CHAPTER 2
LITERATURE REVIEW

Transportation is crucial to the development and defense of societies and nations. This is the reason why many countries are developing unmanned systems such as unmanned fighter jets to replace piloted fighter jets, robots to replace human soldiers and autonomous vehicles to transport materials. The tire-road friction coefficient, $\mu$, plays a significant role in vehicle stability control. With estimated friction coefficients, vehicle motion can be estimated more accurately. Dieckman (1992) developed a method to determine the road surface variation based on the measurements of the wheel slip ratio.

Gustafsson (1977) designed an algorithm to estimate the tire-road friction during normal driving using the measured wheel slip ratio along with Kalman filter. Eichhorn & Roth (1992) introduced the estimation of road friction using optical and noise sensors near the front-end of the tire, and stress-strain sensors inside the tire tread. In terms of control, Hallowell & Ray (2003) proposed a nonlinear estimator and controller to estimate the vehicle state, and friction coefficient, and implement the stability controller by distributing driving torques on the wheels.

Road surface can be generally classified into five possible conditions: Asphalt, Cement, Sand, Grass and rough. Each of these conditions has distinct characteristics in friction coefficient. There is a relationship
between the surface friction coefficient and the vehicle’s wheel slip ratio. The relationship has been experimentally determined by Gustafsson (1977).

A model can be designed to estimate the vehicle velocities and yaw rate when direct measurements are not available, or inaccurate due to signal noise and drift. However, the existing models are either for linear systems or requiring exact knowledge of dynamic system model.

The braking system must accomplish three different tasks:

- To stop the vehicle completely; this function entails braking moments that are as strong as possible on the wheel.
- To control speed, when the natural deceleration of the vehicle due to mechanical friction and motion resistance is not sufficient; this function entails braking moments on the wheels that are moderate, but applied for a long time.
- To keep the vehicle stopped on a slope.

Because of the nature of these tasks the braking system is one of the safety systems of the vehicle. As a consequence, the State Authority and, later, the European Union have introduced regulations that describe design conditions and minimum operational requirements for this system.

Vehicle manufacturers and their component suppliers are, therefore, responsible for compliance of their products to regulations, including correct fabrication and system reliability for a reasonable period of time. Users, too, must play their role because many parts of this system are subject to wear and the safety functions cannot be assured without the necessary maintenance and replacement of parts. A periodic compulsory control is addressed to assessing the correct operation of this system.
If regulations determine minimum performance for this system, each manufacturer considers this only as a starting point, because more stringent requirements are demanded by the market and can be remarkable selling points.

Because of this, braking systems have reached in normal practice high levels of performance and reliability.

It should be remarked that the relationship between brake reliability and accident probability is not very evident; statistics on road accidents show, in fact, that less than 2% of road accidents are caused by inadequate operation of the braking system.

Within this total, 90% of accidents are estimated to be due to insufficient maintenance and 10% to dynamic instability, consequent to a braking event incompatible with transverse accelerations.

The wide application of anti-lock systems (ABS) represents an improvement for braking safety, even if accident statistics contain insufficient witness of this fact.

Studies of accidents in Germany, after the introduction of this system in 1976, showed a reduction for vehicles equipped with it; most recent data show that the presence of ABS leads users to overrate its contribution and, therefore, to expose themselves to dangers, particularly in situations such as icy roads or driving with reduced safety distance.

Another German study on taxis show no relevant difference between cars with and without ABS.

Before starting the description of braking system components, prefer to introduce some preliminary considerations on braking system
design. These will be better explained in the second volume of this book, which is dedicated to system design.

The maximum longitudinal force exchanged with the ground depends upon many factors, the vertical load on the wheel foremost among them. This force is influenced by the nature of ground, the speed of the vehicle and the coexistence of cornering forces.

The maximum performance of this system is, therefore, conditioned by many factors outside the system itself; while the recognition of the latter is the job of drivers, who are in charge of limiting vehicle speed and controlling the distance between close vehicles, vertical loads cannot be easily understood.

