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IMPLEMENT CONTROL AND ITS EFFECT ON THE DYNAMIC PERFORMANCE OF TRACTOR-IMPLEMENT COMBINATIONS

by

DAVID ANTHONY CROLLA, B.Tech.

A Doctoral Thesis

submitted in partial fulfilment of the requirements for the award of Ph.D. of the Loughborough University of Technology.

July 1976

Supervisors: Professor F.T. Hales
E.G. Jenkins

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Preface

The work described in this thesis was carried out at N.I.A.E. between 1972 and 1975 as part of project 1410, 'An investigation of the performance of tractor draught controls. Thanks are due to the Director of N.I.A.E. for permission to publish the work.

So many friends and colleagues at N.I.A.E. helped me in various ways throughout the preparation of this thesis that to list them all would be impossible. However, I would like to take this opportunity of thanking them all for their assistance. In particular, I am especially grateful to Mike Dwyer for his continued support, advice and encouragement, and to Geoff Pearson and Pete Seward for their assistance with the field measurements and analysis. I would also like to thank Professor Hales and Mr E.G. Jenkins of Loughborough University both for their suggestions and for the many helpful discussions we had during this work.

Finally, Sue Daly and Marilyn Bonner deserve a special mention for their patient efforts in typing the manuscript.

The work presented here is original, except where otherwise stated, and has not been submitted for a degree at any other university.
IMPLEMENT CONTROL AND ITS EFFECT ON THE DYNAMIC PERFORMANCE OF TRACTOR-IMPLEMENT COMBINATIONS

by

DAVID ANTHONY CROLLA, B.Tech.

SUMMARY

Tractors incorporate draught controls for two reasons, to limit variations in implement depth and to limit fluctuations in the load, i.e. the draught force, imposed on the tractor by the implement. Efficient operation of this control is important because many farmers demand a consistent tillage depth and also, if the load is controlled, the tractor can be operated at its maximum output without excessive wheelslip or engine stall.

The work for this thesis involved a theoretical investigation of draught control response and its effect on overall tractor performance. A computer model was devised to simulate draught control and its effect on dynamic performance of the tractor-implement combination. Input data for the simulation was provided by laboratory measurements. Two series of field work were carried out to measure response of a linkage force sensing experimental control to sinusoidal and random draught variations and the response of a control which sensed driveline torque. Comparisons between measured and predicted results confirmed the validity of the theoretical analysis.

Suggestions for improving control performance, particularly stability, are made and the likely improvement predicted using the computer simulation. Other possible methods of implement control or improvements to draught controls are discussed in relation to the likely requirements of future tractors.
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1.1 INTRODUCTION AND REVIEW OF PREVIOUS WORK

1.1.1 General

The main source of power in modern farming is the agricultural tractor. On average, a typical tractor spends about one third of its life on cultivation, the rest being spent mainly on transport and harvesting operations. The cultivation operations can be divided into two categories; primary cultivations in which the soil is loosened or inverted after the previous crop, and secondary cultivations in which the disturbed soil is further broken down to a fine tilth in preparation for planting. This thesis is concerned with the performance of the tractor while it is operating primary cultivation implements.

Typically, a plough or rigid tine cultivator is used for primary cultivation and it is connected to the tractor by a three point linkage, consisting of one upper and two lower links. The implement is then fully mounted on the tractor and the implement weight is transferred to the tractor rear wheels. Implements can also be semi-mounted, i.e. attached to the lower links only, in which case only some of the implement weight is transferred to the tractor, or the implement can simply be trailed by attaching it to the tractor drawbar, in which case no weight is transferred.

To control implement depth, nearly all tractors incorporate a draught control which senses the force in either the top or lower link and raises or lowers the linkage to adjust implement depth. This system is used for fully mounted implements; for semi-mounted equipment the linkage is used to control depth at the front of the implement and a depth wheel controls depth at the rear. For trailed implements, depth is usually controlled by depth wheels at both the front and rear of the implement.

There are two benefits to be gained from improved implement depth control; one is a more consistent tillage depth and the other is a more consistent draught force. The benefits due to a more accurate control of tillage depth are not immediately apparent although in
surveys carried out by N.I.A.E. (1) it was found that over half the tractor operators observed were not satisfied with the accuracy of depth provided by draught controls and were prepared to make continual manual adjustments to improve it. Although these operators were attempting to keep the depth variation to within ±10% of the mean depth, the relationship between accuracy of tillage depth and crop yield is not known. The success of direct drilling (2,3) and minimum cultivation techniques (4) tends to suggest that there are certainly some soil conditions where accurate tillage depth is relatively unimportant. Equally, in many soil conditions, inaccurate control of ploughing depth results in subsoil or unburied trash remaining on the surface, both of which have a detrimental effect on crop yield (4).

The effect of more consistent draught force is that the tractor can be operated nearer its maximum rate of work. If the draught variations are large then tractor power must be kept in reserve to prevent tractor stall or excessive wheelslip. The N.I.A.E. survey previously mentioned, also showed that some operators did not consider the control of draught force to be accurate enough because some of the manual interventions to the control were to prevent tractor stall or excessive wheelslip.

1.1.2 Cost benefit

The cost of primary cultivation in the U.K. is high. There are about 5 million hectares (12.5 m acres) of arable land and another 2.4 million ha of temporary grassland (5). Assuming the temporary grassland is cultivated once every four years on average, the total area cultivated per annum is 5.6 million ha. Assuming that the average time spent cultivating 1 ha is 4 hours, the total time spent is $22.4 \times 10^6$ h/annum. With labour and fuel costs of about £2.50/h, the total direct cost of primary cultivation is £44.8 $\times 10^6$/annum. It can be seen therefore that any improvements to tractor draught controls which enable work rates to be improved by even a few percent represent a large saving in national terms.
There is a trend towards higher tractor operating speeds to improve work rates and the benefits resulting from using more powerful, faster tractors, taking into account initial cost depreciation repairs and operating costs, has been estimated as £10 m/annum (6). Higher speeds however are not practical for heavy drought cultivations unless improvements are made to present controls, which have been shown to be inadequate above about 2 m/s (7). The other problems of high speed operations, namely tractor ride vibration and implement design, both of which are the subject of present research (8,9), also affect the implement control and must be included in any analysis of implement controls for higher speeds.

Because this thesis is concerned with implement controls and their effect on tractor performance, it is convenient to divide the following review of previous work into two sections. In the first section the forces acting on cultivation implements and the performance of both commercial and experimental controls are reviewed. In the second section the attempts that have been made to analyse and predict tractor-implement performance are reviewed and discussed.

1.2 REVIEW OF IMPLEMENT CONTROLS

1.2.1 General

Until 1936 when Ferguson introduced top link sensing draught control (10) the only method of controlling implement depth was with depth wheels attached to the implement. Although this was a satisfactory method of maintaining constant depth it had disadvantages. The tractor had to be capable of pulling the implement through the heaviest parts of the field where the soil resistance was highest, and was therefore operating below maximum power over the rest of the field. Also, the combination was not very manoeuvrable, the turning circle was large and reversing was difficult. Mounting the implement on the tractor three-point linkage and using the draught control overcame these problems and had the added advantage of transferring implement weight to the tractor rear wheels to increase tractive
effort. Its only drawback was that depth was not controllable as accurately, because when the implement reached heavier soil, the control reduced the depth slightly so that the draught remained constant.

Since their introduction, top link sensing controls have changed little. Refinements have been incorporated to make the sensing unit respond to both tensile and compressive forces rather than compressive forces only (11), and controls to vary the rates of lifting and lowering have been added in attempts to obtain the optimum response without hunting (12). Top link sensing remains the most common method of control on tractors of under 75 kW which form 95% of tractors in use in the U.K. (5).

Lower link sensing has become increasingly common, especially on larger tractors in the U.S.A. (13). Its main advantage is that it can be used with semi-mounted as well as mounted implements. Also, since the lower link force is always tensile, the sensing unit does not have to sense both compressive and tensile forces and can therefore be slightly simpler than a top link sensing unit.

Another common type of control is pressure control or "traction control" (14,15) which keeps the pressure in the hydraulic lift ram constant and hence the force on the lift rods and weight transferred to the tractor constant. Tractors fitted with this control use it to supplement top or lower link sensing rather than replace them. The control is usually used in conjunction with depth wheels on the implement or with semi-mounted implements since it does not strictly control either depth or draught.

The only other control available commercially is torque sensing control which has been introduced by one manufacturer (16), again as an added extra rather than a replacement for top or lower link sensing. It has the advantages that it can be used with mounted, semi-mounted or trailed implements.

