ABSTRACT

This research project sponsored by the Centre for Agricultural Technology(CAT) is seen as assistance to the Agricultural and Beef Industries which are among the major contributing industries to the Australian economy. The project involves a study of the grass seed industry in Central Queensland by conducting a market survey to determine the existing mechanisation in the seed industry and the problems faced by this group of farmers. The main aim of the study was to arrive at a design of an improved and more effective novel grass seed harvester, specifically designed by considering all the characteristics unique to grass seeds and the details which earlier research had indicated would help to increase the quality and the total yield, from the current 40-60% to about 80-95%. Earlier research on an air-assisted brush type of harvester indicated the problem of having to draw the seeds through the fan before separation. This seemed to cause trauma to the seeds which revoked the seed germination capacity. This research work has therefore been focused on designing a seed separator to be located ahead of the fan which could successfully separate the seeds of various shapes and sizes, from the large volume of conveying air required for suction, before entering the fan. A curved duct concentrator and a uniflow cyclone were selected as part of the experimental work and a theoretical approach was developed to understand and define the problems unique to the separation of the grass seeds. The results of these experiments and their comparison with the previous existing separators and harvesters have been discussed in this thesis. The technique for fan design for the harvester and finally the design specifications of a self driven harvester and a tractor mounted harvester were developed by incorporating all the harvester units, i.e the header unit, the separator and the fan.

DEVELOPMENT OF A NOVEL HARVESTER FOR GRASS SEEDS AND CEREAL CROPS

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Dissertation submitted in partial fulfilment
of the requirements for the degree of
Master of Engineering

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DECLARATION

This thesis covers the development and sets a specification for a Novel Vacuum Harvester for pasture seeds. The main area concentrated upon and developed is the air-solid separator unit of this harvester. The development of such a harvester would be of invaluable contribution specifically to the Pasture seed industry and the Beef industry and to the Australian Agricultural industry at large as 50% of its total income is from the pasture seed industry both directly by sale of quality seeds and indirectly by pasture improvement to the beef industry. If the separators designed prove to be only 50% as successful, as indicated in the laboratory experiments, it would help in solving many of the problems presently experienced by farmers in economising the harvester costs, eliminating considerable time lost in post harvest treatments and in stabilising the pasture seed industry.

This new harvester has been designed for a project supported by the Central Queensland University, C.A.T and D.P.I. It has been developed by taking into consideration the various problems faced by the farmers, seed merchants and consumers, to demonstrate the efficiency of this new harvester. Once the efficiency, easy operation and maintenance free parts of this harvester are fully established, it is believed that this harvester will become more popular and widely used in the pasture seed industry.

The work contained in this thesis is a direct result of the experiments carried out by the author and has not been previously submitted for a degree or diploma at any other tertiary institution to the best of my knowledge. This thesis contains no material previously published by another person except where due reference is made.

Leena Mary John

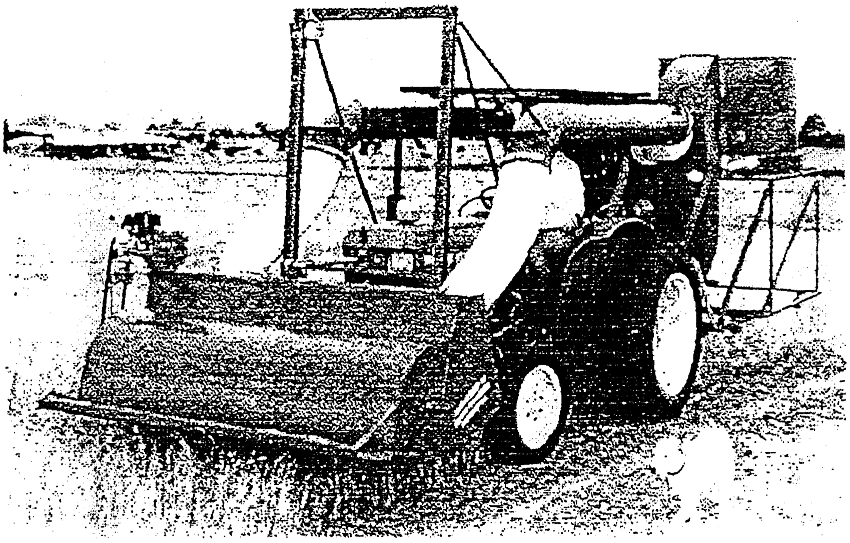
1.0 INTRODUCTION

This thesis concerns with the design of a novel vacuum pasture seed harvester and was funded by the Centre for Agricultural Technology (CAT), the Central Queensland University (CQU) and the Queensland Department of Primary Industries (DPI). The project was put forward as a necessity for the farmers of Central Queensland by the request of CAT and DPI. The research plan includes developing of a scaled up model of the harvester, to be designed and tested at CQU. Later the technology was to be transferred to CAT, for further tests to be conducted to investigate its performance in the field. Before arriving at a design, it was decided to conduct a market survey to study the pasture seed industry and collect information from the farmers and the merchants, to enable in identifying the problems that could be resolved within the period of a post graduate study.

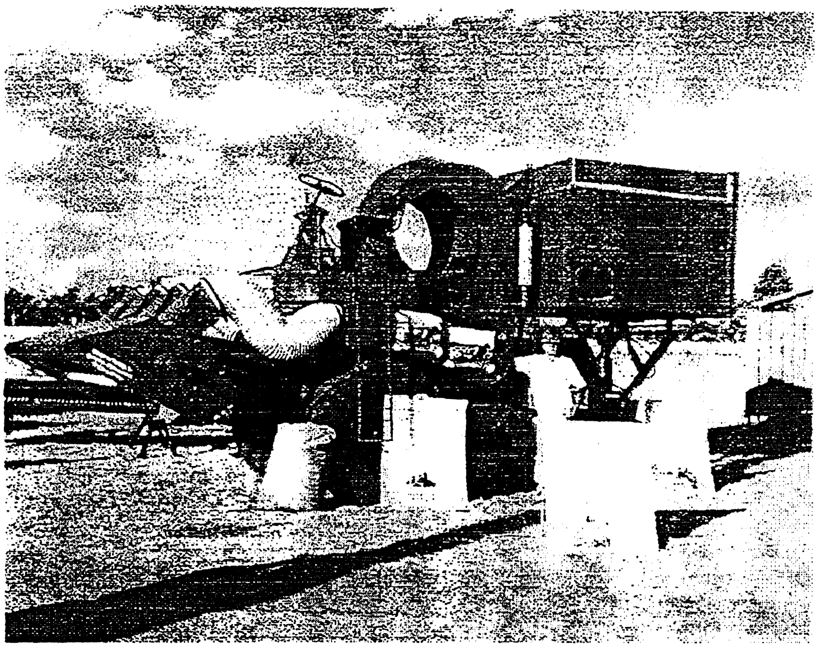
This chapter has been furnished to present the substance of the work that has been presented in this thesis. It discusses the data obtained in the market survey conducted among the pasture seed farmers and the merchants. The market survey assisted in defining the main problems faced by the grass seed industry with the currently practised harvesting techniques and post harvest technology. Once the existing problems were established, the next step was to decide on methods to overcome these problems.

The latest grass seed harvesters incorporate, a picking head which gathers only the ripe seeds from the grass seed head; fans to furnish air assistance at the picking head to harvest and convey the grass seeds to the collection point; separators to abstract the small volume of seed from the large volume of conveying air and temporary storage sites on the harvester. An example of such a harvester is shown in Figure 1.1.

Some developmental work has already been done on the design and specifications of header unit for the vacuum harvester. In view of the available data on the header unit, the air velocities required at the header unit and the power availability, design details for the separator and, the fan and ducting dimensions had to be developed. It was possible to decide upon a particular type of separator and conduct experiments on it and achieve excellent advancement in the separation of a mass of seeds from a mass flow rate of air on a laboratory scale. However further tests will have to be carried out to achieve a balance between the high volumetric flow rate of air and the size of the separator. This could be accomplished by keeping the air velocity constant (as we have noticed from experiments conducted at CQU, the air velocity at the picking head is the main criteria for lifting the seeds off the seed heads) and trying to reduce the volumetric flow rate of air at the header to an acceptable proportion, to be able to build more economic separators of smaller sizes. On that account after completion of this work, it was realised that further research work would have to be done on the harvester header to get an optimum flow rate with required velocity to reach an optimum size for the separator. Though these problems were recognised it was not possible to address all of them within the time frame provided for a post graduate course. The project was therefore confined to identifying a separator able to satisfy the above mentioned requisite and to perform a detailed



Tractor mounted, air assisted brush harvester designed and produced at Kawana Engineering.



Grain Header converted to Pasture seed Harvester at Kawana Engineering.

Figure 1.1

study on it and to explain the separation process. The main objective in selection of this separator were,

i. a separator that could cope with such a large mass flow rate of air and be successful in abstracting the small mass flow rate of seeds from the large mass flow rate of air.

ii. a separator that could be positioned ahead of the fan to avert the needless stress, experienced by the seeds on passing through the fan as is the current practice in all the air-assisted grass seed harvesters.

The following paragraph summarise the contents of the different chapters in this report.

Chapter 2 concerns with the market survey report and the inference arrived at from the collected data. For conducting the market survey report, a questionnaire was prepared and copies were sent out to 696 farmers having properties along the Dawson and Mc Kenzie rivers. Another questionnaire was prepared which was personally taken to the pasture seed merchants in the Rockhampton area. They were interviewed on a personal basis. On receiving the completed questionnaire the responses were reviewed to assess the problems and needs of the pasture seed farmers and the merchants from their view point.

The feedback was of considerable help in establishing the problems experienced by the farmers. It helped in identifying the machinery that are presently employed for harvesting the various pasture seeds and the equipment used for other post harvest treatments. The most outstanding element of this survey was that, as yet, no distinct harvester had been designed or built targeted at the original characteristics of pasture seeds and the needs of the pasture seed industry. Most machinery and equipment in contemporary use are either modified forms of other grain harvesters and post harvest equipment or equipment built by farmers themselves, which was the main source contributing to the problems being encountered by the farmers and the merchants. These problems were also adversely influencing the relationship between the farmer, the merchant and the consumer. The pasture seed industry contributes to nearly fifty percent of the gross income of the agriculture industry in Australia. In view of the above, it was recognised that manufacturing of such a harvester, essentially for harvesting pasture seeds by taking into account its various unique characteristics, would be of considerable assistance to these farmers, the agricultural industry, the beef industry and to Australia at large.

Experimentation had already been initiated and details perfected on the header and fan units of a vacuum harvester. It was decided to use these available data and pursue the research work on to the separation unit of the harvester and, to develop detailed specifications for the fan.

Chapter 3 primarily describes the fundamentals of fluid mechanics theory applied to fan design. This was intended at assisting any non-technical person who would be interested in designing a fan according to their needs. The different fan theories and equations, and the diverse types of fans available in the market has been described in some detail. An attempt was also made to present the various aspects considered in terms of the advantages and disadvantages, influencing the selection of fan for the harvester along with the significant characteristics of the grass seeds. The approach taken for the fan design was based on the velocity triangle theory. The details and equations of the theory have been described in this chapter. This also assisted in identifying the fan to be considered for the experiments and also in establishing the fan specification for the harvester.

Chapter 4 is the central chapter of this thesis and deals comprehensively with the different types of harvesters used in pasture seed industry, their history and development over the course of years and their advantages and disadvantages. Finally the latest innovation in the harvester and the modification required have been discussed in detail.

The discussion then proceeds to the adversity experienced by the farmers with the existing model, of the air-assisted grass seed harvester. In view of the positive features of the existing harvester, it was realised to improve or try and eliminate the negative aspect of the harvester. The fundamental model regarded was the Woodward Flail-Vac type harvester which was researched and improved upon by the use of air assistance to get about 60-80% of harvesting efficiency (i.e of the total harvested seeds 60-80% of seeds were live seeds and would germinate) by the year 1990. Although the efficiency improved, it was observed that further development could be accomplished on this model in areas of redesigned harvester picking head, improved fan design and separator. All of these three factors had been the subject of earlier research. In this research work the emphasis was on the development of an efficient separator unit. The chapter then deliberates on the factors governing the selection of the separator unit for the Master research project. A significant amount of research had already been concluded by the Kawana Engineering at Rockhampton and the CQU on the header and fan units. Therefore reasonably reliable information were available on the flow requirements and air velocity required for achieving good harvesting efficiency. The main flaw observed in the harvester was that although 85% harvesting efficiency was achieved, a total of only 5-10% of live seeds were obtained from this 85% of harvested seeds [1]. This could only be attributed to the fact that currently in all harvesters the separator is placed behind the fan. The harvester drawing the seeds along with the large volume of air therefore required all of the seed to have to go through the fan unit before entering the separator and get separated. Results in the earlier work indicated that this seemed to traumatise the seeds to such an extent as to affect its germination. Hence it was concluded that the this small volume of seeds should be separated from the large volume of air, before the air seed mixture is transmitted through the fan, thus preventing any loss to the harvested seed by having to pass through the fan.

At this point it would be prudent to accentuate on the limitations faced while working on this project to optimise the size of the separators. Based on the existing data the picking head mechanisms developed up until now were designed to accommodate the variations in the pasture seed head heights, so as to bend maximum seed heads to the same height to get maximum harvest yield at each pass of the harvester over the grasses. Thus having to avoid making many passes. The header aperture height is what determines the air flow rate required for a particular harvester with a given head width. This factor i.e the header length and width had already been determined and established by experiments carried out earlier. Therefore according to the data obtained from the previous research the separator to be designed, would have to be capable to handle air velocities of 12-20 m/s and volumetric flow rates of around 30 m^3/s.

Although there are many pneumatic (Air-solid) separators separating solids from air, the ratio between the two volumes is generally same, but in the case of a vacuum grass seed harvester the volume of seed harvested is much lower than the volume of air required to harvest and convey the seeds. This is so because from previous research it was realised that the separation of the seeds from the seed heads had to be done with minimum mechanical use (such as rotating brushes or beaters) to avoid damage to the seeds. Therefore the only other option was to harvest the ripe seed heads by using high velocity of air.

Based on this the chapter proceeds with a definition of the separation. process, the type of separators and their classification. As this thesis deals with air as the conveying fluid, the discussion has been limited to only the different types of air-solid (aero-mechanical) separators. The advantages, disadvantages and the operation of the different types of air-solid separators have been covered in the supplement to the thesis.

Finally considering the unusual phenomena of having to separate a small volume of seeds or solids from a large volume of air, an attempt has been made to show the criteria by which the separators for the thesis were selected. The theory applied to explain the separation of grass seeds from the transporting air flow by means of forced path curvature followed by gravitational separation is discussed in detail.

As grass seed harvesting deals with an unusually large volume of air in comparison with a smaller volume of solid particles, the seeds had to be concentrated into a small portion of the total cross section of the transporting air flow by some form of centrifugal separator and then further be separated from the remaining underflow air by means of a deceleration chamber or settling chamber. Further on, the parameters affecting the separation process and the theory applied have been discussed to explain the separation process. Ways to calculate the drag coefficient experienced by different seeds and then to arrive at the drag force that would be acting on these seeds are developed. This would enable any interested person, to arrive at a value for the centrifugal force, needed to be developed to compensate for the drag force. Thus be able to design the length and size of a separator unit, needed for any system. The rest of the chapter then deals with the development of the experimental models, the experiments conducted and the results with the tables, graphs and figures. Discussions of the same along with conclusions arrived at and suggestions for further improvement are included.

In chapter 5, attempt has been made to arrive at a scaled up model of the actual harvester and to arrive at specifications for the header, separator and fan units of the same. The problems encountered while scaling up have been discussed. Suggestions have been made to overcome some of these problems. Finally the conclusions for the thesis at large has been deliberated upon.

In the early stages of the project, specifications for a Prototype (self driven) and Production model (self driven), budget figures for manufacturing costs for a Prototype (self driven), hybrid model from second hand parts (self driven) and tractor mounted Pasture Seed Harvester were drawn up in collaboration with Kawana Engineering, Rockhampton. The suggested specifications are given in Appendix 2 and the budget figures for manufacturing costs have been given in Appendix 3.

2.0 MARKET SURVEY REPORT

2.1 INTRODUCTION

Australia is essentially an agricultural oriented country with the agricultural and cattle industry being its major source of income. Of the total 50% income from agriculture, a good percentage is contributed by the pasture seed industry, both directly by the sale of quality seeds and indirectly by providing a boost to the beef industry, with the supply of high standard, good quality feed. Depending on the type of pasture from 3 to 40 hectare of native pasture are needed to safely carry one beef animal whereas, 0.1 to 3 hectare of a productive sown pasture can support the same stock. Marketable age for meat animals are also more than halved with these intensive sown pasture and forage systems [2].

The narrowing margins between production costs and returns have increased the need for a low cost pasture technology, and management strategies to maintain such pastures. A more competitive economic climate also implies a need for more efficient pasture management and better integration of sown pasture, native pasture and forage crop feed sources on properties.

Contemporary harvesters, used to harvest most of the pasture seed on the market are modified Headers, Cotton Pickers, Wheat Strippers and so on. Very few (e.g Flail-Vac) specialised machinery have been as yet developed for harvesting of quality pasture seeds.

Pasture seed market was always very inconsistent. It underwent sudden growth and development into the present form in the early to middle nineteen sixties.This change was in response to the rise in demand of beef and dairying industries. This industry in Australia has been observed to be a very unregulated industry. Unlike other industries it is not subject to any commodity boards or marketing authorities, has no restrictions in production, no specific subsidies or incentives or price guarantees. Production is determined by individual decisions, natural calamities and market forces. The only regulation is for seed quality at sale and the quarantine aspect of import. Commercial seed is mostly produced by private medium scale farmers (100 - 500 hectare), mixed farmers having cattle and growing other crops with pasture seed production forming a part of the overall enterprise. Very little seed is contracted to be sold in advance. The greater bulk of seed is sold to merchants and a small amount is sold between farmers. Each farmer negotiates independently with merchants. Few farmers know at planting or even at harvesting if they have an assured market or price for their seed. The industry has no union to represent them and they only have small associations linked to Australian Seed Producer's Federation to represent their interests at a national level. All this has contributed to a deep mistrust between the different sectors of the industry i.e the producer, the merchant and the members of official bodies. Market intelligence is kept secret and when divulged, not believed. All this lack of information and co-ordination in turn has contributed to the large fluctuations in prices and uncertainty in the pasture industry.

2.2 PROBLEMS FACED BY PASTURE SEED INDUSTRY

A brief summary of the problems faced in this industry are outlined here. The problems encountered may broadly be classified into:-

1 Engineering

2 Agricultural

3 Marketing.

4 Ancillary Services.

2.2.1 Engineering

While designing harvesting machinery the problems encountered are:-

1 Unsynchronised crop / ripening.

2 Variable seed structure.

3 Post harvest treatments.

4 Quality tests.

2.2.1.1 Unsynchronised crop and unsynchronised ripening

Unlike other agricultural crops, there are many varieties of grasses in which maturity of seeds does not occur at the same time. Some seed heads ripen before others. There is differential ripening even within each head. It has been observed that in a panicle type head, seed ripening occurs progressively from the top of the head and tips of the lateral branches towards the bottom and centre of the head over a period of 10 to 14 days. This sometimes takes longer in the autumn. The practice of heading grasses before the seed head is fully mature in the hope of collecting more seeds results in a large percentage of unripe seeds being collected. These immature seed will never germinate and hence reduce the overall quality of the harvested seed. The potential yield and the actual harvested yields vary greatly according to the crop and the method of harvesting. In some tropical grass the potential seed yield is found to be as high as 1 to 1.5 tons/hectare which includes all the seeds produced over a period of some weeks. Because of the unsynchronised ripening of the seeds the quality of the harvested samples is generally much below the potential, as some immature seeds are often found with the harvested seeds.

Hence, in countries with cheap labour, harvesting is done by hand picking which gives high quality seeds/hectare. However this process is very labour intensive and time consuming and is not a viable proposition for a country like Australia. In countries like Australia where labour is costly, mechanisation is a necessity. Most of the machinery used till the present times collect a lot of trash along with the ripened seeds, in some cases cutting off the head of crop with both mature and immature seeds, along with the leaf, stems and often tangled material. This considerably reduces the total yield well below the potential yield and often live seed product is found to be no more than 5% of the total bulk [1]. By direct heading with a All Crop Header which is the accepted harvesting system in practice, the efficiency is found to be 40 to 60% of the total seed yield carried on the crop at harvest.

2.2.1.2 Variable seed structure

Pasture seeds cover a large variety of grass seeds and legumes which vary in sizes and makeup from light fluffy ones to large heavy ones. Hence harvesters had to be modified to harvest the variety of seeds. For green tangled type of crops, harvesters with an open front are preferred. Whereas for heavier seeds harvesters with narrow front are preferred, to reduce overloading of threshing and cleaning units by the bulk of material harvested. The vacuum harvesters using a brush at the picking head were found to be good for light fluffy to medium sized grass seeds but not for the heavy legumes.

So far no machine has as yet, been designed and found to be effective in harvesting the large variety of pasture seeds.

2.2.1.3 Post harvest treatments

Existing harvesters tend to collect a lot of trash along with the mature seeds. Most of them have no provision for drying during harvest. Hence the seeds require a lot of post harvest processing such as:-

1. Recovering of quality seeds from the trash.
2. Drying of harvested seeds soon after they are harvested, as the presence of immature seeds cause rapid heating and deterioration of the batch of harvested seeds. Generally the seed is harvested and is accumulated in the bin, then transferred to a truck, which is again loaded with seeds from several

bins, before transported for drying. This may take 6 to 12 hours between harvest and commencement of the drying process. During this period the seed may suffocate, that is get damaged by overheating or by accumulation of waste gases (CO_2) by the continuing metabolic activity of the seeds. Generally a harvested seed contains 60% of moisture content which has to be reduced to 10% to alleviate the problems of suffocation specified above. Therefore it is most desirable to commence the drying process as soon as possible prior to the harvesting [4][5].

2.2.1.4 Quality tests

Quality tests for the seeds such as the percentage purity, germination tests [6] as prescribed by the DPI should be complied with as this facilitates a better understanding of the efficiency of the machinery in terms of the quality of the harvested seeds. This also helps in establishing the overall performance of the harvester and in identifying the defects. This input is important, as designing is an ongoing process.

2.2.2 Agricultural

Agricultural problems can be classified as-

1 Seed quality control / testing.

2 Storage.

2.2.2.1 Seed quality control and testing

Farmers should be convinced of the importance of quality tests and encouraged to get them done by offering these services at accessible distances.

Farmers should be educated about the benefits of the tests, which are-

-Stabilising the relation between the produce and seed merchants, the seed merchants-the graziers.

-In the setting of minimum standards.

-Protecting Consumers.

-Controlling of diseases and noxious weeds, which in turn helps in protecting the farmlands.

2.2.2.2 Storage

Some seeds have better germination when stored for longer periods of up to more than one year. Hence storage of seeds is very important. The main factors affecting seeds are the percentage of moisture content in the seed, relative humidity and temperature. Hence the seed moisture should be reduced to 10% before storing. Within a seed storage facility, the appropriate relative humidity and temperature should be maintained [7].

Government or other agency should provide, such an infrastructure for the storage of seeds in different areas at a minimum cost and co-ordinate the collection, storage and sale of the harvested seeds, for the farmers in that area.

2.2.3 Marketing

There are two major problem areas identified in seed marketing.

1 Sale Outlets.

2 Marketing Strategy / Policy.

2.2.3.1 Sale outlets

In the pasture industry there are no organised societies or co-operatives which could provide stable outlets for farmers to sell their seeds at a fair price. The merchants buy the seeds from the farmers, and sell it to other farmers and graziers at higher prices, making a profit. But the producers are given no percentage of this profit. This happens as there are no fixed prices for the seeds and each farmer negotiates independently with the merchants. So on the whole it is the farming community who lose on both ends. Realising this many of the producers store the seeds and try to sell directly to other farmers, which is not easy because of the difficulty in co-ordinating these sales and which in any case is only marginally profitable.

2.2.3.2 Marketing strategy

Government has not evolved any strategy or policies for marketing of pasture seeds. There are no fixed prices or sale outlets. Hence the farmers are disadvantaged and do not receive an equitable share of the profits.

2.2.4 Ancillary services

Some of the ancillary services that could be provided are-

1 Repairs and maintenance

2 Hire / Purchase facility of harvest and post-harvest equipment

3 Bank Loans

2.2.4.1 Repairs and maintenance

During the harvesting season the machinery tend to breakdown due to continuous operation. So there should be workshops at accessible distances, offering repairs, maintenance services and spare parts. Breakdowns can cause a heavy loss of mature seeds and even of the total yield.

2.2.4.2 Hire / purchase facility

For farmers with small holdings, purchasing costly machine that remains unused for much of the year is uneconomical and may involve a loss until its cost is recovered. Hence Government or other agencies should be encouraged to offer the required machinery on hire, thus assisting the farmers to make use of their money in other ways without tying the funds in buying large machinery which may not be utilised to its full capacity. Sugar industry has such infrastructures where the farmers can hire sugar cane harvesting contractors.

2.2.4.3 Loan systems

Banks should have loan schemes with lower interest rates for purchasing these machines.

At present the reduction in export of beef has also affected the prices of pasture seeds. In view of all the above mentioned problems, especially the engineering problems, CQU in collaboration with the CAT and the DPI have designed a grass seed harvester suitable for all types of pasture seeds and which most probably can be extended to harvest other grain crops.

To identify the prospective market for such a harvester a limited market survey was conducted. The survey assisted in assessing, the magnitude of the pasture seed industry in Central Queensland and the degree of mechanisation within the industry. This information would on review, assist in defining the size and other design criteria of the harvester to be fabricated. The survey would also provide a basis for evaluating the performance of the designed machine with the machinery currently used by the farmers.

2.3 MARKET IDENTIFICATION

The main markets which provided an impetus for growth of pasture grasses were:-
- The dairying and beef cattle in the higher rainfall areas of eastern Australia from northern N.S.W to about Cooktown in northern Queensland growing both grass and

legumes.

- The predominantly beef cattle producing drier inland areas of southern and central Queensland with more areas of grasses like Green Panic, Rhodes and Buffel and limited areas growing legumes.

- The Townsville Stylo grown across northern Australia in the extensive cattle rearing area of Kimberley of W.A, the top end of N.T and inland north eastern and central Queensland.

From a merchants point of view the nucleus of the seed industry has always been Rockhampton. Rockhampton has the advantage of being situated centrally, with reasonable freight and communication links north and south, its proximity to grass seed harvesting districts and a climate suitable for storing seeds. The present survey has therefore been restricted to the Central Queensland area to get an impression of the market.

2.4 ANALYSIS OF THE MARKET SURVEY

For the survey, 696 questionnaires were sent out; 448 to farmers living along the Dawson river and 248 to farmers living along the Mackenzie river. Questionnaire prepared for the farmers is shown in Appendix 1. There were 102 replies from farmers. Of which only 30 forms contained all of the necessary details.

On analysing these 30 Questionnaires the following statistics were compiled. Each question has been dealt below in the same order and sequence, using the same numbering system as in the Questionnaire prepared for the farmers as shown in Appendix 1.

1.0	*Area under cultivation between*	*0-2000 hectare*
	On an average, Area / farmer	*230 hectare*
2.0	*Total areas harvested in the past year, between*	*30-300 hectare*
	On an average area harvested / farmer	*106 hectare*
3.1	*Total yield in past year between*	*620-20000 kg*
	On an average, Total yield / farmer	*4415 kg.*
3.2	*Total yield per hectare in past year between*	*8-133 kg / hectare*
	On an average, Total yield / hectare by a farm	*49 kg / hectare*

It can be concluded that most of the farmers have medium holdings (100-500 hectare), with cattle for grazing and also growing other crops along with the pasture crops. Some farmers do not grow pasture seeds on their own farm, or grow only partially and contract harvest for other farmers, for which they get 50% of the total harvested seeds.

On comparing data 1.0 and 2.0 above it is clear that, only 46% of the total area under cultivation has been harvested by each farmer.

4.0	*The type of machine used for harvesting.*	
4.1	*All crop header*	*14*
4.2	*Tractor beater Harvester*	*2*
4.3	*Tractor brush Harvester*	*2*

The most commonly used harvesters are the All crop Header. A few farmers are using machinery modified by themselves. The most popular of such harvesters are Cotton pickers, Wheat Strippers Header Beaters and Corn Thresher / Harvesters.

5.0 The post harvest treatments

Drying:

i. On hessian.

ii. On floor.

iii. Forced air draft (heated/unheated)

All three methods seem to be equally popular.

Duration of drying

Duration of drying depends on the drying technique applied. Most of the farmers seem to dry for 2 days. Some dry for 72 hours and those who dry the seeds on the floor under the sun take up to two to three months.

Cleaning

i.	*On farm*	*13*
ii.	*By Merchants*	*13*

Estimated seed cleaning cost

Seed cleaning costs were found to range between 0.25 cents to $ 2.00 / kg.

On an average cost / kg $ 1.00.

Seed cleaning machine used

The different seed cleaning machinery used by farmers in Central Queensland are:-

Buffel Cleaner Baggers

Shaker type for Green Panic

Header TR 85 New Holland

Eckarts Pre Cleaner

Pre Cleaners Boddington Indent Cylinder

Home made Cleaner Bagger

Agitator and Sieves

Andrew Adam's Shaker Cleaner

Blowers

Seed Cleaner Grades.

Percentage of commercial seed from harvested seed

The claimed percentage of quality seeds obtained from the total harvested seeds using the available harvesters were seen to range between 60-100%, i.e on an average 78%.

Storage of seeds

Farmers after harvesting have to store the seeds before they are able to market it. Some have storage facilities in their own farm and some store it at the facilities provided by the merchants on rental basis.

i.	*On farm*	*18*
ii.	*By Merchants*	*11*

Farmers seem to store both on farm and at merchants.

Average duration of storage

Duration of storage ranges between 90 to 400 days. Most seem to store for about one year.

Germination test for commercial seeds

These tests can be done at harvest and at sale. Most of the farmers do not get the seeds tested both times.

i.	*At harvest*	*14*
ii.	*At sale*	*11*
iii.	*Test done both at harvest and at sale*	*5*

The results of the test are indicated in the following table.

S.No.	*Tests conducted on seeds*	*At harvest*		*At sale*	
		Range	*Average*	*Range*	*Average*
1	*% Purity*	*12-99*	*89*	*12-99*	*86*
2	*% Germination at 21 days*	*5-70*	*39*	*25-85*	*53*
3	*No.Tetrazolium (TZ)*	*4-92*	*48*	*10-70*	*42*

6.0 Preferred Harvester

The farmers were asked to tick the harvester they would prefer most among the three mentioned types-

i.	*Harvester for fluffy grass type like Buffel*	*8*
ii.	*Harvester for all pasture grass seeds*	*13*
iii.	*Harvest all pasture grass seeds, Wheats, Oats and Sorghum*	*8*

Preference for all three types seem to be about the same.

7.0 The preferred front width and the price for a power air flow self contained auto harvester.

The above question was asked, to estimate the size of the harvester to be designed and to estimate the price the farmers were willing to pay.

Front width

i.	*2 m*	-
ii.	*4 m*	*3*
iii.	*6 m*	*7*
iv.	*8 m*	*4*
v.	*10 m*	*1*

Apart from the above mentioned sizes about 8 farmers have requested for 5 m front width.

RETAIL PRICE

i.	*$ 50,000*	*15*

ii.	*$ 75,000*	*1*
iii.	*$ 100,00*	-
iv.	*$ 150,000*	-

Five farmers have requested the retail price to be reduced between $ 30,000 to $ 40,000. Conclusion would be preference for a self contained auto harvester with minimum acceptable front width of 4 m, preferably 5 m with retail price of $ 50,000 or less, which has been shown to be impracticable by the design work outlined later in this thesis.

8.0 The preferred front width and price of a new Tractor mounted power air flow harvester for grass seeds.

The above question was asked to estimate the size of the harvester to be designed and to estimate the price the farmers were willing to pay.

Front width

i.	*2 m*	-
ii.	*4 m*	*9*
iii.	*6 m*	*9*
iv.	*8 m*	*3*
v.	*10 m*	*1*

Five farmers having requested for a 5 m front width.

8.2 RETAIL PRICE

i.	$ 10,000	15
ii.	$ 20,000	6
iii.	$ 40,000	-
iv.	$ 60,000	-

Two farmers have requested for less than $ 10,000. Looking at the above information it can be concluded that the preference is for a Tractor mounted power air flow harvester with minimum front width of 4 m to 6 m and with a retail price of $ 10,000. Four farmers have requested for 8 m front for both self contained and Tractor mounted harvester. But they are ready to pay only $ 50,000 for the self contained and $ 20,000 for the Tractor mounted air flow harvester.

2.5 CONCLUSION OF MARKET SURVEY

From the survey report it is quite obvious that there are many features in the existing machinery which need some improvement. Some of the features that need consideration are:-

- The harvesting units which tend to take in a lot of trash along with the mature seeds. In some harvesters, of the total seeds dislodged only 60% get collected while the rest fly in the air and are lost.
- After harvest, seeds have to be cleaned on separate units sometimes risking great loss.
- In most of the harvesters drying facilities are not available, to commence the drying process soon after harvest to prevent seed suffocation. Normally it takes nearly 6 to 12 hours prior to harvesting for the drying process to start causing great loss to the

seed quality.

- The driver's seat position is not in a position where the driver has a clear view of where he/she is going, which can risk major damage to the vehicle and loss of seeds.
- As mentioned in section 1.1 the seeds ripen over a period of 10 to 14 days, hence using a vacuum type harvester the seeds are harvested by covering the same area two to three times over a period of 14 days. But most of the machines presently used have low clearance causing the crop to bend which may cause loss of seeds, because on the second run the harvester may not be able to pick the seeds from these dislodged crops.

On comparison of the harvester designed by the CQU with the existing machines, it is quite obvious that most of the above mentioned problems have been considered and the deficiencies in the existing harvesters have been improved upon..

2.6 ADVANTAGES AND DISADVANTAGES OF THE CQU HARVESTER

The various advantages of the harvesting machine designed by the University of Central Queensland based on the market survey findings with respect to the existing machinery are:

1. Speed of harvesting and a wider harvesting front unit ensures more coverage of area in any one run.
2. Only ripe and mature seeds are dislodged from the heads, hence germination is high.

3. Weather permitting, two or more harvests of the same crop can be accomplished as seed ripening progresses, thus increasing the total yield.

4. Driver's seat is positioned over one of the wheels, to permit better viewing for the driver.

5. The harvesting front has been designed to pick all of the dislodged seeds with no loss.

6. The harvested seeds are clean without trash thus eliminating the tedious process of seed cleaning.

7. These seeds are collected in a bin with false floor through which air heated by the engine is circulated to start the drying process of the seeds from the time of harvest thus reducing the losses from seed suffocation. This feature also helps in reduced compaction and sticking of the harvested seeds.

8. Optimum ground speed allows better harvest of all mature seeds.

9. This harvester is seen to work efficiently for different sized seeds by varying the air speed and speed of the header unit.

10. Simplicity- The working/moving parts are few and easy to operate and maintain.

11. Efficiency is high as loss of seed is minimised and as only ripened seeds are picked up without any trash.

The only disadvantage of the designed harvester is the cost of a self contained auto harvester being $ 75,000 and more, which may be very high for small scale and medium holding farmers.

2.7 REVIEW OF THE PASTURE SEED INDUSTRY

Some general observations made with regard to the grass seed industry in Central Queensland as a result of the survey conducted are listed below.

1. Pasture seed industry is a very disorganised industry.
2. The grass seeds in Central Queensland are mostly produced by farmers with small to medium holdings with cattle and growing other crops along with the grass seeds.
3. There has been no mechanisation in the pasture industry in any of the areas such as sowing and harvesting. Most of these processes are being done by machinery developed for other grain crops. Such machines are not quite as efficient, as they have not been designed by taking into consideration the unique qualities of the pasture seeds. Hence it can be said with confidence that the harvester designed by the CQU has a bright future in this market.
4. Since most of the farmers in Central Queensland have medium holdings, so the size and cost of harvester should be suitable to their land area and finances.

2.8 HARVESTER SIZES BASED ON THE SURVEY REPORT

Suggestions of the design criteria for the harvester to be manufactured based on the survey report are as given below.

As mentioned earlier most Central Queensland farmers have small to medium holdings, hence the market would be for,

Tractor mounted power air flow harvester

1. 45 kW tractor
2. Front width 5 m
3. Retail price $ 60,000

But since nearly an equal number of the farmers have requested for a harvester that can harvest all pasture seeds and a harvester that could harvest both pasture seeds and grain crops; with clearance of 1.5 m being stressed upon in some cases, the suggestion would be for,

Power air flow self contained auto harvester

1. Power 90 kW
2. Clearance 1.2 m
3. Front width 5 m
4. Retail price $ 85,000

This harvester has a bright future, if as envisaged it is capable of harvesting all varieties of pasture seeds. The market would be greater with more consumers ready to buy, even if it is bigger and dearer.

3.0 FAN THEORY AND DESIGN

3.1 INTRODUCTION

Fans are generally used to move air continuously against moderate pressures and they are used both as a blower or for creating a vacuum, by suction. This project is mainly concerned with the design of a fan to be used to draw air through a harvester system. Large portions of the design and operation of fans involve application of the theories of fluid mechanics. Hence some fundamentals of fluid mechanics theories applied have been included in the supplement to the thesis, to assist farmers or any non technical people to follow the theory of vortices and fan designs detailed in this thesis. These theories can also be found in [8] and [17]. Some of the books, journals and papers referred to are [18] to [21]

3.2 VORTEX FLOW

When a mass of fluid is rotated it is said to have a vortex flow. This type of flow is generally encountered in fan, cyclone and radial duct separators, which are the main topics of this thesis. Vortex flows can be classified into three main categories.

3.2.1 Free vortex

Free vortex is when the total energy remains constant as expressed by Bernoulli's equation,

$$p + \frac{1}{2}\rho u^2 = \text{constant} \tag{3.1}$$

which can further be derived to get the equation, by differentiating this with respect to a radial direction and then on further integration to get,

$$u\ r = \text{constant} \tag{3.2}$$

where p is the static pressure, ρ is the density and u is the tangential velocity which varies inversely as radius r, ie, with increase in r, u will reduce.

