THE LIFT BRAKE

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Abstract

The European Standard EN.81.1 under paragraph 12.4.2.1. recommends that the machine brake should consist of two jaws operating independently. These jaws must be sized in such a way that in the event of a breakage of one of them, the other must be able to retard the downward travel of the fully loaded car.
This paper analyses the behaviour of the drum brake (which is used on lift systems) in the event of a failure of one jaw under unfavourable load conditions. In addition a comparison is made with ultimate disk brake.

1. INTRODUCTION

As it is known a lift brake consists of a drum, generally made of cast iron, keyed onto the driving shaft, on the lateral surface of which two braking sectors are pushed by means of a non-return spring fixed to the free ends of the two jaws (Figure 1) .

The operating movement of the brake-shoes towards the drum is a translatory movement. In fact the relative brake-shoe movement, compared to the drum, consists of a rotating motion around a point belonging to the line joining the rotating centre of the jaw (O_1 or O_2) and the rotating centre of the drum. Each movement acting around such a point can be split into a rotating motion acting around the drum centre, which has no effect on the brake-shoe approach, and a translatory movement acting in normal direction to the line joining the two centres.

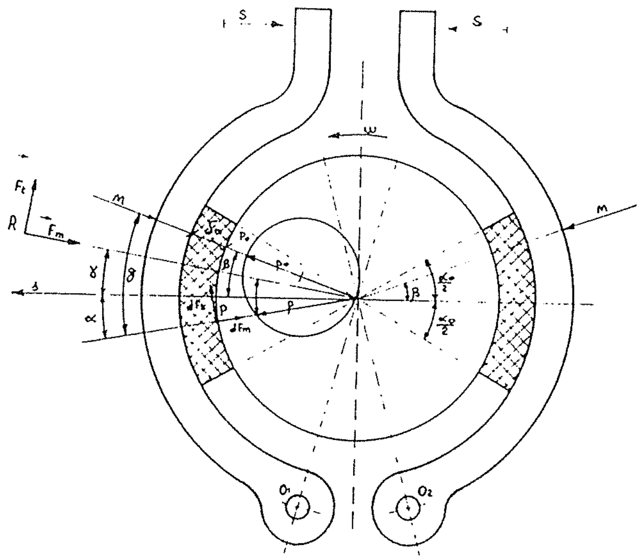

Figure 1

If such a direction is indicated by a "n", it can be noted that it does not clash with the bisecting line of the angle in the centre, which subtends the braking sector, but is inclined, in relation to it, by an angle which will be called β . It is evident that the wear of the lining will be greater in the straight line direction n .
To have an idea of how the pressure exerted by the brake-shoe onto the drum is shared, Reye's hypothesis can be used.

It can be seen that the bodies in contact will wear out in proportion to the friction work. Therefore the maximum pressure is at point P_0 , where the maximum wear δ_0 lays. On any other point P , belonging to an inclined radius of an angle θ ,as compared to n , the wear is equal to the component of the maximum wear in the OP radius direction, namely

$$\delta = \delta_o \cos\theta \qquad \text{..... (1)}$$

Therefore the pressure on point P will be

$$p = p_o \cos\theta \qquad \text{..... (2)}$$

This relationship can be represented by a circumference,which passes through the centre of the drum, tangentially to the line joining O_1 O , whose diameter is equal to p .
To calculate the braking couple it is necessary to note that the brake-shoe exerts a radial elementary thrust on the drum, at each contact point, given by

$$d F_n = p \cdot r \cdot d\theta \cdot l \qquad \text{..... (3)}$$

where l indicates the width of the brake-shoe and a tangential elementary force equal to

$$d F_t = f \cdot d F_n \qquad \text{..... (4)}$$

where f is the friction coefficient, directed in the opposite direction to the drum rotation.