Before reviewing the work carried out into the performance of the
above-mentioned commercial controls and later the performance of experimental controls, it is relevant to briefly summarise the work on soil forces on implements and the forces in the tractor linkage. These forces affect the performance of existing controls and also govern the control strategies adopted to improve performance with experimental controls.

1.2.2 Forces on implements

Only the forces acting on ploughs and cultivators will be considered since these are the two implements relevant to this study.

(i) Plough

The conventional mouldboard plough operated by cutting and inverting slices of soil. The main components of the draught force are the cutting force and frictional force as the soil slides over the mouldboards. Much work has been done to minimise the draught by maintaining correct cutting angles and ensuring that just enough energy is imparted to the soil to invert it (17-20). Measurements have been made on various parts of the plough body and frame (21-23) and the effect of altering the angles of the bodies has been studied (24). Other measurements have included the frequency and distribution of loads to investigate fatigue life (25,26). Dimensional analyses have to some extent explained the relationship between draught force and soil parameters (27, 28).

In general, it is not unreasonable to assume that plough design has been optimised (29,30) although it must be noted that a particular plough can only produce optimum performance at one speed. This is most noticeable with high speed bodies because above the optimum speed, too much energy is imparted to the soil and it is thrown sideways, and below this speed not enough energy is imparted to the soil to invert it (31).

Draught has been found to increase with depth in a relationship which depends on the particular plough and soil condition (32). Vertical force tends to increase slightly with depth until the average
working depth is reached when it decreases as the soil reaction forces on the disc coulters, hubs and landside increase. Skalweit (33) considered that the supporting force, \( S \), on the landside was the most important of these over the normal range of working depths and its relationship with depth is shown in fig. 1.1. Measured values of draught and vertical force are also shown.

The relationship of draught force with speed for a particular plough can be described by the expression (34-36),

\[
H = K_1 + K_2 v^n
\]  

(1.1)

where \( K_1 \) and \( K_2 \) are constants, depending on the plough body design and soil characteristics, \( v \) is forward speed and the exponent \( n \) has been found to vary between 1 and 2 depending on the soil condition (35). This equation is a modified form of Goryachkin's earlier equation (37),

\[
H = f g + K a + \gamma a b v^2
\]  

(1.2)

where \( f g \) = a function of plough weight and soil metal friction,

\( K \) = coefficient of soil resistance to deformation,

\( a \) = furrow depth,

\( b \) = furrow width,

\( \gamma \) = coefficient which is a function of soil density and plough geometry.

Equation (1.1) however is better able to fit the experimental results because variation of the value of the exponent, \( n \), enables better correlation.

(ii) Cultivators

For cultivator tines, draught has been found to increase with depth in a relationship similar to that for ploughs, but again depends on the particular tine and soil condition (38,39). The vertical force also increases with depth (40) and typical relationships of both \( H \) and \( V \) with depth are shown in fig 1.2.

Draught has been found to be approximately proportional to
Fig. 1. Soil support force / depth relationship for a 3 furrow plough in field 3, with different linkage arrangements (after Stalweit (33))
forward speed over a typical range of working speeds from 1 to 4 m/s
(41,42).

\[ H = K_3 + K_4 v \]  \hfill (1.3)

where \( K_3 \) and \( K_4 \) are constants depending on the particular tine
dimensions and the soil condition.

1.2.3 Linkage forces

The tractor three point linkage has been the subject of a consider-
able amount of work. The forces exerted in the links have been
measured during ploughing (43-45), cultivating (46), which lifting
and lowering implements (47) and with various methods of mounting (48)
in attempts to optimise the linkage configurations. There have been
two important aims behind these measurements, firstly maximising the
amount of weight transfer during operation (49) and secondly minimising
the hydraulic force necessary to raise the implement clear of the ground
(50,51).

Measurements of the linkage forces are also necessary for
analysing draught control because top or lower link force is commonly
sensed and is therefore the input to the control. A typical example
of top and lower link forces, \( F_T \) and \( F_L \) measured by Skalweit (33)
when a three furrow plough was being used is shown in fig 1.3.

Other analyses (52-55) have shown that for certain linkage angles
an implement control is not required and the implement will naturally
remain at a given depth. The instantaneous centre of rotation of an
unsupported linkage occurs where the lines drawn along the links
intersect. If the resultant forces on the implement pass through this
point then the implement is in equilibrium and remains at that depth.
By arranging the angles of the links to be adjustable by moveable
mounting points on the tractor, this system was used as a depth control
for one small tractor (56).

1.2.4 Performance of conventional implement controls

Many of the early measurements of the performance of commercially
available controls were carried out to compare depth controls with
Fig 1.2 Typical relationships of cultivator tie draught and vertical force with depth
draught controls. The reason for this was that traditionally, accurate tillage depth, especially when ploughing, was important and when hydraulic draught controls were introduced farmers wanted to know if they were as good as their traditional method of using depth wheels.

Hawkins and Boa (57) compared the following combinations,

(a) trailed two furrow plough,
(b) two furrow plough with depth wheels,
(c) two furrow plough with draught control,
(d) a single furrow plough drawn from a real hitch point under the belly of the tractor.

Their performance criterion was that the plough should follow the surface contour but ignore minor surface irregularities, so they measured ploughing depth on land with ridges and furrows of amplitude 10 cm. They concluded that (d) gave the best depth control and (c) the worst. For (a) and (b) the plough followed the surface contour but smoothed out the depth variations so that the amplitude of depth was smaller than the amplitude of the surface undulations.

Seifert (58) compared a depth control, arranged by fitting depth wheels to a plough, with two top link sensing controls. Again using depth variation as a performance criterion he concluded that the depth wheels were better than either of the draught controls. Other workers reported similar results (59, 60). In earlier work Seifert (61) had concluded that the acceptable depth variation when ploughing was ±10% of the mean working depth in the range 18-25 cm.

In a comprehensive series of surveys by Dwyer, Osborne and Rogie (1), it was found that many farmers were attempting to control depth within these limits. The average standard deviation of depth when ploughing with the draught control operating was 2 cm. This meant that the depth variation was less than ±4 cm over 95% of the field. For a mean depth of about 20 cm this represented a ±20% variation about the mean. However, over half the operators observed were making continual manual adjustments to the draught control lever, and by doing this the average standard deviation was improved to 1 cm. The depth variation was then
Fig 1.3 Top and lower link force/depth relationships for a 3 furrow plough (after Skalweit (33))
+10% over 95% of the field, which suggested that Selfert's limits were acceptable to many farmers. For cultivating, a comparison was made between draught control and depth control using depth wheels. Using draught control, the average standard deviation of depth was 3.5 cm but this was reduced to 2.5 cm when depth wheels were fitted.

The comparison already mentioned showed quite clearly that depth wheels provided better control of depth than draught control. However no work has been reported in which the same comparisons are carried out using draught force as a performance criterion. The only reported measurements of the draught variations of a depth controlled implement (62) showed that the fluctuations were large. Draught force varied by as much as +50% over 100 m of ploughing at constant depth and by +100% over an area of 1 ha. This variation is far greater than typical maximum values of +30% (64) for ploughing with the draught control operating and it suggests that a realistic comparison of draught and depth control must include measurements of both depth and draught.

The performance of various draught controls has also been compared. Cowell and Len (7) in 1967 compared the field performance of two top link sensing controls, one having a variable rate of lifting and the other a variable rate of lowering. The tractors pulled a three furrow plough over an artificially prepared surface which was described by a sine wave of wavelength 10 m and amplitude 10 cm. Their performance criterion was depth variation measured with a level and staff and the main conclusions drawn were that, (a) the response rate had little effect on performance, and (b) performance deteriorated rapidly with speed. The plough came out of the ground at a forward speed of 2 m/s with one control and at 2.6 m/s with the other. They also carried out some measurements with depth wheels attached to the plough and found that the depth control was then improved. This was an important result because the soil resistance was fairly constant in the experimental plot and therefore the depth variations allowed by the draught control were not due to variations in soil resistance but due to the inadequacy of the control. So it showed that over an undulating
surface where soil resistance was constant, depth wheels provided better depth control than top link sensing draught controls. Previous workers who had obtained similar results had not shown whether the greater depth variations allowed by draught controls were due to changes in soil resistance or inadequate response of the control.

They concluded that the depth errors with the draught control operating were due to the fact that large vertical forces were induced on the plough due to the tractor vertical movement and it was these, rather than the draught force which operated the control valve. They did not however have a relationship between vertical force and plough vertical motion and so they could not analyse its effect further.

