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Advance development in Dual Mass Flywheel

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Abstract- The growing concerns for the environment and the adoption of more stringent regulations have marked the development of new engines technologies. This result in the development of more efficient engines and also allowed to deliver more torque and power at low revolutions in these engines. However, this caused an increase in the level of vibration. The traditional clutch discs are unable to absorb such vibrations. This is where the Dual Mass Flywheels play a key role.

In the present study the focus is on development of new flywheel for two stroke engine system using helical springs and multi-mass system to improve inertia of flywheel and to improve the engine efficiency. The aim of this project is to do the comparative analysis of spring mass flywheel Vs Conventional flywheel. This includes fabrication of a prototype, design of prototype components, analysis testing of prototype at different speed and load, modification in prototype if required. It is observed that multi mass flywheel system improves power output and fuel efficiency of the two stroke engine.

Keywords- *Engine, Dual mass flywheel, Two Stroke Engine, Prototype, Parameters.*

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1. INTRODUCTION

The principle of the flywheel is found in the Neolithic spindle and the wheel. The flywheel as a general mechanical device for equalizing the speed of rotation. In the Industrial Revolution, James Watt contributed to the development of the flywheel in the steam engine, and his contemporary James Pickard used a flywheel combined with a crank to transform reciprocating into rotary motion. The flywheel is effectively a weight which is fastened to the end of the crankshaft of the engine. The power from the pistons tends to be created in pulses and the weight of the flywheel smoothes out these pulses by providing inertia to the rotating engine, As well as providing a weight the flywheel has a gear around its circumference on which the starter motor operates and is a convenient means of attaching the clutch which provides a variable connection to the transmission.

Flywheels are often used to provide continuous energy in systems where the energy source is not continuous. In such cases, the flywheel stores energy when torque is applied by the energy source (here 2 stroke-engine), and it releases stored energy when the energy source is not applying torque to it. For example, a flywheel is used to maintain constant angular velocity of the crankshaft in a reciprocating engine. In this case, the flywheel which is mounted on the crankshaft stores energy when torque is exerted on it by a firing piston, and it releases energy to its mechanical loads when no piston is exerting torque on it. Other examples of this are friction motors, which use flywheel energy to power devices such as toy cars. [1]

2. DUAL MASS FLYWHEEL

The rapid development of vehicle technology over the last few decades has brought ever higher performance engines paralleled by an increased demand for driver comfort. In addition, lean concepts, extremely low-speed engines and new generation gearboxes using light oils contribute to this. Since the middle of the 1980s, this advancement has pushed the classic torsion (spring mass) damper as an integral part of the clutch driven plate to its limits. With the same or even less installation space available, the classic torsion damper has proved inadequate to outbalance constantly increasing engine torques. Extensive development by LuK resulted in a simple, but very effective solution – the Dual Mass Flywheel (DMF) – a new torsion damper concept for the drive train shown in fig.1

Fig.1.Dual mass flywheel

The dual-mass flywheel is actually a great piece of engineering. This relatively new piece of equipment has been a 'must have' fixture to most modern day engines as standard equipment. Any engine that is properly balanced is prone to vibration in a number of ways. These vibrations are almost impossible to eradicate due to the repetitive and stringent combustion forces acting on the pistons, connecting rods and crankshaft at regular intervals as per the firing order of a particular engine as shown in fig.2. The most damaging of these vibrational modes experienced is torsional and the effect gets worse at the lower engine RPM range.

DMF is a device which is used to dampen vibration that occurs due to the slight twist in the crankshaft during the power stroke. The torsional frequency is defined as the rate at which the torsional vibration occurs. When the torsional frequency of the crankshaft is equal to the transaxles torsional frequency an effect known as the torsional resonance occurs. The vibration caused by the torsional resonance when the operating speed of the engine is low can be avoided using dual mass flywheel. [2]

