

**BUREAU OF SUGAR EXPERIMENT STATIONS
QUEENSLAND, AUSTRALIA**

**FINAL REPORT
SRDC PROJECT BS210S
LIGHTWEIGHT ELEVATOR AND
ADVANCED SECONDARY CLEANING
SYSTEM FOR CANE HARVESTERS
by
C P Norris and R J Davis
SD01011**

**Principal Investigator: Mr C P Norris
Senior Research Officer
BSES
Private Bag 4
BUNDABERG DC QLD 4670
Phone: (07) 41325200**

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1.0 SUMMARY

The goal of this project was the development of a prototype lightweight harvester elevator and integrated secondary cleaning system. The goal was commercially driven and proposed transfer and adaptation of high-speed conveyor technology (from mining and from other agricultural materials handling applications) and advanced pneumatic cleaning concepts to the cane harvester. The design brief targeted enhancing machine performance (ie cane loss, extraneous matter and pour rates) whilst reducing machine weight and improving machine stability.

The development has not, however, resulted in a commercially viable prototype. Because of unforeseen difficulties relating to the design adopted for the hugger belt system, the final goals could not be met within the framework of the initial project. When 'pushing the limits' of machine development, problems such as this must be expected, particularly when the development is being undertaken on limited manpower and fiscal budgets.

The project, however, has in no way been unsuccessful. Whilst the nominated final goal has not been met, the project has very significantly progressed knowledge relating to the cleaning of cane, and has clearly shown that the concepts embodied in this project have very considerable potential.

Key outcomes of the project included the following.

- The concept of alternative design of the harvester elevator bowl, which potentially enhances the performance the current primary extractor, has been demonstrated.
- The ability to feed billets into a hugger belt system at commercially viable rates has been demonstrated.
- The ability of a hugger belt system to present billets in a configuration more suitable for effective cleaning than the current chain and slat elevator has been demonstrated.
- The enhanced cleaning performance offered by the blower type secondary cleaning module, the design of which was optimised by using computational fluid dynamics modelling, has been graphically demonstrated.
- The lighter weight, and more importantly lower overturning moment of the elevator assembly, has been demonstrated, with extremely positive comment from machine operators.

The project also demonstrated that the complementary strengths of different organisations could be used to enhance the outcomes of a project.

The difficulties encountered in the final design relate to the operational reliability of the hugger belt configuration adopted. Initial trials of the final prototype in the workshop indicated major problems with the belts 'running off' requiring continual adjustment of belt alignment. The severity of the problem was such that a field trial program could not be contemplated. Although belt tracking had been a significant issue with the initial proof of concept module, discussions with conveying technology experts had indicated that close attention to alignment of rollers and careful attention to optimisation of the 'crowning' of rollers would mitigate the problems. The project team had been sufficiently

confident that this would not be a problem because, in the design of the final unit, the 'limits had been pushed' even further in terms of the use of the lower belt configuration to optimise the trajectory of the billets as they exit the conveying module. This confidence was clearly in error; however, the prototype did allow the demonstration of the potential of the concept.

It is strongly recommended that further work be undertaken to investigate potential solutions to the belt tracking issues. A solution to this problem would allow the industry to capture the significant developments made throughout this project and allow the commercial development and availability of a high-speed elevator and advanced cleaning system.

2.0 OBJECTIVES

The objectives of the project were to develop, to a commercial prototype, a lightweight cane harvester elevator incorporating an advanced concept for secondary cleaning of the cane. Key features of the prototype were to include the following structure.

- The incorporation of an elevator bowl designed to give more efficient entry of air into the cleaning chamber (complementing the approaches of SI49) and more 'available space' for other cleaning chamber developments (eg projects NE1 and SI61).
- A lightweight high speed elevator system. The proposal was to operate the elevator on a steeper angle than current elevators, therefore allowing a layout that placed fewer size constraints on the design of primary cleaning chambers than the current elevator design. It was also anticipated the system would potentially offer lower 'whole of life' costs.
- The secondary cleaning system concept was designed to capitalise on the presentation of material afforded by the high-speed elevator. Whilst the current secondary extractor performance would be improved if fed with the more evenly presented thin layer of material offered by the high speed conveyor, the design of a lightweight cleaning system, based on blowers for the air supply, was a primary objective of this project.

The project was conducted in conjunction with key commercial stakeholders, including Gough Plastics, Beltreco and I.H. Austoft. Staff at James Cook University undertook the engineering design and staff at SRI undertook the computational fluid dynamics (CFD) modelling to optimise the performance of the system.

3.0 BACKGROUND

A significant increase in harvester pour rates has been witnessed throughout the Australian sugar industry in recent years, even where harvest contract sizes have not increased. This increase in pour rate has been primarily facilitated by an increase in harvester engine power. Unfortunately, the inability of the harvester to effectively clean the cane at these pour rates, whilst maintaining acceptable cane loss, has been clearly

demonstrated from trials conducted in north Queensland from 1997-2000 by Whiteing et al (2001). The increase in extraneous matter (EM) in the cane supply resulting from higher pour rates has been identified as a primary cause of the low CCS problem in north Queensland (Leslie and Wilson, 1996).

Increases in harvester pour rates are, however, essential for the continuing viability of the cane harvesting sector, and subsequently the industry. Notwithstanding these pour rate increases, EM content in the cane supply must be reduced to enable the industry to meet sugar quality targets and minimise CCS reductions. Similarly, cane loss must also be reduced to more acceptable levels for the continuing viability of the Industry.

The fundamental problem is the inability of the current cleaning system to achieve both high EM removal and low cane loss at the high pour rates the industry now demands. Actual cane loss of over 15 t/ha (measured by replicated 'mass balance' trials as distinct from 'blue tarp' tests) from harvesters operating in 'ex-factory' set-up was not uncommon as shown by Whiteing *et al* (2001). The inadequacies in the conceptual design of current cleaning systems for high pour rate cleaning are exacerbated by inappropriate presentation of the cane to the cleaning chamber directly from the choppers (Hobson, 1995; Quick D 1982). Additionally, the presentation of cane to the secondary extractor by the conventional chain and flight conveyor, whilst somewhat better than the presentation to the primary extractor, is still highly inappropriate for effective cleaning to occur. This inappropriate presentation, along with a range of design constraints, leads to relatively low cleaning efficiencies and excessive cane loss under conditions of high vegetative EM in the material being presented. Whiteing (2001) indicates that the secondary extractor typically removes approximately 20% of the vegetative EM it is presented with. Cane loss can vary from very low >0.5 t/ha, under conditions requiring little secondary cleaning, to indicated cane losses of in excess of 7 t/ha under high EM conditions and at high pour rates.

Considerable resources have been committed by SRDC to projects addressing more appropriate cleaning chamber designs and alternative cleaning system concepts. To achieve the full potential offered by any of these advanced concepts, a major redesign of the harvester is necessary and, as such, this is a medium to long-term development. Significant effort is, however, being expended to manipulate the concepts within the constraints of the current harvester layout. The presentation of material to these systems (directly from the choppers) is such that the potential of the concepts is unlikely to be fully realised. If, however, more space were available at the rear of the harvester, better designs exploiting more of the potential performance gains of these concepts would be possible.

In addition to the issues of cleaning and cane loss, machine weight and stability are also crucial issues. Whilst best engineering practice is currently used by the manufacturers to minimise elevator mass, its mass and the mass of the secondary extractor not only add to machine weight, but also adversely impact on machine stability when operating across slopes. Previous attempts to redesign the elevator from non-ferrous metals to reduce weight have introduced seemingly insurmountable problems with wear, electrolysis/corrosion, durability and cost (Williams J, R & D Engineering, Austoft, pers. com. 1992

The SRDC mechanisation scholarship on reducing harvester weight found that a major constraint was the weight of the elevator. Its weight and weight distribution dictated the required strength and weight of harvester if acceptable stability and durability were to be achieved. A conclusion from the research was that if a lighter weight elevator with reduced overturning moment was achievable, significant further weight reductions in other components of the harvester would be possible.

The genesis of this project, therefore, arose from the identified need by industry to develop a lightweight elevator with enhanced cane cleaning performance. The concepts embodied in the project follow on from concepts developed as a result of work associated with BS152S, and the potential to transfer high speed conveyor technology from other industries.

Whilst the development of the next generation of cane harvesters is still some time off, the development of an elevator system offering lightweight and enhanced cleaning performance was seen as offering significant advantages for all sectors of the sugar industry.

To the harvester manufacturer such a system would offer:

- a reduction in weight of a major component of the harvester. The reduction in elevator weight will allow subsequent weight reduction in other harvester components;
- potential to increase the dimensions available for the cleaning chamber, and the potential to move the elevator further to the rear, allowing incorporation of further equipment in the primary cleaning area. The project directly assists the developments associated with projects NE1, SI49 and SI61;
- improved performance of the current harvester, due to the improved performance of the proposed secondary cleaning system. This will allow the current primary extractor systems to operate in a mode consistent with lower cane loss, and is complementary to all current development programs relating to enhancement of primary cleaning system performance;
 - an opportunity to facilitate the evaluation of concepts for the next generation of harvester. This facilitation relates to primary cleaning, conveying technology and secondary cleaning systems.

To harvester operators such a system would offer:

- a significant opportunity to reduce both cane loss and EM levels. It is anticipated the system would be a cost-effective retrofit to current harvesters;
- ideally, a lower whole of life cost than the current chain and slat elevators. The use of a blower-based cleaning system rather than the current extractor system would further reduce maintenance costs.

This was a joint project between manufacturers and research organisations. It was focused on a commercial outcome, but was to include both technology transfer and technology development.

It was anticipated the project would involve the integration of two significant and complementary developments into one harvester component.

4.0 DESIGN DEVELOPMENT

The industry standard cane harvester includes associated mechanical componentry to deliver cane billets to a cane receival vehicle whilst additionally offering multi-stage cleaning. After the chopper system severs whole stalks into billets, the flow of billets pass across the primary extraction zone. This system is based on an axial flow fan (nominally three bladed, although some newer designs incorporate a four blade design) which induces an upward air flow through the cleaning chamber, nominally in the range 15-20 m/sec (Joyce and Edwards), 1994. This upward air flow separates trash and EM from the cane billets as they pass across the chamber. The billets then fall into the conveying system, which elevates the billets. On exiting this conveyor, a second cleaning system offers some additional cleaning of the billets as they are delivered to the receival vehicle. The elevating and secondary cleaning components form the basis of the development undertaken in this project

4.1 Industry-standard elevator and secondary cleaning system

4.1.1 Elevator

The standard harvester elevator consists of a truss and pressed metal frame in which a chain and flight conveying system are located. The elevator can rotate through 160 degrees in the horizontal plane (slew) to enable the harvester to cut a one-face operation. Additionally, the elevator is adjustable in height to provide high clearance for various haulout vehicles and has lowering provision to allow clearance from overhead obstacles during travel and has to minimise height when conducting maintenance work.

The elevator can typically be described as an S-shape structure, nominally 850 mm wide, and supporting a chain and slat system operating over a stationary, typically perforated, floor to convey the product. Key points include the following.

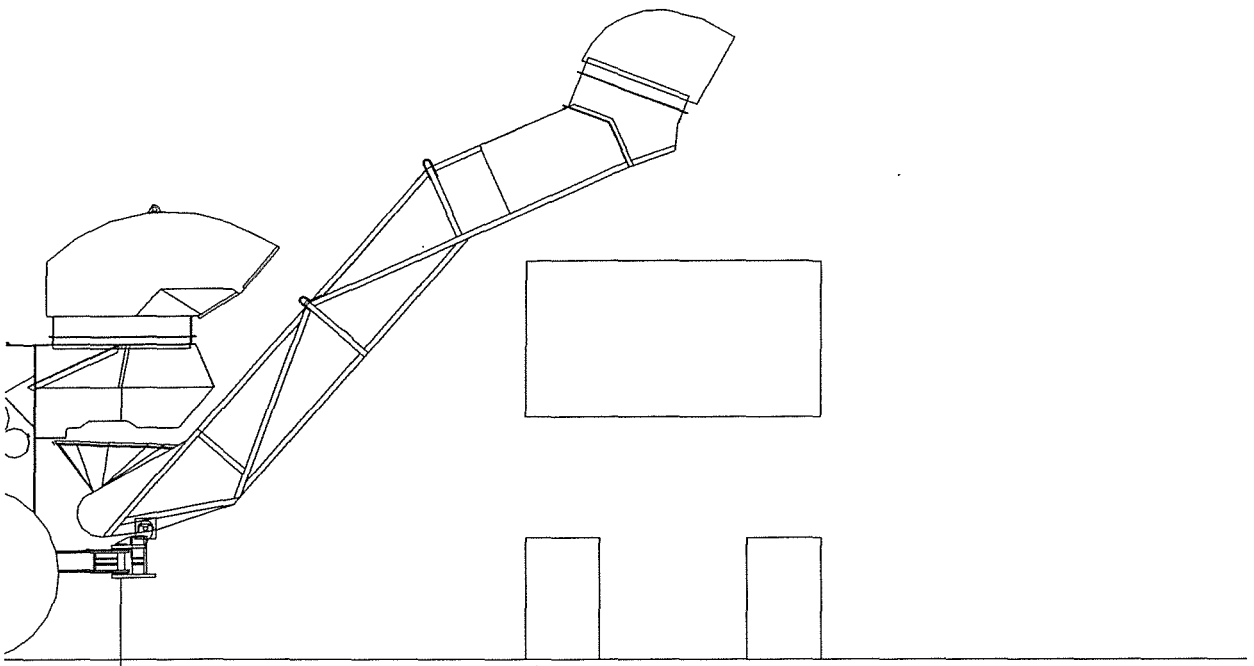
- At the lower end of the elevator, cane falls into a bowl, which acts as a buffer storage (storage volume of approximately 0.55 m³ of material) as well as constraining the material as it is taken away by the chain and slat elevator system.
- The chain and slat componentry consists of twin lengths of 2 inch pitch (51 mm) roller chain connected together by slats (flights) approximately 150 mm high. Between 20 and 22 slats, depending on the elevator model, are spaced at approximately 510 mm intervals. The chain and flights are driven by two sprockets on the top shaft and powered by two hydraulic motors. The unit is fully reversible to clear blockages. A nominal storage volume of 0.15-0.25 m³ is also achievable on the inclined section of

the elevator (back calculated from measurements of 3 to 10 kg/slat, depending on material density and billet length, 6.5 kg average, 7 active flights).

- Whilst earlier designs were essentially straight, more recent designs incorporate an 'S' shape, to allow the lower section to operate at a shallower angle, thus enhancing fill of slats, and at the top end giving the elevator further reach over the sides of transport equipment.
- The total storage volume of the conventional elevator is approximately 0.7-0.8 m³.

An additional issue with the design of the current elevator system is the space constraints the current elevator configuration places on the design of the primary cleaning system. All current projects relating to the development of and adaptation of advanced cleaning concepts to the current harvester (eg NE1, SI49, SI61) face undesirable layout constraints because of 'available space' limitations. The current elevator design limits the 'plan area' available for the cleaning system, and the constraints of machine height and the need for adequate clearance for the elevator bowl severely restrict the vertical space available. Some proposals involve moving the elevator backwards to partially alleviate these space problems, but at the expense of an accentuation of weight and stability problems.

Figure 1 illustrates the overall layout of the standard harvester chain and flight elevator, as fitted to a standard harvester. The constraints on primary cleaning chamber size caused by the elevator are evident.



**Figure 1 - Conventional elevator including elevator bowl and secondary extractor.
Note the 'S' configuration to maximise reach over the bin.**

4.1.2 Secondary cleaning system

The standard design cane harvester has a secondary cleaning system based on an extractor fan mounted at the exit point of the billets from the chain and slat elevator. The purpose of the system is to offer final stage cleaning of remaining EM from the cane billets as they exit the elevator and fall into the haulout vehicle. The standard system comprises a three-bladed axial flow fan housed in a barrel and discharge of the extracted material is achieved via a directional discharge hood. The fan is driven via a centrally mounted hydraulic motor. Significant issues for the industry are the cleaning performance of the current unit, its weight and location of centre of gravity, and the impact of this on machine stability. These are outlined below.

- The presentation of cane to the secondary extractor by the conventional chain and flight conveyor, whilst somewhat better than the presentation to the primary extractor, is highly inappropriate for effective cleaning to occur. This inappropriate presentation, along with a range of design constraints leads to relatively low cleaning efficiencies and excessive cane loss under conditions of high vegetative EM in the material being presented. Whiteing (2001) indicates that the secondary extractor typically removes approximately 20% of the vegetative EM it is presented with. His data also indicate cane loss can vary from very low (<0.5 t/ha), under conditions requiring little secondary cleaning, to indicated cane losses in excess of 7 t/ha under high EM conditions and at high pour rates.
- The cleaning system unit including fan, shaft, barrel, discharge hood and associated hydraulic equipment typically weighs in excess of 280 kg. The location of the cleaning system unit, coupled with the combined weight of the unit produces a substantial overturning moment at the rear of the harvester. This unbalancing force generates a degree of instability of the harvester during harvesting on uneven terrain and during road travel for wheeled harvesters.

The issues of weight (and associated overturning moment) and cleaning performance are major issues. Optimisation of the performance and reductions in weight of the current secondary cleaning system would offer significant advantages to the industry. The secondary cleaning system of a standard cane harvester is illustrated in Images 1 and 2.



Image 1 – External view of secondary extractor module. The mass of the current unit is in excess of 280 kg.



Image 2 - View of fan assembly of secondary extractor. The trash flow through the fan absorbs considerable power, reduces the aerodynamic efficiency of the fan and caused accelerated wear.

4.2 Lightweight elevator

4.2.1 Design goals

To allow significant improvement over the conventional elevator and secondary cleaning system, the primary design parameters were defined by the following criteria.

(a) *Lightweight:* Whilst best practice engineering approaches are used to minimise the weight of the current elevator system, the inherent design coupled with the structural requirements, including the need to support the current secondary extractor, dictate the size and weight of the unit. The weight of the unit, and the position of the centre of gravity when harvesting means that considerable strength must be designed into the support structures on the machine. The criterion for light weight can then be further defined to incorporate these two major components.

- The complete unit must be of lower weight than the conventional system to reduce overall harvester weight to address issues such as soil compaction and trafficability under wet conditions. This can be achieved utilising alternative conveying technologies and via reducing secondary cleaning system weight.
- The unit must achieve a substantially more desirable position of the centre of gravity of the elevator. This can best be effected by significant reductions in the mass of the secondary elevator. By reducing weight and improving the position of the centre of gravity, the weight transfer from the outer wheel to the inner wheel is reduced while harvesting. Apart from machine mobility and compaction issues, this enhances machine stability during harvesting and during travelling.

(b) *Improved cleaning:* The secondary cleaning system that would be developed in association with the lightweight elevator must offer significantly enhanced performance over the existing system, to increase overall machine performance with respect to reduced EM and reduced cane loss in the cane supply. Effective cleaning is dependent on a number of factors including pour rate and EM levels; however, the presentation of material to the cleaning system can dramatically impact on cleaning potential. The goal was for the elevator system to present the material in such a way as to maximise the potential efficiency of the cleaning system.

(c) *Reduced impact by the elevator on the primary extractor:* The elevator bowl must enhance air flow to the primary extractor, relative to the air flow regime resulting from the current elevator bowl design. This would result in enhanced performance of the primary extractor P Hobson, SRI. pers. com.

Redesigning the current chain and flight elevator and extractor cleaning system to incorporate these advances would be extremely difficult.

For the development of a new concept for the elevator/secondary cleaning system to be acceptable to the industry, it must have no significant operational disadvantages over the current system, and must be able to demonstrate lower whole of life costs.

4.2.2 Potential system configurations

To achieve the above criteria, lightweight flexible belting systems were considered to offer the most appropriate technology. A number of potential flexible belting systems were evaluated, as listed in Table 1.

Despite the potentially significant issues relating to feeding billets into the elevator section at an acceptable, even rate, the twin hugger belt system appeared to be the option which best met the requirements for the development.

Table 1
Advantages and disadvantages of different flexible belt elevator systems

Concept	Potential advantages	Potential disadvantages	Comments
Simple trough conveyor	<ul style="list-style-type: none"> • Simple to design and construct • High conveying/delivery speeds potentially available • Non-overloading therefore simple systems to control flow of material on to belt • Some 'on belt' storage volume available 	<ul style="list-style-type: none"> • Low maximum conveying angle • Difficulty in accelerating material to high belt speeds 	Low maximum conveying angle makes the system not viable
Simple trough conveyor with slats/flights	<ul style="list-style-type: none"> • Greater angle of transfer available than simple system • Still simple system with non-overloading characteristics • Significant 'on belt' storage volume 	<ul style="list-style-type: none"> • Flights/slats increase belt loadings and significantly reduce maximum belt speeds 	Reduced maximum conveying speed means that essential potential advantages cannot be realised
Twin hugger belts	<ul style="list-style-type: none"> • Wide range of potential operational angles • Necessary high speeds possible because of simple belts • Negligible 'in belt' storage volume 	<ul style="list-style-type: none"> • Control of feed rate essential to prevent overloading • Usually simple 'straight line' systems 	Only system to meet requirements of desirable conveying speed and angle of operation

The elevator design conceptualised used a hugger belt system, fed from an elevator bowl developed to feed the hugger belts. The proposed elevator bowl design would offer enhanced air flow to the primary extractor, as well as fewer space constraints on its design. This system offered a number of potential advantages, as follows.

1. The weight reductions achievable utilising rubber belts in the proposed design allowed potential for a reduction in machine weight, centre of gravity and overhang considerations impacting on the strength and durability requirements of the machine superstructure. If the elevator was needed to be moved further to the rear of the machine (eg to facilitate fitting of some proposed cleaning chamber technologies such as NE1), the associated adverse consequences would be reduced.
2. The high-speed hugger belt conveyor was seen as potentially capable of presenting an even, thin layer of material to the secondary cleaning system at an optimised velocity. Additionally, the mixing and agitation of the billets as they are transferred on to, and conveyed by, the hugger belts, were seen as offering loosening of the attachment between leaf material and the billets (particularly important under wet harvesting conditions). It was also envisaged this process would reorientate the material for the secondary cleaning. The 'buffer storage' effect of the elevator as it feeds would reduce the variability in the material flow rate.
3. Given that all pneumatic cleaning systems have an inherent relationship between pour rate, EM and cane loss, the incorporation of more effective secondary cleaning would allow total cane loss and EM to be minimised over the widest possible range of harvesting conditions. This would be achieved by allowing the primary cleaning system to operate in a higher EM/lower cane loss mode, and removing a greater proportion of the EM by the more effective secondary system. An effective secondary cleaning system is a logical adjunct, if the cleaning ability of the current harvester is to be maximised.

With any transfer of technology there are, however, limitations. The hugger belt concept has three limitations in this application, all of which had to be accounted for in the design of the elevator system. The limitations with the hugger belt concept are:

- (a) minimal or no storage of cane material can be achieved between the two belts. On conventional elevators the flights on the inclined section can store approximately 0.15-0.25 m³ of material. With no storage volume possible on the proposed high speed elevator, and to maintain or increase overall storage levels compared with conventional elevators, an increase in the storage volume of the lower bowl section must be achieved;
- (b) the 'S'-type geometry of the conventional elevator configuration cannot be achieved using hugger belt concepts. Therefore, to obtain the desired 'S'-type arrangement the elevator must include two mechanically independent sections. The two sections are the lower section or bowl, which contains storage volume and the inclined section incorporating the hugger belt system. The top component of the 'S' could be achieved by the trajectory of the cane after leaving the belts;
- (c) the technology of feeding material as non-homogeneous as billeted sugarcane, including various amounts of trash and other material into inclined, high speed hugger belts offered considerable challenge.

The design details of the relevant components of the proposed lightweight elevator and secondary cleaning system will be discussed in the following sections.

4.2.3 Design requirements: elevator bowl, transfer zone and belt module

The initial conceptual design was performed by BSES. The design development incorporated the following considerations.

- (a) *Spatial envelope of bowl and belt module:* The constraints of physical size to allow retrofitting to existing machines. To achieve full compatibility with existing harvesters (for retrofit), the physical limits of the lower sections of the elevator were constrained to the similar physical dimensional envelope as the current elevator design. Significant gains in utilisation of available space were identified by constraining the bowl to rotation only versus rotation and tilt as for the current system.
- (b) *Conveyor component design:* The design of the mechanical components of the bowl including no specialised manufacturing requirements and low weight. Additionally, the initial conveying components to allow feed from the full bowl condition, and to achieve the optimum delivery layer of cane, being presented to the hugger belts. The ability of the bowl to unload material (at realistic pour rates) from the full condition was a major concern.
- (c) *Conveyor exit conditions:* The design of the mechanical components in the transition zone between the elevator bowl and the hugger belt system. Exit and entry conditions of the bowl and the hugger belt were seen to be critical to ensure even and continuous flow of material.

Additional considerations relating to the design included the following.

- (a) *Material storage:* The ability to store cane billets in the bowl of the elevator and meter them to the conveying system during 'cold start up', ie with a full elevator bowl. At start up, billets have to be fed from the storage in the bowl to the hugger belts at a controlled rate.
- (b) *The ability to effectively feed cane from the elevator bowl in to hugger belts running at a steep vertical angle:* Delivery of material to the hugger belts at a continuous even rate is obviously critical to the performance and operational reliability of the system.
- (c) *Material flow rate:* The current capacity of the conventional chain and slat elevator is approximately 70-90 t/h when conveying uncleaned cane (fans off treatments in BS189 and BS227 trials) and at in excess of 200 t/h in burnt cane. Therefore, it is paramount that the hugger belt system can deliver material at a rate equivalent-to or greater than these levels.
- (d) *Operational reliability and low whole of life costs:* The concept of high-speed hugger belts is used in numerous applications in other industries. The proposed design should ensure maximum reliability and a minimum whole of life cost. Flexible belting systems are used in a wide range of applications, from conveying of agricultural produce to conveying of rock and minerals, and have a high degree

of reliability under adverse environmental conditions. The conventional elevator has very high mechanical reliability. This, coupled with the ability to effectively rebuild high wear components, gives the units a life span in the order of 200,000 to 500,000 tonnes. High mechanical reliability and lower whole of life costs than the conventional chain and slat elevator are essential if the proposed design is to be successful. Similarly, the proposed design for a secondary cleaning system utilising blowers rather than extractor fans offered potential for worthwhile operating and maintenance cost savings.

