Stresses Analysis of Various Force for Front Axle Beam in Heavy Duty Vehicle

Khaing Zarli Wint, Aung Myo Htoo, Si Thu Thwin

***Abstract*— Front axle beam of heavy duty vehicle is the important component of the vehicle. Up to 40% of the vehicle load carrying capacity is taken up by front axle beam. The model selected is that of a heavy duty vehicle which has a gross vehicle mass of 24 tons. The collapse of front axle while static and dynamic loading conditions is of huge apprehension to both goods and human lives, hence it becomes essential to scrutinize the structural integrity of the axle to endure characteristic such loading which can build up stresses in the same being consequential to fracture and finally failure.**

***Index Terms*—Axle Beam, Static and Dynamic Loads, Stresses, Failure**

I. Introduction

The automotive industry is one of the fastest growing sectors all over the world. The automobile components are used in commercial vehicles, passenger cars, two wheelers and auto related parts. A front axle is a rotating shaft at the front of a vehicle that turns the front wheels [1]. Front axle includes the axle beam, spring pad and king pin bore, etc.

The front axle experiences static and dynamic load conditions due to irregularities of road, mostly during its travel on and off roads. In the static conditions, the axle might be considered as beam supported vertically upward at the ends. Under the dynamic conditions, vertical bending moment is increased due to road roughness [2]. So, axle must be resistant to tolerate additional stresses and loads. A misaligned front axle cause in improper turning of tires, reduced tire life, difficulty in driving and unsafe the vehicle.

Earlier research with respect front axle beam mainly focused on large cases. According to the early research, an existing front axle is modified in given load condition.

This paper intends to provide safe working conditions, effective stress concentration and maximum deflection in various conditions of front axle beam. The design is based on the strength of material and stress analysis. The front axle must be determined by using the design of machine's elements and strength of materials theory.

***Manuscript received October, 2018*.**

***Khaing Zarli Wint**, Department of Mechanical Engineering, Pyay Technological University, Ministry of Education, Pyay, Myanmar, Phone/ +959423681465*

***Aung Myo Htoo**, Department of Mechanical Engineering, Pyay Technological University, Ministry of Education, Pyay, Myanmar, Phone/ +09422454664*

***Si Thu Thwin**, Department of Mechanical Engineering, Pyay Technological University, Ministry of Education, Pyay, Myanmar, Phone/ +0943044185*

II. Construction and Operation

Front Axle Beam

For design purpose, the front axle beam of ISUZU CYZ 400 vehicle was chosen. The front axle beam have I cross section in the middle and elliptical cross-sections at the ends. The I cross-section has lower section modulus and hence gives better performance with lower weight. This type of construction produces an axle that is light weight and great strength so I cross-section type is chosen for this paper.

The I cross-section front axle beam is as shown in Figure1.

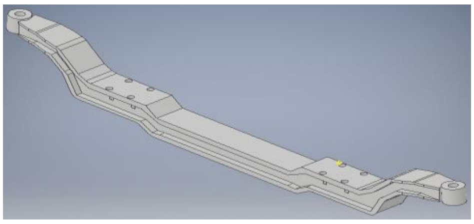

Fig. 1 I Cross-Section Type Front Axle Beam for Heavy Duty Vehicle

III. Methodology

In this design, it includes of three main loads. They are

A. Vertical
B. Braking
C. Cornering

The vertical loads cause the static condition and the cornering loads cause the dynamic condition. To calculate the front axle beam, the design data of ISUZU CYZ 400 illustrated in table I are taken as the specification data of this paper.

A) Calculation of Vertical Forces

In order to calculate vertical load, driving force equation is used. The velocity 50Km/hr is chosen for heavy duty vehicle.

$$F_{Rr} = F_{total} - F_{Rf} \tag{1}$$

$$\text{length, } l_1 = \frac{F_{Rr} \times l_{total}}{F_{total}} \tag{2}$$

Where,

F_{Rr} = rear reaction force [N]
F_{Rf} =front reaction force (or) normal force [N]
F_{total} =total force [N]

Table 1. Specification Data of ISUZU CYZ 400

Technical category	**Dimensions**	**Units**
Gross vehicle mass (maximum and laden)	235.440	KN
Overall length of vehicle	7.65	m
Overall Width of vehicle	2.49	m
Overall Height of vehicle	3.13	m
Total chassis cab tare mass (unladen)	80.98155	KN
Front (chassis cab tare mass)	44.9298	KN
Rear (chassis cab tare mass)	36.05175	KN
Wheelbase (or) total length	4.3	m

