DESIGN AND FABRICATION OF TRIANGULAR AIR COMPRESSOR WITH COMMON COMPRESSION CHAMBER

A Project Report

Submitted By

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"ME 1347-Design and Fabrication Project"

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SYNOPSIS

A need exists for an air compressor that consumes less energy and produces high pressure. This need is fulfilled by triangular air compressor with common compression chamber. In this project, a maximum pressure of seven bar is developed and the discharge is also optimized. The power required for developing this outlet pressure and discharge is just one horse power (1 h.p.).

The triangular air compressor with common compression chamber is a reciprocating type compressor. It consists of three cylinders and all the cylinders have their own connecting rod, crank shaft, pistons and chain sprockets. It is driven by a chain drive. This compression chamber will have an inlet and outlet valve. From this valve, air is sucked in and then delivered out. The common compression head will be in triangular shape with an angle of 60°between the sides.

The main advantage of this project is that it consumes less power and the vibration is comparatively less when compared to the existing ones. The optimized discharge is about 0.0864cub.m/min.Thus with this arrangement the objective of the project is fulfilled.

The specification of the compressor is

Maximum pressure	=	7 Bar
Displacement of compressor	=	100 cc.
Delivery rate	=	0.0864cub.m/min.

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1. INTRODUCTION

There is a need to improve the performance of the compressors. This is done by considering several aspects such as design, construction etc...The power required to compress the air to a known pressure of 7bar is given in the attached tables. One cubic root of air equals approximately 28 liters its volume.

It consists of three cylinders placed radially and equally apart such that the cylinder openings tends to meet on a common triangular compression chamber that has one inlet and outlet valve. There is a tank for receiving the air in the compressed state.

All the 3 pistons are made to compress the air simultaneously and deliver it to a common triangular chamber over shorter stroke, so that the isothermal efficiency will be better than the single cylinder of bore 50 mm. There will be no vibration due to the triangular forces acting towards the center at the same time.

Adiabatically, this will be advantageous since the displacement of three cylinders is at the same velocity. This is achieved by using three sprockets of same diameter and keeping them in contact using chain drive.

For every rotation of crank, each piston will move once from **Top Dead Centre (TDC)** to **Bottom Dead Center (BDC)** and from **BDC** to **TDC** (i.e. two strokes). Therefore, theoretically 98cc of air will be taken in and compressed to the volume of smaller space in the common compression chamber. If the space is smaller, the pressure will be more and vice versa.It finds its application in Air conditioning and Refrigeration systems, Vacuum pumps and General purpose usages.In positive displacement compressors, the volume of the air delivered should be constant for every cycle of operation.

2. GENERAL ASPECTS OF A RECIPROCATING COMPRESSOR

- SIGNIFICANCE OF COMPRESED GASES AND VAPOUR
- COMPRESSOR DEFINITION
- CLASSIFICATION
- RECIPROCATING COMPRESSOR TERMINOLOGY
- THERMODYNAMICS- PRINCIPLE
- WORKDONE BY COMPRESSION
- COOLING OF COMPRESSORS
- SUMMARY

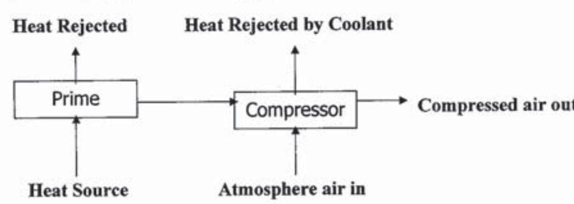
2.1 SIGNIFICANCE OF COMPRESSED GASES AND VAPOUR:

In industry and in economic life, the use of compressed gases and vapours are steadily increasing. Pressurized gases are used for many chemical and industrial purposes. The compressed air has numerous productivity in some fields such as metallurgy, chemical plants, hospitals, inflation of tyres, hot air guns, to operate air driven hand tools such as die polishers, die grinders etc…

Infact, it would be difficult to find a branch of industry where utilization of compressed air would not affect a material rationalization of manufacturing process. The advantages of pneumatic machines and tools are their safety, simplicity and ruggedness combined with comparatively low weight. The compressed vapours are used in refrigerating plants, which improves our standard of living by economic handling of perishable foods.

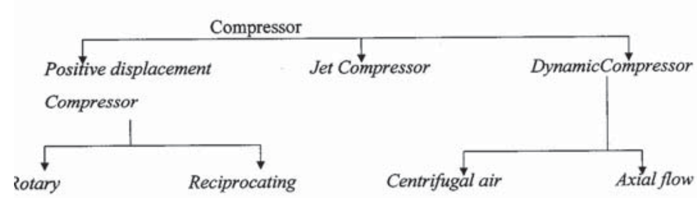
2.2 COMPRESSOR DEFINITION:

Compressor is a machine which provides air at high pressure and the work is done upon the air compressor by external agency. An air compressor takes atmospheric air in, compresses it and then delivers the high pressure air to a storage vessel. It is conveyed through pipelines to the supply centre from the vessels.



The above figure shows schematically the general arrangement of a compressor. Of the total energy is received by the compressor from prime movers, some will be lost by radiation and the rest will be maintained within the air and delivered at high pressure. The coolant is provided to reduce the heat dissipation through the compressor.

2.3 CLASSIFICATION:-



In dynamic compressors, high velocity nozzles are used to impart the kinetic energy and later this is converted to pressure energy by means of diffusers.

2.4 RECIPROCATING COMPRESSORS:-

The working of the compressor comprises of four phases namely,

- Expansion
- Suction
- Compression
- Discharge

This is accomplished by the reciprocating motion of the piston inside the cylinder. From this, we can achieve high pressure at relatively low Capacities. The basic elements of the reciprocating Compressors are piston, cylinder, valves, etc… The piston inside the cylinder is used to pressurize the gas. The connecting rod and crank mechanism are incorporated for the Conversion of rotary motion to linear motion.

When the piston moves down, the pressure inside the Cylinder falls below the atmospheric pressure. This makes the inlet valve open, and the air rushes in. When the piston comes to BDC the inlet valve closes. During the upward motion of piston, the air gets compressed. At certain pressure (The outlet valve opening Pressure) the outlet valve opens. This air is stored in the tank and used whenever necessary.

2.4.1 RECIPROCATING COMPRESSOR TERMINOLOGY:

1) SINGLE ACTING COMPRESSOR:-

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

2) DOUBLE ACTING COMPRESSOR:-

In this type, suction, compression and delivery of air takes place on both sides of the piston. Such compressors have two delivery strokes per revolution of the crank shaft.

3) SINGLE STAGE COMPRESSOR:-

In this compressor, the compression of air from the initial pressure to the final pressure is carried out in one cylinder.

4) MULTISTAGE COMPRESSOR:-

In this compressor, the compression of air from the initial Pressure to the final pressure is carried out in more than one cylinder. The air is passed in series through these cylinders.

