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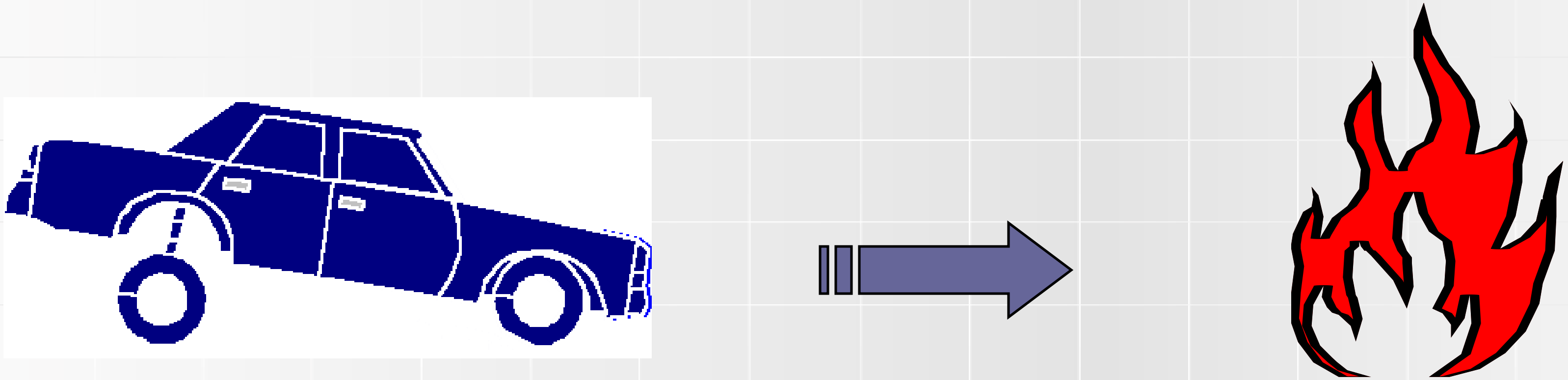
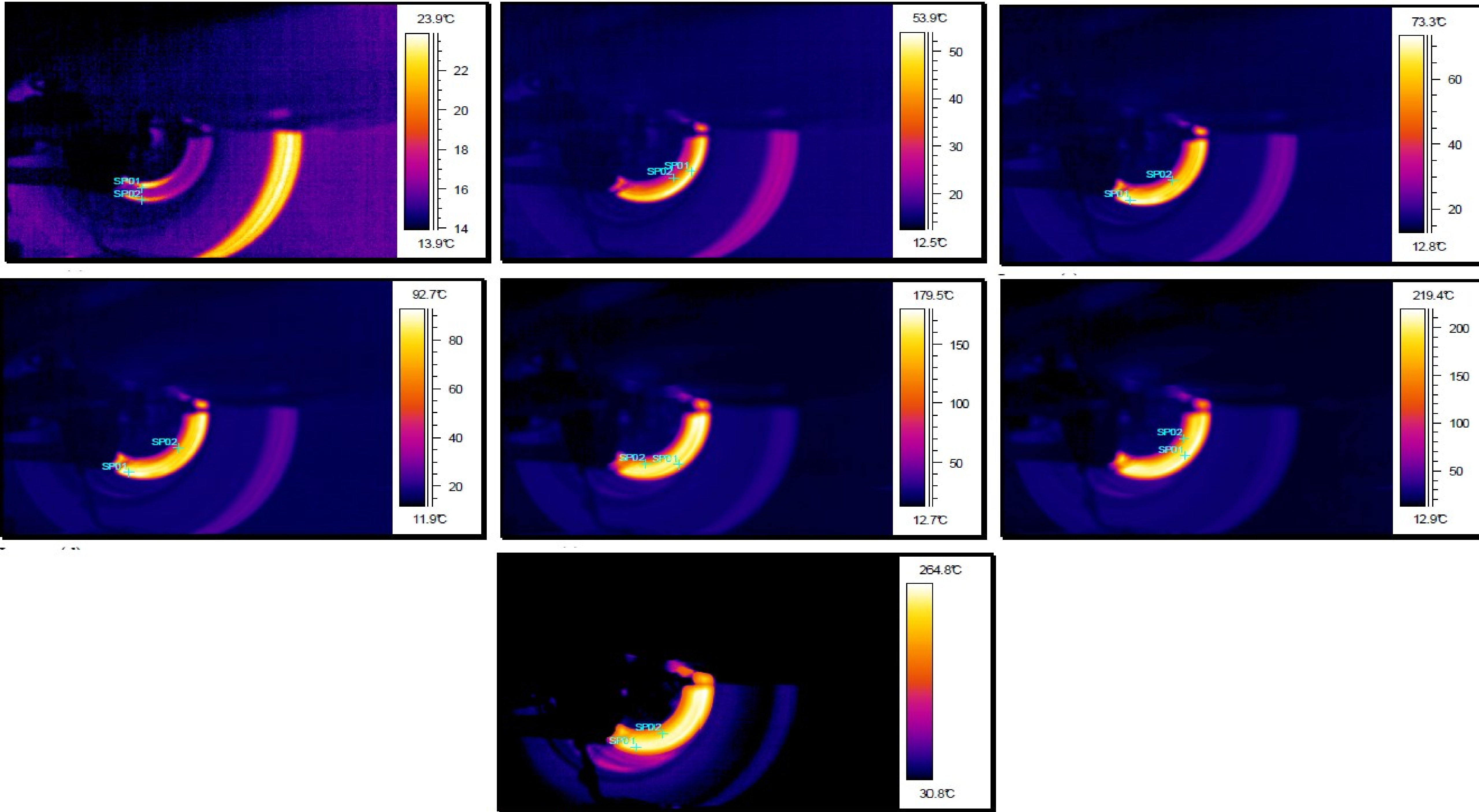
Braking performance

Energy Conversion



The brake system converts the kinetic energy of vehicle motion into *heat*


23.9℃
22
20
18
16
14
13.9℃
SP01
SP02
53.9℃
50
40
30
20
12.5℃
SP01
SP02
73.3℃
60
40
20
12.8℃
SP02
SP01
92.7℃
80
60
40
20
11.9℃
SP02
SP01
179.5℃
150
100
50
12.7℃
SP02
SP01
219.4℃
200
150
100
50
12.9℃
SP02
SP01
264.8℃
30.8℃
SP02
SP01

The functions and conditions of a brake system

1. The braking system must decelerate a vehicle in a controlled and repeatable fashion and when appropriate cause the vehicle to stop.

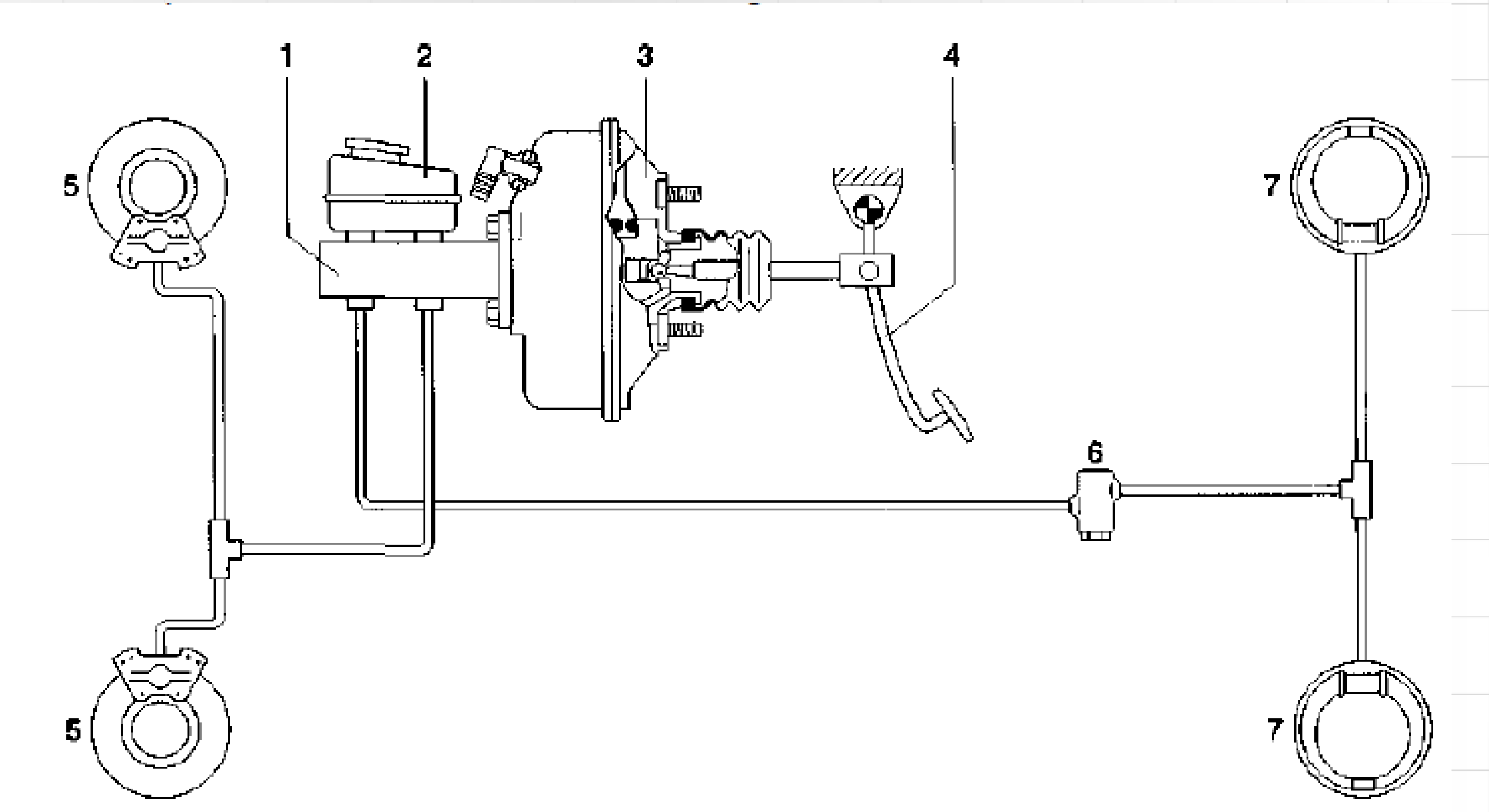
2. The braking system should permit the vehicle to maintain a constant speed when travelling downhill.

