

DESIGN AND APPLICATION OF FEEDERS FOR THE CONTROLLED LOADING OF BULK SOLIDS ONTO CONVEYOR BELTS

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SUMMARY

The efficient operation of conveyor belts for bulk solids handling depends to a significant extent on the performance of the gravity feed system. The attainment of controlled feeding with a minimum of spillage and belt wear is of major importance. In addressing this problem, this paper focuses attention on the design requirements of the mainly used feed system comprising a gravity flow hopper, feeder and chute. The specific function of these three components is briefly outlined and the need for the hopper and feeder to be designed as an integral unit is stressed. Various types of feeders are reviewed and methods for determining feeder loads and power requirements are presented. The interaction between the flow pattern developed and wall pressures generated in mass-flow hoppers and the manner in which these influence feeder loads is discussed. A simplified methodology is presented for the design of belt feeders and feed hoppers with extended skirtplates for feeding directly onto conveyor belts. The design procedures are illustrated by example 4. Mention is made of the basic design requirements of feed chutes with particular reference to the need for careful consideration of the bulk solid flow properties and the friction characteristics of chute lining materials.

1. INTRODUCTION

Of the various modes at transporting bulk solids, belt conveying is clearly one of the most effective and reliable and well suited to handling bulk solids over a wide range of tonnage rates. The success of belt conveyors depends on a number of factors, not the least of which is the initial feeding of the bulk solid onto the belt and to the efficient transfer of solids from one belt to another at conveyor transfer stations. With the future trend towards higher belt speeds and narrower belts in order to achieve higher economic efficiency [1-8], much attention will need to be given to the design of belt feeding systems which will guarantee high feeding rates with minimum of spillage and belt wear.

While the basic objectives of an ideal feeding arrangement for loading conveyor belts are fairly obvious [9-11], it is important that they be noted. Such objectives may be summarised as follows:

- Free and uniform flow of material without segregation at a pre-determined flow rate in the same direction as the belt travel and preferably at the same speed.
- Uniform deposition of material about the centre of the belt,
- Avoidance of material spillage and dust problems.
- Minimisation of abrasive wear and impact damage.

The feeding of bulk solids onto belt conveyors is normally controlled by a gravity flow hopper/feeder combination and, in the majority of cases, the solids are finally directed onto the belt through a gravity flow chute. The feed hopper may be a part of a surge bin as in Figure 1(a) or a part of a stockpile reclaim system as in Figure 1(b). Alternatively it may be a separate dump hopper for unloading trucks or rail wagons as in Figures 1(c) and 1(d) respectively.

Feed rates are controlled by the hopper and feeder as an integral unit while the feed chute in the flow directing and feed velocity controlling device. It is important that the interactive roles of these three components as an integrated system be understood; hence some elaboration is warranted:

- Gravity Flow Hopper - The hopper geometry and internal wall friction characteristics in conjunction with the flow properties of the bulk solid establishes the type of discharge flow pattern and maximum potential rate of discharge.

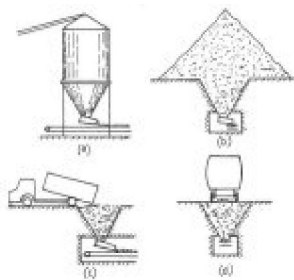


Figure 1 - Hopper/Feeder combinations for loading conveyor belts

- Feeder - The feeder controls and meters the flow of bulk material from the hopper to meet the specified discharge flow rate. The importance of the hopper and feeder to be designed as an integral unit cannot be too greatly emphasised. A well designed hopper may be prevented from functioning correctly if the feeder is poorly designed, and vice versa.

- iii. Chute - While primarily a flow directing device, the chute, when properly designed, can control the velocity of material entering the belt in a way which ensures uniform distribution of bulk material on the belt with minimum belt wear, spillage and power losses. In view of their obvious simplicity feed chutes have all too often received little attention to their design. There have been many instances where feed chutes are the "weakest link in the chain" in that lack of attention to design detail has led to major problems such as flow blockages, spillage's and accelerated belt wear.

Over recent years considerable advances have been made in the development of theories and associated design procedures for gravity flow storage and feeding systems for bulk solids handling. A selection of relevant references [9-45] are included at the end of this paper. The purpose of the paper is to review the overall requirements for designing gravity flow feeding systems for bulk solids handling with particular emphasis on the feeding operations in association with belt conveying. The paper outlines the characteristics of gravity flow storage/feeder systems, presents an overview of the more common types of feeders used and discusses the determination of feeder loads and power requirements. Brief mention is made of the role of chutes and skirtplates in directing and containing the motion of bulk solids.

2. GRAVITY FLOW OF BULK SOLIDS

While the general theories and design requirements for gravity storage and feeding systems are well documented, [9-17] it is useful to review those aspects of particular relevance to the design and operation of feeding systems for belt conveying operations.

2.1 General Design Philosophy

The design of gravity flow storage bin/feeder combinations for controlling the flow of bulk solids onto conveyor belts involves the following basic steps:

- Determination of the strength and flow properties of the bulk solids for the worst likely conditions expected to occur in practice.
- Determination of the bin geometry to give the desired capacity, to provide a flow pattern with acceptable characteristics and to ensure that discharge is reliable and predictable.
- Selection of most appropriate type of feeder and determination of feeder geometry to achieve a satisfactory flow pattern with a fully live hopper outlet.
- Estimation of loading exerted on the bin walls and the feeder under operating conditions.
- Design and detailing of the bin structure and feeder components.

It is important that all bin and feeder design problems follow the above procedures. When investigating the required bin geometry, it should be assumed that gravity will provide a reliable flow from storage. Not until it has been demonstrated that the gravity forces available are insufficient to provide reliable flow should more sophisticated reclaim methods be investigated.

2.2 Bin Flow Patterns

Following the definitions of Jenike, there are two basic modes of flow, mass-flow and funnel-flow. These are illustrated in Figure 2.

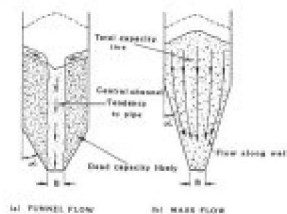


Figure 2 - Bin Flow Characteristics

In mass-flow the bulk material is in motion at substantially every point in the bin whenever material is drawn from the outlet. The material flows along the walls with the bin and hopper (that is, the tapered section of the bin) forming the flow channel. Mass-flow is the ideal flow pattern and occurs when the hopper walls are sufficiently steep and smooth and there are no abrupt transitions or inflowing valleys.

Funnel-flow (or core-flow), on the other hand, occurs when the bulk solid sloughs off the surface and discharges through a vertical channel which forms within the material in the bin. This mode of flow occurs when the hopper walls are rough and the slope angle α is too large. The flow is erratic with a strong tendency to form stable pipes which obstruct bin discharge. When flow does occur segregation takes place, there being no re-mixing during flow. It is an undesirable flow pattern for many bulk solids. Funnel-flow has the advantage of minimising bin wear.

Mass-flow bins are classified according to the hopper shape and associated flow pattern. The two main types are conical hoppers, which operate with axisymmetric flow, and wedge-shaped or chisel-shaped in which plane-flow occurs. In plane-flow bins the hopper half angle α will usually be, on average, approximately 10° larger than that for the corresponding conical hoppers. Therefore, they offer larger storage capacity for the same head room than the conical bin but this advantage is somewhat offset by the long slotted opening which can cause feed problems.

The limits for mass-flow depend on the hopper half angle α , the wall friction angle ϕ and the effective angle of internal friction δ . The relationships for conical and wedge-shaped hoppers are shown in Figure 3. In the case of conical hoppers the limits for mass-flow are clearly defined and quite severe while the plane-flow or wedge-shaped hoppers are much less severe. Consider, for example, a conical hopper handling coal with $\delta = 45^\circ$. If the hopper is of mild steel it is subject to corrosion, the angle ϕ is likely to be approximately 30° . On this basis, from Figure 3, the limiting value of $\alpha = 13^\circ$. A margin of 3° is normally allowed making $\alpha = 10^\circ$ which is a very steep hopper. If the hopper is lined with stainless steel, the friction angle ϕ is likely to be approximately 20° . On this basis $\alpha = 26 - 3 = 23^\circ$. This results in a more reasonable hopper shape. The corresponding angles for plane-flow are $\alpha = 22^\circ$ for $\phi = 30^\circ$ and $\alpha = 35^\circ$ for $\phi = 20^\circ$.

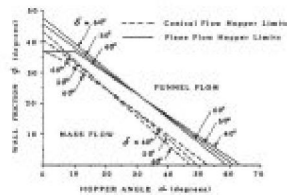


Figure 3 - Limits for mass-flow in conical hoppers

Funnel-flow bins are characterised either by their squat hopper proportions or their flat bottoms. For funnel-flow bins to operate satisfactorily is necessary for the opening size to be at least equal to the critical pipe dimension D_f . This will ensure that the material will not form a stable pipe or rathole but rather will always collapse and flow. However, for many materials the minimum pipe dimension D_f is too large, rendering funnel-flow bins impracticable. This is certainly the case, for example, with most coals and mineral ores which, at higher moisture levels, are known to have critical pipe dimensions of several metres.

Where large quantities of the bulk solid are to be stored, the expanded-flow bin, as illustrated in Figure 4 is often an ideal solution. This bin combines the storage capacity of the funnel-flow bin with the reliable discharge characteristics of the mass-flow hopper. It is necessary for the mass-flow hopper to have a diameter at least equal to the critical pipe dimension D_f at the transition with the funnel-flow section of the bin. This ensures that the flow of material from the funnel-flow or upper section of the bin can be fully expanded into the mass-flow hopper. The expanded-flow bin concept may also be used to advantage in the case of bins or bunkers with multiple outlets.

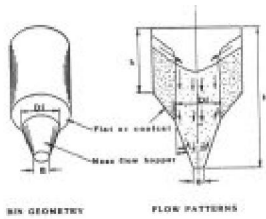


Figure 4 - Expanded-Flow bin

The expanded flow concept may also be employed in gravity reclaim stockpile systems such as that illustrated in Figure 5.

As a general comment it is noted that symmetrical shaped bins provide the best performance. Asymmetric shapes often lead to segregation problems with free flowing materials of different particle sizes and make the prediction of wall loads very much more difficult and uncertain.

