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Study the Need of Designing a Special Purpose Machine to Improve the Production of Engine Block

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ABSTRACT : Special purpose machine tools are designed and manufactured for specific jobs and such never produced in bulk such machines are finding increasing use in industries the techniques for designing such machine would obviously be quite different from those used for mass produced machine. A very keen judgment is essential for success of such machines. A special purpose machine is needed to be designed and manufactured at a Company which found beneficial in increasing production quantity & reducing manpower.
The company sponsored the project to be undertaken and need of SPM machine is to be studied under this sponsorship.
Keywords: Special Purpose Machine, Engine Block.

I. INTRODUCTION

Special purpose machine tools are designed and manufactured for specific jobs and such never produced in bulk such machines are finding increasing use in industries the techniques for designing such machine would obviously be quite different from those used for mass produced machine.

Broadly the special purpose machine tools could be classified as those in which jobs remain fixed in one position and those in which job moves from one station to other (Transfer machine).

In this case the product may be either moving continuously (as in the case of spraying, polishing, sanding etc) or intermittently (the most usual case in machining operation). Rotary intermittently motion Transfer machine is very popular production machine.

It is essential that all movements be completely synchronized in order to obtain desired product it is essential that all tools and units must have completed their operation and come to rest before tools and units begin their work.

II. LITERATURE REVIEW

[1] S. Kalpakjian and S. R. Schmid, Manufacturing engineering and technology. Upper Saddle River, N.J: Pearson Prentice Hall, 2006 Due to the increasing demand for new products and greater competition as a result of globalization, manufacturing companies face unpredictable changes in the market. For this reason, manufacturing systems must be designed to meet the factors that enable the companies to remain competitive. These factors are high quality of products, low product cost, and flexible response to changes in the market and consumer needs.

[2] M. Tolouei-Rad and S. Zolfaghari, "Productivity improvement using specialpurpose modular machine tools," IJMR, vol. 4, pp. 219-235, 2009 These factors are very important for achieving greater productivity in manufacturing systems.

[3] U. H. Farhan, "An integrated computer-aided modular fixture design system for machining semi-circular parts," 2013 Traditional machinery was used to carry out manufacturing operations until the beginning of the 1950s. This included lathes, drill presses, milling machines, and other equipment for operations such as shaping, forming, and joining. However, using traditional machinery and equipment was relatively inflexible and a high level of skilled labor was required to operate and produce parts with the required specifications. These disadvantages led to high production costs. Therefore, production cost needed to be reduced by improving the flexibility and efficiency of manufacturing systems.

[4] M. P. Groover, Automation, Production Systems, and Computer-Integrated Manufacturing: Pearson Education, Limited, 2014 This led to meeting the requirements of the major factor in manufacturing, which is productivity. In order to improve the productivity of manufacturing systems, some important techniques have been implemented. One of these techniques is automation, which is a process to

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automate the operation of a machine by following a predetermined sequence of processes.

Automation has various levels, starting with simple hand tools and continuing on to computer numerical control machine tools (CNC) and, ultimately, the implementation of expert systems. Automation has been implemented in many areas, such as manufacturing processes, material handling and movement, inspection, assembly, and packaging.

[5] T.-C. Chang, H.-P. Wang, and R. A. Wysk, Computer-aided manufacturing. Upper Saddle River, N.J: Pearson Prentice Hall, 2006
The main advantages of automation are improving the productivity and quality of products, reducing human errors and work piece damage, arranging machines and other equipment efficiently, and integrating various aspects of manufacturing operations.

III. DESIGN OF PARTS

Broadly the special purpose machine tools could be classified as those in which jobs remain fixed in one position and those in which job moves from one station to other (Transfer machine). In first case the machine may perform either only one operation or more .In the second case, the product may be either moving continuously (as in the case of spraying, polishing, sanding etc) or intermittently (the most usual case in machining operation). Rotary intermittently motion Transfer machine is very popular production machine and is described in brief bellow.

1. DESIGN OF SPINDLE UNIT:

The operational capability of a machine tool in term of productivity as accuracy and finish of machining parts largely depends upon the extent to which these function is qualitatively satisfied. They also determine the important design requirement to spindle unit.

The following recommendations for selecting spindle material may be formulated:

- For normal accuracy spindles, plain carbon steels C45 and C49 (AiSi C1050 and C1050) hardened and tempered to R_C =30.
- For above normal accuracy spindles –low alloy steel 40Cr 1 Mn 60 Si 27 Ni25(AiSi5140) induction hardened to Rc 50-56;if induction hardening of above accuracy spindles difficult due to complex profile then low alloy steel 50 Cr 1 Mn 60 Si 27 Ni 25 (Aisi 5147) is used with hardening to Rc =55-60.
- For spindle of precision machine tools, particularly those with sliding bearing –Low alloyed steel 20 Cr Mn 60 Si 27 Ni25 (Aisi 5120) case hardened to Rc =63-68.
- For hollow, heavy –duty spindles –grey cast iron or spheroidal graphite iron.

MATERIALS AND HEAT TREATMENT

Carburising steel for machine tools 15 Ni2Cr1Mo15

Steel designation		15Ni2Cr1Mo15							
Chemical composition %		C	Si	Mn	Ni	Cr	Mo	S	P
		0.12-0.18	0.1-0.35	0.6-1	1.5-2	0.75-1.25	0.1-0.2	0.05 Max.	0.035 Max.
Form of material		Bars, billets and forgings							
Supply condition		Rolled or forged							
Condition of material		As supplied, annealed	Refined and quenched						
Limiting ruling section mm		—	90	60	30				
Tensile strength kgf/mm² Min.		67	95	100	110				
Yield strength kgf/mm² Min.		57	80	90	95				
Elongation in % Min.	Gauge length $5.65\sqrt{S_0}$	17	—	—	9				
	Gauge length $4\sqrt{S_0}$	—	—	—	12				
Izod impact strength kgf.m Min.		—	—	—	3.5				
Hardness	Brinell HB	193*	230	230	300				
	Vickers HV		Ref. hardenability curve						
	Rockwell HRC		59-63 case						
Fatigue limit kgf/mm² 10^7 reversals		25-31	40-47	40-50	44-55				
Machinability and Machinability rating		Fair	80						
Application		Heavy duty components, gears, etc.							
Remarks		*Hardness value should not exceed 217							
Equivalent steels		BS	CSN	DIN	AISI	JIS	GOST	AFNOR	
		En 354	16220	13CrNi6	4317	SNCM22	12CbN2	18NCD6	

Table no.4.1 Materials & Heat Treatment

Calculations of forces on spindle shaft:

Inputs:

