DEVELOPMENT OF A COMPUTER PROGRAM FOR FRICTION WINDING SYSTEM DESIGN

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DEVELOPMENT OF A COMPUTER PROGRAM FOR FRICTION WINDING SYSTEM DESIGN

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ABSTRACT

DEVELOPMENT OF A COMPUTER PROGRAM FOR FRICTION WINDING SYSTEM DESIGN

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As the trend to deeper mines continues, mine hoists and associated equipment will become more sophisticated, complex, large and expensive. Correct selection of the right type of hoist is imperative. In this vital link between underground and surface, crude estimates of hoist capacity are not good enough, and the mining engineer must design and select the right hoisting system to meet the design specifications and establish the most suitable operating parameters. This study aims to constitute a software model, which results all required design parameters of friction type winding system on minimum required power. The computer program has been structured on Microsoft Visual Basic programming language. The program requires user inputs (winding depth, hourly hoisting capacity) and selections (type and number of motors, type of friction wheel mounting) to run macros and equations so that the operating parameters such as skip capacity, rope type and diameter, hoisting speed, acceleration, cycle period, friction wheel diameter are determined to give the minimum motor power requirement.

Keywords: Hoisting capacity, Koepe friction winding, Computer program for friction winding, Operating parameters, Motor power

ÖZ

SÜRTÜNMELİ KUYU İHRAÇ SİSTEM TASARIMI İÇİN BİLGİSAYAR PROGRAMI GELİŞTİRİLMESİ

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Derin madenciliğe yönelme sürdükçe, kuyu ihraç sistemleri ve donanımları daha karmaşık, büyük ve pahalı olacaktır. Gerekli ihraç sisteminin en doğru olarak seçimi şarttır. Yeraltı ve yerüstü arasındaki bu önemli bağlantıda, sadece ihraç kapasitesinin tahmin edilmesi yeterli olmayıp, maden mühendisi tasarım özelliklerini karşılayacak şekilde ihraç sistemini tasarlayıp seçimini yapmalı ve en uygun çalışma parametrelerini saptamalıdır. Bu çalışmanın amacı, sürtünmeli ihraç sistemleri tasarımının tüm yönlerini kapsayan ve sonuç olarak, tasarım parametrelerini veren bir bilgisayar programı geliştirmektir. Bu model Microsoft Visual Basic Program tabanlı oluşturulmuştur. Söz konusu yazılım, bazı kullanıcı girdileri (kuyu derinliği, saatlik kuyu taşıma kapasitesi) ve kullanıcı tercihleri (motor tipi ve sayısı, sürtünme tamburu yerleşim konumu) ile hazırlanmış makro ve işlemleri çalıştırmak suretiyle, en düşük motor gücü gereksinimini ve bu gereksinimi sağlayan skip kapasitesi, halat türü ve çapı, taşıma hızı ve ivmesi, tam tur zamanı, sürtünmeli tambur çapı gibi çalışma parametrelerini elde etmeyi mümkün kılmaktadır.

Anahtar Kelimeler: Kuyu ihraç kapasitesi, Koepe sürtünmeli ihraç sistemi, Sürtünmeli sistem için bilgisayar programı, Çalışma parametreleri, Motor gücü

Dedicated to My Father Ahmet Ünal
& My Fellow Burak Sencer

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HP	Horse power, hp
3D	Three dimensions
FW	Friction winder / wheel
GP	Guide pulley
HS	Headsheaves
TSE	Turkish Standards Institute
TS	Turkish Standards
VB	Visual Basic
SF	Safety factor
M	Mass of loaded conveyance, kg
D	Depth of wind, m
PL	Payload, kg
C	Mass of skip, kg
LC	Locked coil
TAS	Triangular strand
MG	Motor generator
AC	Alternating current
DC	Direct current
FWD	Friction wheel diameter, m
RPM	Revolutions per minute, rpm
HSD	Headsheave diameter, m
F	Friction, N
H	Mass of each headsheave referred to rope center, kgm^2
RMS	Root mean square
DCDD	Direct coupled direct drive
DCGD	Direct coupled geared drive
MRS	Maximum rope speed, m/s
MRPMFW	Maximum revolutions per minute of friction winder
PMP	Preliminary motor power, kW
RGR	Reduction gear ratio

MRPM	Motor revolutions per minute
GPD	Guide pulley diameter, m
AP	Calculated acceleration, m/s^2
MoI	Moment of inertia, kgm^2
T1	Load on ascending rope, kg
T2	Load on descending rope, kg
P1	Tensions on ascending rope, N
P2	Tensions on descending rope, N
R1	Mass of Rope on ascending side, kg
R2	Mass of Rope on descending side, kg
R3	Mass of rope between headsheaves and friction Wheel, kg
L1	Length of rope on ascending side, m
L2	Length of rope on descending side, m
L3	Length of rope between friction winder and headsheave, m
M1	Total dynamic load on ascending rope, N
M2	Total dynamic load on descending rope, N
Cm	Mass constant of rope
Cs	Strength constant of rope
L_a	Linear acceleration, m/s^2
R_a	Radial acceleration, rad/s^2
T_s	Static torque, Nm
T_a	Dynamic torque, Nm
T_e	Equivalent time, second
P_a	Required power for acceleration, kW
P_c	Required power for constant velocity, kW
P_r	Required power for retardation, kW
Ta / ta	Acceleration time, second
Tc / tc	Constant velocity time, second
Tr / tr	Retardation time, second
Tcr / tcr	Creeping time, second
Td /td	Decking time, second
ΔV	Rate of change of velocity, m/s
Δt	Rate of change of time, second
g	Gravitational acceleration, $9.81 m/s^2$

d	Diameter of rope, mm
a	Actual acceleration, m/s^2
r	Retardation, m/s^2
a^*	Slippage acceleration, m/s^2
e	Naperian base, 2.7182
μ	Coefficient of friction, 0.09
θ	Arc of contact, radians

CHAPTER 1

INTRODUCTION

1.1 General

Over the past decades, mine hoisting equipment and systems have moved from steam engines to static rectification of alternative current to direct current, and electronic controls are the standard.

The mine hoist can be a bottleneck between the underground mine and the surface preparation plant. Today, more consideration is given to the problems concerning winding techniques than to the mining methods applied to get the coal and minerals. The reasons for this priority are increased international competition, greater economy in production, increased productivity and rationalization. Mine shaft sinking is one of the most critical and technically difficult aspects of underground mine development and construction. A mine shaft must be completed and commissioned for a deep mine before any other underground mine development can commence (DMC Mining, 2008). The importance of shaft design and construction might be detailed; however, it is obvious that the shaft design and construction phases have a vital place among other mine development processes.

When a shaft design or sinking is considered, the name of hoisting system pursues immediately as a critical design parameter. Hoisting or winding systems are the main legs of any constructed shaft. Their functions in mine shafts might be compared as heart in a biologic body. They are used to raise and lower conveyances inside of the mine shaft. There are different principal types of hoists; might be listed in order of importance; koepe/friction, drum and blair multi-rope types. Friction systems are used mostly in Europe, Asia, and Australia (Vergne, 2003).

In consideration of any hoist, here as friction type, its design and compatibility with the other shaft parameters are very important since they make dependent all other facts in any shaft design. For instance, the diameter or required power to run friction hoist must be considered before any construction practice. On the other hand, there are very limited studies about the hoisting/winding design in the literature. This shortage has made the available studies indispensable. Moreover, there are lots of limitations and parameters on the focus of winding design. These limitations put design process very crucial. Lack of scientific studies and many limitations as a characteristic of winding design makes it very complex. So, any proper prepared study would be very helpful about this subject.

There are some risks or difficulties in hoisting design, which might be listed shortly;

- Hoist design is very important stage in the whole design study of any underground shaft,
- There are obvious lack of number of studies about this issue when compared other stages of mining and its development,
- The shaft sinking and operating is very costly process, so each parameters and facts should be determined carefully,
- There are many limitations in design of hoisting systems, and most of them are indispensible,
- Any mistake in design stage of hoisting system might be costly and has various effects till the closure of mine.

This list might be extended, however, the importance of any shaft design and directly related hoisting design can be understood, clearly. As it is seen, any hoist design requires long time, depends on many factors, has many limitations and has great impacts in consideration of mine life.

1.2 Objective and Scope of the Thesis

Nowadays, many newly developed technologies are applied on mining industry. Many 3D simulations, recently developed softwares or new practical design programs might be seen within mining operations. These modern technology benefits and applications should be made intersect with the requirements of any hoist design.

Thus, this study has been chosen as the thesis subject to put as a representative study in mining literature in terms of hoist design by usage of modern technology. The subject is limited to friction winding due to very wide scope of hoisting systems.

1.3 Prior Expectations from Designing Model

The friction (Koepe) winding system is one of the most applied and operated hoisting system in mining in last decades. This system has some significant advantages and characteristics when comparing with the other hoisting systems. Besides of its advantages and properties, friction winding system design has very complex, attention required, long time calculation characteristics. On the other hand, the design outputs should be as optimal as and maximum safety since shafts are main arteries of operated mine.

The study is aimed to form a complete design of friction winding system by aid of computer skills. It is expected that this study (software program) would give the most correct and the safest results in the shortest time as available. The convenient applications might be differed than the theoretical design results, however; the bases of practical applications are being on the theoretical results. So, some of the requirements and expectations from this subjected study (thesis study) might be figured.

1.3.1 Quick

As stressed above; friction winding design calculations are dependent on many different parameters. Most of these parameters also depend on each other among the design calculations. Such characteristic makes the design calculations very complex. Complexity of engineering side prolonged the calculation stage. The whole considered calculations under this thesis can take around 30 minutes to 2 hours or more if the calculations would be made by hand.

To illustrate these dependencies, it can be figured that rope diameters and rope unit masses, which will be used at almost all stages of design process, are calculated at the beginning. The friction winder diameter, which has absolute effect on motor selection stage, is directly dependent on rope diameter. Motor selection stage is considered at the final section of this design process. To imagine that any variation at the rope diameter will affect the motor selection, so the whole calculations should be started from the beginning if any changes were made. As a result, all calculated elements and time would be wasted. Under these conditions, hand-made calculations get longer and longer since any variation might be easily occurred in this process. On the other hand, computer technology can prevent this time loss. Proper established software can consider all possible variation and gives a solution in a shorter time. So, this subjected study should save significant amount of time in calculation process of winding design.

1.3.2 Efficient

Efficiency is the quality of being able to do a task successfully, without wasting time or energy. At the present time, efficiency is the basic and required factor from all engineering applications. Time considerations have a direct relation with the financial aspects. Energy has also direct relation with them and should be always considered.

The aimed results are acquired by efficient run of design process in engineering discipline. For this reason, efficient process is important for running design. Friction winding design should also be efficient and provide correct result; giving the expected in a possible shorter time and with less energy.

Friction winding design study has been based on the Microsoft Visual Basic software and directly utilizes the latest computer skills. This will cause of saving significant amount of time. The correct establishment of process flow will also ensure the accurate results. Meanwhile, the possible least energy is going to be consumed.

1.3.3 Correct

Accuracy is the number one expectation from any engineering design. Accuracy in design process, accuracy in calculations and accuracy in determining best choice are the upcoming prospects.

The computer based friction winding design process should acquire all above illustrated expectations. This study aims to provide the most accurate results in terms of friction hoisting system. The results of design study might have important facts on real-life applications. These facts and their effects might have irreversible impacts on them. Due to their importance, the results must be absolutely accurate.

This study is based on scientifically accepted theories and methodologies. Each calculation is researched in literature whether they are commonly accepted or not. As a result of these searches, the most common ways, approaches and concepts are used as bases of this design process. Additionally, the human factor, namely human errors or misjudgments, are eliminated by usage of computer skills. It is clearly expected that the results of design process by application of this study will be accurate as possible after considering these whole factors.

1.3.4 Easy

The outcome of this study would be a software program, which will provide accurate results in a short time with a high efficiency. Beside these, utility is important to be simple. This model, as a product of this study, serves any user a standard utilization by its simplicity.

It is considered that any member of engineering disciplines should use and understand outcomes of this program without any assistance, easily. So, user-friendliness is determined at the design stage of this study.

Evaluation of all these issues, expectations are very high from the produced software, generally from this study. The results must be accurate, which is natural expectation from any scientific study. The results should be obtained in a short time. For a while, the all process should run efficiently. Users can expect to achieve the design outputs by least effort via this formed model.

This master study was sustained under the headlines of these expectations. The direct or indirect effects of such expectations were considered and this study was concluded accordingly.

1.4 Features of Studied Design Program

The based methodologies and applied calculations are mainly sourced from S.C. Walker’s (1988) Mine Winding and Transport book by Elsevier publisher.

There are many parameters in hoisting system design and most of them are important in terms of their effect. This study is aimed to develop software for friction winding design. This developed software is expected to provide most of the design outputs and prepare a base for practical applications.

The studied program is expected to calculate and yield some of the parameters as a result. There are calculated many design variables by program. These can be listed below; however, their functions and places in the design process will be discussed in the following sections.

- Rope type
- Rope diameter
- Required total rope length
- Skip capacity
- Friction wheel (FW) diameter
- Headsheaves (HS) diameter
- Guide pulley (GP) diameter
- Calculated safety factor
- Slippage acceleration
- Total cycle time
- Maximum speed of conveyances
- RMS power
- Required motor power

These variables are calculated while the design process and some of them are presented as outcome. Since some of these variables are important in terms of friction winding design and some of others are only a media in terms of design outputs.

The program requires decision of user in some stage of calculations to achieve these design outputs. These options can be seen as flexibility to user. User should decide the mounting type (tower or ground), angle of contact (between 210-230 degrees), length of distance between FW and HS (only for ground mounted/between 30-50 m) and some of others.

By decision of these design variables, any user should provide basic required inputs to this design program. These inputs have vital role since friction winding design program runs on these inputs. These inputs are also basic figures for any considered underground mine. These inputs are classified as depth of wind (shaft), tons per hour (hourly planned hoisting production (ore/waste)). The program runs on these provided inputs, basically.

There are also reference tables. Model runs calculations by sourcing required data from these tables. These provide general characteristic data of issued materials. Rope tables are referred to *Turkish Standards Booklet, TS 1918,* “Steel Wire Ropes for General

Purposes", published by Turkish Standard Institute (TSE); moment of inertia tables of FW, HS and GP, moment of inertia tables of A.C. type motor, D.C. type direct drive motor, D.C. type geared drive and winder reduction gear tables are referred to "*Mine Winding & Transport"* book, written by Walker (1988). "The Turkish Standard for Steel Wire Ropes for General Purposes", TS 1918, covers the all applicable rope types and their specifications for friction winding systems for underground mines. This standard is admitted as a formal rope standards for underground mine hoisting ropes. The inertia tables, which are illustrated at Mine Winding and Transport Book, are grouped and arranged to cover all applicable motor types to friction winding systems and hosted all required specifications. These applied tables might be modified or updated as a future application, which is an important properties of studied program as given a chance to modifications.

Safety factor is another important parameter, which should be included in design of this program. Most of the developed or developing countries have some mine legislations and this safety factor for hoisting in underground mines put into force as a must. This program is decided to evaluate only skip as a main conveyance. Skips are used only for material or bulk transportation, generally. So, their determined safety factors are lower than cages since cages can be used for human hoisting, too. The constant safety factor in this study is decided as 7.0, which has been written in forced regulation related to *worker health and job safety in mine and quarry enterprises and tunnel constructions of Turkish Republic [13/08/1984 - 84/8428]* (Council of Ministers, 1984). The program gives a chance to modify this factor in future applications.

1.5 Outline of the Thesis

This thesis is divided into six chapters including the Introduction Chapter 1. Literature Review is presented in Chapter 2. Principles and Framework of the Designed Software Model is given in Chapter 3. Mathematical Approach and Methodology are given in Chapter 4. Software Approach- Program Design Results and Discussions are given in Chapter 5. Lately, Conclusions and Recommendations are given in Chapter 6.

CHAPTER 2

LITERATURE REVIEW

2.1 Software Model Studies

There are conducted some researches and reviews for the subject of mine winding transport design both in internet and written references. However, it was resulted that there are not any sufficient studies on winding design in terms of software applications. Then, these researches are extended to cover the mine shaft design with current computer skills. There are some basic software model studies on underground mine shaft design but mostly they are focused on shaft sinking and construction subjects and mainly comprise financial aspects.

Thus, it can be concluded that there could not be found any relevant studies about mine winding transport designs with applying computer skills. At that point, this study is very important to start a survey for design stages of friction winding systems with using of current computer skills.

2.2 Mine Shafts

Harmin (2001) expressed, “Shaft” as a term in mining as; vertical or inclined underground opening through which a mine is worked. Shafts are like aortas in human body as their duties and importance. They are worked as main entry to underground mines where they are established. All transport duties are supplied via shafts. Mainly, shafts are used for accessing an ore body, transporting men and materials to and from underground workings, hoisting ore and waste from underground, serving as intake and return airways for the mine "*ventilation*", providing a second egress as required by mining law, storing of nuclear waste (Matunhire, 2007).

Many underground operations consist of several tunnels acting as accesses, haulages, production levels, and airways but there is only a limited number of shafts that can be developed for any given ore body and these shafts must be sunk in the right place with the correct configuration to get optimum operational benefit. (Matunhire, 2007)

Two different descriptions can be used; (*I*) a vertical, deep, restricted cross-section excavation and (*II*) a vertical or inclined primary opening in rock that gives access to and serves various levels of a mine (Brucker, 1975).

There are several classifications that can be used to differentiate shafts by type. For the purpose of this discussion, four commonly used classifications are presented. Shafts can be classified; (i) by purpose, (ii) by configuration, (iii) by ground support, (iv) by excavation method.

In case of purposes of shafts can be listed as illustrated below;

i. Production: ore and waste handling
ii. Service: personnel and materials handling
iii. Ventilation: upcast or downcast airflow
iv. Exploration: for defining mineral deposits
v. Escape: for emergency
vi. Combinations of the above (Edwards, 1992).

They might be classified by their size and configurations. Circular, rectangular and elliptical are some of them. Size of the shafts can be small as (3 to 15 m^2) or greater than (200 m^2) (Lineberry, 1992).

According to Edwards (1992), the ground support requirements might be used as a classification of shafts. The timber shafts, concrete-lined shafts or steel-lined shafts are some of them.

2.3 Mine Shaft Design

Since shafts play a major role in the general planning of mine development, their location is usually pre-determined. The location of a shaft can be changed when adverse geotechnical site conditions are encountered (Unrug,1984). However; after designing stage of a mine shaft, changing the location of construction is very costly and time consuming. The design of mine shaft is an iterative process, which requires several variables and options to be considered in order to arrive at an economic decision (Matunhire, 2007).

The first stage in selecting a shaft location is the surface topography. The shaft must be placed in an area where supporting infrastructure is able to be located in close proximity. Special considerations must be made when, for example, the mine is to be located under a lake or close to major faults (Queen's University, 2009).

Placement of the shaft is a trade-off between development costs/traveling distance and ore recovery. In flat lying tabular ore bodies with single seam mining at moderate depths, placing the shaft in the centre of the ore body is the most efficient solution. This reduces haulage distances to the shaft underground, as well as ventilation airflows to the production faces. However, the central placement results in the required safety pillars around the shaft reducing the recovery of ore, as they cannot be mined. An alternate solution to increase the recovery of ore is to place the shaft outside of the ore body, with safety pillars consisting of waste material. However, in this situation there is another trade off as haulage and traveling distances to the shaft as well as ventilation requirements are dramatically increased. In fact, placement of the shaft at side locations can increase development and transportation costs underground by a margin of 50% (Unrug, 1992) when compared with the central scenario (Queen's University, 2009).

The shaft design process is very critical stage in developing of a mine after consideration of underground mine production. There are some important facts, which can clarify the importance of this application.

The design study consumes very long times by its detailed and complex engineering calculations, evaluations, and its long listed parameters.

Mine shaft design and construction are very expensive works when compare to other stage of mine development. The design, installation, operation and maintenance costs are very high and important place in operating an underground mine.

Safety in shaft design and operations is another significant parameter. Since the constructed shafts are designed for servicing for the whole life of mine. The safety considerations of shafts should be determined from the construction of opening the mine till the closure.

Under considering these parameters; shaft design and construction are very long, costly, attention required process and has significant place in a decision of mine life.

2.4 Shaft Hoisting Systems

Two general types of access should be considered: (*I*) vertical to near-vertical shafts using hoists and cable suspended conveyances (Figure 2.1), or (*II*) horizontal or inclined openings using rail, trucks, conveyors, or cable-operated conveyances.

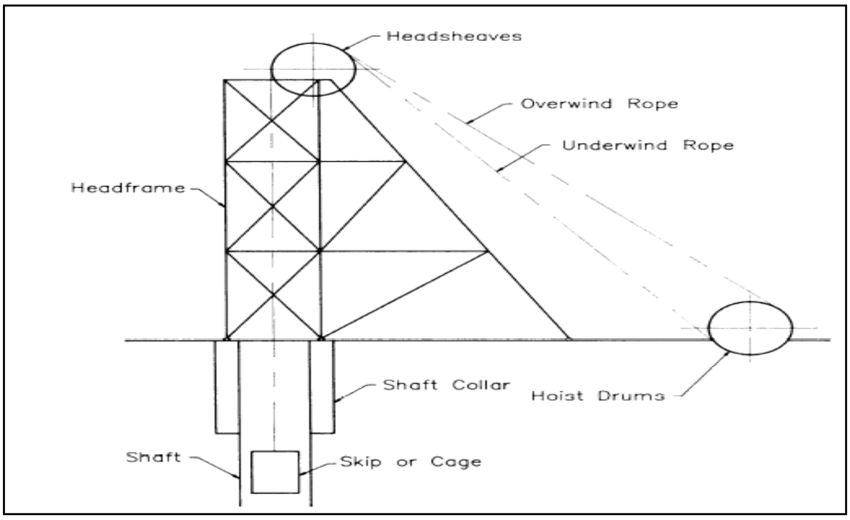


Figure 2.1 Typical Shaft Hoisting System (Edwards, 1992)

Since a shaft often provides the most direct access over the longest period of time there is an advantage to designing a shaft for maximum duty, consistent with economy. The current trend in shaft design is to provide multipurpose shafts. These shafts contain facilities for handling ore, waste, materials, personnel, services, manways, and ventilation.

