DESIGN OF NOVEL LOAD TEST SIMULATOR FOR PRIME MOVERS

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Abstract— It is essential for a driver and sometimes also an observer to drive the vehicle on a road through various starting, acceleration, speed, pulling, deceleration and braking tests in order to determine the operating condition and mechanical fitness of a vehicular semi-automatic, change gear transmissions,. This technique of transmission testing often does well where the malfunction is manifest, but there are drawbacks in that more slight mechanical or hydraulic defects cannot be precisely determined. Seldom does a mechanic interconnect test instruments such as an engine tachometer, a hydraulic oil pressure gauge, or a manifold vacuum gauge to the automobile prior to an actual road test, yet it is precisely these instruments, which must be used to ascertain specific defective components in the transmission. Even if the test instruments were interconnected to the automobile, a second mechanic must accompany the driver-mechanic to observe the various gauges and meters during the driving test. Lastly, this method does not produce a permanent graphical record of the measured variables in which each variables is time correlated. In short, a graphical record which depicts the relationship between the dynamic variables affecting transmission performance will often reveal much more than separate observations of each variable, which are impossible to correlate.

Design of a functional load simulator, which overcomes all demerits of conventional existing testing technique, is taken up in this project. The proposed simulator is stationary with which a road test can be accurately simulated. This facility consists of an anchor for holding the automobile stationary and with the traction wheel in engagement with a roller coupled to a rotary – linear conversion mechanism. After arriving at the configuration all the subsystems will be designed. Design adequacy will be confirmed by performing structural analysis using Finite Element Method (FEM).

Keywords— Modeling, Designing, analysis, simulation

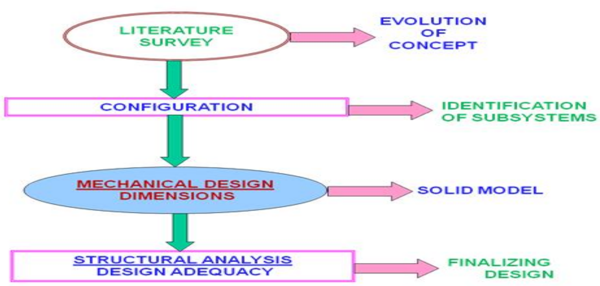
INTRODUCTION

PROBLEM DEFINITION:

It is essential for a driver and sometimes also an observer to drive the vehicle on a road through various starting, acceleration, speed, pulling, deceleration and braking tests in order to determine the operating condition and mechanical fitness of a vehicular semi-automatic, change gear transmissions,. This technique of transmission testing often does well where the malfunction is manifest, but there are drawbacks in that more slight mechanical or hydraulic defects cannot be precisely determined. Seldom does a mechanic interconnect test instruments such as an engine tachometer, a hydraulic oil pressure gauge, or a manifold vacuum gauge to the automobile prior to an actual road test, yet it is precisely these instruments, which must be used to ascertain specific defective components in the transmission. Even if the test instruments were interconnected to the automobile, a second mechanic must accompany the driver-mechanic to observe the various gauges and meters during the driving test. Lastly, this method does not produce a permanent graphical record of the measured variables in which each variables is time correlated. In short, a graphical record which depicts the relationship between the dynamic variables affecting transmission performance will often reveal much more than separate observations of each variable, which are impossible to correlate. Design of a functional load simulator, which overcomes all demerits of conventional existing testing technique, is taken up in this project. The proposed simulator is stationary with which a road test can be accurately simulated. This facility consists of an anchor for holding the automobile stationary and with the traction wheel in engagement with a roller coupled to a rotary – linear conversion mechanism. After arriving at the configuration all the subsystems will be designed. Design adequacy will be confirmed by performing structural analysis using Finite Element Method (FEM).

PROJECT EXECUTION:

It is planned to execute the project in three stages as shown below



THEORY:

PRIME MOVER TESTING PARAMETERS:

Prime mover needs test

- To find out performance before mass production and fitting it into a vehicle.
- To improve the design and configuration, to integrate new materials and technology
- Historically, the test basically was to find out the power and fuel consumption, also to test effectiveness of cooling, vibration and noise, lubrication, controllability, etc.
- Modern regulations force prime movers to reduce harmful emission and comply stringent regulations, therefore, test is getting more and more sophisticated.

