TURBO BRAKE ASSISTS – TURBOTOR: CONCEPTUAL DEVELOPMENT AND EVALUATION

G. Ramkumar and Amrit Om Nayak
Student, Department of Mechanical Engineering, Thiagarajar College of Engineering, Madurai, India

ABSTRACT
'Turbotor' employs the principles of turbo machinery and automotive engineering in order to achieve effective braking assistance for the modern day disc as well as drum brakes. These brakes are intended for the purpose of braking assists during high speed motion. The major components include a turbine or rotor, pump, nozzle, enlarged brake fluid chamber, DOT 5.1 braking fluid. The functioning involves directing a jet of high velocity brake fluid (DOT 5.1 in this case) at the blades of the rotor which is coupled with the wheel. The braking circuit is used as such with the enlargement of the brake fluid chamber. A pump is added to the set up to enable increase of mass flow rate of fluid and also to ensure the continuity of fluid flow in the entire circuit. A speed sensor is integrated along with the circuit to activate the pump such that the braking assists are put to use only after a particular velocity is reached as braking assists are not necessary at lower speeds.

KEYWORDS: *brake Assists, brake fluid, mass flow rate, and nozzle*

I. INTRODUCTION

The modern era has seen unprecedented technological growth and has propelled automotives on an ever rising curve of speed. Still, despite chargers, turbochargers, twin turbochargers or NO_X, there are limits which cannot be surpassed by a land based vehicle in terms of speed. Brakes on the other hand are a part of automobiles that abide by only those limitations imposed to them by the ability of human body to withstand rapid decelerations. Otherwise, it would be a lot easier stopping a car than making it go insanely fast.
The drum based braking system can be considered to be the forefather of the modern day brake.The man largely credited with the development of the modern day drum brake is French manufacturer Louis Renault, in 1902. This braking systems was external, a feature which soon turned into a problem. Dust, heat and even water rendered them less effective. It was time for the internal expanding shoe brake. By placing the shoes inside the drum brake, dust and water were kept out, allowing the braking process to remain effective.
As the vehicles spilled out the assembly plants, they started becoming both faster and heavier. The earlier brakes were effective, but they had a tendency to ineffectively distribute heat. This feature made room for the creation of the disc braking system**.**
The '*turbotor'* will take the development process in the field of automotive braking a step further. It is an attempt to reduce our dependence on frictional braking and thereby reduce wear and tear of brake components. The '*turbotor'*utilizes the concepts governing turbo machinery and automotive engineering in order to achieve effective braking assistance for the modern day disc as well as drum brakes. The principle involved encompasses the targeting of a jet of viscous fluid onto the blades of a turbine which is attached to the wheel. This jet of fluid retards the motion of the turbine and in the process damps or slows down the rotation of the wheels. The '*turbotor'* is a potential brake assist in high speed conditions thereby reducing the work done by the disc or drum brakes. This results in decreased braking distance as well as braking time. The major components include a turbine or rotor, pump, nozzle, enlarged brake fluid chamber, DOT 5.1 braking fluid. The functioning involves

directing a jet of high velocity brake fluid (DOT 5.1 in this case) at the blades of the rotor which is coupled with the wheel. The braking circuit is used as such with the enlargement of the brake fluid chamber. A pump is added to the set up to enable increase of mass flow rate of fluid and also to ensure the continuity of fluid flow in the entire circuit. A speed sensor is integrated along with the circuit to activate the pump such that the braking assists are put to use only after a particular velocity is reached as braking assists are not necessary at lower speeds.
The concept development and evaluation of *"turbotor"* is organised into six sub-sections. The preliminary section describes the *Theory* behind the functioning of turbotor, and is followed by the second section which discusses the *Construction* of Turbo Brake Assist system and the integral components that form the heart and soul of the system. The third section illuminates the *Working* principles and driving concepts of the system. The detailed *Mathematical study of existing braking systemand turbotor* is illuminated in the fourth section. The fifth section consists of *Results and Discussion* which throws light on the inferences from mathematical study of the brake assist system. The sixth section is the *Conclusion* which elucidates the practicality of the concept and performance of turbotor as an effective brake assist.

