# STUDY OF SAFETY DESIGNS & MODIFICATIONS TO JUTE COMPOSITE CARD

CHARACTER IS LIFE

## Project Report 1998 - 99



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In partial fulfilment of the requirements
for the award of the Degree of
BACHELOR OF ENGINEERING
in Mechanical Engineering Branch
of Bharathiar University, Coimbatore

Department of Mechanical Engineering
Kumuraguru College of Technology

MILLTEX

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## CERTIFICATE

This is to certify that the following final year Mechanical Engineering students of Kumaraguru College of Technology, Coimbatore have successfully carried out the project entitled

**"JUTE CARD MANUFACTURE"**

Mr. BERNARD.G. RUBAN
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They had shown keen interest in the design and development of the Machine and their performance in the project work was excellent. We wish them all success in their career.

**For MILLTEX ENGINEERS P. LTD.**

A. SHANMUGASUNDARAM
Managing Director

Department of Mechanical Engineering

# Kumaraguru College of Technology

Coimbatore - 641 006

## Project Work 1997 - 98

Name ............................................................ Register No. ..............................

Certified that this is the Bonafide Record of the Project Work Done by

Mr. Dept Copy ............................................................

In partial fulfilment of the requirements for the Degree of Bachelor of Engineering in Mechanical Engineering Branch of the Bharathiar University.

.................................................. Head of the Department

12/3/99 .................................................. Guide

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Submitted for the University Examination held on 15.3.99 ..........

.................................................. Internal Examiner

15/3/99 .................................................. External Examiner

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# Synopsis

In the existing jute carding machines, the production rate are low and moreover, the quality of jute fibre is not as expected by the customer. We tried of improving the quality as well as increasing the production rate.

**To improve the production rate**

We have replaced the bush type bearing by self-aligned spherical roller bearing. The spherical roller bearings have two rows of roller which run on a common sphered raceway in the outer ring. The two inner ring raceways are inclined at an angle to the bearing axis. In addition to radial loads, the bearing can also accomodate axial load acting in both directions.

The production increases from 50 tonnes/day to 80 tonnes/day

**To improve the quality**

The husk and broken pieces are accumulated on the underside of the cylinder. We have introduced the covers at the underside of main cylinder, worker and stripper and have provided a blower to catch all the dust, and sent them to the dust collector. They are periodically removed. The provision of dust collector have reduced the mixing of the dust with the good jute. Thus increasing the quality of the fibre. Since the dust is not allowed to escape out, persons working nearby are not affected by dust thus maintaining the healthy atmosphere for the workers.

The density of staves and their angle of inclination determine the working quality of the jute fibre. We have changed the existing angle and density by trial and error method and fixed these optimum values.

| | Pins/square inch | Pin angle |
|---|---|---|
| Cylinder | 3.15 | 70o |
| Worker | 5.0 | 40o |
| Strippers | 5.0 | 40o |
| Duffer | 5.0 | 40o |

The existing mechanical brake is replaced by the A.C. solenoid brake, which is an electro - mechanical brake. It stops the machine in just 10 seconds giving at most safety to the machine components in case of emergency and at certain breakdowns.

# CONTENTS

## 1. INTRODUCTION

The pilot plant facility will go a long way in dissemenating the technology for small scale jute spinning, Training of Executives, Supervisors and workmen in Jute processing and assist new Entreprenuer to set up Small Scale Spinning units.

The technology of small scale spinning includes cutting of jute to stable length of 10 - 12 inches, softening and processing, in the mini Jute Cards (Breaker and Finisher), with suitable doublings in the Finisher card.

The carded sliver is then processed through the Mini Jute Drawing Frames (1 st Drawing, 2 nd Drawing) Open Gill type followed by (3 rd Drawing & Finisher Drawing with Bobbinor) Intersecting Gill type. Number of doublings to be suitably incorporated depending on the count of yarn to be spun.

The drawing rove is then spun into yarn in the Ring spinning frame followed by twisting in case of plied yarn followed by Reeling to conventional hanks of 54"/72"/90" or in spool form as required.

**JUTE FIBRE**

Jute is the common name given to the fibre produced from the stem of the plant belonging to the genus "Corchorus", family "Tiliaceae",. It has been the most important industrial fibre and has been used for many years mainly for textile applications. Jute for fibre production occupies the land, for about 4 -5 months and is commonly rotated with food crops.

The fibre lies along the length of the plant stem in the form of an annular meshwork composed of more than one fibre layer.

The commercial fibres, as obtained from the plant are 1.5 to 3.0 meters long, and when viewed in the transverse section under the microscipe, show from 6 to 20 or even as many as 50, single, thick walled polygonal cells (Ultimates) each containing a central canal or lumen.

It is not long single filament but minute fibre cells of about 2.5 mm cemented together to form the "Reed" with branches all through its length. The linear density of jute is in the range of 1.3 to 2.4 Tex and the number of ultimate cells in a bundle is 8 to 25.

The average yield of dry fibre varies from 1.6 tonnes / hectare for white jute to 2.0 t / ha for tossa jute. The green plants are composed of 15% leaves, 40% wood and 45% ribbin by weight. The plant contains about 6% dry fibre. An area of about 0.5 ha is needed to produce one ton

## FLOW CHART FOR JUTE AGRICULTURE

Sowing : ------>----- :Weeding/Thinning : ----->------- :Harvesting :------:

:

:----------:Retting :----------<--------:Drying/Defoliation : -----------------------:

:

:----->-------:Fibre Extraction :---------Fibre Washing :---->----: Fibre Drying

## QUALITY EVALUATION TECHNIQUES

| | | | |
|---|---|---|---|
| 1. | Colour & Lusture | : | Visual observation |
| 2. | Fineness | : | Measured as denier (Gravimetric method ie., cut and weigh) |
| 3. | Strength | : | Measured as Bundle strength in G/Tex using bundle strength tester. |

4. Grading kits are available at IJIRA & JTRL, Calcutta.

denier --------> weight in grams of 9000 m of fibre

G/Tex ---------> weight in grams of 1000 m of yarn

## THE JUTE PROCESSING SYSTEM

JUTE FIBRE SELECTION
|
SOFTENING
|
CARDING
|
DRAWING
|
SPINNING
|
WINDING
|
BEAMING
|
WEAVING
|
FINISHING
|
SEWING
|
PACKING
|
BAILING

## JUTE CARDING MACHINE

The primary function of the cards is to convert the reeds of jute into a uniform supply of fibrous material which can then be drafted and finally twisted into yarn. There are basically three stages of carding.

1. Breaker / Teaser Card
2. Inter Card
3. Finisher Card

Jute fibre when passed through the breaker card gets cut, carded and parallelised and transformed into a thin web of separate fibres emerging as a fleece which is then condensed into a sliver. In the three stages of carding, the work of weight reduction by drafting and weight levelling by doubling and mixing of various types of jute are carred out.

The jute carding machine consists of different cylinders performing different operations.

1. Main cylinder (1 No.)
2. Worker (3 Nos.)
3. Stripper (3 Nos.)
4. Doffer (1 No.)

## JUTE GAINING IMPORTANCE

It was always under an impression that jute was a neglected sector compared to all other fibres. Jute was regarded as a sunset industry. Everyone was concerned with textile exports, readymade garment quota issues, international policy implications of the multi - fibre policy, fluctuating prices of cotton, problems of NTC, propsects of development of sericulture in all parts of the country and thrust areas for the development of wollen and man - made fibre industry. In west Bengal, jute is most important cash crop.

The fluctuating prices of jute and the distress sales resorted to by the farmers were generaly the subject of matter of detailed reporting in Economic dailies with very little supportive or innovative action emenating from Government quarters. Every intervention was a palliative and evey decision was connected with the solving of the problem on a short - term basis. It was not appreciated that encouragement to synthetic packaging materials would gradually deprive the jute industry of its substantial share of the markets particulary in the cement and fertilizer sectors. As different arms of the Government were dealing with different aspects of jute and synthetic industries, it was never possible to harmonise the interests of both these industries and ultimately it was realised that the synthetic packing sector was growing at the cost of jute. For some of us, dealing with jute, it was a difficult to advocate aggressively the cause of jute as the new sector always topped in terms of new technologies, innovative products and new markeging strategies. It was realised gradually that the threat and challenge posed by the synthetics to jute could be converted into an opportunity for growth and development of jute on new lines hitherto unexplored.

**Fibre Question and its versatilty :**

In the beginning, the cause of jute was stated vociferously by the minisitry of textiles in different forums in terms of possible dangers to the jute industry. The cause of any industry in India could be convincingly advocated if the underlying interests of the farmers behind the industry could be properly projected. On this basis, the textile and the jute industry deserve to be well protected because the fate of millions of cotton growing, jute growing, sericulture farming and sheep rearing farmers, is twined with the fortunes of these industries. The synthetic or man - made industry may have its place in the scheme of things in the modern context in which we are placed but it cannot be at the cost of the natural fibres.

From a production level of 4.5 million bales in the beginning of 50's. the farmers have been able to achieve the production level of nine - million bales in the beginning of 90's. From the angle of soil climate, social needs and economic advantage, jute will always be cultivated irrespective of the campaign against jute by misguided policy makers and vested industrial interest.

Introduction of the concept of minimum support price during 1970 - 71 under the ages of jute corporation of India (JCI) Government of India undertaking, was a partial support mechanism for the jute growing farmers as it was and has also been over the last 25 years for the cotton growing farmers for whose benefit cotton corporation of India was set up around the same period. In case of JCI the operations were by and large confined to minimum support price (MSP) only and it could not exercise the option of remunerative price options as the customers coming to it for purchase of the crop on their behalf on commercial terms were few and far between. This was not surprising because except for the jute mills largely loacted in calculta and a few of them in Orissa, Andhra Pradesh, Assam and Tripura, there are no additional or major consumers of raw jute in the country. Cotton is also traded in international markets to several destinations and therefore the interests of the cotton growing farmers, intermediate consumers and end - users always get entangled if not harmonised.

It was therefore, becoming increasingly clear to us that if the fate of the jute farmer is solely linked to the fate of the jute mills, there was the certainty of a decision emanating from the higher quarters for phasing out jute cultivation, as it was an uneconomic proposition to the farmers. It was possible and feasible to utilise the fibre in textile and serveral non - textile areas. In the textile areas production of jute fine yarn or blender yarn could lead to production of attractive textile fabrics for furnishing, upholstery, floor covering and even apparel end - uses. Utilisation of fibre production of jute - based superior grades of paper as well as newsprint and composite material for funiture and construction applications as a substitute for wood.

It is not by accident that today the raw jute prices are at a level above Rs.1000 per quintal whereas these were below Rs.400 until 1987. For the first time over the last two years beginning from 1994 raw jute prices are becoming beyond the reach of the industry and there appears to be a nexus between the jute growing farmers and the traders.

## CARDING PROCESS

The reduction of jute from stick form to a sliver of suitable weight per length and sufficiently broken down for the subsequent process of DRAWING & DOUBLING, is called CARDING.

**BREAKER FEEDING : There are two types of Breaker - feeding :-**

1. Hand Feeding and 2. Roll - Feeding

1. In hand - fed Breakers, two operatives known as Breaker - feeders, spread out the "morahs" of jute on the Feed - cloth -- Root - ends first. Behind each pair of operatives, a bench - divided into 2 or 4 parts - is placed, on which another operative known as a Dollop - weigher places the bunches of jute "Morahs" after weighing. The weight of each of these bunches, is half that of the full dollop weight ; each of these weights of jute is to be spread out on the feed - cloth while the Clock - pointer moves through half - a - circle. Therefore, since two of these weights make a full dollop, the latter quantity will be spread over the Feed - cloth while the Clock - pointer moves through a full circle.

2. Rolls made in the Jute Goods Spreader m/c., are fed in the Roll - fed Breakers. In this case, the weight is regulated at the feed - end of Goods machine. One operative can easily look after two Breakers.

Carding may be accomplished by :

| | | | |
|---|---|---|---|
| 1. | DUAL CARDING | ... ... | Breaker & Finiser Cards operating independently. |
| 2. | MONO or SINGLE CARDING | ... ... | One Card (Combined Breaker Finisher) reducing the long sticks of Jute to the required condition for the Drawing process. |

3. TANDEM CARDING ... ... Breaker & Finisher Cards operating as one unit, the latter being, in some cases, driven by the former.

**CARDING ACTION**

The action of the different rollers of a Breaker Card and their effect on the ultimate fibre is given here under :-

**FEED - ROLLER**

The Feed Roller passes in the material from the Feed - cloth at a definite rate governed by the 'Clock - length' and 'Draft' on the machine. The back - pointing pins of the Feed - Roller in conjunction with the Cast - iron Shell providing the necesary restraining action. The angle of the pins of the Feed - Roller is such that the back of the pins is sufficiently keen to pull in the fibre but, nevertheless sufficient to hold the fibre against the 'pull' of the Cylinder and prevent the material from being gulped. As the jute is fed in, the pins of the fast moving cylinder split up and vigoriously card away the ribbons of fibre over the edge of the shell, so that it is fleeced out and carried off on the cylinder pins. The cylinder pins are set at such an angle that the material is worked by the points, as too flat an angle would permit the fibres to jam into the bottom of the pins which would

**WORKER :**

The inclination of the pins together with the slow surface speed of the Worker, is such that the material may be carried away from the action of the cylinder and not stripped by it. Thus the back - pointing pins of the Worker, opposing the cylinder pins point against point, catch up the loose fibre and at the same time open out and card fibre, held by the cylinder, as it is carried past. The fibrous "tow" retained by the Worker, is carried round by it and is ultimately combed away by the pins of the faster running Stripper. The stripper pins are forward - pointing.

**STRIPPER :**

The attenuated fibrous tow carried round by the Stripper, is carried away by the faster running cylinder, the fleece being thinned down considerably ; so much so is this the case that the cylinder takes the material from the Stripper mainly in the form of individual fibre which merge with the bulk on the cylinder. The angle of pins assists the effectiveness of Stripper - action.

**DOFFER :**

As the angle of pinning and surface speed are comparable to those of the Workers, there is a similarity of action. In this case, however the fleece is stripped by the Drawing - roller and the Pressing - roller and condensed into sliver by the V - shaped conductor.

