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Design and production of belt drive elevator traction machine: modelling of double side belt system

Kayış tahrikli asansör çekiş makinesinin tasarımı ve üretimi: çift taraflı kayış sisteminin modellenmesi

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Design and Production of Belt Drive Elevator Traction Machine: Modelling of Double Side Belt System

Highlights

- *Belt drive elevator traction motor design*
- *Double side belt system and belt calculations*
- *Verification of design by mechanical analyses*
- *Selection of the components*
- *Prototype production and hoistway placement*

Graphical Abstract

In this study, a gearless traction motor with double-sided belt connection was designed for elevator systems and a prototype was produced.

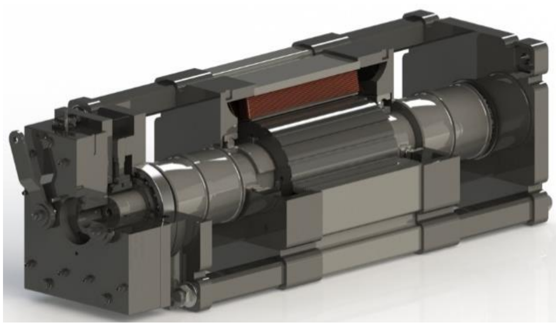

Figure. Sectional view of the elevator traction motor

Aim

It is aimed to design a belt drive elevator motor with a car speed of 1.6 m/s and a load capacity of 1175 kg and to select all components according to elevator standards.

Design & Methodology

3D modelling software was used during the design process and the strength of the motor under loads was verified using the finite element analysis (FEA).

Originality

A double side-belt drive elevator motor design, selection of components, materials and mechanical verification are presented for the first time in this article.

Findings

The motor shaft, covers, body and stator teeth are located in the safe working area according to the standards.

Conclusion

In this study, the selection and strength of the all components of the belt drive elevator machine were made according to the standards and the prototype production of the designed motor was carried out.

Declaration of Ethical Standards

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Design and Production of Belt Drive Elevator Traction Machine: Modelling of Double Side Belt System

Araştırma Makalesi / Research Article

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ABSTRACT

Elevator traction motors are critical components of elevator systems. After the structural analysis of the traction motor, it is necessary to proceed to the production processes for the comfortable and safe use of elevator systems. In this study, the static load values for bearing lifetime calculation, belt selection, and finite element analyses (FEA) were determined. According to calculated loads; materials, bearings, belt, pulley, brake, and encoder were selected. In order to obtain a design in accordance with elevator standards, motor parts were examined for structural analysis. Finite element analyses were carried out according to the determined maximum static load values, and a new belt elevator traction motor design that provides the safety coefficients has emerged. When the FEA analysis results of bending moments are examined; the maximum Von Mises stress acting on the motor frame and shaft was obtained 17.1 MPa and 27.62 MPa respectively. The maximum Von Mises stress was obtained under the torsional moment as 71.4 MPa and the shaft is 6.1 times safe against a torsional moment. This value is the minimum safety factor of the designed system and safety factors of bending moments are higher than this value. The designed elevator machine has got 12.9 kW rated power, 1.6 m/s cabin speed, and 1175 kg carrying capacity. The advantage of the proposed design is to eliminate the casting process with modular structure via double side belt system. The prototype of the designed motor was produced and it was observed that the motor provided the desired constraints as a result of the loading experiments.

Keywords: Elevator traction machine, gearless machine, belt drive elevator motor, finite element analysis, FEA, design calculation.

Kayış Tahrikli Asansör Çekiş Makinesinin Tasarımı ve Üretimi: Çift Taraflı Kayış Sisteminin Modellenmesi

ÖZ

Asansör tahrik motorları, asansör sistemlerinin kritik bileşenleridir. Asansör sistemlerinin konforlu ve güvenli kullanımı için, tahrik motorunun yapısal analizleri yapıldıktan sonra üretim süreçlerine geçilmesi gerekmektedir. Bu çalışmada rulman ömür hesabı, kayış seçimi ve sonlu eleman analizleri (SEA) için statik yük değerleri belirlenmiştir. Hesaplanan yüklere göre; malzemeler, rulmanlar, kayış, kasnak, fren ve enkoder seçilmiştir. Asansör standartlarına uygun bir tasarım elde edebilmek için motor parçaları yapısal analiz ile incelenmiştir. Belirlenen maksimum statik yük değerlerine göre sonlu elemanlar analizleri yapılmış ve emniyet katsayılarını sağlayan yeni bir kayışlı asansör tahrik motoru tasarımı ortaya çıkmıştır. Eğilme momentlerinin SEA analiz sonuçları incelendiğinde; motor gövdesine ve mile etki eden maksimum Von Mises gerilmesi sırasıyla 17,1 MPa ve 27,62 MPa olarak elde edilmiştir. Burulma momenti altında maksimum Von Mises gerilmesi 71,4 MPa olarak elde edilmiş olup mil burulma momentine karşı 6,1 kat emniyetlidir. Bu değer, tasarlanan sistemin minimum emniyet faktörüdür ve eğilme momentlerinin emniyet katsayıları bu değerden yüksektir. Tasarlanan asansör makinesi 12,9 kW anma gücüne, 1,6 m/s kabin hızına ve 1175 kg taşıma kapasitesine sahiptir. Önerilen tasarımın avantajı, çift taraflı kayış sistemi ile modüler yapı ile döküm işlemini ortadan kaldırmasıdır. Tasarlanan motorun prototipi üretilmiş ve yapılan yükleme deneyleri sonucunda motorun istenilen kısıtlamaları sağladığı görülmüştür.

Anahtar Kelimeler: Asansör tahrik makinesi, dişlisiz makine, kayış tahrikli asansör motoru, sonlu elemanlar analizi, SEA, tasarım hesabı.

1. INTRODUCTION

Elevators are designed to transport passengers or other goods from their starting floors to their destinations safely, comfortably and efficiently. Due to the increase in the number of passengers and cargo loads in high-rise buildings, there was a need to increase cabin capacities. The capacity increase is a more suitable method since the installation of another elevator system will be costly [1]. Since increases in car load have a direct effect on the motor shaft and connection parts, they should be taken into account in the design and selection of the elevator motor.

