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INSTITUTION NOTES

Agricultural Engineering Symposium

The Institution will hold an Agricultural Engineering Symposium from 11-14 September 1967 at the National College of Agricultural Engineering, Silsoe, Bedford. The purpose of this Symposium is to bring together agricultural engineers and other scientists from research stations, the advisory services, universities, agricultural colleges, farm institutes and the industry. Papers of a high technical standard will be presented to delegates who will be drawn both from the U.K. and overseas. The Symposium will provide a unique opportunity for the detailed discussion of agricultural engineering research design and other technical matters by informed specialist groups to whom the subjects are of vital importance.

The Symposium will be divided into five divisions covering the following fields: (I) Mechanical handling and farm buildings; (II) Research and design; (III) Machine performance; (IV) Soil mechanics; (V) Water application and control.

It is hoped that by giving more than a year's advance notice the maximum attendance will be achieved. Detailed information will appear in the *Journal* in due course. Enquiries should be addressed to the Honorary Organizing Secretary, Agricultural Engineering Symposium, Institution of Agricultural Engineers, Penn Place, Rickmansworth, Herts.

Journal Publication Dates

In order to establish a close liaison with arrangements for Open Meetings and to accommodate in the Journal technical articles available from other sources, the four issues which appear each year will with immediate effect be classified *Spring*, *Summer*, *Autumn* and *Winter* instead of being linked to specific months.

Annual Subscriptions

The small proportion of members who have as yet not remitted their annual subscription for 1966 are cordially reminded that the Institution must be able to rely on receiving regularly subscription income in order to be able to budget for expanding activities. The facility of payment by Banker's Order is worthy of emphasis, and the appropriate form may be obtained from the Secretary.

Income Tax relief on the whole of the Annual Subscription may normally be obtained by completing form P358, obtainable from the Tax Office.

1966/67 Session Programme of Open Meetings

Autumn National Open Meeting

Plant Soil and *Water:* an all-day meeting on Monday 26 September 1966 commencing 10 a.m. at the Essex Institute of Agriculture, Writtle, Chelmsford. Speakers will be drawn from the National Vegetable Research Station, the Drainage Division of the Ministry of Agriculture, Fisheries and Food, the National Agricultural Advisory Service, and the CSIRO Irrigation Research Laboratory, Commonwealth of Australia.

Spring National Open Meeting

The Contribution of Agricultural Engineering in Developing Countries: an all-day meeting on Thursday 16 March 1967 at the University of Reading, Berks.

Annual Conference 1967

Mechanization of Cattle Feeding: an all-day conference on Thursday 11 May 1967 in the Conference Hall of the Institution of Mechanical Engineers, BirdcageWalk, London SW1.

Complete programmes for the above events will be announed in due course.

Forthcoming Conferences and Events

Fourteenth International Course on Rural Extension: 4-29 July, Wageningen, The Netherlands. The course is organized by the International Agricultural Centre under the patronage of the Food and Agriculture Organization of the United Nations and the Organization for Economic Co-operation and Development in collaboration with the United Nations Educational Scientific and Cultural Organization. Programme and application form are obtainable from the Director International Agricultural Centre, Post Box 88, Wageningen, The Netherlands.

CIGR Section Four: International Conference on Electricity in Agriculture: 17-20 September 1966 at Lindau/Bodensee. Subjects covered in the conference will be light and productivity with reference to animals, edible and decorative plants, and the use of electricity in farm buildings with special reference to heating and ventilation. Programme and registration form are obtainable from the Secretary, Institution of Agricultural Engineers, Penn Place, Rickmansworth, Herts.

Steel Congress 1966—'Steel in Agriculture': 25-27 October 1966 at Luxembourg. Correspondence should be addressed to the Steel Congress, E.C.S.C. Luxembourg.

PUBLICATIONS AND REVIEWS

Crop Husbandry
by H. C. Mason
MSC, NDA, AMI AGR E
The English Universities Press Ltd;
1965; 12*s*. 6*d*.

Published by the English Universities Press, Ltd, this book is one in a General Technical Series under the general editorship of Air Commodore J. R. Morgan, OBE, RAF (RETD), BSC, MI MECH E, FRAES and has been written 'specially to meet the requirements of students studying Syllabus No. 265, of the City and Guilds of London Institute on Crop Husbandry'.

Primarily of use therefore for the first stages of study, the book, which extends to some 180 pages and includes a large number of illustrations, covers a wide range of the principles and practice of Crop Husbandry and deals briefly with some of the machinery required in the various stages of crop production. A considerable amount of useful information is included in each of the 27 chapters under such chapter headings as 'Land', 'Plant Food', 'Botany of Agricultural Crops', 'Potatoes', 'Pulse Crops', 'Grassland Management' etc. Most chapters are short, that on 'Sugar Beet' extending to 5 pages for example—although that on 'Cultivation and Soil Fertility' covers 14 pages—consequently there is little space for much discussion of many of the points touched on. It is unfortunate that the lists of examples of crop varieties quoted in the appropriate chapters do not include in a few cases some of the more recently introduced varieties which have achieved popularity. M.G.B.

John Fowler and the Business he Founded
by Theo Davies
John Fowler & Co (Leeds) Ltd;
Limited edition,
first published 1951,
reprinted 1964

Except for a short biography by Rolt, little has been written about Fowler. Now a hundred years after his death some of us are beginning to appreciate his greatness both as an engineer and as a much-loved man, dedicated, generous and without guile.

It is strange that George Stephenson the father of mechanized transport should be famous while Fowler the father of mechanized food production remains unknown.

Mr Davis has written a book which is needed and readable. Less than half of it is about John Fowler but this is because of his early death and the vigour with which his family and friends carried on his work after it. The John Fowler story to be complete cannot end with the fatal accident in 1864.

It is a browsing book which can be opened anywhere and enjoyed. Unfortunately, it is insufficiently illustrated. D.R.B.

Fundamentals of Soil Science
fourth edition
by C. E. Millar,
L. M. Turk and H. D. Foth
John Wiley & Sons;
1965; 75*s*.

This is the fourth edition of a textbook on soil science which first appeared in 1943. It has been considerably revised and improved, and can be recommended as an introduction to soil science and its application to agriculture.

The emphasis throughout is on the agricultural implications of the subject but the fundamentals of the science are adequately covered.

Although the basic scientific concepts necessary to the understanding of soils are introduced and explained as they are required, such accounts are necessarily brief and presumably the reader is expected to have a general scientific background. The chapters on the physics, chemistry and biology of soils are followed by an account of the origin and classification of soils, and the great soil groups of the world. Having discussed topics mainly concerned with the soil itself, then the nutrient requirements of plants are considered and this leads on naturally to chapters on fertilizers and manures. The final chapters deal with soil erosion, soils of arid regions, irrigation, and a short account of soil resources.

Since the book is American, the many examples are naturally taken from that country and this does somewhat reduce the value of the book to the reader not familiar with America.

At the end of each chapter there is a list of questions and problems. This should be a help to the student, as should also be the glossary at the end of the book.

The book is very well illustrated and produced, and has an excellent index. It is a very useful elementary account of soil science. D.P.

PUBLICATIONS AND REVIEWS

(continued)

Water Engineer's Handbook 1965
edited by D. Wilkinson and N. Squire of WATER AND WATER ENGINEERING;
Guardian Technical Journals Ltd;
1965; 25*s*.

A very useful reference book and directory for everyone working within and connected with the water supply industry. It will be of interest to a reader wishing to obtain outline knowledge of this industry or to anyone seeking information about water supplies and undertakers in some part of the United Kingdom.

The directory information found in this book can be summarized as follows:

Water Undertakers
Names and addresses of chief officials, sources, available yields, limits of distribution, hardness and charges for water.

Reservoir (Safety Provisions) Act 1930
Names of qualified Civil Engineers on the various panels as reconstituted under the above Act.

River Authorities and River Purification Boards in Scotland
Names and addresses of chief officers.

Government Departments and associated bodies
A considerable amount of information about the Water Resources Board, Government Departments, Research Organization and Committees, and the North of Scotland Hydro-electric Board which deal with water supply and its use.

Institutions and Associations
Addresses and membership of the various professional bodies but no description of their objects is given except in the case of the Water Research Association.

Buyers' Guide
A comprehensive index of the manufacturers of a wide range of plant, instruments, equipment and material.

British Standards
A list of standards applicable to Water Engineering.

The technical section of this is provided in a handy form and picks out that information most useful to a water engineer. It is not, however, a text-book. R.H.M.

PUBLICATIONS RECEIVED

British Standard 1521; 1965 (revised edition): Waterproof Building Papers.
British Standard 1578; 1965 (revised edition): Rubbers for the Dairy Industry.
British Standard 1608; 1966 (revised edition): Electrically-driven refrigerant condensing units.
British Standard 3810; Part 2; 1966; Glossary of terms used in materials handling. Part 2, Terms used in connection with conveyors and elevators (excluding pneumatic and hydraulic handling).
British Standard 3817; Part 1; 1965; Rotary cultivators (other than pedestrian controlled). Part 1, Bolts and nuts for blade attachment.
British Standard 3976; 1966; Capacity and the performace of refrigerated farm milk tanks.
British Standard 3986; 1966; Methods of test for agricultural grain driers.
British Standards Institution; PD 5686; 1965; The Use of SI Units.
Commission Internationale du Génie Rural; Proceedings of Farm Buildings Conference held in Cambridge, September 1965.
Czechoslovak Scientific and Technical Society—Agriculture and Forestry Section; Hydrological and Technical Problems of Land Drainage. Proceedings of the international symposium held in Prague, October 1965. (*In English*).
British Electrical Development Association; Rural Electrification Conference 1965—Proceedings and Supplement.
Centaxial Fans Limited: Axial Flow Fan Catalogue.
Norman Kark Publications Ltd: British Tractors & Farm Machinery (the Green Book) 1966.
J. & F. Pool Ltd; The Metal Perforators Catalogue.

All publications reviewed or listed above have been acquired for the Institution Library.

A REVIEW OF POWER TRANSMISSION IN FARM MACHINERY

by

T. Sherwen, MI MECH E, MSAE, MI AGR E*

A paper presented at an Open Meeting of the Institution at the National College of Agricultural Engineering, Silsoe, Bedford on 28 September 1965

The transmission of power by mechanical means is an ancient art and has been known literally for thousands of years. There are two main forms, namely, rotary and linear and in both cases power can be transmitted by mechanical, electrical or fluid means.

Probably the earliest example of linear transmission was when prehistoric man first used a tree branch as a lever to move a boulder. Chains have been known since about 800 B.C. and started in ancient China, whereas gears came later and were thought to have been used in Greece between 400 and 300 B.C.

The use of fluids for measurement of time was known before 1000 B.C., but their use to transmit power from one source to another did not come until much later.

The protagonists of that great genius, Leonardo da Vinci, believe that some of his sketches show what appears to be a hydraulic press, but there is no record of it having been made and it seems that the first practical press did not appear until the end of the eighteenth century.

It is interesting to note that a vane pump was projected in 1588 by Ramelli and a gear pump was illustrated in the 1600's, but they could not be made because of the lack of machinery processes to form them.

Rotary fluid power transmission followed much later. The first attempts were limited as the rotary motion was achieved by rack and pinion. Continuous rotary motion was not possible until the development of the fluid motor about 1865.

Apparently the first fluid motor was a two-cylinder oscillating motor used to rotate a capstan. This was succeeded by three-cylinder motors which were used for gun turret operation on warships. By 1874 a relatively sophisticated three-cylinder radial motor whose stroke could be changed was illustrated in the proceedings of the Institution of Mechanical Engineers.

Public hydraulic power transmission preceded its electrical counterpart and a mains supply was available in Hull in 1877. In the electrical field, it is curious to see that the first practical transmission of power occurred in rotary form, unlike the order in which the mechanical and fluid categories were developed. This took place in the late 1800's, and it was not until the present century that linear transmission by electrical means became practical.

This brief initial review is intended to set the background of the early development of power transmission in general, most of which was in fixed installations. It was not until the mid-1800's that a mobile source of power became available to the agricultural industry in the form of the steam engine.

Mobile Prime Movers

First of all, steam engines were used to replace horses to drive threshers, but the early pioneers soon thought of connecting the engine to the wheels to provide locomotion. One of the first uses of the traction engine was for ploughing and the cable system was introduced in England about 1859. Although it was tried in the United States of America, it did not find favour there and the Americans preferred to draw the plough directly behind the engine.

One of the first examples of this system was the Fawkes steam plough in 1858. The ploughs were mounted behind the engine and had lifting gear, but presumably the depth control was manual.

It was only ten years after this that we have what must be one of the earliest forms of power-driven implements. This was the Standish steam plough which had groups of blades which revolved round a near vertical axis, like the later Gyro Tiller. These were driven by shafts from the main engine and were mounted on a framework which was raised by chains.

These two examples of early pioneer work came from the United States of America, but a third one, albeit in a different field, was Thomson's rubber-tyred steam tractor. We are told that it was not very successful in England and France, but performed well in California.

The next major step in the development of farm machinery, which ultimately was to have a big effect on transmissions, was the invention of the internal combustion engine. The first engine to be patented was in 1794 by an Englishman, Robert Street, who used turpentine mixed with air as a fuel. However, it was nearly eighty years later, after much work in the United States, France and Germany, that the Otto engine of 1876 appeared and this was the first practical engine with a reasonable fuel consumption.

Another fifteen years were to pass before this form of motive power was applied to the farm tractor, and this took place in the United States of America. To begin with, the speed of these engines was no greater than that of the steam engines they replaced, so similar transmissions by open chain and cast gears were used, but by the time of the First World War engine speeds had started to rise and this, together with the necessity of protecting

* Consulting Engineer

the transmission from earth and stones, made it essential to cover it up.

Most of these early tractors were virtually stationary engines mounted on a frame with wheels, but by the First World War the form of the tractor was being influenced by the commercial vehicle which, together with the passenger car, were at the start of fifty years of continuous production development.

So far, we have dealt with the progress of the wheeled tractor, which was the logical first step from the farm cart. At an early stage, however, difficulties were experienced in bad soil conditions, and it was quickly realized that large wheels gave better results. An attempt to utilize the advantages of large wheels was made by John Fowler of Leeds about 1877 with a tractor having 12 ft driving wheels. We are told that this was not successful, partly because stones and clods of earth were deposited on the driver's head and in the driving gears!

However, early thoughts on tracklaying systems had started in 1770 when R. L. Edgworth patented a machine laying its own rails. Cayley followed in 1825 and produced a steam tractor which was supported by a belt of wood and iron. The English appear to have been first in this field, and by 1854 the Army were using Boydell steam tracklaying engines in the Crimean War. Like their wheeled counterparts, the early tracklaying tractors had open gears and chain transmission, and the development of these followed the same lines as those of the wheeled tractors.

Where power was transmitted on threshers and other farm machinery or from a tractor to such equipment, this was done by belts, usually flat. The larger torques were transmitted by chain.

These methods were universally used up to the Second World War with very little change. Then the continuous reinforced rubber V-belt replaced to a large extent the woven flat belt, and the need for better safety conditions brought about the use of covers and guards. The provision of a rearward facing p.t.o. became standard and this, together with new machinery such as pick-up balers and forage harvesters, has demanded greater power which has necessitated the use of shafts and gearing.

Hydraulic Transmission Systems

We have already seen that the earliest forms of hydraulic transmission were applied to capstan and gun turrets. The first major step forward in this field was the Rigg variable-stroke hydraulic engine, in 1885. The object of this was to economise on high pressure water when under light load.

This was followed in 1896 by what appears to be the first true hydrostatic transmission, the Hall variable-stroke gear. This used separate piston valves for each cylinder and the cylinders rotated about the crankshaft.

Eleven years later the well known Williams-Janney gear appeared in America and this was followed in the United Kingdom by the work of Professor Hele-Shaw in 1908. This was a period of great ingenuity and activity in the hydraulic world, and the possibilities of hydraulic drive were seized upon by vehicle designers. Many patents were taken out, and although few of the ideas were actually put into practice these early patents have proved awkward to present-day designers when they appear as 'citations'.

Some thirty different systems of transmission appeared at about this time, of which about a dozen were adapted for vehicles, but none survived for any length of time, partly due to the competition from electric transmissions at that time.

In those early days, mechanical design had outstripped sealing knowledge, and it is fair to say that the lack of good sealing prevented the use of high pressures, and this in turn imposed size, weight and cost restrictions on the exploitation of this type of transmission.

The next stage in the reduction of leakage was the use of finer fits and closer tolerances, which increased the cost still further and were dependent on the gradual evolution of machining processes which could produce the desired results economically.

It was many years before good seals became readily available and, even then, the extra friction that accompanied the use of these seals lowered the overall efficiency in spite of a higher volumetric component. It is only in the last decade that the art of nearly positive sealing with low friction is becoming more widely known.

The first hydrokinetic transmission was produced by Dr Föttinger in Germany for marine use and had a maximum efficiency of 83% with a torque conversion ratio of 5 to 1. These forms of transmission, together with the simple fluid coupling, have found their way into vehicle transmissions, both commercial and passenger car, largely because of the inherent slip which allows smooth take-up of drive. At this stage, one should mention the Lysholm-Smith hydrokinetic system which was used on buses for many years, and for the same reason some industrial tractors and earthmoving equipment still use this type.

However, the low overall efficiency over a usable range has always been a handicap to the hydrokinetic transmission. Great efforts have been made to improve this by refinement in blade design and lock-up mechanisms etc., but the system has not established itself in any application where economy is of prime importance.

Electrical Transmission Systems

The earliest form of electric transmission that the author could find was the Patton system in the United States of America in 1890. This used an engine driven generator to charge batteries which supplied current to a pair of electric motors connected respectively to the two axles of a tramcar. In the next twenty to thirty years a number of electrical systems appeared for vehicle propulsion, and great ingenuity was displayed in the variations to improve the low basic efficiency of around 65% by such means as electrical lock-up for high speeds and regeneration when going downhill.

The first tractor to have such a transmission was the Backer tractor of 1912. This had electric motors in the wheels fed by an engine-driven generator. However, the sophistication seems to have ended there, as we are told the steering was by means of reins!

During this development period some systems adopted electrical and mechanical steps and differential gears to extend the range of speed variation. These extra complications were brought into help when a.c. current was used instead of the original d.c. and also to help cope with increasing road speeds. Except for use in certain large earthmoving equipment, most electric systems have been discontinued, due to the extra cost, weight (especially where batteries were involved) and complications, compared with the spur gear type of transmission.

One ought to mention here the M'Karski transmission system using compressed air; it is believed that this was in use for many years on the tramcars in Nancy.

Tractor Layout

We have seen that the first half-century of tractor, and consequently transmission, development brought great progress. The first source of power, the steam engine, was well on the way to being replaced by the internal combustion engine and transmissions, apart from a few exceptions such as friction drive, had developed from crude cast open gears and chains to machine-cut gears, mostly working enclosed and lubricated.

The general form of today's mechanical transmissions was also beginning to take shape, as the type with fore-and-aft engine axis, spur gear reduction and either bevel or worm right-angle drive was gradually ousting the many other types, including the all-spur or chain types with transverse engines.

The latter really started with the steam engine whose most convenient position was with the cylinder on top of the front of the boiler and the fly-wheel lying in a vertical plane to one side. Thus, the driven axle shaft was parallel to the driving shaft.

This type of transmission has persisted in small numbers for many years, but largely because it was associated with engines with large flywheels and the only convenient way to place the latter was vertically with a transverse axis.

It has been mentioned previously that early in this century tractor transmissions began to be influenced by the progress in the vehicle industry. Since the distance between the engine and the driven axle was usually too great for gear or chain drives, the shaft was found to be the most economical method of connecting these units. Thus, the fore-and-aft axis position for the engine centre line became almost universal and this meant that with higher working speeds, gear tooth materials, heat treatment and accuracy had to be improved to stand up to the torques available. All this know-how helped the tractor manufacturers to reduce the size of their gears and consequently of the housings.