Dwyer (64) in 1969 analysed the forces acting on a plough body moving vertically in the ground and showed that the vertical force had two components, a steady state term which was a function of depth and a dynamic term equal to \( \frac{Hdy_0}{dx} \) where \( y_0 \) is plough depth and \( x \) is horizontal displacement. He was then able to derive expressions for the linkage forces and the response of the control.

He compared predicted ploughing depth with measured results from two tractors, one having an on-off control and the other a proportional control. Like previous workers he used ridge and furrow pasture land with the turf removed which provided a sinusoidal surface of wavelength 8 m and amplitude 75 cm. Depth was measured using a Cowley level and no significant differences were found between the performance of each control. Nor was the depth variation affected by small changes in the control response rates. More importantly however, agreement between measured and predicted draught force, vertical force and top link force was good enough to suggest that the theoretical analysis, particularly for the vertical force was valid. Agreement between measured and predicted ploughing depth was not however as good because the effect of the time delay inherent in the hydraulic lift was not included.

In later work, Dwyer (65) extended the theoretical analysis to
include the tractor dynamics and compared the performance of a top link and lower link sensing control. To provide a repeatable sinusoidal draught variation the tractor wheels were set eccentric by 5 cm and draught, vertical force, speed and wheelslip were recorded when ploughing and cultivating. His conclusions from the field measurements were that there were no significant differences between the top and lower link sensing controls, cultivators were more difficult to control than ploughs and higher control sensitivities would be required to retain present control performance at higher speeds.

He measured the basic control parameters of each tractor on a track treadmill (66) and obtained values of the deadband, rate of lift and sensitivity of the proportional response (rate of lift or lower per unit changed in sensed forces). He was then able not only to predict the draught and vertical force fluctuations when the tractor linkage was maintained in a fixed position, but also with reasonable accuracy when the draught control was operating. Having validated the analysis, it was possible to estimate the optimum parameters for the draught control and his results are summarised in fig 1.4. The performance criteria used was

\[
\frac{\text{draught amplitude with control operating}}{\text{draught amplitude with no control}}
\]

and is plotted on the ordinate in each case. He concluded that the optimum rate of lift was about 150 mm/s but this figure was already exceeded on most tractors anyway, and that a deadband of 500 N was acceptable. The effect of sensitivity shown in the top graph of fig 1.4 was calculated for top link sensing assuming zero deadband and an infinite maximum rate of lift and indicated that performance could be improved by increasing sensitivity. In fact, this implies that on-off control, where the sensitivity can be assumed to be infinite, provides the best performance. However this analysis took no account of the delay time in the hydraulics or its effect on control stability which would be the limiting factor in increasing response or sensitivity and decreasing the deadband.
Fig 1.4 Effect of control parameters on amplitude ratio calculated for a top link sensing control (after Dwyer(65))
1.2.5 Performance of experimental depth controls

Several attempts to provide depth control using electro-hydraulic components have been made (67-69). A depth wheel or skid is used to provide an electrical signal proportional to depth which then controls the operation of an electro-hydraulic valve to lift or lower the linkage. It is possible to filter out the higher frequency components (70) of the signal which are due to ground irregularities and leave only the low frequency components which represent gradual variations in the mean depth. Attempts to control depth in this way have generally been successful but have the disadvantages of high cost of the electro-hydraulics components and the need for another connection between tractor and implement.

In Germany, Hesse and Moller (71) have examined more sophisticated depth controls using one wheel at the front of a long plough and another at the rear. The signal from the front wheel was used to control lower link height and the signal from the rear wheel to control the length of a hydraulic ram which replaced the top link. In this way, the angle between implement and tractor could be altered to prevent changes in tractor pitch causing excessive depth variations particularly at the rear of a long implement. Again, although the control was successful, the increased cost and complexity were difficult to justify. Zevelev (72) in the U.S.S.R. has described a similar improved depth control which adjusts top link length using a hydraulic ram.

There is a problem with long or wide implements that small changes in tractor pitch or roll cause large depth variations particularly at the extremities of the implement. Also their ability to follow short wave-length ground undulations is inadequate in some field conditions where smaller implements are able to follow the surface contour better. One solution to this problem is to have a flexible implement (73), each part of which is controlled by depth wheels. This improves its surface-following ability but still has the disadvantage of other depth wheel controls in that no weight is transferred to the tractor.
In Japan (74), where rotary tillage is more common, a rotary tiller depth control was developed. Two parameters were sensed; depth by a depth wheel or skid and power take-off (p.t.o.) torque by a torque transducer. The electrical signals could be added in various proportions and then used to control the operation of an electro-hydraulic valve to lift or lower the linkage. It was concluded that the combination of both torque and depth signals provided better control than either of them separately. One interesting result was that torque was assumed to be proportional to both speed and depth, but if the depth was changing, then there was another term proportional to rate of change of depth to include.

\[ T = k_1 v y_0 + k_2 \frac{dy_0}{dt} \]  

(1.4)

where\( T \) = torque  
\( k_1, k_2 \) = constants  
\( v \) = forward speed  
\( y_0 \) = rotary tiller depth

This equation is analogous to the sensed force for top link sensing control (64), because the vertical force on a draught implement has a term proportional to rate of change of depth. Work is still being carried out on this control and so the effect of the rate term on control performance has not yet been established.

1.2.6 Performance of experimental draught controls

Besides the conventional controls available commercially there have been other attempts at designing a control where other forces proportional to draught force are sensed.

After measuring the individual linkage forces, Pisar (75, 76) concluded that control could be improved by sensing the sum of the link and right hand (looking from the back of the tractor) lower link force (77). He built an experimental control and showed that it resulted in a smaller standard deviation of draught and reduced fuel consumption compared with a top link sensing control. By adding the forces in the two links, the vertical force is eliminated. This control
is similar in principle to the pure draught sensing control proposed by Cowell (78) but differs slightly in that the sensed signal is not exactly proportional to draught because any side force on the implement has a component in one lower link force. If the forces in both lower links were added then any side force would be eliminated because it would appear as a positive component in one link and negative in the other.

The system proposed by Cowell (78) is shown in fig 1.5. His reasons for proposing this control were that the constraints imposed on the plough due to hitch geometry, traction motion and surface contour cause sensing control and impaired performance. This conclusion was the result of previous field work (7). From fig 1.5 the sensed force can be found by adding the forces in the lower links and the small angled link.

\[ F = \left[ 1 + \frac{y_H}{m} \right] H - \left[ \frac{x_V}{m} - \frac{a}{b \tan \theta} \right] V \]  

(1.5)

Therefore the vertical force components is eliminated if,

\[ \frac{x_V}{m} - \frac{a}{b \tan \theta} = 0 \]

i.e. \( \tan \theta = \frac{am}{bx_V} \)  

(1.6)

This analysis assumes that the links are parallel and the lift rods are perpendicular to them otherwise the force, \( F \), has other, albeit small components. For space reasons the extra linkage has to be small but then small spring deflections cause significant changes in the angle, \( \theta \), and again a small vertical component is re-introduced.

Cowell tested the principle of pure draught sensing by constructing an experimental control which was then used to control the depth of a single tine in a soil bin at Newcastle University. The tine was mounted on rollers restrained in the horizontal direction by a spring. Movement in this direction was proportional to draught force and was transmitted to a variable displacement hydraulic pump via a bowden cable. Oil flow from the pump was proportional to movement of the cable and so a proportional rate of linkage movement was achieved.
Fig 1.5 Pure draught sensing system proposed by Cowell (78)
This control was compared with a top link sensing control. A sinusoidal depth disturbance was applied using a specially designed rig (79) which consisted of a chassis on which a vertical motion carriage could be driven up or down by a scotch yoke mechanism. The tractor linkage was attached to the chassis and the tine to the carriage so the vertical movement of the carriage altered the depth of the tine.

Previously this rig had been used during measurements of draught and vertical force when a sinusoidal depth disturbance was applied in order to compare their dynamic and static values. The amplitude ratio \( \frac{\text{dynamic draught force}}{\text{static draught force}} \) was found to be unity over the frequency range 0.16 to 1.3 Hz and the phase shift was approximately zero. This was important for the theoretical analysis because if there had been any attenuation or phase shift under dynamic conditions it would have had to be included in the analysis as an implement transfer function.

Performance of the pure draught sensing control was better than the top link sensing control up to depth disturbances of 0.6 Hz after which it was similar. For a 5 cm amplitude disturbance at 0.3 Hz, the amplitude ratio \( \frac{\text{actual depth amplitude}}{\text{input depth amplitude}} \) was 0.63 for pure draught sensing and 0.68 for top link sensing. From the theoretical analysis, depth could be predicted to within 17% for the experimental control.