Today, there are four main types of elevators. These are electric, hydraulic, pneumatic, and maglev elevators. The

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most commonly used elevator systems are electrically driven geared or gearless elevator systems [2]. These elevators can have rope or belt traction. Rope traction elevators have ropes running over a pulley connected to an electric motor above the shaft. The main function of the ropes is to raise and lower the elevator car. A gear traction elevator consists of a motor with a gearbox attached to it. The main function of gears is to provide power to the pulley that moves the ropes. A gearless traction elevator does not have a gear system for speed regulation. Motor efficiency has also increased with gearless elevator motors with direct drive. These motors need high torque at low speeds. In gearless elevator motors, elevator car and counterweight are moved by ropes or belts attached to pulleys. In order to prevent fracture in steel ropes, a ratio of 40:1 between the drive pulley and the rope diameter is mandatory [3]. Today, pulleys with a minimum diameter of 240 mm are used in the acceptable risk range in rope elevator motors. However, it has been observed that some companies reduce pulley diameters by up to 210 mm by using specially certified ropes [4, 5]. In order to move the same loads with low torques at high speeds, pulley diameters need to be further reduced. For this purpose, belt mechanisms have been used instead of rope mechanisms in recent years [6]. With the commercial use of belts made of steel and rubber, fracture defaults in belt mechanisms have been eliminated and pulley diameters have been reduced [7]. In this way, motor sizes with similar rated loads have been reduced, and smoother and quieter travel has been achieved [2].

Different types of electric motors are used for traction in elevator systems. These can be listed as induction motors, permanent magnet synchronous motors, linear motors or reluctance motors [8-12]. Passenger comfort, motor efficiency and production costs are the main optimization parameters for the selection and design of these motors [13-20]. For this purpose, magnet geometry, stator winding, air gap, slot-pole ratio, stator tooth structures etc. are generally optimized. In addition, design improvements and weight reductions have been investigated for the frame, rails, door assembly and hoistway equipments [20-23]. Different suspension ratios are used in the traction system to balance the load according to the motor speed, and the cabin speeds and loads are balanced according to the design criteria.

Today, elevator machines with a car speed of 1.6 and a carrying capacity of 1175 kg are widely used in medium-height buildings. According to these parameters, the number of counterweights and belts according to the carrying capacity of the motor has been determined for the first time in this publication. The weights of the parts that are not limited by standards such as cabin weight, suspension weights and counterweight frame are determined according to the design of the hoistway. Mass production of the designed motor will be possible only with laser, milling and lathe processes without the need of casting process. So the major difference of the proposed design from other products in the industry is to eliminate the casting process with modular structure via double side belt system.

In this study, a permanent magnet synchronous motor with a power of 12.9 kW, a cabin speed of 1.6 m/s and, a carrying capacity of 1175 kg (16 people) in the IE4 efficiency class has been designed as the belt drive elevator traction system. Structural analyses were carried out with FEA in order to demonstrate that the motor is safe according to its carrying capacity. In addition, the materials of the motor such as bearings, pulleys, belt, brake and encoder were selected and then prototype production was carried out. It has been proven that the designed and manufactured belt driven elevator motor can be used safely in elevator systems.

2. MACHINE DESIGN

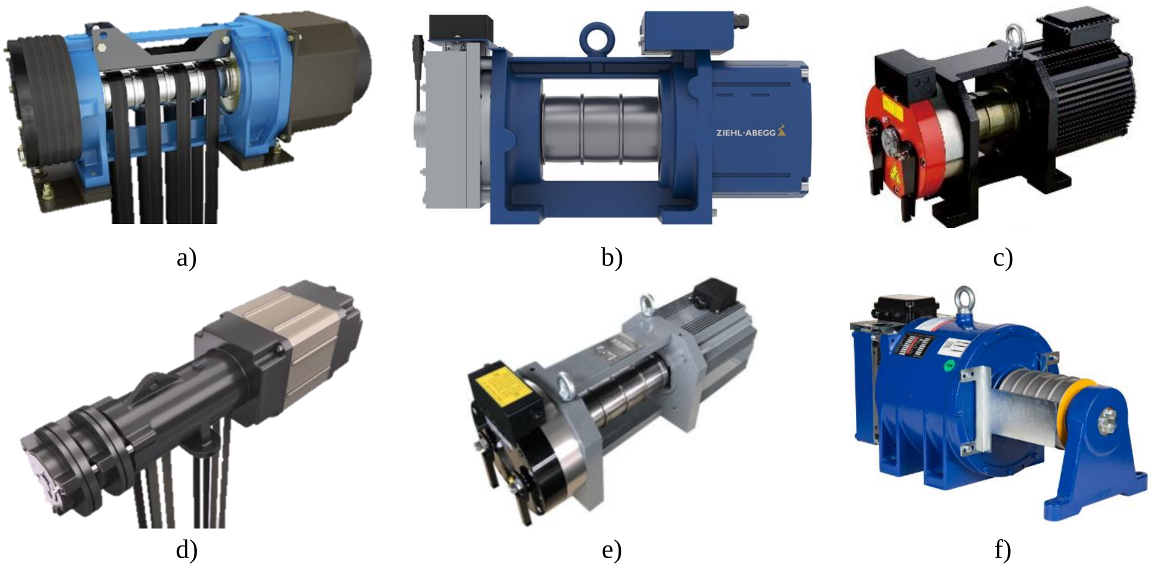


Figure 1. Belt Drive Elevator Machines: a) Otis-Gen2 [24], b) Ziehl-Abegg ZAtopx [25], c) Faxi 100A [26], d) Schindler 3300 [27], e) Kisa KA100 [28], f) Akış S200 K2 [29]

2.1.Modelling of Double Side Belt System

In the field of belt traction elevator motors, companies have different machine designs. In most of these designs, belt pulleys are located on one side of the motor, while in a few designs they are located on the right and left side. Belt elevator traction machines with different structures are shown in Figure 1. Only the Schindler 3300 model has the belts running from the right and left sides of the center of these designs. The advantage of the design in this study over the Schindler 3300 model is that it eliminates the casting process and has a modular structure.

Considering the motor static and dynamic loads, the elevator traction machine should be placed in the balance center in the hoistway. For that purpose, the electric motor is placed in the middle of the traction system, and the belt pulleys attached to the shaft are positioned to be on the right and left sides of the electric motor. Belt drive elevator motor design is seen in Figure 2. The advantages of the belt drive elevator motor design are listed below:

- Belt elevator motor will be placed in the balance center in the hoistway and thus uncomfortable travels caused by the balance center in roped gearless elevator motors will be prevented.
- Elevator loads accumulated on the motor are transmitted to the building and the ground in a smooth and balanced manner.
- There will be no need for a cast body in the production of the proposed belt elevator motor, and a lighter and more compact motor will be produced.
- Compared to rope systems of the same power, a lower volume and lower weight, more efficient and more comfortable elevator traction system has been designed.

