ELEVATOR

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Project	: ELECTROMECHANICAL ELEVATOR
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Location	:
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Date	: MARCH 2007
Project Manager	: RICHARD WHITE
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Notes	:
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1. INTRODUCTION

This study is based upon EN 81.2. Furthermore, the following literature was also used:

a) Greek Standard ELOT EN 81.2.

b) Elevators Study-Calculations, F. Dimopoulos, Athens 1990.

c) Technical Manuals of KLEEMANN.

2. ASSUMPTIONS & Rules of Calculation

a) General Data of Elevator

Car surface area (F): For Passengers lift it is calculated according to Table 1.2 EN 81-2, if the user has not entered another value.

Lift nominal weight (Q): Depending on the lift kind and if the user has not entered another value, it is calculated as follows:

a) Passengers lifts:

i) Number of passengers < 20: Q = (75 x Number of passengers) (Kp)

ii) Number of passengers ≥ 20: Q = (500 x Car Surface Area) (Kp)

b) Hospitals lifts: Q = (200 x Car Surface Area) (Kp)

c) Vehicles lifts: Q = (200 x Car Surface Area) (Kp)

d) Loads lifts: Q = (300 x Car Surface Area) (Kp)

Car weight: If the user has not entered another value, it is being calculated as follows:

a) Persons lifts: P = 100 + (50 x Passengers Number) (Kp)

b) Other lift kinds:

i) Q ≤ 500 Kp: P = 100 x (3 + Car surface area) (Kp)

ii) Q > 500 Kp: P = 100 x (3 + (1.5 x Car surface area)) (Kp)

b) Guide Rails

1. General

1.1 In order to meet the requirements of 10.1.1, the calculations of the guide rails, as shown below, are valid only if no special load distribution is specified.

1.1.1 The rated load - **Q** - is assumed not to be homogeneous distributed around the car area, see section 2.2.

1.1.2 It is assumed that safety devices operate coincidently on the guide rails and that the braking force is equally distributed.

2. Loads and forces

2.1 The acting point of the empty car mass and the components supported by the car, as the piston, the flexible cable section and the wire ropes / balancing chains (if they exist) - **P** - must be the center of gravity of the car mass.

2.2 According to section 8.2, in cases of “normal use” and “safety device operation”, the rated load - Q - must be equally distributed in the 3/4 of the car area, which is located in the most malign position, as described in paragraph 7 (Examples of calculation methods).

However, if different load distribution conditions are provided (0.2.5) after negotiations then the calculations should be based upon these conditions.

2.3 Buckling force - F_k - of the car must be calculated using the equation:

$$k_1 * g_n \cdot * (P+Q)$$

F_k = -------------------------
n

where:

$\boldsymbol{k_1}$ = collision coefficient according to **Table Z.2**,

$\boldsymbol{g_n}$ = constant acceleration of gravity (9,81 m/s^2),

P = empty car mass and the components supported by the car, e.g. the piston, the flexible cable section and the wire ropes / balancing chains (if they exist) in kg,

Q = *rated load, in kg,*

n = number of guide rails.

2.4 The counterweight buckling force or the balancing weight of the safety gear - $\boldsymbol{F_c}$ - should be calculated using the equation:

$$F_k = \frac{k_1 * g_n * (P+q * Q)}{n} \quad or \quad F_c = \frac{k_1 * g_n * P * q}{n}$$

where:

$\boldsymbol{k_1}$ = collision coefficient according to **Table Z.2**,

$\boldsymbol{g_n}$ = constant acceleration of gravity (9,81 m/s^2),

P = empty car mass and the components supported by the car, e.g. the piston, the flexible cable section and the wire ropes / balancing chains (if they exist) in kg,

Q = *rated load, in kg*,

q = balancing factor, indicating the balancing amount of the rated load by the counterweight or the car mass balancing amount by the balancing weight,

n = number of guide rails.

2.5 While the car is loaded and unloaded, it must be assumed that a force - F_s - is acting on the center of the threshold of the car door. The force acting on the threshold must be:

$F_s = 0,4 * g_n * Q$ for lifts with rated load less than 2500 Kg in houses, office buildings, hotels, hospitals, etc,

$F_s = 0,6 * g_n * Q$ for lifts with rated load equal or more than 2500 Kg,

$F_s = 0,85 * g_n * Q$ for lifts with rated load equal or more than 2500 Kg in the case of forklift vehicle loading

When the force acts on the threshold, we must assume that the car is empty. The force must only act on the most malign door in cars which have more than one door.

2.6 The guidance forces of a counterweight or a balancing weight – G – must be calculated considering:

- the acting point of the mass
- the suspension and
- the forces due to wire ropes / balancing chains (if they exist), tensile or not.

In a counterweight or a balancing weight, which is hung and has a central guidance, it must be taken into account an eccentricity between the mass acting point and the horizontal cross section center of gravity of the counterweight or the balancing weight, equal to at least 5% of the width and 10% of the depth.

2.7 It must be taken into consideration the forces per guide rail, owing to auxiliary equipment adjusted to the guide rail - **M** - except for the overspeed governors and their relative components, switches or calibration equipment.

2.8 The wind loads - WL - of the lifts operating at the external of the buildings with an incomplete cell should be considered and calculated after a discussion with the designer of the building (0.2.5).

3. Loading conditions

3.1 The forces and loads and the loading conditions that must be taken into account, are shown on **Table Z.1**.

Table Z.1: Forces and loads that must be taken into account on different loading conditions.

Loading conditions	Forces and loads	P	Q	G	F_s	F_k or F_c	M	WL
Normal use	Running	+	+	+	-	-	+	+
	Loading + unloading	+	-	-	+	-	+	+
Safety device operation	Safety devices or similar devices	+	+	+	-	+	+	-
	Breaking valve	+	+	-	-	-	+	-

3.2 In documents destined for the first test and inspection, it is adequate to contain just the calculation of the most malign loading condition.

4. Collision coefficients

4.1 Safety device operation

Collision coefficient, $\mathbf{k_1}$, owing to the operation of the safety device, depends on the type of the safety device.

4.2 Car

In the loading condition «Normal use, Running», the vertical moving masses of the car (**P+Q**) must be multiplied by the collision coefficient $\mathbf{k_2}$ so that the rough brake owing to activation of the electric safety device or a random stop of the power, is taken into account.

4.3 Counterweight or balancing weight

The forces, acting on the guide rails of the counterweight or the balancing weight, as defined in section 2.6, should be multiplied by collision coefficient $\mathbf{k_3}$, so that the heavy bounce of the counterweight or the balancing weight when the car stops with deceleration greater than 1 g_n, is taken into account.

4.4 Collision coefficients values

The collision coefficients values are shown on Table **Z.2**.

Table Z.2: Collision coefficients

Collision in:	**Collision Coefficient**	**Value**
Instantaneous safety gear or jam device operation, without an activation cylinder type	k_1	5
Instantaneous safety gear or jam device operation, with two activation cylinder types or ratchet device with energy accumulation buffer or energy accumulation buffer		3
Progressive safety gear operation or progressive jam device or ratchet device with energy scattering buffer or energy scattering buffer		2
Breaking Valve		2
Running	k_2	1,2
Auxiliary components	k_3	(....) [1]
[1] The value must be specified by the manufacturer during the actual installation.		

5. Calculations

5.1 Calculations Range

The size of the guide rails should be calculated considering the bending forces.
In cases where safety devices will act on the guide rails, their size must be calculated taking into account the bending and buckling stresses.

With hung guide rails (attached to the top of the well) instead of buckling forces, tensile forces should be taken into account.

5.2 Bending stresses

5.2.1. Depending on:

- the suspension of the car, the counterweight or the balancing weight
- the position of the car's guide rails, the counterweight or the balancing weight
- the load and its distribution on the car,

the suspension forces - $\mathbf{F_b}$ - on the guidance blocks generate bending stresses on the guide rails.

5.2.2 For the calculation of bending stresses in a particular axis of the guide rail (**figure Z.1**) it must be assumed that:

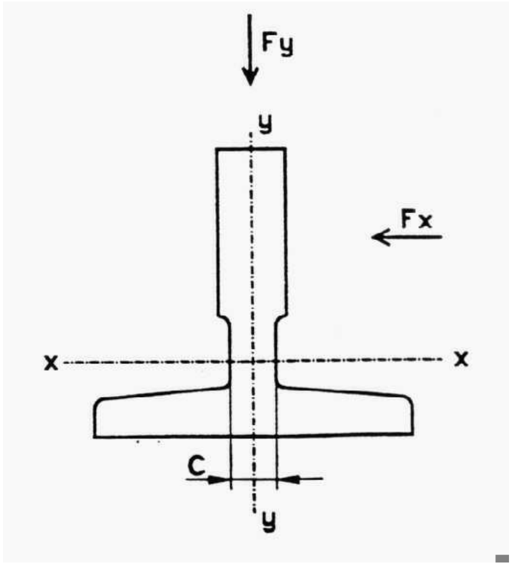


Figure Z.1: Guide rail's axis

- the guide rail is a continuous beam with elastic fix points which have a distance of length l,
- the resultant of forces, which causes bending stresses, acts in the middle of two adjacent fix points,
- bending moments act on the neutral axis of the guide rail's cross section.

For the calculation of bending stress – $\boldsymbol{\sigma_m}$ – from the forces acting perpendicular to the cross section's axis, the following equations should be used:

$$\sigma_m = \frac{M_m}{W}$$

where:

$$M_m = \frac{3 * F_b * l}{16}$$

where:

$\boldsymbol{\sigma_m}$ = bending stress in Newton per square mm,

$\boldsymbol{M_m}$ = bending moment in Newton per mm,

W = strain torsion of the cross section in cubic mm,

$\boldsymbol{F_b}$ = force acting from the blocks to the guide rail in different loading conditions, in Newton,

l = maximum distance between the guide rails arms, in mm

This calculation does not apply if the loading condition is "normal use, loading" on the assumption that the relative position of the guide rails blocks towards their fix points has taken into account.

5.2.3 The bending stresses in different axes must be combined by taking into consideration the guide rail's cross section.

If the usual values of the tables ($\boldsymbol{W_{xmin}}$ και $\boldsymbol{W_{ymin}}$) are used respectively for W_x and W_y and the acceptable stresses are not exceeded, there is no need for any further documentation.

Otherwise, the external endpoint of the guide rail's cross section in which the tensile forces reach their maximum value must be defined.

5.2.4 If more than two guide rails are used, it is allowed to consider an equal distribution of the forces between the guide rails, on the premise that their cross sections are equal.

5.2.5 If more than one safety gears are used, according to 2.8.2.2, it can be assumed that the total braking force is equally distributed on them.

5.2.5.1 In the case of multiple vertical safety gears, which act in the same guide rail, it must be assumed that the braking force of the guide rail acts in a single point.

5.2.5.2 In the case of multiple horizontal safety gears, the braking force of the guide rail must be according to 2.3 or 2.4.