No attempt was made however to predict performance for the top link sensing control which would be more difficult because of the effect of the vertical soil forces. Also the experimental control used a double-acting ram to lift or lower the linkage. This system would not be feasible in practice because the weight on the tractor rear wheel could be decreased when the control was forcing the implement into the ground, especially for ploughs whose rate of penetration is limited by the mouldboard design. All commercial draught controls are single-acting to prevent weight being supported by the implement and not by the rear wheels.

In later experiments, the ram was altered to single-acting
operation and the leg of a plough restrained by a spring to provide a similar pure draught control to that used with the single tine. Field trials (80) indicated that control was again better than for top link sensing but response was limited by the rate of entry of the plough.

1.2.7 Performance of other experimental controls

Apart from the driveline torque sensing control which has been introduced commercially by one tractor manufacturer (16), there have not been many other attempts recorded of novel controls.

Engine speed sensing control has been suggested by one manufacturer (81) and in fact work done as part of this project prompted him to patent the idea. The best method of sensing engine speed appeared to be the fuel rack position on the governor and some position feedback would be necessary to prevent over-correction. It would also have the advantage that it could be used to control semi-mounted or trailed implements with external rams, or even p.t.o. driven implements providing a sufficient range of sensitivities was provided.

Other ideas have been considered, particularly when developing the driveline torque sensing control (82). These included sensing torque at other points in the driveline, i.e. the input to the transmission or the rear axle, rear axle housing bending, front axle load, wheelslip, pressure in a hydrostatic transmission. Some of the relative advantages and disadvantages of these suggestions are discussed by Wilson (82). His conclusions were that most of these controls were technically possible but all had functional limitations which made it impossible to select a 'best' system. Most controls had advantages for some particular implements or method of mounting but no control met the ideal specification of optimum performance with mounted, semi-mounted and trailed implements. In fact, the question of what was optimum performance was still a subject for discussion and the ideal control would provide an option of depth or draught control.

Hesse and Moller (83,84) investigated a control which did not strictly control either depth or draught. The inputs to this control
were signals proportional to either front axle load or pressure in a hydraulic ram fitted in place of the top link. These controlled the operation of an electro-hydraulic valve which in turn controlled the top link length. From field measurements with a four furrow plough fitted with a depth wheel at the rear to support some of the weight, they concluded that the control which sensed top link pressure was better and an accumulator in the hydraulic circuit was advantageous for removing high frequency pressure peaks. The disadvantages of this control are the same as for top link sensing controls in that the vertical force still has a component in the sensed signal. Also, since it does not control the height of the lower links, depth variations will occur when the tractor pitch changes as it crosses undulating land.

Another approach to the implement control problem is to allow the draught variations due to the implement but minimise the effect of them on the tractor. Ryan (85) has described a control fitted to a hydrostatic tractor which varies the overall gear ratio as the draught changes to maintain the engine at maximum power. This approach is discussed in more detail in section 1.3.3.

1.3 TRACTOR-IMPLEMENT PERFORMANCE

1.3.1 General

Attempts to analyse tractor-implement performance can be divided into two categories, those assuming steady state conditions and those assuming dynamic conditions.

In the steady state analysis, the assumption is made that the load acting on the tractor does not vary and its main use is for comparing the performance when a particular parameter is varied, for example, tyre size. It also has application for drawbar tests where conditions can be maintained constant. A special test track is usually used for these tests and load is applied by a loading vehicle capable of maintaining a steady load.

In field conditions, a dynamic analysis is more appropriate since
the load, rolling resistance and traction characteristics are continually varying and the relative effect of each varying parameter tends to vary according to the particular field condition.

This section is divided into a first part dealing with previous work on steady state performance and a second part with dynamic performance.

1.3.2 Steady state performance

Several workers have attempted to predict the optimum operating condition for a tractor and implement using steady state analyses (86-88). One of the more recent and comprehensive attempts is by Zoz (89) who included operating and capital equipment costs in his calculations to find the conditions for the least cost per unit area cultivated. He made the assumptions that tractive performance could be predicted from a simple soil measurement, the cone index value (90), and that the tractor was operated at a particular slip and coefficient of traction.

The main conclusion was that a cost/unit area within 5% of the minimum could be achieved for a wide range of tractor power levels and speed. For example, a 150 kW tractor at 3.2 m/s had the same cost/unit area as a 75 kW tractor at 1.3 m/s. Thus the analysis was not conclusive about any particular optimum operating point but was useful for predicting the effect of the many variables considered, for example, soil resistance, ploughing depth, tractor power to weight ratio.

The main criticism of this type of analysis is that although it involved many parameters it treated them in isolation. For example speeds of 1 to 4.5 m/s were used in the calculations but for reasons such as operator comfort, implement design or implement control, it is probably not possible to operate within the upper half of this range. No account is taken in the economic analysis of the possible extra cost of a tractor capable of operating at these speeds. The analysis is therefore limited to those conditions which can be extra-
polated from present levels involves no new technology, providing tyres capable of similar tractive efficiencies are available, and the cost of doing this can be predicted fairly accurately.

Other steady state analyses have been used to investigate the effect of particular parameters on performance, e.g. tyre size (91,92), weight transfer (93,94), tractor power levels (95,96) and forward speed (97).

Dwyer (91,91a), compared the performance of tyres measured by a single-wheel tester (98) with that measured during normal tractor field work (99). For two wheel drive tractors he found tractive efficiencies for the tractors was lower than that measured by the tester. The reason for this he concluded, was that the tyres fitted as standard to the tractor were too small. Using the lowest inflation pressure recommended for the tyre gives maximum tractive efficiency at a particular coefficient of traction (COT) usually about 0.4, where

$$\text{COT} = \frac{\text{Pull}}{\text{Weight on rear wheels}} \quad (1.7)$$

The rear wheel weight is limited by the recommended load at the lowest inflation pressure and so the pull is also limited by the above eqn 1.7. The pull calculated by this method results in too high a tractor forward speed if the tractor operates at maximum power. That is, the tractor cannot operate at that speed because of a deterioration in operator comfort, quality of work or implement control. Working at a higher pull will solve the problem because then maximum tractor power is used at a lower speed. But if pull is increased, rear wheel weight must also be increased (eqn 1.7)-to keep COT constant, and also inflation pressure must be increased to support the greater weight. This reduces tractive efficiency. The answer is to fit bigger tyres, then pull and ballast can be increased, but inflation pressure and tractive efficiency kept constant at their maximum values.

The reason that manufacturers continue to fit tyres which are too small is to keep the initial cost of the tractor down. Operators such as contractors whose use the tractor continually for high draught
operations often fit bigger tyres themselves. Published work on tractor power levels and operating speeds (95-97) has isolated tractor tyres as the limiting factor in producing high draughts since their overall efficiency is only about 70%. The limitations to working at higher speeds were summarised as operator comfort, quality of work and implement control. The main conclusions reached by these authors echoed the general trend in tractor design of increasing power levels and to a lesser extent increasing operator speeds.

Attempts in America to predict tractive performance from simple soil measurements have been generally in favour of the cone penetrometer method from which a cone index can be calculated (90,100). This method was used by Zoz (89) as described previously and it was also used by Berenyi (101) who analysed individual runs of tractor operation which represented one pass along the length of a field. Using an iterative computational scheme (101a) to match the required power from the implement and the available power from the tractor, Berenyi calculated the times taken for various runs, some of which included the effect of a sloping field. The time to accelerate up to speed and decelerate at the end was calculated but during the rest of the run the load was assumed to be constant.

1.3.3 Dynamic performance

Earlier work on the N.I.A.E. project described here had led Dwyer (65) to analyse tractor dynamic performance. He developed the basic equations for motion of the tractor-implement combination in the longitudinal direction and wrote a computer programme to simulate the tractor motion, which formed a basis for the programme developed in this work.

Smith (102) has also analysed the tractor motion in the longitudinal direction and he considered seven degrees of freedom: longitudinal and vertical displacement of the tractor centre of gravity, tractor pitch, angular rotation of the inner and outer parts of the driving
wheel, angular rotation of the engine and angular motion of the
governor fly-weights. He used Lagrangian dynamics to develop the
equations of motion and his main conclusion was that the tractor
behaviour could be simplified to a linear second order differential
equation, provided the engine remained in the governed range. For
a tractor in the over 75 kW range subjected to a drawbar pull,

\[ P = P_m - P_a (1 - \cos \omega t) \]

where \( P_m \) is the minimum pull and \( P_a \), half the peak to peak amplitude,
the velocity amplitude ratio was calculated. This ratio is defined as

\[ \frac{V_m - V_{\text{min}}}{V_m - V_a} \]  \hspace{1cm} (1.8)

where \( V_m \) = steady state velocity at a pull, \( P_m \)
\( V_{\text{min}} \) = minimum velocity during loading cycle
\( V_a \) = steady state velocity at a pull \((P_m + 2P_a)\)

It is plotted against drawbar loading frequency, \( \omega \), in fig. 1.6.

