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Design & Analysis of Multi Mass Spring flywheel

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ABSTRACT

All engines have flywheels or weighted crankshafts that balance out compression and power strokes, maintain idle speed, aid starting and reduce parts wear. If the flywheel weight is more, then motorcycle needs more force to start, idles badly and is prone to stalling. Weight is not the more important factor here, but inertia. Inertia is gained energy, and is not directly proportional to flywheel weight. It's possible to have a less weight flywheel with much more inertia than a high weight flywheel. The arrangement of the multi mass spring flywheel is an suitable answer to the above problem statement where in the inertia is increased using four set of masses phased opposite to each other. The arrangement of the multi mass spring flywheel is best. Multi Mass Spring Flywheel (MMF) is primarily used for dampening of oscillations in automotive power trains and to prevent gearbox rattling. This Paper explains the Multi Mass spring Flywheel mechanics along with its application and parts. Afterwards a detailed model of the MMF dynamics is presented. This mainly includes a model for the four arc springs in the MMF Multi mass spring flywheel is a clutch device which is used to dampen vibration that occurs due to the slight twist in the crankshaft during the working stroke of I C engine. With the help of Multi Mass spring Flywheel, we can reduce weight of flywheel. Then automatically reduce overall weight of vehicle & Multi mass flywheel gives required performance as per our requirement

Keywords- **Design & Development of Optimized flywheel, Test & Trial on optimized flywheel using Test rig, Plot Performance Characteristic Curves, Analysis of model**

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I. INTRODUCTION

A flywheel is an inertial energy-storage device. It absorbs mechanical energy and serves as a reservoir, storing energy during the period when the supply of energy is more than the requirement and releases it during the period when the requirement of energy is more than the supply

The main function of a fly wheel is to smoothen out variations in the speed of a shaft caused by torque fluctuations. If the source of the driving torque or load torque is fluctuating in nature, then a flywheel is usually called for. Many machines have load patterns that cause the torque time function to vary over the cycle. Internal combustion engines with one or two cylinders are a typical example. Piston compressors, punch presses, rock crushers etc. are the other systems that have fly wheel. Flywheel absorbs mechanical energy by increasing its angular velocity and delivers the stored energy by decreasing its velocity The amount of power a motor develops is not related to flywheel weight. Heavy flywheels do NOT "make more torque", this is completely fictional. The power is merely stored by the flywheels, and they only have what is diverted from the primary. Obviously there's a certain minimum amount of flywheel inertial that should be present for several reasons:

1. Idle stability
2. Tolerance of high compression, cam overlaps etc.
3. Better clutch operation for low speed and traffic operation
4. Fewer load reversals on the driveline during low speed
5. Better traction
6. The carburetor's accelerator pump and off-idle circuit

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settings are closer to "real world"

7. Damps vibration out some
8. Oil pressure is more consistent

Lighter flywheel offers the following advantages

1. Improves acceleration
2. Improves braking.
3. Better suspension compliance in non-IRS where flywheel gyro wraps up the springs under brakes
4. Reduced overall weight

On the other hand lighter flywheel leads to following problems;

1. Is harder to kick through
2. Requires slightly higher idle speed screw setting for stable idle.
3. Is more likely to stall when cold/out of tune
4. Is easier to shift.
5. Has better braking (unless you disconnect the motor by pulling the clutch in while braking)
6. Requires more delicate touch with the clutch in traffic
7. Harder on the primary chain.
8. Less tolerant of "walking speed" in gear

Thus it is safe to interpret from above discussion that the flywheel inertia plays a major role in vehicle optimized performance and by suitable modifying the flywheel mass of flywheel can be reduced by still maintaining the inertia. The arrangement of the multi mass flywheel is an suitable answer to the above problem statement where in the inertia is increased using four set of masses phased opposite to each other. The arrangement of the multi mass flywheel is best the flywheel inertia plays a major role in vehicle optimized performance and by suitable modifying the flywheel mass of flywheel can be reduces by still maintaining the inertia.

