DESIGN & FABRICATION OF SLIDING VANE ROTARY COMPRESSOR

PROJECT REPORT

P-125

SUBMITTED IN PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE AWARD OF DEGREE OF

BACHELOR OF MECHANICAL ENGINEERING

OF THE BHARATHIYAR UNIVERSITY
COIMBATORE 641 046

SUBMITTED BY

B. SRIKUMAR
K. SIVAKUMAR
R. CHANDRASEKAR
K. ANANDARAJ

UNDER THE GUIDANCE OF

Dr.ALI UDDIN ANSARI M.E., Phd., (U.S.A.)

1990 - 1991

DEPARTMENT OF MECHANICAL ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY
COIMBATORE 641 006

Department of Mechanical Engineering
Kumaraguru College of Technology
COIMBATORE - 641 006

CERTIFICATE

This is to certify that the report entitled

"DESIGN AND FABRICATION OF SLIDING VANE ROTARY COMPRESSOR"

has been submitted by

Mr. ....

in partial fulfilment for the award of Bachelor of Engineering in the MECHANICAL ENGINEERING branch of the BHARATHIAR UNIVERSITY, COIMBATORE during the academic year 1990-91.

Guide **Head of the Dept.**

Certified that the candidate was examined by us in the Project work viva-voce examination held on ____________ and the University register number was ____________

Internal Examiner **External Examiner**

ACKNOWLEDGEMENT

We have great pleasure in expressing our gratitude to Prof.R.Palanivelu, Principal for providing all the facilities to complete this project.

We wish to express our profound gratitude to our guide Dr.Ali Uddin Ansari, Head of the Mechanical Engineering Department who has been a source of inspiration and encouragement.

We wish to express our special thanks to Mr.R.Selvaraj, Design Engineer, Shanti gears for giving us valuable ideas and information and who helped us in the timely completion of our project.

We also wish to thank our staff Mr. V.Gunaraj M.E., and Mr. P.Suresh Prabhu M.E., for their guidance and encouragement.

Lastly we thank all those who directly or indirectly helped us in completing our project.

SYNOPSIS

This project deals with he design and fabrication of Rotary Sliding Vane Compressor.

This slide Vane Compressor is a rotary compressor which works on a sliding vane principle. The main advantages of the compressor are:

Single rotor
Direct Drive
Low Noise level
Integral and compact design.

The compressor was designed for a max working pressure of 3 bar and free air delivered of 22 lit/min. Various other components were designed and a suitable drive was selected.

Fabrication work was carried out according to the design and various performance tests were carried out in the Heat Power Lab at Kumaraguru College of Technology.

The slide vane compressor is suitable for large discharge and air delivery is continuous. It can be used to draw gases or vapours even below atmospheric pressure.

CONTENTS

GENERAL INFORMATION

1.1. INTRODUCTION:

A compressor is a device used to increase the pressure of a compressible fluid.

The inlet pressure level can be any value from a deep vacuum to a high positive pressure. The discharge pressure can range from subatomospheric levels to high values. The inlet and outlet pressure are related corresponding to the type of compressor and its configuration. The fluid can be any compressible fluid, either gas or vapour and can have a wide molecular weight range.

A few typical Applications

1. Air separation
2. Vapour Extraction
3. Refrigeration
4. Steam recompression etc.

1.2 COMMON FEATURES:

Rotary compressors are classified as positive displacement machines. This group of compressors has several features in common despite differences in construction. They are similar in operation to rotary pump. Probably the most important feature is the lack of valves as used on the reciprocating compressor.

The rotary is light in weight than the reciprocator and does not exhibit the shaking forces of reciprocating compressor making the foundation requirements less rigorous. Even though rotary compressors are relatively simple in construction, the physical design can vary widely. Both multiple rotor and single rotor construction is found.

Rotor design is one of the main items that distinguish the different types, size and operating range is another area unique to each type of rotary.

1.3. SLIDING VANE COMPRESSOR

Sliding vane compressors can be used to draw gases or vapour from spaces under pressure below atmospheric. Such a compressor is referred to as a vacuum pump. Sliding vane vacuum pumps create a 95% vacuum.

1.3.1. THE HEART OF THE PACKAGE:

The major moving part is a single slotted rotor which is held eccentric to a cylindrical bore. When the rotor is driven extending vanes trap pockets of air and the reducing volumes created by the eccentricity cause the air to compress.

The air enters inside the compressor through an air filter which filters dust and other impurities present in air.

"Simplicity" is the key to reliability and long life. That's an engineering fact. And an examination of any fluid air compressor shows how thoroughly we have absorbed its implications.

For instance there is no oil pump, no high speeds, less stress, lower temperature. And you will experience none of the 'usual' problems caused by cylinder head valves, pistons, rings, bearings as there are none.

1.4. GENERAL OUTLINE

The sliding vane compressor consists of a single rotor mounted in a cylinder slightly larger than the rotor. The eccentricity is maintained in the rotor end cover so that while in operation the rotor rotates eccentrically inside the cylindrical housing.

The rotor has a series of radial slots which hold a set of vanes. The vanes are free to move radially within the rotor slots. They maintain contact with the cylinder wall by centrifugal force generated as the rotor rotates.

1.5. FEATURES:

1.5.1. Single Rotor:

Minimum 50,000 running hours bearing life, pulsation free output and low noise level.

1.5.2. Direct drive:

High power conversion rate minimises wear of moving parts.

1.5.3. Integral design:

All major parts are safely enclosed and tamper free. All maintainable items are easily accessible.

1.5.4. Compact design:

Easily installed and saves space.

1.5.5. Sliding vanes:

The vanes are made of Tefflon, graphite, hylam or gun metal strips. Their life and cylinder wear are sensitive to the quantity and quality of the oil applied.

WORKING PRINCIPLE OF ROTARY COMPRESSOR

Rotary compressors are classified as positive- displacement machines they are similar in operation to rotary pumps.

Most common are rotary sliding-vane compressors. The basic design of a sliding vane compressor is as; As rotor mounted eccentrically in casing rotates, the gas is trapped in closed spaces formed by vanes and conducted from the suction to the discharge chamber. With this compressor design characterized by adequate balancing of moving masses, a high rotative speed can beimparted to the rotor of the machine directly connected to an electric motor.

A running sliding vane compressor gives off large amount of heat due to mechanical friction. Therefore, at compression ratios above 1.5 the compressor casing is water-cooled.

Sliding vane compressors can be used to draw gases or vapours from spaces under pressures below atmospheric. Such a compressor is referred to as a vacuum pump. Sliding vane vacuum pumps create a 95% vacuum.

The capacity of a sliding vane compressor depends on its geometry and rotative speed. Assuming that the vanes are radial type and the volume of gas contained between two vanes will be $V = fl$ (where f = maximum cross sectional area of space between vanes l = vane length.

We can assume as an approximation that

$$df = \frac{rd\theta+(r+2e)d\theta}{2}\, 2e = 2e(r+e)d\theta$$

Hence,

$$f = \int_0^{\beta/2} 4e(r+e)d\theta = 2e(r+e)\beta$$

Since $r+e=R$ and $\beta=2\pi/z$ (where z=number of vanes),

$$f=\frac{4\pi er}{z}$$

According to fig. the volume of gas between vanes is

$$V=2eRl=\frac{4\pi eR}{z}\, l$$

During one revolution of compressor shaft each space between vanes is filled with gas once, and therefore the actual capacity of compressor is

$$Q=2eRl\beta z\eta\,\lambda_{vol} \qquad = 4\pi eRl\eta\lambda_{vol}$$

where λ_{vol} = capacity coefficient, in the range from 0.5 to 0.8.

The capacity coefficient depends on the internal leakage of gas through radial and axial clearances, the thickness and number of rotor vanes.

14.2 POWER AND EFFICIENCY

The power of a water-cooled rotary compressor stage is calculated from the isothermal work.

$$N=\frac{N_{iso}}{\eta_{iso}\eta_m} = \frac{P_1 Q_1 \ln\varepsilon}{1{,}000\,\eta_{iso}\eta_m}$$

Where P1 = Initial Pressure

Q1 = Capacity under suction conditions

For compressors with high rate air cooling

$$N = \frac{Na}{1000\,\eta_a \eta_m}$$

For an adiabatic process, power is determined from the formula.

$$Na = \frac{k}{k-1} P1Q1 \left[\left(\frac{P2}{P1} \right)^{\frac{k-1}{k}} - 1 \right]$$

For sliding-vane compressors, the products of efficiencies and are in the range from 0.5 to 0.6 to 0.7 respectively.

14.3 CAPACITY REGULATION OF ROTARY COMPRESSORS

As can be seen from the rotary compressor capacity equation, capacity is proportional to the rotative speed of compressor shaft. Hence Q can be adjusted by varying n.

Generally, sliding vane compressors are directly connected to electric motors running at 1,450; 960; 735 rpm. Here, capacity regulation requires that a variable speed gear be interposed between the shafts of motor and compressor.

The methods of capacity regulation used for rotary compressor of both types are suction throttling, recycling the compressed gas into the suction line, and start-stop opertion.

14.4 ROTARY COMPRESSOR DESIGNS

Sliding vane compressos are built for capacities upto 500 m3/min and capable of generating pressues upto 1.5 MPa by two stage compression with intercooling.

To reduce fricional forces in the rotor slots, the vanes are tilted 7 to 10° from the radial position in the direction of rotation. As a result the direction of force applied to the vanes by the casing and relief rings becomes neaer to the direction of vane movment in th slots and the frictional force decreases.

To reduce gas leakage through axial clearances, the rotor hub accomodates sealing rings that are held by springs against the head faces.

A spring-loaded stuffing box is installed at the point where the shaft extends through the head.

The construction employs roller bearings. Lubrication is done with medium-viscosity engine oil through sight-feed lubricators. The parts subject to lubrication are the relief rings, end-face sealing rings, and stuffing box.

MECHANICAL CONSTRUCTION

2.1 MECHANICAL CONSTRUCTION

The housing is generally constructed of cast iron. The hose is machined and brought to a good finish to reduce the vane sliding friction. The inlet connection is screwed to the housing-end-cover. The rotor and shaft extension may also be machined from a single piece of bar. The material is carbon steel for single piece models. The larger compressor, using the two piece rotor arrangements use carbon steel for the shaft and cast iron for the rotor body.

The rotor body is keyed to the shaft by means of a paralled key vanes attached to the rotor body by means of milled slots. For a non-lubricated design carbon or hylam is used. The vane number influence the differential pressure between adjacent vane cells. This influence becomes less as the number of vanes increases.

Roller bearings or ball bearings are generally widely used. Ralls are either a packing or mechanical contact type. Since this is of a non-lubricated dry type compressor lu-bricnt should be applied to the vanes frequently so as to reduce the wear of the vanes due to sliding friction. Vane wear should be monitored in order to schedule replacement before the vanes become too short and wear the rotor slots.

If the vanes are permitted to become too worn on the sides or too short, the vane may break and wedge between the rotor and the cylinder wall at the point of eccentricity possibily breaking the cylinder. Shear pin couplings ae some times used to prevent damage from a broken vane under sudden stall conditions. The bearing are seated in the housing end cover.

2.3. DISTINGUISHABLE FEATURES FROM RECIPROCATING COMPRESSORS:

1. The reciprocting compressors reciprocating motion is transmitted to a piston which is free to move in a cylinder. The displacing action of the pistion together with the inlet valve causes a quantity of gas to enter the cylinder where it is in turn compressed and discharged.

   In the sliding vane compressor as rotor mounted eccentrically in the casing rotates the gas is trapped in closed spaces formed by vanes and conducted from the suction to the discharge chamber.

2. The sliding vane compressor performs the compression in an intermittent mode which is not in the case of a reciprocating compressor.

