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Theory of Vibrating Conveyors

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Theorie der Schwingförderer
Théorie des convoyeurs oscillants
Teoría de transportadores vibrantes
振動コンベアの理論
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نظرية الناقلات بالسير

Theorie der Schwingförderer

Zur Förderung von vorwiegend körnigem Schüttgut werden Schwingrinnen eingesetzt, die auf Federn abgestützt oder hieran aufgehängt sind.

Die Rinnen werden durch Kurbel-, Unwucht- oder Elektromagnetantriebe in eine schwingende Bewegung mit hoher Frequenz und kleiner Amplitude versetzt, wodurch das Fördergut mit einer Mikrowurfbewegung bewegt wird. Verglichen werden Baulängen der Rinnen, Fördergeschwindigkeiten und Schwingungsgrößen.

Bestimmend für die Rinnenauslegung ist die *Wurfkennziffer,* die das Verhältnis der maximalen vertikalen Rinnenbeschleunigung zur Fallbeschleunigung angibt.

Das Schwingungssystem wird meist unterkritisch in Resonanznähe abgestimmt mit dem Ziel, daß eine Änderung der Rinnenbeladung möglichst wenig die Fördergeschwindigkeit beeinflußt.

Die Fördergeschwindigkeit wird berechnet aus der Erregerfrequenz, dem Wurfwinkel und der Wurfzeit im Verhältnis zur Dauer der Rinnenberührung unter Berücksichtigung von Materialeigenschaften, Schichthöhe und Rinnenneigung.

Zur exakten Auslegung der Schwingrinnen wurden die Dämpfungskonstanten und Ankopplungsfaktoren in Abhängigkeit von den Betriebsbedingungen ermittelt.

Théorie des convoyeurs oscillants

Des couloirs vibrants qui s'appuient sur des ressorts ou y sont suspendus, sont installés pour transporter des matières en vrac qui sont surtout granuleuses. Les couloirs où les matières à transporter sont animées par une micro-projection, sont soumis à un mouvement oscillant à haute fréquence et faible amplitude sous l'impulsion d'une manivelle, d'un balourd ou d'un électro-aimant. On compare ici les longueurs hors tout des couloirs, les vitesses de transport et les amplitudes des oscillations.

L'*indice de projection* qui donne le rapport de l'accélération verticale maximale des couloirs par rapport à l'accélération de la chute, est déterminant pour la conception des couloirs. Le système vibratoire est déterminé, la plupart du temps, sous-critique en voisinage résonant avec l'intention qu'une modification de la charge des couloirs influence aussi peu que possible la vitesse de transport. La vitesse de transport est calculée à partir de la fréquence des excitateurs, de l'angle de projection et du temps de projection par rapport à la durée de contact avec les couloirs en tenant compte des propriétés du matériel, de l'épaisseur de la couche et de la déclivité des couloirs. Pour une conception exacte des couloirs oscillants, on a determiné les constantes d'amortissement et les facteurs d'accouplement selon les conditions de fonctionnement.

Teoria de transportadores vibrantes

Para el transporte de materiales a granel predomina el empleo de canales vibrantes, los cuales estan apoyados a muelles ó cuelgan de ellos.

Los canales son movidos a manivela, exéntricamente ó por electromagnetismo, transmitiendo así movimientos vibradores de alta frecuencia y pequeña amplitud, de forma que los materiales a transportar sean desplazados con movimientos de micro-proyección. Las comparaciones se obtienen por medio de la longitud de la construcción de los canales, de la velocidad y de la magnitud de las vibraciones.

Determinante para el diseño de los canales es el *indice de proyección,* que dá la relación entre la aceleración vertical máxima de los canales y la aceleración de la caída.

El sistema vibrador es ajustado mayormente en la zona sub-crítica, cerca del punto de resonancia, con el fin que las variaciones en la carga influyan lo menos posible a la velocidad de transporte.

Los cálculos para determinar la velocidad de transporte se obtienen estimulando la frecuencia excitante, el ángulo y el tiempo de proyección, en relación al tiempo de contacto de los canales; teniendo en cuenta las características de los materiales, el espesor de la capa de material y la inclinación de los canales.

Para el exacto diseño de los canales se determinaron las constantes de amortiguación y los factores de acoplamiento, en relación de dependencia de las condiciones de trabajo.

Summary

Vibrating conveyors are installed to transport mainly granular goods; they are supported or suspended by springs. The troughs are driven and controlled by connecting rods, eccentric masses, or electromagnets, to perform a vibrating motion at a high frequency with a small amplitude, transporting the goods by micro-bounces.

The different controls are compared with respect to trough length, conveying speed, vibration values.

Characteristic for the conveyor design is the dynamic material coefficient, showing the ratio of maximum vertical trough acceleration and the gravitational constant.

The vibrating system usually is tuned subcritical near resonance, so that change of trough loads have only a small effect on the conveying speed.

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This conveying speed is determined by the exciting frequency, the angle of trough motion, the ratio of projection time to contact time, taking into account material properties, loading height and trough inclination.

For an exact conveyor design, the damping constants and the mass coupling factors of the load were determined, depending on operating conditions.

I. Introduction

For horizontal or slightly inclined transportation of granular bulk materials, over relatively short distances, vibrating troughs are often used. They usually consist of a trough with trapezoidal cross-section, hanging on or supported by springs, Fig. 1. The trough is forced by exciter units to vibrate (with high frequencies and small amplitudes) while inclined upward with an angle of incidence β at about 30° to the horizontal in the conveying direction, moving the product by successive micro-bounces. The usually open trough may be closed or of a tubular cross-section to transport dusty materials.

The advantage of this type of conveyor is its construction of steel or light metal plate being highly resistant against hot and chemically aggressive materials so enabling careful handling and transportation. Furthermore, there are few maintenance costs, because the trough can be easily cleaned and there are no rotating or returning elements to pick up the load.

The trough vibrates with a frequency of 5 to 100 Hz and an amplitude of 15 to 0.1 mm. By this motion, the bulk materials are accelerated in the direction of vibration in such a way that they lift off the trough and continue on a trajectory of micro-bounces. During this parabolic projection, the trough swings back and then accelerates again the impinging materials on their next forward motion. For the theoretical deduction, the path of the trough s_T is reduced into its horizontal and vertical components x_T and y_T.