2. THE CALCULATION OF THE RESULTANTS

2.1. The resultants of the elementary efforts

The resultants of the elementary efforts dFn and dFt will be applied on a point R whose position can be located by calculating the anomaly γ and the module OR in relation to a polar reference system, which has the drum centre and the straight-semi-line s as the reference radius. It is assumed that the anomaly is positive, when it is consistent with the rotating direction of the drum.
Let P be one of the contact points between the braking sector and the drum.
From Equations (2) and (3) is obtained:

$$d F_n = p_o l r \cdot \cos\theta \cdot d\theta \qquad \text{..... (5)}$$

The resultant of the dFn components in F_t , according to the Varignon thorem, must be zero. Therefore:

$$p_o l r \int_{-\frac{\alpha_o}{2}}^{+\frac{\alpha_o}{2}} \cos\theta \cdot \text{sen}\,(\alpha - \gamma) \cdot d\theta = 0 \qquad \text{..... (6)}$$

It can be observed that $\theta = \alpha - \beta$, and resolving, the following result will be obtained

$$tg\,\gamma = \frac{\alpha_o - \text{sen}\,\alpha_o}{\alpha_o + \text{sen}\,\alpha_o} \cdot tg\,\beta \qquad \ldots\ldots (7)$$

As

$$|\text{sen}\,\alpha_o| < |\alpha_o|$$

it can be seen that γ and β always have the same sign, and also that $|\gamma| < |\beta|$.

2.2. Calculating the dFn resultant
Calculating the dFn resultant in the Fn direction, the following will be obtained:

$$F_n = p_o\, l\, r \int_{-\frac{\alpha_o}{2}}^{+\frac{\alpha_o}{2}} \cos(\alpha - \beta) \cdot \cos(\alpha - \gamma) \cdot d\alpha \qquad \ldots\ldots (8)$$

thus

$$F_n = \frac{1}{2} p_o\, l\, r\, (\alpha_o - \text{sen}\,\alpha_o) \frac{\text{sen}\,\beta}{\text{sen}\,\gamma} \qquad \ldots\ldots (9)$$

from which can be derived that the resultant Ft is equal to f . Fn.
At this point the module (b) of the application point of the two resultants can be calculated. To do this the braking moment can be obtained by integrating the following expression

$$d M_f = r \cdot d F_t = f \cdot r \cdot d F_n \qquad \ldots\ldots (10)$$

from which

$$M_f = f \cdot r \cdot \int_{-\frac{\alpha_o}{2}}^{+\frac{\alpha_o}{2}} d F_n = f \cdot r^2 \cdot p_o \cdot l \cdot 2 \cdot \text{sen}\,\frac{\alpha_o}{2} \cdot \cos\beta \qquad \ldots\ldots (11)$$

But as:

$$Mt = b \,.\, Ft \qquad \ldots\ldots (12)$$

and using Equations (11), (8) and (4), the following conclusion can be drawn:

$$b = \frac{4 \cdot r \cdot \text{sen}\,\frac{\alpha_o}{2}}{\alpha_o + \text{sen}\,\alpha_o} \cdot \cos\gamma \qquad \ldots\ldots (13)$$

Let the same procedure be adopted to determine the resulting action exerted on the drum by the right hand brake-shoe, keeping in mind that the vector Ft is always orientated to a different direction from the rotating direction of the drum. It can be noted that with an equal thrust exercised by the brake closing spring on the two brake-shoes, the braking effect is different. In fact whilst the Q thrusts exerted by the spring have the same intensity and an equal lever arm as compared to the relevant fulcrums, O_1 and O_2 , the two resultants F have a different lever arm and therefore the

resultant that has a smaller lever arm will have a greater intensity in the equilibrium of the moments.
In the case represented in Figure 2 it is the right hand brake-shoe which exercises a greater thrust on the drum and therefore also its Ft tangential component will be stronger. So the right hand brake-shoe exercises a braking moment greater than the left hand brake-shoe. Naturally the role of the two brake-shoes will reverse when the rotating direction of the drum is reversed.