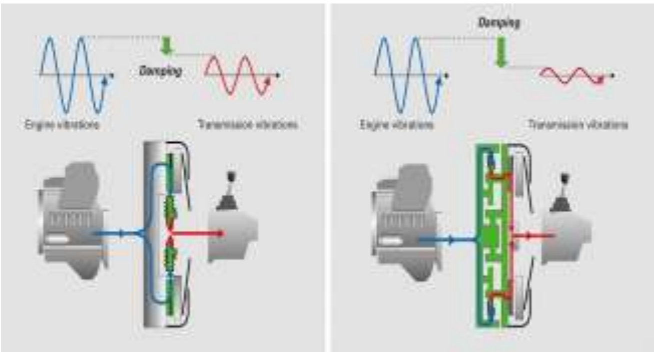

Fig.2. Vibration damping from conventional Flywheel and DMF

In manual transmission, the vibration of engine torque causes rattle noise due to backlash between teeth of transmission gears. While booming noise is generated due to resonance which is produced when vibration frequency of engine matches with natural frequency of transmission.

Hence it becomes interesting and worth to study DMF component design and comparison of DMF with conventional flywheel on the basis of speed, torque, power and efficiency.

A standard DMF is as shown in Fig.3. It consists of the primary flywheel and the secondary flywheel. The three decoupled masses are connected via a spring/damper system and supported by a deep groove ball bearing so they can rotate against each other. The primary mass with starter ring gear is driven by the engine and tightly bolted to the crankshaft. It encloses, together with the primary cover a cavity which forms the arc spring channel.

At the heart of the torsion damper system are the arc springs. They sit in guides in the arc spring channels and cost effectively fulfills the requirements of an "ideal" torsion damper. The guides ensure correct guidance of the springs during operation and the grease around the springs reduces wear between the guides, channels and the springs. Torque is transferred via the flange. The flange is bolted to the secondary flywheel with its wings sitting between the arc springs. The secondary flywheel helps to increase the mass moment of inertia on the gearbox side. Vents ensure better heat dissipation. As the DMF has an integral spring/damper system, a rigid` clutch disc without torsion damper is normally used. [7]

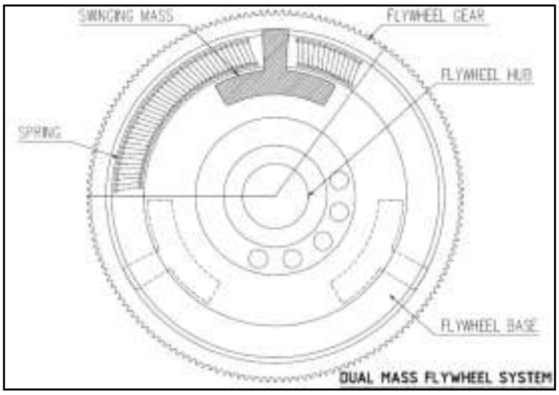

Fig.3.Dual mass flywheel (DMF)

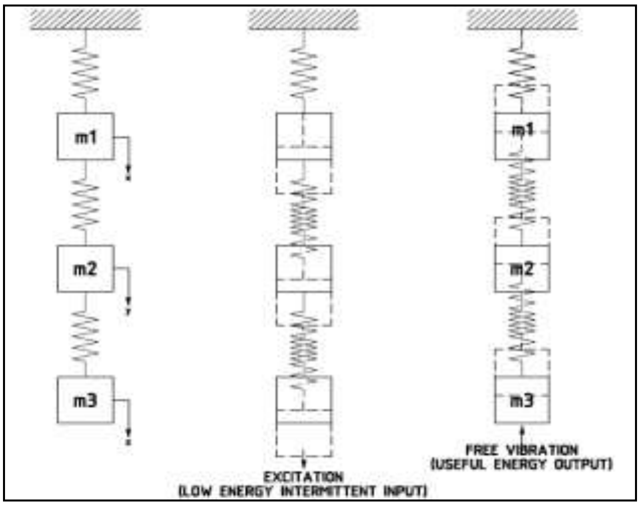

Fig.4.Principle of operation of Dual mass flywheel (DMF)

Fig.4 elaborates the principle of DMF operation. This set up also called as free un-damped vibrations set up of two mass- two spring system. The input to the system is in the form of an low energy intermittent input from any power source (excitation), this results in free un-damped vibrations are set up in the system. It results in the free to and fro motion of the mass m1 and m2. This motion is assisted by gravity and will continue until resonance occurs, i. e. the systems will continue to work long after the input which is intermittent; has ceased. Hence the term free energy is used.

