Design Considerations of Steering and Braking Systems for Mazda Jeep

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Abstract- In automobile, there are two primary control systems. They are at the driver's disposal as the steering system and the braking system. The fundamental problem in steering is to enable the vehicle to transverse an arc such that all four wheels about the identical center point. The automotive brake system is a group of components designed to show and stop each of the four wheels of the automobiles. The braking action begins when the driver pushes on the brake pedal. In this paper, Mazda jeep steering gear is a recirculation ball nut type. This type contains the worm diameter is 12 mm and gear diameter is 120 mm with face width of 9 mm. Steering mechanism can be designed with wheel base 426.82 cm and turning radius of front wheel in outer and inner conditions is 8210.62 mm. In braking system for front wheel braking pressure and torque are 528.1902 KN/m² and 630.311 N-m with Diameter of brake master cylinder is 19 mm.

Indexed Terms- braking, brake pedal, recirculation, and steering, turning radius.

I. INTRODUCTION

Steering is the collection of components, linkages, etc. which allows any vehicle (car, motorcycle, bicycle) to follow the desired course. An exception is the case of rail transport by which rail tracks combined together with railroad switches provide the steering function. The primary purpose of the steering system is to allow the driver to guide the vehicle. The basic aim of steering is to ensure that the wheels are pointing in the desired directions. This is typically achieved by a series of linkages, rods, pivots and gears. One of the fundamental concepts is that of caster angle – each wheel is steered with a pivot point ahead of the wheel; this makes the steering tend to be self-centering towards the direction of travel. Many modern cars use rack and pinion steering mechanisms, where the steering wheel turns the pinion gear; the pinion moves the rack, which is a linear gear that meshes with the pinion, converting circular motion into linear motion along the transverse axis of the car (side to side motion). This motion applies steering torque to the swivel pin ball joints that replaced previously used kingpins of the stub axle of the steered wheels via tie rods and a short lever arm called the steering arm.

In an automobile vehicle, a braking system is an arrangement of various linkages and components (brake lines or mechanical linkages, brake drum or brake disc, master cylinder or fulcrums etc.) that are arranged in such a fashion that it converts the vehicle's kinetic energy into the heat energy which in turn stops or de accelerate the vehicle. The conversion of kinetic energy into heat energy is a function of frictional force generated by the frictional contact between brake shoes and moving drum or disc of a braking system.

The subject of steering geometry is very complex one in its own right and because of this, many manufactures import the knowledge and skills of specialists. However, the basic principles are relatively simple to understand and apply to most vehicles. The development of the steering and suspension (to which it is very closely linked) is based on past experience; much of the work has been through an evolutionary process learning from mistakes and modifying systems to suit varying applications. Many of the terms used are applied only to the steering and have no alternative; hopefully most of these are explained in the next. The geometry of steering may be best understanding by looking at Fig.1. A swinging beam mounted on a turntable frame turns the wheels.

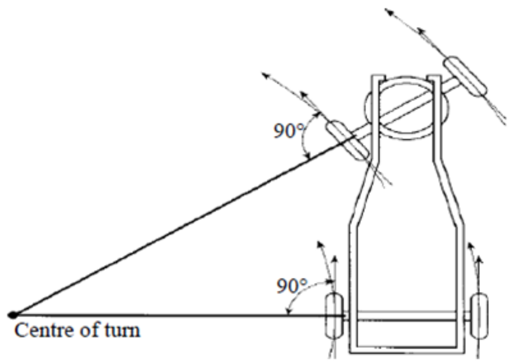

Fig.1 Steering Geometry

II. TYPES OF STEERING GEARBOX SYSTEM

The steering mechanism which provides the necessary leverage may be of several forms, however the important and common types are as follows.