**PINS & PINNING :**

The cylinder of the card may be either of wood or of cast - iron construction and the rollers may be wood or mild steel tubes, with steel shafts shrunk into cast - iron end - flanges.

## Comparison of Conventional Design and the proposed design

1. The conventional design of breaker feed allows the fibre to be attacked too harshly by the cylinder, resulting in compacting of the fibre on to the shell nose and resistance to the cylinder pins. The shel with modified angled breakers provides a gentle nose nip where the individual fibres are free to slip past their neighbours. This results in minimum fibre breakage and the production of longer and finer fibres.

2. In orthodox designs when the shell nose wears, the settings are upset and the complete shell must be removed and remachined, resulting in loss of production for an appreciable time. The proposed breaker shell nose is a replaceable hardened insert which has an extremely high resistance to the abrasive action of the fibre.

The JF4 finisher feed is a three - roller arrangement of plain roller, pinned roller and feed stripper. All three rollers are rigidly mounted in one stand, the arbors run in sealed - for - life ball bearings and the relationship of the rollers to each other and to the cylinder can be adjusted from the outside of the machine.

3. Many production hours have been lost because of fire between the cylinder ends and the shrouding. In many instances, fires have resulted and caused extensive damage to the whole machine.

In the proposed method, patented shrouding that has been specially designed to prevent such chokes. This design eliminates the fire hazard and increases the card efficiency. The smooth inner surfaces are completely free from screw holes and projections where fibres could be retained.

4. The Cylinder is a light weight, welded steel construction, with 100mm diameter arbors, running in sealed - for - life ball bearings. An extremely powerful electro - mechanical break is incorporated which reduces the stopping time from the usual 2 minutes to a mere 10 seconds.

5. Power consumption is low because all revolving members are mounted in ball bearing. The standard h.p. size of motors have been deliberately retained to allow an increase in the production of the card above the normal by either increased speed or increased sliver weight without fear of motor overload.

6. With only the gear teeth and the chains requiring occasional lubrication the cost of this routine procedure has been greatly reduced. This also ensures that breakdowns do not occur due to important lubrication points being overlooked.

7. Patented 'Metalag' light alloy clothing is used on all rollers. This type of clothing reduces pin consumption and repinning costs by

approximately 50% and because the same diameter of pin can be used for at least ten repinnings, there is a considerable reduction is the quantity and sizes of pins that require to be stocked.

The staves interlock circumferentially enabling the number of securing screws to be reduced with a consequent saving in time when restaving.

The light alloy material and the mutually supporting design prevents the staves from shattering and causing extensive damage to the machine.

To enhance the quality of jute fibre and to increase the production rate, proper selection of pin angle is necessary, using trial and error method, these pin angles are fixed for the various cylinders.

| Cylinder | Pin angle |
|---|---|
| Main cylinder | $70^\circ$ |
| Worker | $40^\circ$ |
| Stripper | $40^\circ$ |
| Doffer | $50^\circ$ |

# A.C. SOLENOID OPERATED BRAKES

AC Solenoid operated brake is designed for life long, easy installation and minimum maintenance. They are electrically released and spring applied providing "Fail-free" operation. The retarding torque developed is directly proportional to the spring pressure.

The brake wheel is of relatively large size in relation to the torque developed by the brake. This permits use of a larger brake shoe lining and low shoe pressure. Low shoe pressure equally distributed over a large lining area, results in even wear of the friction surfaces and even braking torque.

Low shoe pressure reduces the stress on all pins and pivot points to extend mechanical life and reduce maintenance cost. There is no unbalance of forces to cause side thrust during braking, because shoe type brake does not depend upon on a wrapping up action of the lining around the wheel.

**Applications**

Typical applications include conveyors, hoisting equipment, machine tools, printing presses, small cranes, overhead doors, vacuum moulding machines.

## Constructional features

Special constructional features such as those listed below account for exceptionally long mechanical life of the brakes.

i) Neoprene shock absorber – Prevents link pin breakage and increases the life of the solenoid.

ii) Hardened steel lever and spring gland – The hardened steel construction of the lever and spring gland reduces wear at the pins and all the other point of contact.

iii) Cast iron wheel – The physical properties of the iron minimise the tendency of the wheel to deposit the metal particles in the lining which would result in the serious scoring of the wheel.

iv) Spring pins – The tight gripping spring pins ensure against the loss of pins due to shock.

v) Inexpensive lining replacement – Linings attached to shoes by removable flat head groove-pins.

vi) Solenoid – The solenoid coil can be removed without disturbing the brake adjustment. Solenoid loading is designed to reduce wear.

# DESIGN MODIFICATION IN THE JUTE CARDING MACHINE

To improve the production rate, we change the bush type bearing to roller bearing.

To design the type of bearing, we need the various loads due to 1. Gear drive, 2. Chain drive, 3. Belt drive.

To find the loads, we design these drives separately.

## Design of Spur gear

Material Selection

$$i = \frac{D}{d} = \frac{Z_2}{Z_1} = \frac{90}{20} = 4.5$$

From P.S.G. Design Data Book pg.no. 8.4., Table 5

for i < 8, HB < 350

Pinion - C45

Wheel - C 35 Mn 75

Tooth Profile

Involute

Pressure angle

From P.S.G. Design Data Book pg.no.8.1

$\alpha = 20$

To Calculate centre distance (a)

From P.S.G. Design Data Book pg.no.8.13

$$a = (i+1) \times \sqrt[3]{\left(\frac{0.74}{[\sigma_c]}\right)^2 \times \frac{E[M_t]}{i\psi}}$$

<u>To find young's modules (E)</u>

From P.S.G. Design Data Book p.g.no.8.14

For Steel

$E = 2.15 \times 10^{6}$ kgf / $cm^{2}$

<u>Design of twisting moment</u>

From P.S.G. Design Data Book p.g.no.8.15

$[M_t] = M_t . K_d . K$

where,

$M_t$ = Nominal twisiting moment

$K_d$ = Dynamic Load factor

$K$ = Load correction factor

Initially assume for symmetric scheme

$K_d K = 1.3$

$$M_t = 97420 \times \frac{KW}{n}$$

KW = nominal power transmitted

= 7.36

n = Motor speed = 1440 rpm.

$$M_t = \frac{97420 (7.36)}{1440}$$

= 498

$[M_t] = 498 \times 1.3$

= 647.4 kgf.

## Design of compressive stress $[\sigma_c]$

From P.S.G. Design Data Book Pg.no. 8.16

$$[\sigma_c] = C_B HB.K_{cl} Kgf/cm^2$$

Where,

$C_B$ = Coefficient depending on the surface hardness.

For surface hardness HB < 350

Wheel material -----> Carbon steel and Alloy steel

$C_B = 25$

HB = Brinell hardness number

From P.S.G. Design Data Book Pg.No. 1.9,
for pinion material - C 45

HB = 229

$K_{cl}$ = Life factor

From P.S.G.Design Date Book Pg.No. 8.17 Table 17.

for material - steel and

Surface Hardness $\leq$ 350

Life in no. of cycles, $N > 10^7$ since $N = 60\ n\ T$

$T$ = Life in hrs

$N = 60 \times 1440 \times 10{,}000$

$= 86.4 \times 10^7$

$> 10^6$

$K_{cl} = 1$

$[\sigma_c] = 245 \times 229 \times 1$

$= 5725\ kg\ f/cm^2$

From P.S.G. Design Data Book Pg. no. 8.14,

Take $\psi = 0.3$

for open type gearing

$$a = 5.5 \times \sqrt[3]{\left(\frac{0.74}{5725}\right)^2 \times \frac{2.15 \times 10^6 \times 647.4}{4.5 \times 0.3}}$$

$= 14.2$ cm

From P.S.G. Design Data Book Pg.no. 7.20.
In Basic series of preferred numbers, choose R 10 series

$a_{min} = 16$ cm

Design of Module

From P.S.G. Design Data Book Pg.no. 8.13

$$m = 1.26 \times \sqrt[3]{\frac{[M_t]}{Y\,[\sigma_b]\;\psi_m Z_1}}$$

To design bending stress $[\sigma_b]$

From P.S.G. Design Data Book Pg.no. 8.18

$$[\sigma_b] = \frac{1.4\,K_{bl}}{n\;k_\sigma}\,\sigma_{-1}$$

Where,

$\sigma_{-1}$ = Endurance limit

From P.S.G. Design Data Book Pg.no. 8.19 Table - 19

For material -----> Cast steel

$\sigma_{-1} = 0.22\,(\sigma_u + \sigma_y) + 500$

Where,

$\sigma_u$ = Tensile Strength

$\sigma_y$ = Yield stress

From P.S.G. Design Data Book Pg.No. 1.9

For C 45 $\sigma_u$ = $67 \times 10^2$ kg f /mm$^2$

$\sigma_y$ = $36 \times 10^2$ kg f /mm$^2$

$\sigma_{-1}$ = $0.22\,(67 + 36) \times 10^2 + 500$

= 2766 kg f / cm$^2$

$K_{bl}$ = Life factor for bending

From P.S.G. Design Data Book Pg.no. 8.20

For material ------> steel

Surface hardness ------> < 350

Life in no. of cycles ------> > $10^7$

$K_{bl} = 1$

<u>To find stress concentration factor [$K_s$]</u>

From P.S.G. Design Data Book Pg.no. 8.19

For material and ------> Steel, Normalised
Heat Treatment -------> Surface hardened.

coeff. X (ie) O< x < 0.1 -----> 1.5

$K_s$ = 1.5

To find factor of safety, n

From P.S.G. Design Data Book Pg. no. 8.19

For material ------> steel

Heat treatment ------> Normalised

$n = 2$

$$[\sigma_b] = \frac{1.4 \times 1}{2 \times 1.5} \times 2766$$

$$= 1290.8 \text{ kg f / cm}^2$$

$Z_1$ = no.of teeth on pinion

= 20 (Assume)

From P.S.G. Design Data Book Pg.no. 8.14

$\psi_m$ = 10 (In general)

To find form factor (y)

From P.S.G. Design Data Book Pg. no. 8.18

Z = 20 (Already assumed),

Addendum modification coefficient x = 0,

Y = 0.389

$$\text{Module} \quad m = 1.26 \times \sqrt[3]{\frac{647.4}{0.389 \times 1290.8 \times 10 \times 20}}$$

$$m = 0.23 \text{ cm}$$

rounding of the value to the next standard value

$m = 2.5$ mm

From P.S.G. Design Data Book Pg. no. 8.2

Recommended series of modules

Preferred series 91,
$m = 2.5$ mm

from P.S.G. Design Data Book Pg. no. 8.22

To find no. of teeth

$$Z_1 = \frac{2a}{m(i+1)}$$

$$= \frac{2(16)}{0.25(4.5+1)}$$

$= 23$ teeth

$$Z_2 = iZ_1$$

$$= 4.5[23]$$

$= 104$ teeth.

To find actual centre distance,

From P.S.G. Design Data Book Pg.no. 8.22

$$a = \frac{m(Z_1 + Z_2)}{2}$$

$$= \frac{0.25(23+104)}{2}$$

$= 15.875$ cm

But the centre distance $= 16$ cm

$16 < 15.75$

Actual centre distance must be greater than centre distance

So Design is not safe

So Increase the Module

$$m = 3 \text{ mm}$$

$$Z_1 = \frac{2a}{m[i+1]} = \frac{2(16)}{0.3(5.5)}$$

$$= 20 \text{ teeth}$$

$$Z_2 = iZ_1 = 90 \text{ teeth}$$

$$a = \frac{m[Z_1 + Z_2]}{2}$$

$$= \frac{3(20+90)}{2}$$

$$= 16.5$$

$$16.5 > 16.0$$

So Design is safe.

To find face width

From P.S.G. Design Data Book Pg.no.8.1

$$\psi = \frac{b}{a}$$

$$b = 16.5 \times 0.3$$

$$= 4.95 \text{ cm}$$

$$\psi_m = \frac{b}{m}$$

$$10 = \frac{b}{0.3}$$

$b = 3$ cm

So take $b = 4.95$ cm

To check for compressive stress $[\sigma_c]$

From P.S.G. Design Data Book Pg.no. 8.13.

$$\sigma_c = 0.74 \times \frac{i+1}{a} \times \sqrt{\frac{i+1}{ib} \times E\,[M_t]}$$

$[M_t] = M_t.K.K_d$

To find K - load correction factor

From P.S.G. Design Data Book Pg.no.8.15

$$\psi_p = \frac{b}{d_1} = \frac{4.95}{0.3 \times 20} = 0.871$$

$$\cong 1$$

$[d_1 = mZ_1 = 0.3 \times 20]$

From Table 14

For $\psi_p = 1$, $k = 1.1$

From P.S.G. Design Data Book Pg.no. 8.15

$$V = \frac{\pi d_1 n_1}{60} = \frac{\pi (3)(20)(1440)}{60 \times 1000}$$

$$= 4.5 \text{ m/s}$$

$K_d$ = Dynamic load factor

For IS quality - cylindrical year = 5

Surface Hardness - $\leqslant$ 350

$K_d = 1.2$

$K_d . K. = 1.32$

$[m_t] = 647.4 \times 1.32$

$= 854.5$ kgf.

$$\sigma_c = 0.47 \times \frac{5.5}{16.5} \times \sqrt{\frac{5.5}{4.5 \times 4.95} \times 2.15 \times 10^6 \times 854.5}$$

$= 3337 \text{ kgf/cm}^2 < 5725$

So Design is safe

To Check for Bending stress

From P.S.G. Design Data Book Pg.no. 8.13

$$\sigma_b = \frac{i+1}{amby} [M_t]$$

$$= \frac{5.5}{16.5 \times 0.3 \times 4.95 \times 0.389} \times 854.5$$

$= 493 < 1290.8$

So Design is safe.