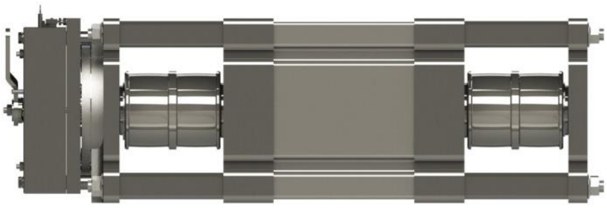


Figure 2. Designed double side belt drive elevator traction machine

Before starting the structural analysis, static loads should be determined by calculating the weights such as cabin, suspension, counterweight, frame weight, belt weights. According to the determined static loads, the mechanical parts of the motor, bearing, material, belt, pulley, brake and encoder selection were made.

2.2. Selection of Bearings According to Lifetime Calculations

The loads required to calculate the bearings' lifetime in the elevator traction motor are given in Table 1. Static loads used in bearings' lifetime calculations and loads used in mechanical strength calculations are different from each other. All static load calculations were made according to the 2:1 elevator suspension system in Figure 3. The expression 2:1 means that 1/2 of the load on the motor in elevators is transferred to the building with the suspension design. Considering that the belts flow from both sides of the motor, the loads on the bearings are divided into two part: the machine cabin side ($\boldsymbol{T_{cs}}$) and the machine counterweight side (T_{cw}). In addition, O_{wb} value which is the total force of the shaft, rotor, pulleys and belt weights also affects the bearings. There are pulleys on the right-left side of the designed elevator machine and bearings on both sides of these pulleys. The total loads on the bearings (T_{sl}) are the sum of T_{cs}, T_{cw} and O_{wb} values. For the lifetime calculation, the loads per bearing are calculated with the equations in (1-6). As a result, the load acting per single bearing was obtained as 5381.88 N.

Table 1. Static load values for bearing lifetime calculation

Notations	Value	Unit
Nominal Load (Q)	1175	kg
Suspension weight (W_s)	185	kg
Frame weight (W_f)	185	kg
Cabin weight (W_c)	1000	kg
Machine cabin side weight (M_{cs})	2360	kg
Machine counterweight side weight (M_{cw})	1772.5	kg
Other loads affecting the bearing (O_{wb})	1257.64	N
Total load on the machine counterweight side (T_{cw})	8694.11	N
Total load on the machine cabin side (T_{cs})	11575.8	N
Total static load affecting to the bearings (T_{sl})	21527.55	N
Load per bearing (for life calculation) (L_{pb})	5381.88	N
Gravity (g)	9.81	m/s^2

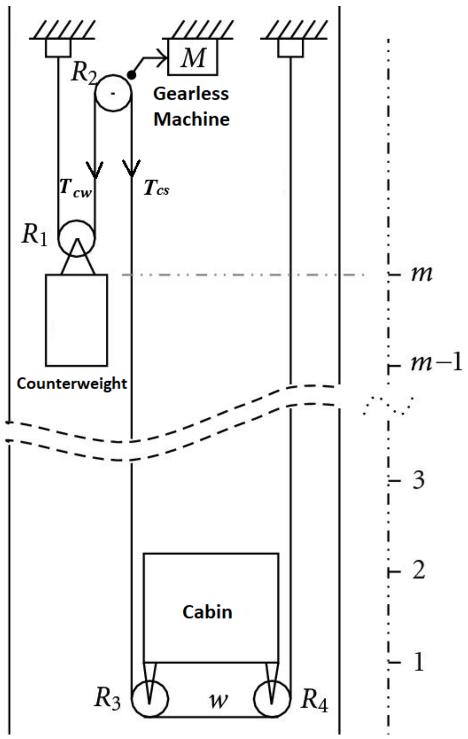


Figure 3. 2:1 elevator suspension system [30]

$$M_{cs} = Q + W_c + W_f \quad (1)$$

$$M_{cw} = Q/2 + W_c + W_s \quad (2)$$

$$T_{cs} = \frac{M_{cs}}{2} \text{ x } g \quad (3)$$

$$T_{cw} = \frac{M_{cw}}{2} \text{ x } g \quad (4)$$

$$T_{sl} = T_{cs} + T_{cw} + O_{wb} \quad (5)$$

$$L_{pb} = \frac{T_{sl}}{4} \quad (6)$$

As seen in Figure 4, four bearings are used in the designed elevator motor and the static load is equally distributed on these bearings. Bearing selection was made according to lifetime value L_h which given in (7). Here, F is the equivalent load, C is the dynamic load number, n is the rotation speed (rpm) and p is the lifetime coefficient. SKF 6212 and SKF 6215 type bearings were selected for both sides of the pulley according to the loads per bearing and the motor cover thickness. From the catalog data, the C value of 6212 and 6215 type bearings is taken as 55.3 kN and 68.9 kN respectively. The p value is taken as 3 for ball bearings. The rotation speed of the motor (n) is 615 rpm.

$$L_h = \frac{10^6}{60xn}\left(\frac{C}{F}\right)^p \quad (7)$$

As a result of the calculations made according to (7), the lifetimes for 6212 and 6215 type bearings are 29430 and 56952 hours, respectively. The designed elevator motor has a duty cycle of S5 and the ratio of the loaded operation time to the total cycle time is 40%. Considering that the elevator motor will operate at full load for 6 hours per day, 6212 and 6215 type bearings can be used without maintenance for approximately 13.6 and 26.3 years, respectively. These values are the results that occur in the case of continuous full-load operation. In practice, the elevator will not always operate at full load. So the bearing lifetimes are expected to be longer.

2.3. Material Selection

Before proceeding to the analysis of the elevator traction motor, the materials used in different types of traction systems were examined. Since all loads will be affected on the shaft, 42CrMo4 (~AISI 4140), which is a quenched and tempered steel, was chosen as the shaft material. The forces are acting to covers and connection parts vertically, so the material of these parts was selected as AISI 1045, which is a medium tensile steel. The properties of these materials are given in Table 2 [31]. The strength of the selected materials has been verified by static analysis.

