Roland Stawinoga

The efficiency of elevator hoistway equipment

Summary

Drive motors and their associated frequency converters can be more accurately sized if they are chosen, by calculation, to be closer to the effective actual load values for the system. The very imprecise assumptions made in practice are often compensated for by choosing motors and drive control systems with large diversity factors.

In keeping with current technological developments, the individual losses in the hoistway equipment (pulleys, bearings, guide shoes, ropes, etc) should be determined more accurately for various payloads (other than the rated load and for all operating conditions (starting reaching rated speed, braking, stopping, etc. - in both directions).

General remarks

In order to determine the individual losses the following basic rule applies in all cases: the frictional losses are dependent upon the *force* (F) and the *friction* (μ) acting in each case.

$$F_R = f(F, \mu) \qquad (1)$$

For the forces in the system a distinction is made between

- static forces (for an elevator at rest and moving at a steady speed)
- dynamic forces (when starting up or braking)

In the case of the friction, a distinction has to be made between

- friction when moving at steady speed
- friction at the beginning of the motion (starting up)

When dimensioning the motor and control units, it is necessary to consider the least favourable case. This generally means starting with a fully loaden cabin from the lowest stop and travelling upwards.

Precise determination of all the forces acting in the hoistway (including dynamic forces and losses) can, however, also be carried out to calculate the traction according to EN 81-1 : 1998 [1] (see Section 9.3 and Annex M). In this case, however, other load cases have to be taken into account such as the cabin with 125% of the rated load at the lowest stop or also the empty cabin at the uppermost stop.

The losses in the hoistway are generally caused by the following components:

1. Cabin guides (sliding or roller guides)
2. Rope pulleys (with sliding or roller bearings)
3. Others (forces required to bend ropes, losses due to airstream)

No realistic methods of calculation have hitherto been available for the forces required to bend the ropes, which means that they cannot be taken into account in this instance. Yet, as elevator safety regulations mean that the diameters of the ropes are very small in relation to those of the rope pulleys, the bending forces required are also very low. They can be further reduced by enlarging the ratio of rope pulley diameter to rope diameter above and beyond the values demanded in the EN

Losses due to wind resistance in the hoistway can only be expected at high velocities. The magnitude of the losses is then also affected by the ratio of the area of the hoistway to that of the car.

Mathematical approach to determining the losses

The following theoretical considerations form the basis for determining the losses due to guide rails and rope pulleys:

Sliding guides - sliding bearings

The loss due to friction of a car guide rail is calculated according to the formula

$$FR_G = F_G \times \mu_G \tag{2}$$

The force F_G is the force with which the guide shoe is pressed onto the rail if the loads are not centred within the cabin. Provided no other more precise values are available, the loads are assumed to comply with Annex G of EN 81-1 : 1998. The forces on the guide shoes can be reduced by suspending the car from its centre of gravity and by having the greatest possible vertical distances for the guide shoes. The values for the friction, μ_G, depend upon the particular material used for the guide shoes (see Table 1 for a selection).

Greased bronze bushes have generally been used for the sliding bearings of the rope pulleys. Due to the low rotational speeds commonly employed in elevator construction, no complete film of grease is formed so that the shaft and bush partially come into contact, in which case one speaks of mixed friction. To ensure that the lubrication is as even as possible, the slide

	Material	Coefficient of friction greased μ	Coefficient of friction dry μ_A	Remarks
Group 1	Vulkollan	0,20	0,50	extremely resistant to abrasion and good damping properties
	Vulkollan with additives	0,15	0,30	improved sliding characteristics
	ACLASYN	0,05	0,05	as per manufacturer's figures (not verified)
Group 2	Polyamide (PA 6)	0,12	0,30	high mechanical strength
	Polyamide (PA 6-G/oil) with additives	0,10	0,18	improved sliding and wear characteristics
	Polyethylene "S" green (PE-UHM)	0,08	0,12	lower strength than polyamide
	Polyethylene "S" 8000 (PE-UHM)	0,05	0,09	improved sliding characteristics
The coefficients of friction for Group 1 and Group 2 were determined under different conditions and therefore cannot be compared directly with one another.				
Coefficients of friction for guide shoes on steel				

Table 1

bearings should be lubricated with a forced feed grease lubrication system. Under these conditions the coefficient of friction lies between approximately $\mu = 0.08$ to 0.10 [2]. As the metallic contact first has to be overcome at the start of every rotational movement, an initial coefficient of friction of approximately $\mu_A = 0.19$ has to be included in the calculation (coefficient of adhesion for bronze on steel, dry). The frictional loss occurs on the outer diameter, d_A, of the axle. The rope force generated by the friction on the circumference of the rope pulley with diameter D_P is lower in relation to the diameter:

$$FR_{P_G} = F_P \times \mu_{P_G} \times \frac{d_A}{D_P} \tag{3}$$

The axle load F_P is given by the masses in the hoistway, that are suspended upon the axle and from the arrangement of the rope pulleys (see also Figure 2).

