Design and Fabrication of a Triangular air Compressor with a Common Compression Chamber

KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE-6
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A Project Report 1998-99

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CERTIFICATE

Name ____________________________________ *Register No* ____________

Certified that this is the Bonafide Record of the Project Work

DESIGN AND FABRICATION OF A TRIANGULAR AIR COMPRESSOR WITH A COMMON COMPRESSION CHAMBER

Done by

Mr ______________________________

In partial fulfilment for the Degree of Bachelor of Engineering in Mechanical Engineering Branch of Bharathiar University, Coimbatore during the academic year 1998 – 99.

________________________ ________________________

Head of the Department *Project Guide*

Submitted for the University Examination held on ____________

________________________ ________________________

Internal Examiner *External Examiner*

SANTHASIVAM INDUSTRIES

CERTIFICATE

This is to certify that the following final year students of Kumaraguru college of Technology success fully carried out their project in our industry, titled " Design and fabrication of a Triangular air compressor with a common compression chamber"

V. P. VIKRAM VETRIVEL

R. KALIL RAHMAN

V. KARTHI KEYAN

They had shown keen interest in designing and fabricate the machine we wish them all success in future career.

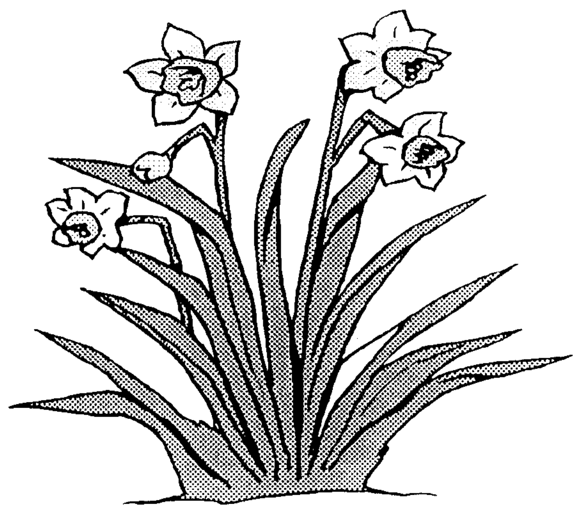
For Santha Sivam Industries

For Sri Santhasivam Industries

Partner.

CONTENTS

SYNOPSIS



SYNOPSIS

Compressors are used to get high pressure air for many industrial and commercial purposes. The triangular air compressor with common compression chamber is a reciprocating type compressor, which delivers air at high pressure with less power consumption than the existing ones.

Triangular air compressor with common compression chamber uses three cylinder and all the cylinders will have their own connecting rod, crank shaft, pistons and chain sprockets and it is driven by a chain drive.

During operation the pistons will move in phase from BDC to TDC and hence air is compressed in the common compression head. This common compression chamber will have one inlet and outlet valve. From this valve air is intaked and delivered. The common compression head will be in a triangular shape with an angle of 60° from one side to other.

The specification of the compressor are

1. Maximum pressure = 7 Bar
2. Displacement of compressor = 100 cc.
3. Delivery rate = 0.864 cub.m/min.

ACKNOWLEDGEMENT

We are totally thankful to each and every sole which has sincerely contributed in the blossoming of our project. The following is a few who have kindly assisted in our work.

Our sincere thanks to our Principal **Dr. K.K. Padmanabhan M.E,Ph.D, Dr. T.L. Seetharama Rao M.E., Ph.D** for their keen cooperation and motivation for the structuring of our project.

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Our special thanks to **Mr. Sasi, Mr. Ganesh Prabhu, Mr. Sukumar,** for their gracious presence with us in spending a lot of their valuable time and giving life to our project.

We are also bound to thank all our staff, friends supervisors and Workers who have made this dream come true.

INTRODUCTION

There is a need to improve the performance of compressors by several methods, Such as design shape and construction are known and screw type compressor are the latest design as on date. The power required to compress the air to a known pressure is given in the attached tables. One cubic root of air equals approximately 28 litres its volume, we brought of a novel idea to design the triangular compressor as per the attached sketches or diagrams with this.

It is a three cylinder compressor placed radially equally apart such that the cylinders openings tend to meet on a common triangular compression chamber that has one inlet and outlet valve to take air from the atmosphere and push on through the outlet value to a tank for receiving air in the compressed state. As an example let us take a 100cc displacement single cylinder compression, e.g. that will have a bore of 50mm and stroke 50mm. The volume displaced is

$50^2/4 \times \pi \times h = 25^2 \times 3.14 \times 50$

$= 98.125$ cc

[nearly] $= 100$ c.c

To have the identical capacity with 3 cylinders for above 100cc capacity in each cylinder, displacement must be 100/3 = 33.33cc each. In the market sprayer 2 stroke engine has a displacement of 34 cc 3 times that equals 102 cc, the bore is 35 and stroke is 34.

i.e Swept volume of each = $35^2/4 \times \pi \times 34$

For single cylinder, swept Volume = 32.695

3 Cylinders = 98.1 cc

The compressing area of a single 100 cc compressor as said above is 50 Q or 1962.5 mm^2.

The Compressing area of 35cc piston in 961.625 mm^2 for 3 pistons are = 2884.875 mm^2

This is bigger by = 922.375 mm^2

Now all 3 pistons are made to compress air simultaneously on to a Common compression triangular chamber over shorter stroke and so the isothermal efficiency will be better than the single cylinder.

Adiabatically also this will be advantages since 3 cylinders displacement is pushed at the same velocity and force on the common chamber because the three piston move at equal velocity driven by one chain to achieve this 3 compressors cranks sprockets as teeth. For every rotations of crank each pistons will move once from TDC to BDC and BDC to TDC two strokes. Therefore theoretically 98cc of air will be taken in and compressed to the volume of smaller space in the common compression chamber which we have made as to **15 bore x 70 long = 18.78 cc or 7 times the total 98cc volume = 7 atmospheres.** If space is smaller, the pressure will be more and bigger the lower. If compressor is made to work at 1400 rpm air taken will be 98 x 1400 = 137.2 litres @ 7 atmospheres pressure the advantage with triangular comprressor will be low vibration, smaller unit giving more output and so cheaper to make, ideal for air compressor in Air conditioning and Refrigeration, Vacuum pumps and General purpose usage.

GENERAL ASPECTS OF A RECIPROCATING COMPRESSOR

1. SIGNIFICANCE OF COMPRESED GASES AND VAPOUR
2. COMPRESSOR DEFINITION
3. CLASSIFICATION
4. RECIPROCATING COMPRESSOR TERMINOLOGY
5. THERMODYNAMICS PRINCIPLE
6. WORKDONE BY COMPRESSION
7. COMPRESSOR COOLING
8. SUMMARY

SIGNIFICANCE OF COMPRESSED GASES AND VAPOUR

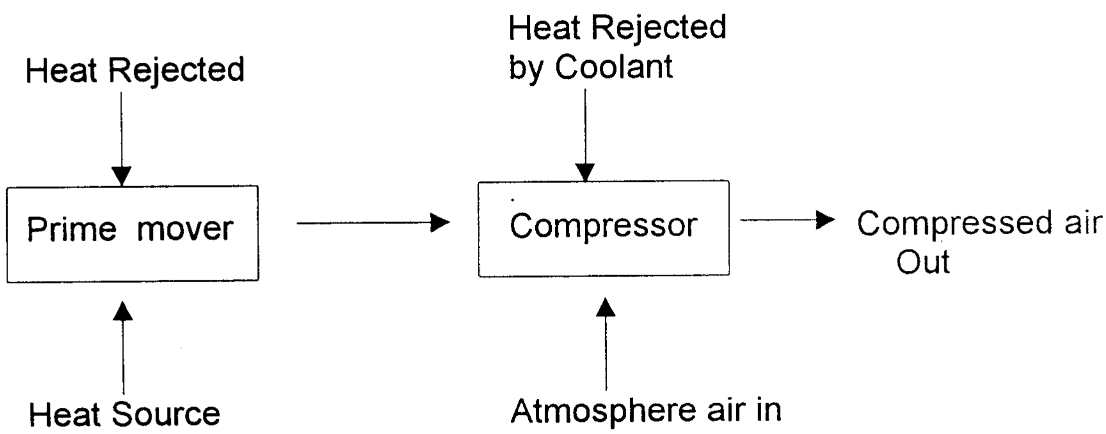
In industry and in economic life, in general the use of compressed gases and vapour are steadily increasing various gases are after required purposes and pressure for many chemical and industrial purpose and compressed air specially has numerous productivity is several fields such as metallurgy, chemical plants, hospitals, inflation of tyres hot air guns, to operate air driven hand tools such as die polishers, diegrinders etc.

In fact it would be difficult to find a branch of industry where utilization on compressed air would not affect a material rationalization of manufacturing process the advantages of pneumatic machine and tools are their safety simplicity and ruggedness combined with comparatively low weight of great importance is the use of compressed vapour in refrigerating plants which improves our standards of living by economic handling of perishable foods.

COMPRESSOR DEFINITION

Compressor is a machine which provided air at high pressure and work must be done upon the air by external agency.

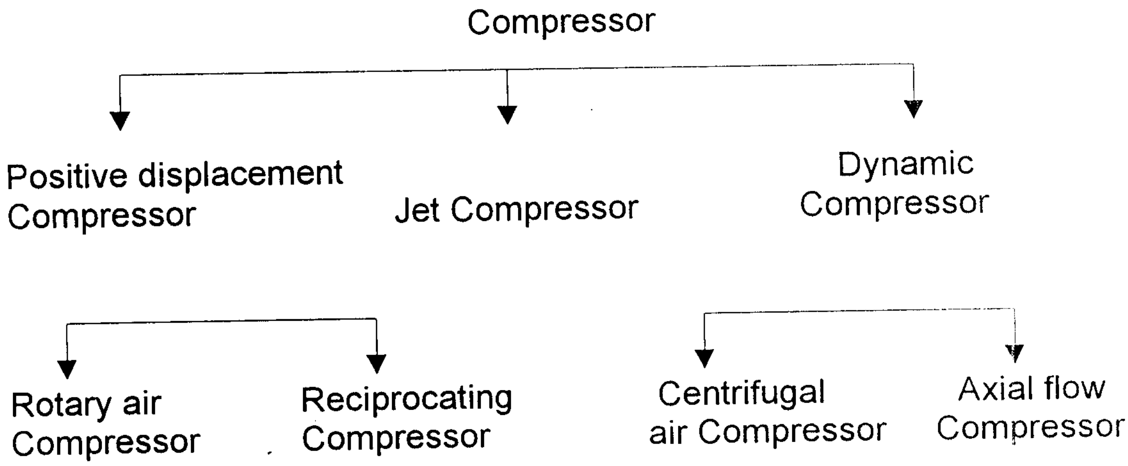
An air compressor takes in atomosphere air compresses it and delivery the high pressure air to a storage vessel from which it may be conveyed by a pipe line where ever supply is required.



The fig shows schematically the general arrangement of compressor set of the energy received by compressor from the prime movers, some will be lost to radiation and any coolant employed to cool the compressor and the rest will be maintained within the delivered air at high pressure.

So far as compressor above is considered the energy which it reaches is that available at the shaft of prime movers.

CLASSIFICATION



In positive displacement compressor the volume of the air reduced there by pressure is increased.

In dynamic compressor make use of high velocity nozzle to impart kinetic energy and later convert it to pressure energy by means of diffusers.