5.3 Buckling

The determination of the buckling stresses uses the "omega" method and the following equations:

$$\sigma_k = \frac{(F_k + k_3 * M) * \omega}{A} \quad or \quad \sigma_k = \frac{(F_c + k_3 * M) * \omega}{A}$$

where:

$\boldsymbol{\sigma_k}$ = buckling stress, in Newton per square mm,

$\boldsymbol{F_k}$ = buckling force acting in a car's guide rail, in Newton , see section 2.3,

$\boldsymbol{F_c}$ = buckling force acting in a counterweight or balancing weight guide rail, in Newton, see section 2.4.

$\boldsymbol{k_3}$ = collision coefficient, see **Table Z.2**,

M = force acting in a guide rail due to auxiliary equipment, in Newton,

A = guide rail's cross section surface, in square mm,

$\boldsymbol{\omega}$ = omega value

The «omega» values can be derived from **Tables Z.3** and **Z.4** or can be calculated using the following polynomial equations:

$$\lambda = \frac{l_k}{i} \quad \text{and} \quad l_k = l$$

where:

$\boldsymbol{\lambda}$ = slenderness,

$\boldsymbol{l_k}$ = buckling length, in mm,

i = the minimum radius of gyration, in mm,

l = the maximum distance between the guide rails arms, in mm.

For steels with tensile strength R_m = 370 N/mm^2:

20	≤	λ	≤	60	:	ω=	$0{,}00012920 * \lambda^{1,89} + 1$
60	<	λ	≤	85	:	ω=	$0{,}00004627 * \lambda^{2,14} + 1$
85	<	λ	≤	115	:	ω=	$0{,}00001711 * \lambda^{2,35} + 1$
115	<	λ	≤	250	:	ω=	$0{,}00016887 * \lambda^{2,00}$

For steels with tensile strength R_m = 520 N/mm^2:

20	≤	λ	≤	50	:	ω=	$0{,}00008240 * \lambda^{2,06} + 1{,}021$
50	<	λ	≤	70	:	ω=	$0{,}00001895 * \lambda^{2,41} + 1{,}05$
70	<	λ	≤	89	:	ω=	$0{,}00002447 * \lambda^{2,36} + 1{,}03$
89	<	λ	≤	250	:	ω=	$0{,}00025330 * \lambda^{2,00}$

The determination of the "omega" values for steels with tensile strength $\mathbf{R_m}$ between 370 N/mm^2 and 520 N/mm^2 must be using the following equation:

$$\omega_R = \frac{\omega_{520} - \omega_{370}}{520 - 370} + \omega_{370}$$

The "omega" values for metal materials of different strength should be given by the manufacturer.

5.4 Combination of bending and buckling stresses

The resultant of bending and buckling stresses must be calculated using the following equations:

$$\sigma_m = \sigma_m + \sigma_y \qquad \leq \sigma_{al}$$

$$\sigma = \sigma_m + \frac{F_k + k_3 * M}{A} \qquad \leq \sigma_{al}$$

or

$$\sigma = \sigma_m + \frac{F_c + k_3 * M}{A} \qquad \leq \sigma_{al}$$

Buckling and bending $\sigma_c = \sigma_k + 0{,}9 * \sigma_m \qquad \leq \sigma_{al}$

where:

- $\boldsymbol{\sigma_m}$ = bending stress, in Newton per square mm,
- $\boldsymbol{\sigma_x}$ = bending stress in the x axis, in Newton per square mm,
- $\boldsymbol{\sigma_y}$ = bending stress in the y axis, in Newton per square mm,
- $\boldsymbol{\sigma_{al}}$ = allowable stress, in Newton per square mm, see 10.1.2.1.
- $\boldsymbol{\sigma\gamma}$ = buckling stress, in Newton per square mm,
- $\boldsymbol{F_k}$ = buckling strength, acting in the car's guide rail, in Newton, see 2.3,
- $\boldsymbol{F_c}$ = buckling strength, acting in the counterweight's or balancing weight's guide rail, in Newton , see 2.4,
- $\boldsymbol{k_3}$ = collision coefficient, see **Table Z.2**,
- $\boldsymbol{M}$ = strength acting in the guide rail due to auxiliary equipment, in Newton,
- $\boldsymbol{A}$ = guide rail's cross section area, in square mm.

5.5 Hinge bending

It should be taken into account the bending of the hinge. For T shape guide rails the following equation should be used:

$$\sigma_F = \frac{1{,}85 * F_x}{c^2} \quad \leq \sigma_{al}$$

where:

$\boldsymbol{\sigma_F}$ = local hinge bending stress, in Newton per square mm.

$\boldsymbol{F_x}$ = force acting in the hinge from a guidance block, in Newton,

c = width of the attached part of the foot with the blade, in mm, see figure Z.1,

$\boldsymbol{\sigma_{al}}$ = allowable stress, in Newton per square mm.

5.6 Examples with instructions, suspension and loading conditions of the car and the respective equations are given in section 7.

5.7 Deflections

Deflections should be calculated using the following equations:

$$\delta_y = 0{,}7 * \frac{F_y * \cdot l^3}{48 * E * I_x} \qquad \textit{Y-Y reference level}$$

$$\delta_x = 0{,}7 * \frac{F_x * \cdot l^3}{48 * E * I_y} \qquad \textit{X-X reference level}$$

where:

$\boldsymbol{\delta_x}$ = deflection towards the X axis, in mm,

$\boldsymbol{\delta_y}$ = deflection towards the Y axis, in mm,

$\boldsymbol{F_x}$ = reaction force towards the X axis, in Newton,

$\boldsymbol{F_y}$ = reaction force towards the Y axis, in Newton,

l = maximum distance between the guide rails brackets, in mm,

E = elasticity, in Newton per square mm,

$\boldsymbol{I_x}$ = moment of inertia towards the X axis, in mm^4,

$\boldsymbol{I_y}$ = moment of inertia towards the Y axis, in mm^4.

6. Allowable deflections

Allowable deflections of guide rails with T shape cross section are given in 10.1.2.2.

Guide rails deflections, which do not have a T shape cross section, should be limited so that 10.1.1 is satisfied.

The combination of allowable deflections and brackets deflections, which move freely along the guidance blocks and the alignment of the guide rails should not affect the requirements of 10.1.1.

7. Examples of calculation methods

The following examples are used for the calculation of the guide rails.

The symbols that follow will be used in an algorithm with a cartesian coordinate system for every possible geometric case.

The following symbols are used for the dimensions of the lift:

$\boldsymbol{D_x}$ = car size on x axis, car depth,

$\boldsymbol{D_y}$ = car size on y axis, car width,

x_c, y_c = position of cabin center (C) towards the guide rail cross section coordinates,

***x_s*, y_s** = position of suspension (S) towards the guide rail cross section coordinates,

x_p, y_p = cabin mass position (P) towards the guide rail cross section coordinates,

x_{cp}, y_{cp} = cabin center weight position (C) towards the guide rail cross section coordinates,

S = cabin suspension point,

C = cabin center,

P = car bending mass – center weight of the mass,

Q = rated load – center weight of the mass,

———> = loading direction

1, 2, 3, 4 = center of the cabin door 1, 2, 3 ή 4,

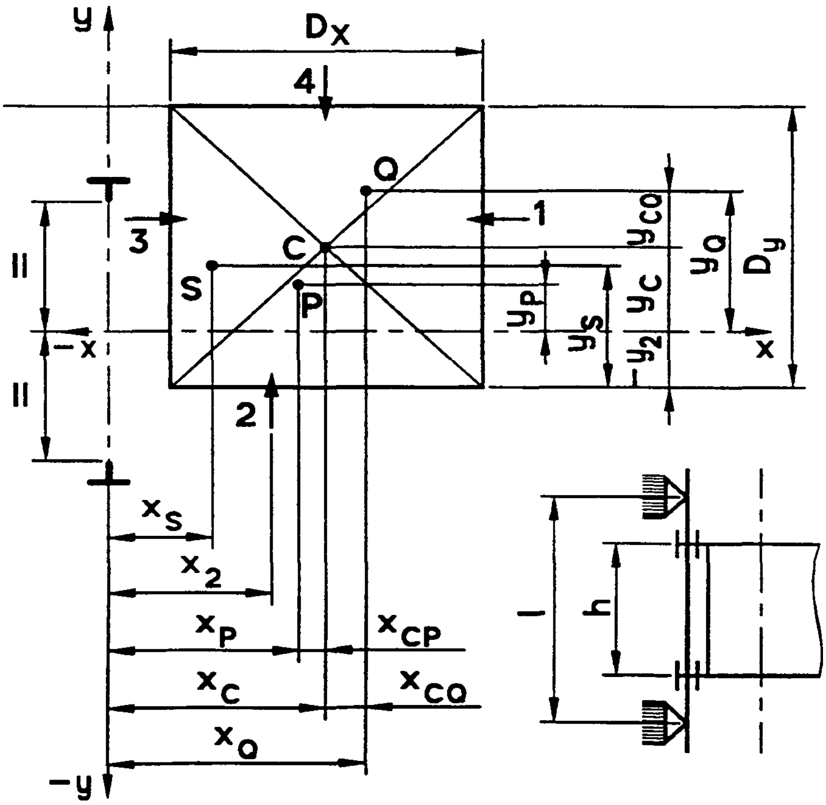
x_i, y_i = position of the cabin door, i = 1, 2, 3 ή 4,

n = guide rails number,

h = distance between the guide rails blocks,

x_Q*, *y_Q = rated load (Q) position towards the guide rail cross section coordinates,

x_{cQ}*, *y_{cQ} = *distance between the cabin center* (C) and rated load (Q) towards X and Y direction respectively



7.1 Safety gear operation

7.1.1 Bending stress

a) Bending stress towards guide rail's Y axis, due to guidance force:

$$F_x = \frac{k_1 * g_n * (Q * x_Q + P * x_P)}{n * h}, \quad M_y = \frac{3 * F_x * l}{16}, \quad \sigma_y = \frac{M_y}{W_y}$$

b) Bending stress towards guide rail's X axis, due to guidance force:

$$F_y = \frac{k_1 * g_n * (Q * y_Q + P * y_P)}{n * h/2}, \quad M_y = \frac{3 * F_y * l}{16}, \quad \sigma_y = \frac{M_x}{W_x}$$

7.1.2 Buckling

$$F_k = \frac{k_1 * g_n * (Q + P)}{n} , \sigma_k = \frac{(F_k + k_3 * M) * \omega}{A}$$

7.1.3 Combined stress

$$\sigma_m = \sigma_x + \sigma_y \qquad =< \sigma_{al}$$

$$\sigma = \sigma_m + \frac{F_k + (k_3 * M)}{A} \qquad =< \sigma_{al}$$

$$\sigma_c = \sigma_k + 0,9 * \sigma_m \qquad =< \sigma_{al}$$

7.1.4 Foot bending

$$\sigma_f = \frac{1,85 * F_x}{c^2} =< \sigma_{al}$$

7.1.5 Deflections

$$\delta_x = 0,7 * \frac{F_x * l^3}{48 * E * I_y} =< \delta_{al}$$

$$\delta_y = 0,7 * \frac{F_y * l^3}{48 * E * I_x} =< \delta_{al}$$

7.2. Operation in normal use

7.2.1. Bending stress

a) Bending stress towards guide rail's Y axis, due to guidance force:

$$F_x = \frac{k_2 * g_n *}{n * h}$$

$$M_y = \frac{3 * F_x * l}{16}$$

$$\sigma_y = \frac{M_y}{W_y}$$

b) Bending stress towards guide rail's X axis, due to guidance force:

$$F_y = \frac{k_2 * g_n *}{n * h/2}$$

$$M_x = \frac{3 * F_y * l}{16}$$

$$\sigma_x = \frac{M_x}{W_x}$$

7.2.2. Buckling

There is no buckling in normal use.

