THE AGRICULTURAL ENGINEER

JOURNAL and Proceedings of the INSTITUTION of AGRICULTURAL ENGINEERS

Volume 31 Summer 1976 No 2

NOTE THIS DATE

THE INSTITUTION OF AGRICULTURAL ENGINEERS AUTUMN NATIONAL CONFERENCE

9 NOVEMBER 1976

THE WASTE MAKERS

The Effect of Machine Design and Operator Efficiency on Crop Losses during Harvesting and Storage.

VENUE: HOTEL NORWICH, Boundary Road (A47), Norwich.

The Chairman will be Sir Peter Greenwell Bt, Chairman, Ransomes Sims and Jefferies, and a prominent Suffolk farmer.

Morning Session

10 00	Registration and coffee.
10 40	The Chairman opens the Conference.
10 45	Paper 1 A statistical review of harvesting and storage losses in cereals and pulses, sugar beet, potatoes, vining peas and beans. T W D Theophilus, Regional Farm Management Advisor, ADAS, Eastern Region.
11 15	Paper 2 The design and use of combine harvesters for minimum crop loss A design engineer from Claas, Maschinenfabrik, GmbH, W Germany.
11 45	Paper 3 The minimisation of crop losses associated with sugar beet harvesting. N B Davis, British Sugar Corporation.
12 10	Questions/discussion.
12 45	Lunch.

Afternoon Session

13 45	Paper 4 The design and operation of pea viners and bean pickers as they affect crop loss. A T Bain, Process Engineer, Birds Eye Foods Ltd.
14 15	Paper 5 The design and operation of potato harvesters for minimum damage and losses. D McRae, Scottish Institute of Agricultural Engineering.
14 45	Paper 6 Recent developments in potato storage methods. W G Burton, Food Research Institute, Norwich.
15 15	Questions/discussion.
15 55	Summing-up and closure by Conference Chairman.
16 00	Tea and depart.

Joint Conference Convenors:	J H Neville BSc (Agric) MSc (AgrE) NDA MIAgrE, National College of Agricultural Engineering, Silsoe, Bedford. U G Curson TEng (CEI) MIAgrE, Ben Burgess and Co, 43 King Street, Norwich, Norfolk.
All enquiries to Conference Secretary:	Mrs E Holden The Institution of Agricultural Engineers, West End Road, Silsoe, Bedford MK45 4DU. Telephone: Silsoe (0525) 61096.

THE
AGRICULTURAL
ENGINEER

JOURNAL and Proceedings of the INSTITUTION
of AGRICULTURAL ENGINEERS

Volume 31 SUMMER 1976 No. 2

Institution and Editorial Office:
West End Road, Silsoe, Bedford MK45 4DU
Tel: Silsoe 61096

Advertisement Manager:
LINDA PALMER

Production Manager:
NORMAN STUCKEY

Advertisement and Production Office:
PO Box 10, Rickmansworth, Herts WD3 1SJ
Tel: Rickmansworth 78877

The AGRICULTURAL ENGINEER is published quarterly by the Institution of Agricultural Engineers

ABC
MEMBER OF THE AUDIT BUREAU OF CIRCULATIONS

Price: £1.45 per copy, annual subscription £5.80 (post free in UK)

Printed by Studio Trade Plates Ltd, Watford WD1 8SA

CONTENTS

Cover: The new headquarters building of the Institution and below the key being handed over to IAgrE Secretary Ray Fryett (right).

The journal today by Brian Stenning

THE retirement of Professor Brian May from the position of Honorary Editor of *The AGRICULTURAL ENGINEER* and Chairman of the Editorial Panel provides us, as members of the Institution, with an opportunity to pay tribute to him formally and sincerely for the work which he has put into the Journal during the past four years. Under his guidance the high standards which one now associates with the Journal have been maintained and reinforced and his continued membership of the Editorial Panel will help to ensure a smooth editorial change.

Thanks are also extended to Mr J C Hawkins and Professor F M Inns, both of whom retire from the Editorial Panel as this issue goes to press.

The form and content of the Journal have undergone a number of changes over the years of its existence, each development no doubt being introduced in order to match the publication to the needs of the times. This open minded attitude is one which must be maintained by any Editorial Panel and members of the Institution may be assured that we shall continue to make every effort to meet their requirements, within the limits which are imposed by financial and similar considerations. At the same time, however, care must be taken to preserve the reputation which the Journal has justifiably attained: this reputation is based on the publication of conference reports and high quality papers which are intermediate in their appeal and approach between articles published in the widely circulated agricultural press on one hand and detailed research reports on the other. Such material, in conjunction with reports from branches and the Secretariat is surely the foundation on which our Journal should continue to grow.

The Editorial Panel, however, will always welcome constructive suggestions which might lead to improvements in the Journal from the members' point of view. And of course the invitation to members to submit papers to the Panel is ever open. Prospective authors who may feel the need of a little provocation or encouragement to put pen to paper are cordially invited to contact the Honorary Editor ———————— ————————— and in that capacity I look forward to an interesting and stimulating period of office.

Official opening of the Institution's new offices

MR FRANK E ROWLAND, the senior past president, opened the new registered office and headquarters building of the Institution, situated on the campus of the National College of Agricultural Engineering, Silsoe, on Thursday 8 April.

In introducing Mr Rowland, the President Mr John Fox remarked that this was a landmark in the Institution's history, being the first time that the Institution had owned property. That it was able to do so was symbolic of the wide and ever growing recognition of the vital importance of agricultural engineering.

"This is the era of the agricultural engineer and it is fitting that the Institution should benefit from the prestige that this building, in this setting, brings us".

Mr Fox expressed appreciation of the work of those who had contributed to the realisation of this project. He mentioned particularly the College Governors, the Cranfield Institute of Technology, the former Principal of the College, Professor P C J Payne and the present Head of School, Professor B A May; also the Head of College Administration Mr C E King. Many Institution members had worked hard and long towards the achievement, not least among them being the Institution Secretary Mr Ray Fryett, and Messrs J Hawkins and J Messer, who had acted as technical advisers during both the design and the construction.

Mr Rowland, in expressing his pleasure at being associated with the opening ceremony, felt that he represented a link between the present and the handful of men who, through their foresight, founded the Institution 38 years ago. They, by their patience and persistence had set it on a course which, in the hands of this and future generations, would carry the Institution forward and its members to wider recognition.

No more appropriate site for the headquarters could have been found than alongside the NIAE and the NCAE.

Mr Rowland declared the building open and handed the key to Mr Ray Fryett, symbolically placing the custody of the headquarters in his keeping.

New president is R&D man

THE new President of the Institution is T C D Manby BSc MSc(Eng) CEng FIMechE FIAgrE, a senior principal scientific officer and head of the machine division at the National Institute of Agricultural Engineering, Silsoe, Bedford.

David Manby joined the NIAE in 1944, shortly after its formation, after graduating with a 1st Class Honours Engineering degree from Leeds University. He entered university after an apprenticeship period at Metropolitan Vickers Electrical Co, Manchester, then abandoned his intention of seeking a career in nuclear physics, in favour of agricultural engineering. In making this decision he was combining home background interests of automobile engineering and farming and he continued to gain experience in partnership with his father on a small Yorkshire farm throughout the 'forties'.

Prior to working at NIAE, Askham Bryan, he had a short period helping supervise large earthmoving projects and this led to an interest in problems concerning construction equipment. Indeed, improvement of traction was the first task for which he gained professional recognition. Then, resulting from tyre research at NIAE, he was awarded an external Master of Science degree in 1948.

During the next 20 years he became best known for his involvement in tractor development and in international standardisation of assessment techniques. He has led British delegations to all International Standards Organisation conferences on these matters, was a consultant to FAO Rome and Vice-Chairman of the Committee of the Organisation of Economic Co-operation and Development "working to assist the flow of agricultural machinery in trade".

The Institution of Mechanical Engineers presented him with the Engineering Applied to Agriculture Award, 1962, the Filtration Prize in 1965 and their Automobile Division highest award — the Crompton Lanchester Medal.

At NIAE David Manby became head of the newly formed testing division in 1965 and subsequently the tractor and machine division, where major interests were cultivation and forage conservation, research on tractor safety cabs and aspects of comfort and convenience leading to improved worker health and machine output. Now within machine division the latter continues with the addition of interests in machine design for vegetable, soft fruit and sugar beet growing, in materials handling, in basic research on wear of soil engaging materials and in tractor-machine drive line problems.

The President has been an active member of the Institution for some 25 years — as a speaker at and a convener of conferences; for the past eight years he has been a member of Council.

Overload protection for p t o driven agricultural machinery

A A W Chestney

Abstract

THE work described is incomplete but is published at the present time to attract interest and attention to a difficult problem area.

Field and laboratory tests have been made to evaluate the performance of a number of overload protection devices used on agricultural machinery. These have included overload clutches of the plunger and ball ratchet, combined plunger and friction, and friction plate types. Different types of overloads imposed on pto driven agricultural machinery are discussed including complete blockages, high intermittent loading due to soil or crop conditions and high starting torque values imposed on the drive lines of high inertia machines when using tractors with hydraulically engaged pto clutches. The need for overrun on high inertia machines is also discussed.

In general, the performance of all the overload devices evaluated was inconsistent and breakaway torque values were spread over a wide range. In cases where extreme overloads occurred some overload protection for the machine was provided. The investigation has shown that 'running-in' of the clutch plates did help towards producing more consistent breakaway torque values and that after periods of storage clutch performance deteriorated for various reasons including contamination, lack of lubrication and adhesion of components caused by corrosion.

1 Introduction

To the farmer, timeliness of operation is important and therefore the reliability of machinery is necessary. Because of the nature of materials handled in agriculture, almost all farm machinery is subjected to load variation and overloads frequently occur. In order to ensure reliability it is therefore necessary to fit overload protection devices. If efficient overload protection is employed the benefits obtained will be:

(a) Increased operational reliability by reduced breakages and lost time when critical seasonal operations are in progress.
(b) The use of more realistic safety factors in machine design which may reduce capital costs.

A survey[1] of the durability of farm machinery has shown that breakages often occur due to overloads which could be prevented by using more reliable and efficient overload protection devices. Work at NIAE was therefore started to evaluate the performance of existing overload protection devices as fitted to agricultural machinery with the object of making improvements where necessary. It has been concerned with measuring the load patterns to which overload devices are subjected, observing their effectiveness in eliminating overloads and investigating by means of laboratory tests those factors which most affect performance.

Whilst this work has been in progress, tractors and machines have become larger, greater use has been made of 1000 rev/min pto shafts and hydraulically engaged pto clutches have been introduced. On some designs the driver has no control over the rate of engagement and often the clutch is fully engaged in less than 0.8 s. This has resulted in a greater need for reliable overload protection devices in general, particularly for controlling the starting torques on high inertia machines.

2 Types of overload

Types of overload occurring in pto driven agricultural machinery are:

(a) Complete and sudden blockages eg when fouling a large stone or tree root. This can be experienced on all types of pto driven machines.
(b) High peak torques due to overloading in normal work or a temporary blockage. Again, this type of overload can be experienced on all types of pto driven machines.
(c) High starting torques caused by the sudden engagement of the tractor pto clutch at high engine speed. This type of overload is prevalent on machines which have high rotary inertias and particularly so when used with tractors having hydraulically engaged pto clutches.
(d) High negative torques in the drive line when high inertia machines overrun the tractor. This can occur when the tractor engine speed is reduced or the pto clutch disengaged.

3 Machine torque patterns

For most machines the torque pattern measured at the pto shaft is of a fluctuating nature. The ratio of the amplitude of the peak to mean driving torque and the frequency of oscillation varies for different groups of machines. For soil working machines such as rotary cultivators, rotary diggers and powered harrows the ratio of peak to mean torque values is high. Measurements have shown that for rotary diggers peak torques of up to twice the mean are frequently present and for rotary cultivators peak torque values of three times the mean are not unusual. Fig 1 shows a typical torque pattern for a rotary cultivator and fig 2 shows the distribution of the positive peak torque values measured during early work with the NIAE experimental rotary digger. The frequency of torque oscillations with significant amplitudes for soil working machines is in the range of 4 to 20 Hz.

For grass machinery such as rotary drum mowers and flail forage harvesters the amplitude of the fluctuating torque is much less than that for the soil working machines, but some high peak torque values often occur during normal working, as shown in table 1. However, the highest peak torque values likely to occur

Table 1 Field test data for a flail forage harvester with a pto speed of 1000 rev/min

Crop	*Maximum torque during run* Nm	*Estimated mean torque during run* Nm	*Speed* km/h	*Mean power consumption at pto* kW
Tall fescue	630	400	3.7	39.6
	850	495	5.6	48.0
	790	580	7.2	51.7
	1110	615	8.6	57.8
	925	380	3.7	38.8
	945	565	5.6	56.8
Cocksfoot	1305	670	7.2	64.9
	1130	640	8.6	64.6
	1115	635	11.2	59.4
	325	130	3.7	13.2
S23	780	360	5.6	35.1
Ryegrass	960	585	7.5	57.1
	1155	805	8.7	85.0
	1005	560	11.1	49.1

A A Chestney is from the Machine Division, NIAE, Wrest Park, Silsoe, Bedford.

