



Design and analysis of bogie and coach of mass rapid transit (metro) for Nepal

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Abstract

Mass Rapid Transit (Metro) has become the ultimate solution for city transportation due to its various advantages including traffic issues solving ability, emission free mode, comfortability, speed and efficiency. In developing countries like Nepal where people are deprived of proper transportation system, Metro railway can bring about lots of positive socio-economic changes. A lot of analysis regarding the technical aspects and designing process adapted throughout the globe has been done and the best suited design for the context of Nepal has been included in the paper. A Bogie is a chassis or framework carrying wheels attached to the train, serving as a modular subassembly of wheels and axles. Old bogies are cast models which are huge and bulky in its nature. In order to overcome the limitations of the existing FIAT bogie, a new bogie namely Y32 Bogie is designed to equip with the coach. The coach with the longitudinal arrangements of seats is designed to accommodate the higher number of passengers. The design of the various bogie components is done following the procedures of machine design and various standards that suits the best for our context. The 3D-modeling of the bogie and the coach is performed in CREO PARAMETRIC 5.0. The design of bogie frame is then imported to ANSYS 19.2 for Finite Element Analysis. The analysis on stresses and maximum deformation induced in the bogie frame structure for different materials is performed and the comparison is made to find the best suited design. To achieve the Finite Element Analysis, the load criteria are considered from Indian Railway Standards.

Keywords: ANSYS 19.2; FIAT Bogie; Y32 Bogie; Creo Parametric 5.0; FEA

1. Introduction

A Mass Rapid Transit system is a transit system using trains of high performance electrically powered rail cars operating in exclusive rights-of-way, usually without grade crossings, with high platform stations. Mass Rapid Transit help catalyze new economic and employment opportunities by acting as nodes of development. The mass rapid transit system is the need of the hours as it offers the benefit of allowing larger number of people to travel from one area to the other area in the predetermined time. Nowadays, pollution has become main concern for the Government as different ways are being explored to reduce the pollution problems. The provision of MRTs as means of public transport offers great opportunity in combating the pollution issues. In contrasts to buses being used for public transport, MRTs offers clean and hygienic mode of transport for commuting from place of residence to their place of work [1].

A bogie is a structure underneath a railway vehicle body to which axles and wheels are attached through bearings. The term "bogie" is used in British English, while a "wheel truck", or simply "truck" is used in American English [2]. The outer shell of the train/metro that carry the passenger or load within it is called coaches. The overall part that lies on the bogie frame is also called coaches. Depending on the feature and requirement the coaches are of different types. Motor coach is a powered vehicle with axles each equipped with a 3 phase asynchronous Traction Motor whereas Trailer coach is a non-powered coach. Generally, Trailer coach is situated behind the motor coach [3].

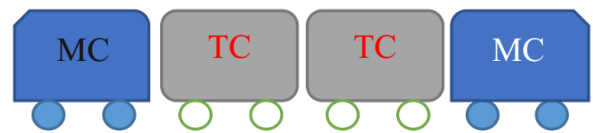


Figure 1: Configuration of coaches.

2. Loadings and calculations

Two motor coach and two trailer coach is designed in Creo Parametric 5.0 following specifications according to the Railway Board of India and its approximate mass is measured by assigning specified materials to all the components. The configuration of the coaches is shown in Fig. 1. The maximum design speed of the metro is 90 km/hr and operating speed is 70 km/hr [4].

Number of Motor Coach (MC) = 2

Number of Trailer Coach (TC) = 2

Mass of motor coach = 63 tons

Mass of trailer coach = 53 tons

Total Mass of Metro (M_{metro}) = 232 tons

Weight of the metro (W_{metro}) = 255 tons

2.1. Power and numbers of motors required [5]

The tractive effort can be calculated by the following equations:

$$\text{Tractive effort (TE)} = F_a \pm F_g + F_r \text{ (Newton)}$$

Force required for Linear and Angular acceleration (F_a) is given by;

$$F_a = 277.8 \times W_e \times a$$

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Where,

$$\text{Accelerating weight } (W_e) = \text{Weight} \\ + \text{Weight} \times \text{Rotational mass in \%}$$

$$W_e = 272.85 \text{ tonne}$$

$$\text{Acceleration } (a) = \frac{\text{Velocity (km/hr)}}{\text{Time (sec)}} = 2.25 \text{ km/hr sec}$$

$$F_a = 170544.8925 \text{ N}$$

Force due to Gradient and Curve Resistance (F_g) is given by

$$F_g = \pm 98.1 \times W \times G + R_{\text{curve}}$$

W is the weight of the train in ton.

$$\text{Gradient } (G) = \sin \theta = \frac{\text{Elevation}}{\text{Distance along the train}}$$

θ is the angle of slope.