Mathematically, the unidirectional path of the trough s_T may be written as:

$$s_T = r\cdot(1 - \cos 2\pi ft)$$

where:

r = amplitude (1)
f = frequency
t = time

The acceleration of the trough in the direction of vibration takes the form:

$$\ddot{s}_T = 4\pi^2 f^2\cdot r\cdot\cos 2\pi ft \qquad (2)$$

The bulk materials lift off the trough, when the vertical component of the trough acceleration $\ddot{y}_R$ becomes greater than the gravitational constant g:

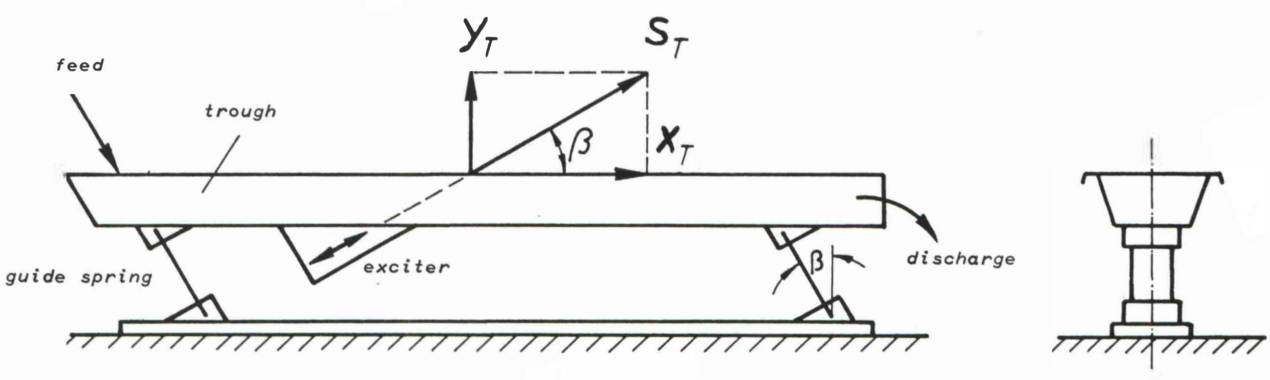
$$\ddot{y}_T = \ddot{s}_T\cdot\sin\beta = 4\pi^2 f^2\cdot r\cdot\cos 2\pi ft\cdot\sin\beta \qquad (3)$$

The ratio of maximum vertical acceleration to the acceleration due to gravity gives the characteristic value for the motion of the load and is called *dynamic material coefficient* [1]:

$$\Gamma = \frac{\ddot{y}_{Tmax}}{g} = \frac{4\pi^2 f^2 r\sin\beta}{g} \qquad (4)$$

Γ has to be larger than unity and depends on the values of amplitude r, frequency f and angle of inclination β, which may be chosen by constructional means.

Fig. 2 shows a section of the trough in its oscillating motion $y = f(x)$ as a function of the path with the angle of inclination β, and also $x = f(t)$ as a function of time. The path of a load particle is drawn, following the sinusoidal motion of the trough from the impact to the lift off, but then the particle flies along a parabolic trajectory to the next impact point and is then forced again to follow the path of the trough. The projected distance is much longer than the carried path, therefore the contact time is small compared to the projection time, so that even for wear resistant materials, e.g.



$s_T = (1 - \cos 2\pi\cdot f\cdot t)$

$\ddot{s}_T = r\cdot(2\pi\cdot f)^2 \cos 2\pi\cdot f\cdot t$

$\ddot{y}_T = s_T \sin\beta$

$\Gamma = \frac{r\cdot 4\pi^2\cdot f^2\cdot\sin\beta}{g}$

r = amplitude of trough path
f = frequency of excitor unit
β = angle of incidence
t = time
Γ = dynamic material coefficent

Fig. 1: Basic sketch of a vibrating conveyor

corundum, there is only a very small trough abrasion. Important is the ratio of projection time t_i — t_l and exciter period T, shown in the middle diagram.

It depends on the correct choice of trough layout, so that the bulk material does not hit the forward swinging trough either too early or too late in order to gain an optimal conveying

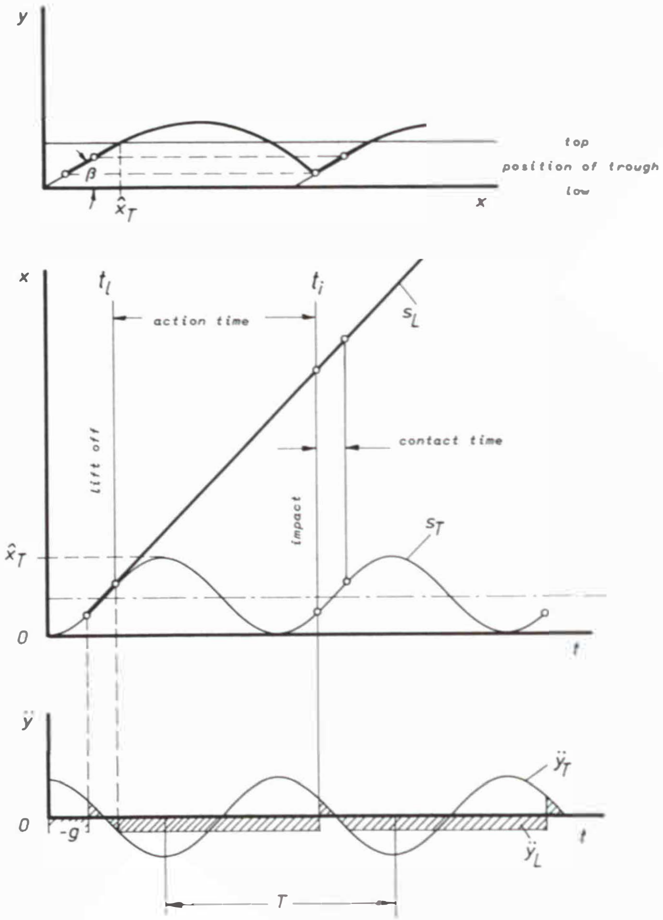


Fig. 2: Trough and load motion as a function of path and time

speed. The lower diagram shows the ratio of maximum vertical trough acceleration relative to the acceleration of gravity g.

Characteristic for the trough application is the *dynamic machine coefficient* K, defined by the ratio of the maximum trough acceleration in the vibration direction to the acceleration due to gravity, i.e., the ratio of dynamic stress compared with static stress [2]:

$$K = \frac{\ddot{s}_{Tmax}}{g} = \frac{4\,\pi^2 f^2 \cdot r}{g} \qquad (5)$$

The relationship between K, r and f is shown in the diagram of Fig. 3. Practically, vibrating conveyors are designed with K = 3 to 10.

The conveying speed may be theoretically computed with frequency f, angle of inclination β and ratio n of projection time t_i — t_s to period T, n being given as implicit function of:

$$n = \frac{t_i - t_l}{T}$$

and

$$\Gamma = \sqrt{\left(\frac{\cos 2\pi n + 2\pi^2 n^2 - 1}{2\pi n - \sin 2\pi n}\right)^2 + 1} \qquad (6)$$

The diagram in Fig. 4 shows n as function of Γ with the value n = 1 for the dynamic material coefficient Γ = 3.3, that means the projection time t_i — t_l is equal to the period T for Γ = 3.3.

$$v = \eta_M \cdot \eta_H \cdot \eta_J \cdot \frac{g}{2} \cdot \frac{n^2}{f} \cdot \cot\beta$$

η_M = constant due to material properties <1 for low load densities and small grain size
= 0.8 to 0.9 for spec. heavy, granular, dry material
= 0.8 to 0.1 for share of grain size up to 0.3 mm >20 %
= 0 for grain size <0.06 mm (no transport)

η_H = constant due to height of load
= 1.0 for small heights of load
to 0.75 for heights of load up to 300 mm

η_J = constant due to inclination >1 for downward conveying
1 for upward conveying up to 15°, depending also on friction between trough and load