3. DESIGN AND ANALYSIS

The study is done on a fabricated testing set up. This set up consists of base frame, engine, fuel tank, rope pulley and flywheel. The construction of the testing set up is as shown in Fig.5 and Fig.6.

Using the same principle of free energy three arc springs with three masses lies in the channel provided inside primary flywheel and supported by the guides. To prevent the arc springs from wear, sliding contact areas are lubricated. One end of this arc spring is fixed on primary flywheel with the help of stopper while at the other free end mass is attached.

These three masses are connected with mass lever with the help of three mass hinge pins. This mass lever is riveted with secondary flywheel. Hence it transfer torque from the primary flywheel via the arc springs and masses to the secondary flywheel; in other words, from the engine to the clutch. Both the flywheels are integrally provided with projections to prevent excessive compression and expansion of arc spring and so prevent the springs from being getting damaged. After doing all these mounting in the set up test and trials are taken by attaching the loads with the help of rope pulley and speed of engine is measured with tachometer. With the help of speed performance graphs are drawn. Hence comparison of this spring mass loaded DMF is done with conventional flywheel.

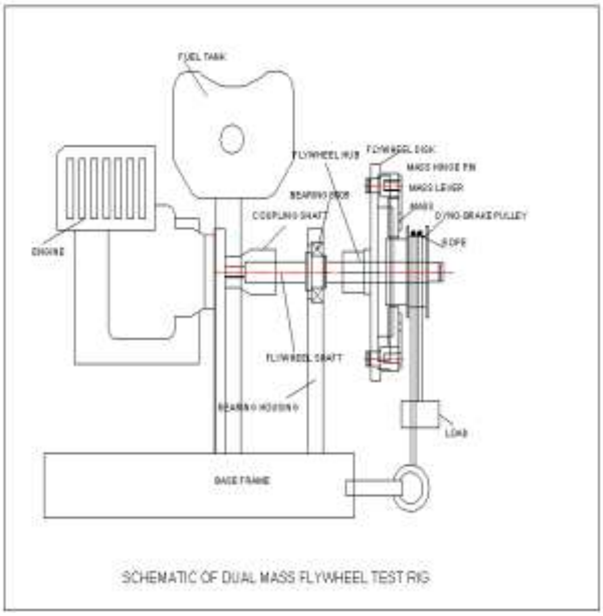

Fig.5.Schematic of Testing set up of spring mass DMF

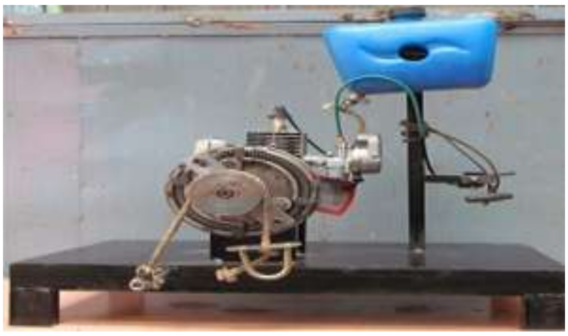

Fig.6.Complete Testing Unit

A. *Prime Mover Selection*

For this study a Crompton Greaves two stroke Spark ignition engine is selected. Other specifications of the engine are as follows:

Make : Crompton Greaves
Model : IK-35
Bore /diameter: 35 mm
Stroke : 35 mm
Capacity : 34 cc
Power output : 1.2 BHP at 5500 rpm
Torque : 1.36 N-m @ 5000 rpm
Dry weight : 4.3 kg
Ignition : Electronic Ignition
Direction of rotation: Clockwise looking from driving end
Carburettor :'B' type
Cooling : Air Cooled engine

The design of various components of set up is done as elaborated in following sections. It consists of engine shaft, coupling shaft, flywheel shaft, bearing, clutch system and flange/ mass lever.

B. *Design of engine shaft.*

The material designation and its mechanical properties are listed in Table 1 below

The design calculations for shaft are done as per the methods in ASME CODE.

