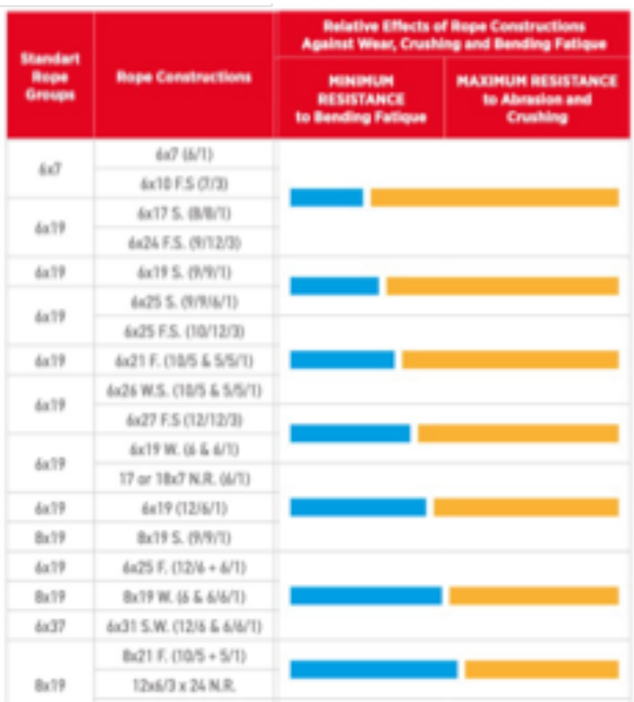
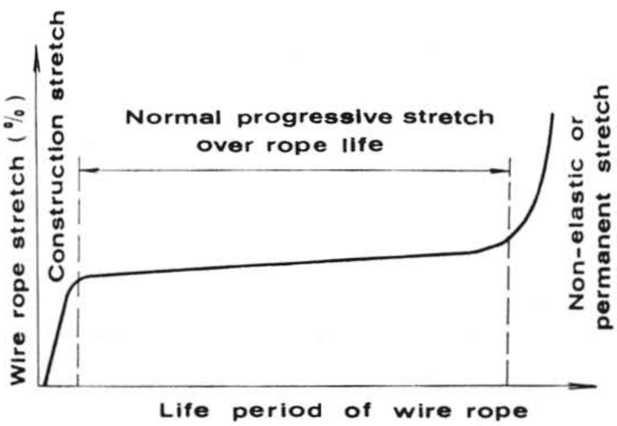
ANSWERS TO QUESTIONS ON REGULATOR ROPE CALCULATIONS

Question 1: Is it compulsory to calculate extension in regulator ropes? Are L% ratio < 1 used in the calculation and E=63000 N/mm² values applicable for all ropes?

The 6 mm fiber-cored ropes that were commonly used as regulator ropes in the past were an application method tested and calculated for lifts with a certain height, one-way safety gear and 15-20 kg weight at the center of the tension pulley. This common system that could be used for a long time without causing any problems for lower speeds, average traffic load, medium height and small tension weights should be re-evaluated following changes in the applications. Speeds have changed, well heights and traffic loads have been increased, and tension weights at the center of the tension pulley that affect the rope tripled or quadrupled with the use of two-way safety gears. In this case, all companies should control L% rope extension for main model elevator applications. The facts that rope safety calculations are made and the rope is within the safe values do not mean that L% extension calculation is suitable. Especially in high-rise buildings with large tension weights, L% calculation should be made depending on rope section. Substantial increase of breaking strength values of the rope fits well with the rope safety calculation, while the fact that the rope section and E value do not increase evenly may not fit well with the rope extension calculation, which may lead to deformation of regulator ropes within a short time.

Ropes are selected by taking the flexibility and wear resistance into consideration. Wear resistance decreases as flexibility increases. Similarly, L% flexible extension ratio decreases as the breaking strength increases and the rope gets thicker. Deformation starts when the rope exceeds its flexible extension area. Relevant graphics from the books Steel Wire Ropes Technical Information and Elevator Mechanical Design (Lubomir Janovski) are given below. As can be seen in the tables, L% extension ratios vary depending on the rope structure, loading ratio and service life.