## Design of Belt Drive

Belt (i)

Power $P = 10\ HP = 7.36\ kw$

$N_1 = 1440\ rpm$

We know that,

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

$$N_2 = \frac{6.25}{14} \times 1440$$

$$= 623\ rpm$$

From P.S.G. Design Data Book Pg.no. 7.58

For $P = 7.36\ kw$

Select Type B

Minimum pulley pitch dia = 125 mm

From P.S.G. Design Data Book Pg.no. 7.61

<u>To find speed ratio i</u>

$$i = \frac{D}{d}$$

Where,

D = Dia of larger Pulley

d = Dia of smaller Pulley

$$i = \frac{14}{6.25}$$

$$i = 2.24$$

From P.S.G. Design Data Book Pg.no. 7.61

For $i = 2.24$

Recommended $\frac{C}{D}$ ratio = 1.0

$$C = 356 \text{ mm}$$

<u>To find norminal pitch length</u>

From P.S.G. Design Data Book Pg.no. 7.61

$$L = 2C + \frac{\pi}{2}(D + d) + \frac{(D - d)^2}{4C}$$

$$= 2 \times 356 + \frac{\pi}{2}(354 + 158) + \frac{(354 - 158)^2}{4 \times 356}$$

$$L = 1544 \text{ mm}$$

<u>To find the standard length</u>

From P.S.G. design data book Pg. No.7.59 for cross section B, L = 1567mm

<u>To find Maximum power capacity</u>

From P.S.G. design data Book Pg. No. 7.62 for section B

$$K.W. = (0.795^{-0.09} - \frac{50.8}{d_e} - 1.32 \times 10^{-4} S^2)\ S$$

Where,

S = Belt speed

$$= \frac{\pi d N_1}{60}$$

$$= \frac{\pi \times 15.8 \times 1440}{60} = 11.9 \text{ m/s}$$

$d_e$ = Equivalent Pitch diameter

From P.S.G. design data book Pg. No. 7.62

$d_e = d_p \times F_b$, Where $F_b$ = Small dia factor

for $\frac{D}{d} = 2.24$, $F_b = 1.13$ ($d_p = d = 158$)

$d_e = 158 \times 1.13 = 179$ mm

$$K.W. = [\,0.79\,(11.9)^{-0.09} - \frac{50.8}{179} - 1.32 \times 10^{-4} \times 11.9^2\,]\; 11.9$$

= 4 K.W.

To check this value

From P.S.G. design data book Pg. No. 7.64 for Belt speed = 12 m/s,
$d_e$ = 180mm
KW = 3.97 to 4

<u>To find No. of Belts</u>

From P.S.G. design data book Page No. 7.70

$$\text{No. of Belts} = \frac{P \times F_a}{KW \times F_c \times F_d}$$

Where,

$F_a$ = correction factor for industrial service.

From P.S.G. design data book Pg. No. 7.69 for heavy duty, machinery running the machine for 10 hrs. to 16 hrs,

$F_a = 1.5$

<u>To find $F_d$ (correction factor for arc of contact)</u>

From P.S.G. design data book Pg. No. 7.69

$$\text{Arc of contact angle} = 2\cos^{-1}\frac{(D-d)}{2C}$$

$$= 2\cos^{-1}\frac{(354 - 158)}{2 \times 356}$$

$$= 148^\circ$$

for this $148^\circ$ ,

From P.S.G. design data book Pg. No. 7.68,
for V belt $F_d = 0.92$

<u>To find $F_c$ (correction factor)</u>

From P.S.G. design data Book Pg. No. 7.59 for cross section B, for nominal pitch length 1567 mm

$$F_c = 0.92$$

$$\text{No. of belts} = \frac{7.36 \times 1.3}{4 \times 0.92 \times 0.92}$$

$$= 2.82$$

Standardising to next value,

= 3 belts

Belt (ii)

Power $P = 10$ H.P. $= 7.36$ KW

$N_1 = 643$ rpm

Dia of larger pulley $D = 635$ mm

Dia of smaller pulley $d = 127$ mm

From P.S.G. design data book Pg. No.7.58 for P = 7.36 KW, select B type.

Minimum pulley pitch diameter = 1.25 mm

## To find speed ratio (i)

From P.S.G. design data book Pg. No. 7.61

$$= \frac{D}{d} = \frac{635}{127} = 5$$

For i = 5

Recommended $\frac{C}{D}$ ratio = 0.9

$$c = 571.5 \text{ mm}$$

## To find nominal pitch length

From P.S.G. design data book Pg. No. 7.61

$$L = 2c + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4c}$$

$$= 2 \times 571.5 + \frac{\pi}{2}(635 + 127) + \frac{(635 - 127)^2}{4 \times 571.5}$$

$$= 2453 \text{ mm}$$

## To find standard length

From P.S.G. design data Book Pg. No. 7.60 for section B

$$L = 2507 \text{ mm}$$

## To find Maxi power capacity

From P.S.G. design data Book Pg. No. 7.62 for cross section B

$$KW = (079\ S^{-0.09} - \frac{50.8}{d_e} - 1.32 \times 10^{-4}\ S^2)S$$

$$\text{Belt speed } S = \frac{\pi d N_1}{60} = \frac{\pi \times 0.127 \times 643}{60}$$

$$= 4.27 \text{ m/s}$$

To find equivalent pitch dia ($d_e$)

From P.S.G. design data Book Pg. No. 7.62

$$d_e = d_p \times F_b$$

To find $F_b$

From P.S.G. design data Book Pg. No. 7.62

for $\frac{D}{d} = 5, \quad F_b = 1.14$ ($d_p = d = 127$mm)

$$d_e = 127 \times 1.14$$

$$= 145 \text{ mm}$$

$$KW = (0.79\,(4.14)^{-0.09} - \frac{50.8}{145} - 1.32 \times 10^{-4} \times 4.12^{2}\ 4.27)\ 4.27$$

$$= 1.46 \text{ KW}$$

To check this value

From P.S.G. design data Book Pg. No. 7.64 for Belt speed = 5 m/s

for $d_e = 140$ mm
KW = 1.54

To find No. of belts

From P.S.G. design data Book Pg. No. 7.70

No. of belts = $\frac{P \times F_a}{\phantom{x}}$

To find $F_a$

From P.S.G. Design Data Book Pg.no. 7.69

Running the machine over 10 hrs to 16hrs.

$F_a = 1.3$

From P.S.G.Design Data Book Pg.no. 7.68

$$\text{Arc of contact} = 2\cos^{-1}\frac{(D - d)}{2C}$$

$$= 2\,\text{Cos}^{-1}\frac{(635 - 127)}{2 \times 571.5}$$

$= 127.2 \qquad 130$

From P.S.G.Design Data Book Pg.no. 7.68

For 130 $F_d$ = 0.86 for V - V belt.

To find correction factor ($F_c$)

From P.S.G. Design Data Book Pg.no. 7.60

For L = 2507, $F_c = 1.02$

$$\text{No.of belts} = \frac{0.90 \times 7.36 \times 1.2}{1.4 \times 0.86 \times 1.02}$$

$= 6.2$

$\simeq$ 6 belts

## Design of Chain Drive

$$\text{Diametral Pitch } P_d = \frac{T}{D}$$

Where,

T = No.of teeth,

D = Pitch circle dia.

<u>Dia of Gear wheel (1)</u>

$$D = \frac{T}{P_d} = \frac{20}{6} = 3.33$$

$$= 8.46 \text{ cm} \simeq 8.5 \text{ cm}$$

<u>Dia of Gear Wheel (2)</u>

$$D = \frac{90}{6} = 15 = 38.1 \text{ cm}$$

Power P = 7.36 kw

Speed of motor = 1440 rpm

From P.S.G. Design Data Book Pg.no. 7.74

$$\text{Transmission ratio } i = \frac{Z_2}{Z_1}$$

where,

$Z_1$ = No. of teeth on sprocket pinion (Driving) = 20

$Z_2$ = No.of teeth on sprocket wheel (driven) = 95

$$i = \frac{Z_2}{Z_1}$$

$$= \frac{95}{20} = 4.75$$

## Selection of pitch

From P.S.G. Design Data Book Pg.no. 7.71

$P = 8$ (Assume simplex chain)

Design power = Rated Power X $K_s$

$K_s$ = Service factor

From P.S.G. Design Data Book Pg.no. 7.76

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

Where,

$K_1$ = Load factor
for constant load
$K_1$ = 1.00

$K_2$ = Factor for distance regulation
For drive using idler sprocket
= 1.1

$K_3$ = Factor for centre distance of sprockets
= 1

$K_4$ = Factor for the position of sprockets
(upto 60 inclination)
= 1

$K_5$ = Lubircation factor
for drop - lubrication
= 1.0

$K_6$ = Rating factor
for single shift of 8 hrs per day

$K_s$ + 1 x 1.1 x 1 x 1 x 1.0 x 1.0

$K_s$ = 1.1

Design Power = 7.36 x 1.1
= 8.096

To determine the type of chain

From P.S.G. Design Data Book Pg.no. 7.71

Assume roller chain.

Take R 830 - rolon type
ISO / DIN - 05 B -1

Other details

From P.S.G. Design Data Book Pg.no. 7.71

Roller dia $D_r$ = 5.00 mm

Width between inner plates W = 3.10 mm

Pin body dia $D_p$ = 2.31 mm

Plate depth = 7.05 mm

Overall over joint $A_1$ , $A_2$ , $A_3$ = 11.10 mm

Bearing area = 0.11 $Cm^2$

Weight per meter = 0.18 kg f

Breaking load = 460 kg f

To find pitch circle dia

From P.S.G. Design Data Book Pg.no.7.78

$$\text{Dia of small sprocket } d_1 = \frac{P}{\sin 180} \text{ mm}$$

$$= \frac{8}{\sin \frac{180}{20}} = 51.13 \text{ mm}$$

Dia of large sprocket $d_2 = \frac{P}{\sin \frac{180}{Z_2}}$ mm

$$= \frac{8}{\sin \frac{180}{95}} = 242 \text{ mm}$$

<u>To find velocity of chain,</u>

$$V = \frac{\pi \, d \, n}{60} = \frac{\pi \quad (51.13) \times 1440}{60} = 3.85 \text{ m/s}$$

<u>To find allowable factor of safety (n)</u>

From P.S.G. Design Data Book Pg.no. 7.77

For Speed of rotation of small sprocket as 1440 rpm

$n = 13.2$

<u>To check for actual factor of safety</u>

From P.S.G. Design Data Book Pg.no. 7.78

$$[n] = \frac{Q}{\varepsilon P}$$

Where,

$\varepsilon P = P_t + P_c + P_s$

$P_t =$ Targential load

From P.S.G. Design Data Book Pg.no. 7.78

$$P_t = \frac{102\ N}{V} = \frac{102 \times 8.096}{3.85}$$

$$= 214.50\ \text{kgf.}$$

$P_c$ = Centrifugal tension

From P.S.G design data book pg. No. 7.78

$$P_c = \frac{W \times v^2}{g}$$

$$P_c = \frac{3.10 \times 3.95^2}{9.81}$$

$$= 4.68\ \text{kgf}$$

$P_s$ = Tension due to sagging of chain

From P.S.G. design data book Pg. No. 7.78

$$P_s = K.W.a$$

<u>To find co-efficient of sag (k)</u>

From P.S.G design data book Pg. No. 7.78 for angle up to 40 K = 4

<u>To find centre distance 'a'</u>

From P.S.G. design data book Pg. No. 7.74

$$a = (30 \text{ to } 50)\ P$$

$$= 40P = 40 \times 8 = 320\ \text{mm.}$$

$\therefore P_s = 4 \times 3.10 \times 0.32$

$P_s = 3.968$ kgf

$\therefore P = P_s + P_c + P_t$

$= 214.5 + 4.68 + 3.968$

$\Sigma P = 223.139$ kgf

$\therefore$ Actual factor of safety

$$[n] = \frac{Q}{\Sigma P}$$

$$= \frac{460}{223.139}$$

$$= 2.06$$

But Allowable factor of safety

$n = 13.2$

$[n] \ngtr n$

So Design is not safe.

So Assume P = 12.7

From P.S.G. design data book Pg. No. 7.71

Take Roller chain - simplex type.

We have

R 1224, R 1230, R 1230H, R 1248,

R 1248H, R 1249, R 1253, R 1264, R 1252,

R 40, R 1278, R 1278H.

Select R 1252.

Rolon - R 1252

ISO/DIN - 083-1

Other details

$D_r$ = 8.51 mm

W = 5.30 mm

$D_p$ = 4.45 mm

G = 11.70 mm

$A_1, A_2, A_3$ = 18.10 mm

Breaking Area = 0.39 $cm^2$

Weight per metre = 0.58 kgf

Breaking load = 1820 kgf

From P.S.G design data book Pg. No. 7.78

$$d_2 = \frac{12.7}{\sin \frac{180}{95}} = 384 \text{ mm}$$

$$d_1 = \frac{12.7}{\sin \frac{180}{95}} = 81 \text{ mm}$$

To find velocity

$$V = \frac{\pi\ d\ n}{60} = \frac{\pi \times 0.81 \times 1440}{60}$$

$= 6.12$ m/s

To check for actual factor of safety.

From P.S.G design data book Pg. No. 7.78

$$[n] = \frac{Q}{\Sigma P}$$

Where, $\Sigma P = P_t + P_c + P_s$

Where $P_t = \frac{102 \times N}{V} = \frac{102 \times 8.096}{6.12}$

$= 134.93$ kgf

Where, $P_c = \frac{W \times v^2}{g}$

$$= \frac{5.3 \times 6.12}{9.81}$$

$= 20.235$ Kgf

Where, $P_s = K.w.a$

$= 4 \times 5.3 \times 0.508$

$= 10.76$ Kgf

(Here $a = 40P = 40 \times 12.7 = 0.508$ m)

$\varepsilon p \doteq 134.93 + 20.235 + 10.76 = 165.925$

$$[n] = \frac{Q}{\Sigma p} = \frac{1820}{165.925}$$

$= 10.96$

$10.96 \nless 13.2$

So Design is not safety since Actual factor of Safety is less than allowable factor of safety.

