

**AN INVESTIGATION INTO THE EFFECTS OF DIFFERENT
BLOW TANK AERATION CONDITIONS ON THE
CONVEYABILITY OF FINE POWDERS IN PNEUMATIC
CONVEYING SYSTEMS**

**A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF REQUIREMENT FOR THE
DEGREE OF**

**MASTER OF ENGINEERING
IN THERMAL ENGINEERING**

BY

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CERTIFICATION

This is to certify that the dissertation entitled "An investigate into the effects of different blow tank aeration conditions on the conveyability of fine powders in pneumatic conveying systems" is an authentic record of my work carried out as requirements for the award of the degree of Master of Engineering in Thermal Engineering at Thapar Institute of Engineering and Technology, Patiala under the supervision of Dr. S.S. Mallick, Associate Professor, Mechanical Engineering and Mr. Atul Sharma, Lecturer, Mechanical Engineering. No part of this thesis has been submitted for the award of any degree.

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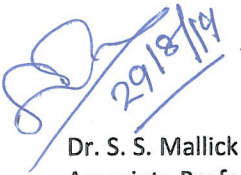


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“The essence of all beautiful art is gratitude”, quoth Freidrich Nietzsche. Though heartfelt always, expression of this beautiful and powerful emotion makes an absolute necessity, for best and rewarding transactions in life that summate to create what comes to be called ‘experience’.

A seeker, I have been fortunate to have received, almost always abundantly, both in instances expected and unexpected. And today, as I meet a milestone, specifically in career, by and large in life, I am afforded an opportunity to spread the sweetness I feel, through humble strokes of my pen.

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DEDICATION

Dedicated to my parents Dr. Mohd. Azhar Masoodi and Tahira Masoodi who are my role models.

In their feet lies my world, in their service my existence finds a purpose.

ABSTRACT

Pneumatic conveying test rigs, one positive pressure system and one negative pressure (vacuum) system, were designed, fabricated and installed at the 'Laboratory for Bulk Solids and Particle Technologies' in the Mechanical Engineering Department of Thapar Institute of Engineering and Technology (T.I.E.T), Patiala. The work was carried out under the aegis of a research collaboration between T.I.E.T and NTPC Ltd., with an aim to rid the pneumatic conveying plants installed in power stations owned by NTPC Ltd. of certain encountered problems. This document reports the entire facility building procedure/activity, for the aforementioned test rigs, undertaken at T.I.E.T.

Presented also are the findings of experimental investigations conducted on a bottom discharge type 0.8 m^3 blow tank feeder, used in the present study, through variation of aeration conditions of this feeder and the air flow rates fed to the system for material conveyance, are presented. Fly-ash (bulk density 1000.7 kg/m^3 , particle density 1384.4 kg/m^3 , $d_{50} = 43.61 \text{ }\mu\text{m}$) was conveyed over a wide range of flow rates, and correspondingly a wide range of air flow rates, to study the effect of fluidization air supply to the blow tank, total air supply to the system, aerating side air supply to the blow tank, location of aerating supply to the blow tank, and pre-pressurization condition of the blow tank on flow rate by mass, of material, in tonnes/hour (tonnage), through the system. Furthermore, the effect of varying relative proportions of side air to the blow tank and conveying air to the system was also evaluated. Pre-pressurization of the blow tank feeder and increase in relative proportion of side air fed to the blow tank feeder were seen to considerably favour the tonnage.

For use in the larger experimental program five sonic nozzles for metering specified mass flow rates of air to the system was designed. The nozzles are arranged to provide parallel flow, to form an assembly called the sonic nozzle bank.

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NOMENCLATURE

H	Riser inlet height above gas distribution plate [m]
Q_f	Fluidization gas flow rate [m^3s^{-1}]
Q_p	Pressurizing gas flow rate [m^3s^{-1}]
Q_s	Supplemental gas flow rate [m^3s^{-1}]
d_r	Riser diameter [m]
d_{50}	Median particle diameter [m]
m_s	Solid mass flow rate [$kg s^{-1}$]
ρ	Density of air [$kg m^{-3}$]
ρ_b	Loose poured bulk density [$kg m^{-3}$]
ρ_t	Tapped bulk density [$kg m^{-3}$]
ρ_p	Particle density [$kg m^{-3}$]
μ	Solid loading ratio
P_0	Stagnation pressure (absolute) at inlet (Pa)
T_0	Stagnation temperature (absolute) at inlet (K)
ρ_0	Stagnation Density at inlet (kg/m^3)
P^*	Pressure at throat (Critical)
T^*	Temperature at throat (Critical)
q_m	Choked mass flow rate required (kg/s)
C^*	Critical Flow Function (from standards)
μ_0	Coefficient of dynamic viscosity at Inlet stagnation conditions
V^*	Sonic Velocity at throat (m/s)
ρ^*	Density at throat
Re^*	Reynolds No. at throat based on inlet conditions of Viscosity and Mass flow rate
Cd	Coefficient of discharge(from Standards)
R	Gas Constant for air (J/kg.K)
A^*	Area of cross-section of throat (m^2)
P_b	Downstream pressure
r_c	Radius of curvature of toroidal surface (m)
d	diameter of throat (m)
k	Specific Heats Ratio
l	Length of the divergent section (m)
θ	Half angle of the frustum of divergent section
A_2	Area of cross-section of exit of divergent section

$P_{2\max}$	Maximum allowable downstream pressure
r_e	Radius of exit section
d_e	diameter of exit section

Chapter 1: Introduction and objectives

1.1 Introduction

As the nomenclature suggests, pneumatic conveying is an industrial transportation process for bulk solids (fly-ash, cement, pharmaceutical products, and agricultural grains) wherein material is conveyed from one position to another through pipelines using a pressurized gas, typically air. Used across the gamut of industry, the advantages that favour the adoption of a material transportation system as such are bound transportation, dry grasp, eco-friendliness, relative economical friendliness, layout flexibility, ease of installation and automation, besides permitting multiple material entry and exodus points. Traditionally pneumatic conveying is operated through a dilute phase or suspension flow mode that involves a magnitude of carrier gas velocity high enough to allow ingress and subsequent suspension of all the particles along the pipeline. However, this mode is subject to certain distinct disadvantages such as a lower solid mass-flow rate for a given mass-flow rate of air i.e. low solid loading ratio, wear of the piping infrastructure (straight lines and bends), demand for higher capacity compressors and bag filters to mention a few. The other mode referred to as fluidized dense phase pneumatic conveying involves a non-suspension flow (inhomogeneously distributed material across pipe cross-sectional area) and presents the merits of high material loading ratios with low gas velocity at pipe inlet. For a well-designed setup, for example, loading ratios as high as 100 are achievable with fluidized dense phase conveying over long distances, compared to typically under 20 with dilute phase conveying, the conveyed product being a ‘Geldart Group A’ categorized solid such as fly-ash or cement. The dense phase conveying however presents the complexities of a highly turbulent flow and formation of non-suspension dunes (Mallick, 2010).

Fly-ash is a ‘Geldart Group A’ categorized fine powder that forms the major tranche (around 80%) of ash generated in coal fired thermal power stations. To give a quick approximation, one might well expect over 6000 tonnes of fly-ash to be generated in one day by running continuously an installed capacity of about 4800 MW. Assumptively, at a bulk density of 600 kg/m³, an estimated 10000 m³ of space a day would be required to store the fly-ash generated (Mills, 2016). Needless to mention the impracticality, the fly-ash thus needs to be transported

outside the plant to a location of temporary storage. From the electrostatic precipitator, the locus of its trapping, the fly ash is transported to a buffer hopper located generally within a distance of 100 meters, through a negative pressure (vacuum) pneumatic conveying process. A positive pressure pneumatic conveying system then transports it onward to the remote silo which is typically located within 500 meters to 2000 meters from the buffer hopper, as per the plant size and layout. In view of the presented economic advantage, through reduced or optimized air flow, fluidized dense phase conveying is currently seen to be the mode of choice for operation of pneumatic conveying systems for fly-ash. However, extensive usage of this particular mode remains restricted, due to non-availability of reliable models and scale-up techniques attributable to aforementioned complexities in fluidized dense phase pneumatic conveying. As such, the design of pneumatic conveying apparatus in power-plants frequently suffers anomalies.

Blow tank, an important component of a pneumatic conveying system, is a feeding device and has been seen to have a bearing on the overall performance of the pneumatic conveying system. Varying the total pressure at the starting position (material entry point) and therefore the total pressure drop in the conveyance pipeline, is responded to, through regulation of product discharge rate (m_s) by the blow tank (Wypych et al., 2006). Xu et al. (2013) found the pre-pressurization condition and initial aeration of the bow tank to have a considerable impact on performance in terms of the material discharge rate (m_s). However, the blow tank feeder, which is the centre-piece of a positive pressure pneumatic conveying system operating in fluidized dense phase mode and meters the quantity of material entering the conveying pipeline, has not been subjected to sufficient study in terms of the effect of aeration conditions like the relative proportions of top/side, fluidizing, conveying air flow rates fed to it, and pre-pressurization condition on material discharge and throughput rates. Thus, these parameters are not optimized for best results.

Taking cognizance of the multiple problems, including the aforementioned ones, in pneumatic conveying, that need to be addressed, NTPC Ltd., India's government owned electric utility enterprise, launched a research project in collaboration with the Thapar Institute of Engineering and Technology, Patiala to study, model and solve or remedy these problems. For the purpose a pilot test facility for pneumatic conveying was developed at the mechanical engineering

department of TIET, Patiala and an extensive testing program carried out. On-going are the analysis and modelling using data obtained from the testing program to furnish the results and conclusion to be delivered to the NTPC Ltd.

1.2 Objectives

With an aim to rid their plants of the aforementioned problems, 'Laboratory for particle and bulk solid technologies' at TIET, Patiala has undertaken research collaboration with the NTPC Ltd. Under aegis of the said project, the following objectives to be met are enlisted for this thesis work.

- 1) To design and develop pilot plant for dense-phase pneumatic conveying of fine powders.
- 2) To carryout conveying trials with fine powders with different blow tank aeration conditions.
- 3) To investigate into the effects of different blow tank aeration conditions on the conveyability of fine powders.

Chapter 2: Review of literature

A review of the literature relevant to pneumatic conveying, blow-tank aeration in particular, is presented here forthwith. At the outset, the theory of pneumatic conveying with development of perspective on construction and operation of pneumatic conveying systems, emphatically the blow-tank feeders, is presented. Subsequently relevant research corresponding to the blow-tank feeders is summarized.

2.1 Pneumatic conveying

Deriving the propulsive force from flow of a gas, either due to pressurization of the gas or by creation of suction (vacuum), the bulk goods can be made to translate through a pipeline; the statement constitutes the principle behind pneumatic conveying. A pneumatic conveying system is thus employed for movement of material between any two requisite positions and as such finds applications in chemical, pharmaceutical, mining and mineral industries to mention a few (Ratanayake, 2005). Majorly, pressurized air is used to propel the bulk solid, unless its use poses any threats such as unwanted chemical reactions with the product being conveyed, explosion and fire hazards. Furthermore, the material well suited for such a mode of transport typically is non-abrading, non-fibrous, and flowing. With the advancement in technology though, systems with low gas velocities operating in dense phase mode have been developed and have extended the compatibility of pneumatic conveying process with cohesive, abrasive and friable materials.

Enlisted are the salient features of a pneumatic conveying system.

- i) Pneumatic conveying allows for material transportation to otherwise difficult locations in terms of accessibility. Relative ease of piping installation and pliability presented by the piping network is not possible to be achieved with the mechanical conveying systems.
- ii) Dimensionally, a pneumatic conveying system is more compact and smaller than mechanical conveyors for example.

- iii) Being totally enclosed the possibility of environmental hazards is markedly less. Moreover, problems like contamination of the product, loss of material and generation of dust are thereby avoided (Ratanayake, 2005).
- iv) There is a possibility to have multiple locations for both, material uptake and deposit (Ratanayake, 2005)
- v) A lower maintenance cost is associated due to comparatively lower ingredience of motive parts.
- vi) Better and easy control and automation is possible due to constructional simplicity of the system in comparison to mechanical systems.
- vii) Dilute phase conveying presents the problems of material degradation, excessive consumption of energy, and erosion of the piping setup (Mallick, 2010).
- viii) Owing to the improper system design, dense phase systems often present piping vibration and blockades Wypych, 1989)
- ix) Due to complexity of flow seen in dense phase systems, design operation and maintenance of these systems requires a skill of high level (Mills, 2004).
- x) There is a lack of standardised literature for system selection specific to a certain use.

2.2 Classification of pneumatic conveying systems

Pneumatic conveying systems can broadly be classified into two types

- a) Positive Pressure Pneumatic Conveying Systems: Flow of gas through the pipeline in these systems is achieved by pressurization of the gas using an air mover, like for example, a compressor.
- b) Negative Pressure Pneumatic Conveying Systems: These systems use a negative pressure or vacuum to propel the gas (majorly, air) through pipelines. Clearly, gas laden with bulk material derives energy for motion through creation of vacuum.

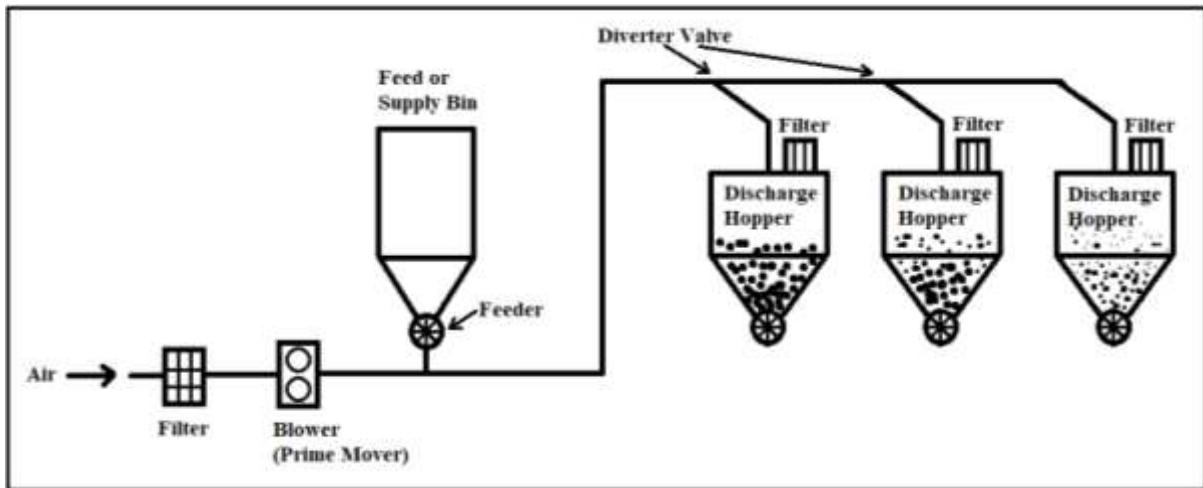


Figure 2.1 – Positive Pressure Pneumatic Conveying System (Klinzing et al., 2011)

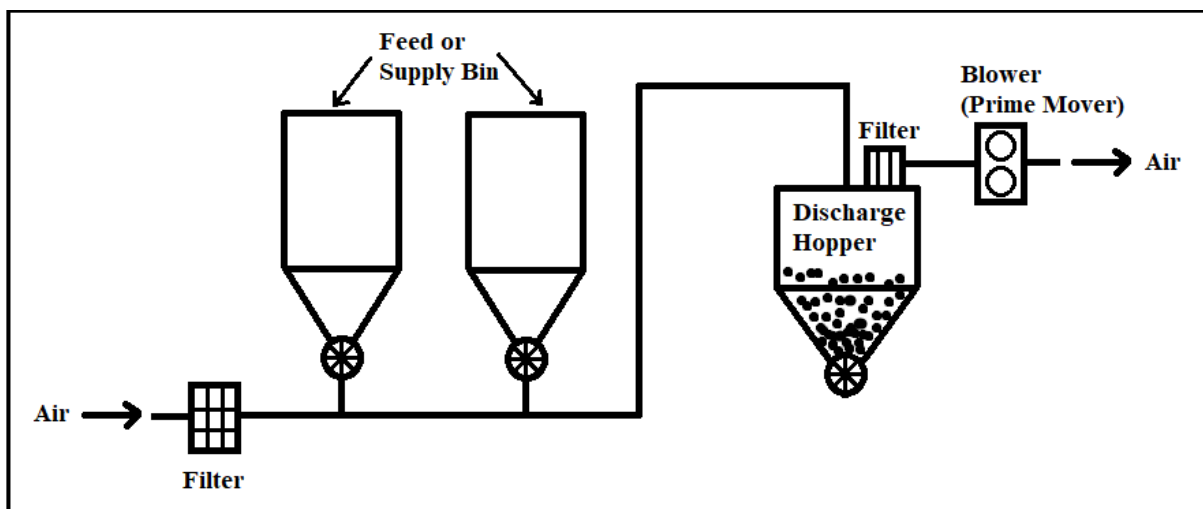


Figure 2.2 – Negative Pressure (Vacuum) Pneumatic Conveying System (Klinzing et al., 2011)

2.3 Major components of a pneumatic conveying system

The constructional ensemble of a pneumatic conveying system essentially consists of the following

- a) An equipment for gas pressurization or vacuum creation
- b) A piping network to channel the flow of this gas between desired locations of pneumatic transport

- c) A feeding device at initial position that feeds bulk material into gas flowing in the pipeline
- d) A separation apparatus at the final position that separates the bulk material from the gas solid suspension emerging from the pipeline.

An air receiver bin to buffer the supply of pressurized gas with fluctuating demand is generally a part of the system and placed between the gas pressurizer and the onward piping network. An air drier unit may be included for applications using air as the transportation medium allowing to dry air sufficiently before loading it with material so that material suffers no damage due to moisture, besides reducing the energy consumption for transport. Moreover, a piping circuit to control the pneumatically operated valves/equipment may also be required. A well designed automated control system is generally in place with most present day pneumatic conveying systems, in addition to the regular measurement instruments.

Gas Pressurization/Vacuum Creation

Pressurization of the gas or creation of suction for use in pneumatic conveying is usually accomplished by a prime mover such as a compressor, a blower or a vacuum pump. Selection of the class of equipment and the particular equipment per se is governed by the intended application and mode of conveying (Klinzing et al., 2009). The pressure requirement, flow rate requirement, demand fluctuation, adjustability to varied loads, response to fluctuating demand are some of the parameters that must be looked into while making the equipment selection. In most cases since air is the transport medium and is drawn directly at ambient conditions from the atmosphere, an air filter and air drier are integrated with this equipment.

Conveyance/Instrumentation Piping

Straight, horizontally or vertically aligned, and bent pipelines are used to route the material from a point of uptake to a desired position. The piping network thus established is called the conveyance piping network. However, it is pertinent to mention that such pipelines must be differentiated from pipelines preceding the feeding point that carry only air, and may also branch to supply air for, fluidization/aeration in the bulk material feeding device. Additionally a network of small pipelines may also be required for supply of air to control pneumatically operated valves/instruments through a control mechanism like a solenoid valve operated pneumatic panel. Generally, steel pipelines suffice for conveyance of majority of the materials. For cases where the conveyed product is abrasive and causes pipeline erosion it could be advantageous to use flexible piping such as rubber hoses. Such pipelines also help check the

degradation of a product, if friable. It is preferable to use a spun alloyed cast iron pipeline in the event of material being highly abrasive (Klinzing et al., 2009).

Material Feeder

In general, at the crux of design of a pneumatic conveying system lies the bulk material feeding device since it determines the overall performance of the system. The device should be selected for compatibility with both the system and the product being conveyed. A detailed discourse on feeding devices is presented in the section to follow.

Apparatus for separation of material from gas

At the position of its destination the transported material needs to be deposited and hence separated from the gas-solid flow or the suspension of which it is an ingredient at that point. This could be achieved using cyclone separators, bag filters or electrostatic precipitation. The method of choice is determined by factors like particle size of the bulk material, its flow rate and device collection efficiency (Mills, 2004)

2.4 Feeding devices

A feeding device or a feeder is one of the basic components of all pneumatic conveying systems that feeds bulk material into the conveyance piping network for onward pneumatic transport. Additively, the device serves to seal the material storage hopper. Figures 2.1 and 2.2 also depict the central nature of a feeding device in a pneumatic conveying system.

Pertinently, as the device begins to feed, solid particles are entrained into a speeding gaseous flow (powered by either gas pressure or vacuum). The particles thus entrained derive momentum from the gas flow, leading to a drop in pressure. This is augmented by presence of highly turbulent flow condition in the feeding zone. The highest fraction of pressure losses in majority of pneumatic conveying systems as communicated by Ratanayake et al. (2008) is attributable to loss of pressure at feeding. However, advantageously, feeding devices present the possibility for accurate metering of solid's flow rate into the conveyance pipeline (Klinzing et al, 2009).

There is a good diversity in devices available for feeding in pneumatic conveying systems and choice for one should be made based on system and material specific parameters. Enlisted in

figure 2.4 are some important feeding devices along with the respective ranges of system pressure under which they are well-suited to operate.

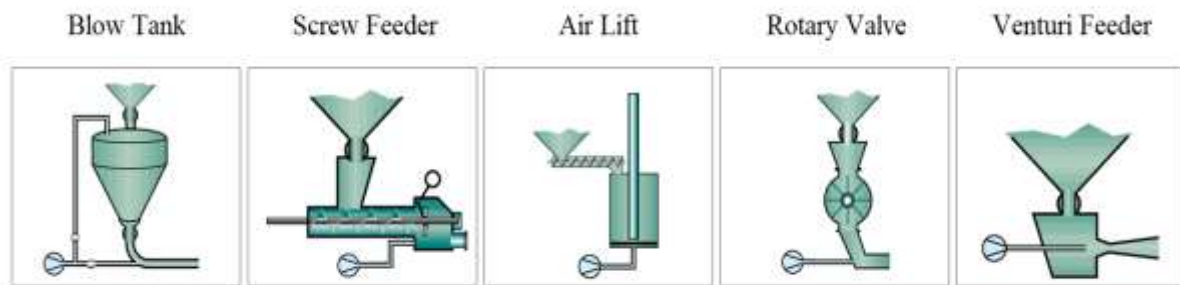


Figure 2.3 – Commonly used feeding devices

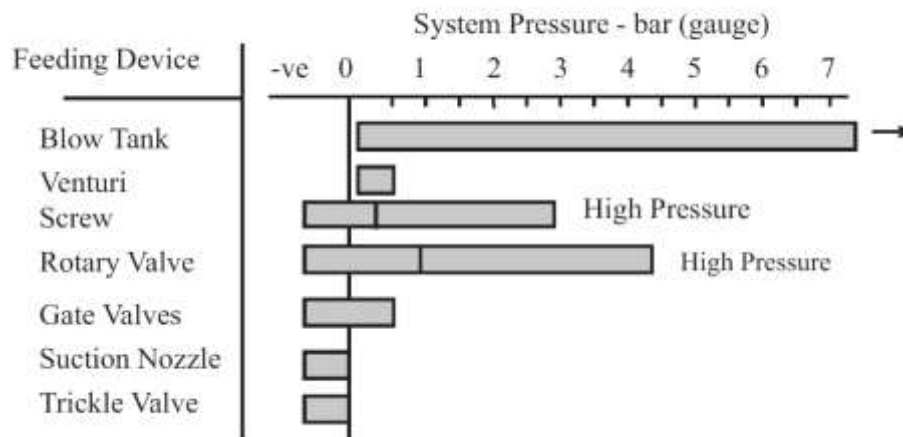


Figure 2.4 – Ranges of operating pressure for feeding devices (Mills, 2016)

As indicated in figure 2.4, system pressure provides one basis to classify the feeding devices. Klinzing et al. (2009) ascribed the low, medium, and high pressure categories to pressure ranges as under:

- a) Low pressure feeders: Vacuum to 100 kPa
- b) Medium pressure feeders: 100 to 300 kPa
- c) High pressure feeders: 300 to 1000 kPa

Further to this a categorization of feeding devices based on feed requirements is possible. For instance, in dilute phase systems a continuous feed rate, as uniform as possible, is required. Surges in feed rate in this case could lead to pipeline blockages. Klinzing et al. (2009) used a

vertical tandem blow tank feeder system to provide a continuous bulk solid supply. Still, certain feeding devices, additively serve the purpose of metering the solid into the conveying pipeline without requirement of any further equipment, unlike some devices that only serve the basic purpose of achieving air lock needed for operation of positive pressure systems.

Rotary valves and blow-tank feeders are the more commonly used feeding devices through an industrial viewpoint. A brief description of both follows.

2.4.1 Rotary valve

Also called the rotary airlock, the rotary valve is known to be the most common feeding device used in industrial pneumatic conveying. It is comprised of a rotor assembly housed in a fixed casing and primarily acts as an airlock. Rotors can be the open-end type or the closed-end type as depicted in figure 2.7. In a closed-end type rotor, two discs are welded to the shaft and ends of the blades giving rise to enclosed pockets, unlike the open-end type rotors in which the blades are directly welded to the rotating shaft. The former configuration, advantageously, presents a more rigid and stronger structure capable of withstanding higher pressures. The pockets formed between rotor blades are also of three types and are shown in figure 2.6. Selection of a particular type is governed by material type and operating efficiency considerations.

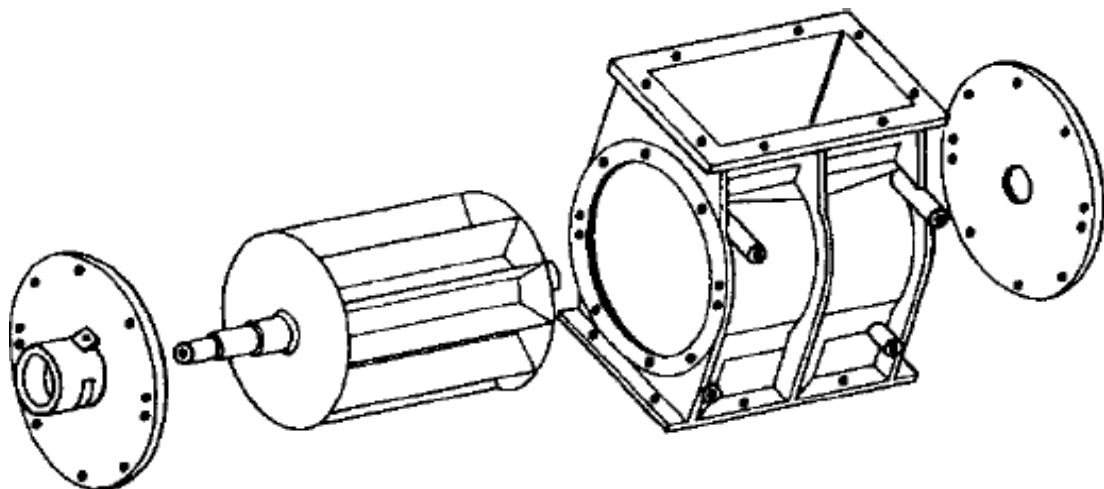


Figure 2.5 – A rotary airlock (Klinzing et.al, 2009)

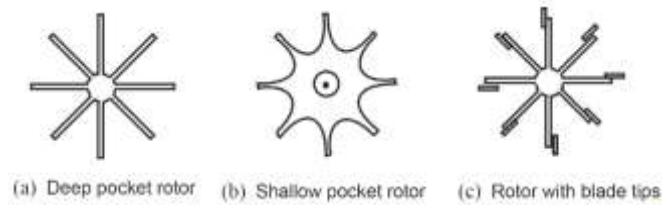


Figure 2.6 – Rotor pocket configurations (Mills, 2016)

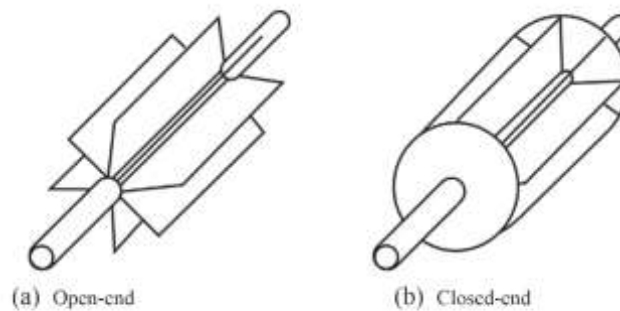


Figure 2.7 – Types of rotor (Mills, 2016)

Inherently, the rotary valves suffer the demerit of gas/air leakage in positive pressure pneumatic conveying and their use in vacuum pneumatic conveying systems presents an equivalent disadvantage (Mills, 2016). While manufacturers extend their best effort to keep the leaks under check through proper design and fabrication, customarily air leakage across the rotary valves is specified, for being taken into account in selection of prime mover being used to pressurize the air. A categorization of the leakages by Wypych et al. (2006) is given below

- a) Carry-over leakage: This is caused due to passage of air through empty pockets that are returning in course of rotation.
- b) Labyrinth Leakage: This occurs through the shaft seals in the casing and gaps between sides of blades and casing.
- c) Clearance leakage: This takes place through the clearances or gaps between blade tips and casing.