Roller bearings [3]

The total rolling resistance of an axially loaded roller bearing consists of the rolling and sliding friction in the roller contacts, in the contact surfaces between the roller bodies and the cage, and in the guides surfaces for the roller bodies or the cage. Friction due to the lubricant and the sliding friction caused by chafing seals in sealed bearings also play a role. The total moment of friction [Nmm] is given by the formula

$$M_{FR_G} = M_0 + M_{1(A)} + M_2 + M_3 \tag{4}$$

The load-independent moment, M_0, depends on the lubrication and speed of the rollers. It can be calculated as

$$M_0 = 10^{-7} f_0 \; (\nu n)^{2/3} \; (\frac{d+D}{2})^3 \tag{5}$$

Where:	d =	the bore diameter of the bearing	[mm]
	D =	the outer diameter of the bearing	[mm]
	f_0 =	a factor dependent upon the type of bearing and lubrication	
	n =	the rotational speed of the bearing (see Figure 2)	[min^{-1}]
	ν =	the kinematic viscosity of the basic oil	[mm^2/s]

The load-dependent moment of friction, M_1, is determined by the elastic deformation and partial sliding in the contact surfaces. It can be calculated from

$$M_1 = f_1 \; P_1^{\,a} \; (\frac{d+D}{2})^b \tag{6}$$

Where:	f_1 =	a factor dependent upon the type of bearing and load on the bearing	
	P_1 =	the load critical for the moment of friction	[N]
	a, b =	exponents dependent upon the bearing	

The load-dependent moment of friction from axial loads, M_2, is no taken into account here. However, if bearings are used in which the seals rub, the moment of friction thus generated, M_3, also has to be taken into account

$$M_3 = (\frac{d+D}{f_3})^2 + f_4 \tag{7}$$

Where: f_3, f_4= factors dependent upon the type of bearing

The initial moment of friction, M_{1A}, is approximately twice as high as the load-dependent moment of friction, M_1.

Using these loss moments, it is possible to determine the frictional loss of the bearing at the circumference of the rope pulley as

$$FR_{P_W} = M_{FR_G} \frac{2}{D_P} \tag{8}$$

As the losses are significantly less for rope pulleys with roller bearings, rope pulleys with sliding bearings are being used less and less frequently. In any case, however, the loss for the bending of the rope has to be included in the calculation (see above).

The roller bearing losses of the roller guides are determined in the same manner as for the rope pulleys, with an extra allowance having to be made for the losses due to flexing of the rubber tyre.

Determination of the static and dynamic forces in the system

The forces determined are those that act on both sides of the traction sheave as rope forces (T_1 and T_2). Here the general case from *EN 81-1 : 1998 [1] Annex M - Example M.3* is used as the basis. The attached Figure 1 has been adapted from Figure M.4 in the standard, as have the terms for the forces occurring in the system.

In the formulae the following terms are used, based on Example M.3:

Heights in the hoistway

H	travel height	[m]
y	vertical distance of the car $y = H$ at the lowest stop $y = 0$ at the highest stop	[m]
y_{car}	vertical distance from centre of rope pulleys for car	
y_{ctw}	vertical distance from centre of rope pulleys for counterweight to centre of traction sheave/rope pulley in the machine room/top of hoistway	
y_M	vertical distance from centre of rope pulley in machine room/top of hoistway to centre of traction sheave when positioned below or to the side next to the hoistway	