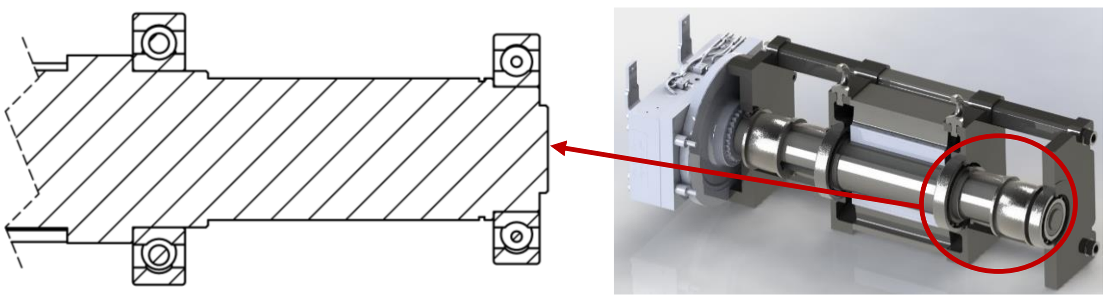
Table 2. Properties of selected materials

Properties	**42CrMo4**	**AISI 1045**	**Unit**
Elastic modulus	210000.0031	205000	N/mm^2
Poisson's ratio	0.28	0.29	N/A
Shear Modulus	79000	80000	N/mm^2
Mass Density	7800	7850	N/m^3
Tensile strength	1000	625	N/mm^2
Yield strength	750	530	N/mm^2
Thermal Expansion coeff.	1.1e-05	1.15e-05	1/K
Thermal conductivity	14	49.8	W/(m.K)
Specific heat	440	486	J/(kg.K)

2.4. Selection of Belt and Pulley

According to IEC 60034-2-1:2014, motors must be loaded at 125% nominal load maximum at the tests [32]. Belt and pulley dimensions are determined by taking into account the loads on them. The maximum force that the belts can encounter at 125% of their nominal load is calculated by F_{bmax} in (8). The W_b value in the equation represents the belt weight on the side where the maximum force occurs and is taken as 20 kg. The number of cords (N) and radius (d), belt thickness and MBL value of the belts have a critical role at the selection stages. Minimum breaking load (MBL) represents the minimum load that can be applied to a belt before it breaks [33]. Since the designed elevator machine has four different belts, the safety factor S_f calculated for a single belt is expressed by (9). According to the EN 81-20 norm, the safety factor in the designed system should be at least 12 [34]. After the calculations, it was determined that the MBL value of the flat belt to be selected should be at least 36.91 kN. Therefore, the belt selection was made as BRUbelt40 kN from Table 3. According to the selected belts, the safety coefficient was obtained as 12.11.

$$F_{bmax} = \left(\frac{W_c + 1.25 \text{ x } Q}{2}\right) \text{x } g + W_b \text{ x } g \quad (8)$$



a) b)

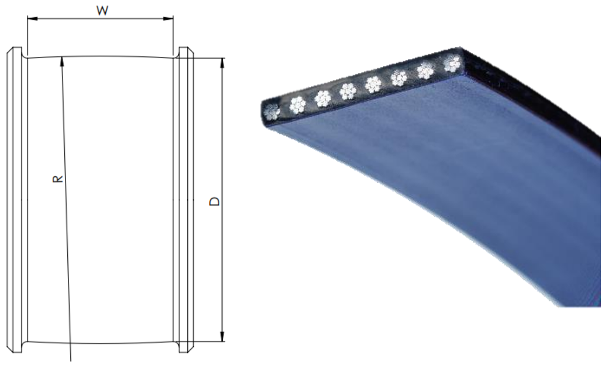
Figure 4. a) Shaft cross section between bearings, b) Section view of designed motor

$$S_f = \frac{MBL}{(F_{bmax}/4)} \tag{9}$$

Table 3. Features of the belts [33]

Product name (BRUbelt)	**Min. Breaking Load (kN)**	**Cross-section (*mm x mm*)**	**Number of steel cords**	**Belt weight (*g/m*)**
3.6 kN	3,6	8 x 2.3	4	30
7.5 kN	7,5	16 x 2.3	8	60
9 kN	9	25 x 2.3	10	90
15 kN	15	33 x 2.3	16	125
40 kN	40	25 x 3.4	8	200
50 kN	50	31 x 3.4	10	250
7 0kN	70	44 x 3.4	14	350
109 kN	109	70 x 3.3	24	549
126 kN	126	100 x 3.0	42	691
218 kN	218	150 x 3.3	48	1136

According to the calculations between the rated motor speed and cabin speeds, the pulley diameter *D* was chosen as 100 mm. The belts can be eccentric during assembly and the belt can move left and right on the pulley during the operation of the elevator motor. To eliminate these problems, the pulley width is made wider than the belt width. This width to be added can be taken as 30 mm. Thus, the pulley groove width *w* becomes 55 mm in total with a belt width of 25 mm and a safe belt tracking width of 30 mm. Crown radius *R* is formed on the surface of the pulleys so that the belts can move better. For a pulley diameter of 100 mm, the *R* value can be taken as 450 mm. A radius edge is formed on the pulley edges to prevent the belts from being cut and surge out of the pulley. Also the surface of the pulleys must have roughness R_a of 2-3 µm for optimal traction performance. In Figure 5, the technical drawing of a pulley to be used in the elevator machine and section view of a belt with 8 cords are given.

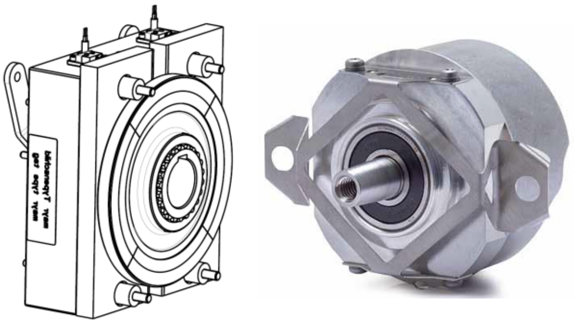


a) b)

Figure 5. a) Drawing of pulley, b) Elevator belt with 8 cords

2.5. Brake and Encoder Selection

According to EN 81-20 norm; electrochemical brake must be capable of stopping the machine when the car is travelling downward at rated speed and with the rated load plus 25 % [34]. As the brake of the elevator motor with a nominal torque of 200 Nm, the Mayr Roba-twinstop size-180 type brake, which has the ability to brake loads of at least 2x200 Nm and can brake axially, is chosen. The selected brake is a customized design for 2x200 Nm and works with a voltage of 207 VDC [35]. The brake hub has a circular space for encoder connection. The motor encoder was chosen as Heidenhain ECN 1313, which is a frequently used encoder in elevator systems. Heidenhain ECN 1313 is a 12-pin rotary encoder with 2048 pulse/rotation resolution used in motor control [36]. Selected electromechanical brake and encoder are shown in Figure 6.



