

**Development of a Large  
Capacity Haulout Bin**

**SRC Project DDI1S**

**DEVELOPMENT OF A LARGE  
CAPACITY HAULOUT BIN**

**SUGAR RESEARCH COUNCIL  
PROJECT DDI1S**

**FINAL REPORT**

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# 1 INTRODUCTION

This report is submitted to the Sugar Research Council as a summary and description of the haulout bin designs which have been generated under project DDI1S.

Progress with the project has been partly detailed in the Progress Report submitted in October, 1989, the paper on the measurement of hitch loads in haulout bins submitted to the 1990 Conference of the Australian Society of Sugar Cane Technologists, and the paper prepared for presentation to the November 1990 Conference on Agricultural Engineering. Copies of these two papers are attached as Appendix D.

The project rationale was based on an analysis by Connell and Borrel (BAE Occasional Paper 101, 1987) of the effects of haulout capacity on the economics of cane growing. Appendix A, taken from the Project Application, details this rationale.

The project was conceived as a design exercise aimed at applying the best of available knowledge and techniques to the task of designing a haulout bin or bins which would be attractive to the industry and act to encourage the changeover to larger and therefore more efficient haulout units. While recognising that the cane industry is particularly innovative in its development of equipment to meet local needs, it was suggested that from an industry wide perspective, there was a need for the promotion of properly designed and targeted haulout units, which would meet the industry's requirements as rationalisation of harvesting groups proceeded, and mill rail fleets changed to take advantage of the economies of scale.

The project has attempted to achieve a synthesis of the varying views and practises found throughout the industry during a period wherein the Project Engineer travelled throughout all the cane growing areas collecting information and assessing the perceived needs. Ultimately, this synthesis has to be expressed as hardware which has been configured to conform to the best suggested practise, within the limits imposed by cost, weight, overall dimensions and effects on the soil. This process has led the project staff to propose two related configurations which it is suggested will allow operators to upgrade their existing equipment in manageable steps, bearing in mind the inevitable changes which will be occurring in the industry during the next five years.

The project staff are also conscious that the industry can also hold very firm views on what is and what is not suitable for haulout duty. The two configurations described here represent our best judgement of the future needs of the industry and the best results of the analysis and design techniques available to us.

## 1.1 DESIGN PHILOSOPHY

The design philosophy employed with this project resulted from the industry survey performed and the initial charter of the project , which was:

1. To establish the most appropriate bin capacity and configuration to suit the wide range of mill bins in the industry.
2. To carry out proper engineering design and analysis of this bin to minimise its cost and maximise its effectiveness.
3. To minimise the adverse effects of this bin on the soil by proper selection and design of tyres and suspension.
4. To promote this bin to the cane industry, and to facilitate its adoption by means of licensing arrangements for its manufacture.

Through talking to industry representative it became clear that several factors should be kept uppermost in the design process.

**1. Cost:** The best haulout possible will not be acceptable to the industry if its cost is not within a reasonable range. There is an upper limit on the price of one of these bins, based on how much can be earned by a harvesting contractor in a season. Consequently, some of the otherwise acceptable bins currently on the market are outside the price range of most contractors. This inhibits their introduction into the market, with overall detriment to the industry. Any haulout proposed must therefore be designed with a primary objective of keeping the initial capital cost low.

**2. Simplicity:** Another perception within the industry is that haulouts, particularly the tipper/elevator, are too complicated. This discourages buyers who would prefer a simpler design (which would give the appearance of less to go wrong). Allied with this perception is a need for any design to look strong, because there is a healthy scepticism of any unit that gives the appearance of being underdesigned.

**3. Durability:** The need for low maintenance is a factor that often arises when discussing haulout units. Users of haulouts are looking for trouble free units which don't require continual maintenance or frequent replacement of wearing parts.

On the basis of the factors detailed above the design philosophy followed can be encapsulated as :

- 1. To use readily available common components in the design where possible. This will enable cheaper manufacture, and ease of replacement should the need arise.**
- 2. To reduce the number of wearing parts, which will improve the simplicity of the design. The design of the units should be as simple as possible without compromising the integrity or efficiency of the structure.**
- 3. To employ engineering design analysis methods to keep the structure weight as low as feasible, keeping in mind the need for the design to 'look strong' and to resist dynamic loads.**
- 4. To use as many components as possible which will be common to the two bin types.**
- 5. To allow for the possibility of assembling the bin "on farm" from a kit of components.**

## **2 INDUSTRY SURVEY**

A brief outline of the industry survey is made below. More specific industry responses have been included in the different design sections of the report.

### **2.1 METHOD**

The industry survey took place between January and June of 1989. During this time a number of trips were made, taking in the cane growing regions from Murwillumbah in Northern N.S.W to Mossman in Far North Queensland. In each region a member of the local Bureau of Sugar Experiment Station (or a similar body) helped out with local expertise and contacts. An effort was made to talk to a broad cross-section of the industry in each area.

From the millers, information concerning the effect of different types of haulout bins on wet-weather cane supply, bin weights, damage to mill rolling stock and their preferences was obtained as well as the likely future direction of the mill in terms of rail bin sizes. Harvesting contractors provided information on the performance of existing large capacity infield haulouts, necessary features to be incorporated in a design and their reservations about different design aspects. The farmers consulted with assisted the survey by volunteering information on wet-weather damage to crops resulting from different haulouts and their opinions on the available designs. Finally the local extension officers were able to assist with a broad overview of the regional needs.

### **2.2 CONCLUSIONS**

The industry survey revealed a broad diversity of opinions. It became obvious that no one design was going to satisfy the needs of all concerned. Opinions varied from region to region and then between millers, contractors and growers. The only common consensus that was reached was that in the current climate of rationalization any design should be as cost effective as possible.

The conclusions which were drawn from the industry survey are summarised in the sections which follow.

### 2.2.1 WHEELS/SUSPENSION

Most industry people recognize the benefits of high flotation tyres in wet weather situations. However where the region is not prone to high rainfall or where the mill tends to shut down in wet weather the need for high flotation is perceived as minimal. If continuous crushing becomes a more usual practise, then the need for high flotation equipment will increase. Conjecture exists within the industry as to what should actually be used as a high flotation tyre, with the main problem being that the traditional high flotation agricultural tyres are not durable enough in this application.

There is resistance in the industry to tandem axles. While it is recognized that legal constraints on axle loads often necessitate the use of tandem axles, their lack of manoeuvrability and perceived stool damage make their total industry acceptance unlikely. However, tandem suspensions are essential if bin loads exceed about eight tonnes. Springs in suspensions are considered an unnecessary addition, but are a requirement imposed by the Department of Main Roads.

### 2.2.2 TIPPERS/TIPPER-ELEVATORS/ROLL-ON-ROLL-OFF'S

A big factor in the resistance to the introduction of large capacity haulout bins is the initial capital outlay. Although the Roll-on/Roll-off's may be antiquated they are still cheap to buy and maintain. In areas where farms or groups are small, the Roll-on/Roll-off trailers may continue to be used, and be supplemented by "mud" trailers in wet weather. The necessarily small tyres are damaging to soil structure, particularly in wet conditions. From the millers' point of view, however, they are subsidising the growers by being forced to maintain their rolling stock, which is often damaged in field by haulout/harvester operators. Most millers agree that the general trend in the future will be toward larger rail bins that stay on the track, thus bringing about a need for large-capacity infields. Most people within the industry recognize that neither type of infield transporter is the ideal solution. A tipper bin will always be the cheaper alternative but is normally restricted to filling only one size of bin. For this reason most see a need for the tipper-elevator, which is more versatile in terms of partial filling of odd size bins, and also allows greater wet weather mobility by making it unnecessary to have a full infield bin to fill rail bins. There is also some concern about the lack of control of cane flow in unloading afforded by tippers.

### **2.2.3 STABILITY**

One of the more distressing faults of current haulouts in the industry is their marked lack of stability when performing infield haulout duties. This was identified by different people as an area that had real potential for improvement with some well thought out design. The reasons for instability are a combination of tyre properties, driver ability, bin weight distribution and weight transfer hitches geometry.

### **2.2.4 OPTIMUM BIN SIZE**

The quantity of cane which can be carried is restricted by allowable axle loads. While the ideal situation would be to carry 12 (or more) tonnes payload on a single axle this is not feasible within the axle load limits imposed by DMR. To achieve the higher capacities desired by a large portion of the industry, tandem axles are a necessity. Twelve tonnes appears to be an ideal weight because it allows a structure weight of about six tonnes, which should be adequate to ensure the durability of the structure. Twelve tonnes is a logical size because it allows filling of two lots of six or three lots of four tonne bins, which are at present the two more common bin sizes in the industry. The large rail bins currently being introduced in limited numbers have a capacity of ten tonnes, with a possible expansion in capacity of up to fifteen tonnes.

## **2.3 BIN DESIGN**

Following the survey of the industry and the development of the conclusions summarised above, the choice of bin capacities and configurations could be made and the design process could then proceed. Bin capacity and unloading method were the primary parameters, followed by the choice of tyres, axles and suspension method. Then followed the detailed design of the chassis, baskets and unloading arrangements as described in the following sections and the attached drawings.

### **2.3.1 TRAILED OR SELF PROPELLED**

Self propelled bins represent the ultimate bin configuration and are the most expensive bins available to the industry. They offer a specialised and dedicated unit which is more manoeuvrable and which may offer faster road speeds than the trailed units. Because the philosophy adopted in the current project is that the haulout bins to be proposed should have the minimal additional cost for the increment in capacity and efficiency achieved, both the configurations being proposed are trailed.

### **2.3.2 DRIVEN OR NON-DRIVEN**

The addition of drive to the trailer adds mechanical complication both in the transfer of power from the PTO to the axle and in the arrangement of the axles and suspension. In the interests of keeping the trailers as simple as possible the bins proposed are non-driven. The use of a front wheel assist tractor for haulout duty is fairly common, and provided that the trailer itself is sufficiently mobile, this lack of drive could only be a problem in the wettest districts.

### **2.3.3 BIN CAPACITIES**

In determining optimum bin capacities it was considered important to take into account future trends in mill bin size. It is apparent that six tonne bins will play an ongoing role in the industry, as evidenced by Bundaberg Sugar's commitment to the six tonne mill bin. Industry trends indicate that bins smaller than six tonnes will eventually be phased out and be replaced with larger size bins although this will take some time to happen, given the high capital cost of changing a bin fleet.

Without question the cheapest way to haul cane to six-tonne mill bins is with trailed double six-tonne tippers. In the regions where wet weather harvesting is infrequent these transporters seem to represent the optimum solution. Any design of a double six tonne bin could be adapted to a double four tonne bin, although with the move away from the four tonne bin operators in regions with the smaller bins would be better advised to move to a tipper-elevator in anticipation of larger mill bin sizes.

In wetter regions and where the mill is in the middle of changing its bin fleet over to a larger size there is a need for tipper-elevators. The optimum size for this bin, based on the evidence that we have, is twelve tonnes. A twelve tonne bin will fill three four tonnes or two six tonne bins, which makes it more flexible than an eight tonne bin which may be continually part-filling bins.

It is a necessary fact that a twelve tonne bin of any description has to be fitted with tandem axles to comply with the road regulations. This tends to alienate a significant proportion of potential buyers whose dislike of tandem axles has been noted previously.

Based on the evidence that had been gathered, and bearing in mind the constraints imposed by Main Roads regulations and the geometry of the overall system, two bin configurations have been proposed and designed:

- 1. A twelve tonne capacity tipper elevator trailed bin**
- 2. A double six tonne side tipper trailed bin.**

### **3 CHOICE AND DESIGN OF BIN COMPONENTS**

There is a number of components whose choice or design is common to both the bin configurations being proposed. These are outlined in this section before the components particular to each configuration are discussed.

#### **3.1 AXLES AND SUSPENSION ARRANGEMENT**

The general attitude of the sugar cane industry towards suspension under haulout equipment is that a solid mount suspension is entirely up to the job. During the industry survey, not one haulout unit was found that incorporated a functional spring/damper system. Any springs which were incorporated into units were there for the express purpose of satisfying MRD regulations that call on tandem units to have damping springs to suppress 'axle-hop'. Where truck suspension systems had been used, the spring component had invariably been replaced with a solid member.

The haulouts using commercially available suspension systems generally retain the rubber bushing components. The general impression was gained that units without some form of rubber bushing incorporated were more prone to wear components quickly.

There were three reasons for the lack of spring/damper suspension systems:

- (1) As a rule these units run on agricultural tyres, which tend to provide a damping effect on the unit.
- (2) It is perceived that any springs introduced into the system will only add to the problem of instability: (a) during travel and (b) during unloading, particularly for side tippers.
- (3) The clearance necessary to incorporate such a system is detrimental to the load carrying capacity of the unit because of the increased intrusion of the wheel arches into the basket.

Conventional bin suspensions are typically based on truck suspension components, and use through axles whether or not drive is required. Such arrangements add substantial weight to the bin, raise the bin chassis, reduce basket height, and reduce bin stability by raising its centre of mass when loaded. By removing the option of drive it is possible to specify a tandem suspension based on a centre pivoted balance or walking beam with stub axles. This allows the chassis to be located at minimum ground clearance and for basket internal height to be maximised. Bin tare weight is also reduced.