It is clear that the conventional calculations and applications cannot be deemed valid if different speeds, different well heights, different traffic densities and different tension weights are used in each lift. If we rewrite the rope extension formula, the stress/stretching ratio gives the elasticity module.

$E = \sigma / \varepsilon$, $\sigma = E*\varepsilon$, $\varepsilon = L/L_0$, $\sigma = E*L/L_0$, $\sigma = F/A$, $F/A = E*L/L_0$

$L = (F*L_0)/(E*A)$

$L\% = (L/L_0)*100$

$L\% = (F_{max}.100)/(E.A)$

σ = Stress N/mm^2

L = Total extension length of rope (mm)

L_0 = Single-side rope length (mm)

L% = Rope extension percentage

ε = Stretching ratio

E = Elasticity module for steel ropes N/mm^2

$A = (\pi.d^2.x)/4$ mm^2 actual rope section area (in general, x = 0.49 for 6x19 ropes and x = 0.44 for 8x19 ropes)

For this calculation, we need to find F_{max} value. F_{max} value is composed of the total force (T_1) that is formed as a result of the rope weight and traction force which is created by the regulator tension weight on rope arm during normal use. The calculation is explained in the main article. In medium-height lifts with low acceleration, F_{max} is found by directly calculating the weights.

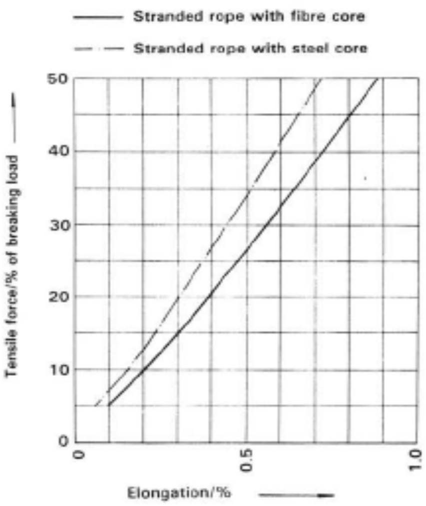
(F_{max}= T1)

During on-site observations, it is seen that rope deformation is more than expected in lifts with high acceleration and deceleration speeds, long travel distance, and medium or higher traffic load. In main suspension ropes, minimum 12x more security is desired, meaning that the rope is loaded around 8% as a maximum. In regulator ropes, the loading ratio can increase up to 12.5% due to 8x more security. High acceleration values in long rope lengths cause a significant additional increase in rope stress when overcoming idleness in the regulator and tension pulley. This additional loading affects rope load, and may load to exceeding the flexible area as rope extension. Therefore, it is recommended that k_2 coefficient is also taken into consideration in lifts with long travel distance and medium or higher traffic load. High acceleration value is an important factor for L_0 rope extension in long ropes. In these lifts, using k_2 coefficient in F_{max} calculation will help to extend the rope life (As can be seen in practice, a greater value can be taken for lifts with high acceleration and heavy traffic load).

$F_{max} = k_2*T1 = 1.2*T_1$ (k_2=1.2 impact coefficient in operating state EN 81-20; 5.7.4.4)

The L% value calculated according to this formula should, in general, be less than 1%. However, the L% value in the calculation given as an example in the article and (x) coefficient are generally accepted for concept projects in the beginning. For ropes used in application projects, actual values of both rope section ratio (x) coefficient and L% value that are suitable for the rope used should be taken into consideration. In EN 12385-5 standard, section ratios for each different rope type is given in the relevant chart. These values are given in the guidelines for the rope used specifically for the application.

Examples for L% values are also given in the abovementioned books. According to the values given in the book Elevator Mechanical Design (Lubomir Janovski), steel-cored ropes have a smaller extension percentage than fiber-cored ropes. The same book gives examples from a rope company for L% extensions in flexible area.



The company Gustav Kocks GmbH states the following elastic elongation for its ropes:

6 x 19 + FC	0.5%,
8 x 19 + FC	0.6%,
DRAKO 300T	0.4%,

all values at the load of 14% of the calculated breaking load.