One interesting point emerged at this time. The form of wheeled tractors was divided, some with a frame on which the engine and gearbox were mounted and some with engine, gearbox and back axle forming one whole backbone member. In many cases, the former was used because most components were bought out and were not designed to mate with each other, but the latter form was used by the larger manufacturers who designed all their own sub-assemblies as a coherent whole. This trend began at the end of the First World War and was typified by the 1917 Fordson whose basic form has not altered to the present day.

Early examples of transmissions for four-wheel drive tractors appeared in the United States about 1912 and one, the Nelson, had chain drive to each axle, and four-wheel steering. One would imagine that the initial cost probably restricted the wide acceptance of this type of tractor at this stage, but the idea must have remained attractive to designers as various models emerged right through the inter-war period.

It is probably fair to say that the high cost of transmitting power by mechanical means to steered wheels, in a reliable and satisfactory way, formed the major obstacle to the four-wheel drive tractor. Several attempts were made to steer by braking, but one imagines that on a tractor that requires to be manoeuvrable steel wheels would absorb too much power in soft ground and pneumatic tyres would suffer too much wear.

The above remarks may appear to generalize too much, but the broad reasons hold good. The present-day pivot or 'knocker' method of steering did not appear until much later, and one feels that the stability problems introduced by this method of steering limit its appeal.

In the earliest days, linear transmission of power was achieved by drum and rope or chain. This satisfied the needs for a tension force and was simple and adaptable. A compressive force was achieved by screw threads or rack and pinion. These methods remained the same for centuries until the invention of the hydraulic press in 1795 by Bramah.

The next ten years saw a limited extension of the uses of hydraulic jacks, but it was not until the mid-1800's that these jacks came into general use and they became the best way of exerting large forces over a short distance. It is not known exactly when the first double-acting cylinder was introduced, but this was probably in the latter half of the nineteenth century.

Progress over the next century has been devoted to better materials, more economical use of these, improved seals and improved fluids. Today's hydraulic jacks are simple, efficient and economical, and they are unchallenged for transmitting linear motion except where the travel is too great for the mechanical stability of the design, or where the need to provide a source of fluid power makes the system uneconomical compared with a screw jack.

The introduction, before the last war, of a hydraulic system as a standard feature on some tractors has made possible the use of hydraulic jacks for tipping trailers, loaders, etc., and represents a tremendous step forward in reducing the effort required to operate and adjust farm machinery.

Engineering Development

At this point we may note some of the factors which promote and stimulate the progress of engineering development. First, there must be a demand for a better product. While an article performs its function satisfactorily and the laws of supply and demand are in balance, there is no stimulus to improve that article. If, however, something upsets the equilibrium, such as change in material or labour costs, or supply exceeding demand which promotes competition for the existing demand, then a reason exists for improving or redesigning the article.

This might be done to obtain a bigger share of an existing market, or to retain a market which has been reduced by increased labour costs. Whatever the reason, the step involves spending money on development, and there usually has to be a 'carrot' as potential reward.

In the case of the tractor and farm machinery industry, there has always been a sizeable 'carrot'. For instance, in the United States tractor production, which had been in hundreds at the turn of the century, had reached 200,000 by 1920, and the present tractor population in that country is around four million. A similar increase has taken place in the other manufacturing countries of the world.

This is the first reason for engineering expenditure in this industry. The second is, of course, that farm labour costs in these countries have risen steadily to the point where manual operations are economically out of the question, and two World Wars have further stimulated progress, giving rise as they did to acute shortages of manual labour regardless of price.

The above progress has been accelerated still further by the results achieved, in that the 'popular' tractor has been turned into a general work horse that can now be used for an enormous variety of operations previously unthought of, and this expanded the demand still further.

Manufacturing Techniques

The result of all this is that all farm machinery, and especially tractors, have had large amounts of money spent on their development. This has given us the very advanced machinery which we use today. Transmissions have received their share of this money and attention, but outside factors have controlled the stages of development. For instance, about 1850, gears were mainly cast iron because no cutting machines were then available (the standard finishing processes were hammer and chisel and filing). Steel castings were too inaccurate to be finished off by hand. Some twenty years later, gear cutting machines had begun to appear, and soon the trend was to use steel gears which were machine-cut.

The principles of generating teeth had not been known for long, so these two inter-related steps allowed higher speeds and smaller gears in transmissions, and this was the first major progress in the use of metal gears.

As mentioned previously, it was soon found desirable to keep dirt out and this, together with the necessity of lubrication, brought about the general use of gear cases.

During the period between the two wars, improvements were in three fields; first, accuracy of cutting the teeth as well as better surface finishes: secondly, in the quality of the steels used; which, together with heat treatment, allowed greater loadings and reduced wear.

Exploiting both these fields enabled hardened teeth to be produced, and then these could be form ground.

Thirdly, lubricants played an ever-increasing part in the uprating of the power which could be transmitted.

At this stage, one must say a word about the complex subject of efficiency. We will start with a quotation from Robertson Buchanan in 1841: 'Rule 1: For a pitch of three inches when the velocity is 3 ft/s at the pitch line, make the teeth as many inches broad as the number of horses power which it has to resist'.

It is perhaps paradoxical that the actual gear transmission efficiency of a pair of well run-in wooden gears lubricated with tallow from, say, a windmill in the eighteenth century was only 1%-2% less than that of a pair of modern, hardened and ground gears, running within the power range they are both capable of transmitting.

The improvements and refinements which have been made over two centuries have allowed larger powers through smaller wheels, higher speeds and quieter operation. Improved shaft and bearing rigidity and lower bearing friction have resulted in an overall improvement in the efficiency of a gear train of some 10%-12% over this period, and it is interesting to note that the use of fine-pitch teeth can halve the actual power loss over a pair of gear wheels which may be between 1% and 2%.

Of the modern finishing techniques, shaving is the most economical way of producing an accurate surface finish and crowning, which can be combined with shaving, concentrates the load towards the centre of the tooth, thus giving a longer, quiet life when distortion exists.

Gearbox Efficiency

It is not generally realized that many of the improvements in gearcutting and design have been directed to smaller sizes, quieter operation and longer life more than a better efficiency. Thus, we see that the actual efficiency which can be achieved across one pair of spur gears, running under ideal conditions, is probably 98%-99% and across a pair of epicyclic gears 96%-97%. It must be emphasized that many other factors detract in practice from these ideal figures, such as oil churning, seal friction, bearing friction, etc., and if any pumps are used to supply oil for clutch operation these can absorb appreciable amounts of power.

So taking the overall efficiency from the flywheel to the rear axle, the following figures represent approximately what can be obtained to-day:

6 to 8 speed spur gear transmission	90%
8 to 10 speed epicyclic transmission with power shift	85%
Hydrostatic transmission back to back ..	80%
Hydrostatic transmission split torque ..	85%
Hydrostatic transmission wheel motor.. ..	85%
Hydrokinetic transmission	65%

The relative cost of various types of transmission is very difficult to assess, mainly because the mechanical gear box is practically the only type which has been produced. This means that comparisons are made between a highly developed article and other possibilities which in most cases have never even been designed for mass production.

However it is thought that the following figures represent the relative costs of producing the best known types at the present moment for the orthodox form of tractor:

1. 6 speed spur gear no syncromesh	100
2. 8 speed spur gear no syncromesh	105
3. 8 speed epicyclic power change	130
4. 10 speed epicyclic power change	140
5. Hydrostatic back to back	150
6. Hydrostatic split torque	170
7. Hydrostatic with wheel motors ..	175-200*

One must say here that the last type is capable of being adapted to suit almost any form of tractor and the high cost shown above could be partially offset by savings on the structure.

The spur gearbox has monopolized the tractor field for fifty years, and one of the reasons why it has held this supremacy is that it has been considered unnecessary until recently to change gear on the move. This has meant that synchromesh was not needed and has resulted in a very simple and efficient assembly.

The vehicle field, however, has also been attracted by the epicyclic type of transmission, and it is largely the extra cost that has held this back. With the demand for automatic transmission in passenger cars after the last war the epicyclic type lent itself to automatic hydraulic control and has become very popular in conjunction with a fluid coupling or torque converter. Now that the demand is increasing for on-the-move tractor changes and particularly power-sustained changes the epicyclic train has appeared in the tractor field.

The present-day situation is undoubtedly controlled by the current concept of the 'popular' tractor as having only the two rear wheels driven. This concept is strengthened by the fact that this same tractor is virtually unidirectional in work and carries the majority of its working tools mounted at the back end.

Outside the popular class where higher powers are required, or in the smaller sizes where greater manoeuvrability is needed, there is a growing trend towards four-wheel drive. In the former category this has been achieved in the United Kingdom by various means, including hinging two tractors together without their front axles; by the use of volume production assemblies this provides a very economical answer for the power produced.

In America, where they enjoy a large domestic market, the bigger size in the orthodox type of rear wheel drive tractor has been preferred. In Europe, the smaller size of tractor, often for use in vineyards, is made with mechanical four-wheel drive, but to avoid the expense of constant velocity joints on the steered wheels a layout with a central vertical pivot point is used. This form of steering has the further advantage that a smaller turning circle can be achieved without the angular limitations of constant velocity joints.

All the above tractors use mechanical gear transmissions and since, as previously mentioned, these have been evolved over half a century and have had a great deal of money spent on their development and refinement, it is likely that this form of transmission will hold its own for many years to come on existing designs.

The Development Potential of Transmission Systems

At this stage one must say a word on the development potential of the various forms of transmission. We have seen that the mechanical gear is giving between 88% and 93% efficiency according to type of box, which means that there is only 7%-12% possibility of improvement. The law of diminishing returns takes effect, and it would appear that the cost of improving this figure by more than 1% or 2% in the foreseeable future would be prohibitive.

Regarding the future improvement of electrical systems, the author has insufficient knowledge in this field to make a forecast, but perhaps this subject will arise in the discussion.

There is much more room for improvement with the hydrokinetic transmission, some 20%-25% in fact, but the problems here are partially out of the control of the engineer. If a new operating fluid could be found with a low, flat viscosity curve and a high specific gravity, great improvements could be made in the efficiency. However, over fifty years only relatively minor steps have been achieved towards its solution, so it seems unlikely that the future will see more than 5% or so improvement in efficiency.

With hydrostatic transmissions, the future is slightly more hopeful. After the last World War, a great number of new variable-delivery pumps appeared as well as many types of motor, including several slow-speed radial-piston motors. Some of the latter have shown some very high overall efficiencies on test, in fact up to 97%. Mention must be made here of the large amount of pioneering work done at the National Institute of Agricultural Enginering, which has stimulated fresh thoughts and activities in the field of hydrostatics.

Furthermore, there is a trend towards the use of higher operating pressures, brought about by a better understanding of the sealing problems and the desire to make even more compact units. These factors, taken together, make it possible, in the author's opinion for an overall transmission efficiency of the order of 88%-90% to be achieved in the not-too-distant future, and this would bring this type of system within striking distance of the mechanical gearbox.

If any new form of tractor should emerge in the future and an open approach were to be made on the transmission, then it is possible that the freedom of form and flexibility of control conferred by the use of hydrostatic type might well weigh heavily in its favour. This could, however, only occur if the units were designed as matched components in the system and full use was made of the latest knowledge of sealing with the resultant economies in production and enhanced efficient working life.

Much of the historical part of this paper has been devoted to work outside the field of farm machinery, but it is felt that this was necessary to provide a background to developments within that industry.

Acknowledgement

The author would like to thank Dr H. E. Merritt for his help in the section on gear efficiencies and his many friends in the industry who have freely discussed the problems involved.

* According to the number of stages on the motors.

DISCUSSION

Mr E. S. BATES (British Petroleum) asked if any work study had been carried out in connection with the hydrostatic tractor to show whether there was, in spite of the lower mechanical efficiency, a saving in overall fuel consumption and manhours due to the greater flexibility of the transmission system. Commenting on this question at Mr Sherwen's invitation Mr H. J. HAMBLIN, (National Institute of Agricultural Engineering) expressed his agreement with the efficiency ratings quoted by Mr Sherwen for hydrostatic transmission and strongly supported Mr Sherwen's confidence in the future of hydrostatic transmission.

Mr H. J. NATION (National Institute of Agricultural Engineering) described the results of some experiments conducted at the NIAE on hydrostatic tractors. He said that in the case of power take-off driven machinery it had been possible to show increases in working rate with a hydrostatic transmission tractor as against its exact mechanical transmission counterpart. When driving hay-making machinery such as mowers and tedders, increases in output of 13%, 15% and 18% had been recorded and when operating power take-off driven potato harvesters figures of 22% and 35% increase in output had been achieved. In the operation of front-loaders increases in output from hydrostatic tractors of 3%, 9% and up to 18% had occurred but in some other cases no more than parity had been achieved. These gains in the case of power take-off driven equipment, Mr Nation continued, were due to the advantage of having an infinite range of gear ratios available; if in fact the gear ratio of the mechanical transmission exactly matched the job it was not possible to improve on it. In the case of front-loader work the advantage was very largely in the more convenient handling and manoeuvrability and the quicker reversibility which was available with hydrostatic transmission. A third class of work was straightforward pulling, drawbar-power work, and here experiments had shown up the expected lower overall maximum efficiency of hydrostatic transmission. In some experiments, Mr Nation added, where the drawbar horsepower test of the hydrostatic transmission model gave a maxmimum of 80% that of the counterpart tractor, varying outputs on ploughing and cultivating work had been obtained. In one set of experiments an average of 92% for the hydrostatic model had been obtained in comparison with the mechanical transmission; the stepless speed facility was giving back 12% of the 20% deficit under variable conditions where the mechanical transmission model was not at full power continuously. Mr Nation regretted that figures for fuel consumption were not currently available although records of this had been kept in some of the comparative tests.

Commenting on the history of the development of mobile prime movers as described by Mr Sherwen, Mr Nation recalled the work of Usher of Edinburgh who had utilized a portable steam engine to drive a rotary cultivator; it appeared that the machine progressed across the field not only through the drive to its back wheels but also through the drive to the cultivating mechanism at the rear. He believed this development occurred about 1849 and thus pre-dated the two American tractors to which reference had been made in the Paper. Mr Nation then spoke of the Hall variable-stroke hydrostatic transmission which he would describe as a differential gear, a split torque gear of the casing reaction type which achieved a high efficiency at 1:1 speed ratio by virtue of the lock-up, in this case an hydraulic lock-up achieved by reducing the motor to zero stroke so that the whole assembly rotated as a solid mass with the input shaft. Mr Nation said that retrospective congratulations were owed to Dr Foettinger of fifty years ago for having obtained 83% from a torque convertor—a figure which apparently could not be bettered today. The 65% overall efficiency quoted in the Paper for a hydrokinetic transmission as a whole implied an 18% gearing loss! He would like to have Mr Sherwen's views on the future possibilities for variable blading torque convertors in the modern tractor.

Mr Sherwen recalled the time when he was engaged on development work on one of the earliest moving blade torque convertors to be applied to a motor car. Assembling and stripping the converter in between tests was a very lengthy operation, due to the complicated construction and although great progress has been made over the last 30 years the advantage in efficiency obtained does not usually justify the extra cost. Mr Nation asked how the peak efficiency figures quoted by Mr Sherwen varied with the load for these types of transmissions. The fall-off with some transmissions could be quite considerable at part-load. For instance with a power-shift type, clutches were kept engaged by hydraulic pumps and a very large number of clutch plates and brake bands rotated in close proximity but were not in fact engaged and the consequent losses represented part of the approximately 10% difference in efficiency between the straight mechanical gear and the power shift gear. Was there in fact constant loss or would other factors cause it to be more or less than the 20% difference in proportion at half load? He believed that one of these power-shift transmissions relied upon a circulation in the region of fourteen gallons per minute to keep the whole system cool. In his reply Mr Sherwen said that the efficiency of a hydrostatic transmission was bound to vary with the load because some of the losses were constant irrespective of the load. It was important to remember however, that a portion of that deficiency would be recovered by the convenience of operating the tractor. He then made reference to a film shown by the NIAE of the handling of potato pallets. The picture was split horizontally showing identical pictures of operations—in one part of the screen was a mechanical tractor with a front loader picking up boxes and underneath was a hydrostatic tractor with front loader picking up boxes. As the film progressed one saw the tractors come off the field and put their boxes on a trailer and return to the next station; after about 4 or 5 stations the hydrostatic tractor was practically one cycle of work ahead of its mechanical counterpart. In economic terms, in fuel costs etc., there was compensation for the lower transmission efficiency of the hydrostatic tractor.

(*continued on page 36*)

TRANSMISSION OF POWER BY POWER TAKE-OFF

by

J. A. Howard, AMI AGR E*

Presented at an Open Meeting of the Institution at the National College ofAgricultural Engineering, Silsoe, on 28 *September* 1965

Introduction

The development of the farm tractor was, in its early days greatly influenced by the desire of engineers and farmers to replace the horse with a mechanical and more powerful means of traction. Early tractors were therefore thought of mainly as traction machines to draw the already extensive range of farm implements which were already used with the horse.

Some implements already existed which required energy to operate them in the form of rotary motion, reaping and binding machines, and reciprocating grass cutters are examples. The usual way of providing the rotary motion was from ground-engaging wheels which were driven round by the soil as the implement moved forward. This in effect created a mechanical transmission link between the horse or tractor and the mechanism of the implement through the medium of the soil. The soil is a very inefficient medium for the transmission of power and consequently only implements requiring a low power input were successfully developed as wheel driven machines.

With the introduction of the tractor it was soon realized that excess power was available beyond that required to move the tractor over the ground and that, given a suitable transmission system, this power could be applied directly to the implement. Many systems of power transfer to the implement have been suggested and tried, but the one that has stood the test of time is the power take-off shaft extending rearwards from tractor rear transmission housing, although this is not the only means of power take-off from the tractor in use today. It can however be said that nearly all modern wheeled tractors in popular use on farming operations throughout the world are equipped with a rear power take-off.

P.t.o. Drive Train

The modern tractor rear power take-off is a branch of the tractor transmission which transmits power directly from the tractor power unit, independently of that part of the transmission unit which carries power at varying speeds to the wheels. The power take-off speed is not dependent on the forward speed of the tractor but is constant for a given engine speed, except in the case of ground speed drive power take-offs which have a speed which is in direct relationship with the forward speed of the tractor.

The transmission from the power unit flywheel to the power take-off is either through the main tractor clutch, in which case the power to the power take-off is interrupted whenever the tractor is stopped by releasing the main clutch or alternatively, through a secondary clutch which is provided as an independent clutch for the power take-off drive.

The independently clutched power take-off is one which may by virtue of the addition of a second clutch, continue to operate when the forward motion of the tractor is stopped. The independent power take-off possesses, for instance, the advantage of permitting the operator to bring the mechanism of a forage harvester up to speed before the forward motion of the tractor is started.

With most power take-off driven implements the speed of operation and hence the power take-off speed is very important to the correct functioning of the implement. In order to achieve implement interchangeability it is necessary that the power take-offs of tractors do conform to standards for speed, size and position in relation to tractor implement attachment points. Such standards have been developed and are laid down in British Standard 1495, where the height of the p.t.o. shaft from the ground is specified as 21 to 26 in. In practice the heights on tractors vary from 18 to 31 in. Until 1964 one standard power take-off speed of 540 rev/min was specified. In 1964 a further power take-off speed of 1,000 rev/min was included in the standard.

The 540 rev/min power take-off was developed when farm tractors were of relatively low power and the power take-off driven implements in general use did not require the power that some modern implements demand. The power take-off itself was either $1\frac{1}{8}$ in. diameter or $1\frac{3}{8}$ in. diameter. Some tractors are equipped with $1\frac{3}{4}$ in. diameter power take-off, but this is largely confined to the bigger crawler or industrial type of tractor.

The power output of the farm tractor in the popular range has risen from 30 hp to as much as 125 hp, all of which are required to power power take-off driven implements. At the same time implements have developed which make use of the full power output of the power take-off. Examples of such implements are rotary cultivators and flail type forage harvesters.

