AN INVESTIGATION INTO IDLER SEALS AND THE EFFECT ON RIM DRAG.

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1. Introduction

This paper is concerned with the operating properties of conveyor belt idlers. In particular the effect of seal type has on properties such as rim drag and its consequence in the overall belt tension. Basically there are three types of sealing methods used, each exhibiting a different rim drag friction characteristic. Some end users of idlers have in the past rejected idlers from manufactures that exhibit a higher than normal rim drag on the basis of increased power consumption. There are however many factors which should govern the selection process. This paper compares three different sealing methods used in relation to the power consumed and discusses other factors which should also be considered when choosing idlers.

1.1 Conveyor Idler Overview

Put simply conveyor belt idlers are a means of supporting a conveyor belt by providing a cylindrical shell which is free to rotate on the underside of the belt. On the carry side of the conveyor the idlers are arranged to form the belt into a trough and thus allow a greater capacity of bulk material to be transported.

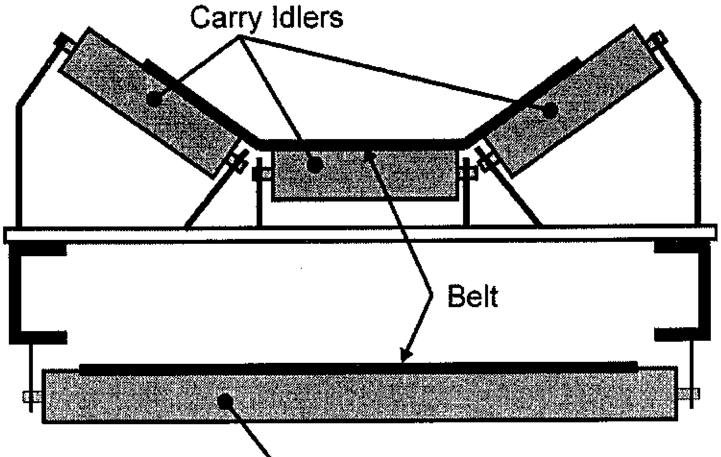

Figure 1: Cross Section of Typical Belt Conveyor

2. Idler Failure

Figure 2 shows a schematic diagram of the components of a conveyor idler. The main components of a conveyor idler that are susceptible to failure are as follows:-

- Shell
  Typically manufactured from steel tube, but other materials such as aluminum, stainless steel may be used where required. The shell is in direct contact with the belt and may wear excessively when rotational velocity is not equal to the velocity of the belt. This can lead to shell thickness failure causing the idler shell to collapse. This wear can be attributed to:-
    - Misalignment
      When idler sets are not fitted to perpendicular to the belt line rubbing of the belt over the idler shell surface can cause wear resulting in premature failure.
    - Vertical curves

When a conveyor is constructed with a vertical curve the outer wing idlers are subject to a no constant radius, and thus a velocity differential is produced between each ends of the idlers. The result is friction between the belt and the shell in certain locations on the shell. The velocity difference is normally very low and not a major problem.

- Foreign matter

  If the underside of the belt is contaminated with either carry back matterial or other foreign matter this can a source of increased friction. The action of grinding hard particles between the belt and the shell can cause an increase in the amount of wear on the shell.

