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Thermal Stress and Load Distribution of Belt Conveyor Drives

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THERMAL STRESS AND LOAD DISTRIBUTION OF BELT CONVEYOR DRIVES

SUMMARY

With regard to load sharing in the case of conventional multimotor A.C. drives for belt conveyor systems during their steady-state operation, in the first part of the contribution it is demonstrated that the more or less different utilization of the installed drive powers may occur with multiple arrangements of driving pulleys. As a consequence, under certain conditions this can lead to the overloading of some of the motors, while at the same time the rated power of other drives is not fully utilized. The main parameters and their fundamental relationships are introduced. The basis is given for calculating the load distribution in two-pulley arrangements, and thus for determining the utilization of the installed rated power of the drives. It is stated that even small differences of the effective pulley diameters exert a large influence on the load distribution. And, in addition, when selecting and designing conventional belt conveyor drives, attention has to be paid to the tolerances and the gradient of their torque-speed-characteristics as well as to the influence of the different types of conveyor belt.

In addition to the load distribution, the problem of the thermal stress on conveyor belt drives during start-up is dealt with in detail. Especially with regard to the frequently used soft-start techniques to reduce belt stress, particularly for high-capacity overland belt conveyor systems with high belt speeds, the resulting thermal energy in conventional drives with non-frequency-controlled A.C. motors is of great significance in the design of motors or fluid couplings and starter resistors. In the last part of the contribution, relationships are introduced between the limitation of the drive forces during start-up and the resulting reduced start-up accelerations, as well as equations for calculating the amount of energy loss and the corresponding thermal stress due to the slip of the drive units during their start-up. In addition, information is given that permits an estimation of the thermal stressability of squirrel-cage motors and fixed filling fluid couplings.

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1. Introduction

In the last 30 years belt conveyor systems have been developed into a means of transport for bulk materials with mass flows up to 37 500 t/h and with centres of pulleys of more than 13 000 m. This increase of parameters, expressing the latest state of continuous conveying technology, was only made possible by systems with higher belt speeds. For example, the great belt conveyor systems in the brown coal mines of Rheinbraun in the Federal Republic of Germany are operated with belt speeds up to 7.5 m/s.

In the design of belt conveyor systems with high belt speeds for the purpose of achieving large mass flows with a certain given cross-sectional area of the material on the belt, attention must be paid to the influence of the belt speed on the power consumption of such systems. This applies in particular to horizontal and slightly inclined belt conveyor systems in which the drive forces required are determined by the frictional resistances to motion. The latter increase with increasing belt speeds, provided that the systems have the same design of the belt and the same area of material cross-section. Under these preconditions, the increase of belt speed with regard to higher values of mass flows causes an increase of the drive forces, and thus also an over-proportional increase of the required drive powers which has to be transmitted to the belt by the driving pulleys.

Belt conveyor systems that have long centre distances and are operated with high belt speeds thus demand the installation of multi-pulley arrangements which do not only permit the installation of greater drive forces and drive powers. In addition, with an increased number of driving pulleys the initial belt forces, and thus the maximum belt tensile forces, can be limited. This advantage of multi-pulley arrangements is contrasted with the problem that, under certain conditions, the installed drive powers at the individual pulleys are utilized unequally. The first part of the contribution deals with the influences on the load distribution of belt conveyor drives and with methods to calculate them.

In the second and last part of this contribution the problem of the thermal stress of belt conveyor drives during start-up is analyzed in detail, since it is more and more frequently the case that long and partly extreme durations of start-up procedures occur. The latter are due to the growing application of soft-start techniques for the purpose of minimizing belt stresses on the one hand, and the increase of the capacity of belt conveyor systems by installations with higher belt speeds on the other hand. As a result of the high slip during these operational phases, the longer start-up times cause increasing energy losses in conventional A.C. drives, which are supplied directly, i.e. without frequency converters from the three-phase network.

2. Distribution of Peripheral Forces of Driving Pulleys in Multi-Pulley Arrangements during Steady Operating Conditions of Belt Conveyor Systems

Belt conveyor systems with large distances between the centres of pulleys and high mass flows exclusively use multi-pulley arrangements. As far as two driving pulleys are concerned, they are frequently part of head stations, in many cases combined with single-pulley tail stations. In the case of additional tail drives, the most favourable distribution of drive forces for achieving minimum belt tensile forces is obtained, when the resistances to motion in the upper strand are overcome by the drives at the head station, and when those of the lower strand correspond with the drive forces at the tail station. The following explanations apply only to driving pulleys with positive peripheral forces, i.e. power transmission from the network to the belt conveyor system. Figure 1 shows two typical arrangements of multimotor two-pulley drives with the positive peripheral forces F_1, F_2 of the driving pulleys 1, 2. The parameters used are dealt with in Section 3.2.

The introduction of multi-pulley arrangements required with respect to the increased parameters of modern belt conveyor systems does not only allow the installation of greater drive powers. But also, with regard to the transmissibility of raised drive forces, the power transmission by an increased number of driving pulleys permits lower initial belt forces, and thus lower maximum belt tensile forces. This advantage when compared with corresponding single-pulley drives is due to the enlarged useable angle of wrap of all the driving pulleys. However, the complete utilization of the total of all angles of wrap is not possible. This gives rise to the question of the transmissibility of the peripheral forces of the driving pulleys. This problem is not covered by the following explanations, since it is assumed that the corresponding aspects are well known.

Independent of the transmissibility of the installed drive powers at each of the pulleys during the steady-state operation of belt conveyors, different utilization of the drives occurs with multiple arrangements of driving pulleys. The reasons for unbalanced load distributions, i.e. ones which are non-proportional to the installed drive powers at the driving pulleys, are differences between the characteristics of the

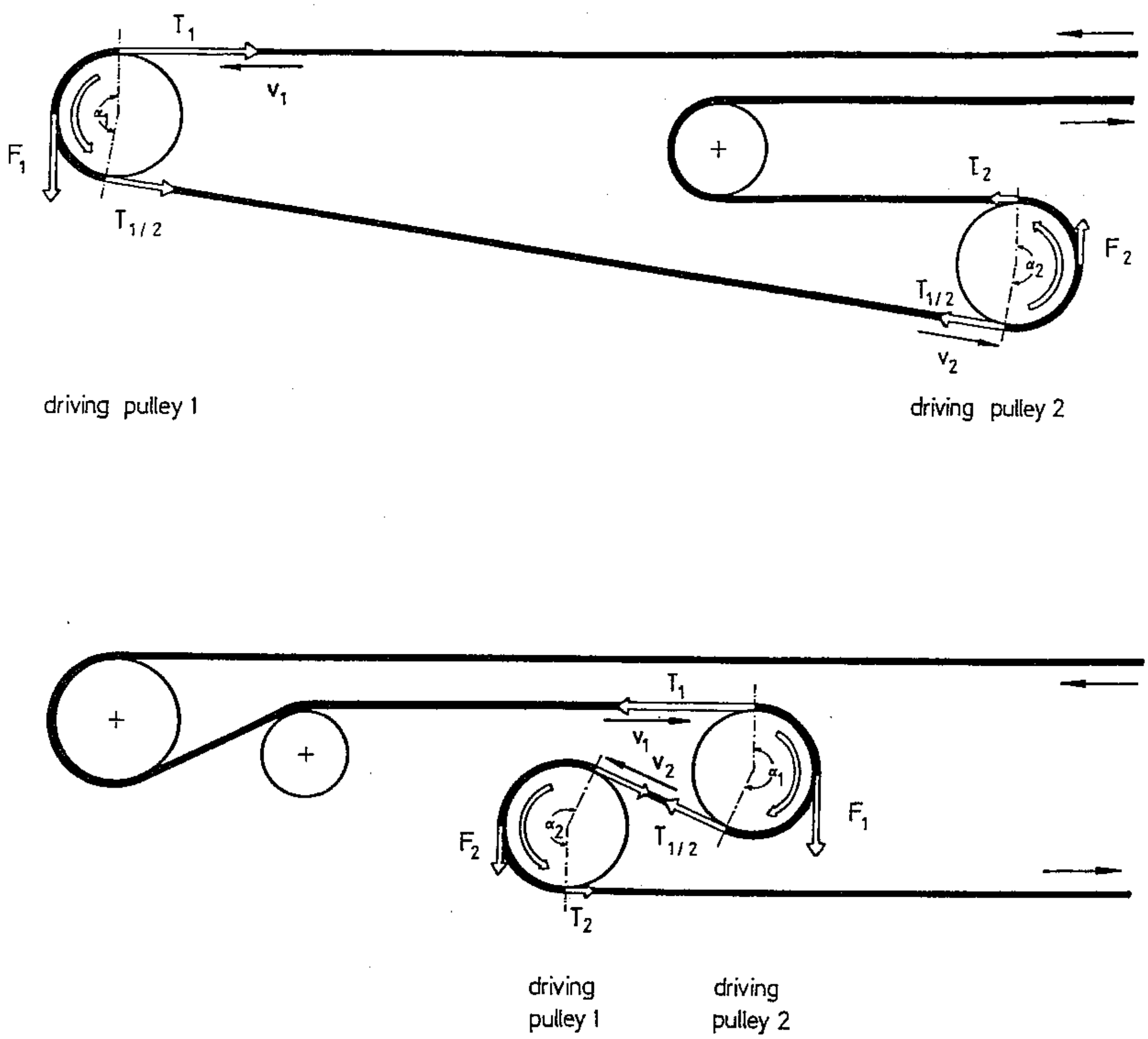


Figure 1 Belt tensile forces T_1, T_2, peripheral forces F_1, F_2 as well as belt speeds v_1, v_2 at the entry side of the pulleys 1, 2 in the case of two typical multimotor two-pulley drives (α_1, α_2: angles of wrap)

drives and between the effective pulley diameters, as well as the elasticity of the conveyor belt. As a consequence, this can lead to the overloading of some of the motors, while at the same time the rated power of other drives is not fully utilized.