* Howard Rotavator Co. Ltd

Fig. 1
Heavy duty Rotary Cultivator capable of cultivating 130 in. wide strip at each pass. Designed for use with tractors up to 125 hp

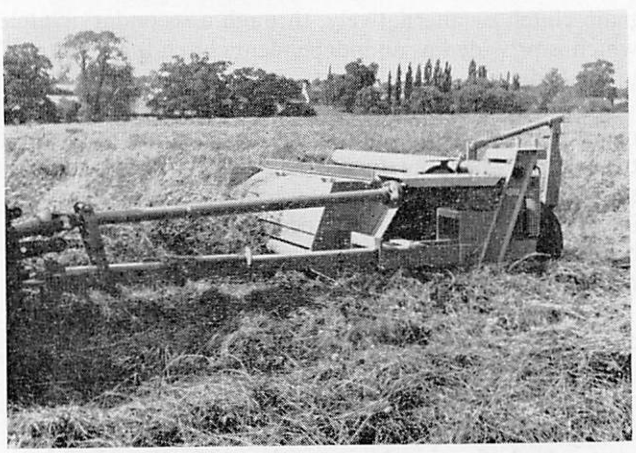

Fig. 2
Example of a trailed type of implement (hay cutting machine)

Fig. 3
Example of a 3-point linkage mounted implement

The maximum engine power which a $1\frac{3}{8}$ in. diameter power take-off shaft turning at 540 rev/min can transmit safely is in the order of 60-65 hp. The introduction of the $1\frac{3}{8}$ in. diameter 1,000 rev/min power take-off was therefore necessary to enable the power of the larger tractors to be transmitted to the implement. Tractors of up to 125 hp on the engine flywheel are now successfully supplying power to implements such as rotary cultivators which demand all the power the engine can provide.

The connection of the tractor power take-off to the implement power input shaft usually presents the implement designer with considerable problems. Except where implements are rigidly fixed to the rear of the tractor, the power take-off to implement link has to be able to accommodate out-of-line movement between the tractor and implement in both vertical and horizontal planes.

Power take-off driven implements fall mainly into two categories: (1) Trailed implements which are towed by the tractor from its drawbar and have wheels which carry the majority of the implement weight, i.e. farmyard manure spreaders, balers etc. (2) Mounted implements which are attached to the tractor by the tractor 3-point linkage and which are lifted by the tractor hydraulic system for transport (i.e. mounted rotary cultivators, reciprocating grass mowers etc.) Both types of implement have similar problems in the transmission of power from the power take-off to the implement.

The method generally used for the connection of the power take-off to the implement is by means of universally jointed transmission shafts. Two types of universal joint can be used on the power take-off to implement shaft. Firstly, the *Hooke* type of joint which suffers from the disadvantage that a single joint when being operated at an angle to the power take-off shaft will not deliver uniform angular velocity at the output end of the joint. Secondly, the constant velocity joint which as its name implies, does not suffer from the disadvantage of the *Hooke* joint, but is not very much used on agricultural machinery as it is costly to manufacture. If constant velocity joints were available at an acceptable cost, they

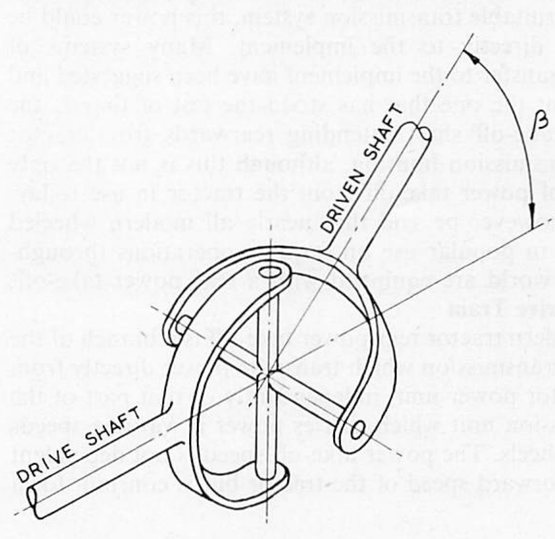

Fig. 4
Hooke type joint shown diagrammatically

would considerably simplify the attachment of power take-off driven implements to tractors.

Hooke's Joint

From Fig. 4 it can be seen that both the input and output shafts terminate in a fork which is freely pivoted on a central cross member. The arms of the cross member are at 90° and serve to transmit rotary motion from the fork on the input shaft to the fork on the output shaft, whilst at the same time allowing angular misalignment by the pivoting of the forks about the cross member journals.

As mentioned previously, this joint suffers from the disadvantage of not transmitting a constant angular velocity. This is because the angular position of the driven fork in relation to that of the driving fork is only 90° when one arm of the cross member lies in the plane of the off-set angle β. i.e. every 90° from the position shown in Fig. 4. At other positions of rotation θ of the driving shaft, the respective rotation φ of the driven shaft is given by the relation: $\tan \varphi = \dfrac{\tan \theta}{\cos \beta}$

It follows that, if there is a difference in rotation of the driven shaft to that of the driving shaft, assuming that the laten rotates at constant angular velocity, there must be a variation in velocity of the driven shaft, and that this is accompanied by both acceleration and deceleration.

joint may be overcome by using two such joints coupled together by an intermediate shaft. This arrangement enables the speed variations of one joint to be cancelled out by the other, providing the joints are in the correct angular relationship and the two joint angles are the same. This is the familiar universal joint as used for the majority of power take-off applications.

In the design of universal joints there are a number of factors to be considered depending upon the type of drive required. Some of these are as follows:

(*a*) The intermediate shaft in most cases has to be made telescopic to allow for changes in joint length. e.g. when lifting an implement mounted on three point linkage, or turning with a trailed implement.

(*b*) The intermediate shaft should be of low mass, since it is subjected to high accelerations as in the case of the single Hooke's joint.

(*c*) The intermediate shaft must be torsionally stiff to prevent angular misalignment of the two joints, particularly in the case of long drives.

(*d*) The torque capacity of the joint has to be higher than that of the rest of the transmission in applications where a large joint angle exists, since there will be large fluctuations in the speed of the intermediate shaft, and the torque is inversely proportional to the speed for a given horsepower.

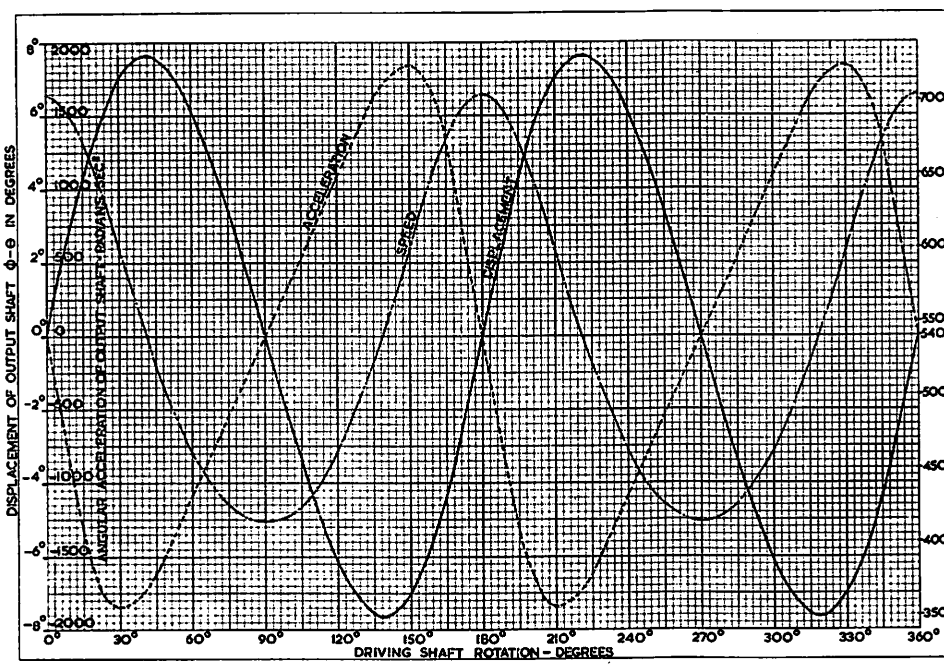

Fig. 5

Graph of a single Hooke's joint characteristics

Fig. 5 depicts graphically the above driven shaft characteristics, for a joint angle of 40° and an input speed of 540 rev/min.

If the driven shaft of this single Hooke's joint was to be rigidly coupled to a mass moment of inertia of, say .25 ft lb/sec the fluctuations in torque transmitted would amount to ±460 lb-ft: which represents ±52 hp at 599 rev/min i.e. the angular velocity at max. acceleration and deceleration.

Fortunately the disadvantages of the single Hooke's

In consideration of the above facts it can be seen that too large a joint angle should be avoided, since it not only complicates the joint design but also reduces the joint life and efficiency due to friction and large journal movements.

It has been found with 3-point linkage mounted rotary cultivators, that the maximum angle of operation of a universal joint in a two joint shaft when working continuously, should not exceed 20° if a satisfactory joint life is to be achieved. The angle can be increased to 45° for short periods when the power being transmitted is small in comparison with the maximum power capacity of the joint at low angles. This condition is produced in a 3-point linkage mounted rotary cultivator when the machine is lifted clear of the ground for turning at headlands.

Constant Velocity Joint

The disadvantages of the single Hooke's joint may be overcome by the use of a constant velocity joint. These fall into two main classes, the first being in the form of a pair of close coupled Hooke's joints, which are mounted within a spherical housing, in such a manner that the angle between the input and output shafts is always halved by the intermediate member.

The second type of joint is a true constant velocity joint, and works on a different kinematic principle to that of Hooke's joint.

For power to be transmitted from one shaft to another at constant velocity, the point of transfer of power must be on a line bisecting the angle between the shafts and passing through the point of intersection of the shaft axes. This is shown diagrammatically in Fig. 6.

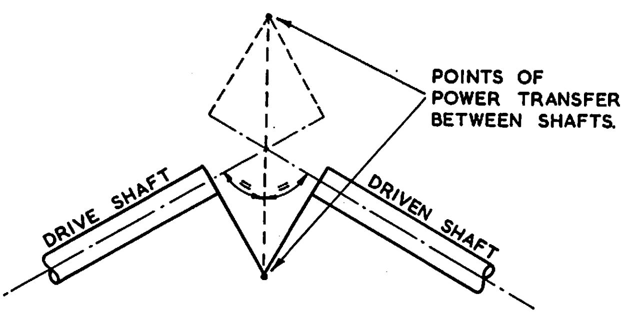

Fig. 6
Principle of constant velocity joint

For this arrangement to be achieved in practice it is necessary to have sliding members, which with their attendant friction reduce the efficiency of the joint. Although the efficiency may be improved by employing rolling balls it is of necessity more complex than the Hooke's type joint and therefore more expensive.

Axial Loads on Universal Joint

The axial movement is provided by telescopic shafts and tubes between the universal joints. The telescopic members are usually in the form of a square or rectangular shaft sliding in a mating sleeve. Splined shafts within splined sleeves are also used particularly on shafts required to carry high powers.

The telescopic section of a universal joint shaft can be a source of trouble when sliding has to take place under high power conditions. The friction present can set up high thrust loads on both tractor power take-off and implement input shaft. For this reason universal joint shafts used on the large Rotary Cultivator and other implements which require high powers are usually of the splined sliding section type which are better when sliding under load than the square or rectangular type. There is also development in progress to produce shafts which slide on balls or rollers and on which axial thrust under high torque conditions is almost negligible.

The reduced thrust of the splined sliding section is most probably due to the reduced bearing pressure aiding the maintenance of the lubrication film.

To determine the actual thrust produced in a universal joint, tests have been made with resistance strain gauges attached to the joint intermediate shaft. The traces obtained in this way are shown in Figs. 7 and 8.

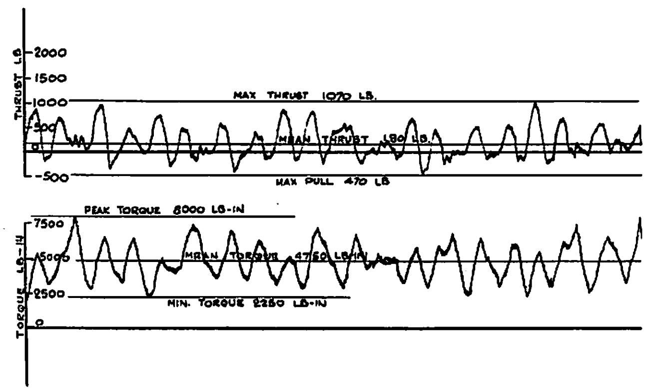

Fig. 7
Torque and thrust trace taken after the implement had been lowered into the ground with the tractor stationary i.e. telescoping the joint under low torque

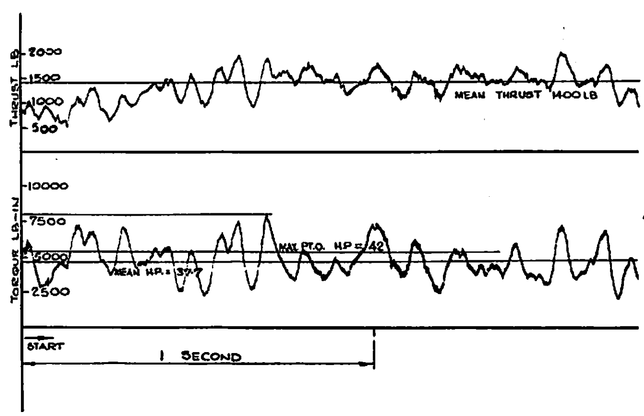

Fig. 8
Torque and thrust trace taken after the implement had been lowered into the ground with the tractor in forward motion i.e. telescoping the joint under high torque

Both traces were taken from a 50 in. rotary cultivator working at 4 in. depth in stubble on clay. Fig. 8 shows a high mean thrust level which is compressive and due to the joint shortening when the machine is lowered to 4 in. depth. The cause of the thrust variations is uncertain, but could be caused either by the implement moving horizontally due to backlash in the mounting arms etc., (or bending of mounting pins) or by vertical movement being resolved horizontally due to the configuration of the geometry of the mounting.

Allowing for the deflections in the mounting arms and pins it would require a total movement of approximately .002 in. to produce 1000 lb thrust, which would be achieved by .15 in. vertical movement at 4 in. depth. The average ratio of thrust to torque is, from Fig. 8, .23 lb/lb in., giving a μ of .3 for a $1\frac{3}{8}$ in. $\times$ $1\frac{1}{4}$ in. shaft. This indicates that under the conditions of small or no movement there is a complete breakdown of the lubrication film.

It is interesting to note that the 1400 lb mean thrust is obtained at 37.7 hp; this could increase to 2400 lb at 65 hp. The value of 2400 lb would exceed the S.A.E. bending limit of 600 lb at the groove of the p.t.o. shaft when the joint angle exceeded 7°, assuming a groove-to-bearing distance of 3 in. and a groove-to-joint centre distance of 3 in.

The foregoing illustrates the need for standardization of the height of p.t.o. shafts from above ground level, in order that universal joints may be as straight as possible under the heaviest working conditions.

Torque Loads on Universal Joints

The selection of suitable universal joint shafts for the transmission of power between the tractor power take-off and the implement input shaft has to take into account not only the average power or torque requirements of the implement, but the maximum instantaneous (high frequency) peak torque which the implement may demand. Tests with electronic torque meters on rotary cultivators indicate that average power or torque requirements have very little value as a basis for designing power take-off and implement drive parts. Maximum instantaneous peak torques are far more important, particularly in soil working implements.

The peak loads vary tremendously and are influenced by a number of factors which include the following:

1. The kinetic energy stored in the rotating parts of the tractor.
2. The power take-off horsepower available from the tractor.
3. The torsional flexibility in the drive line between the tractor power unit and the power consuming components of the implement.
4. The horsepower to operate the implement.

Items 1 and 3 have a great deal more influence on peak torque than have the power take-off horsepower of the tractor, or the horsepower required to operate the implement. This is particularly true when power take-off drives are used to operate implements such as rotary cultivators or forage harvesters, which are capable of causing very rapid changes in power take-off speed when operating in adverse conditions.

From Figs. 7 and 8, it can be seen that the torque transmitted is not even, but consists of a considerable number of short term variations, which in this case are mainly due to individual and groups of blades hitting the ground (since the soil was a heavy clay presenting an even load).

The peak torque shown on these traces is 8000 lb/in. which represents 63.5 hp, exceeding the maximum tractor hp of 42 by 51%. The ratio of peak to mean torque is $\frac{8000}{4400}=1.8$. Experience in other conditions indicates that this ratio is a useful design figure, even though at times the peak torque can reach three times the mean torque. The torque part of the traces was obtained with a torque meter mounted inside the implement gearbox shown in Fig. 9. The actual test in progress can be seen by referring back to Fig. 3.

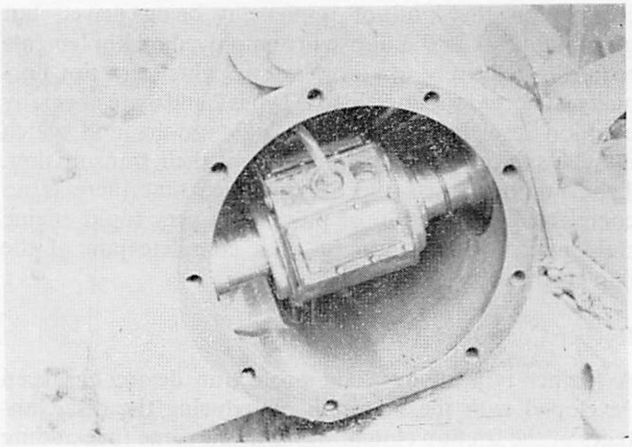

Fig. 9
Torque Meter

Overload Protection

Many power take-off driven implements include in their transmission system a safety over-load device to prevent excessive over-loads being imposed on the transmission system. Because of varied conditions under which agricultural machinery must operate, it is not always possible to provide transmissions which are capable of carrying all the over-loads which are liable to be imposed on them. It is therefore necessary to provide protection devices to relieve excessive loads should they occur.

Protection devices are usually one of the following type:

1. Those that depend upon the shearing of a replaceable link in the drive.
2. Flexible couplings.
3. Units in which spring force holds two dogs together utilizing the principle of the inclined plane.
4. Friction clutches.

The virtues of these devices are summarized as follows:

1. *Shear Link*. Advantages: simple and cheap, drive completely disconnected. Disadvantages: tends to fail through hammering effect of small overloads, takes time to replace. Danger of stronger material being used for replacement link.

2. *Flexible Coupling*. Advantages: smooths out torque fluctuations, capable of dealing with short term overloads without loss of drive. Disadvantages: no complete disconnection in event of complete jam-up, engine stalls.
3. *Spring loaded dogs*. Advantages: disengaging torque can be accurately set, relatively inexpensive. Disadvantages: will not re-engage unless relative speed of dogs is low, wear effects disengaging torque.
4. *Friction clutches*. Advantages: continues to transmit torque when slipping, automatic take up of drive when jamming torque is released. Disadvantages: wears when slipping, slipping torque is not equal to the break-away torque.

The selection of a suitable protection device depends entirely on the kind of implement being driven but usually types 1 and 3 above are employed on implements requiring relatively low powers and which are not normally subjected to shock loads. Types 2 and 4 are normally used on implements requiring high powers and which may be subjected to shock loading of their transmission. Type number 2 has the disadvantage that there is no complete disconnection of power and very rapid engine stall may result in the case of a complete jam of the implement drive.

Rubber Controlled Torque Limiter

A further type of overload protection device has been developed with the intention of reducing the disadvantages of the friction clutch, whilst at the same time adding the advantages of the flexible coupling.

The torque limiter in Fig. 10 consists basically of a multi-plate friction clutch surrounded by a number of rubber segments, which transmit the drive from the clutch to the implement. Fig. 11 shows the actual arrangement of the rubber segments.