These are determined by different factors, such as:

- Payload and its distribution in the vehicle
- Road slope
- Longitudinal acceleration, in particular, the same braking acceleration

To highlight this fact, let us consider a vehicle of mass m, moving on a flat road, inclined by a slope angle $\alpha$. 
Assume that the vehicle is symmetric with reference to its median plane, while its center of gravity G has a distance a from the front axle and b from the rear axle; the center G is high $h_G$ on the ground (see Figure. 2.1); finally:

$$l = a + b,$$

is the vehicle wheelbase.

Now assume that all acting forces are negligible, except for braking forces $F_{x1}$ and $F_{x2}$, applied to the front and rear axle. Each of these forces represents the sum of those acting on the wheels of the same axle, which in assumptions are here equal.

Indicate with $F_{z1}$ and $F_{z2}$ the total vertical force on each axle and with $ax$ the braking acceleration (it should be remarked than it is negative, because it is braking).
\[ F_{x1} + F_{x2} - mg \sin \alpha = \text{max} \]
\[ F_{z1} + F_{z2} - mg \cos \alpha = 0 \]
\[ F_{z1}a - F_{z2}b + mghG \sin \alpha = mhGax \]

From this system:

\[ F_{z1} = \frac{m}{L} (gb \cos \alpha - ghG \sin \alpha - hGax) \]
\[ F_{z2} = \frac{m}{L} (ga \cos \alpha + ghG \sin \alpha + hGax) \]

These equations indicate clearly that vertical reaction forces depend not only on the center of gravity position, but also on the vehicle braking deceleration.

In the chapter on tires the maximum obtainable longitudinal force is proportional, through the friction coefficient \( \mu_{xp} \) and the peak value of the curve \( \mu_{x}(\sigma) \), to the vertical load on the wheel.

As a consequence, the following conclusions can be drawn:

- Maximum braking forces are determined by m, b, a, hG (load conditions).
- These forces are also determined by the braking deceleration ax.
- Braking force distribution between the two axles is also determined by the obtained longitudinal acceleration.

Therefore, a braking system should be able to change its geometry to adapt braking forces to existing vertical loads. In particular the braking
force on the rear axle should be progressively limited when the total braking force is increased.

This function is performed by the brake distributor valve, which will be described in the following paragraphs; the ABS system can also manage this operation with precision.

2.1 CAR BRAKES

According to regulation, the functions mentioned at the beginning of the previous paragraph are to be accomplished in a vehicle by three different systems, which cannot be directly matched to them; these are:

- **Service braking system**, able to reduce speed or stop the vehicle in normal driving conditions
- **Emergency or secondary braking system**, suitable for the same above function, but to be used in case of failure of the service brake
- **Parking braking system**, suitable for parking only, adaptable to slopes

All these systems must exert an adjustable braking force on the vehicle. Many components are common to the first two braking systems, but some of them are specific; redundancies guarantee reliability and availability of the braking function.

2.1.1 Service and Secondary Systems

Figure 2.2 shows the scheme of a hydraulic braking system; it includes the service and secondary systems.
The front and rear brakes are actuated by two completely independent hydraulic circuits A and B; this separation is also present in the master pump C, actuated by the pedal and enlarged in the picture at the lower left.

This feature, imposed de facto by law, allows the two circuits to be defined, when they operate together in normal conditions, as service circuits and, when they operate separately (when one of the two has failed) as emergency circuits.

The kind of failure considered in this approach is the rupture of one of the flexible tubes connecting the brake actuators, moving with the wheels, with the rest of the circuit, fixed to the vehicle chassis structure; if, for example, the tube connecting the left front wheel should fail, at the first braking, a defined quantity of oil may be squeezed out, but the rear circuit would be still able to function.

Figure 2.2 Pictorial scheme of a car braking system
The homologation rules for a car request that, on a perfectly paved road, this stopping distance must be obtained:

\[ s \leq 0.1V + \frac{V^2}{150}, \]

where \( s \) is the stopping distance, measured in (m) and \( V \) is the car speed, measured in (km/h), at the onset of braking.

Front and rear brakes are operated by two independent hydraulic circuits A and B. C shows an enlargement of the master pump and D the distributor.

This condition must be met by car service system; an increased distance:

\[ s \leq 0.1V + \frac{2V^2}{150} \]

is allowed when one of the circuits is broken, at which point only the emergency system is functional.