E value also varies for each rope. 63000 N/mm^2 in the calculation is the value given for 6 mm 8*19 fiber-cored steel ropes that are generally used. <u>This is a value that can be used in concept projects for middle-height lifts.</u> These values vary for each rope type. Actual values to be used in application should be taken from the charts or rope brochures. Some exemplary values are given in the article "Lift Ropes".

Composition	E (N/mm^2) For %0-%20 Load	E (N/mm^2) For %21-%65 Load
6x7-FC	80.700	90'000
6x19-FC	75.000	83'000
6x37-FC	68.000	76'000
8x19-FC	56.000	62'000
6x19-IWRC	93.000	103'500
6x37-IWRC	85.000	96'500

Lubomir Janovski also gave the elasticity module values (E) for ropes in the book.

According to Schweizerische Seil-Industrie AG the modulus of elasticity of steel wire is 196 kN/mm^2 and varies with steel ropes:

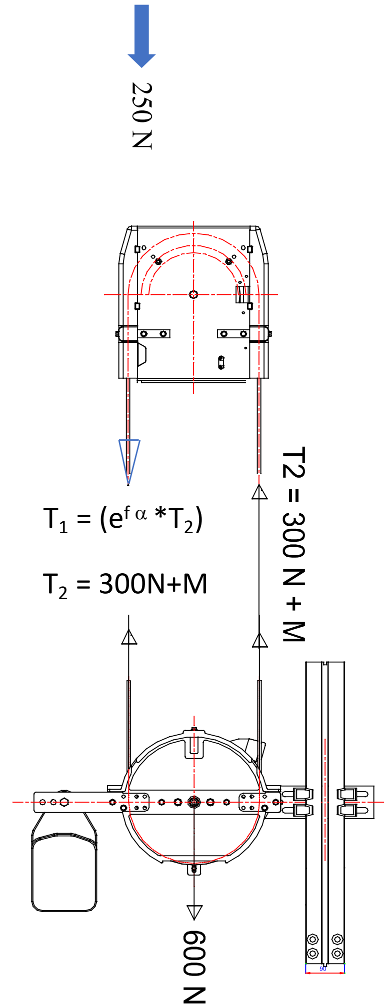
1.0 - 1.25 x 10^5 N/mm^2 for stranded wire ropes with steel core,
0.7 - 1.0 x 10^5 N/mm^2 for stranded wire ropes with fibre core.

As can be seen, F_{max} for each lift should be calculated according to the tension weight, regulator rope and well height by taking the actual E, A values into consideration, and compared to the L% value suitable for the rope. Although the rope safety coefficient is suitable, the rope can be deformed quickly if the extension does not fit well with the L% value. Therefore, L% calculation should be made at least once for each lift applied as a model.

Question 2: Should regulator traction sheaves be V-channel?

This is a conventional application method. Previously, only downward safety gears were used in lifts. Regulators were assembled to be effective only in downward locking. Regulator tension weights were designed for this purpose. Generally, weights that were suspended directly at the center of the pulley and systems that ensured rope tension with moving joint arm were used. ~~Idle~~ Tension pulleys that were not intended for traction and were therefore used as U-channels fulfilled the tensioning function with weights suspended directly at them, as seen in the adjacent pictures.

Considering that (T_2= half of the tensioning weight and rope weight M) will fulfill the tensioning function at the time of locking (T_1), $e^{f\alpha}$ value should be high in order that the regulator's gear arm traction force is greater than (T_2) at the time of locking ($T_1= T_2 * e^{f\alpha}$). (f) friction value should be high in order that $e^{f\alpha}$ coefficient is high when 3.14 is a constant radian equal for 180 degree of α angle. Narrow angled V-channels with the highest friction value were preferred to create a greater tractive power with smaller weights. Moreover, the same V angled pulley can be used for 6 mm, 8 mm and 10 mm ropes in V-channels, which is considered as an advantage (Pulley angles are restricted in the standard; and it is required to remain within these values).