3. The sliding vane compresor do not use inlet and discharge valves which are present in the reciprocating compressor.

4. The sliding vane compressor gives out continuous flow of air which is not possible in the reciprocating compressor.

5. The pressure range of a reciprocating compressor is the boardest in the compressor family on tending from vacuum to 300 bar. The sliding vane compressor can be used to 3.5 bar in single stage form and when staged can be used to 8 bar.

6. Rotary compressors are suitable for large discharge at low discharge pressure reciprocating compressors are suitable for low discharge at very high discharge pressures.

7. There is no balancing problem in rotary compressor where as it is a major problem in reciprocating compressor.

8. Air delivered by rotary compressors is more clean as it does not come in contact with lubricating oil.

9. Lubrictaing system is more complicated in reciprocating compressor which is very simple in rotary compressor.

2.4 ADVANTAGES OF SLIDING VANE COMPRESSOR

1. The most important feature is the absence of valves used on the reciprocating compressor.

2. The rotary is lighter in weight than the reciprocator and hence can be handled manually.

3. It does not exhibit the shaking forces of the reciprocating compressor making the foundation requirements less rigorous.

4. You will experience none of the usual problems caused by cylinder head valves, piston, rings and bearings.

5. Continuous air low is possible which is absent in the reciprocting compressor.

6. It does not need storage tanks to store compressed air as the flow is continuous.

7. It is more compact and has an integrated design.

8. Simplicity is the key to reliability and long life. That's an engineering fact which is in the care of a sliding vane compressor as it is simple in construction.

2.5. APPLICATIONS:

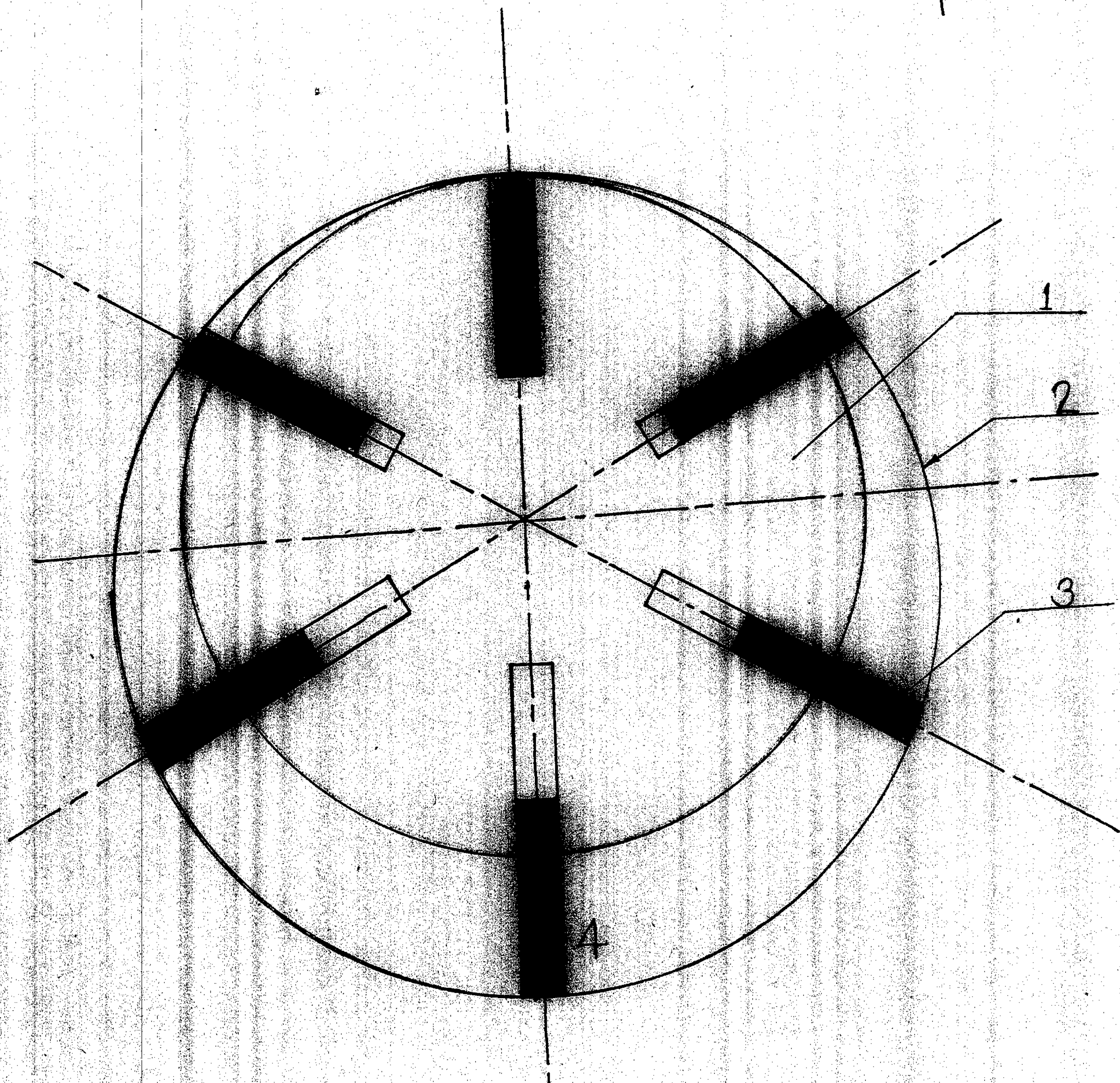
1. An often over looked application for the sliding vane compressor is that of vacuum service where in single stage form, it can be used to 0.90 bar.

2. The sliding vane compressor is used in gas gathering and gas boosting applications in direct competition with the reciprocting compressor.

3. The compressor is also widely as a vapour extraction machine in a wide variety of applications which include steam turbine condenser service for air extraction.

4. Sliding vane compressor can be used to draw gases or vapours from spaces under pressures below atmospheric.

1. Rotor
2. Casing
3. Vane
4. Enclosed spac[e]



DESIGN FACTORS TO BE CONSIDERED FOR ROTARY COMPRESSORS

Rotary compressors are normally suitable for only low compression ratios(as two stage machines, compressing fom 1 to 11 ata at the most) and for small and medium amounts of free gas delivered. They have no reciprocating masses, permitting high rotational speeds and are consequently of small dimensions. Since they do not cause any mechanical vibrations that present little difficulty from the point of view of foundation material, but they may be the source of an objectionable noise of high frequency. They have no suction valves and, usually no discharge valves, and possess small clearance volume. On the other hand, friction and leakage losses are greater, their parts comparision with turbo-compressos, they have the advantage of high efficiencies when pumping small quantities of gas.

According to their design we distinguish the following type:

1. Vane (or blade) compressors (multi-vane)
2. Rotating eccentric type(or Pendulum type)Compressors (single vane)
3. Liquid piston type compressos.
4. Double-impeller compressors.

ROTARY VANE COMPRESSORS

These machines(also termed sliding vane compressors) have the rotor seated eccentrically in such a way that it almost contacts the cylinder wall. Deep slots are milled into the rotor. mostly radial but sometimes oblique in which move vanes(plates)subdividing the crescent shaped space between the rotor and the cylinder into a number of compartments. When the rotor is running, centrifugal force drives the vanes out to the cylinder wall. Thus the action is that initially the space between the rotor, the cylinder and two adjacent vanes is enlarged, and the gas is drawn into this space. When the position opposite to the eccentricity of the rotor is reached, suction has finished, and during further rotation the enclosed space decreases, compression of the gas occurs (as in reciprocating compressors) and finally it discharges. These compressors exhibit a very uniform torque, so that a flywheel can be dispensed with. They've a low weight.

Advanced types are equipped with bronze or cast iron rings of a somewhat smaller diameter than that of the cylinder these rings rotate together with the piston, taking up the centrifugal force of the vanes. A small clearance remains between the vane and the cylinder. Otherwise the wear of the vanes on the friction surfaces would be considerable.

In order to prevent leakage of the compressed gas from delivery to suction along the retaining rings, they've small grooves(15 to 30) on their outer periphery. Small plates of insignificant mass are inserted. These produce a very small centrifugal force resulting in little wear. The peripheral grooves and the opening in them, facilitate the supply of oil to the inner surface of the rings.

Even when the centrifugal force of the vanes is taken up by the rings, the vane wear is very high, as are the friction and leakage losses. The effect of centrifugal force can be eliminated,however, in the system suggested by Kozousk(52) combining two opposite vanes into one Unit, the cylinder and the rotor with the through slots. Opposite vanes are combined into double-vanes. The outer sliding edges are enlarged and bear against the bars which have a segment shaped cross section. These rotate concentrically with the cylinder at the same angular velocity as the rotor. During rotation the double-vane slides in relation to its guide remaining perpendicular to it. This friction path for one revolution is equal to four times the eccentricity consequently it is very small. If separate double vanes are to be used, one or two double-vanes are usually employed. In the patent application by Kozousek there are also descriptions of methods of balancing the centrifugal and the Coriolis accelerating forces by linking the vanes. Vane Compressors constructed in this way are balanced and are thus suited for very high rotational speeds.

NUMBER, MATERIAL AND CONSTRUCTION OF VANES:

It is apparent that the utilization of the capacity of the cylinder improves as the number of vanes increases. With a large number of vanes, however the volume occupied by the vanes must be considered.

Where as in a single-vane compressor the crescent shaped space between cylinder and rotor is filled once in each revolution, in a double vane machine a space only little smaller than that of the single-vane compressor is filled twice in each revolution.

In a four-vane compressor the space between two vanes is further reduced in comparison with the double-vane machine, but the number of gas intakes is doubled, again increasing the volume pumped. As the number of vanes continues to increase the quantity pumped further increases, but he relative gain in this respect steadily diminishes.

A further advantage of having a large number of vanes is that the pressure difference between adjacent spaces is smaller resulting in smaller leakage losses. Thus, to cater for larger pressure differences, as for example in refrigerating ammonia compressors, the rotor is usually equipped with 20 to 30 vanes.

The vanes are mostly made of steel. If they are made of synthetic fibrous materials, no retaining rings are employed. If a large number of steel vanes are used, their thickness is usually 1 to 3 mm. Machines which have to supply oil-free compressed air are usually fitted with carbon vanes. The clearance of the vanes in the slots is only 0.01 to 0.04mm. depending upon he thickness of the vanes, which varies from 3 to 10 mm. The thickness depends upon the pressure difference and the radial width of the vanes. The surfaces of the carbon vanes in contact with the cylinder are rounded with a radius equal to or slightly smaller than the radius of the cylinder. Cylinder and rotor should be made of corrosion-resistant metal as hard as possible, with surfaces contacting carbon finely ground, and the interior of the cylinder honed.

The coefficient of friction between carbon and polished hard metal is about 0.10.

As opposed to reciprocating compressors, vane compressors can be made with no clearance volume. This is effected by grinding the cylinder, along the side of the eccentricity, to the same radius as that of the rotor and adjusting the eccentricity so that the rotor nearly contacts the cylinder in this region. This arrangement besides eliminating clearance volume also has the greater advantage that sealing of the greatest pressure difference does not take place in the line of contact but across a relatively wide surface, here a thin oil film contributes to good sealing.

Low power inputs and high volumetric efficiencies generally depend upon the maintenance of minimum clearance between the rotor with its vanes and the cylinder with its covers, especially at higher compression ratios. For this reason, great care must be taken with calculations of thermal expansions.the precise manufacture of the parts and restriction of all clearances due to wear.