II. THEORY

We deem the '*turbotor'* to be a success even if it caters to at least 35 per cent of the braking force from the disc as well as the drum brakes. The main purpose of this system is to function as a braking assist. If the '*turbotor'* is capable of providing excessive braking which is close to that obtained from the primary brake system, this would lead to locking of the wheels and very sudden deceleration resulting in utmost discomfort for the driver. Therefore the working of this system will be limited to a velocity range in order to accommodate for the above mentioned features.
The total braking force obtained from the front disc brakes and the rear drum brakes is calculated independently. The braking force obtained from the '*turbotor'* is calculated by evaluating the pressure density or force of the jet of viscous fluid (DOT 5.1 braking fluid) on the turbine blades. The weight distribution between front and rear axles during dynamic braking is calculated. This will enable us to decide the amount of braking assist required at the front and the rear wheels of the vehicle. The use of DOT 5.1 braking fluid will ensure that heat generated during damping of turbine motion will not pose a threat as it is hygroscopic in nature. The concept is explained in detail in the following sections.

III. CONSTRUCTION

The integral parts of the Brake Assists are depicted in the schematic diagram Fig.1and the components are explained subsequently.

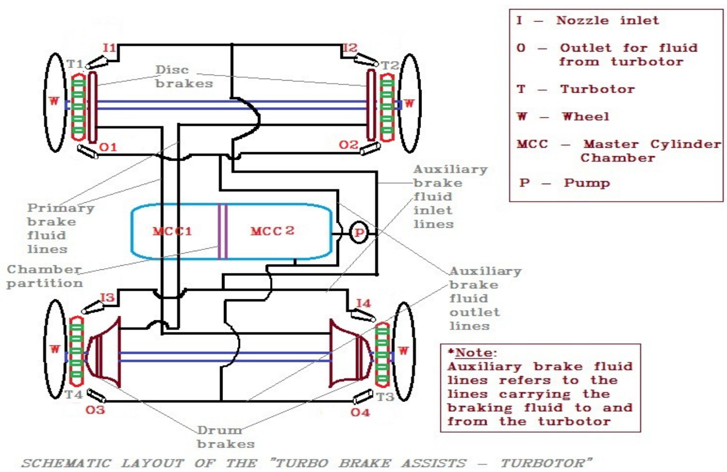

Fig.1 Turbotor schematic layout

3.1.Master cylinder chambers:

The *Turbotor* setup consists of a master cylinder which is partitioned into two chambers in order to provide undisturbed volumes for normal braking system as well as for *Turbotor*. The position of the master cylinder chambers is the same as in the existing braking system. The master cylinder chamber for *Turbotor* system is provided with pipelines to carry brake fluid to and fro from the rotor setup. These pipelines for the *Turbotor* are called auxiliary brake fluid lines(inlet and outlet). The other chamber of master cylinder is provided with pipelines for disc and drums brakes and is thus left undisturbed.

3.2. Auxiliary brake fluid pipelines:

The auxiliary brake fluid pipe lines are connected to the master cylinder chamber for turbotor at one end and the other end consists of nozzle to impart high velocity to the brake fluids at the striking end to the turbotor. There are four inlet and outlet brake fluid lines attached to the master cylinder chamber for turbotor. The outlet pipe lines from the turbotor carrythe brake fluid that is accumulated after striking the turbotor to the master cylinder chamber for turbotor. The auxiliary inlet pipelines to the turbotors at the front and rear is symmetric at the point along its longitudinal length as shown in the Fig. 1. This symmetric arrangement ensures equal mass flow rate of fluid to both left and right turbotors at front and rear axles.

3.3.Hydraulic pump:

The hydraulic pump that is going to be used here is a diaphragm type positive displacement reciprocating pump. A diaphragm pump is a positive displacement pump that uses a combination of the reciprocating action of a rubber, thermoplastic or teflon diaphragm and suitable non-return check valves to pump a fluid. Sometimes this type of pump is also called a membrane pump.