In extreme cases during infield use and particularly during the unloading of side tipper bins this suspension beam and its supporting pivot structure will be highly loaded, but it has proved possible to analyse and design a suitable beam and pivot arrangement. Rubber or polyurethane bushes as used in truck suspensions have been specified for the beam pivot because of their reported long service life, beneficial effects on bin ride and ready availability. The suspension structure is applicable to both types of bin, and its design is based on the possibility that all the bin weight might at some time be transferred to one side of the unit - i.e be carried by one set of wheels. To overcome the lack of space afforded between the chassis and tyres the pivot structure was placed between the two wheels, with the rubber bushes fitted at either end of the pivoting shaft. This arrangement reduces the forces on the bushes to values equivalent to those found in truck service. The pivot arrangement can be detached by the removal of two retaining caps.

## **3.2 TYRE SELECTION**

The selection of suitable tyres for the two bin configurations has been based on a number of factors which include industry perceptions and experience and an engineering analysis.

### **3.2.1 INDUSTRY PERCEPTIONS**

**3.2.1.1 Tyre Wear:** If the operating surface was bitumen rather than soil then the best tyre would be one with the highest possible ply rating and wall stiffness. Unfortunately softer tyres suitable for low compaction seem to be more prone to wear. For this reason a lot of contractors have turned to earth-moving tyres, which while being detrimental to the soil last for a long period of time. The tendency for the softer tyres to wear out quickly is accentuated in tandem axle setup, where the scrubbing effect makes tyres wear at a greater rate. Tractor tyres while not being as indestructible as earthmoving tyres seem to last longer than the low-compaction "ground-hog" tyres.

Another associated problem being experienced in the field is the tendency for the tyres to blow off around the rim, especially with the tractor tyres which are not designed as a rule for the weights they are carrying. Also the rims are needing to be heavily reinforced to cope with the loads. "Olympic" have recently introduced a tyre into the industry which is 14 ply and doubled beaded in an attempt to address these problems. One can notice from **Table 1** that some of the tyres under consideration are not made to take the loads required for this application (4.2 tonnes based on permissible axle loadings).

**3.2.1.2 Compaction/stool damage:** Both compaction and stool damage are critical factors in tyre selection. Stool damage is the more obvious of the two, with the effects of compaction being hard to quantify. With the earthmoving tyre a "square stool" syndrome has been identified i.e the tyres significantly effected the shape of the stool. This effect is not as pronounced with other 'softer' types of tyres. The width of any tyre must be such that it will fit comfortably in the interow. From this point of view the "Groundhogs" are thought to be of some advantage because of they are shaped much like the interow space.

**3.2.1.3 Rutting:** Rutting of the soil in the wet seems a fairly predictable process, but in the course of talking to different operators in the industry an interesting observation was made. The soils ruts made by the haulout unit are dependent largely on the type of harvesting gear used. If the soil is wet enough a wheeled harvester will create a rut which subsequent haulouts will only worsen, dependent on the tyre size of the haulout. However if a full track harvester is used it is not likely to cause that same initial rut, therefore lessening the likelihood of the haulout causing its own rut. It also appears that a driven haulout tyre is more likely to cause a rut than a non driven tyre.

**3.2.1.4 Haulout stability:** Another important factor in tyre selection that often escapes attention is their contribution to unit stability. Tyres with soft sidewalls, will by nature cause a degree of "wallowing" to be introduced into the bins motion. While a tyre with stiffer sidewalls is less likely to do this, the inherent suspension ability of softer tyres is lacking. The tyre profile also affects unit stability.

**3.2.1.5 Trash blanket Effect:** With the advent of green cane harvesting and trash blankets, some contractors have found that the smaller tyres tend to bulldoze the trash rather than ride over it.

**3.2.1.6 Geographical considerations:** Tyre needs are highly dependent on the area involved. In wet regions harvesting would not continue without the use of high flotation tyres. However, in the drier regions, it is harder to persuade operators that a high flotation tyre is needed.

### **3.2.2 DESIGN CONSIDERATIONS**

**3.2.2.1 Size:** The diameter of the wheel is a factor which has a major effect on any proposed design. A large diameter wheel restricts the size of the bin that can be fitted between the bin chassis and the harvester elevator. The section width of the tire is constrained by the spacing between the rows. Ross Ridge (B.S.E.S. Bundaberg) has noted that a 23.1 \* 26 tyre is about as wide (587 mm) as one should go before stool damage starts to be a problem.

TYRE	DIAMET. mm	WIDTH mm	LOAD tonnes	PRESS. kPa	PLY	CRR
14 x 17.5	907	361	2.77	700	14	0.398
9 x 20	1019	259	4.56	700	14	0.247
38 x 20 x 16.1	993	498	2.3	275	10	0.286
18 x 19.5	1096	457	4.39	550	18	0.274
23.1 x 26	1605	587	3.8	180	12	0.161
23.1 x 34	1808	587	3.7	140	10	0.144
22.5 x 26	1621	551	5.44	175	10	0.166
23.1 x 30	1707	587	4.04	180	12	0.152
20.0 x 20	1240	483	4.13	250	14	0.223
18.4 x 26	1394	467	4.36	280	12	0.214
23.1 x 26	1548	587	4.63	140	8	0.166
500/60 - 22.5	1170	500	4.2	253	8	0.242
600/55 - 26.5	1350	600	4.19	140	8	0.186
400/55 - 22.5	1000	400	4.23	550	16	0.333
600/55 - 22.5	1350	600	4.2	250	14	0.186
600/60 - 26.5	1495	600	4.2	228	14	0.169
500/60 - 38	1550	500	4.2	278	14	0.185
600/65 - 34	1650	600	4.2	190	12	0.155

TABLE 1

**3.2.2.2 Rolling Resistance:** An important factor in our considerations is the rolling resistance of any particular tyre, which gives us an estimate of the mobility of the tractor/bin combination. A useful measure of rolling resistance is the co-efficient of rolling resistance as defined by Gee-Clough (1980) and expanded upon by McAllister (1983). It is the rolling resistance force expressed as a fraction of the load on the tyre. The lower the rolling resistance of the tire the more chance the bin has of being towable in marginal soil conditions. Values for Crr are included in Table 1 for a wide range of tyres and it is obvious that the larger diameter tyres are preferable.

**3.2.2.3 Ground pressure exerted:** To minimize soil compaction and its detrimental effects the ground pressure exerted by the tyre should be minimised. Taking the approach of Plackett (1984), we say that the ground pressure is equal to the inflation pressure plus the carcass pressure of the tyre (which is a function of the ply rating and wall stiffness of the tyre.)

It has been shown that the majority of compaction done in agricultural soils is effected on the first pass. With this in mind, the ideal situation for the bin tyres is for them to exert less ground pressure than the harvester/tractor tyres (ie. those tires that effect the first pass). If the bin tyres are exerting a higher ground pressure than the tractor/harvester tyres then further unnecessary compaction will be taking place. Harvester/Tractor tires are often 23.1 \* 26 's which have an inflation pressure of around 180 kPa (maximum recommended for a 6.7 tonne rear axle load) and an estimated carcass pressure of 30 kPa (pg.163 Plackett)

giving a total ground pressure of 210 kPa. The ideal bin tyre then should have inflation pressure less than 180 kPa and a relatively low carcass stiffness.

It has previously been proposed that the mean ground contact pressure of haulout tyres should be kept below 100kPa to avoid compaction of the soil. Given the loads to be carried and the available tyre capacities, it is not possible to achieve this. Even tracks are not necessarily the answer, because of pressure peaks below the track rollers, and the fact that deep compaction is a function of axle load only, not contact pressure.

**3.2.2.4 Economics:** It goes without saying that a tyre that meets all of the above requirements may well be economically unrealistic. One important factor which will have to be considered is whether it is worth the extra cost of a superior tyre to ensure harvesting for every day of the season, or if it is more economic to settle for a cheaper tire with less mobility and stop harvesting on the days in the season when the ground is too wet. A high flotation tire is undoubtedly justified in the Tully region but may not be in the drier Isis region.

On the basis of these considerations there appear to be three types of tyre available for use.

- 1). 23.1 x 26 12 ply tractor tyre
- 2). 22.5 x 26 10 ply implement tyre.
- 3). 600/55-26.5 "Trelleborg Twin" low profile trailer tyre.

Of these three that meet the criteria, the "Twin" tyre probably meets the requirements best. It is a specifically designed trailer tyre, which easily meets the load criteria, is designed with long service life in mind, runs at a low ground pressure with low carcass stiffness, but also has a low profile which negates to some degree the possible instability due to soft sidewalls. The "groundhog" implement tyre would seem to have a lot of advantages, but from viewing these tyres in operation in the industry there seem to be problems with firstly the life span (they wear quickly in a tandem setup) and secondly that they appear to induce bin instability.

Although not ideal, the tractor tyres are readily available and meet most of our criteria. The normal tractor tyre is overloaded in this application, but with the development of a slightly more rugged variant of this tyre (eg Olympic's 14 ply double beaded model), they should stand up to the application well. They are longer wearing than the groundhogs and less likely to contribute trailer instability. On the basis of their cheaper price (approx. \$1100 5/90) compared with the Trelleborg tyre (\$1358 11/89) and more ready availability (the Trelleborg tyres must be shipped in from overseas) the tractor tyres are being specified in our design. However it is strongly recommended that the Trelleborg tyres be trialled in a design for it

may be well be worth the extra cost in terms of long-life and the extra bin space that would be gained.

### 3.3 HITCH ARRANGEMENT

In the interests of maintaining as many components in common as possible for the two bin configurations, the same hitch arrangement is being proposed for both. There is a wide variety of hitch types and arrangements being used on current haulout bins, and the industry survey found many examples.

Earlier side tipper bins used a simple hitch which connected to a tractor drawbar, which was connected to the tractor underneath the transmission housing, and supported vertically behind the rear of the tractor PTO housing. These simple hitches allowed motion about the three axes (turn, pitch and roll) and transferred weight from the front axle of the tractor to the rear axle, thereby increasing traction. Based on simple calculations of the statics of the tractor/bin combination, between one and two tonnes of bin weight were placed on the tractor drawbar.

When bins carry twelve tonnes of cane and the tandem suspension has a legal capacity of 2x8.4 tonnes, then bin tare weight in excess of 4.8 tonnes has to be carried by the tractor. Conventional bins of this capacity use what is known as a "weight transfer" hitch, which has a pitch axis underneath and slightly forward of the tractor axle, and the roll and turn axes behind the tractor. The term "weight transfer" is, strictly speaking incorrect, because no weight is transferred from the front axle of the tractor to the rear and the front axle is more heavily loaded by the bin mass. This type of hitch places the majority of the bin excess weight on the rear axle, and only a fraction of a tonne extra on the front axle.

A qualitative analysis of hitch function will show that the if the turn axis is forward of the components of the roll axis then when the bin is cornering it will drop the inside front corner and may become unmanageable. If the pull line of the hitch is too high then the tractor and the bin will roll excessively when cornering. Examples of these faults were found during the industry survey.

The "weight transfer" hitches being used in the industry appear to be both overdesigned and underdesigned. Hitch loads have been measured (Pearce, Harris and Bennett 1990) to establish their magnitude and investigate any possible frequency dependence. Horizontal and vertical forces at the pitch axis (underneath the rear axle) were found to have no frequency dependence and to be of the same magnitude as might be calculated from statics for the vertical forces and from estimates of rolling resistance for the horizontal forces. Dynamic effects can increase these values by a factor of about two.

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The design of a suitable hitch is constrained by the need to keep the hitch as small and as light as is possible while allowing for the static and dynamic loads which are imposed on it.

The measurements of hitch loads in the field throughout the haulout cycle have enabled us to specify design loads for the hitch.

If the hitch is seen as similar to a universal joint, then it is desirable to keep the bearings on each axis as close together as is possible. In order to resist the imposed shear force and bending moment (taken as 35kN and 35 kNm respectively) the bearings have to be sized to allow for the loads in the shafts or pins. An idea of the magnitude of these loadings may be gained from the fact that a PTO drive supplying 105kW at 1000rpm is transmitting only 1kNm of torque (which is equivalent to bending moment) and no shear load.

Using plain spherical bearings, the design which eventuates has bearings with a load capacity ten times that necessary, with shafts 150mm in diameter. Obviously, a compromise has to be made, and the hitch layout specified in the drawings represent our best conclusions. The RHS members external to the hitch carry the bending moment, so that the bearing pins are only loaded in shear. The front of the hitch is a cross member which it is intended will attach to the longitudinal beams which pivot under the tractor rear axle. The rear of the hitch is a clevis which will attach to the bin drawbar. Both these attachments and their positioning depend on the particular tractor being used, and are therefore not specified further.

Slewing rings were also considered as a possible basis for a hitch, as has been done in a number of current designs. While these rings can provide an adequate moment and shear capacity within a sufficiently small envelope, they are not stocked in numbers in Australia, and are relatively expensive. In contrast, the four plain spherical bearings specified are readily available and cost relatively little.

The central cross component which mounts the four bearings has not been subjected to a full stress analysis, because of time limitations and computer hardware failures. It is recommended that this should be done, possibly within the scope of the new project "Hitch Design and Dynamic Stability of Infield Haulouts", project number USQ1S. Hitch reliability is very relevant to the question of operator safety.

### **3.4 CHASSIS ARRANGEMENT**

The bin configurations which have been chosen are intended to provide the maximum possible increment in operating efficiency for operators wishing to upgrade their haulout capacity. To achieve a legal capacity of twelve tonnes within the constraints imposed by axle loadings, weight transfer to the tractor rear axle, and overall dimensions of the bin, it is necessary to reduce the tare weight as much as possible, both by choice of the configuration and by careful analysis of the stresses.