Since vanes and rotor are heated not only by the compresed gas but also by friction their temperature is taken to be rather high in calculating the clearance (upto 2/3 of the final adiabatic compression temperature) The length of the clearance gap between rotor(or vane) and cover must be

$$Sd = \frac{2}{3} \alpha L (tad - ts) + f \text{ mm}$$

where α is the coefficient of thermal expansion

l the length of rotor(or vane) in mm

ts temperature of induced gas(assuming that this is equal to the temperature of the cooled cylinder),

f thickness of the oil film(0.1 to 0.2 mm)

By substituting the radius r into the formula instead of the length, L, we obtain the necessary radial clearance, r between rotor and cylinder.

If the outer edges of the vanes are shaped in compliance with the theory of liquid friction, mechanical losses, mainly the wear of the vanes, are considerably reduced.

Lefevre(56) claims that vanes of this design permit "mean" peripheral velocity of the vanes on the cylinder (v = w.R) of 12 to 16 m/sec, while Mashinostroieniye(60) recommends for compressors with rings, a velocity of 10 to 12 m/sec, giving 13 m/sec as the maximum. With carbon vanes, however, a maximum velocity of 10 m/sec is recommended.

For medium compression ratios the overall adiabatic efficiency of vane compressor is approximately the same as that of reciprocating machines. Their biggest advantage becomes apparent when operating at a low compression ratio because valve losses are largely eliminated. Discharge valves are sometimes mounted between the cylinder and the delivery chamber allowing variation of the compression ratio which would otherwise be governed by the positions of the port. The shaft seal is similar to the simpler type of seal used in refrigerating compressors, i.e. sealing is not effected on the cylindrical surface of the shaft but on a ring perpendic- ular to the axis of the shaft.

CHECK VALVE

In order to prevent escape of the compressed air from the receiver during idle running of the compressor or the rotor being driven in the reverse direction after the com- pressor has been stopped, a single ring valve with high lift and light spring is mounted in the discharge chamber. This must present the least possible resistance to the normal flow. The continuous delivery of gas by the compressor means that a high lift can be used and this in turn allows the use of a single ring valve. The single-ring valve offers much better sealing than the normal multiport valves.

TWO-STAGE TYPES

For compression ratio higher than 4, rotary vane compressors are built as two-stage units, usually having both stages in line. The desire to reduce the overall length of the machine has recently given rise to the construction of air- cooled units with the second stage mounted above the first, both stages being linked by a pair of spur gears.

Rotary vane-type compressors can be used as single-stage machines for pressures of upto 3 atm and as two-stage compressors upto 8 atm. The maximum vacuum when operating as a vacuum pump amounts to 97% for single-stage and upto 99% for two-stage machines.

LUBRICATION OF VANE COMPRESSORS:

The oil is supplied to the cylinder through the covers at a point near the shaft. The oil flows out to the cylinder wall under the influence of centrifugal force. In a Czecho-slovak invention, the oil is supplied through the bored shaft to the bottom of the rotor slots, the flow being caused by the difference between the discharge and suction pressure. This invention will probably eliminate radial vibration of the vanes, which often cause rippling of the interior surface of the cylinder.

The lubricating oil for vane compressors is usually heavier than that for reciprocating compressors, particularly when the vanes are not held by the retaining rings discussed earlier. If the oil feed is not brought about by the gas pressure. the cylinders are fitted with force-feed lubricators. The bearings and the shaft seal must be lubricated as well as the vane. An efficient oil separator must be placed in the discharge line from a vane compressor. After filtration, the oil can be re-injected into the cylinder at a very high rate in order to reduce the leakage losses and to provide for internal cooling of the cylinder.

OVERALL VOLUMETRIC EFFICIENCY OF VANE COMPRESSORS AND VACUUM PUMPS:

Vane-type machines of normal design with moderate lubrication usually have an overall volumetric efficiency between 0.6 and 0.9; with internal oil; cooling and no clearance volume, large machines can reach a value, 0.95 even with a compression ratio of 5.

CAPACITY REGULATION OF VANE COMPRESSORS

Vane compressors are regulated by either

a) variation of the rotational speed or

b) closing the suction by a pneumatically controlled valve, closing the delivery by a check valve or by-passing from delivery to suction, or

c) stop-start control in the case of small machines.

DESIGN OF THE PROJECT

3.1 DESIGN CONSIDERATIONS

The relative dimensions of vane type compressor are as a rule for low pressure compressors.

$$\frac{r}{R} = 0.86$$

[Or] $e = 0.14R$

where

'r' is the radius of the rotor

'R' is the radius of the cylinder

'e' is the eccentricity of the rotor

If the final compression temperature does not exceed 60 °C no cooling is used,

3.1.1. PISTON-SWEPT VOLUME:

The largest volume between two vanes is given by the product of the length of the cylinder 'L' and the hatched area as shown in the fig (3.1)

$$fk = R1\ V1\ V2\ R2$$

where R1, R2 , V1 and V2 refer to the appropriate points on the diagram.

The distance of the vane contact point V from the axis of the rotor 'or' will be

$$\rho = e \cos \beta + \sqrt{R^2 - e^2 \sin^2 \beta} \qquad ; E = e/R$$

$$\rho = R(E\cos\beta + \sqrt{1-E^2 \sin^2 \beta} \qquad = R(1+[E\cos\beta - E^2/a \sin^2 \beta])$$

$$\rho^2 = R^2 (1+2E \cos\beta + E^2 \cos 2\beta) \qquad \rightarrow (3.01)$$

neglecting the terms containing E^3 and E^4 and putting

$\cos 2\beta = \cos^2 \beta - \sin^2 \beta$

The area fk is given by the relation

$$fk = \int_0^{\beta} {}^2 \, d\beta - \beta r^2 = \int_0^{\beta} {}^2 \, d\beta - \beta (1-E)^2 R^2$$

$$fk = R^2 E [(2-E)\beta + 2\sin\beta + E/2 \sin 2\beta) \qquad \rightarrow (3.020)$$

The angle $\beta = \pi/m$, m being the number of vanes in the rotor. if the number of vanes is greater than 12 sufficient accuracy is obtained using

$$\sin \beta = \pi/m \text{ and } \sin 2\beta = 2\pi/m$$

Thus the area fk becomes

$$fk = \frac{4R^2 E \pi}{m} \ m2$$

and the total swept volume during one revolution.

$$V_{theory} = fk.ML = 4\pi ReL \ m3/rev$$

In this calculation we have neglected he volume occupyed by the vanes, which is for one chamber.

$$S(\rho - r)\, L = SR\, E\, (1+\cos\beta - E/2 \sin^2 \beta)L \quad \text{-> (3.05)}$$

Where 'S' is the thickness of the vane.

For a large number of vanes we can put $\cos\beta \approx 1$ and neglect the last term.

The loss of volume in one chamber is then

$$S(\rho - r)L = 2SREL = 2SeL \rightarrow (3.6)$$

Therefore the the swept volume per revolution is

$$V = 4\pi ReL - 2m SeL = 2eL(\pi D - ms)\ m^3/rev \rightarrow (3.7)$$

3.2 Design Calculations :

3.2.1 Calcualation of pressure

r = Rotor dia = 105 mm radius = 52.5 mm

R = Housing dia 125mm radius = 62.5 mm

$r/R = 0.86$

The area of fK becomes

$$fK = 4R^2 E\pi/m * m^2 \ ; E=e/R$$

$$fK = 4*(0.0625)^2 *0.16*\pi / 6$$

$$fK = 1.309 * 10^{-3}\ m^2$$

3.2.2. Swept Volume Per Revolution

$$V = 2EL\ (\pi D-ms)\ m^3/rev$$

$$V = 2*0.01*200*10^{-3}\ (\pi*125*10^{-3} - 6*6*10^{-3})$$

$V = 1.426 * 10^{-3}\ m^{3}/rev$

The Intermediate Area Between two vanes $f_Ø$

$$f_Ø = Re\ [\ (2-E)\beta + 2\sin\beta\cos\beta + E/2*\sin 2\beta\ \cos 2\beta\ ]$$

$$= 62.5*10^{-3} *0.01[(2-0.16)30+2\sin(30)*\cos(150) +$$

$$0.16/2*\sin 2(30)\ \cos 2(150)]$$

$$=62.5*10^{-4}\ [0.96-0.015+6.04*10^{-4}\ ]$$

$$f_Ø = 5.9*10^{-4}\ m^{2}$$

3.2.3. Compression Ratio

$$f_Ø/f_K = [Po/P]^{1/n}$$

$$= (5.9*10^{-4})/(1.309*10^{-3}) = [(1.03*10^{5})/P]^{(1/1.4)}$$

3.2.4. Angle of the Discharge Edge:

When the chamber moves from the position of the largest volume where the cross sectional area is a maximun f_K by the angle $Ø$ the hatched area changes to the volume $f_Ø$ and at the sametime the gas pressure rises from the volume Po to P. For polytropic compression ($Pv^{n}=C$) the ratio of the shaded area is detemined from the compresion ratio

$$f\phi/fK = [Po/P]^{1/n}$$

The Reciprocal of the area can be found for various compression ratio P/Po by means of the diagram.

For a given angle ϕ the value of $f\phi$ can be determined by adapt-ing the equation

$$f\phi = 1/2 \int_{\phi-\beta}^{\phi+\beta} {}^{2}\, d\phi - \beta r^{2}$$

$$= R^{2} \;\; E[(2-E)\beta + (\sin(\phi+\beta) - \sin(\phi-\beta) +$$

$$E/4 \sin 2(\phi+\beta) - \sin 2(\phi-\beta)$$

on substituting

$$\sin(\phi+\beta) - \sin(\phi-\beta) = 2 \sin\beta \cos\beta$$

We obtain,

$$f\phi = Re\;[\;(2-E)\beta + 2\sin\beta\cos\phi + E/2\, \sin 2\beta\, \cos 2\phi$$

The angle ϕ for the given values of 'E' can be read for various number of vanes 'm' from the diagram in fig (3.2)

$$(0.45)^{1.4} = \frac{1.03 * 10^{5}}{P}$$

$$0.32\,P = 1.03 * 10^{5}$$

$$\text{or } P = 3.22 * 10^{5}\ N/m^{2}$$

$$\text{(i.e)} = 3.22\ kgf/cm^{2}$$

Centrifugal force calculations

Weight of each vane 'W' = 0.0584 kg.

$$\text{Centrifugal force (C.F)} = \frac{mV^2}{r} = \frac{W}{g} \cdot \frac{r^2}{r}$$

$$= \frac{W}{g} w^2 r$$

$$W = \frac{2\pi N}{60} = 100.5$$

$$C.F = \frac{0.0584}{9.81} * (100.5)^2 * 37.5 * 10^{-3}$$

C.F = 2.25 N = 0.2 kg.