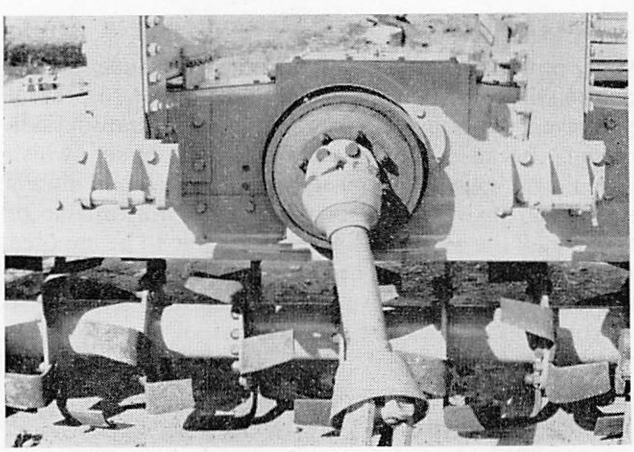

Fig. 10
Torque Limiter

The rubber blocks impart a degree of flexibility in the drive and also act as a torque sensor. Since rubber is virtually incompressible in bulk, once the initial clearances have been taken up, it will exert an axial pressure proportional to the torque applied.

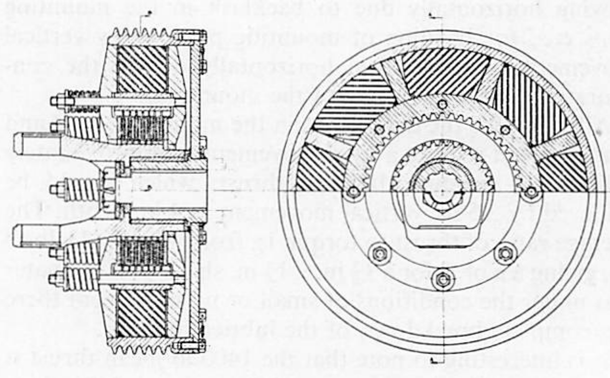

Fig. 11
Diagram of torque limiter

This axial pressure is used to off-load the clutch control springs until the point is reached when the clutch plates slip.

The theory of the torque limiter is as follows:

Let T = torque transmitted.
μ = Coefficient of friction of clutch plates.
W = Force acting on clutch plates.
P = Force produced by clutch springs.
R = Mean radius of clutch plate.
n = Number of working surfaces of plates.
K = Factor, depending upon the side force produced by the torque T acting upon rubber segments.

Then: slipping torque of clutch $T = W \mu Rn$

but $W = P - KT$

$$\therefore T = (P - KT)\, \mu Rn$$

$$\therefore T = \frac{P \mu Rn}{1 + K \mu Rn} \quad \text{or} \quad T = \frac{P}{\frac{1}{\mu Rn} + K}$$

This indicates that effects of μ have been reduced providing either μ, R or n is large. The maximum value of the slipping torque would occur if μ=infinity, in which case $T = \frac{P}{K}$.

These advantages can be seen in Fig. 12 in which the curve for a rubber-controlled torque limiter is superimposed upon that for a conventional friction clutch.

The use of rubber as the control medium has the effect of increasing the slipping torque slightly under conditions of rapidly rising torque, since rubber is effectively stiffer under these conditions. This is an advantage since, if the clutch is set to slip without stalling the engine, it would cut off all the peak torque, above the maximum p.t.o. torque and thus reduce the average power transmitted. This device reduces the peak torques and only slips if these are maintained for more than a very short duration.

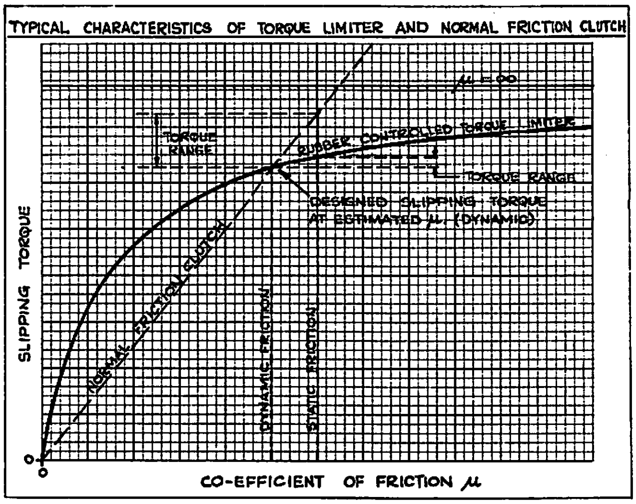

Fig. 12
Graph of torque limiter characteristics

Guarding of Universal Joints

A further point to which the implement designer must pay careful attention is that of the safety guarding of the power take-off shaft. Existing regulations in the U.K. require that the power take-off shaft be covered over its length by a guard which may revolve with the shaft, but which must stop on contact.

Some European and Scandinavian countries require that power take-off shaft guards remain stationary.

Conclusion

Mechanical power take-off to implement drives using universal joints are by far the most widely used implement drives. In recent years hydrostatic drives have been introduced on implements such as fertilizer spreaders and implements requiring low powers. The extension of the hydrostatic drive to higher power implements is probable, due to the flexibility of the system from the implement attachment point of view, but cost of such a drive system is at the moment very high where high powers and high transmission efficiencies are required.

It would appear that for some time to come the transmission of power from the tractor power take-off to implement, will largely be carried out by means of universal joints. The introduction of constant velocity joints which can be used on agricultural implements, will be a considerable step forward and will ease the burden on the power take-off driven implement designer very considerably.

DISCUSSION

Dr A. R. Reece (University of Newcastle-upon-Tyne) expressed interest in the graph showing large compressive and tensile loads on the power take-off shaft. He said that it had been suggested that this was due to the relative motion induced by the implement being lifted up and down. He considered, however, that if the two ends of the power shaft and the virtual hitch point of the linkage were in line the movement up and down would cause no outward movement of the shaft; if the last joint was above that line then it would cause a tensile movement when it went up—whatever happened going up was equal to what happened coming down. Mr Howard said that in the tests conducted by his company, where the power take-off on the tractor was above the input shaft of the rotavator the load was normally compressive and *vice versa* when the shaft was the other way round. This was not necessarily always the case but in normal work where there were no tremendous shock loads this appeared to happen. When the blades of the rotavator entered the ground they tended to lift the machine a very small amount as each blade struck the ground, harder working conditions increasing the lift. When the machine went up the lift was at the point where the blades were trying to penetrate the ground and that was the point where most torque was being transmitted; once the machine started coming down again the torque was relieved on the shaft to a certain extent with the effect, as shown by test, of releasing the friction to a certain extent on the universal joint. Thrust curves that went past zero point were unusual. A joint that was working absolutely straight and which did not move should not put any thrust on the power take-off shaft at all. He could not offer an exact explanation as to why the thrust was produced only in those curves but related this phenomenon to the fact that the torque was reduced on the shaft when the implement started to descend.

Mr P. H. Bailey (National Institute of Agricultural Engineering) made reference to the Institute's experience with power take-off shafts with anti-friction sliding members in research equipment in which the fore-and-aft loading was of particular importance. The limiting axial loading was approximately proportional to the torque and was of course much greater in ordinary square or splined shafts. Units with ball bearings rolling on the shafts had been made at the Institute as commercial units were at an early stage of design and were expensive. He asked if there were any forthcoming practical applications which could affect the price.

Mr Howard said that one of the first applications would be on rotavators, particularly the higher powered types. Four different types of frictionless joint were at present on test, all of which were giving good results and reducing the thrust on power take-off shafts produced with rotary cultivators. Frictionless shafts would soon be introduced on the bigger type of rotavator and the present intensive development meant that the price of the shafts was already much more realistic than 12 months ago.

Mr Howard agreed with Captain E. N. GRIFFITH (Rotary Hoes) that it was possible for the power take-off shaft to have to transmit a greater horsepower than the engine; instantaneous powers had been recorded with rotary cultivators of up to three times the normal maximum output of the tractor power take-off. This could occur when there was a rapid deceleration of the rotor shaft on the rotary cultivator and the back-up of the inertia of the tractor engine went through the transmission of tractor and rotavator to overcome the jamming load which had caused the deceleration. In normal agricultural conditions the maximum peak torque or peak powers going through power take-offs with a rotavator was on the average 1.8 times the normal output torque of the tractor power take-off. Under certain conditions Mr Howard continued, for instance in forestry work, for cutting of firebreaks etc. and land clearance work, the figure could reach three times the normal output of the tractor power take-off.

MR J. H. W. WILDER (PRESIDENT) discussed the revised British Standard recommending that power take-off speeds should be in the vicinity of 540 rev/min. This was very satisfactory in the case of the rotavator, but he wondered whether manufacturers of implements that were attached to the rear of the tractor and used to absorb very little power were encountering difficulties as a result of the modification of the British Standard. Mr Howard agreed that the specification that the power take-off speed of 540 rev/min should be within 80% of the maximum tractor speed was a great advantage to the rotavator. Near maximum tractor power was now obtained at normal power take-off speed, whereas in the past tractor power take-off speeds had varied a great deal in relation to the point where maximum power was given; it had been almost impossible to cope with all the speed variations.

MR J. BURNS (International Harvester Co Ltd) invited Mr Howard's views on the secondary British Standard specification of 1,000 rev/min. Mr Burns considered this specification was better suited to tractor transmissions and joints etc., throughout the whole transmission train. Manufacturers of power-driven machinery, however, appeared reluctant to accept this specification. Mr Howard agreed that the 1,000 rev/min power take-off presented problems to machinery manufacturers particularly if slow-moving shafts were required on the implement or machine, necessitating a much greater reduction in speed from the tractor power take-off shaft to the operating mechanism of the implement and thus adding to cost. It was normally possible to make economies, however, by using lighter universal joints, because of the higher speed at which they ran. Referring to the introduction of 1,000 rev/min power take-off in the United States Mr Howard said that the problems initially envisaged in running universal joints at that speed had not materialized. From the point of view of the rotavator 1,000 rev/min shafts must be welcomed and would be utilized as far as possible.

MR R. A. JOSSAUME (Cleales Ltd) asked Mr Howard to comment on the future of power take-off drives in view of the introduction of the hydraulic motor into agriculture. Mr Howard said that he could not envisage an immediate solution to the difficulties of transmitting high powers from the tractor engine to an implement by hydrostatic means. The design of transmissions to rotavators required an overload allowance of at least 1.8 times normal tractor power. In the case of a 50 hp tractor, hydraulic pumps and motors carrying 90 hp would be required, thus carrying the cost far beyond that of universal joint methods. His company had built rotavators with hydrostatic transmission, replacing £15 of universal joints by approximately £1,000 of hydraulic equipment. He believed that there was much more scope on tractors for sophisticated transmissions like the hydrostatic unit than on implements, where the cost had to be kept relatively low. Admittedly some implements already employed a hydrostatic transmission—usually in the form of a gear pump and a gear motor—but these were implements such as fertilizer distributors which required relatively low power and could accept the power at a relatively high speed. For these reasons he believed it would be a long time before mechanical power take-off disappeared. Mr Jossaume then drew attention to the advantages of a hydraulic motor produced by a tractor manufacturer that could be used with a range of implements, thus satisfying the requirements of both low and high horsepower implements. Mr Howard while not doubting that many more implements would be introduced employing a hydrostatic drive and the normal hydraulic pump of the tractor, did not envisage that a manufacturer would provide an additional pump, in, say, a 50 horsepower tractor, giving a hydraulic horsepower output of approximately 50. Such a step would be most expensive and although the mechanical power take-off shaft—the universally-jointed shaft—caused many mechanical problems, it served a useful purpose at a reasonable cost. He would thus predict that its future was assured for some time to come.

MR A. BLOOMFIELD (Plant Protection Ltd) asked whether, in view of much higher power output of tractors and the increasing width of implements, torque-limiting safety device were designed purely to protect the transmission and gearing in the implement or whether it was possible to protect individual implement tines. Mr Howard said that there was much discussion of this question at the present time. In the case of a rotavator it was difficult to protect individual tines; blade bolts were provided to bolt on the blades of the machine and these could normally be expected to shear before the blade broke. Protection of the transmission of the rotavator had generally been limited to the expensive transmission components. A replacement for a broken blade was not very costly but it was obviously preferable that the inexpensive bolts should break before the blades. This was difficult to achieve however, not only because the balance between the strength of the blade and the strength of the bolt was very fine but also because of production tolerances.

In response to a request from MR H. FAIL (University

(continued on page 37)

TYRES MOBILITY AND THE TRACTOR

by

W. Wortley, B SC (MECH ENG), AMI AGR E and E. G. Dean*

Presented at an Open Meeting of the Institution at the National College of Agricultural Engineering, Silsoe, on 28 September 1965

The paper examines the inter-relationship between the soil and the tyre and its influence on mobility followed by a description of current tyre designs and their properties. Characteristics of traction aids, tracked tractors, two and four wheel drive tractors are discussed and the paper is concluded with consideration of soil consolidation effects.

Intoduction

The agricultural tractor is becoming more and more complicated as time goes by with sophisticated power take-offs and hydraulic drives, etc. for manipulating implements, but these are all useless if it does not have the ability to travel over all types of terrain. The final, but essential, item in the transmission train is the wheel with its tyre.

The pneumatic tyre was first used in agriculture for the wheel barrow. Work carried out in 1929 showed that a load of 400 lb could be moved with comparative ease over uneven surfaces whereas a 200 lb load was the limit when using a barrow fitted with an iron tyred wheel.

Later further tests were carried out on pneumatic tyre equipment for farm carts. The tyres were of smaller diameter than the standard iron tyred cart wheel and again a vast reduction in draught was obtained. Increased pay load varying from 35% to 100% could be pulled by a horse depending on ground conditions.

From these beginnings the vast range of sizes and types of tyre for the farmer has grown.

Traction in Soils

Traction on the highway does not present the same problem as off the highway because the road surface has high strength and when dry, a high coefficient of friction exists between it and the rubber compounds used for tyre treads. Modern efficient tread designs wipe away any lubricants, the most common being water, to maintain dry contact.

Travel on soils is quite different as the strength of the soil is much lower and instead of breakdown occurring between the tyre and the ground surfaces it often takes place within the soil itself. Tread designs for off-the-road duty must therefore bring into play the maximum quantity of the low strength soil to enable high tractive forces to be generated.

There are two basic types of soils—frictional and cohesive. A frictional soil consists of relatively large particles which are quite discrete and have no form of cementation binding them together. An extreme case is wind blown sand. Cohesive soils have much smaller particle size and the particles tend to stick together. These form the clays. Most agricultural soils have a mixture of these two types blended to different degrees to give the various known classifications of loams.

To obtain purchase on soft soil it is desirable to interlock the driving member with the soil. When a bar is thrust into the soil and horizontal force applied to it, before maximum shear resistance is obtained relative movement between the shear member and the soil must take place. The resistance increases with displacement to a maximum and then falls away with a cohesive soil. In a frictional soil it builds up to a maximum and then remains constant. In a frictional soil the shear resistance varies with the vertical load, whereas in a cohesive soil it is dependent on the area of the shearing surface. The loams are a mixture of frictional and cohesive soil and will therefore have a shearing resistance which varies with both loading and area.

Since soils do not have high strength there is a tendency for the tyre to sink in forming a rut and hence create resistance to motion. With the solid wheel there is no means of adjusting the contact pressure between the rim and the soil and the wheel will sink in until sufficient area of the rim is in contact with the soil to support the load. The amount of penetration of a tyre depends upon the nature of the soil, the load carried by the wheel and the contact pressure of the tyre. In the first approximation, the average contact pressure of a tyre is equal to the inflation pressure with a small addition due to the structural stiffness of the tyre. On hard unyielding surfaces the deformation will take place wholly in the tyre. If the ground supporting pressure is appreciably lower than the tyre contact pressure, the tyre will behave as a solid wheel. Between these limits deformation will be shared to varying degrees. Furthermore the tread surface is highly flexible and will readily deform to absorb small irregularities in the soil surface.

Tyre pressure cannot be reduced indiscriminately as there is a limit to the deformation of the tyre casing for satisfactory structural life. Rear tractor tyre pressures range from 12-26 lb/in^2. Section widths of approximately 9 in. to 15 in. are most commonly used. These features, coupled with adequate load carrying capacity, require a

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tyre diameter of four to five feet. The high ratio of diameter to section width has other advantages as explained later.

Rolling Resistance

The rolling resistance of the tyre is built up from the inherent rolling resistance of the tyre itself and the work done in compacting the soil and forming the rut. A tyre is not completely elastic and work must be done in flexing the sidewall as it rolls along. This may amount to 30 lb per ton of vertical load. Incidentally, this damping effect is useful when ride characteristics are considered as the tyre is usually the only springing medium on the tractors. Work done in overcoming soil resistance is usually greater, being dependent on the nature of the soil, and can become more than 400 lb per ton. Minimum rolling resistance will be achieved where the deformation is confined to the tyre and none takes place in the soil.

The depth and width of the rut formed is a measure of the work done on the soil. Increased load-carrying capacity for the same rut depth requires either an increase in tyre diameter to give a larger contact length or an increase in tyre width. With the former, rolling resistance will remain the same as the rut width is unchanged but it will be increased for the latter. The larger diameter tyre is therefore the more efficient as rolling resistance expressed as a percentage of load is reduced. Wider tyres can however be beneficial if by their use, rut depth is reduced.

Tread Pattern Design

Tread patterns for off-the-road tyres must bring into play the maximum volume of soil. At first thought a transverse bar would appear to be best but it has the disadvantage that once the soil is sheared it tends to remain packed between the bars, particularly when the soil is sticky, and no further penetration is obtained the next time the bars are ready to enter the soil. On hard surfaces the ride will be very bumpy. A diagonal bar is better, as there is a tendency for the soil to be ejected at the side of the tyre but it is not completely satisfactory as there is a side force created, which will tend to push the tractor sideways. A combination of two diagonal bars to produce a chevron overcomes all these objections and is now generally accepted as being the most suitable design. The bars are deep to penetrate and key in with the softer soils; to penetrate through a soft upper layer to obtain purchase on firm underlayer; to wipe slippery surfaces clean.

Tyre Development and Testing

Tyre development is a very lengthy process and starts with new ideas being converted into fact by hand cutting the tread design in a plain treaded tyre. If these tyres prove promising in intial screening tests, a mould is prepared to provide a quantity of tyres, which are placed on farms throughout the country with varying soil types and methods of cultivation. Only after a successful output test of this nature is full production entertained.

Accurate road wear measurement is not easy because of the many variables which can influence the result such as ambient temperatures, proportion of wet weather, driver technique, differences between tractors, etc. Two identical tractors are therefore used, which are run in tandem around typical country lanes. During each tyre test the variables are smoothed out by systematic alteration of tractor position i.e. leading or trailing and changing drivers and tyres from machine A to machine B.

The user will be the final judge of the tyre and for field traction tests we have simulated field ploughing as it usually requires the highest drawbar pull under the most difficult soil conditions in autumn and spring. Again two tractors are used, the leading tractor is equipped with the test tyres and the drawbar load is applied by the trailing tractor which has a mounted plough. The drawbar load is measured by a dynamometer which is interposed in the chain link between the tractors. Drawbar pull can be varied by manipulation of the engine controls of the rear tractor. All data, such as wheel revs., distance travelled and drawbar pull are recorded and from this information drawbar pull versus slip graphs are plotted.

As tyre manufacturers, we are primarily interested in comparing sets of tyres. No attempt is normally made to measure the rolling resistance of the front wheels and it is assumed constant for each test. All testing is carried out in natural conditions. Since soil conditions do vary with moisture content, a cloudy wind-free day is most suitable. Nevertheless to be certain that soil conditions have not changed, a standard tyre is used before and after all tests and the performance of this tyre must be unchanged before the results of the test tyre are accepted.

Slip is expressed as a percentage which is defined as follows:

If the distance travelled by the unladen tractor per wheel revolution is X and the distance travelled per revolution under drawbar pull is Xp then X—Xp is the slip distance and

$$\text{percentage slip} = \frac{X - Xp \times 100}{X}$$

It is often useful when comparing results on different tractors and/or tyres sizes to express the results as a coefficient of traction which is the ratio $\frac{\text{drawbar pull}}{\text{vertical axle load.}}$

The vertical axle load should include any weight transfer and not simply static axle load.