The driving force can be calculated as follows:

$$\text{Driving force} = \text{motive force} + \text{friction force} \quad (3)$$

$$\text{the magnitude of torque}, T = F_d \delta \quad (4)$$

Where, F_d = driving force [N]
δ = the drop from the spindle axis to section [m]

B) Calculation of Braking Forces

Braking force occurs due to inertia load.

$$\text{Braking force}, F_b = \mu \times \text{normal force} \quad (5)$$

Where, F_N = normal force [N]
μ is the coefficient of friction for tires in various conditions.

a) Combine (Vertical and Braking) Force

Inertia loads present while applying brakes

The combined (vertical and braking) force diagram acting on the front axle beam are shown in Ftgure 2.

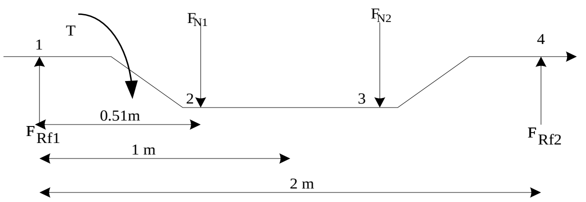

Fig. 2 Combine (Vertical and Braking) Force Diagram of Front Axle Beam

Where,
F_N = normal force [N]
F_{Rf} = front reaction force [N]
T = torque due to braking [N m]

C) Calculation of Cornering Forces

In cornering equations, the tire must develop a lateral force, the tire will also experience lateral slip as it rolls [6]. At high-speed,

$$\delta_i \cong \delta_o$$
$$\delta = \frac{\delta_o + \delta_i}{2} \quad (6)$$

Where,
δ_0 = outer steer angle
δ_i = inner steer angle

At radius of the turn, R=7.5m

$$\delta = 5.73\frac{l}{R} + \alpha_f - \alpha_r \quad (7)$$

Where,
α_f = front slip angle
α_r = rear slip angle

The lateral force coefficient is used for this equation.

$$\text{cornering friction force}, F_f = \mu F_N \quad (8)$$

Cornering Equation,

$$F_{yt} = F_{yf} + F_{yr} = \frac{mv^2}{R} \quad (9)$$

$$F_{yr} = \frac{ml_1 v^2}{lR} \quad (10)$$

b) Center of Height

In a cornering maneuver the lateral forces act in the ground plane to counterbalance the lateral acceleration acting at the CG of the vehicle.

$$F_{yfo} = F_{yfi} = \frac{Wl_2 v^2 h}{2lgRa} \quad (11)$$

Where,
F_{yt} = total lateral (cornering) force[N]
F_{yfo} =outer lateral (cornering) force at front wheel [N]
F_{yfi} =inner lateral (cornering) force at front wheel [N]
F_{yf} =lateral (cornering) force at the front axle [N]
F_{yr} =lateral (cornering) force at the rear axle [N]
a = front track width [m]
h = center of height [m]
m = mass of the vehicle [N]

The center of height is calculated from lateral (cornering) at the front axle equation.

Table 2. Typical Values of CG Height

Vehicle Type	**CG Height(inches)**	**CG Height(m)**
Sports Car	18-20	0.46-0.51
Compact Car	20-23	0.51-0.58
Luxury Car	20-24	0.51-0.61
Pickup Truck	30-35	0.76-0.89
Passenger Van	30-40	0.76-1.02
Medium Truck	45-55	1.14-1.40
Heavy Truck	60-85	1.52-2.16

Table 2 shows the typical values of CG height [6].

The center of height 1.85 is suitable for ISUZU CYZ 400 heavy duty vehicle.

$$F_{NO} = \frac{gmR\omega_s + hmv^2 + lv}{R\omega_s} \quad (12)$$

$$F_{Ni} = \frac{gmR\omega_s - hmv^2 - lv}{R\omega_s} \quad (13)$$

$$(F_{NO} - F_{NI})\omega_s - (F_{yfo} + F_{yfi})h = \frac{lv}{R} \quad (14)$$

Where,

F_{NO} =outer normal force[N]

F_{NI} =inner normal force[N]

L = constant magnitude angular momentum of wheel

R =radius of the turn[m]

c) Combined (Braking and Cornering) Force

The cornering of a vehicle is an important performance and lateral acceleration will be present to counteract the lateral acceleration the tire must develop lateral forces.

The lateral forces denoted by F_y is called the cornering force.