Material of spindle shaft case hardened steel

Chemical composition: 15 Ni 2 Cr 1 Mo15

Shear strength (fs) = 5.3kgf /mm2
= 51.67 N /mm2

D = diameter of gear on spindle shaft =90

T= Number of teeth on gear = 30

Module m = 3

Torque $T = p \times 60 / 2\pi\ n$

$= 0.826 \times 103 \times 60 / 2 \times \pi \times 424.41$

$= 18.58 \times 103$ N-mm

$= 1.89$ kgf-m

Tangential component of force acting on gear (Pt)

$Pt \times PCD = 2 \times$ Torque

$Pt = 2 \times$ Torque /PCD

$= 2 \times 18.58 \times 103 / 90$

$= 232.25$ N

$= 23.682$ kgf

Radial component of force acting on gear (Pr)

$Pr = Pt \times \tan 20$

$= 232.25 \times \tan 20$

$= 84.53$ N

=8.6198kgf

2. Forces on spindle shaft :

a) Inputs:

- Shear strength (fs) = 5.3kgf /mm2

  = 51.67 N /mm2

  D = diameter of gear on spindle shaft =90 ; T= Number of teeth on gear = 30

  Module (m) = 3; Torque (T) $= p \times 60 / 2\pi\ n = 1.89$ kgf-m;

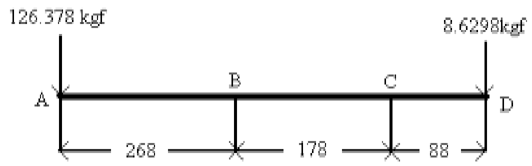
- Tangential component of force acting on gear (Pt)

$Pt \times PCD = 2 \times$ Torque =23.682 kgf

- Radial component of force acting on gear (Pr)

$Pr = Pt \times \tan 20 = 8.6198$kgf

3. Vertical loadings :



Free body diagram for vertical loading diagram of spindle shaft

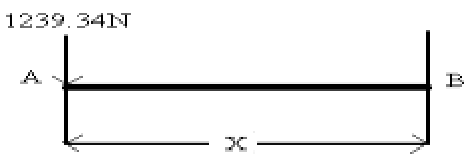
RB+RC =134.99kgf;

RB=312.39kgf;

RC=177.39kgf.

Bending Moment Calculations :

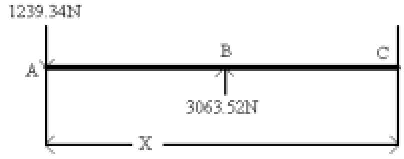
a) SECTION AB 0<X<268 :



Bending moment at (X=0) =0; B (X=268) = 33869.16 kgf-mm

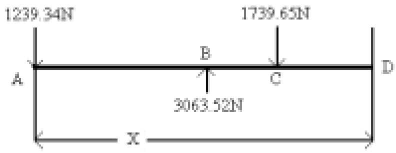
b) SECTION BC 268<X<446 :

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252



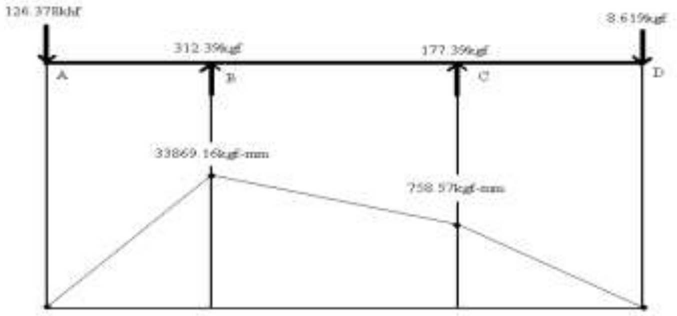
Bending moment at B(X=268) =33869.16 kgf-mm; Bending moment at C (X=446) = 758.57 kgf-mm

c) SECTION CD 446<X<534 :

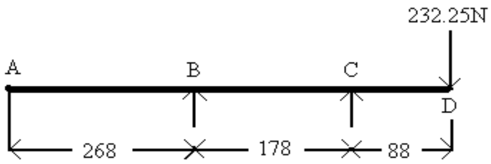


Bending moment at C (X=446) =758.57 kgf-mm; Bending moment at D (X=534) =0

Vertical Bending Moment Diagram:



4. Horizontal Loading :

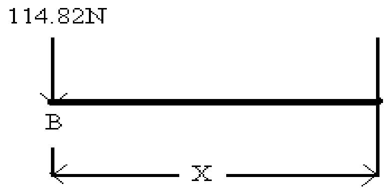


Free body diagram for horizontal loading diameter of spindle shaft

RB+RC=23.68 kgf ;
∑MB =0
RC=35.39 kgf; RB= -11.70 kgf
Negative sign indicates that direction of RC is opposite and should be changed from ↑ to ↓

Bending Moment Calculations :

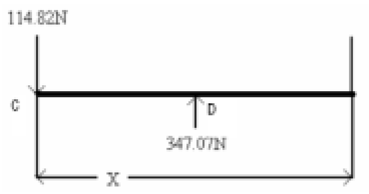
a) SECTION BC 0<X<178 :



BENDING MOMENT AT B (X=0) = 0;
BENDING MOMENT AT C(X=178) = 2084.09 kgf-mm

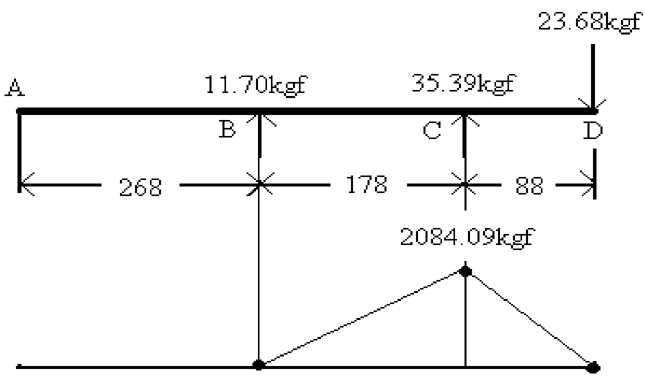
International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

b) SECTION CD 178<X<266 :



BENDING MOMENT AT C (X=178) = 2084.09 kgf-mm;
BENDING MOMENT AT D (X=266) =0;
BENDING MOMENT AT A=0
BENDING MOMENT AT B = $\sqrt{(R_{BV}^2+R_{BH}^2)}$ =33869.16Kgf-mm.
BENDING MOMENT AT C = $\sqrt{(R_{CV}^2+R_{CH}^2)}$ =273213.269 Kgf-mm
BENDING MOMENT AT D = 0;
BENDING MOMENT IS MAX. AT B = 33869.16 kgf-mm

Horizontal Bending Moment Diagram :



5. Calculations of min. shaft diameter :

$\pi/16 \times fs \times d^3 = \sqrt{(Mt)^2 + (T)^2}$

$(D)_{min} = 32.59$ mm

Multiply $(d)_{min}$ by factor of safety 2

d = 65.18 mm

6. Deflection of Spindle Shaft :

Shafts when subjected to bending loads are subjected to bending stresses. If induced bending stress exceeds the permissible stress values either for compression or tensile, then the shafts undergoes the failure.