Therefore, for a service circuit, an average deceleration:

\[ a \geq 2.9 \text{ m/s}^2, \]

is accepted, while with the emergency circuit it can be increased to:

\[ a \geq 5.8 \text{ m/s}^2; \]

Circuits 1 and 2 describe the still working brakes, when each of the flexible tubes, at a time, is broken.

These prescriptions are applicable for any load condition within those allowed by the homologation form.
The braking circuit can be organized in different ways; the schemes described in Table 2.1 are accepted.

Table 2.1  Braking circuits allowed by the regulations of the European Union

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Circuit 1</th>
<th>Circuit 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>TT</td>
<td>Front axle</td>
<td>Rear axle</td>
</tr>
<tr>
<td>K</td>
<td>Front right and rear left wheels</td>
<td>Front left and rear right wheels</td>
</tr>
<tr>
<td>HT</td>
<td>All wheels</td>
<td>Front axle</td>
</tr>
<tr>
<td>LL</td>
<td>Front axle and rear right wheel</td>
<td>Front axle and rear left wheel</td>
</tr>
<tr>
<td>HH</td>
<td>All wheels</td>
<td>All wheels</td>
</tr>
</tbody>
</table>

Circuits 1 and 2 represent the possible configurations of the emergency circuit when the brake hydraulic flexible of one pipe of the wheels is out of order.

The choice among these schemes must take into account the vehicle load breakdown on the axles.

The scheme TT, for example, is suitable for a car whose center of gravity is located approximately at the middle of the wheelbase, as can happen in a front engine, rear drive two seater.

Scheme K is, instead, suitable to a front wheel driven car, where loaded with driver only, it is necessary that at least one of the front wheels can brake, to satisfy the requested performance.

The following schemes, such as HH, have the advantage of granting the emergency braking system a complete, or almost complete, braking circuit at a slightly increased cost; this fact allows the torque applied
by the unbalance of braking forces to the steering wheel to be kept to a minimum.

The safety margin in fading conditions must factor into this choice. Fading will be discussed later; fading is defined as a loss of braking efficiency caused by a partial evaporation of the braking fluid, the result of high temperatures.

When the fluid evaporates, it increases its elasticity dramatically and loses its ability to transmit pressure by compression. This phenomenon occurs near a heat source and, therefore, near the wheel. Because of this, circuits HT, LL, HH are more vulnerable when the fluid is evaporating near the front wheel.

Referring again to Figure 2.2 see valve D, including the pressure distributor, which will be explained later.

In addition to the prescribed stopping distance and braking acceleration, a further condition applies to the control force needed to achieve such a stopping distance. This force, applied to the pedal by the driver’s foot, must be:

$$F_p \leq 500 \text{ N}$$

this condition often requires the application of a power assistance device.

### 2.1.2 Parking System

The parking braking system, operated by a hand lever or by a further pedal, is needed to keep the vehicle stopped on a slope when the driver is not in the vehicle.
The regulation specifies for this system a mechanical, non-hydraulic, connection between the driver control and the brake; tie rods and cables are allowed.

The rationale for this prescription is to build up an additional emergency system, increasing system reliability by adding a backup in the rare event of a simultaneous failure of both service and emergency braking systems.

The parking braking system must be designed to keep the vehicle stopped on a slope of:

\[ i \geq 18\% , \]

or to reduce its speed, on a flat road, by an acceleration of:

\[ a \geq 1.5 \, \text{m/s}^2 , \]

after having applied on the control a force:

\[ F_p \leq 500 \, \text{N} ; \quad F_m \leq 400\, \text{N} ; \]

the first value applies to pedals, the second to hand levers.

The braking system must be able to generate an adjustable force that can be also maintained by driver controls; it works on the wheels of a single axle. The brakes can be the same as used for the service system.

Figure 2.3 shows a hand lever parking system: The control lever 1 moves the tie rod 17 and can be held in any position by the spring loaded pawl 27 engaging in the ratchet sector 23; the disengagement button 26 is used to unlock the pawl 27.
Figure 2.3 Scheme of a parking system with hand lever control

The tie rod 15 pulls the cable 6, through the sliding equalizer 16, dividing the force on the two brakes into two equal parts. The cable is connected to the rear brakes through the pin 8, which is connected to the two shoes.