a) b)

Figure 6. a) Electromechanical brake, b) Encoder

3. DESIGN and MECHANICAL ANALYSES

The bearing and material selections for the elevator traction system has been made. During the operation of the elevator motor, it is necessary to measure the resistance of the machine to maximum loads. The total force F_s, where the cabin is at 125% load and acting on the machine parts, is given by (10). F_s is similar to T_{sl} in bearing calculation, the difference is that 1.25 times the nominal load value is taken in this equation. The reason for this is that the deformation analyses should be performed against overloads. However, it is sufficient to consider the nominal loads in the calculation of the bearing life. In addition, the total force value in this equation was multiplied by the dynamic load coefficient while performing the structural analysis. When the values are substituted in (10), the F_s value is obtained as 22.96 kN. In structural analyses, the total force acting on the motor is implemented as 46 kN (Dynamic load factor: 2).

$$F_s = \left(\frac{1.25 \text{ x } Q + W_c + W_f}{2} + \frac{Q/2 + W_c + W_s}{2}\right) \text{ x } g + O_{wb} \tag{10}$$

3.1. Design, Sizing and Hoistway Layout

While determining the motor design; a design that is easy to manufacture, whose mass production costs can compete with rival products, and which meets the loads in a balanced way has been achieved. Figure 7 shows the final design of the elevator machine. The general dimensioning of the motor whose 3D design is completed is available in Figure 8.a. The motor has a weight of 164.5 kg plus an additional 25 kg of brake weight. Thus, the total weight of the elevator machine including the brake is expected to be 189.5 kg. Its length is 797.68 mm, its width is 250 mm and its height is 250 mm. The shaft design was made according to the motor design and the motor was centered in the middle of the right and left belt connections. The placement of the motor in the hoistway is shown in Figure 8.b.

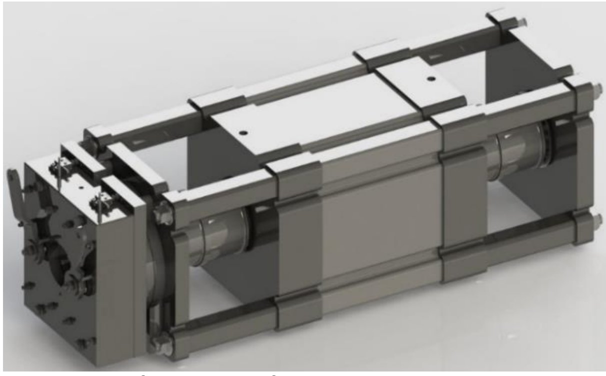


Figure 7. 3D final design of the elevator machine

3.2. Mechanical Analyses

In order to clarify the final design of the elevator traction motor, the strength of the motor parts under load and the amount of deformation need to be measured. For this purpose, FEA were conducted on motor parts. In low-speed rotating motors, the rotational speed is much lower than the critical speed and rotor inertia is accordingly low. Therefore, there is no problem due to dynamic effects in electric motors rotating at low speeds [37]. Since the rated speed of the designed motor is as low as 615 rpm and the produced motor will not operate at variable speeds, it was not necessary to carry out dynamic analyses. The stresses affecting to motor parts are shown in Table 4.

Table 4. The stresses affecting to motor parts

Motor part	**Impact type**
Motor frame	Bending
Motor shaft	Bending + Torsion
Stator teeth	Torsion

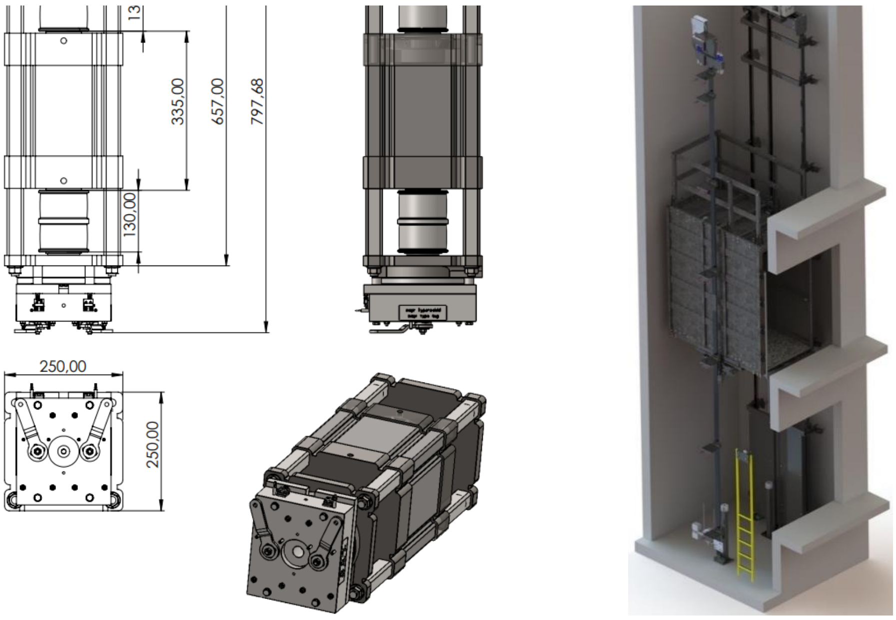
The strength of the parts under these stresses, the amount of deformation and whether they remain within the yield limits were tested by FEA. Mesh properties of analyses are shown in Table 5. The most adequate mesh size was selected according to different analysis results which have various mesh dimensions. Mesh refinements were done until the results came stable. After we came a limit, analysis results did not vary significantly even when we refine our mesh. So the element size that gave a mesh independent solution was selected. In the design calculations, the factor of safety was taken as 5 [38].

3.2.1. Motor frame bending analysis

In the production of the motor prototype, AISI 1045 material, one of the medium carbon manufacturing steels, was used for the chassis elements. The fixing areas of the system are the connecting bolts under the covers, and the

Table 5. Mesh properties of analyses

Mesh type:	Motor frame	Shaft bending	Shaft torsion	Stator torsion
Mesh Quality:	High	High	High	High
Jacobian points:	16 points	16 points	16 points	16 points
Element Size:	4,86906 mm	2,30055 mm	3,38114 mm	4,14377 mm
Tolerance:	0,243453 mm	0,115027 mm	0,169057 mm	0,207189 mm
Total Nodes:	1282146	1376555	547712	744937
Total Elements:	831698	979591	384940	491522



a) b)

Figure 8. a) General dimensioning of the motor, b) Hoistway assembly

loading points are the bearings, which are the contact points of the shaft. The system has a modular structure and the chrome-plated grinded shafts located along the motor in 4 corners are connected by the nuts. Total of 46 kN of force was applied to the motor frame as a static load. Figure 9 shows the results of the bending analysis performed on the motor frame using SolidWorks software.

