

Design and analysis of elevator wire ropes

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Abstract. In this research paper design and analysis of wire ropes used in elevator have been presented. The main objective of this study is to find the best practices on handling wire ropes for safety and to reduce the downtime of elevators due to wire rope failures. Design calculations have been done to calculate the apt dimensions and design is constructed on Auto Desk inventor software and the analysis have been performed using Ansys. Two configurations of the wire ropes have been studied and from this result it can be concluded that the design of wire rope is possible to relieve mechanical handling and to improve the design of wire ropes making it more durable by possible inclusion of new materials which in turn increases the efficiency and life of wire ropes without compromising safety.

Keywords: Mechanical engineering / safety research / numerical analysis / transportation

1 Introduction

Wire ropes are used for lifting and hoisting in cranes and elevators. They are commonly used for hoisting in elevator cars. The different types of wire ropes have been designed and tested for various applications to carry a tensile load. So, these types of wire are tested for both static and fatigue loading and these loads are transmitted through a flexible member [1,2]. Wire ropes are structural elements commonly used in industries and civil engineering domain. They consist of several steel wires twisted on the central core to improve the axial strength, stiffness and bending flexibility which in turn increases the mechanical properties. Wire ropes are usually made of many structures and they can fulfil their task of carrying tensile load without breaking [3]. Wire ropes are usually subjected to varying axial tensile loads or torsional moments resulting in unstable performances [4]. So, in this context wire ropes are usually analysed for stress and strain to study its mechanical strength [5] and hence research on this wire ropes are significant.

Wire ropes are composed of several twisted wires in a specific arrangement strands and then wound in a helical pattern around the central core. Imrak and Erdonmez [6] have studied new techniques in modelling of wire ropes. Initially they developed a modelling of wire rope using simple straight strand and they adapted this procedure for whole wire rope structure using double helical geometry.

They concluded this modelling procedure is easier and effective. Their results showed good agreement with other available literature and is considered simpler.

Fatigue performance in case of wire ropes depends on crack propagation which occurs in the internal and external cross sections of the wire rope and its ability to withstand abrasion and plasticity after long use [7]. So, the design of rope is carefully selected and constructed, and the lubricants are used to reduce the localized friction of individual wires in the strand. Kim et al. [8] have studied effect of wire rope diameter and tensile stress to study the life of wire rope. They considered five wire rope of 8 mm and 10 mm wire diameter and applied a constant tensile load and observed bending fatigue strength for different cycles. They concluded that the fracture surface area was larger in case of larger wire diameter and these data can be used to study the structural integrity in case of wire rope. The applications and the method of manufacturing wires are different for different equipment and it depends upon surface finish, texture lay, whirling grade, stress capacity, tolerance etc. [9]. So, it has become quite normal to expect wire rope construction from varying manufacturers, it becomes necessary to perform some mandatory tests before to determine various responses and reactions to various loading conditions. Barkany et al. [10] have studied non-destructive evaluation and discard criteria of wire rope giving importance to internal damage. They used the electromagnetic non-destructive and other methods to study the broken wire used in crane ropes and hoist ropes. They recommended to conduct magnetic test every year to check the failure criteria of wire rope. Sukhorukov et al. [11]

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have studied electromagnetic non-destructive inspection to inspect the wire ropes commonly used in oil and gas and mining industries. They concluded that electromagnetic non-destructive methods can be used to meet the challenges to study the failure and discard criteria. Rostami et al. [12] have studied the condition monitoring of a steel wire rope which was broken due to corrosive and fatigue used in case of hoisting elevator. They used finite element analysis to study wave propagation in wire ropes. They concluded by recording those wave propagations to analyse the wire rope. Zhang et al. [13] have studied electromagnetic techniques to study the wire rope. They designed a sensor to detect the broken wire using magnetic flux leakage method. They concluded that the designed sensor was optimized, and the performance of magnetic concentration have been analysed using Finite element method.