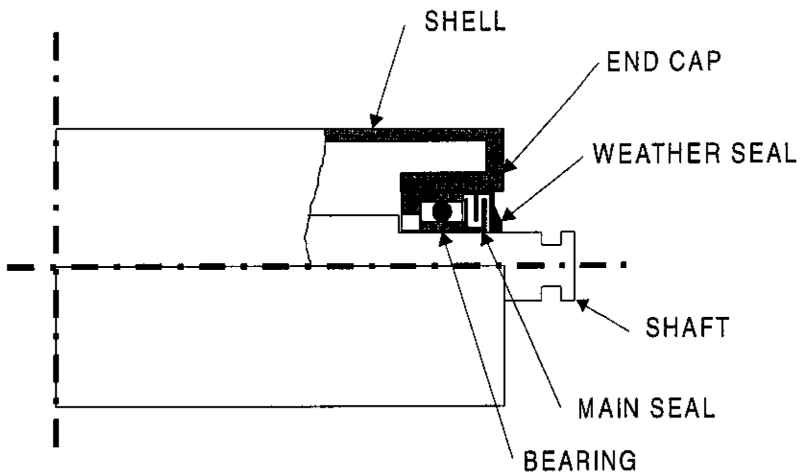

Figure 2: Main Components of a Conveyor Idler

- End cap separation

  This is caused by either wear, corrosion, or mechanical failure of the joint between the shell and the end cap. The end cap separates from the shell and may result in a sharp edge of the end cap causing injury to the belt. This type of failure has the potential to cause many thousands of dollars damage, far in excess of the cost of a replacement idler.

- Bearing Failure

  A seized bearing is probably the most common type of idler failure. Bearing may seize from the following causes:-

  - Overloading of the bearing may cause the bearing to collapse if the incorrect bearings are selected.
  - Incorrect tolerances for the bearing fit may result in the bearing overheating.
  - Misalignment during construction of the idler. If the bearings are not fitted square and concentric with the shell premature failure may occur.
  - Damage to the idler during installation on site. Rough treatment of idlers by maintenance personal when fitting into idler frames. The use of a hammer to fit idlers into a bent or damaged frames or into the incorrect type of idler frame has caused bearings to begin their working life prematurely faulty.
  - Foreign matter entering the bearing as a result of seal failure. The life of an idler relies on the ability of the seal to keep the bearing from contamination. The operational environment of a conveyor along with maintenance procedures (such as wash down with high pressure water) present every opportunity for foreign matter to get past the seal and into the bearing.

3. Sealing Arrangements

The three types of idler seals discussed in this paper are:-

- Labyrinth
- Full contact
- Integral moulded

3.1 Labyrinth Seals

Labyrinth seals are the most common seals used in conveyor idlers. The non-contact construction gives this seal its low frictional characteristics. Manufacturers claim low wear rates due to this lack of contact. Figure 3 shows a schematic diagram of a typical labyrinth seal idler.

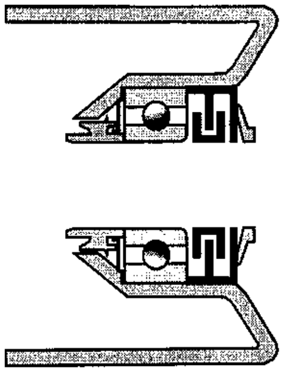

Figure 3: Schematic Diagram of typical Labrynth seal idler

3.2 Full Contact Seals

Typically made from polyurethane these seals have lips that make contact with the shaft on which it is mounted. This contact produces friction, which results in typically higher friction than labyrinth seals. Figure 4 shows a schematic diagram, of a full contact seal idler.

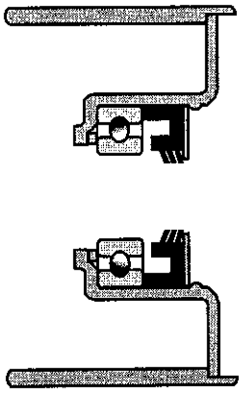

Figure 4: Schematic Diagram of typical full contact seal idler

3.3 Integral moulded Seals

Unlike Labyrinth or full contact seals that are fitted to housing in the end cap of the idler, this system uses a moulded polyurethane end cap to house the bearing with the seal built into the design. While these types of idlers have exhibited the highest friction of the idlers tested the manufacturer claim other benefits such as the vibration absorption of the polyurethane end caps and the ability to recondition rather than totally replace idlers a major advantage. Figure 5 shows a schematic diagram of a typical integral moulded seal idler.