Later, applications varied with the use of safety gears in two ways. In lifts that use upward safety gear as two-way safety gear in the car, a minimum tractive power of 300 N is required to operate the safety gear upwards. While the force T_1 acting downwards grows with $e^{f\alpha}$ coefficient ($T_1= T_2 * e^{f\alpha}$), only the force T_2 is effective upwards directly to pull the safety gear arm, as explained in the main article. In this case, in older aplications the T_2 force used to be around 100-150 N and with the help of the narrow V-channel angle, the T_1 force can increase to 500 N with a high $e_{f\alpha}$ coefficient, whereas in the bidirectional safety gear application, the T_2 force must be at least 300 N in order to be effective upwards. This has necessitated a counterweight pulling force of at least 62 kg at the center of the tension pulley.

This is achieved by either directly using a large weighted suspension device or by extending the moment arm by extending the knuckle arm. But the problem here is that the T_2 force itself is now three to four times larger than before. Some of the different practical problems that will arise as a result of this should be taken into account.

As a result of the growth of the T_2 force, the value of T_1 in the equation ($T_1= T_2* e^{f\alpha}$) will also grow a lot. This leads to the selection of a rope with a larger diameter due to the requirement that the minimum breaking force of the rope must be at least 8 times greater than the T_1 force. To avoid this, the $e^{f\alpha}$ coefficient must be reduced. Since the angle α is fixed, V-channels with large channel angles or U-channels with undercuts are used, which have a smaller (f) value. Sheaves of this kind give a far lower friction (f) values when compared to V-channels with narrow channel angles. However, the $e^{f\alpha}$ value must be greater than 2 (At least T_1-T_2>300 N at the moment of pulling the mechanism arm during braking).

The regulator sheaves channel shape and angles must be selected according to (f) friction value, out of complete necessity. Whether it is a V or U channel and channel angles are determined entirely by calculations, factors such as rope holding, slippage, minimum breaking force must be taken into account. A rope that is not sufficiently safe in a narrow angle V channel may be sufficiently safe in a wide angle V-channel or U-channel. As the weight tensions increase, it would be correct to recalculate and reconfigure the channel angles accordingly. Care should be taken to use the regulator and the appropriate tensioning systems for which these calculations are made together.

Question 3: Is 300 N traction force sufficient for the regulator.

The 300 N tractive power value is the lower value given by the standard, but if you read the article carefully, it says that the main point is that the pull force of the regulator must be at least twice the brake tractive force used. To use a regulator with a 300 N traction force, you need to use a brake with a maximum traction force of 150 N. This is not a very common value for bi-directional brakes. If 250 N traction force is taken into consideration in a brake that can be used in an average medium capacity lift, it would be more accurate to make the regulators to be used for these lifts according to an average traction force of 500 N. At high rated loads, special care must be taken to ensure that the traction force of the regulator and brake are compatible with each other.

Question 4: Can the regulator rope thickness be less than 6 mm?

In the previous standard, EN 81-1+A3, it was stated that regulator rope should be steel rope and that its thickness should be at least 6 mm in the articles related to speed regulator rope.

"EN 81-1+A3; ***9.9.6 Overspeed governor ropes***
9.9.6.1 *The overspeed governor shall be driven by a wire rope designed for that purpose.*

9.9.6.3 *The nominal rope diameter shall be at least 6 mm."*
However, these articles have changed in the new standard that replaced it and we still use which is TS EN 81-20 standard. The first term is that instead of steel rope, the rope which is in conformity with TS EN 12385-5 standard was put into use and 6 mm restriction was removed.

"EN 81/20; ***5.6.2.2.1.3 Overspeed governor ropes***
The rope of an overspeed governor shall satisfy the following conditions:
a) the overspeed governor shall be driven by a wire rope as specified in EN 12385-5."

