DEVELOPMENT AND PERFORMANCE EVALUATION OF A DRIED GRAIN PACKING MACHINE

BY

ADEYEMO Adeniyi Gbenro

MEng/SEET/2017/6985

DEPARTMENT OF AGRICULTURAL AND BIORESOURCES ENGINEERING,

FEDERAL UNIVERSITY OF TECHNOLOGY, MINNA

NOVEMBER, 2021

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THESIS SUMITTED TO THE POSTGRADUATE SCHOOL, FEDERAL UNIVERSITY OF TECHNOLOGY, MINNA.NIGERIA, IN PARTIAL FIULFILLMENT OF THE REQUIREMENTS FOR THE AWARD OF THE DEGREE OF MASTER OF ENGINEERING (MENG) IN AGRICULTURAL AND BIORESOURCES ENGINEERING.

NOVEMBER, 2021

ABSTRACT

Efficient post-harvest handling of grains can tremendously contribute to socio-economic empowerment in developing nations. The challenges usually encountered during post-harvesting operations have limited the large scale farming activities especially in rural areas. The manual method being employed to dry and pack grains is so tedious that it discourages productivity.

Sometimes, it introduces impurities to the grain, causes grain damages and even may reduce the quality of grain. As a result, a mechanical packer was designed and fabricated. Research on mechanization of large scale production of cereal grains, however, is very low especially in the rural areas. The main objective of the study is to develop and evaluate the performance of a grain packing machine. The packing machine consists of frame, cyclone, cyclone air discharge, conveying pipe, blower, suction inlet, bagging section and petrol engine. The diameter of the suction pipe was determined to be 211 mm while the total pressure in the pipe to overcome the friction been generated in the pipe was 653.4 Pa, the volume of grains that was flowing from the blower curve according to the pressure drop measured at the inlet of the pipe was $1.15 m^3/s$. The performance test of the machine was carried out, the grains that were used to test the machine include Rice, Millet and Guinea corn which were spread evenly on the concrete floor. Grain thicknesses of 2.5, 2.0 and 1.5 cm by using calibrated scale for various thicknesses on the smooth ground, operator speeds of 2.0, 1.5 and 0.7 m/s were chosen because the average human working speed is 1.5 m/s and machine speeds by using different pulleys of 2880, 1440 and 720 rpm were used as independent variables and their effects were shown on dependent variables such as machine capacity, collection efficiency, percentage of broken grains and percentage of whole grains. The ANOVA results showed that the independent variables does not have a significant effect on the machine capacity while machine speed and interaction between AB has significant effect on collection efficiency. Also, operator speed (A), grain thickness (B), machine speed (C) and interaction (AB) has significant ($p<0.05$) effect on the percentage broken grain and operator speed, grain thickness and machine speed has significant effects on percentage whole grain.

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CHAPTER ONE

1.0 INTRODUCTION

1.1 Background to the Study

The traditional methods of packing dried grains are so tedious that it discourages productivity. It introduces impurities to the grain, causes grain damage (visible and internal) and also reduces the grain quality. Therefore, a study of development of a grain packing machine becomes necessary to overcome the aforementioned problems. Physical, mechanical and aerodynamic properties of grain are necessary for the design of equipment to handle and package the grains. The immediate adoption of new technologies to aid storage system was as a result of a greater demand to increase production to cope with the fast-growing population of Nigeria which was estimated to grow to about 200 million by the year 2019 (UN, 2019). Another factor for adopting the technology for packing was as a result of government encouragement for the citizen to patronise local production of grains (rice) which also lead to increased production (Adeyemo *et al.,* 2014).

The adoption of improved production technology increases yield and likewise gives birth to new challenges on how to deal or handle tons of wet grains that needs to be dried to maintain good quality, storability and high commercial value.

Drying is the process that reduces grain moisture content to a level where it is safe for storage. Drying is the most critical operation after harvesting a grain crop. Delays in drying, incomplete drying or ineffective drying reduce grain quality and result in losses. Drying and storage are related processes and can sometimes be combined in a piece of equipment (instore drying). Storage of incompletely dried grain with moisture content higher than the acceptable level leads to grain deterioration regardless of storage facility used. In addition, the longer the desired grain storage period, the lower the required grain moisture content must be (Sony *et al.,* 2013).

Packing of grain seeds from the concrete floor is the process of using either manual method or mechanical methods to pack grain seeds into the storage system after being dried for some time. A long-standing problem in managing the behaviour of a collection of solid grains concerns the nature of the grain packing, a property that is typically controlled by how the grains are poured or shaken (Chen *et al.,* 2006). Packing problems have been much studied in the past decades, in particular, to their wide range of applications in many settings of theoretical and practical interest, including packing/loading, scheduling and routing (Pergola *et al.,* 2015).

Confronted with different postharvest challenges, the government of Nigeria activated various agencies like National Cereals Research Institute (NCRI), The Nigeria Stored Product Research Institute (NSPRI), Institute of Agricultural Research (IAR) and other institutions to take steps to ease the problems. To date with all the postharvest technologies being developed and offered by the government, there are grey areas in the postharvest aspect of drying grains that should harmonize with the practice of small scale farmers as well as large scale farmers and even traders.

Several drying technologies were introduced to small scale farmers, large scale farmers and traders. The rate of return from sun drying operation is high while the rate of return from the best mechanical dryers available in the country is low. Farmers unanimously use sun drying and none adopts mechanical dryers. In the light of this development and present practices, it is obvious that sun drying will stay as one of the postharvest technologies in the Nigeria.

There is a great importance in mechanising the process of collecting the grains spread on the wide pavement and also worthy of note that the difficulty of the manual collection of grains was stressed as one of the major problems of most grains packer because of the lack of technology that can be used for that project and the speed they require. This is important when packing up the grains when the rain is about to start, it will take more than an hour or more when manually collecting the grains depending on the size of the field. The larger the drying field, the more man power you need to quickly collect the grains since the process of collecting is only by sweeping the grain.

Angeles, (1987) stated that the inappropriateness of imported technologies over the country's socio-economic conditions had created awareness of developing our own equipment and machine out of local materials using locally manufacturing technologies and manpower. Owing to significant development of sun drying as a socially accepted technology and its possibility of development through mechanization, he also added that continuous efforts have to be undertaken to conduct development studies of local machinery based on the appropriate features of existing commercial machinery from developed countries and emerging economies. It is for this reason that this research was undertaken to develop and fabricate a dried grain packing machine out of local materials using locally manufacturing technology and man power that would help farmers and traders to contribute to the reduction of losses, save time, labour, and cost of collecting and bagging.

1.2 Statement of the Research Problem

The major factor limiting the production of grain seeds is the post-harvest processing problems particularly seed drying and storage. There is a great importance in mechanizing the process of packing grains spread for drying. It is worthy of note that the difficulty of the manual collection of grains is one of the major problems farmers faced because of the lack of efficient technology that can be used.

1.3 Aim and Objectives of the Study

The aim of this research is to develop a dried grain packing machine while the objectives of this project are;

i to determine the air velocity, pressure drop, and power requirement in relation to pneumatic conveying characteristics

ii to design and fabricate a dried grain packing machine

iii to carry out performance evaluation of the dried grain packing machine

1.4 Justification of the Study

The essence of the study is encapsulated in the review of the past and current efforts made in Nigeria to achieve national development through the application of technology. When a large quantity of grains is required for consumption and for industrial purposes, they should be packed mechanically. Traditional and mechanical methods used presently are not encouraging. The process takes much time and at the same time, the output using these methods are low in quantity and even sometimes reduces the quality. About 1000kg of grains spread on concrete floor would take almost one day hour of work to pack while the motorised one will not be more than 30 to 40 minutes. Therefore, there is the need to develop a machine for packing grains.

1.5 Scope of the Study

The scope of the study is to design, fabricate, test and carry out performance evaluation of dried grain packing machine. Using statistical analysis to determine the effect of the independent variables (operator speed, grain thickness and machine speed) on the dependent variables (machine capacity, grain collecting efficiency, percentage of damaged grains and percentage of whole grains).

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1 The Concept of Conveyance of Agricultural Produce

Conveyance is the movement of grains from one stage to another. It is a post harvesting process. Grains are removed from their pods and dried. The quality of seeds produced is a factor of the soil type, seeds variety and the cultivation care applied before harvesting (Agatha, 2013). Various local methods have been developed using available materials. In some areas, storage is restricted to the amount that can be dried on a heat supply similar to that available from a kitchen fire (Sureguard, 2010).

A conveyor system is a common piece of mechanical handling equipment that moves materials from one location to another. Conveyors are especially useful in applications involving the transportation of heavy or bulky materials. Conveyor systems allow quick and efficient transportation for a wide variety of materials, which make them very popular in the material handling and packaging industries. They also have popular consumer applications, as they are often found in supermarkets, airports, constituting the final leg of item/bag delivery to customers. Many kinds of conveying systems are available and are used according to the various needs of different industries. There are chain conveyors (floor and overhead) as well. Chain conveyors consist of enclosed tracks, I-Beam, towline, power and free, and hand pushed trolleys (Malek *et al.*, 2015).

Conveyor systems are used widespread across a range of industries due to the numerous benefits they provide (Michael, 2012). Conveyors are able to safely transport materials from one level to another, which when done by human labour would be strenuous and expensive. They can be installed almost anywhere, and are much safer than using a forklift or other machine to move materials. They can move loads of all shapes, sizes and weights. Also, many have advanced safety features that help prevent accidents. There are a variety of options available for running conveying systems, including the hydraulic, mechanical and fully automated systems, which are equipped to fit individual needs. Conveyor systems are commonly used in many industries, including the Mining, automotive, agriculture, computer, electronic, food processing, aerospace, pharmaceutical, chemical, bottling and canning, print finishing and packaging. Although a wide variety of materials can be conveyed, some of the most common include food items such as beans and nuts, bottles and cans, automotive components, scrap metal, pills and powders, wood and furniture and grain and animal feed. Many factors are important in the accurate selection of a conveyor system. It is important to know how the conveyor system will be used beforehand. Some individual areas that are helpful to consider are the required conveyor operations, such as transportation, accumulation and sorting, the material sizes, weights and shapes and where the loading and pickup points need to be (Malek *et al.,* 2015).

An operation is composed of processes designed to add value by transforming inputs into outputs (Amadi and Pellissier, 2013). Seed processing is a vital part of the seed production needed to move the improved genetic materials of the plant breeder into commercial channels for feeding the rapidly expanding world population. The farmer must get the quality seed that is free from all undesired materials because farmer's entire crop depends on it. It is easier, safer, faster, more efficient and cheaper to transport materials from one processing stage to another with the aid of material handling equipment devoid of manual handling. Handling of materials which is an important factor in manufacturing is an integral part of facilities design and the efficiency of material handling equipment add to the performance level of a firm. Conveyor systems are durable and reliable in materials transportation and warehousing. Based on different principles of operation, there are different conveyor systems namely: gravity, belt, screw, bucket, vibrating, pneumatic/hydraulic, chain, spiral, grain conveyor systems etc. The choice however depends on the volume to be transported, speed of transportation, size and weight of materials to be transported, height or distance of transportation, nature of material, method of production employed. Material handling equipment ranges from those that are operated manually to semi-automatic systems and to the ones with high degree of automation. The degree of automation however depends on handling requirements.

Material handling involves movement of material in a manufacturing section. It includes loading, moving and unloading of materials from one stage of manufacturing process to another. A belt conveyor consists of an endless and flexible belt of high strength with two end pulleys (driver and driven) at fixed positions supported by rollers.

2.2 Pneumatic Conveying Systems

A pneumatic conveying system is a process by which bulk materials of almost any type are transferred or injected using a gas flow as the conveying medium from one or more sources to one or more destinations. Air is the most commonly used gas, but may not be selected for use with reactive materials and/or where there is a threat of dust explosions (Bhatia, 2015).

A well designed pneumatic conveying system is often a more practical and economical method of transporting materials from one point to another than alternative mechanical systems (belt conveyors, screw conveyors, vibrating conveyors, drag conveyors and other methodologies) because of three key reasons:

i. First, pneumatic systems are relatively economical to install and operate,

ii. Second, pneumatic systems are totally enclosed and if required can operate entirely without moving parts coming into contact with the conveyed material. Being enclosed these are relatively clean, more environmentally acceptable and simple to maintain,

iii. Third, they are flexible in terms of rerouting and expansion. A pneumatic system can convey a product at any place a pipe line can run.

Pneumatic conveying can be used for particles ranging from fine powders to pellets and bulk densities of 16 to 3200 kg/m^3 (1 to 200 lb/ft^3). As a general rule, pneumatic conveying will work for particles up to 2 inches in diameter at typical density. By "typical density" I mean that a 2-inch particle of a polymer resin can be moved via pneumatic conveying, but a 2-inch lead ball would not (Bhatia, 2015).

Pneumatic conveying systems are basically quite simple and are eminently suitable for the transport of powdered and granular materials in factory, site and plant situations. The system requirements are a source of compressed gas, usually air, a feed device, a conveying pipeline and a receiver to disengage the conveyed material and carrier gas. The system is totally enclosed, and if it is required, the system can operate entirely without moving parts coming into contact with the conveyed material. High, low or negative pressures can be used to convey materials. For hygroscopic materials, dry air can be used, and for potentially explosive materials an inert gas such as nitrogen can be employed. A particular advantage is that materials can be fed into reception vessels maintained at a high pressure if required. Plate 2.1 shows simple pneumatic conveying system.

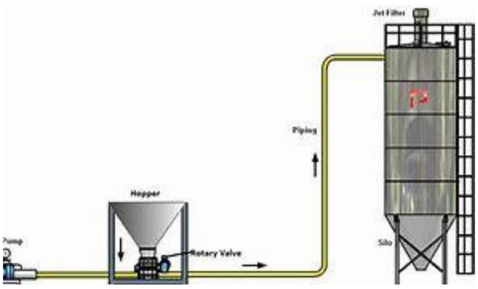

Plate 2.1: A Pneumatic Conveying System

Source: Vacuumafarin.com

The pneumatic conveying characteristics of agricultural materials such as the length, width, thickness, arithmetic mean diameter, geometric mean diameter, sphericity, volume, thousand seed mass, bulk density, true density, porosity, projected area, terminal velocity, drag coefficient, pressure drop, power requirement, and seed damage are used in handling, processing, and designing equipment (Güner, 2006; Kılıçkan and Güner, 2006). In order to optimize various factors, threshing efficiency, pneumatic conveying, and storage pertaining to grain seed, the physical properties are essential (Konak *et al.,* 2002). Aerodynamic properties of agricultural materials are used in the handling and processing of various agricultural products. Terminal velocity is one of the most important aerodynamic properties for the separation, the pneumatic transportation, and the cleaning of seed grains (Song and Litchfield, 1991). Cleaning, automatic weighing, and hulling are among many operations that require certain pneumatic handling systems for conveying grain seeds. Pneumatic cleaners are used in the cleaning process, and they also describe the action of air and shakers in combination (Tabak and Wolf, 1998). In order to design equipment for cleaning, handling, aerating, storing and processing of grain seeds, it is necessary to study their pneumatic conveying characteristics. Thus, the objective of this study was to determine the physical properties of grain seeds and air velocity, pressure drop, and power requirement in relation to pneumatic conveying characteristics.

2.2.1 Types of pneumatic conveying

There are several methods of transporting materials using pneumatic conveying. In general, they seem to fall into three main categories: dilute phase, dense phase, and air conveying (Bhatia, 2015).

i. Dilute-phase conveying is the process of pushing or pulling air-suspended materials from one location to another by maintaining a sufficient airstream velocity. Dilute phase conveying is essentially a continuous process, characterized by high velocity, low pressure and low product to air ratio,

ii. Dense-phase conveying relies on a pulse of air to force a slug of material from one location to another. Dense-phase system is essentially a batch process, characterized by low velocity, high pressure and high product to air ratio unlike dilute phase which is a low product to air ratio,

iii. Air-activated gravity conveying is a means of moving product along a conveyor on a cushion of air.

2.2.2 System flexibility

With a suitable choice and arrangement of equipment, materials can be conveyed from a concrete floor in one location to another location some distance away. Considerable flexibility in both plant layout and operation are possible, such that multiple point feeding can be made into a common line, and a single line can be discharged into a number of receiving hoppers. With vacuum systems, materials can be picked up from open storage or stockpiles, and they are ideal for clearing dust accumulations and spillages. Pipelines can run horizontally, as well as vertically up and down, and with bends in the pipeline any combination of orientations can be accommodated in a single pipeline run. Conveying materials vertically up or vertically down presents no more of a problem than conveying horizontally. Material flow rates can be controlled easily and monitored to continuously check input and output, and most systems can be arranged for complete automatic operation. Pneumatic conveying systems are particularly versatile. A very wide range of materials can be handled and they are totally enclosed by the system and pipeline. This means that potentially hazardous materials can be conveyed quite safely. There is minimal risk of dust generation and so these systems generally meet the requirements of any local Health and Safety Legislation with little or no difficulty. Pneumatic conveying plants take up little floor space and the pipeline can be easily routed up walls, across roofs or even underground to avoid existing equipment or structures. Pipe bends in the conveying line provide this flexibility, but they will add to the overall resistance of the pipeline. Bends can also add to problems of particle degradation if the conveyed material is friable, and suffer from erosive wear if the material is abrasive (Utkarsh, 2018).

2.3 Mode of Conveying

Much confusion exists over how materials are conveyed through a pipeline and to the terminology given to the mode of flow. First it must be recognized that materials can either be conveyed in batches through a pipeline, or they can be conveyed on a continuous basis, 24h a day if necessary. In batch conveying, the material may be conveyed as a single plug if the batch size is relatively small. For continuous conveying, and batch conveying if the batch size is large, two modes of conveying are recognized. If the material is conveyed in suspension in the air through the pipeline it is referred to as dilute phase conveying. If the material is conveyed at low velocity in a non-suspension mode, through all or part of the pipeline, it is referred to as dense phase conveying (David, 2004).

2.3.1 Dilute phase

According to David (2004), almost any material can be conveyed in dilute phase, suspension flow through a pipeline, regardless of the particle size, shape or density. It is often referred to as suspension flow because the particles are held in suspension in the air as they are blown or sucked through the pipeline. A relatively high velocity is required and so power requirements can also be high but there is virtually no limit to the range of materials that can be conveyed. There will be contact between the conveyed material and the pipeline, and particularly the bends, and so due consideration must be given to the conveying of both friable and abrasive materials. With very small particles there will be few impacts but with large particles gravitational force plays a part and they will tend to 'skip' along horizontal pipelines. Many materials are naturally capable of being conveyed in dense phase flow at low velocity. These materials can also be conveyed in dilute phase if required. If a high velocity is used to convey any material such that it is conveyed in suspension in the air, then it is conveyed in dilute phase. Plate 2.2 shows a dilute phase pneumatic conveying system

Plate 2.2: A Dilute Phase Pneumatic Conveying System

Source: made-in-china.net

2.3.2 Dense phase

According to David (2004), in dense phase conveying, two modes of flow are recognized. One is moving bed flow, in which the material is conveyed in dunes on the bottom of the pipeline, or as a pulsatile moving bed, when viewed through a sight glass in a horizontal pipeline. The other mode is slug or plug type flow, in which the material is conveyed as the full-bore plugs separated by air gaps. Dense phase conveying is often referred to as non-suspension flow. Moving bed flow is only possible in a conventional conveying system if the material to be conveyed has good air retention characteristics. This type of flow is typically limited to very fine powdered materials having a mean particle size in the range of approximately 40–70m, depending upon particle size distribution and particle shape. Plug type flow is only possible in a conventional conveying system if the material has good permeability. This type of flow is typically limited to materials that are essentially mono-sized, since these allow the air to pass readily through the interstices between the particles. Pelletized materials and seeds are ideal materials for this type of flow. A dense phase pneumatic conveying system is shown in Plate 2.12.

Plate 2.3: Dense Phase Pneumatic Conveying System

Source: nol-tec.com

2.3.3 Conveying air velocity

According to Bhatia (2015), for dilute phase conveying, a relatively high conveying air velocity must be maintained. This is typically in the region of 12m/s for a fine powder, to 16m/s for a fine granular material, and beyond for larger particles and higher density materials. For dense phase conveying, air velocities can be down to 3m/s, and lower in certain circumstances. This applies to both moving bed and plug type dense phase flows. These values of air velocity are all conveying line inlet air velocity values. Air is compressible and so as the material is conveyed along the length of a pipeline the pressure will decrease and the volumetric flow rate will increase. For air, the situation can be modelled by the basic thermodynamic equation as shown in Equation 2.1.

$$\frac{V_1 P_1}{T_1} = \frac{V_2 P_2}{T_2} \qquad (2.1)$$

where p is the air pressure (kN/m^2.abs), V is the air flow rate (m^3/s), T is the air temperature (K) and subscripts 1 and 2 relate to different points along the pipeline. If the temperature can be considered to be constant along the length of the pipeline this reduces to:

$$V_1 P_1 = V_2 P_2 \qquad (2.2)$$

Thus, if the pressure is one bar gauge at the material feed point in a positive pressure conveying system, with discharge to atmospheric pressure, there will be a doubling of the air flow rate, and hence velocity in a single bore pipeline. If the conveying line inlet air velocity was 20m/s at the start of the pipeline it would be approximately 40m/s at the outlet. The velocity, therefore, in any single bore pipeline will always be a minimum at the material feed point. It should be emphasized that absolute values of both pressure and temperature must always be used in these equations. These velocity values are also superficial values, in that the presence of the particles is not taken into account in evaluating the velocity, even for dense phase conveying. This is universally accepted. Most data for these values, such as that for minimum conveying air velocity are generally determined experimentally or from operating experience. It is just too inconvenient to take the presence of the particles into account.