During the process of identifying the purpose of the shaft, it should be realized that once a shaft is excavated and equipped, it cannot be enlarged easily in the future. Therefore, the shaft's initial and ultimate requirements must be defined during the design phase.

A shaft hoisting system has been composed of generally five main components: (1) hoist (wind), (2) conveyance, (3) rope, and (4) headframe (Edwards, 1992).

Edwards (1988) has also identified an additional 277 subcomponents. The number of subcomponents and their interrelationship with the main components are indicative of the complexity involved with the design of shaft hoisting systems.

2.4.1 Certain Components of Hoisting Systems

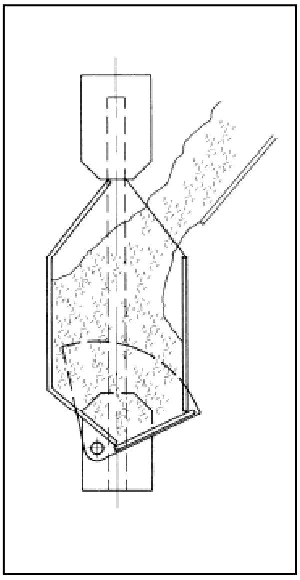
2.4.1.1 Wheels

Generally, it is pronounced two types of wheels; (*I*) Drum wheel and (*II*) Friction (Koepe) wheel. The main difference between them is, the hoist rope is stored on drum, however; the hoist rope passes over friction winder in hoisting operation. There are several sub-types are available in each winder types.

Drum winders are usually located at some distance from the shaft and require a headframe and sheaves to center the hoisting ropes in the shaft compartment. Friction winders may also be located directly over the shaft and, depending upon the wheel diameter, may require deflection sheaves to center the rope in the shaft compartment (Edwards, 1992).

2.4.1.2 Conveyances

Skips and cages are defined as conveyances in mine hoisting. They are classified by their uses. Handling personnel and material are defined as cages. Handling of broken ore or waste material is defined as skips. Counterweights are also defined as conveyances.



(A)

(B)

Figure 2.2 (A) Ordinary Skip / (B) Ordinary Cage

2.4.1.3 Ropes

There are three common applications of steel ropes in mine hoisting systems.

Table 2.1 Ropes by their usages and common constructions (Edwards, 1992)

Rope Use	Rope Construction
Hoist Rope	Round-Strand Flattened-Strand Locked Coil
Balance Rope	Non-Rotating
Guide Rope	Half-locked Coil

Also; the three types of steel ropes used mostly in mine hoisting are round-strand, flattened-strand and locked coil.

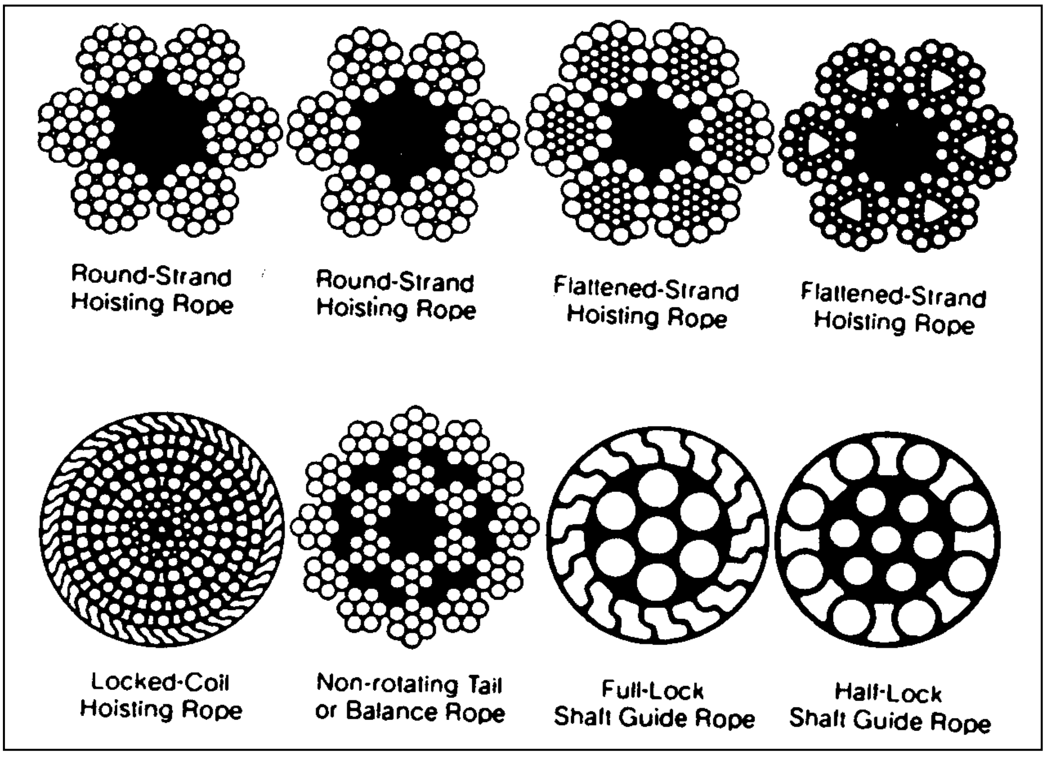


Figure 2.3 Steel rope types by their particular construction (SIEMAG Nordberg Hoisting Technology, 2001)

2.4.1.4 Headframes

Headframes are built with timber, steel, concrete, and a combination of steel and concrete. Wood headframes are no longer built in industrialized countries, but they still have application in the developing world. The question as to whether a steel or concrete headframe will be best for a particular project is a problem often encountered (Vergne, 2003) (b).

2.5 Winding in Mine Hoisting Systems

2.5.1 Drum Winding

Where it is established that shaft hoisting is required for a proposed mine, a determination is often required whether to employ a drum hoist or a Koepe (friction) hoist. A friction hoist system differs from a drum hoisting system in performance as well as components. Therefore, when attempting to decide which type of hoist to use, it is necessary to compare the two complete systems rather than the two hoists alone (Edwards, 1992).

Brucker (1975), Schulz (1973) and Tudhope (1973), among others, have discussed drum hoist and friction hoist applications. The following general statements help distinguish between these two hoisting systems: (i) double-drum hoists are the preferred hoist for shaft sinking; (ii) double-drum hoists are the best choice for hoisting in two compartments from several levels; (iii) drum type hoists are best suited for high payloads from shallow depths; (iv) the limitation on a drum hoist employing a single rope is the ultimate strength of the rope, because large ropes are difficult to manufacture and handle; (v) the depth capacity of drum hoists can be extended by using two ropes per conveyance (Blair-type hoist), and with this arrangement, Blair hoists can be used for depths exceeding those of either single-rope drum hoists or friction hoists); (vi) friction hoists with multiple ropes can carry a higher payload and have a higher output in tons per hour than drum hoists within a range of depths from 460 to 1520 m; (vii) friction hoist mechanical operation is very simple, has a low rotational inertia, and is less costly than a drum hoist; (viii) friction hoists have a lower peak power demand than drum hoists with the same output; and (ix) the friction hoist can operate on a relatively light power supply (Edwards, 1992).

2.5.2 Friction Winding

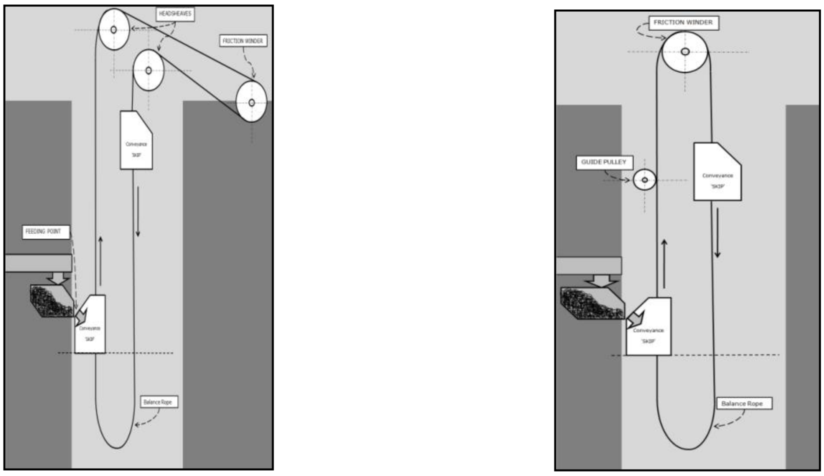
The Koepe or friction hoist was developed by Frederick Koepe in 1877. It consists of a wheel with a groove lined with friction material to resist slippage (Edwards, 1992). The Koepe system is applicable to those cases where there is no possibility of rope slip and the difference of the tensions between the loaded and empty sides is sufficiently small not to cause slip. In order to make this difference as small as possible, the use of a tail or balance rope is desirable (Walker, 1988).

The friction hoist is a machine where one or more ropes pass over the drum from one conveyance to another or from a conveyance to a counterweight. In either case, separate tail ropes are looped in the shaft and connected to the bottom of each conveyance or counterweight. The use of tail ropes lessens the out-of-balance load and hence the peak horsepower required of the hoist drive (Vergne, 2003) (c).

2.5.2.1 Comparisons of Ground-Mounted and Tower-Mounted Friction Windings

In early installations, the hoist was mounted on the ground, and a single rope was wound around the drum and over the headsheaves to the conveyances, in a balanced arrangement. In addition, a tail-rope of the same weight per unit length as the head-rope was suspended in the shaft below each conveyance. Thus the only out-of-balance load was the payload; Ground-mounted hoist (Figure 2.4 (A)).

As hoisting loads became larger, the number of head-ropes and headsheaves increased to the point where it became more practical to install the hoist in the headframe directly over the shaft. In North America, many friction hoists are mounted in this way. In order to bring the rope centers in line with the compartment centers, deflection sheaves must also be installed in the headframe below the hoist; Tower-mounted (Headframe-mounted) hoist (Edwards, 1992) (Figure 2.4 (B)).



(A) (B)

Figure 2.4 (A) Ground Mounted Hoist / (B) Tower Mounted Hoist

Ground-mounted friction hoist systems and tower-mounted friction hoist systems have some significant advantages when compared to each other. The conditions should be determined and decided for achieving the best solution. Some of the advantages of ground-mounted friction hoist are given below:

- Steel headframe (concrete is preferred in tower mounts for rigidity, reinforced concrete is not subject to residual stresses).
- Shorter headframe.
- An elevator is not required in the headframe.
- An overhead bridge crane may not be required.
- Easier access for maintenance.
- A water supply to the top of the headframe is not required.
- Shorter runs of power cables.
- Less susceptible to damage from over winds, mine explosions, lightning, and earthquakes.
- The longer rope between the hoist and the highest point of conveyance travel makes rope surge and possible subsequent structural upset less likely.
- Most efficient use of available space in the shaft for conveyances.
- Generally believed to be less susceptible to operating problems.

This may be partly due to the fact that it is more forgiving with respect to differential in hoist drum groove diameter because of the greater distance between the high point of travel for the conveyance and the hoist wheel (Vergne, 2003) (b).

The tower-mounted friction hoist system has some significant advantages, which are listed below:

- Zero or one deflection sheave is required.
- Installing and changing head ropes is less complicated.
- Rope vibration (whip) is less of a concern.
- The headframe tower may be more aesthetically pleasing.
- The headframe shell can be used for shaft sinking simultaneous with Koepe hoist installation above the sinking sheave deck (Vergne, 2003) (c).

2.5.3 Comparisons of Drum Winding and Friction Winding

Generic lists may be formed in a format that separates advantages and disadvantages of both drum and friction winding systems (Edwards, 1992).

2.5.3.1 Advantages of Drum Winding

- The drum hoist requires less downtime for routine maintenance.
- The maintenance regime for a drum hoist is less sophisticated.
- The drum hoist can continue to operate normally when the shaft bottom is flooded.
- Less shaft depth is required beneath the loading pocket.
- Less over-wind and under-wind protection is required.
- Since the drum winding can be used as unbalanced system, it is suited to multi-level hoisting.
- The drum hoist is less subject to nuisance trip-outs because it is equipped with fewer control and safety devices.
- Less investment in spare rope inventory is required of a drum hoist.
- If one conveyance is jammed in the shaft, emergency access may be had with the other conveyance of a double drum hoist.
- If a shaft wreck occurs, it is typically less catastrophic with a drum hoist than with a friction hoist.

- The drum hoist has a more liquid market and higher salvage value when it needs to be replaced or is no longer required.

2.5.3.2 Disadvantages of the Drum Winding

- The drum hoist generates power at the end of the wind, which goes back into the power grid. If the grid is provided by generated power, this can become a problem because generators are designed to produce and not receive power.
- This problem is more acute with multiple generators fighting to maintain synchronization. The problem is alleviated if an independent steady load is included in the generator grid (to act as a sink for power generated by the hoist).
- The spikes of the drum hoist cycle are also a problem for generators. They do not react well to rapid fluctuations in demand, particularly if the generators are not over-sized for the application.
- A drum hoist takes up more space than a friction hoist, for the same service.
- To change the rope diameter on a drum hoist requires a new drum sleeve or shell, while on a Koepe hoist, only the tread liners need to be replaced.
- For application underground, the drum hoist may have to be specially manufactured with sectioned drums to fit travel ways.

2.5.3.3 Advantages of the Koepe (Friction) Hoist

- A new Koepe hoist is less expensive to purchase than a new drum hoist for the same service.
- The delivery time for a new Koepe hoist may be less than a new drum hoist for the same service.
- More competition exists in the manufacture of friction hoists.
- A multi-rope Koepe hoist has a capacity to lift a heavier payload than a single-rope drum hoist.
- The peak power consumption is less, requiring a drive of smaller nameplate horsepower for equivalent service.

- The energy consumption and peak power recorded by a demand meter are virtually the same for a Koepe or drum hoist for equivalent service, but the effects on a sensitive power grid are less for a Koepe hoist.
- The Koepe hoist does not regenerate significant power into the grid, which may be of consequence when the power is supplied by on-site generators.
- The Koepe hoist is of smaller diameter than a drum hoist for the same service, hence easier to transport and erect for an underground blind shaft (winze).
- Rope life is usually much longer than for a drum hoist.
- A Koepe hoist can operate at higher speed than a drum hoist.

2.5.3.4 Disadvantages of the Koepe (Friction) Hoist

- A balanced Koepe system is not satisfactory for hoisting from loading pockets at different horizons in the shaft. For this service, a skip/counterweight configuration is required.
- A Koepe hoist is generally not suited to shaft deepening.
- A Koepe hoist is not satisfactory for sinking deep shafts.
- The braking effort is restricted by the requirement to maintain friction between the head ropes and drum.
- If the shaft bottom is flooded, the Koepe hoist is automatically slowed to creep speed.
- A used Koepe hoist is difficult to find to fit a particular application.
- Rope replacement is accomplished with great effort and may require a mid-shaft rope changing station if the shaft is deep.

CHAPTER 3

FUNDAMENTALS OF DESIGNED MODEL

3.1 Introduction

This subjected friction winding design model is based on Microsoft Visual Basic. Visual Basic (*VB*) is the third-generation event-driven programming language and integrated development environment (*IDE*) from Microsoft for its COM programming model (Randal et.al., 2006). This program allows using and applying tables, functions and other benefits of Microsoft Excel application. The main reason in chosen this program language as a programming basis is that this program enables to reach Microsoft Excel features directly and has simple and improvable properties. It is also used commonly, nowadays.

It is determined that the use of this constituted program should be friendly, so the statements and forms are established on that way. The advantages of Microsoft Excel program are also determined because it might be developed in the future by any relevant body.

The fundamentals of design calculations and general concept are based on book, “Mine Winding and Transport” by Walker (1988). The main reason to refer this book as a basis of design calculations is that there are very limited sources/references in literature giving detailed data related to hoist design.

The process flow of design model has been illustrated in Figure 3.1. This figure might be called as a general process flow of designed model. Each stage has some details and will be discussed, separately. Generally, user must type some inputs related to actual operation parameters and basically, program has some default parameters, which are decided after many calculations, legislation surveys and scientific researches. Meanwhile, user should decide some of the characteristic parameters of shaft infrastructure and then the program starts to run several prepared macros, sequentially as shown in Figure 3.1.

It is designed that program is opened with a front window to user, so all user inputs can be typed and selected on that screen. There are some small boxes each referred to individual input parameter. All required user inputs can be keyed in these specific boxes.

On the other hand, program has some default inputs such as wire grade of rope, safety factor and so on. These default parameters are recorded into this program and user has no right to change these parameters. However, the program is provided with flexibility such that any administrator can change these parameters.

In general, when the user inputs and selections are done and the model let be run, designed model starts to run and compute several macros and series of formulas. These all macros and calculations are done on the screen background, so user cannot see these computations. The first macro, which the model starts to run is the rope macro process. Then, velocity, moment of inertia of headsheaves (if ground mounted selected), slippage acceleration, moment of inertia of guide pulley (if tower mounted selected) and friction wheel and motor macros are run, consequently. There are also calculated some parameters in Excel sheets by comprised formulas. The all design outputs according to essential parameters and minimum required motor power are listed in an output screen in the output cards corresponding to all available rope types, finally. The processes and these input and output screens are discussed in coming chapters but the general flow of these processes are drawn in Figure 3.1.

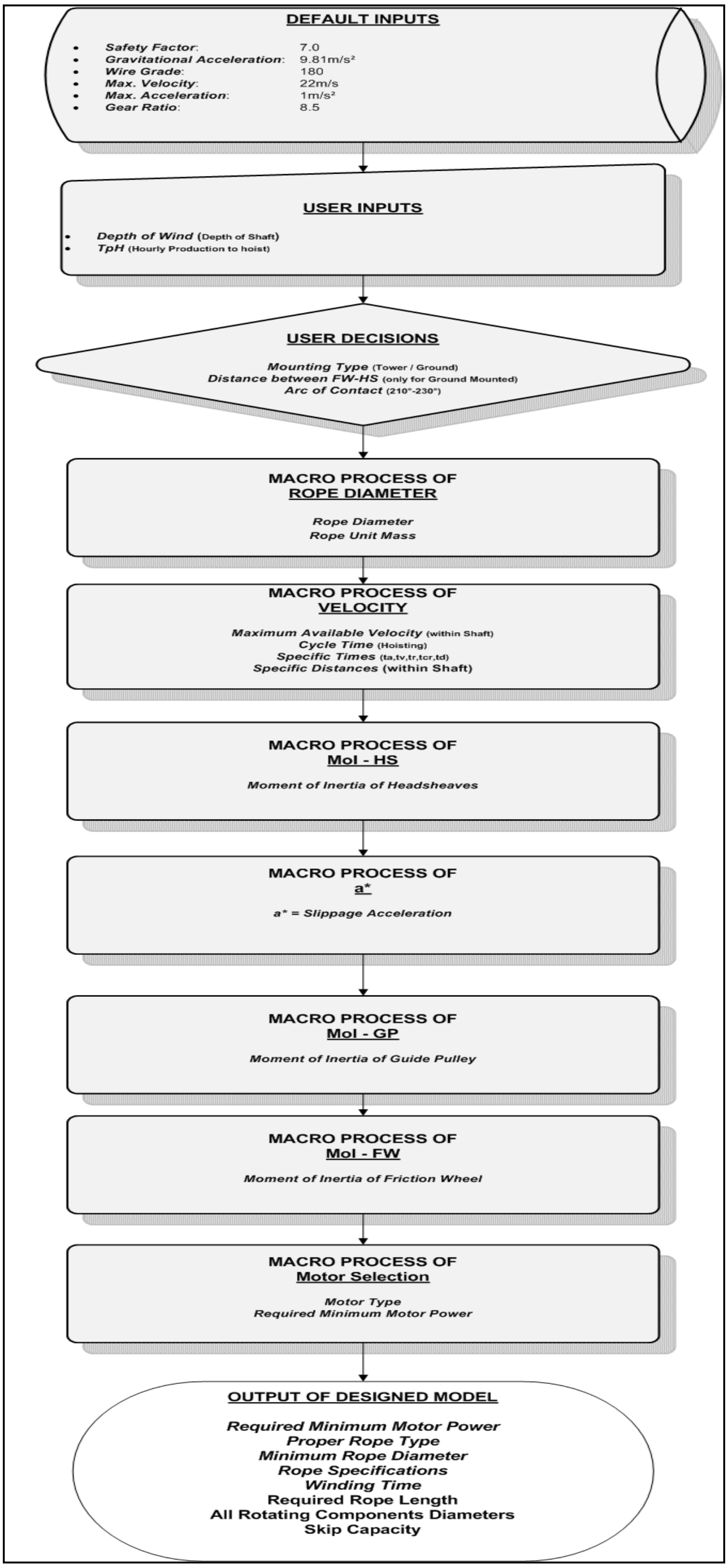


Figure 3.1 General Process Flow of Software Model

3.2 Winding Rope Design Process

The designed program is started to compute "*Winding Rope Macro*" at first, as shown in Figure 3.1. Main outputs of this macro are rope specifications. It is aimed to compute, by aid of this macro, proper rope diameter and its unit mass. The mathematical base of this macro is referred to book "*Mine Winding & Transport* ". In this book, the wire rope formula is denoted as in (Equation 3.1) (Walker, 1988);

$$\mathbf{SF.\ (M + (D.C_m.d^2)).g/1000 = C_s.d^2}$$

(Equation 3.1)

The descriptions of the values are;

SF:	**Safety Factor = 7.0 [constant]**
M:	**Mass of Loaded Conveyance [kg] =Mass of Empty Conveyance + Mass of Payload**
D:	**Depth of Wind [m]**
Cm:	**Mass Constant of Rope**
Cs:	**Strength Constant of Rope**
g:	**Gravitational Acceleration [m/s^2]**
d:	**Diameter of Rope [mm]**

The rope diameter is highly dependent on some specific variables, these are; safety factor, mass of loaded conveyance (summation of its mass and its payload), depth of wind, gravitational acceleration.

