamic balancing machine is a machine on which a rotor can be rotated and the unbalance of the rotor is indicated on suitable instruments .

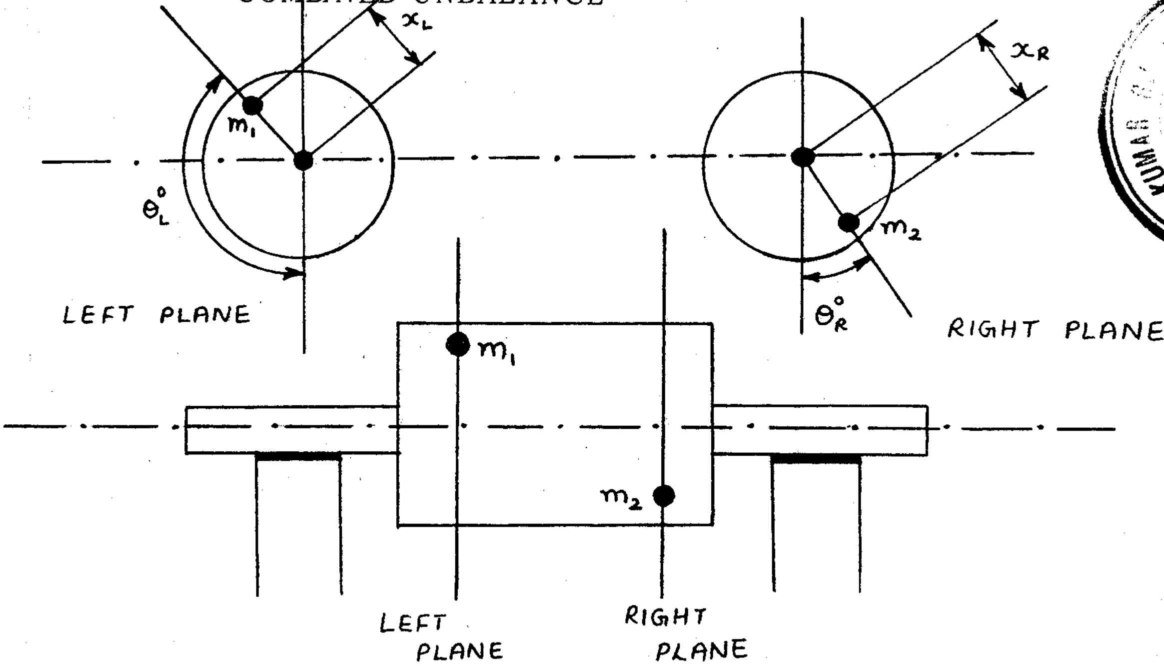
The rotor is normally supported on two supports and is rotated using a belt drive . Two independant measurements at the two supports yield the required information required to define the unbalance of the rotor . These measurements are normally done in one direction and the machines may be classified into three types , depending on the type of support .

For a fixed unbalance , the support displacement as a function of speed is given in fig.4. Balancing machines are classified into three basic categories depending on their region of working with respect to the resonance area that is indicated in fig.4

FIG. 4 Variation of support displacement with speed



COMBINED UNBALANCE



(a) <u>HARD BEARING MACHINES</u>

Machines which work well below resonance are called hard bearing machines. In addition to this , balancing of all kinds of rotors is done without going through any caliberation or trial runs and only rotor dimensions need be dialled into the machine .

(b) <u>SOFT BEARING MACHINES</u>

Machines whose pedestals work well above resonance are called Soft bearing machines. The job supports of such machines are normally very flexible and can easily be moved by hand . Hence the name .

(c) <u>RESONANCE TYPE MACHINES</u>

Machines which work at or close to resonance are called Resonance type machines. The displacement of job support and therefore the readings on these machines change drastically with small speed changes . Therefore the exact measurement of unbalance is difficult if we try to directly correlate the unbalance with the electrical output of the sensors or pickups . However , there are methods by which this problem can be overcome .

This project involves the design and fabrication of a Soft Bearing machine which measures the amplitude of vibrations .

THE SOFT BEARING MACHINE

THE SOFT BEARING MACHINE

A simple form of this machine was one of the first machines to be evolved.In this type of a machine,the two work supports (carriages) of the balancing machine are mounted on springs as illustrated in fig.5 .

CARRIAGE OF A SOFT BEARING MACHINE

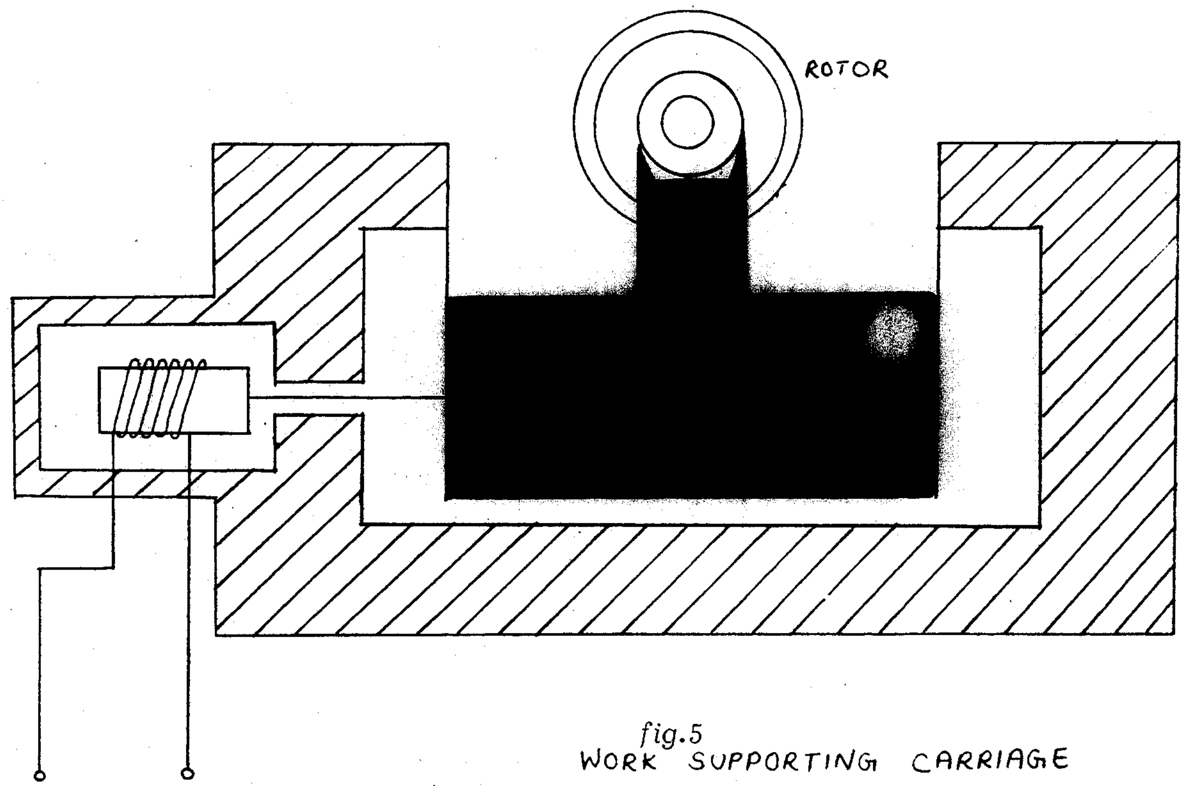


fig.5
WORK SUPPORTING CARRIAGE

ANALOG OUTPUT

(Free to oscillate in the horizontal plane)

An analog output of the vibrations was obtained using an electromagnetic transducer.

These carriages,along with the rotor to be balanced,can therefore oscillate in the horizondal plane.When the rotor is rotated,the centrifugal forces generated by the unbalance,make it oscillate to and fro along with the supporting carriages.The amplitude of this oscillation depends on the amount of force.Since the force is directly propotional to the unbalance (to be mathematically proved in a later section of this report),measurement of amplitude gives an indication about the extent of unbalance.

This design however has a few short comings,which are elaborately discussed below.

(1) No possibility of permanent caliberation

On a Soft Bearing machine,it is necessary to caliberate the machine upon shifting from one type of rotor to another.This is because the amplitude of oscillation that is measured does not depend only on the force.It also depends on the mass that vibrates and the distribution of the mass,which includes the rotor mass and the supporting carriage mass.As this changes from rotor to rotor,the same unbalance on two different rotors give different readings.Hence everytime a new rotor is used,the machine has to be caliberated.Caliberation is quite a tedious process.Known unbalance(or weights) have to be added to the rotor and the magnification of the measuring system adjusted so that the meter on which the unbalance is read shows the unbalance on the rotor.The process is further complicated due to the fact that the rotor has its own unbalance.Even in the most sophisticated Soft Bearing machine incorporating a computer and special aids for caliberation,two or three caliberation runs are required every time a new type of rotor is taken up.Hence balancing on this machine is not only time consuming but also requires a real expert to do the job.

(2) Inherently fragile in design

In order to measure small forces,the carriages of the balancing machine must be quite flexible.This makes the oscillating structure very fragile and unsuitable for rough conditions that generally prevail in workshops.In general, it requires delicate care and maintenance.For example,the pedestals are normally locked before starting the machine to avoid damage during acceleration.After the acceleration phase is over, they are unlocked for measuring the amplitude of the oscillation.

(3) Small weight range

A Soft Bearing machine cannot handle jobs of widely varying weights.Optimum accuracy is achieved only in the upper weight range of the machine and accuracy suffers as the mass of the rotor is reduced.This is because of the fact that the large weight of the machine carriage must also be moved along with the small rotor that is being balanced.The large weight of the machine carriage reduces the amplitude of oscillations,thereby reducing accuracy.The weight of the carriage is normally referred to as Parasitic mass and is more prominent for smaller rotors,where the rotor mass is comparable or smaller than the parasitic mass.

The above shortcomings are overcome in the Hard Bearing machine (beyond the scope of this project report),which employs a force measuring transducer.In this type of machines,there are no oscillating parts and the movement of the machine carriages is negligibly small.The Hard Bearing machine,though more expensive,has been found more useful in most applications and is therefore fast replacing the Soft Bearing machine.

ANALYSIS OF UNBALANCE

QUANTITATIVE ANALYSIS OF UNBALANCE

The unbalance 'U' of the rotor of fig.1 is given by

$$U = mr$$

where m = unbalance mass

r = radius at which the mass is located

It is worth noting here that the unbalance 'U' is independant of speed and therefore exists even when the rotor is stationary.

The centrifugal force that develops when the unbalanced rotor rotates is given by 'F'.

$$F = (m v^2)/r$$

where v = linear velocity

If 'w' is the angular velocity,then $v = r w$

Therefore $F = (m r) w^2$

but unbalance 'U' $= m r$

which implies that $F = (U w^2)$

Thus the centrifugal force 'F' is directly propotional to the unbalance 'U'.Hence a reduction in the unbalance leads to a reduction in the force.Vibrations in the system are thereby effectively suppressed to a large extent.

Besides determining the amount of unbalance,it is also necessary to locate the area where the unbalance mass exists.The location is generally expressed as an angle at which the unbalance is located.

For instance the unbalance in fig.3 can be expressed in the following manner.

Unbalance in the left plane = $U = m_1$ at θ_L°

Unbalance in the right plane = $U = m_2$ at θ_R°

GENERAL DISTRIBUTION OF UNBALANCE

Consider a rotor with a distribution of mass as shown in fig.6

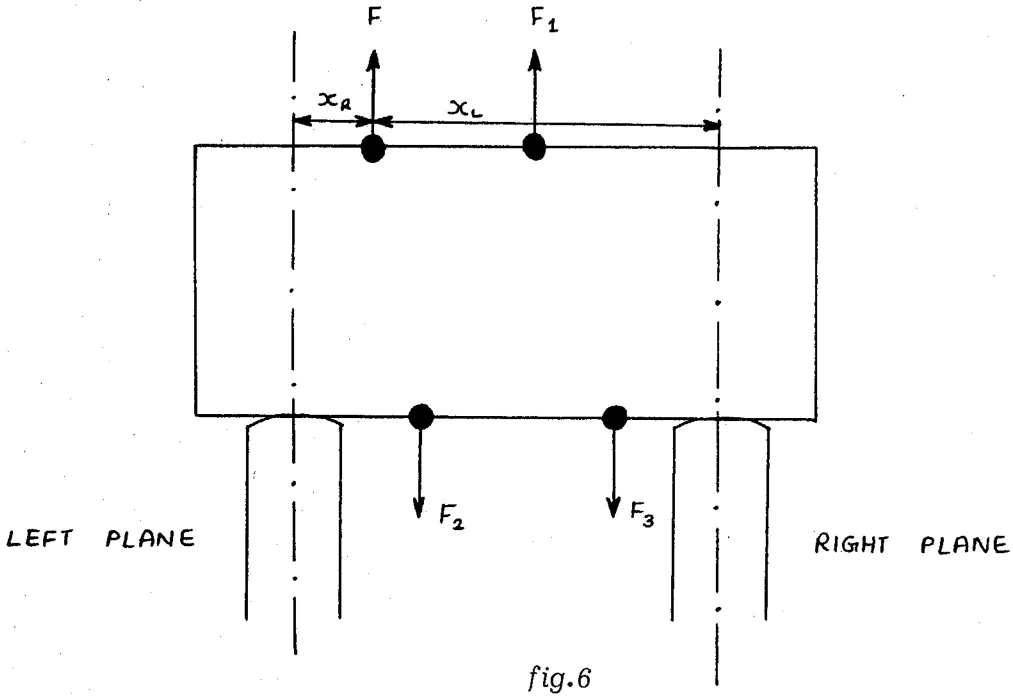


fig.6

When the above rotor rotates,each unbalance on it gives rise to a centrifugal force,as indicated in fig.6.Assuming that the rotor under consideration is a rigid body,each force on the rotor can be resolved into two planes.For example,resolution of the force F in the left and right planes yield