2.3.4 Particle velocity

In dilute phase conveying, with particles in suspension in the air, the mechanism of conveying is one of drag force. The velocity of the particles, therefore, will be lower than that of the conveying air. It is a difficult and complex process to measure particle velocity, and apart from research purposes, particle velocity is rarely measured. Once again it is generally only the velocity of the air that is ever referred to in pneumatic conveying. In a horizontal pipeline, the velocity of the particles will typically be about 80% of that of the air. This is usually expressed in terms of a slip ratio, defined in terms of the velocity of the particles divided by the velocity of the air transporting the particles, and in this case, it would be 0.8. The value depends upon the particle size, shape and density, and so the value can vary over an extremely wide range. In vertically upward flow in a pipeline a typical value of the slip ratio will be about 0.7. These values relate to steady flow conditions in pipelines remote from the point at which the material is fed into the pipeline, bends in the pipeline and other possible flow disturbances. At the point at which the material is fed into the pipeline, the material will essentially have zero velocity. The material will then be accelerated by the conveying air to its slip velocity value. This process will require a pipeline length of several metres and this distance is referred to as the acceleration length. The actual distance will depend once again on particle size, shape and density. There is a pressure drop associated with acceleration of the particles in the air stream and it has to be taken into account by some means. It is not only at the material feed point that there is an acceleration pressure drop. It is likely to occur at all bends in the pipeline. In traversing a bend, the particles will generally make impact with the bend wall and so be retarded. The slip velocity at exit from a bend will be lower than that at inlet and so the particles will have to be re-accelerated back to their steady-state value. This additional element of the pressure drop is usually incorporated in the overall loss associated with a bend (Bhatia, 2015).

2.3.5 Solids loading ratio

According to Woodcock and Mwanbe (1984), solids loading ratio, or phase density, is a useful parameter in helping to visualize the flow. It is the ratio of the mass flow rate of the material conveyed divided by the mass flow rate of the air used to convey the material. It is expressed in a dimensionless form as shown in Equation 2.3.

$$\emptyset = \frac{m_p}{3.6\, m_a} \qquad (2.3)$$

where $\emptyset$ is the solids loading ratio (dimensionless), m_p is the mass flow rate of material (tonne/h) and m_a is the mass flow rate of air (kg/s). Since the mass flow rate of the conveyed material, or particles, is usually expressed in tonne/h and the mass flow rate of the air is generally derived by calculation in kg/s, the constant of 3.6 in Equation (2.3) is required to make the term dimensionless. A particular useful feature of this parameter is that its value remains essentially constant along the length of a pipeline, unlike conveying air velocity and volumetric flow rate, which are constantly changing. For dilute phase conveying, maximum values of solids loading ratio that can be achieved are typically of the order of about 15. This value can be a little higher if the conveying distance is short, if the conveying line pressure drop is high, or if a low value of conveying air velocity can be employed. If the air pressure is low or if the pipeline is very long, then the value of solids loading ratio will be very much lower. For moving bed flows, solids loading ratios need to be a minimum of about 20 before conveying at a velocity lower than that required for dilute phase can be achieved. Solids loading ratios, however, of well over 100 are quite common. For much of the data presented in this, Design guide on materials such as cement and fine fly ash, solids loading ratios in excess of 100 are reported, whether for horizontal or vertical flow. In conveying barytes vertically up the author has achieved a solid loading ratio of about 800 with a short pipeline. Conveying at very low velocity is necessary in order to achieve very high values of solids loading ratio in moving bed flow. This is because air flow rate is directly proportional to air velocity and air flow rate is on the bottom line of Equation (2.3). For plug type flow the use of solids loading ratio is not as appropriate, for the numbers do not have the same significance. Since the materials have to be very permeable, air permeates readily through the plugs.

2.4 Recent Developments

Although pneumatic conveying systems have numerous advantages over alternative mechanical conveying systems for the transport of materials, they do have drawbacks, particularly for materials that can only be conveyed in dilute phase. Particle degradation and erosive wear of pipeline bends are particular examples. Due to the high conveying air velocity required, energy requirements are also high. In recent years there have been many developments of pneumatic conveying systems aimed at increasing their capability for conveying a wider range of materials in dense phase, and hence at low velocity. This has generally been achieved by conditioning the material at the feed point into the pipeline, or by providing a parallel line along the length of the pipeline to artificially create either permeability or air retention in the material (Bhatia, 2015).

2.5 System Types

Pneumatic conveying system types can be divided into conventional and innovatory types. In conventional system, the material to be conveyed is simply fed into the pipeline and it is blown or sucked to the discharge point. It must be realized that low velocity, dense phase, conveying in conventional pneumatic conveying systems is strictly limited to materials that have the necessary bulk properties of good air retention or good permeability. The use of high pressure air is not synonymous with dense phase conveying. It is dictated entirely by the properties of the material to be conveyed in a conventional conveying system. Probably the majority of materials that are conveyed have neither of these properties. Figure 2.1 shows diagram to illustrate the wide range of conveying systems available for conventional systems operating with a single air source.

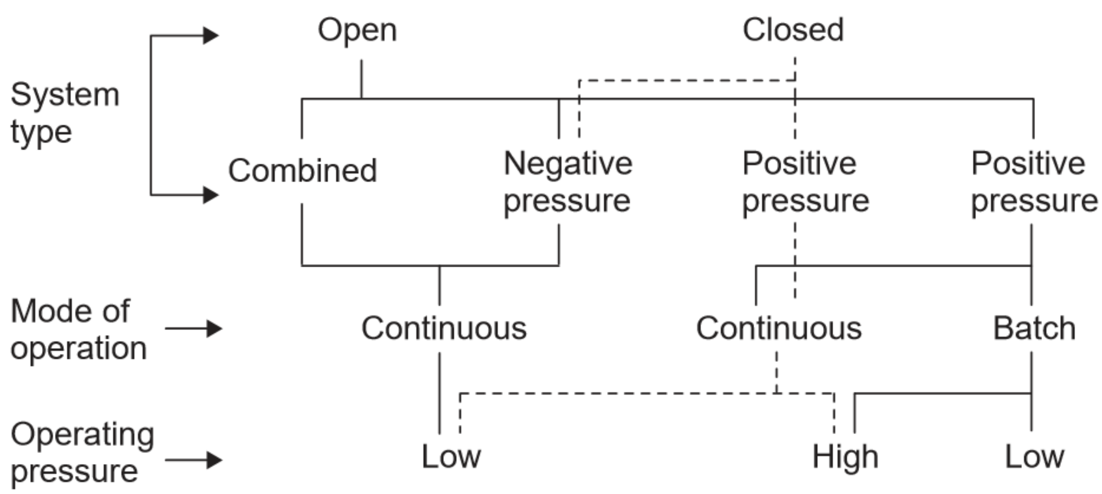

Fig 2.1: Diagram to illustrate the wide range of conveying systems available for conventional systems operating with a single air source.

Source: David Mills (2004)

There has, therefore, been much research undertaken into pneumatic conveying with a view to developing systems that are capable of conveying a much wider range of materials in dense phase and hence low velocity. Making these systems more suitable for abrasive and friable materials has provided a particular driving force (David, 2004).

2.6 System Design

This is where basic modelling for pneumatic conveying begins. Materials are added to the pipeline and the influence of the materials is considered and compared. Scaling parameters and design procedures are then introduced and these are reinforced with two case studies. Some first approximation design methods are presented to allow feasibility studies and system checks to be undertaken quickly, and the possibilities of multiple materials and multiple distance conveying are considered (David, 2004).

2.7 Conveying Characteristics

Conveying characteristics for a material provide a valuable aid to system design. They provide the design data in terms of air flow rate and air supply pressure for a given material flow rate and quantify the effect of pipeline bore and conveying distance. In addition, the conveying characteristics identify the minimum conveying conditions and provide the means to determine power requirements, thus enabling comparisons to be made for different conveying systems. Conveying characteristics are presented for representative materials and, in addition to total pipelines, data is also presented for individual sections of pipeline, as well as bends (David, 2004).

2.8 Conveying Capability

It has already been mentioned that pneumatic conveying systems are capable of conveying almost any material. Distance, however, does impose a practical limit. Although hydraulic conveying systems are capable of conveying material at a flow rate in excess of 100 tonne/h, over a distance of 100 km, or more in a single stage, the limit for pneumatic conveying is typically about $1\frac{1}{2}$ km for most applications. With water having a density that is about 800 times greater than that of air, at free air conditions, the difference in density between the conveyed material and that of the conveying fluid is widely different. As a consequence, conveying air velocities are a factor of about ten times greater than those required for water in order to convey material in suspension (David, 2004).

2.9 Flow Rate Capability

The capability of a pneumatic conveying system, in terms of achieving a given material flow rate, depends essentially on the conveying line pressure drop available and the diameter of the pipeline. As mentioned above, the use of pressure is generally limited in the majority of applications to about 5 bar and so pipeline bore is increased to achieve an increase in material flow rate if this is required. In many cases, pressure capability is set by the desire to use a particular type of compressor or blower. In most cases the duty of conveying a given flow rate of material can be met by a wide range of combinations of pressure drop and pipeline bore. There is rarely a single solution to the design of any pneumatic conveying system. Where there is a choice it is well worthwhile comparing the systems in terms of operating cost as well as capital cost. Only if a very high material flow rate is required will the options be limited (David, 2004). Litigant (2010) reported on a pneumatic system for off-loading cement from bulk carriers at 800 tonne/h, and its onward conveying to silos 500 m distant through twin pipelines. Castle Cement had a need to import up to one million tonne/year of cement at a terminal 20 km east of London on the River Thames. As the river is tidal (7m) it was necessary to build a jetty in the river against which the ships could berth, and hence the long conveying distance. A single vacuum nozzle was employed to off-load at 800 tonne/h, but it was decided to use two pipelines at 400tonne/h each for the transfer to the silos, as it was considered that a single bore pipeline would be more expensive to build. It was estimated that the power required for conveying the cement at 800tonne/h to the silos was 2400 kW (David, 2004).

2.10 Material Influences

It has already been mentioned that different materials have different conveying capabilities in terms of the minimum value of conveying air velocity required, and hence air flow rate. Different materials can also achieve very different mass flow rates when conveyed through the same pipeline under identical conveying conditions. And it is not just different materials! Different grades of exactly the same material can exhibit totally different performances. Thus, a conveying system designed for one material may be totally unsuitable for another (David, 2004).

2.11 Suction Nozzles

According to David (2004), a specific application of vacuum conveying systems is the pneumatic conveying of bulk particulate materials from open storage and stockpiles, where the top surface of the material is accessible. Vacuum systems can be used most effectively for the offloading of ships and for the transfer of materials from open piles to storage hoppers. They are particularly useful for cleaning processes such as the removal of material spillages and dust accumulations. In this role, they are very similar to the domestic vacuum cleaner. For industrial applications with powdered and granular materials, however, the suction nozzles are rather more complex. It is essential with suction nozzles to avoid filling the inlet tube solidly with material, and to maintain an adequate flow of air through the conveying line at all times. To avoid blocking the inlet pipe, sufficient air must be available at the material feed point, even if the suction nozzle is buried deep into the bulk solid material. Indeed, the vacuum off-loading system must be able to operate continuously with the nozzle buried in the material in order to maximise the material flow rate. Sufficient air must also be available for conveying the material through the pipeline once it is drawn into the inlet pipe. In order to obtain maximum output through a vacuum line it is necessary to maintain as uniform a feed to the line as possible with the absolute minimum of pulsations. To satisfy these requirements two air inlets are generally required, one at the material pick-up point and another at a point downstream. A sketch of a typical suction nozzle for vacuum pick-up systems is shown in Figure 2.2.

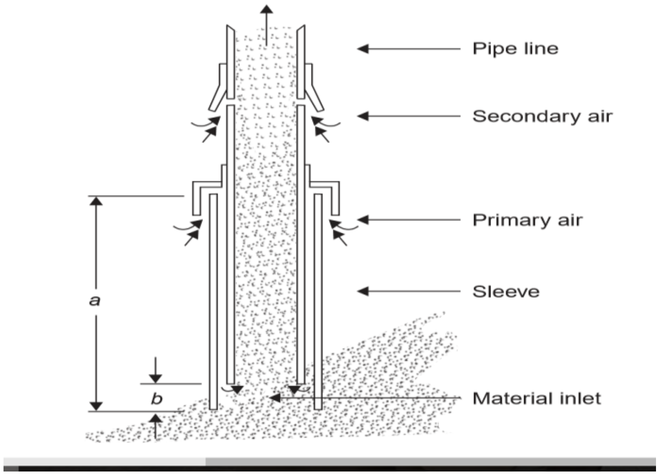

Fig 2.2: Suction Nozzle for Vacuum Pick-up Systems

Source: David (2004)

2.12 Pipelines

Decisions do have to be made with regard to the pipeline. Material, wall thickness, surface finish, steps and bends to be used, all have to be given due consideration. One of the most critical parameters with regard to the successful operation of a pneumatic conveying system is maintaining a minimum value of conveying air velocity for the material to be handled. For the dilute phase conveying of granulated sugar, for example, this is about 16 m/s. If the velocity drops to 15 m/s the pipeline is likely to block (David, 2004).

2.13 Hoses

Where flexibility is required in a pipeline, and this cannot be conveniently achieved with a combination of straight pipe and bends, flexible hose can be used. Where a single line needs to feed into a number of alternative lines, and a flow diverter is not wanted to be used, a section of flexible hose of the steel braided type can be used to provide the link.

Where road and rail vehicles and boats need to be off-loaded, flexible rubber hose is ideal. It is available in natural rubber and a variety of synthetic materials come in a wide range of sizes. The author has conveyed various drilling mud powders through hoses at pressures of up to 6bar gauge to obtain data for transferring these materials from boats to oil rig platforms in the North Sea. Flexibility is generally needed in ship off-loading applications with vacuum systems, and hoses provide the necessary flexibility here. Care must be taken if the material is abrasive and has a large particle size, because the wear rate of rubbers can be excessive with such materials (David, 2004).

2.14 Erosive Wear

If an abrasive material is to be conveyed in a pipeline, consideration must be given to the use of schedule 80 pipeline or higher. For very abrasive materials conventional mild steel pipeline is unlikely to be suitable, and spun alloy cast iron pipeline would be preferred. An alternative to this, which is commonly adopted, is to line a conventional steel pipeline with basalt. If a more wear resistant material is required, then alumina ceramics can be used, but this is likely to be very much more expensive. A usual combination is to line the straight pipeline with basalt and to use alumina for the bends. Erosive wear of bends tends to be more severe than straight pipeline and so a much higher degree of protection needs to be given to them (David, 2004).

2.15 Material Degradation

Friable materials need to be conveyed 'gently' and this is best achieved by controlling the conveying conditions. In terms of pipeline influences most of the problems of material degradation occur at the bends in the pipeline. It is the deceleration of particles on impact with bends that causes much of the damage. Decelerating forces are significantly lower with materials such as urethane and rubber, because of their resilience. It is generally a matter of compatibility with the conveyed product as to whether these materials can be incorporated into the pipeline (David, 2004).

2.16 Bends

Bends provide a pneumatic conveying pipeline with considerable flexibility in routing, but are the cause of many problems. Each bend will add to the overall resistance of the pipeline, and hence to the conveying air pressure required. Figure 2.3 shows Some special bends developed for pneumatic conveying systems; (a) the blinds tee (b) the booth bend, (c) the portico ell, (d) the flow bow, (e) the expanded bend and (f) the gamma bend.

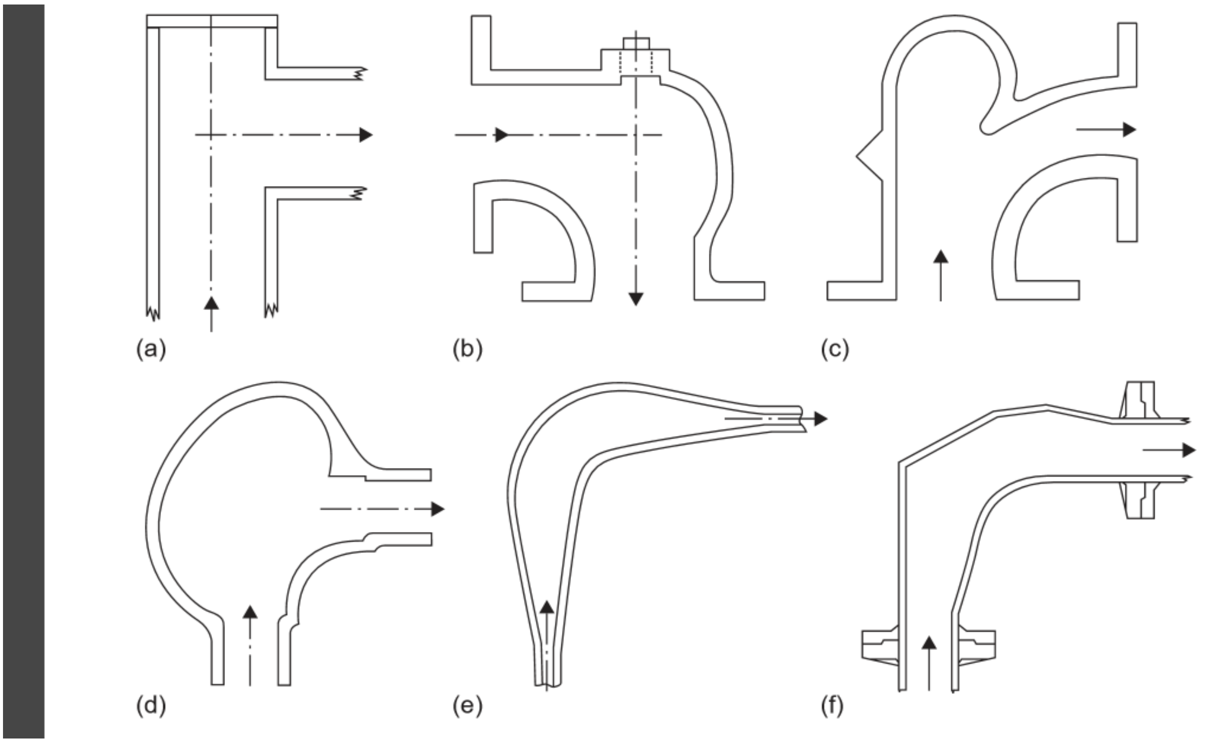

Fig 2.3: Some special bends developed for pneumatic conveying systems; (a) the blinds tee (b) the booth bend, (c) the portico ell, (d) the flow bow, (e) the expanded bend and (f) the gamma bend.

Source: David (2004)

If the conveyed material is abrasive an ordinary steel bend could fail within 2h. An abrupt change in direction will add to the problem of fines generation with friable materials, and angel hairs will be generated in long radius bends with many synthetic materials. Numerous different bends are available, to minimize each of the above problems. Many of these are made of, or lined with, basalt, cast iron, rubber, etc, and some have a constant bore and a constant radius, as with conventional bends. Another group of bends that have been developed, specifically for pneumatic conveying system pipelines, have neither constant bore nor constant radius. Some of these bends are shown in Figure 2.6. Care must be taken in selecting such bends, for account must be taken of their suitability for the material being conveyed and the pressure drop across the bend with that material (David, 2004).

2.17 System Considerations

Being at the end of the conveying process, its importance is often overlooked, but incorrect design and specification can cause endless problems in the conveying system. It is also important that the separation system is not considered in isolation. The influence that the system can have on the filter, and the influence that the filter can have on the system need to be considered in addition (David, 2004).

2.18 Centrifugal Fans

The airflow for a centrifugal fan is different from that of axial flow fans. For a centrifugal fan, the airflow is drawn into a rotating impeller and discharged radially from the fan blade into a housing. The resulting flow of air is perpendicular to the axial rotation or parallel to blade motion (Hartman, 1997) and the housing is used to direct the airflow to the desired location. Plate 2.4 shows different types of centrifugal fan.

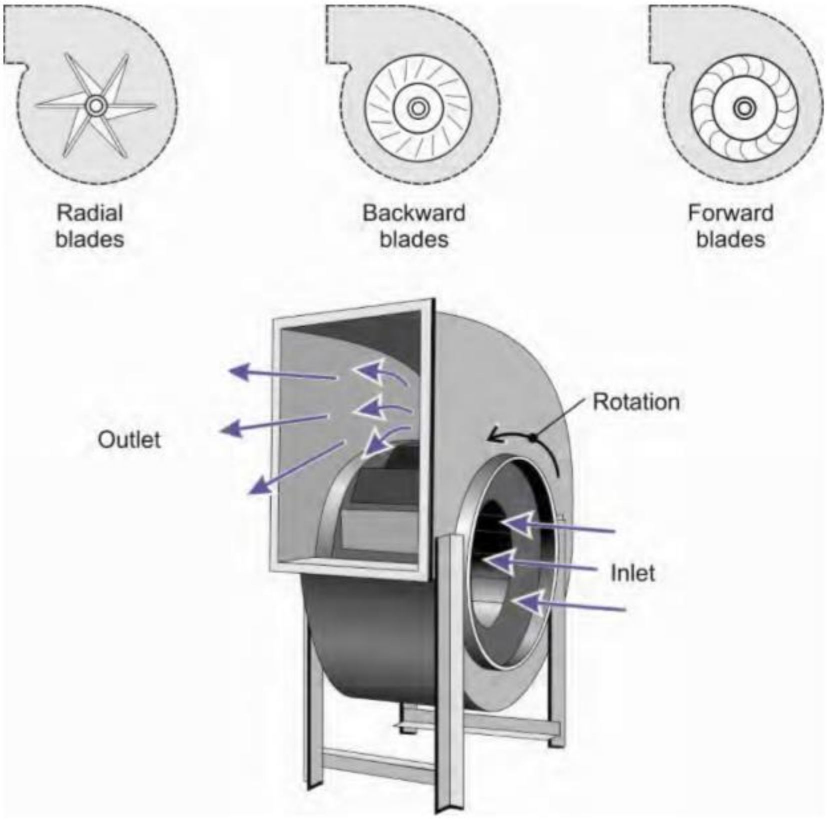

Plate 2.4: Types of Centrifugal Fan

Source: DHHS (NIOSH), 2012

There are numerous types of centrifugal fans. The flow through the fan is basically the same for all types, the difference being in the configuration of the blades. Each blade type has its advantages for different applications. Air foil type blades have the best mechanical efficiency and lowest noise level. Backward curved blades have slightly lower efficiencies compared to air foil blades. These blades are better suited to handle contaminated air because they are single thickness and can be made of heavier material that can resist the effects to fan blades by the contaminated air. Backward inclined blades have lower structural strength and efficiencies. They are easier to produce due to the elimination of the blade curvature (Agarwal, 2011).

Radial tip blades are curved at the tips. These types are used mainly in large diameters (30 to 60 inches) under severe conditions of high temperatures with minimal air contamination (Bleier, 1998).

Forward curved blades produce airflow rates higher than other centrifugal fans of the same size and speed. This allows for the fan to be more compact than other types of centrifugal fans. These fans are often used in furnaces, air conditioners, and electronic equipment cooling. Radial blades are rugged and self-cleaning, but have low efficiencies. They are suited for airflows containing corrosive fumes and abrasive material from grinding operations (Agarwal, 2011).

