ENERGY ASSISTED GRAVITY FLOW:

AN OVERVIEW OF VIBRATIONAL EFFECTS ON BULK SOLIDS AND STORAGE VESSELS

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1. INTRODUCTION

The ideal design for a bulk solids storage vessel is one in which the material will flow by gravity alone. However, continuous "mass flow" may not always be possible due to height restrictions, cost limitations, loading practices, bulk material variations and/or changes in hopper characteristics. It is in these cases where energy assistance can be used effectively to promote material flow.

Vibration is one of the most widely used means for promoting material flow from bins and hoppers. Applications using tools as simple as sledge hammers to more sophisticated live bottom bins are examples. However, little is known about how vibrational energy is transmitted through the hopper structure and into the bulk solid to re-establish material flow. The objective of this paper is to provide an overview of basic vibrations concepts as they apply to bulk solids and structures in order to gain insight on the positive affects of applied vibration

2. BULK MATERIAL FLOW CONCEPTS

Bulk material flow theory is based on the premise that if an arch or rathole cannot occur in an hopper or bin, satisfactory continuous flow will result by gravity alone. Two factors contributing to flowability are material strength and sliding friction.

2.1 STRENGTH AND SLIDING FRICTION

Strength is the ability of the material to retain shape, or degree of cohesiveness. Increases in consolidation pressure lead to higher strength and reduced flowability.

Sliding friction is the dynamic friction between the bulk solid and hopper wall surface. Sliding friction and hopper wall inclination determine the ability of the material to flow at the wall. Both strength and sliding friction are influenced by temperature, moisture content, consolidation pressure, and storage time. For applications involving the storage of bulk solids, controlling these elements to maintain gravity flow is often impractical.

2.2 VIBRATION EFFECTS ON FLOWABILITY

Properly applied vibration causes both a reduction in strength and sliding friction. Dynamic vibrational shear testing conducted at the University of New Castle demonstrated these benefits. Hopper materials with identical average roughness values produced different wall friction characteristics. RMS roughness was proposed as a means for accounting for both the magnitude and number of the peaks and valleys along the surface profile. Dynamic shear tests indicated a direct correlation between reduced sliding friction and an apparent reduction in RMS wall roughness. Both decreased by identical magnitude with increased vibration frequency. It was concluded that high frequency vibration (100-200 Hz) is best for reducing sliding friction.

Dynamic shear test measurements of material strength showed similar effects. For the same consolidation conditions, the application of vibration at different frequencies during shear leads to a reduction in strength. Dynamic shear strength was affected by both the magnitude and frequency of the applied vibratory force. In addition, bulk material resonance effects were observed at high frequencies (100 – 200 Hz). It has been suggested that the bulk material within the hopper contains a multitude of fundamental frequencies due to both a wide range in consolidation and normal pressures.

3. VIBRATION CONCEPTS AS RELATED TO HOPPER VIBRATION EXCITERS

Still today, a great deal of research is needed to correlate the vibrational shear testing conducted at the University of New Castle to field installations in order to develop accurate sizing procedures. Current vibrator sizing methods are based on principles 35 years old developed prior to bulk material flow theory. Relatively little is known about what type of vibrational energy to apply, how the vibration energy propagates along the inner hopper surface and into the bulk material, how much energy is required to develop flow, or when and where to apply it. Assumptions that vibration always adversely affects hopper structures and compacts bulk materials often preclude the use of vibratory flow aids. An overview of vibrational effects on bulk materials and hopper structures provides insight into how vibration can be best utilized to re-establish material flow.

3.1 FORCING FUNCTIONS – TYPES OF VIBRATION EXCITERS

The response of any system under the influence of vibration is dependent upon the type, magnitude and frequency of the forcing function. Vibration exciters for bins and hoppers develop either sinusoidal or pulsed type forcing functions.