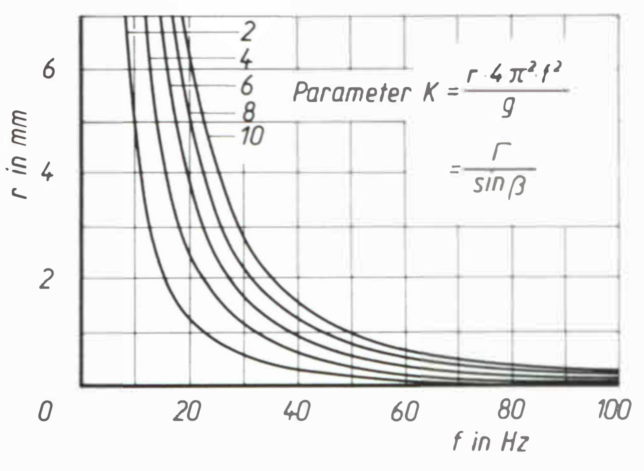


Fig. 3: Relation between dynamic machine coefficient K, amplitude r and frequency f

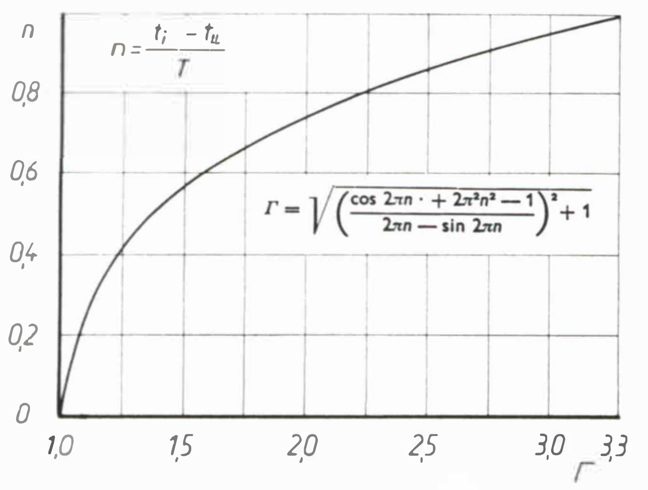


Fig. 4: Conveying speed v determined by dynamic material coefficient Γ and empirical operating constant η

Material properties, height of load, and trough inclination are considered by a constant η with empirical values.

The conveying speed

$$v = \eta_M \cdot \eta_H \cdot \eta_J \cdot \frac{g}{2} \cdot \frac{n^2}{f} \cdot \cot\beta \qquad (7)$$

increases quadratically with the relative projection time and with decreasing angle of action. The conveying speed relates inversely proportional to the frequency.

Fig. 5 shows the optimum angle of action β as a function of the dynamic machine coefficient K, in order to gain a high conveying speed. The related dynamic material coefficient is also shown. It is now possible to compute the mass flow as the product of conveying speed, bulk cross-section, and bulk density.

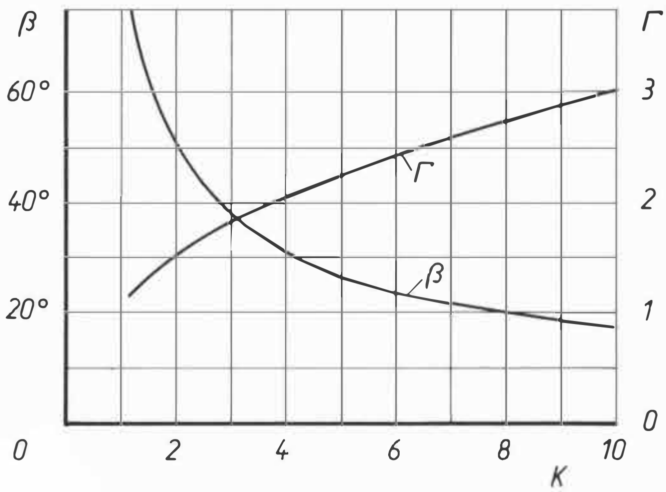


Fig. 5: Optimum angle of incidence β for achieving a high conveying speed v, depending on the dynamic machine coefficient K, and the related dynamic material coefficient Γ

2. Vibration Control

The following types of control are used to vibrate and cause motion:

— Control by connecting rod,
— control by eccentric masses and
— control by electro-magnet.

2.1 Control by Connecting Rod

This control is used typically for longer vibrating conveyors. Often the supports exist of inclined plate springs of steel or glass fiber reinforced plastics. The natural frequency of the vibrating system, determined by the vibrating masses and the spring constants, is always tuned low to protect the environment against the vibrating conveyor. The conveyor may also oscillate near resonance, if a potential spring supports the trough against a compensating frame, which itself is supported to the ground in such a manner that the vibrations are isolated. In these cases it is an advantage to imply a coupling spring between the connecting rod and the trough in order to have a free mobility of the trough without a fixed vibration path (Fig. 6, upper part).

