24

BRAKING DYNAMIC PERFORMANCE

The study of braking on straight road is performed using mathematical models similar to those seen in Chapter 23 for longitudinal dynamics. But in this case, the presence of suspensions and the compliance of tires are neglected and the motion is described by the longitudinal equilibrium equation (23.1) alone

$$m\ddot{x} = \sum_{\forall i} F_{x_i} .$$

Apart from cases in which the vehicle is slowed by the braking effect of the engine, which can dissipate a non-negligible power (lower part of the graph of Fig. 22.2), and by regenerative braking in electric and hybrid vehicles, braking is performed in all modern vehicles on all wheels. Subscript i thus extends to all wheels or, when thinking in terms of axles, as is usual for motion in symmetrical conditions, on all axles.

24.1 BRAKING IN IDEAL CONDITIONS

Ideal braking can be defined as the condition in which all wheels brake with the same longitudinal force coefficient μ_x.

The study of braking forces the vehicle can exert will follow the same scheme seen in Section 23.5, the only obvious difference being that braking forces, like the corresponding longitudinal force coefficients and the longitudinal slip, are negative. Normal forces between road and tires can be computed using the equations seen in Chapter 23.1, remembering here as well that the acceleration is negative.

G. Genta, L. Morello, *The Automotive Chassis, Volume 2: System Design*, Mechanical Engineering Series,

The total braking force F_x is thus

$$F_x = \sum_{\forall i} \mu_{x_i} F_{z_i} \ , \tag{24.1}$$

where the sum extends to all the wheels. The longitudinal equation of motion of the vehicle is then

$$\frac{dV}{dt} = \frac{\sum_{\forall i} \mu_{x_i} F_{z_i} - \frac{1}{2}\rho V^2 S C_X - f \sum_{\forall i} F_{z_i} - mg\sin(\alpha)}{m} \ , \tag{24.2}$$

where m is the actual mass of the vehicle and not the equivalent mass, and α is positive for uphill grades. The rotating parts of the vehicle are slowed directly by the brakes, and hence do not enter into the evaluation of the forces exchanged between vehicle and road. These parts must be accounted for when assessing the required braking power of the brakes and the energy that must be dissipated.

Aerodynamic drag and rolling resistance can be neglected in a simplified study of braking, since they are usually far smaller than braking forces. Also, rolling resistance can be considered as causing a braking moment on the wheel more than a direct braking force on the ground.

Since in ideal braking all force coefficients μ_{x_i} are assumed to be equal, the acceleration is

$$\frac{dV}{dt} = \mu_x \left[g\cos(\alpha) - \frac{1}{2m}\rho V^2 S C_Z \right] - g\sin(\alpha) \ . \tag{24.3}$$

On level road, for a vehicle with no aerodynamic lift, Eq. (24.3) reduces to

$$\frac{dV}{dt} = \mu_x g \ . \tag{24.4}$$

The maximum deceleration in ideal conditions can be obtained by introducing the maximum negative value of μ_x into Eq. (24.3) or (24.4).

The assumption of ideal braking implies that the braking torques applied on the various wheels are proportional to the forces F_z, if the radii of the wheels are all equal.

As will be seen later, this may occur in only one condition, unless some sophisticated control device is implemented to allow braking in ideal conditions. If μ_x can be assumed to remain constant during braking, the deceleration of the vehicle is constant, and the usual formulae hold for computing the time and space needed to slow from speed V_1 to speed V_2:

$$t_{V_1 \to V_2} = \frac{V_1 - V_2}{|\mu_x| g} \ , \qquad s_{V_1 \to V_2} = \frac{V_1^2 - V_2^2}{2|\mu_x| g} \ . \tag{24.5}$$

The time and the space to stop the vehicle from speed V are then

$$t_{arr} = \frac{V}{|\mu_x| g} \ , \qquad s_{arr} = \frac{V^2}{2|\mu_x| g} \ . \tag{24.6}$$

The time needed to stop the vehicle increases linearly with the speed while the space increases quadratically.