3.1.1 SINUSOIDAL VIBRATION EXCITERS

Sinusoidal vibration exciters develop their forcing functions through either rotational or linear motion. The force output of the rotational type is generated by an unbalanced weight rotating about a central axis. The magnitude of the centrifugal force output is a function of the static torque (or unbalance) and square of the rotational speed.

$$F=\frac{WxR_G(\frac{\pi}{30}FREQ)^2}{g}$$

(Eq 1)

Typical rotational vibration exciters are the pneumatically driven ball, roller, turbine and unbalance weight, the electric motor driven unbalanced weight (electromechanical) and the hydraulic driven unbalance weight. Force outputs range from 2lbs to 30,000 pounds with frequency ranging from 900 cpm to 12,000 cpm (15 – 200 Hz).

The force output of the linear type sinusoidal exciters is a function of the work developed by a linear reciprocating mass (piston) and the square of the oscillation speed. The moving mass is spring returned at each end of its stroke to produce sinusoidal output. The developed work is the mass times the displaced distance during one half cycle and its analogous to the static torque of a rotational type exciter. Force outputs range from 20 to 5000 pounds at oscillating frequencies of 600 cpm to 3600 cpm (10 – 60 Hz).

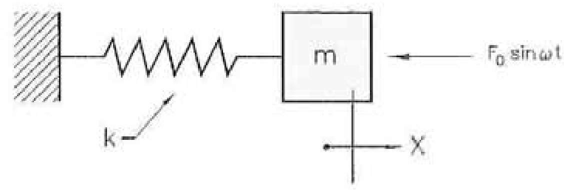
3.1.2 PULSED VIBRATIONS EXCITERS

Pulsed vibrations exciters are commonly referred to as “impacting vibrators” since metal to metal contact is used to transmit the force. Exciters can be of the single or repetitive pulsed type and include both linear pneumatic and electric (electromagnetic) type vibration exciters. A mass, either a piston or magnet, is moved linearly and strikes a metal base mounted to the hopper wall surface. The transmitted force for linear pulsed vibration is dependent on the momentum (mass times velocity) and the time duration of contact. Repetitive impulse vibration exciters operate at frequencies from 900 – 3600 cpm (50 – 60 Hz).

3.2 ELEMENT DISTURBANCE – SPRING MASS MODEL

The disturbance of an element depends upon the type of forcing function applied. The element, whether bulk material or hopper structure, in simplest form, is represented as a single degree of freedom spring mass system model. The spring is considered to be weightless and the mass a point mass in which all the motion is in one direction. Although greatly simplified, the response of such a system to different forcing functions is easy to define.

SINGLE DEGREE OF FREEDOM - SPRING/MASS SYSTEM



$$ma=-kx+F_0\sin\omega t$$

$$m\frac{\delta^2 x}{\delta^2 t}+kx=F_0\sin\omega t$$

3.2.1 SYSTEM MODEL FOR A SINUSOIDAL FORCING FUNCTION

The application of a sinusoidal force to the system model results in a displacement response dependent upon both the magnitude and frequency of the applied force. The steady stated displacement solution to Newton’s Second Law of Motion is a sinusoidal oscillation at the frequency of the applied force with amplitude equal to the static displacement times a magnification factor (eq 2). The static displacement is the force divided by the spring stiffness. The magnification factor is the relationship of the forcing frequency to the natural frequency of the system.

$$x=\frac{F_0}{k}\left(\frac{1}{1-\frac{w^2}{w_0^2}}\right)\sin wt$$

(Eq 2)

3.2.1.1 FORCE, SPEED AND TORQUE

A relatively low forcing frequency compared to the natural frequency of the system results in a magnification factor slightly greater than one and the systems maximum displacement is approximately the static displacement. Systems such as these are said to be “brute forced” into motion.

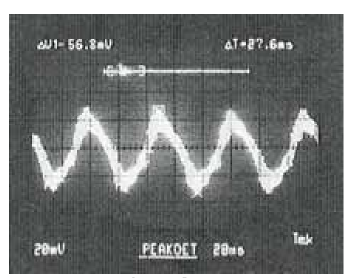


Figure 2: Response of a simply supported plate, linear sinusoidal exciter

Rearranging the displacement solutions and representing peak force as work times speed squared, reveals that the displacement response is related to the static unbalance, not force output (eq 2).

$$X = \frac{T}{W}\left(\frac{1}{\dfrac{K_o}{W_\omega^2} - 1}\right)\sin\omega t$$

(Eq 3)

For example, a rotary vibration exciter with a static unbalance of 10 in lbs and rotational speed of 3,600 rpm produces the same force output as one running at 900 rpm and a static unbalance of 160 in lbs. Although the two exciters have identical force outputs (3,700 lbs), the displacement responses for the systems defined above are completely different. The first exciter produces low amplitude high frequency displacement, while the second produces high amplitude, low frequency displacement. A high forcing frequency relative to the natural frequency, results in small peak displacement compared to the static displacement. This principle is used in the design of vibration isolators and does not apply to vibration excitation.