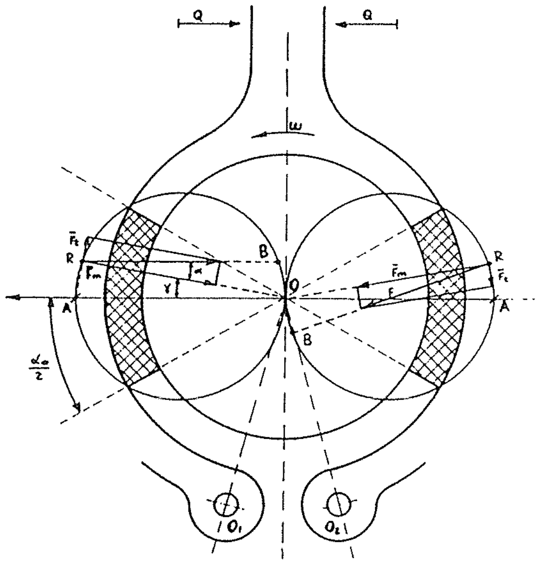

Figure 2

2.3. Braking effect

It could be wondered what technique is to be used in order that the two brake-shoes have the same braking effect. It is necessary to examine the equilibrium of the moments, around the fulcrum, of all the forces acting on each brake-shoe. The lever arms of forces Ft and Fn will be called bt and bn; the forces acting on the left hand brake-shoe will be indicated by Apex 1, and those acting on the right hand brake-shoe will be indicated by Apex 2.
With regard to the equilibrium of the left hand brake-shoe there will be

$$F_{n1} \cdot b_n + F_{t1} \cdot b_t = Q \cdot b \qquad \text{..... (14)}$$

Remembering that $Fn1 = \frac{Ft1}{f}$ the following result can be obtained:

$$F_{t1} = \frac{f \cdot b}{b_n + f\, b_t} \qquad \text{..... (15)}$$

As to the right hand brake-shoe there will be

$$F_{n2} \cdot b_n - F_{t2} \cdot b_t = Q \cdot b \qquad \text{..... (16)}$$

hence

$$F_{t2} = \frac{f \cdot b}{b_n - f\, b_t} \cdot Q \qquad \text{..... (17)}$$

which confirms that Ft2 > Ft1, and also indicates that to make Ft2 = Ft1 it is necessary that bt = O , namely the

brake-shoe fulcrums should be applied on the straight lines of the tangential components. This would require that the brake-shoe approach direction should have a very wide inclination as compared to the bisecting line of the braking sector, β angle, and the brake-shoe would wear out in an asymmetric way and would be replaced more frequently.
The movement of the fulcrums on the opposite side could be considered, i.e. on the side of the convergency of the tangential components, but in such a case the brake-shoes would become two disadvantageous levers, a higher stiffness for the spring and an oversizing for the electromagnet employed to open the brake would be required.
It may be wondered what happens when the brake shoe approach direction clashes with the bisecting line of the braking sector, namely with the straight-semi-line s . As $\gamma = \beta = 0$ and being

$$\lim_{\beta \to 0} \frac{\text{sen}\,\gamma}{\text{sen}\,\beta} = 1 \qquad \text{..... (18)}$$

it would derive

$$F_n = \frac{1}{2} p_o\, l\, r\, (\alpha_o - \text{sen}\,\alpha_o) \qquad \text{..... (19)}$$

It can be observed that for $\beta = 0$, Fn becomes minimum whilst Mf becomes maximum. The effort exerted on the drum is lower but the braking effect is higher. This depends on the fact that whilst the intensity of Ft decreases, as β decreases its lever arm however increases as compared to the drum rotating centre. In fact for $\beta = \gamma = 0$ the lever arm takes on its maximum length which is given by

$$b_o = \frac{4 \cdot \text{sen}\,\frac{\alpha_o}{2}}{\alpha_o + \text{sen}\,\alpha_o} \cdot r \qquad \text{..... (20)}$$

The function which links the lever arm b to the angle γ , in the polar reference system having the centre in point O , can be represented through a circumference whose diameter is bo (Figure 2). To determine point R , on the straight-semi-line s a point A is taken where OA = bo .
A circumference with a diameter OA is drawn. From point O a segment inclined by γ as compared to s is drawn. The intersection point of such a segment with the circumference is the point of application of the resultant of the actions exerted on the drum by the braking sector. The vector F_n lays on the RO segment and is directed in a centripetal direction. The straight-line application of the vector Ft = f . Fn , as it is perpendicular to RO , must necessarily pass through point A .

3. THE BRAKING COUPLE

If it is assumed that $\gamma = 0$ in order to make the calculation easier, the distribution of forces on the brake jaws is shown in Figure 3.