2. DESIGN METHODOLOGY

Design of multi mass flywheel system

In our attempt to design a special purpose device we have adopted a very a very careful approach, the total design work has been divided into two parts mainly;

- System design
- Mechanical design

System design mainly concerns with the various physical constraints and ergonomics, space requirements, arrangement of various components on the main frame of machine no of controls position of these controls ease of maintenance scope of further improvement; height of m/c from ground etc.

In Mechanical design the components are categories in two parts.

- Design parts
- Parts to be purchased.

For design parts detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work.

The various tolerances on work pieces are specified in the manufacturing drawings. The process charts are prepared & passed on to the manufacturing stage .The parts are to be purchased directly are specified &selected from standard catalogues

PRIME MOVER SELECTION

Make : Crompton Greaves

Model : IK-35

Engine is Two stroke Spark ignition engine with following specifications:

Bore ‘diameter : 35 mm

Stroke : 35 mm

Capacity : 34 cc

Power out put : 1.2 BHP at 5500 rpm

Torque : 1.36 N-m @ 5000 rpm

Dry weight ; 4.3 kg

Ignition : ELECTONIC IGNITION

Direction of rotation : Clockwise ..looking from driving end

Carburetor :'B' type

Cooling : Air Cooled engine

DESIGN OF ENGINE SHAFT.

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 24	800	680

ASME CODE FOR DESIGN OF SHAFT.

Since the loads on most shafts in connected machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

$fs_{max} = 0.18\ fult$

$= 0.18 \times 800$

$= 144\ N/mm^2$

OR

$fs_{max} = 0.3\ fyt$

$= 0.3 \times 680$

$= 204\ N/mm^2$

considering minimum of the above values ;

$\Rightarrow fs_{max} = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow fs_{max} = 108\ N/mm^2$

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

$\Rightarrow T\ design = 1.36 \times 10^3\ N.mm.$

Check for torsional shear failure of shaft.

Engine shaft is provided with M8 x 1.2 pitch threads at the output side hence the diameter of shaft to be checked in torsional failure is 6.8 mm

$\Rightarrow d = 6.8\ mm$

$Td = \Pi/16 \times fs_{act} \times d^3$

$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times d^3}$$

$$= \frac{16 \times 1.36 \times 10^3}{\Pi \times (6.8)^3}$$

$\Rightarrow fs_{act} = 22\ N/mm^2$

As $fs_{act} < fs_{all}$

$\Rightarrow$ Engine shaft is safe under torsional load

DESIGN OF COUPLING SHAFT.

Material selection : -Ref :- PSG (1.10 & 1.12) + (1.17)

ASME CODE FOR DESIGN OF SHAFT.

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 24	800	680

Since the loads on most shafts in connected

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 24	800	680

machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

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considering minimum of the above values ;

$\Rightarrow fs_{max} = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow fs_{max} = 108\ N/mm^2$

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

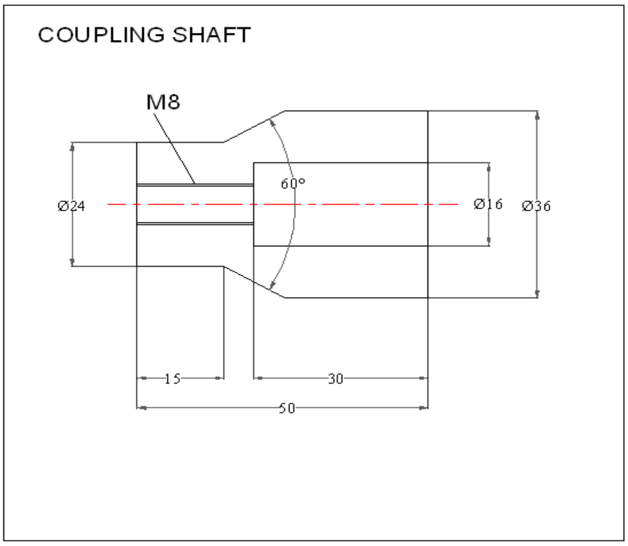
$\Rightarrow T\ design = 1.36 \times 10^3\ N.mm.$

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

Coupling haft is provided with M8 x 1.2 pitch threads at the engine side where as it is hollow at the flywheel shaft end hence the coupling shaft is to be checked in torsional failure as hollow shaft

