THE TONGAAT SHREDDER

By B. ST. C. MOOR
Tongaat Sugar (Pty.) Ltd.

Abstract

The design, development and commissioning of the first of a new type of shredder is described. While utilising the results of recent research and experimentation with conventional shredders, the Tongaat design incorporates several departures from the traditional type of construction. These innovations yield significant benefits in construction, maintenance and performance. The excellent results from the first season's operation are described, and numerous features of the design are listed.

Genesis

During a tour of several of the world's major cane sugar producing countries towards the end of 1970, it was apparent that one area of recent technological development from which the South African industry could derive considerable advantage was cane preparation. Both milling in Australia and diffusion in Hawaii yielded high extractions largely by using very fine preparation to ease the task of the extraction plant.

With the relatively high fibre content (15% - 16% Cane) and "tough" nature of the fibre at Tongaat, the potential gains from better preparation were likely to be high. However, the 0,9% improvement in extraction from better shredding predicted by Murry[1] of the Queensland S.R.I., using the MILSIM computer programme, applied to a shorter milling train and could presumably not be expected on the larger of Tongaat's two tandems, which has seven mills. Tongaat nevertheless investigated the replacement of the existing 330 kW 2 100 × 1 050 mm Gruendler shredder by a 2 100 × 1 500 mm Walkers shredder, but, in order to effect certain improvements in mechanically troublesome carrier arrangements, the standard shredder casing would require modification. It was during consideration of this problem that ideas were formed for the design of a machine somewhat different from any of those commercially available, but which appeared to have numerous advantages over conventional shredders.

Main design criteria

As preliminary design data, a machine having a rotor 2 150 mm wide with a hammer tip diameter of 1 500 mm and running at approximately 1 200 rpm was decided upon. Study of the work of Shaw and Shann,[2] Crawford[3] and Payne[4] confirmed the suitability of these dimensions for the duty required on Tongaat's larger tandem, and work on the drawing board was commenced.

Design proceeded rapidly, with engineers, draughtsmen and foremen all contributing constructive criticism and suggestions. The new machine was commissioned in May, 1972, having been conceived, designed and built in less than a year, with about 85% of the fabrication being done in Tongaat's fairly well equipped maintenance workshops.

The most important feature of the Tongaat shredder is the use of alternately staggered profiled plates instead of the conventional disc and spacer construction used in all other models. This device affords numerous very important advantages, several of which are listed in Appendix A to this paper. With eight rows of hammers desired, an attempt was made to use simple square plates for the rotor of this first machine, but it proved necessary to notch out the sides of the squares slightly (See Figures 1 and 2) in order to accommodate ideal hammer size and proportions. Appendix B describes a useful "check formula" for hammer proportioning. Straight-sided triangular plates have in fact been used in a 1 300 mm version (See Figure 3) subsequently built for the 1973-74 season. This smaller machine was designed because its rotor would have provided an emergency replacement for the severely damaged one of the previous shredder on this tandem. Fortunately, the old shredder did manage to survive the rest of the season.

Returning to the original 1 500 mm shredder, it was desirable from the maintenance point of view to have hammers of simple rectangular construction rather than the more complicated shapes favoured by the manufacturers of the Gruendler and Silver machines, or that tested by Shann and Cullen[5] or the double-edged one of Clarke and McCulloch.[6] Thanks to the staggered plate rotor and correct proportioning, hammers were designed which would be effective but could be simply cut from a standard size of flat bar, drilled and hard-surfaced.