2.19 Bagging

According to Cecala *et al.,* (2019), to address problems associated with the bagging process, a number of different dust sources need to be addressed and controlled, specifically product blowback, product "rooster tail," and contaminated bags. When manually bagging and stacking 50 to 100 bags, these dust sources directly affect the worker's exposure. Two different types of bags that are used to transport product within this weight range are open-top bags and closed bags with an internal valve.

For valve-type bags, three major dust sources need to be addressed for effective dust control. The first dust source is from product blowback, which occurs during bag filling and results from product spewing out of the bag valve. Product blowback occurs as excess pressure builds inside the bag during bag filling and is then relieved by air and product flowing out of the bag valve around the fill nozzle. The second major dust source is product spewing from the fill nozzle and bag valve as the bag is ejected from the filling machine. Plate 2.5 shows the exhaust ventilation system used during loading of flexible intermediate bulk containers

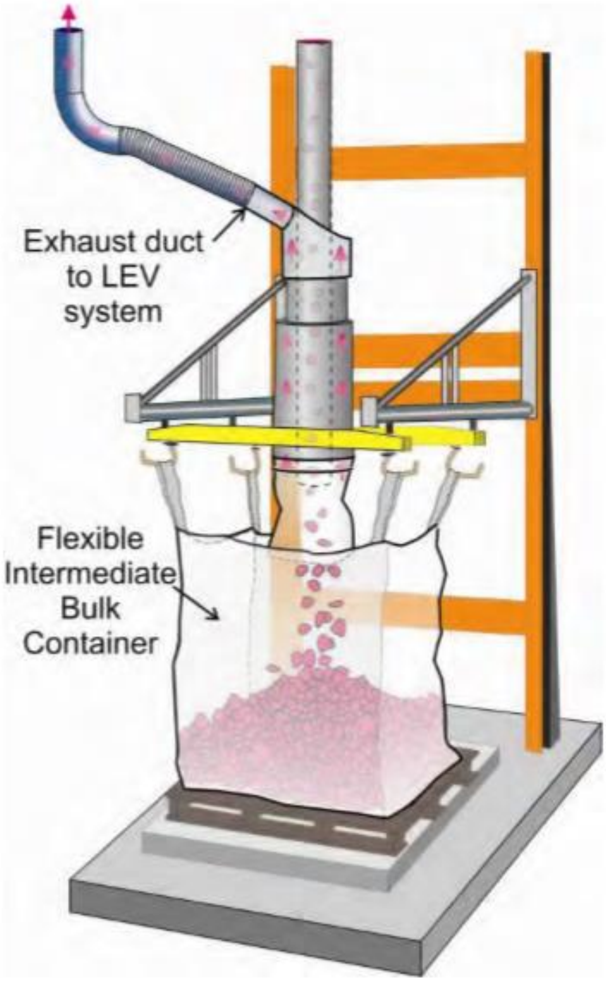

Plate 2.5: Exhaust Ventilation System used during Loading of Flexible Intermediate Bulk Containers

Source: DHHS (NIOSH), 2012

Both release dust into the air and contaminate the outside surface of the bag. The contaminated bags then become the third significant source of dust exposure for the bag stackers, or for any other individuals handling the bags, including the end user of the product. Product blowback during bag filling, the product "rooster tail" as the bag is ejected from the fill nozzle, and dust contamination on the outside of the bag after loading is completed. If bags are undersized, they create a greater amount of product blowback and "rooster tail" during the bagging process, and this needs to be considered when evaluating these dust sources (Agarwal, 2011).

2.20 Properties of Grains

Cereal grains are edible seeds and, as such, would eventually be released from the plant when fully mature. Grains can be divided into three groups; cereals (maize, wheat, millet and rice), pulses (beans, peas and cowpeas), and oil seeds (soyabeans, sunflower and linseed) (Ghafori *et al.*, 2011).

2.20.1 Moisture content

The moisture content of a crop is normally given on a 'wet basis' (wb) and is calculated as follows (%mc wb):

$$\frac{Weight\ of\ wet\ sample}{Weight\ of\ moisture\ on\ wet\ basis\ (wb)} \times 100 \qquad (2.5)$$

Occasionally 'dry basis' (db) moisture content is given and it is important to know which has been used. For example, if 100 kg of moist grain is dried and loses 20 kg of water, the moisture content is:

$\frac{20 \times 100}{100}$ = 20% on wet basis (wb) or

$\frac{20 \times 100}{25}$ = 25% on dry basis (db)

The physical and engineering characteristics of cereal crop (grains) is very important to optimize the design parameters of agricultural equipment used in their production, handling and storage processes. So, it is essential to determine and recognize the database of physical and engineering (aerodynamic and mechanical) properties of these agricultural products because these properties play an important role in designing and developing of specific machines and their operations such as sorting, separating and cleaning, also to determine the optimum in seed metering device in pneumatic planter and precision sowing machine to suite every size of these grains. Sitkei (1987) reported that the functioning of many types of agricultural machines (sifters, sowing machines, pneumatic transport systems) is influenced by the physical properties of the objects participating, and so in order to study a given process they must be described accurately, Also the quality of processing (in chopping and milling) may be characterized by a products mean size and mean standard deviation, or these data may be used to organize a technological process or in designing certain structural elements (mesh dimensions of sifters or dimensions of screen holes). He added that during the treatment of agricultural materials air is often used the transport medium, pneumatic transport and cleaning of various agricultural products have been known for a long time, during this process aerodynamic properties play an important role and must be known for optimum design and the operation of the equipment. The two most important aerodynamic properties of a body are drag coefficient and terminal velocity.

The geometric properties such as size and shape are one of most important physical properties considered during the separation and cleaning of agricultural grains. In theoretical calculations, agricultural seeds are assumed to be spheres or ellipse because of their irregular shapes (Mohsenin, 1980). Ahmadi and Mollazade (2009) determined the physical and mechanical properties of funnel seed as a function of moisture content. They found that there was a parabolic mathematical equation for sphericity, true density, and deformation on both seed length and width sections with changes of moisture content.

The value of aerodynamic drag coefficient, which is used for determining the aerodynamic drag force (Fd), acting upon a particle moving through air depends upon particle characteristics (mass, projected area, shape and terminal velocity) as well as the conditions of airflow. The projected areas and drag coefficient of agricultural grains changes because of irregular shape and continuous the variation of positions. In studies carried out, the projected area and drag coefficient of grains were usually determined by using the diameter of the sphere equivalent to seed (Mohsenin, 1980; Gorial and O’Callaghan, 1990.

2.20.2 Physical properties of crop seeds varieties

Grain dimensions (L, W and T), mass of thousand grain, volume, geometric diameter, arithmetic diameter, bulk and real densities, percent of sphericity and projected area. Standard Deviation, coefficient of variation, maximum, minimum and arithmetic mean for grains varieties under studies. The calculated equations according to EL–Raie *et al.*, (1996) studied the size of the three varieties of corn in terms of length (L), width (W) and thickness (T). The size was used to calculate the volume (V), geometric diameter (Dg), arithmetic diameter (Da), percent of sphericity (S), area of surface (Af), and area of transverse surface (At) of the individual seeds. The following equations will be used to calculate the values of the above-mentioned properties (Ghafori *et al.*, 2011):

$$V = \frac{\pi}{6} LWT \qquad , mm^3 \qquad 2.6$$

$$D_g = (LWT)^{1/3} \qquad , mm \qquad 2.7$$

$$D_a = \frac{(L+W+T)}{3} \qquad , mm \qquad 2.8$$

$$S = \frac{(LWT)^{1/3} \times 100}{L} \qquad , \% \qquad 2.9$$

$$A_f = \frac{\pi LW}{4} \qquad , mm^2 \qquad 2.10$$

$$A_t = \frac{\pi TW}{4} \qquad , mm^2 \qquad 2.11$$

Where,

L = length of seed, mm

T = thickness of seed, mm

W = width of seed, mm

$$\rho_b = \frac{m}{v} \qquad 2.12$$

Where,

ρ_b = bulk density of the grain, g/cm^3

V = bulk volume of the grain, cm^3

M = mass of the grain, g

2.20.3 Aerodynamic properties of crop seeds varieties

The terminal velocity of crop seeds is determined by measuring the air velocities, required to suspend a seed in a vertical air stream by using terminal velocity apparatus. Drag coefficient and Reynold's number were calculated according to equations of Hexing (1989) as follows:

$$C_d = \frac{2gF_d}{A_p \rho_a V_t^2} \qquad 2.13$$

$$N_{Re} = \frac{\rho_a V_t \sqrt{A_p}}{\mu} \qquad 2.14$$

Where,

V_t = terminal velocity, m/s

A_p = projected area of particle, m^2

μ = dynamic viscosity of the air (18×10^{-6})

ρ_a = density of air (1.28 kg/m^3)

g = gravity, m/s^2

F_d = drag force, N

C_d = drag coefficient

CHAPTER THREE

3.0 MATERIALS AND METHODS

3.1 Material Selection

The machine was constructed using the following materials:

i. Angle iron (mild steel) – 40 mm x 40 mm

ii. Steel sheet (gauge 16)

iii. Shaft steel rod

iv. Pulleys

v. Leather belt

3.2 Design Consideration

Design requirements were synthesized based on the analysis of findings in the various literatures reviewed and from patented and commercial pneumatic grain collectors. Some of the identified design requirements are the following:

i. the machine collect grains at varying thickness under sun drying condition and bag it;

ii. the machine help reduce drudgery and quicken collection and bagging of grains after sun drying; and

iii. the machine should be of intermediate technology, made from local materials, using local manufacturing technology, simple and safe to operate and maintain, functionally and structurally sound, and with minimum tooling.

In designing the grain packing machine, the basic factors considered include the choice of materials, in addition to their availability and cost which are always of primary consideration. These materials were chosen on the basis of their properties such as:

i. the availability of these materials within our locality which will reduce constructional cost and hence will make the price comparatively low

ii. making it affordable by the target customers

iii. shape, size, density and weight of grains were examined as physical characteristics

iv. drag coefficient and terminal velocity of grains were examined as aerodynamic characteristics

3.3 Design Analysis, Theories and Calculations

The process design analysis was carried out to determine the necessary design parameters for the selection of various machine parts and this was done in order to avoid failure of machine parts during the required working life of the equipment as well as to minimize cost by avoiding under or over design of parts for the fabrication of the equipment. Essential design calculations were made in order to determine and select appropriate strength and sizes of the component parts of grain collecting machine. This was done with the aid of the results of the preliminary investigation that was conducted, established and using conventional formula.

3.4 Capacity of the Machine

An assumed capacity of 1500 kg of grains was made for the machine to pack within the hours of operation per day. With this, the high labour intensive operation of packing of grains being practiced manually will be reduced thereby boosting productivity.

3.5 Design Calculations

The suction pipe which pack spread grain by suction and conveyed it to the bag in air-grain mixture from the starting point to the delivery point was designed, two sections of the pipe which were placed horizontally and vertically were considered. The vertical height 800 mm was chosen in such a way to accommodate the bag to which the grains will be packed and the horizontal pipe of 200 mm was selected based on the size of the blower.

It was found from reviewed researchers that the conveying velocity for most grains ranges from 22.86 m/s to 33 m/s. Therefore, 33 m/s of the conveying velocity of rice was selected so that the machine would be able to pack other grains such as millet and guinea corn which their conveying velocity is not up to that of rice (Steinke and Kandlikar, 2005).

Also, 0.3 solid-air loading ratio was selected because it is a continuous process that is characterized with high velocity, low pressure and low product to air ratio.

3.5.1 Determination of the Diameter of Suction Pipe

The diameter of the suction pipe was determined to know the type and size of pipe that can be used to pack grains effectively. It is a function of conveying distance, suction air velocity and pipeline bends. Suction pipe is shown in Figure 3.1.

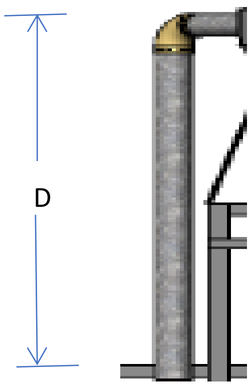

Figure 3.1: Suction Pipe

The suction pipe diameter was determined from Akhil Raj, (2017) suction pneumatic conveying system design guide as shown in Equation 3.1.

$$m = \varphi \times \rho \times A \times V \qquad (3.1)$$

Where:

φ is the solid – air loading ratio = 0.3 (for dilute phase pneumatic conveying)

ρ is the density of air = 1.2 kg/m^3

m is the required mass flow rate of grains =1500 kg/h = 0.42 kg/s

V is the velocity of air = 33 m/s

A is the area of suction pipe = $\frac{\pi}{4} \times D^2$

From Equation 3.1,

Where D is the diameter of the suction pipe

Therefore, by substitution and making D subject of the formular

$$0.42 = 0.3 \times 1.2 \times (\frac{\pi}{4}) \times D^2 \times 33$$

$$D = 0.211m$$

$$\mathrm{D} = 211mm$$

3.5.2 Determination of pipe pressure in the system

The airflow (without materials) generates friction in the pipe which must be overcome. The performance of a pneumatic conveying system in terms of achieving a given material flow rate, depends essentially on the system resistance. The effect of pressure on the pipe is very important because it can make the pipe to fail. The higher the system resistance, the higher will be the pressure drop in the system or higher will be the static pressure of the fan. The system resistance (pipe wall friction per unit area) can be estimated using the equation:

The velocity pressure is given as shown in Equation 3.2.

$$V_p = \frac{1}{2}\rho V^2 \qquad (3.2)$$

Where:

ρ is the density of air = 1.2 kg/m^3

V is the velocity of air = 33 m/s

Therefore, by substitution

$$= \frac{1}{2} \times 1.2 \times 33^2$$

$$= 653.4Pa$$

3.5.3 Determination of the size of aperture of the collector

The air velocity was calculated to know the volume of grains that is flowing from the blower curve according to the pressure drop measured at the inlet of the pipe. To determine the air velocity, the volumetric airflow rate in m^3s^{-1} of the root blower corresponding to the pressure drop established throughout the entire conveyor was determined by multiplying the pipe cross sectional area m^2 with the velocity of air as cited by Hauch, (2005); Agarwal, (2005) and Ghafori *et al.*, (2011) as shown in Equation 3.3: A collector aperture is shown in figure 3.2.

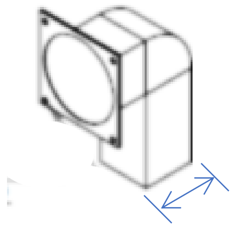

Fig. 3.2: Collector Aperture

$$Air\ discharge\ through\ the\ blower = A \times V \quad (3.3)$$

Where:

A = Cross sectional area m^2 = $\frac{\pi D^2}{4}$

V = Velocity of air is 33m/s

$$Air\ discharge = \frac{\pi D^2}{4} \times V \quad (3.4)$$

D = is the diameter of the conveying pipe = 0.211m

$$Air\ Discharge = \frac{\pi \times (0.211)^2}{4} \times 33$$

$$Air\ Discharge = 1.15m^3/s$$

3.5.4 Determination of frictional factor

The frictional factor was determined to know actual pressure loss due to the viscous nature of fluid such as air. The factor is termed fanning coefficient. The ratio of the wall stress to the flow kinetic energy per unit volume according Steinke and Kandlikar, (2005). The frictional head loss in pipes with full flow was calculated by using the formular in Equation 3.5.

$$f = \frac{0.331}{\left[\log_n\left(\left(\frac{\varepsilon}{3.7 \times D}\right) + \left(\frac{7}{N_{Re}}\right)\right)\right]^2} \tag{3.5}$$

Where:

ε is the pipe roughness factor which can be estimated as 0.00015 for smooth pipes or 0.0005 for shot-peened pipes.

D = Pipe inside diameter (m)

N_{Re} = Reynold's number

$$N_{Re} = \frac{DV_g\rho_g}{\mu_g} \tag{3.6}$$

Where:

D = Pipe inside diameter (m)

Vg = Gas velocity (m/s)

P_g = Gas density (kg/m^3)

$\mu_g$is the gas viscosity in 18.5 kg/ms at stp (Calısır *et al.*, 2005)

$$N_{Re} = \frac{0.2119 \times 33 \times 1.2}{18.5}$$

$$N_{Re} = 0.452$$

Therefore, by substitution

$$f = \frac{0.331}{\left[\log_{10}\left(\left(\frac{0.00015}{3.7 \times 0.211}\right) + \left(\frac{7}{0.452}\right)\right)\right]^2}$$

$$f = 0.23$$

3.5.5 Determination of actual pressure loss

The actual pressure loss was determined because grains are conveyed in the gas stream at a velocity that is greater than saltation and choking velocities. Friction losses as the result of the solids being in contact with the inside of the pipe are usually very small and can be neglected when considering dilute phase transport. Head losses experienced in pneumatic conveying systems are the result of the following forces.

Friction of the gas on the inside of the pipe + forces required to move the solids through the pipe + forces required to support the weight of the solid and the gasses in vertical pipe runs + forces required to accelerate the solids + friction between the solids and the inside of the pipe

The total pressure loss of the parameter system can be expressed in Equation 3.7 according to (Bhatia, 2015).

$$\Delta P_T = \Delta P_{acc} + \Delta P_g + \Delta P_s + \Delta H_g + \Delta H_s + \Delta P_{misc} \tag{3.7}$$

Where:

ΔP_T = total pressure loss in the system

ΔP_{acc} = pressure loss due to accelerate of the solids from their 'at rest' condition at the pick-up point

ΔP_g = frictional pressure loss of the gas

ΔP_s = frictional pressure loss of the solids

ΔH_g = elevation pressure loss of the gas

ΔH_s = elevation pressure loss of the solids

ΔP_{misc} = pressure loss from miscellaneous equipment

Pressure loss due to acceleration of the solids is given as (Bhatia, 2015):

$$\Delta P_{acc} = \frac{W\,V_p}{144.g} \tag{3.8}$$

Where:

W = solids mass velocity = m x v = 1.2 x 33 = 39.6 kg/sec/m^2

V_p = particle velocity = 0.8 x V_g

g = acceleration due to gravity

Grains Particles also move at a velocity lower than the gas velocity due to drag forces. The difference between these velocities is called the slip factor. For most course or hard solids, the slip factor is around 0.80. Slip factor is the difference between the gas velocity and lower velocity the particle moves caused by drag force

V_p = 0.8 x V_g = 0.8 x 33 = 26.4 m/s

Therefore, by substitution:

$$\Delta P_{acc} = \frac{39.6 \times 26.4}{144 \times 9.81}$$

$$\Delta P_{acc} = \frac{1045.44}{1412.64}$$

$$\Delta P_{acc} = 0.7401m$$

Also, frictional Pressure loss of the gas is given as (Bhatia, 2015):

$$\Delta P_g = \frac{4f.l.P_g\ V_g^2}{2g.D.144} \tag{3.9}$$

Where:

ΔP_g = frictional pressure loss of the gas

F = fanning friction factor = 0.2337 (calculated)

L = equivalent length of pipeline (H)

P_g = Gas density = 1.2 Kg/m^3

V_g = Gas velocity = 33m/s

g = acceleration due to gravity = 9.8 m/s^2

Note: Equivalent length for 90^o bend can be determined by multiplying 40 x diameter of the pipe = 40 x 0.272 = 10.88

Vertical + Horizontal length = 1000cm = 1m

Equivalent = 1 + 10.88 = 11.88m

Therefore, by substitution:

$$\Delta P_g = \frac{4 \times 0.2337 \times 11.88 \times 1.2 \times 33^2}{2 \times 9.81 \times 0.211 \times 144}$$

$$\Delta P_g = \frac{14512.56808}{596.13408}$$

$$\Delta P_g = 24.34\ m$$

Also, frictional Pressure loss of the Solid is given as (Bhatia, 2015):

$$\Delta P_s = \Delta P_g . K . R \quad (3.10)$$

Where:

ΔP_s = frictional pressure loss of the solids

ΔP_g = frictional pressure loss of the gas = 24.34m

K = friction multiplier for the solids conveyed = 0.8

R = solid to gas flow ratio

$$R = \frac{W}{V_g P_g} \quad (3.11)$$

Where:

W = solids mass velocity

M = the solid mass flow in ib/s

A = the pipe cross sectional area

P_g = Gas density (kg/m^3)

V_g = Gas velocity (m/s)

$$R = \frac{0.42}{\frac{\pi D^2 \times 33 \times 1.2}{4}}$$

$$R = 0.3032$$

Therefore, by substitution:

$$\Delta P_s = 24.34 \times 0.8 \times 0.3032$$

$$\Delta P_s = 5.9m$$

Also, elevation Pressure loss of the gas is given as (Bhatia, 2015):

$$\Delta H_g = \frac{\Delta_z . P_g . g}{144 . g_i} \quad (3.12)$$

Where:

ΔH_g = Elevation pressure loss of the gas

Δ_z = Elevation change in pipe line

P_g = Gas density

g = acceleration due to gravity

g_i = Constant (32.174 ft ib/ibs^2) = 9.8

Therefore, by substitution:

$$\Delta H_g = \frac{1 \times 1.2 \times 9.8}{144 \times 9.8}$$

$$\Delta H_g = \frac{11.760}{1411.2}$$

$$\Delta H_g = 0.0083m$$

Also, elevation Pressure loss of the solid is given as (Bhatia, 2015):

$$\Delta H_s = \frac{\Delta_z . W . g}{144 . V_p . g_i} \quad (3.13)$$

Where:

ΔH_s = elevation pressure loss of the solids

Δ_z = Elevation change in pipe line

W = Solid mass velocity = 39.6 kg/s/m^2

g = acceleration due to gravity 9.8 m/s^2

V_p = particle velocity = 26.4 m/s

g_i = constant

Therefore, by substitution:

$$\Delta H_s = \frac{1 \times 39.6 \times 9.8}{144 \times 26.4 \times 9.8}$$

$$\Delta H_s = \frac{388.08}{37255.68}$$

$$\Delta H_s = 0.01042\ m$$

Finally, total pressure loss:

$$\Delta P_T = \ 0.7401 + 24.34 + 5.9 + 0.0083 + 0.01042$$

$$\Delta P_T = 30.999\ m$$

3.5.6 Determination of power required

The choice of power source and vacuum blower was based on the ability of the blower to provide adequate suction and discharge pressures to overcome the pressure losses (air friction losses, losses due to acceleration of the grain, lift of the grain and the grain flow) in the system (Srivastava *et al.*, 2006). Delivery pressure and volumetric flow rate are the two main factors that influence the power requirement of a blower. Power delivered at the output of the blower is the product of density of solid material conveyed, volumetric rate of the material movement, acceleration due to gravity and total head of mixture. The power required to ascertain the volumetric discharge and drives the materials is presented in Equation 3.14 (Agarwal, 2005).

$$P_{out} = \rho \times Q \times g \times H \qquad (3.14)$$

Where:

P_{out} is the power required, kW

ρ is the material density, kg/m^3

Q is the volumetric discharge, m^3/s

H is the total head losses, m

But

$$Q = \frac{M}{\rho}$$

Where:

Mass flow rate, M = 0.42 kg/h

Grain materials density, ρ = 1.2kg/m^3

Therefore, by substitution:

$$Q = \frac{0.42}{1.2}$$

$$Q = 0.35\, m^3/s$$

Thus, power output is

$$P_{out} = 1.2 \times 0.35 \times 9.8 \times 30.999$$

$$P_{out} = 127.59W$$

$$P_{out} = 0.1276KW$$

Considering factor of safety, 1.5 was considered suitable for this design, the safe power output is:

$$P_{out} = 0.1276 \times 1.5$$

$P_{out} = 0.1914KW$

Power to be supplied at the input of the blower will be the ratio of the output power to the efficiency of the blower. 60% blower efficiency was elected to ensure optimum performance of the blower (Ghafori *et al.*, 2011).

