MEASUREMENT OF THE POWER SPLIT ON A FIVE ROLLER MILLING UNIT

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Abstract

Using a technique well tried in the Australian sugar industry the power split between the pressure feeder and mill of five roller Walker milling units was measured. Measurements were taken on the first and last units of a five unit milling tandem. The equipment used, some of the problems encountered and results obtained are described.

Introduction

During the 1978 milling season the SMRI was requested by the SZ engineering staff to measure the power split between the pressure feeder and mill of their five roller Walker mills. Each milling unit of the Walker tandem has a single steam turbine as a power source. The pressure feeder section and mill section are fed by separate shafts after two reduction gear boxes. There are five such milling units in this tandem and the power split was determined by measuring the input torque on the tail bars of the respective sections of each unit. Torque measurements were made on No. 1 and 5 milling units. No attempt was made to measure the fibre throughput during the tests to relate power and fibre, or power difference between the back and front mills.

This paper describes the equipment used, some of the problems encountered and the results obtained.

Measuring device

There are a number of ways of approaching a torque measurement problem. The most obvious and perhaps most common torque measurement method would be using strain gauges. This method is especially suitable for round shafts where it is relatively easy to theoretically relate strain and torque. Unfortunately this relationship is a bit more complicated for square shafts such as tail bars.

An alternative method would be to measure the angle of twist of a given length of shaft. A technique of this method is to attach two arms to the shaft and to measure the relative change of position between the arms. This approach has been used by the Australian sugar industry and their results indicated that theoretically calculated deflections agreed very well with actual measured deflections for a given torque. Macey and McGinn[1] stated that their experiments indicated that the difference between theoretical and measured deflections was never more than 5%. Rather than physically calibrate the measuring equipment by twisting the tailbars it was decided to use this proven technique for torque measurement. As in the Australian industry the movement between the arms was measured using displacement transducers. The displacement transducers were of the differential transformer type with a DC input and output. These transducers were fed from stabilised power supplies using an integrated circuit (LM 340 series voltage regulator) and supply battery.

An attempt was made to use a short range FM telemetry system to link the transducer output to the recording equipment. Interface equipment had to be built to accept the transducer output which was not pure direct current but had a high frequency ripple. The system as a whole, transducers, radio link and recorders when set up in the workshop, functioned well. However this was not the case when installed in the factory.