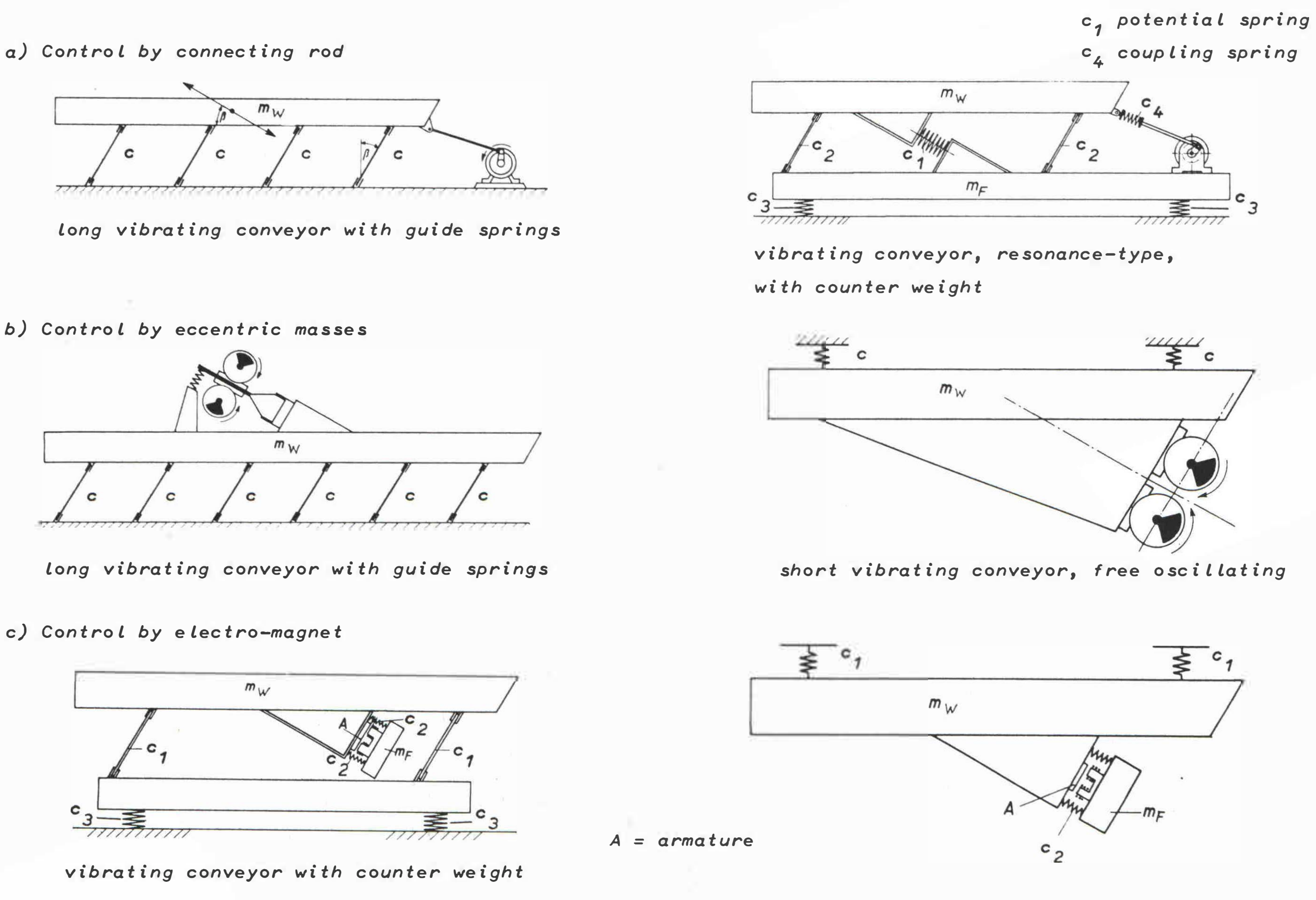


Fig. 6: Controls for vibrating conveyors

2.2 Controls by Eccentrics

These controls are composed of two eccentric rotating masses of equal size. The centrifugal forces result in a force in the direction of the vibration, the transversal components effectively cancel each other out. Supports exist in the form of plate or spiral or rubber springs. If the vibration direction is fixed by using plate springs as supports, there should also be installed a helper spring — as shown in Fig. 6, center — transverse to the vibration direction, so that the resultant centrifugal force of both eccentric masses does not induce large tensile forces in the plate springs when the inertia force resultant is not exactly perpendicular to these leaf springs.

2.3 Electro-magnetic Control

When using an electro-magnet for control, this has to be connected to the vibrating trough mass by strong springs and has also to be weighted by an additional counterweight, as it works against an armature plate, fixed to the trough (Fig. 6, lower part).

The magnet is supplied with 50 Hz directly from the alternating current mains respectively with 25 Hz from a halfwave rectifier. The electro-magnetic force is proportional to the square of the current, and vibrators controlled by these electro-magnets oscillate with 100 respectively 50 Hz.

By using thyristor control, the vibration frequency of the electro-magnets may be reduced to 25 Hz. The amplitudes are very small, namely 0.1 to 3 mm; correspondingly, only a small conveying speed from 0.01 to 0.3 m/s may be realized.

Due to the tuning near resonance, there follows an advantage of control by electro-magnets. Switching off the exciting current, the decreasing vibration acceleration stops the conveying motion after several finite periods within a very short time.

For comparison, Fig. 7 shows frequencies, amplitudes, conveying speeds and usual trough lengths for all three types or modes of control:

Controls by connecting rod work at 5 to 15 Hz with amplitudes from 3 to 15 mm and with conveying speeds from 0.2 to 0.8 m/s.

Controls by eccentrics realise frequencies from 10 to 25 Hz with amplitudes from 5 to 0.5 mm for conveying lengths from 0.5 to 10 m and conveying speeds from 0.4 to 0.05 m/s.

Furthermore, Fig. 7 gives a combination of the equations for exiting force, static and dynamic foundation force, as well as the tuning of the vibration system.

The 8 m long trough, shown in Fig. 8, controlled by eccentrics, serves to convey dusty materials and therefore is designed with a closed box-section. A larger number of supporting leaf springs takes care of a vibration motion of the trough without causing bending, such are situated close together and near the control unit. The potential springs are tuned subcritical and the eccentric masses are synchronized by gears to run with 16 Hz exciter frequency.

Fig. 9 shows a vibrating conveyor controlled by eccentrics in the same way. The centrifugal forces are 2 x 470 kN and excite a 4 m wide trough filled with hot sinter material.

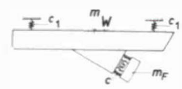
control: / unit	connecting rod 	eccentric masses 	electro-magnet 
frequency f — Hz	5...15...(25)	1o...25...(5o)	25; 5o; (100)
stroke s — mm	6...3o $s_W = 2r$	1...1o $s_W = s_D \cdot \frac{m_D}{m_W + m_E}$	o,1...2 $s = \frac{F_{exc}}{c_2} \cdot \frac{1}{1 - (f_{exc} / f_{nat})^2}$ $s = s_W + s_F$ $s_W \cdot m_W = s_F \cdot m_F$
conveying speed v — m/s	o,3...o,7	o,o5...o,4	o,o1...o,15
trough length l — m	2...2o spec. design up to 5o m	o,5...1o spec. design 3o...5o m	o,1...5 spec. design up to 1o m
exciting force F_{exc} — N	$F_{exc} = m_W \cdot 4\pi^2 \cdot f_{exc}^2 \cdot r$	$F_{exc} = (m_W + m_E) \cdot 4\pi^2 \cdot f_{exc}^2 \cdot r$	$F_{exc} = m_N \cdot 4\pi^2 \cdot f_{exc}^2 \cdot r_W \cdot \frac{1 - (f_{exc}/f_{nat})^2}{(f_{exc}/f_{nat})^2}$
static foundation force F_{stat} — N	$F_{stat} = (m_W + m_L) \cdot g$	$F_{stat} = (m_W + m_E + m_D + m_L) \cdot g$	$F_{stat} = (m_W + m_F + m_L) \cdot g$
dynamic foundation force F_{dyn} — N	$F_{dyn} \approx \Sigma c \cdot r + m_W \cdot 4\pi^2 \cdot f_{exc}^2 \cdot r$	$F_{dyn} = \Sigma c \cdot r$	$F_{dyn} = \Sigma c_1 \cdot r$
tuning of the vibration system	$f_{exc} \gg f_{nat}$ $f_{nat} = \frac{1}{2\pi} \sqrt{\frac{\Sigma c}{m_W}}$ $\Sigma c \ll 4\pi^2 \cdot f_{exc}^2 \cdot m_W$	$f_{exc} \gg f_{nat}$ $f_{nat} = \frac{1}{2\pi} \sqrt{\frac{\Sigma c}{m_W + m_E + m_D}}$ $\Sigma c \ll 4\pi^2 \cdot f_{exc}^2 \cdot (m_W + m_E + m_D)$	subcritical: $f_{exc} \approx 0{,}9\, f_{nat}$ supercritical: $f_{exc} \approx 1{,}1\, f_{nat}$ $f_{nat} = \frac{1}{2\pi} \sqrt{\frac{c_2 (m_W + m_F)}{m_W \cdot m_F}}$ $\Sigma c_1 \ll 4\pi^2 \cdot f_{exc}^2 \cdot (m_W + m_F)$
m_W = working mass m_F = free mass m_E = exiter mass m_D = double eccentric masses m_L = mass of load			