When the analysis results are examined; the maximum Von Mises stress acting on the motor frame under a force of 46 kN was obtained as 17.19 MPa. This value is well below the yield limit of the material, 530 MPa, and the material is 30.8 times safe under the applied load. The maximum displacement of the material is 5.1 µm, which is a very low value. Therefore, the motor body operates in the safe zone under maximum bending load. The motor frame safety coefficient was obtained at a high level. The places on the frame that can be revised are the motor covers and outer plates. However, there are bearings in the hubs of these elements and cover selection should be made according to the thickness of the bearings and the lifetime calculation. The minimum cover and plate thicknesses have already been determined according to the bearing thicknesses, and the inside of the cover has been milled to reduce weight.

3.2.2. Bending and torsion analysis of shaft

The elevator car, counterweight and passenger weights, which are connected to the belts on the pulley create a bending moment on the shaft. In addition, as the shaft forces the load to rotate, torsional moment occurs on it. Therefore, the reaction of the shaft when forced against both torsion and bending was tested by FEA. In the production of the prototype, 42CrMo4 material, which is a quenched and tempered steel, is predicted to be used for the shaft. For the bending stress, Figure 10 gives the mesh structure of the shaft, applied force, Von Mises stress results and displacement results.

When the analysis results are examined; the maximum Von Mises stress acting on the shaft under a force of 46 kN was obtained as 26.09 MPa. This value is well below the yield limit of the material, 750 MPa, and the material is 28.7 times safe under the applied load. The maximum displacement of the material is 3.8 µm, which is a very low value. As a result, when there is maximum bending stress, the shaft operates in the safe zone. A torsional torque of 200 Nm which is the maximum output torque of the motor is applied to the shaft.

The moment is applied to the rotor keyseat of the shaft and the fixings are made from the pulley keyway regions. The applied torsional moment analysis results are shown in Figure 11. The maximum Von Mises stress on the shaft was obtained as 71.4 MPa under the effect of applied torsional moment of 200 Nm. The material's shear yield strength is taken into consideration when the torsion effect will be analysed. This amount is roughly 0.58 times the yield strength for bending [39]. The material's shear yield limit in this instance is 435 MPa. Thus, the shaft is 6.1 times safe against a torsional moment. The maximum displacement in the shaft was determined 20

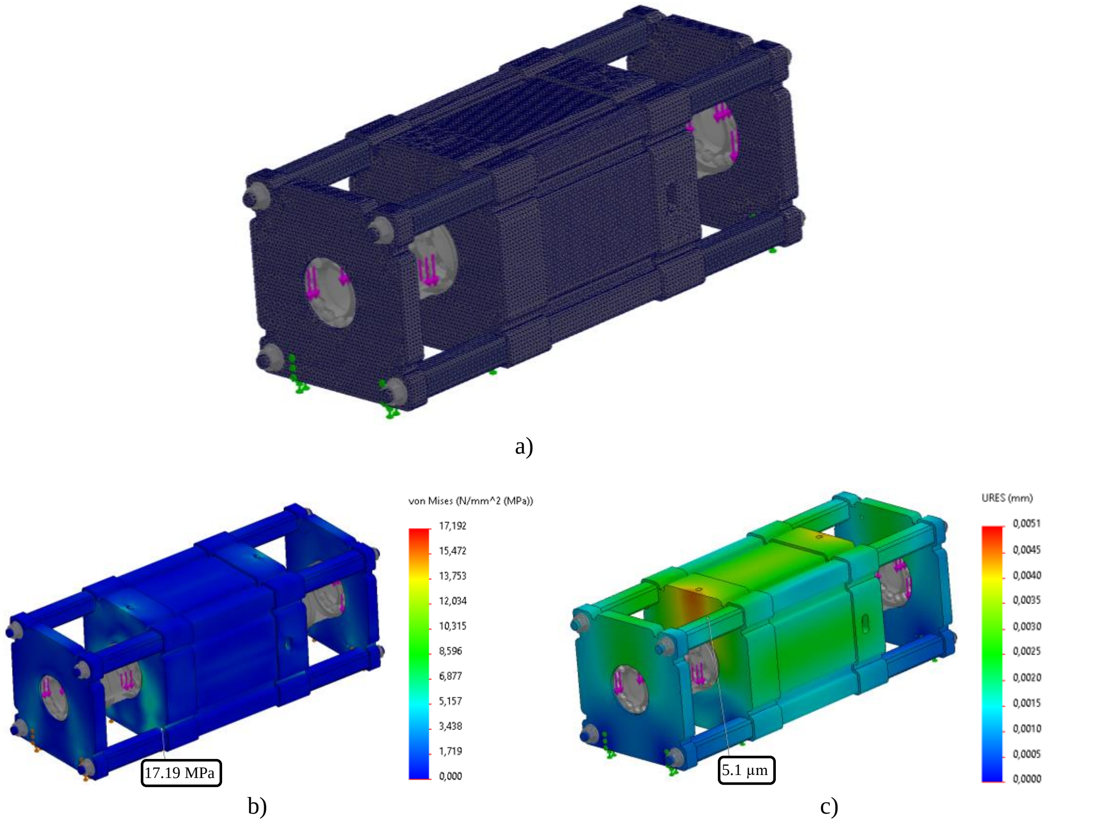


Figure 9. Motor frame bending analysis a) Applied force and mesh structure, b) Von Mises stress analysis results (N/mm^2), c) Displacement analysis results (mm)

μm at the shaft's end points. As a result, when the maximum output torque is applied, the shaft operates in the safe zone. Although the shaft has a high safety factor in bending analyses, it is 6.1 times safe under the effect of torsion. In this case, no additional reduction was required in the shaft design.

3.2.3. Torsion Analysis of stator teeth

To determine the deformations that may occur in the stator teeth, a torque of 200 Nm was applied. JFE Steel 50JN350 (0.5 mm sheet equivalent of M350) was used as the stator sheet material in the motor analysis, and the yield strength of this material is 389 MPa. According to analysis results, the maximum Von Mises stress was obtained as 1.31 MPa. It was observed that the torsion and displacement in the stator teeth were very low.

