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INTEGRATED UNIT OF POWER TRANSMISSION HAVING COMBINATION OF CLUTCH AND BRAKE WITH ANALYTICAL DESIGN

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Abstract - *A combination clutch-brake unit is a single unit which provides both functions of clutching and braking simultaneously. There are many high frequency industrial manufacturing machines such as automatic punching machines, press brakes, printing machines, shears, stamping & forming presses, woodworking machines, looms, combers, roving and ring spinning machines, and warping machines in which there is a requirement of frequent start and stop of the machine but the motor driving the machine is to be kept running continuously. There is also a requirement of a machine to be stopped suddenly during an emergency situation such as machinery failure and accidents caused due to unsafe acts of the worker. So during such requirements of frequent start-stop and sudden stopping of machine, the use of a combination clutch-brake unit is needed. The above requirements involve high power transmission and high braking torque which is generated by using multi-disc friction clutch and disc brake in the combination clutch-brake unit. The combination clutch-brake unit is hydraulic actuated which is more efficient in providing high power transmission and high braking torque than pneumatic actuation. Thus the design using uniform pressure theory and analysis using C-programming, of hydraulic actuated combination clutch-brake unit involving multi-disc friction clutch and disc brake, is done here.*

***Key Words*:** Frictional torque, Shear stress, Yield strength, Axial thrust, Intensity of axial pressure, Locating coordinate.

1. INTRODUCTION

There are various operations in manufacturing industry requiring frequent or continuous start and stop. The requirement is such that when the machine starts it should achieve the required speed instantly and when it is stopped, it should halt immediately. One way to achieve this is to continuously switch the power supply to the machine on and off. But this is not feasible as it would quickly deteriorate the condition of the motor because of the continuous fluctuation of the voltage supply during on and off. Another way is to introduce clutch and brake in the system. So to start the operation the clutch is engaged and brake is disengaged and to stop the operation the brake is engaged and clutch is disengaged. In this a hydraulic actuated piston assembly is used to provide the simultaneous engagement and disengagement process. The assembly is such that the engagement of the clutch and the brake cannot happen at the same time.

1.1 Working

A motor transmits motion to the input shaft usually through V-belt. A drum is rigidly mounted on the input shaft which carries a set of discs A or pressure plates through bolts. The output shaft is splined and coaxial to the input shaft. Two sets of discs (friction plates B and a brake disc C) are placed on the splined sleeve of output shaft, so that they are free to move axially. A stationary outer shaft is coaxial to the output shaft which carries a couple of brake shoes D through bolts. The arrangement of discs in the clutch is such that the discs A and B are alternately placed with clearance. Similar is the case with disc C and brake shoes D. From the output shaft, power is transmitted to the machine. When motor is stopped, a piston moves towards right. The discs B disengages from discs A (clutch is disengaged); followed by the brake shoes D contacting the disc C (braking), thus the output shaft and machine will halt rapidly. While starting the machine, the motor first starts rotating; after the motor acquires enough torque, the piston moves towards left. This disengages the brake first, followed by engagement of clutch (power transmission).

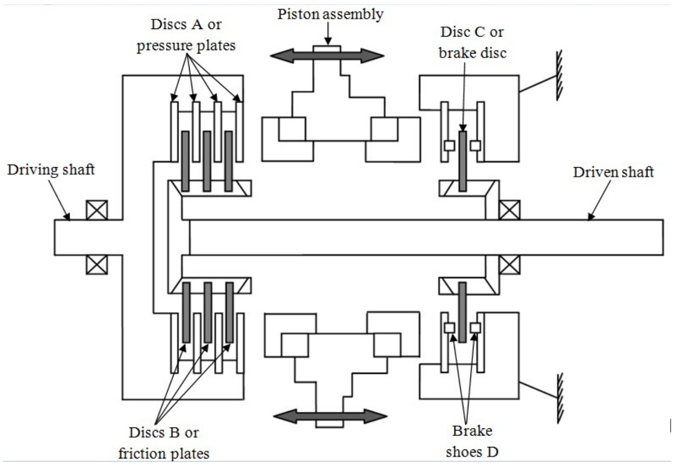

Fig -1: Combination of Clutch and Brake

1.2 Applications

The Combination Clutch-Brake can be used on the main shafts of looms, comber, roving & ring spinning machines & warp beam in warping machines. It can also find its application in punching machines, press brakes, printing machines, shears, stamping & forming presses, and woodworking machine. Therefore, it can be used in machines which requires frequent start and stop and also in situations where machine needs to be stopped suddenly and immediately (but motor needs to be running) such as in the case of an emergency situation.