Safety factor or factor of safety is a figure used in structural applications that provides a design margin over the theoretical design capacity. The factor of safety allows for uncertainty in the design process, such as calculations, strength of materials, duty and quality. The factor of safety is equal to the strength of the component divided by the load on the component (Burkot et.al., 2011). Constructions should be strong enough to resist loads and disturbances exceeding those that are intended. A common way to obtain such safety reserves is to employ explicitly chosen, numerical safety factors (Hanson, 2007). The static safety factor in friction winding systems must not be least than 7.0 for material hoisting and 9.5 for human hoisting (Council of Ministers, 1984). The model is designed to consider only for material hoisting, so the parameter for safety factor in this model is confirmed 7.0 as a constant.

Shaft conveyances shall be constructed of steel or other metal of equivalent strength material (CDIR, 2004). The skip is the most efficient way to hoist ore from underground to surface (Atlas Copco, 2002). The two most important values in determining skip capacity (mass of payload which is loaded in it) for a given hoisting rope are the allowed static factor of safety and the skip factor; ratio between the empty skip mass and rock payload (Rebel et.al., 2006). The ratio between empty conveyance mass and payload of conveyance is confirmed as 1 for this designed model. This confirmation is sourced by Walker (1988). This ratio might change according to sizes of the skip and whether is made of. However, general assumption is stated that the skip mass equals to skip payload by Walker (1988);

Skip capacities and sizes are varied belong to their payload capacity and other design features such as diameter of shaft, skip filling station sizes, material amount (tonnage/time) to load a skip in one occasion. The size of any skip is dependent mainly on its capacity. Proper skip sizing is required to ensure production rates can be met and depends mainly on cycle times and hoisting availability. Particular skip sizes are chosen on hoist system type, required daily tonnage, deepest hoisting distance and optimum line speed. It is important to consider waste tonnages as well as ore tonnages. Skip sizes are often limited by shaft diameter and other necessary shaft compartments (Queen's University, 2009).

It might be stated that the skip capacity is more effective than skip sizes on conveyance parameter determinations. In skip capacity selection, the most important factor is the amount of tonnage per hour for winding cycle. The skip capacity should not be too small (i.e. 100 kg) or too large (70 tons) due to other design parameters. Motor powers, diameter of shaft and some others might be listed for these factors. Hence, the skip masses are chosen to vary from 1,000 kg to 30,000 kg for this model.

The bottom limit of mass of skip is selected 1,000 kg and the upper one is selected 30,000 kg. These limits are determined to cover any design of feasible underground shaft system in real mining operations and applied equipments. Due to this determination, the mass of loaded conveyance (M) is herein between the limits of 2,000 kg and 60,000 kg.

The depth of wind of any mine depends on some factors. The main one is the depth of ore. It directly affects the depth of wind. The size and position of the ore are important factors, too. Surface conditions, like topography or overburden formation, decision of access type, rock conditions, plans of mine layout and design are the other remaining factors which affect the depth of wind.

In common applications, a friction hoist with two skips in balance may be suitable for a hoisting distance as shallow as 400 m (Queen's University, 2009). The practical operating depth limit for a friction hoist is 1,700 m for balanced hoisting and 2,000 m for counterweight hoisting. Beyond these depths, rope life may be an expensive problem (Vergne, 2003) (a). It can be stated that the friction winding systems are proper and effective for the moderate depth interval of 400 m to 2,000 m. However, the designed model program gives a permission of to determine any depth for its users but these practical limits should be in mind.

A wire rope is made up of the basic components illustrated. The terms used to describe these component parts should be strictly adhered to, particularly when reporting on the conditions of ropes. Size, production method, lay and type of rope are significant to identify any wire rope property. The main components of rope are illustrated below (Figure 3.2).

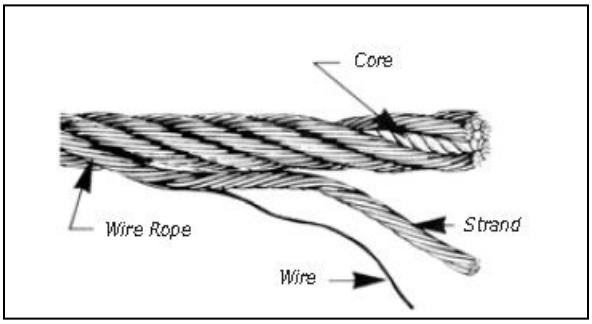


Figure 3.2 Composition of Wire Rope
(A. Noble & Son Ltd., 2011)

Each individual wire is arranged around a central wire to form a 7-wire strand. Six of these strands are formed around a central core to make a wire rope (Figure 3.3).

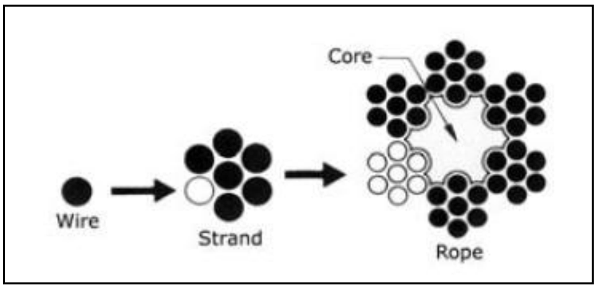


Figure 3.3 Components of Wire Rope
(A. Noble & Son Ltd., 2011)

The rope is specified as 6x7 (6/1) – i.e. six strands each of seven wires. The size and number of wires in each strand, as well as the size and number of strands in the rope greatly affect the characteristics of the rope. In general, a large number of small size wires and strands produce a flexible rope with good resistance to bending fatigue. The rope construction is also important for tensile loading (static, live or shock), abrasive wear, crushing, corrosion and rotation (A. Noble & Son Ltd., 2011). Since hoisting in underground mines requires extra safety and engineering precautions, steel is used as the common rope material. This raw material usage has impact on mass values of rope types.

Mass of rope is a self-mass and characteristic of each individual rope, which has a direct effect on hoisting system design. Each rope type has specific mass and this value depends on the rope type and the diameter.

Every rope used on a friction winder should be non-destructively tested to determine that the rope has not fallen below its required factor of safety (NSW Department of Mineral Resources, 2002). There are many types of rope, which are divided into groups by their strand and laying or spinning wires around this strand. The wires used in the making of winding and haulage ropes in mines are made from steel rods which are drawn through dies of tungsten carbide. After drawing, the wires are laid up in special machines into strands and the strands again laid up into the finished rope. For practical purposes six or more strands are arranged around a core. The center core may be a further steel strand (for heavy duty especially), a moulded plastic extrusion or a fiber core (for lighter duties) (Walker, 1988). The strength constant "Cs" which depends on rope type and wire grade, and the mass constant "Cm" which depends on rope type are important constants. The breaking strength of the rope in kN can be estimated by the relation of "Cs x d^2" where d is the rope diameter in mm and the rope mass per unit length in kg/m can be estimated by the relation “Cm x d^2”. The tensile strength (known as Grade) will depend on the purpose for which finished rope is to be employed. The grade is expressed in different terms; kgf/mm^2, i.e. 180 Grade indicates tensile strength of 180 kgf/mm^2, or in SI units, 1765 N/mm^2 (Walker, 1988). The following grades of hoisting ropes according to their applications have been found satisfactory (Table 3.1).

Table 3.1 Common Purposes and Related Grades of Hoisting Ropes (Walker, 1988)

Grade [N/mm^2]	**General Purpose**
1765	Winding and Balance Rope
1570	Winding and Haulage Rope
1470	Aerial Track Rope
1079	Balance Rope
773-883	Guide and Rubbing Rope

In general, steel wire ropes can be grouped into three; stranded, flat and locked coil. Each rope group can be detailed in subgroups. Flat ropes have expensive form of construction and have rare application (Walker, 1988). However, flattened strand and locked coil types have applications in friction winding systems in real mining applications due to convenient experiences and operations.

Flattened (triangular) strand ropes; consist of six strands laid around a centre core. The wires forming the strands are laid on a triangular shaped core consisting of a single shaped wire or three or more round wires. This compact form, when used in conjunction with a main fiber core, has a cross section of which about 62% consists of steel. This is about 10% more steel than that in a round strand rope of equal size and is stronger by the same amount when the material used has a tensile strength of the same order. This type is favored in koepe winding (Walker, 1988). This type of rope is only produced in Lang’s lay (Figure 3.4).

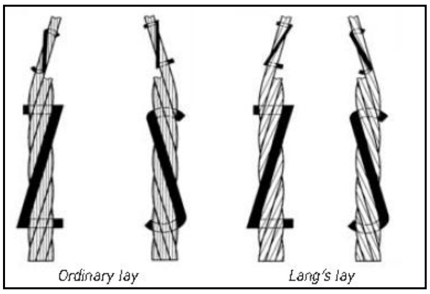


Figure 3.4 Lay Types of Wire Ropes (A. Noble & Son Ltd., 2011)

This construction has improved wear and crushes resistance and has wide application in winding and haulage systems (A. Noble & Son Ltd., 2011). Hence, triangular (flattened) strand ropes show less wear and more life span than the equivalent size round strand ropes (Figure 3.5).

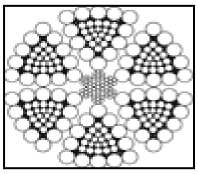


Figure 3.5 Section View of Triangular Strand (Severstal Metiz, 2010)

Locked coil ropes; may be considered as a single straight strand containing such number of wires as are necessary to produce the desired strength. The rope is built up round a single central wire around which are laid a number of concentric layers, having a variety of shapes and sizes. The outer layer is always constructed of full-lock wires, which by their interlocking action impart a smooth exterior to the rope, thus minimizing external wear. The full-lock wires are retained in the event of breaking of some ropes. Lubricant is also more effectively sealed the rope (Figure 3.6).

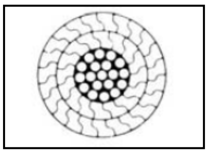


Figure 3.6 Section View of Locked Coil (Severstal Metiz, 2010)

Flattened (triangular) strand and locked coil type wire ropes are selected for friction hoisting design program. Turkish Standard Institute's standard book, "TS 1918", has been based on for these rope kinds. (TSE, This standard has been published for steel wire ropes for general purposes. This definition covers the hoisting ropes in underground mines, too. Although there are other types of steel wire ropes for flattened strand and locked coil types in the literature, this study is limited within this standard. The main reason for this is that this study aims to be applicable and to support convenient operations. The remaining types other than coverage of this standard cannot be applicable in mining operations in Turkish Republic due to legislative sanctions.

The existing rope types in this study are;

- Locked Coil 18 x 7 (LC_18x7)
- Locked Coil 36 x 7 (LC_36x7)
- Locked Coil 10 x 10 (LC_10x10)
- Triangular Strand 6 x 8 (TAS_6x8)
- Triangular Strand 6 x 9 (TAS_6x9)
- Triangular Strand 6 x 22 (TAS_6x22)
- Triangular Strand 6 x 23 (TAS_6x23)
- Triangular Strand 6 x 25 (TAS_6x25)
- Triangular Strand 6 x 28 (TAS_6x28)
- Triangular Strand 6 x 31 (TAS_6x31)

Wire ropes with 180 grade are considered in the model since they are the type generally used for hoisting.

3.3 Velocity and Time Parameters Design Process

Velocity is an important parameter in terms of hoisting systems since most of the parameters are dependent on it and it has a limit factor. In today's mining applications, the hoisting systems are working in the limits of this parameter. Velocity is also determined with the available motor powers and also limited with the slippage acceleration in friction hoisting systems.

Slippage acceleration (*will be marked as a* in the following sections*) might be identified as the limit acceleration value. If the actual system acceleration is higher than this value, the rope, which contacts with the friction wheel starts to slip. Hence, velocity is very important parameter for the purposes of safety and economic considerations.

Suchard (1999) stated that the maximum attainable speed for a friction hoist that can be safely obtained with today's technology is 19 m/s. The upper limit is taken as 22 m/s in the model considering the available motor power.

On the other hand, there should be a lower limit for velocity in hoisting systems. This lower limit was considered according to creep velocity. The creep velocity, which is the lower limit, has been assumed as 1m/s according to Walker (1988).

The hoisting cycle can be clarified as starting to loading operation and finished unloading operation within the shaft. This cycle can be explained by illustrated typical tower mounted system configuration, figured (Figure 3.7).

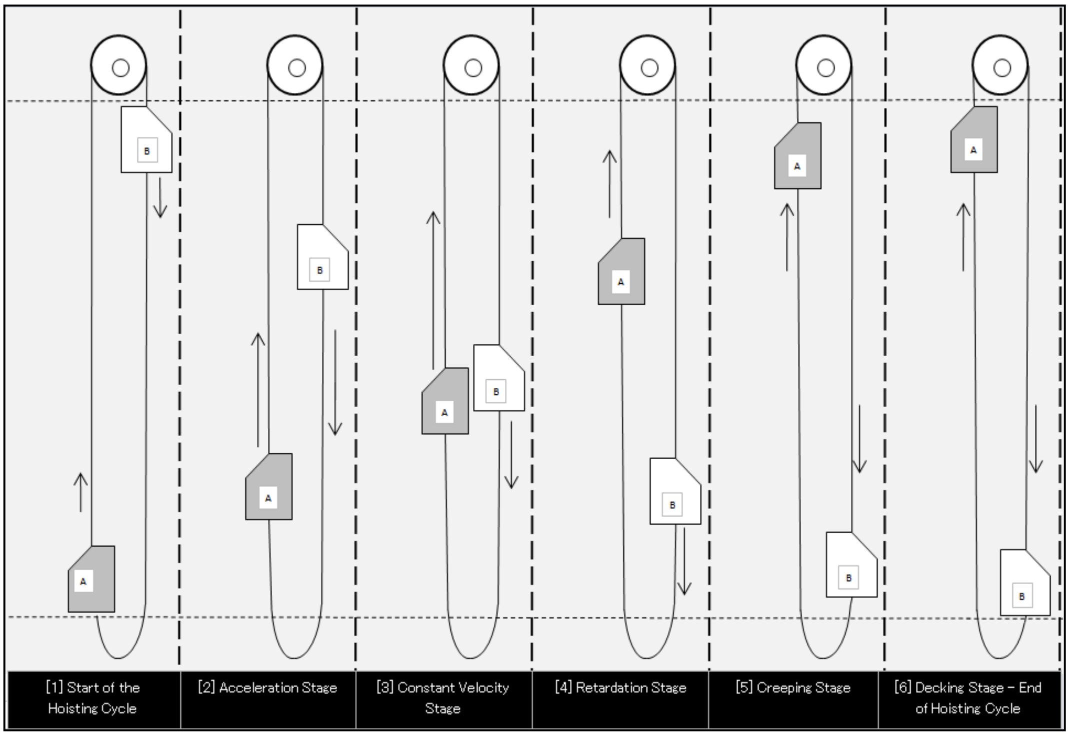


Figure 3.7 Hoisting Cycle Periods in terms of Velocity Parameter

In Figure 3.7, a typical tower-mounted friction hoisting system configuration has been drawn and shown the specific stages of hoisting cycle in terms of velocity and related parameters. From the left to the right; [1st] first sub-figure shows the starting of hoisting cycle. The skip, labeled “A”, represents a loaded skip. The skip was loaded and ready to be lifted. The bottom dashed line shows the level of the chute; where the other remaining rope below this line is called the rope below the lowest skip position. The steady loaded and the empty skip, skip “B”, at the other side of the rope, are started to move. This stage [2nd] is called as “acceleration stage”. Skip “A” is started to lift with an acceleration value and the skip “B” start to be lowered. This acceleration stage continues until reaching a constant velocity value. After reaching constant velocity value, the movement continues steady at this velocity till the retardation stage [3rd]. When the retardation stage starts, the velocity of both skips is started to decrease, steadily [4th]. This stage continues till attain the creep velocity.

Creep can be stated as “the tendency of a solid material to slowly move or deform permanently under the influence of stresses (Wikimedia Foundation Inc., 2011). This special movement should be considered for the characteristics of the steel wire rope due to the tendency of tension oscillation. This ad hoc movement takes some significant time and assumed its velocity as 1 m/s (Edwards, 1992). Hence, this creep time should be considered as a part of cycle time [5th]. In comparison to the other particular times, creep time is significantly smaller. When the skips stop totally, it is started to decking period, which occurs in steady position, where velocity has zero value.

The typical velocity vs. time graph of a hoisting cycle is shown in Figure 3.8. There is only briefed the velocity aspect of this hoisting cycle, on the other hand, there is a time aspect of this cyclic movement. To remind the definition of that hoisting cycle; “the total time it takes to move a conveyance from the bottom of its wind to the top” stated Edwards (1992).

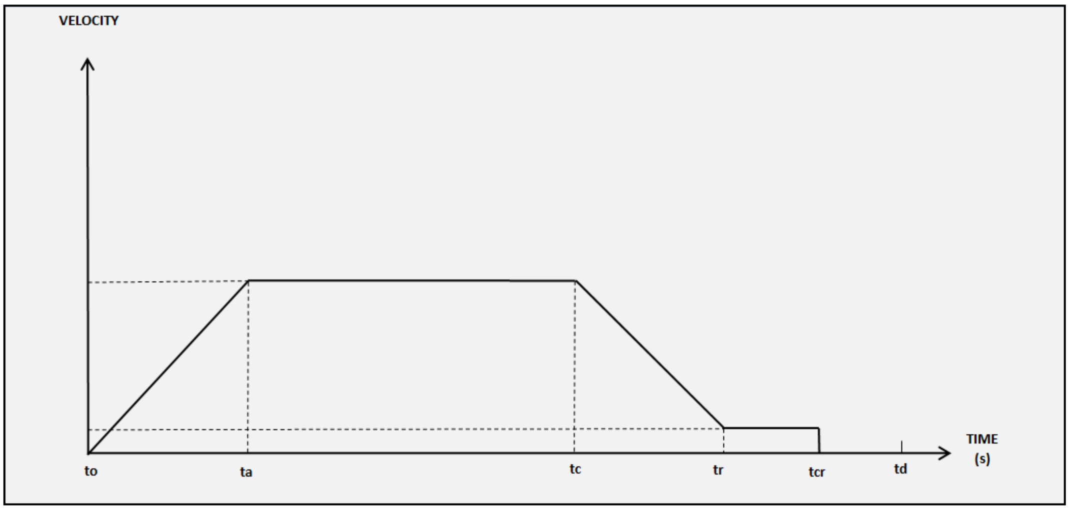


Figure 3.8 Velocity vs. Time Graph of Hoisting Cycle

Each velocity value meets with a time value in this graph. As stated in above paragraph, hoisting cycle has some parts. These periods might be identified and symbolized as "*Acceleration* / t_a ", "*Constant Velocity* / t_c ", "*Retardation* / t_r ", "*Creeping* / t_{cr} " and "*Decking* / t_d ". These periods are symbolized to ease explanations.

3.4 Headsheave(s) and Related Moment of Inertia Process

Friction hoists might be located directly over the shaft (*in tower-mounted systems*) or might require headsheave to center the rope in the shaft compartment (*in ground-mounted systems*) (Edwards, 1992). Headsheaves are called sometimes deflection sheaves and they are only installed for ground mounted friction winder operations in terms of friction hoisting. They are used mainly for positioning of the hoisting ropes in ground mounted operations. They are also used when the pitch circle diameter of the friction hoist pulley wheel (sitting above the shaft) is greater than the center to - center distance of compartments. They have the advantage of increasing the angle of contact of the ropes on the wheel and permitting a higher tension ratio before slippage occurs (Unrug, 1992).

However, headsheaves have some disadvantages; requiring additional torque during the hoisting cycle, increasing the height of headframe and putting reverse bending into the ropes which can reduce their life (Edwards, 1992).

Headsheaves diameter is very important due to its effect on the rope life. It should also be determined that it has direct effect on safety conditions, too. Hence, United States, the *Code of Federal Regulations* (CFR) specifies minimum requirements for ratio of winder/sheave diameter to rope diameter relevant to mine hoisting. It was thought that bending fatigue is the most dominant factor in rope degradation. On the other hand, research carried out independently in South Africa and USA has shown that bending fatigue is not the most dominant factor in rope degradation contrarily a corrosion-assisted fatigue process or corrosion itself is more influential on rope failure.

3.5 Slippage Acceleration Decision Process

Acceleration, as a ratio of velocity change over time, is very important parameter in hoisting design studies. The safety conditions and also economical aspects are directly affected by acceleration in friction hoisting. Friction hoisting is worked on the "friction" feature as a natural case. Friction is the force; resisting the relative motion of solid surfaces, fluid layers, and/or material elements sliding against each other (Beer and Russel, 1996). Friction occurs between the steel wire ropes and the friction wheel.