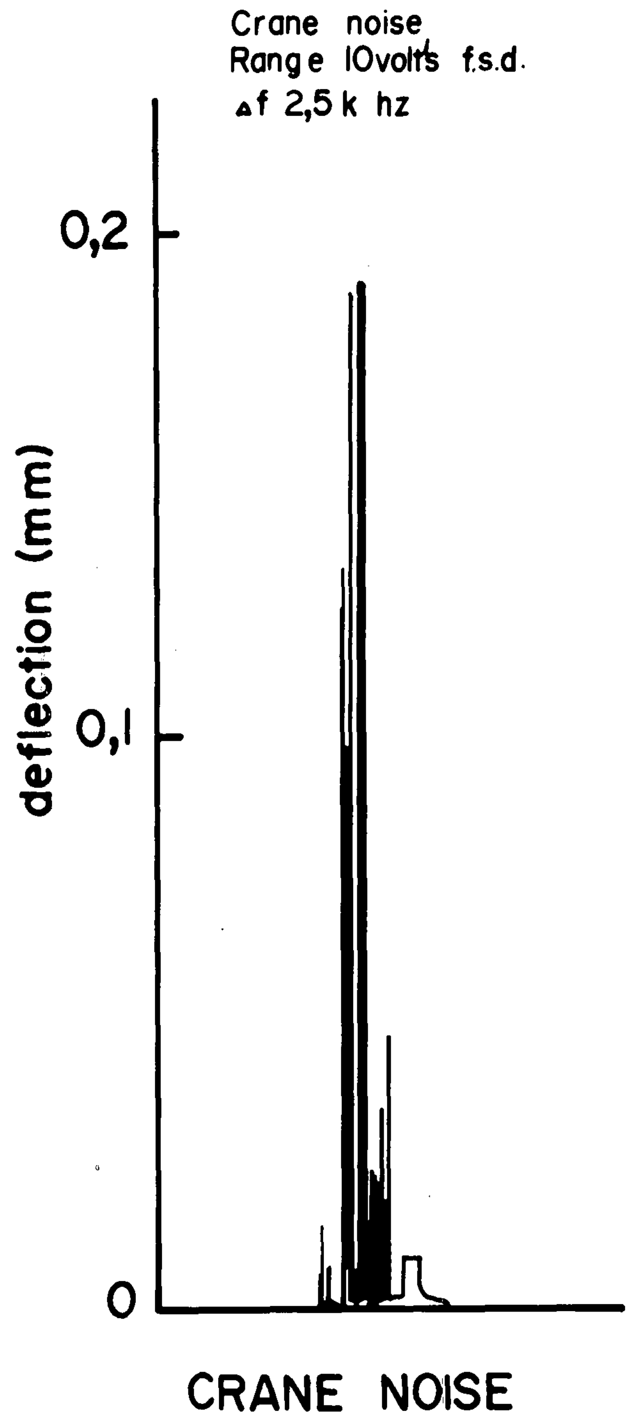


FIGURE 1 Radio interference from crane.

Various aerials were used to improve signal strength but to no avail. Fig. 1 shows the signal picked up by the receiver when a crane passed overhead. As can be seen this signal was proportional to 0,2 mm displacement of the torque arms. This value is very significant as the maximum deflections measured were of the order of 0,4 mm. After the failure of the telemetry system a continuous wire system was used.

Test procedure

Displacement arms with transducers were attached to the two shafts which were being measured for torque. The displacements arms, transducers and power packs are shown in Figs. 2 and 3.

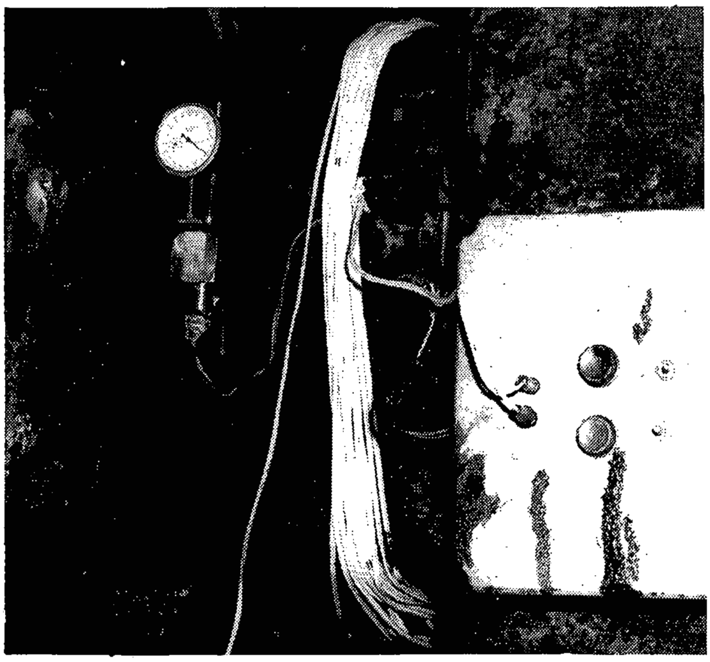


FIGURE 2 Measuring equipment on mill drive shaft.