2. DESIGN FORMULATION

2.1 Design of Shaft

Power required to be transmitted, $P = \frac{2\pi NT}{60}$
where,

N = Speed (rpm) of shaft
T = Frictional torque on clutch

The design of solid shaft is based on twisting moment only.

Frictional torque on clutch, $T = \frac{\pi}{16}\tau d^3$
where,

$\tau =$ Shear stress
d = Diameter of shaft

Shear stress, $\tau = \frac{S_y}{N_f}$
where,

S_y = Yield strength
N_f = Factor of safety

The range of factor of safety, N_f is between 1 and 5.

For design of shaft we consider medium and high carbon steels.

Table -1: Yield stress of material

Sr. No.	Material	Yield strength, S_y N/mm^2
1.	40C8	330
2.	45C8	360
3.	50C4	380
4.	55C8	400
5.	60C4	420

2.2 Design of Clutch

Frictional torque on clutch, $T = n_c \mu_c W_c R$
where,

n_c = Number of contact surfaces for clutch
μ_c = Coefficient of friction for clutch
W_c = Axial thrust with which the clutch discs are held together
R = Mean radius of friction surfaces

μ_c = 0.35 to 0.37 (The friction material used for clutch lining is Kevlar© due to its longevity & smooth engagement properties.)

Now, $R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$ and $W_c = \pi p (r_1^2 - r_2^2)$
where,

r_1, r_2 = Outer & Inner radius respectively of friction surface in clutch
p = Intensity of axial pressure with which the clutch discs are held together

r_2 = (d + 10) mm (Clearance between shaft and friction plate is kept 10 mm for safe design considerations.)

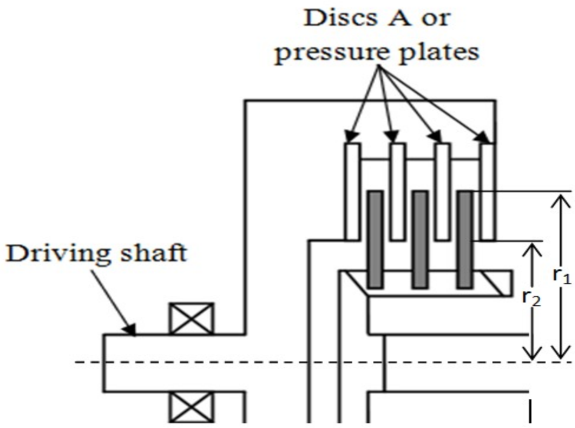

Fig -2: Clutch

2.3 Design of Brake

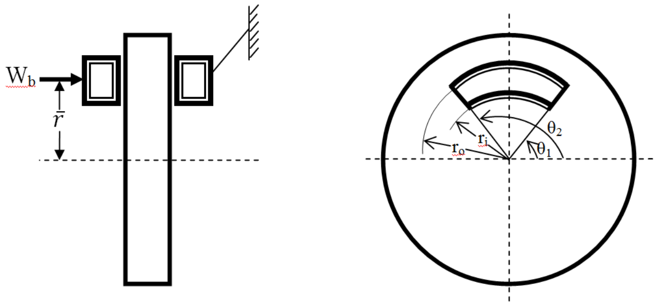

Fig -3: Brake

Locating coordinate of actuating force W_b,
$$\bar{r} = \frac{2}{3}\left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right]\left[\frac{(\cos\theta_1 - \cos\theta_2)}{\theta_2 - \theta_1}\right]$$
where,

r_o, r_i = Outer & Inner radius respectively of brake shoe

θ_1, θ_2 = Angle made by the brake shoe with the centre of brake disc

Also, $\bar{r} = \frac{r_1 + r_2}{2}$ where, r_1 & r_2 are obtained from design of clutch

Actuating force for braking, $W_b = \frac{1}{2}(\theta_2 - \theta_1)p_a(r_o^2 - r_i^2)$ and

Braking torque, $T_b = \frac{2}{3}(\theta_2 - \theta_1)\mu_b p_a(r_o^3 - r_i^3)$

where,

p_a = Largest allowable contact pressure in brake

μ_b = Coefficient of friction for brake

μ_b = 0.5 to 0.55 (The friction material used for brake lining is Feramic due to its quick lock-up property)

3. ANALYTICAL CALCULATION

3.1 Design of Shaft

Input values:

- P=5kW
- N=100rpm
- S_y=330N/mm^2
- N_f=3

$$\tau = \frac{S_y}{N_f} = \frac{330}{3} = 110\ N/mm^2$$

$$P = \frac{2\pi NT}{60} \Rightarrow 5000 = \frac{2\pi \times 100 \times T}{60}$$
$$\therefore T = 477.46\ N.m = 477460\ N.mm$$

$$T = \frac{\pi}{16}\tau d^3 \Rightarrow 477460 = \frac{\pi}{16} \times 110 \times d^3 \therefore d = 28.06\ mm$$

3.2 Design of Clutch

Input values:

- n_c=5
- μ_c=0.35
- W_c=5kN
- p=1N/mm^2

$$r_2 = (d + 10) \Rightarrow r_2 = (28.06 + 10) \therefore r_2 = 38.06\ mm$$

$$T = n_c\mu_c W_c R \Rightarrow 477460 = 5 \times 0.35 \times 5000 \times R$$
$$\therefore R = 54.57\ mm$$

$$W_c = \pi p(r_1^2 - r_2^2) \Rightarrow 5000 = \pi \times 1 \times (r_1^2 - r_2^2)$$
$$\therefore (r_1^2 - r_2^2) = 1591.55\ mm^2$$

$$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right] \Rightarrow 54.57 = \frac{2}{3}\left[\frac{r_1^3 - 38.06^3}{1591.55}\right]$$
$$\therefore r_1 = 57.02\ mm$$

3.3 Design of Brake

Input values:

- $\frac{r_o}{r_i} = 1.5$
- θ_1=30° : θ_2=150°
- p_a=1N/mm^2
- μ_b=0.5

$$\bar{r} = \frac{r_1 + r_2}{2} = \frac{57.02 + 38.06}{2} = 47.54\ mm$$

$$\bar{r} = \frac{2}{3}\left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right]\left[\frac{(\cos\theta_1 - \cos\theta_2)}{\theta_2 - \theta_1}\right]$$

$$\Rightarrow 47.54 = \frac{2}{3}\left[\frac{(1.5r_i)^3 - r_i^3}{(1.5r_i)^2 - r_i^2}\right]\left[\frac{(\cos 30 - \cos 150)}{\frac{\pi}{180}(150 - 30)}\right]$$
$$\therefore r_i = 45.38\ mm$$

$$\frac{r_o}{r_i} = 1.5 \Rightarrow \frac{r_o}{45.38} = 1.5 \therefore r_o = 68.07\ mm$$

$$W_b = \frac{1}{2}(\theta_2 - \theta_1)p_a(r_o^2 - r_i^2)$$
$$= \frac{1}{2}(150 - 30) \times 1 \times (68.07^2 - 45.38^2) \therefore W_b = 2.69\ kN$$

$$T_b = \frac{2}{3}(\theta_2 - \theta_1)\mu_b p_a (r_o^3 - r_i^3)$$

$$= \frac{2}{3}\left[\frac{\pi}{180}(150 - 30) \times 0.5 \times 1(68.07^3 - 45.38^3)\right]$$

$$\therefore T_b = 154.95\ N.m$$

4. PROGRAMMING IN DESIGN

The programming done here is with C language. The open-source software used is Code::Blocks – The open source, cross-platform IDE (Release 17.12 rev 11256 (2017-12-28 10:44:41) gcc 5.1.0 Windows/unicode - 32 bit).