The deflection may be calculated by any of the following methods;

- Macaulay's Method
- Area-moment method
- Conjugate beam method
- Castiglione's method
- Graphical integration method

Macaulay's method:

The bending moment expression obtained shall be equated to

$E \times I\, d^2y/dx^2$

Subsequently the entire expression shall be integrated once to obtain the expression for dy/dx i.e. the slope of beam at any cross section location .it shall be integrated once again to obtain the expression for y, the deflection.

Hence $E \times I\, dy/dx = \int M \times dx$ ----(1)

$E \times I \times Y = \iint M \times dx$ --- (2)

While integrating, the constants of integration shall be coupled with the expression, and shall serve as the constants for entire expression for the valid zones. The constants of the integration C_1 and C_2 shall be solved by applying suitable end conditions as y=0 at X=1 etc. to the basic expression for y. The actual values of y or dy/dx shall be obtained by substituting given value of E and I_{XX}.

International Journal of Advances in Engineering and Management (IJAEM)
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7. SELECTION OF BEARINGS :

The selection of the type of bearing in a particular application depends upon the requirement of the situation and the characteristics of different types of bearing. The guidelines for selecting a proper type of bearing are as follows.

1) For low and medium radial loads, ball bearings are used, where as for heavy loads and large shaft diameters, roller bearings are selected.
2) Self-aligning ball bearing and spherical roller bearing are used in applications where a misalignment between the axes of shafts and housing is likely to exist.
3) Thrust ball bearings are used for medium Thrust loads where as for heavy Thrust loads, cylindrical roller Thrust bearings are recommended. Double acting Thrust bearings can carry the Thrust load in either direction.
4) Deep groove ball bearing, angular contact bearing and spherical roller bearing are suitable in application where the load acting on the bearing consists of two components- radial and Thrust.
5) The maximum permissible speed of the shaft depends upon the temperature rise in the bearing. For high speed applications, deep grove ball bearings, angular contact bearing, and cylindrical roller bearings are recommended.
6) Rigidity controls the selection of bearings in certain application like machine tool spindles. Double row cylindrical roller bearing or taper roller bearings are used under these conditions. The line of contact of these bearings, as compared with the point of contact in ball bearings, improves the rigidity of the system.
7) Noise becomes the criterion of selection in application like household applications. Under these circumstances, deep groove ball bearings are recommended. Knowledge of the design characteristics of different types of bearing and proper appreciation of the needs of application enables the designer to select a proper type of bearing.

All dimensions in *mm*

IS designation	d	D	T	B	r	r_1	Basic load rating Dynamic C kgf	Basic load rating Static C_o kgf	d_b Min.	d_a Max.	D_a Min.	D_b Min.	r_a Max.	Limiting speed rpm Lubrication Grease	Limiting speed rpm Lubrication Oil	SKF designation
30 KB 22	30	62	21.25	20	1.5	0.5	3200	2750	36	37	52	58	1	6300	8500	32206
35 KB 22	35	72	24.25	23	2	0.8	4300	3650	42	43	61	67	1	5300	7000	32207
40 KB 22	40	80	24.75	23	2	0.8	4750	4050	47	48	68	75	1	4800	6300	32208
45 KB 22	45	85	24.75	23	2	0.8	5200	4650	52	53	73	80	1	4500	6000	32209
50 KB 22	50	90	24.75	23	2	0.8	5300	4800	57	58	78	85	1	4300	5600	32210
55 KB 22	55	100	26.75	25	2.5	0.8	6700	6300	63	64	87	95	1.5	3800	5000	32211
60 KB 22	60	110	29.75	28	2.5	0.8	8000	7650	69	69	95	104	1.5	3400	4500	32212
65 KB 22	65	120	32.75	31	2.5	0.8	9800	9300	74	76	104	115	1.5	3000	4000	32213
70 KB 22	70	125	33.25	31	2.5	0.8	9800	9300	79	80	108	119	1.5	2800	3800	32214
75 KB 22	75	130	33.25	31	2.5	0.8	10400	10200	84	85	114	125	1.5	2600	3600	32215
80 KB 22	80	140	35.25	33	3	1	12000	11600	90	90	122	134	2	2400	3400	32216

Table 5.1 Basic Load Ratings of bearing

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

BEARING MOUNTING ARRANGEMENTS

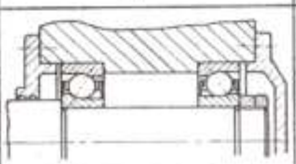
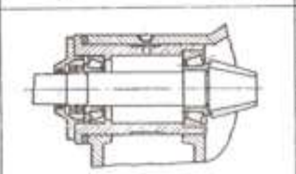
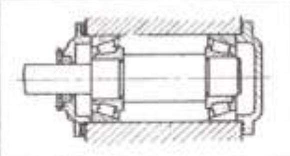
Application	Figure	Loading conditions	Drive conditions	Axial location	Relative axial expansion	Seating fits	Remarks
Two single row angular contact ball bearings (back to back mounting) Matched pair		Radial load in both positions, axial load in one direction at each position	Housing rotating, shaft stationary, direction of radial load constant on inner ring	By one bearing in each direction	Adjusted by shaft nuts	Inner ring sliding outer rings interference	
Two tapered roller bearings (back to back)		Radial and axial load in one direction at each position	Shaft rotating, housing stationary, constant direction of riadial load on outer rings	By one bearing in each direction	Adjusted by shaft nuts	Inner rings interference, outer rings sliding	
Two tapered roller bearings (face to face mounting)		Radial and axial load in one direction at each position	Shaft rotating, housing stationary, constant direction of radial load on outer ring	By one bearing in each direction	Adjusted by shims between flange and housing faces	Inner ring interference outer ring sliding	

Table no.5.2 Bearing Mounting arrangement

Select the parameter for taper roller bearing and cylinder roller bearing are as follows:
For Internal diameter (d) = 65 mm
Dynamic load (C) = 9800 kgf
Static load (C_o) = 9300 kgf
Outer diameter (D) = 120 mm
Thickness (t) = 32.75 mm
Hence ,
I select taper roller bearing with SKF designation of 32213.

