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Structural design of track re-creator for functional testing of race cars

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Abstract - Race cars will be sensitive to disturbance originated from race track waviness. This waviness results into two kinds of motions which are rotation about two orthogonal planes. These two motions will occur independently and simultaneously. Ultimately these motions will drive the race car to undergo periodic motion with frequency of excitation varies from 1 to 50 Hz. The frequency of periodic motion depends on the track. This periodic motion will affect the subsystems in the race car like transmission system. Once the race car is designed it needs to be qualification tested for race track environment before it is put in use. To carry out the qualification testing a device which can replicate the motion on race track is required. This project aims at design of such race track motion re-creator for testing the performance of race cars. Unique feature of the proposed design which is not available in the systems which are presently being used is giving the periodic motion with continuous variation of the frequency. The frequency range of this system will be from 1 to 50 HZ (Disturbance range of race track). The intended system will be evolved using all mechanical elements without depending on electronic systems which make it cost effective. This project aims at design of such re-creator which using which performance of the race car can be tested. After evolving the basic configuration, sizing will be done for all subsystems by carrying out design. Then this configuration will be converted into solid model using 3D CAD software tool i.e. SOLIDWORKS and followed by structural analysis using Finite Element Method (FEM) using ANSYS software.

Keywords – race track, frequency, motion, testing, simulation, CAD, FEM analysis.

IINTRODUCTION

The banking angles of the race track applies different types of motions on the race car. It give three types of motions in three axes. Roll, pitch and yaw are the types of motions which act on the car due to banking angles present in the race track. Therefore race cars should be designed in such a way that it withstands all these forces. We generally design a car chassis based on the bending stresses induced in it, but we neglect torsional stresses. By performing this test, we can certify that the car's chassis can also withstand the torsional stresses which act on it during it's performance on the track. This project aims at design of such re-creator which using which performance of the race car can be tested. After evolving the basic configuration, sizing will be done for all subsystems by carrying out design. Then this configuration will be converted into solid model using 3-dimensional computer aided design software tool i.e. solidworks and followed by structural analysis using Finite Element Method using ANSYS software.

IITHEORITICALBACKGROUND

BEAMS

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

A beam which is shown above is a structural element that is capable of withstanding load primarily by resisting bending. The bending force induced into the material of the beam[1] as a result of the external loads, own weight and external reactions to these loads is called a bending moment.

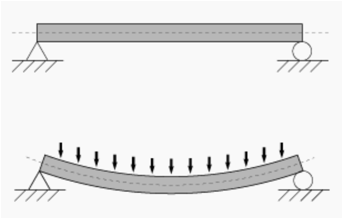


Figure 1 – Loads on Beams

Beams generally carry verticalgravitationalforces but can also be used to carry horizontal loads (i.e., loads due to an earthquake or wind). The loads carried by a beam are transferred to columns, walls, or girders, which then transfer the force to adjacent structural compression members[2]. In Light frame construction the joists rest on the beam.

Beams are characterized by their profile (the shape of their cross-section), their length, and their material. In contemporary construction, beams are typically made of steel, reinforced concrete, or wood. One of the most common types of steel beam is the I-beam or wide-flange beam (also known as a "universal beam" or, for stouter sections, a "universal column").

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[3]. Other materials such as aluminium 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.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.

III DESIGN OF TRACK RE-CREATOR

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.

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

- Inner platform
- Gear
- Shaft for inner platform
- Bearing for inner platform
- Outer platform
- Shaft for outer platform
- Bearing for outer platform

The following design inputs are considered.

- Engine weight = 100 Kg
- Engine dimensions = 400 mm x 320 mm x 433 mm
- Number of engine mounting holes = 6
- Angular rotation on either side = 60^0 = 1.047 rad/sec
- Speed of gear, N = 3000 rpm (Speed corresponding to maximum frequency i.e. 50 Hz)
- Diameter of gear, D = 100 mm
- Desired FOS = 1.5
- Grade 10 finish
- Total Angle of rotation = 120^0 (60^0 on either side)
- Total time for one revolution = 20 seconds
- Moment of inertia (Inner frame assembly calculated from software) = 20 Kg – m^2

Design of inner platform

Engine load basically induces bending stress on inner platform. The bending stress can be expressed as follows.

$$f_b = \frac{M.y}{I}$$

Where
f_b:Bending stress
M: Bending moment
y: Distance from neutral axis
I: Moment of inertia

The above expression indicates that bending stress varies inversely with moment of inertia. Hence cross section having higher moment of inertia should be chosen for inner platform.