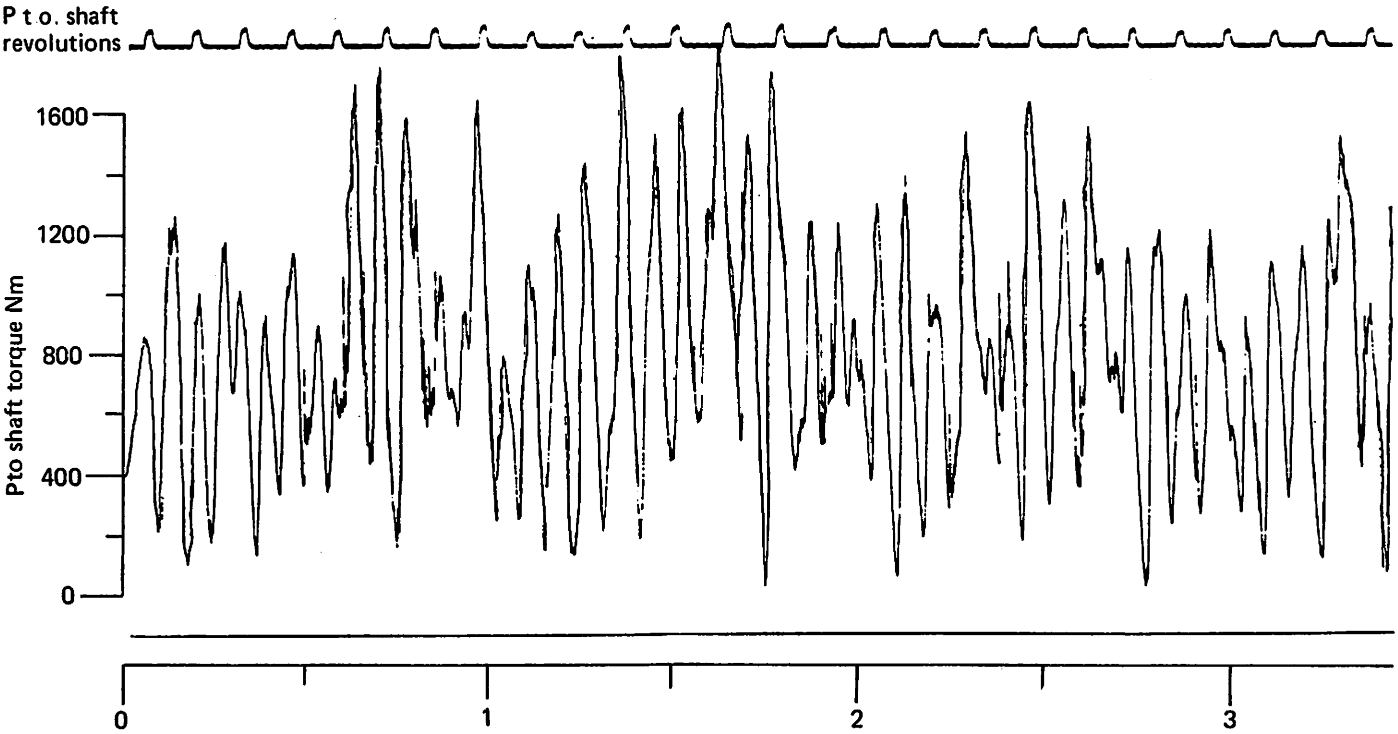


Fig 1. Pto torque pattern for rotary cultivator working on heavy grassland

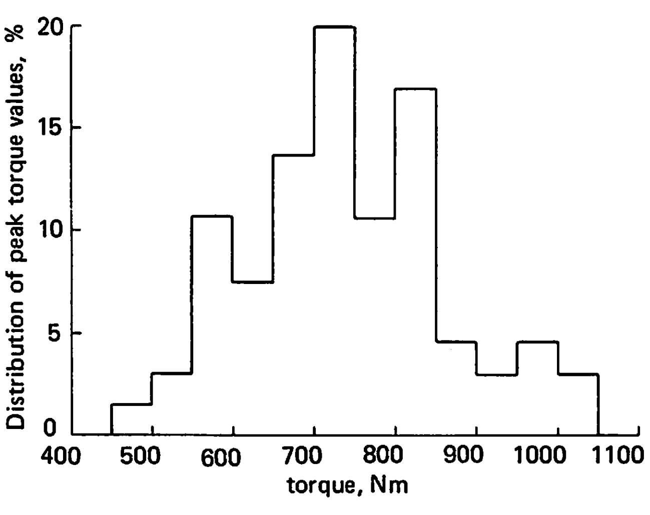


Fig 2. Distribution of peak torque values for NIAE experimental rotary digger with a pto speed of 540 rev/min: mean driving torque 520 Nm.

with this type of higher inertia machine are those during starting[2], and particularly so if used with tractors having hydraulically engaged pto clutches. Table 2 gives some values of torques measured

Table 2 Data for starting tests on a drum mower without slip and overrun clutches

Tractor		*A*	*B*	*C*
Maximum peak torque value	Nm	1650	2150	2490
Mean value of peak torques for a range of engine speeds	Nm	1370	1760	2310
Minimum torque value during starting	Nm	-520	-1750*	-2220*
Minimum torque value during overrun	Nm	-780	-1010	—

*Engine stalled

when starting a 2.13 m wide — 2-drum rotary mower with three different tractors. These measurements were made for a range of engine speeds with each tractor. Tractor A had a manually operated mechanical pto clutch and tractors B and C had hydraulically operated pto clutches. At the time this machine had neither slip or overrun clutches and when the engine stalled whilst attempting to start the machine, severe torque reversals were applied to the whole drive line as shown in table 2. During overrun, high values of negative torque were recorded for tractors A and B but none were recorded for tractor C because the pto brake was not working. These starting torque values represent over 4 times the mean working torque and over twice the maximum peak working torque value.

Baling machines have a different type of load pattern where high torques are produced during the compression stroke. These machines are usually fitted with flywheels to help provide energy for compression and to smooth out the load cycle. Without slip and overrun clutches, some high positive and negative torque values would be produced.

The above examples of machine torque patterns show the need for overload and overrun protection devices to control the maximum torques being induced in machine drive shafts and gear boxes.

4 Overload protection devices

The main types of overload protection devices used on pto driven agricultural machines are:

(a) Shear pins. These are used to prevent breakages due to occasional excessive overloads when it is necessary to stop and clear the obstruction.

(b) Ball or plunger ratchet type overload clutches which are designed to slip and give an audible warning when a machine blockage occurs. It is usually necessary to stop and clear the blockage before re-engaging the drive. Some designs, fig 3, have a combined friction lining and plunger ratchet arrangement, which has the effect of providing a higher mean slipping torque. This type is therefore more suitable for removing overload peak torques whilst continuing to drive the machine.

(c) Friction plate clutches, an example of which is shown in fig 4, are used on machines where it is necessary to control starting torques and also to prevent overloads occurring during normal work. These are designed to slip enough to control the torque at a set value but are intended to continue to transmit torque during slip periods. The friction materials may be either organic based or sintered metal and the clamping pressure is applied either by a series of coil springs around the periphery or by a central disc spring.

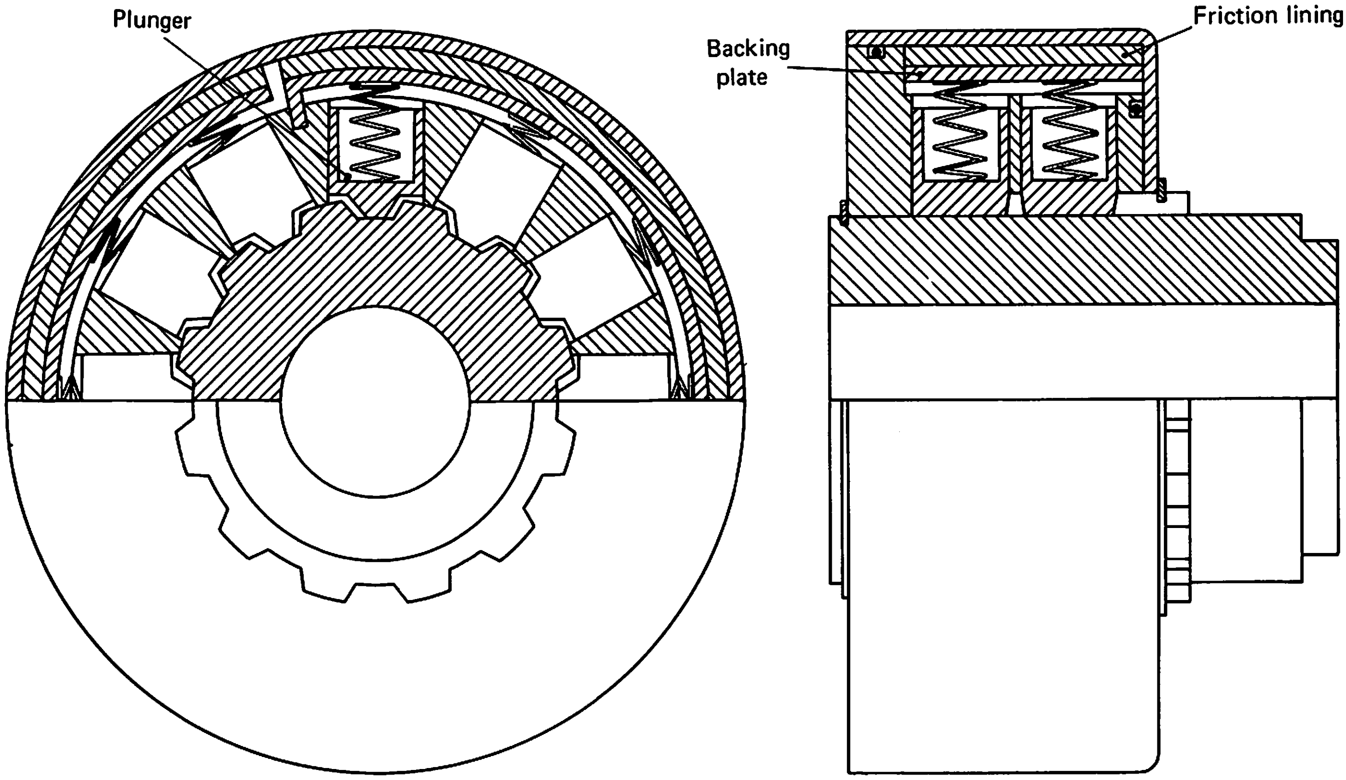
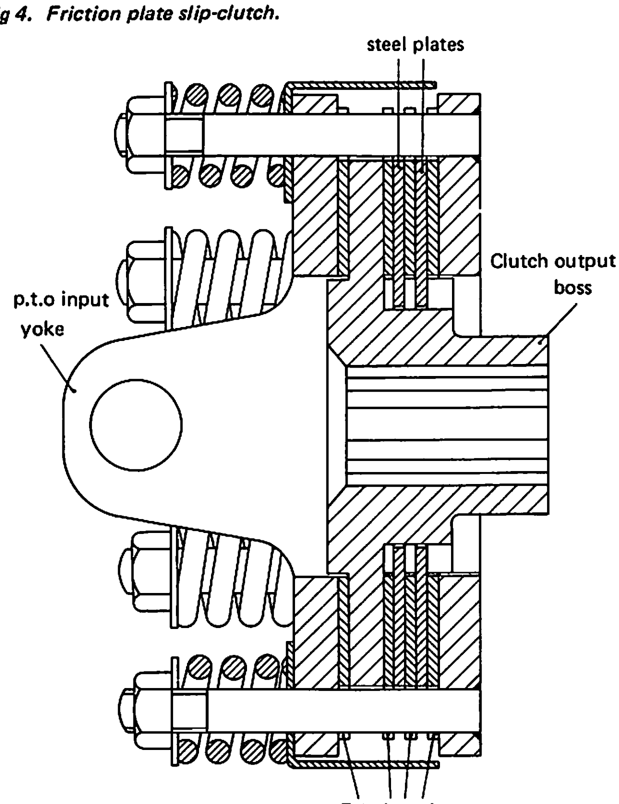


Fig 3. Combined plunger ratchet and friction type overload clutch.

(d) Overrun or freewheel clutches are designed to allow high inertia machines to come to rest without driving the tractor through the pto, when the tractor speed is reduced. Various designs are in use on agricultural machinery including
(i) axial spring loaded pins which engage with a cam plate;
(ii) radial spring loaded keys which engage with an outer circumferential cam arrangement;
(iii)hinged pawls which engage with a circumferential ratchet due to centrifugal force.

Fig 4. Friction plate slip-clutch.



5 Experimental work

5.1 Field measurements

Several machines were instrumented in order to measure the torque patterns at the pto shaft and the effectiveness of the overload protection devices fitted.

5.1.1 Pick-up baler

On this machine the pto shaft was connected to a boss which carried two hinged pawls of the overrun clutch. These pawls engaged with any two of the ten ratchet recesses on the flange of the centre of the torque limiting friction clutch, the rear output plate of which was attached to the flywheel. The flywheel was connected to the output shaft through a shear bolt. The organic-based friction material was attached to both faces of the steel centre plate of the torque limiting clutch. The front and rear steel plates were connected together by seven bolts while axial loading of the plates was provided by adjustable compression springs.

Torque patterns measured at the pto shaft appeared to fit into two categories: those showing two large torque peaks per ram cycle (corresponding to medium baling rates) and those showing three large torque peaks (corresponding to baling rates greater than 9 t/h). In static slip tests, breakaway torque values in any one series were consistent only to within 17% of the nominal clutch setting of 540 Nm and maximum torques transmitted were sometimes 27% above this. The torque range through which clutch slip occurred during normal baling varied from 400 Nm to approximately 800 Nm ie from 74% to 148% of the nominal. Fig 5 shows the distribution of slipping torque values measured during normal baling. The amount of slip associated with these torques varied between 0.01 and 0.60 of a revolution. Clutch slip was more prevalent during the ram compression stroke. Despite the wide range of torque at which the clutch slipped, the performance of the machine did not appear to be adversely affected and torque limitation was probably adequate for this machine.

5.1.2 Rotary cultivator

On this machine the torque limiting friction clutch was fitted between the pto shaft and the gearbox input shaft. The clutch was a multi-plate unit with sintered metal friction plates, with axial loading provided by adjustable helical compression springs.

Test runs were made on both grassland and stubble with the clutch set according to the manufacturer's instruction book. The effect of the machine's engaging a substantial buried obstruction was

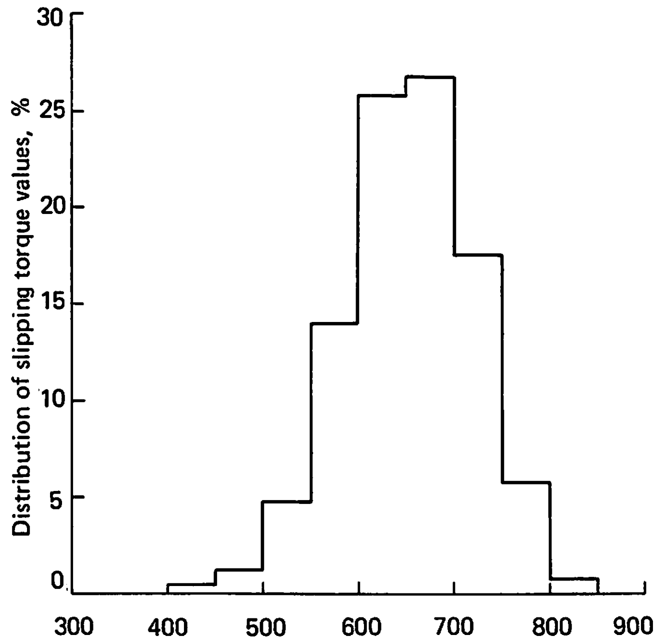


Fig 5. Distribution of the slipping torque values for a baler clutch during field tests; nominal setting 540 Nm.

also investigated but with the clutch set to a lower torque value. The pto shaft torque fluctuations, under most circumstances, were regular in nature and often contained high amplitude peaks of a predominant frequency corresponding to the fundamental natural frequency of the transmission. With the clutch set according to the manufacturer's handbook, no clutch slip was recorded during normal work although torque peaks as high as 1890 Nm were measured. Table 3 shows the maximum peak torques and mean torques measured during these field experiments. With the clutch set to slip statically at a nominal torque of 910 Nm some slip occured when the machine encountered the buried obstruction, but even so torques as high as 1510 Nm were recorded. The clutch fitted to this machine was clearly ineffective in preventing the occurrence of peak torque values much in excess of the mean values when set to the manufacturer's instructions but some degree of protection was obtained at a lower setting.