4.2.3.1 Concept test rig development: elevator componentry

The development of the designs of the elevator bowl, the transition components between the bowl and the hugger belts, and the hugger belt system was undertaken simultaneously in the initial stages of the project because of the interactions that occur between components.

The design of the secondary cleaning system is discussed separately in Section 4.3 of this report.

The approach taken in the development of all components was to start with the simplest possible system and develop the design as an active learning process. For this reason, the first concept test rigs manufactured were of extremely simple design. The design layout was based on a simple, parallel sided sheet metal structure with all the components attaching directly to it.

The first concept test rig represented a simple elevator bowl incorporating the horizontal belt, and a fixed gate designed to limit the maximum rate of feed of material to the two offset parallel hugger belts positioned on fixed centres. The system is detailed in Figure 2 and Image 3.

The bowl section incorporated an agricultural conveyor belt with 'Z' surface pattern, of 900 mm nominal width, running on 160 mm rollers at 750 mm centre distance.

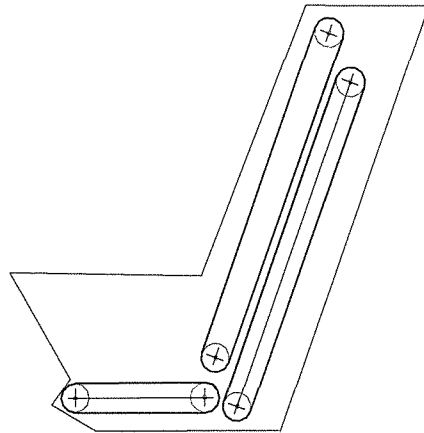


Figure 2 - Side view of concept development rig.

The inclined hugger belts ran between rollers at 1,900 mm centre distance, and were of similar material to the horizontal belt.

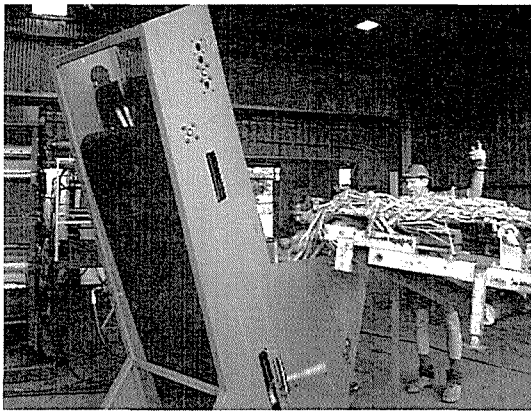


Image 3 - View of concept development rig being loaded with billeted material.

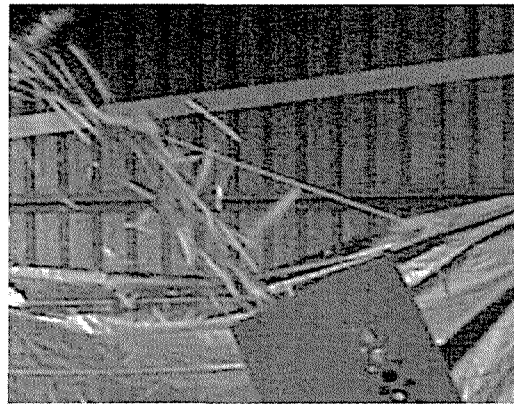


Image 4 - The billet trajectory from the hugger belts, indicating potential suitability for effective cleaning.

Initial calculations of billet trajectories by the dilute phase model indicated belt velocities in the order of 5-7 m/sec would be desirable. Discussions with belt manufacturers indicated that this velocity was acceptable providing bonded joints were used in the belts.

Drive for all rollers was by a Char-lyn hydraulic motors embedded in one end of the nominated drive roller for each belt, using similar mechanical componentry to the harvester feedrollers. Rollers were 'crowned' as per industry recommendations to minimise the chance of belt tracking problems.

4.2.3.2 Concept test rig evolution

Trials indicated that whilst the system was incapable of feeding billets from a stationary mass (simulating start up with cane in the bowl), a surprising level of success was achieved when the unit was fed at a controlled rate with billets from a pre-loaded conveyor belt. The test rig clearly demonstrated the ability of a hugger belt system to present billets in a manner ideal for further cleaning (Image 4).

The primary reasons for the unsuitability of the simple layout of components were:

- insufficient aggression in the floor of the bowl to achieve material movement to the outlet gate with any significant mass of material in the bowl;
- bridging of the cane at the outlet gate, preventing feed from the hopper, when any significant mass of material was in the bowl;
- difficulty in achieving effective feed from the exit of the bowl section into the inclined hugger belts due to the constraints on available space and the significant change in direction required;
- high levels of billet damage as the billets transferred from the horizontal belt to the hugger belt system;
- concerns that the billets were not adequately constrained between the hugger belts, and that reliable conveying would not always occur up the incline required.

A systematic program of development was then undertaken to evolve a design that met the goals of the project.

(a) Elevator bowl design and material flow control

Key issues identified were bridging of material and difficulty in controlling feed of the material into the hugger belts. An additional consideration was the need to maximise the angle of the base of the elevator bowl to minimise the transition angle of the material from the bowl into belts. A number of concepts were evaluated in an attempt to evolve a suitable concept for development into a prototype system. Although many arrangements were trialed, the concept fitted into three broad systems.

- *Scalper roller*. The concept of a scalping roller, rotating against the direction of flow of the material was to control the feed of billeted material into the transition zone and hugger belts, and to spread the billeted material evenly across the throat of the belt system. A number of different concepts were trialed. Whilst this system was effective in controlling maximum feed rates into the belt system, particularly when being fed via the pre-loaded conveyor belt, it did not alleviate the problems relating to bridging and lack of feed when the bowl was full.

- *Scalper roller and oscillating wall.* The next development was the incorporation of an oscillating wall on the elevator bowl to discourage bridging and create a 'rotation' of material in the bowl. This concept achieved only limited success.
- *Flow control rollers.* A series of eccentric rollers mounted above the floor of the elevator bowl were trialed on the basis that the rollers would support the cane mass, and allow it to feed on to the bottom belt at a controlled rate. Whilst problems were envisaged with trash wrapping around the rollers, the system appeared to work reliably, and was selected for further development.

(b) Transition zone

The transition zone is primarily aimed to change the direction of the billets as well as assisting in the acceleration of the billets to the velocity of the hugger belts. Initial trials had demonstrated the need for an appropriate transition system; however, the constraints to system geometry whilst incorporating simple belt systems created considerable difficulties.

- The concept of an intermediary roller, between the bottom feeder belt and the bottom hugger belt was developed, along with a large diameter roller on the top belt to enhance entry conditions. The initial concept had been for the transition roller to rotate with a surface speed approximately midway between the surface speed of the feed belt and the hugger belts. Trials indicated this concept was worthy of further consideration.
- A second system involving a much more complex system of a flexible backing system, which allowed a very smooth transition of the billets, was also conceptualised.

Using two-dimensional dynamic modelling software, JCU staff modelled the two different transition zone concepts incorporating the interaction between the cane billets and the conveyor components in this zone.

Whilst the forces and accelerations predicted by the model were lower for the more complex system, the modelling was able indicate ways to improve the layout of the simpler concept to enhance the flow of material from the bowl into the hugger belt system, including:

- subtle changes to component locations and, more importantly;
- ensuring the rotation of the intermediate roller was at the same surface speed as the hugger belts, rather than at an intermediate speed.

This optimisation allowed good transition of billets from the bowl to the hugger belt system. Figure 3 illustrates the 'proof of concept' elevator bowl design that evolved. A detailed account of the dynamic simulation modelling can be found in Appendix B: Dynamic Simulation of a Conveyor Belt System for Cane Harvesters.

(c) Hugger belts

Whilst the concept of hugger belts appeared to offer considerable merit, conventional belt configurations did not appear compatible with the requirement for a simple, robust agricultural elevator. Two standard features of conventional hugger belt configurations of straight line movement of material (ie straight parallel belts); and the use of flexibly mounted constraining rollers to hold the belts together to maintain material conveying effect up inclines were particular constraints to any proposed design. Designs, which complied with accepted practice, limited the belt layout to a straight configuration, or were excessively complex.

Within the constraints of the proposed high speed elevator, there was a requirement for:

- light weight and simplicity;
- the ability to feed at large differences in instantaneous flow rates, effectively meaning that some continuous adjustment in belt tension would be required;
- the desirability of being able to convey material around a curve to optimise outlet trajectories.

To minimise the constraints, a hugger belt concept that evolved, included:

- a curved backing for the lower belt, which meant that the reaction caused by belt tension in the top belt effectively constrained the material without additional constraining rollers or other systems to ensure contact between the belts and the material;
- a large diameter 'floating' lower roller. This enhanced entry conditions to the belts, allowed the flexibility to absorb 'glut feeding' events, as well as allowing the roller centres to change. Thus, near constant belt tension could be maintained while allowing changes in material thickness and the ability to cope with 'glut' feeding events.

The system developed included the top hugger belt tensioned by the large diameter tail roller spring-loaded in two planes. This allows the opening between the hugger belts to adjust to the amount of cane being conveyed and maintains automatic adjustment of the belt tension.

Whilst this arrangement did not comply with normal industry protocols for the design of high-speed hugger belt systems, all aspects of the design could be rationally argued. The advantages of the system appeared considerable over any other alternative layout considered. The system was built into the proof of concept test rig and trialed.

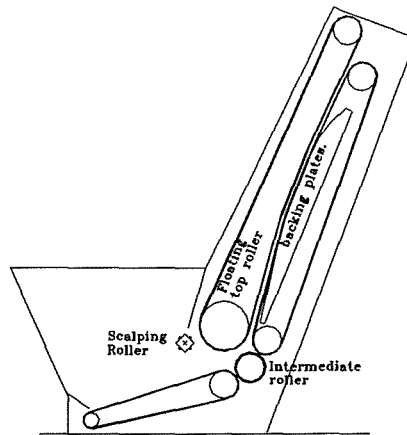


Figure 3 - A schematic of the proof of concept elevator bowl and hugger belt modules, as detailed in Images 5 and 6.

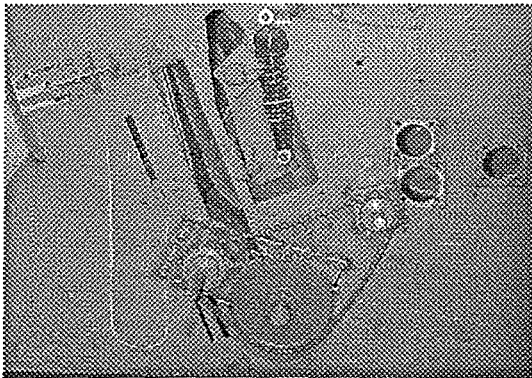


Image 5 - Details of intermediate roller, backing plate for bottom belt, spring-loaded roller for top belt and drive for scalper roller.

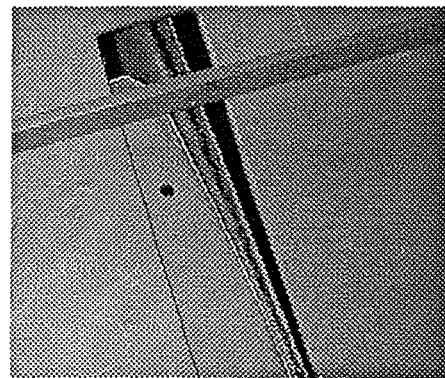


Image 6 - Details of top lower belt backing plate.

During the workshop trial program, the system demonstrated considerable promise. Whilst operational reliability of the belt system through tracking instability was the major problem encountered, it was believed that this was primarily related to difficulties in maintaining both top and bottom belt alignment. Discussions with Beltreco staff (P Birkbeck) indicated that close attention to alignment of rollers and to crowning of the rollers could be expected to overcome these problems.

4.2.3.3 Proof of concept review

The process of designing relatively simple systems to evaluate concepts and develop potential designs had progressed knowledge on the potential for a high-speed hugger belt system for conveying sugarcane billets. It also gave the stakeholders the confidence to progress to the development of a prototype system. Key results had been that:

- there was a high degree of confidence that billets could be metered from a full elevator bowl with sufficient reliability and evenness to progress the project;
- the concepts developed for the transition of the cane from the elevator bowl to the hugger belt appeared to meet required performance criteria;
- the hugger belt concept demonstrated the ability to convey material at a steep angle and to deliver the material in a thin layer suitable for further cleaning.

The next step was to develop a prototype unit suitable for field testing.

4.2.4 Prototype design

The successful development of proof of concept systems had established design concepts to be included in the prototype. The agreed distribution of tasks for the production of the prototype were as follows.

- BSES: Design, manufacture and testing the elevator bowl material flow control and transition components.
- JCU: Design the hugger belt module. It was to be manufactured by Austoft, with final assembly undertaken by BSES.
- Gough Plastics: Manufacturing of the secondary cleaning module, to a design developed in consultation with JCU and BSES.

BSES would then assemble and workshop test the prototype, and finally evaluate the field performance of the system.

4.2.4.1 Elevator bowl design and manufacture

Key elements of the design of the elevator bowl material flow control and transition components were:

- spatial design to maximise storage volume available for billeted material;
- aerodynamic design to maximise performance of the current primary cleaning system;
- ability to feed the hugger belts reliably when full of billeted material of varying compositions and flow characteristics.

Spatial constraints and opportunities

The three primary spatial constraints on the bowl design were:

- clearance of the base of the bowl over the grousers on tracked machines. The problem was the encroachment by the top of the track into the elevator area when the machine had the basecutters fully raised. This defined the lower envelope of available space;
- clearance of the sides of the bowl from the wheels of wheeled machines. This constrained width and component layout considerations;
- clearance from the machine frame to rotate, and height constraints so as to fit under the current cane cone. Some consideration was given to attaching the cane cone to the elevator bowl; however, this was rejected on the basis that it made the elevator less appropriate as a retrofit.

Given these constraints, a design was developed incorporating:

- concentric cone sections of perforated metal (12 mm diameter perforations, 46% open area) formed the side walls of the bowl. The cone sections were constructed in tiered layering, which allowed for maximum air flow into the primary extractor and to maximise storage volume;
- an inclined feeder belt as the base of the bowl. This design allowed clearance of the frame over the track grousers while also minimising the change of angle of material as it passed through the transition to the hugger belts.

The final bowl volume was approximately 0.9 m^3 , which compares well with the conventional elevator's total storage volume of $0.7\text{-}0.8 \text{ m}^3$.

Aerodynamic modelling

A proposed design was developed and subjected to aerodynamic modelling of the prototype design utilising the computational fluid dynamics code FIDAP, which was undertaken by SRI. This was carried out to investigate and optimise the air flow in the region of the primary extractor when the new elevator bowl was fitted and was undertaken to maximise the cleaning performance of the primary extractor.

A report on the modelling outcomes is presented in Appendix C: Computational Fluid Dynamics (CFD) Modelling of the Air flow around the Primary Extraction Chamber.

The outcome of this modelling was the prediction that the proposed design should significantly enhance the performance of the current primary cleaning systems.

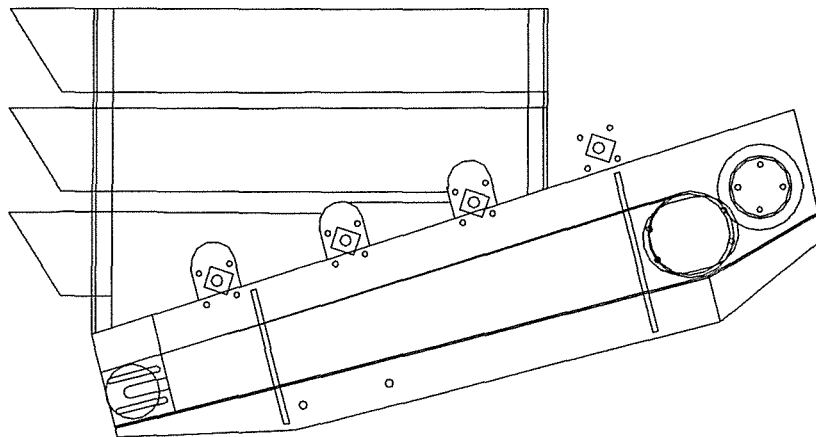


Figure 4 - Layout of components in final elevator bowl design. Note the concentric cone sides for maximum air flow to the primary extractor.

Prototype bowl design

The bowl design, which evolved to meet the requirements of the project including reliable feed at start up and adequate storage capacity, is represented in Figure 4, and in Images 7 and 8. Key features include:

- the bowl rotates only on its vertical axis, thus maximising storage capacity and the ability to optimise air flow to the primary extractor;
- three rollers to support the material above the feeder belt, yet actively feed and control the flow of material onto the feeder belt when operating;
- the counter-rotating 'scalping' roller was maintained to control the feed of cane into the transition onto the hugger belts and spread material across the whole belt, particularly when the unit was being fed from one side, as happens during normal harvesting operations.

The final prototype, which was developed as part of BS210, demonstrated the ability, in workshop trials, to empty the bowl from full at commercially acceptable flow rates.

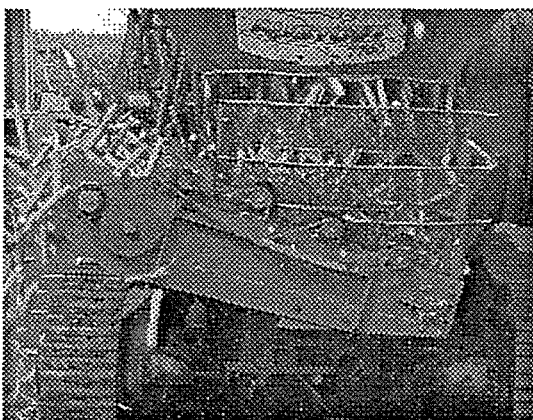


Image 7 - Rear view of bowl assembly showing support rollers and drives for scalper roller and intermediate roller.



Image 8 - A view of the bowl feeding from full. Note the openness of the design for maximum air flow to the primary extractor.

4.2.4.2 Prototype hugger belt module design

Initial proof of concept design pushed the limits of conventional knowledge. The primary concern had been achieving reliable tracking of the high-speed hugger belts. Discussions with experts in the field drew the project team to the conclusion that providing appropriate crowning was applied to all rollers, and that adequate rigidity was built into the machine frame, belt tracking should not be a problem

The design adopted for the prototype 'pushed the limits' even further in terms of the use of the hugger belt configuration, particularly with respect to the lower belt. Whilst the proof of concept test rigs had experimented with a curved backing plate, the prototype design pushed this concept further to optimise the trajectory of the material as it left the belts.

The lower belt geometry was extended from a shallow curved path (curve radius 5.0 m) to a more complex curved path (initially 5.0 m tightening to 1.0 m radius at the discharge end of the elevator), to optimise the trajectory of the billets as they exit the conveying module. Additionally, increased tension between the two belts for more aggressive holding of material was achieved using this approach.

Key points relating to the design of the hugger belt module were:

- a module belt width of 900 mm was selected. This is the same width as the conventional chain and slat elevator;
- the design layout was based on a truss frame with all the components attaching directly to it via fabricated brackets, which also enhanced the stiffness of the frame;
- the hugger belt arrangement consisted of a top belt running against a bottom belt, which in turn ran on a curved removable backing plate;
- the top hugger belt was to be tensioned by the large diameter tail roller, spring-loaded in two planes. This was to allow the opening between the hugger belts to adjust to the amount of cane being conveyed and maintains automatic adjustment of the belt tension.

On adoption of the prototype design a full detailed design was undertaken by JCU. A detailed account of the design details as well as detailed assembly drawings of the prototype elevator are presented in Appendix D: Mechanical Design of a High Speed Elevator for a Cane Harvester.

The frame and all associated components were fabricated at Austoft and installation of components was undertaken by BSES. Figure 5 illustrates the prototype hugger belt module for the high speed elevator system.

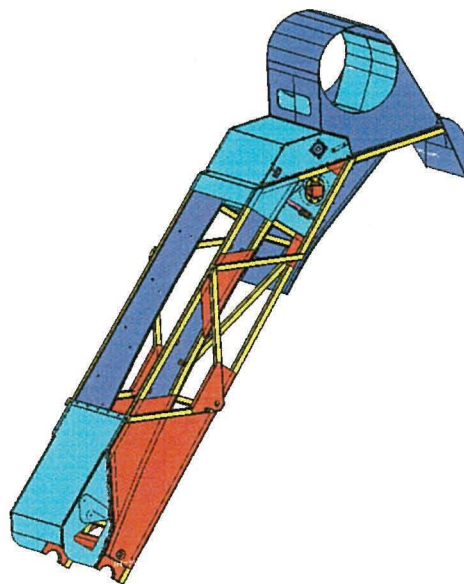


Figure 5 - Layout of frame components for hugger belt module for the high-speed elevator.

4.3 Advanced secondary cleaning system

4.3.1 System requirements

All pneumatic cleaning systems have an inherent relationship between pour rate, EM and cane loss; however, optimisation of design can enhance trash removal efficiency while reducing cane loss.

The development of a cleaning system in conjunction with a high-speed hugger belt system offered the potential to present an even layer of material to the secondary cleaning system at a velocity that could be optimised for the subsequent cleaning operation. This system was seen to allow effective cleaning from a range of potential secondary cleaning system designs.

With this system of material presentation the conventional cleaning system would be improved; however, the main objective of the project was to develop a lightweight cleaning system. Initial investigations indicated that the use of blowers for the air supply had considerable potential including:

- high efficiency;
- low maintenance as clean air passes through the fan, minimising wear;
- potentially lower weight.

The conventional extractor type cleaning system has an inherent aerodynamic efficiency in the order of 30-40% (Quick, 1982) with additional power being used to process the trash and billets passing through it. The power consumption for this function typically at least equals that required for the pumping of air (P Hobson, SRI, pers. com: J Williams, Austoft, pers com.). Typically, the hydraulic power required for the secondary cleaning system is in the order of 20-33 kW (assuming 1,500-2,500 psi operating pressure, 30.6 gpm). In contrast, well designed blower systems demonstrate aerodynamic efficiencies in the order of 80% (Bleier). Thus, if cleaning potential is related to total energy in the air flow, significant potential reductions in the energy required for the cleaning process could be achieved by a blower-based system whilst maintaining similar cleaning potential. Alternatively, more air power is available for cleaning whilst maintaining the same installed power.

The secondary cleaning section was developed in tandem with the developments for the base of the boot section of the elevator. 'Potential layouts', including a range of novel approaches were developed by the project team. The potential of each of these options was explored using simple trajectory modelling at BSES. A proof of concept testing rig was manufactured by BSES to facilitate initial performance assessment and more sophisticated computational fluid dynamics modelling at SRI.

After analysis of potential performance and an assessment of design, layout and operational considerations, the design of the best option was then further developed and refined by additional FIDAP modelling at SRI, and manufactured and tested at Bundaberg.

4.3.2 Concept development

The test rig to evaluate initial concepts was developed and manufactured by BSES. The unit incorporated an air supply, associated ducting and delivery nozzles. Billeted cane was supplied to the test rig by the elevator bowl/hugger belt module, which had been fitted with a deflector plate to give a billeted material trajectory of approximately 30 degrees off horizontal. When trials were undertaken, the hugger belt module was fed with billeted cane via a pre-loaded belt conveyor. The system was fabricated with the flexibility to determine the performance envelope of flow characteristics including air speed, flow direction, nozzle spatial arrangement and the effect this performance envelope has on the separation of trash from the billet stream exiting the high speed hugger belt system.

The air supply component incorporated two double entry centrifugal fans with backward curved blades on a common shaft, and powered by a 5 kW electric motor. This complete system had been acquired by BSES from Massey Ferguson (MF) after the closure of the Bundaberg based cane harvester division in the 1980s. Massey Ferguson utilised this unit in their cane harvester research & development division throughout the late 1970s. The fan impellers were 380 mm in diameter and performance charts of similar fans (Richardson) indicated their combined output would be approximately 6.4 m³/second of air against a head of 250 Pa at 2,175 rpm. Workshop testing indicated that an output of 7.7 m³/second was achieved against no head, which was consistent with the expectations of the unit.

The air supply system was located in a discrete unit above the proof of concept hugger belt system, and flexible ducting was utilised to deliver the air supply to the outlet components. The outlet components were interchangeable to allow testing of varying nozzle widths and locations. Images 9 and 10 illustrate the proof of concept secondary cleaning test rig fabricated at BSES.



Image 9 - Hugger belt module set up to be fed with billeted cane at a controlled rate by a belt conveyor.

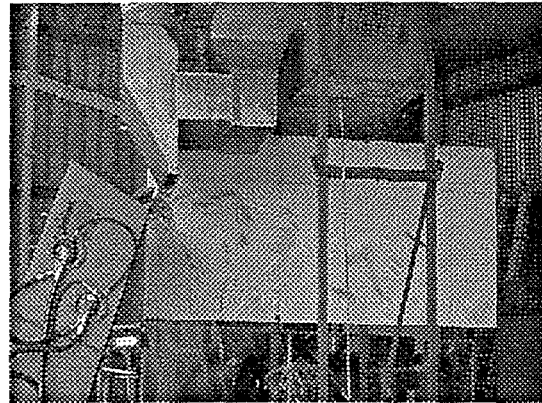


Image 10 - Overview of test facility illustrating narrow and a broad air supply nozzles in operation. The billets are deflected to the correct trajectory by a deflector plate fitted to the top of the hugger belt module.

4.3.3 Aerodynamic modelling and evaluation

Aerodynamic modelling of the proposed secondary cleaning system was investigated using a bipartisan approach. Firstly, a basic dilute phase model was utilised to investigate the air flow characteristics and their effects on cleaning performance. The system was optimised using this phase and then the findings were confirmed utilising a more rigorous computational fluid dynamics (CFD) modelling approach. A detailed account of the modelling phases undertaken are described in the following sections.

4.3.3.1 Dilute phase modelling

A model that predicted trajectories of materials with different mass and aerodynamic properties had previously been developed by the project leader, and this model was further refined for BS152. This model offered the potential to undertake preliminary investigations into probable particle trajectories in a 'dilute phase' situation, where interactions between components could be ignored. Given the proposed high 'launch' speed of particles in a proposed thin layer, it was believed that the particle trajectory model was an appropriate tool. Aerodynamic and other physical properties of cane components were derived from published data. An example of the inputs to this model are presented in Appendix E: Dilute Phase Model Inputs for Secondary Cleaning System.

Dilute phase model verification

As the concept development continued, a number of cleaning system options were investigated using the objective of this approach, which was to use dilute phase modelling and supplement it with limited laboratory testing for verification of findings. It was believed that the findings of the dilute phase modelling would be a good predictor of the performance of the cleaning system because the presentation of billets to the cleaning chamber was proposed to be as close to single layer presentation as possible.