I – section is having higher moment of inertia, it is weak in shear and moreover mounting space is restricted. In contrast with this moment of inertia of box section is nearly close to that of I – section and mounting space is more i.e. objects can be mounted on four sides of the section. Hence box section is chosen for inner platform.

To arrive at dimensions of inner platform engine will be the dictating element as it is directly mounted on inner platform whose dimensions are fixed. To accommodate engine comfortably (Having space left free on either side of the engine as 25 mm), box section having dimensions of 50 mm x 50 mm x 5 mm is chosen.

Hence, inner width of inner platform = Width of engine – Width of box section = 320 – 50 = 270 mm

Outer width of inner platform = Width of engine + Width of box section = 320 + 50 = 370 mm

Inner length of inner platform = Length of engine – Length of box section = 400 – 50 = 350 mm

Outer length of inner platform = Length of engine + Length of box section= 400 + 50 = 450 mm

Height of inner platform = Height of box section = 50 mm

Wall thickness = 5 mm

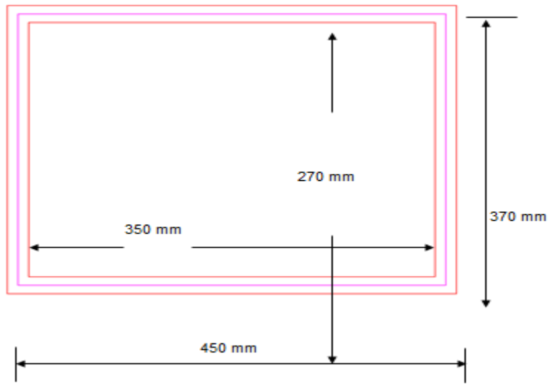


Figure 2 - Dimensional model of inner platform

As discussed earlier inner platform will experience a force which is known as centrifugal force due to high C.G which can be expressed as follows.

$F = m \times r \times \omega^2$

Where
m: Mass of the engine = 100 Kg
r: C.G. offset from inner platform to engine = 0.2415 m
ω: Angular velocity of the motor = 1.047 rad/sec (60^0)

Substituting we get

F = 26.47 N

As this force acts on the platform as uniformly distributed load (UDL), moment due to this force can be expressed as follows

$$M = \frac{F \times l^2}{8}$$

Where
l: Arm length = 0.37 m

Substituting we get

M = 1.22 N-m

Maximum bending stress can also be expressed as follows.

$$f_b = \frac{M.y}{I}$$

Where
f_b:Bending stress
M: Bending moment = 1.22 N-m
y: Distance from neutral axis = Height /2 = 0.025 m
I: Moment of inertia

For the inner platform moment of inertia is calculated to be 6.12×10^{-7} m^4

Substituting we get, maximum bending stress = 22.4 MPa

Factor of safety (FOS) can be expressed as follows

$$FOS = \frac{\text{Allowable (Yield) stress}}{\text{Maximum bending stress}}$$

As steel is considered as material for all subsystems of this design, allowable stress = 200 MPa.

Factor of safety > 10

Design of gear

Torque, T = I x α

Where I: Moment of inertia

$$\text{Angular Acceleration}, \alpha = \frac{\theta}{t^2}$$

Where θ = Angular rotation in radians

$$= \frac{120 \text{ x } \prod}{180} = 2.09 \text{ radians}$$

Angular Acceleration,

$$\alpha = \frac{2.09}{20^2} = 0.0052 \text{ rad/sec}^2$$

T= 20 x 0.0052 = 0.104 Kg – m = 1.02 N-m

Velocity can be expressed as

$$V = \frac{\pi D N}{60}$$

In which N: Speed of gear.

From which, V = 15.7 m/sec
Beam strength of gear tooth can be expressed as

Where

Beam Strength of gear tooth, $S_b = mb\sigma_b y$
m: Module
σ_b: Bending stress = σ_u /3 = 330 MPa
b : Face width
Y: Lewis form factor = 0.34 (For preliminary design stage)

In the design of gears face width will be expressed in terms of module. If the face width is too large, there is a possibility of concentration of load at one end of gear tooth. Small face width result into poor capacity to resist the shock and absorb vibrations. In practice the optimum range of face width is

8m < b < 12m

Where m: Module of gear tooth

In preliminary stage of gear design, the face width is assumed as ten times of module
b = 10m

Beam strength of gear tooth = $1.1 \times 10^9 m^2$

Effective load of gear tooth can be expressed

$$\text{Effective Load}, P_{eff} = \frac{C_s \times P_t}{C_v}$$

as

Where
C_s: Service factor
P_t: Tangential tooth load
C_v: Velocity factor

For precision gears, the following expression is chosen for velocity factor.

$$C_v = \frac{6}{6+V} = 0.276$$

$$\text{Effective Load}, P_{eff} = \frac{C_s \times P_t}{C_v} = 73.9\,\text{N}$$

For the present design desired Factor of safety is specified as 1.5, which can be expressed as

$$FOS = \frac{S_b}{P_{eff}}$$

From which m: Module = 0.317 mm

From the design data bookAll the dimensions of the gear are summarized below.

