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School of mechanical and industrial engineering

DESIGN OF VARIABLE SPEED VIBRATING PLATE SOIL COMPACTOR

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Declaration

I hereby declare that the work which is presented in this thesis entitled “***design of variable speed vibrating plate soil compactor***” is original work of my own, has not been submitted elsewhere for a degree/diploma and that all sources of material used for the thesis have been duly acknowledged.

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This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

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Abstract

When soil is dug up or moved, it becomes aerated and loses some of its density. It must be compacted or allowed to settle before a structure (house, buildings, roads etc.) is built on top of it. Potholes in roads, sinkholes in parking lots, and severe cracking or settling in foundations are very often the result of a poorly compacted base. Now a day's different compactor machine are used to compact different soil texture. Each compaction machines are designed to compact for specific type of soil.

The motive of this research is to design variable speed vibrating plate soil compactor that can bring uniform compactness and can increase the load bearing capacity of different soil texture. This thesis research focuses on design of variable speed vibrating plate compactor, which has the property and characteristics of four deferent compactor machines such as sheep's foot roller, smooth wheel roller, rammer and vibrating plate compactor.

Therefore it will avoid selecting of different compaction machine for different soil texture to bring uniform soil compactness. If this design finally be manufactured, it will work for any kind of soil texture and save space, time, and cost to bring uniform soil compactness even if soil texture is different from distance to distance.

In this thesis research there are best solutions principles based on weighted objective method for different components of the variable speed vibrating plate compactor and choose different best option by computing based on different criteria.

This thesis research has addressed detail design of different components of variable speed vibrating plate compactor such as design of Rotating shaft, Hydraulically controlled variable eccentricity Eccentric Rotating Mass, gear, Pulley, Belt, springs, Vibration Isolator (Damper) and Hydraulically controlled variable surface Vibrating Plate, selection bearings, pump and prime mover And also indicted the vibration analysis and vibration response of the machine to soil damping and spring constants.

Variable speed vibrating plate compactor has the ability of maximum efficiency to compact 100m² areas within 600second on single trip which is within acceptable range of compaction time when it compared with exiting compaction machine.. The estimated total market cost is 223,314 birr which is relatively low-priced since it replaces four different compactor machines.

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List Symbols and Abbreviation

γ_d	Dry unit weight
G_s	Soil grains
E	Compaction energy
e	Void ratio
ω'	Angular speed
θ -	Angel to which the two ERM meet from the vertical
C_t -	Coefficient of elastic uniform shear and
C_z	Coefficient of elastic uniform compression
C_θ	Coefficient of elastic non-uniform compression
C_φ	Coefficient of elastic non-uniform shear
K_z	Spring coefficient For vertical motion
K_t	Spring coefficient horizontal (or sliding) motion
K_θ	Spring coefficient for rocking motion
K_φ	Spring coefficient For torsion motion (rotation about vertical axis)
ξ	Damping ratio
ψ	Damping angle
γ_p	Plastic strain
τ_o	Shear stress
T	Transmissibility
n	Factor of safety
M_a	Amplitude moment component

M_m	Midrange moment component
σ_m	Midrange stress component
σ_a	Amplitude stress component
σ_{max}	Maximum stress
σ_{min}	Minimum stress
S_e	Endurance limit of the material
S_{ut}	Ultimate strength of the material
k_f	Stress concentration factor subjected to bending load
$k_{f.s}$	Stress concentration factor subjected to torsional load
C_{10}	Catalog rating in lbf or KN
F_D	Desired radial load in lbf or KN
L_R	Rating life in hour
L_D	Desired life in hour
n_R	Rating speed in rev/min
n_D	Desired speed in rev/min
Z.O.R	Zero void ratio
w_{opt}	Optimum Moisture Content
ERM	Eccentric rotating mass

CHAPTER ONE

1. INTRODUCTION .

1.1 Backgrounds

Potholes in roads, sinkholes in parking lots, and severe cracking or settling in foundations are very often the result of a poorly compacted base. With much attention focused on a structure, the condition of the ground beneath is frequently neglected. This can cause a myriad of problems for homeowners, business owners, and road districts in the years to come [1].

When soil is dug up or moved, it becomes aerated and loses some of its density. It must be compacted or allowed to settle before a structure (i.e., house, building, parking lot, sidewalk, etc) is built on top of it. When the soil naturally settles beneath a structure, a void remains between the bottom of the structure and the base material itself. This void provides no support for whatever structure rests upon the base, thus beginning a chain reaction of structural deterioration [3].

If the soil is not compacted beneath a foundation, it may crack and deteriorate causing the house or building to "settle". Cracks and leaks in basement walls, cracks in drywall, and sagging concrete joints are examples of a poorly compacted base beneath a house. Little or no compaction of soil beneath a driveway or parking lot may eventually lead to sinkholes and potholes that can be very expensive to repair [1].

A wide range of machines are available for soil compaction such as rammers, vibratory plate compactors, and trench rollers. Rammers (often called tampers or jumping jacks) use a shoe that pounds the soil to compact it. Rammers are vertical machines that are well suited for narrow trenches where a plate compactor may not fit. Vibratory plate compactors are walk behind compaction machines that use a wide vibrating steel plate that compacts the base. Plate compactors are often used with water on asphalt applications. Reversible plate compactors are usually bigger than their forward-direction-only counterparts, and are better suited for bigger jobs. Vibratory trench rollers (usually remotely controlled or ride-on) have vibrating drums with feet that compact soil in deep trenches where it can be dangerous for workers [13].

The motive of this research is to design variable speed vibrating plate soil compactor that can bring uniform compactness and can increase the load bearing capacity of different soil texture.

Therefore it will avoid selecting of different compaction machine to different soil texture to bring uniform soil compactness. If this research finalized, it will work for any kind of soil texture and save space, time, and cost to bring uniform soil compactness even if soil texture is different from distance to distance.

1.2 Problem statement

The desired level of compaction is best achieved by matching the soil type with its proper compaction method. While making foundation on site there is problem of different soil load carrying capacity or different soil texture in 2 or 5m distance. [2] This means when the soil texture differs with in this distance it implies that it has different load carrying capacity according to specified soil texture. Soil compaction is defined as the method of mechanically increasing the density of soil. In construction, this is a significant part of the building process. If performed improperly, settlement of the soil could occur and result in unnecessary maintenance costs or structure failure. Almost all types of building sites and construction projects utilize mechanical compaction techniques. Now a day's different compactor machine are used to compact different soil texture [1].

As mention above soil structure may vary from distance to distance so that to prepare foundation for 100m² area it should be used at list two type of compactor machine [9]. On market there are different compactor machine in according to soil texture. This means it consume time, cost and space to maintain similar compacted soil. Each compacting machines are designed to compact for specific type of soil. [2] since each compacting machines deliver different level of energy according to different soil texture depending on different level of speed; to overcome this problem it require design of variable speed vibrating plate compactor which deliver all level of energy required to different soil texture to come up with uniform level of compactness.

1.3 Objectives of the research

This research aims to design of variable speed operating vibrating plate compactor as an alternative method of increasing density/compaction of any type of soil.

General objective

- Design of variable speed vibrating plate compactor which provide all level compaction energy and criteria such as pressure, kneading, impact and vibration as per required by varying the speed, amplitude and surface characteristic of vibrating plate .

Specific objectives

- Selecting the best solution for components of variable speed vibrating plate compactor based solution principle.
- Design different components of the variable speed vibrating plate compactor based computed solution principle.
- Improving the time required to compact 100m² areas in comparison with existing mini compactor machines. .
- Calculating the total market cost of variable speed vibrating plate compactor and compare it with the existing compactor machines.

1.4 Significance of the research

The majority of construction enterprise uses different kind of compaction machines such as oil injected rammer, double drums roller and single speed operating vibrating plate compactor simultaneously and at least they require 6 persons to operate those compaction machines. The uses of all listed compactor have disadvantages such as high cost that used to rent or to buy compactors and salary to pay operators, consume time and power require to compact soil and space used to store equipments. But if it is used variable speed vibrating plate compactor it can overcome better and uniform compactness even if soil texture is varied from distance to distance by varying speed, amplitude and compaction energy. Use of variable speed vibrating plate compactor avoids requiring of huge compaction machines for different type of soil and also saves time, cost and space.

1.5 Scope and limitation of the research

The scope of this research is limited to design of variable speed operating vibrating plate compactor. There are limitations to manufacture and conduct Series of experiment on the model to test the feasibility of compactor for different soil textures and identify factors that affect the overall efficiency of the compactor and also determine the performance of compactor per area due to manufacturing cost.

1.6 Research methodology

The following methodology will be followed in order to achieve the objectives of the research.

a. Gathering information

Literature review

Since the variable speed operating vibrating plate compactor compact different soil texture based energy required to bring uniform compactness, Reading different literatures and journals written on the area of soil mechanics and soil texture types and properties of soil.

b. Site Data collection

Since compaction energy is varied by varying the speed eccentricity of ERM, there are requirement of onsite data gathering. It requires testing of load bearing capacity of different soil textures and the energy required to bring uniform compactness throughout the soil textures. Based on this energy analysis, design of variable speed vibrating plate compactor will start.

c. Designing model

One of the methodologies is to develop the physical model. The design of vibrating plate compactor operate at variable speed would include determining of the appropriate weight required to absorb energy, the maximum and minimum range of amplitude and frequency, the correct amount of power input of prime mover.

CHAPTER TWO

2. LITERATURE REVIEWS

Soil compaction

Many types of earth construction, such as roads, railways, embankments, retaining walls, earth dams, and airport, require man-placed soils, or fills. These soils are loose (weak) and must be compacted to increase their strength characteristics.

Soil compaction is defined as the method of mechanically increasing the density of soil by reducing volume of air. It is the process in which a stress applied to a soil causes densification as air is displaced from the pores between the soil grains. When stress is applied that causes densification due to water (or other liquid) being displaced from between the soil grains then consolidation, not compaction, has occurred. Normally, compaction is the result of heavy machinery compressing the soil, but it can also occur due to the passage of (e.g.) animal feet [7].

In construction, Soil compaction is a vital part of the construction process. It is used for support of structural entities such as building foundations, roadways, walkways, and earth retaining structures to name a few. For a given soil type certain properties may deem it more or less desirable to perform adequately for a particular circumstance. In general, the preselected soil should have adequate strength, be relatively incompressible so that future settlement is not significant, be stable against volume change as water content or other factors vary, be durable and safe against deterioration, and possess proper permeability.

- The densification is accomplished by pressing the soil particles together into a close state of contact with air being expelled from the soil mass in the process, thereby increasing its unit weight.
- The degree of compaction is measured in terms of the dry unit weight
- For given water content, the maximum degree of compaction that can be achieved is when all of the air voids have been removed.
- The dry unit weight correlates with the degree of packing of the soil grains:
- The smaller the void ratio will be, the higher the dry unit weight will be

When an area is to be filled or backfilled the soil is placed in layers called lifts. The ability of the first fill layers to be properly compacted will depend on the condition of the natural material

being covered [14]. If unsuitable material is left in place and backfilled, it may compress over a long period under the weight of the earth fill, causing settlement cracks in the fill or in any structure supported by the fill. In order to determine if the natural soil will support the first fill layers, an area can be proof rolled. Proof rolling consists of utilizing a piece heavy construction equipment (typically, heavy compaction equipment or hauling equipment) to roll across the fill site and watching for deflections to be revealed. These areas will be indicated by the development of rutting, pumping, or ground weaving.

To ensure adequate soil compaction is achieved, project specifications will indicate the required soil density or degree of compaction that must be achieved. These specifications are generally recommended by a geotechnical engineer in a geotechnical engineering report.

In earlier days, embankment design and construction were not given adequate attention. Embankments were constructed and left for compaction by natural process. Due to loads imposed by heavier axle loads, very high degree of sub-grade support have become necessary in present scenario which requires fast and heavy compaction by suitable compacting equipment's. Many types of compacting equipment's are available now a day for compacting different types of soils to be used in earthwork [9].

Compaction is the process of increasing the density of soil by mechanical means by packing soil particles closer together with reduction of air voids and to obtain a homogeneous soil mass having improved soil properties. Compaction brings many desirable changes in soil properties as follow [2]:

- Helps soil to acquire increase in shear strength
- Reduces compressibility thus minimizing uneven settlement during service
- Increase density and reduces permeability, thereby reducing susceptibility to change in moisture content
- Reduction in erodability
- Results in homogeneous uniform soil mass of known properties
- Reduction in frost susceptibility in cold regions
- Helps the pavement designer in assessing the sub-grade strength to a reasonably accurate strength and thereby produce a safe and economical design

- Results in little change in volume under traffic loads, thus minimizing deformation and maintaining good ride ability characteristics of the pavement
- Reduces expenditure on maintenance of formation during service

The densification is accomplished by pressing the soil particles together in to a close state of contact with air being expelled from the soil mass in the process, thereby increasing its unit weight.

The degree of compaction is measured in terms of the dry unit weight (γ_d). For given water content (w), the maximum degree of compaction that can be achieved is when all of the air voids have been removed. The dry unit weight correlates with the degree of packing of the soil grains(G_s) [1]:

$$\gamma_d = \frac{G_s * \gamma_w}{1 + e} \dots \dots \dots \text{eq 2.1}$$

- The smaller the void ratio (e) will be, the higher the dry unit weight (γ_d) will be.

The dry density of the soil is increased by compaction. The increase in the dry density depends upon the following factors [7]:

Five factors affecting compaction

1. Physical & chemical properties
2. Moisture content
3. Method of compaction
4. Amount of compaction effort
5. Thickness of layer or “lift” being compacted

Water content:

At low water content, the soil is stiff and offers more resistance to compaction. As the water content is increased, the soil particles get lubricated. The soil mass becomes more workable and the particles have closer packing. The dry density of the soil increases with an increase in the water content till the optimum water content is reached. At that stage, the air voids attain approximately a constant volume. With further increase in water content, the air voids do not decrease, but the total voids (air plus water) increase and the dry density decreases. In soils, compaction is a function of water content (w).The soil particles will slip more on each other causing more reduction in the total volume, which will result in adding more soil and, hence, the

dry unit weight (γ_d) will increase, accordingly, to a maximum point $(\gamma_d)_{\max}$. Beyond certain moisture content, any increase in moisture content tends to reduce the dry unit weight. The moisture content at which the maximum dry unit weight is attained is generally referred to as the Optimum Moisture Content (w_{opt}) [13].

The laboratory test generally used to obtain the maximum dry unit weight of soil compaction and the optimum moisture content is called the Proctor Compaction Test (Proctor, 1933).

Thus the higher dry density is achieved up to the optimum water content due to forcing air voids out from the soil voids. After the optimum water content is reached, it becomes more difficult to force air out and to further reduce the air voids [16].

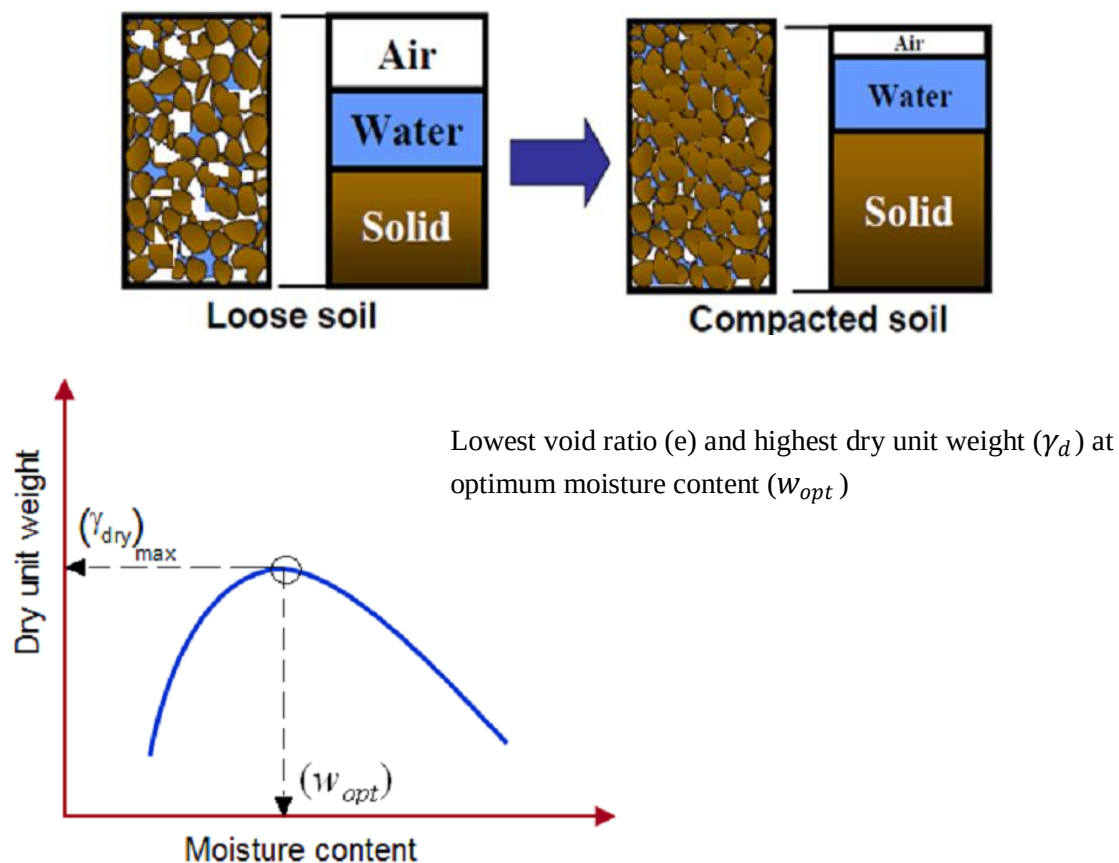


Figure 2.1:- graph of moisture content vs. dry unit weight [2]

Proper control of moisture content in soil is necessary for achieving desired density. Maximum density with minimum compacting effort can be achieved by compaction of soil near its OMC (Optimum Moisture Content). If natural moisture content of the soil is less than OMC, calculated amount of water should be added to soil with sprinkler attached to water tanker and mixed with

soil by motor grader for uniform moisture content. When soil is too wet, it is required to be dried by aeration to reach up to OMC [2].

Amount of compaction:

The effect of increasing the amount of compaction effort is to increase the maximum dry density and to decrease the optimum water content. At water content less than the optimum, the effect of increased compaction is more predominant. At water content more than the optimum, the volume of air voids become almost constant and the effect of increased compaction is not significant.

It may be mentioned that the maximum dry density does not go on increasing with an increase in the compaction effort. For a certain increase in the compaction effort, the increase in the dry density becomes smaller and smaller. Finally a stage is reached beyond which there is no further increase in the dry density with an increase in the compaction effort [15].

The line of optimums which join the peaks of the compaction curves of different compaction efforts follows the general trend of the zero-air void. This line corresponds to air voids of about 5% [2].

In modern construction projects, heavy compaction machinery is deployed to provide compaction energy. Types of machinery required are decided based on type of soil to be compacted. The method of compaction is primarily of four types such as **kneading, static (pressure), and dynamic (impact) and vibratory compaction.**

Different type of compaction is effective in different type of soils such as for

- Cohesive soils; sheep's foot rollers or pneumatic rollers provide the kneading action.
- Silt soils can be effectively compacted by sheep foot roller/pneumatic roller or smooth wheel roller.
- Compacting sandy and gravelly soil, vibratory rollers are most effective.
- If granular soils have some fines, both smooth wheel and pneumatic rollers can be used.

The compaction energy (E) per unit volume used for the standard Proctor test can be given as [21]:

$$E = \frac{\left[\begin{array}{c} \text{Number of blows} \\ \text{per layer} \end{array} \right] \times \left[\begin{array}{c} \text{number} \\ \text{of layers} \end{array} \right] \times \left[\begin{array}{c} \text{weight of} \\ \text{hammer} \end{array} \right] \times \left[\begin{array}{c} \text{height of drop} \\ \text{of hammer} \end{array} \right]}{(\text{volume of mold})} \dots \dots \dots \text{eq 2.2}$$

As the compaction effort increases, the maximum dry unit weights increases and the optimum moisture content decreases.

- **Line of Optimum:** A line drawn through the peak points of several compaction curves at different compaction efforts for the same soil, almost parallel to zero-air-void (z. a. v) curve.

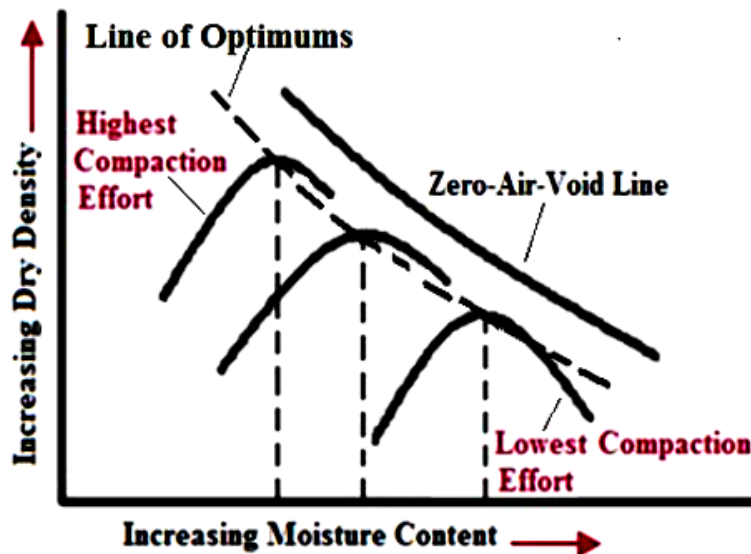


Figure 2.2:- graph of moisture content vs. dry unit weight vs. corresponding compaction effort [2]

Type of soil:

Type of soil has a great influence on its compaction characteristics. Normally, heavy clays, clays and silt offer higher resistance to compaction where as sandy soils and coarse grained or gravelly soils are amenable for easy compaction. The coarse-grained soils yield higher densities in comparison to clays. A well-graded soil can be compacted to higher density.

The dry density achieved depends upon the type of soil. In general, coarse grained soils can be compacted to higher dry density than fine-grained soils. With the addition of even a small quantity of fines to a coarse-grained soil, the soils attain a much higher dry density for the same compaction effort. However, if the quantity of the fines is increased to a value more than that required to fill the voids of the coarse-grained soils, the maximum dry density decreases. Well graded sand attains a much higher dry density than a poorly graded soil [7].

Cohesive soils have high air voids. These soils attain a relatively lower maximum dry density as compared with the cohesion less soils. Such soils require more water than cohesion less soils and

therefore the optimum water content is high. Heavy clays of very high plasticity have very low dry density and very high optimum water content. Fine grain soil needs more water to reach optimum and coarse grain soil needs less water to reach optimum.

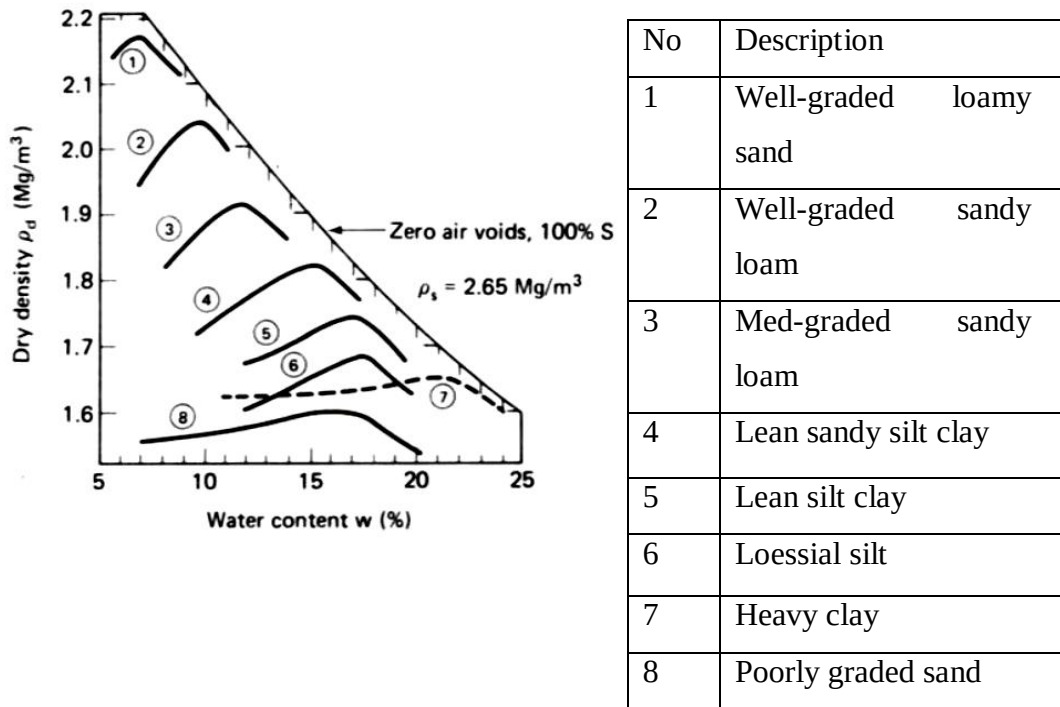


Figure 2.3:- water content vs. different dry density with corresponding soil type [2]

Layer thickness

The more the thickness of layer of earth subjected to field compaction, the less the energy input per unit weight of soil and hence, less is the compaction under each pass of the roller. Suitable thickness of soil of each layer is necessary to achieve uniform thickness. Normally, 200-300 mm layer thickness is optimum in the field for achieving homogeneous compaction [5].

Since this thesis project title is new even if I put a lot of effort on to this, I cannot find the literature or journal based on this title. But different compactor machines characteristics are summarized as table.

The following table is summarized that shows different compactors in accordance with recommended soil type and their specifications [1].

Table 2.1:- compactor machine, soil type & contact pressure [1]

Type of compacting machines	Type of soil	Contact pressure	Type of compaction effort	Depth of compaction
Smooth-wheel roller	All except rocky soil	380kPa	Static weight	200-300mm
Pneumatic rubber-tired roller	Granular and fine grained soil	700kPa	Static weight & kneading	100-250mm
Sheep's-foot roller	Clayed soil	1400-700kPa	Static weight & kneading	250-500mm

Table 2.2:- compactor machine, soil type & blows per minute [1]

Type of compaction machine	Type of soil	Type of compaction effort	No. of blows per minute
Vibratory roller	Clean sand and gravels	Static weight and vibration	2000-6000
Vibratory plate compactor	Granular, rocky soil, clay soil	Static weight and vibration	1500-2500
rammer	Granular, rocky soil, and clay soil	Impact and vibration	800

Table 2.3:- recommended compactor machine according to soil type [5]

Equipment application				
Compactor machines	Granular soil	Sand and clay	Cohesive clay	Asphalt
rammers	Not recommended	Testing recommended	Best application	Not recommended
Vibrating plate	Best application	Testing recommended	Not recommended	Best application
Reversible plate	Testing recommended	Best application	Best application	Not recommended
Vibratory roller	Not recommended	Best application	Testing recommended	Best application
Rammax roller	Testing recommended	Best application	Best application	Not recommended

Contact Pressure

Contact pressure depends on the weight of the compactor and the contact area. In case of pneumatic roller, the tire inflation pressure also determines the contact pressure in addition to wheel load. A higher contact pressure increases the dry density and lowers the optimum moisture content [1].