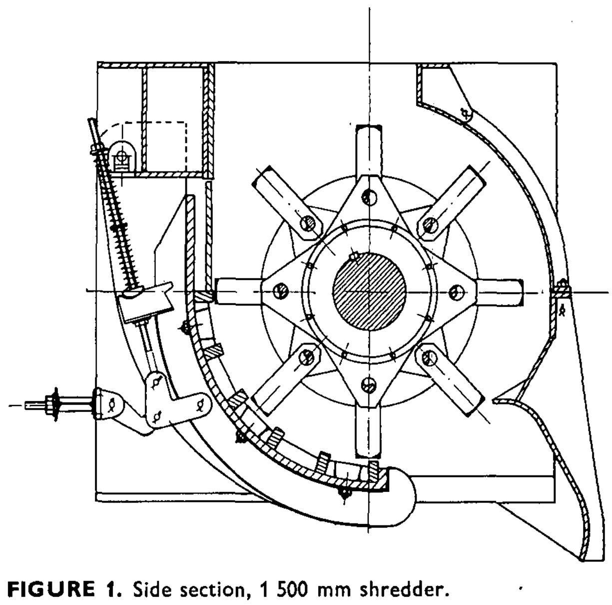

FIGURE 1. Side section, 1 500 mm shredder.

The grid plate had to be fully adjustable, preferably with facilities for adjustment on the run. Despite the exceptional angle of hammer swing-back available with the staggered plate rotor, it was also deemed advisable to provide "give" in the grid plate suspension to pass any very large items of tramp or abnormal cane surges without damage or choking. The unique spring loaded toggle arrangement met both these requirements.

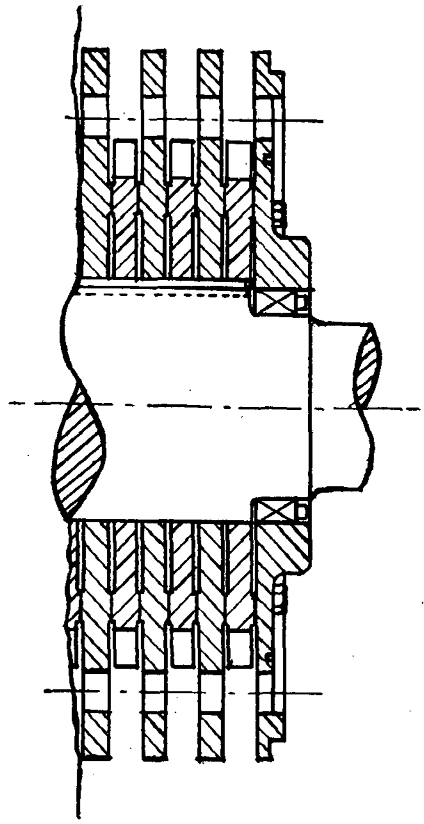

FIGURE 2. Part front section, 1 500 mm rotor.

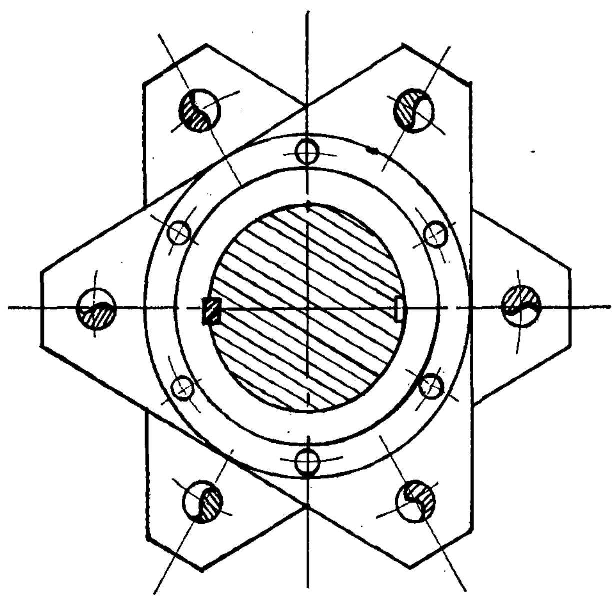

FIGURE 3. Side section, 1 300 mm rotor.

Greenwood's paper[7] comparing different types of shredder grids came to hand just as this element of the design was about to commence. While not in full agreement with Greenwood's "Column Theory", his practical experimentation was sound and convincing, and confirmed the conclusions derived from less carefully controlled experimental work at Tongaat. A pocketed-type grid was therefore decided upon, but again with the simplest possible form of wearing bars and provision to use four corners of each bar.