A simple analysis of the operating cycle of a haulout bin will show that it is in the unloading phase that the bin and the chassis are most highly stressed. In this phase the loads on the chassis are concentrated at pivot points rather than being distributed along the chassis. Consequently some proportion of the tare weight of the bin results from components or load capacity which are only necessary while unloading, and this proportion of the tare weight is higher for tipper/elevator bins than for side tippers because of the two elevators being carried and the limitations inherent in the placement of rams and pivots. The type of suspension being used, the provision of drive to the bin wheels, and the number of wheels will also influence the tare weight.

The layout of the bin chassis for each configuration has therefore been given careful consideration because it is a major component which is highly stressed only during unloading. The initial designs were based on a single central tubular member, because of the perceived advantages of simplicity and commonality. However, as analysis proceeded, it was necessary to revert to a more conventional layout incorporating parallel main members. The concentrated loadings resulting from bin tipping were calculated from computer simulations of the unloading cycles, and a finite element package was then used to design the structure and optimise its layout.

### **3.5 UNLOADING AND CONTROL OF CANE FLOW**

Conventional side tipper bins unload by being rotated through approximately 140 degrees about a pivot located along an upper edge. This is achieved by a ram placed at each end of the bin. Cane flow into the rail bin is controlled only by the skill of the operator, and can be quite discontinuous. Fold out chutes and bin lips are used to assist the pouring of the cane into the rail bins with minimal loss.

Tipper/ elevator bins are tipped forward about a pivot placed towards the top of the front wall onto a primary elevator which forms the front wall of the bin. This elevator is used to lift and control the flow of the cane onto a cross elevator which folds out to extend sideways from the bin and above the rail bin. The cross elevator then controls the flow of cane into the rail bin. In most current tipper elevator bins the tipping pivot is lifted either by slides or a rotating pivot frame so that varying rail bin heights can be accommodated.

The two configurations will now be discussed separately to show how their particular unloading arrangements, chassis design and basket design have been derived, these being the functions and components which are not common to both.

## 7 SUMMARY

The attached drawings summarise the general layouts and dimensions of the two bin configurations, with some details of particular components. Because some details or manufacturing procedures will be particular to a particular manufacturer or haulout tractor, no attempt has been made to provide a complete set of working drawings. The same approach has been taken to the provision of hydraulic tanks, pumps lines and valves, which are left to the discretion of the manufacturer. However, it was noted during the survey of the industry that PTO mounted pumps provided a neat and convenient solution to the provision of hydraulic services.

The chassis and basket details are sufficiently complete to allow reasonably accurate estimates of unit weight and provide a basis for estimating costs. The proposed powered roller and variable geometry floor of the tipper/elevator bin have been fully detailed because their satisfactory functioning is dependent on the solution of the inherent design problems.

The suspension beam and its pivot are highly loaded components, requiring a full analysis and detailing to ensure that when manufactured they will function properly. Braking arrangements and the provision of springs have been omitted because each manufacturer has particular solutions to these problems.

The tare weights calculated on the basis of the design details are lower than those previously achieved for similar units. The double side tipper has identical basket structures symmetrically placed about the suspension, which loads the trailer axles to their limit and allows for about 0.5 tonnes to be transferred to the tractor rear axle. This allows for ballasting of the tractor to achieve adequate traction

The tipper elevator bin has less than the maximum load on the suspension, and transfers about three tonnes to the tractor rear axle. This is of the same order of the loads being transferred by current designs, but the trailer axle loads are less.

The hitch design proposed is adequate for the forces involved, using a simplified analysis of the stresses. However, it is recommended that a full finite element analysis be carried out to ensure that it will function as desired.

## **4 TIPPER/ELEVATOR BIN**

### **4.1 UNLOADING ARRANGEMENT**

The primary elevator in current tipper/elevator bins is part of the inclined front floor of the bin. This inclination of the front floor reduces the angle through which the bin must be tipped, because the primary elevator has some capacity for moving cane upwards and thereby controlling its flow onto the cross elevator. The inclined front floor also reduces the volume of cane which can be carried in the bin, thereby making the bin longer than the equivalent rectangular bin.

This elevator also introduces substantial extra weight and complication to the bin, which increases its tare weight and cost. We have therefore sought alternative configurations which would provide the same functions without the complications.

The inclined front wall of existing designs reduces the angle through which the bin must be tipped before the primary elevator is able to lift the cane onto the secondary elevator. Some earlier designs provided for the bin to tip cane directly onto the secondary elevator, but this involves long rams and the unnerving placement of the tipped bin above the tractor cab. The final 50 degrees of tipping angle is only necessary for unloading the cane which is trapped in the front corner of the bin. It was therefore reasoned that if the geometry of the front wall of the bin could be altered on demand, then the tipping angle could be reduced, and the primary elevator could be eliminated.

We are therefore proposing in the present design that the bin should be rectangular and that the primary elevator should be replaced by a variable geometry front wall and floor which move in the latter stages of the tip cycle to allow the cane in the front of the bin to tip into the cross elevator. This wall and floor have been designed and detailed to function as desired.

This variable geometry floor has four functions. It replaces the inclined front floor associated with conventional primary elevators, and thereby increases the volume in the bin for a given overall length. It is lighter than the primary elevator, it reduces the cost of the bin, and it reduces the necessary tipping angle by about 60 degrees. The floor has two hinges and its leading edge is supported by sections of elevator chain running on rails which are part of the basket structure. It is actuated when the basket has reached its maximum tip angle and enables the last two or three tonnes of cane to be unloaded in a controlled fashion.

The second function of the conventional primary elevator is that it controls the flow of cane onto the cross elevator. Along with the variable geometry

wall and floor, we are proposing that the flow of cane should be controlled by a powered roller coaxial with the bin pivot, which is an adaptation of the method also proposed to be used for cane flow control in the side tipper bin. This is a method used for side tipper bins by a Bundaberg contractor, Doug Young. It is very effective in retaining cane and releasing it in a controlled flow as desired for topping up bins, and has been in use for many years.

This combination of a variable geometry front wall and the powered roller permits bin tipping angles (and therefore ram stroke) to be only slightly greater than those of conventional tipper/elevator bins, but without the weight and complication of a primary elevator.

Existing tipper/elevator bins have a variety of tipping ram locations, ranging from a single vertical ram at the rear to one ram along each side to a horizontal ram underneath the bin. Whatever the arrangement, the ram forces on the chassis and the bin pivots create stresses in the chassis which exceed those generated during the transport phase. More recent bins also provide for the bin pivot to be elevated at the beginning of the unloading phase so that the unfolding secondary or cross elevator will clear the side of the rail bin. This is achieved by sliding the pivot upwards with separate rams, or by rotating upwards a frame to which the pivots are attached, again using separate rams.

Any ram arrangement which has an other than vertical ram will exert a horizontal force on the bin pivots. This force adds to the bending moment in the chassis caused by the weight of the cane being supported by the pivots, so that the chassis has to be strengthened to a greater degree.

For the present bin, a tipping arrangement has been devised consisting of a ram on each side of the bin which operates a tipping linkage. This arrangement has been optimised using a computer simulation so that the stresses in the chassis during the tipping cycle are minimised. The initial lift of the bin pivot is also achieved with these tipping rams, and the necessary ram stroke has been reduced by using the linkage. The inner width of the basket is reduced by having the tipping rams along the sides, but the vertical rather than inclined front wall more than compensates for this loss. Because the tipping ram remains essentially vertical throughout the tipping cycle, the forces on the bin pivots are minimised.

## **4.2 PIVOT REGION**

In the pivot region of the tipper/elevator bin are located the unloading roller with its supports and drive, the bin pivot with its elevating slide and secondary elevator support, and the top hinge of the variable geometry floor.

The unloading roller is shown with five axial lugs, based on the forms already being used in side tipper bins. Experience may show that this is not the most appropriate geometry.

The unloading roller is supported at each end by the motor bearings, which have an adequate capacity. It may be that only one motor will be necessary to drive the roller, in which case the other may be replaced by a suitably arranged bearing and housing.

The drive motors are housed within the ends of the roller and can be installed from the outside of the bin. The splined motor shaft connects with and supports the roller.

The pivot region of the bin structure is formed by a plate to which is attached the bin structure side members, the mounting for the roller drive motors, the bin pivot structure and the top hinge of the variable geometry floor.

The bin pivot lugs are attached by bolts so that the bin structure may be easily separated from the chassis for servicing of the pivot area.

The bin pivots on a slider assembly which allows the tipping rams to lift the pivot point and the secondary elevator by up to 600mm before tipping commences. The secondary elevator will be mounted on the lugs attached to the pivot slide, and as noted previously has not been detailed in this work.

#### **4.3 VARIABLE GEOMETRY FLOOR**

The variable geometry floor is in two parts, connected by a simple hinge, and is operated by two small rams supported on a cross member at the front of the bin. The leading edge of the floor is supported by roller chain links running on wear strips on rails rather than by some sliding arrangement. The floor has a 10mm clearance on each side to the inside of the bin. The floor will be operated in the later stages of the tipping cycle to empty the remaining cane from the bin. With the bin tipped to 80 degrees, the front floor angle will be 45 degrees to the horizontal.

#### **4.4 TIPPING RAM AND LINKAGES**

The tipping rams have not been detailed. They will be telescopic, with a closed length of 1100mm and a stroke of 2600mm. Their maximum diameter is 190mm and the ram force required is 11 tonnes. Details of the ram base lugs and their attachment to the chassis outriggers have been given.

The two linkage arms require on either side of the bin have been detailed, showing the lug arrangement at either end.

#### **4.5 SECONDARY OR CROSS ELEVATOR**

The design of a secondary elevator is a problem of function rather than analysis. The secondary elevators being used in the industry are quite similar, and it was felt that the project could not offer any improvements, and should instead concentrate its efforts on other aspects. The general specification is that the secondary elevator should have a width of approximately 1.4 metres.

#### **4.6 BASKET DESIGN**

Baskets are conventionally constructed from steel sheet and mesh supported by a RHS framework. The basket for the two bins described here followed this convention, but have been analysed using the finite element software, and designed so as to minimise their tare weight. Loadings were calculated for the transport phase, the initial tipping phase and the tipping configuration calculated to just initiate cane flow.

Each side of the tipper/elevator basket is supported at the bin pivot and the upper ram pivot. There is also a large internal tensile force between these two points because of the forces developed in the tipping linkage. A basket structure has been devised which allows for these forces and transfers them to the rest of the structure. A number of basket structures have been examined so that the resulting stresses and deflections are minimised. Calculated maximum deflections occur in the base of the basket and at the rear where rotations are a maximum.

The basket has been narrowed in the front portion to allow more clearance for the tipping rams and linkage, and lengthened by 200mm overall to compensate for this loss of volume. The tipping pivot area and the lift pivot area have been reinforced with a backing plate to distribute the concentrated forces in these areas. An attached table lists the members, their sizes and their mass. (Appendix C)

## **5 DOUBLE SIX TONNE SIDE TIPPER BIN**

### **5.1 UNLOADING ARRANGEMENT**

The unloading arrangement proposed for the double six tonne tipper uses trunnion mounted rams at each end of the baskets, in accordance with common practise. These rams achieve the required rotation with the minimum of stroke and fit neatly down the front and rear of each bin.

Control of cane flow from each basket is achieved using the powered roller technique noted previously . Each basket is divided in two longitudinally by a near vertical wall, at the top of which is a powered roller. When the bin tips, the cane in the upper compartment is retained by the roller, and will not empty until the roller is activated. The tipping sequence has been simulated so that ram loads, basket loads and chassis stresses can be calculated.

The basket pivots and the ram pivots are supported on identical frames attached to the chassis. These frames have been optimised for geometry and member size using the finite element software.

### **5.2 BASKET DESIGN**

It works out conveniently that the two baskets are the same shape and size. The basket structure has been analysed using the finite element software, and checked for loadings in the transport, initial lift and tipping phases. The arrangement of the framework and the member sizes has been chosen to optimise the stresses and minimise the deflections. It has been found possible to minimise the weight of the structure by ensuring that stresses are approximately equal in all the members. A table is provided listing the basket members, their sizes and their mass.

### **5.3 TIPPING RAMS**

The tipping rams are trunnion mounted in pivot housings with bolted caps, to allow them to be assembled and removed. The ram and bin pivot support structure has been simplified to achieve the necessary functions with minimum weight and adequate strength.

## **5.4 UNLOADING CONTROL ROLLER**

The unloading control roller is a development of the design currently being used. It assembles into its support bearings by being inserted diagonally, and will be driven by a suitable motor attached by a coupling to its longer end.

## 6 TARE WEIGHTS

Estimated masses of the bin components are listed below, together with an estimate of the weight transfer to the tractor and the bin axle loadings.

### 6.1 DOUBLE SIDE TIPPER

Chassis	1500
Wheels and rims	800
Axles	160
Front basket	580
Rear Basket	580
Rams	400
Hydraulics	300
Hitch	250
Suspension	500
Rollers	200

**Total**            **5270** or a weight of 5.27 tonnes  
 Axle load total is 16.8 tonnes, weight transfer to tractor  
 approximately 0.5 tonnes.

### 6.2 TIPPER / ELEVATOR

Chassis	1500
Suspension	500
Axles	160
Wheels and Rims	800
Basket	1010
Rams	500
Hydraulics	300
Front Roller	250
Front Floor	220
Hitch	250
Primary Elevator	1000

**Total**            **6490** or a weight of 6.49 tonnes  
 Axle load total is 15.5 tonnes, weight transfer to tractor  
 approximately 3.0 tonnes.