To compute the forces F_x the wheels must exert to perform an ideal braking manoeuvre, forces F_z on the wheels must be computed first. This can be done using the formulae in Section 23.1. However, for vehicles with low aerodynamic vertical loading, such as all commercial and passenger vehicles with the exception of racers and some sports cars, aerodynamic loads can be neglected. Drag forces can also be neglected and, in the case of a two-axle vehicle, the equations reduce to

$$F_{z_1} = \frac{m}{l}\left[gb\cos(\alpha) - gh_G\sin(\alpha) - h_G\frac{dV}{dt}\right] , \tag{24.7}$$

$$F_{z_2} = \frac{m}{l}\left[ga\cos(\alpha) + gh_G\sin(\alpha) + h_G\frac{dV}{dt}\right] . \tag{24.8}$$

Since the values of μ_x are all equal in ideal braking, the values of longitudinal forces F_x can be immediately computed by introducing Eq. (24.3)

$$\frac{dV}{dt} = \mu_x g\cos(\alpha) - g\sin(\alpha)$$

into equations (24.7) and (24.8)

$$F_{x_1} = \mu_x F_{z_1} = \mu_x\frac{mg}{l}\cos(\alpha)\left(b - h_G\mu_x\right) , \tag{24.9}$$

$$F_{x_2} = \mu_x F_{z_2} = \mu_x\frac{mg}{l}\cos(\alpha)\left(a + h_G\mu_x\right) . \tag{24.10}$$

By adding Eq. (24.9) to Eq. (24.10), it follows that:

$$F_{x_1} + F_{x_2} = \mu_x mg\cos(\alpha) , \tag{24.11}$$

and then:

$$\mu_x = \frac{F_{x_1} + F_{x_2}}{mg\cos(\alpha)} . \tag{24.12}$$

By introducing the value of μ_x into equations (24.9) and (24.10) and subtracting the second equation from the first, it follows that

$$F_{x_1} - F_{x_2} = \frac{b-a}{l}(F_{x_1} + F_{x_2}) - \frac{2h_G}{lmg\cos(\alpha)}(F_{x_1} + F_{x_2})^2 . \tag{24.13}$$

A relationship between F_{x_1} and F_{x_2} is readily obtained. It is an equation expressing the relationship between the forces at the front and rear axles that must hold to make ideal braking possible,

$$(F_{x_1} + F_{x_2})^2 + mg\cos(\alpha)\left(F_{x_1}\frac{a}{h_G} - F_{x_2}\frac{b}{h_G}\right) = 0 . \tag{24.14}$$

The plot of Eq. (24.14) in the F_{x_1},F_{x_2} plane is a parabola whose axis is parallel to the bisector of the second and fourth quadrants if $a = b$ (Fig. 24.1).

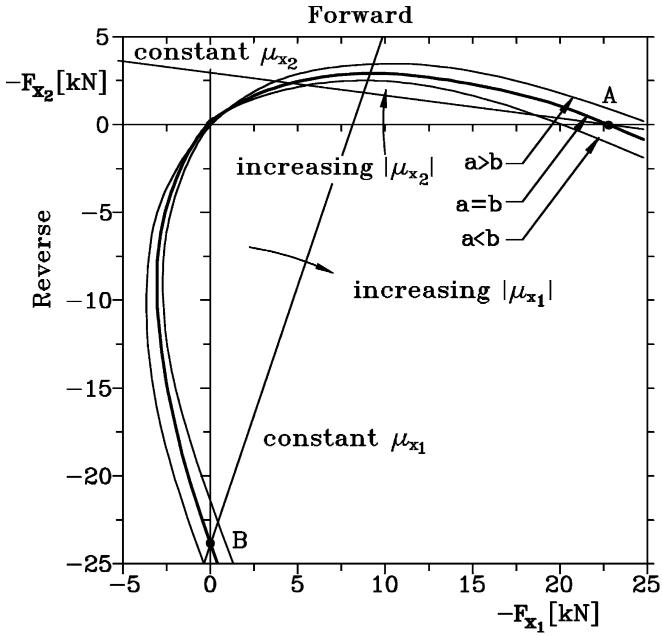

FIGURE 24.1. Braking in ideal conditions. Relationship between F_{x_1} and F_{x_2} for vehicles with the centre of mass at mid-wheelbase ($a = b$), forward ($a < b$) and backward ($a > b$) of that point. Plots obtained with $m = 1000$ kg; $l = 2.4$ m, $h_G = 0.5$ m, level road.

The parabola is thus the locus of all pairs of values of F_{x_1} and F_{x_2} leading to ideal braking.

Only a part of this plot is actually of interest: That with negative values of the forces (braking in forward motion) and with braking forces actually achievable, i.e. with reasonable values of μ_x (Fig. 24.2).

On the same plot it is possible to draw the lines with constant μ_{x_1}, μ_{x_2} and acceleration. On level road, the first two are straight lines passing, respectively, through points B and A, while the lines with constant acceleration are straight lines parallel to the bisector of the second quadrant.