Fig. 7: Vibrating conveyors, controls and operating data

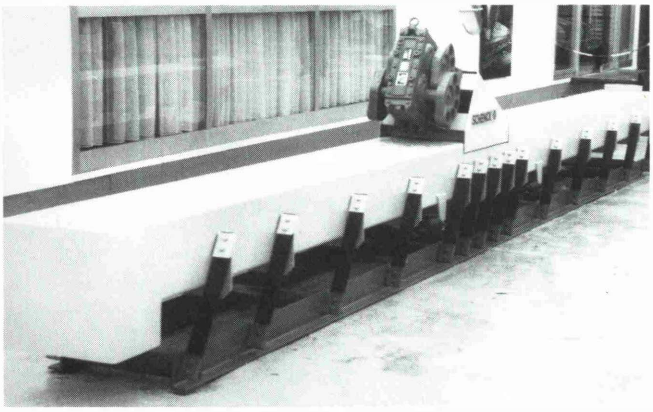


Fig. 8: Vibrating conveyors controlled by eccentric masses (Courtesy of Schenck)

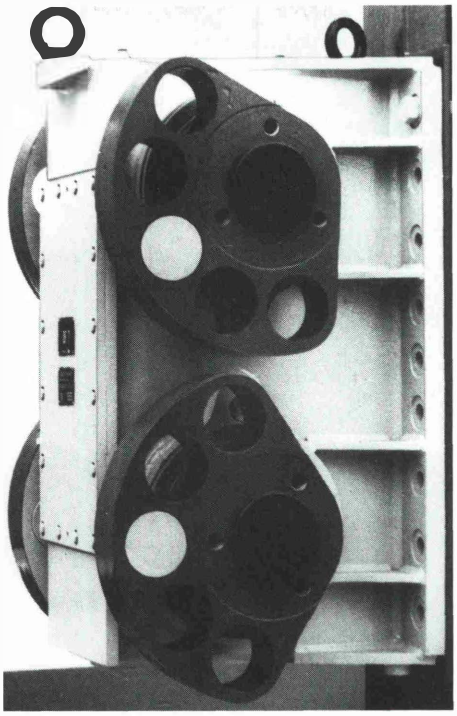


Fig. 9: Control unit by eccentrics for 2x470 = 940 kN centrifugal force, acting on a 4 m wide trough with hot sinter (Courtesy of Schenck)

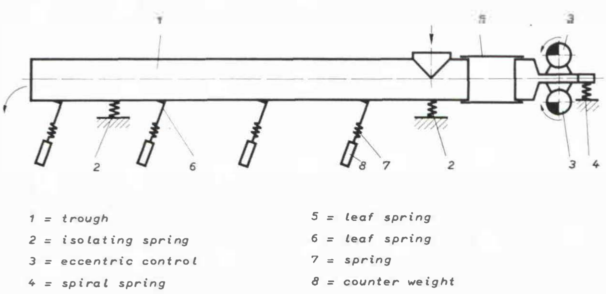


Fig. 10: Vibrating tubular trough with damping devices (Courtesy of Uhde)

A tubular vibrating conveyor is shown in Fig. 10. The reversely rotating eccentric masses synchronize each other during the starting period, since they are connected to the trough by two horizontal oriented leaf springs, supported by a vertical spiral spring.

Damping devices work perpendicular to the designed projection direction and support the relatively flexible conveyor tube. In this way, conveying lengths of up to 30 m may be run in one unit, the amplitudes remaining unaffected by superimposed tube bending vibration.

In Fig. 11, a vibrating conveyor is composed of several tubular troughs as mentioned before, working as collecting

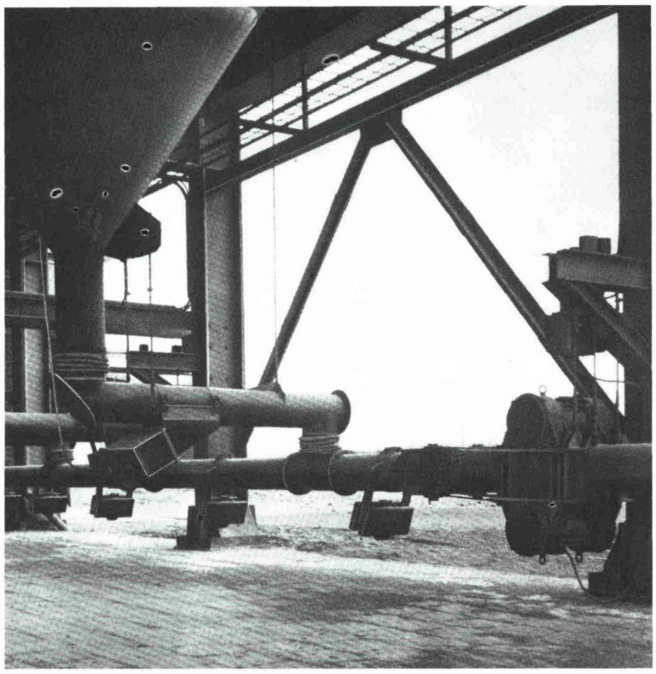


Fig. 11: Vibrating conveyor in a LD-steel plant (Courtesy of Uhde)

section in a LD-steel plant. At the transfer points, packings are located to separate the vibrations of each conveyor unit.

Fig. 12 shows several dosing troughs in a steel plant. Here, the vibrating conveyors are controlled by electro-magnets, and they remove different materials from bunkers to a belt conveyor.

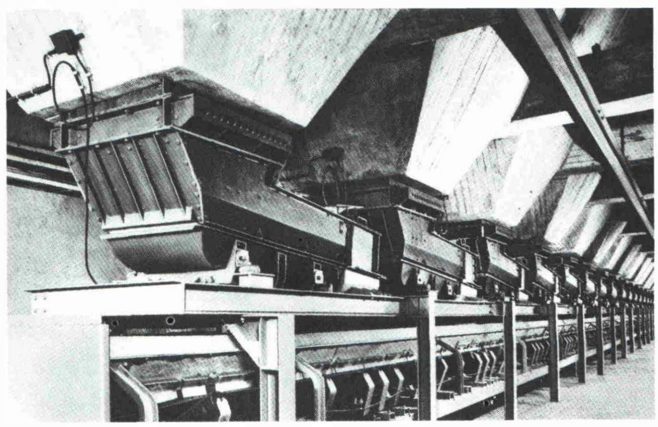
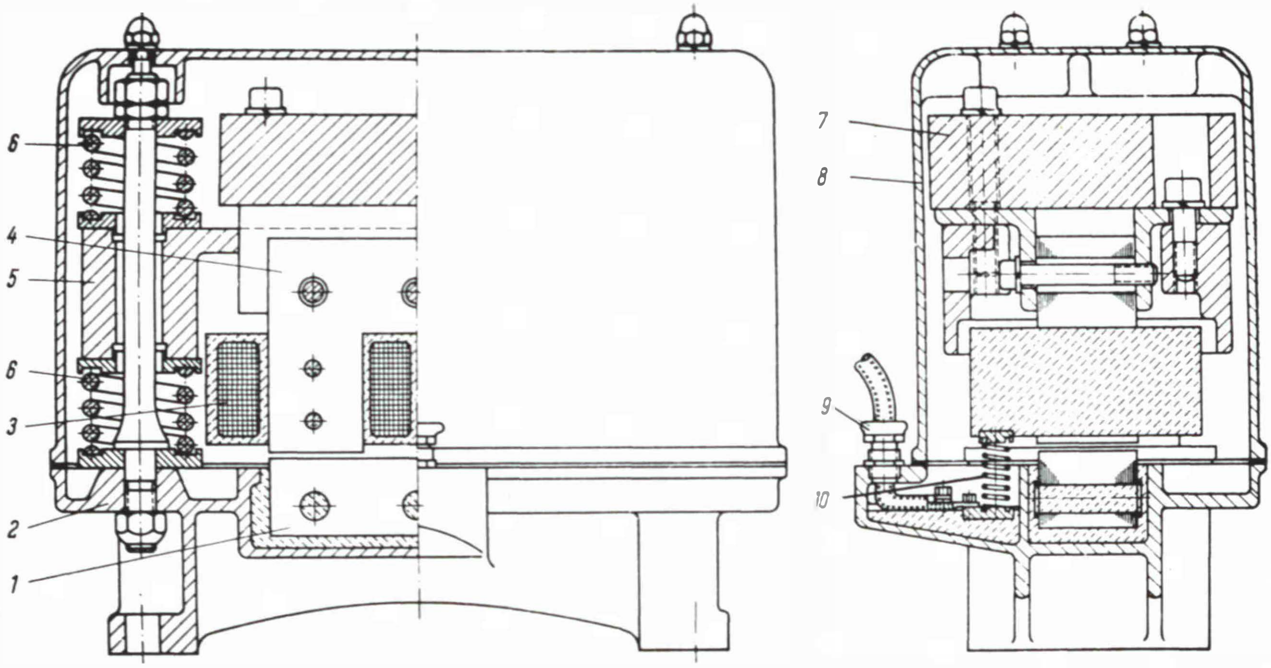