Table 3 Field test data for a rotary cultivator fitted with an overload clutch set according to the manufacturer's handbook

Crop	*Mean depth of cultivation* mm	*Mean forward speed* km/h	*Nominal rotor speed* rev/min	*Maximum torque peak during run* Nm	*Mean torque value during run* Nm	*Total clutch slip during run*
	75	2.3	122	1180	390	None
	75	3.3	122	1140	440	None
Grass	75*	2.3	153	1260	420	None
ley	50	3.3	153	1370	530	None
	90	2.1	172	1270	530	None
	65	3.3	172	1750	570	None
	90	2.1	216	1490	710	None
	90	3.3	216	1890	720	None
	90	2.3	122	880	420	None
	75	3.5	122	900	510	None
Barley	90	2.2	153	1270	580	None
stubble	90	3.4	153	1170	660	None
	90	2.2	172	1200	550	None
	100	3.4	172	1300	700	None
	100	2.1	216	1670	720	None
	90	2.9	216	1550	800	None

5.1.3 Drum mower

An axial pin type overrun clutch was fitted at the tractor end of the pto shaft and a friction plate slip clutch at the gearbox end. This clutch had two organic friction plates mating with steel plates and pressure was applied to the plates by means of central disc springs. The slip clutch had a nominal setting of 740 Nm.

Test runs were made at several forward speeds when cutting S23 rye grass and also some measurements were made when starting the machine at different tractor engine speeds using a tractor with a hydraulically engaged pto clutch. The torque fluctuations in the drive shaft were of a random nature although at the higher forward speeds there was a tendency for a low frequency oscillation of about 2 Hz to develop. Table 4 gives the maximum

Table 4 Field test data for a drum mower fitted with slip and overrun clutches

Forward speed km/h	*Maximum peak torque value during run* Nm	*Mean torque value* Nm	*Mean power* kW	*Clutch slip*
3.4	490	195	11.4	None
5.8	690	215	12.3	None
7.6	750	265	15.3	None
8.9	880	295	16.5	None

and mean torques recorded during the mowing runs, where no clutch slip was recorded. Table 5 gives the maximum torque and amount of clutch slip when starting-up the machine and shows that the clutch has effectively controlled the starting torque to an acceptable level. (See table 2 for comparison without slip clutch). However, little is yet known of the performance of this clutch over a period of time.

Table 5 Data for starting tests on a drum-mower fitted with slip and overrun clutches using tractor with hydraulically engaged pto clutch

Tractor engine speed rev/min	*Maximum torque* Nm	*Clutch slip* No of revs
1000	930	3.5
1200	900	7
1400	850	11.5
1600	900	16.5
1800	870	23.5
1900	840	26.5

5.2 Laboratory work

5.2.1 Clutch rig

A rig, fig 6, was developed to allow various tests to be made on overload protection devices in the torque range 0 — 1360 Nm. The rig was instrumented to measure torque, clutch slip and shaft speed and could be used for static or dynamic tests. For static tests, with one end of the shaft locked, the desired torque was applied at the other end by a linear electro-hydraulic actuator acting on a torque arm. For continuous rotation tests, the shaft was driven from the pto shaft of a tractor. A servo-controlled disc brake which had a frequency response of up to 32 Hz was used to allow synthesised or recorded transmission torque spectra to be applied to the test clutch.

Tests made on this rig have included the measurement of the performance of a range of proprietary clutches of both ratchet and friction plate types and also an investigation into the factors affecting the performance of friction plate clutches.

5.2.2 Ratchet type overload clutches

Several plunger ratchet and ball ratchet type overload clutches were tested to determine their performance. Tests were made at various speeds of rotation up to 720 rev/min and loads were applied to simulate a range of practical conditions. Clutch components were examined for wear both before and after the tests. The breakaway

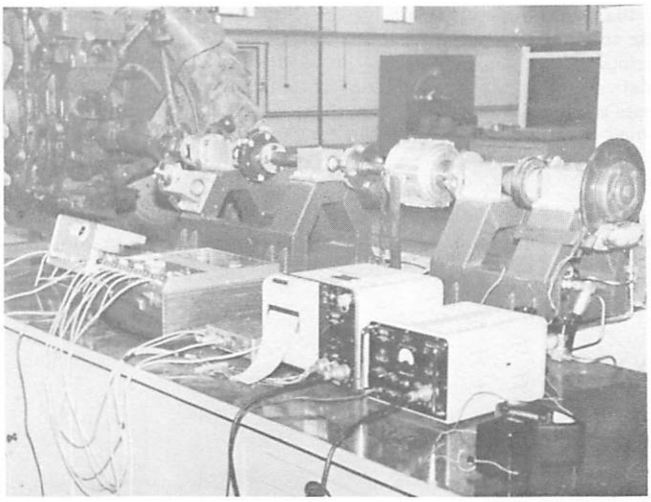
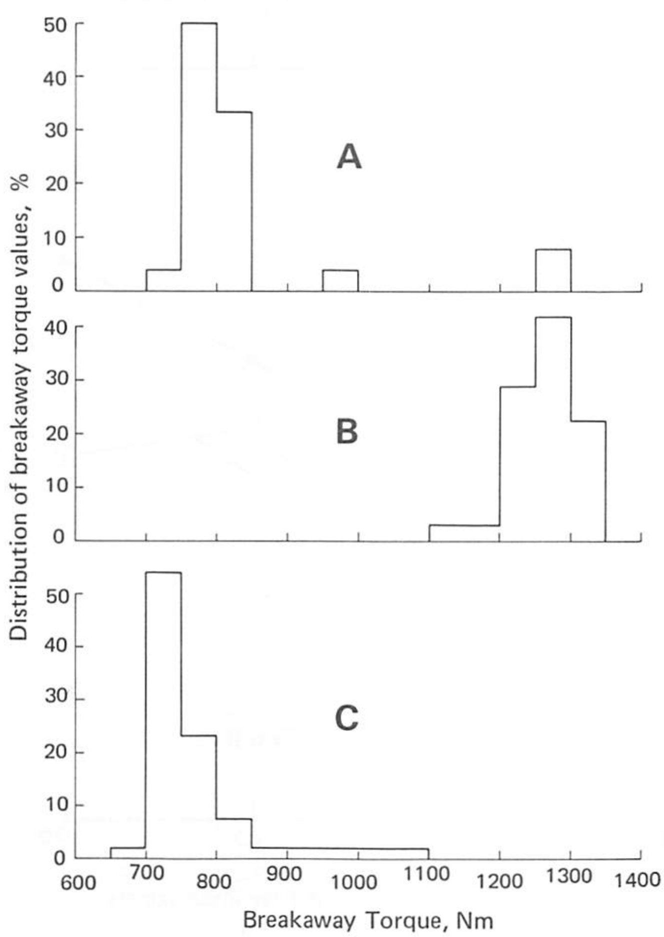


Fig 6. Laboratory test rig.

torque values recorded for each of the clutches was inconsistent and spread over a wide range. This clearly limits the use of these designs if close control of torque is required to protect a machine. The distribution of breakaway torque values for a plunger ratchet clutch is shown in fig 7 (A). The breakaway torque values were higher when load was suddenly applied than when a more gradual loading cycle was used. During tests when the clutches were subjected to random loadings, once slip had commenced it continued until the end of the run, despite the fact that the subsequent torque level was lower than that at which slip first occurred. In order to re-engage the drive it was necessary to stop the machine to allow the balls or plungers to locate their sockets. Low mean slipping

Fig 7. Distribution of values of breakaway torque for a plunger ratchet clutch (A), combined plunger ratchet and friction type (B) and friction plate clutch (C), when running at 540rev/min and subjected to suddenly applied overloads.



torque values make these designs of clutch unsuitable for removing overloads whilst continuing to drive the machine. For example, on soil working machines it is necessary to remove momentary overloads but provide a high mean torque to keep the blades rotating. Shaft speed had little effect on the breakaway torque values but lower mean slipping torques were measured at the higher shaft speeds. Whilst slipping, these clutches had fluctuating torque patterns caused by the plungers or balls moving over the ramps and sockets. The maximum peak torque values were as high as double the dynamic breakaway values, while negative troughs were frequently recorded. This highly fluctuating torque pattern may be an undesirable characteristic of these clutches when considering fatigue life of machine components. Static breakaway torque values were usually more consistent and lower in value than those measured dynamically.

Existing designs are far from ideal but they may provide some degree of overload protection when a complete blockage of a machine occurs, making it necessary to stop and clear the blockage before re-engaging the drive. The experiments were made with new clutches and therefore little is known of their performance over a period of time. The main advantage with these designs is that when slipping they provide an audible warning to the driver.

5.2.3 Combined plunger ratchet and friction clutch

The performance of a combined friction and ratchet type overload clutch was determined by subjecting it to a series of triangular and suddenly applied overloads designed to simulate practical conditions at shaft speeds of 540 and 720 rev/min. The breakaway torque values for the triangular ramp function inputs were more consistent than those for the suddenly applied loads and were also lower. The distribution of breakaway torque values for suddenly applied load is shown in fig 7 (B). Shaft speed had little effect on the breakaway torque values. Whilst slipping, the clutch had a fluctuating torque pattern with maximum and minimum torques of the order of 2100 Nm and −450 Nm respectively, and a mean value of approximately 630 Nm. The higher mean slipping torque characteristic caused by the increased load applied to the friction linings when the plungers are moving over the ramps (see fig 3) makes this design of clutch more suitable for moving peak torque overloads, while at the same time continuing to drive the machine. Overload protection when a complete blockage occurs is also provided by this design but the temperature of the components will rise more rapidly than for clutches discussed in 5.2.2 if it is allowed to slip continually for more than a few seconds. However, whilst slipping an audible warning will be provided which should enable the driver to stop the pto before any damage is done to the clutch components. Again, the highly fluctuating torque pattern produced when slipping may be undesirable when considering fatigue life of components.

During the last series of runs there was a gradual rise in the average value of breakaway torque. Static breakaway torque values measured at intervals throughout the tests were within $\pm$ 10% of the initial value except for those measured after the last series of runs which were extremely high (35% above nominal setting). On inspection, the plungers were found to be in need of lubrication. No provision was made for adding lubricant after assembly due to the difficulty of preventing it coming into contact with the friction linings.

The main disadvantage with the design of clutch tested was that the plungers were inadequately lubricated and that the friction material may adhere to the mating surface causing the clutch to become inoperative. As the lubrication problem was revealed during the short period of rig testing it can only be assumed that the performance of this design of clutch would deteriorate rapidly in use. It may be possible to obtain a more consistent and reliable performance by using friction linings which are designed to be lubricated.

5.2.4 Friction plate clutches

Tests were made on four torque limiting friction plate clutches. The tests were designed to simulate two types of overload which can occur: firstly a sudden jamming of the machine caused by a complete blockage and secondly an intermittent overload caused by heavy working conditions in a situation of fluctuating load. Table 6 gives the breakaway and maximum peak torque values for the four clutches when subjected to the suddenly applied overloads. The mean of the maximum torque values transmitted whilst slipping are 19, 9, 10 and 250% above the mean breakaway torques for clutches A, B, C and D respectively. This high increase for clutch D is partly due to inertia effects caused by its large size. The distribution of breakaway torque values for suddenly applied overloads for clutch A at another clutch setting is shown in fig 7 (C). When subjected to the random loading the clutches slipped at fairly consistent torque levels, but the torque continued to rise during slip. For those clutches which were adjustable, consistency of performance depended upon the extent to which even loading of the pressure springs was achieved. Clutches which used a central disc spring gave a better performance than those having several coil springs around the periphery. This is illustrated in table 6 by the relatively consistent performance of clutches B and C. However, these values are much higher than those quoted by the manufacturer which were 540 Nm and 195 Nm for clutches B and C respectively. (for table 6 see page 33).

Fig 8. Mean torque values for running-in check using cast iron and organic friction material mating plates: contact pressure 227 kN/m$_2$ (top) and 345 kN/m$_2$ (bottom). Plates taken apart for measurement between tests A and B.

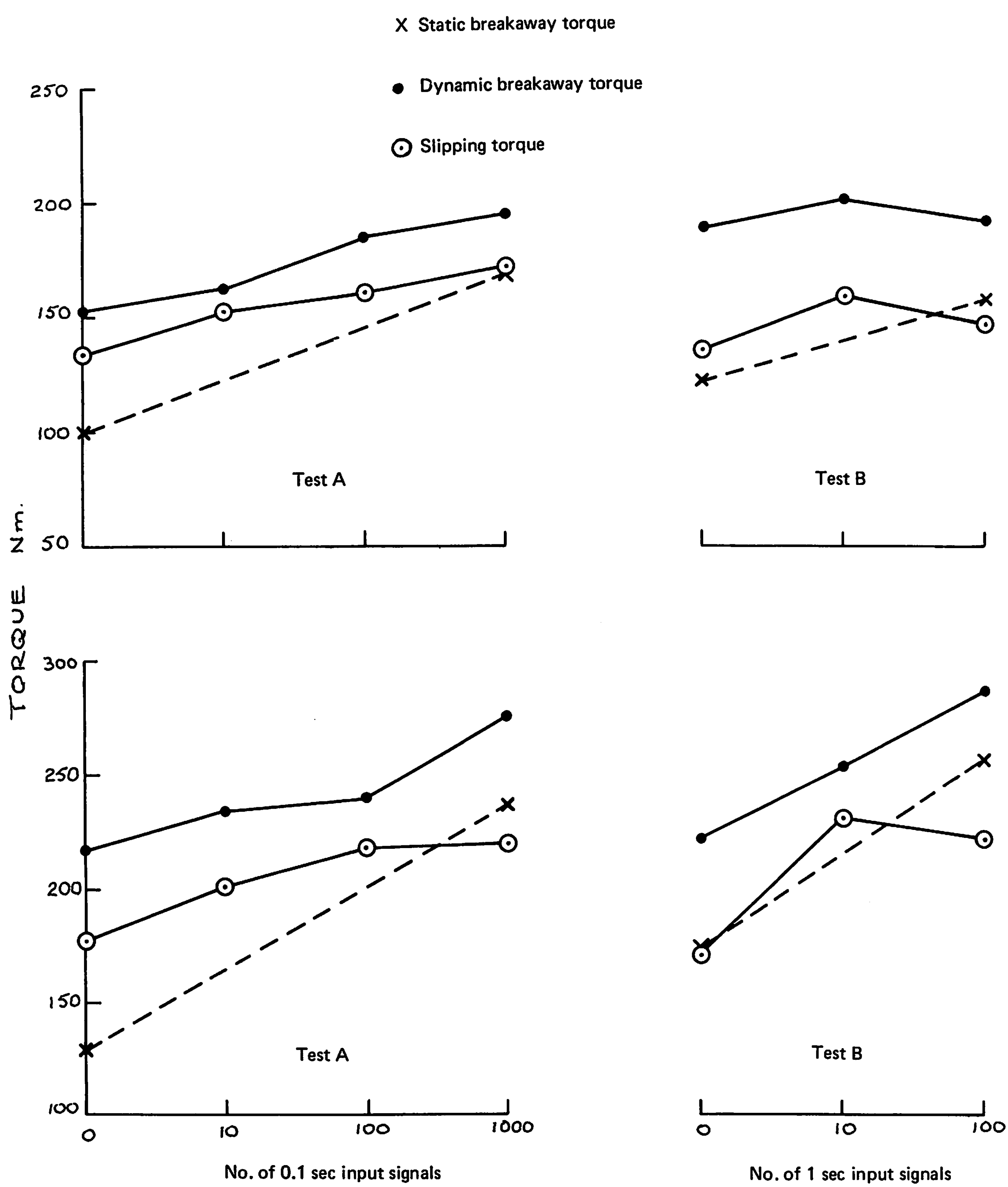


Table 6 Data for 4 friction plate clutches when subjected to suddenly applied overloads

Clutch		*A*	*B*	*C*	*D*
Breakaway	minimum	1090	700	290	420
torque	maximum	1200	750	320	460
Nm	mean	1155	725	310	430
Maximum	minimum	1320	770	310	920
torque transmitted during slip	maximum	1460	820	390	1440
Nm	mean	1375	790	340	1090

5.2.5 Factors affecting the performance of slip clutches

The effects of mating materials and surface finish on the performance of slip clutches were studied when using an experimental clutch[3]. Running-in checks were made with organic and sintered metal friction materials mating with cast iron, hardened steel and mild steel plates. The components were subjected to a standard series of tests and measurements were made, at regular intervals, of static and dynamic breakaway torque values and also of mean slipping torques. Each combination of materials was investigated at two contact pressures. Fig 8 shows the running-in effect for cast iron and organic mating material. The experiments showed that the breakaway and slipping torque values became more consistent after the running-in period but considerable fluctuations still occurred. It was therefore difficult to establish when the components had become run-in but the period was likely to vary for different material combinations and surface finishes. However, there appeared to be a greater tendency to stabilise at the lower contact pressure. In most cases a carbon type dust was present between the plates at the end of a test and this may have had some influence on clutch behaviour. With regard to wear, cast iron was shown to be a good mating material for both organic and sintered friction materials. The hardened steel and mild steel were satisfactory when used with organic material at the lower pressure but surface finish deteriorated at the higher pressure.