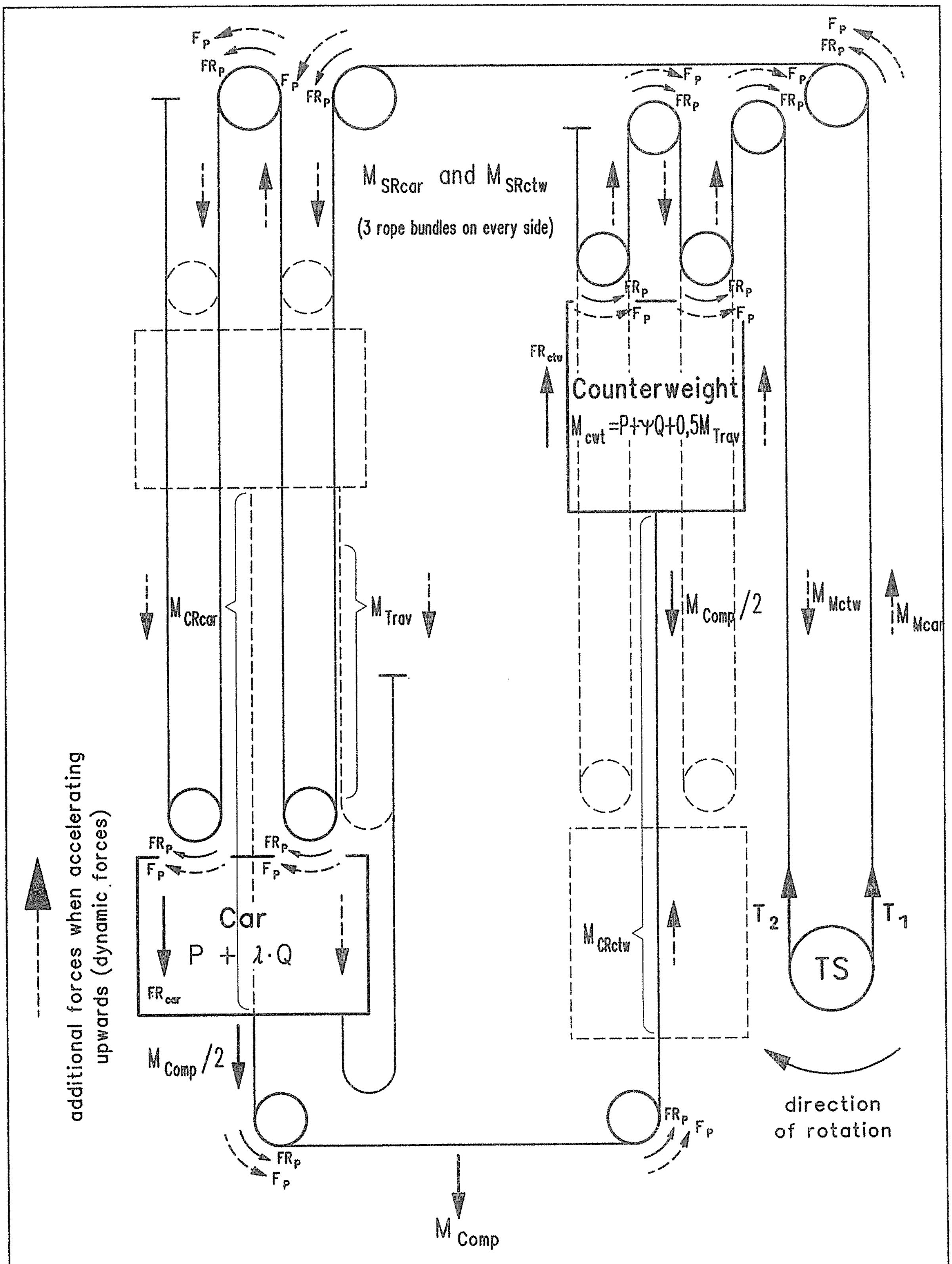
Velocity, retardation, rotational speed of rope pulleys

r	reeving factor	
D_P	diameter of the rope pulleys	[m]
v	car velocity	
v_S	rope velocity (see Figure 2) $n_P = 60 \times v_S / (D_P \times \pi)$	[m/s]
g_n	standard acceleration of free fall	[m/s²]
a	acceleration or retardation of the car	[m/s²]

Masses and moments of inertia

independent of the car position

Q	rated load	[kg]
λ	factor for the proportion of the rated load $\lambda = 0$ for the empty car $\lambda = 1$ for 100% of the rated load	