Typical traction coefficients are given in the following Table.

Soil	*Coefficients of Traction*	
	30% Slip	*Max. Sustained Pull*
Clay loam *very wet*	.41	.51
Light stoney loam *moist*	.49	.55
Clay loam *very moist*	.55	.62
Sandy loam *dry*	.55	.70
Medium loam *moist*	.66	.76
Stiff clay *dry*	.65	.79
Tarmac *dry*	—	.95

Tread Pattern Types

The outcome of test and development has been several types of rear tractor tyres.

For heavy soils a deep bar pattern with an open pitch is essential to make best use of the cohesive properties of the soil.

When operating in pure sand the deep bars become an embarrassment since, if excessive wheel slip is encountered, the bars will tend to scoop out the sand under the tyre and it will become bogged down. A very simple ribbed design of shallow pattern dimensions backed by the most flexible case gives optimum performance.

Again when travelling on hard roads the relatively high bar contact pressure of the deep bars, together with their tendency to flick back when moving out of contact, produces wear.

Tyre road life is increased by up to 50% in this design which has a greater number of bars of reduced height. The bar area has been re-distributed to give an increased pattern density at the centre of the tyre. Its tractive performance in the heavy clays is not as good as the deep bar design but in sandy loams it is certainly equal.

In recent years the agricultural tractor has been adapted for many industrial and earthmoving duties and a further tread design has been developed specifically for this duty. Road life is approx. 100% greater than the agricultural tyre and the bars are very robust to withstand the arduous work involved and provide high tractive effort.

Casing Construction

Cross ply tyre casings are made up of layers of rubbered cord plies which are laid at an angle to the circumferential centre line of the tread and each consecutive layer is laid at the opposite hand to the adjacent ply. The plies are anchored by passing them under a bead coil of many wire strands.

By far the greater proportion of tractor tyres have the above cross ply construction but some are now made in radial ply.

The main casing plies in the radial ply tyre run at a right angle to the tread centre line. Under the tread of the tyre the casing is reinforced by several layers of cord in cross ply form. The reinforcement is made very rigid transversely by laying the plies at a small angle to the tread centre line.

The radial ply tyre has different characteristics to the cross ply. One of the main advantages is an improvement in tread life. To examine to what extent this feature may apply to tractor tyres we can draw some conclusions from on-the-road tyres. On cars the tread life advantage can be up to 70%, in truck duty approx. 30%. Tread wear on-the-road takes place largely when cornering and is proportional to the square of the speed. If we now consider cornering speeds, representative values for cars are 50 mile/h, for trucks 30 mile/h and for tractors 10 mile/h. The tread wear advantage to tractor radial ply tyres is likely therefore to be small. This conclusion has been supported by controlled experimental work by my company.

Deflection and contact patch characteristics of the radial ply tyre give a small tractive advantage in sandy conditions but tread bar design is the more important feature in the heavy clays and this advantage is not maintained. To some extent in very sticky soils the rigidity of the tread tends to prevent good self-cleaning action.

Oversizing

Oversizing may be effected in two ways.

(*a*) By increasing section width and diameter, maintaining rim diameter constant.

(*b*) By increasing section width and reducing rim diameter, maintaining overall diameter approximately constant.

Obviously method (*a*) will alter gear ratios and tractor geometry but as a new rim is not absolutely necessary it is frequently used.

If there is no change in inflation pressure and/or load, traction will not be increased. The smaller tyre actually will have the advantage on account of its lower bulldozing resistance and the greater penetration of the smaller bars.

Oversized tyres have a greater load carrying capacity and where ballasted to their maximum will give a traction increase in proportion to the added weight above that of the standard tyre. Alternatively pressure reduction, within the permissible limits, will improve flotation. A reduction from 18 to 12 lb/in^2 on a very moist stubble on clay loam gave an increase of 10% in coefficient of traction.

In conditions promoting rapid tread wear, such as flint-containing soils, oversizing without weight additions may be considered to prolong tyre life.

Traction Aids

Numerous traction aids can be used to increase drawbar pull such as ballasting—by liquid or solid weights, tyre chains, girdles, strakes, half tracks and twin tyres.

The effects of these have been investigated by Southwell[1] and from his paper it can be seen that strakes are most effective in clay soils and the half track for sandy soils. For the other combinations the increase in pull produced is roughly proportional to the increase in axle weight.

One further aid which has now become a standard fitment is the differential lock. Without a differential lock the maximum drawbar pull available is twice the traction of the tyre with the lowest tractive force. On firm uniform conditions, therefore, engagement of differential lock will not give great advantage, but in conditions where there is wide variation in the ground surface e.g. ploughing with the land wheel on a wet slippery surface, marked difference will be available.

Tracks

Within any vehicle size the track can provide a greater contact area with the ground than any wheeled machine. The contact area has a long narrow aspect. These features combine to give an improvement in drawbar pull in frictional soils due to the lower motion resistance and in cohesional soils due to the increase in soil shearing area.

Steering of tracked vehicles is commonly achieved by braking one track which is very wasteful and the full increase in drawbar pull is only available in straight ahead travel.

A tracked machine is essentially a low speed tool because track inertia produces destructively high forces at quite moderate speeds.

Mobility between sites may be restricted due to the damage caused to permanent roads by the grousers.

Noise level is high and ride poor.

While the tracked machine has the highest drawbar pull capability the more rapid methods of cultivation now being developed together with some of its disadvantages mentioned above are tending to restrict its use to special applications.

Front Wheel Drive

With the single axle drive tractor, the rolling resistance of the front tyres must be overcome by the tractive force produced by the rear wheels, which diminishes the useful effort for implement haulage. As drawbar pull is increased load transfer will take place adding weight to the rear driving wheel and reducing that on the front steering wheels. Complete transfer of front axle weight to the rear driving axle is undesirable as steering control will be lost and there is the danger of the tractor overturning. A proportion of the tractor weight and the contact area of the front tyres is not therefore used to provide tractive force.

Four wheel drive eliminates the loss and provides a more efficient machine but to make best use of this configuration, weight must be re-distributed. Preferably tyres of equal size, front and rear, should be used and at designed working drawbar pull the total weight should be distributed evenly on all tyres. This means that in the bare tractor condition the major proportion of the weight must be on the front axle, which is the antithesis of the single driven axle machine. In addition to the four wheel drive tractor having the capability of a higher drawbar pull, it will also be able, at the lower ranges of pull, to operate with a lesser wheel slip than the comparable two-wheel drive machine. Directional control of the machine may be achieved

(*a*) by skid steering i.e. braking one side of the machine in a similar manner to a tracked tractor. This method has the same disadvantages as apply to the tracked tractor.

(*b*) by turning the front wheels in the conventional manner.

(*c*) by pivot steering where the machine is hinged at a point between the front and rear wheels. Centre pivot steering has the advantage that the rear wheels follow in the same track as the front wheels.

Soil Consolidation

Inflation pressures of agricultural tyres are low and their compacting effect is small. For the majority of conditions the compaction effect is not sufficient to retard growth but a reduction in crop yield can occur under certain conditions.

The inter-relationship between plant and soil is affected by such factors as air voids, oxygen content, mechanical impedance to root penetration and moisture movement. Although it is difficult to isolate the reasons for these effects, Bateman[2] has concluded that crop growth may expect to be retarded when the air voids at maximum soil moisture content are near the 10% value.

Tractor tyre traffic at normal loads and inflation pressure (around 12 lb/in^2) has been observed to reduce the air voids to the critical 10% and vulnerability of the soil increases with moisture content as it acts as a lubricant to allow the particles to slide into closer relationship. There is some indication that susceptibility to compaction increases with clay content. It was shown on a silty clay loam that once air voids were reduced to 10% this proportion was not increased during the process of drying out.

An experiment where deliberate compaction of a silty clay loam by

(*a*) two passes of the tractor tyre over the whole plot after drilling and two passes between the rows after last cultivation

(*b*) two or four passes in the furrow bottom plus compaction of the land wheel on top prior to turning furrow

resulted in a reduction in yield averaging 12 bushels per acre.

A further effect of the tractor tyre is a result of the slip which is inherent in its tractive properties. The sliding motion between the tread surfaces and the soil can in wet conditions induce a smearing action, particularly on cohesive type soils, which is thought to seal off the soil pores and therefore restrict the vertical movement of soil air and moisture. The effect could be most pronounced in a furrow bottom.

Since the effects of tread compaction are dependent upon soil condition the first safeguard must be to ensure that crop cultivation is not undertaken at a time when weather conditions gives soil moisture contents which will allow consolidation to the 10% air void level. In difficult weather close attention must be given to inflation pressure to ensure over-inflation does not aggravate the situation. For critical soils, oversizing should be considered. There is no advantage, however, for oversized tyre equipment unless the tractor axle weight is such that the wider section tyre can operate at a much lower inflation pressure than the smaller sections of the same diam. Otherwise, the user is covering more ground with the wider sections and effectively increasing the area of compaction.

Acknowledgement

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DISCUSSION

Mr P. H. BAILEY (National Institute of Agricultural Engineering) recalled earlier Papers presented by Mr A. Senkowski on wheel performance and by Dr A. R. Reece on the theory of off-the-road locomotion, and welcomed the present contribution from tyre manufacturers. In relation to a statement by the authors that the effective ground pressure of a tyre was equal to the inflation pressure plus a small addition due to the structural stiffness of the trye, Mr Bailey suggested that for conventional tyre construction the addition could be quite a large one, as indicated by the results obtained on hard ground by Dr Söhne. He asked if Mr Dean and Mr Wortley had additional evidence on the softbed character of soils to show that the addition was only small. Mr Bailey queried the definition of slip distance employed in the Paper and said that in tractor testing it was normal (as a useful convention) to take the average of the free running distance and the towed distance in order to eliminate as far as possible the effect of the rolling resistance of the front wheels which gave a force which produced a small amount of slip in the 'free running' condition. Turning to strakes, Mr Bailey said that in work conducted at the NIAE on a single wheel tester conclusions had been drawn similar to those of Mr Southwell, i.e., that strakes were at their best in heavy soil. Strakes always caused an increase in rolling resistance but on heavy soil the increase in traction outweighed the increase in rolling resistance. On sandy soils it would seem that there was much to be said for pure ballast, as the rolling resistance produced by strakes was considerable while the increase in traction was only proportionate to the weight.

In reply Mr WORTLEY agreed that the effect of the structural stiffness of the tyre could have a not insignificant effect on the ground pressure, generally between 5 per cent and 25 per cent depending on the construction of the tyre. On the question of slip definition he believed that the ideal way was to obtain the mean between the towed and the driven wheel and thus find the exact effective diameter. In the case of tyre testing to determine the differences between variations in tread or casing design, the method of measurement of slip distance was not normally significant. It was most important that the method should be the same for all tyres in the test to obtain a valid comparison of performance properties. Mr BAILEY expressed his interest in the authors' comments on their experience of road wear with radial-ply tyres. He suggested that if a tractor was used in a hilly district where transport work involved a drawbar load there was a possibility of better road performance with radial-ply tyres. NIAE results had indicated that the sideways force characteristics of radial-ply tyres were parallelled by their tractive force characteristics—i.e. just as there was later but sharper breakaway with radial-ply tyres on a car there was likewise better retention of tractive force and therefore less slip at a given pull, producing less roadwear with tractive force conditions. With radial-ply tyres it seemed that on hard ground the tractive force coefficient was comparable with the sideways force coefficient in that both differed in a similar way from the comparable coefficient obtained with conventional tyres. Finally he invited the authors' comments on the difficulties of separating out differences in performance as between tyres of slightly different tread design.

Mr Wortley said that radial-ply tyres were advantageous where high speed and high side forces were being generated. In tractor operation, however, speeds were normally low so that the same degree or proportion of sideways forces could not be generated and thus the benefit from the radial-ply tyre was not as great. He would agree with Mr Bailey however that on hillside work where a greater proportion of sideways force was produced then the differences would tend to be somewhat greater than in normal operation. The investigation of small differences between tyres was problematical. Traction testing was both difficult and wasteful in that experiements could be conducted with a variety of soils and differences might not emerge. Knowledge of particular soils was valuable in traction testing in that positive results could be obtained by waiting for moisture conditions in which traction differences could be measured. He believed it was important to work in natural soil conditions in order to take into account the effect of root structure and similar factors which would not be present in an artificial soil.

Mr K. M. THOMAS (Goodyear Tyre & Rubber Co. Ltd.) invited the authors to comment on the angle of the lug bar in relation to traction and to the adhesive properties of a tyre. Mr Wortley said that shearing and packing of the soil was at a maximum with straight transverse bars whilst the tendency to pack was at a minimum with circumferential ribs. In between these extremes he felt there was a general area in which it was possible to achieve a blend of maximum traction and minimum road-wear.

Mr R. A. JOSSAUME (Cleales Ltd) referring to the authors' statement on soil compaction asked if it was in fact correct to assume that when, say, 2000 lb in weight was added to the wheels the compaction remained the same. Mr Wortley said that he felt the tyre was a self-compensating device and if the load was increased, the area of contact would be extended, more soil would be encompassed and in that sense loading would not affect compaction. On the other hand when an area of soil was being subjected to a total load there must be some small effect, but nothing like the effect there would be in the case of a solid wheel, with which much higher point loads would be possible.

Mr J. V. FOX (Bomford & Evershed Ltd) asked for further information on the question of reduced tyre pressures. It appeared that a lower pressure under soft ground conditions produced the three distinct benefits of greater traction, reduced rolling resistance and less harmful consolidation. If this were in fact the case still lower tyre pressures would seem to be generally beneficial. Was any work being done to produce tyres capable of being operated at still lower pressures?

In his reply Mr Wortley said that assuming that any tyre was capable of giving a certain horizontal force and that this horizontal force was reduced by rolling resistance and certain other factors, there was thus a definite correlation between traction and rolling resistance. If tyre pressure was reduced, rut penetration would be less with the result that rolling resistance would be in turn reduced. The lower the pressure exerted on the soil the less was likely to be the consolidation. With regard to the availability of tyres operable at lower pressures this could be achieved within certain limitations through oversizing. However, cost was a limiting factor as a significant drop in inflation pressures would alter the footprint area and require more tyre. For this reason there had been little incentive for further investigation into the provision of lower pressure tyres.

Mr R. A. Jossaume (Cleales Ltd) said that in practice the use of oversize tyres was governed by the width of furrow. A furrow width of more than 12 in. was not normally acceptable under heavy land conditions, i.e. where problems of traction arose. Thus 12 inches seemed to be a maximum width for the cross-section and if the wheel diameter was increased then the geometry and hydraulics of the tractor unit would be thrown out of balance.

Mr Wortley accepted that there were these limitations but said that in many overseas markets much larger section tyres were in common use, perhaps because of the methods of cultivation employed. It would not be appropriate for him to quote a hard and fast rule regarding oversizing, which must vary according to the particular job.

Mr T. C. D. Manby (National Institute of Agricultural Engineering) said that experienced tractor operators knew when to reduce inflation pressures in difficult conditions. The question was what minimum pressure could safely be suggested by a manufacturer in operating instructions; even so, there was a risk of the operator going below the figure suggested. Even assuming that operators would not reduce pressure below 8 lb/in^2 it had been shown from recent NIAE experiments that although this pressure was generally very satisfactory, a very heavy mounted implement on heavy land would cause over-deflection where all the weight was carried by the tractor. Approximately 20 years ago 100% water filling of tyres was recommended specifically so that tyres did not over-deflect when greater weights acted upon them. 100% water fill should still be recommended in areas where there were difficulties with heavy mounted equipment. In conclusion, Mr Manby wished to amplify previous statements concerning the very complex situation underneath the tyre tread by emphasizing that the pressure immediately below the tread bar differed from that near the surface and there were problems of determining the pressures exerted on the relevant tread-bars.

Mr Wortley commented that the addition of loads to 95% or 100% water-filled tyres resulted in lower deflections than would occur on air-filled tyres owing to the increase in inflation pressure which occurred. This increase in pressure would apply greater stressing to the casing cords.

Mr R. Downs (International Harvester Co Ltd) said that from the farmer's standpoint performance and cost were the significant criteria in the context of tyres. He drew attention to the introduction by Dutch tyre manufacturers of a relatively inexpensive new type of tyre for use in greasy conditions. This had a widely spaced lug-bar and was probably the modern successor to the old steel lug type of wheel still favoured by some farmers for use in greasy wintertime conditions. There did not appear to be a comparable tyre in this country. Secondly, with regard to flinty conditions, many farmers were concerned that as tractor performance was improved and power outputs and weight increased, as was the case with some of the bigger tractors now imported from the U.S.A., high expenditure on tyre repair was occasioned through damage by flint and stones. He would welcome information on work being done to improve the component materials of tyres and asked if the tyre walls were in any way being improved by the use of appropriate modern materials.

Mr Wortley replied that the very deep-treaded tyre produced in Holland had been in use for many years overseas in rice and paddy fields. There had not been sufficient demand in the U.K. to justify production of a very deep lug tyre for home use. Whilst he would agree that this type of tyre was certainly useful in clay soils, it would not be useful on sandy soils because of its deep bars which, once slipping began, would dig the sand straight out. One of the problems in tyre manufacture was to have a suitable compromise in properties to suit most conditions and this 'cane-and-rice' tyre was too near one end of the scale to be universally acceptable. In order to provide a tyre which would produce a significant uplift in wear in flinty conditions, more material would have to be used to strengthen the side walls and the tread. A tyre of this type could certainly be made but only by increasing costs. A closer-pitch tyre of oversizing could be beneficial in flinty conditions, but improvement of the wear characteristics of the same size tyre introduced a cost factor.

A final question related to the difficulties of combine-harvester operation and it was suggested that larger size tyres should be used. Another solution proposed was the use of double wheels. In reply Mr Dean said he believed that narrow-section tyres on earlier combine models had created problems some years ago but he felt that this had now been solved by fitting tyres of 15×30 or 15×26 section. The practical advantages of twinning were probably also worthy of investigation.

MECHANICAL POWER TRANSMISSION BETWEEN TRACTOR AND IMPLEMENT

by L. H. FREEMAN*

The following paper was presented to a meeting of the West Midlands Branch of the Institution on 1 March 1965. It is now published, following the historical treatment of power transmission in farm machinery by M. T. Sherwen and the general discussion of power take-off problems by Mr J. A. Howard, to provide a detailed analysis of aspects of the subject.

The most convenient method of coupling the tractor output shaft (p.t.o.) to the implement input shaft (p.i.c.) is to make use of the simple Hooke's joint. There are, however, certain problems associated with this, one of which concerns the non-constant velocity characteristics of the joints. I do not propose to enter into a long and involved technical explanation of the reasons for this, but a few words in the simplest terms may well be of interest.

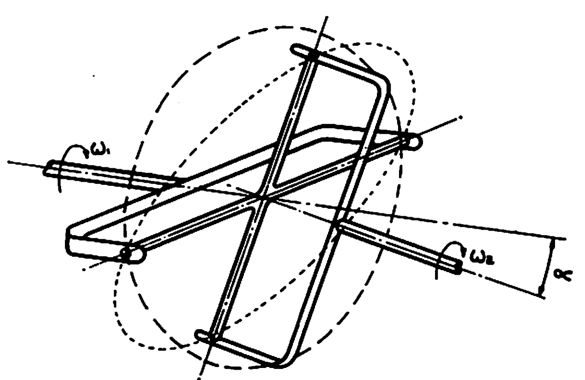

Fig. 1

As indicated in Figure 1, a Hooke's joint consists basically of three members—two yokes and a connecting member, usually in the form of a cross. The four arms of the cross are formed as bearings, opposite pairs of arms being restrained in the yokes. The yokes are thus constrained radially with respect to each other, but are individually free to oscillate about the bearings on the journal cross. Usually the bearings are formed with uncaged needle rollers, although sometimes plain bush type bearings are utilized.