$F\ (left) = x_L\ F$

$F(right) = x_R\ F$

On the same lines,all the forces shown in the above figure can be resolved in the two planes.Further,these two forces can be represented simply by the unbalance at those points since these unbalances would result in the two forces.This case is illustrated by fig.7

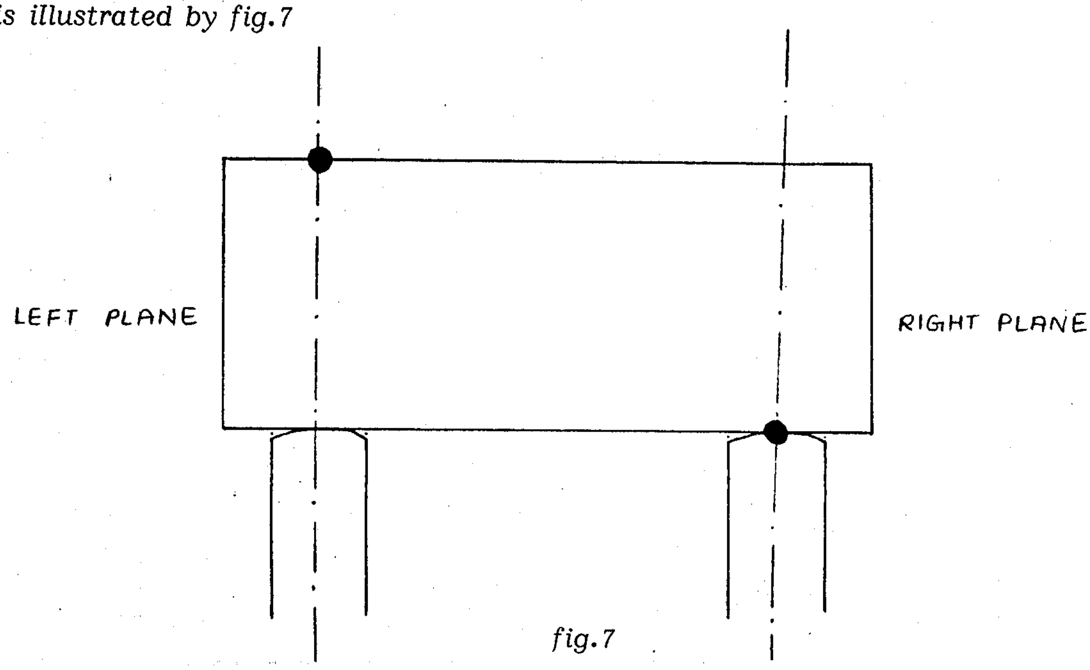


fig.7

The above principle however is not the most general case but it illustrates that the total unbalance of a rigid rotor can always be represented by two weights (or unbalances) in any two desired planes.

The dynamic balancing machine operates on the above principle.The rotor to be balanced is placed on the two work supporting carriages of the balancing machine and is rotated using a belt drive.The centrifugal forces created due to the unbalances act on the carriages and are measured.These two forces give an indication to the total unbalance of the rotor in the two planes (the planes along which the carriages are placed) .While measuring the unbalance,it has to be ensured that the rotor is rotated about its normal axis of rotation.If the rotor is rotated about any other axis,spurious unbalance will be created by the displaced mass of the rotor.The simplest method of achieving the above mentioned objective is to rotate the rotor while supporting it on its bearing surface .

CORRECTION OF UNBALANCE

THE MEASUREMENT AND CORRECTION PLANE

As already explained in the previous section,the dynamic balancing machine measures unbalance in two planes.These are the planes in which the rotor is supported on the balancing machine and generally the rotor has its bearings along these planes.But the unbalance correction is made along a different plane;where it is convenient to add or remove weights.The former is called the measurement plane while the latter is called the correction plane.

Thus the unbalance corrections to be carried out differ from the measured unbalance values.

Fig.8 illustrates the above theory,wherein the unbalances are measured and corrected in different planes.

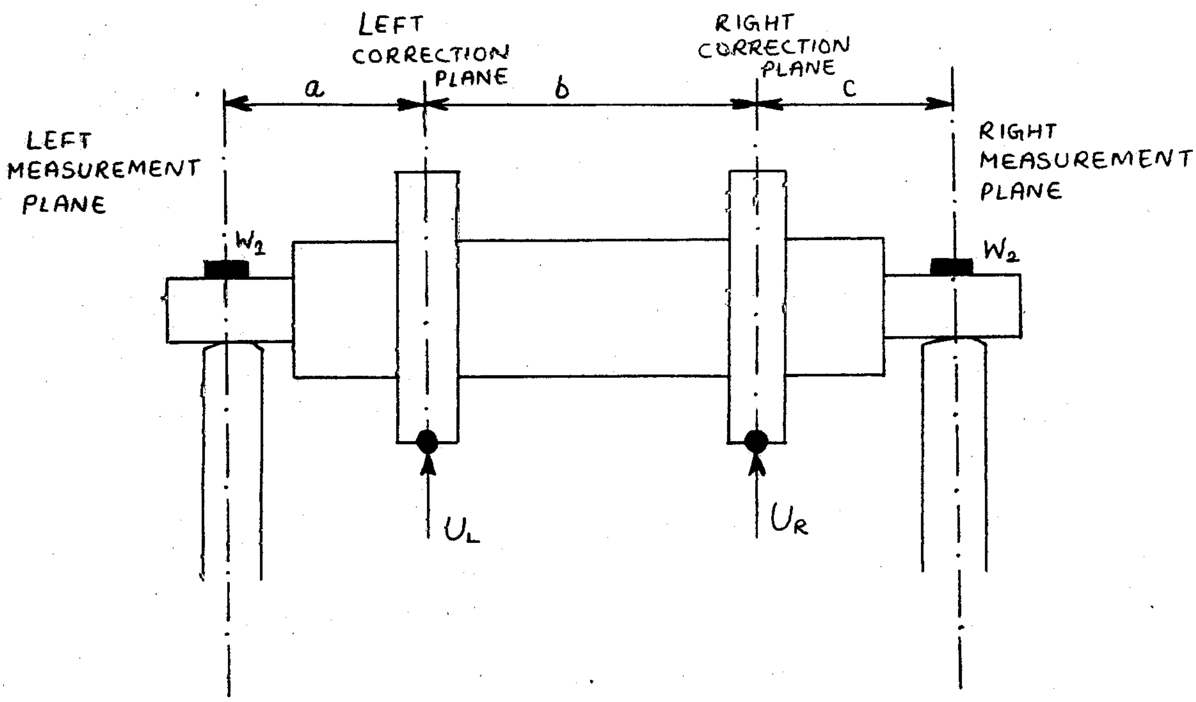


fig.8

In order to calculate the correction to be carried out,moments are taken about any two points and are then equated to zero.Taking moments about the left correction plane ,

$$U_R\, b = W_2 (b + c) - W_1 a$$

$$\Rightarrow U_R = W_2 (b + c) / b - W_1 a / b$$

where U_L = unbalance required in the left correction plane

U_R = Unbalance correction needed in the right correction plane

W_1 = Unbalance in the left measurement plane

W_2 = Unbalance in the right measurement piane

Now the actual weight m to be added has to be found out. This is evidently dependent on the radius r at which the correction has to be made.

The weight for the right plane is given by m

$$m_R = U_R / r_R$$

Similarly, the correction required in the left correction plane is also found out.

$$U_L = W_1 (a+b) / b - W_2\, c / b$$

Also $m_L = U_L / r_L$

where m_L = mass correction needed in the left correction plane

m_R = mass correction needed in the right correction plane

r_L = radius at which the correction is made on the left side

r_R = radius at which the correction is made on the right side

In the above example, all the unbalances were taken on the top side of the rotor to make the calculations simple. This was just to illustrate the principle. In actual practice the unbalance weights could be found anywhere on the rotor and therefore complex calculations have to be carried out involving complex vectors. This process is called plane seperation

An example of such a case has been discussed in a subsequent section.

BALANCING ACCURACY AND BALANCING SPEED

Accuracy of Balancing

Just as in a machined component each dimension has a tolerance (the permissible deviation from a specified value), a rotor to be balanced always has a certain *permissible residual unbalance.Residual unbalance is the unbalance that is left over after the rotor is balanced.Obviously,lower the residual balance,higher is the accuracy of the balancing operation.The accuracy to which the rotor has been balanced depends on various factors.One important factor is the speed at which the motor has to work.As the rotor goes to hither speeds,the centrifugal force increases as $F = (U w^2)$.An attempt is therefore made to keep the force 'F' within a limit.The speed is accordingly chosen.The level to which any rotor should be balanced can be decided with the help of ISO 1940 .*

WORKING SPEED AND BALANCING SPEED

Operating speed and balancing speed

Operating speed is the speed at which the rotor is ultimately going to work whereas balancing speed is the speed at which the balancing machine works in order to sense the unbalance of the rotor.These two speeds need not be the same.A rigid rotor when balanced at one speed will be balanced at all speeds.As discussed earlier,the unbalance 'U' is independent of speed.

BALANCING SPEED AND BALANCING ACCURACY

Accuracy of speed has no relation to the balancing speed.Sensitive machines can measure small unbalances at slow speeds while a cruder machine may measure this unbalance at higher speeds.

Balancing at lower speeds has a lot of advantages such as faster acceleration and decceleration,low wear and tear of the moving parts,greater safety,lower power requirements etc .Therefore it is advisable to select the lowest speed at which the desired accuracy can be achieved.

the above analysis is however not true for flexible rotors which change shape with speeds.Such rotors form a very small fraction of rotors in industry and a completely different system of balancing must be followed for these rotors.

SIGNAL ANALYSIS

THE ELECTRONIC SYSTEM

The electronic system of a dynamic balancing machine comes into play after the job has been accomodated and rotated on the balancing machine.In modern machines,the unbalance is first converted into an equivalent electrical signal.The commonly used pickup is a transducer.We have used an electromagnetic transducer for the conversion of the unbalance into an equivalent electrical signal.The apparatus mainly consists of a moving coil and a magnet.The relative motion between the two gives rise to an electrical signal which is propotional to the rate of change of flux in the coil.

The following four basic functions must be performed by the electonic system of a dynamic balancing machine.

(1) Sensing the unbalance

(2) Elimination of noise

(3) Analysis of signal

(4) Display of values

SENSING THE UNBALANCE

The unbalance is usually sensed and converted into an equivalent electrical signal by an electromagnetic transducer.

NOISE ELIMINATION

Listed below are some of the major reasons for noise generation.

(1) Roughness of journals

(2) Non-smooth running of bearings in the roller supports

(3) Air-turbulence

(4) Electrical interference

(5) Vibrations of adjacent machinery (transferred through the machine foundation)

(6) Vibrations generated due to driving belt,gears or other driving systems.

In a balancing machine, the primary interest is to measure the signal which has the same frequency as the rotational speed. All other frequencies are referred to as noise.

The output signal of the transducer (pickup) is a sum of various sinusoidal signals.

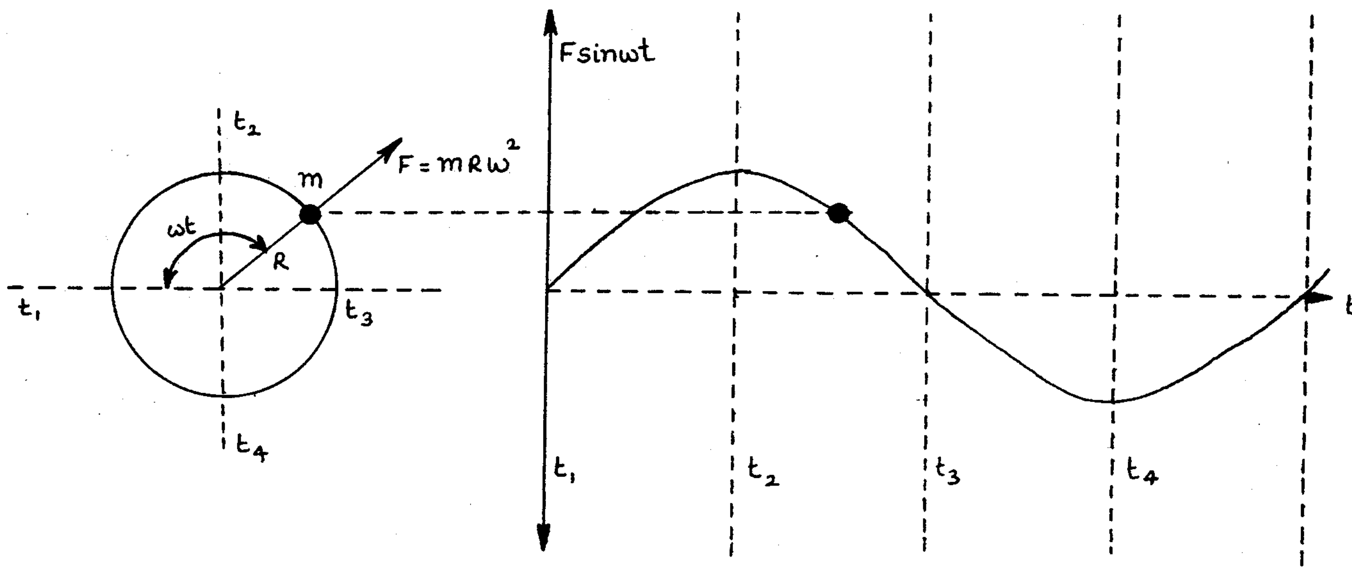


Fig.9

Fig.9 shows the horizontal component of the centrifugal force due to the unbalance on the rotating body. This is exactly what the transducer measures. The amplitude of the sine wave is propotional to the amount of unbalance and its phase is directly related to the angle of unbalance. (The sine wave has a frequency which is exactly the same as the rotational speed). Thus for an accurate analysis of the signal, a pure sinusoidal output is required from the transducer. However this is not the case as noise generation is unavoidable. Evidently, the presence of noise leads to inaccurate balancing and therefore noise elimination is a necessity in balancing machines.