3.2.5. Power Calculation :

$$Power = \frac{n}{n-1} P1 * D \left(\frac{P2}{P1} \right)^{\frac{n-1}{n}} - 1 \; X \; \frac{N}{60000}$$

Where $P1 = 1.03 * 10^5$ Mm2

Where D = Displacement = $1.426 * 10^{-3}$ m3/rev

N = 960 rpm

$$\text{Power} = \frac{1.4}{1.4-1} \times 1.03 \times 10^{5} \times 1.426 \times 10^{-3} \times \left[\left(\frac{3.22 \times 10^{5}}{1.03 \times 10^{5}} \right)^{\frac{1.4-1}{1.4}} - 1 \right] \times \frac{960}{60000}$$

$$= 3.16 \text{ KW}$$

Minimum Shaft Diameter Calculations

$$\text{Power} = \frac{2 \pi N T}{60000}$$

$$3.16 = \frac{2 \pi \times 960 \times T}{60000}$$

Torque $T = 31.4$ N.m

$$\frac{\tau}{r} = \frac{T}{J}$$

Where τ = Shear Stress

r = Radius of the Shaft

T = Torque

J = Mass Movement of inertia

$$= \frac{\text{Pressure}}{\text{Area}}$$

$$\frac{\text{Pressure}}{\text{Area}} = \frac{3.22 \times 10^{5} \text{ N/m}^{2}}{\frac{\pi \times (125 \times 10^{-3})^{2}}{4}}$$

$$= 26.24 \times 10^{6} \ N/m^{2}$$

$$\frac{\tau}{r} = \frac{T}{j} = \frac{26.24 \times 10^{6}}{d/2} = \frac{31.14}{d^{4}/32}$$

$$= \frac{26.24 \times 10^{6} \times d^{4}/32 \times 2}{d} = 31.14$$

$$d^{3} = 6.04 \times 10^{-6}$$

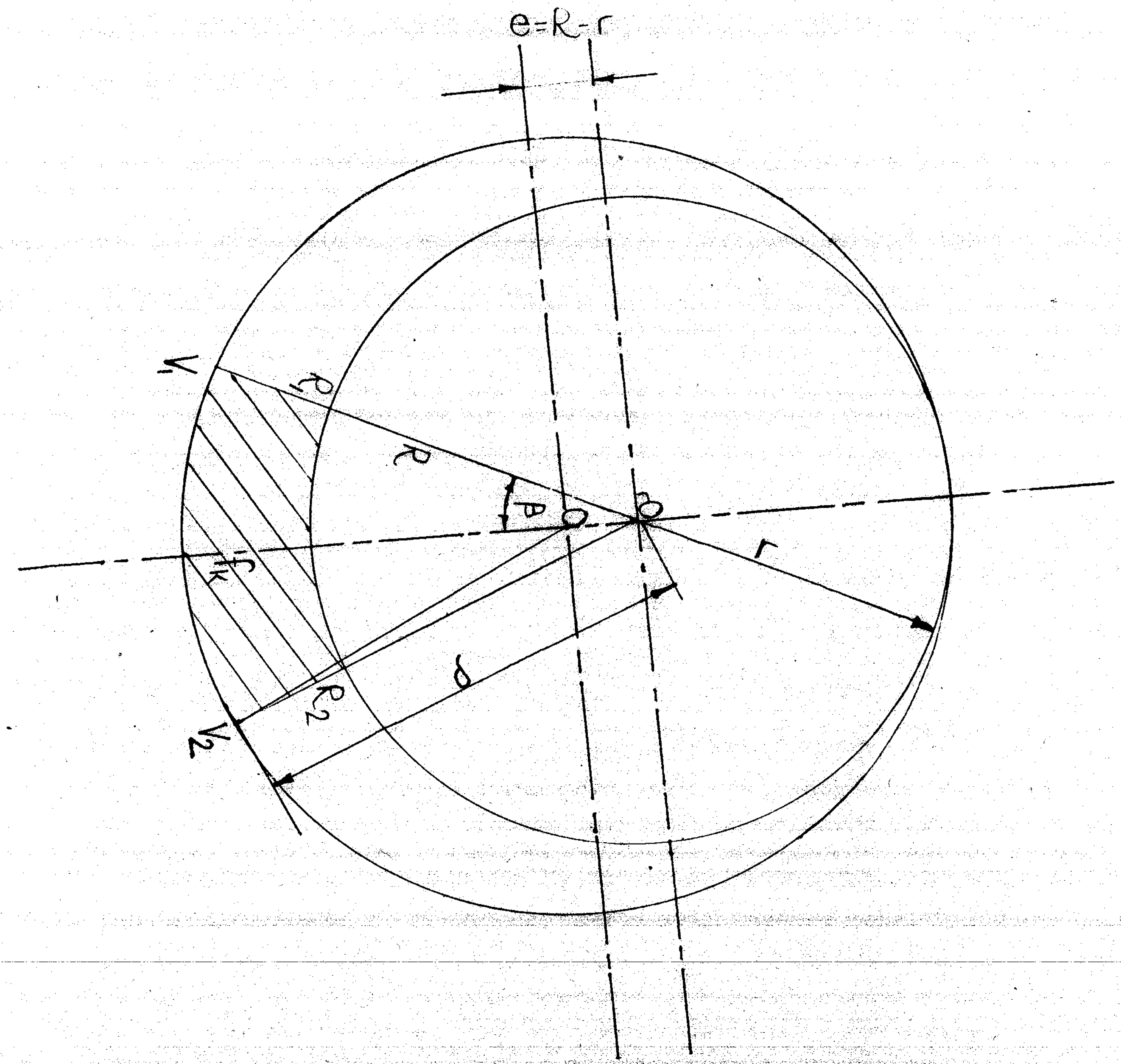
$$d = 0.018 \ m \quad \text{i.e.} \quad d = 18 \ mm$$

3.2.6 Free Air delivered:

$$\frac{1.426 \times 10^{-3} \times 960}{60} = 22.8 \ lit/sec$$

3.3 Compressor Specifications

Housing bore dia	= 125 mm
Housing length	= 200 mm
Rotor dia	= 105 mm
Vane Thickness	= 6 mm
Vane length	= 200 mm
Vane width	= 30 mm
Eccentricity	= 10 mm
Speed of the motor	= 960 rpm
Power of the motor	= 3 kw
Max.working pressure	= 3 bar
Free air delivered	= 22 lit/sec
Overall weight of the Compressor	= 33 kg



e=R-r
V1
V2
R
R1
R2
r
ρ
β
O
O1
fk

4.1 FABRICATION:

Machining of various Components:

4.1.1 HOUSING:

A wooden pattern in first prepared for moulding operation. After moulding in the foundry, molten cast iron from the cupola furnace is poured into the mould. Hence a cylindrical block is obtained.

The outer surface of the housing is first machined in a lathe. Turning operation is carried out using a turning tool. Then facing operation is done. The inner bore of the cylinder is machined in a boring machine using a boring tool. AFter machining to the required dimensions the inner bore of the cylindrical housing is ground in a internal grinding machine to obtain a very fine surface finish to reduce the vane sliding friction. The grinding is limited to 10 or 20 microns. Refer to drawing no.04.

In the both sides of the housing flange six equal holes of 10 mm diameter an angle of 60 between each hole is drilled in the radial drilling machine.

A delivery hole of 8.5mm diameter at an angle of 160 from the suction hole is drilled in a radial drilling machine. Tapping operation is done using a 10 mm tap of pitch 1.5 mm.

A circular groove of 165mm pitch circle diameter, dia 5m and depth 2mm is machined in a lathe is order to seat the 'O' ring.

4.1.2 ROTOR:

A wooden pattern is prepared and metal is poured into the mould in the foundry. The cylindrical solid block is first machined in a turret lathe. Initially metal is removed by turning and facing operation in the lathe. Boring operation is done in the Boring machine according to the shaft diameter. A key way is machined along the length of the rotor in a key way milling machine. Radial slots at an angle of 60 apart is marked on the rotor and slots are milled in a milling machine along the length of the rotor. Thus the rotor is machined to the required dimensions. Ref: Drg.No.02.

4.1.3 SHAFT:

The shaft is machined in Lathe. According to the drawing as shown in the figure step turning, facing operation is done according to the dimensions as in the drawing. The bearing diameters are ground in a cylindrical grinding machine to j6 tolerance. Refer to drawing no.03.

4.1.4 HOUSING END COVERS:

A pattern is made for the end cover and rough casting is prepared in the foundry. The center of the end cover boss is shifted from the end cover centre to a distance of 10 mm thereby maintaining an eccentricity of 10mm.

Machining is carried out in Lathe. Boring operation is also carried out in the lathe.

Drilling is carried out in Radial Drilling Machine. To the pitch circle diameter of 165mm six equal holes of 10mm diameter at an angle of 60 between each hole is drilled.

A suction hole is drilled in one of the end covers. In the end cover boss 4 equal holes of 4.5 mm diameter at an angle of 90 apart is drilled in the universal bench drilling machine. Tapping operation is done using a 6mm tap of pitch 1.5mm. Refer to drawing no.01 and 0.5.

A circular groove of 165mm pitch circle diameter, diameter 5mm, depth of 2mm is machined in a lathe in order to seat the 'O' ring.

4.1.5 BEARING END CAP:

Machining is carried out in lathe as per the drawings. To a pitch circle diameter of 65mm, 4 equal holes of 6mm diameter at an angle of 90 apart is drilled in a universal bench drilling machine. Refer to drawing no.06 and 07.

4.2 DYNAMICS

4.2.1 INTRODUCTION:

One of the basic problems with any machine is unwanted vibration. Because the machine dynamic rotation vibration will be present primarily because of unbalance.

In the rotary compressors the design has more control as they have no inherent shaking force other sources such as driver induced ibration or component problems can contribute to the machine shaking.

If the vibrations are left unchecked damage to the compressor will occur such as premature failure of bearing and in extreme case component fatigue and failure. Secondary effects may result from the shaking forces being transmitted through the frame to the foundation.

4.2.2 SHOP BALANCE MACHINE

This is where most balancing of rotors takes place. The component can be individually balanced in a precision mandrel. By precision it means the runout is 0.001 inch. The runout high spot should be scribed on the mandrel. The new component now can be reasonably well balanced.

4.2.3 ROTARY SHAKING FORCES:

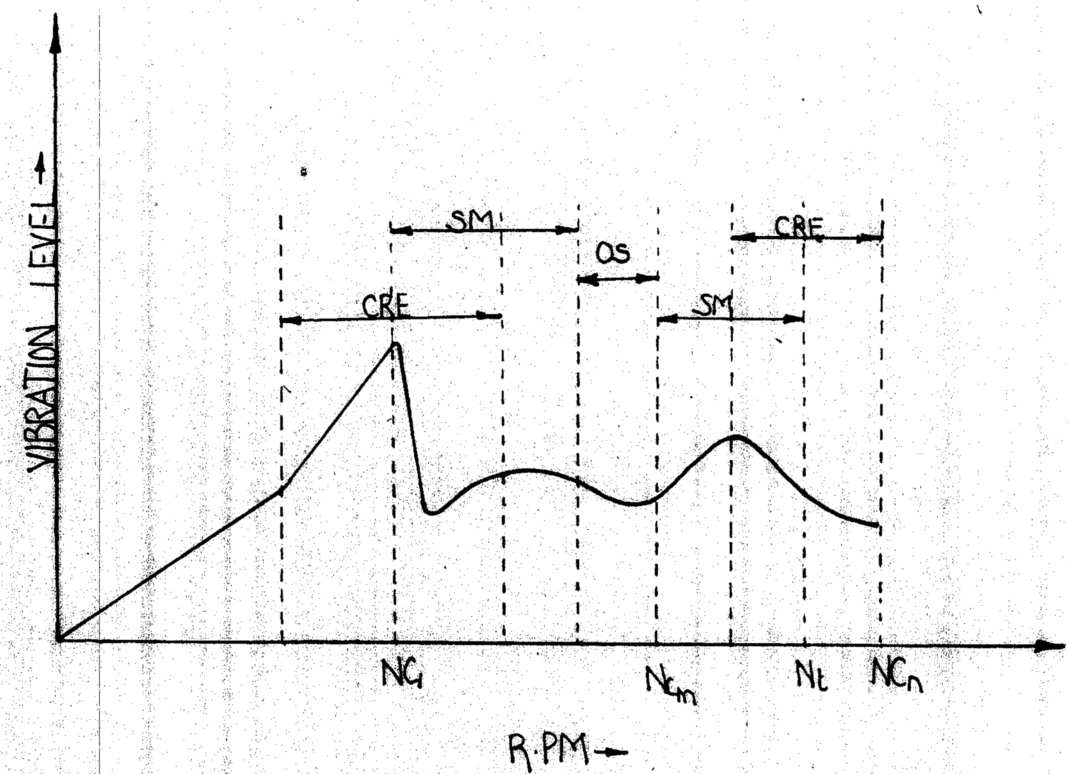
Normally these forces are rather insignificant relative to the weight of the compressor.Rotary and dynamic compressor have in common the force due to an balance. The objective is to balance the rotor to 0.19 or better a force equal to or less than 10% of the rotor weight.