1) Rack and pinion steering mechanism
2) Worm and wheel steering gear
3) Screw and nut type steering gear
4) Pinion and sector steering gear
5) Helical grooved cam steering gear
6) Worm and Roller steering

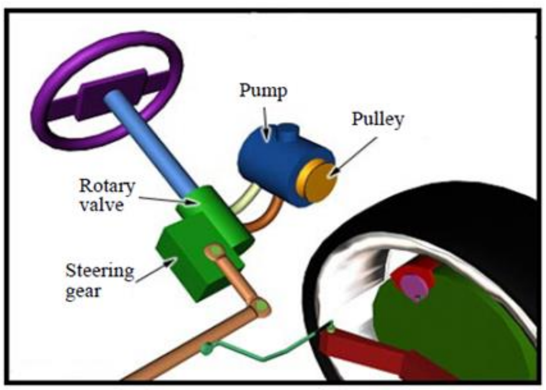

Fig.2 Types of Steering Gearbox System

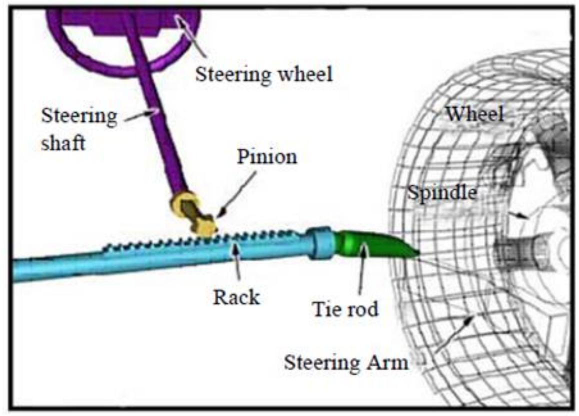

Fig.3 Steering Mechanisms

III. TYPES OF BRAKE

There are in general two main classes of drum brake system, one is internal expanding drum brake and other is external contracting drum brake. Disc brakes differ in construction and operate in different manner from the drum type brake. These brakes consist of a metal disc instead of a drum and a pair of pads, instead of the curved shoes. Pressure on the friction pads may be actuated by pull rod or by hydraulic system. In case of rear wheel brake the brake assembly is attached to the axle housing while of a front wheel brake, it is secured to the steering knuckle.

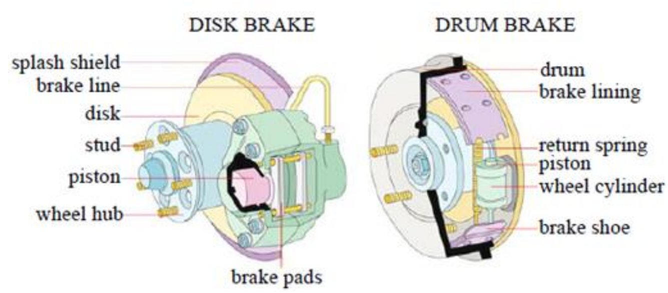

Fig.4 Types of Brakes

The main purposes of fitting brakes on motor vehicle are as given under.

1) In emergencies to bring the vehicle to rest in the shortest possible distance.
2) To control the vehicle when it is descending along the hills.
3) To keep the vehicle in desired position after bringing it in complete rest when there is no driver. To full fill the above needs two independent braking systems are provided in the vehicle.

4) "Service brake" which is operated by foot pedal in regular operation.
5) "Emergency brake" which is operated by hand lever while parking the vehicle.

When a fluid enclosed in a pipe is used to transmit the pedal effort to road wheel instead of the use of rod or cable, the system is called hydraulic brake. The outline and arrangement of the system is clear from the Fig.5. The system consists of a master cylinder and piston which is connected by steel piping to hydraulic wheel cylinders, provided at each wheel. A wheel cylinder consists of two pistons which are pivoted to brake shoes.