Wire ropes that operate on sheaves are subjected to various stresses due to which possible failures may take place. The life of the wire rope tends to decrease when it is subjected to increasing axial load and decreasing sheave bend radii. At the same time, the bending of the ropes on elevator pulleys will increase the cross section of the wire rope at the outer cross section whereas the inner most wires tend to shorten and is considered to be the main cause of damage and wear in the elevator operation units [14,15]. In this context Shi et al. [14] have studied a 3D model using finite element method considering 6+1 wire rope strand to investigate the traction between wire and sheave. They studied the tensile distribution in the wire. They concluded that contact wrap angle is critical in traction capability calculations. Nabijoe and Hobbs [15] have studied the frictional behaviour in case of heavily loaded wire ropes bent over sheaves considering different diameters. They concluded that the reduction in the ratio of D:d (diameter of the sheave and diameter of wire rope) exhibit larger travel distance of the individual wires and the multiple strands of the unit leading to increasing wear and in turn reduction in the life of the rope. Hence, careful consideration of the wire rope selection plays an important role for various types of applications. Vogel [16] have studied the endurance and the lifetime of the wire rope over the sheave. He studied the influence of rope considering the rope length and size factors in case of lift application operations. They concluded the influence of rope length and size factors on the life of the wire rope. Xi et al. [17] have studied the wire rope failures using Non-destructive testing based on magnetic and infrared information. They considered the broken wire for their testing, and they concluded that using this method it is possible to estimate the residual lifetime of the wire rope. From this literature review it is observed that researchers have studied wire ropes which are commonly used in lifting and hoisting in case of cranes and elevators. Most of the researchers have designed wire ropes to study tensile loading considering both static and fatigue loading apart, stiffness and bending flexibility to improve the performance (Mechanical strength) of wire rope. Also, some researchers have designed and studied the strand in case of wire ropes to study its ability to withstand abrasion and plasticity to improve the life of wire ropes. In this research paper wire ropes have been studied for safety

purposes to reduce the downtime of elevators and the analysis have been performed using ANSYS considering two configurations of wire ropes by introducing new materials to increase the efficiency and life of wire ropes.

2 Methodology

The base model of the wire rope is Autodesk inventor and the analysis are performed using Ansys software. Two configurations of wire ropes have been designed and their diameters is selected as 22 mm and 24 mm and the corresponding model is as shown in Figure 1. The corresponding dimension and other properties of the wire rope is as shown in Table 1. The maximum load considered is 45 kN and the wire rope diameter selected is 22 mm. The wire rope configuration selected for the analysis is as shown in Table 2 and Table 3.

The efficiency of wire rope is calculated using the following equation.

$$\eta_s = (\eta_r)^i \eta_f = (\eta_r)^i \left(\frac{1}{n}\right) \left(\frac{1 - (\eta_r)^i}{1 - \eta_r}\right) \quad (1)$$

where η_f is efficiency of pulley block, η_r is efficiency of one pulley block = 0.96 to 0.98, i is the number of pulleys = 6 and n is the number of ropes = 5.

Diameter of individual wires of the wire rope (d_w) is calculated using

$$d_w = \frac{d}{1.5 \times \sqrt{i}} \quad (2)$$

The minimum rope diameter is determined using the following formula:

$$d = C \times \frac{\sqrt{F}}{\sqrt{\eta_s}} \quad (3)$$

where d (mm) is Minimum Rope Diameter, C is Strength factor, F is Calculated traction Force, η_s is Efficiency of wire rope.

The Diameter ratio (For Car Sheaves, Counterweight Sheaves and Drive Sheaves):

$$x = \frac{D_p(\text{Diameter of Pulley})}{D_r(\text{Diameter of Rope})} \quad (4)$$

The Friction coefficient that is experienced by the rope (m_y)

$$m_y = \frac{0.1}{\left(\frac{1+v}{10}\right)} \text{ where } v = \text{rope speed (m/s)}.$$

The Rope Friction factor (f) during operation situation is calculated using

$$f = m_y A \left(\frac{1 - \sin \beta}{\pi - \beta - \sin \beta} \right) \quad (5)$$