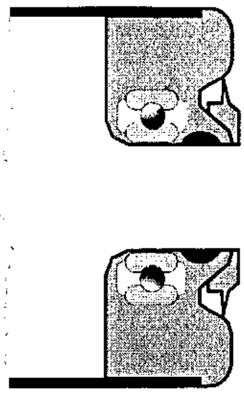

Figure 5: Schematic Diagram of typical integral moulded seal idler

4. Measurement of Rim Drag

Measurements of the rim drag of idlers from each of the seal types mentioned was conducted at the University of Newcastle, Australia. The testing involved mounting the idlers in a lathe and while rotating the shaft the resistance produced at the outer shell was recorded. One end of a cord was attached to the shell and the other connected to a load cell. The forced produced while keeping the shell from rotating was measured when the shaft was rotating at an equivalent rotational velocity to the velocity of rotation the idler would experience on a conveyor belt. The velocity chosen was a belt speed of 5m/s. Figure 6 shows and idler being tested for rim drag.

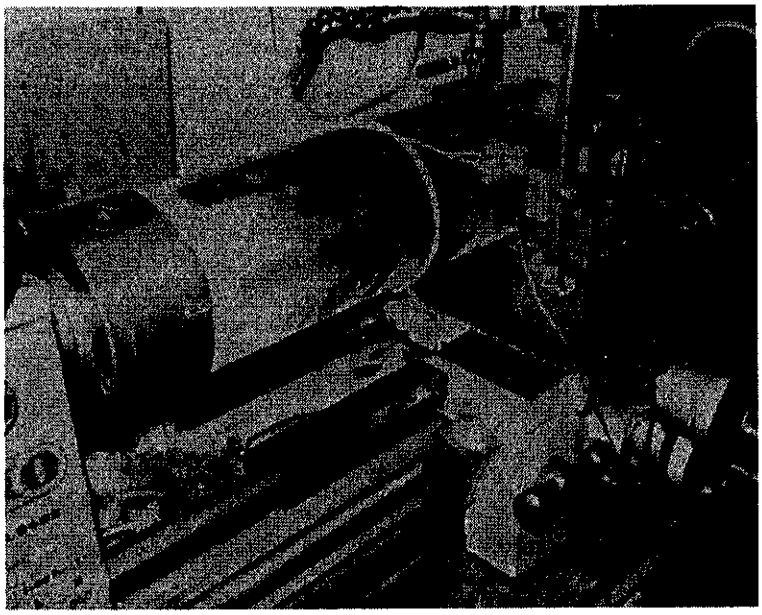

Figure 6: Idler undergoing rim drag friction testing

5. Results of rim drag testing

The results from the rim drag testing are shown in Figure 7. It can be seen the labyrinth type seal exhibits by far the lowest friction while the integral moulded type shows a friction of approximately 20 times greater than the labyrinth seal.

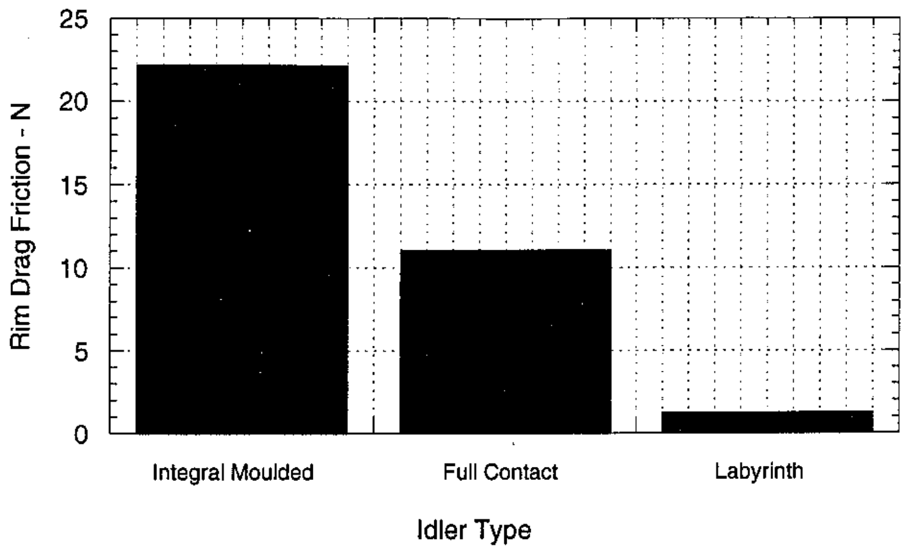

Figure 7: Rim drag measurement for the 3 types of idlers

6. Implications of Seal Types on Conveyor Power

To gain an appreciation of the effect of rim drag on the operational power requirements of a conveyor belt a number of example calculations are presented.