3.2.2 Forced vortex

Forced vortex is when the fluid rotates at constant angular velocity ie, angular velocity ω = constant. Therefore $v = \omega \times r$ or $v \propto r$, ie, as radius increases, the tangential velocity also increases. By considering a fluid element in vortex, the equation given below can be derived

$$\frac{dp}{dr} = \frac{\rho u^2}{r} = \rho \omega^2 r \tag{3.3}$$

which on integration between limits of radius gives,

$$p_2 - p_1 = \rho\frac{\omega^2}{2}(r_2^2 - r_1^2) = \frac{1}{2}\rho u_2^2 - \frac{1}{2}\rho u_1^2 \tag{3.4}$$

In other words that the equation 3.4 conveys that the difference in static pressure is equal to the difference in tangential velocity pressure across the vortex in this type of flow.

3.2.3 Compound vortex

In a compound vortex the vortex is of the form u^2r = constant. By experiment this vortex is seen to apply to centrifugal dust separators very closely and which was observed by the experiments carried out in this thesis. By considering a fluid element in vortex, the equation derived is,

$$p_2 - p_1 = \rho u_1^2 - \rho u_2^2 \qquad (3.5)$$

This equation 3.5 then helps us to find the pressure loss Δp in the vortex of the centrifugal separator, by using Bernoulli's equation,

$$p_1 + \frac{1}{2}\rho u_1^2 = p_2 + \frac{1}{2}\rho u_2^2 + \Delta p \qquad (3.6a)$$

$$\Delta p = \frac{1}{2}\rho u_2^2 - \frac{1}{2}\rho u_1^2 \qquad (3.6b)$$

where u_1 the tangential velocity at the outer radius of separator body (is often very nearly the same as the linear velocity in the inlet duct) and u_2 the tangential velocity at a radius of about half that of the exit duct, within which this law of vortex is not applicable.

3.3 DEFINITION OF TERMS USED

The different pressure terms used with respect to air flow through a fan system have been discussed below.

Velocity or dynamic pressure

Velocity or dynamic pressure usually denoted by 'p_v' is a measure of the kinetic energy available in an airstream and is always taken to be positive.

Static pressure

Static pressure is a measure of the potential energy available in the airstream and is seen to act equally in all directions irrespective of the velocity and is denoted by 'p_s'.

Total pressure

As the term implies it is the algebraic sum total of static and velocity pressure at any point and is the measure of the total energy available in an air stream.

Fan total pressure

Denoted by 'p_t' is the algebraic difference between the average total pressure at fan outlet and the average total pressure at fan inlet.

Fan static pressure

For rating of a fan this term is used and has been defined as the fan total pressure minus the velocity pressure corresponding to the average air velocity at the fan outlet and is not the difference between the static pressure at the outlet and static pressure at the inlet.

Fan efficiency

This is the ratio of output power to mechanical input power and is expressed in percentage.

3.4 FAN LAWS

To get the performance ratings of different sized fans working with different speeds and with fluids of different density, a lot of tests are required which can be a very difficult, long, time consuming and an arduous job. Hence some fan laws were developed to assist, in being able to predict the performance of geometrically similar fans of different sizes and speeds quite accurately for particular practical purposes. However it is important to remember that these laws are applicable only to a given point of operation on the fan characteristics and may not be used for predicting other points in the characteristic curve. These laws are generally used for calculating the variations in flow rate, pressure and fan power when the size, rotational speed or gas density are changed. In the equations suffix 1 and 2 have been used to denote the known values and the resulting calculated values respectively.

$$Q_2 = Q_1 \times \left(\frac{N_2}{N_1}\right) \times \left(\frac{D_2}{D_1}\right)^3 \tag{3.7a}$$

$$p_2 = p_1 \times \left(\frac{N_2}{N_1}\right)^2 \times \left(\frac{D_2}{D_1}\right)^2 \times \frac{\rho_2}{\rho_1} \tag{3.8a}$$

$$W_2 = W_1 \times \left(\frac{N_2}{N_1}\right)^3 \times \left(\frac{D_2}{D_1}\right)^5 \times \frac{\rho_2}{\rho_1} \tag{3.9a}$$

where Q is the flow rate, p is the pressure (total, static or velocity), ρ is the gas density, N is the fan rotational speed, D is the impeller diameter and W is the impeller power. This thesis deals with fans operating at pressures below 2.5 kPa, hence fan laws are applicable while considering the inlet volume flow. If one or two variables remain constant, as is seen to be true for the system considered (we found the change in gas density to be negligible and the diameter of the impeller was kept constant), the above three equations can further be simplified to [20],

$$Q_2 = Q_1 \times \frac{N_2}{N_1} \tag{3.7b}$$

$$p_2 = p_1 \times \left(\frac{N_2}{N_1}\right)^2 \tag{3.8b}$$

$$W_2 = W_1 \times \left(\frac{N_2}{N_1}\right)^3 \tag{3.9b}$$

3.5 POINTS TO BE CONSIDERED FOR FAN DESIGN

Some of the important points to be considered while designing a fan are discussed below.

3.5.1 System resistance

When air is moved through any duct system, energy or pressure supplied to the air by the fan is lost due to friction between the duct walls and the fluid and due to separation and eddies formed at fittings. All of these losses are together known as system resistance and for practical purposes are taken to be proportional to the square of the velocity at the point of loss. Although there are some exceptions such as pressure losses in filters. But for the purpose of this thesis, it can be considered that for moving a given volume of fluid through the system the pressure required will be directly proportional to the square of the volumetric flow i.e,

$$p \propto Q^2$$

Therefore to double the air flow, a fan will have to develop four times the pressure. The graph for pressure loss versus volumetric flow for a given system is called the system resistance curve or the system pressure loss curve.

3.5.2 Operating point

This is the point of intersection of the fan characteristic and the system resistance curve and the flow is said to stabilise at this point. The Figure 3.1 shows such a point 'A' called the operating (or duty) point. The dotted curves show the changed system resistance curves caused by either a change in the system or variation in practice from the calculated system resistance, resulting in different sets of operating points shown at 'B' and 'C'. If this gap in operating point is very far from the design values either a reduced or excessive utilisation of fan power can result.

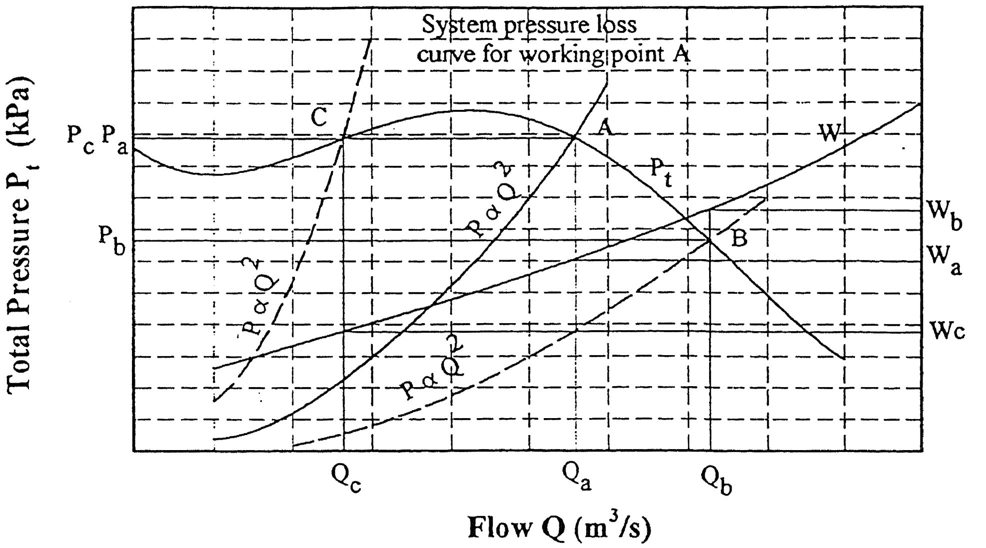


Figure 3.1 Effect of change in system resistance

3.5.3 Fan speed

From the fan equations it is obvious that a change in fan speed for a fixed system can result in an equal percentage change in the volume of air drawn through, pressure varies as square of the speed change and power varies as cube of speed change.

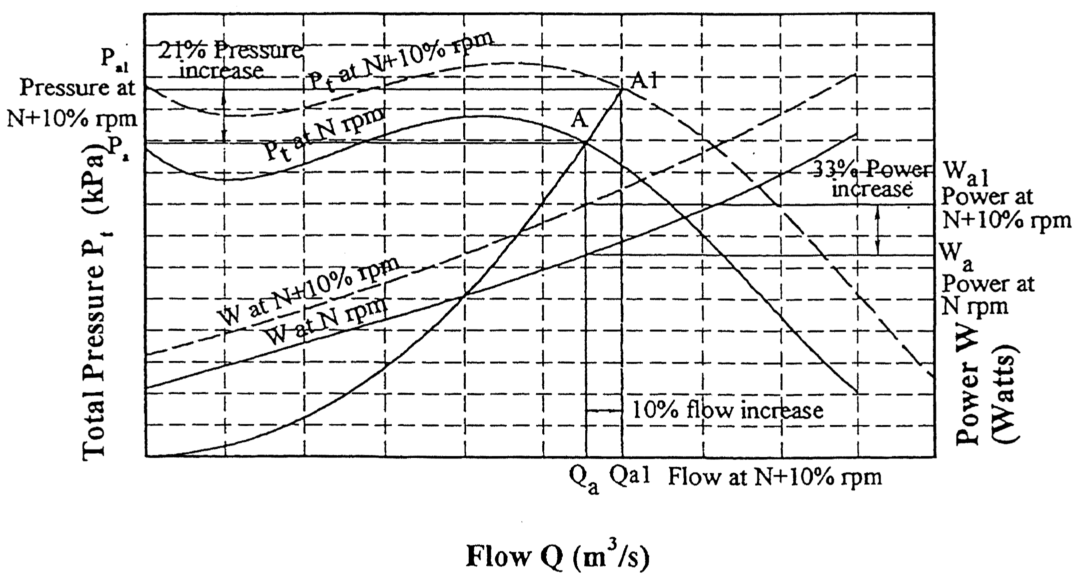


Figure 3.2 Variation of performance with change of fan speed (10%)

Hence if proper equipment is available and if flow can be increased by say 10% then speed increases by 10%, pressure by 21% and power by 33% respectively. Figure 3.2 shows this change in full and dotted lines.

3.5.4 Fan size

While designing a fan if flow volume cannot be increased to the required value by just increasing the speed then the only alternative is to design larger sized fan. But then the intersecting point of the fan characteristic and the system resistance curve change, thus changing the operating point. This then makes it difficult for us to predict the performance of this larger fan until a new operating point and the power requirement has been determined. Figure 3.3 illustrates this effect for a 10% increase in fan size at constant speed.

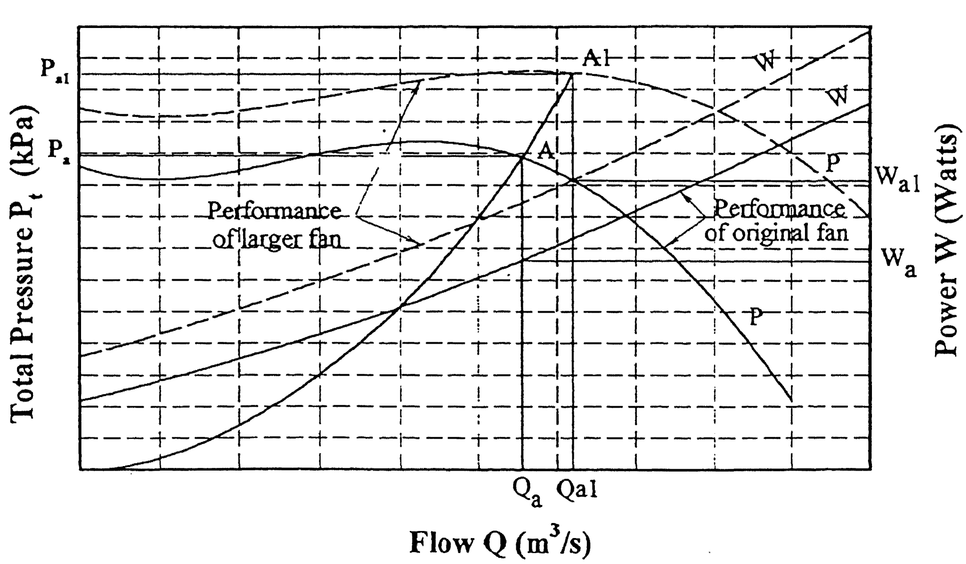


Figure 3.3 Effect of increase in fan size (speed constant)

3.5.5 Fan control

This is a very important criterion to be considered while designing the harvester system. Since this harvester is designed to harvest a variety of different types and sizes of seeds, varying from light and fluffy seeds to heavy spherical shaped legumes, it becomes necessary to be able to control the flow volume and the velocity of the moving air. This control system may be an occasional control type for e.g, just needed in summer or winter or whenever the harvest season is or a continuously variable adjustment type. There are many methods available to achieve this variable flow volume or flow velocity, but we are mainly interested in control by variation of fan performance. As mentioned earlier this variation in fan performance can either be continuously varied type or which need to be adjusted occasionally. Variation of fan performance can be achieved in many ways but we are mainly interested only in the method of speed control.

Speed control methods

Varying the running speed of the fan has been found to be the most efficient way of controlling the fan performance. It has been observed that if the same duct system is considered, the point of operation is seen to move down along the duct system curve as the speed is varied. This helps in maintaining the efficiency of the fan and results in a corresponding maximum drop in power consumption and noise level as the speed is reduced and has been shown in Figure 3.4.

To achieve this provision has to be made to control the speed of the fan. Some methods recommended for the harvester are-

i. Variable speed electric motors.

ii. Variable speed gear box.

iii. Variable ratio pulleys and belt drive.

Some of the fan speed controls presently being used are-

i. DC motor with simple voltage control.

ii. Hydraulic drives for both revolving brush header unit and the fan.

There are various other methods also available. Considering the above options the variable ratio pulleys and belt drive and the types being presently used, would be more economical to maintain a minimum harvester cost.

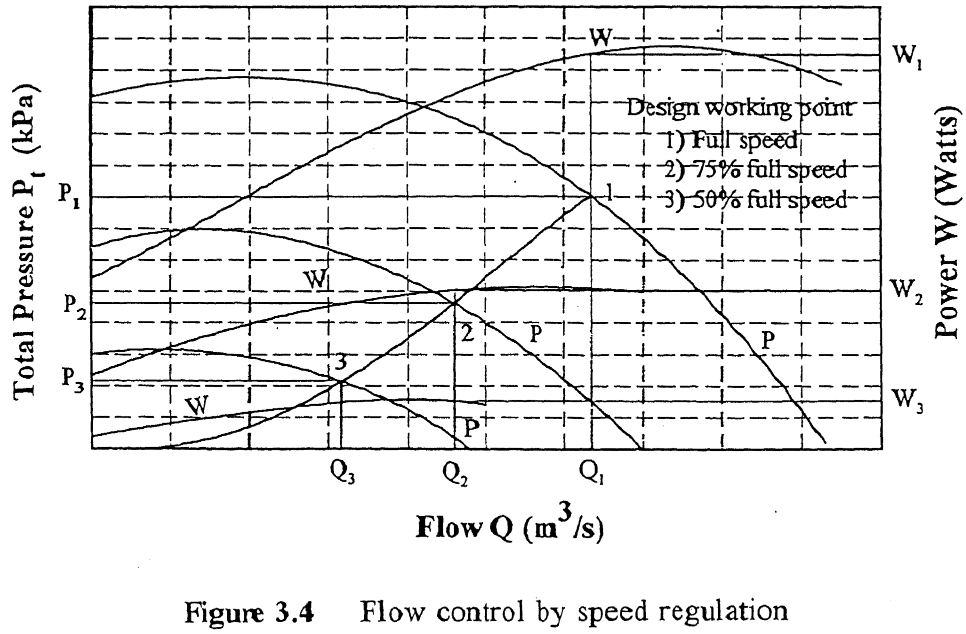


Figure 3.4 Flow control by speed regulation

A point to be kept in mind while trying to determine power saving by these methods, is by giving due account to the changes in the efficiency of the prime mover and the drive which may change with speed or load due to electrical, mechanical or magnetic slip involved. The overall effect is generally seen to be a reduction in the power input to the prime mover and which is calculated as the square of the speed. The efficiency of the speed conversion is taken as the ratio of the output and input speeds. Selection of speed control equipment was based on costs of power involved over the life of the machine and against the initial and maintenance costs. Keeping in mind the amount of control required and the frequency of operation, the main factors influencing the choice are:-

i. Power costs.

ii. Amount of control required whether stepped or continuous.

iii. Initial buying and installation costs.

iv. Maintenance and replacement costs.

v. Accuracy and repeatability of the control equipment.

vi. Whether automatic systems are needed.

vii. The flow ranges over which this control is needed.

viii. Noise levels.

3.5.6 Fans in series and parallel

For some systems two or more identical sized fans may have to be used in parallel or in series, depending on the requirements. In certain cases the total pressure to be developed may be quite high, so instead of using a large fan two or more fans may be connected in series to develop the required pressure. Whereas for certain other cases the total flow volume required may be high, hence instead of using a large fan two or more fans may be connected in parallel to achieve the same required volumetric flow rate at the same total pressure. Connection and operation of fans in series and parallel are discussed below.

Fans in series

When two or more fans are connected in series the flow volume through each fan unit remains same, but the overall total pressure will be the sum of the individual fan total pressures less the losses in interconnections.

Fan in parallel

When two or more fans are connected in parallel the total flow volume is the sum of the individual volumes flowing through each fan at the same effective fan total pressure. In this case however, the effective fan total pressure remains the same across each fan between the common connection points downstream and upstream less the loss of total pressure due to individual fan connections. Fan characteristic for such an assembly may be drawn by summing up the flow volume of each fan at same effective fan total pressure. For achieving this a reverse flow characteristic of the second fan operating in normal direction is required.

For the harvester system the total pressure required is 2.4 to 3 kPa but the volumetric flow rate required is high about 30 m^3/s. Hence in the design considered in this thesis, it has been suggested to use two or more fans, connected in parallel to achieve the required flow rate at constant total pressure.

3.5.7 Fan selection with respect to noise

Noise is an important criteria for selection of fan especially for a harvester, because the operator working with the harvester has to spend practically most of his working hours on the harvester. Therefore if noise level is very high it can be hazardous to the operator's health, which is unacceptable. This is another main reason for selecting centrifugal fans over axial and other fans for the harvester. Because centrifugal fans with few blades produce low frequency sounds whereas axial produces high frequency sounds and most people seem to prefer low frequency sounds as they are less damaging. If a fan has B number of blades and it runs at N rpm, then there are NB/60 traverses of a blade across the outlet aperture per second, each producing a fluctuation in the air pressure. For example, a 10 blade fan operating at 3000 rpm would produce about 500 impulses/s, a frequency to which human ear is very sensitive. This is why fans especially high speed ones tend to be noisy. The human ear is relatively insensitive to frequencies below about 150 Hz, so if NB/60 lies below this frequency, the fan will appear to be much quieter. Hence if a fan has 6 blades and is operated at 1500 rpm, it would have a frequency of 150 Hz, which is the maximum limit. So while designing or selecting a fan compromise will have to be made on the number of blades or fan speed selected.

Figure 3.5 makes a comparison between the noise spectra of two fans. But the Figure 3.6 shows that fan generated noises vary with fan duty and is most quiet while operating at or near peak efficiency and is noisiest close to stalled condition.

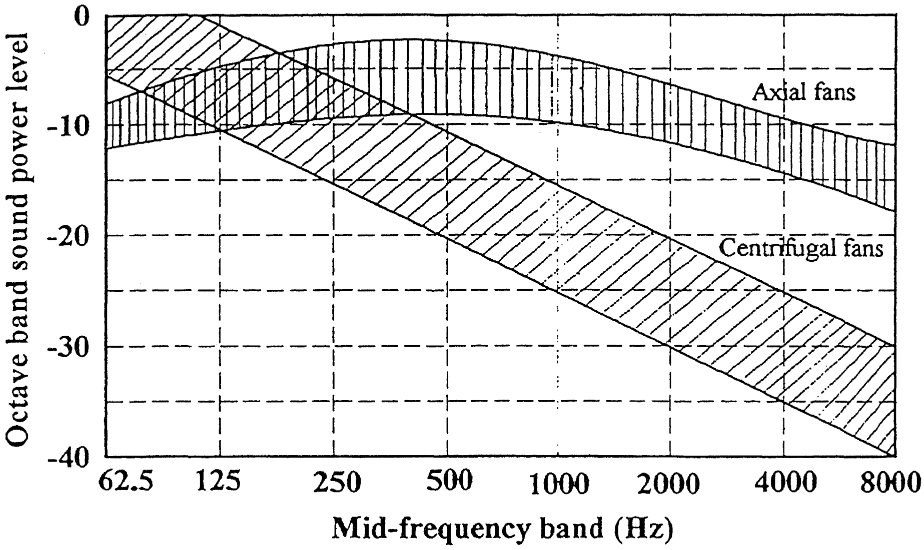


Figure 3.5 Comparison of Centrifugal and Axial fan noise spectra

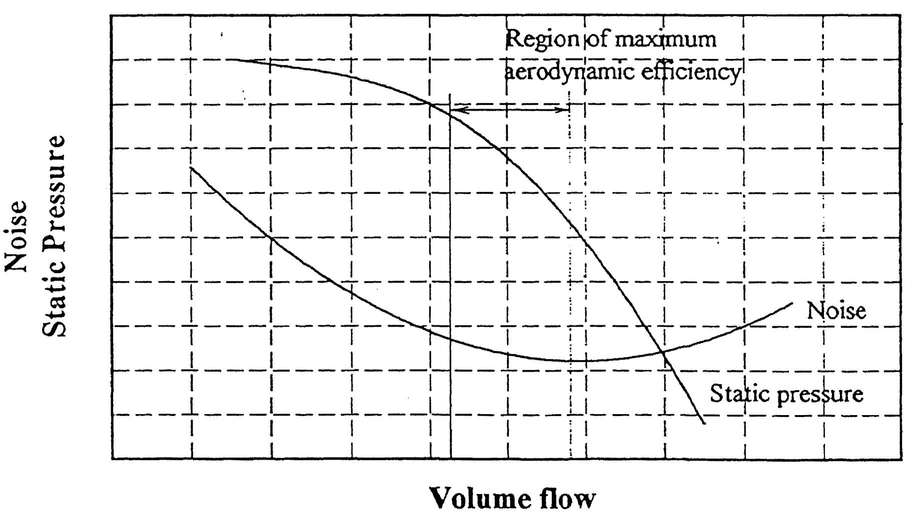


Figure 3.6 Fan generated noise versus duty

Another cause of excessive fan noise may be due to highly turbulent inlet flow created due to placement of bends or other such fittings too close to the inlet. But this can be easily overcome by attaching longer straight ducts near the inlet providing enough length for the flow to develop and for inlet flow to be uniform.

3.6 FAN TYPES AND THEIR CHARACTERISTICS

Over the years fans have been used for a variety of applications. Hence there are a wide range of different types of fans available in the market. They can be categorised under 4 main types.

1 Centrifugal fans.

2 Axial fans.

3 Propeller fans.

4 Mixed flow.

3.6.1 Centrifugal fans

In such fans the air enters the fan axially and is discharged radially into the volute casing. Such fans generally cover medium to low volumetric flow rates at medium to high pressures. Among the centrifugal fans the different types are characterised by the different angles of the impeller. Three main forms of blade angles with respect to direction of rotation at the impeller periphery are most commonly in use. The blade angle is the main criterion in determining the amount of work done on the air or the amount of pressure developed by the fan.

The total pressure of centrifugal fans are composed of two main components-

i. The pressure increase across the impeller, which is usually greatest for fans with backward blade curvature.

ii. The pressure increase due to diffusion in the volute casing, where velocity head is converted to pressure rise. This is greatest for fans with forward blade curvature which have only a small pressure rise across the impeller.

This helps to explain the high efficiency of the backward curved type, since the frictional and other velocity related losses must clearly be greater for (ii) above.

The different types of centrifugal fans are-

Backward inclined blade

These are centrifugal fans with blade tips inclined away from the direction of rotation such that the blade angle β_2 is said to be less then 90°. These blades may be aerofoil type or single thickness blades with the latter being either straight or curved in shape. Such impellers generally have only few blades. The characteristic curve generally has a pressure curve which is quite steep over the working range and has a good power curve with a non overloading profile. Hence very high efficiencies can be achieved.

Radial blade

These are fans with blade tips or the whole blade radial to direction of rotation such that the blade angle β_2= 90°. The characteristic curve shows such blades to develop a power curve which is almost a straight line rising from a minimum at zero flow to a maximum at maximum flow, but this curve is not as steep as that observed for forward inclined blades. Consequently for such fans it will be difficult to achieve high efficiencies. But advantage is that such fans are observed to be self cleaning hence could handle moderately dirty conditions.

Forward inclined or curved type

These are fans with blade tips inclined towards the direction of rotation such that the blade angle β_2 > than 90^0. Generally these type of fans are observed to have large number of curved blades. The characteristic curve shows a very steep power curve, rising from minimum at zero flow to maximum at maximum flow. This means that these types of fans are liable to overload the driving motor if operated beyond its rated duty point. Hence the efficiency that can be achieved is greatly reduced. The advantage is that such fans can handle large volumes of air at low operating speeds. Figures 3.7a and 3.7b makes a comparison of the characteristic curves of the three fans.

3.6.2 Axial fans

In such fans the clearance between the tips of its blades and the cylindrical casing is kept to minimum as practically possible. The air in such fans enters and leaves axially in a straight through configuration. It is able to handle medium to high flow rates at medium to low pressures. In such fans the basic operation is by drawing the air in axially and discharging it with a rotational component due to the work done by the impeller torque. Therefore the absolute velocity of the discharged air is found to be greater than the axial velocity, resulting in a loss of the fan total pressure developed by the impeller.

To minimise this loss advanced designs started providing-

Guide vanes- These were fixed downstream of impeller to convert some of these excess velocity pressure to more useful static pressure.

Pre-rotational vanes- These were provided upstream of the impeller and were rotated

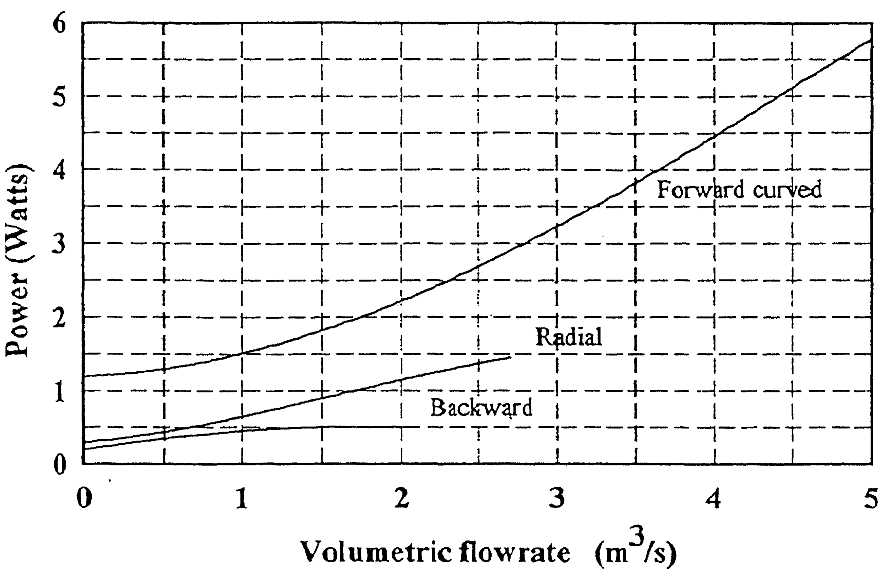


Figure 3.7a Centrifugal fan characteristics

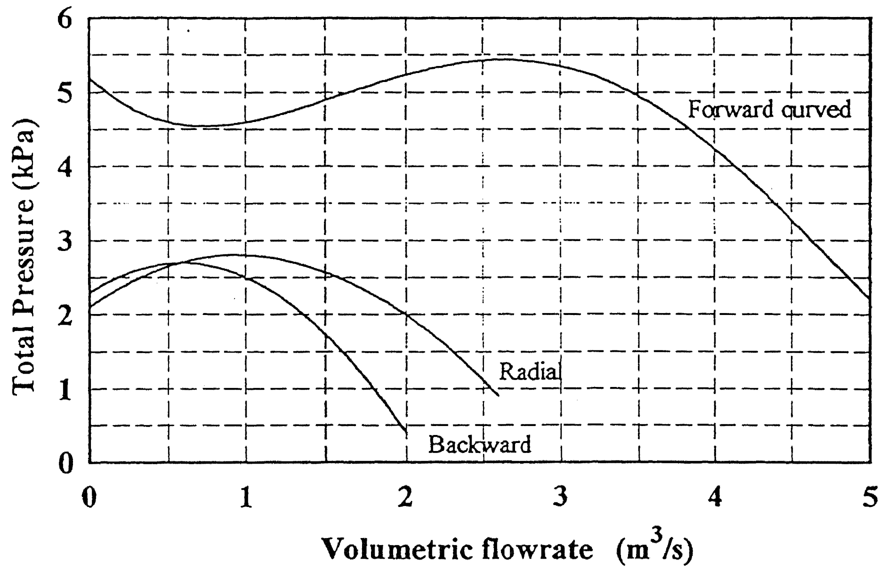


Figure 3.7b Centrifugal fan characteristics

in a direction opposite to the impeller. This if designed properly is seen to assist the discharged air to leave the fan in an axial direction thus achieving maximum useful pressure.

Contra-rotating fans- Such fans make use of a second impeller downstream of the first but rotating in an opposite direction. Thus this second impeller acts like an upstream guide vane unit.

The characteristic curves of such fans indicate that although they are not able to develop pressures as high as centrifugal fans of same impeller diameter and speed, they have non-overloading power curves and so can achieve good efficiencies. Comparison of characteristic curves of the above mentioned 3 types of axial fans are given in Figure 3.8a and 3.8b [8].

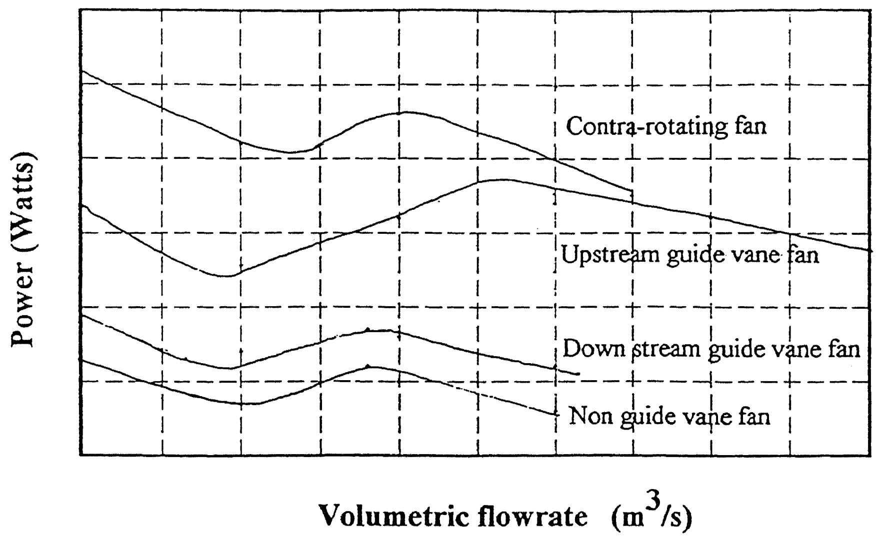


Figure 3.8a Axial fan characteristics

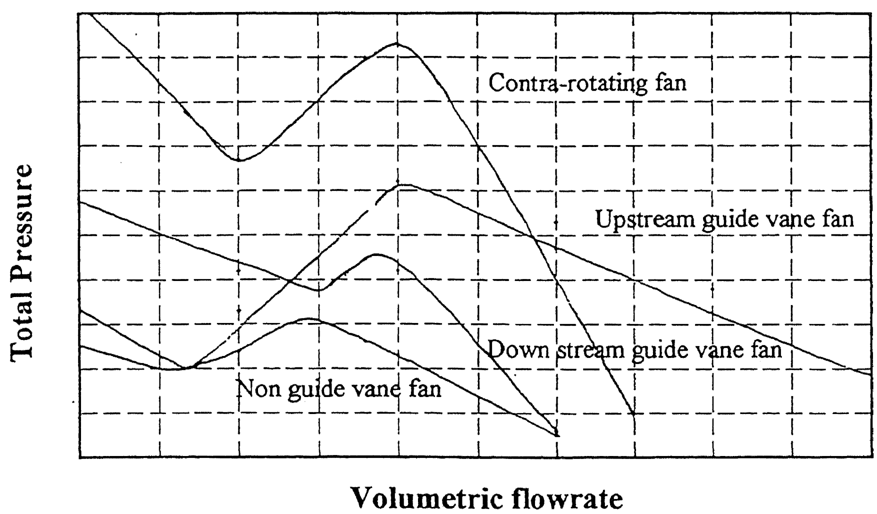


Figure 3.8b Axial fan characteristics

3.6.3 Propeller fans

It is a simple form of axial fan with impeller fixed in a ring, a orifice or a diaphragm plate and discharges air both as axial and radial components. These fans can be closely associated with axial fans. They are able to handle high volume and low pressure flows and is recommended to be used for pressures below 125 Pa. This generally is good at moving air from one space to another and so is good for ventilation systems requiring no duct work.

Typical characteristic curves are as shown in Figure 3.9 which shows the power curve to increase markedly as zero flow is approached which may be due to adverse wind conditions or badly designed highly resistant exhaust hoods. The efficiency ranges from 60-75% depending on smaller or larger units.

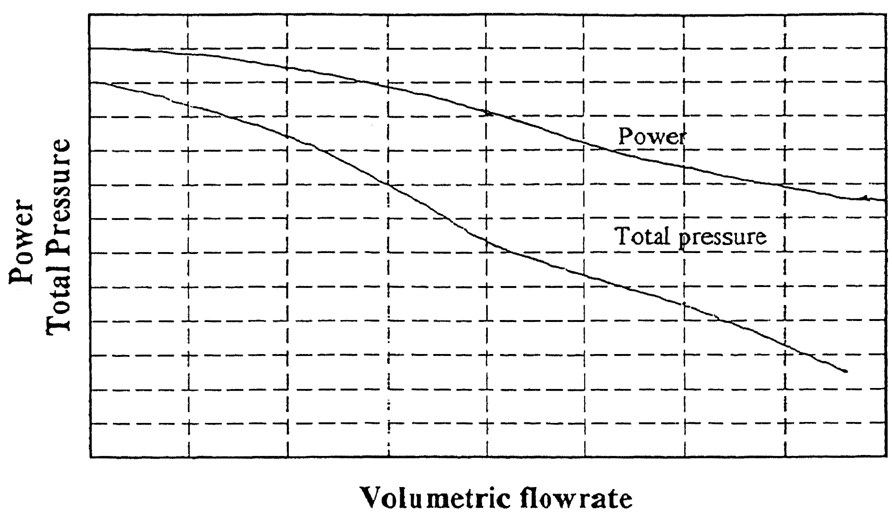


Figure 3.9 Propeller fan characteristics

3.6.4 Cross flow or mixed flow or tangential flow fans

These are fans having blades somewhat similar to forward curved centrifugal fan impellers. The air path through the impeller is somewhat between that of an axial and centrifugal type. The impellers of such fans are sealed at both ends and fitted into a casing such that the air enters at the periphery on one side, passes through the impeller and leaves from the periphery on the other side. Hence the flow is not exactly axial but follows a curved path with axes of inlet and outlet being roughly at right angles. These type of fans can develop more head than a comparable axial fan. Volume flows in such fans can be unlimited as the impeller can be made of any practicable length and also as throttling at the inlet end is not a problem as observed for a centrifugal fan. But because of the fact that the pressure developed by it is mostly in the form of velocity pressure and also efficiency being low, not much

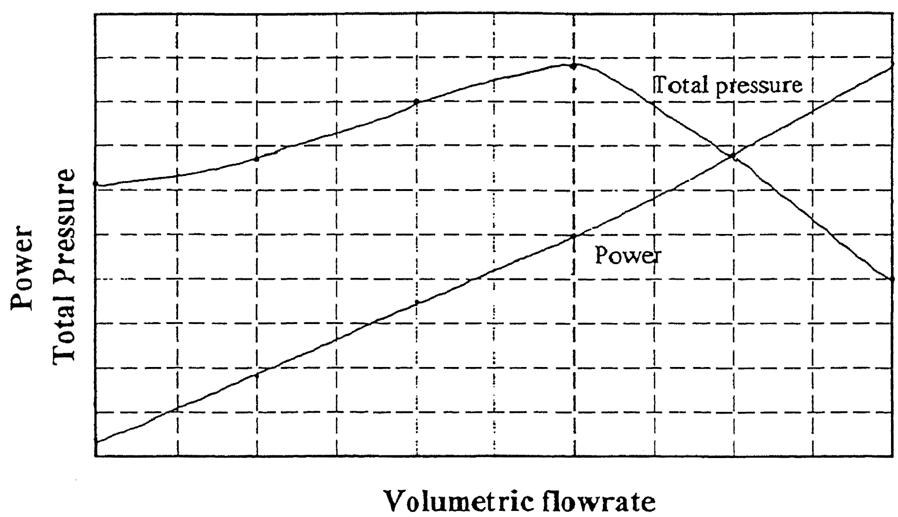


Figure 3.10a Cross flow fan characteristics

development has been made on these fans. These are generally used in small sizes as domestic heaters and so on. Characteristic curves for small domestic equipment is shown in Figure 3.10.

3.7 FAN SELECTION

Fan selection for the harvester was based on the advantages and disadvantages with respect to harvester.