The basic measurements to be undertaken to evaluate the performance of an prime mover on almost all tests are the following:

(a) Speed
(b) Fuel consumption
(c) Air consumption
(d) Smoke density
(e) Brake horse-power
(f) Indicated horse power and friction horse power
(g) Heat going to cooling water
(h) Heat going to exhaust
(i) Exhaust gas analysis.

In addition to above a large number of other measurements may be necessary depending upon the aim of the test.

Measurement of Speed:

One of the basic measurements is that of speed. A wide variety of speed measuring devices are available in the market. They range from a mechanical tachometer to digital and triggered electrical tachometers.

The best method of measuring speed is to count the number of revolutions in a given time. This gives an accurate measurement of speed. Many prime movers are fitted with such revolution counters. A mechanical tachometer or an electrical tachometer can also be used for measuring the speed. The electrical tachometer has a three-phase permanent-magnet alternator to which a voltmeter is attached. The output of the alternator is a linear function of the speed and is directly indicated on the voltmeter dial. Both electrical and mechanical types of tachometers are affected by the temperature variations and are not very accurate. For accurate and continuous measurement of speed a magnetic pick-up placed near a toothed wheel coupled to the prime mover shaft can be used. The magnetic pick-up will produce a pulse for every revolution and a pulse counter will accurately measure the speed.

FUEL CONSUMPTION MEASUREMENT:

Fuel consumption is measured in two ways :

- The fuel consumption of an prime mover is measured by determining the volume flow in a given time interval and multiplying it by the specific gravity of the fuel which should be measured occasionally to get an accurate value.
- Another method is to measure the time required for consumption of a given mass of fuel.

Accurate measurement of fuel consumption is very important in prime mover testing work. As already mentioned two basic types of fuel measurement methods are :

• Volumetric type

• Gravimetric type.

Volumetric type flow meter includes Burette method, Automatic Burrette flow meter and Turbine flow meter. Gravimetric Fuel Flow Measurement The efficiency of an prime mover is related to the kilograms of fuel which are consumed and not the number of liters. The method of measuring volume flow and then correcting it for specific gravity variations is quite inconvenient and inherently limited in accuracy. Instead if the weight of the fuel consumed is directly measured a great improvement in accuracy and cost can be obtained. There are three types of gravimetric type systems which are commercially available include Actual weighing of fuel consumed, Four Orifice Flowmeter, etc.

Measurement of Air Consumption:

One can say the mixture of air and fuel is the food for an prime mover. For finding out the performance of the prime mover accurate measurement of both is essential. In IC prime movers, the satisfactory measurement of air consumption is quite difficult because the flow is pulsating, due to the cyclic nature of the prime mover and because the air a compressible fluid. Therefore, the simple method of using an orifice in the induction pipe is not satisfactory since the reading will be pulsating and unreliable. All kinetic flow-inferring systems such as nozzles, orifices and venturies have a square law relationship between flow rate and differential pressure which gives rise to severe errors on unsteady flow. Pulsation produced errors are roughly inversely proportional to the pressure across the orifice for a given set of flow conditions. The various methods and meters used for air flow measurement include

(a) Air box method, and

(b) Viscous-flow air meter.

BEAMS:

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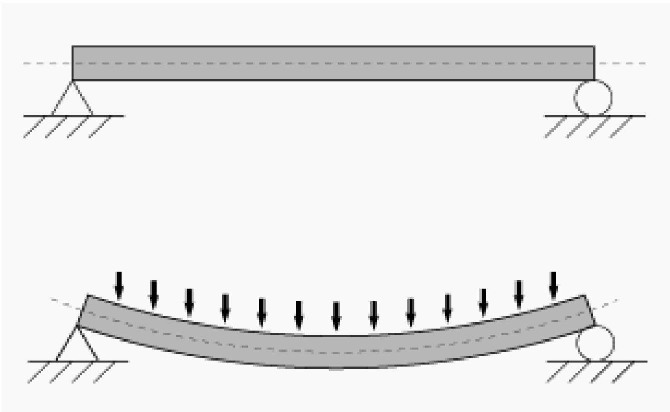
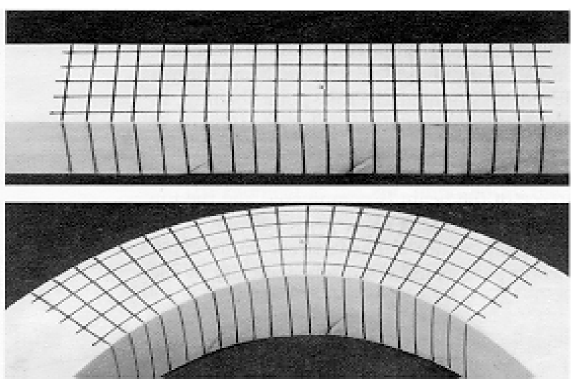
As most of the loads bearing members of the proposed design are beams, salient features of beams are discussed in this section.