The variant with pedal is similar; the pawl can be unlocked by a push pedal or by a button on the dashboard.

The connection with drum and disc brakes is explained in the following paragraphs.
2.1.3 Disc Brakes

This brake includes a disc, rotating with the wheel, on whose two faces two brake linings made of high friction material can be pressed.

On these linings, also called pads, work one or more hydraulic cylinders mounted on a suitable caliper.

The disc brake, like the drum brake will be described in the following paragraph, can be mounted in two different ways, shown in Figure 2.4, where the disc is placed:

- On the wheel hub, directly or
- On an auxiliary hub on the half shaft, at the differential box

The examples in this figure refer to a rear driving axle, but can also be applied to a front axle or a rear idling axle, by adding in this case, a shaft dedicated to the braking force.

The left figure shows the assembly of a disc brake on a guided trailing arm rear suspension. The brake caliper is fixed on the suspension strut, where the reaction to the braking force will be applied.

In the right figure, the caliper 2 is instead fixed to the differential box, or to the transmission, if differential and gearbox are integrated in a front wheel driven vehicle; the reaction to the braking force is applied, in this case, to the car body directly or through a suspension. The disc 1 is fit to the shaft 4 that transmits both driving and braking torque.
Figure 2.4 Examples of application of a disc brake to a rear axle

The disc can be mounted on the wheel itself (left) or attach to the differential box, working on the half shaft (right) to reduce the unsprung mass.

This kind of expensive configuration allows a reduction of sprung mass, with benefits for suspension comfort.

There are two kinds of architectures for disc brakes:

- Fixed caliper
- Sliding or floating calipers

In fixed caliper architectures, shown in Figure 2.5, pads are pressed against the disc by two pairs of independent hydraulic cylinders inside the caliper, connected in parallel to the same pressure source; the double cylinder arrangement can be exploited to separate the service circuit from the emergency circuit.

Inside the disc are radial channels of suitable shape; these behave as a radial ventilator that activates air for brake cooling. The transversal drilling on the disc also improves the direct cooling of the disc’s working surface.
In lighter cars with lower performance and therefore smaller discs, a single pair of cylinders might be sufficient. Ventilation channels and drilling could then be avoided.

Discs are made of iron alloy; in high performance cars, expensive carboceramic discs are now appearing, prized for their property of surviving high operating temperatures.

The floating caliper disc brake, in Figure 2.6 shows, instead, a single cylinder 4, working on the inside pad 5, while the outside pad 6 is pressed on the disc by the caliper body 1, able to slide on its mounts 7 along a direction perpendicular to the disc surface. This solution has the advantage of reducing cost and size, allowing the king-pin offset to be reduced to negative values.

![Figure 2.5 Ventilated disc brake with fixed caliper](image)

The two pads are pressed against the disc by four hydraulic cylinders, two for each pad.
Figure 2.6 Floating caliper disc

The caliper is assembled so as to be able to move in a direction perpendicular to the disc surface; the hydraulic cylinder can be single. An enlarged cross section of the piston and cylinder is shown with the detail of the sealing.

Hydraulic cylinders are operated by the fluid pressure applied by the master cylinder; as soon as the pedal is released, the piston of the master cylinder returns to the initial position by means of a return spring, and the hydraulic pressure is set to zero.

The brake piston is pulled away from the disc by the elastic force, determined by the lateral deformation of the sealing ring, which is invested on each piston with a certain radial load; the shape of the groove (see the enlarged detail of the sealing ring in rest position) allows piston motion without sliding on the sealing. A similar design feature is also applied to fixed calipers.
Floating calipers have the inconvenience that cross sliding can be impaired or blocked by mud sediments or by corrosion. For this reason protection bellows and special coatings are applied to the slide pins.

Impaired sliding of the caliper can cause different degrees of wear on internal and external pads; in some cases the caliper can be blocked by asymmetric braking, affecting vehicle path.

In disc brakes, the parking function is obtained with an additional coaxial drum brake; this solution is necessary if the force to be applied to the disc pad is too large.

This solution is shown in Figure 2.7, at left: Inside the disc bell 1, two additional braking shoes are installed, as on drum brakes; control force is applied as will be explained in the following paragraph.