## APPENDIX A PROJECT RATIONALE

In 1987 Connell and Borrell published BAE Occasional Paper 101 entitled "Costs and Regulation of Cane Harvesting Practices", which recognised that cane harvesting and transport is the single most costly operation in sugar cane farming (representing about 27 percent of total grower costs), and considered the possibility of reducing harvesting costs by improving the utilisation of available cane harvesters. Currently, the average cane harvester in Queensland cuts about 15000 tonnes of cane per year while it has an annual capacity on a one shift basis of 45000 tonnes. On a two shift basis, throughput could be 80000 tonnes, and harvesting costs would be reduced by 42 percent.

The economic model used in their analysis assumed that changes could be made in the regulatory structures to allow much larger harvesting groups and so improve the utilisation of harvesters. It was noted that the current harvesters are capable of annual rates of 121000 tonnes using an extended season, two shifts per day and a 13 day fortnight. Harvesting of cane also involves haulout units which transport the billets of cane from the field to the tramway siding or other delivery point. The majority (about 75 percent) of haulout units are of the roll on/roll off type, consisting of a simple trailer with rails to transfer the mill bins to and from the siding. About 15 percent of units are of the tipper or elevator type, with the balance being trucks which transport the cane directly to the mill. Harvesting costs are therefore also a function of haulout arrangements and haulout distance. The larger the haulout bins, and the more there are of them, the longer the haulout distance can be before the harvesting rate is limited by the haulout cycle time rather than the harvester capacity. The larger the capacity of the haulout units the greater is the possible annual throughput. If proper advantage is to be taken of the potential capacity of existing harvesters, then it is essential that the capacity of haulout units be increased. Roll on/roll off trailers are typically limited to using small high pressure tyres because the bins have to be rolled on and off over the top of the tyres, and so these trailers exert large ground contact pressures and have limited mobility in wet conditions. There is also evidence that the constant handling of the bins damages the bin structure as well as the wheel bearings. The model used by Connell and Borrell allowed for a maximum of three 8 tonne haulout units (two 4 tonne bins on a trailer), this assumption being based on industry practice.

Over the last fifteen years alternative haulout means have been developed in an attempt to overcome the limitations of the roll-on/roll-off's types. Increasing recognition of the damage being done to soil structure by the small high pressure tyres has encouraged the adoption of large diameter low pressure tyres, which offer much lower ground contact pressures. These bins are towed by conventional tractors, sometimes with drive to the trailer wheels. Configurations have included side tipping, double side

tipping, forward tipping onto a cross elevator, split tipping onto a central elevator, bins with and without drive, self propelled bins, and many other forms. Capacities have been limited to about 8 tonnes by axle loadings and gross trailer weights, although self propelled and trailing bins of a nominal 12 tonne capacity have been built. In Northern New South Wales, which harvests only 5 percent of Australia's sugar cane, the side tipper types have been industry standard for fifteen years, because there is no tramway system, and mill transport is by large truck-mounted containers. These innovations in the design of haulout units have typically improved mobility as well as decreased compaction and field damage. occurred in the smaller workshops involved in one facet or another of the cane industry. These workshops have had the ideas and the skills to be able to produce haulout bins which work and represent solutions to some of the problems inherent in the simpler trailers. However, these workshops have not had and do not have the access to the engineering and design skills and techniques which would enable them to address the concept of a haulout bin of large capacity applicable to all cane growing areas of the State.

A further development which will affect the design of haulout units is that many mills are also re-evaluating their mill transport arrangements, and moving towards bins of larger capacity. This is a change which will continue over the next five or ten years, and presents a further challenge to the designers of haulout equipment intended to be easily adaptable to all mill areas.

With the current moves towards de-regulation and rationalisation into more economic groups within the Queensland cane industry, there is substantial current grower interest in haulout units of 12 tonne capacity. This interest is understandable in the light of Connell and Borrell's analysis. As amalgamation proceeds, the traditional roll on/roll off trailers should be replaced by more efficient units which do not damage the soils, have much better mobility under wet conditions, are more manoeuvrable and are flexible enough in their configuration to be used with the wide variety of mill transport which exists in the industry. There is a real need for the development of large capacity haulout units which respond to this general specification.

To achieve a 12 tonne capacity in units which are of legal width, low enough to fit under harvester elevators, short enough to be easily manoeuvrable, which have a track and tyre width to suit row spacings and which do not exceed wheel and axle load limits, is a daunting proposition. The bin itself is a complex structure, the chassis is highly stressed in the unloading phase, analysis of hitch loadings and trailer stability is difficult, and the effect of such a unit on the soil is unknown. A substantial engineering effort in the analysis and design of these large capacity haulouts is necessary to allow them to be more efficient in operation, to reduce their tare weight, to reduce their cost, to make them more reliable, to minimise their effects on the soil and to promote them to the industry as

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a means of making harvesting more efficient. This application, then, seeks funding to allow a properly arranged engineering effort to be focused on the economics, analysis, design and development of large capacity haulout bins, which are so arranged as to be suitable for unloading into the large and diverse range of mill bins in use in Queensland. The need for this type of bin will increase as the present population of roll on/roll off trailers is phased out by the rationalisation of group sizes and harvesting operations, and the steadily increasing recognition of the damage being done to productive soils by inappropriate wheels and suspensions. It would be of benefit to the industry as a whole that there should be haulouts available which match in capacity and design sophistication the current mechanical harvesters.

## **APPENDIX B INDUSTRY MILL BIN SIZES**

We have endeavoured to list the bin sizes for all mill bins within the industry. The dimensions important to infield transporters i.e the height to top of bin from the rail and the internal length of the bin have been listed. Where the fleet is composed of various size bins we have listed the bin size which will eventually be the predominate bin in that region. Overall height of the bin determines the necessary lift height for any transporter and length determines the maximum allowable length for a tipper bin. The tipper bin proposed in this project fits comfortably inside the minimum length of the six tonne bins in the industry.

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MILL	CAPACITY	HEIGHT FROM RAIL TO TOP OF BIN	BIN LENGTH (INTERNAL)
MOSSMAN	10	2560	6070
HAMBLEDON	4	2130	2740
MULGRAVE	4	2270	2620
BABINDA	6	2428	3759
	4	2130	2740
MOURILYAN	6	2413	3670 **
STH JOHNSTONE	4	2100	2710
TULLY	4	2120	2750
	10	2120	6000
VICTORIA	4	2130	2740
MACKNADE	4	2130	2740
INVICTA	4	2200	2425*
PIONEER	4	2610	2440
	4	2500	2650*
KALAMIA	4	2320	2840
INKERMAN	3.5	2400	1930
	5.5	2445	3300*
PROSERPINE	4	2150	2640
FARLEIGH	4	2130	2650
RACECOURSE	5.5	2260	3250
PLEYSTOWE	6	2250	3720
MARIAN	4	2210	2350
CATTLE CREEK	3.5	2210	2180
PLANE CREEK	4	2204	2500
	9.6	3250	2420
FAIRYMEAD	6	2274	3658
MILLAQUIN	5	2260	2896
	6	2274	3658
BINGERA	6	2260	3660
ISIS	6	2260	3660
ROCKY PT	24	2512#	11400*
MORETON	4	2240	2680
BROADWATER	20	2600#	11500*
CONDONG	20	2700#	11500*
HARWOOD	20	2700#	11500*

# FROM GROUND \* EXTERNAL

\*\* ROUGH

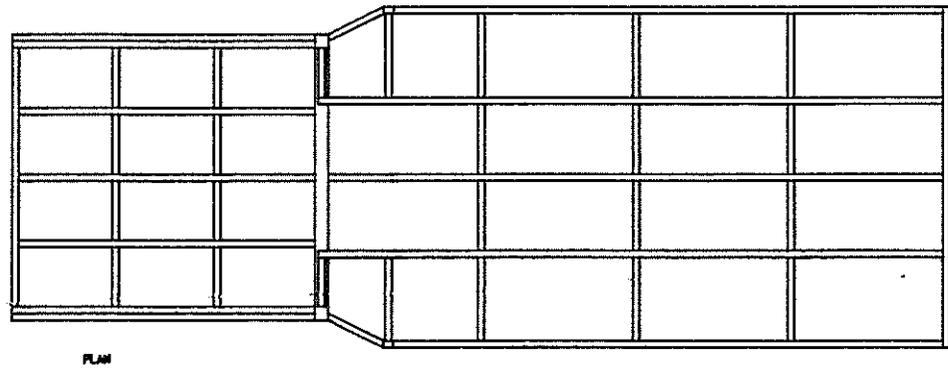
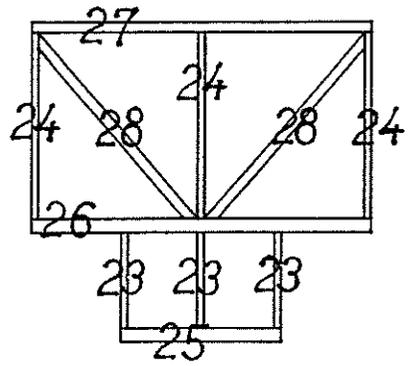
## APPENDIX C BEAM SIZES

Below are listed the beam sizes for the baskets for the double six tonne tipper and tipper elevator bins. the item numbers correspond to those found on the diagrams.

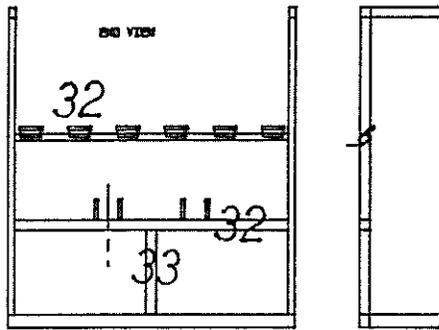
### BEAMS FOR TIPPER-ELEVATOR

ITEM	QTY	SECTION	LENGTH	WEIGHT	WEIGHT TOTAL
1	3	100 x 50 x 3.5	2200	16.57	49.70
2	1	100 x 50 x 3.5	1900	14.31	14.31
3	1	100 x 100 x 9	1900	44.65	44.65
4	8	100 x 50 x 3.5	482.5	3.63	29.07
5	2	100 x 100 x 4	2350	27.26	54.52
6	6	100 x 50 x 3.5	512.5	3.86	23.15
7	3	100 x 50 x 3.5	4580	34.49	103.46
8	2	75 x 50 x 3	1367	7.41	14.82
9	2	75 x 50 x 3	1630	8.83	17.67
10	2	75 x 50 x 3	2505	13.58	27.15
11	2	75 x 50 x 3	1200	6.50	13.01
12	2	100 x 100 x 5	2175	30.89	61.77
13	2	75 x 50 x 3	2098	11.37	22.74
14	2	75 x 50 x 3	542	2.94	5.88
15	2	75 x 50 x 3	2350	12.74	25.47
16	2	75 x 50 x 3	481	2.61	5.21
17	2	75 x 50 x 3	1429	7.75	15.49
18	2	75 x 50 x 3	1370	7.43	14.85
19	2	75 x 50 x 3	4293	23.27	46.54
20	2	75 x 50 x 3	4205	22.79	45.58
21	2	100 x 50 x 3.5	481	3.62	7.24
22	2	100 x 50 x 3.5	4205	31.66	63.33
23	11	75 x 50 x 3	705	3.82	42.03
24	3	75 x 50 x 3	1370	7.43	22.28
25	1	100 x 50 x 3.5	1175	8.85	8.85
26	1	100 x 50 x 3.5	2500	18.83	18.83
27	1	75 x 50 x 3	2500	13.55	13.55
28	2	75 x 50 x 3	1805	9.78	19.57
29	2	100 x 50 x 3.5	463	3.49	6.97
30	8	100 x 50 x 3.5	613	4.62	36.93
31	2	100 x 50 x 3.5	7255	54.63	109.26
32	2	75 x 50 x 3	2000	10.84	21.68
33	1	75 x 50 x 3	619	3.35	3.35
				TOTAL	1006

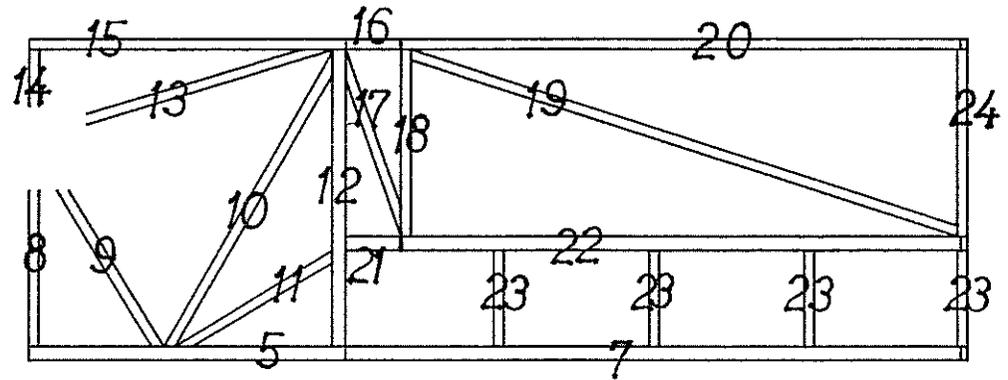
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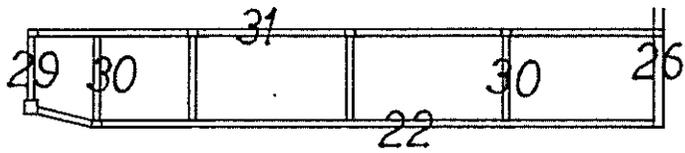
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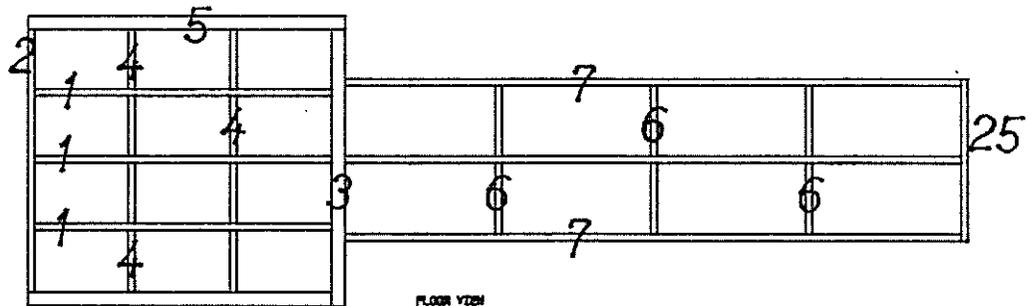
FRONT VIEW