In EN 12385-5 lift rope standard, regulator ropes can be chosen as suspension ropes among the Chart 6 “6*19 ropes” and Chart 7 “8*19 ropes” as the well as thicker ropes in Chart 9 and Chart 10. Attention must be paid to choose suspension ropes from among Chart 6, Chart 7 and Chart 8. The thinnest rope to be used is given as 6 mm in these charts. For regulators, 6 mm rope restriction was removed from the standard but it is given based on to the other standard. There is not a rope description thinner than 6 mm in EN 12385-5 standard.

The same lift can create different conditions as in the situation of suspension ropes. TS EN 81-20 standard states that suspension ropes to be used in lifts based on the TS EN 12385-5 standard cannot be thinner than 8 mm. Nevertheless, ropes with different properties are certificated **as lift suspension ropes** after being tested by Notified Bodies and obtaining the approval of “can be used as lift suspension rope” and the ropes thinner than 8 mm which are 6.5 mm can be used as lift suspension ropes. Ropes thinner than 6 mm which are tested by the Notified Bodies and obtained the approval of “can be used as lift suspension rope” and “certificated as lift suspension trope” can also be used as regulator ropes in the same way. However, they have to ensure the relevant strength restriction of Article 5.6.22.2.1.3.

“TS EN 81/20 ***5.6.2.2.1.3 Overspeed governor ropes***
The rope of an overspeed governor shall satisfy the following conditions:
a) the overspeed governor shall be driven by a wire rope as specified in EN 12385-5.
b) the minimum breaking load of the rope shall be related by a safety factor of at least 8 to the tensile force produced in the rope of the overspeed governor when tripped taking into account a friction factor μ_{max} equal to 0,2 for traction type overspeed governor.”

Studies related to this calculation was published in the article "Regulator Rope Choice and Calculation". The purpose of this article is not for the use of ropes thinner than 6 mm, but for the use of thicker ropes according to the calculation. A rope thinner than 6 mm ensuring the general terms of Article 5.6.2.2.1.3 and sufficient strength value and having the certificate of “lift rope” obtained from a Notified Body can be used in this situation, but cost of this rope will be higher than the normal 6 mm rope. A thinner steel rope with normal restriction and without and lift certificate cannot be used.

Another question that is asked is that “why the whole rope weight is taken in the calculation of regulator stretching”. Yes, in the calculation of the weight that has effect on the rope itself, it requires to take its half since the load is distributed equally but many tensioning weight pulls with a strength more than 600 N. The whole weight of the rope was taken to form the worst condition and facilitate the calculation. Moreover, the shakings during the movement are thus taken into account as well. However, it is required to avoid from solutions very close to the 8x safety coefficient. This part is not one of the parts that should be included in a fine cost calculation.

Question 5: Should regulator weight tension contact be a locked contact?

Answer: First, we should understand the operating principle of a lift. One of the primary properties that distinguish a lift from a lifting crane is the ensuring of the car’s impact security through a safety gear (parachute system) connected to the car. As long as the safety gear is not active, the car is not moved in any way. This rule also applies during the installation. Lift companies should write this term into the installation instructions and control it. Car frame should not be moved before the regulator and safety gear is activated. Many incidents are caused by the operations performed before ensuring this condition. Operation of the safety gear depends on the regulator; only if the regulator is active, it can control the safety gear. Operation of the whole system depends on if the regulator tension pulley keeps the regulator rope tense. If the rope is tense and hangs with a sufficient weight, only then the regulator can operate and activate the mechanic brake device. Thus, regulator tensioning system is among the most important components.