Code:

```
#include<stdio.h>
#include<math.h>
#define PI 3.14159
void main()
{
float
P,T,Tb,N,muc,mub,Wc,Wb,p,pa,rbar,theta1,theta2,R,r1,
r2,ro,ri,nc,nb,n1,n2,tau,d,Sy,Nf, ratiob;
char mu=230,theta=233;
/*DESIGN OF SHAFT*/
printf("\nEnter the following:\n");
printf("  P(kW)= ");
scanf("%f",&P);
printf("  N(rpm)= ");
scanf("%f",&N);
printf("  Sy(N/mm^2)= ");
scanf("%f",&Sy);
printf("  Nf= ");
scanf("%f",&Nf);
printf("\n\t\t\t\tShaft:\n");
tau=Sy/Nf;//N/mm^2
T=(60*P*1000*1000)/(2*PI*N);//N.mm
d=pow(((16*T)/(PI*tau)),0.3333333333);//mm
printf("\t\t\t\t d= %f mm\n",d);
/*DESIGN OF CLUTCH*/
printf("\n\nEnter the following:\n");
printf("  nc= ");
scanf("%f",&nc);
printf("  %cc= ",mu);
scanf("%f",&muc);
printf("  Wc(kN)= ");
scanf("%f",&Wc);
printf("  p(N/mm^2)= ");
scanf("%f",&p);
printf("\n\t\t\t\tClutch:\n");
r2=d+10;//mm
R=T/(nc*muc*Wc*1000);//mm
r1=pow((((R*3*((Wc*1000)/(p*PI)))/2)+
(pow(r2,3))),0.333333333);//mm
printf("\t\t\t\t r1= %f mm\n\t\t\t\t r2= %f mm\n",r1,r2);
/*DESIGN OF BRAKE*/
printf("\n\nEnter the following:\n");
printf("  ro/ri= ");
scanf("%f",&ratiob);
printf("  %c1(degrees)= ",theta);
scanf("%f",&theta1);
theta2=180-theta1;
printf("  %c2(degrees)= %3.0f\n",theta,theta2);
printf("  pa(N/mm^2)= ");
scanf("%f",&pa);
printf("  %cb= ",mu);
scanf("%f",&mub);
printf("\n\t\t\t\tBrake:\n");
rbar=(r1+r2)/2;//mm
ri=(rbar*3*PI*(theta2-theta1)*((pow(ratiob,2))-1))/
(2*180*((cos((PI*theta1)/180))-(cos((PI*theta2)/180)))*(
(pow(ratiob,3))-1));//mm
ro=ratiob*ri;//mm
printf("\t\t\t\t ro= %f mm\n\t\t\t\t ri= %f mm\n",ro,ri);
Wb=(PI*(theta2-theta1)*pa*((pow(ro,2))-(pow(ri,2))))/
(2*180*1000);//kN
printf("\t\t\t\t Wb= %f kN\n",Wb);
Tb=(2*PI*(theta2-theta1)*mub*pa*((pow(ro,3))-(pow(ri
,3))))/(3*180*1000);//N.m
printf("\t\t\t\t Tb= %f N.m\n\n\n",Tb);
}
```

5. RESULT DISCUSSION

The following calculations can be obtained by fixing the values of S_y, N_f, n_c, μ_c, W_c, p, $\frac{r_o}{r_i}$, θ_1, θ_2, p_a, μ_b; and varying the values of P and N.

Table-2: Fixed values of certain design parameters for calculations

Sr. No.	Design Parameter	Value
1.	S_y	380 N/mm^2
2.	N_f	3
3.	n_c	5
4.	μ_c	0.36
5.	W_c	5 kN
6.	p	1 N/mm^2
7.	$\frac{r_o}{r_i}$	1.5
8.	θ_1	30°
9.	θ_2	150°
10.	p_a	1 N/mm^2
11.	μ_b	0.5

Table -3: Calculations

Sr. No.	P (kW)	N (rpm)	d (mm)	r_1 (mm)	r_o (mm)	W_b (kN)	T_b (N.m)
1.	10	100	33.74	69.59	81.14	3.83	262.40
		200	26.78	56.08	66.48	2.57	144.37
		300	23.39	49.55	59.38	2.05	102.88
		400	21.25	45.44	54.91	1.75	81.35
		500	19.73	42.53	51.74	1.56	68.03
		600	18.57	40.32	49.32	1.41	58.93
		700	17.64	38.55	47.39	1.31	52.28
		800	16.87	37.10	45.80	1.22	47.19
		900	16.22	35.87	44.46	1.15	43.17
		1000	15.66	34.82	43.30	1.09	39.89
2.	20	100	42.50	86.68	99.66	5.78	486.21
		200	33.74	69.59	81.14	3.83	262.40
		300	29.47	61.30	72.15	3.03	184.52
		400	26.78	56.08	66.48	2.57	144.37
		500	24.86	52.38	62.46	2.27	119.69
		600	23.39	49.55	59.38	2.05	102.88
		700	22.22	47.30	56.93	1.89	90.66