Initial runs of the test rig indicated good correlation between the observed trajectories and the predicted trajectories by the dilute phase model.



Image 11 - The trajectory of cane billets and trash from the high speed elevator at a belt speed of 6 m/sec and pour rate of approximately 70 t/h. The grid lines are at 300 mm intervals. No air flows have been introduced.

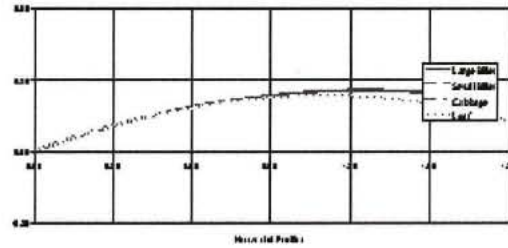


Figure 6 - Dilute phase model output of the anticipated trajectories of components of cane (large and small billets, 'cabbage' and trash) under the conditions of Image 11. (Grid lines at 300 mm intervals).

Repeating the runs at a lower belt speed of approximately 5 m/sec. (Image 12 and Figure 7) indicated poorer control over the trajectory and indications that the momentum of the billets may not be adequate to achieve delivery of the billets from the belt exit to the bin.



Image 12 - The trajectory of cane billets and trash from the high-speed elevator at a belt speed of 4.5 m/sec and pour rate of approximately 50 t/h with no introduced air flows.

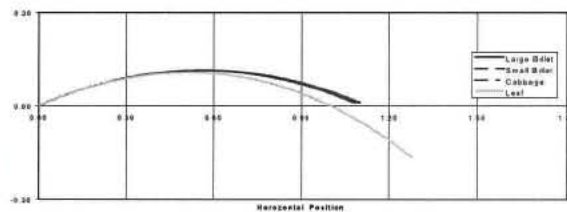


Figure 7 - Dilute phase model output of the anticipated trajectories of components of cane (large and small billets, 'cabbage' and trash) under the conditions of Image 12.

The initial concept proposed for the cleaning system was for a design incorporating a near vertical air flow, generated by a blower system, with the trash being blown downwards and in to a trash collection chute. This gave the flexibility of a wide range of potential air flow configurations, ranging from wide low velocity air streams to narrow high velocity air jets. Dilute phase and CFD modelling conducted under BS156 had indicated high velocity air streams were the most effective method to remove trash and leaf from cane; however, the option of alternative systems was still investigated for this project.

Dilute phase assessment of different cleaning options

A dilute phase model was used to assess the difference in trajectories, which could be expected from two different air flow regimes, each with equivalent air power, thus equivalent power requirements. The two options investigated were a broad downward air flow of 26 m/sec, with a chamber length of 700 mm (and chamber width of 900 mm). To achieve this air flow with a well-designed axial flow fan system would absorb approximately 9 kW. Realistically, however the actual power consumption to achieve this air flow configuration would be higher than this because of the losses associated with flow straightening, etc.

The second option investigated was two high velocity air curtains separated by a short distance. The air curtains have a width of 50 mm and an air velocity of 50 m/sec. The nominal air power was again 9 kW; however, this configuration could be more easily achieved, in a compact package, with considerably lower potential for losses. The responses predicted by the dilute phase model of the various particles to the differing air flow regimes are given in Figures 8 and 9.

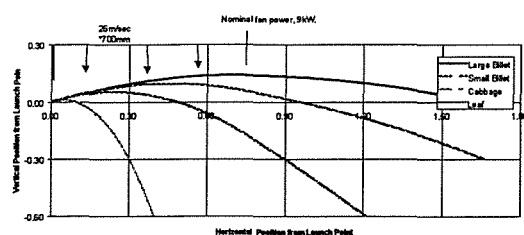


Figure 8 - Predicted impact of a low velocity air stream on trajectories of different cane particles (air power 9 kW).

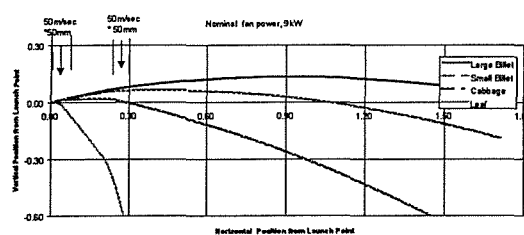


Figure 9 - Predicted impact of narrow high velocity air curtains on trajectories of different cane particles (air power 9 kW).

From this, it can be seen that the dilute phase model predicts that high velocity air curtains had less effect on the billets and more effect on the trash particles than the low air velocity system. For example, at a plane 600 mm below the entry point of the material, the separation of trash from cabbage is approximately 800 mm for the low velocity system and approximately 1,200 mm for the system utilising air jets.

Given the inherently more compact nature of a cleaning system incorporating higher velocity, lower volume cleaning system concepts, it was then decided to further evaluate high velocity options.

The characteristic of the MF test fans gave opportunity to test a range of potential velocity/width scenarios of approximately equal air power, ranging from 50 m/sec with a nozzle width of 50 mm to 20 m/sec with a nozzle width of 200 mm.

Image 13 and associated Figure 10 illustrate the actual and predicted trajectories when two vertical downwards air jets of 50 m/sec with 50mm width and 20 m/sec with 200mm width are impinged on the flow of material from the hugger belts at a velocity of approximately 6.5 m/sec.



Image 13 - The trajectory of cane billets and trash from the high-speed elevator at a belt speed of 6.5 m/sec and pour rate of approximately 50 t/h with vertical air streams of 50 m/sec and 20 m/sec as indicated.

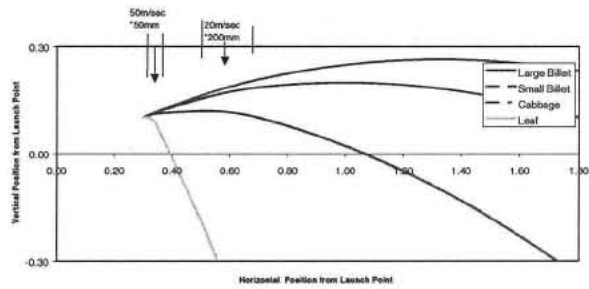


Figure 10 - Dilute phase model output of the anticipated trajectories of components of cane (large and small billets, 'cabbage' and trash) under the conditions of Image 3.

Further trials were conducted to assess the impact of alternative arrangements of the air jets. Image 14 and associated Figure 11 illustrate the actual and predicted trajectories when the second air jet is increased in velocity to approximately 30 m/sec and angled into the billet stream at 45 degrees to the vertical. The flow of material from the hugger belts at a velocity of approximately 6.5 m/sec.



Image 14 - The trajectory of cane billets and trash from the high-speed elevator at a belt speed of 6.5 m/sec and pour rate of approximately 50 t/h with a vertical air stream of 50 m/sec and a second at 30 m/sec angled at 45 degrees, as indicated

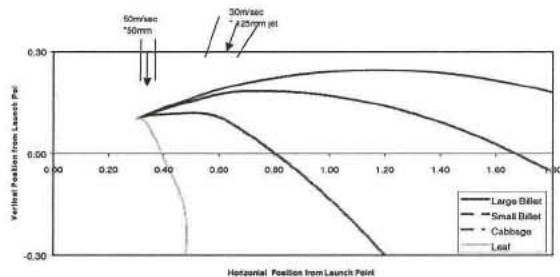


Figure 11 - Dilute phase model output of the anticipated trajectories of components of cane (large and small billets, 'cabbage' and trash) under the conditions of Image 4.

The model output and still image, along with video recordings, of the trajectories both indicate that the separation of trash from the billeted material was enhanced by directing the air jet in to the billet stream more aggressively.

Optimum cleaning strategies indicated by dilute phase modelling

The dilute phase modelling, therefore, indicated that a high level of cleaning performance could be achieved from relatively narrow high velocity air flows, across the trajectory of cane billets travelling at high speed. Both the dilute phase model and the workshop tests indicated that directing the air stream to cross the material stream at a relatively acute angle enhanced the apparent separation of EM from the billet stream.

The dilute phase model was then used to predict the material trajectories when air flows and velocities, which could be expected from a commercial axial flow fan arrangement, were imposed. The trajectories are presented in Figure 12.

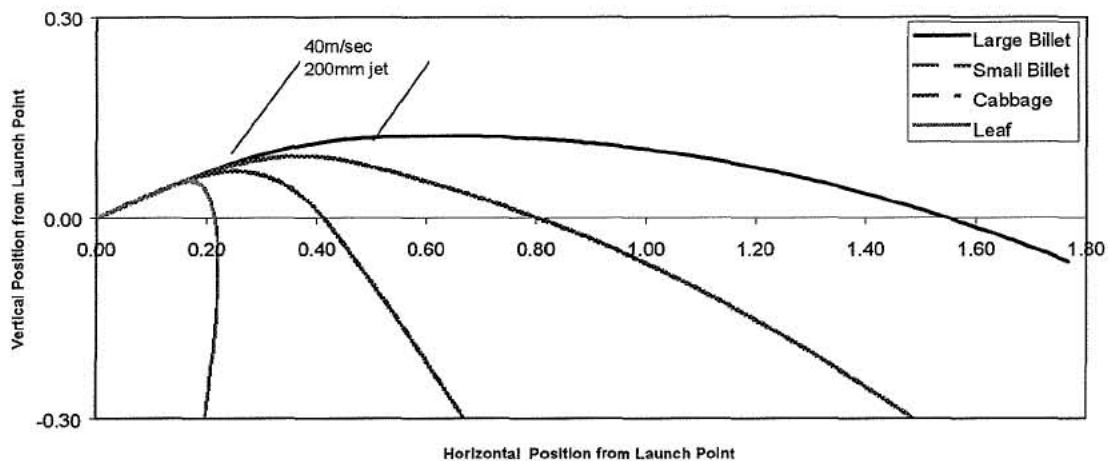


Figure 12 - Trajectories for components predicted for a transverse air jet of 40 m/sec, 200 mm width and at an angle of 45 degrees to vertical.

The aggressively diverging trajectories of trash and billets indicated by the dilute phase model, when a broad transverse air jet was applied across the billet stream, indicated this system offered potential for enhanced cleaning with minimal cane loss.

The SRI CFD models were then used to validate and further develop the design of the secondary cleaning system.

4.3.3.2 Computational fluid dynamics (CFD) modelling

Further detailed modelling of the secondary cleaning system using CFD was undertaken by SRI. Initial CFD modelling of options confirmed the dilute phase and trajectory model predictions of the performance benefits of high velocity transverse air flows when the cane billets are travelling at high velocity.

The CFD modelling has focused on optimisation/verification of designs that were developed from dilute phase modelling and trials.

CFD trajectory model set-up

The trajectory model developed under SRDC project SRI16S and SRI (mill funded) project 1293 was modified and used to investigate a number of options for cleaning cane off a rapidly moving (~ 6 m/s) elevator. The model assumed dense phase conditions and simulated both aerodynamic forces on and impact forces between cane particles. Hobson (1995 and 1996) provides a detailed account of the model structure and validation. The model in its initial form was configured to investigate cleaning within a confined duct (primary cleaning systems). The modifications carried out as part of the current investigation included:

- the removal of ‘wall’ boundary conditions other than those representing the ground and a plane corresponding to the height of the air jet source;
- the implementation of a free jet. This was configured such that the direction and magnitude of air velocities within as well as the height and width of the jet could be varied as required. For simplicity, the envelope of the jet was assumed to be vertical regardless of the angle of the air flow. This arrangement was considered adequate at this stage in the investigation;
- an additional wall boundary was incorporated to simulate a separation plate for the billets and trash. The height and vertical positioning of the plate can be set at any required value. The program keeps a tally of the number and type of cane particles landing on either side of this partition and by this means determines EM in the cleaned cane supply and cane loss. This facility has not been used in this initial investigation, although it is anticipated that it will be used at the more detailed design stage.

Simulated cleaning strategies using CFD

A number of different cleaning strategies were investigated. These have involved an evaluation of the effect of jet width, air velocity, position from end of the elevator and angle of the air jet. The parameters varied are illustrated in the schematic below where:

W	=	Jet width (m)
V	=	Air jet velocity (m/s)
x	=	Horizontal position of the nearest plane of the jet envelope (m)
A	=	Angle of inclination of the air jet measured in an anticlockwise direction from the horizontal (degrees). So a jet directed vertically downwards has a value of $A = 90$ degrees

In varying the width and velocity of the air jet, a constant volume flow rate of $7.5 \text{ m}^3/\text{s}$ was assumed. The jet was also assumed to act across the full width of the conveyor (0.9 m).

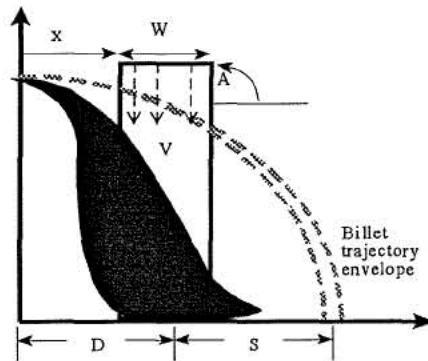


Figure 13 – Generalised diagram of modelled air flow and billet and leaf trajectories for CFD modelling.

A pour rate of 100 t/h was simulated with a trash level of 11% equivalent. This latter figure assumes some (low level) cleaning has already been carried out by the primary system. A range of leaf characteristics is present in any cane supply presented to the cleaning system. The leaves simulated in the following analysis have aerodynamic characteristics typical of 60% by mass of the leaf present in a common variety (Q124).

The simulated cane launch speed, angle and launch height were set at 6.5 m/s, 30 degrees and 2.7 m, respectively, to coincide with the proof of concept system.

The ten configurations considered are given in Table 2.

TABLE 2
Details of secondary cleaning configurations

Configuration	W (m)	v (m.s ⁻¹)	x (m)	A (degrees) [+ magnitude relative to launch angle]
1	0.238	35	0	90 [60]
2	0.119	70	0	90 [60]
3	0.476	17.5	0	90 [60]
4	0.238	35	2	90 [60]
5	0.119	70	2	90 [60]
6	0.476	17.5	2	90 [60]
7	0.238	35	0	60 [30]
8	0.238	35	2	60 [30]
9	0.238	35	0	30 [0]
10	0.238	35	2	30 [0]

CFD modelling outputs

The results are presented in terms of two measurements taken from the simulated trajectory data. These are:

- D (m) The horizontal distance between the launch point ($x = 0$) and the estimated centre of the leaf envelope at ground level. This is intended to give a measure of the relative 'compactness' of the cleaning system.
- S (m) The horizontal distance between the estimated centre lines of the leaf and billet envelopes. This dimension gives a measure of the relative effectiveness of the configuration in separating the two streams.

The simulated particle trajectories for configurations 1 to 10 are given in Figures 14 to 23, respectively.

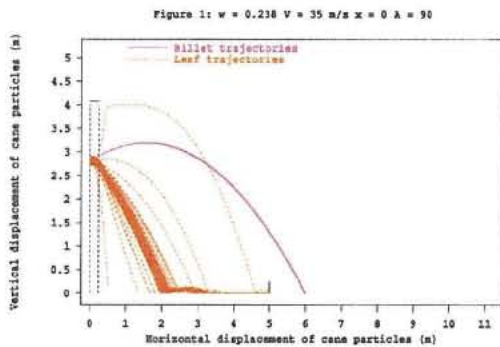


Figure 14 - Modelled trajectories, configuration 1

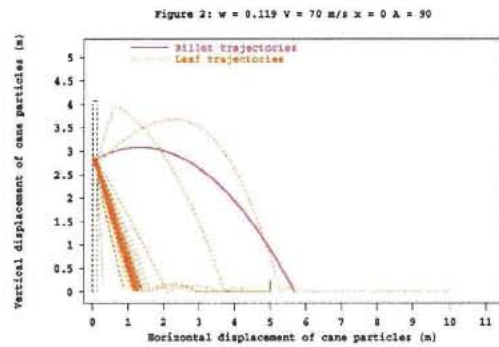


Figure 15 - Modelled trajectories, configuration 2

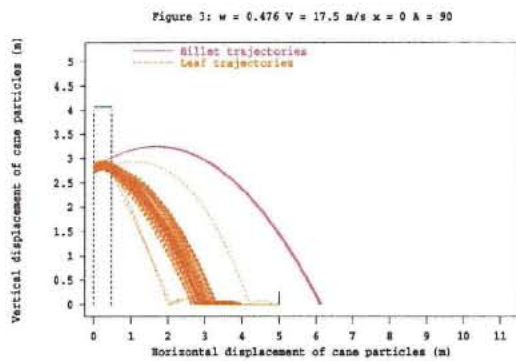


Figure 16 - Modelled trajectories, configuration 3

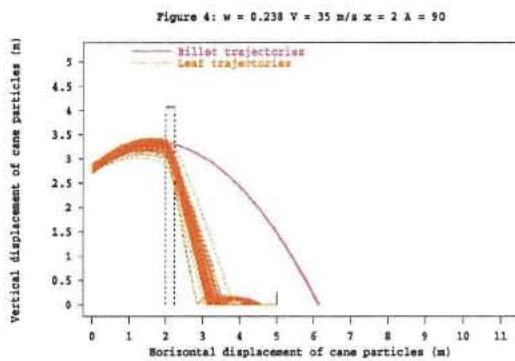


Figure 17 - Modelled trajectories, configuration 4

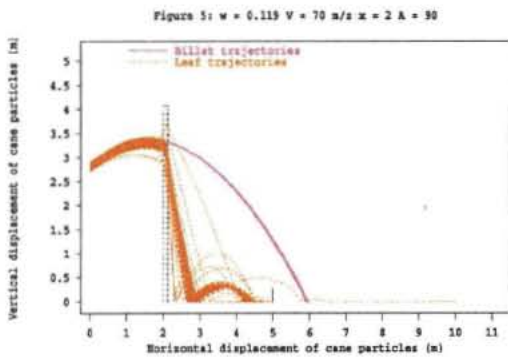


Figure 18 - Modelled trajectories, configuration 5

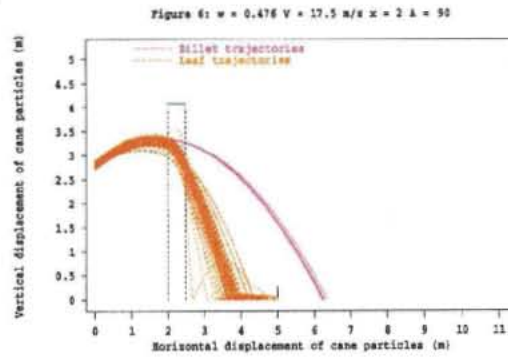


Figure 19 - Modelled trajectories, configuration 6

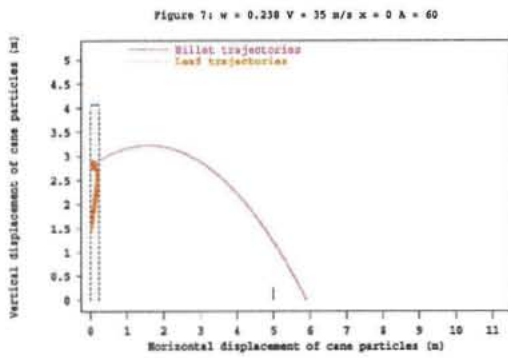


Figure 20 - Modelled trajectories, configuration 7

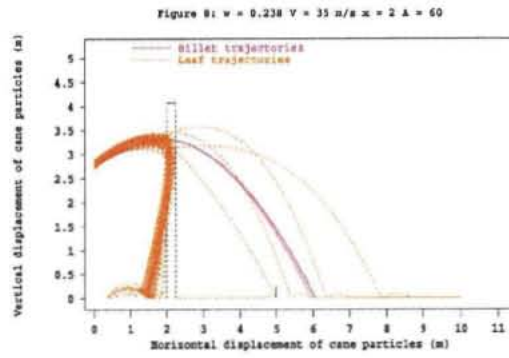


Figure 21 - Modelled trajectories, configuration 8

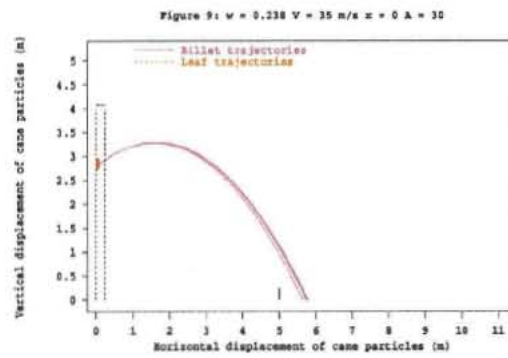


Figure 22 - Modelled trajectories, configuration 9

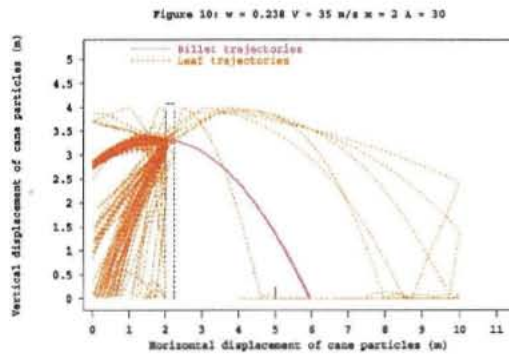


Figure 23 - Modelled trajectories, configuration 10

Measurements taken from these figures are given in Table 3.

TABLE 3
Predicted cleaning characteristics

Configuration	D (m)	S (m)
1	2.1	3.9
2	1.2	4.6
3	3.0	3.2
4	3.3	3.0
5	2.8	3.2
6	3.8	2.5
7	<0.1	>5.8
8	1.5	4.5
9	<0.1	>5.7
10	0.5	5.5

Conclusions and recommendations from CFD modelling

For the secondary cleaning configurations considered, the following conclusions are drawn from Figures 14 to 23, and Table 3.

- For the same volume flow of air, a narrow high velocity jet (configuration 2) produces a more compact (low D-value) system with better separation (high S-value) compared with the lower velocity broader jet (configurations 1 and 3).
- Positioning the air jet closer to the launch point (configurations 1 to 3) gives improved compactness and separation compared with more remotely positioned jets (configurations 4 to 6). The latter (configurations 4 to 6) were an attempt to utilise the more rapid loss of momentum of the leaves during free flight prior to applying the air jet. This effect can be seen by the greater deflection of the leaf by the same air jet if the stream is allowed free flight before entering the jet (compare Figures 1 and 4, 2 and 5, 3 and 6). This greater deflection is due to a reduced component of horizontal leaf velocity before the jet is encountered. Despite this increased angle of deflection, the net effect is a reduction in cleaning effectiveness (higher D-value, lower S-value) relative to a configuration in which the air jet is applied closer to the elevator.
- Inclining the air jet to oppose the initial direction of the billet motion (configurations 7 to 10) significantly improves separation and compactness for the same jet width and velocity (configurations 1 and 4). Table 2 indicates that the air jet directly opposing the billet motion (configuration 10, ie $A = 30^\circ$ [0° relative]) gives improved cleaning relative to the less steeply inclined air jet (configuration 8, ie $A = 60^\circ$ [30° relative]). Inspection of the leaf trajectories, however, indicates a greater degree of random scattering at lower angles. This occurs as the direction of leaf which has just entered the air jet gets 'blown back' along a path which is closer to that of the leaf still approaching or just entering the air jet. This causes a build up in the concentration of leaves within the air jet. This phenomenon has been observed in practice in tests run on the 'horizontal' cleaning chamber at SRI. This effect results in either cane loss or the carry-over of 'lumps' of trash as the trash builds up on billets.

- This increased interaction (and therefore reduced cleaning effectiveness) at ultra-shallow air jet angles combined with the competing effect of improved cleaning at moderate air jet angles, suggests an optimum cleaning angle. The buildup of leaf concentrations at ultra-shallow angles is such that it exceeds the levels of particle-particle interaction for which the model will produce reliable results (the model does not indicate the cane loss or EM carry-over observed in practice under these conditions). This potential optimum has not, therefore, been investigated using the model.

The initial analysis indicates the following features should be implemented or investigated as cleaning options for the high speed elevator.

- A single, narrow, high velocity air jet should be utilised. A simulated air velocity of 70 m/sec through a 238 mm duct did not produce any appreciable cane loss.
- The air jet should be positioned as close as possible to the cane stream emerging from the elevator. This and the above recommendation is in preference to a number of air jets positioned along the cane billet trajectory path (as has been discussed).
- The jet should be inclined so that it opposes the direction of cane motion. An air jet angle of 60° applied at the cane launch point produced improved cleaning with a relatively low indicated degree of trash build up in the cleaning chamber. The cleaning may be improved even further with a directly opposing (30°) air jet. The model, however, indicates that this latter option gives a high degree of indicated trash build up and potential cane loss or carry over of EM. An optimum angle should be investigated (experimentally) between these two (60° and 30°) jet configurations.

4.3.4 Prototype design

4.3.4.1 Design objectives

The design scope of the secondary cleaning system is now discussed. The primary design parameters were defined by the following criteria.

- (a) Lightweight. The mass of the new cleaning system should be kept to a minimum to effect a forward and downward shift of the centre of gravity and to reduce the overall mass of the harvester.
- (b) Physical size. The physical size should be kept to a minimum and be no greater than the conventional elevator for minimal problems with height regulations, infield and off field travel.
- (c) Minimise whole of life costs. The design should ensure minimum whole of life costs for the system including no specialised manufacturing requirements and minimal maintenance costs to the end user.

4.3.4.2 Air supply module development

The CFD and dilute phase modelling had indicated the desirability of a broad air jet of moderate to high velocity, typical of the output of a centrifugal flow fan rather than an air stream of the lower velocities usually associated with an axial flow fan. The length to width requirement of the air jet, along with the desirability of parallel air flow also made an axial flow fan in conventional configuration not the fan system of choice.

Centrifugal fans offer an efficient method of achieving the required air flow stability and ease of ducting to achieve the desired air jet configuration; however, to incorporate a fan of this type into this cleaning system would be extremely difficult because of the weight and size of suitable fans. For example, the most space efficient and lightweight fan assembly to meet the air flow requirements would be two double-sided 'Richardson CYD 365' fans, on a common axis similar to the arrangement of the fans used for the MF fan units. Two fans configured in this way would weigh approximately 260 kg and have a side profile of 855 mm by 725 mm. Transition ducting, etc would add to the size of the fan assembly.

Alternative methods were then sought to meet the requirements of high efficiency, light weight and robustness, whilst minimising space requirements.