$\Rightarrow$ Inner diameter (di) = 16mm

Outer diameter (do) = 36



Check for torsional shear failure:-

$$T = \frac{\Pi \times fs_{act} \times}{16} \left(\frac{Do^4 - Di^4}{Do} \right)$$

$$1.36 \times 10^3 = \frac{\Pi \times fs_{act} \times}{16} \left(\frac{36^4 - 16^4}{36} \right)$$

$\Rightarrow fs_{act} = 0.154\ N/mm^2$

As; $fs_{act} < fs_{all}$

$\Rightarrow$ Coupling shaft is safe under torsional load

DESIGN OF FLYWHEEL SHAFT.

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

ASME CODE FOR DESIGN OF SHAFT.

Since the loads on most shafts in connected machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

$fs_{max} = 0.18\ fult$

$= 0.18 \times 800$

$= 144\ N/mm^2$

OR

$fs_{max} = 0.3\ fyt$

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$=0.3 \times 680$

$=204\ N/mm^2$

considering minimum of the above values ;

$\Rightarrow \quad fs_{max} = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow \quad fs_{max} = 108\ N/mm^2$

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

$\Rightarrow T\ design = 1.36 \times 10^3\ N.mm.$

Check for torsional shear failure:-

$$T = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{Do^4 - Di^4}{Do} \right)$$

$$1.36 \times 10^3 = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{36^4 - 16^4}{36} \right)$$

$\Rightarrow fs_{act} = 0.154\ N/mm^2$

As; $fs_{act} < fs_{all}$

$\Rightarrow$ Coupling shaft is safe under torsional load

DESIGN OF FLYWHEEL SHAFT.

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

ASME CODE FOR DESIGN OF SHAFT.

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 24	800	680

Since the loads on most shafts in connected machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

$fs_{max} = 0.18\ fult$

$= 0.18 \times 800$

$= 144\ N/mm^2$

OR

$fs_{max} = 0.3\ fyt$

$=0.3 \times 680$

$=204\ N/mm^2$

considering minimum of the above values ;

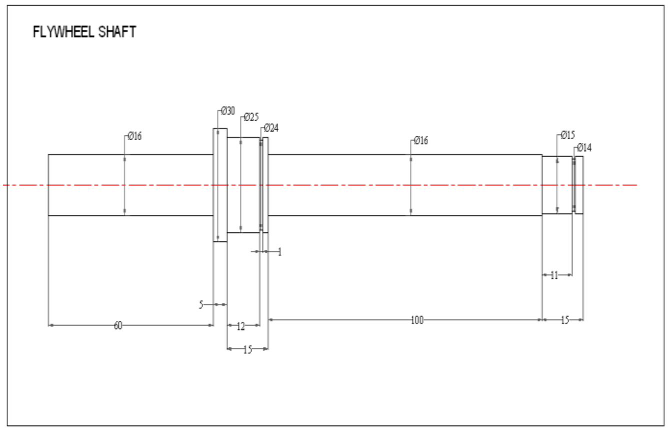
$\Rightarrow \quad fs_{max} = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow \quad fs_{max} = 108\ N/mm^2$

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

$\Rightarrow T\ design = 1.36 \times 10^3\ N.mm.$



CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

Minimum section on the flywheel shaft is 14mm in diameter hence

$\Rightarrow d = 14\ mm$

$Td = \Pi/16 \times fs_{act} \times d^3$

$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times d^3}$$

$$= \frac{16 \times 1.36 \times 10^3}{\Pi \times (14)^3}$$

$\Rightarrow fs_{act} = 2.52\ N/mm^2$

As $fs_{act} < fs_{all}$

$\Rightarrow$ Engine shaft is safe under torsional load

Selection of Bearing on Flywheel shaft

Input shaft bearing will be subjected to purely medium radial loads; hence we shall use ball bearings for our application.

Selecting ; Single Row deep groove ball bearing as follows.