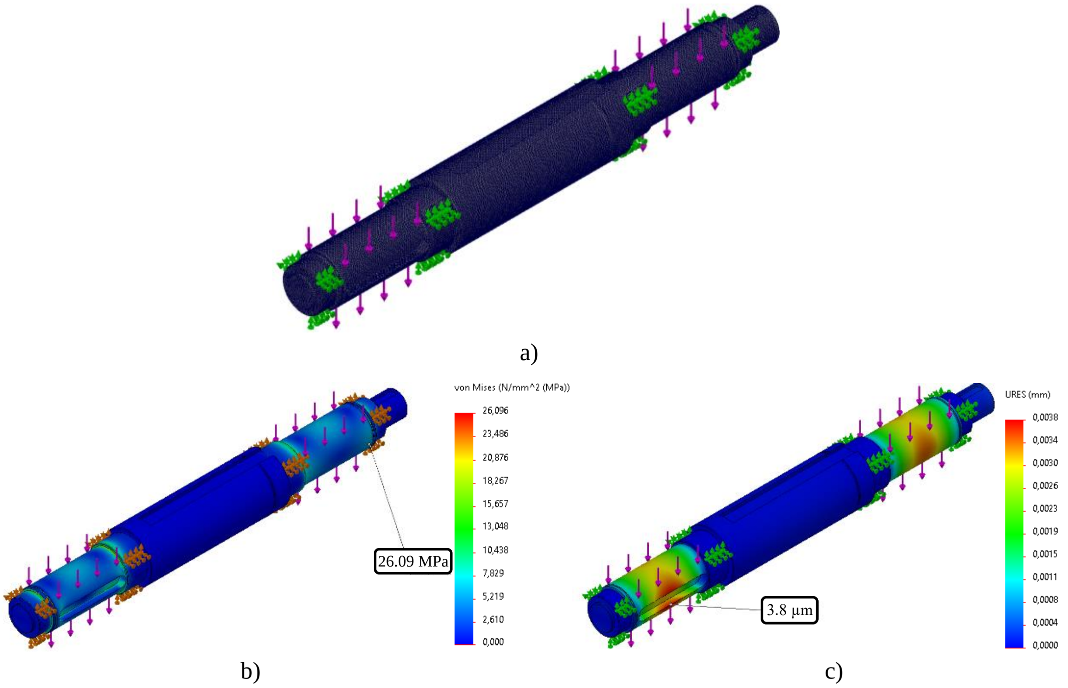


Figure 10. Motor shaft bending analysis a) Applied force and mesh structure, b) Von Mises stress analysis results (N/mm^2), c) Displacement analysis results (mm)

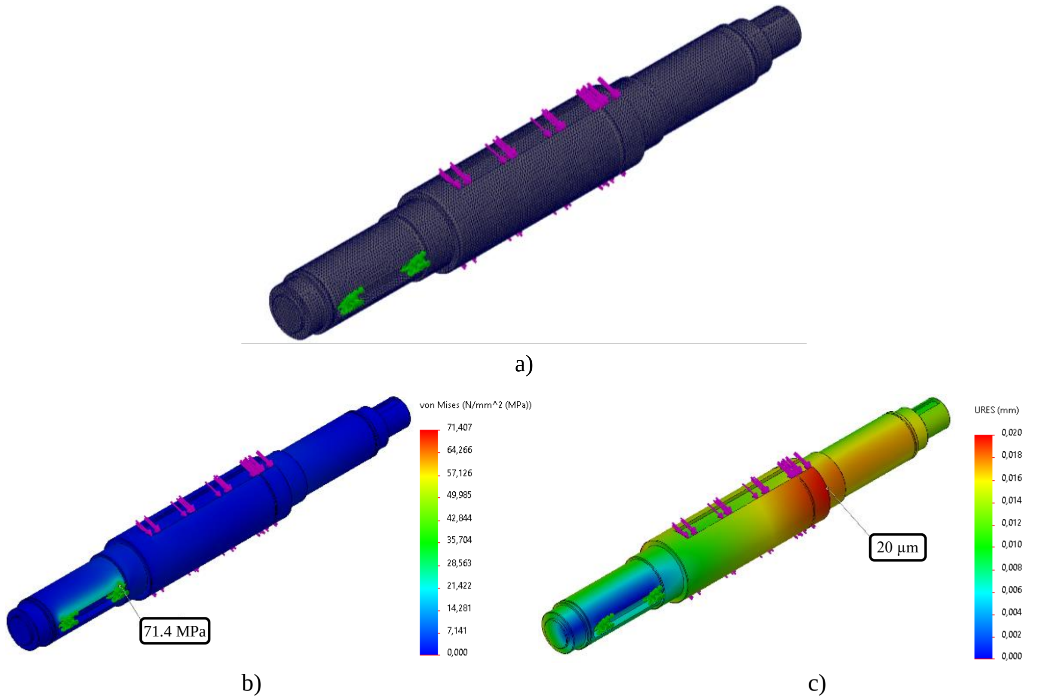
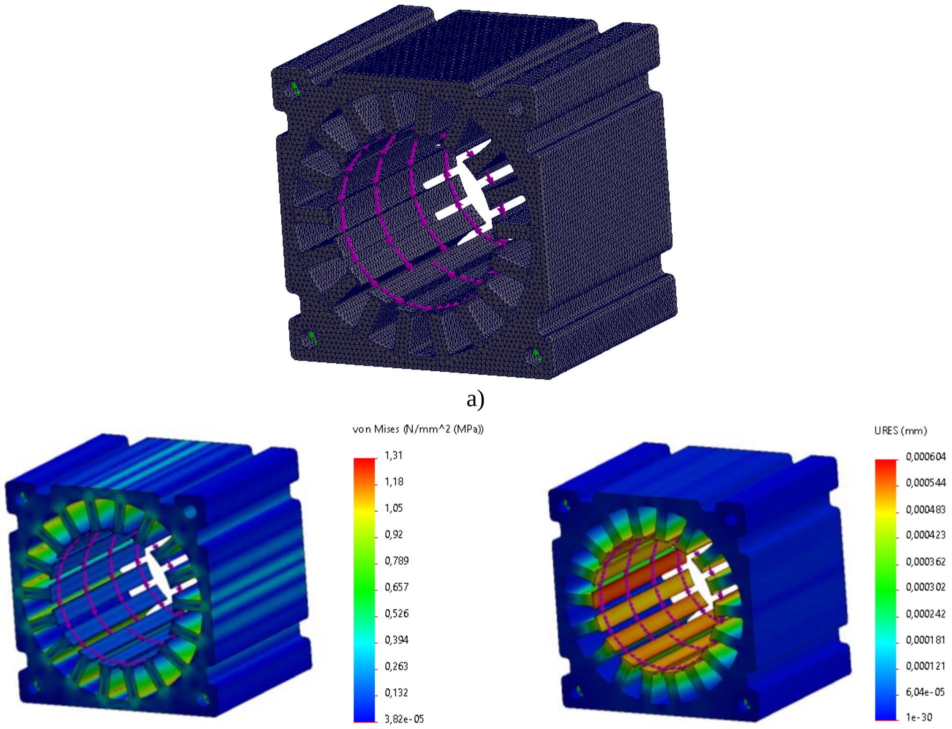


Figure 11. Motor shaft torsion analysis a) Applied force and mesh structure, b) Von Mises stress analysis results (N/mm^2), c) Displacement analysis results (mm)



a)

b) c)

Figure 12. Stator torsion analysis a) Applied force and mesh structure, b) Von Mises stress analysis results (N/mm^2), c) Displacement analysis results (mm)

Figure 12 shows the analysis results of the torque applied to the stator teeth.