3. The braking system must hold the vehicle stationary when on a flat or on a gradient.

System design methodology

1. **Energy source**
   - This includes all those components which generate, store or release energy required by the braking system.
   - standard passenger cars -muscular pedal effort
   - Alternative sources of energy include power braking systems, surge brakes, drop weight brakes, electric and spring brakes.
2. **Modulation system**
   - those elements of the brake system which are used to control the level of braking effort applied to each brake.
3. **Transmission system**
   - The components through which energy travels to the wheel brakes.
   - Brake lines (rigid tubes) and brake hoses (flexible tubes)
   - rods, levers, cams and cables to transmit energy

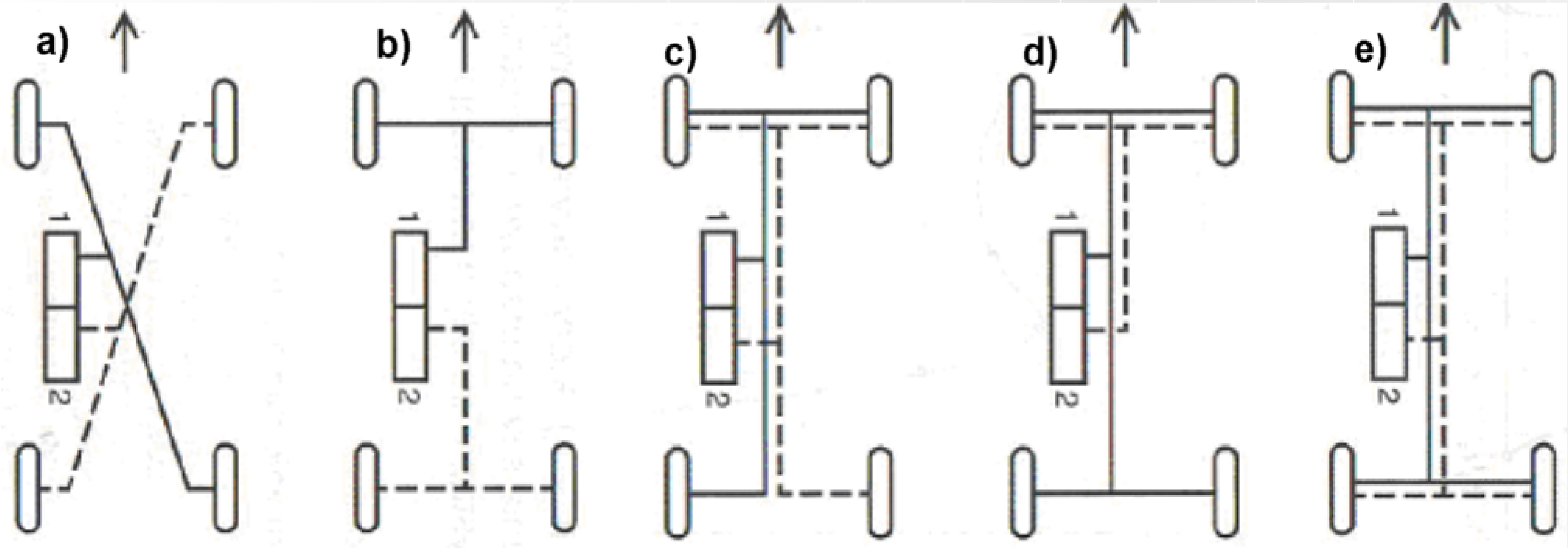
1– master cylinder,
2 – brake fluid reservoir,
3 – power brake device,
4 – brake pedal,
5– brake disc,
6 – brake proportioning,
7 – drum brakes.



Brake system components and configurations

Brake system layouts

- Legislative requirements demand a dual circuit transmission system to be installed on all road vehicles.
- The *II design* is characterized by separate circuits for both the front and rear axles. The II design is often found on vehicles that are rear heavy
- in the **X configuration**, each circuit actuates one wheel at the front and the diagonally opposed rear wheel. the X layout has application on vehicles that are front heavy



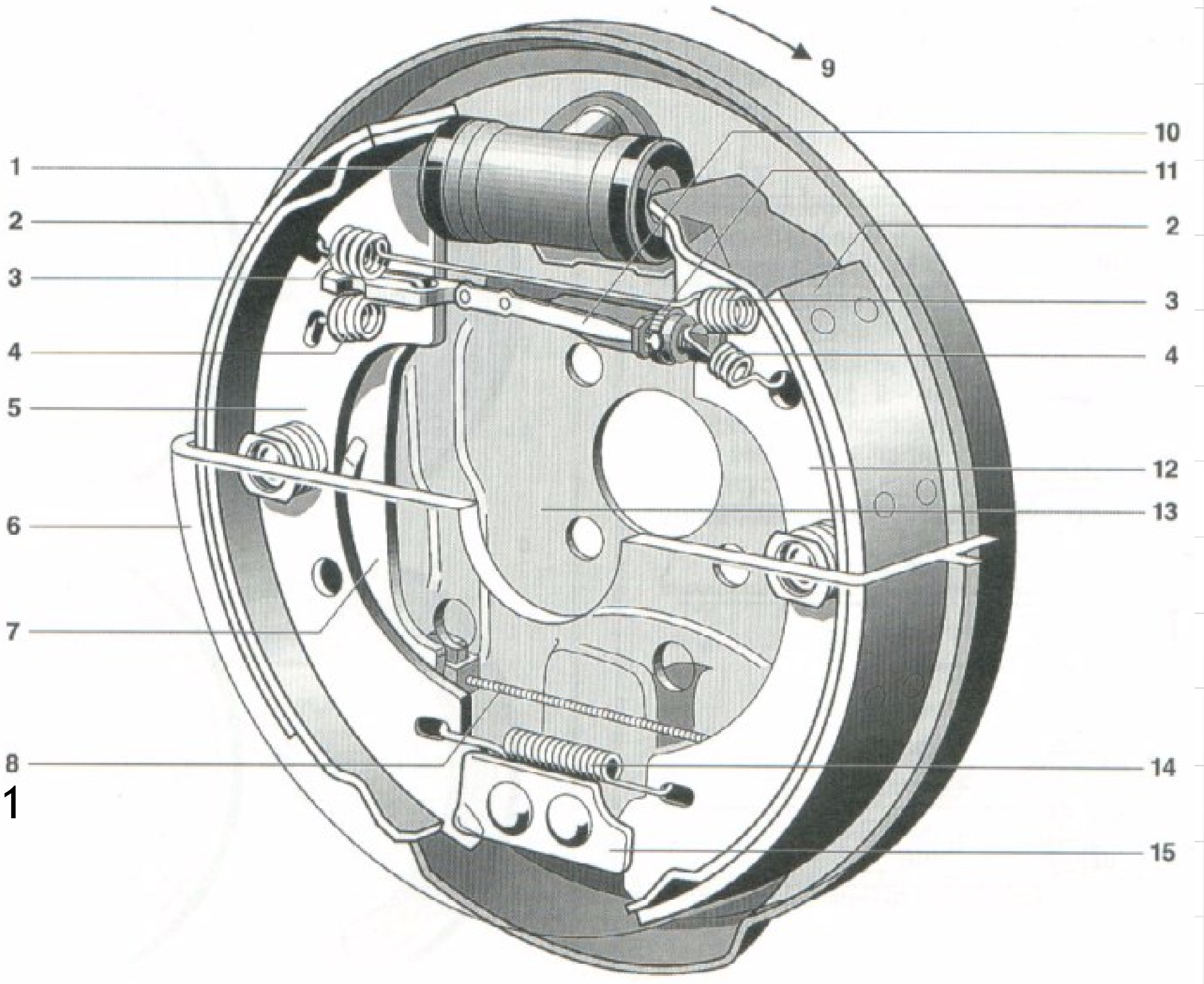
a- X layout b – II layout c – LL layout d – HT layout e – HH layout

1 - first brake circuit, 2 – second brake circuit

Drum brakes construction

Drum brakes – friction mechanism where brake force is created between brake drum (rotational) and brake shoe (stationary);

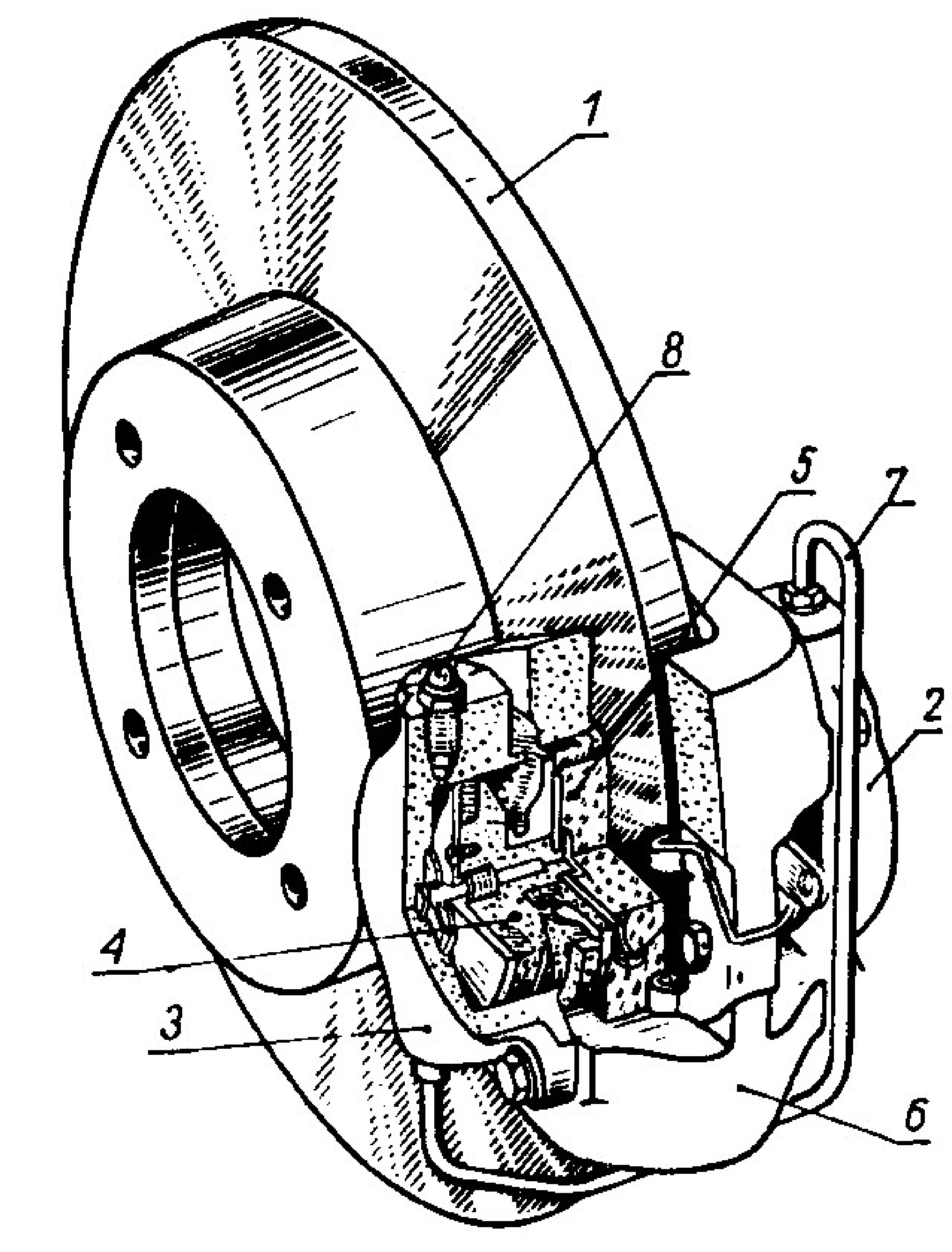
1. Hydraulic cylinder;
2. Friction lining;
3. Spring;
4. Regulation mechanism spring;
5. Brake shoe (backward);
6. Brake drum;
7. Emergency brake leaver;
8. Emergency brake linkage;
9. Drum direction of rotation;
10. Regulation mechanism thermal element; 1
11. Regulation nut;
12. Brake shoe (concurrent);
13. Carrier;
14. Spring;
15. Brake shoe support



Disc brakes

Disc brakes - friction mechanism in which brake force is created between the surface of brake disc and brake pad.

1. brake disc,
2. cylinder - inner
3. cylinder - outer
4. pistons,
5. brake pad (with lining),
6. caliper yoke,
7. hose connecting the cylinders
8. bleeding bolt



Brake disks addition cooling

Fundamentals of braking

A straightforward kinematic analysis assuming straight line (one-dimensional) motion and constant deceleration provides a ready indication of stopping distance.