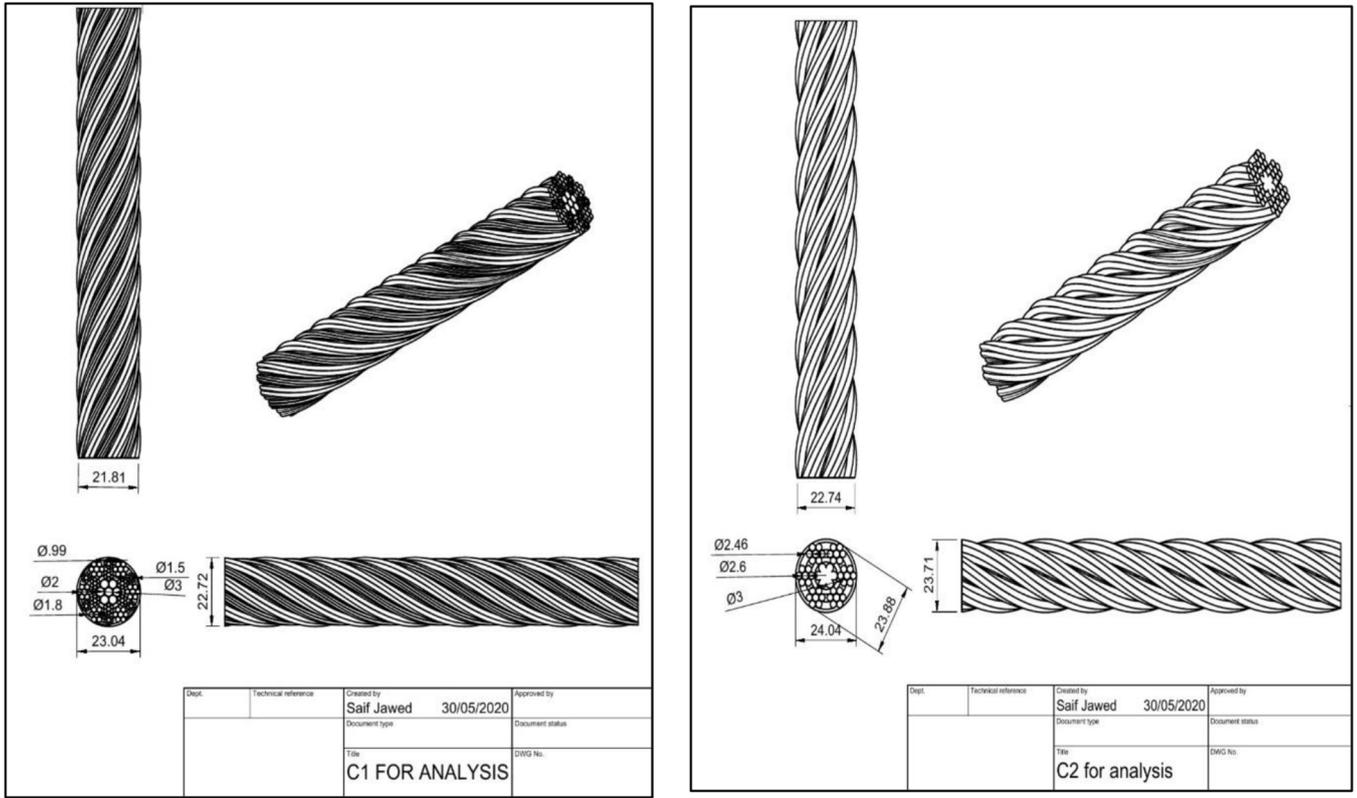


Fig. 1. Wire rope design configuration 1 and 2.

Table 1. Design selection and characteristics.

Diameter of the rope (mm)	Approximate weight N/m	Nominal breaking tensile strength of rope (N/mm ²)
22	17.75	241.24
24	20.594	278.51

Table 2. Wire rope configuration 1 material compositional specifications.

CONFIGURATION 1 – Material Compositional Specifications	
Primary Strand	Stainless Steel AISI 440C (1.20% C)
Secondary Strand	Stainless Steel AISI 440C (1.20% C)
Tertiary Strand	Stainless Steel AISI 440C (1.20% C)
Central Core Wires	Stainless Steel AISI 440C (1.20% C)
Central Core Core	Carbon Steel AISI 410

where m_y = friction coefficient (0.5 for operation situation and 1 for braking situation and stand still situation), β = undercut angle = 105°.