Some important and widely used rotary valve types are

- 1) Drop-through Valve: Well suited for solids the rotor pockets in this type of rotary valve are filled through an inlet port in the form of feeding tee, located above the rotor which is motor driven to transfer its contents to the outlet and thus to conveyance pipeline.

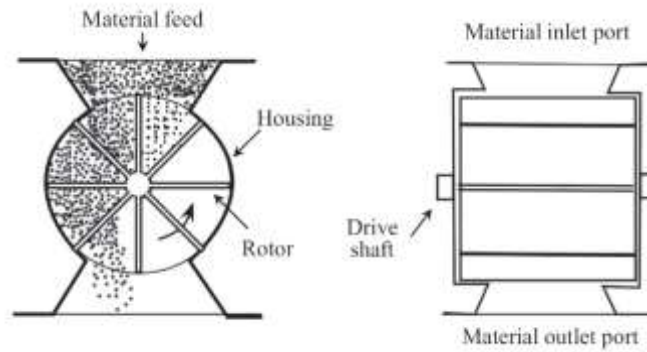


Figure 2.8 – Drop-through rotary Valve (Mills, 2016)

- 2) Offset Valve: Used for pelletized goods, this valve ensures that shearing of the material is circumvented. The inlet of such a valve is offset to its side, so that flow angle of the material prevents complete filling of the rotor pockets which rotate causing product flow into the rotor trough, thereby preventing shearing.

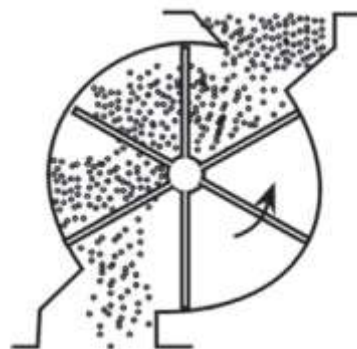


Figure 2.9 - Offset rotary valve (Mills, 2016)

- 3) Blow-through Valve: Suitable for cohesive products, the assembly of this valve permits the conveying air to pass through and purge the pockets discharging material so that material is entrained into the conveying pipeline essentially within the valve.

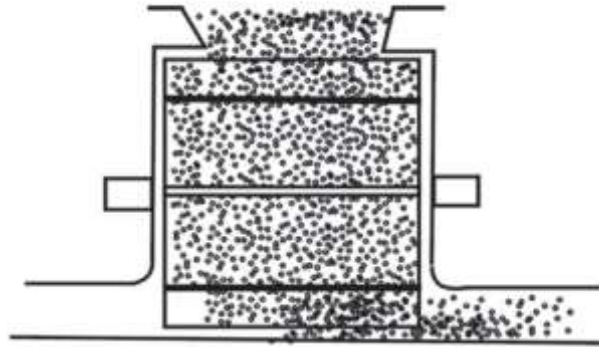


Figure 2.10 – Blow-through rotary valve (Mills, 2016)

2.4.2 Blow Tank

Also known as the airlock vessel, the blow tank is basically a pressure vessel with a conical geometry at the bottom as shown in figure 2.11

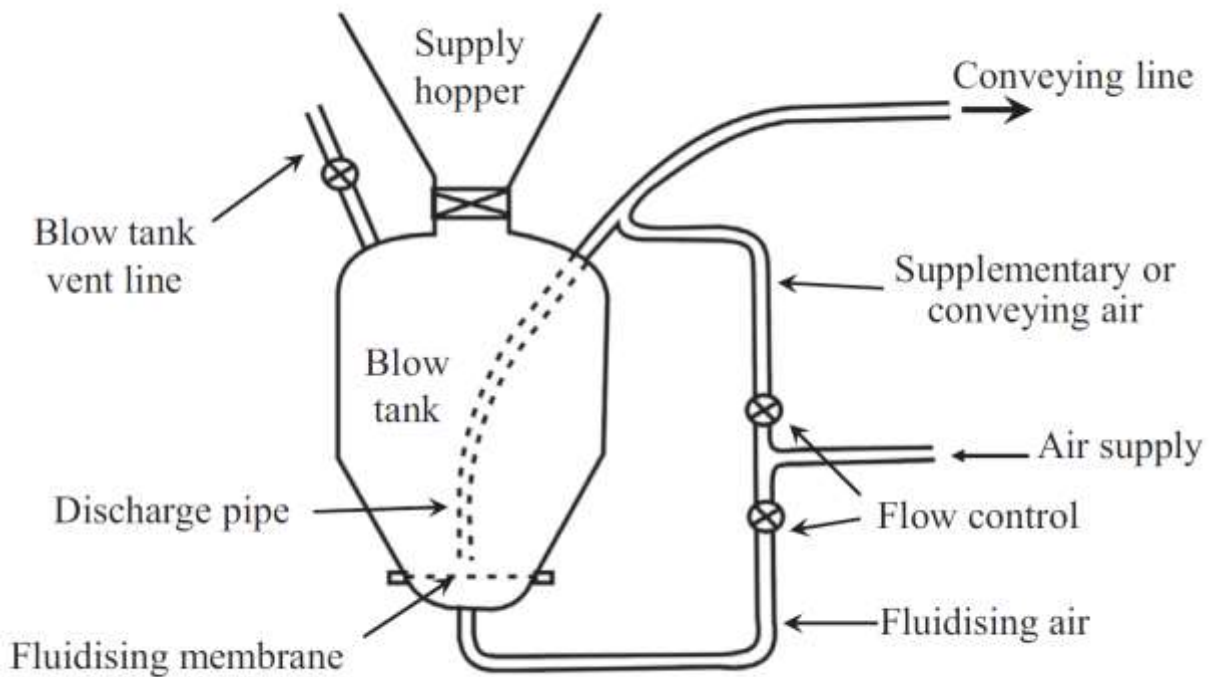


Figure 2.11 – A Blow Tank Feeder (Mills, 2016)

The capability of blow tanks to use highly pressurized air makes them well suited for use as feeding devices in pneumatic conveying. Furthermore, blow tanks present the merit of flexibility and may also be used in low pressure applications. The fact that a blow tank has no moving parts yields the merit of substantial reduction in wear and material degradation. Blow tanks are of numerous types and each can be operated in alternative configurations. Having similar basic features, these different types however, vary in the conveying capabilities offered and control characteristics.

Two basic types of the blow tank are the bottom discharge blow tank and the top discharge blow tank. In the top discharge variant a very efficient fluidizing mechanism is required for discharge of material through the top and its subsequent ingress into the conveyance line. The bottom discharge blow tank, which is in wider use, discharges material through its bottom, generally by operation of a material discharge valve.

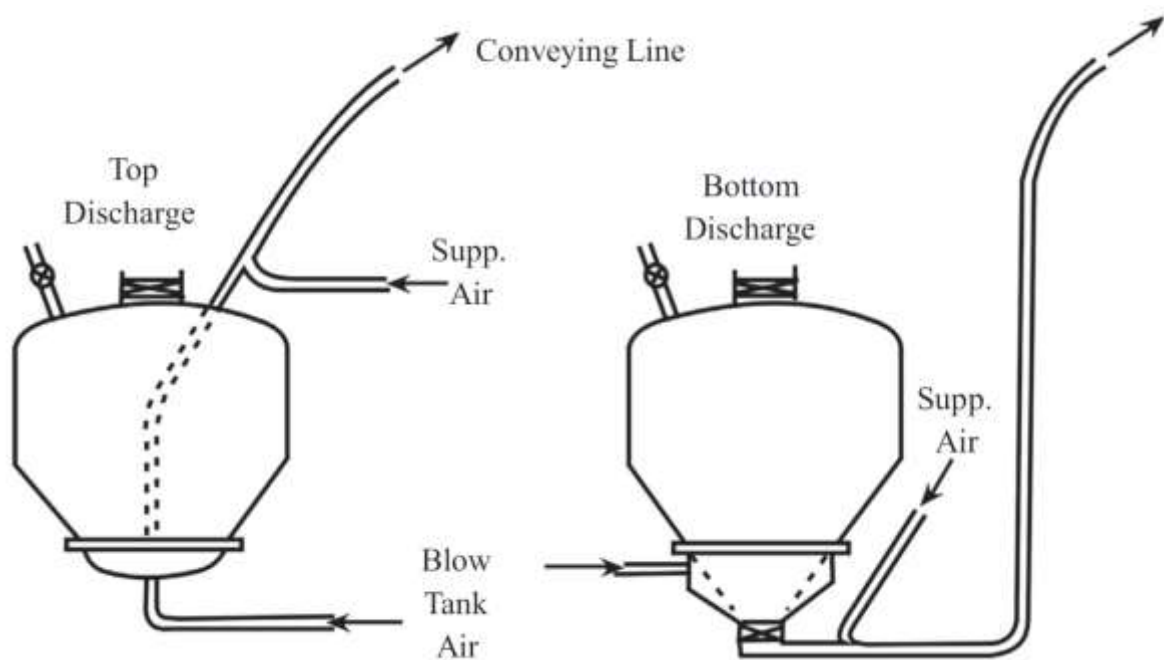


Figure 2.12 – Top and bottom discharge type blow tank feeders (Mills, 2016)

For a bottom discharge blow tank fit with a discharge valve, figure 2.13 depicts typical plots of air flow rate, pressure and weight against time.

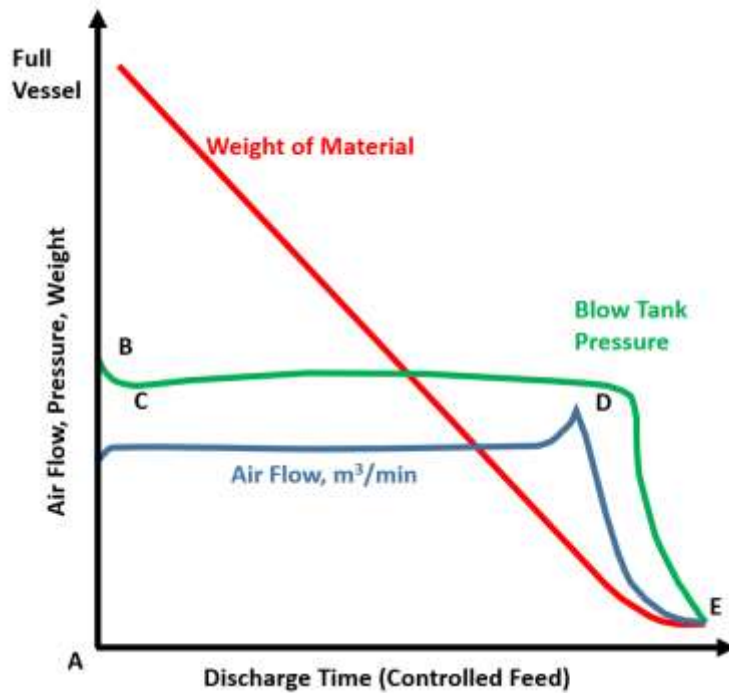


Figure 2.13 – Characteristic curve of the blow tank (Klinzing et al., 2009)

Salient points on the figure indicate the important stages in blow tank operation and the plots define the performance of the blow tank through these stages. Prior to discharge the blow tank is pressurized to a point B which represents the initial pressurization P_{initial} . A drop in pressure indicated by the transition from point B to C takes place across the discharge pipe of the blow tank on opening of the discharge valve. Pertinently, the value of C gives an average pressure within the vessel, as it is feeding a gas-solid flow to the conveyance pipeline. Clearly the major portion of the material to be fed within a cycle of operation of the blow tank is accomplished in the phase marked by transition from point C to D. Drop in blow tank pressure commences at point D while the material still remnant in the vessel is flushed out into the conveyance line. Flow of material at high velocity and in dilute concentration, at this stage can cause a violent surge. This factor must be accounted for in selection of the appropriate separation system for separation of solid from gas (Klinzing et al., 2009). The transition from D to E on x-axis gives the depressurization time, while the transition from C to E marks the total discharge time.

Three design parameters affecting the blow tank performance are the cone angle, vessel pressure and vessel volume (Mc Lean, 1985). Improper design leads to problems such as satholing (prevention of material flow due to formation of a hole through the material) and

bridging (also called arching, leads to prevention of material flow due to formation of a barrier across the vessel outlet).

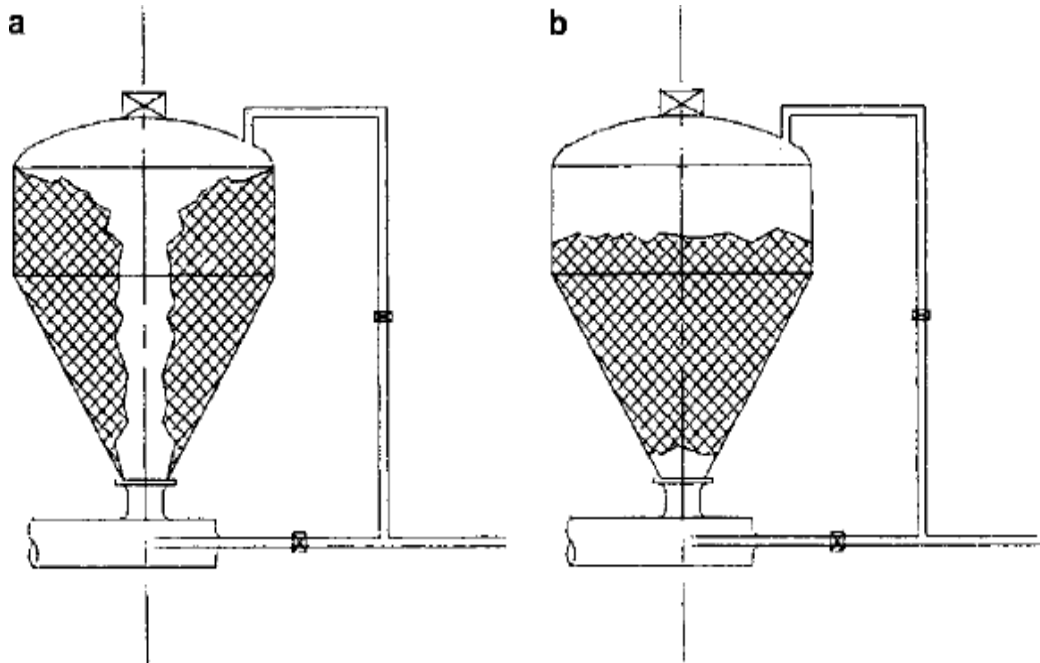


Figure 2.14 – Ratholing (a) and arching (b) in blow tank feeders (Klinzing et al., 2009)

The cubic capacity of the blow tank vessel and the nature of the material being conveyed have a considerable bearing on the performance of the vessel. For materials that undergo fluidization with ease, a higher cubic capacity vessel, as far as possible, is preferred since this ensures a greater time component of discharge compared to clean-out and pre-pressurization in a blow tank operation cycle, thereby leading to improved efficiency in conveying. For the case of free flowing materials, selection for volume of the blow tank is made accounting for the fact that the product bears a flooding tendency and may overflow the conveying pipe causing blockage, conditions which need to be avoided inevitably. Higher pre-pressurization leads to higher discharge rates of the material, and augments long distance conveying, while other system parameters are held constant.

2.5 Summary of blow tank research literature

Tomita et al (1978) experimented using a top discharge semi-cylindrical blow tank with a transparent frontal plane in pneumatic conveying of bulk solids like glass beads, cement raw materials, phosphorous ore, polythene pellets and PVC powder. They observed the mass flow rate of solid discharge from the blow tank to be independent of drop in pressure in the conveyance pipeline. However, solid discharge rate was seen to be dependent on the velocity of air supply to the blow tank. The value of pressurization required in the blow tank for a particular conveying trial was seen to be determined by both the discharge mass flow rate of material from the blow tank and the drop in pressure in the conveyance pipeline. The feeding configuration of the blow tank differed between fine powders and the granular material. The former were discharged into the pipeline in the form of blocks. For the latter a continuous flow of particles into the pipeline was observed. In the range of low velocities of air supplied to the blow tank, the solid discharge rate from the blow tank increases proportionately with increase in supply velocity of air. However, at high air supply velocities to the blow tank, material in proximity of the nozzle is fluidized, due to which bubbling occurs and causes solid discharge rate from the blow tank to increase less rapidly with increasing velocity of air supply to the blow tank.

Mc Lean (1985) extended the design theory of gravity flow bins to come up with a procedure of design for both top and bottom discharge blow tanks. The key to modification of design was incorporation of fluid pressure gradients existing in blow tanks. Recommendations were made to design the blow tank for use with cohesive as well as non-cohesive bulk solids. Parameters such as volumetric capacity of the blow tank, its outlet diameter and cone angle were seen to have a considerable effect on the blow tank performance and problems such as arching and rat-holing encountered in bins in general. The design principles presented by the author were intended to serve as a guide in selection of blow tank suited to a specific application besides reducing the cost implications and increasing efficiency.

Jones et al. (1987) carried out experimentation to directly compare the feeding potentiality of top and bottom discharge configuration blow tanks. Same blow tank, that could be arranged to

operate in both, top and bottom discharge configurations, was used for the purpose while minimal change to pipeline geometry was ensured. Remainder of the conditions/parameters were held constant for both the experiments. From a comparison of blow tank and conveying characteristics, thus generated, it was observed in conveyance of pulverized fuel ash (a material with good aeration properties) that for the same air supply rate to the blow tank i.e. at same blow tank air ratio (BTAR) and same mass flow rate of air, operation in top discharge configuration yielded significantly greater feed rate into the pipeline. A probable reason cited was the ineffective fluidization of material in the bottom discharge blow tank. This conclusion was mentioned to be even more significant to dense phase conveying since the product conveyed i.e. pulverized fuel ash is representative of products suited for conveyance through dense phase mode. The pneumatic conveying characteristics in two experiments did not exhibit any difference except in the fact that for the top discharge mode the characteristics are spread over a larger range of operating conditions.

Wypych (1999) investigated the possibility of long distance pneumatic transport of powders with a high throughput. Suitability of blow tank for the purpose was assessed with specification of an appropriate blow tank design. Long distance transport at a high capacity, using a blow tank feeder, resulted in incomplete/inefficient feeding from each of the blow tanks connected in series or parallel. This was explained to be due to rat-holing in the feeder or funnel-flow caused by low level of pressure drop across the material bed, nature of material or faulty method for air injection into the feeder. Use of a cone dosing valve was seen to make an effective provision for the method of control and metering of material feed into the conveyance pipeline, in particular at start-up, besides helping sense and therefore address pipeline blockages. Further using a semi-empirical approach total pressure drop in the pipeline has been predicted with better accuracy, that has helped design-optimization of stepped diameter pipelines used in the experiment, to yield the merits of reduction in air flow, velocity, power, product damage and wear. A back pressure unblocking system used has led to marked reduction in down time for instances of pipeline blockage.

Wypych et al. (2006) conveyed material consisting of mono-sized poly pellets having a loose poured bulk density $\rho_b = 565 \text{ kg/m}^3$, particle density $\rho_s = 918 \text{ kg/m}^3$ and particle diameter $d_p = 5 \text{ mm}$, to investigate feed rate capacities obtainable in low velocity pneumatic transport, using

a drop-through type rotary valve and a bottom discharge blow tank feeder, through a common conveyance pipe. The rotary valve used has a nominal bore of 250 mm and the volume of the bottom discharge type blow tank was 1 m³. It was established that using the blow tank feeder, by far a higher magnitude of material mass flow rate for a given mass flow rate of air is obtainable, compared to the rotary airlock. This is explained by the restriction posed by the rotary valve in feeding of product into the conveying pipeline, besides metering the material in 'lumps' unlike the blow tank that feeds unrestricted, directly and continuously. Also, attributable is the fact that pre-conveying the blow tank is pressurized unlike the rotary valve which relies on gravity filling. Experiments with varying aeration conditions in terms of top, conveying and cone air quantities supplied to the blow tank were conducted and it was concluded that aeration conditions have a strong bearing on blow tank performance and control. Contrary to the theoretical expectations, speeding up the rotor of the rotary valve did not yield corresponding increase in feed rate beyond a certain limit due to carryover of air and hence resistance to material flow besides incomplete filling of valve pockets due to increased speed. An arrangement was made to modify the rotary valve, which was now pressurized at the top side to a value in nearness of back pressure of the system. Even though the modification improved the feed rate of the rotary valve, it was still far lower than what was obtained using the blow tank.

Ratnayake et al (2008) came up with an equation that calculates the drop in pressure at the entrant part of pipeline feeding the blow tank. The blow tank was of top discharge configuration and the materials conveyed were alumina, ilmenite, cement and barite. Buckingham's pi theorem in dimensional analysis, was used to relate 11 variables and arrive at a functional relationship between them that enables to obtain the aforementioned drop in pressure through formalization of a theoretical model. Combinations of different pipeline diameters with different materials were used to carry out further experimentation data so generated was recorded. This was then validated against the results obtained theoretically by use of the previously devised model. Experiments were seen to be in agreement, within 10% accuracy level, with the modelled equation in case of ilmenite. For the other three materials (alumina, cement and barite), agreement between experiments and the modelled equation within 15% accuracy level was observed.

Xu et al. (2012) conveyed glass beads and pulverized coal to study the effect of properties of the material on performance in pneumatic conveying. Also studies was effect fluidizing and pressurizing gas flow rates have on the mass flow rate of material (m_s) and solid to gas ratio (m^*). With an increase in flow rate (Q_f) of fluidization gas, m_s and m^* , registered at first an increase, and subsequently a decrease owing to an incomplete fluidization of material at riser inlet at low Q_f . An increasing flow rate (Q_p) of pressurizing gas cause m_s to increase to undergo a gradual increase, while m^* was seen to increase at first and decrease subsequently, explained by a slightly increased blow tank pressure due to increased Q_p , and thus an accompanying increase in m_f . The values for m_s and m^* for pulverized coal were seen to be lower than the corresponding values for glass beads, suggesting better flowability of glass beads. At the powder bed heights, above a critical surface near the conjunction of cylindrical and conical parts, and at low Q_f , funnel flow was observed. At large Q_f however, the flow regime seen mass flow. At powder bed heights below the critical surface, essentially funnel flow was observed. At powder height in the range of the height of riser inlet intense fluidization of the remaining powder could be seen with formation of bubbles

Xu et al. (2013) experimented to determine the effect on conveyance properties, like the mass flow rate of material (m_s) and solid to gas ratio (m^*), by a variation in parameters like riser diameter (d_r), height of the riser inlet relative to gas distribution plate (H), flow rate of the supplementary gas supply (Q_s) and the content of external moisture in material conveyed (pulverized coal). A see-through blow tank made up of plexi-glass was used in experimentation, with an aim to provide the possibility of observing the flow patterns of the product in the vessel. It was found that an increase in height of the riser inlet (H), led to an increase in both, mass flow rate of material (m_s) and solid loading ratio (m^*) first, and subsequently a decrease in both. A probable explanation could be the fact that a lower H means closer proximity between riser and the gas distribution plate, permitting escape of a part of the gas through the riser with no involvement in fluidization. Moreover, at higher values of H , the gap between the riser and gas distribution plate increases causing it partly to remain unaffected by fluidization. An increase in moisture content also caused both m_s and m^* to first increase and then decrease. An increase in the rate of flow of supplementary gas led to decrease in both m_s and m^* . A model for prediction of m_s within -25% to 15% accuracy level was also presented.

Chapter 3: Experimental work

As a part of the broader experimental plan, experimentation for this thesis work was conducted using the test rig built at ‘Laboratory for Particle and Bulk Solids Technologies’, with fly ash obtained from the remote silo of a power plant located in north India. Experimentation was aimed at generation of data, which will help understand the effect of variations in side, top, fluidizing and conveying air flow rates on material discharge rate delivered by the blow tank feeder. Furthermore, experiments leading to characterization of the product intended to be conveyed were carried out.

3.1 Experimental setup

Two test rigs, one for vacuum conveying and one for positive pressure conveying were built, as a major objective of this thesis work, and made available for experimentation. The positive pressure test rig is representative of a typical pneumatic conveying pilot setup, employing a 0.8 m³ blow tank feeder which is fitted with trough type fluidizers, and feeds through a double disc valve into the conveyance pipeline. On top of blow tank is mounted the material receiver bin that stores the bulk solid and transmits it into the blow tank, by gravity assisted flow, as the knife gate valve and the pneumatically operated double disc valve linking the two vessels are opened. A bag filter assembly mounted on top of the receiver bin is employed for gas separation from material and subsequently its exodus into the atmosphere. Figure 3.1 gives the layout of the positive pressure system.

Load cells placed underneath both the material storage and feeding vessels at specified points, transmit the load to a data logging device, from which the data is retrievable through a computer. Pressure transducers at salient points, as shown in figure 3.1, report data in a similar way. The piping and instrumentation diagram, shown in figure 3.2, depicts the pipeline connections between various components of the positive pressure system and the relative locations of various flow controlling, metering and measurement equipment.

3.2 Cycle of system operation

Varied combinations of top, side, fluidizing, conveying and total air flow rates, obtainable through valve adjustments, may be used for experimentation depending upon the research goal. Furthermore, the supply pressure of air is controllable using pressure regulating valves and can be set to a value desired. For a particular selected combination, for one cycle of experimentation, the manually and pneumatically operated material inlet valves are both opened with a certain delay after admittance of air into the blow tank and conveyance pipeline, to allow for achievement of desired blow tank pre-pressurization. As the valves are opened, the manually preloaded bulk solid in the receiver bin begins to flow downwards into the blow tank under the effect of gravity where it undergoes fluidization augmented by supply of fluidizing air, and experiences a resultant downward push due to supply of top/side air to the blow tank feeder. Subsequently the valves between the two vessels are closed, which is followed by opening of the material discharge valve at the exit of the blow tank, at which point feed of material into the conveyance pipeline starts and subsequently the entire material is deposited back into the receiver bin after transportation through the conveyance piping loop. Opening of the vent valve in the pipeline connecting the blow tank and the receiver bin causes release of pressure remnant in the blow tank to the receiver bin where added income of pressurized air augments fluidization process. At this point a cycle of system operation is completed. Pertinently, fluidizing air is also supplied to the receiver bin to cause material fluidization and therefore facilitate material passage to the blow tank.

Furthermore, it is instructive to include the layout drawing of the negative pressure pneumatic test rig also. Figure 3.3 shows the system and piping loop layout of the vacuum pneumatic conveying test setup.