5) COMPRESSION RATIO (OR) PRESSURE RATIO:-

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

6) FREE AIR DELIVERED (FAD):-

It is the volume of air Delivered under the conditions of temperature and pressure existing at the compressor intake. (i.e.) Volume of air delivered at surrounding air temperature and pressure.

7) 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 the radius of the cylinder bore, **L** is the stroke of piston.

8) ACTUAL CAPACITY OF THE COMPRESSOR:-

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

9) 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.

10) COMPRESSOR EFFICIENCY:-

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

11) THEORETICAL HORSE POWER:-

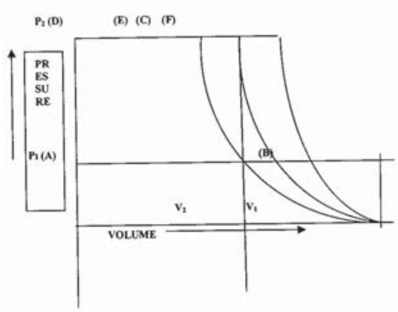
This is the horse power required by the compressor for compressing the air and delivering it through a specified pressure range, without provision for loss of energy.

12) THERMODYNAMICS:-

Isothermal, adiabatic and polytrophic processes are the different types of processes considered. In an isothermal process PV=Constant, as T remains 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.

When a gas is compressed or expanded, it has been established that the pressure will vary to an exponential power of the volume (ie PV = Constant).This type of relationship for the change of state where no heat is lost or gained is known as adiabatic process. A perfect adiabatic process, which is reversible, is called as isotropic process. Industrial compressors can reject heat and have valve and ring leakages. They are also subjected to generation of frictional heat.

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



The ideal indicator diagram in fig ABCD illustrates the working of the compressor. Assuming, water jacket cooling bath and with no internal losses by friction or eddies, the graph is drawn. During the suction stroke AB, Volume (V_1) of air flows into the cylinder at atmospheric 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 is given by,

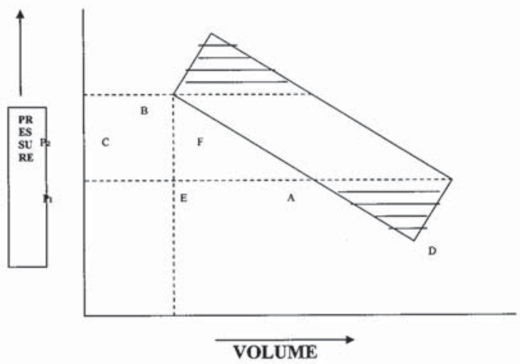
$$W = (P_2\, V_2 - P_1 V_1) / n\text{-}1$$

When there is water around the cylinder, 'n' may vary from 1.35 to 1.25. For efficient cooling and special cases of cooling and for the air which is cooled by spraying water into the cylinder during compression the value may be 1.2 or even lower. ED is the delivery of air from the cylinder to the receiver at constant pressure P_2 and work done during this stroke is P_2V_2.

2.4.2 WORK ON COMPRESSION

2.4.2.1. Adiabatic compression:-

In the reciprocating gas compressor, work is performed by the piston. The gas is sucked from the suction line till it is conveyed to the discharge line for complete cycles. Piston is restricted by the pressure difference between the discharge line and the suction line. The work done by the piston on the gas is,



$$dw = P_a\, dc$$

Where 'P_a' is the max pressure, 'PL' is the displacement, The expression of work can be related as

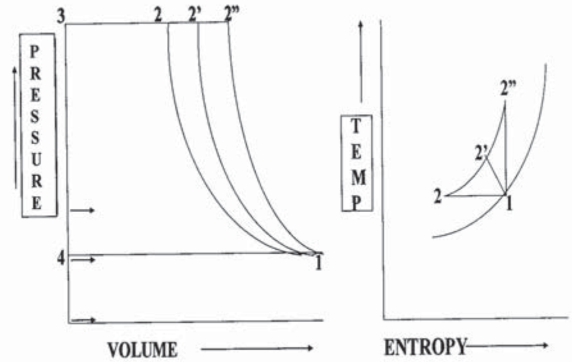
$dw = Vdp$, Where V is the volume

$\omega_1 - z = Vdp$

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 figure. The ordinate P_1 & P_2 represents the suction and discharge line pressures respectively.

PV & TS Diagram



1 – 2 Isothermal Compression

1 – 2' Polytropic Compression

1 – 2" Isentropic or Reversible Adiabatic Compression

The ideal path of the AB and CD is shown as follows. The adiabatic process,

$$P_1V_1^n = P_2V_2^n = \text{Constant}$$

By applying adiabatic relationship and integrating above equation the work on

$$\text{Work on compression} = \frac{n}{n-1} \times \left[\frac{P_2}{P_1}\right]^{\frac{n-1}{n}} - 1$$

Where P_1 is the initial pressure (atm.pressure)

V_1 is the volume occupied by air.

(Stroke volume + Clearance Volume)

Where P_2 is the final pressure after compression. To calculate the Power required for driving the compressor we have the expansion stroke.

Power = P_m is the mean effective pressure.

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 = Work done / Volume

$$= \frac{n}{n-1} P_1 \{R_C^{\frac{n-1}{n}} - 1\} E_V$$

Where,

R_C is the compression ratio (P_2/P_1)

E_v is volumetric efficiency

Volumetric efficiency = free air delivered / stroke volume.

2.4.2.2. Compressor cooling:-

When a gas is compressed adiabatically from pressure P_1 to P_2, the temperature of gas raises from T_1 to T_2. This increase in temperature occurs according to the following relation.

$$\frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{\frac{n-1}{n}}$$

Therefore when the stage pressure ratio is high, the effect of the temperature raise is prominent. Therefore cooling of the gas is essential. The general types of cooling arrangements available are

- **Water jacket cooling**
- **Air cooling**
- **Inter Cooling and after cooling**

Here air cooling is done by providing proper fan arrangement.

2.4.2.3. 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.

3 DESCRIPTIONS OF COMPONENTS

The main components of the triangular air compressor with common compression chamber are given below.

a) Common Compression head
b) Cylinder
c) Connecting rod
d) Crank shaft
e) End bearings
f) Valve

g) Power transmission
h) Air filter
i) Gasket
j) Service valve
k) Lubrication system
l) Cooling fan

3.1 DESIGN CONSIDERATIONS

The design considerations of various parts of triangular air compressor with common compression chamber are given below.

a) Common compression head:-

The cylinder head is made of high grade alloy of aluminium. The cylinder head here serves as the common compression chamber, which holds the entire three cylinders together. The head is of triangular shape, and hence the machining will be some what difficult. The pressure of compression may amount to as much as 5kgf/unit. 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 is made up of cast iron. The cast iron must be done enough to prevent the seepage of air through it. The compressor was casted 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 uses some means to change the rotary motion of the motor into reciprocating motion of the compressor. The crank shaft in this design is in forged steel. This compressor uses an eccentric, fastened to a straight shaft and it removes the need for connecting rod caps and bolts.