Up to about 0.2 Hz, the dynamic loading had little or no effect, the tractor behaved as if the load were applied statically at each instant of time. At 0.6 Hz the dynamic loading had the greatest effect and the minimum velocity reached during the dynamic cycle was lower than if the load had been applied statically. Above this frequency the dynamic loading had less effect and the minimum velocity reached during the dynamic loading was not as low as if the load had been applied statically. The lower curve on this graph shows the effect of the tractor engine having to operate in the non-governed region. Response is slower than for the other conditions shown where the engine always operates in the governed region.

Fig. 1.7 shows the variations in tractor forward speed plotted against time for \( P_a = 6.67 \text{N} \) and at loading frequencies of 0.2, 0.6 and 4 Hz. At 4 Hz the mean tractor velocity was equal to its velocity at
Fig 1.6 Effect of drawbar loading frequency on velocity amplitude ratio [after Smith (102)]

Fig 1.7 Effect of drawbar loading frequency on tractor forward speed [after Smith (102)]
a constant pull of \((P_m + P_a)\) which was the mean of the dynamic pull. The variations about the mean velocity are small and it is the minimum velocity when the response has settled down that is used as \(V_{\text{min}}\) in equation 1.8, rather than the actual minimum velocity at time \(t = 0.6\) s. The velocity overshoot from the initial condition where \(P - P_m\) is 69%. An important point when considering draught control performance is that although under these particular conditions it does not need to attenuate draught variations above 4 Hz, its response must be sufficiently fast to minimise any effects of velocity overshoot which occur when a load change is applied suddenly.

Ryan (85) developed a theoretical analysis of longitudinal tractor motion to design a forward speed for a hydrostatic transmission tractor. Engine speed was sensed by measuring fuel rack position and used to control the transmission ratio to vary forward speed. Thus the control could be used with either p.t.o. driven or draught implements because load variations of either type of implement cause changes in engine speed. Its disadvantage when used with draught implements was that performance may still be limited by wheelslip in conditions where the traction characteristics vary.

Field tests were carried out when operating a forage harvester and a plough. Response to the torque fluctuations of the p.t.o. driven forage harvester was satisfactory. The relatively low inertia in the driveline ensured that changes in the machine torque requirement were quickly sensed as engine load changed and hence fuel rack displacement. For ploughing, the control was tested with the tractor position and draught control operating. With position control, he found that response to the large draught variations was inadequate. The reason for this was probably the increased inertia in the system compared with the p.t.o. implement operation. Draught variations were not sensed as quickly as engine load changes because of the high inertia of all the moving parts involved, i.e. the tractor and implement mass, rear wheels and driveline. This high inertia also resulted in overshoot.
With draught control operating, response was much improved. Ryan claimed that this was because the load/speed characteristic was flat, i.e. a change in forward speed did not affect the load acting on the tractor. In position control, this was not the case because the changes in forward speed were sufficient to cause significant changes in plough draught as the speed varied. The implication of his conclusion is that the response of the draught control was faster than the speed control and load variations were attenuated rather than causing the forward speed control to operate. In these circumstances, the advantages of incorporating a forward speed control whose response is limited by tractor inertia would seem to be marginal providing the draught control is reasonably effective. Forward speed control is however of far greater benefit for p.t.o. operations where its response is not limited by large inertia and where there is no other control of the load variations imposed on the tractor.

1.4 CONCLUSIONS

A few general conclusions can be made about previous work on tractor and implement control performance.

For implement controls, most of the work has involved ad hoc measurements of either the more basic parameters such as the forces on implements and linkage forces or of overall performance of implement controls often using depth variation as a performance criterion. Analyses of draught control performance are few and those that have been developed are of limited application. Cowell (7,78) applied his analysis to pure draught sensing only and his reasons for choosing this type of control were not based on any conclusive theoretical comparisons with top or lower link sensing control. Dwyer (64,65) was able to predict the forces sensed by top or lower link sensing controls, but because insufficient measurements of the control response under controlled conditions were available, it was not possible to specify the optimum control parameters accurately. The exclusion of the effect of delay time for example meant that effectively there was no limitation to
increasing response rates and sensitivities other than excessive operation of the hydraulics. The work of Hesse and Moller (69-71) was restricted to electro-hydraulic controls, which may become more important in the future but are commercially unattractive at present. Their derivation of the transfer functions of some electro-hydraulic components are useful and are referred to later.

The conclusion can be drawn that no comprehensive analysis which embraces the overall implement control problem exists, despite the fact that much of the basic data, e.g. implement and linkage force measurements, are already available.

A similar situation exists for tractor performance. Most of the work has assumed steady state conditions and the analyses have often been used to examine the economics of various tractor-implement systems. This has involved making broad assumptions about tractor performance, such as the assumption that they can always be operated at maximum tractive efficiency, which field measurements (99) have shown to be often unattainable. Much of the present knowledge of tyre sizes, ballasting, power to weight ratios, effect of implement size etc. however, has resulted from these simplified analyses.

To investigate the effect of implement control on tractor performance, it is necessary to consider dynamic conditions but attempts to analyse tractor dynamic performance are either limited in scope or more concerned with other aspects of performance. Berenyi (101) just considered the tractor acceleration and deceleration at the start and end of a run, Smith (102) considered the effect of a sinusoidal draught variation only and Ryan's analysis (85) was more concerned with forward speed control than implement control.

Dwyer (65) started to investigate the effect of implement control on tractor performance but otherwise no attempts have been made despite the fact that one of the fundamental objects of an implement control is to optimise tractor performance by attenuating the draught fluctuations.
1.5 **OBJECTIVES**

The objectives of the programme of work at N.I.A.E. (103), of which this thesis reports a major part were:

a) To develop a method of analysing the operation of existing draught controls.

b) To compare the relative effectiveness of the different types of existing draught controls.

c) To examine methods by which better implement control may be obtained.

Work on objectives (a) and (b) was already in progress when the research reported in this thesis was initiated (104). The forces sensed by draught control units had been derived and some response rates were known (64). Field performance of commercial controls, both top and lower link sensing had been compared (65). A rig had been designed to measure control performance in laboratory conditions (105) and an experimental control on which control parameters could be varied had been built (105). No results however from either of these pieces of experimental equipment were available.

The third objective is the most relevant to this thesis, since an understanding of present controls should lead to suggestions for improving them. However, as mentioned previously, the implement control cannot be treated in isolation, since it is only part of a bigger system, the tractor-implement combination. Improving implement controls therefore, not only involves optimising their response but also their effect on tractor and implement performance.
2. TRACTOR-IMPLEMENT DYNAMICS

2.1 INTRODUCTION

The analysis of the tractor-implement combination is divided into four sections. In the first section, the dynamics of the tractor engine, driveline and running gear describe how the tractor responds in the longitudinal direction to changes in the external forces applied to it (107). The external forces are due mainly to the forces exerted by the implement which is analysed in the second section. The soil forces on both ploughs and cultivators and the dynamic motion of these implements in the soil are derived for various methods of mounting (107, 108).

In the third section, motion of the tractor and implement in the vertical direction is analysed and the interaction between tractor and implement in the low frequency range is discussed. This has important implications as tractor operating speeds tend to increase; because the increased tractor pitch and bounce cause problems not only of operator comfort but also implement control.

In the fourth section, a method of predicting the performance of tractor-implement combinations is outlined. A computer programme for predicting performance, based on the analyses of the previous sections, is described and the effect of draught fluctuations on overall performance is shown by comparing dynamic behaviour with steady state predictions where the draught is assumed to be constant.

2.2 TRACTOR DYNAMIC MOTION IN THE LONGITUDINAL DIRECTION

This analysis applies to a typical two wheel drive tractor which is representative of the majority of those used on British farms. The layout is probably well-known but briefly, a diesel engine, usually with a flyweight governor, drives two large rear wheels through a conventional clutch, gearbox and differential.

Longitudinal motion is governed by two basic equations, one obtained by balancing the torques acting at the rear axle and the other, the forces acting at the rear wheels in the horizontal direction (Fig 2.1)

At the rear axle, the torque, $T_g$, must overcome the torque due to the rolling resistances, $RR$, of the wheels and produce a thrust, $P$, at the rear wheels

$$I_{eq} \ddot{\phi} = T_g - (P + RR)r$$

(2.1)
Fig 2.1 Forces acting on tractor

Fig 2.2 Typical engine torque/speed characteristic
where \( I_{eq} \) is the inertia of the engine, driveline and rear wheels referred to the rear axle.