In part 1the total distance travelled by the vehicle moving with constant velocity U is:

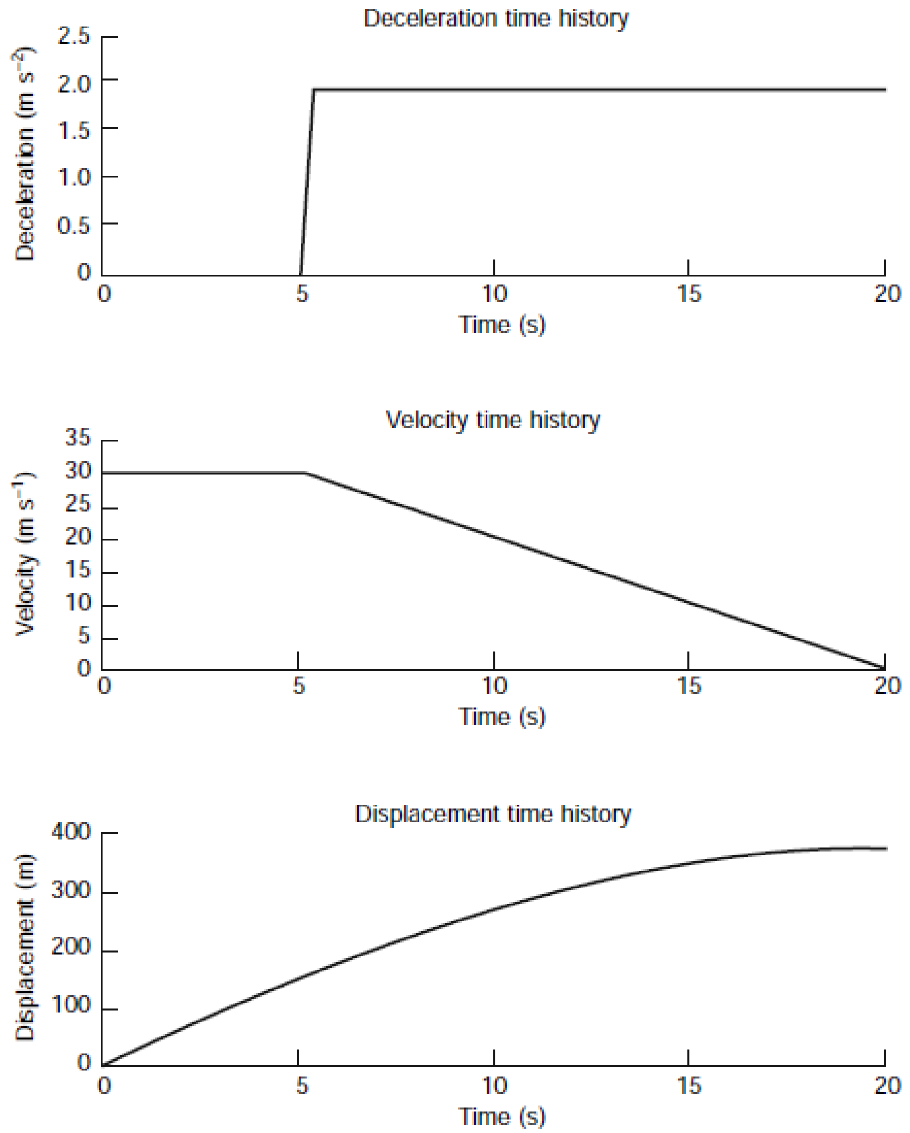
$$S_1 = Ut_1$$

In part 2 the vehicle is decelerated at a constant rate until such time as the vehicle comes to rest.

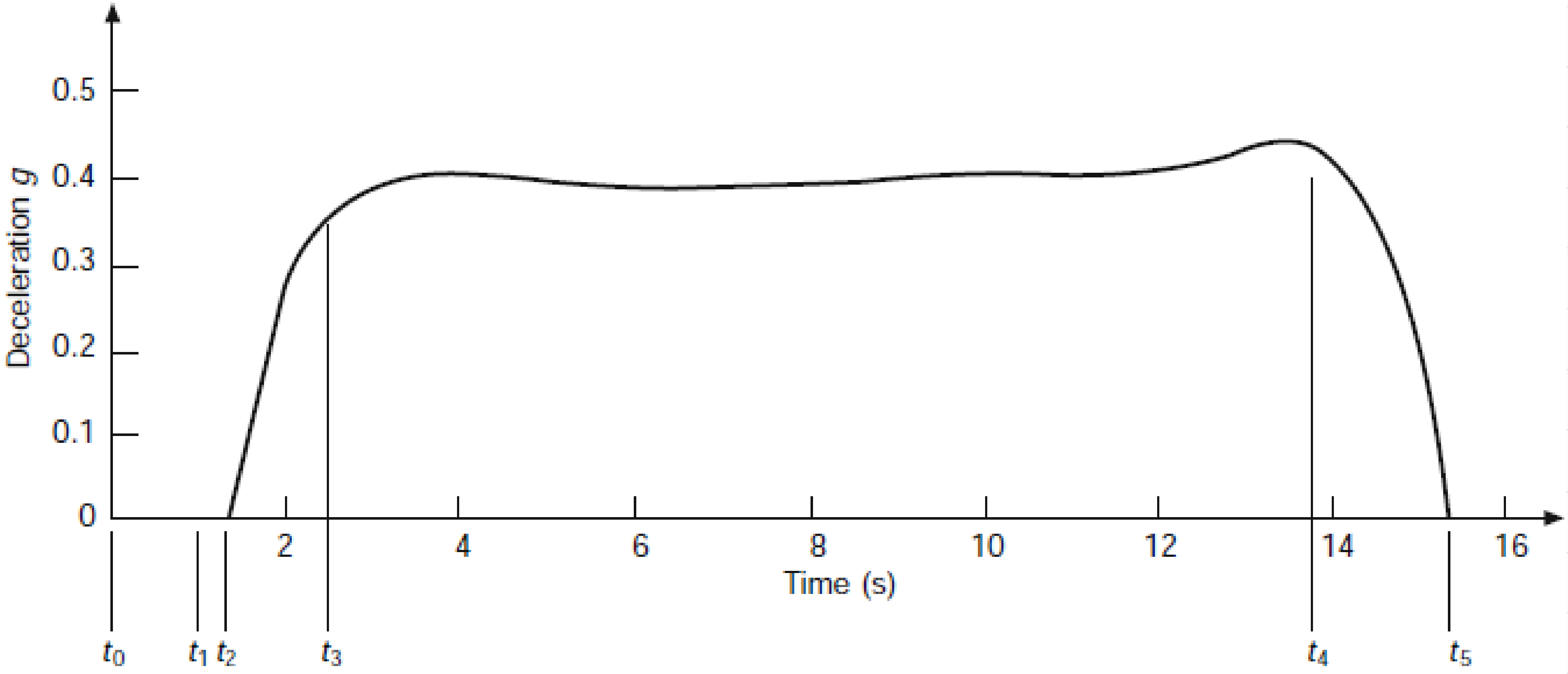
$$S_2 = \frac{Ut_2}{2} = \frac{U^2}{2a}$$

Thus, the total stopping distance is simply

$$S_i = S_1 + S_2 = Ut_1 + \frac{U^2}{2a}$$



Fundamentals of braking



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Fundamentals of braking

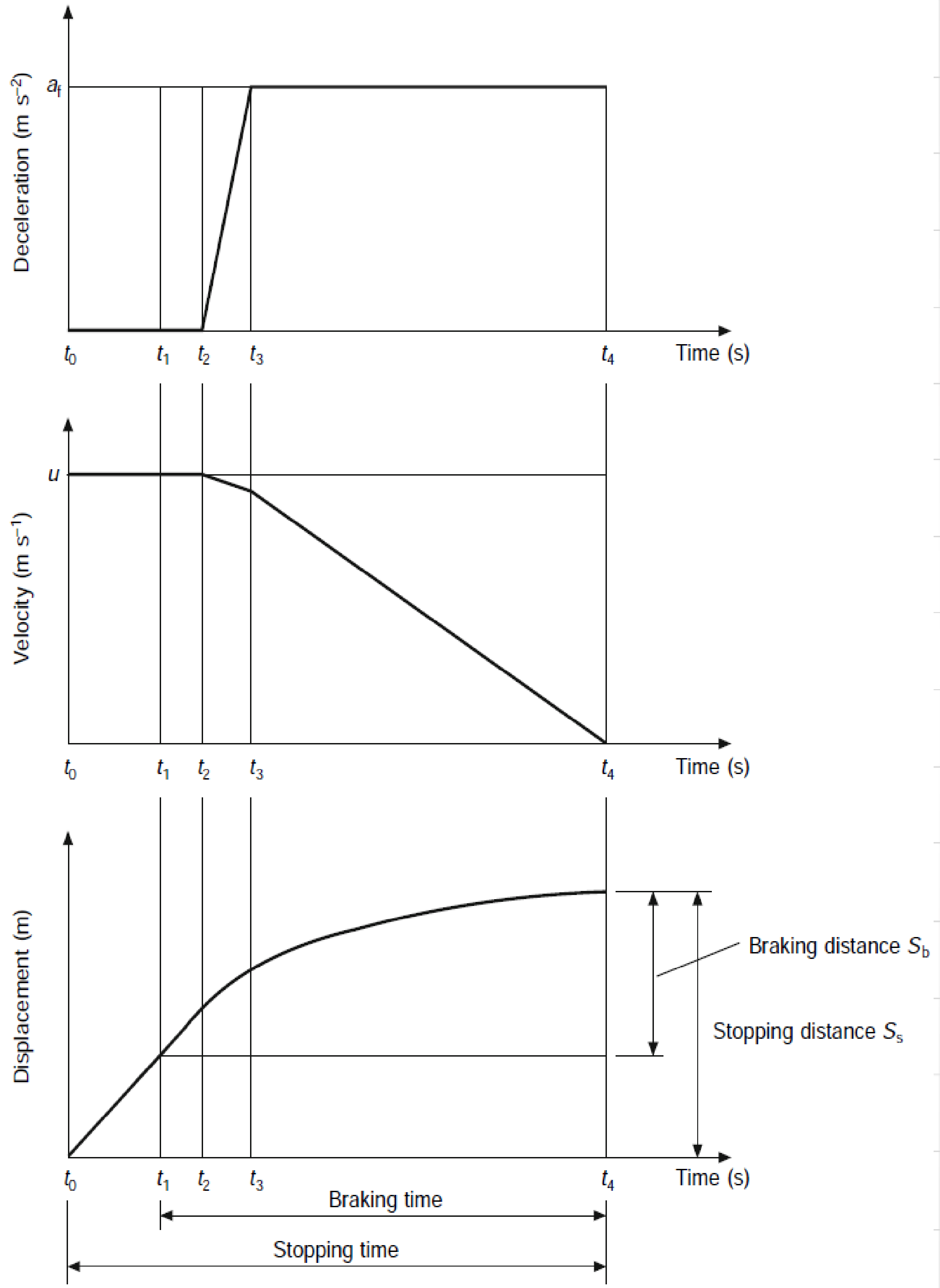
$$S_1 = U(t_1 - t_0)$$

$$S_2 = U(t_2 - t_1)$$

$$S_3 = U(t_3 - t_2) - \frac{a_f (t_3 - t_2)^2}{6}$$

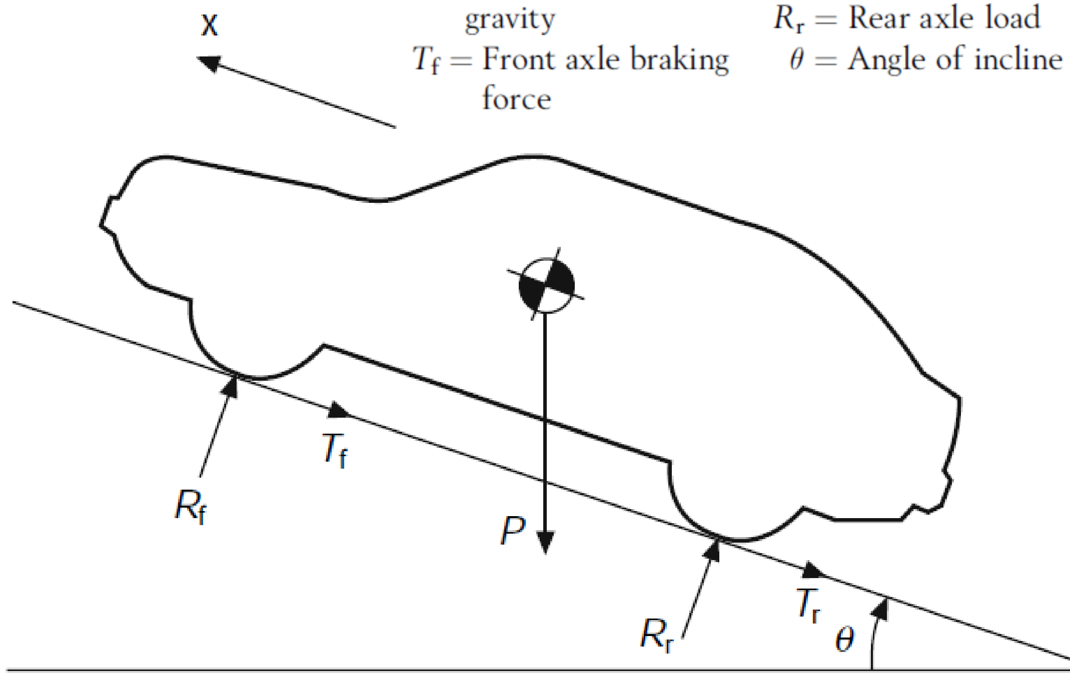
$$S_4 = \frac{1}{2a_f}\left[U^2 + \frac{a_f^2 (t_3 - t_2)^2}{4} - U a_f (t_3 - t_2)\right]$$

$$S_s = \sum_{i=1}^{4} S_i$$



Kinetics of a braking vehicle
Braking duration tb

M = Vehicle mass
P = Vehicle weight
g = Acceleration due to gravity
T_f = Front axle braking force
T_r = Rear axle braking force
R_f = Front axle load
R_r = Rear axle load
θ = Angle of incline