A limited experiment was also made to determine the effect of adhesion when materials are left clamped together for a period of storage, as in the case of seasonal machines. Two types of organic friction material were used, one had a stiction inhibitor incorporated during manufacture and the other one was untreated. Table 7 shows the effect after different periods of storage. The increased force required to slip the components after storage was due to corrosion and not fungal growth. This appears to be one of the greatest problems with organic friction materials making it necessary to remove the pressure and run the clutch before it can be reset and put to use after storage. It may be possible to overcome this problem and to improve the consistency of performance of friction plate clutches by using materials which can be immersed in oil.

Table 7 Breakaway force for adhesion test specimens

Mating materials	*Force required to cause slip, N*			
	At start of test	*After period of storage*		
		3months	*5months*	*7months*
Cast iron — Untreated friction material	56	—	—	370
Cast iron — Treated friction material	56	—	—	430
Mild steel — Untreated friction material	73	450	—	—
Mild steel — Treated friction material	73	650	—	—
Mild steel — Untreated friction material	73	—	560	
Mild steel — Treated friction material	82	—	710	
Mild steel — Untreated friction material	87			510
Mild steel — Treated friction material	73			770

5.2.6 Overrun clutches

Some designs of overrun clutches have recently been found to have high wear rates. This has mainly occurred on machines running at 1000 rev/min although some clutches running at 540 rev/min have also failed. The problem, in addition to the increased speed, seems to be due to the high inertia of machines and long heavy pto shafts which are supported at one end by the overrun clutch. Most overrun clutches in common use on agricultural machines have no internal bearings and therefore considerable lateral loads due to the heavy drive shafts are imposed. When the shaft and clutch are rotating, vibration and whirling effects can occur which increases wear on clutch components. Some wear could be reduced if bearings were fitted to this design of clutch.

6 Conclusions

From work to date it has been shown that the performance of torque limiting and overload clutches used on agricultural machines is usually inconsistent. Any performance figures specified by manufacturers usually refer to a clutch in new condition as tested in the factory and cannot be taken as indicative of performance under practical conditions. Where extreme overloads occur some protection is given to machines using clutches in new conditions but after use or periods of storage their performance deteriorates for various reasons including contamination, lack of lubrication and adhesion or corrosion.

The work described in this paper has discussed the need for reliable and efficient overload and overrun devices to protect pto driven machinery from overloads and has shown that most existing devices are inadequate. Work is to continue at NIAE on three aspects of overload protection for machinery;

(a) By making further measurements of machine torque patterns in order to understand more fully the types of overload likely to be encountered.

(b) By making a theoretical analysis of the dynamics of typical drivelines.

(c) By continuing to investigate the factors affecting the performance of overload and overrun clutches by laboratory and field tests.

When considering possible improvements in the design of overload protection devices for agricultural machines the cost of materials and manufacture must be borne in mind. There is a need for clutches with a greatly improved performance and durability but the cost must be kept as low as possible.

7 References

1 **Richardson R C D; Jones M P; Attwood D G.** A pilot survey of the durability of farm machinery. Paper No 2/F/26. Proceedings of Agricultural Engineering Symposium of the IAgrE 1967.

2 **Chestney A A W.** The measurement of starting torques on tractor pto driven machines with high inertias and the effect of using a torque limiter to control them. Nat Inst Agr Engng. DN/ER/592/1172 May 1975.

3 **Chestney A A W.** An investigation into various factors affecting the performance of slip clutches: I Experiments to determine the running-in and adhesion characteristics. Nat Inst Agri Engng. DN/ER/600/1172 May 1975.

Acknowledgements

The author is grateful to Mr C J Moss, Director, National Institute of Agricultural Engineering, for permission to publish this paper. Thanks are expressed to K Cochrane for his assistance with the experimental work.

IAgrE Newsletter No 2

ALL registered members of the Institution are advised that the Executive Committee decided, on 8 April 1976, that in order to comply with budgetry provisions for postage, the next issue of the Newsletter (No 2) can only take place in October 1976 (when sending out 1977 membership subscription renewal notices), unless there is an earlier general mailing.

It is intended that the Newsletter will *not* be mailed at the same time as the Journal.

Energy economics and the diesel engine

A K Haddock and W Tipler

Abstract

THE high speed diesel engine is recognised as the most economic source of motive power; it will continue to hold this position against competition by the gasoline engine, and other prime movers which are currently the subjects of prolonged development programmes.

Recent advances in the understanding of the combustion system coupled with optimisation of cooling system design offer reductions of fuel consumption of over 5%. Improvements of the order of 10% could be achieved by reducing engine design speed, although the additional bulk and cost of the engine might offset this saving. The development of new materials of construction, which would enable the engine to approach the adiabatic ideal, might permit a 15% improvement in fuel economy.

It is therefore evident that there is considerable potential for further development of the diesel engine.

Introduction

ENERGY economics has been defined as the most effective means of satisfying the energy needs of an application from available sources (Wilson and Tee, 1974). In this paper it is applied to the process of converting the chemical energy of naturally occurring fuels to useful shaft power, through the intermediate stages of fuel refining to produce commercially saleable fuel and then the combustion of that fuel in a diesel engine.

In order to keep the scope of the paper within manageable proportions, it is limited to diesel engines of the small (< 400 bhp) high speed (> 1800 rev/min) type widely used in automotive applications, both on and off highway. The scope of the paper ends at the engine output shaft, there being no discussion of the problems of converting engine shaft power to power at the wheels in any of the many applications to which high speed diesels are harnessed.

The reasons for the inherent high thermal efficiency of the diesel engine are discussed, means of enhancing its efficiency inspite of the handicaps placed upon it by environmental legislation are described, and some longer term developments are suggested.

Fuels and engines are complementary products and it is appropriate to discuss fuels since the objective of energy policies should be to attain the maximum overall efficiency of use of basic natural energy not merely of commercial fuels. Considerations of this type are necessary since it is now recognised that the world's resources of crude oil are likely to be exhausted in a few decades. (Table 1, Fells *et al*, 1973).

Diesel engine thermal efficiency and its improvement

It is generally accepted that the high speed diesel engine is the most efficient power unit available for automotive applications although the Stirling engine (Carlqvist *et al*, 1975; van Beukering, 1975) and the gas turbine (Philips, 1975) are not without their advocates. A well balanced overall survey is also available (Huebner, 1975).

It is also widely held that scope for improvement of the fuel economy of the high speed diesel engine is severely limited. Having reviewed the basic reasons for the economy of the diesel engine, this contention will be rebuffed. At the same time, it is necessary to retain the reputation of the diesel engine for reliability; as Carnot said in 1824: "Fuel economy is only one requirement of a combustion engine; in many cases it is secondary to reliability, durability ......".

The net (or brake thermal) efficiency of an engine can be expressed as the product of several contributory efficiency terms:—

$$E = E_{as}\, E_{di}\, E_c\, E_m$$

where E = brake thermal efficiency
E_{as} = air standard efficiency
E_{di} = diagram factor
E_c = combustion efficiency
E_m = mechanical efficiency

Air standard efficiency (E_{as}) is discussed in detail in many excellent text books (eg Lichty, 1951); diagram factor (E_{di}) is similarly well covered; nothing new can be said on these subjects except to comment that the advent of the computer has enabled the prediction of cycle efficiency to be carried out with some precision in spite of the complexities implicit in these quantities (eg Whitehouse and Abughres, 1975).

Such calculations made at the design point of a high speed diesel engine do not reveal a very marked advantage in thermal efficiency over its only established rival, the gasoline engine. The benefit of higher compression ratio is offset by a lower mechanical efficiency resulting from the higher pressure loadings. However if the fuel consumptions of the two power plants are compared under actual operating conditions then the diesel engine is seen to yield a marked advantage. Figure 1 shows results obtained in the USA for a range of vehicles powered by gasoline engines, and then for the same vehicles converted to diesel power. Various standard operating cycles current in the USA were used. It will be seen that the advantage of the diesel engine becomes progressively more marked as the duty cycle becomes lighter. This gain derives almost entirely from the differing combustion characteristics of the two power units.

The diesel engine runs unthrottled through its operating range, and at air/fuel ratios which rarely fall below 20:1. Consequently, the combustion efficiency (E_c) is very close to 100% under all conditions. While the high air/fuel ratio is beneficial in this respect it does impose a penalty upon the engine size for a given power output.

In contrast, the gasoline engine, operating on a more volatile fuel, uses a premixed combustion system. This enables operation very close to stoichiometric to be achieved without emission of visible smoke, but close control of air/fuel ratio is necessary to maintain a combustible mixture. Consequently, part load operation is obtained by throttling the air flow; and thereby reducing fuel flow through the carburettor in approximately the same proportion. The inlet depression imposed on the engine by the throttle, plus imperfect fuel vaporising and mixing cause serious deterioration in both E_{as} and E_c under part load conditions.

The combustion process

During the past five years the development of high speed diesel engines has been dominated by the necessity to satisfy widespread legislation concerning smoke, gaseous emissions and noise. While the ideals which have stimulated such restrictions must be supported, the fragmented nature of the legislation has caused unnecessary difficulties for the engine manufacturers, particularly those with world wide commitments. The lack of co-ordination between legislation in different territories implies that one engine type may require to be certified repeatedly if a world wide sales drive is to be sustained.

Control of gaseous emissions (particularly hydrocarbons and oxides of nitrogen) has necessitated detailed studies of the combustion process and the development of new combustion systems (eg Bertodo *et al*, 1975). It is now well known that the formation of oxides of nitrogen (NO_x) is strongly temperature dependent and that this formation is virtually negligible if the gas temperature is below about 1700K. This requirement can be satisfied by various means including water injection, exhaust gas recirculation and retarded fuel injection. It is now generally agreed that the last

A K Haddock and W Tipler are from the Research Department, Perkins Engines Company, Peterborough.

Paper presented to Western Branch of Institution of Agricultural Engineers, on 19 November 1975.

Table 1 World energy reserves: Variation in estimates (Fells et al, 1973)

		Present Known			Potential Future		
		Reserves: (10^9 tons oil equivalent)	Life (years at 1971 consumption rates)	Life (years at future consumption rates)	Reserves: (10^9 tons oil equivalent)	Life (years at 1971 consumption rates)	Life (years at future consumption rates)
Oil	Lowest	80	32	16	250	100	30
	Highest	90	36	18	360	140	40
Natural Gas	Lowest	28	33	15	80	90	25
	Highest	39	45	19	280	330	40
Shale/ Tar/Sand	Lowest	97	39	Add 9 to Oil	280	110	Add 10 to Oil
	Highest	120	48	Add 11 to Oil	500	200	Add 17 to Oil
Coal	Lowest	90	60	30	750	500	150
	Highest	1500	1000	190	3300	2200	250

method is most acceptable from aspects of first cost, operational simplicity and reliability; but since a higher proportion of heat release occurs during the expansion stroke there is a distinct loss of thermal efficiency, and increased exhaust smoke (fig 2).

This led to an intensive re-examination of the combustion process, the aim being to speed up the rate of combustion so that heat release was completed early in the expansion stroke while conditions were most favourable thus reducing smoke formation; similarly early completion of heat release was advantageous to thermal efficiency. At the same time it was essential to avoid the high pressure/temperature peak which characterises many combustion systems, since appreciable formation of NO_x can occur in this high temperature region.

These requirements were met by a re-entrant bowl combustion system, the geometry of which is compared with that of a conventional toroidal bowl in figure 3(a) and 3(b). Results obtained with this system are shown in fig 2, compared with those from the conventional system. It will be seen that at retarded injection timing (dynamic) the re-entrant system gives remarkable reductions in smoke level and specific fuel consumption, achieving levels comparable with the optima obtained with the toroidal system under much more advanced timing. For a given timing the re-entrant system gives higher NO_x levels than the conventional system, but for a given smoke level it shows a distinct advantage.