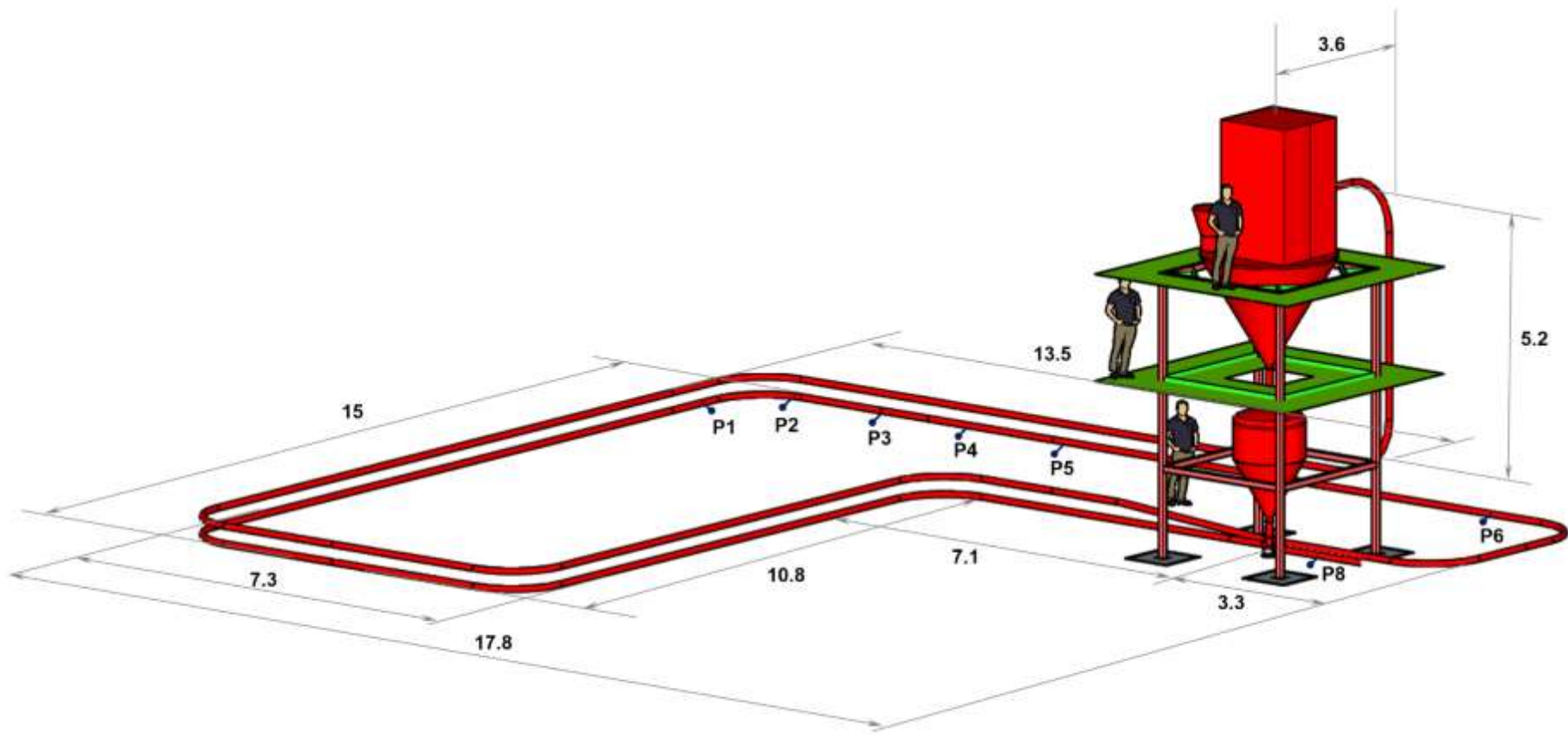


Figure 3.1 – Conveyance pipeline layout of the positive pressure pneumatic conveying test rig

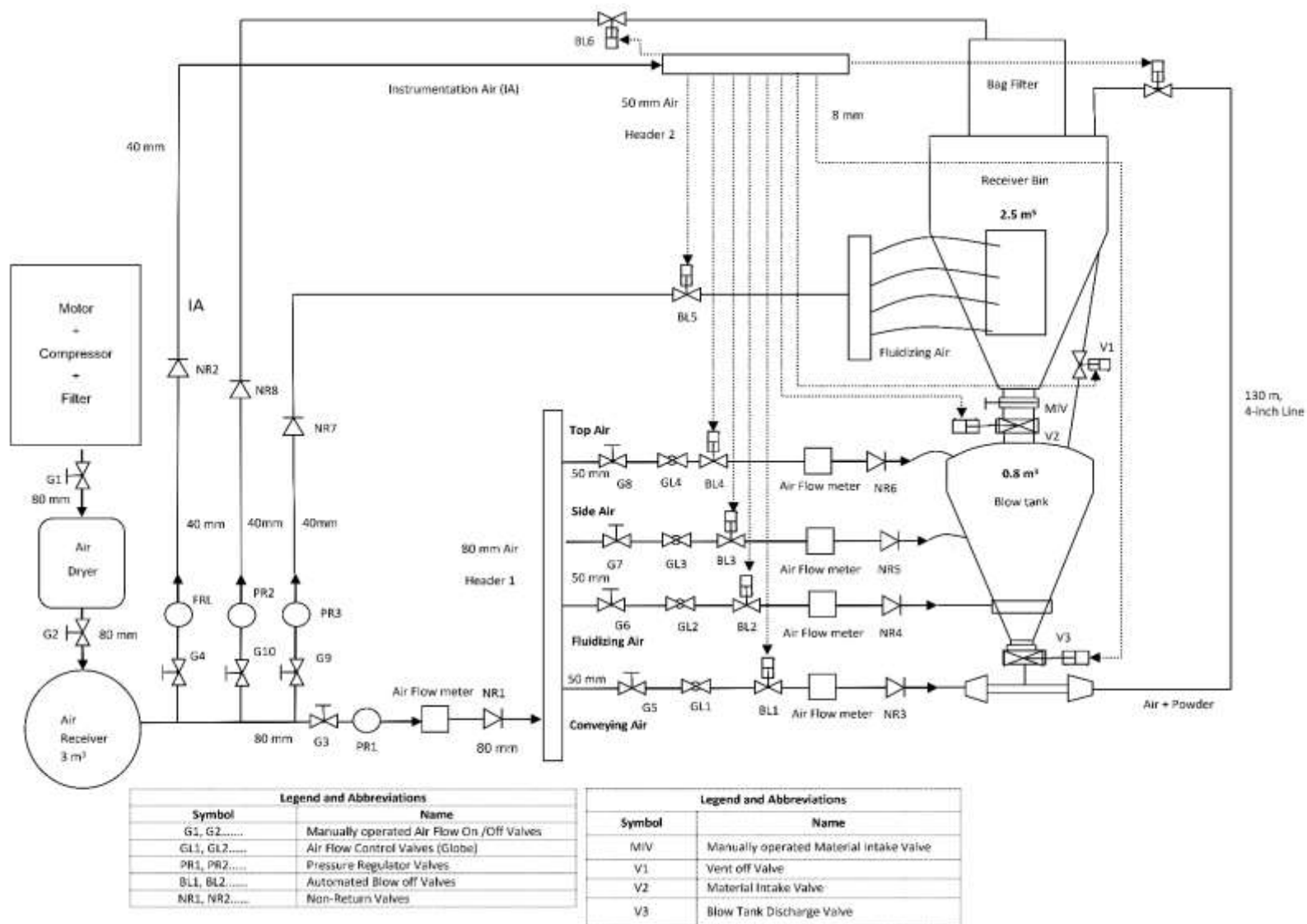


Figure 3.2 – Piping and instrumentation diagram of the positive pressure pneumatic conveying test rig

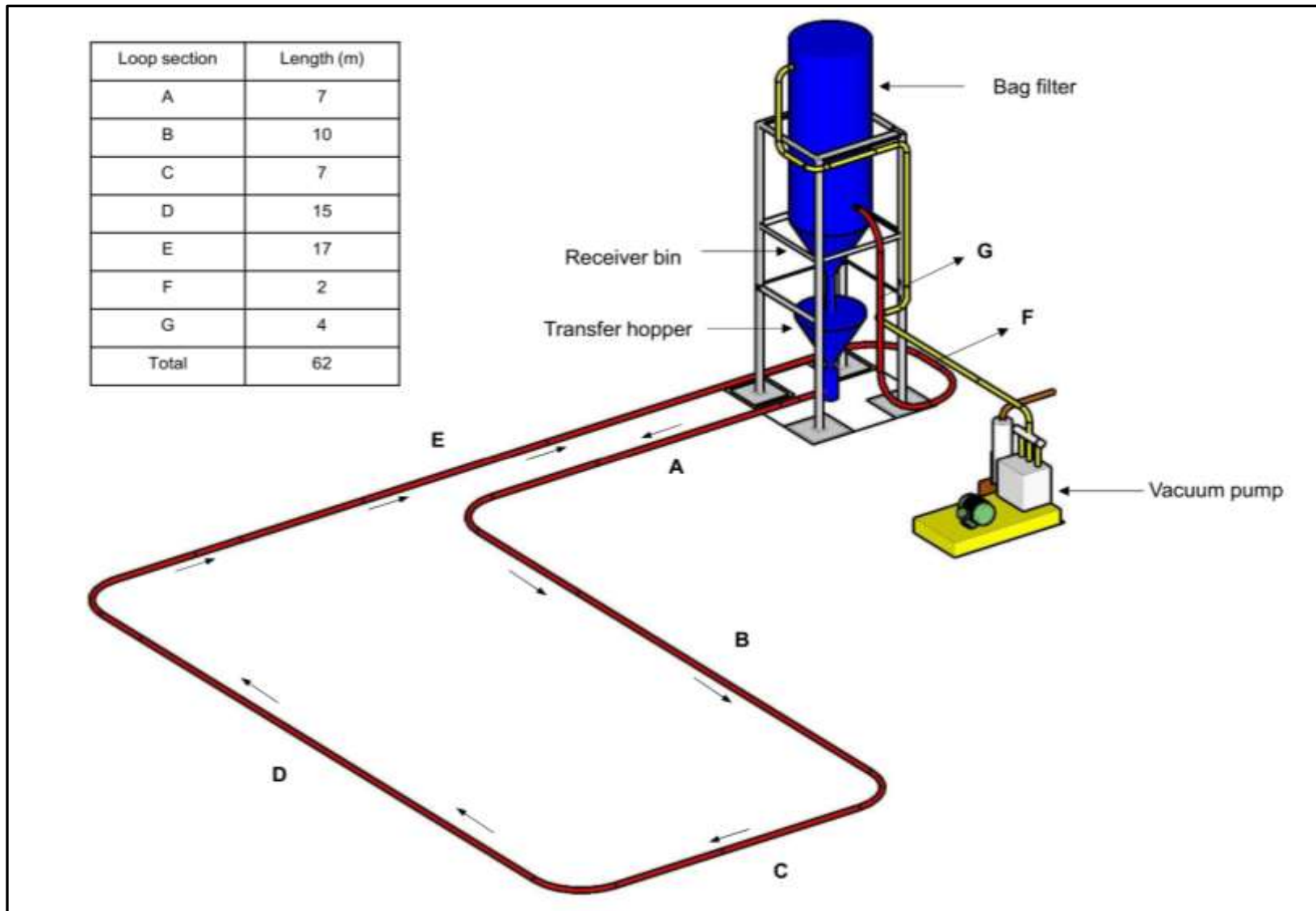


Figure 3.3 – Conveyance piping layout of the negative pressure pneumatic conveying test rig

3.3 Experimental program

With the compressor set at maximum delivery pressure of 7 bars the air receiver tank is initially pressurized to 7 bar, the system being in non-operating condition. The supply pressure to the system can thus be set at or below 7 bar using the pressure regulating valve. Experiments were conducted with conveyance of material through the system (gas-solid flows), using different combinations of various air flow rates. Table 3.1 furnishes the developed experimental plan, well suited to generate data for carrying out the proposed study.

Table 3.1 – Experimental plan for pneumatic conveying experimentation

Experiment No.	Total air	Aeration	Pre-pressurization	Conveying Air	Side/Top Air	Fluidizing air
1	25%	side air	10%	80%	20%	10%
2	25%	side air	10%	80%	20%	10%
3	25%	side air	20%	80%	20%	10%
4	25%	side air	30%	80%	20%	10%
5	25%	side air	30%	80%	20%	10%
6	25%	side air	30%	80%	20%	0%
7	25%	side air	30%	80%	20%	20%
8	25%	side air	30%	80%	0%	20%
9	25%	side air	30%	80%	20%	10%
10	25%	side air	30%	60%	40%	10%
11	25%	side air	30%	40%	60%	10%
12	25%	side air	30%	20%	80%	10%
13	25%	side air	30%	10%	90%	10%
14	25%	side air	30%	0%	100%	10%
15	15%	side air	30%	20%	80%	10%
16	20%	side air	30%	10%	90%	10%
17	30%	side air	30%	20%	80%	10%
18	30%	side air	30%	10%	90%	10%
19	15%	side air	30%	30%	70%	10%
20	50%	side air	30%	10%	90%	10%

21	20%	side air	30%	20%	80%	10%
22	25%	side air	30%	10%	90%	20%
23	25%	side air	30%	10%	90%	20%
24	25%	top air	30%	80%	20%	10%
25	25%	top air	30%	60%	40%	10%
26	25%	top air	30%	40%	60%	10%
27	25%	side air	30%	80%	20%	10%
28	25%	side air	30%	80%	20%	10%
29	25%	side air	30%	80%	20%	10%
30	25%	side air	30%	80%	20%	10%
31	25%	side air	30%	80%	20%	10%
32	25%	side air	30%	80%	20%	10%
33	25%	side air	50%	80%	20%	10%
34	25%	side air	70%	80%	20%	10%
35	30%	side air	10%	80%	20%	20%
36	25%	side air	10%	80%	20%	20%
37	20%	side air	10%	80%	20%	20%
38	30%	side air	10%	20%	80%	20%
39	25%	side air	10%	20%	80%	20%
40	20%	side air	10%	20%	80%	20%

3.4 Characterization of the product

Characterization of the bulk solid to be conveyed, through its physical properties relevant to design of the system, is absolutely essential and important. Conveying capability of the system could show a marked change as the material conveyed is changed. It is thus instructive to identify a material using these important properties rather than names, which for example may be soda-ash or fly-ash (Mills, 2016). Alongside the pneumatic conveying experimentation, powder characterization of the used fly ash sample through following properties was carried out.

3.4.1 Particle size distribution (PSD)

Particle size distribution gives the distribution of fractions of particle size that make up the totality of the product. Typically, it is represented on a percentage mass basis and is given as the percentage by mass of material less than or greater than a specified size. Mean particle size is the size corresponding to 50% value. A Malvern Mastersizer 2000, manufactured by Malvern Instruments, was used to obtain the particle size distribution of the product conveyed. The instrument has a measurement range of 0.020 to 2000 μ m. With refractive index 1.52 and absorption index 1, it works on the principle of laser diffraction for measurement of particle size and specific surface area of bulk solids.

3.4.2 Loose poured bulk density

Bulk density is the ratio of mass of a sample of material to the volume it occupies. In absence of any compacting force it is referred to as the ‘as poured’ or ‘loose poured bulk density’. For calculating sample weight, the weight of hollow cylinder subtracted from the weight of cylinder containing sample. The volume of sample noted from cylinder dimensions. Loose poured bulk density measured by using formula –

$$\rho_b = \frac{\text{Mass of sample}}{\text{Volume of loose sample}} \quad (3.1)$$

3.4.3 Tapped bulk density

The bulk density attained after mechanical compaction due to some compacting force is called tapped bulk density. A container containing the powder sample was tapped to provide for the compacting force for measurement of tapped bulk density in the present study. It was then calculated as

$$\rho_t = \frac{\text{mass of the sample}}{\text{Tapped volume of powder}} \quad (3.2)$$

Minimum volume obtainable after tapping depends on no. of parameters such as particle size distribution, true density, particle shape and cohesiveness due to surface forces. Tap density thus has an important bearing on behaviour of the bulk solid.

3.4.4 Particle density

Particle density is the ratio of mass of a particle to the volume occupied by it. The classic method to measure particle density called the water displacement method has been employed in this study. Water is taken in a measuring tube and its mass and volume recorded. A known mass of the sample is now introduced into this tube. After the air bubbles collapse and particles settle down, mass and volume of the contents of the tube are again recorded. Particle density is then given as

$$\rho_p = \frac{\text{Mass of the sample and water} - \text{mass of water}}{\text{Volume of the sample and water} - \text{Volume of water}} \quad (3.3)$$

Chapter 4: Flow Metering – Design of sonic nozzle bank

4.1 Introduction

A sonic nozzle, also called a critical flow venturi nozzle (CFVN), is a nozzle through which a gas can be made to flow at a constant flow rate by mass, at and beyond a certain minimum drop in pressure across the section of the nozzle up to its throat (minimum area of cross-section). The maximum value allowable for pressure at the throat of the nozzle to attain this constant rate of mass flow is called the critical pressure, and on its attainment, the nozzle is said to be in choked condition, metering a constant and maximum flow rate of mass, for a given pressure differential across the nozzle, with flow velocity at throat being equal to speed of sound at local conditions of pressure and temperature. Sonic nozzles are widely used in industry as calibration standards for flow measurement devices, particularly gas flow meters, and also as flow metering and measurement devices. Sonic nozzles have been used in pneumatic conveying for admission of air/gas at pre-set mass flow rates through the pipelines of experimental interest (Kennedy,1998). Wpych and Hastie (2002), used an assembly of six sonic nozzles for supply of air to the test rig, as demanded by the range of operating conditions prescribed in the test program. Hastie et al. supplied compressed air to the test rig for their experimentation in pneumatic conveying through an assembly of six sonic nozzles, for control of mass flow rate and its adjustment to various required values. This chapter furnishes the design of an assembly of five sonic nozzles that are required to be used to meter specific values of mass flow rate of

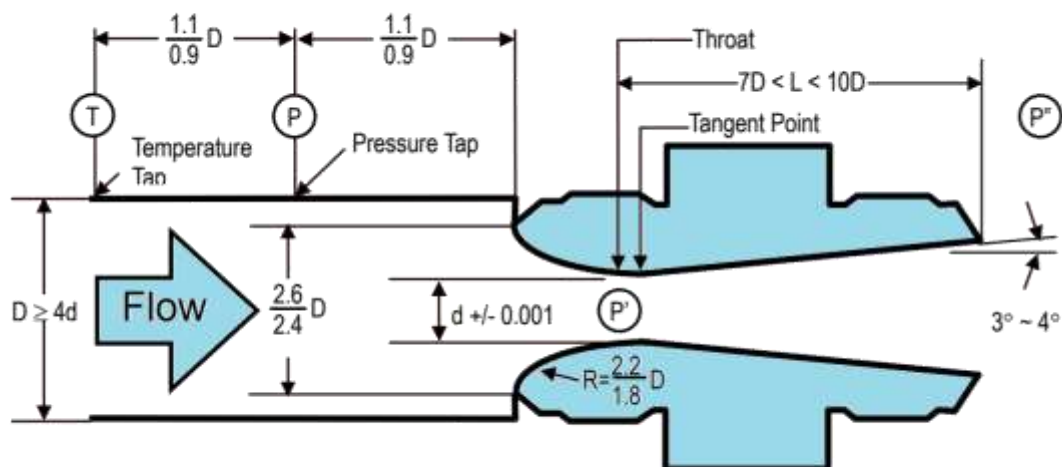


Figure 4.1 – A Critical Flow Venturi Nozzle

air for the broader pneumatic conveying test program to be undertaken at the ‘Laboratory for Particle and Bulk Solids Technologies’.

4.2 Theory

The functioning of a sonic nozzle is essentially based on the behaviour of a compressible flow through a duct with changing area of cross-section. With the assumptions of flow being steady, one-dimensional with no body forces acting in its direction, and occurring through a reversible adiabatic mode (isentropic flow), the governing equations thus obtained may be used to arrive at the following results.

$$\frac{dP}{\rho V^2} = \frac{dA}{A} \frac{1}{(1-M)^2} \quad (4.1)$$

$$\frac{dV}{V} = -\frac{dA}{A} \frac{1}{(1-M)^2} \quad (4.2)$$

Equations 4.1 and 4.2 respectively define the change undergone by pressure and velocity of flow, as the area of cross-section of flow is changed along the axis of flow, starting from the locus with cross-sectional area A , flow velocity V and Mach number M . A little consideration will show that for subsonic flow ($M < 1$), a convergent channel acts as a flow accelerator while as divergent channels decelerate the flow. In case of the supersonic flow ($M > 1$), it may easily be inferred that the vice versa occurs. For the case of sonic flow ($M = 1$), as sonic velocity is approached, from either initially a subsonic, or a supersonic flow condition, by equations 4.1 and 4.2 an infinite change in pressure and velocity respectively, must be registered. This, however, is clearly unrealistic. Sense can only be made if area of cross-section is required to be constant, i.e. $dA = 0$, as flow attains sonic velocity ($M = 1$). Furthermore, a transition from subsonic to sonic conditions requires a flow acceleration using a converging cross-section of flow provided by a converging duct, while as a transition from supersonic flow to sonic requires flow deceleration, which in this case is again provided by a converging duct. Clearly, sonic speed occurs not only at a constant area of cross-section but also at the minimum cross-sectional area, which is provided by the throat of the duct.

When the flow at the throat of a duct attains the sonic velocity, the condition of flow at throat is termed critical, since it is at this juncture that the maximum possible velocity has been attained at the throat and the fate of onward flow depends on the difference of pressure applied across the duct and duct geometry. For an isentropic flow, critical parameters can be shown to be related to the corresponding stagnation conditions near inlet by following equations

$$\frac{P_0}{P^*} = \left[\frac{k+1}{2} \right]^{k/(k-1)} \quad (4.3)$$

$$\frac{T_0}{T^*} = \left[\frac{k+1}{2} \right] \quad (4.4)$$

$$\frac{\rho_0}{\rho^*} = \left[\frac{k+1}{2} \right]^{1/(k-1)} \quad (4.5)$$

$$V^* = RT_0 \left[\frac{2k}{k+1} \right]^{1/2} \quad (4.6)$$

$$m_{choked} = \rho^* A^* V^* \quad (4.7)$$

$$\frac{A}{A^*} = \frac{1}{M} \left[\frac{1 + \frac{k-1}{2} M^2}{\frac{k+1}{2}} \right]^{(k+1)/2(k-1)} \quad (4.8)$$

Pertinently, at this stage the mass flow rate through the duct is at maximum and stays constant (flow velocity is sonic and stays constant at that value), even if the downstream pressure is changed or fluctuates, given that it does not exceed the critical pressure. This is because the signal of pressure change downstream, is conveyed upstream at sonic speed, and so is not able to cross the nozzle throat, the flow velocity at throat being sonic. A plot of area ratio A/A^* against the Mach number of flow M , at the locus of area A , as related by equation 4.8 is shown in figure 4.2. It can be seen that corresponding to each Mach number, there exists a unique area ratio, while as each area ratio (with the exception $A/A^* = 1$) corresponds to a pair of possible mach numbers of flow, one subsonic and the other supersonic. The plot thus translates to convergent divergent geometry of a nozzle that can be used to accelerate a subsonic flow to supersonic.

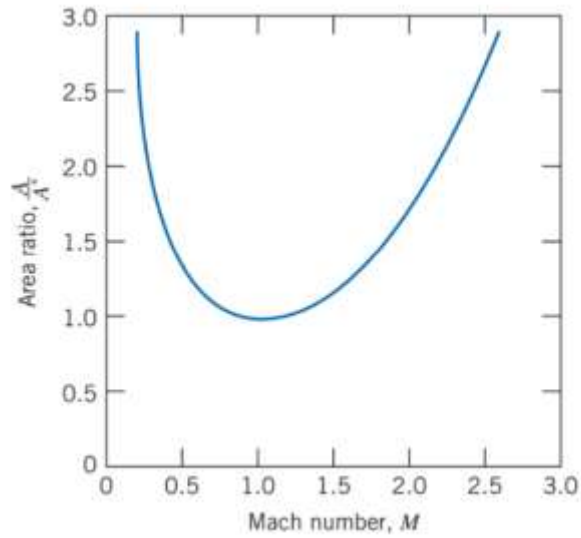


Figure 4.2 – Variation of A/A^* with Mach number for isentropic flow of an ideal gas with $k=1.4$ (Fox, 2011)

Isentropic flow through a convergent divergent nozzle:

Ensuing to the discussion so far, flow through a convergent divergent nozzle is described here forthwith.

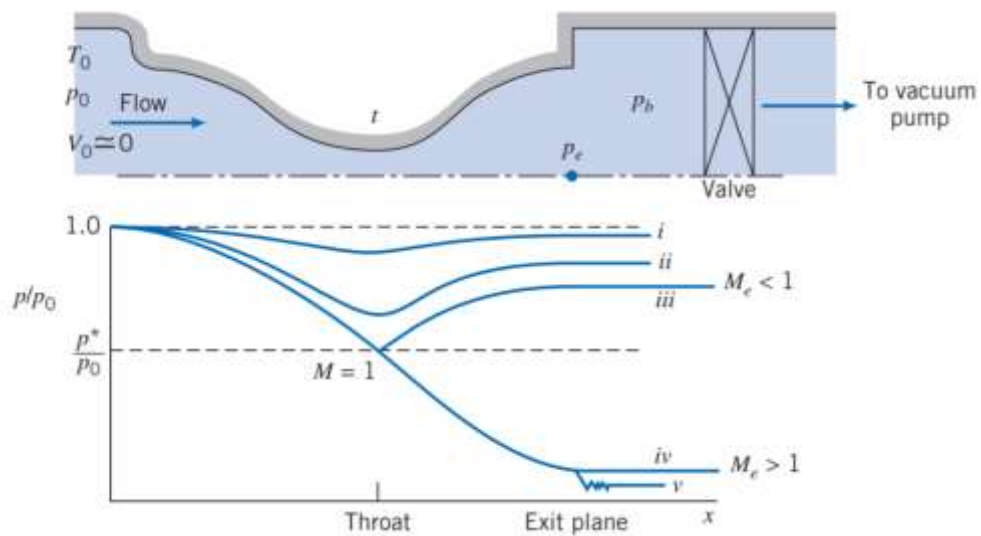


Figure 4.3 – Operation of a converging-diverging nozzle through various back pressures (Fox, 2011)

Distribution of pressure corresponding to a given condition near nozzle inlet and various conditions near the nozzle outlet, along the length of the nozzle is depicted in figure 4.3. Flow commences from stagnant condition (P_0, T_0) near the inlet of the nozzle and is discharged to a back pressure P_b near the nozzle exit, ($P_0 - P_b$) being the pressure differential applied across the nozzle that drives the flow. P_e is the pressure in the exit plane of the nozzle and P_t the pressure at its throat. Depending upon the value of P_b outside the nozzle exit, the behaviour of flow through the nozzle can be described by salient curves of figure 4.3. Curve i signifies an essentially subsonic flow through the nozzle with P_b not low enough to cause P_t to attain critical value. P_b equals P_e in this case and the flow is nearly incompressible. Flow behaviour represented by curve ii is similar to that represented by curve i, the difference being that a lower back pressure P_b has been employed in the latter case, causing the throat pressure P_t to be comparatively lower (still not critical), and so the flow velocity at throat assumes a comparatively higher value resulting in appearance of compressibility effects, that now are not ignorable. Lowering of P_b to a particular value, causes P_t to drop to the critical value P^* , as signified by curve iii, at which point the flow velocity at throat becomes sonic ($M=1$). As discussed earlier, this condition corresponds to the maximum mass flow rate obtainable through the nozzle, and lowering of back pressure beyond this point will yield no further increase in mass flow rate. Substantial lowering of back pressure causes the flow to be signified by curve iv. Interestingly, after chocking of the nozzle (attainment of $M=1$ at throat), onward flow through the divergent section, as represented by curves i,ii and iii undergoes deceleration, thereby exhibiting a subsonic flow behaviour. It must be noticed that a very low back pressure causes the flow in the divergent section to accelerate and so exhibit supersonic character. Pertinently, post chocking, mass flow rate through the nozzle remains at the already attained constant and maximum value for a range of back pressures, provided that back pressure P_b does not exceed the critical pressure P^* . This allows for employability of sonic nozzles for metering constant mass flow rates, even when the downstream pressure fluctuates

4.3 Design

Five sonic nozzles to be assembled into a sonic nozzle bank were designed as per the recommendations made in ISO 9300:2005, the standard for design of critical flow venture

nozzles. Design areas of throat, to obtain certain required mass flow rates at choking, as the nozzle is made to operate between given conditions of pressure, were obtained. A toroidal throat type design was chosen for the sonic nozzles and its geometry is outlined by the following figure given in the standard.