Centrifugal fan

i. Can handle medium to low volumetric flow rates.

ii. Can develop medium to high pressures.

ii. Backward inclined blades

For fans with backward inclined blades, where blade outlet angle $\beta_2 < 90°$, power curve is non overloading type hence, η = 80% and if aerofoil sections are used η = 90%. They are able to develop the least flow volume and total pressure. They have narrower and fewer blades with greater radial depth. The number of blades generally considered for such fans are in the range of 6-16.

iv. Radial blades

Centrifugal fan with radial blades, where blade outlet angle $\beta_2 = 90°$, η = 75%. Such a fan is able to develop total pressure and flow volume slightly more than backward inclined blades, as total pressure p_T and volumetric flow rate Q increase with increase in blade angle for impellers of same given proportions. The power curve is slightly overloading type but not as extreme as forward bladed fans. These are self cleaning type of fans and can handle moderately dirty conditions. Blades, same as for backward inclined blades are narrow with fewer blades of greater radial depth. The number of blades considered are in the range of 6-16.

v. Forward inclined blades

Centrifugal fan with forward inclined blades, where blade outlet angle $\beta_2 > 90°$, $\eta = 40\text{-}50\%$. The total pressure and volumetric flow rate developed are quite high when compared with the other two types. But a very small increase in flow rate can cause a considerable increase in fan power causing overloading of the motor drive, which generally is basically selected for a particular duty rather than for maximum possible fan power. Hence it is obvious that these fans have an overloading power curve. Therefore maximum power required is greater than power absorbed by the fan while working at its maximum efficiency. High flow volume is achieved because the small radial depth of blades allow a larger inlet diameter which in turn prevents throttling of the higher flow volumes entering the impeller. Hence number of blades range from 30 - 60 generally for such fans.

But both radial and forward curved centrifugal fans tend to require more blades than backward curved types. They also need to run at higher speeds to provide large flow rates. In consequence they tend to be noisy in comparison to backward curved centrifugal type and also have lower efficiencies. It should also be realised that a portion of the power loss due to low efficiency appears as noise.

Axial fan

i. Can handle medium to high volumetric flow rates.

ii. Can develop medium to low pressures. Is able to develop pressures lower than centrifugal fans of same diameter and speed. Hence is good for moving air with no suspended solids in it.

iii. Efficiency is about 75% for non guide vane fans which is the minimum and keeps on increasing for fans in the order of down stream guide vane fans, upstream guide vane fans and contra rotating fans as can be seen from the characteristic graph. It goes upto 87% for large downstream guide vane units.

iv. True axial discharge from any axial unit is possible for only a single operating condition i.e may be just a particular flow rate or total pressure.

Conclusion on fan selection

Fan requirement for the air assisted harvester is a fan that can handle high volumetric flow rate (12-30 m^3/s) and develop enough suction pressure to be able to move air with seeds through the header, ducts into the separator where the seeds gets separated from the air and then the air itself enters and gets discharged through the fan into the atmosphere. The fan should be able to operate at different operating conditions i.e, different flow volumes hence different total pressures as is required for harvesting different seeds.

On comparing all 4 types of fans the 2 latter types could be excluded as they were small fans. On comparing the centrifugal and axial fans, centrifugal fan with either backward inclined or radial blades were selected as these develop good efficiency, can handle moderately dirty conditions, have non overloading power curves, produce low frequency noise. Axial fans when connected in parallel seem to present a very real potential noise problem hence is another reason for disregarding then in favour of centrifugal fans.

3.8 CRITERION FOR FAN DESIGN

The criterion used to design a fan were-

1. Economical criteria
2. Tractor power/power (if self driven) availability
3. Size of header units available
4. Required volumetric flow rates and velocity of air
5. Size of separators and the duct system required
6. Total fan pressure to be developed

Once the system requirements were charted, the design of the fan proceeded by back calculation. The one dimensional and the velocity triangle theory was considered. The fluid flow through an impeller is a function of positional coordinates r, θ and z. But to simplify, the flow through the impeller has been considered only along one dimension [17].

Hence the assumptions made:-

i. The number of blades considered are many, so that the variation of velocity across the blade passage tends to zero, ie, the flow is axisymetrical and there is perfect symmetry with respect to the axis of impeller rotation. Hence,

$$\frac{\delta \nu}{\delta \theta} = 0$$

ii. The blades are infinitely thin and pressure difference across them is replaced by an imaginary body force acting on the fluid to produce a torque.

iii. The area where triangles of energy takes place in the impeller along the blade passage, the variation of velocity in the meridional plane (z) ie, across the width of the impeller is negligible. Hence,

$$\frac{\delta v}{\delta z} = 0$$

Hence when in reality, fluid flow through impeller is

$$v = f(r,\theta,z)$$

For one dimensional flow, it is reduced to

$$v_\infty = f(r)$$

only, where suffix infinity indicates assumptions of infinite number of blades, hence axisymmetry. Although while drawing we show only finite blades they are not taken into account and from assumption, (i) fluid streamlines are confined between infinitely narrow inter-blade passages, therefore their paths are similar to the shape of the inter-blade centre line as indicated by dotted line in Figure 3.11. Assumption (ii) helps us to treat space between impeller inlet and outlet as a 'black box' choosing input and output in the form of inlet velocity triangle and outlet velocity triangle. Such a velocity triangle at a constant angular velocity 'ω' for a backward inclined centrifugal impeller is shown in the Figure 3.11.

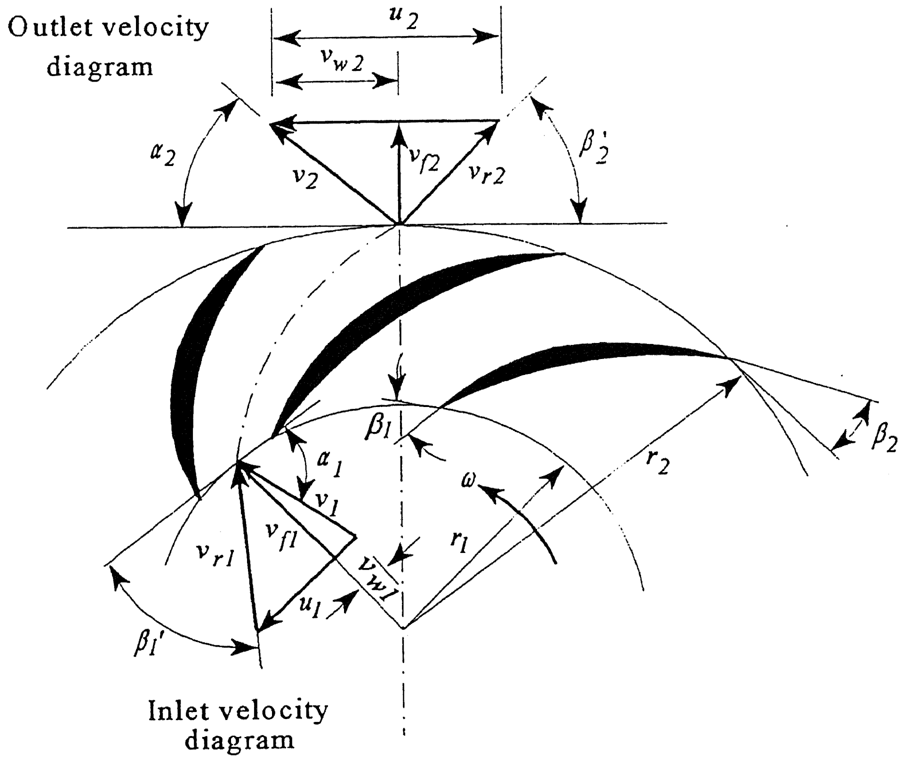


Figure 3.11 One dimensional flow through a centrifugal impeller

The Figure 3.11 shows at inlet, fluid moving with absolute velocity v, entering the impeller through a cylindrical surface of radius r, making an angle α_1 with the tangent at the radius. At outlet the fluid leaves the impeller through a cylindrical surface of radius r_2 with an absolute velocity v_2 inclined at an angle α_2 to the tangent at the outlet.

Velocity triangles

Method of drawing the inlet and outlet velocity triangles have been illustrated.

Inlet velocity triangle

i. Draw a vector to represent absolute velocity v_1 at angle α_1 with the tangent at inlet.

ii. Tangential velocity of impeller u_1 is then subtracted from it vectorialy to get the relative velocity of fluid (v_{r1}) with respect to impeller blades at radius r_1.

iii. Then absolute velocity v_1 is resolved into two components in radial direction called velocity of flow v_f, and the other perpendicular to it, ie in tangential direction called whirl velocity v_{w1}.

Outlet velocity triangle

i. Vector representing absolute fluid velocity v_2 at angle α_2 with the tangent at the outlet is drawn.

ii. u_2 is subtracted vectorially from v_2 to get v_{r2}. he absolute velocity v_2 is again resolved in to two components to get v_{f2} and v_{w2} the radial and tangential components.

Euler's turbine equation is then derived for the energy transfer between the impeller and fluid based on this one dimensional theory and given by the equation, Work done per second,

$$E_t = \frac{W}{g}(u_2 v_{w2} - u_1 v_{w1}) \tag{3.10}$$

S.I unit for above expression is Watts. Since work done/sec by impeller on the fluid, as is the case, is the rate of energy transfer, hence rate of energy transfer per unit weight of fluid flowing

$$E = \frac{E_t}{W}$$

$$E = \frac{1}{g}(u_2 v_{w2} - u_1 v_{w1}) \tag{3.11}$$

This equation 3.11 is known as Euler's equation. Units for this equation is Joules/Newton. This expression further simplifies to metres and is same as for all terms of Bernoulli's equation and E may be used in conjunction with it. By the mode of derivation of this equation, it is obvious that it is applicable to a pump as well as a turbine although in the latter $u_1 v_{w1} > u_2 v_{w2}$. In equation 3.10 and 3.11, W/g is the fluid entering impeller per second.

The fan design based on the above mentioned theory is shown in Chapter 5.

4.0 SEPARATOR

4.1 INTRODUCTION

A separator is a unit used for separating one material from the other for example like separating solids from gases, solid from liquids or liquid from liquid. Such separators irrespective of whether the end product is the clean gas or the separated solids are generally known as a gas cleaning or de-dusting equipment with respect to the gas. The main substance of this thesis is the separation of grass seeds from a large volume of air. The thesis deals with the selection, design and development of a model for this purpose and has been referred to as the separation system. Gas cleaning plays an important role in our lives and is a necessity either for environmental reasons, for health or for technological requirements, where the separated gas or solid or both are needed as separate identities or for further processing.

This project is solely concerned with separation of solids in the form of grass seeds from a fast moving air stream. The role of the separation system in the air-assisted vacuum harvester is to separate seeds from volumetric air flow rates of 12-30 m^3/s travelling at high velocity. This high air velocity at the picking head is imperative for harvesting the ripened seeds from the grass heads (as explained earlier). Previous studies [1] have shown that for air-assisted brush type harvesters, air velocities ranging from 12-20 m/s are required at the picking head for efficient seed stripping. In the studies conducted previously air-seed separators were always placed after the fan. The harvested seeds thus had to pass through the fans. The studies showed that such seeds experienced trauma which in turn had a dramatic effect on seed germination. A total of only about 5-10% of live seeds were obtained while 85% of the seeds on the heads were harvested by this technique. This project has therefore been concentrated on improving the separating capability and reducing any trauma to the seeds by placing the separator ahead of the fan. Consequently improving the efficiency of the harvester by enhancing the quality ie the germination capacity of the entire harvested seeds.

In chapter 2 it has already been mentioned that most perennial tropical grasses produce seed heads over a period of four weeks. The seeds ripen and mature over a period of one to two weeks after inflorescence. The main problem therefore was to time the harvesting such as to recover maximum mature seeds by one pass. Using grain harvesters which are the header type, reduced the effective collection of mature seeds due to the collection of inert materials and immature seeds along with the mature seeds. Grain harvesters were also seen to be ineffective for harvesting of light and fluffy seeds such as Buffel and Blue grasses. Although mechanical harvesting of grass seeds had commenced 53 years ago, a successful design for harvesting the large variety of grass seeds was yet to be developed. A number of Stripper harvesters operating on the principle of a striking force on the seed heads dislodging the seeds onto a collecting bin has been described in [9]. Some modified versions using the same principle are still being used called Beater harvesters. But as mentioned earlier, they tended to collect a lot of trash and so required costly post harvest treatments such as cleaning and drying to meet the commercial seed quality standards. In 1970 and 1979, Vacuum or Suction harvesters were developed to transport the dislodged seeds to the collecting bin [10][11]. Here the seeds were separated from the air flow by using a dust collector initially and later by using a screen separator. This harvester consisted of cylindrical brush with nylon bristles arranged spirally on a central metal case of 102 mm diameter. During harvesting this unit rotates upwards stripping and conveying the seeds into a rear bin. Loss of seeds flying in air was minimised by the design of a curved shroud and a wind cut off board, but comparative experimental studies conducted with different brush designs and brush speeds suggested efficiency of such harvesters to be limited. In 1983, it was found that harvesters with different brush designs and operated at a relatively high speed of 750-1000 rpm yielded only 0.9-14.6 kg/ha [12]. Kawana Engineering at Rockhampton produced working models of a modified form of Woodward Flail-Vac harvester for harvesting Buffel and Blue grasses. The variation in the design being in the arrangement of four rows of brushes, instead of the spirally arranged design. The air discharge opening at the top was widened and the seed bin size was reduced. Provision of a bent to the upward curved header front assisted in gathering the seed heads to about the same height. Finer bristles of 0.375 mm in 12.4 mm tufts 140 mm in length at 55 mm centres were used. The overall brush diameter was 500 mm. Through field tests the optimum operating brush speeds required was found to be 280-350 rpm for Buffel, Rhodes and Blue grasses to minimise trash being stripped off along with the mature seeds. Tests conducted on both spiral configuration and the four row type designed by Kawana showed no difference in the air velocities. This type of harvester was also found to have limitations on its application and efficiency. From the results of the previous experiments it was concluded that to improve the working efficiency a higher velocity airflow at the stripping front of the harvester was required, to improve the stripping action, the gathering up of all the stripped seeds and the transportation of the same into the storage. Since 1990 the development of such Power Assisted Air Flow harvester has improved the yield of mature grass seeds and their quality from 60% to 85% as has been already mentioned in chapter 2.

In Australia much of this development was done by Kawana Engineering at Rockhampton and later the Queensland Department of Primary Industries Engineering Group at Toowoomba who together produced a continuous flow harvesting system using the Flail-Vac system in combination with a mechanical and pneumatic conveyor for transporting the seeds to the collecting bin. To remove the leaves and stalks from the harvested seeds a pre-cleaning wheel was incorporated into the pneumatic conveying system. A strong airflow at the stripper front was required to overcome the vortex problem and for streamlining the flow continuity as well as to improve seed quality and quantity over a wider range of grass seed types. Experiments on these prototype harvesters were conducted by the Mechanical Engineering Department, at the Central Queensland University. The experiments [1] established the range of air velocities required to dislodge and transport a variety of grass seed types. While 2 m/s or less air velocities were required for seeds of Rhodes grass, Buffel and Bluegrass, other heavier seeds such as Jap Millet, Silk Sorghum, Siratro required up to 6 m/s and more to dislodge and transport the seeds. But a minimum velocity of 12 m/s was required to bend the seed heads towards the brush unit and to overcome the vortex which tended to push the seed heads away and disperse the seeds on the ground. Hence it was realised that 12 to 20 m/s air velocity would therefore be required at the harvester front to cover the range of perennial grasses normally harvested. The wide range of velocity is due to the fact that pasture grass seeds consist of a wide variety of seeds with different shapes, size and weight.

A field test was conducted in 1990 to establish the enhanced harvesting efficiency of a forced air draught combined with the brush stripping mechanism. In comparison with a unassisted air draught harvester, the tests revealed an increase in harvested seeds for Green Panic from 10 kg/ha to 24 kg/ha, for hatch Creeping Bluegrass from 15 kg/ha to 45 kg/ha and for Angleton grass from 18 kg/ha to 45 kg/ha for air assisted harvester with fan speed of 1200 rpm and brush speed of 500 rpm. Tests for seed quality also showed a significant increase for purity, germination and pure live seed. These results provided an added incentive to develop power airflow harvesters with improved harvesting and conveying components and to establish mechanical engineering standards for the design of these harvesters.

Although the results proved to be very positive it was felt that this harvester could be further perfected in areas of harvester picking head and in the design of a fan. This would then enable the harvesting of a wide range of seeds from the light fluffy type to the large heavy type of seeds like the legumes, by controlling the volume flow and the flow velocity. It was also observed that although mature seed yield had increased, the germination of seeds had not increased by the same percentage. This was attributed to the fact that in these early prototypes the seeds had to pass through the fan before getting separated from the air flow. This traumatised the seeds to such an extent as to affect its germination.

It was thus decided to extend the research to develop an improved form of separator that could be connected ahead of the fan. The seeds would then be prevented from experiencing the trauma of having to pass through the fan which seemed to reduce its germination capacity. There was a need to improve the separating efficiency of the seeds from the large volume of air flow. This project therefore is an attempt at achieving these improvements in fan design, separator unit efficiency and at setting up of mechanical engineering design standards for the harvester. Few other books, journals and papers dealing with the subject and which were referred for reading are [22] to [35].

4.2 AIR-SOLID SEPARATION

The air-solid separation process consists of three main phases-

1. Separation which is the movement of solid particles to the separator surface.
2. Particle fixation which involves the collection of these separated particles into a collecting bin.
3. Bulk material handling which deals with the handling or removal of the collected materials from the gas cleaning equipment.

Generally such a equipment is known as a gas cleaning equipment irrespective of whether the end product is clean gas or separated solids. This thesis is mainly concerned with the first two phases of the gas cleaning operation. The third phase although a part of bulk material handling, plays an important role in the efficiency of any separating unit. Experience has shown that ignoring the third phase of bulk material handling has many times caused losses in the working efficiency of the separating unit. Some suggestions have been included in chapter 5 for dealing with such problems for an air assisted harvester. Further research can be done on the bulk handling aspects of the separating unit.

4.3 TYPES OF SEPARATORS

As mentioned earlier a separator could be used for separating any one material from another either in the form of solid from gas, solid from liquid or liquid from liquid. To achieve separation, most separators are built on a principle dealing with any one characteristic of the separated particle, which may be electrostatic charge as in precipitators, mass, density as in cyclones and centrifuges.

4.4 PRINCIPLE OF SEPARATION OF PARTICLE SEPARATORS

The particle separators operate on the basis of the type of forces applied onto the solid particles.

1. External forces acting external to the gaseous suspension such as for example gravity, magnetic or electrostatic forces.
2. Internal forces which are due to forces or effects taking place within the gaseous suspension. These for example may be inertial or centrifugal forces, electrostatic effect between charged particles and diffusion.

4.5 CLASSIFICATION OF GAS CLEANING EQUIPMENT

Such equipment operate either on any one principle or on a combination of two or more of the above mentioned principles. In the most common types of classification used, they are mainly placed into one of the four groups.

1. Aero mechanical dry separators- which operate on the principle of gravity or inertia.
2. Aero mechanical wet separators- which are also known as scrubbers are based on the principle of diffusional and inertial effects.
3. Electrostatic precipitators- which operate on the principle of electrostatic or gravity forces.
4. Filters- which operate on the principle of inertial and diffusional effects.

For better and efficient separation, most separator systems combine two or more units in series or otherwise build them as a single unit.

This project mainly deals with aero-mechanical dry separators, as separation of harvested seed is a dry operation with solid particles being the seeds having neutral charge and having no chemical characteristics as such. Therefore only the names of the different types of aero-mechanical separators have been mentioned here. The centrifugal separator have been discussed briefly. The description of the rest of aero-mechanical separators are dealt with in the supplement provided to the thesis. The description can also be found in [13] and [16]. The other articles and books that were referred for this topic are [32][33].

4.5.1 Aero-mechanical dry separators

Such separators generally operate on the principle of separation by gravity or by inertial effects upon the particles or by a combination of both. For those operating on gravity, the particles settle out of the gas stream due to their weight. In momentum separators, the suspended particles are thrown out of the gas stream due to the inertial effect that the particles are subjected to by a sudden change in the direction of the gas stream. Collectors that are based on centrifugal force such as cyclones are also special cases of the inertial effect.

The rate at which the particles are separated is proportional to the force exerted on them. For small particles, smaller than 100 μm in size, collectors based on gravity are found to be generally ineffective and slow. In such cases the separation can be greatly enhanced by employing the inertial effect. This also helps in the reduction of equipment size and increased effective collection down to particles of up to 20 μm in size. In some cyclones it can go down to 5-10 μm [13].

In this category there are mainly five types of gas cleaners.

1. Cyclones
2. Dual vortex separators
3. Settling chambers
4. Inertial separators
5. Fan collectors (or mechanical driven cyclones)

There are others which are not in popular use such as dust collection centrifuge or scroll collectors.

Two separators were selected for the experiment operating on the principle of centrifugal force. They are Curved duct separator and Uniflow cyclone separator. As both these separators function as concentrators, for the experiments they were set up in combination with a settling chamber. Hence the working and the advantages/disadvantages of the various cyclones have been mentioned here in brief.

4.5.2 Cyclones or Centrifugal separators

In centrifugal separators the gas is given a spinning or vortex motion to impart a centrifugal force to the particles. It is this centrifugal force caused by the change in flow direction that drives the particle in the gas stream to the wall. It is very popular and very widely used because of its simple structure, with no moving parts and low initial expense.

4.5.2.1 Cyclone classification and description

Cyclones can be classified into two main categories based on the direction in which the clean gas leaves the main body or cylinder.

- Reverse flow cyclone
- Uniflow cyclone

These two types each can further be classified into two based on the position of the entry of the dust laden air into the main body. They are the axial entry type and the tangential entry type.

The axial entry type of cyclones are used in parallel mainly to deal with large volumes of gas at low pressure drop, but the collection efficiency is not high. In axial flow devices, the gas stream enters at one end of the cylinder. It flows through vanes which impart the spinning motion. The separated and collected particles are carried out by a peripheral stream, while the gas exits through a central outlet at the opposite end from the entry. This type of cyclone can be mounted in any position.

In the tangential entry type the gas stream enters tangentially into the cylinder thus giving the gas stream a spinning motion. These are generally mounted vertically. Particles thrown out against the wall are collected by sliding down into a hopper below.

Reverse flow cyclones

These are more commonly used with tangential entry than axial entry. The gas stream laden with solids enter the cylinder tangentially creating a strong vortex inside the cylinder body. Particles in the flow are acted upon by centrifugal force which causes them to move radially outwards against the wall of the cyclone, where they are separated and slide down into the hopper below. The gas flows downwards in a vortex and reverses its direction of rotation at the bottom of the cone and leaves as clean gas from the central exit duct placed at the top. There are various other inlet types such as spiral, helical or axial, leading to some variation in performance. But tangential entry is seen to be most effective and simple in construction. Hence tangential entry is more popular than others.

Uniflow cyclones

As mentioned in the previous case, these devices may have axial or tangential entry. The only difference being in axial entry the gas enters at centre of cylinder and flows through vanes which impart the spinning motion, while in tangential entry, the gas stream enters the cylinder tangentially thus creating a strong vortex inside the cyclone. In either cases, the solids, thrown outwards to the wall are carried out with the peripheral stream, while the clean gas exits through a central outlet placed at the opposite end of the entry.

The schematic diagram of the above mentioned cyclones are shown in Figures 4.1 and 4.2 respectively. These devices can be built of any metal or even ceramic material if necessary to withstand high temperature, abrasive particles or corrosive atmosphere. A most important point to be kept in mind while fabricating such a device is to have the interior surface very smooth. There are no moving parts, so operation is usually simple and relatively free of maintenance.

While in reverse flow separators, the solids are collected totally free of air, in uniflow the separated solids leave with 5-10% of the main gas flow here termed as the 'underflow'. It is therefore more of a concentrator and should be combined with a secondary stage collector to separate the solids from this underflow.

In geometrically similar cyclones of different sizes the pressure drop is seen to depend on inlet velocity and not on the size of the cyclone. Collection efficiency is

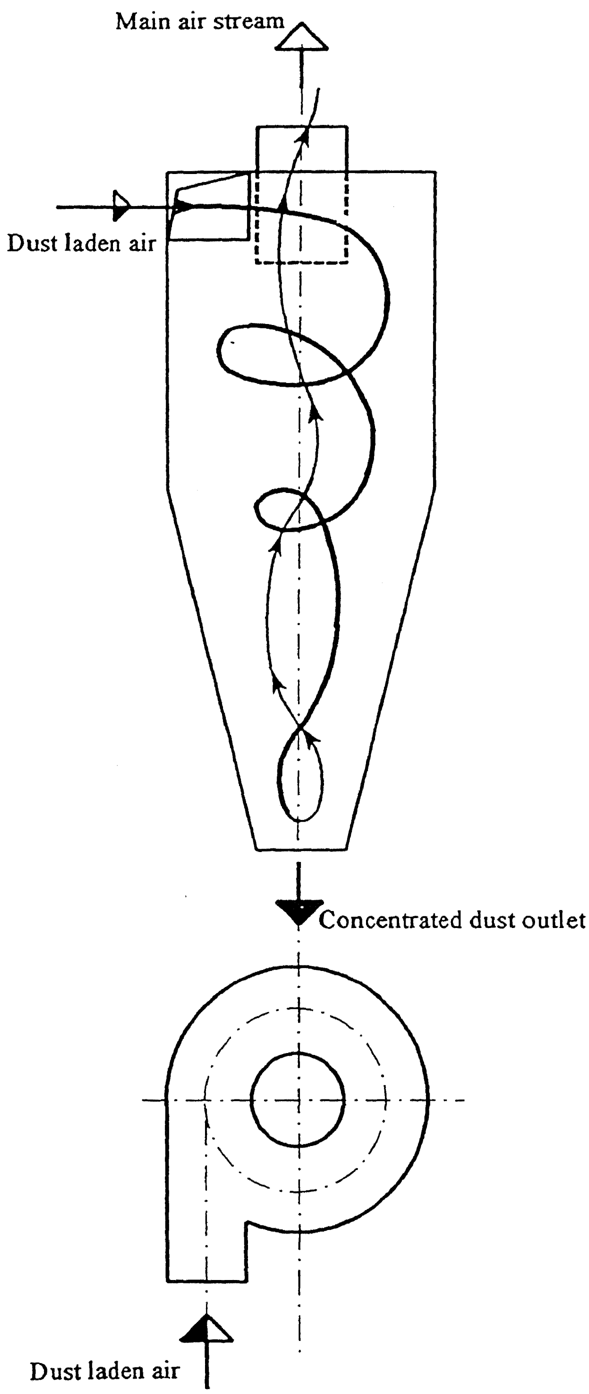


Figure 4.1 Schematic diagram of reverse flow cyclone with tangential inlet

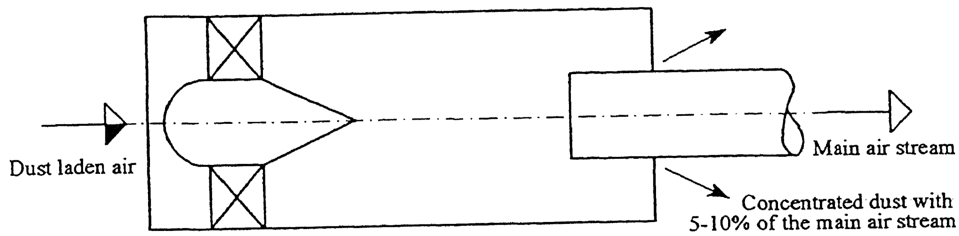


Figure 4.2 Schematic diagram of Uniflow cyclone

noticed to improve with reduction in cyclone size. While dealing with the same volumetric flow rate of air, the collection efficiency and pressure drop are noticed to decrease at elevated temperatures. But as dust loading or solid particle concentration increases, pressure drop reduces and collection efficiency is observed to improve [14].

General features

1.	Type	Cyclone
2.	Removal particle size	< 1 μm
3.	Pressure drop	>2000 Pa
4.	Maximum temperature	1000 ^{0}C
5.	Inlet concentration	>1000 gm/m^3
6.	Initial capital cost	Medium
7.	Remarks	Convenient

Graph with cut size diameters (D in meters) of standard cyclones verses diameter of particles is shown in Figure 4.3.

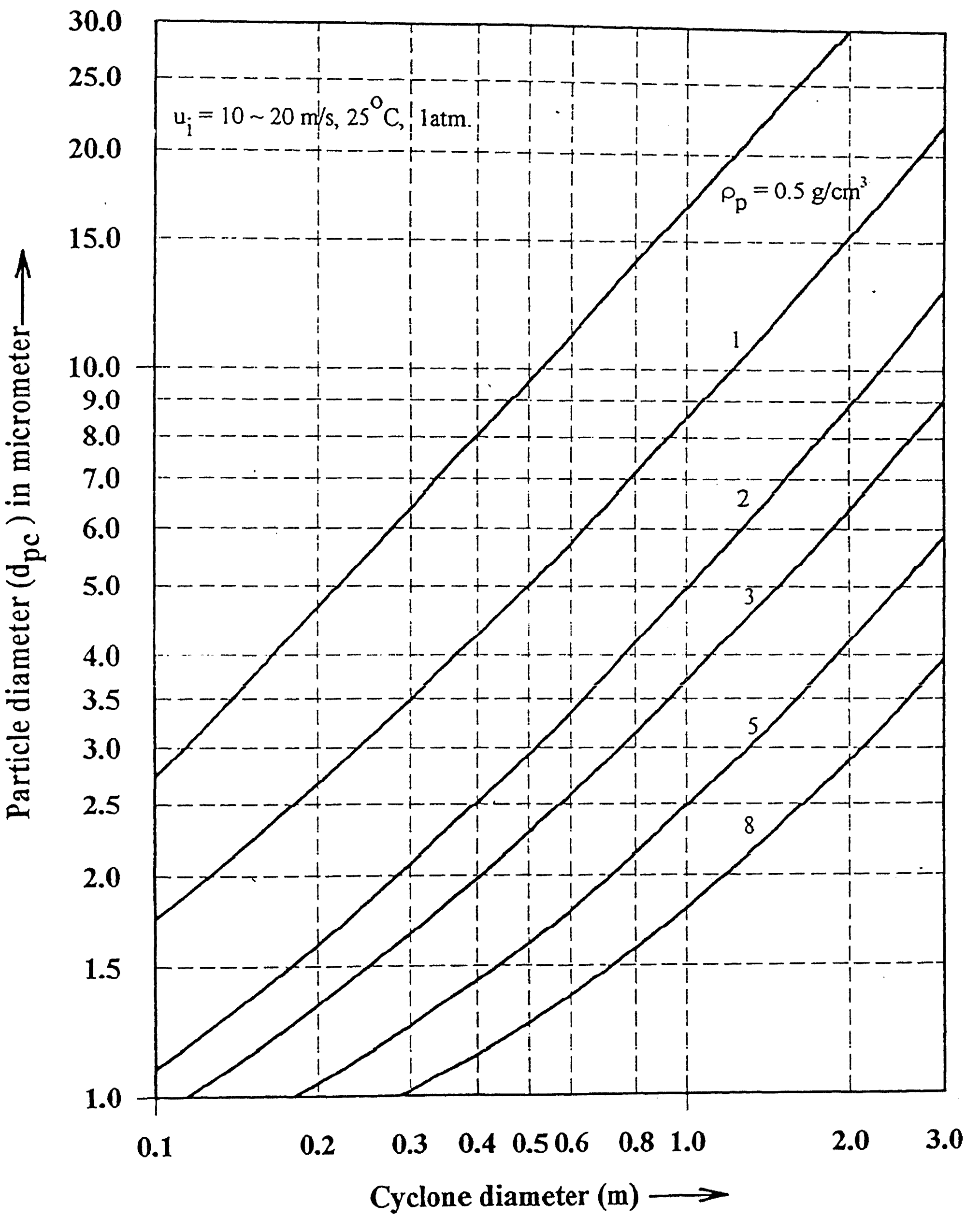


Figure 4.3 Cut size of standard cyclone

4.5.2.2 Advantages of the cyclone separators

i. Although very effective for separating coarser particles, for certain cyclones effective collection has been observed for particles as small as 5-10 μm[13].

ii. Simple in construction.

iii. No moving parts so operation is simple and is relatively free of maintenance.

iv. Can withstand high temperatures, abrasive particles, or corrosive atmospheres as any type of metal or even ceramic may be used if necessary.

v. Low running costs.

vi. Reliability in use.

vii. Relatively inexpensive in comparison with other types of collectors.

viii. Low pressure drop, the acceptable levels of pressure drop for cyclone operation are generally less than 20 cm water gauge (i.e. about 2 kPa) [13]. Pressure drop is observed to be dependant on the inlet velocity and not on cyclone size and is seen to reduce with increase in feed concentration.

ix. Collection efficiency is seen to be as high as 80% for particles as fine as 10 μm[13]. Collection efficiency is affected more by diameter of the cyclone, feed concentration and air inlet velocity. It is observed to improve with reduction in cyclone size and increase in feed concentration. An increase in temperature reduces the collection efficiency.

4.6 TYPES OF SEPARATORS SELECTED FOR THE EXPERIMENT

The discussion and comments with regard to the separators were based on particulate materials such as dust, powder and other solids. In this thesis the same principles have been applied to a new material, i.e the grass seeds which have very different and unique characteristics. The seed types vary from seeds with light fluffy hair attached, seeds having a parachute type extension on one end of its solid body to almost spherical smooth seeds. Some examples of the shapes of different pasture

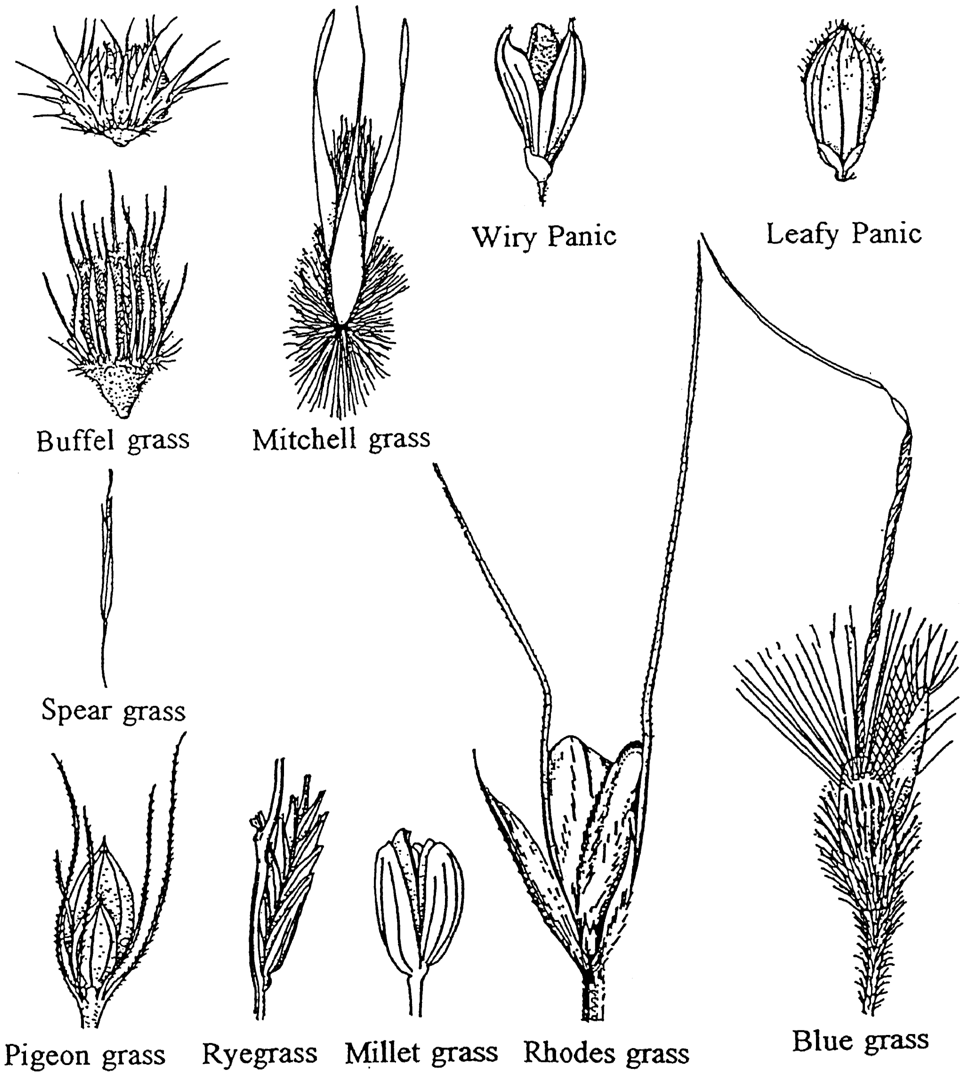


Figure 4.4 Different pasture seed head

seed head are shown in Figure 4.4. Hence it was necessary to conduct more experiments and apply special theory to account for these great differences in the separation process.

The points that were taken into consideration for choosing an appropriate separator for the harvester were-

1. Separator able to handle a wide variety of grass seeds.
2. Simple in construction, hence low in capital cost.
3. Low running cost.
4. Reliability in use.
5. Maintenance free, i.e, should be easy to operate with minimum or no moving parts.
6. Low pressure drop, not more than 2 kPa, hence energy required for operation is also relatively low.
7. Should be able to handle large flow rates.
8. High separation efficiency.
9. Should be small to medium in size and still be able to deal with flow rates ranging from 12 to 30 m^3/s and with 0.13 to 1 kg/m^3 feed concentration.
10. Separator should be such that it can be placed before the fan, thus preventing the seeds from being damaged by having to go through the fan.

After considering the various aero-mechanical dry separators and based on the selection criteria it was concluded to design and test separators operating on the principle of centrifugal separation. It was felt that separation efficiency would improve if the length of the curved path through which the seed could travel was increased. Earlier research had involved a curved tube separator and although that work was unpublished [1], satisfactory results were claimed and it was shown that grass seed could be separated before the fan by this type of inertial separator. In a curved tube the number of turns the seeds can take is limited. A centrifugal separator can provide a larger number of turns to the seed path giving them enough time and length of curvature to separate out from the main flow. A Uniflow cyclone was also selected for testing. Since both the uniflow and the curved tube separators function like a concentrator so the test models were set up in series with a settling chamber.

4.7 SEPARATOR THEORY

The theory applied to separation of particulate material from a transporting airflow by means of forced path curvature followed by gravitational separation is discussed below. In this study, particulate material of irregular shape is to be first concentrated into a small portion of the total cross-section of a transporting airflow by some form of 'centrifugal' concentrator, and then, having been removed from the main airflow together with some air, further separated from the remaining airflow, by means of an aspirated deceleration separator or settling chamber.

The dynamic parameters of interest are likely in consequence to be as follows.

1. The average mass M of individual particles, and the ratio of this average mass to the effective cross-sectional area A_e of an average particle. This effective cross-sectional area will usually be greater, perpendicular to the direction of motion of the particle relative to the surrounding air, than the measured area of the particle itself, due to the existence of a boundary layer, stationary relative to the particle, around each projection of the irregular shape.