5.1 Calculation of Bearing Life**:-**

$L = 60 \times n \times L_h / 10^6$

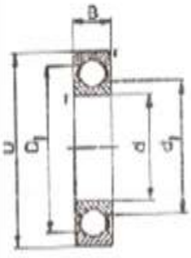
Where
L =Bearing life in million of revolution.
N = speed in rpm
L_h =Life in hours
= 60000 hours (Working in three shifts, 24 hours)

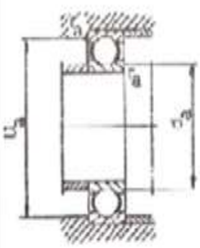
$L = 60 \times n \times L_h / 10^6$
$= 60 \times 420 \times 60000 / 10^6$
L =1512million revolutions.

Calculation of Radial Load on Bearing:-

For d =65mm
$L = (C/P)^{10/3}$
$C/P = L^{0.3}$
$P = C/L^{0.3}$
$P = 9800/1512^{0.3}$
(Dynamic load, C=9800kgf for d= 65 mm)
P =1089.83 kgf

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252





DEEP GROOVE BALL BEARINGS

All dimensions in *mm*

IS designation	Boundary dimensions				Abutment dimensions			Basic load rating kgf		Limiting speed rpm		Equivalent DIN & CSN designation
										Lubrication		
	d	D	B	r Nom.	d_a Min.	D_a Max.	r_a Max.	Dynamic C	Static C_0	Grease	Oil	
10 BC 02	10	30	9	1	14	26	0.6	475	270	24000	30000	6200
12 BC 02	12	32	10	1	16	28	0.6	540	315	22000	24000	6201
15 BC 02	15	35	11	1	19	31	0.6	600	364	19000	24000	6202
17 BC 02	17	40	12	1	21	36	0.6	752	453	17000	20000	6203
20 BC 02	20	47	14	1.5	25	42	1	997	621	15000	18000	6204
25 BC 02	25	52	15	1.5	30	47	1	1088.6	707	12000	15000	6205
30 BC 02	30	62	16	1.5	35	57	1	1519.58	1016	10000	13000	6206
35 BC 02	35	72	17	2	41.5	65.5	1	1995.8	1406	9000	11000	6207
40 BC 02	40	80	18	2	46.5	73.5	1	2404	1655	8500	10000	6208
45 BC 02	45	85	19	2	51.5	78.5	1	2585	1882	7500	9000	6209
50 BC 02	50	90	20	2	56.5	83.5	1	2766	1995	7000	8500	6210
55 BC 02	55	100	21	2.5	63	92	1.5	3401	2540	6300	7500	6211
60 BC 02	60	110	22	2.5	68	102	1.5	3764	2857	6000	7000	6212
65 BC 02	65	120	23	2.5	73	112	1.5	4377	3469	5300	6300	6213
70 BC 02	70	125	24	2.5	78	117	1.5	4808	3855	5000	6000	6214
75 BC 02	75	130	25	2.5	83	122	1.5	5171	4150	4800	5600	6215
80 BC 02	80	140	26	3	89	131	2	5669	4535	4500	5300	6216
85 BC 02	85	150	28	3	94	141	2	6363	5352	4300	5000	6217
90 BC 02	90	160	30	3	99	151	2	7284	5914	3800	4500	6218

Table 5.3 Deep Groove ball bearing

5.2 Arrangements of Bearing Mounting

The following basic principles should be satisfied for satisfactory performance of bearing mounting arrangements.

1. The shaft or rotating component must be axially located in both directions.
2. The bearing arrangements must be able to accommodate relative axial; expansion between the shaft and housing.
3. Proper fits on shaft and housing demanded by the mounting requirements, conditions of load and rotation must be provided.
4. Proper lubricating systems to provide lubricant circulation or retention should be used.
5. Actual load conditions and load carrying capacity of bearing should be compatible.

Preloading of Bearing

The total deflection of spindle nose depends upon its own stiffness spindles-steels as well as the stiffness (compliance) of the spindle supports. The discussion which follows reveals the positive role played by preloading of rolling element in reducing total deflection

The variations of the spindle deformed and due to radial force P is depicted in if the bearing is assembled with the clearance, a reversal of the direction of the applied force results in a abrupt change of deformation. This is highly undesirable from the point of view machining accuracy. Bearing assembled with interference are free of this shortcoming and are distinguished by a smooth SP curve.

The methods of applying in radial and angular contact ball bearing that are generally mounted in pairs. A constant preloading is achieved either by grinding of the faces of the inner races by inserting spacing rings of different widths between the inner and outer races. If the bearing rotate at high rpm, in the initial preload as a tendency to weaken. In such cases especially when the bearings are small, the preloading can be applied by means of springs which ensures the constant preload that can be accurately adjusted precision bearings.

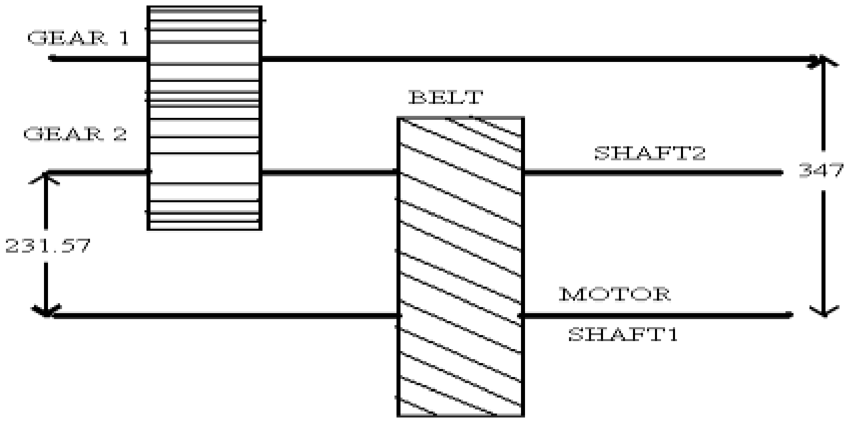
Taper roller bearings are preload by the methods shown in housing inner and outer races are axially displaced with the help of nuts. Races get skewed in the gamut bearing arrangement the outer race is axially displaced by means of springs whereas in the Timken bearings this is achieved by supplying oil or air under controlled pressure.

8. DESIGN OF GEAR BOX

Gear box provide for a wide range of cutting speeds and torques from a constant speed power input enabling proper cutting speed or torques to be obtained at the spindle as required in the case of cutting drives and the desired feed rates

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

in the case of fed drives. Gear boxes are also used to affect inter-related motions between the work piece and the tool as in the case of screw cutting and gear cutting.