From P.S.G. design data book Pg. No. 7.72 Assume roller chain

ISO / DIN - 10 B -1
Rolon - R 1595

<u>Other details</u>

Pitch = 15.875mm

$D_r$ = 10.16 mm

W = 9.85mm

$D_p$ = 5.08 mm

G = 14.30 mm

Bearing area = 0.67 Cm

Weight / meter = 0.91 Kgf

Breaking load =2270 Kgf

From P.S.G. design data book Pg. No. 7.78

Dia of small sprocket $d_1$ = $\frac{P}{\sin(180 / Z_1)}$ mm

$$d_2 = \frac{15.875}{\text{Sin}\,(180 / 95)} = 480.13 \text{ mm}$$

Dia of large sprocket $d_2 = \dfrac{P}{\text{Sin}\,(180 / Z_2)}$ mm

$$= \frac{15.875}{\text{Sin}\,(180/20)} = 101.48 \text{ mm}$$

<u>To find velocity of chain</u>

$$V = \frac{\pi\, d\, n}{60} = \frac{\pi\,(101.48) \times 1440}{60}$$

$$V = 7.65 \text{ m/s}$$

<u>To check for Actual factor of safety</u>

$$[n] = \frac{Q}{\Sigma p}$$

where, $\Sigma p = P_t + P_c + P_s$

$$P_t = \frac{102\, N}{V} = \frac{102 \times 8.096}{7.65}$$

$$= 107.9 \text{ Kgf.}$$

$$P_c = \frac{W \times v^2}{2}$$

$$= \frac{0.91 \times 7.65^2}{9.81}$$

$$= 5.428 \text{ Kgf.}$$

$P_s = K.w.a$

From P.S.G. design data book Pg. No. 7.78
for angle upto 40, K = 4

Centre distance (s)

From P.S.G. design data book Pg. No. 7.74

$a = (30 \text{ to } 50)\ P$

$= 40\ p = 40 \times 15.875$

$= 635mm = 0.635\ m$

$P_s = 4 \times 0.91 \times 0.635$

$P_s = 2.31$ Kgf.

$\Sigma p = P_t + P_c + P_s$

$= 107.9 + 5.628 + 2.311$

$\Sigma p = 115.84$ Kgf

Actual factor of safety

$$[n] = \frac{Q}{\Sigma p}$$

$$= \frac{2270}{115.84}$$

$$= 19.5$$

Allowable factor of safety is
$n = 13.2$

So $[n] > n$

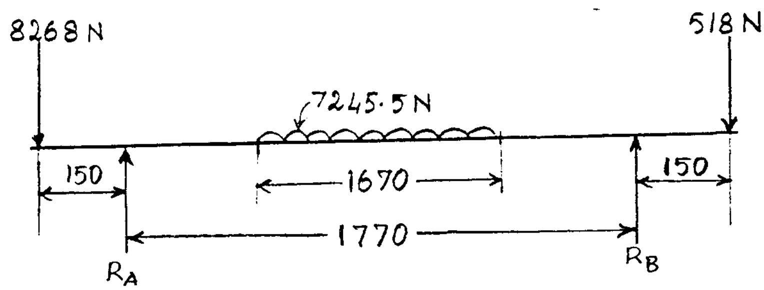
(ie) Actual factor of safety is greater than allowable factor of safety.

Design is safe.

So we choose R 1592 rolon chain Simplex chain

## Design of bearings

### (a) for main cylinder




Uniformly distributed load

$$1210 \times 9.81 = 12100N$$

$$= \frac{12100}{1.67}$$

$$= 7245.5 \text{ N/m}$$

Load due to belt

$$T = T_1 + T_2 + 2T_c$$

$$\frac{T_1}{T_2} = C^{\mu\theta \text{ cosec } \beta}$$

Cosec 20°= Cosec 0.698

$$= C^{0.25 \times 2.22 \times \text{Cosec }(0.698)}$$

$$= 5.06$$

Where, $2\beta = 40°$

$$= 40 \times \frac{\pi}{180} = \frac{0.698}{2} = 0.349$$

Arc of contact angle

$$\theta = 127^\circ = 127 \times \frac{\pi}{180} = 2.22$$

$\mu = 0.25$

$T_1 = 5.06\, T2$ ----------> 1

To find power transmitted by belt

$P = (T_1 - T_2)V$

$1460 = (T_1 - T_2)\, 4.27$

$T_1 - T_2 = 341.92$ ---------> 2

Solving 1 and 2

$5.06\, T_2 - T_2 = 341.92$

$T2\,(4.06) = 341.92$

$T2 = 84.21$ N

$T1 = 426.13$ N

We know $T_c = m\, v^2$ m = 0.212 Kg/m

$= 0.212 \times 4.27$

$= 3.8$ N

$T = T_1 + T_2 + 2T_c$

$= 4216.13 + 84.21 + 2\,(3.86)$

$= 518$ N

$8268 + 7245.5 + 518 = R_A + R_B$

$R_A + R_B = 16031.5$ N -----------> 1

Taking moment about A

$$578\,(192) + \frac{7245.5 \times 167}{167/2}\,(93.5) = R_B \times 177 + 8268 \times 15$$

$R_B = 1774.9$ N

$R_A = 14256$ N

Bearing at B

$F_r = 1774.9$ N $= 181$ Kgf.

$$F_a = \frac{181}{4} = 45.25 \text{ Kgf.}$$

$$\frac{F_a}{F_r} = \frac{1}{4} = 0.25$$

From P.S.G. design data book Page No. 4.4.

Take $\dfrac{F_a}{C_o} = 0.04$

for

$$\frac{F_a}{C_o} = 0.04 \quad \text{=======>} \quad e = 0.24$$

From this we see $\dfrac{F_a}{F_r} > c$

So from the table

for $\dfrac{F_a}{F_r} > c$

$X = 0.56$

$Y = 1.8$

$S = 1.1$

From P.S.G. design data Book Pg. No. 4.2

P = Equivalent load

$P = (XF_r + YF_a)\ s$

$= (0.56\ (180.9)\ ) + 1.8\ (45.25)\ \ 1.1$

$= 200.99$ Kgf.

From P.S.G. design data Book page no.43

$L = 60{,}000$ hrs

Maxi speed = 200 rpm

$L = 60{,}000 \times 200 \times 60$

= 720 mr (mr = million revolution)

From P.S.G. design data book book Pg. No. 4.2

$$\text{Dynamic capacity } C = \left(\frac{L}{L_{10}}\right) P$$

$$\text{Here } K = \frac{10}{3} \text{ for roller bearing}$$

$L_{10} = 1$ mr

$$C = \left(\frac{720}{1}\right) \times 201$$

$C = 1801.5$

From SKF - general catalogue Pg. No. 246

Choose 21304 CC type spherical roller bearing with cylindrical bore.

**Bearing at A**

$F_r$ = 14256 N = 1453.2 Kgf

$F_a$ = 3564 N = 363.3 Kgf

$$\frac{F_a}{F_r} = \frac{1}{4} = 0.25 > 0.24$$

From P.S.G. design data book Pg. No.44

for $$\frac{F_a}{F_r} > 0.24$$

X = 0.56

Y = 1.6

S = 1.1

From P.S.G. design data Book Pg. No.4.2

$$P = (XF_r + YF_a)\ S$$

$$= (0.56 \times 1453.2 + 1.6 \times 363.3)\ 1.1$$

$$= 1395 \text{ Kgf.}$$

From P.S.G. design data book Pg. No.4.4

L = 60,000 hrs.

Max rpm = 200.

$$L = 60{,}000 \times 200 \times 60$$

$$= 720 \text{ mr.}$$

From P.S.G. design data Book Pg. No. 4.2

$$C = \left(\frac{L}{L_{10}}\right) P$$

Here $K = \frac{10}{3}$ for ball bearing

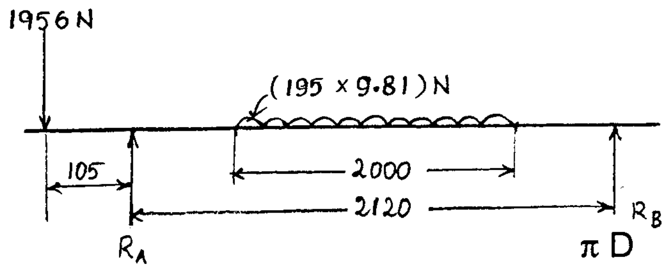
$$C = \left(\frac{720}{1}\right) \times 1395$$

$$= 12,503$$

From SKF - general catalogue Pg. No. 246

Choose 21304 CC type spherical roller bearing with cylindrical bore.

**Design of Bearings for worker cylinder**




We know $P = \frac{\pi D}{T}$

$90T \quad \frac{5''}{8} \; p$

1" = 2.54 cm

for $\frac{5''}{8} \implies 2.54 \times \frac{5''}{8}$

$$\frac{5}{8} \times 2.54 = \frac{\pi D}{90}$$

We know $P = \dfrac{2\pi NT}{60}$ (1 HP = 0.736 KW
N = 158 rpm)

$$10 \times 73 = \frac{2\pi}{60} \times 158 \times T$$

$$T = 445\ Nm$$

$$D = 45.5\ cm$$

$$R = 22.75\ cm$$

$$F_t = \frac{T}{R} = \frac{445}{22.75 \times 10}$$

$$= 1956\ N$$

<u>To find reactions</u>

Total upper load = Total down load

$$R_A + R_B = 1956 + (195 \times 9.81)$$

$$= 3868.95\ N$$

<u>Taking moment about A,</u>

$$R_B \times 2120 + 1956 \times 105 = \frac{195 \times 9.81}{r} (1060)$$

$$R_B = 859\ N$$

$$R_A = 3009\ N$$

**Bearing at B**

$$F_r = (R_B)^2_V + (R_B)^2_H$$

$F_r = R_B$ since $(R_B)V = 0$

$$\frac{F_a}{F_r} = 0.25$$

$F_a = 214.75$ N

From P.S.G. design data Book Pg. No.4.4

$$\frac{F_a}{C_o} = 0.04$$

for $\frac{F_a}{C_o} = 0.04 \implies c = 0.24$

$$\frac{F_a}{F_r} > c$$

($F_r$ = 859 N = 87.50 Kgf

$F_a$ = 214.75 N = 21.84 Kgf)

$X = 0.56$

$Y = 1.8$

$S = 1.1$

From P.S.G. design data book Pg. No. 4.2

$P = (XF_r + YF_a)\ S$

$= (0.56 \times (87.56) + 1.8\ (21.89)\ )1.1$

$P = 97.28$ Kgf

From P.S.G. design data book Pg. No. 4.4

L = 60,000 hrs

Max rpm = 158

$L = 60{,}000 \times 158 \times 60$

= 568.8 mr

From P.S.G. design data book Pg. No. 4.2

$$C = \left(\frac{L}{L_{10}}\right) P$$

Here

K = 3

$$C = \left(\frac{568.8}{1}\right)(97.25)$$

$$= 806$$

From SKF general catalogue Pg. No. 246

Choose 21340 CC type spherical roller bearing with cylindrical bore.

**Bearing at A**

$F_r$ = 3009 N = 306.7 Kgf.

$F_a$ = 752.25 N = 76.68 Kgf

$$\frac{Fa}{Fr} = \frac{1}{4} = 0.25 > 0.24$$

for

$$\frac{F_a}{F_r} > 0.24$$

From P.S.G. design data book Pg. No.4.4.

X = 0.56

Y = 1.6

S = 1.1

From P.S.G. design data book pg. No.4.2

$$P = (XF_r + YF_a) S$$

$$= (0.56 \times 306.7 + 1.6 \times 76.68)\ 1.1$$

$$P = 323.9 \text{ Kgf}$$

From P.S.G. design data book Pg. No.4.4

L = 60,000 hrs

Max rpm = 158 rpm

$$L = 60,000 \times 158 \times 60$$

$$= 568.8 \text{ mr}$$

From P.S.G. design data book Pg. No.42.

$$C = \left(\frac{L}{L_{10}}\right) P$$

Where $K = \frac{10}{3}$ for roller bearing

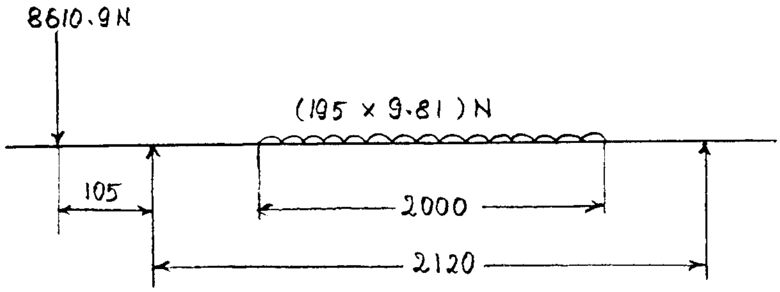
$$C = \left(\frac{568.8}{1}\right) \times 324$$

$$= 2173$$

From SKF general catalogue Pg. No. 236

Choose 21304 CC type spherical roller bearing with cylindrical bore.

## Design for bearing for stripper




$$21T \quad \frac{5''}{8} \; P \qquad\qquad P = \frac{\pi D}{T}$$

$$1'' = 2.54 \text{ cm}$$

$$\frac{5}{8} \times 2.54 = \frac{\pi D}{21}$$

$$D = 10.6 \text{ cm}$$

$$\text{We know } P = \frac{2\pi NT}{60}$$

$$10 \times 736 \times 2 \quad \times \frac{154}{60} \times T$$

$$\Longrightarrow \quad T = 456.4 \text{ Nm}$$

<u>Load due to chain</u>

$$Ft = \frac{T}{R_C} = \frac{456.38}{(10.6/2) \times 10} = 861.09 \text{ N}$$

$$= 877.7 \text{ Kgf}$$

To find reactions

Total upward load = Total downward load

$R_A + R_B = 8610.9 + (195 \times 9.81)$

$= 10524\ N$

Taking moment about A

$8610.9 \times 105 + R_B \times 2120$

$$= (195 \times 9.81)\left(\frac{2000}{2} + 60\right)$$

$R_B = 530\ N$

$R_A = 9994\ N$

**Bearing at A**

$F_r = 9994\ N = 1018.7\ Kgf$

$F_a = 2498.5\ N = 255\ Kgf$

$$\frac{F_a}{F_r} = \frac{1}{4} = 0.25$$

From P.S.G. design data book Pg. No. 4.4

for $\frac{F_a}{C_o} = 0.04 \implies c = 0.24$

$$\frac{F_a}{F_r} > c$$

for $\frac{F_a}{F_r} > c$

$X = 0.56$

$Y = 1.8$

$S = 1.1$

From P.S.G. design data Book Pg. No. 4.2

$P = (XF_r + YF_a)S$

$= (0.56 \times 1018.7 + 1.8 \times 255)\ 1.1$

$P = 1132.45$ Kgf

From P.S.G. design data book Pg. No.4.4

$L = 60{,}000$ hrs

Max. rpm = 154 rpm

$L = 60{,}000 \times 154 \times 60$

$= 554.4$ mr

From P.S.G. design data Book Pg. No.4.2

$$C = \left(\frac{L}{L_{10}}\right) P$$

Where $K = \frac{10}{3}$ for roller bearing

$$C = \frac{554.4}{1} \times 1132.41$$

$C = 7536.$

From SKF general catalogue Pg. No.246

Choose 21304 CC type spherical roller bearing with cylindrical bore.