Reciprocating Compressors

This type machinery work in your discrete phase namely.

1. Expansion
2. Suction
3. Compression
4. Discharge

2. **Double acting Compressor**

In this type suction, Compression and delivery of air takes place on both side of the piston such compressor have two delivery strokes per revolution of the crank shaft.

SINGLE STAGE COMPRESSOR

The Compression of air from initial pressure to the final pressure is carried out in one cyclinder only.

MULTISTAGE COMPRESSOR

The compression of air from the initial pressure to the final pressure is carried out in more than one cylinder, the air passing in series through these.

Compression ratio (or) Pressure ratio :

It is the ratio of the absolute discharge pressure to the absolute inlet pressure.

Free air delivered (FAD)

The volume of air delivered under the condition of temperature and pressure existing at the compressor in take (ie) volume of air delivered at surrounding air temperature and pressure.

Which are accomplished by the reciprocating motion of the piston inside the cylinder by this we can achieve high pressure at relatively low Capacities.

The basic elements of a reciprocating Compressor are shown in fig. This piston inside the cylinder is used to pressure the gas.

The connecting rod and crank mechanism is incorporated for the Conversion of rotary motion to linear motion.

When the piston is moved down, the Pressure inside the Cylinder falls below the atmospheric pressure this makes the inlet valve to opens, when the piston comes to BDC the inlet valve closes. During upward motion of piston the air get compressed. At a certain pressure (The outlet valve opening Pressure) the outlet value opens when the pressure inside the Cylinder in discrete is required. This air can be stored in the tank and can be used when ever necessary.

RECIPROCATING COMPRESSOR TEMERMINOLOGY

1. Single Acting Compressor

In this type the suction, compression and delivery of air take place on one side of the piston only. Such compressor have one delivery stroke per revolution of the crank shaft.

Displacement of the Compressor

The swept volume of the piston in the first cylinder is known as displacement of the compressor.

It is given by $\pi R^2 L$

Where R is radius the cylinder bore.

L is stroke of piston.

Actual Capacity of the Compressor

The actual free air delivered by the cylinder per minute is known as capacity of the compressor. It is given in cubic meter of free air.

Volumetric efficiency

The ratio of actual free air delivered by the compressor per stroke to the displacement of the compressor is known as volumetric efficiency of the compressor.

Compressor Efficiency

This is the ratio of theoretical horse power to brake horse power.

Theortical horse Power

This is the horse power required to compress adiabatically the air delivered by a compressor through the specified pressure range, without provision for loss of energy.

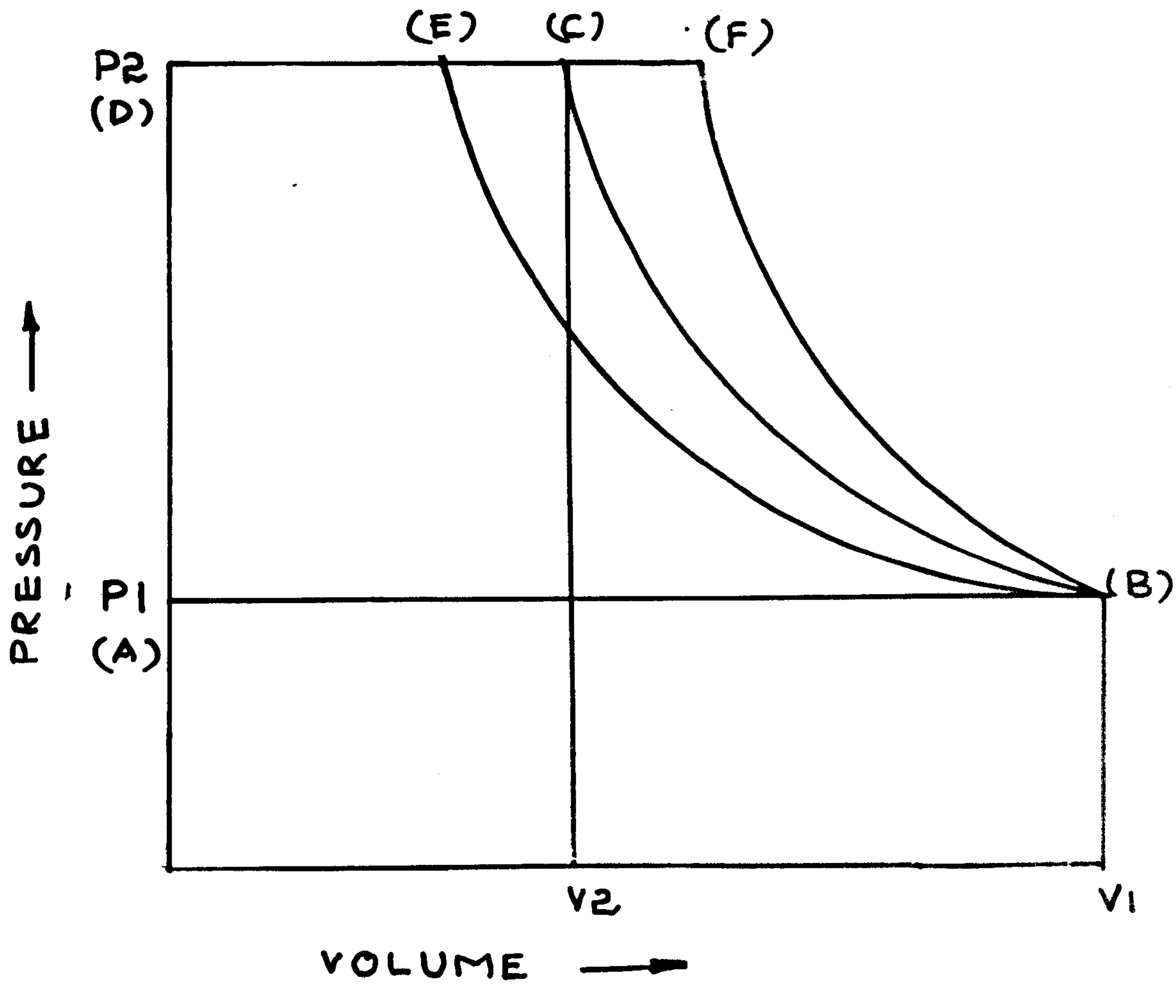
Thermo Dynamics

Isothermal, adiabatic and polytropic process are the different types of process considered. In an isothermal process PV = Constant as T remain Constant. As the compressor is running at high speed, Temperature cannot be maintained constant. So practically it is impossible to have isothermal process in a compressor. But when a gas is compressed or expanded, it has been established that the pressure will vary to an exponetial power of the volume is PV = Constant.

This type of relationship for the change of state where no heat is lost or friction is in current is known as adiabatic process. A perfect adiabatic process which is reversible is called as an isotropic process. Industrial compressors reject heat and have valve and ring leakage and are also subjected to generation of frictional heat.

Thus actual compression and expansion process differs from the ideal isotropic process and are known as polytropic process. A polytropic process differs from an adiabatic process in that change of state does not take place at constant entropy.

They are Shown in Fig.



The ideal indicator diagram in fig ABCD illustrates the action in the compressor. Assuming water jacket cooling bat with no internal losses by friction or eddies.

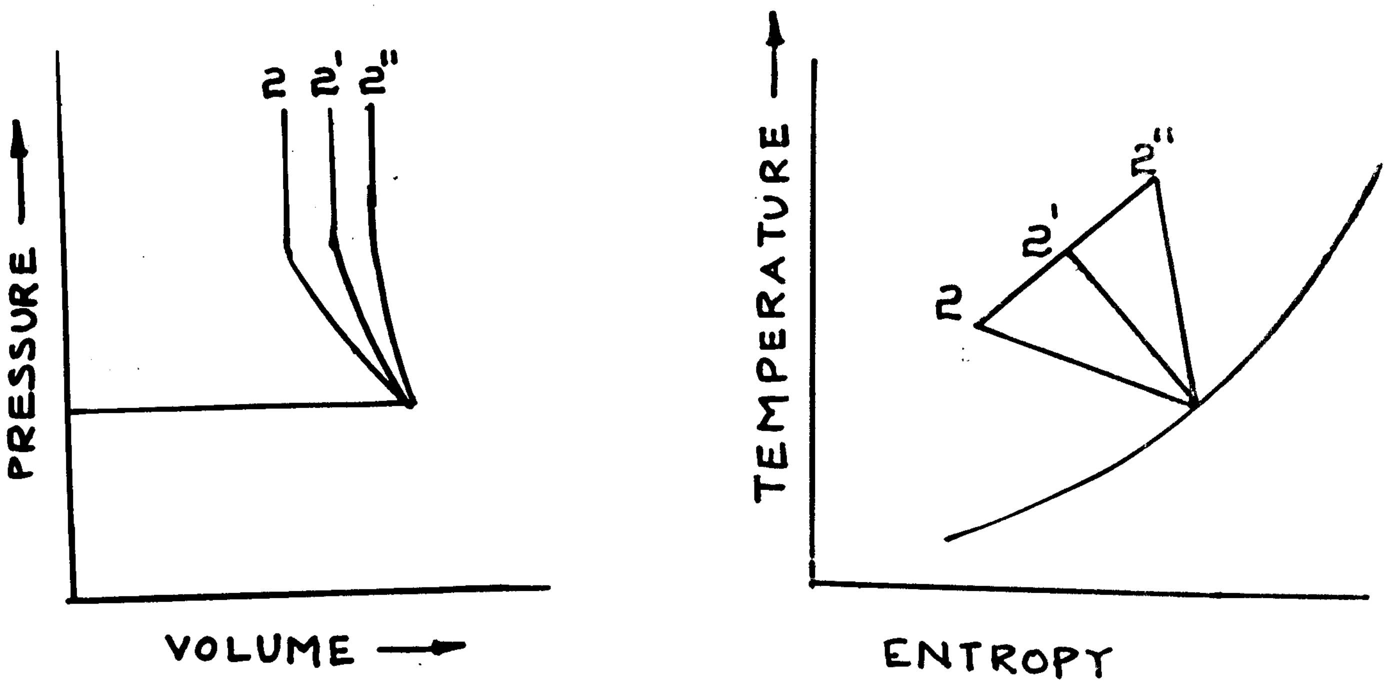
During the suction stroke AB, Volume (VI) of air flows into the cylinder at atomospheric pressure (P_1) absolute and the work on the Piston is P_1V_1.

During the compression stroke BC($PV^n = C$) the work done on the air during compression to absolute pressure (P_2) and volume V_2

$$W = (P_2V_2 - P_1V_1) / n-1$$

When there is a water around the cylinder 'n' may vary from 1.35 to 1.25 for thoroughly efficient cooling and in the special case of cooling the air during compression by spraying water into the cylinder which drained away from the discharge air, may be 1.2 lower value of are probably due to leakage part of the Piston or Valves.

ED is the delivery of air from the cylinder to receiver at constant pressure P_2 and work done during this stroke is P_2V_2.