Weights, losses and dynamic forces in the hoistway

General case: r = 4 , car in lower position, accelerating upwards

Figure 1

P	mass of car	[kg]
ψ	counterweight compensation factor, in general $\psi = 0.5$	
M_{cwt}	mass of counterweight $M_{cwt} = P + \psi Q + 0.5M_{Trav}$	[kg]
M_{Comp}	weight of tension device	[kg]
n_{SR}	number of suspension ropes	
m_{SR}	actual weight of a suspension rope per 1 m length	[kg/m]
M_M	weight of the suspension rope from the upper pulley to machine located underneath at the side $M_M = n_{SR} \times m_{SR} \times y_M$	[kg]
J_{Pcar}	moment of inertia of the rope pulleys on the car side	[kgm²]
J_{Pctw}	moment of inertia of the rope pulleys on the counterweight side	[kgm²]
J_{PTD}	moment of inertia of the rope pulleys on the tension device	[kgm²]
i	number of rope pulleys on the tension device	

dependent upon the car position

M_{SRcar}	mass of one bundle of suspension ropes above the car $M_{SRcar} = m_{SR} \times (y_{car} + y)$	[kg]
M_{SRctw}	mass of one bundle of suspension ropes above the counterweight $M_{SRctw} = m_{SR} \times (y_{ctw} + y)$	[kg]
n_{Trav}	number of travelling cables	
m_{Trav}	mass of a travelling cable per 1 m of length	[kg/m]
M_{Trav}	total mass of the travelling cables $M_{Trav} = n_{Trav} \times m_{Trav} \times 0.5y$	[kg]
n_{CR}	number of the rope weight compensation elements (belt, chain, rope)	
m_{CR}	mass of a rope weight compensation element per 1 m of length	[kg/m]
M_{CRcar}	total mass of the rope weight compensating gear below the car $M_{CRcar} = n_{CR} \times m_{CR} \times y$	[kg]
M_{CRctw}	total mass of rope weight compensating gear below the counterweight $M_{CRctw} = n_{CR} \times m_{CR} \times (H - y)$	[kg]

Forces

The forces are given by multiplying the masses with the accelerations, e.g.

$$F_{Comp/2} = M_{Comp/2} \times g_n \qquad [N]$$

The force required on the circumference of a rope pulley to accelerate the rope pulley to its rotational speed when the car is travelling at its rated velocity may be calculated as