Series 60

No	Bearing of basic design No (SKF)	d	D 1	D	D_2	B	Basic capacity	
25 AC 02	6005	25	28	47	44	12	5200	7800

$P = X F_r + Y F_a$

Neglecting self weight of carrier and gear assembly

For our application $F_a = 0$

$\Rightarrow P = X F_r$

where $F_r = Pt$ = Maximum load at dyno-brake pulley

Maximum load = Torque / Radius of dyno-brake pulley

$= 1.36 \times 10^3 / 30 = 45$

Max radial load = F_r =45 N. (Tension in belt)

$\Rightarrow \quad P = 45N$

Calculation dynamic load capacity of brg

$L = \left(\frac{C}{P} \right)^p$, where p= 3 for ball bearings

For m/c used for eight hr of service per day;

L_H = 4000- 8000hr

But ; $L = \frac{60\ n\ L_H}{}$

10^6

$$L = \frac{60 \times 5000 \times 4000}{10^6}$$

$L = 1200$ mrev

$$\text{Now};1200 = \frac{(C)^3}{(45)^3}$$

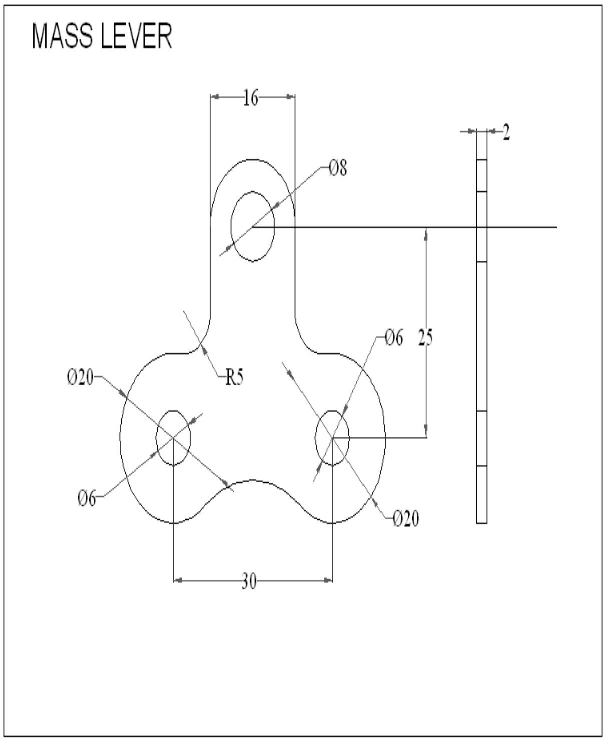
⇒C= 478N

⇒As the required dynamic capacity of brg is less than the rated dynamic capacity of brg;

⇒ Brg is safe

DESIGN OF MASS LEVER

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)



Lever is subjected to bending due to the force at the pin (98.5 N) , the thickness of the lever is 2mm and width of link at hinge pin end is 16mm, this section is decided by the geometry of link, we shall check the dimensions for bending failure

Let; t= thickness of link = 2mm

B= width of link =16 mm

Bending moment;

Section modules; $Z = 1/6\ t\ b^2$

$$Fb = m/z = \frac{PL}{1/6\ t\ B^2}$$

$$= \frac{6PL}{tB^2}$$

Maximum effort applied by hand(P)= 98.5 N

$$\Rightarrow fb = \frac{6 \times 98.5 \times 35}{2 \times 16^2}$$

$fb = 40.4\ N/mm^2$

As $fb_{act} < fb_{all}$

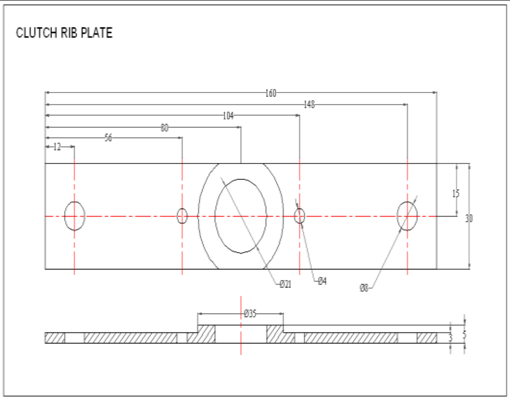
Thus selecting an (16x 2) cross-section for the link.