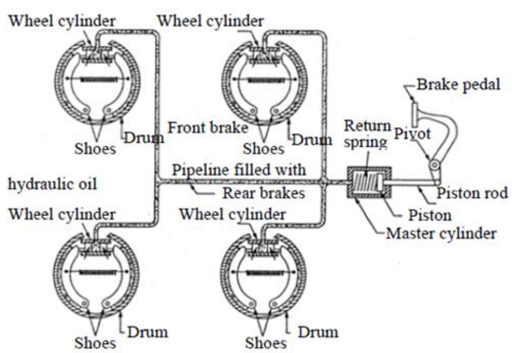

Fig.5 Hydraulic Braking System

IV. DESIGN CALCULATIONS OF STEERING AND BREAKING SYSTEMS

The specifications for MAZDA jeep motor vehicle in steering and braking system are as follows:
Vehicle Weight = 1540 kg
Gross Vehicle Weight = 2105 kg
Payload = 400 kg
Wheel Type = 5.00E×16 Drop Center
Diameter of Wheel = 530 mm
Tyre Type = 7.00-16-6 PRLT
Steering Type = Recirculation ball nut
Reduction Ratio = 29:1
Free Motion of Steering Wheel = 10 to 30 mm
Maximum Steering Angle:
Inner Wheel = 32
Outer Wheel = 26°30'
Steering Geometry:
Kingpin Inclination = 9° 30'
Camber = 1°30'
Caster = 2°
Toe-In = 2 to 3 mm
Caster Trail = 12.1 mm
Break Type = Hydraulic Internal Expansion, No servo
Diameter of Drum = 260 mm
Width of Drum = 45 mm
Thickness of Drum = 0.0635 mm
Pedal Free Travel = 15 to 20 mm
Wheel Cylinder Bore:
Front = 25.4 mm
Rear = 20.64 mm
Parking Break Type = Mechanical Internal Expansion
Overall Length of Vehicle = 3910 mm
Overall Width of Vehicle = 1695 mm
Overall Height of Vehicle = 1955 mm
Speed Range = 50 km/hr to 120 km/hr

Centre distance between axis of worm and axis of gear, C = 66 mm
Pitch diameter of the worm,

$$D_w = \frac{C^{0.875}}{3.48} = 0.02664 \text{ m} \quad (1)$$

Axial module,

$$m_a = \frac{D_w}{3\pi} = 2.83 \text{ mm} \quad (2)$$

Diameter of gear,

$$D_g = 2C - D_w = 105 \text{ mm based on } D_w$$

(3)

Gear transmission ratio (practically), R = 30.

$$R = \frac{D_g}{m_a N_w} \quad (4)$$

TABLE I
RELATIONSHIP BETWEEN NUMBER OF WORM AND PITCH DIAMETER OF GEAR

Nw	1	2	3	4	5
Dg, (mm)	60	120	180	240	300

$$D_w = 2C - D_g \quad (5)$$

This is close to desired proportions. Since the worm will have a quadruple thread.

$$\tan\alpha = \frac{m_a N_w}{D_w} \quad \alpha = 26.565° \quad (6)$$

The face width,

$$b = 0.73D_w = 8.76 \text{ mm, use } b = 9 \text{ mm} \quad (7)$$

Then, the capacity and strength of the gear calculations are as follows.

Phosphor bronze ultimate stress = 210 MN/m^2

Factor of safety = 3.5 (dead loads)

Involutes of tooth from, ϕ□= 20° stub

Number of teeth year, Ng = 10 × Nw = 10 × 2 = 20

From Ng = 20 and ϕ□= 20° stub,

Form factor, y = 0.125

$$\text{Working Stress} = \frac{\text{Ultimate Stress}}{\text{Factor of Fafety}} = 60 \text{ NM/m}^2 \quad (8)$$

Normal module,

$$m_n = \frac{m_a}{\cos\alpha} = 3.35 \text{ mm} \quad (9)$$

Permissible tangential load,

$$F = sby\pi m_n = 711.26 \text{ N} \quad (10)$$

Worm for hardened steel and gear for phosphor bronze,

B = 550 kN/m^2

Allowable wear load,

$$F_w = D_g bB = 594 \text{ N} \quad (11)$$

Endurance load,

$$F_0 = S_0 by\pi m_n = 829.81 \text{ N} \quad (12)$$

Dynamic load,

$$F_d = \left(\frac{S_0}{S}\right) F = 237.09 \text{ N} \quad (13)$$

The allowable values of Fo and Fw must be greater than dynamic load. This is satisfied design condition.