Remark 24.1 *All forces here relate to the axles and not to the wheels: In the case of axles with two wheels their values are then twice the values referred to the wheel.*

The moment to be applied to each wheel is approximately equal to the braking force multiplied by the loaded radius of the wheel: If the wheels have equal radii, the same plot holds for the braking torques as well. If this condition does not apply, the scales are simply multiplied by two different factors and the plot, though distorted, remains essentially unchanged.

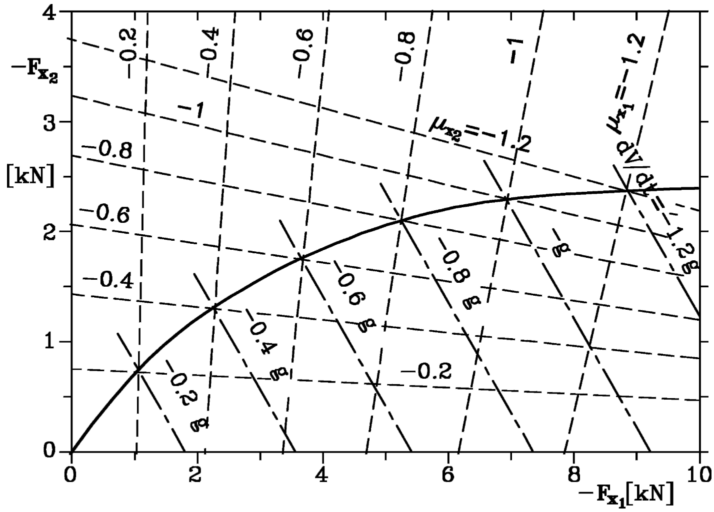

FIGURE 24.2. Enlargement of the useful zone of the plot of Fig. 24.1. The lines with constant μ_{x_1}, μ_{x_2} and acceleration are also reported.

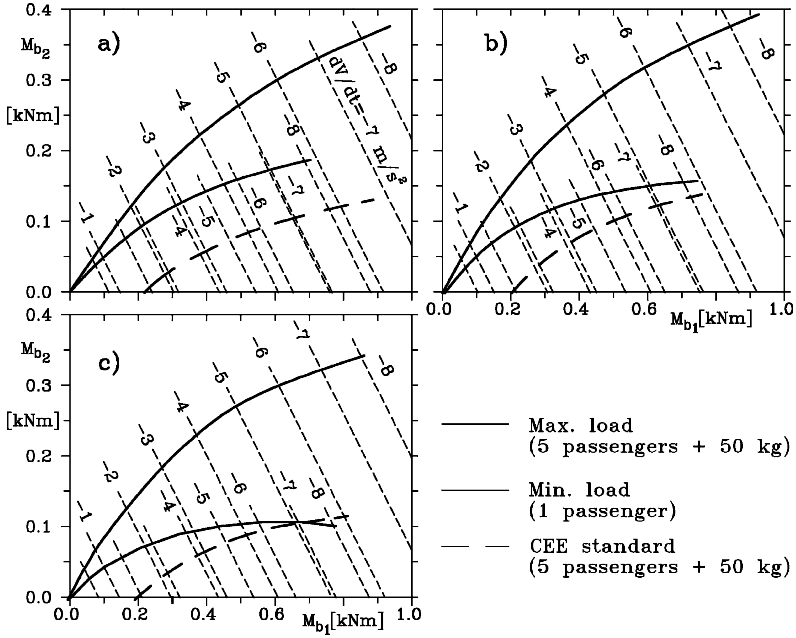

FIGURE 24.3. Plots $M_{b_2}(M_{b_1})$ for ideal braking. (a) typical plot for a rear drive car with low ratio h_G/l; (b) typical plot for a front drive saloon car with higher ratio h_G/l; (c) plot for a small front drive car, sensitive to the load conditions and with high value of ratio h_G/l.

Remark 24.2 *To perform a more precise computation, the rolling resistance, which is a small correction, should be accounted for and the torque needed for decelerating the rotating inertias should be added. This correction is important only for driving wheels and braking in low gear, but in this case the braking effect of the engine, which is even more important and has the opposite sign, should be considered.*

As stated before, the law linking F_{x_1} to F_{x_2}, i.e. M_{b_1} to M_{b_2} to allow braking in ideal conditions, depends on the mass and the position of the centre of mass. For passenger vehicles, it is possible to plot the lines for the minimum and maximum load and to assume that all conditions are included between them; for industrial vehicles, the position of the centre of mass can vary to a larger extent, and a larger set of load conditions should be considered.