The crank shaft main bearing supports the crank. They must carry the end load of the crank shaft. The connecting rod bearings are fitted with greater accuracy.

e) End bearings:-

Anti-frictional bearings are the most suitable for end bearings. They are available in standard sizes, are maintenance less, durable, early replaceable and they have good load carrying capacity. They can withstand variable loads. With the above advantages, they are most suited for compressor end bearings. Selection of the appropriate bearings for the compressor depends on their life in working hours and their format of loading.

f) Valve:-

The valve assembly consists of a valve plate an intake valve, a delivery valve and the valve retainer. The valve plate is made of hardened steel that can be thinner with lower wearing valve seats. Compressor valves are made of high carbon alloy steels.

They are heat treated which gives them the properties of spring steel and are ground to a perfectly flat surface. The intake valve is kept in place by small pins or by the clamping action between the compressor head and valve plate. Exhaust valve is also clamped in the same way. The valve disks must be perfectly flat and a defect of 0.00254mm will cause valves to leak. Of the two valves, the intake gives the least trouble. This is because it operates at a relatively cold temperature. The delivery value must be fitted with special care as it operates at high temperature and must be leak proof against a relatively high pressure difference.

g) Power transmission:-

The compressor being of maximum discharge and less pressure, the horse power required is also 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 at 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 as cylinder heads, valve plates, crank case opening etc are sealed with gaskets. The gaskets are made of special paper and are free from moisture.

j) Service Valve:-

In order to attach gauges and service manifolds to the system, airflow lines must be tapped. Special tapping valves are also available. These are changed to a tube, so that the valve piece forms the tube and at the same time provides necessary gauge and service connections.

k) Lubrication parts:-

The Lubrication part consists 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.

L) Cooling Fan:-

The cooling fan is provided with the crankshaft 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 cylinder and piston, which may lead to increase in the temperature of outlet air. By providing a cooling fan the temperature up to 3-5°C can be reduced.

4. WORKING PRINCIPLE

This triangular air compressor with common compression chamber consists of three cylinders placed at 120° to each other. The main aim of this compressor is to generate same amount of air with same capacity as the other compressors, but with less power input and with low vibration.

The drive is of chain type drive, because to overcome the slipping encountered while using the belt drive.When the motor drives the crank shaft, **the three pistons will move in phase from Bottom Dead Centre to Top Dead Centre.** When the piston reaches TDC, compression takes place and discharge is done. When the piston reaches the BDC, suction conditions are obtained and suction takes place.

The compressed air from the cylinders is allowed to enter into the common compression chamber and is taken out through the central hole provided in the chamber. From here it is conveyed to the requirement areas through the pipe lines.

(Annexure 1. Page No. 75)

5. COMPARISION OF CONVENTIONAL COMPRESSORS WITH TRIANGULAR AIR COMPRESSOR

We know that there are several types of compressors made in many renowned factories for various applications. But three pistons compressing air into a common compression chamber is not known to be an existing one and so we decided to make this design to prove its superiority over others in terms cost and performance.

Any new product must be useful, superior in performance and reliable when compared to other known models. This compressor can be cited as an example for this.

The cost of making must be comparatively low and in the market it should be capable of competiting with other similar products and we hope this compressor has that capability. Range will be based on the perspective of customers. However, first working model is only to prove its 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 more reliable than the conventional ones because of having less vibration.

6. DESIGN DETAILS

DESIGN OF MAJOR COMPONENTS

6.1.DESIGN OF PISTON:

We know the diameter of the piston which is equal to **35 mm**.

The thickness of the piston head is calculated from flat plate theory

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

Where,

P - The maximum compression pressure which is equal to 7 bars.

F - The Permissible stress in tension.

Here piston material is aluminum alloy.

So permissible tensile stress = 34.6 N/mm²

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

$$t = 2.15 \text{ mm}$$

Say, $t = 3$ mm

Number of piston Rings = $2\sqrt{D}$

"D" should be in inches, D = 35 mm=1.49 inches

∴Number of piston rings = $2 \times \sqrt{1.49} = 2$.

∴ we adopt 2 compression ring and one 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 =35/20 =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 piston lands between rings= 0.75 x 1.75

= 1.312 mm

≈ 1.5 mm

Length of piston = 1.625 x D

= 1.625 x 35

= 56.875 mm

Length of the Piston skirt = Total length – Distance of first ring From top of the piston – (Number of Landing between rings x width of land) – (No. of Compression ring x Width of the ring).

= 56.875 – 3.5 – (2 x 1.5) – (2 x 1.75)

= 46.875 mm

Centre of piston pin above the centre of the skirt = 0.02 D

= 0.02 x 35 = 0.7

≈ 1 mm.

Therefore the distance from the bottom of the piston to the centre of the piston pin,

= ½ x 46.875 – 1.5 = 24.937 mm.

The thickness of the piston walls at open ends

= ½ x 3 = 1.5mm

The Bearing area provided by piston skirt

= 46.875 x 35π

= 1640.625π mm².

6.1.1 SPECIFICATION OF PISTON:

Diameter of Piston	=	35 mm
Thickness of the Piston head	=	3 mm
Number of piston rings	=	2 Nos.
Number of compression ring	=	1 No.
Number of oil ring	=	1 No.
Thickness of wall under piston ring	=	3 mm
Thickness of ring	=	1.09 mm
Width of the ring	=	1.75 mm
Distance of the first ring from top of the piston	=	3.5 mm
Width of piston of land between rings	=	1.5 mm
Length of piston	=	56.875 mm
Length of Piston skirt	=	46.875 mm
The centre of the piston pin above the Centre of the skirt	=	1 mm
Distance from the bottom of the piston to the centre of the piston pin	=	24.934 mm
The thickness of the piston walls at open end	=	1.5 mm
The bearing area provided by piston skirt	=	1640.625π mm²

6.2 DESIGN OF CYLINDER

Considerations:

Required stroke length	=	34 mm
Maximum Pressure (P)	=	7 kg/cm²
Material used (cast iron) F_t	=	1730 kg/cm²
Diameter of the piston, d_i	=	35 mm
Force acting on cylinder during compression (F)	=	$\Pi/4 \times di^2 \times P$
	=	$\Pi/4 \times 3.5^2 \times 7$
	=	68 kgf

Thickness of the Cylinder wall (t) $= r_i \{[\sqrt{(f_t + P)/(f_t - P)}] - 1\}$

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

= 1.57 mm

Thickness factor of safety as 3 mm

∴ Thickness t = 1.5 + 3 = 4.5 mm

∴ Outside diameter of Cylinder, d_o = 35 + (2 x 4.5)

d_o = 44 mm

Length of the cylinder = Stroke length + Length of the piston

= 34 + 56.875 mm

= 90 mm

6.4.1 Specification of cylinder:

Inner diameter	=	35 mm
External Diameter	=	44 mm
Thickness of the wall	=	4.5 mm
Length of the cylinder	=	90 mm

6.3 FIN DESIGN

As the temperature produced during the compression process is nearly around 200°C, it is necessary to design the fin to conduct the heat produced during the process. Since the cylinder is made up of cast iron, the fin also made up of cast iron.