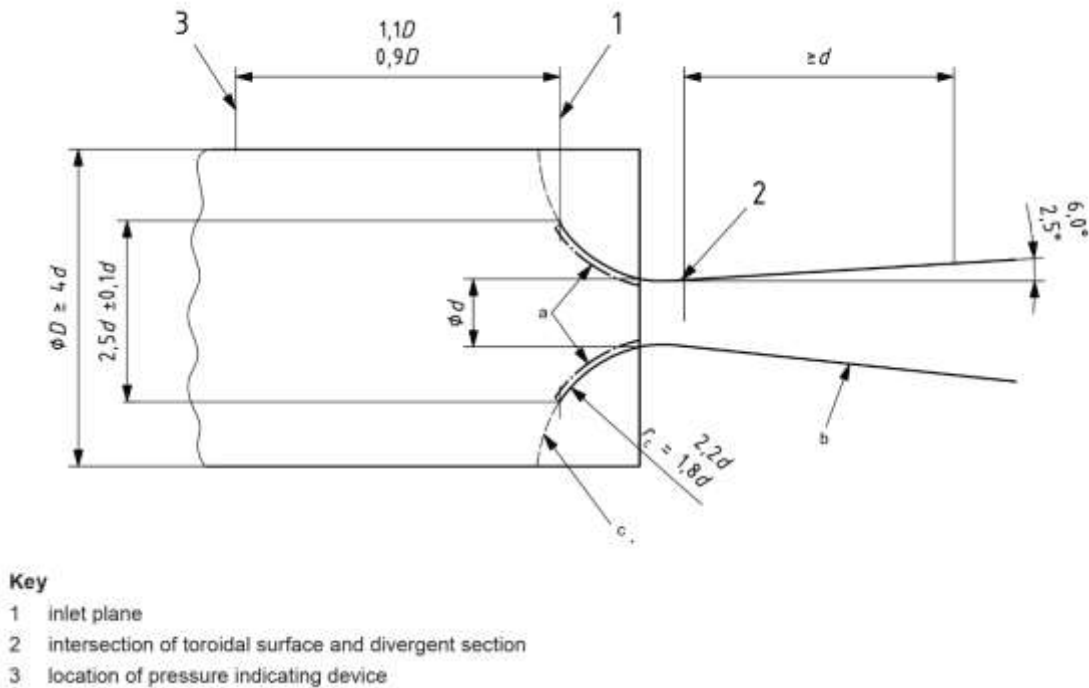


Figure 4.4 – Design geometry of a Sonic Nozzle (ISO 9300, 2005)

The design is based on the following equations

$$q_m = \frac{A^* C_d C_* P_0}{\sqrt{\left(\frac{R}{M}\right) T_0}} \quad (4.9)$$

$$\frac{A_2}{A^*} = \left[\frac{2l \tan \theta}{d} + \frac{2r_c}{d} (1 - \cos \theta) + 1 \right]^2 \quad (4.10)$$

$$\left(\frac{P_2}{P_0}\right)_{max} = 0.8 \left[\left(\frac{P_2}{P_0}\right)_i - r_* \right] + r_* \quad (4.11)$$

Where

$$r_* = \left[\frac{2}{k+1} \right]^{k/(k-1)} \quad (4.12)$$

Table 4.1 – Values of design parameters for sonic nozzles (S.I units unless specified)

q_m	Re^*	Cd	A^*	D	d(cm)	4d(inches)	r_c	l	θ (radians)	A_2	r_e	d_e
0.01	265958	0.993	0.000007	0.002993	0.299	0.471	0.005986	0.017957	0.0785	0.000027	0.002927	0.005855
0.02	376121	0.993	0.000014	0.004232	0.423	0.666	0.008465	0.025395	0.0785	0.000054	0.00414	0.00828
0.04	531755	0.992	0.000028	0.005987	0.598	0.942	0.011975	0.035924	0.0785	0.000108	0.005856	0.011713
0.06	651100	0.992	0.000042	0.007335	0.733	1.155	0.01467	0.044009	0.0785	0.000162	0.007174	0.014349
0.1	840567	0.992	0.000072	0.009469	0.946	1.491	0.018938	0.056815	0.0785	0.00027	0.009262	0.018524

Values of the critical flow function C^* and coefficient of discharge C_d , for each nozzle, are obtainable from appendices/relations provided in the standard. Values of maximum permissible downstream pressure may be obtained from a plot between (A_2/A^*) and $(P_2/P_0)_{\max}$ given in the standard and were found to be equal to $0.89574 P_0$ for each nozzle. The values for parameters of design for the five nozzles are tabulated in table 4.1 and subsequently a design drawing for each nozzle furnished.

Pertinently, each of the five nozzles was designed for operation under an upstream pressure (P_0) equal to 600000 Pa and a down-stream pressure equal to 400000 Pa. Stagnation conditions were assumed at nozzle inlet with the inlet density (ρ_0) equal to 6.96 kg/m^3 and inlet temperature (T_0) equal to 300 K. The values of pressure (P^*), temperature (T^*), density (ρ^*) and velocity (V^*) at throat were found to be 316969 Pa, 250 K, 4.41 kg/m^3 and 316 m/s respectively. Values of C^* and μ_0 for each nozzle, obtained from standards, are equal to 0.7 and 0.000016 Pa.s respectively. Gas constant for air (R) was taken to be equal to 287 J/kgK and specific heat ratio (k) was assumed to be equal to 1.4. Through an applicable clause in the standards, the value of half angle of the frustum of divergent section of the nozzle (θ) was selected to be 4.5 degrees and radius of curvature of the toroidal surface was taken to be equal to two times the throat diameter ($2d$). The exit area to throat area ratio (A_2/A^*) for each nozzle thus worked out to be equal to 3.83 and thus maximum permissible downstream pressure was calculated to be equal to 573444 Pa.

Legend:

P_0 - Stagnation pressure (absolute) at inlet (Pa)

T_0 - Stagnation temperature (absolute) at inlet (K)

ρ_0 - Stagnation Density at inlet (kg/m^3)

P^* - Pressure at throat (Critical)

T^* - Temperature at throat (Critical)

q_m - Choked mass flow rate required (kg/s)

C^* - Critical Flow Function (from standards)

μ_0 - Coefficient of dynamic viscosity at inlet stagnation conditions

V^* - Sonic Velocity at throat (m/s)

ρ^* - Density at throat

Re^* - Reynolds No. at throat based on inlet conditions of Viscosity and Mass flow rate

C_d - Coefficient of discharge(from Standards)

R - Gas Constant for air (J/kg.K)

A^* - Area of cross-section of throat (m^2)

d - Diameter of throat (m)

k - Specific Heats Ratio

P_b - Downstream pressure

P_2 - Pressure at nozzle exit

r_c - Radius of curvature of toroidal surface (m)

l - Length of the divergent section (m)

θ - Half angle of the frustum of divergent section

A_2 - Area of cross-section of exit of divergent section

P_{2max} - Maximum allowable downstream pressure

r_e - Radius of exit section

d_e - Diameter of exit section

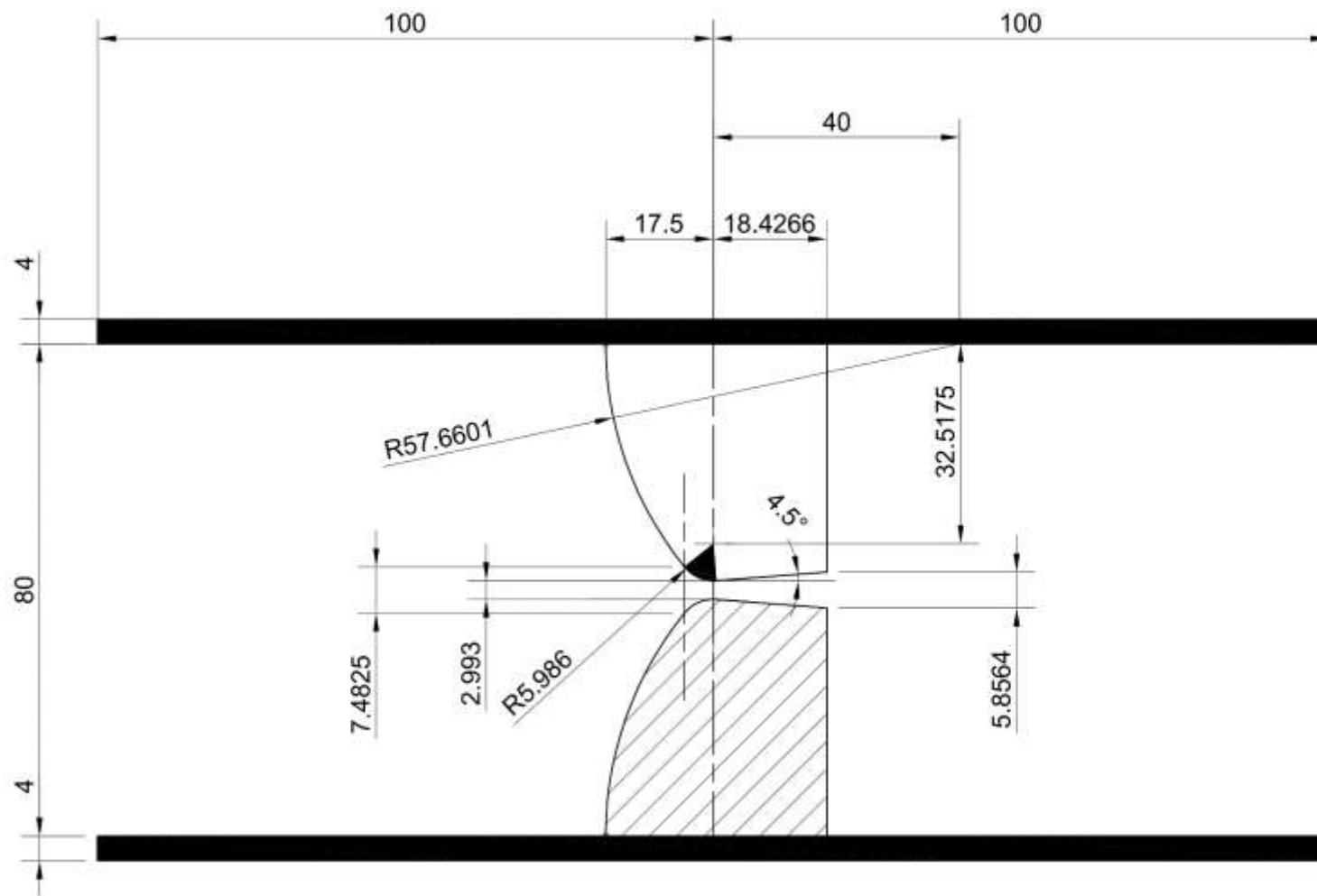


Figure 4.5 - Design drawing of sonic nozzle for metering an air flow rate of 0.01 kg/s

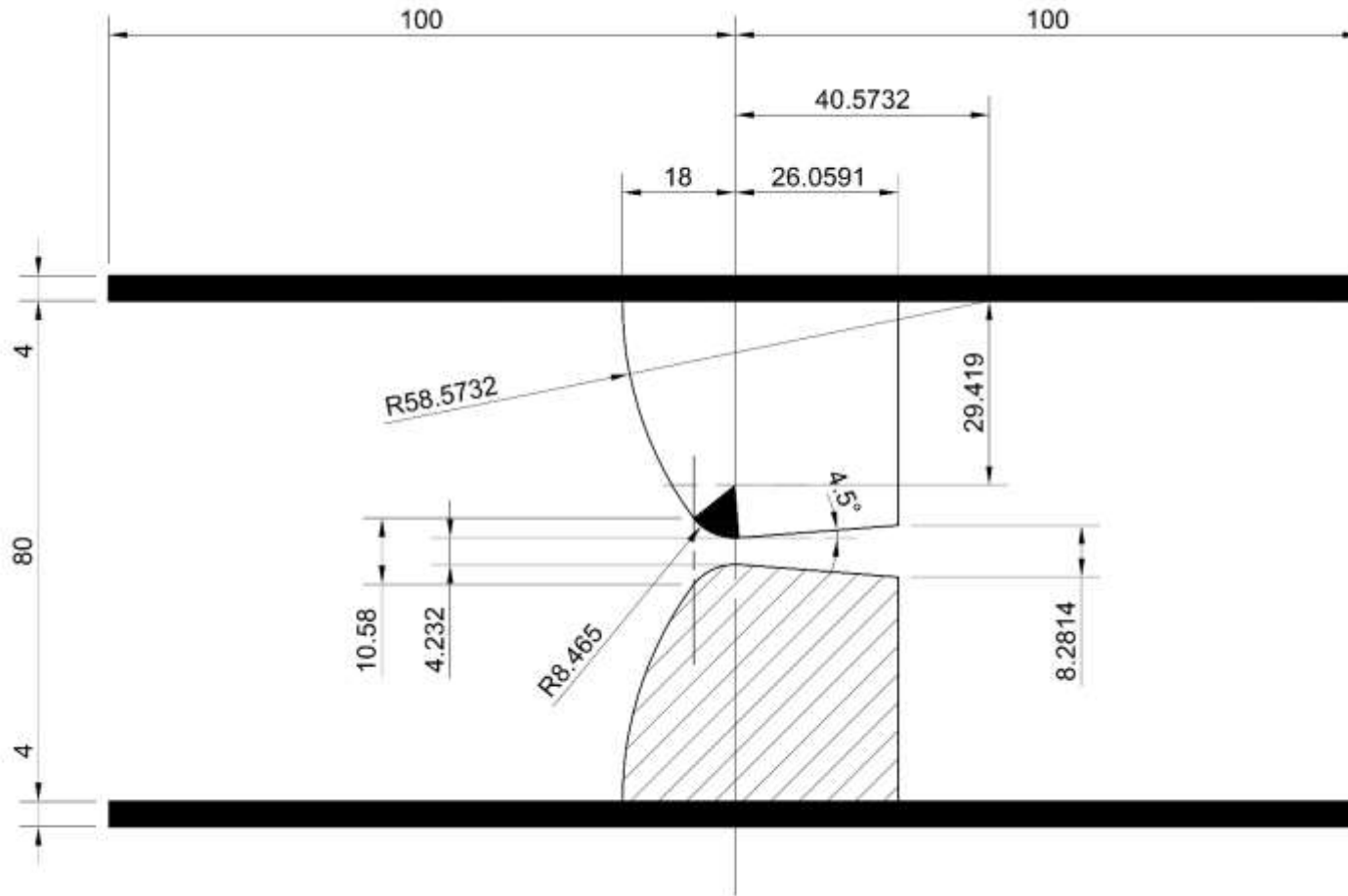


Figure 4.6 – Design drawing of sonic nozzle for metering an air flow rate of 0.02 kg/s

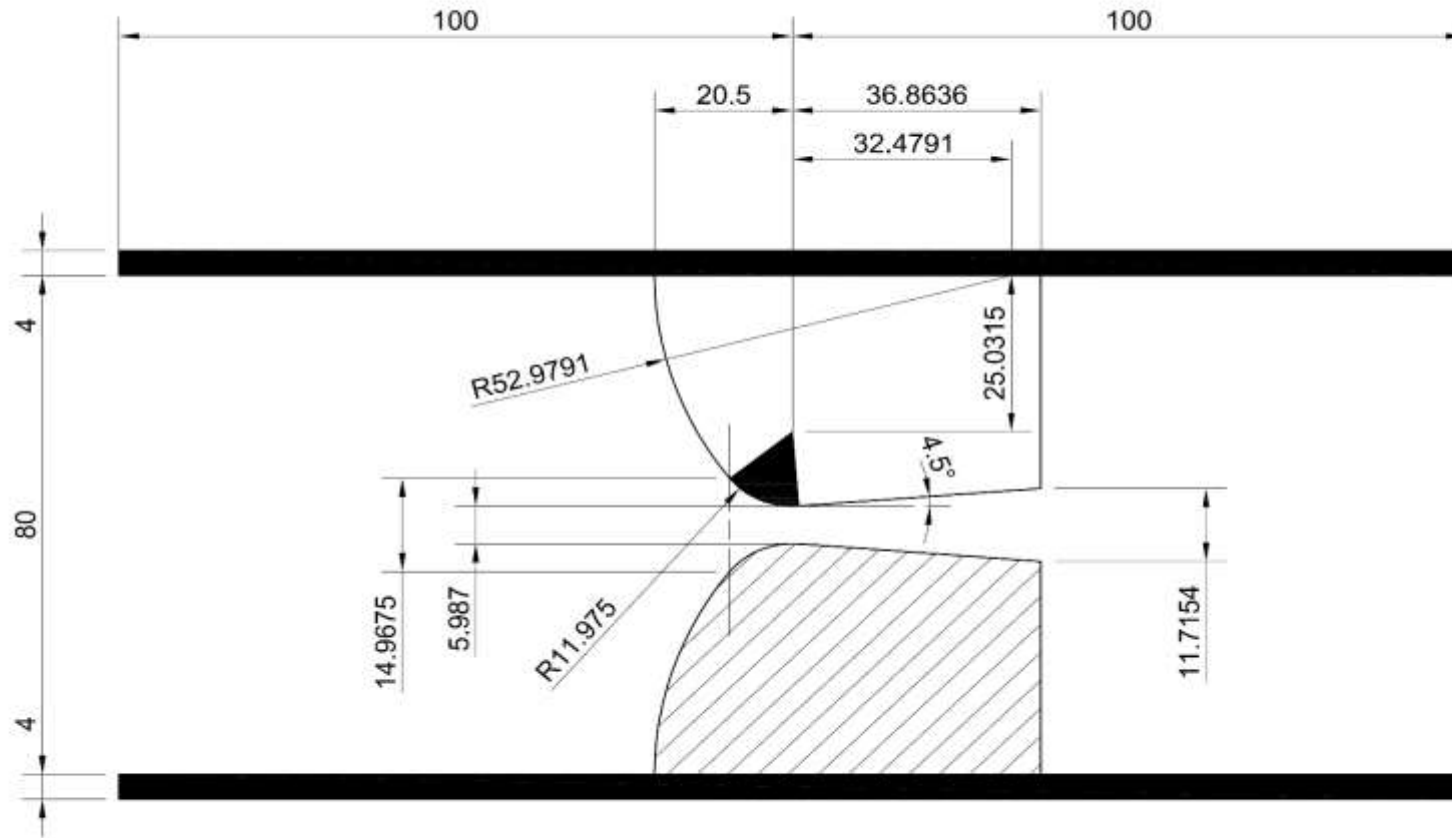


Figure 4.7 - Design drawing of sonic nozzle for metering an air flow rate of 0.04 kg/s

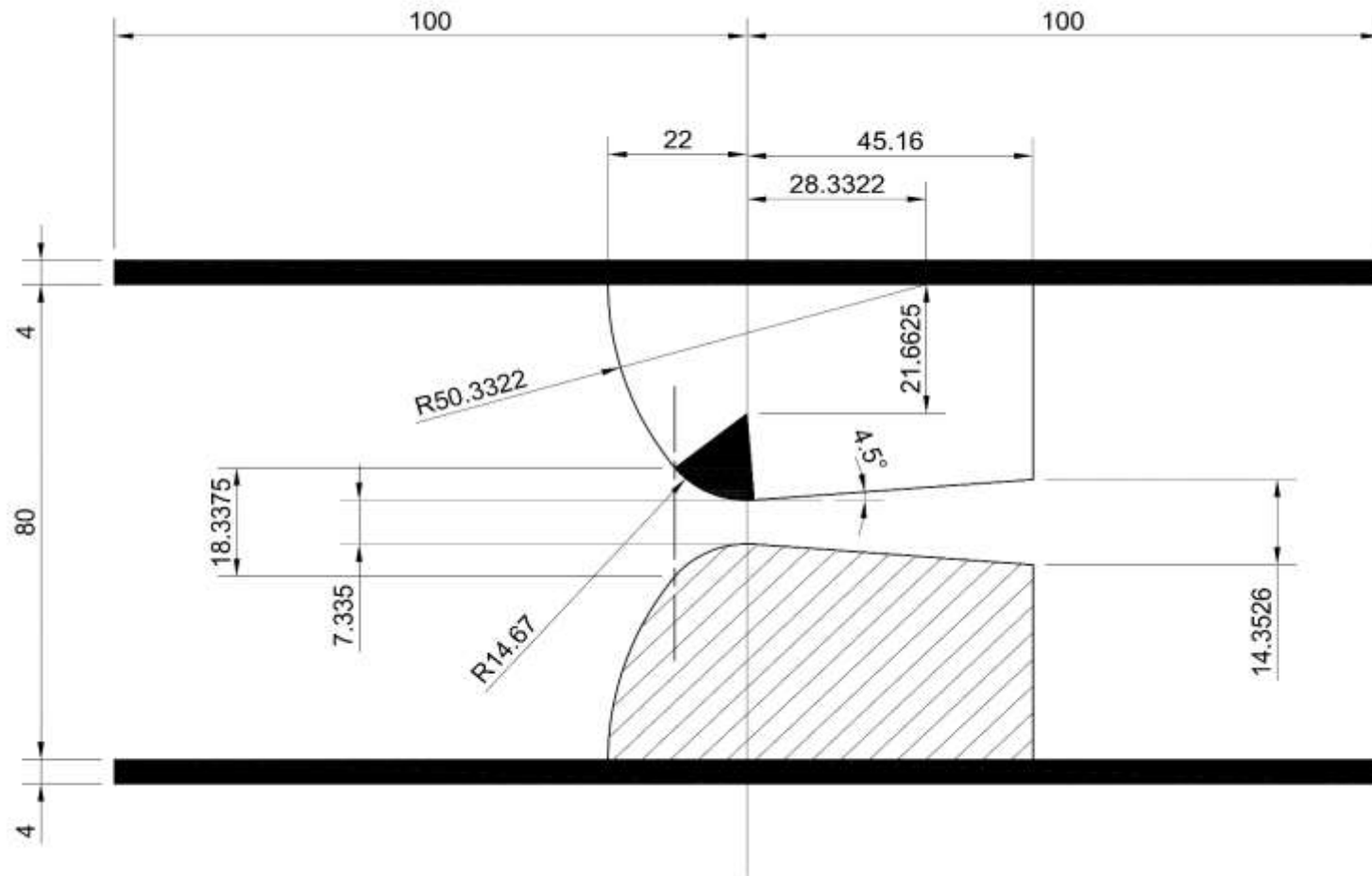


Figure 4.8 - Design drawing of sonic nozzle for metering an air flow rate of 0.06 kg/s

Chapter 5: Design and development

5.1 Introduction to design

Enlisted as the foremost and major objective of this thesis work, development of a pneumatic conveying pilot plant was carried out at the ‘Laboratory for Particle and Bulk Solids Technologies’ in mechanical engineering department of Thapar Institute of Engineering & Technology, Patiala. Post kick-off, planning of the project was initiated with establishment of specifications of the system required, to achieve research goal at hand. Ensuingly, after design or selection of system components as applicable, vendor drawings/documents were reviewed (based on the design/selection by the author) and subsequently manufacture and/or purchase of equipment carried out. The availability of dimensions of the equipment thus selected, enabled assessment of available space for various possible layouts. A judgement between the merits presented by various options was made to finalize the general layout. A representative 3D model is shown below.

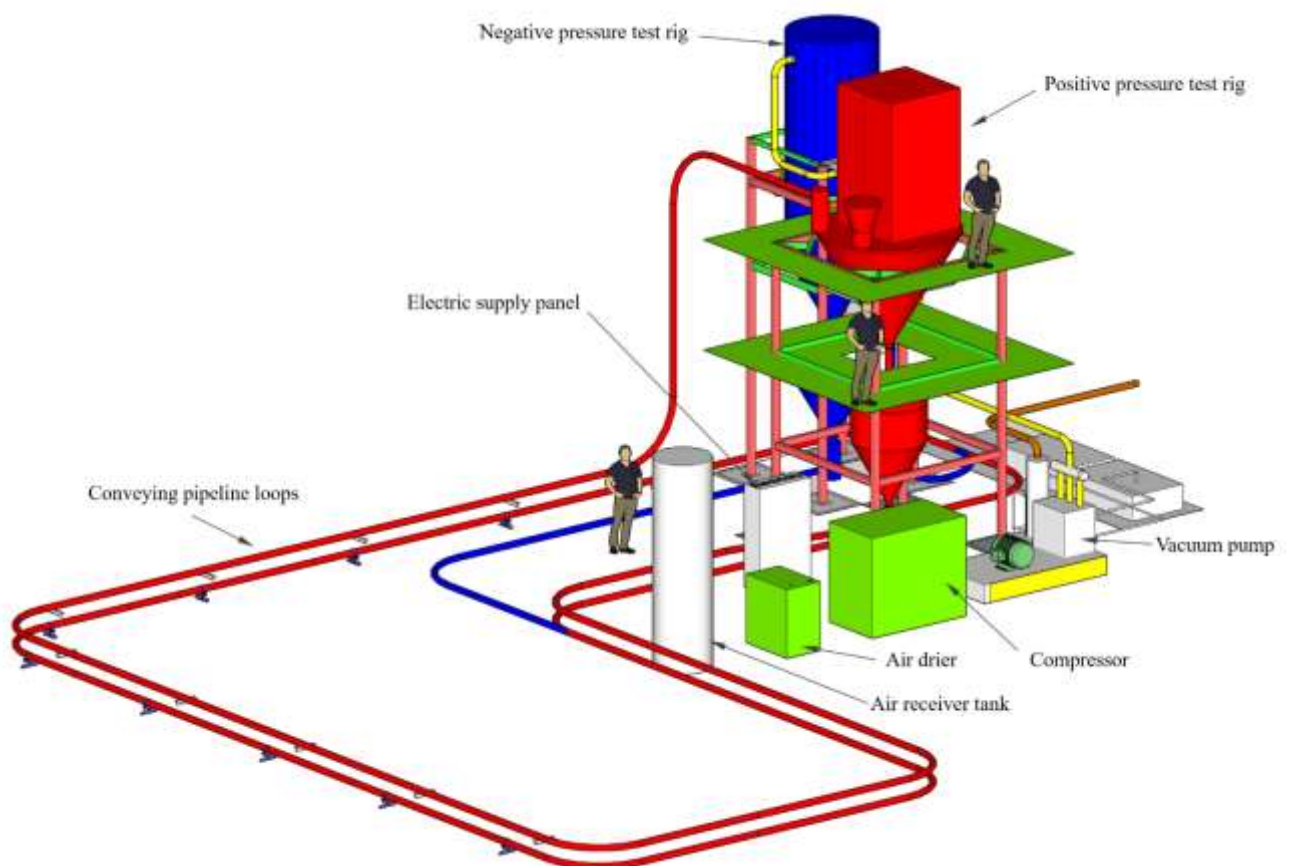


Figure 5.1 – General system layout

Table 5.1 gives a basis for sizing/designing various components of the two pneumatic conveying test rigs developed.