The design that evolved was based on a combination of axial and centrifugal air flow principles. Twin axial flow fans on a common shaft supply air to each end of a concentric cylinder, the air exiting this cylinder via a tangential draw-off duct. In this arrangement, the air swirl component generated downstream of the fans can be captured and utilised via the scroll effect of the cylinder and draw-off duct. This effect is used to advantage on some small hovercraft, and is also the basis for the MF 405 cleaning system.

In order to confirm that the concept could be designed to give an air jet with appropriate air velocity and acceptable evenness of velocity profile, a concept test rig was constructed using a single 600 mm diameter axial flow fan to represent one-half of the proposed system. Images of this proof of concept test rig are presented in Images 15 and 16. A number of different baffle and scroll systems were trialed in the cylinder to assess the impact on air flow profiles; however, the most even air velocity profiles were achieved with no internal baffles or scrolls.

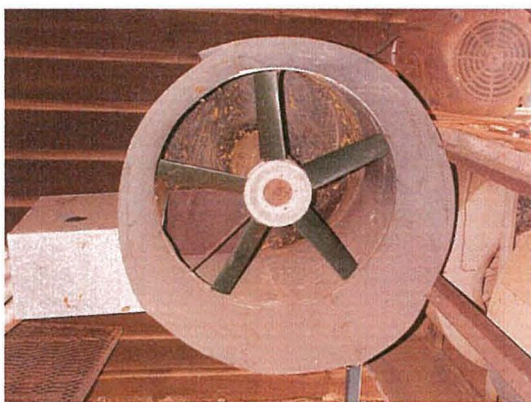


Image 15 – 600 mm diameter axial fan set up in a concentric cylinder with the tangential draw-off duct.



Image 16 - The tangential draw-off duct with grid wires to facilitate measurement of velocity profiles.

After the suitability of velocity profiles was established, design proceeded on the secondary cleaning fan module.

4.3.4.3 Secondary cleaning module layout

The final design of the cleaning system incorporated two off the shelf industrial axial flow fans. The specifications included one left- and one right-hand, 9 bladed, 30 degree pitch blades, constructed from lightweight polypropylene. Both fans are 675 mm in diameter.

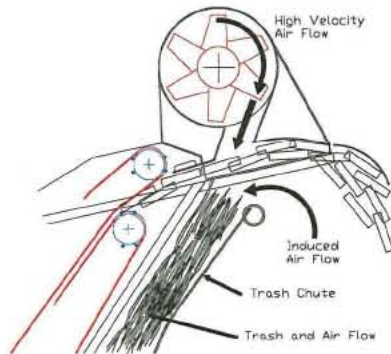


Figure 24 - Sketch of prototype secondary cleaning module layout.

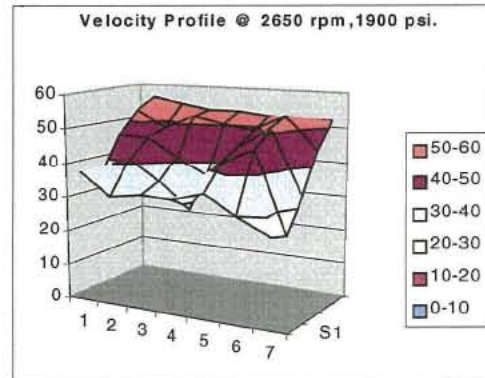


Figure 25 - Measured velocity profile (m/sec) from the outlet of the tangential duct on prototype.

The fans are driven at 2,650 rpm, using a Commercial M50 motor, similar to that used in the conventional secondary extractor, but with a smaller capacity cartridge in lieu of the cartridge usually fitted. The fans operate on the same oil supply circuit as the current secondary extractor. Hydraulic circuit pressure was measured at 1,900 psi which is similar to or lower than that expected when the conventional secondary extractor is operating.

4.3.4.4 Cleaning module design evaluation

This system delivers an air curtain that is the width of the elevator and approximately 250 mm long. The air velocity profile across the width of the air curtain is highly uniform, with a graduated velocity profile in the direction of travel of the billets. The measured velocity profile is presented in Figure 25. The billets pass through this air curtain, the trash exiting via the trash duct to the ground (Figure 24). The cylinder and draw off-duct were fabricated from plane sections by Gough plastics, with the intention of using rotational moulding to manufacture production units. The prototype cylinder, fans and draw-off duct are contained in a lightweight steel frame and the complete unit with associated hydraulic circuitry weighs approximately 102 kg. Attached to the rear of the unit is a hydraulically operated rectangular convex flap to direct billets into the receival bin.

CFD modelling of the completed design focused on the final specification of the layout, as well as on the characteristics of the (trash and air) flow as it leaves the cleaning chamber (Appendix F). The modelling predicted that both high levels of cleaning would be achieved, along with stable flow of the trash to the ground with limited ducting.

5.0 RESULTS AND DISCUSSION

After fabrication of the hugger belt module at Austoft and secondary cleaning module by Gough Plastics, these modules were mated to the bowl of the elevator in the workshop. Commissioning trials after assembly were undertaken with the elevator mounted on a base, that had been previously designed for trials on the configuration of the elevator bowl in the workshop. Image 26 illustrates the unit mounted on the base in the workshop during the testing program and Image 27 gives an overall view of the assembly mounted on the harvester.



Image 26 - The elevator during workshop testing. The PVC pipes are used to duct air to between the belts to reduce trash buildup on the tail rollers. The shade cloth 'bag' collects trash from the trash duct during trials.



Image 27 - Overall layout of the elevator bowl, elevator and secondary cleaning system mounted on the harvester.

Because of the previously discussed delays, the testing program was behind the target schedule for the project. Initial trials, with the aim of 'fine tuning' the unit before fitting it to a harvester for field testing, could not therefore be undertaken until mid October. Billeted cane was supplied to the unit via a conveyor system, giving controlled feed at predetermined pour rates.

Initial test runs indicated major problems with belt tracking instability, thus requiring continual adjustment of belt alignment. Some misalignment was found in the main frame and this was partially corrected, although to achieve total repair would have required substantially more aggressive modifications. Considerable effort was expended in the relocation of idler rollers to alleviate the belt tracking problem and significant gains were

made. The reliability of belt operation was not, however, of the order required to allow field trials to proceed. Substantial further development was necessary before a very limited field trial program could be conducted.

This section reviews the final design/performance of the unit with respect to key performance criteria.

5.1 Overall weight and weight distribution

The design and finite element analysis undertaken by JCU has likewise clearly demonstrated that appropriate engineering design can result in a strong, rigid, lightweight structure. The weight of the complete system (with secondary cleaning system) is approximately 1,000 kg, compared with over 1,300 kg for the conventional elevator. Most significantly, approximately 185 kg of this weight reduction is in the secondary cleaning system (approximately 100 kg vs 287 kg). It is confidently anticipated that, with further development, including the fabrication of more components from plastics, the final weight will be further reduced.

Since most of the weight reduction in the current design is associated with the secondary cleaning system, the centre of gravity for the high speed elevator is closer to the harvester than for the conventional elevator. The resulting 30% reduction from approximately 31 kNM to approximately 22 kNM in the overturning moment is even more important from harvester operational considerations. This 30% reduction very significantly enhances the stability of the machine, particularly in sloping conditions. The final design weighs approximately 70% of the weight of the conventional elevator, but with a substantially more desirable position of the centre of gravity, further enhancing the advantage of the reduced weight.

Comments from operators of the harvester are extremely positive as to the noticeable improvement in machine stability both while harvesting and during road travel.

5.2 Elevator bowl performance

5.2.1 Enhancement of primary cleaning system performance

CFD Modelling by SRI indicated that the design of the elevator bowl adopted would enhance the performance of the primary extractor. This has not yet been confirmed by any appropriate trial process; however, limited field observations indicate that the performance of the primary extractor has been enhanced.

The data from the limited field trial program discussed in Section 5.4 of this report indicate the performance of the primary extractor to be significantly better than expected under the conditions prevailing.

5.2.2 Start-up performance: full elevator bowl

At start up, billets have to be fed from the 'storage' in the elevator bowl to the hugger belts at a controlled rate. The design of the elevator bowl that was developed has given reliable

feed at start up under a very wide range of EM levels and operating conditions, even when the material is wet or contains high proportions of mud and stool.

The current design is not, however, the final solution. The pour rate at start up with a full elevator bowl is limited to approximately 50 t/h because of the interactions between the cane and the feed control rollers. As the hopper empties and the billet mass becomes more 'fluid', the emptying rate increases significantly. The maximum pour rate achievable by the elevator with the feed control system fitted to the elevator bowl is approximately 100 t/h with semi-clean cane.

Whilst achieving the goal of light weight, the elevator boot design incorporating a belt as the base to feed the cane suffers from three primary constraints, namely:

- achieving sufficiently aggressive feed;
- belt tracking issues; and
- collection of material between the belts, causing buildup on the rollers and further operational problems.

Although the supply of air from the secondary cleaning fans somewhat mitigated the problem of buildup of material between the belts, it was only a partial solution.

It is now believed that viable alternatives exist to the use of the flexible belt on the bottom of the elevator bowl, with alternative systems allowing more aggressive feed and greater mechanical reliability.

Although the performance criterion has been met, further development would allow increases in pour rates, whilst further enhancing evenness of feed, resulting in better performance of the cleaning module.

5.3 Hugger belt system performance

5.3.1 Hugger belt billet entry

The ability to feed billets from the elevator bowl into hugger belts operating at a steep vertical angle effectively was seen as a key requirement for the success of the project.

The use of 'proof of concept' test rigs, and utilisation of results from the particle model at JCU, have allowed the development of a design that has achieved the reliable feed of billets into the hugger belts at a wide range of feed rates.

In the light of constantly increasing knowledge, further optimisation is clearly possible.

5.3.2 Hugger belt capacity

The feed control system to allow the elevator to start with a full elevator bowl restricts maximum 'steady state' pour rates of semi-cleaned cane to approximately 100 t/h. To assess the performance of the hugger belt system, the feed control system was removed

and the elevator bowl belt was directly supplied with material via a pre-loaded conveyor belt (see Image 29), giving controlled feed at predetermined pour rates.

In trials, the hugger belt system demonstrated the capacity to convey uncleaned cane (including tops and trash) at a total material flow rate of over 100 t/h during trials and individual runs of over 130 t/h. This is very significantly higher than the capacity of the current chain and flight elevator, which has a maximum capacity of approximately 70-90 t/h when conveying uncleaned cane 'fans off' treatments in BS189 and BS227 trials (Whiteing, pers. comm. 2001). The maximum pour rate in clean cane has not been determined. It is anticipated it will meet requirements.

5.3.3 Hugger belt reliability

The prototype was initially trialed under controlled workshop conditions. These trials highlighted a major problem with belt tracking, with both the top and bottom belts running off-centre and demonstrating significant tracking instability. Repeated adjustment of roller alignment provided only temporary correction before the belts again ran off-centre.

Belt tracking instability had been a significant issue with the initial proof of concept module, even before the backing plate system had been installed. After discussions with P Birkbeck (Beltreco), it had been confidently believed close attention to alignment of rollers and the careful attention to optimisation of the 'crowning' of rollers would mitigate the problems. At this stage it was believed that the backing system under the lower belt was not a contributor to the belt tracking problems.

The project team had been sufficiently confident that belt tracking instability would not be a problem and, in the design of the final prototype, it had been agreed to 'push the limits' even further. The lower belt configuration was therefore used to optimise the trajectory of the billets as they exited the conveying module.

The belt tracking instability caused a number of operational problems in the prototype.

- The edges of the belts rubbed against the side structures, causing rapid wear of the belts, as well as accelerated wear to the plastic wear plates.
- The combination of off-centre belts, exacerbated by belt wear, allowed billets and other material to fall down and be trapped between the forward and return sides of the belts, as well as causing rapid buildup on the drive and idler rollers. This then further exacerbated the belt instability problem.

The result of this was that the reliability of belt operation was not of the order required to allow field trials to proceed as anticipated. A major redesign of the hugger belt system was deemed necessary to overcome operational difficulties relating to belt alignment.



Image 28 - Belt tracking instability allowed the belts to contact the sides of the frame, causing rapid wear. This then allowed material to get between the belts as well as build up on the backing plate and rollers.

5.3.3.1 Short-term strategies undertaken to facilitate testing

The problem was attributed to a number of areas including the following.

1. *Frame misalignment.* On inspection of the main structural frame it was discovered that the frame had been manufactured 'out of true' and the head and tail rollers could not be set up to be parallel. Minor modifications to the frame allowed the misalignment to be reduced; however, the problem could not be totally alleviated without major corrective surgery.
2. *Active trash removal from 'behind' the belts.* The build up of trash and other material around the tail rollers of each of the belts (and subsequent build up on the tail rollers) was reduced by ducting air from the secondary cleaning system to blow loose trash from between belts (see Image 27 and 28). This solution was most effective in dry conditions.
3. *Idler roller location.* The idler rollers in both belts were relocated on the basis of minimising adverse impacts of misalignments if dirt build up occurred on the rollers. Some mitigation of the problem was achieved, although not sufficient to achieve reliable operation.
4. *Roller crown.* Initial advice on the degree of crowning required on the rollers indicated a crown of approximately 2-3 mm (diameter) was ample for the application. Subsequent advice was that a crown of approximately twice the initial magnitude was acceptable. Key rollers were modified to the more aggressive crown, but with limited impact on the problem.

The combination of these actions controlled the belt tracking problem to a sufficient degree to allow a workshop performance testing program to be undertaken, along with a very limited field trial program.

Workshop trials characterised the performance of the integrated hugger belt/secondary cleaning system, by feeding it with billeted cane at controlled pour rates via a conveyor system. The first pass represented using the system as a primary cleaning system and the second and/or subsequent passes as secondary or tertiary cleaning system. The results of these trials are detailed in Section 5.4.

5.3.3.2 Potential solutions to belt tracking instability

At a meeting of the project team an analysis of the problem was undertaken. The hypothesis which best explained the characteristics of the problem was that small changes in the coefficient of friction between the belts and the low friction backing plate (caused by moisture, dirt or temperature effects) caused instability in belt tracking. It was agreed that we had 'pushed the envelope too far' in the way that we had attempted to use the belt backing system to also act as the system to achieve the required trajectory of the cane as it exits the belts.

It was agreed that, given the problems encountered, it was appropriate to adopt a more conservative approach to the belt layout. It was agreed that a redesign of the top section of the machine (deflector plate, secondary cleaning system, etc) to accommodate the simplified belt layout was needed, and that this was outside the scope of the current project.

On the basis of the knowledge gained in the project to that time, it was confidently believed that a design could be developed which would achieve the goals of the project, ie the development of a hugger belt system with lower whole of life costs than the current chain and slat elevator system.

5.4 Cleaning system performance

Given the limited available time before the end of the season, the imminent departure of the project leader overseas, and the belt tracking problems, which did not appear to have an immediate solution, the proposed field testing program presented some difficulties.

It was then decided that rather than undertake a field test program, which would probably be plagued with reliability issues, more was to be gained by undertaking preliminary evaluation of the performance of the system in the workshop, where the anticipated belt reliability problems would be less of an issue.

5.4.1 Workshop testing

Trials were conducted to characterise the performance of the system by feeding it with billeted cane at controlled pour rates via a conveyor system. The test protocol used was as follows.

- The cane used was a light crop of Q124 (60-70 t/ha), with a stalk diameter of 20-30 mm. Initial EM levels (trash plus tops) was approximately 25% (% of total weight).

- Cane was hand-cut, fed through a stationary harvester where it was billeted to a length of 250-275 mm, weighed and loaded onto a variable speed conveyor feeding the bowl of the elevator. By adjusting the speed of the conveyor and the loading per metre of conveyor, the pour rate at which the elevator was fed by the conveyor could be controlled. During test runs, the actual time to feed the nominated mass of cane is measured so actual pour rates could be accurately determined. In all tests billeted full cane was used, ie tops and green leaf were included in the billeted material.
- Typically 250 to 300 kg of material are used for each test.
- The first pass using 'whole cane' represented using the system as a primary cleaning system, and was primarily designed to produce material for testing the system in a secondary cleaning configuration.
- Second and/or subsequent passes assessed the performance of the system as secondary or tertiary cleaning system.
- The cane and trash were not 'reused', ie separated billets and trash were not 'reconstituted' for reuse. This approach was believed to be necessary to maximise the 'reality' of the test program.

After initial testing, minor modifications were undertaken to mitigate potential problem areas, particularly with respect to billet trajectories and clearing of trash from the cleaning module.



Image 29 - The elevator bowl being fed by pre-billeted whole cane. Pour rates of whole cane in excess of 100 t/h were achieved.



Image 30 - Test 7b. The billets and trash exit the hugger belts and pass through the air curtain, with the trash being taken down the trash chute via the induced air stream.

5.4.1.1 Primary cleaning simulation: 'full trash' cane

The unit was not intended to function in a primary cleaning role; however, its performance in this role was benchmarked. Figure 27 presents data on the cleaning efficiency of the high-speed elevator system when operating as a primary cleaning system, ie when operating with cane with high trash levels. Cleaning efficiency is calculated as the trash removed by the cleaning system as a percentage of the initial trash level. Because the system is not meant to operate as a primary cleaning system, conservative pour rates were used in all but trial 8a, in which a pour rate of 100 t/h was used. The tests were actually conducted to pre-clean cane for secondary extractor tests.

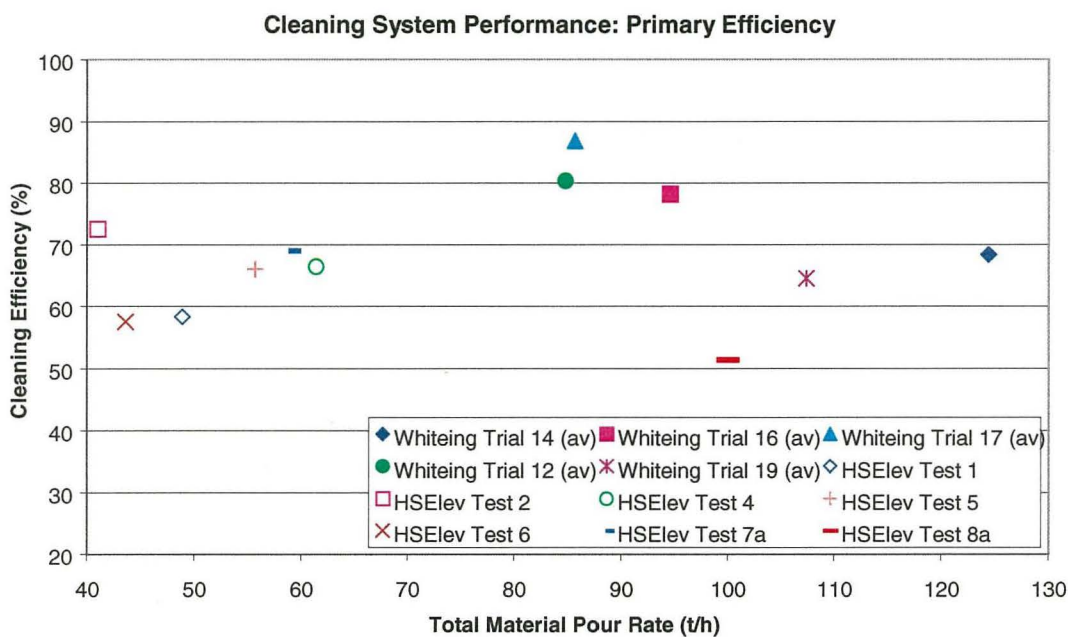


Figure 27 - Performance of the high-speed elevator system when operating as a primary cleaning system, expressed as % of initial trash removed.

In a primary cleaning role, Figure 27, the efficiency of trash removal for the high-speed elevator system was, as can be expected, pour rate related (70% @ 60 t/h, 52% @ 100 t/h). Despite the compact dimensions of the cleaning module, cleaning efficiency was impressive.

By comparison, trash removal efficiencies recorded by Whiteing (in the nominated trials associated with BS189) range from 80-87% (85 t/h) to an estimated 60-70% at 120 t/h. All of the Whiteing trials nominated were under dry harvesting conditions and represent 'above average' performance of the standard harvester.

In the 'sugar balance' trials undertaken by BSES in conjunction with Mulgrave Mill, the removal of trash (leaf material) achieved by the cleaning systems (both primary and secondary) on the 42 harvesters in the trial program ranged from approximately 30% to 80%, under the range of harvesting conditions monitored.

Primary Cleaning System Performance: Cane Losses

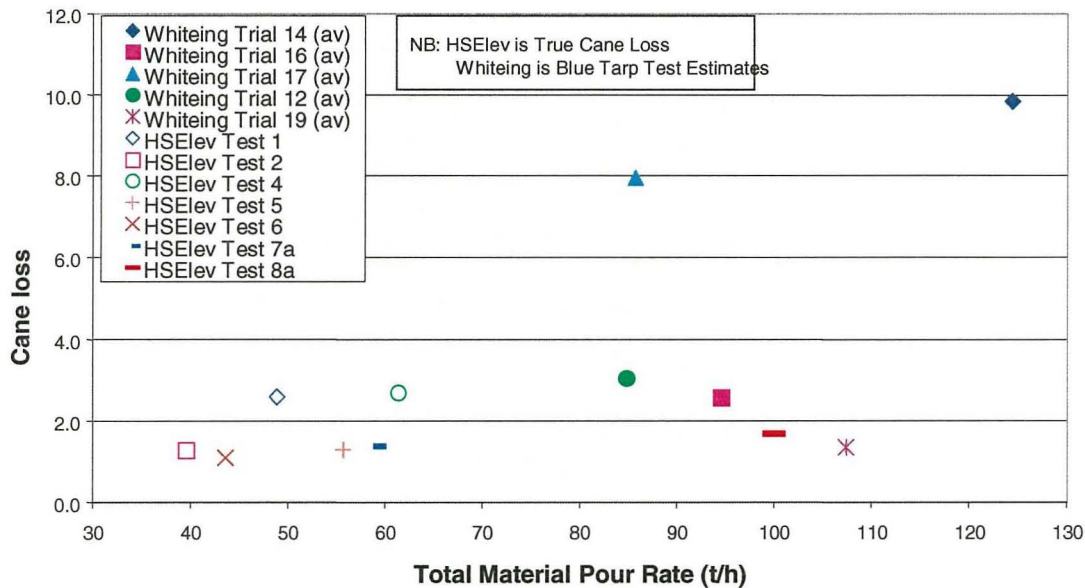


Figure 28 - Cane loss for high-speed elevator system operating in a primary cleaning mode and the cane loss of a conventional harvester.

Figure 28 presents data on cane loss for the high-speed elevator and Whiteing's trials. Because billets do not pass through an extractor fan, cane loss can accurately be determined accurately by simply determining the weight of billets rejected with the trash as a proportion of the total weight of billets before the cleaning pass. Cane loss was typically between 1% and 3% for the high speed elevator.

Again, given the modifications undertaken after trial 4, the cane loss for the high speed elevator appeared conservative compared with the cane loss estimated from the 'blue tarp' tests on the conventional harvester. It should be remembered that the blue tarp has been shown to significantly underestimate cane loss as pour rate increases above approximately 60 t/h (BS189 data).

Similarly, trials were conducted by Ridge and Dick (1987), using a stationary harvester being fed with wholestalk cane at a controlled rate. Cane losses in the order of 15% were noted with a 'standard' harvester a pour rate of 60 t/h.

5.4.1.2 Secondary cleaning performance

Figure 29 details the results from a number of test runs of the high-speed elevator system, at a range of pour rates, operating in a secondary cleaning mode. Also presented in Figure 29 are data from Whiteing (BS189) on the measured performance of secondary extractor systems in field trials.

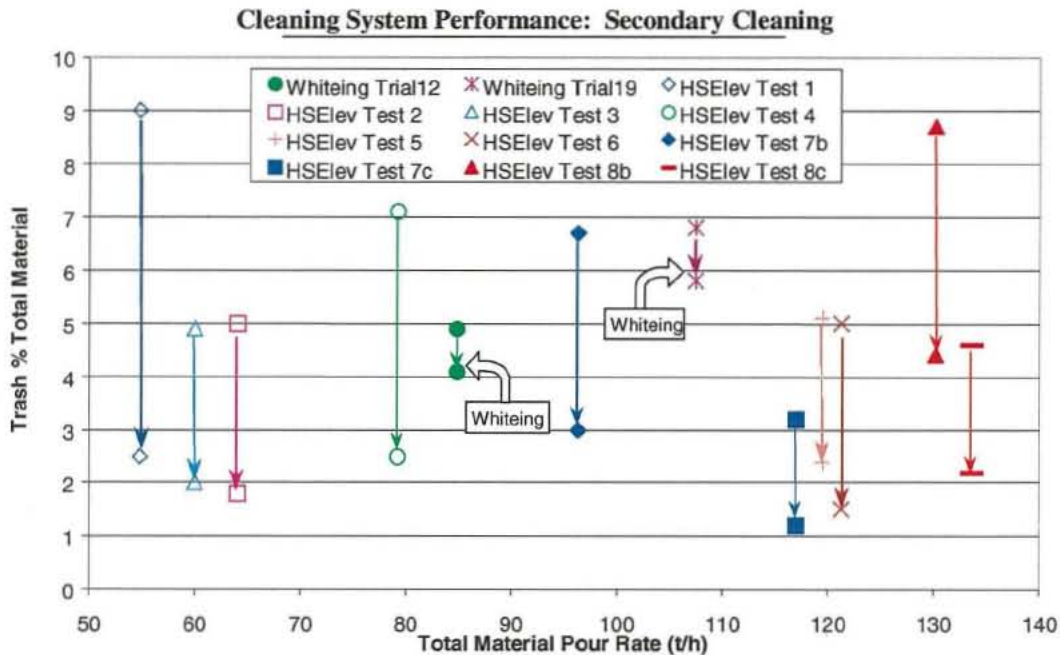


Figure 29 - Cleaning performance of high-speed elevator relative to conventional secondary extractor over a range of pour rates. The upper point is the trash level before passing through the cleaning system, and the lower number the measured trash level after the cleaning pass.

It should be noted that a number of minor modifications to the configuration of the cleaning system and the feed system were undertaken after test 4. This allowed higher pour rates to be used and enhanced the escape of trash from the cleaning module. For comparison, the data for two field trials on secondary extractor performance by Whiteing (BS189) are included for reference. These trials were conducted under dry harvesting conditions, with the final EM levels being better than 'mill average'.