This represents a shaking force at a frequency equal to rotor speed. For foundation design, a value of 5 to 10 times the residual unbalance or 1/2 to 1 times rotor weight at operating speed would be a resonable design value. The direction of the force is perpendicular to the shaft, and operates as a rotating vector which can be centered between the bearings.

4.2.4 ROTOR DYNAMICS:

Lateral critical speeds are some what misnamed since they are now thought of as a damped response to some form of rotor excitation. When the rotor geometric data shaft diameter, lengths, element size and weights ae put together, the mass static rotor system can be modelled.

The fig 4.1 shows a typical rotor response plot (courtesy of the American Petroleum Institute.)



Nc1 = Rotor 1st Critical centre frequency cycles/min

Ncn = Critical speed nth

Nt = Trip Speed

Wmc = Max continous speed 105%

SM = Seperation Margin

CRE = Critical response envelope

ASSEMBLY AND TESTING

5.1 ASSEMBLY:

Assembly is the final work to be done after machining of all the components. To start with first the bearings are fixed to the housing cover by means of hammering it. The bearing gets seated on the step provided in the cover.

The shaft is keyed to the rotor by means of a rectangular key.

The cover is fixed to one end of the shaft where there is a step to arrest the outer rays of the bearings, by means of hammering it with hard plastic headed hammer. The rotor is now introduced into the housing. Now bolts are introduced connecting the housing to the cover. Similarly the other cover is fixed to the other end of the bearing by hammering it and the cover is tightened to the housing by means of hexagonal round headed bolts. Bolts are tightened equally first on both sides of the cover and finally they are made jam tight. Bearing end caps are screwed to either side of the shaft ends.

The housing is now placed on the base plate and is fastened to the base plate by means of two metal sheets screwed to the base plate. Flexible coupling is keyed to one end of the shaft. Now the compressor is coupled with the motor .Refer to drawing no.00.

5.2 DIFFICULTIES ENCOUNTERED DURING ASSEMBLY:

When the rotor is placed inside the housing and the end covers fixed and fastened to the base plate the rotor does not rotate freely.

1. It was because the end coves should be tightened equally on both sides.

2. Secondly all the vanes have to be ground equally and has to go into the rotor slot without protruding outwards.

Fven if one of the vanes protrude out of the rotor slot when placed inside the slot the rotor doesn't rotate because at the point of minimum eccentricity the design is almost of 0.1mm clearance between the rotor and the housing hence, if the vanes protrude out of the slot it gets jammed with the inner cylinder walls.

5.2.1 VIBRATION:

Vibration was encountered when the compressor was not aligned properly on the bore plate and if it was not fastened to the bore plate rigidly.

Also it has to be securely tightened to the foundation moreover the compressor has to be coupled to the motor accurately. There has to be a clearance between both the compressor and motor couplings.

5.2.2 BACK FLOW:

There was some amount of back flow of air inside the compressor, this was due to leakage in the suction side housing covers when made jam tight the back flow was reduced to a considerable extent.

TESTING

5.3 INTRODUCTION:

The word test is quite broad in its definition, and many of the inspection steps in the course of the compressor manufacturing cycle can approxpriately be called tests.

5.3.1 OBJECTIVES:

The question generally arises "Why test?"

There are reasons why from both the manufacturers and users point of view Testing is expensive and manufacturers have no incentive to test low value items except where consequential damage can be caused to the equipment supplied.

The testing has to be reviewed with regard to

1. Verification of design parameters.
2. Proof testing of all functions.
3. Correction of faults found.

The longest time required for modification is in the ordering of raw material. The extent of this should be inversely proportional to the manufacturers experience and directly proportional to the financial consequences of lost production.

5.3.2 Speed :

One of the most important parameters is speed. Capacity is directly proportional to speed on all compressors. Because of the significane of the speed parameter, it is important that accurate speed measurements be made all during test. The speed is measured using a tachometer.

5.3.3 Pressure :

The most important of all the tests is the testing of presuure which is essential in all the compressors. The discharge pressure is measured using a sensitive pressure measuring device in the case of a sliding vane compressor to other compressors.

5.3.4 Vibration measurement :

For the Vane compressor, , the vibration that was carried using mechanical vibration meter. The vibration was measured at various points on the housing . The amplitude of vibration was found to be 22 to 32 amplitudes.

5.3.5 Temperature measurement :

The temperature at the discharge end was measured using Digital thermometer. The Digital themometer was attached to the housing and the maximum temperature was measured as 60 C. At he time of measurement the room temperature was found to be 30 C.

5.4 Suggestions for further Improvement.

5.4.1 Housing:

The housing inner surface was ground on internal grinding machine. If it was honed in a Honing machine the surface finish would be much better and it may reduce the vane sliding friction to a considerable extent.

5.4.2. Vanes:

The vanes used here is made up of hylam. If carbon was used as vanes it offers much more resistance to wear than hylam. Carbon was not used in our project because of its exorbitant price. Also carbon is a self lubricant.

5.4.3 Cooling:

In this project the compressor is of a dry type. Hence more heat is generated during operation due to the sliding vane friction. If proper oil cooling system is coupled with the compressor it can cool the compressor to a great extent.

5.4.4. Rotor:

Rotor, if made of lighter materials like aluminium or plastic the weight of the compressor can be drastically reduced.

ROTARY COMPRESSORS AS AN ADIABATIC OR ISENTROPIC MACHINE

The rotary compressor is the least known and least popular type of gas compressor. Rotary compressors have been proposed as the answer to the compressor dilemma a machine suitable for variable compression ratio and a wide capacity range, one that operates at electric motor speeds and has a potential maintenance history equal to that of a centrifugal compressor.

This project considers the rotary compressor to be an adiabatic or isentropic machine. It has no valves. It operates by confining a volume of gas in a rotating pocket. The confined gas pockets coverage into the discharge chamber. The action is that simple. The speed of the gas is nominal and not great enough to cause polytropic effects such as are experienced in centrifugal compressors when near sonic gas velocities impinge against the vane and volute obstructions. Why then does the discharge temperature exceed the adiabatic rise?

The premise of this chapter is that this type of machine has three aerodynamic losses. The first loss is that incurred in charging and exhausting the pockets which increases the compression ratio within the pocket. The second loss concerns the leakage between the rotors and between the rotors and the case. The third loss is the thermal burden caused by the intake warm-up.

The pocket changing losses are considered to be a function of the rotor tip speed. The gas must follow in the wake of the pocket tip which is continuously opening in the inlet chamber. It is necessary to expand a certain head to match the rotor speed. This is equivalent to the Net Positive Suction Head (NPSH) required to charge a centrifugal pump.

TABLE 5.1

CHARGING AND EXHAUSTING PRESSURE LOSSES FOR ROTARY TYPE COMPRESSORS

TIP SPEED		VELOCITY HEAD	DIFFERENCIAL PRESSURE		
			SUCTION	DISCHARGE	
MACH	fps	$U^2 / 2g$	PSI	$R_c = 2$	$R_c = 3$
0.71	807	10,000	5.35	8.69	11.70
0.45	510	4,025	2.15	3.50	4.70
0.38	430	2,875	1.50	2.46	3.32
0.26	296	1,360	0.71	1.16	1.57
0.14	162	400	0.21	0.34	0.46
0.10	108	180	0.10	0.15	0.21

Table 5.1 shows the pressure differential that must be sacrificed to attain a velocity equivalent to the rotor tip speed. The sonic velocity of the air considered herein is 1,130 fps at a suction temperature of 60 °F. These velad losses are applied to Equation 5.1 to produce the "B" correction factor. This modifies the visual, line pressure Rc to the more

realistic intrinsic pressure differential as it exists in the converging screws. This example considers two cases: One where Rc = 2 or P = 29.4 and the other where Rc = 3 or P2 = 44.1 with P1 = 14.7

B=(1.0+ P2/P2)/(1- P1/P1)

ηad=(RcØ-1)/(BRØ-1)

These equations apply the "B" correction to Rc and give the compression efficiency in reference to the minimal adiabatic 0 power requirement.

The compression efficiencies produced from one K of resistance are too optimistic. Those produced from two K are too conservative. A compromise that uses 2 velads of suction resistance and one velad of discharge resistance appears to be realistic. This resolution of compressor performance does not have the supporting test data that should be allocated to such a project. It does present the rudiments and suggest the outline for such a test program. In the interim it offers the most comprehensive procedure for projecting the performance of high speed rotary compressors.

Presume that one velad is consumed in overcoming the resistance of the compressor chamber, entering and dead heading into the rotor pocket. The absolute velocity of the air must also match that of the rotor tip. This totals 2 velads for the charging operation or 2(2.15) = 4.30 psi. This loss is expressed as a decimal fraction, 4.3/14.7 = 0.293 or 29.3 percent of the suction system pressure.

It is further presumed that the built-in clearance releases the trapped air at or very close to the discharge chamber pressure, so that no overcharging or undercharging exists in the exhaust pocket. The absolute velocity of the gas leaving the rotor and entering the discharge chamber represents another velad expenditure. The analytical difference between the suction and discharge velad is the change in the gas specific volume. The discharge V_{sd} is Vos/Rc1/k = 13.1/2.19 = 5.95 cf/lb, and gas head per psi is 144(5.95) = 857 ft/psi. The gas is dispelled into the discharger chamber without the benefit of any convergence devices to recover the velocity energy of 4,025/857 = 4.7 psi per velad. This energy is wasted and represents 4.7/44.1 = 0.106 decimal fraction;on of the discharge system. This data is applied to Equation 5.1 to evaluate B = (1 + 0.106)/(1 - 0.293) = 1.106/0.707 = 1.565. This makes the intrinsic BRc = 4.7. This value applied to Equation 5.2 makes the adiabatic compression efficiency nad = $(3.0^{\sigma} - 1)/(4.7^{\sigma} - 1)$ = 0.37/0555=66.5 percent.

This procedure can be resolved into the following simplified equations:

$$B_i = 2.5\, m(U^2)/T(10^5) \qquad \text{-> (5.3)}$$

where m represents the molecular weight of the gas, U is the male rotor tip speed in fps and T is the gas temperature °R.

This equation includes two velads of resistance. The exhaust loss equation below includes one velad of resistance.

$$B_e = B_i/2(R_{cr})^{\sigma} \qquad \text{-> (5.4)}$$

The two resistances, θ_i and θ_e are used to evaluate the intrinsic pressure correction factor "B".

$$B = (1.0+\theta_e)/(1.0 + \theta_i) \qquad \rightarrow (5.5)$$

LEAKAGE

Leakage is the next step to evaluate. The rotary compressor configuration offers three longitudinal lines of escape. Two lines follow each rotor and the casing, and the third line is the addendum of the converging rotors. The internal leakage for rotary machinery follows the equation:

$$W_L = 23LP_2 G/(T_2)^{0.5} \quad \text{lb/min} \qquad \rightarrow (5.6)$$

The displacement of a rotary machine is:

$$QD = d\,LUX,\ efm \qquad \rightarrow (5.7)$$

The capacity of the machine is dependent upon the charging pressure. Equation 5.3 has established the percentage of the suction pressure that is lost in the charging operation. The volumetric efficiency simply follows as:

$$E_{vr} = 100 - (\theta_i + W_s/R_c^{\sigma}) \qquad - > (5.8)$$

Where the suction charging loss is not available, use the approximate express, $E_{vr} = 98 - V\,Rc$. Table 5.3 gives an average V factor for each type of rotary compressor.