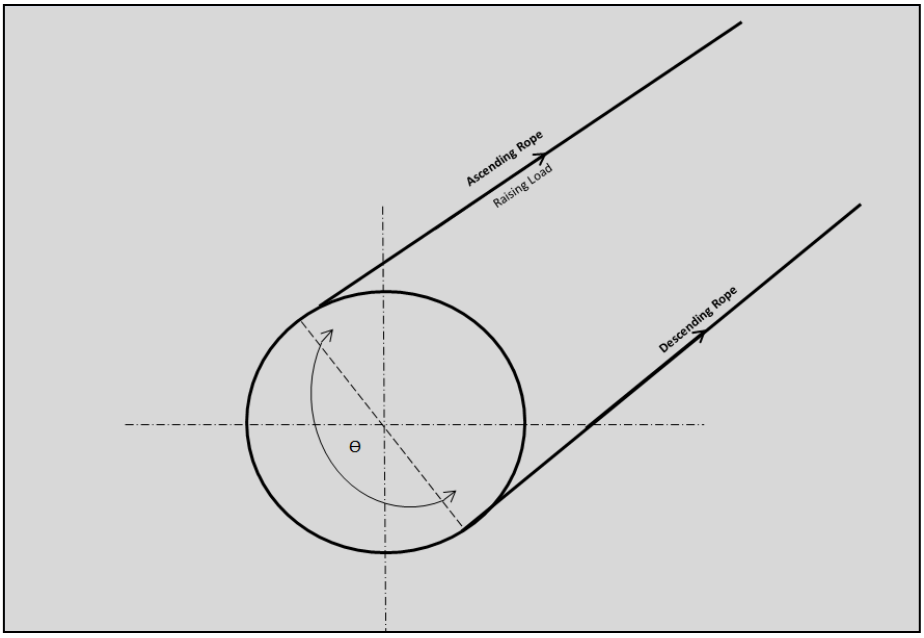


Figure 3.9 Steel Wire Rope Contact Point on Rotating Components

Slippage, where the required friction quantity anyway cannot be satisfied at this contact area (*wire rope and headsheaves or winder*), causes of losing main characteristic of friction hoisting and causes of to lose coupling effect of ropes and drums (winders or headsheaves).

Exceeding acceleration limits in friction hoisting causes of slippage which should be surveyed in detail. Hence, identifying such limits are very critical in terms of to provide safe design parameters. The upper and lower limits of acceleration are recommended as 10 m/s^2 and 0.4 m/s^2, respectively for both drum and friction winding (Walker, 1988).

These limits are very significant for the mathematical and software approaches to the slippage acceleration process. After detecting of these limits, detailing slippage acceleration would be benefit.

The calculated slippage acceleration, which is also identified as *maximum permissible acceleration* in this study, should remain within these limits for friction winding. This value is symbolized as written below in this thesis (Equation 3.2).

$$\mathbf{a^* = Slippage\ Acceleration = Maximum\ Permissible\ Acceleration\ [m/s^2]}$$

(Equation 3.2)

It is the acceleration value, where any attained acceleration over this value causes slippage in friction winding system. The acceleration is generated in the starting stage of the cycle time, which might be decided from the velocity process section. There also exists retardation period, which can be called as a negative acceleration. Acceleration and retardation periods are generally assumed to be the same, giving equal acceleration and retardation rates (Walker, 1988).

3.6 Guide Pulley and Related Moment of Inertia Process

Tower mounted and ground mounted friction winding systems differ among themselves especially not in their operation principles but in applied and installed components. Main difference is that the friction wheel (winder) is installed at ground level in ground mounted system but it is direct on the top of the shaft alignment in tower mounted system.

3.7 Friction Winder and Related Moment of Inertia Process

As mentioned at the beginning of this thesis, the drive from drum or wheel with Koepe and friction winders depends entirely on friction. The main rope is in frictional contact with a groove. Each end of the main rope is attached to the top of a conveyance and the bottoms of the conveyances are connected with a balance rope which is normally of the same mass per unit length as the main rope but is generally of a more flexible construction.

For friction hoists, the drum is called usually as friction wheel or friction winder. The friction wheel consists of a groove lined with friction material to resist slippage. In the past, this material was wood or leather. At present, polyurethane, PVC, or combination blocks are used (Edwards, 1992).

In early installations, the hoist was mounted on the ground, and a single rope was wound around the drum and over the headsheaves to the conveyances, in a balanced arrangement. In addition, a tail-rope of the same weight per unit length as the head-rope was suspended in the shaft below each conveyance. Thus the only out-of-balance load was the payload. As hoisting loads became larger, the number of head-ropes and headsheaves increased to the point where it became more practical to install the hoist in the headframe directly over the shaft. In order to bring the rope centers in line with the compartment centers, deflection sheaves must also be installed in the headframe below the hoist (Edwards, 1992).

The diameter of friction winder is very important. For friction hoists, the drum must be sized to meet statutory requirements for rope-to-drum ratios and must be wide enough to carry the required number of ropes (Edwards, 1992).

Friction wheel diameters have been established considering the recommended drum/rope diameter ratios for the locked coil and flattened strand type ropes used in friction winding (Walker, 1988).

3.8 Motor Selection and Required Power Considerations

The traditional mode of driving the winding system was by means of the steam winding engine which has now virtually disappeared. Three types of drives are currently for powering the winder; the a.c. slip-ring motor, the d.c. motor with a Ward Leonard controlled motor generator (MG) and a d.c. motor with a thyristor convertor (Walker, 1978).

That would be better to specify each individual motor type at the beginning of this section. The motor types, which are commonly used in winding systems, are going to be explained next.

3.8.1 The A.C. Motor Drive

The simplest electrical drive presently available for mine hoists is the a.c. system. The common drive for winders is by means of slip-ring induction motors. When used for winding, the stator is switched direct to the three phase network, the current and torque being limited by the resistance. The supply most commonly used for motors is three phases, 50 or 60 Hz, at 415 V, 3.3 kV / 6.6 kV / 11 kV. The choice of voltage is dictated by the supply available, the size of the motor and by economic factors (Walker, 1988).

The a.c. slip-ring motor is supplied via an automatic circuit breaker and stator reversing contractors designed to control the forward and reverse direction of wind. Motor torque and speed are controlled by controlling the resistance in the rotor circuit. A liquid controller controls the resistance in the rotor circuit of the motor and thus the motor torque and speed. It will be seen that the control circuitry must perform two basic functions; control of speed and the torque (Walker, 1988).

3.8.2 The D.C. Motor Ward Leonard Drive (Direct Coupled)

The primary power supply to most mines is a.c., the conversion to d.c. by the Ward Leonard method involves the use of a motor generator (MG) set. This is a well proven system and can be applied to the largest electric winders.

Drives of this type are generally employed in the 70 to 7000 hp range. In the lower hp ranges, up to 250 hp, d.c. motor voltage usually is 250 V. For larger motors 500 V, 600 V or 700 V or more may be used (Walker, 1988).

This system is economic in operation since the resistance losses are only in the field circuit and it provides a very exact control of the winder speed from maximum down to zero. It also allows regenerative braking to be applied. Among the major disadvantages of using a d.c. drive is that the motor may be directly coupled to the drum shaft and thus does not need gears as does the a.c. motor drive.

A d.c. drive with Ward Leonard system is mostly costly in comparison to an a.c. drive motor. The main reason for that is; a d.c. motor requires three large electric machines however a.c. geared winder requires only one of them.

3.8.3 The D.C. Drive with Gears

D.C. drive can also be applied with gear units. Since the motor is not directly coupled, there are speed limitations; on the other hand, the system is cheaper because the required d.c. motor has much less inertia as compared to a direct coupled motor.

3.8.4 The D.C. Motor Thyristor Convertor Drive

The above remarks for Ward Leonard drives are equally applicable to the thyristor convertor drive; the differences are in the method of supplying d.c. power (Walker, 1988).

In essence, the thyristor is a controllable ½ wave rectifier but is smaller, cheaper, and more efficient and more rapidly used in full wave operations (Walker, 1988). The thyristor is a four layer PNPN, three terminal device which blocks in the same way as a diode in the reverse direction but also blocks in a forward direction until a signal (*firing pulse*) is applied to the third terminal (*gate*) (Rushall, 1979).

The three phases bridge, thyristor configuration (employing a non-reversing armature convertor with an anti-parallel field convertor) is the common mode employed in the mining industry for winder motor drives (Walker et.al., 1974).

The cost of installation, in terms of actual application experiences, of an a.c. drive is less than any d.c. drive. If it is looked from the viewpoint of capital cost in mining applications, an a.c. winder has benefits in contrasting to d.c. winders. However, d.c. winders are less expensive than a.c. winder in terms of operating expenditures. The choice of motor winder type might be dependent on these facts by conditions of decision makers.

3.9 Gear Drives

Drives using reduction gears should be provided with minimum number of gear units since each unit causes an additional power loss. In determining the required number of reduction gear unit, the reduction gear ratio, which is the ratio of actual motor rpm to the drum/friction wheel rpm, should be considered. The upper limit of the reduction gear ratio is given as 10:1 above which double units will be required. Gear ratio is recommended to be less than 10:1, preferably 7:1 to 8:1 for an efficient operation (Walker, 1988). This ratio has been accepted as a default and retained at “8.5:1” in the model.

CHAPTER 4

MATHEMATICAL BASES & METHODOLOGY

4.1 Winding Rope Design Parameters Calculations

The rope formula is revised for this study and separated into three equations to establish a new approach to rope specification calculations (Equation 4.1).

$$\mathbf{SF. (M + (D.C_m.d^2)).g/1000 = C_s.d^2}$$

(Equation 4.1)

This formula is separated into three sub-formulas as shown below;

$$\mathbf{A = SF. M. g / 1000 / Cs}$$

(Equation 4.2)

$$\mathbf{B = SF. D. g. Cm / 1000 / Cs}$$

(Equation 4.3)

$$\mathbf{d = \sqrt{} A / (1 - B)}$$

(Equation 4.4)

This grouping aids to compute rope diameter in one equation as it is seen at last row (Equation 4.4) for “d”, which is rope diameter.

This wire rope formula can also be re-arranged for safety factor values. After this arrangement, the equation is stated as;

$$SF = 1000.\ Cs.\ d^2 / ((M + (D.\ Cm.\ d^2)).\ g$$

(Equation 4.5)

The original wire rope formula is now ready to process in Microsoft Excel program after these grouping and re-arrangements. These operations aid to apply this rope formula in Excel, effectively. It is expected that to have the aimed value, rope diameter, by using Excel applying these operated equations.

Each rope type has a characteristic rope table. These tables include mass constant "Cm", strength constant "Cs" and calculated unit mass (kg/m) and the breaking strength (kN) values according to rope diameters, illustrated an example table in Table 4.1.

Table 4.1 Example Wire Rope Table (6 x 28 Triangular Strand Wire Rope)

6 x 8 Triangular Strand			
Rope Diameter	**Cm**	**Cs**	**Unit Mass**
(mm)		**Wire Grade 180**	**[kg/m]**
13	0.0042	0.71	0.709
14	0.0042	0.71	0.822
16	0.0042	0.71	1.070
18	0.0042	0.71	1.360
19	0.0042	0.71	1.510
20	0.0042	0.71	1.680
21	0.0042	0.71	1.850
22	0.0042	0.71	2.030
24	0.0042	0.71	2.420
26	0.0042	0.71	2.830
28	0.0042	0.71	3.280
29	0.0042	0.71	3.530
32	0.0042	0.71	4.300
35	0.0042	0.71	5.140

The data in these wire rope tables are sourced from TSE-TS 1918 "*Steel Wire Ropes General Purposes*" Standard Booklet (TSE, 1997).

The re-arranged equations are set into Excel sheets specifically to each rope type. Each rope type has specific Excel sheet in this program. The re-arranged equations are modified and set into by these tables. So, the tabulated equations for "*A", "B"* and *"d"* are computed in these Excel sheets.

4.2 Time and Velocity Parameters Calculations

The total cycle time per wind can be calculated by hourly production to be hoisted which will be provided by program user. The unit of this value is in tons.

First of all; number of hoisting cycles per hour should be calculated as shown below in (Equation 4.6);

$$\mathbf{CpH = TpH / PL}$$

(Equation 4.6)

CpH : Winding cycles per hour

TpH : Hourly production to be hoisted [kg/hr]

PL : Payload [kg]

Then, total time of one cycle (wind) is calculated.

$$\mathbf{CT = 3600 / CpH}$$

(Equation 4.7)

CT : Cycle Time [second]

The cycle time of friction hoisting is composed of *Acceleration, Constant Velocity (Maximum Velocity), Retardation, Creeping* and *Decking,* generally. This winding time starts immediately after loading or unloading but it starts from full-stop position (where velocity value equals to zero). Then, it is assumed that all moving units of hoisting system (skips, ropes and other small attachment components) are gained speed at a constant acceleration. When these components are reached the maximum attainable velocity within the shaft through winding engine(s), the velocity remains constant till the next stage. The next stage is retardation (deceleration), where these components are started to slow down. They are retarded till the creeping velocity, which is assumed as

1m/s for this study according to Walker (1988) and Vergne (2003). The creeping stage has smaller time in comparison to other stage time but it cannot be ignored. After creeping, the decking time starts. Decking is happened at zero-velocity, which the all components are in steady position. The loading – unloading operations will be in operation at this stage. After completing decking stage, a new cycle or winding is started within the shaft. The position of a premier skip might be at the top of the shaft or at the bottom of it. It might be loaded or unloaded. The whole cycle starts whenever the skips and other components are started to move from zero-velocity.

The cycle stages should be identified and explained in details since the all required outputs of designed program are highly dependent on these time and velocity values.

4.2.1 Acceleration Period

It is the positive rate of change of velocity over time, physically. Acceleration happens while the hoisting units are at full-stop condition and started to gain velocity till the constant velocity. As explained earlier in Chapter 3.5, the lower and upper limits of acceleration are taken as 0.4 and 1.0 m/s^2. As a reminder, Figure 4.1 shows the response of the acceleration stage both in shaft figure and velocity versus time graph.

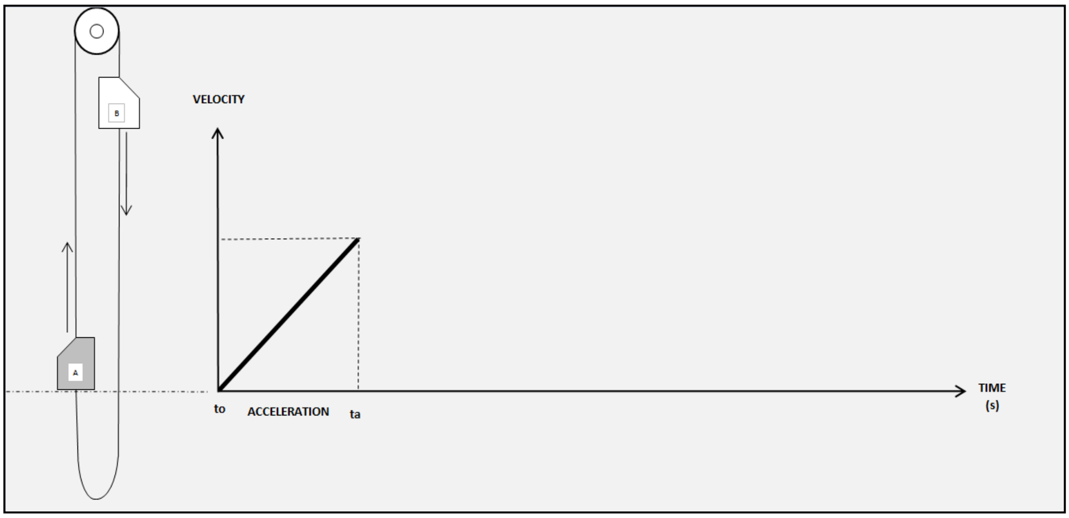


Figure 4.1 Acceleration Period

In terms of formulation, the acceleration can be shown as (Equation 4.8;

$$\mathbf{a = V_{max} / t_a}$$
(Equation 4.8)

V_{max}: Maximum velocity [m/s]

t_a: Acceleration time [s]

4.2.2 Constant Velocity Period

In this stage, the velocity remains at constant value. This constant velocity value would be the attainable maximum value of velocity in hoisting system. Figure 4.2 shows the constant velocity period.

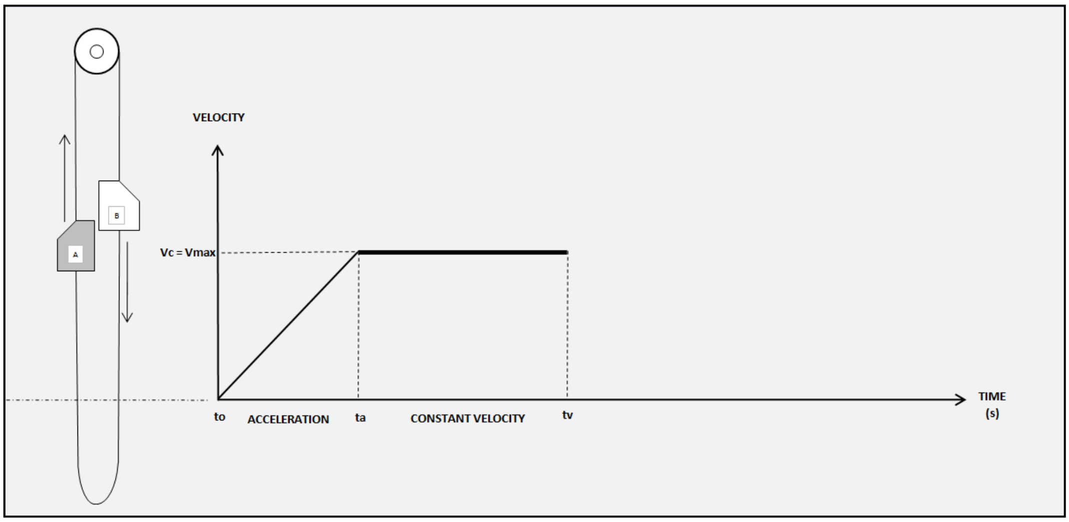


Figure 4.2 Constant Velocity Period

As it seen in the Figure 4.2 Constant Velocity Periodthe constant velocity period is started after acceleration stage. The whole system starts from the steady condition and reaches the attainable maximum velocity and continues its motion at constant velocity. This value is also called maximum velocity. The lower and upper limits of hoisting speeds are taken as 1.0 - 22.0 m/s as explained in Chapter 3.3.

4.2.3 Retardation Period

Retardation or deceleration means the negative rate of change of velocity over time, physically. In terms of winding operations, retardation is the slowing down of the velocity at constant rate. This period continues until the creeping period. Retardation period shown in the Figure 4.3.

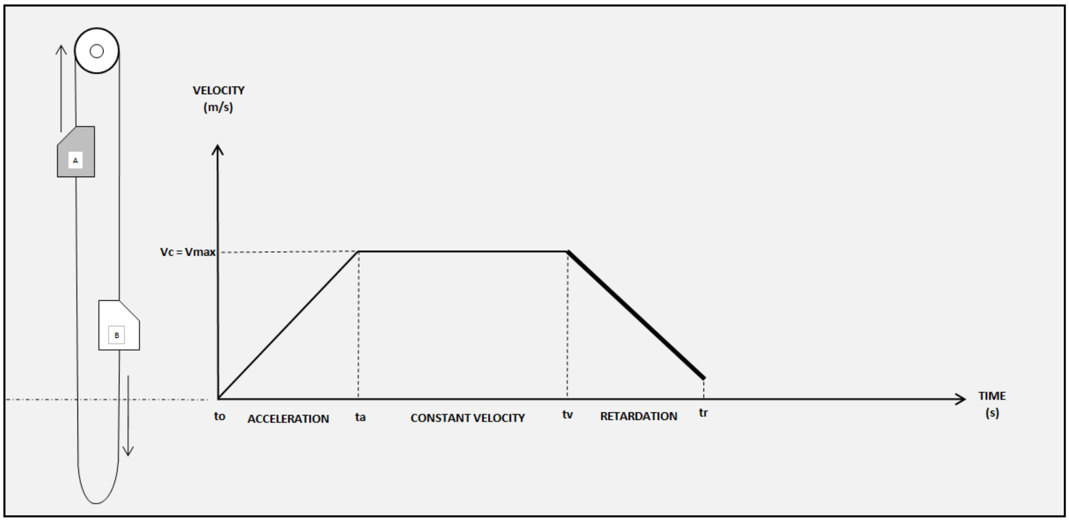


Figure 4.3 Retardation Period

$$\mathbf{r = (V_{max} - V_{cr}) / t_r}$$

(Equation 4.9)

r: **Retardation [m/s^2]**

V_{max}: **Maximum velocity [m/s]**

V_{cr}: **Creeping velocity [m/s]**

T_r: **Acceleration time [s]**

4.2.4 Creeping Period

Creeping period is recommended at the end of the hoisting for smooth landing, during which the rope runs at low speed (1 m/s) for about 3 seconds (Walker, 1988). Figure 4.4 shows the creeping period.

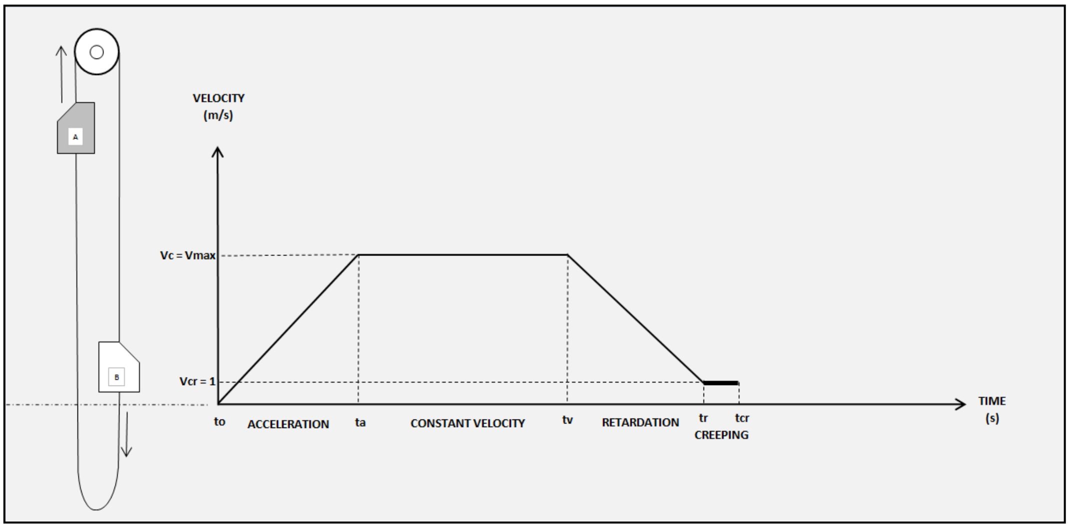


Figure 4.4 Creeping Period

4.2.5 Decking Period

After creeping, the whole system stops and remains steady. The velocity value is zero at this stage, during which loading and unloading operations are carried out.