Input power is therefore related to the output power as presented in equation 3.15:

$$P_{in} = \frac{P_{out}}{Blower\ efficiency} \tag{3.15}$$

$$P_{in} = \frac{0.1914}{0.6}$$

$$P_{in} = 0.3KW$$

3.5.7 Calculation of bending moment on shaft

The maximum bending moment of the wheel shaft is essential in determining its minimum required diameter to withstand twisting and bending. The size of the bending moment depends on the amount of the load it can carry and the distance it acts upon. The bending moment was calculated from the reaction of forces acting on the impeller shaft as shown in the Figure 3.3.

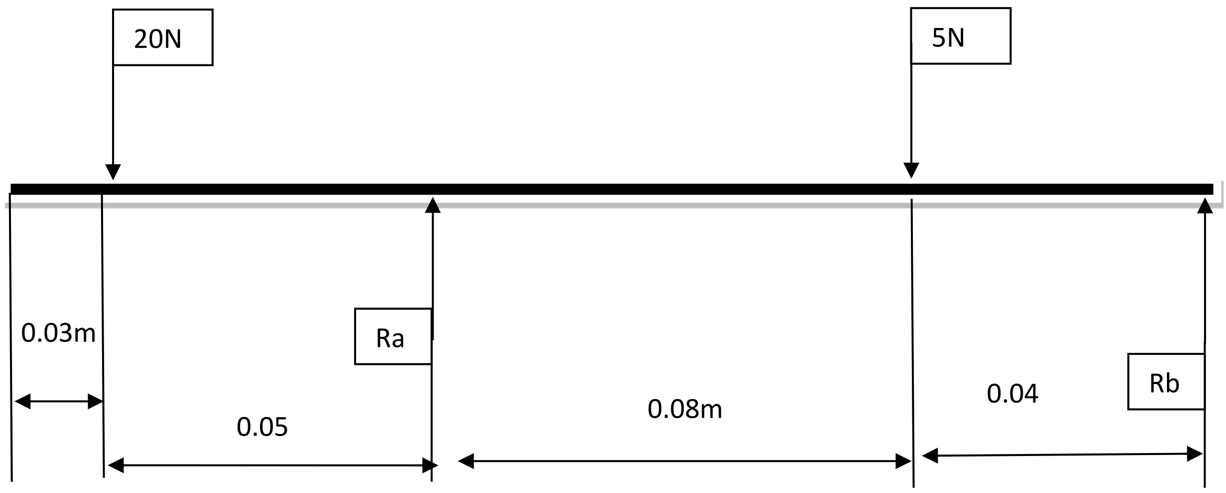

Figure 3.3: Free Body Diagram of the Shaft

$$\Sigma M_B = -20(0.17) + Ra(0.12) - 5(0.04) = 0 \quad (3.16)$$

$$0.12Ra = 3.6$$

$$R_a = 30N$$

Using the Sum of forces to calculate the reaction R_b

$$R_b = -20+30-5 = 5N \quad (3.17)$$

Also, the shear forces was calculated from the diagram, as shown in figure 3.4. From the diagram, the maximum force was calculated to be 10N

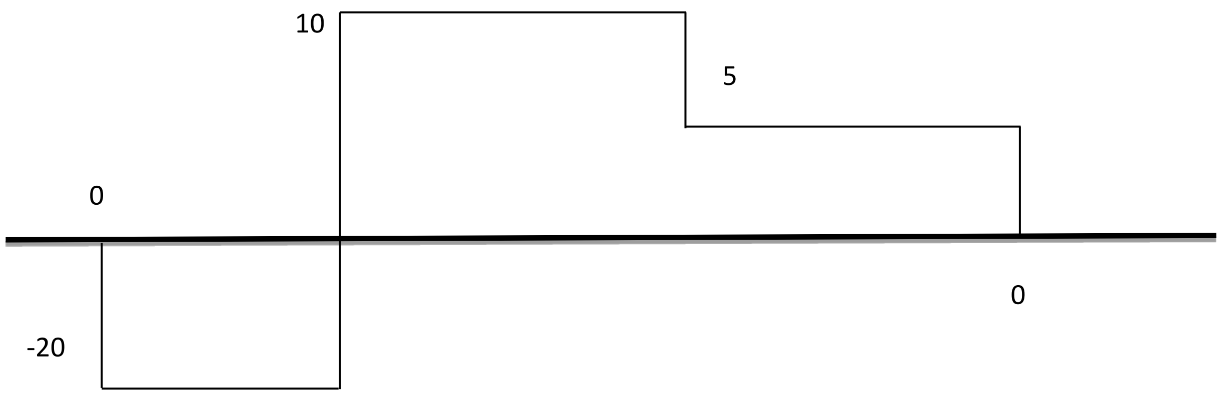

Figure 3.4: Shear Force Diagram

For bending moments

At (-20N), Bm = -20 x 0.05= 1 Nm

At (10N), Bm = 1 + 10x 0.04 = 1.8 Nm

At (5N), Bm = 1.8-5 x 0.04 = 1.6 Nm

The maximum bending moment is 1Nm.

The maximum bending moment was calculated from Figure 3.4 to be 1.8 Nm

The diameter of the impeller shaft was calculated from the relations.

But, maximum permissible load on the wheels as stipulated in ASAE (2005) is 90.7 kg. This turns the shafts about the radius of the wheel (R).

For the purpose of design, maximum load was taken to be 25 N.

Load on one shaft = 20N + 5N (pulley).

Also, the torque on the shaft is essential in estimating the power that is transmitted to the blower shaft and in determining the minimum size of diameter required for both the motor shaft and blower. It is given as shown in Equation 3.18

Ts= Force (F) x distance (3.18)

$$Ts = 25 \times 0.13m$$

$$Ts = 3\ Nm$$

3.5.7.1 Diameter of the shaft

The diameter of shaft was determined in order to know the minimum shaft diameter that will withstand the twisting and bending moments as a result of weights on the blower device shaft. Diameter of the shaft was determined to know the strength of the shaft according Vijarayaghavan and Vishnupriyan, (2010) as shown in Equation 3.19

$$\tau_{max} D^3 = \frac{16}{\Pi}\sqrt{(Mb + Mt)^2} \qquad (3.19)$$

$\tau_y = \frac{\sigma y}{2}$

$\tau_y = \frac{380}{2}$

$\tau_y = 190 N/mm^2$

$\tau_{max} = \frac{\tau y}{n}$

$\tau_{max} = \frac{190}{2}$

$\tau_{max} = 95\ N/mm^2$

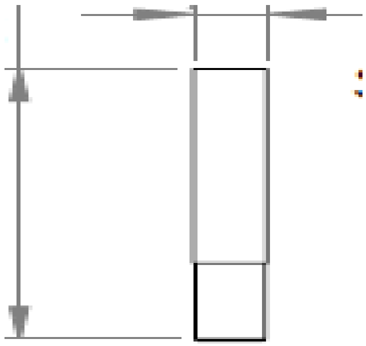

Figure 3.5: Diameter of Shaft

Where:

τ_{max} = Shear stress associated with the shaft, N/mm^2

n = factor of safety applied to the shaft (material used for the shaft is plain carbon steel C45 with yield strength of 380 N/mm^2

Mb = Bending moment, Nm

σy = Yield strength of the material used for the shaft, Nm

Mt = Torsional moment, Nm

3.5.8 Design of shaft

The size of the shaft to transmit power from drive wheel to the sucking device is dependent on the twisting moment (torque) and the maximum bending moment on the shafts as well as the allowable stress of the material of make of the shaft. The minimum shaft diameter was obtained from the following relationship reported by Gbabo, *et al.*, (2013).

$$d^3 = \frac{16}{S_a \pi}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad (3.20)$$

Where: d = diameter of shaft (m)

S = allowable shear stress ($40 \times 10^6\ N/m^2$ for shaft with key way)

Kb = combined shock and fatigue factor for bending = 1.5 (Oluwole *et al.*, 2012)

Kt = combined shock and fatigue factor twisting = 1.0 (Oluwole *et al.*, 2012)

Mb = maximum bending moment (1Nm)

Mt = twisting moment (3Nm)

Impeller shaft diameter d_m was calculated to be 16mm. Therefore, 20 mm diameter is adopted.

3.5.9 Determination of length of belt

In order to choose the suitable belt for the machine, the length of the belt was calculated by adding the pulley circumference to twice the centre distance between the pulleys because when the belt is used to reduce speed, pulleys of different diameters are used. The total length (L) was obtained using Equation (3.22) as given by (Oluwole *et al.,* 2012).

$$L = 2(C) + \pi(\frac{D2+D1}{2}) + \frac{(D2+D1)^2}{2} \tag{3.21}$$

Where:

L= Belt length, mm

C = Center line between the two pulleys, mm

D_2 = Pitch diameter of the first pulley, mm

D_1 = Pitch diameter of the second pulley, mm

C= 400mm

D1= 50mm

D2= 80mm

Therefore, by substitution,

$L = 2(400) + \pi(\frac{80+50}{2}) + \frac{(80+50)^2}{2}$

L = 1000 mm

The closest value to the calculated length of the belt form from standard table of belt are 800 mm and 1000 mm and the larger was selected because it is generally good practice to choose the next larger than the next smaller size (Taye, 2000).

3.5.10 Design of Cyclone

The cyclone is a separator and was designed and fabricated to provide means by which grains will be removed from air or other gas stream at low cost and low maintenance. The cyclone type used for the separation of air from the grain was the tangential feeding type because of the nature of the grains it conveys. The standard parameters for the design are shown in the figure 3.6.

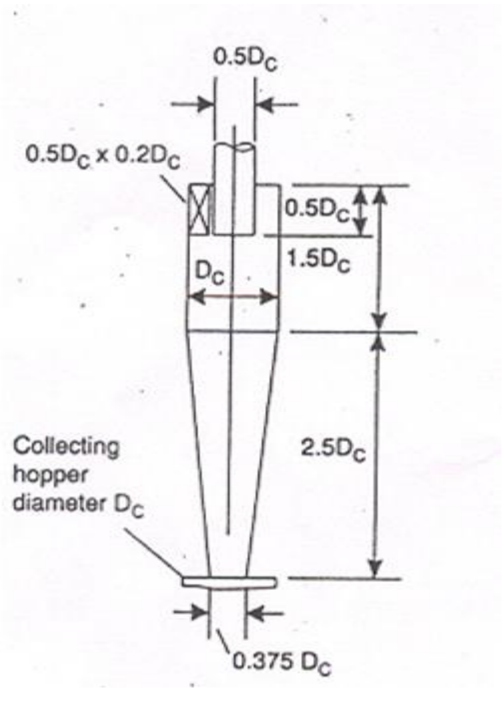

Figure 3.6: A Cyclone

D_c = 500 mm

Vortex finder = $0.5D_c$ = 250 mm

Cylinder height = $1.5D_c$ = 750 mm

Tangential inlet length = $0.5D_c$ x $0.2D_c$= 25000 mm

Conical length= $2.5D_c$ = 1250 mm

Discharge = $0.375D_c$ = 187.5 mm

3.6 Machine Description

The grain packing machine consists of the following major parts. (Plates 3.1 – 3.8)

3.6.1 Machine frame

This is the skeletal structure of the grain packer on which all other components are mounted. It was constructed from angle iron bar 40 mm by 40 mm with the dimension of 1261mm x 947mm x 1638mm in order to give the required strength. Provisions were made for various other component parts to either be welded or bolted to it so as to make up the machine. During operation and for movement from one place to another, the whole frame is mounted on a mobile tyres for easy mobility. The structure of the frame is presented in plate 3.1 below

Plate 3.1: The Frame

3.6.2 The Cyclone

The cyclone type used for the separation of air from the grain was the tangential feeding type because of the nature of the grains it conveys. Tangential velocity is the dominant velocity component that determines the centrifugal force applied to the air stream. The cyclone was centered at the middle of the frame supported by iron rod which was bolted to the frame and welded to the body of the cyclone. This was done to make it easy for any adjustment. It is dimensioned 497.04 mm by 100 mm with the inlet opening diameter of 76.2 mm and the outlet opening of 62.53 mm. The structure of the cyclone is presented in plate 3.2 below

Plate 3.2: The Cyclone

3.6.3 Cyclone air discharge

This is the 3 inches galvanised steel pipe used to channel the air sucked together with the grains into the cyclone so as to pave the way for the grains to be delivered into sacks through the outlet opening. The cyclone air discharge is presented in plate 3.3 below

Plate 3.3: Cyclone Air Discharge

3.6.4 Conveying pipe

It is made up of galvanised steel pipe which was the channel by which the mixtures of air and grains pass through into the cyclone. Elbow joint metal of 3 inches was used to take care of the bends. The conveying pipe is as shown in plate 3.4 below

Plate 3.4: Conveying Pipe

3.6.5 The Sucker (suction unit)

This is a mechanical device for moving air in a direction at an angle to the incoming fluid. The blades of the blower were made up of thick mild steel flat bar cut in order to fit the shape of the blower. The impeller type is of radial flat blade of diameter 180 mm and the thickness of blade was 3.79 mm and the height of blade was 40 mm. The structure of the blower is presented in plate 3.5 below

Plate 3.5: The Blower

3.6.6 Sucking inlet

This is the rectangular shaped like structure by which mixtures of air and grains were sucked by the effect of the blower into the cyclone though the conveying pipe, the suction inlet was dimensioned 191.29 mm x 191 mm x 281/90 mm and it is presented in plate 3.6 below

Plate 3.6: Sucking Inlet

3.6.7 Bagging plate

The bagging plate below the cyclone supports up to 50kg of sack to be filled. The tray was made up of 2.3mm thick steel flat bar welded across the frame in order to sit the sack during the packing of grain. A framed wire mesh was provided between the flat base part of the bagging section and prime mover. Also, a hook welded to the body of the cyclone was used to hold the sack during the packing of grain. The structures of bagging section is presented in plate 3.7 below

Plate 3.7: Bagging Plate **Plate 3.8:** A Hook

The developed grain packing machine is presented in plate 3.9,

Plate 3.9: Developed Dried Grain Packing Machine

3.7 Machine Fabrication Procedure

SolidWorks(CAD) software was used for the drawing of the grain packing machine. The workings and assembled drawings are presented in the appendix. The materials bought are listed in Table 1 detailing the cost and quantity needed. The machine was fabricated at Technology Incubation Centre, Minna, Niger State. Major tools used for the fabrication include cutting machine, welding machine, angle cutter and measuring tape. The frame was first fabricated according to the design; this was where other parts were coupled. Various parts were cut based on their measurement and welded using electrodes and welding machine. The cyclone was also fabricated and thereafter placed at the middle top of the frame. Iron rods were used to support them by bolting the iron rods to the frame and welded the rods to the body of the cyclone. The blower was fabricated; radial flat blades were cut at the designed height and thickness which makes up the blower. The blower is of centrifugal fan. Galvanized steel pipe was used to join the blower and cyclone together which is the channel by which the grains will pass through. Suction inlet was fabricated according to the design taken the shape of a rectangle so as suck the mixtures of air and grains. The plate for the sack was also designed and fabricated at the lower part of the frame below the cyclone to support the sack during operation. After the fabrication, the machine was tested and evaluation was carried out to determine the efficiency of the machine.

3.8 Procedure of testing the machine/Performance evaluation

The samples of cereal grains used were obtained from a local market in Bosso in Niger State. 50 kg bags each of paddy rice, millet and guinea corns were obtained to carry out the performance evaluation on the machine. The quality of the machine was noted before being packed by the machine in order to know the effect of the machine on the grains.

The grains were spread at different grain thicknesses of 2.5, 2 and 1.5 cm on the smooth cemented ground by using a calibrated scale so as to allow the machine to move from one point to the other by the operator. After that, the machine was powered and operated by the operator at different operator speed of 2.0, 1.5 and 0.7 m/s because the average working speed of human being is 1.5 m/s and also at the different machine speed of 2880, 1440 and

720 rpm by using different pulleys. The machine was then evaluated to know the performance of the machine based on the following dependent variables;

i. Mass of Collected Grains, W_{pc} (kg)

This is the quantity of grains collected within a specified time frame. The grains were being packed at different time interval so as to monitor the efficiency of packing of the grains by the machine. It is taken in kilogram (kg).

ii. Sucking Loss, S_l **(kg)**

This is the differences between the total quantity of grains spread (kg) and the quantity of grains packed (kg) by the machine. Any grains not packed by the machine is termed suction loss (kg) and it is always minima in comparison to the mass of grains collected. It is taken in kg

iii Machine Capacity (Mc), kg/h

This refers to the quantity of grains collected per unit time. Collecting capacity of the machine was determined using Equation 3.22

$$\mathrm{Mc} = \frac{W_{pc}}{T} \qquad (3.22)$$

Where:

Mc = Collecting capacity, kg/min

W_{pc} = Mass of packed grains, kg

T = Time taking for packing, min

iv. Grain Collecting Efficiency (Ce), %

The collecting efficiency of the machine is the ratio of grains packed (W_{pc}) to the sum of grains collected and suction losses. The collecting efficiency of the machine was determined using Equation 3.23.

$$C_e = \frac{W_{pc}}{W_{pc} + S_l} \times 100 \qquad (3.23)$$

where:

C_e = grain collecting efficiency, %

W_{pc} = mass of grains collected, kg

S_l = Suction loss, kg

v. Percentage of Broken Grain P_B (%)

This is the ratio of the mass of the broken grains to the total mass of the sample of the grains collected expressed in percentage.

$$P_B(\%) = \frac{Mass\ of\ the\ broken\ grains\ (kg)}{Total\ Samples\ of\ Collected\ Grains\ (kg)} \times 100 \quad (3.25)$$

vi. Percentage of Whole Grains P_W (%)

This is the ratio of the mass of whole grains to the total mass of the sample of grains collected expressed in percentage

$$P_W(\%) = \frac{Mass\ of\ the\ whole\ grains\ (kg)}{Total\ Samples\ of\ Collected\ Grains\ (kg)} \times 100 \quad (3.26)$$

3.9 Experimental Design

In order to evaluate the performance of the packing machine, the variables considered for the performance evaluation were categorized into independent and dependent variables. The independent variables include the operator speed, grain thickness and machine speed which were used to determine the effect of changes on the independent variables which include collecting capacity, collecting efficiency, percentage of damaged grains and percentage of whole grains

The data were subjected to statistical analysis using design expert 11 software

3.10 Cost Analysis

The cost of constructing the machine was calculated as follows:

(i) Material cost: This is the cost of materials used in the construction of the machine (Table 1 in the Appendix)

(ii) Labour cost: This is the cost of services rendered by human being during the construction of the machine and it is always be thirty percent of the material cost

(iii) Overhead cost: This includes the cost of feeding, transportation and miscellaneous expenses incurred during the construction of the equipment, it was assumed to be twenty percent of the material cost

Total cost of material = ₦49,625

Cost of Labour = $\frac{30}{100}$ x ₦49625 = ₦14,888

Overhead cost = $\frac{20}{100}$ x ₦49625 = ₦9,925

Total cost of fabrication = Material cost + Labour cost + Overhead cost

Total cost of fabrication = ₦49,625 + ₦14,888 + ₦9,925 = ₦74,435

CHAPTER FOUR

4.0 RESULTS AND DISCUSSION

4.1 Collecting Efficiency at Various Speeds

The collecting efficiency of the machine at different machine speed is presented in Table 4.1

Table 4.1: Collecting efficiency at different speed

Machine Speed (rpm)	Operator Speed (m/s)	Machine Capacity (kg/min)	Percentage Broken (%)	Percentage whole grain (%)	Collecting Efficiency (%)
2880	2.0	1.1	0.2	99.8	91.7
1440	1.5	0.59	1.02	99.0	87.8
720	0.7	1.14	1.07	98.9	90.6

The results show that the collecting efficiency of the machine was 91.7%, 87.8%, 90.6% at 2880, 1440, 720 rpm respectively. In normal condition, increasing the speed of the machine would result to an increased in collecting efficiency of the machine. Results revealed that the collecting efficiency decreased from 91.7% to 87.8% when operated from 2880 to 1440 rpm and further increased from 87.8 to 90.6% when operated from 1440 to 720 rpm. Consistency in the operation of the machine might be reason for the decrease in the collecting efficiency when the machine was operated from 2880 to1440 rpm. This result is similar to the work of Hellen *et al.,* (2018) where the collecting efficiency are 99.33%, 98.77%, 98.99% at 4200, 4000, 3800 rpm respectively. The effects of machine speed and operator speed on machine capacity, percentage of broken grains, percentage of whole grains and collection efficiency were respectively presented in Figures 4.1 and 4.2.