		800	21.25	45.44	54.91	1.75	81.35
		900	20.43	43.88	53.21	1.65	73.99
		1000	19.73	42.53	51.74	1.56	68.03
3.	40	100	53.55	108.29	123.0	8.81	914.98
		200	42.50	86.68	99.66	5.78	486.21
		300	37.13	76.20	88.30	4.54	338.24
		400	33.74	69.59	81.14	3.83	262.40
		500	31.32	64.89	76.04	3.36	215.99
		600	29.47	61.30	72.15	3.03	184.52
		700	28.00	58.44	69.05	2.78	161.71
		800	26.78	56.08	66.48	2.57	144.37
		900	25.75	54.09	64.32	2.41	130.72
		1000	24.86	52.38	62.46	2.27	119.69
4.	80	100	67.47	135.57	152.5	13.54	1743.3
		200	53.55	108.29	123.0	8.81	914.98
		300	46.78	95.04	108.7	6.87	631.04
		400	42.50	86.68	99.66	5.78	486.21
		500	39.46	80.74	93.22	5.05	397.91
		600	37.13	76.20	88.30	4.54	338.24
		700	35.27	72.58	84.38	4.14	295.10
		800	33.74	69.59	81.14	3.83	262.40
		900	32.44	67.06	78.40	3.58	236.72
		1000	31.32	64.89	76.04	3.36	215.99
5.	100	100	72.68	145.78	163.5	15.57	2150.1
		200	57.69	116.39	131.7	10.11	1121.6
		300	50.39	102.11	116.3	7.88	773.77
		400	45.79	93.10	106.6	6.61	595.06
		500	42.50	86.68	99.66	5.78	486.21
		600	40.00	81.79	94.36	5.18	412.72
		700	38.00	77.88	90.13	4.73	359.64
		800	36.34	74.66	86.63	4.37	319.43
		900	34.94	71.93	83.68	4.07	287.87
		1000	33.74	69.59	81.14	3.83	262.40

It can be observed from the above calculation table that for a highest value of power transmission (P) and corresponding lowest value of speed of shaft (N), the maximum values of various design parameters is obtained. So, from above calculations it can be observed that the values of d, r_1, r_o, W_b, T_b, are maximum for P=100kW (highest value of power transmission) and N=100rpm (corresponding lowest value of speed of shaft).

It can also be observed that for obtaining the minimum values of various design parameters, the lowest value of power transmission (P) and corresponding highest value of speed of shaft (N) is required. So, from above calculations it can be observed that the values of d, r_1, r_o, W_b, T_b, are minimum for P=10kW (lowest value of power transmission) and N=1000rpm (corresponding highest value of speed of shaft).

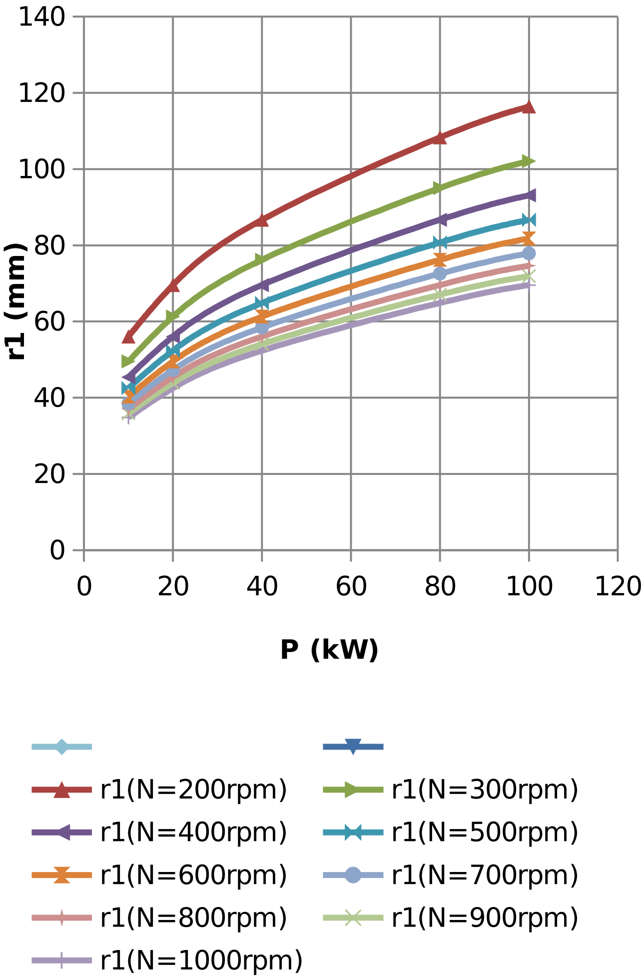

Fig -4: Chart for r_1 vs P in clutch

The above graph is r_1 vs P, for constant values of N in case of clutch which gives a positive slope. This shows that for a constant value of N, P and r_1 are directly proportional. So for a maximum value of P, a maximum value of r_1 will be obtained which is required for safe design consideration. The above graph also shows that for N=100rpm the values of r_1 obtained are maximum as the slope is maximum.