Cane feeding arrangement

The importance of an even feed tangentially into the mouth of the shredder cannot be over-stressed. Cameron[8] has pointed out the waste of available power which can result from poor feeding, and it is widely accepted that "surging" is most detrimental to shredder performance (see also Greenwood[7] and Shaw and Shann[2]). The chosen configuration of final knives discharging directly onto the feed plate to the shredder (see Figure 4) is very similar to that at Farleigh, Queensland, described by Clarke and McCulloch[6] in their paper already quoted. In conjunction with an automatic control maintaining the carrier at a primary speed inversely proportional to the depth of cane entering the final knives (see Figure 4), the arrangement achieves the objectives of the feed metering rolls used on the Silver "Buster" and favoured also by Crawford,[9] but without the extra mechanicals involved. The direct feed from the final knives also has the advantage of eliminating one carrier completely — however, it is essential to have a sufficiently robust shredder to handle everything leaving the final knives, since no tramp or other foreign material can be removed between the two!

Drive

Quotes were initially called for a 1 200 kW geared turbine drive. However, there was concern about the quantity of high pressure steam available for this drive, and a 900 kW Elliot turbine (which would be

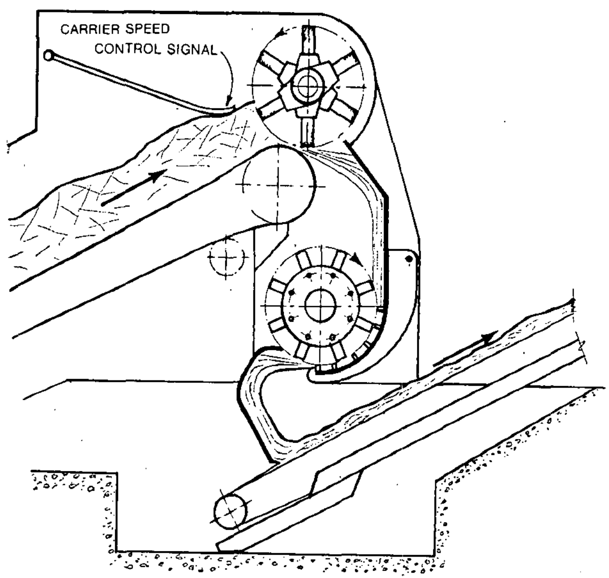

FIGURE 4. Shredder feeding arrangement.

of use elsewhere in future planning) was ordered instead. The gearbox (a 4 : 1 David Brown double-helical unit) was also a well proven model from a most reputable manufacturer, so all anxiety was confined to the shredder itself — until two months before scheduled start-up.

At this stage Tongaat learned that the turbine had been mislaid somewhere between the American manufacturers and the Durban firm who were to supply the drive. This was the start of a tragi-comic saga which culminated in the Tongaat Factory Manager travelling half-way round the world in urgent search of a replacement. He was actually in the offices of a turbine manufacturer in Canada ten days before scheduled start-up when he learned that the turbine had been found, having been in Durban docks for three months!

Commissioning

Start-up was delayed one day by the missing turbine. Fortunately, time was saved in that, when the bare 10 metric ton rotor (the hammers weigh a further 3,3 tons) was finally run up to speed, it ran dead steady and vibration-free without a single balance weight having been added. (The same perfect dynamic balance occurred with the rotor of the 1 300 mm shredder installed the following off-crop).

There was considerable tension amongst the onlookers as the first cane approached the new shredder. All went well, however, and it was immediately apparent that the preparation was exceptional, satisfying Payne's[4] requirements of "... producing a mixture of shreds and fine pith tissue".