Number of roller passes (blows)

Density of the soil increases with the number of passes of rollers or blows but after optimum number of passes, further increase in density is insignificant for additional number of cases. For determination of optimum number of passes for given type of roller and optimum thickness of layer at a predetermined moisture content, a field trial for compaction is necessary as indicated in Para 6.3.5 of “Guidelines for Earthwork in Railway Projects, July 2003” and para 5.0 of this report.

Speed of rolling

Speed of rolling has a very important bearing on the roller output. The greater the speed of rolling the more the length of embankment that can be compacted in one day. Speed was found to be a significant factor for vibratory rollers because its number of vibrations per minute is not related to its forward speed. Therefore, the slower the speed of travel, the more vibrations at a given point and lesser number of pass required to attain a given density [1].

CHAPTER THREE

3. PRODUCT PLANNING

Product planning is a systematic search for, selection and development of product ideas. Product planning is a process used to identify and develop new products. The purpose of planning is to make choices about which product ideas a company should invest in [20].

3.1 Market Analysis

Around 95% of Ethiopian construction companies or enterprises rent or bought different kind of construction equipment [according to defense construction enterprise]. Amongst this construction equipment one is for compaction. In order to compact or increase the load bearing capacity of small area of soil they use different kind of compaction machinery such as rammer, sheep's-foot roller, reversible vibratory plate compactor and the like due to the variation of soil type and property; since each compaction machinery are used according to different soil type, the desired level of compaction is best achieved by matching the soil type with its proper compaction method. While making foundation on site there is problem of different soil load carrying capacity or different soil texture in 2 or 5m distance [2]. This means when the soil texture differs with in this distance it implies that it has different load carrying capacity according to specified soil texture.

According to different soil type it is recommended to use different compaction machines as per soil type. Hence using these all compaction machinery to compact small area of soil increases the cost, time and energy wastage. Since the demand for quality construction is increasing now days in Ethiopia; and also quality construction start with quality compaction, variable speed vibrating plate compactor is also unquestionable choice since it offers different kind of compaction energy depending on soil type. Small or in house construction is emerging now days and they are facing failures like crack and path holes. This occurs due to improper soil compaction at base foundation because they use manual compaction method like compacting soil with wide hammer. Because of this reasons variable speed vibrating plate compactor have more efficiencies with less energy, less cost, less time consumption, avoiding cracks and better quality compacted soil. The demand of compactor machines in Ethiopia is now emerging since the construction industries are also developing, improving the quality, strength and efficiency of variable speed vibrating plate compactor is the first and for most priority holder to improve our construction industries.

3.2 Economic policy

The construction industries in Ethiopia are now one of the emerging and developing economic sectors. Since the development of this industry would have an economic impact on the country GDP, the improvement and development of this industry must be kept on the track. Starting the production of construction machineries like variable speed vibrating plate compactor in Ethiopia avoids foreign currency and brings a significant change to GDP by avoiding the cost allocated to buy different compaction machine. Since these industries dominated by subsistence, costly and time consumable tiresome compaction system; construction industry holders or enterprises should give emphasis and strong commitment to improve their compaction mechanisms.

Product selection

For product selection there has to be a feasibility study to assess the quality viability of the proposed business. The feasibility study needs to answer the question does the idea make economic sense. Since the current Ethiopian policy encourages technologies sector such as manufacturing construction machinery is the perfect and irresistible idea. Design and manufacturing of variable speed vibrating plate soil compactor here in Ethiopia would increase job opportunity and decrease foreign currency and perfectly economical. Plus if it is used variable speed operating vibrating plate compactor it can overcome better and uniform compactness even if soil texture is varied from distance to distance by varying speed and amplitude. Using of this product also saves operating cost, operating time and since it can be operated by one person it also avoids manpower usage. So that variable speed vibrating soil compactor is selected.

3.3 Product Definition

Product definition involves the specification of most important features and requirements of the final product. The important features that variable speed vibrating soil compactor have power source, handle, dumper (shock absorber), vibrating plate, eccentric rotating mass, bearings, bearing house, pulley, belt, rotating shaft, case cover, hydraulic piston, hydraulic pump, valves, hydraulic hose, electric cable, power switch and different bolts and nuts.

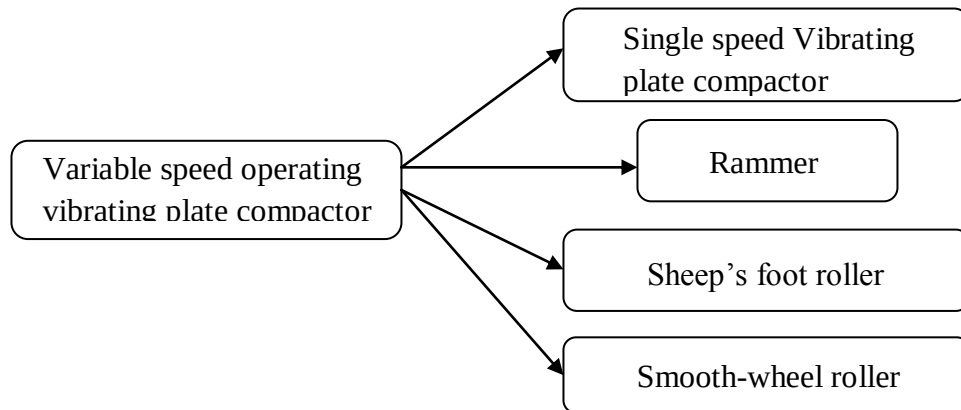


Figure 3.1:- different property of variable speed vibrating plate compactor

3.3.1 Power source

Power source is the part of variable speed vibrating plate compactor where it gets primary source of power. Power source sometimes called prime mover since it is key source of motion. Prime mover is either diesel engine or electric motor.

3.3.1.1 Electric motor

An **electric motor** is an electric machine that converts electrical energy into mechanical energy. Electric motors are used to produce linear or rotary force (torque), in normal motoring mode, most electric motors operate through the interaction between an electric motor's magnetic field and winding currents to generate force within the motor. Electric motors may be classified by electric power source type, internal construction, application, type of motion output, and so on.

The design of AC induction and synchronous motors is optimized for operation on single-phase or poly phase sinusoidal or quasi-sinusoidal waveform power such as supplied for fixed-speed application from the AC power grid or for variable-speed application from VFD controllers. There are different kinds of motor, among these:-

3.3.1.1.1 Induction motor

An induction motor is an asynchronous AC motor where power is transferred to the rotor by electromagnetic induction, much like transformer action. An induction motor resembles a rotating transformer, because the stator (stationary part) is essentially the primary side of the transformer and the rotor (rotating part) is the secondary side. Poly phase induction motors are widely used in industry.

As the motor accelerates, the slip frequency becomes lower, and more current is in the interior of the winding.

3.3.1.1.2 Torque motor

Torque motor is a specialized form of electric motor which can operate indefinitely while stalled, that is, with the rotor blocked from turning, without incurring damage. In this mode of operation, the motor will apply a steady torque to the load. The torque motors can also achieve fast-forward and rewind operation without requiring any additional mechanics such as gears or clutches. In the computer gaming world, torque motors are used in force feedback steering wheels.

3.3.1.1.3 Synchronous motor

A synchronous electric motor is an AC motor distinguished by a rotor spinning with coils passing magnets at the same rate as the AC and resulting magnetic field which drives it. Another way of saying this is that it has zero slip under usual operating conditions. Contrast this with an induction motor, which must slip to produce torque. One type of synchronous motor is like an induction motor except the rotor is excited by a DC field. Slip rings and brushes are used to conduct current to the rotor. The rotor poles connect to each other and move at the same speed hence the name synchronous motor.

3.3.1.2 Diesel engine

The diesel engine (also known as a compression-ignition engine) is an internal combustion engine that uses the heat of compression to initiate ignition and burn the fuel that has been injected into the combustion chamber. This contrasts with spark-ignition engines such as a petrol engine (gasoline engine) or gas engine (using a gaseous fuel as opposed to gasoline), which use a spark plug to ignite an air-fuel mixture.

The diesel engine has the highest thermal efficiency of any standard internal or external combustion engine due to its very high compression ratio and inherent lean burn which enables heat dissipation by the excess air. Diesel engines have several advantages over other internal combustion engines:

They burn less fuel than a petrol engine performing the same work, due to the engine's higher temperature of combustion and greater expansion ratio. Gasoline engines are typically 30% efficient while diesel engines can convert over 45% of the fuel energy into mechanical energy. They have no high voltage electrical ignition system, resulting in high reliability and easy adaptation to damp environments. The absence of coils, spark plug wires, etc., also eliminates a

source of radio frequency emissions which can interfere with navigation and communication equipment, which is especially important in marine and aircraft applications.

The longevity of a diesel engine is generally about twice that of a petrol engine due to the increased strength of parts used. Diesel fuel has better lubrication properties than petrol as well. Diesel fuel is considered safer than petrol in many applications.

3.3.2 Vibration isolator (Dumper)

Damping is the phenomenon by which mechanical energy is dissipated (usually by conversion in to internal thermal energy) in dynamic systems. Knowledge of the level of damping in a dynamic system is important in the utilization, analysis, and testing of the system. This effect could be further magnified by low-frequency support structures and panels with low damping. This example shows that knowledge of damping in constituent devices, components, and support structures is important in the design and operation of complex mechanical systems. The nature and the level of component damping should be known in order to develop a dynamic model of the system and its peripherals. Knowledge of damping in a system is also important in imposing dynamic environmental limitations on the system (that is, the maximum dynamic excitation the system can withstand) under in-service conditions [11].

Several types of damping are inherently present in a mechanical system. If the level of damping that is available in this manner is not adequate for proper functioning of the system then external damping devices may be added either during the original design or during subsequent design modifications of the system [11].

Three primary mechanisms of damping are important in the study of mechanical systems. They are:

1. Internal damping (of material)
2. Structural damping (at joints and interfaces)
3. Fluid damping (through fluid –structure interactions)

Internal (material) damping results from mechanical energy dissipation within the material due to various microscopic and macroscopic processes. Structural damping is caused by mechanical energy dissipation resulting from relative motions between components in a mechanical structure that has common points of contact, joints or supports. Fluid damping arises from the mechanical

energy dissipation resulting from drag forces and associated dynamic interactions when a mechanical system components move in a fluid [11].

Two general types of external dampers may be added to a mechanical system in order to improve its energy dissipation characteristics. They are:

1. Passive dampers
2. Active dampers

Passive dampers are devices that dissipate energy through some kind of motion, without needing an external power source or actuators. Active dampers have actuators that need external sources of power. Dampers may be considered as vibration controllers [11].

3.3.3 Eccentric rotating mass (ERM)

Eccentric mass vibrators are attached to equipment and structures to determine their response to sinusoidal forcing over a range of frequencies in order to estimate resonant frequencies, damping, mode shapes, and other dynamic properties. Eccentric mass vibrators use one or more rotating eccentric weights to produce a force which increases in direct proportion to the eccentricity and to the square of the rotating frequency [4].

The centripetal force of the offset mass is asymmetric, resulting in a net centrifugal force, and this causes a displacement of the machine. With a high number of revolutions per minute, the machine is constantly being displaced and moved by these asymmetric forces. It is this repeated displacement that is perceived as a vibration [4].

The vibration produced by ERMs is an example of “Driven Harmonic Vibration”. This means there is an external driving force causing the system to vibrate, and this is also sometimes called forced vibration. The term ‘harmonic’ means that the system is forced to vibrate at the frequency of the excitation [4].

A single eccentric weight produces a unidirectional rotating force vector, which can be expressed as a force two perpendicular sinusoidal forces, with a 90 degree phase lag. The use of equal counter rotating eccentric weights produces a unidirectional sinusoidal force. Most eccentric weights are adjustable from zero to full eccentricity, so as to allow a wide range of frequencies to be covered with approximately the same force. Two or more vibrators, spaced apart on a structure, can provide torsional forcing, or enhance the motion of one particular mode over another..

3.3.4 Rotating shaft

A shaft is the component of a mechanical device that transmits rotational motion and power. It is integral to any mechanical system in which power is transmitted from a prime mover, such as an electric motor or an engine, to other rotating parts of the system [11].

A shaft is a rotating member, usually of circular cross section. It provides the axis of rotation, or oscillation, of elements such as gears, pulleys, flywheels, cranks, sprockets, and the like and controls the geometry of their motion. Rotating steel shafts are used throughout the materials handling, mining and port industries wherever power transmission or the mounting of rotating components in a mechanical system is required. Shaft failure can be potentially disastrous and can lead to large repair costs and long down times [11].

General Considerations

1. To minimize both deflections and stresses, the shaft length should be kept as short as possible and overhangs minimized.
2. A hollow shaft has a better stiffness/mass ratio (specific stiffness) and higher natural frequencies than a comparably stiff or strong solid shaft, but will be more expensive and larger in diameter.
3. Try to locate stress-raisers away from regions of large bending moment if possible and minimize their effects with generous radii and relief.
4. General Low carbon steel is just as good as higher strength steels (since deflection is typical the design limiting issue).

3.3.5 Vibrating plate

Vibrating plate is plate which is a key part of vibrating plate compactor. It is hydraulically controlled flat and sheep's footed plate which gives kneading, static and vibratory effort to soil to be compacted. It is contact surface between the compactor machine and soil.

3.3.6 Belts

A belt is a loop of flexible material used to mechanically link two or more rotating shafts, most often parallel. Belts may be used as a source of motion, to transmit power efficiently, or to track relative movement. Belts are looped over pulleys. In a two pulley system, the belt can either drive

the pulleys normally in one direction, or the belt may be crossed, so that the direction of the driven shaft is reversed

Belt drive is simple, inexpensive, and does not require axially aligned shafts. It helps protect the machinery from overload and jam, and damps and isolates noise and vibration. Load fluctuations are shock-absorbed (cushioned). They need no lubrication and minimal maintenance. They have high efficiency (90-98%, usually 95%), high tolerance for misalignment, and are of relatively low cost if the shafts are far apart. Clutch action is activated by releasing belt tension. Different speeds can be obtained by stepped or tapered pulleys [8].

There are different type of belt widely used such as flat belt, V belt, multi-groove belt and timing belts:-

3.3.6.1 Flat belts

The flat belt is a simple system of power transmission that was well suited for its day. It can deliver high power at high speeds (500 hp at 10,000 ft/min or 373 kW at 51 m/s), in cases of wide belts and large pulleys. But these wide-belt-large-pulley drives are bulky, consuming lots of space while requiring high tension leading to high loads, and are poorly suited to close-centers applications [8].

3.3.6.2 V belts

V belts (also style V-belts, vee) solved the slippage and alignment problem. It is now the basic belt for power transmission. They provide the best combination of traction, speed of movement, load of the bearings, and long service life. They are generally endless, and their general cross-section shape is trapezoidal (hence the name "V"). The "V" shape of the belt tracks in a mating groove in the pulley (or sheave), with the result that the belt cannot slip off. The belt also tends to wedge into the groove as the load increases, the greater the load, the greater the wedging action, improving torque transmission and making the V-belt an effective solution, needing less width and tension than flat belts [8].

3.3.6.3 Multi-groove belts

A multi-groove or poly groove belt is made up of usually 5 or 6 "V" shapes alongside each other. This gives a thinner belt for the same drive surface, thus it is more flexible, although often wider. The added flexibility offers an improved efficiency, as less energy is wasted in the internal friction of continually bending the belt. In practice this gain of efficiency causes a reduced heating effect on the belt and a cooler-running belt lasts longer in service.

A further advantage of the poly groove belt that makes them popular is that they can run over pulleys on the ungrooved back of the belt. Though this is sometimes done with Vee belts with a single idler pulley for tensioning, a poly groove belt may be wrapped around [8].

3.3.6.4 Timing belts

Timing belts, (also known as toothed) are a *positive* transfer belt and can track relative movement. These belts have teeth that fit into a matching toothed pulley. When correctly tensioned, they have no slippage, run at constant speed, and are often used to transfer direct motion for indexing or timing purposes (hence their name). They are often used in lieu of chains or gears, so there is less noise and a lubrication bath is not necessary. Timing belts need the least tension of all belts, and are among the most efficient. They can bear up to 200 hp (150 kW) at speeds of 16,000 ft/min (4,900 m/min) [8].

3.3.7 Pulley

Pulleys are a circular shaped or thick disk shaped components which used to transmit power, torque and energy to the shaft. Pulleys are also assembled as part of belt and chain drives in order to transmit power from one rotating shaft to another. It usually helps by multiply either speed or torque from the prime mover.

A pulley is a wheel on an axle that is designed to support movement and change of direction of a cable or belt along its circumference. Pulleys are used in a variety of ways to lift loads, apply forces, and to transmit power. In nautical contexts, the assembly of wheel, axle, and supporting shell is referred to as a "block."

Pulley is usually subjected to radial and hoop stress since it is the rotating part and connected to prime mover with belt in moderated tension [11].

3.3.8 Bearings

A **bearing** is a machine element that constrains relative motion and reduces friction between moving parts to only the desired motion. Many bearings also facilitate the desired motion as much as possible, such as by minimizing friction. Bearings are classified broadly according to the type of operation, the motions allowed, or to the directions of the loads (forces) applied to the parts. Bearings are manufactured to take pure radial loads, pure thrust loads, or a combination of the two kinds of loads. In a rolling bearing the starting friction is about twice the running friction, but still it is negligible in comparison with the starting friction of a sleeve bearing. Load, speed, and

the operating viscosity of the lubricant do affect the frictional characteristics of a rolling bearing [11].

3.3.9 Hydraulic pump

Hydraulic pumps are used in hydraulic drive systems and can be hydrostatic or hydrodynamic. A hydraulic pump is a mechanical source of power that converts mechanical power into hydraulic energy. It generates flow with enough power to overcome pressure induced by the load at the pump outlet. When a hydraulic pump operates, creates a vacuum at the pump inlet, which forces liquid from the reservoir into the inlet line to the pump and by mechanical action delivers this liquid to the pump outlet and forces it into the hydraulic system [19].

Pumps that discharge liquid in a continuous flow are non positive-displacement type. Pumps that discharge volumes of liquid separated by periods of no discharge are positive-displacement type.

Non positive-Displacement Pumps:- With this pump, the volume of liquid delivered for each cycle depends on the resistance offered to flow. A pump produces a force on the liquid that is constant for each particular speed of the pump. Resistance in a discharge line produces a force in the opposite direction. When these forces are equal, a liquid is in a state of equilibrium and does not flow [19].

Some pumps are made with variable displacement, pressure compensation capability. Such pumps are designed so that as system pressure builds up, they produce less flow. Pressure relief valves are not needed when pressure-compensated pumps are used.

Advantages over dynamic pumps:

- High pressure capability (up to 800 bar)
- Small, compact size
- High volumetric efficiency
- Small changes in efficiency throughout the design pressure range
- Great flexibility of performance

Three main types of positive displacement pumps:

- Gear
- Vane and
- piston

Vane and piston can be of variable displacement types.

3.3.10 Hydraulic cylinder

A hydraulic actuator receives pressure energy and converts it to mechanical force and motion. An actuator can be linear or rotary. A linear actuator gives force and motion outputs in a straight line.

Cylinders:-They extract energy from the pressurized fluid and convert it to mechanical energy to perform linear. Hydraulic cylinders extend and retract a piston rod to exert a force on an external load along a straight line path. A cylinder is a hydraulic actuator that is constructed of a piston or plunger that operates in a cylindrical housing by the action of liquid under pressure. Cylinder housing is a tube in which a plunger (piston) operates [19].

➤ **Single-Acting Cylinder:**-Single acting design is the simplest type of hydraulic cylinder; It can exert a force only in the extending direction. Retraction is accomplished by gravity or spring. This cylinder only has a head-end port and is operated hydraulically in one direction. When oil is pumped into a port, it pushes on a plunger, thus extending it. To return or retract a cylinder, oil must be released to a reservoir. A plunger returns either because of the weight of a load or from some mechanical force such as a spring.

3.3.11 Valve

Control is the most important consideration in any system. If control system components are not properly selected, the entire system will not function as required. Valves are used in fluid power system to control direction, pressure and flow [19].

Valves regulate pressure by creating special pressure conditions and by controlling how much oil will flow in portions of a circuit and where it will go. The three categories of hydraulic valves are pressure-control, flow (volume) control, and directional-control. Valves are rated by their size, pressure capabilities, and pressure drop/flow.

There are three basic types of valves:

- a) Directional control
- b) Pressure control
- c) Flow control

Pressure Control Valves:-A pressure-control valve may limit or regulate pressure, create a particular pressure condition required for control, or cause actuators to operate in a specific order. All pure pressure-control valves operate in a condition approaching hydraulic balance. Usually the balance is very simple: pressure is effective on one side or end of a ball, poppet, or spool and is opposed by a spring [19].

Flow control valves:-Primary function: controlling rate of flow, velocity control of cylinders, and speed control of hydraulic motors. Such a device can be used as a flow meter by measuring a pressure drop across the orifice. This control of the fluid path is accomplished by orifice, needle valve [19]

Directional-Control Valves: - Directional-control valves also control flow direction. However, they vary considerably in physical characteristics and operation. The valves may be a

- Poppet type, in which a piston or ball moves on and off a seat.
- Rotary-spool type, in which a spool rotates about its axis.
- Sliding-spool type, in which a spool slides axially in a bore. In this type, a spool is often classified according to the flow conditions created when it is in the normal or neutral position. This control of the fluid path is accomplished by check valves, shuttle valves, two-way, three-way, four-way and five-way valves [19].

3.4. Clarifying Design Objectives

The need of variable speed vibrating plate compactor is basic for the current growing of construction industries and construction enterprise in our country. The main objective of this project is to design variable speed vibrating plate compactor locally in order to satisfy the demand of the market and avoid the implementation of different compaction machine. By doing this we can save foreign currency which is put to bring compaction machine and it also create work opportunity to the country.

Product measurement

- Efficient
- Safety
- Machine must be safe from failure
- Low risk of injury to operators
- Low power in put (man power)
- Environmental friendly
- Ease to operate
- Can be handled by one person
- Low cost of manufacturing
- Ease of assembly
- Ease of manufacture
- required weight

- Low vibration to operator
- High rigidity

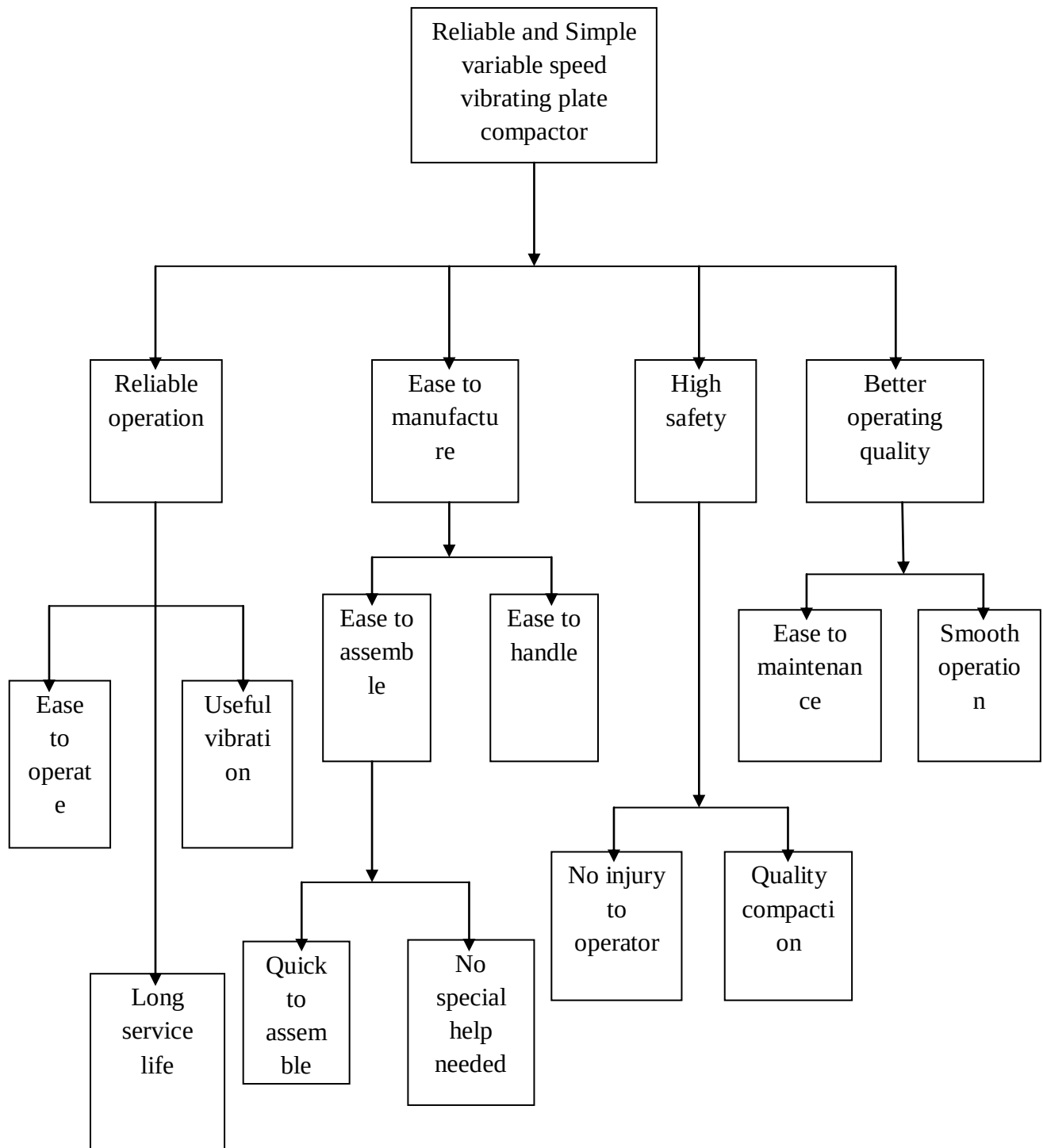


Figure 3.2:- product measurement criteria tree for variable speed vibrating plate compactor

The measurement criteria tree shows how variable speed vibrating plate compactor should fulfill the entire above listed criterion. Once the variable speed vibrating plate compactor is designed it should be checked whether it under go this measurement criterion.

Requirements List (Design Specification)

Setting up the requirements list based on subsystems (functions or assemblies)

W:- Wishes also implies us the part has no effect on variable speed vibrating plate soil compactor if it is left.