The second solution, on the right of the same figure, combines service and parking functions in a single unit; brake pads are pushed by a crank and rod mechanism 1 (the crank is made by a cam fit to the control lever 2) operated by cables. Usually this control includes an automatic backlash adjuster.

In parking brakes a difference in braking force between two wheels of the same axle is not significant for safety.
At left, the function is obtained by an additional coaxial drum brake; at right, a lever 2 and a cam 1 act directly on the braking cylinder.

2.1.4 Drum Brakes

Figure 2.8 shows a cross section of a drum brake and some design details. This brake is made of a rotating hollow cylinder 7 (at left) fit to the wheel hub; on its inside surface work two symmetric shoes 6, on which are riveted or bonded brake linings. The shoes are pushed against the drum by the double piston cylinder 1.

At one end of the shoe one of the pistons pushes, while the other end is linked with a hinge or rests on a suitable surface. The two shoes are kept away from the drum by two springs 5.

Because of the necessary machining tolerance the drum surface and of the possible thermal deformations, the clearance between linings and drum must be well above that between lining and disc in disc brakes which, because of their simplicity, can be machined with higher accuracy and show no thermal deformations in the direction of motion of the pads.
In the same figure, on the right, a cross section of the control cylinder is shown: Two pistons 2, the oil feed drilling 3 and the return spring 6 can be seen; two pistons are assembled into the cylinder 9 and show their seals 4. The bellows 1 prevents water or dust contamination on the cylinder sliding surface.

The cylinder is connected to the oil circuit through the nipple 8; the bleeding valve 7 is used to remove air that could enter the circuit.

In modern cars drum brakes are applied to rear wheels only and supply, therefore, the mechanical parking brake control. This is performed by the push rod 3 and the crank 3, moved by cables.

![Figure 2.8](image) **Section of a drum brake (at left) and its actuation cylinder (at right)**
This function is accomplished by two levers 9 and 10 gearing through a tooth sector, outlined by the oval line S.

The friction between linings and drum causes them to wear. This fact is obviously present in disc brakes too, but the particular shape of the seal (see again the detail in Figure 2.6) eliminates the effect of the additional clearance caused by wear.

In drum brakes, where the shoes are returned against a rest by a spring, this wear could cause an increase of the distance between linings and drums and a consequent increase of brake pedal stroke, which could become unacceptably large.

The rest position of the shoe must therefore be adapted to the actual wear, so as to maintain a constant clearance between shoe and drum. This
adjustment can be made by manually rotating the two cams 6 by a nut and a
lock nut.

In most modern brakes this adjustment is made automatically. A
possible solution is shown in Figure 2.9.

According to this system (Bendix system), one of the two shoes is
made by two pieces 9 and 10 with tooth sectors gearing together, according to
the detail shown within the oval line S, in this figure. The rest of the left shoe
is made by the push rod 12; the clearance between rod and shoes determines
the clearance between lining and drum. Until this last clearance is higher than
the shoe stroke, the two pieces 9 and 10 work as a rigid system; as soon as the
stroke is higher than the clearance, the upper part is retained by the push rod.
This fact causes two sectors to rotate to a different position, producing a
reduction of the distance between linings and drum. The same push rod 12
can be used for parking brake control.

2.1.5 Control System Components

**Pump:** The pedal works on the pump piston, through the push rod
on the right of Figure 2.10.

Between this push rod and the pump the power brake is set, as
explained in the next paragraph.

The master pump or tandem pump is made by two pistons in series
in the same cylinder; in this cylinder wall the openings T are connected to the
oil tank, and the openings A and B to the two braking circuits. This particular
pump feeds two completely separated circuits (service and emergency
circuits), each of which can be connected to the brakes, according to the
schemes in Table 2.1.
Figure 2.10 Cross section of the master pump and vacuum power unit

Below is shown an enlarged detail of the valve between the vacuum chamber and the ambient air, used to modulate the power assistance.

In rest condition, the two pistons are kept to the right by the coil springs shown in the figure; the two piston chambers are connected to the tank at atmospheric pressure; in this way the additional fluid necessary to adapt to a clearance increase due to lining wear can be supplied.