THE FILTER SECTION

Noise can be nulled by eliminating all frequencies except the frequency of rotation.Frequency selectivity is usually achieved through a filter system. Filter systems are broadly classified into two categories as given below.

(1) Tuned filters - with manual tuning to the speed in use.

(2) Synchronous filters - with automatic speed tracking feature.

In this project,we have employed a tuned filter for the purpose of noise elimination.Th properties of tuned filters have been elaborately discussed.

TUNED FILTERS

These filters have a fixed gain in relation to the frequency of the machine.The filter is manually tuned to the balancing speed in use.

A certain minimum bandwidth has to be inherent in order to account for minor speed fluctuations.Bandwidth is the frequency range where the filter gain remains constant.It is a general practice to provide a bandwidth of 5% to 10% .Reduction in bandwidth will make the balancing operation very unreliable,as small changes in speed result in large changes in the filter gain and phase response,thereby yielding erronous results.Moreover the increase in bandwidth results in the noise going through.Bandwidth of 5% to 10% has been found to be a compromise solution.A typical characteristics of a tuned filter is shown in fig.

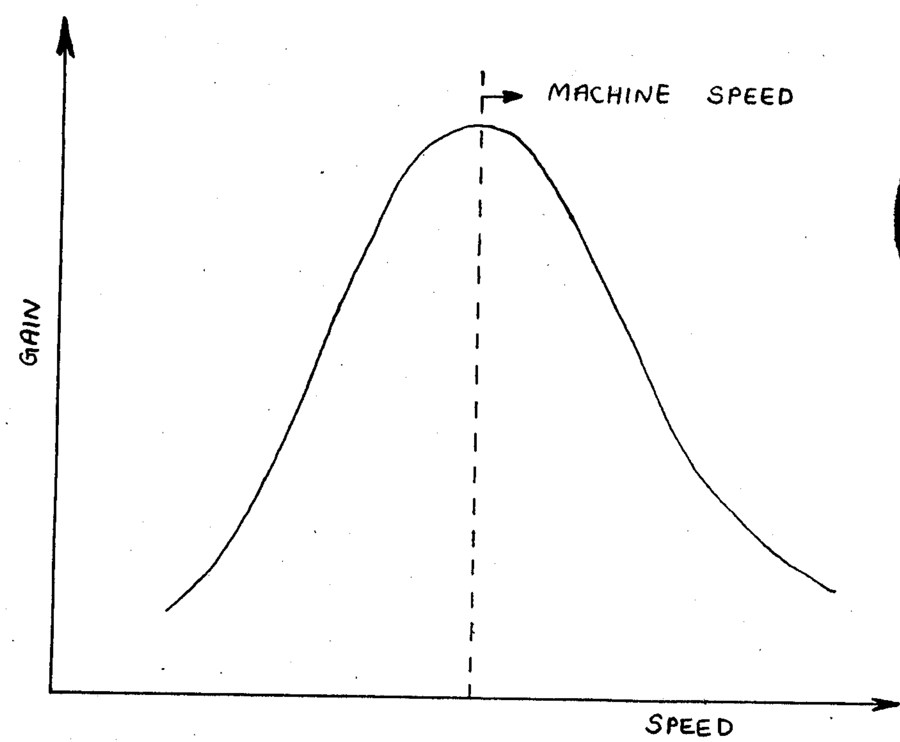


Fig.10

SIGNAL ANALYSIS

The balancing machine has to perform the job of displaying the amount and the angle of unbalance.It may be necessary to know the unbalance values for the required balancing planes or we may need to define the unbalance for making corrections at a few fixed points.It may be also essential to estimate rotor deflections for flexible rotors or carry out plane translations for balancing crankshafts.

A general analysis of the electronic system we have employed is given below.

FUNCTIONAL DESCRIPTION OF A STROBOSCOPIC ELECTRONIC SYSTEM

The schematic diagram for the electronic system used in this project is given in fig.11 and fig. 12

The electromagnetic transducer (pickup) is mounted on the rotor support of the balancing machine.A unidirectional pickup has been used to sense the rotor unbalance.The force (or) vibration is measured along the horizontal plane.As explained earlier,besides picking up the unbalance signal,it also picks up noise signals.

The output of the transducer is fed to an amplifier where the signal is amplified in order to increase the efficiency of analysis.The amplified signal is then fed to a tuned filter stage ; which has been tuned to the operating speed.This section obviously eliminates the noise.Thus a relatively clean sinusoidal signal which represents the unbalance is obtained at this stage.This signal is rectified and fed to a meter to indicate the amount of unbalance.This signal is also fed to a trigger circuit which gives one pulse per revolution which in turn is flashed on to the rotor with the help of a stroboscopic lamp.The rotor has various number markings at different angles.The stroboscope flashes so as to illuminate one number depending on the angle of unbalance.This is the position on the rotor where weight addition / removal is required.This type of indication where both the amount as well as the angle of the unbalance is displayed is classified as polar indication .

BASIC BUILDING BLOCKS OF THE ELECTRONIC SYSTEM

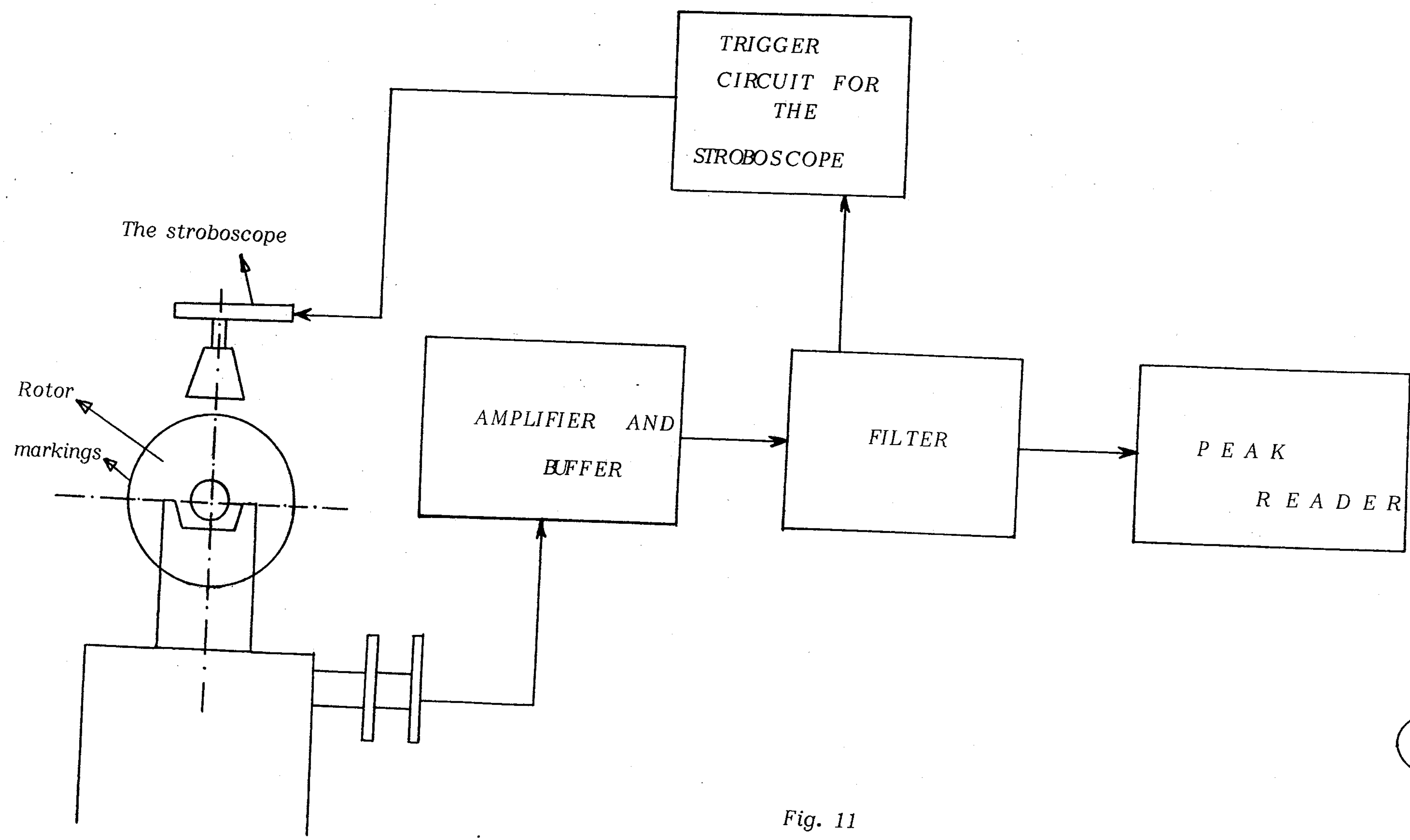
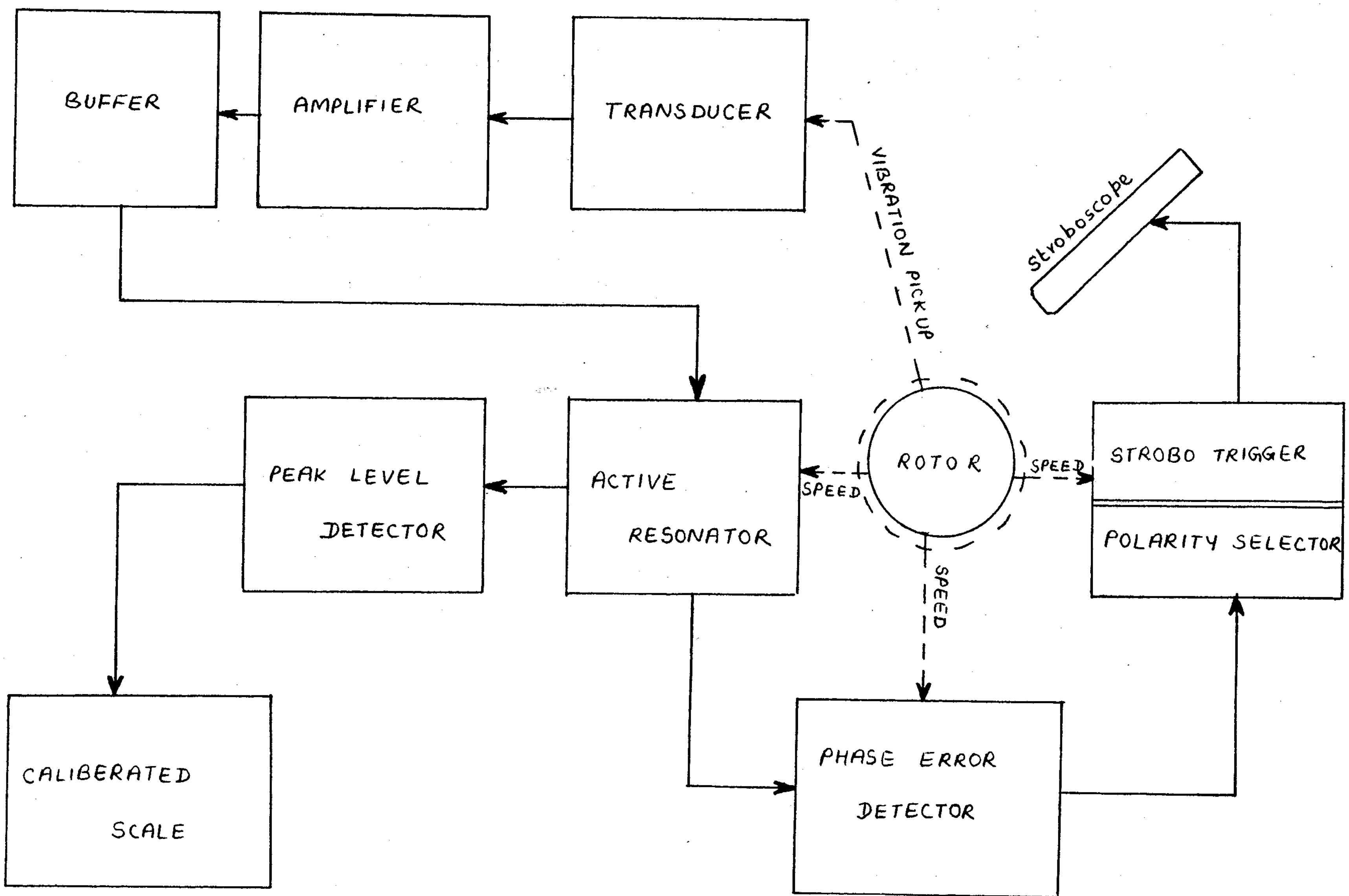


Fig. 11

Modified Electronic system : Building blocks



(The modified electronic system)

THE PHASE ERROR DETECTOR

WORKING OF THE PHASE ERROR DETECTOR

The electronic system of this project requires the measurement of the phase difference between two signals of the same frequency.The circuit given in the previous page makes this measurement accurately eventhough the input amplitudes are different.The output V_o will be zero if the phase difference between the two input signals V_A and V_B is zero.If the phase of V_B leads the phase of V_A then V_o will be positive.The dc output voltage V_o will vary linearly from zero to $+V_{z2}$ as $\phi_B - \phi_A$ varies from 0 to 180° .Likewise, V_o will vary linearly from zero to $-V_{z1}$ as $\phi_B - \phi_A$ varies from 0 to -180°.This is shown for several cases in fig.