BENDING OF BEAMS:



BEAMS:

I section beam

I-beams (also known as W-beams, for "wide flange", RSJ, or double-T esp. in Polish and German) are beams with an I- or H-shaped cross-section. The horizontal elements are flanges, while the vertical element is the web. The Euler-Bernoulli beam equation shows that this is a very efficient form for carrying both bending and shear in the plane of the web. On the other hand, the cross-section has a reduced capacity in the transverse direction, and is also inefficient in carrying torsion, for which hollow structural sections are often preferred [4].

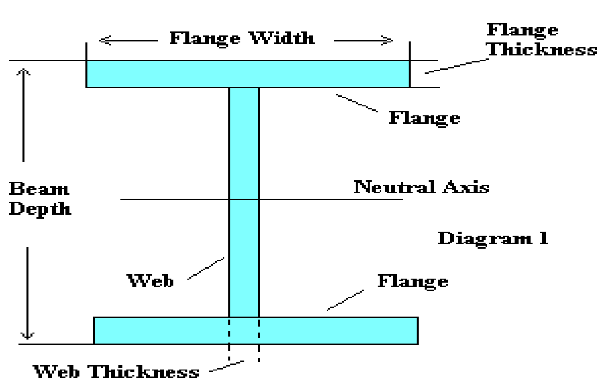
A steel I-beam is a type of joist or girder made from structural steel. I-beams are used as major support trusses in building, to ensure that a structure will be physically sound. Steel is one of the most common materials used to make I-beams, since it can withstand very heavy loads. Other materials such as aluminum are sometimes used to make I-beams, depending on their intended use. Composite I-beams are also available, with layers of other materials encasing the outside of the steel I-beam to disguise it as something else, such as wood.

The shape of a steel I-beam strongly resembles a capital “I” in cross section, which explains the name. The steel I-beam has a strong central core capped with flanges on either side. Various lengths of beam are available to suit construction project needs, and each steel I-beam also carries a rating, indicating how large it is, and how much weight it is able to bear. When engineers are designing a structure, they determine what the load limits of the I-beams used in the structure should be.

There are advantages and disadvantages to using steel I-beams in construction. They are much less likely to bend or warp than wood, allowing builders to use steel I-beams to create large open spaces which would not be possible with ordinary wooden beams. A steel I-beam also does not need to be as large as a wooden beam bearing the same load, so the support beams in a structure do not need to be so obtrusive. However, a steel I-beam can fail catastrophically if exposed to heat, making it necessary to insulate the beam for safety.

Several fabrication techniques are used to create steel I-beams. Many are rolled or extruded, processed on metalworking machinery which creates standardized beams very rapidly. Other steel I-beams, sometimes called plate girders, are made by riveting or welding together sections of steel plate. However the steel I-beam is manufactured, it will be extremely heavy, requiring strong workers and specialized equipment to be handled.

When a house is renovated, steel I-beams may be used to replace old structural supports. When removing structural supports, it is crucial to know how much weight they are supporting, and how they have been installed. For this reason, it is highly recommended that homeowners use professionally certified individuals for construction work which may involve removing such supports, to ensure that they do not compromise the structural integrity of the home.



STRUCTURAL ANALYSIS:

Structural analysis comprises the set of physical laws and mathematics required to study and predicts the behavior of structures. The subjects of structural analysis are engineering artifacts whose integrity is judged largely based upon their ability to withstand loads; they commonly include buildings, bridges, aircraft, and ships. Structural analysis incorporates the fields of mechanics and dynamics as well as the many failure theories. In practice, structural analysis can be viewed more abstractly as a method to drive the engineering design process or prove the soundness of a design without a dependence on directly testing it.