∴ Thermal conductivity of cast iron (K) = 67.5×10^{-3} W/m°K

Heat transfer Co-efficient (h) = 25.4 W/m² °K

Perimeter of the fin (P) = Π (d + Thickness)

= Π (44 + 4.5) =0.152 m

Area of the fin (A) = Π x (d²/4) = Π x 44²/4= 1936 mm²

Heat transferred (Q) = hKA x (T_0-T_1)

= $25.4 \times 0.152 \times 67.5 \times 10^{-3}$

x 1.936 x 171

= 86.27 KW

6.3.1 Fin efficiency:

(Lc) = L + (t/2) ------ (From HMT data book)

= 100 + (4.5/2)

= 0.102 m

As we are having rectangular fin

As = 2 Lc -------------- (From HMT data book)

= 2 x 102.25

= 204.5 mm² = 0.2045 m³

Am = t Lc ------------------ (From HMT data book)

= 4.5 x 102.25

= 460.125 mm²

= 0.460 m²------ (From HMT data book)

$Lc^{1.5} (h/KAm)^{0.5}$ = $(0.102)^{1.5}\ (25.4/\{67.5 \times 10^{-3} \times 0.460\})^{0.5}$

= 0.032 x 28.60

$Lc^{1.5} (h/KAm)^{0.5}$ = 0.915

By checking the value from the graph the fin efficiency is found to be η = 79%

Actual heat transferred = $\eta\, Q_{min}$

= 0.79 x 86.27 KW

Qact = 68.15 KW

To find the No. of fins

Qact = η As h (T_0-T_1) n

n = Q_{act}/ {η As h (T_0-T_1)}

= $(68.15 \times 10^3) / (0.79 \times 204.5 \times 25.4 \times 171)$

= 6

6.3.2 Specification of the fin:

Total number of fins	=	6
Type of fin	=	Rectangular type
Perimeter of the fin (P)	=	0.152 m
Area of the fin (A)	=	1936 mm²
Efficiency of the fin (η)	=	79%
Actual heat transferred	=	68.15 KW

6.4 DESIGN OF VALVE

To find the inlet valve diameter:-

During the suction stroke, the suction effect take place and the pressure drop created is calculated as

Head h_m = 0.24×10^{-3} m of Hg

Therefore, h_a = $h_m (P_m/P_a - 1)$

= 0.24×10^{-3} { (13.1 x 1000)/ (1.2-1)}

= 2.83 m of air

Velocity (V) = $C_v \sqrt{2 g h_a}$

= $0.96 \sqrt{2 \times 9.81 \times 2.83}$

= 7.15 m/sec

Discharge (Q) = A V

= π / 4 d² LN

= $\pi / 4\ (57 \times 10^{-2})^2 \times 56 \times (1440/60)$

= 4.38×10^{-3} m³/sec

∴ Area of inlet valve (A) = Q/V = $(4.38 \times 10^{-3}) / 7.153$

Π/4 d² = 6.164×10^{-4}

∴d = 12.7 mm

Therefore diameter of the inlet valve = 12.7 mm

To find delivery valve diameter:

Head (h) = $(P_1 - P_2)/ Pg$

Where, P_1 = Atmospheric pressure

P_2 = Suction Pressure

$\therefore h$ = $(1.01325 \times 10^{-5} - 0.8 \times 10^{3}) / (1.2 \times 9.81 \times 9.81)$

= 870.4 m of air

Velocity (V) = $Cv \sqrt{2 g h}$

= $0.96 \sqrt{2 \times 9.81 \times 870.4}$ = 125.4 m/sec

Discharge (Q) = A V

Where, A = Area of delivery valve= Q/V

= $(4.38 \times 10^{-3}) / 125.4$

$\Pi/4\ d^2$ = 3.49×10^{-5}

d = 14.9×10^{-3} mm= 14.9 mm

Therefore diameter of delivery valve = 14.9 mm

6.4.1 Specification:

Inlet Valve:

Velocity at which air enters = 7.15 m/sec

Discharge (Q) = $4.38 \times 10^{-3}\ m^3/sec$

Diameter of Inlet valve = 12.7 mm

Outlet Valve

Velocity at which air delivers = 125.4 m/sec

Diameter of outlet Valve = 14.9 mm

6.5 DESIGN OF HEAD

As we are doing Triangular air compressor with common compression chamber, the head design plays an important role. The Cylinders are at an angle of 120° to each other. As the width of the cylinder Block is about 56mm. Therefore the sides of the head are designed to 56mm.The drill at the centre where the valve should be fitted will be at the size of 12.7mm. Because the diameter of the inlet valve is 12.7mm and the outlet valve is about 14.9mm.

Then the air from the cylinder should reach the common compression chamber. Hence the drill is done at the diameter of 15mm at exactly centre of the piston to the common compression chamber.

Then to fix the cylinder with the Compression chamber the groove of 7mm thickness is taken at the three sides.

6.5.1 Specifications of the Head:

Length of the Head = 70mm

Breadth of the Head = 56mm

Diameter of Central drill = 12.7mm

Diameter of the drill which connects the piston with central drill = 15mm

Distance from centre to one side = 30mm

6.6 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

Speed (max) = 1440 rpm

Compression ratio = 7:1

Maximum pressure = 7 bar = $0.7\ N/mm^2$

We know that radius of the crank (r)= Stroke length/2

= 34 / 2 = 17 mm

r = 0.017 m

Ratio of length of connecting rod to the radius of the crank (n) = l / r

n = 80/17 = 4.7

We know that maximum force on the piston due to pressure (F_L) = $\Pi/4\ D^2 P$

F_L = $\Pi/4 \times 35^2 \times 0.7$

= 0.673×10^3 N

Maximum angular speed = ω_{max}

= $(2 \pi N_{max})/ 60$

ω_{max} = 150.8 rad/sec

We know that maximum inertial force of reciprocating parts,

F_L = $m\omega^2_{max} X \{Cos\theta + (Cos2\theta/n)\}$

The inertia force of reciprocating parts is maximum, when the crank is at inner dead centre,

(i.e.) When θ = 0^0

F_L = $m\ \omega^2 r\ \{ 1+ (1/n)\}$

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

= 0.14×10^3 N

Force in the connecting rod Fc equal to the maximum force on the piston due to gas pressure F_L.