Trials such as 7b and 7c, and 8b and 8c represent the cleaning achieved in multiple passes. Notwithstanding the difference between field trials and workshop tests, the data clearly indicates the high levels of trash removal by the high-speed elevator relative to that achieved by the harvester fitted with a conventional extractor. The research by Ridge and Dick (1987) would indicate that, providing operating parameters are properly controlled, results from 'workshop' trials closely reflect the performance that should be achieved in the field, under similar cane conditions.

Figure 30 presents this data as cleaning efficiency, ie the ratio of trash removed as a proportion of total trash entering the cleaning stage. It is evident that there is a general the reduction in cleaning efficiency as pour rate increases (noting also that improvements were made after Test 4), eg 60-70% at 55 t/h vs 50-60% at 120-130 t/h. At a given pour rate, the cleaning efficiency is relatively constant despite significant changes in the composition of material entering the cleaning chamber (eg 7b vs 7c, 8b vs 8c). The cleaning efficiency of the conventional secondary extractor (Whiteing, BS189 trials 12 and 19) is very significantly lower than the results for the high-speed elevator.

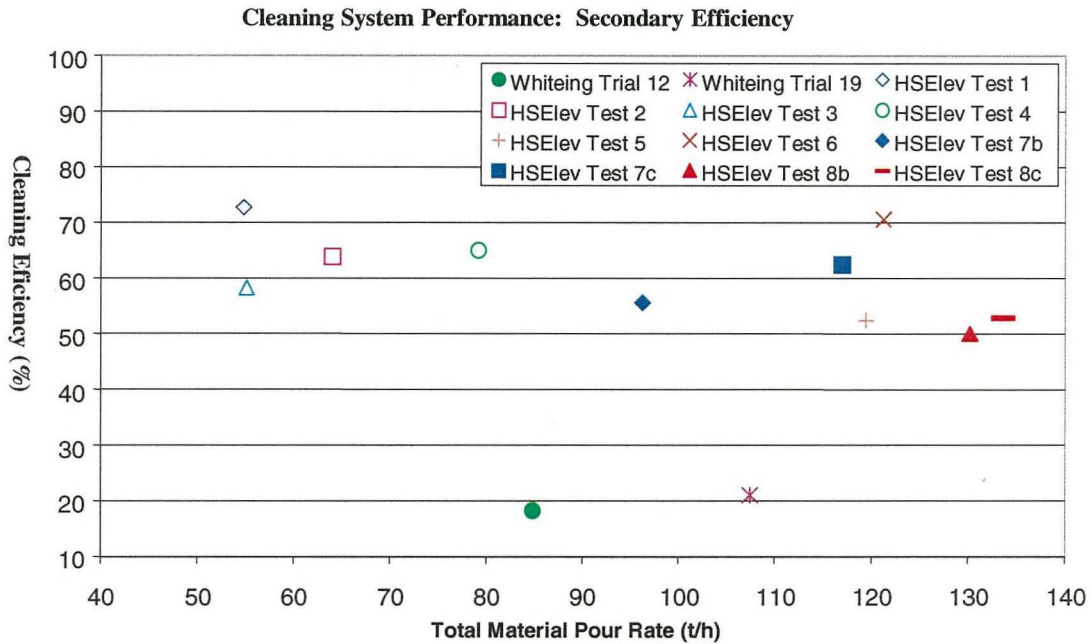


Figure 30 - Trash removal efficiency for high-speed elevator system operating as a secondary cleaning system.

Figure 31 presents data on cane loss from the trials. As previously noted, the modifications made to the high-speed elevator system after trial 4 appear to have dramatically reduced cane loss. Whilst the cane loss recorded by Whiteing is apparently very low, it is estimated from 'blue tarp' measurements. Other trials (Whiteing, pers. com., and Paton pers. com.) indicate cane loss from secondary extractors can reach several per cent.

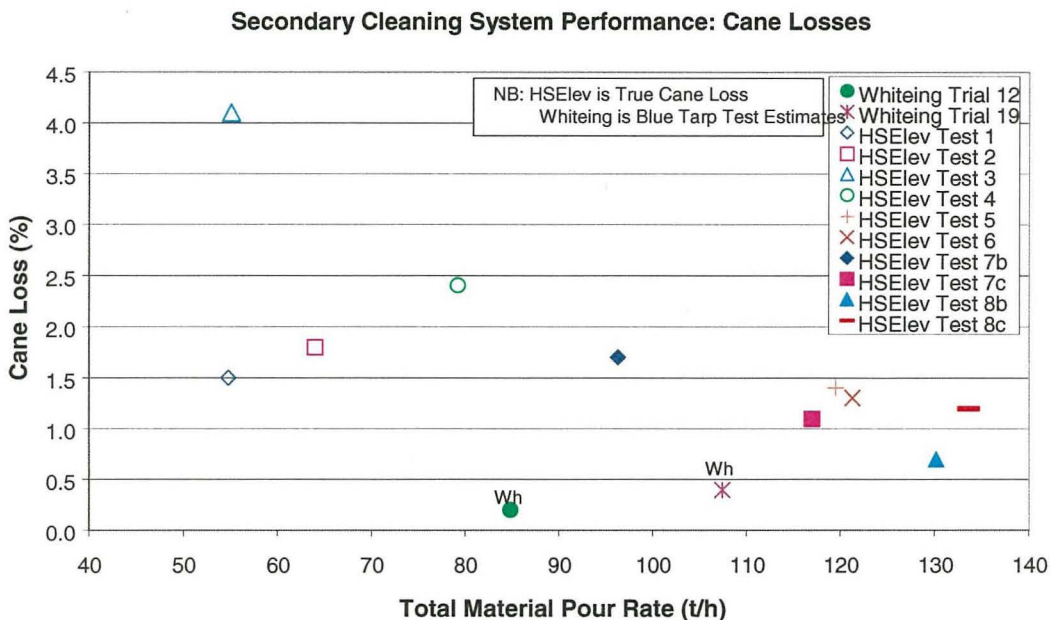


Figure 31 - Cane loss from the high-speed elevator. Modifications to the cleaning module were undertaken after trial 4, which reduced cane loss.

5.4.2 Performance in field trials

A limited field trial was undertaken with the harvester fitted with the high speed elevator in a crop of Q124 under dry, hot harvesting conditions. Cane supply was limited, and there was almost no potential for a significant trial because of belt tracking issues.



Image 31 - The high-speed elevator operating. Note the trash chute and some disturbance of trash on the ground by the air from the secondary cleaning system.



Image 32 - The shade-cloth bag on the secondary trash chute allowed trash extraction efficiency to be measured.

Figure 32 presents information on the trash removal and pour rates for these trials and Figure 33 expresses the data as cleaning efficiency.

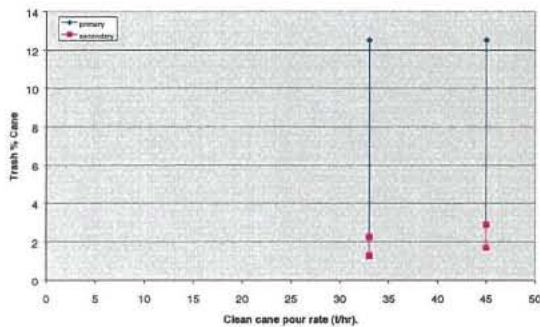


Figure 32 - Data from limited field test with high-speed elevator. Pour rates were lower than anticipated due to a mis-estimation of cane yield.

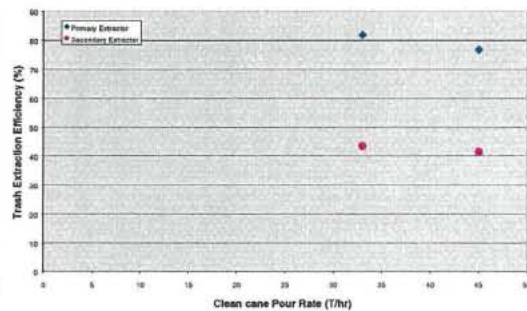


Figure 33 - Cleaning system performance for the primary and secondary systems expressed as trash removal efficiency.

The extremely limited field trial program was conducted at unrealistically low pour rates, however, the following comments can be made.

- Measured cane loss was very low, <1 t/ha.
- Whilst the low pour rates meant that the predicted cleaning efficiency of the primary extractor would be anticipated to be high, it was operated at a very low fan speed

approximately (1,000 rpm.) At this fan speed, it appeared to perform above expectations. This is consistent with the prediction that the modifications to the elevator bowl could be expected to enhance the performance of the primary cleaning system.

- Given the low levels of available trash, the cleaning efficiencies achieved by the secondary cleaning system were within expectation.

5.5 Conclusions from available data

Indications are that cane loss from the high-speed elevator system will be low relative to the level of cleaning achieved. Significantly, the system offers the option of optimising the primary extractor for minimum cane loss and allowing the secondary cleaning system to take a greater role in controlling final EM levels.

The system removed a relatively constant proportion of trash each pass. This is consistent with the Brazilian and Colombian experience with multi-pass cleaning of cane in cane cleaning plants.

The high-speed elevator and secondary cleaning system can be confidently expected to offer a significant improvement in the levels of EM delivered, relative to a harvester fitted with a conventional secondary extractor.

This is achieved within the constraints of a compact cleaning module that is light and has few wearing parts. The high efficiency of the blower fans is maintained because trash does not pass through them causing wear or adversely impacting on blade aerodynamic profile.

6.0 DIFFICULTIES ENCOUNTERED DURING PROJECT

A number of difficulties were experienced throughout the duration of the project which all impacted on the final outcome. These difficulties are detailed below.

1. Delays in the supply of components for the initial development of the proof of concept hugger belt arrangement were experienced. This included access to conveyor belts and the unavailability of the desired length rollers through Beltreco. This required a range of strategies including the purchase of wider rollers and subsequently cutting them down to the required length at a local machine shop, and the fabrication of other rollers and components.
2. Delays in delivery of the prototype hugger belt system frame from the manufacturer Case IH Austoft. This was due to two factors including:
 - delays in the initial supply of drawings by JCU (primarily because the magnitude of the design task was somewhat underestimated) compounded by the two facilities utilising different three-dimensional drawing software. Incompatibility then arose during the transfer of drawings causing further delays;
 - the heavy workload in the Austoft R&D facility because of commitments to NCEA 01 (Jetclean) and the prototype 'Maxi-Haul' that was being developed at the time.

This manifested into a delay in the manufacture of the frame and subsequent significant extension of the delivery date to BSES.

3. These difficulties meant that by the time the prototype was assembled, the 1999 crushing was nearing completion. These time constraints also meant that the project leader was not available to undertake initial trials of the system, or to run the testing program because of other commitments.
4. The belt tracking problems, that were identified very early in the workshop testing program delayed the development of the complete system and resulted in only limited field evaluation being undertaken.

7.0 RECOMMENDATIONS FOR FURTHER RESEARCH

Whilst the prototype development undertaken during this project confirmed the potential for very significant gain, major reliability problems were encountered in the design developed for the high-speed belt system. The project was clearly too optimistic with respect to the fast tracking of technology and the aims of expanding the envelope of hugger belt elevator technology.

The development team strongly believes the system offers major gains in the potential to supply clean cane off the harvester, whilst allowing minimisation of cane loss. However, to achieve this the following criteria would need to be addressed.

- (a) To achieve reliable operation, a significant redesign of the hugger belt system is necessary. The new design would probably adopt a more conventional approach to lower belt support, and subsequently would involve a major redesign of the positioning of the cleaning module. On the basis of current knowledge, it is believed that redesign of the hugger belt module and repositioning of the cleaning module can be undertaken with no detriment to the performance of the cleaning system.
- (b) Feed of cane into the hugger belt system. Initially, this was seen as the most critical component of the entire development, as well as being the component with the highest risk profile. The potential value of the system to industry is clearly dependent on this being achieved. The pre-production prototype has determined that a successful design of a feed system is possible.

Further work is required to build on the significant developments made throughout this project, to allow the commercial development and availability of a high-speed elevator incorporating an advanced cleaning system.

8.0 APPLICATION OF THE RESULTS TO INDUSTRY

This was a joint project between manufacturers and research organisations. It has focused on a commercial outcome, but included both technology transfer and technology development. The development process will be conducted in conjunction with key commercial stakeholders including Gough Plastics, CASE IH Austoft and limited input by Beltreco. It involves the integration of two significant and complementary developments into one harvester component.

The goals of this project, if achieved, would have offered significant potential advantages for all sectors of the sugar industry.

To the harvester manufacturer the concepts offer:

- a reduction in weight of a major component of the harvester. The reduction in elevator weight improves machine stability and will allow subsequent weight reduction in other harvester components;
- an increase in dimensions available for the cleaning chamber, and the potential to move the elevator further to the rear, allowing incorporation of further development in primary cleaning technologies.

To harvester operators the concepts offer:

- a significant opportunity to reduce cane loss and EM levels. It is anticipated the unit would be a cost-effective retrofit to current harvesters.
- a lower whole of life cost than the current chain and slat elevators. The use of a blower-based cleaning system could be anticipated to further reduce maintenance costs.

These advances flow on directly to the industry through better harvester performance and lower harvester weight and advanced cleaning.

Additionally, the prototype elevator bowl is a major departure from current designs. It rotates in a single plane about the vertical pivot supporting the cradle of the current elevator. The vertical movement of the elevator is achieved by rotation about a secondary horizontal pivot at the junction of the bowl and the elevator. This design allows the substantial lowering of the bowl and a dramatic increase in efficiency of space use around the rear of the machine. The design will substantially enhance the flexibility and the length, width and depth available for the primary cleaning system. This could have significant implications for a number of projects including the following.

- NE1: Improving the cleaning of cane.
- SI49: Aerodynamic optimisation of extraction chambers for high pour rate pneumatic cleaning devices.
- SI61: Integrating an improved cane delivery system and chamber aerodynamics.

A significant constraint for the development associated with all three of these projects has been the clearances required for the conventional elevator and bowl, including the requirement to minimise rear overhang for weight transfer constraints. The proposed high-speed elevator and bowl design will dramatically reduce these constraints and enhance the outcomes of all of the above projects.

The use of a high-speed elevator also impacts on research such as the development of mass-flow sensors to monitor cane flow. The energy consumption of a high speed elevator can be anticipated to be much more closely related to mass flow rate than is the case for conventional low speed chain and flight elevators.

9.0 PUBLICATIONS ARISING

This project was a joint venture between a syndicate of a number of companies and organisations. As such, all information was considered confidential and no publications arose from this work.

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11.0 ACKNOWLEDGMENTS

The funding support from the Sugar Research and Development Corporation for this project is gratefully acknowledged. The project was conducted in conjunction with key commercial stakeholders, including BSES, JCU, Gough Plastics, Beltreco, CASE IH Austoft and SRI. The tasks undertaken by each stakeholder included the following.

- BSES facilitated the project by undertaking a key role in project development, particularly design and development of the proof of concept and prototype rigs and data collection throughout the all stages of the project (proof of concept rigs, prototype laboratory testing and field testing).
- Mechanical engineering staff at JCU undertook the detailed mechanical design of the hugger belt frame. This included the development of the design, including structural analysis, preparation of all drawings and component designs for the manufacture of the final prototypes.
- Gough fabricated all plastic componentry required for the proof of concept and prototype developments relating to the cleaning system.
- Case IH Austoft undertook the manufacture of components for the prototype hugger belt system.
 - SRI undertook the numerical modelling for analysis of a number of cleaning chamber concepts, as well as the development of the final design for optimum air flow. They also modelled the elevator bowl design to optimise the improvements in air flow entering the primary cleaning chamber.

The authors would like to thank those involved from these respective organisations including Andrew and Simon Gough (Gough Plastics), Peter Birkbeck (Beltreco), Mal Baker and Don Helmrich (CASE IH Austoft), David Kaupilla (JCU) and Phil Hobson (SRI) for their contributions.

In addition, the assistance from BSES officers Peter Gaul who undertook the construction and development of the prototype elevator and cleaning system and Cam Whiteing and Peter Hockings who assisted with testing is gratefully acknowledged.

APPENDIX A

APPENDIX A PREDICTED FLOW IN A CONVENTIONAL HARVESTER PRIMARY EXTRACTION CHAMBER

Dr P Hobson, SRI

A research program has been undertaken at SRI with the aim of developing computer based models of the pneumatic cane cleaning process. The development of these models has taken place in two main areas; namely, particle trajectory simulation at high pour rates and the use of computational fluid dynamics (CFD) to predict the flow of air in a heavily loaded (high pour rate) harvester extraction chamber. The CFD air flow models have been developed under SRDC project SRI049.

The CFD model developed under SRI049 has been extensively validated using a statically mounted but otherwise unmodified Austoft 7000 harvester primary extraction chamber. The general approach in the validation procedure has been to compare measured aerodynamic forces on statically mounted artificial cane billets within the extraction chamber, with corresponding predicted forces. Using this facility, all the major flow features predicted for the primary extraction chamber have been verified.

Both two- and three-dimensional CFD models have been developed. The following flow predictions are from a two-dimensional model in which the flow simulation plane is positioned through the centre line of the primary extraction chamber as shown schematically in Figure A.1. The simulated components include the fan, extraction chamber wall, cone baffle, cane billets and elevator boot. The simulated billets are assumed to emerge from the chopper and pass (butt first) through the simulation plane.

A pressure-flow rate response has been built into the fan boundary condition to simulate the primary extractor fan characteristics. These data have been extrapolated from measurements made at the CSIRO on a scale model of an Austoft fan (Downie, R J 'Scale model testing of sugar cane harvester trash extraction fan rotor', CSIRO Div. of Build., Const. and Eng., Doc 90/4, 1990).

The predicted flow field for a 150 tonnes/hour pour rate is given in Figure A.2. Inspection of the predicted flow field reveals a number of features that characterise the current extraction chamber design; namely:

- a highly non-uniform flow is predicted which varies in magnitude from less than 3 m/s to greater than 38 m/s within the chamber;
- a significant region of recirculation exists just inside the extraction chamber. This region is found (depending on pour rate) to vary in extent to cover between 10% to 30% of the extraction chamber. The velocities within this region (< 3 m/s) are too low for effective trash removal;
- the combined effect of the elevator boot and cane cone are to direct a jet of air through the centre section of the extraction chamber. The air velocities within this jet (>30 m/s) are well above the terminal velocities of the billets (typically ~15 m/s) and therefore represent a source of potential cane loss;

- comparing the predicted flow rate and therefore pressure drop across the fan boundary even at moderate pour rates (80 tonnes/hour) with the measured fan data of Downie (1990) indicates that most harvesters operate well within a region in which the fan is aerodynamically stalled. The significance of the stalled state is that radial flows become significant and in the extreme, the fan will operate as a centrifugal impeller or mixer. This will result in a high degree of swirl around the chamber. This phenomenon has been observed in practice.

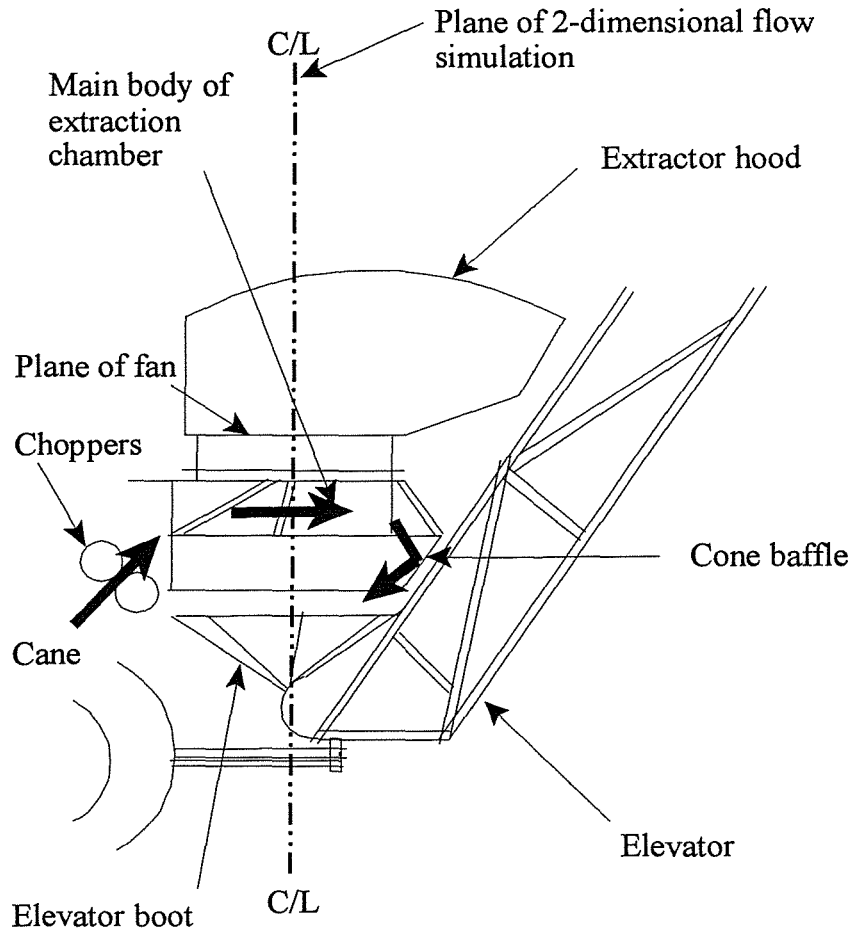


Figure A.1 - Schematic of harvester section showing flow simulation plane through the primary extraction chamber.

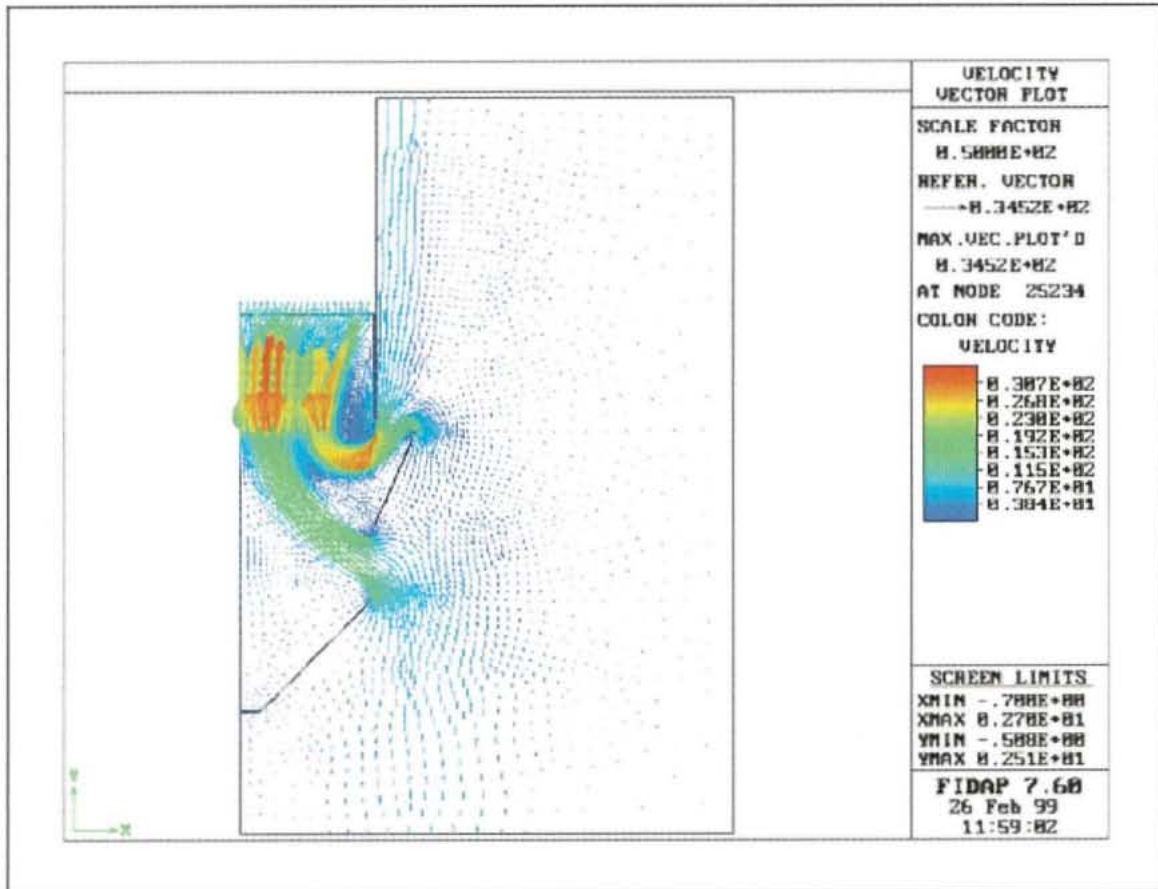


Figure A.2 - Predicted flow around and through a primary extraction chamber operating at 150 t/h.

APPENDIX B

APPENDIX B DYNAMIC SIMULATION OF A CONVEYOR BELT SYSTEM FOR CANE HARVESTERS

Dr Peter Murdoch, JCU. December 1998

B.1 Summary

A proposed sugarcane harvester conveyor system was analysed using 'Working Model 2D' software. The dynamic forces on billets were investigated, and the geometry of the conveyor roller system was optimised to reduce the cane impact forces and increase the robustness of the system with respect to variations in the cane billet sizes. The analysis of four simulations is reported.

A computer model was developed to simulate the current prototype conveyor system as shown in Figure B.1. The dynamics of this system were verified by observing a video of the actual system in operation. The video and simulation observations both confirm that the cane billets experience multiple impacts with the rollers as they make the transition between the horizontal to the inclined belts. The size, speed and relative positions of the rollers in the current design were altered in an effort to reduce the magnitude of the cane billet impacts within the transition region.

The original design was modified and a second computer model was developed. This alternative design repositions the same number of rollers in order to reduce the impact forces by a factor of 10. The original design has a curved backing plate behind the rear conveyor belt that applies a compressive hold on the billets as they move up between the belt pair. By repositioning the curved backing plate behind the front conveyor belt the transition angle can be reduced from 75 degrees to 30 degrees, whilst the structure still fits within the allowable spatial design envelope.

In conclusion, the current design is considered insensitive to minor geometric modifications as no notable improvements were observed in either billet impact force or billet size sensitivity. However, the altered design was observed to be more robust and insensitive to variations in billet size. It was also seen to be far less damaging to billets in the transition region.

B.2 Modelling

The conveyor system was modelled using 'Working Model 2D' dynamic simulation software. The moving conveyor belts were simplified to zero friction plates without adversely affecting the results. Each of the five cane billets used in the model varied in weight and diameter as to best describe the expected range. The billets were given an elastic restitution coefficient of 0.22, which was determined by experimentation, and an initial velocity of 3 m/s.

The area of major concern is the junction between the feed and delivery belts where the billets are required to change direction. Once the billets make this transition and are confined between the hugger belts, further analysis becomes unnecessary.