Table 5.1 - Basis of sizing the components of pilot plant

1. Compressor sizing	
Highest pressure	700 kPa (g)
Maximum blow tank pressure	300 kPa (g)
Air mass flow rate (for compressor FAD estimation)	To be able to maintain initial gas Froude number of 8 at the product feed point against 150 kPa (g) pressure or initial Froude number of 6 at the feed point against 250 kPa (g) pressure, whichever is higher
Air receiver sizing	To be able to maintain a minimum of 10 seconds of steady state flow condition to obtain good experimental data for 35 t/h of solids flow rate
2. Vacuum pump sizing	
Highest vacuum	24 inch Hg vacuum
Operating vacuum	16 inch Hg vacuum
Air mass flow rate	To be able to maintain initial gas Froude number of 18 at the product feed point
3. Bag filter sizing in vacuum and pressure systems	
Cloth size	To be able to achieve maximum 1 m/min of face velocity corresponding to 10% margin on air mass flow rate of compressor and vacuum pump
4. Blow tank sizing	
Sizing of blow tank	To be able to maintain a minimum of 10 seconds of steady state flow condition to obtain good experimental data for 35 t/h of product flow rate
Material inlet valve sizing	To achieve unassisted (gravity flow) of Geldart Group A fly ash into blow tank using Jenike design approach
Material outlet valve sizing	To achieve a maximum of 75% filling of pipe cross section or 25 t/h of solids flow rates, whichever is larger, to obtain good experimental data

Fluidizing pad selection	To achieve a maximum of 20% of total air flow through the pads with a maximum pressure loss across the pad of 20 kPa; able to withstand 400 kPa (g) pressure differential
5. Feeder sizing in vacuum system	
Sizing of feeder	To be able to maintain a minimum of 10 seconds of steady state flow condition to obtain good experimental data for 35 t/h of product flow rate
Material inlet valve sizing	To achieve unassisted (gravity flow) of Geldart Group A fly ash into blow tank using Jenike design approach
Material outlet valve sizing	To achieve 15 t/h of feed rate
5. Pipeline sizing in vacuum and pressure systems	
Pipe size	To be able to maintain a minimum of 10 seconds of steady state flow condition to obtain good experimental data for 35 t/h of product flow rate for pressure system and 20 t/h of product flow rate for vacuum system
Pipe layout	To be able to install pressure tappings atleast 6 m apart after 8 m of clearance from bend exit to obtain good quality experimental data
6. Receiver bin sizing in vacuum and pressure systems	
Bin size	To provide 2.5 times the corresponding feeder size considering loose poured bulk density is 2.5 times the fluidized bulk density
Discharge valve sizing	To achieve unassisted (gravity flow) of Geldart Group A fly ash into feeder using Jenike design approach
Hopper half angle	To achieve unassisted (gravity flow) of Geldart Group A fly ash into feeder corresponding to the feeder inlet valve sizing using Jenike design approach
7. Fluidizing blower sizing	
Air flow rate	To obtain 20 cm/sec of air velocity at fluidizing pad outlet to achieve minimum fluidization condition

5.2 System design and component selection of ash handling plant

5.2.1 Air supply system

Selection of an air supply is critical to the desired performance of a pneumatic conveying system, since compressed air supply is the source of momentum, for transfer to the material, and thus, the energizer of pneumatic transport. Correct specification of conveyance duty, in terms of free air delivery and the delivery pressure, is essential for correct selection of an air mover and so the sine qua non of successful operation of the entire system. These two parameters are in turn dependent on the nature of the material being conveyed, the rate of its flow and the distance of conveyance. In the present context, the design of experimentation to be conducted in pneumatic conveying of fly-ash, with set research goals, warrants a maximum free air delivery of 600 m³ suppliable at a delivery pressure of 6 bar. While this duty can be met by a variety of options, a positive displacement type rotary screw compressor was selected. Favourable to this selection are two important factors. First, the constant speed characteristic of positive displacement machines, by and large, does not flatten out within ranges of high pressure, and therefore an increase in pipeline pressure drop, due to variation in solid loading condition for example, often seen in pneumatic conveying, will not cause a drastic decrease in volumetric airflow rate supplied by the compressor as is clearly depicted by the figure 5.2.

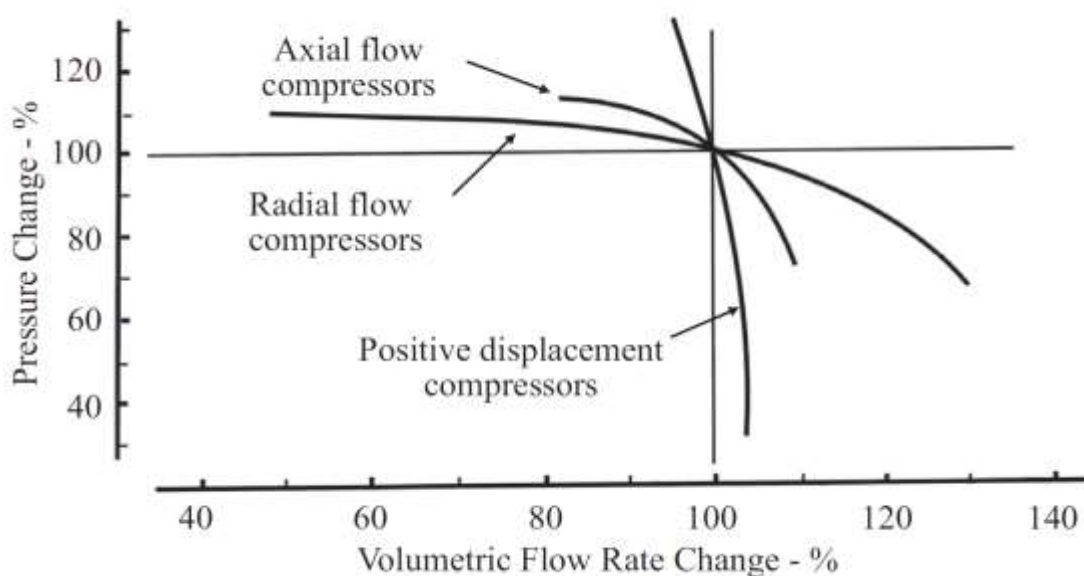


Figure 5.2 – Constant speed characteristics of positive displacement compressors (Mills, 2016)

Volumetric airflow rate may be considered to be almost constant (changes are so small that they may be incorporated into safety margins) with changing delivery pressure which changes corresponding to a changing drop of pressure in the conveyance pipeline. This is inevitable for blockage avoidance, particularly in the dilute phase mode of pneumatic conveying. Secondly, a rotary screw type positive displacement machine was selected since this equipment is well suited amongst the positive displacement machines, for medium to high delivery pressure requiring applications, and gives a supply of about 4 – 700 m³/min with a maximum delivery pressure of about 9 bar. Enlisted in the table 5.1 are the specifications of the unit selected.

Table 5.2 – Specifications of the compressor used in positive pressure system

Description	Unit	Particulars
Manufacturer name		Chicago Pneumatic Compressor, A div. of Atlas Copco India Ltd
Model No.		CPD 75 G 7
Type of compressor		Oil lubricated screw air compressor
No of stages		Single stage
Total weight	Kgs.	1162(approx.)
Dimensions	Mm	1850*1060*1630
Capacity-FAD	Cfm	356
Discharge pressure at compressor outlet	Bar	7
Discharge temperature at design condition	Deg C	45
Shaft input power required at compressor shaft at rated pressure	kW	55.2
Shaft input power required at compressor shaft at unloaded condition	kW	13.3
Guaranteed power consumption at motor input terminals	kW	64.1
Rating of main motor	kW	55
Motor RPM full load	Rpm	2968

A refrigeration type air drier to ensure elimination of moisture to a satisfactory level was selected to draw air input from the compressor. Table 5.2 enlists the specifications of the selected drier

Table 5.3 – Specifications of the air dryer unit

Data components	Description
Equipment and model	Refrigeration dryer , CPZ 380
Make	Chicago Pneumatic Compressor, A Div. of Atlas Copco India
Service	Compressed air
Inlet flow	356 cfm
Inlet pressure	7 bar
Max. working pressure	13 bar
Inlet air pressure	55 deg C
Ambient temperature	25 deg C
Dew point pressure	3-7 deg C
Power consumption	1.70 Kw
Voltage	230/1/50v/ph/hz
Compressor type	Rotatory hermitically sealed
Refrigerant gas	R410A
Refrigerant suction pressure	69-70 psi
Refrigerant discharge pressure	280 – 360 psi
Condensing temperature	40 deg C
Overall dimensions	1025 x 660 x 1180 mm

An air receiver tank generally forms a constitutive part of any compressed air system since it functions to be a reservoir of compressed air helping meet peak demands, provides allowance time for the compressed air to cool, enables settling down of water, if any, is contained in the compressed air and helps minimise the effects of pulsations or fluctuations that may be caused due to the nature of operation of the air mover. For the present system the receiver tank show in figure 5.3 was designed. The tank, made of mild steel (IS-2062), passed a hydro-test at 14 kg/cm² of pressure and was designed for a pressure of 8 kg/cm², with a working pressure of 7 kg/cm².

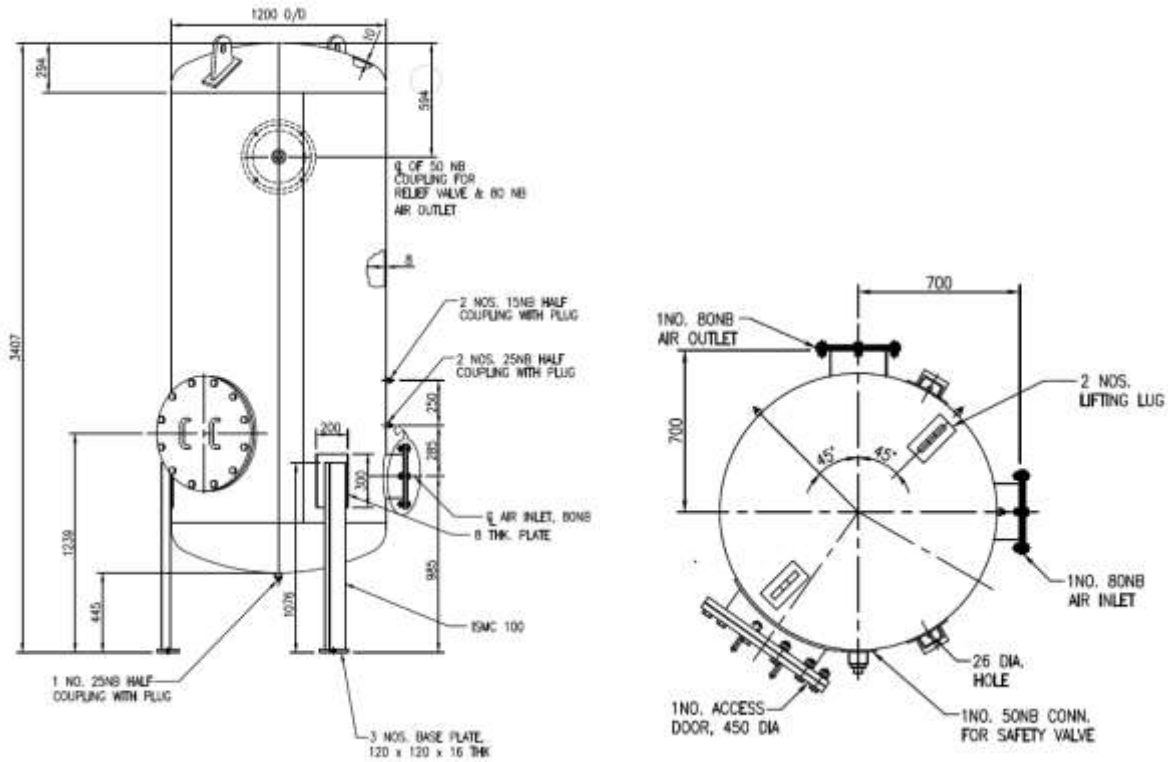


Figure 5.3 – Design drawings of the air receiver tank

A vacuum pump is used to power the flow of air in negative pressure pneumatic conveying. This is analogous to the use of air mover (compressor) in a positive pressure pneumatic conveying system. Selection of the vacuum pump for a particular application is made by specification of flow rate at inlet of the machine and amount of vacuum required at inlet of the machine, which typically is equal to the pressure drop across the pipeline, needed for material conveyance, added the allowances making up for margins and losses. A vacuum pump of the make ‘PPI Pumps’ (model no. PL 400) with an isothermal efficiency of 33.8%, volumetric efficiency of 38.1% and mechanical efficiency of 38.1% was selected for use in the rig constructed. The performance charts of the pump are presented in figure 5.4

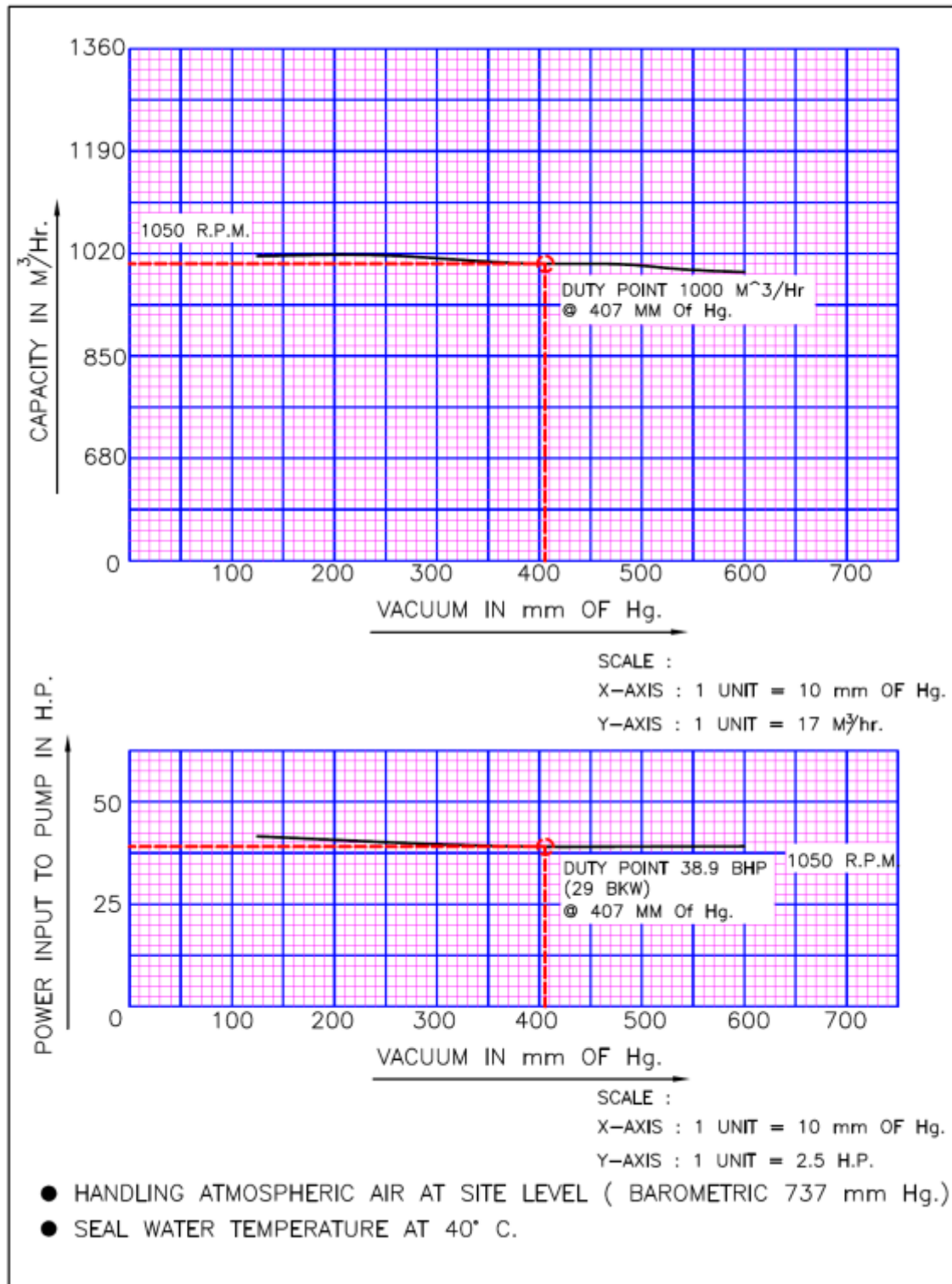


Figure 5.4 – Performance curve for the water ring vacuum pump model PL-400

The performance parameters of the pump with their values are enlisted in table 5.3

Table 5.4 – Specifications of the vacuum pump

Manufacturer	PPI Pumps
Model no.	PL-400
Type of Pump	Water ring vacuum pump
Guaranteed volume (rated capacity) of air to be handled at rated inlet vacuum on dry air basis.	1000 m ³ /h
Rated vacuum at pump inlet.	407 mm of Hg (16 inch)
Shaft power (bkW) (At 50 degree C inlet air temperature and 40 degree C cooling water inlet temperature)	32 brake kW (500 mm OF Hg set point of safety valve)
Vacuum developed by the pump across range of operation	0 to 508 mm of Hg
Max. vacuum at which the machine can be operated at selected motor rating with cooling water temperature 40 degree C and inlet air temperature 50 degree C.	640 mm of Hg
Pump speed at rated capacity & vacuum	1050 RPM
Pump direction of rotation from drive end side	Clockwise

5.2.2 Feeders

A bottom discharge type blow tank feeder was used for feeding the material in the positive pressure pneumatic conveying system and a conical vessel called the transfer hopper was used for material feeding in the vacuum pneumatic conveying system.

A 0.8 m³ blow tank feeder (IS 2062) made up of mild steel and fit with trough type fluidizers was designed for use in the system under consideration. Subjected to a hydro-test pressure of 10 kg/cm² the tank was designed for a working pressure of 5kg/cm². Following figure illustrates the design of the blow tank.

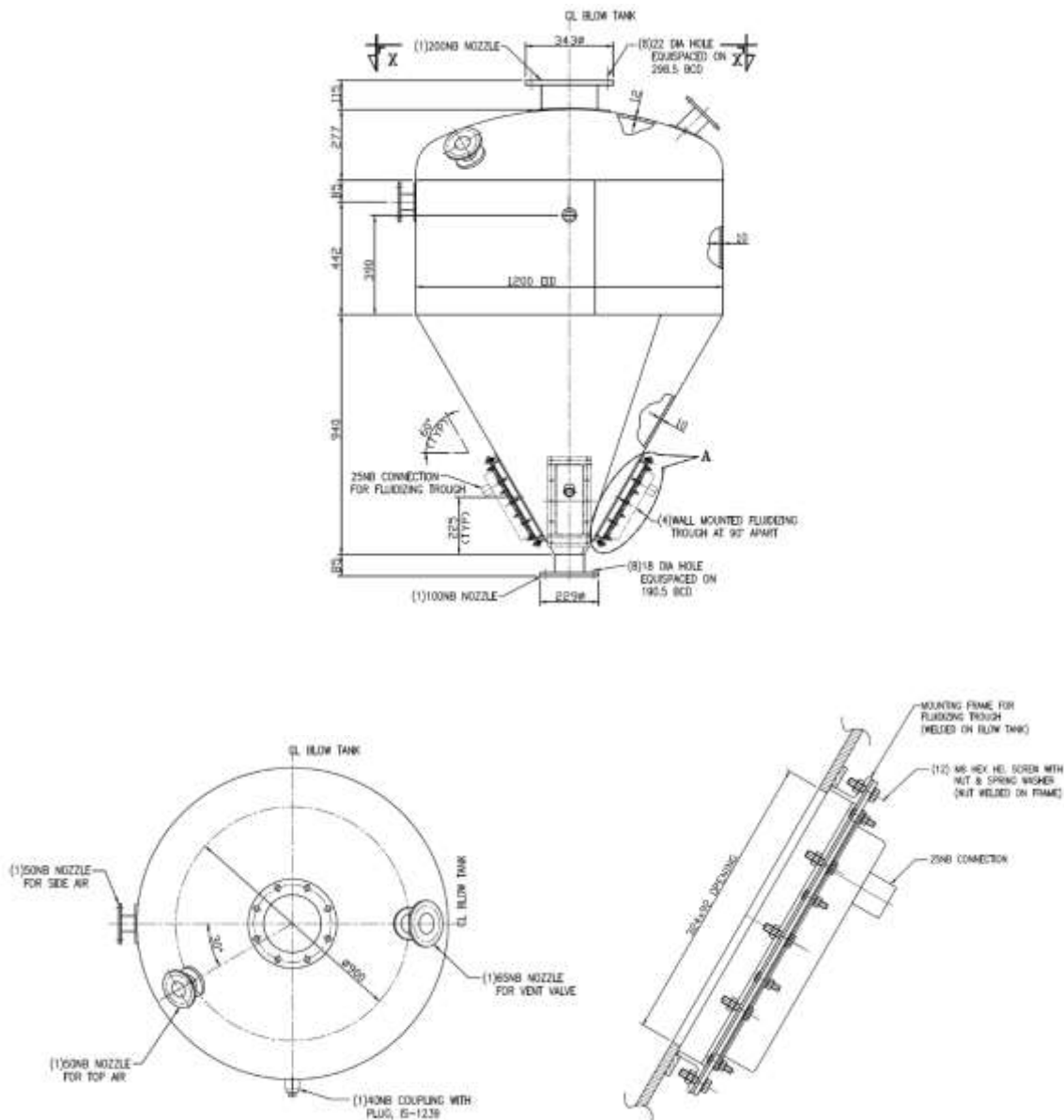


Figure 5.5 – Design drawings of the blow tank feeder and trough type fluidizer

A 0.4 m³ ash feed conical transfer hopper vessel made of mild steel (IS-2062) with a trough type fluidizing arrangement was designed for feed of material in the vacuum pneumatic conveying system. The following figure illustrates the design of the feeder vessel.

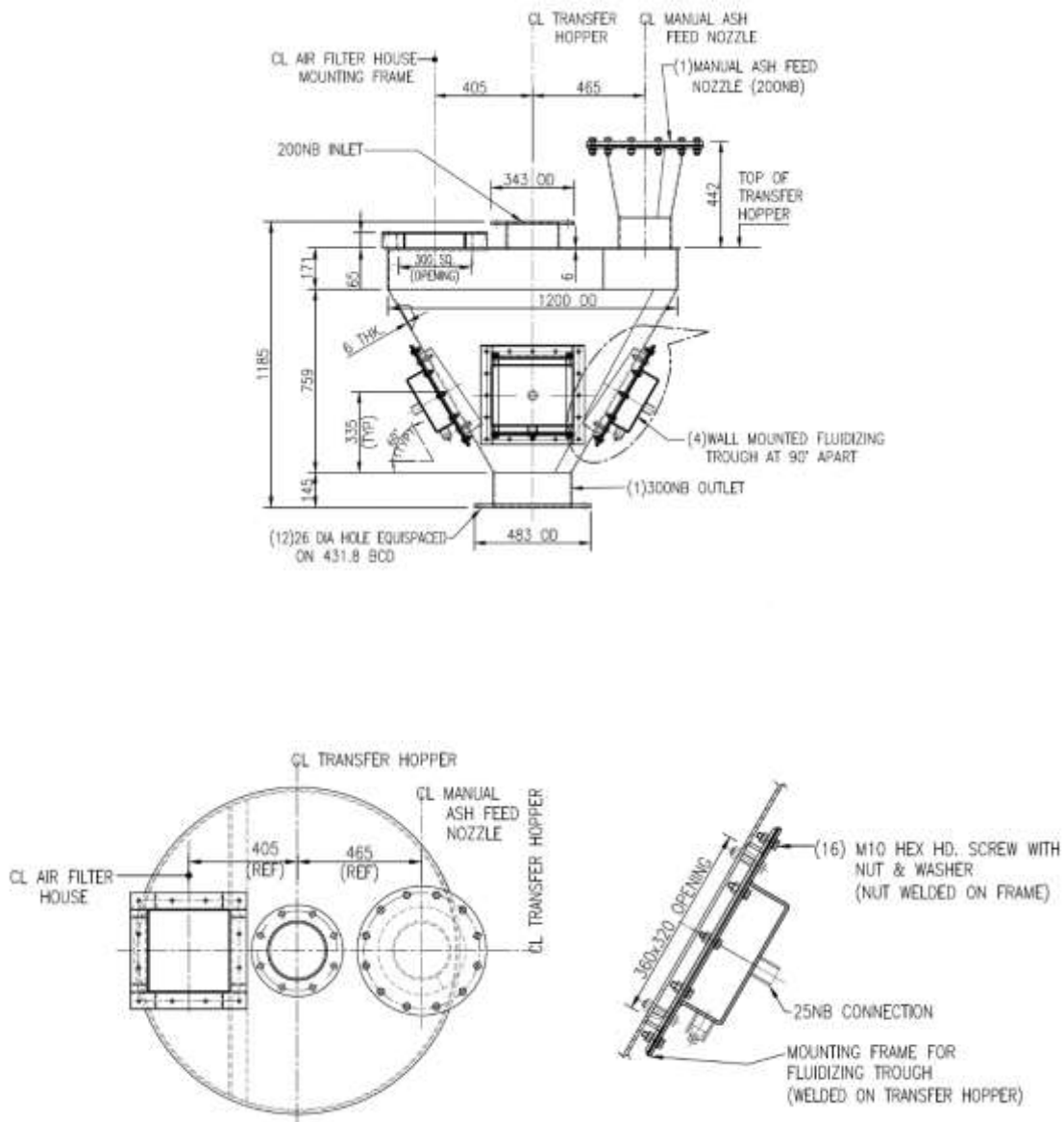


Figure 5.6 – Design drawings of the transfer hopper vessel and trough type fluidizer

5.2.3 Storage bins and material separation equipment

A conical vessel made of mild steel (IS-2062) and with a wall thickness of 6 mm, referred to as the receiver bin was used in the positive pressure pneumatic conveying system for bulk material storage enabling subsequent passage of material to the blow tank for material feed into the conveyance pipeline. The vessel with a volume of 2 m³ was provided with a trough type fluidizing arrangement. Following figure illustrates the design of the vessel.

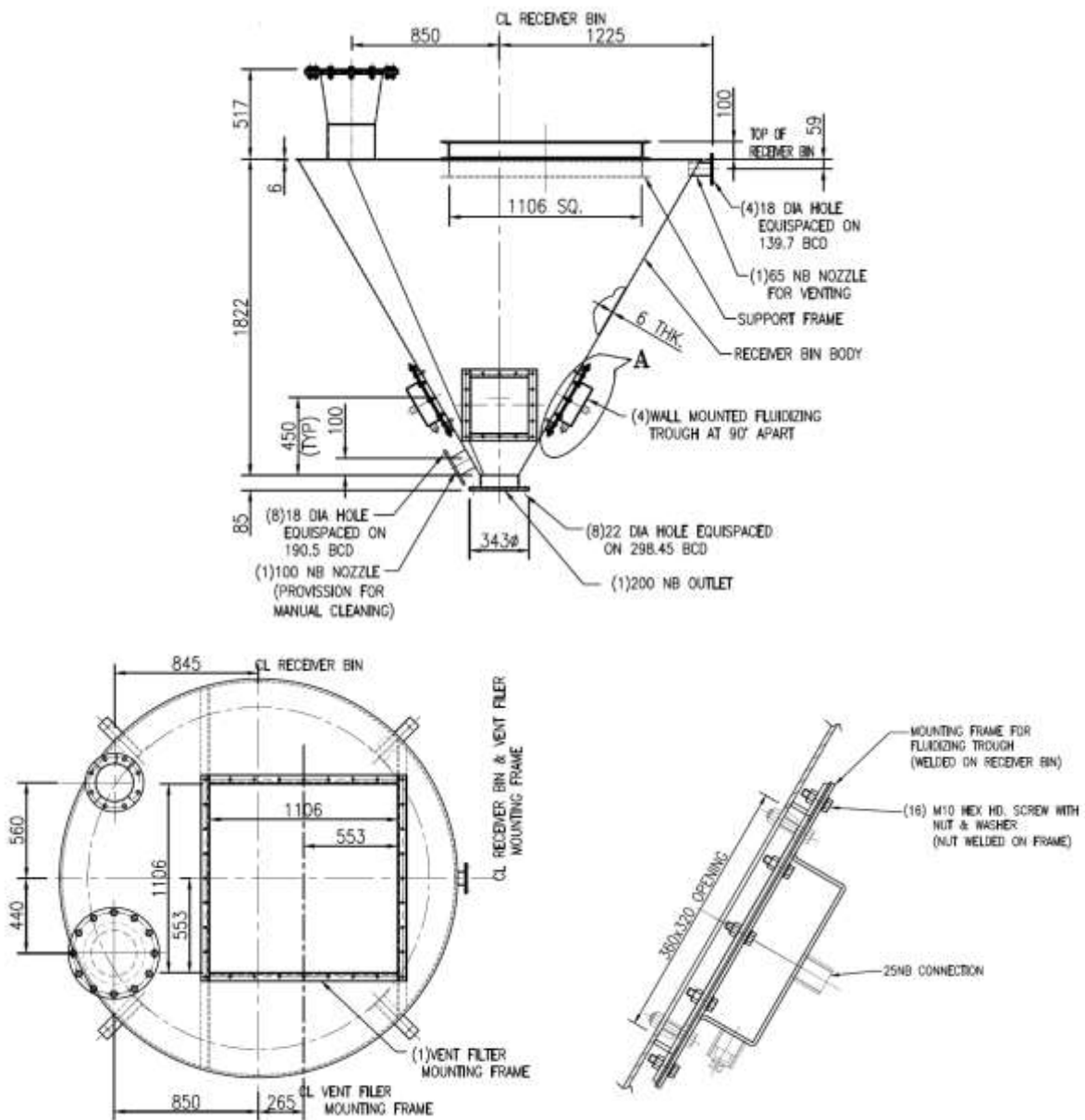


Figure 5.7 – Design drawings of the receiver bin and trough type fluidizer

For use in the positive pressure pneumatic conveying system, a bag filter assembly to be mounted on top of the receiver bin, for receipt of gas-solid flow, separation of material from the gas, and subsequent passage of material into the receiver bin for storage was designed. Comprising of 25 filter bags made of non-woven needle felted polyester fabric, the design required an air to cloth ratio of 1 m³/min, cloth area per bag equal to 0.441m² for a capacity of 11 m³/min. The filter bags, each contained in a cage made of galvanized mild steel, were housed in a casing of cold rolled steel (IS-1079) having a wall thickness of 3.5 mm, through a bag holding plate(CR Steel, 3.5mm thick) to form the filter assembly. Following is the design drawings of the assembly.