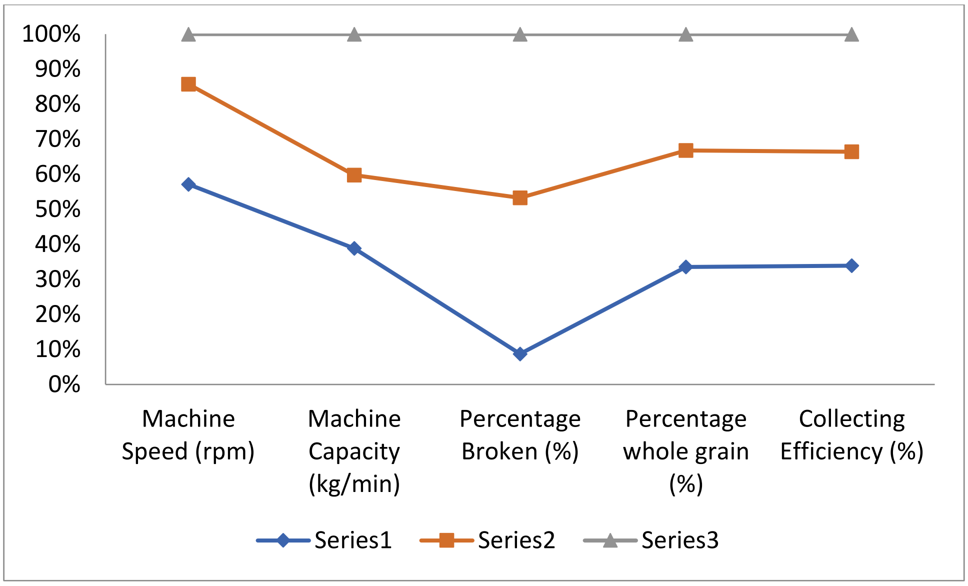

Figure 4.1: Effect of Machine Speed on machine capacity, percentage of broken grains, percentage of whole grains and collection efficiency

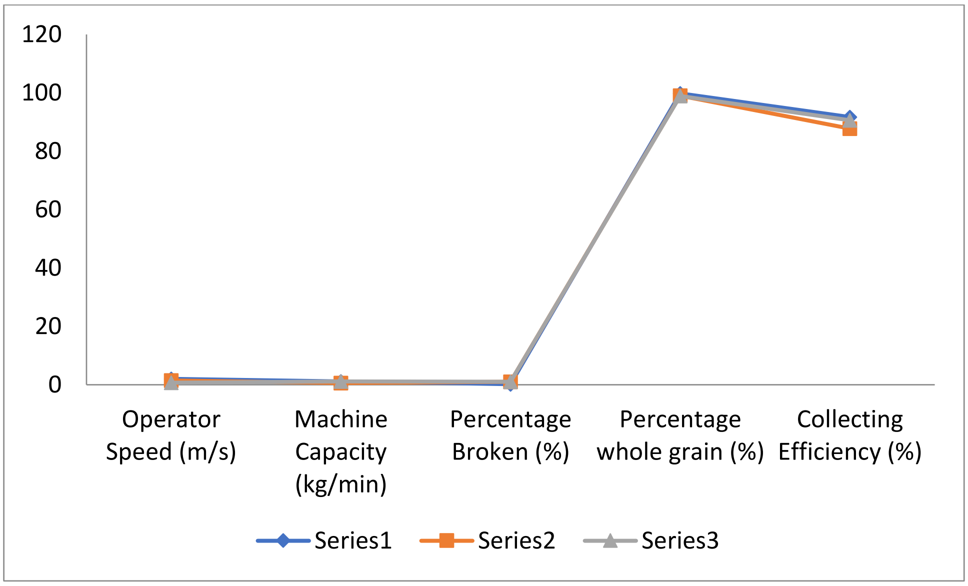

Figure 4.2: Effect of Operator Speed on machine capacity, percentage of broken grains, percentage of whole grains and collection efficiency

4.2 Effects of Operator Speed, Grain Thickness and Machine Speed on Machine Capacity

The effects of operator speed, grain thickness and machine speed on the machine capacity is presented in Figure 4.3. The minimum machine capacity of 0.98 kg/min was obtained at operator speed of 0.7 m/s while the maximum was 1.16 kg/min at operator speed of 2.0 m/s with machine speed of 2880 rpm. Also, the minimum machine capacity of 0.5 kg/min was obtained at 0.7 m/s operator speed while the maximum was 0.7 kg/min at 2.0 m/s operator speed with machine speed of 1440 rpm. At machine speed of 720 rpm, the minimum machine capacity of 1.02 kg/min was obtained at 0.7 m/s operator speed while the maximum was 1.12 kg/min at 2.0 m/s operator speed. This implies that the machine capacity increased with increased in operator speed at all machine speeds (Table 4.2, Appendix A).

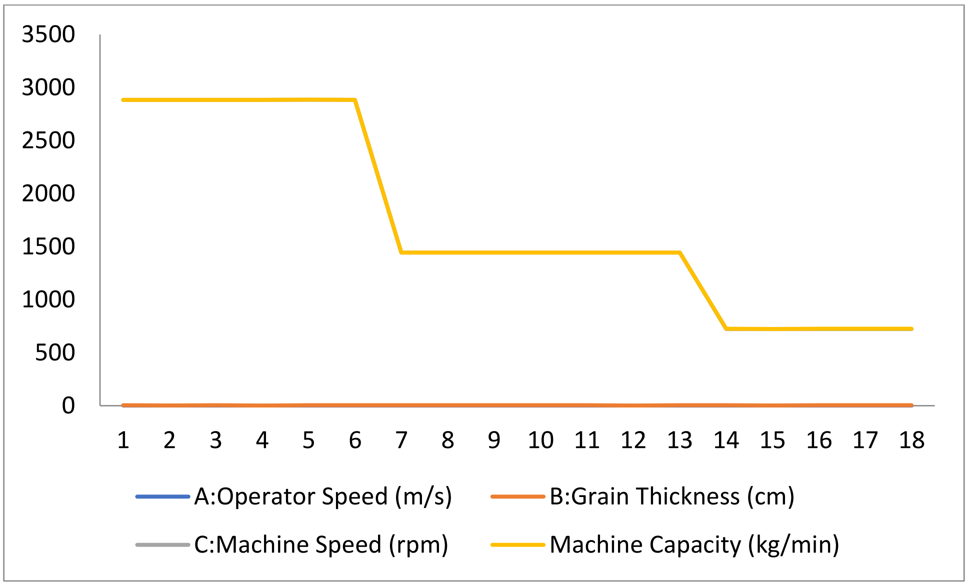

Figure 4.3: The Effects of Operator Speed, Grain Thickness and Machine Speed on the Machine Capacity

Statistical analysis (Table 4.4, Appendix A) shows that the operator speed, grain thickness and machine speed does not have significant ($p<0.05$) effect on the machine capacity. Rather only C^2 is significant.

The 3-D response surface for machine capacity in terms of machine speed and operator speed is presented in Figure 4.4. The plot gave the optimum machine capacity of 1.2 kg/min at 2880 rpm machine speed.

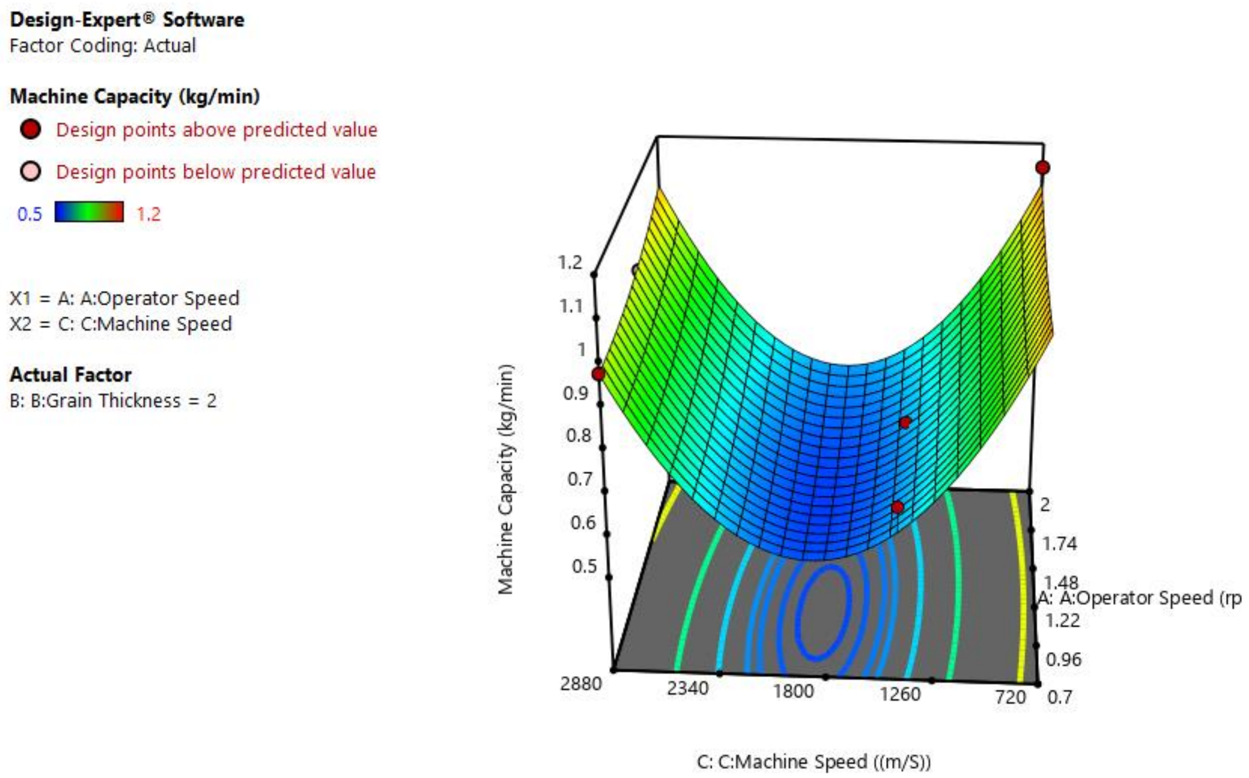

Figure 4.4: 3-D Response Surface for Machine Capacity in terms of Machine Speed and Operator Speed

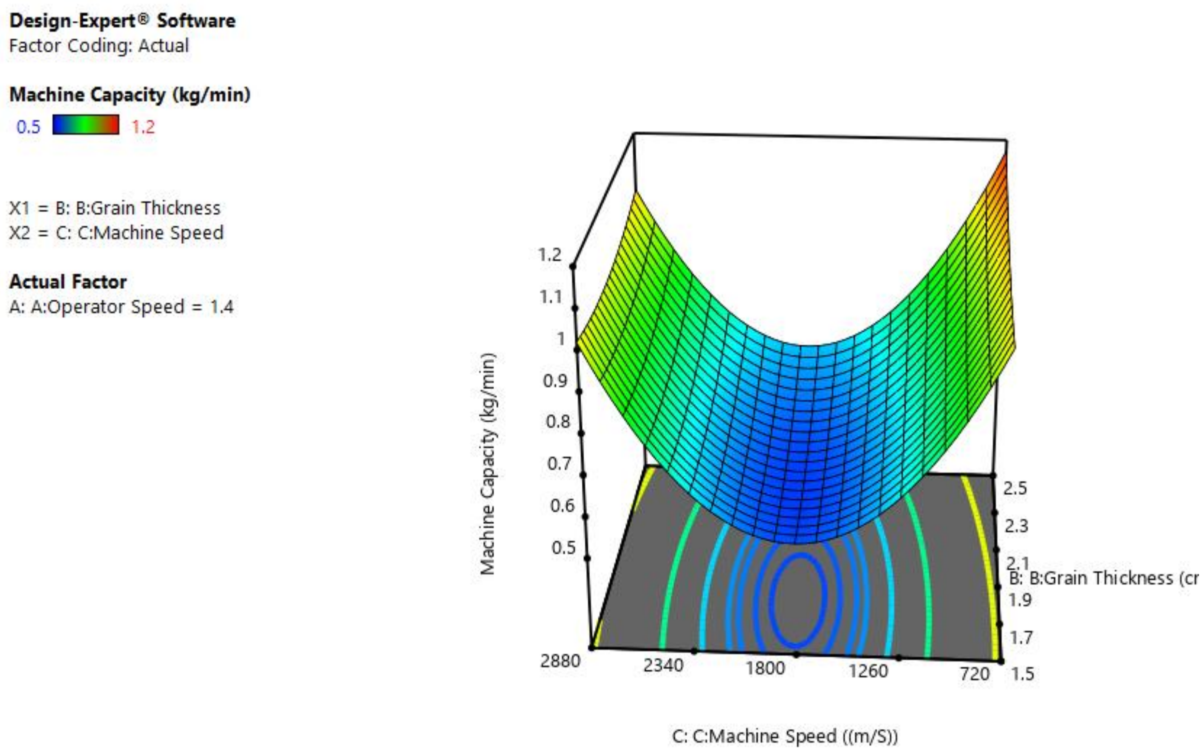

Figure 4.5: 3-D Response Surface for Machine Capacity in terms of Machine Speed and Grain Thickness

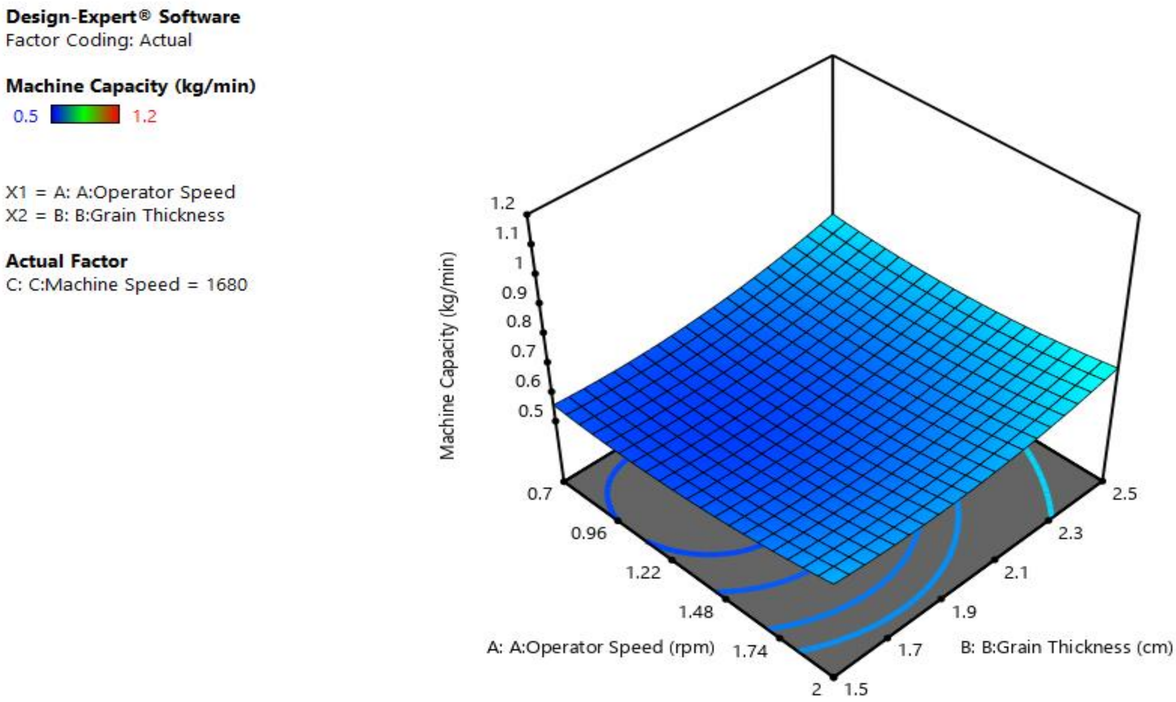

Figure 4.6: 3-D Response Surface for Machine Capacity in terms of Operator Speed and Grain Thickness

The regression equation describing effects of operator speed and grain thickness on machine capacity is given as;

$$Y_{mc}=2.36173-0.184204A-0.460395B-0.001451C-0.026154AB+0.000031AC-0.000051BC+0.081075A^2+0.166573B^2+4.10568E^{-07}C^2 \quad (4.1)$$

4.3 Effects of Operator Speed, Grain Thickness and Machine Speed on Collection Efficiency

The effects of operator speed, grain thickness and machine speed on collection efficiency is presented in Figure 4.7. The minimum operator speed was obtained at 82.0% while the maximum was at 91.7% (Table 4.2, Appendix A). The collection efficiency increased with increase in operator speed, this is in agreement with the work of Hellen *et al.*, (2018). Analysis of variance as shown in Table 4.5 (Appendix A) revealed that machine speed and interaction AB are significant. The response surface plot in Figure 4.8 shows the collecting efficiency in terms of operator speed and machine speed. The condition 2 m/s and 2880 rpm gave the optimum collecting efficiency of 91.7%. The regression equation describing effects of operator speed on collecting efficiency is given as;

$$Y_{CF} = + 94.65478 - 4.55086A + 1.74733B - 0.009639C - 4.62153AB + 0.001502AC + 0.001100BC + 4.55956A^2 + 0.928120B^2 + 1.11850E^{-6}C^2 \quad (4.2)$$

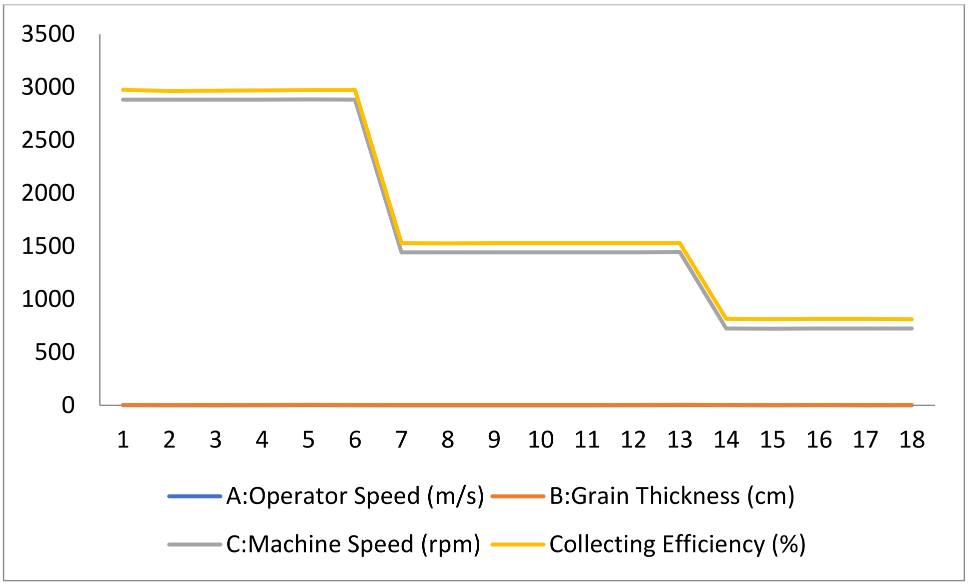

Figure 4.7: The Effects of Operator Speed, Grain Thickness and Machine Speed on Collection Efficiency

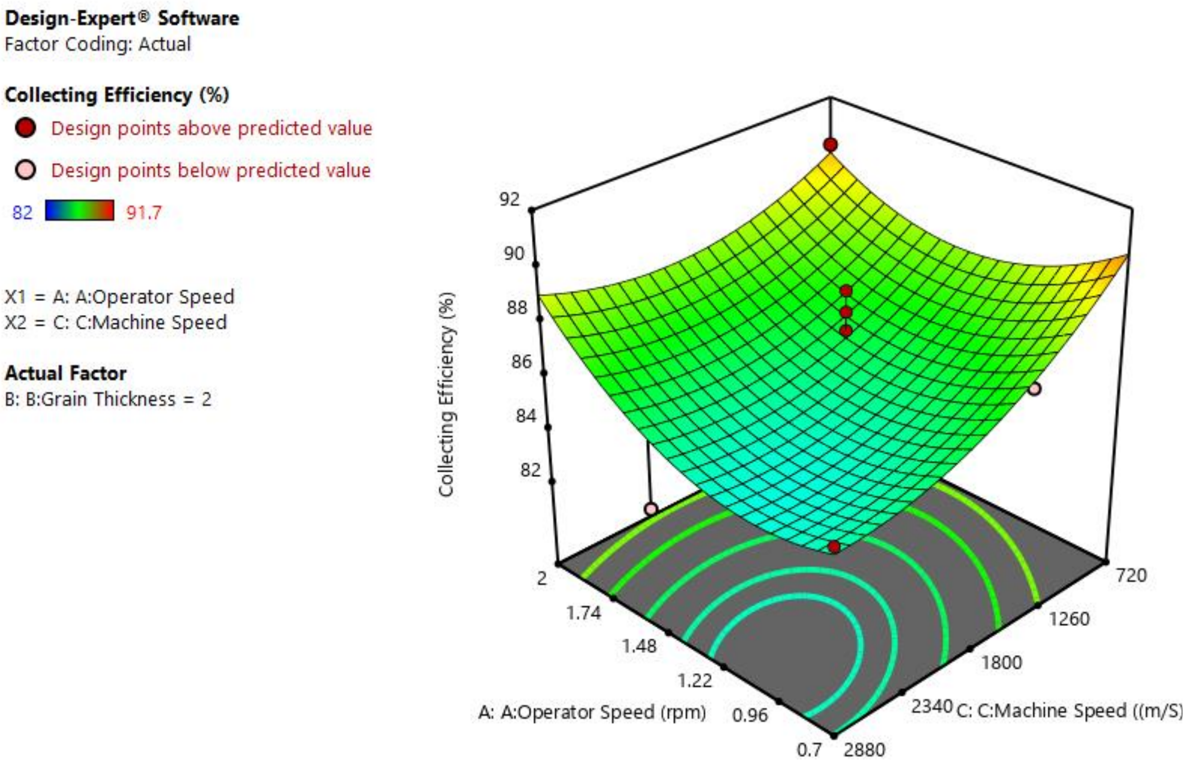

Figure 4.8: 3-D Response Surface for Collecting Efficiency in terms of Machine Speed and Operator speed