7.2.3. Combined stress

$$\sigma_m = \sigma_x + \sigma_y =< \sigma_{al}$$

$$\sigma = \sigma_m + \frac{k_3 * M}{A} =< \sigma_{al}$$

7.2.4. Foot bending

$$\sigma_f = \frac{1{,}85 * F_x}{c^2} \quad =< \sigma_{al}$$

7.2.5. Deflections

$$\delta_x = 0{,}7 * \frac{F_x * l^3}{48 * E * I_y} \quad =< \delta_{al}$$

$$\delta_y = 0{,}7 * \frac{F_y * l^3}{48 * E * I_x} \quad =< \delta_{al}$$

7.3. Loading in normal use

7.3.1. Bending stress

a) Bending stress towards guide rail's Y axis, due to guidance force:

$$F_x = \frac{g_n * P * (x_P - x_S) + F * (x_i - x_S)}{n * h}$$

$$M_y = \frac{3 * F_x * l}{16}$$

$$\sigma_y = \frac{M_y}{W_y}$$

b) Bending stress towards guide rail's X axis, due to guidance force:

$$F_y = \frac{g_n * P * (y_P - y_S) + F * (y_i - y_S)}{n * h/2}$$

$$M_x = \frac{3 * F_y * l}{16}$$

$$\sigma_x = \frac{M_x}{W_x}$$

7.3.2. Buckling

There is no buckling in normal use.

7.3.3. Combined stress

$$\sigma_m = \sigma_x + \sigma_y \quad =< \sigma_{al}$$

$$\sigma = \sigma_m + \frac{k_3 * M}{A} \quad =< \sigma_{al}$$

7.3.4. Foot bending

$$\sigma_f = \frac{1{,}85 * F_x}{c^2} =< \sigma_{al}$$

7.3.5. Deflections

$$\delta_x = 0{,}7 * \frac{F_x * l^3}{48 * E * I_y} =< \delta_{al}$$

$$\delta_y = 0{,}7 * \frac{F_y * l^3}{48 * E * I_x} =< \delta_{al}$$

c) Suspension Wire Ropes

For the selection of the wire ropes diameter the following check is made:

$v = C_m * n * F_g / (P+Q) \geq v_{al}$

where $\boldsymbol{v_{al}}$: minimum safety coefficient with values:

12 for lifts with 3 or more wire ropes

16 for lifts with 2 wire ropes

12 for lifts with rundle

12 for lifts with suspension chain

$\boldsymbol{C_m}$: suspension ratio

n: number of pulling wire ropes

$\boldsymbol{F_g}$: car's suspension wire ropes breaking force (Kp)

P: car weight (Kp)

Q: rated load (Kp)

d) Friction Pulley

The following equations are used:

$$\frac{T_1}{T_2} \leq e^{f * a}$$ for loaded car and emergency braking conditions.

$$\frac{T_1}{T_2} \geq e^{f * a}$$ for idle car conditions

where:

f = friction coefficient.

α = wire ropes winding angle in the friction pulley.

$\boldsymbol{T_1, T_2}$ = forces in the wire ropes sections that are set in both sides of the friction pulley.

It is reminded that

e: base of napierian logarithms e=2.71

Estimation of T_1 and T_2

<u>Loaded car condition</u>

The static ratio T1/T2 must be estimated for the most malign condition, in relation to the car's position inside the well, with 125% of the rated load. Condition 8.2.2 requires special treatment, if it's not covered with the 1,25 load factor.

Emergency braking condition

The dynamic ratio T1/T2 must be estimated for the most malign condition, in relation to the car's position inside the well and the loading conditions (empty or with rated load).

Every moving part must be taken into account with it's own acceleration, considering the fastening ratio of the installation.

Under no circumstances should the acceleration taken into account be less than:

- 0,5 m/s² in normal cases
- 0,8 m/s² when buffers with reduced drive length are used

Stop condition of the car

The static ratio T1/T2 must be estimated for the most malign condition, in relation to the car's position inside the well and the loading conditions (empty or with rated load).

The equations above become:

i) Car loading down

$$\frac{T_1}{T_2} = \frac{\quad + P_{wire} * g}{G * g/C_m} \le e^{f*a}$$

ii) Emergency braking conditions.

a) Full load – car down

$$\frac{T_1}{T_2} = \frac{\quad +}{G * (g - \gamma_b)/C_m} \le e^{f*a}$$

b) Empty car up

$$\frac{T_1}{T_2} = \frac{(P + P_{wire}) * (g_n - \gamma_b)/C_m}{\quad +} \le e^{f*a}$$

iii) Stop condition of the car – counterweight in its buffers and the driving gear rotates with upper direction:

$$\frac{T_1}{T_2} = \frac{(P + P_{wire}) * g}{P_{wire} * g} \ge e^{f*a}$$

where:

Q = Rated load,

P = Car weight,

g = Acceleration of gravity,

C_m = Suspension ratio,

P_{wire} = Wire rope weight,

G = Counterweight weight,

γ_b = Braking acceleration.

Friction coefficient estimation

Issues related to the sheavings of the pulleys

Semicircular sheaving and semicircular with undercut.

The following equation must be used:

4 *

$$f = \mu * \frac{}{\pi - \beta - \gamma - \sin\beta + \sin\gamma}$$

where:

β = undercut angle value,

γ = sheave angle value,

μ = friction coefficient,

f = friction factor.

The maximum value of the undercut angle β must not exceed 106° (1,83 rad.) which is equivalent to 80% of the undercut.

V type sheaving

When the sheave has not undergone a hardening process in order to reduce the pulling deterioration due to wear, it is vital that an undercut is made.

The following equation applies:

- in the case of a loaded car or emergency braking:

$$f = \mu * \frac{4 *}{\pi - \beta - \sin\beta}$$ for sheaves without hardening,

$$f = \mu * \frac{1}{\sin(\gamma/2)}$$ for sheaves with hardening,

- in the case of a stopped car:

$$f = \mu * \frac{1}{\sin(\gamma/2)}$$ for sheaves with or without hardening

where:

β = undercut angle value,

γ = sheave angle value,

μ = friction coefficient,

f = friction factor.

The maximum value of the undercut angle β must not exceed 106° (1,83 rad.) which is equivalent to 80% of the undercut. Under no circumstances, should angle γ be less than 35° for lifts.

Friction coefficient estimation

Valid values are the following

- loading conditions: $\mu = 0{,}1$
- emergency braking conditions: $\mu = \frac{0{,}1}{1 + (v/10)}$
- Stop condition of the car: $\mu = 0{,}2$

where:

μ =friction coefficient,

v =wire rope speed in the rated speed of the car.

WIRE ROPE SAFETY COEFFICIENT ESTIMATION METHOD

The estimation method of the wire rope safety coefficient takes into account:

- The traditional materials used in the design of the wire ropes drive for components such as steel/cast-iron friction pulleys.

- Steel wire ropes according to European norms
- A satisfactory lifetime of the wire ropes presuming that they are normally checked and serviced.

Equivalent number of pulleys N_{equiv}: The number of bendings and the extent of each bending cause wear to the wire rope. This is affected by the sheaving type (U or V sheaving) and the reversal or not of the bending.

The extent of the bending can be simulated with a number of single bendings.

A single bending is specified from the wire rope moving on a semicircular sheaving, where the sheaving radius is approximately 5% to 6% bigger than the rated radius of the wire rope.

The number of single bendings is equal to an equivalent number of pulleys N_{equiv}, which can be calculated by the equation:

$N_{equiv} = N_{equiv(t)} + N_{equiv(p)}$

where:

$N_{equiv(t)}$ = equivalent number of friction pulleys.

$N_{equiv(p)}$ = equivalent number of deflection pulleys.

$N_{equiv(t)}$ estimation: $N_{equiv(t)}$ values can be derived from **Table IC.1**

TABLE IC.1

V type sheavings	V angle (γ)	--	35°	36°	38°	40°	42°	45°
	$N_{equiv(t)}$	--	18,5	15,2	10,5	7,1	5,6	4,0
U type sheavings with undercut	U angle (β)	75	80°	85°	90°	95°	100°	105°
	$N_{equiv(t)}$	2,5	3,0	3,8	5,0	6,7	10,0	15,2

For U type sheavings without undercut: $N_{equiv(t)} = 1$

$N_{equiv(p)}$ estimation: A reverse curvature is taken into consideration only if the distance from wire rope contact point in two fixed adjacent pulleys doesn't exceed more than 200 times the diameter of the wire rope.

$N_{equiv(p)} = K_p * (N_{ps} + 4 * N_{pr})$

where:

N_{ps} = number of pulleys causing single bendings.

N_{pr} = number of pulleys causing reverse bendings.

K_p = coefficient of the ratio between the friction pulley diameter and the pulleys diameter.

with

$K_p = (D_t / D_p)^4$

where:

D_t = friction pulley diameter

D_p = average pulleys diameter except for the friction pulley.

Safety coefficient: For a given design of the drive wire rope, the minimum value of the safety coefficient can be chosen from figure IC.1 taking into consideration the exact ratio D_t / d_r and the calculated value of N_{equiv}.

Calculations are base on the following equations:

$$S_f = 10^{\left[2,6834 - \frac{\log\left[\frac{695,85\cdot 10^6 \cdot N_{\iota\sigma o\delta}}{\left(\frac{Dt}{Dr}\right)^{8,567}}\right]}{\log\left[77,09\left(\frac{Dt}{Dr}\right)^{-2,894}\right]}\right]}$$

where:

$\boldsymbol{S_f}$ = safety coefficient.

$\boldsymbol{N_{equiv}}$ = number of equivalent pulleys.

$\boldsymbol{D_t}$ = friction pulley diameter.

$\boldsymbol{d_r}$ = wire ropes diameter.

Therefore, by entering the appropriate values in the window, the estimated value of the safety coefficient of the suspension wire ropes is calculated.

e) Motor Power

The required motor power is calculated by the equation:

$N = F * V_c / (75 * n_1 * n_2 * n_3)$ (in HP)

where:

n_1: friction pulley performance rate

n_2: friction pulley benches performance rate

n_3: wormscrew performance rate

F: active force $F = (Q + P - G)/C_m$ (Kp)

V_c: car rated velocity (m/sec)

f) Gear regulator

For the selection of the wire ropes and the limiter pulley diameter the following checks are made:

Wire rope strength check

$v' = n' * F_g' / (G'/2 + F_1) \geq 8$

where:

$\boldsymbol{n'}$: number of pulling wire ropes

$\boldsymbol{F_1}$: required force for setting the safety gear into operation (Kp)

$\boldsymbol{G'}$: wire ropes strain force (Kp)

$\boldsymbol{F_g'}$: breaking force of the regulator wire ropes (Kp)

Also the following calculations are made:

Wire ropes strain force

It's calculated by the equation

2 F1*

G' = -------------

$(e^{f' * \alpha'} - 1)$

Semicircular sheaving with undercut.