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.

Table.1.Material properties

Designation	**Ultimate tensile strength N/mm2**	**Yield strength N/mm2**
EN 24	800	680

According to ASME code permissible values of shear stress may be calculated from various relations.

$$fs_{max} = 0.18\, f_{ult}$$
$$= 0.18 \times 800$$
$$= 144\ N/mm^2$$

OR

$$fs_{max} = 0.3\, f_{yt}$$
$$= 0.3 \times 680$$
$$= 204\ N/mm^2$$

Where,

fs_m : Maximum shear stress;
f : Ultimate Tensile Strength
: Yield Strength

Considering minimum of the above values;

$$fs_{max} = 144\ N/mm$$

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

$$fs_{all} = 108\ N/mm$$

Where,

fs = Allowable shear stress

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

$$T_d = 1.36x10^3\ Nm$$

Where,

= Design Torque

Check for torsional shear failure of engine 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

$$d = 6.8\ m$$

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$$T_d = \frac{\pi}{16} fs_{act} d^3$$

$$fs_{act} = \frac{16\,T_d}{\pi\,d^3}$$

$= 22$ N/mm^2

Where,

fs_i = Actual torsional shear stress

As, $fs_i < fs$;

Engine shaft is safe under torsional load

C. *Design of coupling shaft*

The material designation and its mechanical properties are listed in Table 1. The design calculations are as per ASME CODE. The construction of coupling shaft is as shown in Fig. 7.

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 from various relations.

$fs_{max} = 0.18\, f_{ult}$

$= 0.18 \times 800$

$= 144$ N/mm^2

OR

$fs_{max} = 0.3\, f_{yt}$

$= 0.3 \times 680$

$= 204$ N/mm^2

Considering minimum of the above values;

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

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

$fs_{all} = 108\ \ N/m$

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

$T_d = 1.36x10^3\, Nm$

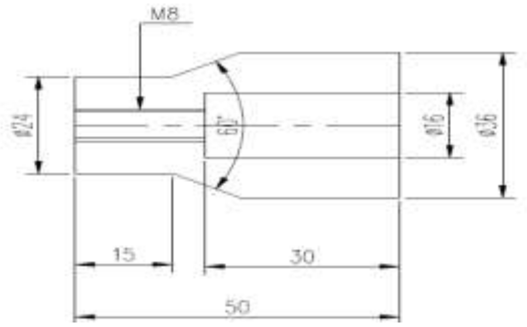

Fig.7.Coupling shaft

Check for torsional shear failure of coupling shaft.

Coupling haft is provided with M8 x1.2 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

Inner diameter $D_i = 16\ m$

Outer diameter $D_o = 36\ m$

Check for torsional shear failure:

$$T_d = \frac{\pi}{16} fs_{act} \frac{D_o^4 - D_i^4}{D_o}$$

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

As $fs_i < fs$

Coupling shaft is safe under torsional load.

D. *Design of Arc Spring*

As loading is continuous so that severe service carbon steel spring with allowable shear stress of 420 Mpa and modulus of rigidity of 80 KN/mm^2 is used.

Outer diameter of spring $(D_o) = 12\ m$

Wire diameter of spring $(d) = 1.6\ m$

Mean diameter of spring $= 12 - 1.6 = 10.4\ m$

Spring coefficient $(C) = D_0/d$

$= 7.5$

Neglecting effect of curvature;

Shear stress factor $(K_s) = 1 + 1/2C$

$(K_s) = 1.066$

Maximum shear stress induced in wire

$$(\tau) = K_s \times \frac{8\,W}{\pi}$$

Where

W axial load

$420 = 1.066 \times \frac{8 \times W}{\pi \times 1.}$

$W = 52.7$

Deflection per active turn

$$\left(\frac{\delta}{n}\right) = \frac{8\,W D_0^3}{G d^4}$$

$$\left(\frac{\delta}{n}\right) = \frac{8 \times 52.77 \times 12^3}{80 \times 10^3 \times 1.6^4}$$

$= 1.39$mm

E. *Design of flywheel shaft.*

The material designation and its mechanical properties are listed in Table 1. The design calculations for shaft are as per ASME CODE. The construction of flywheel shaft is as shown in Fig. 8.