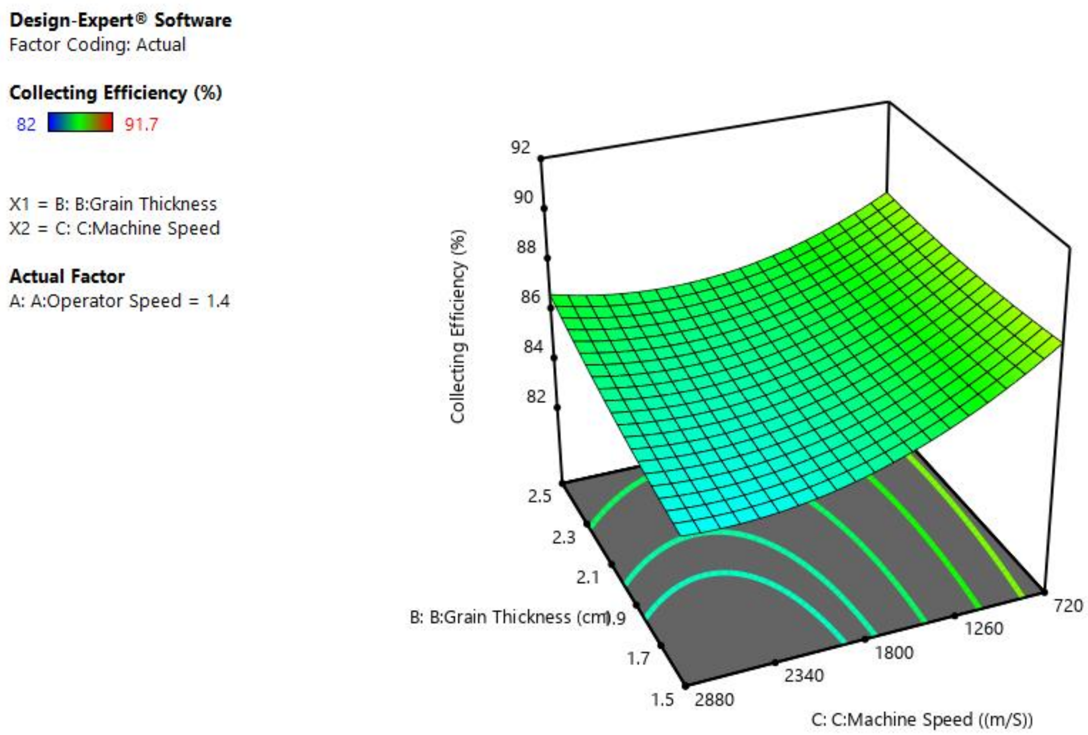

Figure 4.9: 3-D Response Surface for Collecting Efficiency in terms of Machine Speed and Grain Thickness

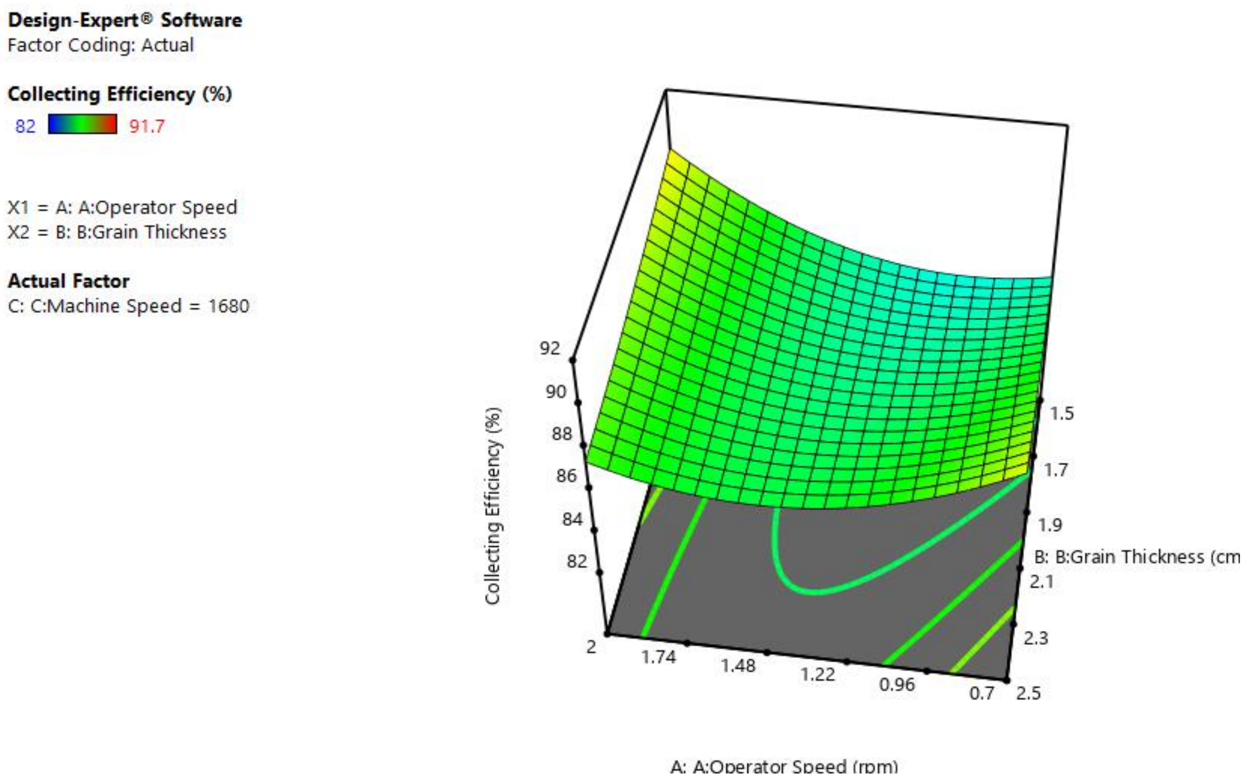

Figure 4.10: 3-D Response Surface for Collecting Efficiency in terms of Operator Speed and Grain Thickness

4.4 Effects of Operator Speed, Grain Thickness and Machine Speed on Percentage of Broken Grains

The effects of operator speed, grain thickness and machine speed on percentage of broken grains is presented in Figure 4.1. The result as presented in Table 4.2 (Appendix A) shows that decreased in operator speed leads to an increase in percentage damage of the grain. The minimum operator speed was obtained at 0.2% while the maximum was at 2.04%. Statistical analysis (Table 4.6, Appendix A) shows that the operator speed (A), grain thickness (B), machine speed (C) and interaction (AB) has significant ($p<0.05$) effect on the percentage broken grain.

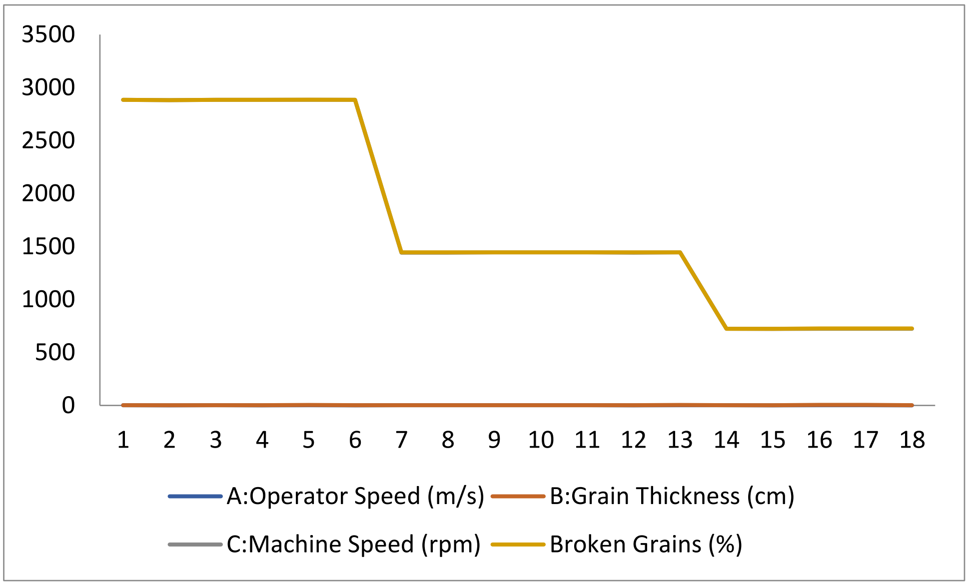

Figure 4.11: The Effects of Operator Speed, Grain Thickness and Machine Speed on Percentage of Broken Grains

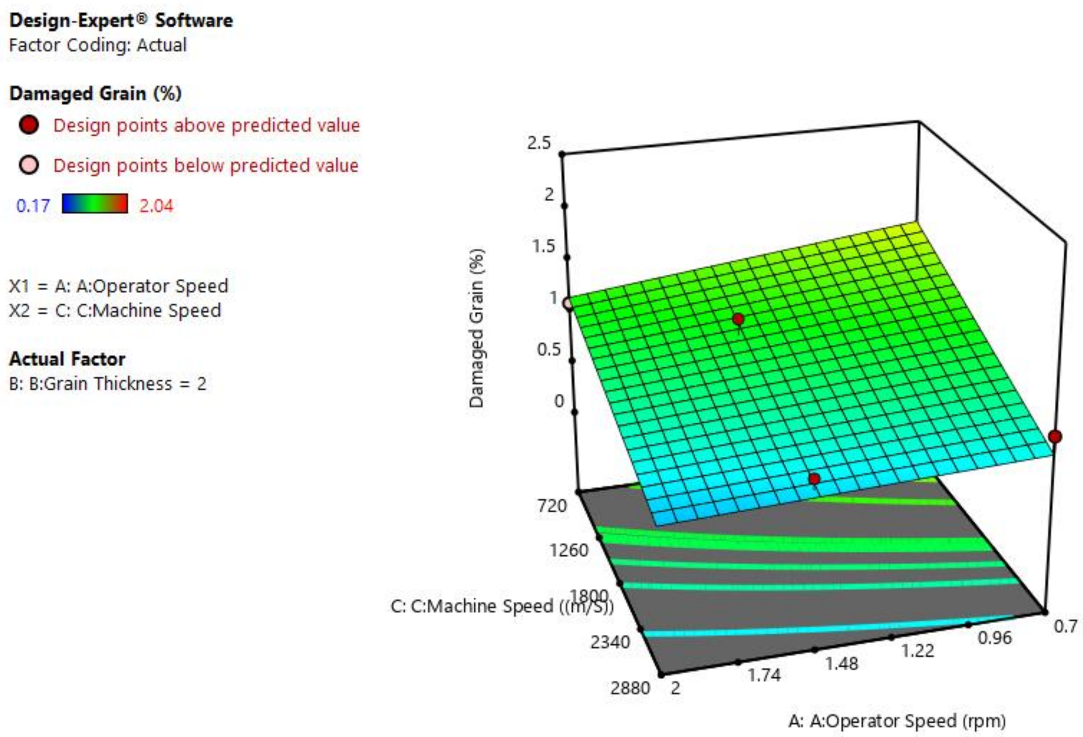

Figure 4.12: 3-D Response Surface for Percentage Broken Grain in terms of Machine Speed and Operator Speed

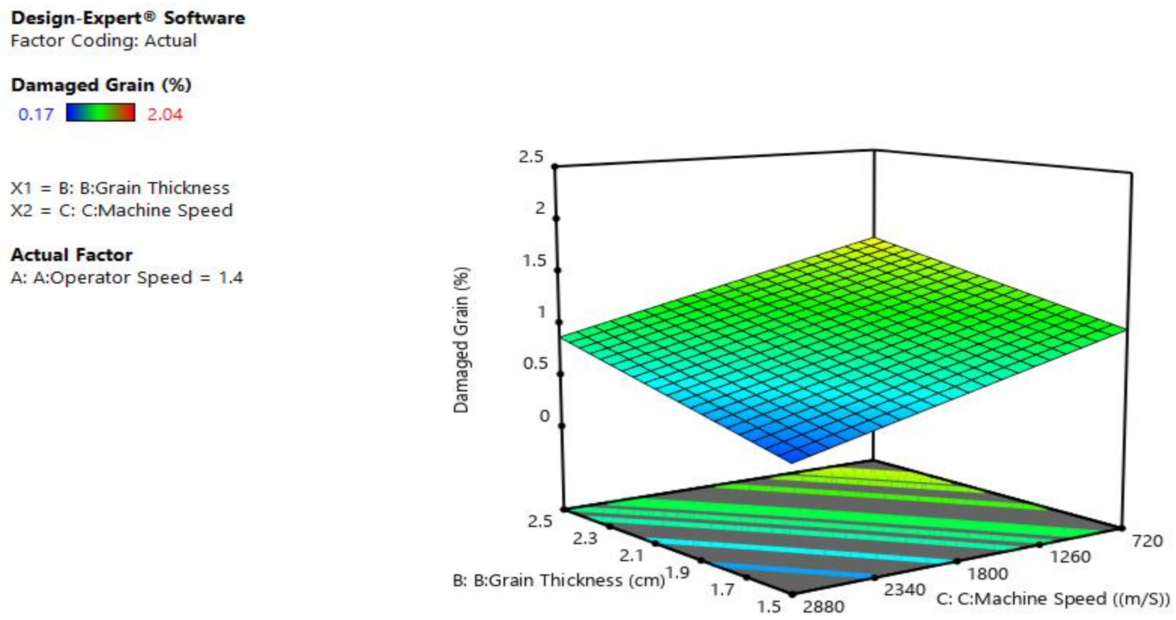

Figure 4.13: 3-D Response Surface for Percentage Broken Grain in terms of Machine Speed and Grain Thickness

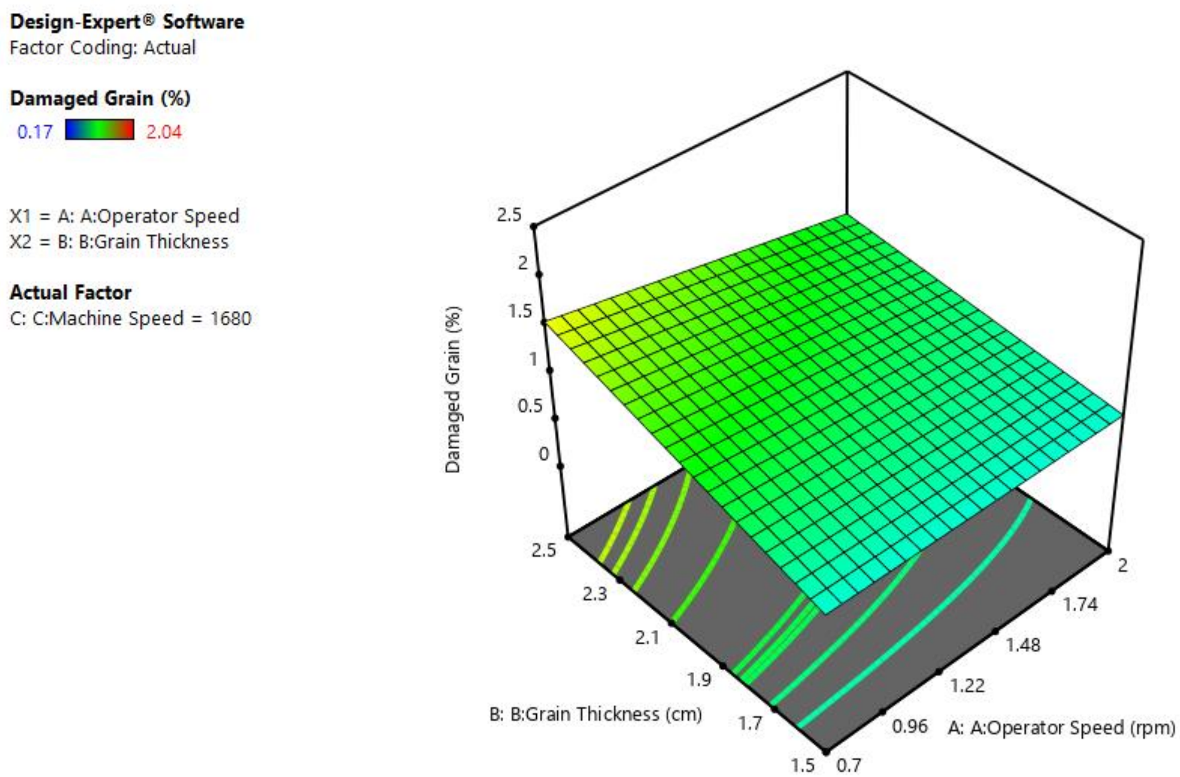

Figure 4.14: 3-D Response Surface for Percentage Broken Grains in terms of Operator Speed and Grain Thickness

The response surface plot in (Figure 4.12) shows the percentage broken grain in terms of operator speed and machine speed. The condition gave optimum percentage broken grain of 2.04% at 0.7 m/s and the minimum of 0.17% at 2.0 m/s. The regression equation describing effects of operator speed on percentage broken grain is given as;

$$Y_{PBG} = -0.143904 + 0.500969A + 1.10394B - 0.000517C - 0.428248AB + 0.000087AC + 0.000026BC \quad (4.3)$$

4.5 Effects of Operator Speed, Grain Thickness and Machine Speed on Percentage Whole Grain

The effects of operator speed, grain thickness and machine speed on percentage whole grain is shown in Figure 4.15 below. Table 4.2 (Appendix A) shows the range of 98% to 99.8%

percentage whole grain was obtained at the operator speed of 0.7 and 2.0 m/s respectively. The statistical analysis (Table 4.7) shows that operator speed, grain thickness and machine speed has significant effects on percentage whole grain. Optimum percentage whole grain of 99.8 % obtained at 2 m/s of operator speed, 1.5cm of grain thickness and 2880 rpm of machine speed.

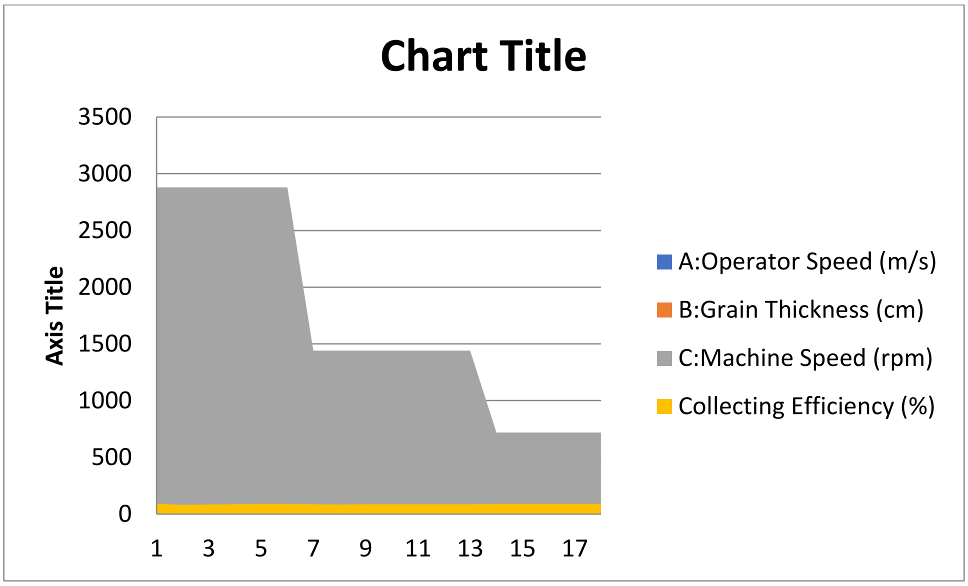

Figure 4.15: The effects of operator speed, grain thickness and machine speed on percentage whole grain

The regression analysis for the model is given as;

$$Y_{PWG} = 100.53715 - 1.49403A - 1.38540B + 0.001069C + 0.340166AB - 0.000035BC + 0.441902A^2 + 0.110674B^2 - 1.44216E^{-7} \quad (4.4)$$

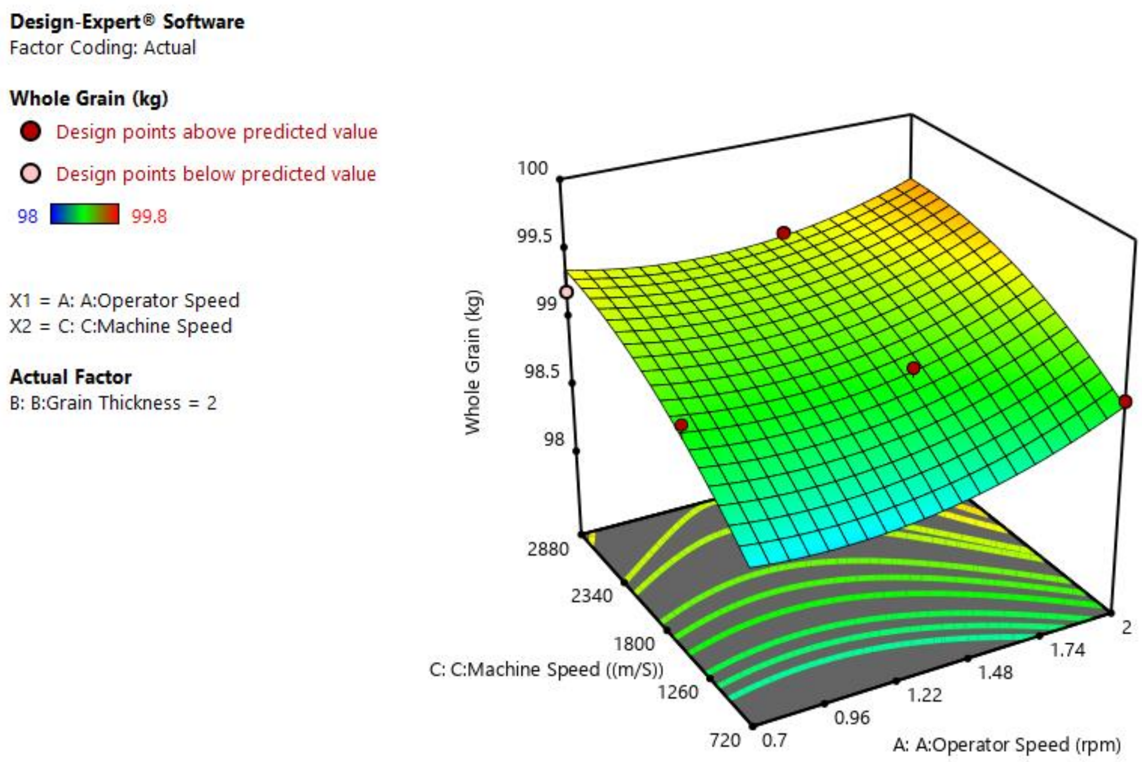

Figure 4.16: 3-D Response Surface for Percentage Whole Grain in terms of Operator Speed and Machine Speed