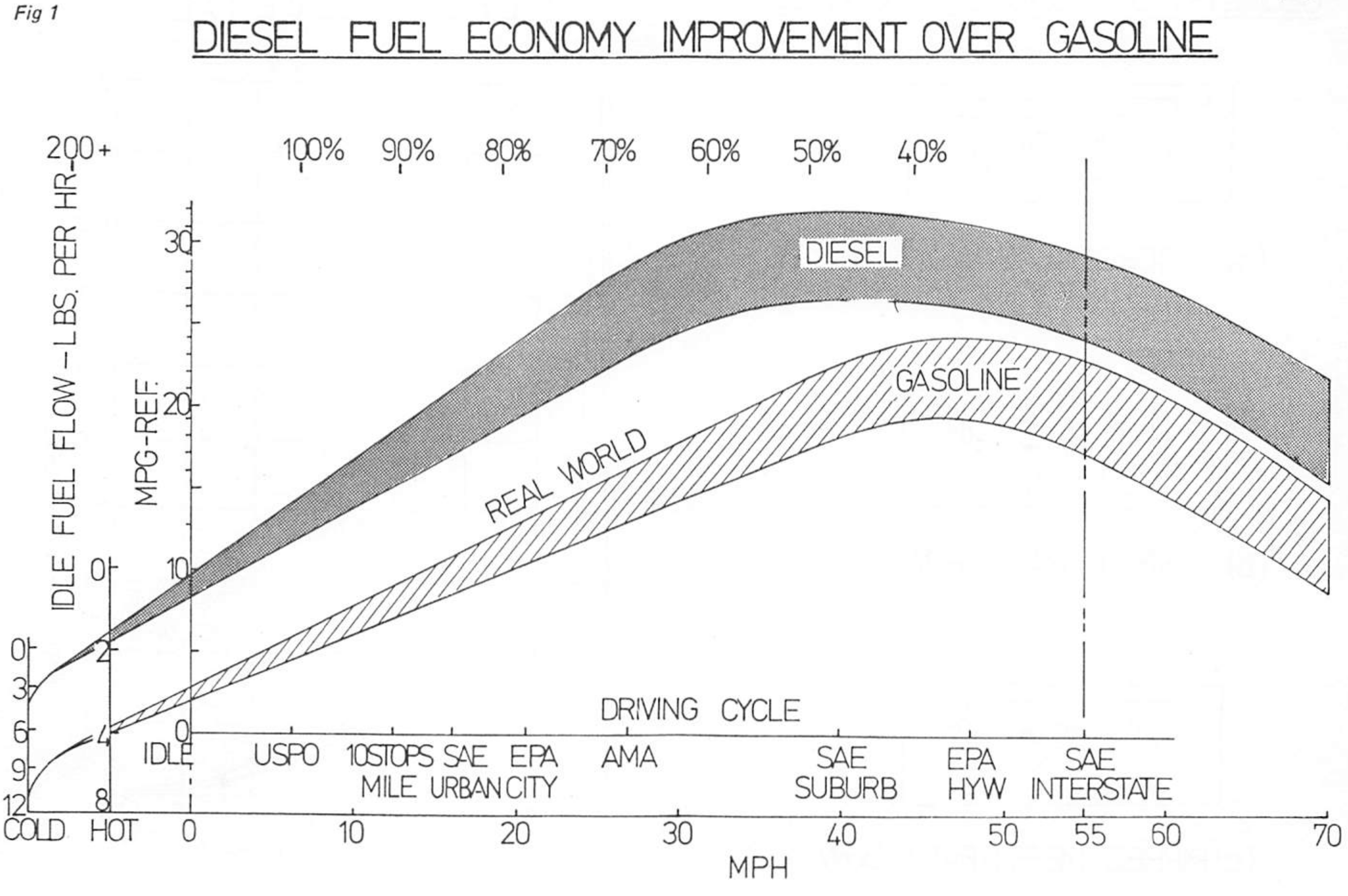
A comparison of heat release rates (fig 5) shows that the re-entrant system gives a high yet controlled rate of heat release, avoiding the high peak pressure which characterises most direct injection systems. This feature is advantageous also in the mechanical design of an engine in that it implies that component strengths (and therefore weights) can be relaxed with no loss of reliability.

Studies of the air flow patterns in a re-entrant system have shown that the air motion is most violent near the circumference of the chamber with a comparatively stagnant region near the axis. This led to the design of a combustion bowl with a large central "pip" (fig 3[c]), the objective being to eliminate the low velocity region and thereby increase the overall effectiveness of air-fuel mixing. Preliminary results (fig 4) show that while this system yielded no advantage in smoke level over the normal re-entrant configuration, it did give a significant improvement in specific fuel consumption. Again this advance was obtained at the cost of a small increase in NO_x production at a given fuel injection timing.

It is thus apparent that pressure of legislation against gaseous emissions has stimulated the development of a very flexible

→ page 36

Fig 1



COMPARISON OF TOROIDAL & RE-ENTRANT COMBUSTION CHAMBER

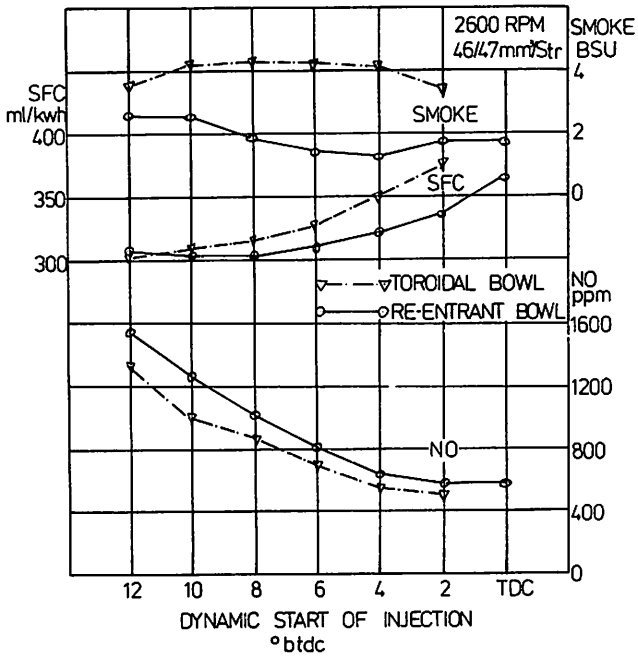
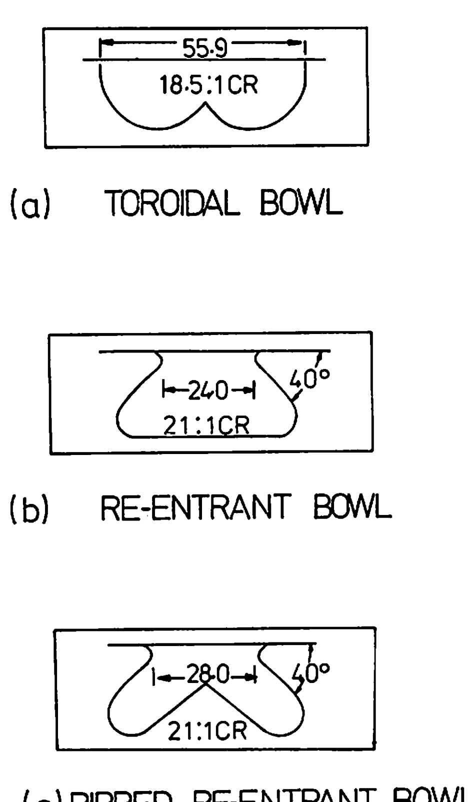


Fig 2.

combustion system. By appropriate selection of combustion bowl geometry and injection timing it is possible to obtain a wide range of combinations of fuel consumption and NO_x emission levels, to meet the requirements of different legislative situations.

Fig 3.

GEOMETRY OF COMBUSTION CHAMBER



COMPARISON OF TOROIDAL & RE-ENTRANT COMBUSTION CHAMBER

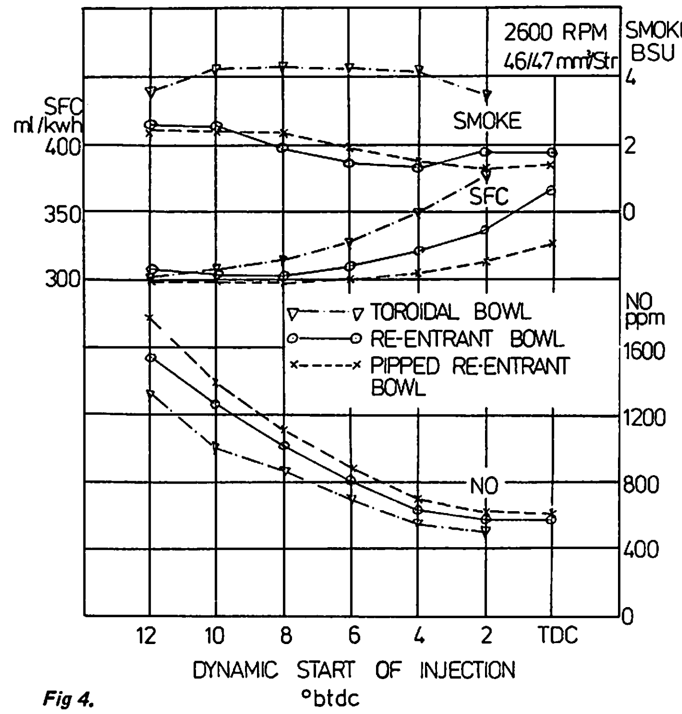


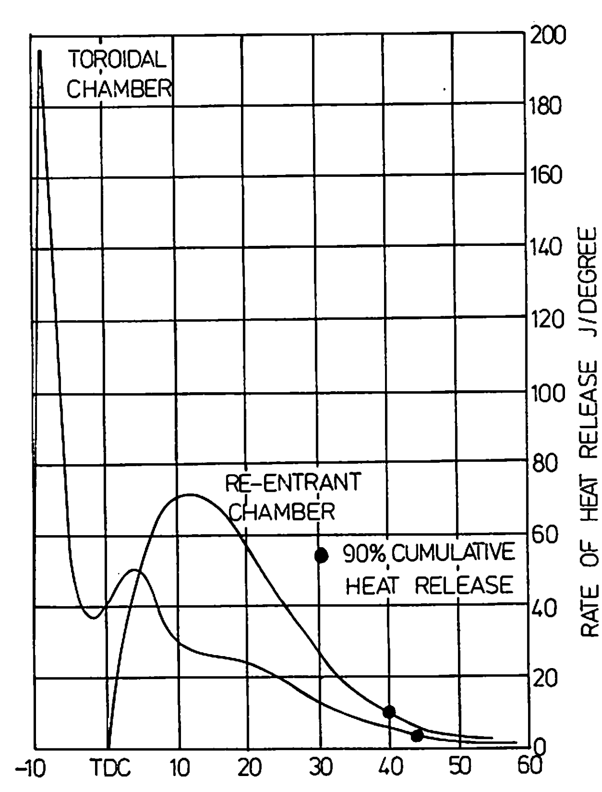
Fig 4.

Heat transfer

A significant contribution to diagram factor (E_{di}) comes from heat transfer between the air and gas and the surfaces of the engine.

Fig 5.

INFLUENCE OF COMBUSTION CHAMBER ON HEAT RELEASE RATE.



Heat transfer to the incoming air reduces the mass of air trapped in the cylinder and consequently limits the mass of fuel which can be burnt (and the power generated) without the emission of smoke. Heat transfer from the hot gases during the expansion stroke represent a loss of thermal energy. This heat, plus any extracted from the exhaust system where it passes through the water jacket, must be removed by the fan/radiator system thereby contributing a further loss by reducing the mechanical efficiency (E_m). Inspection of a typical heat balance (table 2) shows that approximately 30% of the heat input to the engine is lost to the coolant. Of this 30%, about 5% is accounted for by heat flow from the exhaust valves and ports to the cylinder head water jacket, but the remaining 25% represents heat losses from the cylinder. It would therefore appear that significant improvements in engine thermal efficiency might be realised if this 25% heat loss could be reduced.

Table 2 Heat balance for a typical D I engine

Speed (rev/min)	*2800*	*2000*	*1000*
Heat to power (%)	31	34	35
Heat to cooling water (%)	28	34	37
Heat to lubricating oil (%)	3	2	–
Heat to exhaust (%)	35	27	28
Heat unaccounted for (%)	3	3	–

Tests and calculations have been made on a large medium speed engine (Zapf, 1970) and further calculations have now been made on a high speed engine. The results of such a set of calculations made on a consistent set of assumptions are shown in table 3. Full load at

Table 3 Effect of cylinder insulation upon engine performance

	Full load at 2800 rev/min			*Full load at 1000 rev/min*		
	Standard engine	*Zero net heat loss*	*Adiabatic*	*Standard engine*	*Zero net heat loss*	*Adiabatic*
Trapped mass (g)	1.020	0.785	1.048	1.029	0.675	1.079
Air/fuel ratio	21.2	21.0	22.0	21.4	20.5	22.7
Power (kW)	62.8	47.8	74.1	22.4	15.4	30.6
Specific fuel consumption (ml/kWh)	300	310	254	294	302	220
BMEP (kN/m^2)	680	519	803	680	468	930
Max cylinder pressure (MN/m^2)	0.83	0.76	0.85	0.80	0.68	0.85
Exhaust temperature (K)	898	1133	1072	743	976	921

two engine speeds have been considered. In each case the first column gives results for a standard engine. The second column shows the corresponding results on the assumption that the piston, liner and cylinder head are insulated so that there is no heat loss to the cooling water. It will be observed that the trapped mass in the cylinder has fallen by over 20% as a consequence of heat transfer from the hot walls to the incoming air charge. This implies that the fuel flow must be reduced by approximately the same proportion to avoid exhaust smoke. Consequently the engine output is also reduced by about 20%. There is also a slight increase in specific fuel consumption since the frictional losses of the engine are unchanged.

As already mentioned these unattractive results derive from the assumption of a finite thermal conductivity for the surfaces containing the air charge. This implies absorption of heat from the hot gases and return of that heat to next incoming charge ie no net exchange of heat.

If the use of wall materials of zero thermal conductivity is postulated, then the engine becomes adiabatic ie no exchange of heat between the walls and either the air or the gas individually. Repeating the calculations, with the same assumptions concerning friction, the results of columns 3 and 6 of table 3 are obtained. In contrast to those of columns 2 and 5, a marked performance advantage over the standard engine is now predicted. The most important gains are a power increase of over 15% and a reduction of 15% in specific fuel consumption.

While this case is beyond practical realisation, it does show that the concept of a zero heat loss engine is not without promise, provided that materials of construction of sufficiently low thermal conductivity can be evolved, to offset the handicap imposed by the heating of the air charge. The complex analysis necessary to define the properties required of these materials is now in progress and will be reported on completion. The objective is to stimulate other research bodies and materials producers to take up the challenge of evolving a suitable material.

It is already clear that current test programmes elsewhere using high temperature materials such as silicon nitride have little to offer, apart from the novelty of using a new material.

A secondary result of reducing heat loss from the working fluid is that the engine cooling system can be reduced if not eliminated. This would simplify engine design (ignoring the complexity of incorporating components in refractory materials and of lubricating their rubbing surfaces!). Furthermore the power absorbed in the water pump and cooling fan would be reduced; this saving would be up to 7% of rated power on typical installations. On a more practical plan an alternative method of reducing power absorption by cooling fans is discussed later; this involves no handicap to engine reliability.

Mechanical efficiency (E_m)

The mechanical losses of an engine are made up of four main parts. These are the frictional power absorbed by the bearings and rubbing surfaces (piston ring/liner interfaces); the pumping power needed to draw air into the cylinders and expel exhaust gas from them against the resistances imposed by the valves and the inlet and exhaust systems; the power absorbed by auxilliary components essential to the operation of the engine (fuel pump, lubrication pump, water pump, radiator fan); the power absorbed by those auxilliary components needed to suit the engine to the requirements of its application (eg alternator, steering pump, exhauster). The total of these powers expressed as a fraction of the indicated horsepower (E_{as} E_{di}) represents ($1 - E_m$).

The power demands of applicational auxiliaries is not always included in the overall mechanical efficiency, but, from the point of view of the end user, it is considered more realistic to include them.

Fig 6 shows the frictional mean effective pressure (FMEP) of a typical direct injection diesel engine as a function of the engine rotational speed. As shown in table 4, at its design speed (2800 rev/min) this engine has a rated brake mean effective pressure of 680 kN/m^2 (98 lbf/in^2) and from fig 6 the FMEP is 280 kN/m^2 (41 lbf/in^2). Thus the mechanical efficiency (E_m) is 71%.