Assuming x is positive in the direction of travel

$$\sum F_x = M\ddot{x}$$

$$-T_f - T_r - D - P\sin\theta = M\ddot{x}$$

If an additional variable for linear deceleration, d, is defined such that

$$d = -\ddot{x}$$

$$Md = T_f + T_r + D + P\sin\theta = T$$

T is the sum of all those forces that contribute to the overall braking effort

$$d = \frac{T}{M} = -\frac{dv}{dt}$$

$$\int_{v_0}^{v_f} dv = -\frac{T}{M}\int_0^{t_b} dt$$

$$v_0 - v_f = \frac{T}{M}t_b$$

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Kinetics of a braking vehicle

stopping distance

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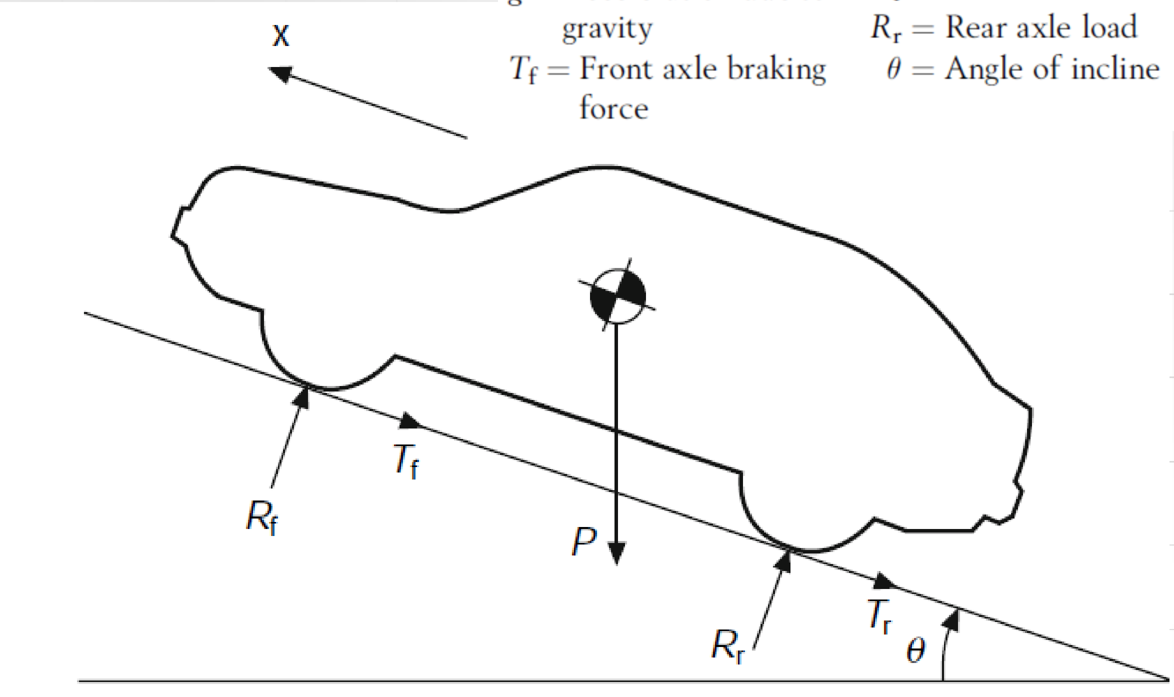


$$d = \frac{T}{M} = -\frac{dv}{dt}$$

$$v = \frac{dx}{dt}$$

$$\frac{T}{M}\int_{x_0}^{x_f} dx = -\int_{v_0}^{v_f} v\,dv$$

$$\frac{T}{M}(x_f - x_0) = \frac{Tx}{M} = \frac{v_0^2 - v_f^2}{2}$$

When considering a stop, the final velocity v_f is zero and so the stopping distance x is,

$$\boxed{x = \frac{Mv_0^2}{2T}}$$

and the time, t_b, taken to stop the vehicle is,

$$\boxed{t_b = \frac{Mv_0}{T} = \frac{v_0}{d}}$$

Tyre–road friction

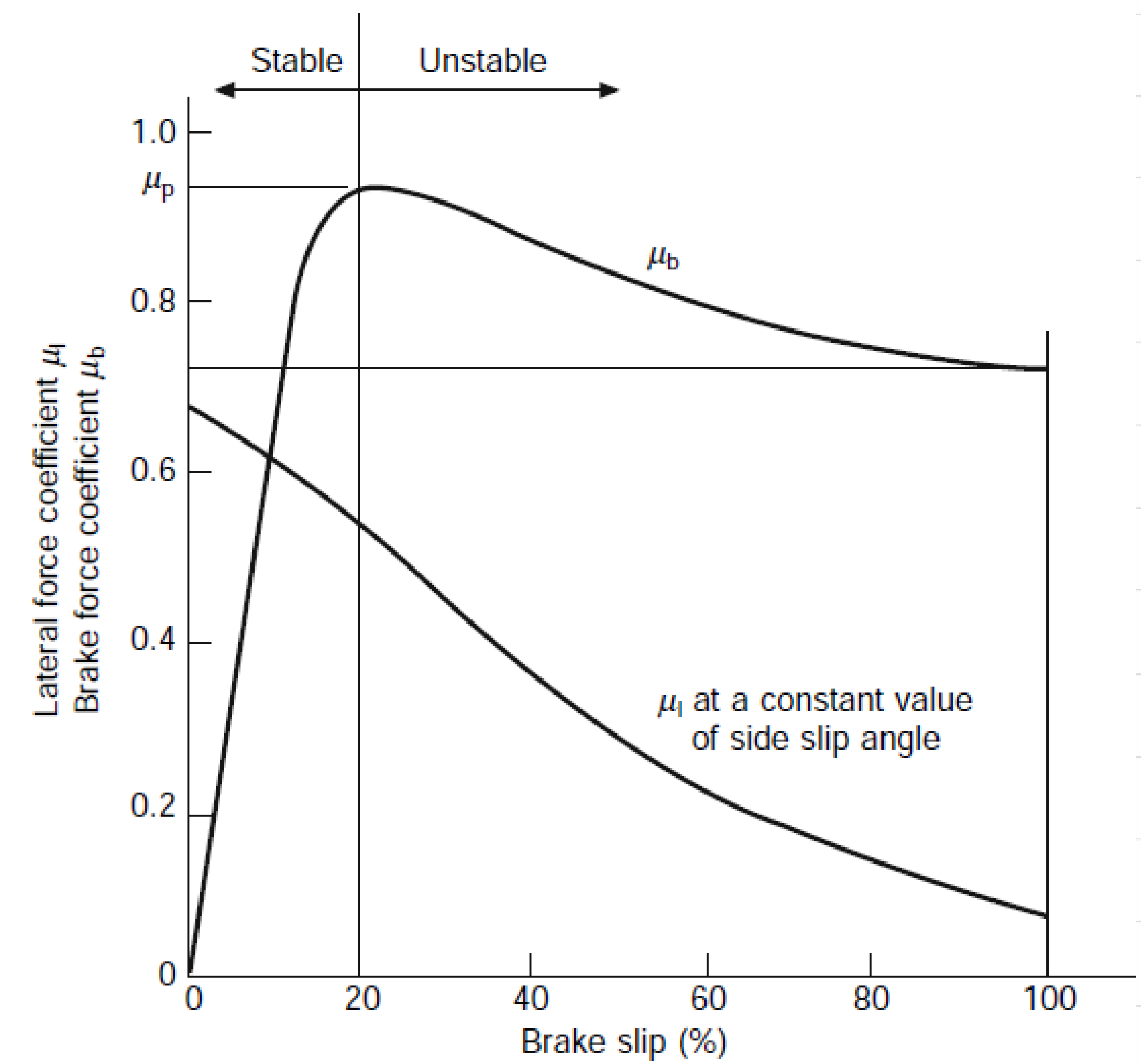
The brake force, F_b, which acts at the interface between a single wheel and the road is related to the brake torque, T_b, ***as long as all the wheels are rolling***.

$$F_b = \frac{T_b}{r}$$

The brake force F_b cannot increase without bound as it is limited by the extent of the friction coupling between the tyre and the road.

Hysteresis and adhesion are the two mechanisms responsible for friction coupling.

$$\text{slip} = \frac{\text{slip velocity in contact patch}}{\text{forward velocity}} = \frac{v - \omega r}{v}$$



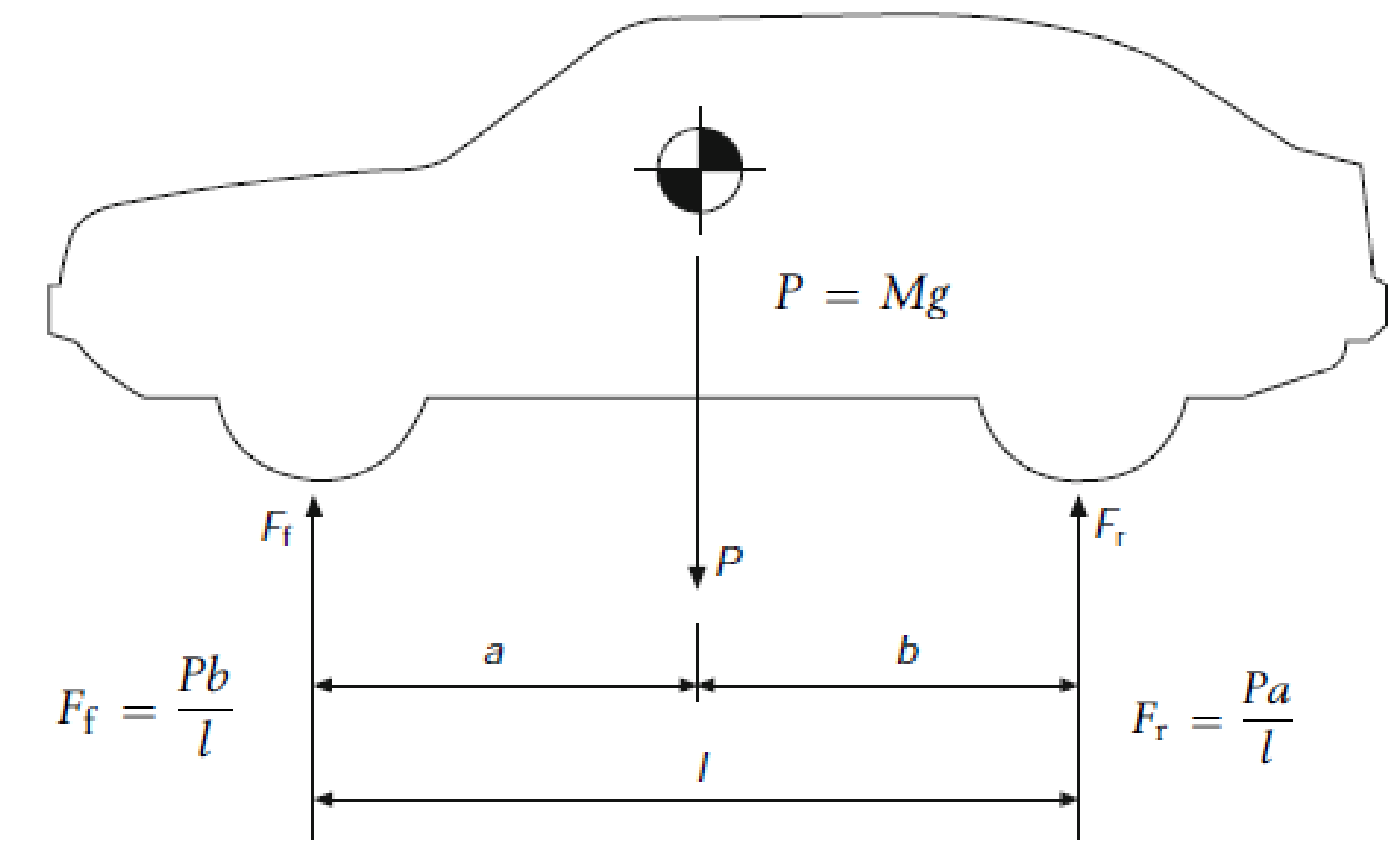
Brake proportioning

The vertical loads carried by the front and rear wheels of a rigid, two axle vehicle are not, in general, equal. In order to efficiently utilize the available tyre–road adhesion the braking effort must be apportioned between the front and rear of the vehicle in an intelligent and controlled fashion.

