SUGAR MILL DRIVE COUPLINGS – AN ALTERNATE DESIGN

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ABSTRACT

The detrimental side effects of tailbar couplings on gearboxes and mill-roller shafts are familiar to all sugar-mill engineers. Various papers have been written since 1968 discussing these effects and possible improvements in the design of tailbar couplings. A completely new concept of coupling is presented in this paper. It totally replaces the tailbar and is virtually maintenance-free. After five successful years of operation the coupling has demonstrated that it has completely eliminated the problems typical of tailbar couplings.

INTRODUCTION

The detrimental side effects of using conventional tailbar-and-collar couplings to drive sugar mill top rolls are familiar to all sugar-mill engineers. Tailbars generate thrust as well as radial loads that apply bending moments to the driving and driven shafts. Since the 1960s various papers have been written qualifying and quantifying these loads.

1. Cullen discussed the operating stresses in mill roll-shafts, including the effects of the tailbar on these stresses.

2. Powers and Clarke quantified the tailbar thrust and described a method of accommodating the thrust.

3. Powers and Clarke described various modifications to the conventional tailbar to eliminate thrust generation.

4. Reid discussed shaft fractures and the contribution that tailbars make to these breakages.

The papers and recommendations presented thus far have not led to any widespread changes, and today mill drive couplings remain unchanged from those used a century ago.
In 1987 the first prototype of a radically new design of coupling was installed at a sugar factory in South Africa. It totally replaces the tailbar coupling, is essentially maintenance-free and has shown that it has eliminated all of the detrimental side effects of tailbars.

Before describing and discussing the new coupling, the side effects of conventional tailbar couplings are briefly considered.

**Thrust**

Tailbar couplings generate substantial variable thrust between mill and gearbox. The amount of thrust is dependent on driving torque, coefficient of friction between materials in contact, condition of the male and female squares, and the angle of misalignment which is in turn a function of the length of the tailbar and the relative positions of driving and driven shafts at any given time.

The amount of thrust generated would typically be 147kN (15 tons) for a 2,134 mm mill with 2,440 mm long tailbar at 12.7 mm offset transmitting 930 kN-m torque, and substantially greater in most large mill drives, because most tailbars are not as long as this and the conditions of the squares is invariably far from perfect. This thrust between mill and gearbox often results in damage to mill and gearbox bearings and has even been known to force shafts through final-drive gearing.

It is necessary at the mill-design stage to allow for this anticipated thrust in the form of heavy duty bearings, sometimes special thrust bearings, and expensive spur gearing that can take small lateral relative movements without suffering any damage. It is not uncommon for mills and gearing to be set up allowing for 3 mm of wear on the thrust fillets of the bearings in one season.

**Radial loads**

Tailbar couplings also generate radial loads that have been estimated to be 40% of the thrust loads, although this figure could be dependent on the clearance between male and female squares of the specific coupling.

These radial loads exert bending moments to the squares of the shafts and there is growing acceptance that these bending moments contribute substantially to the breaking of top-roll shafts. These loads can be quantified in a well-lubricated tailbar coupling with newly machined working faces. As the faces wear lubrication deteriorates resulting in ever higher bending moments until the point of coupling seizure when the loads become unquantifiable.
The misalignment is only in the vertical plane, and thus the bending moments applied to the roller-shaft are in the same plane. When the mill shaft is lower than the gear shaft, this additional bending moment reduces the stress level in the shaft, but when the mill shaft is higher than the gear shaft the generated bending moments increase the maximum stresses at the critical points in the shaft.

Cullen1 discussed shaft-stresses under operating conditions. His closing recommendation to improve shaft-life is “Dispensing with the muff coupling and using a coupling capable of accommodating lift without imposing bending moments onto the shaft would eliminate all problems of misalignment”.

**Mill-shaft fractures**

When mill-shafts fracture, they invariably break either at the inner journal radius, or just inside the shell seating. The fractures are seldom straight. The continued rotation, after the break, of the broken stub-shaft relative to the now-stationary roller can generate massive thrust forces not only between the mill cheeks but also between the mill and the gearbox, transmitted by the tailbar. Serious damage can result to mill and gearbox bearings and even to the gears themselves.

**NEW DESIGN**

In 1986 the author designed a completely new type of coupling for driving sugar mills. This design satisfies the requirements of a sugar-mill coupling, in that:

- It is inexpensive compared to other commercial couplings of similar torque capacity.
- It is maintenance-free.
- It can be uncoupled in only 2 to 3 minutes.
- It can accommodate 100 mm of radial misalignment, 300 mm of end-float and 3° of angular misalignment, all far more than required of a sugar mill coupling.
- It cannot generate or transmit thrust between the mill and gearbox.
- It applies only quantifiable radial loads between shafts (see Appendix I). However, arrangements can be made to ensure that these loads do not increase the stresses in the shafts.