The circuits of A_1 and A_2 are zero-crossing detectors with hysteresis. A_2 is of the inverting type so that V_2 lags V_B by 180°.Because of the high gain of these zero-crossing detectors, V_1 and V_2 are rectangular waveforms as shown in fig.

C_1 , R_7 and C_2 , R_8 are differentiation networks.The diodes D_1 and D_2 select the positive pulses that result from this differentiation.The pulses from D_1 make A_3 go into the low state such that $V_5 = -V_{z1}$ .Pulses from D_2 makes it go into the other state so that $V_5 = V_{z2}$ .If V_A is exactly in phase with V_B ,then the circuit will spend equal amounts of time in the high and low states.The voltage at V_o will be zero.

If the phase of V_B leads the phase of V_A ,then the circuit will spend more time in the high state and therefore V_o will be positive.Likewise,if the phase of V_B lags that of V_A ,then V_o will be negative.The scale factor,i.e.,volts/degree,is set only by the choice of V_{z1} and V_{z2}.If these are identical diodes,the scale factor will be given by the following expression.

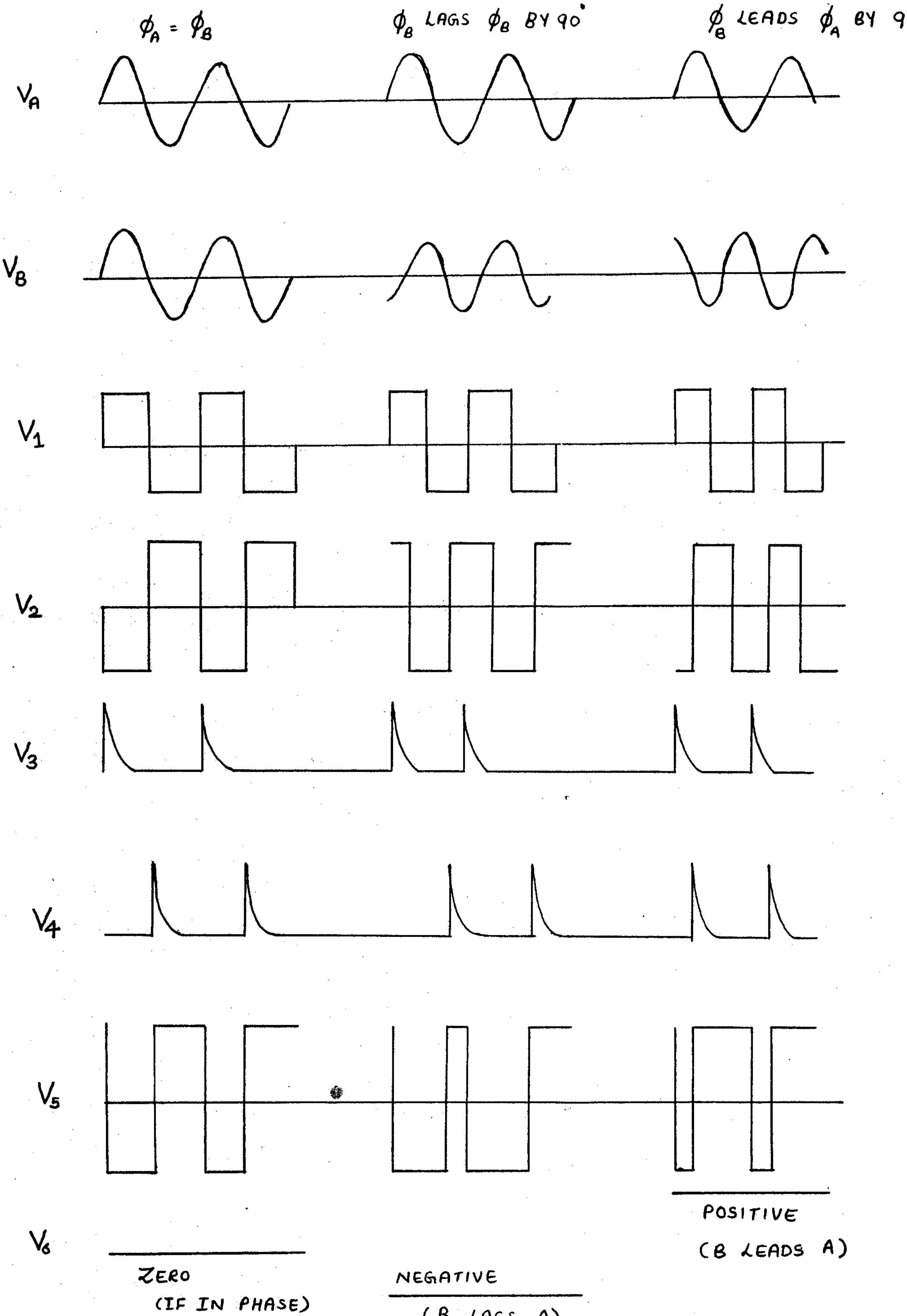
$$V_o = \frac{(\phi_B - \phi_A)V_{z1}}{180} \text{ volts/degree}$$

The range of frequencies over which accurate performance can be guaranteed *depends on different factors at the low and high ends of the spectrum. At low frequencies the risetimes of* V_1 *and* V_2 *may not be fast enough to transfer adequate trigger pulses through the differentiation network to the circuit. Also, the output filter* R_{15} *and* C_3 *becomes less efficient at low frequencies. These deficiencies result in a low* V *which is noisy or temporarily saturated at* $\pm V_{z1}$.

At high frequencies the slew-rate limits of A_1 *and* A_2 *start to reduce the peak-to-peak amplitude of* V_1 *and* V_2. *This will cause the trigger pulses* V_3 *and* V_4 *to diminish in amplitude also until the circuit no longer triggers. This must not be allowed to occur, since the circuit will again hang up in one state or the other.*

Waveforms at different locations of the circuit is given by fig. 13

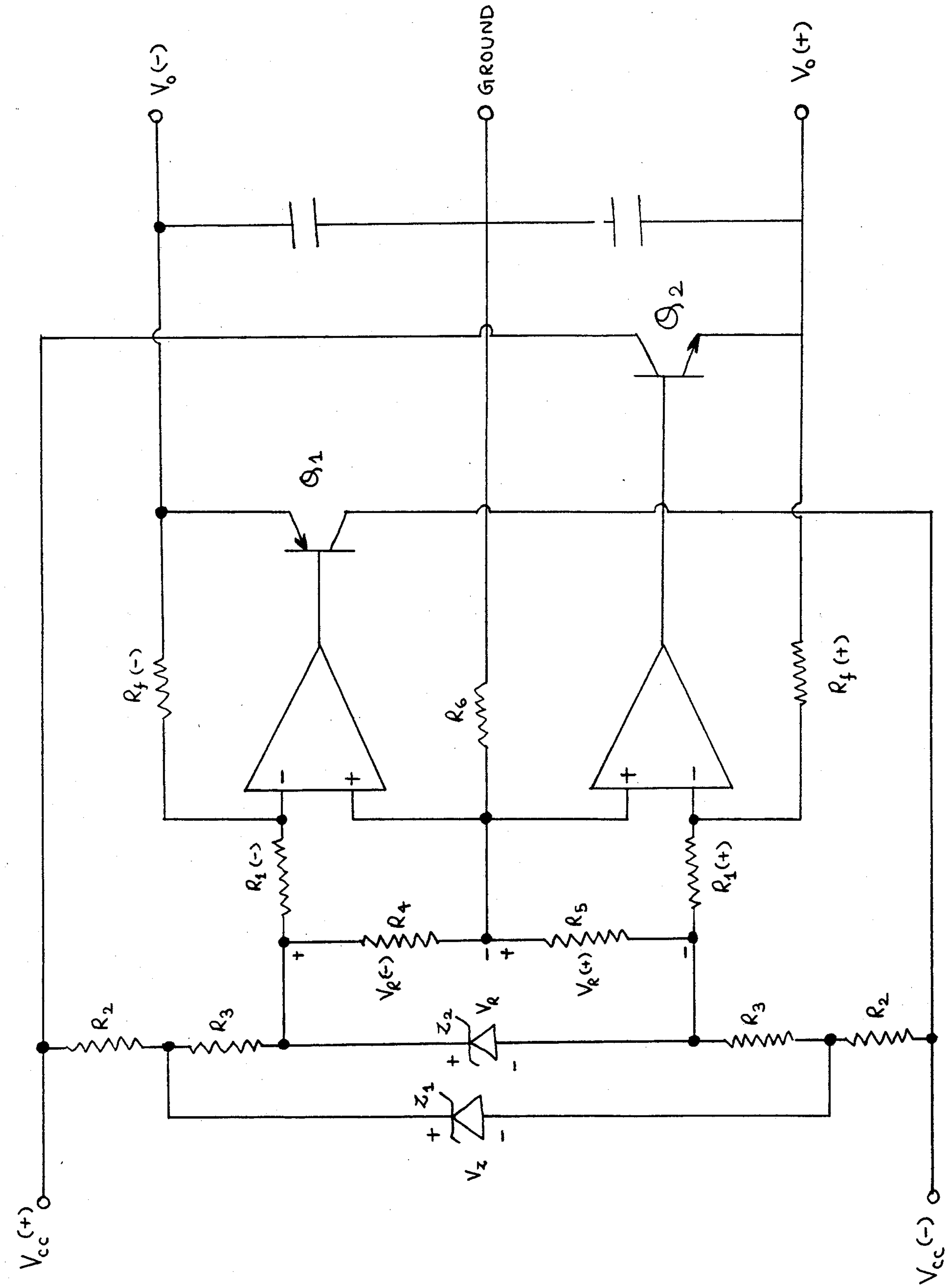
Fig.13 Waveforms of different sections of the phase error detector



11. Optimum value for R_4:

$$R_4 = R_5 R_6 / (R_5 + R_6)$$

THE TRACKING REGULATOR



$V_o(-)$
GROUND
$V_o(+)$
Q_1
Q_2
$R_f(-)$
R_6
$R_f(+)$
$R_1(-)$
$R_1(+)$
R_4
R_5
$V_R(-)$
$V_R(+)$
Z_2
V_R
Z_1
V_Z
R_2
R_3
R_3
R_2
$V_{cc}(+)$
$V_{cc}(-)$

OPERATION OF THE TRACKING REGULATOR

The stability of regulators ultimately depends on the quality (and cost)) of the reference diode.Sometimes,both positive as well as negative supplies are required.This is often implemented by using two seperate reference diodes and associated power-boosting circuitry.The circuit shown in fig. 15 uses only one reference diode for both supplies,which thereby results in considerable cost savings.

The + and - output voltages need not have the same absolute magnitude.Since the reference diode that is most likely to be chosen is the 6.4 V variety, R_4 and R_5 will each have 3.2 V applied across them.The operational amplifiers merely amplify these two voltages to produce $V_o(+)$ and $V_o(-)$.R_f and R_1 in each operational amplifier circiut can therefore be independently chosen to produce nearly any + or - voltage combination.

Z_1 is a preregulator made of a standard low cost zener.It helps maintain a constant current through Z_2 ,which is required for a highly stable V_R. R_6 is chosen to lessen the effects of input offset-current drift in the operational amplifiers.If $R_1(+)$ and $R_f(+)$ have resistances much different from $R_1(-)$ and $R_f(-)$, a seperate R_6 may be needed for each operational amplifier.

DESIGN EQUATIONS FOR THE TRACKING REGULATOR

1. Positive output voltage $V_o(+) = R_f(+)\ V_R(+)\ /\ R_1(+)$
2. Negative output voltage $V_o(-) = R_f(-)\ V_R(-)\ /\ R_1(-)$
3. Reference voltages .

$$V_R(+) = V_R R_5 / (R_4 + R_5)$$

$$V_R(-) = V_R R_4 / (R_4 + R_5)$$

4. The offset drift-reducing resistor :

R = average of $(R_1(-)\, R_f(-))/(R_1(-) + R_f(-))$ and

$$R_1(+)\, R_f(+) \,/\, (R_1(+) + R_f(+))$$

5. Resistor values :

$R_3 = (V_Z - V_R)/ (2 I_R)$, where I_R is the optimum Z_2 current.

$R_2 = (V_{CC}(+) - V_{CC}(-) - V_Z) / (2 I_Z)$, where I_Z is the optimum Z_1 current.