- A vehicle being unable to generate the necessary deceleration for a given pedal pressure.
- Front axle lock, in which the vehicle remains stable yet suffers from a loss of steering control.
- Rear axle lock that causes the vehicle to become unstable.

Brake proportioning

Static analysis

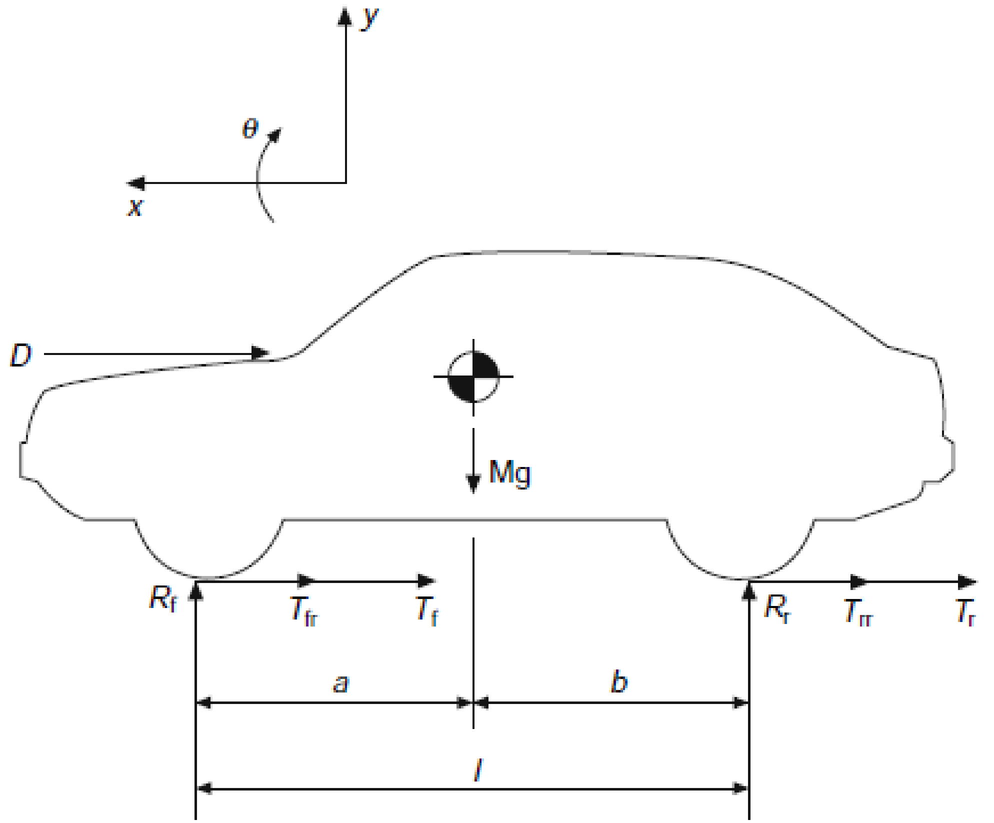


Brake proportioning

Static analysis - brake ratio

If the front and rear axles are to be on the point of locking, then the braking forces T_f and T_r acting at each axle must be in proportion to the vertical loads being carried, R_f and R_r.

$$M\ddot{x} = \sum F_x$$
$$= -D - T_f - T_r - T_{fr} - T_{rr}$$
$$Md = D + T_f + T_r + T_{fr} + T_{rr}$$



If the aerodynamic drag and rolling resistance forces are assumed to be negligible

$$Md = T_f + T_r$$

By defining z to be the vehicle deceleration as a proportion of g:

$$z = \frac{d}{g}$$

$$Mgz = T_f + T_r = Pz$$

Brake proportioning

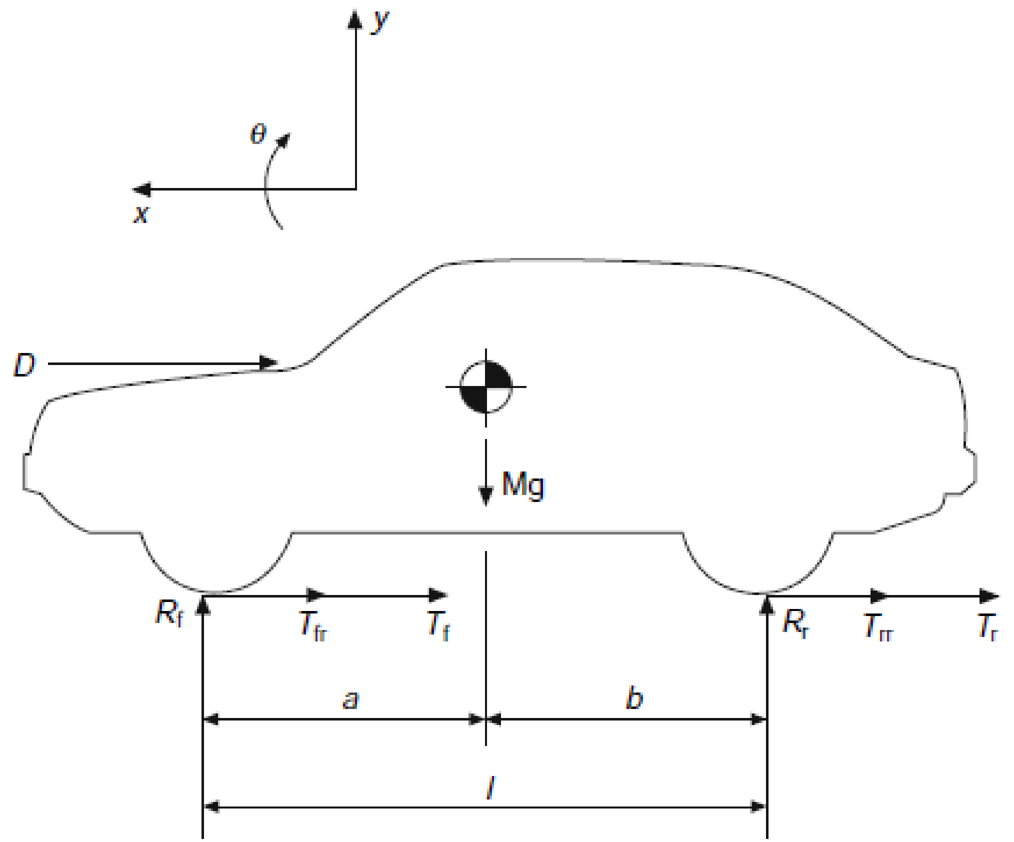
Static analysis - brake ratio

The vertical direction

$$M\ddot{y} = \sum Fy = R_r + R_f - Mg = 0$$

the Ø direction, taking moments about the center of gravity of the vehicle leads to

$$I\ddot{\theta} = \sum M_{cg} = R_f a - R_r b - T_f h - T_r h = 0$$



$$R_f = \frac{Mgb}{l} + \frac{h}{l}(T_f + T_r)$$

$$R_r = \frac{Mga}{l} - \frac{h}{l}(T_f + T_r)$$

$$F_f = \frac{Pb}{l}$$

$$F_r = \frac{Pa}{l}$$

$$R_f = F_f + \frac{Pzh}{l}$$

$$R_r = F_r - \frac{Pzh}{l}$$

Brake proportioning

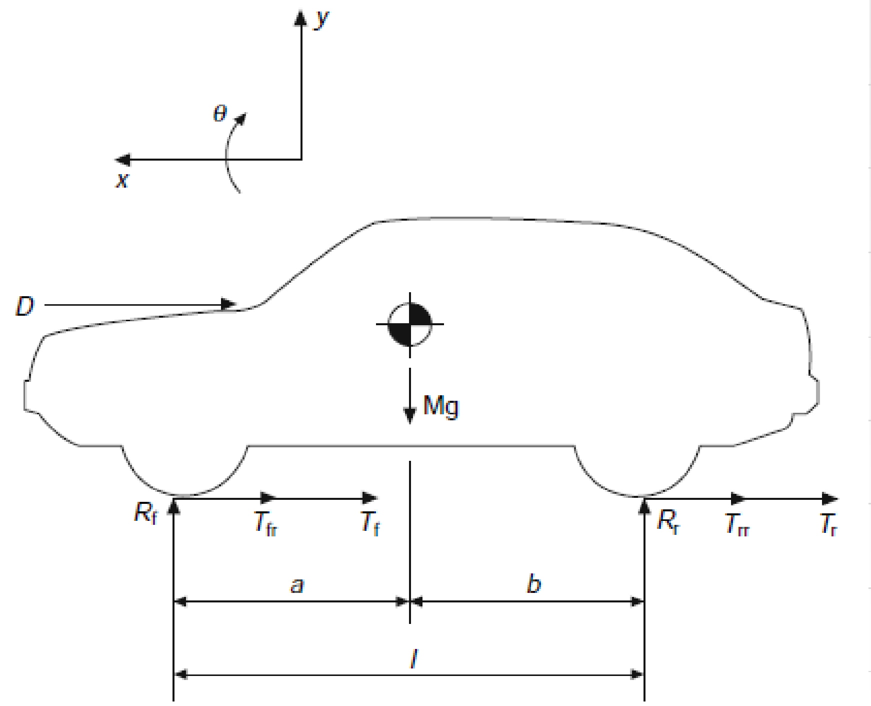
Static analysis - brake ratio

- a change in axle load in favour of the front axle occurs during a braking manoeuvrer.
- in order for each axle to be simultaneously on the verge of locking, the brake force generated at each axle must be in direct proportion to the vertical axle load
- fully utilize the available tyre–ground adhesion, the braking system must support an infinitely variable brake ratio