8.1 Calculations for the design of gears

When pinion and gear are made of same material then pinion is weaker. So design calculations are to e done for pinion.

Material: C-45 Steel

Ultimate tensile strength (fu) = 174 kgf/m
=1706.36 N/mm2

Allowable static stress (fo)= fu /3
=174/3
=58 kgf /mm2

BHN = 495

P=5.5 kW

Flexural Endurance limit (fe) =BHN ×1.75
=495×1.75
=866.25 N/mm2
=88.33 kgf /mm2

Motor speed (rpm) =1440

Speed at the spindle (rpm) =420

Thus I have to reduce the speed from 1440 rpm to 420 rpm

No. of stages to reduce the speed = 2

1440/420 =3.4

Reduction in the first gear = 3.4/2
= 1.7

Reduction in the second gear = 3.4/1.7 = 2

Speed of the shaft for first reduction (pulley)
=1440/1.7
= 847.058

Speed of shaft for second reduction (gear) = 847.058/2
= 423.52≈420

The profile dimensions for gear teeth are based on Module. From table no.54 of Machine tool design hand book by CMTI. I select standard module of gear as 3.

The above figure shows profile of the gear of motor shaft on which driver gear is to be mounted. So the PCD of this gear can be calculated as follows.

Bore diameter of gear =Φ35 mm

Deddendum is =1.25 ×m
=1.25× 3
=3.75 ≈4 mm

And normally for case of manufacturing of gears the distance between root and Deddendum is taken to be 10 mm

Therefore, Pitch Circle Diameter of the Driver gear can be calculated as

35/2 + 10 + 4 = 31.5 mm

PCD = 31.5×2

=63 mm

No of teeth can be calculated as,

Module = D/T

T = D/M

= 63/3

=21≈25≈30

For the ease of manufacturing I take number of teeth as 25

By considering the number of teeth T= 25 the design of gear for static and Wear load in relation to Dynamic load. The static and Wear load must be greater than dynamic in relation to dynamic load. Even by taking the number of teeth T=26 up to T=29 it does not satisfy the condition for safety. But by taking of teeth T=30 the design becomes safe.

So the new PCD for the driver gear can be calculated as,

PCD = m×T

= 3×30

= 90 mm.

The compound gear train for gear box is as shown I the figure. The number of gears on every shaft as follows,

No. of gears on shaft first (Φ35 mm) =1

No. of gear o shaft second (spindle shaft) =1

No. of teeth on driver gear shaft 1: T1 =30

No. of teeth on driven gear shaft 2: T2= 30 × reduction ratio (1st stage)

= 30× 2

= 60

Design of driver gear on shaft T=30

P =2πNT/60

T =P × 60/2πN

=5.5 ×103 ×60/2π×1440

Design of gear Box Shaft

T = P×60/2πN

T =5.5 ×60×103/2π×847

= 62.00×103 N-mm

PCD =3×T

=3×30

=90 mm.

Tangential Component of gear

Pt =2×T/PCD

=2×62×103/90

=1377.7 N

=140.48 kgf

Radial Component of gear

Pr =Pt× tan 20

=1377.7 ×tan20

=501.44 N

=51.13 kgf.

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

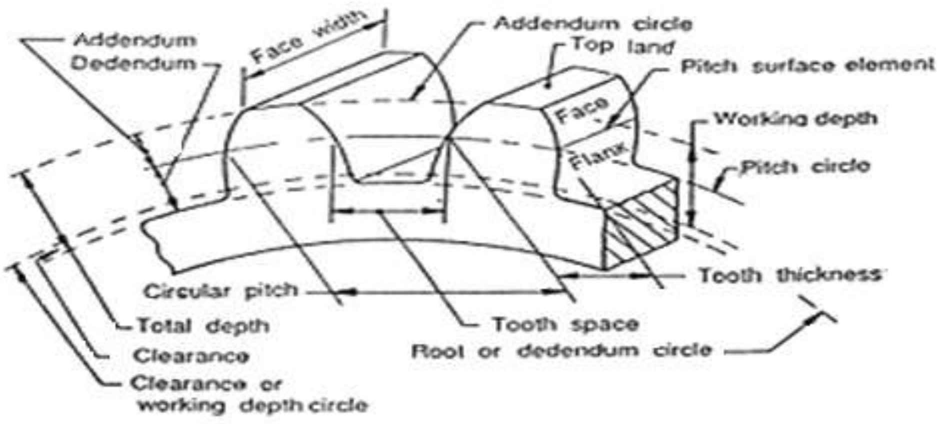


Fig8.2 Gear Terminology

Addendum = 1×m
=1× 3
= 3 mm
Deddendum =1.25 ×m
=1.25 ×3
= 3.75 mm
Working Depth = 2×m
= 2×3
= 6 mm
Minimum tool depth
=2.25 ×m
=2.25× 3
=6.75 mm
Tool thickness =1.5708×m
=1.5708× 3
=4.7124 mm
Minimum clearance
=0.25 ×m
=0.25×3
=0.75 mm
Fillet radius at root
=0.4 ×m
=0.4×3
=1.2mm
Tangential tooth load (Wt):
Wt = P/v ×Cs
Where,
P = Power transmitted in watts.
V = Pitch line velocity in m/s
= π×D×N/60
D =Pitch Circle diameter in mm
N = Speed in rpm
V =π×0.090×1440/60
= 6.78 m/s
Cs = Service factor
=2.0 (for heavy shock continuous 24 hours per day)
Wt =P/V×Cs
=5500/6.78 ×2.0
=1622.42 N

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

≈165.44 kgf

Applying Lewis Equation

WT = fw ×b × Pc ×y

Where,

Fw = Permissible working stress = fo ×Cv

fo =Allowable static stress

Cv =Velocity factor

=3/3 +V (For ordinary cut gears operating at velocities up to 12.5 m/s)

Fw =fo ×Cv

= 568.79 ×3/3 +6.78

=568.79×0.31

=176.32 N

Y = tooth form factor

=0.154 -0.912/ T

=0.154 -0.912/30

=0.124

Circular pitch,

Pc = π ×m

=π × 3

=9.42

WT =fw×b×Pc×y

b =WT/ fw×Pc×y

=1622.42/176.32×9.42×0.124

=8 mm.