Initial problems

However, half an hour later the lubrication system of the drive overheated and oil pressure was lost. The oil cooler supplied was obviously totally inadequate for the turbine and gearbox, and the shredder was by-passed while this was remedied. Other problems with this lubrication system were subsequently found — wrongly set relief valves, inadequately rated filter, oil pump and suction strainer and restricted oil supply line to the white-metal bearings of the gearbox. These deficiencies took some two weeks to identify and eliminate before the shredder could finally be put on sustained trial.

At this stage the one and only serious problem with the shredder itself was encountered — it could only handle about 140 - 150 metric tons cane/hour, beyond which the 900 kW turbine pulled up. To counteract this, it was found necessary to open the grid setting to 20 - 25 mm all round and to remove the last 2 of the 6 grid bars. With these two modifications the drive was able to cope with over 200 tons/hour of 15,5% fibre cane quite adequately. It is estimated that between 1 200 kW and 1 500 kW would be required to drive the shredder at 1 200 rpm on 32 tons fibre/hour with all 6 grid bars installed. At this condition a D.I. (by the S.M.R.I. method developed by Buchanan[10]) of between 93% and 95% would be achieved.

Other problems with the new shredder were minimal. This was undoubtedly due to the contributions of the experienced staff, who had previously maintained Tongaat's Searby and Gruendler shredders and were both to fabricate and to maintain the new shredder. Their influence is evident in many design details, eliminating over-elaboration and maintenance problem areas. Furthermore, the free discussion overcame the snags at the drawing stage, not after commissioning. In fact, during the first season it was found necessary to make more modifications to the "standard" turbine and gearbox drive than to the shredder itself. The only changes made to the shredder were a change in the discharge plate (the Chief Engineer had initially insisted on an over-elaborate arrangement seen in Australia!), a minor change in the locking arrangements for the hammer bars, and chamfering one edge of the side wearing plates.

Performance appraisal

The basic internationally patented features of the Tongaat Shredder are extremely simple, yet the result is a unique and extremely effective machine.

Keynotes of its construction are ease of maintenance and robustness. Numerous maintenance features are listed in Appendix A — a list too long to discuss in the main body of this paper. Its robust power is best demonstrated in reducing cane knives of 125 mm × 20 mm silicon manganese flat bar to pieces about the size of matchboxes. The largest item of tramp so far to have survived the shredder intact is the 300 mm diameter ring of a set of sister hooks from one of the cane unloading cranes. This ring of 50 mm round bar remained intact, but all four hooks and their associated chains were stripped from it and shattered. The gratifying aspect of this incident was that absolutely no evidence of the passage of this tramp could be identified when the shredder was opened at the next week-end stop.

However, the most important criterion by which to judge the shredder is its performance, and some comments on this aspect are also included in Appendix A. Despite the power limitations discussed above, the new shredder consistently averaged over 90% D.I.[10] throughout its first season of operation, compared with a figure of 76 - 78% achieved from the 335 kW Gruendler shredder during the previous season. (A 335 kW Searby shredder on the smaller tandem achieved 78 - 80% D.I.). With only the shredding arrangements altered, fibre throughput on the large tandem increased by between 3% and 5% for the same mill settings; and for the same imbibition % fibre, the extraction of this seven-mill train was increased by approximately 0,5%.

At this level of performance, the first of the new Tongaat Shredders will have completely paid for itself early in its second season of operation.

Acknowledgements

The debt owed to engineers of the Australian industry is obvious from the References quoted. However, many unquoted South African and other Australian technologists have contributed to the general "know-how" required for the design of the

Tongaat Shredder. A special tribute must also be paid to the full Tongaat team who contributed to the design, fabrication and a few early modifications of the first machine.