The Rope friction factor during Emergency braking situation is given as

$$f = m_y 4 \left(\frac{1 - \sin(1.83)}{\pi - 1.83 - \sin(1.83)} \right) \quad (6)$$

Table 3. Wire rope configuration 2 material compositional specifications.

CONFIGURATION 2 – Material Composition Specifications	
Primary Strand	Stainless Steel AISI 440C (1.20% C)
Central Core Wires	Stainless Steel AISI 440C (1.20% C)
Central Core's Core	Stainless Steel AISI 440C (1.20% C)

The Rope friction factor during stand still situation is given as

$$f = m_y \left(\frac{1}{\sin\left(\frac{\gamma}{2}\right)} \right) \quad (7)$$

where γ = groove angle = 30°.

Maximum possible pressure experienced by the wire rope is given by

$$P_{\max} = \frac{12.5 \times 4 \times V_s}{1 + V_s} \quad (8)$$

where P_{\max} = Max. specific pressure of the rope [N/mm²], V_s = Rope speed of the traction sheave [m/s] = 1 m/s.

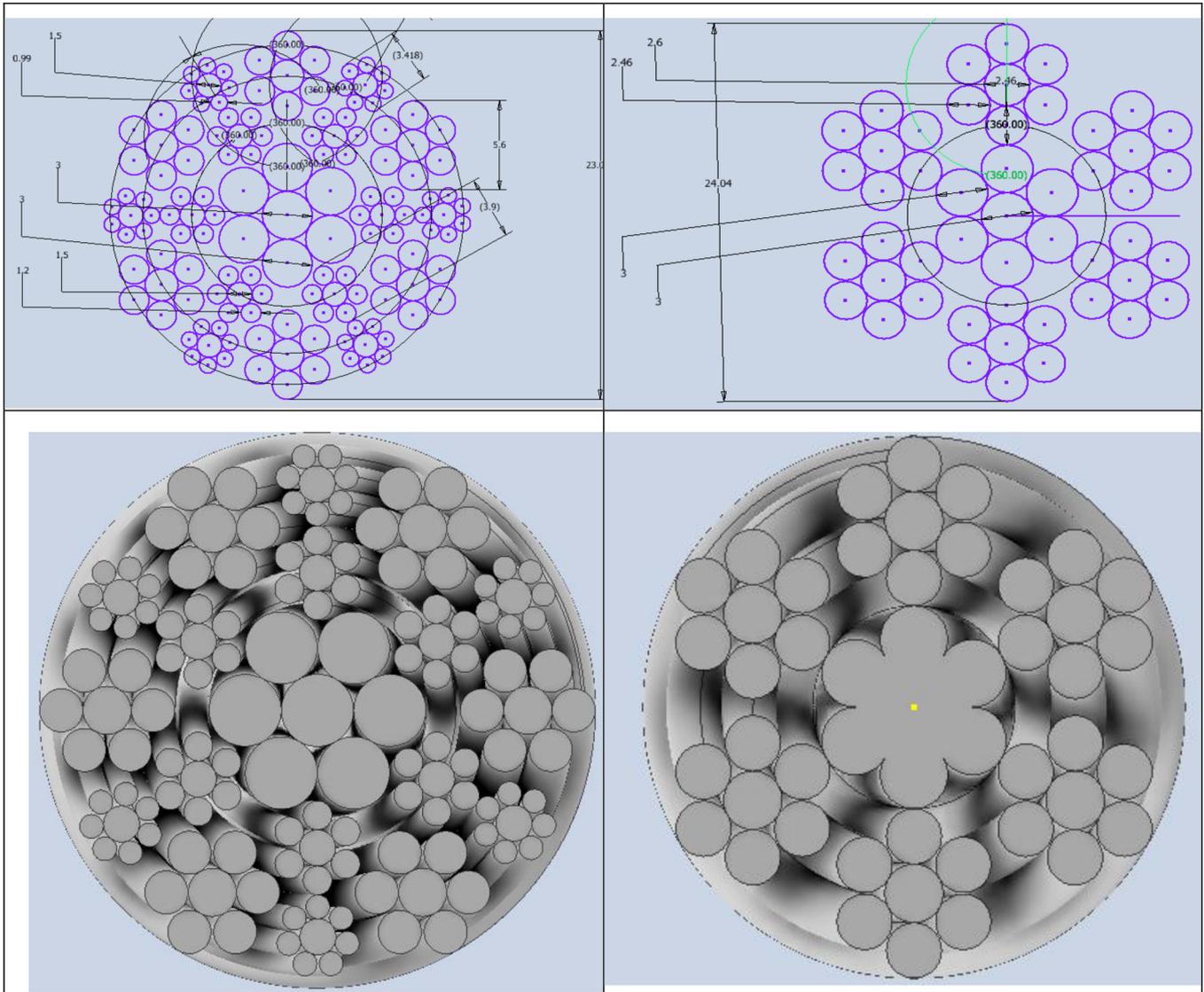


Fig. 2. Dimensions and front view of wire ropes considered (Configurations 1 and 2).

Actual specific pressure experienced by the wire rope is given as

$$p = \frac{T}{\left(\frac{(N_r \times D_r \times D_t) \times 8 \times \cos\left(\frac{\beta}{2}\right)}{\pi - \beta - \sin \beta} \right)}$$