1-2 → ISOTHERMAL COMPRESSION

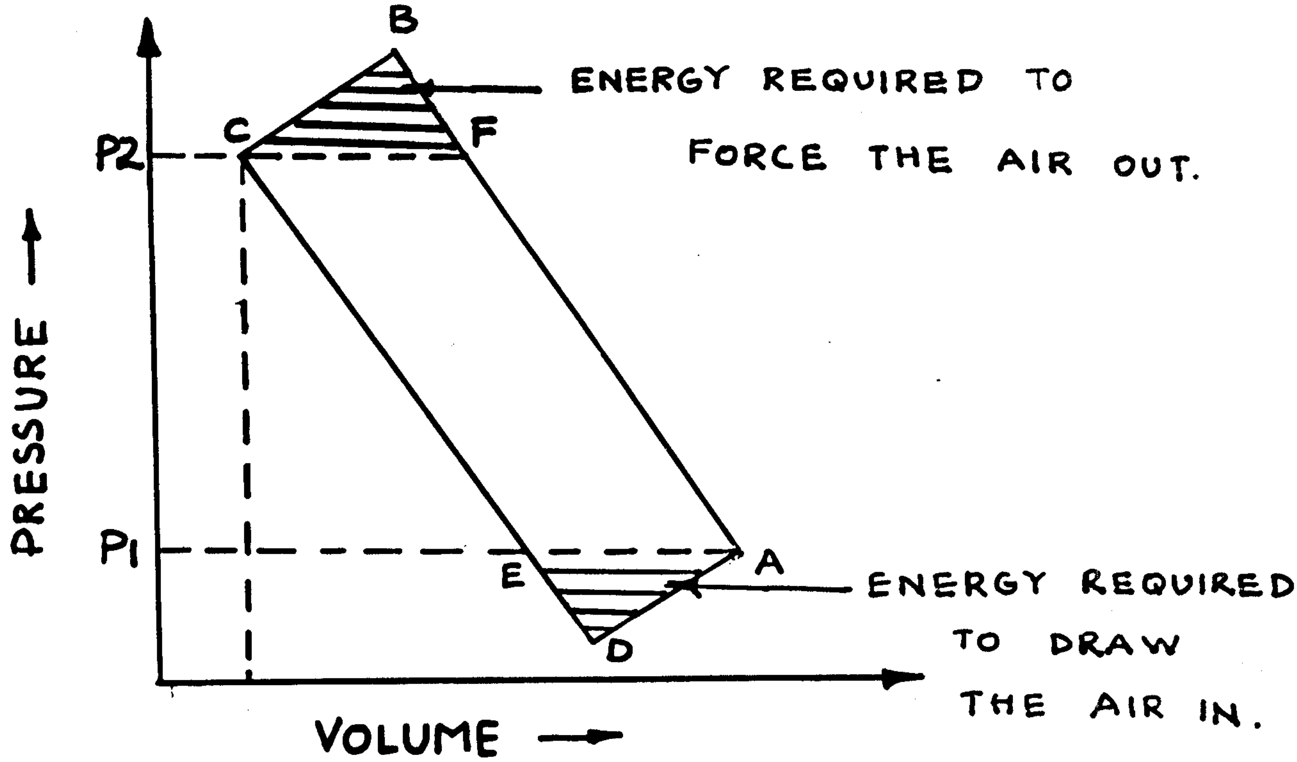
1-2' → POLYTROPIC COMPRESSION

1-2'' → ISENTROPIC OR REVERSIBLE ADIABATIC COMPRESSION

WORK ON COMPRESSION

Adiabatic Compression

In the reciprocating gas compressor work is performed by the piston, during gas from the suction line and displaying it into the discharge line for complete cycles. Piston is restricted by the pressure difference between the discharge line and the suction line.



In the above wire drawing valve pressure losses are indicated in shaded section below suction pressure AE and above discharge, pressure CF.

The ideal work involved in moving the piston through the cylinder stroke is represented by the area AFCE in fig. The ordinates P_1 & P_2 represent the suction and discharge line pressure.

The ideal path of the AB and CD is known to follow. The adiabatic process.

$P_1V_1^n = P_2V_2^n$ = Constant

By applying adiabatic relationship and intergrating above equation the work of compression becomes.

Work on compression = $n/n-1 \; p_1v_1 \, (P_2/P_1)^{n-1/n} - 1)$

Where P_1 is the initial pressure (atms pressure).

V_1 is the volume occupied by air.

(Stroke volume + Clearence Volume)

Where P_2 is the final pressure after compression to calculate the power required to drive the compressor we have the expression.

Power = P_m LAN/60 Kw.

Where

P_m is the mean effective pressure KN/hr.

L is the stroke length in m.

A is the cross section area of piston in m^2

N is Speed in rpm.

Mean effective pressure = Workdone / Volume

$P_m = n/n\text{-}1 \quad P_1 \, (R_c^{n-1/n} - 1) \times \eta_{vol}$

Where

R_c is the compression ration (P_2/P_1)

R is volumetic efficciency

Volumetric efficiency = free air delivered / stroke volume.

Compressor Cooling

When a gas is compressed adiabatically pressure P_1 to P_2 the temperature of gas raised from T_1 to T_2. The increase in temp Occurs accordingly to the following relation

$$\frac{T_2}{T_1} \quad (P_2 / P_1)^{n-1/n}$$

Therefore when the stage pressure ratio is high the effect of the raise temp is prominent therefore cooling of the gas is essential.

The general types of cooling arrangements available are

a. Water jacket cooling

b. Air Cooling by providing fins

c. Inter cooling and after cooling

here air cooling is done by providing for arrangement.

Summary

From the theory of reciprocating Compressor it is clear that single stage reciprocating compressor working in polytropic process is suitable for low pressure compressor.

DESCRIPTIONS AND FUCTIONS

1. Working principle of compressor with Common Compression Chamber.
2. Design Consideration
    a. Common Compression head
    b. Cylinder
    c. Connecting rod
    d. Crank shaft
    e. End bearing
    f. Power transmission
    g. Power transmission
    h. Air filter
    i. Gasket
    j. Service Value
    k. Lubrication system
    l. Cooling fan
3. Merits & Demerits
4. Applications

2. Design Considerations

The design consideration of various parts of compressor with common compression chamber are given below .

a. Common Compression Head

The cyclinder head is made of high grade alloy of aluminium. They cyclinder head here serves as the common compression chamber and to hold all the three cylinder together. The head is made up of triangular shape hence the maching will be some what difficult. the valve plate must therefore have good support so that there will be no leakage at the gaskets on either side of the valve. The cylinder head is attached to the cylinder.

b. Cylinder

The compressor cylinder made up of cast iron. The cast iron must be done enough to prevent the leakage of air though it compressor has fins cast with cylinder and cylinder head to provide better air cooling.

C. Connecting Rod

The connecting rod which connects the crank shaft and piston, is made up of mild steel.

D. Crank Shaft

Reciprocating compressor must use some means tochange the rotary motion of the motor into reciprocating motion of the compressor. The crank shaft in these design is forged steel. This compressor uses an eccentric fastered to a straight shaft in place of connectional crank shaft. This construction is used to reduce vibration and to remove the need for connecting rod caps and bolts.

The crank shaft main bearing supports the crank. They also must carry away the end loads crank shaft and connecting rod bearings are fitted with greater accuracy.

E. End Bearings

Antifrictional bearings are the most suitable for end bearings. They are available in standardsizes, maintenance less, durable, early replaceable and they have good load carrying capacity. They take up variable loads. With the above advantages they are most suited for compressor end bearings. Selection of the appropriate bearings for the compressor depends on life in working hours and the format of loading.

F. Valve

The valve assembly consists of a valve plate an intake valve, a delivery valve retainer, valve plate is made of hardened steel that can be thinner with carban wearing valve seats. Compressor valves are made of high carbon alloy steels. They are heat treated to give them the properties of spring steel and ground to a perfectly flat surface.

The intake valve is kept in place by small pins or the clamping action between the compressor head and valve plate. Exhaust value is also clamped in the same way the value needs or disks must be perfectly flat. A defect of 0.00254 mm will cause values to leak.

Of the two values, the intake gives the least trouble. This is because it operator at a relatively cold temperature. The delivery valve must be fitted with special care it operator at high temperature and must be leak proof against a relatively high pressure difference.

G. Power Transmission

The compressor being of maximum discharge and the pressure is less. But there should not be slip because the piston should move in phase. So we go for chain drive.

H. Air Filter

Dry filter is fitted to the air suction of the compressor to eliminate the atmospheric dust. The filtering elements must be cleaned periodically.

I. Gasket

The joining surface between bolted parts such an cyclinder heads, valve plates, crank case opening etc are sealed with gaskets. The gaskets are made of special paper gaskets which are free from moisture.

J. Service Valve

The lubrication part consist of a small oil sump which is connected to the inlet valve, which on running will drop few drops of oil on the inlet valve which enter the cylinder and lubricate the necessary parts.

Cooling Fan

The cooling fan is provided with the flywheel to reduce the temperature of the working range of the compressor. Since the compressor runs at 1440 rpm extensive heat may be produced due to friction between the cyclinder and piston which may lead to increase the temperature of outlet air. By providing a cooling fan the temperature upto 3 to 5 can be reduced.

Comparision of Convential one with ours

We know of several types of compressors made in many renowned factories at Coimbatore for various applications piston type, screwed type value type etc., But three pistons compressing air into a common compression chamber is not known to be existing and so we decide to make this novel design to prove its superiority to other in cost and performance.

Any new product must be novel and useful it must be superior is performance, reliable and dependable compared to known models.

The cost of making must be comparatively low and in the market it should be capable of Competiting similar products, is every way, colour appearance, value etc.

Range must be based on from prospective users and customers. However first working model is only to prove superiority as per maker's claims.

When it is compared with the existing ones it has very lesser vibration and lesser oil consumption and it is move reliable than the conventional ones because havings less vibration and also it is an Novel-idea.

DESIGN

DESIGN OF TRIANGULAR COMPRESSOR COMPONENTS

1. Theoritical horse power calculation
2. Mass flow rate of air
3. Temperature rise
4. Design of piston
5. Design of Cylinder
6. Design of fins
7. Design of common compression head
8. Design of Connecting rod
9. Design of Crank Shaft
10. Design of Chain drive & Sprockets
11. Selection of bearing

HORSE POWER CALCULATION

For One Cylinder of 35mm Bore and 34mm Stroke

i) Swept Volume

Swept Volume $= \pi/4 \times D^2 x L$

$= \pi/4 \times 35^2 \times 34$

$V_S = 32711.8 \text{ mm}^3$

$V_S = 32.7 \text{ cm}^3$ (or) 32.7 cc

In this compressor, 3 cylinder of same stroke and bore are used.

So, Total swept Volume = 3 x V_S

$V_S = 32.7 \times 3$

$V_S = 98.135 \text{ cc} \cong 100 \text{ cc}$

ii. Clearance Volume

Here the common compression chamber is taken for clearance volume

Bore = 15mm

Stroke = 70mm

$V_C = \pi/4 \times D^2 x L$

$= \pi/4 \times 15^2 \times 70$

$V_C = 12.37 \text{ cc}$

Volumetric Efficiency

$$\eta_V = \frac{\text{Actual free air delivered}}{\text{Displacement of Compressor}}$$

$$\eta_V = \frac{V_a}{V_s}$$

$$\eta_v = 1 - \frac{V_c}{V_s}\left[(P_2/P_1)^{1/n} - 1\right]$$

$$= 1 - \frac{12.37}{98.135}\left[(7/1)^{1/1.3} - 1\right]$$

$$\eta_v = 56.29\%$$

REFERENCE BOOK : GOTHANDARAMAN

iv Initial Volume

Volume of the air to be Compressed

$$V_1 = \eta_{v/100} \, x V_2$$

$$= \frac{56.29}{100} \times \pi/4 \times 35^2 \times 34 \times 3$$

$$V_1 = 55.24 \text{ cc}$$

Where,

V_2 = Swept Volume, D = 35mm, L = 34 mm

$$V_2 = \pi/4 \, D^2 L$$

V. Workdone by the Compressor

Workdone $W = \frac{n}{n-1} P_1 V_1 [(P_2/P_1)^{n-1/n} - 1]$

$$\frac{1.3}{1.3 - 1} \times 1.013 \times 10^5 \times \frac{55.24}{1440 \times 60} [7/1.013)^{1.3-1/1.3} - 1]$$

W = 0.157 KW

Converting kw into HP,

Power = 0.7 HP ≅ 1 HP

DESIGN OF HEAD

In Triangular air compressor with common compression chamber, the head design plays on important role.