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 from various relations.

$fs_{max} = 0.18\, f_{ult}$

$= 0.18 \times 800$

$= 144$ N/mm^2

OR

$fs_{max} = 0.3\, f_{yt}$

$= 0.3 \times 680$

$= 204$ N/mm^2

Considering minimum of the above values;

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

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

$fs_{all} = 108\ \ N/m$

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

$T_d = 1.36x10^3\, Nm$

Check for torsional shear failure of flywheel shaft.

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

$$d = 14mm$$

$$T_d = \frac{\pi}{16} fs_{act} d^3$$

$$fs_{act} = \frac{16\ T_d}{\pi\ d^3}$$

= 2.52 N/mm^2

Where,

fs_t = Actual torsional shear stress

As, $fs_t < fs$;

Flywheel shaft is safe under torsional load

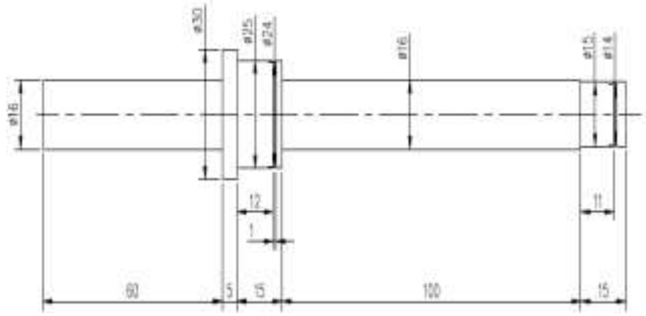

Fig.8.Flywheel shaft

F. *Selection of Bearing on Flywheel shaft*

Input shaft bearing will be subjected to purely medium radial loads; hence we are Selecting Single row deep groove ball bearing from manufacturer's catalogue.

Dynamic equivalent radial load (W) for radial bearing under combined constant radial load (W_R) and constant axial load (W_A) are

$$W = X\,W_R + Y W_A$$

Where

X = Radial load factor

Y = Axial load factor

Neglecting self weight of carrier and gear assembly

For our application $W_A =$

$W = X$ I

Where

$$W_R = Pt = Maximum\ load\ at\ dyno - brake\ pull$$

$$Maximum\ load = \frac{Torque}{Radius\ of\ dyno - brake\ pull}$$

$$= \frac{1.36\ x 10^3}{30}$$

$$= 45$$

$$Max\ radial\ load = W_R = 45\,N.(Tension\ in\ be$$

$$W = P = 45N$$

Calculation of dynamic load capacity of bearing

$$L = (\frac{C}{P}$$

Where, $p = 3\ for\ ball\ bearin$

$L = Rating\ li$

$C = Basic\ dynamic\ load\ ratin$

$P = Equivalent\ dynamic\ lo$

For agriculture 2 stroke engine working life of bearing is $L_H = 4000 - 8000$ engine is used for eight hr of service per day.

$$L = \frac{60\ n}{10}$$

Where,

n = speed in rpm

= Working hrs

$$L = \frac{60 \times 5000 \times 4}{10^6}$$

L =1200 rev

Now;

1200 = c^3/45^3

$$C = 478$$

As the required dynamic capacity of bearing is less than the rated dynamic capacity of bearing;

Bearing is safe

G. *Design of mass lever*

The construction of mass lever is shown in Fig. 9. The material designation and its mechanical properties are listed in Table 2 below

Table.2.Material Specification

Designation	Ultimate Tensile Strength N/mm^2	Yield 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 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;