Decking time depends on the skip type and the loading facilities in practice but decking time is recommended to be considered as one second for each ton of skip capacity plus one second extra (Walker, 1988).

$$\mathbf{t_d = (PL / 1000) + 1}$$

(Equation 4.10)

t_d: **Decking time [s]**

PL: **Skip payload [kg]**

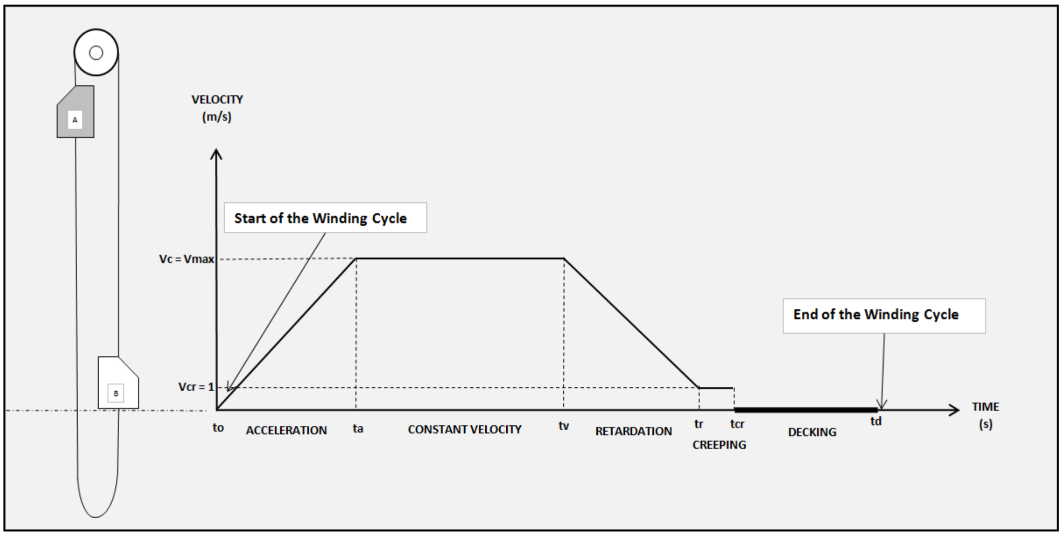


Figure 4.5 Decking Period

4.3 Headsheave and Its Moment of Inertia Design Calculations

Ground mounted friction systems require two headsheaves situated at the top of the shaft. Headsheaves diameter can be taken as equal to the friction wheel diameter.

HSD = FWD

(Equation 4.11)

HSD: Headsheave diameter [mm]

FWD: Friction wheel diameter [mm]

Moment of inertia of each headsheave is calculated from its diameter, using the moment of inertia table for headsheaves. For mid-size headsheaves, moment of inertia is determined by interpolation.

4.4 Guide Pulley and Its Moment of Inertia Design Calculations

A single guide pulley will be required to increase the contact angle if the friction wheel is tower mounted. Guide pulley diameter can be taken as 0.9 x friction wheel diameter (Walker, 1988). (Equation 4.12 shows the calculation of guide pulley diameter in the model.

$$\mathbf{GPD = FWD \times 90\%}$$

(Equation 4.12)

GPD: Guide Pulley Diameter [mm]

FWD: Friction Wheel Diameter [mm]

Moment of inertia of a guide pulley is calculated from its diameter, using the moment of inertia table for guide pulleys given in Appendix B. For mid-size headsheaves, moment of inertia is determined by interpolation.

4.5 Friction Wheel and Moment of Inertia Design Calculations

As the other rotating wheels in friction hoist system, moment of inertia of friction wheel is directly related to its diameter, and the diameter is related to the rope type and diameter. Table 4.2 shows the recommended friction wheel to the rope diameter ratio intervals for locked coil and flattened strand ropes (Walker, 1988).

Table 4.2 Relation of Friction Wheel Diameter and Rope Diameter

Rope Type	Recommended Diameter
Locked Coil	FWD = d x (100-115)
Triangular Strand	FWD = d x (80-90)

(Equation 4.13 shows the accepted calculation of friction wheel diameter in the model.

FWD = d x 115 (for Locked Coil)

FWD = d x 90 (for Triangular Strand)

(Equation 4.13)

d: Rope diameter [mm]

FWD: Friction wheel diameter [mm]

Moment of inertia of a friction wheel is calculated from its diameter, using the moment of inertia table for friction wheels. For mid-sizes, moment of inertia is determined by interpolation.

4.6 Slippage Acceleration Criteria Decision and Calculations

The acceleration stage can be explained as lifting the loads at the start of the winding. At the start of the lifting, there are some effective loads due to the dynamic and static forces (Figure 4.6).

To establish a safe operation with friction systems, there must be no possibility of slip. The ratio of the tensions between the loaded and empty sides should be sufficiently small not to cause slip. To achieve this; the system should be provided with a tail or balance rope.

To guard against slipping of the rope on the Koepe wheel; the ratio of the pulls on the ascending and descending parts of the rope under worst conditions of loading must not exceed an amount determined by the angle of contact and the coefficient of friction (Please see Figure 3.9) (Walker, 1988).

$$\mathbf{P1 / P2 = e^{\mu . \theta}}$$

(Equation 4.14)

e: **Naperian base (=*2.7183*) [Constant]**

μ: **Coefficient of friction between the rope and the friction wheel**

θ: **Arc of contact [Radians]**

P1: **Tensions in the ascending rope**

P2: **Tensions in the descending rope**

The worst condition, in terms of the nearest circumstance to slip, is at the start of the wind during which dynamic force due to acceleration exists and causes an increase in the ratio of full and empty sides of the hoisting rope.

The effective loads at the start of hoisting have been shown in Figure 4.6.

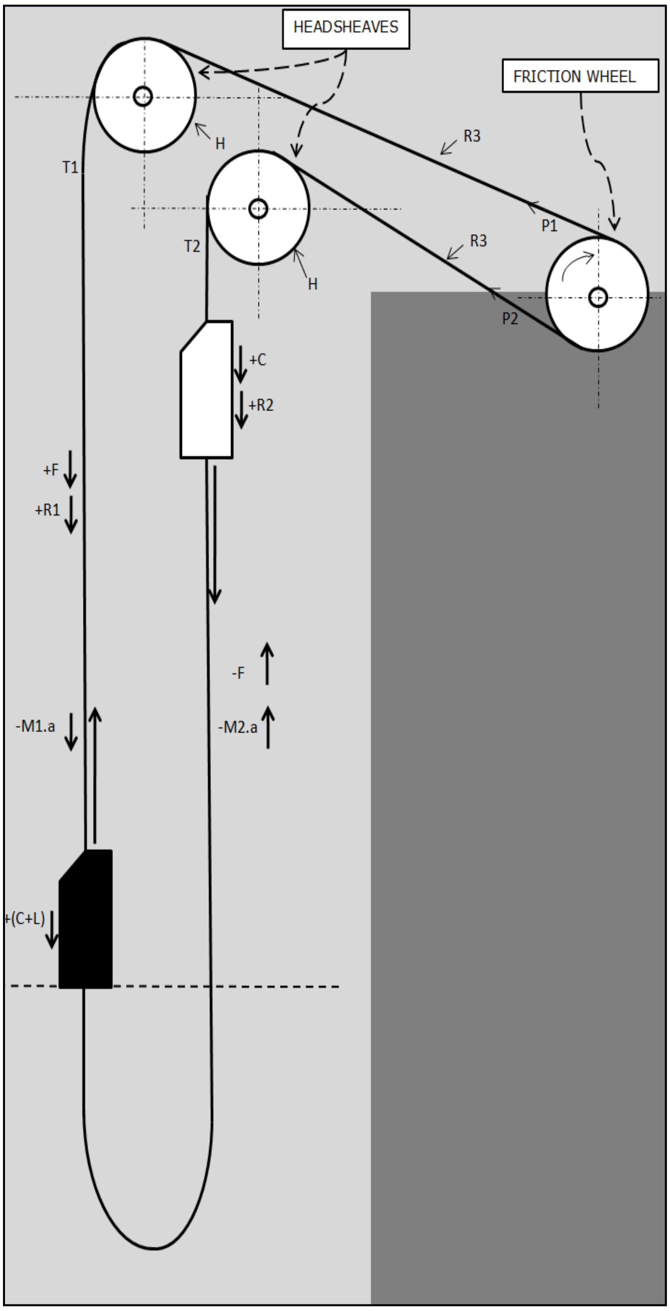


Figure 4.6 Effective Forces at Lifting Load at the Start of Winding (Walker, 1988)

R1: **Mass of rope on ascending side [kg]**
R2: **Mass of rope on descending side [kg]**
R3: **Mass of rope between headsheaves and friction wheel [kg] (*for ground mounted only*)**
C: **Conveyance mass (*tare*) [kg]**
PL: **Payload (*mass of ore or waste in skip*) [kg]**
T1: **Static load on ascending rope *(= (R1 + C + PL).g)* [N]**

T2: **Static load on descending rope *(=(R2 + C).g)* [N]**
F: **Guide friction *(= (0.18/2 x PL).g)* [N]**
H: **Mass of each headsheave referred to rope center**
***(=(MoI /r²).g)* [N] (*for ground mounted only*)**
Mass of guide pulley referred to rope center
***(=(MoI /r²).g)* [N] (*for tower mounted only*)**
G: **H + R3 [kg] (*for ground mounted only*)**
H [kg] (*for tower mounted only*)
r: **Radius of each headsheave [m] (*for ground mounted only*)**
Radius of guide pulley [m] (*for tower mounted only*)
g: **Gravitational acceleration (=9.81) [m/s^2]**

These explanations belong to static loads; the dynamic loads due to the acceleration are the result of the total dynamic loads by multiplying acceleration. If static loads are re-arranged as M1 and M2;

$$\mathbf{M1 = (C+PL+R1+H+R3) / g}$$
$$\mathbf{M2 = (C+R2+H+R3) / g}$$

(Equation 4.15)

M1: **Total static load in ascending rope [kgm/s^2]**

M2: **Total static load in descending rope [kgm/s^2]**

g: **Gravitational acceleration [m/s^2]**

$$\mathbf{P1 = T1 + (T1+G)a/g + F}$$
$$\mathbf{P2 = T2- (T2+G)a/g - F}$$

(Equation 4.16)

P1: **Static + dynamic force in ascending rope**

P2: **Static + dynamic force in descending rope**

Rope slip will occur if the ratio is exceed shown in (Equation 4.14. So;

$$\mathbf{e^{\mu.\theta} = [T1.g + (T1+G).a/g + F.g] / [T2.g- (T2+G).a/g – F.g]}$$

(Equation 4.17)

The maximum permissible acceleration or slippage acceleration is obtained finally as in Equation(Equation 4.18;

$$\mathbf{a^* = [e^{\mu\theta}.(T_2\text{-}F) - (T_1+F)]\ .g\ /\ [e^{\mu\theta}.\ (T_2+G) + (T_1+G)]}$$
(Equation 4.18)

The above expression for slippage acceleration can be used only for ground mounted friction hoisting system.

R3 is zero value for tower mounted Koepe systems and G=H of the guide pulley. Maximum permissible acceleration equation can be obtained by rearranging as in (Equation 4.19;

$$\mathbf{a^* = [e^{\mu\theta}.(T_2\text{-}F) - (T_1+F)]\ .g\ /\ [e^{\mu\theta}.\ (T_2+G) + T_1]}$$
(Equation 4.19)

Since the limits of acceleration rate are established as 0.4-1.0 m/s^2 as explained before, slippage acceleration cannot be lower than $0.4 m/s^2$. The applied acceleration rate can be as high as 1.0 m/s^2 on the condition that it does not exceed the calculated slippage acceleration.

4.7 Motor Selection and Required Power Calculations

The motor selection stage has a special place in winding system design. Motor type and its required power directly affect of capital and operational costs. Some of motor types have lower cost than other types however they consumed more energy than others. That means, motor selection should be surveyed very carefully and in detail.

Three different types of winding motors are considered under the scope of this study. These three motors have some significant advantages and disadvantages in comparison with each other as discussed in Chapter 3.8.

The motor types considered in this study are; A.*C. drive type, D.C. with geared drive and D.C. direct drive*.

The maximum velocity is determined by the following (Equation 4.20:

$$\mathbf{V_{max} = [D - (t_{cr} \text{ x } V_{cr}) / (0{,}5 \text{ x } t_a + t_c + 0.5 \text{ x } t_r)]}$$
(Equation 4.20)

V_{max}: Maximum Rope Velocity [m/s]

D: Depth of Wind [m]

t_{cr}: **Creep Time [s]**

V_{cr}: **Creep Velocity [m/s]**

t_a: **Acceleration Time [s]**

t_c: **Constant Velocity Time [s]**

t_r: **Retardation Time [s]**

After V_{max} calculation, it is required to find out maximum rpm of friction wheel. It can be acquired by;

$$\mathbf{MRPMFW = V_{max} / Circumference = [(V_{max} \times 60) / (2\pi r)]}$$

(Equation 4.21)

MRPMFW: **Maximum rpm of friction wheel [rev/min]**

r: **Radius of friction wheel [m]**

π: **Pi constant [3.14159]**

When the maximum rpm of friction winder is attained, then linear and radial accelerations are calculated.

$$\mathbf{L_a = a = V_{max} / t_a}$$

(Equation 4.22)

L_a: **Linear Acceleration [m/s^2]**

$$\mathbf{R_a = L_a / r}$$

(Equation 4.23)

R_a: **Radial Acceleration [radians/s^2]**

The next step will be to calculate the preliminary motor power so that estimation can be made on the required motor power, and the type of the drive system can be selected. Preliminary motor power can be calculated by the following equation (Walker, 1988).

$$\mathbf{PMP = (V_{max} \times g \times PL) / Efficiency}$$

(Equation 4.24)

PMP: Preliminary motor power [kW]

PL: Payload [kg]

Efficiency: Constant [taken as 0.6 for skips]

The other important parameter is motor rpm. This parameter depends on the type of drive system, friction wheel rpm and the gear ratio if it is not direct coupled. (Table 4.3)

Table 4.3 Motor RPM According to Selected Motor Type

Motor Type	**Motor RPM**
DCDD	MRPM = MRPMFW
DCGD	MRPM = MRPMFW x RGR
A.C.	MRPM = MRPMFW x RGR

Motor rpm equals to rpm of friction wheel for DCDD type motors. If geared a.c. or geared d.c. drive systems are selected, reduction gear ratio of 8.5 will be a default value in the model as explained in Chapter 3.9.

The following step in motor power calculations is to determine the rotating and travelling inertias.

The rotating elements, which differ according to mounting type (ground or tower) are shown in Table 4.4.

.

Table 4.4 Different Rotating Components According to Mounting and Motor Types

<table>
<tr><th>Mounting Type</th><th>Motor Type</th><th colspan="4">Considered MoI of Rotating Inertias</th></tr>
<tr><td rowspan="2">Ground Mounted</td><td>AC</td><td>Headsheaves</td><td>Friction Wheel</td><td>Gear</td><td>Motor Rotor</td></tr>
<tr><td>DCGD</td><td>Headsheave</td><td>Friction Wheel</td><td>Gear</td><td>Motor Rotor</td></tr>
</table>

	DCDD	Headsheaves	Friction Wheel	N/a	Motor Rotor
Tower Mounted	AC	Guide Pulley	Friction Wheel	Gear	Motor Rotor
	DCGD	Guide Pulley	Friction Wheel	Gear	Motor Rotor
	DCDD	Guide Pulley	Friction Wheel	N/a	Motor Rotor

When the motor type is selected as *A.C.*, then total rotating inertia will be as in (Equation 4.25;

TOTAL ROTATING INERTIA (*if AC Motor Drive is selected*)
= [(MoI of Motor Rotor*) + (MoI of FW) + (MoI of Gear) + (MoI of HS / MoI of GP*)]
(Equation 4.25)

MoI of GP* (*referred to Koepe Winding*) = MoI of GP x (FWD/GPD)^2
(Equation 4.26)

The designed program presents an option to select motor number either single or double motor for program user. This option is valid both for AC and DCGD type motors. The moment of inertia of gear unit is also considered when these (AC and DCGD) motor types are selected. The consideration of motor rotor inertia for these types of motors is differed than DCDD type motor inertia. The effective motor rotor inertia of AC or DCGD type motors is calculated as in (Equation 4.27;

MoI of Motor Rotor* = MoI Rotor x RGR^2
(Equation 4.27)

If the motor number is selected as double, the motor rotor inertia is going to be multiplied by two since each motor requires a separate gear unit.

Total rotating inertia is calculated for d.c. drive direct coupled motors as illustrated below;

TOTAL ROTATING INERTIA (*if DCDD Motor Drive is selected*) = [(MoI of Motor Rotor) + (MoI of FW) + (MoI of HS / MoI of GP*)]

(Equation 4.28)

When the motor type is selected as *geared drive d.c. type motor (DCGD),* then total retarding inertia will be;

TOTAL ROTATING INERTIA (*if DCGD Motor Drive is selected)*
= [(MoI of Motor Rotor*) + (MoI of FW) + (MoI of Gear) + (MoI of HS / MoI of GP*)]

(Equation 4.29)

It is also necessary to compute the travelling inertias to obtain the total inertia in the system.

Total travelling loads in the system is;

TOTAL TRAVELLING LOAD = Mass of Payload + Mass of two Skips + Mass of all Ropes

(Equation 4.30)

Total travelling inertia can be calculated as;

TOTAL TRAVELLING INERTIA = Total Travelling Load x (FWD/2)^2

(Equation 4.31)

Total inertia of the system can be determined as in (Equation 4.32.

Σ (SYSTEM INERTIA) = Σ (ROTATING INERTIAS) + Σ (TRAVELLING INERTIAS)

(Equation 4.32)

This total system inertia value is used in dynamic torque calculation. There are two different torque kinds in winding system; static and dynamic. Dynamic torque is also defined as acceleration torque.

$$\mathbf{T_s = PL \times (FWD/2) \times 1.18 \times g \times 10^{-3}}$$
(Equation 4.33)

T_s: Total static torque [kNm]
PL: Payload [kg]
FWD: Friction winder diameter [m]
g: Gravitational acceleration [m/s^2]

$$\mathbf{T_a = Total\ System\ Inertia \times R_a \times 10^{-3}}$$
(Equation 4.34)

T_a: Dynamic torque [kNm]
R_a: Radial acceleration [rad/s^2]

To mark that dynamic, in other words acceleration torque equals to retardation torque. Power is calculated separately for each hoisting period (acceleration, constant, speed, retardation periods).

$$\mathbf{Power = [(Torque \times 2\pi \times MRPMFW) / (60 \times 0.98)]}$$
(Equation 4.35)

Torque: Total torque at each period
MRPMFW: Maximum RPM of friction wheel
0.98: Reduction gear loss factor
[*Not considered for DCDD systems*]

To simplify this power equation, the constant values are gathered and calculated. This power equation (Equation 4.36) is modified as below. The (Equation 4.35 is re-arranged as follows;

Power = [(Torque x MRPMFW) x (0.1069)] – [for AC&DCGD type motors]
Power = [(Torque x MRPMFW) x (0.1047)] – [for DCDD type motors]
(Equation 4.36)

Table 4.5 shows the total (net) torques and the power calculations at each period.

Table 4.5 Power Requirement Equations According to Time Periods

Time [s]	Net Torque [kNm]	Constant	Required Power [kW]
Ta	**Static + Dynamic**	**0.1069** (for AC&DCGD) **0.1047** (for DCDD)	**$Power_a$** =Net Torque x Constant
Tc	**Static**	**0.1069** (for AC&DCGD) **0.1047** (for DCDD)	**$Power_c$** =Net Torque x Constant
Tr	**Static - Dynamic**	**0.1069** (for AC&DCGD) **0.1047** (for DCDD)	**$Power_r$** =Net Torque x Constant
Tcr	**Ignored**	**N/a**	N/a

The process to conclude the power requirement for each stage has been tabulated. Required power refers to each specific time period within the winding operation are called as $Power_a$, $Power_c$ and $Power_r$.

Since a continuously rated motor is required in hoisting operations, RMS power is calculated considering the available equivalent time (Te) during the hoisting cycle (Equation 4.37). (Walker, 1988)

$$\mathbf{T_e = 2/3t_a + t_c + 2/3t_r + 1/3t_d + 1/3t_{cr}}$$

(Equation 4.37)

T_e: Equivalent Time [s]

t_a: Acceleration Time [s]

t_c: Constant Velocity Time [s]

t_d: Decking Time [s]

t_{cr}: Creeping Time [s]

RMS power is calculated using the (Equation 4.38;

$$\textbf{RMS Power} = \sqrt{[((P_a^2 \times t_a)+(P_c^2 \times t_c)+(P_r^2 \times t_r)) / (2/3 \times (t_a+t_r)) + (1/3 \times (t_d+t_{cr}))+t_c)]}$$

(Equation 4.38)

RMS Power: Root Mean Square Power [kW]

P_a: Required power during acceleration period [kW]

P_c: Required power during constant velocity period [kW]

P_r: Required power during retardation period [kW]

RMS power is increased by 5% considering the motor efficiency to find the required motor power (Equation 4.39).

REQUIRED MOTOR POWER = RMS POWER x (5% Tolerance) [in kW]

(Equation 4.39)

As it is stated, the required motor power is calculated on the basis of the values of preliminary motor power and the rpm values of motor. When the series of above explained calculations are done, the required power for each period (acceleration, constant velocity and retardation) are calculated. Then, these values are gathered and obtained a total required power. Due to the tolerance and efficiency determinations, this value processed in RMS value and finally the required power value is obtained.