At the rear wheels, the thrust produced must overcome the draught force,

\[
M''_x = P-H
\]  
(2.2)

The torque produced by the engine is approximately proportional to rack displacement in the governed range (85) and is a function of engine speed, \( \dot{\theta}_e \), outside the governed range. The relationship of torque with engine speed varies with the engine and a typical example is shown in Fig 2.2

\[
T_e = f_\dot{\theta}_e (\dot{\theta}_e)
\]  
(2.3)

The engine speed is related to wheelspeeds by the overall gear ratio, \( \zeta \),

\[
\dot{\theta}_\omega = \zeta \dot{\theta}_e
\]  
(2.4)

Rack displacement, \( x_r \), is controlled by the governor flyweights, the equation of motion for which is (85)

\[
\frac{x_r^2}{\omega_n^2} + \frac{2x_r}{\omega_n} + x_r = k_j y_r - k_4 \dot{\theta}_e^2
\]  
(2.5)

where \( k_j \) and \( k_4 \) are constants depending on the flyweight mass, pivot arm length, fuel rack spring stiffness and set operating speed. The steady state governor performance is therefore:

\[
x_r = k_j y_r - k_4 \dot{\theta}_e^2
\]  
(2.6)

Knowing governor droop, that is, the difference in engine speed between full and no load at the same governor setting, \( y_{\omega} \), and the rack displacements at minimum and maximum torques, \( x_{\min} \) and \( x_{\max} \), for a given nominal engine speed, \( \dot{\theta}_{e1} \), the constants, \( k_j \) and \( k_4 \), can be found from the following:

\[
k_4 = \frac{x_{\max} - x_{\min}}{\dot{\theta}_{e1} - (\dot{\theta}_{e1} - \text{Droop})^2}
\]  
(2.7)

\[
k_j = \frac{x_{\min} + k_4 \dot{\theta}_{e1}^2}{y_r}
\]  
(2.8)

This approach is used to find \( k_j \) and \( k_4 \) because the droop and rack displacements are more easily measured than the flyweight masses, pivot arm lengths etc.

Equations 2.1 and 2.2 can now be re-arranged to give expressions for
Fig 2.3 Typical sup/coefficient of traction characteristics for three surfaces
forward acceleration and wheel acceleration, and by integrating these expressions, forward speed and wheelspeed can be found. The equation linking forward speed and wheelspeed defines the wheelslip, \( S \),

\[
S = \frac{\dot{x}_r - \dot{x}}{\dot{x}_r} \quad (2.9)
\]

The thrust, \( P \), at the rear wheels is a function of slip and the weight on the rear wheels (114).

\[
P = W_R f_4 (S) \quad (2.10)
\]

This relationship varies with the particular tyre and soil condition and is usually plotted as slip against coefficient of traction, \( \text{COT} \), where:

\[
\text{COT} = \frac{P}{W_R} \quad (2.11)
\]

A typical relationship is shown in fig 2.3 (114).

2.3 IMPLEMENT DYNAMICS

2.3.1 General

Two implements will be analysed in this section, a mouldboard plough and a rigid-tine heavy cultivator, since these are the most common draught implements used for primary cultivation. They both require high draught forces and are therefore important to control to ensure efficient tractor operation.

The soil forces acting on each implement are described and the resultant forces exerted on the tractor, which depend on the method of mounting the implement, can then be found. The behaviour of the implement moving vertically in the ground depends on the linkage, soil forces and method of mounting.

2.3.2 Plough

The forces acting on a plough as it moves through the soil are shown in Fig 2.4 (32). They can be resolved into two components, a draught force, \( H \), and a vertical force, \( V \). \( H \) is a function of implement depth,

\[
H = f_1 (y_o) \quad (2.12)
\]

and \( V \) can be found by resolving vertically,

\[
V = V' + W_1 - S_1 \quad (2.13)
\]

where \( V' \) is a function, \( f_2' \), of implement depth, rate of change of depth and draught force (64).
Fig 2.4 Forces acting on plough

Fig 2.5 Typical draught, vertical force/depth relationships (medium soil) for steady state conditions
\[ V' = f'_2(y_o) - H \frac{dy_o}{dx} \]

where \( x \) is displacement along the ground. Hence,

\[ V = f_2(y_o) - H \frac{dy_o}{dx} \] (2.14)

where \( f_2(y_o) \) includes the constant terms. Typical relationship of \( H \) and \( V \) with depth for steady state conditions, \( \frac{dy_o}{dx} = 0 \), are shown in Fig 2.5 \((108)\).

2.3.3 Cultivator

The forces on a cultivator are similar to those on a plough (fig 2.6), and typical steady state relationships of \( H \) and \( V \) with depth are shown in fig 2.7 \((108)\). The main differences from the plough forces are that there is no soil support force since there is no landside and the term, \( f_2(y_o) \), continues increasing with depth up to the point when the cultivator frame contacts the ground. For the plough, this term gradually decreases with depth as more of the plough weight is supported on the disc coulters, hubs and skims.

2.3.4 Effect of speed on implement forces

For a plough, the following equation has been found to represent the increase in specific draught with speed \((35-37)\)

\[ H_s = K1 + K2 \frac{x^2}{s} \] (2.15)

where \( K1 \) is the specific draught at zero forward speed and \( K2 \) is a function of \( K1 \), defined by Zoë \((86)\) for American conditions as

\[ K2 = 0.01K1 + 0.04 \] (2.16)

Since very little data as yet exists for British conditions, these relationships are assumed to be valid.

For a cultivator, draught increases linearly with speed

\[ H_s = K3 + K4 \frac{x}{s} \] (2.17)

where \( K3 \) and \( K4 \) are constants for a particular soil and cultivator. Typical curves of \( H_s \) against speed for a plough and cultivator are shown in fig 2.8 \((86)\).
Fig. 2.6 Forces on a Cultivator

Fig. 2.7 Typical steady state draught, vertical force/depth relationships for a cultivator
Fig 2.8 Effect of speed on the draught of a plough and cultivator (medium soil)
2.3.5 Methods of mounting

There are three conventional methods of attaching an implement to a tractor, (1) fully mounted, (2) semi-mounted and (3) trailed. The forces exerted on the tractor, weight transfer and sensed force at a top or lower link sensing draught control are analysed for each method of mounting. Also the effect of the linkage on the rate of entry of the implement into the ground is analysed;

(1) Fully mounted

Fig 2.9 is a diagram of the forces acting on a tractor and fully mounted implement. H has been found (32) to act somewhere between a depth of \( y_0 \) and \( y_0/2 \) depending on the tillage depth and soil condition. Throughout this work it is assumed to act at \( 3y_0/4 \). The line of action of \( V \) depends on the relative magnitudes of \( V, W_i \) and \( S_i \) and will therefore change under dynamic conditions as \( V \) varies. However, changes in the line of action are small compared with the distance \( l_2 \) (115) and \( V \) is assumed to act at a constant distance \( l_2 \) from the lower link ends. \( H \) and \( V \) can be resolved into a component \( I \) at an angle \( \theta_L \) to the horizontal.

Assuming the links are not supported, the instantaneous centre of rotation of the linkage is at \( i \) since all joints are pin jointed. If \( L \) does not pass through this point then the unbalanced moment \( L l_6 \) exists and the plough will raise or lower until an equilibrium is obtained when \( L \) passes through \( i \). In practice the lower links are restrained by the lift rods and the virtual hitch point (VHP) is then at the intersection of \( F_T \) and the resultant of \( F_R, F_L \) and \( F_V \), where \( F_T \) and \( F_L \) are the horizontal components of the top and lower link forces, \( F_V \) the vertical component of lower link force and \( F_R \) the lift rod force. The unbalanced moment of \( L l_6 \) can no longer be corrected by movement of the plough since it is constrained by the lift rods, so the force in the lift rods alters and moves the position of VHP so that \( L \) then passes through it. The advantage of fully mounting the implement is that weight is transferred to the tractor rear wheels and maximum tractive effort increased.