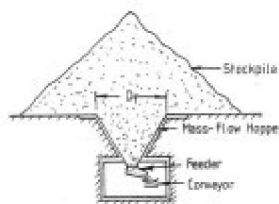


Figure 5 - Expanded flow principal in stockpile reclaim

2.3 Potential Flow Rate from Mass-Flow Bins

The flow rate from mass-flow bins depends on the hopper geometry, the wall lining material and the flow properties of the bulk solid. However, for a well designed mass-flow bin, the discharge is uniform and the flow rate predictable. To illustrate the influence of mass-flow hopper geometry on the potential unrestricted flow rate, consider the example given in Figure 6. This applies to a typical coal at 15.2% moisture content (dry basis) where the hopper lining material is stainless steel type 304 with 2B finish. The graphs show the :

- i. variation in hopper half angle α at a function of opening hopper dimension B and
- ii. the corresponding variation in flow rate for unrestricted discharge.

The full lines refer to a conical axi-symmetric-shaped hopper while the chain- dotted lines refer to a plane-flow wedged shape hopper. In the latter case the flow rate is given in tonnes/hr x 10³ per metre length of slot. The graphs indicate the following:

- The hopper half angle α , for mass-flow increases initially with increase in opening size B and then approaches a constant value.
- The hopper half angle for a plane-flow hopper is approximately 12° larger than that for a conical hopper for the same outlet dimension B,
- The potential flow rate depending on the outlet dimension is quite significant. While with train loading operations where flood loading is required there is a need for high discharge rates, for the majority of cases the potential flow rate may be excessive rendering the need to employ feeders as flow controlling devices.

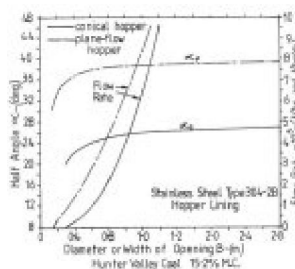


Figure 6 - Mass-Flow hopper geometry's and flow rates for typical coal

3. FEEDERS FOR BULK SOLIDS HANDLING

3.1 General Remarks

Feeders for controlling the flow of bulk solids onto conveyor belts require certain criteria to be met:

- Deliver the range of flow rates required.
- Handle the range of particle or lump sizes and flow properties expected.
- Deliver a stable flow rate for a given equipment setting. Permit the flow rate to be varied easily over the required range without affecting the performance of the bin or hopper from which it is feeding.
- Feed material onto the belt in the correct direction at the correct speed with the correct loading characteristic and under conditions which will produce minimum impact, wear and product degradation. Often a feed chute is used in conjunction with the feeder to achieve these objectives.
- Fit into the available space.

It is important that the flow pattern be such that the whole outlet of the feed hopper is fully active. This is of fundamental importance in the case of mass-flow hoppers. When feeding along slotted outlets in wedge-shaped hoppers the maintenance of a fully active outlet requires the capacity of the feeder to increase in the direction of feed. To achieve this condition special attention needs to be given to the design of the outlet as vertical skirts and control gates can often negate the effect of a tapered outlet. Gates should only be used as flow trimming devices and not as flow rate controllers. Flow rate control must be achieved by varying the speed of the feeder.

Noting the foregoing comments, the salient aspects of the various types of feeders commonly used to feed bulk solids onto belt conveyors are now briefly reviewed.

3.2 Vibratory Feeders

3.2.1 General Remarks

Vibratory feeders are used extensively in controlling the discharge of bulk solids from bins and stockpiles and directing these materials onto conveyor belts. They are especially suitable for a broad range of bulk solids, being able to accommodate a range of particle sizes and being particularly suitable for abrasive materials. However they are generally not suited to fine powders under 150 to 200 mesh where flooding can be a problem. Also 'sticky' cohesive materials may lead to build-up on the pan leading to a reduction in flow rate.

Bulk solids are conveyed along the pan of the feeder as a result of the vibrating motion imparted to the particles as indicated in Figure 7.

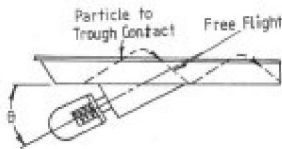


Figure 7 - Movement of particles by vibration

The pan of the feeder is driven in an approximate sinusoidal fashion at some angle θ to the trough.

The conveying velocity and throughput depend on the feeder drive frequency, amplitude or stroke, drive angle and trough inclination, coefficient of friction between the bulk solid and the pan as well as the bulk solid parameters such as bulk density, particle density and general flow properties.

3.2.2 Types of vibrating Feeders

In general vibrating feeders are classified as 'brute force' or 'tuned' depending on the manner in which the driving force imparts motion to the pan.

As the name implies 'brute force' type feeders involve the application of the driving force directly to the pan as illustrated in Figure 8. These feeders have the following characteristics.

- Lower initial cost but higher operating costs.
- Greater forces to be accommodated in the design.
- Impact loads on the pan are transmitted to bearings on which out-of-balance weights rotate.
- Delivery rates are dependent on the feeder load due to bulk solids.
- Generally confined to applications requiring only one feed rate.

On the other hand 'tuned' vibrating feeders are more sophisticated in their operation in as much as the driving force is transmitted to the pan via connecting springs as indicated in Figure 9. In this way they act essentially as a two mass vibrating system and employ the principle of force magnification to impart motion to the pan. The primary driving force is provided by either an electromagnet or by a rotating out-of-balance mass system.

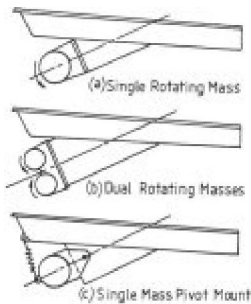


Figure 8 - 'Brute' force type vibratory feeders Ref. [18]

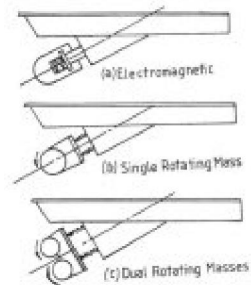


Figure 9 - Tuned type vibratory feeders Ref. [18]

Following the work of Rademacher [19] some general comments may be made. Normally the trough mass is designed to be 2.5 to 3 times as large as the exciter mass. Figure 10 shows typical magnification curves for the tuned feeder for two damping ratios ξ_1 and ξ_2 . Normally the feeders operate below the resonance frequency with $\omega/\omega_0 = 0.9$ where ω = driving frequency and ω_0 = natural frequency of the system. It is often claimed that the 'tuned' feeder maintains its feed rate when the head load varies. The validity of this claim may be examined by reference to Figure 10. The increased head load effectively increases the pan mass lowering the natural frequency ω_0 and increasing the frequency ratio ω/ω_0 . This corresponds to a shift from A to B in the diagram. At the same time, the damping increases due to the increased load resistance causing a shift from B to C. Thus the claim that the feeder maintains its performance regardless of the head load, basically, is not true. However in many instances the negative effect of the increase in damping approximately compensates for the change in magnification factor. This means that points A and C in Figure 10 are approximately of the same magnification factor so that the trough stroke is kept approximately constant.

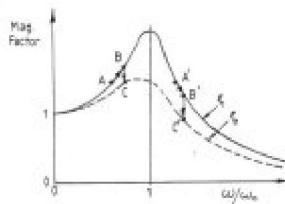


Figure 10 - Typical magnification curve for a tuned two mass system $E_1 < E_2$ Ref.[19]

A similar analysis in the case of the over critical operation with $\omega/\omega_0 > 1$ indicates that increases in ω/ω_0 and ξ leads to a smaller, magnification factor corresponding to a smaller stroke. For this reason over critical operation of this type of feeder must be avoided.

3.2.3 Hopper/Feeder Configurations

There are several aspects to note when designing feed hoppers for use with vibrating feeders. These are discussed at some length by Colijn and Carroll [20]. Some important aspects are noted here.

Figure 11 shows a typical vibrating feeder arrangement. The effectiveness of the feeder, as with all feeders, depends largely on the hopper which must be capable of delivering material to the feeder in an uninterrupted way.

For a symmetrical hopper there is a tendency for the feeder to draw material preferentially from the front of the hopper. Uniform draw can be achieved by making the hopper outlet asymmetrical with the back wall at the correct hopper half angle α and the front wall at an angle of $\alpha + (5^\circ \text{ to } 8^\circ)$.

(The angle α is obtained from Figure 3). Alternatively a symmetrical hopper may be made to feed approximately uniformly by using a rougher lining material on the front face. Other recommendations include

- Dimension E to be at least 150 cm.
- B to be large enough to prevent arching or ratholing.
- Slope $\emptyset$ to be sufficient for the required flow rate
- Gate height H to be chosen primarily to achieve an acceptable flow pattern rather than to vary the flowrate.
- For high capacity feeders skirtplates extending to the outlet of the trough may be required as in Figure 12.

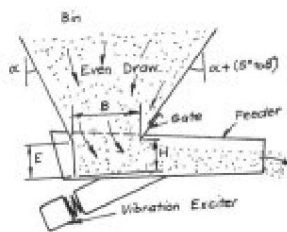


Figure 11 - Typical arrangement for vibratory feeder

In the case of wedge-shaped plane-flow bins problems arise when it is necessary to feed from the long slotted outlets. It is always theoretically better to feed across the slot as in Figure 13(a) but the cost of wide feeders to achieve this goal often becomes prohibitive. A better solution may be to use a multi-outlet arrangement as in Figure 13(b) and employ several narrower vibrating feeders to feed across the slot as shown.

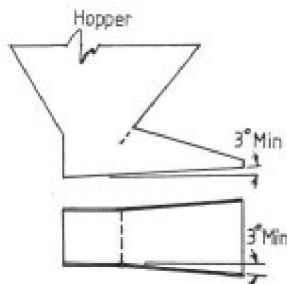


Figure 12 - Tapered skirts Ref. [20]

Where it is necessary to feed along the slot as in Figure 14 (a) , then tapering of the outlet in the direction of the feed is required as indicated in Figure 14(b). Once again it is reiterated that the adjustable gate is a flow trimming device rather than a flow rate controlling device.