Fig. 12: Dosing troughs in a steel plant, controlled by electromagnets (Courtesy of AEG Telefunken)



1 = armature
2 = connecting frame
3 = coils
4 = iron core
5 = counter weight
6 = spiral springs
7 = additional weight
8 = aluminium cage
9 = power supply cable
1o= spiral springs

Fig. 13: Electro-magnetic control (Courtesy of Uhde)

The drawing of a vibrating conveyor, controlled by electromagnets, is shown in Fig. 13. The iron core with its coils is supported to both sides by spiral springs, and a counterweight at the core serves for the tuning of the exciting frequency. The armature is fixed to the base of the trough.

In Fig. 14, this control unit is shown without cowl, and the strong version of springs and cage may be seen.

Due to very small amplitudes of higher frequency electromagnetic controls, it is necessary to design the trough base particularly for bending resistance.

Fig. 15 shows a free vibrating trough with spiral springs and three control units, which induce the exciting forces to the trough by triangular pushing plates. Transverse fins care for the lateral force transmission.

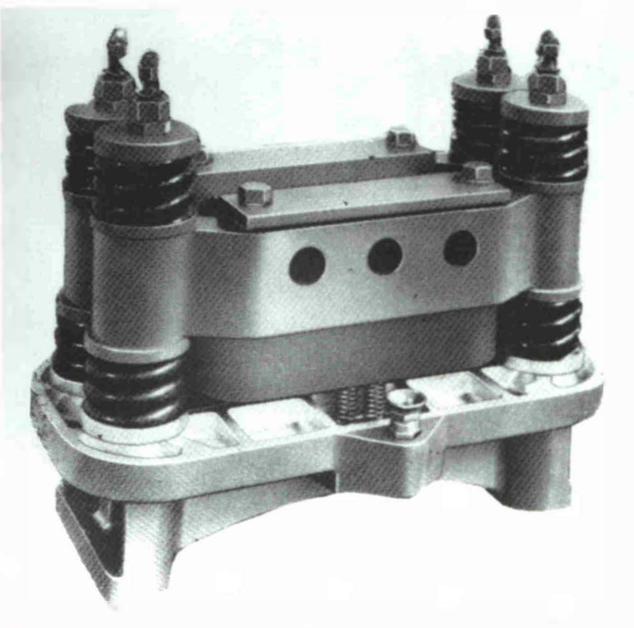


Fig. 14: Electro-magnetic control (Courtesy of Uhde)

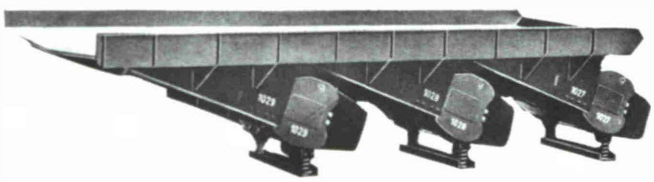


Fig. 15: Free vibrating conveyor with 3 electro-magnetic control units

The top of Fig. 16 shows the simplified equivalent system of a vibrating conveyor as free two-masses-vibrator, controlled by electromagnets. It exists with a working load and a counterweight connected by stiff springs, and a weak suspension of the working load to the foundation. The damping of the conveying load is assumed to act between working load and supports (or foundation). At most, 20 % of the conveying load is added to the mass of trough and armature plate to result the working mass, since only a small part of the carried materials is coupled to the vibration motion, depending on the dynamic material coefficient and the angle of action referring to extensive research work [3].

In order to keep the vibrating conveyor running with an exciting force as small as possible, the system is tuned to oscillate near resonance at

$$f_{exc} = (0.85 \text{ to } 0.90) \cdot f_{nat} \quad (8)$$

The bottom of Fig. 16 shows resonance curves for linear spring characteristics, from which the amplification V of the

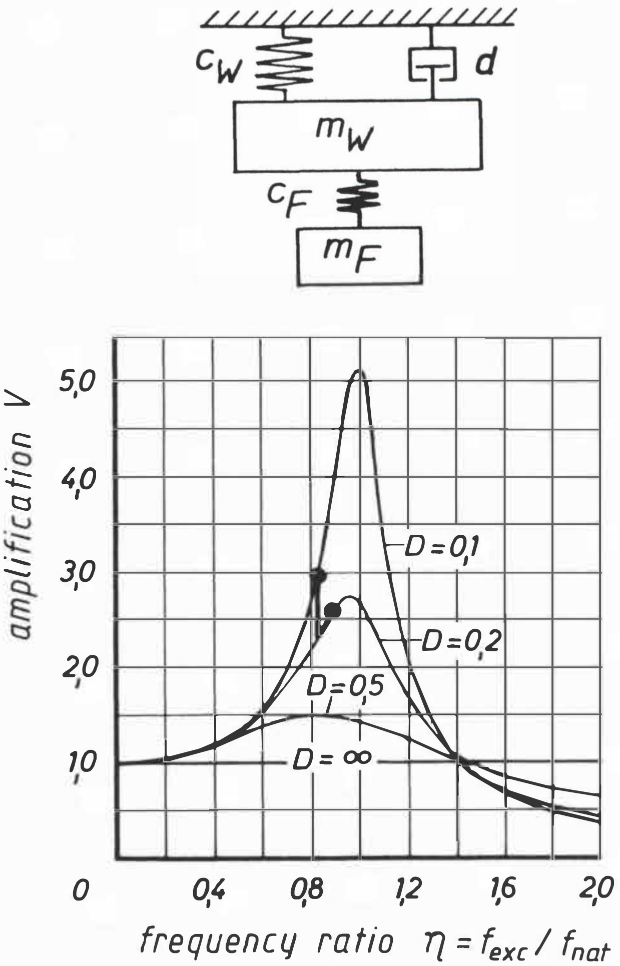


Fig. 16: Tuning of a vibrating conveyor controlled by electro-magnets

amplitudes relative to the static deflection may be seen as a function of the frequency ratio η between exciting frequency f_{exc} and natural frequency f_{nat} and of the damping constant D as parameter.