The curves for three different types of passenger vehicles are shown in Fig. 24.3 as an example. The curve $M_{b_2}(M_{b_1})$ defined by CEE standards and the lines at constant acceleration are reported on the same plot.

24.2 BRAKING IN ACTUAL CONDITIONS

The relationship between the braking moments at the rear and front wheels is in practice different from that stated in order to comply with the conditions needed to obtain ideal braking, and is imposed by the parameters of the actual braking system of the vehicle.

A ratio

$$K_b = \frac{M_{b_1}}{M_{b_2}}$$

between the braking moments at the front and rear wheels can be defined. If all wheels have the same radius, its value coincides with the ratio between the braking forces.

Remark 24.3 *This statement neglects the braking moment needed to decelerate rotating parts. This can be adjusted by considering M_b as the part of the braking moment that causes braking forces on the ground; the fraction of the braking moment needed to decelerate the wheels and the transmission must be added to it.*

For each value of the deceleration a value of K_b allowing braking to take place in ideal conditions can be easily found from the plot of Fig. 24.2. K_b depends on the actual layout of the braking system, and in some simple cases is almost constant.

In hydraulic braking systems, the braking torque is linked to the pressure in the hydraulic system by a relationship of the type

$$M_b = \epsilon_b(Ap - Q_s) \;, \tag{24.15}$$

where ϵ_b, sometimes referred to as the efficiency of the brake, is the ratio between the braking torque and the force exerted on the braking elements and hence has the dimensions of a length. A is the area of the pistons, p is the pressure and Q_s is the restoring force due to the springs, when they are present.

The value of K_b is thus

$$K_b = \frac{\epsilon_{b_1}(A_1 p_1 - Q_{s_1})}{\epsilon_{b_2}(A_2 p_2 - Q_{s_2})} , \tag{24.16}$$

or, if no spring is present as in the case of disc brakes,

$$K_b = \frac{\epsilon_{b_1} A_1 p_1}{\epsilon_{b_2} A_2 p_2} . \tag{24.17}$$

In disc brakes, ϵ_b is almost constant and is, as a first approximation, the product of the average radius of the brake, the friction coefficient and the number of braking elements acting on the axle, since braking torques again refer to the whole axle. If the pressure acting on the front and rear wheels is the same, the value of K_b is constant and depends only on geometrical parameters.

The behavior of drum brakes is more complicated, as restoring springs are present and the dependence of ϵ_b on the friction coefficient is more complex. As stated in Part I, shoes can be of the *leading* or of the *trailing* type. If leading, the braking torque increases more than linearly with the friction coefficient and there is even a value of the friction coefficient for which the brake sticks and the wheel locks altogether.

The opposite occurs with trailing shoes and ϵ_b increases less than linearly with the friction coefficient.

The efficiency of the brakes is a complex function of both temperature and velocity and, during braking, it can change due to the combined effect of these factors. When the brake heats up there is usually a decrease of the braking torque, at least initially. Later an increase due to the reduction of speed can restore the initial values. This "sagging" in the intermediate part of the deceleration is more pronounced in drum than in disc brakes. With repeated braking, the overall increase of temperature can lead to a general "fading" of the braking effect.

If K_b is constant, the characteristic line on the plane M_{b_1}, M_{b_2} is a straight line through the origin (Fig. 24.4).

The intersection of the characteristics of the braking system with the curve yielding ideal braking defines the conditions in which the system performs in ideal conditions. On the left of point A, i.e. for low values of deceleration, the rear wheels brake less than required and the value of μ_{x_2} is smaller than that of μ_{x_1}. If the limit conditions occur in this zone, i.e. for roads with poor traction, the front wheels lock first.

On the contrary, all working conditions beyond point A are characterized by

$$\mu_{x_2} > \mu_{x_1}$$

and the rear wheels brake more than required, i.e., the braking capacity of the front wheels is underexploited. In this case, when the limit conditions are reached, the rear wheels lock first, as in the case of Fig. 24.4.

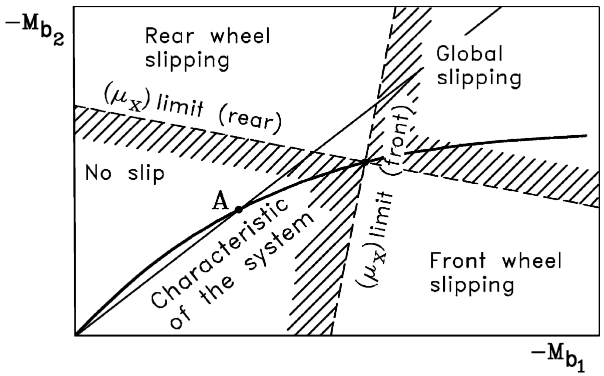

FIGURE 24.4. Conditions for ideal braking, characteristic line for a system with constant K_b and zones in which the front or the rear wheels lock. In the case shown the value of μ_p is high enough to cause sliding beyond point A.