For the straight and level track condition the gradient force can be neglected since the value of gradient is zero.

$$\text{Degree of curvature} = \frac{1750}{\text{Radius of curvature in metres}}$$

Assuming the radius of the curvature to be 300 m.

$$\text{Degree of curvature} = 5.833$$

The curve resistance is given by

$$R_{\text{curve}}$$

$$= \frac{0.4 \times \text{Degree of curvature} \times g \times \text{Mass}_{\text{train}}(\text{tonne})}{1000} \text{ kN} \\ = 5.31 \text{ kN}$$

Therefore,

$$F_g = \pm 98.1 \times W \times G + R_{\text{curve}} = 5.31 \text{ kN}$$

Tractive force to overcome the Train Resistance (F_r) is given by

$$F_r = W \times r$$

W is the weight of the train in Newton.

r is the specific train resistance in N/tonne.

Train resistance using W.J. Davis formula:

$$\text{Train resistance } (R) = \\ = A + BV + CV^2 \\ 1.2 + 0.001 \times 100 + 0.0001 \times 100^2$$

$$\text{Train resistance } (R) = 29 \text{ kg/tonne}$$

$$\text{Specific train resistance } (r) = 22.45 \text{ N/tonne}$$

Therefore,

$$F_r = 5724.75 \text{ N}$$

Finally, the total tractive force can be calculated using equation

$$\text{Tractive effort (TE)} = F_a \pm F_g + F_r = 181.58 \text{ kN}$$

Now, the power required to run the metro at a speed of 90 km/hr with acceleration of 2.25 m/s is given by the equation;

$$\text{Power } (W) = \text{Tractive effort (N)} \times \text{Speed (m/s)} = 4539.495 \text{ kW}$$

The required power is higher (i.e. 4539.495 KW) because the tractive force is higher initially but when the train gets speed the tractive force get decreased resulting in the low amount of power consumption.

The above power is generated by motor situated in bogie.

Number of motor coach (MC) = 2

Number of motor in each bogie = 2 (One motor per axle)

Number of bogie in single coach = 2

$$\text{Total number of motor in a single coach}$$

$$= \text{Number of bogie per coach} \times \text{Number of motor in each bogie} \\ = 4$$

$$\text{Total number of motor in train}$$

$$= \text{Number of coach} \times \text{Number of motor in one coach} \\ = 8$$

Therefore, the above power is distributed along the eight motors.

$$\text{Power of each motor} = \frac{\text{Total power required}}{\text{Total number of motors}} \\ = 567.43 \text{ kW} = 760.94 \text{ hp}$$

Hence, to drive a metro train including all passenger and train weight we will need eight numbers of motor each of 765 hp.

For the further design of axle and other components of the bogie we need the torque transmitted by the motors which can be obtained by using equation;

$$\text{Torque developed by motors } T = \frac{T.E \times D_{\text{wheel}}}{2 \times \eta \times \beta} = 28195.622 \text{ Nm}$$

Where,

η is the transmission efficiency assumed to be 92%.

β is gear ratio assumed to be 3.5.

T is the torque developed by all motors.

$$\text{Torque developed by a single motor, } T_{\text{one, motor}} = 3524.45 \text{ Nm}$$

2.2. Forces for FIAT and Y32 Bogie Frame [6]

Vertical force on bogie frame is given by;

$$F_Y = \frac{1.4 \times g(M_V - 2m^+)}{2}$$

Transverse force on bogie frame is given by;

$$F_Z = 2 \times (10^4 + \frac{(M_V + C_1)g}{3n_a n_b})$$

Longitudinal force on bogie frame is given by;

Table 1: Load comparison for FIAT and Y32 bogie frame.