V. DESIGN CALCULATION OF STEERING MECHANISMS

Wheel tread of vehicle, a = 1400 mm

The inner wheel stub axle, θ = 32°

Distance between the pivot centre, c = 136 cm

Length of track rod, d = 126 cm

Length of track arm, r = 20 cm

$$\sin\alpha = \frac{c-d}{2r}, \alpha = 14.5°$$

$$\sin(\alpha + \theta) + \sin(\alpha - \emptyset) = 2\sin\alpha$$

$$\emptyset = 27.52°$$

$$\text{Permissible tangential load}, F = sby\pi m_n = 711.26 \text{ N}$$

For correct steering,

$$\cot\emptyset - \cot\theta = \frac{c}{b}$$

Turning radius of the outer front wheel,

$$= \frac{b}{\sin\emptyset} + \left(\frac{a-c}{2}\right) \quad (14)$$

Turning radius of the inner front wheel,

$$= \frac{b}{\sin\emptyset} - \left(\frac{a-c}{2}\right) \quad (15)$$

Turing radius of the outer rear wheel,

$$= (b \times \cot\emptyset) + \left(\frac{a-c}{2}\right) \quad (16)$$

Turning radius of the inner rear wheel,

$$= (b \times \cot\emptyset) - \left(\frac{a-c}{2}\right) \quad (17)$$

TABLE II
RESULTS OF STEERING MECHANISMS

Wheel based, b	426.8243 cm
Turning radius of outer front wheel	9216.0218 mm
Turning radius of inner front wheel	8034.5156 mm
Turning radius of outer rear wheel	8210.6166 mm
Turning radius of inner rear wheel	8170.6166 mm

TABLE III
RESULTS FOR THE BRAKING EFFICIENCY

Speed (km/hr)	Braking Distance (m)	Efficiency (%)
50	16	19.664
60	23	28.316
70	31	38.541
80	41	50.339
90	52	63.711
100	64	78.654
110	77	95.172
120	92	113.2632

$$S = \frac{V^2}{\mu g}, \; \mu = 0.6, \; S = 1.083\, V^2 \quad (18)$$

$$\eta_b = \frac{f}{g} \times 100\% \quad (19)$$

Where, f = retardation due to braking force
S = brake distance
η_b = braking efficiency

VI. DESIGN CALCULATION OF FRICTION FORCE AND BRAKING TORQUE

The total kinetic energy absorbed is,

$$E_k = \frac{mv^2}{2} \quad (20)$$

= 241608.8 N-m

Changes in vehicle potential energy (assume practically of slope = 0.2),

$$E_p = mgh \times slope = 245.74 \text{ kN} - m$$

Total energy absorbed by the brake is,

$$E = E_k + E_p = 487.35$$

Tangential braking force is,

$$F_t = \frac{E}{S} = 9746.99 \text{ N}$$

The average braking torque is,

$$T_b = F_t + R_b \quad (21)$$

Frictional force on the brake drum is,

$$F_b = \frac{F_t \times R_t}{R_b} \quad (22)$$

Normal force on the brake shoe is, μ_b = 0.5

$$W_b = \frac{F_b}{\mu_b} \quad (23)$$

In equilibrium position,

Radius of frictional force, k = 24 cm
Distance between anchor pin and actuating forces, m = 30 cm
Distance of the anchor pin from the drum centre, n = 16 cm
Free ends of the two shoes exerted force, W_α = 32 kg
Equilibrium state as $W\alpha = Wt = W1$ and $\theta = 90°$, μ_f = 0.5

Normal force between shoe and drum for leading shoe is,

$$P_1 = \frac{W_1 \times m}{n \sin\theta + \mu_f(k - n\cos\theta)} \quad (24)$$

Braking torque due to leading shoe is,

$$T_1 = \frac{W_1 \times m \times k \times \mu_f}{n \sin\theta - \mu_f (k - n\cos\theta)} \quad (25)$$

Normal forces between shoe and drum for trailing shoe is,

$$P_t = \frac{W_t \times m}{n \sin\theta + \mu_f(k - n\cos\theta)} \quad (26)$$

Braking torque due to trailing shoe is,

$$T_t = \frac{W_t \times m \times k \times \mu_f}{n \sin\theta + \mu_f (k - n\cos\theta)} \quad (27)$$

TABLE IV
RESULTS OF FRICTION FORCE AND BRAKING TORQUE

Average braking torque	1267.1082 N-m
Frictional force on the brake drum	19.8688 kN
Normal force on the brake shoe	39.6877 N
Shoe and Drum for leading shoe	2354 N
Braking torque due to leading shoe	282.528 N-m
Shoe and Drum for trailing shoe	336.343 N
Braking torque due to trailing shoe	40.3611 Nm

A. Design calculation of reaction forces on front and rear wheels

When brakes are applied to the rear wheels;
Weight of vehicle, W = 15107 N
Wheel based, b = 2.250 m
Vehicle CG (x, y) = (1955, 848)
Lengthwise distance of centre of gravity from the rear axle is, l = 0.295 m
Reaction force between the ground and the front wheel,
Cos θ□= 1 (approx.)

$$R_f = \frac{(1 + \mu h)\cos\theta \times W}{(b + \mu h)} \quad (28)$$

Reaction force between the ground and rear wheel,

$$R_R = \frac{(1 - l)\cos\theta \times W}{(b + \mu h)} \quad (29)$$

Reaction force between the ground and front wheel,

$$R_f = \frac{l \cos\theta \times W}{(b - \mu)} \quad (30)$$

Reaction force between the ground and the rear wheel is,

$$R_R = \frac{(b-l-\mu h)\cos\theta \times W}{(b-\mu h)} \qquad (31)$$

When brakes are applied to all the four wheels.