The weight transfer is calculated as follows:

(a) Tractor alone
Fig 2.9 Effect of implement on tractor weight distribution
Taking moments about rear wheel contact point,
\[ W_F = W \left( \frac{w - b}{w} \right) \]

Resolving vertically,
\[ W = W_F + W_R \]
Therefore \[ W_R = \frac{Wh}{w} \] \hspace{1cm} (2.18)

(b) Tractor and implement

Taking moments about rear wheel contact point,
\[ W_F = W \left( \frac{w - b}{w} \right) - Lh \]

Resolving vertically
\[ W = W_F + W_R - L \sin \theta_L \]
Therefore \[ W_R = \frac{Wh}{w} + Lh + L \sin \theta_L \] \hspace{1cm} (2.19)

Hence the overall effect of fully mounting the implement is to transfer a load \( Lh \) from the front wheels to the rear wheels and to add a load \( L \sin \theta_L \) to the rear wheels.

For a top link sensing tractor, the force in the top link is \( \frac{F_T}{\cos \alpha} \) and the sensed force is \( F_T \) assuming that the sensing unit is mounted horizontally. \( F_T \) is also the horizontal force acting at the top hitch point (implement end of the top link) Taking moments about the lower hitch point

\[ F_T = \frac{H}{1.3} \left( \frac{1}{5} - \frac{y_0}{4} \right) - \frac{V_{1.2}}{1.3} \] \hspace{1cm} (2.20)

for a lower link sensing tractor, the sensed force, \( F_L \), is found by taking moments about the top hitch point,

\[ F_L = \frac{H}{1.3} \left( \frac{1}{5} - \frac{y_0}{4} + \frac{1}{3} \right) - \frac{V_{1.2}}{1.3} \] \hspace{1cm} (2.21)

\( F_T \) and \( F_L \) have been calculated from the relationship of \( H \) and \( V \) with depth for a light soil (fig 2.10) and a heavy soil (fig 2.11). The figures refer to a three furrow plough.

Rate of Entry

The rate of entry of a mounted mouldboard plough is a function of draught, vertical force and a factor governed by plough geometry (107). From field
Fig 2.10 Draught, vertical force and force at top and lower links for a three furrow plough in light soil.
Fig 2.11 Draught, vertical force and force at top and lower links for a three furrow plough in medium/heavy soil.
measurements (107) it was found that the following empirical equation described the plough penetration
\[ y_o = A (1 - e^{-Bx}) \]  
(2.22)
This relationship was found to be independent of forward speed over a range of typical ploughing speeds as shown in fig 2.12 (107). Differentiating and re-arranging this equation gives the rate of entry in terms of the plough depth
\[ \frac{dy_o}{dx} = B (A - y_o) \]  
(2.23)
The constant, \( A \), is the maximum depth and can be calculated knowing the draught and vertical force relationships with depth and the linkage geometry. Maximum depth is reached when the resultant force, \( L \), passes through the instantaneous centre of rotation of the linkage. A FORTRAN computer programme has been written to calculate maximum depth of any plough and linkage combination (Appendix 2.II)

From the equation (2.23), the initial rate of entry
\[ \left( \frac{dy_o}{dx} \right) = BA \]  
(2.24)
\[ y_o = 0 \]
so that the plough factor, \( B \), can be calculated if \( A \) and the initial entry rate are known. From field experiments (107) the initial entry rate was found to be independent of soil condition but varied with the plough inclination as it first entered the ground. This angle is controlled by the top link length and a relationship between initial entry rate and plough inclination for a particular linkage is shown fig 2.13. The initial entry rate was also affected by the amount of wear of the plough share points, so the above relationship is only true if the share points are in good condition.

The draught and vertical forces are also affected by plough inclination (fig 2.14) which is most easily changed by altering the top link length. The greater the angle of the plough to the horizontal, \( \delta \), (defined in fig 2.16), the greater are both the draught and vertical force.

For cultivators, the rates of entry have been found to be constant over a wide range of field conditions (108). Entry rate does not vary with depth because the rake angle of the tine does not change significantly and so the rate remains reasonably constant until the plough frame contacts the ground.
Fig 2.12 Path of penetration of three furrow plough measured (solid lines) and predicted (dotted lines) at three forward speeds (107)
Fig 2.13 Effect of plough inclination on initial rate of entry
Fig 2.14 Variation in draught and vertical force with depth for three plough inclinations.
The actual value of entry rate depends again on the wear of the points and the
tine angle but assuming that the points are in good condition a typical value is
0.25 m/m (108).

(2) Semi-mounted

Fig 2.15 is a diagram of a tractor and semi-mounted plough. Resolving the
forces on the plough horizontally gives the force acting at the lower link
sensing unit.

\[ F_L = H + R_w \quad (2.25) \]

The vertical force is supported partly by the wheel at the rear of the plough
and partly by the lower links. Resolving vertically,

\[ V = V_L + V,w \quad (2.26) \]

and taking moments about the ground contact point of the plough wheel,

\[ V_L (1 - \frac{1}{p} - 2) = V_L \frac{1}{p} - H \frac{3y_o}{4} - F_L h \quad (2.27) \]

from which \( V_L \) can be calculated. The weight transferred to the tractor rear
wheels is then \( V_L (1 + e/w) \), calculated by taking moments about the tractor
front wheel ground contact point.

Rate of change of inclination

The rear wheel of a semi-mounted plough maintains the rear of the plough
at a constant height above the ground. Movements of the tractor lower links
raise or lower the front of the plough and alter its inclination, \( \delta \). From
fig 2.16,

\[ \tan \delta = \frac{y_{f1} - y_d}{l_{f1}} = \frac{y_{f2} - y_d}{l_{f2}} = \frac{y_{f3} - y_d}{l_{f3}} \quad (2.28) \]

\[ \delta = \frac{y_{fn} - y_d}{l_{fn}} \quad \text{for a plough with } n \text{ furrows} \quad (2.29) \]

For small changes in inclination, this can be approximated to

\[ \delta \approx \frac{y_{fn} - y_d}{l_{fn}} \quad (2.30) \]

Assuming that the equation describing the rate of entry of each furrow
Fig 2.15 Tractor and semi-mounted plough
Figure 2.16 Semi-mounted plough
is of the same form as eqn 2.23,

\[ \frac{dy_{f_n}}{dx} = B(A - y_{f_n}) \]  

(2.31)

Therefore the maximum possible rate of entry of each furrow is proportional to the furrow depth, \( y_{f_n} \). Providing that \( \delta > 0 \), which will be true for normal ploughing conditions,

\[ y_{f_n} > y_{f_{n-1}} > y_{f_{n-2}} \quad \text{etc} \]

and

\[ \frac{dy_{f_n}}{dx} > \frac{dy_{f_{n-1}}}{dx} > \frac{dy_{f_{n-2}}}{dx} \]

Therefore the maximum rate at which the plough inclination can change is proportional to the maximum rate of entry of the nth furrow. Differentiating equation 2.30

\[ \frac{d\delta}{dx} = \frac{1}{l_{f_n}} \frac{dy_{f_n}}{dx} \]

(2.32)

substituting for \( dy_{f_n} \) from eqn 2.31

\[ \frac{d\delta}{dx} = \frac{B}{l_{f_n}} (A - y_{f_n}) \]

and substituting for \( y_{f_n} \) from eqn 2.30

\[ \frac{d\delta}{dx} = \frac{B}{l_{f_n}} (A - \delta l_{f_n} - y_d) \]

(2.33)

From fig 2.13 the relationship between initial entry rate and plough inclination can be assumed linear over a small range of inclinations and is approximately,

initial entry rate = 1.33 \( \delta \) + 0.42

(2.34)

Hence \( B = \frac{1.33 \delta + 0.42}{A} \) from equation (2.24)

and substituting in eqn (2.33) gives

\[ \frac{d\delta}{dx} = \frac{(1.33 \delta + 0.42)(A - \delta l_{f_n} - y_d)}{L_{f_n}} \]

(2.35)
Fig 2.17 Variation in draught and vertical force with plough inclination at various depth wheel settings on semi-mounted three furrow plough.
Fig 2.18 Tractor and trailed implement
The draught and vertical force also vary with plough inclination and the relationship depends on the soil condition. A typical set of relationships for various plough depth wheel settings is shown in fig 2.17 (108)

(3) Trailed

Fig 2.18 is a diagram of a tractor and trailed implement connected to the drawbar. The vertical force is totally supported on the two depth wheels and the draught force exerted at the drawbar is,

$$H_d = H + R_{w1} + R_{w2}$$

(2.36)

Because the drawbar is a distance $h_d$ above the ground a force $\frac{H \cdot h_d}{w}$ is transferred from the front wheels to the rear wheels.