CHAPTER 5

DESIGNED MODEL PRINCIPLES & RESULTS

5.1 Winding Rope Macro Process

Explained theoretical approach should be conveyed to programming language to construct a proper macro model. This model is targeted to be ease of operation both for user and computer sides. Hence, the constructed software model should be in a frame of clear flow. To access this clear process flow, some default inputs are set into this model for rope macro, which are the basis of this design. The default inputs are shown in Table 5.1.

Table 5.1 Default Data of Designed Software Model for Rope Macro

Data Type (Name)	Data Value	Unit
Safety Factor	7	-
Gravitational Acceleration	9.81	m/s^2
Gear Ratio	8.5	-
Wire Grade	180	-

Table 5.1 shows the data type, their value and related units for rope macro as default data. Some of the values are constant and has no units. Rope safety factor is taken as 7.0 referred to Turkish mining legislation. Wire grade is important to determine the related steel rope properties. These properties are determined for wire grade at 180. The

gear ratio, which is the rate between rpm of selected motor and friction winder rpm, is taken as 8.5 as explained before.

To understand the rope macro, above explained default data are important. Besides these default data there are required some user inputs. These inputs should be supplied by program user to run the designed software. Some of them are illustrated in previous sections but it should be reviewed in detail. Depth of wind value is used in rope macro as a primary value. This data is keyed in by a user in this model.

The macro process for steel rope consideration is run on after determining these data. The written Visual Basic language is started with the selection of some appropriate sheet and Excel row & column coordinate. At first, the sheet, which is called "*Rope*", in this software Excel document, is opened and the first row of "B" column is selected. It is issued a command that put first "*M*" value, mass of loaded conveyance, as 2,000 kg. Then, the loop command is adjusted by counter of 30. This command means that this "*M*" value will be increased till 60,000 kg by increasing 2,000 kg increments. So, it means that the skip capacities are ranged 1,000 kg-30,000 kg and this order is increased 2,000 by 2,000. In that case, the available skip capacities are 1,000 kg, 2,000 kg, 3,000 kg till 30,000 kg included.

Since each available rope type has individual sheets, called like LC_18x7, LC_36x7 and TAS_6x8, the all available characteristic data of them are written in tables in these sheets. As an example, Table 5.2 shows the included data and properties of rope type of "*Triangular Strand 6x31*", which is labeled as TAS_6x31. These sheets have been formed for each specific rope type (TSE, 1997).

Table 5.2 Winding Rope Sheet after Running Winding Rope Macro

6 x 8 Triangular Strand							
Rope Diameter	Cm	Cs	Unit Mass				
(mm)		Wire Grade 180	[kg/m]	d (calc)	A	B	SF
13	0.0042	0.71	0.709	100.0274	5966.204	0.403707	**0.20**
14	0.0042	0.71	0.822	100.443	5996.535	0.405625	**0.23**
16	0.0042	0.71	1.070	100.0321	5975.495	0.402834	**0.29**
18	0.0042	0.71	1.360	99.97663	5961.292	0.403592	**0.37**
19	0.0042	0.71	1.510	100.3528	5996.535	0.404556	**0.41**
20	0.0042	0.71	1.680	100.4931	5996.535	0.406217	**0.45**
21	0.0042	0.71	1.850	100.4523	5996.535	0.405734	**0.50**
22	0.0042	0.71	2.030	100.4458	5996.535	0.405657	**0.55**
24	0.0042	0.71	2.420	100.5045	5996.535	0.406351	**0.65**
26	0.0042	0.71	2.830	100.3819	5996.535	0.404901	**0.75**
28	0.0042	0.71	3.280	100.3598	5996.535	0.404638	**0.87**
29	0.0042	0.71	3.530	100.4717	5996.535	0.405964	**0.93**
32	0.0042	0.71	4.300	100.4867	5996.535	0.406141	**1.12**
35	0.0042	0.71	5.140	100.4597	5996.535	0.405822	**1.32**

Each of this specific rope sheets have been included “a*ctual available rope diameter*”, referred to standard book, “*Cm and Cs values*”, “*unit mass of each available rope diameter of each specific rope type*”, “*results of equations of A and B*” and “*calculated actual safety factor of rope*” under given conditions. The given conditions are defined and default program data, which have been explained in previous sections.

In designed macro, the main criterion of chosen proper rope diameter for each specific rope type is the actual “*calculated safety factor*”. This value is used as comparison value in determining whether the stated rope diameters are proper or not in given conditions.

The calculated safety factor is calculated in each Excel rope sheet and placed in the last column in to the available rope diameter row. The written macro code has been set up to compare each block; whether the calculated safety factor is greater or equal than 7.0 or not. If calculated safety factor yields above conditions then macro engine enters the yield value of safety factor, unit mass of rope and rope diameter into the appropriate blocks in “Rope” sheet. If the calculated safety factor value is smaller than 7.0, then model puts a blank into the appropriate blocks. At the end of this operation, a well-established table is drawn in the sheet of rope. An example of this sheet can be seen in Table 5.3.

Table 5.3 Well-established Rope Sheet Including all Rope Design Results

	LC_18x7			LC_36x7			LC_10x10			TAS_6x8			TAS_6x9			TAS_6x22			TAS_6x23			TAS_6x25			TAS_6x28			TAS_6x31		
M	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF	Unit Mass	d	SF
2000	1.30	**18**	7.67	1.26	**18**	7.50	1.41	**18**	7.55	1.36	**18**	7.02	1.51	**19**	7.52	1.38	**18**	7.04	1.38	**18**	7.04	1.38	**18**	7.04	1.38	**18**	7.04	1.38	**18**	7.04
4000	2.31	**24**	7.16	2.64	**26**	7.68	2.50	**24**	7.05	2.83	**26**	7.16	2.83	**26**	7.26	2.88	**26**	7.21	2.88	**26**	7.21	2.88	**26**	7.21	2.88	**26**	7.21	2.88	**26**	7.21
6000				4.00	**32**	7.72	4.45	**32**	7.79	4.30	**32**	7.20	4.30	**32**	7.30	4.35	**32**	7.26	4.35	**32**	7.26	4.35	**32**	7.26	4.35	**32**	7.26	4.35	**32**	7.26
8000				5.06	**36**	7.49										6.15	**38**	7.49	6.15	**38**	7.49	6.15	**38**	7.49	6.15	**38**	7.49	6.15	**38**	7.49
10000				6.24	**40**	7.43										7.15	**41**	7.19	7.15	**41**	7.19	7.15	**41**	7.19	7.15	**41**	7.19	7.15	**41**	7.19
12000																8.24	**44**	7.02	8.24	**44**	7.02	8.24	**44**	7.02	8.24	**44**	7.02	8.24	**44**	7.02
14000																9.80	**48**	7.11	9.80	**48**	7.11	9.80	**48**	7.11	9.80	**48**	7.11	9.80	**48**	7.11
16000																11.11	**51**	7.04	11.11	**51**	7.04	11.11	**51**	7.04	11.11	**51**	7.04	11.11	**51**	7.04
18000																12.45	**54**	7.03	12.45	**54**	7.03	12.45	**54**	7.03	12.45	**54**	7.03	12.45	**54**	7.03
20000																13.80	**57**	7.05	13.80	**57**	7.05	13.80	**57**	7.05	13.80	**57**	7.05	13.80	**57**	7.05
22000																15.34	**60**	7.08	15.34	**60**	7.08	15.34	**60**	7.08	15.34	**60**	7.08	15.34	**60**	7.08
24000																17.40	**64**	7.26	17.40	**64**	7.26	17.40	**64**	7.26	17.40	**64**	7.26	17.40	**64**	7.26

As it is seen above illustrated table (Table 5.3), the rope macro produces each proper value of rope diameter, its calculated safety factor for given conditions and the unit mass of each rope type and its ordered rope diameter for equivalent loaded mass of skip. For instance, under the given conditions, Locked Coil 18x7 rope type can assure safe conditions (means applicable) only for 1,000kg capacity skip with the 26mm diameter rope. This rope with figured diameter has 2.72kg/m unit mass and assure a 7.56 constant as calculated factor of safety. However, this rope cannot assure any safe operations for other greater payload masses under given conditions, which can be proved.

This rope table is filled for all available rope types and for all limited loaded conveyance masses according to above explained process. At the end of this process, all calculated safety factors, available, proper rope diameters and their unit masses are filled in appropriate blocks.

5.2 Time and Velocity Macro Process

A winding cycle is composed of different periods within the hoisting shaft. The whole system is started to be hoisted by acceleration. When the system reaches the targeted constant velocity, system moves at that constant velocity till the retardation stage. This retardation continues till the stopping time. Creeping comes just before the full-stop. System stops after creeping and decking operation is started.

Each of these stages is important to consider required power hence should be investigated in detail. These operation times are approached by estimation (Walker, 1988). Most of winding design might be considered by this theory. However, time periods within the winding cycle has been considered different than this estimation approach. The main reason for this difference is to apply winding design in programming language. Due to the nature of programming, there are required specific approaches rather than any estimation in any program design. Hence, this approach might be unique for this thesis study.

Each section in cycle time should have specific time and velocity values. These sections compose of the one total cycle. Macro for velocity and acceleration calculations has been constructed on the essential physic hypothesis.

To start this process, there is a requirement to consider total number of cycles in one hour "CpH" and one cycle time period "CT". Calculations of CpH and CT have been explained in mathematical approach section of this macro process. So, the constructed macro program has already recovered these parameters.

The macro program is started to run to find out these essential parameters, first of all. Then, it begins to identify each specific time period, such as t_a, t_d and also velocity values, like V_a, V_c.

Designed macro process flows by calculating variables. These variables were named differently to ease and get proper flow within the macro. This process flow might be seen in detail. Before stating the macro process, explanation for the approach might simplify the process. The written commands in time and velocity macro are stated in following series of mathematical equations; (Equations 5.1).

$$\mathbf{V_2 = V_1 = 1}$$
$$\mathbf{t_1 = V_1 / A}$$
$$\mathbf{x_1 = 0.5 * A * (t_1 \wedge 2)}$$
$$\mathbf{t_3 = (V_2 - V_3) / A}$$
$$\mathbf{t_2 = CT - t_1 - t_3 - DT - 3}$$
$$\mathbf{x_2 = V_2 * t_2}$$
$$\mathbf{x_3 = V_2 * t_3 - 0.5 * A * (t_3 \wedge 2)}$$
$$\mathbf{DV_1 = x_1 + x_2 + x_3 + 3}$$
$$\mathbf{TV_1 = t_1 + t_2 + t_3 + 3 + DT}$$

(Equations 5.1)

Variables, illustrated in above, are calculated by written commands. That should be proper to explain each variable in these commands.

In the beginning, these variables, defined specifically for macro, should be identified. The symbols have been used in macro for defining the parameters.

V_1: **Maximum velocity attained at the end of acceleration period; V_{max}**
V_2: **Maximum velocity; constant velocity; V_{max}**
V_3: **Creeping velocity; V_{cr}**
t_1: **Time interval for acceleration period; t_a**
x_1: **Hoisting distance travelled in acceleration period**
t_3: **Time interval for retardation period; t_r**
t_2: **Time interval for constant velocity period; t_c**
x_2: **Hoisting distance travelled in constant velocity period**
x_3: **Hoisting distance travelled in retardation period**
DT: **Decking time; t_d**
A: **Actual acceleration; a**
DV_1: **1^{st} depth verification; (designated for macro process)**

TV_1: 1^{st} time verification; (designated for macro process)

CT: Cycle time

First velocity value is determined by macro after defining CpH and CT. The program starts to run by defining first velocity value "V_{max}". It was commanded as set this first velocity value to maximum available velocity, which is programmed as 22m/s in the model. Creeping velocity "V_{cr}" has been already inputted as a default value (1m/s).

Time interval in acceleration stage "t_a" can be calculated by designated velocity and default inputted acceleration. The last variable for acceleration zone is the hoisting distance travelled in acceleration process "x_1". This distance is calculated by default maximum acceleration "a" and time value for acceleration process. So, all required variables are calculated by written formulas in macro for the acceleration, constant velocity and retardation periods.

At the end of these series of calculations, macro is commanded to acquire designated first depth "DV_1" and time "TV_1" verification values according to required hoisting capacities.

When the DV_1 value is obtained by starting with maximum available hoisting velocity (22 m/s), macro starts to check this verification value by comparing actual depth of wind "D". The program has been directed to check whether DV_1 value is smaller or greater than original depth of wind, which was already inputted by user as an user input (D). If this "DV_1" is greater than "D", which means 22 m/s is too high than standard velocity; then program starts to compute a new depth verification value, designated as "DV_2" by reducing the velocity as assigned decrement. This velocity reduction process continues to find the exact ideal maximum velocity, which will derive the nearly same value of "DV_1" and user decided "D". The limit for this reduction was established as finding out the difference of both depth verification reaches to 0.1 x 10^{-6}; (Equation 5.2).

$$|D - DV_1| < 0.0000001$$

(Equation 5.2)

DV_1: 1^{st} Depth Verification

D: Depth of Wind

When macro finds the exact velocity value according to this verification interval, then it is commanded to fill all required (*already computed*) variables into the each specific block in Excel sheet. As a result of this macro process, each time, velocity and hoisting distance variables, belonging to each operating stage, are obtained according to used rope type.

As a second possibility, when macro finds out that "DV_1" is smaller than "D", which means maximum available velocity value (22 m/s) is not enough to catch cycle time, then, it is commanded to put a "*No Solution*" sign for relevant block in Excel sheet. This "*No Solution*" warning is also written if calculated velocity value is smaller than "1m/s". In this way, all of these variables, such as t_a, t_d, V_r, x_c, are calculated and written if they are available.

These series of calculations, verifications and evaluations are gone for each skip size in output sheet of each rope type. At the end of this running macro, all available variables are resulted and gathered in these output sheets. An example Excel result table has been drawn below (Table 5.4).

Table 5.4 Example Output Table as a Result of Velocity Macro

M	Xacc	Tacc	V1	Xcons	Tcons	V2	Xret	Tret	V3	Xcreep	Tcreep	V4	Xdec	Tdec	V5	Xtotal	Ttotal
2000	No Solution																
4000	No Solution								1.00								
6000	No Solution								1.00								
8000	95.49	13.82	13.82	806.52	58.36	13.82	94.99	12.82	1.00	3.00	3.00	1.00	0.00	8	0.00	1000.00	96.00
10000	52.01	10.20	10.20	893.47	87.60	10.20	51.51	9.20	1.00	3.00	3.00	1.00	0.00	10	0.00	1000.00	120.00
12000	33.53	8.19	8.19	930.44	113.62	8.19	33.03	7.19	1.00	3.00	3.00	1.00	0.00	12	0.00	1000.00	144.00
14000	23.62	6.87	6.87	950.26	138.25	6.87	23.12	5.87	1.00	3.00	3.00	1.00	0.00	14	0.00	1000.00	168.00
16000	17.61	5.94	5.94	962.27	162.13	5.94	17.11	4.94	1.00	3.00	3.00	1.00	0.00	16	0.00	1000.00	192.00
18000	13.67	5.23	5.23	970.16	185.54	5.23	13.17	4.23	1.00	3.00	3.00	1.00	0.00	18	0.00	1000.00	216.00
20000	10.93	4.68	4.68	975.64	208.65	4.68	10.43	3.68	1.00	3.00	3.00	1.00	0.00	20	0.00	1000.00	240.00
22000	8.95	4.23	4.23	979.60	231.54	4.23	8.45	3.23	1.00	3.00	3.00	1.00	0.00	22	0.00	1000.00	264.00
24000	7.47	3.86	3.86	982.57	254.27	3.86	6.97	2.86	1.00	3.00	3.00	1.00	0.00	24	0.00	1000.00	288.00
26000	6.33	3.56	3.56	984.85	276.89	3.56	5.83	2.56	1.00	3.00	3.00	1.00	0.00	26	0.00	1000.00	312.00
28000	5.43	3.30	3.30	986.64	299.41	3.30	4.93	2.30	1.00	3.00	3.00	1.00	0.00	28	0.00	1000.00	336.00
30000	4.71	3.07	3.07	988.08	321.86	3.07	4.21	2.07	1.00	3.00	3.00	1.00	0.00	30	0.00	1000.00	360.00
32000	4.13	2.87	2.87	989.24	344.25	2.87	3.63	1.87	1.00	3.00	3.00	1.00	0.00	32	0.00	1000.00	384.00
34000	3.65	2.70	2.70	990.20	366.60	2.70	3.15	1.70	1.00	3.00	3.00	1.00	0.00	34	0.00	1000.00	408.00
36000	3.25	2.55	2.55	991.01	388.90	2.55	2.75	1.55	1.00	3.00	3.00	1.00	0.00	36	0.00	1000.00	432.00
38000	2.91	2.41	2.41	991.68	411.18	2.41	2.41	1.41	1.00	3.00	3.00	1.00	0.00	38	0.00	1000.00	456.00
40000	2.62	2.29	2.29	992.26	433.42	2.29	2.12	1.29	1.00	3.00	3.00	1.00	0.00	40	0.00	1000.00	480.00
42000	2.37	2.18	2.18	992.75	455.64	2.18	1.87	1.18	1.00	3.00	3.00	1.00	0.00	42	0.00	1000.00	504.00
44000	2.16	2.08	2.08	993.18	477.84	2.08	1.66	1.08	1.00	3.00	3.00	1.00	0.00	44	0.00	1000.00	528.00
46000	1.97	1.99	1.99	993.55	500.03	1.99	1.47	0.99	1.00	3.00	3.00	1.00	0.00	46	0.00	1000.00	552.00
48000	1.81	1.90	1.90	993.88	522.19	1.90	1.31	0.90	1.00	3.00	3.00	1.00	0.00	48	0.00	1000.00	576.00
50000	1.67	1.83	1.83	994.16	544.35	1.83	1.17	0.83	1.00	3.00	3.00	1.00	0.00	50	0.00	1000.00	600.00
52000	1.54	1.76	1.76	994.42	566.49	1.76	1.04	0.76	1.00	3.00	3.00	1.00	0.00	52	0.00	1000.00	624.00
54000	1.43	1.69	1.69	994.64	588.62	1.69	0.93	0.69	1.00	3.00	3.00	1.00	0.00	54	0.00	1000.00	648.00
56000	1.33	1.63	1.63	994.85	610.74	1.63	0.83	0.63	1.00	3.00	3.00	1.00	0.00	56	0.00	1000.00	672.00
58000	1.24	1.57	1.57	995.03	632.86	1.57	0.74	0.57	1.00	3.00	3.00	1.00	0.00	58	0.00	1000.00	696.00
60000	1.15	1.52	1.52	995.19	654.96	1.52	0.65	0.52	1.00	3.00	3.00	1.00	0.00	60	0.00	1000.00	720.00

5.3 Diameter and MoI of Headsheave Macro Process

The macro list of moment of inertia evaluation for headsheaves is written on the basis of above explained mathematical approach. As it is said that each available rope type has specific rope sheet and this MoI macro process is repeated for each rope type since the evaluated available rope diameters are changed according to each rope type.

The commanded macro selects the exact value of headsheave diameter, which has been already calculated by Excel written formula. Then, macro opens the exact moment of inertia table of headsheave (Appendix B).

Macro tries to find out the headsheave diameter interval to start to compute moment of inertia of headsheave by interpolation process. Finally, this macro fills these evaluated moment of inertia values in the allocated blocks in the exact rope sheet.

This process is repeated for each specific rope type and their calculated rope and headsheave diameters. At the end, all available rope diameters in each rope sheet is included these evaluated moment of inertia values under the block of "*MoI-HS*" in output rope sheets as moment of inertias of headsheave. To illustrate the name abbreviation for these output rope sheets, they are designated as like "*O TAS_6x23*".

5.4 Slippage Acceleration Macro Process

The limits of acceleration are important in determining of the slippage acceleration. The approach to find out the slippage acceleration is available after the studies on mathematical approach. Each of these expressions are fitted into the excel sheet. As a reminder, each rope type has output sheet, therefore, each of previous explained variables and expressions has been written in specific columns, separately.

The moment of inertia and diameter of headsheave(s) are already set into these sheets, as it was explained in previous section. Next, the variables (i.e. PL, C, F) are separated and written in these sheets. Table 5.5 shows the slippage acceleration variables calculated.

Table 5.5 Table Format of Slippage Acceleration Variables (Sample)

L1	R1	T1	L2	R2	C	T2	PL	F	MoI-HS	HS Dia	H	L3	R3	G	e^μθ

"L_1" is the rope length suspended at the ascending side and determined from the winding depth, rope length between the headsheave (or friction wheel) and the top position of the skip, and the rope length below the lowest skip position (Figure 5.1).

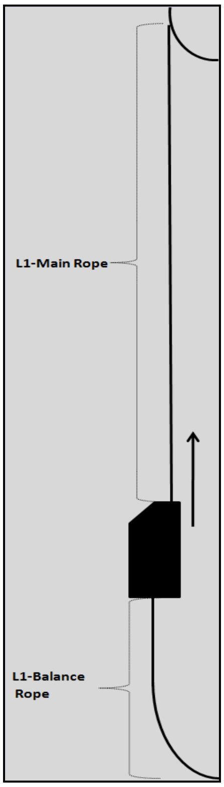


Figure 5.1 Rope Length on Ascending Side

Table 5.6 shows the determination of total rope length suspended at either side.