Table 4 Influence of engine rated speed on rated output, and brake thermal efficiency

	Engine speed (rev/min)	
	2000	*2800*
BMEP (kN/m^2) to fixed exhaust smoke level	790	680
FMEP (kN/m^2) by motoring test	190	280
IMEP (kN/m^2)	980	960
Mechanical efficiency E_m(%)	81	71
Specific fuel consumption (ml/kWh)	275	300
Brake thermal efficiency (%)	37.1	34.0

At 2000 rev/min, the BMEP is 790 kN/m^2 (115 lbf/in^2) for the same exhaust smoke level and the FMEP is 190 kN/m^2 (28 lbf/in^2). Thus the mechanical efficiency is 81%.

The corresponding specific fuel consumptions are 300 and 275 ml/kWh. Thus despeeding an engine from 2800 rev/min to 2000 rev/min would offer an improvement in fuel consumption of

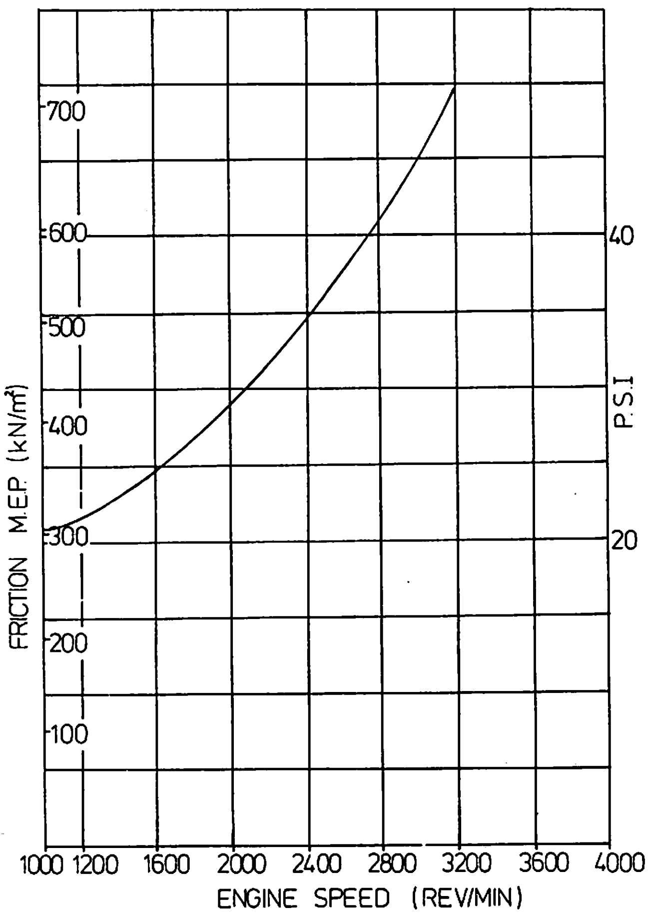


Fig 6

almost 10%. At the same time the available power is reduced by 17%. In some automotive applications this loss of power would result in only a small increase in journey time with the correct choice of gear ratios, but the driver would be involved in many more gear changes.

Nevertheless, this simple analysis does show that appreciable fuel savings are possible by using reduced engine design speeds. In order to regain the power of the original 2800 rev/min engine, the bore and stroke of the lower speed engine must be increased by about 6%, or the stroke by 20% with the original bore, or the bore by 10% with the original stroke.

To the first order of approximation each of these "new" engines would cost about 20% more than the original, it being assumed that cost is proportional to swept volume. Whether this increase, plus the overall increase in vehicle weight resulting from a 20% larger engine (in weight and volume) would represent an economic advantage to the operator when weighed against the potentially improved fuel consumption, would require detailed analysis of individual applications. Past analyses of this type showed the desirability of moving towards the higher speed engines in current use, but further increases in fuel prices might indicate a reversal of this trend to be desirable.

Accepting current engine rated speeds, many detailed studies are in progress aimed at improving fuel economy by the reduction of engine losses.

As has already been mentioned the re-entrant bowl combustion system results in the elimination of the severe high cylinder pressure peak which typifies many combustion systems. Representative figures are 10 MN/m^2 and 8 MN/m^2 respectively; the maximum rate of pressure rise is also reduced with the re-entrant system.

These lower loadings upon the power train imply that the design of pistons, gudgeon pins, connecting rods and crankshafts, together with the associated bearings and piston rings, can all be re-assessed. It is estimated that this would result in a 13% reduction of FMEP, and therefore a 3% increase in available power.

Detailed studies are also being made of the power consumed by the various auxilliary components, since their total offtake can be a significant fraction of the output of the engine, table 5. Those classified as engine auxilliaries impose a reasonable constant drain upon the engine; the demands of the applicational components are lower but with short peaks (particularly the steering pump).

Table 5 Typical power consumptions of various auxilliaries at engine rated speed, as percentage of engine rated power

Engine auxiliaries			
Lubrication pump		2	
Water pump		2	
Radiator fan		5	
	Total	9	
Applicational auxiliaries		*On load*	*Off load*
Alternator		2.5	0.2
Compressor		4.0	1.5
Exhauster		1.5	1.5
Steering pump		12.0	1.5
	Total	20.0	4.7

Noteworthy progress has been made in reducing the power absorbed by the cooling fan through a study to match the fan and radiator together thus obtaining optimised overall operation (Hebard and Tebby, 1974). In many typical installations the use of this procedure has resulted in a 50% reduction of fan power absorption, and thus a 2,5% increase in engine output. The same process yields significant reductions in noise emission since there is a close relationship between fan power absorption and noise emission.

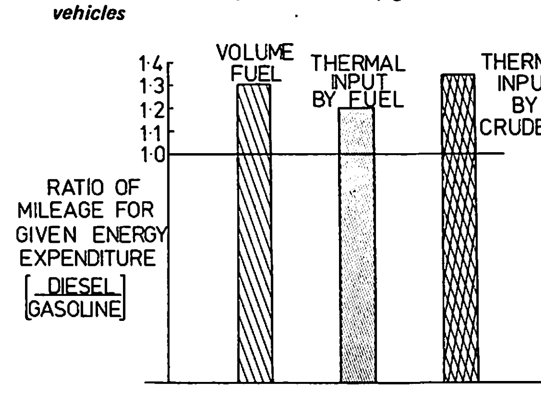
Overall energy economy

In the preceding sections various means whereby the thermal efficiency of a diesel engine can be improved have been discussed. In the eyes of the power plant user this may be sufficient, since it offers him the chance of reduced operating costs, provided that his reduced outlay on fuel is not offset by increased first cost and maintenance charges.

But, with the present realisation that our reserves of fossil fuels are not inexhaustible (table 1) the above attitude is no longer acceptable. An appreciation of the total energy consumption in shaft power generation is far more appropriate than the direct fuel consumption of the engine. The total energy concept has been applied to passenger cars (Hirst and Herendeen, 1973) analysing the energy consumed in every aspect of car manufacture, operation, maintenance and final demolition. A similar analysis could be applied to each aspect of diesel engine application, but within the scope of this paper it is only appropriate to consider the system beginning with crude supplied to the refinery and ending at the engine output shaft.

It is widely accepted that the diesel engine can offer over-the-road improvements in volumetric fuel consumption of 30% (Barnes-Moss and Scott, 1975) although as shown in fig 1 this advantage may be considerably higher. But the relative density of a typical gasoline is 0,74 against 0,83 for diesel fuel; thus on a mass basis the diesel engine offers an advantage of only 16%. On the other hand the lower calorific value of gasoline is about 45,0 MJ/kg, as opposed to 43,4 MJ/kg for diesel fuel; therefore on the basis of energy input the advantage of the diesel engine is 20% as shown in fig 7.

Fig 7 Comparison of energy utilisation by gasoline and diesel vehicles



But to produce one unit of heat as gasoline requires an input to the refinery of 1,21 units as crude oil (Hirst and Herendeen, 1973), while the corresponding ratio for diesel fuel is 1,07. Therefore on the basis of energy input by crude oil the advantage of the diesel engine is 35%.

Considerations of this type, coupled with the reinstatement of coal in certain applications lost in the recent unbridled competition between oil and coal can extend the "life" of the world's reserves of crude oil for about a decade beyond the estimates given in table 1. Further supplies of fuels for automotive applications can be derived from coal using known technology, although this subject requires in-depth study if optimum energy economy is to be achieved (Tipler, 1975).

Conclusions

From the work which has been briefly reviewed in this paper, it is evident that the following improvements in diesel engine fuel economy are attainable on the basis of available knowledge:—

Re-optimisation of combustion system under low emissions operating conditions	3–5%
Redesign of power train system	3%
Re-optimisation of engine cooling system	2,5%
Redesign of engine for lower speed operation (ignoring cost considerations)	10%

In the longer term, development of new materials of construction and of improved lubricants may enable an engine approaching the adiabatic ideal to be contemplated, with a potential gain of fuel economy of 15%.

Advances of these magnitudes will ensure that the diesel engine maintains its pre-eminent position as the most economic source of motive power.

References

1 **Barnes-Moss, H W** and **Scott, W M**: "The High Speed Diesel Engine for Passenger Cars" — IMechE Conference "Power Plants and Future Fuels", London, 1975, p75.

2 **Bertodo, R** *et al:* "A Method of Direct Injection Diesel Emission Control" — IMechE Conference "Combustion in Engines", Cranfield, 1975, p49.

3 **Cralqvist, S G** *et al:* "Stirling Engines: Their potential use in commercial vehicles and their impact on fuel utilisation" — IMechE Conference "Power Plants and Future Fuels", London, 1975, p35.

4 **Fells, I** *et al:* "Energy for the Future" — Institute of Fuel, 1973.

5 **Hebard, P J** and **Tebby, S W**: "The Design and Optimisation of Engine Cooling Systems for Low Noise and Power Consumption" — FISITA, Paris, May 1974.

6 **Hirst, E** and **Herendeen, R A**: "Total Energy Demand for Automobiles" — SAE paper 730802.

7 **Huebner, G J**: "Alternative Power Plants" — SAE President's Address to Automobile Division of IMechE Coventry, June 1975.

8 **Lichty, L C**: "Internal Combustion Engines" — McGraw Hill, New York, 1951.

9 **Phillips, P A**: "Prospects for the Gas Turbine Car" — IMechE Conference "Power Plants and Future Fuels", London, 1971, p25.

10 **Tipler, W**: "Energy Economics of Automotive Power Generation" — SAE paper 750761.

11 **van Beukering, H C J**: "Stirling Engines for private cars: modern developments and potential future applications" — IMechE Conference "Power Plants and Future Fuels", London, 1975, p159.

12 **Whitehouse, N D**, and **Abughres, S M**: "Calculation of Fuel Air Mixing in a Diesel Engine with Swirl for the Purpose of Heat Release Prediction" — IMechE Conference "Combustion in Engines", Cranfield, 1975, p147.

13 **Wilson, S S**, and **Tee N D C**: "Energy Economics and Transport" — Urban Transport Studies Group, University of Warwick, 28 March 1974.

14 **Zapf, H**: *"Einfluss der Kuhlmittel — und Zylinderraumoberflachentemperatur auf die Leistung und den Wirkungs — grad von Dieselmotoren"* — MTZ, 1970 (December), 31, pp499-505. (The influence of coolant and cylinder surface temperature on the performance and efficiency of diesel engines).

Around the branches

Northern Ireland

DURING February 1975 a meeting of Institution members was held and it was agreed that further meetings ought to be arranged. Since then further meetings have been held and judging by enquiries and attendance a viable group is emerging.

Last May Mr E Sudron of Shell Marketing Ltd gave an interesting talk about recent changes in the oil production industry. Having summarised the statistics of demand for fuel he then outlined the history of North Sea oil.

"Oil exploration started in the early 1960's after continental shelf laws had been drawn up, deep sea drilling technology developed and after continued drilling for gas had located oil. There were many unsuccessful bore holes and when combined with drilling time lost due to bad weather and the need to take on supplies onto the rigs the costs are high. However it is hoped to save £4000m on the UK balance of payments and make money by exporting refined oils. Nevertheless North Sea oil is only 2% of world reserves and reproduction from oil wells normally slows down after seven years".

In January 1976, Dr P McNulty of University College, Dublin spoke about grass cutting machines.

Considerable differences in energy requirement had been reported for various types of moving machines and the effect of these machines on drying rate was indicated. Recognising these differences in power requirement, it had been decided to investigate the energy required to cut grass at various blade speeds and it was found that most energy was consumed in accelerating the stem at cutting.

It was stated that most drum mowers are driven through a system of bevel gears, consequently they were liable to damage by impact of the blades or rotors on the ground at high forward ground speeds. There was, therefore, a need for considerable strength in these gears and some manufacturers had opted for belt drives. To minimise such stresses, drum mowers might be supported underneath the rotors, or the mower could be placed where it is more easily seen by the operator.

Field trials had been made to investigate the effect of various parameters on drum mowers and detailed results were to be available soon. The factors investigated were crop type, density and height, mower rotor speed, inclination and inertia, land gradient, tractor speed and blade type.

It had been noted that whilst the minimum pto power recommended was 22 kW, the average power measured at the cutting rotor shafts was 7.7 kW. Cutting power was taken as the product of cutting energy, tractor speed and mower width and average values of cutting energy were recorded as 40 J/m^2 for ideal low speed double shear cutting and 846 J/m^2 for impact mowers.

Attention was then drawn to the transmission arrangement in disc mowers and the possible damage to the spur gear system by impact loads on the teeth and the deflection of the cutter bar along its length. One manufacturer tackled these problems by off setting the discs so that each had an independent drive. Another halved the diameter and encountered the problems of high speed movement. Subsequent doubling of the disc diameter still did not resolve the problem. O'Keefe at UCD developed a driving system incorporating a long drive shaft and individual floating gearboxes and this mechanism was proving successful.

Some aspects of combined mowing and conditioning machines were then outlined. This included the effect of various types of machine on drying rate, and the relationship of pto power to forward speed.

In conclusion it was anticipated that there would now be a period of consolidation in mower design and that this would involve a greater use of precision engineering to accommodate the high speeds involved.

Scottish

THE 1976 Scottish Branch Conference, held on 18 February 1976, and entitled "Tractors — have you a choice?" was attended by 42 members and 47 visitors. In the morning Dr B D Witney, Head of the Agricultural Engineering Department at the East of Scotland College of Agriculture, spoke on application of power to the land and Dr A R Reece of the Department of Agricultural Engineering at the University of Newcastle-upon-Tyne discussed tractor transmissions and traction. In the afternoon Mr A Blyth, Head of the Farm Management Unit in the East of Scotland College of Agriculture, spoke on tractor replacement policy and, finally, Mr R McD Graham, Massey Ferguson Ltd, spoke on the selection of mechanisation systems.

Dr Witney said that the average tractor power rating on farms had increased from 28 kW (38 hp) in 1965 to 41 kW (55 hp) in 1973 but that the number of registered wheeled tractors had fallen from 500 000 in 1967 to about 400 000 today. Though machinery prices had doubled in the past five years, he argued that price should be the final rather than initial factor to be taken into account. The potential user should first weigh up his requirements in terms of tractor power output, gear ratios, size of tractor tyres, driver comfort and safety, and tractor operating costs.