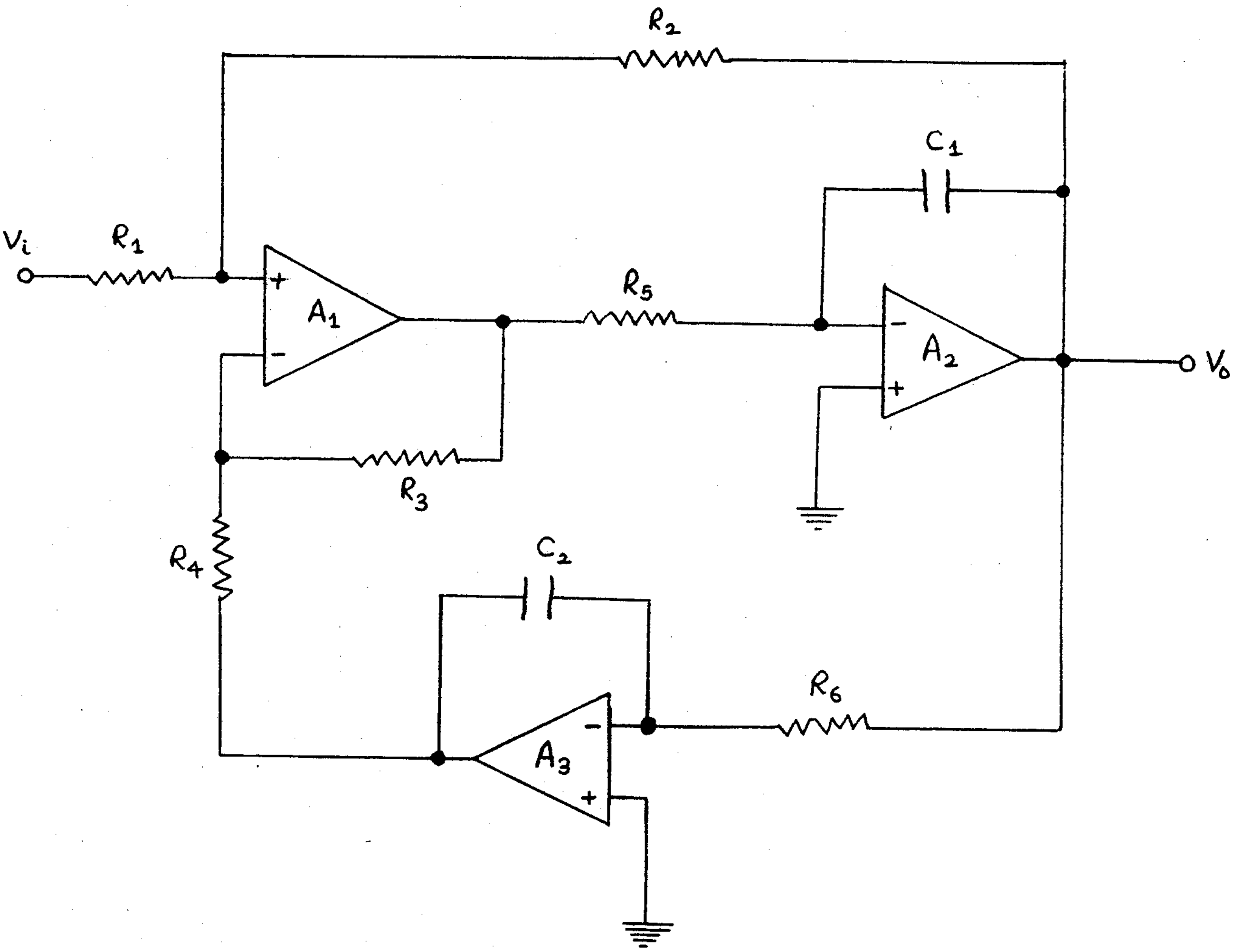
$R_4 = R_5 \geq 10\, V_R / I_R$

$R_1(+)$ or $R_1(-) \geq 10\, V_R / I_R$

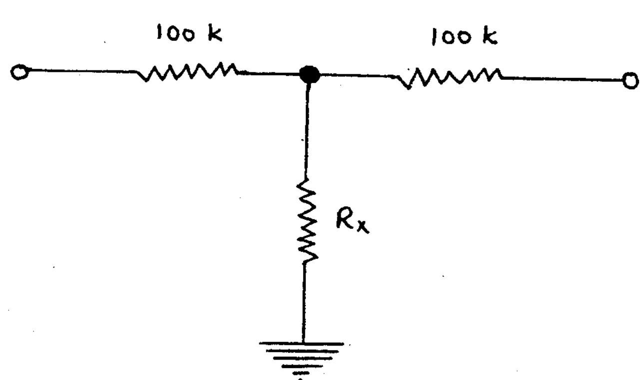
The above two equations prevent excessive loading of V_R .

THE ACTIVE RESONATOR

Fig.15 *The active resonator*



Optional tee network for R_2, R_5, and R_6



DESIGN EQUATIONS AND PRINCIPLE OF OPERATION OF THE ACTIVE RESONATOR

Analog-computer technology provides a very stable band pass filter as shown in fig.15 The circuit requires three operational amplifiers which,preferably,should be in one packing for thermal tracking.This circuit has several advantages over single operational amplifier band pass filters.First,if the components are properly selected,the pass-band central frequency f_o can be made independent of the Quality factor Q of the circuit.Second,the sensitivity of f_o and Q to parameter variations is very low in the state-variable filter.Third,high circuit Q's are possible. ($Q \gg 5$).

The filter is made up of two integrators (A_2 and A_3) and a summing amplifier.The passband center frequency is

$$f_o = \frac{1}{2\pi}[R_3/(R_4R_5R_6C_1C_2)]^{1/2}$$

The circuit Q is

$$Q = \frac{(1+R_2/R_1)\,[(R_3R_5C_1)/(R_4R_6C_2)]^{1/2}}{(1+R_3/R_4)}$$

If we initially set $R_3 = R_4$(fixed resistors) and $C = C_1 = C_2$(fixed capacitors), the above equations reduce to

$$f_o = 1/[2\pi(R_5R_6C_2)]^{1/2}$$

and $$Q = [(1+R_2/R_1)(R_5/R_6)]^{1/2}/2$$

Suppose R_5 and R_6 are equal resistances. In this case,R_5/R_6 will always be unity and Q depends only on R_1 and R_2 If the common value of R_5 and R_6 is R,the equations reduce to

$$f_o = 1/(2\pi RC)$$

$$Q = (R_1 + R_2)/(2R_1)$$

Thus R_5 and R_6 are used to set f_o while R_2 is used for Q adjustment.

The transfer function for the circuit shown in fig. is given by

$$A_{vc} = V_o / V_i = (-sA) / (s^2 + sB + C)$$

where

$$A = \frac{1}{R_5 C_1} \frac{(1 + R_3 / R_4)}{(1 + R_1 / R_2)}$$

$$B = \frac{1}{R_5 C_1} \frac{(1 + R_3 / R_4)}{(1 + R_2 / R_1)}$$

$$C = \frac{R_3}{R_4} \frac{1}{R_5 R_6 C_1 C_2}$$

Also, the circuit gain at $f_o = (R_2 / R_1)$

Radian frequency of passband $= \omega_o = 2\pi f_o$

THE POLARITY SELECTOR

Fig.16 *The inverting precision half wave rectifier*

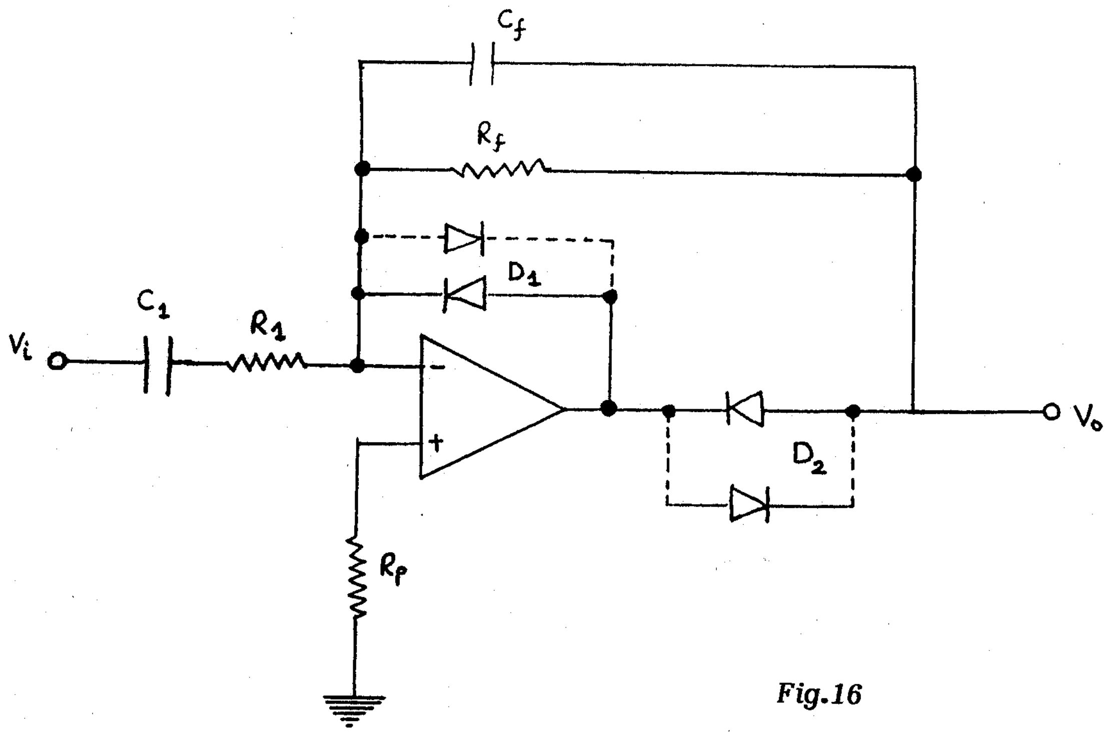


Fig.16

The noninverting precision half-wave rectifier

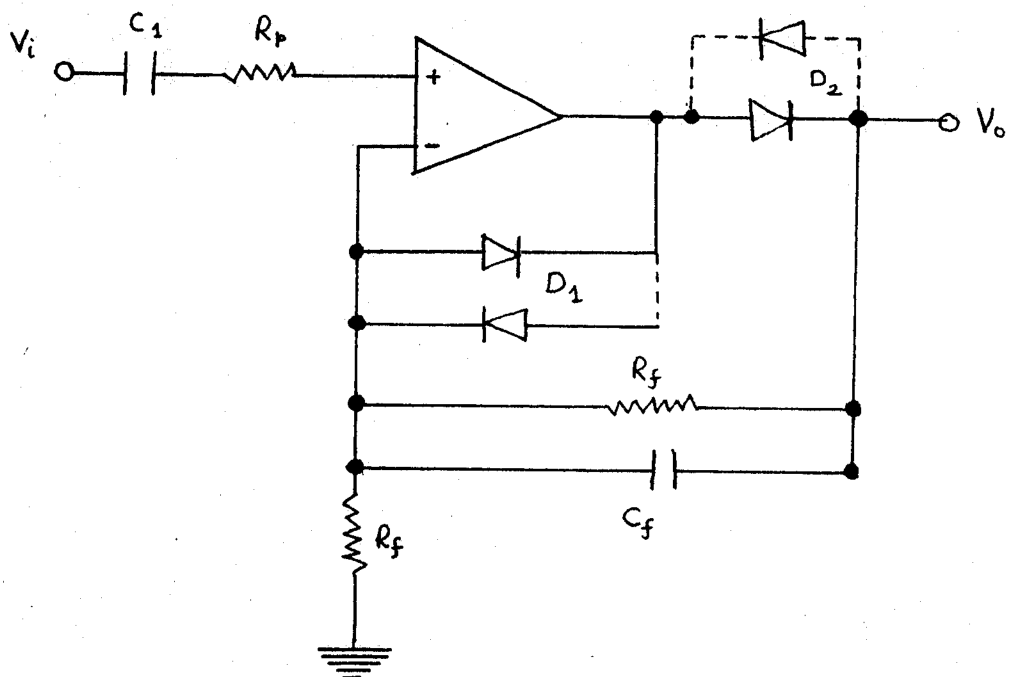


Fig. 17

PRINCIPLE OF OPERATION OF THE POLARITY SELECTOR

The circuit shown in fig.16 comes in four basic conifigurations.Fig. shows pictorially the input-output relationship of these four basic circuits along with a plot of each transfer function.A precision half-wave rectifier performs very closely to the expected response of an ideal diode.Evidently,the ideal diode possesses several advantages relative to the silicon diode.First,the ideal diode can rectify signals down to zero volts amplitude.Second,the forward-conduction region of an ideal diode is linear.

In the inverting half-wave precision rectifier circuit shown in fig.16,the two ideal properties discussed above can be approached with nearly zero error.The circuit will rectify low-level signals with peak voltages of only $0.7/A_v$ *. If* $A_v = 1000$*,precision linear rectification of a 0.7mV signal is possible.*

The circuit operates by providing two gains.For one polarity of input,D_1 *is reverse-biased and diode* D_2 *is forward-biased.Under these conditions the gain of the circuit is* $\pm R_f / R_1$*. (+ for fig.16 and - for fig.17). If the opposite-polarity input is applied,*D_1 *is forward-biased and* D_2 *is reverse-biased.The gain of the circuit then becomes zero.*

Fig.17 shows the circuit configuration for the noninverting precision half-wave rectifier.

Several sources of error are possible in the circuits shown in fig.16 and fig. 17 .If the op amp output offset voltage approaches 0.7V, D_1 *or* D_2 *may begin to conduct.This will add a dc component to* V_o *which may be falsely interpreted as a rectified signal.The portion of this error voltage due to the op amp input offset voltage may be eliminated by adding a coupling capacitor in series with* R_1*. This will cause the dc gain of the circuit to equal unity.Eventhough mid-band ac signals will be amplified by* R_f / R_1*,the input offset voltage will be multiplied by 1.If* R_p *is made equal to* R_f *, a further reduction in the offset is possible by* *cancellation of the effects of each input bias current.*

The capacitor C_f is added to the circuit if a dc output voltage propotional to the peak input voltage is required.