The following method will be in general used for carrying out structural analysis.

- Analytical methods
- Classical methods
- Elasticity methods
- Finite Element Method (FEM)

STRUCTURAL DESIGN OF NOVEL LOAD TEST SIMULATOR FOR PRIME MOVERS

INTRODUCTION: Based on the design philosophy a configuration is identified. All the subsystems of the configuration are to be designed taking functional loads into account. While designing the subsystems various mechanical design aspects are considered. The outcome of the structural design would be solid models of all subsystems of the intended system. The total design process is concluded with mention of details for the design, which will be ascertained further by finite element method details of which would be mentioned in later chapters. This chapter brings out the details of the design procedures adopted, solid models of all subsystems.

DESIGN SPECIFICATIONS:

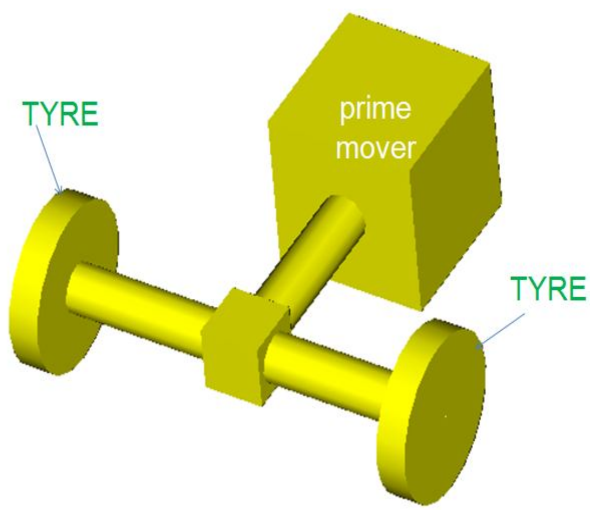
- Maximum speed of vehicle = 150 KMPH
- Time taken to attain maximum speed = 10 sec
- Time taken to decelerate = 20 sec

- Time taken to come to stand still = 3 sec (Braking)

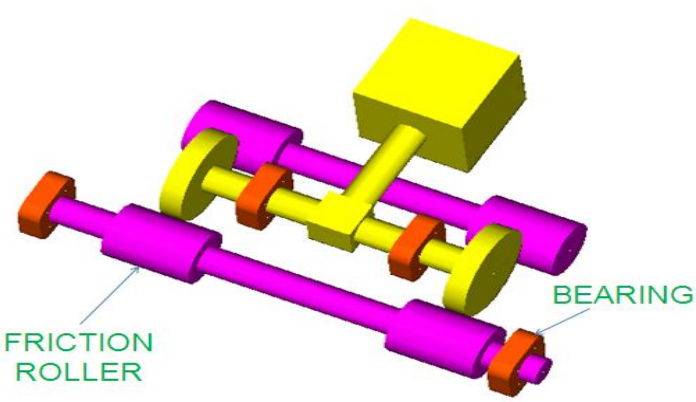
DESIGN PHILOSOPHY:

As it was mentioned in the previous chapter, all these test facilities presently being used are meant for static condition. Moreover realistic loading situation is not considered.

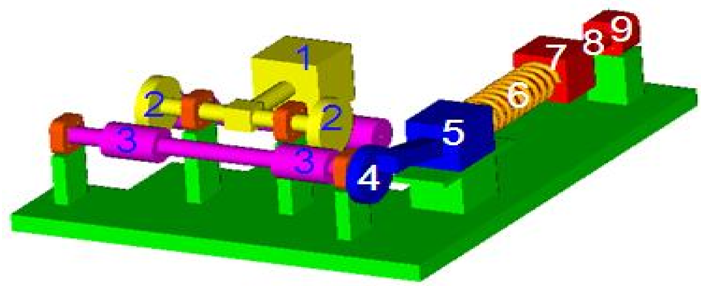
Hence it is planned to design a test facility which can simulate various loads on prime mover by adjusting the stiffness of spring. Construction of proposed design of test facility is shown in various stages. Orientation of prime mover along with tyres is shown in figure