The equation below must be used:

$$f' = \mu' * \frac{4 *}{\pi - \beta' - \gamma' - \sin\beta' + \sin\gamma'}$$

where:

β' = undercut angle value,

γ' = sheave angle value,

μ' = friction coefficient,

f' = friction factor.

The maximum value of the undercut angle β' should not exceed 106o (1,83 rad.), which is equivalent to 80% of the undercut.

The value of the sheave angle γ' should be given by the manufacturer according to the sheave design. Under no circumstances should it be less than 25o (0,43 rad.).

V type sheaving

When the sheave has not undergone a hardening process in order to reduce the pulling deterioration due to wear, it is vital that an undercut is made.

The equation below applies:

- in the case of a loaded car and an emergency breaking:

$$f' = \mu' * \frac{4 *}{\pi - \beta' - \sin\beta'} \quad \text{for sheavings without hardening}$$

$$f' = \mu' * \frac{1}{\sin(\gamma'/2)} \quad \text{for sheavings with hardening,}$$

where:

β' = undercut angle value,

γ' = sheave angle value,

μ' = friction coefficient,

f' = friction factor.

The maximum value of the undercut angle β' must not exceed 106° (1,83 rad.) which is equivalent to 80% of the undercut. Under no circumstances, should angle γ' be less than 35° for lifts.

Friction coefficient estimation

The following equation applies:

- emergency braking conditions: $\mu' = \frac{0,1}{1 + (v'/10)}$

where:

v' = wire rope speed in the rated speed of the car.

It is reminded that

e: base of napierian logarithms e=2.71

F_1 : required force for setting the safety gear into operation (Kp)

G' : wire ropes strain force (Kp)

a' : wire rope spooling angle on the limiter pulley

g) Buffers

Car and counterweight buffers

Lifts must be equipped with buffers that are set to the lower limit of the car's drive and the counterweight.

The operation point / points of the buffer / buffers under the car's projection must be characterized by an obstacle (base) of such a height that 5.7.3.3 is satisfied. For buffers with their active surface center within 0,15 m from the guide rails and the similar fastening devices except for the walls, these devices are considered to be obstacles.

Apart from the previous paragraph's requirements, rundle or chain lifts must be equipped with buffers on the top of the car, which operate in the upper limit of the drive.

Accumulation energy buffers with linear and non linear characteristics must only be used when the rated speed does not exceed 1 m/s.

Accumulation energy buffers with decelerating restoration must only be used when the rated speed of the lift does not exceed 1,6 m/s.

Energy scattering buffers can be used for lifts of any rated speed.

Accumulation energy buffers with non linear characteristics and with decelerating restoration as well as energy scattering buffers are considered to be safety components.

Car and counterweight buffers drive

The following calculations are made for the selected buffers:

1.1 Accumulation energy buffers type

1.1.1 Buffers with linear characteristics

1.1.1.1 The total effective drive of the buffers (in meters) should be at least double of the bounce height of the gravity energy, which is equal to 115% of the rated speed (0,135 * v2)(*).

However, the drive should not be less than 65 mm.

1.1.1.2 Design of the buffers must be such that will cover the drive set in 1.1.1 with the static load energy between 2,5 and 4 times the total mass of the car increased by its rated load (or counterweight's mass).

(*) $\frac{2 \cdot (1,15 \cdot v)^2}{2 \cdot g_n} = 0,1348 \cdot v^2$, rounded to $0,135\, v^2$

Calculations are based on the following equations:

$S = 0,135 * V_c^2$ (mm) (1)

where:

V_c: Rated speed of the car (m/sec)

$V_c \leq 1$ m/s

Since S < 65 mm, we choose S = 65 mm.

Furthermore,

$2,5 * (P + Q) < F_m \leq 4 * (P + Q)$ (2)

where:

F_m: Buffers static load energy

1.1.2 Buffers with non linear characteristics

1.1.2.1 Energy accumulation buffers with non linear characteristics must satisfy the following requirements:

- In the free fall case, when the car hits the car's buffer with its rated load and a speed equal to 115% of the rated speed, the average deceleration should not be more than 1 g_n.
- A deceleration more than 2,5 g_n should not last more than 0,04 s.

- Return speed of the car should not exceed 1 m/s.
- After the activation no permanent deformation should exist.

1.1.2.2. The term “fully compressed” means compression 90% of the installed buffer’s height.

Calculations are based on the following equations:

$V_c \leq 1\ m/s$

and

$V_t = 1{,}15 * V_c$

where:

V_t : Speed of the car during the fall.

V_c : Rated speed of the car.

$(1{,}15 * V_c)^2 = 2 * \gamma * S \Rightarrow$

$$\gamma = \frac{(1{,}15 * V_c)^2}{2 * S} < g_n$$

thus,

$$S > \frac{(1{,}15 * V_c)^2}{2 * g_n}$$

where:

γ = deceleration during the fall

1.1.3 Energy accumulation buffers with decelerating restoration

The requirements of 1.1.1 apply for this type of buffer.

However: $1 \leq Vc \leq 1{,}6$

1.1.4 Energy scattering buffers

1.1.3.1 The total effective drive of the buffers (in meters) should be at least equal to the stop distance due to gravity, which is equal to 115% of the rated speed (0,0674 v2).

1.1.3.2 If a system controlling the deceleration exists in the edge stops of the lift’s drive, then according to 12.8 for the calculation of the buffer’s drive in accordance with 1.1.3.1 the car’s (or counterweight’s) speed during its contact with the buffers can be used, instead of the rated speed. Although, the drive should not be smaller than:

- half of the drive calculated according to 1.1.3.1, if the rated speed does not exceed 4,0 m/s.

  The drive should not, in any case, be smaller than 0,42 m,
- one third of the drive calculated according to 1.1.3.1, if the rated speed exceeds 4,0 m/s.

  The drive should not, in any case, be smaller than 0,54 m.

1.1.3.3 Energy scattering buffers must satisfy the following requirements:

- in the free fall case, when the car hits the car’s buffer with its rated load and a speed equal to 115% of the rated speed, the average deceleration should not be more than 1 g_n.
- a deceleration more than 2,5 g_n should not last more than 0,04 s.
- after the activation no permanent deformation should exist.

1.1.3.4 The normal operation of the lift must be possible only when buffers are in their normal loose position after their operation. for this to be achieved an electric safety device must be used.

1.1.3.5 If buffers are hydraulic, then they must be constructed in such a way that the inspection of the fluid level is easy.

Calculations are based on the following equations:

$S = 0{,}0674 * V_c^2$ (mm) (1)

where:

V_c: Rated speed of the car (m/sec)

If $V_c \leq 4$ m/s, then it's essential that $S \geq 0{,}42$ m.

If $V_c > 4$ m/s, then it's essential that $S \geq 0{,}54$ m.

ELECTRIC LIFTS DATA CALCULATION

1. CONSTRUCTION DATA

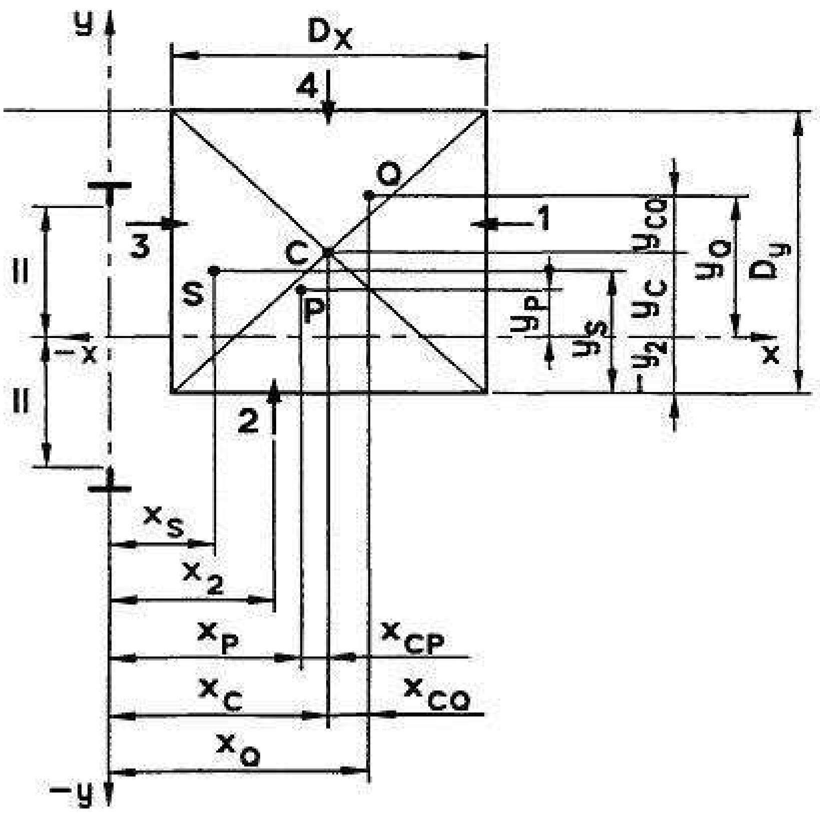
Elevator kind : ELEVATOR PERSONS

C_m = suspension ratio 1:1, 2:1 etc.	C_m = 2
D_x = Car size along dimension x	D_x = 1400.00 mm
D_y = Car size along dimension y	D_y = 1100.00 mm
Lg = Car drive length	Lg= 11.50 m
Number of stops : 4	
P = sum of car and frame force	P = 500 kg
Q = nominal load (persons x 75 kg, 8 persons)	Q = 600 kg
V_c = car nominal speed	V_c = 0.63 m/sec
n = number of pulling wire ropes	n = 4
d = diameter of pulling wire ropes	d = 8.0 mm
F_g = breaking load of pulling wire ropes	F_g = 3810 kg
P_{wrp} = Weight of wire ropes	P_{wrp}=10.95 kg
P_{cbl} = Weight of cable	P_{cbl}=3.45 kg
Dt = friction pulley diameter (Dt ≥ 40 d)	Dt = 360.0 mm
Dp = deviation pulley diameter (Dp ≥ 40 d)	Dp = 320.00 mm
Kind of Pulleys : V type sheaving with undercut	
alpha= wire rope overlapping angle on friction pulley	alpha =180°
beta= friction pulley undercut angle	beta =97°
gamma= friction pulley sheavig\ng angle	gamma =38°
Nps = number of pulleys, causing simple bendings	Nps = 1
Npr = number of pulleys, causing reverse bendings	Npr = 0
A = guide rail crosscut T 50 x 50 x 9	A = 706.00 mm²
N_r = number of guide rails	N_r = 2
l_k = bowing length (maximum distance between guide rail fastenings)	l_k = 1100.0 mm
A_{co} = counterweight guide rail crosscut T 50 x 50 x 9	A_{co} =706.00 mm²
V' = speed limiter activation velocity	V' = 0.72 m/sec
G' = Weight of Strainer	G '=50 Kg
d' = speed limiter wire ropes diameter	d' = 6.0 mm
F_g' = breaking load of speed limiter wire ropes	F_g' = 2140 kg
D' = friction pulley diameter of limiter (D' >= 30 d')	D' = 180.0 mm
Dp' = pulley diameter of strainer (Dp' >= 30 d')	Dp'= 180.0 mm
Kind of Limiter Pulleys : V type sheaving with undercut	
alpha´= wire rope overlapping angle on speed limiter pulley	alpha´=180°
speed limiter pulley	beta´ =97°
gamma´= speed limiter pulley sheave angle of non standart form	gamma´=35°
n' = number of speed limiter wire ropes	n' = 1