Therefore, force in the connecting rod

F_c = F_L = 0.673×10^3 N

The section of the connecting rod is **'I'** section. Connecting rod is designed for buckling about perpendicular axis. Assuming both the ends is hinged taking a factor of safety as 6.

The buckling load

W_{cr} = $F_c \times 6$

= $0.673 \times 10^3 \times 6$

= 4.04×10^3 N

Area of cross section

A = $2\ (4t \times t) + (t \times 3t)$

= $11t^2$

Moment of inertia about perpendicular axis

I_{xx} = $\{ [\ 4t(5t)^3 / 12\ ] - [\ 3t(3t)^3 / 12\ ] \}$

I_{xx}	=	$419\ t^4 / 12$
Radius of gyration K_{xx}	=	$\sqrt{I_{xx} / A}$
	=	$\sqrt{(419\ t^4 / 12) \times (1 / 11t^2)}$
	=	1.78 t

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

L	=	80 mm
Taking the values for F_c	=	320 N/mm²
a	=	1/7500

For connecting rod material, By Rankines formula,

W_{cr}	=	$F_c A / \{1 + a [L/K_{xx}]^2\}$
4.04×10^3	=	$[320 \times 11t^2] / \{[1+ (1/7500)] (80/1.78t\}^2$
1.14	=	$t^2 / [1+ (0.269/t^2)]$
1.14	=	$t^4 / [t^2 + 0.269]$
$1.14t^2 + 0.3066$	=	t^4
t^2	=	$1.14 \pm \sqrt{[1.4^2 - (4 \times 1 \times (-0.3066))]} / 2$
	=	$1.6 \pm (1.78/2) = 1.69$
t	=	1.3 mm

The dimensions of cross section of the connecting rod at mid section are

Height	=	5t	=	5 x 1.3	=	6.5mm
Width	=	4t	=	4 x 1.3	=	5.2 mm
Thickness of the flange and web			=	1.3 mm		

Depth of the small end	=	6.5/1.1 =5.90 mm

6.6.1 Specification of the connecting rod:

Length of connecting rod from centre to centre	=	80 mm
Height of the Connecting rod	=	6.5 mm
Width of the Connecting rod	=	5.2 mm
Thickness of the flange and web	=	1.3 mm
Depth at the big end	=	7.15 mm
Depth at the small end	=	5.90 mm

6.7 SELECTION OF BEARING

Deep groove ball bearing is selected for end bearing which support crank shaft.

Bearing Life (L)	=	3 Years = 3 x 300 x 5
	=	4500 x 1440 x 60 revolutions
	=	388.8×10^6 revolutions
Total load of the bearings	=	0.673×10^3 N
Central load acting on each bearing	=	0.336×10^3 N

Referring to Design data Book,

Equivalent load (P)	=	$(X_f + Y_f)$
f_r- radial load	=	0.336×10^3 N
f_a-axial load	=	0 N (From Design Data Book,)

X=1 and Y=0

Service factors (S)	=	1.3 for reciprocating M/C's
P	=	$(1 \times 0.336 \times 10^3 + 0) \times 1.3$
	=	0.436×10^3 N

From the relation,

Dynamic Capacity (C)	=	$(L/L_{10})^1$ Ck
K	=	3 for ball bearings
L_{10}	=	10^6 revolutions
C	=	3.119×10^3 N

Therefore referring to the Data Book we select SKF 6202 (ISI 25B CO3) for big end of the connecting rod.

6.7.1 Specifications of the bearings:

Total load on bearing	=	0.673×10^3 N
Equivalent load	=	0.436×10^3 N
Dynamic Capacity	=	3.119×10^3 N
Bearing Life	=	3 Years
Number of revolutions	=	388.8 x 10□ Revolutions
Types of Bearing	=	SKF 6202
Code	=	ISI 25 B CO3

6.8. DESIGN OF CRANK SHAFT

Let 'P' be the force acting on the Crankpin

l	=	Length of crank pin
d	=	Diameter of the crank pin
f	=	Permissible tensile stress intensity in the pin material
P	=	Safe bearing pressure on the projected area.

Assuming the Crank pin to be a simply supported beam, the maximum bending Moment is given by,

M	=	Pl / 8 ---------------- (1)

The equation derived from bearing consideration will be

P	=	pdl ------------------ (2)
From (1) and (2) we get, M	=	P² / 8 Pd

By equating the resisting moment of the crank pin to bending moment, we get

$\Pi/32\ d^3$ F	=	$p^2 / 8$ Pd
d	=	$4 \sqrt{(32\ p^2 / 8\pi\ f)}$
d	=	$\sqrt{(4\ p^2 / \pi pf)}$
Force acting on the crank pin, P	=	$\pi\ d^2 / 4 \times P$
	=	$(\pi \times 3.5^2/4) \times 7$
	=	67.34 Kg

Safe bearing pressure for the material is,

p	=	280 Kg/cm²

Permissible tensile stress for the material is,

f	=	1730 Kg/cm²
From data book, d	=	$4 \sqrt{\{4 \times (67.34)^2 / (\pi \times 280 \times 1730)\}}$
d	=	0.33 cm

d = 3.3 mm

On safer side the diameter of the crank shaft 'd' may be taken as 14 mm

Thickness of the web (t) = 0.7 d

= 0.7 x 4

t = 2.8 mm

Say (t) = 3 mm

Distance between web = 1.1 d

= 1.1 x 4=4.4

≈ 5 mm

Width of the web (w) = 1.14 d

= 1.14 x 4

= 4.56

w = 4.6 mm

6.8.1 Specifications of the Crank Shaft:

Diameter of the Crank shaft (d) = 14 mm

Thickness of the Web (t) = 3 mm

Distance between web = 5 mm

Width of the Web (W) = 4.6 mm.

6.9. DESIGN OF PISTON PIN

Total load acting on the pin = $\pi/4 \times (0.035)^2 \times 7 \times 10^2$

P = 0.673×10^3 Kgf

Equating this load to shearing resistance

P = $2 \times \pi/4 \times (d_1^2 - d_2^2) \times fs$

Here, 2-Double Shear

d_1-External Diameter

d_2-Internal diameter

The ratio between the diameters is taken as

$d_1 = 2\ d_2$

$0.673 \times 10^3 = 2 \times \Pi/4\ \{(2d_1)^2 - d_2^2\} \times fs$

Available shear stress for the material is 840 kg/cm²

$0.673 \times 10^3 = 2 \times \pi / 4\ \{(4d_2)^2 - d_2^2\} \times 840$

$4\ d_2^2 - d_2^2 = (0.673 \times 10^3) / (\pi/2 \times 840)$

d_2 = 0.42 cm = 5 mm

d_1 = 2 x 5 = 10 mm

Length of the pin between supports = $1.75\ d_1$

= 1.75 x 10 = 17.5 mm

Check for bending stress. If 'f' be the bending stress procedure

$\Pi/32 \times (d_1^3 - d_2^3) \times f$ = T l / 8

Where, T = P d l

T = 10 x (1-0.5) x 1.75

T = 8.75 Kg

$\Pi/32\ (1^3 - 0.5^3) \times f$ = (8.75 x 1.75) / 8

1.914 = 0.085 x f

f = 22.28 Kg/cm²

But allowable bending stress is 630 kg/cm². So the design is safe for this material.