Measurements carried out on drives in operation have shown that under certain conditions the existing load distributions may deviate considerably from those that are expected from the distribution of installed powers. In the following, the criteria for the resulting load distributions and equations for determining them are analyzed in detail.

2.1 Mathematical Relationships and Preconditions for Calculating Load Distributions

In order to simplify the procedure, it is necessary to make the following assumptions when deriving equations for determining the load distribution of multimotor two-pulley drives in belt conveyor systems during their steady-state operation:

- linear-elastic behaviour of the conveyor belts in the region of their operational load,
- installation of drive units of exactly the same size and with almost linear torque-speed-characteristics in the region of their operational load,
- transmissibility of the peripheral forces of the pulleys due to the mathematical interrelationships describing the load distributions of the drives.

When describing forces and motions of a driving two-pulley arrangement mathematically, the following parameters are used:

B belt width

E_G modulus of elasticity per unit width of a conveyor belt

F total resistance two motion of a belt conveyor system

F_1, F_2 peripheral forces of pulley 1, 2 (see also figure 1)

F_N rated (nominal) peripheral force of one driving pulley due to the total rated torque of its drives (related to its effective radius r, taking into account the transmission ratio and efficiency of the gearboxes)

k_N rated (nominal) rupture force of the belt per unit width

q quotient or ratio between parameters referring to pulley 1 and those referring to pulley 2

r effective radius of a driving pulley (see figure 3 in Section 2.3)
($\neq r_0$: geometric radius of a pulley including its lagging)

R reserve factor
(ratio of the total of rated torques of all drive units to the total of required torques of the drives in their steady operating conditions)

s slip of all the drives of a pulley according to their torque-speed-characteristics
(drives with slipping couplings: total slip of motors and couplings)

T belt tensile force (see also figure 1)

v belt speed at the entry side of a driving pulley (see also figure 1)

ε belt strain corresponding with the elongation of the belt due to its force T

ε_{r1} reduction of belt strain due to the peripheral force F_1

Subscripts:

1,2 parameter of the driving pulleys 1, 2 and their connected drives

M q_M: ratio of the rated drive torques at pulley 1 and 2

N rated (nominal) load of the drives

r q_r: ratio of the radii r_1 and r_2

s q_s: ratio of the slips s_1 and s_2

Relationships between belt speeds, belt strains and belt tensile forces:

$$\frac{v_1}{v_2} = \frac{1 + \varepsilon_1}{1 + \varepsilon_2} \approx 1 + \varepsilon_1 - \varepsilon_2 \tag{1}$$

$$\varepsilon = \frac{T}{E_G \cdot B} \ll 1 \tag{2}$$

$$\varepsilon_{r1} = \varepsilon_1 - \varepsilon_2 = \frac{F_1}{E_G \cdot B} \tag{3}$$

The value of the parameter E_G can be estimated with the help of the following equation:

$$E_G = c_E \cdot k_N \tag{4}$$

where:

c_E coeffient of elasticity

$c_{E/St}$: between 60 and 80 in the case of steel cable belts

$c_{E/EP}$: between 10 and 20 in the case of belts with polyester/polyamide plies

Relationships between peripheral forces, belt speeds and slips of the drives

$$\frac{v_1}{v_2} = q_r \cdot \frac{1 - s_1}{1 - s_2} \tag{5}$$

$$F_1 = F_{1N} \cdot \frac{s_1}{s_{1N}} \tag{6}$$

$$F_2 = F_{2N} \cdot \frac{s_2}{s_{2N}} \tag{7}$$

$$F = F_1 + F_2 \tag{8}$$

The given relationships provide the following equations for the ratio q_F of the constrained peripheral forces F_1 and F_2 and for these forces themselves, if the reduction of belt strain ε_{r1N}, due to the peripheral force F_{1N} at pulley 1, is introduced:

$$\varepsilon_{r1N} = \frac{F_{1N}}{E_G \cdot B} \tag{9}$$

$$q_F = \frac{F \cdot r_2 \cdot \left[1 + \frac{F_1}{F_{1N}} \cdot \varepsilon_{r1N}\right] \cdot \frac{q_M}{q_s \cdot q_r} + r_1 \cdot \frac{F_{1N}}{s_{1N}} \cdot \frac{q_r - 1}{q_r}}{F \cdot r_1 \cdot \left[1 + \frac{1}{q_r} \cdot \frac{\varepsilon_{r1N}}{s_{1N}}\right] - r_1 \cdot \frac{F_{1N}}{s_{1N}} \cdot \frac{q_r - 1}{q_r}} \tag{10}$$

$$F_1 = \frac{F \cdot \left[1 + \frac{F_1}{F_{1N}} \cdot \varepsilon_{r1N}\right] \cdot \frac{q_M}{q_s \cdot q_r^2} + \frac{F_{1N}}{s_{1N}} \cdot \frac{q_r - 1}{q_r}}{1 + \left[1 + \frac{F_1}{F_{1N}} \cdot \varepsilon_{r1N}\right] \cdot \frac{q_M}{q_s \cdot q_r^2} + \frac{1}{q_r} \cdot \frac{\varepsilon_{r1N}}{s_{1N}}} \tag{11}$$

$$F_2 = \frac{F \cdot \left[1 + \frac{1}{q_r} \cdot \frac{\varepsilon_{r1N}}{s_{1N}}\right] - \frac{F_{1N}}{s_{1N}} \cdot \frac{q_r - 1}{q_r}}{1 + \left[1 + \frac{F_1}{F_{1N}} \cdot \varepsilon_{r1N}\right] \cdot \frac{q_M}{q_s \cdot q_r^2} + \frac{1}{q_r} \cdot \frac{\varepsilon_{r1N}}{s_{1n}}} \tag{12}$$

Equations (10), (11), (12) show that the peripheral forces F_1 and F_2 of a two-pulley drive unit and their distribution ratio are dependent in a very complex way on the parameters of belt, drives and pulleys. In the following these relationships are thus to be examined in certain combinations of the influencing parameters. In order to simplify the calculation, but without any significant loss of accuracy, the expression $(F_1/F_{1N}) \cdot \varepsilon_{r1N}$ is neglected with regard to its value less than 1 %.

2.2 Influences of Belt Elasticity and Drive Characteristics on the Load Distribution in the Case of Pulleys with Identical Radii ($r_1 = r_2 = r$ and thus $q_r = 1$)

Under the conditions introduced above, the parameter q_F and the peripheral forces F_1 and F_2 are only dependent on the parameters q_M, q_s and the quotient ε_{r1N}/s_{1N}. Figure 2 contains the relationship between the forces F_1, F_2 and this expression, as well as the slip ratio q_s as an additional parameter. This relationship is given separately for the torque ratios $q_M = 1$ and $q_M = 2$. The peripheral forces F_1 and F_2 are related to those that would exist in the case of ideal load distribution, which are defined as follows:

$$(F_1)_{id} = \frac{q_M}{1 + q_M} \cdot F \tag{13}$$

$$(F_2)_{id} = \frac{1}{1 + q_M} \cdot F \tag{14}$$

It can be seen that an increasing load of the drives at pulley 2 and a corresponding reduction of load of those at pulley 1 occurs with increasing values of the slip ratio q_s and with increasing values of the quotient ε_{r1N}/s_{1N}, i.e. with a

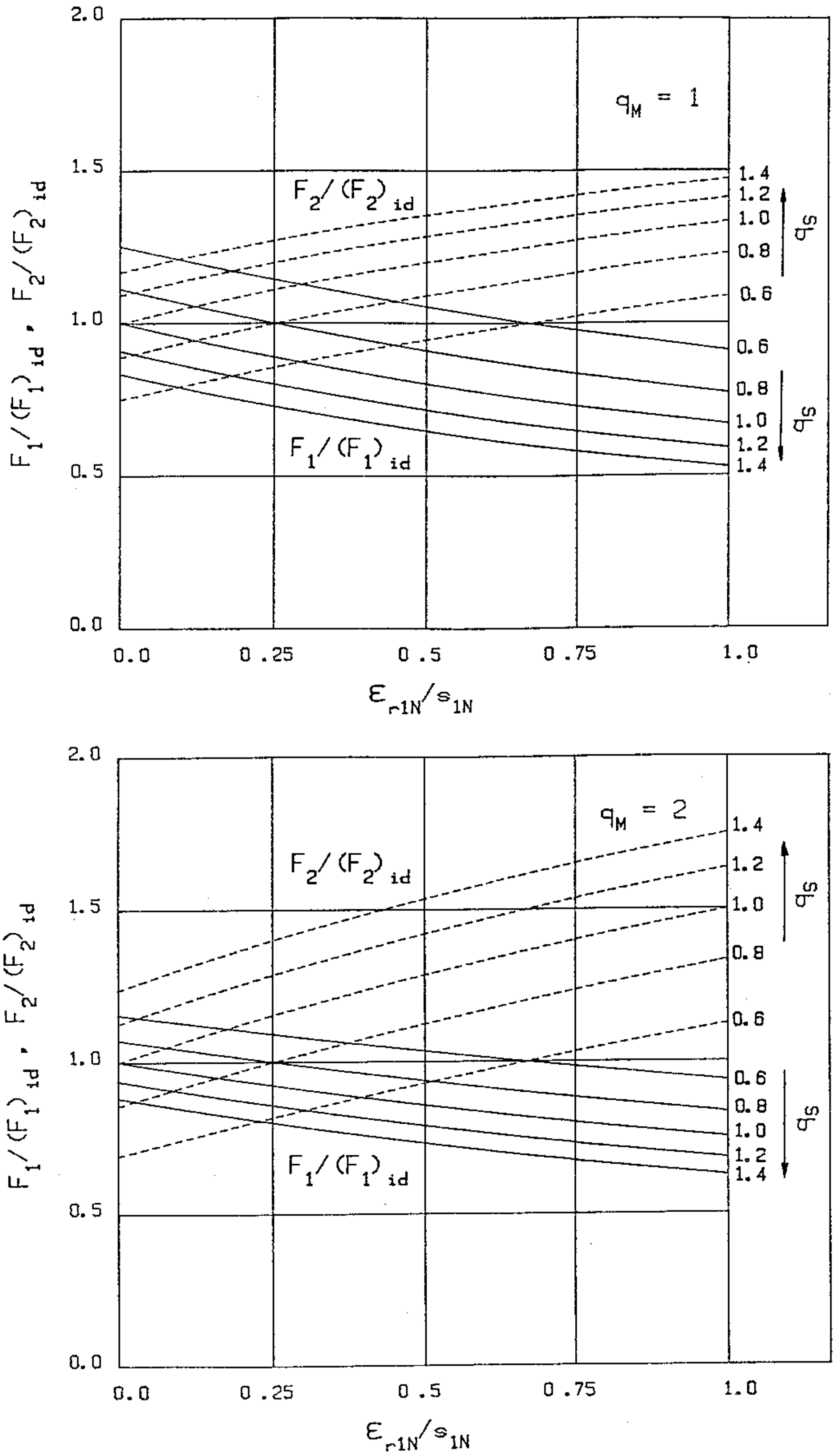


Figure 2 Dependence of the peripheral forces F_1 and F_2 on the reduction of belt strain ε_{r1N} at drive pulley 1 related to the rated slip of its drives, as well as on the slip ratio q_s in the case of multimotor two-pulley drives. Ratio of the rated drive torques: q_M = 1 (top) and q_M = 2 (bottom)