where p = Specific Pressure of the rope [MPa], N_r = number of the ropes = 5, D_r = rope diameter [mm] = 22 mm, D_t = diameter of traction sheave [mm] = 400 mm, T = Static Rope force car side (Rated load bottom floor) [N] = 36910, s = Rope Speed on Sheave [m/s] = 1 m/s, Gamma (γ) = groove angle = 30°, beta (β) = undercut angle = 105°.

2.1 Checking for the sufficiency of the pulleys

$Kp = (Dt/Dp)$ where Kp = Ratio factor for difference in sheave and wire dia, D_t = Dia. of Sheave [mm] = 400 mm, Dp = Avg. Dia. Of all the sheaves [mm] 400/5 = 80 mm.

$Neq^p = (Nps + 4Npr) Kp$ where N = Number of pulleys, Neq^p = Number of Drive pulleys, Nps = No. of sheaves with eased bends = 6, Npr = No. of sheaves with negative bends = 0.

Since the Kp value equals the Number of pulleys drives as calculated, the number of pulleys so assigned is appropriate.

The effective loads experienced by the wire rope is calculated using

$$T = F_{car}(\text{operation}) = \frac{(m_y(\text{operation}) \times Q + P_{\max}) \times g}{r} \tag{9}$$

where m_y is the friction coefficient and Q is the load.