As soon as the pedal is depressed the two holes T are closed and the pressure inside the circuits is increased, in proportion to the pedal force; this pressure will act on pistons working on the pads or shoes.

This kind of arrangement for the pump guarantees the operation of one circuit when the other has failed (spilling as a consequence of a pipe
rupture); in fact, if one of the circuits should spill the two independent pistons will contact each other and the pressure would increase in the still working circuit. The increase in pedal stroke warns the driver about the failure.

**Braking fluid:** The transmission of the pedal force to the braking surfaces is performed hydraulically.

The working fluid for this purpose must have particular features and satisfy the following specifications:

- In normal working pressure conditions the fluid must be incompressible.
- Its boiling point must exceed a certain minimum value, in order to maintain its properties after a lengthy braking.
- The fluid must have low viscosity at very low temperature, in the range of -40°C.
- It must have suitable lubrication properties for parts in relative motion (pistons, seals and cylinders).
- It must be chemically stable and non-aggressive to metal and elastomeric components.

These conditions are fulfilled by some organic oils. These oils must be changed after a certain period of use because they are hygroscopic; water is present in the air as humidity and molecules of water can contaminate the oil in the atmospheric tank, when the level decreases as a consequence of wear.

Water in solution decreases the boiling temperature of the oil. When brakes heat, water in solution changes to vapor; vapor bubbles decrease oil compressibility and increase pedal stroke at the same level of pressure. At
critical conditions the pedal stroke is insufficient to allow the desired braking force. This lack of braking efficiency is called fading or vapor-lock.

Water absorption speed is largely dependent on climate; in humid and hot climates a percentage of 3% of the oil can be absorbed water, with a consequent 80% reduction in boiling temperature.

The United States Department of Transportation certifies fluids as DOT3, DOT4 and DOT5, defining different boiling points as a function of water content.

**Distributor:** Because of vertical load transfer due to vehicle deceleration, the braking force applied to the front wheels must increase as compared to the static value; for the same reason the braking force applied to the rear wheels must decrease.

Static load conditions also affect braking force distribution between the axles, because of the different position of payloads on the vehicle, with reference to the axle positions.

The function of adapting the braking force shared between axles is performed by the brake distributor.

The braking circuit is designed to grant the rear wheels the maximum braking force necessary, usually at full static load; the distributor is designed so as to reduce this pressure to the suitable value, corresponding to the actual static load and load transfer.

This target is achieved according to the following rules:

- When the circuit pressure is lower than a threshold value, the rear pressure is not reduced.
• When this threshold is exceeded, front pressure and rear pressure increase, according to a preset value lower than one.

This function is achieved by the valve sketched in Figure 2.11. The nipple 7 is connected with the pump and the front circuit, while the nipple 6 is connected with the rear circuit. In this valve a moving spool 1 responds to the rear suspension stroke, through the tip 2.

The scheme at the right of this figure shows installation on the vehicle; a suitable leverage pushes on the tip 2 when the suspension moves to compression (displacement in direction a) and vice versa in the extension direction (displacement in direction b). The vehicle suspension acts as a dynamometer measuring the axle load through the suspension stroke.

When the spool 1 is compressed in the upper direction, the valve 4 is lifted, opening the passage 3; when the spool descends, at a given position the valve 4 closes, interrupting the connection between the pump circuit and the rear brake circuit. In this condition the pressure at the nipple 6 will be reduced with reference to the pressure at nipple 7, according to the ratio between the areas on the spool 1.

The stiffness of the spring 5 determines the suspension load at which the braking pressure is reduced, while the ratio between the surfaces determines the value of this reduction.
Figure 2.11  Cross section of a brake distributor and its installation on the rear axle

Figure 2.12 shows an example comparing the curve of the ideal braking pressure distribution and the actual distribution made by a valve of the kind shown in Figure 2.11. By an ideal distribution, each wheel can brake at the maximum value of the friction coefficient. The parameter for this comparison is the pressure in the front and rear circuits.
Figure 2.12  Comparison between the ideal distribution curve of braking pressure and the actual distributor curve

The disadvantage of a valve of this kind is that the real distribution curve is at any condition lower than the ideal, preventing the rear wheels from reaching their maximum braking capacity; on the other hand, slip is completely avoided.