1 safety gear of both directions selected, of type :
Instantaneous braking, chock type

UNITS: 1 kW = 1.341 * HP Joule = Ntm

2.GUIDE RAILS CALCULATION



rails technic data

Dimensions : T 50 x 50 x 9

Material : St 37

Rated load Q = 600.00 kg

Cabin weight P_{car} = 500.00 kg

Frame weight P_{fr} = 0.00 kg

Door weight 1 P_{T1} = 0.00 kg

Door weight 2 P_{T2} = 0.00 kg

Car weight $P = P_{car} + P_{fr} + P_{T1} + P_{T2}$ = 500.00 + 0.00 + 0.00 + 0.00 = 500.00 kg

x position of cabin center towards guide rail crosscut x coordinate X_c = 0.00 mm

y position of cabin center towards guide rail crosscut y coordinate Y_c = 0.00 mm

x position of frame mass towards guide rail x coordinate x_{fr} = 0.00 mm

y position of frame mass towards guide rail y coordinate y_{fr} = 0.00 mm

x position of door 1 towards guide rail x coordinate x_1 = 700.00 mm

x position of door 2 towards guide rail x coordinate x_2 = 0.00 mm

y position of door 1 towards guide rail y coordinate y_1 = 0.00 mm

y position of door 2 towards guide rail y coordinate y_2 = 0.00 mm

x position of car mass towards guide rail x coordinate

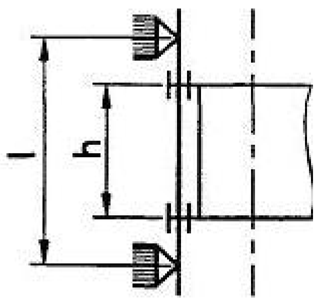
$x_P = (P_{car}*X_c + P_{fr}*X_{fr} + P_{T1}*X_1 + P_{T2}*X_2) / P =$

= (500.00*0.00 + 0.00*0.00 + 0.00*700.00 + 0.00*0.00) / 500.00 = 0.00 mm

y position of car mass towards guide rail y coordinate

$y_P = (P_{car}*Y_c + P_{fr}*Y_{fr} + P_{T1}*Y_1 + P_{T2}*Y_2) / P =$

= (500.00*0.00 + 0.00*0.00 + 0.00*0.00 + 0.00*0.00) / 500.00 = 0.00 mm



Distance between Rail Brackets l : 1100.0 mm

Vertical distance of chassis drive h : 2700.0 mm

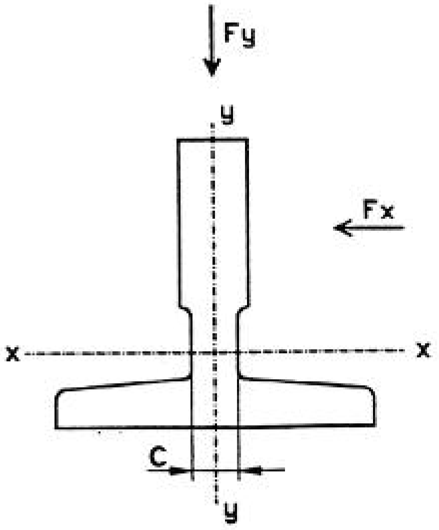
Number of guide rails n = 2

Car size along dimension x Dx = 1400.00 mm

Car size along dimension y Dy = 1100.00 mm

Running gear vertical drive distance h = 2700.00 mm

Guide rail fastenings distance l = 1100.00 mm
Crosscut A = 706.00 mm^2
Traction torsion W_x = 5060.00 mm^3
Traction torsion W_y = 2600.00 mm^3
Stagnancy radius i_y = 9.61
Buckling coefficient lamda = l/i_y = 114.46
Based on material and lamda, manufacturer's tables give bending coefficient omega(lamda) = 2.231



CASE OF 1/8 LOAD TRANSPORTATION ON (X) AXIS
$_q$ = X_c +D_x / 8 = 175.00 mm
$_q$ = Y_c = 0.00 mm

2.1. Safety gear operation

2.1.1. Bending stress
safety gear operation, collision coefficient is k_1 = 5.00
) Guide rail bending stress on Y axis, caused by driving force:

$$F_x = \frac{k_1 * g_n * (Q * x_Q + P * x_P)}{n * h} = \frac{5.00 * 9.81 * (600.00 * 175.00 + 500.00 * 0.00)}{2 * 2700.00} =>$$

F_x = 953.75 Nt

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 953.75 * 1100.00}{16} = 196710.94 \text{ Nt * mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{196710.94}{2600.00} = 75.66 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_y = \frac{k_1 * g_n * (Q * y_Q + P * y_P)}{n * h/2} = \frac{5.00 * 9.81 * (600.00 * 0.00 + 500.00 * 0.00)}{2 * 2700.00 / 2} =>$$

F_y = 0.00 Nt

3 * F_y * l   3 * 0.00 * 1100.00

$$M_x = \frac{}{16} = \frac{}{16} = 0.00 \text{ Nt * mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{0.00}{5060.00} = 0.00 \text{ Nt / mm}^2$$

2.1.2 Buckling

$$F_k = \frac{k_1 * g_n * (Q + P)}{n} = \frac{5.00 * 9.81 * (600.00 + 500.00)}{2} = 26977.50 \text{ Nt}$$

$$sigma_{Gk} = \frac{(F_k + k_3 * M) * omega}{A} = \frac{(26977.50 + 0.000 * 0.000) * 2.231}{706.00} = 85.25 \text{ Nt / mm}^2$$

2.1.3. Combined stress

$sigma_m = sigma_x + sigma_y \le sigma_{al}$ => $75.66 = 0.00 + 75.66 \le 205.00$ Nt / mm^2

$$sigma = sigma_m + \frac{F_k + k_3 * M}{A} \le sigma_{al} \Rightarrow 113.87 = 75.66 + \frac{26977.50 + 0.000 * 0.000}{706.00} \le 205.00 \text{ Nt / mm}^2$$

$sigma_c = sigma_k + 0.9 * sigma_m \le sigma_{al}$ => $153.34 = 85.25 + 0.9 * 75.66 \le 205.00$ Nt / mm^2

2.1.4. Hinge bending

Width of hinge-bar connection c = 7.50 mm
Moment of inertia on x axis J_x = 167000.00 mm^4
Moment of inertia on y axis J_y = 65200.00 mm^4

$$sigma_f = \frac{1.85 * F_x}{c^2} \le sigma_{al} \Rightarrow 31.37 = \frac{1.85 * 953.75}{7.50^2} \le 205.00 \text{ Nt / mm}^2$$

2.1.5. Bending vectors

$$delta_x = 0.7 * \frac{F_x * l^3}{48 * E * J_y} \le delta_{al} \Rightarrow 1.378 = 0.7 * \frac{953.75 * 1100.00^3}{48 * 206010 * 65200.00} \le 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} \le delta_{al} \Rightarrow 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 167000.00} \le 5 \text{ mm}$$

2.2<u>. Normal use operation</u>

2.2.1. Bending stress
normal use operation, collision coefficient is $k_2 = 1.2$
) Guide rail bending stress on Y axis, caused by driving force:

$$k_2 * g_n * (Q * (x_Q - x_S) + P * (x_P - x_S))$$

$$F_x = \frac{\qquad}{n * h} =$$

$$\frac{1.2 * 9.81 * (600.00 *(175.00 - 0.00) + 500.00 * (0.00 - 0.00))}{2 * 2700.00} = 228.90 \text{ Nt}$$

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 228.90 * 1100.00}{16} = 47210.63 \text{ Nt} * \text{mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{47210.63}{2600.00} = 18.16 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_y = \frac{k_2 * g_n * (Q * (y_Q – y_S) + P * (y_P – y_S))}{n * h/2} =$$

$$\frac{1.2 * 9.81 * (600.00 * (0.00 - 0.00) + 500.00 * (0.00 - 0.00)}{2 * 2700.00 / 2} = 0.00 \text{ Nt}$$

$$M_x = \frac{3 * F_y * l}{16} = \frac{3 * 0.00 * 1100.00}{16} = 0.00 \text{ Nt} * \text{mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{0.00}{5060.00} = 0.00 \text{ Nt / mm}^2$$

2.2.2. Buckling:
is no buckling while normal use.

2.2.3. Combined stress

$$sigma_m = sigma_x + sigma_y <= sigma_{al} \quad => 18.158 = 0.00 + 18.16 \quad <= 165.000 \text{ Nt / mm}^2$$

$$sigma = sigma_m + \frac{k_3 * M}{A} <= sigma_{al} \quad => 113.87 = 18.158 + \frac{0.000 * 0.000}{706.00} \quad <= 165.000 \text{ Nt / mm}^2$$

2.2.4. Hinge bending

$$sigma_F = \frac{1.85 * F_x}{c^2} <= sigma_{al} \quad => 7.53 = \frac{1.85 * 228.90}{7.50^2} \quad <= 165.000 \text{ Nt / mm}^2$$

2.2.5. Bending vectors

$$delta_x = 0.7 * \frac{F_x * l^3}{48 * E * J_y} <= delta_{al} \quad => 0.331 = 0.7 * \frac{228.90 * 1100.00^3}{48 * 206010 * 65200.00} \quad <= 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} <= delta_{al} \quad => 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 167000.00} \quad <= 5 \text{ mm}$$

2.3. Normal use loading

2.3.1. Bending stress

) Guide rail bending stress on Y axis, caused by driving force:

$F_S = 0.40 * g_n * Q = 2354.40$ Due to the fact that nominal load is less than 2500 Kg

$$F_x = \frac{g_n * P * (x_P - x_S) + F_S * (x_i - x_s)}{n * h} =$$

$$\frac{9.81 * 500.00 * (0.00 - 0.00) + 2354.40 * (700.00 - 0.00)}{2 * 2700.00} = 305.20 \text{ Nt}$$

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 305.20 * 1100.00}{16} = 62947.50 \text{ Nt} * \text{mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{62947.50}{2600.00} = 24.21 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$g_n * P * (y_P - y_S) + F * (y_i - y_s)_y = \frac{}{n * h/2} =$$

$$\frac{9.81 * 500.00 * (0.00 - 0.00) + 2354.40 * (0.00 - 0.00)}{2 * 2700.00 /2} = 0.00 \text{ Nt}$$

$$M_x = \frac{3 * F_y * l}{16} = \frac{3 * 0.00 * 1100.00}{16} = 0.00 \text{ Nt} * \text{mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{0.00}{5060.00} = 0.00 \text{ Nt / mm}^2$$

2.3.2. Buckling:

is no bowing while normal use.