Table 5.6 Total Rope Length in Decision of Designed Software

Rope Distances	**Length (in meter)**
Depth of Wind (Shaft Depth)	User Input
Distance from Top Position of the Skip to Rotating Wheel (*Headsheave / Friction Wheel*)	**15** (*Default Input*)
Length of Rope Loop below the Lowest Position of Skip (*Balance Rope Portion on Ascending Side*)	**10** (*Default Input*)

The central positions of headsheaves are nearly at the same level in ground mounted system applications, operationally. The level of these positions might be changed, however; it is accepted that the level of the centers of headsheaves are the same in this study. So that L1 = L2 (Figure 5.2)

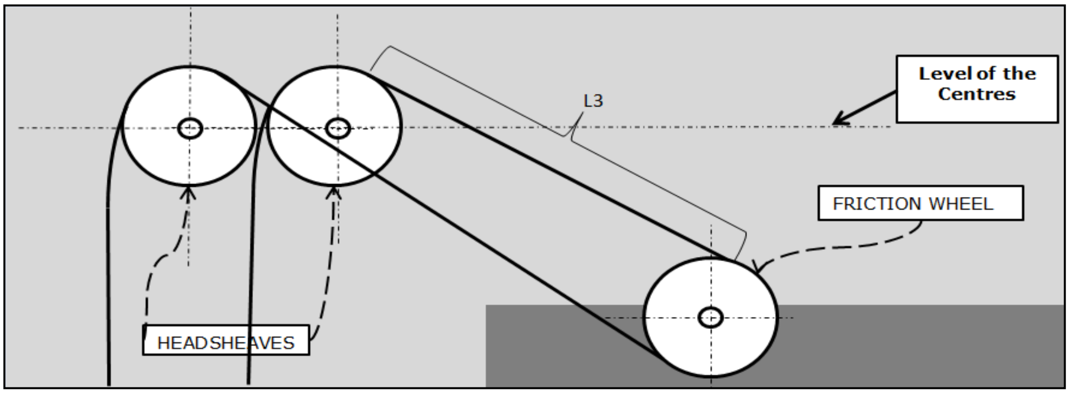


Figure 5.2 Level of the headsheaves and the distance for L3 (Assumed at this study)

In addition to these rope lengths, there is an extra rope in ground mounted systems in addition to tower mounted systems. This extra rope length is symbolized as "*L3*".

This rope is located between the friction wheel to the top of the headsheave (Figure 5.2). This rope length is determined by the program user and typed in user inputs. However; there is a limitation in terms of this distance. The distance between the friction wheel and headsheaves can be neither greater than 50 m nor closer than 30 m, proven conventionally, (Equation 5.3). Hence, the designed program has been put this limitation in this situation.

$$\mathbf{30m \leq L3 \leq 50m}$$

(Equation 5.3)

R1 is the total mass of rope on ascending side.

$$\mathbf{R1 = L1 \times Cm \times d2}$$
$$\mathbf{R2 = L2 \times Cm \times d2}$$
$$\mathbf{R3 = L3 \times Cm \times d2 \text{ (only for ground mounted)}}$$

(Equation 5.4)

The unit mass value of the rope is varying according the rope type and rope diameter. Thus, R1, R2 and also R3 values are calculated for each rope type and rope diameter. The outputs of these calculations are written in the blocks in each rope sheet in their specific places. Empty mass of conveyance "C" is assumed to be equal to the payload "PL"

The values of T1 and T2 are calculated according to written expressions since the all required values are gathered and already calculated like R1 or C values in each output sheet of rope types. Moreover, the results of T1 and T2 would be in kilograms but it should be required to convert these values in Newton. Hence, these values are multiplied by 9.81 m/s^2 and by 10^{-3} to conclude its in kN (kilo Newton).

$$\mathbf{T1 \times 9.81 = [N]}$$

(Equation 5.5)

Guide friction "F" is calculated in the Excel sheets using the following relation (Equation 5.6) (Walker, 1988).

$$\mathbf{F = PL \times 0.18/2 \times 9.81 = [N]}$$

(Equation 5.6)

The values of moment of inertias and diameter of headsheaves have been calculated already by the MoI - Headsheave macro. So, these values have been already written in their belonging blocks nearby of these values.

The values of “H” and “G” are also evaluated by Excel. The equations are established as resulted in these values.

When the macro results the MoI values of headsheaves and the diameter of them, then the written Excel equations calculate the “H” value according to the following formula.

$$\mathbf{H = \mathit{MoI} / \mathit{r}^2}$$

(Equation 5.7)

On the same basis, the written Excel expression calculates the corresponding “G” value as;

$$\mathbf{G = H + R3}$$

(Equation 5.8)

The above explained variables are required to calculate slippage acceleration. At the end of these series of calculations, each rope type sheet, established in Excel tables, contains prior maximum permissible acceleration values.

Since the all required variables would be already calculated (*including $a_{slippage}$ (a^*)*), the written macro is able to compare default initial upper acceleration ($a_{default}$ = 1m/s^2) limit with calculated one ($a_{slippage} = a^*$) to set the actual system acceleration limit (upper acceleration limit for design). There might be three possible conditions in a result of comparison of these two acceleration values after computing. These possible conditions and required actions are tabulated in Table 5.7.

Table 5.7 Possible Conditions for Acceleration Calculations

Acceleration Varieties		
Default Maximum Acceleration (1 m/s^2)		**Calculated Acceleration [m/s^2]**
$a = a_{default} = 1m/s^2$		$a^* = a_{slippage} = a_{calculated}$
THREE POSSIBLE CONDITIONS		
No	**Possible Condition**	**Required Action**

I	$1 < a^*$	*No need for action; process continues as "a=1"*
II	$0.4[m/s2] \leq a^* < 1$	*Running Slippage Acceleration Macro*
III	$a^* < 0.4$ [m/s2]	*No possibility for winding operation*

If the calculated slippage acceleration "a^*" is greater than default acceleration limit "1 m/s^2", then designed model accepts the default acceleration value as an upper limit (1 m/s^2).

If the calculated slippage acceleration is between the default limits "$0.4 \leq a^* < 1$", then written slippage macro runs and identifies the slippage acceleration as upper limit for the designed system. This means that system acceleration must be smaller than slippage acceleration (a^*).

If the calculated slippage acceleration "a^*" is smaller than default lower acceleration limit "0.4 m/s^2", then model stops to decide the acceleration limits since there is no possibility to design any hoisting system has got lower maximum permissible acceleration value than lowest value of acceleration limit (0.4 m/s^2).

5.5 Diameter and MoI of Guide Pulley Macro Process

Guide pulley diameter is related to the friction wheel diameter which is related to the rope diameter as discussed before. The MoI-GP macro has been commanded to open the specific rope sheet, to select the blocks, where guide pulley diameter data are placed, and to get these values. Then, the macro is issued to command to open the "MoI GP" sheet, which is included required moment of inertia and guide pulley diameter data. Macro is started to check whether the calculated guide pulley diameter value is fallen into between the intervals of two diameter values.

When the macro finds the appropriate two values of guide pulley diameter, it calculates the MoI by interpolation.

As a result of these commands, macro can obtain the required moment of inertia values for aimed guide pulley diameters for the selected rope type and evaluated friction wheel. Meanwhile, this macro is commanded to write these evaluated moment of inertia values in the allocated blocks against in each proper guide pulley diameter in each output sheet.

All available rope diameters in each rope sheet is included these evaluated moment of inertia values under the block of "*MoI of Guide Pulley*" in output rope sheets as moment of inertias of guide pulley.

5.6 Diameter and MoI of Friction Wheel Macro Process

The moment of inertia process of friction wheel (MoI-FW) macro is worked on the basis of series of interpolations. The logic and theory of these calculations have been explained in mathematical approach section (Chapter 4.6).

There are constructed specific output sheets for all available rope type. Each output sheet contains already calculated diameter of friction wheel. As it is stated above, friction wheel diameter is calculated according to rope diameter. The applied rope diameter has direct effect on statement of diameter of friction winder. Thus, it can be accepted that each Excel rope sheet includes the calculated rope diameter, so, the diameter of friction wheel diameter, too. This calculation can be possible by written a formula in rope sheets as illustrated in above section, which is changed according to rope type.

Since the diameter of friction winder has been already calculated, the available moment of inertia value for each available friction wheel diameter is achieved by series of interpolation calculations similar to MoI-HS and MoI-GP.

Macro starts its operation by selection of first rope sheet. This output sheet contains required friction wheel diameter, already calculated by constituted Excel formula. So, the proper friction winder diameter is located in this sheet.

The MoI-FW macro has been commanded to open the rope sheet, to select the blocks, where friction winder diameter data are placed, and to get these values. Then, the macro is issued to command to open the "MoI FW" sheet, which is included required moment of inertia and friction winder diameter data. Macro is started to check whether the actual friction winder diameter value is placed in between which two written diameter values.

When the macro finds the appropriate intervals, between two values of friction wheel diameter, it starts interpolation processes.

As a result of these calculations, macro can obtain the required moment of inertia values for aimed friction winder diameters for the selected rope type and evaluated friction wheel. Meanwhile, this macro is commanded to write these evaluated moment of inertia values in the allocated blocks against in each proper friction winder diameter in each output sheet.

All available rope diameters in each rope sheet is included these evaluated moment of inertia values under the block of “*MoI of Friction Winder*” in output rope sheets as moment of inertias of friction winder.

5.7 Motor Selection and Minimum Required Power Macro Process

The designed software has been prepared to designate design parameters of friction winding system. The last and the most important parameter is to determine required power. Determining of required power of any winding system is crucial.

The calculations and approaches for motor power selection are described in detail in previous sections (Chapter 4.7).

The designed “Motor Macro” requires some user inputs before starting to run series of calculations. These inputs are varied by the selected motor type. For instance, user has to provide the kV and number of motor inputs if a.c. type motor is selected. The user inputs are shown in Table 5.8 according to selected motor type.

Table 5.8 User Inputs for Required Power Design Macro Process

<table>
<tr><th rowspan="2">MOTOR TYPE</th><th>REQUIRED USER INPUTS</th><th>OUTPUT OF MOTOR MACRO</th></tr>
<tr><td><u>Common User Inputs for all Motor Types</u>:
• Mounting Type (ground/tower)
• Motor Type (AC/DCGD/DCDD)</td><td rowspan="4">REQUIRED MOTOR POWER [kW]</td></tr>
<tr><td>A.C. Type Drive [AC]</td><td>• kV (3.3/6.6/11.0)
• Number of Motors (single/double)</td></tr>
<tr><td>D.C. with Geared Drive [DCGD]</td><td>• Number of Motors (single/double)</td></tr>
<tr><td>D.C. with Direct Drive [DCDD]</td><td>• N/a (except for common user inputs)</td></tr>
</table>

These illustrated user inputs specify the calculated parameters in inertia, torque and power calculations. Hence, user must decide the motor type and then the other belonging inputs.

The “Motor Macro” makes it possible to select the motor type; AC (*a.c. type motor*), DCGD (*d.c. with geared drive type motor*), DCDD (*d.c. with direct drive type motor*) by the program user.

When the selected motor type is identified, macro opens the necessary motor macro. The motor macros (*excluding DCDD*) look first of all for the entered number of motors and kV values (*only for AC*). Motor macro open the first output sheet and starts to identify both already calculated preliminary motor power “*PMP*” and motor rpm “*MRPM*”. These two values would be calculated in each output sheet for all possible skip masses and other parameters. Table 5.9 shows the limitations on PMP and MRPM depending on the type of motor drive (Walker, 1988).

Table 5.9 Limitations in Motor Inertia Tables for PMP & MRPM

MOTOR TYPE		**LIMITS FOR PMP [kW]**	**LIMITS FOR MRPM [rpm]**
AC	**3.3kV**	$200 \leq PMP \leq 2000$	$300 \leq MRPM \leq 500$
	6.6kV	$400 \leq PMP \leq 2000$	$375 \leq MRPM \leq 500$
	11.0kV	$600 \leq PMP \leq 2000$	$375 \leq MRPM \leq 500$
DCGD		$300 \leq PMP \leq 4000$	$300 \leq MRPM \leq 500^{*}$
DCDD		$750 \leq PMP \leq 5000$	$45 \leq MRPM \leq 60$
**Due to the limits of Gear Unit Inertia table*			

The valid PMP values are changed when user selected the number of motors double. This relation can be seen below description to understand macro process in consideration of motor number. Preliminary motor power “PMP” requirement is determined by equation (Equation 5.9) can be halved by selecting double motor instead of one.

SINGLE MOTOR **PMP / Motor = PMP**
DOUBLE MOTORS **PMP / Motor= PMP / 2**
(Equation 5.9)

After designation of PMP and MRPM values, macro check these values whether they are within the limits or not. If these values are out of the limits, then macro cannot continue its run and identify this condition as “Out of Range”. On the other hand, if these values procure these limits, then macro continues its run to find out required motor power.

Afterwards, macro select the inertia values appropriate to selected motor type, and calculated PMP and MRPM values. The motor inertias are used in torque calculations thus they are important to be identified, exactly. Inertia tables have been given in the Appendices.

The determined values are integrated into the output sheets. If the motor number is selected as double, then these inertia values are multiplied by two.

Number of Motors: DOUBLE

MOTOR INERTIA = MOTOR INERTIA x 2

(Equation 5.10)

When these motor inertia values are calculated, output sheets start to compute total inertias, both dynamic and static torques, required power in all time stage while winding, RMS power and finally total required power to operate designed friction winding system (Chapter 4.7).

These all macro processes end up with selecting and writing all commanded required output parameters and values in a general output sheet. This sheet can be printed out by user. To conclude this motor macro process, motor macro scanned the all calculated required motor power values and identifies the minimum required motor power. This means that the software model gives the minimum motor power according to stated user inputs at the end of this macro process.

5.8 Designed Model Run

The model is based on Microsoft Excel and Visual Basic as explained in previous sections. Thus, the model is started by opening the Excel file. When opening this file, there is a front page at which there are some list and combo boxes for input parameters and selections as it is seen in Figure 5.3.

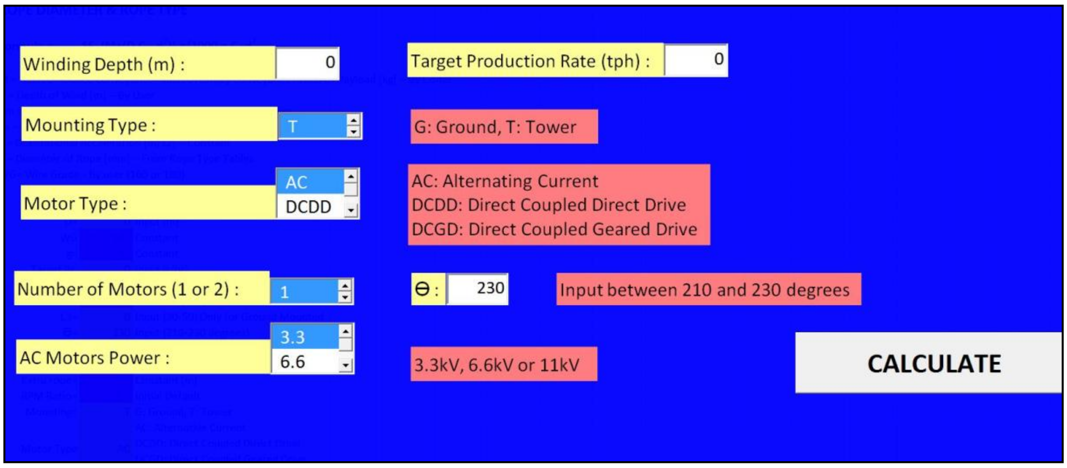


Figure 5.3 Front Page of Designed Model

There is required that user keys depth of wind [meter], hourly hoisting capacity [tons] and angle of contact value [degrees]. It is also necessary to select mounting type [ground/tower] motor type [DCDD/DCGD/AC], number of motors [single/double] and if AC type motor is selected, the kV power supply [3.3/6.6/11.0] (Figure 5.3).

When these selections are done and the required values are typed then the calculate button will be clicked. Then the all formed macros, formulas and calculations are completed by designed model. This process takes around 5 seconds. Finally, the model results a whole output screen as shown in Figure 5.4. The all available minimum required power and accordingly all other design parameters are filled for each rope type. Corresponding to all applicable hoisting rope types, there are 10 different types, depending to calculated minimum power depending to user inputs each design parameters are calculated and then resulted in this screen. To see these design parameters, Figure 5.5 is illustrated. As it can be seen there are all user inputs and resulted output parameters are written in this output sheet. All of these output sheets are comprised in an output screen (Figure 5.4).

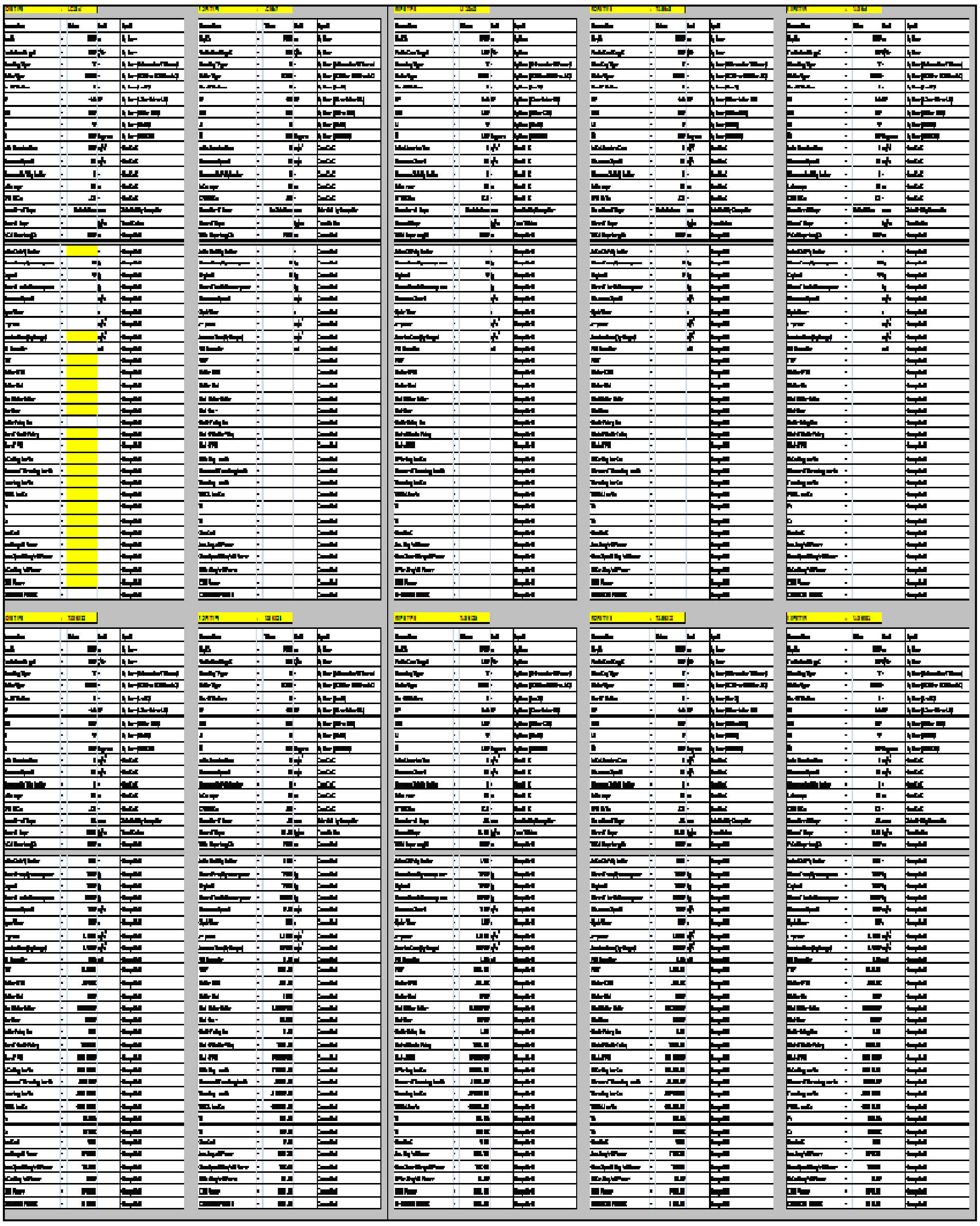


Figure 5.4 Output Screen of Results

ROPE TYPE	:	TAS 6X22		
Parameters		**Value**	**Unit**	**Input**
Depth	=	750	m	by User
Production Target	=	200	t/hr	by User
Mounting Type	=	G	-	by User (G:Ground or T:Tower)
Motor Type	=	DCDD	-	by User (DCDD or DCGD or AC)
No. Of Motors	=	1	-	by User (1 or 2)
kV	=	3.3	kV	by User (3.3 or 6.6 or 11)
WG	=	180		by User (160 or 180)
L3	=	40		by User (30-50)
θ	=	230	Degrees	by User (210-230)
Initial Acceleration	=	1	m/s²	Constant
Maximum Speed	=	22	m/s	Constant
Minimum Safety Factor	=	7	-	Constant
Extra rope	=	25	m	Constant
RPM Ratio	=	8.5	-	Constant
Diameter of Rope	=	60	mm	Selected by Computer
Mass of Rope	=	15.34	kg/m	from Tables
Total Rope Length	=	1590	m	Computed
Actual Safety Factor	=	7.44	-	Computed
Mass of empty conveyance	=	13000	kg	Computed
Payload	=	13000	kg	Computed
Mass of loaded conveyance	=	26000	kg	Computed
Maximum Speed	=	3.54	m/s	Computed
Cycle Time	=	234	s	Computed
a - prime	=	0.4484	m/s²	Computed
Acceleration (by design)	=	0.4483	m/s²	Computed
FW Diameter/HS Diameter	=	5.40	m	Computed
PMP	=	752.42		Computed
Motor RPM	=	106.42		Computed
Motor MoI	=	13300		Computed
MoI Motor Rotor	=	960925.00		Computed
MoI Gear	=			Computed
Guide Pulley Dia	=	4.86		Computed
MoI of Guide Pulley	=	15209.87		Computed
MoI of FW	=	134200.00		Computed
Retarding Inertia	=	183700.00		Computed
Masses of Travelling Loads	=	63390.60		Computed
Travelling Inertia	=	462117.47		Computed
TOTAL Inertia	=	645817.47		Computed
Ts	=	406.31		Computed
Ta	=	107.24		Computed
Constant	=	0.10		Computed
Acc.Req.'ed Power	=	673.18		Computed
Cons.Speed Req.'ed Power	=	532.61		Computed
Retar.Req.'ed Power	=	392.04		Computed
RMS Power	=	533.80		Computed
REQUIRED POWER	=	560.49	kW	Computed

Figure 5.5 Example Design Outputs for TAS_6x22 Rope Type

5.9 Discussions of Program Test Runs and Results

The all applied methodology and model features in this software model have been explained in above sections.