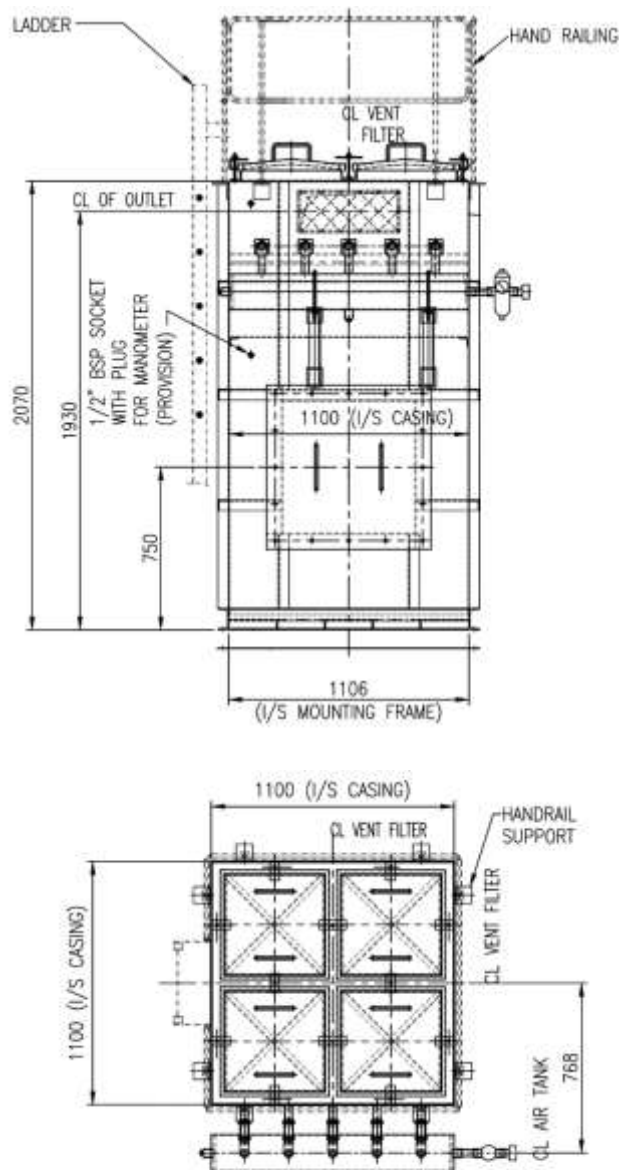


Figure 5.8 – Design drawings of the bag filter assembly

In the vacuum pneumatic conveying test rig a 0.8 m³ conical vessel made of mild steel was extended into a cylinder from its top surface for housing the bag filter assembly. The vessel provided with four trough type fluidizers stores material, for onward passage to the transfer hopper vessel, which subsequently feeds it into conveyance pipeline for vacuum conveying. The bag filter assembly housed in the cylinder that runs in continuity with the top of aforementioned conical vessel, consists of filters made of non-woven needle felted polyester fabric. The filter assembly is designed for a capacity of 18 m³/min, air to cloth ratio of 1 m/min and cloth area/bag equal to 0.586 m². Figure 5.9 illustrates the design of material storage and filtering unit employed in the negative pressure test rig.

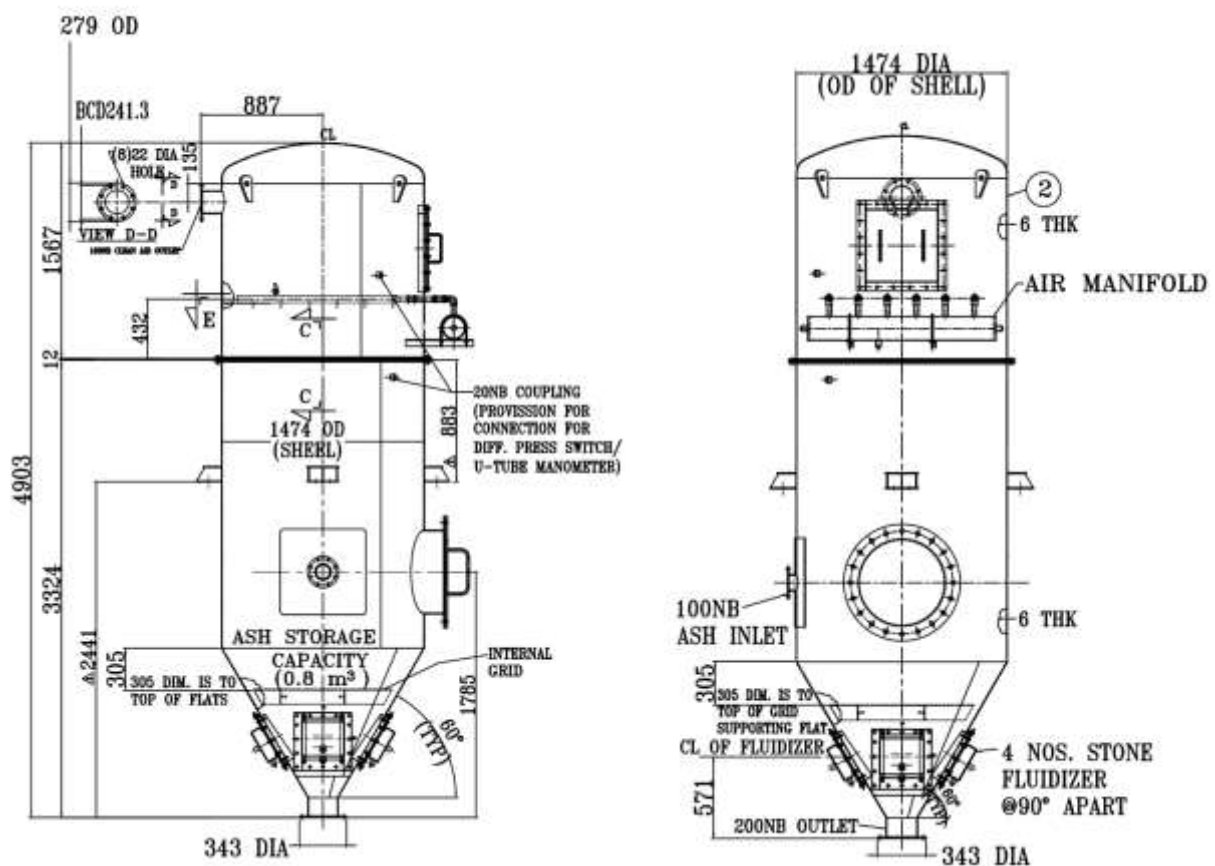


Figure 5.9 – Design drawings of the storage hopper vessel and bag filter assembly

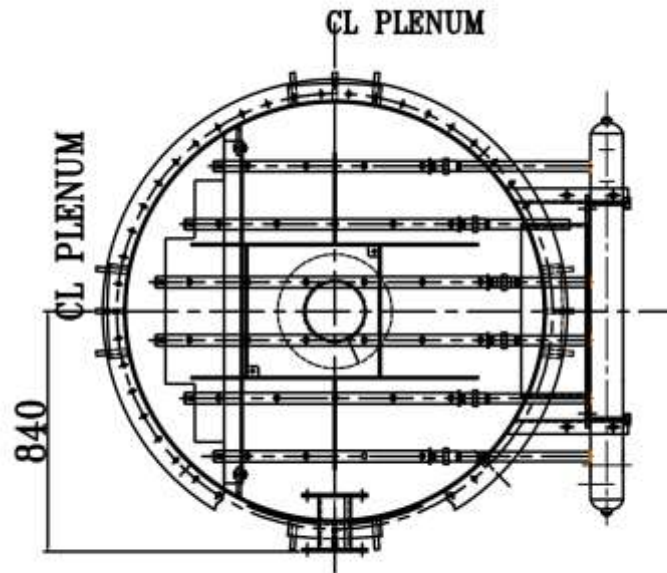


Figure 5.10 – Plan of the bag filter assembly used in vacuum test rig

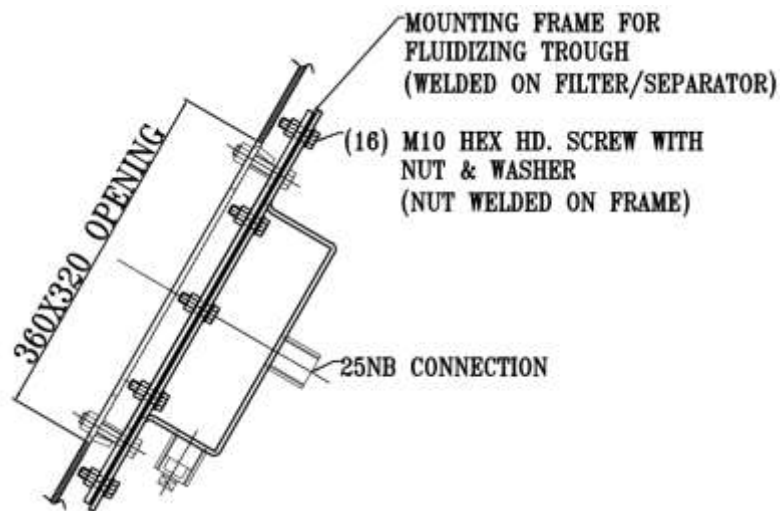


Figure 5.11 - Design drawing of the trough type fluidizer used in vacuum test rig

5.2.4 Valves

Various valve types such as double disc valves, blow-off valves, globe valves, seat valves, knife gate valves were bought out as per the system piping specifications and flow control desired. Most of these valves present the possibility of being operated pneumatically. The location of these valves in the system and their utility are illustrated by the piping and instrumentation drawing to be discussed in the upcoming section.

5.3 Measurement, automation and control

Like any other flow system, the test rig under consideration also employs many measuring instruments for flow pressure and load measurement. A sonic nozzle bank for intended employment to meter specified mass flow rates in the broader test program has been designed. Orifice flow-meters are in use, at relevant places indicated in the piping and instrumentation drawings, for measurement of air flow rates in pressure and vacuum test rigs. Fourteen single ended shear beam load cells, each with a measuring capacity of 5 tons, were bought out and employed to weigh the storage and feed vessels, both in pressure and vacuum test rigs, as will be illustrated through the stacked-up assembly drawings of the two rigs. Pressure transducers of the make 'Wika', with pressure measurement ranges were plugged into the threaded tapings at salient loci in the pipeline network, where pressure measurement was desired. A data logging unit for data retrieval, connectible with a computer for data presentation and storage, and compatible with the aforementioned load cells and pressure transducers was used. The data logger along with a supplied channel offers connectivity to 49 sensors, each of which is connected using two wires. It has a data sampling rate of 1 sample per 2 seconds for each channel and comes with a software through which it is operable from a computer.

Control of the pneumatically operated valves is possible through a pneumatic panel which admits air to various pneumatic valves through solenoid valves housed inside this panel. The solenoid valves are operable either manually through knobs located on the pneumatic panel or else derive operational inputs from a programmable logic controller unit housed in a panel box.

The PLC supplied by the firm 'ABB', as the name suggests is programmable, allowing for automatic system operation with desired valve timings pre-set, and imposition of any conditions, if required, on the operation of a particular valve.

5.4 Stacked-up system assemblies and piping

The stacked-up system assembly of the pressure and vacuum conveying test rigs under consideration are shown in figures 5.12, 5.13, 5.14 and 5.15. As is evident from the drawings, in case of the positive pressure system the top of the receiver bin (storage bin) is connected to the separation equipment (bag filter assembly) and the bottom of the receiver bin connects with the blow tank through a knife gate valve, a load isolator (expansion bellows) and a pneumatically controlled double disc valve in the same succession. Manual loading of material into the receiver bin is carried out through an ash feed inlet located at the top of the bin, from where the material flows by gravity into the blow tank as the knife gate and double disc valves are opened. The material is then fed to the conveying line and subsequently returned to the receiver bin, again through a double disc material discharge valve located at the exit (bottom) of the blow tank. Post feed, depressurization of the blow tank vessel is carried out by opening the double disc vent valve in the pipeline connecting the blow tank and the receiver bin, so that mobilization of material in the receiver bin is facilitated due to receipt of air from the blow tank. The arrangement of stacked-up assembly of the vacuum test rig is very similar to that of the pressure system with the storage transfer placed at top being connected to the feeding hopper at bottom through a knife gate valve, a load isolator (expansion bellows) and a pneumatically controlled double disc valve in the same succession. The notable differences between the two systems are, presence of a seat valve for material discharge in vacuum rig unlike a double disc valve in the pressure system, absence of a venting line connecting feeding vessel to the material storage vessel in vacuum system and presence of a vent filter at the top of feeding hopper in vacuum system.

Medium thickness (Class B) pipes made of mild steel, with nominal bores as per the requirement derivable from piping loop layouts and the piping and instrumentation diagram presented in chapter 3, were used for conveyance piping. The applicable standards for specifications and testing of these pipes are IS 1239 and IS 3589. Lengths (maximum 7 meter)

of these pipelines which were well suited for use with the range of pressure being employed in the test rigs, and present the advantages of high strength, corrosion resistance and high toughness, were coupled with flanges welded onto them, to create the required conveyance pipeline network. Furthermore, a flexible piping with a pressure bearing capacity high enough to safely conduct the pressurized flow encountered in the system was used to supply air required by the instruments/control apparatus present at various locations in the test rigs.

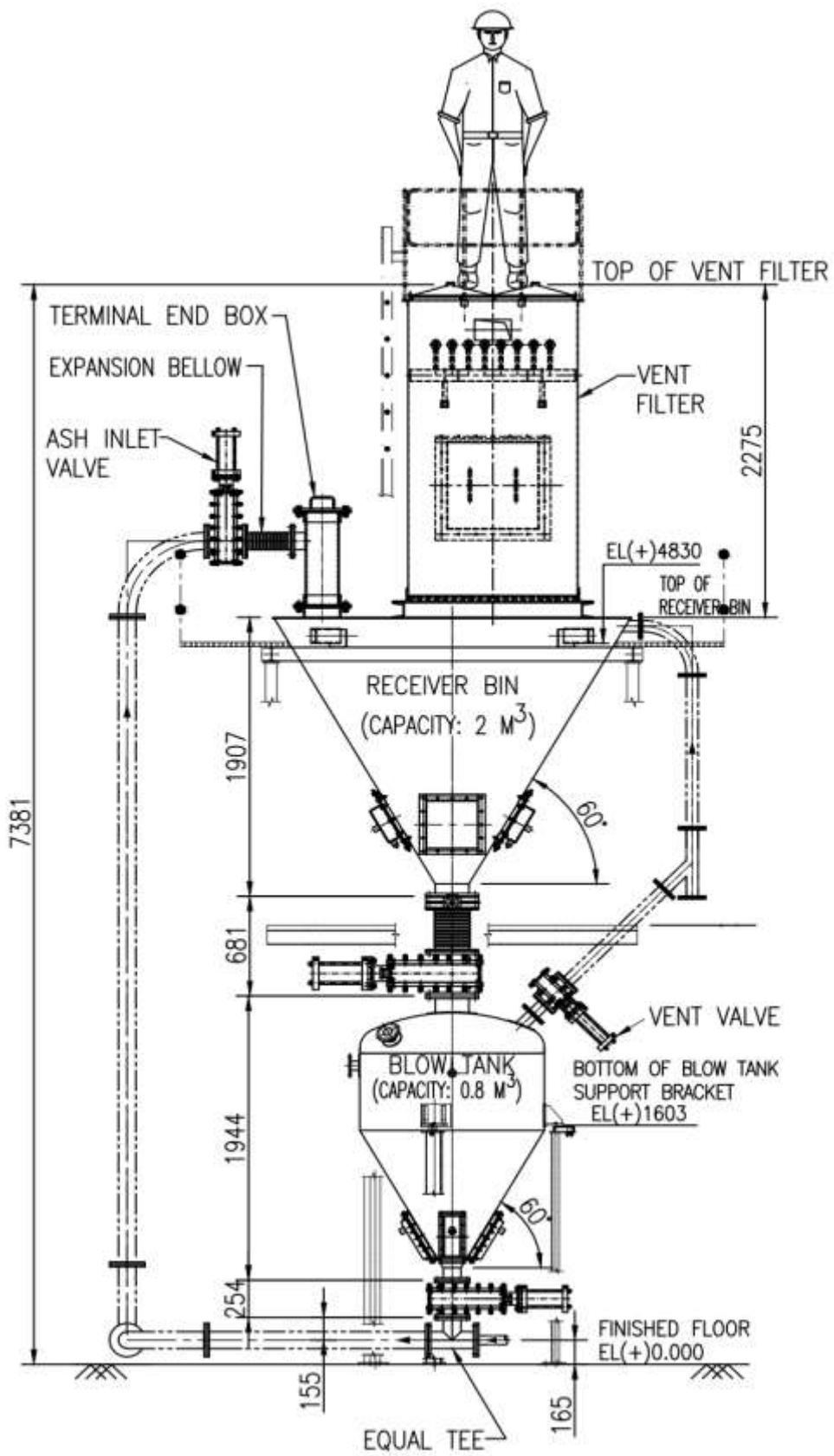


Figure 5.12 – Stacked-up system assembly of the positive pressure pneumatic conveying test rig

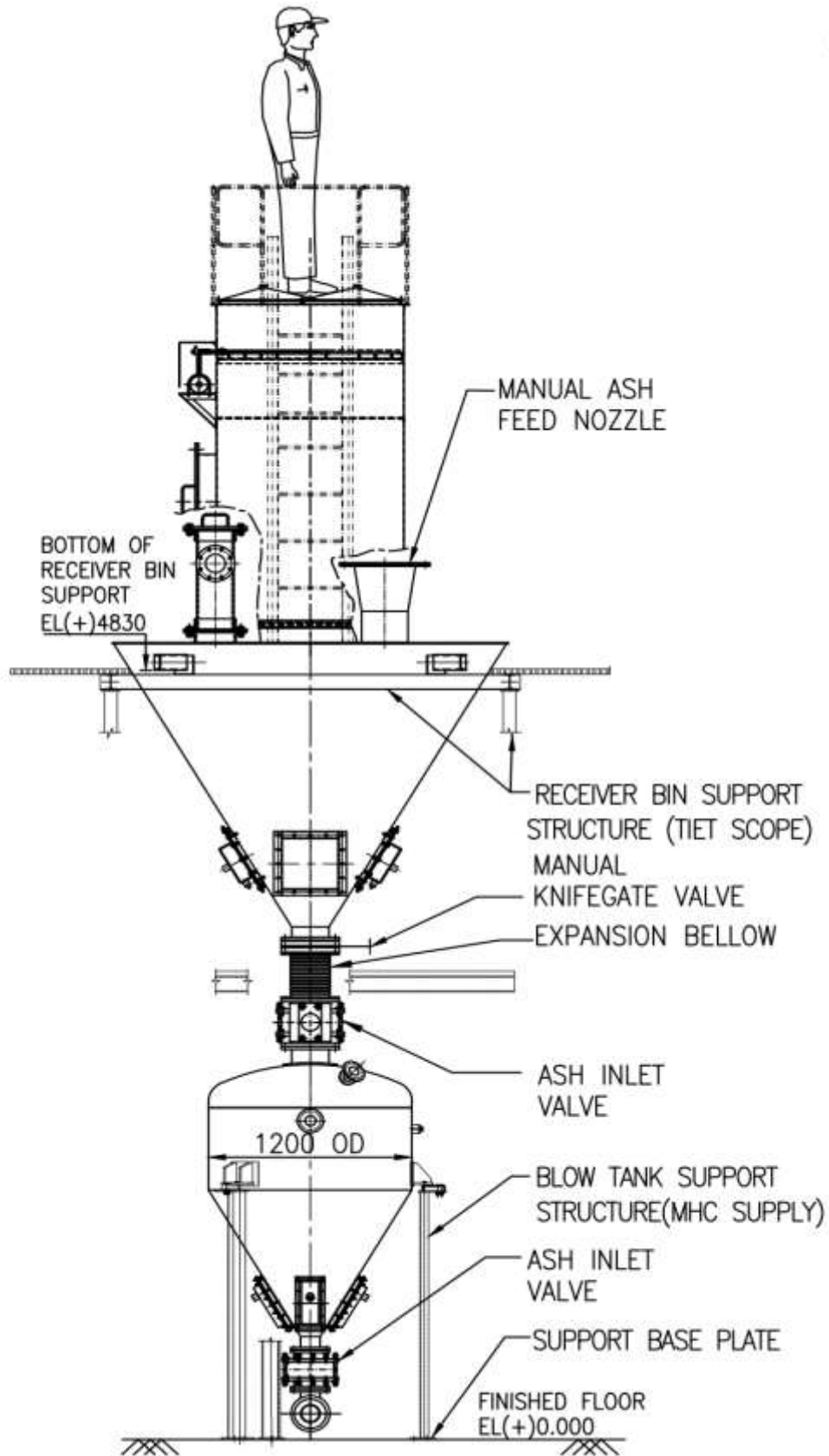


Figure 5.13 – Side view of the positive pressure test rig stacked-up system assembly

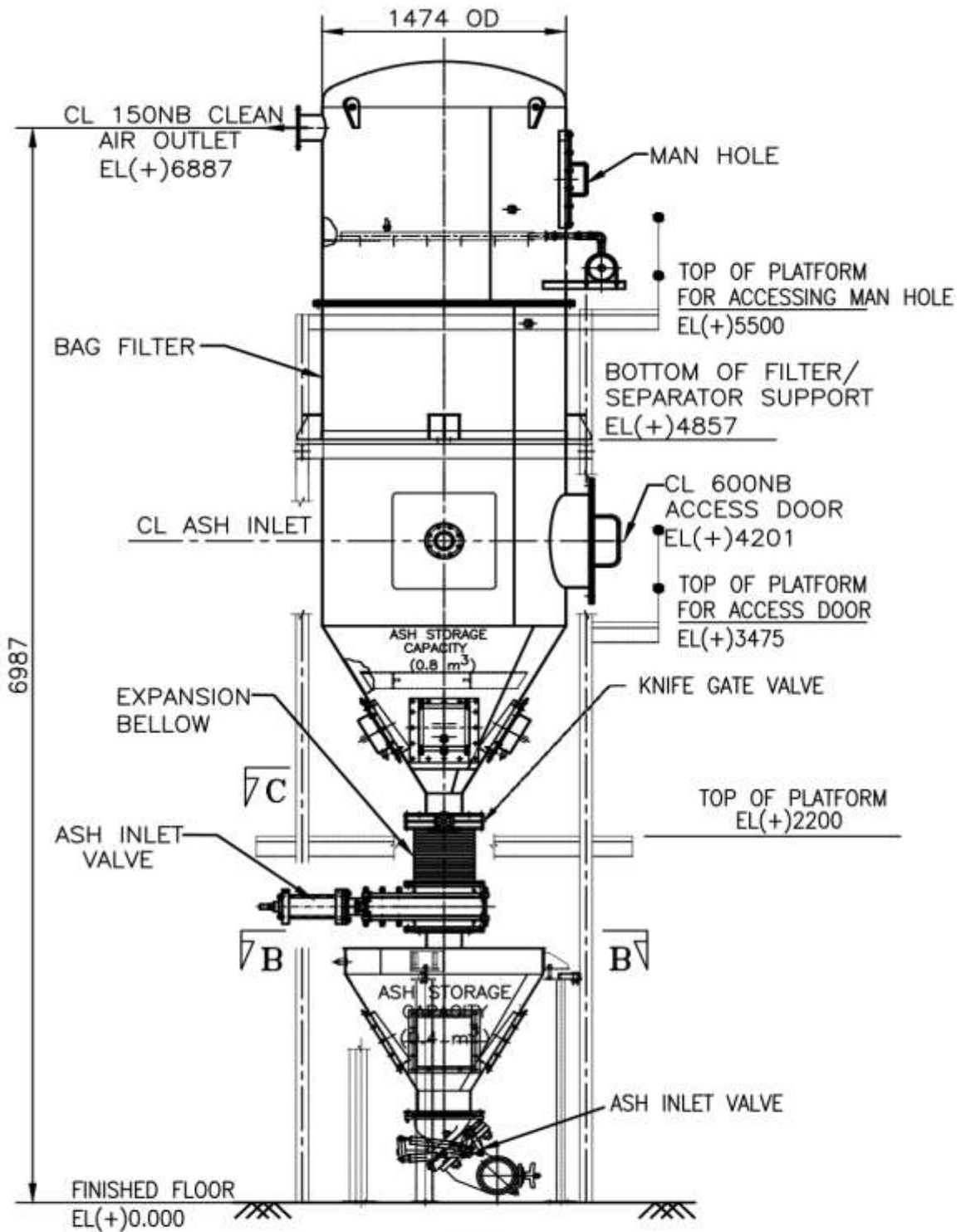


Figure 5.14 - Stacked-up system assembly of the negative pressure pneumatic conveying test rig

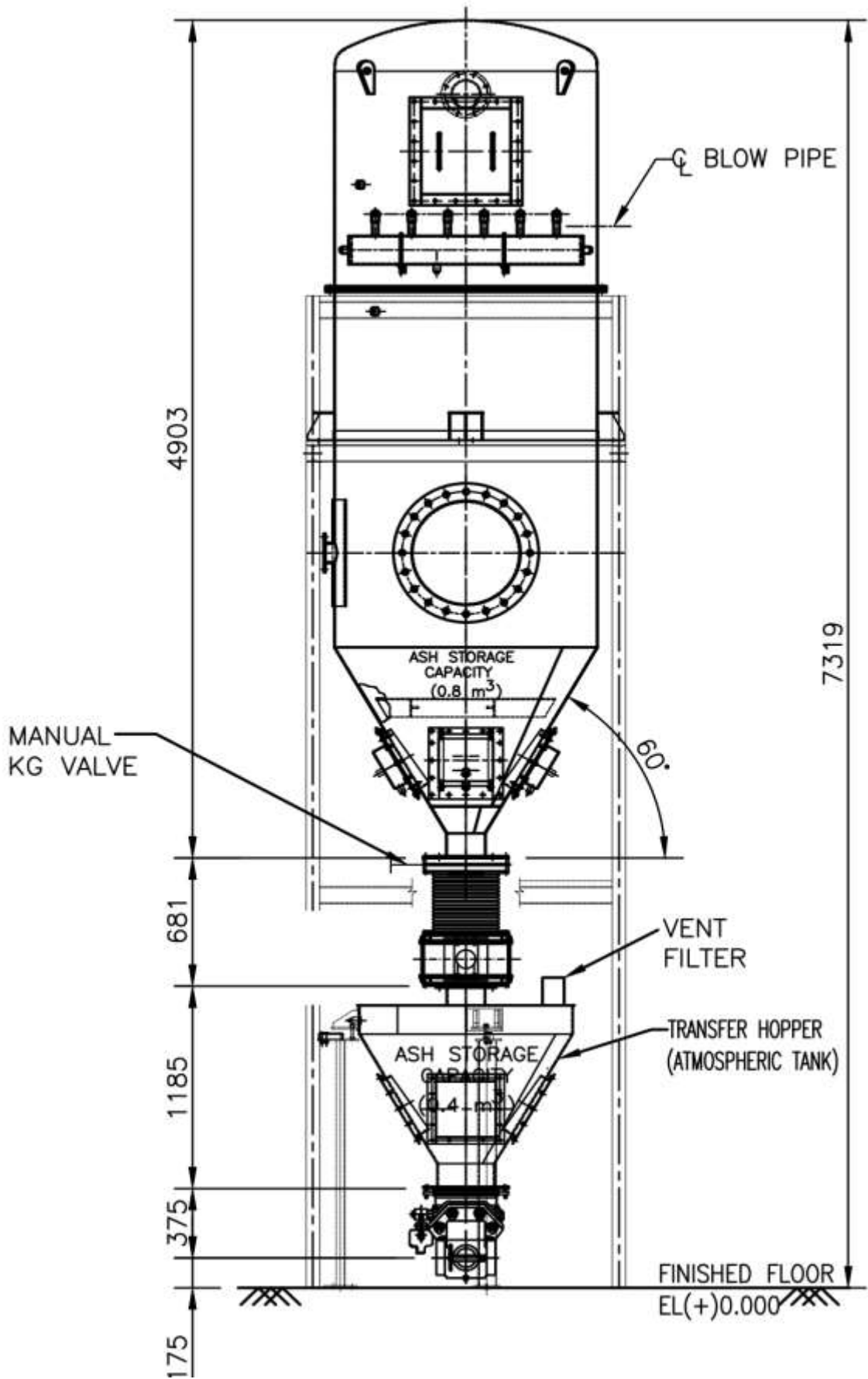


Figure 5.15 - Side view of the negative pressure test rig stacked-up system assembly