2.3.3. Combined stress

$$sigma_m = sigma_x + sigma_y \quad <= sigma_{al} => 24.211 = 0.00 + 24.21 \quad <= 165.000 \text{ Nt / mm}^2$$

$$sigma = sigma_m + \frac{k_3 * M}{A} <= sigma_{al} => 24.211 = 24.211 + \frac{0.000 * 0.000}{706.00} <= 165.000 \text{ Nt / mm}^2$$

2.3.4. Hinge bending

$$sigma_f = \frac{1.85 * F_x}{c^2} <= sigma_{al} => 10.04 = \frac{1.85 * 305.20}{7.50^2} <= 165.000 \text{ Nt / mm}^2$$

2.3.5. Bending vectors

$$F_x * l^3 \qquad 305.20 * 1100.00^3$$

$$delta_x = 0.7 * \frac{}{48 * E * J_y} \leq delta_{al} \Rightarrow 0.441 = 0.7 * \frac{}{48 * 206010 * 65200.00} \leq 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} \leq delta_{al} \Rightarrow 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 167000.00} \leq 5 \text{ mm}$$

CASE OF 1/8 LOAD TRANSPORTATION ON (Y) AXIS

$_q = X_c = 0.00$ mm
$_q = Y_c + D_y / 8 = 137.50$ mm

2.1. Safety gear operation

2.1.1. Bending stress
safety gear operation, collision coefficient is $k_1 = 5.00$
) Guide rail bending stress on Y axis, caused by driving force:

$$F_x = \frac{k_1 * g_n * (Q * x_Q + P * x_P)}{n * h} = \frac{5.00 * 9.81 * (600.00 * 0.00 + 500.00 * 0.00)}{2 * 2700.00} \Rightarrow$$

$F_x = 0.00$ Nt

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 0.00 * 1100.00}{16} = 0.00 \text{ Nt * mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{0.00}{2600.00} = 0.00 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_y = \frac{k_1 * g_n * (Q * y_Q + P * y_P)}{n * h/2} = \frac{5.00 * 9.81 * (600.00 * 137.50 + 500.00 * 0.00)}{2 * 2700.00 / 2} \Rightarrow$$

$F_y = 1498.75$ Nt

$$M_x = \frac{3 * F_y * l}{16} = \frac{3 * 1498.75 * 1100.00}{16} = 309117.19 \text{ Nt * mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{309117.19}{5060.00} = 61.09 \text{ Nt / mm}^2$$

2.1.2 Buckling:

$$F_k = \frac{k_1 * g_n * (Q + P)}{n} = \frac{5.00 * 9.81 * (600.00 + 500.00)}{2} = 26977.50 \text{ Nt}$$

$$sigma_k = \frac{(F_k + k_3 * M) * omega}{A} = \frac{(26977.50 + 0.000 * 0.000) * 2.231}{706.00} = 85.25 \text{ Nt / mm}^2$$

2.1.3. Combined stress

$$sigma_m = sigma_x + sigma_y <= sigma_{al} \Rightarrow 61.09 = 61.09 + 0.00 <= 205.00 \text{ Nt / mm}^2$$

$$sigma = sigma_m + \frac{F_k + k_3 * M}{A} <= sigma_{al} \Rightarrow 99.30 = 61.09 + \frac{26977.50 + 0.000 * 0.000}{706.00} <= 205.00 \text{ Nt / mm}^2$$

$$sigma_c = sigma_k + 0.9 * sigma_m <= sigma_{al} \Rightarrow 140.23 = 85.25 + 0.9 * 61.09 <= 205.00 \text{ Nt / mm}^2$$

2.1.4. Hinge bending

Width of hinge-bar connection c = 7.50 mm
Moment of inertia on x J_x = 167000.00 mm^4
Moment of inertia on y J_y = 65200.00 mm^4

$$sigma_f = \frac{1.85 * F_x}{c^2} <= sigma_{al} \Rightarrow 0.00 = \frac{1.85 * 0.00}{7.50^2} <= 205.00 \text{ Nt / mm}^2$$

2.1.5. Bending vectors

$$delta_x = 0.7 * \frac{F_x * l^3}{48 * E * J_y} <= delta_{al} \Rightarrow 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 65200.00} <= 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} <= delta_{al} \Rightarrow 0.846 = 0.7 * \frac{1498.75 * 1100.00^3}{48 * 206010 * 167000.00} <= 5 \text{ mm}$$

2.2. Normal use operation

2.2.1. Bending stress
normal use operation, collision coefficient is $k_2 = 1.2$
) Guide rail bending stress on Y axis, caused by driving force:

$$F_x = \frac{k_2 * g_n * (Q * (x_Q - x_S) + P * (x_P - x_S))}{n * h} =$$

$$\frac{1.2 * 9.81 * (600.00 *(0.00 - 0.00) + 500.00 * (0.00 - 0.00))}{2 * 2700.00} = 0.00 \text{ Nt}$$

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 0.00 * 1100.00}{16} = 0.00 \text{ Nt * mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{0.00}{2600.00} = 0.00 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_y = \frac{k_2 * g_n * (Q * (y_Q - y_S) + P * (y_P - y_S))}{n * h/2} =$$

$$\frac{1.2 * 9.81 * (600.00 * (137.50 - 0.00) + 500.00 * (0.00 - 0.00)}{2 * 2700.00 / 2} = 359.70 \text{ Nt}$$

$$M_x = \frac{3 * F_y * l}{16} = \frac{3 * 359.70 * 1100.00}{16} = 74188.13 \text{ Nt * mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{74188.13}{5060.00} = 14.66 \text{ Nt / mm}^2$$

2.2.2. Buckling:
is no bowing while normal use.

2.2.3. Combined stress

$$sigma_m = sigma_x + sigma_y <= sigma_{al} \Rightarrow 14.662 = 14.66 + 0.00 <= 165.000 \text{ Nt / mm}^2$$

$$sigma = sigma_m + \frac{k_3 * M}{A} <= sigma_{al} \Rightarrow 14.662 = 14.662 + \frac{0.000 * 0.000}{706.00} <= 165.000 \text{ Nt / mm}^2$$

2.2.4. Hinge bending

$$sigma_F = \frac{1.85 * F_x}{c^2} <= sigma_{al} \Rightarrow 0.00 = \frac{1.85 * 0.00}{7.50^2} <= 165.000 \text{ Nt / mm}^2$$

2.2.5. Bending vectors

$$delta_x = 0.7 * \frac{F_x * l^3}{48 * E * J_y} <= delta_{al} \Rightarrow 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 65200.00} <= 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} <= delta_{al} \Rightarrow 0.203 = 0.7 * \frac{359.70 * 1100.00^3}{48 * 206010 * 167000.00} <= 5 \text{ mm}$$

2.3. Normal use loading

2.3.1. Bending stress
) Guide rail bending stress on Y axis, caused by driving force:

$F_S = 0.40 * g_n * Q = 2354.40$ Due to the fact that nominal load is less than 2500 Kg

$$F_x = \frac{g_n * P * (x_P - x_S) + F_S * (x_i - x_s)}{n * h} =$$

$$\frac{9.81 * 500.00 * (0.00 - 0.00) + 2354.40 * (700.00 - 0.00)}{2 * 2700.00} = 305.20 \text{ Nt}$$

$$M_y = \frac{3 * F_x * l}{16} = \frac{3 * 305.20 * 1100.00}{16} = 62947.50 \text{ Nt * mm}$$

$$sigma_y = \frac{M_y}{W_y} = \frac{62947.50}{2600.00} = 24.21 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_y = \frac{g_n * P * (y_P - y_S) + F * (y_i - y_s)}{n * h/2} =$$

$$\frac{9.81 * 500.00 * (0.00 - 0.00) + 2354.40 * (0.00 - 0.00)}{2 * 2700.00 /2} = 0.00 \text{ Nt}$$

$$M_x = \frac{3 * F_y * l}{16} = \frac{3 * 0.00 * 1100.00}{16} = 0.00 \text{ Nt * mm}$$

$$sigma_x = \frac{M_x}{W_x} = \frac{0.00}{5060.00} = 0.00 \text{ Nt / mm}^2$$

2.3.2. Buckling:
is no buckling while normal use.

2.3.3. Combined stress

$sigma_m = sigma_x + sigma_y <= sigma_{al} => 24.211 = 0.00 + 24.21 <= 165.000$ Nt / mm^2

$$sigma = sigma_m + \frac{k_3 * M}{A} <= sigma_{al} => 24.211 = 24.211 + \frac{0.000 * 0.000}{706.00} <= 165.000 \text{ Nt / mm}^2$$

2.3.4. Hinge bending

$$sigma_f = \frac{1.85 * F_x}{c^2} <= sigma_{al} => 10.04 = \frac{1.85 * 305.20}{7.50^2} <= 165.000 \text{ Nt / mm}^2$$

2.3.5. Bending vectors

$$delta_x = 0.7 * \frac{F_x * l^3}{48 * E * J_y} <= delta_{al} => 0.441 = 0.7 * \frac{305.20 * 1100.00^3}{48 * 206010 * 65200.00} <= 5 \text{ mm}$$

$$delta_y = 0.7 * \frac{F_y * l^3}{48 * E * J_x} <= delta_{al} => 0.000 = 0.7 * \frac{0.00 * 1100.00^3}{48 * 206010 * 167000.00} <= 5 \text{ mm}$$

3. CAR SUSPENSION-COUNTERWEIGHT WIRE ROPES CALCULATION -

4 6 x 9 SEALE wireropes of 8.0 mm diameter, breaking limit Fg=3810 Kg and weight P_{wrp} = 10.95 Kg are chosen.
pulley diameter and pulleys diameter ratio coefficient.
$_p$ = $(Dt/Dp)^4 = (360.00 / 320.00)^4 = 1.60$
wire ropes equivalent number :
ropes safety factor limit:
$_f = 10^k$, where :

$$\frac{\log((695.85 * 10^6 * N_{equiv}) / (D_t / d)^{8.567})}{\log(77.09 * (D_t / d)^{-2.894})} = 2.6834 - \text{-----}$$

log((695.85 * 10^6 * 12.10) / (360.00 / 8.00)$^{8.567}$)

$= \; 2.6834 \; - \; \dfrac{\quad}{\log(77.09 * (360.00 / 8.00)^{-2.894})} = 1.221$

:

coefficient : **v** = $n*F_g / (((P+Q)/ C_m) + P_{wrp})$

thus : **v** = 4x3810/((500+600)/2 + 10.95) = 27.168

and $v >= S_f$

4. FRICTION PULLEY CALCULATION

i) Loading conditions :

Coefficient my :

$my_1 = 0.1$

For V type thread fittings without hardening it is:

$f_1 = 4*my_1*(1 - \sin(beta/2)) / (pi - beta - \sin(b)) = 4*0.1*(1 - \sin(97/2)) / (pi - 97/180 - \sin(97)) = 0.220$