**Bearing at B**

$F_r$ = 530 N = 54 Kgf

$F_a$ = 132.5 N = 13.5 Kgf

$$\frac{F_a}{F_r} = \frac{1}{4} = 0.25 > 0.24$$

From P.S.G. design data book Pg. No. 4.4

for

$$\frac{F_a}{F_r} > 0.24$$

X = 0.56

Y = 1.6

S = 1.1

From P.S.G. design data book Pg. No.4.2

$P = (XF_r + YF_a)S$

P = (0.56 x 54 + 1.6 x 13.5) 1.1

= 57 Kgf

From P.S.G. design data book Pg. No.4.4

L = 60,000 hrs

Maxi rpm = 154 rpm

L = 60,000 x 154 x 60

L = 554.4 mr

From P.S.G. design data book pg. No. 4.2

$$C = \left(\frac{L}{L_{10}}\right) P$$

(Where k = 10/3 for roller bearing)

$$= \left(\frac{554.4}{1}\right) 57$$

$$= 379.33$$

From SKF general catalogue Page 246

Choose 21304 CC type spherical roller bearing with cylindrical bore.

## SPHERICAL ROLLER BEARING

Spherical roller bearing have two rows of rollers which run on a common sphered raceway in the outer ring. The two inner ring raceways are inclined at an angle to the bearing axis. The bearings are self-aligning and insensitive to minor angular misalignments of the shaft relative to the housing, or to shaft bending. In addition to radial loads, the bearing can also take up axial loads acting in both directions.

SKF spherical roller bearings have a large number of long, symmetrical rollers of large diameter and consequently very high load carrying capacity. The rollers are guided by the raceways, the cage and a non-integral guide ring which is positioned between the two rows of rollers. The internal design of SKF spherical roller bearings represents the latest state of the art. The special geometry and the optimum finish of the raceway surfaces ensures a minimum of friction for bearings. These bearings operate at lower temperatures or can accommodate heavier axial loads or can run at higher speeds than earlier designs.

**Limiting Speeds :**

The reduced friction of the new SKF spherical roller bearing designs has permitted the limiting speeds for the majority of these bearings to be increased compared to those of the earlier designs. In some cases, this increase is quite substantial so that SKF spherical roller bearings can now be used in applications where self - aligning bearings are to advantage, but where previously other bearing types had to be used for speed capacity reasons. If desired, a check of the suitability can be made using a computer programme developed by SKF to predict operating temperatures for spherical roller bearings running under defined operating conditions in a known arrangement.

The new bearing designs not only have lower friction and are able to operate at higher speeds, they can also accommodate appreciably heavier axial loads. For these bearings, therefore, no reduction to the limiting speed is necessary as long as the ratio of the axial to the radial load is $F_a / F_r \leqslant 1$.

For the large - size bearings which have not yet been converted to the latest design, the correction factors previously applied to the limiting speed for bearings operating under combined load are still vaild. They are as follows.

| | | |
|---|---|---|
| $F_a / F_r$ | 0.3 | 0.6 |
| Factor | 1 | 0.8 |

## Misalignment

By virtue of their design, spherical roller bearings are self - aligning, i.e., they can accommodate a certain amount of misalignment between outer and inner rings without detrimental consequences. Under normal loads and operating conditions and where the inner ring rotates, the permissible angular misalignment is shown in the adjacent table. Whether this self - aligning capacity can be fully utilised depends, however, on the bearing arrangement design, the type of seals used etc.

## Heat Treatment

SKF spherical roller bearing are subjected to a spe°cial heat treatment so that they can be used at temperatures of up to +200 C without any inadmissible dimensional changes occuring.

## Masses

The masses given in the bearing tables are for bearing with cylindrical bore; the masses of bearings with tapered bore are listed in the tables showing bearings together with sleeves.

## Appropriate Housings

Appropriate housings for spherical roller bearings with cylindrical bore and for bearings with adapter sleeves will be found in the section "Bearing Housings".

**Axial load carrying ability of bearings mounted on adapter sleeves :**

If spherical roller bearings with adapter sleeves are mounted on smooth shafts with no fixed abutment, the magnitude of the axial load which can be carried is dependent on the friction between shaft and sleeve. Provided the bearings are correctly mounted, the permissible axial load can be calculated from :

$F_{ap} = 3\,B\,d$

Where

$F_{ap}$ = Maximum permissible axial load, N

$B$ = Bearing width, mm

$d$ = Bearing bore diameter, mm

Equivalent dynamic bearing load

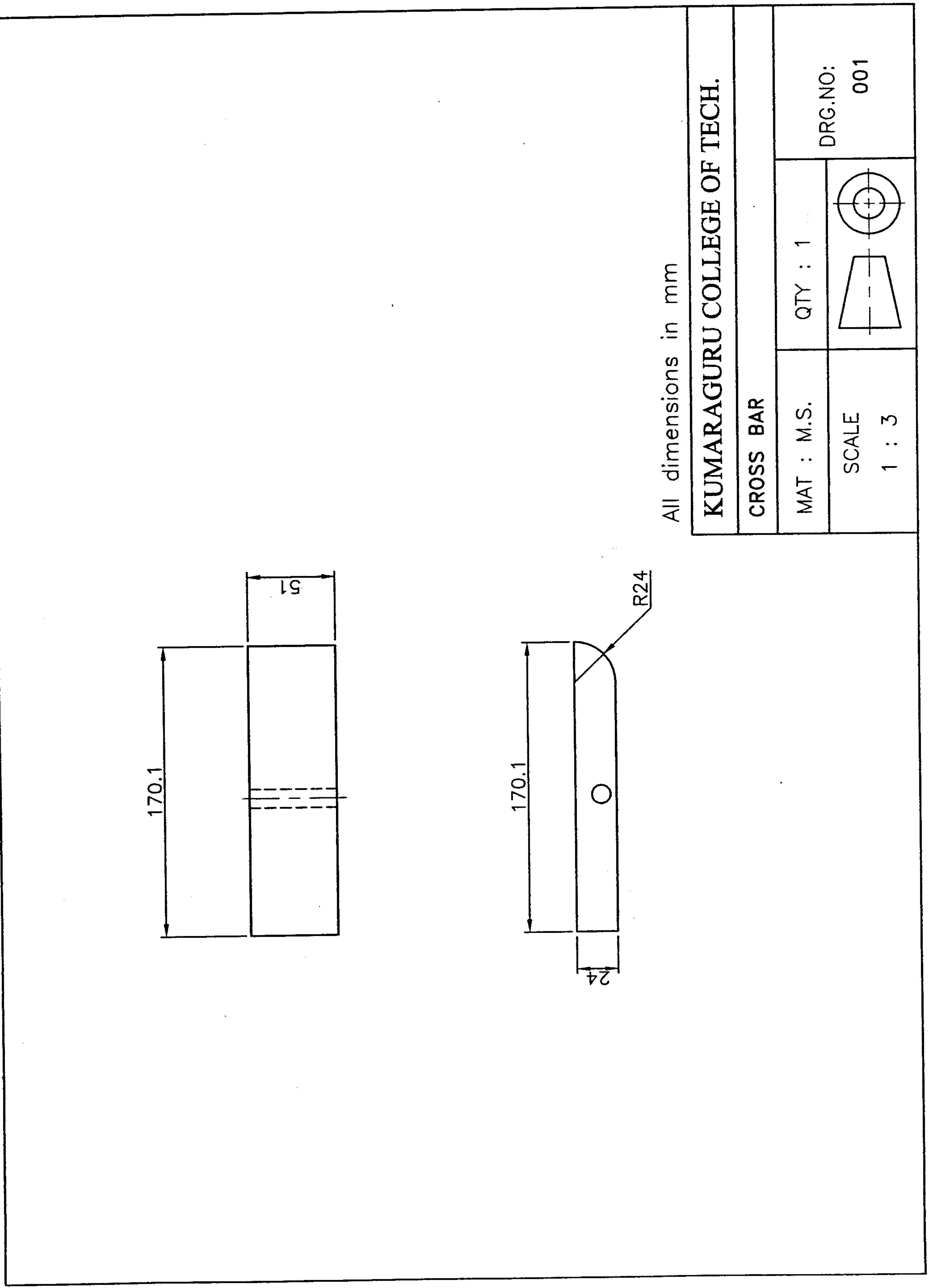
$P = F_r + Y_1 F_a$ When $F_a / F_r \leqslant e$

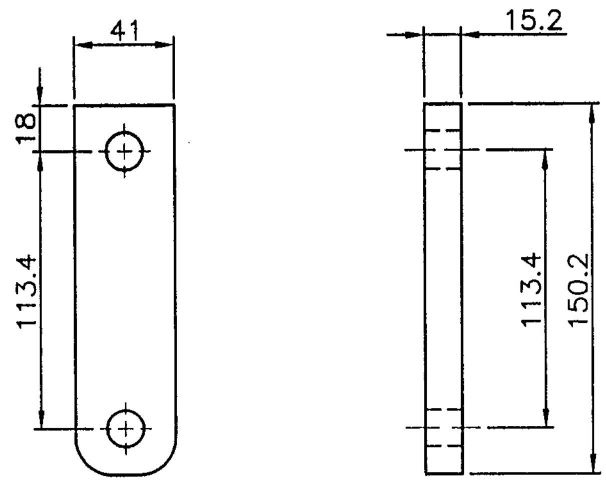
$P = 0.67\,F_r + Y_2 F_a$ When $F_a / F_1 \geqslant e$

Appropriate values of the calculation factors $e, Y_1$ and $Y_2$ will be found in the bearing tables for each individual bearing.

Equivalent static bearing load

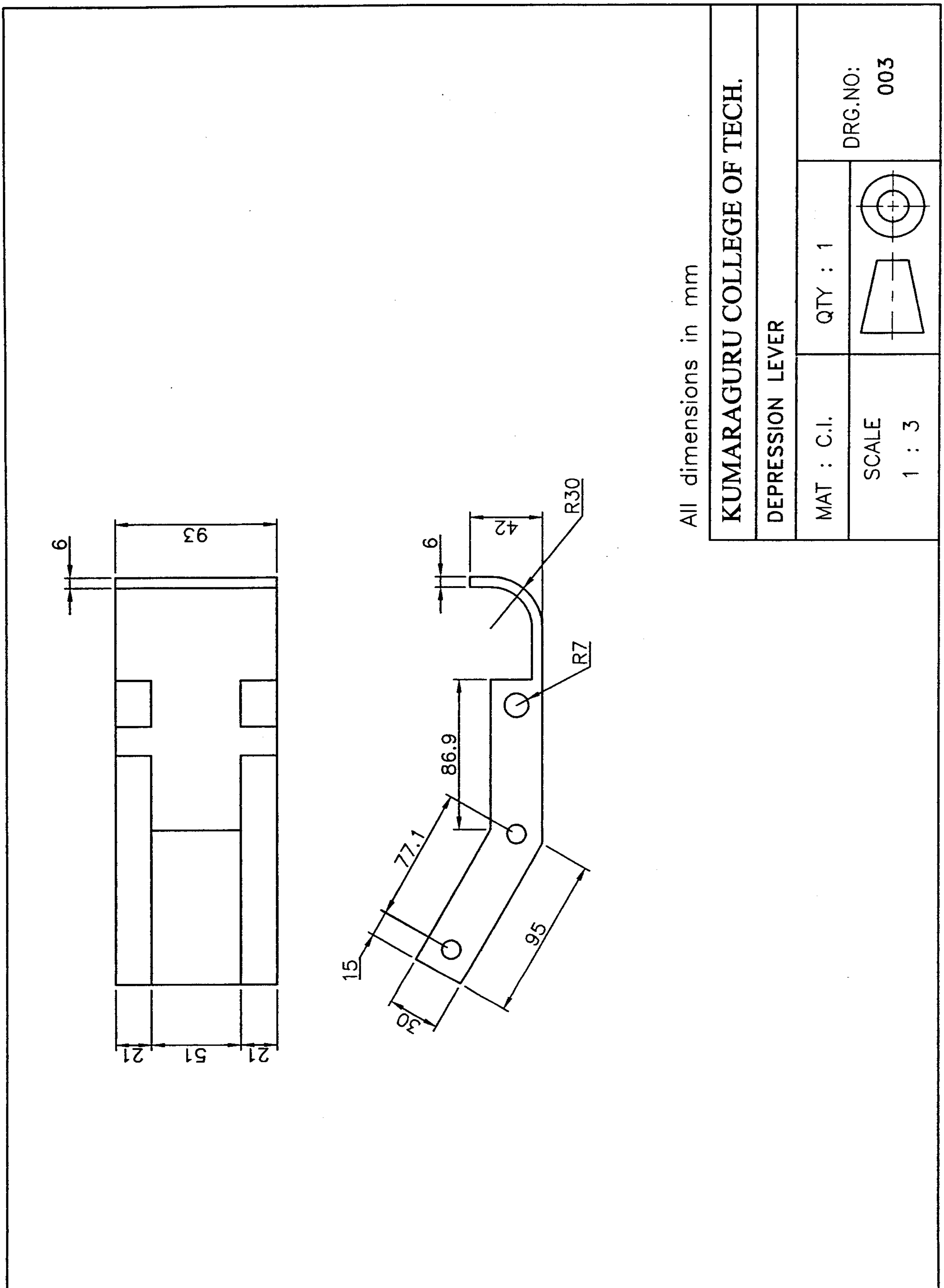
$P_o = F_r + Y_o F_a$


All dimensions in mm
KUMARAGURU COLLEGE OF TECH.
CROSS BAR
MAT : M.S.
QTY : 1
SCALE
1 : 3
DRG.NO:
001
51
170.1
R24
170.1
24