$Z = 20$

$m = 5\,mm$

$b = 50\,mm$

$D = 100mm$

$Addendum = \text{m} = 5\,mm$

$Dedendum = 1.25xm = 6.25\,mm$

$Clearence = 0.25 \times m = 1.25\,mm$

$\text{Tooth thic kness} = 1.5708\,\text{X m} = 7.854\,\text{mm}$

$\text{Fillet radius} = 0.4\text{ x m} = 2\text{mm}$

Design of shaft for inner platform

The load diagram of the shaft is shown in figure 3.7.

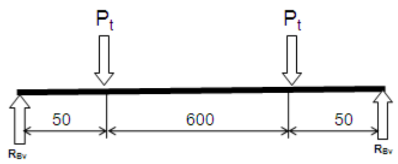


Figure 3 - Load diagram of shaft

Where

P_t: Tangential tooth load of gear = 20.4 N
R_{BV}: Reaction at bearing = 20.4 N
Maximum Bending moment, M = 20.4x650+20.4x50=14280 N-mm
Torque, T = 1.02 N-m = 1020 N-mm

Equivalent Twisting moment,

$$T_e = \sqrt{T^2 + M^2} = 14316\,\text{N-mm}$$

Shear strength of shaft material, $f_s = 140\,\text{N/mm}^2$

$$T_e = \frac{\pi}{16} f_s d^3 \Rightarrow d = 8 = 10\,\text{mm}$$

Design of bearing for inner platform

Type of bearing plays vital role in any dynamic system, which has bearings. Basic classification of bearings is shown below.

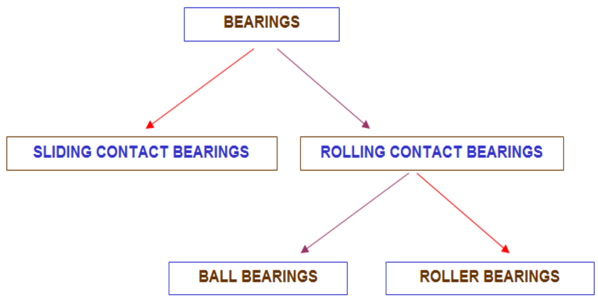


Figure 4 – Types of bearings

Among the above sliding contact bearings will have large starting friction and they are suitable for huge installations. Where asfor the proposed design rolling contact bearings will be suitable because, they are of high precision type and will have very low starting friction.

Total reaction

$$R = \sqrt{R_V^2 + R_H^2} = 20.4\,\text{N}$$

Bearing reactions are in radial direction

Radial load, $F_r = R = 20.4\,\text{N}$

There is no axial thrust on bearing

$\therefore F_a = 0$

Equivalent dynamic load, $P = XF_r + Y\,F_a$

$X = \text{Radial Factor}$

$Y = \text{Thrust Factor}$

When no axial load and purely radial

$X = 1$

$Y = 0$

$\therefore P = 1 \times 20.4 + 0 = 20.4\,\text{N}$

From standards load factor is taken as 1.2

Relation between dynamic load carrying capacity, Eq. dynamic load & bearing life

$$L = \left(\frac{C}{P \text{ X Load factor}} \right)^{p}$$

C = Dynamic load capacity

$$P = 3 \rightarrow \text{Ball}(b)$$
$$= 10/3 \rightarrow \text{Roller}(b)$$

$$\Rightarrow C = L^{\frac{1}{3}} P \text{ X load factor} = P(L)^{\frac{1}{3}} \text{ X } 1.2$$

$$C = 375.18 \text{ N}$$

For 10 mm diameter shaft the selection of bearing can be made from the following table from standards.

d	D	B	C	Designation
10	19	5	1480	61800

where

d: Diameter of shaft = 10 mm

D: Outer diameter of bearing = 19 mm

B: Axial width of bearing = 5 mm

No choice for number of balls

From the above table

1480 > C=375.18

Hence bearing No. 61800 is suitable.

The same procedure is followed for the outer platform and the required dimensions and design parameters are found accordingly.