6.1 Belt Tension Predictions

Roberts, [1] gives a number of resistances used in calculating the belt tension in conveyor belt. It was decided for the purpose of this paper to use only the 3 most dominant resistances for the comparison of different idler types. This was deemed not to be an over simplification but would give a reasonable approximation for comparative purposes. f

- Resistance to Rotation of Idlers and movement of empty belt F_{H1}

$$F_{H1} = \left(q_{ro} + q_{ru} + 2q_b \cos\alpha\right)\mu_{r1} Lg \qquad \text{(N)}$$

- Resistance to movement of material horizontally F_{H2}

$$F_{H2} = q_m \mu_{r2} Lg \cos\alpha \qquad \text{(N)}$$

- Slope resistance F_{st}

$$F_{st} = q_m Hg \qquad \text{(N)}$$

where

q_{ro}	=	Mass per unit length of rotating parts of idlers along carry side of conveyor (kg/m)
q_{ru}	=	Mass per unit length of rotating parts of idlers along return side of conveyor (kg/m)
q_b	=	Mass per unit length of belt (kg/m)
α	=	Angle of elevation of conveyor (degrees)
L	=	Conveyor length (m)
g	=	Acceleration due to gravity (9.81m/s2)
μ_{r1}	=	Artificial friction coefficient due to idlers for unloaded belt.
μ_{r1}	=	Artificial friction coefficient due to idlers for loaded belt.

q_m = Mass of material conveyed per unit length on belt (kg/m)

H = Net change in elevation (m)

6.2 Parameters used for the predictions

A number of different tension senarios were calculated to be used in the comparison between the different idler types. All tenson calculations used the following values for each of the variables:-

q_{ro} = 20 kg/m q_{ru} = 7 kg/m q_b = 30 kg/m g = 9.81 m/s^2 μ_{r1} = 0.025 μ_{r1} = 0.030

Velocity of the belt v =5m/s Width of belt b = 1200mm

Number of carry idlers per set = 3 Carry idler troughing angle 35°

Carry idler spacing = 1.2 m Number of return idlers per set = 1

Return idler spacing = 3.0 m Material surcharge angle = 25°

Bulk material density = 800 kg/m^3 and 2500 kg/m^3

Conveyor Length = 100 m, and 1000 m

Conveyor inclination angle = 0°, 7°,and 15°

6.3 Results of Comparisons

For each of the various conveyor lengths, inclination angles and material bulk densities an approximate total belt force was calculated. These results were then compared to the total rim drag forces based on the total number of idlers in each system. A tension ratio was then calculated by dividing the rim drag force by the total belt force predicted.

ie:

$$\text{Tension Ratio} = \frac{\text{Total rim drag force}}{\text{Calculated total belt force}}$$

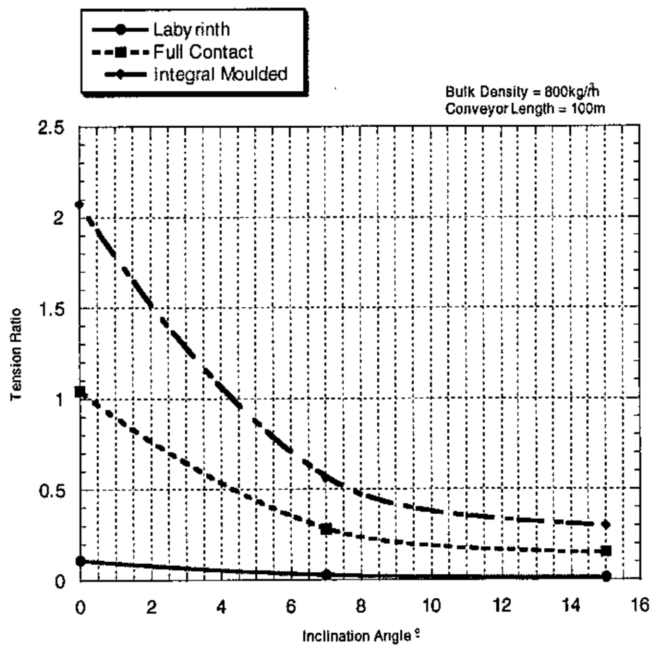

Figure 8: Tension ratio vs inclination angle for bulk density of 800 kg/m^3 and conveyor length of 100m