The cylinder are at 120° to each other. As the breath of the cylinder block is about 56 mm. Therefore the sides of the head is designed to 56 mm.

The drill at the centre where the valve should be fitted will be at the size of 12.7 mm. Because the dia of the inlet value is 12.7 mm and the outlet is about 14.9 mm.

To fix the cylinder with the common compression chamber, the groove of 8mm thickness is taken at the three sides.

Specification of the Compressor head

Length	=	70mm
Breath of head	=	56 mm
Dia. Of central drill	=	12.7 mm
Dia of the drill which connected The piston with central drill	=	15 mm

Mass Delivered per Min

$P_1.V_1 = P_2V_2^n$

$(V_2 / V_1)^n = P_1 / P_2$

$V_2 = [P_1/P_2]^{1/n} \times V_1$

$[1.013 / 7]^{1/1.3} \times 55.24$

$= 12.49$ cc / stroke

$V_2 = 0.0012\ m^3$ / stroke

$V_2 = 0.0012 \times 1440/2\ m^3/min$

$V_2 = 0.9\ m^3/min$

We know that,

$P_2V_2 = MRT_2$

$7 \times 10^5 \times 0.9 = m \times 287 \times 474.04$

$m = 4.5$ Kg/min

Where,

$T_2 = T_1 \times [P_2/P_1]^{n-1/n}$

$= 303 \times [7/1]^{1.3-1/1.3}$

$T_2 = 461.04°K$

$T_1 = 201.04°K$

DESIGN OF PISTON

We know that,

Diameter of Piston = Bore dia = 35 mm

Thickness of the piston head = t

$t = D\sqrt{(3/16 \times P/F)}$

Where,

P = Max. Compression Pressure = 7 bar

Piston is made up of aluminum alloy. The permissible tension stress = 34.6 N/mm^2.

REFERENCE BOOK : MACHINE DESIGN, PAGE NO. 912, PANDYA & SHAH

$t = 0.035 \sqrt{(3/16 \times 7 \times 10^5 / 34.6 \times 10^6)}$

$t = 2.5 \text{ mm} \cong 3 \text{ mm}$

No. of piston rings = $2\sqrt{D}$

Where 'D' should be in inchas

D = 35 mm = 1.49 inchas

D = $2\sqrt{1.49}$

No. of rings = 2.4

2 - Compression ring

1 - Oil ring

Thickness of wall under piston ring = 3 mm

Thickness of the ring = D/32 = 35/32

= 1.09 mm

Width of the ring = D/20

= 1.75 mm

The distance of the first ring from top of the piston equals

= 0.1D

0.1 x 35

= 3.5 mm

Width of the piston land between rings,

= 0.75 x 1.75

= 1.5 mm

Length of piston = 1.6250

= 56.8 mm

Length of the piston skirt =

Total length – Distance of first ring from top – (No. of landing between rings x width of land) – (No. of Comp. Ring x width of the ring) =

= 56.8 - 3.5 – (2 x 1.5) – (2 x 1.75)

= 46.8 mm

Centre of the piston pin above the centre of the skirt equals to

$0.02D = 0.02 \times 35$

$= 0.7 \cong 1$ mm.

Distance between the centre of the piston pin and the bottom of the piston $= \frac{1}{2} \times 46.87 + 1.5$

$= 24.9$ mm

Thickness of piston wall at open ends.

$= \frac{1}{2} \times 3 = 1.5$ mm

The bearing area provided by piston skirt = $46.875 \times 35 = 1640.6$ mm^2

Specification

Dia of the piston	=	35 mm
Thickness of the piston head	=	3 mm
No. of compression ring	=	2
No. of oil ring	=	1
Thickness of wall under piston ring	=	3 mm
Thickness of ring	=	1.09 mm
Width of the ring	=	1.75 mm
Length of Piston	=	56.8 mm
Length of piston Skirt	=	46.8 mm

DESIGN OF CYLINDER

Known Value

Stroke Length = 34 mm

Max. Pressure = 7 bar = 7 Kg/cm^2

Material used (Cast Iron) F_t = 1730 Kg/Cm^2

Dia of piston, d_p = 35mm

Force acting on Cylinder during Compression

$F = \pi/4 \; d^2_p \times P$

$\pi/4 \times 3.5^2 \times 7$

$F = 68$ kgf = 667 N

REFERENCE BOOK : MACHINE DESIGN, PAGE No. 906, AUTHOR PANDYA & SHAH

Thickness of the Cylinder wall,

$T = r_p \{\sqrt{f_t} + P) / (f_t - P) - 1\}$

$= 17.5 [\sqrt{} [(1730 + 7 / 1730 - 7) - 1]$

$t = 1.57$ mm

Taking factor of safety as 3mm

$t = 1.5 + 3 = 4.5$ mm

Out side dia of the cylinder

$d_o = 35 + 2 \times 4.5$

$d_o = 4$ mm

Length of the Cylinder,

= Stroke length + Length of the piston

= 34 + 56.8

= 90 mm

Specification

Internal diameter	=	35 mm
External diameter	=	44 mm
Thickness of the wall	=	4.5 mm
Length of the Cylinder	=	90 mm

FIN DESIGN

During compression stroke, more amount of heat will be produced. To dissipate the heat from the compressor, the fins are designed.

Material for fin ——▶ Cast Iron

The following values are obtained from fleat & mass transfer data book.

- Thermal conductivity, K = 72.7 w/m°k
- Fleat transfer Co. efficient, h = 24.6 w/m^2k

HEAT & MASS TRANSFER DATA BOOK – PAGE NO. 41

Perimeter of the pin

$P = \pi$ (d + thickness)

π (44 + 4.5)

P = 0.156 m

Area of the fin,

$A = \pi \times d^2/4$

$= \pi \times 44^2/4$

$A = 1.530 \times co^{-3}\ m^2$

Amount of heat transfer , $Q = \sqrt{hpkA}\,(T_o - T_1)$

$Q = \sqrt{(25.4 \times 0.150 \times 72.7 \times 1.52 \times 10^{-3}} \times (470 - 201)$

$Q = 174$ Watts

REFER HMT DATA BOOK PAGE No. 42

Fin Efficiency

Length, $L_c = L + t/2$

$= 100 + 4.5/2$

$= 102.25$ mm

$L_c = 0.102$ mm

Surface Area,

$A_S = 2 \times L_c$

$= 2 \times .102$

$A_s = 0.204\ M^2$

$A_m = tL_c$

$= 4.5 \times 102.25$

$Lc^{1.5}(h/Kam)^{0.5} = (0.102)^{1.5}\ (25.4/\ 72.7 \times 0.46)^{0.5}$

$= 0.28$

$r^2/r^1 = 1022 + 22\ /\ 22$

$= 124.2 / 22$

$= 5.6$

For the value $r_{2c} / r_1 = 5.6$ and $L_c^{1.5} (h/kAm)^{0.5} = 0.28$

The fin efficiency is ontained from the graph given in HMT data book.

$$\eta_{Fin} = 84\%$$

Therefore, the actual heat transfer, $Q_{act} = \eta \times Q$

$$Q_{act} = 0.84 \times 175$$

$$= 147 \text{ watts}$$

Specification

Type	=	Rectangular type
Perimeter	=	0.152 m
Area of the fin	=	1520 mm^2
Efficiency of fin	=	84%
Heat Transfer, Q	=	147 Watts

DESIGN OF CONNECTING ROD

- Diameter of Piston = 35 mm
- Mass of Reciprocating Parts = 0.3 Kg
- Length of connecting Rod from Centre to centre L = 80 mm
- Stroke length = 34 mm
- Max speed = 1440 rpm
- Compression Ratio = 7.1
- Max. pressure = 7 bar = 0.7 N/mm^2

We know that radius of the crank.

$$r = \text{stroke length} / 2$$

$$r = 34/2 = 17 \text{ mm}$$

Ratio of length of connecting rod to the radius of the cranks.

$$n = l/r = 80/17$$

$$n = 4.7$$

DATAS AND FORMULA → P.S.G. DATA BOOK, PAGE NO. 7.123

- Max pressure force acting on piston,

$$F_G = \pi/4\ D^2P$$

$$= \pi/4 \times 35^2 \times 0.7$$

$$F_G = 0.673 \times 10^3 N$$

- Max angular speed $= W_{max}$

$$= \frac{2\pi N_{max}}{60}$$

$$W_{Max} = 150.8 \text{ rad/sec.}$$

- Max. Raciprocating Inertia force

$$F_I = mw^2_{max}(\cos\theta + \text{Cos } 2\theta/n)$$

When $\theta = 0$, ie., when the crank is at inner dead centre, the force acting on piston will be maximum.

$$F_I = Mw^2r \quad (1 + 1/n)$$

$$= 0.3 \times 150.8^2 \times 0.017\ (1 + 1 / 4.7)$$

$$F_I = 0.14 \times 10^3 N$$

Force acting on the connecting rod is equal to the maximum pressure force on the piston.

$$F_c = F_l = 0.673 \times 10^3 N$$

The section is 'I' section. Connecting rod is designed for buckling about perpendicular axis. Assuming both ends are hinged,

Taking factor of safety = 6

The buckling Load, $W_{cr} = F_c \times 6$

$$W_{cr} = 0.673 \times 10^3 \times 6$$

$$W_{cr} = 4.04 \times 10^3 N$$

- Radius of Gyration, $K_{xx} = \sqrt{(I_{xx}/A)}$

$$\sqrt{(419\, t^4/12) \times 1/11t^2)}$$

$$K_{xx} = 1.78\, t$$

We know that equivalent length of the connecting rod for both ends hinged,

L = 80 mm

F_c = 320 N/mm^2

a = 1/7500

By Rankines formula

W_{cr} = $F_cA / (1 + a (L / K_{xx})^2$

4.04×10^3 = $320 \times 11t^2 / [(1 + 1/7500) (80/1.78t)^2]$

1.14 = $t^4 / [t^2 + 0.269]$

$1.14t^2 + 0.3066 = t^4$

t^2 = $1.14 \pm \sqrt{(1.4^2 - (4x1x(-0.3066) / 2))}$

t^2 = $1.14 \pm 1.75/2$

t^2 = 1.96

t = 1.3 mm

Specifications

- Height = 5t = 5x 1.3 = 6.5 mm
- Width = 4t = 4 x 1.3 = 5.2 mm
- Thickness of Web and Flange, t = 1.3 mm
- Depth of big end = 1.1 x 6.5 = 7.15 mm
- Depth of small end = 6.5/1.1 = 5.90 mm
- Length of connecting rod from centre to centre = 80mm

SELECTION OF BEARING

Type : Deep Groove ball bearing

REFERENCE : PSG DDB PAGE NO. 4.2

- Bearing Life = 3 years
- Working hours = 1500 hours / year

Life, L = 3 x 1500 = 4500 hours

L = 4500 x 1440 x 60 revolution

L = 388.8×10^6 revolution

- Total load on the bearings = $0.673 \times 10^3 N$
- Central Load acting on each bearing = $0.336 \times 10^3 N$
- Refering from Design Data Book,

Equivalent Load, $P = (XF_r + YF_a)\ S$

F_r, Radial Load = $0.336 \times 10^3 N$

F_a, axial load = 0

From P.S.G. Data book,

X = 1 and Y = o

S = 1.3 for reciprocating m/c's

$P = (1 \times 0.336 \times 10^3 + 0)^{1.3}$

$P = 0.436 \times 10^3 N$

Dynamic Capacity

$$C = (L/L_{10})^{1/k} \times P$$

Where,

$K = 3$ for ball bearing

$L_{10} = 10^6$ revolution

$$C = (388.8 \times 10^6/10^6)^{1/3} \times 0.436 \times 10^3$$

$$C = 3.119 \times 10^3 N$$

Therefore refering from Data Book we select "SKF 6202 (ISI 25BC03)' for big end of the connecting rod.