REFERENCES

1. Murry, C. R., 1971. Computer Aided Studies of Crushing Train Performance, Proc. Qd. Soc. Sug. Cane Tech. Thirty-eighth Conf., p. 228.
2. Shaw, M. and Shann, D. S., 1970. Preliminary Experiments in Cane Shredder Research, Proc. Qd. Soc. Sug. Cane Tech. Thirty-seventh Conf., p. 171.
3. Crawford, W. R., 1969. Mechanics of Swing-Hammer Shredders, Proc. Qd. Soc. Sug. Cane Tech. Thirty-eighth Conf., p. 329.
4. Payne, J. H., 1968. Cane Diffusion — The Displacement Process in Principle and Practice. Proc. I.S.S.C.T., 13th Congress, p. 103.
5. Shann, D. S. and Cullen, R. N., 1968. Some Aspects of Shredder Hammer design, Proc. Qd. Soc. Sug. Cane Tech. Thirty-fifth Conf., p. 95.
6. Clarke, C. D. and McCulloch, R. J., 1970. The performance of a Heavy Duty Shredder. Proc. Qd. Soc. Sug. Cane Tech. Thirty-seventh Conf., p. 181.
7. Greenwood, W. A., 1971. A Comparison of Shredder Grids, Proc. Qd. Soc. Sug. Cane Tech. Thirty-eighth Conf., p. 239.
8. Cameron, C. D., 1970. Experiences with a Heavy Duty Shredder, Proc. Qd. Soc. Sug. Cane Tech. Thirty-seventh Conf., p. 197.
9. Crawford, W. R., 1970. Cane Extraction by Milling, C. W. Murray Award-Winning Paper, 1970.
10. Buchanan, E. J., 1961. S.M.R.I. Quart. Bull., 19 and 20, 1961.

(*Note*: All references to Displaceability Index in this paper refer to Buchanan's method, using a 200 g sample of prepared cane, tumbled in 2 000 g water and measuring the pol extracted.)

APPENDIX A

Features of the Tongaat shredder

1. HAMMERS

1.1 There is more than 100% coverage of hammers across the shredder width (46 hammers, each 50 mm + hardfacing, across 2 100 mm shredder).

1.2 Hammers are of simple rectangular construction — no "club heads" required for width coverage.

1.3 Reasonable gap between the hammers reduces side wear due to friction between hammer flanks.

1.4 Each week, 2 of the 8 rows of hammers are reversed and two rows removed for rewelding. In a five week cycle, each set (two rows) of hammers thus has four weeks in service and one week being reconditioned.

1.5 Hammers need no bushes — very little wear on bore. (approximately 1,25 mm in direction of loading in hammers used for 800 000 tons cane.)

2. ROTOR

2.1 Extremely rigid construction, with 350 mm shaft (on 1 500 mm shredder) and no spacer discs.

2.2 Rotor elements (discs) are flogged up solid with large through-bolts, then locked together with "Ringfeeder" double-taper clamps, giving a solid assembly.

2.3 Having no spacer discs greatly reduces overall machining required during construction.

2.4 The profiled discs require less plate area than round discs, and the off-cuts are usable. The design also saves external machining of the discs.

2.5 The mass of construction renders a flywheel unnecessary. The inertia of the 1 500 mm shredder rotor with hammers is 2 060 kg m^2.

2.6 The construction allows for thick discs giving more than double the bearing area of conventional discs for carrying the hammer bars. This in turn allows for smaller diameter bars and consequently a stronger hammer section across the hole.

2.7 The hammer bars are secured by simple locking plates which lock the bars to prevent their turning in the discs. This virtually eliminates wear in the disc holes. Nevertheless, these holes may be fitted with replaceable bushes if desired.

2.8 The leading tip of the hammer can be deflected to 150 mm inside the normal extended hammer tip position to allow for tramp to pass through the machine. In this position the hammer is firmly supported at its tip, with no bending force tending to create abnormal stresses at the pivot point. (No signs of stress in any hammers after full season of use.)

2.9 The first critical speed of both the 1 300 and the 1 500 mm rotors is well over 2 000 rpm. (Even neglecting the stiffness of the solid disc construction, the 1 500 mm rotor has been calculated out for a $N_{crit_1} = 2\ 185$ rpm).