$$t = thickness\ of\ lever = 2m$$

$$b = width\ of\ lever = 16mm$$

$$Bending\ Moment(M) = PL$$

Where,

Maximum effort applied by hand (*P*) = *98.5 N*

Length of lever (*L*) = *35mm*

$$Bending\ Moment(M) = 3447.5Nmm$$

$$Section\ modulus\ (Z) = \frac{1}{6} tB^2$$

$$Section\ modulus\ (Z) = 85.33\ mm^3$$

$$Bending\ Stress\ (\sigma_b) = \frac{M}{Z}$$

$$Bending\ Stress\ (\sigma_b) = 40.4N/mm^2$$

As $fs_t < fs$

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

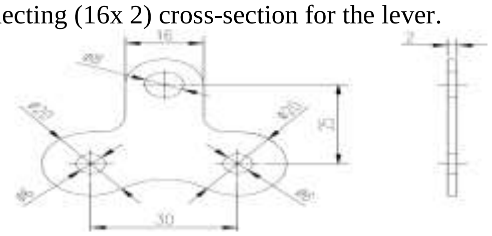

Fig.9.Mass lever

H. *Design of clutch rib plate*

The material designation and its mechanical properties are listed in Table 2.

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

Let;

$$t = thickness\ of\ plate = 3m$$

$$b = width\ of\ plate = 30mm$$

$$Bending\ Moment(M) = PL$$

Where,

Maximum effort applied by hand (*P*) = *98.5 N*

Length of lever *(L) = 80 mm*

$$Bending\ Moment(M) = 7880\ Nmm$$

$$Section\ modulus\ (Z) = \frac{1}{6}tB^2$$

$$Section\ modulus\ (Z) = 450\ mm^3$$

$$Bending\ Stress\,(\sigma_b) = \frac{M}{Z}$$

$$Bending\ Stress\,(\sigma_b) = 17.5\ N/mm^2$$

As $fs_i < fs$

Thus selecting (30x3) cross-section for the rib plate.

I. *Design of unidirectional clutch*

One way SKF clutch is selected for the present study. One way clutch is of the same dimensions of ball bearing 6202, it will be subjected to purely medium radial loads;

Dynamic equivalent radial load (W) for clutch under combined constant radial load (W_R) and constant axial load (W_A) are

$$W = X\,W_R + Y\,W_A$$

Where

X = Radial load factor

Y = Axial load factor

Neglecting self weight of carrier and gear assembly

For our application $W_A =$

$W = P = X\,I$

As $X =$

$W = P = W_R$

$P = Maximum\ radial\ load = 98.5$

Calculation of dynamic load capacity of clutch

$$L = (\frac{C}{P}$$

Where, $p = 3\ for\ ball\ bearin$

$L = Rating\ li$

$C = Basic\ dynamic\ load\ rati$

$P = Equivalent\ dynamic\ lo$

$$L = \frac{60\ n}{10}$$

Where,

n = speed in rpm

= Working hrs = 4000-8000hr

$$L = \frac{60 \times 5000 \times 4}{10^6}$$

L =1200 rev

Now; $1200 = \frac{C}{98}$

$$C = 1046.7$$

As the required dynamic capacity of clutch is less than the rated dynamic capacity of clutch.

J. *Design of clutch housing*

The material designation and its mechanical properties are listed in Table1.Clutch housing can be considered to be a hollow shaft subjected to torsional load.

$$fs_{all} = 108\ N/m$$

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

$$T_d = 1.36x10^3\,Nm$$

Check for torsional shear failure of clutch housing:

Inner diameter $D_i = 30\ m$

Outer diameter $D_o = 54\ m$

$$T_d = \frac{\pi}{16} fs_{act} \frac{D_o^4 -}{D}$$

$$fs_{act} = 0.06\ N/m$$

As $fs_i < fs$

Clutch plate is safe under torsional load.

K. *Design of output shaft.*

The material designation and its mechanical properties are listed in Table 1.

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 from various relations.

$$fs_{max} = 0.18\,f_{ult}$$

= 0.18 x 800

= 144 N/mm^2

OR

$$fs_{max} = 0.3\,f_{yt}$$

=0.3 x 680

=204 N/mm^2

Considering minimum of the above values;

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

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

$$fs_{all} = 108\ N/m$$

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

$$T_d = 1.7x10^3\,Nm$$

Check for torsional shear failure of output shaft.

$$T_d = \frac{\pi}{16} fs_{act}$$

$$fs_{act} = \frac{16}{\pi}$$

$$fs_{act} = \frac{16 \times 1.7 \times}{\pi \times 16^3}$$

$= 2.11$ N/mm^2

As, fs_t < fs;

Output shaft is safe under torsional load

As Mass lever, Mass hinge pin, Flywheel base and shaft are critical components, their design is validated with the help of ANSYS.

