BELTCON 1

BELT CONVEYORS - DESIGN, OPERATION AND OPTIMIZATION

PAPER A3

THE DESIGN OF TROUGHING IDLERS

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The design of troughing idler components, to SABS 1313-1980 and complying with C.E.M.A. recommendations, is critically examined. Proposals for standard idler load ratings are presented, and criteria for improved component design given.

1.0 Introduction

With the advent of the new SABS standard specification for conveyor belt idlers, it has become necessary to critically examine the design of idler components.

The success or failure of a design may often be decided in advance by the way in which the design problem is presented. Many a design has failed to make its mark because the design specification was wrongly prepared.

It is vital to clarify the assignment by defining the functions of the item to be designed and the requirements of the job it must perform. The machine designer must clearly distinguish between the essentials and the presumptions.

What, then, are the essential aspects of troughing idler design?

2.0 What governs idler spacing?

The spacing of idlers along the conveyor belt is a very important factor in the over-all economy of the conveyor. Spacing greatly influences the life of both the belt and the idlers. Idler spacing may also influence the drive-power required, as well as the tension rating and cost of the belt.

The factors affecting idler spacing are often not clearly understood, even by idler manufacturers.

According to the Conveyor Equipment Manufacturers Association (C.E.M.A.):

The best spacing for carrying idlers depends on:

1. the weight of the belt plus the weight of the material load that it carries, and

2. the catenary sag of the belt between the idlers.

Spacing should be calculated to observe the following limitations:

1. Maximum of 3% sag when the belt is operating under normal load.

2. Maximum of 4,5% sag when the loaded belt is standing still.

3. The idler spacing should not exceed the values shown in Table 1.

4. The load on any idler should never exceed the idler load ratings.

For optimum economy, a belt conveyor should be fitted with the most suitable, most cost-effective belt and use the least number of idlers. Assuming that the sag limitations are complied with, what are the required idler load ratings to permit maximum idler spacing to be used?

3.0 Standard Idler Load Ratings

Once an idler spacing has been fixed the load carried by each idler is determined by the weight of the belt and the weight of the material carried on the belt.

TABLE 1 : TROUGHING IDLER SPACING

Belt width, mm	Max. spacing *mm	Min. spacing mm
450	3400	700
600	3000	600
750	3000	600
900	3000	500
1050	2800	450
1200	2800	450
1350	2800	450
1500	2400	450
1800	2400	400

Table based on CEMA recommendations.

*

CEMA prefers to limit maximum spacing to 2100 mm, except in special circumstances.

The volume of materials supported by each idler depends on the idler spacing and the cross-section of the material on the belt. The cross-sectional area varies with the degree of troughing (inclination of side rolls), and the amount of surcharge (angle of surcharge).

The average density of material (kg/m^3) carried by belt conveyors varies tremendously. Bagasse, bark and wood-chips have an average density of 110 - 480 kg/m^3, coal and soil have densities of 720 - 1600 kg/m^3, while crushed rock, cast iron chips and lead ores can have average densities of 2000 - 4300 kg/m^3.

A graph, showing loads carried by idlers at maximum spacing and with different material densities, is shown in Fig 1. Some rationalization in standard idler rating is obviously desirable if idler manufacturers are to provide a practical product range. It is suggested that the required range could be covered using three categories:

i) coal and light materials

ii) average ores

iii) heavy metals

Secondly, it is noted that the cross-sectional area of material on a belt is proportional to the <u>square of belt - width</u>. It follows that an idler of half the belt width would carry only one-quarter of the material load.

Taking the 1500 mm belt as a starting point, and considering three standard idler loads of 500 kg, 1000 kg and 1500 kg per idler, a table of standard idler ratings can be calculated. Such a table is shown in Table 2. The data is plotted in Fig. 1. This table provides a rational basis for engineering design of idlers.