The main aspect of this model is to shorten calculation time and to give the minimum required power and related components design parameters with the available user inputs. There were applied some test runs according to explained frame at this model. The results of these runs are also discussed to clarify and verify the model.

First of all, there were run several tests on variation of depth of wind and hourly hoisting capacity values for only one rope type. Triangular strand 6x22 rope type was chosen for these tests. The motor was selected as d.c. direct drive motor with ground mounting type. Under these inputs, the model was run for the depths starting with 750m to 1500m. For each depth of wind value, the hourly hoisting capacity was keyed as 200tons and 350tons. The all results of these runs are tabulated in Table 5.10.

Table 5.10 Results of Test Runs for TAS_6x22

USER INPUTS									
ROPE TYPE:	TAS_6X22								
MOTOR TYPE:	DCDD								
Mounting Type:	Ground								

PROGRAM OUTPUTS: DEPTH [m]	TpH [ton]	Skip Capacity [kg]	Cycle Time [s]	Rope Diameter [mm]	a* [m/s²]	FWD [m]	Vmax [m/s]	MINIMUM REQUIRED POWER [kW]
750	200	13000	234	60	0.4484	5.4	3.54	560.49
	350	14000	144	64	0.4539	5.76	6.58	1147.24
1000	200	12000	216	64	0.6015	5.76	5.17	761.72
	350	12000	123	64	0.6015	5.76	10.99	1714.32
1200	200	11000	198	64	0.7101	5.76	6.85	938.04
	350	11000	113	64	0.7101	5.76	15.38	2330.1
1500	200	8000	144	64	0.8829	5.76	12.6	1387.84
	350	No Solution	No Solution	No Solution	No Solution	No Solution	No Solution	No Solution

There are only some significant design parameters are written in Table 5.10 such as skip capacity, cycle time and friction wheel diameter according to required minimum power corresponding to user inputs.

When the depth of wind is determined as 750 m and hourly hoisting capacity as 200 tons; minimum required power resulted as 560.49 kW with 13 tons skip. On the other hand, when the hourly hoisting capacity was increased to 350 tons then the required minimum power increased 1147.24 kW with 14tons skip.

Besides, when the depth of winding was increased double for the 200 tons hourly production; the required minimum power resulted as 1387.84 kW with an 8 tons skip capacity.

When the results are determined, it can be recognized that the skip capacity might be varied independently than depth of wind and hourly hoisting capacity to get minimum required power.

It is also remarkable that the cycle time is independent on depth of wind but it is directly dependent on maximum velocity value.

On the second series of test run, the depth of wind and hourly production values remained as constant (1000 m and 250 tons), also mounting type was selected as tower mounted but the motor type selected for alternative current drive (AC), direct current direct drive (DCDD) and direct current with geared drive (DCGD). The results and parameters are gathered in a table to see all outputs (Table 5.11)

Table 5.11 Results of Test Runs for all Motor Types

USER INPUTS	DEPTH [m]:	1000	DEPTH [m]:	1000	DEPTH [m]:	1000
	TpH [ton]:	250	TpH [ton]:	250	TpH [ton]:	250
	Mounting Type:	Tower	Mounting Type:	Tower	Mounting Type:	Tower
	MOTOR TYPE:	DCDD	MOTOR TYPE:	AC	MOTOR TYPE:	DCGD
			kV:	6.6		
PROGRAM OUTPUTS	MINIMUM REQUIRED POWER [kW]	980.52	MINIMUM REQUIRED POWER [kW]	1263.67	MINIMUM REQUIRED POWER [kW]	1114.51
	Rope Type	TAS_6x22	Rope Type	TAS_6x22	Rope Type	TAS_6x22
	Rope Diameter [mm]	64	Rope Diameter [mm]	51	Rope Diameter [mm]	54
	Cycle Time [s]	173	Cycle Time [s]	115	Cycle Time [s]	130
	a* [m/s2]	1.8189	a* [m/s2]	1.7778	a* [m/s2]	1.7742
	Vmax [m/s]	6.6	Vmax [m/s]	10.66	Vmax [m/s]	9.2
	FWD [m]	5.76	FWD [m]	4.59	FWD [m]	4.86
	Skip Capacity [kg]	12000	Skip Capacity [kg]	8000	Skip Capacity [kg]	9000
	Actual Safety Factor	7.26	Actual Safety Factor	7.04	Actual Safety Factor	7.03

The designed model recommends using triangular strand 6x22 rope type in case of each different motor type selection. However, minimum motor power requirement will be resulted for d.c. direct drive motor type selection among three motor alternatives. As it can be seen in Table 5.11, the required power will be 980.52 kW in d.c. direct drive motor application under these conditions. The required rope diameter should be 64 mm with 12 tons skip. On the other hand, the greatest required power should be supplied in comparison of all available motor alternatives in case of a.c. motor type application under given inputs.

It should be reminded that these outputs are gathered and presented in output screen of the model and tabulated for comparison in these cases in this section. Additionally, model cannot present any design outputs if any parameter is below or above identified limits. Hence, the all presented results can be identified that applicable in terms of practice applications. To illustrate, the calculated safety factor values are all higher than 7.00.

CHAPTER 6

CONCLUSIONS & RECOMMENDATIONS

The nature of friction winding design process is very complex and takes considerable time by manual calculation since various factors affect the design process.

The friction winding design software program developed gives the solutions for the design parameters within about five seconds saving significant time. Moreover, since it is based on the computer skills, the human error during manual calculation is eliminated.

The program requires some user inputs (winding depth, hourly hoisting capacity) and selections (type and numbers of motors, type of friction wheel mounting) which can be changed by user. Therefore, the program makes it possible to change and compare different conditions in designing the friction winding systems.

The program considers the minimum power requirement as the main criteria and determines the corresponding design parameters such as skip capacity, hoisting speed, acceleration, friction wheel, headsheave, guide pulley diameters, cycle period and motor size for each available rope type. If the system cannot be operated by the given input, model presents this result as “no solution”.

The fundamentals of this model are constructed to make future changes and modifications at certain inputs. The inputs in the program related to rope safety factor, available rope types, rotating elements, motor types, velocity and acceleration limits, skip capacity can be changed if required.

This model might be developed for designing drum winding systems with skips and cages. There might be complete design software program for hoisting systems.

The economic and financial aspects of friction winding design can be developed in the future studies. Such a financial model can be adapted this design model and give a chance to study the design process both in engineering and finance phases.

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APPENDIX A

STEEL WIRE ROPE TABLES

A.I. LOCKED COIL 18X7

Table A.1. Rope Table (Locked Coil 18 x 7)

18 x 7 Locked Coil			
Rope Diameter	**Cm**	**Cs**	**Unit Mass**
(mm)		**Wire Grade 180**	**[kg/m]**
4	0.0040	0.77	0.064
5	0.0040	0.77	0.100
6	0.0040	0.77	0.145
7	0.0040	0.77	0.197
8	0.0040	0.77	0.257
9	0.0040	0.77	0.326
10	0.0040	0.77	0.402
11	0.0040	0.77	0.486
12	0.0040	0.77	0.579
13	0.0040	0.77	0.679
14	0.0040	0.77	0.788
16	0.0040	0.77	1.030
18	0.0040	0.77	1.300
20	0.0040	0.77	1.610
22	0.0040	0.77	1.950
24	0.0040	0.77	2.310
26	0.0040	0.77	2.720
28	0.0040	0.77	3.150

A.II. LOCKED COIL 36X7

Table A.2. Rope Table (Locked Coil 36 x 7)

36 x 7 Locked Coil			
Rope Diameter (mm)	Cm	Cs Wire Grade 180	Unit Mass [kg/m]
12	0.0039	0.74	0.562
13	0.0039	0.74	0.659
14	0.0039	0.74	0.765
16	0.0039	0.74	0.999
18	0.0039	0.74	1.260
20	0.0039	0.74	1.560
22	0.0039	0.74	1.890
24	0.0039	0.74	2.250
26	0.0039	0.74	2.640
28	0.0039	0.74	3.060
32	0.0039	0.74	4.000
36	0.0039	0.74	5.060
40	0.0039	0.74	6.240

A.III. LOCKED COIL 10X10

Table A.3. Rope Table (Locked Coil 10 x 10)

10 x 10 Locked Coil			
Rope Diameter (mm)	Cm	Cs Wire Grade 180	Unit Mass [kg/m]
12	0.0043	0.78	0.625
13	0.0043	0.78	0.734
14	0.0043	0.78	0.851
16	0.0043	0.78	1.110
18	0.0044	0.78	1.410
20	0.0044	0.78	1.740
22	0.0043	0.78	2.100
24	0.0043	0.78	2.500
26	0.0043	0.78	2.940
28	0.0043	0.78	3.400
32	0.0043	0.78	4.450

A.IV. TRIANGULAR STRAND 6X8

Table A.4. Rope Table (Triangular Strand 6 x 8)

6 x 8 Triangular Strand			
Rope Diameter (mm)	Cm	Cs Wire Grade 180	Unit Mass [kg/m]
13	0.0042	0.71	0.709
14	0.0042	0.72	0.822
16	0.0042	0.71	1.070
18	0.0042	0.71	1.360
19	0.0042	0.71	1.510
20	0.0042	0.71	1.680
21	0.0042	0.71	1.850
22	0.0042	0.71	2.030
24	0.0042	0.71	2.420
26	0.0042	0.71	2.830
28	0.0042	0.71	3.280
29	0.0042	0.71	3.530
32	0.0042	0.71	4.300
35	0.0042	0.71	5.140

A.V. TRIANGULAR STRAND 6X9

Table A.5. Rope Table (Triangular Strand 6 x 9)

6 x 9 Triangular Strand			
Rope Diameter (mm)	Cm	Cs Wire Grade 180	Unit Mass [kg/m]
19	0.0042	0.72	1.510
20	0.0042	0.72	1.680
21	0.0042	0.72	1.850
22	0.0042	0.72	2.030
24	0.0042	0.72	2.420
26	0.0042	0.72	2.830
28	0.0042	0.72	3.260
29	0.0042	0.72	3.530
32	0.0042	0.72	4.300

A.VI. TRIANGULAR STRAND 6X22

Table A.6. Rope Table (Triangular Strand 6 x 22)

6 x 22 Triangular Strand			
Rope Diameter	**Cm**	**Cs**	**Unit Mass**
(mm)		**Wire Grade 180**	**[kg/m]**
13	0.0042	0.72	0.709
14	0.0043	0.72	0.834
16	0.0043	0.72	1.090
18	0.0043	0.72	1.380
19	0.0042	0.72	1.530
20	0.0043	0.72	1.700
21	0.0042	0.72	1.870
22	0.0043	0.72	2.060
24	0.0043	0.72	2.450
26	0.0043	0.72	2.880
28	0.0043	0.72	3.340
29	0.0043	0.72	3.580
32	0.0042	0.72	4.350
35	0.0043	0.72	5.210
38	0.0043	0.72	6.150
41	0.0043	0.72	7.150
44	0.0043	0.72	8.240
48	0.0043	0.72	9.800
51	0.0043	0.72	11.110
54	0.0043	0.72	12.450
57	0.0042	0.72	13.800
60	0.0043	0.72	15.340
64	0.0042	0.72	17.400

A.VII. TRIANGULAR STRAND 6X23

Table A.7. Rope Table (Triangular Strand 6 x 23)

6 x 23 Triangular Strand			
Rope Diameter	**Cm**	**Cs**	**Unit Mass**
(mm)		**Wire Grade 180**	**[kg/m]**
13	0.0042	0.72	0.709
14	0.0043	0.72	0.834
16	0.0043	0.72	1.090
18	0.0043	0.72	1.380
19	0.0042	0.72	1.530
20	0.0043	0.72	1.700
21	0.0042	0.72	1.870
22	0.0043	0.72	2.060
24	0.0043	0.72	2.450
26	0.0043	0.72	2.880
28	0.0043	0.72	3.340
29	0.0043	0.72	3.580
32	0.0042	0.72	4.350
35	0.0043	0.72	5.210
38	0.0043	0.72	6.150
41	0.0043	0.72	7.150
44	0.0043	0.72	8.240
48	0.0043	0.72	9.800
51	0.0043	0.72	11.110
54	0.0043	0.72	12.450
57	0.0042	0.72	13.800
60	0.0043	0.72	15.340
64	0.0042	0.72	17.400

A.VIII TRIANGULAR STRAND 6X25

Table A.8. Rope Table (Triangular Strand 6 x 25)

6 x 25 Triangular Strand			
Rope Diameter	Cm	Cs	Unit Mass
(mm)		Wire Grade 180	[kg/m]
13	0.0042	0.72	0.709
14	0.0043	0.72	0.834
16	0.0043	0.72	1.090
18	0.0043	0.72	1.380
19	0.0042	0.72	1.530
20	0.0043	0.72	1.700
21	0.0042	0.72	1.870
22	0.0043	0.72	2.060
24	0.0043	0.72	2.450
26	0.0043	0.72	2.880
28	0.0043	0.72	3.340
29	0.0043	0.72	3.580
32	0.0042	0.72	4.350
35	0.0043	0.72	5.210
38	0.0043	0.72	6.150
41	0.0043	0.72	7.150
44	0.0043	0.72	8.240
48	0.0043	0.72	9.800
51	0.0043	0.72	11.110
54	0.0043	0.72	12.450
57	0.0042	0.72	13.800
60	0.0043	0.72	15.340
64	0.0042	0.72	17.400

A.IX TRIANGULAR STRAND 6X28

Table A.9. Rope Table (Triangular Strand 6 x 28)

6 x 28 Triangular Strand			
Rope Diameter	**Cm**	**Cs**	**Unit Mass**
(mm)		**Wire Grade 180**	**[kg/m]**
13	0.0042	0.72	0.709
14	0.0043	0.72	0.834
16	0.0043	0.72	1.090
18	0.0043	0.72	1.380
19	0.0042	0.72	1.530
20	0.0043	0.72	1.700
21	0.0042	0.72	1.870
22	0.0043	0.72	2.060
24	0.0043	0.72	2.450
26	0.0043	0.72	2.880
28	0.0043	0.72	3.340
29	0.0043	0.72	3.580
32	0.0042	0.72	4.350
35	0.0043	0.72	5.210
38	0.0043	0.72	6.150
41	0.0043	0.72	7.150
44	0.0043	0.72	8.240
48	0.0043	0.72	9.800
51	0.0043	0.72	11.110
54	0.0043	0.72	12.450
57	0.0042	0.72	13.800
60	0.0043	0.72	15.340
64	0.0042	0.72	17.400

A.X. TRIANGULAR STRAND 6X31

Table A.10. Rope Table (Triangular Strand 6 x 31)

6 x 31 Triangular Strand			
Rope Diameter	Cm	Cs	Unit Mass
(mm)		Wire Grade 180	[kg/m]
13	0.0042	0.72	0.709
14	0.0043	0.72	0.834
16	0.0043	0.72	1.090
18	0.0043	0.72	1.380
19	0.0042	0.72	1.530
20	0.0043	0.72	1.700
21	0.0042	0.72	1.870
22	0.0043	0.72	2.060
24	0.0043	0.72	2.450
26	0.0043	0.72	2.880
28	0.0043	0.72	3.340
29	0.0043	0.72	3.580
32	0.0042	0.72	4.350
35	0.0043	0.72	5.210
38	0.0043	0.72	6.150
41	0.0043	0.72	7.150
44	0.0043	0.72	8.240
48	0.0043	0.72	9.800
51	0.0043	0.72	11.110
54	0.0043	0.72	12.450
57	0.0042	0.72	13.800
60	0.0043	0.72	15.340
64	0.0042	0.72	17.400

APPENDIX B

MOMENT OF INERTIA TABLE FOR HEADSHEAVES & GUIDE PULLEYS

Table B.1. Moment of Inertia Table for HS & GP

Diameter (mm)	Inertia (kgm²)				
	1-rope	2-ropes	4-ropes	6-ropes	8-ropes
1750	600	1200	2400	3600	4800
2000	750	1500	3000	4500	6000
2250	950	1900	3800	5700	7600
2500	1150	2300	4600	6900	9200
2750	1360	2700	5400	8160	10900
3000	2000	4000	8000	12000	16000
3250	2300	4600	9200	13800	18400
3500	3060	6120	12240	18360	24500
3750	3600	7600	15200	22800	30400
4000	5300	10600	21200	31800	42400
4250	6800	13600	27200	40800	54400
4500	9100	18200	36400	54600	72800
4750	11000	22000	44000	66000	88000
5000	14000	28000	56000	84000	112000
5250	16000	32000	64000	96000	128000
5500	19500	39000	78000	117000	156000
5750	23000	46000	92000	138000	184000
6000	29000	58000	116000	174000	232000
6250	32000	64000	128000	192000	256000
6500	37000	74000	148000	222000	296000
6750	43000	86000	172000	258000	344000
7000	51000	102000	204000	306000	408000

APPENDIX C

MOMENT OF INERTIA TABLE FOR FRICTION WHEEL

Table C.1. Moment of Inertia Table for Friction Wheel

Diameter (m)	Inertia (kgm^2) 1-rope
1.75	3000
2	4000
2.25	5400
2.5	6900
2.75	9500
3	16300
3.25	21000
3.5	28000
3.75	36000
4	44000
4.25	55800
4.5	78000
4.75	92000
5	115000
5.25	121000
5.5	143000
5.75	171000
6	199000
6.25	212000
6.5	225000
6.75	270000
7	285600

APPENDIX D

MOMENT OF INERTIA TABLES OF WINDING MOTORS & REDUCTION GEARS

D.I. A.C. DRIVE MOTOR (3.3 kV)

Table D.1. Moment of Inertia Table for AC Motor / 3.3 kV

kW	RPM				
	300	333	375	428	500
200	520	485	460	320	220
400	950	720	650	520	350
600	1480	1150	950	550	490
800	2200	1600	1300	1100	700
1000	3050	2250	1750	1500	850
1200	4100	3000	2250	1900	1000
1400	N/A	4000	2750	2350	1200
1600	N/A	N/A	3350	2850	1400
1800	N/A	N/A	N/A	3400	1650
2000	N/A	N/A	N/A	3950	1950

D.II. A.C. DRIVE MOTOR (6.6 kV)

Table D.2. Moment of Inertia Table for AC Motor / 6.6 kV

kW	RPM				
	300	333	375	428	500
400	N/A	N/A	1150	1210	1260
600	N/A	N/A	1600	1500	1300
800	N/A	N/A	2300	1800	1400
1000	N/A	N/A	2950	2230	1650
1200	N/A	N/A	3650	2650	1750
1400	N/A	N/A	4200	3450	2200
1600	N/A	N/A	4840	4000	2450
1800	N/A	N/A	5580	4640	2730
2000	N/A	N/A	6440	5380	3040

D.III. A.C. DRIVE MOTOR (11.0 kV)

Table D.3. Moment of Inertia Table for AC Motor / 11.0 kV

kW	RPM				
	300	333	375	428	500
600	N/A	N/A	1900	1800	1500
800	N/A	N/A	2700	2160	1680
1000	N/A	N/A	3500	2680	1980
1200	N/A	N/A	4380	3180	2100
1400	N/A	N/A	5000	4100	2600
1600	N/A	N/A	5800	4800	2800
1800	N/A	N/A	6600	5500	3200
2000	N/A	N/A	7600	6300	3600

D.IV. D.C. DRIVE DIRECT COUPLE MOTOR (DCDD)

Table D.4. Moment of Inertia Table for DCDD Motor

kW	RPM				
	45	47.5	50	55	60
750	9500	9000	8550	7770	7120
1000	13300	12600	11970	10880	9970
1500	22380	21200	20140	18300	16780
1750	24380	23100	21900	20000	18300
2000	26390	25000	23750	21600	19800
2500	32800	31100	29500	26900	24600
3000	39300	37200	35340	32100	29450
4000	65700	62200	59100	53700	49200
5000	92000	87100	82800	75300	69000

D.V. D.C. DRIVE GEARED MOTOR (DCGD)

Table D.5. Moment of Inertia Table for DCGD Motor

kW	RPM					
	300	400	500	600	700	800
300	470	350	280	233	200	175
400	630	470	380	315	270	235
500	790	590	470	390	340	300
600	940	700	560	470	400	350
700	1100	825	660	550	470	410
800	1260	945	760	630	540	475
900	1410	1060	850	700	600	530
1000	1570	1170	940	785	670	590
1200	1890	1420	1130	945	810	710
1400	2200	1650	1320	1100	940	820
1600	3000	2250	1800	1500	1290	1125
1800	3400	2550	2040	1700	2460	1280
2000	4000	3000	2400	2000	1710	1500
3000	5800	4350	3480	2900	2490	2200
4000	8000	6130	4900	4080	3500	3060

D.VI. REDUCTION GEAR

Table D.6. Moment of Inertia Table for Reduction Gear

kW	RPM				
	300	333	375	428	500
300	4000	3500	2000	1500	0
400	6000	5000	4000	2500	1500
500	8000	7000	5050	4000	2500
600	9500	8500	7000	5000	4000
700	11500	10500	8000	6500	5000
800	13500	12000	10000	8000	6000
900	15000	14000	12000	9500	7500
1000	17000	15500	13500	11000	9000
1200	21000	19000	16500	14000	11500
1400	24500	22000	19500	17000	14000
1600	28000	25500	23000	19500	16500
1800	32000	29000	26000	22500	19000
2000	35500	32000	29000	25000	21500