Sliding safety limit

$e^{f1*alpha} = e^{0.220*180} = 2.00$

Sliding safety

$T_1 / T_2 = (((1.25*Q+P)*g/c_m) + P_{wire.}*g) \, / \, (G*g) = (((1.25*600+500) / 2) + 10.95) \, / \, (800) = 1.59$

thus

$1.59 = T_1 / T_2 \le e^{f1*alpha} = 2.00$

ii) Emergency braking conditions :

Coefficient my :

$my_2 = 0.1 / (1 + V_c/10) = 0.1 / (1 + 0.63/10) = 0.089$

For V type thread fittings without hardening it is:

$f_2 = 4*my_2*(1 - \sin(beta/2)) / (pi - beta - \sin(b)) = 4*0.089*(1 - \sin(97/2)) / (pi - 97/180 - \sin(97)) = 0.196$

Sliding safety limit

$e^{f2*alpha} = e^{0.196*180} = 1.85$

a) Car at the lower stop - Full load :

Sliding safety

$T_1 = (Q+P)*(g + gamma_{pi})/c_m + P_{wrp.}* (g + c_m * gamma_{pi}) = (600+500)*(9.81+0.50) / 2 + 10.95*(9.81 + 2*0.50) = 5788.85$ N

$T_2 = G*(g - gamma_{pi})/ c_m = 800*(9.81-0.50) / 2 = 3724.00$ N

$T_1 / T_2 = 1.55$

therefore

$1.55 = T_1 / T_2 \le e^{f2*alpha} = 1.85$

b) Empty car at the upper stop :

Sliding safety

$T_1 = (P + P_{cbl})*(g - gamma_{pi})/c_m = (500 + 3.45)*(9.81-0.50) / 2 = 2343.56$ N

$T_2 = G*(g - gamma_{pi})/ c_m + P_{wrp.}* (g + c_m * gamma_{pi}) = 800*(9.81-0.50) / 2 + 10.95*(9.81 + 2*0.50) = 4242.35$ N

$T_2 / T_1 = 0.70$

therefore

$1.81 = T_2 / T_1 \le e^{f2*alpha} = 1.85$

) Empty car - Superjacent counterweight:

Coefficient my :

$my_3 = 0.2$

For V type thread fittings without hardening it is:

$f_3 = my_3$ / sin(gamma/2) = 0.2 / sin(38/2) = 0.614

Sliding safety limit
$e^{f3*alpha} = e^{0.614*180} = 6.89$

Sliding safety
$T_1 / T_2 = (P*g) / (P_{wire.} * g*c_m)$ = 500 **/** 10.95 = 22.99

thus
$22.99 = T_1 / T_2 \geq e^{f3*alpha} = 6.89$
diameter chosen:= 360.0 mm
is:>= 40 * d <=> 360.0 mm >= 40 * 8.0 mm = 320.0 mm
pulley diameter chosen: = 320.00 mm
is: >= 40 * d <=> 320.00 >= 40 * 8.0 mm = 320.0 mm Dp<=Dt
?

5. ELECTRIC MOTOR POWER CALCULATION

power is:

N = $F*V_c*C_m/(75*n)$ sigma HP, F = (Q+P-G)/Cm

where : **n_1** : friction pulley performance rate = 0.8
n_2 : friction pulley benches performance rate = 0.7
n_3 : wormscrew performance rate = 0.6 **n** : performance rate of the whole system = $n_1*n_2*n_3$=
= 0.8x0.7x0.6 = 0.34

Thus : **N** = 150x0.63x2/(75x0.34) = 7.5 HP

N = 7.5 HP or5.59 KW
}

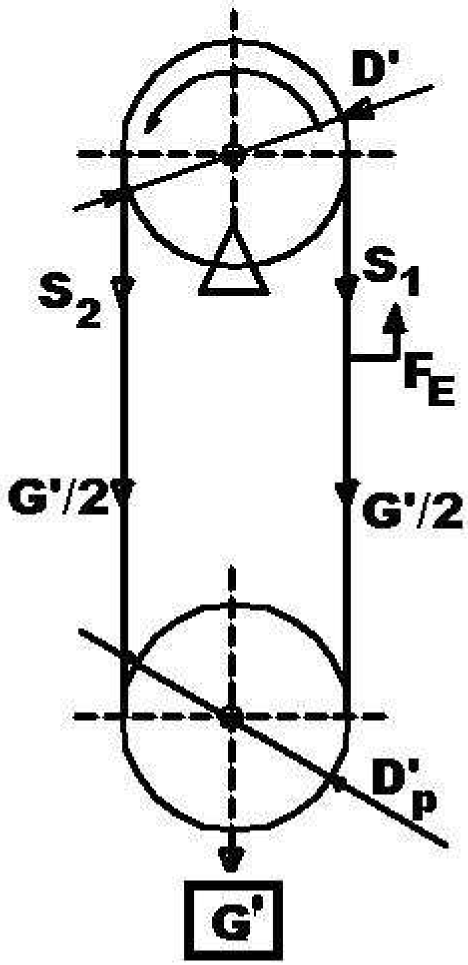
6. SPEED LIMITER CALCULATION

Friction coefficient between wire ropes and speed limiter pulley:

$$\mathbf{my' =} \frac{0.1}{1 + V'/10} = \frac{0.1}{1 + 0.72/10} = 0.093$$

For V type thread fittings with hardening, without undercut, the friction coefficient between the wire rope and the speed limiter pulley sheaves is:

$$\mathbf{f'} = my' * \frac{1}{\sin(\text{ gamma}'/2\ )} = 0.093 * \frac{1}{\sin(\ 35/2\ )} = 0.310$$

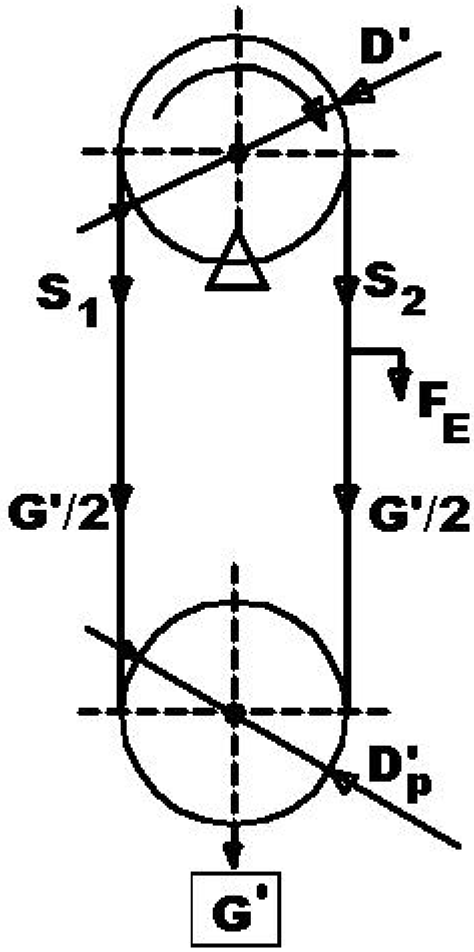


Safety gear activation force during ascending : $_{Eup.}$= G'*(e$^{f' * alpha'}$ - 1)/2 =41.22 kg

acting on the wire rope during ascending :$_{2up}$ = F_{Eup} + G'/2 = 66.22 kg

Safety gear activation force during descending : $_{Ed.}$= (G’/2)*(1 - 1/e$^{f´ * alpha´}$) =15.56 kg

acting on the wire rope during descending :$_{2d}$ = G’/2 =.25.00 kg



Wire rope safety coefficient calculation :

v' = n'*F_g'/ S_{2max}

thus :

v' = 1x2140/66.22 = 32.31 >= 8

of pulley chosen:’ = 180.0 mm mm
’ >= 30 * d’ <=> 180.0 mm >= 30 * 6.0 mm = 180.0 mm
’ >= 30 * d’ <=> 180.0 mm >= 30 * 6.0 mm = 180.0 mm Dp’<=D

7. BUFFERS CALCULATION

Car and counterweight buffers :of buffer selected: drive length required S:

S = 135*V_c*V_c = 135*0.63x0.63 = 53.58 mm

As long as S < 65 mm, we set S = 65 mm
have been designed to cover the drive length, mentioned above,
having a static load energy per buffer of :

2.5*(P+Q+P_{wrp})/n < f_m < 4*(P+Q+P_{wrp})/n =>

$=> 2777.37 < f_m < 4443.79$

}

8. COUNTERWEIGHT GUIDE RAILS CALCULATION

weight G(kg) = 800.00 kg

guide rails technic data

Counterweight guide rails Nr 5380

Dimensions : T 50 x 50 x 9

Material : St 37

Crosscut A_{cnt}: 706.00 mm^2

counterweight rail brackets l_g = 1100.00 mmvertical drive distance h_g = 2700.00 mmradius i_y = 9.61coefficient lamda = l_g/i_y =

Calculation for normal use-operation

a) Guide rail bending stress on Y axis, caused by driving force:

$$F_{Gx} = \frac{k_2 * g_n * G * x_G}{n_G * h_G} =$$

$$\frac{1.2 * 9.81 * 800.00 * 15.00}{2 * 2700.00} = 26.16 \text{ Nt}$$

$$M_{Gy} = \frac{3 * F_{Gx} * l_G}{16} = \frac{3 * 26.16 * 1100.00}{16} = 2.231 \text{ Nt * mm}$$

$$sigma_{Gy} = \frac{M_{Gy}}{W_{Gy}} = \frac{5395.50}{2600.00} = 2.08 \text{ Nt / mm}^2$$

) Guide rail bending stress on X axis, caused by driving force:

$$F_{Gy} = \frac{k_2 * g_n * G * y_G}{n_G * h_G/2} =$$

$$\frac{1.2 * 9.81 * 800.00 * 25.00}{2 * 2700.00 /2} = 43.60 \text{ Nt}$$

$$M_{Gx} = \frac{3 * F_{Gy} * l_G}{16} = \frac{3 * 43.60 * 1100.00}{16} = 8992.50 \text{ Nt * mm}$$

$$sigma_{Gx} = \frac{M_{Gx}}{W_{Gx}} = \frac{8992.50}{5060.00} = 1.78 \text{ Nt / mm}^2$$

) Combined stress

$sigma_{Gm} = sigma_{Gx} + sigma_{Gy} <= sigma_{Gal}$ => 3.85 = 2.08 + 1.78 <= 165.00 Nt / mm^2

d) Hinge bending

$$sigma_{Gf} = \frac{1.85 * F_{Gx}}{} <= sigma_{Gal} => 0.86 = \frac{1.85 * 26.16}{} <= 165.00 \text{ Nt / mm}^2$$

c_G^2 7.50 ^{2}

) Bending vectors

$$delta_{Gx} = 0.7 * \frac{F_{Gx} * l_G^3}{48 * E * J_{Gy}} <= delta_{Gal} => 0.04 = 0.7 * \frac{26.16 * 1100.00^3}{48 * 206010 * 0.00} <= 10 \text{ mm}$$

$$delta_{Gy} = 0.7 * \frac{F_{Gy} * l_G^3}{} <= delta_{Gal} => 0.02 = 0.7 * \frac{43.60 * 1100.00^3}{} <= 10 \text{ mm}$$

?