**Design**

The design of the coupling is based on the principle of applying the torque between driving and driven shafts through flexible members acting tangen-
tially in a plane normal to the shafts. The coupling is shown in Figure 1, and consists of the following components.

1. Driving and driven yokes fitted to the shaft squares. The heads on the end of the arms each have two ferrule housings for holding the ends of the steel ropes.

2. Two pairs of steel tension ropes for transferring force from the heads of the driving yoke to the heads of the driven yoke. The line of action of the ropes is tangential and normal to the shafts.

3. A floating “compression-plate” with rope-grooves in the radiused ends. It is clamped between and supported by the ropes when driving and bends the ropes into two parallel sets of flexible members at right angles to one another.

4. A support-pipe that carries the mass of the compression plate, when the mill is not being driven, and also acts as a safety feature in the event of a pair of ropes breaking.

5. A reversing mechanism for freeing a choked mill or barring the mill for surface-maintenance on stop days.

FIGURE 1. A new mill coupling design.
Principle of operation

The deflection of the steel ropes into two parallel formations provides misalignment absorption properties to the coupling. Considered in static mode and referring to Figure 1, the vertical ropes permit x-x axis misalignment, while the horizontal ropes allow y-y axis movement. The flexibility of the ropes also permits relatively large z-z or axial movement. The coupling is therefore totally flexible in the three normal planes.

Dimensions

The overall dimensions of the coupling are large when compared to a tailbar coupling. Typical swing radius for a 1.6 MN-m coupling driving a 2,134 mm mill would be 1,350 mm. This is necessary because of the minimum radius required for any steel rope under flexing conditions.

Design life of the steel ropes

A steel rope is considered discardable by the manufacturers when only ten visible wires have failed within a length of six rope diameters. The life of a steel rope is directly dependent on angle of deflection, number of flexing cycles per minute, and pre-tensile stress level (i.e. the working load as a percentage of the Absolute Breaking Force—A.B.F.).

The amount of misalignment determines the angle of flex required of the ropes every cycle. Most large sugar-mill top-rolls can be misaligned at the journals by up to 32 mm ("worst conditions"), but under "normal" operating conditions would seldom be more than 10 mm misaligned. At the center of the coupling, it would therefore need to accommodate 50 and 16 mm respectively to accommodate "worst" and "normal" conditions.

Driving torque

For a 2,134 mm mill, normal required driving torque would be 1,200 kN-m, while 1,600 kN-m would normally be available from the prime mover. Figure 2 shows predicted life in cycles plotted against load for 1.5° and 3° deflection in the ropes selected for the coupling.

In the first coupling built 1.5° and 3° deflections corresponded to approximately 16 and 32 mm of misalignment at the centerline of the coupling, or approx. 10 and 20 mm at the journals.
The 18.5% and 14% pre-tensile stress lines correspond to maximum and "normal" torque conditions, and indicate, for example, that "normal" operating conditions would result in a life of $19 \times 10^6$ cycles of operation, equivalent to 103 months of operation @ 5 rpm and 85% overall time efficiency. At twice this misalignment the life would reduce to $70 \times 10^6$ cycles of operation, or 38 months of operation. Rope manufacturers claim that the life of the ropes can theoretically be trebled by periodically rotating them 120°.

**Radial load generated**

As mentioned earlier, tailbars apply bending moments to the shaft squares, and there is little doubt that these bending moments contribute to the breaking of mill roll shafts.

Rope couplings apply only radial loads to the shafts, and these loads in themselves do translate into bending moments along the shaft. However, their magnitude is quantifiably low. (see Appendix I).

Typical radial load for a coupling transmitting 1,200 kN-m of torque with the shafts misaligned 16 mm at the coupling centerline would be 32 kN or 3.26 tons. This figure would be linear with respect to misalignment, i.e. 1 mm of misalignment generates 2 kN of radial load.

To this figure, one must add half the mass of the coupling or approx. 35 kN. Naturally this component acts downwards on both shafts. As mentioned earlier, the direction of the radial loads attributable to misalignment would depend on...
the relative shaft positions. These forces act vertically up when the mill shaft lines up below the gear shaft at the centreline of the coupling, and vice versa.

It is only when the mill-shaft is higher than the gear shaft that the resulting bending moment increases the stresses in the shafts at the journal fillet and shell-seating areas, where they are most likely to fail. This additional bending moment can be totally negated or minimized simply by ensuring that the gearbox shaft is generally higher than the mill shaft. (i.e. lift the gearbox ± 20 mm).

Operational experience

There are currently two units operating. The first or prototype unit was installed in April 1987 at the Umfolozi Sugar Factory in South Africa, to drive a 2,134 mm Walker Mill at the end of a seven-mill tandem.

Prior to the installation this particular mill had broken three top rolls in one season, and in addition had always had problems with breaking holding-down bolts on the final-drive output-shaft bearings.