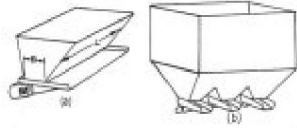


Figure 13 - Alternative arrangements for feeding across the slot wedge-shaped hopper

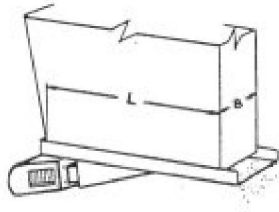


Figure 14 - Arrangement for feeding along slot of wedge-shaped hopper Ref. [18]

3.3 Belt Feeders

Belt feeders are used to provide a controlled volumetric flow of bulk solids from storage bins and bunkers. They generally consist of a flat belt supported by closely spaced idlers and driven by end pulleys as shown in Figure 15. In some cases, hoppers feed directly onto troughed conveyors as in the case of dump hoppers used in conjunction with belt conveyors.

Some particular features of belt feeders include

- Suitable for withdrawal of material along slotted hopper outlets when correctly designed.
- Can sustain high impact loads from large particles.
- Flat belt surfaces can be cleaned quite readily allowing the feeding of cohesive materials.
- Suitable for abrasive bulk solids.
- Capable of providing a low initial cost feeder which is dependable on operation and amenable to automatic control.

With respect to the first point, the hopper and feeder geometry for long slots are critical if uniform draw is to be obtained. While normally feeders are installed horizontally, on some occasions a feeder may be designed to operate at a low inclination angle β . The outlet should be tapered as shown in the plan view of Figure 16. Research [21-24] has shown that the taper angle ϕ and downslope angle β together with the gate opening H are very sensitive as far as obtaining efficient performance is concerned. In particular, as stated previously, the gate opening H should be used to train the flow pattern and not to control the flow rate. As has been demonstrated by experiment [23], incorrect setting of the gate will cause non uniform draw with funnel-flow occurring either down the back wall or down the front wall. In one series of experiments using a free flowing granular type material, merely increasing the gate setting H causes the flow to move progressively towards the front. The final gate setting needs careful adjustment if uniform draw is to be achieved. Thus in belt feeders flow rate variations must be achieved by varying the belt speed. This requirement places some limitations on belt feeders when very low flow rates are required, especially if the bulk solid is at all cohesive or contains large lumps.

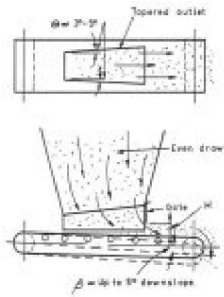


Figure 15 - Arrangement for belt feeder

Particular care is needed with the design of the hopper/feeder arrangement when handling fine powders in order to ensure that problems of flooding are avoided. If the bulk material tends to stick to the belt, spillage may be a problem with belt feeders. Therefore if sufficient headroom is available, it is desirable to mount the feeder above the belt conveyor onto which it is feeding material in order that any material falling from the return side of the belt will automatically fall onto the conveyor belt.

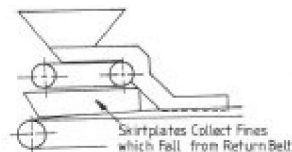


Figure 16 - Belt feeder sited above conveyor to minimise spillage Ref. [9]

Belt feeders can also have applications where a short speed-up belt is used to accelerate the material at the loading point of a high speed conveyor as illustrated in Figure 17. The accelerating conveyor avoids wear that would otherwise occur to the cover of the long conveyor.

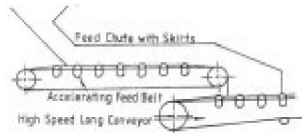


Figure 17 - Belt feeder as acceleration conveyor Ref. [25]

3.4 Apron Feeders

Apron feeders are a version of belt feeders and are useful for feeding large tonnages of bulk solids being particularly relevant to heavy abrasive ore type bulk solids and materials requiring feeding at elevated temperatures. They are also able to sustain extreme impact loading. The remarks concerning the need for uniform draw and gate settings applicable to belt feeders are also applicable to apron feeders. Figure 18(a) shows an apron feeder with parallel outlet which is inducing funnel-flow down the rear wall of the hopper. Apart from the obvious flow problems, the funnel-flow pattern developed will accelerate the wear down the rear wall. The tapered outlet of Figure 18(b), when correctly designed will induce uniform draw, minimising segregation and minimising hopper wall wear.

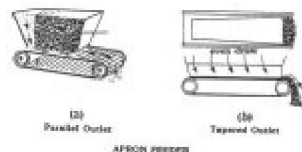


Figure 18 - Apron feeders

3.5 Plough Feeders

Rotary plough feeders are generally used in long reclaim tunnels under stockpiles where they travel along the tunnel as in Figure 19 or fixed under stockpiles and large storage bins, as shown in Figure 20.

In the case of the stockpile slot reclaim system of Figure 19, it is necessary for the diagonal dimension of the slot to be at least equal to the critical rathole diameter D_f of the

bulk solid in order to prevent ratholes from forming under the high storage pressures [15]. In this way the gravity reclaim efficiency is maximised. The tie beams between slots should be steeply capped. Furthermore the slot width B_f must be large enough to prevent arching.

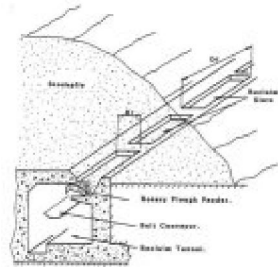


Figure 19 - Typical stockpile with paddle feeder reclaim system

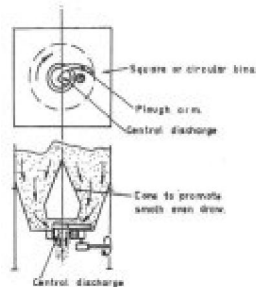


Figure 20 - Fixed plough feeder

The basic concept of the traveling plough feeder is to allow bulk solids to flow by gravity onto a stationary shelf and then remove the solids from the shelf either with a linear drag plough or a traveling rotary plough. It is important that high penetration of the plough is achieved and that there is a small vertical section behind the plough to prevent material build-up on the sloping back wall. A high penetration rotary plough feeder is shown in Figure 21.

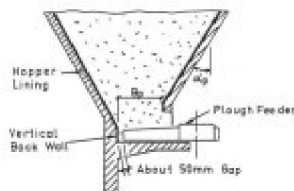


Figure 21 - High penetration plough feeder

Colijn and Vitunac [26] have reviewed the application of plough feeders in some detail. They recommend the following limiting values for plough feeders:

- a. Rotary Plough
 - Tip speed < 20 m/s
 - Tip diameter range 1.8 to 4 m
 - Carriage speed 0.01 to 0.08 m/s (does not contribute significantly to capacity).
- b. Linear Drag Plough
 - Traversing speed 0.13 to 0.76 m/s
 - Cut width 1.4 to 2.3 m.

3.6 Rotary Table Feeders

The rotary table feeder can be considered as an inverse of the plough feeder. It consists of a power driven circular plate rotating directly below the bin opening, combined with an adjustable feed collar which determines the volume of bulk material to be delivered. A typical rotary feeder arrangement is shown in Figure 22. The aim is to permit equal quantities of bulk material to flow from the complete bin outlet and spread out evenly over the table as it revolves. The material is then ploughed off in a steady stream into a discharge chute.

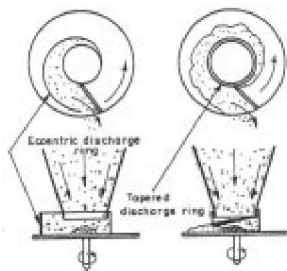


Figure 22 - Rotary table feeder

This feeder is suitable for handling cohesive materials which require large hopper outlets, at flow rates between 5 and 125 tonnes per hour. Feed rates to some extent are dependent on the degree to which the material will spread out over the table. This is influenced by the angle of repose of the material which varies with moisture content, size distribution and consolidation. These variations prevent high feed accuracy from being obtained.

Rotary table feeders are suitable for bin outlets up to 2.5 m diameter; the table diameter is usually 50 to 60% larger than the hopper outlet diameter. With some materials a significant dead region can build up at the centre of the table. This can sometimes be kept from becoming excessive by incorporating a scraping bar across the hopper outlet. It is important to ensure that the bulk material does not skid on the surface of the plate, severely curtailing or preventing removal of the bulk material.

3.7 Screw Feeders and Dischargers

3.7.1 Screw Feeders

Screw feeders are widely used for bulk solids of low or zero cohesion such as fine and granular materials which have to be dispensed under controlled conditions at low flow rates. However, as with belt feeders, design difficulties arise when the requirement is to feed along a slotted hopper outlet, Figure 23. An equal pitch, constant diameter screw has a tendency to draw material from the back of the hopper as in Figure 23(a). To counteract this, several arrangements are advocated for providing an increasing screw capacity in the direction of feed as in Figure 23(b) to (f). The arrangements shown are:

- Stepped pitch
- Variable pitch
- Variable pitch and diameter
- Variable shaft diameter.

Pitch variation is generally limited to a range between 0.5 diameters minimum to 1.5 diameters maximum. This limits the length to diameter ratio for a screw feeder to about six, making them unsuitable for long slots.

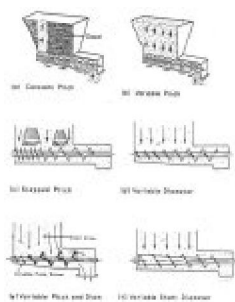


Figure 23 - Screw feeding along a slot

The section of the screw leading from the hopper to the feeder outlet is fundamental in determining the quantity of material discharged per revolution of the screw. At the point where the screw leaves the hopper, it is essential for control purposes to cover the screw,

normally by a 'choke' section having the same radial clearance as the trough. This choke section should extend for at least one pitch to prevent material cascading over the flights.

As a screw feeder relies on friction to transport material it has a very low efficiency in terms of the energy requirements. Furthermore, the volumetric efficiency is impaired somewhat due to the rotary motion imparted to the bulk material during the feeding operation [27].

Since screw feeders are generally fully enclosed, relatively good dust control is achieved. However due to the high frictional losses abrasive type bulk solids can effectively reduce the life of the feeder due to abrasive wear. Fine powders that tend to flood are difficult to control in a screw feeder in flooding situations.

3.7.1 Screw Dischargers

Screw dischargers are variations of the normal screw feeder. Two of the more commonly used versions are shown in Figure 24. Figure 24(a) shows a single screw which is forced to circle slowly around the bottom of a flat bottom storage silo. The screw rotates at the same time and slices the bulk material, transferring it to a central discharge chute. In Figure 24(b) the whole floor of the silo rotates about a fixed axis. The bulk material is forced against the rotating screw as the silo bottom rotates.