3.2.1.2 RESONANCE

Resonance is the condition when the forcing and natural frequencies are identical and is shown graphically in Figure 3. System failure will occur if the amplitude of motion stresses the material element beyond its fatigue limit. This condition for bulk materials would be desirable at the free surface of the arch or rathole formation. Tuning the forcing frequency to the natural frequency of the bulk material at its free surface is only possible if the natural frequency at that location is known. Tuning without knowledge of the free surface natural frequency could cause resonance at other locations within the bulk material mass and lead to compaction.

DISPLACEMENT RESPONSE – SINUSOIDAL FORCE
NO DAMPING

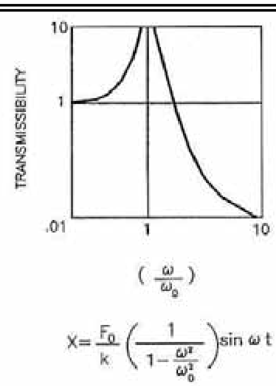


Figure 3: Transmissibility vs. Frequency Ratio spring mass system without damping

Resonance of hopper structure is caused by matching the forcing frequency of high frequency sinusoidal exciters to hopper natural frequencies, and can adversely affect the structural integrity. Hopper structural natural frequencies are in the upper range of vibration exciters (>100 Hz). Small high frequency rotational vibrators on large hopper structures have led to structural failure because of resonance.

3.2.2 SYSTEM MODEL FOR A PULSED FORCING FUNCTION

The application of force as a short time duration pulse function in our ideal spring mass system, results in a steady state solution where resultant displacement is periodic and oscillates about its equilibrium position at its natural frequency (eq 4). In reality, the amplitude decays logarithmic due to damping as shown in Figure 4.

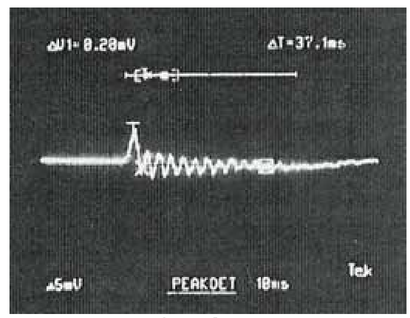


Figure 4: Response of a simply supported plate short time duration pulse

$$X = \frac{F}{k}\left(1 - \cos\omega_\eta t\right)$$

(Eq 4)

3.2.2.1 IMPULSE MOMENTUM RELATIONSHIP

The displacement amplitude depends upon the time duration of the applied force, and therefore cannot be determined from the vibration exciter alone. This is the best illustrated by differentiating Newton's Second Law to show that the impulse, force times the time duration, is equal to momentum, mass times velocity (eq5)

$$F(t_1 - t_2) = Mv$$

(Eq 5)

For example, consider a pneumatic piston impacting vibrator mounted to a flexible hopper wall, modeled for one half cycle where the piston travels horizontally acted on by a constant pressure force. At the time of contact between the piston and hopper wall, the piston has momentum mass times velocity. The piston continues to travel as the wall flexes, and is decelerated until the piston's forward velocity is zero. At this point in time, the pistons momentum is zero, and therefore, the change in momentum is equal to the initial momentum. Likewise, the impulse is also equal to the initial momentum. Now consider the same example vibrator except mounted to a stiff hopper wall. In this case, the piston again is traveling under the same pressure force. At the time of contact, the piston has the same initial momentum as the first case. During the time of contact, the piston's velocity will reach zero at which time its momentum is also zero. The difference between both cases is time of contact due to the stiffness of the hopper. Since both cases produce the same change in momentum, the impulse in each case must also be the same. Therefore, the transmitted force in the case of the stiff hopper wall is much higher due to the shorter time duration of contact, even though the applied force is identical in both cases as shown graphically in Figure 5.