The displacement multiplied by the volumetric efficiency gives the capacity of the compressor. The capacity divided by the specific volume gives the capacity in terms of pounds per minute:

$$W = 0.093\ dm\ E_{vr}\ LUX\ P_1/T_1 \qquad \rightarrow (5.9)$$

The slip leakage is the ratio of (Equation 5.6/Equation 5.9) which reads:

$$W_s = 30.7\ (0.577\ G + 0.00038)\ R_c^{1.9}\ d/m\ E_{vr}\ LXU^{0.5} \qquad (5.10)$$

The manufacturer usually determines the volumetric efficiency in a unique manner. The speed required to sustain the design pressure at zero flow is known as the slippage. The decimal fraction of the slippage speed divided by he rated speed is the volumetric efficiency. For example, the static slippage for an 8-inch, straight-lobe compressor is 120 rpm at 3 psig and 144 rpm at 10 psig. The percent slip in reference to the rated speed of 1,200 rpm is (120/1,200) or 10 percent for the 3 psig static system. The slippage Nz for the 10 psig system is 12 percent and the manufacturers' volumetric efficiency is 88 percent.

This method does not consider the dynamic loss which is evaluated by Equations 5.3, 5.8 and 5.10. The tip speed loss in most other types of rotary compressors is so low that it is negligible. Table 5.4 shows that the tip speed must exceed 0.12 Mach before it is significant. These zero-flow speeds provide valuable checks on the clearance gap and a means of estimating the zero-flow pressue. Having the percent of slip speed, Ns*, the likely discharge pressure is

$$R_c^{1.9} = N_s{*}m\ LE_{vr}\ X(U_s^{0.5})/30.7d\ (0.5779+0.0038)$$

Using G as the actual gap = 0.133 and G* as the effective gap = 0.0095, Us = rpm0d)/229 = 144(8)/229 = 5, Ns* = 0.12 E_{vr} = 0.98 assume L = d; X = 0.265, $U_s^{0.5}$ = 2.23 and m = 29:

$R_c^{1.9}$ =0.12(29)0.98(0.265)2.23/30.7(0.0095)=7

R_c = 2.80, P_2 = 2.80(14.7)= 41.2 psia or 26.5 psig

Having the speed and the discharge pressure, it is possible to solve or the clearance gap G in the same equation.

The average clearance gap G for small (3 to 4 inch) rotors is 0.010, 0.020 for 13-inch rotors and 0.030 for 20-inch and larger rotors. A_e is a geometric coefficient based upon the open-end area divided by the square of the male rotor diameter, d is the diameter and L is thelength of the rotor, both in inches. The A_e and X coefficients for the various rotary configurations are given in Table 5.3 (X = 0.1235Ae) The exponent 1.9 is an average value suitable for diatomic and heavy polyatomic gases. The complete exponent is 1 + (k +1)/2k, where k is the ratio of specific heat at the mean compression temperature.

TABLE 5.3
SUMMARY OF ROTARY DISPLACEMENT COMPRESSOR PERFORMANCE DATA

	Helical Screw	Spiral Axial	Straight Lobe	Side Vane	Liquid Liner
Configuration Features(Male* Female	4 x 6	2 X 4	2 x 2	8 Blades	16 Sprockets
Max Displace -ment, icfm	20,000	13,000	30,000	6,000	13,000
Max Dia., in	25	16	28	33	48
Min Dia., in	4	6	10	5	12
Limiting Tip speed y Mach	0.30	0.12	0.05	0.05	0.06
Normal Tip Spe -ed y Mach	0.24	0.09	0.04	0.04	0.05
Max.L/D Low pressure	1.62	2.50	2.50	3.00	1.1
Normal L/D High Pressure	1.00	1.50	1.50	2.00	1.0
V factor for Volumetric eff	7	3	5	3	3
X factor for displacement	0.0612	0.133	0.27	0.046	0.071
Normal Overall effect.,%	75	70	68	72	50
Normal Mech effect.at +/- 100 hp ,%	90	93	95	94	90
Normal ratio of Compression Rc	2/3/4	3	1.7	2/3/4	5
Normal Blank-Off Rc	6	5	5	7	9
Dispalcement factor, A	0.462	1.00	2.00	0.345	0.535

Note : Mach =1250 fps air at 185 F

TABLE 5.4
ADIABATIC SLIP AND THERMAL EFFICIENCIES

Mach No. (Tip speed)	Efficiencies	Rc 2.0	Rc 2.5	Rc 3.0	Rc 4.0
0.35 M* (392 fps)	n_{ad}	60.5	66.3	69.5	73.4
	n_s	96.0	93.8	91.3	84.8
	n_t	99.0	98.3	96.7	92.4
	n_o	57.5	61.0	61.4	57.5
	n_{ad}	73.3	78.0	80.5	83.5
	n_s	95.9	93.7	91.2	84.5
	n_t	98.9	98.0	96.7	92.2
	n_o	69.5	71.7	71.0	65.2
	n_{ad}	93.5	94.5	95.6	96.2
	n_s	94.5	91.4	88.0	79.1
	n_t	98.5	97.2	95.4	89.5
	n_o	87.1	84.0	80.3	68.1
	n_{ad}	96.0	96.8	97.3	97.6
	n_s	94.1	90.9	87.2	77.6
	n_t	98.7	97.1	95.6	88.8
	n_o	89.2	85.5	81.2	67.3

Table 5.4 gives the percent leakage for tip speed variation of 0.1 to 0.35 Mach in reference to the total weight flow for a typical 6-inch rotary compressor having a nominal clearance gap of 0.0133 inch. The leakages were calculated from Equation 5.10. A variation in the diameter or the gap creases the leakage in direct proportion. The R_c is the most significant factor that affects the leakage.

TABLE 5.5

ANTICIPATED LEAKAGE PERCENTAGE OF TOTAL WEIGHT FLOW

Mach No. (Tip Speed)	R_c 2.0	2.5	3.0	4.0
0.35 (392 fps)	4.0	6.2	8.7	15.2
0.26 (300 fps)	4.1	6.3	8.8	15.5
0.12 (136 fps)	5.5	8.6	12.0	20.9
0.10 (113 fps)	5.9	9.1	12.8	22.4
Average	5.0	7.5	10.6	18.5
a) Temp Rise	7 °F	12 °F	22 °F	48 °F
Power Loss	1.3 %	2.3 %	4.1 %	8.9 %

Note (a) The above leakage raises the suction temprature by the amount shown.

Table 5.5. also shows the amont of warm-up resulting from bypass leakage to the incoming flow. The percent of power loss is directly proportional to the increase in the Rankine tempeature. This leakage percentage is also shown in Table 5.5. A simple expression for evaluating the leakage warm-up effect for each percentage of leakage is

$$Xt = 0.12(Rc) + 0.02 \qquad \rightarrow (5.11)$$

Equations 5.10 and 5.11 offer a means of evaluating the leakage and warm-up losses. Neither are, in reality, dynamic losses and should not be corrected by a gas head alteration. The leakage and the thermal burden directly affect the horsepower and can be applied as an efficiency factor in the denominator of the power equation. The leakage effect is given in Equation 5.12. The thermal effect of the suction warm-up is given in Equation 5.13:

$$n_s = 1 - W_s \qquad \rightarrow (5.12)$$

$$n_t = 1 - W_s(0.12R_c) + 0.021 \qquad \rightarrow (5.13)$$

$$Adhp = 0.0468(R_c^{\sigma} - 1)\ W\ Z\ T/m\sigma \qquad \rightarrow (5.14)$$

The basic adiabatic horsepoweris the same for all types of compressors, namely:

Where W is the gas flow in pounds per minute, Z is the compressibility factor and r is (k -1) /k

$$Gas\ hp = Adhp/n_{ad}\ (\eta_s)n_t \qquad \rightarrow (5.15)$$

$$Bhp = Gas\ Hp + friction(gas\ hp)^{0.4} \qquad \rightarrow (5.16)$$

JACKET - WATER REQUIREMENT

The slide vane and piston compressors have common difficulties in their lubrication and jacket temperature. There is a scarcity of credible reference data on both subjects and for that reason these data are included. The quantity of heat absorbed by the jacket water is best determined from these equations:

$$H_j = 4(t_{ag} - t_{aw}) + 100 \text{ , BTU /bhp-hr} \quad \rightarrow i$$

The letters t_{ag} represents the average of the gas temperature

$(t_{out} + t_{in}$ and t_{aw}) is the average jacket water temperature.

The heat rejection is 3.12 % of the mechanical heat equivalent, 2545 btu/bhphr. Statements advocating 20 and even 30% of the total power heat equivalent are absured. One oil companv has operated its entire system of gas compressor with dry jackets for over 15 years. They have no detrimental experience. There has never been a creditable reduction in the power as a result of jacket water heat rejection.

There has been considerable damage experienced as a result a cold cylinder jacket water system in handling wet gases at or gas close to the saturation;on line. The cold liner causes the liquids in gas to condense and wash of the lubricant. reducing the life of the sliding vanes to a matter of weeks. In instances of this order, it is advisable to reduce the water circulation so that the temperature leaving the cylinder is within 10 °F of t_{ag} . The jacket water temperature rise in equation (ii) should be 2° & 5° for a warm (140° F)system.

The sliding vane compressor is not suitable for handling saturated and supersaturated vapour. It is particularly inapt with cold jackets. This increases the cylinder wall condensation. The best operation requires a warm jacket and use of lubrication with a solvent resistant inhibitor such as rapeseed oil,lanolin or tallow.The manufactures advocate rather generous quantity of cold jacket water lubricant.

The oil rate is doubled for the 20 inch cylinder and tripled for cylinders over 30 inches in diameter. The vanes are made of laminated phenolic resins.Their life and cylinder wear are sensitive to the quantity and quality of the lube oil applied. The most success application of the slide vane compressors has been with the 100 hp 360 g/m portable - construction type air compressor. The rotors are cooled with lubricant. These oil rates are on the order of 10 times that required for equivalant piston machinery service. The table was compiled from 20 years data accumulated on reciprocating compressor by a California Oil company's gas plant.

QUANTITY OF LUBE OIL REQUIRED FOR COMPRESSOR CYLINDER LUBRICATION

Range of Cylinder diameter	Normal discharge Pressure	Film Thickness Microinch	bhp-hr/ gallons (Thousands)	Compressor Lube Oil,
24.36	80	16	13	1.5
15.23	200	17	19	1
10.14	800	23	24	0.8
7.9	2000	24	32	0.6
4.7	8000	47	24	0.8
3.5	20000	57	27	0.7

SAFETY PRECAUTIONS

Various safety regulations, depending to a large extent upon the gas handled, must be observed when operating a compressor plant.

The main hazards are:

Rursting of vessels and piping by the compressed gas.

Accidents due to unprotected moving parts of compressors and motors.

Poisoning by toxic gases.

Burns by hot surfaces on the delivery side of the compressor and by various gases.

Fire caused by inflammable and explosive gases.

Accidents due to electric current where electric motors are used.

Accidents by parts of damaged compressors.

Accidents in mounting heavy parts of the compressor plant.

In certain industrial fields, the safety rules stipulated for compressor plants are incorporated into various codes.

In mining these are the Board of Mines Regulations(4) and the newer Technical-Operational Regulations for the Ostrave-Karvina Region(85) and further successively introduced specifications.

In the field of refrigeration there is the Safety Code for Refrigerating Plants of an output of upto 40000 kcal/hr according Czechoslovak Standard CSN 14 0645 and the Safety Code for Refrigerating Plants of an output exceeding 40000 kcal/hr contained in Czechoslvak Standard CSN 14 0646.