DESIGN OF CLUTCH RIB PLATE

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN9	600	380

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN9	600	380



Lever is subjected to bending due to the force at the pin (98.5 N) , the thickness of the lever is 3mm and width of link at hinge pin end is 30 mm, this section is decided by the geometry of link, we shall check the dimensions for bending failure

Let; t= thickness of link = 2mm

B= width of link =30 mm

Bending moment;

Section modules; $Z = 1/6\ t\ b^2$

$$Fb = m/z = \frac{PL}{1/6\ t\ B^2}$$

$$= \frac{6PL}{tB^2}$$

Maximum effort applied by hand(P)= 98.5 N

$$\Rightarrow fb = \frac{6 \times 98.5 \times 80}{3 \times 30^2}$$

$fb = 17.5 N/mm^2$

As $fb_{act} < fb_{all}$

Thus selecting an (30x 3) cross-section for the link.

DESIGN OF UNIDIRECTIONAL CLUTCH

SELECTION OF ONE WAY CLUTCH CSK_15

One way clutch is of the same dimensions of ball bearing 6202, it will be subjected to purely medium radial loads; Selecting

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IsI No	Bearing of basic design No (SKF)	d	D1	D	D_2	B	Basic capacity	
CSK-15	6202	15	19	35	31	11	3550	4650

$P = X F_r + Y F_a$

For our application $F_a = 0$

$\Rightarrow P = X F_r$

As; $F_r < e \Rightarrow X = 1$

$\Rightarrow P = F_r$

Max radial load = F_r = 98.5 N.

$\Rightarrow$ P= 98.5 N

Calculation dynamic load capacity of brg

$L = \left(\frac{C}{P}\right)^p$, where p= 3 for ball bearings

When P for ball brg

For m/c used for eight hr of service per day;

L_H = 4000- 8000hr

But ; $L = \frac{60\ n\ L_H}{10^6}$

L=1200 mrev

Now; $1200 = \left(\frac{C}{98.5}\right)^3$

$\Rightarrow$C= 1046.7 N.

$\Rightarrow$As the required dynamic capacity of brg is less than the rated dynamic capacity of brg;

DESIGN OF CLUTCH HOUSING : -

Clutch housing can be considered to be a hollow shaft subjected to torsional load.

Material selection

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
EN 24	800	680

As Per ASME Code;

$\Rightarrow fs_{max} = 108\ N/mm^2$

Check for torsional shear failure:-

$$T = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{Do^4 - Di^4}{Do}\right)$$

$$1.7 \times 10^3 = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{54^4 - 30^4}{54}\right)$$

$\Rightarrow fs_{act} = 0.06\ N/mm^2$

As; $fs_{act} < fs_{all}$

$\Rightarrow$ Hub is safe under torsional load.

DESIGN OF OUTPUT SHAFT.

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 24	800	680

ASME CODE FOR DESIGN OF SHAFT.

Since the loads on most shafts in connected machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

fs_{max} = 0.18 fult

= 0.18 x 800

= 144 N/mm^2

OR

fs_{max} = 0.3 fyt

=0.3 x 680

=204 N/mm^2

considering minimum of the above values ;

$\Rightarrow fs_{max} = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow fs_{max} = 108\ N/mm^2$

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

$\Rightarrow$ T design $= 1.7 \times 10^3$ N.mm.

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

$Td = \Pi/16 \times fs_{act} \times d^3$

$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times d^3} = \frac{16 \times 1.7 \times 10^3}{\Pi \times (16)^3}$$

$\Rightarrow fs_{act} = 2.11\ N/mm^2$

As $fs_{act} < fs_{all}$

$\Rightarrow$ Engine shaft is safe under torsional load

3. EXPERIMENTAL PROCEDURE

Effect of increased inertia of Dual mass flywheel

The effect of inertia augmentation can be seen by the difference in the fluctuation of energy in the Dual mass flywheel and the Conventional flywheel

Let,

Maximum fluctuation of energy of Dual mass flywheel =

$\Delta E_{dmf} = m R^2 \omega_{dmf}^2 Cs$

Where , m =mass of flywheel =1.9 kg

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R= Mean Radius of rim = 68 mm =0.068