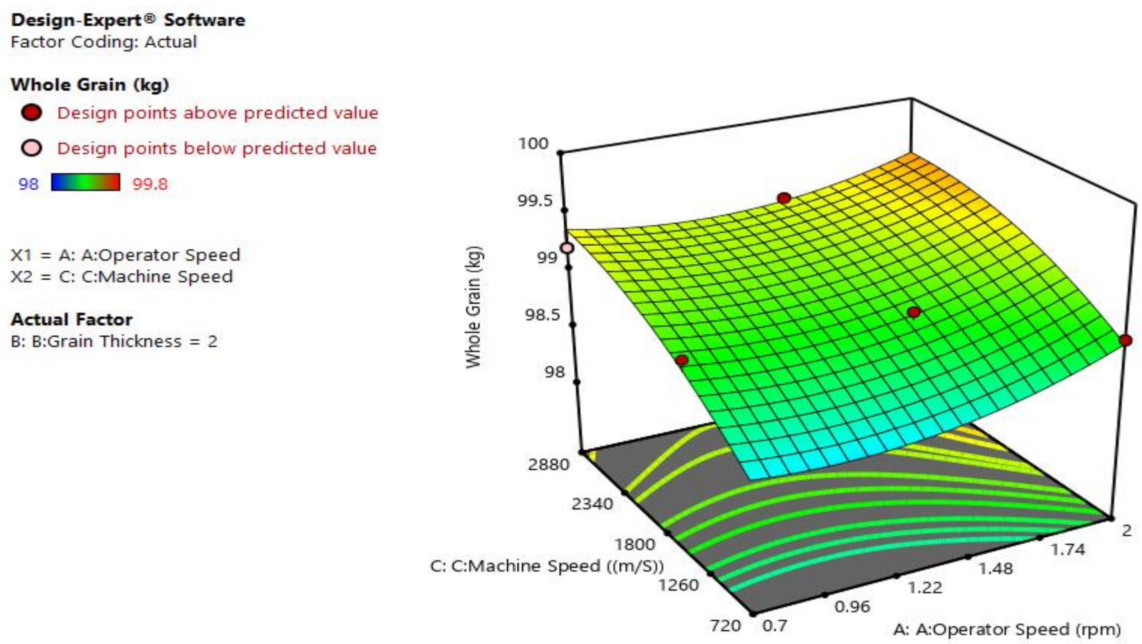

Figure 4.17: 3-D Response Surface for Percentage Whole Grain in terms of Operator Speed and Machine Speed

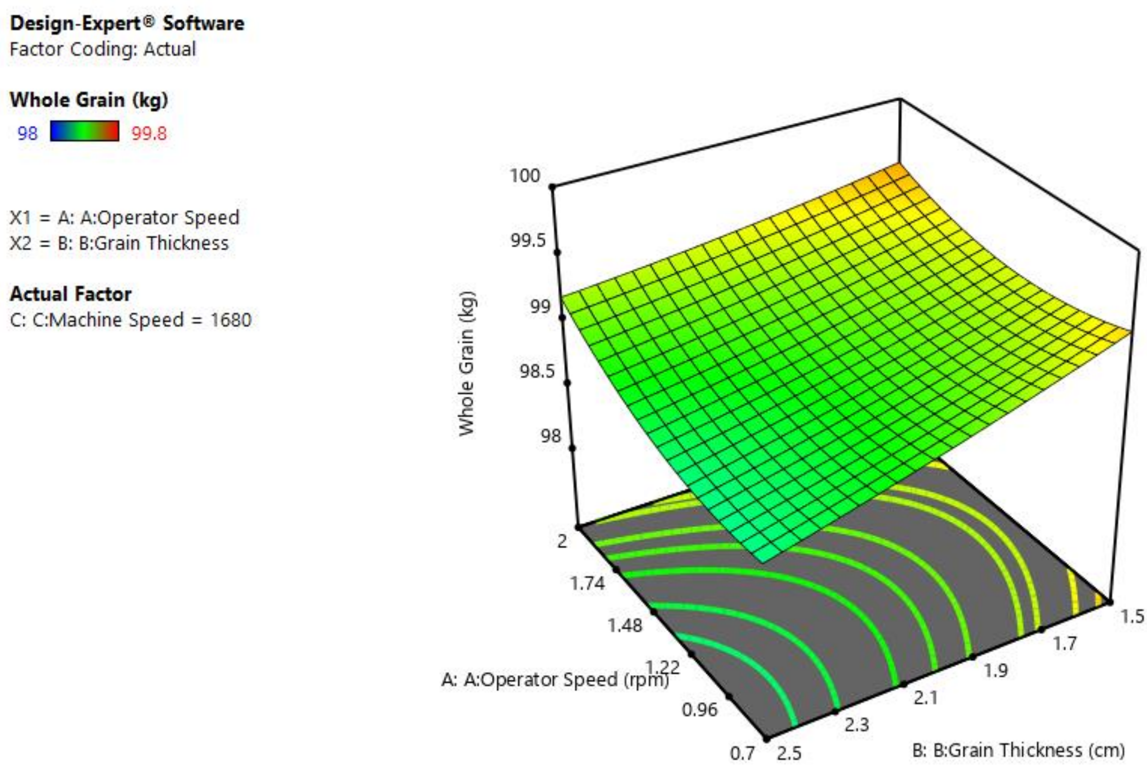

Figure 4.18: 3-D Response Surface for Percentage Whole Grain in terms of Operator Speed and Grain Thickness

CHAPTER FIVE

5.0 CONCLUSION AND RECOMMENDATION

5.1 Conclusion

A grain packing machine capable of packing grains within a stipulated time was designed, fabricated and tested.

The grain packing machine works on the principle of pneumatic conveying of products from the ground to a storage medium. Centrifugal forward curved fan was used as a blower so as to reduce the damages on the grains.

Diameter of suction pipe, air velocity, pressure drop and power requirement were determined in relation to pneumatic conveying characteristics to design the grain packing machine.

The machine was successfully constructed and evaluated. The performance evaluation of the machine was carried out at three different machine speeds of 2880, 1440 and 720 rpm and three different operator speeds of 2.0, 1.5 and 0.7 m/s.

i. The machine capacity increased with increase in operator speed at all machine speeds and statistical analysis revealed that operator speed, grain thickness and machine speed does not have significant effects on the machine capacity

ii. The collection efficiency increased with increase in operator speed and analysis of variance revealed that machine speed and interaction between operator speed and grain thickness are significant effect on the collection efficiency.

iii. A decrease in operator speed leads to an increase in percentage of broken grain and analysis of variance shows that operator speed and the interaction between operator speed and grain thickness has significant effect on the percentage of broken grains.

iv. Optimum percentage of whole grains was obtained at the operator speed of 2 m/s, grain thickness of 1.5 cm and machine speed of 2880 rpm and the analysis of variance shows that all the independent variables have significant effects on the percentage of whole grains.

5.2 Recommendation

To further enhance the performance of the dried grain collector machine, the following are recommended:

i. Conduct a study on design of suction pipe that effectively and efficiently collect grains spread on a concrete pavement. A wider pipe design could be used; the upstream portion of the pipe of scoop (Panake) type which could collect the grain by pushing the nozzle until it reaches the downstream side of the pipe in which the velocity of air is high enough to suck the grain. The pipe could be attached to the pneumatic grain collector in order to reduce labour during operation.

ii. Conduct a study to assess the performance of pneumatic grain collector to unload the grain on flatbed dryers. This could maximize the utilization of the machine even in rainy season when sun drying on pavement would not be possible.

iii. Adjust the length of the pipe so as to reduce the suction losses by bringing the cyclone discharge closer to the sucker.

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APPENDICES

Appendix A

TABLE 1: BILL OF MATERIALS

S/N	PARTICULAR	SPECFICATION	QUANTITY	UNIT COST(#)	TOTAL COST(#)
1	Mild steel plate	1mm	Half plate	3500	3500
2	V-belt		1	1500	1500
3	Galvanised pipe	3"	1	4000	4000
4	Elbow joint	3"(Metal)	1	500	500
5	Tyers		4	1000	4000
6	Sack	100kg	1	100	100
7	Angle iron	1.5"	1	1700	1700
8	Angle iron	1"	1	1400	1400
9	Flat bar	1"	1	1000	100
10	Bolt and Nut	17mm	2	100	200
11	Bolt and Nut	14mm	4	50	200
12	Bolt and Nut	12mm	2	50	100
13	Prime mover	3.5hp, 3600rpm	1	27,000	27,000
14	Fuel	Petrol	5liters	165	825

15	Grains	Rice	50Kg	7400	7400
16	Grains	G/corn	50Kg	4800	4800
17	Grains	Millet	50Kg	4300	4300
18	Labour				14,888
19	Miscellaneous				9,925
	TOTAL				74,438

TABLE 2: COMPONENTS AND SPECIFICATION

COMPONENTS	SPECIFICATIONS
Overall Dimension and Weight	
LXBxH	1261mmX947mmX1638mm
Weight	120kg
Discharge Cyclone Separator	
Inlet Opening	76.2mm
Outlet Opening	62.53mm
Cylinder Dimension ф, H	497.04mm, 100mm
Inverted Cone Dimension	
Material	Mild steel, 1mm thickness
Air movers	
Type	Centrifugal fan
Overall Dimension	
Weight	20kg
Impeller	
Type	Radial flat blade

Dimension

Diameter	180mm
Height of blade	40mm
Thickness of blade	3.79mm

Suction Side

Type	Rectangle
Dimension	101.26 X 101 X 281.50

Discharge Side

Type	Circle
Dimension	3"
Material	Galvanized

Wheel

Front

Type	
Size, Diameter X Width	80mm
Material	Rubber

Rear

Type	
Size, Diameter X Width	80mm
Material	Rubber

Prime mover

Type

Rated Power	3.5hp
Rated speed	3600rpm
Cooling system	Air cooling
Starting System	Starter recoil
Dry weight	15.1kg
Fuel type	Petrol
Machine performance parameters	
Collecting capacity	100kg
Average Collecting efficiency	84.695%

Table 3: DIFFERENT FORMULAS USED IN THE DESIGN

PARAMETERS	**UNIT**	**FORMULARS**
Diameter of the Pipe Section	mm	$m = \varphi \times \rho \times A \times V$
Velocity Pressure	Pa	$V_p = \frac{1}{2} P v^2$
Blower Air Discharge	m^3/s	$Air\ Discharge = A \times V$
Fanning Friction		$f = \frac{0.331}{\left[\log_n\left(\left(\frac{\varepsilon}{3.7 \times D}\right) + \left(\frac{7}{N_{Re}}\right)\right)\right]^2}$
Reynolds Number		$N_{Re} = \frac{D V_g \rho_g}{\mu_g}$
Head Losses inside the Pipe	m	$\Delta P_T = \Delta P_{acc} + \Delta P_g + \Delta P_s + \Delta H_g + \Delta H_s + \Delta P_{misc}$
Particle Velocity	m/s	$V_p = 0.8 \times V_g$

Power Required	KW	$P_{out} = \rho \times Q \times g \times H$

RUN	Time (min)	A:Operator Speed (m/s)	B:Grain Thickness (cm)	C:Machine Speed (rpm)	Machine Capacity (kg/min)	Collecting Efficiency (%)	Damaged Grain (%)	Whole Grain (%)	Mass of Collected Grain (kg)	Suction Loss (kg)	Mass of Damaged Grain (kg)	Mass of Whole Grain (kg)
15	5	2	1.5	2880	1.1	91.7	0.2	99.8	5.5	0.5	0.012	5.488
1	10	0.7	1.5	2880	1	82	0.17	99.8	10	2.2	0.017	9.98
10	15	1.5	2	2880	0.99	83.2	0.74	99.3	14.85	3	0.11	14.74
8	20	0.7	2	2880	0.98	85.6	0.82	99.2	19.6	3.3	0.16	19.44
18	25	2	2.5	2880	1.16	88.4	0.72	99.3	29	3.8	0.21	28.79
9	30	0.7	2.5	2880	1.03	89.8	0.94	99.1	30.9	3.9	0.29	30.61
3	35	1.5	1.5	1440	0.8	87.5	0.79	99.2	28	4	0.22	27.78
14	40	1.5	1.5	1440	0.5	84	0.95	99.1	20	3.8	0.19	19.81
5	45	1.5	2	1440	0.56	86.3	1.22	98.8	25.2	3.99	0.31	24.89
6	50	1.5	2	1440	0.59	87.8	1.02	99	29.5	4.1	0.3	29.2

RPM of Motor (N)	rpm	$v = \frac{\pi \times N \times D}{60}$
Collecting Capacity	Kg/h	$Fc = \frac{W_{pc}}{T}$
Collecting Efficiency	%	$C_e = \frac{W_{pc}}{W_{pc} + S_l} \; x\, 100$

Table 4.2: Data analysis

11	55	1.5	2	1440	0.62	87	1.03	99	34.1	5.1	0.35	33.75
16	60	0.7	2	1440	0.69	87.3	1.02	99	41.4	6	0.42	40.98
4	65	2	2.5	1440	0.7	87	0.95	99.1	45.5	6.8	0.43	45.07
7	70	2	1.5	720	1.06	90.3	1.02	99	74.2	8	0.76	73.44
2	75	0.7	1.5	720	1.02	90.6	1.07	98.9	76.5	7.98	0.82	75.68
17	80	2	2	720	1.14	90.2	1.09	98.9	91.2	9.8	0.99	90.21
13	85	1.5	2.5	720	1.2	89.9	1.57	98.4	102	11.5	1.6	100.4
12	90	0.7	2.5	720	1.2	90	2.04	98	108	12	2.2	105.8

Table 4.4: ANOVA for Response Surface Quadratic Model (Machine Capacity)

Source	Sum of Squares	df	Mean Square	F-value	p-value	
Model	0.9051	9	0.1006	12.45	0.0008	significant
A-A:Operator Speed	0.0068	1	0.0068	0.8402	0.3861	
B-B:Grain Thickness	0.0166	1	0.0166	2.06	0.1894	
C-C:Machine Speed	0.0125	1	0.0125	1.55	0.2487	

AB	0.0006	1	0.0006	0.0701	0.7979	
AC	0.0042	1	0.0042	0.5239	0.4898	
BC	0.0060	1	0.0060	0.7397	0.4148	
A²	0.0029	1	0.0029	0.3644	0.5628	
B²	0.0061	1	0.0061	0.7502	0.4116	
C²	0.4928	1	0.4928	61.02	< 0.0001	
Residual	0.0646	8	0.0081			
Lack of Fit	0.0178	5	0.0036	0.2283	0.9272	not significant
Pure Error	0.0468	3	0.0156			
Cor Total	0.9697	17				

Table 4.5: ANOVA for Response Surface Quadratic Model (Collecting Efficiency)

Source	**Sum of Squares**	**df**	**Mean Square**	**F-value**	**p-value**	
Model	89.73	9	9.97	2.17	0.1441	not significant
A-A:Operator Speed	6.68	1	6.68	1.46	0.2619	
B-B:Grain Thickness	3.78	1	3.78	0.8246	0.3904	

C-C:Machine Speed	23.70	1	23.70	5.17	0.0527	
AB	17.67	1	17.67	3.85	0.0854	
AC	9.69	1	9.69	2.11	0.1842	
BC	2.83	1	2.83	0.6168	0.4549	
A^2	9.31	1	9.31	2.03	0.1922	
B^2	0.1881	1	0.1881	0.0410	0.8446	
C^2	3.66	1	3.66	0.7970	0.3980	
Residual	36.71	8	4.59			
Lack of Fit	29.46	5	5.89	2.44	0.2470	not significant
Pure Error	7.25	3	2.42			
Cor Total	126.44	17				

Table 4.6: ANOVA for Response Surface Quadratic Model (Percentage Broken Grain)

Source	Sum of Squares	df	Mean Square	F-value	p-value	
Model	2.77	6	0.4612	21.12	< 0.0001	Significant
A-A:Operator Speed	0.1849	1	0.1849	8.47	0.0142	
B-B:Grain	0.8806	1	0.8806	40.34	<	

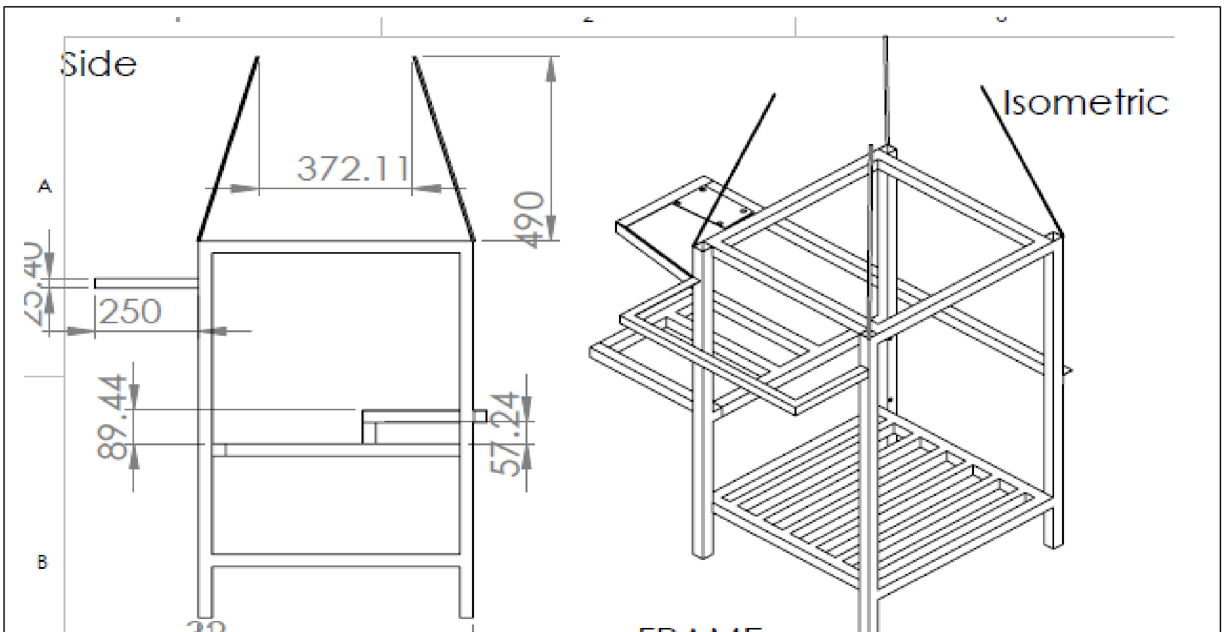
Thickness					0.0001	
C-C:Machine Speed	1.61	1	1.61	73.58	< 0.0001	
AB	0.1555	1	0.1555	7.12	0.0218	
AC	0.0343	1	0.0343	1.57	0.2362	
BC	0.0016	1	0.0016	0.0734	0.7915	
Residual	0.2402	11	0.0218			
Lack of Fit	0.2020	8	0.0252	1.98	0.3101	not significant
Pure Error	0.0382	3	0.0127			
Cor Total	3.01	17				

Table 4.7: ANOVA for Response Surface Quadratic Model (Percentage Whole Grain)

Source	Sum of Squares	df	Mean Square	F-value	p-value	
Model	2.83	9	0.3141	21.26	0.0001	Significant
A-A:Operator Speed	0.2169	1	0.2169	14.69	0.0050	
B-B:Grain	0.7830	1	0.7830	53.00	<	

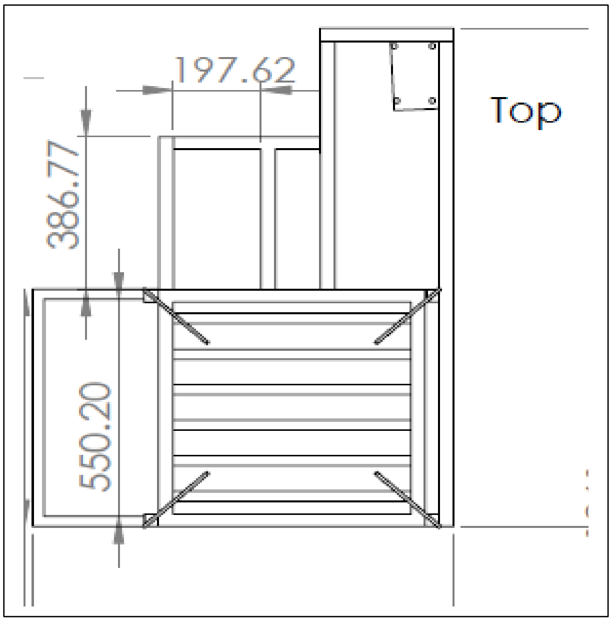
Thickness					0.0001	
C-C:Machine Speed	1.60	1	1.60	108.37	< 0.0001	
AB	0.0957	1	0.0957	6.48	0.0344	
AC	0.0338	1	0.0338	2.29	0.1687	
BC	0.0029	1	0.0029	0.1937	0.6715	
A^2	0.0874	1	0.0874	5.92	0.0410	
B^2	0.0027	1	0.0027	0.1811	0.6817	
C^2	0.0608	1	0.0608	4.12	0.0770	
Residual	0.1182	8	0.0148			
Lack of Fit	0.0865	5	0.0173	1.64	0.3630	not significant
Pure Error	0.0317	3	0.0106			
Cor Total	2.94	17				