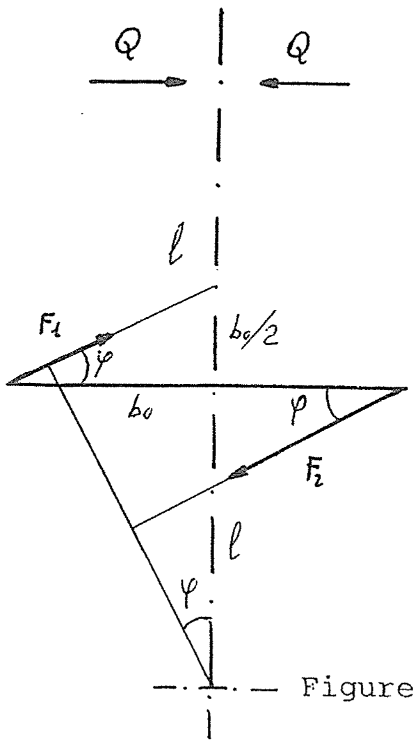

Figure 3

The braking linings currently in use, are made of thickly weaved cotton without metallic insertions. These have a friction coefficient of 0-5 , which remains virtually constant within a temperature range of 30°C to 80°C. Therefore the straight-lines for the application of the friction forces intersect the vertical at points where the distance to the origin 0 is bo/2 (Figure 1).

With regard to the equilibrium of rotation around the fulcrums the following results:

$$Q \cdot 2l = F_1\,(l + bo/2)\cos\varphi$$
$$Q \cdot 2l = F_2\,(l - bo/2)\cos\varphi \qquad \ldots\ldots (21)$$

If $\alpha_0 = \pi/2$ and $l = 1.3 \cdot r$ are assumed (where r is the radius of the braking drum), from Equations (20) and (21), then $bo = 1.1r$ and $F_2 \simeq 2.5\,F_1$.
This means that the braking effort F_2 represents about 70% of the total. And as the total braking moment is given by

$$Mf = (F_1 + F_2) \cdot \frac{bo}{2} \cos\varphi \qquad \ldots\ldots (22)$$

in the event that a failure occurs to the right hand jaw, the remaining braking effect, resulting from the operation of the left hand jaw, decreases to 30% of the total.

The moment that the drive motor is de-energized, the whole system globally possesses a certain kinetic energy,which will be indicated by Ec. The brake has the task to absorb this energy by friction. Keep in mind the mechanical performance of all the system in motion; of the friction resistances of the reduction couple; of the thrust bearing and the bushings; of the ventilation effect of the rotating parts; and the unit resistance, which influence the components in the translatory movement.
Finally the positional energy change which occurs to the masses moving in a vertical translatory movement, must be taken into account.
Let J_1 be the inertia moment of the gear high speed shaft and of all associated components (motor, fan, shaft etc.), and J_2 the inertia moment of the gear low speed shaft and of all associated components (rim, friction pads, etc.), with ω_1 and ω_2 being the relevant angular speeds.

Let Ca be the car weight, Cp the counterweight weight and Q the car load. Indicating with v the car translatory speed, the kinetic energy of the entire system in motion is expressed by:

$$Ec = \frac{1}{2}\left[J_1 \omega_1^2 + J_2 \omega_2^2 + \frac{Ca + Cp}{g} v^2 + \frac{Q}{g} v^2 \right] \quad \ldots\ldots (23)$$

Indicating with $\tau = \omega_2 / \omega_1$ the reduction ratio and with r the radius of the friction pulley, as $\omega_2 = v/r$ and $\omega_1 = v/\tau \cdot r$ then:

$$Ec = \frac{1}{2}\left[\frac{J_1}{\tau^2 r^2} + \frac{J_2}{r^2} + \frac{Ca + Cp}{g} + \frac{Q}{g} \right] v^2 \quad \ldots\ldots (24)$$

If $x = \frac{J_1}{\tau^2 r^2} + \frac{J_2}{r^2}$ $\quad Ca \simeq Q \; , \; Cp \simeq Ca + \frac{1}{2} Q$

then:

$$Ec = \frac{1}{2} \left[x + \frac{3.5}{g} Q \right] v^2 \quad \ldots\ldots (25)$$

where x has the units of a mass.