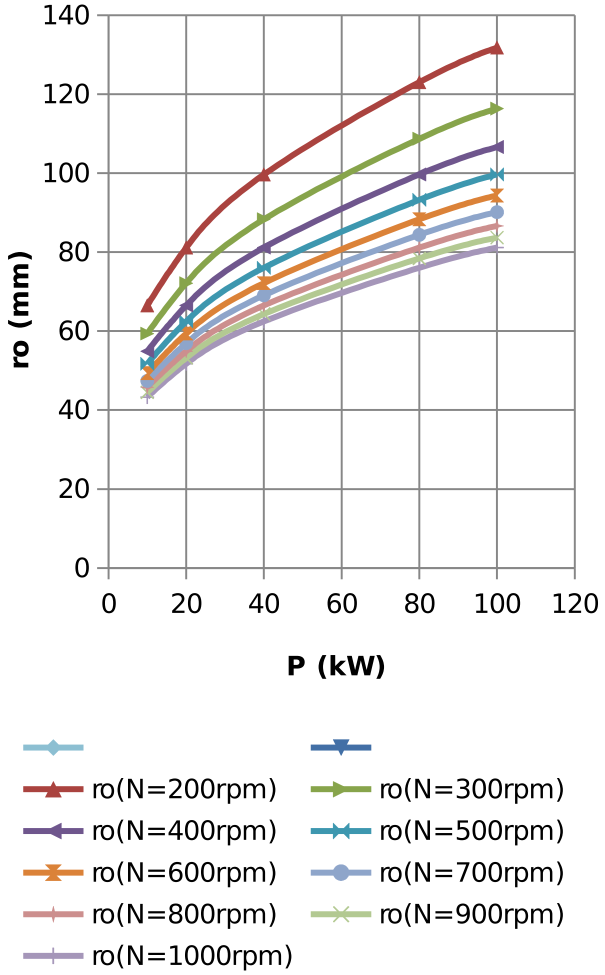

Fig -5: Chart for r_0 vs P in brake

The above graph is r_o vs P, for constant values of N in case of brake which gives a positive slope. This shows that for a constant value of N, P and r_o are directly proportional. So for a maximum value of P, a maximum value of r_o will be obtained which is required for safe design consideration. The above graph also shows that for N=100rpm the values of r_o obtained are maximum as the slope is maximum.

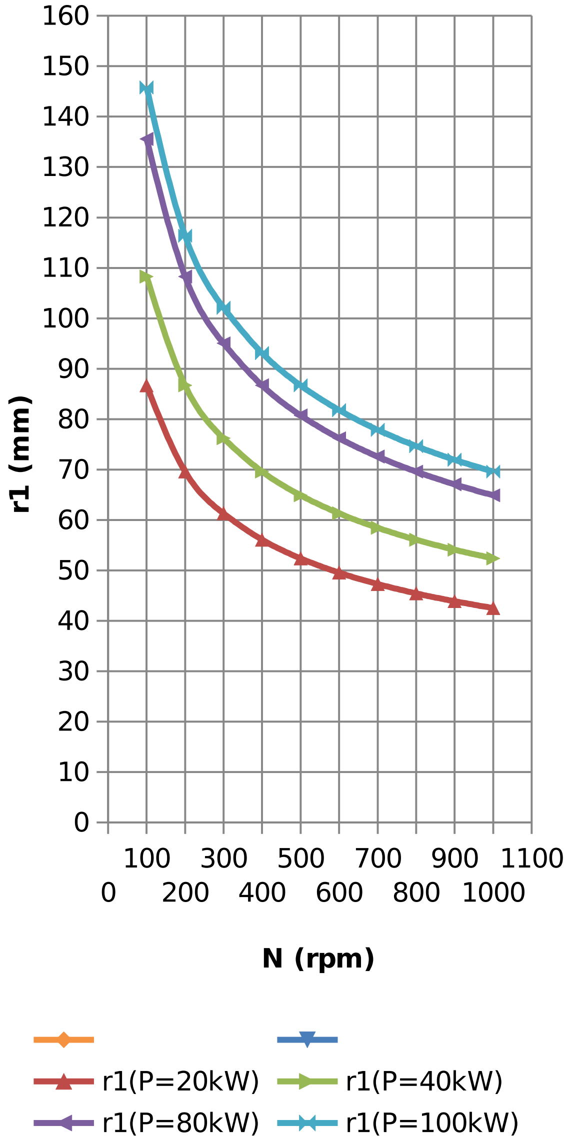

Fig -6: Chart for r_1 vs N in clutch

The above graph is r_1 vs N in case of clutch for constant values of P, which gives a negative slope. This shows that for a constant value of P, N and r_1 are inversely proportional. So for a minimum value of N, a maximum value of r_1 will be obtained which is required for safe design consideration. The above graph also shows that for P=100kW the values of r_1 obtained are maximum as the slope is maximum.