APPENDIX B

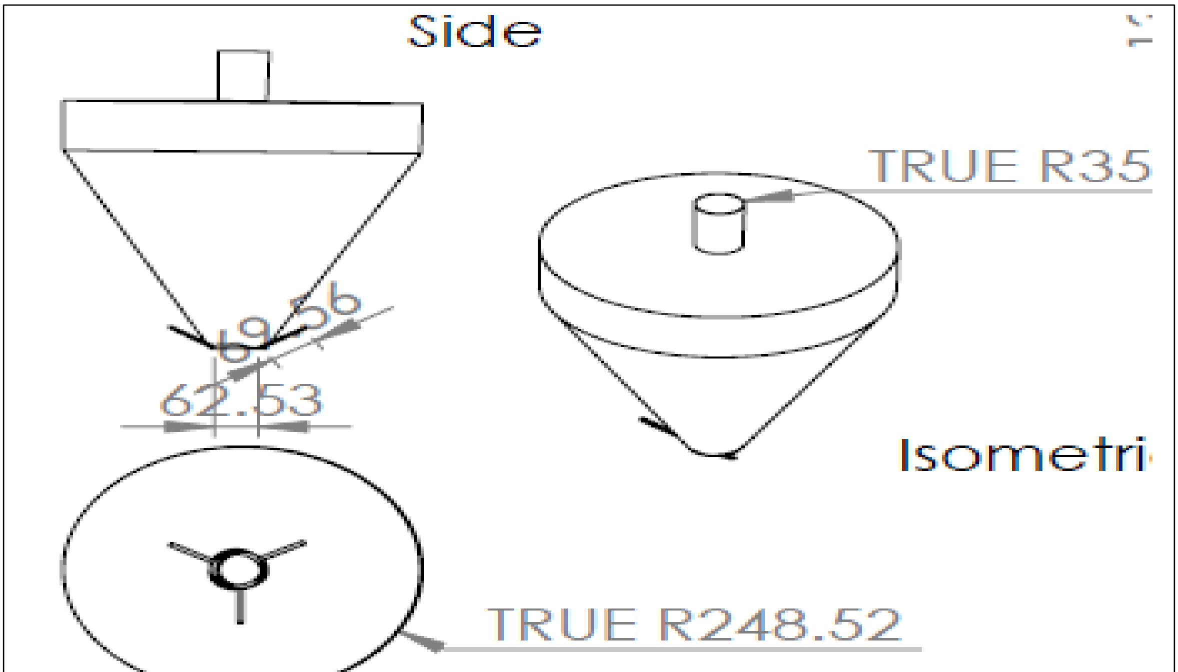


S

B1: Front View and Isometric View of the Packer Frame

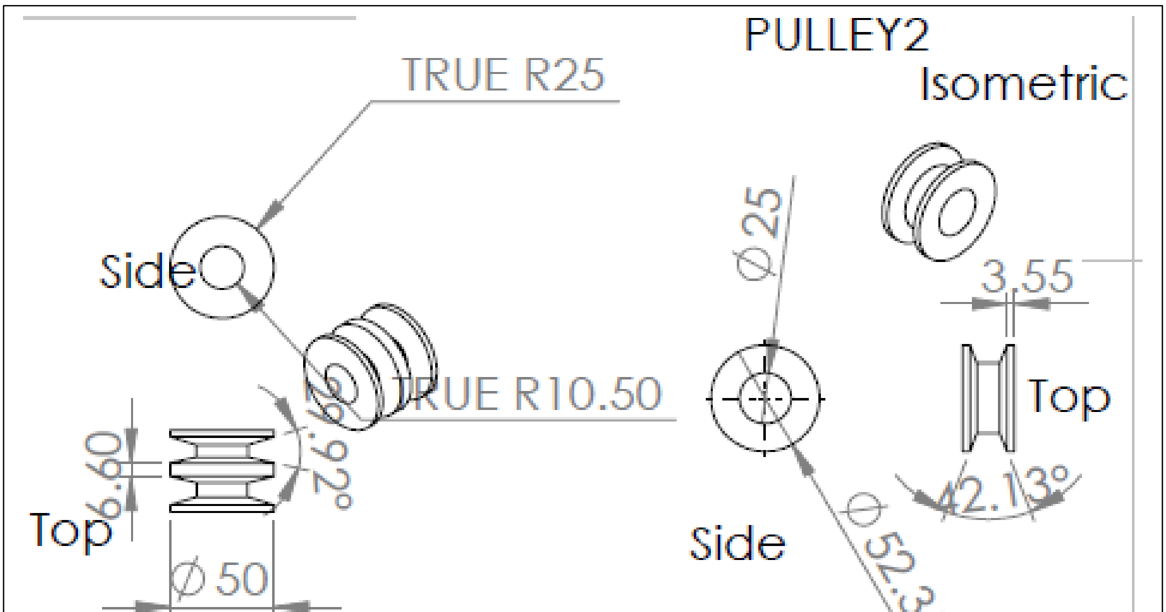


B2: Top View (Schematic) of the Packer Frame

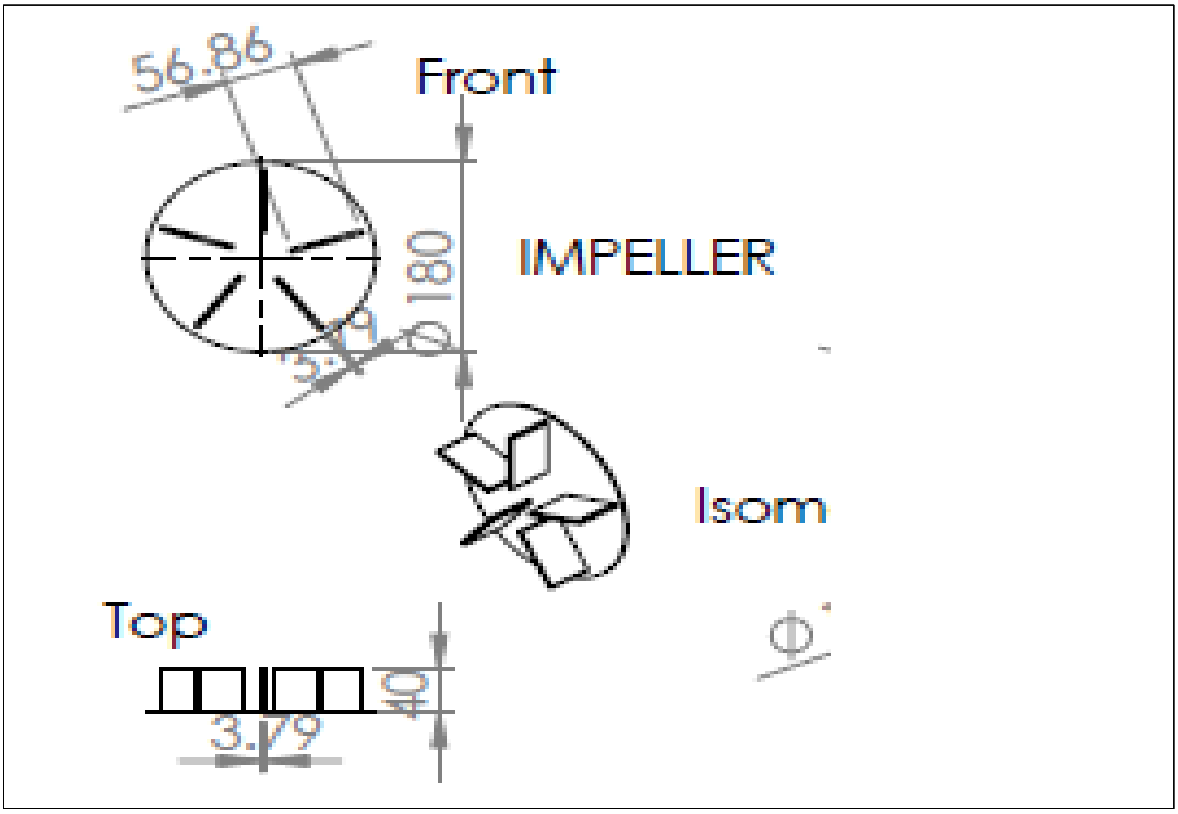


B3: Side View, Top and Isometric View (Schematic) of the Cyclone

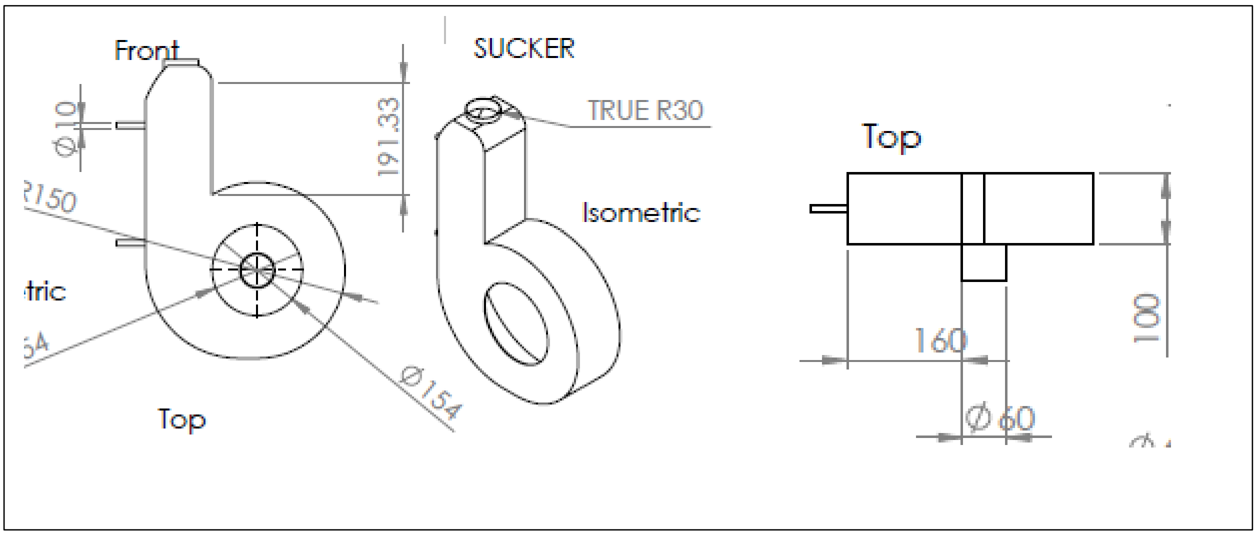
B4: Side View, Top and Isometric View (Schematic) of the Blower Shaft



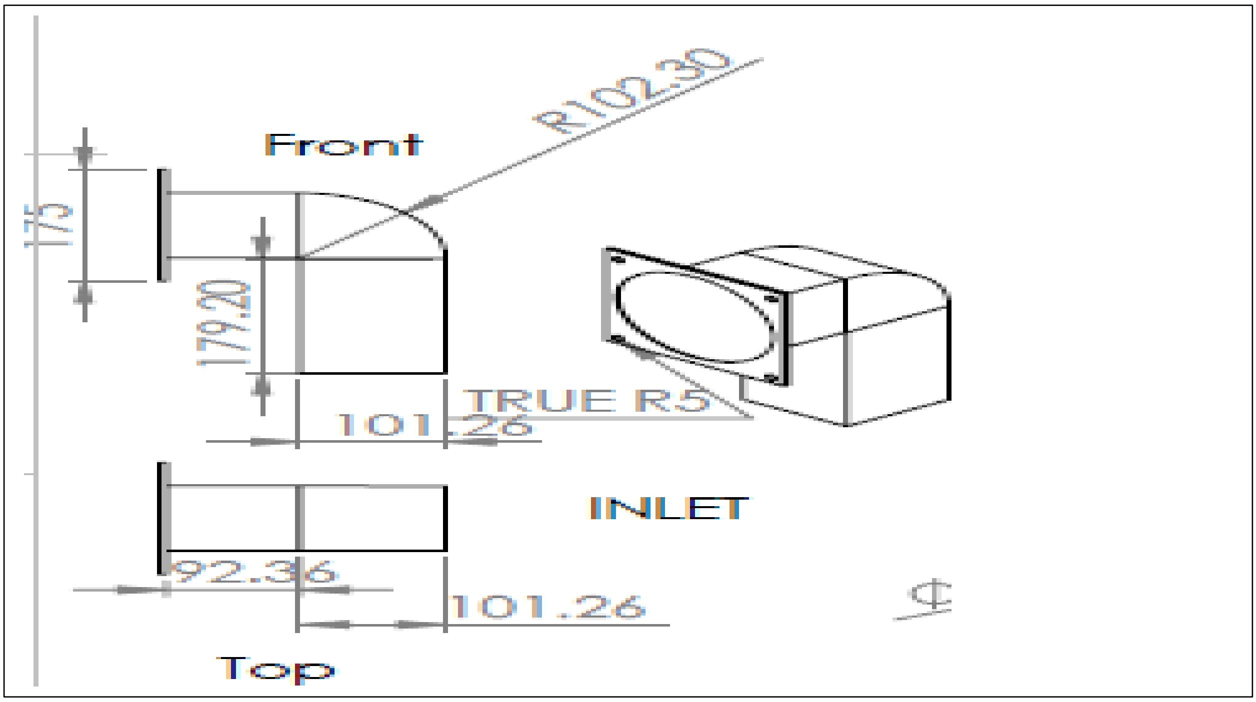
B5: Side View, Top and Isometric View (Schematic) of the Pulleys



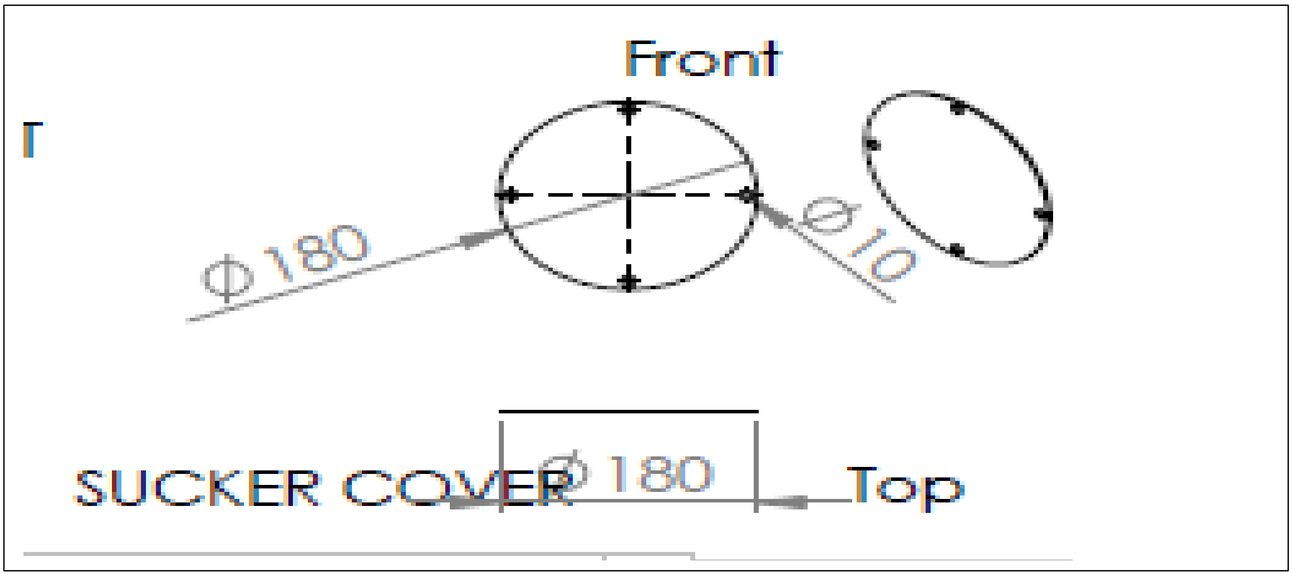
B6: Side View, Top and Isometric View (Schematic) of the Impeller Blade



B7: Side View, Top and Isometric View (Schematic) of the Blower Case

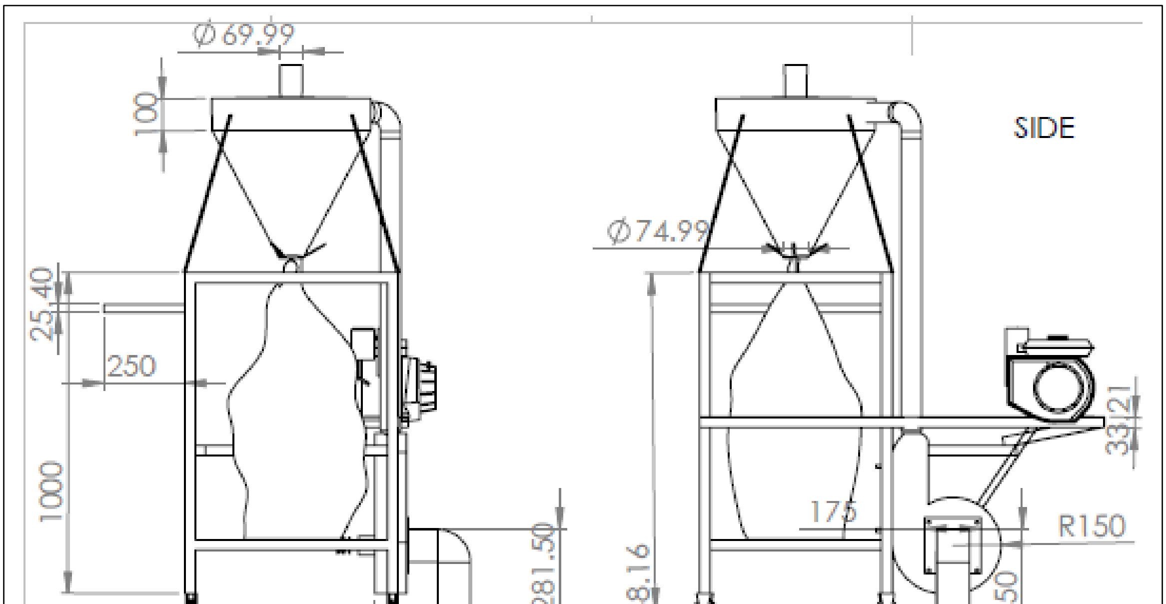


B8: Side View, Top and Isometric View (Schematic) of the Sucker Inlet

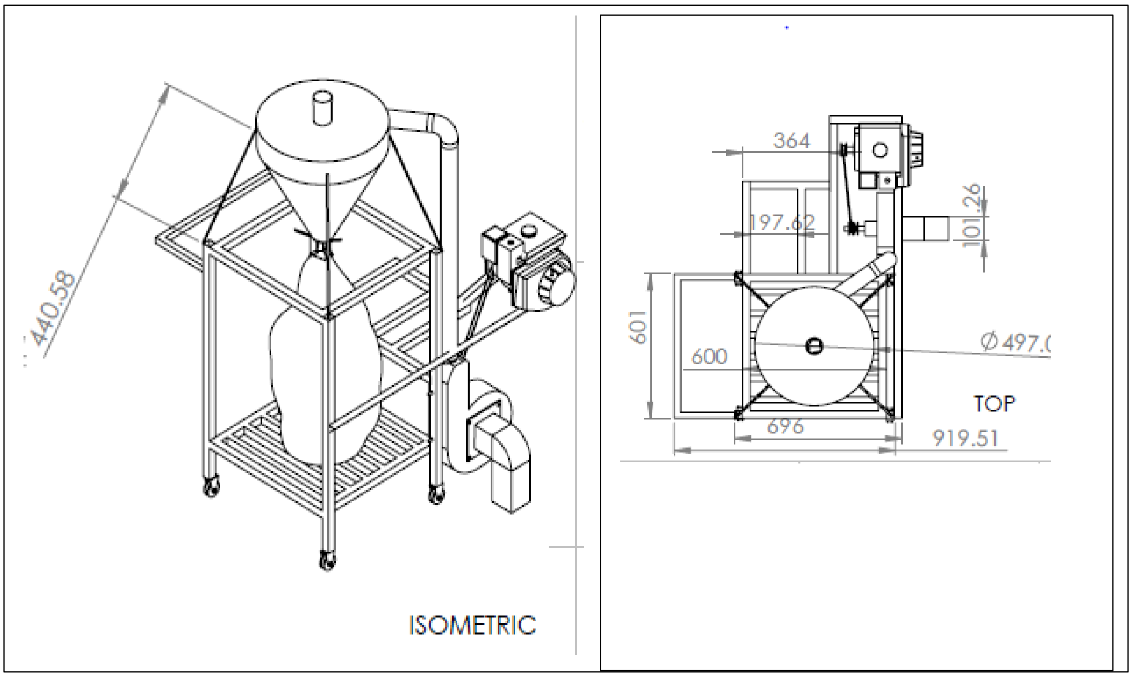


B9: Side View, Top and Isometric View (Schematic) of the Sucker Cover

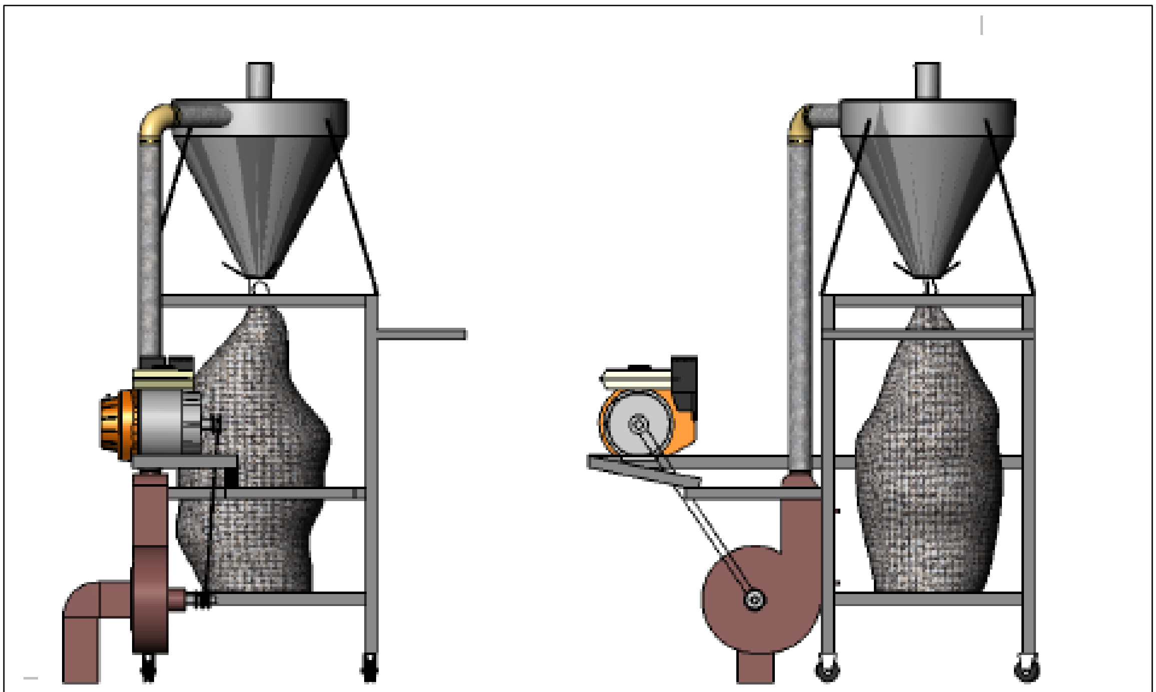
B10: Side View, Top and Isometric View (Schematic) of the Elbow Joint



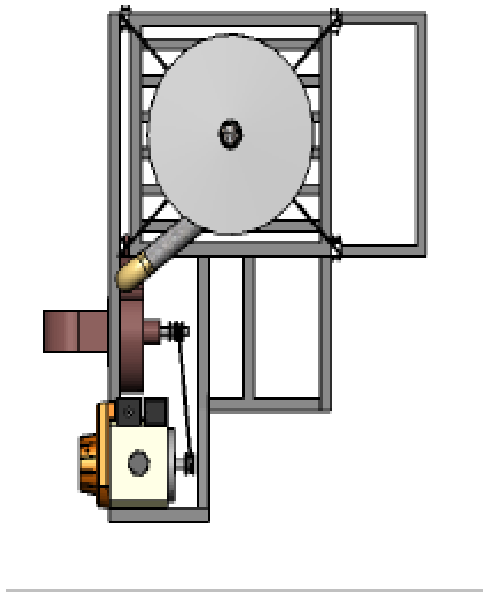
B11: Side Views (Schematic) of the Machine



B12: Isometric and Top View (Schematic) of the Machine



B13: Front and Side Views (Animated) of the Machine



B14: Isometric and Top Views (Animated) of the Machine

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