$$F_P = J_P \times (2/D_P)^2 \times r \times a \qquad [N]$$

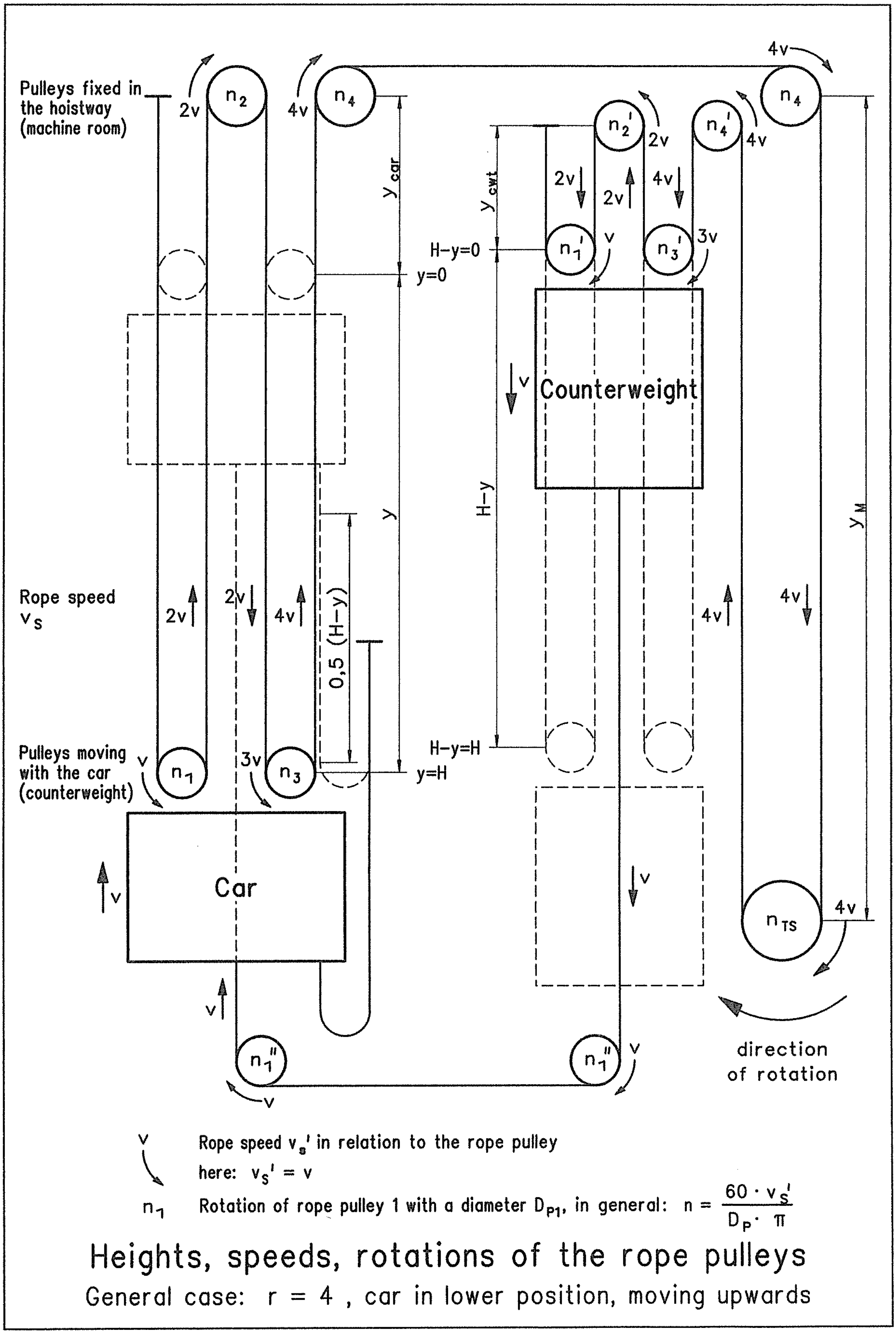

Pulleys fixed in the hoistway (machine room)
Rope speed v_S
Pulleys moving with the car (counterweight)
Car
Counterweight
direction of rotation
Rope speed v_s' in relation to the rope pulley
here: $v_S' = v$
n_1 Rotation of rope pulley 1 with a diameter D_{P1}, in general: $n = \frac{60 \cdot v_S'}{D_P \cdot \pi}$
Heights, speeds, rotations of the rope pulleys
General case: r = 4 , car in lower position, moving upwards

Figure 2

Rope forces from			car side: static	car side: moving steadily	car side: with initial acceleration	counterweight side: static	counterweight side: moving steadily	counterweight side: with initial acceleration
Translational masses	Independent of the hoisting height	rated load car counterweight	$F_{car}/r = (\lambda\cdot Q+P)g_n/r$	1) $(F_{car}+FR_{car})/r$	1) $(F_{car}+FR_{car})\frac{(+a)}{r}$	$F_{ctw}/r = (\gamma Q+P+\frac{M_{Trav}}{2})g_n/r$	1) $(F_{ctw}-FR_{ctw})/r$	1) $(F_{ctw}-FR_{ctw})\frac{(-a)}{r}$
		rope tension device	$F_{Comp} / 2\cdot r = M_{Comp}\cdot g_n / 2\cdot r$			$F_{Comp} / 2\cdot r = M_{Comp}\cdot g_n / 2\cdot r$		
		ropes to machine	$F_{M\,car} = (-M_{M\,car})\cdot g_n$		$F_{M\,car}\cdot(-a)$	$F_{M\,ctw} = (+M_{M\,ctw})\cdot g_n$		$F_{M\,ctw}\cdot(+a)$
	Dependent upon the hoisting height	travelling cables	$F_{Trav} = M_{Trav}\cdot g_n / r$		$F_{Trav}\cdot(+a)$	——	——	——
		suspension ropes	$F_{SR\,car} = M_{SR\,car}\cdot g_n$		$F_{SR\,car}\cdot(+ra)$	$F_{SR\,ctw} = M_{SR\,ctw}\cdot g_n$		$F_{SR\,ctw}\cdot(-ra)$
		rope weight compensation gear	$F_{CR\,car} = M_{CR\,car}\cdot g_n / r$		$F_{CR\,car}\cdot(+a)$	$F_{CR\,ctw} = M_{CR\,ctw}\cdot g_n / r$		$F_{CR\,cwt}\cdot(-a)$
Rotational masses		rope pulleys on car and ctw side	——	2) $+(\Sigma FR_{P\,car})$	3) $+(\Sigma F_{P\,car})$	——	2) $-(\Sigma FR_{P\,ctw})$	3) $-(\Sigma F_{P\,ctw})$
		rope pulleys of tension device	——	2) $+FR_{P\,TD}\cdot i/2\cdot r$	3) $+F_{P\,TD}\cdot i/2\cdot r$	——	2) $-FR_{P\,TD}\cdot i/2\cdot r$	3) $-F_{P\,TD}\cdot i/2\cdot r$
Total			$T_{1\,stat}$	$T_{1\,FR}$	$T_{1\,dyn}$	$T_{2\,stat}$	$T_{2\,FR}$	$T_{2\,dyn}$