steeper characteristic of the drives and with belts of higher operational strains. In contrast, a steeper characteristic of the drives at pulley 1 compared with those at pulley 2 ($q_s < 1$) is the reason for an additional load on the drives at pulley 1. Thus the influence of the expression ε_{r1N}/s_{1N} can be compensated over a wide range by suitably modified characteristics.

With regard to the influence of the ratio q_M of the rated torques, it can be seen that, with the value $q_M = 1$, the additional load on the drives at pulley 1 corresponds to an equally large reduction on those at pulley 2. In contrast, with the value $q_M = 2$, the parameter q_s and the quotient ε_{r1N}/s_{1N} have a greater effect on the additional load of the single drive at pulley 2 than on the load reduction of the two drives at pulley 1. This means, assuming installations with the torque ratio 2:1 between pulleys 1 and 2, the drives at pulley 2 have a greater risk of being overloaded than in the case of the ratio 1:1.

When interpreting figure 2 with regard to the different operational strains of steel cable belts and fabric belts, it can be stated that, because of their higher operational strains (lower values of the modulus of elasticity), the latter cause higher deviations from the rated load of the drives than steel cable belts. Thus, for the purposes of practical application, great importance must be attached, in the case of slip-ring motors, to the correct design of their continuous slip resistors in the rotor circuit, and in the case of squirrel-cage motors with fluid couplings as starting aids, to the filling of the couplings themselves.

On the other hand, in the case of steel cable belts (higher values of the modulus of elasticity and thus lower values of the parameter ε_{r1N}) and belt conveyor drives with a flatter gradient of their characteristics (higher values of the slip s_{1N}), and thus smaller values of the quotient ε_{r1N}/s_{1N}, the influence of the elasticity of the belts and the gradients of the drive characteristics with regard to the load distribution is distinctly smaller than the influence of different gradients of the characteristics (parameter $q_s \neq 1$). Thus, for example in the hypothetical case $\varepsilon_{r1N}/s_{1N} = 0$, a tolerance of the rated slip of only ±10 % leads to the slip ratio $q_s = 1.1/0.9 = 1.22$ in the most unfavourable case. This ratio corresponds to an additional load of the drives at pulley 2 amounting to 14 %, provided that the torque ratio $q_M = 2$ and that the total rated power of the drive units is needed in order to overcome the resistances to motion of the belt conveyor system.

In summary, it can be stated that when selecting and designing belt conveyor drives, attention has to be paid to the tolerances and the gradient of their torque-speed-characteristics as well as to the influence of the types of conveyor belt with regard to the load distribution. In this way, mismatching of torque and power of the drives can definitely be avoided.

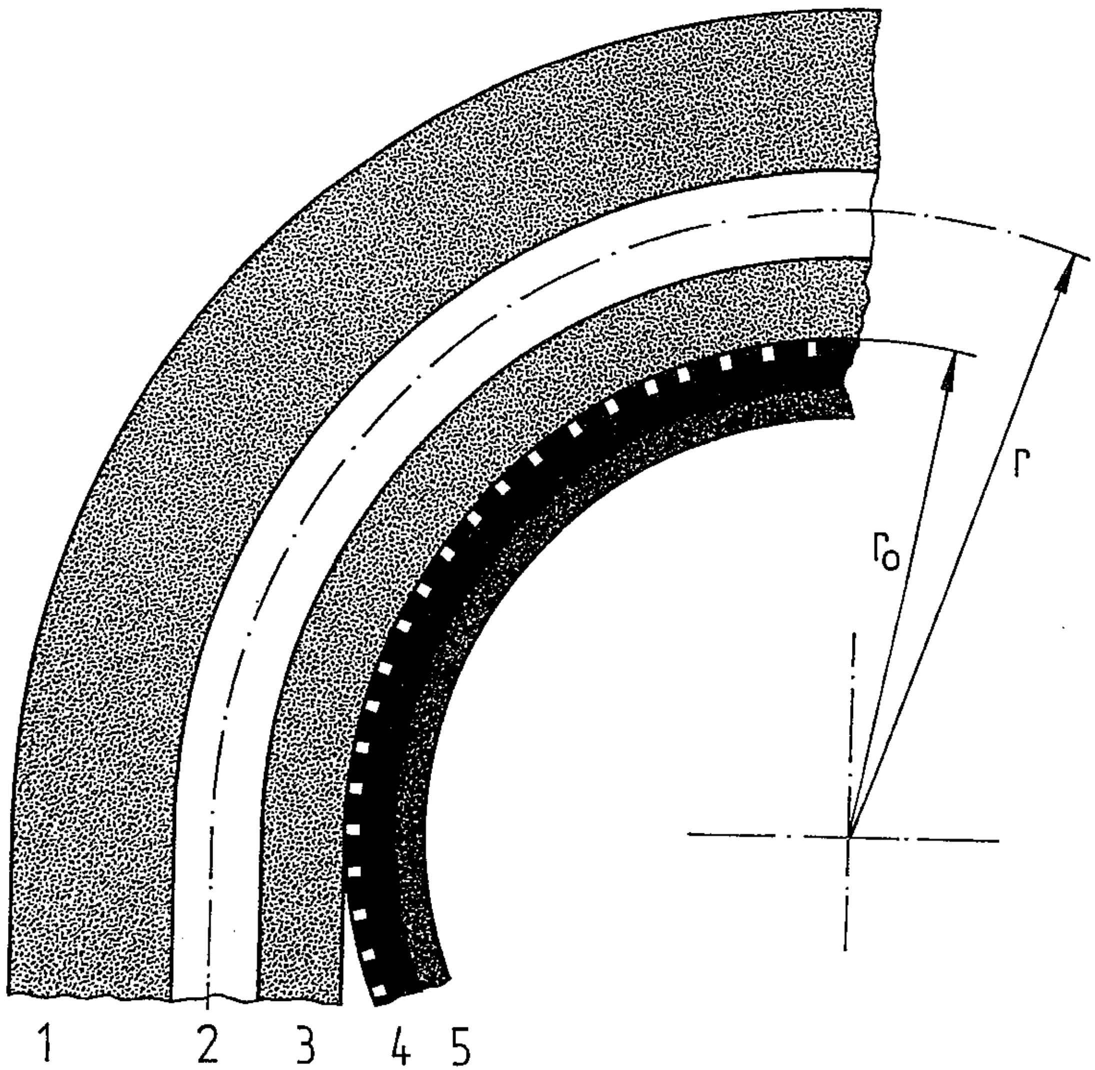
2.3 Influences of Belt Elasticity and Differences between Pulley Diameters on the Load Distribution in the Case of Drives with Identical Characteristics ($s_{N1} = s_{N2} = s_N$ and thus $q_s = 1$)

Before analyzing the influence of differences between effective pulley diameters on the load distribution in two-pulley drives, the meaning of the term "effective radius" of a driving pulley is to be explained. According to figure 3, this parameter is the radius r of the "quasi-neutral" fibre in the tensile member of the belt. This is the fibre that is not additionally stretched when the belt runs round a pulley. It must be noted that the quasi-neutral fibre is not always identical with the centre line of the belt.

When using conveyor belt types with a thicker cover plate at the top and a less thick plate at the bottom, differences can arise in the effective radii of driving pulleys, depending on the design of the multi-pulley arrangement. If the transmission of drive forces into the tensile member takes place via the thicker top cover plate of the belt, the effective radius is greater than in the case of power transmission by a less thick bottom plate. This demonstrates the influence of the pulley arrangement on the values of the effective radii. This is to be discussed in detail with the help of figure 1. The pulley arrangement in the upper part of this figure is characterized by the fact that the peripheral forces F_1, F_2 are transmitted into the belt, in both cases by its bottom cover plate. In contrast, as demonstrated by the other arrangement in figure 1, the forces F_1, F_2 are transmitted by different cover plates: force F_1 by the top and force F_2 by the bottom plate.

The effective radius of a pulley is subject to changes during the operation of belt conveyor systems. This is due to the wear of the belt cover plates, but especially to the wear of and dirt accumulation on the lagging of the pulley. Under unfavourable conditions of operation and maintenance, this can lead to considerable differences in the values of the effective pulley radii.

After these general remarks on the effective pulley radius, the influence of differences between pulley diameters and, additionally, of belt elasticity on the load



Legend

Belt:

1 cover plate (top)

2 tension member with quasi-neutral axis

3 cover plate (bottom)

Driving pulley

4 lagging

5 shell

r_o geometric radius

r effective radius

Figure 3 Definition of the effective radius of a driving pulley distribution is to be considered in greater detail. Here, it is assumed that the gradients of the torque-speed-characteristics of the drives have the same values, i.e. $s_{N1} = s_{N2} = s_N$ and thus $q_s = 1$.