If the braking space is indicated by h , the change in the positional energy which occurs in the system during braking is given by:

$$Ep = (Ca + Q - Cp) \cdot h \simeq \frac{1}{2} Q\, h \quad \ldots\ldots (26)$$

Indicating by the symbol α the rotating angle of the drum during the braking phase, the result will be:

$$\overline{Mf \cdot \alpha = Ec + Ep} = \frac{1}{2}\,\eta \left[x + \frac{3.5}{g} Q \right] \cdot v^2 + \frac{1}{2} Q \cdot h \quad \ldots\ldots (27)$$

where η is the mechanical performance.
If the value of x is considered neglegible in relation to all the masses in movement, and if it is noted that $h = \alpha \cdot R$, where R is the radius of the braking drum, deriving Equation (27) against time, the result is:

$$Mf \cdot \omega = \eta \cdot \frac{3.5}{g} Q \cdot v \cdot a + \frac{1}{2} Q\, \omega\, R \quad \ldots\ldots (28)$$

Where $\omega = \frac{d\alpha}{dt}$ is the angular speed, $v = \omega R$ is the linear speed, a is the deceleration (which is normally around 1 m/sec^2, namely about 1/10 of g) and R is the radius of the friction pulley.
If $\eta = 0.7$ is assumed, the following result will be obtained:

$$Mf \simeq 0.25 \cdot \ell - R + 0.5 \cdot \ell \cdot R \quad \ldots\ldots (29)$$

This means that the posistional energy to be absorbed affects the total by about 67% . But as stated above in the case of a failure occuring to the right hand jaw the braking effect decreases to about 30% on the total. This means that the left hand jaw absorbs only part of the positional energy lost by the system and the remaining part becomes kinetic energy.

Therefore not only does the car not slow down during the braking phase, but it continues its downward movement with an increasing speed.

4. THE DISK BRAKE

Suppose a disk brake is used whose braking sectors have the shape of a circular rim segment whose opening angle is indicated with α_0 ,and the internal and external radius with ri and re respectively (Figure 4).

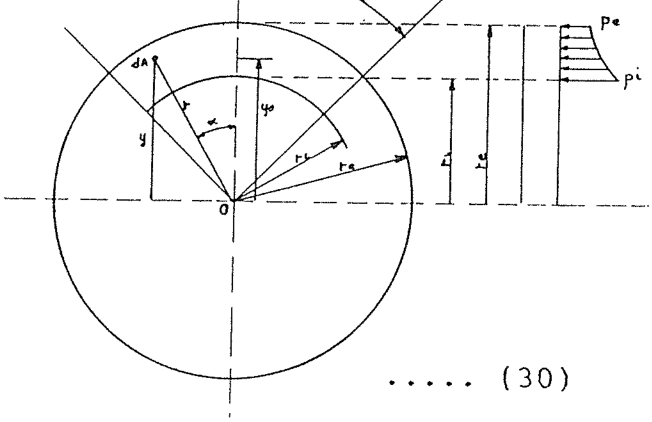

Figure 4

Reye's hypothesis applied to the brake can be expressed by the following: "The wear of the lining is directly proportional to the frictional work done", indicating with ds an elementary surface of lining, with δ the thickness consumed over the unit of time, the volume of lining worn out in time t will be $\delta \cdot ds \cdot t$. The frictional work absorbed in the same time interval will be

$$L = f \cdot p \cdot ds \cdot v \cdot t \qquad \text{..... (30)}$$

where v is the peripheral speed of the contact point between the disk and the lining.

By virtue of the Reye hypothesis there will be:

$$\delta \cdot ds \cdot t = K \cdot f \cdot p \cdot ds \cdot v \cdot t \qquad \text{..... (31)}$$

where K is the coefficient of proportion. From the previous equation is obtained $\frac{\delta}{f \cdot K} = p.v.$ But $v = \omega \cdot r$, moreover δ and K are constants and therefore

$$p \cdot r = \cos t \qquad \text{..... (32)}$$

The pressure of the lining on the disk grows under a hyperbolic law as r decreases. The resulting action exerted by the lining on the disk is given by

$$N = \int_A \cdot p \cdot dA \qquad \text{..... (33)}$$

where

$$dA = r \cdot d\alpha \cdot dr \qquad \text{..... (34)}$$

and therefore

$$N = \int_{r_i}^{r_e} \int_{-\frac{\alpha_0}{2}}^{+\frac{\alpha_0}{2}} p \cdot r \cdot dr \cdot d\alpha \qquad \text{..... (35)}$$

thus

$$N = \alpha_o \cdot p_i \cdot r_i (r_e - r_i) \qquad \text{..... (36)}$$

The lining behaves as though it exerted a thrust of intensity p_i ,evenly divided on a rectangular surface of base $\alpha_0 \cdot ri$ and height h = re - ri. The N application point, because of the symmetry of the system, must be on the besecting line of the braking sector, at a distance yo from the centre which can be obtained applying the Varignon theorem.