2. The frictional force which resists travel of the particle relative to the surrounding air. This force will be proportional to the effective area A_e of the particle as well as the square of its velocity, and also to the drag coefficient, C_d, which tends to reduce with increase of Reynolds number of the airflow around the particle.

3. The 'driving' force F_r which causes motion of the particle relative to the surrounding air in which it is transported. When the transporting airflow is forced to follow a curved path due to curvature of the duct or other constraint, the particle, which is assumed to be at rest or to have a very low velocity relative to the transporting flow in the absence of curvature of the flow, must be subject to a force directed toward the centre of the flow curvature if it is to follow a curved streamline so as to remain at rest relative to the transporting flow. Such a force can only be applied to the particle by the transporting flow of air itself, and is in fact the drag force due to the motion of the particle relative to the flow.

In consequence, the particle must move radially outward, across the transporting airflow, toward the 'outer' wall of the duct or other constraint, and will accelerate radially to a velocity V_r at which the 'drag' force upon the particle is equal to the 'driving' force F_r as shown in Figure 4.5. The drag force F_D is given by

$$F_D = \frac{C_d \, A_e \, \rho \, V_R^2}{2} \tag{4.1}$$

While the 'driving' force at any radius R_p of the particle path is

$$F_r = M\frac{V_f^2}{R_p} \tag{4.2}$$

where V_F is the flow velocity of the transporting airflow. While values of C_d, appropriate to various particle shapes can usually be obtained to at least moderate accuracy from published results, the effective cross-sectional area A_e is clearly of great importance in determining the radial velocity which a particle may attain under given conditions.

Fortunately, an approximate value of A_e may be obtained from free-fall tests, though it must be realised that the value obtained at low velocities will not be likely to remain constant at higher ones, since as the velocity of flow past the particle

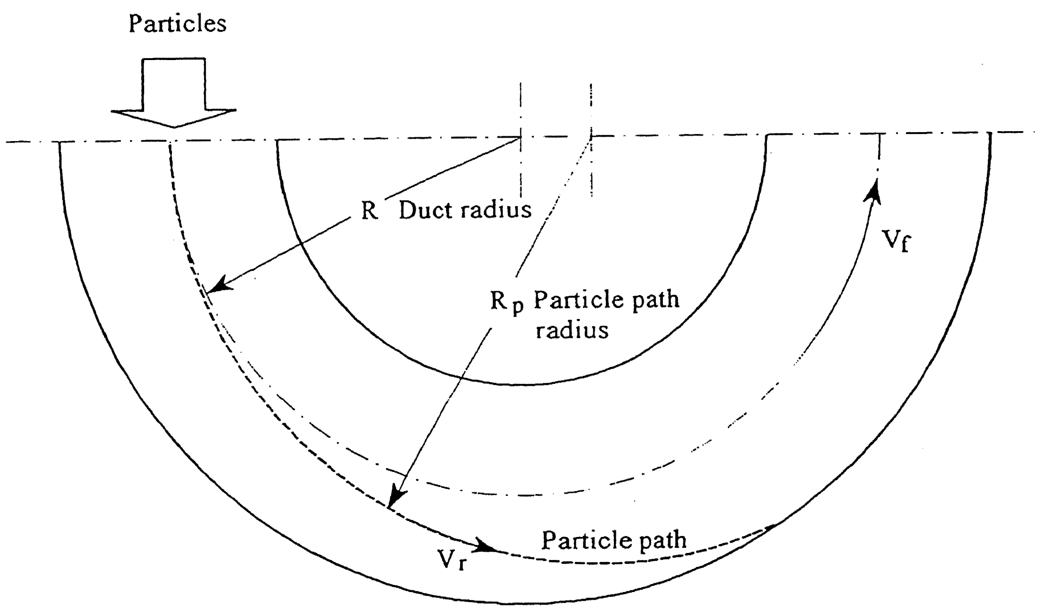


Figure 4.5 Typical particle trajectory in the curved duct concentrator

increases, the thickness of the boundary layer is likely to decrease, particularly if the flow becomes turbulent, so that A_e at high values of Reynolds number may be considerably smaller than that at low velocities. Any calculation based on free fall tests will then be conservative.

The arrangement of a free-fall experiment is extremely simple. A suitable known number of typical particles are supported by a mesh screen, and an airflow is arranged to flow upward through the screen, the air velocity being measured by a Pitot-static tube immediately above the screen. It was considered essential that the air flow be laminar, or at least, not fully turbulent, so that the screen was arranged to be no more than a few diameters above the inlet to the tube. In order to keep the value of the Reynolds number in the tube as low as possible, the tube and mesh diameter was reduced to 25 mm for particles whose terminal velocity V_T in free-fall was likely to be greater than 5 m/s, giving Reynolds number to be approximately 10^3. The method used involved noting the upward air velocity at which 10% of the particles began to levitate in the air flow, and also the velocity at which only 10% remained on the mesh, the arithmetic mean of these velocities being taken as the average free-fall terminal velocity. Since the propulsive force on the particle is its weight Mg under gravity, it follows that this must be equal to the total drag at the terminal velocity measured, so that

$$Mg = C_d A_e \rho \frac{V_T^2}{2} \tag{4.3}$$

Measurement of the particles sizes, most of which were varieties of grass seeds, allowed calculation of Reynolds number of the air flow around the particle, and thus a value for C_d to be found, if only approximate, by examination of published results by Stokes. The shape of many particles was such that a value of C_d intermediate between that for a sphere and that for a flat plate perpendicular to the direction of the flow was felt to be appropriate as shown in Figure 4.6a and 4.6b, while a few of the seeds used were almost perfect spheres. Though these latter type of seeds tended to exhibit terminal velocities in the range of 4 to 10 m/s, the very irregular ones tended to have lower terminal velocities in the range 1 to 5 m/s.

As an example, a particular variety of White French Millet used in the experiments with a shape very close to spherical, which exhibited a terminal velocity in free-fall of about 5 m/s and had on an average a relevant dimension, the half circumference, of about 1 mm, gave a Reynolds number 'Re' of

$$Re = \rho \frac{V_T}{\mu} D = \frac{1.2 X 5 X 0.001}{19.05 x 10^{-6}} = 315$$

showing that flow past the particle to be laminar, and from published results shown in Figure 4.5, the drag coefficient C_d is found to be about 3. In this case, from equation 4.3, the effective cross-sectional area was given by

$$A_e = \frac{2Mg}{3\rho V_T^2} \tag{4.4}$$

and since for this variety, the average mass found to be 0.008 gm, the value of A_e was calculated as $1.74x10^{-6}$ m^2, slightly greater than twice the cross-sectional area obtained by direct measurement. When the same particle is transported through some form of 'centrifugal' concentrator, it is likely to be subject to a much higher value of centripetal acceleration than gravity. In consequence, its terminal velocity in a radial direction will be higher than in a free fall experiment, so that not only is the

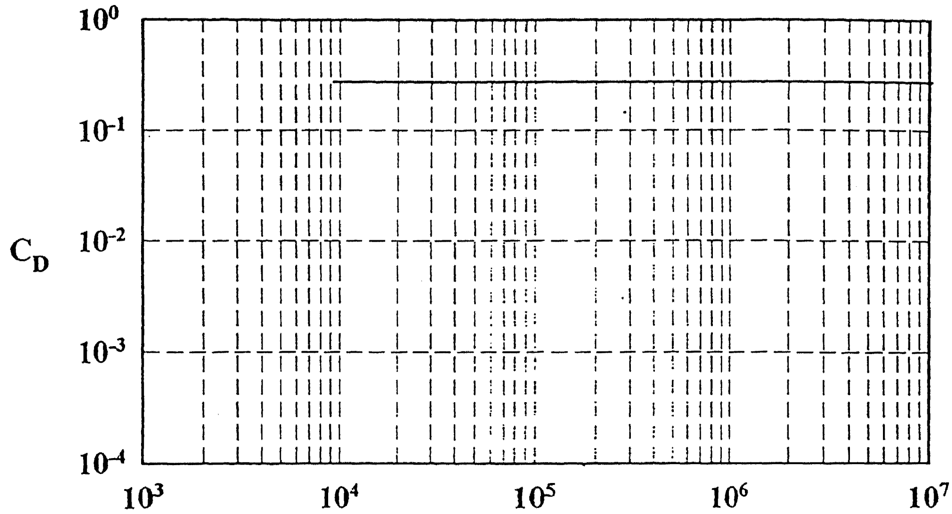


Figure 4.6a Drag coefficient as a function of Re number for a flat plate normal to the flow

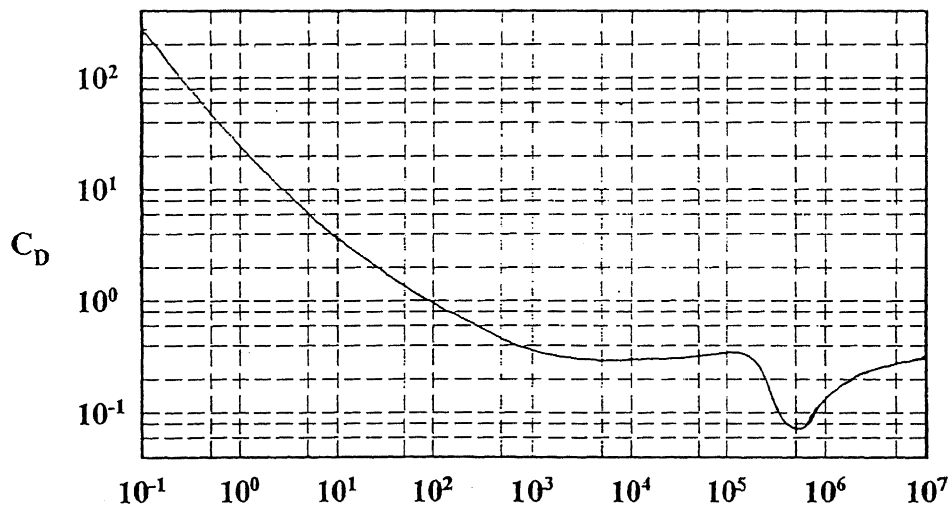


Figure 4.6b Drag coefficient as a function of Re number for a smooth sphere

Reynolds number of the flow past it in a radial direction much larger, but also the boundary layer responsible for its increased effective area is likely to be reduced in width, to the point where its effective area may be only a little greater than its measured cross section. While the drag coefficient for spherical particles may still be obtained from published work at the higher value of Re, it is more difficult to estimate the effective area A_e. It is in consequence useful to first calculate, using the theory above, a range of duct angle, in the case of a curved duct, within which full concentration is expected to occur for a particular particle type, and then to compare this with experimental results. No generally accepted definition of 'full concentration' appears to be available, but it is here suggested that the term be applied at the point where, having started with particles fairly evenly distributed in the transporting flow, 90% of the particles have been concentrated within 10% of the duct cross-sectional area, in the case of a curved duct.

Initial results from experiments with curved ducts which are described in detail later in the report appear to indicate that for fairly spherical particles, either the value of the $C_d x A_e$ product obtained from free fall experiments may have been too high, or alternatively that A_e decreases more rapidly with increasing radial velocity than had been anticipated, in that full concentration appeared to occur within a somewhat smaller angle of duct curvature than predicted. In these experiments, a 0.2x0.2 m curved duct having a radius of 0.4 m was used, with a velocity of transporting airflow of 15 m/s, so that the centrifugal acceleration of a particle on the duct centre line was approximately 563 m/s^2. It must, of course be understood that the radius of curvature the path of the particle, rather than that of the duct centre line, should be used in calculation of the particle acceleration. This, however is very difficult to observe, but in view of the fact that full concentration appeared to occur within 360^0 of duct curvature, whereas 90^0 had been predicted, the acceleration imposed on the particles, in this case millet seed, must have been of the order of 50 times gravity and more.

Other phenomena of interest were also observed when velocity profiles of transporting flow were measured across a radial section of the curved duct at various points which have been discussed in the experimental section.

Following the 'centrifugal' concentrator, the concentrated particulate together with a volume of air sufficient only to transport it effectively is led by a short, smooth duct to a deceleration separator or settling chamber, into which it is projected in a downward direction. Air is extracted from this chamber, which has a large horizontal cross-section, by an entrainment device consisting of a venturi contraction following the concentrator and close to the fan inlet A shaped nozzle is positioned in this contraction and its position and area adjusted to provide flow of entrained air in the deceleration separator whose upward velocity is insufficient to support the particulate in free fall, but is adequate to transport it in the narrow duct between concentrator and separator, the particulate material thus remains in the separator chamber, which can also act as the delivery hopper for the harvester. Due to the unavailability of funds and facility to built the venturi, the venturi contraction was not used for the experiments.

4.8 INSTRUMENTATION

1. Pitot static tube

2. Gauge meter used along with the Pitot static tube (0 - 1 kPa and 0 - 10 kPa)
3. Velocimeter/Hot wire anemometer (velocity 0 to 40 m/s, volumetric flow for maximum circular duct size of 255 cm)
4. Tachometer
5. Balance (maximum-6000 gm)

Other equipment used

1. Transistor inverter- Hitachi HFC-VWS (Variable Speed Drive used to vary the speed of the fan)
2. Fan
3. 10 hp, 400 V, 3 ph, 1400 rpm ac motor used to drive the fan

4.9 CURVED TUBE CONCENTRATOR AND UNIFLOW CYCLONE

For the experiments two concentrators were considered to be used in conjunction with a settling chamber.

1. Curved tube concentrator.
2. Uniflow cyclone.

Both operate on the principle of centrifugal force and are centrifugal separators.

4.9.1 Curved tube concentrator in combination with a settling chamber

A Curved tube concentrator consists of a curved duct as the name implies with two outlet ducts provided at the exit of the concentrator. The first is the air duct, is meant for removing the clean air from the concentrator into the main duct leading into the fan. The second, a smaller duct, called the concentrated seed duct, conveys the concentrated seed along with the 5-10% of underflow air into the settling chamber. A second duct of same size as the concentrator seed duct joins the settling chamber to the main air outlet duct , to convey the 5-10% of underflow air back into the main duct. For improving the concentrated seed flow a pressure drop is needed between the take off point of the concentrated seeds and the point where the underflow air duct is connected back into the main duct. Initially this was to be achieved by a venturi in the main duct with the nozzle attachment to the underflow duct placed at the centre of the venturi. As facilities for building such a venturi was not available, it was decided to just have an entrainment nozzle attached to the underflow duct and place it at the centre of the main duct to achieve the effect of creating a pressure drop. This pressure drop was to be sufficient, to improve the flow of separated seeds into the settling chamber, for separating the seeds from this underflow air in the settling chamber and to draw this underflow air back into the main duct.

4.9.1.1 Equipment Description

The Figure 4.7 as shown in page 92 shows the details of the final testing model developed for the investigation. The test rig provided flow conditions up to 1.05 m^3/s which is much less than the actual expected flow of 12 to 30 m^3/s in a commercial harvester, however flow conditions are similar to those expected in a commercial harvester with the turbulent regimes in the same range of Reynolds number. Therefore it would be a valid experiment.

The equipment consists of a curved duct concentrator, square in section (0.2x0.2 m) with a radius of curvature of 0.4 m and a 360^0 arc of curvature to give a full circle. Thus achieving a length of curvature of 2.5 m (2πx0.4). This was built in a spiral form to facilitate the positioning of the inlet and the outlet ducts at the two ends of the concentrator as shown in Figure 4.7. This spiral curved tube concentrator was then placed in a horizontal plane with its axis parallel to the ground.

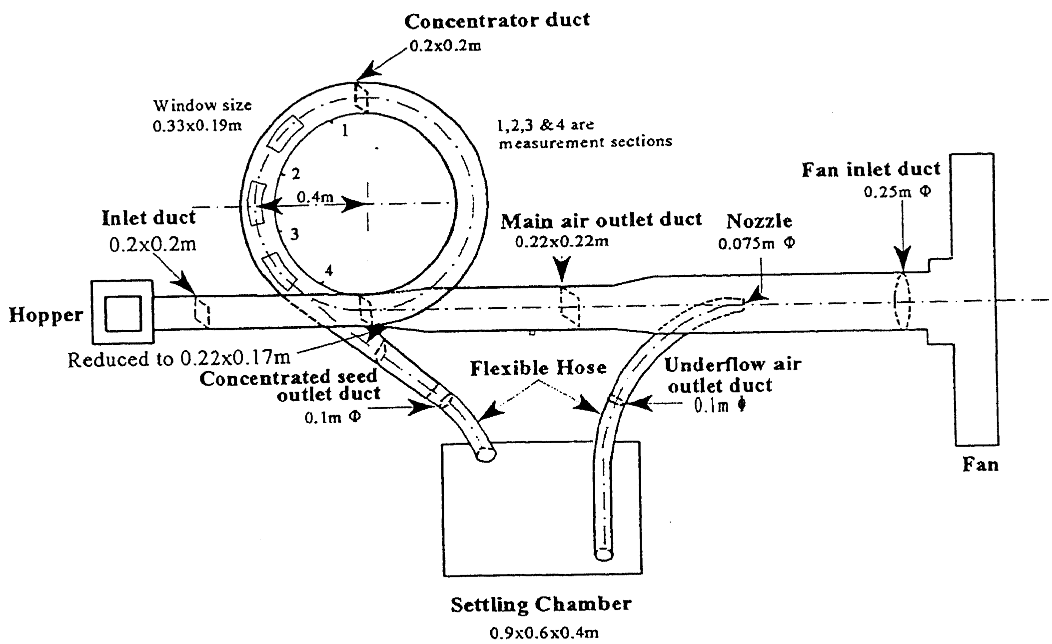


Figure 4.7 Top elevation of Spiral Duct Concentrator system placed in horizontal plane parallel to the ground

The various other ducting system consisted of -

- the inlet duct leading to the curved concentrator duct 0.2x0.2 m in size.

- the main air outlet duct conveying the major flow of air from the concentrator to the fan inlet duct consisted of a duct 0.22x0.22 m in size. This was reduced to 0.22x0.17 m in size to prevent the concentrated seed outlet from being in line with the fan inlet. This also promoted unhampered flow of separated seeds by increasing the concentrated seed outlet by a width of 5 cm. In effect reducing the clogging at the take off point of the concentrated seed duct. This rectangular duct was transformed to a circular duct of diameter 0.1 m over a distance of 0.16 m to be joined to the concentrated seed outlet duct.

- the concentrated seed outlet duct consisted of a circular flexible duct with a diameter of 0.1 m. This duct conveyed the separated seeds from the concentrator to the settling chamber. The concentrated seed outlet duct was taken off tangentially and given a downward incline to promote smooth air-seed flow.

- the underflow air outlet duct was provided to remove the underflow air from the settling chamber back into the main air outlet duct. It consisted of a circular flexible duct with a diameter of 0.1 m. To improve the suction of the seed-air flow through the concentrated seed duct into the settling chamber and of the underflow air through the underflow duct into the main air outlet duct, a nozzle of size 1/3 of the main outlet duct was connected to the end of the underflow duct. The nozzle was aligned to the centre of the main duct, parallel to the air flow and facing the fan inlet. This was done to achieve a pressure drop in the settling chamber, in effect causing good entrainment, resulting in the enhancement of the concentrated air-seed flow between the seed take off point in the concentrator and the nozzle.

- a fan inlet duct circular in cross section with a diameter of 0.25 m connected the main air outlet duct to the fan inlet.

The other components were the fan and the settling chamber-

- a centrifugal fan with radial blades having an inlet diameter of 0.25 m and a rectangular outlet area of 0.046 m^2 (0.25x0.184 m), capable of delivering 1.04 m^3/sec of air at the maximum speed of 1200 rpm. The total pressure developed is 755 Pa with an outlet velocity of 21 m/sec. The fan is directly driven by a motor.
- the settling chamber consisted of a box of size 0.9x0.57x0.32 m.

4.9.1.2 Experimental method

The object of the experiment was to provide test data for assessment of the performance of the curved duct concentrator. To accomplish this two types of tests were considered to be conducted on each experiment section.

1. Flow performance test
2. Particulate collection efficiency test

The first flow performance test was performed to determine the air flow characteristics which would indicate the mode of seed separation and the forces causing the separation. The second, particulate collection efficiency test was performed to determine its collection efficiency at different operating conditions, for example at varying fan speeds and varying seed concentration.

Flow performance test

To perform this test, the seeds were dropped through the system and the seed separation was observed. If good separation was observed then the experiment was proceeded on to the next step, that of studying and analysing the air velocity profile. The air velocity profile in the concentrator was investigated by using a pitot-static tube. The velocity or dynamic pressure (p_v), the static pressure (p_s) and the total pressure (p_t) drop across the curved concentrator duct were measured using the pitot-static tube. The measurements were always taken from the outer edge to the inner edge of the curve by inserting the pitot-static tube through the holes provided and as shown in the respective figures. The air velocity was calculated using the equations given below,

$$p_v = \frac{1}{2} \rho V^2 \tag{4.5}$$

$$V = \sqrt{\frac{2p_v}{\rho}} \tag{4.6}$$

To measure the velocity/dynamic pressure, the pitot-static tube were oriented to face the direction of the flow. The air velocities were then measured from the outer to the inner edge of the curve in finite steps (position 1 to 20 with a gap pf 1 cm between each step). At the least two and sometimes more test runs were repeated while taking the measurements in order to obtain reliable results. A hot wire anemometer was also used to measure the flow rate and the air velocity profile. The static and the total pressure drop across the sections were also measured at times, using the same pitot-static tube. Profiles of the air flow were then plotted on a graph to explain the separation and the forces preventing and assisting the separation.

Particulate collection efficiency

Particulate collection test was conducted only on obtaining good collection of seeds. The seed flow through the concentrator was observed through the windows provided on the inner curve and on one side of the concentrator. To perform this test, a measured amount of seeds are dropped into the inlet duct, using a scoop and they are drawn in through the system. Then the amount of seeds collected in the settling chamber is taken out and weighed. The method used is mentioned below.

Measured amount of seed drawn into the system,

Weight of scoop	$= W_1$ gm.
Weight of scoop + seed, is measured	$= W_2$ gm.
The number of scoops drawn in is taken	$= n_1$.
Therefore amount of seed dropped	$= (W_2\text{-}W_1) \times n_1$ gm.

Then the seed collected in the settling chamber was taken out and measured.

Measured weight of seed + scoop	$= W_3$ gm.
Number of scoops collected	$= n_2$.
Total amount of seed collected	$= (W_3\text{-}W_1) \times n_2$ gm.
Total amount of seeds separated	$= (W_3\text{-}W_1) \times n_2$ gm.
Collection efficiency (%)	$= (W_3\text{-}W_1) \times n_2 \times 100/(W_2\text{-}W_1)n_1$.

The tests were repeated for different flow rates. Varying flow rates were obtained by operating the fan at speeds of 1170 rpm and 880 rpm using a transistor invertor. The tests were also repeated for different concentration of seeds. This was achieved by varying the amount drawn for a particular flow rate. Using the dust loading as a variable, to determine the capacity of a particular size of the concentrator was difficult. This was due to lack of facility to build a mechanical control for the seed flow and the facility to measure the feed concentration in order to provide a uninterrupted and continuous operation of the system.

However for an harvester operating in the field, the seed concentration on the separating system will vary continuously as the picking head encounters seed clumps on the field. Some seed varieties tend to grow in segregated clumps while others are more uniformly dispersed throughout the field.

The settling chamber was cleaned each time before performing the collection efficiency test. The fan was operated at speeds of 1170 rpm and in some case at 880 rpm. When operated at speeds lower than 880 rpm, the air flow velocity was observed inadequate to draw any seeds through the system. This was verified for all the experiments.

The seeds were fed through a square shaped hopper 0.2x0.2 m in size when the concentrator was oriented in a horizontal plane. A hessian bag was fixed at the outlet of the fan to capture the escaped seeds.

Two type of seeds were considered for all the experiments-

- White French Millet, is a relatively heavy, almost spherical shaped seed with a diameter of 0.5 mm and a mass of 0.005 gm/seed.
- Buffel seeds are light fluffy type of seeds with a diameter of 0.3 mm and a mass

of 0.002 gm/seed. The Buffel seeds consists of a solid central core with fluffy hair all around it as can be seen in Figure 4.4.

4.9.1.3 Experimental development

Earlier research had indicated that satisfactory separation could be achieved by a simple curved tube of less than 180^0 arc length. The experimental equipment was developed starting with this simple configuration and leading finally to the unit shown in Figure 4.7 on page 95. The following description gives details of the commissioning of the experimental apparatus. The curved tube concentrator was designed and moved through a series of prototype changes before the final working version was developed. The following sections describe the evolution of the design and the reasons for the design changes. The equipment design evolved through three major stages and three minor stages.

It was in stage 1 that the equipment went through three minor changes. During this phase of the experiment the fan was operated at 1770 rpm. Initially the equipment consisted of a curved tube concentrator with 180° arc of curvature and was set up as shown in Figure 4.8, oriented in a vertical plane. Instead of collecting the escaped seeds at the fan outlet, it was decided to trap them before the fan, thus preventing the seeds from entering the fan. To accomplish this a fine wire gauze and a hopper were fixed in the main air outlet duct ahead of the fan. The equipment was placed vertically, hoping to establish a well distributed air-seed flow across the inlet duct due to gravity, well before reaching the concentrator.

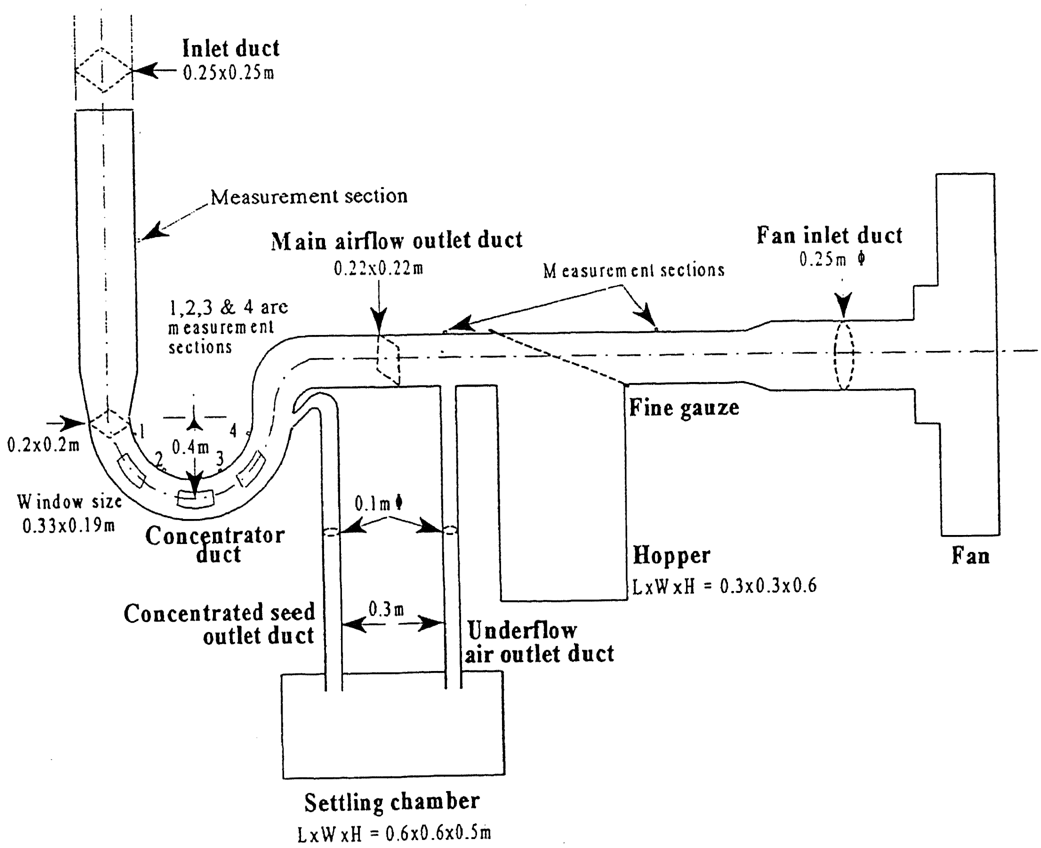


Figure 4.8 Side elevation of Curved Concentrator duct system with gauze

The concentrated seed outlet duct was connected at a 90° angle instead of tangential to the concentrator, due to space constraints and difficulty in building, as both concentrated seed outlet and main air outlet duct were close to one another. The underflow air duct was fixed to the base of the main air outlet duct without a nozzle. The seed collection efficiency test was performed by observation. In this case only 30 to 40 % seeds were observed to concentrate on the outer curve of the concentrator. Those that did not concentrate and were carried beyond the concentrated seed duct and into the main air duct were stopped at the fine screen, blocking the air flow sufficiently to stall the entire separation process, till the seeds on the screen dropped into the hopper below. Some photographs taken of the seed flow through the windows are shown in Appendix 4. It was concluded to observe the seed separation after making some minor changes.

As in the previous experiment the seeds were seen to be spread across the concentrator, a flap was fixed at the concentrated seed outlet hoping to capture more seeds and to prevent the concentrated seed from escaping. Again only 30 to 40% seed separation was observed. In this case also the seeds were observed to block the concentrated seed outlet entry. The flap was observed to block the air flow causing turbulence at the concentrated seed outlet. Eddies were observed causing seeds to flow in a circular path at this section, instead of drawing them into the settling chamber. The gauze was still obstructing free air flow.

The air velocity in the concentrated seed duct was observed to decelerate, indicating that the suction through this duct had to be enhanced. On that account it was concluded to conduct the collection efficiency test after removing the flap, gauze and the hopper.

The collection efficiency test was conducted on the experimental equipment after removing the flap, gauze and the hopper. The test was conducted for the White French Millet and the values are shown in Table 4.1. A collection of around 59% was obtained. Nearly 40% of seeds were still lost through the fan. This could be attributed to three main factors.

The first factor being the orientation of the equipment in a vertical plane. The seeds

Sl.no	Wt. of scoop (gm)	Amount of seed dropped into the system		Amount of seed collected in the settling chamber		Collection efficiency (%)
		(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	
1	429.42	1501.2	642.36	806.95	378.00	58.9
2	429.42	1501.2	645.40	810.00	381.00	59.0
3	429.42	1501.2	643.00	808.54	379.12	59.0

Table 4.10 Collection efficiency test with W.F.Millet and without flap, gauze and hopper

entering the inlet duct were observed to drop straight into the concentrator due to gravity, collide with the outer curved surface of the concentrator and then get scattered across the duct due to the sudden alteration in the flow direction.

The second factor was the angle at which the concentrated seed outlet duct was fixed. This obstructed the smooth flow of the concentrated seed, because the outlet was not constructed at a tangential to the curve of the concentrator in other words parallel to the curved path of the air-seed flow. This caused temporary impediment to the otherwise expected, uninterrupted flow of the seeds and turbulence at the take off point of the concentrated seed duct, forcing some of the separated seeds to be drawn back into the main air flow. The third factor was the inadequate air velocity through the concentrated seed outlet duct to carry the seeds smoothly into the settling chamber.

In view of the above, it was decided to change the orientation of the equipment to a horizontal plane. The take off duct was to be built at a tangential to the outer curvature of the concentrator, with a gradual downwards incline to provide smooth air-seed flow. It was decided to smoothen the existing rough edges. To create an entrainment effect and a pressure drop between the seed take off point and the outlet point of the underflow duct a nozzle with 1/3 the size of the main outlet duct was to be connected at the end of the underflow duct. The nozzle was to be aligned to the centre of the main air outlet duct facing the fan.

In stage 2, as decided the whole equipment was dismantled and placed in a horizontal plane parallel to the ground as shown in the Figure 4.9. A feed hopper was placed above the inlet duct to allow a more controlled feed of the seeds. As proposed the

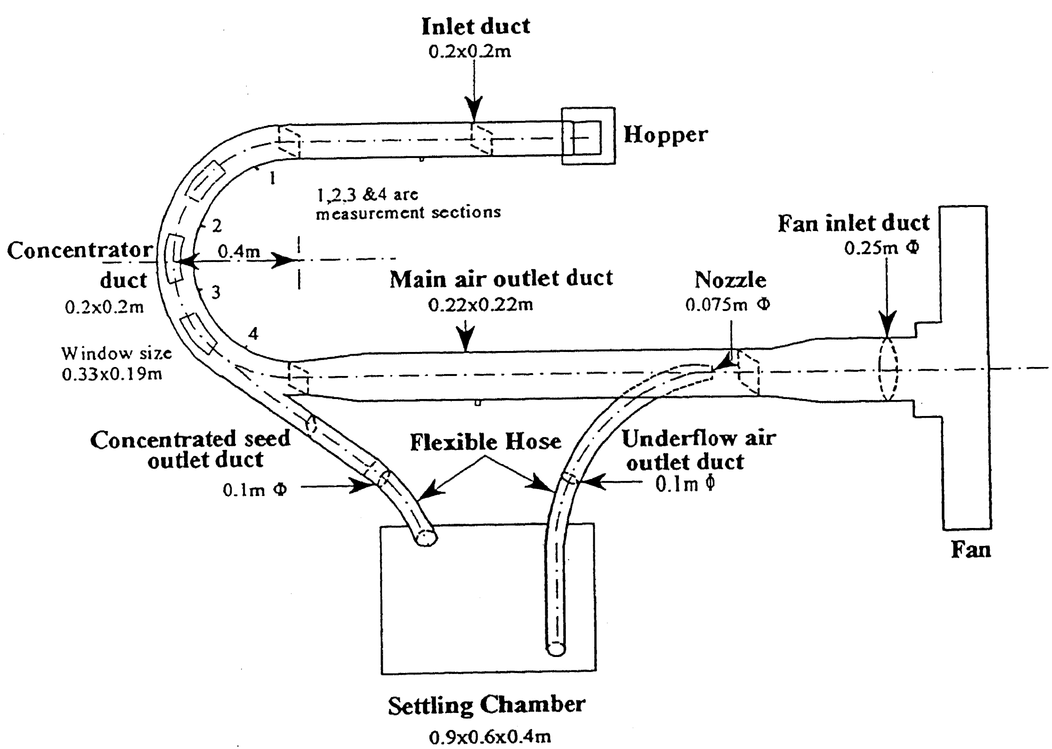


Figure 4.9 Top elevation of Curved Duct Concentrator system placed in horizontal plane parallel to the ground

concentrated seed outlet was built tangentially to the outer curvature of the concentrator, leading into the settling chamber. The underflow duct leading from the settling chamber is coupled with the nozzle and connected parallel to the air flow, at the centre of the main air duct facing the fan in order to achieve a pressure drop in the settling chamber and to get a good entrainment effect. This would in turn improve the concentrated air flow between the seed take off point in the concentrator and the nozzle. The turbulence and blockade observed at the mouth of the seed take off point of the concentrator in the previous experiment would also be eliminated by this technique.

The test for collection efficiency was then performed on this rig to verify its performance for both White French Millet and Buffel grass seeds. The values for both White French Millet and Buffel seeds are shown in Table 4.2 and 4.3. A collection efficiency of 83-84% was obtained. This improvement could be attributed to the fact that air flow had improved in the concentrated seed duct due to its tangential orientation and also because of the smooth downward incline provided to the duct. The nozzle improved the flow velocity through the duct by creating a significant pressure drop across this section. The loss of 16 to 17% of seeds could be contributed to the concentrated seed take off being in direct line with the fan inlet. Hence some of the seeds concentrated on the outer curvature on reaching the outlet region of the concentrator, were being drawn back into the main air outlet duct and into the fan due to the stronger fan suction at this point. The loss could also be due to the considered length of curvature, not being sufficient for all the seeds to concentrate on the outer wall of the curve. Placing the rig horizontally helped in

Sl.no	Wt. of scoop (gm)	Amount of seed dropped into the system		Amount of seed collected in the settling chamber		Collection efficiency (%)
		(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	
1	429.3	6308	2873.6	5832.70	2398.32	83.5
2	429.3	6308	2873.6	5840.00	2405.60	83.7
3	429.3	6308	2873.6	5828.48	2394.08	83.3

Table 4.2 Collection efficiency test conducted after rig was placed in horizontal plane with White French Millet seeds

Sl.no	Wt. of scoop (gm)	Amount of seed dropped into the system		Amount of seed collected in the settling chamber		Collection efficiency (%)
		(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	(Wt. of seed + scoop) x no. of scoops (gm)	Wt. of seed (gm)	
1	429.3	3600	165.6	2713.98	138.20	83.4
2	429.3	3600	165.6	2714.82	139.02	83.9
3	429.3	3600	165.6	2715.24	139.44	84.2

Table 4.3 Collection efficiency test conducted with Buffel grass seeds

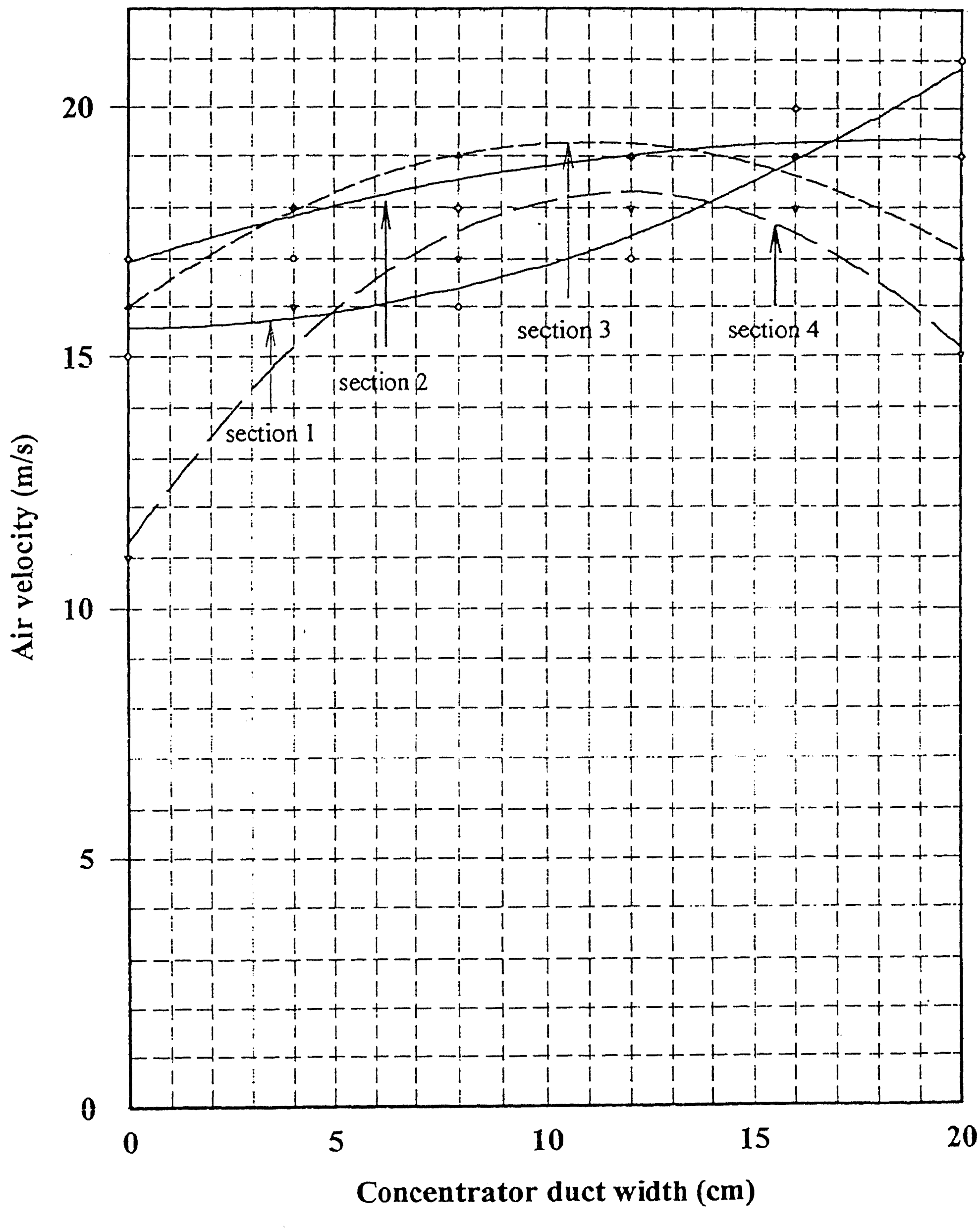


Figure 4.10 Variation of air velocity across the Concentrator duct width when in horizontal position, with improved tangential concentrated seed outlet and an effective pressure drop across the take off point and the main duct

reducing the effect of gravity in the inlet duct, which was a problem in the previous rig. Since the collection efficiency was good, it was decided to proceed to the next step of studying the changes in the air velocity profile of the rig used in this experiment.