Figure B.2 shows the original design as described by the drawing in Figure B.1. The red roller has a diameter of 200 mm and drives the feed belt with a velocity of 3 m/s. The green intermediate roller also has a diameter of 200 mm and spins with a surface velocity of 6 m/s. The dark blue roller is the base roller for the rear delivery conveyor and travels with a surface speed of 6 m/s. The light blue roller is 280 mm in diameter, has a surface velocity of 6 m/s, and is positioned 320 mm above and 20 mm aft of the red roller. The grey roller simulates the forward delivery conveyor belt and expels the billets once they have passed through the transition region.

The design shown in Figure B.3 is a slight modification to the original. The diameters of the rollers remain unchanged, as do the speeds with the exception of the intermediate (green) roller, whose surface speed is increased to 7.9 m/s. The light blue roller is repositioned to a point 320 mm above and 8 mm forward of the red roller.

The design shown in Figure B.4 is another slight modification to the original. In this instance the dark blue roller has its surface speed increased to 7.9 m/sec and the light blue roller is repositioned to a point 390 mm above and 52 mm aft of the red roller.

The main emphasis of the design shown in Figure B.5 was to reduce the angular change that the billets were required to make in the transition region. The original design as shown in the drawing (Appendix 1) shows the billets entering the delivery conveyor at an angle of 75 degrees and then exiting at an angle of approximately 45 degrees giving a total angle path change of 30 degrees, courtesy of the backing plate. This alternative arrangement places the convex backing plate on the forward delivery belt. This allows the transition angle to drop to 30 degrees with the exit angle of 60 degrees. The green intermediate roller in the original design was replaced with a 280 mm roller with the rear delivery belt attached. The dark blue roller may or may not be necessary depending on the chosen backing plate geometry. The light blue roller is positioned 290 mm above and 90 mm aft of the red roller. The green roller is placed 40 mm below and 250 mm aft of the red roller. The red roller has a surface velocity of 3 m/s while the others have a surface velocities of 6 m/s. The hinged frictionless beams simulate the action of the conveyor belt as the billets track around the three upper rollers.

B.3 Results and observations

Numerical results are shown in the graphs on each of the attached figures. They quantify the X, Y and rotational velocities of the first billet as it travels through the transition region. The graphs are recorded over a time period from 0.1 to 0.3 seconds. The range of each X velocity graph is from -4 to 2 m/s (the -ve sign indicating a velocity from right to left). Each Y velocity graph ranges from -1 to 3 m/s and the rotational velocity graph is from -2,400 to 1,000 degrees/s.

Comparing the graphs for Figures B.2, B.3 and B.4 there does not appear to be much of a difference in terms of the variations in the velocity of the first billet. Looking closely at the original design in Figure B.2 it shows the billet striking the dark blue roller and actually reversing its direction. At this point the billet is accelerated forward again by the red roller and by contact with the following billets.

In order to avoid this direct collision with the dark blue roller, the angular velocity of the green roller was increased in Figures B.3 and B.4 with the result being that the billet would rotate up and clear the dark blue roller. However, in the original design the light blue roller knocked the billet down into the dark blue roller. Therefore, the position of the light blue roller in Figures B.3 and B.4 was altered in an attempt to avoid this.

The original design and the two variations were observed to be sensitive to variations in billet size. The random nature of the collisions between the billets and the green roller produced vast variations in billet path and trajectory. The conclusion is that the original design is not robust in terms of the smooth handling of billets and the two variations showed little improvement due to the nature and magnitude of the random fluctuations.

The alternative design shown in Figure B.5 displays a significant improvement both in the reduction of X and Y velocity variations and in the degree of robustness.

The overall geometry of this configuration still fits within the dimensional limits of the harvester conveyor system and is no more complicated to either manufacture or maintain.

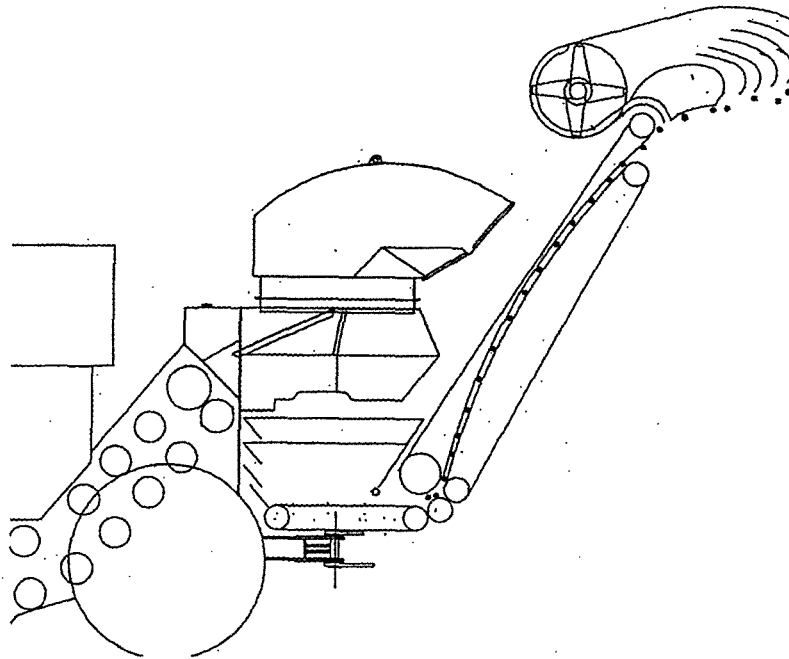


Figure B.1 - Original design

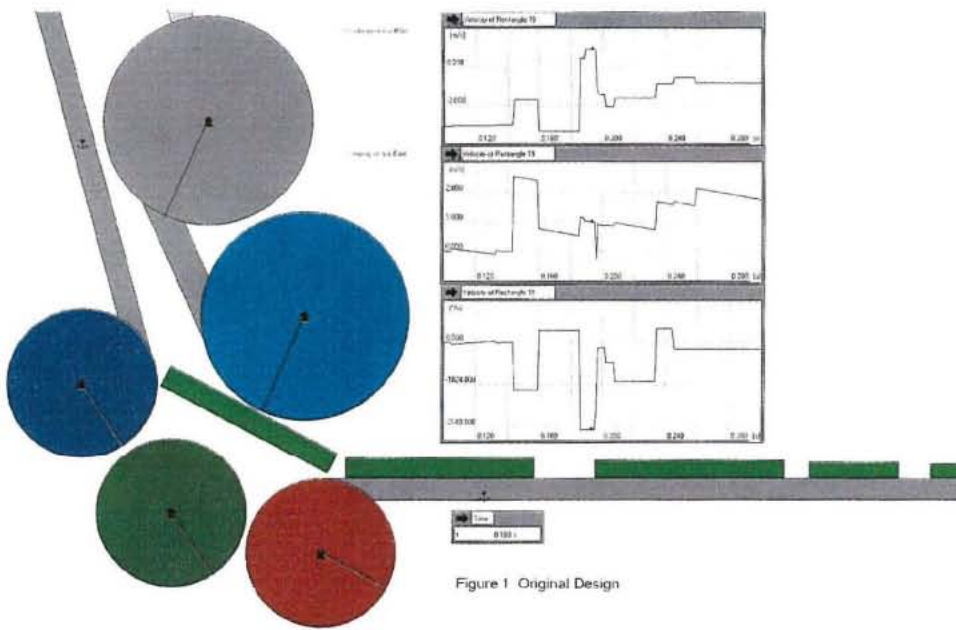


Figure 1 Original Design

Figure B.2 - Original design

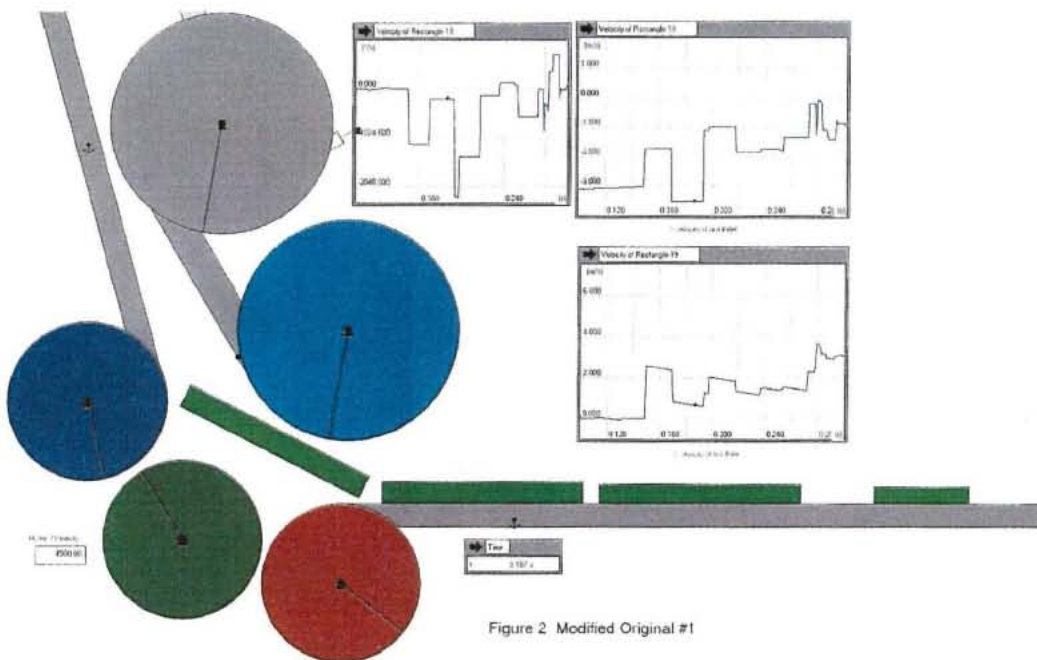


Figure 2 Modified Original #1

Figure B.3 - Modified original #1

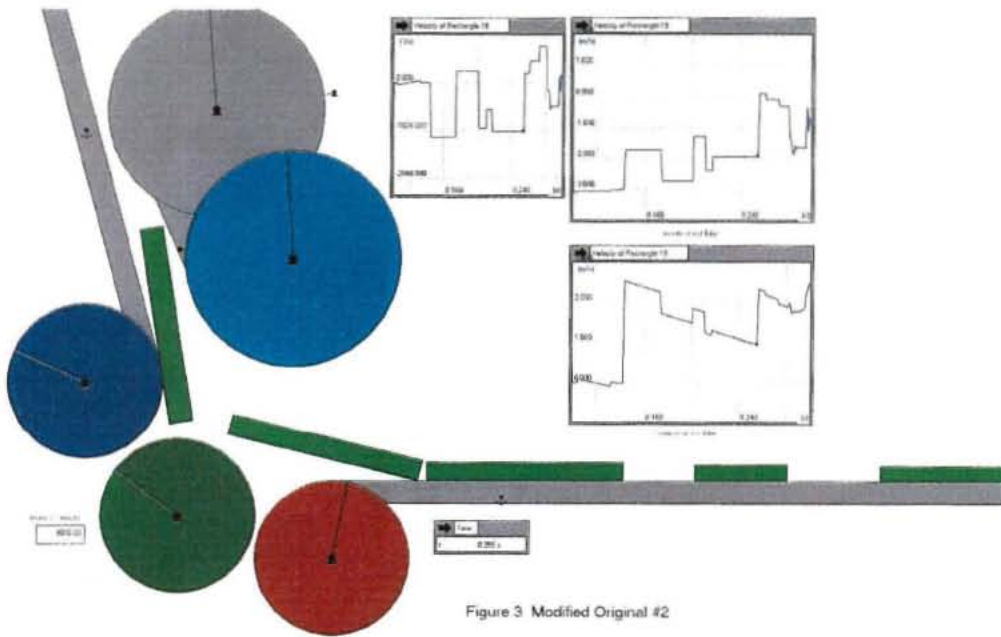


Figure 3 Modified Original #2

Figure B.4 - Modified original #2

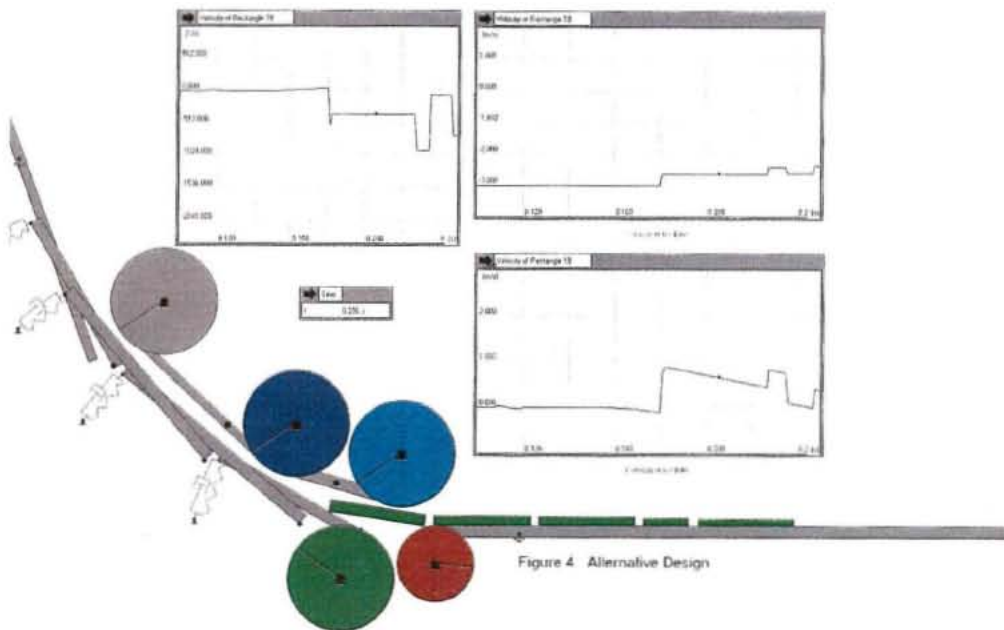


Figure 4 Alternative Design

Figure B.5 - Alternative design

APPENDIX C

APPENDIX C COMPUTATIONAL FLUID DYNAMICS (CFD) MODELLING OF THE AIR FLOW AROUND THE PRIMARY EXTRACTION CHAMBER

Dr P Hobson, SRI

The CFD code FIDAP has been used to investigate and improve air flows in the region of the primary extractor with the modified high-speed, lightweight elevator installed.

In the investigation, two main issues have been investigated:

- the blockage effect of the sideways-slewed elevator and its effect on the symmetry of flow through the primary chamber. This includes an investigation of the influence of the exposed high-speed elevator belt on the air flow;
- the flow distribution as a result of air entering through the back section of the primary extraction chamber. During normal operation, air is prevented from entering the front section of the extraction chamber by the main body of the harvester.

In practice, all effects on the flow will, to varying degrees, be interdependent. However, in the interests of keeping the computational time and memory demands of the model within manageable proportions, this investigation has been carried out using two two-dimensional simulations. To investigate the above two flow effects, the extraction chamber has been simulated in vertical planes at right angles and parallel respectively, to the length of the harvester.

C.1 Open flow configuration

The first configuration investigated assumes the maximum open flow area between the extraction chamber and the elevator boot. Previous practical and CFD studies indicate that this approach delivers a uniform air flow without the (trash) blockage effects associated with the installation of baffles to control the flow. This 'open' design is only possible to implement at the 'sides' of the extraction chamber. At the back of the extraction chamber, a baffle has to either extend down from the chamber or up from the elevator boot in order to 'catch' cane being ejected by the chopper.

The open sections at the side of the extraction chamber have been assumed to take the form of a 239 mm gap between the lower edge of the unmodified cane cone and the upper edge of the elevator boot. At the rear of the extraction chamber, the elevator boot is extended upwards to the level of the lower edge of the cane cone. This extended section is angled backwards at 55° to give a (horizontal) clearance of 167 mm between the lower edge of the cane cone and the upper edge of the boot. This extension of the back of the elevator boot corresponds to the maximum clearance (and therefore maximum possible open flow area) which permits free movement of the elevator if the boot is designed to be static or alternatively free movement of the boot if the boot is designed to move with the elevator.

The models have been run initially assuming a minimal (~0 tonnes/hour) pour rate. The fan inlet total pressure fan (as in all the subsequent simulations) is calculated within the model according to the measured characteristics of an Austoft 7000 extractor fan (Downie, R J, January 1990, 'Scale model testing of sugar cane trash extraction fan rotor', CSIRO report DBCE Doc. 90/4). The conveyor belt speed is set at 6 m/s and assumed to be exposed (uncovered). The resulting air flow is shown as a vector plot in Figure C.1. Peaks in the air velocity of around 15 m/s are indicated in the central region of the extraction chamber. Although high relative to the mean velocity of ~9 m/s, this peak is moderate compared with those for a conventional extraction chamber (~20 m/s) with the same near-zero pour rate. In addition, regions of recirculation on the inside of the extraction chamber wall are small relative to those predicted for the conventional configuration.

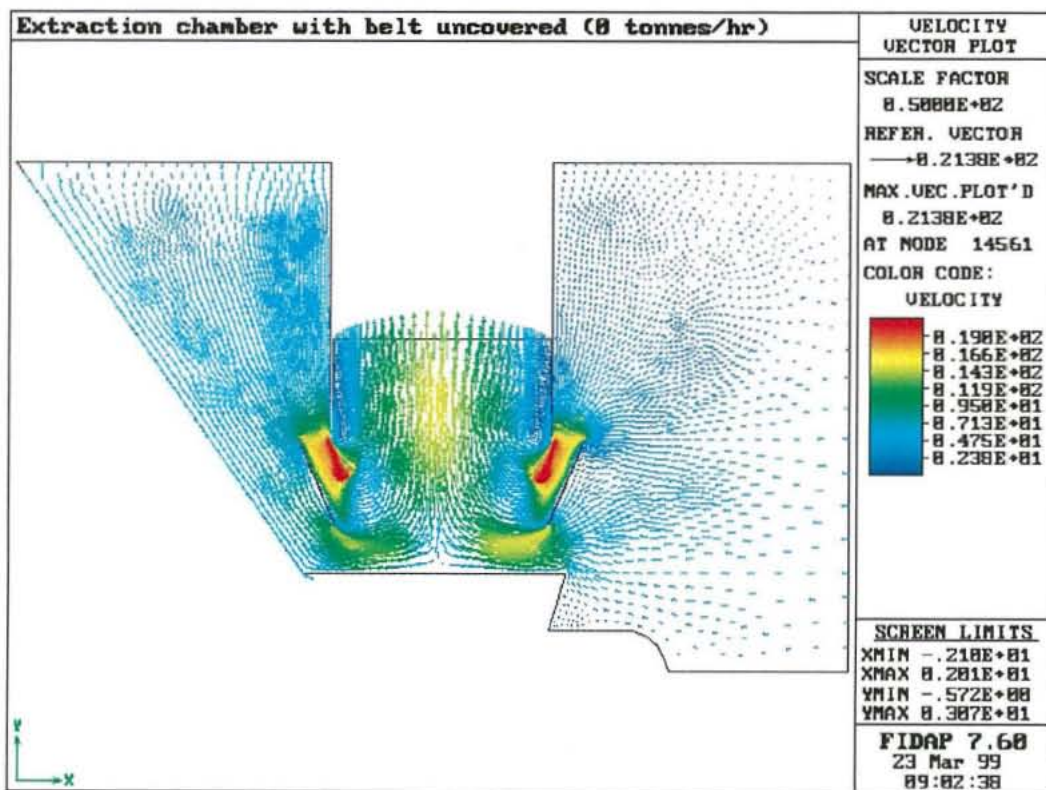


Figure C.1 - Vector plot of extraction chamber air flow with belt speed of 6 m/s.

To determine the effects of the elevator belt on the flow field in the chamber, the belt speed was set to 0 m/s (simulating a covered belt). The resulting predicted flow (Figure C.2) indicated a negligible effect on peak velocities and flow distribution. A small decrease in the mean air velocity in the chamber has been predicted (8.6 m/s compared with 8.9 m/s for the case with the uncovered belt).

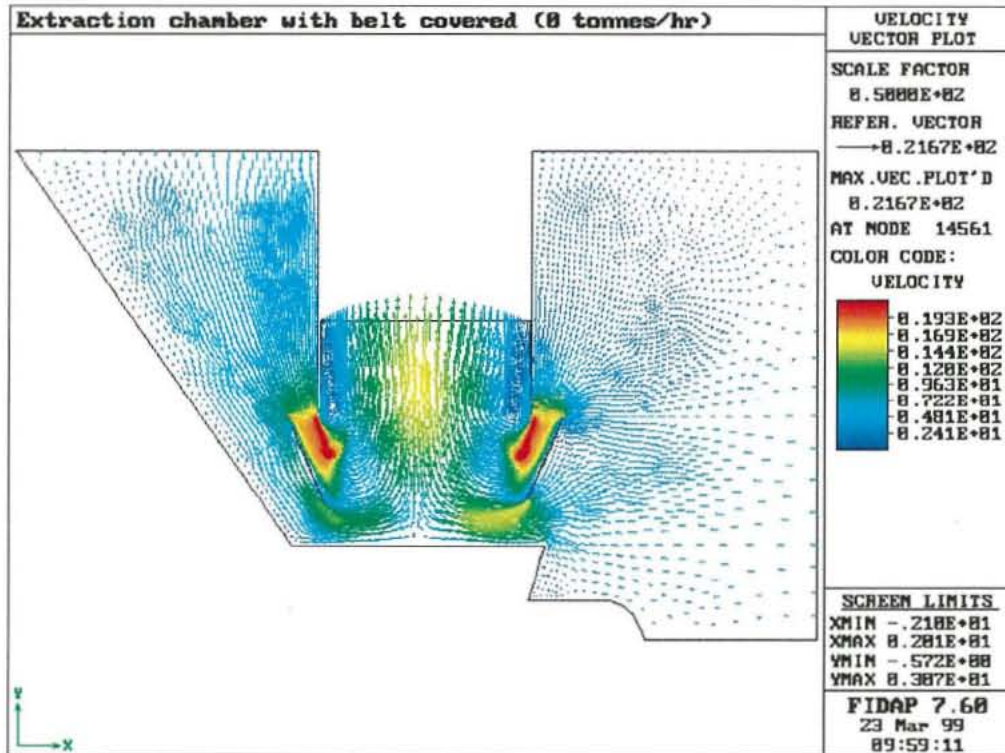


Figure C.2 – Vector plot of extraction chamber air flow with belt speed of 0 m/s

Details of the predicted flows at the higher simulated pour rate of 180 tonnes/hour are shown in Figure C.3 (for the uncovered belt) and Figure C.4 (for the covered belt). The peaks in the flow essentially disappear at this higher pour rate. In neither case does the belt-induced air flow have any significant effect on flow distribution or total flow rate.

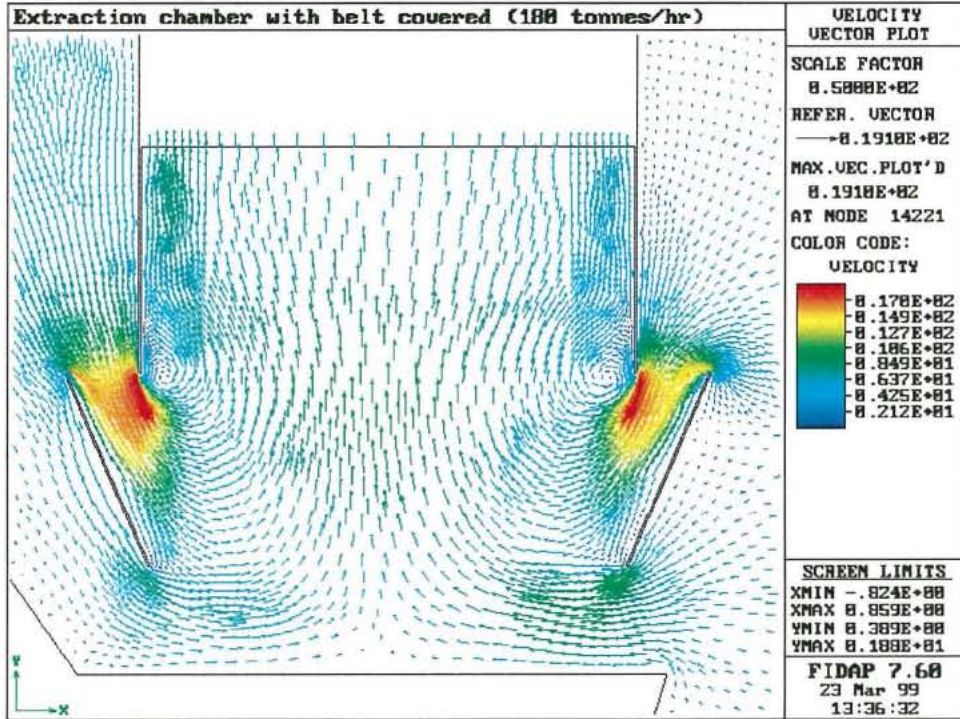


Figure C.3 – Vector plot of extraction chamber air flow with uncovered belt and pour rate 180 t/h.

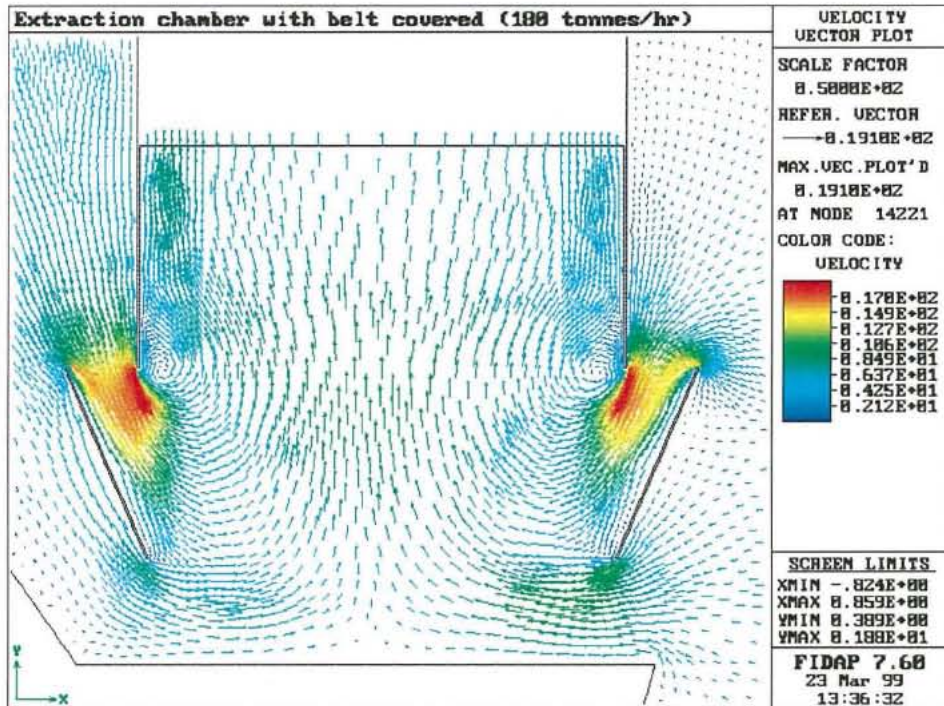


Figure C.4 – Vector plot of extraction chamber air flow with uncovered belt and pour rate 0 t/h.