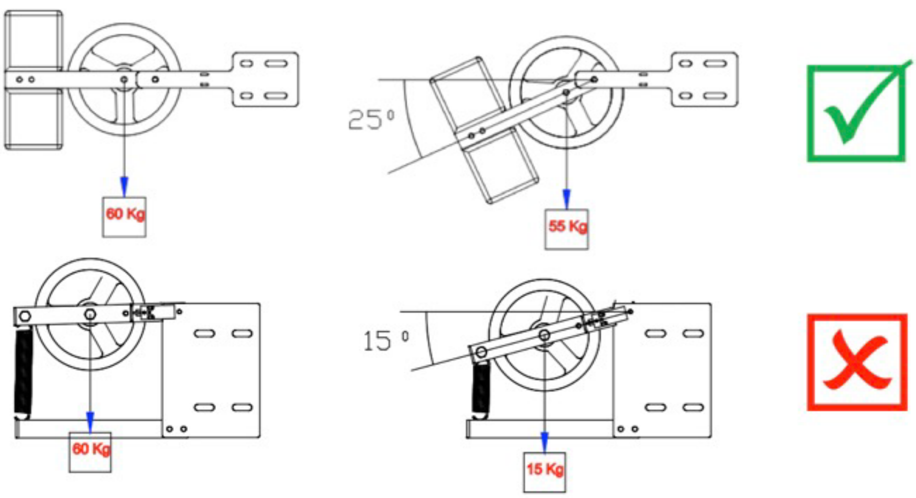
Control of whether the regulator rope is tense is executed with a safety contact. This contact must engage in when the regulator rope breaks off or stretches considerably and suspend the lift engine (*EN 81-20; 5.6.2.2.1.6 c) Electrical checking*). Since it is a machinery enabling the operation of safety system, it is a contact that is not short circuited in no case and is in the lead of safety chain. As it is known that although the regulator upper contact, safety gear contact, final limit switches, buffer contacts and door contact for leveling can be short circuited for electrical emergency retrieval operations, the regulator tensioning contact is not short circuited under any circumstances. Its safety circuit is stopped when it cannot operate, the lift stops and is not moved before adjusting that contact, because operation of the safety gear depends on whether the tensioning system is active. Short circuit should not be used by no means.

During braking, bouncing as well as stretch in the regulator tensioning weight due to the hitting or bouncing due to the movement may occur. Following this situation, if the rope is not in a state to operate, the lift should not be driven. However, if the rope turns back to its place again after bouncing, retrieval can be performed in the lift through electrical emergency retrieval, because the retrieval in synchronized engine systems commonly used today or high-capacity lifts can only be possible through electrical emergency retrieval. In case a locked contact is used in regulator weight, an electrical emergency retrieval cannot be made since the safety circuit will be switched off in the first place upon bouncing. Short circuiting this circuit will extremely be a false application since it is a circuit including the stop switches. The suggestion that “it is required to control this circuit before retrieval” is an abstract thought and does not apply for field practices. Who goes down 20 floors before every breakdown reporting, look at the contact and then climb up again and perform retrieval? This should be reflected upon. If a locked contact has to be used in tensioning system, then changing the form of tensioning equipment should be considered in this case. A separate contact that will control the rope stretching or breaking off and will be placed on top of the security circuit chain must be used; this contact must certainly engage in when the rope stretches or breaks off while the locked contact used in the joint should be displayed on the diagrams in a way not to prevent electrical emergency retrievals upon bouncing through connecting in the area where the manual retrieval is short-circuited (to the parachute contact, boundary cutters section). It is not correct to use a locked contact in the tensioning system without such an additional device.

Question 6: Can regulator tension systems be spring-loaded?

Answer: Regulators have been progressing to a very problematic state in tensioning systems recently. One of the responsibilities of tensioning system is to keep the rope tense but another important responsibility is to ensure at least a pulling of 300 N for the rope arms connected to the braking device in the systems performing especially the upward braking in the cabin. As a result of an approximate calculation, if there is a need for at least 30 kg suspension load on a rope arm, a suspension load of 60 kg at the center of the tension pulley since 30+30 kg load will be required on each arms. This load is the one that should be ensured in the position at a point where the tensioning contact will engage in upon the rope stretching, not while the tension pulley stays parallel to the ground, because the tension weight has to continue the tensioning capacity like this so long as the tension pulley does not switch of the circuit. Tensioning angle changes after the rope stretching in jointed arm weighted systems but we can still observe that a 90% part of the hung weight is still active since the Cos25 value give 0.90 value even in the angles of 25^0 after the stretching. Thus, it will be enough to keep the requested weight 10% more in the weighted systems for solving this problem. However, the situation differs for spring-loaded systems. So, this becomes more of an issue.