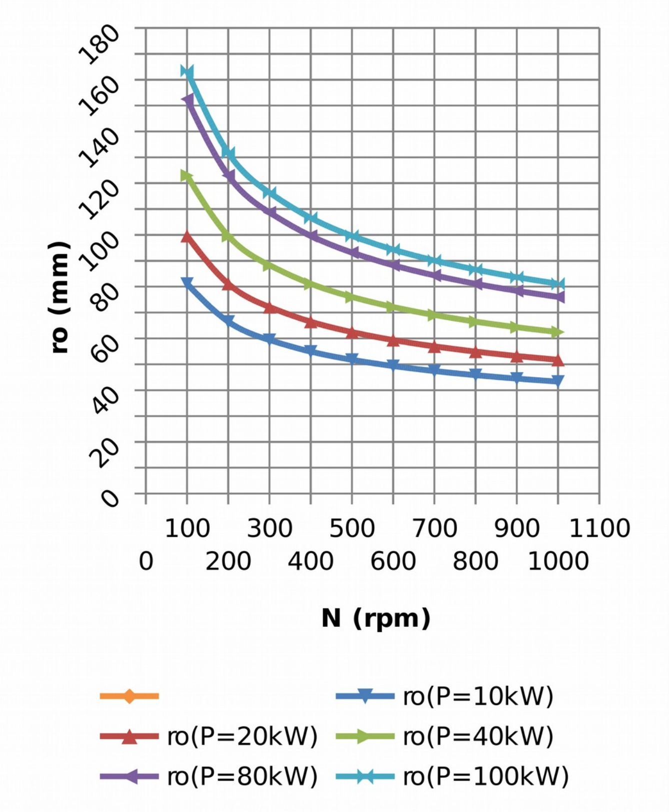

Fig -7: Chart for r_0 vs N in brake

The above graph is r_o vs N, for constant values of P, which gives a negative slope. This shows that for a constant value of P, N and r_1 are inversely proportional. So for a minimum value of N, a maximum value of r_1 will be obtained which is required for safe design consideration. The above graph also shows that for P=100kW the values of r_o obtained are maximum as the slope is maximum.

6. CONCLUSIONS

For the safety of any mechanical component, the maximum values of design parameters are considered.
From the Chart-8, Chart-9 and Chart-10 it can be observed that for constant value of N, the values of r_1, r_o and T_b increases with increase in P. So, the maximum values of r_1, r_o and T_b are obtained from maximum value of P. Therefore, for maximum values of design parameters the maximum value of Power transmission is P=100kW.

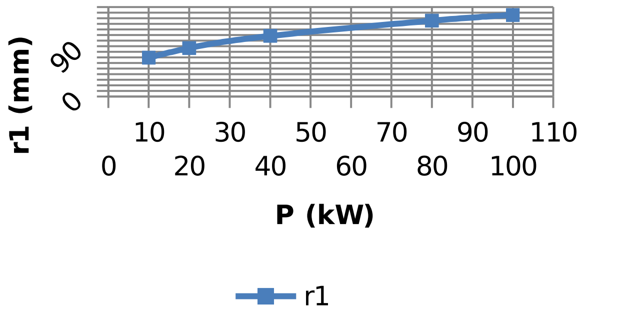

Fig -8: Chart for r_1 vs P (for N=100rpm)in clutch

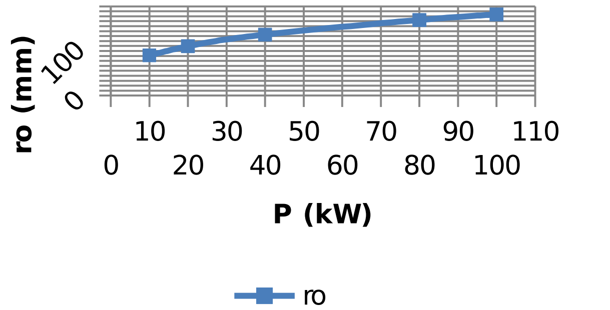

Fig -9: Chart for r_0 vs P (for N=100rpm) in brake

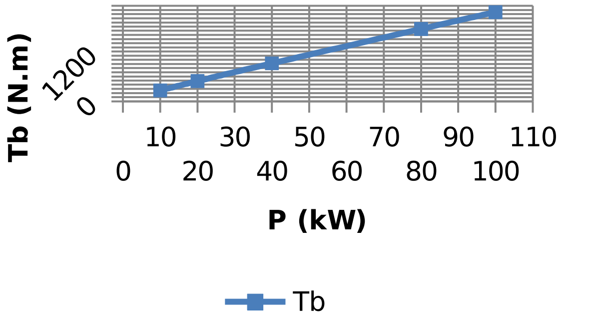

Fig -10: Chart for Tb vs P (for N=100rpm)