Under the conditions introduced above, the parameter q_F and the peripheral forces F_1 and F_2 are dependent on the parameter q_M as well on the expressions $(q_r - 1)\cdot R/s_N$ and ε_{r1N}/s_N. In the first expression the reserve factor R is a newly-introduced parameter:

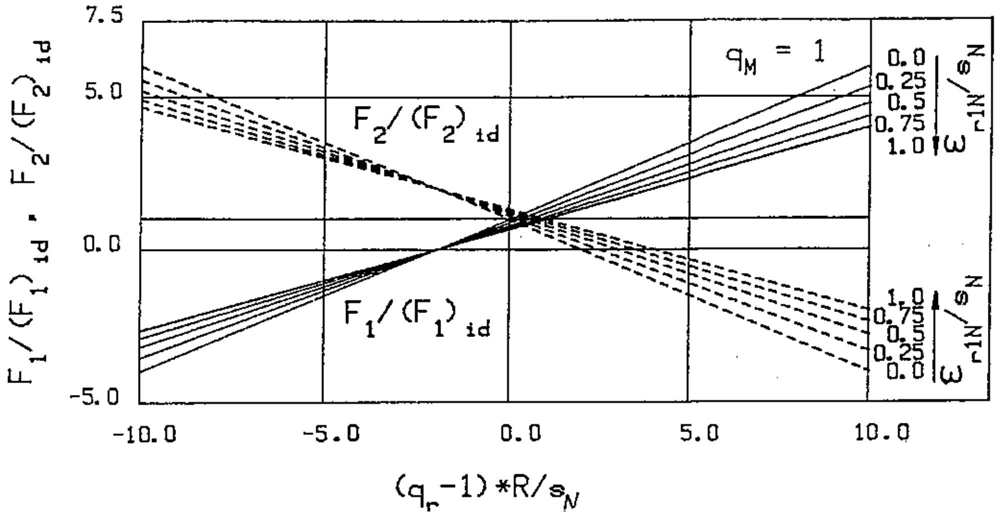
$$R = \frac{F_{1N} + F_{2N}}{F} \qquad (15)$$

In contrast to the relationships shown in figure 2, the reserve factor appears here as a parameter that considerably influences the load distribution of the drives.

In figures 4 and 5, the peripheral forces F_1 and F_2 are shown dependent on the expression $(q_r - 1)\cdot R/s_N$ and the reduction of belt strain ε_{r1N} related to the rated motor slip s_N. While figure 4 demonstrates the load distribution of the drives in the range between partial and rated load, figure 5 demonstrates the load distribution in the rated load region.

Figure 4 provides information on the fact that low utilization of the rated motor power (R in the region of 10), as well as relative differences between the effective radii of pulleys 1 and 2 (expression $q_r - 1$) in the region of the rated slip s_N, cause extreme deviations from the ideal load distribution, as defined by equations (13) and (14). It must be noted that high values of the expression q_r-1 are the reason for the fact that the drives at the smaller of the two pulleys are operated with oversynchronous speed, and thus regeneratively. In these cases, power transmitted into the system by the drives at the pulley with the larger radius is returned to the network. These phenomena are not purely theoretical. The reason why they are not always recognized in the day-to-day operation of belt conveyor systems is connected with the fact that the ammeters used for monitoring are unable to indicate the return power transmission to the network.

With regard to the overloading of drives operated in the region of their rated power, information is provided by figure 5 which represents an enlarged section of figure 4 for values of the parameter R approximately equalling 1. Both figures reveal for the ratio of rated torque $q_M = 1$ that an additional load on the drives



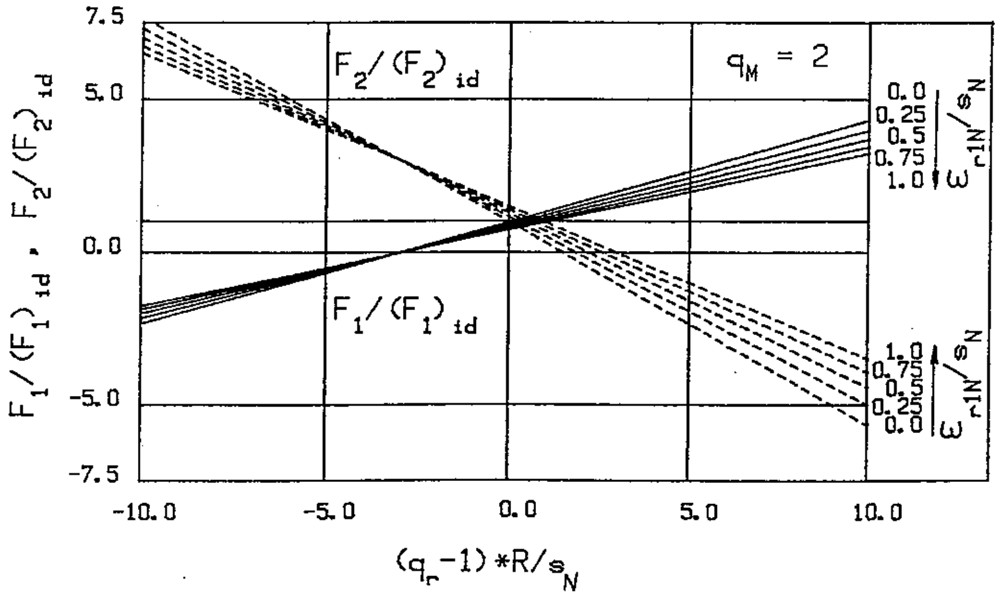
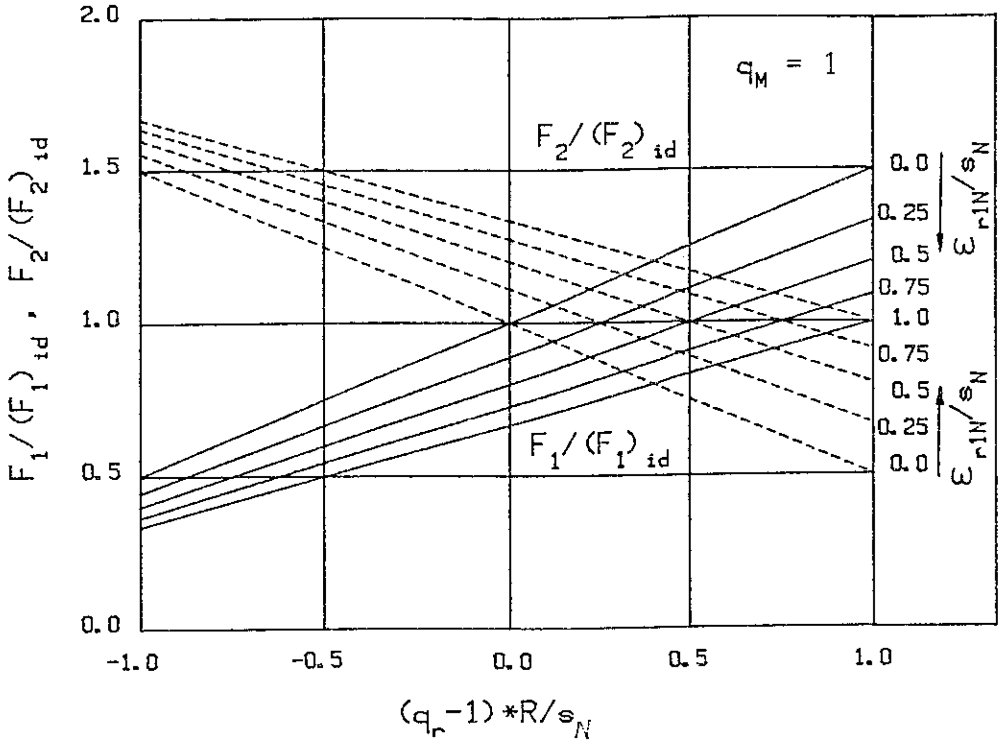


Figure 4 Dependence of the peripheral forces F_1 and F_2 on the ratio q_r of the pulley radii, reserve factor R and rated slip s_N of the drives, as well as the reduction of belt strain e_{r1N} at pulley 1 in the whole of the load region of the drives ($0.1 \leq R \leq 1$) in the case of multimotor two-pulley drives. Ratio of the rated drive torques:

$q_M = 1$ (top) and $q_M = 2$ (bottom)



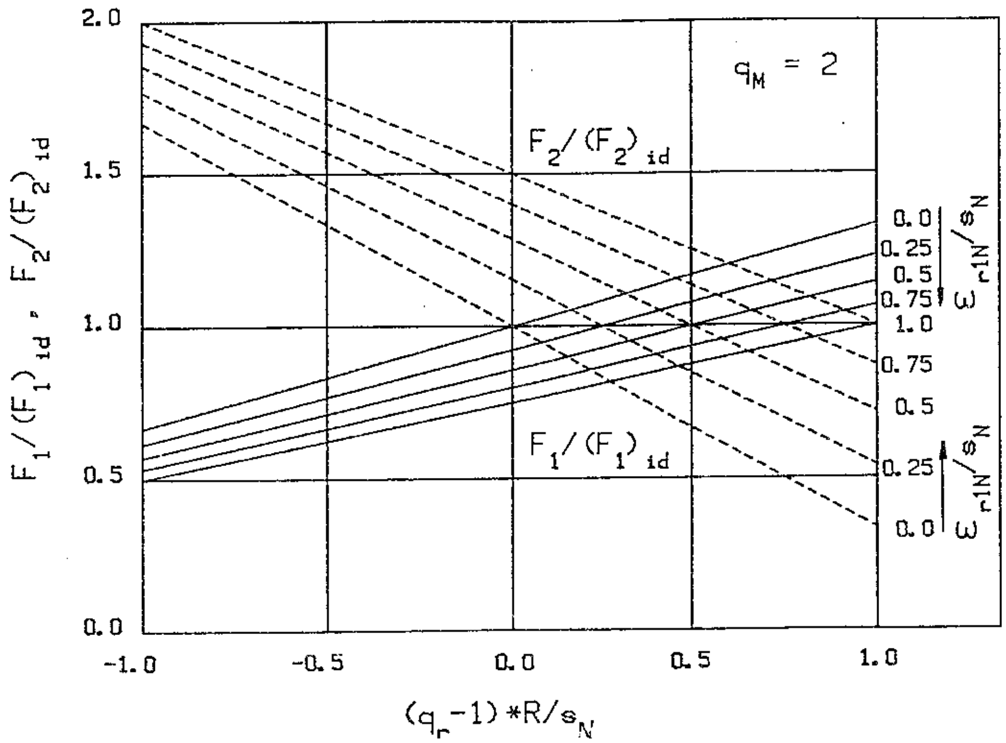


Figure 5 Dependence of the peripheral forces F_1 and F_2 on the ratio q_r of the pulley radii, reserve factor R and rated slip s_N of the drives, as well as the reduction of belt strain e_{r1N} at pulley 1 in the region of the rated load of the drives ($0.8 \le R \le 1$) in the case of multimotor two-pulley drives. Ratio of the rated drive torques:

$q_M = 1$ (top) and $q_M = 2$ (bottom)

at one pulley corresponds to an equally large load reduction on the other drives. In contrast, for the ratio q_M = 2, the load difference - and thus the danger of overload - is distinctly greater in the case of the single drive at pulley 2 when compared with the two drives at pulley 1. The following is an example of this: assuming a relative difference of the effective pulley diameters of only 1 % (q_r = 0.99: the effective diameter of pulley 2 is larger than that of pulley 1), and the rather small rated slip s_N = 2 %, the single drive at pulley 2 is overloaded by 34 %, when the total rated power of all the drives is needed to overcome the resistances to motion of the conveyor (R = 1), and when the reduction of belt strain ε_{r1N} is neglected. The latter alters the load distribution to the disadvantage of the single drive. - In contrast, assuming a doubling of the rated motor slip to s_N = 4 %, the overload of the single drive at pulley 2 would be reduced to approximately 17 %, and thus halved, provided that the other parameters are unaltered.

In summary, it must be stated in conclusion that even small differences of the effective pulley diameters exert a large influence on the load distribution in two-pulley drive units, especially when the ratio between the rated torques of the drives at pulleys 1 and 2 is 2:1 (q_M = 2). This is why the uneven wear of and dirt accumulation on the pulley laggings greatly influence the load distribution of multi-pulley drive units. This large influence of even small differences in the effective pulley diameters can be counteracted by the installation of drives with higher values of rated slip. In the case of slip-ring (wound rotor) motors, this is made possible by suitable design of the continuous slip resistors, while in the case of squirrel-cage motors with fluid couplings the latter must have the appropriate fillings.

2.4 Summary

Independent of the transmissibility of the installed drive powers at each of the pulleys, a more or less different utilization of the drives occurs in the case of multiple arrangements of driving pulleys. They are due to manufacturing tolerances of drive components, conditions during operation, as well as to the mechanical coupling of the motors - connected with different pulleys - by the belt. Mainly those influences are important from which different effective diameters of the driving pulleys and different torque-speed-characteristics result. As a consequence, this can lead to the overloading of some of the motors, while at the same time the rated power of other drives is not fully utilized.

In the case of slip-ring (wound rotor) motors, the possibility exists of influencing the gradient of their characteristics for the steady-state operation by appropriate continuous slip resistors in such a way that torque and power matching takes place. In the case of squirrel-cage motors with fluid couplings as a starting aid, it is possible to influence the totally effective drive characteristics by matching the filling of the couplings.

The matching of the torque-speed-characteristics of conventional belt conveyor drives as described can only be realized for a certain load condition. On the other hand, this means that, under different load conditions, unbalanced operation of the drives cannot be prevented. Due to the restricted possibilites for matching the characteristics of belt conveyor drives, it is necessary to prevent overload by appropriate design of the system, especially of the drive system. The basis is given for calculating the load distribution in driven two-pulley arrangements, and thus for determining the utilization of the installed rated power of the drives.

3. Thermal Stress of Drive Units during the Start-Up of Belt Conveyor Systems

The forces introduced into belt conveyor systems by their drive and brake units during non-steady operating conditions, i.e. during start-up and shut-down procedures, are of exceptional importance for the reliable design of these systems, both with regard to the investment costs and the working life of its belt. For this reason a smooth start with a given limitation of the acceleration, as independent of the load conditions of the system as possible, is always called for above a certain size of the belt conveyor system.

However, the limitation of the belt acceleration during start-up is the reason for long acceleration processes, especially in systems with high belt velocities during their steady operating conditions. The corresponding high values of start-up times cause high thermal stresses of the drives. In the past, especially in connection with the ill considered application of cheap electronic soft-start systems, a number of breakdowns and cases of damage have occurred that were due to the fact that these aspects were not taken into account to a satisfactory extent. In the following, a short introduction will thus be given into the relationships between the limitation of the drive forces during start-up and the resulting reduced start-up accelerations, as well as information on calculating the amount of energy loss and the corresponding thermal stress due to the increased slip of the drive units during start-up procedures.

3.1 Fundamental Equations to Calculate Belt Accelerations during Quasi-steady Operating Conditions of Belt Conveyor Systems

During the start-up procedures of belt conveyor systems, the magnitude and distribution of the forces generated by the drive systems on the one hand and the breakaway and motional resistances on the other hand, cause additional dynamic belt tensile forces as a result of the accelerations of the belt occuring in these transient operating conditions. The accelerations occuring can be considered being the same in all sections of the upper and lower strands, and thus the system behaviour as quasi-steady, only when the rate of rise of the pulleys' peripheral forces in these operating conditions is limited. If this prerequisite is not satisfied, precise calculations of the transient local belt tensile forces and belt velocities require a relatively large amount of effort. However, this does not apply for the approximate calculation of the start-up time, which is generally completely satisfactory for practical requirements, and which assumes a mean acceleration of all moving parts of the belt conveyor system, and thus a quasi-steady operating behaviour. Under these conditions, the additional dynamic forces can be regarded as mass forces, and the following fundamental equations can be deduced under the assumption - also made in DIN 22101 - that the resistances to motion during the transient operating conditions are approximately equal to those occuring during the corresponding steady-state operation of the sytem:

$$\frac{F_A}{F} = p_A = 1 + \frac{a_A}{a_n} \qquad (16)$$

$$a_A = (p_A - 1) \cdot a_n \qquad (17)$$

a_A	belt acceleration
a_n	"natural" acceleration of the belt due to the force F and all moved masses in the upper and lower strands
F	total of the resistances to motion
F_A	sum of the peripheral forces of all driving pulleys during start-up
p_a	force ratio with regard to the value of the force F_A

Provided that the belt conveyors, which are to be investigated with respect to their non-steady operating conditions, have a steady inclination and a continuous load distribution over their entire length, but no special resistances due to idler tilting, discharge plows and friction between load and skirtboards outside the loading points, the following defining equation for the "natural" acceleration a_n can be deduced:

$$a_n = \beta \cdot a_{no} \qquad (18)$$

with

$$a_{no} = c_m \cdot C \cdot f \cdot g \qquad (19)$$

$$\beta = 1 + \frac{\sin\delta}{C \cdot f} \cdot \eta \qquad (20)$$

for $\cos\delta \approx 1$.

The following parameters apply:

c_m mass ratio
(ratio between the masses which are in relation to the frictional resistances to motion and the total of translatorily and rotatorily moved masses (the latter reduced to their periphery) without directly driven components of the belt conveyor system

$C \cdot f$ parameter product (according to the standard DIN 22101) which characterizes the main and secondary resistances of belt conveyors in the upper and lower strands

g gravitational acceleration ($g = 9.81\ m/s^2$)

β acceleration ratio

δ average slope angle of a belt conveyor

η ratio between the masses due to load and the total mass in the upper and lower strands. (The total mass is the one in relation to the frictional resistances to motion)

Of those parameters which determine the value of the acceleration a_{no} the product $C \cdot f$ is the most important. Belt conveyor systems, which have centre distances of more than 500 m and which are designed and operated in accordance with the relevant standards, are generally characterized by values of the product $C \cdot f$ in the range $0.015 \leq C \cdot f \leq 0.030$. The parameter c_m ist generally slightly less than 1.

The acceleration ratio β is to be regarded as a factor by which the acceleration of inclined belt conveyors ($\delta \neq 0$) in their non-steady operating conditions is increased in comparison with horizontal belt conveyors ($\delta = 0$) with the same force ratio p_A. The ratio β is essentially determined by the slope angle δ and the mass ratio η , and it varies over a wide range:

The quotient $\eta/(C \cdot f)$ is usually in the range $0 \leq \eta/(C \cdot f) \leq 50$. The lower limiting value is characteristic of unloaded belt conveyors, and the upper one of loaded belt conveyors with low frictional resistances to motion and high values of the mass ratio η. Usual slope angles δ between $-18°$ and $+18°$ yield acceleration ratios in the range $-14.5 \leq \beta \leq 16.5$.

Completely different results will be found when calculating the parameter β for slightly inclined belt conveyors. Analyzing for exsample a downhill conveyor with a slope angle δ between $-1°$ and $-1.5°$, additionally characterized by $\sin \delta = -C \cdot f$ and the mass ratio $\eta = 0.65$, the equation (20) yields $\beta = 0.35$. Thus this belt conveyor is accelerated approximately with only one third of those values which are typical for a horizontal conveyor with the same force ratio p_A.