Five seasons (or 45 operational months) later, there have been no further broken top rolls on this mill, and no more problems with holding-down bolts. The mill engineer also reports that the condition of the final-drive gear-teeth has actually improved since the coupling installation, no doubt due to the elimination of periodic gear misalignment resulting from broken bolts.

Also, shortly after the coupling was installed, hydraulic caps were fitted to the mill to afford some measure of relief in the event of tramp-iron passing through the mill. This mill was originally equipped with discharge hydraulics only.

Shortly after the hydraulic caps had been fitted, one cap failed because of a material flaw, resulting in more than 200 mm of misalignment in the coupling. No damage resulted to either the mill (other than the caps), the coupling, or the gearbox.

Had such a failure occurred with the original tailbar in place the final-drive gearing would almost certainly have sustained severe damage, and the roller-shaft square would probably have broken off.

No tension ropes have had to be replaced at Umfolozi because of rope-strands breaking from fatigue.
FACTORY ENGINEERING

A second coupling was installed in April 1991 at the Malelane Sugar Factory in South Africa, also to drive a 2,134 mm Walker Mill. No problems have been reported by the mill staff.

Design changes

Figure 1 shows the original general design, and how the coupling could be adapted to replace different length tailbars. The latest designs now incorporate the following changes:

- The method of fixing to shafts has been changed from rigid keys to a simpler “slide-on-and-lock” system. The locking system comprises reverse jacking screws in the four half-faces of the square bore not subject to driving loads. The screws pre-load the driving faces.

- The reversing ropes have been replaced by a simple chain connection through lugs under the yoke-arm heads.

- The rope ferrules are now larger in diameter and the rope-fitting is simplified. Disconnecting the coupling can be done within three minutes.

- The flange-connection between yoke and rigid-half coupling has been eliminated to reduce costs.

CONCLUSION

The coupling has achieved all it was designed to do. The incorporation of this coupling into the design of the drive can change design philosophy as follows:

- It permits the use of lighter spherical-roller bearings in the final-drive gearbox. No continuous lubrication is necessary.

- It permits the use of alternate, less expensive, gearing such as double-helical, herringbone, or even high-quality hardened and ground helical or planetary gears.

- In the event of a transverse mill-shaft break, the mill cheeks and gearbox could be totally protected from damage caused by generated thrust, but it would be necessary to modify the mill bearings in one of the following ways.
The mill-bearing fillets could be reduced or even eliminated on the inside of the journals, alternately both fillets could be removed on the driven side of the mill. Another possibility would be to eliminate the retaining flange on the inside of the bearings.

REFERENCES


APPENDIX I

Calculation of Radial load generated by the coupling.

Consider the 1,600 kN-m coupling installed. The effective working radius of the ropes is 1 m.

The force P acting on each arm is therefore

\[ \frac{1600}{2 \times 1} = 800 \text{ kN} \]

The angle \( \theta \) is a function of the actual misalignment and the distance between rolling points of the ropes on the compression plate and yokes, which is 600 mm.
Therefore for 16 mm misalignment at the centerline of the coupling (caused by journal misalignment of 10 mm),

\[ x = \arctan \frac{16}{600} = 1.53° \]

\[ R = P \tan x \]

Radial component of load at maximum torque of 1600 kN-m will be \( 2 \times R \)

\[ = 2P \tan 153° = 4267 kN \]

Therefore typical radial load applied by the coupling absorbing 16 mm of misalignment at normal torque of 1200 kN-m will be 32kN.

**UNE CONCEPTION DIFFERENTE POUR LA LIAISON DE L’ENGRENAGE ET DU MOULIN**

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**RESUME**

Les liaisons engrenage/moulin faites par tail-bar ont des désavantages qui sont connus par les ingénieurs sucriers. Plusieurs travaux, depuis 1968, discutent ces effets et proposent de nouveaux concepts pour le tail-bar. Ce papier propose une conception nouvelle. Le tail-bar est éliminé complètement et le système reclame très peu d’entretien. Après cinq ans d’opération, sans problème, ce mode de liaison a éliminé complètement tous les problèmes associés avec un tail-bar.
ACOPLES DE MANDO EN MOLINOS AZUCARERO – UN DISEÑO DISTINTO

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RESUMEN

Los efectos negativos que se derivan del uso de las barras de acople entre los reductores y los ejes de las mazas, son muy conocidas por todos los ingenieros de la industria azucarera. Varios escritos se han presentado desde 1968 discutiendo estos efectos y las posibles mejoras en el diseño de las barras de acople. Un concepto, completamente nuevo de estas barras de acople es presentado en este escrito, este reemplaza totalmente las barras y elimina casi en su totalidad su mantenimiento. Después de cinco años en operación, los resultados han sido satisfactorios. El acople ha demostrado que ha eliminado los problemas típicos de las barras de acoples.