ω_{dmf} = mean angular speed of dual mass flywheel
= 2π (N1 +N2)/2 = 2π (1430 +930)/2
ω_{dmf} =7414 rad/sec
Cs = Coefficient of fluctuation of speed = N1-N2 /N
Where N= (N1 +N2)/2 = 118

Cs = 1430-930 /1180 =0.423
$\Delta E_{dmf} = m R^2 \omega_{dmf}^{\ 2}$ Cs
= 1.9 x 0.068^2 x 7414^2 x 0.423 =204.27 KJ

Maximum fluctuation of energy of Conventional flywheel = $\Delta E_{cnv} = m R^2 \omega_{cnv}^{\ 2}$ Cs
Where , m =mass of flywheel =1.9 kg
R= Mean Radius of rim = 68 mm =0.068
ω_{cnv} = mean angular speed of dual mass flywheel
= 2π (N1 +N2)/2 = 2π (1315 +910)/2
ω_{cnv} =6990 rad/sec
Cs = Coefficient of fluctuation of speed = N1-N2 /N
Where N= (N1 +N2)/2 = 1112
Cs = 1315-910 /1112 =0.364
$\Delta E_{cnv} = m R^2 \omega_{dmf}^{\ 2}$ Cs
= 1.9 x 0.068^2 x 6990^2 x 0.364 =156.25 KJ
Effectiveness (έ) = $\Delta E_{dmf} / \Delta E_{cnv}$ = 204.27 /156.25 =1.30
Thus the Dual mass flywheel is 1.3 times effective than the Conventional flywheel

OBSERVATION TABLE :
Conventional mount flywheel
ENGINE SPEED = 1300 rpm
Engine Power = 205 watt
Sample calculations :

a) Output Torque = W x 9.81 x Radius of dyno-brake pulley

Top = 4 x9.81 x 0.032 =1.26 N-m

b) Output power = 2 π N Top / 60

Pop =2 π x 1155 x 1.26 /60 = 152.39 watt

c) Efficiency = (Output power/ Input power) x 100 = (152.39 /205) = 74.33

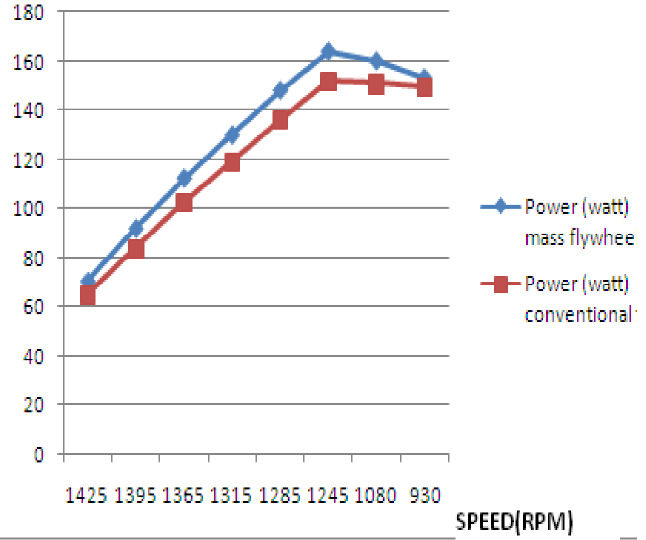
RESULT TABLE CONVENTIONAL MOUNT

SR.	LOAD (gm)	SPEED	TORQUE	POWER	EFFICIENCY	ACC
1	1500	1315	0.47088	64.85	31.63	31.5
2	2000	1275	0.62784	83.83	40.89	40
3	2500	1245	0.7848	102.33	49.91	50
4	3000	1205	0.94176	118.85	57.97	63
5	3500	1185	1.09872	136.36	66.51	80
6	4000	1155	1.25568	151.89	74.09	100
7	4500	1020	1.41264	150.90	73.61	125