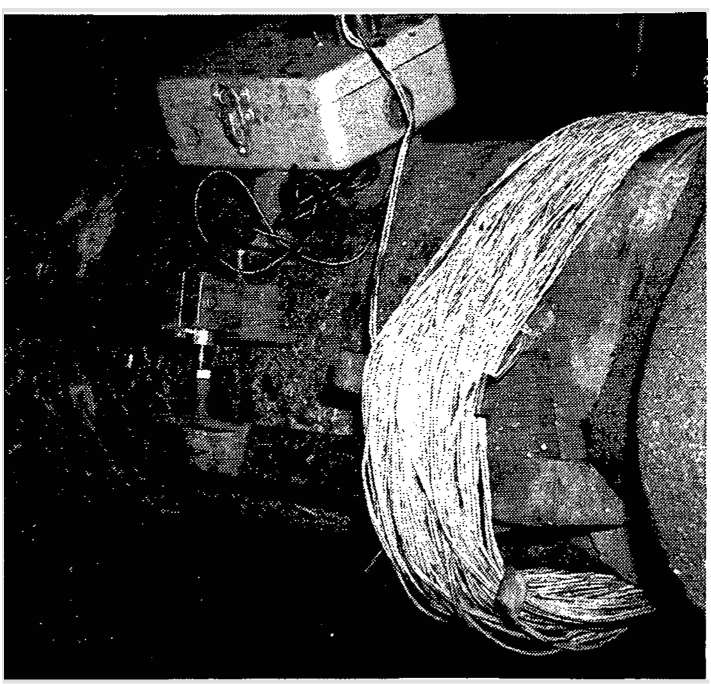


FIGURE 3 Measuring equipment on pressure feeder drive shaft.

The wire flex was wound onto the shaft in such a direction that it first unwound from the rotating shaft, then rewound onto the shaft allowing a total testing time of about 35 minutes. Unfortunately the linearity of the transducer output voltage was dependent upon the output load so this was checked for various displacements before each test. During the tests the turbine speed and first stage pressure readings were noted and the time of readings marked on the recorder chart so that the delivered power and absorbed power could be related (and provide a form of check on the accuracy of the results).

Results and discussion

As the shafts with the transducer devices were not physically calibrated and because the transmitted torque was theoretically determined from a measured angle of twist, the developed and absorbed power were compared to validate the results. Unfortunately the power absorbed by the gearboxes was not known. The David Brown representative was consulted and he indicated that a power loss of 15 %, between turbine output and mill input could be expected. The Australian sugar industry[3] has, using similar measurement methods, calculated gearbox losses of 25 %. However, unless the conditions of the gear trains is known it is difficult to relate the losses.

Shown below are lists of results from the two test runs. Various points were selected at random on the recorder charts. For each point, the mill, pressure feeder and turbine power were calculated.

No. 1 Walker Mill (test date 23.10.78)

Mill	Power (kW)			Percentage basis		
	P.F. i*	Gearbox ii†	Turbine Output iii‡	Mill	P.F.	Gearbox
1. 463,0	90,6	101,4	655	71	14	15
2. 513,9	89,1	68,4	671	77	13	10
3. 416,9	96,5	81,6	595	70	16	14
4. 412,3	84,9	94,8	592	70	14	16
5. 522,1	105,9	54,0	682	77	15	8

No. 5 Walker Mill (test date 4.12.78)

Mill	Power (kW)			Percentage basis		
	P.F. i*	Gearbox ii†	Turbine Output iii‡	Mill	P.F.	Gearbox
1. 265,0	22,5	75,1	363	73	6	21
2. 262,6	23,5	63,8	350	75	7	18
3. 322,3	30,6	84,3	417	74	7	19
4. 260,0	25,0	67,9	353	74	7	19
5. 274,7	24,5	84,2	383	72	6	22
6. 266,8	24,5	92,1	383	70	6	24
7. 292,5	25,0	79,4	397	74	6	20

* i PF ± Pressure feeder
† ii Gearbox absorbed power = Turbine output — (Mill + PF absorbed power)
‡ iii Calculated from turbine expected performance curves

The above power values were calculated using the formulae shown in the appendix. The appendix also includes a sample calculation.