Force type	Force Magnitude (kN)	
	FIAT Bogie Frame	Y32 Bogie Frame
Vertical Force	274	293
Transversal Force	106	114
Longitudinal force	25.75	28
Impact Force	2575.125	2803.3

$$F_X = 0.1 \times \frac{M_V g}{n_a}$$

Impact force on bogie frame is given by;

$$\text{Impact force} = 5g \times M_V$$

For FIAT Bogie Frame;

Mass of locomotive in running order (M_V) = 52500 kg

Mass of bogie (m^+) = 6300 kg

Acceleration due to gravity (g) = 9.81 m/s²

Number of axles (n_a) = 2

Number of bogies (n_b) = 2

Mass of the driver (C_1) = 85 kg

For Y32 Bogie Frame;

Mass of locomotive in running order (M_V) = 57152.6 kg

Mass of bogie (m^+) = 7257.48 kg

Acceleration due to gravity (g) = 9.81 m/s²

Number of axles (n_a) = 2

Number of bogies (n_b) = 2

Mass of the driver (C_1) = 85 kg

From Table 1, it can be observed that all the forces acting on the y32 bogie frame is higher than flat bogie frame because both the moving mass and locomotive mass of the y32 bogie is higher. The load bearing capacity of y32 bogie is much higher than the flat bogie.

2.3. Axle load and diameter [7]

Case I: Shafts Subjected to Combined Twisting Moment and Bending Moment.

The diameter of the gear (attached to the motor) is assumed to be 100 mm. In above calculation, we the gear ratio of 3.5. So, the diameter of the gear (attached to the axle) is 350 mm.

Speed of the driven gear (pinion) is given by:

$$N_{\text{pinion}} = \frac{60 \times v}{2\pi r_{\text{pinion}}} = 1364.18 \text{ rpm}$$

Torque transmitted by pinion (T_{pinion}) = $\frac{P \times 60}{2\pi N_{\text{pinion}}} = 3.97 \times 10^6 \text{ Nmm}$

Tangential force on the pinion, $F_t = \frac{2T_{\text{pinion}}}{D_{\text{pinion}}} = 22.697 \times 10^6 \text{ N}$

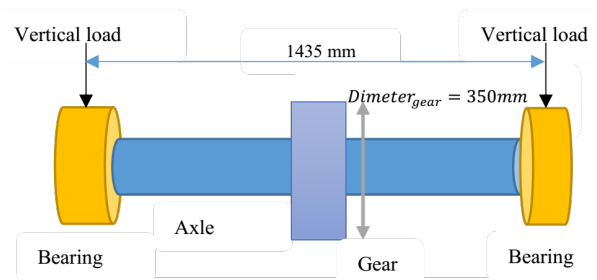
Maximum bending moment of a simply supported shaft carrying a central load (Fig. 2) is given by;

$$M = \frac{W \times L}{4} = 55.41 \times 10^6 \text{ Nmm}$$

Equivalent twisting moment is given by;

$$T_e = \sqrt{(M^2 + T^2)} = 55.55 \times 10^6 \text{ Nmm}$$

The diameter of the axle can be calculated by using design data book.

**Figure 2:** Load on bearing and gear.

$$\text{Equivalent twisting moment, } T_e = \frac{\pi}{16} \times \tau \times d_{\text{shaft}}^3$$

Where, τ is maximum shear stress theory = 60 Mpa .Then, we get;

$$d_{\text{shaft}} = 167.68 \text{ mm}$$

Case II: Size of the shaft when subjected to gradually applied load;

Equivalent twisting moment is given by;

$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2} = 83.208 \times 10^6 \text{ Nmm}$$

K_m is the combined shock and fatigue factor for bending moment whereas K_t is the combined shock and fatigue factor for torsion [Values are taken from design data book].

Since,

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

So, $d = 191.86 \text{ mm}$

Equivalent bending moment is given by;

$$M_e = \frac{1}{2} [K_m \times M + \sqrt{(K_m \times M)^2 + (K_t \times T)^2}] = 83.16 \times 10^6 \text{ Nmm}$$

Diameter of the shaft is given by;

$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3$$

$$d = 211.13 \text{ mm}$$

The maximum value is taken, so the diameter of the shaft will be 211.13 mm. [Standard diameter of 220 mm is taken]