$$N \cdot y_o = \int_A p \cdot dA \cdot y \qquad \ldots\ldots (37)$$

$$N \cdot y_o = p_i \cdot r_i (r_e^2 - r_i^2) \cdot \mathrm{sen}\frac{\alpha_o}{2} \qquad \ldots\ldots (38)$$

and recalling the N expression the following conclusion is drawn

$$y_o = (r_e + r_i)\frac{\mathrm{sen}\frac{\alpha_o}{2}}{\alpha_o} = r_m \frac{\mathrm{sen}\frac{\alpha_o}{2}}{\frac{\alpha_o}{2}} \qquad \ldots\ldots (39)$$

where rm is the medium radius. In particular for α_o tending to zero, y_o tends to the medium radius. For α_o tending to 2π, y_o tends to zero, and N becomes coaxial with the disk. In any case the application point of N clashes with the barycentre of the circumference arch having a radius r_m. To calculate the braking moment, note that the specific tangential action exerted by the lining at any point is given by $\tau = f \cdot p$ and therefore $Mf = \int_A \tau \cdot dA \cdot r$ from which the following is derived:

$$M_f = f \cdot p_i \cdot r_i \cdot \frac{\alpha_o}{2}(r_e^2 - r_i^2) = f \cdot N \cdot r_m \qquad \ldots\ldots (40)$$

The result is as though a tangential force of intensity $T = f \cdot N$ were applied to a point rm distant from the drum centre.

5. CONCLUSIONS

The brake lining of the disk brake wears out evenly.
For shoe brakes the peak wear occurs in the approach direction and it decreases in the opposite direction.
With equal values of α_o and brake lining weights, the replacement rate for shoe brakes is higher than that related to disk brakes. In this respect it is convenient to decrease the amplitude of α_o for shoe brakes, increasing their width l, but keeping in mind that in such a case overall dimension problems may arise. With regard to disk brakes it would be more convenient to increase the amplitude of α_o. Moreover, in the event of a jaw failure the efficiency of the brake-shoe brake decreases by about 70%, whilst the disk brake retains an efficacy amounting at least to 50%. On the other hand it is not impossible to develop a mechanical system, so that in the event of a failure of one jaw, it could increase the pressure on the working jaw.
As regards the maximum pressure exerted between the two members of the braking couple, manufacturers recommend keeping the values between 8 and 30 N/cm^{2}.
Consider that each braking action removes all the energy from the moving masses and that such energy is dissipated

in the form of heat. Then it can be understood why the brake linings are easily subjected to the risk of overheating, with the resulting reduction of the friction coefficient and therefore of their braking effect. This phenomenon is known in mechanics as " fading" .
To avoid risk of overheating some researchers recommend using the following empirical formula $p \cdot v < 250$ where p is the brake lining pressure expressed in N/cm^2 and v is the relative speed of either the drum or the disk compared to the brake lining (expressed in m/sec). Consider then that the drum brake operates on the surface, where there is the peak peripheral speed, which contains the peak pressure lower values. Moreover the pressure is not evenly spread and therefore the brake is forced to operate below its best capabilities. As regards disk brakes the relative speed is,because of the different contact points, always lower than the disk peripheral speed. Moreover, as seen above, it results $p \cdot r = \cos t$ and therefore $p \cdot \omega \cdot r = \cos t$ for each contact point between the braking sector and the disk. That means that the disk brake operates evenly on all contact surfaces.

BIOGRAPHICAL DETAILS

Italian Eng. Carlo Distaso was born near Foggia . He completed his grammar school education in Urbino, attended high school in Rimini,and graduated in mechanical engineering in Rome.
He started his career as a teacher of mechanical technologies at a professional institute where in his last years he took up also the post of chief of the technical office and deputy headmaster.
Eng. Distaso participated successfully in a competition launched by ENPI (the former Italian national body for safety at work) which resulted in a post in its Milan offices.
It was in those years that he became involved in the lift technology.
He attended many international conferences as a speaker, many of the ISO Technical Commission No. 178 meetings, and is a member of the UNI (Italian Unification Body) sub-committee which is involved in lifts.
Eng. Distaso has been the Editor-in-Chief of the Elevatori magazine for about fifteen years. Finally he founded and manages in Italy the Technical Control Institute, ICT, which handles consultancy and tests on lifts and vertical transportation systems.