The combined (braking and cornering) force diagram acting on front axle beam are shown in Figure 3.

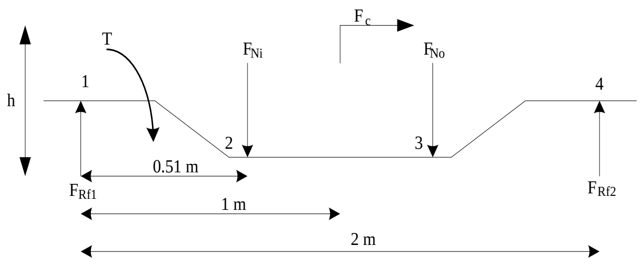

Fig. 3 Combine (Braking and Cornering) Force Diagram of Front Axle Beam

Where,

F_{Rf1}= front reaction force at inner wheel[N]

F_{Rf2}=front reaction force at outer wheel [N]

F_c= centrifugal force [N]

h=center of height [m]

The various forces are calculated and chosen the maximum allowable bending and shear stress for front axle beam.

Figure 4 shows the stresses in the x-y plane.

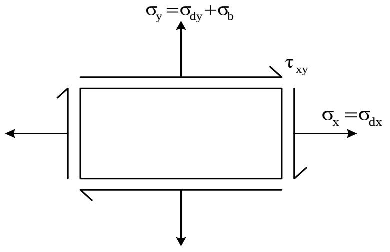

Fig. 4 Stresses for X-Y Plane

i) Calculation of maximum bending stress

$$\sigma_b = \frac{M_{max} y}{I} = \frac{M_{max}}{z} \quad (15)$$

ii) Calculation of direct stress at y-axis

$$\sigma_{dy} = \frac{F_N}{A} \quad (16)$$

Where,
F_N = normal force
A = area of the spring pad

iii) Calculation of total stress at y-axis

$$\sigma_y = \sigma_{b+} \sigma_{dy} \quad (17)$$

Where,
σ_b = bending stress
σ_{dy} = direct stress at y-axis

iv) Calculation of direct stress at x-axis

$$\sigma_{dx} = \sigma_x = \frac{F_d}{A} \quad (18)$$

Where,
F_d = driving force
A = area of spring pad

v) Calculation of shear stress at king pin bore

$$\tau = \frac{Tr}{J} \quad (19)$$

vi) Calculation of shear stress at I-section spring pad

$$\tau = \frac{VQ}{It} \quad (20)$$

Where,
V=Maximum shear force
Q=moment of area, m^3
I= Moment of Inertia [8]

Table 3. Materials Properties for Front Axle Beam of Alloy Steel AISI 4140

Type of material	**Ultimate stress (MN/m^2)**	**Yield stress (MN/m^2)**	**Modulus of Elasticity (GN/m^2)**
Alloy steel AISI 4140	1250	1140	205

Table 3 shows the mechanical properties of alloy steel.

1) Strength Check on Front Axle Beam

Front axle beam have to withstand various load in these conditions. This is fluctuation load may cause the variable stress which fail the material below the yield point. Therefore, the materials have to be analyzed by variable stress theory.

a) Mean stress, σ_m
The mean stress as follow:

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad (21)$$

b) The alternating stress, σ_{alt}

$$\sigma_{alt} = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{22}$$

c) Endurance Stress, σ_e

$$\sigma_e = 0.5\sigma_u \tag{23}$$

Where,
σ_u = ultimate stress

d) Safety factor, N

$$N = \frac{1}{\frac{\sigma_{alt}}{\sigma_e} + \frac{\sigma_m}{\sigma_u}} \tag{24}$$

The tensile stress σ_u is 1250MN/m^2 for alloy steel AISI 4140 from material properties.

Table 4. Mean Stress, Alternative Stress and Factor of Safety for Front Axle Beam of Alloy Steel AISI 4140

Type of Material	**Mean Stress, σ_m (MN/m^2)**	**Alternating Stress, σ_{alt} (MN/m^2)**	**Factor of safety for allowable stress**
Alloy Steel	256.43	212	1.8

The result data of mean stress and alternating stress for front axle beam of Alloy Steel AISI 4140 as shown in TableIII. The Goodman diagrams are as follows:

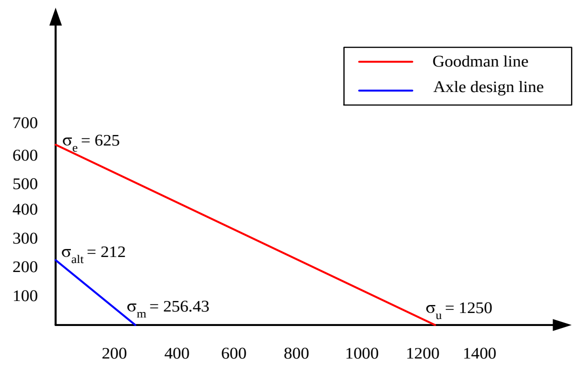

Fig. 5 Goodman Diagram for Front Axle Beam

Figure 5 shows the point at the intersection of mean stress 256.43 MN/m^2 and alternating stress 212MN/m^2 is under the goodman line. Therefore, the design of front axle beam is satisfied.

e) For Principal stresses

$$\sigma_{1,2} = \frac{1}{2}\left(\sigma_y + \sigma_x\right) \pm \frac{1}{2}\left[\left(\sigma_y + \sigma_x\right)^2 + 4\tau^2\right]^{\frac{1}{2}} \tag{25}$$

f) For von-Mises stress

$$\bar{\sigma} = [\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau^2]^{\frac{1}{2}} \tag{26}$$

g) For maximum shear stress

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} \tag{27}$$

Table 5. Result Data Stresses

Stress	symbol	Vertical condition	Braking condition	Cornering condition
Maximum Principle stress	σ_1	423.09	274.5	1000
Minimum Principle stress	σ_2	-336.57	-214.6	-534.6
Von-Mises stress	σ	400.87	217.5	862.7
Maximum shear stress	τ_{max}	379.83	252.1	789.2

h) The Deflection of Front Axle Beam

The deflection of front axle beam by using Macaulay's method,[10]

$$EI\frac{d^2y}{dx^2} = M \tag{28}$$

The section X-X is the deflection of front axle beam,

$$EI\frac{d^2y}{dx^2} = R_1x - F_{Ni}(x - 0.51) - F_{No}(x - 1.49) \tag{29}$$

Figure6 shows front axle beam is subjected loading condition. To determine the deflection of front axle beam by using Macaulay's method,

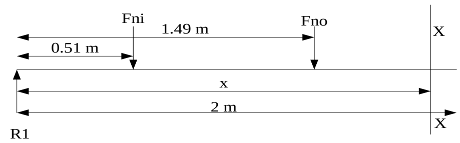

Fig. 6 Maximum Deflection of Front Axle Beam

4) Conclusion

In this research, I cross-section axle beam for heavy duty vehicle was analyzed by analytical method for alloy steel AISI 4140 at various forces. The results showed the modified GoodMann Line Diagram. The results of von-Mises stress are above table 5 and maximum deformation on front axle beam is 0.00103m. The results indicate von-Mises stress below yield strength of the material, which satisfies the design. In all global shapes, the structure is safe so our designed I cross-section type of front axle beam is safe.

Acknowledgment

The author would like to express her gratitude to Professor Dr. Aung Myo Htoo, Department of Mechanical Engineering, Pyay Technological University, for his invaluable guidance, support and encouragement throughout the entire process of this research work. Special thanks goes, U Si Thu Thwin, Lecturer Department of Mechanical Engineering, and Pyay Technological University for his valuable guidance and very helpful suggestions for this research. And, the author would like to express her parents for their understanding, kindness and helps during this research process.

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Khaing Zarli Wint received her BE (Mechanical) Degree from Pyay Technological University, Pyay, Myanmar in 2008. She works a demonstrator at Mechanical Department in PTU from 2009 to 2015. After that, she works at GTHS(Pyay) from 2015 until now. Now, she is currently researching ME thesis in PTU from 2016. Her research work is in solid and structural mechanics.

Aung Myo Htoo received his BE (Mechanical) Degree and ME (Mechanical) from Mandalay Technological University, Mandalay, Myanmar in 2001 and 2003, respectively. After that, he works at Department of Mechanical Engineering in TU(Dawei) from 2003 to 2004. He also finished his Ph.D degree from MTU, Mandalay Technological University, Mandalay, Myanmar in 2007.Now, he is currently working as a Professor at the Department of Mechanical Engineering, PTU. His research field is in fluid and dynamics.

Si Thu Thwin received his BE (Mechanical) Degree from Mandalay Technological University, Mandalay, Myanmar in 2002. He also finished his ME (Production) from Russia in 2006.Now, he is currently working as a Lecturer at the Department of Mechanical Engineering, PTU. His research field is in Production Technology.