DESIGN OF CRANK SHAFT

Let 'P' be a force acting on the Cylinder

l = Length of crank pin

d = Diameter of crank pin

f = Permissible tensile stress intensity in the pin material

p = safe bearing pressure

Assuming crank pin to be a simply supported beam,

MACHINE DESIGN - PAGE NO : 947, PANDYA & SHAH

Max, bending moments is,

$$M = pl/8$$

Where,

$$P = p.d.l, \quad l = P/pd$$

Therefore,

$$M = p^2/8\ pd$$

By equating resisting moment with the bending moment,

$$\pi/32x4\ d^3f = p^2/8pd$$

$$d = 4\ \sqrt{32P^2/8\pi pf}$$

$$d = 4\sqrt{4P^2/\pi pt}$$

Force acting on the crank pin,

$$P = \pi d^2 4 \times p$$

$$= \pi/4 \times 3.5^2 \times 7$$

$$P = 67.34 \text{ kg}$$

Safe bearing pressure for the material,

$$P = 280 \text{ Kg / cm}^2$$

Permissible tensile stress for the material,

F = 1730 Kg/cm^2 (From Data Book)

$$d = 4\sqrt{(4x (67.34^2 / \pi \times 280 \times 1730)}$$

$$d = 0.33 \text{ cm}$$

$$d = 3.3 \text{ mm}$$

- Thickness of the Web. $T = 0.7d$

  $= 0.7 \times 4 = 2.8$ mm

  Say $t = 3$ mm

- Distance between web $= 1.1\, d$

  $1.1 \times 4 = 4.4$

  Say $= 5$ mm

- Width of the Web $= 1.14\, d$

  $1.14 \times 4 = 4.56$

  $W = 4.6$ mm

DESIGN OF CHAIN DRIVE

⇒ Power = 1 HP = 736 watts

⇒ Speed = 1440 rpm

⇒ Centre distance between the pulley = 250 mm

⇒ Transmission ratio = 1

REFERENCE BOOK - MACHINE DESIGN - PAGE NO. 712
PANDYA & SHAH

Design of Sprocket

Let us consider Z_1 = 14 (Spare available is minimum)

No. of teeth on the Driven sprocket.

$Z_2 = iZ_1$

$Z_2 = 14$

P_{max} = a/30 = 250/30 = 8.33 mm

P_{min} = a/50 = 250/50 = 5 mm

Near to the P_{max} Value, we (Choose the standard pitch length 12.7 mm.

P = 12.7 mm

We, choose, 6.15 M84 (Page No. 7.73)

Datas

- Roller Dia = 7.75 mm
- Width, W = 6.45 mm
- Bearing Area = 0.39 cm^2
- Bearing Load = 1500 Kgf
- Weight per meter = 0.49 Kgf

Tangential Force, P_t :-

P_t = 75M/V

P_t = 75 x 1 / 4.26

$$\text{Velocity, } V = \frac{\text{No. of teeth on sprocket x pitch x n}}{60 \times 1000}$$

$$= \frac{14 \times 12.7 \times 1440}{60 \times 1000}$$

V = 4.26 m/c

P_t = 75 x 1 / 4.26 = 17.57 kgf

P_t = 176.2 N

Centrifugal Tension

$$P_c = WV^2/g = \frac{0.49 \times (4.26)^2}{9.81}$$

P_c = 9.0N

Tension due to Sagging

P_s = K.W.Q.

K = 6

P_s = 6 x 0.49 x 6.25 x 10

P_s = 7.3 N

P_T = $P_t + P_s + P_c$

= 176.2 + 9.0 + 7.3

P_T = 192.5N

- **Design Load**

= K_s * P_t

K_s = $K_1 \times K_2 \times K_3 \times K_4 \times K_5 \times K_6$

K_1 = 1.25 (for mild stock)

K_2 = 1.25 (Fixed support)

K_3 = 1.25 (a_p < 25P

K_4 = 1 (Horz. Drive)

K_5 = 1 (drop lube)

K_6 = 1.0 (for 5 hrs/day work)

K_S = 1.25 x 1.25 x 1.25 x 1 x 1 x 1.0

K_s = 1.95

Design Load = 1.95 x 192.5

375.9 N

Factor of safety = Breaking load/design Load

= 15000 / 375.9

n = 39

From page No. 7.77, the actual factor of safety is 12.7, we get the actual factor of safety greater and hence design is safe.

Bearing stores on Rollers,

Induced Stress, δ = (Power x 75 x K_S / A x V)

$$= \frac{1 \times 75 \times 1.95}{0.39 \times 4.26}$$

δ = 88.02 Kgf /cm^2

δ = 88.02 kgf/cm^2 = 0.88 kgf/mm^2

Allowable, bearing stress,

δ = 1.97 kgf/mm^2

So induced stress (0.88 kgf/mm^2) is less than allowable stress (1.97 kgf/mm^2).

So Design is Safe.

Length of the Chain

Length = ARC 'AB' + BC + ABC 'CD' + DE + ARC 'EF'

+ GF + ARC 'HG' + AH

= 122.5 + 180 + 66 + 55 + π/180

(110 + 75 + 132 + 118) X 25 / 2

Length, L = 518.4mm

$$\text{No. of Links} = \frac{\text{Length}}{\text{Pitch length}}$$

= 518.4 / 12.7

= 40.8

No. of links on chain = 42 links

- No. of teeth of drive on contact

  = 14/360 x 75°

  = 4 teeth

- No. of teeth of Sproket (1) on contact

  = 110/360 x 14

  = 4.27

  = 5 teeth

- No. of teeth of sprocket (2) on contact

  = 14/360 x 132°

  = 5.05

  = 6 teeth

- No. of teeth of sprocket (3) on contact

  = 14/360 x 118°

  = 4.58

  = 5 teeth

Chain Drive Data

Simplex Drive 6.15 M84

Pitch	=	12.7 mm
No. of teeth	=	14
Length of chain	=	518 mm
No. of links	=	42 links

MATERIAL SELECTION

It is an important part of engineering practise the choice of the material for our machine requirements, the proper use of those materials including the devlopment of new ways of using them for greater effectiveness, all were or direct responsibility.

The selection of the material for the machine is reduced three broad constraints.

1. Service requirements
2. Fabrication requirements
3. Economic requirements.

The service requirements, of course are permanent the material costs stand upto service demand. Such demands commonly include dimensional stability, corrosion resistance strength, hardness, toughness, heat resistant.

Fabrication requirements include the possibility to shape the material and to join it to other materials. The assessment of fabrication requirements concern the question of machinability hardenability, ductility, castability and weldability, qualities which are some times difficult to asses.

Finally there are the economic requirement it is essential that the overall cost of machining and fabrication to be maintained to an optimum level without compromising on the quality.

Other factors which are considered for the selection of materials were

1. Availability
2. Economic (i.e) Lower initial cost
3. Easy to fabricate
4. Capacity to meet service demands
5. Easy handling
6. Durability
7. Appearance

The material used for fabrication of various parts of our compressor are : **Mild steel, Cast Iron, Cast Aluminimum.**

Mild Steel

This is low Carbon Steel with no precise Control over the Composition or mechanical properties. The cost is low in comparision with other steel and it with stand high pressure.

Cast Iron

It is the least expanision of all the metal that could be used for casting and hence it is considered first when a cast metal is being selected other metals are selected only when the mechanical and physical properties of grey cast iron are in adequate. Elastic modulus is only 9×10^4 N/mm^2.

Cast Aluminium

Cast Aluminium designation is 4250 it has good fluidity / pressure tightness and resistance to corrosion suitable for intricate casting, weldable. The chemical composition of IS 4250 contains 0.1% Cu 0.5% Mn, 5% Si, 0.6% Fc, 5% Mn, 0.1% Ni, 0.1% Zn, 0.2% Sn, 0.1% Pn,)-0.5% Sn rest aluminium.

FABRICATION DETAILS OF THE COMPRESSOR

1. Compressor Head
2. Cylinder Block
3. Connecting Rod
4. Crank Shaft Assembly
5. End Bearing
6. Stand

1. **Compressor Head**

The material chosen for the body of the compressor head is high grade aluminium alloy. The head is of triangular shape with an angle of 60° to each side. First an Aluminium rod of 80mm diameter and length of 70 mm is taken and it is turned and faced in Lathe as a triangle shape of required dimension.

Then groove is taken on the three sides inorder to fix the cylinder block to it. Then Central drill of dia of 12.7 mm is done in its length and the 3 drills of 15mm dia is done from exactly centre of the side to the central drill.

2. **Cylinder Block**

The material for cylinder block is Cast iron the Cylinder Block is first cast and then machined and also required surface are finished with suitable methods like boring and horning.

The outer diameter turning and facing of the top and bottom surfaces are done in a lathe. When boring is done in boring machine, care should be taken to machine it with accurate dimension.

3. Connecting Rod

The material chosen for connecting rod is mild steel. It is hardened to with stand the pressure of 7 bar. First the template is made with the actual dimensions and then gas cutting is done on the mild steel plate for the template dimension and then machined.

Drilling and then boring is done on the big end for the bearing dimension.

4. Crankshaft Assembly

The Crankshaft is of solid type made of cast iron to withstand the pressure developed by the bearing and the compressor components.

The Crank shaft is turned, hardened, grand and the ends are tapered. All the made tappers are checked using ring gauge upto 30° taper angle is provided.

The web is seperately machined and welded the end bearings are restricted between the web and end cover. The main connecting rod with its centre bearing is fitted with tight fit to the eccentric crank shaft assembly.

The Crank shaft has its centre of gravity off set from the axis of rotation of the Crank shaft. Hence it act as balancing weight.

5. End Bearings

The thrust ball bearings is provided on the cover plate to safe guard the crankshaft. The bearing takes up the variable loads at the reasonable speeds care should be taken while placing the bearing in the cover in the centre portion of the body thickness plate.

6. Steel

The stand for the Compressor is made in mild steel to withstand all the loads of the Compressor and motor assembly.

OPERATION AND USE

Examine the unit for transit damages

Electrical Connection

The electrical power supply must be connected to the motor through starter. Start the unit momentrily and observe the direction of rotating rotor. The direction should be clockwise. If the direction of rotation is not correct, change the direction by interchanging and of the two places in the starter.

Adjusting Chain Tension

The chain tension between the motor and Crankshaft flywheel must be correctly adjusted with proper tension. Otherwise the Compressor will not run of the required speed.