D:- Demand implies us the component is critical to variable speed vibrating plate soil compactor

Table 3.1:- requirement list

AAIT		Requirements list For Tip-up variable speed vibrating plate compactor	Issued on July 24/2014
Changes	W D	Requirements	Responsible
	D	Prime mover ➤ Electric motor ➤ Diesel engine	Primary source of power
	D	Vibrating plate ➤ 850x600x15mm	Contact surface between soil and machine
	D	Rotary shaft	Rotating ERM
	D	Pulley	Power transmission b/n shaft and prime mover
	D	Belt	Connecting prime mover pulley with rotating shaft
	D	Eccentric rotating mass	Source of forced vibration

	D	Bearings	Act as support and smooth rotation
	D	Bearing housing	Mounting bearings
	D	Pump	Source of hydraulic power
	D	Valve	Switch to vary radii of ERM& adjust the vibrating plate figure from flat to sheep's foot surface
	D	Piston	Actuator to vary the radii of ERM& vibrating plate from flat to sheep's foot surface.
	W	Case cover	Cover the internal figure and part of the machine system
	D	Bolts and Nuts	Fastening
	D	Washers	
	D	Spacers	
	D	Retaining rings	Used to lock bearing
		Replaces issue of	

Note:-The requirement lists implies the critical components of the machine and parts which don't have basic impact on the machine if they are omitted from machine.

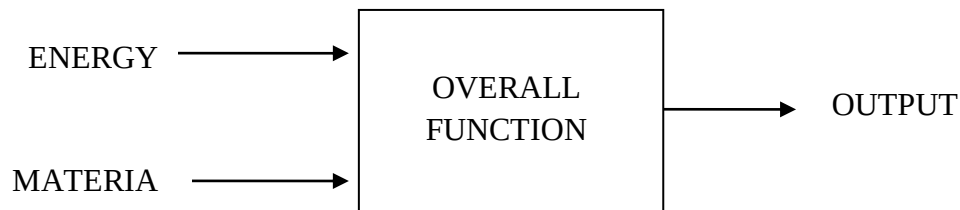
CHAPTER FOUR

4. CONCEPTUAL DESIGN

Conceptual design is that part of the design process in which, by identification of the essential problem through abstraction, by establishment of function structure and by the search for appropriate solution principles and their combination, the basic solution path is laid down through elaboration of solution concepts [20].

4.1 Establish functional structure of variable speed vibrating plate compactor

The conversions of energy, material and signals, Task or function are described on the basis of input and output. The overall function of variable speed vibrating plate compactor:-



The overall function is usually possible to link sub-function in various ways. Meaningful and compatible combination of sub-function into an overall function produces so called function structure. Function structure is useful to distinguish between main and auxiliary function. Main functions are those sub-functions that serve the overall function directly and Auxiliary functions serve the overall function indirectly [20].

Variable speed vibrating plate compactor has main function that serve the overall function of the process and also it does have Auxiliary function in the process.

The main function is manufacturing components and it shows how the materials flow for each component production. The auxiliary function is the design and sizing component.

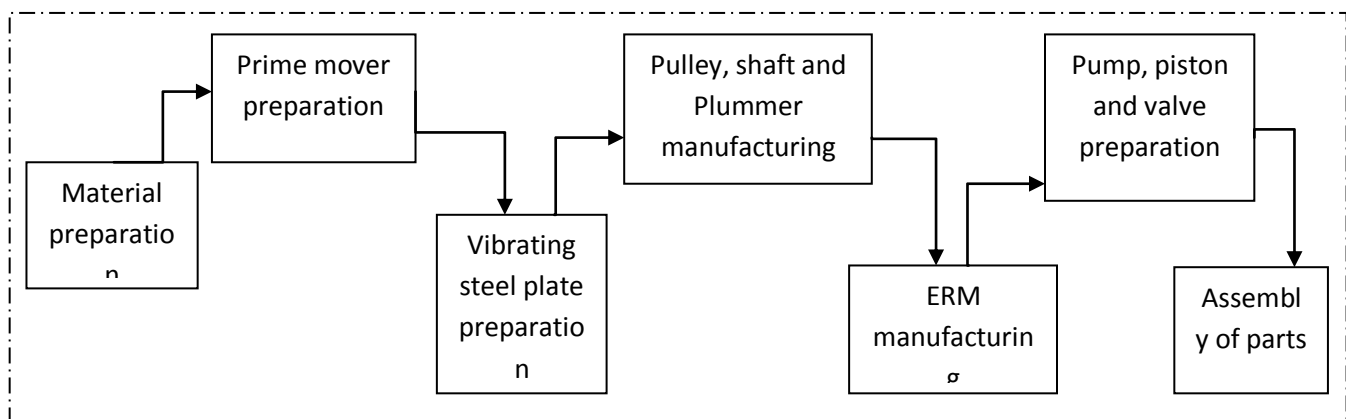


Figure 4.1:- Flow of main function of development of variable speed vibrating plate compactor

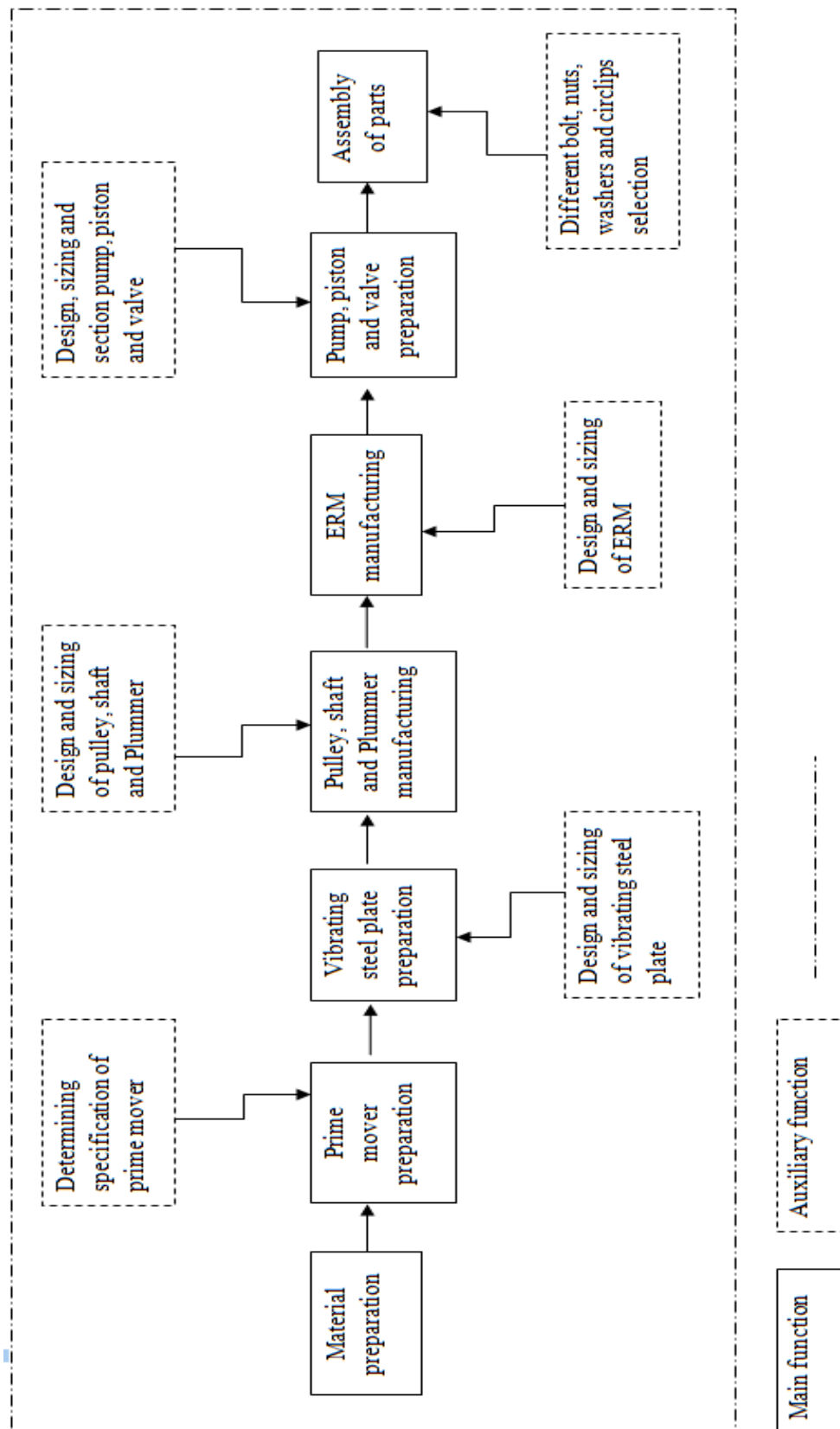


Figure 4.2:- overall function of development of variable speed vibrating plate

4.2 Solution Principles

To accomplish the required function, the variable speed vibrating plate compactor should have power source, vibrating plate, rotating shaft, eccentric rotating mass, For each of its components, we suggest solution principles in the following sequence by using weighted objective method to come up with solution:-

4.2.1 Power source

Power source is the part of variable speed vibrating plate compactor where it gets primary source of power. Prime mover is either diesel engine or electric motor. In this solution principle it going to be evaluated to select the appropriate method of prime mover which best suited based on the criteria as shown in the following table:-

Table 4.1:- computing power source type

Power source type						
Design criteria		Weight (%)	Diesel engine		Electric motor	
			S	U	S	U
Efficiency		0.15	8	1.2	8	1.2
Weight		0.10	9	0.9	6	0.6
Portability		0.10	9	0.9	6	0.6
Easily Availability		0.15	5	0.75	10	1.5
Assembly	Ease	0.10	5	0.5	9	0.9
	Time	0.10	4	0.4	8	0.8
Cost		0.15	5	0.75	9	1.35
Maintainability		0.10	5	0.5	9	0.9
Aesthetics		0.05	9	0.45	7	0.35
Total utility		1		6.35		8.2
Rank			2		1	
Key :						
W=percentage weight of each criterion (from 100%)						
S= score of quality of each design(out of 10)						
U=utility(weighted score of each design)=W*S						

Table 4.2:- computing electric motor type

Electric motor type					
Design criteria	Weight	Induction motor		Synchronous motor	
		S	U	S	U
Efficiency	0.45	8	3.6	9	4.05
Slip	0.3	6	1.8	9	2.7
Cost	0.25	7.5	1.875	8.5	2.125
Total utility	1		7.275		8.875
Rank		2		1	

Note: - prime mover which is electric motor of synchronous motor type, is selected from computed above table.

4.2.2 Vibrating plate

Vibrating plate is plate which is a key part of vibrating plate compactor. It is contact surface between the compactor machine and soil. In order to give the compaction of the soil it should have the property of soil compaction criterion. And hence it should have the best material property and compaction criterion.

Table 4.3:- computing vibrating plate material type

Vibrating plate material type									
Design criteria	Weight	Steel plate		Plastic polymer plate		Concrete plate		Composite material plate	
		S	U	S	U	S	U	S	U
Efficiency	0.3	8	2.4	6	1.8	3	0.9	5	1.5
weight	0.2	10	2	5	1	8	1.6	4	0.8
Cost	0.2	5	1	6	1.2	8	1.6	7	1.4
Durability	0.15	10	1.5	7	1.05	3	0.45	5	0.75
Impact resistance	0.15	10	1.5	7	1.05	3	0.45	5	0.75

Total utility	1	8.4	6.1	5	5.2
Rank	1	2	4	3	

Table 4.4:- computing vibrating plate type surface property

Vibrating plate surface property						
Design criteria		Weight (%)	Flat plate		Adjustable sheep's foot plate	
			S	U	S	U
Efficiency		0.4	6	3.2	9.5	3.8
Assembly	Ease	0.15	9	1.35	7	1.05
	Time	0.1	8	0.8	6	0.6
Cost		0.15	8.5	1.275	7	1.05
variability		0.2	4	0.8	9	1.8
Total utility		1		7.425		8.3
Rank			2		1	

Table 4.5:- computing vibrating plate type control type

Vibrating plate					
Design criteria	Weight (%)	Manual controlled adjustable sheep's foot plate		Hydraulically controlled adjustable sheep's foot plate	
		S	U	S	U
Efficiency	0.45	8	3.6	9	4.05
Time	0.3	6	1.8	9	2.7
Cost	0.25	8.5	2.125	6.5	1.625
Total utility	1		7.525		8.375
Rank		2		1	
Key :					
W=percentage weight of each criterion (from 100%)					
S= score of quality of each design(out of 10)					
U=utility(weighted score of each design)=W*S					

Note: - vibrating plate which is hydraulically adjustable sheep's foot steel plate, is selected in order to give static, kneading and vibratory impact to soil to be compacted.

4.2.3 Rotating shaft

A shaft is the component of a mechanical device that transmits rotational motion and power. It is integral to any mechanical system in which power is transmitted from a prime mover to other rotating parts of the system. Also it is spar on which eccentric vibrating mass is attached so that it should withstand the bending and shear stress exerted by centrifugal force due to rotation of ERM. In this solution principle it going to be evaluated to select the appropriate type of rotating shaft which best suited based on the criteria as shown in the following table:-

Table 4.6:- computing rotating shaft material type

Rotating shaft material									
Design criteria	Weight	Steel shaft		Plastic polymer shaft		Carbon fiber shaft		Composite material shaft	
		S	U	S	U	S	U	S	U
Efficiency	0.3	8	2.4	6	1.8	7	2.1	5	1.5
weight	0.2	10	2	5	1	8	1.6	4	0.8
Cost	0.2	5	1	6	1.2	8	1.6	7	1.4
Durability	0.15	10	1.5	7	1.05	7	1.05	5	0.75
fatigue resistance	0.15	10	1.5	7	1.05	7	1.05	5	0.75
Total utility	1		8.4		6.1		7.4		5.2
Rank		1		3			2		4
Key :									
W=percentage weight of each criterion (from 100%)									
S= score of quality of each design(out of 10)									
U=utility(weighted score of each design)=W*S									

Table 4.7:- computing rotating shaft type

Rotating shaft						
Design criteria		Weight	Solid shaft		Hollow shaft	
			S	U	S	U
Material		0.1	8	0.8	8	0.8
Variability		0.1	7	0.7	8	0.8
Stiffness		0.3	9	2.7	8	2.4
Manufacturing	Ease	0.1	7	0.7	9	0.9
	Time	0.1	9	0.9	8.5	0.85
	Cost	0.1	7	0.7	8	0.8
Assembly	Ease	0.1	8	0.8	8	0.8
	Time	0.1	8.5	0.85	7.5	0.75
Total utility		1		8.15		8.1
Rank			1		2	
Key :						
W=percentage weight of each criterion (from 100%)						
S= score of quality of each design(out of 10)						
U=utility(weighted score of each design)=W*S						

Note: - Rotating shaft which is made of steel and solid shaft is selected to transmit power and carry eccentric rotating mass which is means vibration.

4.2.4 Eccentric rotating mass

Eccentric rotating mass is mass in which the center of mass is eccentrically located at eccentricity “e” so that when it is subjected to rotation it produces vibration in which the amplitude and energy is directly proportional to the eccentricity e and the mass of ERM. The centripetal force of the offset mass is asymmetric, resulting in a net centrifugal force, and this causes a displacement of the machine. With a high number of revolutions per minute, the machine is constantly being displaced and moved by these asymmetric forces. It is this repeated displacement that is perceived as a vibration. In this solution principle it going to be evaluated to select the appropriate type of ERM which best suited based on the criteria as shown in the following table:-

Table 4.8:- computing ERM material type

Eccentric rotating mass material type									
Design criteria	Weight	Steel ERM		Plastic polymer ERM		Carbon fiber ERM		Composite material ERM	
		S	U	S	U	S	U	S	U
strength	0.3	8	2.4	6	1.8	7	2.1	5	1.5
density	0.2	9	2	7	1.4	6	1.2	4	0.8
Cost	0.2	5	1	6	1.2	9	1.8	7	1.4
Durability	0.15	9	1.35	7	1.05	6	0.9	5	0.75
Manufacturability	0.15	9	1.35	7	1.05	7	1.05	7	1.05
Total utility	1		8.1		6.5		7.05		5.5
Rank		1		3		2		4	

Table 4.9:- computing ERM type

Eccentric rotating mass type						
Design criteria		Weight	Fixed ERM		Telescopic ERM	
			S	U	S	U
Variability		0.25	1	0.25	10	2.5
Stiffness		0.25	9	2.7	8	2.4
Manufacturability	Ease	0.1	9	0.9	7	0.7
	Time	0.1	9	0.9	7	0.7
	Cost	0.1	8	0.8	6	0.6
Assembly	Ease	0.1	8	0.8	7	0.7
	Time	0.1	9	0.9	7.5	0.75
Total utility		1		7.25		8.35
Rank			2		1	
Key :						
W=percentage weight of each criterion (from 100%)						
S= score of quality of each design(out of 10)						
U=utility(weighted score of each design)=W*S						

Table 4.10:- computing ERM type control method

Eccentric rotating mass					
Design criteria	Weight (%)	Manually controlled ERM		Hydraulically controlled ERM	
		S	U	S	U
Efficiency	0.35	7	2.45	9	3.15
Ease	0.2	4	0.8	9	1.8
Time	0.25	5	1.25	9	2.25
Cost	0.2	8.5	1.7	5	1
Total utility	1		6.2		8.2
Rank		2		1	

Table 4.11:- computing ERM retraction type

Eccentric rotating mass retraction type					
Design criteria	Weight (%)	Double acting ERM		Single acting ERM	
		S	U	S	U
Efficiency	0.35	8	2.8	7	2.45
manufacturability	0.2	7	1.4	7	1.4
Controllability	0.25	5	1.25	8	2.25
Cost	0.2	7	1.4	7	1.4
Total utility	1		6.85		7.5
Rank		2		1	

Key :

W=percentage weight of each criterion (from 100%)

S= score of quality of each design(out of 10)

U=utility(weighted score of each design)=W*S

Table 4.12:- computing number of ERM

Number of Eccentric rotating mass					
Design criteria	Weight (%)	Double ERM		Single ERM	
		S	U	S	U
Efficiency	0.3	8	2.4	6	1.8
Self drivability	0.2	9	1.8	6	1.2
Controllability	0.1	7	0.7	8	0.8
Cost	0.2	6	1.2	7	1.4
Linearity	0.2	8	1.6	6	1.2
Total utility	1		7.7		6.4
Rank		1		2	

Key :

W=percentage weight of each criterion (from 100%)

S= score of quality of each design(out of 10)

U=utility(weighted score of each design)=W*S

Note: -The centrifugal force is the main factor of vibration. The centrifugal force exerted by eccentric rotating mass is directly proportional to eccentricity radius, square of rotational speed and mass of ERM. In order to overcome different ranges centrifugal forces, it can be achieved by varying the eccentricity of ERM. Hence Eccentric rotating mass which is single acting, hydraulically controlled double steel ERM is selected to give variable amplitude, linear force and energy of vibration.

4.2.5 Belts

Belts may be used as a source of motion, to transmit power efficiently, or to track relative movement. Belts are looped over pulleys. In variable speed vibrating plate compactor, belts are used to transmit power and torque. Since the diameter of driver pulley from the prime mover is less than the diameter of driven pulley at the rotating shaft it multiply the torque and reduces the speed. It could be of different type as mentioned above but In this solution principle it going to be

evaluated to select the appropriate type of belts which best suited based on the criteria as shown in the following table:

Table 4.13:- computing belt type

Belts type										
Design criteria		Weight (%)	Timing belt		flat belt		Multi-V groove belt		V-belt	
			S	U	S	U	S	U	S	U
Efficiency		0.2	7	1.4	6	1.2	8	1.6	7	1.4
Slippage		0.15	9	1.35	6	0.9	8	1.2	7	1.05
Durability		0.15	7	1.05	6	0.9	8	1.2	7	1.05
Cost		0.2	6	1.2	8	1.6	7	1.4	7	1.4
Assembly	Ease	0.05	7	0.35	8	0.4	8	0.4	8	0.4
	Time	0.05	7	0.35	8	0.4	8	0.4	8	0.4
Stretch ability		0.1	8	0.8	6	0.6	9	0.9	9	0.9
Noise isolation		0.1	7	0.7	9	0.9	8	0.8	8	0.8
Total utility		1		7.2		6.9		6.4		7.9
Rank			2		3		4		1	
Key :										
W=percentage weight of each criterion (from 100%)										
S= score of quality of each design(out of 10)										
U=utility(weighted score of each design)=W*S										

Note:- belts which is selected based on the criteria it should meet, V grooved belt is selected. Minimal slippage, less noise, tension resistance and efficient power and torque transmission is the basic characteristics of V grooved belt.

4.2.6 Pulley

A pulley is a wheel on an axle that is designed to support movement and change of direction of a belt along its circumference. Since it transmits power and torque it is going to use two pulleys one at the electric motor and one at rotating shaft. It also multiplies torque or speed of power source

by using different diametrical ratio. It could be of different type. In this solution principle it going to be evaluated to select the appropriate type of pulley which best suited based on the criteria as shown in the following table:-

Table 4.14 computing pulley material type

Pulley material type									
Design criteria	Weight	Steel pulley		Plastic polymer pulley		Aluminum pulley		carbon fiber pulley	
		S	U	S	U	S	U	S	U
strength	0.25	8	2	6	1.5	7	1.75	5	1.25
density	0.2	8	1.6	6	1.2	7	1.4	5	1
Cost	0.2	6	1.2	6	1.2	9	1.8	7	1.4
Durability	0.1	9	0.9	7	0.7	8	0.8	6	0.6
Manufacturability	0.1	7	0.7	8	0.8	9	0.9	7	0.7
Shear resistant	0.15	9	1.35	7	1.05	6	0.9	7	1.05
Total utility	1		7.75		6.45		7.55		6
Rank		1		3		2		4	
Key :									
W=percentage weight of each criterion (from 100%)									
S= score of quality of each design(out of 10)									
U=utility(weighted score of each design)=W*S									

Note: - pulley which is made of steel V-grooved solid pulley is selected from the above computed solution principle.

4.3 Summery Result Of Solution Principle

1. Prime mover which is electric motor of synchronous motor type, is selected based on the design criteria of efficiency, slip resistance, cost and availability computed from the above table.
2. Vibrating plate which is hydraulically controlled adjustable sheep's foot steel plate is selected in order to give static, kneading, vibratory and impact to soil to be compacted based on the design criteria of efficiency, cost, durability, impact resistance and variable surface characteristics computed from the above table.
3. Rotating shaft which is made of steel and solid shaft is selected to transmit power and carry eccentric rotating mass which is means vibration based on the design criteria of efficiency, fatigue resistance and manufacturability computed from the above table.
4. In order to overcome different ranges centrifugal forces, it can be achieved by varying the eccentricity of ERM. Hence Eccentric rotating mass which is single acting hydraulically controlled double steel ERM is selected to give variable amplitude, linear force and energy of vibration based on the design criteria of strength, variability, efficiency and cost computed from the above table..
5. Belts which is selected based on the criteria it should meet, V grooved belt is selected. Minimal slippage, less noise, tension resistance and efficient power and torque transmission is the basic characteristics of V grooved belt.
6. Pulley which is made of steel V-grooved solid pulley is selected based on the main design criteria of strength, durability, shear resistance and belt type computed from the above table.

5. DETAIL DESIGN

5.1 Soil compaction energy

Due to the development of much heavier earth moving and vibratory roller compaction equipment, densities in the field are reaching levels that are not attainable in the laboratory. Higher compaction efforts, routinely seen in the field, not only result in higher unit weights but also lower optimum moisture contents than those found by the Modified Proctor test. The data illustrated in Figure show that this result is experienced in the field due to the higher compaction energies produced by modern heavy compaction equipment. Due to this phenomenon, the optimum moisture content (OMC) obtained in the laboratory is often higher than that in the field compaction. Consequently, in the field compaction the maximum density compacted using the laboratory OMC will be lower than that obtained using the field OMC (point A versus point B in Figure) [21]. In addition, the impact compaction method does not work well with the pure sandy soil.

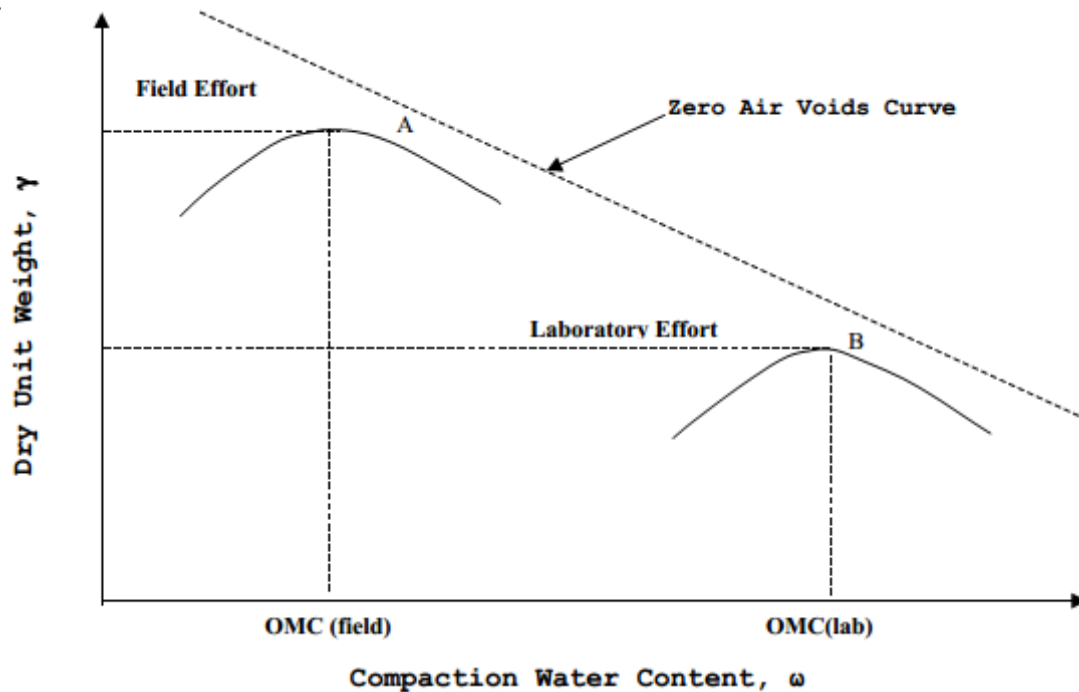


Figure 5.1: - effect of compaction effort (compaction energy) on compaction curve [21]

Accordingly, in order to determine the compaction energy (compaction effort) in the laboratory taste in present state of the art, the above relationships are established in the laboratory. But this approach has serious intrinsic limitations, because field compaction is achieved by different modes and at different energy levels than in the laboratory and more variability exists in all

variables in the field. In order to simulate the different field compaction methods, a number of techniques have been developed to compact soil in the laboratory – most of the tests fall into four types [21]:

- Impact compaction tests in which a standard weight is repeatedly dropped on the soil sample for a prescribed number of blows. The weight is adjusted to achieve the desired compaction effort.
- Static compaction tests in which a uniform pressure is applied to the soil and maintained long enough for the soil to compact under the pressure.
- Kneading compaction tests in which a small “foot” is loaded, then unloaded, at various locations on the surface of the sample being compacted; the soil is effectively kneaded with this procedure.
- Vibratory compaction tests in which the soil is vibrated as it is compacted, which is particularly effective in compacting cohesionless soil such as sand and gravel.