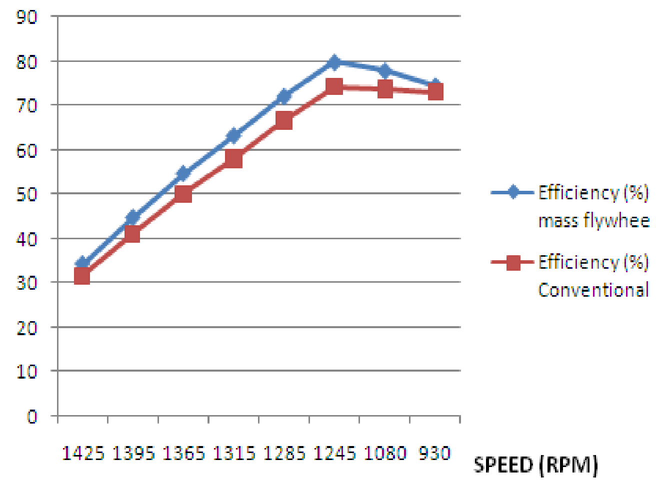
RESULT TABLE MULTI MASS MOUNT

SR	LOAD (gm)	SPEED	TORQUE	POWER	EFFICIENCY	ACC
1	1500	1425	0.490	66.55	35.60	32.6
2	2000	1395	0.630	86.83	45.89	41.5
3	2500	1365	0.803	106.33	55.90	51.2
4	3000	1315	0.954	125.85	62.50	64
5	3500	1285	1.205	145.36	71.50	84
6	4000	1245	1.300	161.80	79.09	102
7	4500	1080	1.645	160.90	7550	128

RESULT AND DISCUSSION:
Comparison of Power output of Conventional and Multi mass flywheel



It is observed that there is approximately 7 to 8 % increase in power output by using the Dual mass flywheel
Comparison of Efficiency of Conventional and Multi mass flywheel



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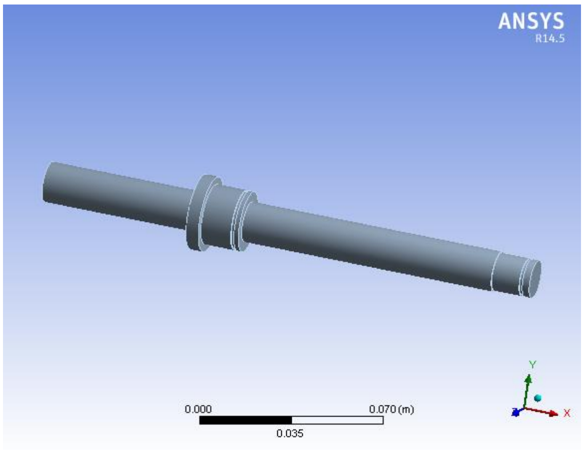
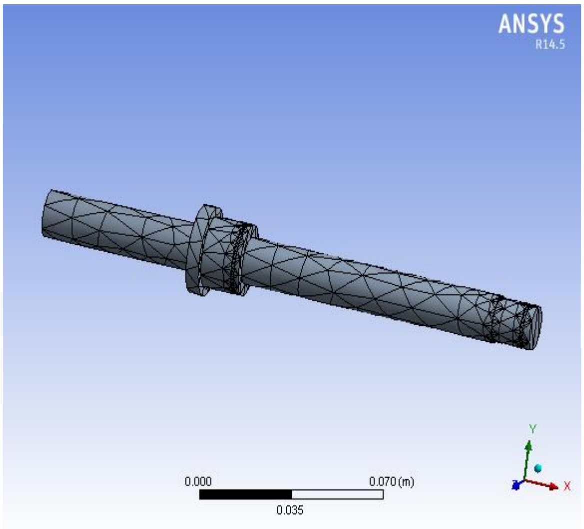
It is observed that the Dual mass flywheel is 5 to 6 % efficient than the conventional flywheel which will also result in increasing fuel economy of the engine.