Further friction rollers are added as shown in figure



PRINCIPLE OF WORKING:



1: Prime mover start

2: Tires will rotate

3: Rollers will rotate

4: Rotary - Linear motion (5)

6: Compression & tension of spring with Free end (5)

7: Nut (Fixed end for spring)

8: Lead screw

9: Motor

- By running the motor (9) lead screw (8) will be rotated
- Rotation of lead screw gets converted into linear motion of nut (7) towards spring
- With this height of spring will vary which in turn will increase the stiffness of spring
- With increased stiffness prime mover has to put more effort to deflect spring and hence more load acts on prime mover

DESIGN OF COMPONENTS:

The following components are identified for which detailed design is carried out.

- Friction rollers
- Shaft
- Bearings
- Mechanism for conversion of rotary – linear motion
- Spring
- Lead screw
- Motor

DESIGN GOAL:

Minimum desired factor of safety $\geq$ 1.5

DESIGN INPUTS:

Utility of the project is chosen as Innova corresponding to which the following design inputs are considered.

- Power: 235 HP = 172866 W
- Torque = 260 N-m
- Speed = 6349 rpm
- Weight = 1665 Kg
- Tire diameter = 381 mm
- Tire width = 205 mm
- Wheel base = 2750 mm
- Coefficient of friction between tire and road = 0.7
- Surface roughness = 6.3 S

CALCULATION OF FORCES ON PRIME MOVER

$$Acceleration = \frac{\text{Maximum speed in m/sec}}{Time} = \frac{150x\frac{5}{18}}{10} = 4.2 m/\sec^2$$

Acceleration force = mass x acceleration = 6993 N

$$Deceleration = \frac{\text{Maximum speed in m/sec}}{Time} = \frac{150x\frac{5}{18}}{20} = 2.1 m/\sec^2$$

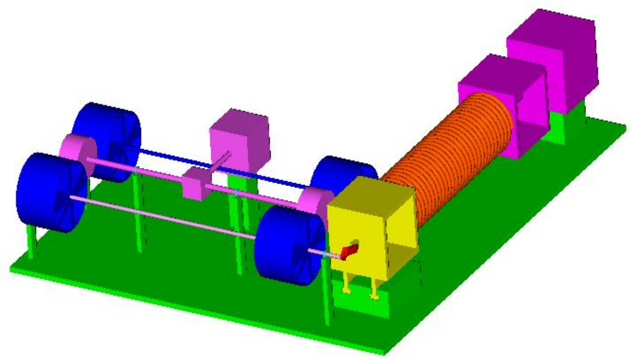
Deceleration force = mass x Deceleration = 3496.5 N

$$Deceleration\ \text{during braking} = \frac{\text{Maximum speed in m/sec}}{Time} = \frac{150x\frac{5}{18}}{3} = 14 m/\sec^2$$

Braking force = mass x Deceleration = 23310 N

SOLID MODELING OF ASSEMBLY:

All the solid models pertaining to the components designed are assembled to get the assembly of the system. Solid model of the assembly is shown in Figure .Figure Solid model of the assembly



STRUCTURAL ANALYSIS USING FINITE ELEMENT METHOD (FEM)

STRUCTURAL ANALYSIS:

Structural analysis is carried out against the functional load i.e. due to weight of the vehicle. Maximum Von Misses stress and displacement are obtained from the analysis. Maximum stress thus obtained is compared with allowable stress and obtained the available factor of safety.

FINITE ELEMENT MODELING:

To begin with geometric model of the test system is built in 3D CAD software from its dimensions evolved as an outcome in the previous chapter.