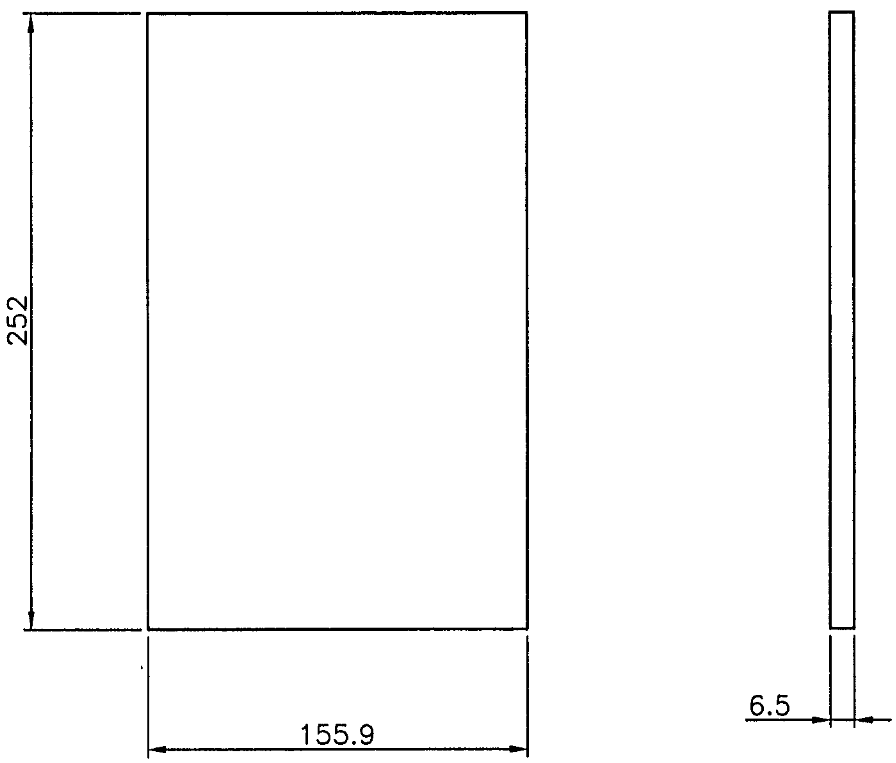



All dimensions in mm

| | | |
|---|---|---|
| KUMARAGURU COLLEGE OF TECH. | | |
| CONNECTOR | | |
| MAT : M.S. | QTY : 1 | DRG.NO: 002 |
| SCALE 1 : 3 | | |



All dimensions in mm
KUMARAGURU COLLEGE OF TECH.
DEPRESSION LEVER
MAT : C.I.
QTY : 1
DRG.NO:
003
SCALE
1 : 3
93
6
21
51
21
42
6
R30
R7
86.9
77.1
15
30
95




All dimensions in mm

**KUMARAGURU COLLEGE OF TECH.**

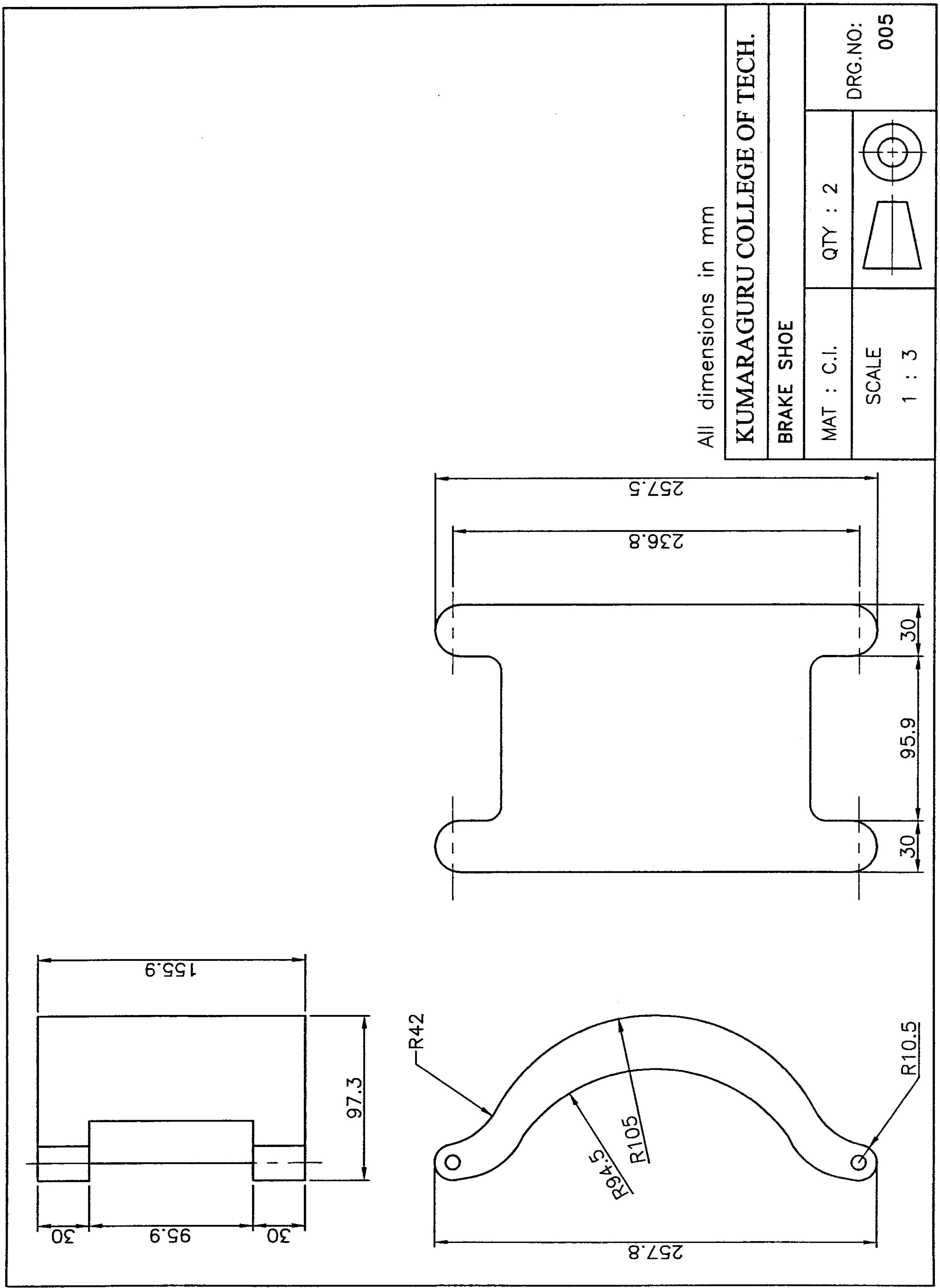
LINING

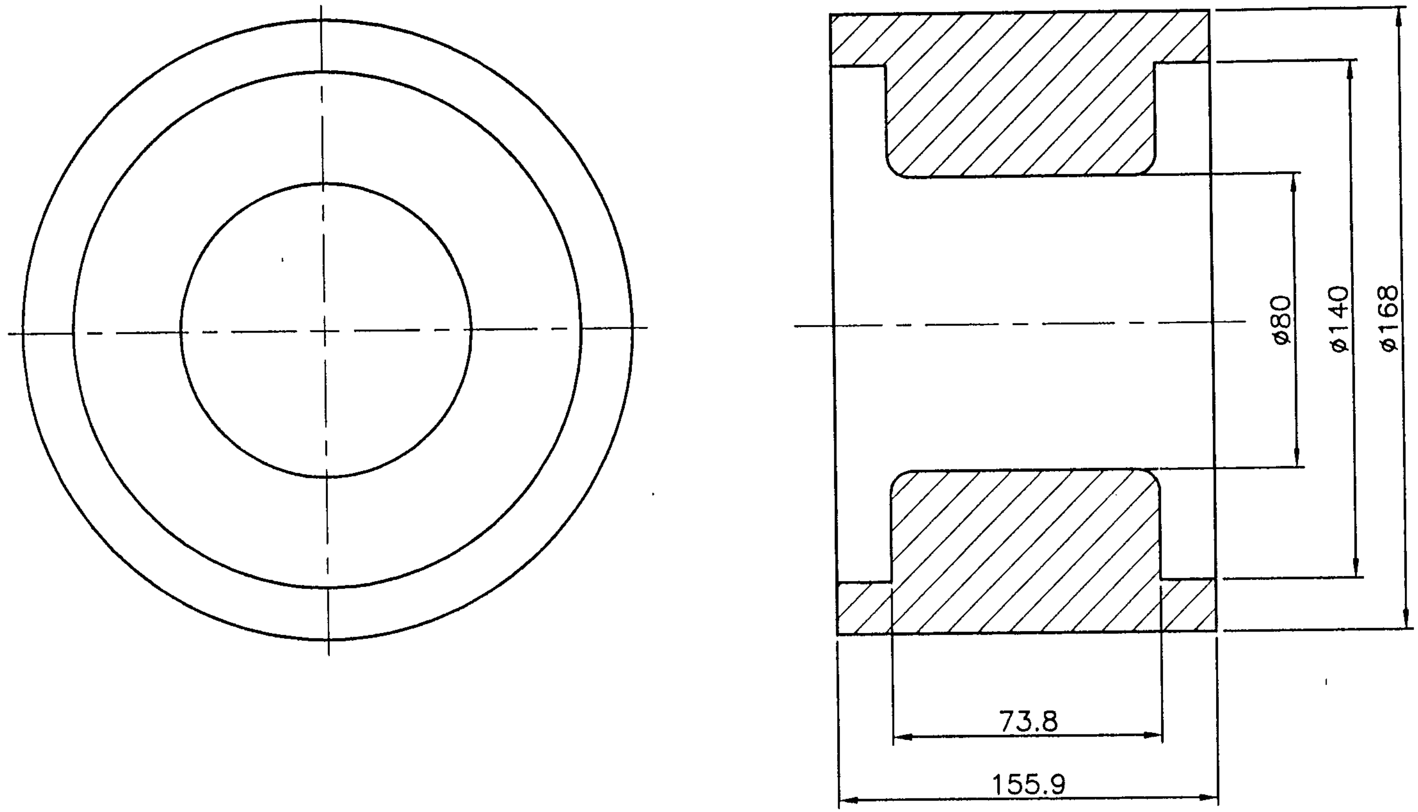
MAT : LEATHER

QTY : 2

SCALE 1 : 3

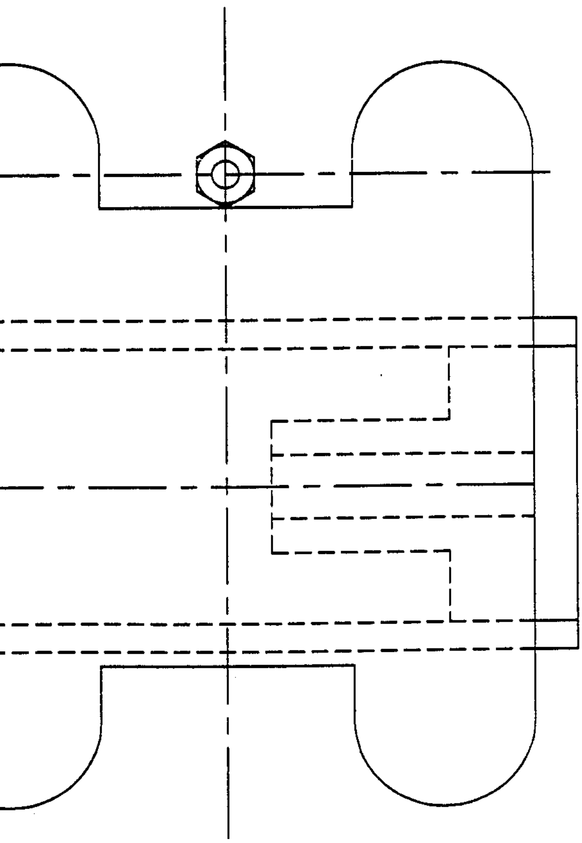
DRG.NO: 004


All dimensions in mm
KUMARAGURU COLLEGE OF TECH.
BRAKE SHOE
MAT : C.I.
QTY : 2
DRG.NO:
005
SCALE
1 : 3
257.5
236.8
30
95.9
30
155.9
97.3
30
95.9
30
R42
R105
R94.5
R10.5
257.8



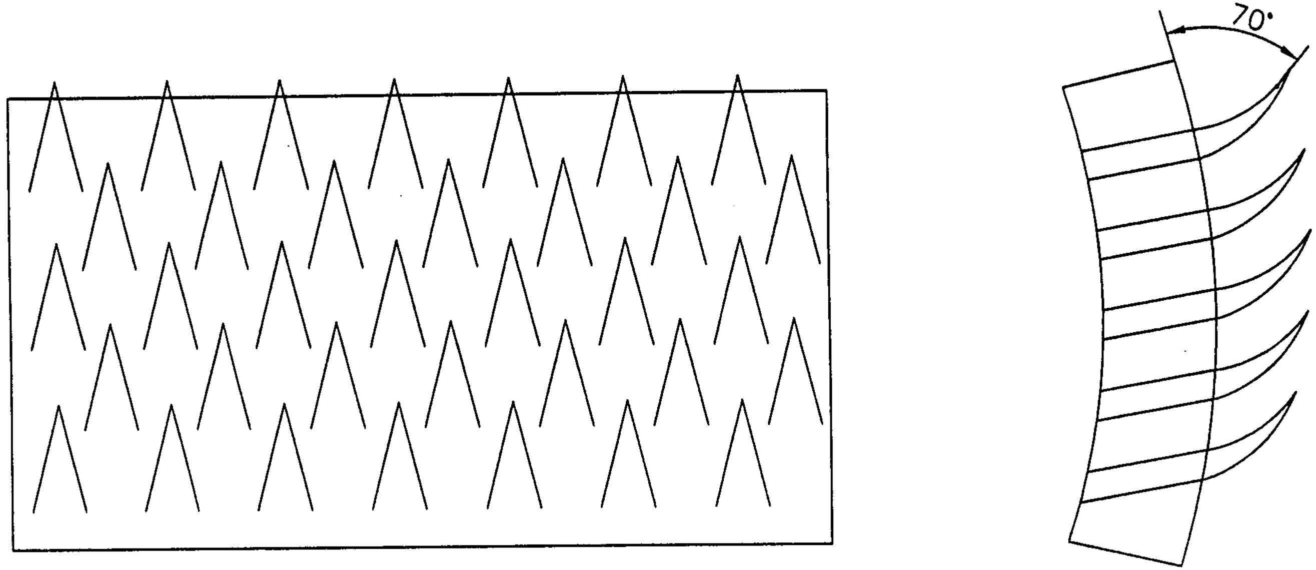

All dimensions in mm

| KUMARAGURU COLLEGE OF TECH. | | |
|---|---|---|
| WHEEL | | |
| MAT : HcHcr STEEL | QTY : 1 | DRG.NO: 006 |
| SCALE 1 : 2 | | |