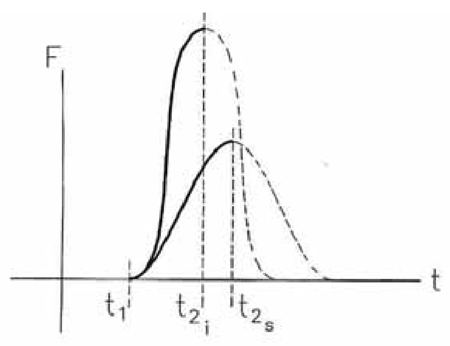


Figure 5: Impulse Momentum relationship

3.2.2.2 RESONANCE

Resonance for the single pulse type vibration exciter cannot occur since repetition of the force does not exist. The displacement response of the system model is excitation at its fundamental frequency. The effect of this occurrence on bulk materials is not fully understood. In regards to the hopper wall, the effect appears to be a reduction in the apparent RMS wall surface roughness and decrease in friction between the wall and bulk material. Resonance can occur for the repetitive pulse type vibration exciter, if the forcing frequency is identical to the natural frequency. For both bulk materials and hopper structures, the same concepts apply as for sinusoidal exciters. Generally repetitive pulsed exciters are considered low frequency devices compared to hopper structures natural frequencies.

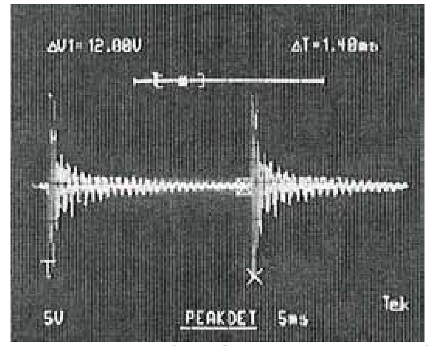


Figure 6: Response of a simply supported plate, repetitive pulse exciter

3.3 DISTRIBUTED DISTURBANCE – SYSTEM MODEL

The propagation of vibration energy through continuous elastic systems can be described by using an infinite freely supported rod as a system model. The solution to Newton's Second Law of Motion for an element of the rod shows that the energy propagates with time from one element to another as a wave. This holds true for most all physical systems. Consider a point disturbance on an infinite rod in the direction of the rod's length. At incremental positions on the rod in the direction of the disturbance, the elements of the rod are compressed. In the reverse direction, the elements are stretched, or strained. This occurrence is that of a compressive wave traveling in the reverse direction and a tensile-stress wave traveling in the reverse direction. Modeling the system as a rod of finite length results in reflection of the compression and tensile waves at the end conditions. The displacement of any point on the rod is the sum of the displacements of the incident and reflected waves, and the result is an infinite set of modal frequency responses, the lowest of which is the fundamental frequency.

3.3.1 PROPAGATION THROUGH BULK MATERIALS

Little is understood about how vibrational energy propagates through bulk materials. However, wave propagation in soils has been thoroughly studied and may be analogous to wave propagation in bulk materials. It is known that the energy radiating from a point source into a three dimensional infinite elastic media with one boundary layer propagates as three separated wave forms: a pressure wave, shear wave, and boundary layer wave (Rayleigh wave). Modeling a homogenous, isotropic, elastic half-space sample under the influence of a sinusoidal disturbance results in a distribution of the total input energy among the 3 waves as 2/3 Rayleigh wave, 1/4 shear and 1/10 compression wave. Most of the energy propagates at the boundary layer surface. Extensive testing on noncohesive and cohesive soils has shown similar distributions. Decreasing the frequency of vibration increases the Rayleigh wave length and penetration depth. Internal damping in soils is normally represented in terms of logarithmic decrement. Large amplitude strains (.0001 in/in) causes changes in soil structure that lead to loss of strength.