Brake proportioning

fixed brake ratio

If the ratio has been set so that the front axle locks in preference to the rear, then the brake force generated at the front axle when about to lock is



$$T_f = \mu R_f = \mu\left(F_f + \frac{Pzh}{l}\right)$$

During the same braking event, the rear axle is also generating a brake force that has not exceeded its limiting value

$$R = \frac{x_f}{x_r} = \frac{T_f}{T_r}$$

Hence

$$T_r = T_f \frac{x_r}{x_f} = \mu\left(F_f + \frac{Pzh}{l}\right)\frac{x_r}{x_f}$$

leading to a total brake force of

$$\begin{aligned} T &= Pz = T_f + T_r \\ &= \mu\left(F_f + \frac{Pzh}{l}\right) + \mu\left(F_f + \frac{Pzh}{l}\right)\frac{x_r}{x_f} \end{aligned}$$

which reduces to:

$$T = Pz = \mu\left(F_f + \frac{Pzh}{l}\right)\frac{1}{x_f}$$

the maximum value of deceleration as a proportion of g is:

$$z = \frac{\mu F_f}{P(lx_f - \mu h)}$$

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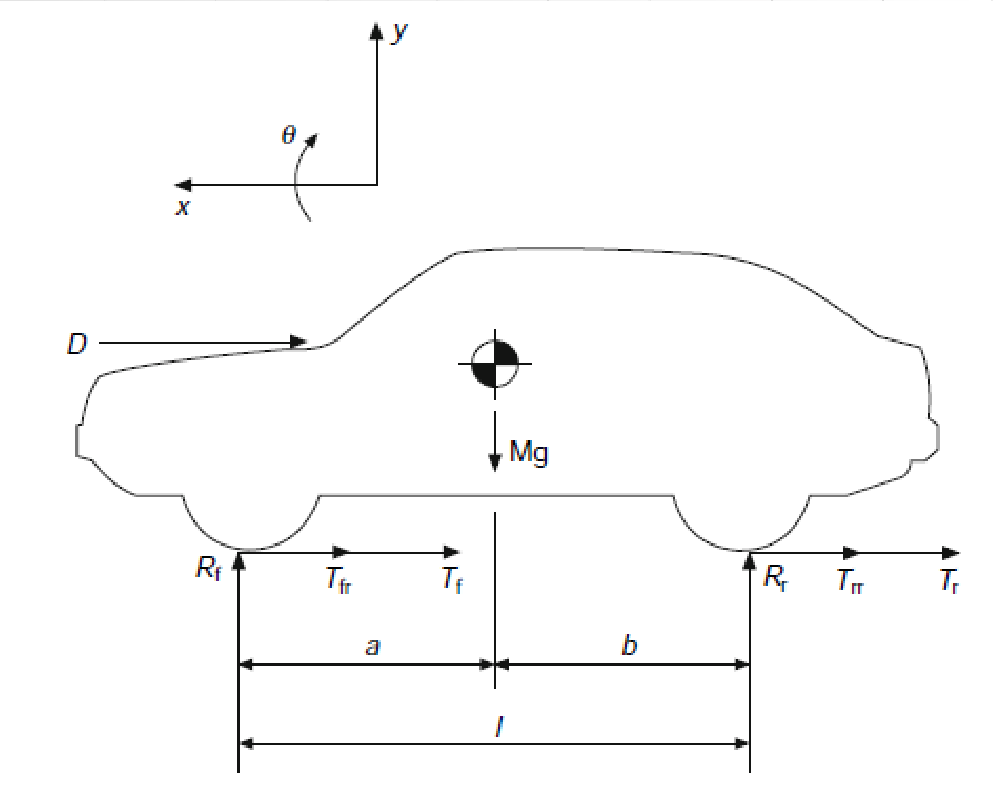
Brake proportioning

fixed brake ratio



If, the ratio has been set so that the rear axle locks in preference to the front, then the brake force generated at the rear axle when about to lock is

$$T_r = \mu R_r = \mu\left(F_r - \frac{Pzh}{l}\right)$$

this case, the brake force that is generated at the front axle is not necessarily the limiting value and its magnitude is found from the brake ratio

$$T_f = T_r \frac{x_f}{x_r} = \mu\left(F_r - \frac{Pzh}{l}\right)\frac{x_f}{x_r}$$

which leads to a total brake force of:

$$T = Pz = T_f + T_r = \mu\left(F_r - \frac{Pzh}{l}\right)\frac{x_f}{x_r} + \mu\left(F_r - \frac{Pzh}{l}\right) = \mu\left(F_r - \frac{Pzh}{l}\right)\frac{1}{x_r}$$

deceleration as a proportion of g

$$z = \frac{l\mu Fr}{P(lx_r + \mu h)}$$

Braking efficiency

The efficiency with which a brake system uses the available tyre–ground adhesion,

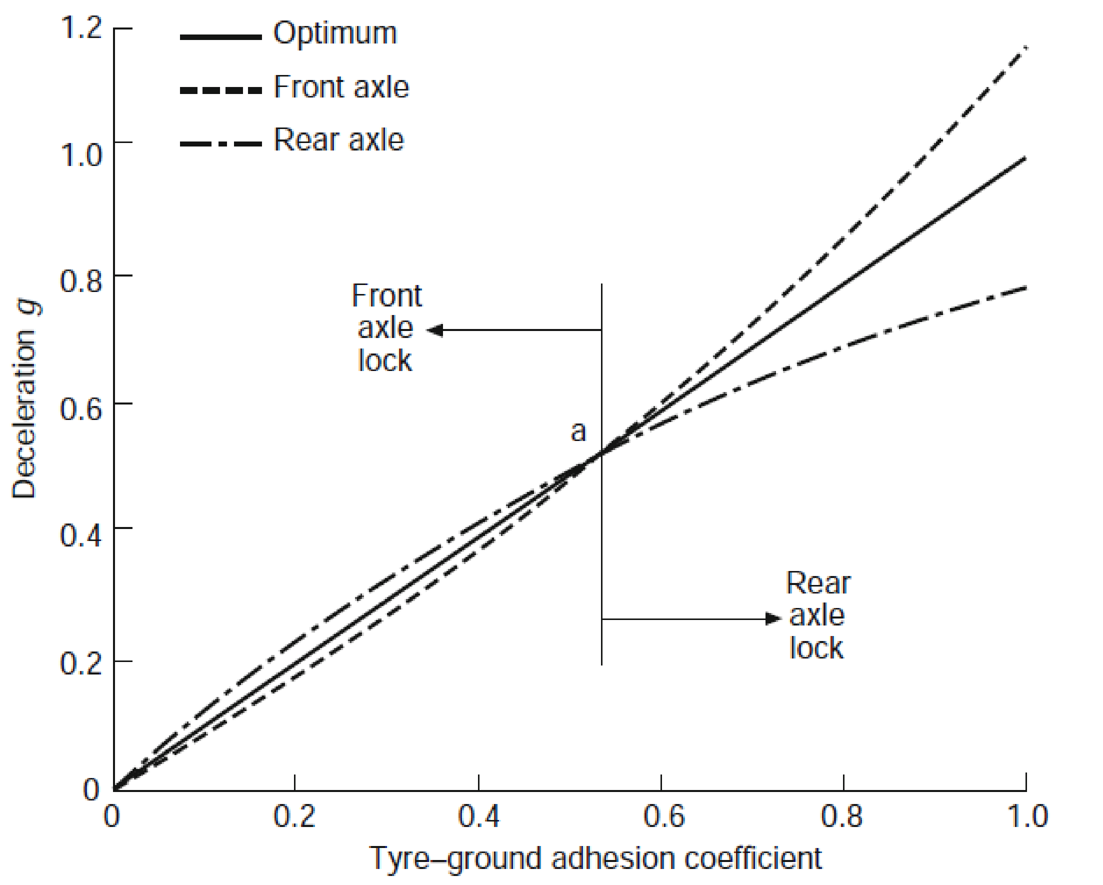
$$\eta = \frac{\text{deceleration,}}{\text{tyre– ground adhesion coefficient}} = \frac{z}{\mu}$$

front axle is about to lock

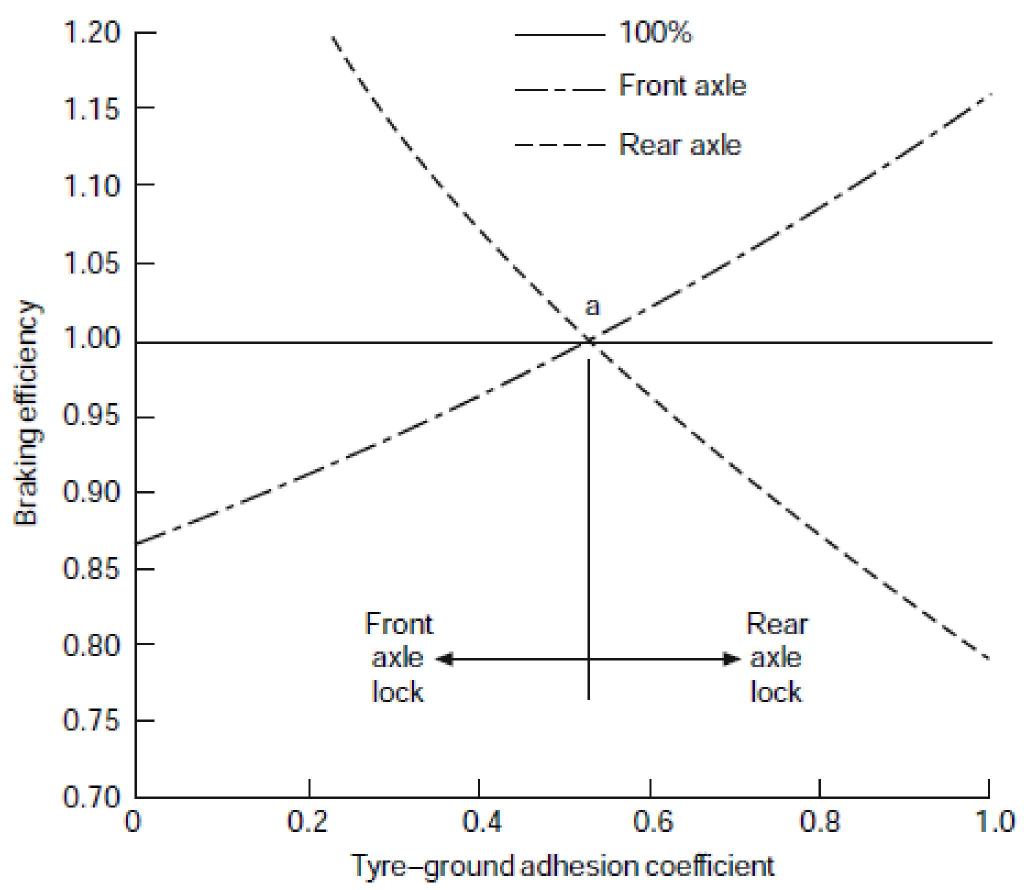
$$\eta = \frac{z}{\mu} = \frac{\dfrac{l\mu F_f}{P(lx_f - \mu h)}}{\mu} = \frac{F_f}{P\left(x_f - \dfrac{\mu h}{l}\right)}$$

rear axle is about to lock

$$\eta = \frac{z}{\mu} = \frac{\dfrac{l\mu F_r}{P(lx_r + \mu h)}}{\mu} = \frac{F_r}{P\left(x_r + \dfrac{\mu h}{l}\right)}$$



Optimum
Front axle
Rear axle
Deceleration g
1.2
1.0
0.8
0.6
0.4
0.2
0
Front axle lock
a
Rear axle lock
0
0.2
0.4
0.6
0.8
1.0
Tyre–ground adhesion coefficient



100%
Front axle
Rear axle
Braking efficiency
1.20
1.15
1.10
1.05
1.00
0.95
0.90
0.85
0.80
0.75
0.70
a
Front axle lock
Rear axle lock
0
0.2
0.4
0.6
0.8
1.0
Tyre–ground adhesion coefficient