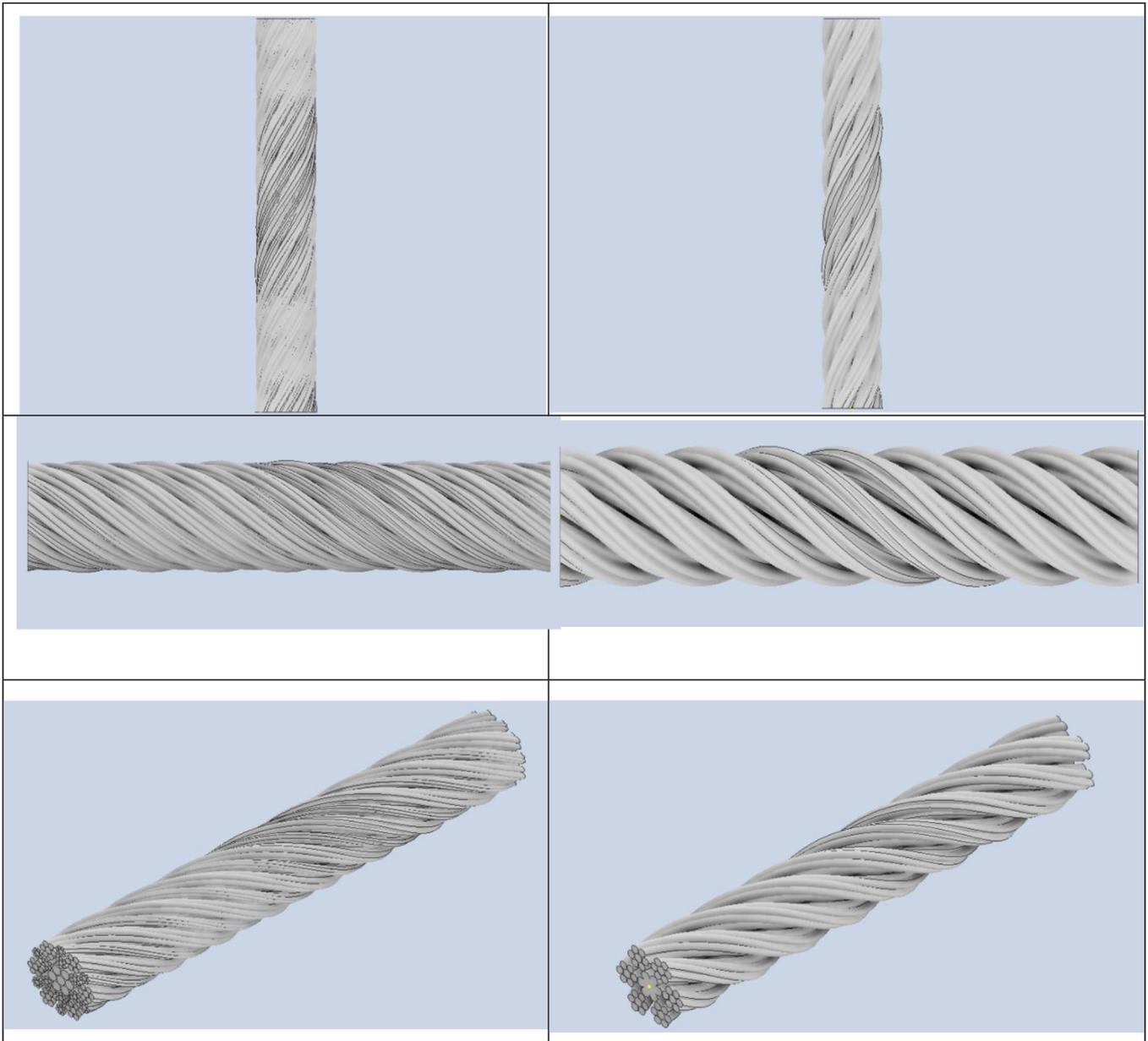


Fig. 3. Master view, right view, and isometric view of the wire rope (Configurations 1 and 2).

2.2 Factor of safety

Since normal breaking strength of Wire Rope C1 ≈ 212 KN and for Wire Rope C2 of designed diameter ≈ 286 KN, and maximum static load considering a freight elevator is 45 kN; The Factor of safety is the ultimate normal breaking strength of rope to the maximum static load ratio. i.e.,

$$\text{FOS} = \frac{\text{Normal Breaking Strength}}{\text{Maximum Static Load}} = \frac{212000}{45000} = 4.71 \quad (10)$$

$$\text{FOS} = \frac{\text{Normal Breaking Strength}}{\text{Maximum Static Load}} = \frac{286000}{45000} = 6.35. \quad (11)$$

3 Results and discussions

3.1 CAD model analysis

The two-wire rope model is constructed, and these wire ropes were tested against static structural analysis by applying a load of 45 kN. Stress analysis have been conducted on both the models (Configurations 1 and 2). The corresponding dimensions and the views of configurations 1 and 2 is as shown in [Figure 2](#) and [Figure 3](#). The analysis is performed by taking a small cross section of the wire rope considering one end as fixed and negative load is applied on the other end in the opposite directions. The model of configurations 1 and 2 and the corresponding meshed model of configurations 1 and 2 is as shown in