To study the air velocity profile the velocity pressure readings were taken along the same points as shown in the Figure 4.9. The plotted air velocity curves are shown in Figure 4.10. In this case readings were taken from outer to inner curvature of the concentrator at points 0-20, at a gap of 4 cm. The comparative study of static and total pressure across the concentrator width is shown in Figure 4.11.

The turbulence in the concentrator is observed to have settled out which is obvious by the smaller range of variation in the air velocity. Looking at the graph, boundary layer effect is obvious at sections 3 and 4. The turbulent flow profile is more developed and evident at these sections. As in most cases in sections 1 and 2 the velocity at the inner curve is observed to be higher which may be due to the air having to travel a shorter distance against boundary layer friction than the air travelling along the outer curve of the concentrator. The increase in velocity at sections 3 and 4 between 12 to 16 cm cross section of duct may be due to the fact that air on outer curve has to accelerate more than the air flowing along the inner curve which actually decelerates to equalise the flow velocities across the duct on its passage into the main air duct which is a straight duct. These points near to the outer curve are actually facing the fan inlet and subjected to a higher air suction, which may be the reason for some of the seeds concentrated on outer curve, being

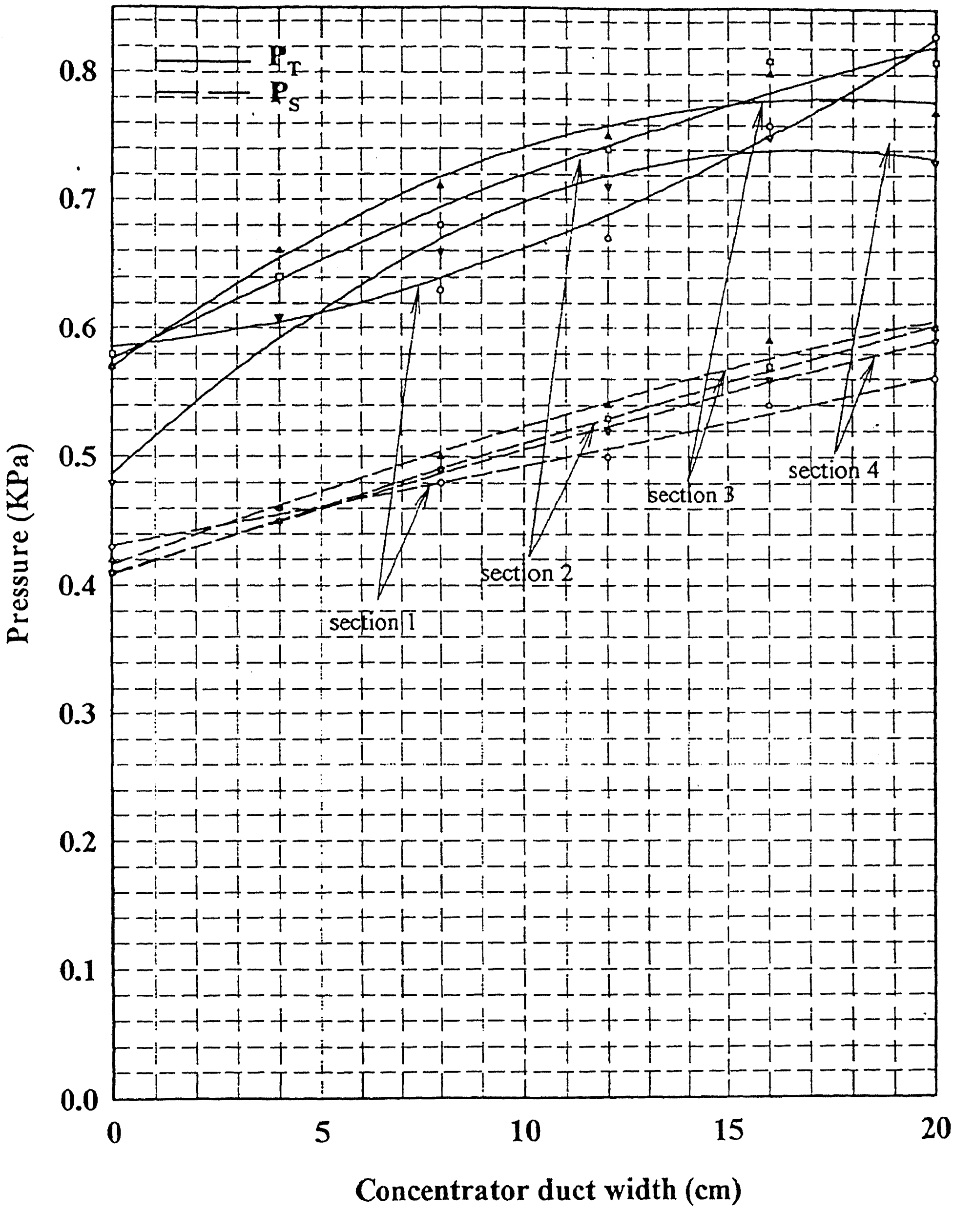


Figure 4.11 Variation of static pressure and total pressure across the Concentrator duct width when in horizontal position

drawn back into the main air outlet duct and carried beyond the take off point of the concentrated seed outlet duct. Overall the flow is observed to be more steady.

Therefore it was concluded that the equipment could be made more efficient by effecting two major changes. The first change by improving the positioning of the concentrated seed outlet by ensuring it was not directly in line with the fan inlet. The second change by increasing the arc of curvature from 180° to 360° i.e a full circle built in a spiral form. This would give more length of travel in the curved section of duct for the seeds to concentrate onto the outer wall of the concentrator. It would also give the opportunity for the air velocity curve to develop into a well developed turbulent flow profile. The final version of the curved duct concentrator was thus established.

Thus the third and the final conclusive design of the curved duct concentrator was established and is as shown in Figure 4.7 in page 92 and as described in section 4.9.1.3. The collection efficiency test was conducted on this experimental model for both White French Millet and the Buffel seeds and the readings are as shown in Table 4.4 and 4.5. The collection efficiency was observed to have improved for White French Millet from 83% to 91% and for Buffel seed from 83% to 97%. This could mean that a greater length of curvature is essential for the centrifugal forces to develop, to promote effective concentration of seeds to the outer curvature of the duct. About 9% of the Millet and 3% of the Buffel seeds are still lost. On observing through the windows, it was quite obvious that this was due to the concentrator seed duct still being in direct line with the fan inlet. This was still forcing some of the

Sl.No.	Wt. of Scoop (gm)	Amount of seed dropped into the system (Wt. of seed + Scoop) X no. of scoops (gm)	Amount of seed dropped into the system Wt. of seed (gm)	Amount of seed collected in the settling chamber (Wt. of seed + Scoop) X no. of scoops) (gm)	Amount of seed collected in the settling chamber Wt. of seed (gm)	Collection efficiency (%)
	W.F.Millet					
1	429.1	2645	1358	2509.05	1222.2	90.0
2	429.1	2645	1358	2507.4	1220.1	89.8
3	429.1	2645	1358	2511.3	1224.0	90.1
	Buffel					
4	429.1	1417	130	1406.4	119.1	91.6
5	429.1	1417	130	1405.8	118.5	91.2
6	429.1	1417	130	1404.6	117.3	90.2

Table 4.4 Collection efficiency test conducted on the Spiral duct Concentrator placed in horizontal plane

Sl.No.	Inlet duct (Area = $0.04 m^2$)		Underflow duct (Area = $0.008 m^2$)		Main air outlet duct (Area = $0.048 m^2$)	
	Q (m^3/s)	V (m/s)	Q (m^3/s)	V (m/s)	Q (m^3/s)	V (m/s)
1	0.61	14	0.078	10.15	0.75	15.0
2	0.68	15	0.080	9.20	0.76	15.1
Average	0.65	15	0.08	9.7	0.76	15
Calculated Q = AxV	0.04 x 15 = 0.6		0.008 x 9.7= 0.08		0.048 x 15 = 0.72	

Table 4.5 Measured and calculated values of volumetric flow rate and air velocity in the inlet, underflow and main air outlet ducts

separated seeds to be drawn back from the outer curve into the main air duct.

It was concluded that the concentrated seed outlet needed to be repositioned away from the fan inlet. The concentrator although fabricated spirally, was built in 2 (180^0 arc) sections and joined together. The concentrator required further improvement in terms of having a steady and continuous slope and the smoothening of the inner surfaces of the duct. The joints and discontinuities caused a lot of turbulence preventing at least 10% of seeds from being concentrated to the outer curve of the concentrator.

Provision of a venturi at the point of nozzle connection within the main air duct would improve the entrainment effect, causing more suction through the concentrated seed outlet duct.

4.9.1.4 Conclusion for the Curved Duct Concentrator

Conclusion arrived at from experimental data obtained for the curved duct concentrator are-

- Curved tube concentrators are very effective in separating the small volume of seed from a large volume of air flow.
- To get maximum efficiency a curved tube with 360° arc of curvature should be substantial to concentrate the seeds. The inner surface of the tube should be smooth.
- The minimum diameter of the duct to handle the expected flow rate should be considered as an important factor. This would provide the minimum radial distance

of travel for seeds to separate onto the outer wall and hence an early separation.

- Smaller radius of curvature would give a better separation as the centrifugal force experienced would be higher, thus assisting separation of the seeds from the air.

The experiment was then proceeded to the second type of separator considered for the investigation. The uniflow cyclone in combination with a settling chamber.

4.9.2 Uniflow cyclone in combination with a settling chamber

The uniflow cyclone consisted of two cylinders, one placed over the other. The upper cylinder acted as the concentrator while the lower cylinder acted as the settling chamber. An inlet duct for drawing in the particulate laden air flow was placed tangentially at the uppermost point of the top cylinder. A centrally placed outlet duct was placed at the lower end of the upper cylinder to remove the clean air to the main duct leading into the fan. An opening was provided along the circumference of the cylinder for the concentrated seeds to be transported into the cylinder below with a 5-10% of underflow air. As in the previous case an underflow air duct was provided in the lower cylinder, leading into the main duct with a nozzle at its re-entry point in the main duct. The underflow duct in this case was mainly for drawing the underflow air back into the main duct thus reducing the agitation of seeds in the settling chamber.

4.9.2.1 Equipment Description

For this experiment two sizes of Uniflow cyclones were developed to investigate the minimum size required for efficient separation of seeds for the provided flow rate. As mentioned earlier the test rig provided flow up to 1.05 m^3/s which is much less than actual expected flow of 12 to 30 m^3/s in a commercial harvester, however flow conditions are similar to those expected in a commercial harvester with the turbulent regimes in the same range of Reynolds number.

The Figures 4.12 and 4.13 in pages 113 and 114 respectively show the two final models developed of the Uniflow cyclones, for the experimentation. The design was partially based on the design methods used for a reverse flow cyclone.

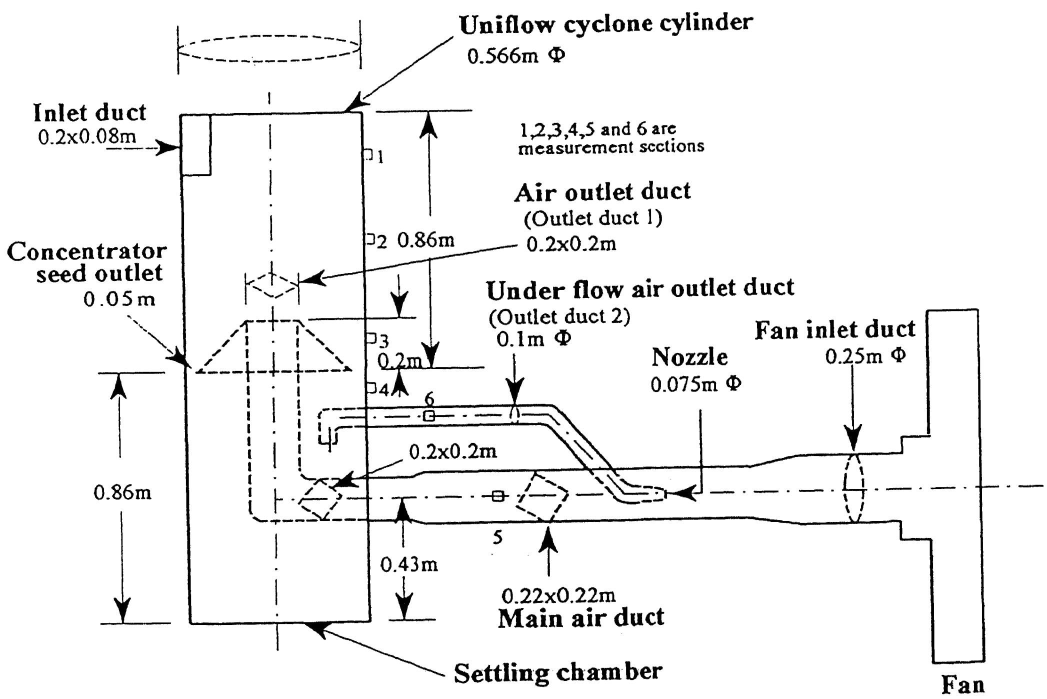


Figure 4.12 Uniflow cyclone model 1 and settling chamber with inlet duct reduced to 0.2x0.08 m and underflow air duct fixed inside the settling chamber at an angle to the air outlet duct

The Model 1 consisted of two cylinders with a diameter of 0.566 m and a height of 0.86 m each, placed one on top of the other and joined together. The top cylinder was to perform as the uniflow cyclone and the lower cylinder as the seed collector or the settling chamber. A window (0.3x0.3 m) was provided on the lower side of both the upper and lower cylinders each, for viewing the seed flow and separation.

The model 2 as shown in Figure 4.13 consisted of two smaller cylinders with a diameter of 0.365 m and a height of 0.59 m each, placed one on top of the other and joined together. The photographs of this model is shown in Appendix 6. The top cylinder was to perform as the uniflow cyclone and the lower cylinder as the seed collector or the settling chamber. A window (0.3x0.3 m) was provided on the lower

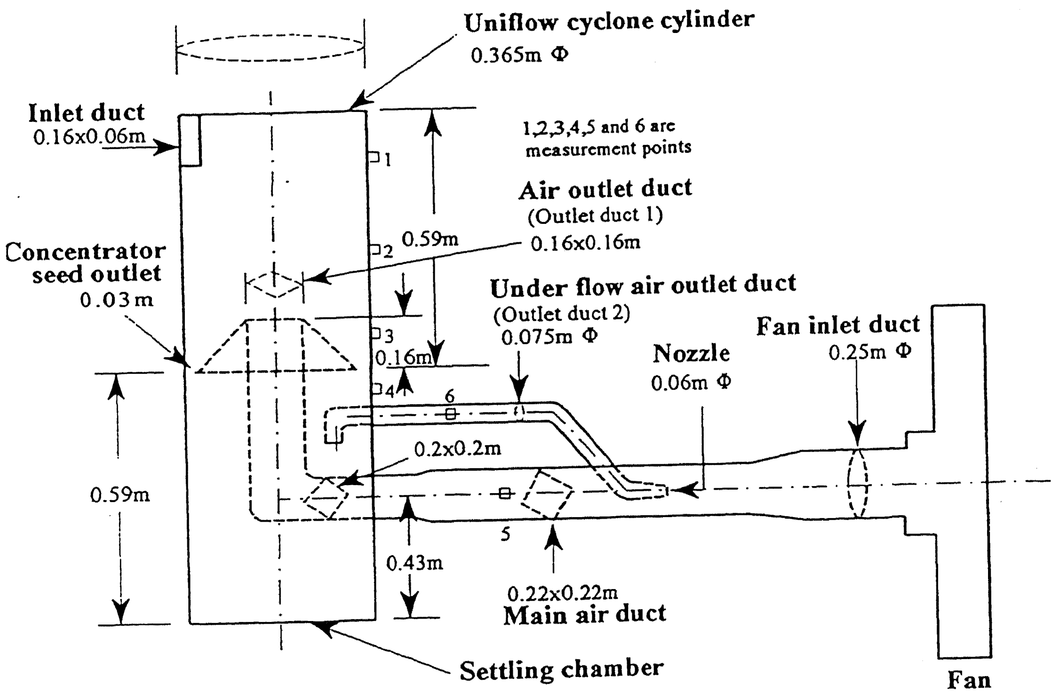


Figure 4.13 Uniflow cyclone model 2 and settling chamber

side of both the upper and lower cylinders each, for viewing the seed flow and separation.

The various other ducting system consisted of:

- for model 1 the air inlet consisted of a rectangular duct 0.2x0.08 m in size with a length of 0.75 m and for model 2 it consisted of a rectangular duct 0.16x0.06 m in size with a length of 1 m. The duct was connected tangentially to the top edge of the upper drum to draw the air stream with the seeds tangentially into the main cylinder body thus providing the desired vortex.
- for both models 1 and 2 the main air outlet duct consisted of a square duct 0.2x0.2 m in size. It was fixed centrally at the bottom of the upper cylinder, protruding 0.2 m into the main cylinder body of the cyclone as shown in the figures. The duct is to draw out the major portion of the clean air from the cylinder and convey it to the main duct of 0.22x0.22 m square section, which is then connected to a circular fan inlet duct of diameter 0.25 m leading into the fan, as in the previous system.
- the underflow air duct consisted of a circular flexible duct with a diameter of 0.1 m for the model 1 and of diameter 0.075 m for the model 2. The underflow air duct is connected to a nozzle which is placed at the centre of the main duct. The size of the nozzle is taken to be approximately 1/3 the diameter of the main duct, ie, 0.075 m for model 1 and 0.06 m for the model 2. It was given a smooth shape to prevent any disturbance to the air flow in the main duct. The nozzle helped to develop enough pressure drop between the seed take off point and the nozzle so as to draw out the concentrated seeds into the settling chamber and the underflow air from the settling chamber back into the main duct. Thus helping to reduce the

turbulence which would be caused if the air is trapped in the lower cylinder when such an outlet is not provided, causing the seeds to settle out from the underflow air.

- the passage for the conveyance of the air-seed from the cyclone to the settling chamber was assembled by attaching a conical hat shaped section to the rim of the outlet duct inside the main cylinder body of the cyclone, leaving an opening of 0.05m for model 1 and 0.03 m for model 2 between the rim of the cone and the wall of the cyclone all along the cylinder circumference.

The other components were the fan and the settling chamber-

- the same centrifugal fan used in the previous experiment was coupled to the Uniflow system and operated at speeds of 1170 and 880 rpm to obtain about the same flow rates as for the curved tube separator system.
- the settling chamber was provided to separate the seeds from the underflow air and to convey the underflow air back into the main air duct. As mentioned earlier the lower cylinder functioned as the settling chamber for both model 1 and 2.

4.9.2.2 Experimental method

The object of the experiment was to provide test data for assessment of the performance of the Uniflow cyclone concentrator. To accomplish this three types of tests were considered to be conducted on each experiment section.

1. Flow performance test
2. Particulate collection efficiency test
3. Concentration test

The first flow performance test was performed to determine the air flow characteristics which would indicate the mode of seed separation and the forces causing the separation.

The second particulate collection efficiency test was performed to determine its collection efficiency at different operating conditions, for example at varying fan speeds and varying seed concentration.

The third test particle concentration test was conducted to identify the maximum concentration of particulate air flow that a particular size of cyclone concentrator, could contend with.

Flow performance test

To perform this test, the seeds were dropped through the system and the seed separation was observed. If good separation was observed then the experiment was proceeded on to the next step, that of studying and analysing the air velocity profile. The air velocity profile in the concentrator was investigated by using a pitot-static tube. The velocity or dynamic pressure (p_v), the static pressure (p_s) and the total pressure (p_t) drop across the Uniflow cyclone concentrator were measured using the pitot-static tube. The measurements were always taken from the outer wall to the centre point of the cylinder by inserting the pitot-static tube through the holes provided and as shown in the respective figures. In some cases both the downward flow and the tangential flow of air were examined. But mostly only the tangential air flow was studied as it was the main air flow conveying and assisting in the separation of the seeds.

The measurements were taken across three sections along the height of the cyclone cylinder and one in the settling chamber cylinder. For model 1 these points are, as shown in the Fig 4.12.

- Section 1 at a distance 15 cm from top of the cylinder.
- Section 2 at a distance 29 cm from section - 1.
- Section 3 at a distance 30 cm from section - 2.
- Section 4 at a distance 22 cm from section - 3.

For model 2 these points are, as shown in Fig 4.13.

- section 1 at distance 23 cm from top of the drum.
- section 2 at distance 13.5 cm from section 1.
- section 3 at distance 17 cm from section 2.
- section 4 at distance 17 cm from section 3.

Pitot-static tube was used to measure the dynamic or velocity pressure and to calculate the velocity of the air flow across the four sections of the cylinder. To study the downward flow of air, the pitot-static tube was placed with the open end pointed upwards facing the downward flow of air and measurements for velocity or dynamic pressure and static pressure were taken across each section along the diameter of the cylinder from points 28 to 6 cams starting from wall to the centre of the cylinder for the model 1 and from points 18 to 0 cams for model 2. The points in between 0 to 5 cams at the centre of the cylinder for model 1 could not be measured as pitot-static tube was not long enough. As mentioned earlier in section 3.2 of chapter 3 the law of vortex is not applicable at a radius of half that of the exit duct hence, this problem did not really affect the air velocity profile. Readings were repeated to verify the measured values.

To study the tangential air flow, the pitot-static tube was placed parallel to the cylinder wall and the open end facing the tangential air flow. The velocity pressure and static pressure readings were taken across each section along the diameter of the cylinder from points 28 to 6 cams starting from the wall to the centre of the cylinder for model 1 and from points 18 to 0 cams for model 2. Readings were repeated to verify the measured values.

Velocity pressure and air flow velocity measurements were also taken across the inlet duct, the air outlet duct and the underflow duct with a pitot-static tube and a hot wire anemometer across the sections shown in the Figure 4.12 and 4.13. The volumetric flow rate in all these ducts were also measured using the hot wire anemometer.

The air flow profiles were then plotted on a graph to explain the separation and the forces preventing and assisting the separation.

Particulate collection efficiency

Particulate collection test was conducted only on obtaining good collection of seeds.

The seed flow through the concentrator was observed through the windows provided on the side of the concentrator. The test conducted is same as explained in section 4.9.1.2.

Particle concentration test

The particle concentration test to be effective required a continuous flow of varying weight of seeds for the same volumetric flow rate, ie the concentration of seeds. This required a good hopper and a mechanically controlled feeder. But due to limited facility the test was done using a small hopper fixed above the inlet section of the inlet duct, with a sliding door and another gate below the sliding gate to control the seed flow or the concentration. The hopper of size L x W x H (0.3x0.06x0.3 m)was fixed over the inlet duct about 0.15 m away from the duct opening. This was done to provide free suction of air along with the seeds into the cylinder.

Fan was operated at both 880 and 1170 rpm. Each time the hopper was filled up with a measured quantity of seeds. Then the upper sliding gate was used to fix the opening area below the hopper. The gate opened across the 0.3 m length of the hopper. The second gate is then opened once the fan was operated. This helped in drawing in of the seeds along with the air flow into the cyclone cylinder. The gate is opened for certain fixed time, which is measured. If no seeds were observed to be lost, the seeds collected in the chamber were weighed and measured and was taken to be same as the amount of seeds drawn into the system. This was confirmed by weighing the seeds left in the hopper. Hence weight of seeds drawn per second into the system was calculated. The air flow rate through the inlet was measured. The concentration of the particulate flow was calculated by dividing the weight of seeds drawn per second by the volumetric air flow rate.

4.9.2.3 Experimental development of Uniflow cyclone model 1

This section deals with the evolvement of the model through the various stages to the conclusion of the final model. Model 1 progressed through 4 stages before culminating to the final model. During the first three stages the inlet duct was a square duct 0.2 x0.2 m in size.

In stage one of the experiment the fan was operated at 1770 rpm. At this point, an observation needs to be made. Due to a error in comprehending the design while assembling the duct system, the underflow duct was actually connected to one side of the air outlet duct, instead of being connected inside the settling chamber. As the inaccuracy in the assembling had occurred within the chamber, it was noticed only after the tests were conducted. The whole experiment was thus performed with no underflow air duct. But instead an additional smaller air outlet duct connected to the larger air outlet duct. Theoretically this parallel duct should have no influence on the performance of the cyclone unit.

The collection efficiency test was performed with both variety of seeds and the values obtained are shown in table 4.6. A collection efficiency greater than 97% was achieved for both varieties of seeds. From the result obtained it could be deduced

Sl. no	Type of seed	Wt. of scoop (gm)	Amount of seed dropped into the system		Amount of seed collected in separator		Collection efficiency (%)
			(Wt. of seed + scoop) x no. of scoops	Wt. of seed (gm)	(Wt. of seed + scoop) x no. of scoops	Wt. of seed (gm)	
1	W.F.M	429.2	5338.9	2334.5	5273.1	2268.7	97.0
2	W.F.M	429.2	5338.9	2334.5	5320.0	2315.6	99.0
3	W.F.M	429.2	5338.9	2334.5	5306.0	2301.6	98.7
1	Buffel	429.2	3161.2	156.8	3161.2	156.8	100
2	Buffel	429.2	3161.2	156.8	3160.5	156.8	100
3	Buffel	429.2	3161.2	156.8	3161.2	156.8	100

Table 4.6 Collection efficiency test readings on Uniflow cyclone model 1 with underflow duct fixed into air outlet duct

that cyclone separator provided a greater length of curvature, for the seeds to develop the essential centrifugal force needed to move outwards towards the cylinder wall and get separated from the main airflow. On observing through the windows the seeds could be seen to be travelling along the wall of the cylinder in an angular curved downward path and being carried off straight into the settling chamber below with a small amount of underflow air. The seeds in the settling chamber were being gently stirred in a circular motion at the bottom of the settling chamber. Although the disturbance was very slight it could be assumed now, that it was caused by the underflow air which was not being drawn back into the main duct as the underflow duct was fixed to the air duct instead of being fixed into the settling chamber. But this did not effect the separation or the settling of seeds in the settling chamber.

As separation was good it was decided to proceed and verify the optimum speed for suction and separation by operating the fan at different speeds and also examine the air flow profile within the cylinder.

A test was then performed to specify the minimum fan speed required for suction and separation of seed. The seeds were not weighed for this part of the experiment as the objective was to establish the lowest fan speed for effective separation. The fan was operated at different speeds of 800 rpm, 560 rpm and 280 rpm.

At 800 rpm nearly 100% efficiency was observed. Whilst at 560 rpm the seeds that were drawn kept rotating around the cone and took about 30-40 seconds to settle down into the settling chamber. At 280 rpm the seeds were not being drawn into the system and just lingered onto the lower surface of the inlet duct and was gently being moved around.

Thus it was concluded that 800 rpm was the minimum and the optimum speed for suction and separation for this particular model. Any speed greater would be a waste and the lowest possible speed that gives the maximum separation can be considered as the ideal speed for the particular unit.

At 560 rpm the centrifugal force was not sufficient to create a vortex in the cylinder so the seeds that were drawn into the cyclone cylinder fell straight down onto the cone due to gravity. Across the cone due to reduction of area, the increased velocity of the air kept the seeds rotating around the cone, till it lost its momentum and settled down into the settling chamber. Hence it was obvious that any speed less than 800 rpm was not applicable in operating this system. At these low fan speeds the cyclone was noticed to function as a settling chamber. However the rate of separation of the seeds in this mode of operation is too slow for this to be of any practical use. It was proposed to operate the system at 1170 rpm and study the downward and tangential air flow profiles inside the cyclone cylinder.

Finally the flow performance test was conducted on the experimental model. The downward and the tangential air flow within the cyclone cylinder and the settling chamber were examined. The plotted curve of the downward air flow are shown in Figure 4.14a and 4.14b respectively. The plotted curve for the tangential air flow velocities versus the cyclone cylinder diameter are shown in Figure 4.15.

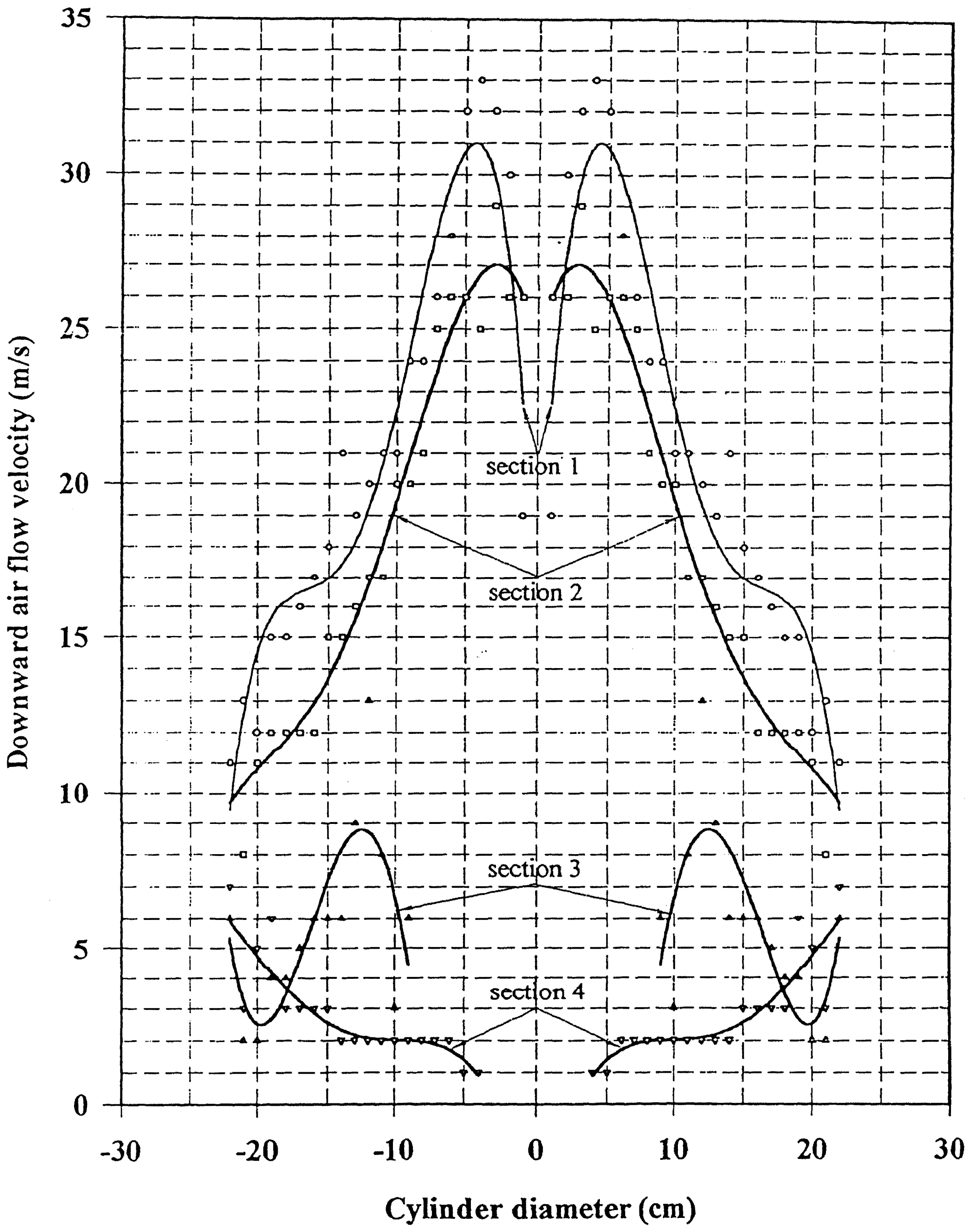


Figure 4.14a Uniflow cyclone model 1. Downward air flow velocity profile across the cyclone cylinder and the settling chamber diameter, when underflow air duct is fixed onto the air outlet duct

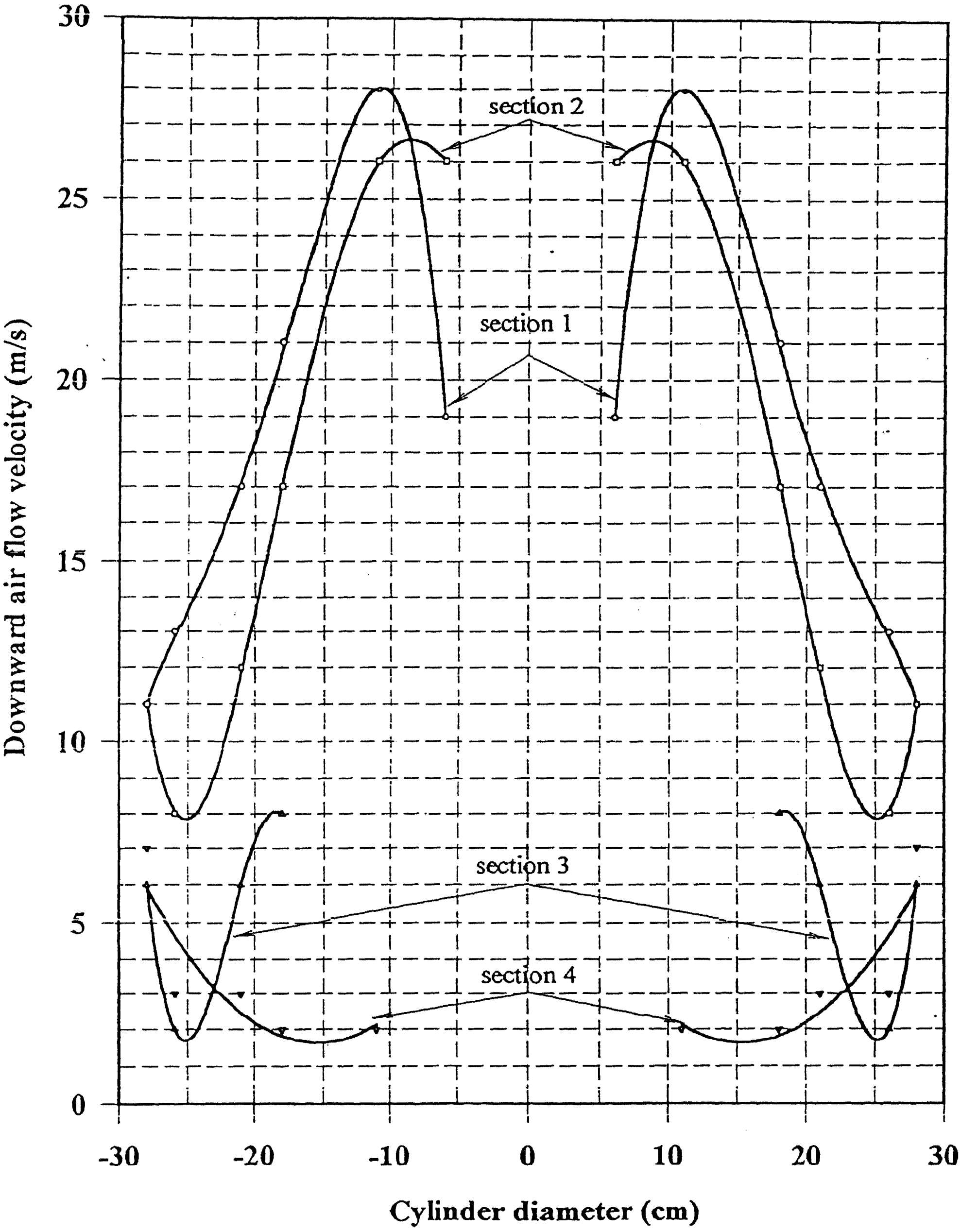


Figure 4.14b Uniflow cyclone model 1. Downward air flow velocity profile across the cyclone cylinder and the settling chamber diameter, when underflow air duct is fixed into the air outlet duct

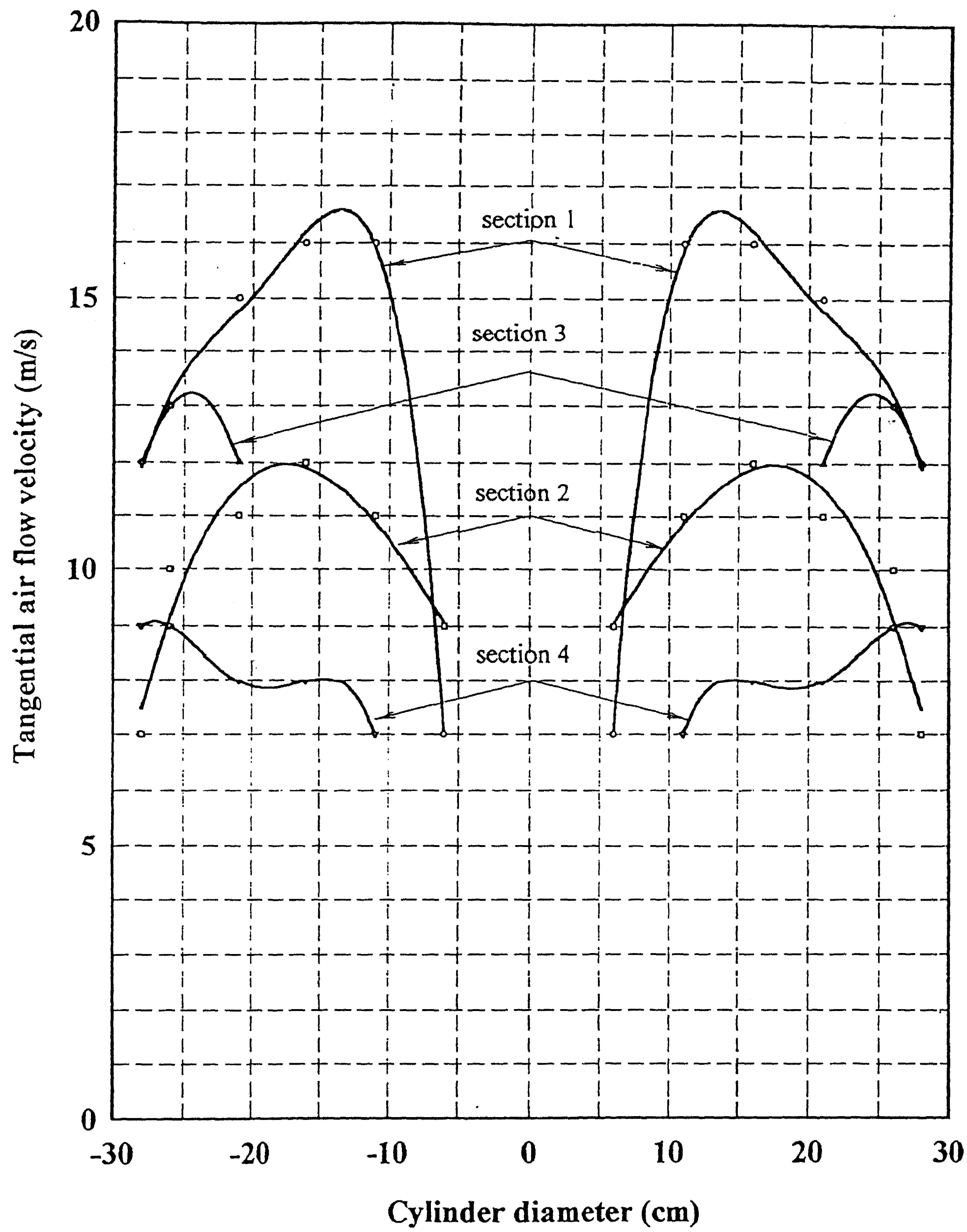


Figure 4.15 Uniflow cyclone model 1. Tangential air flow velocity profile across the cyclone cylinder and the settling chamber diameter, when underflow air duct is fixed into the air outlet duct

Velocity pressure and air flow velocity measurements were also taken across the inlet duct, the air outlet duct and the underflow duct with a pitot-static tube and a hot wire anemometer across the sections shown in the figure. The volumetric flow rate in all these ducts were also measured using the hot wire anemometer. The values are given in Tables 4.7, 4.8, 4.9, and the plotted curves are shown in Figures 4.16, 4.17 and 4.18 respectively. Graphs were plotted against velocity versus duct width.