From the viewpoint of handling, it is advisable that

$$\mu_{x_2} < \mu_{x_1} ,$$

since this increases the stability of the vehicle; the characteristics of the braking system should lie completely below the line for ideal braking. Locking of the rear wheels is a condition that must be avoided since it triggers directional instability.

In A the ideal conditions are obtained: If the limit value of the longitudinal force coefficient occurs at that point, simultaneous locking of all wheels occurs.

The values of ratio K_b for which the ideal conditions occur at a given value of the longitudinal force coefficient μ_x^* are immediately computed,

$$K_b^* = \frac{b + h_G|\mu_x^*|}{a - h_G|\mu_x^*|} . \tag{24.18}$$

It is possible to define an efficiency of braking as the ratio between the acceleration obtained in actual conditions and that occurring in ideal conditions, obviously at equal value of the coefficient μ_x of the wheels whose longitudinal force coefficient is higher,

$$\eta_b = \frac{(dV/dt)_{actual}}{(dV/dt)_{ideal}} = \frac{(dV/dt)_{actual}}{\mu_x g} , \tag{24.19}$$

where the last expression holds only on level road for a vehicle with negligible aerodynamic loading.

The total braking force acting on the vehicle when the rear wheels lock is

$$F_{x_1} + F_{x_2} = F_{x_2} (1 + K_b) , \tag{24.20}$$

and thus the deceleration on level road is

$$\frac{dV}{dt} = \frac{F_{x_2} (1 + K_b)}{m}. \tag{24.21}$$

Eq. (24.8) yields

$$F_{x_2} = \frac{\mu_{x_2} g}{l} \left[am + h_G F_{x_2} \left(1 + K_b\right) \right] \,, \tag{24.22}$$

and then

$$F_{x_2} = \frac{\mu_{x_2} gam}{l - \mu_{x_2} h_G \left(1 + K_b\right)} \,, \tag{24.23}$$

$$\frac{dV}{dt} = g \frac{\mu_{x_2} a \left(1 + K_b\right)}{l - \mu_{x_2} h_G \left(1 + K_b\right)} \,. \tag{24.24}$$

If on the contrary the front wheels lock, the total braking force acting on the vehicle is

$$F_{x_1} + F_{x_2} = F_{x1} \left(1 + \frac{1}{K_b}\right) \,. \tag{24.25}$$

Operating as already seen with rear wheels lock, the value of the acceleration can be found,

$$\frac{dV}{dt} = g \frac{\mu_{x_1} b \left(1 + K_b\right)}{l K_b - \mu_{x_1} h_G \left(1 + K_b\right)} \,. \tag{24.26}$$

The braking efficiency is then

$$\eta_b = \min \left\{ \frac{a(K_b + 1)}{l - \mu_p h_G (K_b + 1)} \,, \, \frac{b(K_b + 1)}{l K_b + \mu_p h_G (K_b + 1)} \right\} \,. \tag{24.27}$$

The first value holds when the rear wheels lock first (above point A in Fig. 24.4), the second when the limit conditions are reached at the front wheels first.

A typical plot of the braking efficiency versus the peak braking force coefficient is plotted in Fig. 24.5.

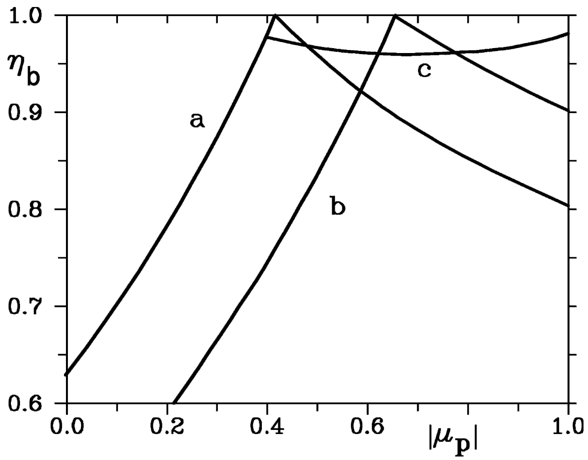

FIGURE 24.5. Braking efficiency η_b as a function of the limit value of μ_x for a vehicle without (a) and (b) and with (c) pressure proportioning valve.