In order that we may differentiate between the two yokes, we denote one as the driving yoke and the other as the driven yoke. The driving yoke is, of course, attached to the p.t.o. shaft of the tractor and the driven yoke is attached to one end of the implement drive shaft, the other end being attached to a second joint with similar characteristics.

The two bearings within the driving yoke rotate in a perfect circle at the same speed as the p.t.o. shaft. When an angle is imposed on the joint, the two bearings constrained in the driven yoke will oscillate as the shaft rotates and this will result in a non-circular path relative to that of the driving bearings. The path traced by the driven bearings, therefore, is elliptical compared with that traced by the driving bearings; and as the period of time taken to complete one revolution is the same in both cases, speed fluctuation must occur between the pairs of bearings. That briefly explains the reasons for non-constant velocity.

This variation of speed can be very undesirable if transmitted to the implement but, as the normal drive line usually incorporates two joints, it is possible to mount them in such a manner that the speed fluctuations of the one are cancelled out by those of the other, thus resulting in near constant velocity on the final output drive. This can only be achieved if the operating angles of each joint are similar and in the same plane. Unfortunately, although standards have been drawn up for the position of the p.t.o. shaft on the tractor, these standards are not yet generally adopted and this results in a wide variation of positions on the tractors available on the present day market. There is a similar lack of standardization in so far as the power input connection to the implement is concerned, and these two factors combined make it almost impossible to arrive at a situation where similar angles on the two joints can be attained in more than, perhaps, one particular combination of implement and tractor. Because of this, the resulting speed fluctuations in the drive shaft are transmitted to the implement and result in vibrations which at the best are extremely annoying and at the worst can be very destructive.

In addition to the problems associated with normal operation of the implement whilst actually doing work there are, of course, special problems arising from the conditions needed for turning at headlands etc. Whilst the amount of power consumed under these conditions is usually small, the angle of the joint is often very large and should there be a great difference between the angles on the two joints, severe vibration can be expected. It should, perhaps, be pointed out that whereas the difference between, let us say, 2° and 3° is very small in terms of speed fluctuation, a 1° difference at 40°, that is 40° on one joint and 41° on the other, can be much more significant.

Another aspect concerning angles, which always has to

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be borne in mind when designing Hooke's joints for agricultural applications, is the problem of storage. When the implement is disconnected from the tractor for storage or transportation, it is desirable that the operator should be able to fold the drive shaft back against the implement, in order to take up less space and avoid damage to the shaft. This necessitates the joint being able to move through 90°, although the joint obviously cannot rotate under these conditions. This requirement for a 90° angle joint results in a long overhang from the end of the p.t.o. shaft or p.i.c. shaft to the centre line of the joint; and consequently additional strength has to be designed into the yoke ears to avoid premature fracture or distortion during the operation.

I now wish to enlarge a little on the problems associated with movements between the tractor and implement and the consequential effect on the drive shaft. Great mobility is required, owing to movement at the hitch due to steering, unevenness of ground etc. This of course applies mainly to towed implements, mounted implements being less of a problem. During operation the tractor and the trailer undergo independent movements and these can occur about both horizontal and vertical axes—horizontal for steering and verticial from uneven ground etc. The result is that the joints are subjected to compound angles operating in planes other than the normal vertical and horizontal and it is, of course, the compound angle which must be taken into consideration when calculating velocity variations. At any given instant during operation it is possible to have completely dissimilar angles on the two joints. Illustrations of this showing the effect of the relationship between the p.t.o. and p.i.c. shafts and the hitching point are contained in Figures 2 and 3.

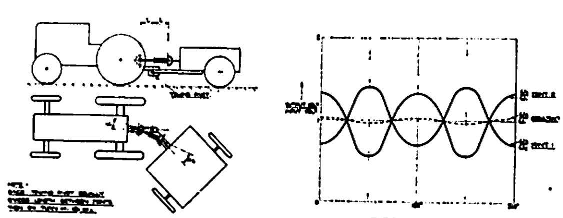

Fig. 2

Figure 2 illustrates an arrangement in which the hitching point lies below the drive shaft; and the p.t.o. and p.i.c. shafts are directly in line both in plan and elevation. The hitching point is mid-way between the two joints and due to the fact that the shaft can telescope, the angles on the joints remain similar whatever angle is contained between the tractor and the implement during turning. A graphical illustration of the velocity variations is shown, the dotted line indicating the effect on the p.i.c. shaft. This, of course, is an ideal arrangement although there will be some variations between the angles due to uneven ground. Let me add that in practice this arrangement appears to be the exception rather than the rule.

Figure 3 is similar, but with the hitch point off centre between the joints. This is rather more common, but results in dissimilar angles during turning. This as you will see from the graph gives more variation at the p.i.c.

Figure 4 shows an arrangement in which the hitching

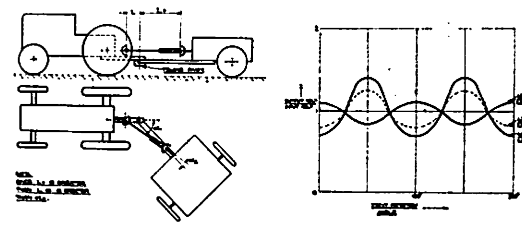

Fig. 3

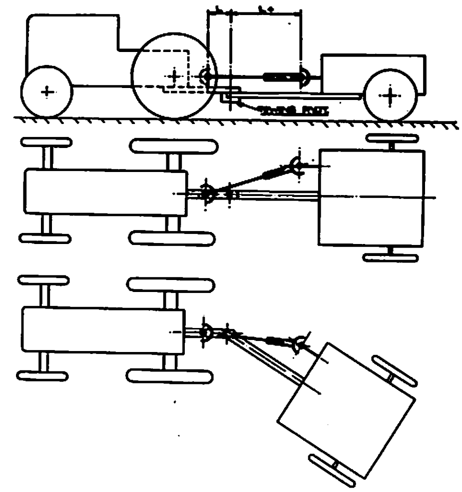

Fig. 4

point still lies below the shaft, but the p.t.o. and p.i.c. shafts are not in line in the plan view, although they are parallel. It will be seen that when taking a left-hand turn, the joint at the tractor end has very little angle whilst the joint at the implement end is working at a far greater angle. When taking a right-hand turn, the opposite is the case, i.e. a larger angle is imposed on the joint at the tractor end, whilst no angle at all appears on the joint at the implement end. A graphical representation would be similar to the previous example, but more pronounced. Here again the position could be either aggravated or assisted by operation over rough ground.

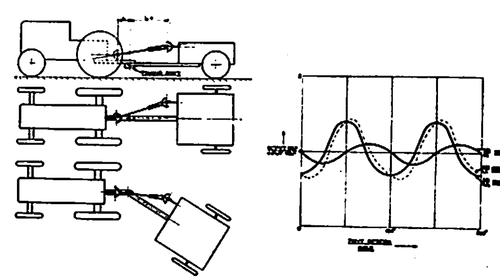

Fig 5

Figure 5 shows a different combination of circumstances with the p.t.o. and p.i.c. shafts offset in both plan and elevation. This can result in plane differences for the angles and the result is shown in the graph. The amplitude of the resultant is greater than the others. Obviously in all these arrangements, it is essential to have some form of collapsible member between the two joints, as the distance between the joints varies continually during operation.

It will be readily understood that the velocity and torque variations resulting from the use of joints at unequal angles will have considerable effect, both on the life of the shaft and the successful operation of the implement. Some indication of how the size of the angle affects the magnitude of these variations is contained in Figure 6, showing curves of velocity variation during one revolution at angles of 15°, 30°, 45° and 60°.

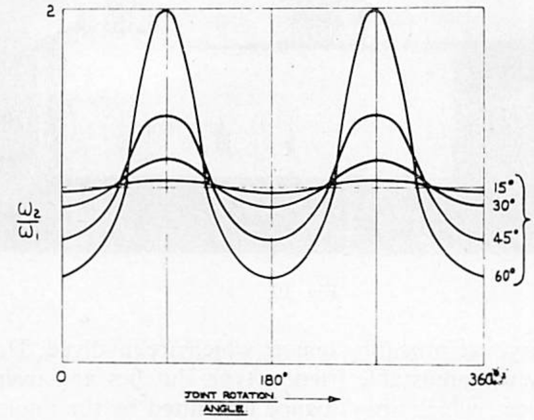

Fig. 6

Before any real steps could be taken in respect of improved drive lines, it was essential to evolve some method of measuring the torque fluctuations occurring in the drive shaft. It should, of course, be pointed out that factors other than unequal angles can create torque fluctuations, among these being the torque fluctuations emanating from the engine itself. In the case of rotary cultivators, there is the difference occurring in the terrain, i.e. varying texture, rocks, tree roots, etc., and, of course, in implements such as manure spreaders, the load is constantly varying. This problem is resolved in the development by Hardy Spicer of torque measuring equipment.

It is therefore possible to obtain torque traces from agricultural implements, of which typical examples are shown in the Figures following and the more important conclusions to be drawn from them are discussed.

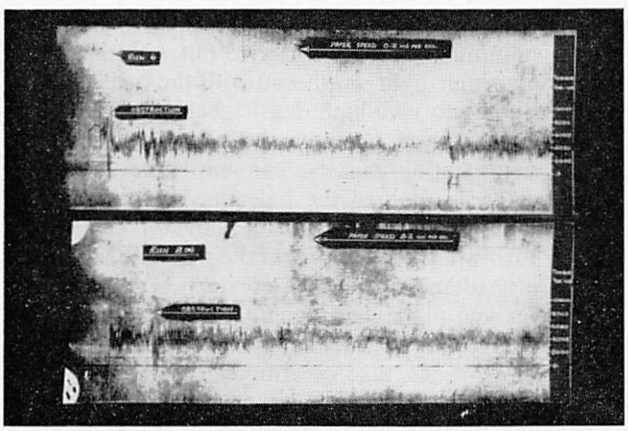

Fig. 7

The first machine (Figure 7) is a rotary cultivator of 60 in. width; this implement imposes some of the most arduous conditions encountered by the universally-jointed shaft. You will see from the traces that it is fairly consistent in amplitude ranging from 1,000 lb-in. to 6,000 lb-in. with a mean of approximately 4,000 lb-in. Obstructions indicated on both the traces, impose torques in the drive of up to 9,000 lb-in. Negative torques approaching 2,000 lb-in. at the point of obstruction are also indicated. These are probably caused by the obstruction lifting the machine clear of the ground and stored energy tending, momentarily, to drive the tractor through the p.t.o. shaft. When confronted with such arduous conditions, the designer's first thought is to install a clutch; but experience has shown that under certain conditions for example, virgin grassland, obstructions can occur frequently thus causing excessive clutch operation and endangering the life expectation of this component. It is most important that the clutch is set at a figure slightly above the normal peak torque of the implement, but below the maximum peaks. Unfortunately, the devices available on the market at the moment are inclined to be unreliable when working within such fine tolerances. It is, therefore, more satisfactory to make the overall transmission sufficiently robust to cater for the more arduous conditions which are likely to be encountered in service.

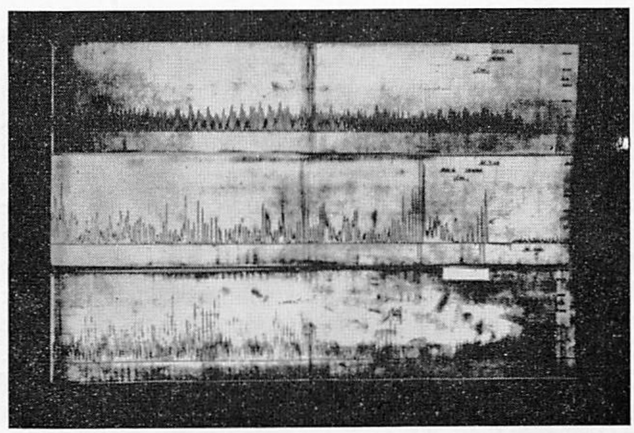

Fig. 8

Figure 8 shows results from a mower crimper combination. The first trace shows the torque demanded by the cutter bar type mower alone, giving torque amplitudes of 1,850 lb-in. maximum down to zero, with a mean figure of approximately 600 lb-in. The second trace shows the torque demanded by the crimper; again this is typical of this type of machine and is the most irregular pattern which I have so far shown. The high positive and negative torques shown at the beginning of the trace are due to the crimper flutes being momentarily out of mesh. The other peaks are created by varying crop densities; the amplitudes being from zero to approximately 3,500 lb-in. with a mean of 920 lb-in. The third trace is of the torque demanded by the combined mower and crimper. An interesting point to note is, that although one would assume that the two separate torques would summate, in fact this does not happen to any significant degree. The mower torque seems to fit in with the crimper and increases the frequency of fluctuation, particularly in the lower part of the range without basically altering the form.

Amplitudes of torque are between approximately 4,000 lb-in. maximum and zero with a mean of 950 lb-in.

The third machine is a manure spreader of the trailer type with shredder at the rear. There are many versions

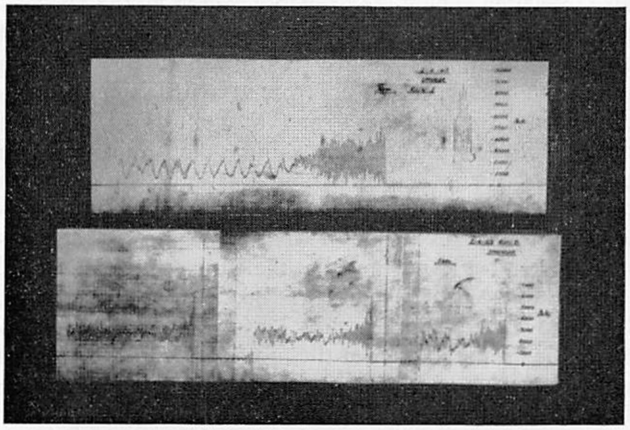

Fig 9.

of these machines on the market at the moment, and it is necessary to take great care with regard to shaft selection, as the power requirement of the different versions can vary considerably. For example, texture of material, degree of shredding, floor speed and capacity of machine are all significant factors. Figure 9 shows traces taken from a typical manure spreader—run one is using the highest floor speed recommended by the manufacturer for spreading with a full load. Run eight—again the highest floor speed with a full load of well rotted manure. In run one, a short length of high amplitude torque of just one second duration was created by attempting a fierce start by sudden engagement of the p.t.o. clutch. With the material packed against the shredders, together with the natural inertia of the load on the floor, there is very high resistance to sudden movement. As a result, the torques recorded peak in one case at 9,000 lb-in. whilst the others are slightly under but approaching this figure. A further crash start was carried out a little later with less marked effect; the machine then settled down to its normal spreading, running at a peak torque of 5,000 lb-in. and the period of instability lasted approximately $4\frac{1}{2}$ seconds. It is obviously most undesirable for the operator to be able to create such conditions and therefore, it is usual to fit some form of ratchet clutch in the p.t.o. shaft, the effect of which is indicated in run eight. Three fierce starts were made and it can be seen that the clutch limited the starting torque to a peak of about 5,500 lb-in. The slip took place for a period of about $1\frac{1}{2}$ seconds on each occasion before settling down to the normal torque. Naturally, the inclusion of the clutch is also a useful means of protecting the machine against overloads caused by obstructions, which also allows the transmission to be lighter, more economic and more in keeping with the normal loads handled by this machinery.

The fourth machine is a baler; this type of implement imposes heavy loads on the transmission system because

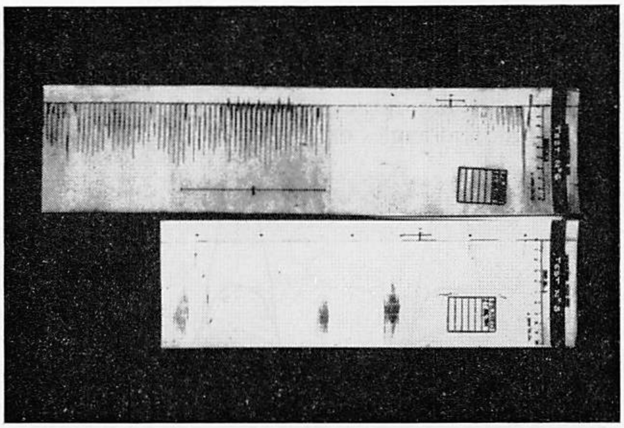

Fig. 10

of the large reciprocating masses which are involved. The majority use adjustable friction type clutches and over-run devices, which are designed and fitted by the implement manufacturers. The object is to prevent damage from overload; and the over-run device is to guard against the possibility of the implement attempting to drive the tractor. Before these particular tests were started, the machine clutch was set to a static slip torque of 4,800 lb-in. Run number three as shown on Figure 10, gives the torque traces obtained during six attempts, to slip the clutch whilst the bale chamber was blocked. On three occasions, namely a, b and e, no slip took place, and the engine stalled. In the other cases, namely c, d and f, slip was achieved. On these occasions the range of oscillation during slip was very extensive, ranging from 10,200 lb-in. down to minus 10 lb-in., the average break away torque being in the region of 6,000 lb-in. Such figures indicate a possible factor relationship between static and dynamic clutch rating, in this case 4,800 to 6,000 or 1:1.25. The three failures to operate the clutch were probably due to a lack of torque surplus from the tractor, relative to the torque required to slip the clutch. Obviously, if such a blockage occurred during normal working, it would be advisable to stop the machine until cleared. A working run shown in number six was carried out with this clutch setting on a single light swath and with the bale chamber tightened down to give high density baling. The torques range from 5,000 lb-in. to zero and the fluctuations caused by the reciprocating masses, can easily be seen. The highest peaks are due to the bale ram and the smaller ones are created by the packer mechanism. The section marked 'turn' was recorded when the machine was taken through a headland turn which created an unbalance of joint angles and planes, producing velocity variations in the drive and having the effect of increasing the magnitude of the peak torques, also creating the instability in the lower sections, while it was also responsible for setting up negative torques up to as much as 1,000 lb-in. A similar test was carried out with a lower clutch setting of 3,600 lb-in. of which the slip test can be seen in test number eight (Figure 11). The same static-to-dynamic clutch factor as before was recorded under these conditions. The oscillation range is still very large, as will be seen from the curves, the decrease from a to b to c being

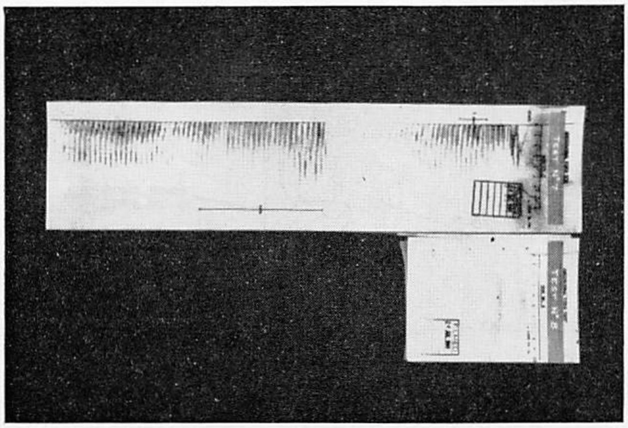

Fig. 11

probably caused by clutch fade. A further run was then carried out with the lower clutch setting on a heavy swath. Here again the average peak torque was 5,000 lb-in. but the effect of velocity variations, due to unequal angles on the joints, is more pronounced than in the other example, which is probably due to their effect on the clutch at these lower settings. A similar effect is created at the start of the run, as a result of initial misalignment between the tractor and the implement. From these working traces you will readily appreciate that the drive shafts are subjected to arduous working conditions, because of the size and frequency of the pulsations. A larger flywheel would probably reduce the amplitude of the torque variations and improve the conditions considerably, but this would have disadvantages both from the physical size and the extra cost involved

The fifth and final machine is a cutter bar type mower, which is included in order to illustrate the effect of large

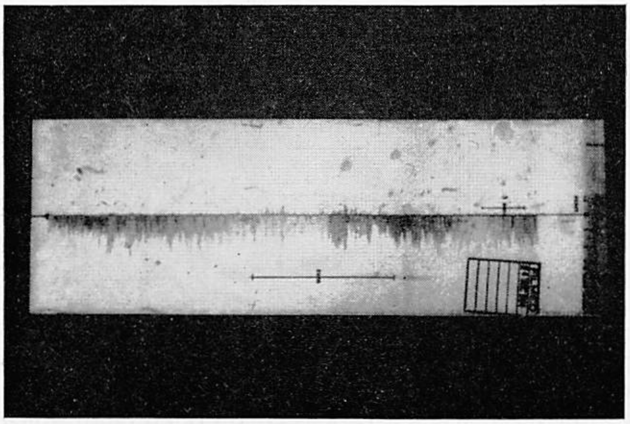

Fig. 12

velocity variations. In test number one (Figure 12), the mower was mounted in accordance with the manufacturer's instructions. The sections before and after the turn give the normal working trace and, during the turn, the cutter bar was raised whilst still operating. This trace is very similar to that of the previous mower in the mower crimper combination. For test number five

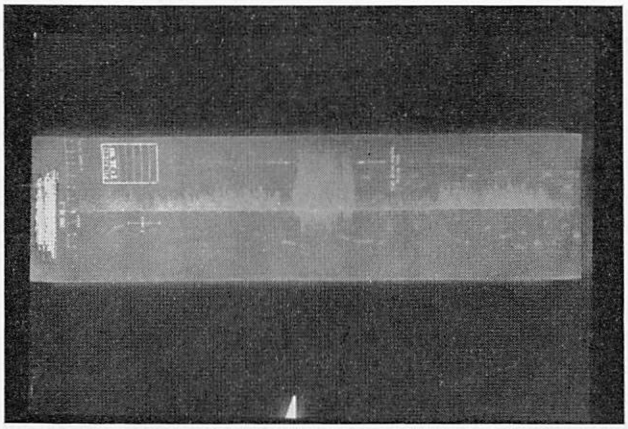

Fig. 13

(Figure 13) the implement was deliberately mounted incorrectly, such as might be the case with an inexperienced operator. This created unbalanced joint angles in the raised position, although in the working position the angles were normal. The result can be seen in the section marked 'turn' where torques of very high frequency ranging between 6,000 lb-in. and minus 2,500 lb-in. were recorded whilst the machine was idling. This is a good illustration of the attention which must be given to the problem of balancing joint angles at the design stage.