Then geometric model is converted into FE model by discretizing with elements in commercial FEM software package ANSYS

Sl. No.	Component	Type of element
1.	Roller	Shell 63
2.	Spring	Combin 40
3.	All other subsystems like Shaft, Lead screw, etc.	Beam 4

MATERIAL PROPERTIES:

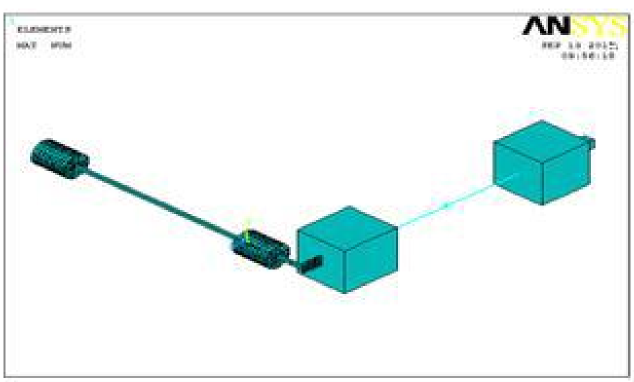
Material	Young's modulus (E)	Poisson's ratio (µ)	Density (ρ)
Spring steel	2.1×10^{11} Pa	0.3	7850 Kg/m^3

BOUNDARY CONDITIONS:

- Node corresponding to bearing support constrained for all DOF except rotation about bearing axis
- Motor end of lead screw constrained for all DOF
- Nodes corresponding to block constrained for all DOF except translation along axis of spring

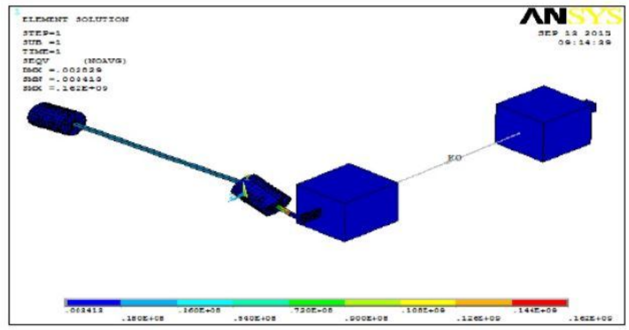
LOAD:

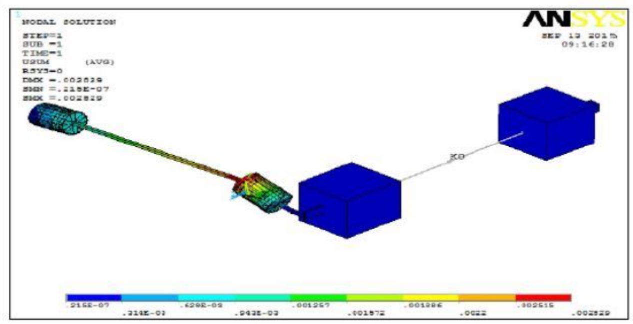
1. STATIC ANALYSIS
Self weight with 1g acceleration
2. STATIC ANALYSIS
Maximum force (Breaking force) of 23310 N on rollers
3. DYNAMIC ANALYSIS
Modal analysis



STATIC ANALYSIS – SELF WEIGHT:

The FE model is then solved for Von Misses stress and displacement using ANSYS software. Maximum stress plot is shown in Figure in which maximum stress location is visible in red color.





OBSERVATIONS:

- Maximum Von Misses stress is observed to be 162 MPa.
- Available factor of safety is observed to be (6.1) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 990 MPa.
- As the available factor of safety (6.1) is more than minimum desired factor of safety (1.5) the design is safe.
- Maximum displacement is observed to be 2.83 mm.

Static analysis – Self weight & Angular velocity:

Same FE model is then solved for Von Misses stress and displacement using ANSYS software. Maximum stress plot is shown in Figure 5.4 in which maximum stress location is visible in red color.

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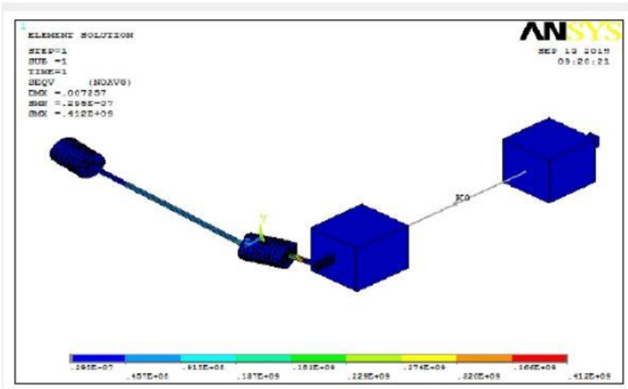