But the face width may be taken as 9.5 m to 12.5 m so I have taken b=10m

b =10×3

=30 mm

Normal load on tooth (WN) = WT /cosΦ

=1622.42 /cos 200

=1726.54 N

=176.06 kgf

Radial load (WR) =WT×sin Φ

=1622.42 ×sin 200

=554.90 N

=56.58 kgf

Strength factor of the gear =fw ×y

= (WT/b×Pc×y)×y

= (1622.42/30×9.42×0.124)×0.124

= 46.30 ×0.124

= 5.74

Dynamic Tooth Load (WD) = WT + WI

= WT + (21 V (b×C+WT)/21V +$\sqrt{}$b×C+WT)

For Calculating Dynamic load the value of tangential load may be calculated by neglecting CS

WT =P/V

=5500/6.78

=811.21 N

C =Deformation /Dynamic Factor

=k / (1/EP+1/EP)

=0.111 for 200

e =0.0525 for V= 6.78

EP =EG =205 ×103 N/mm2

C =0.111×0.0525/2×(1/205×103)

=597.32 N /mm

Now,

WD can be calculated as,

WD =811.21 + [21×6.78(30×597.32 +811.21)/21 ×6.78 +$\sqrt{\ }$30×597.32 + 811.21]
=811.21 + (2782392.81/279.24)
=10361.74 N
=1056.56 kgf

Static load /beam strength (Ws)

Ws =fe ×b×Pc×y
=866.25×30×9.42×0.42×0.12
=30355.48 N
≈3095.40 kgf

Ws ≥1.5 WD
30355.48>15542.61
So, design is safe for tooth breakage.

Wear tooth load (Ww) = Dp×b×Q×k
Where, DP = (PCD of pinion) =90 mm
B = 30 mm
Q = Ratio Factor =2 TG/TG+TP
=2×42/(42+30)
=1.16

K =Load Stress Factor
= 0.16(BHN/100)2
=0.16(495/100)2
=3.92

Ww =90×30×1.16×3.92
=12277.44 N
=1251.95 Kgf

WW>WD
12277.4 > 10361.71 N
So the design is safe.

Tangential Component of gear
Pt =2×T/PCD
=2×62×103/90
=1377.7 N
=140.48 kgf

Radial Component of gear
Pr =Pt× tan 20
=1377.7 ×tan20
=501.44 N
=51.13 kgf.

9. LUBRICATION & LUBRICANTS :

Machine tools generally work to a high degree of accuracy and are expected to sustain this accuracy over a long period. This requires proper control of friction and Wear of parts which are in relative motion and calls for effective lubrication of the vital elements like bearings, gears etc.

Effective lubrication is achieved by using proper lubricants. Lubricants not only reduce the friction and the consequent heat generation, they also aid in transporting the heat generated.

9.1 Types of lubricants:

Types of lubricants commonly in machine tools are

1. Oil
2. Grease
3. Oil mist
4. Solid lubricants

Of the above types oil & grease are more commonly used.

Mist lubrication is restored to only for very high speed spindles. Solid lubricants are used in situations where lubrication point is increasable or likely to be neglected.

9.2 Modes of lubrication:

A proper of the lubricants depends up on the mode of lubrication. Operating conditions like speed, load, lubricant properties, surface quality etc. determine the mode of lubrication. There are three modes of lubrication:

1. Boundary lubrication
2. Mixed lubrication
3. Fluid film lubrication

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

9.3 Selection of Lubricants:-

The general choose of lubricant is between oil and grease. The advantages and disadvantage between oil and grease are as follows.

Lubricant	Advantages	Disadvantages
Oil	Effective cooling carries away dirt, Wear debris, etc. Suitable for wide operating conditions Easy to drain and refill. Coefficient of friction & frictional torque is low.	Invariably requires a pump smyce. Requires good sealing. High maintenance and initial cost.
Grease	Simplicity in design. Easy sealing arrangements. Easily retained in the housing. Low maintenance cost.	Poor cooling property. Not suitable for very high speed applications. Retains dirt and Wear debris.

Table no.7.1Advantages & Disadvantages of Lubricants

The most common mode of lubrication for lubricating spindle elements and gears of gear box is grease, because of its above mentioned properties of low maintenance and initial cost. So I select grease for the lubrication purpose.

10. SEALING OF ROLLING BEARING :

Seals are be used to retain lubricant and to prevent dirt and other forging matter from entering element .In machine tools the space adjacent to work heads is usually limited , making it may be necessary to have external sealing arrangements in chucks in addition to seals in the bearing housing . Different types of sealing arrangements are in vogue. Rubbing type seals generally provide effective sealing but result in increased friction and heating, thus imposing restrictions on the speed of operation labyrinths or clearance seals on the other hand are suitable higher speeds and can be designed to suit both oil and grease lubrication.

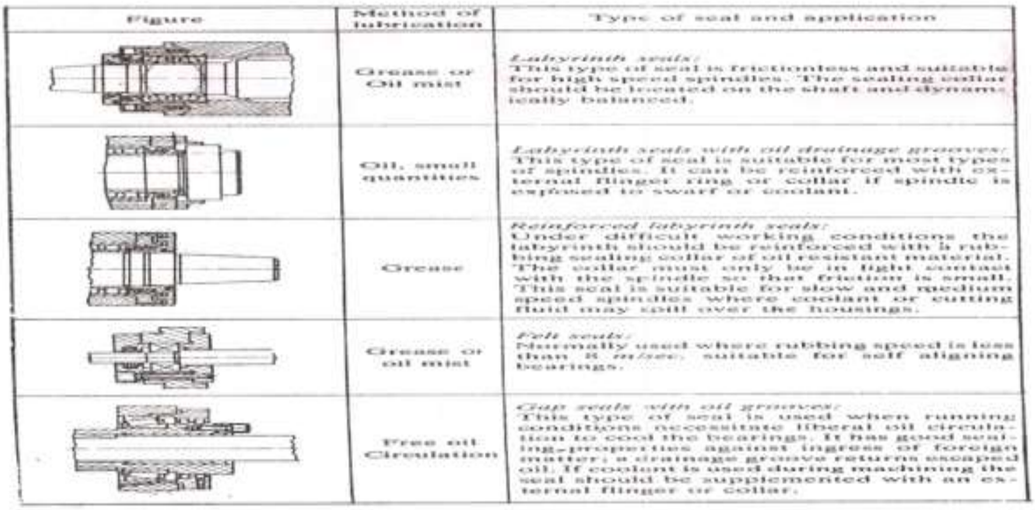


Figure	Method of lubrication	Type of seal and application
	Grease or Oil mist	*Labyrinth seals:* This type of seal is frictionless and suitable for high speed spindles. The sealing collar should be located on the shaft and dynamically balanced.
	Oil, small quantities	*Labyrinth seals with oil drainage grooves:* This type of seal is suitable for most types of spindles. It can be reinforced with external flinger ring or collar if spindle is exposed to swarf or coolant.
	Grease	*Reinforced labyrinth seals:* Under difficult working conditions the labyrinth should be reinforced with a rubbing sealing collar of oil resistant material. The collar must only be in light contact with the spindle so that friction is small. This seal is suitable for slow and medium speed spindles where coolant or cutting fluid may spill over the housings.
	Grease or oil mist	*Felt seals:* Normally used where rubbing speed is less than 8 *m/sec*, suitable for self aligning bearings.
	Free oil Circulation	*Gap seals with oil grooves:* This type of seal is used when running conditions necessitate liberal oil circulation to cool the bearings. It has good sealing properties against ingress of foreign matter; a drainage groove returns escaped oil. If coolant is used during machining the seal should be supplemented with an external flinger or collar.