With the formulae deduced, a relationship can be derived between the acceleration a_A during start-up and the force ratio p_A, which characterizes the limitation of the peripheral forces of the pulleys, and thus the maximum belt tensile forces. This relationship is co-determined only by very few typical and generally known parameters of the belt conveyor system. It permits the calculation of the start-up time t_A even in the case of time- and velocity-dependence of the ratio p_A in the following way:

$$t_A = \frac{1}{a_n} \cdot \int_0^v \frac{dv}{p_A - 1} \tag{21}$$

In practical application it is frequently the case that the ratio p_A is permitted to be approximated by the mean value p_{Am}. Equation (21) thus can be simplified as follows:

$$t_A = \frac{v}{a_n \cdot (p_{Am} - 1)} \tag{22}$$

where:

t_A start-up time

v belt speed during steady operating conditions

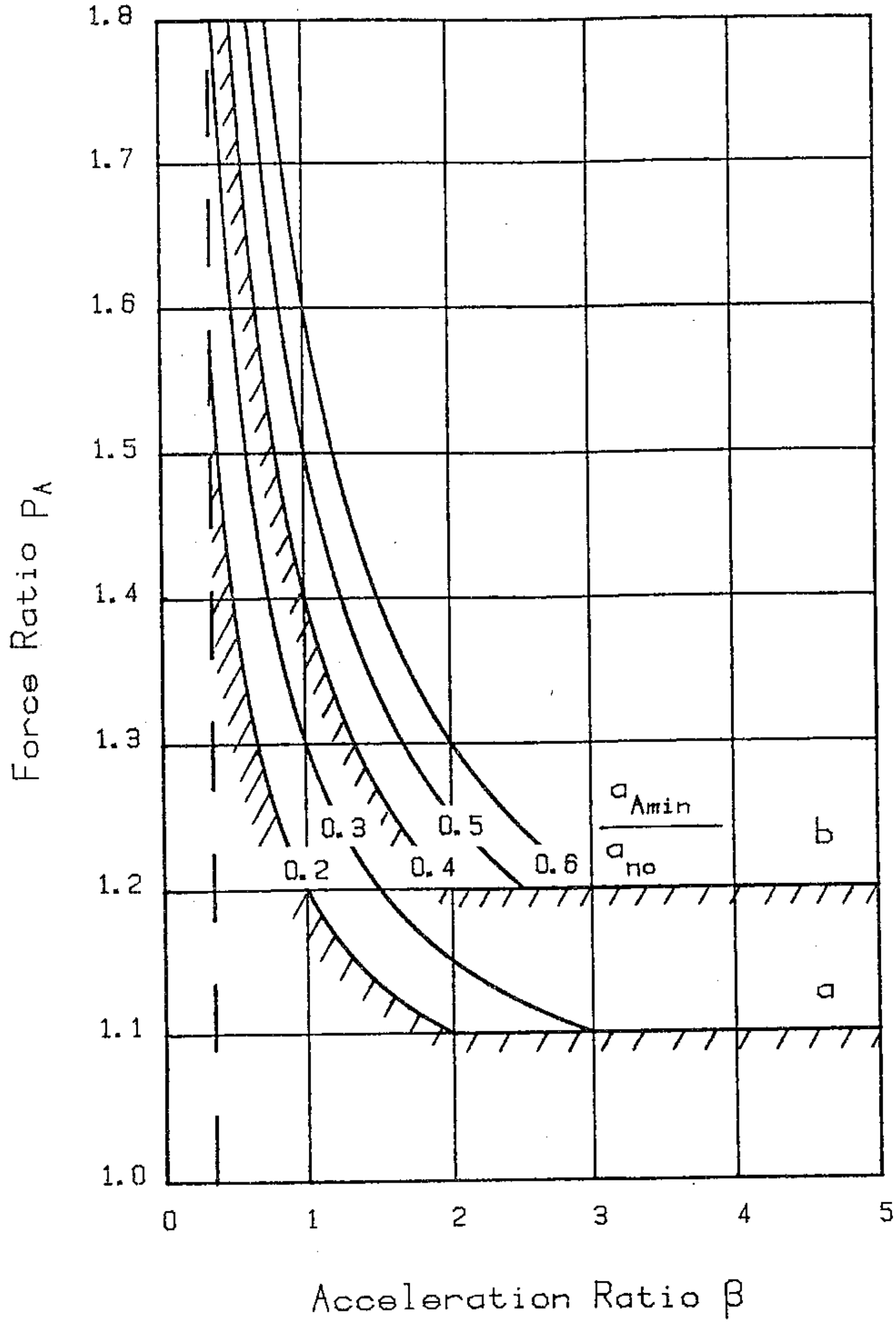


Figure 6 Diagram for the optimization of the force ratio p_A and the starting acceleration a_A for horizontal, slightly inclined and steep uphill belt conveyor systems (lower limiting value β_{min} = 0.35 according to equation (20) for sin δ = −C·f and η = 0.65)

It is frequently the case that constancy of the force ratio p_A is assumed in the design of belt conveyor systems. Figure 6 can be used as an aid for the optimum choice of this parameter. The figure was first published in this form for BELTCON 4. With regard to its use, it must be noted that the selection of force ratios p_A less than the minimum values p_{Amin}, defined by the limiting curves a and b, should only be allowed in justifiable, exceptional cases.

The higher limiting curve b, determined by the minimum starting acceleration $a_{Amin} = 0,4 \cdot a_{no}$ and the minimum force ratio $p_{Amin} = 1,2$ applies to most of the belt conveyors with uneven loading and to those with uncontrolled and non-regulated drive units during the starting phases. - On the other hand, the lower limiting curve a, determined by the minimum starting acceleration $a_{Amin} = 0,2 \cdot a_{no}$ and the minimum force ratio $p_{Amin} = 1.1$, only applies to belt conveyors with controlled loading and such drive units whose torque during starting is controlled or regulated depending on the load of the conveyor, and whose thermal stressability is sufficient.

The pracitical design of drive units by optimizing the force ratio p_A with regard to minimum belt stress and permissible thermal stress of the driving units has, additionally, to take into consideration that the drive units' torques during starting generally shows peaks and valleys. In these cases, optimally designed units should under all conditions provide force ratios p_A, which remain above the discussed limiting curves during the whole of the starting phases.

On the basis of the considerations introduced above with regard to the calculation and selection of the force ratio p_A, the thermal stress of the drives of belt conveyor systems during start-up is to be analyzed in the following with respect to this parameter.

3.2 Basic Equations for Calculating Thermal Stress of Belt Conveyor Drives

A distinction must be made between the conventional A.C. drives for belt conveyor systems with and without slipping couplings as starting aids with regard to the thermal stress of their components during start-up. A distinct motor heating effect occurs both in the stator- and rotor-components during the start-up of "directly" driving three phase A.C. motors, i.e. those without slipping couplings. The heating up of components in the rotor-circuits frequently determines the design of the motors, in the case of slip-ring (wound-rotor) motors it mainly determines that of the starter resistors. The thermal stress of the components in the rotor-circuits during start-up is almost exclusively the result of the energy loss due to the increased slip in this operational phase. On the other hand, in the case of drive units with fluid or eddy current couplings, assuming their optimum design, the losses due to slip occur predominantly in the couplings themselves.

In the following, equations are initially derived to calculate the energy losses due to slip and the corresponding thermal stress in the case of directly driving A.C. motors. Subsequently, it is explained how these equations can also be applied for drive units with slipping couplings.

Independent of the method of influencing the torque-speed-characteristics of A.C. motors considered here, the following fundamental equations have to be taken into account:

Energy loss due to slip during start-up:

$$W_s = \int_0^{t_A} s \cdot P_D \; dt$$

$$= 2 \cdot f_I \cdot f_L \cdot (W_{kin})_M \qquad (23)$$

Temperature rise due to slip during start-up:
(assumption: no heat dissipation during this operational phase)

$$\Delta\vartheta = \frac{W_s}{C_H} \qquad (24)$$

where:

$$(W_{kin})_M = \frac{1}{2} \cdot \omega_o^2 \cdot I_M \qquad (25)$$

The parameters used have the following meaning:

C_H heat capacity

f_I ratio between the total of effective moments of inertia of a belt conveyor system and the total moments of inertia of all drive motors of the system. (The effective moments of inertia are related to the motor shafts under consideration of the rotating parts of the drive and brake units and the efficiency of their gear boxes)

f_L load factor as defined by equation (26)

I_M moment of inertia per motor

P_D rotating field power per motor

s slip of motor (time dependent during start-up)

t time

W_s energy loss due to slip during start-up per motor

$(W_{kin})_M$ kinetic energy per motor rotor at synchronous speed

$\Delta\vartheta$ temperature rise

ω_0 angular velocity of motor rotor at synchronous speed n_0 ($\omega_0 = 2\cdot\pi\cdot n_0/60$)

It can be seen that the energy loss due to slip during start-up is approximately proportional to the kinetic energy of the system during steady operating conditions, and thus to the square of the belt speed in this state. As a consequence of this, for example, an increase of the belt speed by a third, e.g. from 4.5 to 6 m/s, causes a doubling of the kinetic energy of the system. The parameters f_I and f_L have a multiplying influence on the energy loss due to slip, and thus on the thermal stress of the drives.

If one expresses both the motor torque M_M due to its characteristic and the load torque M_L by the rated motor torque M_{MN} with the aid of the equations

$$M_M = p_{Ao} \cdot M_{MN}$$

$$M_L = M_{MN} / R$$

then one obtains the following equation for the load factor f_L:

$$f_L = \int_{s_{sta}}^{1} \frac{p_{Ao}}{p_{Ao} - \frac{1}{R}} \cdot s \cdot ds \qquad (26)$$

where:

p_{Ao} torque ratio with regard to the torque-speed-characteristics of the motors

s_{sta} slip of the motors under load during steady operating conditions

R reserve factor (cf. explanation of the symbols in Section 2.1) (ratio of the total of rated torques of all drive units to the total of required torques of the drives in the steady operating conditions of a belt conveyor system)

The equation shows that the factor f_L may become very large when the parameter p_{AO} approaches the value 1/R. There is a risk of this, particularly when there is a distinct dip in the lower slip region of the torque-speed-characteristics of the motors and if the motors are operating under high load, i.e. R ≈ 1. It is not unusual to see that a designer has failed to take this fact into consideration. A common consequence of this is that a conveyor will start under heavy load, but on reaching between 15 to 20% of full speed will not accelerate further and thus a distinct thermal stress of the drives will occur before the system is tripped out. Provided that the torque-speed-characteristic and the load of the motors are known, the relationships introduced above permit the calculation of the energy loss due to slip and the corresponding thermal stress of the components in the rotor-circuits of directly driving asynchronous motors. While these losses arise mainly in the resistors in the case of slip-ring motors, in squirrel-cage motors without slipping couplings they lead to correspondingly high temperatures of the rotor parts.