IV STRUCTURAL ANALYSIS USING FINITE ELEMENT METHOD (FEM)

Structural analysis of track recreator for race cars is carried out using Finite Element Method (FEM) in ANSYS software. Analysis is carried out against the functional load. The objective of the structural analysis is to assess the design adequacy against the functional load and to validate the design calculations. This chapter brings out the details of FE modeling and analysis results.

Structural analysis is carried out against the self weight and functional loads[4] i.e. due to movement of the system. Maximum Von Misses stress is obtained from the analysis. Maximum stress thus obtained is compared with allowable stress and obtained the available factor of safety.

Criteria

Static analysis

- Minimum available factor of safety should be more than the desired factor of safety (1.5).

Modal analysis

- First natural frequency should be above the frequency associated with operation of mechanism i.e. 0.166 Hz

The basic approach followed in finite element analysis is represented in form of flow chart below.

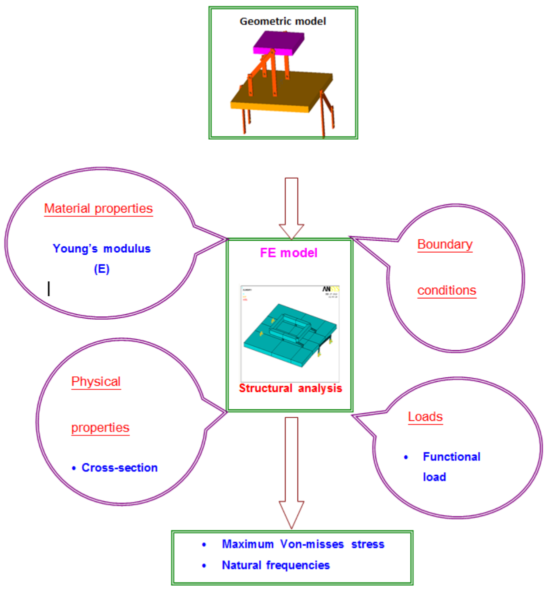


Figure5 – Finite Element Modelling

To begin with geometric model of the recreator is built in 3D CAD software from its dimensions evolved as an outcome in the previous chapter. However load bearing members are only considered for analysis.Then geometric model is converted into FE model by discretizing with elements in commercial FEM software package ANSYS.

Following elements given in table 4.1 were used to discretize various subsystems.

Sl. No.	Component	Type of element
1.	Outer platform	Four nodded quad shell (SHELL63)
2.	Links	
3.	Inner platform	Linear beam (BEAM4)
4.	Engine	MASS21

Material Properties

As all sub systems of the re-creator are made of steel, its material properties are considered for the analysis[5]. Material properties are given in Table.

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

Boundary conditions - Base of links constrained for all DOF in the FE model.
Load - Following loads given in table 4.3 are applied to the FE model.

STATIC ANALYSIS – Load due to self weighti.e. 1g acceleration.

STATIC ANALYSIS – Self weight and angular velocity of 1.047 rad/sec

DYNAMIC ANALYSIS – Modal analysis

FE Model is shown in figure below

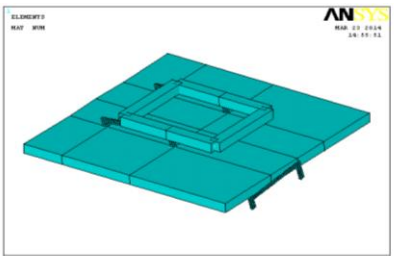


Figure 6 – Finite Element Model

Static analysis – self weight

The FE model is then solved for Von Misses stress using ANSYS software. Maximum stress plot is shown in Figure.

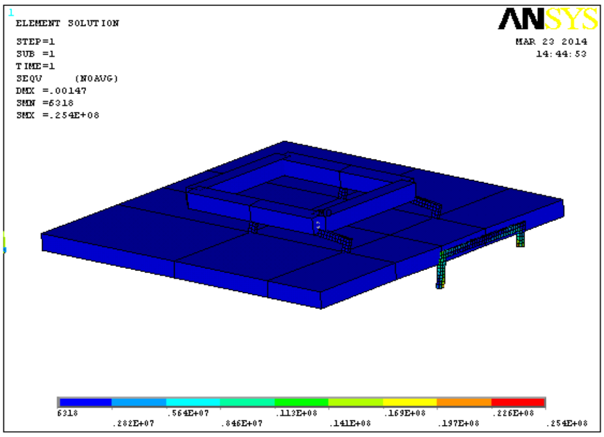


Figure 7 – Stress Plot

Observations - Maximum Von Misses stress is observed to be 25 MPa.
Available factor of safety is observed to be (> 10) by comparing the maximum stress with that of allowable stress (Yield) of steel i.e. 330 MPa.
As the available factor of safety (> 10) is more than minimum desired factor of safety (1.5) the design is safe.