2.4. Brake Design [7]

The main condition imposed is that braking forces at the wheel-rail contact surface $F_{b, \text{max}}$ must not exceed the wheel-rail adhesion force F_a , for designing purposes considered in normal conditions:

$$F_{b, \text{max}} \leq F_a$$

Considering a vehicle having W_v weight being equipped with n brake shoes;

$$F_{b, \text{max}} = \sum_{i=1}^n (\mu_s \cdot P_{si}) \leq \mu_a \cdot W_v$$

Respectively with n brake discs;

$$\frac{4 \cdot \mu_d \cdot r_m \sum_{i=1}^n (P_{d,i})}{D_O} \leq \mu_a \cdot W_v$$

Where,

μ_a is the wheel/rail adhesion coefficient

P_s and P_d are the clamping force on a brake shoe, and pad respectively

μ_s and μ_d the friction coefficient between brake shoes and wheel tread and brake pad and disc respectively,

The wheel diameter ($D_o=300\text{ mm}$) and $r_m = 0.15\text{ mm}$, the medium friction radius.

In the case of plastic brake shoes the friction coefficient is about 0.25, while for brake pads is about 0.35. Here, $\mu_s = 0.25$ and $\mu_d = 0.35$

Using equation to get clamping force by pad, P_d ;

$$\frac{4 \cdot \mu_d \cdot r_m \sum_{i=1}^n P_{d,i}}{D_o} = \mu_a \cdot W_v$$

$$P_d = 7354.98\text{ N by each brake}$$

In practical calculus, for the friction coefficient between cast iron brake shoes and wheel tread there are recommended different empirical relations, determined by experiments, depending on most important influencing factors, meaning mainly the running speed V [km/h], the applied forces on a break shoe p_s [kN] or the surface contact pressure p_s [N/mm^2].

Using UIC formula,

$$\mu_s(V, p_s) = 0.49 \frac{\frac{10}{3.6} \cdot V + 100}{\frac{35}{3.6} \cdot V + 100} \frac{\frac{875}{g} \cdot p_s + 100}{\frac{2860}{g} \cdot p_s + 100}$$

We get,

$$\text{Surface contact pressure } (p_s) = 0.1291\text{ N}/\text{mm}^2$$

Using Karvatzki formula;

$$\mu_s(V, P_s) = 0.6 \frac{V + 100}{5V + 100} \frac{\frac{16}{g} \cdot P_s + 100}{\frac{86}{g} \cdot P_s + 100}$$

$$\text{Applied force on the brake shoe } (P_s) = 2306.08\text{ N}$$

3. CAD design of coach, its interiors and bogie

The exterior and the interior of a coach are shown in Fig. 3 and Fig. 4 respectively which are designed using Creo Parametric 5.0. The design of inner aesthetics of a coach refers to the arrangements of seats and the optimization of space to accommodate the maximum number of passengers during the peak traffic hours. The interiors of coach has been designed based on the fact that the floor has a total area of 64 m^2 . The maximum number of passengers that a coach can accommodate is 90. In order to maximize the passenger carrying capacity, longitudinal seating arrangement has been adopted. Criteria for the calculation of standing passengers are 1 person per square meter of standing floor area in normal state and 2 persons per square meter in crush state of peak hour. The CAD model of bogie, Fig. 5 has been designed as per the calculations and standards followed by Indian Railways.

4. Static analysis of the bogie frame

Bogie frame is a chassis carrying wheels, suspension system, brakes and attached under the railway vehicles for running. Generally two pieces of bogies used for one passenger coach and a railway vehicle consists of several coaches. Bogies are critical components of a railway vehicle because of being running components. In this study the static structural analysis of Y32 bogie is performed under two different loading conditions for different materials and the results are interpreted.

The static structural analysis of the bogie frame involved the following steps:

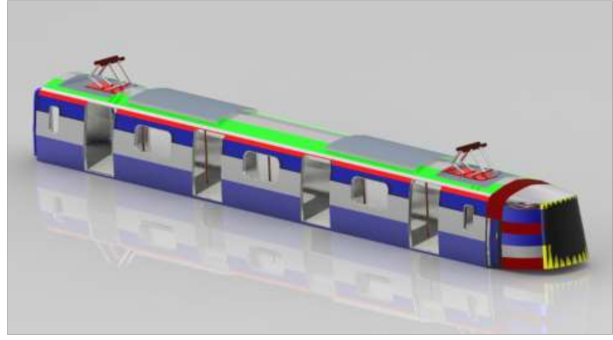


Figure 3: CAD model of exterior of coach.