In other industrial fields no detailed safety regulations concerning compressor plants have been issued. Receivers, compressed gas bottles and other pressure vessels used in compressor plants should be designed, manufactured and tested in accordance with the pertinent safety rules. The necessary certificates should be delivered with these components.

In order to eliminate the causes of accidents, the following rules must be observed:

1. Mechanization and automation must include a means of signaling dangerous conditions.

A safety valve is essential for each stage of a compressor. If a pipe closure is inserted between compressor and safety valve it must be open whenever the plant is operating or is about to be started.

2. Enclosing of all moving machine parts, especially belts, in the proximity of personnel. Enclosing big couplings by railings.

3. Efficient sealing of all pipe and vessel joints, good ventilation and lighting of the compressor room.

4. Covering or insulating of hot surfaces of compressors and pipes. Ducts for piping and for cables, pits for coolers and separators etc., should have covers of adequate strength.

5. Correct design and mounting of electric installations.

6. Building platforms for maintenance of compressors and parts out of reach from the floor. These platforms should have a railing at least 1 m high and reaching down to 18 cm or less, above the platform.

7. Instruction of operators in safety regulations and conditions for safe working and a system of ensuring that the regulations are carried out.

8. Unconditional prohibition of smoking or the use of an open flame wherever inflammable and explosive gases are handled.

9. Clear notices on all doors of the compressor room forbidding entry to all unauthorized persons.

10. Systematic, regular maintenance of the entire compressor plant.

11. Cleanliness of the compressor plant, especially the floor. Petrol and other easily inflammable materials should not be used for cleaning the internal parts of compressors.

12. The machine and piping should be adjusted prior to assembly, making sure that no spanners, nuts, cleaning material or other objects have been left in them.

13. Welding of pipes and vessels for inflammable and explosive gases only after they have been evacuated and filled with an inert gas.

14. Wherever a positive gas pressure has been operating on any parts such as covers, flames, screwed connections etc., this pressure must be relieved before dismantling. Toxic and inflammable gases must be evacuated before opening the plant. Stop valves of high pressure piping must be opened very slowly to avoid sudden pressure changes and possibility of explosion.

15. Steps must be taken to prevent the machine being started during cleaning and repair work. Before dismantling a machine following breakdown the main switch should be disconnected and the fuses removed. A warning notice should be placed in a prominent position to the switchboard to indicate that repairs are in progress.

16. Lubrication with an adequate quantity of oil recommended by the manufacturer.

17. Monthly inspection and cleaning of the compressor valves.

18. No work should be carried out on the machine while in operation.

19. Use of suitable working cloths for normal operation and of protective clothes, goggles and masks for repairs to the plant.

20. Adequate insulation between pipes and electric cables.

21. Safe thermal insulation of the discharge line whenever it passes through the walls of inflammable material.

22. A trained operator must be constantly in attendance unless the plant is fully automatic.

CHAPTER SIX

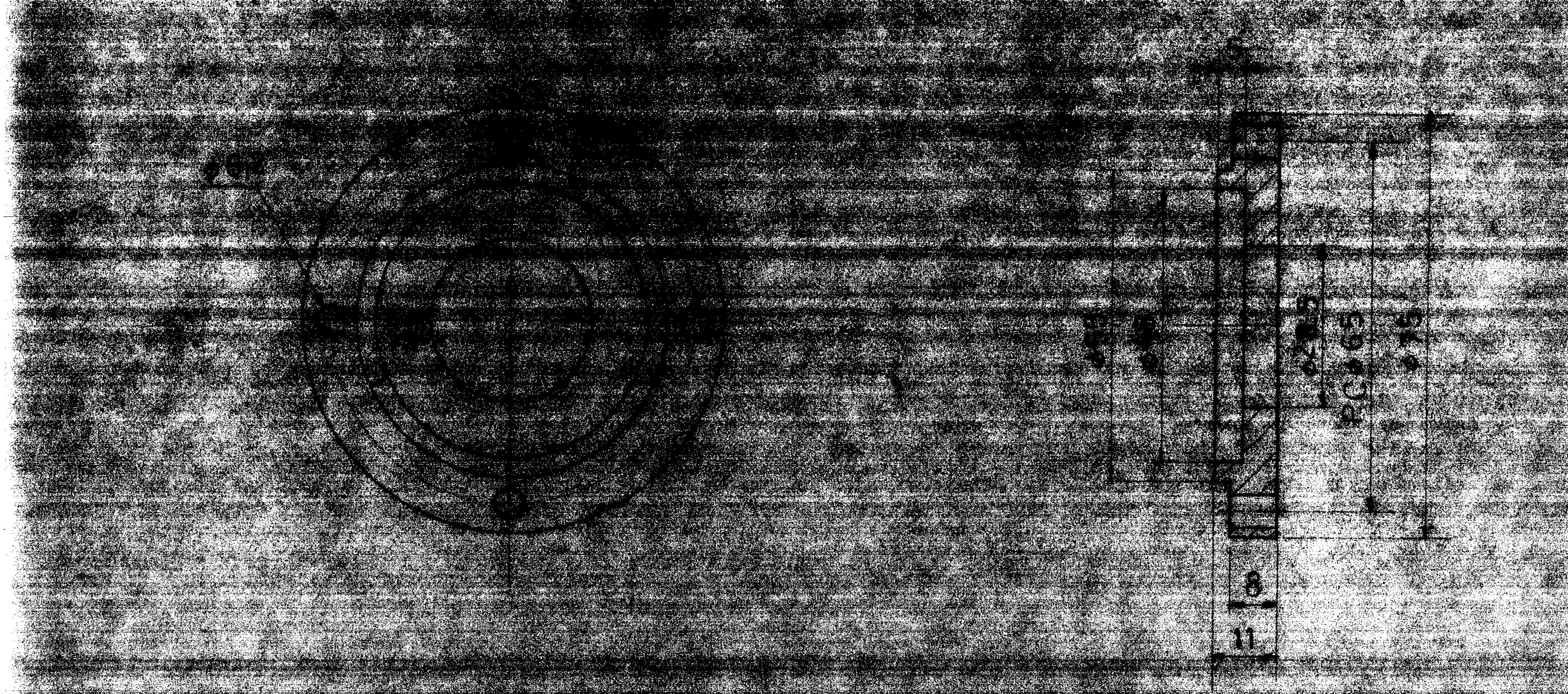
6.1. Bill of Materials and cost Estimation:

S.No.	Part Name	Mat1	Wt. Kg.	Mfg/ Bo	No. off	Total Cost
1.	Housing	FG 200	8	Mfg	1	200.00
2.	Rotor	FG 200	10	Mfg	1	250.00
3.	Vanes	Hylam	.324	BO	6	50.00
4.	End Covers	FG 200	4	Mfg	2	90.00
5.	Bearing End Cup	FG 200	0.8	Mfg	2	20.00
6.	Shaft	C 15	2	Mfg	1	45.00
7.	Base Plate	FG 200	5	Mfg	1	40.00
8.	Coupling	FG 200	2	BO	1	30.00
9.	Socket head M6x20 screw	MS	0.3	BO	8	4.00
10.	Socket head M10x40 Screw	MS	0.6	BO	12	12.00
11.	Bearings	-	-	BO	2	150.00
12.	Air Filter	-	-	BO	1	30.00
13.	'O'-Rings	-	-	BO	2	24.00
	Total -					945.00