COST ESTIMATION

1. Cost of Components
2. Cost of machining
3. Turning
4. Milling
5. Drilling
6. Boring
7. Grinding
8. Honing

   Cost of labour

   Over head cost

   Total Cost

Cost is very important factor for any product by which the customer has to be satisfied. The cost of a product mainly depends upon the following factors.

1. Materials selected
2. Method of manufacturing
3. Accuracy required
4. Number of product produced
5. Labour hours required
6. Fixed costs involved eds.

The cost analysis give the various expenditure incurred and the ways by which the costs can be minimize.

COST OF COMPONENTS

S.No.	Component	Qty	Cost/Comp.	Total Cost
1.	L34 Crank Case Assembly	3	300	900.00
2.	6202 SKF Bearing	6	60	360.00
3.	Piston assembly	3	195	585.00
4.	Cylinder block L 34	3	265	795.00
5.	Oil Seal L 34	6	13	78.00
6.	Magneto key	3	2	6.00
7.	Crank shaft L-34 Connecting Rod Set	3	740	2220.00
8.	Mounting plate MS	1	120	120.00
9.	Aluminimum block for head	1	75	75.00
10.	Chain	1	195	195.00
11	Sproket	4	110	440.00
12.	Valves	2	120	240.00
13.	Air tank	1	400	400.00
14.	Pressur	1	125	125.00
15.	Safety Valve	1	100	100.00
16.	Motor mount platform	1	300	300.00
17.	Switch and control	1set	325	325.00
18.	Motor 1 HP.	1	2800	2800.00
19.	Bolts & Nuts			139.50
				10,203.50

MAINTENANCE

Lubrication

The Crankshaft and Connecting Rod assembly is lubricated by lubricating oils as mentioned in the recommended lubricants charts oil should be filled in the oil tank which sprays oil into cylinder during the suction stroke. Care should be taken to see that the oil level is correctly maintained.

Daily

Check the oil level

Check the chain tension

Every hours of operation

Clean the suction filter, to ensure long life for the values and the piston assembly.

Every 200 hours of Operation

Check and adjust the chain tension.

Check all the bolts for tightness clean.

Check the developed pressure clean and adjust the safety valve if necessary.

Check the bearing sound.

Every 1000 Hours

The suction and delivery valves should be removed and the valve seats seating should be inspected for any scores or damange. The valves may be lapped if necessary in their respective seats using fine lapping compound.

Dismantle the crank shaft bearing and assemble it by new one or by applying grease.

Every 3000 Hours

The entire unit must be diamantled by an experienced hand who knows about the compressor in details and a general overhaul should be done. This will include inspection of all parts for wear and tear and replacement of damaged components, checking clearance between various components and assembling.

FURTHER IMPROVEMENTS

The triangular air compressor with common compression chamber can be improved in following areas.

1. Lubrication method to be improved to reduce wear and tear of the chain drive and moving parts.

2. Inter cooling can be used to reduce the workdone when multistage compression is used.

3. Pressure rise can be increased by reducing the clearence volume

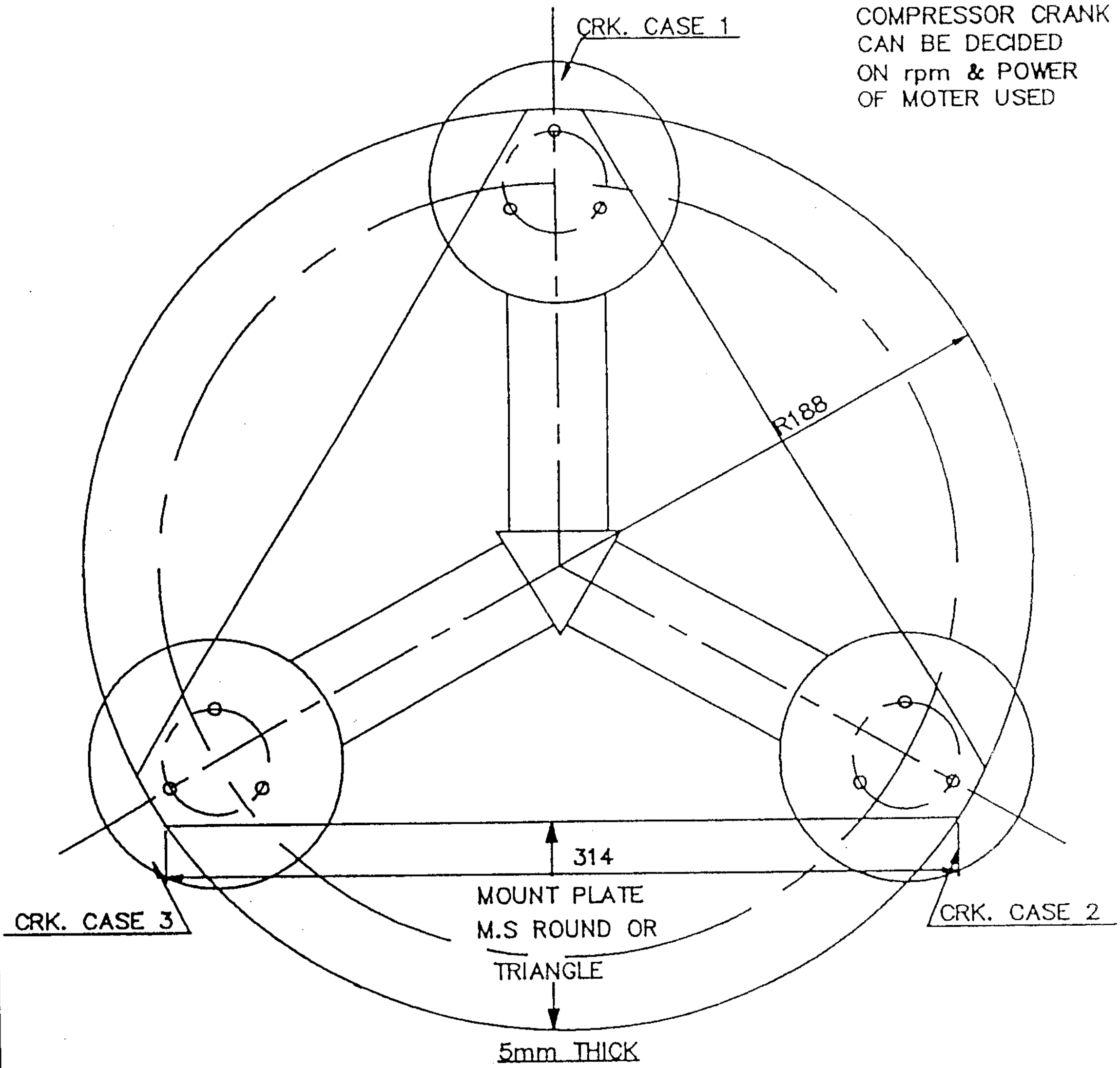
4. The production cost can be reduced by casting the cylinder and the head assembly in to a single piece.

CONCLUSION

The design and fabrication of triangular air compressor with the common compression chamber have been successfully completed and over all assembly of the compressor is drawn in this report.

The performance of the compressor was found to be satisfaction and the output of the compressor is continuous it is upto the level expected further improvements can be done on the radial compressor as dealed is previous chapter.

NOTE
SPEED OF EACH
COMPRESSOR CRANK
CAN BE DECIDED
ON rpm & POWER
OF MOTER USED

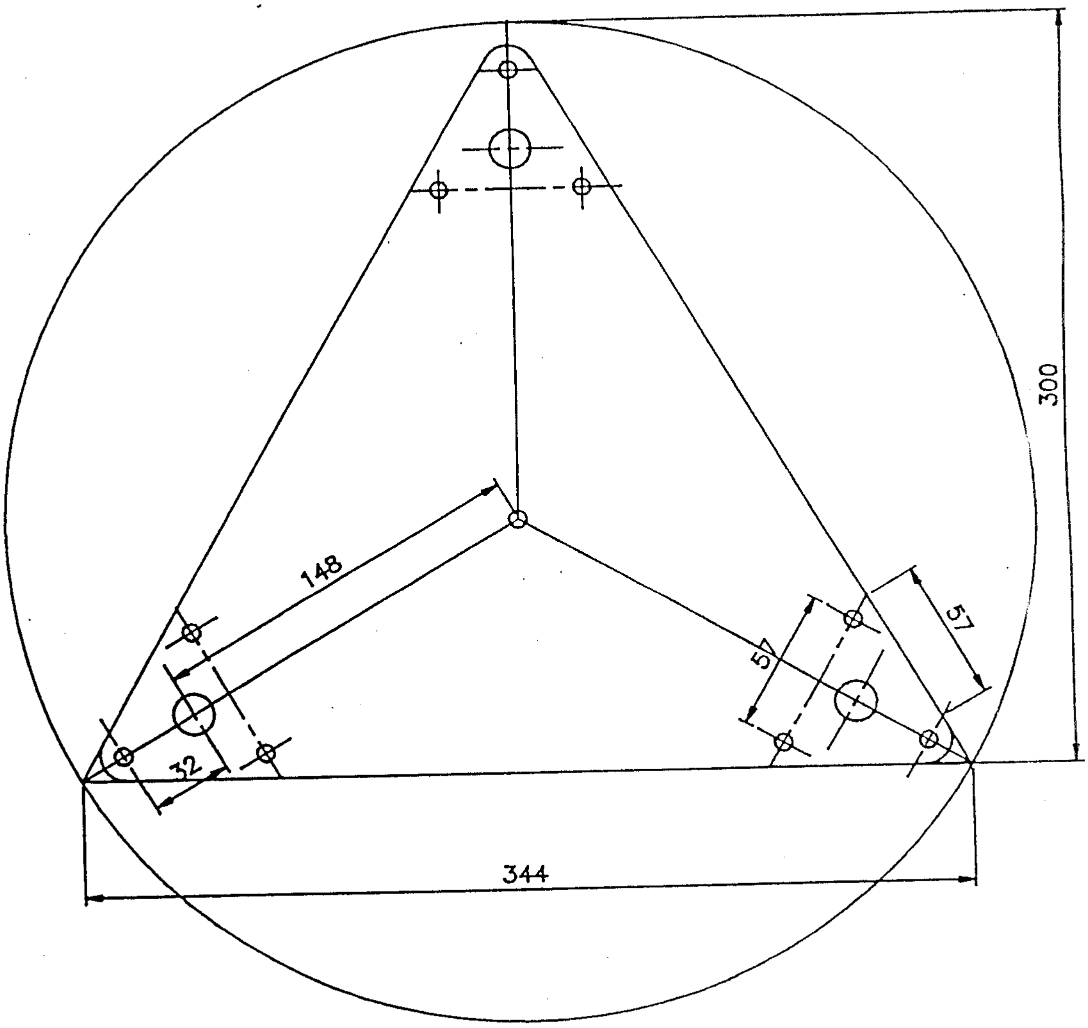


TRIANGULAR COMPRESSOR
LAY-OUT WITHOUT
AIR TANK & MOTOR

LAY-OUT OF
3 COMPRESSORS
ASSEMBLED

ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK

		NAME	DATE
KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE	DESIGNED		
	APPROVED		
SCALE NTS	TITLE GENERAL LAY OUT OF TRIANGULAR COMPRESSOR	DRG NO.	