The effect of the belt was discounted at this stage and has not been investigated further.

Using the two-dimensional simulation of flows through the back of the extraction chamber indicates severe asymmetry of flow through the chamber. Figure C.5 shows the predicted flow field at a pour rate of 180 tonnes/hour. This asymmetry is caused primarily by the streaming of air across the chamber from the more restricted flow area produced by the extended boot at the back of the chamber and the (effective) absence of air entering from the forward section of the extraction chamber.

C.2 Enclosed configuration

A second configuration has been investigated in which the 'open' design has been replaced by baffles which have the dual role of enclosing the region between the elevator and cane cone to form a larger cane storage volume as well as producing an acceptable air flow distribution. In determining the dimensions of and spacing between the enclosing baffles, three criteria were applied; namely:

- that the open space between baffles should be at least equivalent to that between the extraction chamber wall and cane cone on the current conventional harvester design (162 mm). This is to ensure that trash does not block these vents to any greater extent than currently occurs in an unmodified harvester;

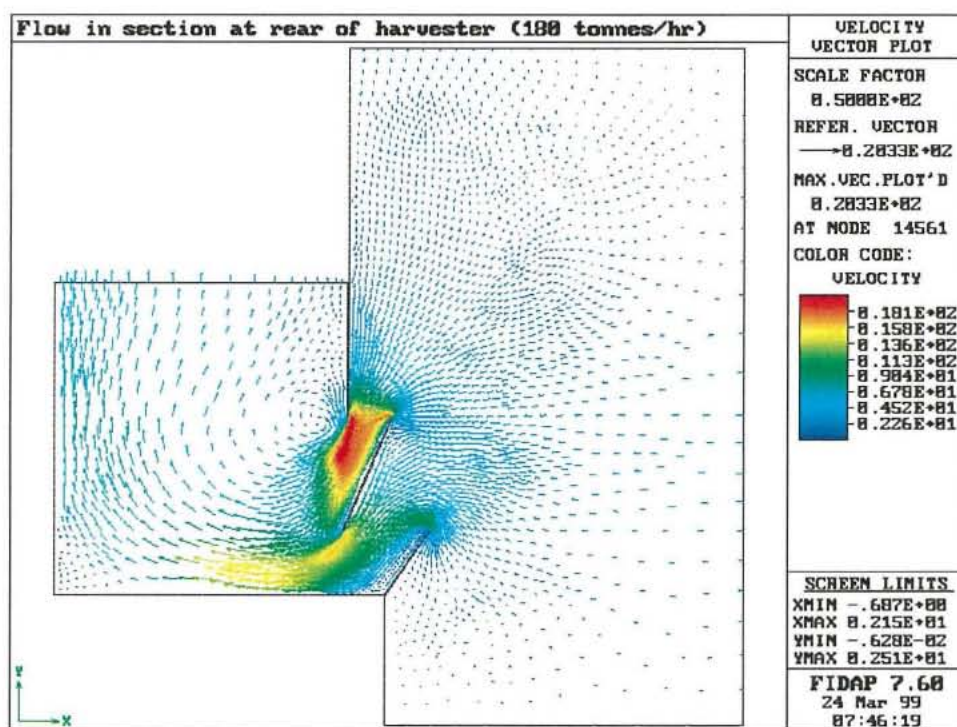
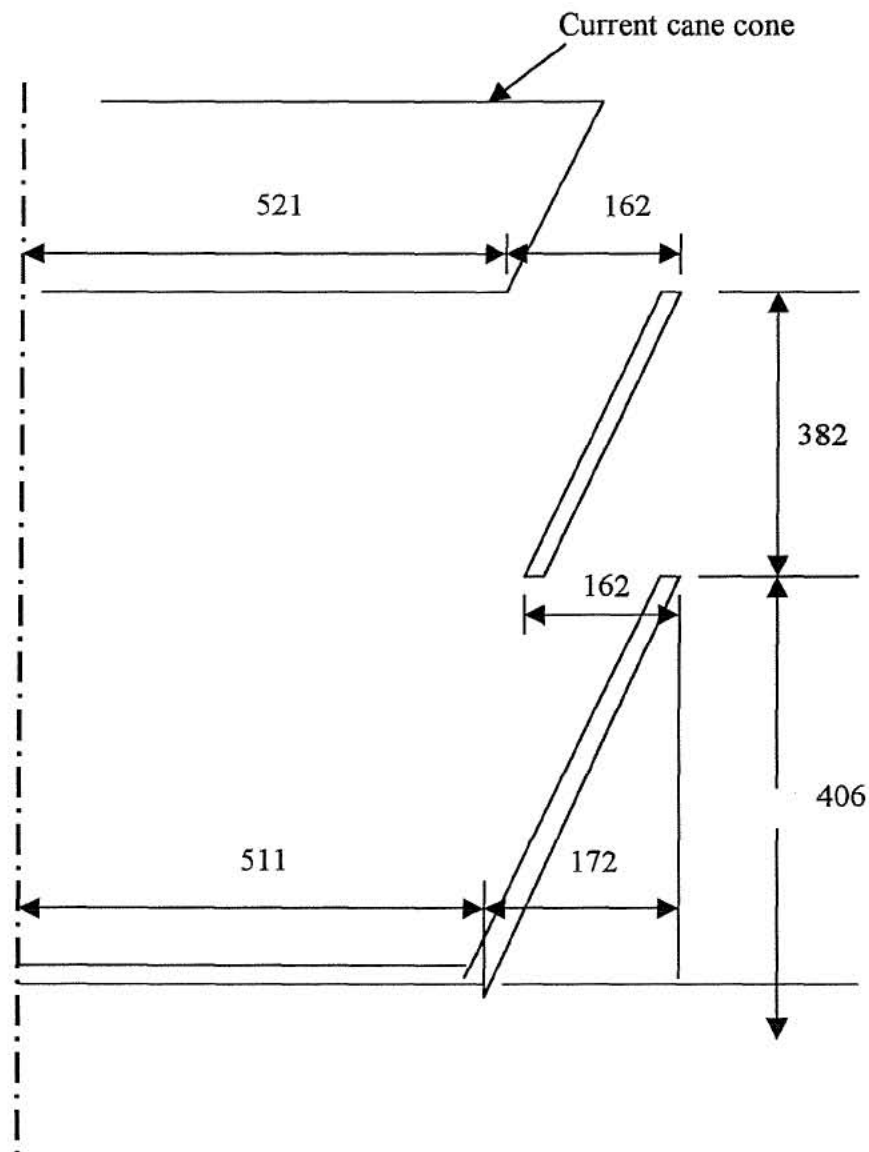


Figure C.5 - 2D vector plot of extraction chamber air flow with a pour rate 180 t/h.

- that the angle of inclination of the baffles be at least equivalent to that of the current cane cone ($\sim 67^\circ$). A baffle surface inclined at a shallower angle could cause cane to 'sit' on these surfaces and disrupt the air flow;
- to prevent the spillage of cane when the enclosed boot is being used as a temporary storage space (turning between rows), the lower edge of any baffle should be in the same plane as the upper edge of the baffle below it.

These three criteria in effect limit the number of baffles to a single set with the basic dimensions shown below (all dimensions in mm).



The profile shown above extends from the lower edge of the cane cone to the lower edge of the angled conveyor that forms the base of the elevator boot. The corresponding opposite side section of the extraction chamber is assumed to be enclosed by the conveyor section but otherwise open and free of baffles below the cane cone.

Figure C.6 shows the simulated flow (in a plane at right angles to the length of the harvester) associated with the enclosed boot configuration. The main conveyor belt is assumed to be exposed. The flow distribution is good with no major velocity peaks and only small regions of recirculation on the inside wall of the extraction chamber. This latter improvement is due to a reduction in the volume flow of air through the annulus between the extraction chamber and the top edge of the cane cone, which in turn is due to the large flow area and low flow resistance offered by the extended baffled side wall. In addition, the total flow through the extraction chamber is increased with a mean velocity that is 5% higher than the 'open' configuration.

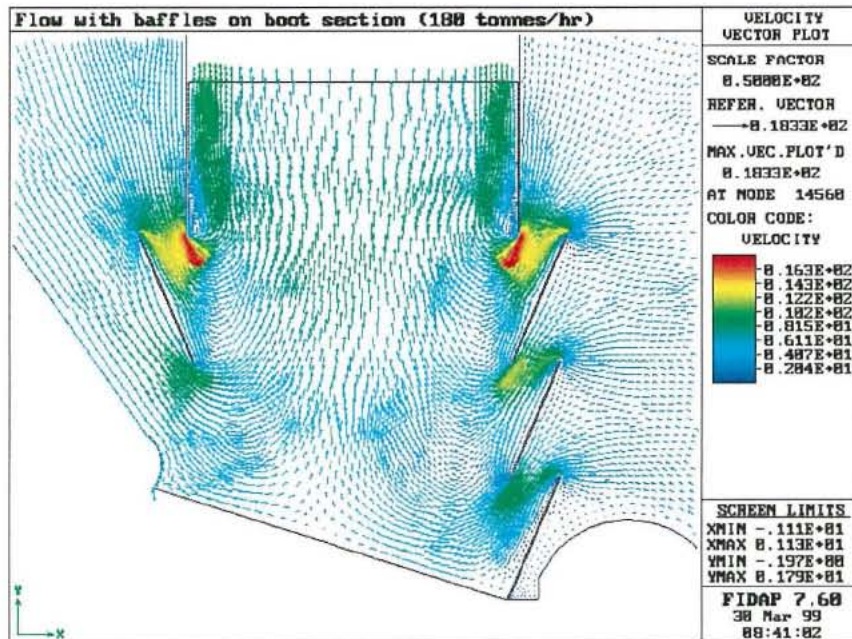


Figure C.6 - Vector plot of extraction chamber air flow in a plane at right angles to the length of the harvester.

The predicted air flow through the baffles at the rear section of the extraction chamber is shown in Figure C.7. Although the flow distribution is improved relative to the 'open' configuration, there is still considerable recirculation behind the back wall of the extraction chamber.

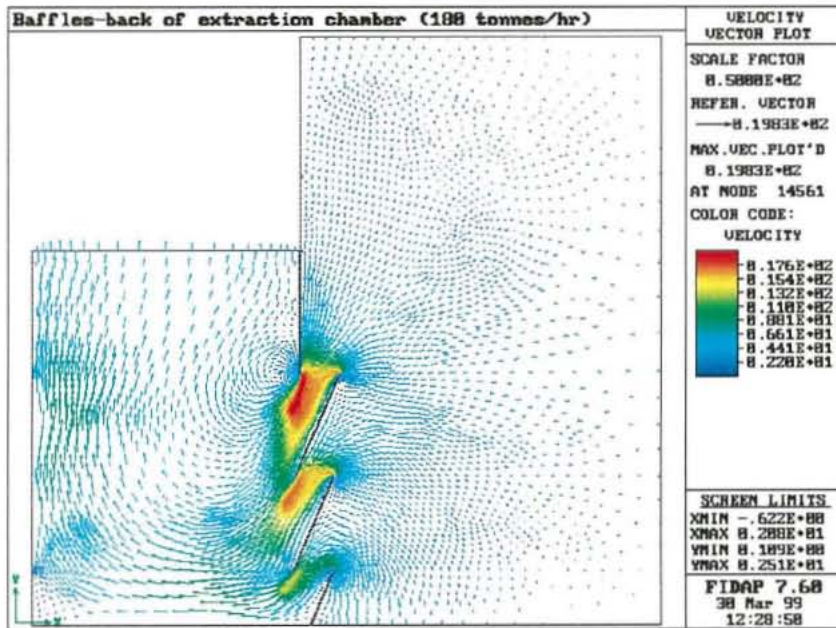


Figure C.7 - Vector plot of the predicted air flow through the baffles at the rear section of the extraction chamber.

An option for increasing the open area at the back of the extraction chamber (without hindering the motion of the elevator as it is slewed), has been investigated. This involves configuring the boot to have a centre of rotation which is displaced backwards by 150 mm relative to its current alignment with the centre of rotation of the primary extractor fan. This configuration permits the use of an elevator boot with a maximum diameter that is correspondingly 150 mm greater than is possible when the centre of rotation of the boot coincides with that of the primary fan axis.

This configuration was investigated using the CFD model. Some improvement (in terms of the flow distribution) is observed with this configuration. However, mounting the conveyor and rotating primary fan hood on different centres of rotation introduces additional mechanical complexity associated with the conveyor slewing mechanism. The improvements in air flow associated with displacing the centres of rotation of these components does not warrant its implementation.

C.3 Conclusions and recommendations

An 'open' boot configuration for the side walls of the boot/extraction chamber gives good flow distribution but does not provide sufficient cane containment capacity.

An enclosed boot configuration with baffles in the side wall provides increased containment capacity relative to the open design without compromising the flow

distribution. A 5% increase (relative to the open system) in the mean velocity is achieved with this configuration (reduced flow losses).

A maximum number of baffles in the enclosed boot system is recommended based on practical considerations relating to the prevention of vent blockages by trash.

Both the open and enclosed configurations produced significant recirculation of air entering through the back section of the extraction chamber. The enclosed system exhibited some improvements in this respect. Modifications to improve the flow in this region introduced increased mechanical complexity that could not be justified by the predicted improvements.

The flow is largely unaffected by the presence of the elevator and the motion of the main elevator belt. The belt, if left uncovered, will assist in feeding 'stray' billets or cane stored while the harvester turns at the end of a row, back onto the main feed belt on the floor of the elevator boot.

APPENDIX D

APPENDIX D MECHANICAL DESIGN OF A HIGH-SPEED ELEVATOR FOR A CANE HARVESTER

David Kaupilla, JCU

D.1 Design scope

This section outlines the mechanical design of a high-speed elevator and advanced secondary cleaning system. The primary design parameters were defined by two main points.

The mass of the new elevator system should be kept to a minimum to effect a forward and downward shift in the centre of mass of the entire harvester. The decisions made during the detail design phase should ensure a minimum 'whole of life' cost for the new component.

The second of these main points can be further expanded to better define the 'whole of life' cost as it applies to this particular project.

While minimum mass is a key design parameter, fatigue resistance must become a dominant factor when selecting materials and manufacturing practices. Manufacturing cost must be kept to a minimum for the project to be commercially successful. Maintenance costs to the end user must be minimised for the product to gain full acceptance in the market place.

D.2 - Design details

D.2.1 - Design constraints

- The *spatial envelope* is defined by current harvester and transporter geometry. The new component must retrofit with only minor harvester modifications.
- *Belt roller* geometry is dictated by computational and experimental work carried out by JCU and BSES.
- *Secondary blower* design is dictated by computational and experimental work carried out by SRI and BSES.
- *Polymer components* will be designed, in consultation with Gough Plastics, with the ability to manufacture by a rotational moulding process as a key consideration.
- The belt width will be 900 mm.
- No specialised manufacturing requirements.
- Low weight.
- Low maintenance[†]

[†] Low maintenance was defined by all parties involved and is discussed further in Section 2.2.

D.2.2 - Conceptual design

The initial conceptual design was performed by BSES with some assistance from Austoft staff. After examining their proposals, two possibilities emerged for the prototype design. The first of these was referred to as concept one, and was similar to the test rig built for BSES, being based around a single sheet metal structure with all components attaching directly to it.

The second possibility was referred to as concept two, and used a truss frame, similar to the current elevator, to which separate sheet metal and polymer 'modules' and other isolated components would attach. Given Austoft's current level of tooling, neither of these designs would have presented any notable manufacturing problems.

However, from a maintenance perspective, it was thought that the concept based on the sheet metal structure would have been more challenging and more time consuming. This was attributed mainly to the difficulties of gaining physical access to the interior of the structure coupled with the necessity to remove and refit each of the minor components individually to effect belt replacement. The open truss frame and component modules of the second concept were seen to be a more 'maintenance- friendly' alternative in comparison.

The mass of the sheet metal structure was also identified as a problem. In order to obtain acceptable stiffness over such long and relatively narrow plate sections, stiffening members would be required. This would entail either building a frame around the sheet metal, as with the current design, or alternatively using multiple plate sections, with folded stiffeners, joined along lapped folded edges. The first of these options could be seen as merely adding excess plate mass to a frame which, if designed correctly, would be robust enough to carry the service loads essentially unaided. The second option introduces additional manufacturing expense and unnecessary complexity. While properly designed free standing plate structures are quite suitable under static loading, their reliability cannot be guaranteed given the complicated dynamic loading conditions experienced during field operation.

In conclusion, an analysis of the stiffness versus mass problem clearly highlights the advantages of employing some form of truss frame as the primary load carrying structure. Given this, and the maintenance concerns mentioned previously, the second concept was adopted.

D.2.3 - Detail design

D.2.3.1 - Preliminary work

The primary considerations when designing a minimum mass, dynamically loaded structure are geometry and fatigue resistance, both of which influence the selection of construction materials.

The appropriate form for the structure was previously defined in the conceptual design phase (ie truss style frame). The geometric aspect of the detail design phase was now reduced to the correct placement of frame members such that the induced stresses were

evenly distributed throughout the structure. With the use of computational design tools, this task was economically achieved during the initial stages of the modelling process.

In contrast, the problem of fatigue resistance requires considerable forethought. Fatigue is by nature a complicated topic. Even with today's advanced analytical and experimental capabilities, our scientific understanding of this phenomenon is relatively limited. If a dynamically loaded structure were designed using only the principles of static stress analysis (ie design based on material yield strength), it would most likely suffer premature failure.

It is true that if the same structure were designed using static design principles with a substantial factor of safety, say $n = 4$, such a failure may be avoided. However, this is a poor option when designing for minimum mass. To cope with this problem in a more structured manner, engineers have devised design principles based on cumulative statistical data from many years of experimental research. One of the key findings of this research is the concept of a stress based *endurance limit*, S_e , for ferrous metals. This value defines the upper stress limit at which the designer can, under conditions of elastic strain, expect infinite fatigue life.

Most polymers, aluminium alloys, magnesium alloys, and other non-ferrous metals do *not* have an endurance limit. While the manufacturers of these materials often quote fatigue strengths, the data presented are commonly based on a finite life of between 1×10^8 and 5×10^8 cycles. This fact is sufficient to exclude polymers and non-ferrous materials from the list of possible candidates for structural members.

To minimise the manufacturing cost of the prototype, commercially available high strength low alloy carbon steel was chosen. Further, to simplify the manufacturing process and to ensure adequate strength in all modes of loading, RHS sections were favoured.

Having established the material and section to be used for the structural components, a starting point was required for the sizing of these members. Initial hand calculations were performed using a simple truss frame with best estimate loads applied. From these results, an initial section size of 50 mm x 50 mm x 2.5 mm was selected. The remainder of the design work was performed using I-Deas™ CAD software.

D.2.3.2 - Solid modelling

The use of solid modelling and FEA software allowed a continual monitoring and refinement of the form, mass, centre of gravity, dynamic loading, and stress state of the design. This software approach also allowed alternative configurations to be examined efficiently, and the implications of any modifications to be assessed immediately. Following this iterative process, a final prototype design was arrived at. An illustration of the solid model is shown in Figure D.1. The relevant solid model output data are listed below.

System mass :	approx. 780 kg (w/o belts, bowl, or blower fans)
COG relative to slew point :	1,932 mm behind and 2,120 mm above
Dynamic load factor :	1.7 (excitation = 300 mm amplitude @ 1 Hz)

This compares favourably with the current design, which has a mass of approximately 1,300 kg with the centre of mass in approximately the same position.

D.2.3.3 - FE analysis

An integral part of the concurrent computational design process was the development of a finite element model of the structural members for the purposes of stress analysis. The model was constructed from generic isotropic steel with the following material properties.

Modulus of Elasticity :	207 GPa
Density :	7,820 kg/m ³
Poisson's Ratio :	0.3

The RHS frame was meshed with 735 beam elements using 761 nodes. The sheet metal components were meshed with 2829 eight node quadrilateral thin shell elements using 3311 nodes. The sheet metal was constrained to the frame with rigid elements suitably positioned to simulate the weldments. The mesh can be seen in Figure D.2.

The midpoints of the two pivot centre lines were restrained to allow axial rotation only. Translational restraints were added at the lift ram pins to simulate operation at a fixed angle of inclination.

The entire structure was loaded by a gravity induced body force (adjusted for dynamic loading). A worst case distributed load of 2.5 kN/m was applied along the entire length of each hood support member. This was intended to simulate the dynamic loading of the blower hood assembly and top belt head assembly (assumed combined mass of 200 kg) including a factor of safety of $n = 2$.

The model was solved using a linear elastic, quasi-static analysis. Stress analysis results, shown in Figure D.3 show a maximum Von Mises stress of 96.9 MPa at the midpoint of the hood support members. A generally accepted value of the endurance limit for welded mild steel ($S_{UT} = 350$ MPa) structures is approximately 80~100 MPa. Since the suggested material specification was for high strength ($S_{UT} > 400$ MPa) steel, and the actual mass of the blower hood will be approximately 60~70 kg (not 100 kg as was assumed for the FE model), this level of stress (96.9 MPa) should be manageable. As a further precautionary measure, hood support gussets (not present in the FE model) were added to the structure. The remainder of the frame showed stress levels below 60 MPa (see inset in Figure D.3).

D.2.4 - Design outputs

The workshop drawings produced for manufacture and assembly of the prototype are presented in a separate volume with both hard and soft copies accompanied by a Drawing Register. The DXF files to produce the sheet metal parts in a NC laser cutter are also contained on the enclosed CD.