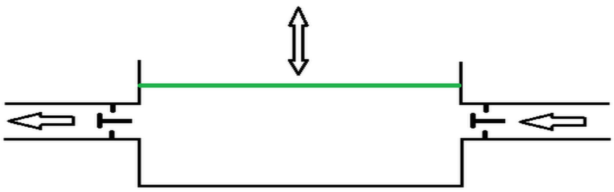

Fig.2. Diaphragm pump schematic

When the volume of a chamber of either type of pump is increased (the diaphragm moving up), the pressure decreases, and fluid is drawn into the chamber. When the chamber pressure later increases from decreased volume (the diaphragm moving down), the fluid previously drawn in is forced out. Finally, the diaphragm moving up once again draws fluid into the chamber, completing the cycle. This action is similar to that of the cylinder in an internal combustion engine.

3.4. Turbotor:

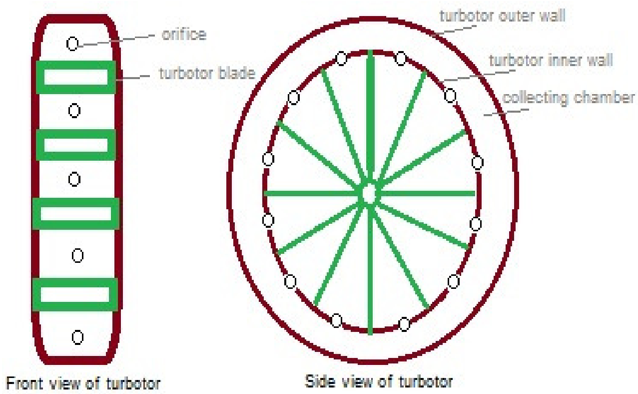

Fig.3. Tubotor schematic

The turbotor schematic is shown in Fig. 3. After the fluid jet hits the turbotor blades, it is allowed to pass through the orifice holes and collect in the collecting chamber. This collected brake fluid is now sent back via the outlet turbotor outlet back to the MCC 2 as shown in Fig. 1.

IV. WORKING

The overall working of the system is quite simple. The brake fluid is pumped from MCC 2 and is sent via the auxiliary inlet lines to the nozzle. High pressure fluid is sprayed on the turbotor blades which in turn decelerates the blades. Since the turbotor is coupled to the axle, it in turn causes overall deceleration of the vehicle. The brake fluid collects in the collecting chamber of the turbotor through the orifices and is sent back to the MCC 2 via the auxiliary fluid outlet lines with the help of the pump. This entire turbotor system aids the primary braking system and thereby leads to reduction in braking time and braking distance. This also reduces the wear and tear experienced by the primary braking system.

V. MATHEMATICAL STUDY OF EXISTING BRAKE SYSTEM

NOTE: The numerals beside the equations are intended to signify the number of equations only.

5.1. Drum brakes braking equations:

The maximum wheel torque is limited by wheel slip and is given by:

$$T_w = \mu_a WR \quad (1)$$

Where,

$$T_w = wheel\ retarding\ torque(Nm)$$
$$\mu_a = adhesion\ factor$$
$$W = vertical\ load\ on\ wheels(N)$$
$$R = Wheel\ rolling\ radius(m)$$

The torque produced at the drum brakes caused by friction between the lining and the drum which is necessary to bring the vehicle to a standstill is given by:

$$T_B = \mu Nr \quad (2)$$

Where,

$$T_B = Braque\ drum\ torque(Nm)$$
$$\mu = coefficient\ of\ frictio\ between\ lining\ and\ drum$$
$$N = radial\ force\ between\ lining\ and\ drum(N)$$
$$r = drum\ radius(m)$$

Both wheel and drum torques must be equal up to the point of wheel slip but they act in opposite direction to each other. Therefore they may be equated.

$$T_B = T_W \quad (3)$$
$$\mu Nr = \mu_a WR \quad (4)$$

Therefore Force between lining and drum is:

$$N = (\mu_a WR)/(\mu r)(\mathrm{N}) \quad (5)$$

The above mentioned equation is the braking force produced by drum brakes in order to bring the vehicle to a stop.