In Figures 4.14a and 4.14b, the air flow path is that of a compound vortex, with the curve of a forced vortex at the centre and that of a free vortex surrounding it. This maybe due to boundary layer affect. Then the velocity is seen to increase steadily and then reduce towards the central part of the duct. This profile is typical of a compound vortex. Since it is a compound vortex flow the velocity at a common radius 'R' should be same for both vortices. This was found to be true for all the experiments conducted with Uniflow cyclone.

Sl.no	Points considered across the duct (cm)	INLET DUCT ($0.2 \times 0.2 = 0.04 m^2$)					
		Volumetric flow rate (m^3/s)	Velocity of air using velocime-ter (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)	p_{static} (kPa)
1	0	0.59	2	0.06	10	0.135
2	5	0.59	12	0.09	12	0.144
3	10	0.59	15	0.11	14	0.144
4	15	0.59	14	0.10	13	0.145
5	20	0.59	11	0.05	9	0.142
Average		0.59	11	-	12	-
Q =AxV m^3/s		0.04 x 11 = 0.44		0.04 x 12 = 0.48		Q_{av}=0.46

Table 4.7 Measurement of dynamic and static pressures, inlet velocity, volumetric flow rate at the inlet duct using a pitot-static tube and hot wire anemometer (Inlet duct size of 0.2x0.2 m)

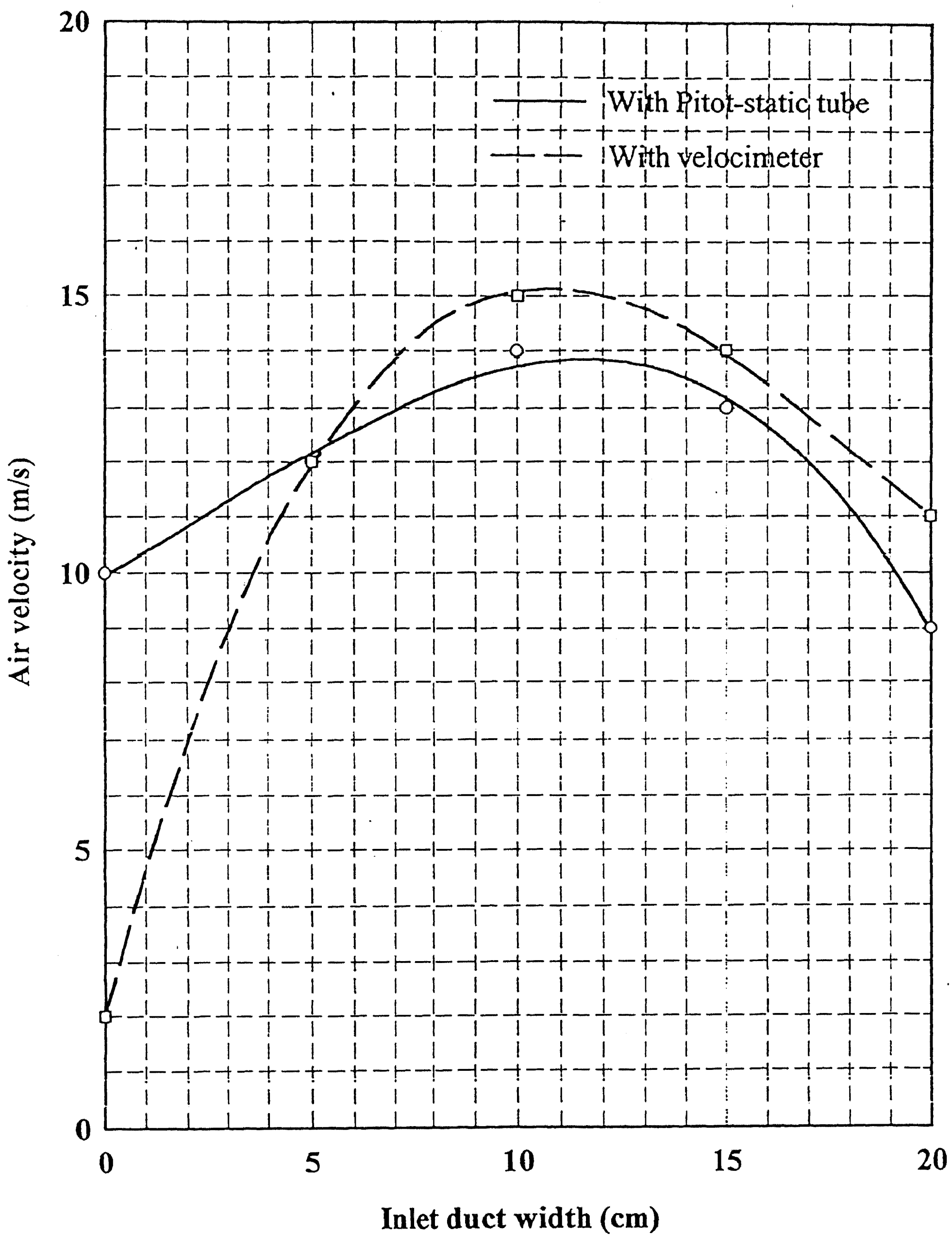


Figure 4.16 Uniflow cyclone model 1. Air velocity profile across the inlet duct width when underflow air duct is fixed into the air outlet duct

Sl.no	Points considered across the duct (cm)	OUTLET DUCT ($0.2 \times 0.2 = 0.04 m^2$)				
		Volumetric flow rate (m^3/s)	Velocity of air using velocime-ter (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)	p_{static} (kPa)
1	0	0.57	23	0.33	23	1.10
2	5	0.57	6	0.024	6	1.10
3	10	0.57	7	0.02	6	1.08
4	15	0.57	10	0.078	11	1.55
5	20	0.57	22	0.23	20	0.96
Average		0.57	14	-	13	-
Q =AxV m^3/s		$0.04 \times 14 = 0.56$		$0.04 \times 13 = 0.52$		Q_{av}=0.54

Table 4.8 Measurement of dynamic and static pressures, volumetric flow rate, velocity at the main air outlet duct (Outlet duct size of 0.2x0.2 m)

Sl.no	Points considered across the duct (cm)	UNDERFLOW DUCT (Φ= 0.1m, Area = $0.008 m^2$)			
		Volumetric flow rate (m^3/s)	Velocity of air using velocimeter (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)
1	0	-	0.71	-	-
2	2	-	1.09	-	-
3	4	-	2.27	-	-
4	5	0.021	2.86	0.004	2.6
5	6	-	2.74	-	-
6	8	-	1.84	-	-
7	10	-	1.16	-	-
Average		0.021	1.81	-	2.6
Q =AxV m^3/s		$0.008 \times 2.6 = 0.021$			

Table 4.9 Measurement of volumetric flow rate and air velocity in the circular underflow duct having diameter 0.1 m

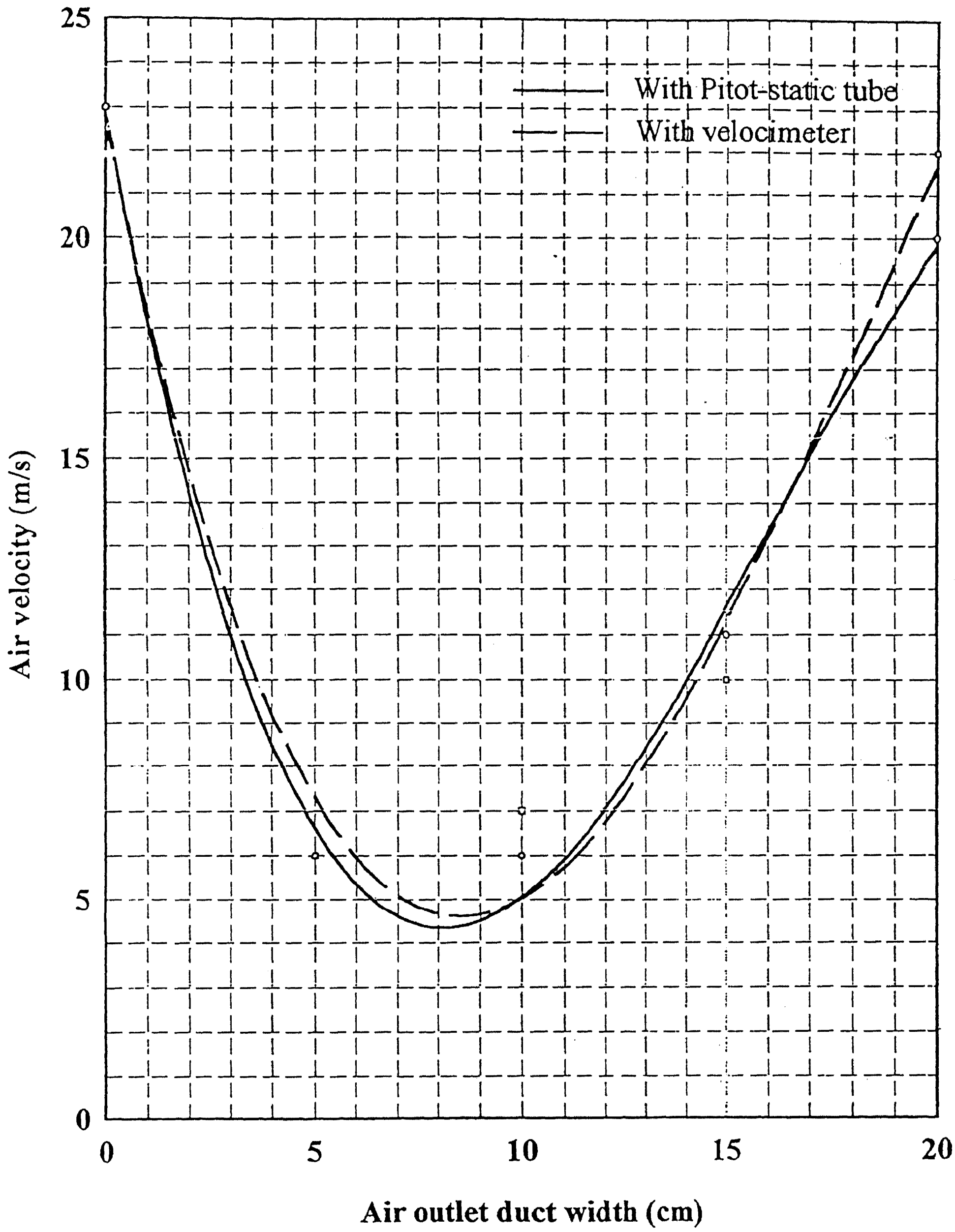


Figure 4.17 Uniflow cyclone model 1. Air velocity profile across the air outlet duct width when the underflow air duct is fixed into the air outlet duct

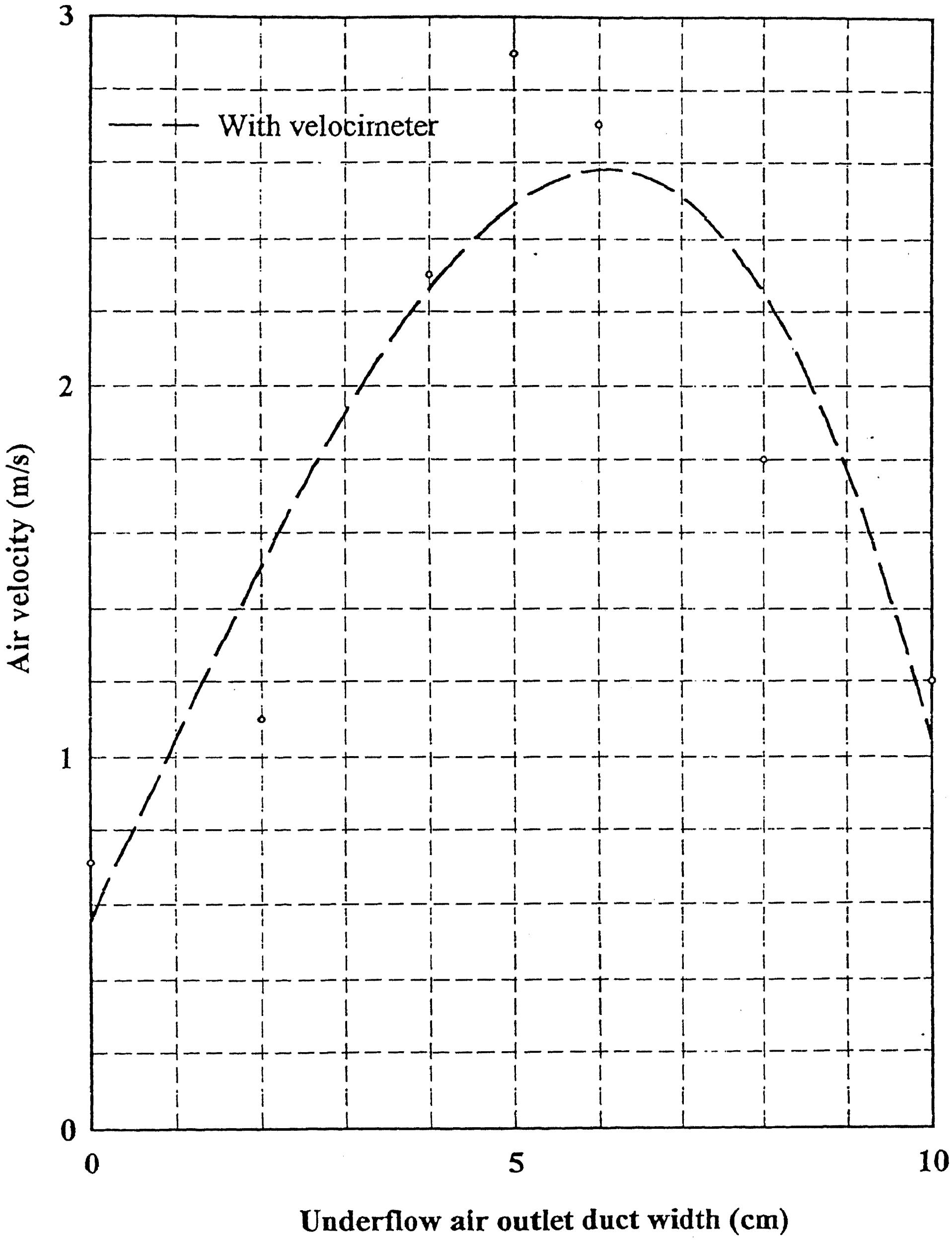


Figure 4.18 Uniflow cyclone model 1. Air velocity profile across the underflow air outlet duct width when the underflow air duct is fixed into the air outlet duct

The flow can be confirmed to be a compound vortex flow by considering values taken from the curve referred to Section 1 of Figure 4.15 and applying them to the theory of a compound vortex mentioned below.

Consider a forced vortex, velocity V at the common radius R where the two curves meet is given by the equation,

$$V = \omega R \qquad (4.7)$$

$$V = \frac{C}{R} \qquad (4.8)$$

whereas for free vortex, velocity at the common radius R is given by the equation, 4.8 (from equation Vr = C, where 'C' is a constant known as the strength of vortex at any radius 'r'). Therefore, common radius at which both the velocities will be same is given by,

$$\omega R = \frac{C}{R} \qquad (3.9a)$$

$$\text{i.e,} \quad R = \sqrt{\frac{C}{\omega}} \qquad (3.9b)$$

Applying the values obtained from the Figure 4.15, at Section 1,

1. at R = 12 cm

   ω = V/R = 16.5/12 = 1.4 for forced vortex.

2. at R = 17 cm

   V = C/R

   i.e, C = VR = 16.5x17 = 280.5 for free vortex.

Now, substituting the obtained values above in equation 3.9b,

$$R = \sqrt{(C/\omega)} = \sqrt{(280.5/1.4)} = 14.2 \text{ cms}$$

This can be verified from the curve at section 1 in Figure 4.15.

Moving down along the cylinder axis the peak, i.e the velocity at the common radius is observed to move outwards towards the wall of the cylinder implying that more and more of the air at the central vortex was converted to a forced vortex surrounded by smaller free vortex.

Figure 4.16 shows a well developed turbulent air flow curve across the inlet duct width with a mean velocity of V = 12 m/s, which is same as the tangential velocity of air inside the cylinder at section 1, as would be expected. Figure 4.17 shows an inverted turbulent curve across the air outlet duct width with higher velocities at the wall of the duct and less velocity at centre of the duct, which may be due to the fact that the readings were taken closer to the bend of the air outlet duct where the turbulence has not yet fully developed, as shown in the Figure 4.17. The mean air velocity was found to be, V = 13-14 m/s. Figure 4.18, shows a well developed turbulent air flow curve across the underflow air duct width with a mean air velocity of V = 1.8 m/s.

Considering Tables 4.7, 4.8 and 4.9, the volumetric flow rate at inlet should be same as the sum of the flow rates in the two outlet ducts. Therefore,

$$Q_{in} = Q_{out.1} + Q_{out.2}$$

Substituting experimental data,

$0.59 = 0.57 + 0.02 = 0.59\ m^3/s$.

So the air drawn in is being drawn out from the cyclone system which attributes to the absence of turbulence in both the cyclone and settling chamber cylinder. The a underflow air flow 0.02 m^3/s is 3.4% of the total inlet air flow. This is what was expected as an underflow air between 5-10% or less, of the total inlet airflow (but in this case the underflow duct was fixed to the side of the air outlet duct).

Although a error was committed during the assembling, the cyclone was observed to perform nearing 100% efficiency. The underflow duct which was connected to the main air duct did not seem to contribute in any way to the airflow. Air in the settling chamber was seen to slightly stir the seeds around in a circular motion and did not have much velocity. It was assumed that air was escaping through the air outlet duct by turning back into the cyclone cylinder. This air movement was not strong enough to lift any of the seeds back into the cyclone cylinder. It was decided to perform the same tests on the model, after fixing the underflow duct back into the settling chamber.

In stage two, the experimental model was set up with the underflow duct fixed into the settling chamber as it was intended, to develop a desirable pressure drop in the section. Photographs of this model are shown in Appendix 5.

The collection efficiency was evaluated by observation. Then the air velocity profile was analysed to study the variation in the air flow profile from the previous experiment. This time only the tangential flow was investigated, as it is the main air flow effecting the separation. The plotted curve is shown in Figure 4.19.

On conducting the collection efficiency test, higher turbulence was observed in the settling chamber. Some seeds in the cyclone cylinder were seen to continue revolving around the air outlet duct cone and took a while to settle out and drop into the settling chamber. This could be because of higher velocity around the cone. This higher velocity at the cone was confirmed by the values at section 3 in Figure 4.19. Also it may be because the air was not being effectively drawn from the settling chamber by the underflow duct, and so it was forced to find its way back into the cyclone cylinder and escape through the air outlet duct. This prevented some of the seeds revolving around the cone from settling out, and caused them to remain afloat for a short duration and finally escape with the air into the air outlet duct.

In the settling chamber the seeds were observed to be churned around at a high velocity and did not settle down. The only conclusion reached could be that more air was coming into the settling chamber as underflow air and which was not being drawn back by the underflow duct causing the air to keep rotating inside the settling chamber. The restriction in area caused by the placement of the air outlet duct and the underflow duct inside the chamber may have helped in increasing the velocity of the air in the chamber. Also fixing the entrance of the underflow duct directly above the bend of the air outlet duct seemed to obstruct efficient and free suction of underflow air through this duct. On considering the Figure 4.19, the profile of a compound vortex is quite obvious. On comparing with the previous Figure 4.15, the observation made is that the overall air velocity of each stream in each section and

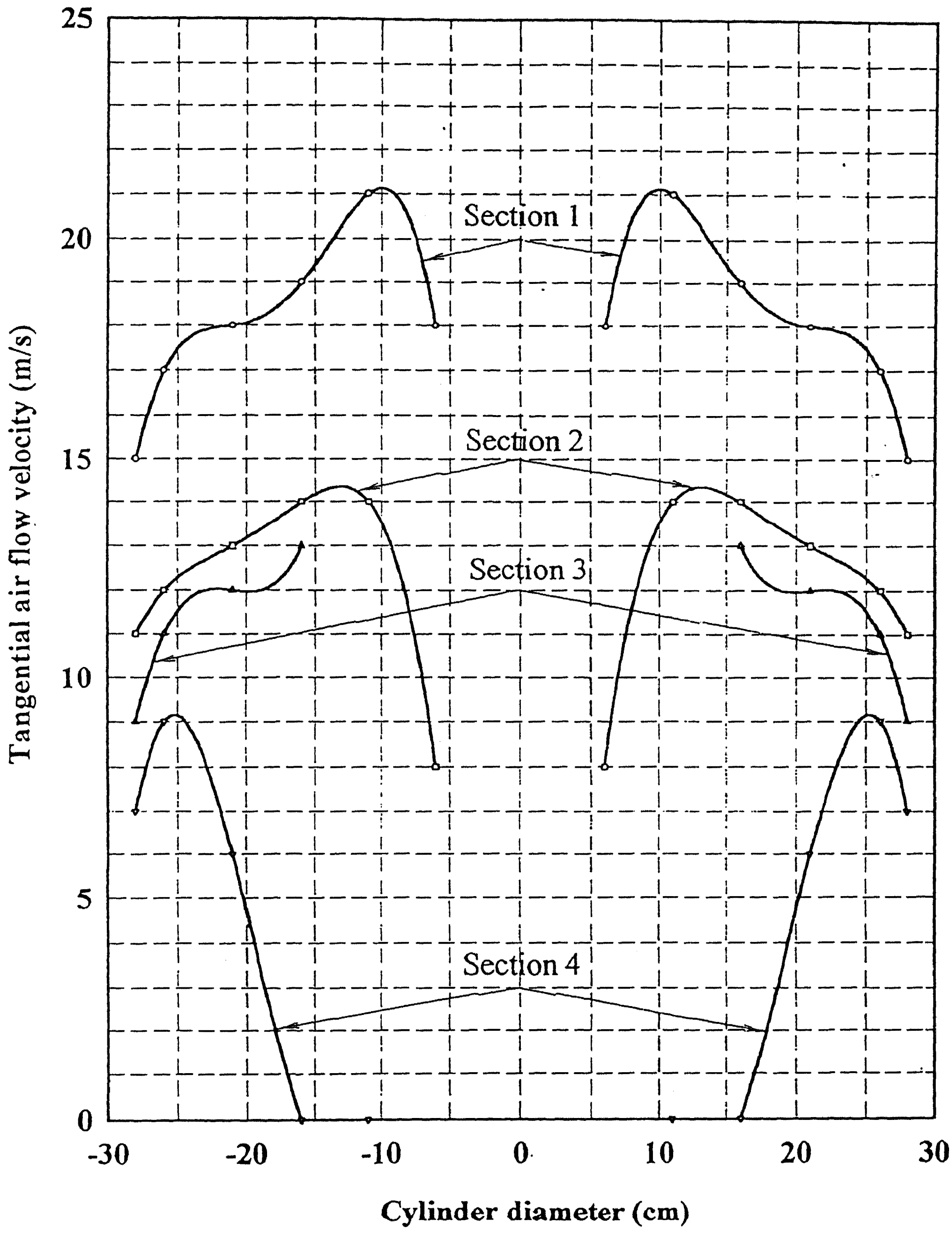


Figure 4.19 Uniflow cyclone model 1. Tangential air flow velocity profile across the cyclone cylinder and the settling chamber with the underflow air duct fixed inside the settling chamber

at each of the points considered in these sections had increased slightly, but the profile was still the same. The common velocity at radius 'R' for the two vortices was observed to be moving outward towards the cylinder wall for each of the different streams considered along the height or axis of the cylinder.

Since the underflow duct did not seem to be effectively removing the air from the settling chamber and since the settling seemed to have been better when the underflow duct was connected to the main air outlet duct, it was decided to test the rig after removing the underflow duct.

The collection efficiency was observed to be 99%. But turbulence in the settling chamber seemed to be higher causing the seeds to be moving in a continuous circular motion inside the settling chamber and forcing few of the seeds revolving around the cone, back into the air outlet duct (less than 1%).

In stage three, the same experimental rig is used after detaching the underflow duct. The collection efficiency was observed without actually taking any measurements. The readings of volumetric flow rate and air velocity in the air inlet and the air outlet ducts were measured to analyse the amount of air being drawn in and drawn out of the system. The readings are given in Table 4.10 and 4.11 respectively.

Sl.no	Points considered across the duct (cm)	INLET DUCT (0.2 x 0.2 = 0.04m²)			
		Volumetric flow rate (m^3/s)	Velocity of air using velocimeter (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)
1	0	0.396	2	0.07	11
2	5	0.515	14	0.08	12
3	10	0.590	14	0.13	15
4	15	0.496	13	0.12	14
5	20	0.397	10	0.11	14
Average		0.48	11	-	13
Q =AxV m^3/s		0.04 x 11 = 0.44		0.04 x 13 = 0.52	
		Q_{av} = 0.5			

Table 4.10 Inlet duct readings taken with hot wire anemometer and pitot static tube (Inlet duct size of 0.2x0.2 m)

Sl.no	Points considered across the duct (cm)	OUTLET DUCT (0.2 x 0.2 = 0.04m²)			
		Volumetric flow rate (m^3/s)	Velocity of air using velocimeter (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)
1	0	0.22	5	0.02	6
2	5	0.40	11	0.05	9
3	10	0.24	6	0.02	6
4	15	0.34	8	0.05	9
5	20	0.36	17	0.19	18
Average		0.31	9	-	10
Q =AxV m^3/s		0.04 x 9 = 0.36		0.04 x 10 = 0.4	
		Q_{av} = 0.38			

Table 4.11 Outlet duct readings taken with hot wire anemometer and pitot static tube (Outlet duct size of 0.2x0.2 m)

Although collection efficiency was observed to be good, the churning of seeds in the settling chamber was higher implying higher turbulence. A small amount of seeds kept rotating around the cone in the cyclone cylinder, of which part of the seeds were being pushed into the clean air outlet duct. This meant that underflow air from the chamber below was trying to escape through the air outlet duct, forcing some of the seeds to remain afloat and escape through the air outlet duct. Small flags attached to the cone were observed to be flapping upwards to the mouth of the air outlet duct confirming the movement of underflow air back into the cyclone cylinder and out through the air outlet duct.

The readings in Tables 4.10 and 4.11 imply that the amount of air drawn in was not being completely drawn out of the cyclone and the settling chamber which would explain for the higher turbulence, observed in the settling chamber.

From Tables 4.10 and 4.11 on calculating the difference in the amount of air drawn in and drawn out of the cyclone cylinder in terms of percentage, the values obtained are, $Q_{in} - Q_{out} = 20\%$ of the total amount of air drawn in.. This amount of air is not being drawn out of the cyclone cylinder and so must be moving down as underflow air. This is quite high because as mentioned earlier the underflow air should only be between 5-10% of the total amount of air drawn into the system. This would attribute to the higher turbulence observed in the chamber. It could also mean that air outlet duct should be slightly bigger than the air inlet duct or the air inlet duct should be slightly smaller than the air outlet duct. This conclusion was arrived at by also considering the results of experiments 1 and 2 in stage 1.

It was concluded to proceed the experiment by using the same rig with some modifications. The inlet duct size was to be reduced to 0.2x0.08 m from 0.2x0.2 m. Therefore the outlet duct area now has twice the area of the inlet duct. Hence the air would be able to escape through the clean air outlet duct before it can move down into the settling chamber causing more disturbance to the seeds.

The underflow air was to be bled from the settling chamber compartment. To achieve this underflow duct with the entrainment nozzle at the point of re-entry into the clean air duct was to be fixed back into the chamber but in a different position. The previous entry had been directly over the bend of the clean air outlet duct. Therefore, this time the underflow duct entrance was fixed at an angle to the main air outlet duct, away from any possible obstructions. This would then facilitate free movement of underflow air from the chamber back into the main duct through the nozzle. This would also prevent any excessive pressure drop in the system.

For stage four, the final stage the same experimental rig is used as in previous experiment with the inlet duct changed from square shape to a rectangular shaped duct and of size 0.2x0.08 m as shown in Figure 4.12 in page 113. Thus obtaining the final model for the Uniflow cyclone model 1. The underflow duct was connected back into the settling chamber with the entrance of the duct fixed at an angle to the air outlet duct. This would promote unobstructed flow of the underflow air from the settling chamber into the main air duct.

The fan was operated at 880 rpm and 1170 rpm. The behaviour and the collection efficiency of the seeds were observed. The volumetric flow rate, velocity pressure and the air velocity across the duct width of the inlet, air outlet and the diameter of the underflow duct were measured. The readings are given in Tables 4.12, 4.13 and 4.14 respectively. The cyclone cylinder air velocity profile was also investigated at fan speeds of 880 and 1770 rpm. The plotted curves are shown in Figure 4.20a for the fan speed of 1770 rpm and Figure 4.20b for the fan speed of 880 rpm. The collection efficiency was observed to be nearly 100% for both White French Millet The turbulence inside the settling chamber had reduced considerably with only slight shifting of seeds at the floor of the chamber.

From the results obtained in the Tables 4.12, 4.13 and 4.14 it is obvious that the amount of air drawn in is same as the amount of air drawn out of the cyclone and the settling chamber, in both cases i.e when fan was operated at 880 and 1170 rpm, and as shown below

$$Q_{In} = Q_{out.1} + Q_{out.2}$$

$$0.26 = 0.24 + 0.03 = 0.27 \text{ m}^3/\text{s} \quad \text{at 880 rpm}$$

and, $$0.402 = 0.34 + 0.06 = 0.4 \text{ m}^3/\text{s} \quad \text{at 1170 rpm.}$$

The percentage of underflow air is 11% of the total amount of air drawn in when fan was operated at 880 rpm and 15% at fan speed of 1170 rpm. This is still higher than the 5-10% range. This could probably be further cut down by reducing the concentrated seed outlet opening from 0.05 m as shown in Figure 4.12 to 0.02 m along the circumference of the cylinder. In the Figure 4.20a for fan speed of 1170 rpm and Figure 4.20b for fan speed of 880 rpm the air profiles are typical of a

Sl.no	Points considered across the duct width (cm)	INLET DUCT ($0.2 \times 0.08 = 0.016 m^2$)					
		Fan speed = 880rpm (30Hz)			Fan speed = 1170rpm (40Hz)		
		Volumetric flow rate, Q using Velocimeter (m^3/s)	$p_{dynamic}$(kPa)	Calculated velocity, V (m/s)	Volumetric flow rate, Q using Velocimeter (m^3/s)	$p_{dynamic}$ (kPa)	Calculated velocity, V (m/s)
1	0	0.128	0.04	8	0.286	0.19	18
2	2	0.235	0.14	15	0.415	0.42	26
3	4	0.268	0.19	17	0.455	0.52	29
4	6	0.354	0.28	22	0.420	0.40	26
5	8	0.322	0.23	20	0.418	0.40	26
Average	-	0.26	-	16	0.402	-	25
$Q = A \times V$ (m^3/s)	$0.016 \times 16 = 0.256$				$0.016 \times 25 = 0.4$		

Table 4.12 Measurement of volumetric flow rate and air velocity across air inlet duct at different fan speeds

Sl.no	Points considered across the duct width (cm)	OUTLET DUCT ($0.2 \times 0.2 = 0.04 m^2$)					
		Fan speed = 880rpm (30Hz)			Fan speed = 1170rpm (40Hz)		
		Volumetric flow rate, Q using Velocimeter (m^3/s)	$p_{dynamic}$(kPa)	Calculated velocity, V (m/s)	Volumetric flow rate, Q using Velocimeter (m^3/s)	$p_{dynamic}$ (kPa)	Calculated velocity, V (m/s)
1	0	0.076	0.002	2	0.224	0.02	6
2	5	0.115	0.005	3	0.277	0.03	7
3	10	0.160	0.010	4	0.276	0.03	7
4	15	0.355	0.050	9	0.354	0.05	9
5	20	0.476	0.070	11	0.556	0.12	14
Average	-	0.24	-	6	0.34	-	9
$Q = A \times V$ (m^3/s)		$0.04 \times 6 = 0.24$			$0.04 \times 9 = 0.36$		

Table 4.13 Measurement of volumetric flow rate and air velocity across outlet duct at differnt fan speeds.

Sl.no	Points considered across the duct diameter (cm)	Fan speed = 880rpm (30Hz)		Fan speed = 1170rpm (40Hz)	
		Q (m^3/s) using Velocimeter	Velocity (m/s) using Velocimeter	Q (m^3/s) using Velocimeter	Velocity (m/s) using Velocimeter
1	0	0.006	1	0.033	4
2	3	0.014	2	0.047	6
3	5	0.055	7	0.080	10
4	7	0.0564	7	0.0795	10
5	10	0.023	3	0.045	6

Table 4.14 Measurement of air velocity across underflow air outlet duct at different fan speeds (ϕ = 0.1m, Area = 0.008m^2).