Ideally, the laboratory compaction tests should simulate the characteristics of soil compaction used in field procedures. The direct consequence of soil compaction is densification of the fill. The quality of compacted material is generally specified in terms of dry unit weight, which is usually expressed as a percentage of the maximum dry unit weight achieved in a specific laboratory compaction test. Construction specifications based on this principle are known as ‘end-result’ specifications. Many laboratory soil compaction procedures are available. Most of these procedures utilize either impact compaction or vibratory compaction. These include the tests based on the Proctor hammer and those using vibratory compaction [21].

5.2. Impact Compaction

The most common impact compaction tests are the standard and modified Proctor tests, AASHTO T 99 and T 180, respectively. Developed in the 1930s and 1940s, these tests were the first to be standardized and as a result a broad base of data exists for comparison. One downfall of the Proctor tests is that impact compaction has proved to be relatively ineffective for the compaction of noncohesive soils because the material displaces under the hammer, and consequently low-density values are obtained. Despite this fact majority of construction use these test procedures in their construction specifications [21].

5.2.1 Standard Proctor compaction procedure

This test procedure covers laboratory compaction procedures used to determine the relationship between water content and dry unit weight of soils compacted in a 4 or 6 in. diameter mold with a 5.5 lb. hammer dropped from a height of 12 in. (Figure 2.1) (AASHTO, T99), producing a compaction effort of 12,400 ft-lb/ft³. A soil at a selected water content is placed in three layers into a mold of the given dimensions, with each layer compacted by 25 blows of the hammer. The resulting dry unit weight is then determined. This procedure is repeated for a sufficient number of water contents to establish a relationship between the dry unit weight and the water content of the soil. This test procedure applies only to soils that have 30% or less by weight of particles retained on the 3/4 in. sieve. Generally a well-defined maximum dry unit weight will be produced for non-free draining soils. If this test method is used on free draining soils the maximum unit weight may not be well defined and can be less than that obtained using the ASTM test procedure D 4253 (vibratory compaction) [21].

5.2.2 Modified Proctor compaction procedure

The Modified Proctor compaction procedure is a test method that covers laboratory compaction procedures used to determine the relationship between water content and dry unit weight of soils compacted in a 4 or 6 in. diameter mold with a 10 lb. hammer dropped from a height of 18 in. producing a compaction effort of 56,000 ft-lb/ft³. Five layers of soil at a selected water content are placed into a mold of the given dimensions, with each layer compacted by 25 blows of the hammer. The resulting dry unit weight is then determined. This procedure is repeated for a sufficient amount of water contents to establish a relationship between the dry unit weight and the water contents of the soil. This test procedure applies only to soils that have 30% or less by weight of particles retained on the 3/4 in sieve. Generally a well-defined maximum dry unit weight will be produced for non-free draining soils. As with the Standard Proctor test procedure, if this test method is used on free draining soils the maximum unit weight may not be well defined, and can be less than that obtained using the ASTM test procedure D 4253 (vibratory compaction) [21].

To calculate the compaction force for both standard proctor test and modified proctor test:

Table 5.1 standard specification of Standard and modified proctor test measurement [21]

Standard proctor test measurement		
Volume of the mold	Hammer mass	Drop of hammer
1000cm ³	2.5kg	300mm
Modified proctor test measurement		
Volume of the mold	Hammer mass	Drop of hammer
1000cm ³	4.9kg	450mm

$$E = \frac{(\text{No. of blows per layer}) \times \left(\frac{\text{number}}{\text{of layers}} \right) \times \left(\frac{\text{weight of}}{\text{hammer}} \right) \times (\text{height of drop of hammer})}{\text{volume of mold}}$$

Substituting the above values in tables for each test for the 3 and 5 layers,

$$\left. \begin{array}{l} E=56250\text{kgm/m}^3 \text{ is for 3 layers} \\ E=93750\text{kgm/m}^3 \text{ is for 5 layers} \end{array} \right\} \text{Standard proctor test [21]}$$

and

$$\left. \begin{array}{l} E=165375\text{kgm/m}^3 \text{ for 3 layers} \\ E=275625\text{kgm/m}^3 \text{ for 5 layers} \end{array} \right\} \text{Modified proctor test [21]}$$

Respectively

Unit conversion

1J/m³=0.1kgm/m³ and hence the above compaction energy (compaction effort) for both test

$$\left. \begin{array}{l} E=5625\text{J/m}^3 \text{ is for 3 layers} \\ E=9375\text{J/m}^3 \text{ is for 5 layers} \end{array} \right\} \text{Standard proctor test}$$

and

$$\left. \begin{array}{l} E=16537.5\text{J/m}^3 \text{ for 3 layers} \\ E=27562.5\text{J/m}^3 \text{ for 5 layers} \end{array} \right\} \text{Modified proctor test}$$

Respectively

If the tested soil or the selected soil is not reach to the specified dry density in relation to optimum moisture content by compaction effort (compaction energy) in either standard proctor test or modified proctor test the geotechnical engineer would recommend to refill the area with proper

soil type. Whereas by varying the compaction effort or compaction energy to different soil type with optimum moisture content can achieve the required dry density.

5.3. Vibration analysis

In this section it is going to address and represent the free body diagram and vibration analysis of variable speed vibration plate compactor. Since the machine is vibrating in forward and upward two directions, the analysis is conducted in two degree of freedom.

Systems that require two independent coordinates to describe their motion are called two degree of freedom systems.

No. of degree of freedom = No. of masses of the system + No. of possible type of motion of each mass

If we give suitable initial excitation, the system vibrates at one of these natural frequencies.

During free vibration at one of the natural frequencies, the amplitudes of the two degrees of freedom (coordinates) are related in a specified manner and the configuration is called a normal mode, principle mode, or natural mode of vibration [18].

Thus a two degree of freedom system has two normal modes of vibration corresponding to two natural frequencies. If we give an arbitrary initial excitation to the system, the resulting free vibration will be a superposition of the two normal modes of vibration. However, if the system vibrates under the action of an external harmonic force, the resulting forced harmonic vibration takes place at the frequency of the applied force.

Hence the variable speed vibrating plate compactor gets its excitation from the centrifugal force exerted by eccentric rotating mass therefore the system vibrates under the action of the exciter harmonic force.

It is important to remember that in the case of the ERM model, the excitation input is not the voltage applied to the electric motor. Instead, it is the rotation of the mass around the central shaft. The mass movement can be modeled as a sinusoidal wave, shown below [18]:

$$A \sin (wt) \dots \dots \dots \text{eq 5.1}$$

Here the function $A \sin (wt)$ is the excitation input, and the frequency of this sine wave is the frequency at which the ERM vibrates. Therefore the frequency, however when modeling the ERM as a system, we will refer to the sinusoid as the input.

To analyze the behavior of the ERM, we will approximate the system to having one degree of freedom (DOF). This means that the vibration will only manifest itself in one direction, and it makes the maths a lot simpler.

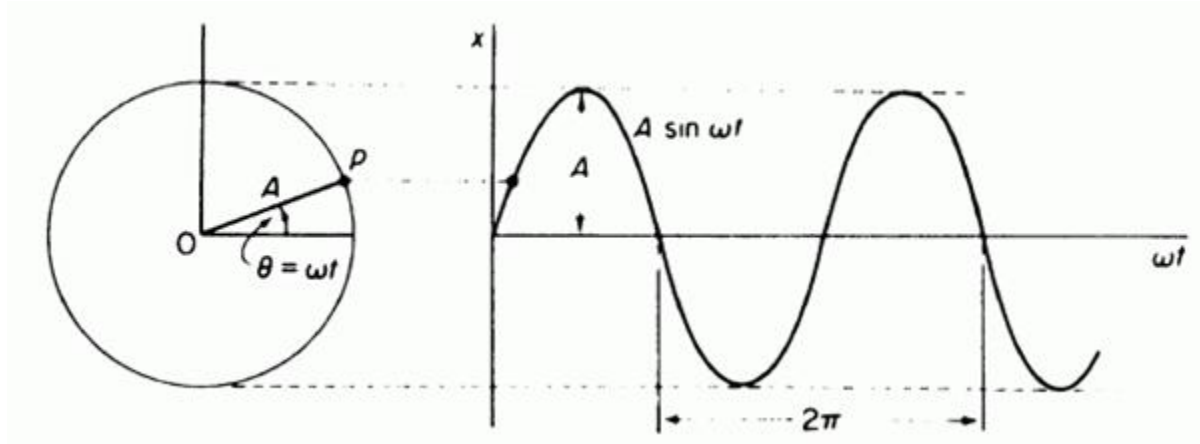


Figure 5.2: - ERM rotation position as a **sin** wave [5]

The centrifugal force exerted by ERM is

$$F_e = m e \omega^2 \dots \dots \dots \text{eq. 5.2}$$

Here 'm' is the mass of the ERM, ' ω ' is the angular speed of rotating shaft and 'e' is the eccentricity from center of rotating shaft to the mass center of ERM since the ERMs are two equal mass.

Therefore the force exertion modeling of ERM is also sinusoidal:-

$$F_o = F_e \sin \omega t \dots \dots \dots \text{eq. 5.3}$$

5.3.1 Modeling the variable speed vibrating plate compactor

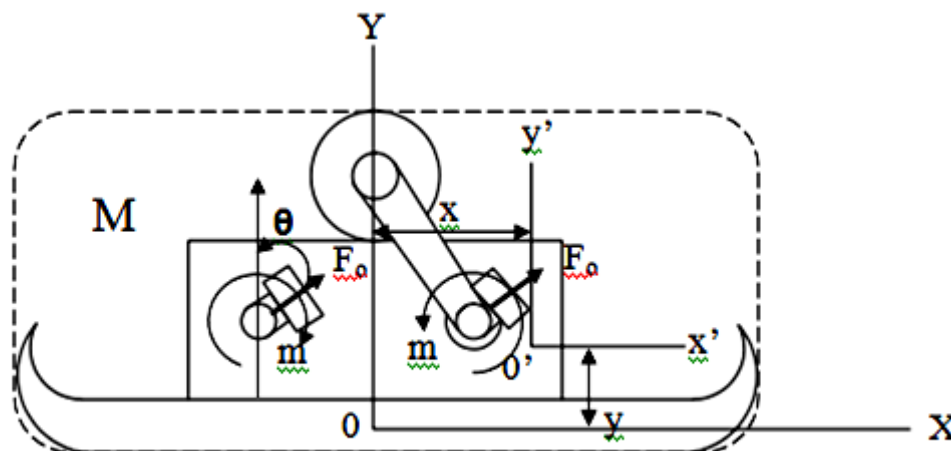


Figure 5.3:-free body diagram of vibrating plate compactor

θ - is the angel to which the two ERM meet from the vertical

M- is total mass of vibrating plate compactor

m- is mass of ERM

F_o- is the centrifugal force of each ERM

Since the two eccentric masses are equal and rotating in opposite direction, the linear upward and downward centrifugal force exerted due to rotation is achieved by setting the two ERM to meet at certain fixed angle from the vertical axis. The rotating ERM turning in opposite directions are phase so that the centrifugal forces add in the desired direction and cancel in other directions. The effective generated force from the two rotating ERM is midway between the two axes of rotation and is normal to a line connecting the two it. The forward movement of the variable speed vibrating plate compactor is caused by the horizontal component of the two centrifugal force of ERM. The upward force or compaction force is caused by the vertical component of the centrifugal force exerted by the ERM. Due to constant rotation of the ERM the compaction force and the forward movement is harmonic or sinusoidal vibration.

The resultant centrifugal force exerted by the two rotating eccentric masses is the direct summation of the centrifugal force exerted by each ERM.

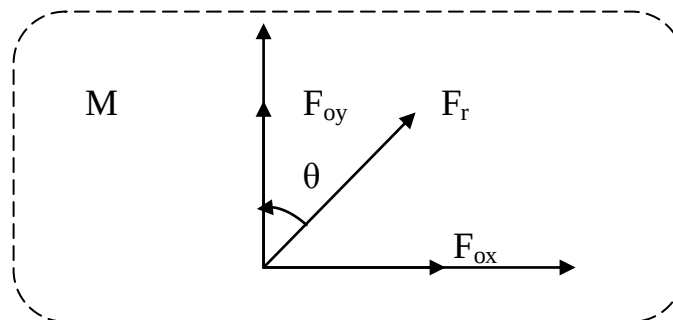


Figure 5.4:- force vector of ERMs

$$Fr = \sum_{k=0}^n Fo \dots \dots \dots \text{eq. 5.4}$$

$$Fo1 + Fo2 = Fr$$

Since the two ERM are equal mass and equal eccentricities from the center of the shaft:-

$$Fr = 2Fe \sin \omega t$$

Therefore

The vertical component force or compaction force is

$$Foy = 2Fe \cos \theta \sin \omega t \dots \dots \dots \text{eq. 5.5}$$

The forward force is

$$F_{ox} = 2Fe \sin \theta \sin \omega t \dots \dots \dots eq. 5.6$$

The forces generated by the rotating ERM are transmitted directly to the vibrating plate without dependence upon a reactionary force against a heavy base or rigid ground connection.

5.3.2. Force analysis

Here we used concept of forced vibration with mass excitation. As mentioned above the eccentric rotating mass is a common source of forced vibrations. The eccentricity in rotating machines is measured in terms of mass 'm' rotating with its centre of gravity at a distance 'e' from the axis of rotation. The centrifugal force due to eccentric mass 'm' acts as harmonic excitation force.

Assumption:-It is assumed that the system is constrained to move vertically and horizontally having two degrees of freedom. Let, Total vertical and horizontal displacement of mass of machine excluding eccentric rotating mass $(M - m) = y$ and $(M - m) = x$

Total vertical and horizontal displacement of eccentric rotating mass 'm' = $(y + e \sin \omega t) \omega t$ and $= (x + e \sin \omega t)$ respectively [10]

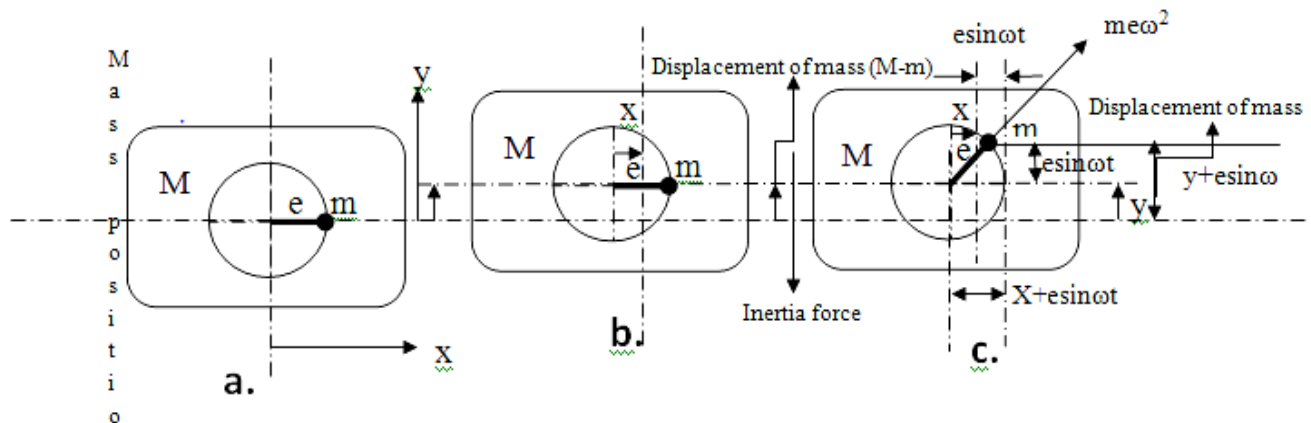


Figure 5.5: - Eccentric rotation mass [10]

- Equilibrium position
- Displaced position with angle of rotation of eccentric rotating mass equal to zero.
- Displaced position with angle of rotation of eccentric rotating mass equal to ωt .

Inertia force in the horizontal and vertical direction due to mass of compactor excluding eccentric rotating mass

$$(M-m) = (M - m) \ddot{x}$$

$$(M-m) = (M - m) \ddot{y} \quad \text{respectively}$$

Inertia force due to eccentric rotating mass 'm' in both directions

$$\begin{aligned}
 &= m \frac{d^2}{dt^2} (x + e \sin \omega t \sin \theta) \\
 &= m \ddot{x} - m e \omega^2 \sin \omega t \sin \theta \quad \& \\
 &= m \frac{d^2}{dt^2} (y + e \sin \omega t \cos \theta) \\
 &= m \ddot{y} - m e \omega^2 \sin \omega t \cos \theta \quad \text{respectively}
 \end{aligned}$$

The gravitational force is constant and act downward all the time which is

$$= Mg$$

Since the variable speed vibrating plate compactor is not subjected to any spring or damper or the plate is not fixed other than soil spring and damping constant it is considered in the response of the soil due to the machine vibration.

$c\dot{x}$ or $c\dot{y}$ and Kx or Ky is negligible

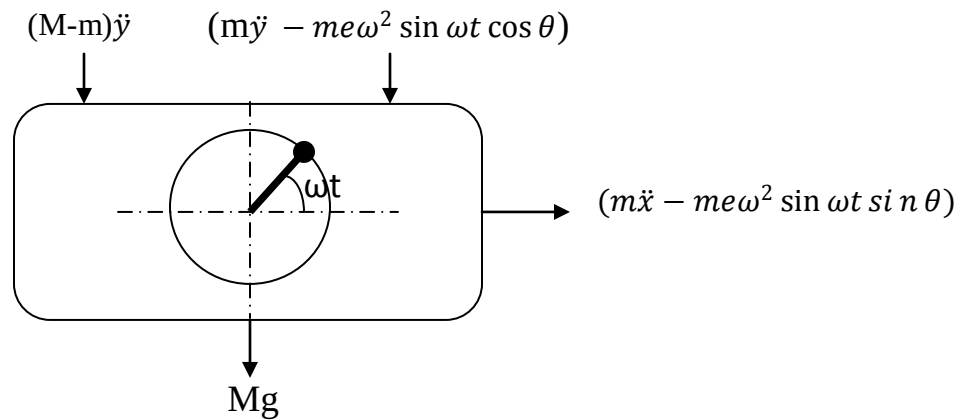


Figure 5.6: - free body diagram showing force distribution [10]

The total summation of inertia and external force applied to the machine is zero in vertical and horizontal direction neglecting soil friction.

$$\begin{aligned}
 \sum [\text{internal force} + \text{external force}] &= 0 \\
 (M - m)\ddot{y} + m\ddot{y} - m e \omega^2 \sin \omega t \cos \theta + Mg &= 0 \\
 M\ddot{y} &= m e \omega^2 \sin \omega t \cos \theta - Mg \dots \dots \dots \text{eq. 5.7}
 \end{aligned}$$

And

$$\begin{aligned}
 \sum [\text{internal force} + \text{external force}] &= 0 \\
 (M - m)\ddot{x} + m\ddot{x} - m e \omega^2 \sin \omega t \sin \theta &= 0 \\
 M\ddot{x} &= m e \omega^2 \sin \omega t \sin \theta \dots \dots \dots \text{eq. 5.8}
 \end{aligned}$$

The general matrix

$$\begin{bmatrix} M & 0 \\ 0 & M \end{bmatrix} * \begin{pmatrix} \ddot{x} \\ \ddot{y} \end{pmatrix} = \begin{pmatrix} me\omega^2 \sin \omega t \sin \theta \\ me\omega^2 \sin \omega t \cos \theta - Mg \end{pmatrix} \dots \dots \dots eq. 5.9$$

Since the compaction effort exerted from vibrating plate compactor is fixed per blows (cycle) per area and in order to increase the compaction effort per area of plate can be achieved by moderately fix the forward speed of vibrating plate compactor. This is because if the forward speed of compactor is kept minimum value, the number of blows per area increase and it directly multiply the compaction effort. Therefore, the slower the speed of travel, the more vibrations at a given point and lesser number of pass required to attain a given density. From the vibrating plate compactor manufacturer's the forward speed of vibrating plate compactors are standardized as shown below:-

Table 5.2:- standard forwarding speed of compactor machine [5]

Name of compactor machine	Forward speed(meter per minute)
Vibrating plate compactor	18 and 21
Rammer	10
Reversible vibrating plate compactor	25

From the standard forward speed of vibratory plate compactor it is selected that our variable speed vibrating plate compactor has 21m/min forward or travel speed

$$\dot{x} = 21m/min \quad \text{Or} \quad \dot{x} = 0.35m/sec$$

And hence the newly designed variable speed vibrating plate compactor should exert compaction effort per blow which is exactly equal to the laboratory modified proctor test's energy. Since the speed and amplitude is adjustable, the design should be based on the maximum compaction energy exerted by variable speed vibrating plate compactor such as:-

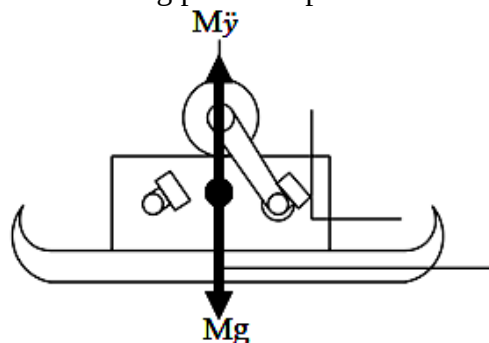


Figure 5.7:- the vertical movement of vibrating plate compactor

$$M\ddot{y} + Mg = 27562.5N/m^2 \dots \dots \dots eq. 5.10$$

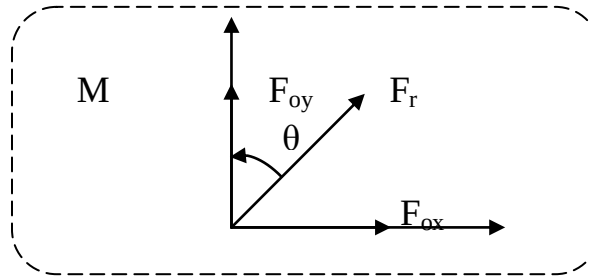
By taking the assumption that the overall weight of variable speed vibrating plate compactor is $M=120\text{kg}$ at max and the plate dimension is $0.6 \times 0.85\text{m}^2$. Substituting the value of the mass in the above equation and multiplying with the area of the plate and we get the vertical acceleration is:

$$\ddot{y} = 107.32\text{m/s}^2 \text{ Or } \ddot{y} = 10.93$$

g -is the gravitational acceleration

Therefore $\ddot{y}$ is the vertical acceleration of the compactor machine.

The angel to which the two ERM meet from the vertical is selected in order to create the travel or forward force.



$$F_{oy} = M\ddot{y} - Mg$$

$$F_{oy} = 120\text{kg} * 107.32\text{m/s}^2 - 120\text{kg} * 9.821\text{m/s}^2$$

$$F_{oy} = 11700\text{N}$$

Therefore in order to determine the forward force

$$\frac{F_{ox}}{F_{oy}} = \tan \theta$$

$$F_{ox} = F_{oy} * \tan \theta$$

Therefore the angle θ is selected to be 45°

$$F_{ox} = 11700\text{N} * \tan 45$$

$$F_{ox} = 11700\text{N}$$

Whereas the drag force or forward force is the resultant of horizontal component force since the variable speed vibrating plate compactor jump forward sinusoidal the friction between the vibrating plate and soil is negligible.

$$F_{ox} = M\ddot{x}$$

$$M\ddot{x} = 11700\text{N}$$

In order to determine the horizontal acceleration of variable speed vibrating plate compactor divides the forwarding force with mass of machine

$$\ddot{x} = 97.5\text{m/s}^2 \quad \text{Or} \quad \ddot{x} = 9.93g$$

However to determine the angular speed of vibrating plate it is require to recall the forward acceleration and forward speed movement derivation:-

$$M\ddot{x} = me\omega^2 \sin \omega t \sin \theta$$

$$M\dot{x} = -me\omega \cos \omega t \sin \theta$$

By taking the ratio of the above two equations

$$\frac{\ddot{x}}{\dot{x}} = \omega \tan \omega t$$

The graph of the acceleration and velocity of forward movement or travel speed is over lap at

$$\omega t = \frac{\pi}{4}$$

Therefore the angular speed of the vibrating plate is

$$\omega = \frac{97.5m/s^2}{0.35m/s}$$

$$\omega = 278.57rad/sec \quad \text{Or} \quad \omega = 2661.5RPM$$

To determine the resultant force exerted by the ERM

$$Fr = \sum_{k=0}^n F_o$$

$$Fr = F_{oxi} + F_{oyj} \quad \text{Substituting the corresponding values}$$

$$Fr = 16546.3N$$

To determine eccentric moment of the ERM again it requires recalling the above equation which represents the resultant force exerted by ERMs

$$Fr = 2me\omega^2 \sin \omega t$$

Since the resultant force of the machine is maximum when $\omega t = \frac{\pi}{2}$

$$me = \frac{Fr}{2\omega^2} \dots \dots \dots eq. 5.11$$

Substituting the values to the above equation we get

$$me = 0.11kgm$$

5.3.3 Vibration response of machine to soil

In order to analyze the vibration response of newly designed variable speed vibrating plate compactor first we should consider the spring constant and damping coefficient of the soil. Since the spring constant and damping ratios are different to vertical and horizontal vibration it should

be analyzed in two degree of freedom vibration analysis. To come up with the solution such as values of spring constant and damping coefficient of the soil it requires to consult the geotechnical engineers and refer different soil dynamics books.

5.3.3.1 Use of soil constants and their relation

The values of the soil constant c'_z , c'_t , etc to be used in the design of actual machine foundation may obtained from the following relation (IS:5249-1969):

$$c'_z = c_z \sqrt{(A_b/A_t)} \dots \dots \dots eq. 5.12$$

$$c'_t = c_t \sqrt{(A_b/A_t)} \dots \dots \dots eq. 5.13$$

Where C_t - is the coefficient of elastic uniform shear and

C_z is the coefficient of elastic uniform compression

And is value obtained from the test on concrete block and A_b is the base area of the concrete block used in the test. Actual test have shown, however, that the inverse area relationship given by the above equation is not applicable beyond a certain area limit which may be taken as $10m^2$, after which the value of C_z may be assumed constant for design purposes. This is in view of the fact that the variation of C_z for large base areas, often encountered in machine-foundation problems is not well established [22].

Where exact data concerning soil are not available, the values of C_z given in the table may be used for preliminary designs. The values given in the table may be used for foundations with base areas of $10m^2$, or more. If the base area is less than $10m^2$, the values tabulated shall be multiplied by $\sqrt{(10/A_t)}$ where A_t is the actual base area of the machine [22].

Table 5.3 Recommended Design Values for C_z (IS: 2974-PLI-1969) [22]

Soil category	Permissible bearing capacity of soil In σ_p kg/cm ³	Coefficient of elastic uniform compression C_z in kg/cm ³
Weak soil	1	2
	2	4
Medium soil	3	5
	4	6
Strong soil	5	7
rocks	>5	>7

Having determined one of the soil constant say C_z from the insitu testing of the soil, the other dynamic soil constants may be evaluated approximately using the following relationship suggested by Barkan [22].