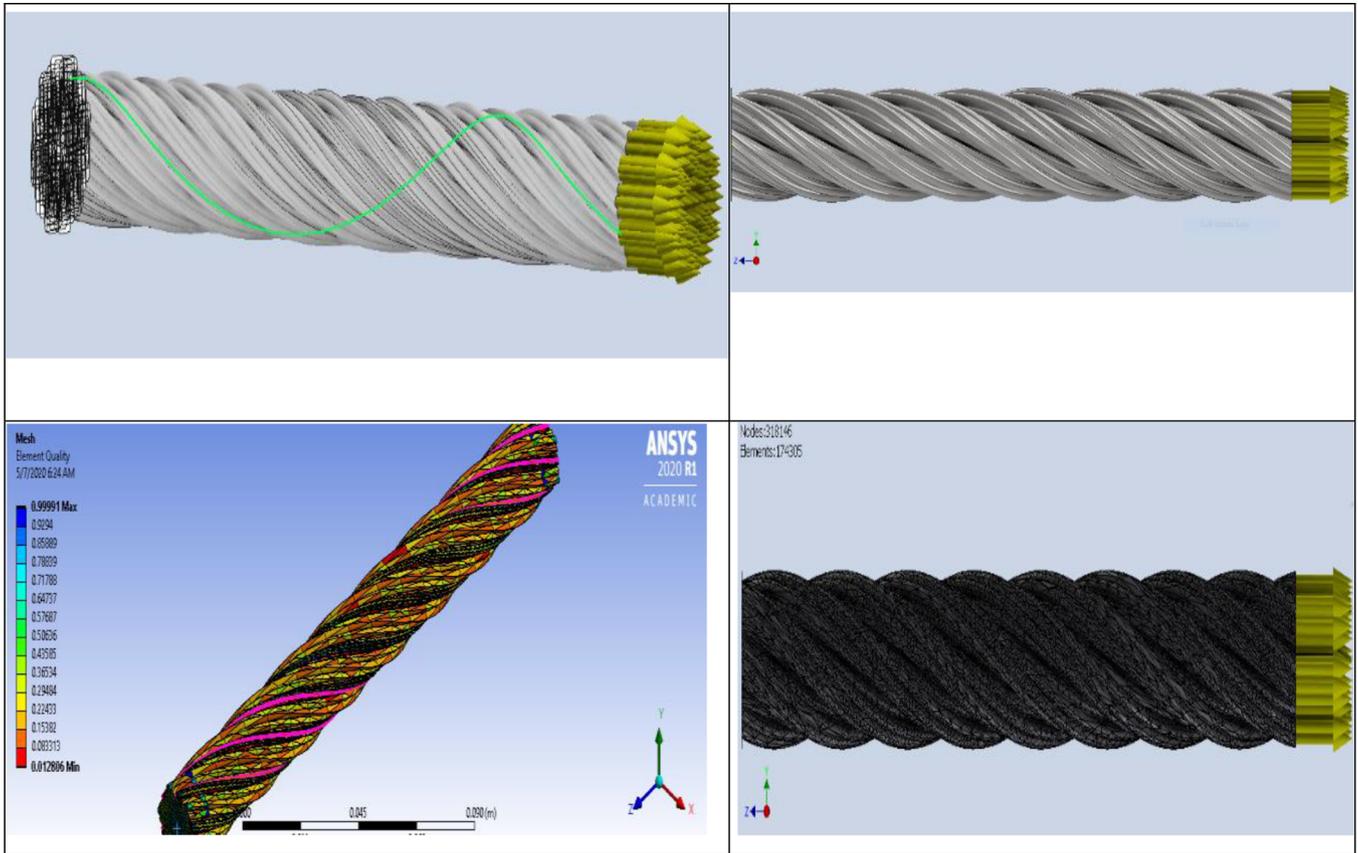


Fig. 4. Cad model and mesh model of configuration 1 and 2.