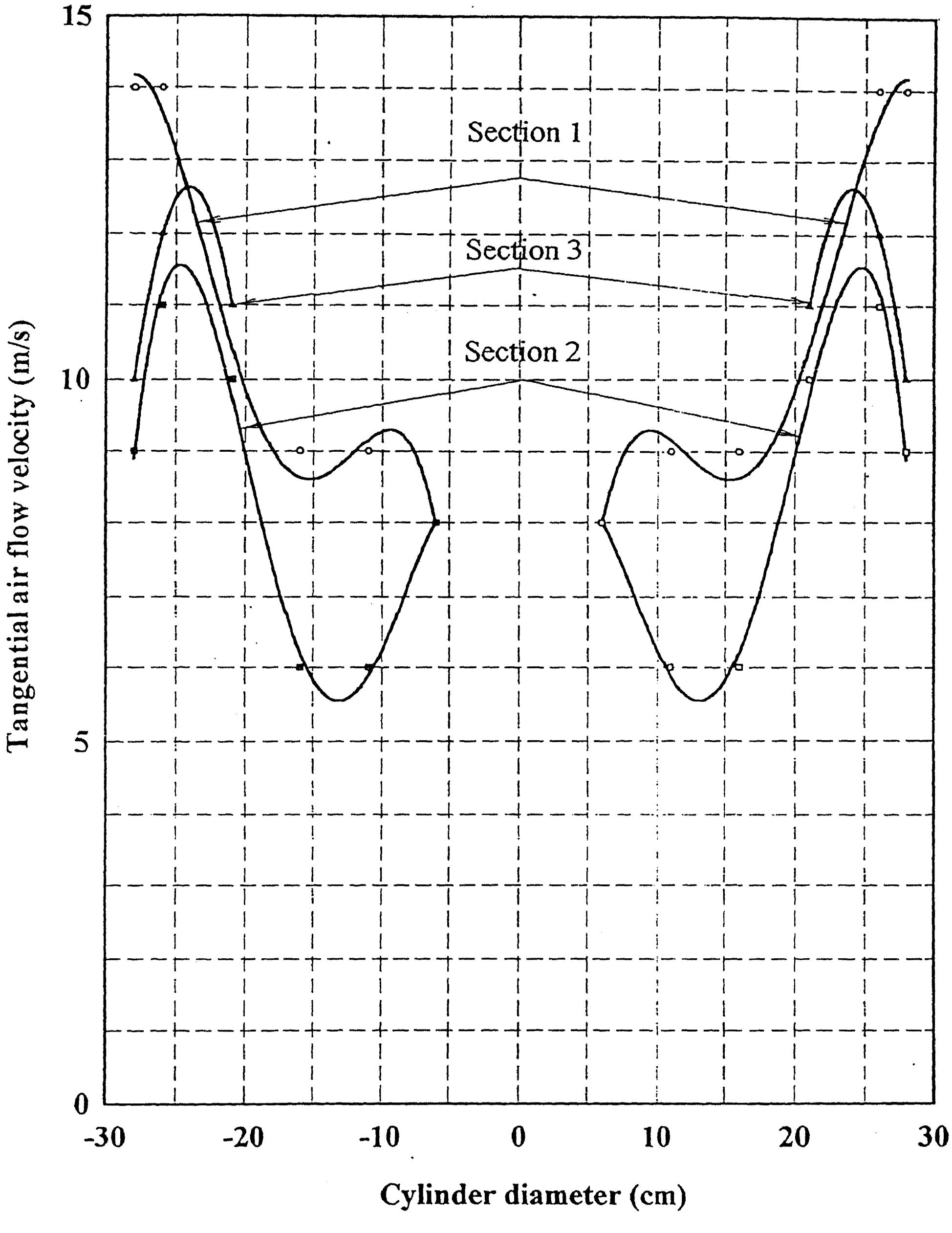


Figure 4.20a Uniflow cyclone model 1. Tangential air flow velocity profile across the cyclone cylinder and the settling chamber with the underflow air duct fixed inside the settling chamber and with the inlet duct size reduced. (Fan operated at 1170 rpm)

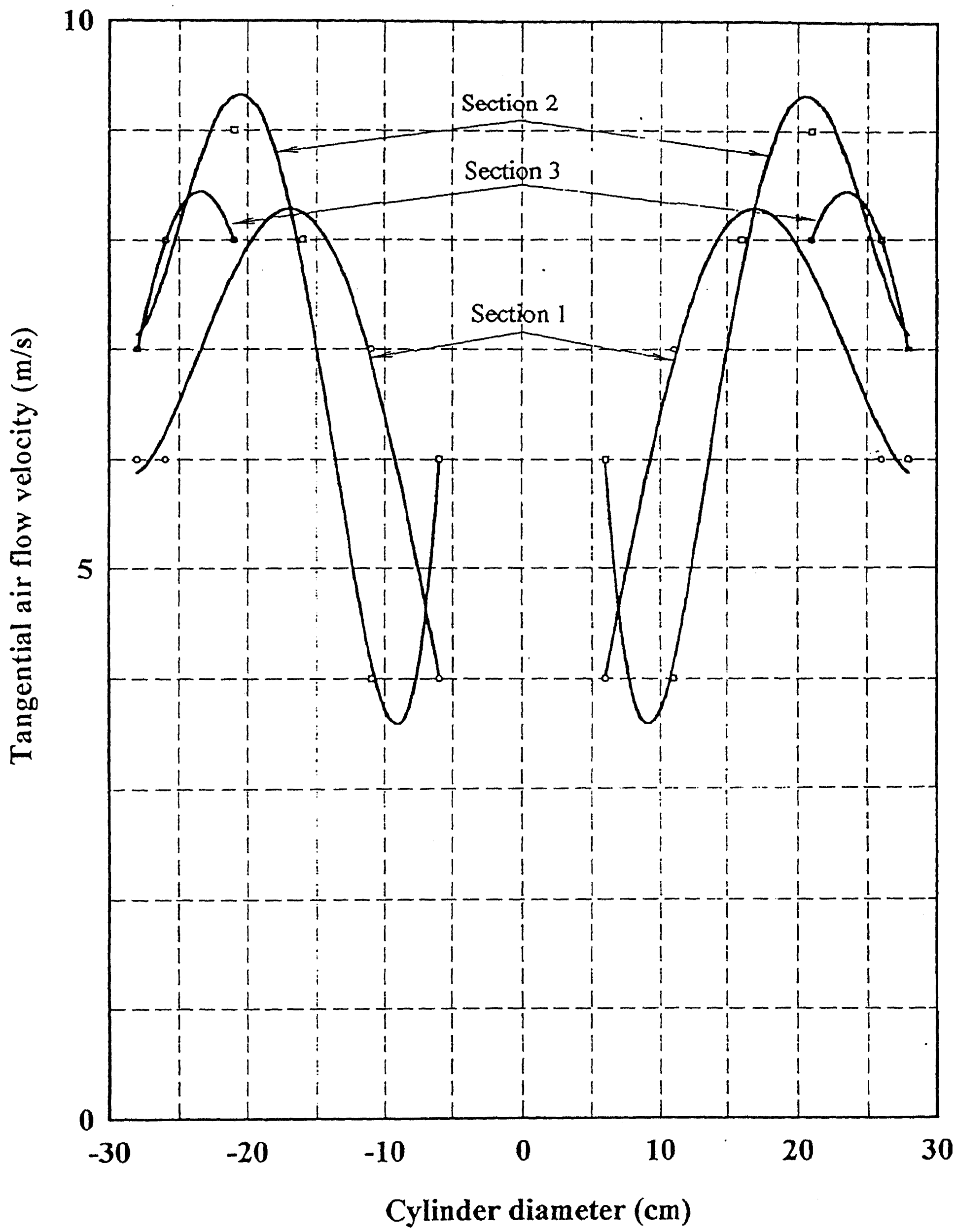


Figure 4.20b Uniflow cyclone model 1. Tangential air flow velocity profile across the cyclone cylinder and the settling chamber with the underflow air duct fixed inside the settling chamber and with the inlet duct size reduced. (Fan operated at 880 rpm)

compound vortex flow. The observed variation in Figure 4.20a is that of a profile of a free vortex at the centre, surrounded by a forced vortex profile. These two vortices are again surrounded by a free vortex near the wall of the cylinder. This type of curve is also observed in section 2 of Figure 4.20b. This variation could be due to the reduction of the inlet duct size and the shape of the duct.

The collection and separation has been observed to have improved to nearly 100%, in terms of accuracy of these experiments. The turbulence in the chamber has also been noticed to have reduced, which means the underflow duct was drawing the underflow air back into the main duct and was functioning properly.

As collection was observed to be approximately 100% when operated at both fan speeds it was decided not to perform any further tests on this rig. It was decided to proceed and conduct the same tests on a cyclone cylinder of smaller size, i.e Model 2 with a smaller diameter. This would be helpful in economising on the size and hence on the cost of the uniflow cyclone, if smaller units could handle the same or larger flows as efficiently.

4.9.2.4 Experimental development of Uniflow cyclone model 2

The design criteria for this model was based on the data obtained and the conclusions drawn from the experiments of the previous model. Hence it was not necessary to go through many different stages to conclude to a final design. But few experiments were conducted on the basic model with slight changes. The tests were mainly repeated on a cyclone of smaller size to confirm whether the performance would be as good and whether it could handle the same concentration and flow. This would facilitate in establishing a optimum size for the cyclone separator.

The experiment model is as explained in section 4.9.2.1 and as shown in Figure 4.13 in page 114. Photographs of the experimental rig as shown in Figure 4.13 are shown in Appendix 6.

The fan was operated at 880 rpm. The collection efficiency test was conducted by observation to test its performance. The tests were proceeded to air velocity profile test on observing good collection. The velocity/dynamic pressure and static pressures were also measured across the four sections considered along the height of the cylinders as shown in the Figure 4.13.

Measurements for downward and tangential air flow were taken. The graphs of the plotted curves are as shown in Figure 4.21 for downward air flow against cylinder diameter and Figure 4.22 for tangential air flow against cylinder diameter. Measurements for velocity of air, volumetric flow rate and velocity pressure were taken across the sections shown in the Figure 4.13 in the inlet duct, air outlet duct and the underflow duct. The values of the same are given in Tables 4.15, 4.16 and 4.17 respectively.

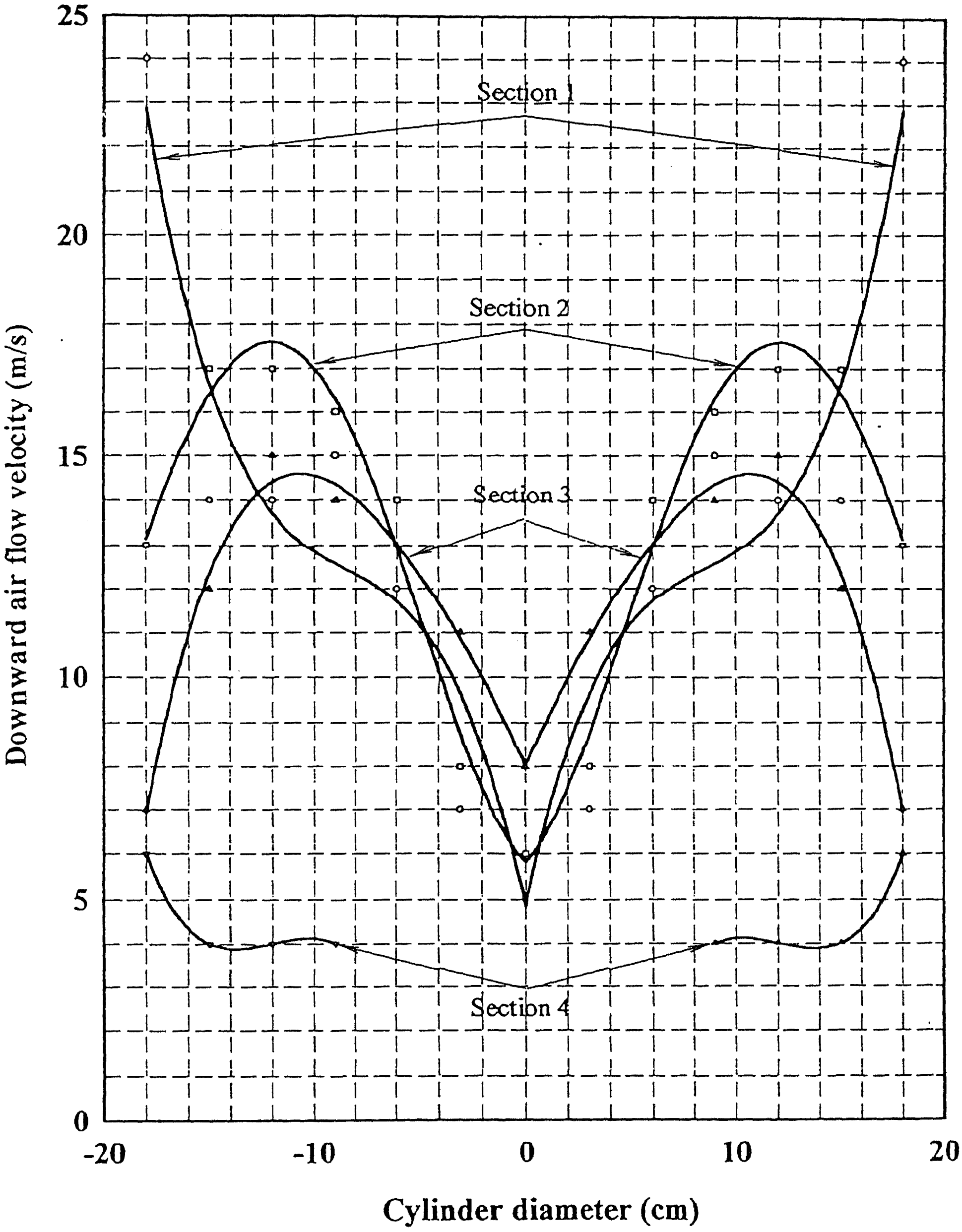


Figure 4.21 Uniflow cyclone model 2. Downward air flow velocity profile across the cyclone cylinder and the settling chamber with the underflow air duct fixed inside the settling chamber (Fan operated at 880 rpm)

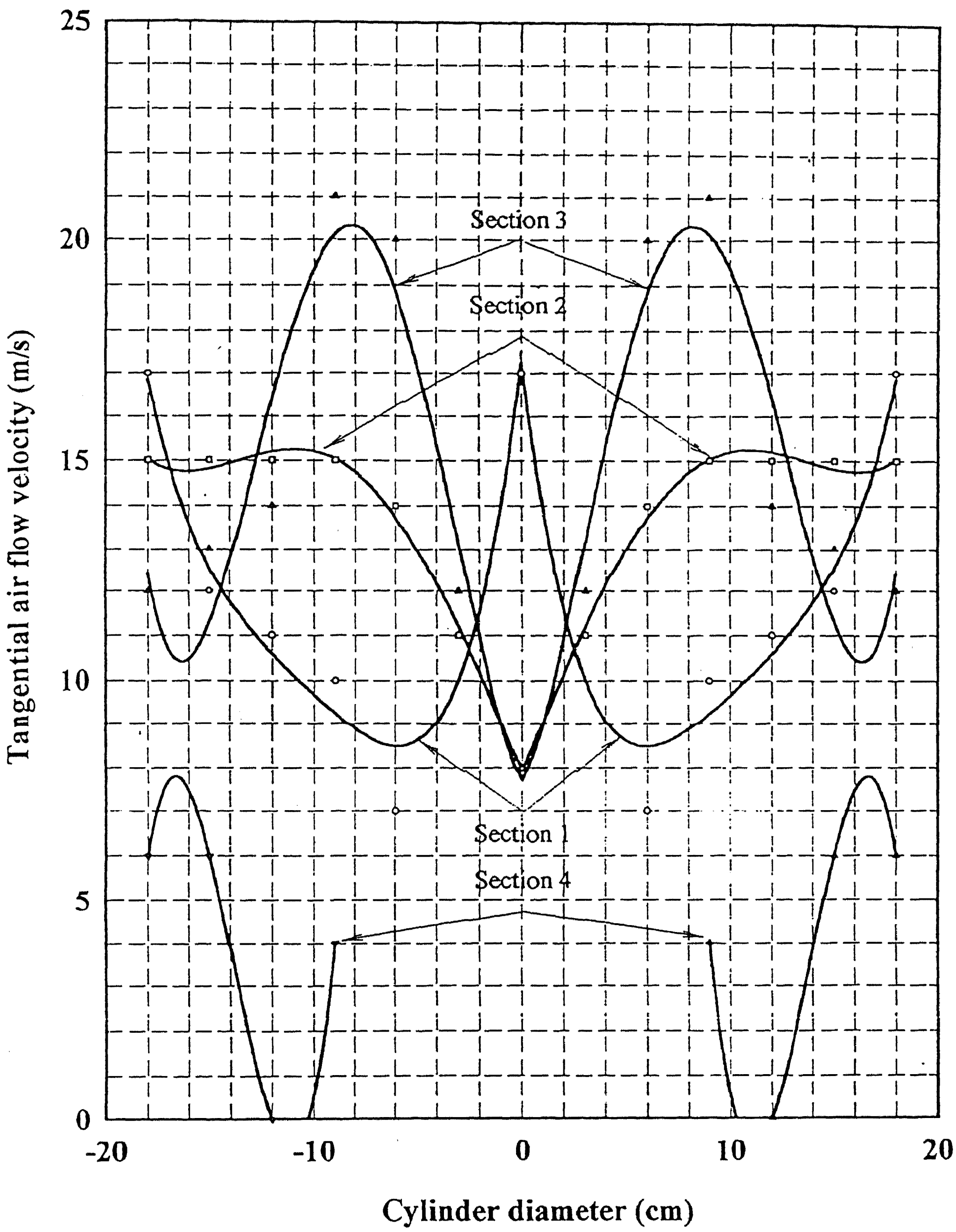


Figure 4.22 Uniflow cyclone model 2. Tangential air flow velocity profile across the cyclone cylinder and the settling chamber with the underflow air duct fixed inside the settling chamber (Fan operated at 880 rpm)

Sl.no	Points considered across the duct width (cm)	INLET DUCT (0.16 x 0.06 = 0.01m^2)			
		Using Velocimeter		Using Pitot Static Tube	
		Volumetric flow rate (m^3/s)	Velocity of air (m/s)	$p_{dynamic}$ (kPa)	Velocity (m/s)
1	0	0.044	4	0.143	15
2	2	0.169	16	0.182	17
3	3	0.177	18	0.193	18
4	4	0.175	18	0.185	18
5	6	0.161	16	0.111	14
Average		0.146	14	-	16
Q =AxV m^3/s		0.01 x 14 = 0.14		0.01 x 16 = 0.16	
		Q_{av} = 0.15			

Table 4.15 Measurement of volumetric flow rate and air velocity in cyclone inlet duct at 880 rpm

Sl. no	Points consid-ered across the duct (cm)	OUTLET DUCT (0.16 x 0.16 = 0.026m^2)					
		Volumetr-ic flow rate (m^3/s)	Velocity of air using velocime-ter (m/s)	$p_{dynamic}$ (kPa)	Calculated velocity (m/s) $V=\sqrt{(2p_{dy})}/\rho$	p_{total} (kPa)	p_{static} (kPa)
1	0	0.064	2	0.005	3	0.687	0.681
2	2	0.129	4	0.004	3	0.689	0.685
3	7	0.131	3	0.003	2	0.692	0.685
4	12	0.133	2	0.001	1	0.692	0.684
5	17	0.104	3	0.004	3	0.689	0.684
6	22	0.055	3	0.004	3	0.689	0.684
Average		0.103	3	-	3	-	-
Q =AxV m^3/s		0.026 x 3 = 0.078		0.026 x 3 = 0.078		Q_{av} = 0.078	

Table 4.16 Measurement of volumetric flow rate and air velocity in air outlet duct at 880 rpm

Sl. no	Points consid-ered across the duct (cm)	UNDERFLOW AIR OUTLET DUCT (Φ = 0.075m, Area = 0.0044m^2)					
		Volumetr-ic flow rate (m^3/s)	Velocity of air using velocime-ter (m/s)	$p_{dynamic}$ (kPa)	Calculated velocity (m/s) $V=\sqrt{(2p_{dy})/\rho}$	p_{total} (kPa)	p_{static} (kPa)
1	0	0.0100	2	0.056	10	0.530	0.482
2	3.75	0.0435	8	0.035	8	0.503	0.478
3	7.5	0.0430	9	0.033	7	0.500	0.479
Average		0.03	6	-	8	-	-
Q =AxV m^3/s		0.0044 x 6 = 0.0264		0.0044 x 8 = 0.0352		Q_{av} = 0.031	

Table 4.17 Measurement of volumetric flow rate and air velocity in the underflow air outlet duct at 880 rpm

The collection was observed to be good, approximately 100%. The Figure 4.21 for downward air flow shows the profile of compound vortex at sections 2 and 3. At section 1 the curve profile is of forced vortex at centre and near the cylinder walls, with a free vortex profile in between. This same profile was observed in Figures 4.20a and 4.20b at the same section. This became noticeable since the previous experiment, after the inlet duct size was reduced and its cross section changed to a rectangular shape which is same as in this case.

Figure 4.22 for tangential air flow shows the profile of a typical compound vortex at sections 2 and 3. At section 1 it is directly opposite of a compound vortex curve, that is with a free vortex profile at the centre surrounded by a forced vortex profile along the cylinder wall. The profile at section 1 could also be explained as forced vortex profiles existing in between the centre of the cylinder and the wall of the cylinder. This change from compound to forced vortex may also be due to the frictional effect due to the proximity of the outer walls. Section 4 profile shows a bit of a turbulence which would not be expected in the settling chamber.

The volumetric flow rate at inlet should be same as the sum of the flow in the air outlet and the underflow ducts, $Q_{in} = Q_{out.1} + Q_{out.2}$. From the measured values the result obtained is approximately true,

$$0.146 = 0.103 + 0.03 = 0.106 \text{ m}^3/\text{s} \quad \text{i.e approximately } 0.11 \text{ m}^3/\text{s}.$$

The slight difference may be due to instrument alignment error.

The percentage of underflow air should be 5-10%. In this case it is,

$$(Q_{out.2} / Q_{in}) \times 100 = (0.03 / 0.15) \times 100 = 20\%$$

This is quite high and the gap provided as concentrated seed outlet should be reduced to the minimum as possible. It was decided to extend the test to concentration test to investigate the maximum seed concentration that could be drawn into the cyclone without losing any seeds, i.e to identify the maximum capacity of particulate air flow that this cyclone could handle.

The concentration test was conducted with the limited facilities by feeding seed from a small hopper into the inlet duct. The narrow opening did not allow adequate feeding with the light fluffy seeds such as Buffel. Hence the test could be performed only on the White French Millet. The fan was operated at both 880 and 1170 rpm.

The measured and calculated readings are shown in Table 4.18.

At fully open condition of the gate and the fan operating at 880 rpm the concentration of seeds was too high for the air flow rate to carry the seeds along. Hence about 584 gm weight of seeds fell out through the inlet opening. It was found that at a fan speed of 880 rpm and a volumetric air flow rate of 0.195 m^3/s the maximum concentration the system could handle was 594 gm/m^3.

At 1170 rpm fan speed and 0.252 m^3/s air flow rate, the system could handle the 1313 gm/m^3 concentration, which was the maximum concentration that could be obtained from the hopper. The collection efficiency was observed to be 100% in each

Sl. No	Fan speed (rpm)	Area of hopper (cm^2)	Wt. of seed to fill the hopper (gm)	Wt. of seed settled through = Wt. of seed collect-ed (gm)	Time for which gate is open (s)	Wt. of seed settled per second (gm/s)	Inlet air flow rate (m^3/s)	Conce-ntration (gm/m^3)
1	880	20x6=120	2648	1281	11	116	0.195	594
2	1170	-do-	2648	2648	12	221	0.252	877
3	880	30x6=180	2648	2064	11	187	0.195	959
4	1170	-do-	2648	2648	8	331	0.252	1313

Table 4.18 Concentration test for White French Millet

case. At 1170 rpm and 0.252 m^3/s air flow rate the system could most probably handle a higher concentration of seeds than the 1313 gm/m^3 and which could not be evaluated due to the lack of facility. As the concentration was increased the collection was noticed to improve and seeds were observed to settle out faster into the settling chamber. It could be concluded that if these values could be used to build a scaled up model, it would bring about a great and positive improvement in the harvesting efficiency of the pasture seed harvester.

4.9.2.5 Conclusion for the Uniflow cyclone concentrator

Conclusions arrived at from the experiments conducted on the Uniflow cyclone concentrator are-

- Uniflow cyclone as a separator is found to be very effective in separating the small volume of seeds from the large volume of air.
- the compound vortex curve shows velocity peak at mid section of the radius of the concentrator. This accelerates the radial movement of the seeds towards the wall resulting in separation of seeds from the large volume of air flow.
-this type of concentrator provides a larger length of curvature for the seeds, to develop the centrifugal forces required to drive them outwards towards the cyclone walls to get separated from the air flow and be carried into the settling chamber by the underflow air.
- the possibility of traumatisation of seed is very less.

5.0 HARVESTER DESIGN-SCALING TECHNIQUES

Results obtained from the experiments can be used as a basis for the practical design of a tractor-operated or a self-driven-vacuum harvester. The basic design may require further modification to achieve optimum commercial performance. The scaling procedures outlined in this section can be used to modify the basic design to better suit individual farmers requirements.

5.1 CONCLUSIONS AND SUGGESTIONS

The results obtained from the experiments previously done [1] on the header unit, fan [1] and those obtained from this thesis on the separator, the conclusions drawn and suggestions made are outlined here.

5.1.1 Header unit

The picking head width should cover all pasture grasses over which the tractor passes. As the wheels of the tractor bend the crops over which they run, this crop must be harvested before passage of the tractor. Usually the next harvest of the same crop will be some days later by which time the grass will return to its free standing position. A minimum width would be of the order of 2 m.

As the same variety or type of pasture grasses are observed to grow at varying heights and also because of undulating ground, the aperture height of the header unit to be considered is very important. The aperture height selection is based on the varying range in the height of crops and selected such as to be able to collect the maximum percentage of ripe seeds in any one run. This range of height may vary from one variety of grass to another. For scaling up the aperture height value has been arbitrarily selected based on Woodward Flail-Vac Seed Stripper. The velocity of air required at the header, the size of the fan and the availability of mechanical power also are a basis in selection of aperture height. The design presently being used with the curved shroud can be used for bending the taller grasses to allow for a reduced aperture height. Reduction of header aperture height leads to reduction in the area of the aperture which in turn increases the air velocity at the header unit for a given air flow rate. It has been found in earlier work that if the air velocity at the header is sufficiently high tall grasses and in some cases lodged grasses will be directed into the aperture due to this suction power. The reduction in the header aperture area also reduces the volumetric flow rate of air.

As mentioned previously there is a limitation for reduction in aperture height which depends on the range of height of the crops. If the curved shroud is bent down too low then it might break the taller grasses or hit the seed head and knock the ripened seeds off taller grasses which would fly into the air and be lost. Previous experiments have shown that to shake and draw the ripened seeds into the harvester the range of air velocity required are between 12 to 20 m/s [1]. The easiest way to achieve high velocity is by reduction of header aperture area. Further research work needs to be conducted in designing of effective header unit to achieve the said range of air velocity by not going beyond the minimum limit of aperture height.

Till recent years, header units of existing harvesters consisted of a revolving brush which dislodged the seeds and provided air flow to propel the seeds into a bin. To improve on this, work was conducted on the header and fan units at CQU to develop a new concept of an air-assisted pasture seed harvester in which a forced airflow would provide both seed transport and at the same time enhance the stripping efficiency. The experiments proved that total yield with forced air assisted brush type seed harvester at same operating conditions was higher than the brush type seed harvester. The decision to use an air assisted brush harvester was based on the observation that the tangential velocity caused by circular motion of the brush although beneficial in gathering seed heads at some place still tended to also push the seed heads away and disperse the seeds onto the ground. This adversely effected the seed quality and yield of mature seeds. The high airflow velocity was required to overcome the vortex which tended to push the seed heads away from the brush. The minimum velocity needed to bend the seed head towards the brush unit was found to be 12 m/s [1].

This problem of bending the seed heads towards the brush unit mainly depended on the stiffness and length of the grass. Although this improved efficiency by two to three fold, it is still believed that since the harvesting is still done by physical contact that contact could cause damage to seeds. A suggestion has therefore been put forward to harvest seeds without any physical contact but by creating a to and fro transverse motion of air. In this case the high velocity at header unit is used to disturb and draw the ripe seeds from the seed head. To draw and disturb the seeds, consider two sheets with lower sheet fixed and upper sheet made to vibrate at high amplitude causing the ripened seeds to joggle. To vibrate the sheet either an electronic device or some mechanical system could be used. This would help in overcoming the problem of seed heads being pushed away by the centrifugal force produced by the brush unit and also prevents any damage caused by the physical contact of the harvesting unit with the seeds.

5.1.2 Air seed separator

The uniflow cyclone model successfully tested in the lab needs to be scaled up for commercial harvesters. For scaling up we have considered air to be an incompressible fluid. A Mach number less than 0.2 has been maintained throughout the design by the necessity to limit system pressure drops. The error in assuming the flow to be incompressible amounts to be less than 1% at such air velocities [17]. Therefore the whole harvester system should be designed to maintain air velocities below 60 m/s. Mach number becomes a significant parameter in flow situations where the ratio of flow velocity to sonic velocity exceeds about 0.25 to 0.3. Since Mach number is a function of density it remains same at any given temperature and pressure even when scaled up. While scaling up the consistency of Mach number or velocity should not exceed a value of 0.3 (as mentioned earlier), but it may have the effect of increasing the Reynolds number as it depends on values of length of duct. But this increase is mitigated by the fact that the duct diameter also increases along with the length in the scaled up model when compared with the test model.

5.1.3 Fan

The selection and design of fan are based on results obtained from previous experiments. Backward inclined fan blades have been considered as explained in Chapter 3. The fan size decision is based on pressure drop developed through the system. As the system considered deals with moving air, the pressure developed in the system is mainly velocity pressure. Hence the maximum velocity developed within the system (at the point of minimum area) is used as the base to calculate this maximum expected velocity pressure drop. The fan is then designed to handle this pressure drop for moving the air and seed through the whole system. The volumetric flow rate and the availability of power have also been taken into consideration.

5.1.4 The settling chamber

Although bulk material handling does not form part of the thesis, it would be wise to remember and understand the importance of designing and maintaining an efficient method of removal of separated seeds from the settling chamber. The efficiency of the separator unit and the whole system can be lowered dramatically by use of improper seed removal methods. To counteract this effect a suggestion has been made for the design of a gate. This aspect in itself can form part of further research work.

The gate through which the seeds are removed out of the settling chamber if designed improperly, could reduce the vacuum pressure within the system.

Consequently it would cause a reduction in the separation of the seeds from the air and effect the operation of the whole system. The gate should be positioned at the end of a long duct leading from the settling chamber sloping downwards preferably at an angle of 45°. The gate considered should be a counter balancing type of gate, i.e a type which is forced to open outwards when a specific weight of seed collects in the duct. The gate is controlled by a spring under tension and when the seed weight on the gate increases by more than the spring force, it opens letting out seed and when the seed weight decreases to the spring force, it closes the gate. A schematic diagram of the proposed gate is shown in Figure 5.1.

A problem which would be encountered when using such a gate is an obstruction to complete closure of gate due to obstruction caused by seeds blocking its return. This in turn would cause a continuous trickle of seeds and also lower the vacuum pressure within the system if air is able to enter through this gate. To overcome this problem it would be wise to always maintain the level of seeds at least halfway along the length of the duct.

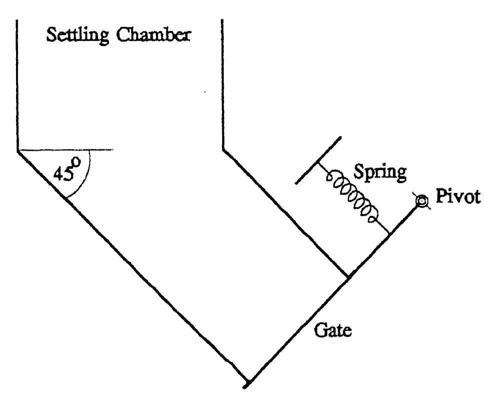


Figure 5.1 Seed removing gate

5.2 CALCULATED SYSTEM PRESSURE

Before scaling up the model the first requirement is to calculate the system pressure drop. The pressure drop at each section could be calculated. Whilst the system is closed i.e, it contains air below atmospheric pressure, it must by its nature be open to atmospheric pressure at both ends. But in between the pressure will be below atmospheric pressure so system walls must be closed. In such a case the best method to calculate the system pressure drop is to calculate the difference in pressure between the atmospheric pressure and the point of maximum expected velocity within the system. This difference between the atmospheric pressure and the considered velocity pressure would be the pressure drop required to be developed by the fan for moving air and seed through the system. Velocity pressure has been considered as the system deals with moving fluid and hence the static pressure would be negligible. Since a Mach number of 0.2 has been assumed the maximum velocity anywhere within the system should be below 60 m/s. Therefore, in a system where Ma = 0.2,

p_V , velocity pressure = $\rho v^2/2 = 1.2 \times 60^2/2 = 2160$ Pa.

Hence the pressure to be developed by the fan would be with added allowance of 50% pressure to overcome friction and other losses.

$\Delta p = 2160 + 0.5 \times 2160 = 3240$ Pa = 3.24 kPa.

Note: Although Mach number 0.2 has been assumed for design purposes a maximum velocity of 40 m/s has been assumed. Hence the pressure required to be developed by the fan would be about 1.5 kPa with the added allowance of 50% for friction and other losses amounting to 2.25 kPa.

5.3 DESIGN OF VACUUM HARVESTER

The header unit size selection was based on information gathered from the market survey, size of tractor and variation in crop height. Design of the Uniflow cyclone has been partly based on Stairmand High efficiency cyclone design [13]&[16]. Fan design was based on the one dimensional flow and the velocity triangle theory as detailed in chapter 2.

5.3.1 Notations

5.3.1.1 Header unit

Q = total volumetric flow rate (m^3/s)

m = total mass flow rate (kg/s)

V_H = velocity of air at header unit (m/s)

H_W = header width (m)

H_{W_r} = reduced header width (m)

H_h = height of header aperture (m)

A_H = area of header aperture (m^2)

5.3.1.2 Uniflow cyclone unit

n_c = number of cyclones considered

Q_c = volumetric flow rate per cyclone (m^3/s)

V_{ic} = velocity of air at cyclone inlet (m/s) (assumed)

D_c = diameter of cyclone cylinder (m)

a = height of duct (m)

b = width of duct (m)

d_{ic} = diameter of cyclone inlet duct (m)

A_{ic} = area of cyclone inlet duct (m^2)

d_{oc} = diameter of cyclone outlet duct (m)

A_{oc} = area of cyclone outlet duct (m^2)

h_c = height of cyclone cylinder (m)

5.3.1.3 Fan unit

n_f = number of fans considered

Q_f = Volumetric flow rate per fan (m^3/s)

N = Revolution per minute (rpm)

ω = angular velocity of impeller (radians/s)

Δp = total pressure developed by the fan (kPa)

A_1 = Area of inlet duct (m^2)

A_2 = Area of outlet duct (m^2)

D_1 = Diameter of fan inlet (m)

D_2 = Diameter of fan outlet (m)

b_1 = inlet blade width (m)

b_2 = outlet blade width (m)

B = number of blades considered

U_1 = inlet tangential velocity of impeller (m/s)

U_2 = outlet tangential velocity of impeller (m/s)

V_{if} = inlet fan velocity (m/s)

V_1 = inlet absolute velocity of fluid (m/s)

V_2 = outlet absolute velocity of fluid (m/s)

V_{f1} = inlet flow velocity of fluid (m/s)

V_{f2} = outlet flow velocity of fluid (m/s)

V_{w1} = inlet whirl velocity of fluid (m/s)

V_{w2} = outlet whirl velocity of fluid (m/s)

V_{r1} = inlet relative velocity of fluid m/s)

V_{r2} = outlet relative velocity of fluid (m/s)

β_1 = inlet blade angle (degree)

β_2 = outlet blade angle (degree)

h_p = Euler head (m)

p = Euler pressure (Pa or kPa)

p_{vf} = pressure drop due to flow velocity of fluid (Pa or kPa)

g = acceleration due to gravity (m/s^2)

p_{atmos} = atmospheric pressure (assumed = 0)

m_f = mass flow rate per fan (kg/s)

T = Torque (N-m)

W_{FP} = fluid power (watts or kW)

W_m = mechanical power (watts or kW)

η = efficiency of fan (%)

W = tractor or motor power available (W or kW)

W_F = available power to each fan (W or kW)

5.3.2 Formulas used

5.3.2.1 Header unit

Assumed values

H_w value is assumed based on the survey report.

H_{wr} is assumed to reduce the net aperture area to increase the velocity of the air that is being drawn in and to reduce the total volumetric flow rate at the same time. This is done by fixing shoes at equal gaps in the header aperture.

H_h value is based on the varying range of the height of crops.

V_H value is based on data odtained from previous research work[1].

The area of the aperture and the total volumetric flow rate are then calculated using the equations given below.

$$A_H = H_{wr} \times H_h$$

$$Q = A_H \times V_H$$

5.3.2.2 Uniflow cyclone unit

In the design of cyclone unit, the number of cyclone units and velocity of air at the inlet were assumed. As the total volumetric flow rate of air would be very high, if a single cyclone was considered, the size of the cyclone unit would have to be very large. Therefore the number of cyclone units assumption was based on the volumetric flow rate of air.

$$Q_c = Q/n_c$$

In most cyclone designs the sizes of the inlet, outlet ducts, height of the cyclone cylinder are all based on the size of the cyclone cylinder diameter. In this case, a

Mach No of 0.2 was assumed to maintain an incompressible fluid flow. Therefore the velocity of air would always have to be below 60 m/s within the system. With this maximum velocity as a basis an air velocity at the cyclone inlet was assumed and the cyclone inlet duct area was calculated.

$A_{ic} = Q_c/V_{ic}$

$d_{ic} = \sqrt{(4 \times A_{ic}/\pi)}$ (if duct considered is circular)

The rest of the cyclone proportions are based on the cyclone diameter. The design is based on C.J.Stairmand high efficiency cyclone design. Usually the cyclone cylinder diameter is assumed and then the other proportions are calculated. In this case using the same proportions and by back calculation the size of cyclone is obtained.

$a = 2.1$ to $2.5\ b$ (if duct considered is rectangular)

$A_{oc} = 2 \times A_{ic}$

$d_{oc} = \sqrt{(4 \times A_{oc}/\pi)}$ (if duct considered is circular)

$a = b = \sqrt{A_{oc}}$ (if duct considered is square type)

$D_c = a/0.5$ ie $a = 0.5D$

$h_c = D_c + 0.5D_c$

5.3.2.3 Fan unit

The performance of the fan is dependent on the impeller action which induce the fluid flow from the inlet to the outlet duct. The fluid which enters axially is directed by the impeller to flow out towards the periphery. Hence it is important to do detailed designing of the impeller and casing. Fluid action can be understood by the study of fluid velocities at the inlet and outlet points of the impeller, as explained in the one dimensional flow and the velocity triangle theory in Chapter 3. The fan size required for handling the particular system pressure is calculated purely by trial and error method.

The assumed values are, D_1, D_2, b_1, b_2, β_2, N&B, η and the rest of the parameters are then calculated using the formulas given below.

Design Requirements:--

i. The condition $A_2 > A_1$ allows all pressure increase to occur across the impeller due to deceleration of flow.

ii. Blade width b_1 and b_2 selection were based on consideration of blade stiffness and flow volume.

iii. Selection of the number of blades B and the speed of N rpm were based on reducing the noise created by the fan. If a blower has B blades and runs at a speed of N rpm, the traverses of each blade across the outlet aperture per second would be NB/60, each producing a fluctuation in the air pressure. Since human ear is insensitive to frequencies below 150Hz, so if NB/60 lies below this frequency the fan would appear quiet.

iv. The difference between fan inlet and outlet diameter helps in developing the high pressure required.