4. RESULT AND DISCUSSION

Engine speed for conventional and DMF checked at various loads. These speeds are noted at various loading and unloading conditions. At the same load the average of speed at loading and unloading condition are calculated.Table.3.shows the observations of conventional flywheel whereas table.5.shows the observations of DMF. Other engine output parameters such as output torque, output power, efficiency. The sample calculation for other engine parameters is also explained. The calculated parameters for conventional and DMF are tabulated in the Table.5 and Table.6 respectively. Also flywheel effectiveness of DMF in comparison with conventional flywheel is checked.

Table.3.Observation Table of conventional flywheel

Sr. No.	LOADING		UNLOADING		AVERAGE
	Load (gm)	Speed (rpm)	Load (gm)	Speed (rpm)	Speed (rpm)
1	1500	1310	1500	1320	1315
2	2000	1270	2000	1280	1275
3	2500	1240	2500	1250	1245
4	3000	1210	3000	1200	1205
5	3500	1180	3500	1190	1185
6	4000	1150	4000	1160	1155
7	4500	1100	4500	1110	1105
8	5000	1070	5000	1080	1075

Table.4.Observation Table of dual mass flywheel

Sr. No.	LOADING		UNLOADING		AVERAGE
	Load (gm)	Speed (rpm)	Load (gm)	Speed (rpm)	Speed (rpm)
1	1500	1420	1500	1430	1425
2	2000	1390	2000	1400	1395
3	2500	1360	2500	1370	1365
4	3000	1310	3000	1320	1315
5	3500	1280	3500	1290	1285
6	4000	1240	4000	1250	1245
7	4500	1075	4500	1085	1080
8	5000	925	5000	935	930

Sample calculations:-

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

Output Torque = 4 x9.81 x 0.032 =1.2556 N-m

b) Output power = 2 π N Top / 60

Output Power =2 π x 1155 x 1.2556 /60 = 151.83 W

c) Efficiency = (Output power/ Input power) x 100

= (151.83 /205)

= 74.33

Table.5.Result Table of conventional flywheel

Sr No	Load	Speed	Torque	Power	Efficiency
1	1500	1315	0.47088	64.843	31.63
2	2000	1275	0.6278	83.83	40.89
3	2500	1245	0.7848	102.33	49.91
4	3000	1205	0.9417	118.80	57.95
5	3500	1185	1.0987	136.31	66.49
6	4000	1155	1.2556	151.83	74.06
7	4500	1110	1.41264	164.17	80.08
8	5000	1075	1.5695	176.62	86.15

Table.6. Result Table of dual mass flywheel

Sr No	Load	Speed	Torque	Power	Efficiency
1	1500	1425	0.47088	70.267	34.27
2	2000	1395	0.6278	91.692	44.72
3	2500	1365	0.7848	112.16	54.71
4	3000	1315	0.9417	129.65	63.24
5	3500	1285	1.0987	147.81	72.10
6	4000	1245	1.2556	163.66	79.83
7	4500	1080	1.41264	159.76	77.93
8	5000	930	1.56	151.89	74.09

The engine output torque of both conventional and DMF are plotted against the average engine speed. As plotted in Fig.10. It is observed that output torque of conventional and DMF are same for different loading conditions as they are tested at same loads.

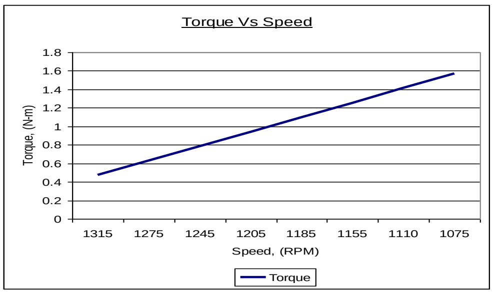

Fig.10.Graph of Torque Vs Speed for conventional and DMF flywheel

The comparison of Power output and Efficiency of conventional and DMF is done as shown in Fig. 11 and Fig. 12. It is observed that there is approximately 7 to 8 % increase in power output of DMF is compared to conventional flywheel. Also 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.