An examination of the gearbox loss for the tests conducted on No. 1 and No. 5 mills, which return average gearbox losses of 13 % and 20 % respectively, reveals a good agreement with the value predicted by the David Brown representative. Judging by the noise and vibration emitted from the No. 5 mill gear train when compared to No. 1 mill gearing a higher gearbox loss would have been anticipated for No. 5 mill gearing. The calculations tend to confirm this.

No. 1 Walker Mill

The calculations indicated that approximately 73 % and 14 % of the developed turbine power is fed to the mill and pressure feeder respectively.

Fig. 4 of deflection (torque) versus time curves clearly indicate that the torque loading for the pressure feeder leads that of the mill. The cyclic variation as well as the non-uniformity of the milling unit loading is very evident from the above curves.

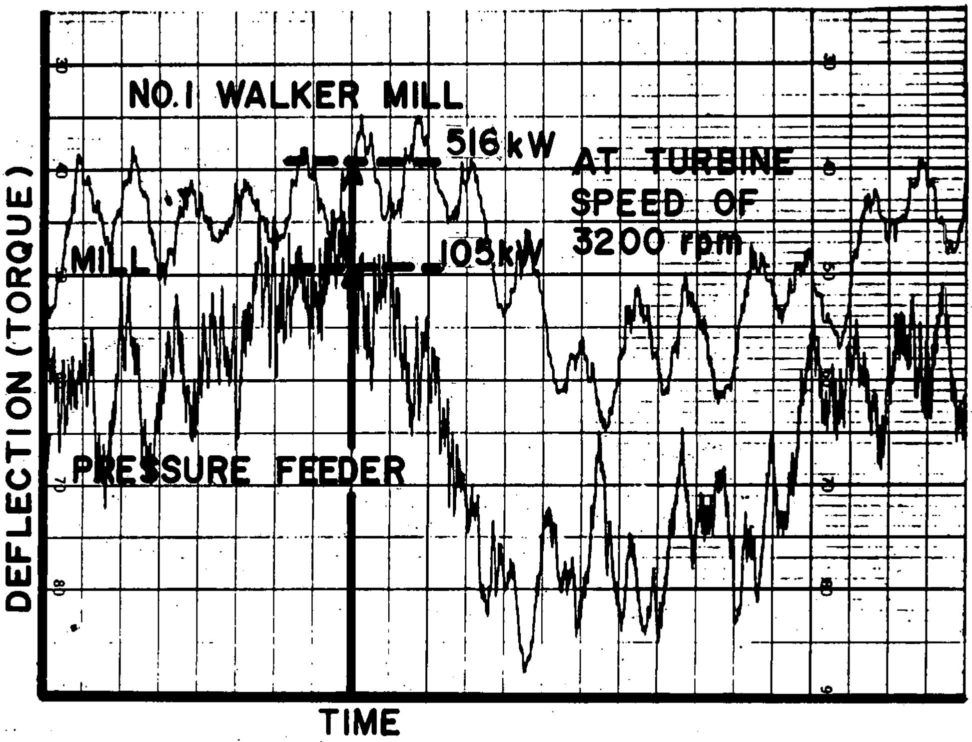


FIGURE 4 Recording of torque from No. 1 milling unit.