Considering both the inlet and outlet velocity triangles-

It is easy to establish the tangential velocities relative to the impeller at the inlet and outlet edges of the blades. Then relating the absolute fluid velocities at the inlet and outlet to these, relative velocities can be represented by two velocity triangles as shown in Figure 5.2. From the velocity triangle the different relations for the velocities are obtained. The radial velocities called as flow velocities are found by

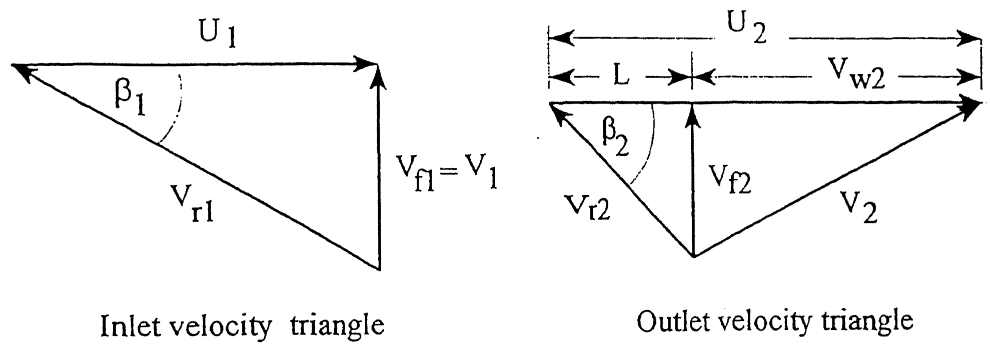


Figure 5.2 Velocity triangles

use of the continuity equation-

$V = Q/A$

$\therefore V_1 = Q/\pi D_1 b_1$

and $V_2 = Q/\pi D_2 b_2$

where $\pi D_1 b_1 = A_1$ is cylindrical inlet area and $\pi D_2 b_2 = A_2$ is cylindrical outlet area.

The velocities of whirl are derived from the velocity triangle. In most cases fluid at inlet has no whirl velocity so,

$V_{w1} = 0$

$V_{w2} = U_2 - L = (U_2 - V_{f2})/\tan \beta_2$

The tangential velocity of impeller is calculated from the relationship

$U = \omega r$

which gives, $U = \pi ND/60$

Therefore, $U_1 = \pi ND_1/60$

and $U_2 = \pi ND_2/60$

Using the inlet and outlet velocity triangles the other values are calculated,

$$V_{r1} = \surd(\ v_{f1}^2 + U_1^2\ )$$

$$\beta_1 = \tan^{-1}(\ V_{f1}/U_1\ )$$

$$V_{r2} = \surd(\ L_2 + V_{f2}^2\ )$$

$$V_2 = \surd(\ V_{f2}^2 + V_{w2}^2\ )$$

The theoretical head developed i.e the Euler head is given by,

$$h_p = (\ U_2 V_{w2} - U_1 V_{w1})/g$$

But because at inlet there is no velocity of whirl, so Euler head takes the form,

$$h_p = U_2 V_{w2}/g$$

when expressed in pressure form,

$$p = \rho U_2 V_{w2}$$

Pressure developed by flow velocity is given by,

$$p_{vf} = \rho(\ V_{f1} - V_{f2}\ )^2/2$$

Therefore the total head developed by the fan is

$$\Delta p = p_{vf} + p$$

On relating torque T to a mass flow rate of a fluid system as shown in Figure 3.14, the torque on the system can be derived as,

$$T = m(V_{w2}r_2 - V_{w1}r_1)$$

where m is the mass flow rate. The volumetric flow rate being known the mass flow rate can be calculated using the relationship

$$m = \rho Q$$

When a torque is applied to any system rotating with angular velocity, a power transfer occurs and since

$$\omega = U/r$$

$$\therefore W = T\omega = m(V_{w2}U_2 - V_{w1}U_1)$$

This is the power imparted on the fluid by the impeller. Fluid power can also be calculated as

$$W_{FP} = \Delta p \times Q$$

By assuming an efficiency η of fan, the mechanical power can be calculated,

$$W_m = W_{FP}/\eta$$

Thus total mechanical power, $W_m = W_m \times n_F$

And tractor or engine power = $W_m + 0.5\ W_m$

5.3.3 Design specifications

The calculated system pressure is 3.24 kPa. But fan should be capable of developing slightly more pressure than the calculated system pressure to avoid it from getting overloaded. If the system velocity can be maintained then the system pressure can be controlled which in turn prevents the fan from being overloaded. There are two critical points in the system,

i. the cyclone inlet cross-sectional area and

ii. the extractor or the venturi point due to reduction in cross-sectional area.

If a square duct is used, then at the air extracting point from the settling chamber, the increase of air velocity could be prevented by bulging the duct at the point of extraction. Therefore considering the mentioned possible problems, a fan capable of developing 3.33 kPa pressure has been considered. Worked examples for design specifications of a tractor mounted vacuum harvester and a self driven vacuum harvester are given below.

5.3.3.1 Tractor mounted vacuum harvester

5.3.3.1.1 Header unit

H_w, header width = 3 m

H_{wr}, reduced header width = 2.4 m

H_h, height of header aperture = 0.5 m

A_H, area of header aperture = 1.2 m^2

V_H, velocity of air at header unit = 16 m/s

Therefore Q, total volumetric flow rate = 19 m^3/s

5.3.3.1.2 Uniflow cyclone unit

M, Mach number assumed = 0.2

Q, total volumetric flow rate = 19 m^3/s

n_c, number of cyclones considered = 4

Q_c, volumetric flow rate / cyclone = 5 m^3/s

V_{ic}, velocity of air at cyclone inlet = 40 m/s

D_c, cyclone cylinder diameter = 1.1 m

A_{ic}, Area of cyclone inlet duct = 0.125 m^2

a x b, height x width of cyclone inlet duct if rectangular duct is considered = 0.55 x 0.22 m

d_{ic}, diameter of cyclone inlet duct if circular duct is considered = 0.4 m

A_{oc}, Area of cyclone outlet duct = 0.25 m^2

a = b, height and width of cyclone outlet duct if square duct is considered = 0.5 x 0.5 m

d_{oc}, diameter of cyclone outlet duct if circular duct is considered = 0.56 m

h_c, height of cyclone cylinder = 1.65 m

5.3.3.1.3 Fan unit

n_f, number of fans considered = 4

Q, total volumetric flow rate = 19 m^3/s

Q_F, volumetric flow rate / fan = 5 m^3/s

Assumptions

Note: To avoid too many pressure changes within the system the diameter of the cyclone outlet has been taken as the diameter of fan inlet duct.

D_1 = 0.56 m	D_2 = 0.76 m
b_1 = 0.21 m,	b_2 = 0.21 m
A_1 = 0.37 m^2,	A_2= 0.50 m^2
N = 1500 rpm	$\beta_2 = 36^0$
B = 6	

Inlet velocity triangle

V_{f1} = 13.5 m/s

U_1 = 44 m/s

V_{w1} = 0 (assumed)

V_{r1} = 46 m/s

β_1 = 17^0

Outlet velocity triangle

V_{f2} = 10 m/s

U_2 = 60 m/s

L = 13.8 m/s

V_{w2} = 46.2 m/s

V_{r2} = 17 m/s

V_2 = 47.3m/s

Pressure developed

p_{vf} = 7.35 Pa

p = 3326 Pa

Δp = 3334 Pa

Power required at full 3.33 kPa total pressure

W_{FP} = 16.7 kW

W_m = 28 kW

W_m = 112 kW

W = 168 kW

5.3.3.2 Self driven vacuum harvester

5.3.3.2.1 Header unit

H_w, header width = 5 m

H_{wr}, reduced header width = 4.5 m

H_h, height of header aperture = 0.5 m

A_H, area of header aperture = 2.25 m^2

V_H, velocity of air at header unit = 16 m/s

Therefore Q, total volumetric flow rate = 36 m^3/s

5.3.3.2.2 Uniflow cyclone unit

M, Mach number assumed = 0.2

Q, total volumetric flow rate = 36 m^3/s

n_c, number of cyclones considered = 4

Q_c, volumetric flow rate / cyclone = 9 m^3/s

V_c, velocity at cyclone inlet = 40 m/s

D_c, cyclone cylinder diameter = 1.5 m

A_{ic}, area of cyclone inlet duct = 0.225 m^2

a x b, height and width of cyclone inlet duct if rectangular duct is considered = 0.75 x 0.3 m

d_{ic}, diameter of cyclone inlet duct if circular duct is considered = 0.54 m

A_{oc}, area of cyclone outlet duct = 0.45 m^2

a = b, height and width of cyclone outlet duct if square duct is considered = 0.67 x 0.67 m

d_{oc}, diameter of cyclone outlet duct if circular duct is considered = 0.76 m

h_c, height of cyclone cylinder = 2.25 m

5.3.3.2.3 Fan unit

n_f, number of fans considered = 4

Q, total volumetric flow rate = 36 m^3/s

Q_F, volumetric flow rate/fan = 9 m^3/s

Assumptions

As mentioned earlier, for avoiding pressure changes within the system the fan outlet duct diameter is assumed the same as cyclone outlet duct.

D_1 = 0.7 m, D_2 = 0.9 m,

b_1 = 0.21 m, b_2 = 0.21 m

A_1 = 0.46 m^2, A_2 = 0.59 m^2,

N =1500 rpm, $\beta_2 = 36°$

B = 6

Inlet velocity triangle

$V_{f1} = 19.6$ m/s

$U_1 = 55$ m/s

$V_{w1} = 0$ (assumed)

$V_{r1} = 58.4$ m/s

$\beta_1 = 20°$

Outlet velocity triangle

$V_{f2} = 15$ m/s

$U_2 = 70.7$ m/s

$L = 21$ m/s

$V_{w2} = 49.7$ m/s

$V_{r2} = 25.8$ m/s

$V_2 = 51.9$ m/s

Pressure developed

$p_{Vf} = 12.7$ Pa

$p = 4217$ Pa

$\Delta p = 4230$ Pa $= 4.23$ kPa

Power required at full 4.23 kPa total pressure

$W_{FP} = 37.8$ kW

$W_m = 63$ kW

Total W_m = 252 kW

Engine power =252 + 0.5 x 252 = 378 kW

In this case, the pressure changes caused within the system may be overlooked and the fan outlet duct diameter need not be assumed to be same as the cyclone outlet duct as by doing so, as seen above, we end up designing a fan that is far too large, able to develop 4.2 kPa pressure which is much more than required.

Either the same fan used for tractor mounted may be used in this case or design a new one with smaller inlet and outlet diameter values.

$D_1 = 0.6$ m	$D_2 = 0.8$ m
$b_1 = 0.21$ m,	$b_2 = 0.21$ m
$A_1 = 0.4$ m^2	$A_2 = 0.53$ m^2
N =1500 rpm,	$\beta_2 = 36°$
B = 6	

From inlet velocity triangle

$V_{f1} = 22.5$ m/s

$U_1 = 47.1$ m/s

$V_{w1} = 0$ (assumed)

$V_{r1} = 52.2$ m/s

$\beta_1 = 26°$

From outlet velocity triangle

V_{f2} = 17 m/s

U_2 = 62.8 m/s

L = 23.4 m/s

V_{w2} = 39.4 m/s

V_{r2} = 28.9 m/s

V_2 = 42.9 m/s

Pressure developed

p_{Vf} = 18.2 Pa

p = 2969 Pa

Δp = 2987 Pa = 2.99 kPa

Power required at full 2.99 or 3 kPa total pressure

W_{FP} = 26.9 kW

W_m = 44.9 kW

Total W_m = 180 kW

Engine power = 270 kW

The system pressure for self driven vacuum harvester with maximum velocity of 60m/s with provision for friction losses comes to 3.24 kPa as shown in section 2.0. Hence the same fan designed for tractor mounted vacuum harvester can be used as it develops a pressure of 3.33 kPa

5.4 SIZES AND LENGTHS OF DUCTS REQUIRED FOR THE VACUUM HARVESTER

Duct sizes and air velocity have been chosen such as to ensure effective transport of seeds. While selecting ducts for the harvester it is important to remember to select smooth ducts to avoid seed damage due to abrasion and to reduce friction losses. The suggested sizes and lengths of ducts between different units for both tractor mounted and self driven vacuum harvester are given below.

5.4.1 Tractor mounted vacuum harvester

1. Main duct drawing air from header unit

   Diameter of duct = 0.8 m

   Area of duct = 0.5 m^2

   Velocity of air = 19/0.5 = 38 m/s

   Length of duct = 1.5 m

2. Ducts leading into the four cyclone inlets

   Diameter of duct = 0.4 m

   Area of duct = 0.125 m^2

   Velocity of air = 5/0.125 = 40 m/s

   Length of duct = 1 m

3. Ducts leading from cyclone to venturi

Diameter of duct = 0.56 m

Area of duct = 0.25 m^2

Velocity of air = 20 m/s

Length of ducts = 0.5 m

4. Venturi size

Minimum diameter = 0.45 m

Area at point of minimum diameter = 0.16 m^2

Velocity of air = 56 m/s

Length of total venturi = 0.25 m

5. Ducts leading from the venturi to fan inlet

Diameter of duct = 0.56 m

Area of duct = 0.25 m^2

Velocity of air = 20 m/s

Length of ducts = 0.75 m

5.4.2 Self driven vacuum harvester

1. Main duct drawing air from header unit

Diameter of duct = 0.87 m

Area of duct = 0.59 m^2 = 0.6 m^2

Velocity of air = 60 m/s

Length of duct = 1.5 m

2. Ducts leading into the four cyclone inlets

Diameter of duct = 0.54 m

Area of duct = 0.225 m^2

Velocity of air = 40 m/s

Length of duct = 1 m

3. Ducts leading from cyclone to point of venturi

Diameter of duct = 0.76 m

Area of duct = 0.45 m^2

Velocity of air = 20 m/s

Length of duct = 0.5 m

4. Venturi size

Minimum diameter = 0.45 m

Area at point of minimum diameter = 0.16 m^2

Velocity of air = 56 m/s

Length of total venturi = 0.25 m

5. Ducts leading from point of venturi or reduction to fan inlet

Diameter of duct = 0.56 m

Area of duct = 0.25 m^2

Velocity of air = 36 m/s

Length of duct = 0.75 m

5.5 CONCLUSION AND SUGGESTIONS

From the above design for tractor operated and self driven vacuum harvester the conclusions arrived at are-

To attain the required velocity at the header of 16 m/s, with the present considered header dimensions of 3 x 0.5 m for tractor mounted and 5 x 0.5 m for the self driven vacuum harvester, four large fans are required which require quite high power to maintain the required flow rate. Hence the design although is viable theoretically may not be viable when viewed on an economic basis for the medium holding farmers.

It has been observed in some previously conducted field tests that on practical operation the fan does not use the power anywhere close to the theoretically calculated power values. But still we can not compromise on the size of the fan based on this observation.

No compromise can be made on the value of the velocity considered at the header unit, as it was assumed based on the values collected from previous research works [1] as the value required for effective harvest of pasture seeds. Hence to reduce the large volumetric flow rate keeping velocity constant, the only other approach left is to reduce the area of the header unit. But this again brings forward some problems as the area can be reduced only upto a certain extent. As explained previously the height of the header is based on the expected varying heights observed among the same variety of pasture grasses and the head width is decided based on the spacing of the tractor wheel and the wheel width. But it should be more economical for designing a harvester for harvesting fluffy seeds as they require smaller velocity of 6 to 12 m/s.

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APPENDIX 1

MARKET SURVEY QUESTIONNAIRE

GRASS SEED HARVESTING
IN CENTRAL QUEENSLAND

(Please indicate responses in the appropriate box)

1. Area under cultivation of pastures seeds ☐ ha

2. Area harvested in the past year ☐ ha

3. Total yield in past year ☐ kg

4. Type of machine used for harvesting

☐ Tractor mounted ☐ Tractor beater harvester

☐ All crop header ☐ Tractor brush harvester

☐ (Other) ____________________

5. Post-harvest treatments

Drying -> ☐ On hessian or ☐ On floor

or ☐ Forced air draft (heated / unheated)

Duration of drying ☐ hours

Seed cleaning -> ☐ On farm ☐ By merchants

Estimated seed cleaning cost ☐ $

List of seed cleaning machinery used:

____

____

____

____

____

____

____

____

Percentage of commercial seed from harvested seed ☐ %

Storage -> ☐ On farm ☐ By merchants

Average duration of storage before use ☐ days

The germination test of commercial seed gave the following results:

At harvest ☐ % Purity

☐ % Germination 21 days

☐ No. TZ

At sale ☐ % Purity

☐ % Germination 21 days

☐ No. TZ

6. Which harvester would you prefer to buy:

One that can

☐ only harvest fluffy grass types like Buffel

☐ harvest all pasture grass seeds

☐ harvest all pasture grass seeds, wheat, oats and sorghum

7. For your personal use in grass seed harvesting, a power air-flow self-contained auto harvester should have:

A seed stripping front width (metres) of:

☐ 2m ☐ 4m ☐ 6m

☐ 8m ☐ 10m

A retail price of:

☐ $50,000 ☐ $75,000 ☐ $100,000 ☐ $150,000

8. A new tractor-mounted power air-flow harvester for grass seed should have:

A seed stripping front width (metres) of:

☐ 2m ☐ 4m ☐ 6m

☐ 8m ☐ 10m

A retail price of:

☐ $10,000 ☐ $20,000 ☐ $40,000 ☐ $60,000

APPENDIX 2

SUGGESTED SPECIFICATIONS

1.0 SPECIFICATIONS FOR PROTOTYPE (SELF DRIVEN)

Head Width	3 m	Transportable + other adv.
Track	2.5 m	Stability
Wheelbase	3.6 m	Estimate
Travel Speed	10 kph (max)	(Variable)
Tyre Width	300 mm (max)	
Clearance	1.2 m	
Operator Position	one track	
Drive	2WD	
Steering	Rear Wheels	Power Assisted
Power	60 hp	45 kW
Must be transportable		
Overall Height	2.85 m (max)	
Weight	2.5 T	

2.0 SPECIFICATIONS FOR PRODUCTION MODEL (SELF DRIVEN)

Head Width	6 m - 8 m	
Track	3 m	(max 3.5 m)
Wheelbase	5 m	

Travel Speed	10 kph	operational + road speed option
Tyre Width	0.38 m	
Clearance	1.2 m	
Operator Position	Over one track	
Drive	2WD	
Steering	Rear Wheel Power Assisted	
Power	120 hp	90 kW
Overall Height	Must be within transport standards max 4.2m	

APPENDIX 3

BUDGET FIGURES FOR MANUFACTURING COSTS

Three models were considered-

1. Production (self driven)
2. Hybrid from second hand parts (self driven)
3. Tractor mounted

1.0 BUDGET FIGURES FOR PROTOTYPE

1.1 Chassis, Accessories and Engine

1.	Wheel rim (0.28 m x 0.61 m) @ $100 x 4	$400.00
2.	Tyres (1.09 m O.D) @ $500 x 4	$2,000.00
3.	Chassis + cabin	
	(material + labour + consumable @ $1,600 + $1,600 + $100)	$3,000.00
4.	Rams @ $400 x 5	$2,000.00
5.	Steering wheel	$50.00
6.	Diesel engine + Radiator	
	(4 cylinder Isuzu 60 hp)	$6,500.00
7.	Pulleys + Bx Belts + Shafting motor	
	[$550 + (4 x $30) + $45]	$715.00
8.	Centrifugal clutch (on the pulley)	$4,000.00
	Total	**$18,965.00**

1.2 Hydraulic Systems

1.	Hydrostatic transmission + Wheel motors + Brakes	\$12,000.00
2.	Hydraulic pump to drive transmission	\$5,000.00
3.	Hydraulic valve	\$2,000.00
4.	Orbitrol steering system (Hydraulic)	\$1,000.00
5.	Piggy back pump (P.B.P)	\$1,000.00
6.	Hydraulic motor on front	\$800.00
7.	Hydraulic hoses and fittings	\$1,500.00
	Total	**\$23,300.00**

1.3 Harvester Unit

1.	Front @ \$2,000 /m, \$2,000 x 3 m	\$6,000.00
2.	Ducting	\$2,000.00
3.	Twin Fan and Separator	\$12,000.00
4.	Bin and Drying Fan	\$4,000.00
5.	Painting and cleaning	\$1,200.00
	Total	**\$25,200.00**
	Total of 1.1, 1.2 and 1.3	\$67,465.00
	Miscellaneous @ 5% (\$67,465 x 0.05)	\$3,373.25
	Grand total	**\$70,840.00**

2.0 BUDGET FIGURES FOR HYBRID FROM SECOND HAND PARTS (SELF DRIVEN MODEL)

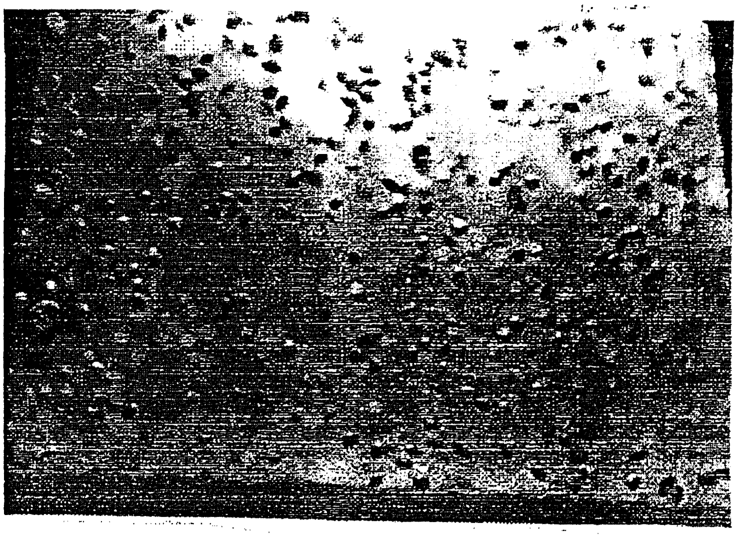
1.	Suitable machine to wreck	$8,000.00
2.	Fan and Separator (Twin 300mm)	$12,000.00
3.	Harvester Front including Hydraulics	$9,000.00
4.	Bin and Drying Fan with Hydraulic tilt and false floor	$4,000.00
5.	Ducting	$2,000.00
6.	Narrow Drive Wheels (300 - 150 mm)	$1,200.00
7.	Chassis	$2,500.00
8.	Lay Shaft and Clutch	$4,000.00
9.	Modification to Steering and controls, Front and Rear Axle, Lift mechanism	$6,000.00
10.	Hydraulic Fitting and Hoses	$800.00
11.	Cleaning and Paint	$1,200.00
	Total	**$50,700.00**
	Other charges @ 7 %	$3,300.00
	Grand Total	**$54,000.00**

3.0 BUDGET FIGURE FOR TRACTOR MOUNTED MODEL

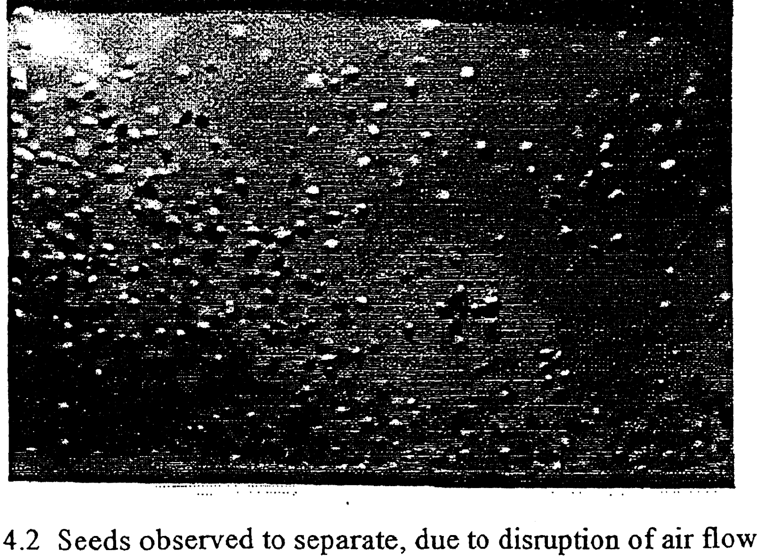
Grand Total	**$24,000.00**

APPENDIX 4

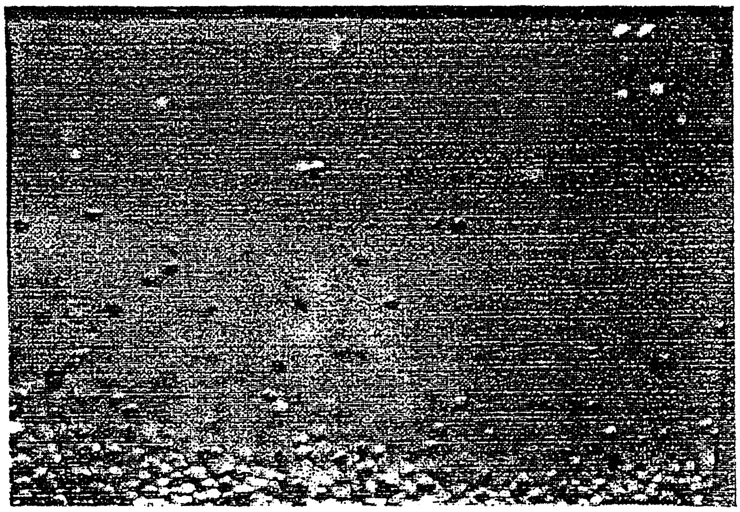
PHOTOGRAPHS OF SEED FLOW THROUGH CURVED DUCT CONCENTRATOR SYSTEM WITH GAUZE



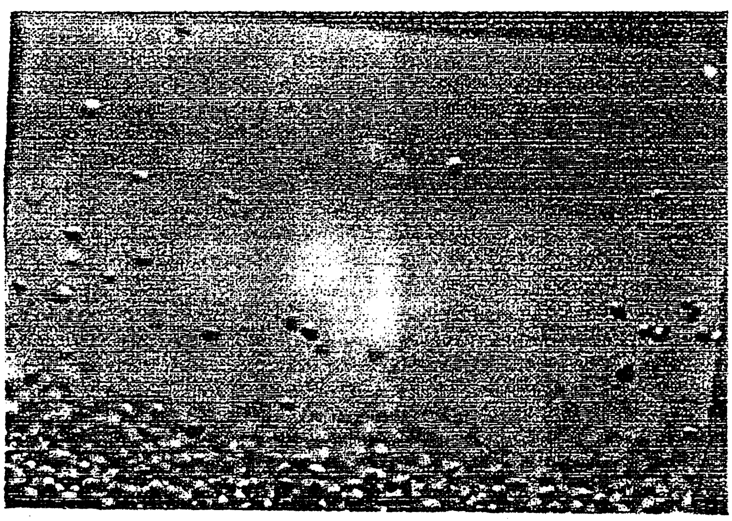
4.1 The seeds are spread as it enters the curved duct.



4.2 Seeds observed to separate, due to disruption of air flow by the escaped seeds at the Gauze (ref. 4.8.4.1)



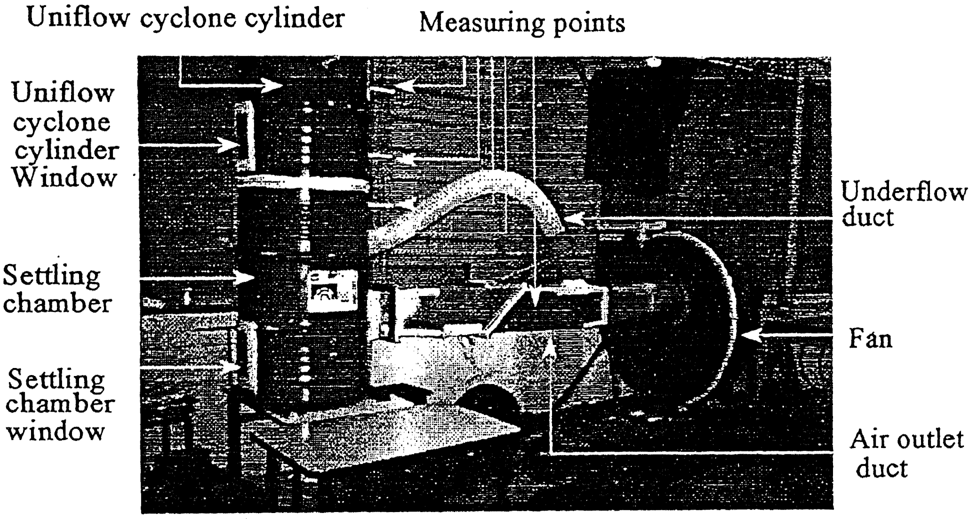
4.3 Seeds settling due to air flow blockade and also effect of gravity rather than centrifugal force.



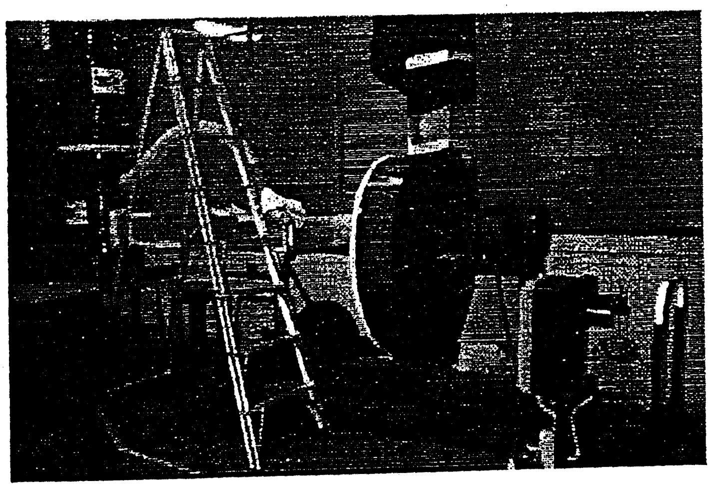
4.4 Air flow is blocked, seeds seem to just settle down on the outer curvature.

APPENDIX 5

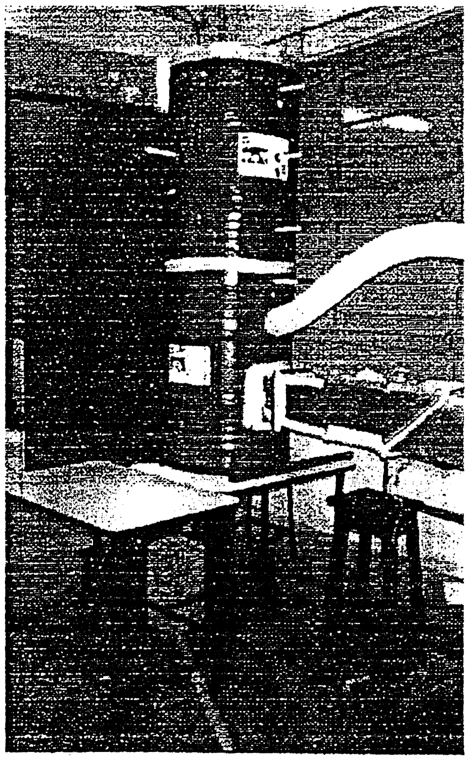
PHOTOGRAPHS OF UNIFLOW CYCLONE MODEL 1



5.1 View of the whole experimental rig of Uniflow cyclone model 1 as shown in Figure 4.29.

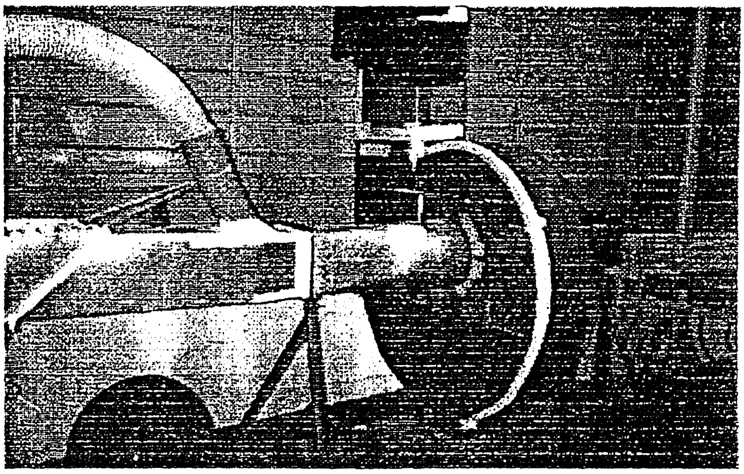


5.2 View of the whole experimental rig of Uniflow cyclone model 1 from fan side.



5.3 View of the Uniflow cyclone model I showing the underflow duct fixed right above the air outlet duct as mentioned in (2) of Results and discussion of section 4.8.5.4.

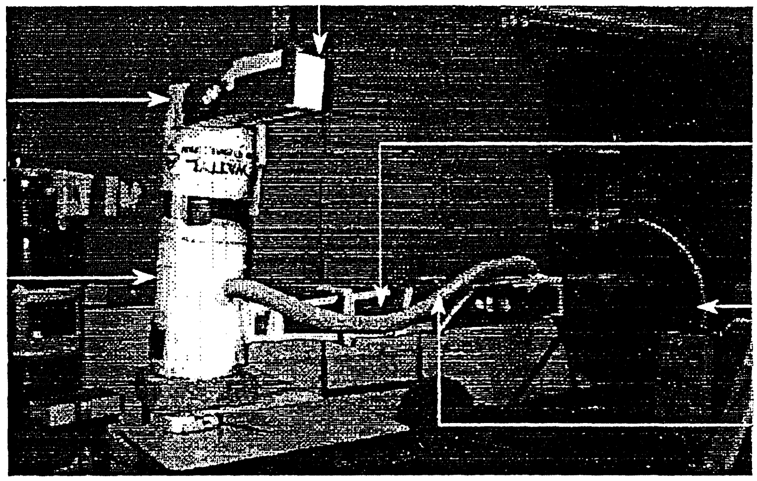
5.4 View of the windows provided in Uniflow cyclone model I for observation.



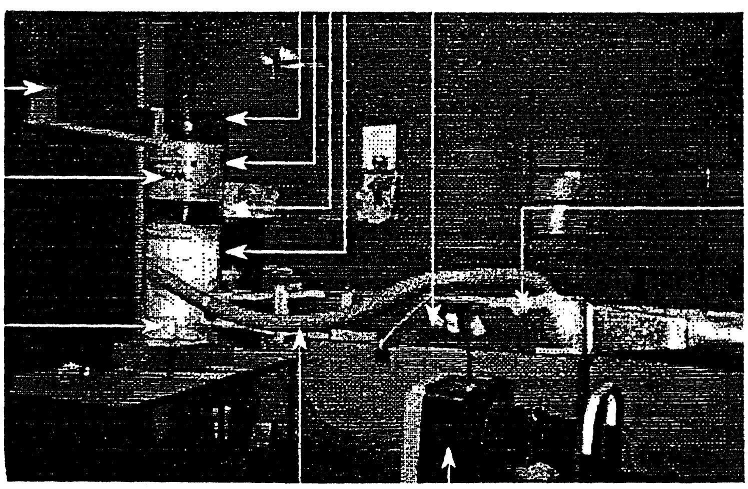
5.5 View of the underflow duct fixed with a nozzle into the air outlet duct.

APPENDIX 6

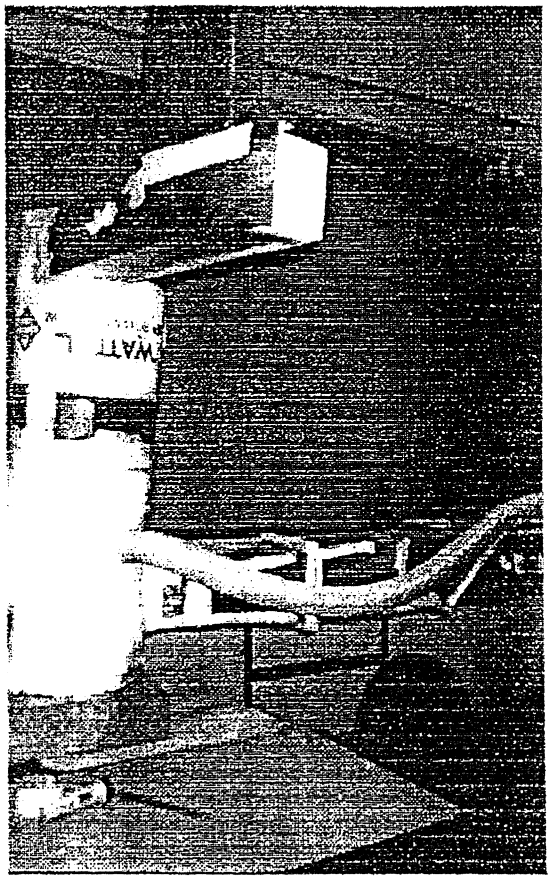
PHOTOGRAPHS OF UNIFLOW CYCLONE MODEL 2



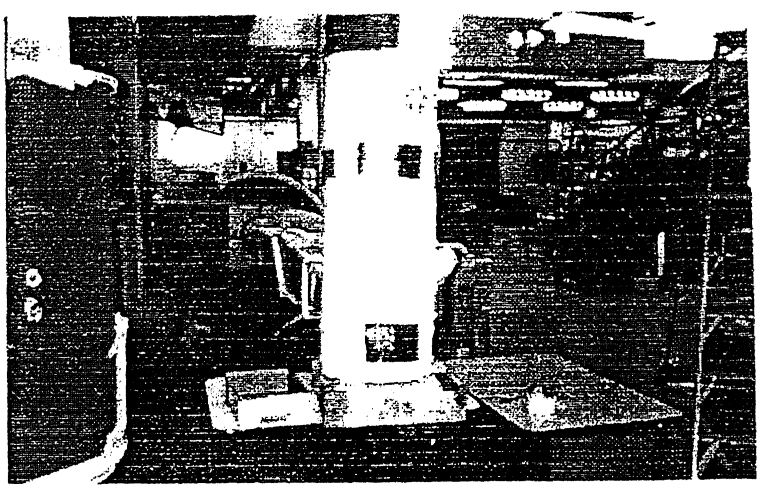
6.1 View of the whole experimental rig of Uniflow cyclone model 2 as shown in Figure 4.34.



6.2 View of the Underflow duct leading from the settling chamber into the air outlet duct with a nozzle at its end.



6.3 View of underflow duct fixed into the settling chamber to air outlet duct to maintain unobstructed suction of underflow air from the settling chamber (ref. 4.8.6.1).



6.4 View of the windows provided on the Uniflow cyclone model 2 for observation.