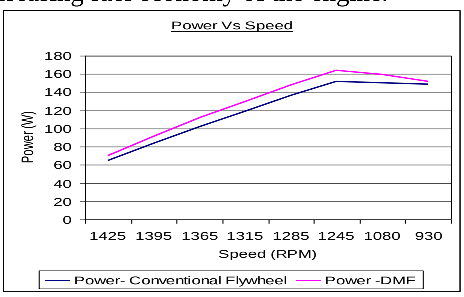

Fig.11. Comparison of Power output of Conventional and Dual mass flywheel

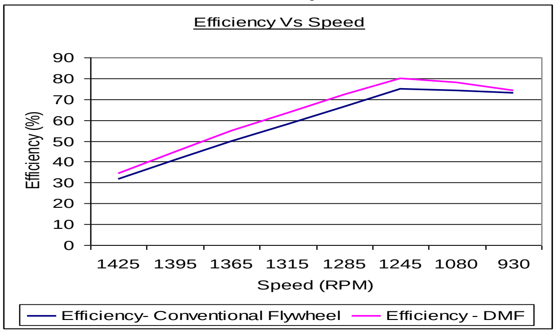

Fig.12. Comparison of Efficiency of Conventional and Dual mass flywheel

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

Maximum fluctuation of energy of Dual mass flywheel

$$\Delta E_{dmf} = mR^2\omega_{dmf}^2 C_s$$

Where,

ΔE_{di}= Maximum fluctuation of energy of DMF

m = mass of flywheel = 1.9

R = Mean Radius of rim = 68 mm = 0.0

ω_{di}= mean angular speed of dual mass Flywheel

$$\omega_{dmf} = \frac{2\pi\ (N_1 +}{2}$$

$$= \frac{2\pi\ (1430 + 930)}{2}$$

$$= 7414\ rad/s$$

= Coefficient of fluctuation of Speed

$$C_s = \frac{(N_1 - N_2)}{N}$$

Where

$$N = \frac{(N_1 + N_2)}{2}$$

= 1180

$$C_s = \frac{(1430\ - 930)}{1180}$$

$$C_s\ = 0.423$$

$$\Delta E_{dmf} = mR^2\omega_{dmf}^2$$

$$= 1.9 \times 0.0682^2 \times 7414^2 \times 0.4$$

$$= 204.27$$

Maximum fluctuation of energy of Conventional flywheel

$$\Delta E_{cnv} = mR^2\omega_{cnv}^2 C_s$$

Where,

ΔE_{cnv} = Maximum fluctuation of energy of conventional flywh

m = mass of flywheel = 1.9

R = Mean Radius of rim = 68 mm = 0.0

ω_{cnv} = mean angular speed of convention Flywheel

$$\omega_{cnv} = \frac{2\pi\ (N_1 +}{2}$$

$$= \frac{2\pi\ (1315 + 910)}{2}$$

$$= 6990\ rad/s$$

C_s = Coefficient of fluctuation of Spe

$$C_s = \frac{(N_1 - N_2)}{N}$$

Where

$$N = \frac{(N_1 + N_2)}{2}$$

$$= 1112\ r$$

$$C_s = \frac{(1315\ - 910)}{1112}$$

$$C_s\ = 0.364$$

$$\Delta E_{cnv} = mR^2\omega_{cnv}^2 C_s$$

$$= 1.9 \times 0.0682^2 \times 6990^2 \times 0.3$$

$$= 156.25\ KJ$$

$$\boldsymbol{Effectiveness}\ (\varepsilon) = \frac{\boldsymbol{\Delta E_{dmf}}}{\boldsymbol{\Delta E_{cnv}}}$$

$$= 1.3$$

Thus the Dual mass flywheel is 1.3 times effective than the Conventional flywheel

CONCLUSION

Use of Dual mass flywheel improves flywheel effectiveness and in turn improves Engine performance characteristics such as speed, torque, power and efficiency. Thus a vehicle loaded with this advanced DMF, offer increased fuel economy.

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