The value of the maximum longitudinal force coefficient μ_p at which the condition $\eta_b = 1$ must hold can be stated and the value of ratio K_b can be easily computed. For values of $|\mu_p|$ lower than the chosen one, the rear wheels lock first while for higher values locking occur at the front wheels.

Once K_b is known, the braking system can easily be designed. The curve $\eta_b(\mu_x)$ can be plotted by assigning increasing values to the pressure in the hydraulic system, computing K_b and then the values of μ_x and η_b referred to the front and rear wheels. The result is of the type shown in Fig. 24.5, curve (a) or (b).

Operating in this way, the rear wheels lock when the road is in good condition. To postpone the locking of the rear wheels, curves of the type of line (b) can be used, but this reduces efficiency when the road conditions are poor.

To avoid locking of the rear wheels without lowering efficiency at low values of μ_x, a pressure proportioning valve, i.e. a device that reduces the pressure in the rear brake cylinders when the overall pressure in the system increases above a given value, may be used. A linear reduction of the pressure on the rear brakes with increasing pressure in the front ones above a certain pressure p_i,

$$\begin{cases} p_2 = p_1 & \text{for } p_1 \le p_i, \\ p_2 = p_1 + \rho_c\,(p_1 - p_i) & \text{for } p_1 > p_i, \end{cases} \tag{24.28}$$

where ρ_c is a characteristic constant of the valve, can be assumed.

Pressure p_i and constant ρ_c must be chosen in such a way that the device starts acting when the efficiency η_b gets close to unity. The reduction of the rear pressure must be such that it does not cause locking of the rear wheels; nor should it be so high as to substantially lower the efficiency (see Fig. 24.5, curve (c)).

To comply with these conditions in all load conditions of the vehicle, p_i and, possibly, ρ_c must vary following the load. A possible way to achieve this is to monitor the load on the rear axle, e.g. by monitoring the vertical displacement of the rear suspension.

The characteristic line in the M_{b_1}, M_{b_2} plane of a device operating along this line is reported in Fig. 24.6.

To prevent wheels from locking, antilock systems (ABS) act directly to reduce the pressure in the hydraulic cylinders of the relevant brakes when the need to reduce the braking force arises. Modern devices are based on wheel speed sensors allowing the actual speed of the wheels and the speed corresponding to the velocity of the vehicle to be compared. If a slip that exceeds the allowable limits is detected, the device acts to reduce the braking torque, restoring appropriate working conditions.

As will be shown in detail in Chapter 27, ABS systems may work in different ways, both in the physical characteristics of the system and in the control algorithms.

The above braking efficiency holds only in the case of rigid vehicles. If the presence of suspensions is accounted for, the load transfer from the rear to the front wheels does not occur immediately, and at the beginning of the braking manoeuvre the vertical loads on the wheels are the same as those at constant

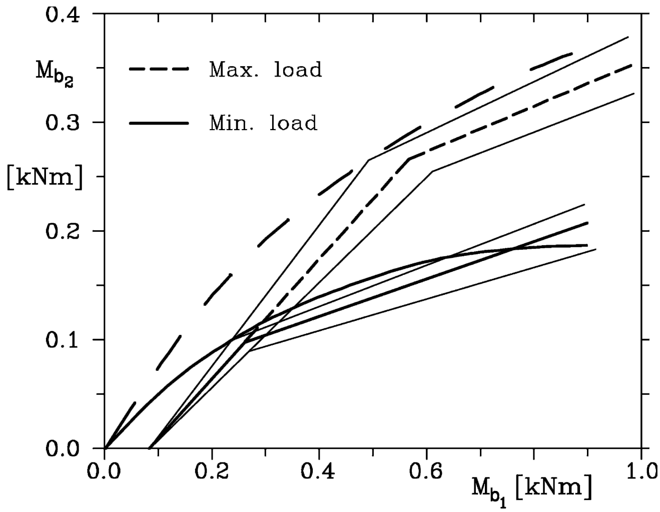

FIGURE 24.6. Characteristic of a braking system in which a pressure proportioning valve operating following Eq. (24.28) is present. To take into account the variability of the parameters of the system, specifically the friction coefficient, a band of characteristics has been considered instead of a single line. The ideal braking lines at the two different load conditions have also been plotted.

speed. The body of the vehicle then starts to dive and the load on front wheels increases, until steady state conditions are reached and the loads take the values given by Equations (24.7) and (24.8). This effect actually depends largely on the characteristics of the suspensions: the rotation of the body can be very small and load shift is almost immediate when antidive arrangements are used.