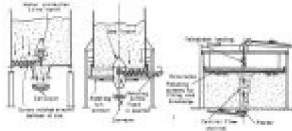


Figure 24 - Various screw discharge arrangements

Screw discharges have been used successfully with some wet, sticky, bulk solids which have not been handled effectively using other means. In addition to providing the necessary flow promotion, these devices also control the feed rate. Problems could arise when devices of this type suffer breakdown need careful investigation when considering a screw discharge device for use in a particular application.

An alternative application of screw discharges is in the Eurosilo [28] which is shown schematically in Figure 24(c). This design of silo, originated in the Netherlands and, as indicated, the screws sweep around the top surface drawing material to the central discharge channel. The screws are also used to distribute the bulk material during filling. The Eurosilo was originally developed as an inexpensive storage facility, for potato starch but it is now being used for other bulk materials, notably coal. It provides a very large capacity, environmentally clean storage facility. Its principal disadvantage is that it

operates on a first-in last-out sequence and hence is not recommended for materials that degrade with time.

3.8 Rotary Feeders

Rotary feeders (also known as drum, vane, star and valve feeders) are generally used for the volumetric feeding of fine bulk solids which have reasonably good flowability.

A rotary drum feeder, Figure 25(a), might be considered an extremely short belt feeder. The drum prevents the bulk material from flowing out but discharges it by rotation. This feeder is only suitable for materials with good flowability which are not prone to aeration. Similar considerations apply to the rotary vane feeder, Figure 25(b), which might be considered as an extremely short apron feeder; Figure 25(c) shows some modifications to the vane. The rotary valve feeder, Figure 26(a), is completely enclosed and aims at preventing powders or fine grained materials from flooding. The star feeder, Figure 26(b), provides a means for obtaining uniform withdrawal along a slot opening.

These feeders are not suitable for abrasive bulk materials as clearances cannot be maintained and the feeders tend to lose control especially when handling aerated powders. Cohesive powders will tend to clog the rotor pockets and reduce feeder capacity.

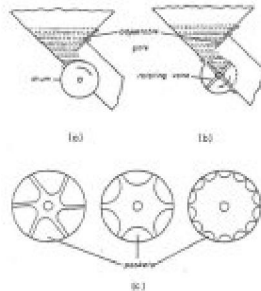


Figure 25 - Rotary drum and vane feeders with various rotating elements Ref [18]

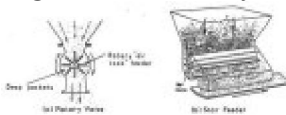


Figure 26 - Rotary valve & star feeders

3.9 Feeder Selection

The selection of a feeder for a particular situation is not always simple, especially if more than one satisfactory solution appears possible. The type and size of feeder for a given application is primarily dictated by the characteristics of the bulk material to be handled and the required capacity. Some general guidelines on feeder selection are given in References [18,19].

4. FEEDER LOADS AND POWER REQUIREMENTS

4.1 General Remarks

From a design point of view it is important to be able to determine with some accuracy the loads acting on feeders in hopper/feeder combinations and the corresponding power requirements. Yet the state-of-the-art has, in the past, been such that the loads and power requirements could not be estimated with any degree of precision. For instance Wright [29] has observed that the majority of formulae published are empirical in nature and derived to predict loads and corresponding power requirements for feeders used in conjunction with funnel-flow bins. These formulae are inadequate when applied to mass-flow bins since, in such cases, the loads and power requirements are often greatly underestimated. This is largely due to the fact that in mass-flow bins the full area of the hopper outlet is presented to the feeder.

The loads acting on feeders can vary considerably. There are many reasons for this, some more obvious than others. As indicated by Reisner and Rothe [18], the shape of the hopper outlet will influence the load on a feeder as illustrated in Figure 27. In Figure 27(a), the full load (not equal to the hydrostatic head) acts on the feeder. In Figure 27(b) the load is partly reduced by changing the shape of the hopper. In Figure 27(c), the load is completely removed from the feeder and only acts on the hopper wall. Although the advantages of Figure 27(b) and (c) appear obvious, the solution may not be as simple as that depicted. It is clear that the flow pattern developed in the feeding operation must be such that uniform, non-segregated flow is achieved at all times.

The loads acting on feeders and corresponding power requirements are influenced by several factors. These include the following:

- Hopper flow pattern, whether mass-flow or funnel-flow
- Flow properties of the bulk solid
- The chosen hopper shape which in the case of mass-flow includes axi-symmetric or conical, plane-flow or transition (combination of conical and plane-flow)
- The actual hopper geometry
- The wall friction characteristics between the bulk solid and hopper walls and skirtplates
- The type of feeder and its geometrical proportions

- The initial filling conditions when the bin is filled from the empty condition and the flow condition when discharge has occurred.

The most efficient and reliable feeding performance is achieved by using a mass-flow hopper/feeder combination. For a given bulk solid and hopper/ feeder geometry the load acting on a feeder varies considerably between the initial load, when the bin is first filled, and the load either during flow or after flow has stopped. Reisner [18] has indicated that the initial load can be 2 to 4 times the flow load. However, research [21,22,24] has shown that the variation is much greater than this with the initial loads of the order of 4 to 8 times that of the flow load. Theoretical predictions show that circumstances can arise whereby the initial/flow load variations can be much higher than those indicated.

A procedure for estimating feeder loads for mass-flow/feeder combinations has been established [13,30]. The general procedure is now reviewed.

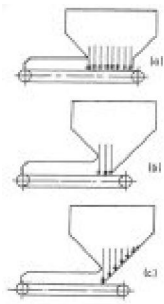
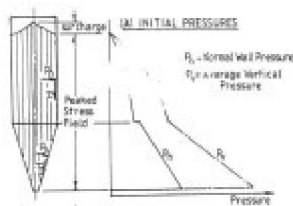


Figure 27 - Varying the load on the feeder by varying the hopper configuration [18]

4.2 Pressure Distributions in Mass-Flow Bins

It is first necessary to examine the pressures acting in mass-flow bins under both initial filling and flow conditions. The pressures in mass-flow bins are discussed in some detail in References [13,31-37].

Figure 28 shows the bin stress fields and corresponding pressure distributions for the initial filling and flow cases. In each case, p_n represents the pressure acting normal to the bin wall while p_v represents the average vertical pressure.



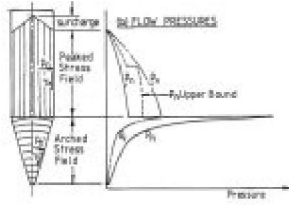


Figure 28 - Pressures acting in mass-flow bins

4.2.1 Initial Filling Case - Figure 28(a)

In this case, vertical support is provided and the major consolidating principal pressure is nearly vertical. A peaked stress field exists in both the cylinder and hopper as indicated.

(i) Cylinder

The normal wall pressure p_n is given by the Janssen equation with an initial surcharge pressure term included.

$$p_n = \frac{\gamma R}{\mu} [1 - e^{-\mu K_j h/R}] + p_{no} e^{-\mu K_j h/R} \quad (1)$$

$$\text{where } R = \frac{D}{2(1+m)} = \text{Hydraulic radius} \quad (2)$$

γ	=	Bulk	Specific	Weight
D	=	Cylinder	diameter	or width
m	=	1	for axi-symmetric	or circular bin
m	=	0	for long rectangular	plane-flow bin
μ	=	$\tan \theta$	= coefficient of wall friction	friction angle
θ	=		friction	

P_{no} = initial pressure at top contact point with wall due to natural surcharge of material caused by filling.

The coefficient K_j relates the normal pressure p_n to the average vertical p_v . That is

$$K_j = p_n/p_v \quad (3)$$

The value of K_j varies depending on the cylinder geometry. For parallel sided cylinders with slight convergence's Jenike suggests

$$K_j = 0.4 \quad (4)$$

For continuously diverging or stepwise diverging cylinders without convergence's, the theoretical value of K_j may be used

$$K_j = \frac{1 - \sin \delta}{1 + \sin \delta} \quad (5)$$

Where δ = effective angle of internal friction for the bulk solid

For continuously converging cylinders K_j is approximated by

$$K_j = 1$$

The initial surcharge pressure p_{no} may be estimated by

$$p_{no} = K_j \gamma h_s \quad (6)$$

Where h_s = effective surcharge.

h_s depends on the bin shape and manner in which the bin is loaded. Assuming central loading, then for an axi-symmetric bin a conical surcharge is assumed; for a plane-flow bin in which the length is greater than the width, an approximate triangular shape would occur if a travelling feeding arrangement, such as a tripper, is used. With these two limits

$$h_s = \frac{H_s}{m_s + 2} \quad (7)$$

where $m_s = 0$ for triangular surcharge
 $m_s = 1$ for conical surcharge
 H_s = actual surcharge

For a plane flow bin, some rounding of the ends of the top surface of material is likely to occur so that an approximate intermediate value of m_s can be used.

The Janssen curves for the cylinder are shown in Figure 28(a).

(ii) Hopper

Following Jenike [37], the normal pressure acting against the wall is related to the average vertical pressure by a parameter K such that

$$K = \frac{P_n}{P_v} \quad (8)$$

$$p_v$$

From an equilibrium analysis for the hopper the following differential equation is obtained.

$$\frac{dp_n}{dz} + \frac{np_n}{(h_o-z)} = \gamma K \quad (9)$$

Solution of this equation leads to

$$p_n = \gamma K \left\{ \left(\frac{h_o-z}{n-1} \right) + \left[h_c - \frac{h_o}{n-1} \right] \left[\frac{h_o-z}{h_o} \right]^n \right\} \quad (10)$$

$$\text{where } n = (m+1) \left\{ K \left(1 + \frac{\mu}{\tan \alpha} \right) - 1 \right\} \quad (11)$$

α = hopper half angle

h_c = surcharge head acting at transition of cylinder and hopper

h_o = distance from apex to transition

h_c is given by

$$h_c = \frac{Q_c}{\gamma A_c} \quad (12)$$

where $\frac{Q_c}{A_c}$ is derived from the Jansen Equation (1)

That is

$$\frac{Q_c}{A_c} = \frac{\gamma R}{\mu K_j} [1 - e^{-\mu K_j H/R}] + \gamma h_{se} e^{-\mu K_j H/R} \quad (13)$$

where Q_c = surcharge force at transition

A_c = area of cylinder

H = height of material in contact with cylinder walls.