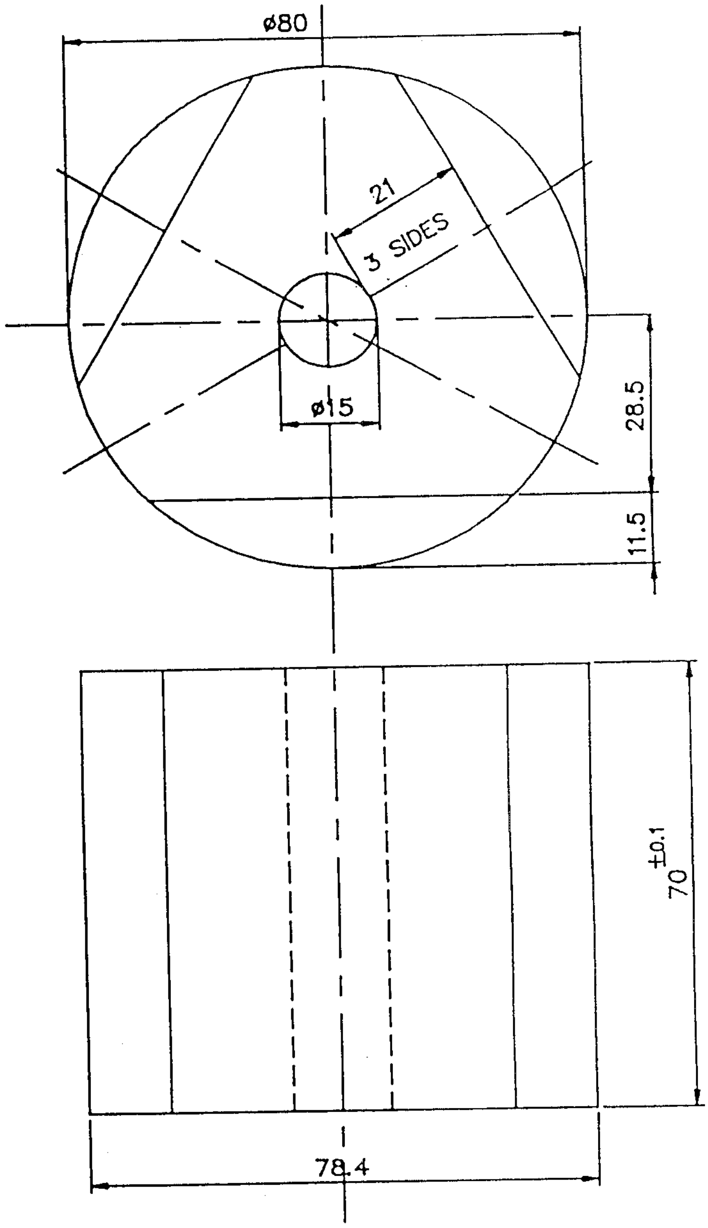


ROUND }
OR } MAKER'S CHOICE
ROUND }

M.S PLATE 5mm THICK
STRAIGHT WITH NO BENDS
350 EACH SIDE ROUGH

ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK

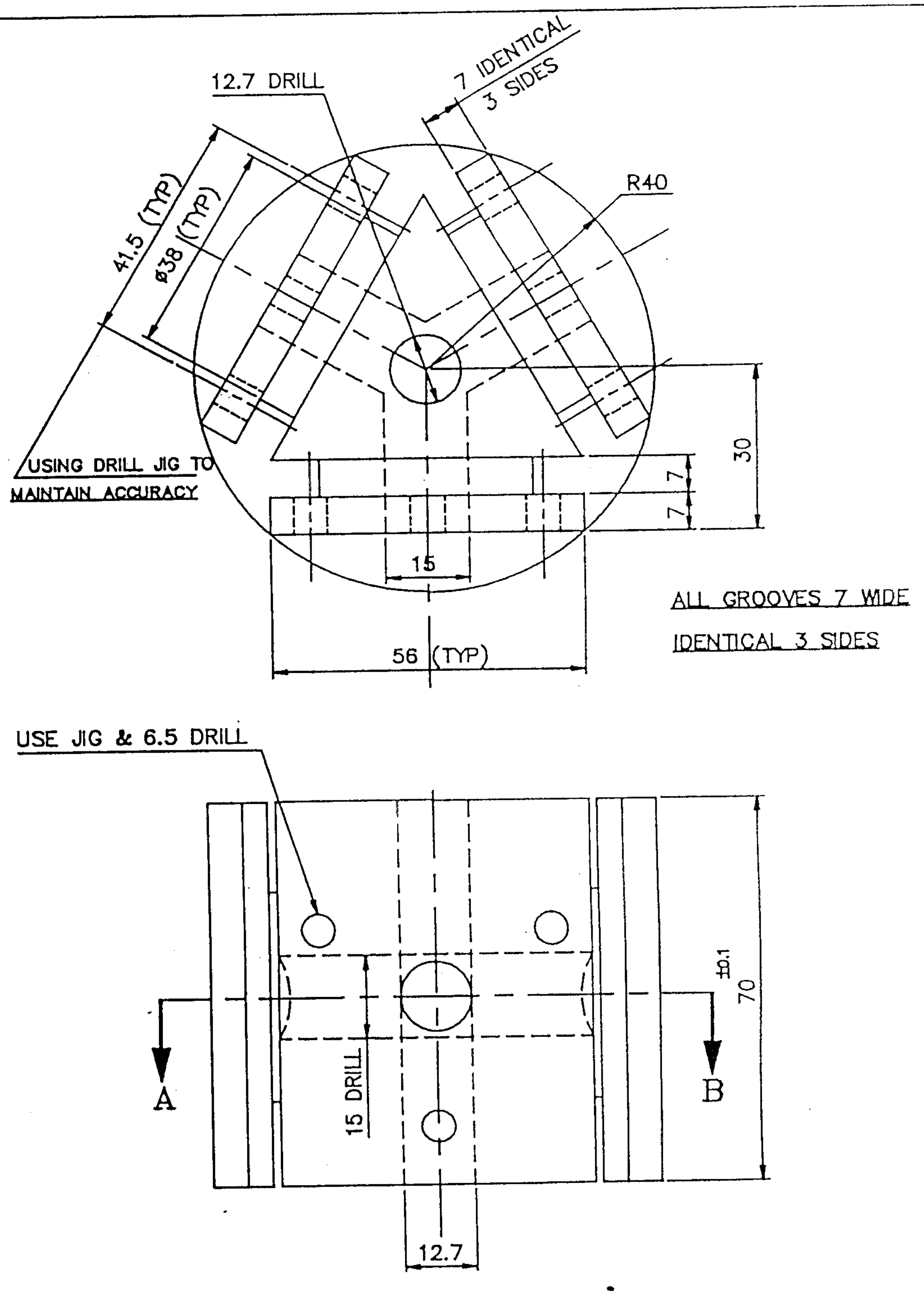
KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE		NAME	DATE
	DESIGNED		
	APPROVED		
SCALE NTS	TITLE TRIANGULAR COMPRESSOR MOUNT PLATE	DRG NO.	



MUST BE EQUILATERAL TRIANGLE
WITHIN ±5 MINUTES
OF 60° CORNERS

1st OPERATION
MACHINING

ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK		
KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE		NAME	DATE
	DESIGNED		
	APPROVED		
SCALE NTS	TITLE BLANK – CENTRAL PRESSURE HEAD – (COMMON)	DRG NO.	



ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK

KUMARAGURU COLLEGE OF TECHNOLOGY
COIMBATORE

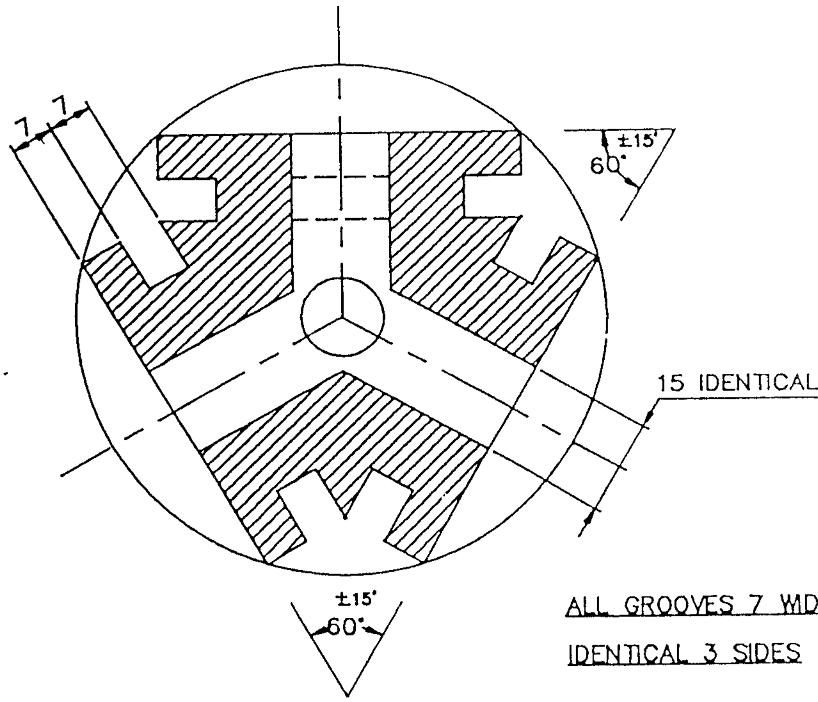
	NAME	DATE
DESIGNED		
APPROVED		

DRG NO.

SCALE NTS

TITLE

CENTRAL PRESSURE HEAD
MACHINED



ALL GROOVES 7 WIDE

IDENTICAL 3 SIDES

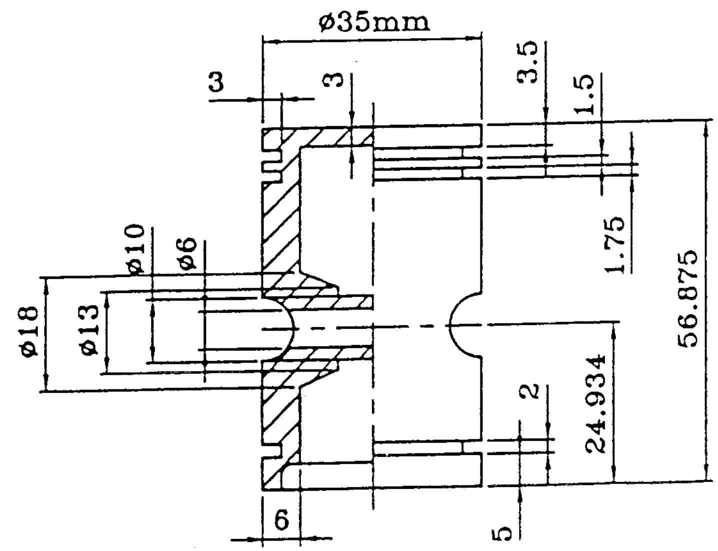
ALL OTHER DIMENSIONS AS

ON DRG : 160494-97-003

SECTION – AB

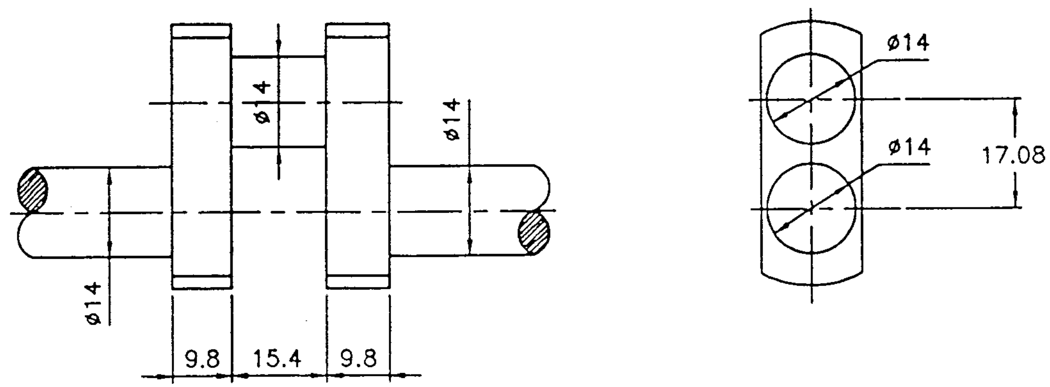
ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK

		NAME	DATE
KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE	DESIGNED		
	APPROVED		
SCALE NTS	TITLE CROSS SECTION OF CENTRAL PRESSURE HEAD	DRG NO.	