The load on the rear wheels is higher and the locking of the rear wheels is more difficult at the beginning of the manoeuvre: This consideration explains the practice of giving short brake pulses, effective when modern braking systems designed to avoid rear wheel locking were not available.

Example 24.1 *Plot the braking efficiency of the car of Appendix E.2, assuming that the braking system is designed to reach the ideal conditions for a longitudinal force coefficient* $\mu_x = -0.4$. *Use a pressure proportioning valve in such a way that the front wheels lock before the rear ones up to a value of* μ_x *equal to unity. Neglect aerodynamic forces and rolling resistance.*

The curve characterizing the conditions for ideal braking in plane F_{x_1}, F_{x_2} *is plotted (Fig. 24.7a). In order to obtain the ideal conditions at a value of the longitudinal force coefficient* $\mu_x^* = -0.4$, *ratio* K_b *is immediately computed from Eq. (24.18):* $K_b = 2.283$. *The braking forces corresponding to the ideal conditions are* $F_{x_1} = 2.265$ *kN and* $F_{x_2} = 0.992$ *kN.*

The pressure proportioning valve is assumed to start acting when values of the forces, equal to 90% of those for ideal conditions, are reached: $F_{x_1} = 2.038$ *kN and* $F_{x_2} = 0.893$ *kN. As the point at which the ideal conditions with* $\mu_x = 1$ *are reached is*

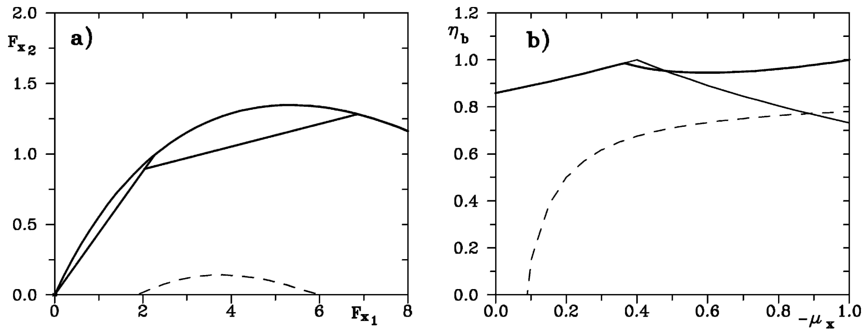

FIGURE 24.7. Braking characteristics of the vehicle of Appendix E.2. (a) Ideal braking conditions and characteristics of the braking system. (b) Braking efficiency with and without pressure proportioning valve. The dashed lines show the minimum conditions stated by CEE standards.

easily computed ($F_{x_1} = 6.861$ kN and $F_{x_2} = 1.282$ kN), the equation expressing force F_{x_2} as a function of F_{x_1} when the valve is operating is immediately found. From its slope, the value of constant $\rho_c = 0.184$ is obtained. The characteristic of the braking system is plotted in Fig. 24.7a.

At each point a pair of values F_{x_1} and F_{x_2} are obtained. From them the deceleration and the maximum value of the longitudinal force coefficient may be computed, ultimately obtaining the braking efficiency. The results are plotted in Fig. 24.7b.

In the same figures the curves related to the CEE standards are also plotted (dashed lines). Note that the position of the centre of mass results in the very low position of the dashed line in Fig. 24.7a.

24.3 BRAKING POWER

The instantaneous power the brakes must dissipate is

$$|P| = |F_x|V = V\left|\frac{dV}{dt}m_e + mg\sin(\alpha)\right| , \tag{24.29}$$

where all forms of drag have been neglected.

The brakes cannot dissipate this power directly; they usually work as a heat sink, storing some of the energy in the form of thermal energy and dissipating it in due time. Care must obviously be exerted to design the brakes in such a way that they can store the required energy without reaching excessively high temperatures and so that adequate ventilation for cooling is ensured. The average value of the braking power must, at any rate, be lower than the thermal power the brakes can dissipate.

Two reference conditions are usually considered: Driving in continuous acceleration-braking cycles, and downhill running in which the speed is kept constant with the use of brakes.

In the first case, neglecting all resistance to motion, the energy to be dissipated during braking from speed V to zero is equal to the kinetic energy of the vehicle. The worst case is a number of accelerations from standstill to speed V, performed in the lowest possible time, followed by braking to standstill. The average power on an acceleration-deceleration cycle is

$$|P| = \frac{m_e V^2}{2(t_a + t_b)} \; . \tag{24.30}$$

The acceleration time t_a increases with V and can be computed with the method used in the previous chapter. Braking time t_b is, at least,

$$t_b = \frac{V}{g\eta_b|\mu_{x_{max}}|} \; .$$

The average braking power first increases with the speed V, and then decreases again since the acceleration time increases far more than the braking energy. When the vehicle is approaching its maximum speed t_a tends to infinity and the average power tends to zero.