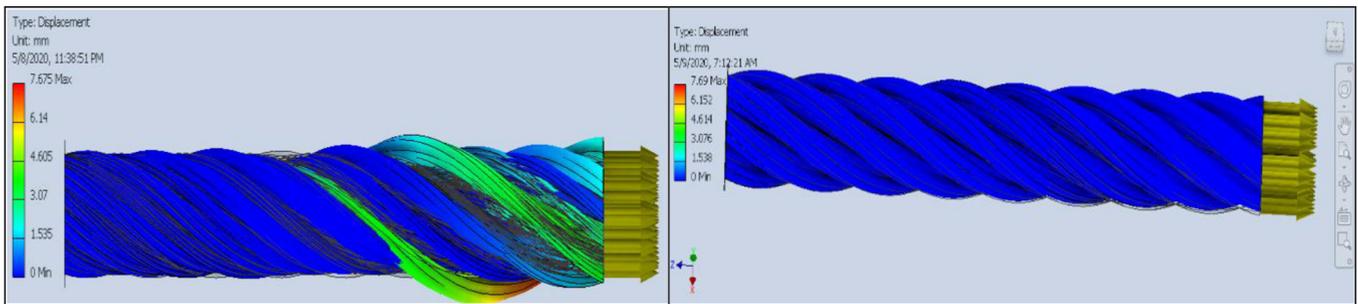


Fig. 5. Displacement diagrams of configuration 1 and 2 at a load of 45 kN.

Figure 4 and the number of nodes for the model meshing is 318166.

3.2 Analysis results of wire ropes

Figure 5 shows the displacement diagrams of wire ropes (configurations 1 and 2) for a load of 45 kN. From Figure 5a it is observed that wire rope is displaced in the low region of deformation, however, the far end of the wire rope shows individual strand displacement of up to 7.675 mm

maximum. This wire rope is of the configuration 7*19 which is considered unfamiliar and unconventional. From this it can be stated that with the usage of wire terminations or wire rope sling terminations the above deflections may be avoided which guaranty a sturdy and secure wire rope with high strength and high level of tolerance.

The wire rope configurations considered for the second case represents wire rope of 6*49 design as compared to configurations 1 is as shown in Figure 5b. This wire rope

Table 4. Wire Rope Physical Properties for GCS and Galvanized Steel (Analysis Values).

Material	Physical Properties	
	GCS (Galvanized carbon steel)	Steel, Galvanized
Density	1 g/cm ³	7.85 g/cm ³
Mass	0.0381284 kg	0.299812 kg
Area	99805 mm ²	58189.6 mm ²
Volume	38128.4 mm ³	38192.7 mm ³
Center of Gravity	X = 0.0000000510449 mm	X = 0.000000040191 mm
	Y = 0.0000000711695 mm	Y = 0.000000122819 mm
	Z = -75 mm	Z = -75 mm

configurations have been analyzed using Auto desk inventor using stress component displacement analysis and the number of nodes considered is 318166. This type of design is unconventional and never been constructed or analyzed for application. From this result it can be concluded that for a load of 45 kN, this wire rope has performed very well, and all the strand separation is intact which can be observed from Figure 5b. From this result it can be stated that the configuration 2 design is highly desirable in elevator lifting wire rope as it is having minimum strand separation and will guarantee maximum safety.

Table 4 shows the wire rope physical properties for GCS (Galvanized carbon steel) and galvanized steel considered for the analysis. From this table it is observed that mass of the wire rope made of GCS has 0.0381284 kg while the mass of the galvanized steel is 0.299812 kg. From this table it is also observed that area of GCS is 99805 mm² whereas the area of galvanized steel is 58189.6 mm². The density as well as center of gravity of both these cases can also be observed in Table 4.

4 Conclusions

This paper discussed about the design and analysis of a wire rope used in elevator system. The analysis has been performed considering two configurations of wire rope considering GCS and Steel Galvanized. From the analysis the following conclusions have been drawn.

- The static structural analysis using ANSYS has been performed by considering the two wire strand configurations of wire ropes to increase the efficiency and life of the wire rope without compromising the safety of rope.
- From this analysis it is observed that wire strand configuration 1 has shown internal strand to strand displacement and the factor of safety was less than 5 which is not recommended as per industry standards whereas, the strand configuration 2 has shown no internal displacement and has performed better as compared to configuration 1 and the factor of safety was higher as compared to the configuration 1 and is good design.
- The wire rope design using Autodesk inventor software and the analysis have been performed using Ansys. From this analysis it can be concluded that design of wire rope

considered for the study increases the efficiency and life of the wire rope without compromising safety.

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