The advantage of this subcritical tuning is as follows: Increasing loading leads to an increased damping, as shown in Fig. 16, which induces smaller amplitudes; but at the same time the working mass is increased and results in a lower natural frequency, so that the unwanted damping effect is partially compensated by tuning towards resonance yielding larger amplitudes.

On the opposite, a supercritical tuning diminishes the amplitudes due to both arguments when changing from partial load to full load.

With progressive spring characteristics, as shown in Fig. 17, the amplification remains nearly constant for partial and full load. In such a case, only subcritical tuning is possible, since for supercritical tuning the instable section causes a jump of the amplitudes to the upper curve part with diminishing tuning.

In 1981, extensive research work will be completed at the *Institut für Fördertechnik der Universität Hannover* (Institute of Conveying Technology) investigating the problem of the retroaction of the conveying load to the vibrating conveyors by theoretical and experimental means [4].

Fig. 18 shows the testing installation, composed of 7

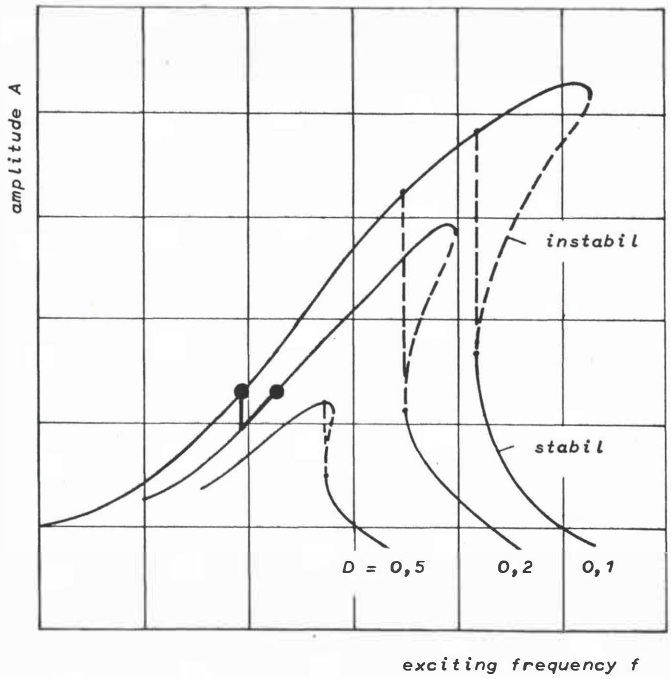


Fig. 17: Resonance curve for progressive spring characteristic

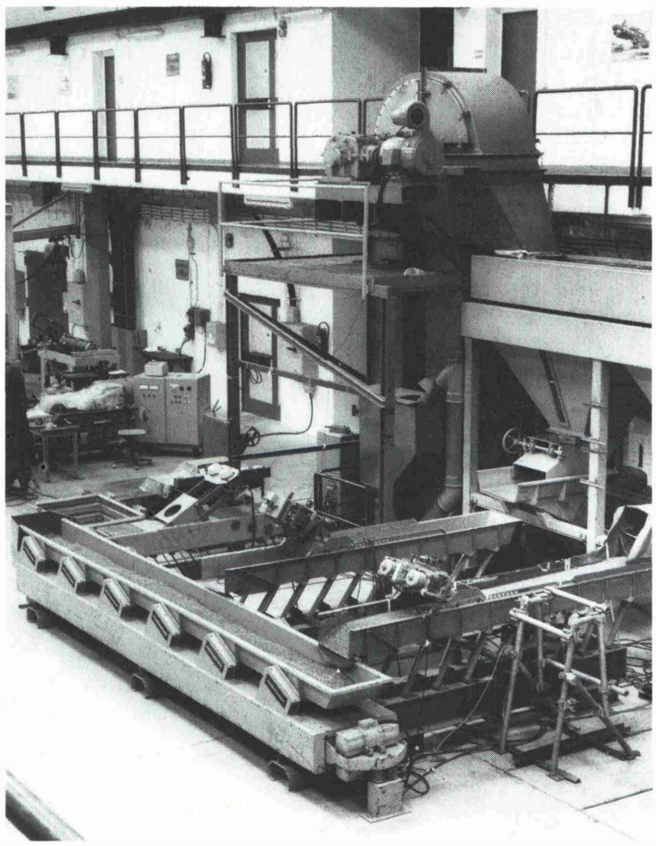


Fig. 18: Conveying cycle with belt-type bucket conveyor and vibrating conveyors

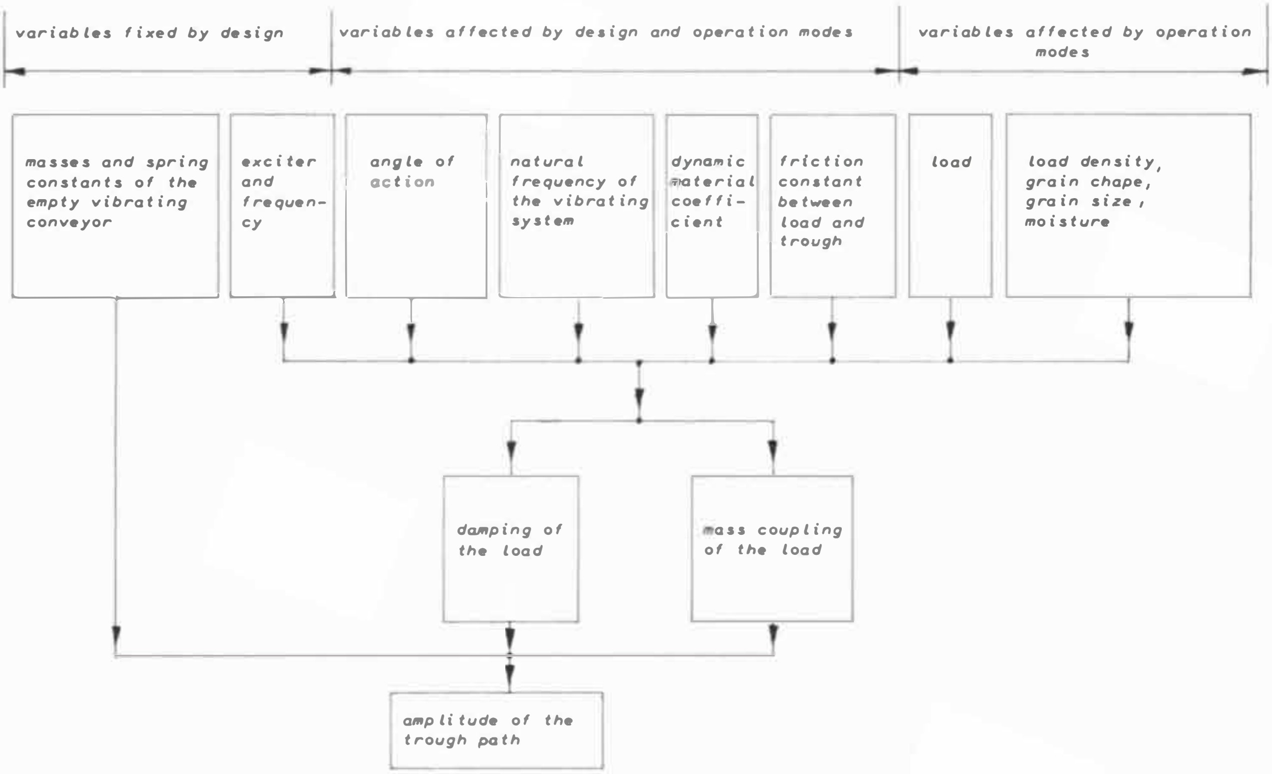


Fig. 19: Retro-action between load and vibrating conveyor (Courtesy of Steinbrück)

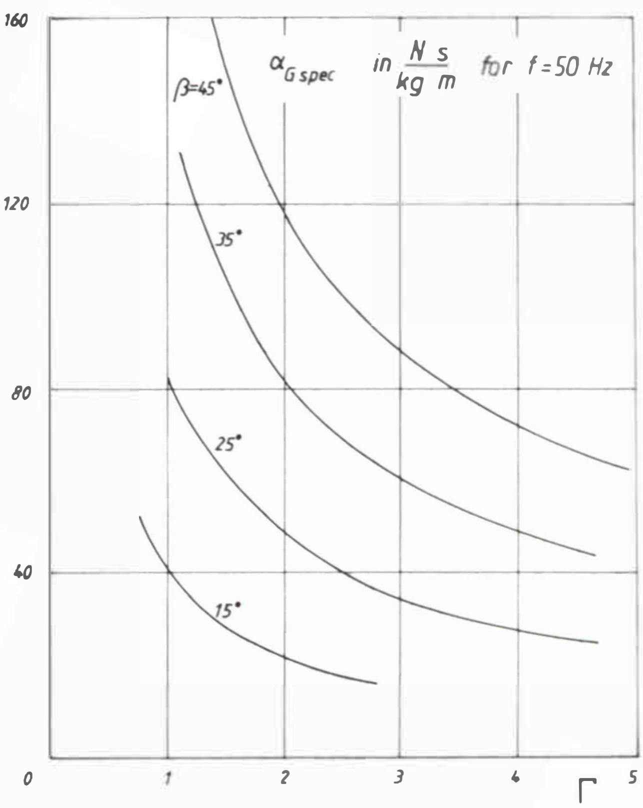


Fig. 20: Specific damping constant α_{Gspec} depending on dynamic material coefficient Γ and angle of incidence β (Courtesy of Steinbrück)

vibrating conveyors with different design and controls and a belt-type bucket conveyor as return run, the transported load running in a cycle. Transported goods, angle of incidence, frequency and amplitudes are variable within a large range. Many electronic instruments and data processing equipment are employed to analyze and evaluate experimental results.