4. PROTOTYPE PRODUCTION AND TESTS

The structural analyses of the designed elevator motor were carried out and its suitability according to the elevator safety coefficients was proven by FEA. The approved parts were produced by laser cutting and machining methods. The produced parts are shown in Figure 13.

Figure 13. Produced motor components

Stator windings were placed and magnets were attached to the rotor. Then, as a result of the assembly of all components, the belted elevator motor prototype took its final form. A competitive and efficient motor has emerged in the field of belt elevator motors. In Figure 14, there is the test setup of the motor, and in Figure 15, there is the final version of the produced elevator machine. Since the lowest safety factor was observed in the shaft torsion analysis, more attention was given to torsion test of the shaft. In the experiments, acceleration, stopping and loading operations were applied to the motor separately. Quick stopping with powder brake was applied to the motor operating at rated speed and 125% load, and the strength of the shaft under the effect of torsion was tested. The tests were completed successfully and no deformation occurred in the shaft.

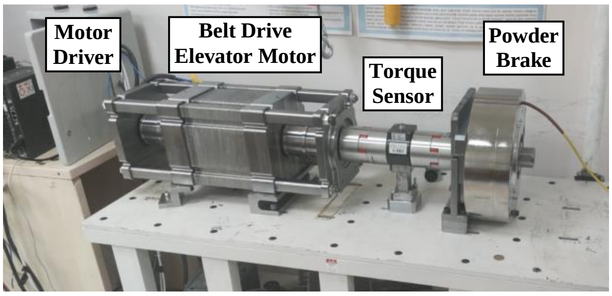


Figure 14. Test Setup

Figure 15. Prototype elevator machine

5. CONCLUSION

In this study, detailed design and prototype production of a double-sided belt elevator traction motor has been made. The advantages of the double-sided belt system and the proposed motor design can be listed as follows:

- Thanks to the machine being placed in the balance center in the hoistway, more comfortable travel will be achieved.
- Mass production will be possible only with laser, milling and lathe processes without the need for a cast body.
- A light and compact motor was produced.
- Compared to rope systems, a lower volume, lower weight, more efficient and more comfortable elevator traction system has been designed.
- The loads in the elevator motor are transmitted to the building and the ground in a smooth and balanced manner.

For bearing lifetime calculations, static loads on the bearings were calculated. After, 6212 and 6215 type bearings were chosen for both sides of the pulleys. It has been determined that the 6212 and 6215 type bearings in the elevator motor operating in the S5 duty cycle can operate without maintenance for approximately 13.6 and 26.3 years, respectively. Since bending and torsion forces affects to the motor shaft, it is produced from 42CrMo4 quenched and tempered steel. Covers and other connecting parts are made of medium tensile steel AISI 1045. The MBL value calculated according to the loads on the belts was found to be 36.91 kN. According to MBL value, BRUbelt 40kN belts with a cross section of 25x3.4 mm were selected. The pulleys were designed with a diameter of 100 mm. The pulley groove width is 55 mm in total with a belt width of 25 mm and a safe belt tracking width of 30 mm. Mayr Roba-twinstop size-180 was chosen as electromagnetic brake and Heidenhain ECN1313 2048 was chosen as the encoder.

The final 3D model of the motor was acquired and inserted into the hoistway design once all the components had been chosen and created. A total of 46 kN static load was applied to the motor. Also 200 Nm torsional moment was applied to the shaft. Bending and torsion effects were investigated for shaft, motor frame and stator teeth via FEA. According to the analysis results, the motor was obtained within the desired safety limits in accordance with the standards. As a result, a brand-new elevator motor design has appeared that can compete with others in the industry.

The maximum safety factor was occurred at bending of the motor frame as 30.8 times and the minimum safety factor was occurred at torsion of the shaft as 6.1 times. Since the minimum safety factor must be taken into account for the reliability of the system, the experiments were also done according to torsion of the shaft. In tests; a loading of 125% of the nominal torque was applied to the shaft with the powder brake system. Afterwards, the tests of the prototype produced motor were carried out and the motor design was verified by the tests. The shaft was examined after the tests and no deformation was observed. In low-speed rotating motors, the rotational speed is much lower than the critical speed, and the rotor inertia remains at low levels accordingly. Therefore, there is no problem related to dynamic effects in electric motors that rotate at low speeds [37]. Since the rated speed of the designed prototype motor is as low as 615 rpm and the produced motor will not operate at variable speeds, there was no need for dynamic analysis.

In this study, all the processes that must be carried out from design to its production of an elevator motor are mentioned. Considering the FEA carried out, the structural suitability of the elevator machine has been proven. It is predicted that this study will be an important source of literature for those who will work on issues such as determining static loads, belt selection, pulley sizing, bearing selection in belt drive elevator systems. In addition, it is expected to bring a new perspective to the elevator industry with design innovations such as the absence of casting processes and the belt placements that ensure the equal distribution of elevator loads. In future studies, belt and pulley selections can be made and new belt drive elevator motor designs with different carrying capacities can be obtained. In addition, corner square profiles can be innovated to improve motor assembly.

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DECLARATION OF ETHICAL STANDARDS

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

AUTHORS' CONTRIBUTIONS

Mücahit SOYASLAN: Wrote the manuscript, made the design, calculations and analyses, produced the motor and performed the experiments.

Yusuf AVŞAR: Wrote the manuscript, made the design, calculations and analyses, produced the motor and performed the experiments.

Ahmet FENERCİOĞLU: Made the design, calculations and analyses, validation of results and designs.

Feyyaz SARIHAN: Made the design, validation of results and designs.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

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