5.2. Disc brakes braking equations

The normal clamping thrust N on each side of the disc acting through the pistons multiplied by coefficient of friction μ generated between the disc and pad interfaces produces a frictional forceon both sides of the disc.

$$F = \mu N \text{in N} \quad (6)$$

If the resultant frictional force acts through the centre of the friction pad the mean distance between centre of the pad pressure and centre of the disc will be

$$(R_2 - R_1)/2 = R \text{in m} \tag{7}$$

Accordingly, the frictional braking torque will be dependent upon twice the frictional force (on both sides) and the distance the pad is located from the disc centre of rotation. That is,

$$Braking\ torque = 2\mu NR(\text{Nm}) \tag{8}$$

i.e. , $T_B = 2\mu NR$(Nm)

5.3. Braking dynamics of Turbotor

The maximum sustained deceleration required by the specifications is 0.65g(where g is the acceleration due to gravity and is taken as 9.81 m/s^2)

The braking force required to bring the car to a stop is given by,

$$F_b = ma \text{in kN} \tag{9}$$

The average power absorbed by the brakes is given by,

$$P_{avg} = F_b V_o/2 \text{in kW} \tag{10}$$

The static weight distribution of the car is given by,

Weight on the front axle:

$$W_f = mgd/(c+d) \tag{11}$$

Weight on the rear axle:

$$W_r = mgc/(c+d) \tag{12}$$

Assuming that the car is established, under steady–state braking, application of Newton's law provides the equations to determine the dynamic weight distribution on the front and rear axles during braking.

$$W_{f'} = mgd/(c+d) + ma'h/(c+d) \tag{13}$$

$$W_{r'} = mgc/(c+d) - ma'h/(c+d) \tag{14}$$

NOTE*: All weight distributions are in kN.

The following diagram indicates the parameters as noted in above formulae

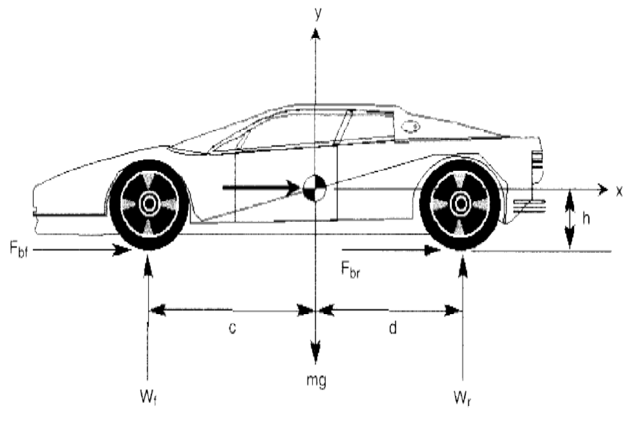

Fig. 4 Vehicle force diagram

5.4. Mathematical study of proposed brake assist system

We know that the braking power obtained at the conventional brakes is the maximum braking power that can be provided at the turbotor for it to act safely as a brake assist. This power is carried to the turbotor by the brake fluid.