2.10 The 175 mm bearings (22 340 CK) fitted with a J7 fit into special solid cast steel housings afford ample load and shock-carrying capacity with low heat generation on a simple circulating oil system. The shaft diameter is increased to 250 mm inside the bearing housing.

3. GRID

3.1 The grid is fabricated completely from steel plate, and stress relieved.

3. Anvil bar and grid bars are identical and of a simple rectangular section with central dowel holes. Recommended material of construction is EN30B, heat treated. However, less costly materials are also being investigated.

3.3 Each grid bar comprises two 1 050 mm sections for ease of handling.

3.4 The rectangular construction allows for 4 wearing corners for each bar.

3.5 The fixing of the bars is by a wedge plate and dowels, which allows for quick changing and secure mounting.

3.6 The grid is completely adjustable (both for gap and sweep). The hinge point is adjustable in and out, and may be packed up and down.

3.7 The gap can be adjusted on the run simply by moving the lower lock-nut on the spring-toggle arrangement.

3.8 The entire grid plate can swing back to pass large items of tramp. The Tongaat 1 500 mm grid can move out 150 mm for this purpose, without any damage to adjusting mechanism, shearpins, etc. Together with the 150 mm hammer deflection, this would permit a total of 300 mm clearance for abnormal tramp.

3.9 The toggle arrangement allows for relatively light springs to locate the grid plate.

3.10 In the event of the grid plate swinging open to clear an abnormal load, the increasing return force of the spring is counteracted by the decreasing mechanical advantage of the toggle. Consequently, the grid plate is returned gently (not violently) to its correct position. (This has already been observed on several occasions.) The return force is in fact almost constant regardless of deflection.

3.11 The grid is effectively "self setting" — with wide, deep pockets formed by the bars, a cushion of compacted fibre builds up and the shredding actually takes place on this apparently self-setting layer (the Australian concept of shredding "cane on cane"). This also greatly reduces wear of the grid bars.

4. PRACTICAL RESULTS

4.1 Achieving an average 90% + D.I. (S.M.R.I. method) on 30 tons (metric) fibre/hour with 900 kW drive.

Knife power on this tandem:

	Installed kW	*Absorbed kW (approx.)*
Levellers	2×135	±120 (can run with 1 motor)
1st knives	335	±200
2nd knives	300	±180
		500

Overall absorbed kW/metric ton fibre = 45 (approx.)

4.2 A failure of the first knives motor led to these being removed for a period. Throughput decreased to 28 tons fibre/hour, and the average D.I. fell about 2%.

4.3 The above results have been achieved using only 4 of 6 grid bars, and 20 mm to 25 mm settings, since insufficient power is available to use all 6.

4.4 No hammer damage other than normal wear after 1 000 000 tons cane. (Still using original sets of hammers, see 1.4 above.)

4.5 Negligible wear on hammer bars. After 1 000 000 tons cane, original bars are down 0,1 mm to 0,24 mm on diameter where they pass through the rotor discs, and down 0,1 mm to 0,4 mm where the hammers are carried.

5. OTHER FEATURES

5.1 The grid van be opened up without disturbing the grid setting, simply by releasing the I-beam carrying the toggle fulcrums. This is done by slacking off the nuts on the two long bolts securing the beam. This feature has proved extremely useful in quickly clearing chokes caused by a faulty control tripping out tbe carrier from the shredder.

5.2 The entire grid is quickly and easily disconnected and removed by crane for checking and changing bars, etc. This also does not involve disturbing grid settings, so that repositioning is also quick (±30 minutes required).

5.3 The 2 100 mm × 1 500 mm shredder is designed to safely take a 1 800 kW drive at 1 200 rpm from one end.

5.4 The hammers are proportioned so as to have a natural period of oscillation on the hammer bar which ensures that each hammer enters the working quadrant in a state of maximum potential energy at the designed operating speed. This ensures optimum work and minimal oscillation of the hammers. (See Appendix B)

APPENDIX B

Hammer proportions

The oscillating motion of a shredder hammer about the rod on which it is carried approximates very closely to that of a compound pendulum, since the effects of the earth's gravitational pull, windage and the friction of the hammer on the rod are negligible (see Crawford[3]).