Soils can fatigue to failure with repeated cyclic loading. The general effect appears to be an accumulation of minor re-arrangements of soil particles at the contact points causing a gradual increase in strain as the number of loading cycles increases. As the number of pulses increase, little increase in deformation may occur and then increases rapidly as the failure condition is approached.

If these principles hold true for bulk materials, then we might assume that the vibration energy tends to concentrate along the boundary surfaces, both the hopper boundary and arch or rathole surface. Damping is primarily geometric and dissipates the vibration energy quickly. Therefore the optimum position of the vibratory exciter would be in the vicinity of the arch or rathole abutment at the hopper wall. In addition, low frequency vibration would penetrate into the bulk material further than high frequency vibration. Fatigue of the bulk material sample would be accomplished by repetition of the force until instantaneous shear occurs.

3.3.2 PROPAGATION THROUGH HOPPER STRUCTURES

The distribution of vibration through beam and plate structures such as hoppers and bins is well established. The energy induced into the hopper plate radiates as waves outward from the point of application in a circular pattern and decreases geometrically. Acceleration measurements taken at radial distances away from repetitive pulsed exciters on conical hoppers confirm a geometric damping. Hopper plates and beams contain fundamental and modal frequencies. A forcing frequency that is a fractional multiple of the fundamental frequency causes the formation of nodes and antinodes, and amplification of motion occurs at the fundamental frequency. Amplification of displacement increases as the ratio of frequencies approaches unity. Figure 7 is a photograph of two oscilloscope traces superimposed one on the other. The traces are the acceleration response of a ½" steel plate freely supported while being vibrated by a sinusoidal exciter. The longer wave from is the acceleration response at a forcing frequency of 1500 cpm. An increase in frequency to 1600 cpm produces a secondary wave form at the fundamental frequency. The response of the system is altered significantly as shown in Figure 7.

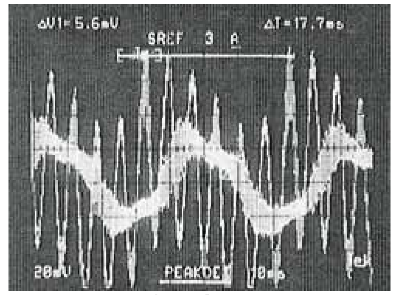


Figure 7: Response of a simply supported plate, liner sinusoidal exciter at two forcing frequencies

Tuning a low frequency sinusoidal vibration exciter to a much higher fundamental frequency of the hopper wall is one way of achieving high frequency excitation at the wall to reduce wall friction without resonating the structure to failure. A number of procedures exist for calculating the fundamental frequencies in structures but are only useful for estimation purposes. Tuning of this type requires a frequency response analysis of the existing hopper structure.

Use of high frequency vibration exciters (>100 Hz) on hopper structures, without knowing the actual fundamental frequencies, may be unwise due to the possibility of developing resonance in the structure, which could cause structural failure. The degree of damping in the structure tends to prevent vibrations of excessive amplitude, and the system model must include the damping influence of the bulk material mass. Strain gage testing performed by AEA O'Donnel Inc. show the damping effect of full versus empty for two conical hoppers under repetitive pulsed vibration. Figure 8 shows a five-fold decrease in strain from the empty to the full conditions.

STRESS MEASUREMENTS – IMPULSE FORCE VERSUS FREQUENCY

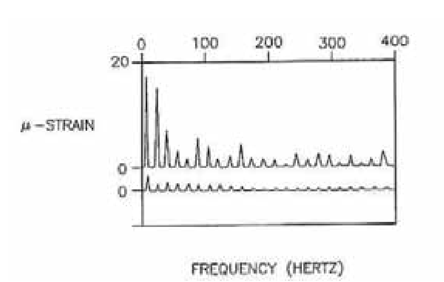


Figure 8: Strain vs. frequency – full and empty hopper, repetitive pulse exciter

The application of a pulsed force is to excite the hopper plate at its fundamental and modal frequencies. The magnitude of excitation decreases with increasing frequency mode. Figure 8 also shows the occurrence of high frequency excitation at the modal frequencies with decreasing strain.