6.9.1 Specifications of piston pin:

External Diameter of the piston pin (d_e) = 10 mm

Internal Diameter of the piston pin (d_i) = 5 mm

Length of the piston pin between supports = 17.5 mm

6.10. DESIGN OF CHAIN DRIVE

Power required to run the Compressor = 1 HP = 736 W

Speed at which compressor run = 1440 rpm

Average running hrs/day = 5 hrs/day

Centre distance between the pulley = 344 mm

Transmission ratio (i) = 1440/1400 = 1

6.10.1 Selection of teeth on the driver Sprocket:

Consulting table 11.4 recommended value

(From the book design of transmission elements)

Z_2 = i Z_1

= 1 x 30

= 30

Optimum centre distance, a = (30-50) P

= 344 mm

P_{max} = a/30 = 344/30 = 11.46 mm

P_{min} = a/50 = 344/50 = 6.88 mm

Assume a standard pitch closer to P_{max} (Larger Pitch is chosen so as to arrive at a quicker solution, but this may not be the best solution, any standard pitch between 6.88 to 11.46 can be chosen and the chain should be checked for strength and bearing pressure).

P = 12.7 mm (Standard is chosen)

Assume the chain to be duplex

The details of the chain to be brought is

08B -2 DR 1278 (Table 11.6 Design Data Book)

Total load on driving side of the chain

$P_T = P_t + P_c + P_s$

Tangential force (P_t) = 1020

$$V = \frac{\text{No. of teeth on the sprocket x pitch x rpm}}{60 \times 1000}$$

= 9.14 m/s

P_t = (1020 x 0.764) / 9.14

= 85.26N

Centrifugal tension,

$P_c = WV^2/g = mv^2$

Here, m = 1.32 (From table 11.6)

$P_c = 1.32 \times 9.14^2$

= 110.27 N

Tension due to sagging

P_s = K. W. a

Where,

K = 6 for horizontal drive

W = mg = 13.2 N/m

a = 0.344 m

P_s = 6 x 13.2 x 0.344

= 27.24 N

P_T = $P_t + P_c + P_s$

= 85.26 + 110.27 + 27.24

= 222.77 N

Design load = K_s x P_T

Where,

K_s = $K_1 . K_2 . K_3 . K_4 . K_5 . K_6$

K_1 = 1.25 for variable shock load (Table11.8)

K_2 = 1 for adjustable supports (Table 11.9)

K_3 = 1 because we have taken

a_p= (30 to 50) p (Table 11.10)

K_4 = 1 drive is horizontal (Table 11.11)

K_5 = 1 for droop lubrication (Table 11.12)

K_6 = 1.0 (for 5 hrs/day work) (Table 11.13)

K_s = 1.25 x 1 x 1 x1 x 1 x 1 = 1.25

Design load = 1.25 x 222.77 = 278.46 N

Factor of safety = Breaking load / Design load

Here, Breaking Load = 31800 N

Factor of safety = 31800/278.46 = 11.4

Consulting table 11.1 we find for smaller sprocket speed of 1440 rpm and pitch 12.7 mm, the required minimum factor of safety is 11. We get the actual factor of safety greater and hence the design is safe.

Bearing Stress on rollers induced stress,

σ = $(P_t / K_s) / A$

A = 100 mm² (Table 11.6)

σ = (85.26 x 1.25) / 100

= 1.06 N/mm²

Allowable stress(σ) = 18.5 for small sprocket speed of 1440 rpm and pitch 12.7 mm from table 11.4. We find that the induced stress is less than the permissible value.

Length of the Chain,

L_p = $2_{ab} + \{(Z_1+Z_2)/Z\} + \{[(Z_2 - Z_1)/2\Pi]^2/a_p\}$

Where, ap = a_0/p = 344 / 12.7

= 27.08

Lp = $2 \times 27.08 + [(30+30)/2] + \{[(30-30)/2\Pi]^2/27.08\}$

= 84.17

= 86 links (rounded off to an even number)

Since we have to connect three pulleys, so we have to multiply it by 1.5 to get exact number of links

130 links is required

Actual length of the chain = 130 x P

= 130 x 12.7

= 1.65 mts.

6.10.2 Specification of the chain:

The transmission ratio	=	1
Pitch of the chain	=	12.7 mm
Tangential force	=	85.26 N
Centrifugal tension	=	110.27 N
Tension due to Sagging	=	27.24
Total load on chain	=	222.77 N
Breaking Load	=	31800 N
Design Load	=	278.46 N
Bearing stress on rollers	=	1.06 N/mm²
Number of links on the chain	=	86 links
Length of the chain	=	1.65 mts
Designation ISO Number	=	C8B-2
Rolon chain Number	=	DR 1278

6.11. DESIGN OF SPROCKET

P_{cd} of smaller sprocket

d_t = P/{Sin (180/Z_1)}

= 12.7 / [Sin (180/30)]

= 121.49 mm

Sprocket outside diameterdo$_1$ = $d_0 + 0.8\ d_r$

Here, d_r = 8.51 mm (Table 11.6)

do = 121.49 + (0.8 x 8.51) =128.30 mm

P_{cd} of larger sprocket d_2 = P / [Sin (180/Z_1)]

= 12.7 / [Sin (180/30)]

= 121.49 mm

do_1 = d_2 + (0.8 d_r)

= 121.49 + (0.8 x 8.51)

= 128.30 mm

6.10.1 Specification of Sprocket:

Pcd of smaller Sprocket	=	121.49 mm
Outside diameter of smaller Sprocket	=	128.30 mm
Pcd of larger sprocket	=	121.49 mm
Outside diameter of bigger sprocket	=	128.30 mm
Number of teethes on sprocket	=	30

7. MATERIAL SELECTION

The most important part of engineering practice is the choice of the material for our machining requirements, and the development of new ways for using them with greater effectiveness. The selection of the material for the machining is reduced to three broad constraints.

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

The service requirements are permanent but the material costs must stand up to service demand. Such demand commonly includes dimensional stability, corrosion resistance, strength, hardness, toughness and heat resistance.