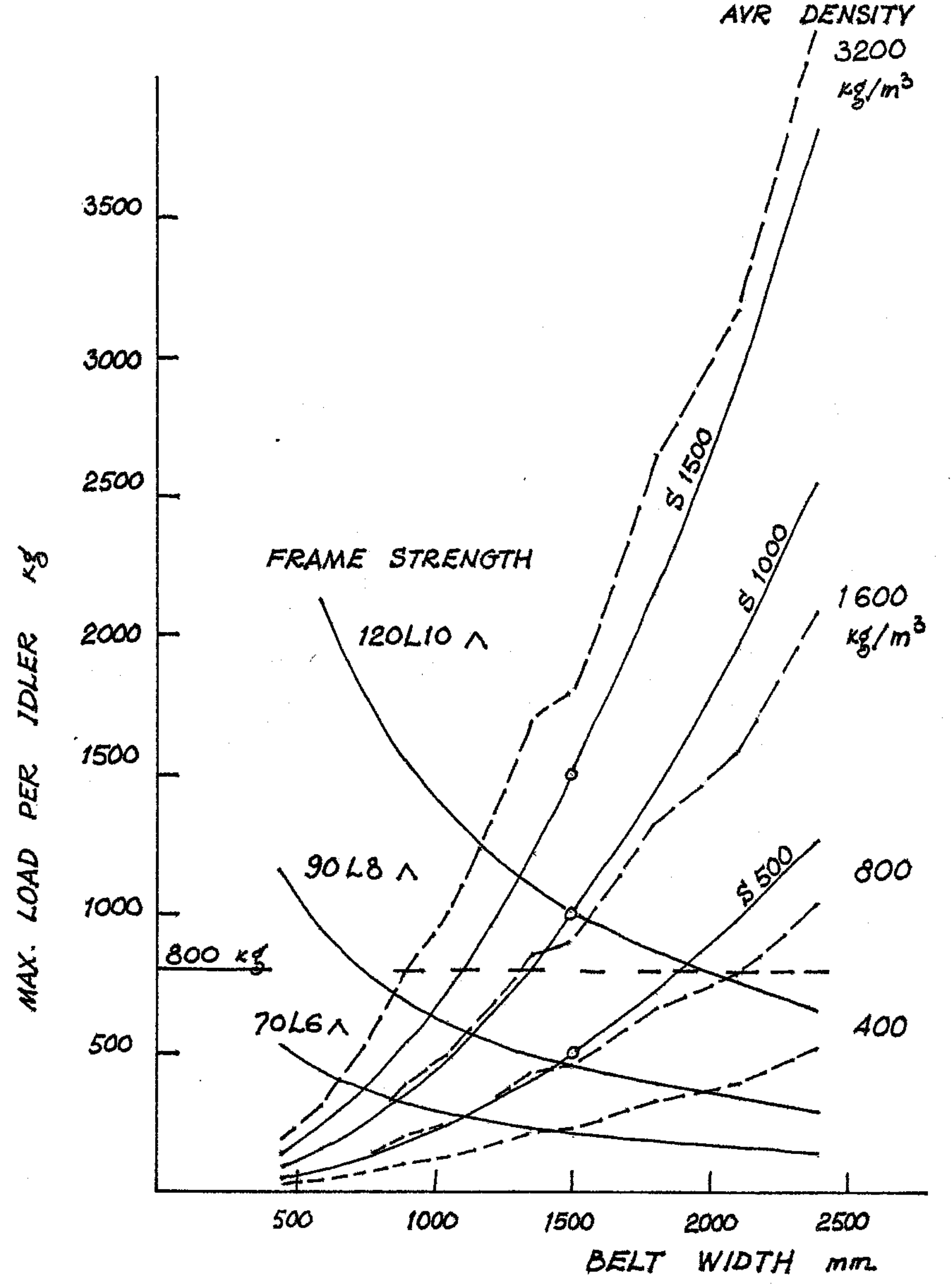

FIG 1 MAXIMUM LOAD PER TROUGHING IDLER

TABLE 2 : STANDARD IDLER LOAD RATINGS				
Belt Width mm	Rating Ratio	Load Rating kg/idler		
		S 500	S 1000	S1500
450	0,09	45	90	135
600	0,16	80	160	240
750	0,25	125	250	375
900	0,36	180	360	540
1050	0,49	245	490	735
1200	0,64	320	640	960
1350	0,81	405	810	1215
1500	1,00	500	1000	1500
1800	1,44	720	1440	2160
2100	1,96	980	1960	2960
2400	2,56	1280	2560	3840

Starting from a basic tabular load rating C.E.M.A. goes on to consider a number of qualifying factors:

1. the speed of the belt,

2. the maximum lump size, and

3. the angle of inclination of the end rolls of troughing idlers.

Consider now the significance of each of these factors.

4.0 The service life of idler bearings

The heart of the conveyor idler is the bearing. The service life of the idler bearing is the total running period during which the properly used and maintained bearing is functional - until the components fail from material fatigue, or the bearing fails to perform its duty due to wear.

Analysis of fatigue life is based on the fundamental equation

$$L = \left(\frac{C}{P}\right)^3$$

Where L is the life of the bearing in millions of revolutions, C is the bearing catalogue load rating, and P the actual load carried by the bearing.

At constant speed it is more convenient to express the fatigue life in operating hours.

$$L_h = \frac{1\ 000\ 000}{60\ n} \left(\frac{C}{P}\right)^3$$

Where L_h is the basic rating life in operating hours, and n is the rotational speed in rpm.

Selecting a belt speed of 3 m/s as reference the bearing fatigue life is inversely proportional to the speed. Thus a bearing running at twice the speed would have half the fatigue life. This is graphically shown in Fig. 2.

A comparison with the C.E.M.A. belt speed factor, plotted on the same graph, shows that belt speed has a far more dramatic effect on bearing life than the C.E.M.A. recommendations would indicate.