We have been carrying out torque measurement tests at Hardy Spicer for the last $3\frac{1}{2}$ years. In this time we have gained much valuable information and experience from this method of measurement, although we are still trying to keep pace with the continuous developments of the implements themselves.

I should like to say a word on the selection of the joint sizes to suit a particular machine. It is only possible to give quite general rules for this procedure, because of the many differing characteristics involved and the experience which is necessary to aid interpretation. Broadly, however, momentary overload torque must be catered for to ensure that the shaft does not suffer permanent damage. The mean working conditions must be taken into account in order that the shaft shall provide a satisfactory service life. If the momentary torque is very high relative to the mean as, for instance, in the case of the manure spreader, then the installation of a clutch will allow the use of a smaller joint, simply because the high peak condition is removed.

When establishing the size of drive shaft required for an application, due care must be taken to ensure that the peak torques attained during operation do not exceed the capabilities of the shaft. In certain applications, occasional peak torques are encountered which are of such magnitude that to specify a shaft large enough to successfully withstand such torques would, inevitably, result in a shaft which was uneconomical and far too heavy for normal operation. To combat these conditions, it is usually possible to incorporate in the drive line some form of torque limiting device; and when choosing a suitable type, due regard must be paid to the fact that, under all normal operating conditions, the drive must not be broken. Should the device be too weak, loss of power will result during normal working and, inevitably, the device will be overloaded and generate heat. This fault can be extremely annoying when the results of a certain operation are patchy and generally unsatisfactory. Furthermore, the operator is by no means consoled when he discovers that the clutch has been ruined and it is necessary for him to buy a replacement.

On the other hand, should the clutch be too powerful and not operate just short of maximum peaks attained, overloading of the shaft will occur with resultant damage to various parts in the transmission line. When choosing the correct size of clutch local conditions should be taken into account. For instance, a rotary cultivator, operating over very stony ground, may require a slightly higher clutch setting than one operating on a sandy soil, where the occasional large boulder or tree root will be encountered. The ability of the clutch to 'break away' is governed by shock loading and, therefore, testing of such clutches in production has to be taken into consideration. As I indicated on the torque traces, a clutch that will 'break away' under a steady torque of 5,000 lb-in. may fail to 'break away' under a sudden shock load of 7,000 lb-in., if the duration of the torque application is sufficiently short. This would mean that in an application where the shaft is continually subjected to torques of around 5,000 lb-in., it would be extremely difficult to fit a clutch that would remove peak loads of 7,000 lb-in., and when selecting the size of shaft required for the application, due consideration would have to be given to this factor.

Figure 14 shows a section of a typical clutch, as fitted

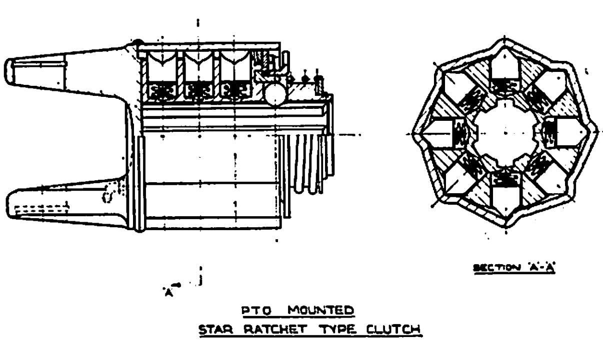

Fig. 14

to a drive shaft. The plungers are springloaded and the number fitted can be varied to give a variety of ratings. The rating cannot easily be altered by the operator, and remains very consistent during operation.

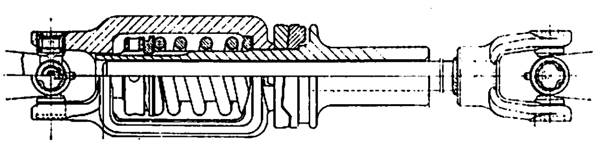

Fig. 15

Figure 15 shows an example of an old form of clutch, which is now largely out of favour. It is bulky, difficult to guard and can be very erratic in operation.

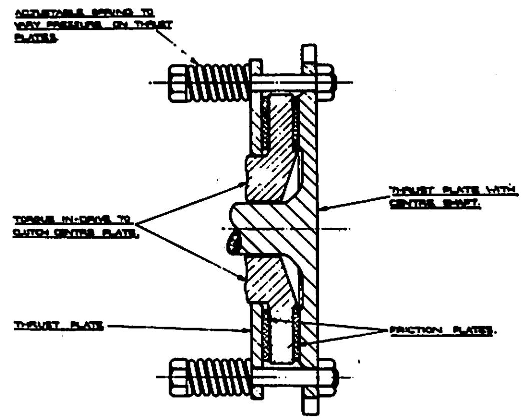

Fig. 16

Finally, Figure 16 shows a plate clutch as fitted to some implements. The pressure can be adjusted by tightening the bolts but this type has the disadvantage that some operators will over-tighten and so effectively prevent operation.

I should now like to give some consideration to the build-up of the standard p.t.o. shaft.

Sealing of Universal Joints

With the adoption, originally, of the basically automotive Hooke's joint for p.t.o. use in the agricultural world, the form of sealing which was currently in force for cars was naturally adopted.

FLAT SEAL

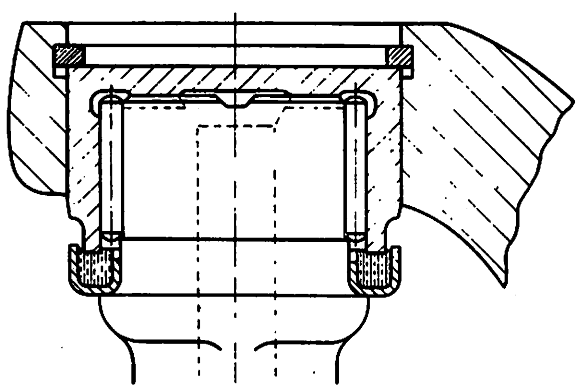

Fig. 17

The seal arrangement, as will be seen from Figure 17, consisted of a circular cork element, located in an annular pressed housing at the base of the trunnion peg, the top side of the seal being in contact with the end of the bearing bush. The journal has cross drillings into which is fed the lubricant from the greaser through to the four bearings. This form of seal has proved to be extremely durable in service and arguments can be put forward for the advisability of retaining the ability to re-grease which, at the same time as expelling the old lubricant, also gets rid of any dirt that may have collected in the bearings. Some variations on this principle were used, noticeably the introduction of a synthetic rubber bonded cork, which certainly improved the all round efficiency, largely because the sealing element itself now became impervious to the passage of oil or grease, this is not the case with plain cork.

Later it was felt that elimination of servicing or, certainly a considerable diminution in the frequency of this operation was desirable and our efforts were turned to the design of a form of sealing which would not only retain the lubricant, or sufficient quantity of lubricant for efficient operation during the life of the shaft but, almost as important, would also be equally effective in keeping out dirt.

A successful form of solution is shown in Figure 18 and this was known as the taper seal. This design necessitated the use of a conical form for the base portion of the trunnion peg, which had the added mechanical advantage of increasing the rigidity of this particular part of the journal. The sealing element was located in the 'L' shaped pressed steel housing in the end of the bearing bush. The element consisted of plastic bonded cork which was not only impervious to the passage of lubricant, but also had higher strength and elasticity than natural cork. As the section of the element was rectangular in its free state, its introduction on to the conical form of the peg formed a short sealing width, as illustrated in Figure 18, which was under the closer control of diametral dimensions, as compared to the 'stack-up' of tolerances which could be encountered on the type described previously.

TAPER SEAL

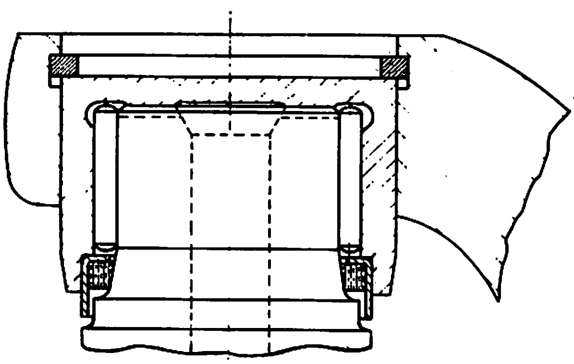

Fig. 18

The Hardy Spicer Walterscheid commonised series of agricultural shafts were fitted with this form of seal from inception of production. As far as I am aware, this was the first step towards the cutting down of maintenance operations in the p.t.o. agricultural shaft.

A later form of permanent seal known as the 'lip seal', is of quite different design and construction (Figure 19). The seal itself is manufactured from synthetic rubber and has an internal lip which bears on the end of the bush to retain the lubricant, whilst the outside of the bush is shaped to form a labyrinth to exclude dirt. It also has the advantage that particles of dirt which are flung outwards by centrifugal force, do not tend to be trapped by any obstruction at the mouth of the bearing bush adjacent to the seal, but are diverted outwards and around the smooth top edge of the seal.

LIP SEAL

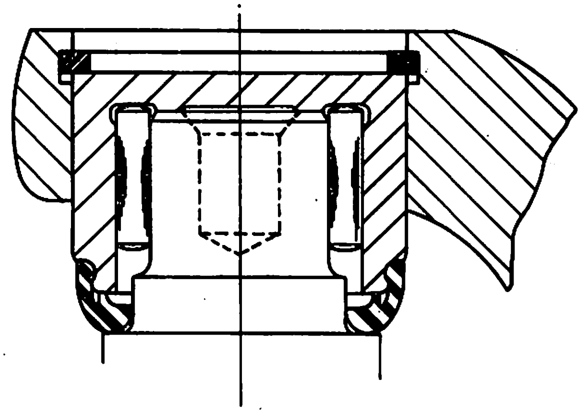

Fig. 19

Telescopic Members

In addition to universal joint angular movement, changes in length of the shaft are necessitated by turns and relative movement between tractor and implement over uneven ground.

On the standard form of propeller shaft fitted to motor cars, the changes in length are accommodated by means of a fairly short splined portion, which is situated either in the shaft itself between the front and rear universal joint, or is adjacent to the front joint sliding on the output shaft from the gearbox. Such arrangements are seldom used in the agricultural industry; the length differences which have to be accommodated are much greater than on road vehicles, also splines being of a high precision nature, because of the need for balance stability at high speeds, are expensive to manufacture.

The solution used on agricultural p.t.o. shafts was originally in the form of a square bar sliding in a hollow square sleeve (Figure 20). In the early days this was quite a satisfactory design, as both the bar and the hollow sleeve could be made cheaply and were of sufficient length to take care of the required movements. However, with the increase of torque loadings, following on the introduction of larger agricultural implements, this arrangement became unsatisfactory because of distortion in the hollow swaged sleeve, which led to binding and pick-up between the relative moving parts. Some alleviation of this problem was possible by the use of one or more

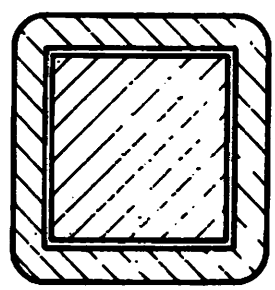

Fig. 20

lubricators in an endeavour to get the grease through to the more heavily loaded areas. Also oils and greases containing additives of the nature of colloidal graphites were used; some of which gave some alleviation to the problem, but did not satisfactorily solve it.

These problems led to the use of a rectangular bar, sliding in a solid sleeve which was broached out to a fitting rectangular form (Figure 21). This much more

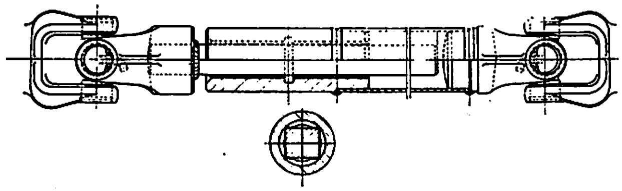

Fig. 21

rigid construction proved to be a great step forward, particularly with the introduction of lubricating grooves in the bore of the sleeve, helping to retain lubricant where required. A further step in the development of this arrangement was the use of an alloy steel rectangular bar; this basically reduced distortion because of the increase in strength, and permitted a higher torque throughput whilst lessening the tendency to bind up and seize. The rectangular form also ensured that the joints could only be assembled in correct phasing. All of the telescopic arrangements so far described can impose a heavy end load on the universal joints when operating under torque and, in some instances, are the cause of overloading when added to the working conditions of torque and angle imposed by the implement.

An interesting stage in the development of the telescopic portion of the shaft was carried out and put into production in Germany, which consisted of the use of fitting inner and outer tubes, lemon shaped in section, as indicated in Figure 22. It is not necessary for me to

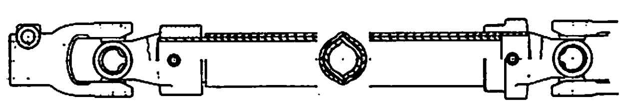

Fig. 22

emphasize the intial difficulties which were experienced in manufacturing the shapes precisely to drawing, in order to give the maximum bearing areas around the 'pips'. This problem was overcome as a result of close co-operation for many years between the p.t.o. shaft manufacturers and the tube mills.

Figure 23 compares the relative coefficients of friction between the arrangements described, from which it will be seen that the lemon section tube shows a considerable advance in this respect. This system is also a cheaper form of construction, as a lesser number of component parts are required in the build up as compared, for instance, with the rectangular bar arrangement. Another advantage is that the farmer, for service purposes, can himself cut the tubing to the required length should it be

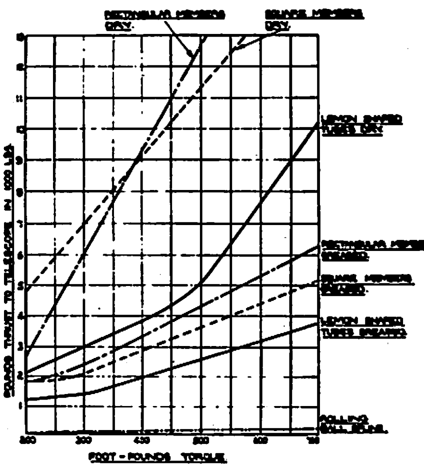

Fig. 23

necessary to shorten the p.t.o. shaft for any particular implement; this can quite easily be carried out by the use of an ordinary hacksaw. Hardy Spicer have this type of shaft in production as part of a programme to standardize the agricultural p.t.o. shaft in Europe. Associated companies are producing the same arrangements in both France and Germany.

A further development is the use of pearlitic malleable iron for the universal joint yokes in the lemon section arrangement, as opposed to the steel forgings which have been traditionally used in this country. This is a further step in the constant endeavour to reduce the cost of this component to the implement manufacturer.

Many programmes have been undertaken with the object of designing more satisfactory forms of telescoping tubular members as illustrated in Figure 24; but they

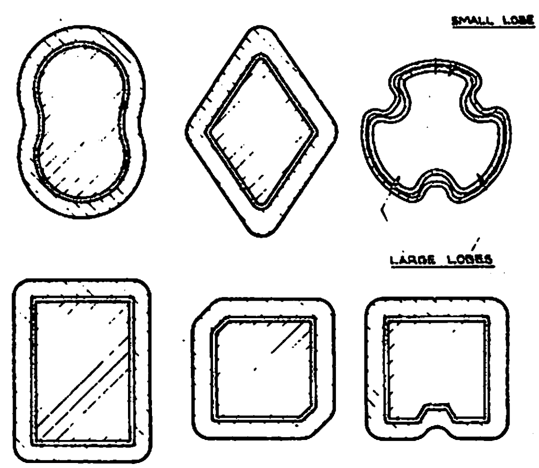

Fig. 24

have all proved to be too expensive due, largely, to the difficulty of forming the shapes within acceptable tolerances on a full production basis.

I have already mentioned the sealing of the universal joints, and the fact that it is really a twofold endeavour, on the one hand to keep the lubricant in and on the other to keep the dirt out. The same endeavours will have to be made to evolve a form of permanent sealing between the telescoping members, if we are to produce a shaft which is virtually service-free—an ideal which is most desirable. Accordingly we are at this moment still working on various forms of seal, with emphasis on the lemon tube form.

One of the later forms of telescopic member under development for agricultural purposes is the frictionless rolling spline. So far as I know this has not as yet been introduced into production to any extent in this country, but it is more common in America. The latest S.A.E. regulation has set down limiting figures for axial resistance compared with torsional loading, a step which seems to be a most sensible acceptance of the situation brought about by the much heavier implements in use in that country. Undoubtedly, we shall soon be faced with a similar situation in this country and in Europe generally.

Figure 25 shows a section of a typical spline now under

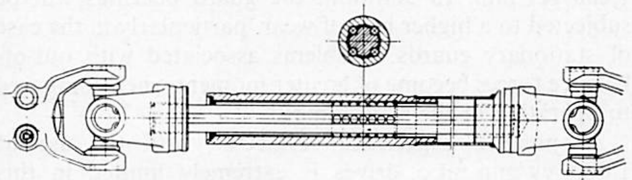

Fig. 25

development. This, as you will see, consists essentially of four tracks in an inner and outer member into which is introduced an appropriate number of balls to accommodate the loading conditions. Such a design is certainly much more expensive than the sliding forms I have described, and is, in fact, a simplified form of the frictionless splines, both ball and roller types that have been used on certain European cars for some years. I think their eventual adoption is inevitable, because increasing torques imply increasing end loading from the sliding arrangements bringing about uneconomic sizes of universal joints because they would be unrelated directly to the power consumed by the implement for useful work.