For the initial filling condition in the hopper, Jenike [37] assumes that the average vertical pressure distribution in the hopper follows the linear hydrostatic pressure distribution. For this condition the value of K in (10) is the minimum value. That is

$$K = K_{\min} = \frac{\tan \alpha}{\mu + \tan \alpha} \quad (14)$$

Substitution into (11) yields $n = 0$.

Hence (10) becomes

$$P_n = \gamma K_{\min}(h_c + z) \quad (15)$$

and

$$P_v = \frac{P_n}{K_{\min}} = \gamma(h_c + z) \quad (16)$$

The linear relationships for p_n and p_v for the hopper are shown in Figure 28(a).

4.2.2 Flow Case - Figure 28 (b)

In this case with the vertical support removed, the load is transferred to the hopper walls and the peaked stress field switches to an arched stress field. As flow is initiated from the hopper the switching from a peaked to arched stress field commences at the hopper outlet and travels upward. There is evidence to suggest that the switch becomes locked at the transition and remains there during continuous flow. In the arched stress field, the major principal pressure acts more in the horizontal direction.

i. Cylinder

During the flow, the peaked stress field remains in the cylinder and for a perfectly parallel cylinder without localised convergences the Janssen stress field remains. However, in practice, localised convergences are difficult to avoid, such convergences being caused by weld projections or plate shrinkages in the case of steel bins or irregular formwork in the case of concrete bins. For this reason an upper bound normal wall pressure based on Jenike's strain-energy stress field [37] is computed for design purposes. The modified upper bound cylinder normal wall pressures is shown in Figure 28.

ii. Hopper

The normal wall pressures for flow in the hopper are given by equation (10) with K set at the maximum value

$$K = K_{\max} \quad (17)$$

where $K_{\max}$ is given in graphical form in Reference [37] or else may be computed using equations derived in Reference [13]. That is

$$K_{\max} = \left[\frac{\sigma_w}{\gamma B} / q \left(\frac{4}{\pi} \right)^m \right] \quad (18)$$

$$\text{Where } \frac{\sigma_w}{\gamma B} = \frac{Y(1+\sin\delta \cos 2\beta)}{2(x-1)\sin\alpha} \quad (19)$$

$$\text{and } q = \left(\frac{\pi}{3} \right)^m \frac{1}{4\tan\alpha} \left\{ 2 \left(\frac{\sigma_w}{\gamma B} \right) (\tan\alpha + \tan\phi) - \frac{1}{1+m} \right\} \quad (20)$$

$$\text{Where } \beta = \frac{1}{2}[\phi + \sin^{-1} \left(\frac{\sin\phi}{\sin\delta} \right)] \quad (21)$$

$$x = \frac{2^m \sin\delta}{1-\sin\delta} \left[\frac{\sin(2\beta+\alpha)}{\sin\alpha} + 1 \right] \quad (22)$$

$$y = \frac{[2(1-\cos(\beta+\alpha))]^m (\beta+\alpha)^{1-m} \sin\alpha + \sin\beta \sin^{1+m}(\beta+\alpha)}{(1-\sin\delta) \sin^{2+m}(\beta+\alpha)} \quad (23)$$

$\phi = \tan^{-1}\mu$ where ϕ = wall pressure angle
 $\delta =$ effective angle of internal friction
 $m =$ 1 for axi-symmetric flow
 $= 0$ for plane-flow.

In equation (22) the plane-flow numerator term $(\alpha+\beta)$ must be in radians.

4.3 Theoretical Estimate of Feeder Loads and Power

The theoretical prediction of the surcharge load acting at the outlet of a mass-flow hopper requires consideration of both the initial and flow consolidation pressures acting on the bulk material. Figure 29 shows diagrammatically the loads acting in a hopper feeder combination.

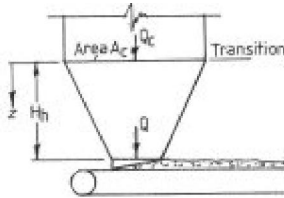


Figure 29 - Feeder loads in hopper/feeder combination

Following the approach adopted by McLean and Arnold [30] the surcharge Q acting at the hopper outlet is given by

$$Q = q \gamma L^{1-m} B^{2+m} \quad (24)$$

where:

$$\begin{aligned} \frac{q}{\gamma} &= \text{non-dimensional surcharge factor} \\ &= \frac{\rho}{\rho_g} = \frac{\text{bulk density at outlet}}{\text{acceleration due to gravity}} \\ \frac{L}{B} &= \frac{\text{hopper outlet length}}{\text{hopper outlet width or diameter}} \\ m &= 0 \text{ for plane-flow or wedge-shaped hopper} \\ m &= 1 \text{ for axi-symmetric or conical hopper} \end{aligned}$$

4.3.1 Analytical Expression for Initial Non-Dimensional Surcharge Factor

The load acting on the feeder may be estimated by assuming the average vertical pressure P_v acts over the whole area of the hopper outlet. From Section 4.2.1, the average vertical pressure is given by equation (16), when multiplied by the hopper outlet area and combined with equation (24) yields the following expression for the initial non-dimensional surcharge factor q_i

$$q_i = \left(\frac{\pi}{2}\right)^m \frac{1}{2(m+1)\tan\alpha} \left[\frac{D}{B} + \frac{2Q_c \tan\alpha}{A_c \gamma B} - 1 \right] \quad (25)$$

where

$$\begin{aligned} \frac{D}{B} &= \frac{\text{bin width}}{\text{hopper opening dimension}} \\ \alpha &= \text{hopper half angle} \\ m &= 0 \text{ for plane-flow} \\ m &= 1 \text{ for axi-symmetric flow} \end{aligned}$$

McLean and Arnold [30] considered the surcharge at the outlet as being the difference between the weight of the material in the hopper section plus the surcharge Q_c at the transition minus the vertical wall support. They used the original Jenike method [13] for the initial normal wall pressures in the hopper. The expression obtained is

$$q_i = \left(\frac{\pi}{2}\right)^m \frac{1}{2(m+1)\tan\alpha} \left[\frac{D}{B} + \frac{2Q_c \tan\alpha}{A_c \gamma D} - 1 \right] \quad (26)$$

It will be noted that equations (25) and (26) are identical except for the denominator of the middle term of the section contained within the square bracket; in (25) the denominator includes B while in (26) it is D.

Evidence suggests that (26) provides a good estimate of q_i . Since (25) will yield higher values of q_i it is suggested that it be regarded as an absolute upper bound. For design purposes it is suggested that equation (26) be used.

4.3.2 Analytical Expression for Flow Non-Dimensional Surcharge Factor

Following McLean and Arnold [30] the flow non-dimensional surcharge factor q_f is given directly by equation (20). That is

$$q_f = \frac{1}{4} \left(\frac{\pi}{3} \right)^m \frac{1}{\tan \alpha} \left[\frac{y}{x-1} \left(\frac{1 - \sin \delta \cos 2\beta}{\sin \alpha} \right) (\tan \alpha + \tan \theta) - \frac{1}{1+m} \right] \quad (27)$$

where β , x and y are given by equations (19), (20) and (21) respectively.

Charts for q_f are also presented in Reference [13].

4.4 Empirical Approaches for Estimating Feeder Loads

In this section the empirical approaches of Reisner [18], Bruff [38] and Johanson [39] are given.

4.4.1 Reisner's Methods

i. Flow Load Based on σ_1

Reisner postulates that, faced with higher experimental values of feeder loads, one should consider the possibility that the stress field shifts and rotates above the feeder causing a different vertical stress condition. The highest value would occur when the major consolidation pressure (σ_1) acts downward. σ_1 is given by

$$\sigma_1 = \frac{\gamma B \text{ ff}}{H(\alpha)} \quad (28)$$

where ff = flow factor. Charts for ff and $H(\alpha)$ are given in [13]. While Reisner suggests that this value should be used for vibratory feeders to predict feeder loads during flow, recent research [21,22,24] suggests that the prediction is also applicable to belt feeders.

ii. Flow Load Based On σ_w

Reisner suggests that the normal wall pressure at the hopper outlet (σ_w) (which is less than σ_1 but greater than the mean consolidation stress σ) provides a good approximation for belt, apron and table feeders during flow. (See comment at end of (i) above.)

$\sigma_w/\gamma B$ is available in chart form [13] or can be calculated from the equation (17) which is repeated below

$$\frac{\sigma_w}{\gamma B} = \frac{y(1+\sin\delta \cos 2\beta)}{2(x-1)\sin\alpha} \quad (29)$$

x and y are given by equations (22) and (23) respectively. For both the above approaches Q_f is calculated from

$$Q_f = \sigma_1 \text{ (or } \sigma_w) \times \text{Hopper Outlet Area.}$$

iii. Initial Loads

Reisner indicates that initial loads are 2 to 4 times higher if the bin is filled from completely empty and only 1.1 to 1.2 times higher if the bin is not completely emptied before refilling.

4.4.2 Bruff's Method

Bruff [38] suggests that Q for flow conditions be approximated by taking the weight of a block of bulk solid of height = 4 x R (where R = hydraulic radius) above the hopper outlet, Figure 30.

$$R = \frac{\text{cross-sectional area of outlet}}{\text{perimeter of outlet}} \quad (30)$$

$$R \text{ for a circular outlet} = B/4 \quad (31)$$

$$R \text{ for slotted outlet} = \frac{LB}{2(L+B)} - \text{including end effects} \quad (32)$$

$$= B/2 - \text{neglecting effects} \quad (33)$$

Q for flow and initial conditions can then be calculated from the following equations -

$$\text{For circular outlet } Q = \pi/4 B^3 \gamma \eta_s \quad (34)$$

For slotted outlet including end effects

$$Q = \frac{2L^2B^2}{L+B} \gamma \eta_s \quad (35)$$

For slotted outlet neglecting end effects

$$Q = 2LB^2 \gamma \eta_s \quad (36)$$

where:

$\eta_s = 4$ for initial filling conditions
 $= 1$ for flow conditions.