For the relatively small hammer swing angles experienced in practice, the period T_1 of one complete oscillation of a hammer subjected to centripetal acceleration at its pivot point is given (very nearly) by:—

$$T_1 = \frac{1}{n}\sqrt{\frac{k^2 + a^2}{Ra}} \text{ secs}, \quad \ldots\ldots\ldots \quad (1)$$

where n = rotational speed of the shredder, revs/sec.
k = radius of gyration of the hammer about its centre of gravity.
a = distance of the hammer c.g. from its pivot point.
R = radius of the pivot point from shredder axis.
(k, a, R in any consistent length units).

Pendulum motion is basically one of repeated conversions of potential energy (at one extreme of the swing) to kinetic energy (at mid-position) and back to potential energy (at the opposite extreme of the swing). One complete cycle of the oscillation of a shredder hammer can thus be analysed in four phases, each of equal duration, as follows:—

Let energy conditions which are capable of doing work (on the cane) in the direction of travel of the hammer in the shredder be positive, and opposite energy states be negative. Commencing with the hammer deflected fully backwards:

Phase 1: From backward extreme position to mid-point — energy state changes from maximum positive potential to maximum positive kinetic.

Phase 2: From mid-point to forward extreme — positive kinetic changes to negative potential.

Phase 3: From forward extreme to mid-point — negative potential changes to negative kinetic.

Phase 4: From mid-point to backward extreme — negative kinetic changes to positive potential.

By considering the swing cycle in this way it becomes clear that the hammer is best capable of doing work on the cane during phase 1, i.e. during the period when it is accelerating forward relative to the shredder rotor. It is therefore desirable to have the hammer enter the working zone (grid sector) during phase 1 of its swing. It can also be shown that the minimum total hammer movement on the pivot, and hence minimum rod and hammer bore wear, is achieved if the hammer enters the working zone at the beginning of phase 1, i.e. in the fully retracted position.

Conversely, minimum effectiveness and maximum wear will occur if the hammer enters the working zone during phase 3 — in fact, unstable harmonics can occur in this case, resulting in excessive wear and possible shredder imbalance.

To determine the conditions necessary for the hammers to enter the working zone at any particular stage of its oscillation cycle, it is necessary to make some realistic assumption about the hammer position at some fixed point of its revolution round the shredder. The best such assumption appears to be that it leaves the working zone (grid) fully retracted.

But this is exactly the position we desire at the *start* of the working zone. The hammer must therefore oscillate as a pendulum through one complete cycle (more than one cycle represents unnecessary wear) in the time between leaving the working zone and re-entering this zone.

This time $T_2 = \frac{360-\theta}{360} \times$ time for one shredder revolution,

$$\text{i.e. } T_2 = \frac{360-\theta}{360} \times \frac{1}{n} \qquad (2)$$

where θ = angle (°) subtended by the grid (working zone)

Putting $T_1 = T_2$, we get

$$\frac{1}{n}\sqrt{\frac{k^2+a^2}{Ra}} = \frac{1}{n}\,\frac{360-\theta}{360}$$

$$\text{or } \boxed{\frac{k^2 \times a^2}{Ra} = (1-\frac{\theta}{360})^2} \qquad (3)$$

This then is the condition that must be satisfied for hammer proportions to be optimum in terms of the joint criteria of maximum effectiveness and minimum wear. Note that this relationship is *independent* of the speed of the shredder.

For a 90° grid, equation (3) reduces to

$$\frac{k^2+a^2}{Ra} = 0{,}56 \qquad (4)$$

For the Tongaat 1 500 mm diameter shredder the value of $\frac{k^2+a^2}{Ra} = 0{,}55$. The minimal wear both on the hammer rods and in the bore of the hammers after handling over one million metric tons of cane of 15,3% fibre tends to point to the validity of the above theory in practice.