Guarding

Regulations for the guarding of p.t.o. shafts existed in some of the continental countries for quite a period before they became statutory in Great Britain. This applied particularly to Scandinavia where the regulations were of a precise nature and made it compulsory for the guard to be stationary, even whilst the shaft was in operation. Because of this requirement, precautions had to be taken in the design of the bearings to eliminate the possibility of any form of seizure, which would cause the guard to become locked with the shaft.

Chains were normally used between the covered portion of the guard and a convenient hitch point on the tractor or implement to prevent rotation (Figure 26). These regulations are still in force and it is interesting to note that similar requirements are being introduced into France and seem likely to be extended to other countries.

The law in this country is not, at present, so demanding rather implying that the shaft must be covered, but that

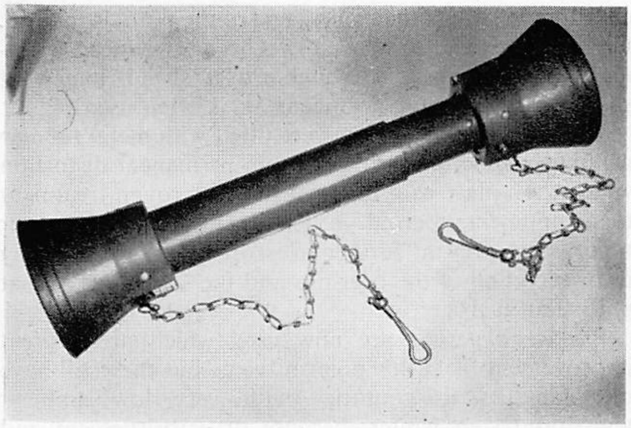

Fig. 26

the guard can rotate with it, provided that there is a sufficiently low coefficient of friction in the bearing arrangement, to ensure that it will stop if someone or something falls against it.

These somewhat less stringent requirements have made it possible to consider a much wider choice of bearings than would be possible with the stationary arrangement and, consequently, they can be less costly in construction. The permanently stationary guard must be fitted with lubricators for service purposes, in addition to having bearings which will be robust enough to operate continually throughout the life of the guard.

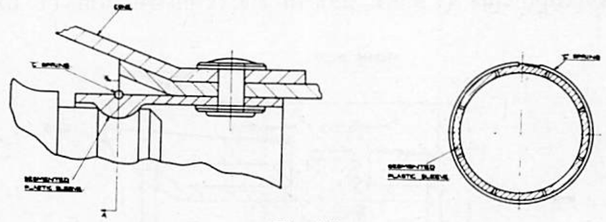

Fig. 27

As will be seen from Figures 27 and 28, the guard normally takes the form of two telescoping tubular members located on bearings, running on to yoke bosses at each end of the shaft. The tubes terminate in flared or conical portions, covering as much of the joint as possible,

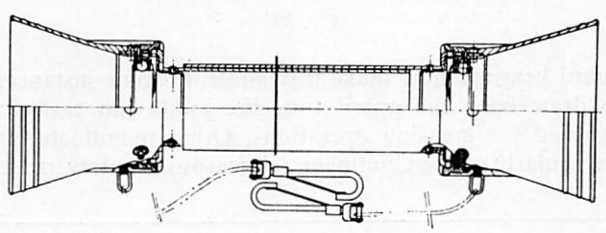

Fig. 28

bearing in mind the restrictions on the length of this part because of the high angularities required by the p.t.o. arrangement. The almost universal use nowadays of flexible cones has helped this situation, because they will distort under pressure from the joint when the implement is, for instance, parked at a high angle. At the tractor end, there is a regulation specifying a fixed platform type of guard under which the conical end of the p.t.o. is located. No precise regulations have as yet been introduced applying to the implement, although a fixed shield, somewhat similar to the tractor arrangement, is often used.

The early guards, which were fitted with metal tubular members, sometimes suffered from permanent distortion which took place as a result of the very rough treatment this equipment received—a permanent dent in the outer tube would prevent further telescoping action, resulting in overloading of the bearings and the scrapping of that particular guard.

Plastics materials are now used which are a great improvement from this point of view, but they still have disadvantages in countries with very low ambient temperatures in winter, because of embrittlement and reduction of strength at these temperatures. I think it is fair to say that the steady development of plastics compounds in all their physical characteristics should produce a solution to this problem.

Continental manufacturers, for the most part, have retained metal tubing of a generous wall thickness and have used it in combination with plastic cones. Such designs are basically more expensive than the plastic versions because of the cost of the steel tube which has to be made to a special size, either inner or outer in order to 'mate' with the other member within the limits of the clearance required for satisfactory sliding in operation.

Early guards, because of the rigid cones, prevented easy access to the universal joints for service purposes; but developments (Figure 29) in the construction of the

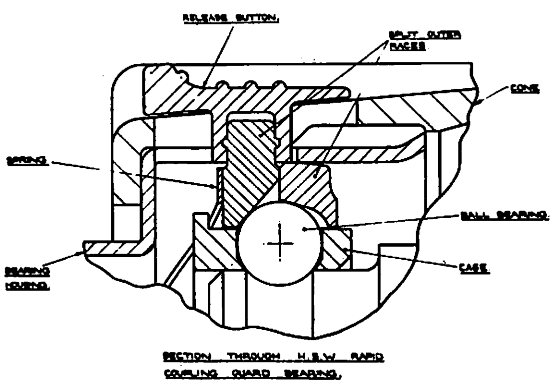

Fig. 29

guard bearings now make it possible, in some instances, to draw back the guard, thus the joints can easily be exposed for greasing operations. Other manufacturers, particularly on the Continent, for reasons of safety, prefer to rely on a form of corrugated plastic cone which can be held back by hand to expose the joints for service, but when released will spring back again into its guarding position.

The Future

Tractor manufacturers are now providing a choice of speeds for p.t.o. shafts, namely the present accepted standard of 540 rev/min, and a new speed of 1,000 rev/min. Whilst this has certain advantages from the implement manufacturers' and the users' point of view, it does create additional problems in connection with the drive line. Obviously, with the variation of angles between the p.t.o. joint and the p.i.c. joint, the velocity variations will be more pronounced and, therefore, that much more unacceptable when working at 1,000 rev/min than at 540 rev/min. It is necessary to take this fact into account when designing implements which are to be driven at 1,000 rev/min. In addition, the guard bearings will be subjected to a higher rate of wear, particularly in the case of stationary guards. Problems associated with out-of-balance forces become of greater moment when compared to operation at the slower speed.

At present, experience connected with the use of 1,000 rev/min p.t.o. drives is extremely limited in this country and, although development work is being carried out on a fairly broad front, it may well be some time before all the problems and their solutions are fully appreciated.

Certain development work is being carried out by the National Institute of Agricultural Engineering and some of this work is directed at the possibility of incorporating the hitch point within the drive shaft and utilizing a single constant velocity joint within the hitch. One of the difficulties is the inability of constant velocity joints to operate at the angles required for turning at headlands, corners, etc.

Another interesting line of development, to enable the tractor driver to connect and mount his implement, without descending from the tractor, is now being pursued by several manufacturers. This obviously saves a great deal of time and trouble in the field; and considerable success for non-powered implements has already been achieved.

The problem of powered implements is, of course, a much larger one and, various designs, some of them complicated and expensive, have been put forward to allow for the coupling up of the drive line at the same time as the hitch is connected. Certainly, within the next year or two, considerable strides will be made in this direction.

Discussion to Paper by T. Sherwen (*continued from page 12*)

The loss in an epicyclic box, Mr Sherwen continued, was due to pump operation. While the controls of the box were being operated during a power sustained change work was being taken from the pump so that momentarily the loss in the pump would be high; it would decrease again once the change had been made to a constant value. There were, moreover, oil churning losses which varied with the speed and oil temperature but not with the torque transmitted.

Mr C. J. Moss (NIAE) said that Mr Bates had emphasized that insufficient data from the field was available on comparative performances of mechanical and hydraulic transmissions. Drawing office and laboratory calculations were inadequate in a controversy of this kind and it was his very earnest wish that a very much better comparison of 'in the field' performance of tractors with different types of transmissions was achieved. The Institute was endeavouring to stimulate an interest in comparative performance and he hoped that it would be possible to do a prolonged and thorough study in the field of the comparative performances of a mechanical and a hydrostatic transmission tractor.

Dr A. R. REECE (University of Newcastle-upon-Tyne) suggested that tractors of a similar price should be compared rather than tractors of similar size and power. The farmer could have either say, a 40 hp hydrostatic tractor which had many advantages or a bigger heavier more powerful conventional tractor with which bigger and more powerful equipment would be required. The comparison should thus be between output per unit cost including fuel and running cost. He felt that the results of the test described by Mr Nation were not favourable to hydrostatic transmission from the point of view of the tractor's work on the farm over the year; if the cost of the equipment were taken into account the results would be even more unfavourable. In conclusion Dr Reece welcomed the comparative tests envisaged by Mr Moss and suggested that the Institute should show what the differences in mechanical performance would mean in economic terms to a representative range of farmers in this country and abroad.

Mr SHERWEN said that whilst he agreed that ultimately it was essential to make comparison between units of the same cost, at this stage of development of the hydrostatic tractor its cost could not be comparable with that of its mechanical counterpart since it was not in large scale production. Mr J. H. W. WILDER (President) said that fifty years of engineering knowledge and production technique had gone into designing and producing gears. Hydrostatic tractors were a new development and thus it was impossible to make present cost assumptions based on present designs. The comparison must be based on the assumption that hydrostatic transmission tractors would benefit from the same amount of research work as had gone into the design and production of mechanical transmission tractors.

Mr BATES invited Mr SHERWEN's opinion on the future possibilities of the turbo-charging of diesel-engine tractors and the likelihood of this improving their overall mechanical efficiency. Mr Sherwen said that he felt that turbo-charging was economical where there was a continuous demand for the extra power as often occurred in the road transport industry. With a farm tractor, however, there was a very wide difference in power demands throughout farm operations and it thus might not prove economical to use turbo-charging as a method of getting a given amount of power.

In response to a question by Mr K. A. L. ROBERTS (Bomford Bros Ltd) as to whether the figures quoted on the relative costs of different forms of transmission were comparable in terms of the amount of work available at the output end, taking into account the effective efficiency of the individual unit, Mr Sherwen said that the figures were based on a constant power input.

Mr D.W. Jewett (University of Cambridge) said that one of the important recent developments in the commercial vehicle field seemed to be the Perkins differentially super-charged diesel engine. The torque/speed characteristic of this unit was an almost perfect rectangular hyperbola. It would appear that this removed many gearing problems and also to be a very cheap method of achieving the desired type of torque/speed curve. The unit employed a torque convertor with the result that it had many of the inherent advantages of a hydrostatic transmission in the sense that the components of the power train were not subject to misuse. Mr Sherwen said that the differentially-charged system went some way towards solving the problems that hydrostatic transmission could solve but that at its present stage of development it must be used in conjunction with some form of gear ratio change as well as the torque convertor.

Discussion to Paper by J. A. Howard *(continued from page 20)*

of Newcastle-upon-Tyne) for precise information on the setting of friction clutches under variable conditions, Mr Howard said that instruction on the setting of safety clutches was a problem shared by many implement manufacturers. Friction clutches needed to be adjusted for almost every job and every tractor. It was most difficult for the manufacturer to convey to operators the correct adjustments for clutches and for this reason his company tended to prefer transmission safety devices which did not require adjustment.

MR J. H. NICHOLLS (David Brown Industries Ltd) drew attention to the American system of two individual power take-off shafts, one for 1,000 rev/min and the other for 540 rev/min as against a single conventional shaft for both speeds. Mr Howard said that he preferred the American system whereby a six spline $\times$ $1\frac{3}{8}$ in. shaft was used for 540 rev/min and a 21 tooth involute spline also $1\frac{3}{8}$ in. for 1,000 rev/min. The system generally adopted in the United Kingdom was to use the same shaft for both 540 rev/min and 1,000 rev/min power take-offs. There was, therefore, a risk of the operator running the shaft at 1000 rev/min when the implements required a drive of 540. Operator instruction could help solve this problem but in general he favoured the American system.

In reply to a further question from Mr Nicholls, Mr Howard said that there were some tractors in use in the United States in which it was possible to withdraw and replace the shaft; the mechanism was designed so that introducing a different size of spline caused the ratio of the drive to be changed automatically. This appeared to be the ideal system where it could be achieved.

ELECTIONS AND TRANSFERS

Approved by Council at its meeting on 13 January 1966

ELECTIONS

Associate Member		Arnold, R. E.	Beds
		Billington, W. P.	Beds
		Byass, J. B.	Beds
		Carpenter, G. A.	Beds
		Charlton, G. K.	Beds
		Chestney, A. A. W.	Bucks
		Coleman, E. P.	Herts
		Comely, D. R.	Beds
		Cottrell, F. B.	Beds
		Cowell, J.	Beds
		Dawson, J. R.	Beds
		French, M. W.	Beds
		Graham, K. A.	Perths
		Johnson, R. A. H.	Herts
		Lindsay, R. T.	Beds
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		Weaving, G. S.	Beds
		Wells, D. A.	Beds
		Williams, N. Y.	Herts
		Winspear, K. W.	Herts
Associate		Anderson, P.	Berks
		Cleaver, G. F.	Beds
		Edgar, R.	London
		Gutteridge, M. W.	Beds
		Macpherson, W. O.	Yorks
		Searle, M. S.	Somerset
		Shed, G. W. Z.	Beds
	Overseas	Marengo, C. E. G.	Republic of S. Africa
	Overseas	Molligoda, A. R.	Ceylon
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		Cartland, M. C.	Beds
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		Ellam, D. F.	Essex
		Hale, O. D.	Beds
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		McLatchie, R. J. T.	Glasgow
		Mayne, I. T.	Warwicks
		Southcombe, E. S. E.	Beds
	Overseas	Onafeko, O.	Nigeria

ELECTIONS AND TRANSFERS *(continued)*

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		Bebb, D. L.	Salop
		Bell, D. A.	Cumberland
		Bowden, C.	Hants
		Carpenter, P. W.	Cambs
		Keightley, M. S.	Surrey
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		Willett, G. T.	Beds
		Yarrow, M. W.	Berks

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Associate Member		Bradley, M. J.	Hants
		Draper, I. C.	Lincs
		Draper, M. J.	Lancs
		Nellist, M. E.	Beds
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Engineering Dept
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WEST MIDLANDS

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The Goodyear Tyre & Rubber Co (Great Britain) Ltd
Bushbury, Wolverhampton
Staffs

YORKSHIRE

T. H. E. HARRISON, BSC(AGRIC), MSC(AGRIC ENG), GR I AGR E
16 Wood Lane, Ashenhurst
Huddersfield, Yorks

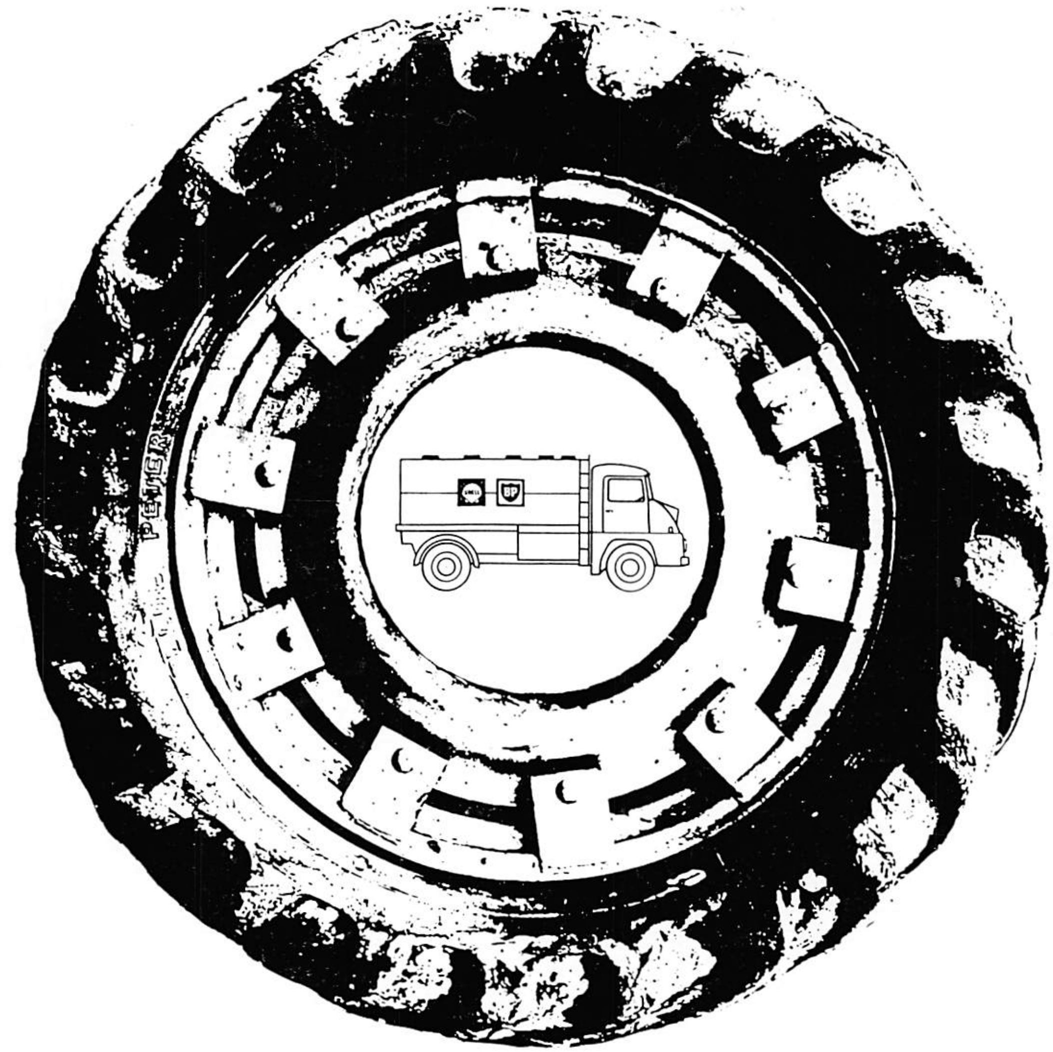
Index to Advertisers

Abbreviations and Symbols used in the Journal

a	year
A or amp	ampere
ac	acre
a.c.	alternating current
atm	atmosphere
b.h.p.	brake horse-power
bu	bushel
Btu	British Thermal Unit
cal	calorie
c.g.	centre of gravity
C.G.S.	centimetre gramme second
cm	centimetre
c/s	cycles per second
cwt	hundredweight
d	day
dB	decibel
D.B.	drawbar
d.c.	direct current
°C, °F, °R	degree Celsius, Fahrenheit, Rankine
deg	degree (temperature interval)
dia	diameter
doz	dozen
e.m.f.	electromotive force
ft	foot
ft^2	square foot (similarly for centimetre etc.)
ft lb	foot-pound
G.	gauge
g	gramme
gal	gallon
gr	grain
h	hour
ha	hectare
Hg	mercury (pressure)
hp	horse-power
h	hour
in.	inch
in^2	square inch
i.d.	inside diameter
kWh	kilowatt hour

l	litre
lb	pound
lm	lumen
m	metre
max.	maximum (adjective)
m.c.	moisture content
m.e.p.	mean effective pressure
mile/h	miles per hour
mill.	million
min	minute
min.	minimum (adjective)
o.d.	outside diameter
o.h.v.	overhead valve
oz	ounce
Ω	ohm
pt	pint
p.t.o.	power take-off
qt	quart
r	röntgen
r.h.	relative humidity
rev	revolutions
s	second
s.v.	side valve
S.W.G.	standard wire gauge
t	ton
V	volt
v.m.d.	volume mean diameter
W	watt
W.G.	water gauge
wt	weight
yd	yard
$>$	greater than
$\ngtr$	not greater than
$<$	less than
$\nless$	not less than
$\propto$	proportional to
$\sim$	of the order of
$^\circ$ $'$ $''$	degree, minute, second (of angles)

The above abbreviations and symbols are based mainly on B.S. 1991 (Part 1), 1954



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