DESIGN EQUATIONS FOR THE POLARITY SELECTOR

1. *The voltage gain of the circuit when solid diode connections are used ; $C_f = 0$: 0 :*

INVERTING

$$A_{vc} = \frac{V_o}{V_i} = -\frac{R_f}{R_i} \quad \text{if} \quad V_i > 0$$

$$A_{vc} = 0 \quad \text{if} \quad V_i < 0$$

NONINVERTING

$$A_{vc} = V_o/V_i = 1 + R_f/R_1 \quad \text{if} \quad V_i > 0$$

$$A_{vc} = 0 \quad \text{if} \quad V_i < 0$$

2. *The voltage gain of the circuit if the dashed diode connections are used ; $C_f = 0$:*

INVERTING

$$A_{vc} = V_o/V_i = -\frac{R_f}{R_1} \quad \text{if} \quad V_i < 0$$

$$A_{vc} = 0 \quad \text{if} \quad V_i > 0$$

NONINVERTING

$$A_{vc} = V_o/V_i = 1 + R_f/R_1 \quad \text{if} \quad V_i < 0$$

$$A_{vc} = 0 \quad \text{if} \quad V_i > 0$$

3. *Input resistance :*

INVERTING : $R_{in} = R_1$

NONINVERTING : $R_{in} = A_v R_{id}$

4. *Size of the filter capacitor C_f :* $\dfrac{1}{2\pi f_c R_f} < C_f < \dfrac{1}{2\pi f_m R_f}$

5. *The magnitude of the dc output voltage when C_f is utilized :*

INVERTING $\quad V_o(dc) = -\dfrac{0.45}{R_i} V_{i(Rms)} R_f$

NONINVERTING : $V_o(dc) = 0.45\ V_{i(RMS)} \left(1 + \frac{R_f}{R_1}\right)$

6. The optimum value for R_p :

If C_1 is used ,

$R_p = R_f$

If C_1 is not used , $R_p = (R_1 R_f) / (R_1 + R_f)$

7. The effective forward voltage drop of the rectifier, 'V_f' :

$V_f = 0.7 / A_V$

where

A_V is the op amp open loop gain

(V_f increases with frequency as A_V drops)

8. Maximum high accuracy frequency of the circuit (error< 10%) 'f_{max}':

$f_{max} = f_u / (100|A_I|)$

where A_I is the op amp dc open loop gain and

f_u is the unity gain frequency of the op amp . (ie the gain cross over frequency)

)

THE PEAK SIGNAL TRACKER

The peak signal tracker

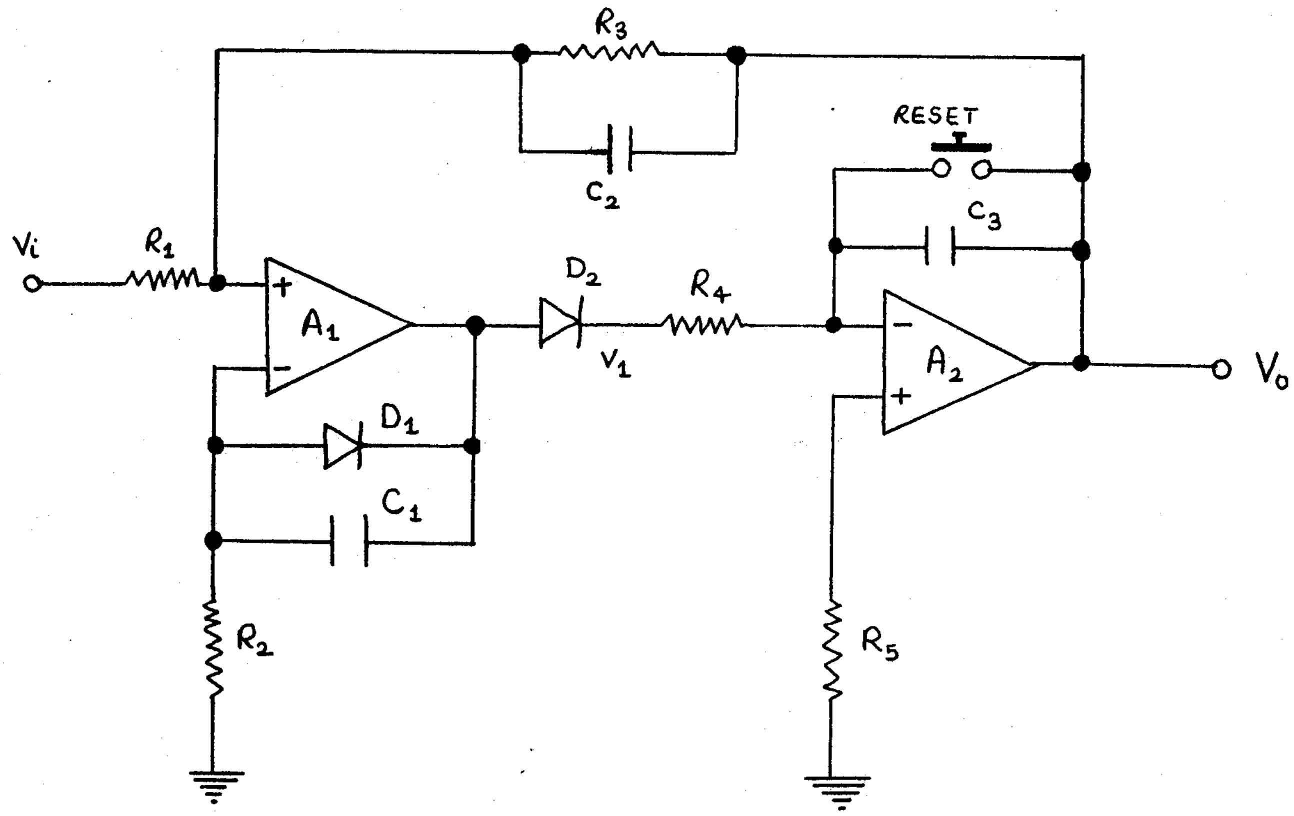


Fig.18

The input and output waveforms of the positive peak detector

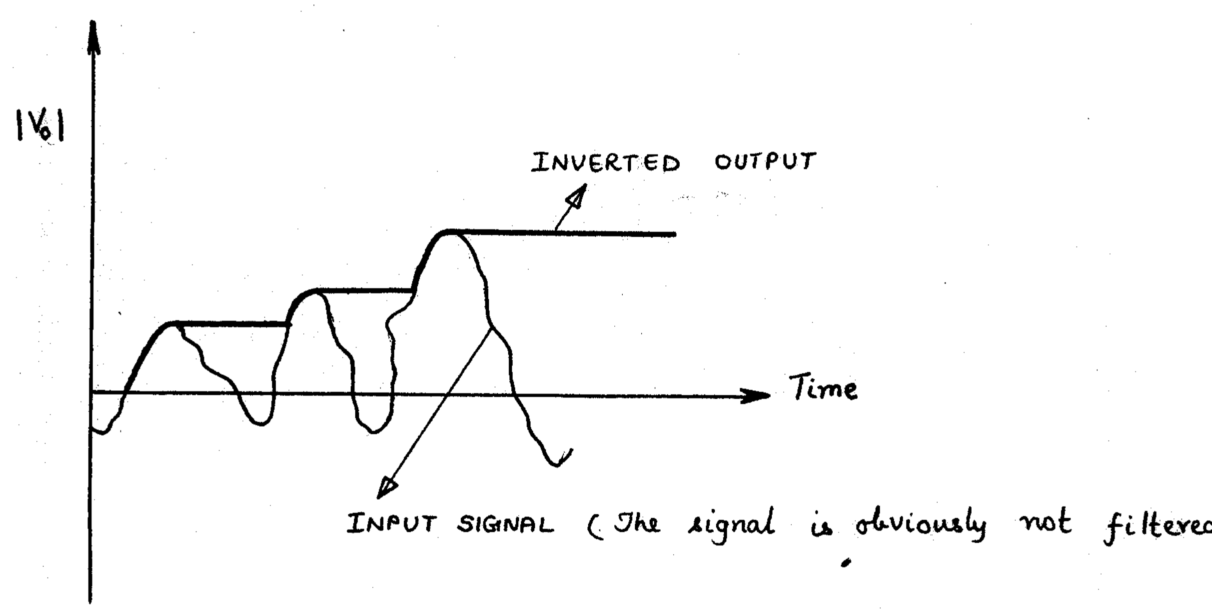


Fig.19

PRINCIPLE OF OPERATION OF THE PEAK-SIGNAL TRACKER

The circuitry of a peak detector can be arranged for positive- or negative- peak detection. For each of these cases, the output can be made positive or negative. The circuit described in fig. 18 selects positive peaks and produces a negative output.

Peak detectors track the input signal and hold the output at the highest peak found since the operation of the reset switch. They continuously compare the input waveform with the stored peak value to determine if the stored value must be updated. This is graphically illustrated in fig. 19 The peak detector is a type of sample/hold circuit. It samples and holds the peak value of the largest peak in a given measurement interval. This is extremely useful in applications where widely spaced transients in a system must be measured.

The circuit of A_1 is similar to the circuit of a polarity selector. However, the feedback resistor R_f and capacitor C_f of fig. ~~19~~ 16 have been replaced by an active feedback network, namely R_3, R_4, R_5, C_2, C_3 and A_2. The circuit A_2 is merely a fast integrator.

The circuit gain is defined as the ratio of peak output voltage to peak input voltage. In terms of circuit components, the gain is

$$A_{vc} = [\ V_o(peak)\ /\ V_i(peak)\ ] = -\ R_3 / R_1$$

After a peak is stored on C_3, the diode D_2 is reverse-biased for all succeeding lower amplitudes. This actually opens the feedback loop. The A_1 ouput will then try to saturate with negative V_i. Diode D_1 prevents this by holding the A_1 output near $-\ 0.7\ V$ if V_i becomes negative.

This circuit may be unstable with some type of op amps because of the large phase shift around the loop. The gain A_{vc} must be critically damped or overdamped to prevent overshoot. An overshoot may be interpreted as a maximum peak. Therefore caution in the feedback design is recommended. C_1 and C_2 are two possible compensation *capacitors.Since the size of these capacitors is critically dependent on the types op amps,an experimental approach has to be carried out.* $C_1 = C_2 = 5pf$ *is used as the starting value and deviations from this value are carried out while all the same observing the overshoot in* V_1 *with* V_i *= a step function.*

If the peak must be stored for long periods of time, A_2 *should be an FET input operational amplifier. C should also be a low-leakage capacitor.The bias current of* A_3 *and the leakage current of* C_3 *will produce a peak hold error of*

$$\Delta V_o = (I \times \text{hold time}) / C_3$$

where

I is the sum of A_2 *input bias current and* C_3 *is the leakage current.*

DESIGN EQUATIONS OF THE PEAK SIGNAL TRACKER

1. *Voltage gain of the circuit :*

This is the ratio of the peak output to the peak input.

$$A_{vc} = [\ V_o(peak) / V_i(peak)\ ]$$
$$= (- R_3 / R_1)$$

2. *Approximate rise time of the integrator :*

(Obviously , the circuit cannot accurately respond to peaks having risetimes faster than this)

$t_r \simeq R_4 C_3$

Slew rate limit of $A_2 = S_2 = I_{o2}(max) / C_3$

3. Optimum value for R_2:

$$R_2 = (R_1 R_3) / (R_1 + R_3)$$

4. Optimum value for R_5:

$$R_5 = R_4$$

5. The error in the stored peak value of V_o due to I_b and I_c :

$$\Delta V_o = \frac{(I_b + I_c)\Delta T}{C_3}$$

BASIC OP-AMP CONFIGURATIONS

This section discusses the basic operational amplifier conifigurations,or building blocks,out of which numerous analog circuits are composed.Most involve feedback to carry out their particular function.The characteristics of these feedback circuits depend primarily on the conifiguration, i.e.,how the circuit is connected,and on the resistors and other passive components.

THE NONINVERTING AMPLIFIER

Perhaps the most important op amp conifiguration is the noninverting amplifier given by fig. 20

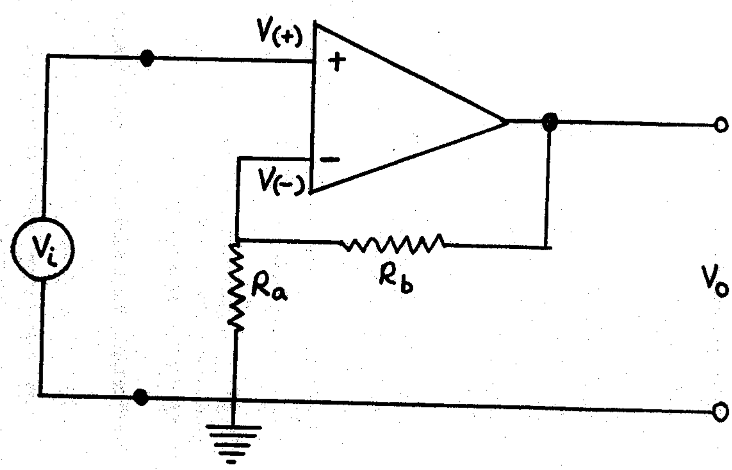


Fig.20

Its purpose is to amplify an input voltage V_i by a factor A to give an output voltage of V_o.A fraction β of the output voltage is fed back to the inverting input of the op amp.Because the sign of the feedback is such as to reduce the magnitude of the output,it is an example of negative feedback.