- | | |
|---|--------------------|
| i. Coefficient of elastic non-uniform compression | $C_\theta = 2C_z$ |
| ii. Coefficient of elastic uniform shear | $C_t = 0.5C_z$ |
| iii. Coefficient of elastic non-uniform shear | $C_\phi = 0.75C_z$ |

Expression for spring stiffness

The spring coefficients (K) of soil for different modes of vibration are calculated as follows [22]

- | | |
|--|-----------------------------------|
| i. For vertical motion | $K_z = C_z A_t$ |
| ii. For horizontal (or sliding) motion | $K_t = C_t A_t$ |
| iii. For rocking motion | $K_\theta = C_\theta I_{x(or y)}$ |
| iv. For torsion motion(rotation about vertical axis) | $K_\phi = C_\phi I_z$ |

Where A_t is area of horizontal contact surface between foundation and the soil, I is the second moment of contact area about the horizontal axis (x or y) passing through the centroid of the base and normal to the plane of rocking, and I_z is the second moment of contact area about the vertical axis passing through the centroid of the base [22].

Since the newly designed variable speed vibrating plate compactor vibrate in vertically and horizontally it is require determining the two spring constant of which are horizontal and vertical loading condition.

The coefficient of elastic uniform compression for soil to be compacted is 0.5771kg/cm^3 . Since area of the vibrating plate is small which is $0.85 \times 0.6\text{m}^2$ In order to determine the coefficient of elastic uniform compression should be multiplied by the factor $\sqrt{(10/A_t)}$. The soil which is ready to be compacted is weak soil and

$$C_z = \sqrt{(10/A_t)} \times 0.5771\text{kg/cm}^3 C_z = 2.56\text{kg/cm}^3$$

To determine the Coefficient of elastic uniform shear

$$C_t = 0.5C_z$$

$$C_t = 1.28\text{kg/cm}^3$$

To change to N/m^3 multiply with gravitational acceleration [22].

To determine the spring constant of the soil for the two loading condition

➤ For vertical motion

$$K_z = C_z A_t$$

$$K_z = 12.8 \times 10^6 \text{ N/m}$$

For the soil ready to be compacted

➤ For horizontal (or sliding) motion

$$K_t = C_t A_t$$

$$K_t = 6.4 \times 10^6 \text{ N/m}$$

For the soil ready to be compacted

Hence the values of ξ_t and ξ_z are the damping ratio of the soil when the loading is horizontal and vertical respectively. Whereas since the newly designed variable speed vibrating plate compactor is designed for the maximum condition. Geotechnical engineering the maximum damping ration of the soil is 0.3 because the damping ratio has direct relation with the damping angle. Usually damping ratio is defined as [22]

$$\xi = \frac{1}{2} \sin \psi$$

Where ψ is the damping angle and its maximum value is 37° since for large deformation the plastic strain γ_p will become almost equal the total strain γ_o and the shear stress τ_o will be almost equal to the maximum shear stress τ_{max} . The value of ψ then will approach its maximum value

$$\sin \psi_{max} = \frac{6}{\pi^2} \frac{\tau_o}{\tau_{max}} = \frac{6}{\pi^2} \left(\frac{\gamma_p}{\gamma_o} \right)^{\frac{1}{2}} = 0.6$$

$$\psi = 37^\circ$$

Therefore substituting the value of ψ we get for both vertical and horizontal loading

$$\xi = 0.3$$

It may well be possible that the value of the damping angle ψ is practically constant in particular when the elastic deformations are very small compared to the plastic deformation. Another possibility is that for small strain the energy dissipation rate tends towards zero. Thus the maximum damping ratio is about 0.3, on theoretical ground. The realistic values of damping ration of soft soils are of the order of 0.1 up to 0.2 [22]. **For this design purpose the damping ratio is selected to be 0.2**

To come up with the solution of the vibration response of the variable speed vibrating plate compactor it must done the vibration analysis

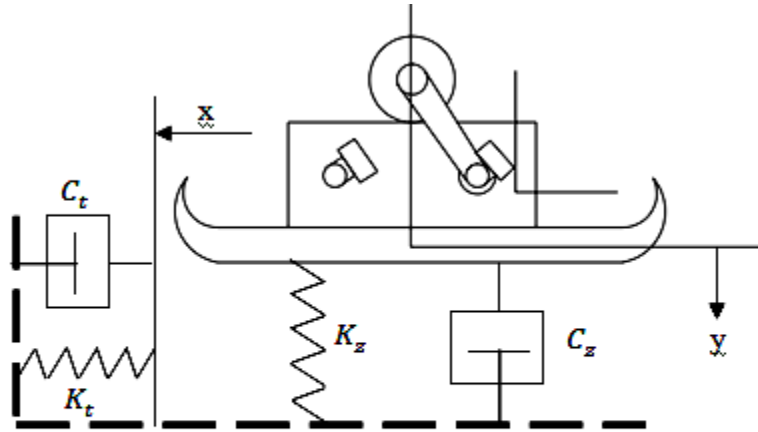


Figure 5.8: free body diagram representing the variable speed vibrating plate compactor subjected to spring and damping constant of the soil in two degree of freedom system.

Since the exciting force exerted by the eccentric rotating mass is the periodic external force causing variable speed vibrating plate compactor vibrates in both vertically and horizontally in sinusoidal manner.

Therefore the general equation of motion representing two degree of freedom of damped motion and external forcing function:-

$$M\ddot{x} + C_t\dot{x} + K_t x = me\omega^2 \sin \omega t \sin \theta \dots \dots \dots eq. 5.14$$

And

$$M\ddot{y} + C_z\dot{y} + K_z y = me\omega^2 \sin \omega t \cos \theta \dots \dots \dots eq. 5.15$$

The matrix form of the above two degree of freedom equation is

$$\begin{pmatrix} M & 0 \\ 0 & M \end{pmatrix} * \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{pmatrix} C_t & 0 \\ 0 & C_z \end{pmatrix} * \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{pmatrix} K_t & 0 \\ 0 & K_z \end{pmatrix} * \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} me\omega^2 \sin \omega t \sin \theta \\ me\omega^2 \sin \omega t \cos \theta \end{bmatrix}$$

Hence the general solution of the above two degree of freedom equation is obtained from any vibration analysis books. Therefore the general solution of these equations is as follows:-

$$x(t) = c_1 x_1(t) + c_2 x_2(t) + A \sin \omega t + B \cos \omega t$$

And

$$y(t) = d_1 y_1(t) + d_2 y_2(t) + C \sin \omega t + D \cos \omega t \text{ Respectively}$$

$$x(t) = x_c(t) + X(t)$$

$$x_c(t) = c_1 x_1(t) + c_2 x_2(t) \text{ And } X(t) = A \sin \omega t + B \cos \omega t \text{ and}$$

$$y(t) = y_c(t) + Y(t)$$

$$y_c(t) = d_1 y_1(t) + d_2 y_2(t) \text{ And } Y(t) = C \sin \omega t + D \cos \omega t$$

Where $\mathbf{x}_c(t)$ is the general solution of the homogeneous equation of horizontal loading,

$\mathbf{X}(t)$ Is the particular solution of the non homogeneous equation horizontal loading,

$\mathbf{y}_c(t)$ Is the general solution of the homogeneous equation of vertical loading

$\mathbf{Y}(t)$ Is the particular solution of the non homogeneous equation vertical loading respectively.

The homogeneous solution $x_1(t)$ and $x_2(t)$ are depend on the root r_1 and r_2 of the characteristic equation

$$Mr^2 + C_t r + K_t = 0$$

$$r = \frac{-C_t \pm \sqrt{C_t^2 - 4MK_t}}{2M}$$

Since M , C_t and K_t are all positive constants, it follows that r_1 and r_2 are either real and negative, or complex conjugates with negative real part.

Thus $\mathbf{x}_c(t) = \mathbf{c}_1 \mathbf{x}_1(t) + \mathbf{c}_2 \mathbf{x}_2(t)$ is called the transient solution. Note however that

$$\mathbf{X}(t) = \mathbf{A} \sin \omega t + \mathbf{B} \cos \omega t$$

is a steady oscillation with same frequency as forcing function. For this reason, $\mathbf{X}(t)$ is called the **steady-state solution, or forced response**. The same is also for vertical loading. With increasing time, the energy put into system by initial displacement and velocity is dissipated through damping force. The motion then becomes the response $\mathbf{X}(t)$ and $\mathbf{Y}(t)$ of the system to the external force $m\omega^2 \sin \omega t \sin \theta$ and $m\omega^2 \sin \omega t \cos \theta$ respectively.

Without the damping of the soil, the effect of the initial conditions would persist for all time.

Therefore without further do of simplification the general solution of the two degree of freedom from mechanical vibration books is as follows

$$X(t) = \frac{m\omega^2 \sin \theta}{K_t \sqrt{(1 - [\omega/\omega_{nt}]^2)^2 + (2\xi \omega/\omega_{nt})^2}} \sin[\omega t - \tan^{-1} \frac{2\xi \omega/\omega_{nt}}{1 - (\omega/\omega_{nt})^2}]$$

$$\omega_{nt} = \sqrt{\frac{K_t}{M}}$$

and

$$\phi_t = \tan^{-1} \frac{2\xi \omega/\omega_{nt}}{1 - (\omega/\omega_{nt})^2}]$$

And

$$\xi_t = c_t / 2\sqrt{MK_t}$$

And

$$Y(t) = \frac{me\omega^2 \cos \theta}{K_z \sqrt{(1 - [\omega/\omega_{nz}]^2)^2 + (2\xi \omega/\omega_{nz})^2}} \sin[\omega t - \tan^{-1} \frac{2\xi \omega/\omega_{nz}}{1 - (\omega/\omega_{nz})^2}]$$

$$\omega_{nt} = \sqrt{\frac{K_t}{M}} \quad \text{And}$$

$$\phi_z = \tan^{-1} \frac{2\xi \omega/\omega_{nz}}{1 - (\omega/\omega_{nz})^2}$$

$$\xi_z = c_t / 2\sqrt{MK_z}$$

Where the ϕ_z and ϕ_t are the phase lag for vertical and horizontal loading

From the above literature the values of spring constant, damping ration and natural frequency

Table 5.4: vibration data of horizontal and vertical loading

Loading condition	Spring constant	Damping ratio	Natural frequency
Horizontal loading	$K_t = 6.8 \times 10^6 \text{ N/m}$	$\xi_t = 0.2$	$\omega_{nt} = 238.05 \text{ rad/s}$
Vertical loading	$K_z = 12.8 \times 10^6 \text{ N/m}$	$\xi_z = 0.2$	$\omega_{nz} = 326.6 \text{ red/s}$

Therefore the general solution of the above expression by substituting tabulated values

$$X(t) = 20.437 \sin[278.57t + 55.54^0] \text{ mm}$$

$$y(t) = 16.763 \sin[278.57t + 55.22^0] \text{ mm}$$

And hence the maximum vertical amplitude of the variable speed vibrating plate compactor is obtained when the time reaches $\sin[278.57t + 55.22^0] = \frac{\pi}{2} \text{ sec}$

Therefore

$$Y_{max} = 16.763 \text{ mm}$$

Finally the graph of the vibration response of the machine to soil damping and spring constant property for both vertical and horizontal movement can be plotted using MatLab command:-

```
>> t=0:0.01:1; % time in second
```

```
>> x=20.437*(sin (278.57*t+0.966)); % the horizontal amplitude movement equation
```

```
>> plot(t,x)
```

```
>> grid on
```

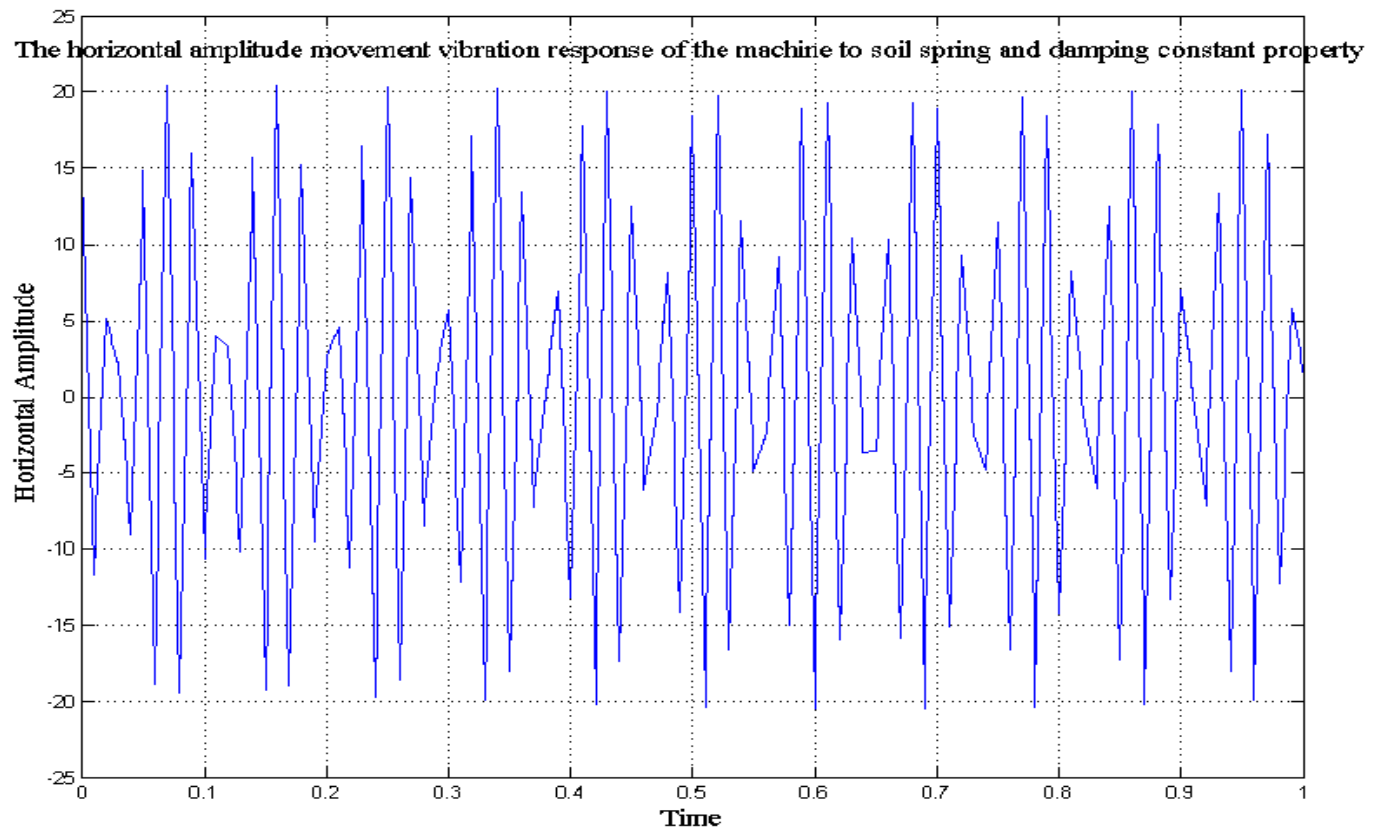


Figure 5.8.1:- Graph of the horizontal movement vibration response of compactor machine to

```
>> t=0:0.01:1; % time in second
```

```
>> Y=16.763*(sin (278.57*t+0.933)); % the vertical amplitude movement equation
```

```
>> plot(t,x)
```

```
>> grid on
```

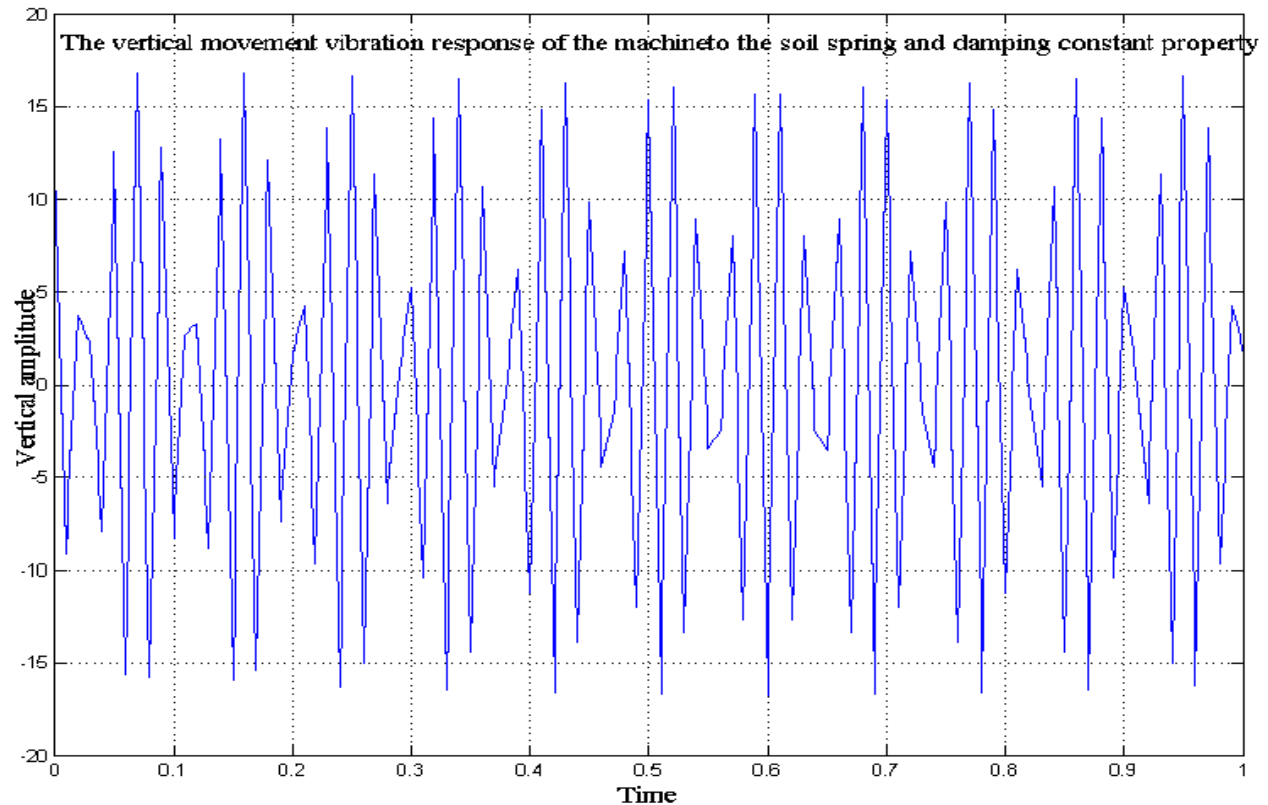


Figure 5.8.2:- Graph of the horizontal movement vibration response of compactor machine to soil

The above two graph results from the close values between the excitation frequency ω and the natural frequency ω_n of the soil. Although this looks like a mess that is incomprehensible, if we look at in the right way, the important features of the graph will come clear. Since $|\omega - \omega_n|$ is much smaller than $\omega + \omega_n$ we can think of as the "amplitude" of the periodic function. The amplitude is not a constant, but rather a *periodic function* itself. This variation of the amplitude is called a *beat or amplitude modulation*.

5.3.4 Vibration isolation

The problem of reducing the level of vibration in constructions and structures arises in various branches of engineering, technology, and industry. In most of today's mechatronic systems, a number of possible devices such as reaction or momentum wheels, rotating devices, and electric motors are essential to the system's operation and performance. These devices, however, can also be sources of detrimental vibrations that may significantly influence the mission performance, effectiveness, and accuracy of operation. Therefore, there is a need for vibration control. Several

techniques are utilized either to limit or alter the vibration response characteristics of such systems. [12]

In vibration isolation, either the source of vibration is isolated from the system of concern (also called “force transmissibility”), or the device is protected from vibration of its point of attachment (also called “displacement transmissibility”). [12]

In the case variable speed vibrating plate compactor it is selected both vibration isolation which are displacement transmissibility and force transmissibility that isolate the upper part from vibrating plate.

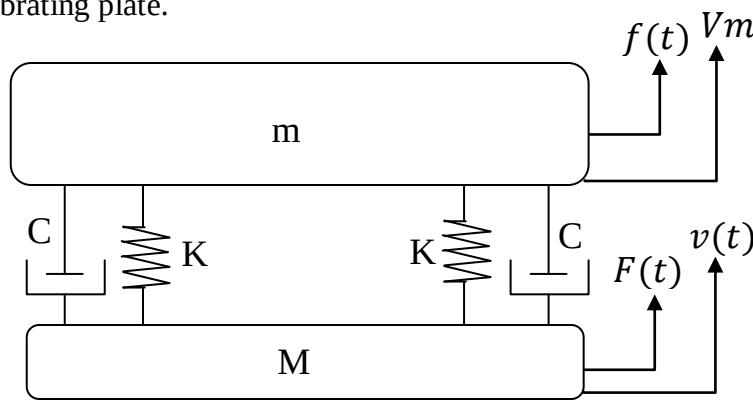


Figure 5.8.3:- free body diagram of vibration isolation

C: - damping constant

M: - mass of vibrating plate

K: - spring constant

m: - Mass of upper part

$F(t)$: - Excitation force

$f(t)$: - Isolated force

$v(t)$: - Excitation velocity

Vm : - Isolated velocity

As a result, the concepts of force transmissibility and motion transmissibility may be studied using just one common transmissibility function T ; the transmissibility function is given by

$$T = \frac{(K + Cj\omega)}{(K - m\omega^2 + Cj\omega)} \dots \dots \dots eq. 5.3.4.1$$

where ω is the frequency of vibration excitation. Note that the model (Equation 5.3.4.1) is not restricted to sinusoidal vibrations. Any general vibration excitation may be represented by a Fourier spectrum, which is a function of frequency ω : Then, the response vibration spectrum is obtained by multiplying the excitation spectrum by the transmissibility function T : The associated design problem is to select the isolator parameters K and C to meet the specifications of isolation. Equation 5.3.4.1 may be expressed as [12]

$$T = \frac{(\omega_n^2 + 2\zeta\omega_n\omega j)}{(\omega_n^2 - \omega^2 + 2\zeta\omega_n\omega j)} \dots \dots \dots eq. 5.3.4.2$$

Where $\omega_n = \sqrt{k/m}$ -----undamped natural frequency

$$\zeta = \frac{b}{2\sqrt{km}} \text{ -----damping ratio of the system}$$

Equation 5.3.4.1 may be written in the nondimensional form:

$$T = \frac{1 + 2\zeta r j}{1 - r^2 + 2\zeta r j} \dots \dots \dots eq. 5.3.4.3$$

Where the nondimensional excitation frequency is defined as:

$$r = \omega/\omega_n$$

The transmissibility function has a phase angle as well as magnitude. In practical applications, the level of attenuation of the vibration excitation (rather than the phase difference between the vibration excitation and the response) is of primary importance. Accordingly, the transmissibility magnitude [12]

$$|T| = \sqrt{\frac{1 + 4\zeta^2 r^2}{(1 - r^2)^2 + 4\zeta^2 r^2}} \dots \dots \dots eq. 5.3.4.4$$

is of interest. It can be shown that $|T| < 1$ for $r > \sqrt{2}$ which corresponds to the isolation region. Hence, the isolator should be designed such that the operative frequencies ω are greater than $\sqrt{2}\omega_n$. Furthermore, a threshold value for [T] would be specified, and the parameters k and c of the isolator should be chosen so that [T] is less than the specified threshold in the operating frequency range (which should be given). [12]

5.3.4.1 Design consideration

The level of isolation is defined as $1 - T$: It was noted that in the isolation region ($r > \sqrt{2}$) the transmissibility decreases (hence, the level of isolation increases) as the damping ratio ζ decreases. Thus, the best conditions of isolation are given by $\zeta = 0$: This is not feasible in practice, but we should maintain ζ as small as possible. For small ζ in the isolation region, Equation 5.3.4.4 may be approximated by [12]

$$T = \frac{1}{(r^2 - 1)} \dots \dots \dots eq. 5.3.4.5$$

Note that T is real, in this case, of $\zeta \cong 0$; and also is positive since $r > \sqrt{2}$. However, in general, T may denote the magnitude of the transmissibility function. Substitute

$$r^2 = \omega^2 / \omega_n^2 = \frac{\omega^2 m}{k}$$

We get $k = \frac{\omega^2 m T}{(1+T)} \dots \dots \dots eq. 5.3.4.6$

The required spring stiffness is found by substituting the values such as for $m = 1.375kg$, $\omega = 278.5 \frac{rad}{sec}$, $T = 0.8$ for $r = 1.5$ which is the smallest $r > \sqrt{2}$

And we get $k = 47.4kN/m$

This equation may be used to determine the design stiffness of the isolator for a specified level of isolation $(1 - T)$ in the operating frequency range $\omega > \omega_0$ for a system of known mass (including the isolator mass). Often, the static deflection δ_s of spring is used in design procedures and is given by:

$$\delta_s = \frac{mg}{k} \dots \dots \dots eq. 5.3.4.7$$

Substituting eq. 5.3.4.6 to eq. 5.3.4.7 we obtain

$$\delta_s = \frac{(1 + T)g}{\omega^2 T} \dots \dots \dots eq. 5.3.4.8$$

$$\delta_s = 0.3mm$$

Since the isolation region is $\omega > \omega_n$ is desirable to make ω_n as small as possible in order to obtain the widest frequency range of operation. This is achieved by making the isolator as soft as possible (k as low as possible). However, there are limits to this in terms of structural strength, stability, and availability of springs. Then, m may be increased by adding an inertia block as the base of the system, which is then mounted on the isolator spring (with a damping layer) or an air-filled pneumatic mount. The inertia block will also lower the centroid of the system, thereby providing added desirable effects of stability and reducing rocking motions and noise transmission. For improved load distribution, instead of just one spring of design stiffness k, a set

of n springs, each with stiffness k/n and uniformly distributed under the inertia block, should be used.

It can be plotted the isolation graph for 0.1 damping ratio using MatLab command

```
>> r=0:0.132:3.5;
>> t=sqrt(((1+(0.04*((r).^2)))./(1-((r).^2)).^2+(0.04*((r).^2))));
>> plot(r,t)
>> grid on
```

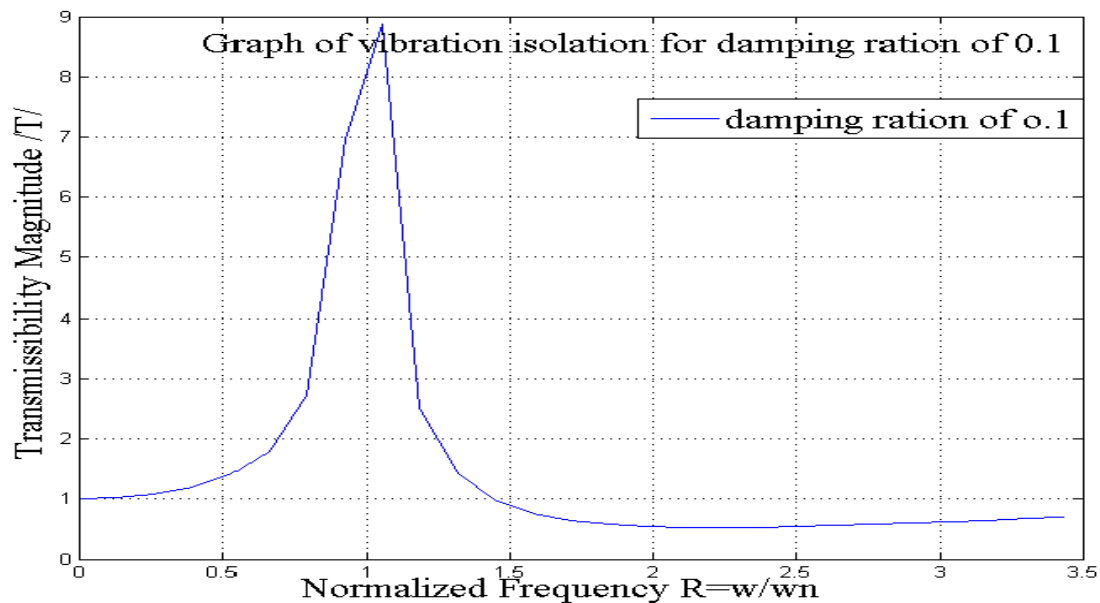


Figure 5.8.3:- graph of vibration isolation

Another requirement for good vibration isolation is low damping. Usually, metal springs have very low damping (typically ζ less than 0.01). On the other hand, higher damping is needed to reduce resonant vibrations that will be encountered during start-up and shutdown conditions when the excitation frequency will vary and pass through the resonances. In addition, vibration energy has to be effectively dissipated, even under steady operating conditions. Isolation pads made of damping material such as cork, natural rubber, and neoprene may be used for this purpose. They can provide damping ratios of the order of 0.01.