A civil structure was designed to support the stacked up assemblies of both pressure and vacuum test rigs. The design for foundation of such a structure was first furnished and subsequently a frame was designed to house and anchor the stacked-up assemblies. Following figures depict the aforementioned design drawings

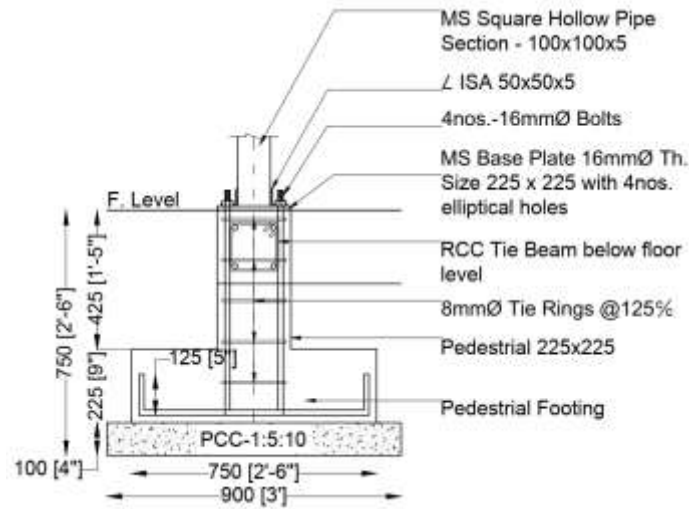


Figure 5.16 – Sectional detail of the foundation

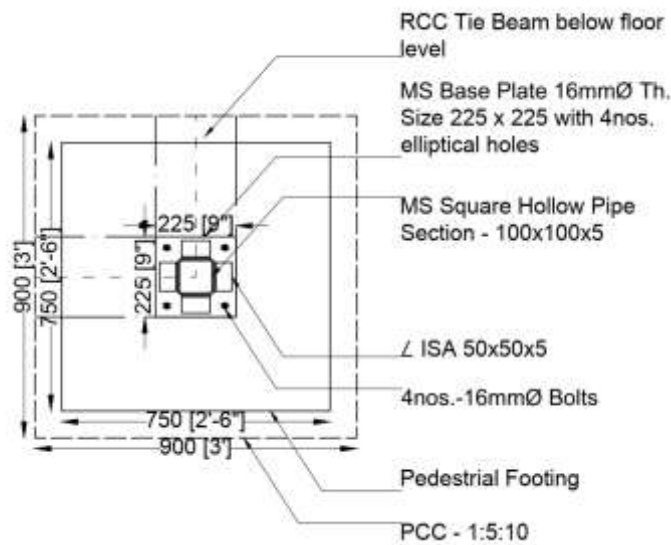


Figure 5.17 – Plan of each member of supporting structure

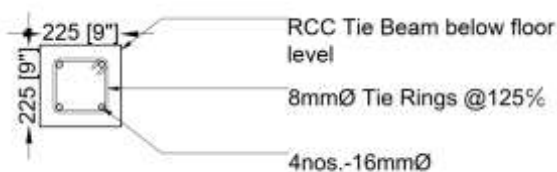


Figure 5.18 – Details of the tie beam

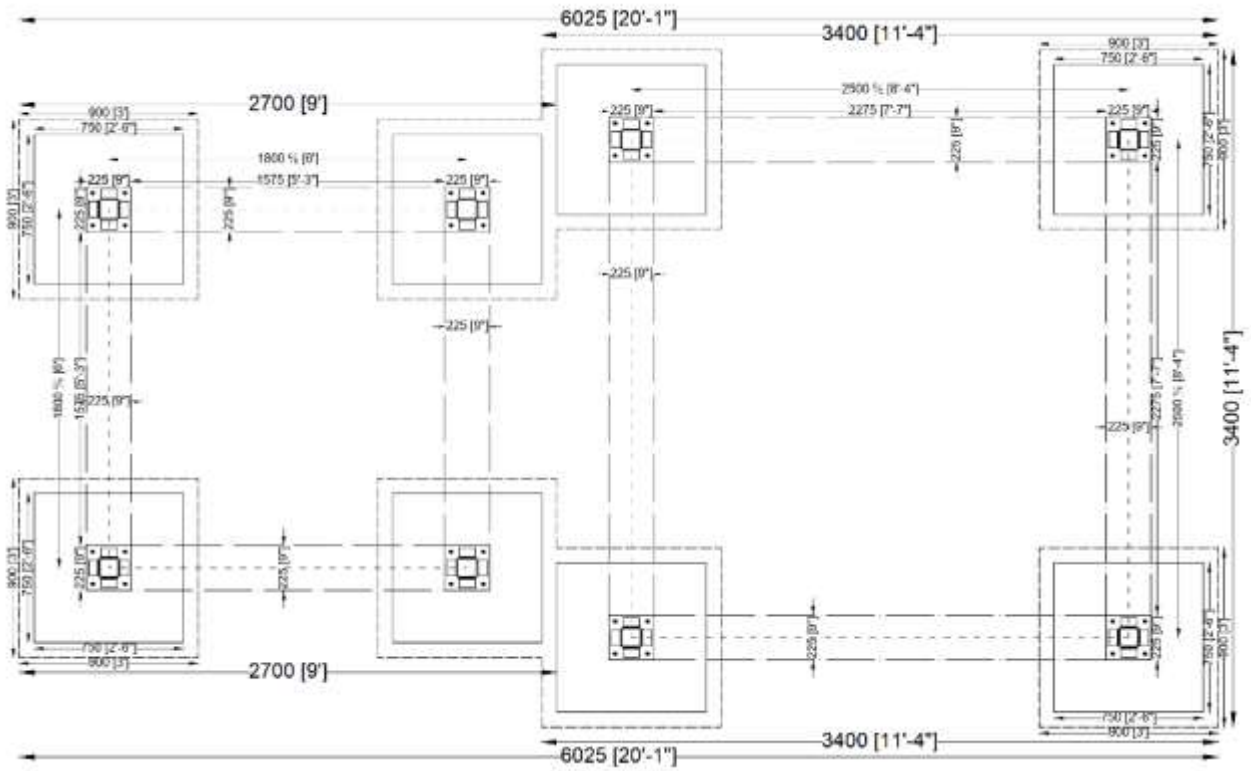


Figure 5.19 – Plan of the foundation of supporting structure

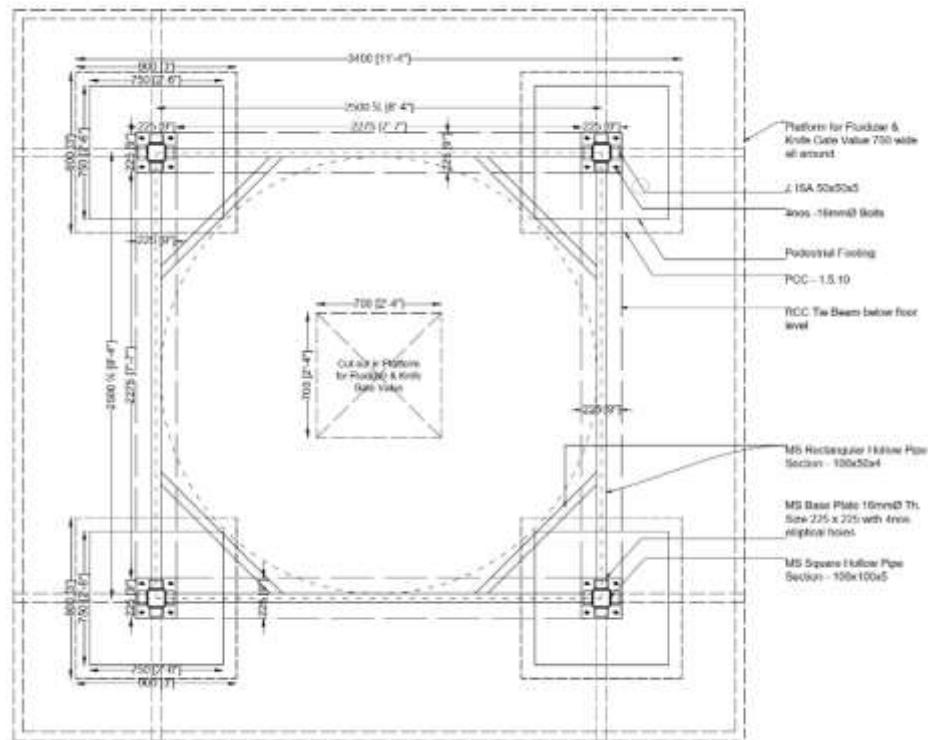


Figure 5.20 Plan of the supporting structure for positive pressure system stacked-up assembly

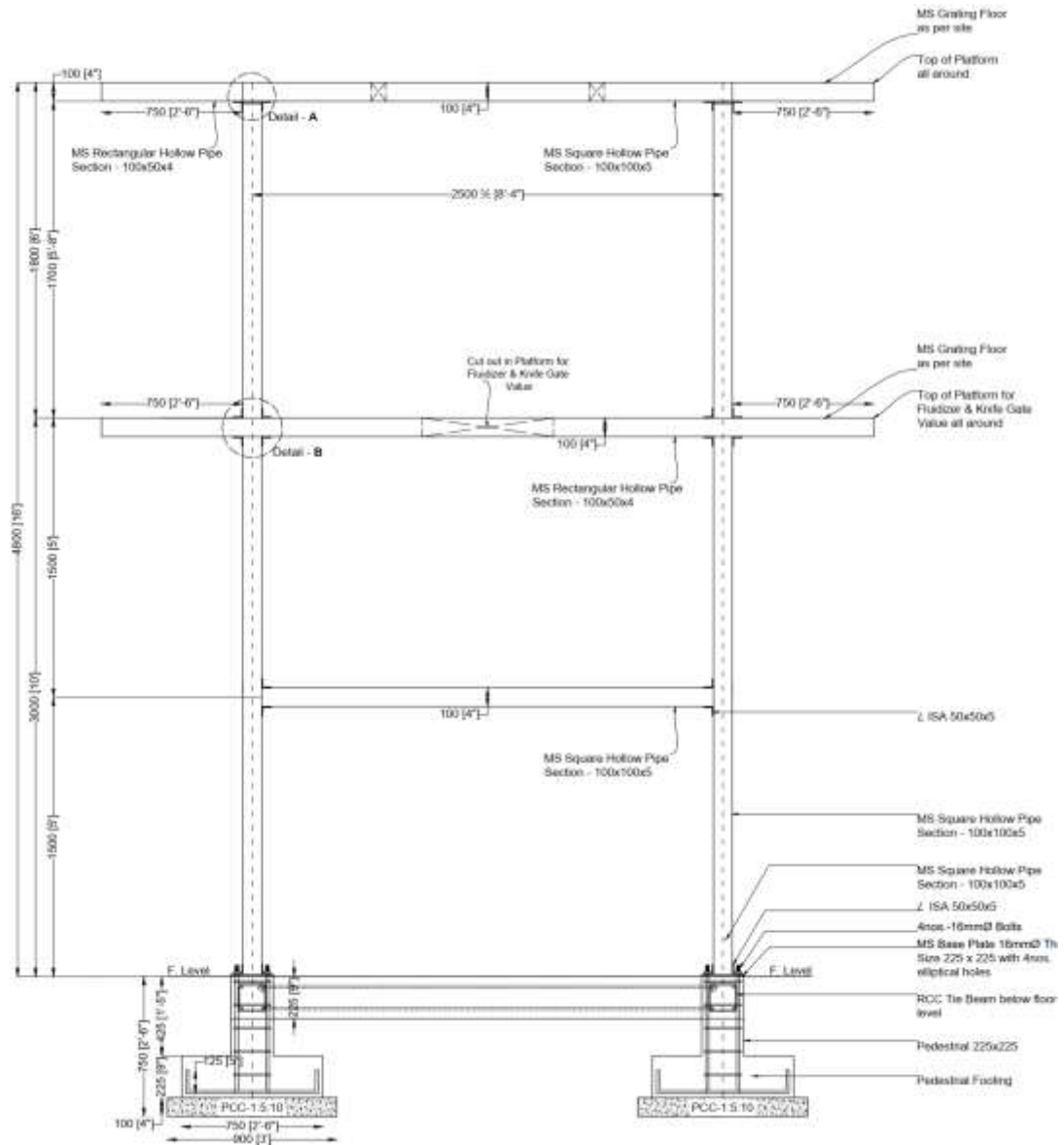


Figure 5.21 – Elevation of the supporting structure for positive pressure system stacked-up assembly

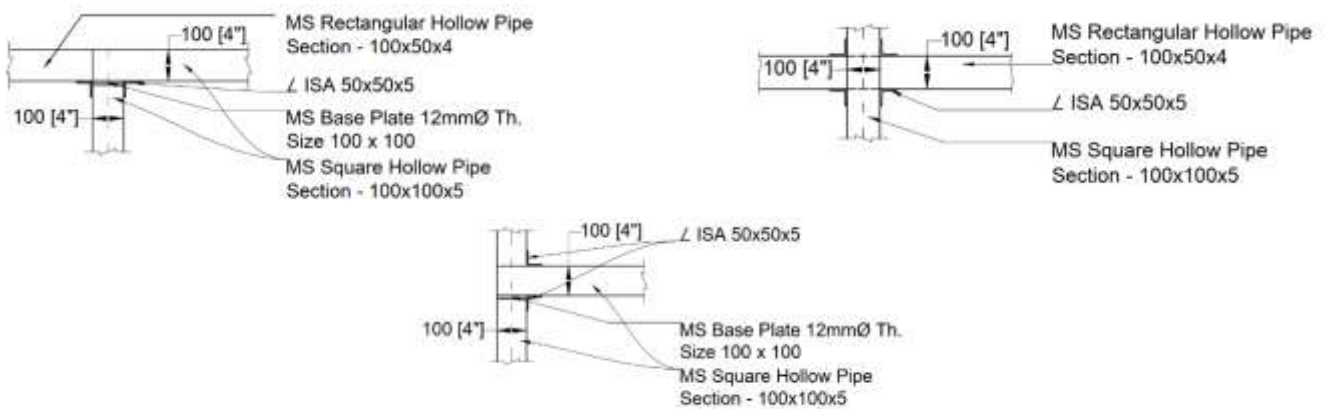


Figure 5.22 – Details of joints and angles used

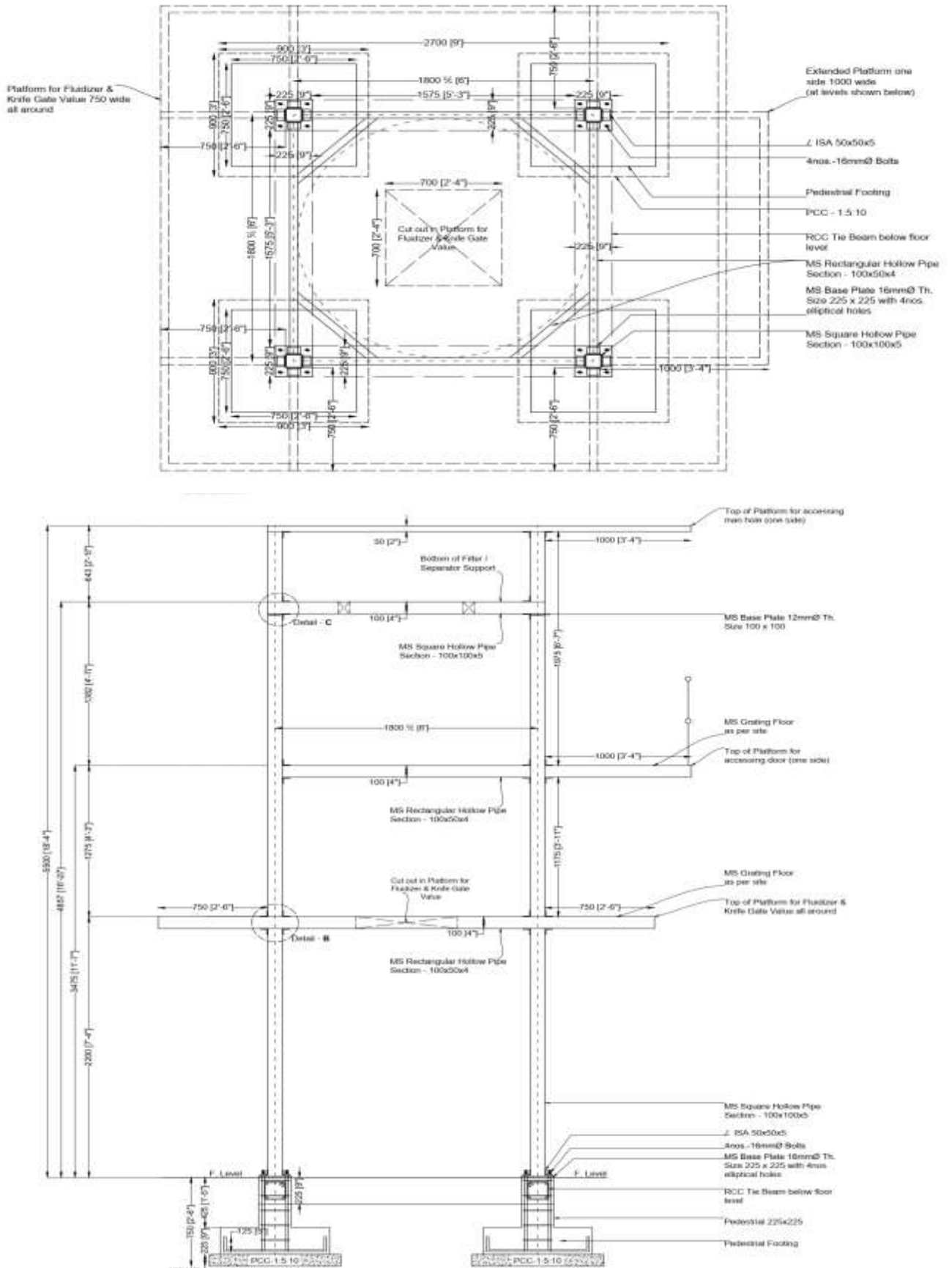


Figure 5.23 – Plan and elevation of supporting structure for vacuum test rig stacked-up assembly

5.5 Electrical connectivity

Electrical connectivity to the system, from the substation is provided through an XLPE/PVC insulated, PVC sheathed, 3.5 core armoured cable of aluminium conductor with a cross-sectional area of 120 mm² and working Voltage of 1100 V. Specifications for the electrical distribution panel, acquired for receipt and onward supply of power to the equipment with sufficient protection, were furnished and tabulate in table --. The total maximum load yielded a maximum amperage of 110 amperes, and the aforementioned cable size was therefore selected from the sizing chart.

Furthermore, amperage of each equipment based on rated loads was calculated and thus cable sizing for supply of power to each equipment from the electrical distribution panel carried out. The following table enlists the specifications of various cables used.

Table 5.5 – Cable sizing and electrical connection scheme for test facility

Equipment	Power Consumed	Motor Rating	Supply Power	Internal Wiring	Connection/Switch
Compressor	BKW 55.2 at duty point	55 KW	415V 3Ph	1100volts grade 120 sq.mm (3 ½ Core)armoured cable	Electrical Distribution Panel
Air Drier	1.7KW	-----	230V 1Ph	3C x 2.5 mm sq. (Cu)	Electrical Distribution Panel
Vacuum Pump	BKW 29KW at duty point	37KW	415V 3Ph	3C X 70 mm sq. (Aluminium)	Electrical Distribution Panel
Heater	3KW	-----	3Ph 4 wire	4C x 2.5 mm sq. (Cu)	Electrical Distribution Panel
Blower	BKW 2.13 at duty point	3.7KW	415V 3Ph	3C x 2.5 mm sq. (Cu)	Electrical Distribution Panel
PLC Panel	1.5KW approx.	-----	3Ph 4 wire. 415V	3C x 2.5 mm sq. (Cu)	Electrical Distribution Panel
Data Logger			1Ph 2 wire. 230V	2C x 2.5 mm sq. (Cu)	PLC Panel
Computer			1Ph 2 wire. 230V	3C x 2.5 mm sq. (Cu)	PLC Panel

Table 5.6 – Specifications of the main electrical feeding panel

INCOMING FEEDER	
	Quantity
250A TP MCCB 36kA,O/L & S/C,TMD Rel.	1
6A SP MCB,10KA	4
Indicating Lights Red (R-Phase,ON)	2
Indicating Lights Yellow (Y-Phase)	1
Indicating Lights Blue (B-Phase)	1
OUTGOING FEEDER	
200A TP MCCB 25kA,O/L & S/C,TMD Rel. for Compressor	1
16A FP MCB,10KA for Heater & PLC Panel	2
25A DP MCB,10KA Air Dryer & Spare	2
37 KW FASD STARTER	
160A TP MCCB 25kA,O/L & S/C,TMD Rel.	1
65A TP Contactor (2NO+2NC)	3
O/L Relay (37-50A)	1
Star-Delta Timer	1
6A SP MCB,10kA	1
Illuminious Push Button Red (ON)	1
Illuminious Push Button Green (OFF)	1
Ammeter (0-150/5A)	1
3.7 KW DOL STARTER	
Component Description	Quantity
16A TP MCB,10KA	1
12A TP Contactor (2NO+2NC)	1
O/L Relay (5.5-8A)	1
6A SP MCB,10kA	1
Illuminious Push Button Red (ON)	1
Illuminious Push Button Green (OFF)	1

Chapter 6: Analysis of data and results

6.1 Findings from pneumatic conveying

Experimentation, conducted as per the experimental plan furnished in chapter 3, led to generation of data, analysis of which yielded the following results of interest.

Effect of Fluidization on material tonnage:

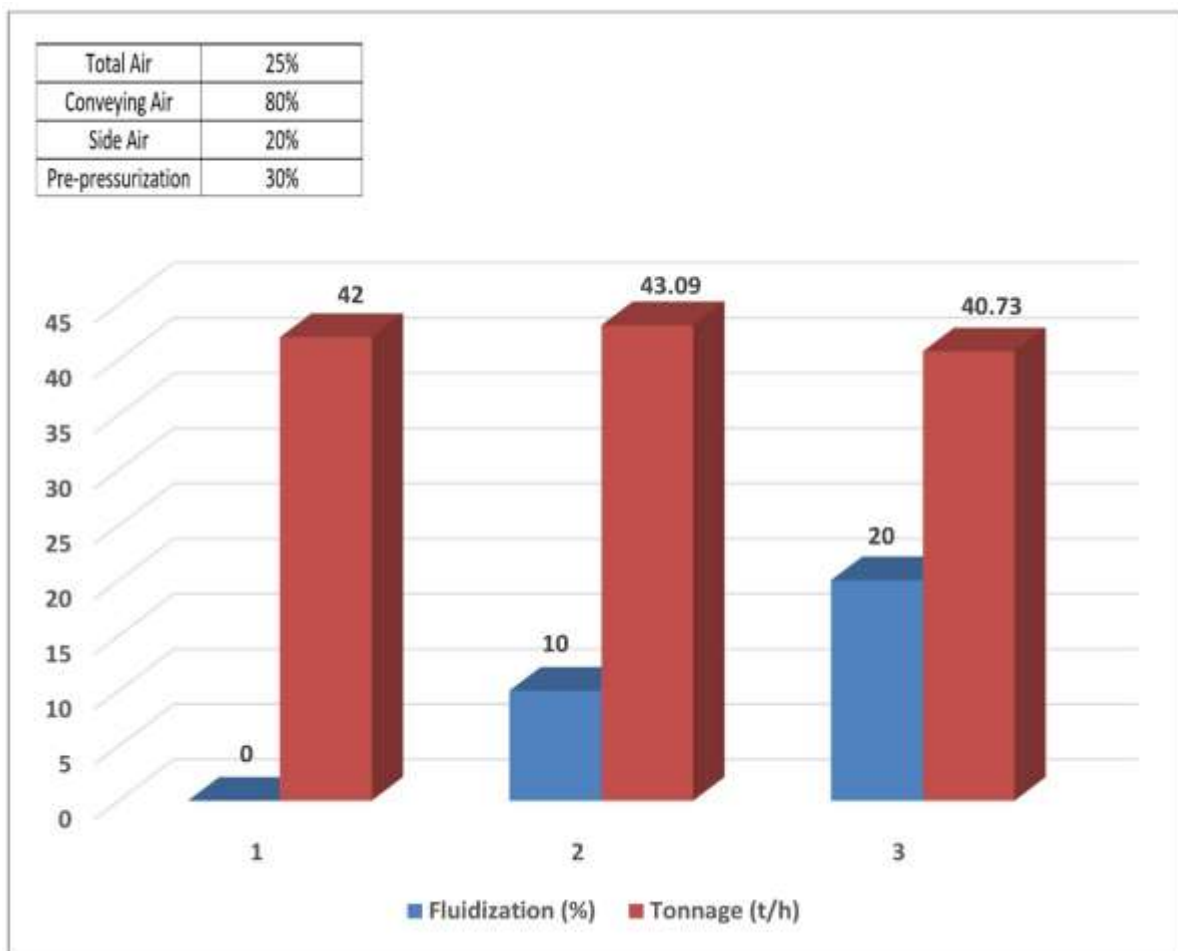


Figure 6.1 – Variation of material tonnage with varied percentage admissions of fluidization air

Figure 6.1 shows a bar graph plotted between percentage of fluidization air admitted to the blow tank feeder and the tonnage (mass flow rate expressed in t/h) of material conveyed. The

other parameters are held constant, with total air supplied being 25%, blow tank pre-pressurization 30%, conveying air 80% and side air 20%. Evidently, fluidization is seen to have no significant effect on tonnage of material, with tonnage at each value of fluidizing air supply showing a negligible departure from the mean value of 41.94 t/h as fluidizing air supply is changed from 10% to 30%. This may be attributable to the fact that the bulk solid under consideration is already aerated and so fluidization does not alter its flow characteristics.