Table No. 8.1 Sealing of Rolling bearing

11. Assembly Procedure for spindle and Gear box :

11.1 Assembly procedure of Spindle unit :

The procedure for assembly of spindle unit is as mentioned below

1) Clean the spindle housing thoroughly by using kerosene and air pressure.
2) Clean the spindle using same method as above.
3) Heat the front side bearing (Φ) in hydraulic /lubricating oil.
4) Insert the front side spacer (no. 1) at the location provided in the spindle housing.
5) Remove the bearing from oil bath after temperature of bearing reaches 60^0 C to 70^0 C.

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

6) By using tongs lift the bearing from oil bath after temperature of bearing reaches 60^0C to 70^0C
7) Then keep the assembly in open for sometimes so that the temperature of assembly attains the room temperature and bearing fits tightly on to the spindle.
8) Insert the spindle into the spindle housing from front side of the housing.
9) The front side bearing gets located at the correct specified location.
10) Now insert the middle spacer (number 1) spacer are mostly made of EN8 material
11) Insert the 2^{nd} taper roller bearing in hot condition on the spindle (Ø90mm).
12) Again keep the spindle unit assembly in open for sometime so that the temperature of assembly attains the room temperature and bearing fits tightly on to the spindle.
13) Insert the 3^{rd} spacer.
14) Then insert cylindrical rolling bearing in the hot condition.
15) Insert the small spacer (number 2).
16) Fit the oil seal cap using Allan bolt (M6).
17) Then tight it.
18) Fit the oil seal using oil seal cap.
19) Tight the check nut with lock nut Between the two nuts lock washer is to be fitted.
20) Check the run out of spindle using a dial gauge. The total indicating reading (TIP) should be within 0.02 to 0.01.
21) Then start preloading of bearing by means of chuck nut.
22) Lock the washer.
23) Then fit the front side cap.

11.2Assembly Procedure for Gear Box :

The procedure for assembly of spindle units is as mentioned below.

1) Insert the key in the keyway on the spindle.

Subassembly of Gear And Gear Holding Bush :

1) Gear should be located on bush by using two do Il pins.

Assembly of Cluster Gear :

1) Insert the ball bearing on both sides of cluster gear

Assembly of Gear Covers Of Gear Box :

1) Insert the baring on the motor gear shaft and fit into the gear box cover.
2) Insert the bush removing spacer into the cluster shaft.
3) Insert the cluster bush. Fit it into round cover on the gear box cover by using 6 bolts.

Assembly of Main Gear Box :

1) Place the motor shaft bearing number 2 into the gear box housing.
2) Fix the cluster shaft by using 4 bolts(M12)
3) Insert the gear holding bush sub-assembly.
4) Insert the spacer number 3.
5) Place the circlip.
6) Insert the cluster gear assembly on the cluster shaft.
7) Then fit the circlip.
8) Insert the motor gear shaft into the motor gear shaft bearing number 2
9) Put the gasket with the help of gasket oil.
10) Assemble the rear cover with proper location in the motor shaft bearing number 1 and cluster bush.
11) Tighten the cover assembly with the help of bolt.
12) Fit the drain plug and oil window glass.
13) Pmy the lubricating oil from top tapped plug.
14) Maintain the oil level.
15) Apply grease in the spindle housing and plug it with grease nipple.
16) Fit the motor in position and tighten the 4 bolts.
17) Connect the supply of motor.
18) Run the motor.
19) Paint the spindle as per requirement.

12. SEQUENCES OF OPERATIONS :

12.1 Spindle Shaft :

Sr. No.	Description	Machine	Location/clamping	Tool used	Inspection details
1	Drop forging	Hydraulic press	Die		Visual
2	Grinding of extra material	Portable grinder		Grinding wheel	Visual
3	Milling Ø 40 mm	Centre lathe	3 jaw chuck	ISO 9 P30 2525	Bore gauge

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

4	Grinding Ø 100 mm	Centre type grinder (plunge grinder)	Between centers	Grinding wheel	Vernier micrometer
5	Grinding Ø 90 mm	Centre type grinder (plunge grinder)	Between centers	Grinding wheel	Vernier micrometer
6	Grinding Ø 85 mm	Centre type grinder (plunge grinder)	Between centers	Grinding wheel	Vernier micrometer

Table No. 12.1 Sequences of Operation of spindle shaft

12.2. Gear Shaft 1

Sr. No.	**Description**	**Machine**	**Location/clamping**	**Tool used**	**Inspection details**
1	Turning Ø35mm	Centre lathe	3 jaw chuck	ISO 2 P30 2525	Vernier caliper
2	Milling keyway	Milling Machine	Vice	Slot mill cutter	Visual

Table No. 12.2 Sequence of operation for Gear Shaft

12.3 Gear Manufacturing Process

Sr. No.	**Description**	**Machine**	**Location/clamping**	**Tool used**	**Inspection details**
1	Gear cutting	Milling Machine	Mounted between two centers with mandrel	Form disc cutter	Vernier Micrometer
2	Broaching a key in a gear center bore	Vertical broaching machine	Broaching fixture	Keyway broach	

Table No. 12.3 Sequence of operation for Gear manufacturing process

13. JUSTIFICATION OF NEW SPM LOCUS CLEARANCE MILLING :

Operation	**Machine**	**Total Cycle time In min**	**No. of Components per month**	**Operators Per shift**	**Machining cost per unit**	**Machining cost per month**
1	OLD SPM	17.11	2103	3	20	1, 01,880
2	NEW SPM	5.5	6792	1	15	42,060

14. TOLERANCES

I have selected following tolerances for various elements like shafts, bearings etc after referring to table 189, and page 494-496 of Machine Tool Handbook by CMTI. The tolerances selected are as follows:-

Sr. No.	Elements	Tolerance Grade	Type of fit
1	Shaft	K6	Transition fit
2	Sliding(spacers)	H7/J6	Transition fit
3	Location	H7/H6	Transition fit
4	Ball bearings in gear box	K7	Clearance fit

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

5	Front bearing on spindle shaft	K6	Transition fit
6	Rest all bearings on spindle shaft	J6	Transition fit
7	Press fit	H7/N6	Transition fit

Table No. 11.1 Tolerances of Shaft & Bearing etc.