Total cost of the compressor = Rs.945.00

3 KW 960 rpm motor cost = 3,600.00

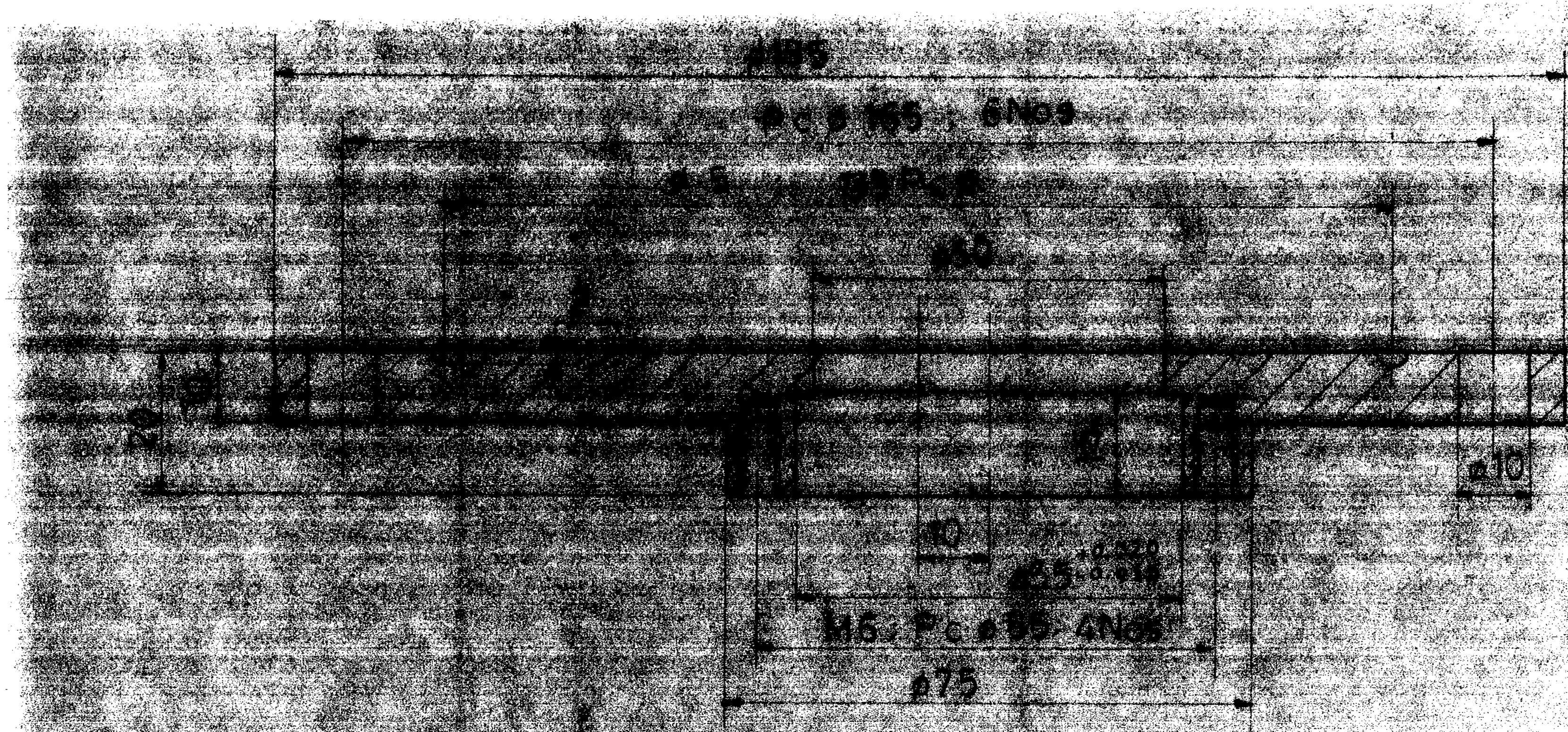
Total cost of the unit = 4,550.00



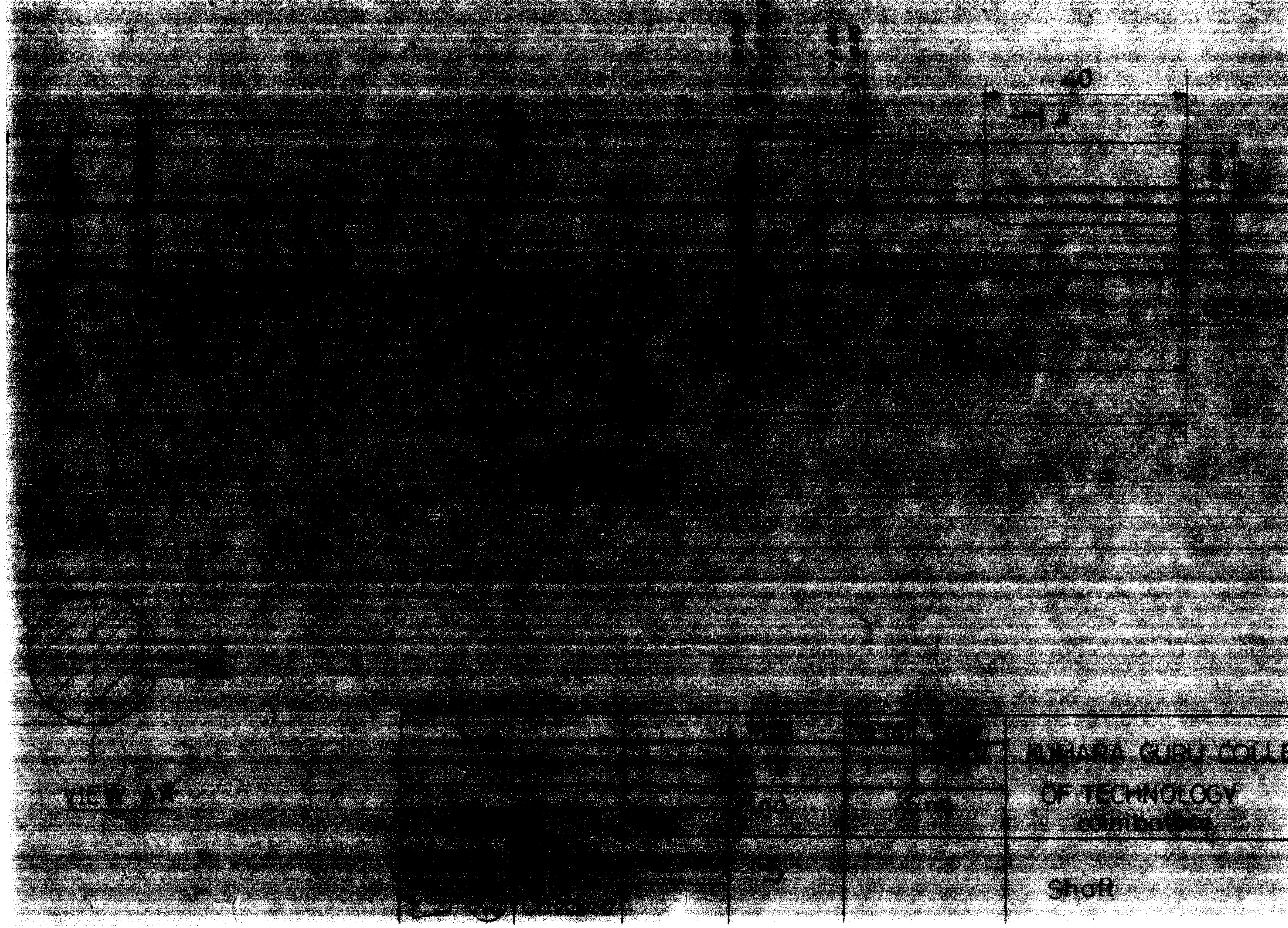
KUMARA GURU COLLE

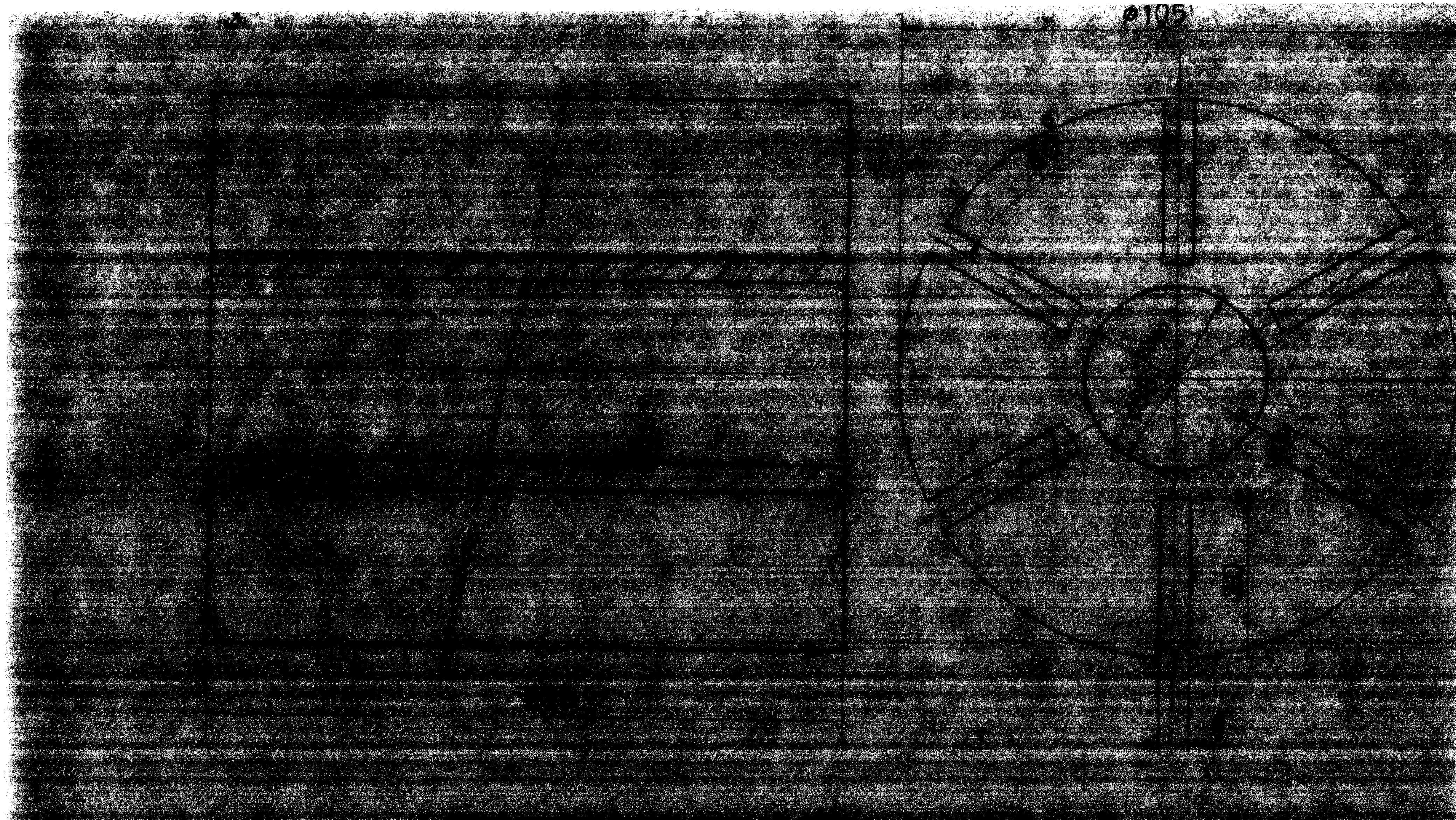
OF TECHNOLOGY

7



		Matl	No off	Toler	KUMARA GURU COLLE
		FG 200	1	IS 2102	OF TECHNOLOGY coimbatore
Revision	date	P.no	S no		
Scale 1:1	Design	6			R.H End cover
	Checked				


40
VIEW AA
KUMARA GURU COLLE
OF TECHNOLOGY
coimbatore
Shaft

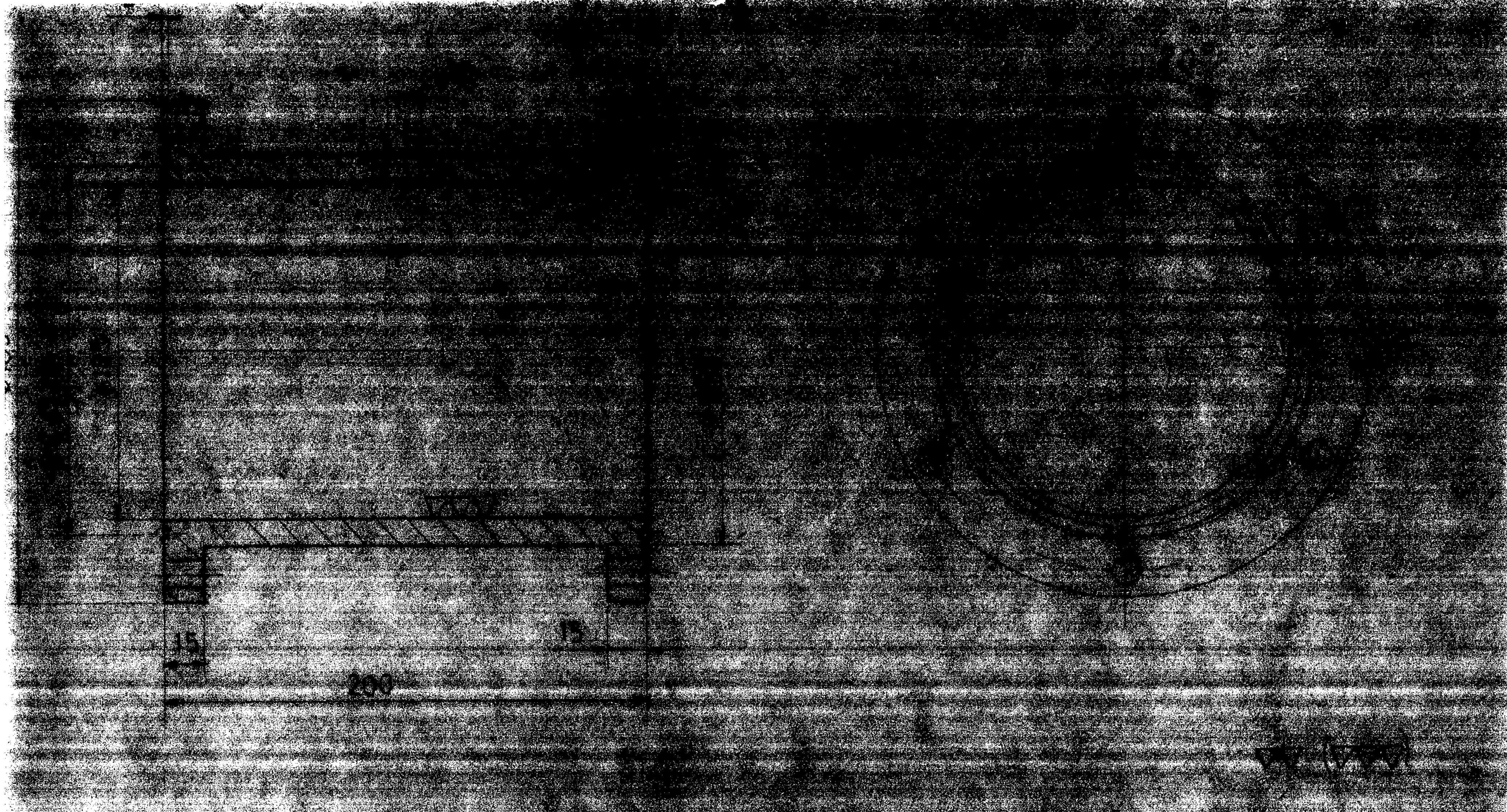


OF TECHNOLOGY

P no.		S no.
4		

Rotor

Checked



		FG 200	1	KUMARA GURU COLLE
		P no	S no	OF TECHNOLOGY coimbatore
Checked		3		Housing