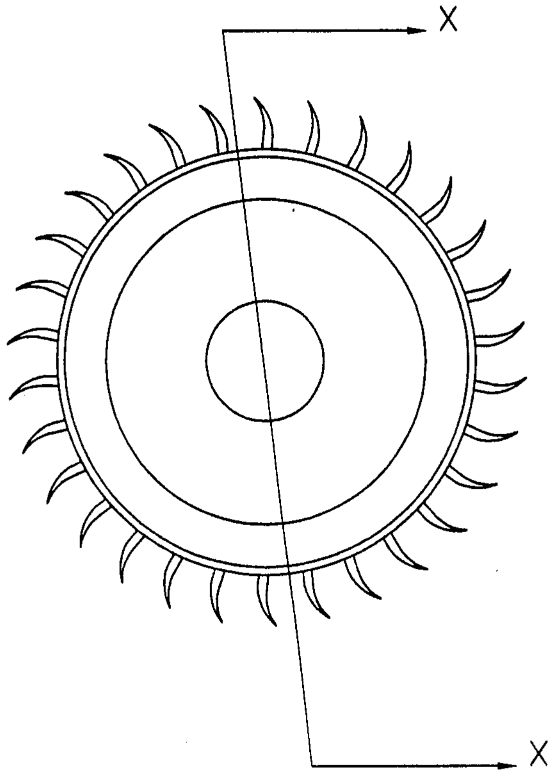
All dimensions in mm

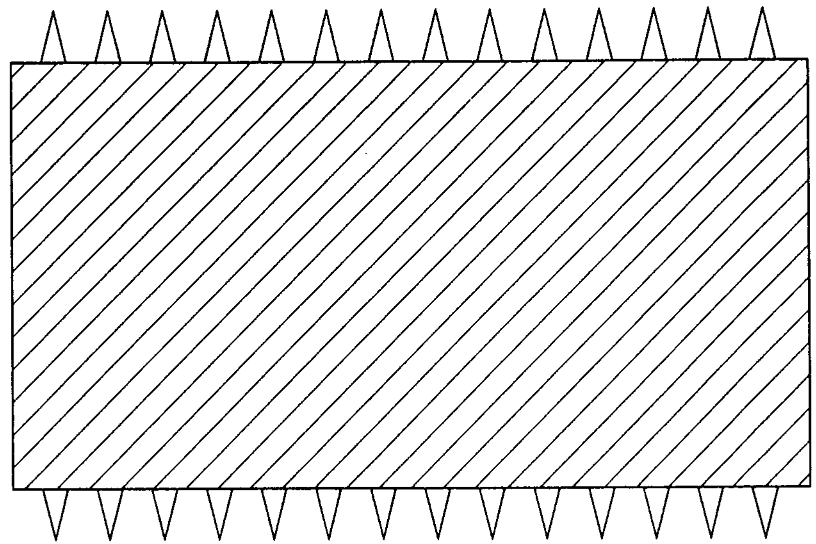
| | | |
|---|---|---|
| KUMARAGURU COLLEGE OF T | | |
| BRAKE ASSEMBLY | | |
| SCALE | QTY : 1 | DRG NO |




All dimensions in mm

| | | |
|---|---|---|
| KUMARAGURU COLLEGE OF TECH. | | |
| STAVES IN WODDEN PLATE | | |
| MAT : M.S. | QTY : 1 | DRG.NO: 007 |
| SCALE | | |

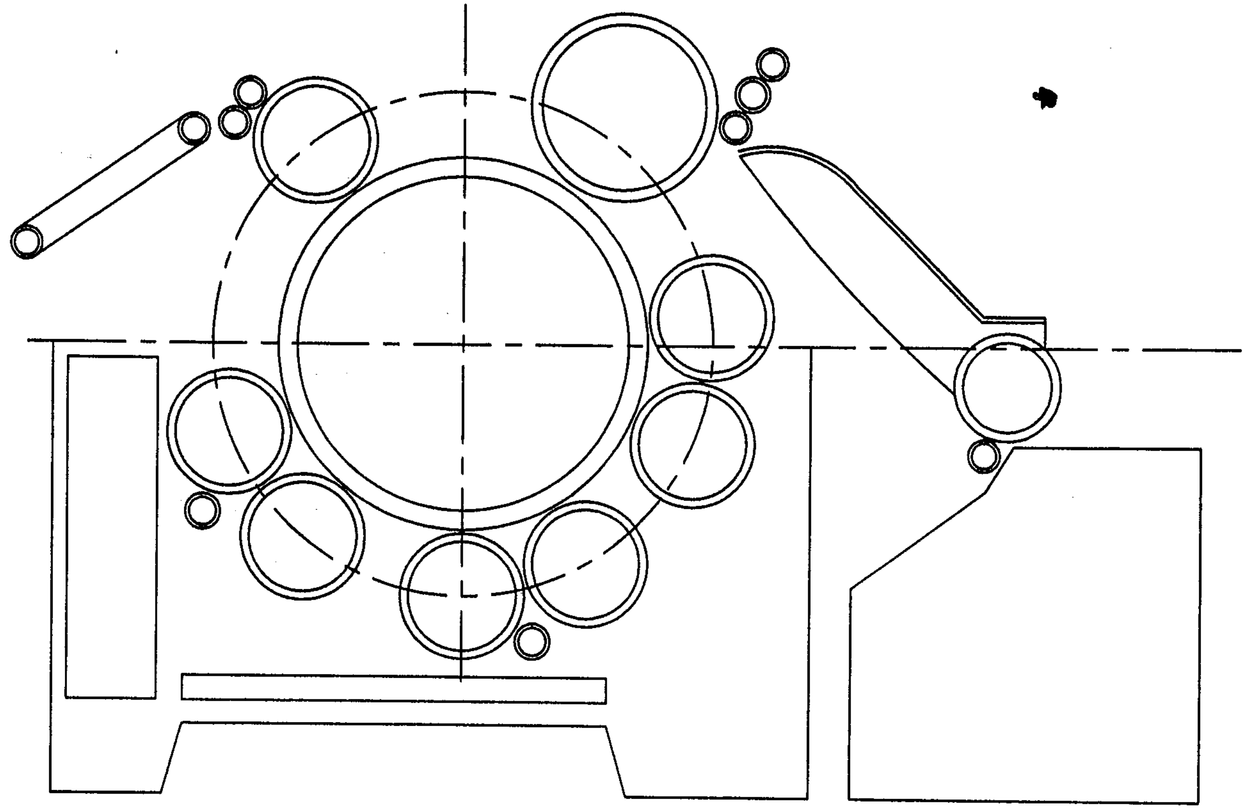


SECTION X–X

All dimensions in mm

| | | |
|---|---|---|
| KUMARAGURU COLLEGE OF TECH. | | |
| STAVES IN CYLINDER | | |
| MAT : M.S. | QTY : 1 | DRG.NO: 008 |
| SCALE 1 : 3 | | |