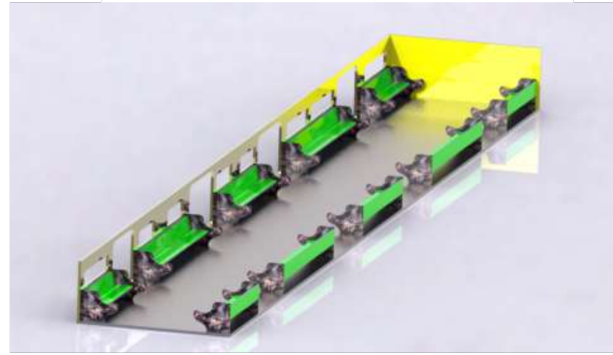


Figure 4: CAD model of interior of coach.

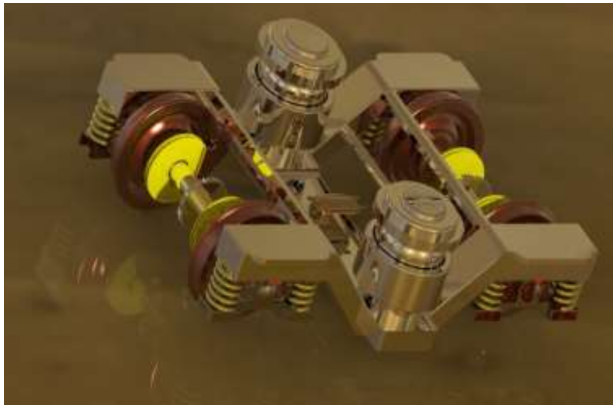


Figure 5: CAD model of Y32 bogie.

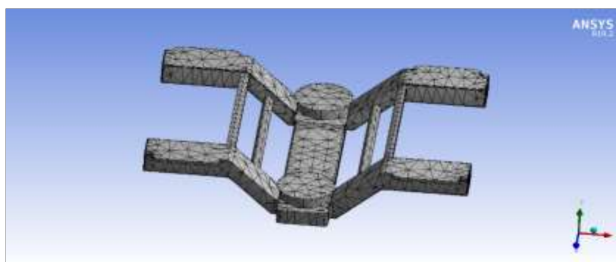


Figure 6: Meshed model of bogie frame.

Table 2: Loads on bogie frame in maximum loading condition.

Particulars	Quantity	Units
Total Mass of a Coach	63	Tonnes
Total Weight of a Coach	630000	N
Total Load on a Bogie Frame	315000	N

4.1. Meshing of the bogie frame using Ansys workbench

The model of the bogie frame is prepared using Creo Parametric 5.0. All the dimensions and geometric properties of Y32 bogie is modelled in the software. The model is then imported to the Ansys Workbench and meshed as shown in Fig. 6.

4.2. Applying boundary conditions

The four ends of the bogie frame are fixed. Boundary conditions are applied with two different loading conditions as shown in Fig. 7. The total load acting on the bogie frame is divided into two half and applied on the either end of the secondary suspension.

4.3. Loading conditions

Two different loading conditions incur during static analysis of a bogie frame. The minimum loading condition occurs when the car body or the coach is empty. The maximum loading condition occurs when the coach is under crush loading or when the coach is full with its maximum limit of passengers. Since a single coach consists of two bogies, the total load is divided into two halves and applied on the bogie frames. The approximate load of the assembly of bogie and coach is obtained by assigning the specific materials to the overall assembly in Creo Parametric 5.0.

Tare mass of the coach = 58.05 Tonnes

Average mass of a passenger = 55 Kg

Maximum permissible number of passengers = 90

Total mass of passengers in maximum loading condition = 4950 Kg

Total weight of passengers in maximum loading condition = 49500 N

4.3.1. Case I (Maximum Loading Condition)

The maximum load is equal to summation of the tare mass of the coach and its pay mass. The loads acting on bogie frame in maximum loading condition are calculated and tabulated in Table 2.

4.3.2. Case II (Minimum Loading Condition)

The minimum load is equal to the tare mass of the coach or the weight of the coach excluding the passengers. The loads acting on bogie frame in minimum loading condition are calculated and tabulated in Table 3.

Table 3: Loads on bogie frame in minimum loading condition.