5.4. Electric motor selection

Since the angular speed required for variable speed vibrating plate compactor to deliver the maximum energy per area which is exactly equal to the modified proctor test energy at the

laboratory test, it is selected the appropriate type of synchronized electric motor which has the angular speed of 2661.5RPM or relatively above.

From synchronized electric motor manufacturers it is found that 6Hp, 2900RPM and three phase synchronized electric motor is selected.

5.5.Rotating shaft design

A shaft is a rotating member, usually of circular cross section, used to transmit power or motion. In our case rotating shaft transmit power from the prime mover to ERM (eccentric rotating mass). Therefore rotating shaft should withstand the loads such as bending, shear stress and fatigue load due to centrifugal force exerted by ERM and the tortional stress comes from the prime mover.

Shaft layout

The general layout of a shaft to accommodate shaft elements, e.g. gears, bearings, pulleys and eccentric rotating mass, must be specified early in the design process in order to perform a free body force analysis and to obtain shear-moment diagrams. The geometry of a shaft is generally that of a stepped cylinder. The use of shaft shoulders is an excellent means of axially locating the shaft elements and to carry any thrust loads.

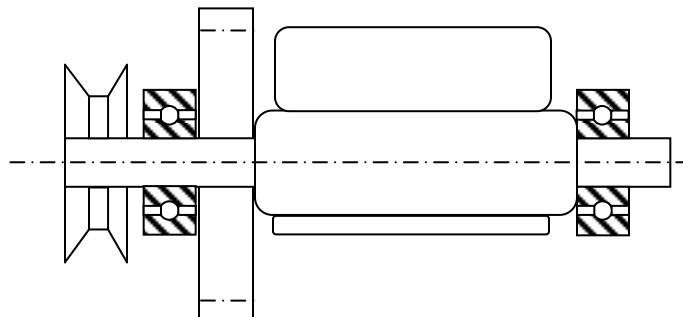


Figure 5.9:- shaft layout

Axial layout of the components

The axial positioning of components is often dictated by the layout of the housing and other meshing components. In general, it is best to support load-carrying components between bearings rather than cantilevered outboard of the bearings. Pulleys and sprockets often need to be mounted outboard for ease of installation of the belt or chain. The length of the cantilever should be kept short to minimize the deflection. Only two bearings should be used in most cases. For extremely long shafts carrying several load-bearing components, it may be necessary to provide more than

two bearing supports. Shafts should be kept short to minimize bending moments and deflections. But in this machine application it is only use two bearings to support the rotating shaft.

Supporting the axial load

In cases where axial loads are not trivial, it is necessary to provide a means to transfer the axial loads into the shaft, then through a bearing to the ground. Often, the same means of providing axial location, e.g., shoulders, retaining rings, and pins, will be used to also transmit the axial load into the shaft.

Providing for torque transmission

Most shafts serve to transmit torque from an input gear or pulley, through the shaft, to an output gear or pulley. Of course, the shaft itself must be sized to support the torsional stress and torsional deflection. It is also necessary to provide a means of transmitting the torque between the shaft and the gears. Common torque-transfer elements are:

- Keys
- Splines
- Setscrews
- Pins
- Press and shrink fits
- Tapered fits

One of the most effective and economical means of transmitting moderate to high levels of torque is through a key that fits in a groove in the shaft and gear. Keyed components generally have a slip fit onto the shaft, so assembly and disassembly is easy. In this machine application key is selected as torque transfer element.

Shaft design stress

It is not necessary to evaluate the stresses in a shaft at every point; a few potentially critical locations will suffice. Critical locations will usually be on the outer surface, at axial locations where the bending moment is large, where the torque is present, and where stress concentrations exist. By direct comparison of various points along the shaft, a few critical locations can be identified upon which to base the design. Since the rotating shaft is subjected to cyclic loading due to rotating eccentric mass, determining the diameter of the rotating shaft is based on fatigue failure criterion of fluctuating stress.

Shaft stress

Bending, torsion, and axial stresses may be present in both midrange and alternating components. For analysis, it is simple enough to combine the different types of stresses into alternating and midrange von Mises stresses. Axial loads are usually comparatively very small at critical locations where bending and torsion dominate, so they will be left out of the following equations. The fluctuating stresses due to bending and torsion are given by

$$\begin{aligned}\sigma_a &= \frac{\sigma_{max} - \sigma_{min}}{2} \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \\ \sigma_a &= k_f \frac{M_a c}{I} \sigma_m = k_f \frac{M_m c}{I} \\ \tau_a &= k_{f.s} \frac{T_a c}{J} \tau_m = k_{f.s} \frac{T_m c}{J}\end{aligned}$$

Assuming a solid shaft with round cross section, appropriate geometry terms can be introduced for c , I , and J resulting in

$$\begin{aligned}\sigma_a &= k_f \frac{32M_a}{\pi d^3} \sigma_m = k_f \frac{32M_m}{\pi d^3} \\ \tau_a &= k_{f.s} \frac{16T_a}{\pi d^3} \tau_m = k_{f.s} \frac{16T_m}{\pi d^3}\end{aligned}$$

Combining these stresses in accordance with the distortion energy failure theory, the von Mises stresses for rotating round, solid shafts, neglecting axial loads, are given by

$$\begin{aligned}\sigma'_a &= (\sigma_a^2 + 3\tau_a^2)^{\frac{1}{2}} = [(k_f \frac{32M_a}{\pi d^3})^2 + 3(k_{f.s} \frac{16T_a}{\pi d^3})^2]^{\frac{1}{2}} \\ \sigma'_m &= (\sigma_m^2 + 3\tau_m^2)^{\frac{1}{2}} = [(k_f \frac{32M_m}{\pi d^3})^2 + 3(k_{f.s} \frac{16T_m}{\pi d^3})^2]^{\frac{1}{2}}\end{aligned}$$

The fatigue failure criteria for the modified Goodman line as expressed

$$\frac{1}{n} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}}$$

Where n — is the factor of safety

M_a — is the amplitude moment component

M_m — is the midrange moment component

σ_m — is the midrange stress component

σ_a — is the amplitude stress component

σ_{max} — is the maximum stress

σ_{min} – is the minimum stress

s_e – Is the endurance limit of the material

s_{ut} – Is the ultimate strength of the material

k_f – Is stress concentration factor subjected to bending load

$k_{f.s}$ – Is stress concentration factor subjected to torsional load

Therefore in order to determine the diameter of the shaft subjected to fatigue loading:

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{s_e} \left[4(k_f M_a)^2 + 3(k_{fs} T_a)^2 \right]^{\frac{1}{2}} + \frac{1}{s_{ut}} \left[4(k_f M_m)^2 + 3(k_{fs} T_m)^2 \right]^{\frac{1}{2}} \right\} \right)^{\frac{1}{3}}$$

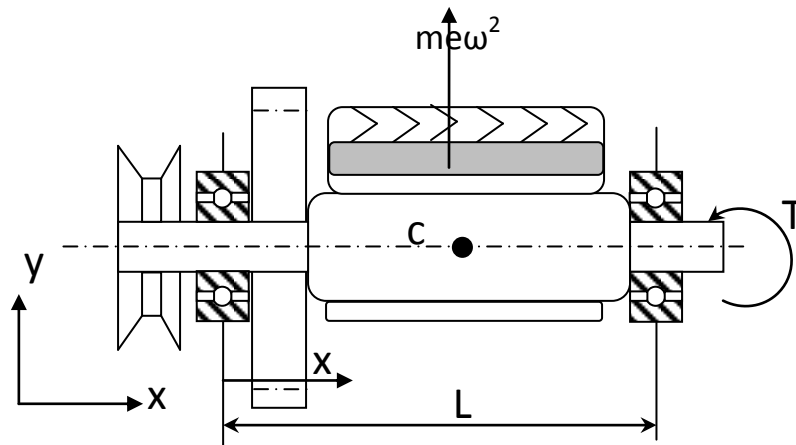


Figure 5.10:- free body diagram of ERM, pulley and rotating shaft

The maximum bending moment due to centrifugal force exerted by the ERM is located at the center 'c' where its value is product of centrifugal force and half of the length of the shaft

$$M = me\omega^2 L/2$$

Since the shaft is subjected to cyclic (sinusoidal) loading the maximum and minimum moment is $me\omega^2 L/2$ and zero respectively. Therefore

$$M_a = M_m = me\omega^2 L/4$$

And also the torque applied to shaft from the prime mover is

$$T = \frac{P * 60}{2\pi N}$$

Since it also cyclic loading the maximum and minimum torque is $\frac{P*60}{2\pi N}$ and zero respectively.

Therefore

$$T_a = T_m = \frac{P * 60}{4\pi N}$$

The shaft material is the quenched and tempered steel is appropriate for preparing shaft. The property of this steel is listed as followed and the endurance limit of the selected steel is $s_e = 700MPa$ for $s_{ut} > 1400MPa$

Table 5.5: shaft material property

Quenched and Tempered, Normalized and Annealed steel							
AISI No.	Material	Treatment	Yield strength (MPa)	Ultimate strength (MPa)	Endurance limit (MPa)	k_f	$k_{f.s}$
4140	Steel	Q&t	1640	1770	700	2.36	1.7

The shaft is subjected to cyclic loading and the values of loading is as shown in the table

Table 5.6:- shaft loading values

Shaft loading values							
M_{max} (Bending moment)(Nm)	$M_a=M_m$	T_{max} Torque (Nm)	$T_a=T_m$	Power (Hp)	Angular speed (RPM)	Length of shaft (mm)	Safety factor (n)
1067.02	533.51	16.07	8.035	6	278.57	250	2.5

Substituting the tabulated values in to the above formula therefore the diameter of the shaft is going to be

$$d \geq 39.99mm$$

In order to manufacture safe and standard shaft diameter it is selected to be

$$d = 45mm$$

5.6. Bearing selection

A bearing is a machine element that constrains relative motion and reduces friction between moving parts to only the desired motion. It may prevent a motion by controlling the vectors of normal forces that bear on the moving parts. Many bearings also facilitate the desired motion as much as possible, such as by minimizing friction.

Bearing life

When the ball or roller of rolling-contact bearings rolls, contact stresses occur on the inner ring, the rolling element, and on the outer ring. If a bearing is clean and properly lubricated, is mounted

and sealed against the entrance of dust and dirt, is maintained in this condition, and is operated at reasonable temperatures, then metal fatigue will be the only cause of failure. In as much as metal fatigue implies many millions of stress applications successfully endured, we need a quantitative life measure. Common life measures are

- Number of revolutions of the inner ring (outer ring stationary) until the first tangible evidence of fatigue
- Number of hours of use at a standard angular speed until the first tangible evidence of fatigue

The commonly used term is bearing life, which is applied to either of the measures just mentioned. The rating life is the 10th percentile location of the bearing group's revolutions-to-failure distribution.

Bearing load life at rated reliability

To establish a single point, load F_1 and the rating life of group one $(L_{10})_1$ are the coordinates that are logarithmically transformed. The reliability associated with this point, and all other points, is 0.90. Thus we gain a glimpse of the load-life function at 0.90 reliability. Using a regression equation of the form

$$F L_1^{\frac{1}{a}} = \text{const.}$$

The result of many tests for various kinds of bearings result in

- $a = 3$ for ball bearings
- $a = 10/3$ for roller bearings (cylindrical and tapered roller)

As declared in the manufacturer's catalog to correspond to a basic load rating in the catalog for each bearing manufactured, as their rating life. We shall call this the catalog load rating and display it algebraically as C_{10} , to denote it as the 10th percentile rating life for a particular bearing in the catalog. It is written as

$$F_1 L_1^{\frac{1}{a}} = F_2 L_2^{\frac{1}{a}}$$

And associate load F_1 with C_{10} , life measure L_1 with L_{10} , and write

$$C_{10} L_{10}^{\frac{1}{a}} = F L^{\frac{1}{a}} \quad \text{where the unit of } L \text{ is revolution}$$

Further we can write

$$C_{10} (L_R n_R 60)^{\frac{1}{a}} = F_D (L_D n_D 60)^{\frac{1}{a}}$$

where

C_{10} – is catalog rating in lbf or KN

F_D – is desired radial load in lbf or KN

L_R – is rating life in hour

L_D – is desired life in hour

n_R – is rating speed in rev/min

n_D – is desired speed in rev/min

$$C_{10} = \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{\frac{1}{a}}$$

Therefore in order to select the required bearing according to loading condition such as by Considering SKF cylindrical roller bearing manufacturer where cylindrical roller bearing have high load rating compared with other bearing type then it is suitable for this application, which rates its bearings for 1 million revolutions, so that L_{10} life is $L_R n_R 60 = 10^6$ revolutions, The $L_R n_R 60$ product produces a familiar number it is $90(10^6)$ revolutions. Then the selected bearing should work 8hr per day, 6days per week for two years life at 2661.5 RPM at the radial load of exactly equal to the centrifugal force exerted by ERM which is 8.537kN with a reliability of 90 percent, hence by calculating the corresponding catalog rating then we would select bearing in an SKF catalog.

Substituting the values shown in the table in to the above formulas

Table 5.7 bearing loading condition

$F_D (kN)$	$L_D (hr)$	$n_D (RPM)$	$L_R n_R 60$	a
8.537	4992	2661.5	10^6	$\frac{10}{3}$

Therefore the catalog rating is

$$C_{10} = 63.35kN$$

Taking $C_{10} = 63.35kN$ to SKF catalog rating to select bearing

Then we get **NUP 2207 ECP, 23mm width, bore diameter 35mm, outer diameter 72mm** and catalog rating is 69.5kN since the calculated catalog rating is much less than the standard one then it is selected little higher one.

5.7. Belt and pulley design

5.7.1 Belt design

Assumptions

➤ Selecting a motor of three phase synchronized type, power 6hp (4.476kw) and speed on an eighth pole type synchronized (ac) motor of 2900rpm, which is as expressed in table. For general

purpose ac motors of speed ranges and corresponding rated power. (Since the speed varies in this range according to the manufacturer's catalogue.

Then the smaller pulley is on the motor shaft.

➤ Belt-type: V-belt, used for smaller center distance drives.

➤ Belt material: leather belt with ultimate tensile strength:

$$\sigma_{ult} = 30MPa \text{ and}$$

$$\text{factor of safety } f = 9$$

➤ Coefficient of friction $\mu = 0.25$

➤ For good life of the belt, allowable tensile strength is taken to be

$$\sigma_{all} = 3.33MPa$$

Design power

The output power on the motor shaft is obtained from its belt efficiency (0.80 - 0.95), rated power, and service factor from table for machinery and normal service 8-10 hr,

Service factor of motors: $C_s = 1.3$ Then design power:

$$P_d = P_m * C_s * \eta_b \dots \dots \dots eq. 5.7.1$$

$$P_d = 4.476Kw * 1.3 * 0.8$$

$$P_d = 4.655KW$$

V-belt cross section

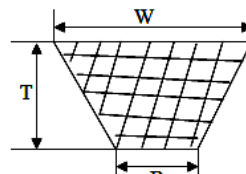


Figure 5.11:- sectional view of v-belt

From standard table, type-A V-belt is used for a rated power range of 0.2– 7.46kw; therefore we use type A V-belt which has a dimension of:

$$W = 13mm, \text{ groove angle } \beta = 34^0, 36^0, 38^0$$

$$T = 8mm \text{ and } B = W - 2T \tan \beta$$

$$B = 7.75mm$$

Weight per meter length = 1.06N/m

$$\text{Density } \rho = \frac{1000k}{m^3}$$

For 6hp motor, shaft diameter will be 28mm and pitch diameter of the smallest pulley will be 75mm, and above.

➤ let smaller pulley diameter, $d_s = 80\text{mm}$

Speed ratio

$$i = \frac{n_m}{n_{ERM}} = \frac{2900\text{rpm}}{2661.5\text{rpm}} = 1.1$$

➤ Pitch diameter of larger pulley

$$D = 88\text{mm}$$

Let centre distance

$$C = 250\text{mm}$$

Angle of contact arc of the belt will be calculated as:

$$\theta = 2 \cos^{-1} \left(\frac{D - d_s}{2C} \right) \dots \dots \dots eq. 5.7.2$$

$$\theta = 178.17^\circ$$

Belt pitch length is calculated as:-

$$L = 2C + \pi(R + r_s) + \frac{(R - r_s)^2}{C} \dots \dots \dots eq. 5.7.3$$

$$L = 763.824\text{mm}$$

From standard v-belt length it is selected little higher from calculated value

$$L = 792.67\text{mm}$$

Since pitch length of the belt is changed the center distance must be recalculated

$$C = 264.425\text{mm}$$

Mass of the belt is calculated as:-

$$M = \rho * A * L$$

Where A is the cross sectional area of the belt and L is pitch length of the belt

$$A = \frac{1}{2}(W + B)T$$

$$A = 82.96\text{mm}^2$$

Therefore

$$M = \frac{1000\text{kg}}{\text{m}^3} * 82.96\text{mm}^2 * 792.67\text{mm}$$

$$M = 0.066\text{kg}$$

Centrifugal tension

$$T_c = MV^2 \quad \text{where } V = 18.5\text{m/s} \quad \text{from belt spesification}$$

$$T_c = 22.59\text{N}$$

The maximum tension that the belt can with stand is:

$$T_{max} = A * \sigma_{all} \quad \text{then} \quad T_{max} = 276.26\text{N}$$

Now it is possible to calculate the tension in the tight side:

$$T_t = T_{max} - T_c \quad \text{then} \quad T_t = 253.67\text{N}$$

Tension in the slack side is calculated as follows:

$$2.3 \log \frac{T_t}{T_s} = \theta \cdot \csc \beta \quad \text{where } \beta = \frac{180 - \theta}{2}$$

$$\frac{T_t}{T_s} = 3.1 \quad \text{therefore} \quad T_s = 81.83\text{N}$$

The initial belt tension would be:

$$T_i = \frac{T_t + T_s}{2} \quad \text{therefore} \quad T_i = 167.75\text{N}$$

The maximum power the belt can transmit is:

$$P_{max} = (T_t - T_s) * V \quad \text{then} \quad P_{max} = 3.2\text{kW or } 4.3\text{hp}$$

Therefore since the maximum power the belt can transmit is less than input power which is 4.476kW or 6 hp, then in order to transmit required amount of power to rotating shaft we need extra belt.

Therefore the required number of belt is calculated as:

$$\text{no. of belts} = \frac{\text{design Hp}}{\text{standard tranmision Hp} * \left[\text{coefficient of arc contact} \right] * \left[\text{coefficient of belt length} \right]} \dots \dots \text{eq. 5.7.4}$$

From the table $\left[\text{coefficient of arc contact} \right] \approx 1$ and $\left[\text{coefficient of belt length} \right] = 1.04$

$$\text{number of belt} = \frac{6.254\text{hp}}{4.3 * 1 * 1.04} \quad \text{then} \quad \text{number of belt} = 1.4$$

Hence it is recommended to use **two numbers of belts**

Actual speed of belt is

$$V_a = \frac{\pi d_s \omega_m}{60} \quad \text{then} \quad V_a = 12.1\text{m/s}$$

Which less than belt specification speed and it is safe.

5.7.2 Pulley

Type of pulley: V-grooved pulley.

- Number of grooves on the pulleys =2, for one belt 3.2kw power rating.
- Material: alloy steel or cast iron.
- Manufacturing of pulleys:

By machining (especially for pulleys made of ductile materials) or casting techniques (for Brittle materials).

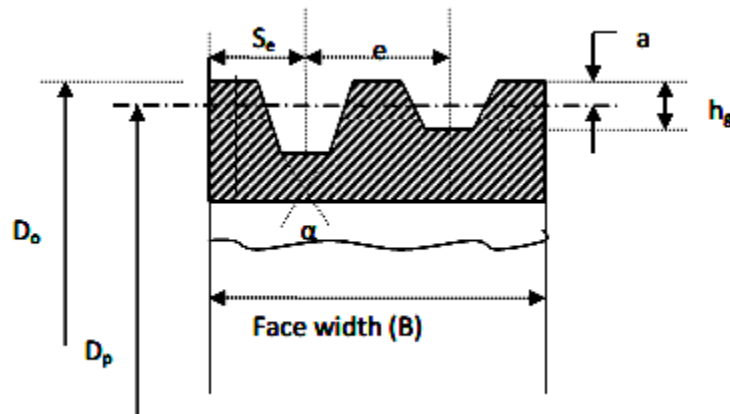


Figure 5.12:- sectional view of pulley

For type-A V-belt grooved pulley:

Groove angle: $\alpha = 38^\circ$

$a = 4\text{mm}$ (minimum)

$H_g = 12\text{mm}$ (minimum)

Face width, $B = e(N - 1) + 2Se$, where N : number of grooves.

$$B = 20(2 - 1) + 2 \times 15$$

$$B = 50\text{mm}$$

Hence the speed of the larger pulley is exactly equal to the speed of rotating shaft.

Therefore the dimension of the pulley is:

Table 5.8:- overall designed dimension of pulley

Smaller pulley		Larger pulley
D_o	88mm	96mm
D_p	80mm	88mm
D_i	30mm	35mm
face width B	50mm	50mm

5.8. Eccentric rotating mass design

The centripetal force of the offset mass is asymmetric, resulting in a net centrifugal force, and this causes a displacement of the machine. In conceptual design single acting, hydraulically controlled variable eccentricity eccentric rotating mass is selected. So that in this detailed design part we are designing the spring which act as retracting and extending eccentricity alliance with hydraulic system.

5.8.1 Spring design

A compression spring is a mechanical device in the shape of a helix made from spring wire. It is used to store or release energy. It can also absorb shock or maintain a force between two surfaces. Compression springs are coil springs that resist a compressive force applied axially.

A compression spring used in variable eccentricity ERM is used to resist the centrifugal force exerted by ERM without the application of hydraulic pressure. However it is selected six compression spring arranged in paralleled connection and having the same spring constant.

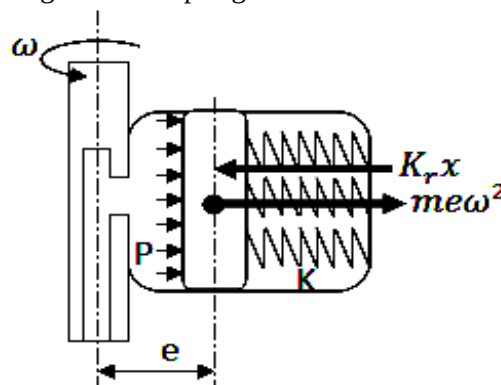


Figure 5.13:- free body diagram of variable eccentricity ERM

The geometry of the ERM is 230mmX45mmX19mm and 1.5kg weight. The spring is initially compressed to 5mm and the maximum compression would be 0.11kgm/1.5kg is 71mm.

Therefore the equivalent spring constant would be:

$$K_r * X_o = me\omega^2$$

Where e is the eccentricity measured from the center of the rotating shaft to mass center of ERM which is equal to $e = 22.5 + 9.5 = 32mm$ and $X_o = 5mm$

$$K_r = me\omega^2/X_o$$

$$K_r = \frac{me\omega^2}{X_o} \quad K_r = 745kN/m$$

Equivalent Spring constant in parallel connection is:

$$K_r = K_1 + K_2 + K_3 \dots \quad K_r = 6K$$

The individual spring constant would be: $K = K_r/6$

$$K = 124.2KN/m$$

Therefore the compression spring having spring constant of $K = 124.2KN/m$ and free length of **53.5mm** is selected.

The pump pressure required to push ERM to its maximum eccentricity i.e. 71mm is calculate as follows

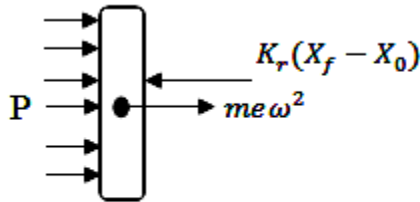


Figure 5.14:- Free body diagram of applied force vectors.

As indicated in the above free body diagram of the ERM the pump pressure is calculated by summation of force acting on the ERM.

$$PA + me\omega^2 = K_r(X_f - X_o)$$

However the final compression of the spring is: $X_f = 71mm + 5mm - 37.5mm$

$$X_f = 15mm$$

Substituting the values to the above expression we would find the pump pressure equal to

$$P = 2.42Mpa \quad \text{or} \quad 351\text{pound per inch}$$

5.9. Vibrating plate design

Vibrating plate is plate which is a key part of vibrating plate compactor. It is contact surface between the compactor machine and soil. In order to give the compaction of the soil it should have the property of soil compaction criterion such as kneading, static, dynamic or impact force

and vibratory . And consequently it should have the best material property and compaction criterion. Vibrating plate which is hydraulically adjustable sheep's foot steel plate is selected in order to give all compaction criterions as per geotechnical order. The retraction of sheep's foot from the plate is achieved by spring and extraction of the sheep's foot from the plate is achieved by hydraulic system.

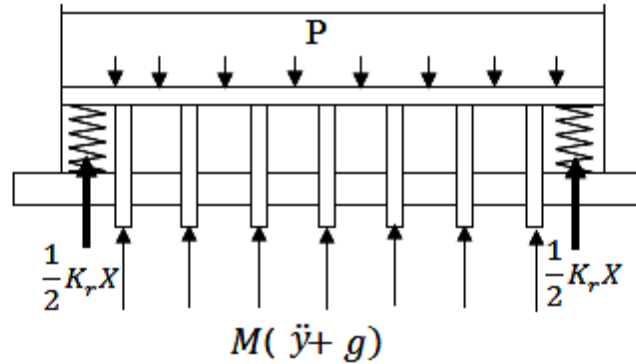


Figure 5.15:- free body diagram of vibrating plate force vectors

The compression spring used in this vibrating plate is used to support the kneading plate not extract by itself. However fourteen springs arranged in parallel with same spring constant is selected. When the vibrating plate is at flat position the compression springs are resisting the upward force exerted by the ERM. To install the compression spring is initially compressed to 10mm.