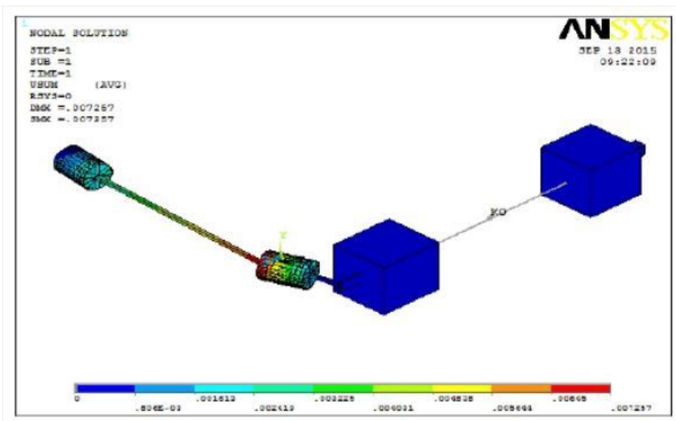

OBSERVATIONS:

- Maximum Von Misses stress is observed to be 412 MPa.
- Available factor of safety is observed to be 2.4 by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 990 MPa.
- As the available factor of safety (2.4) is more than minimum desired factor of safety (1.5) the design is safe.
- Maximum displacement is observed to be 7.2 mm.

DYNAMIC (MODAL) ANALYSIS:

Modal analysis is the study of the dynamic properties of structures under vibration excitation. Modal analysis is the field of measuring and analyzing the dynamic response of structures and or fluids when excited by an input. Examples would include measuring the vibration of a car's body when it is attached to an electromagnetic shaker, or the noise pattern in a room when excited by a loudspeaker.

In structural engineering, modal analysis uses a structure's overall mass and stiffness to find the various periods that it will naturally resonate at. These periods of vibration are very important to note in earthquake engineering, as it is imperative that a building's natural frequency does not match the frequency of expected earthquakes in the region in which the building is to be constructed. If a structure's natural frequency matches an earthquake's frequency, the structure could continue to resonate and experience structural damage.

Although modal analysis is usually carried out by computers, it is possible to hand-calculate the period of vibration of any high-rise building by idealizing it as a fixed-ended cantilever with lumped masses.

A modal analysis calculates the undamped natural modes of a system. These modes are given in decreasing order of period and are numbered starting from 1.

The analysis calculates the natural modes of the discretised model, not those of the real continuous system. However the discretised modes are close to the continuous ones and for a mode number the accuracy improves as more and more elements are used to model the system. For any given level of discretisation the accuracy is better for the lower modes and progressively worsens as you go to higher and higher modes. The highest numbered modes are unlikely to be realistic since they are oscillations whose wavelengths are of the same order as the segment length.

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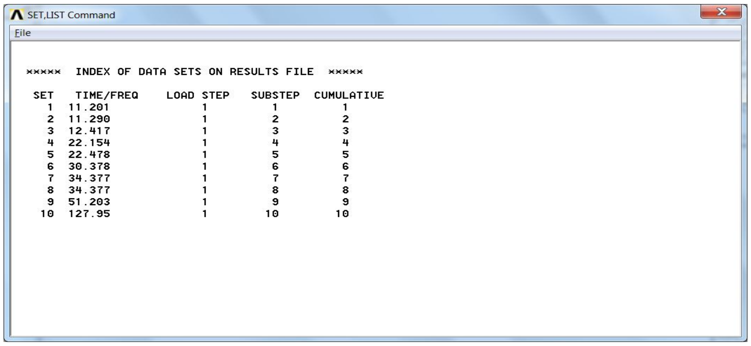
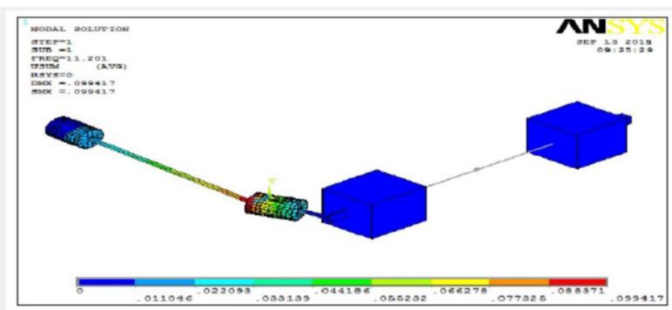
Same FE model used for static analysis is extended for modal analysis. The list of frequencies are given below.