The brake power (in kW) that needs to be imparted to the brake fluid is given as:

$$P = F * V' \tag{15}$$

Where F = braking force in kN and

V'= velocity of the brake fluid in m/s

This is the velocity of the brake fluid (in m/s) obtained at the nozzle outlet in the auxiliary inlet fluid line. The head developed in the nozzle is given be

$$V'^2 = 2gH \tag{16}$$

Where g = acceleration due to gravityin m/s^2

And H = head developed in m

The necessary power to be given to the brake fluid is obtained with the help of a suitable reciprocating pump (as described in Fig. 1) which is similar to the fuel pumps in use today. The power rating (in kW) of the pump is given by

$$P = (\rho g * ALN * H')/(60 * 1000) \tag{17}$$

Where ρ = density of the brake fluid in kg/m^3

A= area of cross-section in m^2

L = stroke length in m

N = rpm of the pump

H' = total head=$H_s + H_d$

Where

H_s = suction head in m

H_d= delivery head in m

The head obtained at the nozzle outlet is the delivery head of the pump (without considering the losses in pipeline). From the above equation we can obtain the value of suction head of the pump. The pump is similar to fuel pump considering Head in terms of pressure. Head in terms of pressure can be obtained using the formula,

$$H = (P * 10.197)/SG \tag{18}$$

Where,

H = head in m

SG = Specific gravity of the brake fluid

P = pressure of the fluid in bar

Since turbotor spins at the same rpm of the wheel, Rpm of the turbotor can be obtained from:

$$vehicle\ speed\ (in\ miles/hr) = (0.00595 * RPM\ of\ the\ wheel * radius\ of\ the\ wheel)/(gear\ ratio * Final\ drive\ reduction\ ratio) \tag{19}$$

$$u = \pi dn/60 \tag{20}$$

Where,

u = tangential velocity of the turbotor in m/s

d = diameter of the turbotor in m

n = Rpm of the turbotor

From the velocity triangle of the turbotor blades, Fig 5, we get the following equations,

$$V_{w1} = V_1 \tag{21}$$

Where,

V_1 = Absolute velocity of the brake fluid at inlet of the blade in m/s

V_{w1}= brake fluid whirl velocity at the inletof the blade in m/s

$$V_{w2} = (V_1 - u)\cos\varphi - u \tag{22}$$

Where,

V_{w2}= whirl velocity of the brake fluid at outlet in m/s

u = tangential velocity of the turbotor in m/s

φ = angle of deflection of the brake fluid from the blade surface

$$F_x = \rho Q(V_{w1} + V_{w2}) \tag{23}$$

Where,

F_x= braking force given by the jet of brake fluid in kN

$$\text{And} \quad Q = a * V' \tag{24}$$

Where,

Q = discharge of the brake fluid in m^3/s

a = area of cross-section in m^2

V'= velocity of the brake fluid at the nozzle outlet in m/s

$$W_x = \rho Q(V_{w1} + V_{w2}) * u \tag{25}$$

Where,

W_x= work done by the brake fluid in kNm

The velocity diagram for the blade is shown below:

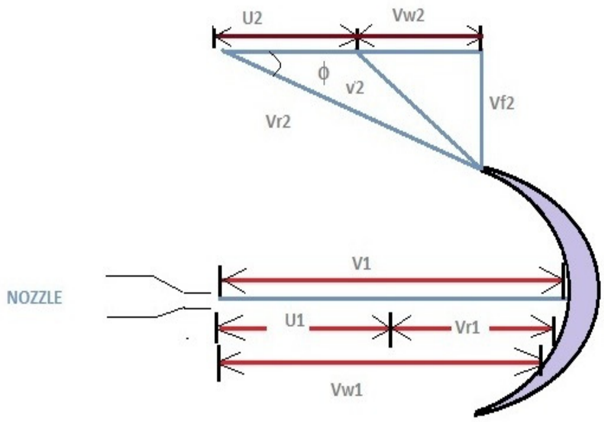

Fig. 5 Velocity diagram for blade

In the above velocity diagram,

V_1= absolute velocity of the brake fluid at inlet in m/s
U_1= turbotor wheel velocity in m/s
V_{w1}= whirl velocity of the brake fluid at inlet in m/s
V_{r1}= relative velocity of the brake fluid with respect to blade at inlet in m/s
V_2= absolute velocity of the brake fluid at outlet in m/s
U_2 = turbotor wheel velocity in m/s
V_{w2} = whirl velocity of the brake fluid at outlet in m/s
V_{r2}= relative velocity of the brake fluid with respect to blade at outlet in m/s
And U_1=U_2= U = turbotor wheel velocity in m/s
V_{r1}= V_{r2}, neglecting the effect of friction between fluid and blade.