For front wheel reaction force,

$$R_f = \frac{(b+\mu h)\cos\theta \times W}{b} \qquad (32)$$

For rear wheel reaction force,

$$R_R = \frac{(b-l-\mu h)\cos\theta \times W}{b} \qquad (33)$$

TABLE V
RESULTS OF REACTION FORCE ON FRONT AND REAR WHEELS

Ground and front wheel	4853.855 N
Ground and rear wheel	10253.145 N
Ground and front wheel	2751.815 N
Ground and rear wheel	12355.185 N
For front wheel	6214.013 N
For rear wheel	8892.9873 N

B. Design calculation of master cylinder and force on wheels

Master cylinder bore = 19.05 mm

Practically foot force, FFoot = 100 lb = 444.947 N

Practically experience data, r1 = 280 mm and r2 = 36 mm

Wheel cylinder area,

$$A_w = \frac{\pi d^2}{4} \qquad (34)$$

$= 5.07\times10^{-4}\ m^2$

Pressure on the wheel cylinder,

$$P_w = \frac{F_w}{A_w} \qquad (35)$$

$= 12.19\ N/mm^2$

By Pascal's Law,

$$\frac{F_w}{A_w} = \frac{F_{MC}}{A_{MC}} \qquad (36)$$

$d_{MC} = 19$ mm

Front wheel master cylinder diameter = 19 mm

Force on wheel cylinder (rear wheel), F2 = 4119.225 N

Wheel cylinder diameter (rear) = 2064 mm = 0.02064 m

Wheel cylinder area,

$$A_w = \frac{\pi}{4} d^2 = 3.45 \times 10^{-4} m^2$$

Pressure on wheel cylinder,

$$P_w = \frac{F_w}{A_w}$$

By Pascal's Law,

$$\frac{F_w}{A_w} = \frac{F_{MC}}{A_{MC}}$$

VII. DESIGN CALCUATION OF BRAKE MEAN LINING PRESSURE AND HEAT GENERATION

Width of braking lining, ω= 45 mm

Constant angle of each lining, α□= 240

Coefficient of friction between shoe and drum, μ= 0.4

Braking mean lining pressure =

$$\frac{180\times f\times r}{8\pi\times\mu\times a\times\omega\times\alpha} \qquad (37)$$

$$\text{Generated heat} = \frac{F\times \text{Speed per minute}}{4\times427} \qquad (38)$$

A. Design Calculation of Vehicle Centrifugal Force

$$\text{Centrifugal force}, F_C = \frac{wv^2}{gc} \qquad (39)$$

TABLE VI
RESULTS FOR MASTER CYLINDER, BRAKE MEAN LINING PRESSURE, HEAT GENERATION AND CENTRIFUGAL FORCE

Wheel cylinder area, Aw	3.346×10-4 m^2
Pressure on wheel cylinder, Pw	12.1137 N/mm^2
Master cylinder diameter, d_{MC}	19 mm
Braking mean lining pressure	0.17325 N/mm2
Generated heat	3251.601 kcal/min at 50 km/hr
Centrifugal force, F_C	29070.865 kg at 50 km/hr

VIII. CONCLUSION

The driver must select the steering wheel angle to keep deviation from the desired course low. However, there is no definite functional relationship between the turning angle of the steering wheel made by the driver and the change in driving direction, because the correlation of the facts (a) turns of the steering wheel, (b) alteration of steer angle at the front wheels, (c) development of lateral tyro forces, and (d) alteration of driving direction. This results from elastic compliance in the components of the chassis. To move a vehicle, the driver must continually adjust the relationship between turning the steering wheel and the alteration in the direction of the travel. The most important information the driver receives comes via the steering moment or torque which provides the driver with feedback on the forces acting on the wheel. Therefore, the job of the steering system to convert the steering wheel angle into as clear a relationship as possible to the steering angle of the wheels and to convey feedback about the vehicle's state of movement back to the steer.

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