A further disadvantage is that is difficult to obtain a result of this kind for any of the possible load combinations of the front and rear axle.

For reduced load variations (as on a two seater) simple pressure reducers can be used.

A third class of valves is responsive to vehicle deceleration. In this case the change of slope of the distributor curve is determined by the braking deceleration. This kind of valve takes into account the weight distribution
change as well. A possible malfunction can be caused by the effect of the internal friction of the valve’s mechanical accelerometer.

ABS systems also perform the function of limiting braking pressure on the rear axle, according to the actual vertical load; in this case the distributor valve is no long necessary.

### 2.1.6 Power Brakes

Power brakes allow a vehicle to be braked with a reduced force on the pedal and a reduced stroke that can be contained within acceptable limits. Their advantages include braking safety and driver comfort.

To design a power brake, the following requirements must be specified.

- A power brake must be sufficiently responsive to allow the driver to modulate the braking force, even with low pedal pressure; internal friction must therefore be limited. The lower intervention point of the power system must be in the range of a pedal force of about 15 ÷ 20 N.
- The effort exerted on the pedal is the feedback of the braking system to the driver; it must be correlated to vehicle deceleration. The force on the pedal must be, in any case, proportional to braking deceleration.
- The power system response time should be lower than 0.1 s; the response time is the time necessary to reach the maximum assistance value during a sudden braking, where the pedal is depressed at a speed of about 1 m/s.
• The passage from the assisted mode to the unassisted mode at the saturation point (see the definition of this condition later on) must be gradual to allow the driver to use further force increments in emergency situations.

• The reliability of this system must be absolute; power system failures can panic the driver.

• The weight and volume must be limited to suit installation into the engine compartment.

The tip-in load of a power brake is the minimum load on the pedal necessary to trigger the assistance of the system during a braking event.

The saturation load is, instead, the value of the load at which the diagram of force on the pedal versus braking pressure changes its slope, because the assistance of the power brake has reached its maximum value. This load should never be reached in actual conditions; it must therefore be over the braking force necessary to stop the vehicle at maximum friction coefficient and maximum vehicle load.

**Vacuum power brake:** The power brakes is placed between brake pump and pedal and amplifies the force applied on the pedal by the driver, exploiting the difference of pressure between two chambers, one connected with ambient air and one with the intake manifold, for a throttled gasoline engine. When the vacuum manifold is not sufficient, as in diesel engines, a vacuum pump is driven by the engine.

With reference to Figure 2.10, already used to explain the braking pump, note the dimension of the actuator, bound to the modest value of the difference in pressure between the manifold and the ambient air.
The actuator includes a cylinder and a piston made of steel sheet, with a suitable membrane seal; front (at left) and rear chambers are found in the actuator.

The front chamber is always in communication with the intake manifold, downstream of the throttle valve or with the vacuum pump.

Three different situations, or phases, may be identified:

- Rest position, with pedal completely released (phase 1)
- Pedal depressed (phase 2)
- Pedal released (phase 3)

**Phase 1:** When the pedal is released or the pedal stroke is zero (as in Figure 2.10), the two chambers are set at the same pressure $p_s$. This pressure is equal to that of the vacuum source.

Because there is no pressure difference between the two faces of the power brake membrane, there will be no assistance.

In the same figure, on its lower side, is shown a detail of the shaft of the actuator’s piston. On this shaft there is a valve which puts the two chambers of the power brake in communication; in this figure the valve is drafted when the pedal is depressed at zero force, or when any play is set to zero.

**Phase 2:** Let us now assume that a pressure is applied to the pedal. After a transient, the valve will cut the communication between the two chambers. In the detail on the left of the same figure, a plunger 1 closes the communication between the two chambers by a rubber surface.
After a short while, the rubber element 4, compressed by the braking force, will assume a given deformation, opening a passage 2 between the rear chamber and the ambient pressure.

The pressure difference between the two chambers will determine the assistance force.

The opening of the passage is a function of the deformation of the rubber element 4, sensitive to the load applied by the pedal. The passage will close as soon as the driver reaches the desired load on the pedal; the pressure in the rear chamber will therefore be proportional to the rubber element deformation and to applied load. The rubber element 4 measures the desired load.