A fabrication requirement includes the possibility to shape the material and to join it with other materials. The assessment of fabrication requirements concerns the question of machinability, hardenability, ductility, castability and weldability, which are some times difficult to assess.

Finally, the economic requirement is essential so that the overall cost of machining and fabrication are maintained to an optimum level without compromising with the quality.

Other factors which are considered for the selection of materials are

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

The materials used for fabrication of various parts of our compressor are: Mild steel, Cast iron, Cast Aluminium.

7.1 Mild Steel:

This is low Carbon steel with no precise control over the composition or mechanical properties. The cost is low in comparison with other steel and it can withstand high pressure.

7.2 Cast Iron:

It is the least expanding one when compared to all the metals which 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 inadequate. Elastic modulus is only $9 \times 10^{\square}$ N/mm^2.

7.3 Cast Aluminium:

Cast Aluminum has good fluidity / pressure tightness and resistance to corrosion and are suitable for intricate castings. The chemical composition of cast aluminium is 0.1% Zn, 0.2%Sn, 0-1%Pn, 0-0.5% Sn and rest aluminium.

8. FABRICATION

8.1 FABRICATION DETAILS OF THE COMPRESSOR

8.1.1. Compressor Head
8.1.2 Cylinder Block
8.1.3. Connecting Rod
8.1.4. Crank Shaft Assembly
8.1.5. End Bearing
8.1.6. Stand

8.1.1. 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 320mm is taken and is turned and faced in Lathe to a triangular shape. (Annexure 3, 4, 5, 6.)

Then groove is taken on the three sides in order to fix the cylinder block to it. Then, a central drill of diameter 12.7mm is drilled in its length and the 3 drills of 15mm diameter are drilled from exactly centre of the side to the central drill.

8.1.2. Cylinder Block:

The material used for cylinder block is cast iron. The Cylinder Block is first cast and then machined. The required surface finish is achieved with suitable methods like boring and honing.

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.

8.1.3. Connecting Rod:

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

(Annexure 2)

8.1.4. 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, grinded and the ends are tapered. All the tapers are checked using ring gauge to check whether 30 ° taper angles is provided.

The web is separately 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 crankshaft has its centre of gravity offset from the axis of rotation of the crankshaft. Hence it acts as balancing weight. (Annexure 7.)

8.1.5. End Bearings:

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

8.1.6. Stand:

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

8.2 OPERATION

Examine the unit for transit damages.

8.2.1. Electrical Connection

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

8.2.2. Adjusting Chain Tension

The chain tension between the motor sprocket and crankshaft sprocket must be correctly adjusted with proper tension. Otherwise, the compressor will not run for the required speed.

8.2.3. Starting Procedure

The following are to be checked before starting the unit.

- The oil level in the oil indicator was checked. If the oil level indicator's level was below the minimum mark, fresh oil was added to the correct grade.
- The chain tension was checked.
- The suction pipe air filler was checked.
- Care should be taken to check whether the unit rotates freely by hand and that there are no mechanical obstructions.
- The unit was started and allow to run for a few minutes.

8.2.4. Checks during operation:

- The running sound was checked during the operation period.
- The pressure developed was checked by opening the manometer safety device.

If all the above preliminary checks are found satisfactory, then the unit may be put to regular use.

9. MERITS:-

1) Best suitable for low pressure application.
2) Less weight compared to other compressors.
3) Production cost is low.
4) Higher efficiency because of less power requirement.
5) High Durability.
6) Compactness of the compressor saves utilization space.
7) Easy to use and portable.

10. APPLICATIONS

1) Used in laboratories and pharmaceuticals.
2) Used for fabrication of plastic structure with hot guess.
3) Used for best control fumigation service.
4) Used for air agitation of photo film processing tanks, electroplating bath and in chemical plants.
5) Used for cooling electronic circuits.
6) Used for inflating tyres and air mattresses.
7) Used for light duty spray painting.
8) Used to operate air driven hand tools such as Die-Polishers, Die-Grinders etc.,
9) Used in bore wells to deliver water from the well.

11. MAINTENANCE PROCEDURE

11.1. Lubrication:

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

11.2. Daily:

- The oil level was checked.
- The chain tension was checked.

11.3. Every two hours of operation:

The suction filter was cleaned, to ensure long life for the valves and the piston assembly.

11.4. Every 200 hours of operation:

1. The chain tension was checked and adjusted.
2. All the bolts are checked for tightness.
3. The developed pressure was checked and the safety valve was adjusted, if necessary.
4. The bearing sound was checked.

11.5. Every 1000 hours:

The suction and delivery valves are removed and the valve seats seating are inspected for any score or damage. The valves may be lapped, if necessary in their respective seats using fine lapping compound. The crank shaft was dismantled and then assembled by applying grease.

11.6. Every 3000 Hours:

The entire unit was dismantled by an experienced hand, who knows about the compressor in detail and a general overhauling was done. This includes inspection of all parts for wear and tear and replacement of damaged components, checking clearance between various components and assembling.

12. FURTHER IMPROVEMENTS

12.1. Further Improvements:

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

1) Lubrication method to be improved to reduce wear and tear.
2) Intercooling can be used to reduce the work done when multistage compression is used.
3) Pressure rise can be increased by reducing the clearance volume.
4) The production cost can be reduced by casting the cylinder and the head assembly in to a single piece.

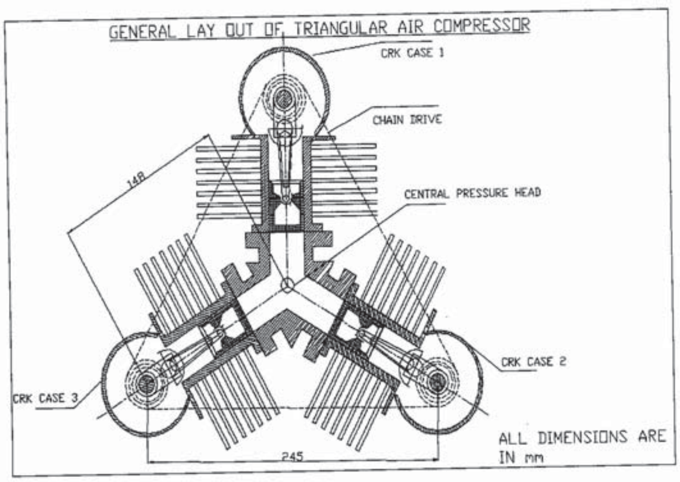
13. CONCLUSION

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

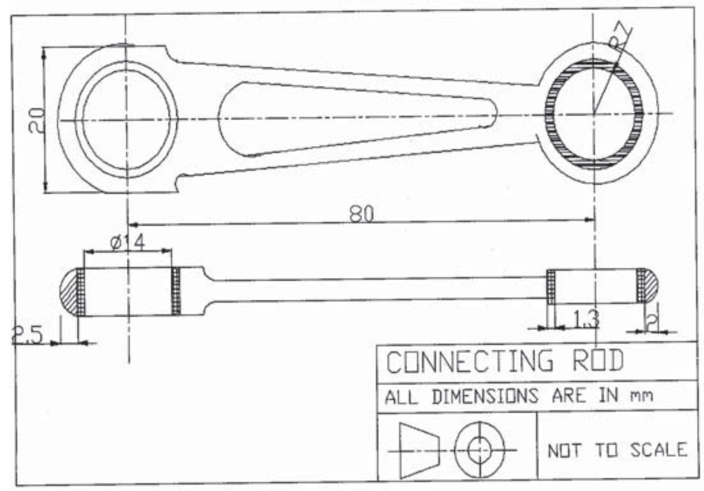
The performance of the compressor was found to be satisfactory and the output of the compressor is continuous. It is up to the level expected and further improvements can be done as mentioned above.