The equations derived above for directly driving A.C. motors can also be applied to drive units with fluid couplings as starting aids. In this case, the synchronous angular velocity of the motor ω_0 has to be substituted by the angular velocity ω_1 of its coupling which corresponds to the input speed of the coupling. The slip s now means the slip of the coupling instead of that of the motor. All the parameters describing the kinetic energy and the moments of inertia now refer to the parameters of the output side of the coupling.

3.3 Practical Calculation of the Thermal Stress of Belt Conveyor Drives

Finally, the equations derived above for the thermal stress of the drives in belt conveyor systems during start-up are to be simplified here with regard to their practical application. They are to be complemented by information with the aid of which the energy losses during start-up, and thus the corresponding temperature rises of the drive components, can be estimated without too much calculating effort.

A relationship is first established between the force ratio p_A, which is generally determined by the conveying technology engineer, and the torque ratio p_{A0}, which is to be realized by the power transmission engineer:

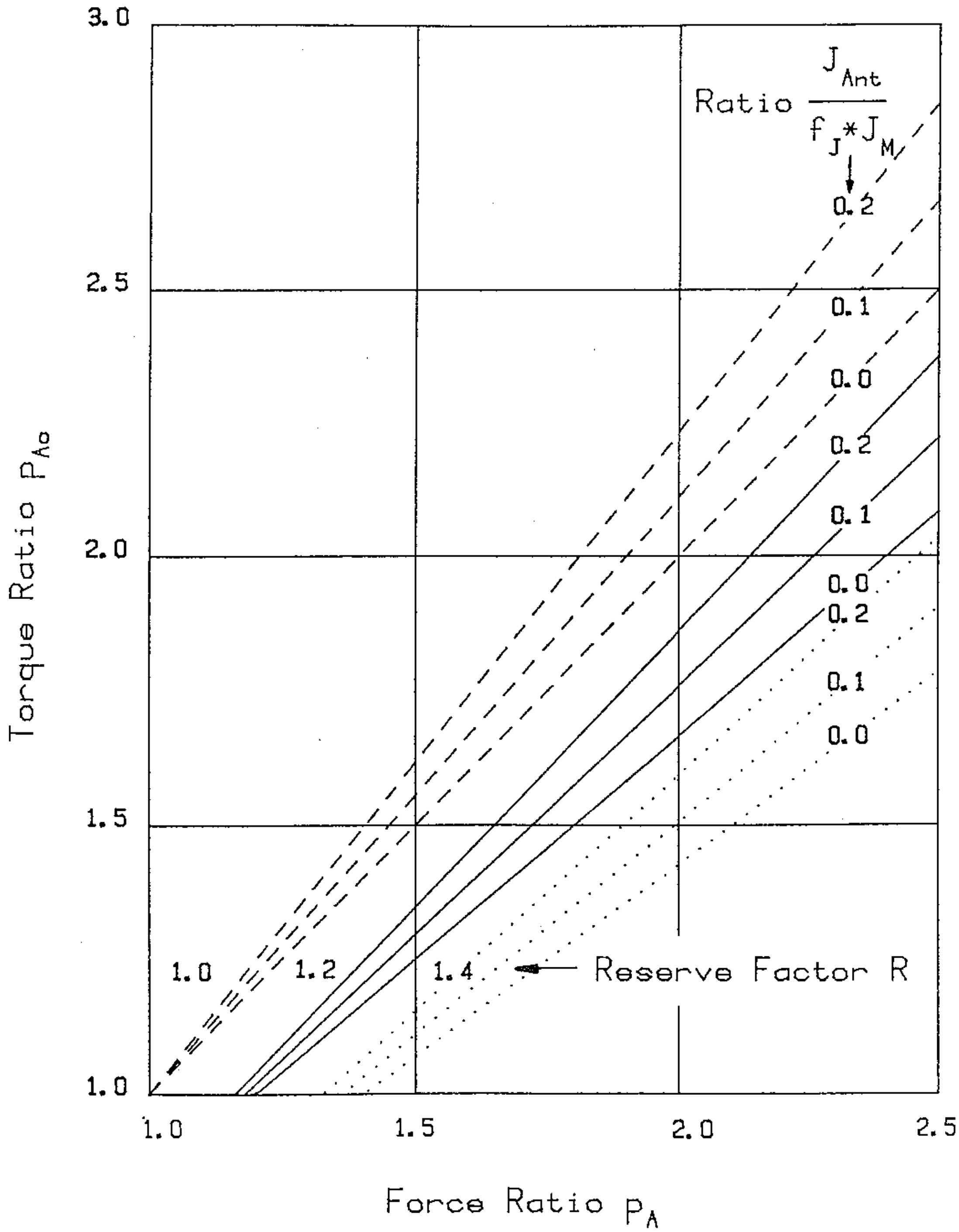


Figure 7 Relationship between the torque ratio p_{Ao}, depending on the characteristics of the drives and the force ratio p_A, representing the pulleys' peripheral forces during quasi-steady acceleration phases, with the reserve factor R and the ratio $I_{Ant}/(f_I \cdot I_M)$ as parameters of loaded belt conveyor systems

$$p_{Ao} = \frac{1}{R} \cdot \left[1 + \frac{p_A - 1}{1 - \frac{I_{Ant}}{f_I \cdot I_M}} \right] \qquad (27)$$

where:

I_{Ant} total of effective moments of inertia of all the rotating parts of the drive and brake units reduced to the shaft of their driving motors

Figure 7 demonstrates graphically the relationship between the parameters p_A and p_{Ao} for belt conveyors with mass flows in the range of their rated load. It is obvious that if $R = 1$, then for all values of the ratio $I_{Ant}/(f_I \cdot I_M) > 0$ the torque ratio p_{Ao} is always higher than the force ratio p_A. If $R > 1$, then p_{Ao} is lower than p_A. This is all the more the case, the lower the ratio $I_{Ant}/(f_I \cdot I_M)$ is. This has to be considered especially when belt conveyors are operated with partial load or unloaded, in which cases the reserve factor increases to $R > 3$ and the mass ratio to $I_{Ant}/(f_I \cdot I_M) > 0.5$. To summarize, it is essential to differentiate between the parameters p_A and p_{Ao}. Only by taking this into consideration, can unproductive discussions be avoided between the experts for the drive units and those for the belt conveyors about the optimum values of the "starting-factors" - which are frequently-used synonyms for the ratios p_A and p_{Ao}.

The equation derived for the calculation of the load factor f_L can be considerably simplified, provided that mean values, independent of the drives' speeds and their slips, can be assumed for the torque ratio p_{Ao} and the reserve factor R:

$$f_L = \frac{1}{2} \cdot \frac{p_{Aom}}{p_{Aom} - \frac{1}{R_m}} \cdot \left[1 - s_{sta}^2 \right] \qquad (28)$$

As deduced in Section 3.2, the factor f_L has the same multiplying effect on the energy loss during start-up as the factor f_I. Figure 8 shows a relationship between the rated power output P_{MN} and the highest value of the factor f_I permissible for three no-load start-ups of initially cold low-voltage A.C. motors.

According to figure 8, the factor $(f_I)_{3max}$ decreases like a hyperbolic function with increasing values of the rated motor power P_{MN}, and is all the lower, the higher

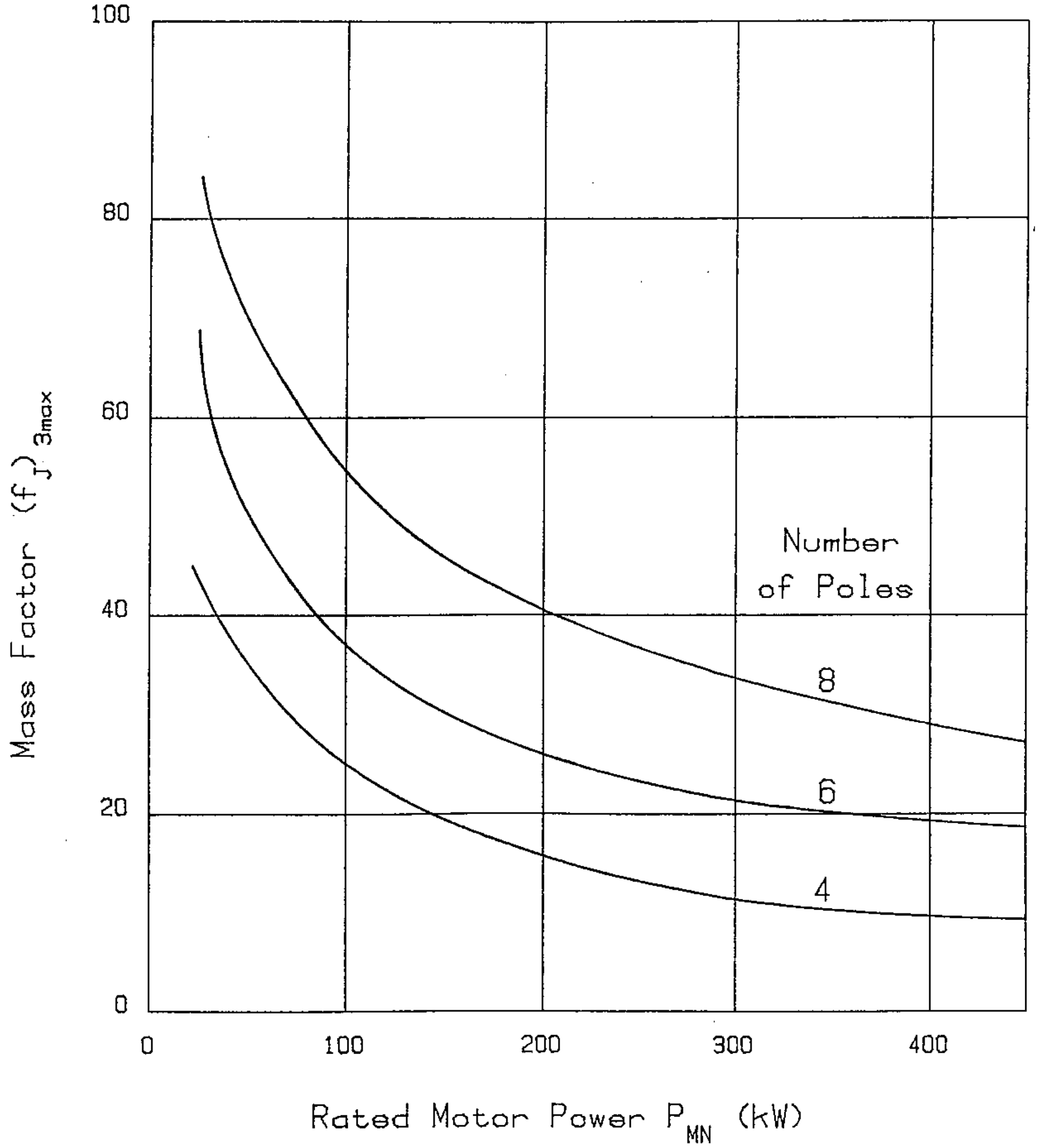


Figure 8 Relationship between the rated power output P_{MN} and the highest value of the factor $(f_I)_{3max}$ permissible for three no-load start-ups of initially cold low-voltage A.C. motors