4.4.3 Johanson's Method

Johanson [39] suggests a similar empirical approach to Bruff, for flow conditions, except that he uses half Bruff's values and always neglects the end effects for a long slotted outlet, Figure 31. Johanson makes no recommendations for initial filling conditions.

4.5 Power to Shear Bulk Solid in Hopper

Knowing the load acting on the feeder, the force required to shear the bulk solid tangentially at the hopper outlet may be estimated. For a belt or apron feeder, the force to shear the material is approximated by

$$F = \mu_1 Q \quad (37)$$

Various authors assume different values of μ_1 . For instance

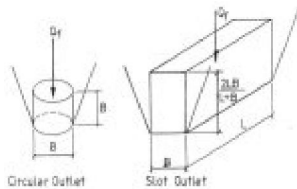


Figure 30 - Bruff's approximation for Q_f

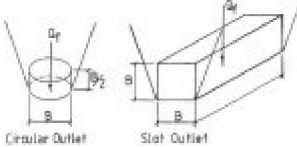


Figure 31 - Johanson's approximation for Q_f

Reisner and Bruff assume $\mu = 0.4$

McLean and Arnold, and Johanson assume

$$\mu = \sin\delta \quad (38)$$

where δ = effective angle of internal friction.

The power required to the shear bulk at the hopper outlet is

$$p = Fv \quad (39)$$

Where v = belt or apron speed.

It is to be noted that depending on the feeder type, additional forces may be exerted on the feeder. For example

- force due to material contained within skirtplates
- additional load due to material on a belt, trough or table.

It may also be necessary to determine the resistances and powers due to other factors such as

- Skirtplate resistance
- belt resistance

These aspects are discussed in more detail in Section 5.

4.6 Example

4.6.1 Problem Description

It is required to determine the loads exerted at the outlet of the plane-flow wedge-shaped bin of Figure 32.

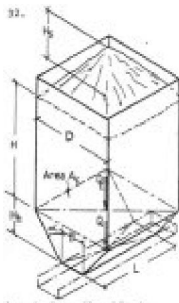


Figure 32 - Wedge-shaped plane-flow bin for feeder load example

The relevant details are as follows:

i. Bin

Opening Dimension $B = 1.5 \text{ m}$

Height $H = 8.0 \text{ m}$

Width $D = 5.0 \text{ m}$

Surcharge $H_s = 1.5 \text{ m}$

Height of hopper $H_h = 2.16 \text{ m}$

Half angle $\alpha = 39^\circ$

Parallel section of bin = mild steel

Hopper section = Lined with stainless steel type 304-2B

Length of opening $L = 5 \text{ m}$

ii. Bulk Solid Type = coal

Effective angle of
internal friction $\delta = 50^\circ$

Angle of friction between
coal and mild steel $\phi = 30^\circ$

Angle of friction between
coal and stainless
steel $\phi_n = 18^\circ$

Bulk density $\rho = 0.95 \text{ t/m}^3$

NOTE : The coal type and hopper geometry is based on that given in Figure 6.

4.6.2 Solution

i. Static Surcharge Factor q_i

Assuming material surcharge on top of the bin is of conical shape, $m_s = 1$. Then from (17)

$$h_s = \frac{1.5}{1+2} = 0.5 \text{ m}$$

Cylinder is square in cross-section. Thus from (2)

$$R = \frac{5}{2 \times 2} = 1.25$$

Surcharge at the transition - from (13), assuming $K_j = 0.4$

$$\frac{Q_c}{A_c} = \frac{0.95 \times 9.81 \times 1.25}{\tan 39 \times 0.4} [1 - e^{-\tan 39 \times 0.4 \times 8 / 1.25}] + 0.95 \times 9.81 \times 0.5 e^{-\tan 39 \times 0.4 \times 8 / 1.25}$$

$$= 40.53 \text{ kPa}$$

From (26) with $m = 0$ for plane-flow

$$q_i = \frac{1}{1 \times 2 \times \tan 39} \left[\frac{5}{1.5} + \frac{2 \times 40.53 \tan 39}{0.95 \times 9.81 \times 5} - 1 \right] = 2.31$$

ii. Flow Surcharge Factor
From (21)

$$\beta = \frac{1}{2} [18 + \sin^{-1} \left(\frac{\sin 18}{\sin 50} \right)] = 20.89$$

iii. From (22) and (23)

$$x = \frac{\sin 50}{1 - \sin 50} \left[\frac{\sin(2 \times 20.89 + 39)}{\sin 39} + 1 \right]$$

iv. $= 8.41$

$$y = \frac{1.045 \sin 39 + \sin 20.89 \times \sin(20.89 + 39)}{(1 - \sin 50) \sin^2(20.89 + 39)}$$

v. $= 5.52$

vi. From (27)

$$q_f = \frac{1}{4 \tan 39} \left[\frac{5.52}{7.41} (1 + \sin 50 \sin 41.8) (\tan 39 + \tan 18) - 1 \right]$$

vii. $= 0.34$

viii. Feeder Loads

Using (24)

$$\begin{aligned}\text{Initial } Q_i &= 2.31 \times 0.95 \times 9.81 \times 5 \times 1.5^2 \\ &= 242.2 \text{ kN}\end{aligned}$$

$$\begin{aligned}\text{Flow } Q_f &= 0.34 \times 0.95 \times 9.81 \times 5 \times 1.5^2 \\ &= 35.93 \text{ kN}\end{aligned}$$

ix. Empirical Values

These have been determined using the methods outlined in Section 4.4. The results, together with those above are summarised in Table 1.

TABLE 1 - SUMMARY OF PREDICTED FEEDER LOADS

Source	Initial conditions Q_i (kN)	Flow Conditions Q_f (kN)	$\frac{Q_i}{Q_f}$	$\frac{Q_i}{Q_{f\sigma_1}}$ *
Theoretical	242.25	35.93	6.74	2.21
Reisner σ_1	438.3	109.57	4.0	4.0
Reisner σ_w	390.0	95.5	4.08	3.56
Bruff Incl. End Effects	645.2	161.30	4.0	5.89
Bruff Excl. End Effects	838.8	209.7	4.0	7.66
Johanson	-	104.8	-	-

* Ratio of Q_i to $Q_{f\sigma_1}$ for Reisner computed on basis of σ_1 .

4.7 Some Comments

For the given hopper geometry and coal properties, the results in Table 1 show the theoretical initial and flow values to be the lowest respectively for all the methods. The ratio of initial load to flow load, Q_i/Q_f is of significant magnitude as shown. As pointed out previously, Reisner [18] indicates that the initial load can be 2 to 4 times the flow load. However research to date [21,22, 24] suggests a ratio of 4 to 8 which confirms the theoretical ratio Q_i/Q_f of 6.78 given in Table 1. Figure 33 shows a typical filling and flow feeder load variation obtained in laboratory tests on a plane-flow stopper and belt feeder.

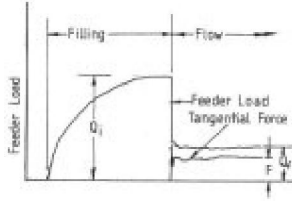


Figure 33 - Feeder load variations for plane-flow hopper and belt feeder

The research [21,22,24] has also indicated that the theoretical value of Q_i given by equation (26) provides a good estimate. However the evidence suggests that the theoretical value of Q_f determined using equation (27) under-estimates the flow load and that Reisner's method, based on the major consolidating pressure σ_1 at the outlet, provides a better estimate. The last column in Table 1 shows the ratio of Q_i to Q_{foi} where Q_{foi} is value given by Reisner's method.

It is useful to examine the influence of variation in hopper geometry on the magnitude of the feeder loads. Figure 34 shows the variation in the non-dimensional surcharge factors while Figure 35 shows the variation of the theoretical feeder loads as a function of hopper opening dimension B for the bin of the previous example. The principal dimension of the bin are maintained with the exception of B , α and H_h . B and α , are varied in accordance with Figure 6, while H_h is adjusted to accommodate these variations. The decrease in Q_i and Q_f with decrease in α and B to maintain mass-flow is clearly evident. Also Q_{foi} for the Reisner method also decreases. However the ratios Q_i/Q_f and Q_i/Q_{foi} increase with decrease in B .

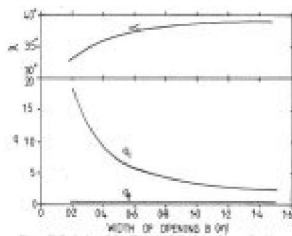


Figure 34 - Variation of non-dimensional surcharge factor for bin of example 4.6.1.

4.8 Controlling Feeder Loads

The high initial loads which may act on feeders are a matter of some concern and where possible steps should be taken to reduce the magnitude of these loads. While it is possible to limit the loads by a suitable arrangement of hopper outlet as depicted in Figure 27, it is important to ensure that the hopper outlet is fully active if mass-flow is to be maintained.

Since the initial load only ever occurs when the bin is filled from the empty condition without discharge taking place, it is good practice to always maintain a cushion of material in the hopper [22]. From a practical point of view this is most desirable in order to protect the hopper surface from impact damage during filling.

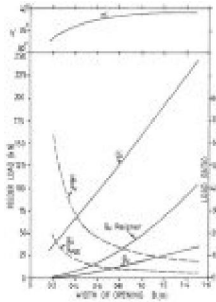


Figure 35 - Variations in feeder loads with hopper geometry plane-flow bin of example 4.6.1

However there is a further advantage; the material left in the hopper as a cushion, having previously been in motion, will preserve the arched stress field. The new material being deposited in the bin will initially have a peaked stress field. This will provide a surcharge load on the arch field, but the load at the outlet will be of lower order than if the bin is totally filled from the empty condition. The stress condition and reduced loading is illustrated in Figure 36.

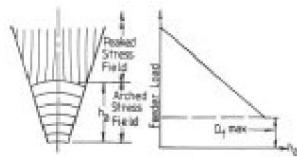


Figure 36 - Cushioning in hopper to reduce feeder load

5. FEEDING OF BULK SOLIDS FROM BIN ONTO BELT CONVEYOR

In the handling of bulk solids, belt feeders with skirtplates are commonly used. In other cases dump bins are used in combination with belt conveyors as illustrated in Figure 37. For design purposes it is necessary to determine the belt loads under initial filling and flow conditions and the corresponding drive powers [11].