From the Chart-11, Chart-12 and Chart-13 it can be observed that for constant value of P, the values of r_1, r_o and T_b decreases with increase in N. So, the maximum values of r_1, r_o and T_b are obtained from minimum value of N. Therefore, for maximum values of design parameters the minimum value of Speed of shaft is N=100rpm.

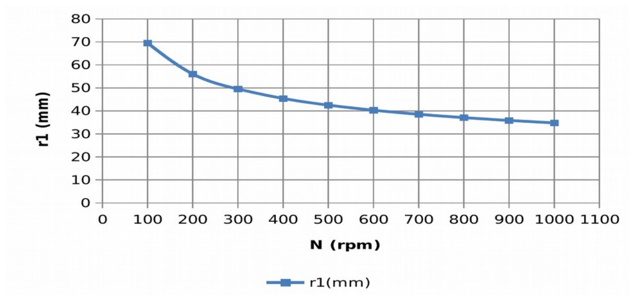

Fig-11: Chart for r_1 vs N (for P=10kW) in clutch

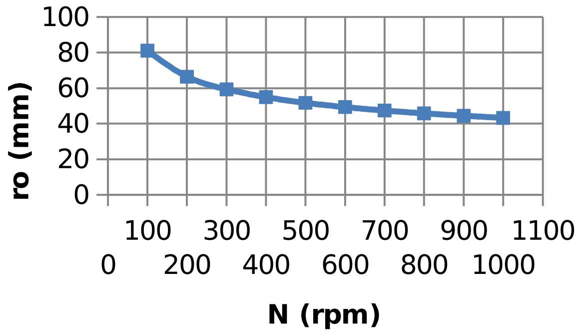

Fig -12: Chart for r_0 vs N (for P=10kW) in brake

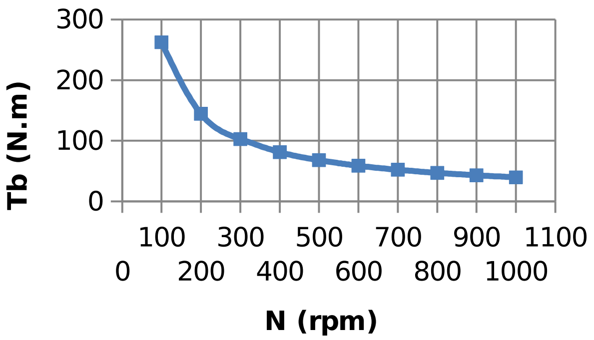

Fig -13: Chart for Tb vs N (for P=10kW)

So for P=100kW and N=100rpm, the maximum values of design parameters are:

Table -4: Maximum values of design parameters

P (kW)	N (rpm)	d (mm)	r_1 (mm)	r_o (mm)	W_b (kN)	T_b (N.m)
100	100	72.68	145.78	163.5	15.57	2150.2

Therefore, for the safe design consideration of Combination Clutch-Brake the maximum values of d, r_1, r_o, W_b, T_b are to be considered which are obtained from the maximum value of power transmission and corresponding minimum value of speed of shaft.Therefore, it can be concluded that the design of Combination Clutch-Brake is done considering the maximum value of the power to be transmitted by clutch and the minimum value of speed of shaft.

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REFERENCES

[1] Joseph E. Shigley, Charles R. Mischke, Richard G. Budynas, Keith J. Nisbett, "Shigley's Mechanical Engineering Design", 8th edition (In SI Units), Tata McGraw Hill Education Private Limited.

[2] Robert C. Juvinall, Kurt M. Marshek, "Fundamentals of Machine Component Design", 5th edition, John Wiley & Sons.

[3] V. B. Bhandari, "Design of Machine Elements", 4th edition, McGraw Hill Education (India) Private Limited.

[4] Robert L. Norton, "Machine Design", 5th edition, Pearson Education.

[5] V. B. Bhandari, "Machine Design Data Book", 1st edition, McGraw Hill Education (India) Private Limited.

[6] E. Balagurusamy, "Programming in ANSI C", 7th edition, McGraw Hill Education (India) Private Limited.

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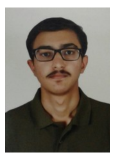

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