Particulars	Quantity	Units
Tare mass of a Coach	58.05	Tonnes
Tare Weight of a Coach	580500	N
Total Load on a Bogie Frame	290250	N

Table 4: Material properties.

Material	Density (Kg/m ³)	Young's Modulus (Pa)	Tensile Yield Strength (Pa)	Tensile Ultimate Strength (Pa)
Cast Steel	7820	2.03×10^{11}	3.49×10^8	5.05×10^8
Low Carbon Steel	7850	2.1×10^{11}	3.29×10^8	4.4×10^8

4.4. Material selection

The commonly used materials for the manufacture of bogie frame are cast steel and low carbon steel. The analysis is performed under the above mentioned materials under different loading conditions and the results are obtained. The materials properties for all the materials are tabulated in Table 2.

After adding the material properties, the results of total deformation for cast steel under maximum and minimum loading conditions are simulated as shown in Fig. 8 and Fig. 9 respectively. Similarly, Fig. 10 and Fig. 11 depict the results of equivalent stresses for cast steel under maximum and minimum loading conditions respectively. The simulated results of total deformation for low carbon steel under maximum and minimum loading conditions are simulated as shown in Fig. 12 and Fig. 13. Fig. 14 illustrates the result of equivalent stresses for low carbon steel under maximum loading condition and Fig. 15 illustrates the result of equivalent stresses for low carbon steel under minimum loading condition.

4.5. Static structural analysis

5. Results and discussion

Table 5 shows the tabulated results of the two tests conducted on same bogie frame but with different materials and different loading conditions. Ansys result for the static structural loading condition for all the materials shows that the maximum induced stress

Table 5: Total deformation and equivalent stress for ccast steel and low carbon steel under different loading conditions.

Materials	Minimum Loading Condition	Maximum Loading Condition	Total Deformation (m)	Equivalent Stress (Pa)
Cast Steel	7.408×10^{-7}	11212	8.0402×10^{-7}	12168
Low Carbon Steel	7.146×10^{-7}	11115	7.755×10^{-7}	12063

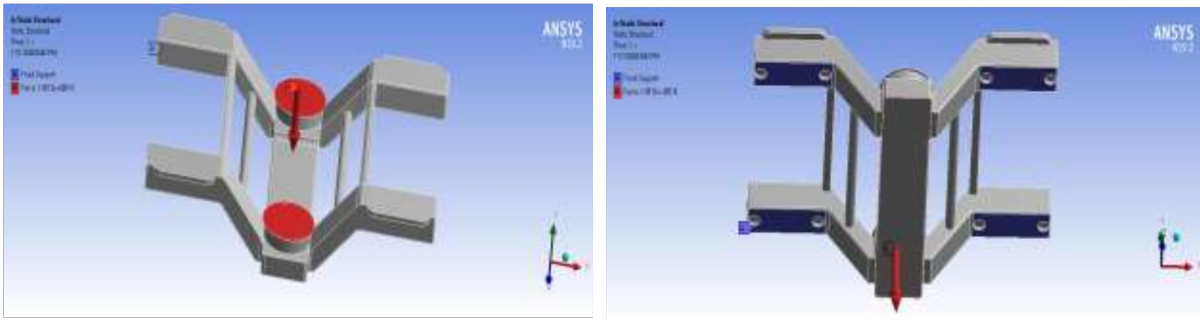


Figure 7: Boundary conditions of bogie frame.

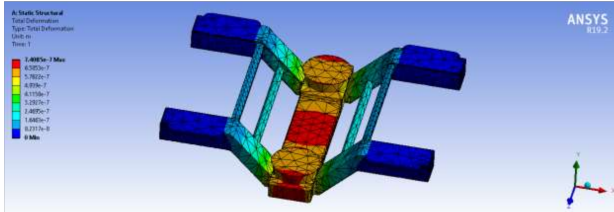


Figure 8: Total deformation of bogie frame for cast steel under minimum loading condition.

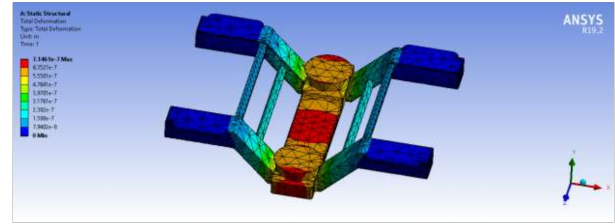


Figure 12: Total deformation of bogie frame for low carbon steel under minimum loading condition.