Tractors should be compared by their pto power output when running at the British standard speed, for this figure was the most consistent for various test procedures and the requirement for particular machines was often well defined. Efficient utilisation of drawbar power depended on three factors: transmission, traction and the operator. A transmission with an unsatisfactory distribution of gear ratios led to much work being done at speeds, and in circumstances, under which the tractor engine could not develop its full power. Some manufacturers attempted to fill gaps between gear ratios by means of a high/low "on the move" gear change but some of these were unsafe because they did not provide engine braking when in the lower ratio. Traction studies suggested that some 2 wheel drive tractors were fitted with tyres too small to transmit the power available; rate of work was increased by 10-20% when larger or dual fully ballasted tyres were fitted and an implement of the width to generate 10% wheelslip at full power was used. Operators were unwilling to tolerate the noise which accompanied work at full power nor would they suffer the jolting which went with doing light work at high speeds. The Q cabs were an improvement but 90 dB(A) was far from quiet. Rubber mounts might reduce noise transmission to the cab but it was doubtful if the ride was much improved. It was also pointed out that the use of larger trailers at higher operating speeds was dangerous unless effective trailer braking systems were fitted and used; these were available at about £150.

Dr Witney concluded that though study of operating costs suggested that the least cost tractor size was 60 kW (80 hp) there was a wide range of solutions for various crop enterprises provided equipment was properly matched to crop area and tractor size. Power rating apart, tractors were designed primarily to develop either high pto power or high drawbar power. Farmers and their tractormen should study the operational characteristics of various makes, especially power/weight ratio, gear ratio spread, tractive performance and operator layout.

Dr Reece restricted his remarks to wheeled tractors for tillage. He pointed out that tractors of a particular type obtained from any manufacturer cost almost the same per pound weight. However, 4-wheel drive tractors cost about 20% more per lb than 2-wheel drive, presumably because more of the tractor consisted of high cost components required for the additional transmission. Since maximum drawbar pull of either kind of tractor was closely related to tractor weight it followed that, having picked a tractor type, one paid a fixed price per unit of drawbar pull.

However, farmers bought tractors for drawbar horsepower rather than drawbar pull and it was apparent that a light, powerful tractor could only develop its full power at high land speed. He found that 6.5 km/h (4 miles/h) was the maximum controllable tillage speed and, with that limitation, he calculated that a 2-wheel drive tractor should weigh 38 kg per drawbar kW (110 lb per drawbar hp) while the 4-wheel drive tractor need only weigh 24 kg per drawbar kW (70 lb per drawbar hp).

Dr Reece then considered current tractors and took light and heavy 2-wheel drive tractors and a light 4-wheel drive tractor as examples. He demonstrated that the light 2-wheel drive tractor could not develop its maximum usable drawbar power, even when fully ballasted, at speeds of 6.5 km/h (4 miles/h) or less. In contrast, the heavy 2-wheel drive tractor already had a sufficient axle load to suit it for tillage. The 4-wheel drive tractor exceeded the necessary axle loading which he had calculated. Thus it followed that only the heavy 2-wheel drive tractor was really suited to tillage; the others were justified only in special circumstances eg much roadwork or pto work for the light 2-wheel drive tractor and much hill work for the 4-wheel drive tractor.

Mr Blyth began by illustrating how the wide range of leasing charges made it impossible to give general advice on whether to lease or buy. In addition there was no general rule about useful tractor life because there was insufficient information on the relationship between the repair costs and life. Decisions based on these factors must necessarily be subjective. However, some factors could be dealt with more rationally. For example, over the last decade inflation had increased the purchase price of tractors by about 250% and spare parts by a similar amount. (In passing he pointed out that combine pulleys had gone up by 621%!) Over the same period the price at which the farmer sold wheat had gone up 144%, milk 131% and beef 180%. Clearly in times of inflation it paid to replace machines early. Another factor was tax liability. The farmer, who had a constant income, paid less than his neighbour who, while on the same average income, suffered annual fluctuations. Thus, in unusually profitable years it was advisable to reduce tax liability by replacing machines early, particularly as the entire purchase cost could be written off in the first year.

He summarised that tractors should be replaced early when they were used extensively, when breakdowns were frequent, when depreciation on repair costs were likely to rise, when the tractor could not cope with the amount of work, and when an improved tractor could save considerable amounts of money.

Mr Graham made use of service data which he had collected himself. He discussed the advantages and disadvantages of working to a system and the need for flexibility. He emphasised that a mechanisation system required machines and labour to be complementary at appropriate levels, which tended to increase the productivity of labour. However, the pattern of a system was often set by circumstances beyond the farmer's control. He illustrated this by displaying a scatter diagram of horsepower per acre plotted against a size of farm. Though the scatter was extreme it became apparent that there was a pattern once similar enterprises in a particular district were picked out, for these were closely grouped.

South East Midlands

G NEWMAN, farm manager of the Institute for Research on Animal Diseases, Compton, addressed a joint meeting of the S E Midlands Branch and the Bedfordshire Discussion Group of The East of England Agricultural Society, at Shuttleworth College on 17 February.

In his opening remarks on "Mechanised Livestock Feeding in Practice" Mr Newman suggested that the ability of the British livestock farmer to produce large quantities of milk and meat efficiently from grass is hardly ever repeated after the crop has been conserved. The recently introduced energy feeding standards, however, measured in terms of metabolisible energy, will help reduce this problem since one can now relate the energy requirements of an animal to its potential dry matter intake and the feed available.

In the light of these observations, Mr Newman reviewed the considerations which had led to the present forage plan at Compton. This involves the production of dried lucerne, lucerne silage, maize silage and ground ear maize. Young stock are grazed in summer and fed dried lucerne and silage in winter; experimental animals are fed conserved forage all the year round; dairy cows are grazed in summer with supplementary zero-grazed maize and they receive maize silage, dried lucerne and ground ear maize in winter.

Provided that the forage is of high quality, chopped and mixed (to prevent selection by the animal) to an accuracy of $\pm$ 5% for complete ration feeding, high yielding cows may receive up to 30% forage in their ration and lower yielders up to 75%. These cows would be carefully rationed each day but with 24 hours access to the feed.

Problems which arise out of this system and which call for the attention of agricultural engineers are

i) the need for accurate weigher mixers
ii) the need for automatic animal-weighing facilities so that feeding regimes can be matched to trends in the weight of cow groups (rather than individual cows) during lactation or the dry period.
iii) the need for a simple automatic system for measuring the milk yield of each group of cows so that any variation in group production is not masked by consideration only of the overall herd output
iv) the need for more co-operative driers. Dehydrated forage is not an alterantive but rather a complement to hay and silage.

Mr Newman also forecast that the small farmer in future will be prepared to spend more on feeding equipment in order to use forage more effectively. He will therefore require easily operated weighing equipment and, as his herd increases in size to more than about 75 cows, he will begin looking for complete diet systems with small feeder wagons.

West Midlands

"THERE could be no accurate research without instrumentation. This is never cheap and never less than 10% of the cost of any total project", W R Wignall of the Instrumentation Services Department of the National Institute of Agricultural Engineering told his audience from west Midlands Branch at Stareton on 26 January 1976. His job was to provide a service to research engineers by designing and fitting instruments on machines and projects under investigation.

Instrumentation should be planned right at the start of any development, and the form of analysis decided before commencement. Five factors are vital with instrumentation:—

1) the equipment must be accurate to an acceptable degree, which may be 1% or 10% (accuracy greater than required pushes up costs);
2) instruments must be resistant to damp, dust and vibration;
3) they must meet the specification laid down by the client;
4) delivery must be on schedule — particularly in agricultural seasonal work — which causes a piece of research to be delayed a year if missed by a few days;
5) cost.

Information can be collected and conveyed by various means, such as wires and cables, telemetry (transmission of signals by radio), hydraulics and optics. Often an unsuitable or weak signal must be greatly amplified or modified and then recorded by analogue or digital means.

Mr Wignall illustrated with colour slides some projects showing the uses of instrumentation such as monitoring the cowman's heart beats in a milking parlour to measure exertion levels, stresswork on rotary diggers and spray booms and environmental control in greenhouses by means of micro-processors.

His forecasts of instrumentation's future trends included smaller, neater instruments, better ergonomics, easier usage and more sophisticated solid-state circuitry.

●

ON 23 February at Stareton, G R A Miller of the European Tractor and Equipment Training Centre of Ford Motor Company spoke on "Controlling heavy implements in work". With the aid of excellent slides, films and animated 3-dimensional models, the meeting was given a review of tractors and trailed implements going back to 1917, and including the latest developments in tractor-mounted and semi-mounted equipment embracing top-link sensing, bottom-link sensing and load monitor (which is operated by torque forces acting on the transmission shafts).

Mr Miller said that top-link sensing was most effective for light and medium horse-power tractors, and bottom-link sensing for heavy tractors and implements with load monitor being a useful addition to both systems.

He gave much useful information on the geometry of cultivating implements — particularly in on-the-land ploughing with heavy wheeled tractors, where he advocated single-rear-wheels ballasted with water as cheap and effective.

Dual-rear-wheels had limited application (he said) except for jobs like heavy rotary cultivation.

Wrekin Branch

MATERIALS handling is now recognised as an important subject in agricultural engineering and its role is likely to increase in the future in step with farmers' requirements to improve productivity per man employed, reduce manual effort and to gain the advantages of timeliness.

An engineer concerned with materials handling in agriculture is faced with a complex package: the handling tasks are closely dictated by the agronomic demands of crops, by soil conditions and the weather; the materials and commodities being handled have a wide variety of physical characteristics, and each farm has individual materials-handling requirements.

Industrial work study techniques and materials-handling theory can be applied to farm operations and probably the most appropriate text-book rule is 'consider materials-handling as a link in an integrated system'. This is particularly relevant for farm transport at harvest time.

→ page 42

Farm transport

The art of managing a transport system is to balance the harvesting, transport and unloading activities so that no one team is waiting for the other. Improved harvester performance in recent years has created the need for larger trailers but there are limitations to trailer size due to such factors as safety and soil compaction. For safety reasons we recommend that trailers with greater than 5 tonnes capacity should be fitted with power brakes. Flotation tyres and tandem axles now enable heavier loads to be transported over wet land but there are still compaction problems and this aspect of materials-handling could well benefit from further fundamental research.

Palletisation

An alternative to trailers is to use unit loads, either as demountable bodies or palletised boxes. Fork-lift equipment has done much to reduce the manual lifting which was once accepted as normal practice on farms. For carrying out loading tasks a farmer can now choose between a tractor fore-loader, a tractor rear-mounted fork-lift, a rough terrain fork-truck, and an industrial loader. Should he use several simple tractors, each one fitted with a handling attachment, and allocated to one job only, or should he have a special tractor designed to accept several attachments each with a fast work rate?

Fertiliser is now palletised by manufacturers in 1½ tonne units although for movement around the farm 1 tonne is likely to provide a safer, more stable and manoeuvrable load. Development of the 750 kg polyproplene sack for handling granular materials is likely, and on larger farms the use of liquid fertiliser will continue to show advantages.

Sanderson SB55TC handling fertiliser bags.

Timeliness

Handling seed potatoes is a good example of a farm task which benefits from the timeliness of operation. A fast turn-round on the headland is essential to gain the advantages of high speed planters and there are developments in the palletisation of seed trays and bulk boxes designed for seed potatoes which can be handled by fork-lift equipment.

Crops susceptible to damage

Handling systems for crops like potatoes, fruit and leafy vegetables require special attention to keep damage levels to a minimum. There are several methods of filling and emptying bulk bins designed with this in mind and some use electronic sensing devices to control the emptying process.

Bales

Bale handling is one of the major problems still confronting farmers. Some 7 million tonnes of hay are made each year in this country and most of this is handled in the conventionally sized bale – a size convenient for lifting manually and for rationing. Much interest is being shown in handling hay and straw in large bales. In most parts of this country hay ought to be handled in such a way that air can be blown through the stack to complete its conditioning, although further developments in the application of acid at the time of baling, if successful, may allow more flexibility in the stage at which hay is baled.

There is likely to be an increase in the quantity of straw used for factory type processing. One of the disadvantages of straw is that it is a very light material and bales are not sufficiently dense for economic road or rail transport. Work is in progress to study the requirements of putting straw bales through a second compression cycle. The double handling would not be good materials-handling practice, and since this operation is likely to be a job for a contractor it might be more logical to sweep the straw to a stationary high density large baler employing a sweep rather like Hoiser used in the 1940's.

Summary of a paper presented to the Wrekin Branch on 3 November 1975, and to the Northern Branch on 9 December 1975 by D A Bull, a mechanisation adviser at the ADAS Liaison Unit, Silsoe.

Yorkshire

"PLEASE don't ask a 540 rev/min power take-off shaft to transmit more than about 60 hp. With a trailed machine in particular, when shaft angles can be high, the torque loadings are just too high at 540 rev/min. The one thousand rev/min is a must," pleaded Brian Spofforth, product specialist for Manns, the UK Claas importers.

Mr Spofforth, speaking at a Yorkshire Branch meeting of the Institution of Agricultural Engineers, at Wakefield, on 26 January, emphasised the importance of a steady and uninterrupted power supply to a precision-chop forage harvester, the machine upon which he based his paper. Inadequate tractor power or shear-bolts or slip-clutches in the harvester transmission were to be avoided because once the power flow even faltered, then the machine efficiency immediately dropped and it could even block solid.

In his broad based review of the precision-chop harvester, it was compared by Mr Spofforth with the cylinder lawn mower, another machine susceptible to damage but highly efficient and relatively low in its power requirements.

The importance of the correct design of the harvester pick-up was emphasised. The precision-chop harvester is primarily a weather-beating tool and, unlike the baler, it has to be able to pick up grass efficiently in any condition between the newly mown swathe and material which is nearly hay.

●

IN his opening remarks H G Penfold, ADAS Regional Architect, speaking on the subject of "The Agricultural Engineer and Farm Buildings", on 11 March, made the case for co-operation between the agricultural engineer and the architect in the planning of farm buildings. He referred to the fact that seldom did the architect get a proper brief from the customer with regard to his forward intentions.

A recent survey of 25 farms had shown haphazard growth of farm buildings from the original ones constructed up to 200 years ago to more recent additions which had been placed in a haphazard manner and not integrated into the farm system, nor unfortunately into the planning of the local area.

Mr Penfold also referred to the need to harmonise farm buildings with the countryside in which they are being placed, with a special emphasis on the needs of National Park areas, and said that in his opinion the approach of colour alone was not adequate. A combination of colour, materials and structure were required to harmonise the farm building with the locality so that an eyesore was not the result.

Returning to the subject of the economics of farm buildings he remarked that in the survey of the 25 farms alluded to above it had been found that from 8% to 40% of the buildings was allocated to the work force who occupied it for less than 4% of the time. This contrasted with an overall investment of £38 000 per man in agriculture and he wondered whether this was really economical.

Mr Penfold then made the plea for the organisation of agricultural buildings to be measured in terms of economics, efficient routines, aesthetics and an adequate working place for the force.