Acceleration factors: in upwards direction: $(+a) = (g_n + a) / g_n$

$(+ra) = (g_n + r\cdot a) / g_n$

in downwards direction: $(-a) = (g_n - a) / g_n$

$(-ra) = (g_n - r\cdot a) / g_n$

1) FR_{car} and FR_{ctw} depend upon the type of guide (sliding or roller guides) and the force on the guides

2) FR_P depends upon the type of bearing (sliding or roller bearing), diameter of pulleys and load on the respective pulley axle

3) F_P depends upon the moment of inertia, the diameter and the rotational speed of the rope pulley concerned

Formulae for determining the rope forces on the traction sheave

Example: Car with payload travelling upwards

Table 2

Table 2 lists all the formulae for determining the rope forces, etc., separated into car and counterweight side. A distinction is made between the static forces and the forces during steady-speed motion and during accelerated motion. If the values for the pull-out friction are used in the accelerated motion, then the pull-out forces required at the beginning of a movement are obtained. The sum of the forces per side results in the rope forces T_1 and T_2, which act directly on the traction sheave, with the larger of the two forces always being designated as T_1, regardless of whether it acts on the car or counterweight side.

The number of rope pulleys changes with the reeving factor. Furthermore, the rope weight compensating gear or tension device should only be inserted when they are actually present.

For the basic formulae it is assumed that the car has to be accelerated from the lowest stop in an upwards direction. The masses of the ropes, the travelling cable and the rope compensating gear change according to the position of the cabin and counterweight within the hoistway. The effective load of the car is determined by the factor λ, changes in the counterweight are catered for by the counterweight compensation factor, ψ. The signs on the acceleration and frictional forces change according to whether the car and counterweight move downwards or upwards, or whether the elevator is starting up or braking.

Determining the efficiency of hoistway equipment

By *efficiency of hoistway equipment* we mean the ratio between the net static and the dynamic rope forces acting on the traction sheave when the car is travelling at a steady speed:

$$\eta_{hoistway} = \frac{T_{1_{stat}} - T_{2_{stat}}}{T_{1_{FR}} - T_{2_{FR}}} \tag{9}$$

Different results are obtained depending on whether the car is moving at a steady speed or is being accelerated, whether it is fully or partially loaded, or whether it is moving upwards or downwards.

Concluding Remarks

The above consideration shows that either the conditions for determining the efficiency of the hoistway equipment must be clearly defined in order to obtain comparable values, or that, as suggested here, the hoistway efficiency should be abandoned and that motor and frequency inverter should be dimensioned by determining the individual losses within the hoistway.

References

[1] EN 81-1 : 1988

[2] Ernst: Die Hebezeuge, Braunschweig 1973

[3] SKF - Katalog 4000/IV T, 1994

About the author

Dipl.-Ing. Roland Stawinoga has been involved in the design of elevators (and escalators) for more than 42 years. He is a member of the IAEE and well-known for his technically authoritative lectures (ELEVCON'93 in Vienna, LifTech'94 in Brussels, ELEVCON'95 in Hong Kong, London Lift Seminar '95, ELEVCON'96 in Barcelona, ELEVCON'97 in Shanghai), which have also been published in the most important elevator trade journals (Elevator World, Elevatori, Lift-Report), as well as for his other contributions to elevator engineering in Lift-Report.

In 1973 he took charge of the Technical Department in the firm of G. A. Koch in Hamburg, where he first became acquainted with V-belt drive systems and played a decisive role in their further development. 1990 saw the founding of his own engineering office, where fully equipped with the latest CAD technology, he employs his extensive and expert knowledge in assisting elevator companies to construct high-quality installations. His special interest continues to focus on the design and development of elevator drive systems in general and on the V-belt drive in particular.