The closed-loop gain = $A = V_o / V_i$

The voltage at the inverting input is a fraction β of the output.

$V(-) = \beta V_o = (R_a V_o) / (R_a + R_b)$

(The very small current flow into the input has been neglected)

Further, $A = V_o / V_i = 1 / (\beta + 1/A_o)$

For a properly designed amplifier, β is chosen to be atleast 100 times greater than $1 / A_o$. This usually does not present any difficulty as A is typically very high (10^4 to 10^6). To a very good approximation, the gain becomes

$A = V_o / V_i = (R_a + R_b) / R_a$

The above expression is independent of A_o but is true only if A_o is sufficiently large. Where accurate gains are desired, close-tolerence resistors (1 %) are used, and closed loop gains A are limited to 100 or less.

An advantage of the noninverting amplifier is its high input impedence. The minimum gain is unity ($R_a = \infty$, $R_b = 0$), and maximum useful gain about 10^2 to 10^3. The best range of R_b is 2 to 100k .

THE INVERTING AMPLIFIER

A very useful amplifier is shown in fig. 21

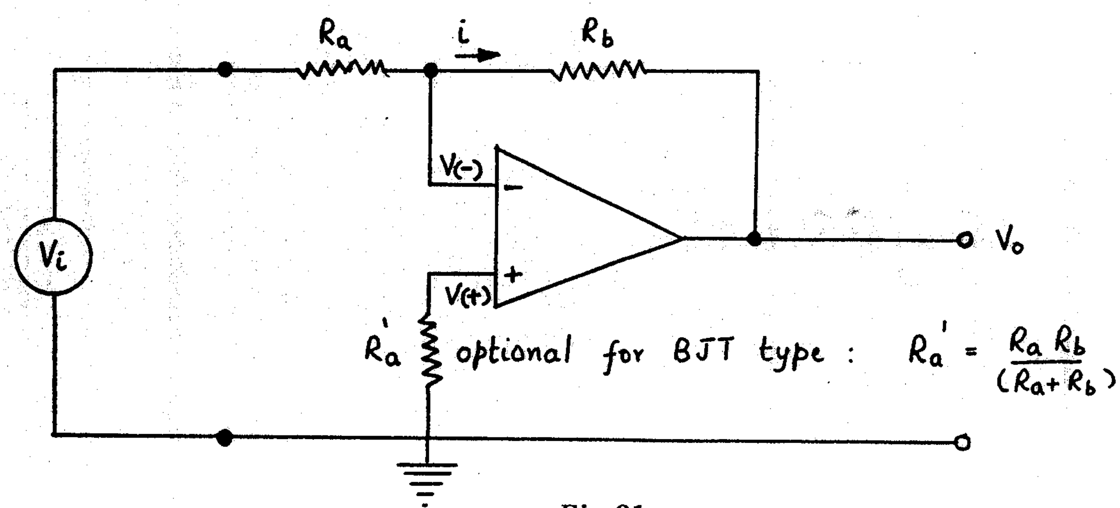


Fig.21

The gain of the above circuit can be easily calculated using the infinite-gain approximation [V(+) = V(-)]. The current i, which flows from the input to the output is

$$i = V_i / R_a = - V_o / R_b$$

Therefore, the closed-loop gain A is given by

$$A = V_o / V_i = -R_b / R_a$$

A disadvantage of the inverting amplifier is its relatively low-input impedence, which is equal to R_a because the inverting input is at virtual ground potential. Input impedence, however, is ordinarily much larger than the op amp output impedence and therefore rarely presents a problem when driven by another op amp. The input resistance R_a must not be too high (over 100K) with BJT-type op-amps or the effect of the bias currents may become too high. If the bias current is a problem, a bias current compensation resistor ($R'_a \simeq R_a$) may be added from the + input to ground. Closed-loop gains A of 0.1 to 100 or 1000 are practical.

THE BUFFER SECTION

The unity-gain amplifier shown in fig.22are specialized versions of the inverting and noninverting amplifiers discussed previously and can be used as buffers.

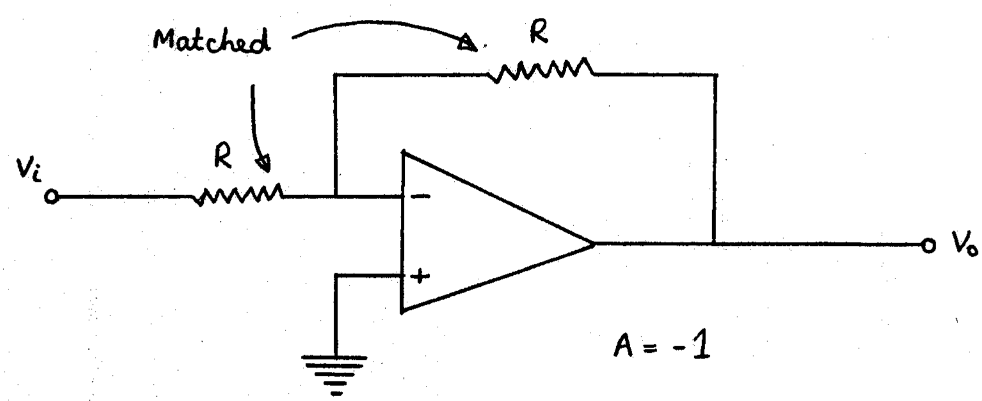


Fig.22

The main use of the noninverting amplifier is as a high-input-impedence buffer which has an ouput impedence low enough to drive subsequent stages.

UNITS OF UNBALANCE

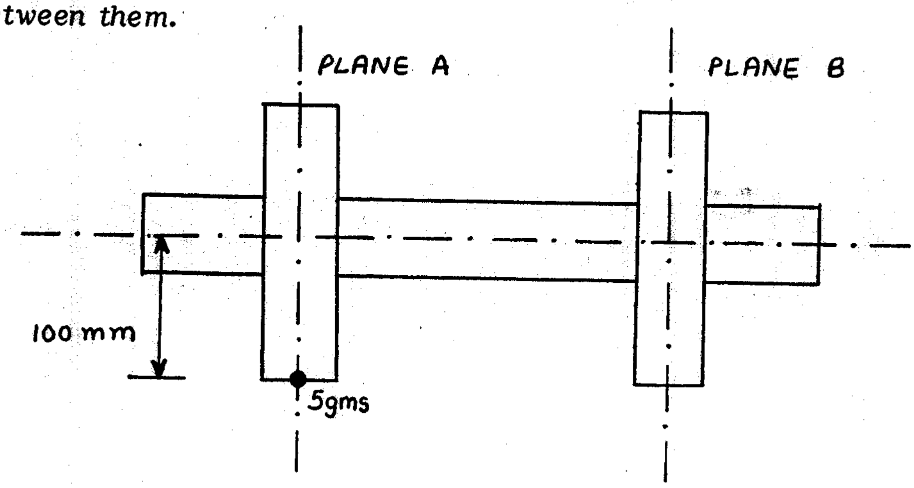
UNITS OF UNBALANCE

Unbalance of any rigid rotor is normally defined in two specified planes. The unbalance of the rotor is the arithmetic sum of the unbalance in the two planes. The unbalance in each plane is the extra mass multiplied by the radius at which it exists.

The unbalance in plane A of the Fig.24 is 5 x 100 = 500 gm.mm. Very often the amount of unbalance may be specified as specific unbalance which is the unbalance per unit weight of the rotor.

$$\text{Specific unbalance (microns)} = \frac{\text{Rotor unbalance (mm.g,)}}{\text{Rotor weight (kg.)}}$$

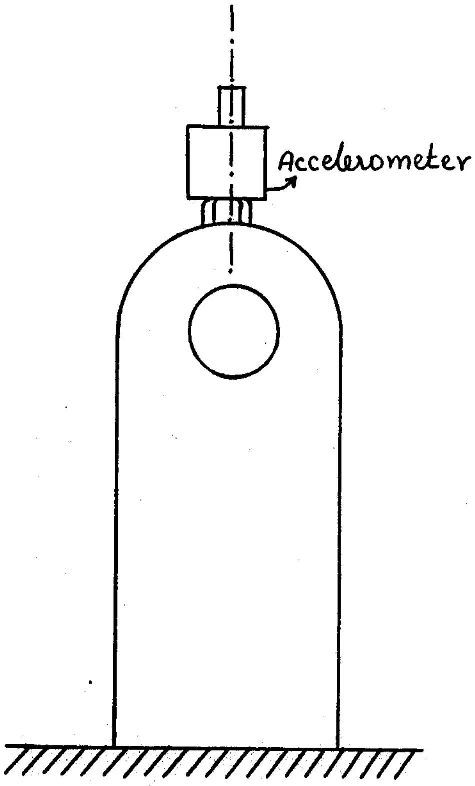
It is important to distinguish between the unbalance in microns to the vibration level in microns. These are two independent things and there is no direct relationship between them.



fig·24

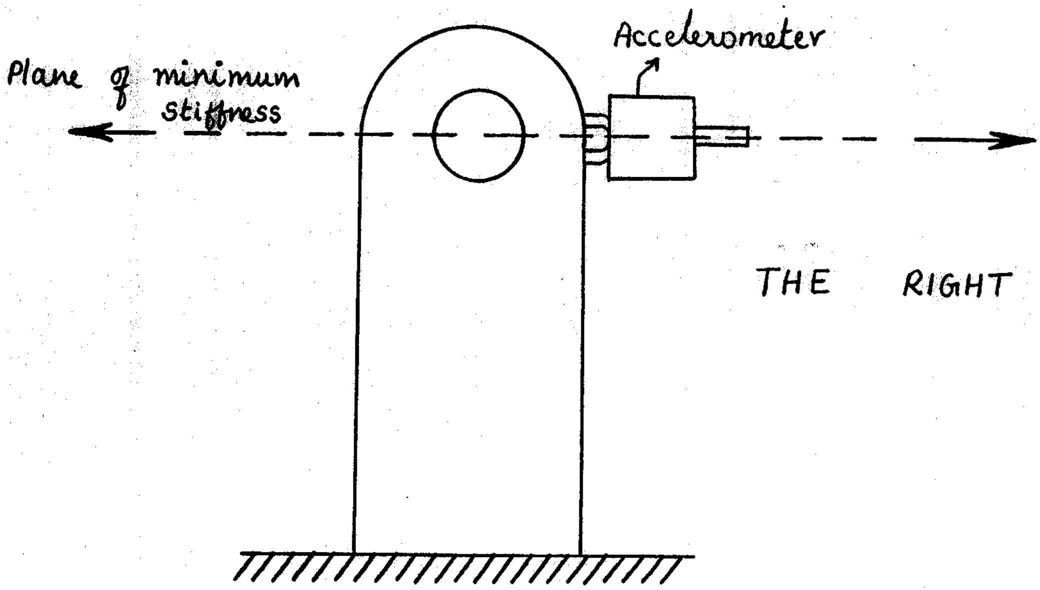
CLASSIFICATION OF RIGID AND FLEXIBLE ROTORS

For the purpose of balancing it is important to make a distinction between rigid and flexible rotors. A rigid motor is a rotor whose unbalance can be corrected in two arbitrary planes and after this correction the unbalance of the rotor will not materially change from zero speed upto its maximum operating speed. On the other hand, the unbalance of a flexible rotor will change materially with speed, unless corrections are made at some specific planes and very often in



A WRONG WAY OF MOUNTING

Fig. 24



THE RIGHT WAY OF MOUNTING

Fig. 24 The transducer is mounted on a plane possessing least rigidity *more than two planes.*

In order to help in laying down the balancing procedures and levels for rotors, ISO-5406 had classified all rotors into 5 classes as given below:

Class 1:

A rotor whose unbalance can be corrected in two (arbitrarily selected) planes so that, after the correction, its unbalance does not change significantly at any speed upto the maximum service speed. Rotors of this type can be corrected by rigid rotor balancing method.

ClaSS II:

A rotor that cannot be considered rigid but that can be balanced using modified rigid rotor balancing techniques.

Class III:

A rotor that cannot be balanced using modified rigid rotor balancing techniques but instead requires the use of high speed balancing methods.

Class IV:

A rotor that could fall into classes 1, 2 or 3 but has, in addition, one or more components that are themselves flexible, or are flexibly attached.

Class V:

A rotor that could fall into class 3 but for some reason, for example economy, is balanced for one speed of operation only.

BALANCE QUALITY FOR RIGID ROTORS

Based on present experience, the International Standards Organisation (ISO) has categorised the normally available rigid rotors into various groups and have suggested the balancing levels for them in the ISO-1940 specifications. Table and Fig 25 that follow, give the level to which a rotor should be balanced.

Care should be taken while applying these standards to actual cases as these standards are only a general indication.

Special considerations may require balancing to much higher accuracy levels.

Practical limitations may also require balancing of an individual component to a lower accuracy level and to control the total unbalance of one assembly by balancing other parts to higher accuracy levels (or to balance a complete assembly).

BALANCE QUALITY CRITERIAN FOR FLEXIBLE ROTORS

In the previous section, the method of specifying the balance quality for rigid rotors was surveyed. This was done using the specific unbalance (mm.g. per unit weight) or rotor. In the case of flexible rotors, this methoid can have little meaning except in a few specific cases. Unfortunately, there is no single universally accepted method for specifying the balance quality criterion for flexible rotors, nor is there any method to work out the acceptable limit. Some of the commonly used methods to specify the balance quality of flexible rotors, are as below:

(1) The maximum permissible level of vibrations may be specified. In this case it is important to specify the conditions under which these vibration levels are measured: for example, a separate rotor on a high speed balancing machine, a separate rotor on a high speed balancing facility where auxillary power is used to run the rotor; an assembled rotor on a test bed; an assembled rotor after its final installation on site etc. In addition, the vibration level will also depend on the type of high speed balancing machine being used; for example - there is the possibility of resonance of balancing machine pedestals in the operating speed range of the rotor. The place of measuring the vibration is also important. We could measure vibrations of the bearing pedestals, the journals, the shaft where it gives maximum vibrations etc. The vibrations can also be measured to two radial directions (horizontl and/or verticle) and in the axial direction.