Fig. 19 summarizes all investigated control factors, changing the trough path amplitude by damping and mass coupling of the transported goods compared to no-load run of the trough. Some results of this work are:

Fig. 20 shows the damping constant α, referred to load mass, depending on the dynamic material coefficient Γ with the angle of action β as parameter. In this case, the trough is controlled by electro-magnets at a frequency of 50 Hz with a loading height of 200 mm. The damping decreases strongly with higher trough acceleration and smaller angle of incidence.

Fig. 21 shows mass coupling factors as a function of dynamic material coefficient Γ; the parameters are friction constant μ between load and trough and different angles of incidence β.

The conveying speed as function of the dynamic coefficient Γ is shown in Fig. 22; as parameter, the exciting frequency is changed from 16 to 60 Hz; the angle of incidence β remains constant at 25° and the loading height was 200 mm.

This investigation essentially expands earlier research works of the Institute, concerning the investigated areas. New knowledge about the phenomenon of damping and mass coupling are published, leading to practical application for design and operating of vibrating conveyors.

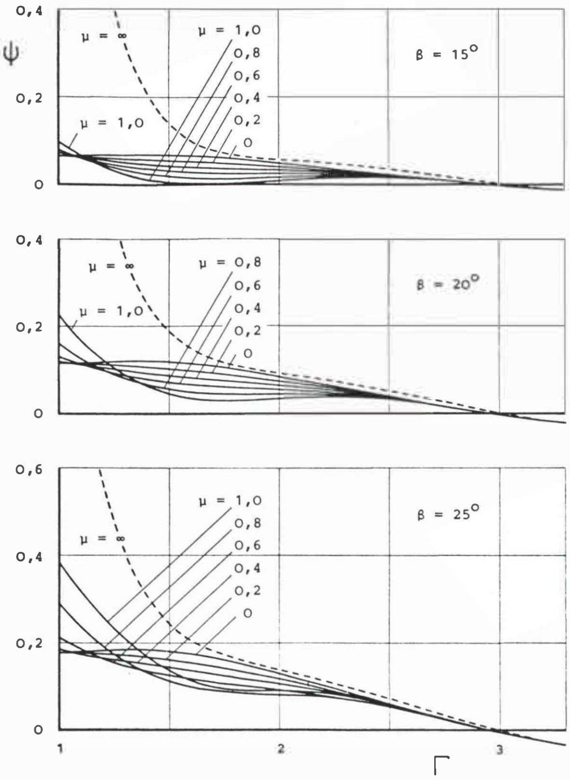


Fig. 21: Mass coupling factor ψ as function of dynamic coefficient Γ and friction constant μ (Courtesy of Steinbrück)

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Fig. 19, 20, 21, 22: Klaus Steinbrück, Frankfurt, W. Germany

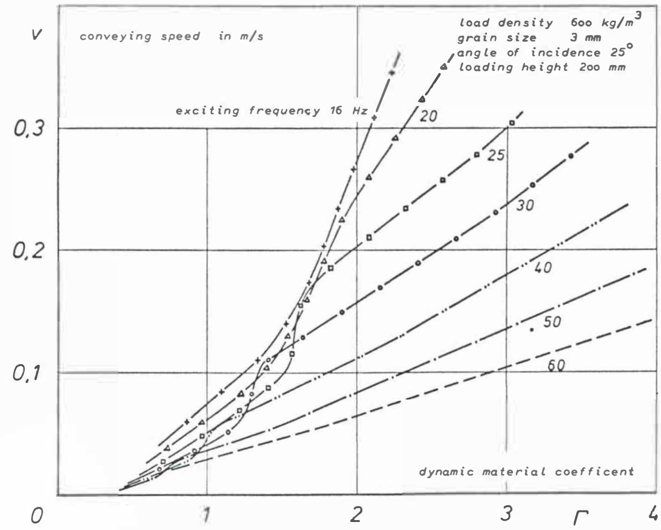

Fig. 22: Conveying speed on a vibrating trough, depending on dynamic material coefficient and exciting frequency (Courtesy of Steinbrück)

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