15. VOLUME AND WEIGHT OF SPINDLE SHAFT

Method selected : Drop forging method.
Raw material : bar stock.
Raw material size: Φ68 ×679.5

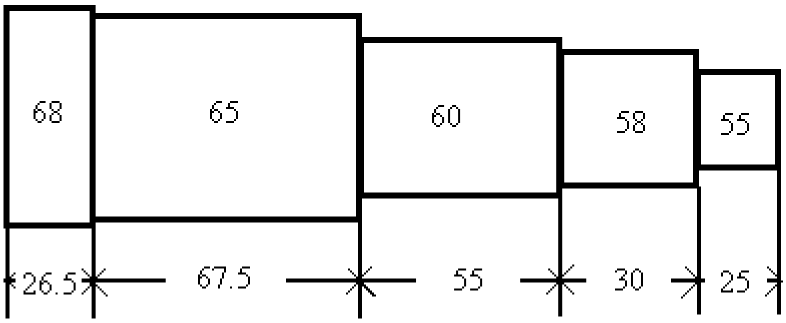


Fig.12.1 Stepped spindle shaft

STEPS:
As the shaft is cylindrical one, its volume is
V=π d2h/4
But the shaft is of different diameters.

Net volume = π /4×682 ×26.5 +π /4 ×652×67.5+602×55×π/4+ π/4×582×30+ π/4×552×25
=782268.5× π/4
=614392.24 mm3
=614.392 cm3

Net weight = 614.392×(7.9/100) = 48.53 kg

Consider the losses, flash loss is shown on the three sides 25 mm on three sides with a thickness of 1.5mm sprue loss is on the last side.

Flash loss = Perimeter ×width ×thickness
= ((172.78+182.21+188.49+204.20+213.62)/5)×679.5×1.5
=130640.67 mm3
=130.64 cm3
weight = 130.64×(7.9/100)
=1.03 kg.
Sprue loss = 7.5 % of 48.53
=3.64 kg
Scale loss = 7.5 % of 48.53
=3.64 kg
Tongs hold a volume of V = πD 2h/4
= π/4 ×682×68
= 246.954 cm3

Hence I have to take maximum diameter of stock and a length of 68 mm for holding the job in tongs.
Volume = 246.956 cm3
Weight = 246.956×7.9/100
=1.9 kg.
Gross weight of stock = 48.53 +1.03+ 3.64+ 3.64+ 1.9
=58.74kg

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

16. VOLUME, WEIGHT AND COST OF HOUSING

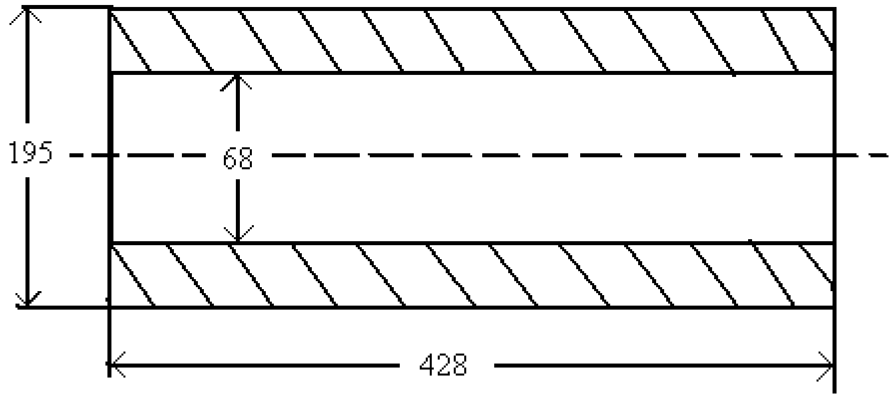


Fig. 13.1 Housing of spindle unit

INPUTS:
Rate of castings in Rs. Per kg. =40
CALCULATIONS:
Volume of A = l×b×h
=488×404×195
=38444640 mm3
=38444.64 cm3
Volume of B = πr2×h
=π×342×488
=177260.38 mm3
=1772.260 cm3
Total volume =Volume of A- Volume of B

=38444.64- 1772.26

=36672.38 cm3

Weight =Volume ×Density

=36672.38×7.2

=264041.13 gm

=264.04 kg.

Total cost = Rate in Rs per kg × Total Weight

=264.04×40
=10561 Rs.

17. JUSTIFICATION OF NEW SPM LOCUS CLEARANCE MILLING

FOR OLD MILLING MACHINE

Working hours/shift	8
No. of working days in Week	6
No. of operators	3
Working days per month	25

STANDARD TIME PER UNIT PER SHIFT PER MONTH

Machine time	12
Operator time	3

International Journal of Advances in Engineering and Management (IJAEM)
Volume 2, Issue 7, pp: 466-485 www.ijaem.net ISSN: 2395-5252

Total time/unit 15
Calculations
Machine time =12/0.85
=14.11
Operator time =3 min.
Total time/unit =17.11
No. of units produced/shift/month
=8×60×25/17.11
=701 units.
No. of units produced per months
=701×3 (Three sifts per day)
=2103 units.
FOR NEW MILLING MACHINE
Working hours/shift 8
No. of working days in week 6
No. of operators 1
Working days per month 25
STANDARD TIME PER UNIT PER SHIFT PERMONTH
Machine time 4.12
Operator time 0.45
Total time/unit 5.3
Calculations
Machine time =4.12/0.95
=4.33
Operator time =1 min.
Total time/unit =5.33
No. of units produced/shift/month
=8×60×25/5.33
=2264 units
No. of units produced per months
= 2264×3 (Three sifts per day)
= 6792 units

Operation	Machine	Total Cycle time In min	No. of Components per month	Operators Per shift	Machining cost per unit	Machining cost per month
1	**OLD SPM**	**17.11**	**2103**	**3**	**20**	**1, 01,880**
2	**NEW SPM**	**5.5**	**6792**	**1**	**15**	**42,060**

Table No. 15.1 Justification of new SPM locus clearance milling

Total cost saving per month : 59,820/-
Total cost saving per year : 7, 17,840/-

III. CONCLUSION :

From the overall procedure I followed in designing of the spindle unit and gear box , I conclude that design is safe , accordingly the design could be brought into practice while designing I have successided in keeping the cost factor to minimum total net savings per year after new SPM is **Rs.7,17,840**

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