Static analysis – Self weight & Functional loads
The FE model is solved for Von Misses stress using ANSYS software. Maximum stress plot is shown in Figure

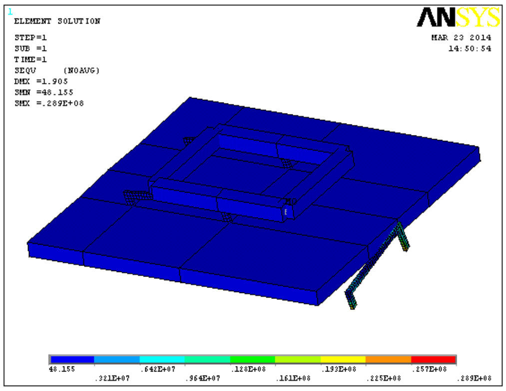


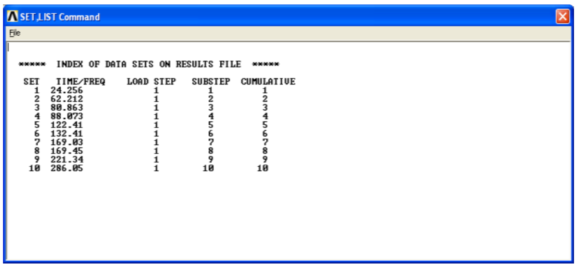
Figure 8 - Stress plot
Observations - Maximum Von Misses stress is observed to be 29 MPa.

Available factor of safety is observed to be (> 10) by comparing the maximum stress with that of allowable stress (Yield) of steel i.e. 330 MPa.
As the available factor of safety (> 10) is more than minimum desired factor of safety (1.5) the design is safe.

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.

The analysis calculates the natural modes of the discretized model, not those of the real continuous system. However the discretized 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 discretization 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.

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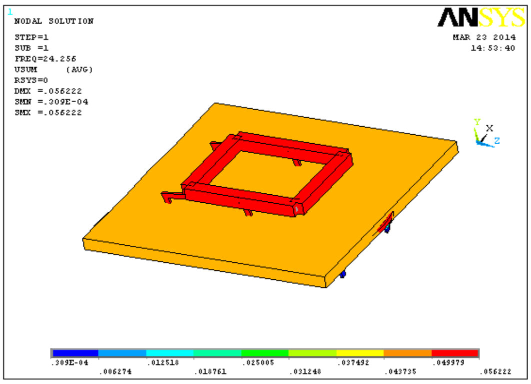
File

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

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	24.256	1	1	1
2	62.212	1	2	2
3	80.863	1	3	3
4	88.073	1	4	4
5	122.41	1	5	5
6	132.41	1	6	6
7	169.03	1	7	7
8	169.45	1	8	8
9	221.34	1	9	9
10	286.05	1	10	10

First bending mode shape is shown in Figure

Figure 9 – First mode shape



Observations - Frequency of the intended system corresponding to first bending mode is found to be 24 Hz which is more than the frequency associated with operation i.e. 0.166 Hz.
Hence system doesn't experience resonance.

V RESULTS AND CONCLUSION

Design of a track re-creator for functional testing of race cars is done which will have provision for providing two orthogonal angular motions to the system under test.

Static analysis – Self weight

Maximum Von Misses stress is observed to be 25 MPa.
Available factor of safety is observed to be (> 10) by comparing the maximum stress with that of allowable stress (Yield) of steel i.e. 330 MPa.
As the available factor of safety (> 10) is more than minimum desired factor of safety (1.5) the design is safe.

Static analysis – Self weight & Functional load

Maximum Von Misses stress is observed to be 29 MPa.
Available factor of safety is observed to be (> 10) by comparing the maximum stress with that of allowable stress (Yield) of steel i.e. 330 MPa.
As the available factor of safety (> 10) is more than minimum desired factor of safety (1.5) the design is safe.

Modal analysis

Frequency of the intended system corresponding to first bending mode is found to be 24 Hz which is more than the frequency associated with operation i.e. 0.166 Hz.

Hence system doesn't experience resonance.

VIRECOMMENDATIONS

It is recommended to incorporate the track recreator for functional testing of race cars proposed in this project for simulating track disturbances associated with course of application of race cars.

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