Therefore

$$K_r * X_o = M\ddot{y} \qquad \qquad \qquad K_r = 1287.84kN/m$$

The individual spring constant

$$K_r = 14K \qquad \qquad \qquad K = 92kN/m$$

The pump pressure is calculated as follows on which sheep's foot is extracted due to hydraulic pressure and the compression spring is compressed to 50mm from the uncompressed position. The area on which the hydraulic pressure applied is 540mmX680mm

$$P * A = K_r(x_f - x_o) + M(\ddot{y} + g)$$

Substituting the corresponding values in the formula

$$P = 178kpa$$

Since the pressure at ERM is much greater than at vibrating plate the pump selection is based on the pressure at ERM whereas pressure relief valve is selected to control the vibrating plate and set at 178kpa (25.82psi).

The pump specification can be read from standard pump graph based on the maximum calculated pump pressure which is pressure exerted at ERM adjustment.

From the graph **400 psi pump having 10gpm 2.5hp** of theoretical horsepower is selected.

And also pressure relief valve is selected to control ERM set at 2.41Mpa (351psi)

Whereas it is also selected that at 10gpm delivering pump, stroke (piston) velocity for both ERM and vibrating plate is 50Ipm (0.021mps).

5.10 Spur Gear design

Spur Gears have teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another parallel shaft. The two gears in variable speed vibrating plate compactor are in one to one gear ratio and are rotating in opposite direction which is used to overcome two sum able centrifugal forces at two opposite direction and omitted at any other point.

Assumptions:

- Material: alloy steel similar to that of shaft, class 40 cast iron.
- Manufacturing process: by machining, or the casting techniques. It is advisable to produce it by machining
- Type of teeth: involutes cut of normal pressure angle:20°
- Number of teeth on the gear: $Z_p = 63$
- Pitch diameter: $d_p = 190mm$.
- Gear ratio: $n=1$
- Module $m= 190mm/63 = 3$

5.10.1 Bending strength (stress)

We can design for strength Using the European standard of design analysis.

$$\sigma_b = \frac{F_t P_d K_v K_o K_m}{b J} \dots \dots \dots eq. 5.10.1$$

F_t – tangential force

b – face width = 15mm

P_d – diametral pitch

J – spur gear geometry factor = 0.466

K_v – velocity factor

K_o – overload factor = 1 for bending

K_m – load distribution (mounting) factor = 1.35

Table 5.9:- geometry of spur gear

Geometry value	
d_p	190mm
b	15mm
Z_p	63
m	3
K_v	0.185
P_d	0.333
K_m	1.35
K_o	1
J	0.466

$$P_d = Z_p / d_p = \frac{63}{190} = 0.333$$

$$K_v = \frac{6}{6 + V}$$

$$V = \frac{\pi d_p n}{60} = \pi * 0.16m * \frac{2661.5rpm}{60} = 26.5m/s$$

$$K_v = 0.185$$

$$F_t = \frac{2T}{d_p} = 201N$$

Then substituting the above value

$$\sigma_b = \frac{201 * 0.444 * 0.212 * 1 * 1.35}{15 * 0.466}$$

$$\sigma_b = 3.654Mpa$$

Check the pinion for bending failure:

Allowable bending stress number:

Sat: taken from table for induction or flame hardened steel, Sat = 22,000lb/in² = 151.74Mpa.

$$\sigma_b \leq \frac{S_{at} Y_N}{S_f K_T K_R} \dots \dots \dots eq. 5.10.2$$

S_f – safety factor = 5

Y_N : stress cycle factor, can be calculated as for the specified steel material

$$Y_N = 2.314(N_C)^{-0.0538}$$

N_C : number of cycles, assuming 10 years of service, number of cycles can be calculated as:

$$N_C = 10\text{yrs} \times 365\text{days/yr} \times 8\text{hrs/day} \times 60\text{min/hr} \times 2661.5\text{RPM}$$

$$N_C = 6.663 \times 10^9 \text{cycles}$$

$$Y_N = 0.7$$

K_R – reliability factor = 1.5

K_T – temperature factor = 1

Then bending stress on gear is calculated as:

$$\sigma_b \leq \frac{151.74\text{Mpa} \times 0.7}{5 \times 1.5 \times 1}$$

$$3.654\text{Mpa} \leq 14.2\text{Mpa which is safe for bending failure}$$

5.10.2 Surface fatigue stress calculation

$$SC = C_p \sqrt{\frac{F_t K_v K_o K_m}{b I_d}} \dots \dots \dots \text{eq. 5.10.3}$$

C_p : elastic coefficient. It has different values for the case of using steel ring gear and also in designing steel pinion on cast iron ring.

Surface fatigue stress is the stress occurs at the contact surface of the two meshing gears. Therefore we are going check the gear is safe from failure due to the surface fatigue stress in two cases. Case-1 checks the gear is safe from surface fatigue due to material property. Case-2 checks the gear is safe from surface fatigue by considering one as pinion.

Case-1

For steel gear on cast iron ring, from table

$$C_p = 179$$

d_p : pitch diameter of pinion gear=190mm

I: geometry factor, for 20° full depth pressure angle straight gears and gear ratio of 1:1 and number of teeth 76, I=0.11

Then;

$$SC = 179 \sqrt{\frac{201 \times 1.35 \times 0.444}{15 \times 190 \times 0.11}}$$

$$SC = 81.7\text{Mpa}$$

Check for surface fatigue failure

$$S_C \leq \frac{S_{ac} Z_N C_H}{S_H K_T K_R}$$

Where:

Sac: allowable contact stress number, from the table for flame or induction hardened, grade1 alloy steel,

$$S_{ac}=170,000\text{lb/in}^2 = 1172.57\text{Mpa}.$$

Z_N : stress cycle factor for pitting resistance

$$Z_N = 2.466 (N_c)^{-0.056}$$

$$Z_N = 0.71$$

SH: safety factor for pitting resistance taken 1.5.

CH: hardness ratio factor for pitting resistance

For gear ratio of 1:1 and ratio of hardness of pinion to hardness of ring gear less than one,

CH= 1.

Then,

$$S_C \leq \frac{1172.57\text{Mpa} * 0.71 * 1}{1.5 * 1.5}$$

$$81.7\text{Mpa} \leq 370\text{Mpa} \quad \text{SAFE}$$

Case-2

If there is an option to produce the ring from steel by machining, we can recheck the design for steel ring gear.

Now, $C_p = 191$.

CH = 1.

And other factors the same, the surface stress will be:

$$S_C = 191 \times 1.91$$

$$S_C = 346.81\text{Mpa}$$

.Check for surface fatigue failure of the pinion,

This is the same as first case $346.81\text{Mpa} \leq 370\text{Mpa} \quad \text{SAFE}$

5.10. Total Market Cost And Compaction Time Analysis

The time required to compact 100m² areas of construction site at maximum speed of the compactor machine is calculated as follow. The area of the variable speed vibrating plate compactor is 0.6X0.85 m². And also the forwarding velocity of the machine is 0.3m/s which is equal to 0.6X 0.3 m²/s area is compacted on single trip. Therefore in order to compact 100m² area it require

$$T_c = \frac{100m^2}{0.6 * 0.3 m^2/s}$$

$$T_c = 555.56sec$$

Table 5.10:- list of compactor machines and compaction time [23]

Compactor machines	Forward speed. (Meter per sec.)	Width of the compactor (m)	Time of compaction per single trip (sec)
Smooth wheel roller	1	2.0066	49.83
Sheep's foot roller	1.667	2.03m	29.551
Rammer	0.13	0.2m	3846.15
Reversible vibrating plate compactor	0.35	0.75m	381

- ❖ Note that the above compaction time is calculated by assuming the time consumed changing the direction and alignment is zero.

Therefore the newly designed variable speed vibrating plate compact is acceptable range of compaction time when it compared with exiting compaction machine.

Calculating Total Market Cost

According to ABAYA MECHANICAL ENGINEERING PLC, The manufacturing cost of variable speed vibrating plate compactor's parts and spares is listed in the table blow.

Table 5.11 manufacturing cost of variable speed vibrating plate compactor

Manufacturing cost of variable speed vibrating plate compactor (birr)	
Vibrating plate	1x30,000
Rotating shaft	2x3,500
Electric motor	1x13,000
Hydraulic pump	1x5,000
Hydraulic valves	2x1200
Hydraulic hose	3m x 800
Bearing	4x 450
Handle	1x 500
Cover	1x1000
Vibration isolator (damper)	4x 1000
ERM	2x4,500
Gears	2x6,500
Pulley	2x 2500
Frames	1x1000
Electric switch	2x300
Man power	15,000
Auxiliary work	12,000
Total	122,700 birr

The total market cost is calculated by multiplying manufacturing cost by 75% profit cost, 20% selling cost, 35% design cost, 10% transportation cost and 5% packing cost.

$$total\ market\ cost = manufacturing\ cost * (100 + 35 + 15 + 20 + 7 + 5)\%$$

$$total\ market\ cost = 122,700 * 1.82\%$$

total market cost = 223,314 birr

Comparing the total market with the exiting compaction machine

Table 5.12:-cost comparison of different compaction machine [23]

Compaction machine	Cost (USD)
Vibratory smooth wheel roller	26700
Sheep's foot roller	29400
Reversible vibrating plate compactor	9400
Rammer	1100

Note that the above tabulated prices of four different compactor machines are listed without including the transportation cost and tax in us dollars.

Since the newly designed variable speed vibrating plate compactor has the ability to offer the characteristics of stated four compactor machines varying the surface characteristics and speed, amplitude and compaction energy, the total cost of variable speed vibrating plate compactor is cheap when it compare with the listed tabulated price of the four different compactor machines.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1. Conclusion

This design come up with the solution of developing the compactor machine that can deliver different level of compaction energy and frequency replacing currently used compactor machines. Since currently used machines are designed to compact as per soil type and recommended compaction energy it is required to use different compactor machine. And hence if the variable speed vibrating plate compactor is manufactured it can deliver different level of energy and compaction criterion such as Kneading, static, impact and vibratory. But it has limitation to cover large and long construction areas.

Variable speed vibrating plate compactor has the property and characteristics of four different soil compactor machines such as sheep's foot roller, Smooth-wheel roller, Rammer (Jumper), Single speed Vibrating plate compactor.

Variable speed vibrating plate compactor has the ability of maximum efficiency to compact 100m² areas within 600second. Therefore the newly designed variable speed vibrating plate compact is within acceptable range of compaction time when it compared with exiting compaction machine.

5.2. Recommendation

The prime mover selected in this design analysis based on the solution principle is synchronized electric motor of 2900RPM and 6hp but since the compactor machines used on field and to increase the portability of the machine it can be replaced by diesel engine with the same specification.

The reversed (backward) movement of the variable speed vibrating plate compactor for this design is achieved by turning the face back manually. But the reversible movement of the machine needs further investigation and solution.

Since the detail design is addressed and every components dimension is specified the manufacturing and development of variable speed vibrating plate compactor should be keep on going.

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APPENDIX

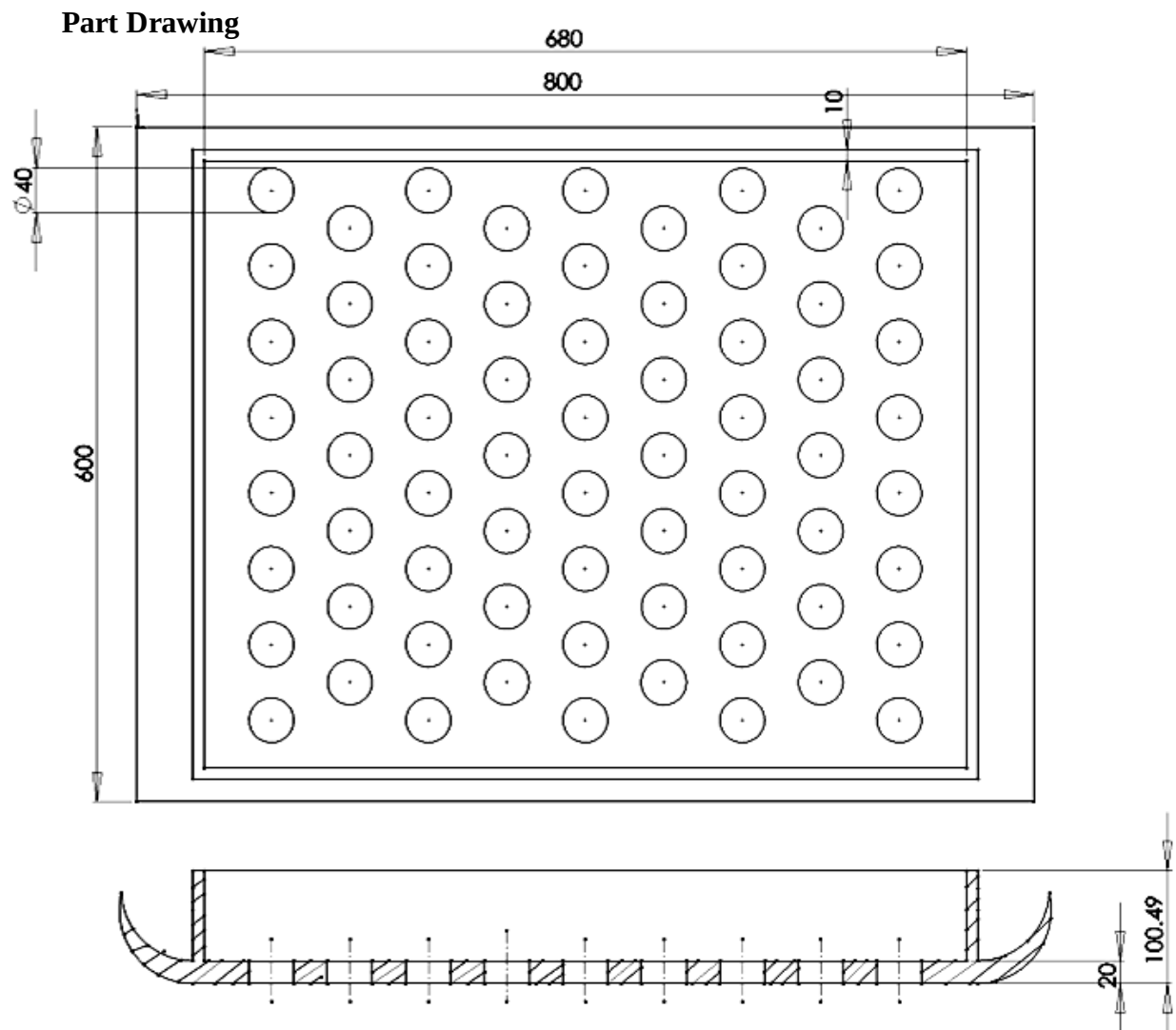


Figure 6.1:- hydraulically controlled vibrating plate

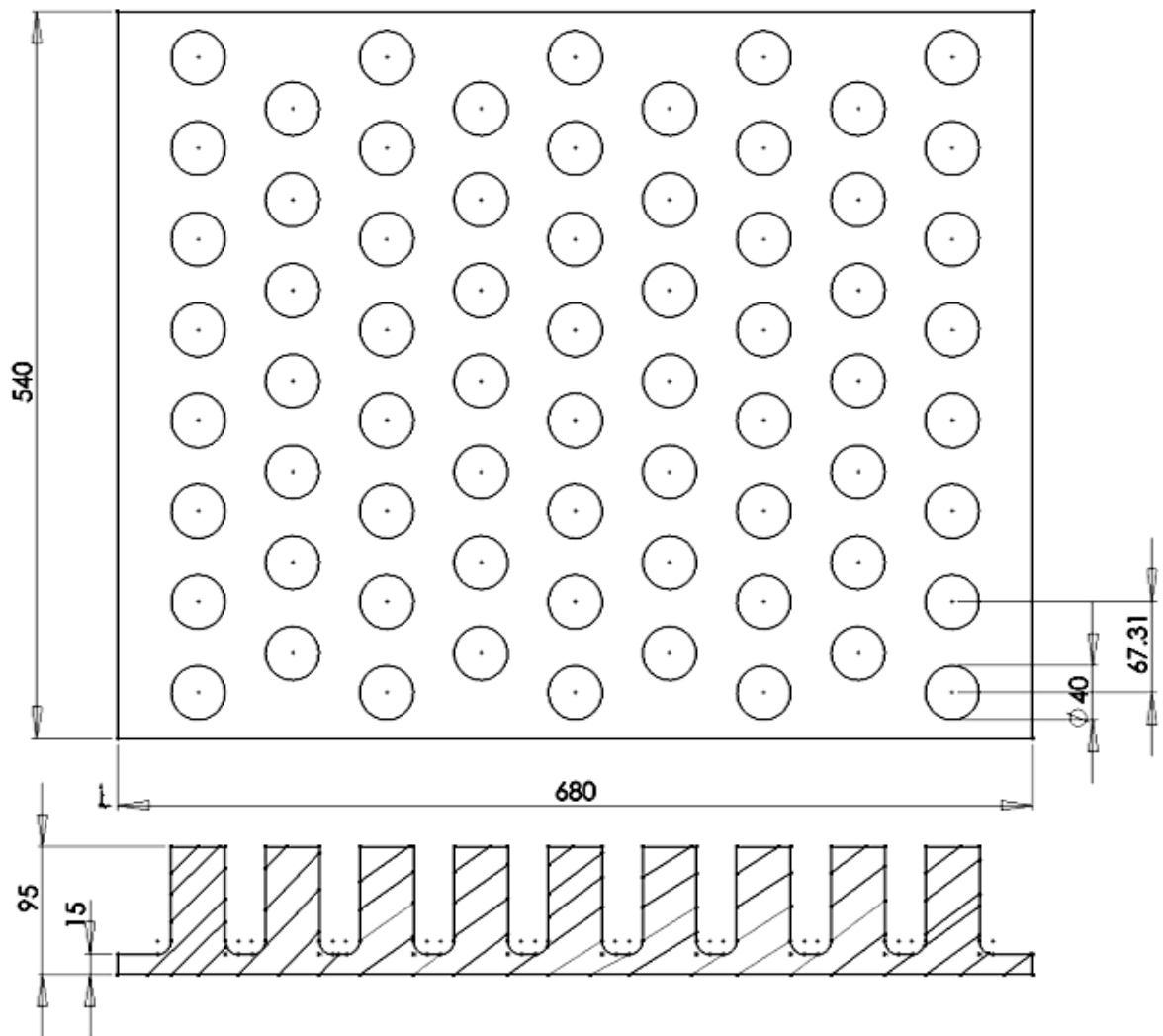


Figure 6.2:- sectional view of kneading

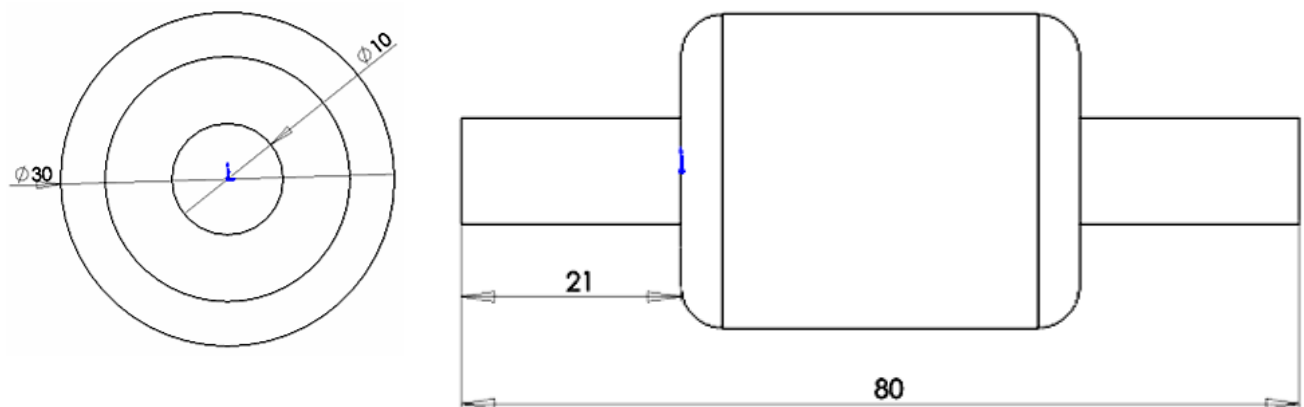


Figure 6.3:- vibration isolator (damper)