SET,LIST Command

File

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	11.201	1	1	1
2	11.290	1	2	2
3	12.417	1	3	3
4	22.154	1	4	4
5	22.478	1	5	5
6	30.378	1	6	6
7	34.377	1	7	7
8	34.377	1	8	8
9	51.203	1	9	9
10	127.95	1	10	10



OBSERVATIONS:

- Frequency of the intended system corresponding to first bending mode is found to be 11 Hz.
- Test system does not have any frequency at 105.8 Hz which is the frequency associated with operating speed of engine i.e. 6349 rpm).

RESULTS AND DISCUSSION

SUMMARY OF DESIGN CALCULATIONS:

Sl. No	Design Parameter	Value
FORCES		
1.	Acceleration force	6993 N
2.	Deceleration force	3496.5 N
3.	Braking force	23310 N

Sl. No.	Design Parameter	Value
FRICTION ROLLER		
1.	Diameter	616 mm
2.	Width	410 mm
3.	Rim thickness	4.05 mm
4.	Number of arms	6
5.	Size of arm	50 mm
6.	Diameter of hub	70 mm
7.	Length of hub	410 mm

Sl. No.	Design Parameter	Value
SHAFT		
1.	Diameter	35 mm
2.	Length	2135 mm

Sl. No.	Design Parameter	Value
BEARING		
1.	Outer diameter	62 mm
2.	Width	14 mm
3.	Designation	6007

Sl. No	Design Parameter	Value
MECHANISM		
1.	Stroke	200 mm
2.	Length of short link	100 mm
3.	Length of long link	200 mm
4.	Width	70 mm
5.	Thickness	25 mm
6.	Size of pin	10 mm

Sl. No.	Design Parameter	Value
SPRING		
1.	Spring index	10
2.	Shear stress factor	1.05
3.	Wahl's stress factor	1.1448
4.	Wire diameter	51 mm
5.	Mean coil diameter	510 mm
7.	Number of turns	32
8.	Deflection	200 mm
9.	Free length	1.83 m
10.	Buckling ratio	3.5

Sl. No.	Design Parameter	Value
LEAD SCREW		
1.	Diameter	120 mm
2.	Length	1500 mm

Sl. No.	Design Parameter	Value
MOTOR		
1.	Torque	379 N-m
2.	Rpm	504
3.	Power	27 Hp
4.	Type of motor	DC

Sl. No.	Analysis	Maximum Value	Allowable value	Factor of safety
Static – Self weight				
1.	Von Misses stress	162 MPa	990 MPa	6.1
Static – Operational load				
1.	Von Misses stress	412 MPa	990 MPa	2.4
Modal				
1.	First natural frequency	11,12,22,30,34,51,127 Hz	105.8 Hz	--

CONCLUSIONS & RECOMMENDATIONS

CONCLUSIONS:

A test simulator is designed which can simulate various loads on prime mover by adjusting the stiffness of spring.

STATIC ANALYSIS – SELF WEIGHT

- Maximum Von Misses stress is observed to be 162 MPa.
- Available factor of safety is observed to be (6.1) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 990 MPa.
- As the available factor of safety (6.1) is more than minimum desired factor of safety (1.5) the design is safe.
- Maximum displacement is observed to be 2.83 mm.

STATIC ANALYSIS – FUNCTIONAL LOAD

- Maximum Von Misses stress is observed to be 412 MPa.
- Available factor of safety is observed to be 2.4 by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 990 MPa.
- As the available factor of safety (2.4) is more than minimum desired factor of safety (1.5) the design is safe.
- Maximum displacement is observed to be 7.2 mm.

MODAL ANALYSIS

- Frequency of the intended system corresponding to first bending mode is found to be 11 Hz.
- Test system does not have any frequency at 105.8 Hz which is the frequency associated with operating speed of engine i.e. 6349 rpm).

RECOMMENDATIONS:

- It is recommended to implement the realistic testing technique brought out in this project for qualification of prime movers.
- It is further recommended to consider other types of loads that will be experienced by prime movers.

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