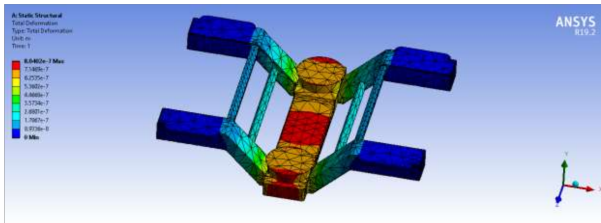


Figure 9: Total deformation of bogie frame for cast steel under maximum loading condition.

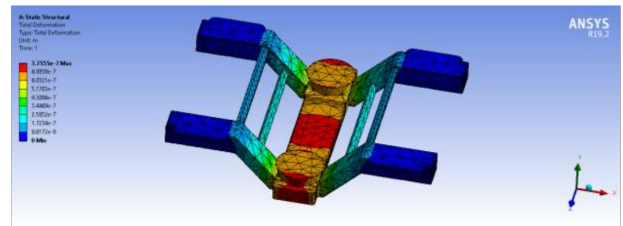


Figure 13: Total deformation of bogie frame for low carbon steel under maximum loading condition.

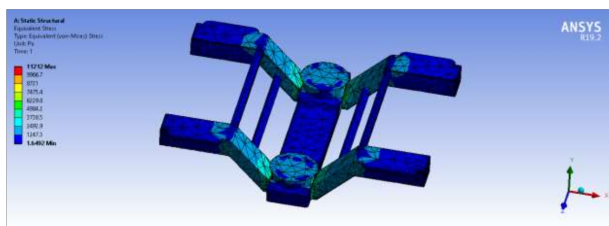


Figure 10: Equivalent stress of bogie frame for cast steel under minimum loading condition.

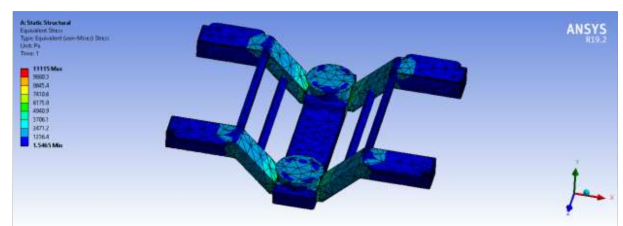


Figure 14: Equivalent stress of bogie frame for low carbon steel under minimum loading condition.

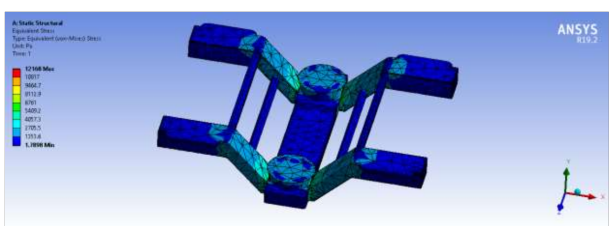


Figure 11: Equivalent stress of bogie frame for cast steel under maximum loading condition.

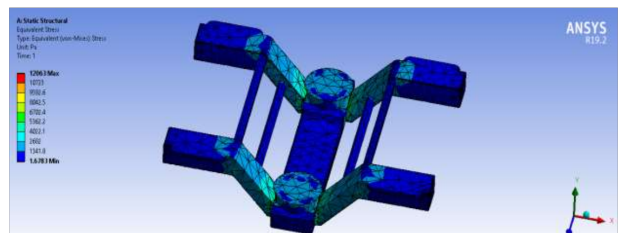


Figure 15: Equivalent stress of bogie frame for low carbon steel under maximum loading condition.

is lesser than its tensile yield strength. The total deformation for Cast Steel is higher than that of Low Carbon Steel in both the loading conditions. The value of total deformation in all the cases is negligible. So, the design of the bogie frame is considered to be safe for both the materials.

6. Conclusion

All the design and calculations of this research are done based on the Indian Railway Standards as various critical factors including traffic conditions, land topography and other socio-economic aspects of Nepal are relatable to that of India. The research reveals the essential elements to be considered while designing the components of metro. This research can be used as a reference for further research regarding the various prospects of metro in our country. The similar methodology can be followed for performing structural analysis on other components of the bogie.

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