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THE ENGINEER

ELECTRIC ELEVATOR BILL OF QUANTITIES

GUIDE RAIL

GUIDE RAIL DIMENSIONS	COST(€/m)	QUANTITY(m)	TOT.COST(€)
Nr 5380	0.00	33.000	0.00

WIRE ROPES

WIRE ROPE DIAMETER	COST(€/m)	QUANTITY(m)	TOT.COST(€)
8 mm	0.00	140.000	0.00
6 mm	0.00	28.000	0.00

PULLEYS

PULLEY DIAMETER	COST(€)
360 mm	0.00
180 mm	0.00

MOTOR

MOTOR KIND	COST(€)
Z 132.19/8.2 HP	0.00

TECHNICAL DESCRIPTION OF ELECTRIC LIFT

Employer :
:
:

Project : ELECTROMECHANICAL ELEVATOR
:
:

Location :
:

Date : MARCH 2007

Project Managers:
:
:
:

Notes :
:

1. ASSUMPTIONS - REGULATIONS

In this project the respective regulations for the installation and operation of persons and loads lifts were followed as well as the standard "EN 81.1: Safety rules for the construction and installation of persons and loads lifts".

2. DRIVE MECHANISM

Drive mechanism will contrain an electric motor suitable for 380V/50Hz network connection and number of rounds less than 1500 rpm. Electric motor will be suitable for the expected use (Power margins and capability of 120 couplings per hour without superheat danger). During the starting phase, current will not exceed 350% of the normal operation current. Starting time (until normal speed) will be at least 4 s.

Revolutions reducer will consist of a special steel worm gear with surface hardening and a helical gear-wheel. The connection of the reducer with the electric motor must be made by a cast-iron dimer bond without intervention of elastic rings and the whole construction should be easily inspected.

Lift brake system will automatically operate in case of interruption of current supply or circuit control current. Brake system will consist of two shoes separated from each other. Friction surface will be sheathed with special material. For the operation of the brake an electromagnet or an auxiliary low noise level electromagnet will be used. There will also be a hand wheel for the opening of the shoes and the hoist motion.

Friction pulley will be from fine quality steel with wire ropes holder channels, processed with great accuracy so that different speed motion of the wire ropes, their slide and excessive wear are avoided.

Drive mechanism installation will be on an appropriate armoured concrete base, while between the base and the mechanism a non-vibrational material will be inserted so that vibrations are not transferred to the building.

3. WELL EQUIPMENT

Well equipment will consist of the aligner bars (guide rails), the suspension wire ropes, the counterweight, the frame and the car and floors doors.

Aligner bars will be used as guides for the car and counterweight motion. They will be constructed of special fine quality steel and will have armoured slide surface. They will be connected with special plates, clamps and connection screws. Suspension of guide rails will be on top with special brackets and their lower end will be free to take contractions and expansions.

Suspension wire ropes of the car and the counterweight will be of the same quality, diameter and type. In their end there will be a firm and safe so that full consolidation of wire ropes is ensured. Cones of their ends will be uniform and the wire ropes length must be equal in every position of the car and the counterweight. Counterweight will move in the special track of the lift's well, which will be protected by a removable steel grid. It will be moved in fixed guide rails of «T» cross section with reinforced and carefully processed sliding surface with the help of blocks and automatically lubricated gaskets.

4. CAR EQUIPMENT

Frame of the car will be constructed from structural iron bars properly reinforced and welded to have inflexibility and no deformation danger will exist in case of operation of the safety device in the guide rails. The frame will have a safety gear device, a wire ropes suspension system and in its upper and lower part 4 complete blocks with sliding gaskets on the guide rails, will be placed. In the lower part of the frame, a rectangular frame of structural iron bars with proper weld will be implemented, on which the lift's car will be assembled.

External doors of the well will open outwards and will be constructed from knock durable sheet iron and will be firmed with strong hinges. They will have the necessary electric contacts and pre-hasp system with electromagnet or electric motor.

Car's floor will be constructed from DKP sheet iron of 2 mm thickness. Amiant sheet will be fixed on it. The front part of the floor in the entrance position will be covered with protective angular plate from tough aluminium. Side walls of the car will be constructed from DKP sheet iron of 2 mm thickness with double backfolding in the unification points for support and assurance of inflexibility.

Car's roof will have a manhole, opening outwards and up, while a 42 V socket and a protective fencing of 10 cm height will be placed on the car's roof. There will be openings to ensure the adequate ventilation of the car. The floor of the lift's car will be movable and will serve as a switch, which will pleclude the lift's motion with external command when it is loaded.

5. ELECTRICAL EQUIPMENT

The main distribution panel will be placed in the machine room near the entrance and will be accompanied with all necessary components. It will have a knife type switch, three slow melting fuses and a protection circuit breaker for the motor.

Lighting panel will be placed near the main panel with all necessary components. It will have a 220/42 transformer for the car lighting. Control panel will be placed in a closed metal box and will have all necessary instruments. Distribution disk of the stops will be moved by a wire rope and chain straight from the car. Control will have the proper contacts and all necessary glow lamps.

6. SAFETY DEVICE

Brake system will be fixed in the frame so that it acts simultaneously on the guide rails during the brake. Safety gear system must be of elastic brake and in case of break or sag of the wire rope or even in the case of exceeding the allowable speed limit by 40% it should automatically be set to operation.

Gear regulator will be placed above the car and will act on the safety gear in cases where descent speed of the car exceeds the allowable limits. In the lower end it will be connected with the necessary wire rope, the binder pulley and the counterweight. A switch will be placed in the linkage point of the suspension wire rope, which will shut off the control circuit when safety gear acts.

In the safety systems there will also be a drive finish switch system which will cut off the current if the car exceeds its drive end limits.

Outside the well two sound devices must be installed for the alarm signal of the respective car button.

Special pre-hasp contacts will be placed in the external doors, which will make the motion of the lift impossible if external doors are not closed. Furthermore, they will preclude the opening of the well door when the car moves or is not behind this door.

An overload checking equipment will preclude the car's motion if it is overloaded more than 5% of the expected limit.

In the lower part of the well, a hammer deposit system for the car and the counterweight will be installed, so that energy absorption from the system will allow stoping of the car with a deceleration less than the gravity's.

A sign will be placed inside the car and in a noticeable point, which will indicate the manufacturer, the production and installation series number, the expected load, the manufacture year, and the number of persons.

7. CHECK – MAINTENANCE

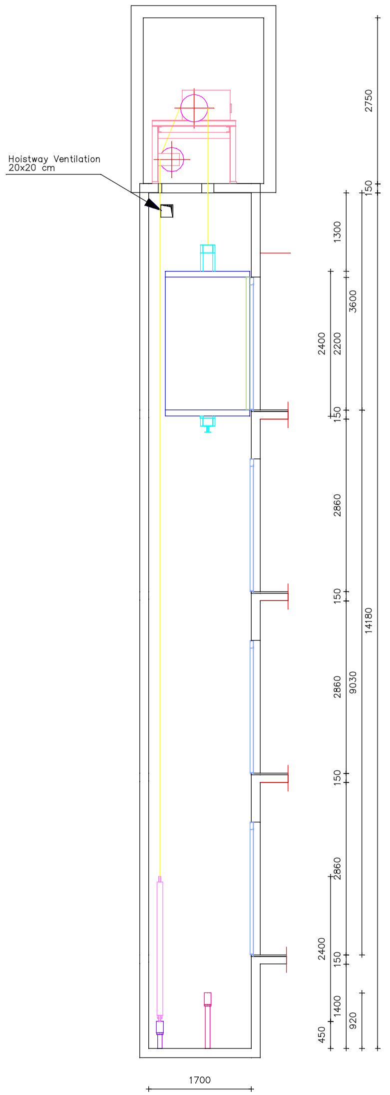
Check and acceptance tests will be performed by authorized personnel (EN81.1 section 16.1).

Lift will be subject to regular check and maintenance from a licensed person according to the regulations (EN 81.1 Appendix E. a). Every modification performed after the lift's distribution must be studied, decided and constructed only by qualified persons who will be registered in the technical part of the record or the lift's record (EN 81.1 section E.2).

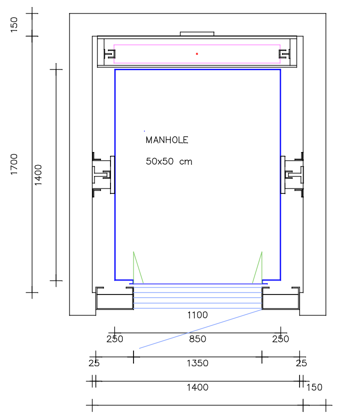
A record must necessarily exist, be constantly updated and it must contain technical and chronological data for every procedure concerning the installation or replacement of a lift's component (EN 81.1 section 16.2.).

Changes or modifications in everything mentioned above can be done only after the written authorization of the project manager.

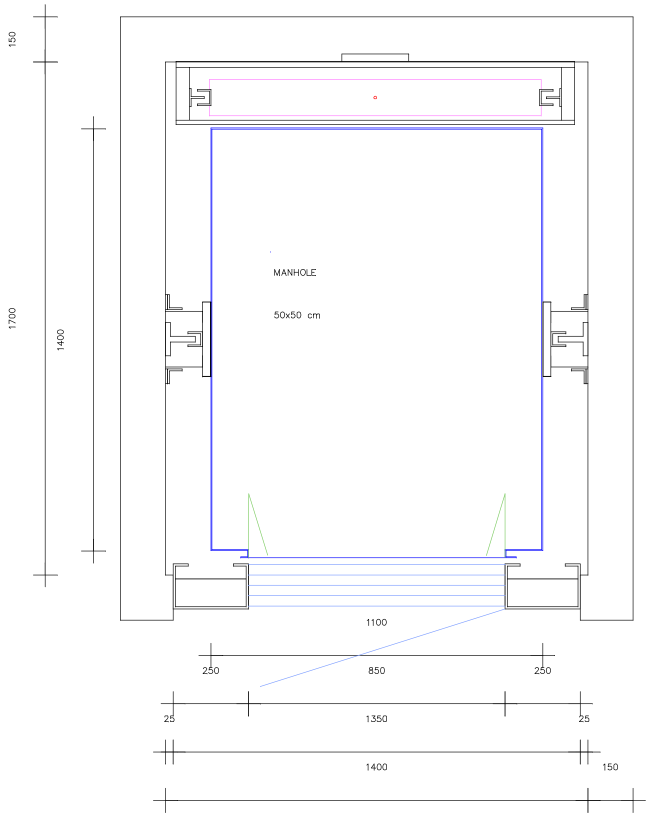
The Author



VERTICAL SECTION (SCALE 1:50)



PLAN HOISTWAY-CAR (SCALE 1:20)



PLAN HOISTWAY-CAR (SCALE 1:10)

LABEL	
LIFT TYPE	ELECTRIC
LIFT KIND	8 Passenger Lift
FLOOR NUMBER	4
PAYLOAD	600 Kp
SPEED	V= 0.63 m/s
ENGINE ROOM	[mhxan_pos]
WIRE ROPES	8 mm
GUIDE RAILS	T 50 x 50 x 9
MOTOR POWER	8.2 HP