All dimensions in mm

| | | |
|---|---|---|
| KUMARAGURU COLLEGE OF TECH. | | |
| MACHINE ASSEMBLY | | |
| N.S | QTY : 1 | DRG.NO: 010 |

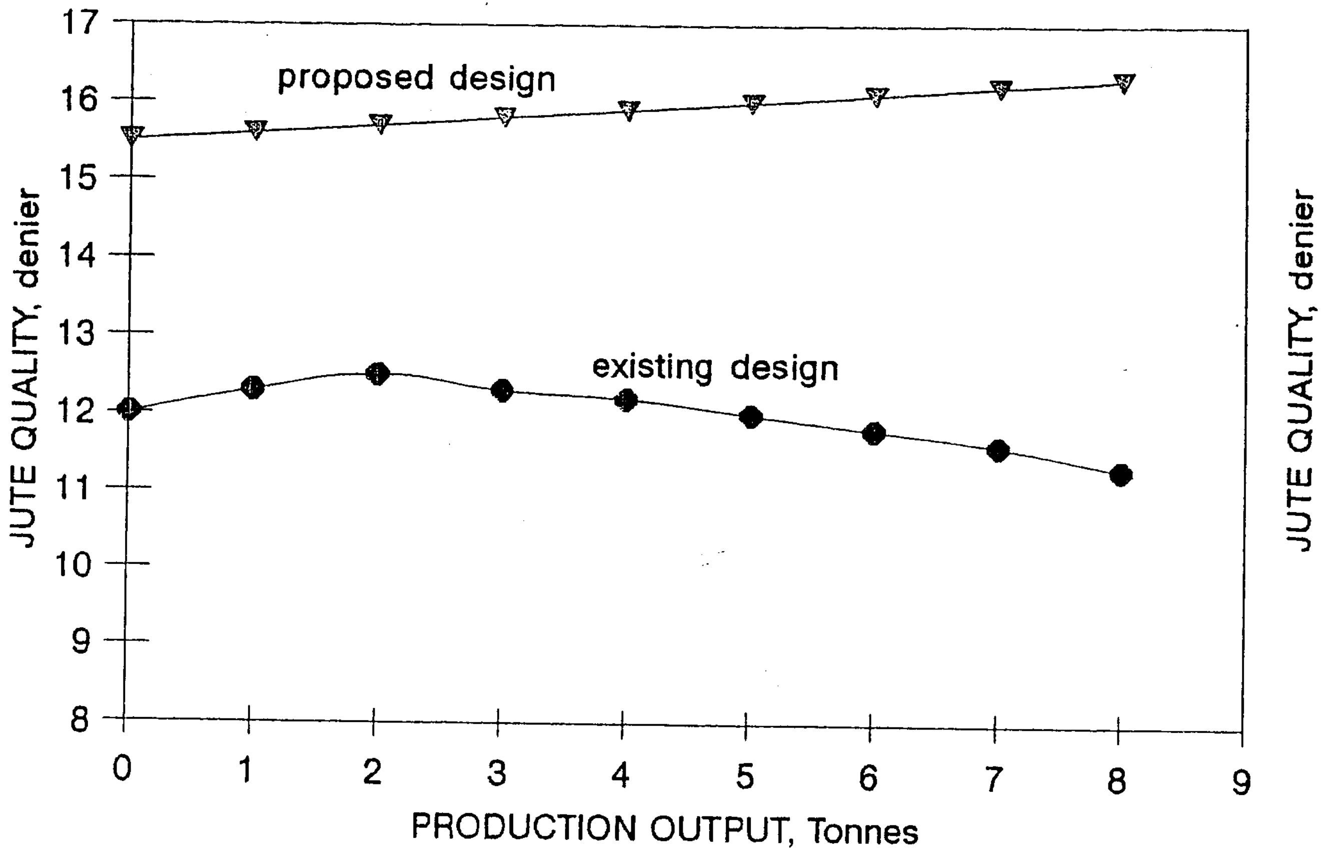



FIG.1 PRODUCTION VS QUALITY FOR THE EXISTING AND PROPOSED DESIGNS

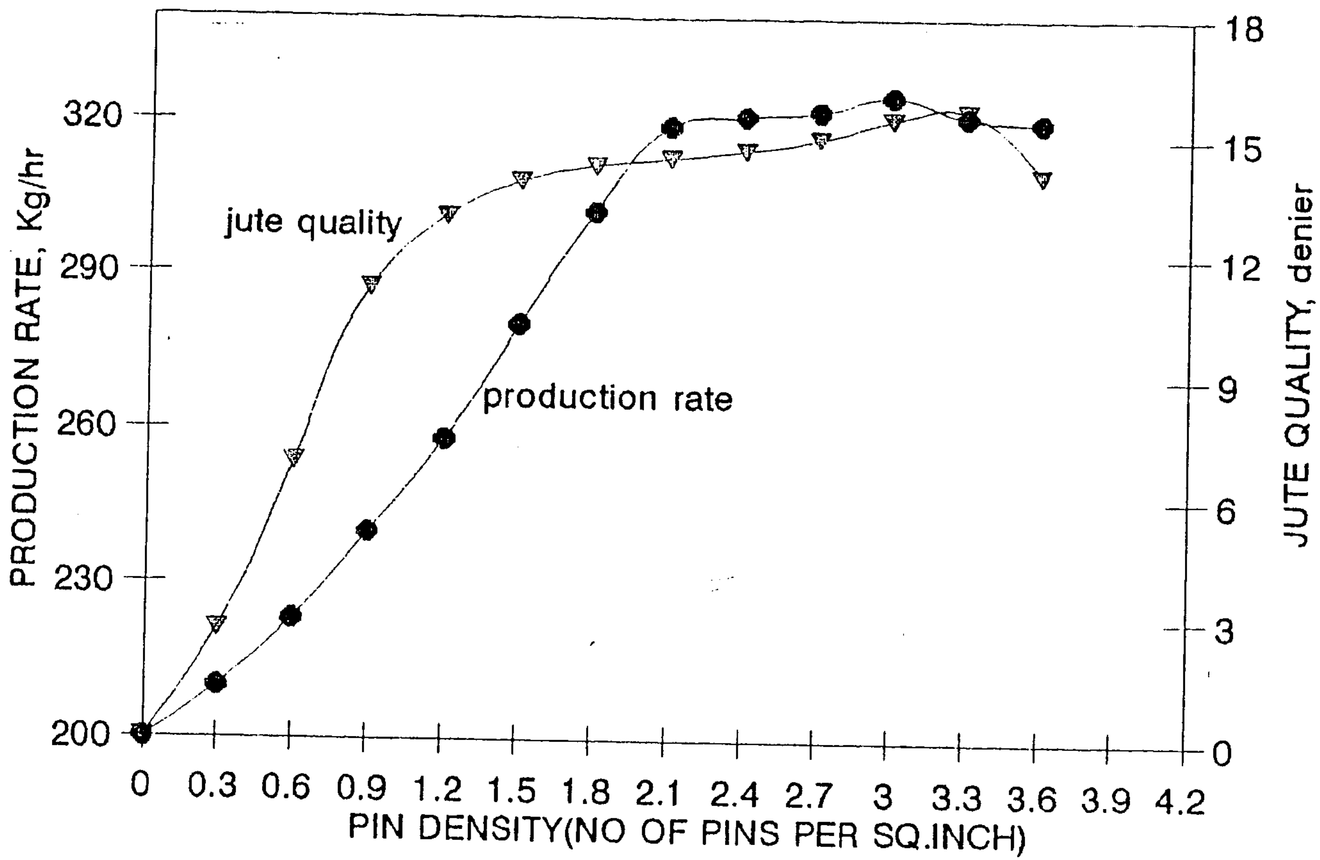



FIG.2 EFFECT OF PIN DENSITY ON JUTE QUALITY AND PRODUCTION RATE FOR MAIN CYLINDER

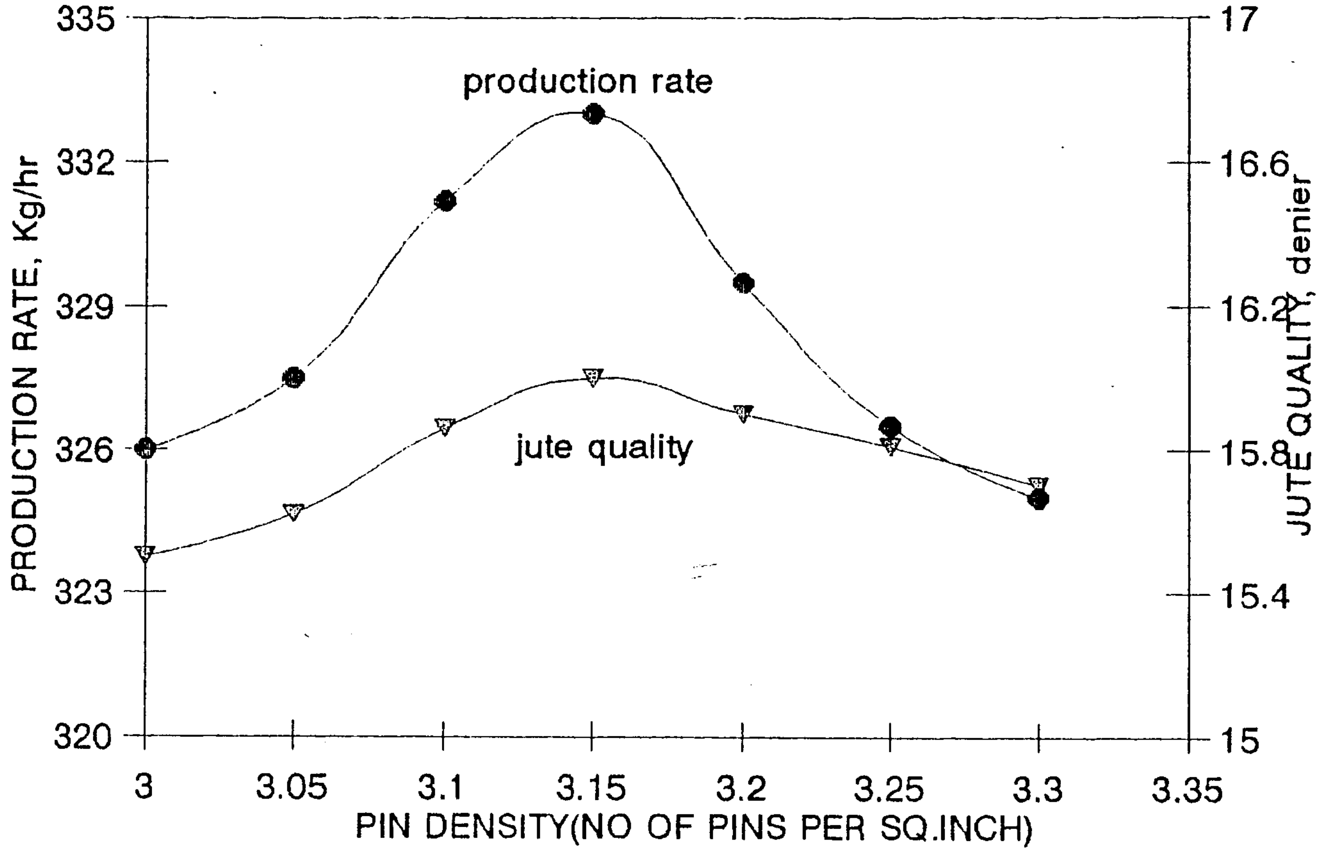



FIG.3 EFFECT OF PIN DENSITY ON JUTE QUALITY AND PRODUCTION RATE FOR MAIN CYLINDER

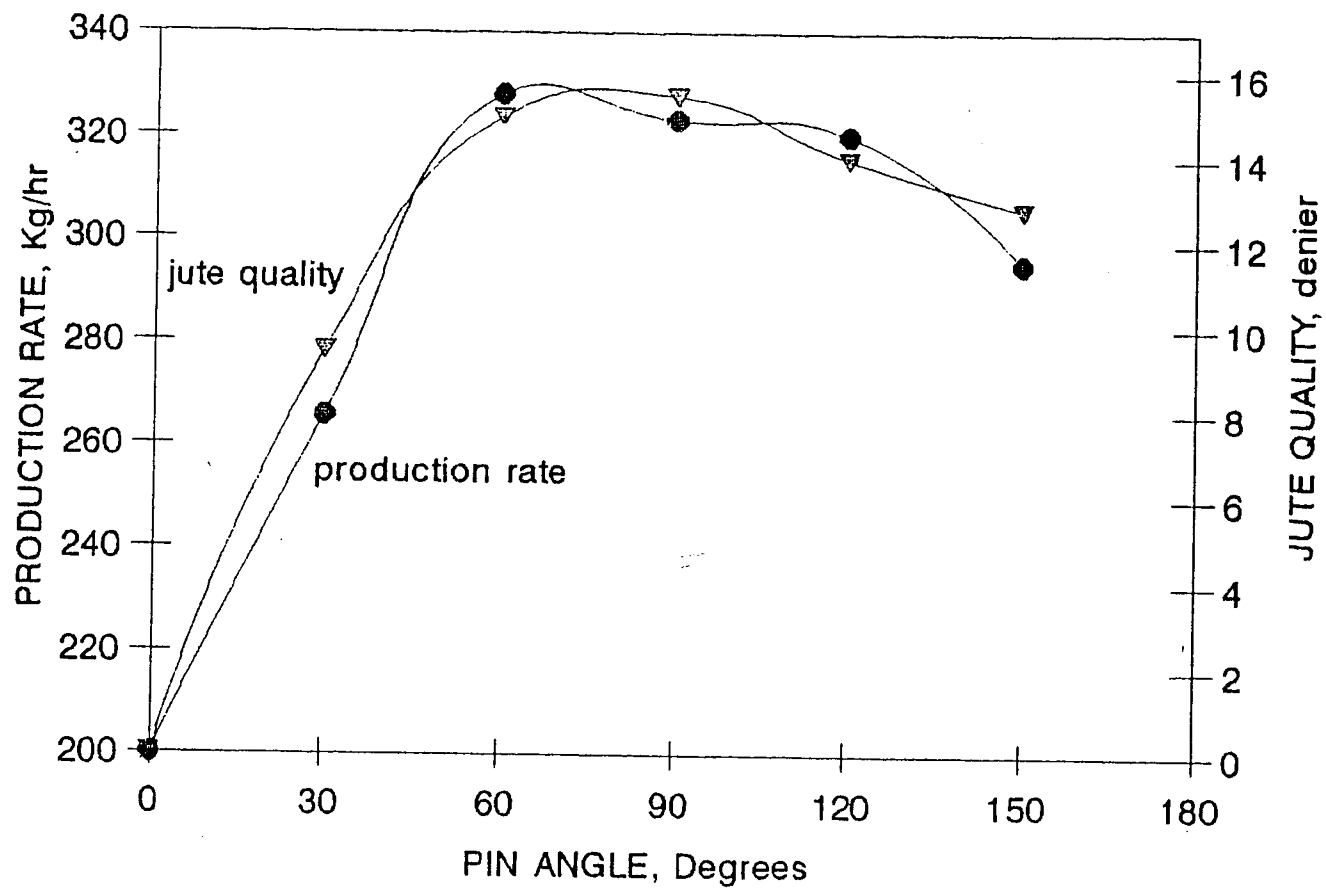



FIG.4 EFFECT OF PIN ANGLE ON JUTE QUALITY AND PRODUCTION RATE FOR MAIN CYLINDER

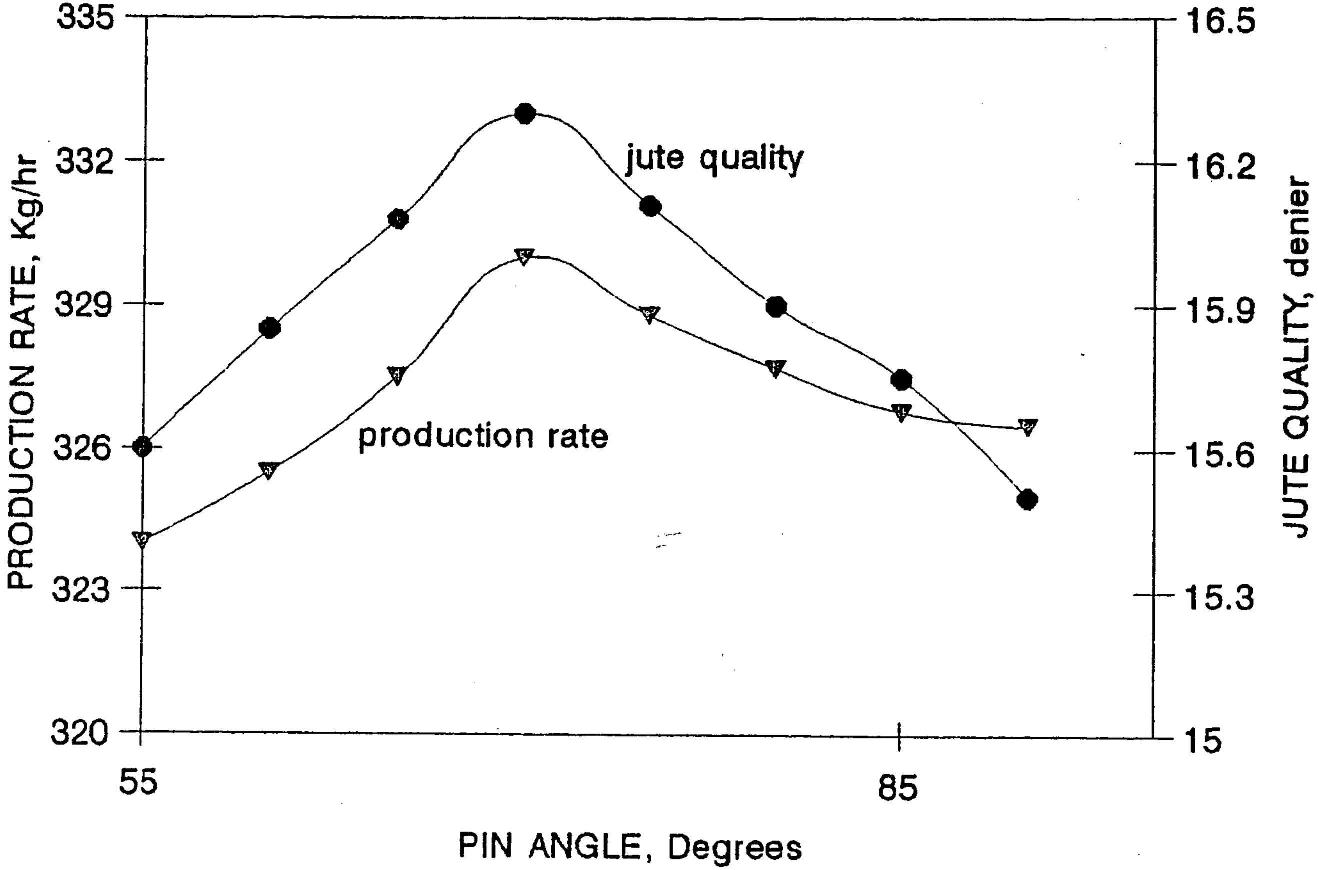



FIG.5 EFFECT OF PIN ANGLE ON JUTE QUALITY AND PRODUCTION RATE FOR MAIN CYLINDER

# 8. ANALYSIS OF RESULTS

**From fig. 1:-**

We find that for the same production, the quantity of jute in the proposed design increases by 28% to that of the existing design.

**From fig. 2:-**

We find that quality and production rat are maximum for the pin density of 3.0 to 3.3 pins/square inch.

**From fig. 3:-**

We find that for the pin density of 3.15 pins/sq. Inch

| | | |
|---|---|---|
| Quality | = | 16 deniers |
| Production rate | = | 333 kg/hour |

**From fig. 4:-**

We find that the quality and production rate are maximum when the pin angle lies between 60° to 90°.

**From fig. 5:-**

We find that for the pin angle of 70°, the quality and production rate are maximum and the values are

| | | |
|---|---|---|
| Production rate | = | 333 kg/hour |
| Quality | = | 16 deniers |

# CONCLUSION

Replacing the bush bearing by spherical roller bearing, the production rate increased from 52 tonnes/day to 80.5 tonnes/day.

Due to the introduction of dust collectors, the dust jute not mix with the good jute and hence the quality became as high as 16 deniers.

The dust collectors provide dust free atmosphere for the workers by collecting all the dust that when let out uncollected would pose severe health problems to the workers.

Changing the density of staves and the pin angle by trial and error method, the production rate and the quality increases.

Replacing the existing mechanical brake by an A.C. Solenoid brake, which an electro-mechanical brake stops the cylinder within 10 seconds, giving at most safety to the machine components in case emergency and at certain break downs.

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