Adhesion utilization *(f)*

minimum value of tyre– ground adhesion required to sustain a given deceleration and is defined as the ratio of the braking force to the vertical axle load during braking

$$\boxed{f_f = \frac{T_f}{R_f}}$$

$$\boxed{f_r = \frac{T_r}{R_r}}$$

Knowing that

$$R_f = F_f + \frac{Pzh}{l}$$

$$R_r = F_r - \frac{Pzh}{l}$$

Front axle brake force is proportional to $\boldsymbol{x_f Pz}$

$$f_f = \frac{x_f Pz}{F_f + \dfrac{Pzh}{l}}$$

$\boldsymbol{x_r Pz}$

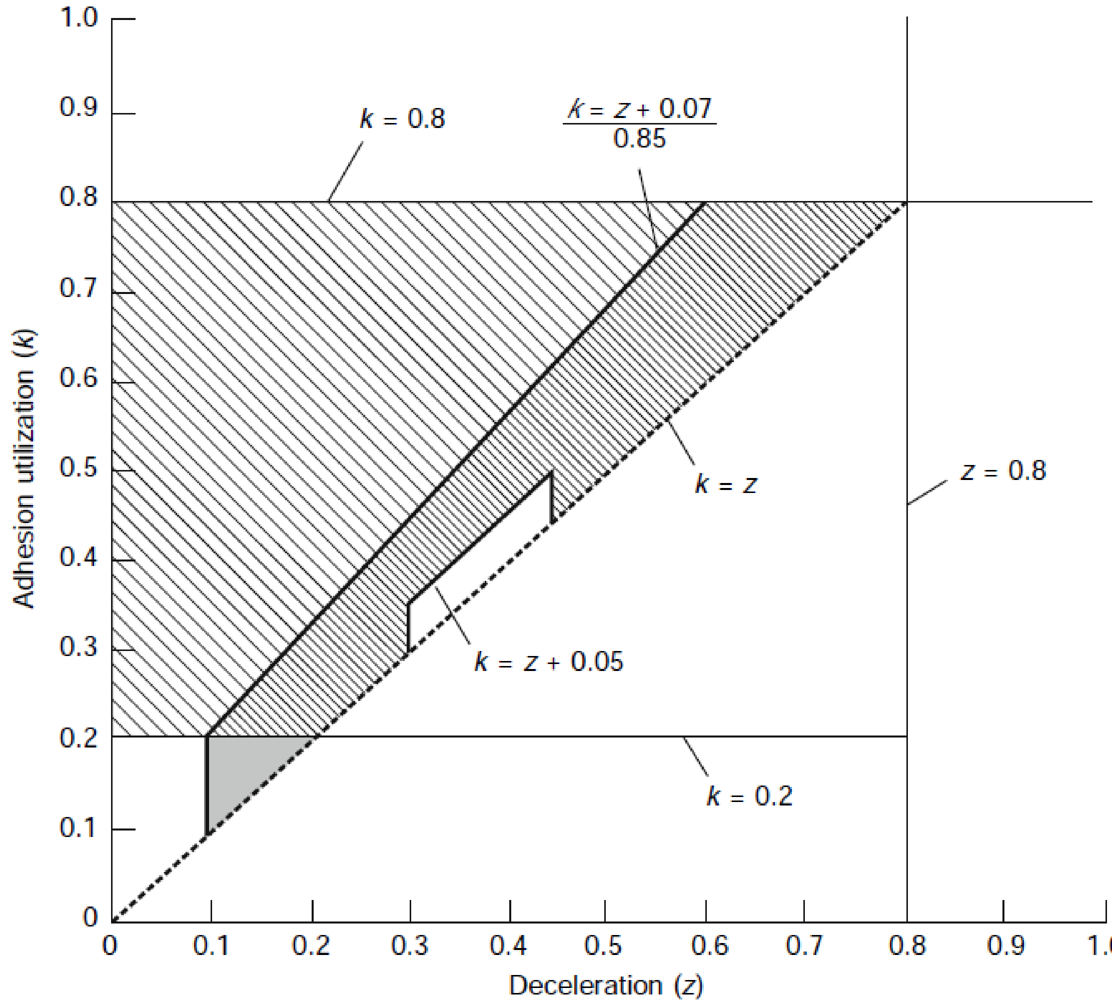
$$f_r = \frac{x_r Pz}{F_r - \dfrac{Pzh}{l}}$$

Adhesion utilization *(f)*

Legislation EEC 91/422/EEC

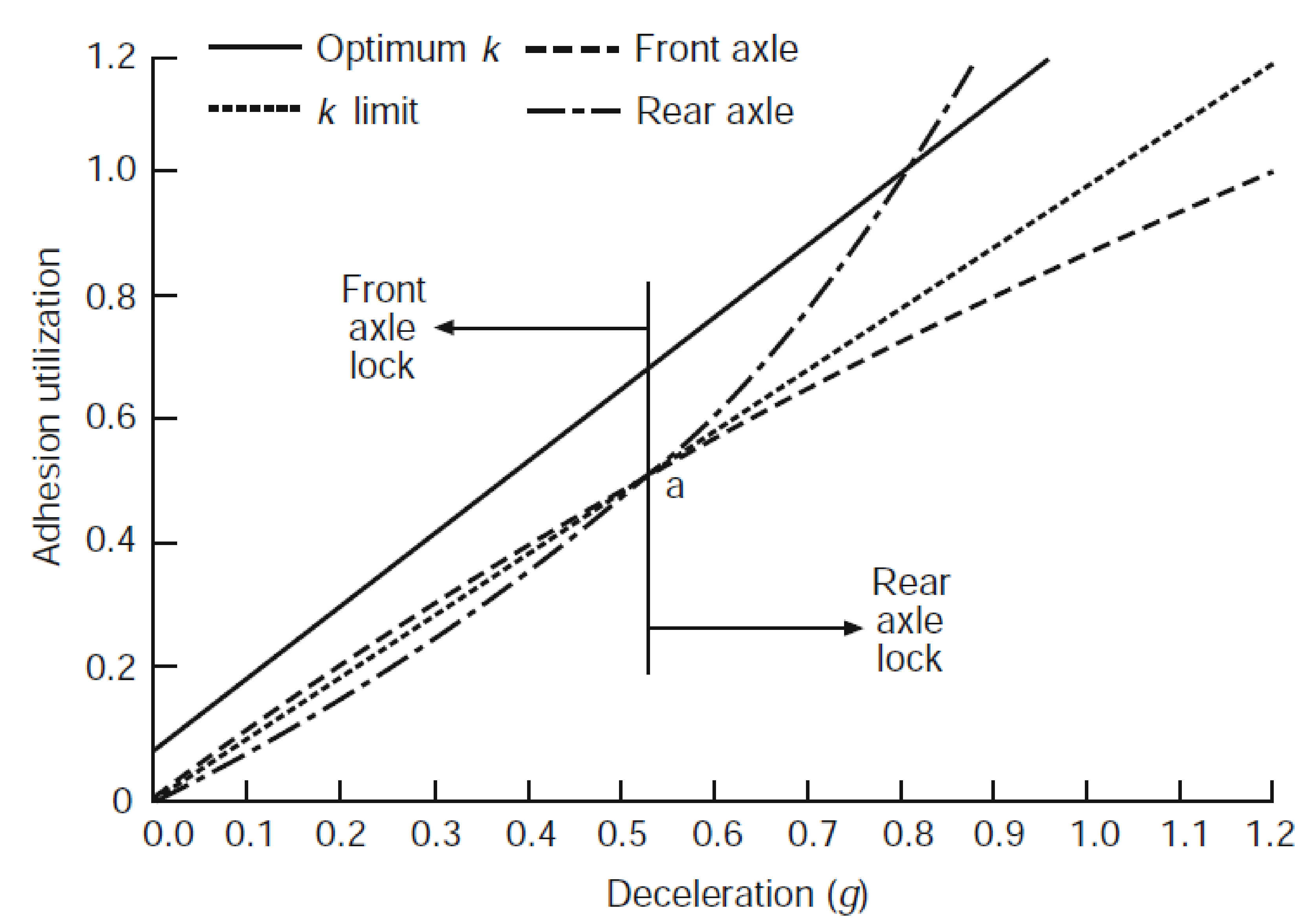
for all categories of vehicle for values of adhesion utilization between 0.2 and 0.8,

$$z \geq 0.1 + 0.85(k - 0.2)$$



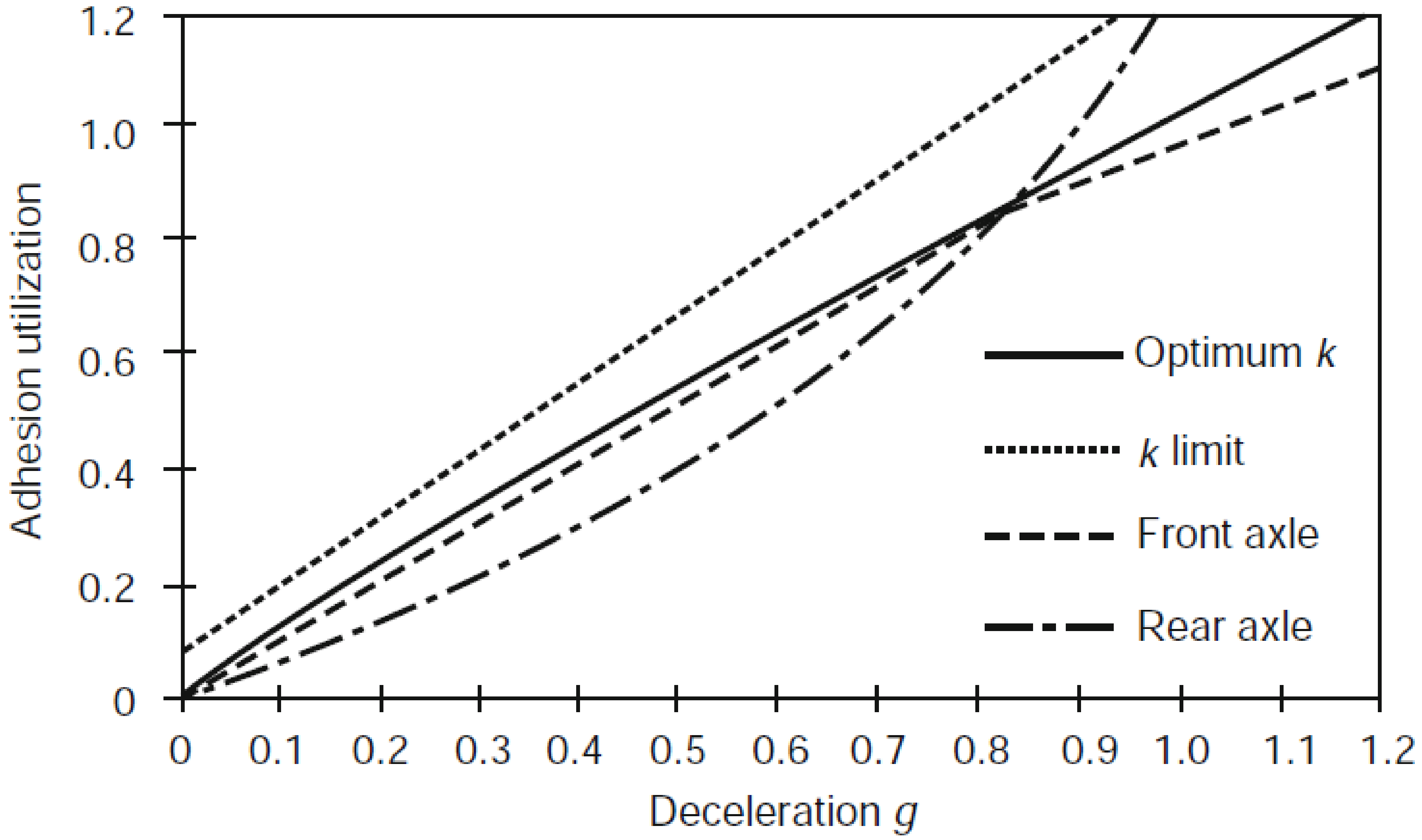
- For category M_1 vehicles, the adhesion utilization curve of the front axle must be greater than that of the rear for all load cases and values of deceleration between 0.15g and 0.8g.
- Between deceleration levels of 0.3g and 0.45g, an inversion of the adhesion utilization curves is allowed provided the rear axle adhesion curve does not exceed the line defined by k=z by more than 0.05
- The above provisions are applicable within the area defined by the lines k=0.8 and z = 0.8