Figure 8 shows the tension ratio versus inclination angle for a bulk density of 800 kg/m3 and a conveyor length of 100 m. The results show the expected low tension ratio of the labyrinth type seal for all inclination angles. At zero inclination the full contact seal has a tension ratio of approximately 1, which suggests if these idlers where fitted than an increase of 100% in the installed power would be required. At 15° inclination the full contact seal requires and increase in power of approximately 20%. The integral moulded seal type shows at zero inclination a power increase of over 200% is required for these idlers, and a 30% increase is required at 15°.

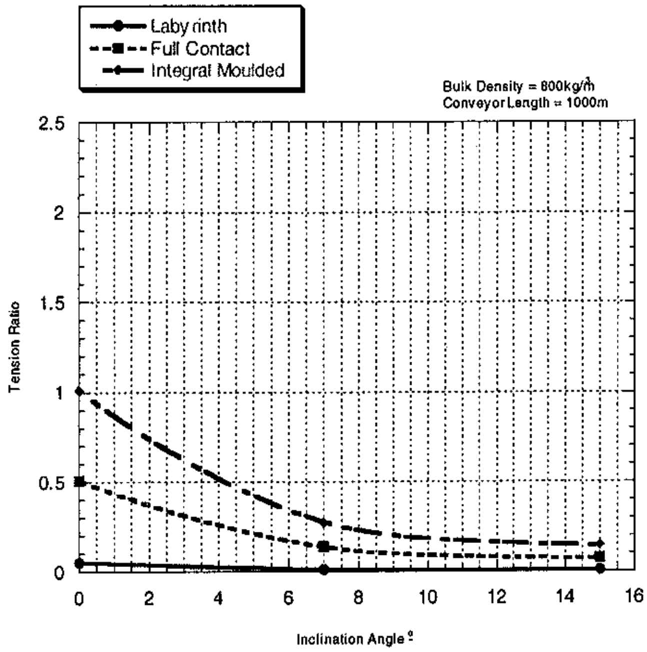

Figure 9: Tension ratio vs inclination angle for bulk density of 800 kg/m³ and conveyor length of 1000m

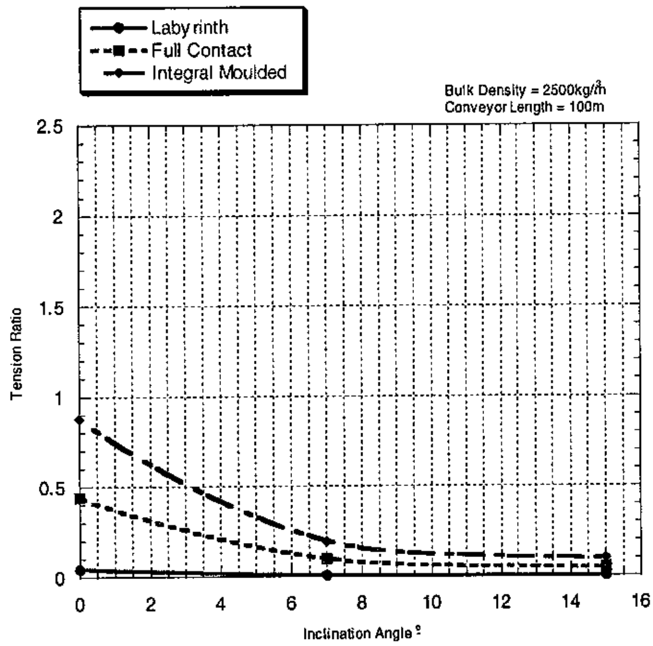

Figure 10: Tension ratio vs inclination angle for bulk density of 2500 kg/m³ and conveyor length of 100m