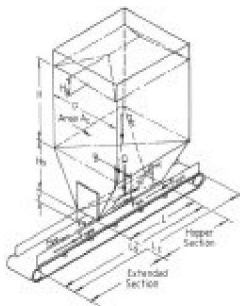


Figure 37 - Schematic arrangement of dump bin and belt conveyor

5.1 Design Equations

5.1.1 Bin and Hopper Surcharge and Corresponding Power

These are determined in accordance with the methods described in Section 4.

5.1.2 Skirtplate Resistance

Assuming steady flow, the skirtplate resistance may be determined as follows:

- i. Hopper Section

$$F_{sph} = \mu_2 K_v(2Q + \rho g B L y) y/B \quad (40)$$

- ii. Extended Section (Section beyond hopper)

$$F_{spe} = \mu_2 K_v \rho g (L_s - L) y^2 \quad (41)$$

where:

Q = feeder loads as determined by equation (24)
 ρ = bulk density
y = average height of material against skirtplates
 K_v = ratio of lateral to vertical pressure at skirtplates
g = acceleration due to gravity = 9.81 (m/s²)
B = width between skirtplates
 μ_2 = skirtplate friction coefficient
 L_s = total length of skirtplates (m)

5.1.3 Belt Load Resistance

- i. Hopper Section

$$F_{bh} = (Q + \rho g B L y) \mu_b \quad (42)$$

- ii. Extended Section

$$F_{be} = \rho g B (L_s - L) y \mu_b \quad (43)$$

Where μ_b = Idler friction.

5.1.4 Empty Belt Resistance

$$F_b = W_b L_b \mu_b \quad (44)$$

Where:

W_b = belt weight per unit length

L_b = total length of belt

5.1.5 Force to Accelerate Material onto Belt

$$F_A = Q_m v \quad (45)$$

Where:

Q_m = mass-flow rate

v = belt speed

It is assumed that

$$Q_m = \rho B v \quad (46)$$

Usually the force F_a is negligible.

5.1.6 initial and Flow Loads and Powers

The foregoing loads and resistances are determined for the initial and flow conditions using the appropriate values of the variables involved. The power is computed from

$$P = (\Sigma \text{ Resistances}). v / \eta \quad (47)$$

Where η = efficiency and v = belt speed.

The condition for non-slip between the belt and bulk solid under steady motion can be determined as follows:

$$\mu_3 (Q_f + w) \geq (F_f + F_{sp}) \quad (48)$$

Where:

μ_3 = coefficient of friction between belt and bulk solid
 Q_f = flow surcharge at hopper outlet

W = weight of bulk material between skirtplates in hopper section of conveyor

F_f = force to shear material at hopper outlet

F_{sp} = skirtplate resistance.

5.2 Example

5.2.1 Problem Description

As an extension of the example of section 4.6.1, consider the bin being used in conjunction with a belt conveyor. referring to Figure 36 the relevant details are

i. Bin

Opening Dimension B	= 1.5 m
Height	H = 8.0 m
Width	D = 5.0 m
Surcharge	H _s = 1.5 m
Height of Hopper	H _h = 2.16 m
Half angle	$\alpha = 39^\circ$
Length of bin	L = 5 m
Cylinder	= Mild steel
Hopper lining	= Stainless steel

ii. Bulk Solid

Type	= coal
Effective angle of internal friction	$\delta = 50^\circ$
Angle of friction between coal and steel	$\phi = 30^\circ$
Angle of friction between coal and stainless steel	= 18°
Bulk density	= $\rho = 0.95 \text{ t/m}^3$

iii. Conveyor

Belt velocity	v = 0.5 m/s
Conveyor length	L _c = 20.0 m
Length of skirtplate	L _s = 20.0 m
Average skirtplate height	y = 0.8 m
Belt width	B _b = 2.0 m

5.2.2 Solution

i. Hopper Surcharges

From Section 4.6

$$Q_i = 242.2$$

$$Q_f = 35.9 \text{ kN}$$

ii. Force to Shear Material at Hopper Outlet

$$F_i = Q_i \sin \delta = 242.2 \sin 55 = 185.6 \text{ kN}$$

$$F_f = Q_f \sin \delta = 35.9 \sin 55 = 27.5 \text{ kN}$$

iii. Skirtplate Resistance - Hopper Section

Equation (40)

$$\text{Assume } \mu_2 = \tan 20^\circ \text{ for polished mild steel} \\ = 0.364$$

$$K_v = 0.4 \text{ for initial case} \\ K_v = 0.6 \text{ for flow case [40]}$$

$$\text{Initial } F_{\text{sph}}(i) = 0.364 \times 0.4 (2 \times 242.2 + 0.95 \times 9.81 \times 1.5 \times 5 \times 0.8) \times 0.8/1.5 \\ = 42.0 \text{ (kN)}$$

$$\text{Flow } F_{\text{sph}}(f) = 0.364 \times 0.6 (2 \times 35.9 + 0.95 \times 9.81 \times 1.5 \times 5 \times 0.8) \times 0.8/1.5 \\ = 14.9 \text{ (kN)}$$

iv. Skirtplate Resistance - Extended Section

Equation (41)

$$\text{Initial } F_{\text{sph}}(i) = 0.364 \times 0.4 \times 0.95 \times 9.81 (20 - 5)0.8^2 \\ = 13.03 \text{ (kN)}$$

$$F_{\text{sph}}(f) = 0.364 \times 0.6 \times 0.95 \times 9.81 (20 - 5)0.8^2 \\ = 19.54 \text{ (kN)}$$

v. Belt Load Resistance - Hopper Section

Equation (42) Assume $\mu_b = 0.06$

$$\text{Initial } F_{bh}(i) = (242.2 + 0.95 \times 9.81 \times 1.5 \times 5 \times 0.8) \times 0.06 \\ = 17.9 \text{ (kN)}$$

$$\text{Flow } F_{bh}(f) = (35.9 + 0.95 \times 9.81 \times 1.5 \times 5 \times 0.8) \times 0.06 \\ = 5.5 \text{ (kN)}$$

vi. Belt Load Resistance - Extended Section

Equation (43)

$$\text{Initial } F_{be} = (i) = 0.95 \times 9.81 \times 1.5 \times (20 - 5) \times 0.8 \times 0.06 \\ = 10.07 \text{ (kN)}$$

$$\text{Flow } F_{be}(f) \text{ same as for initial condition} \\ \text{i.e. } F_{be}(f) = 10.07 \text{ (kN)}$$

vii. Empty Belt Resistance

Equation (44)

$$\text{Assume } w_b = \frac{60 B_b}{1000} g = \frac{60 \times 2.0 \times 9.81}{1000}$$

$$= 1.18 \text{ (kN/m)}$$

$$F_b = 1.18 \times 2 \times 20 \times 0.06 \\ = 2.83 \text{ (kN)}$$

(Actual belt weight will need to be checked after final belt selection is made).

viii. Acceleration Resistance

$$Q_m = 0.95 \times 1.5 \times 0.8 \times 0.5 = 0.57 \text{ t/s}$$

$$F_A = 0.57 \times 0.5 = 0.285 \text{ (kN)}$$

ix. Checking Non-Slip

From (48)

$$\mu_3 \geq \frac{F_f + F_{sph}(f) + F_{spe}(f)}{F_{bh}(f) + F_{be}(f) + F_b + F_A}$$

$$\begin{aligned}
& Q_f + w \\
\geq & \frac{27.5 + 14.9 + 19.54}{35.93 + 0.95 \times 9.81 \times 20 \times 1.5 \times 0.8} \\
\geq & 0.24
\end{aligned}$$

For non-slip $\mu_3 \geq 0.24$. For coal on rubber belting this condition is easily satisfied.

x. Total Power

Assume $\eta = 90\%$

$$\begin{aligned}
\text{Initial case } P_i &= (185.6 + 42.0 + 13.0 + 17.9 + 10.1 + 2.83) \times 0.5/0.9 \\
&= \underline{151 \text{ kW}}
\end{aligned}$$

$$\begin{aligned}
\text{Flow case } P_f &= (27.5 + 14.9 + 19.5 + 5.5 + 10.1 + 2.83 + 0.29) \times 0.5/0.9 \\
&= \underline{44.8 \text{ kW}}
\end{aligned}$$

The value of P_f will be larger if the Reisner value of Q_f is used. The total initial power is based on the assumption that the total initial load acts with belt velocity v during start-up from the initial filling condition. In practice, the initial power is most likely to be less due to the starting characteristics of the drive. Furthermore, once flow conditions have been established, and the bin is kept nominally full, then start-up from a stopped condition will most likely occur at a much lower power corresponding to the flow condition.

5.3 Belt Feeders - More Rigorous Analysis

In order to obtain satisfactory draw of material uniformly distributed over the full hopper outlet, as previously discussed, it is usually necessary to taper the hopper bottom in the direction of feed. A detailed study of the forces and power requirements for belt feeders has been presented by Rademacher [19,40].

5.4 Accelerated Flow in Skirtplate Zone

The skirtplate analysis presented in the example of Section 5.2 is based on the assumption of uniform flow. When directing bulk solids onto conveyor belts through feed chutes, it is necessary to accelerate the solids to belt speed. The motion is resisted by the drag imposed by the skirtplate. This problem has been analysed by Roberts and Hayes [2,11].

As material accelerates, the depth h of the bed decreases as indicated in Figure 38.

In view of the shallow bed conditions, it may be assumed that the lateral pressure distribution is proportional to the hydrostatic pressure distribution. For each wall the average pressure is $K_v \rho g h / 2$, where K_v is the pressure ratio coefficient at the walls of skirtplates. Hence for the two walls the total average lateral pressure is $\rho g h K_v$.

Here h is the depth of the material and within the acceleration zone varies inversely with the velocity v_s . That is, for a constant throughput Q_m (kg/s), it follows that

$$h = \left(\frac{Q_m}{\rho b} \right) \frac{1}{v_s} \quad (49)$$

where ρ is density of the material (kg/m³), b is width between skirtplates (m), and v_s is velocity at section considered (m/s). Here $(Q_m / \rho b) = \text{constant}$.