14. ANNEXURES:-

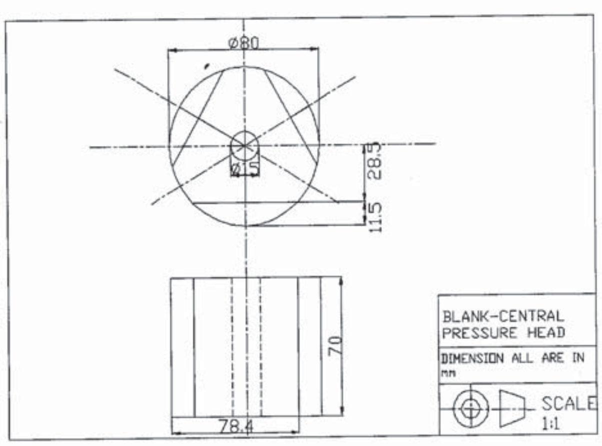
Annexure – 14.1



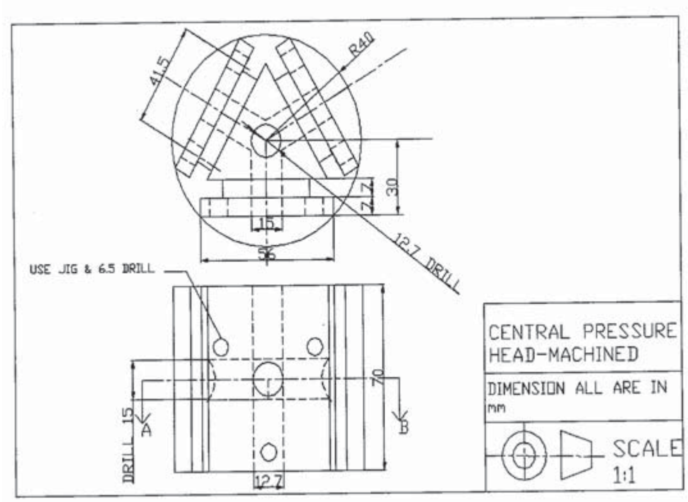
Annexure – 14.2



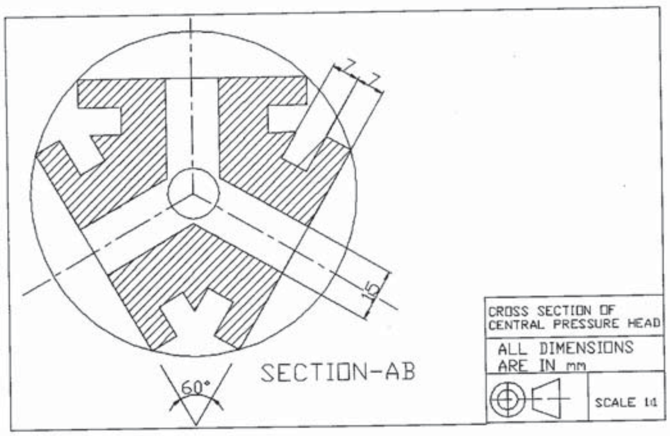
Annexure – 14.3



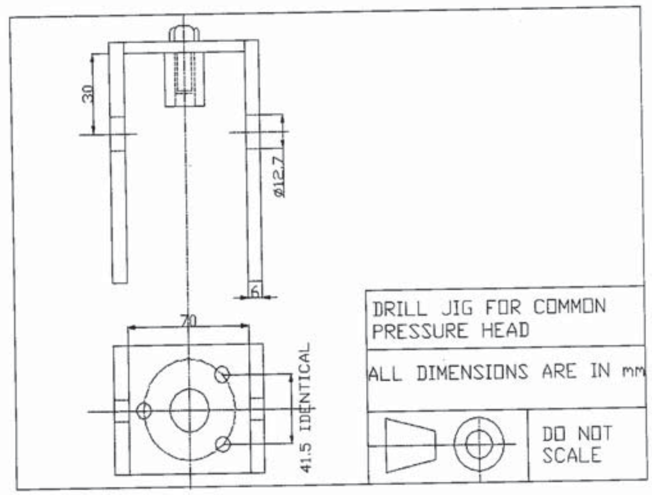
Annexure -14.4



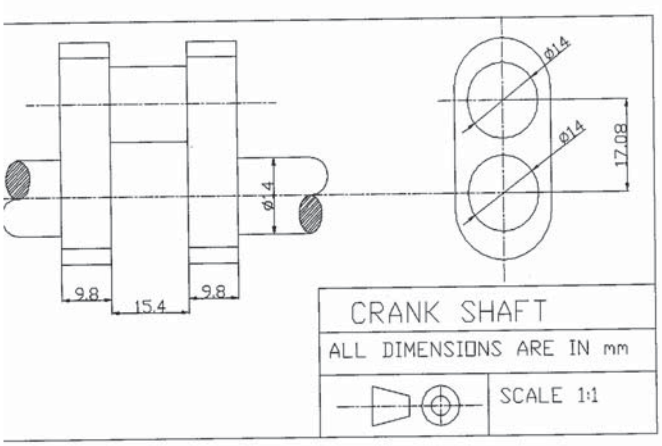
Annexure – 14.5



Annexure – 14.6



Annexure – 14.7



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17. COST ESTIMATION

Sl. No.	*COMPONENT*	Qty.	*COST / COMP*	*TOTAL COST*
01	L34 Crank Case Assembly	3	200	600
02	6202 SKF Bearing	6	50	300
03	Piston Assembly	3	500	1500
04	Cylinder block L34	3	400	1200
05	Oil seal L34	6	50	300
06	Crank Shaft L-34 connecting Rod set	3	120	360
07	Mounting plate MS	1	600	600
08	Chain	1	150	150
09	Sprocket	4	35	140
10	Valves	2	100	200
11	Motor mount platform	1	150	150
12	Motor 1 HP	1	1000	1000
			Total	6500