the rated motor speeds are (lower numbers of poles). This is the reason why, solely with regard to the motor heating effect, the application of directly driving standardized squirrel-cage motors is restricted to small-sized belt conveyor systems with relatively small rated power of the drives. In these systems, it is frequently the case that little attention is paid to the stress of the belts during start-up when compared with bigger systems. The extension of the field of application of these motors is thus only possible by oversizing or a special design as well as by the combination with slipping couplings, on condition that the latter have sufficient thermal stressability.

Figure 9 provides information on the relationship between size and heat capacity C_H of fixed filling fluid couplings with their maximum permissible fillings. The lower limiting curve applies to couplings without and the upper limiting curve to those with high-capacity delayed filling chambers. With these heat capacity values the temperature rise of the types of couplings mentioned can be estimated from the energy loss due to slip with the aid of equation (24). Then this temperature rise can be compared with the permissible temperature of the couplings.

Unfortunately, no general statements can be made with regard to the heat capacity of squirrel-cage motors due to the different types of construction of their rotors.

Again assuming mean values p_{Aom} and R_m, it is possible to derive a comparatively simple equation for the evaluation of the start-up time:

$$t_A = f_I \frac{\omega_o \cdot I_M}{M_{MN}} \cdot \frac{1 - s_{sta}}{p_{Aom} - \frac{1}{R_m}} = \frac{\omega_{sta} \cdot I_M}{M_{MN}} \cdot \frac{f_I}{p_{Aom} - \frac{1}{R_m}} \qquad (29)$$

If this equation is combined with equations (23), (25), (26) the following simple expression can be derived for the energy loss W_s due to slip during start-up:

$$W_s = \frac{1}{2} \cdot p_{Aom} \cdot P_{DN} \cdot t_A \, (1 + s_{sta})$$

$$\approx \frac{1}{2} \cdot p_{Aom} \cdot P_{MN} \cdot t_A \frac{1 + s_{sta}}{1 - s_N}$$

$$\lessapprox \frac{1}{2} \, p_{Aom} \cdot P_{MNel} \cdot t_A \qquad (30)$$

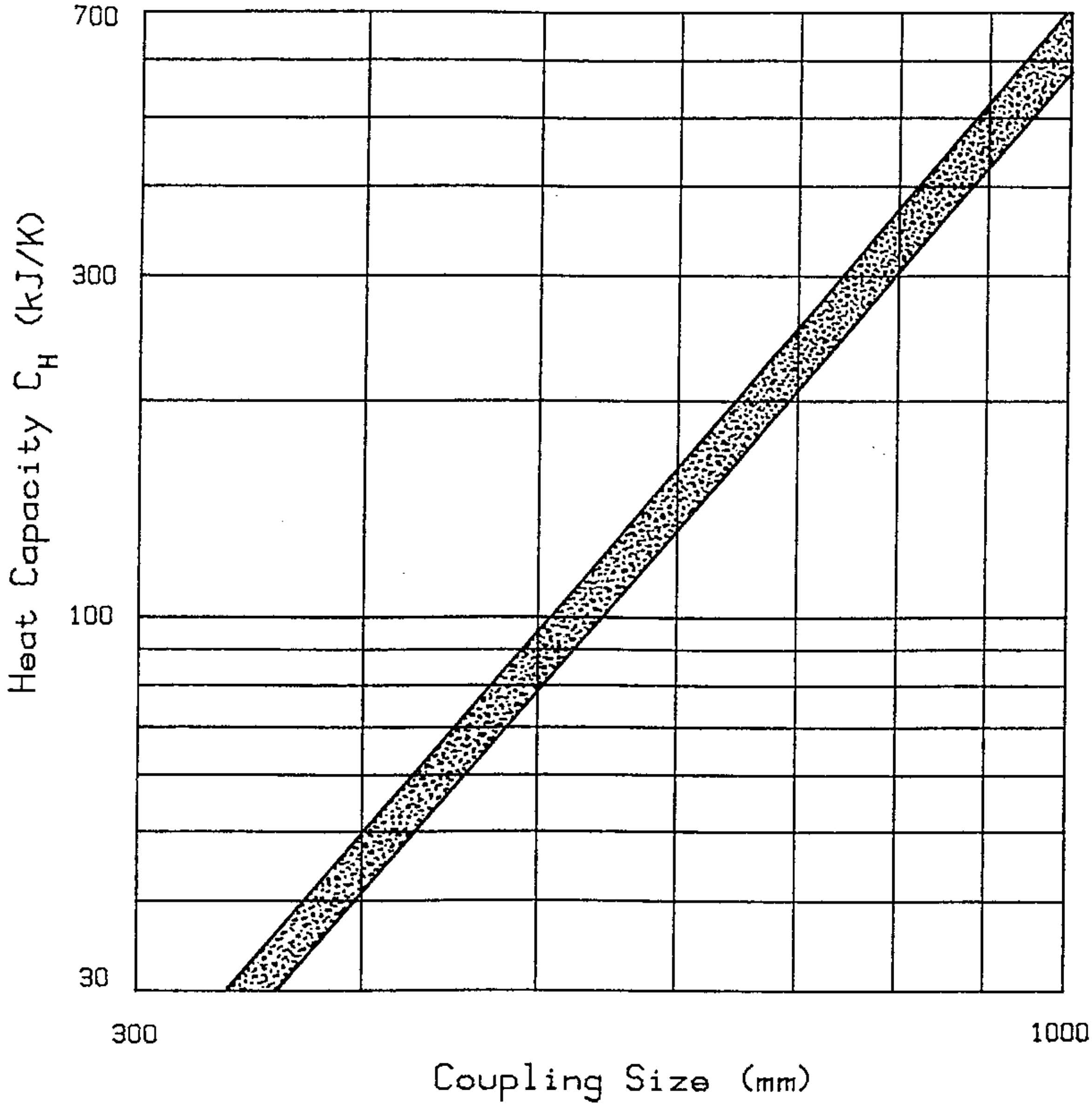


Figure 9 Relationship between the size (≈ profile diameter) and heat capacity of fixed filling fluid couplings with their maximum permissible fillings (data provided by VOITH)

where:

P_{MNel} electric power input per motor under rated load

Finally, a formula for evaluating the energy loss W_s is introduced, which can be used on site with little effort and without knowledge of the torque-speed-characteristic of the drives. When this formula is being derived, the assumption is made, as in Section 3.1., that the resistances to motion of the system during start-up are independent of speed and thus have the same values as during steady operating conditions of the system, when the load of the drives consists of the torque M_{MN}/R. Under this precondition the following applies:

$$W_s = f_I \cdot (W_{kin})_M \cdot \left[1 - s_{sta}^2\right] + \frac{P_{Msta} \cdot t_A}{1 - s_{sta}} \cdot \left[1 - \frac{l_A}{v_o \cdot t_A}\right] \qquad (31)$$

In this equation, the parameters that have not yet been defined have the following meaning:

P_{Msta} power output per motor at its shaft during steady operating conditions

l_A start-up distance of the belt conveyor system

v_0 belt speed due to the synchronous speed of the motors

In order to reduce the effort for the calculation, it is possible to approximate the slip $s_{sta} = 0$ and the power output P_{Msta} equals the electric power input of each drive during its steady-state operation.

3.4 Summary

With the growing application of soft-start techniques for the purpose of minimizing belt stress on the one hand, and the increase of the capacity of belt conveyor systems by installations with higher belt speeds on the other hand, it is more and more frequently the case that long, and partly extreme, durations of start-up procedures occur. Due to the high values of slip during these operational phases, these longer start-up times cause higher energy losses in conventional A.C. drives, which are supplied directly, i.e. without frequency converters, from the three-phase network.

The high energy losses due to slip are the reason for correspondingly high drive heating effects. In the case of directly driving A.C. motors, i.e. those without slipping couplings, this energy loss is the reason for considerable temperature rises of the components in the rotor circuit, which, particularly in the case of squirrel-cage motors with high rated power, leads to restrictions to their use in belt conveyor systems. In contrast, in drive units with slipping couplings, the energy loss occurs in the couplings themselves.

In the foregoing, methods are described for the approximate calculation of the thermal stress of drive units of belt conveyor systems during start-up. These methods also include a comparatively simple relationship between the torque ratio p_{A0}, essentially determined by the characteristics of the drive units, and the force ratio p_A, which strongly influences the peripheral forces of the driven pulleys during start-up, and thus both the values of the maximum belt tensile forces and the duration of the start-up procedures. In addition, information is given that permits an estimation of the thermal stressability of squirrel-cage motors and of fixed filling fluid couplings.