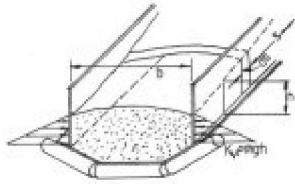


Figure 38 - Skirtplates in acceleration zone

A dynamic analysis of the motion of the material in contact with both the belt and skirtplates shows that the acceleration of the material is non-uniform and that the velocity v_s as a function of distance l is non-linear in form. However for simplicity in this case it will be assumed that the average height of the material is inversely proportional to the average velocity based on linearity. That is, from equation (49),

$$h_{av} = \left(\frac{Q_m}{\rho b} \right) \left(\frac{2}{v - v_o} \right)$$

Hence the drag force due to side plate friction can be obtained as follows:

$$F_A = \mu_2 \rho g \ell_b K_v (h_{av})^2$$

or

$$F_a = \frac{4\mu_2 g \ell_{bl} Q_m^2}{\rho b^2 K_v (v - v_o)^2} \quad (50)$$

where μ_2 is friction coefficient between material and skirtplates = 0.5 to 0.7, ℓ_{bl} is the acceleration length.

Assuming, for simplicity, a block-like motion of the bulk solid, an analysis of the forces due to the belt driving the material forward and the skirtplates causing a resistance to forward motion shows that the acceleration of the bulk solid is

$$a = \mu_1 g - \mu_2 g K_v h/b \quad (51)$$

where:

$$\begin{aligned} \mu_1 &= \text{belt friction} \\ \mu_2 &= \text{skirtplate friction} \end{aligned}$$

It is to be noted that the component $\mu_1 g$ in (51) is the acceleration in the absence of skirtplates. Substituting for h from (49) gives

$$a = g(\mu_1 - \mu_2 \frac{K_v Q_m}{\rho b^2 v_s}) \quad (52)$$

where:

v_s = bulk solid velocity at distance s from point of entry to belt.

Equation (52) shows that for a given throughput and skirtplate configuration the acceleration decreases as v_s decreases. This implies that there is a minimum value of v_s to achieve the required feeding at the rate Q_m onto the belt. The critical condition will be at the point of entry where $s = 0$ and $v_s = v_o$. Putting $a = 0$ in (52) yields

$$v_{o\min} = \frac{\mu_2 K_v Q_m}{\mu_1 \rho b^2} \quad (53)$$

Writing $a = v_s \frac{dv_s}{ds}$ in (52) and substituting from (53) yields

$$\frac{dv_s}{ds} = \mu_1 g \left(\frac{1}{v_s} - \frac{v_{o\min}}{v_s^2} \right) \quad (54)$$

Integration of (54) over the limits $v_s = v_o$ to $v_s = v$ where v is the belt speed yields the following expression for the acceleration length

$$\ell_{b1} = \frac{1}{\mu_1 g} \left\{ \frac{(v^2 - v_o^2)}{2} + v_{o\min} (v - v_o) + v_o^2 \min \ln \left[\frac{v - v_{o\min}}{v_o - v_{o\min}} \right] \right\} \quad (55)$$

Example:

Coal of density $\rho = 0.9 \text{ t/m}^3$ is fed onto a conveyor belt at a rate of $Q_m = 800 \text{ t/h}$ with an initial velocity of $v_o = 0.5 \text{ m/s}$. The conveyor has skirtplates in the acceleration zone. It is assumed that $\mu_1 = 0.5$, $\mu_2 = 0.4$ and, for conservative design, $K_v = 1.0$. The width between

skirtplates $b = 1.0$ m. It is required to determine the acceleration length. The belt speed $v = 3$ m/s.

Solution:

From (52) the minimum initial velocity is

$$v_{\text{omin}} = \frac{0.4 \times 1.0 \times 800}{0.5 \times 0.9 \times 1 \times 1 \times 3600} = 0.198 \text{ m/s}$$

From (54)

$$\ell_{b1} = \frac{1}{0.5 \times 9.81} \left\{ \left(\frac{3^2 - 0.5^2}{2} \right) + 0.198 (3 - 0.5) + 0.198 \ln \left(\frac{3 - 0.198}{0.5 - 0.198} \right) \right\}$$

$$\ell_{b1} = 1.011 \text{ (m)}$$

6. FEED CHUTES

6.1 General Remarks

As outlined in the introduction, the role of feed-chutes is to direct bulk solids from bins and feeders onto conveyor belts in a manner which will minimise spillage and belt wear. The chute may also be designed in a manner which will ensure the component of the exit velocity tangential to the belt v_T is matched as closely as possible to the belt speed.

A typical chute arrangement is shown in Figure 39.

While the normal component V_N of the exit velocity should be as small as possible to minimise impact damage to the belt, it is necessary to ensure continuity of feed with sufficient chute slope to maintain flow and prevent choking.

The flow characteristics of bulk solids in chutes has been the subject of considerable research, and general design procedures have been presented [41-45]. It is beyond the scope of this paper to discuss the design of chutes; reference may be made to the literature cited for details. However it is worth mentioning the importance of the friction characteristics of chute lining materials and the influence of friction on the selection of chute geometry.

6.2 Chute Friction and Slope Angles

There are many lining materials available and these need to be selected on the basis of their frictional and wear resistance properties. It is also important to consider any corrosive influence of the bulk solid on the hopper wall. By way of illustration, Figure 40

shows the wall yield loci or friction characteristics for coal at 19% moisture content (d.b.) on stainless steel, polished mild steel and rusted mild steel for the instantaneous condition as well as the polished mild steel surface after 72 hour storage. The increase in friction in the latter case is quite considerable. It has been found that

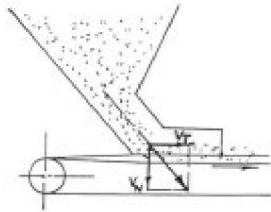


Figure 39 - Feed Chute for Belt Conveyor

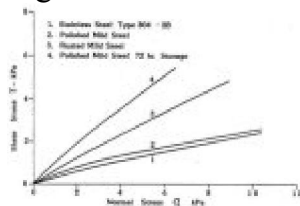


Figure 40 - Wall yield loci for coal at 19% moisture content (d.b.)

certain coals, for example, will build up on mild steel surfaces even after a short contact time of a few hours. The type of behavior found to occur in practice is illustrated in Figure 41. Moist coal from a screen has been found to adhere to vertical mild steel surfaces as indicated, particularly where the initial velocity of the coal in contact with the surface is low.

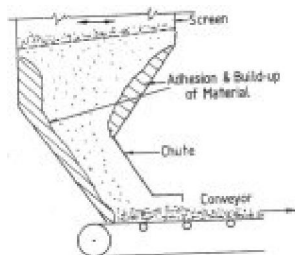


Figure 41 - Build up of cohesive material on chute surfaces

As indicated by Figure 42, the wall yield loci are normally slightly convex upward in shape. Also, depending on the surface "roughness" or rather "smoothness", moisture bulk solids may exhibit an adhesion component adhesion often occurs with a smooth surface. The effect of the adhesion and/or the convexity in the wall yield locus is to cause the wall friction angle to decrease as the consolidation pressure increases. This is illustrated in Figure 43.

The variation of friction angle with consolidation pressure must be taken into account when determining chute slope angles. Since bed depth on a chute surface is related to

consolidating pressure it is useful, for design purposes, to examine the variation of friction angle with bed depth. Figure 43 shows, for a range of moisture contents, the considerably high friction angles that can occur at low bed depths, the decrease in friction angle being significant as the bed depth increases.

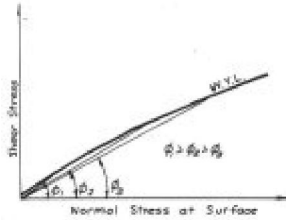


Figure 42 - Wall yield locus and wall friction angles

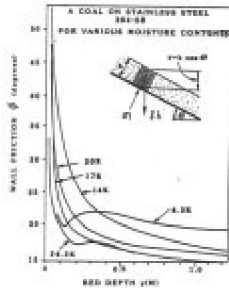


Figure 43 - Wall friction versus bed depth

For a chute inclined at an angle $\emptyset$ to the horizontal, the relationship between bed depth and consolidation pressure at the chute surface is

$$h = \frac{\sigma_1}{\gamma \cos \emptyset} \quad (56)$$

The slope of the chute $\emptyset$ should be at least 5° larger than the maximum friction angle measured. Often moist bulk solids will adhere initially to a chute surface particularly if the initial velocity tangential to the chute surface is low. However, as the bed depth increases, the corresponding decrease in friction angle will cause flow to be initiated. In such cases flow usually commences with a block-like motion of the bulk solid. This is depicted in Figure 44.

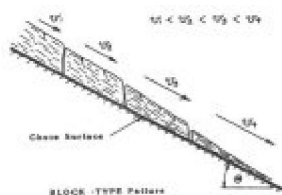


Figure 44 - Block-like flow of bulk solid

By far the greatest component of the drag force in a chute occurs along the chute bottom; the side walls contribute to a lesser extent. Where possible the side walls should be

tapered outward or flared as indicated in Figure 45 with gussets in the corners to prevent, or at least reduce, the build-up of material in the corners.

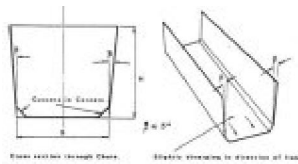


Figure 45 - Recommended chute configuration

7. CONCLUDING REMARKS

This paper has focused attention on the interactive role of storage bin, feeders and chutes in providing efficient and controlled feeding of bulk solids onto conveyor belts. Various types of feeders have been reviewed and methods for determining feeder loads and power requirements have been presented. The theoretical expressions given by equations (24) and (26) appear to provide a good estimate of the initial feeder load while the combinations of equations (24) and (27) give a theoretical prediction based on the radial flow stress theory for the flow loads. However the flow loads tend to be underestimated by this procedure and accordingly the method due to Reisner based on the major consolidation stress σ_1 as given by equation (28) is recommended as providing a more realistic estimate. It is clear that more research is necessary to validate the predictions proposed.

As a general comment it is worth noting that the modern theories of storage and feeding system design have been developed over the past 30 years with many aspects still being subject to considerable research and development. It is gratifying to acknowledge the increasing industrial acceptance, throughout the world, of the modern materials testing and design procedures. These procedures are now well proven, and while much of the industrial development has, and still is, centered around remedial action to correct unsatisfactory design features of existing systems, it is heartening that in many new industrial operations the appropriate design analysis and assessment is being performed prior to plant construction and installation. It is more important that this trend continue.

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