Adhesion utilization *(f)*



Adhesion utilization *(f)*

$$\frac{x_f}{x_r} = \frac{0.803}{0.197}.$$



Wheel locking

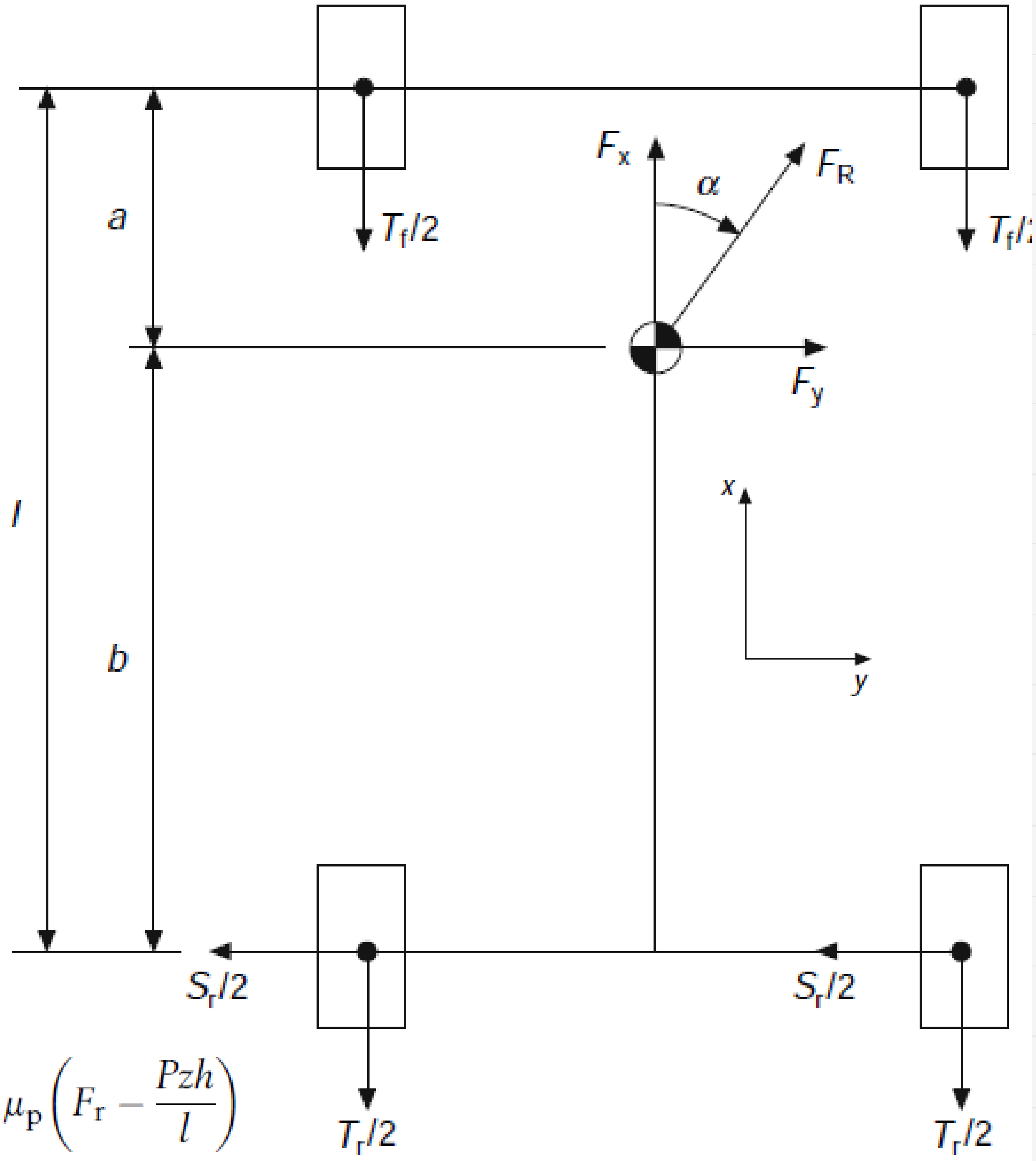
- Irregularities in the road surface or lateral forces can cause the vehicle to deviate from its direction of travel.

- The nature of the ensuing motion, which is rotational about the vehicle vertical axis, depends on which axle has locked together with the vehicle speed, tyre–ground friction coefficient, yaw moment of inertia of the vehicle body and the vehicle dimensions.

- By considering the two cases of front and rear axle lock it is possible to derive useful insight into the stability problem

Wheel locking

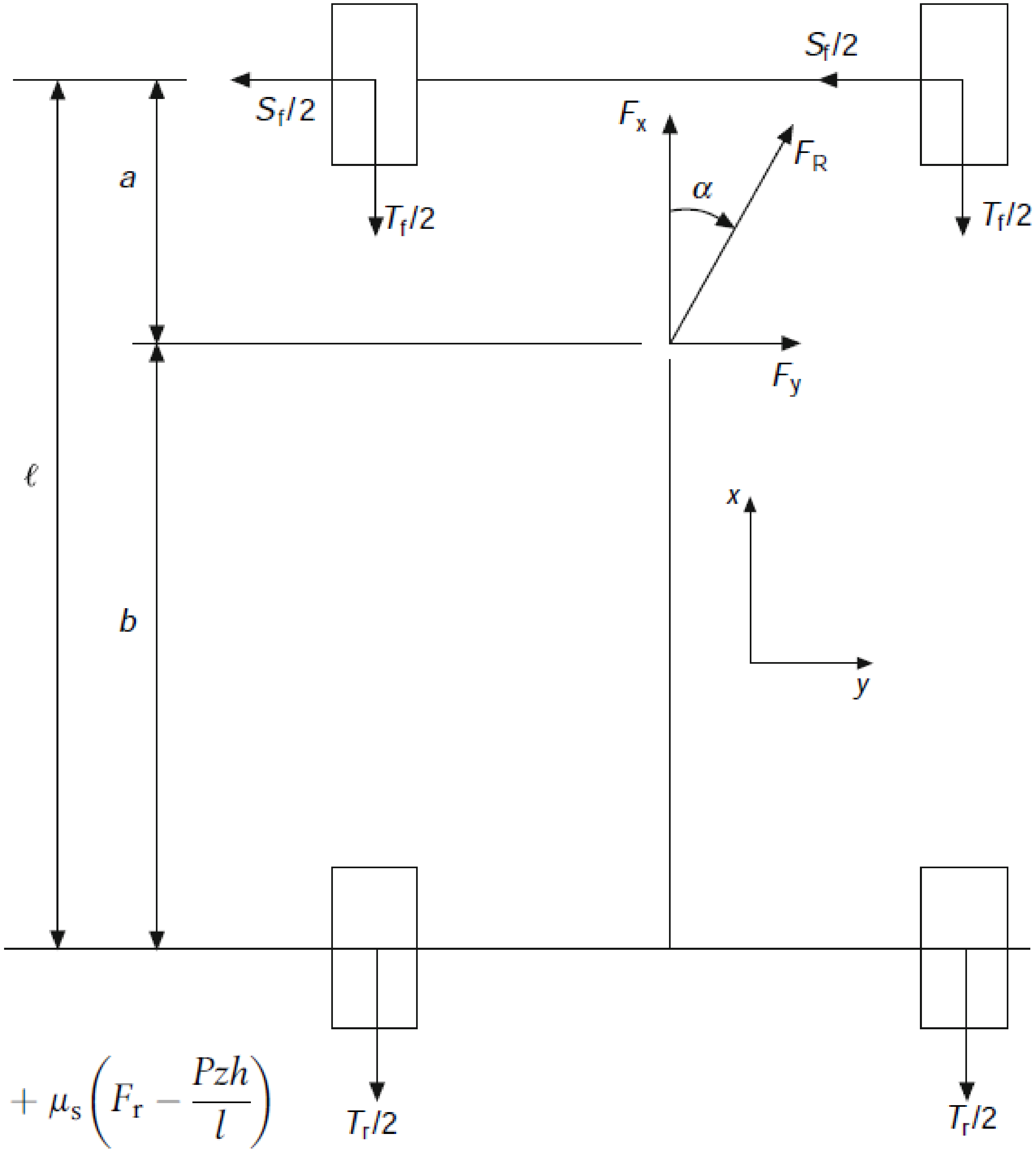
Front axle



$$Pz = \mu_s\left(F_f + \frac{Pzh}{l}\right) + \mu_p\left(F_r - \frac{Pzh}{l}\right)$$

Wheel locking

Rear axle



$$Pz = \mu_p\left(F_f + \frac{Pzh}{l}\right) + \mu_s\left(F_r - \frac{Pzh}{l}\right)$$

- **It is thus preferable, from a safety point of view, for the front axle to lock in preference to the rear as this is a stable condition and the driver is able to regain directional control of the vehicle simply by releasing the brakes**.
- If the rear axle has locked and the vehicle has begun to spin, driver reaction must be rapid if control of the situation is to be regained.
- In a collision situation, a frontal impact, linked to front axle lock, will usually result in less serious occupant injury than the possible side impact that could well be associated with the uncontrolled yawing of the vehicle that results from rear axle lock.

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It is therefore feasible to apply the preceding ideas to the formulation of a fixed brake ratio that will invariably lead to front axle lock and this is commonly applied to the design of brake systems found on passenger vehicles.

The fixed brake ratio is chosen such that for the unladen case both front and rear axles are on the verge of lock when the vehicle undertakes a 1g stop on a road Surface that has a tyre–ground adhesion coefficient of unity

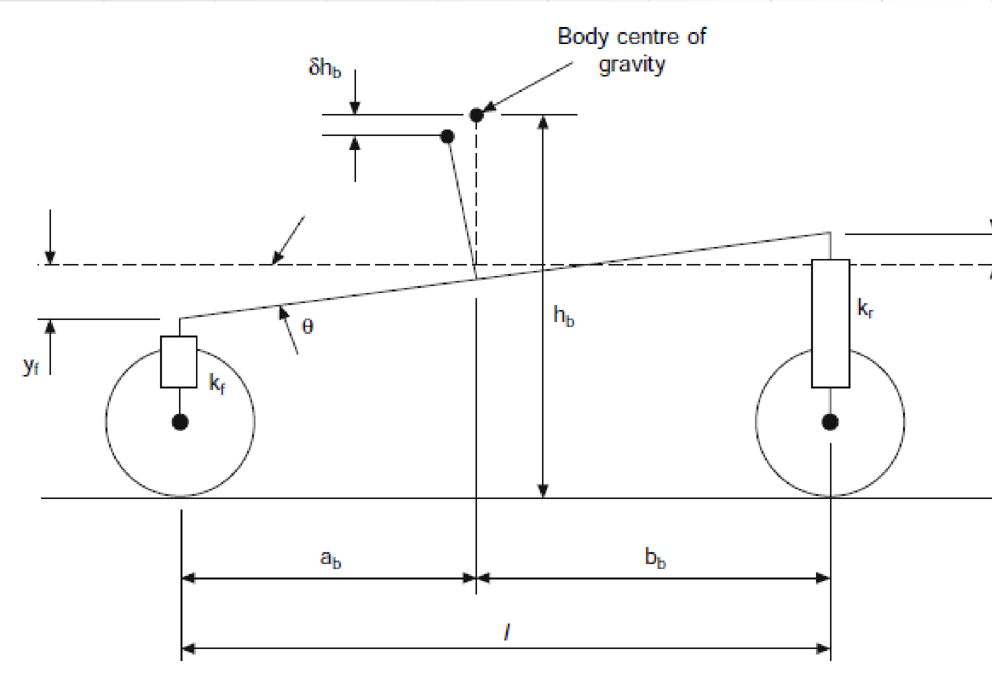
$$\frac{x_\mathrm{f}}{x_\mathrm{r}} = \frac{F_\mathrm{f} + \frac{Ph}{l}}{F_\mathrm{r} - \frac{Ph}{l}}$$

On all surfaces where the tyre–ground adhesion is less than unity, the braking will be limited by front axle lock

Pitch motion

The opposed spring forces generated during a braking event are equal to the load transfer that takes place and so are equal to

$$y_r \pm \frac{Pzh}{l}$$



Thus, on the assumption of linear springing, the compression travel at the front and rear are:

$$y_f = \frac{\frac{Pzh}{l}}{k_f} \qquad y_r = \frac{\frac{Pzh}{l}}{k_r}$$

The pitch angle, q, in degrees, adopted by the vehicle body is therefore given by:

$$\theta = \left(\frac{y_f + y_r}{l}\right) \times \frac{360}{2\pi}$$

$$\delta h_b = -y_f \frac{F_{bf}}{F_b} + y_r \frac{F_{br}}{F_b}$$

where

$$F_{bf} = F_{sf} + F_{af}$$
$$F_{br} = F_{sr} + F_{ar}$$
$$F_b = F_{bf} + F_{br}$$

Literature

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