The repeated application of the driving force introduces the possibility of fatigue. For a repetitive pulse type exciter, the peak stress due to impact will be higher than the peak stress caused by an equivalent sinusoidal type exciter. Figure 9 is a strain gage recording of a 6" diameter repetitive pulse exciter mounted on an empty hopper. Distinguishable are the sinusoidal stress components produced by the work of the piston mass and the impact stress component resulting from the kinetic energy transfer. The impact stress is roughly 5 times the magnitude of the sinusoidal component of stress.

STRESS MEASUREMENTS – IMPULSE FORCE VERSUS TIME

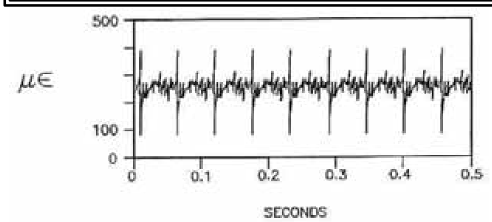


Figure 9: Strain vs time, conical hopper, repetitive pulse exciter

4.0 SELECTION CRITERIA

Ideally, selection of vibration exciters would include an analysis of bulk material and hopper structure characteristics. Unfortunately, predicting the required magnitude, frequency, and type of vibrational force to eliminate flow restrictions, considering all significant characteristics, has not been developed. Selection of vibration exciters for hoppers today is still largely empirical.

4.1 SIZING

The primary consideration for the pneumatic piston sinusoidal and pulse type vibration exciters is hopper wall thickness. The assumption is that the exciter must induce maximum stress on the hopper wall adversely affecting the hopper structure. The maximum fiber stress in the hopper plate is a function of the applied force and inverse square of the plate thickness. For sinusoidal piston type vibrators driven by a pressure force acting on the piston diameter, this yields a linear relationship between wall thickness and required piston diameter. The secondary consideration is mass of material in the hopper. The assumption is that the vibrational force must apply energy to a system which includes some portion of the bulk material mass. Charts are provided by most manufactures of vibrator piston size vs. hopper mass and wall thickness. Frequency and work are rarely considered. Sizing for rotational sinusoidal vibration exciters is based strictly on hopper mass. The empirical relationship is that the required vibrational force output is equivalent to 10 percent of the mass. Frequency and torque are not considered, and therefore consistent sizing practices between manufacturers is not available. Several manufacturers use a 5 percent rule with fixed maximum frequency of 1800 cpm.

4.2 MOUNTING REQUIREMNTS

4.2.1 LOCATION

Most manufacturers recommend the vibration exciter to be mounted approximately ¼ to 1/3 the length of the hopper wall as measured from the discharge. The assumption is that the arch or rathole formation normally occurs in the lower half of the hopper section. However, the location of the flow obstruction may not be readily apparent. Such factors as material loading direction, the use of inserts, and poor feeder interface can cause flow obstructions at other locations within the hopper. The ideal case is one in which the vibration exciter is located adjacent to the abutment of the flow obstruction.

4.2.2 HOPPER REINFORCEMENT

Both repetitive pulsed and sinusoidal exciters produce linear force vectors perpendicular to the hopper wall and are mounted directly. The application of force is perpendicular to the wall, and therefore stresses on the mount assembly area are in tension and compression and forcing frequencies are low and generally well below hopper natural frequencies.

Rotational type exciters require structural reinforcement along the length of the hopper section to reduce shear stresses, increase wall stiffness, and increase hopper natural frequency. The rotational force vectors align parallel to the hopper wall placing the mount in direct shear. The stiffener, usually a structural channel, enlarges the area of the welded connection reducing stress. The channel stiffener also increases wall stiffness to minimize acceleration of the exciter. Acceleration levels above 10g's on electromechanical exciters can cause excessive slip between the rotor and stator leading to failure. Increasing the stiffness of the wall increases the natural frequency and thus minimizes the possibility of resonance.

4.2.3 CONNECTION TO THE HOPPER

The connection of the vibration exciter to the hopper wall must maintain effective contact between the exciter and wall surface during operation. Mounting plates and structural stiffeners should be skip welded as opposed to continuous welded. This prevents a poor weld joint from fracturing the entire joint connection. For carbon steel, generally low hydrogen

(E7018) electrodes are used for high ductility. Fasteners should contain high strength and ductility (ASTM 325 or ASE Grade 5). Course threads are normally preferred. Fastener joints should be torque tightened to manufacturer recommendations upon initial installation and re-tightened after several hours of operation.