The heat developed in turbotor during braking is given by the following set of equations:

$$Deceleration\ due\ to\ turbotor = (35\%\ of\ B_f)/m \tag{26}$$

Where,

B_f = braking force in kN

m = mass of the vehicle in kg

$$t = (v - u)/a \tag{27}$$

Where,

t = time taken for deceleration in s

v = velocity of the vehicle before braking in m/s

u = velocity of vehicle after braking in m/s

$$E = P * t \tag{28}$$

Where,

E = heat energy absorbed by brakes in kJ

P = power absorbed by the brake fluid during braking in kW

t = time taken for deceleration in s

Considering the energy absorbed by a single turbotor at front axle, we have:

$$E_{turbotor} = (P * weight\ distribution\ ratio)/2 \tag{29}$$

The temperature change in the turbotor during braking is given by:

$$E = mC_v\Delta T \tag{30}$$

Where,

E = heat energy generated in turbotor in kJ

m = mass of the turbotor in kg

C_v= specific heat at constant volumeof material of the turbotor in kJ/kg.K

ΔT = change in temperature of the turbotor

VI. RESULTS AND DISCUSSION

Here Ford Fiesta hatchback edition is taken as a case-study for the Turbo Brake Assists. The velocity for actuation of turbotor is taken as 19.44 m/s. We now assume that the vehicle is moving with an initial velocity of 27.78 m/s. With mass of the car as 1110kg and deceleration of 6.3765 m/s^2 and the braking force obtained from the conventional braking system is 7.0779 kN.

Now considering 35 per cent of the existing braking force to be obtained by Turbo Brake Assists, the braking force is calculated as 2.4773 kN. This is the braking force necessary to be impacted by the brake fluid to the turbotor.

Considering 59.7 per cent of braking force at front axle (as the weight distribution ratio for Ford Fiesta is 59.7/40.3 in percentage), the braking force at the front wheels is obtained as 1.4789 kN. At each turbotor in the front axle, the braking force is obtained as 0.7395 kN.

The average power absorbed by the conventional brakes during braking is the average power absorbed by the brake fluid during braking in Turbotor.Therefore the average power to be imparted to the brake fluid at each turbotor in the front wheels is calculated withF_b= 0.7395 and V_o= 27.7778 m/s. and is obtained as 10.27036 kW.This average power has to be given to the brake fluid in order to perform braking efficiently.The power to be absorbed by the brake fluid during braking is equal to the power given to the brake fluid before striking the turbotor, and the velocity is calculated as 13.8889 m /s. This is the velocity obtained at the nozzle out let before striking the turbotor.

The above calculations give the necessary design conditions and values for desired efficient performance of Turbo Brake Assists i.e., 35 per cent of existing braking force. And in order to achieve this, a power of 10.27036 kW is to be given to the brake fluid for obtaining a velocity of 13.8889 m/s. The power is provided by the reciprocating hydraulic pump.The Head developed at the nozzle outlet is calculated as 9.8319m. This head is the delivery head for the pump. The total head for the pump is calculated usingρ = 1050 kg/m^3(property of DOT 5.1 brake fluid), g = 9.81 m/s^2, Diameter of the cylinder = 64 mm, L = 175 mm, N = 5000 rpm, P = 10.27036 kW and the total head is calculated as 20.61456 m. In terms of pressure, the total head, suction head, delivery head are calculated as 2.1227 bar, 1.11031bar and 1.01240 bar respectively. Now considering the turbotor blade part, the absolute velocity of the brake fluid at inlet of the blade is obtained as 13.88890001 m/s. The tangential velocity of the wheel is obtained as 85.426471 m/s. With radius of the wheel = 7.5 inches, gear ratio at 5th gear is 0.756, final drive ratio is 4.07:1, and vehicle speed is 62.137119 miles/hr and value of n = 4284.3909 rpm.The whirl velocity of brake fluid at the blade outlet is obtained as 18.2031434 m/s. Here angle of deflection, is taken as 20 degrees.