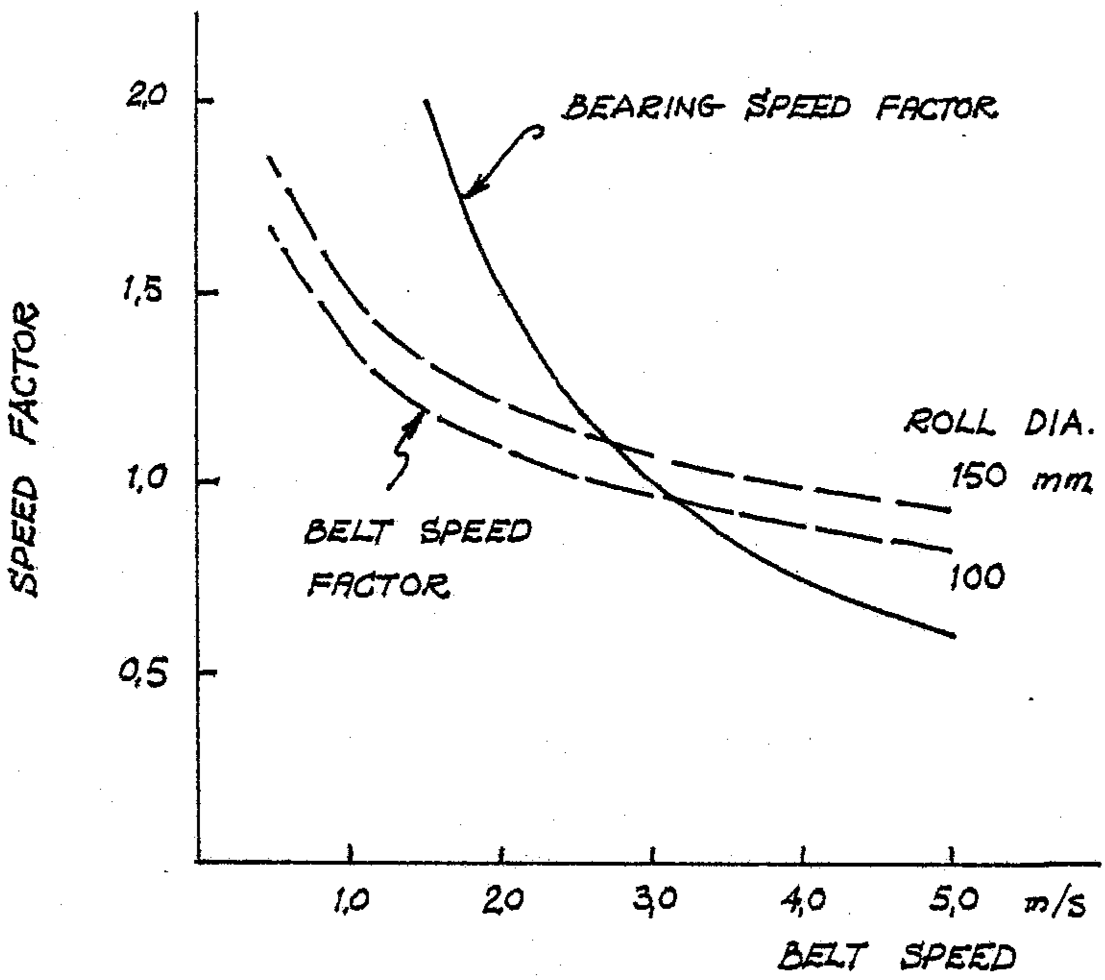

FIG 2 BELT SPEED FACTOR

However, lest bearing life prediction be considered as too much of a science one bearing supplier qualifies his catalogue data as follows:

"The fatigue life represents the upper limit of service life. Due to the influence of wear this limit is not always reached ...... Wear is primarily caused by contaminations which, in the course of time, penetrate into the bearing, but also inadequate lubrication and corrosion due to condensation. Thus the amount of wear suffered by a bearing is primarily dependent on operating conditions, environment, efficiency of sealing systems and lubrication".

The bearing service life is thus the lesser of fatigue life and wear life. Typical wear life figures are shown in Table 3.

5.0 Bearing Selection Vital to Conveyor Life

For long bearing life far more direct emphasis should be placed on the actual bearing selection (C/P ratio), rather than on arbitrary idler spacing limitations.

If W is the load per meter of conveyor and S_i the idler spacing then the fatique life

$$L_h = \frac{1\,000\,000}{60n} \left\{ \frac{C}{WS_i} \right\}^3 \text{ hours}$$

For example, halving the idler spacing will double the cost of the idlers but increase fatigue life 8 times. However, by proper bearing selection the same increase in fatigue life can be achieved without substantial increase in idler costs. When practical limits of wear life are considered the apparently dramatic increases in fatigue life may be purely academic.

TABLE 3 : WEAR LIFE OF IDLER BEARINGS

Operating Conditions	Hours	Years
Excellent		20
Good		10
Average	50 000	6
Bad	10 000	1
Extreme	4 000	0,5

6.0 Idler shaft sizes

SABS 1313-1980 "The dimensions and construction of conveyor belt idlers and rolls" stipulates a range of nominal shaft diameters.

Required shaft diameters for different standard idler load ratings are shown in Table 4. The shaft sizes have been calculated to give adequate bending strength for the various application.

7.0 The impact of lumps

Heavy, sharp lumps will cause more damage to high speed belts due to greater impact at the idlers when there is too much sag in the belt. Occasional large pieces imbedded in fines and centrally loaded on a belt travelling at slow speed are less severe on the idlers than the same size or smaller lumps without the cushioning effect on fines. Lumps not loaded centrally or travelling at high speed have a more destructive effect.