SIDE VIEW



WHEEL HELL FLOOR

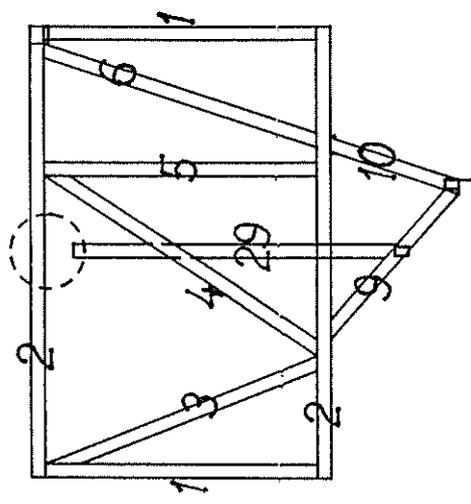


FLOOR VIEW

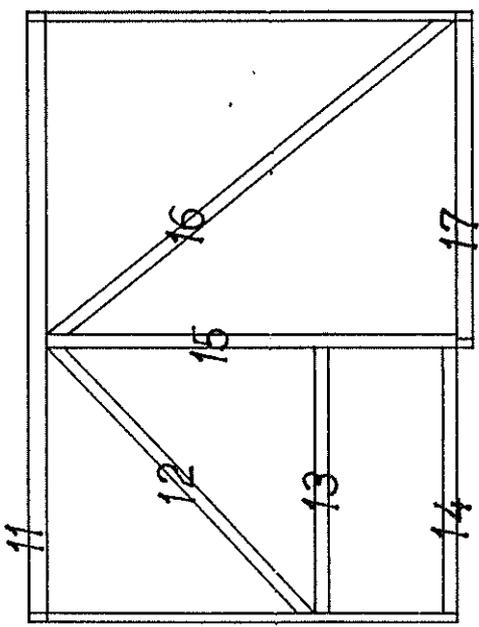
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**BEAMS FOR TIPPER BIN**

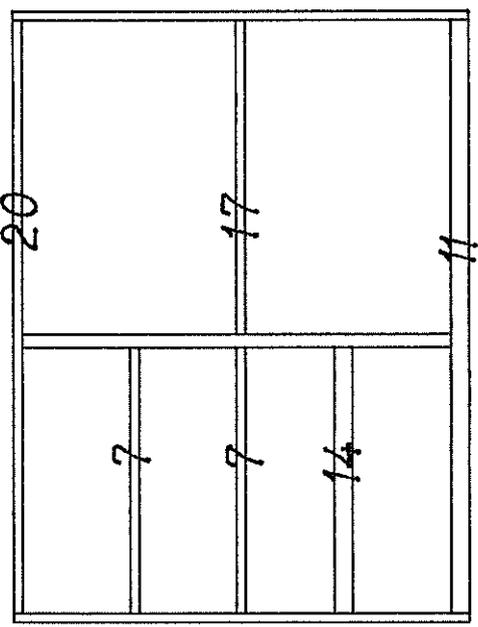
ITEM	QTY	SECTION	LENGTH	WEIGHT	WEIGHT TOTAL
1	1	75 x 50 x 4	1500	10.38	10.38
2	2	75 x 50 x 4	2500	17.30	34.60
3	1	75 x 50 x 4	1616	11.18	11.18
4	1	75 x 50 x 4	1794	12.41	12.41
5	1	75 x 50 x 4	1501	10.39	10.39
6	1	75 x 50 x 4	1581	10.94	10.94
7	2	75 x 50 x 4	1475	10.21	20.41
9	1	75 x 50 x 4	1094	7.57	7.57
10	2	75 x 50 x 4	763	5.28	10.56
11	1	100 x 100 x 5	3300	46.86	46.86
12	2	75 x 50 x 4	2085	14.43	28.86
13	2	100 x 100 x 5	1475	20.95	41.89
14	1	75 x 50 x 4	1475	10.21	10.21
15	2	75 x 50 x 4	2250	15.57	31.14
16	2	100 x 50 x 4	2849	24.19	48.38
17	3	100 x 50 x 4	1778	15.10	45.29
18	2	100 x 50 x 4	2350	19.95	39.90
19	1	100 x 50 x 4	2693	22.86	22.86
20	1	100 x 50 x 4	3300	28.02	28.02
21	1	100 x 50 x 4	2362	20.05	20.05
23	3	100 x 50 x 4	2500	21.23	63.68
25	1	75 x 50 x 4	715	4.95	4.95
26	1	75 x 50 x 4	700	4.84	4.84
27	1	75 x 50 x 4	1094	7.57	7.57
28	1	75 x 50 x 4	627	4.34	4.34
29	1	75 x 50 x 4	1777	12.30	12.30
30	1	75 x 50 x 4	2060	14.26	14.26
			total	weight	577



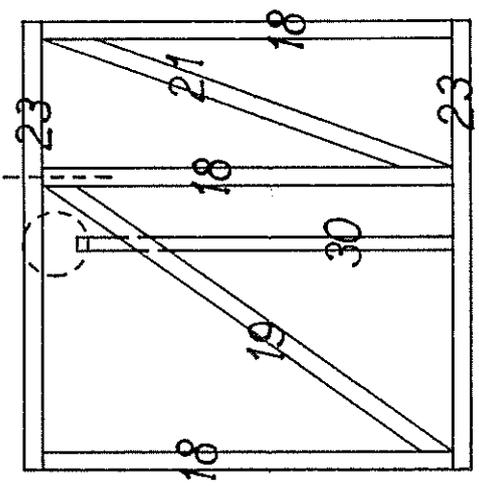
END FRAME / INSIDE END



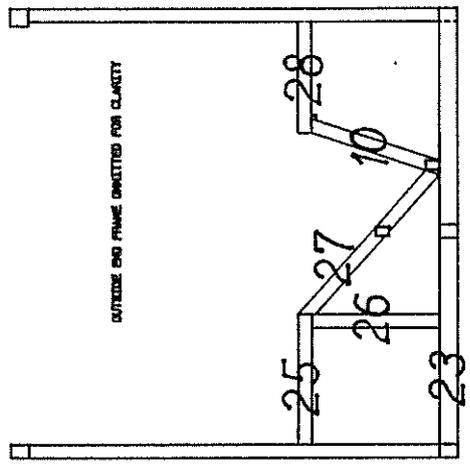
SIDE VIEW



PLAN



END FRAME / OUTSIDE END



SECTION A-A

BEAMS

## APPENDIX D PUBLISHED PAPERS

# MEASURING HITCH LOADS ON AN INFIELD HAULOUT BIN

By  
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## Introduction

As part of the ongoing research associated with the Sugar Research Council project "Development of a Large Capacity Haulout Bin", it was deemed necessary to measure some of the loads associated with the operation of some existing trailed haulout bins.

The loads of interest were those between the tractor and the bin at the weight-transfer hitch, which are more convenient to find than those at other parts of the structure. In this situation the existing connecting pins were simply replaced with two instrumented load-measuring pins used in conjunction with a micro-processor based data acquisition system.

The positioning of these pins (i.e under and slightly forward of the rear axle) yielded the following results:

- Static and dynamic weight transfer from the bin to the rear axle of the tractor;
- Horizontal loading on the tractor from the bin i.e rolling resistance;
- An indication of the dynamic variation of the loads in transit and during the various operations e.g unloading.

## Acquisition system design

Two sets of load measuring pins were designed, one for a McLean 12 tonne tipper-elevator, and one for a Greaves Enterprises 10 tonne combination tipper bin.

The approximate location of the pins on both bins is indicated by Figure 1. They are subjected primarily to a shear stress through a vertical section of the pins in the plane at the connection of the hanger and the hitch. The replacement pins were machined to give a shear-web structure across the critical section. The strain gauges were then positioned on this surface to give a measurement of shear strain (Figure 2). The measured strain could then be linearly related to the shear stress by bulk modulus, which in turn could be linearly related to the shear load by a simple stress formula. Commercially available strain gauge pairs mounted at 90° to each other (Figure 2) were used. The pins were designed to give a shear stress of 160 MPa under a load of six tonnes. The material used was X4140, a heat treated alloy steel with a yield strength of 410 MPa.

On each pin, surfaces were machined to give both a horizontal and a vertical load measuring facility. For the McLean bin, because only two signal conditioners were available, the horizontal and vertical signals were summed electronically to give total horizontal and total vertical load outputs. On the second set of pins for the Greaves Bin, the vertical and horizontal loads were measured separately at each pin. This removed the risk of cancellation of total loads by summation of individual pin loads of opposite sign.

For reliability and simplicity, a commercially available strain-gauge signal conditioner (Analog Devices 1B31AN) was used to enable the pin output signal to

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**KEYWORDS:** Haulout, Hitch, Load, Force, Measurement, Infield

be amplified and offset to the 0-5 volt range usable by the data acquisition computer, to enable adjustment of the bridge excitation voltages for optimum outputs, and to give a remote sensing option which ensured a constant bridge excitation voltage.

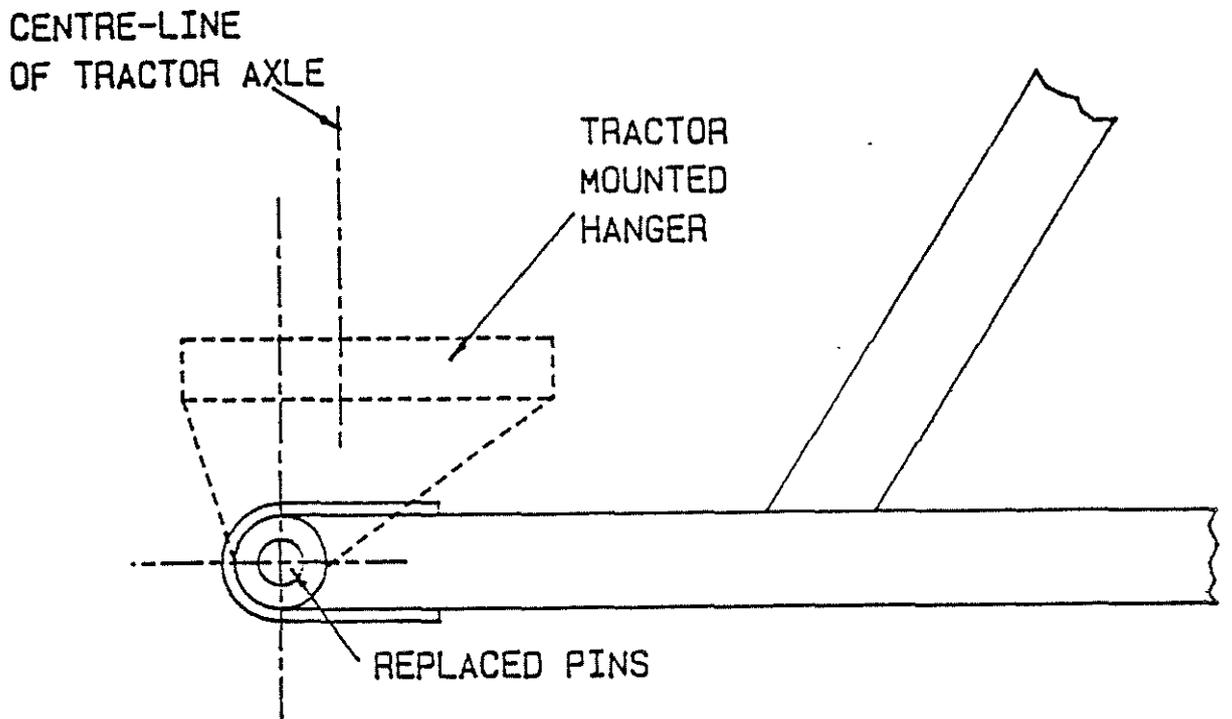
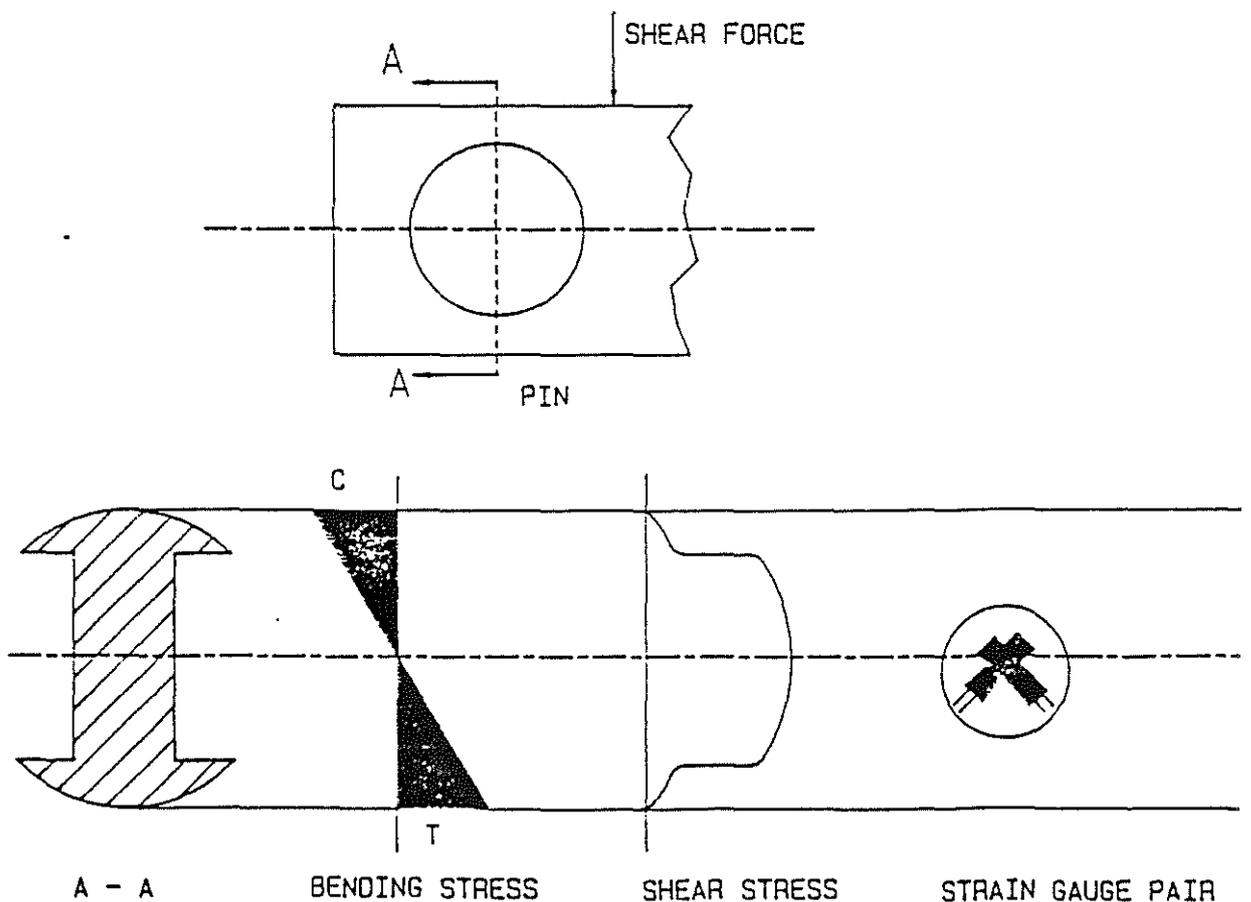