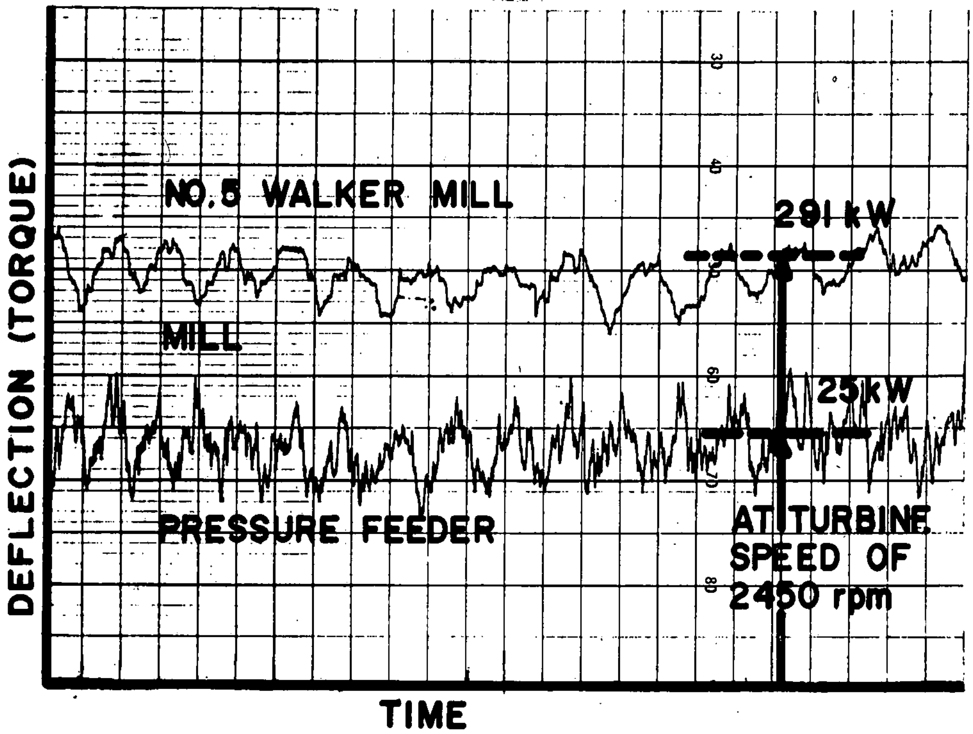


FIGURE 5 Recording of torque from No. 5 milling unit.

No. 5 Walker Mill

The calculations indicated that approximately 73 % and 6 % of the developed turbine power is fed to the mill and pressure feeder respectively.

Fig. 5 of the curves of deflection (torque) versus time clearly indicate a cyclic variation. However the torque loading on both the mill and pressure feeder are fairly uniform.

The calculations indicate that the pressure feeder loading is fairly low, especially on No. 5 Walker Mill. The power ratio split is approximately 5:1 and 12:1 for No. 1 and No. 5 Walker Mills respectively. Initially it was felt that some of the cyclic variation could have been due to slight bending of transducer arms. However, when No. 5 Walker Mill was rotated under no load conditions it indicated negligible cyclic variation while considerably more variation was returned underload condition. No load condition is shown in Fig. 6.

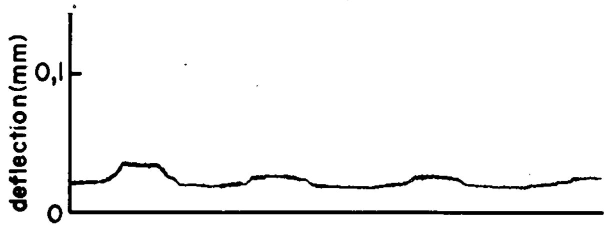


FIGURE 6 Recording of torque under no load conditions.

REFERENCES

1. Macey, D. and McGinn, J. A. (1975). Torque requirements of a roller bearing mill. QSSCT Proc. **42**, 167-170.
2. Timoshenko, S. (1969). Strength of materials, Part 1 (third edition) Van Nostrand Reinhold. 281-300.
3. Sugar Research Institute (Mackay) (1961). Technical Report No. 69. An investigation of the performance of No. 2 pressure feeder and mill at Kalamia, Australia.
4. American Society of Metals. Metals Handbook Volume I (eighth edition). 132.

Appendix A

Theory:

If a twisting couple (torque) is applied to a shaft the degree of twist per unit length is proportional to the twisting moment. Timoshenko[2] gives the following relationship for a rectangular cross sectional shaft:

$$\theta = \frac{Mt}{B\,b\,c^3\,G} \text{ radians/metre} \quad . \; . \; . \; . \; . \quad \text{(i)}$$

Mt = twisting moment

b is the longer and c the shorter side of the solid rectangular cross-section.