In the case of a spring-loaded system, the spring should ensure the tension weight needed for the rope arms in the last stretching point where the contact will engage. However, though the springs ensure this load when they stretch and the tension pulley arm is parallel to the ground in many spring-loaded system, the loads that will stretch the rope in the tension pulley rapidly decreases together with the closing of the springs when the rope stretching starts. After some time, a balance develops between the pulling force of the springs and stretching force of the rope while the stretching of the rope stops. However, the tension force of tension pulley decreases to 150 N value instead of staying as 600 N and thus a value which is far below the tension value that will enable braking as a tractive power occurs on the rope arms.

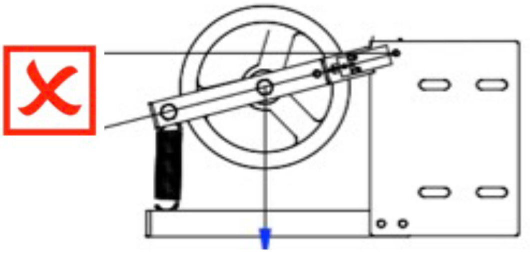


Even if the rope stretches in a weighted tensioning system, the weight changes are around 10%.

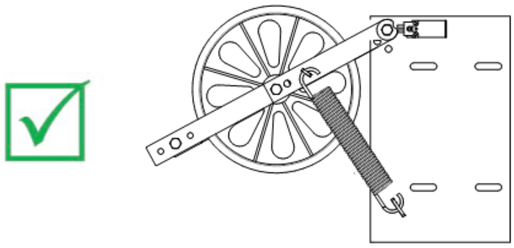
However, even a lesser rope stretching removes the tension weight in many spring-loaded system.

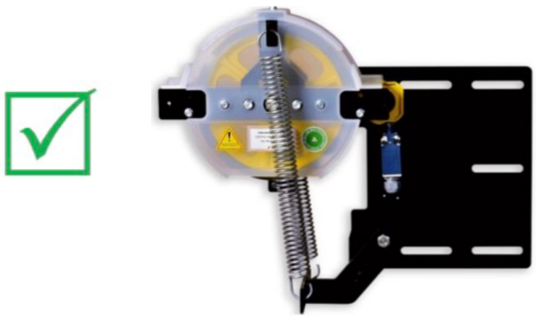
Since this balance situation develops especially after the stretching, spring-loaded tensions are started to be used widely by the service men who are freed from distress of shortening the regulator rope. The most deceptive point of it is that the regulator rope stays as if it is tense when the stretching of the rope reaches to the balance point and does not show any need for an intervention. While, in fact, the rope's tractive power that develops on the tension pulley has already lost the tension that can engage the brake. The service men are happy to be freed from the trouble of shortening the rope while the lift has already become far from being safe anymore.

Thus, the brakes are not activate especially upwards as well as downwards in each brake test for the lifts that use these systems. When everybody calls the brake manufacturers and asks the reason of it, the manufacturers recommend loosening the car brake mechanism's springs or brake springs. But then, since the lift switches to the brake at every sudden movement, the springs are restored after the control. Nobody elaborates on the fact that the regulator tensioning system does not actually work.



In this case, most of the brake and regulator systems of lifts that use short springs and in which a spring-loaded tension that lost its capacity to pull during the slightest rope stretching is used are in a state of shut down. The ones that use this system should constantly perform controls, otherwise they may experience troubles beyond measure.





It is required that the springs still develops min. 60 kg pulling force at the center of the regulator pulley in case that tension pulley reaches to the cutting point of regulator tension contact as a result of regulator rope stretching if the spring-loaded tensions are used. There is no objection to use tension spring systems that ensures this condition. Thus, it is required to use systems which are long-bowed and in which the pulley arm's tension margin stays within the powerful flexing margin of the spring and does not lose its force at the end of the tension margin. Otherwise, undesirable and insecure situations will occur.

In safety gears, the condition of being able to provide safety should be more important than the price. We need system with good operating conditions.

Best regards,
Serdar Tavaslıoğlu
Electr. Eng.