It must also be remembered that most often the total vibration level for the rotor may be specified and this would be the sum of all possible RMS vibration velocities of different frequencies. We have then to consider what should be the level of vibration due to unbalance alone so that the overall vibration level of the rotor remains within the specified level.

(2) The maximum permissible pedestal forces may be spedified. Here also, the conditions of operation are important as already discussed in the preceding section.

(3) The maximum permissible unbalance may be specified under specified operating conditions. One of the methods is to assume a completely rigid pedestal and work out the unbalance. Under these c onditions, the unbalance is the rotor and the unbalance force on the pedestals is related directly by the following formula:

$$F = U\omega^2$$

where

'U' is the rotor unbalance in the planes of the pedestals. We have to remember the 'U' is also a function of speed in flexible rotors.

Fig. 25

Balance quality grades for various groups of representative rigid rotors.

Balance quality Grade G	ew (1.2) mm/s.	Rotor types - General examples
G 4000	4000	Crankshaft-drives of rigidly mounted slow marine diesel engines with uneven number of cylinders .
G 1600	1600	Crankshaft-drives of rigidly mounted large two cycle engines.
G 630	630	Crankshaft-drives of rigidly mounted fast four cycle engines; crankshaft-drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft-drives of rigidly mounted fast four cylinder diesel engines .
G 100	100	Crankshaft-drives of fast diesel engines with six or more cylinders . Complete engines (gasoline or diesel) for cars, trucks and locomotives .
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts, crankshaft drives of elastically mounted fast four-cycles engines (gasoline or diesel) with six or more cylinders - crankshaft-drives for engines of cars, trucks and locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery, individual components of engines (gasoline or diesel) for cars, trucks and locomotives, crankshaft-drive of engines with six or more cylinders under special requirements.
G 6.3	6.3	Parts of process plant machines. Marine main turbine gears (merchant service) centrifuge drums,, fans, assembled aircraft gas turbine rotors, fly wheels, pump impellers, machine tool and general machinery parts normal electrical armatures, individual components of engines under special requirements.
G 2.5	2.5	Gas and steam turbine including marine main turbines (merchant service), Rigid turbo-generator rotors, turbo-compressors, machine-tool drives, medium and large electrical armatures with special requirements, small electrical armatures, turbine driven pumps.
G 1	1	Tape recorder and gramophone drives, grinding machine drives, small electrical armatures with special requirements.
G 0.4	0.4	Spindles, disks and armatures of precision grinders, Gyroscopes.

FIELD BALANCING

FIELD BALANCING

INTRODUCTION :

Field Balancing is performed on assembled rotor, which tend to go out of balance in service. Examples of such rotors are slurry pump impellers, fans or blowers operating in corrosive or highly laden media, fragile rotors which are liable to damage due to extraordinary conditions (like turbines etc.), or rotors subject to excessive non uniform wear and tear (e.g. centrifuges etc). In all such cases, where the disassembly of the rotor is not feasible, field balancing has to be resorted to. However, for this benefit, the price paid is in terms of time required for balancing and the difficulty level in achieving a satisfactory correction ratio. Balancing with a field balancing set may prove to be more difficult and inconvenient when compared to using a balancing machine, and is therefore almost always directed towards the maintenance applications (and not production).

PRINCIPLE OF FIELD BALANCING

Field balancing is based on the principle of calculating the constants which relates synchronous vibration to unbalance and arriving at correction masses that would reduce the synchronous vibration to zero.

Since vibration is dependent upon the unbalance through a variety of variables: viz. dynamic mass, dynamic stiffness, the sensitivity of the transducer, etc. it is obvious that field balancing can be carried out by measuring the effect (vibration at rotational frequency), by introducing a known cause (a known unbalance, or a trial mass placed temporarily on the rotor). Calculations are made to determine the amount of mass and its position to reduce the resultant vibration to zero.

THEORY OF FIELD BALANCING : THE " INFLUENCE COEFFICIENT METHOD ":

(P.T.P)

Although there are a number of practical ways of calculating the correction mass, like graphical method, arithmetical method etc., the theory behind these methods is essentially the same. The influence Coefficient Method which is described here is easy to understand and universal in its application. It can be used both for one-plane and two-plane balancing. One-plane balancing can be resorted to in most cases, where the diameter of the rotor is large compared to its length.

The method which is widely used is the two-plane balancing for rigid rotors. Let the unbalance on a rotor shaft be defined as U. U is a vector quantity processing magnitude as well as phase, the latter being the angle measured from a fixed point on the rotor.
Similarly let the vibration vector be ***V***.

Field balancing essentially consists of the following steps:

1. *Choose 2 measurement planes at appropriate positions. The planes so chosen should be as close to the rotor bearing housing as possible and the transducers should be attached in the plane of minimum stiffness. Let these planes be designated as planes 1 and 3.*
2. *Choose 2 arbitrary correction planes, depending upon the convenience of adding correction and/or trial masses, which may for convenience be done at the same radius (Else, scale up or down, keeping mass x radium a constant).*
3. *Measure $\underline{V}10$ and $\underline{V}20$ these being the initial vibration levels on plane 1 and 2 respectively.*
4. *Mount a trial weight M (Vector, possessing magnitude = mass, and phase = angular position) in correction plane 1. The mass may be so chosen as to achieve a significant influence in the vibration levels. (Usually a good thumb rule is to*

select a weight about 5 - 10 times the ultimate unbalance that can be tolerated). Measure the vibration vectors V_{11} and V_{21} being the vectors in planes 1 and 2 caused by the placing of a trial weight in the correction plane 1.

5. Remove the trial weight from plane 1 and place it on the correction plane 2.

Measure the resulting vibration vectors V_{12} and V_{22} being the vibration vectors in planes 1 and 2 caused by the placing of trial weight M_2 in correction plane 2.

6. Calculate the unbalance as shown below and then correct.

The assumption of a linear system is made here, which implies that the response (vibration is directly proportional to unbalance, a situation which is almost universally applicable. Thus, in vector form,

$$\overline{V} = \overline{K} \cdot \overline{U}$$

Since we are operating in 2 planes, we can convert this equation into a matrix equation i.e.:

$$\begin{vmatrix} V_{10} \\ V_{20} \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix} \begin{vmatrix} U_1 \\ U_2 \end{vmatrix} \quad \text{— (1)}$$

Which essentially relates the vibration in plane 1 to unbalance at the 2 correction planes through constants of proportionality K_{ij} which are known as the influence coefficients. Thus for a given set up, these coefficients will remain constant for a given set of conditions. Obviously these depend on speed, stiffness and dynamic mass of the system, but we need not go into that here.

Now these constants can be determined on the basis of experimental results:

$$\begin{vmatrix} V_{11} \\ V_{21} \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix} \begin{vmatrix} U_1 + M_1 \\ U_2 \end{vmatrix} \quad - (2)$$

Substracting (1) from (2), we get

$$K_{11} = \frac{(V_{11} - V_{10})}{M_1}$$

and

$$K_{21} = \frac{(V_{21} - V_{20})}{M_1}$$

Similarly we get:

$$K_{22} = \frac{V_{22} - V_{20}}{M_2}$$

and

$$K_{12} = \frac{V_{12} - V_{10}}{M_2}$$

Now that we know the matrix K we can calculate the unbalance U by simply inverting the K matrix:

$$\overline{V} = \overline{K} \cdot \overline{U} \quad \text{and} \quad \overline{U} = \overline{K}^{-1} \cdot \overline{V}^{-1}$$

$$\begin{vmatrix} U_1 \\ U_2 \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix}^{-1} \begin{vmatrix} V_{10} \\ V_{20} \end{vmatrix}$$

It is thus possible to calculate the K^{-1} coefficients only once for a given machine, and keep them in record, so that these may be used for any later run, provided all the runs are carried out at the same rotational frequency. It will be only necessary to measure the vibration levels in the 2 planes and compute correction masses using the K^{-1} constants by simple multiplication. It must be remembered that K's are complex numbers (or vectors).

LIMITATIONS OF FIELD BALANCING

Although field balancing can be performed in a majority of real cases, there may be rare cases where it fails to yield a satisfactory correction ratio. This may happen if any of the following conditions exist:

INTERFERING VIBRATIONS

These may often be systematically related to rotational frequency and it may not be possible to eliminate all interfering frequencies by filters, especially if these lie close to the rotational frequency, (e.g. in case of pulleys, gears etc.)

However, if high frequencies are predominant, better results can be obtained by measuring displacement rather than velocity.

RESONANCE

If balancing is performed at speeds which cause resonance in some elements i.e. either the bearing housing members or the transducer itself, the resulting phase could falsify results. Theoretically speaking, these should not affect the results if the speed is absolutely constant because all phase readings will be correspondingly shifted, but in real situations, there are minor speed vibrations which make it difficult to measure the phase accurately.

LOOSENESS OF PARTS

If there are loose pulleys, gears, pinions, or any other parts on the rotors, no amount of balancing will help, as the rocking of the loose parts will drastically change the unbalance.

HIGHLY RIGID MEMBERS AND LOW UNBALANCE VALUES

Practical limitations may enter vibration measurement if the entire structure is too rigid to cause a measurable vibration due to a given unbalance. A simple test may indicate such a situation: place a trial weight equal to the desired unbalance *on the rotor. If the vibration readings change in a measurable manner, it should be possible to balance the rotor.*

PHASE INSTABILITY

Phase instability may occur because of the use of materials like rubber etc. in flexible couplings which allow relative movement. In such cases a visual average of the phase position may be taken.

NON LINEARITY

The assumption of linearity of the system may not hold good. This may be so if balancing is attempted at close to rotor-critical speeds. If such conditions occur, the balancing should be performed at a different speed or the bearing mounting should be stiffened or loosened temporarily. Non-linearity may be unavoidable due to non-linear rigidity of foundations and mounting materials, bearing clearances,, hydrodynamic bearing non-linearities etc. This lowers the unbalance correction ratio.

FIELD BALANCING SETS

These are commercially available sets which consist of some method of measuring the vibration amptitude and phase with reasonable level of sensitivity. Filters are built-in to eliminate undesirable frequency components. The sensitivity of the pickup is not critically important per se if the same pickup is used to measure the vibration in either plane both at the time of determination of K constants and subsequent correction ratio.

In fact, for very heavy and rigid machines, special high sensitivity pickups may be used to increase accuracy of measurement. However, calibrated pickups are desirable if the residual vibration is to be measured for determining acceptable levels.

The phase measurement system broadly comprises of the following methods:

BY A STROBOSCOPIC LAMP

The rotor is lit once per revolution (or more for high frequencies) at the zero crossing point of the vibration signal. Thus the rotor appears stationary at a specific phase angle which can be measured musing numbered tapes stuck on the rotor periphery. The method is not very accurate since it involves reading the angle off the rotor.

BY AN ELCTRONIC REFERENCE PICKUP

These involve electronically picking up a zero reference mark on the rotor by an optical, intra-red or an inductive proximity pick-up. Angle is usually indicated either on a digital display or an analogue meter. Such systems are usually easier to use and more reliable than the stroboscopic method. However stroboscopic methods may be necessary if certain areas of the rotor are inaccessible for mounting the electronic reference pickups.

COMMENT

The prime motive in choosing " THE DESIGN AND FABRICATION OF A DYNAMIC BALANCING MACHINE " as our project work has been our itch to gain an exposure to the practical difficulties associated with hardware design and testing. Though we have not been fully successful with regards to the proposed result, we have acquired sufficient technical knowhow on the aspect of balancing. To be precise, we have essentially been a group of " researchers " and " experimenters ", rather than " manufacturers". All the same, this project has forced us to explore in depth the principle of operation of various popular,important and conventional electronic circuit configurations. We strongly feel that our project can easily be a trend-setter for a new type of balancing machine.

We have discussed in detail (design,schematic layout and principle of operation), various sections of a machine we had hoped to construct. A little more impetus on our presently concluded work could very much result in a very efficient dynamic balancer, which could obviously be commercially exploited.

In this project report,practically everything concerned with dynamic balancing (including field balancing and balancing standards) has been elaborately discussed. We sincerely hope that our project work would be carried through to a future batch of enterprising and hardworking "researchers" who, we're sure would convert our " research " produce into a commercial reality.

REFERENCES

1. *ISO-1940 / VDI-2060 : Balance quality of rotating rigid bodies.*

2. *ISO-5406 : The mechanical balancing of flexible rotors.*

3. *" Applications Manual for Computing Amplifiers "*

   - George A.Philbrick Researches, Inc.

4. *" Operational Amplifiers - Design and Applications "*

   - Tobey,J.G.Graeme and L.P.Huelsman

5. *" Modern Operational Circuit Design "*

   - J.J.Smith

6. *" AC - to - DC Converters for Low Level Input Signals "*

   - R.Kreeger

7. *" Build a Dual Voltage Regulator For $11 "-*

   - H.T.Jones , Electron.Des

8. *Circuit Design for Electronic Instrumentation*

   - Darold Wobschall