Fig. 1—Location of replaced hitch pins.



### Data acquisition computer

A JED-STD 800 computer was used to acquire the data infield. The JED is a single board CMOS computer, designed specifically for data logging applications. Analog input from the pins was fed into the onboard Analog to Digital converter, and the acquired signal was subsequently stored in a 64 kb "Datasafe" which was used in conjunction with the JED. A laptop computer was used infield to download the data to floppy-disk for future analysis.

The JED computer uses a simple on board BASIC to control acquisition processes. Two modes of data collection were used:

1). A slow rate process which made use of the onboard clock to read the input channels (i.e. horizontal and vertical) ten times a second then store the average of these ten values every second.

2). A fast rate process which used an assembly language routine called from the BASIC program to read the input channels at a much higher rate. For the first set of pins a rate of 80 Hz was used. This was chosen on the basis of an estimated dynamic signal in the 10 Hz region. A dynamic signal should be sampled at six to ten times the highest frequency in the signal. (Kocher and Summers, 1987). It was decided, however, on the basis of the results obtained from using this rate that a rate of 20 Hz was adequate. The acquisition program was subsequently rewritten for a fast rate of 20 Hz to be used in conjunction with the Greaves pins.

### Calibration of the system

Four testing jigs (i.e two per pin set) (Figure 3) were made up in the University College of Southern Queensland (UCSQ) workshop to simulate infield loading conditions. These were used to load the pins in a universal testing machine. The individual pins were loaded up to 60 kN in intervals of 10 kN, and then two pins were loaded together to 120 kN in 10 kN intervals.

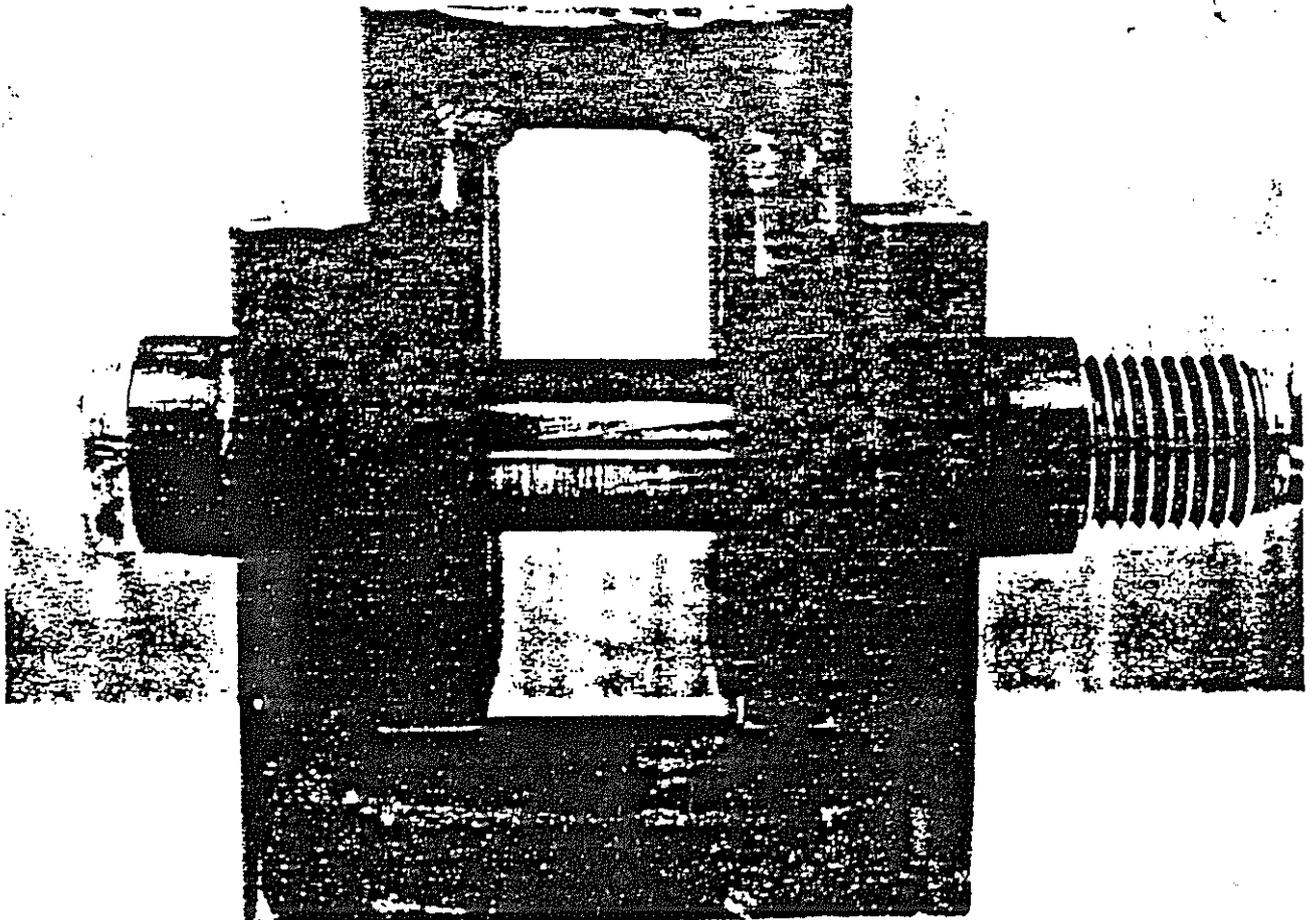


Fig. 3. Testing jig and pin

same point each time it would be unreasonable to expect the same average values during each test.

**TABLE II**—Average vertical forces, Greaves bin.

Test run number	Sample rate	Pin A force (kN)		Pin B force (kN)		Pin A + Pin B
		Average	SD	Average	SD	
1	SLOW	9.9	4.4	9.2	3.7	19.1
2	SLOW	9.2	3.8	8.4	3.1	18.0
3	SLOW	9.1	3.6	8.1	2.9	18.2
4	FAST	10.1	3.9	9.7	3.6	19.8
5	FAST	8.2	3.8	8.2	3.2	16.4
6	FAST	8.8	4.3	7.9	3.4	16.7

**TABLE III**—Average horizontal forces, Greaves bin.

Test run number	Sample rate	Pin A force (kN)		Pin B force (kN)		Pin A + Pin B
		Average	SD	Average	SD	
1	SLOW	15.3	3.9	4.6	3.4	19.9
2	SLOW	14.6	3.2	4.8	3.4	19.4
3	SLOW	14.3	3.2	4.8	2.7	19.1
4	FAST	15.3	4.0	2.3	4.2	19.5
5	FAST	15.1	4.2	3.5	3.5	18.6
6	FAST	13.5	3.9	4.3	4.0	17.8

There is some uniformity in the results from the tests associated with the Greaves bin, since all the readings associated with this test were undertaken in reasonably similar circumstances. Of interest with the Greaves bin readings is the large difference in horizontal forces between the two pins. The possibility of instrumentation errors was considered but discounted. It appears as if the geometry is such that one side consistently takes most of the load.

The vertical and horizontal forces associated with a complete haulout cycle for the Greaves bin are shown in Figures 4 and 5. It is possible from this to note the critical stages of the operation of the bin. The major vertical forces occur, as expected, when the front bin is being loaded.

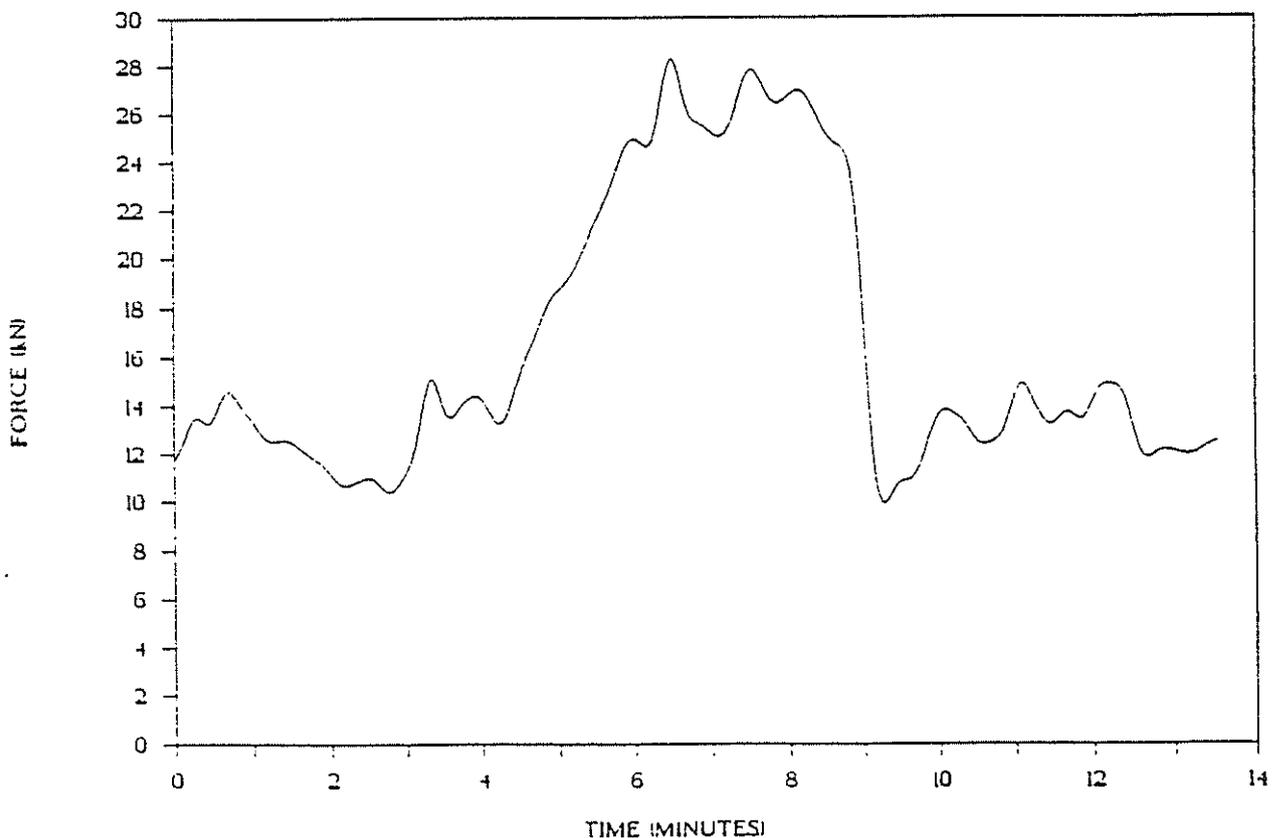


Fig. 4—Vertical hitch forces during loading cycle. Run 5, Greaves Bin.

The first set of pins gave a response significantly different from that expected. The pins displayed a marked cross-talk characteristic, i.e., under a pure horizontal load, outputs were given by both the horizontal and vertical channels. This characteristic could be quantified by defining an electrically neutral axis for each channel of each pin, defined by the angle through which the pin must be turned to obtain a zero output from one of the channels. With this axis defined, the calibration results could be used to convert the raw signals into meaningful readings, although uncertainties still existed about summing forces from the separate pins.

The second set of pins had the advantage of two analog outputs per pin, i.e., one for each channel on each pin. Consequently, the uncertainties introduced by summing signals were alleviated. These pins displayed minimal cross-talk.