TABLE 4 : IDLER BEARING LOADS

Belt Width mm	Load on centre bearings kg* S 500	S 1000	S 1500	Shaft dia. mm
450	11	23	34	
600	20	40	60	
750	31	63	94	
900	45	90	135	
				20
1050	61	123	184	
1200	80	160	240	25
1350	101	203	304	
1500	125	250	375	
1800	180	360	540	30
2100	245	490	735	
2400	320	640	960	
				40

* Assuming each centre bearing takes 25% of rated idler load.

Of necessity such impacts are primarily a horizontal, rather than vertical, effect on the idler components.

These impact effects are very poorly handled by the C.E.M.A. "lump size factor". This is graphically demonstrated in Fig. 3.

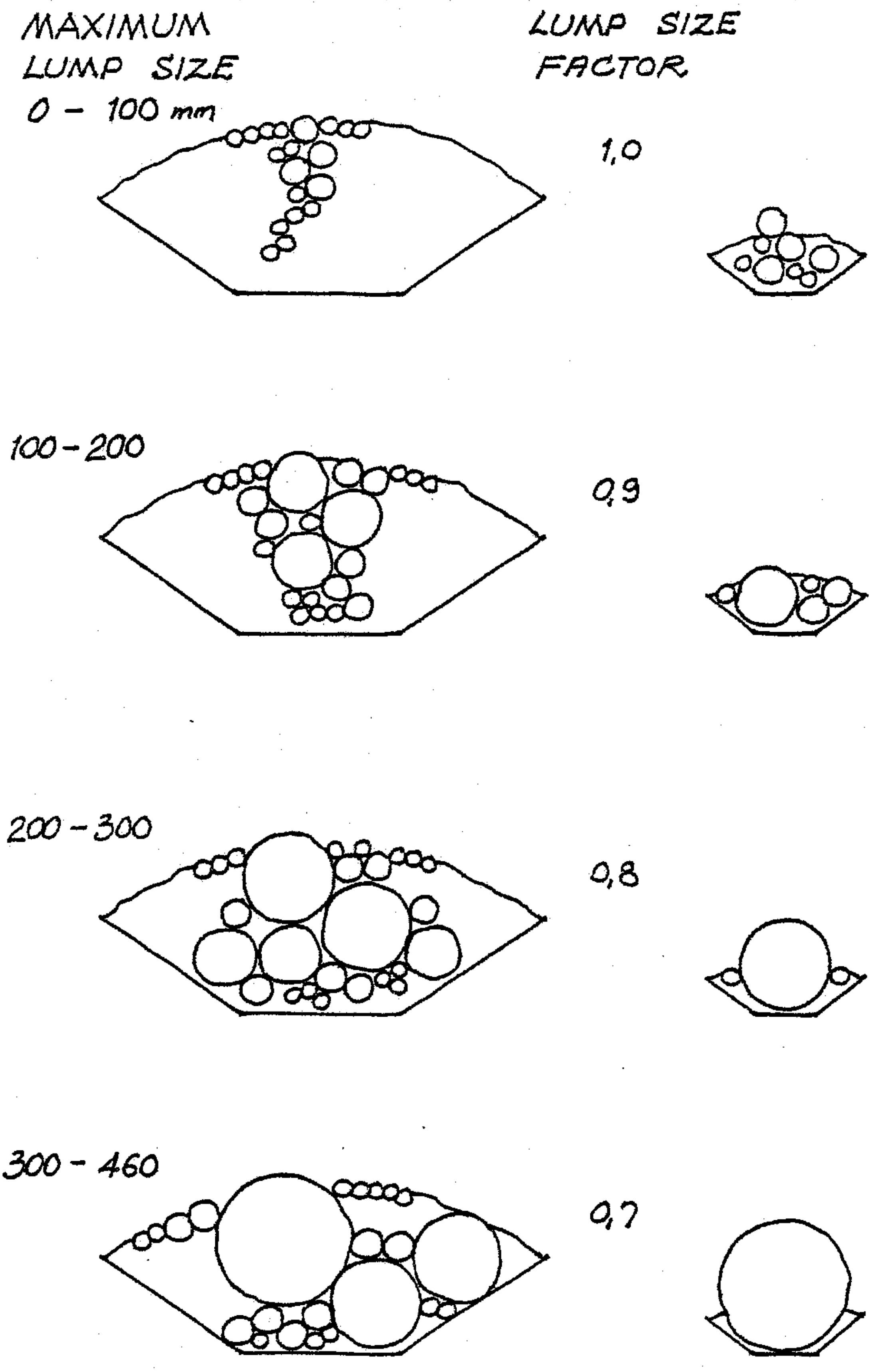

FIG 3 — LUMP SIZE FACTOR

The lump size factor must be related to belt width for two reasons:

1. The ratio of maximum lump size to belt width is clearly significant.
2. Wider idler frames are torsionally weaker than narrow belt frames.

8.0 Idler angle factor

The C.E.M.A. recommendation to derate 35° troughing idlers to 93% of 20° idlers, and 45° troughing idlers to 90%, appears to be highly arbitrary. Which component of the idler is this factor designed to protect? Is it the bearing, the shaft, the frame, or the belt?

If the idler angle factor is intended to give a measure of the horizontal "pinching effect" then a more appropriate formula might be:

$$F_\theta = \frac{1 + 2\cos\theta}{1 + 2\cos 20°}$$

Where θ is the idler angle, the idler angle factors would then be 92% for 35°, and 84% for 45° troughing idlers. It is assumed that this factor will be used to give some "gut-feel" measure of the dynamic load effects on bearing life, and shaft and frame strength.

9.0 How strong the idler frame?

Idler roll support frames can be designed as simply supported beams. Such frames carry the whole of the idler load without inducing secondary stresses in the conveyor stringers. In most cases the stringers have very little torsional strength, therefore the simply - supported assumption is justified.

In some special cases, where tube stringers are used for example, idler support frames could be designed assuming some end rigidity. In such cases lighter support frames could be used.