Figure 9 shows the tension ratio versus inclination angle for a bulk density of 800 kg/m3 and a conveyor length of 1000 m. While the results show a reduction in the required percentage power when compared with the results for a conveyor length of 100 m, considerable power increases (100% at zero inclination), are required for the integral seal type idlers. At zero inclination the full contact seal type idlers require a 50% increase in power and at 15° inclination an increase of 10% power is required

Figure 10 shows even with a material bulk density of 2500 kg/ m3, for conveyor length of 100 m the integral moulded seal type idlers at zero inclination, an increase of almost 90% in power is required, while the full contact seal type need an increase of over 40%. For inclination angles above 8° all idler types require less than 10% increase in power.

From Figure 11 at zero inclination angle with a material bulk density of 2500 kg/ m3 and a conveyor length of 1000 m, the integral moulded and full contact seal types require an increase in power of 40% and 20% respectively. At inclination angles above 7° all idler types require less than 10% increase in power.

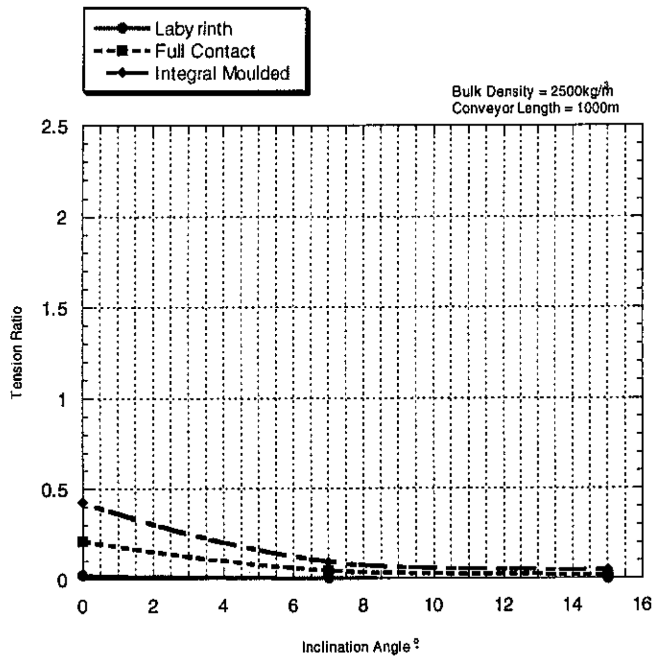

Figure 11: Tension ratio vs inclination angle for bulk density of 2500 kg/m³ and conveyor length of 1000m

7. Conclusions

The type of sealing arrangement used on in a idler can have a significant impact on the power consumption of the conveyor. The selection of idlers for a conveyor system requires thorough investigation. The selection process should include a cost analysis involving such criteria as:-

- Purchase cost
- Installation cost
- Maintenance cost
- Potential cost of idler failure
- Operational cost (power consumption)

It is the maintenance cost and the potential cost of idler failure, which pose the greatest challenge to accurately predict. The job would be made easier if answers to questions such as these were readily available.

- Does seal type greatly effect the life span of an idler?
- Is it more economical to recondition idlers than to replace with new?
- Does an increase in friction mean an increase in sealing ability?

While no rigorous investigation has been done at The University of Newcastle on the reliability of different idlers, preliminary testing equipment has been trialed. Further studies are continuing in this area and it is hoped a testing procedure will be in operation in the near future.

8. References

[1] Roberts, A.W. "Modern Developments in Belt Conveying", Centre for Bulk Solids and Particulate Technologies, The University of Newcastle, August, 1996.

9. Author

Wayne Madden is a professional engineer employed at The Key Centre for Bulk Solids Handling and Particulate Technologies at the University of Newcastle, Australia where he has been involved for the past four years with all facets of bulk materials handling. Specialist areas include, the design, modifications and modelling of various materials handling systems including the operation of feeders, conveyors, and chutes. Currently Wayne is studying part time for a Masters of Engineering on the topic of belt conveyors. Prior to his graduation, Wayne was foreman of a fabrication company servicing the mining and agricultural industries.

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