## Fieldwork

The first set of pins were used in conjunction with the McLean haulout owned by J. Russo and Sons of Childers. The second set were used with the Greaves bin operated by Mr L.G. Hodgetts, a Bundaberg contractor.

The data were recorded under normal field operation, with few concessions being made to the acquisition process. All the acquisition equipment was contained unobtrusively within the tractor cabin. Data were taken in both the slow and fast collection mode, with the times corresponding to different operations noted. A full range of haulout operations was performed during testing.

For some of the time during testing the McLean tipper-elevator was being employed in a purely infield role, loading roll-on/roll-offs at the headland. However this bin was also used as a road-going unit as well.

## Problems with infield testing

Under field conditions the leads from one of the first set of pins were sheared off, necessitating repairs under less than ideal conditions. One of the pins also rotated from its locked position through about 15°. It was also found that with the acquisition computer connected to the tractor battery, voltage transients on starting resulted in a loss of stored data.

## Results

The test results consisted of six runs for each bin in both the slow and fast storage modes. These values were subsequently processed with a data analysis package "Asystant". The resulting averages and standard deviations are shown in Table I, Table II and Table III. Two runs from the McLean bin were discarded because of questionable data.

**TABLE I**—Average forces, McLean bin.

Test run number	Sample rate	Horizontal forces (kN)		Vertical forces (kN)	
		Average	SD	Average	SD
1	SLOW	11.2	4.2	20.1	3.2
2	FAST	11.3	5.2	19.5	2.7
3	FAST	13.3	4.8	21.4	2.6
4	FAST	23.3	9.4	25.7	6.1

There is a relatively wide range of average values associated with the McLean Bin readings. This is because the readings for the first three runs were taken while the bin was being used infield, and the last run involved the bin performing complete haulout cycles, where greater speeds and so greater accelerations contributed to higher horizontal forces. The average vertical forces remain fairly

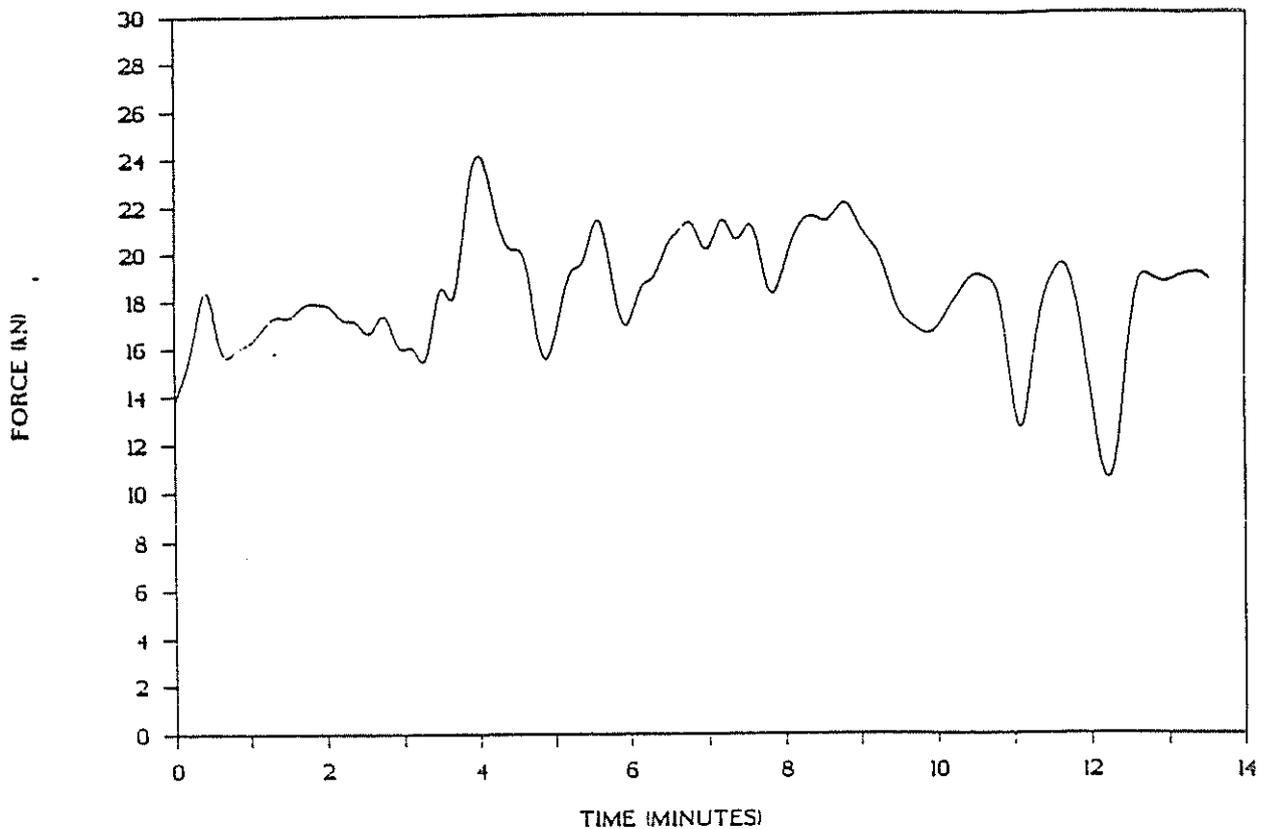


Fig. 5—Horizontal hitch forces during loading cycle. Run 5, Greaves Bin.

A benefit of measuring the forces separately at each pin is that it enables examination of forces encountered in turning. Figure 6 shows the differences between the horizontal forces in the two pins, with the high peaks corresponding to sharp turns. Differential forces of up to 20 kN are shown. From a knowledge of the hitch geometry the torque applied to the hitch can be calculated for use in the design process. In this case, moments of up to 10.4 kNm are being applied to the hitch in the vertical plane. In the worst scenario, this moment would combine at the vertical turning axis, with the moment due to the vertical force at the connecting pins. A vertical hitch force of 25 kN would cause a moment in the horizontal plane of around 23 kNm.

Examination of the horizontal forces in the light of a formula developed by McAllister (1983) for rolling resistance of trailed agricultural machinery, is useful. Employing a value of 300 kPa for the soil cone index, the coefficient of rolling resistance ( $C_r$ ) can then be calculated for any tyre. Multiplying the weight of the trailed unit by  $C_r$  gives an estimate of rolling resistance and therefore horizontal hitch forces.

The McLean bin was fitted with 23.1 \* 26 and 22.5 \* 26 tyres which gave a  $C_r$  of 0.161 from McAllister's formula. For an estimated axle weight of 16 tonnes a maximum horizontal hitch force of 25.8 kN could be expected. This corresponds well with the measured maximum sustained hitch forces of approximately 28 kN. The Greaves bin was fitted with 18 \* 19.5 tyres which gave a  $C_r$  of 0.274. A horizontal hitch force of 30.1 kN is predicted using an estimated axle weight of 11 tonnes. The actual maximum horizontal hitch forces were about 24 kN, the difference probably being attributable to different parameters for such factors as soil type and axle weights than were assumed.

Average forces and standard deviations are useful in gaining a broad perspective of the results, but they are not enough to use in a design situation. A more useful representation of these values is in terms of histogram which gives both the magnitudes of the forces and their frequencies of occurrence. Shown in

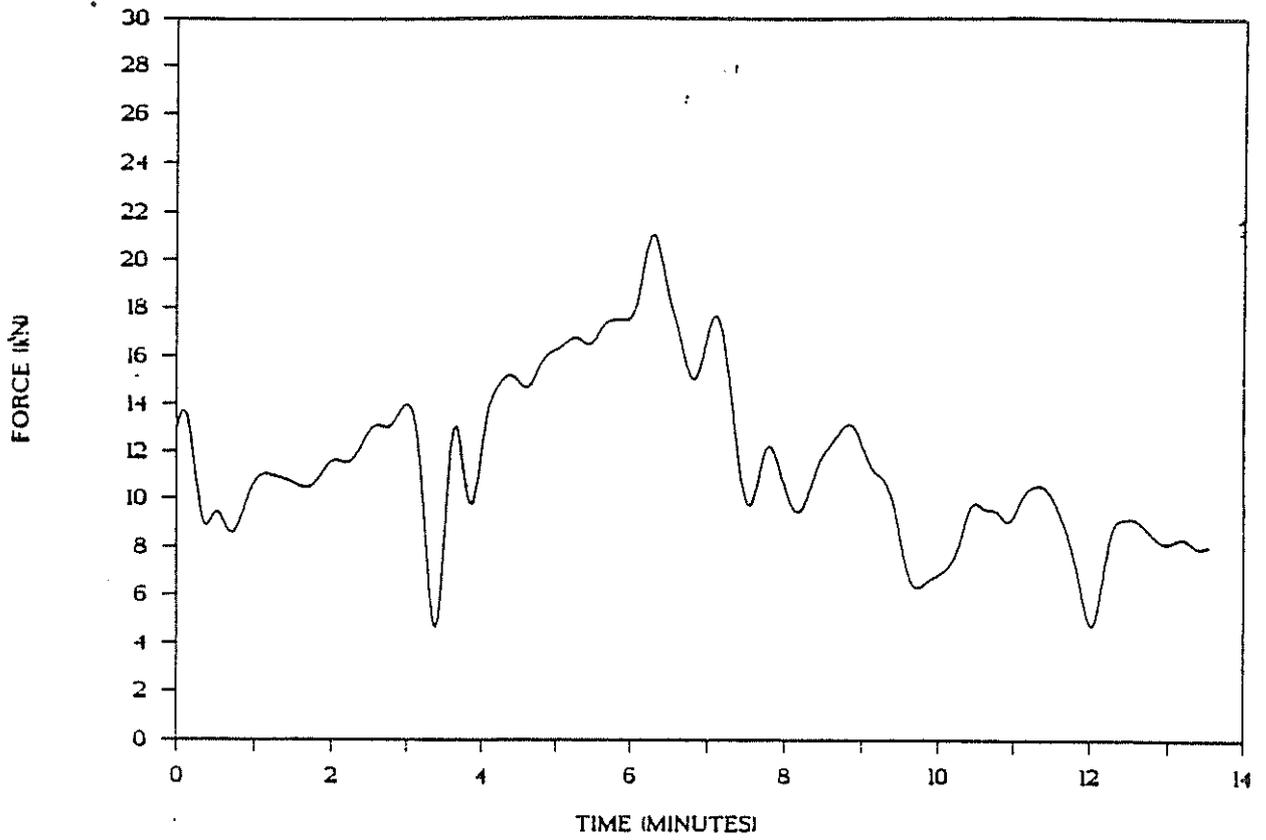
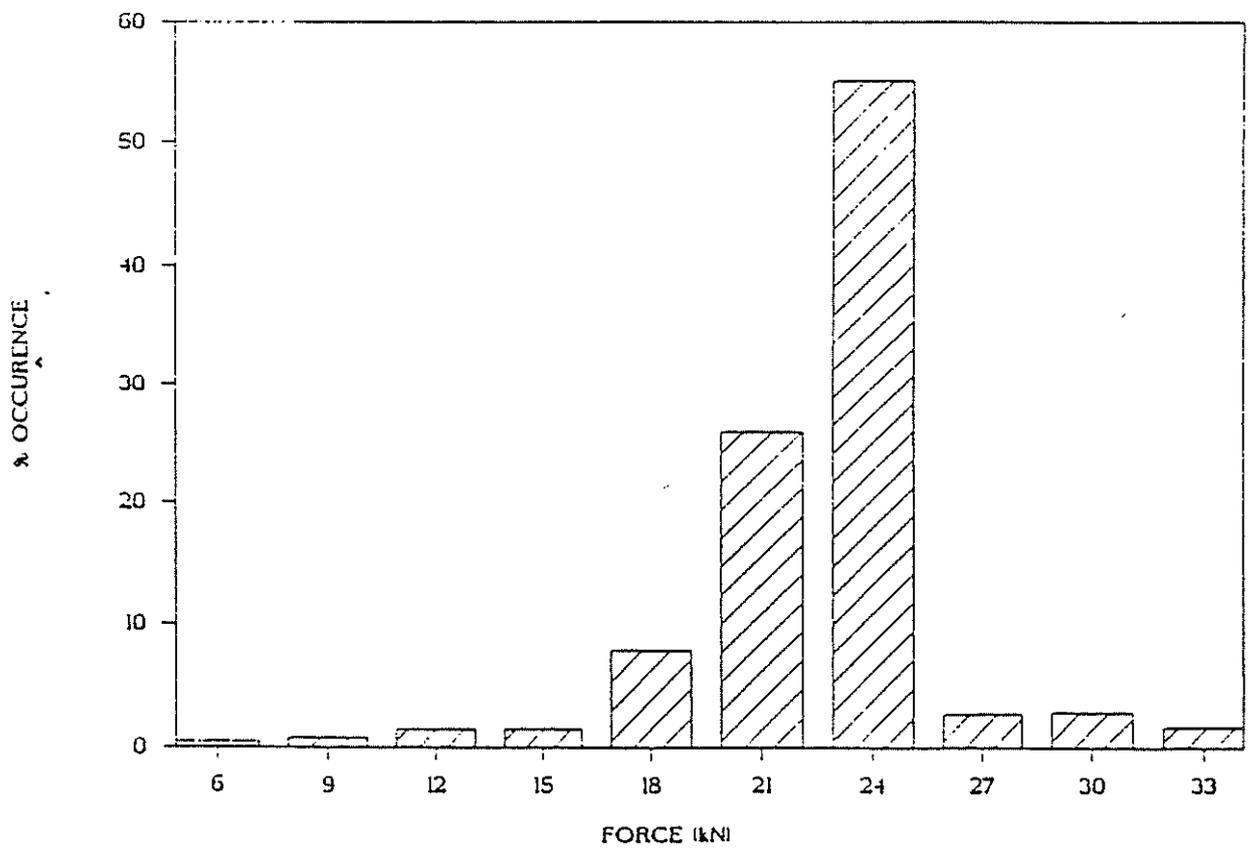


Fig. 6—Differences in horizontal loads between pins. Run 5, Greaves Bin.