NOTE*: Since turbotor tangential velocity is greater than brake fluid velocity, the negative sign obtained in the values are omitted and only magnitudes are considered.

The force imparted by the jet of brake fluid onto the blade surface is obtained as 0.58782 kN. Work done by the brake fluid is obtained as 50.21538792 kW. The above value is obtained for a nozzle diameter of 0.04m. The time taken for deceleration is calculated as 3.73398 s, considering v = 27.7778 m/s and u = 19.444 m/s. This time is considered for the braking force achieved using turbotor system entirely as shown previously i.e. 2477.27025 N.

The heat energy absorbed by the turbo amounts to 128.47335 kJ. And each turbotor at front absorbs 38.3493 kJ.Therefore the temperature change in turbotor is obtained as 20.18384 K with m = 4 kg, C_v = 0.475 KJ/kg-K.

VII. CONCLUSION

From the mathematical results obtained, when a nozzle diameter of 0.04 m (which is a practical figure) is considered the DOT 5.1 brake fluid is able to impart necessary force i.e. 2.35128 kN, which is close to the desired performance value of 2.47727025 kN (35 per cent of existing braking force from conventional braking system), on the turbotor blades there by causing deceleration of the vehicle as a whole and bringing about effective braking assistance to the existing braking system. This reduces the work done by the conventional braking system at high speeds. Hence Turbotor can be practically employed with the existing braking system as effective braking assistance. Another important inference from the mathematical study is that the velocity of brake fluid remains constant at any ratio of braking power and braking force. By varying the nozzle diameter and angle of deflection, the braking force of the brake fluid can be varied accordingly.

REFERENCES

[1]. Richard Stone and Jeffrey K. Ball (2004), “Automotive Engineering Fundamentals”, SAE International, Warrendale, Pa.
[2]. Dr.R.K.Bansal (2005), “A Textbook of Fluid Mechanics and Hydraulic Machines S.I.Units”, Laxmi Publications (P) Ltd., New Delhi.
[3]. Heinz Heisler (2002), “Advanced Vehicle Technology”, Butterworth-Heinemann, Woburn, MA.
[4]. Yuvan.S.W, Foundation of Fluid Mechanics, Prentice Hall, 1998.
[5]. Ellis.J.R, Vehicle Dynamics, Business Books Ltd., London, 1991.
[6] .Rogers, G.F.C. and Mayhew, Y.R. (1967), Engineering Thermodynamics Work and Heat Transfer, Longman.
[7]. Limpert, R. (1999), Brake Design and Safety, Society of Automotive Engineers, Warrendale, Pa.
[8]. SAE Handbook (1989), Volume 2, Society of Automotive Engineers, Warrendale, Pa.
[9]. Gillespie, T.D. (1994), Fundamentals of Vehicle Dynamics, Society ofAutomotive Engineers,Warrendale, Pa.

Authors

G. Ram Kumar was born in Chennai, Tamil Nadu, India, in 1991. He is currently pursuing Bachelor of Engineering degree in sixth semester with the Department of Mechanical Engineering, in Thiagarajar College Of Engineering, Madurai, Tamil Nadu, which is affiliated to Anna University, Tamil Nadu. His research interest includes Automotive Technology and Engineering, Thermal Engineering, blended wing design in aircrafts, turbo machinery and theoretical physics.

Amrit Om Nayak was born in Bhubaneswar, Odisha, India, in 1991. He is currently pursuing Bachelor of Engineering degree in eighth semester with the Department of Mechanical Engineering, in Thiagarajar College of Engineering, Madurai, Tamil Nadu, which is affiliated to Anna University, Tamil Nadu. His research interests includes Automotive Technology and Engineering, Thermal Engineering, blended wing design in aircrafts, turbo machinery and theoretical physics.