He referred to the fact that unfortunately in this country we have no integrated approach at this moment and advocated a cost check on the expertise in other disciplines. He made the case for a design centre specialising in agricultural buildings.

Whilst it was admitted that the broiler chicken industry showed a planned approach to the mechanisation of that industry, good planning of farm buildings in general could save up to £20 million per year for the industry.

In his view therefore it was necessary to approach this from a management point of view and to design buildings which were adaptable to the changes of farming practice which in turn were affected by the changing economic patterns in the country.

Part of this adaptability could be aided by better design of the units and instances of bad design of asbestos cladding units were quoted. It was stated that there was no pressure on manufacturers at this point of time to improve design.

Mr Penfold concluded by saying that he backed the philosophy of the President of the Institute of Architects, namely that of long life, loose fit and low energy, defining this as a life of 25 to 40 years, loose fit to be adaptable to change of farming practice, and low energy requirements for heating etc.

Eire

A SEMINAR on "Agricultural Engineering" was held in Dublin on 3 and 4 February under the sponsorship of the Institution of Engineers of Ireland. The objectives of the seminar were:

(a) To create a general awareness of agricultural engineering and its role in the Irish economy.

(b) To determine how and to what extent, agricultural engineering is being applied at present and to define future needs.

(c) To examine the need for agricultural engineering in the future and the educational and training facilities required to meet this need.

The seminar was opened by the Minister for Agriculture and Fisheries, Mr Mark Clinton, who briefly described how envineering through mechanisation had revolutionised farming over the past 50 years. He recalled his own early association with Harry Ferguson and how he had demonstrated the first Ferguson tractor in this country.

The seminar consisted of four sessions each having a main speaker, three to four reply speakers and an open discussion.

1 Engineering in agriculture — a perspective

Dr T Walsh, Director, *An Foras Taluntais* (Agricultural Institute) emphasised the importance of agriculture in the Irish economy. Agriculture, he described, as more than just farming — in fact, it embodied a broad sophisticated range of production, processing, supply and service industries.

The engineering input to this industry had been directly through agricultural services (eg electricity supply, land drainage, forestry) and indirectly through a host of industries either serving or dependent on farming; engineers were employed in fertiliser manufacture, in milling and provending and in the concrete products industry. The scope for engineers had widened considerably in recent years in the processing of foods such as milk, meat, cereals, sugar and vegetables.

2 Engineering on the farm

Professor J O'Callaghan, The University, Newcastle-Upon-Tyne, declared that, in spite of the major advances in farm mechanisation, the problems in the long run had remained the same ie how to deploy resources of land, labour and capital to provide a reasonable income for the farmer with reasonable certainty. In this context, we should be conscious of the management conflict, which influenced both mechanisation and the structure of farming, between the economics of specialisation and the biological soundness of mixed farming.

Among the conclusions drawn by Professor O'Callaghan were:

(a) There was reliable evidence from field trials that considerable increases in production per unit were still possible. While part of the increase might be from better genetic inputs, the major contribution lay in better control of the other inputs such as drainage, cultivations, nutrients, control of weeds and disease, reduction of harvesting losses, factors which depended on mechanisation.

(b) The range of machinery available to farmers was likely to decrease as manufacturers concentrated production in fewer lines with longer runs. The problems of operating and managing expensive high output machines were such as to repay investment in training operators. National opportunities in agricultural engineering lay in selling, servicing and adapting international lines to meet local needs, in producing specialised equipment for which there was a regional demand and in providing a technical service of advice for both producing and processing the whole range of products in the agricultural-food chain.

(c) Agricultural engineering was now the main tool of management in agriculture; biology told us what to do, economics told us what we should do, and engineering did it!

3 Agricultural processing

Mr M Sheehy, Chief Executive, Irish Sugar Co Ltd defined agricultural processing to include both the processing of agricultural produce and the manufacture of input materials for the agricultural production sector. In reviewing his company's progress, Mr Sheehy demonstrated how the initial enterprise of sugar manufacture had been extended to cater for the specific needs of the Irish grower ie improving the actual seed, its method of planting, nutrition and protection, and the manufacture of production and harvesting machinery. From importing 100% manufacturing technology in 1926, the wheel had turned full circle and the company was now able to export far more expertise than had to be originally sought from abroad. More recently the Irish Sugar Company had diversified into the food processing area especially in root crops and vegetables.

In a broader context, Mr Sheehy showed how and why the overall Irish food processing industry had expanded rapidly over the past 15 years. For example, Ireland's food exports now outranked all their EEC partners as a percentage of total national exports, ie Ireland 43.3%, Denmark 35.3%, Netherlands 22%, France 18.4%, Italy 8.1%, UK 7%, Germany 4%, Belgium 2%.

In all these developments the role of technologists, including engineers was essential.

4 Education and research

An international flavour was provided by Ir F Coolman, Director IMAG, who outlined the education and research system employed in the Netherlands. He described how agricultural engineering education was provided at various levels in the primary, secondary and higher agricultural schools as well as in the agricultural university.

Research at IMAG and the experimental stations was conducted in close co-operation with the trade and industry. Mr Coolman concluded that adequate advisory services follow naturally from the research work. These services comprise not only government advisers but also the information officers from trade and industry and the seller of the machine. The adviser must be able, with the help of the research institution to recommend the right machine to the farmer and to discuss with him its proper place in the farming system. The farmer had to be an adequate party to such discussions, which meant that his education should provide him with the necessary knowledge. It was quite evident from Mr Coolman's address that agriculture in the Netherlands was well served by engineering, especially at farm level.

Conclusion

The seminar was attended by a wide range of interests including the Department of Agriculture and Fisheries and its Advisory Service, the Agricultural Institute, farm machinery manufacturers and traders, universities and technical colleges, dairy co-operatives and food processing firms. The discussion from the floor after each session was rewarding and the general conclusion seemed to be that Irish agriculture could benefit enormously from increased agricultural engineering expertise. In particular such expertise was urgently required in the advisory services.

The Institution of Engineers of Ireland now intended to submit to the Minister for Agriculture and Fisheries a report including recommendations on how increased agricultural engineering expertise can best be provided and incorporated into Irish agriculture.

Corrections

IT is regretted that several errors crept into the last issue (31/1) of *The AGRICULTURAL ENGINEER*.

The rates of work on p 17 (last but one paragraph) were given as varying from 2.5 to 3.5 hectares/hour. These should have been 2.5 to 3.5 hectares/day.

The caption to fig 2 (Mechanical harvesting of raspberries) on p 18 should have read:— "Effect of stroke length and frequency on yield of fruit".

ADMISSIONS

The undermentioned have been admitted to the Institution in the grades stated:—

Fellow (FIAgrE)

Wheeler J V T	Salop	9	8 1 76	
White D J	Beds	5	8 1 76	RD

Member (MIAgrE)

Aked L G	Zambia		29 11 75	
Blowers W J	Kenya		26 11 75	CS
Chark Y B	Singapore		11 12 75	AD
Demian T F	Sudan		29 11 75	ED/RD
Erami A	Leics	2	29 11 75	
Lee G D	Cumbria	3	30 12 75	ED
Oogarah P	Mauritius		1 12 75	
Willcock D B	Kenya		5 1 76	FE/CS

Technician Associate (AIAgrE)

Broomer K D	Hants	13	5 1 76	ED
Goodman M J	Surrey	13	1 12 75	PR
Onwuji J I	Nigeria		23 9 75	ED

General Associate (AIAgrE)

Bankole J O	Nigeria		8 11 75	
Carter D	Surrey	13	30 12 75	DM
Cullen M J G	Midlothian	4	5 1 76	
Hanmer H E G	Warks	8	30 9 75	AD/FE
Mendham J T	Suffolk	1	15 2 76	AD
Muir A P	Somerset	6	5 1 76	CS/DM/FM
Sully C M J	Wilts	7	23 10 75	

Graduate

Cham P A	Gambia		5 1 76	AD/TS/ED
Cole R J	Sussex	13	30 12 75	TS
Lamb S J P	Australia		5 1 76	TS
Lee C J T	Staffs	9	23 10 75	TS
Thakrar H T	Middx	13	22 4 75	

Student

Wilson S C	Beds	5	15 2 76	ED

TRANSFERS

The undermentioned members have been transferred to the grades stated:—

Fellow (FIAgrE)

Nation H J	Beds	5	8 1 76	RD
Whitehouse A E	Worcs	8	8 1 76	AD

Member (MIAgrE)

Back H L	Iran		29 11 75	FM
Barber A	Kenya		29 11 75	AD
Bateman G H	Sussex	13	5 1 76	FE
Campbell C	Herefords	8	5 1 76	ED/AD
Carvill J V	Eire		29 11 75	RD
Counsell G	Cornwall	6	29 11 75	FM/FE
Craven J S	Yorks	10	26 11 75	TS
Culy D S	Berks	13	5 1 76	CS/FE
Defeu D P	Somerset	6	2 8 74	
Foulger S R	Salop	9	30 12 75	
Green B	Ayr	4	11 12 75	TS
Holland D	Canada		29 11 75	BD/DM
Inwood P A	Norfolk	1	26 11 75	AD/FE
Sims B G	London		10 11 75	AD/ED
Taylor M J	Northumberland	3	29 11 75	FE
Tullett K N	Sussex	13	10 11 75	ED
Wilkes J M	Kent		9 12 75	ED
Wyles P W	Notts	2	5 1 76	ED

Technician Associate (AIAgrE)

Poolman J R	Berks	13	5 1 76	
Vinden M H	Bucks	5	10 11 75	TS

General Associate (AIAgrE)

Perera D A L P	Sri Lanka		29 9 75	TS

Graduate

Kidson D I	Yorks	10	8 11 75	

RE-INSTATEMENTS

The undermentioned have been re-instated:—

Graduate

Brown S R H	Malawi		6 8 73	

RESIGNATIONS

The undermentioned have resigned from membership of the Institution:—

Algar S M	London		31 12 75	
Bell L J	Cornwall	6	31 12 75	Retired
Brazil M	Eire		31 12 75	Ill health
Dalton J W	Cambs	5	31 12 75	Economic
Edmonds D J B	Cards		31 12 75	Retired
Fuller H R	Norfolk	1	31 12 75	Economic
Higgins L H	Worcs	8	31 12 75	
Pepper A T	Peterborough	2	23 3 76	Left industry
Saywood A E	Essex	12	31 12 75	
Shepherd W R	Essex	12	31 12 75	
Travis J	Yorks	10	8 3 76	
Webb A S	Warwickshire	8	31 12 73	

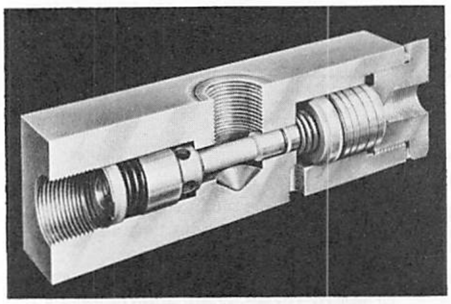
DEATHS

We regret to announce the death of the undermentioned members:—

Knight B J (Mrs)	Devon	6	2 76	
Newcombe-Baker A	Norfolk	1	28 1 76	(Date advised)
Ransome R B	Suffolk	1	16 3 76	(Date advised)
Williams E L	Warwickshire	8	22 1 76	

Occupational classification

AD — Advisory Services; BD — Buildings; CS — Consultancy; FE — Field Engineering; PR — Technical Journalism; TS — Technical Sales, Distribution and Service; DM — Design and Manufacture; ED — Education; EL — Electrification; FM — Farming; RD — Research and Development.



TWO new pilot operated check valves — for hydraulically locking a cylinder or component by preventing return flow until pressure is admitted to pilot port — with port sizes of 3/8 in BSPP and 3/4 in BSPP, both with a 1/8 in BSPP external drain — for flows of 8 Imp gall (36 litres)/min and 25 Imp gall (113 litres)/min respectively are available from Hydraulic System Products Ltd.

Each valve has a maximum working pressure of 5000 lbf/in^2 (350 kgf/cm^2) with check valve cracking pressure of 5 lbf/in^2 (0.35 kgf/cm^2). Piston to check valve ratio is 4:1; the pilot system is isolated from the media.

Valve body is of steel, with case-hardened and ground poppet and pilot system.

Hydraulic System Products Ltd, Queensgate House, 23 North Park Road, Harrogate HG1 5PF (tel Harrogate [0423] 3442).

Branches

1 East Anglia; 2 East Midlands; 3 Northern; 4 Scotland; 5 South East Midlands; 6 South Western; 7 Western; 8 West Midlands; 9 Wrekin; 10 Yorkshire; 11 North Western; 12 South Eastern; 13 Southern.

ENGINEERS REGISTRATION BOARD

The undermentioned have been placed on the ERB (CEI) Register:—

Technician Engineer (TEng [CEI])

Abarikwu O I	Nigeria		20 11 75
Cermak-z-Uhrinova J P	Aberdeen	4	20 11 75
Cobbald T E	Avon	7	20 11 75
Eu W C	Singapore		20 11 75
Hickling J A	Kent		1 3 76
Kitching R B	Lancs	11	18 12 75
Lee A	Co Down	NI	20 11 75
McGrath D M	Eire		20 11 75
MacPherson W O	Northumberland	3	20 11 75
Newman J R	Dorset	13	28 10 75
Sloan W E	New Zealand		20 11 75
Suttie J S O	Warks	8	1 3 76
Teh S W	Malaysia		20 11 75
Thanki S B	India		20 11 75

Technician (Tech [CEI])

Okoye S O	Nigeria		20 11 75
Percival J H	Berks	13	18 12 75

"Shell" award to students

THE President and Members of Council of the Institution are pleased to announce that it is proposed to introduce a series of Awards for Papers and other activities during the forthcoming Session, 1976/77.

The first of these Awards applies specifically to Student members of the Institution; Shell UK Oil have kindly donated three prizes of £100, £75 and £50 which will be awarded for the best papers covering improvements in the handling and transportation of material and produce on the farm. A panel of adjudicators will be appointed by the Institution.

Any registered Student of the Institution is invited to apply to the Secretary for further details.

Closing date for any submissions will be 1 September 1976. The Awards will be presented to the winners at the Autumn Conference being held at Norwich on 9 November 1976.

Details of other Awards will be announced in the next issue of the Journal.

Members of Council 1976/77

THE undermentioned members were elected to Council on 11 May 1976 for the 1976/77 Session.

President	T C D Manby
President-Elect	J C Weeks
Immediate Past President	J V Fox
Past President	J M Chambers
Vice President	J C Turner
Vice President	W T A Rundle
Vice President	B D Witney
Honorary Treasurer	U G Spratt
Honorary Editor	B C Stenning
Fellow	G L Reynolds
Fellow	P Wakeford
Fellow	G E E Tapp
Member	J H Neville
Member	U G Curson
Member	J G Shipman
Companion	A J Gane
Companion	D MacMillan
Associate	E H Mander
Associate	R Chambers
Associate	J Kinross
Associate	G W J Goddard
Graduate	A L Cox
Special Representative for Scotland	B D Witney
British CIGR Association Representative	D P Evans