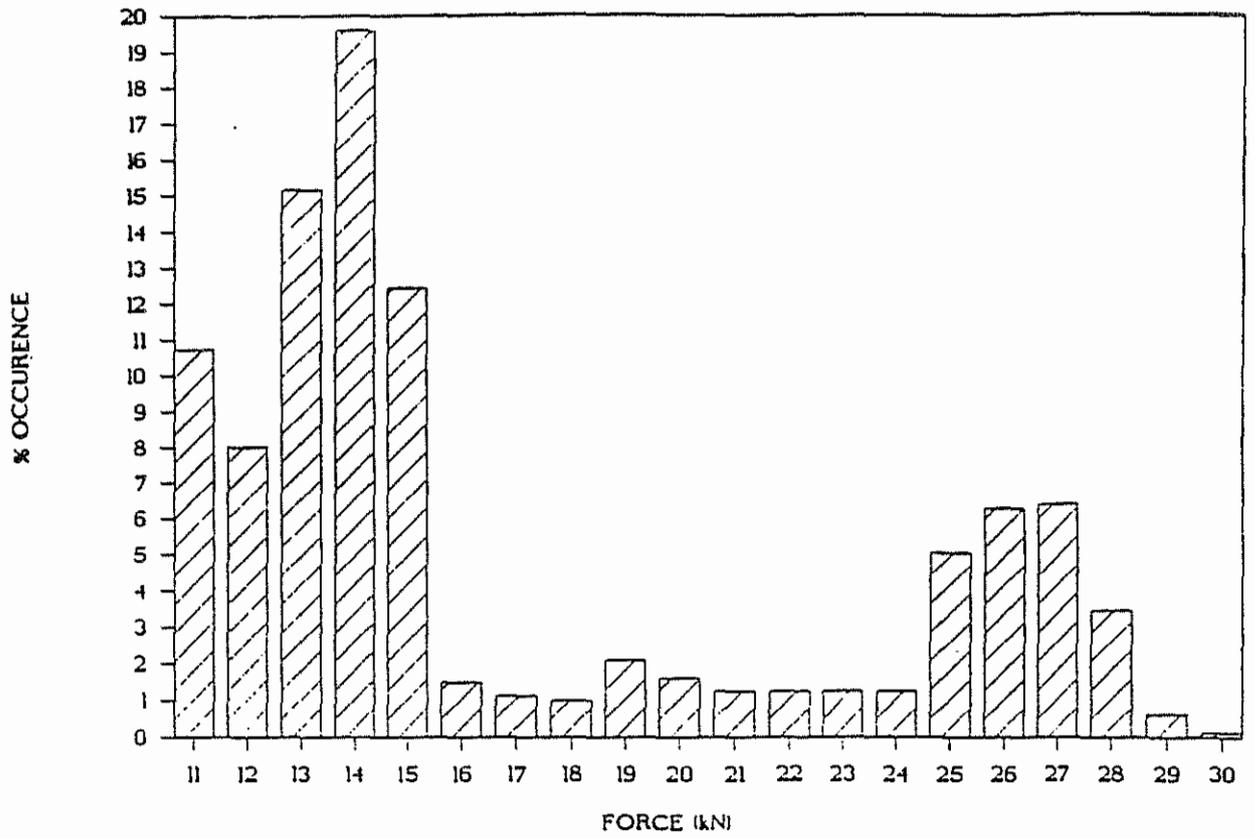
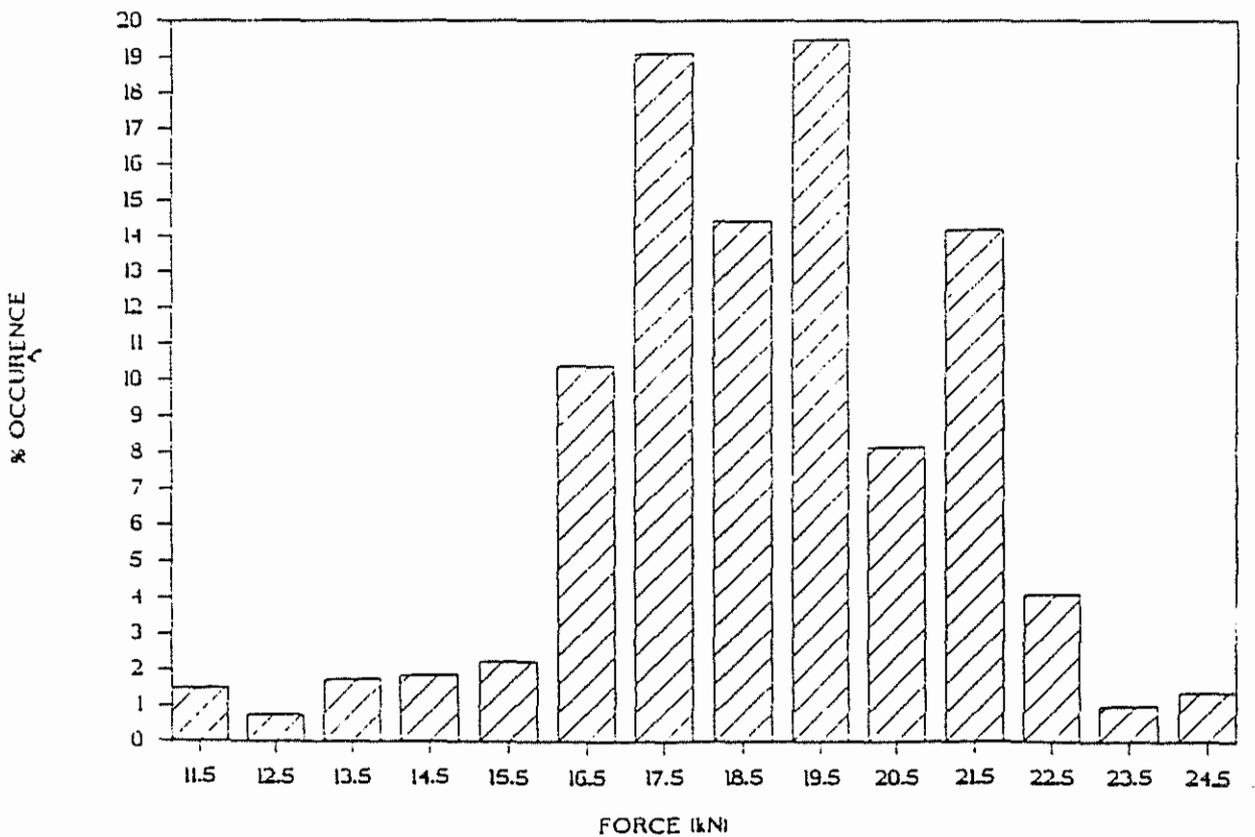


Fig. 8—Histogram — vertical loads. Run 5, Greaves Bin.



From the design perspective, it is important to identify any cyclic dynamic loading. This is best done by means of a power spectral density (PSD) of the resulting forces, which is a frequency domain analysis. The PSD shows which frequencies occurred in the signal and their relative magnitudes.

Despite what might have been expected there was not a marked cyclical loading. No dominant frequencies were discovered in the first set of test data. It was for this reason that the sampling rate was dropped from 80 to 20 Hz for the second set of tests. However there is a danger that if out of phase dynamic signals occurred between the two pins, this dynamic effect might have been cancelled in the summation of the two signals.

In the second set of test results (Greaves bin), dynamic load effects were identified for the horizontal channel on the B-pin, for short periods of time during the travelling to the siding phase. This particular PSD is shown in Figure 10. The dominant frequency is around the 0.5 Hz range which is a low frequency. It is hypothesised that because this pin was only carrying a small portion of the total horizontal load, it was relatively free to experience a motion related dynamic loading. On the whole though, it appears that hitch forces, possibly because of the varying nature of the load and terrain, are totally random.

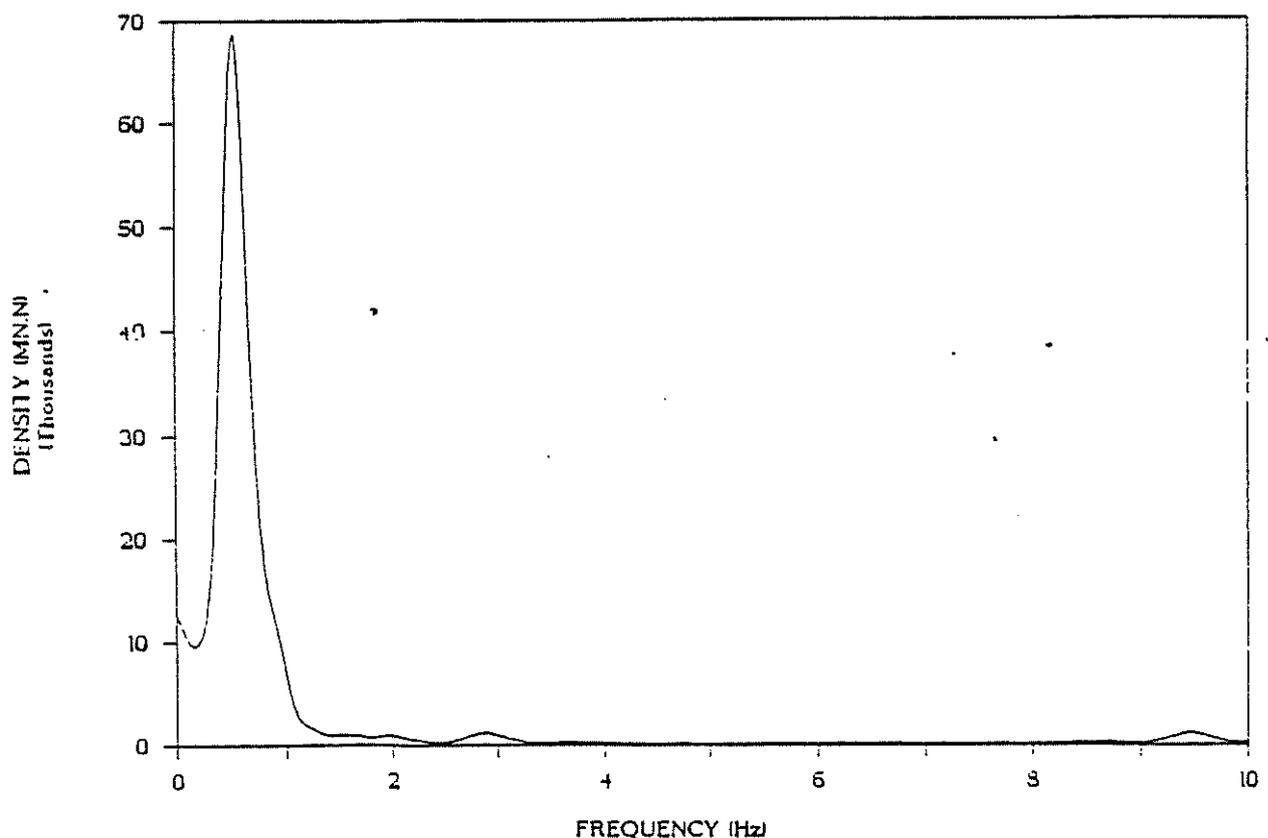


Fig. 10—Power Spectrum Density — Run 5, Greaves Bin.

## Conclusion

The two load measuring exercises undertaken proved most useful in characterising the forces involved in towing trailed haulouts. The second test based on the learning experiences of the first appeared to be more reliable and useful than the first.

From the tests, critical levels of forces and their associated operations can be identified. The forces appeared to be of a mainly random nature lessening the need to consider fatigue failure in the design. The horizontal rolling resistances correlated reasonably well with the predicted results.

It must be borne in mind that the forces measured correspond only to a few of the many potential situations that arise in haulout work. Long distance hauling

and undulating terrain are some common situations not really covered in the scope of this work. Any conclusions drawn from the measured forces must then be taken with the knowledge that larger forces will most probably be encountered. Bearing in mind these points, the forces measured provide a starting point for an initial hitch (and haulout) design.

#### **Acknowledgements**

The authors wish to thank J. Russo and Sons of Childers and Mr. L.G. Hodgetts and his haulout team for their willing assistance. Thanks are also due to the technical staff of the UCSQ for their help in the fabrication and calibration of the components necessary for testing.

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THE ANALYSIS AND DESIGN OF INFIELD HAULOUT BINS FOR THE AUSTRALIAN CANE INDUSTRY.

KEYWORDS: design, haulout, bin, cane, stress analysis

ABSTRACT: A survey of present and future needs for more efficient haulout bins in the Australian cane industry has led to the specification of two bin layouts, a double tipper and a tipper elevator. Designs have been prepared with the aim of reducing tare weight and cost, as the basis for the introduction of more efficient haulout means into the industry. This work was funded by the Sugar Research Council as a design study in an effort to increase the rate of adoption of more efficient haulout units throughout the industry, by proposing designs which used common components and which would offer the lowest cost upgrade path for haulout users.

REFERENCE: Harris, H.D. and Pearce, F., 'The Analysis and Design of Infield Haulout Bins for the Australian Cane Industry'.

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