It is, of course, possible to design frameless idlers. However such idlers are beyond the scope of SABS 1313-1980.

The most common idler frame member is an angle used as an inverted Vee. The safe - load on various angle sizes is shown in Fig. 1.

10.0 Minimum Belt Tension and Belt Sag

So far it has been assumed that the belt tension is adequate to limit the belt sag to within acceptable limits. If the belt tension, when the loaded belt is standing still, is too low the belt will tend to sag excessively, causing spillage of material, decreased belt life, and increased power to drive the conveyor.

What determines the minimum belt tension?

While a belt conveyor is composed of many important parts, none is more economically important than the conveyor belt itself. Generally the belt will represent a substantial part of the initial cost of the belt conveyor. The belt must be selected with the greatest care.

Belt selection is primarily controlled by the effective tension (T_e), and the drive-wrap factor (C_w). The slack side tension T_2 is, by definition:

$$T_2 = (T_e)\ (C_w)$$

The tight side tension T_1 is:

$$T_1 = T_e + T_2$$

The belt cost is largely determined by the maximum belt tension (T_1), while the idler spacing is controlled by T_2. Since T_2 is established by the conveyor designer there is little the idler manufacturer can do about idler spacing. However he can do something about the cost of idlers, at different spacings and carrying different idler loads, by sound, economical engineering product design.

Conclusion

"A great deal of new technology is not new knowledge. It is new perception. It is putting together things that no one had thought of putting together before, things that by themselves had been around a long time". (Peter F. Drucker).

The design of conveyor idlers to SABS 1313-1980, and to the recommendations of the Conveyor Equipment Manufacturers Association, has been critically examined. The techniques of value analysis can now be applied to enable conveyor idlers to be redesigned for greater value.

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