In the case of downhill driving, the speed is assumed to be held constant by the use of brakes. The average power is then coincident with the power to be dissipated in each instant, since it is not possible that in the long run large quantities of heat are stored in the brakes. It then follows:

$$|P| = |Vmg\sin(\alpha)| \; . \tag{24.31}$$

The power that must be dissipated increases linearly with V. The speed must then be limited, and the braking effect of the engine must be exploited on long downhill slopes.

Industrial vehicles are sometimes supplied with devices to maintain constant the speed when driving downhill to prevent the brakes from over-heating. By limiting the speed as a function of α, that is by stating a function $V = V(\alpha)$, the average power can be expressed as a function of the speed of the type shown in Fig. 24.8.

Acceleration-deceleration cycles are usually the critical condition for passenger vehicles and, above all, for sports cars, while for industrial vehicles the worst condition is downhill driving. Plots of the type seen in Fig. 24.8 give an indication of the maximum value of the average power the brakes must dissipate, making them useful for designing their cooling system.

If the road conditions or the driving style require significant use of the brakes, they may be required to store much heat and become very hot, with consequent thermo-mechanical problems. To give an idea of the magnitude of the temperatures reached by some components of the braking system, some experimental temperature readings obtained on mountain and hill roads are reported in Fig. 24.9.

In vehicles with regenerative braking capabilities, the average power computed above gives an idea of how much energy can be stored, and thus determines

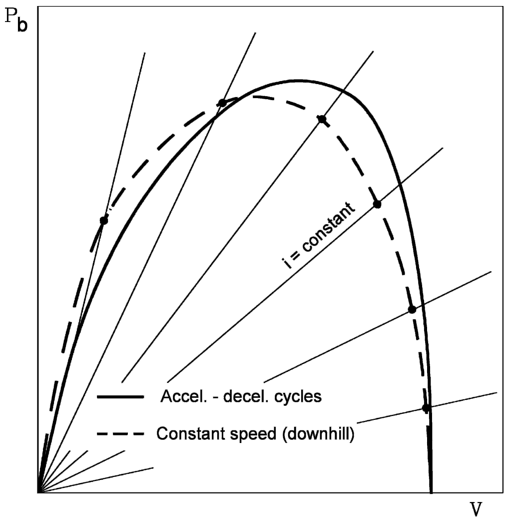

FIGURE 24.8. Power to be dissipated by brakes.

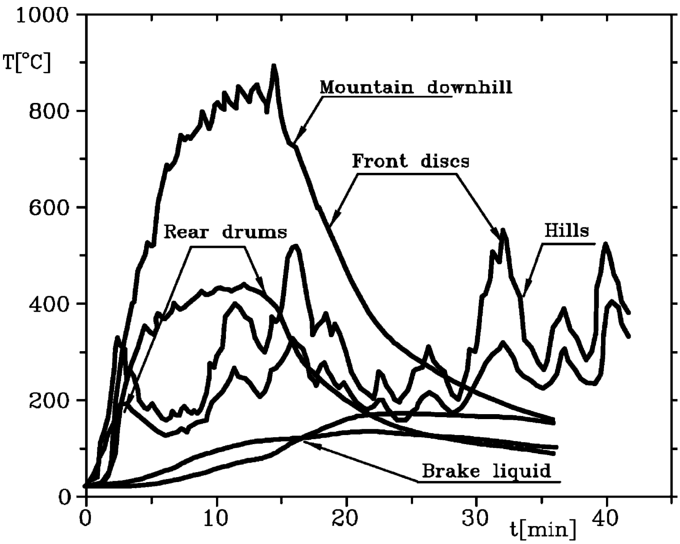

FIGURE 24.9. Time history of the temperature of the brakes and of the braking fluid during testing of a car on different roads.

the capacity of the accumulator. An accumulator able to store braking energy is large enough to provide true hybrid capabilities, i.e. to uncouple the requirements of the vehicle from the instantaneous power of the engine.

At any rate, vehicles with regenerative braking must have a conventional braking system as well. Regenerative braking is usually performed on only one axle, usually the driving axle, with the exception of schemes such as that shown in

Fig. 22.7a1. Braking power is limited by both the transmission and the ability of the accumulator to accept high power levels. The conventional braking system works in less demanding conditions, since it provides emergency braking only rather than frequent slowing in normal use.