B = constant

= 0,141 for a square shaft.

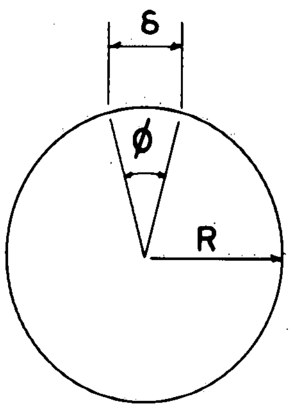
G = shear modulus of shaft

$= 7{,}72 \times 10^{10}$ N/m² $(11{,}2 \times 10^{6}$ lb/in²)[4]

For a solid circular shaft the following relationship holds

$$\theta = \frac{32\,Mt}{\pi\,d^4\,G} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{(ii)}$$

d = shaft diameter



If we consider a shaft as shown above to have twisted through a small angle φ and we measure the deflection δ at some radius R the following relationship exists:

$$\varphi = \frac{\delta}{R} = \theta L \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{(iii)}$$

L = length over which deflection is measured.

Now rearranging (i) and using (iii)

$$Mt = \frac{\delta}{R\,L}\,B\,b\,c^3\,G \quad . \; . \; . \; . \; . \; . \quad \text{(iv)}$$

Power (W) = Mt ω

ω = angular velocity

$$W = \frac{2\,\pi\,N}{60}\,Mt \quad . \; . \; . \; . \; . \; . \quad \text{(v)}$$

N = rpm of shaft

Therefore power transmitted by shaft using equations (iv) and (v)

$$W = \frac{2\,\pi\,N\,\delta}{60\;RL}\,B\,b\,c^3\,G$$

$$= \frac{N\,\delta\,2\,\pi}{RL\;60} \times 0{,}141 \times 0{,}432^4 \times 7{,}72 \times 10^{10}$$

$$= \frac{N\delta}{RL}\,3{,}97 \times 10^4 \text{ kW (for 0,432 m square shaft)}$$

Using the following variables:

N = 3,65 rpm

$\delta = 3{,}71 \times 10^{-4}$ m

L = 0,381 m

R = 0,305 m

W = 461 kW

It can similarly be shown that the power transmitted by a circular shaft can be represented by:

$$W = 6{,}87 \times 10^3 \frac{N\delta}{RL} \text{ kW for a 0,305 m diameter shaft}$$

Using the following variables

N = 4,78 RPM

$\delta = 3{,}3 \times 10^4$ m

L = 0,505 m

R = 0,236 m

W = 91 kW

Appendix B

Recorded Data

Deflection and Turbine details

No. 1 Walker Mill (23.10.78)

	Deflection ($m \times 10^{-4}$)		*Turbine Details*	
	Mill	P.F.	1st Stage Nozzle press (kPa)	Speed rpm
1.	3,71	3,30	517	3200
2.	4,19	3,30	537	3150
3.	3,45	3,63	469	3100
4.	3,53	3,30	483	3000
5.	4,19	3,86	552	3200

Arm Spacing

Mill = 0,381 m

Press. feeder = 0,505 m

Radius at which displacement was measured

Mill = 0,305 m

Press. feeder = 0,236 m

No. 5 Walker Mill (4.12.78)

1.	2,89	1,19	359	2200
2.	2,87	1,24	345	2200
3.	2,87	1,32	372	2700
4.	2,72	1,27	331	2300
5.	2,64	1,14	345	2500
6.	2,57	1,14	345	2500
7.	2,87	1,19	359	2450

Arm Spacing

Mill = 0,365 m

Press. feeder = 0,505 m

Radius at which displacement was measured

Mill = 0,297 m

Press. feeder = 0,236 m

Two test runs were made on No. 5 mill. The first test results were rejected as a certain amount of difficulty was experienced with the calibration of the transducers. This was not rectified before the scheduled start up.