Effect of variation of total air supply on material tonnage:

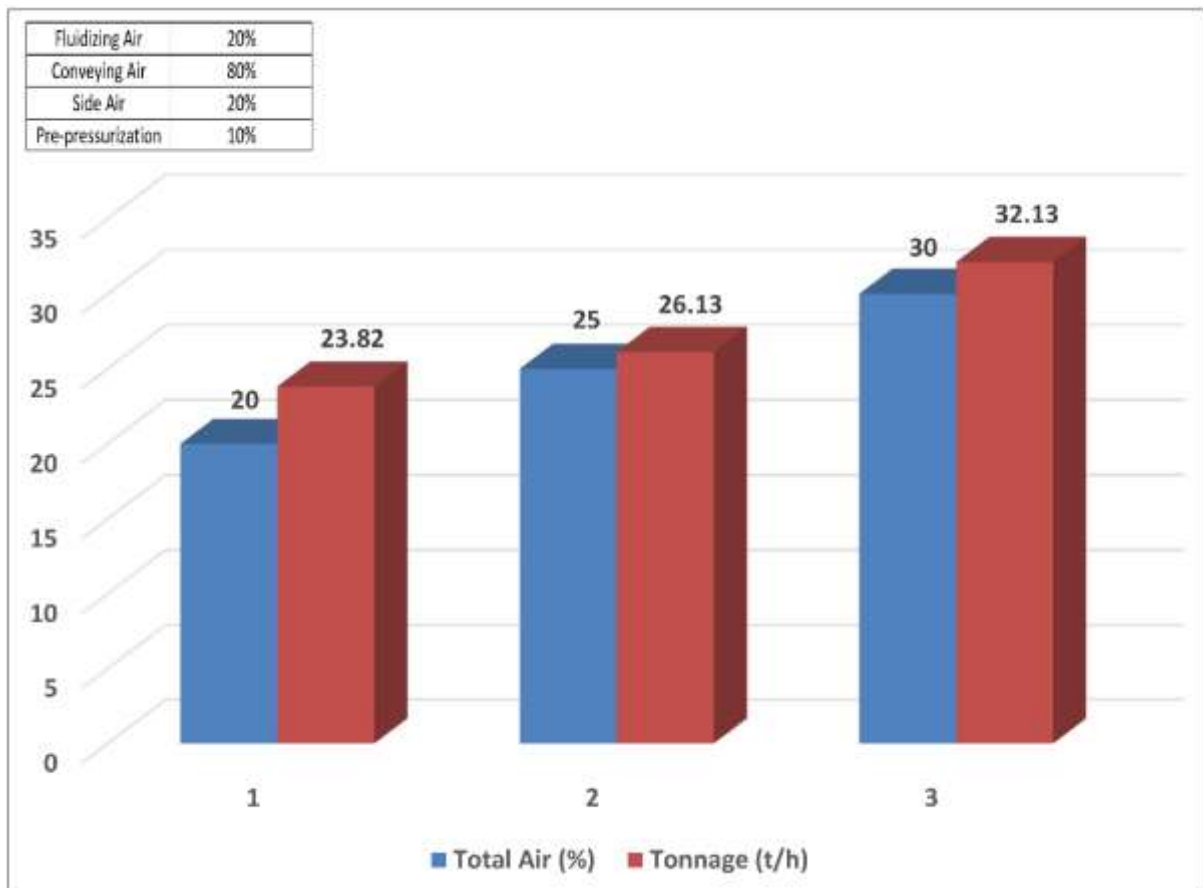


Figure 6.2 – Variation of material tonnage with varied percentage admissions of total air supply

With pre-pressurization of the blow-tank held at 10%, conveying air at 80%, side air at 20%, fluidizing air at 10%, admission of total air supply to the system was varied from 20% to 30%. As can be seen from figure 6.2, with an increase in supply of total air, a steady rise in the mass flow rate of material was achieved. An explanation for this behaviour is given by the fact that

more air means more carrying capacity for a given solid loading ratio (ratio of mass flow rate of material to mass flow rate of air). Furthermore, an increased solid loading ratio is possible with admission of greater quantity of air, since a sizeable portion of the increased supply is passed into the blow tank which leads to an increased discharge rate from the blow tank feeder.

Effect of blow tank aeration with side air on material tonnage:

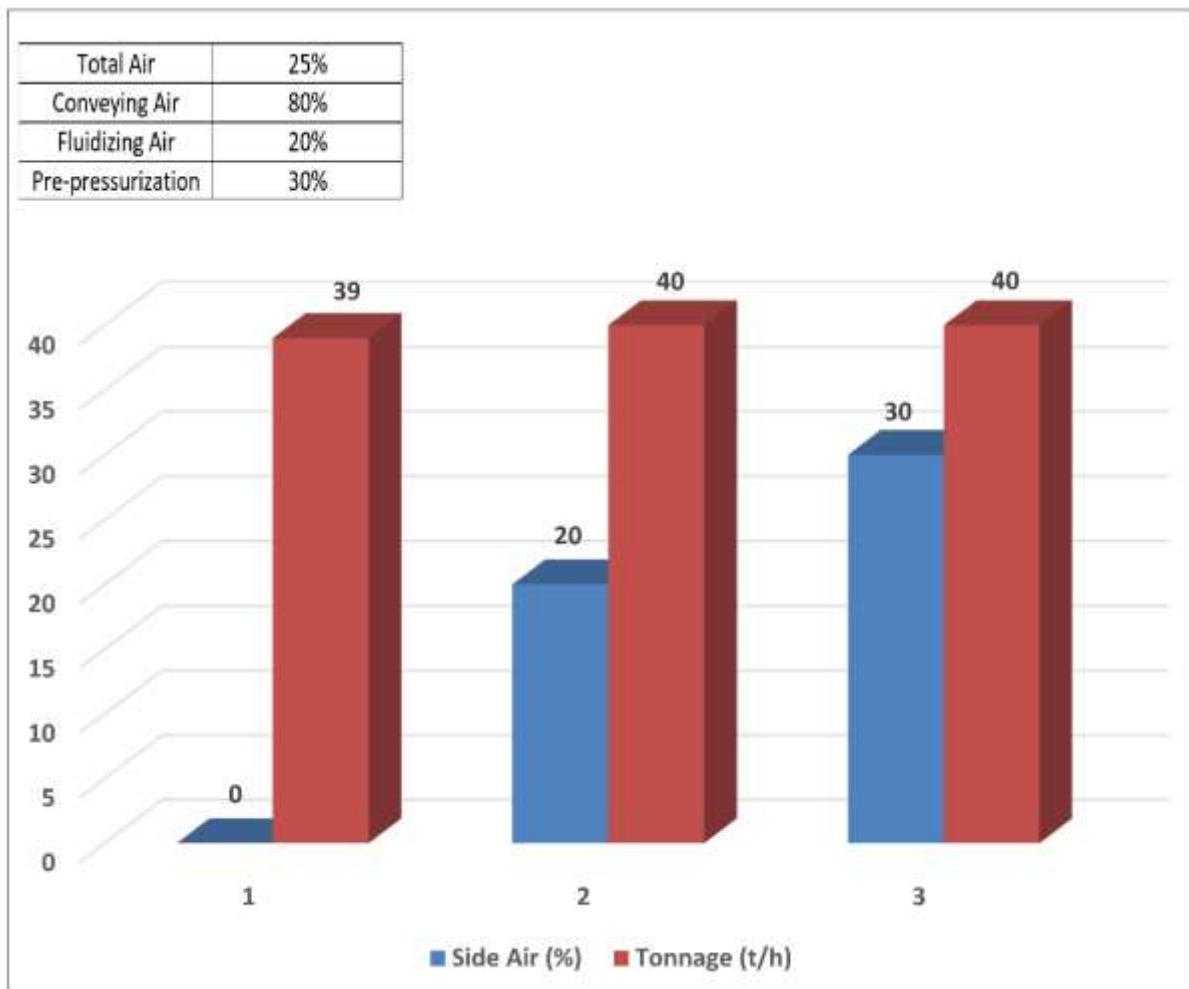


Figure 6.3 – Variation of material tonnage with varied percentage admissions of side air

Aeration of blow tank with side air did not exhibit a significant effect on tonnage conveyed. The most likely cause attributable to this finding is the nature of the product. The product being conveyed is already enough aerated to augment flow to a level maximum possible with aeration

of the material. This behaviour is consistent with the findings of effect of fluidization on material flow rate.

Effect of varying the aeration configuration of blow tank on material tonnage:

Supply of air to the blow tank with two different approaches, first through the inlet at top, and next through the inlet on side of the blow tank, while all other parameters are held constant, showed that aeration through the side yields a very mild increase in material tonnage. The effect is so small, as can be seen in figure 6.4, that no physical or experimental significance can be ascribed to it.

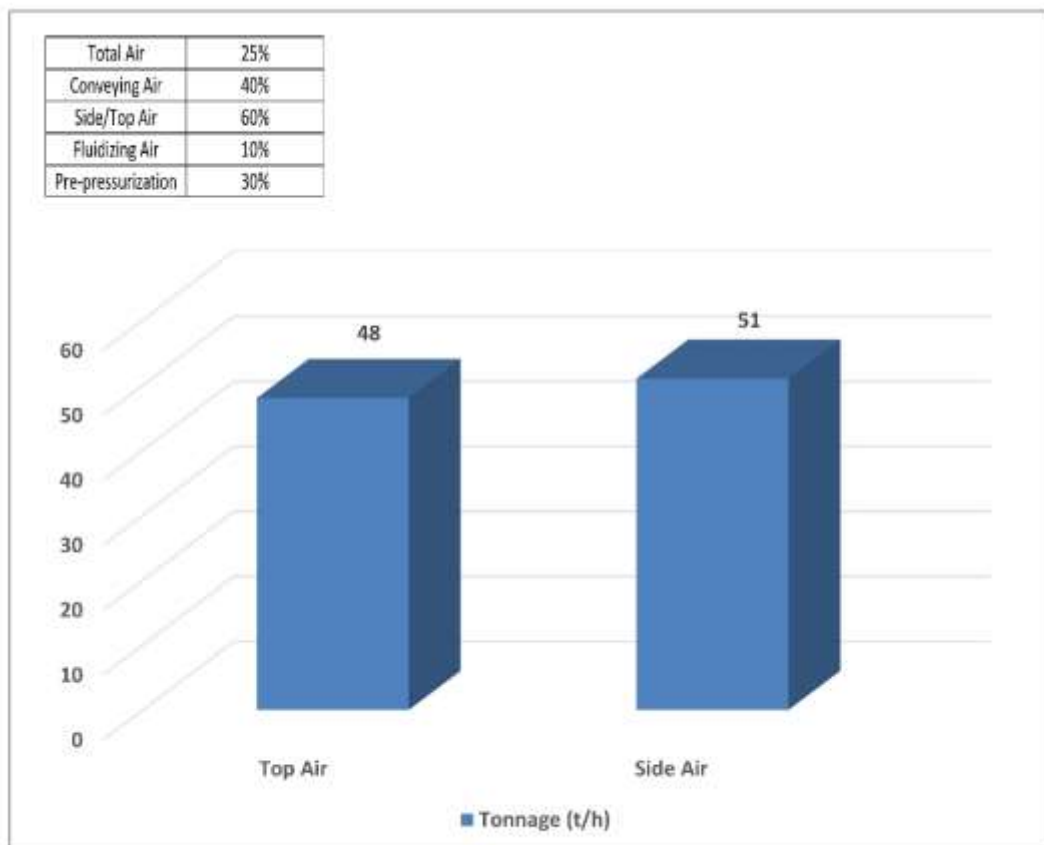


Figure 6.4 – Variation in material tonnage due to admission of side air through different inlets

Effect of pre-pressurization of the blow tank feeder on material tonnage:

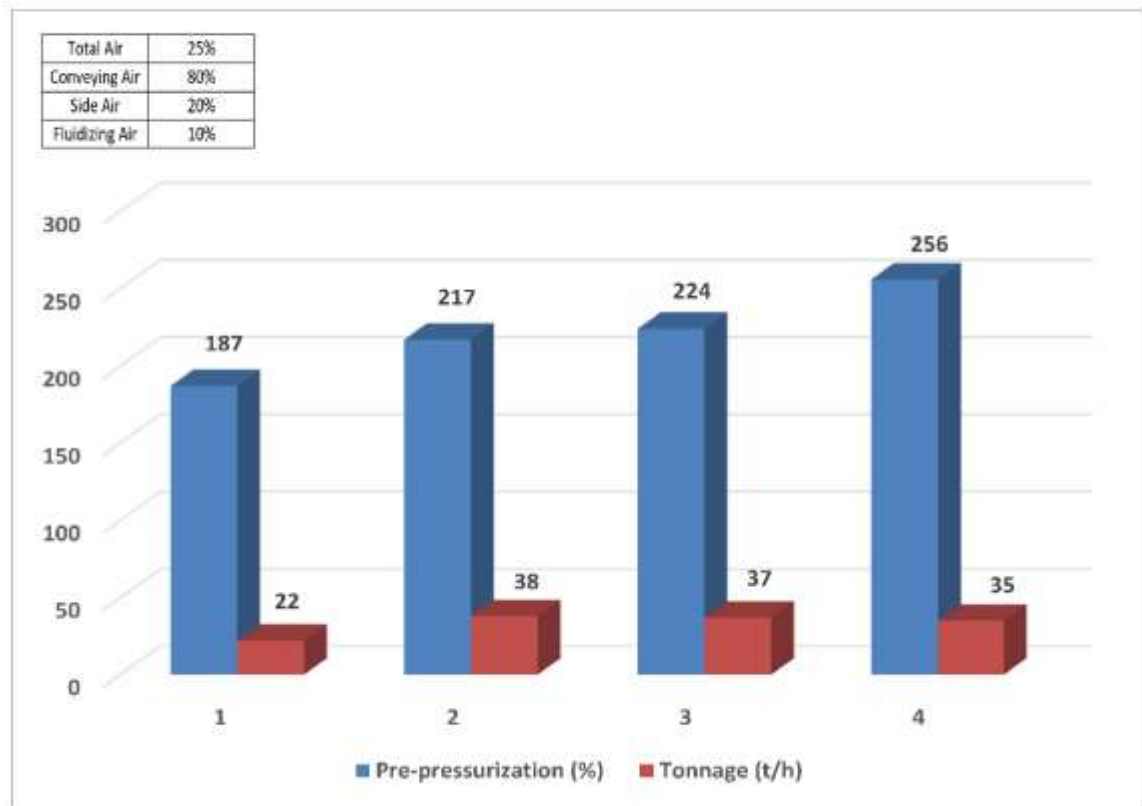


Figure 6.5 – Variation in material tonnage with varied pre-pressurization conditions of the blow tank feeder

Pre-pressurization implies raising of the pressure of the blow tank feeder over and above the total pressure drop experienced in the conveying before material feeding commences. It is carried out with the aim of bringing about larger initial acceleration of air flow at the feeding location to effect a greater material conveyance tonnage. In the present context, as can be inferred from figure 6.5, a significant effect on material tonnage due to pre-pressurization is seen initially with tonnage rising from 22 t/h to 38 t/h as the initial absolute pressure in the blow tank, pre-feeding, is raised from 187 kPa to 217 kPa. However, no significant effect on tonnage is observed as the initial pressure in the blow tank, pre-feeding, is increased beyond 217 kPa. This can be explained by the fact that as pre-pressurization is increased beyond a certain value, material compaction near the bottom of the blow tank feeder begins to occur leading to an increase in local bulk density. This causes increased friction which now

counteracts the greater initial push being provided by pre-pressurization for material discharge. Further pre-pressurization thus does not have much effect on material tonnage.

Effect of varying relative proportions of conveying and side air on material tonnage:

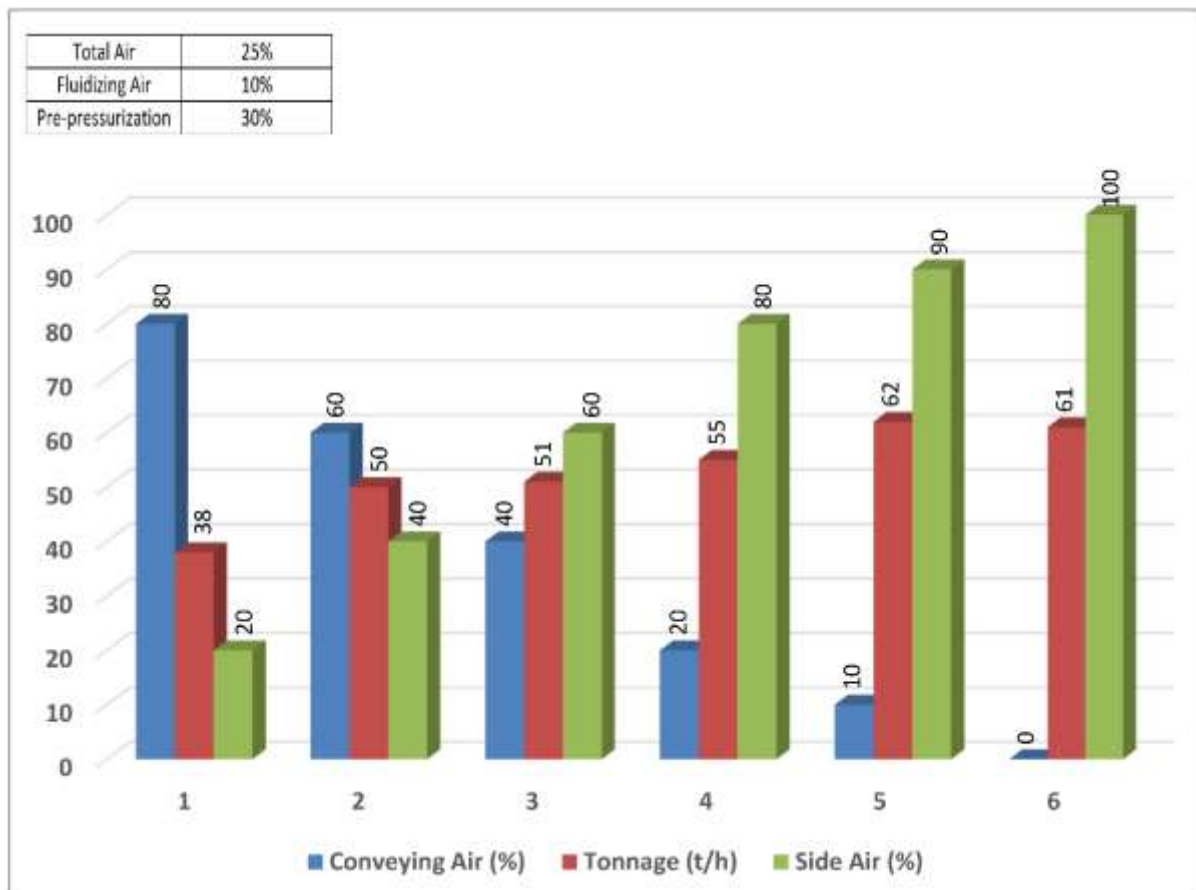


Figure 6.6 – Variation in material tonnage with varied relative percentages of admission of side air to the blow tank feeder and conveying air to conveyance pipeline

With other parameters held constant, relative proportions of conveying air sent to the conveyance pipeline and side air fed to the blow tank feeder were varied. It is observed that tonnage undergoes a gradual increase as the percentage of conveying air is curtailed, and made up for, using side air. This leads to admission of a greater amount of air into the blow tank feeder where in the material is constantly pushed from the top while it is being discharged and the gas-solid suspension flowing out of the blow tank is conveyed as such, resulting in higher material discharge rate and thereby higher tonnage.

6.2 Characterization results

The results of characterization through properties mentioned in the experimental program are tabulated below.

Table 6.1 – Values of properties for product characterization

Properties →	d_{10} (μm)	d_{50} (μm)	d_{90} (μm)	ρ_b (kg/m^3)	ρ_t (kg/m^3)	ρ_p (kg/m^3)
Product ↓						
Remote Silo	5.4	43.61	170.7	1000.7	1383.4	2103.62

d_{10} (μm), d_{50} (μm) and d_{90} (μm) represent sizes (diameters) of particles below which lie the particle diameters of 10%, 50% and 90% mass of the sample respectively. A particle size distribution of the material sample under consideration is thus obtained. ρ_b (kg/m^3), ρ_t (kg/m^3) and ρ_p (kg/m^3) represent the values of loose poured bulk density, tapped density and particle density respectively.

A scanning electron microscopic image of the conveyed fly-ash sample is shown below.

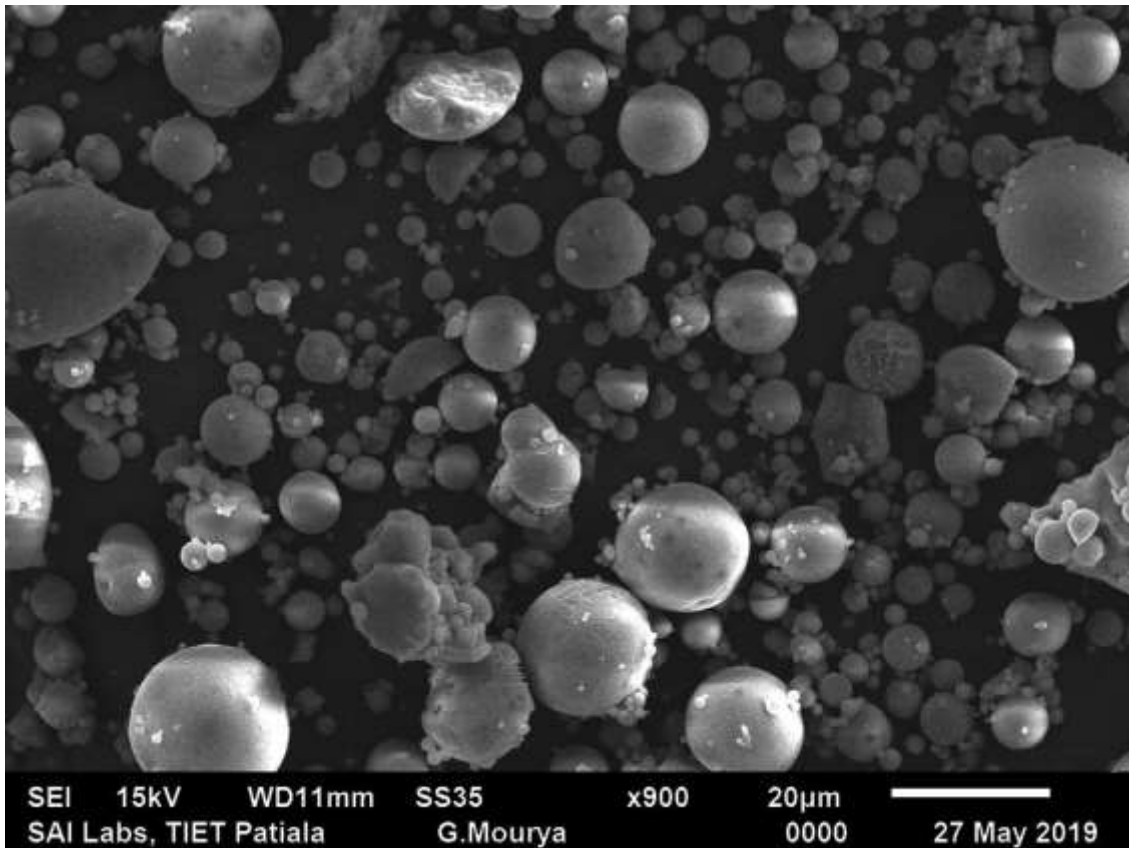


Figure 6.7 – Scanning electron microscopic image of the conveyed product

The image captured in the scanning electron microscope at SAI Labs, TIET Patiala shows the smooth and spherical particles of the product which implies that the product will show an ease in fluidization/aeration. Keeping in view this fact, the product will have a tendency to be well aerated, unaided. This validates the reasoning given for behaviour of flow observed as fluidization air supply and side air supply to the blow tank feeder are varied.

Chapter 7: Conclusions and future scope of work

7.1 Conclusions

Conclusively, fluidization does not seem to play much role in augmentation of mass flow rate of material. This is so due to the nature of the sample of fly-ash (Geldart Group A) conveyed and the observation thus holds good only for this product. For the same reason, aeration of the blow tank feeder with side air is also observed to have no effect on material tonnage. However, routing the major portion of total air supply to the side air inlet of the blow tank, at the expense of conveying air fed directly to the conveyance pipeline, results in an increased material tonnage. Pre-pressurization of the blow tank feeder before commencement of material feeding into the conveyance pipeline also leads to a substantial augmentation in material tonnage.

7.2 Future scope of work

Following are cited to be potential problems subjectable to onward research

- i) In this work pneumatic conveying experimentation with only one product belonging to the Geldart classified group 'A', that is fly-ash obtained from NTPC coal fired power station, has been carried out. An extensive experimental program could be run with a variety of Geldart group 'A' bulk solids to ascertain flow behaviour and possible generalizations.
- ii) An investigation into the working and therefore influence of fluidizing arrangements fitted onto the vessels on flow behaviour is required. Such an investigation will elucidate if fluidization air is admitted at all operating conditions with equal ease and enlist possible factors that can retard the process of fluidization in course of system operation.

REFERENCES

- Jones, M.G., Mills, D. and Mason, J.S. (1987). A Comparison of the Performance of Top and Bottom Discharge Blow Tank Systems. *Bulk Solids Handling*. 7(5): 701-706.
- Jones, M.G. and Williams, K.C. (2003). Solids friction factors for fluidized dense phase conveying. *Particulate Science and Technology*. 21: 45-56.
- Keys, S. and Chambers, A.J. (1993). Scaling pneumatic conveying characteristics for pipeline pressure drop. In the proceedings of National Conference on Bulk Materials Handling, Preprints, Capricorn International Resort, Yeppoon, Queensland, 22-25 September: 205- 213.
- Keys, S. and Chambers, A.J. (1995). Scaling pneumatic conveying characteristics for Pipeline Pressure. *Powder Handling and Processing*. 7 (1): 59-62.
- Klinzing, G.E., Rizk, F., Marcus, R. and Leung, L.S. (2009). *Pneumatic Conveying of Solids*. Publ. Springer. 3rd Ed.
- Lech, M. (2001). Mass Flow Rate Measurement in Vertical Pneumatic Conveying of Solid. *Powder Technology*. 114: 55-58.
- Weber, M. (1991). Friction of the air and the air/solid mixture in pneumatic conveying. *Bulk Solids Handling*. 11 (1): 99-102.
- Williams, K.C. and Jones, M.G. (2004). Numerical model velocity profile of fluidized dense phase pneumatic conveying. In the proceedings of 8th International Conference on Bulk Materials Storage and Transportation, Wollongong, NSW, Australia, 5-8 July: 354-358.
- Wypych, P.W. and Arnold, P.C. (1984). The use of powder and pipe properties in the prediction of dense phase pneumatic transport behaviour. In the proceedings of Proceedings of the Technical Program, Pneumatech 2, International Conference on Pneumatic Conveying Technology, University of Kent, Canterbury, England, 4-6 September, Organised by the Powder Advisory Centre, London, England.
- Wypych, P.W. (1989). PhD. Dissertation: Pneumatic Conveying of Bulk Solids. University of Wollongong, Australia.

- Wypych, P.W. and Arnold, P.C. (1989a). Meeting the demands of long distance and large throughput pneumatic transportation. In the proceedings of 3rd International Conference on Bulk Materials Storage and Transportation, Newcastle, Australia, 27-29 June: 78-83.
- Wypych, P.W., Kennedy, O.C. and Arnold, P.C. (1990). The future potential of pneumatically conveying coal through pipelines. *Bulk Solids Handling*. 10 (4): 421-427
- Wypych, P.W., Kennedy, O.C. and Arnold, P.C. (1990a). Pneumatic conveying of pulverized and crushed R.O.M. coal. In the proceedings of 4th International Conference on Pneumatic Conveying Technology, Glasgow, Scotland, 26-28 June.
- Wypych, P.W. (1999). Pneumatic Conveying of Powders over Long Distances and at Large Capacities. *Powder Technology*. 104: 278-286.
- Wypych, P.W., Hastie, D.B., Frew, I. and Cook, D.M. (2006). An Experimental Investigation into the Feed Rate Capacity of Rotary Valve and Blow Tank Feeders. *Particulate Science and Technology*. 24: 165-179.
- Xu, G.L., Chen, X.P., Liang, C., Liu, D.Y., Xu, P., Cai, J.Y. and Zhao, C.S. (2013). Conveying Characteristics of Pulverized Coal in a Top Discharge Blow Tank System. *Chemical Engineering Technology*. 36(5): 773-780.
- Xu, G.L., Chen, X.P., Liang, C., Xu, P., Cai, J.Y. and Zhao, C.S. (2012). Effect of Operating Conditions and Powder Properties on Pneumatic Conveying Characteristics of a Top Discharge Blow Tank. *Chemical Engineering Technology*. 35(11): 1998-2006.

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Publication

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Wenhao Pu, Peng Lu, Chen Yue, Dong Han. "Research on Flow Characteristics of Pulverized Coal in a High Pressure Blow Tank", Volume 1: Aircraft Engine; Ceramics; Coal, Biomass and Alternative Fuels; Controls, Diagnostics and Instrumentation, 2012

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47

Cecilia Arakaki, Ali Ghaderi, Arild Sæther, Chandana Ratnayake, Gisle G. Enstad. "Air mass balance for mass flow rate calculation in pneumatic conveying", Powder Technology, 2010

Publication

48

C Burne. "Pneumatic conveying of abrasive bulk solids at Anglesey Aluminium Metals Limited: four case studies", Proceedings of the Institution of Mechanical Engineers Part E Journal of Process Mechanical Engineering, 01/01/2000

Publication

49

Submitted to University of Bradford

Student Paper

50

B. K. Datta. "Influence of Fines Concentration on Pneumatic Transport Capacity of Alumina", Particulate Science And Technology, 1/2003

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