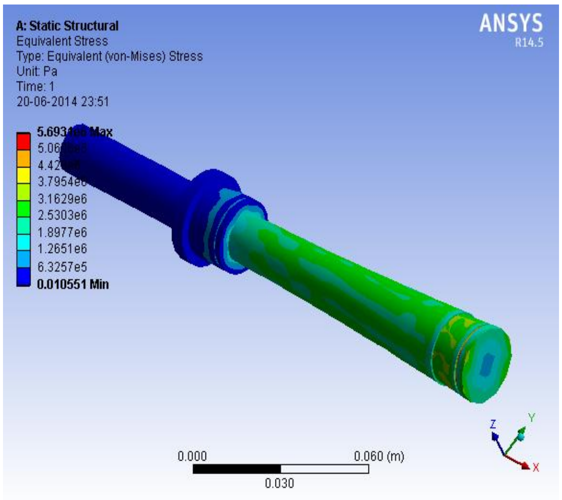
DESIGN VALIDATION OF FLYWHEEL SHAFT GEOMETRY



MESHING

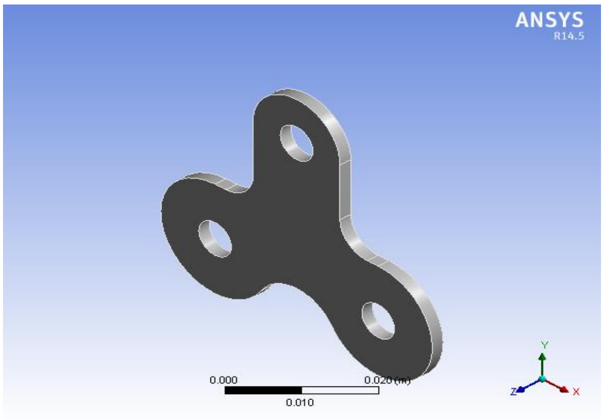


SOLUTION

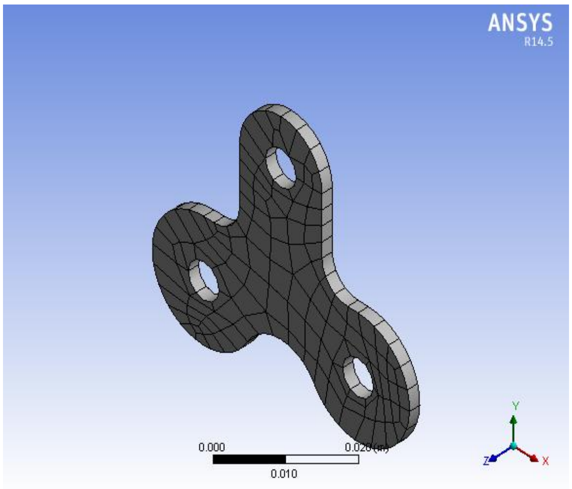


Maximum torsional shear stress induced in the flywheel shaft = 5.69 N/mm2 which is less than the allowable streess hence the pin safe under Torsional shear failure

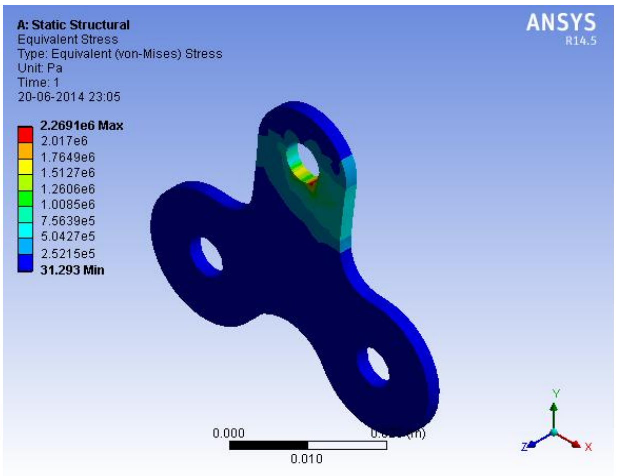
DESIGN VALIDATION OF MASS LEVER GEOMETRY



MESHING



SOLUTION



Maximum bending stress induced in the pin = 2.26 N/mm2 which is less than the allowable streess hence the pin safe under bending failure

CONCLUSIONS

1) Lowered weight of fly wheel system will reduce system weight thereby leading to better fuel economy of vehicle
2) Compact size : The size of the flywheel will lead to better cabin space of vehicle

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3) Lowered weight of flywheel will reduce the overall material cost
4) Engine life increases due to balanced power output
5) Improve Overall performance of engine due to less weight
6) Damping out vibration of the engine shaft & it gives better performance
7) Flywheel mass optimization will lead to better acceleration characteristics of the vehicle

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