4.3 CONTROL

The proper control of vibration exciters is key in eliminating an arch or rathole formation and re-establishing material flow. Vibration exciters on bins and hoppers should only be operated when the bulk material is in a potential flow mode: the hopper discharge gate must be open, the discharge feeder activated, and the flow of material impeded by the arch or rathole formation. If the material does not have access to expand its volume, the vibrational energy may consolidate the bulk material and increase strength. Feedback devices such as level detectors or flow switches can be incorporated into the control logic to monitor the flow status and activate the exciter accordingly.

5.0 EXAMPLE CASES

5.1 CHARCOAL MANUFACTURER, SOUTH CAROLINA

An installation of pneumatic piston impacting vibrators on a 60 degree coal silo hopper at a charcoal manufacturing plant in South Carolina has allowed for substantial savings. Coal was transplanted to the facility by rail car and loaded into a 20 ft diameter silo with vibrating bin bottom. For years consistent material flow was the norm, until the purchasing department contracted a supplier of coal at half the cost. The drawback was that the new coal contained high moisture content, up to 15% and consisted mostly of fines. Reliable flow through the bin activator could only be achieved with coal at less than 6% moisture. Above 6%, arching would occur completely shutting down the process until manual sledge hammering re-established flow. A new carbon steel 70-degree cone section with 8” discharge opening and 5” pneumatic repetitive impacting vibrators were installed. The vibrators are activated several seconds after start-up of the discharge feeder. Sequential cycling of the vibrators provides consistent flow of coal with moisture levels up to 25%. The vibrators are mounted 120 degrees apart at various elevations on the lower cone section.

5.2 CHEMICAL MANUFACTURER, OHIO

A chemical manufacturing facility in Ohio was faced with a serious problem of removing ultra fine calcium carbonate powder from several storage silos. The material characteristics were constantly changing preventing consistent gravity flow. The cone section on the existing silos had been severely damaged by repeated sledge hammering. A study was conducted to find the most cost effective solution. Bin activator and external vibrators were compared and because of the low initial cost, and ability to test the vibrators on the actual installation, vibratory flow aids were installed over the weekend. A 5” diameter pneumatic piston repetitive pulse vibrator was installed near the discharge opening. A backing plate between the vibrator and hopper wall was added to increase wall stress. Two (2) additional vibrators were located at higher elevations 120 degrees apart. The vibration system is activated via the rotary air lock start signal. The vibrators are sequentially cycled. Since the initial start up four years ago, the facility has not experienced any flow problems. Consistent material throughput has resulted in more reliable production scheduling, higher production output and improved operator safety.

6.0 SUMMARY

Properly applied vibration to bins and hoppers increases bulk solid flowability without causing adverse affects to the hopper structure. The objective to develop flow is to reduce sliding friction at the hopper wall and strength within the bulk material. The response of the hopper wall and bulk material are dependent upon the type, magnitude and frequency output of the vibration exciter.

Using simple system models and correlating principles of soil dynamics to bulk solids, it appears that the repetitive pulse exciter produces the optimum response. The short time duration pulse provides high frequency excitation at the hopper wall necessary to reduce sliding friction while the low frequency high amplitude repetition of the pulse allows the energy to penetrate into the bulk material, causing the necessary shear disturbance. Low frequency excitation also prevents the possibility of resonating the hopper structure to failure.

7.0 NOMENCLATURE

C	Damping coefficient	lb-sec/in
F,F	Peak Force	lbs
FREQ	Forcing frequency	Rev/min
g	Gravitational acceleration, G=384/in/s^2	in/s^2
K	Spring stiffness	Lb/in
M	Mass	Lbm
R_G	Radius of Gyration	in
T	Torque	In-lb
t	Time	Sec
x	Displacement	In
v	Velocity	In/s
W	Weight	Lbs
w	Forcing frequency	rad/s
w_2	Natural frequency	rad/s

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