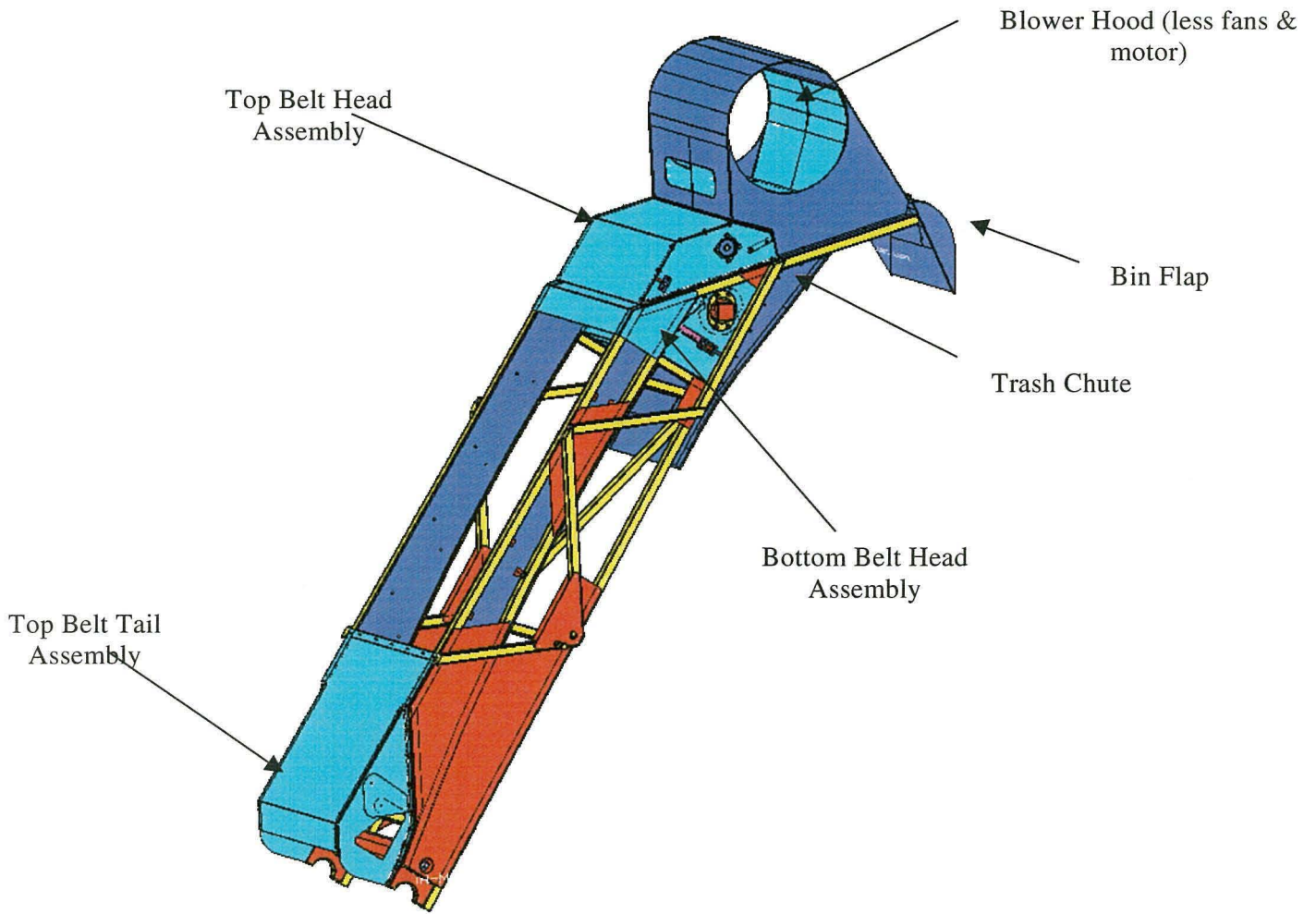


Figure D.1 - Solid model of element

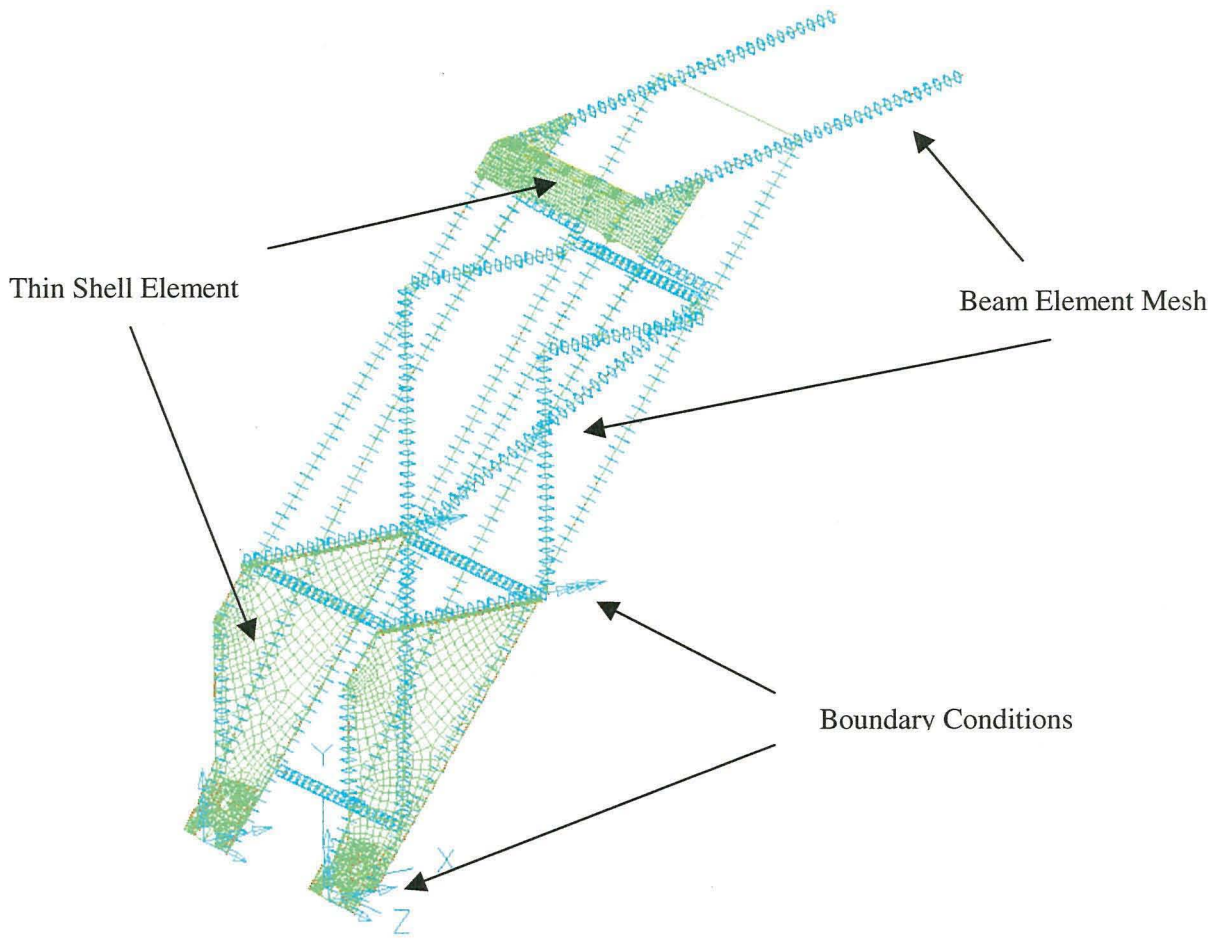


Figure D.2 - Finite element mesh and boundary conditions

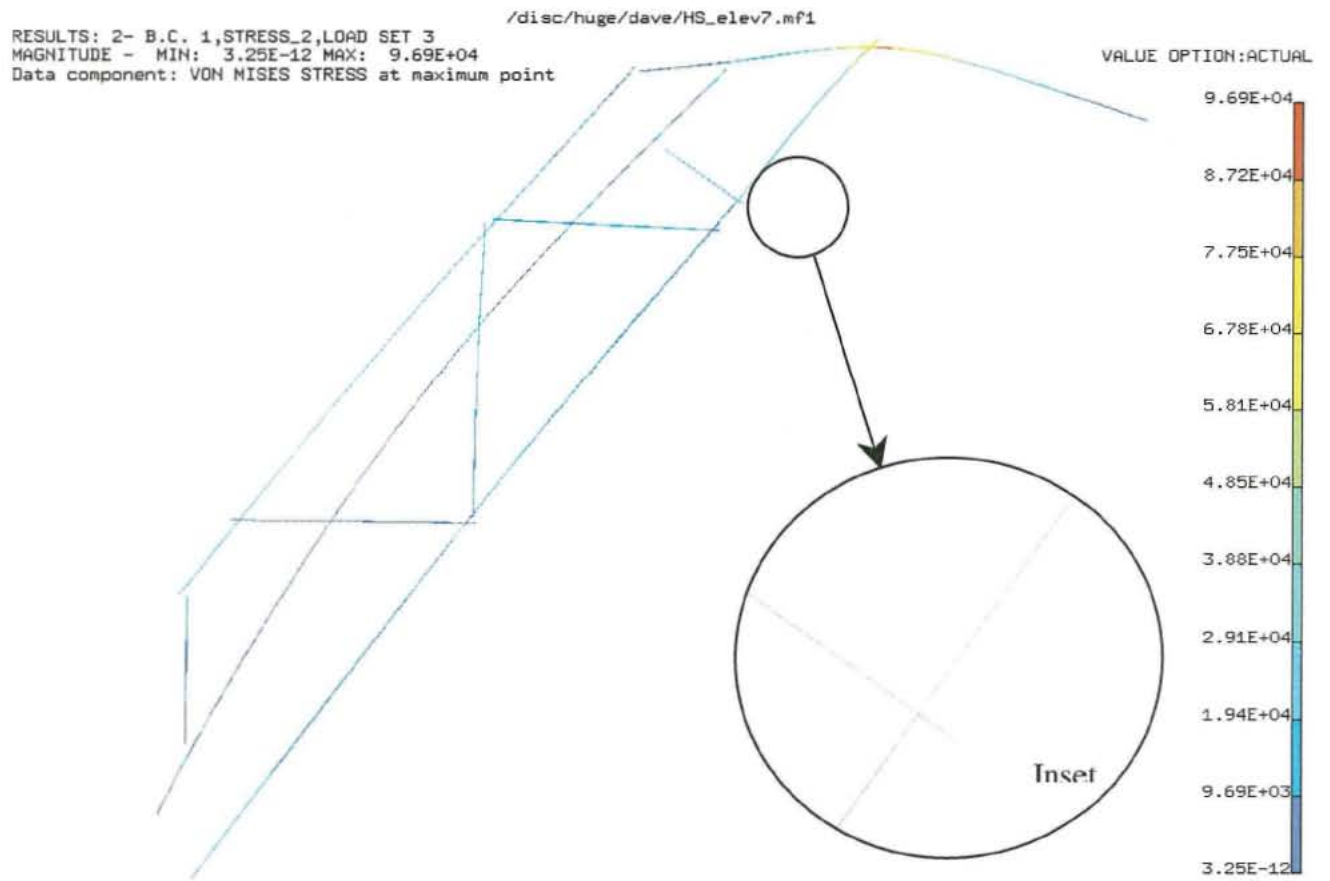


Figure D.3 – Stress analysis results

APPENDIX E

APPENDIX E DILUTE PHASE MODEL INPUTS FOR SECONDARY CLEANING SYSTEM

C P Norris

Trajectory of Particles
Particle Classification:

Particle Mass: (kG)
 Terminal Velocity, VF: m/sec
 Axis perpendicular to airflow (90°)
 Terminal Velocity, VF: m/sec
 Axis parallel to airflow (0°)

Large Billet	Small Billet	Cabbage Billet	Leaf
0.2	0.100	0.05	0.008
25	18	12	5
35	25	20	5

Rem: $\text{airdrag } F = M \cdot G = k \cdot \text{Area} \cdot VF^2$

Rem: $k \cdot \text{Area}$ constant for a given particle, kA
 Rem: $G = 9.8$

Particle input conditions

Initial Position: Sx
 Sy
 Initial Angle (axis to horizontal)

Rem: Use conventional Cartesian co-ordinates

0
0
30

Trajectory Angle degrees
 Trajectory Angle rads
 Billet Velocity m/s

Airflow Conditions:

Air Velocity: (m/sec)
 Airflow angle (deg)
 Airflow angle (rad)
 Cartesian Airflow Vax
 Cartesian Airflow Vay

Zone 1	Zone 2	Zone 3	Zone 4
0	-40	0	0
0	70	0	0
0.00	1.22	0.00	0.00
0.00	-13.68	0.00	0.00
0.00	-37.59	0.00	0.00

Vbx m/s
 Vby m/s

Time	Lrge Bill X	Lg Bill Y	Sml Time	Sml Bill X	Sml Bill Y	Cabb time	Cabb X	Cabb Y	Leaf Time	Leaf X	Leaf Y
0	0.00	0.00	0	0.00	0.00	0	0.00	0.00	0	0.00	0.00
1.5	0.01	0.00	1.5	0.01	0.00	1.5	0.01	0.00	2	0.01	0.00
3	0.02	0.01	3	0.02	0.01	3	0.02	0.01	4	0.03	0.01
4.5	0.03	0.01	4.5	0.03	0.01	4.5	0.03	0.01	6	0.04	0.01
6	0.04	0.01	6	0.04	0.01	6	0.04	0.01	8	0.05	0.02
7.5	0.05	0.02	7.5	0.05	0.02	7.5	0.05	0.02	10	0.06	0.02
9	0.06	0.02	9	0.06	0.02	9	0.06	0.02	12	0.08	0.03
10.5	0.07	0.02	10.5	0.07	0.02	10.5	0.07	0.02	14	0.09	0.03
12	0.08	0.03	12	0.08	0.03	12	0.08	0.03	16	0.10	0.04

APPENDIX F

APPENDIX F COMPUTATIONAL FLUID DYNAMICS MODELLING OF SECONDARY EXTRACTOR AERODYNAMICS

Dr P Hobson, SRI

F.1 - Aim of the aerodynamic analysis

Earlier cane particle trajectory predictions and tests indicated that narrow jets provided improved separation of trash and leaf from cane billets relative to broader jets. The trajectory model analysis has shown that from the consideration of simple aerodynamic drag on cane particles and the more complex particle-particle interactions, a continued reduction in the width of the air jet (ie higher air jet velocities for a constant volume flow rate), produces progressively improved cleaning with no apparent limit on this effect. The lack of cane loss predicted by the particle model at the higher air velocities investigated is attributed to the lower residence time of the cane in the narrower air jet.

In practice, there may well be limits on the minimum width of the air jet. The energy from thinner, higher velocity air jets will dissipate more rapidly compared with a broader jet of the same volume flow rate. This energy dissipation may be particularly rapid as air passes through the cane mat. With these issues in mind, an analysis of the aerodynamics of the interaction of the air jet and cane was carried out to determine:

- the extent to which the air jet is dissipated or deflected as it passes through the moving cane stream (deflection of the jet would entail the careful design of additional duct work to direct the trash after separation); and
- the ability of the air jet (in terms of the magnitude of the air velocities) to transport trash removed during cleaning to a location remote from the cleaned cane supply (ie the ground between haul out and harvester).

F.2 - The model

As with the primary extraction chamber, the computational fluid dynamics (CFD) code FIDAP has been used to model air flows from a notional secondary cleaning system. In addition FIDAP is used to simulate the aerodynamic effects (entrainment) due to the presence of billets in the air flow. To investigate this latter effect, the model is set up such that there is a two-way exchange of momentum between the cane particles and the air. This means that not only does the air have a retarding effect on the billet, but the billet has an accelerating effect on the air. This model will simulate aerodynamic and gravitational forces on the cane billets but (unlike the particle model) does not simulate direct mechanical interaction (impacts) between the particles.

A two-dimensional model has been set up to simulate the harvester in normal operation. The elevator is assumed to be in the sideways-slued position and feeding cane into an adjacent haulout vehicle. The distance between the centre lines of the two vehicles is set at 4.8 m. The dimensions of the harvester and installed lightweight elevator are as supplied by BSES. A value of 3.750 m is assumed for the maximum bin height. These

combined dimensions give a minimum (horizontal) distance of 0.49 m between the elevator belt and the nearest point on the body of the haulout.

The simulation domain consists of the region bounded in the horizontal direction by the adjacent harvester and haulout vehicles. In the vertical direction, the domain extends between the ground and a plane set (appropriately far removed but otherwise arbitrarily) at a distance of 1.5 m above the main body of the secondary air jet. The main bodies of the two vehicles are assumed to be impervious to flow, whereas air is free to flow out of the domain through the chassis region (ie the top of the wheels downwards). The elevator belt is assumed to be uncovered and to have a downwards surface speed of 6 m/s.

The cane is assumed to be deflected immediately after launch such that it has an effective initial trajectory angle of 30° to the horizontal. The undeflected launch angle of 60° was found to produce a cleaning chamber, which was excessively extended in the direction of launch. The air jet is positioned such that the centre line of the jet is located 1.042 m horizontally and 0.609 m vertically from the cane launch point. These distances are the minimum required such that the air jet can be directed to flow through the smallest distance between the harvester and haulout and in a direction parallel to the elevator belt. These distances are also chosen to ensure that the cane passes just beneath the air jet exit without striking the main body of the jet.

F.3 - Simulated flows

Three air jet widths have been investigated using a single volume flow rate of 11.25 m³/s and an assumed duct depth (normal to the plane of the simulation) of 900 mm. A cane billet pour rate of 180 tonnes/hour (50 kg/s) is simulated in all cases. These jet widths and corresponding air velocities are given in Table 1.

TABLE F.1
Air jet widths and corresponding velocities investigated
using the FIDAP model

Run	Jet width (mm)	Jet air velocity (m/s)
1	250	50
2	400	31
3	100	125

F.3.1 - Run 1

Figure F.1 shows speed contours for run 1. It is immediately evident from an inspection of Figure F.1 that there is minimal predicted disruption of the air jet by the moving cane. There is also little evidence of air entrainment due to the motion of the cane billet stream; any evidence there is occurs at the initial launch point. An adjustment of the contour colour scale indicates that entrainment velocities are below 3 m/s and do not extend more than 100 mm on either side of the cane stream.

The simulation of billets only may underestimate the entrainment effects of the cane stream. The sensitivity of this entrainment effect has been tested by increasing the billet pour rate by a factor of 4 to 200 kg/s. Figure F.2 clearly indicates that even at this enhanced pour rate there is little discernible effect on the air jet.

With regards to the persistence of the jet and the its ability to transport separated leaves to the ground, Figure F.1 indicates air velocities of around 7 m/s at ground level and extending over much of the ground between the haulout vehicle and harvester. These velocities are more than adequate to transport the leaf after separation. Under dry conditions the air velocities involved may cause dust problems.

Figure F.1 also indicates a tendency for the jet to attach itself to the moving conveyor belt surface. This 'self-guiding' effect will facilitate the directing of the air and trash away from the haulout.

F.3.2 - Run 2

The predicted air flow for the broader, slower jet is shown in Figure F.3. Again there is little indicated disruption of the jet by the cane stream. It is evident from Figure F.3 that further duct work is required to ensure that the air is directed between the two vehicles.

F.3.3 - Run 3

The predicted flows for run 3 exhibit similar traits to those from run 1 although velocities are higher. There is little evidence that the cane significantly dissipates the energy of the jet.

Predicted ground level velocities of around 14 m/s are almost certain to cause a dust problem. In addition, at these velocities, trash on the ground may become airborne and become re-entrained into (or ensnared by) the primary cleaning system.

F.4 - Conclusions and recommendations

Predictions indicate that even at high air velocities (where dissipation of the flow energy is relatively high), there is little indication of distortion or dissipation of the air stream by the passage of cane through the jet. This feature considerably simplifies the design process in that the jet can be positioned from purely geometrical considerations, ie a jet positioned at 55° to the horizontal will produce an air flow the centre line of which can be assumed (on the grounds of the above results) to continue along a line inclined at 55°.

The model indicates that both high velocity jets (runs 1 and 3) are suitable secondary cleaning systems. Although the high velocity jets are 'self-directing' in terms of the passage of the jet between the two vehicles, a duct, which follows the contours of the outer edge of the jets, should be installed as a trash guide. The air velocities below the top of the bin are such that the trash guide need only extend down to a point just clearing the bin.

The previous particle simulation work indicates that the highest velocity jet (run 3) will produce the greatest cleaning effect. The current analysis indicates that ground velocities

The previous particle simulation work indicates that the highest velocity jet (run 3) will produce the greatest cleaning effect. The current analysis indicates that ground velocities for this system are such that re-entrainment of leaf and dirt into the primary cleaning system may become an issue.

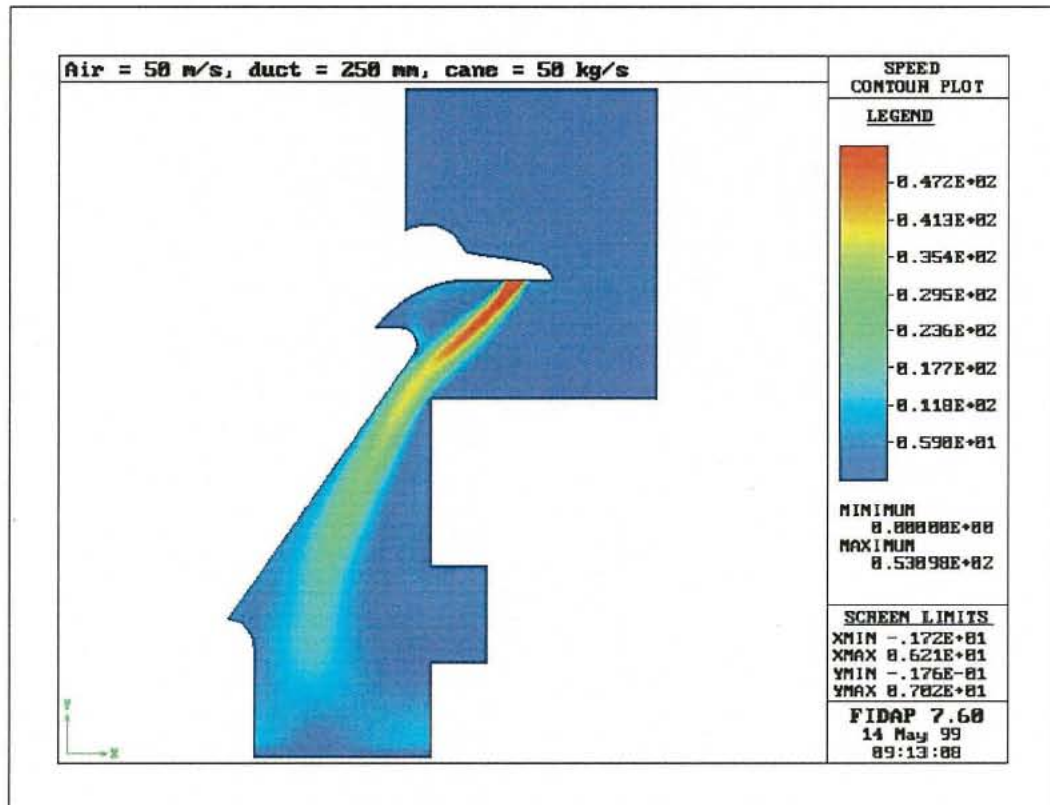


Figure F.1 - Shows velocity contours for run 1

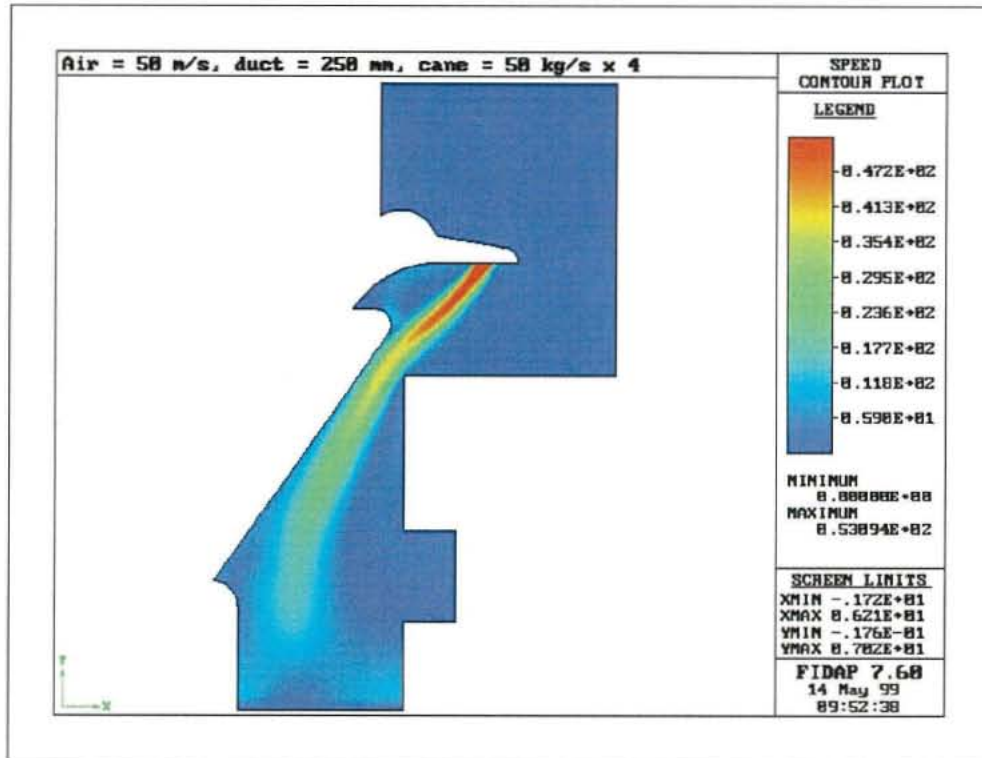


Figure F.2 – Shows velocity contours for a pour rate of 200 kg from run 1

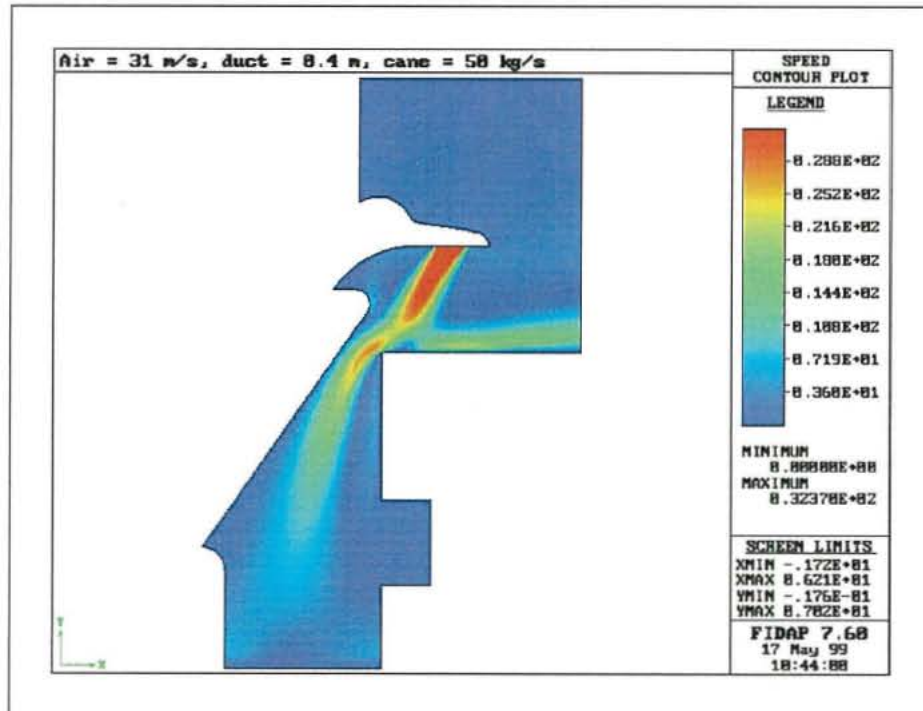


Figure F.3 – Shows velocity contours for run 2

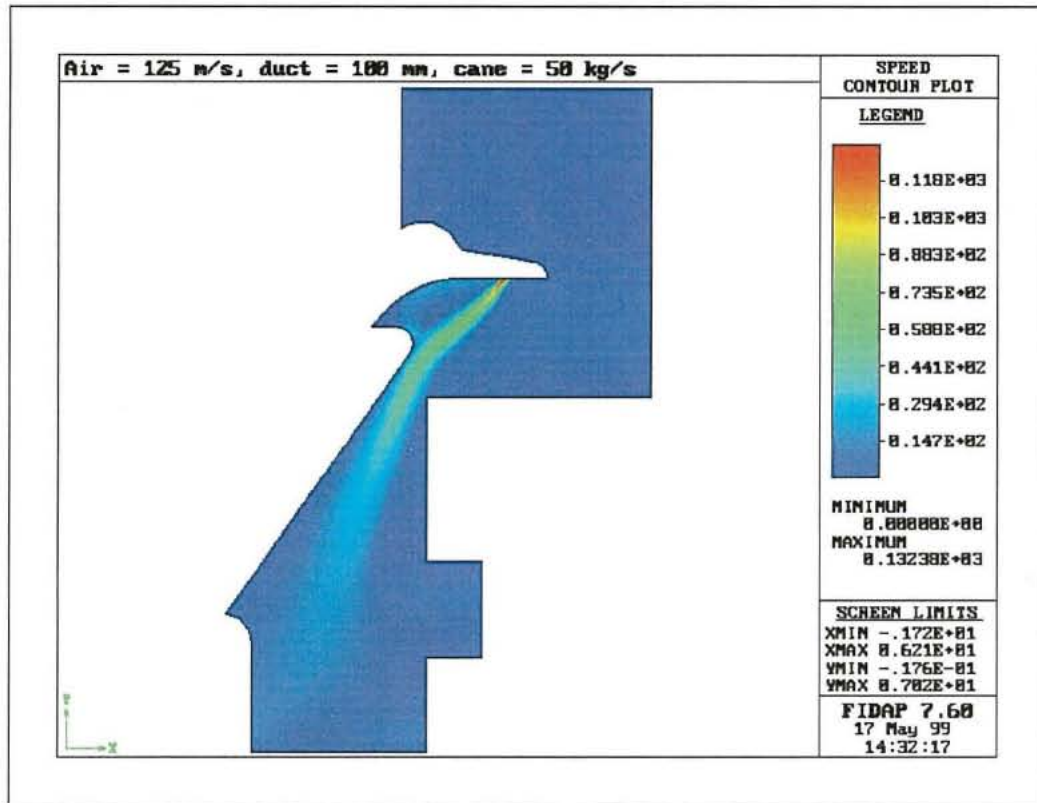


Figure F.4 – Shows velocity contours for run 3