KUMARAGURU COLLEGE OF TECHNOLOGY
COIMBATORE

ALL DIMENSIONS ARE IN mm				
	SCALE : 1:1		NAME	DATE
		DESIGNED :		
		APPROVED :		
TITLE : PISTON		DRG No:		



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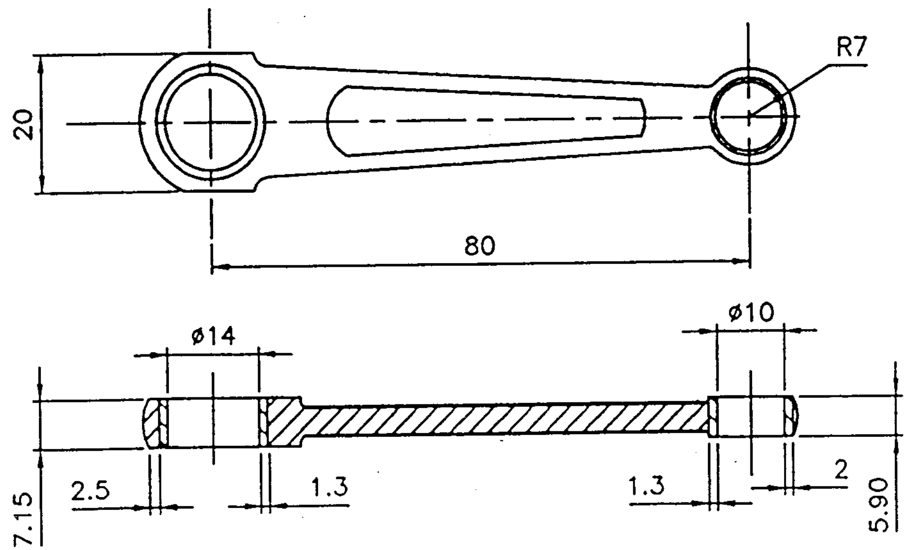
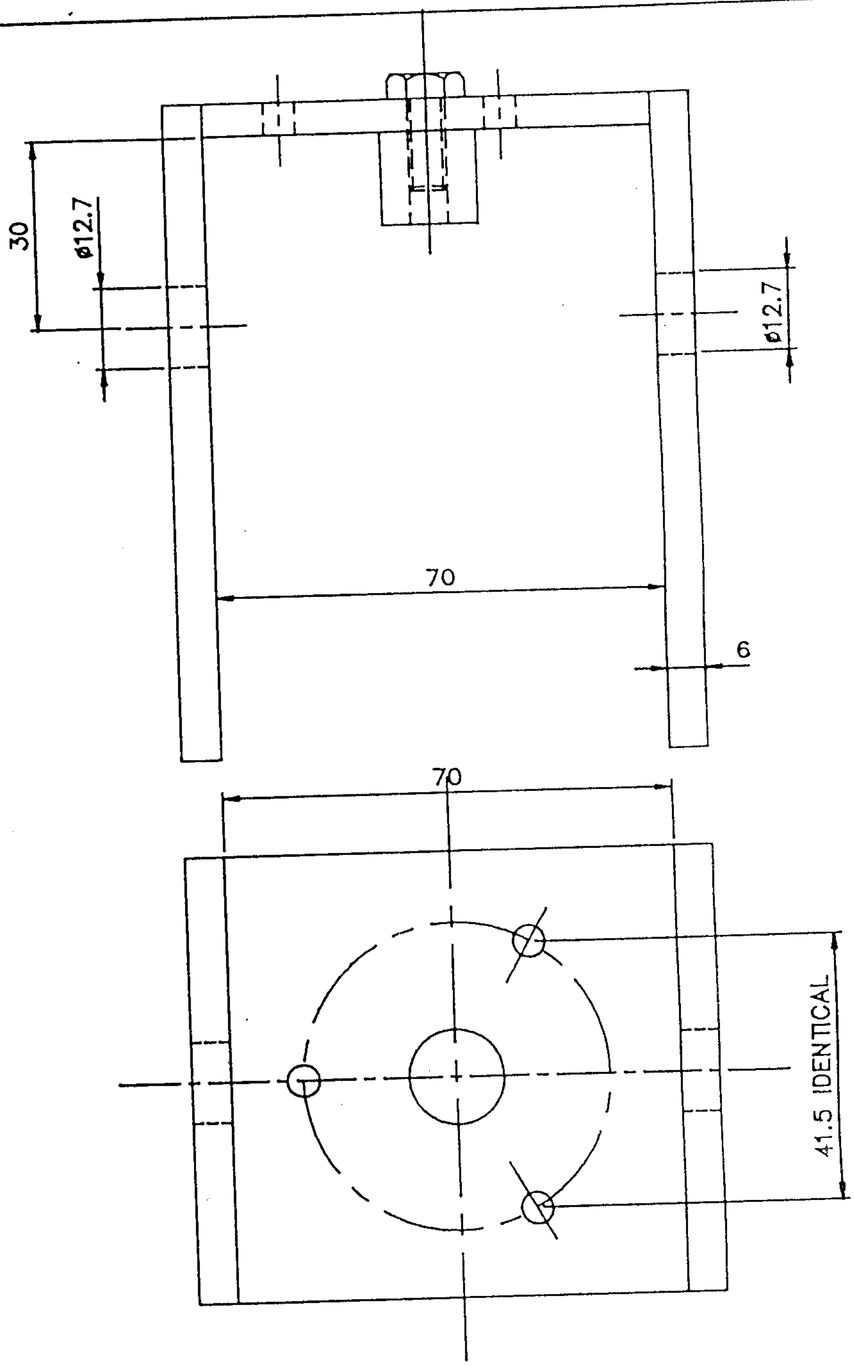
ALL DIMENSIONS ARE IN mm				
	SCALE : 1:1		NAME	DATE
		DESIGNED :		
		APPROVED :		
TITLE : CRANK SHAFT		DRG No:		



KUMARAGURU COLLEGE OF TECHNOLOGY
COIMBATORE

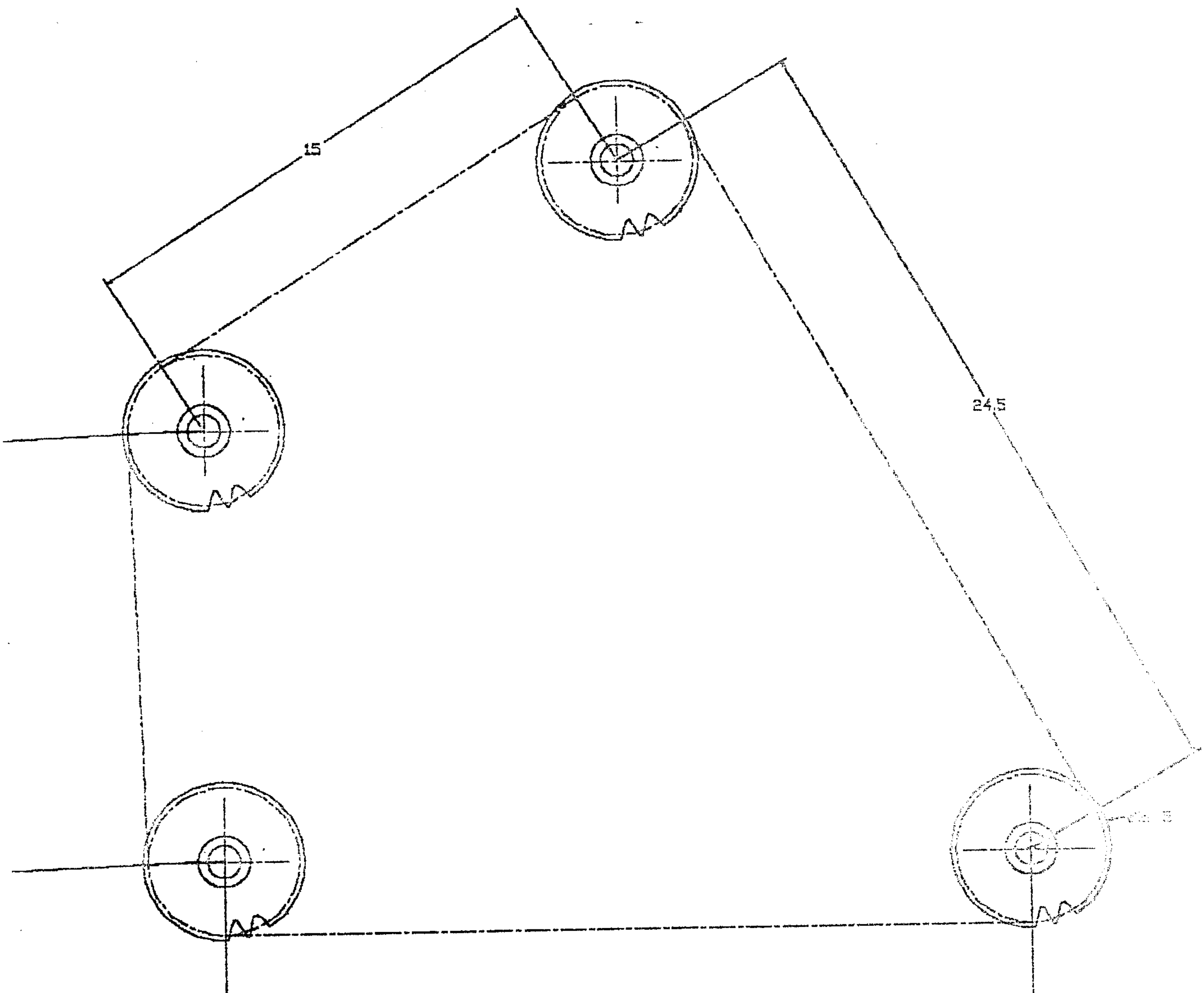
ALL DIMENSIONS ARE IN mm				
	SCALE : NTS		NAME	DATE
		DESIGNED :		
TITLE : CONNECTING ROD		APPROVED :		
		DRG No:		



BOTTOM VIEW

ALL DIMENSIONS ARE IN MM	DO NOT SCALE IF IN DOUBT ASK		NAME	DATE
KUMARAGURU COLLEGE OF TECHNOLOGY COIMBATORE		DESIGNED		
		APPROVED		
SCALE NTS	TITLE DRILL JIG FOR COMMON PRESSURE HEAD	DRG NO.		

CHAIN DRIVE MECHANISM



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3. Machine Design - E. wilson, Michels
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