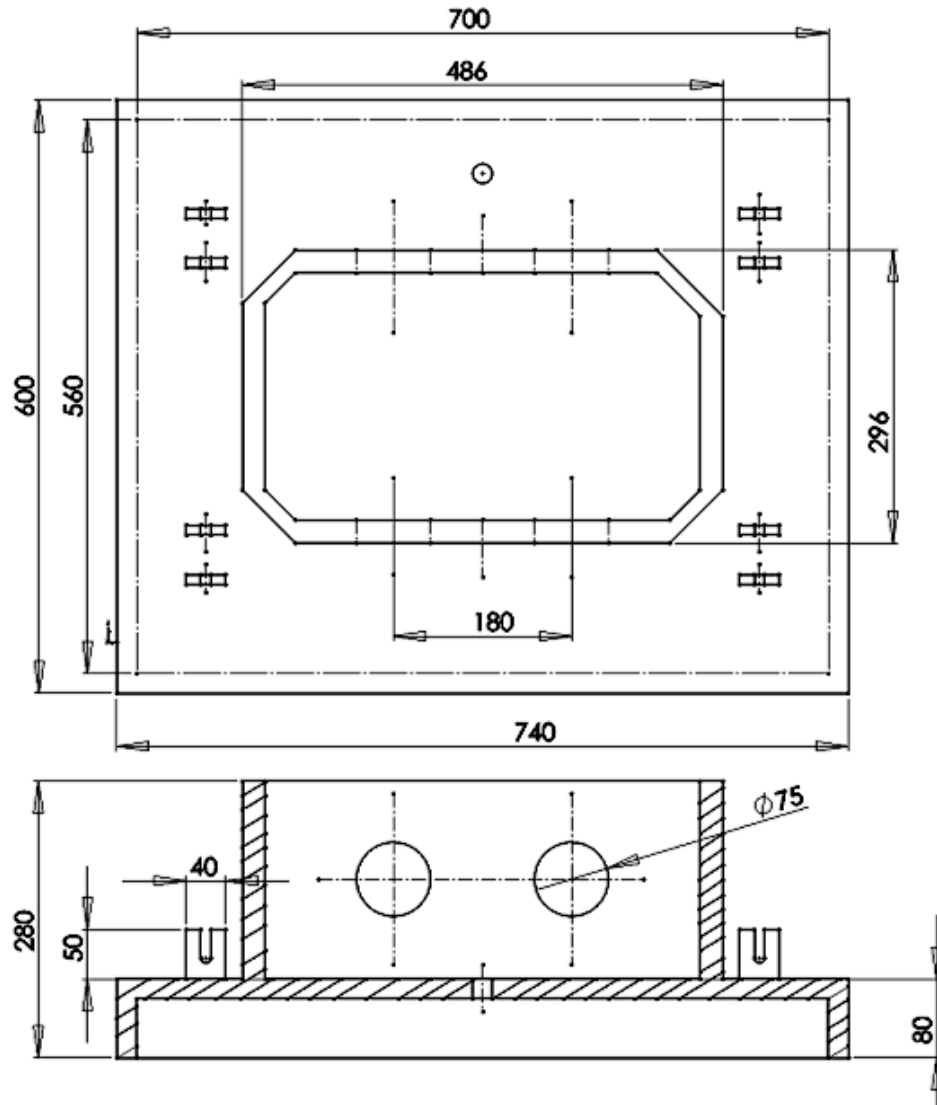


Figure 6.4 :-sectional view of vibrating plate cover

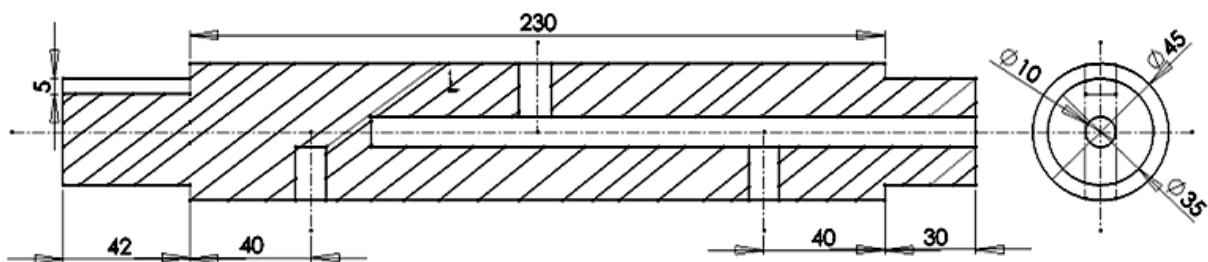


Figure 6.5:- sectional view of rotating shaft

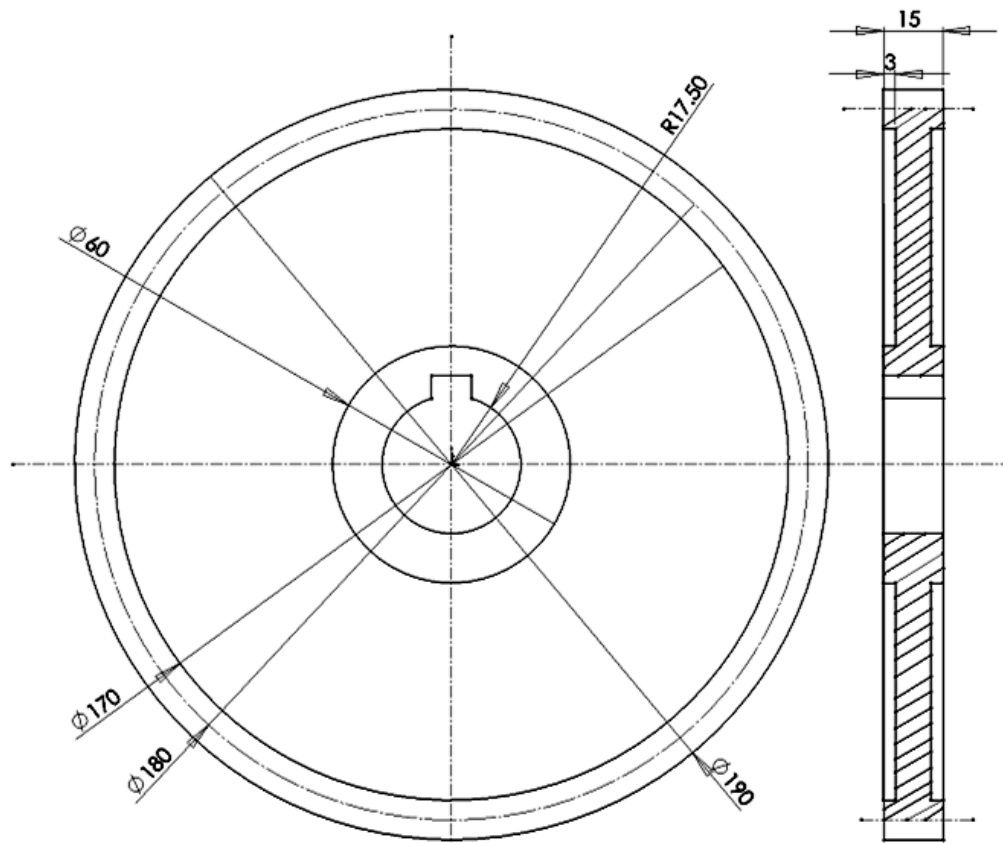


Figure 6.6: - sectional view of gear

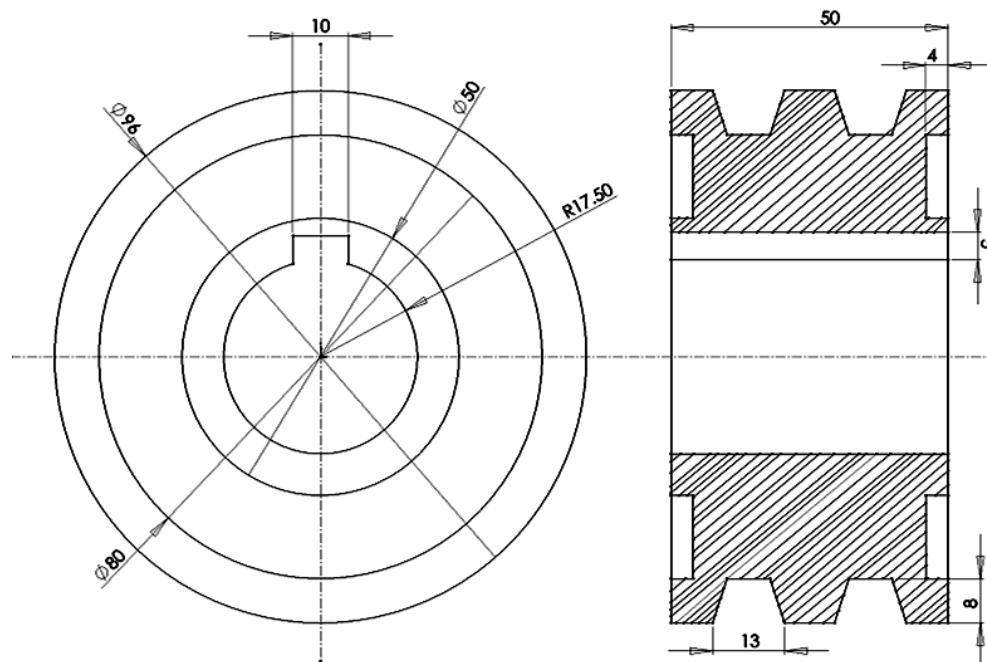


Figure 6.7: - sectional view of pulley

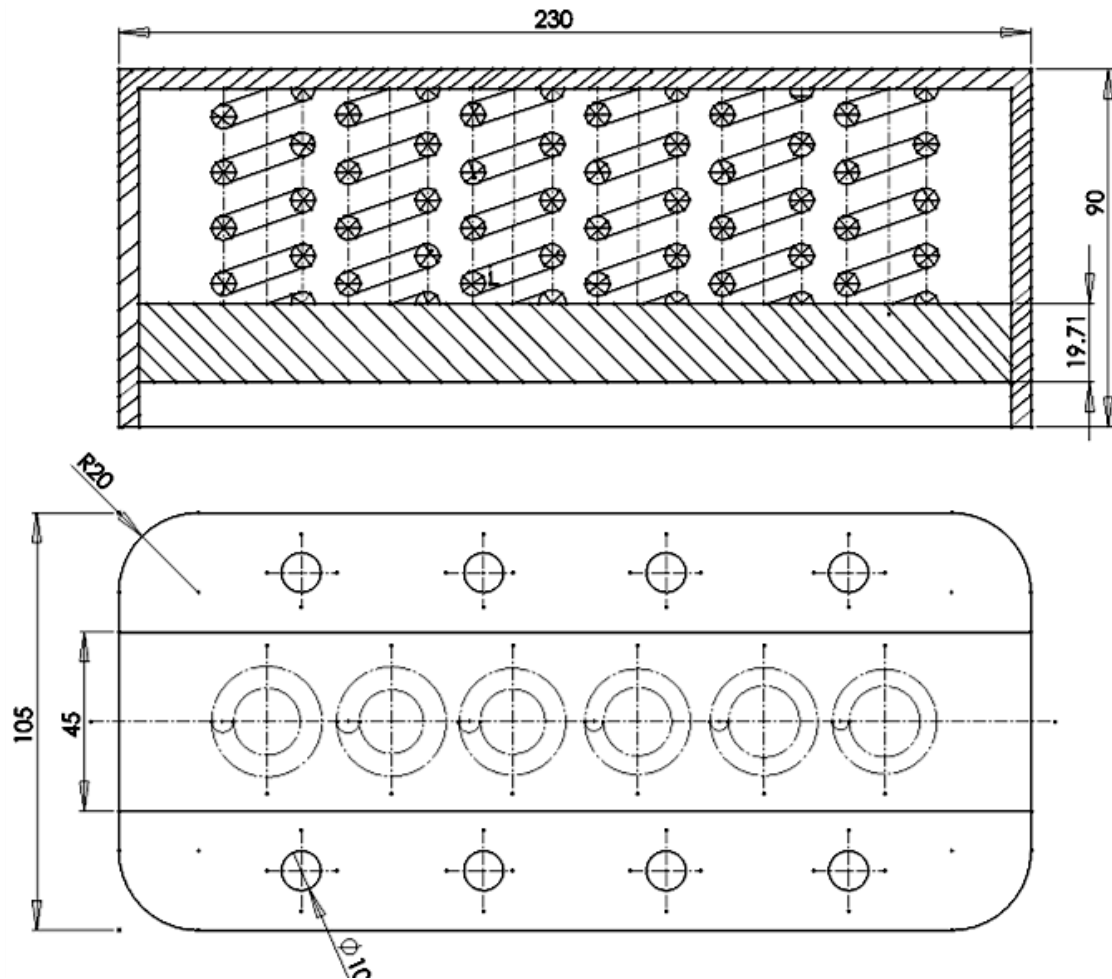


Figure 6.8: - sectional view of hydraulically controlled ERM

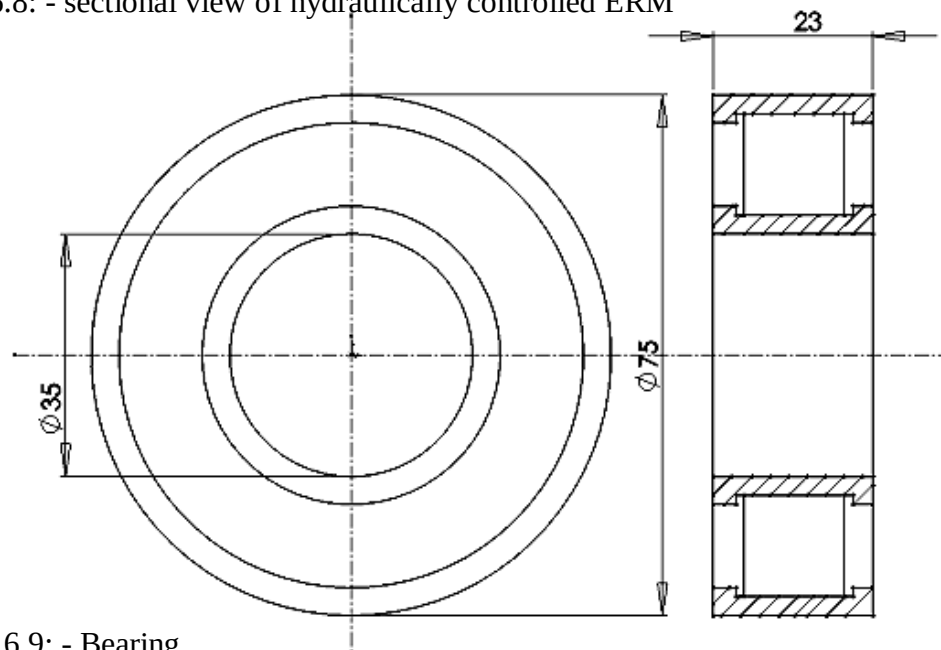


Figure 6.9: - Bearing

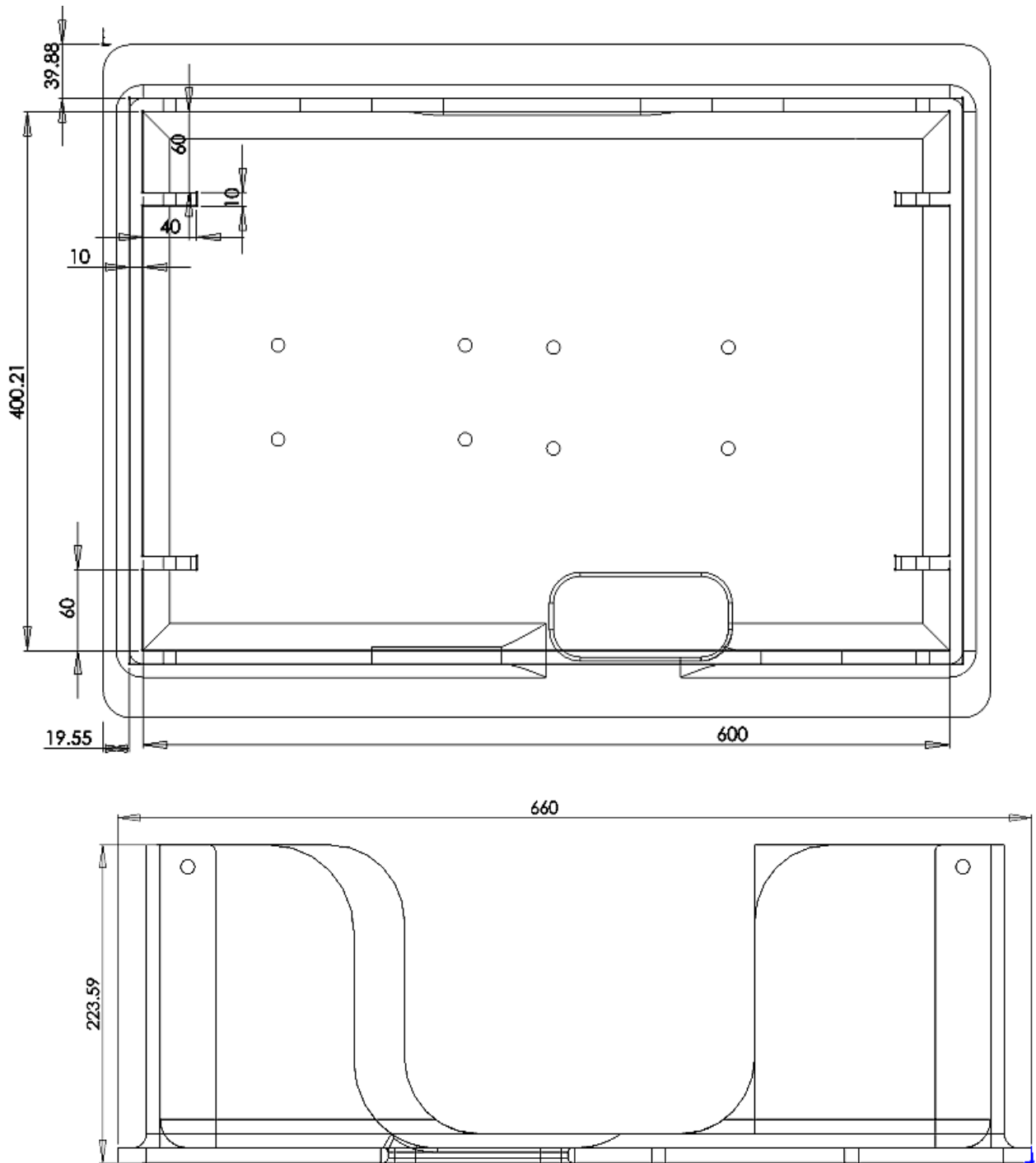


Figure 6.10:- base

Hydraulic circuits

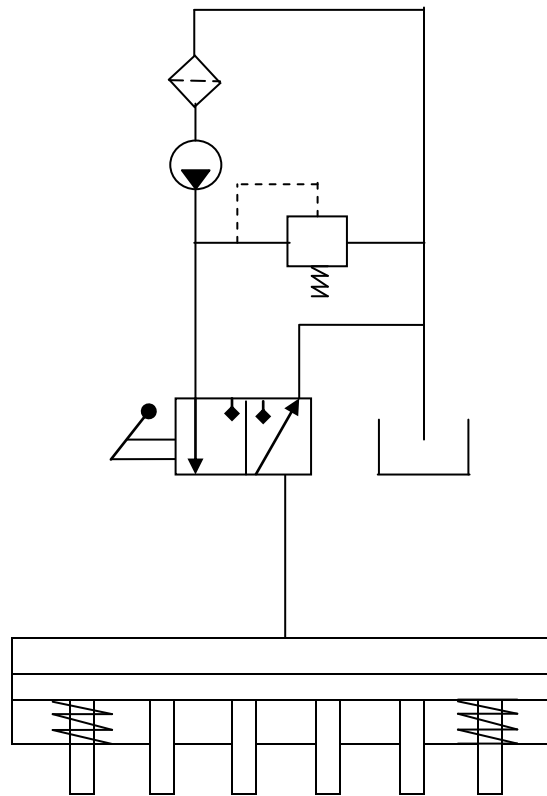


Figure 6.11: - hydraulic circuit for vibrating plate

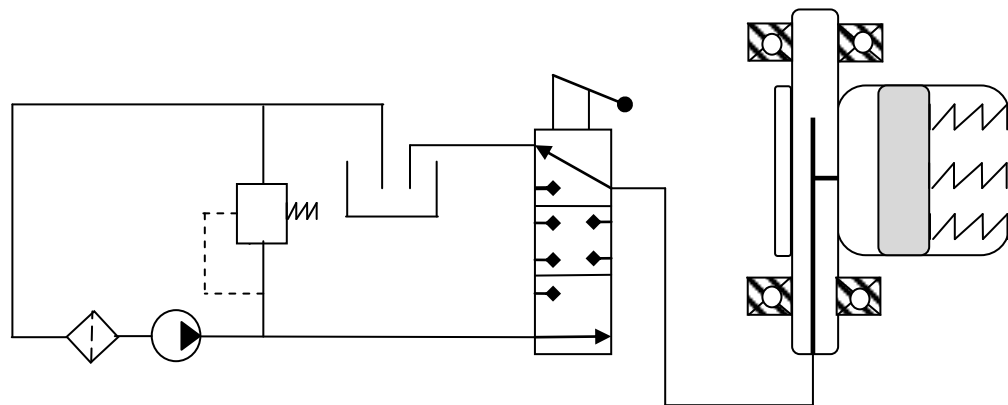


Figure 6.12: - hydraulic circuit for ERM

Pictorial Drawing

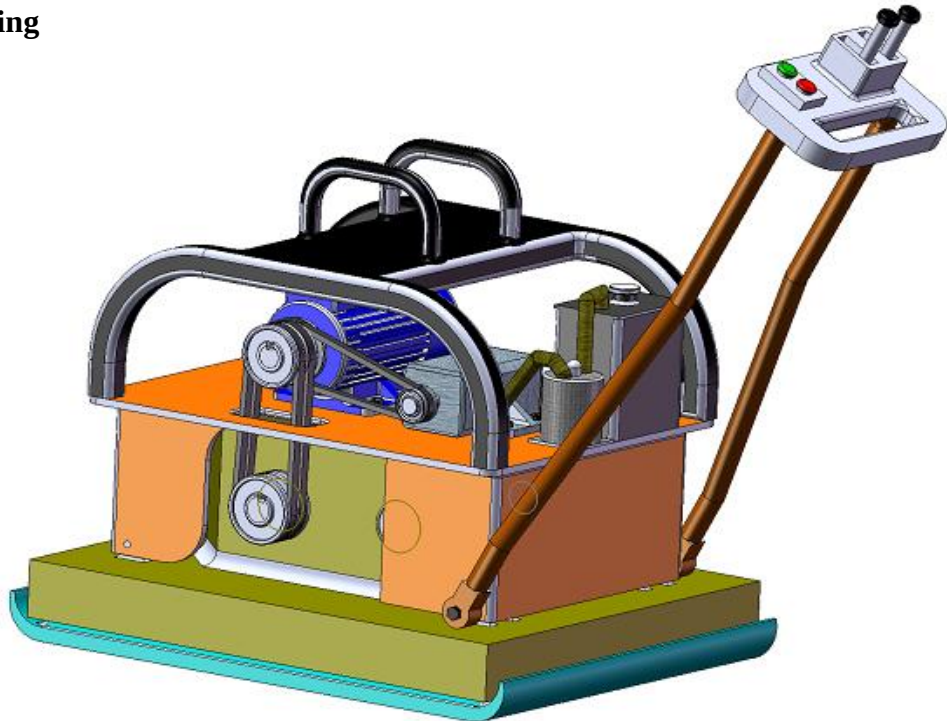


Figure 6.13:- 3D Assembled view

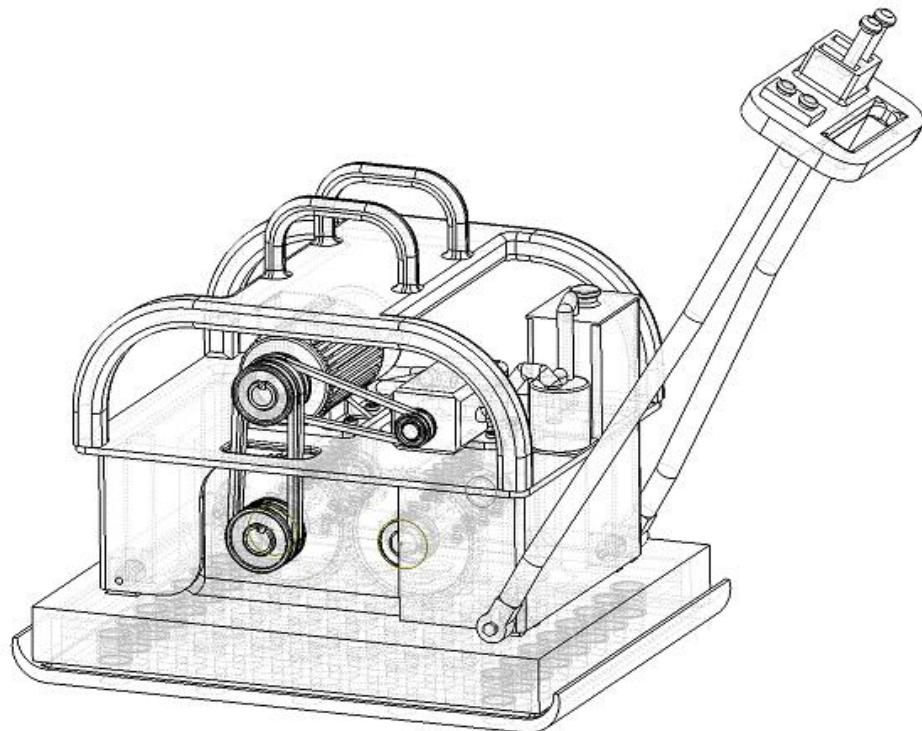
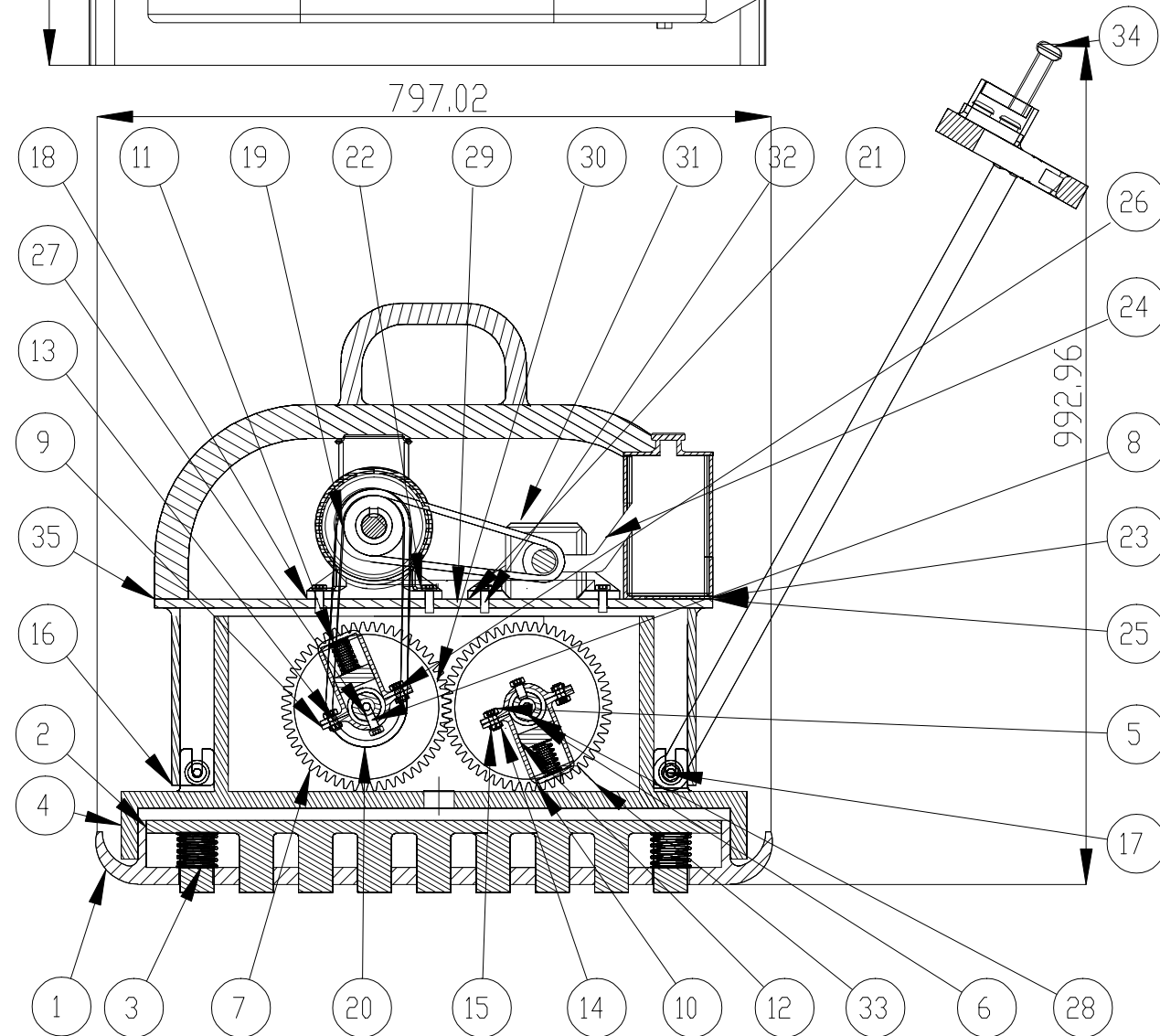
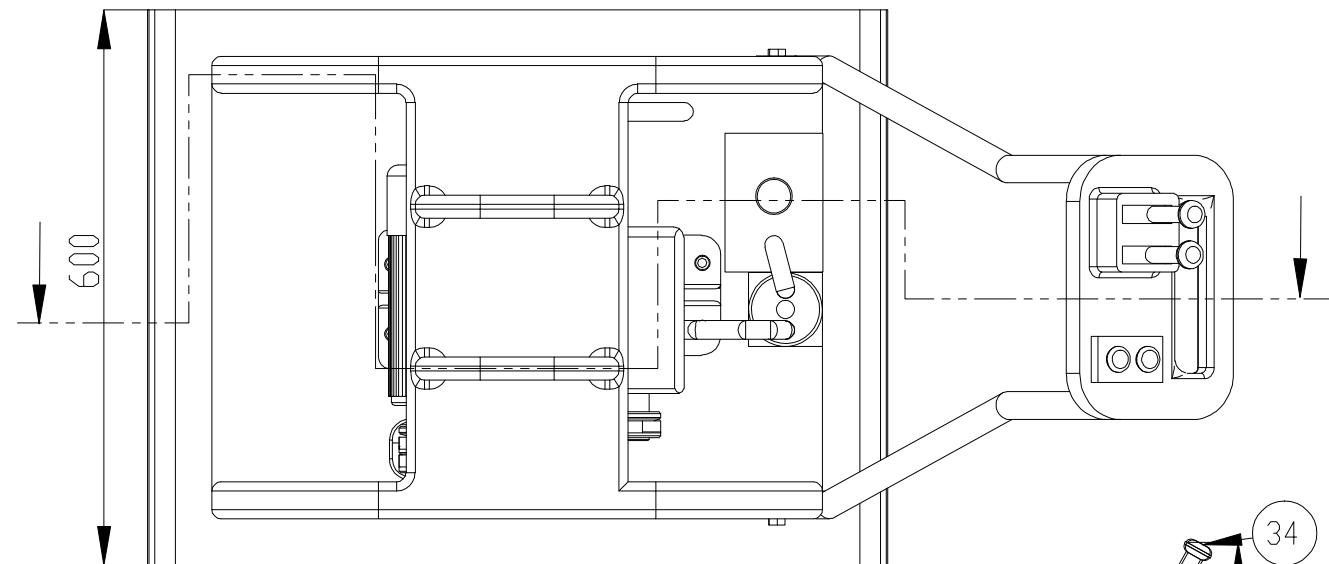


Figure 6.14:- 3D Hidden line Assembled view



ITEM NO.	PART NUMBER	QTY	material property
1	vibrating plate	1	1144 annealed steel
2	kneading	1	1018 annealed steel
3	spring vb	16	4142 steel
4	cover	1	iron sheet
5	bearing	4	
6	shaft L	1	1045 quenched steel
7	gear	2	1011 annealed cast iron
8	shaft v	1	4142 quenched steel
9	ERM mate	2	high carbon steel
10	ERM	2	gray castt iron
11	spring	12	
12	weight	2	gray cast iron
13	pulley,l	1	A1045 tempered steel
14	ALLEN KEY	28	
15	NUT	16	
16	damper	1	ruber
17	real damper	4	ruber
18	motor	1	
19	pulley motor	1	1011 tempered steel
20	belt	2	
21	pump	1	
22	beltpum	1	
23	filter	1	
24	hose	1	flexible rubber hose
25	tanker	1	low carbon steel
26	hose2	1	flexible rubber hose
27	conL	1	ABS cast iron
28	conS	1	ABS cast iron
29	conF	2	cast iron
30	connector	1	low carbon steel
31	hose 5	1	flexible rubber hose
32	hose3	2	flexible rubber hose
33	hose4	1	flexible rubber hose
34	handel	1	low carbon steel
35	cover handel	1	low carbon steel

NAME	SIGNATURE	DATE	TITLE:
DRAWN Anteneh Tilahun			Detail drawing of variable speed vibrating plate compactor
CHK'D			
APPV'D			
			DWG NO. 1
			A3
			SCALE:1:1
			SHEET 1 OF 1