**Phase 3:** When, at the end of braking, the pedal is released, communication with the outside will be closed and that between the two chambers will be reopened. Both chambers will be set at the same pressure, with no force applied to the braking pump.

In summary, the possible states of the power brake are shown in Table 2.2, as a function of the load applied to the pedal and of its derivative over time.

Figure 2.13 shows an example of the characteristics of a power brake. This curve shows the braking pressure as a function of the load on the pedal. The pressure can be calculated by the following formula:

\[ p = \frac{(F_s + F_p \tau)}{A_p} \]

where \( F_s \) is the assistance supplied by the power brake;
\( F_p \) is the force on the pedal;
Table 2.2 Description of the states of a vacuum power brake, as a function of the pedal force $F$ and of its derivative over time $\frac{dF}{dt}$

<table>
<thead>
<tr>
<th>$F$</th>
<th>$\frac{dF}{dt}$</th>
<th>Outside duct</th>
<th>Inside duct</th>
<th>Rear Pressure</th>
<th>Assistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>$=0$</td>
<td>$=0$</td>
<td>Open</td>
<td>Closed</td>
<td>$p_s$</td>
<td>No</td>
</tr>
<tr>
<td>$&gt;0$</td>
<td>$&gt;0$</td>
<td>Closed</td>
<td>Open</td>
<td>$p_s &lt; p \leq p_o$</td>
<td>Yes</td>
</tr>
<tr>
<td>$&gt;0$</td>
<td>$=0$</td>
<td>Closed</td>
<td>Closed</td>
<td>$p_s &lt; p \leq p_o$</td>
<td>Yes</td>
</tr>
<tr>
<td>$&gt;0$</td>
<td>$&gt;0$</td>
<td>Open</td>
<td>Closed</td>
<td>$p_s &lt; p &lt; p_o$</td>
<td>No</td>
</tr>
</tbody>
</table>

- $\tau$ is the leverage ratio between pedal and plunger;
- $p$ is the braking pressure;
- $A_p$ is the useful area of the master pump.

Figure 2.13 Characteristic curve of a power brake
The braking pressure is shown as a function of the load applied on the pedal. This characteristic curve can be measured in a bench test, where the vacuum value is kept constant, while the braking force is set to different values.

In this diagram, four zones are outlined.

- **Zone 1:** the braking force is insufficient to win the resistance of the springs that keep the brake in rest position. The braking pressure is thus zero. The load value, where the power brake begins to supply its contribution, is called tip-in load.

- **Zone 2:** after the tip-in load is reached there is a sudden increase of the assistance force; this phase is called jump-in. Jump-in pressure is the pressure value reached at the end of this phase.

- **Zone 3:** in this phase there is a constant amplification of the force applied by the pedal. The ratio $G$ between the pressure and the applied load is called power brake gain.

- **Zone 4:** the power brake has reached the maximum value of pressure difference between the ambient pressure $p_0$ and the vacuum source $p_s$. The pressure increment in this area is caused solely by an increase of the force applied by the driver to the pedal. This value is called saturation pressure.

**Hydraulic power brake:** In hydraulic power brakes the energy source is supplied by a pressurized fluid. In general, the pressure source is the same as for the power steering system and the two circuits share the same fluid. The braking servo other than the actuator is identical to what have been seen.
The higher working pressure allows a reduction in system dimensions and makes this system available to heavy cars and medium size industrial vehicles, where the vacuum pressure is insufficient.

The assistance system is made up of a simple hydraulic cylinder set in series with the master cylinder. It is fed by the power steering pump, through suitable valves.

A solenoid valve puts the power steering pump in communication with the braking circuit. When the brakes are in rest condition the pressure is available for the power steering system. During braking, priority is given to this system. A second valve modulates assistance pressure, according to the pedal force.

A pressure accumulator contains a quantity of pressure oil suitable for 2 or 3 braking, in case of the failure of the pressure source or engine stall.

For heavier vehicles, where greater storage of energy is necessary, an additional electrical pump is applied; this is used when the normal flow of oil from the steering pump is interrupted.

The power system fluid and the braking fluid are different and should not be mixed; specialized seals avoid contamination.