2.4 TRACTOR-IMPLEMENT DYNAMIC MOTION IN THE VERTICAL DIRECTION

Previous work (8) on tractor dynamic motion in the vertical direction has concentrated on the tractor alone in attempts to investigate its ride characteristics. This is perhaps surprising since tractors are nearly always operated with either an implement or trailer. In this section, tractor motion in the vertical plane and its interaction with implement motion are analysed. The analysis is restricted to cultivation implements since these are the most important to the implement control problem. Field results are presented to support the analysis and the implications for operator ride comfort and implement control particularly at high speed are discussed.

2.4.1 The tractor/implement model

Fig 2.19 is a diagram of the tractor and implement with the external soil forces acting on it. The four degrees of freedom, or generalised coordinates are:

- $z$ - tractor bounce
- $\phi$ - tractor pitch
- $\theta_1$ - rotation of lower links
- $\theta_2$ - rotation of implement

The tractor tyres, compliance in the hydraulic lift and top link sensing unit are all modelled by equivalent springs and dampers. Also the following assumptions are made:
Fig 2.19 Tractor and implement showing generalised coordinates and soil forces.
1) the top and lower links are parallel,
2) the lift rods are perpendicular to the lower links,
3) the links have negligible mass compared with that of the tractor or implement,
4) the spring $k_a$ is always in tension because the vertical force on the plough is always positive and is therefore considered to be double-acting, as in the damper $c_a$. If the vertical force on the plough was not always positive then the assumption would be invalid because tractor linkages are not restrained in the upward direction.

The two external soil forces acting on the implement, the draught force $H$ and the vertical force $V$ are defined in equations 2.12 and 2.14. The implement depth, $y_o$, is defined as a mean depth, $d$, and a displacement $z_p$ about the mean. Hence, assuming that eqn 2.12 is approximately linear for small depth changes

$$H = k_1 (d + z_p) \quad (2.40)$$

and equation 2.14 can be written

$$V = f_2 (d + z_p) - H \frac{dz_p}{dx_p} \quad (2.41)$$

The first term, $f(d + z_p)$ does not change significantly for small changes in depth, $z_p$, and can be approximated to a constant term $V = f(d)$. Also, assuming that $H$ remains constant at its value at the mean depth, $d$, $V$ can be approximated to

$$V = V_o - k_1 \frac{\dot{z}_p}{\dot{x}} \quad (2.42)$$

And for a constant forward speed

$$V = V_o - c_s \frac{\dot{z}_p}{\dot{x}} \quad (2.43)$$

The figures used in this analysis refer to a medium power tractor and three furrow mouldboard plough and values of all parameters used are given in Appendix 2.1

2.4.2 Derivation of the equations of motion

The method of Lagrange is used to obtain the differential equation of
motion of the tractor and implement. The problem must first be expressed in generalised coordinates \( q_i \). These are independent of one another and \( i \) is the minimum number of them necessary to specify any part of the system.

Lagrange's equation is a differentiation of the energy of the system written in the generalised coordinates

\[
\frac{\partial}{\partial t} \left( \frac{\partial T}{\partial q_i} \right) - \frac{\partial U}{\partial q_i} + \frac{\partial D}{\partial q_i} + \frac{\partial Q}{\partial q_i} = 0 \quad (2.44)
\]

where
- \( T = \text{kinetic energy} \)
- \( U = \text{potential energy} \)
- \( D = \text{dissipative energy} \)
- \( Q_{q_i} = \text{generalised external force for the } q_i \text{ coordinate} \)

This method has advantages for systems with multiple degrees of freedom because only the system energies need to be found. The forces of constraint which do no work but which often complicate the force-mass-acceleration equations are excluded. Also, the calculation of energies requires only the velocities and not the accelerations of the system components which are often more difficult to compute. Substituting each of the generalised coordinates in turn into equation (2.44) gives four equations of motion of the system.

To calculate the kinetic \((T)\), potential \((U)\) and dissipative \((D)\) energies we need the velocities of the main components. The tractor velocity is simple but the implement velocity, \( v_p \), is more complicated because it involves all the generalised coordinates. The vector diagram for the velocity components due to each coordinate is shown in fig 2.20

Hence,

\[
v_p^2 = \dot{x}_p^2 + \dot{y}_p^2 \quad (2.45)
\]

\[
v_p^2 = (\dot{\theta}_1 l_1 \cos \theta_1 + \dot{\theta}_2 l_2 \cos \theta_2 + \dot{\theta}_1 l_4 \cos \beta + \dot{\theta}_2 l_4 \cos \beta)^2
\]

\[
+ (\dot{\theta}_1 l_1 \sin \theta_1 + \dot{\theta}_2 l_2 \sin \theta_2 + \dot{\theta}_1 l_4 \sin \beta)^2 \quad (2.46)
\]

which becomes

\[
v_p^2 = \dot{\theta}_1 \dot{\theta}_1 l_1^2 + \dot{\theta}_2 \dot{\theta}_2 l_2^2 + \dot{\theta}_1^2 l_4^2 + 2\dot{\theta}_1 l_1 \dot{\theta}_1 + 2\dot{\theta}_2 l_2 \dot{\theta}_2 + 2\dot{\theta}_1 l_4 \dot{\theta}_4
\]

\[
+ 2\dot{\theta}_1 \dot{\theta}_2 \dot{\theta}_4 + \dot{\phi}^2 - \dot{\phi}^2 (\dot{\theta}_1 l_1 + \dot{\theta}_2 l_2 + \dot{\phi} l_4) \quad (2.47)
\]

assuming that \( \cos (\theta_1 - \theta_2), \cos (\theta_1 - \beta) \) and \( \cos (\theta_2 - \beta) \)

all equal unity because the angular displacements are small.
Fig 2.20 Calculation of resultant plough velocity, \( v_p \)
The energies of the system can now be calculated.

Kinetic energy \( T = \frac{1}{2}M_2^2 + \frac{1}{2}b_2^2 + \frac{1}{2}m_1v_2^2 + \frac{1}{2}I_2^2 + \frac{1}{2}I_1^2 \) (2.48)

Potential energy \( U = \frac{1}{2}k_x(z + b_1)^2 + \frac{1}{2}k_y(z - a_1)^2 \)

\[ + \frac{1}{2}k_a \frac{\theta_1}{L}^2 + \frac{1}{2}k_b \frac{\theta_2}{L}^2 \] (2.49)

Dissipative energy \( D = \frac{1}{2}c_x(\dot{z} + b_1)^2 + \frac{1}{2}c_y(\dot{z} - a_1)^2 \)

\[ + \frac{1}{2}c_a \frac{\dot{\theta}_1}{L}^2 + \frac{1}{2}c_b \frac{\dot{\theta}_2}{L}^2 \] (2.50)

Before substituting these into Lagrange equation (2.44) the external force \( Q_{q_1} \) for each coordinate must be known.

The external forces are most easily found by calculating the work done by the horizontal force \( H \) and the vertical force \( V \) for a small movement \( \delta q_1 \).

Note that positive work is done by the external forces on the system when the coordinate displacement is in the same direction as that in which the force acts.

For a displacement \( \delta z \)

\[
\text{work done} = Q_z \delta z = V \delta z
\] (2.51)

Where \( Q_z \) is the external force for the \( z \) coordinate.

\[
Q_z = V = V_o - c_s \delta \dot{z}^P = V_o - c_s (\dot{z} - \theta_1^L - \theta_2^L - \phi_1^L - \phi_2^L)
\] (2.52)

Constant terms, in this case \( V_o \), can be ignored because they only affect the static deflections of the springs and not the dynamic analysis. There is no term involving \( H \) because although \( H \) changes its magnitude and line of action for a movement \( \delta z \), there is no work done in the \( z \) direction.

For a displacement \( \delta \phi \)

\[
\text{work done} = (-H(h_1^g + d + z_1^p) - Vl_4) \delta \phi
\] (2.53)

\[
Q_\phi = -H(h_1^g + d + z_1^p) - Vl_4
\] (2.54)

Substituting for \( H \) and \( V \) gives

\[
Q_\phi = -k_1 (d + z_1^p) (h_1^g + d + z_1^p) - (V_o - c_s \delta \dot{z})^L_4
\] (2.55)

Ignoring constant terms as before and also second order terms,

\[
Q_\phi = -k_1 (2d + h_1^g) (\dot{z} - \theta_1^L - \theta_2^L - \phi_1^L - \phi_2^L) + c_s l_4 (\dot{z} - \theta_1^L - \theta_2^L - \phi_1^L - \phi_2^L)
\] (2.56)

Using a similar analysis for a displacement \( \delta \theta_1 \).
Work done = \(- V_1 \dot{\theta}_1\) \hspace{2cm} (2.57)

\[ Q_{\theta_1} = \frac{c_s}{s} \left( \ddot{z} - \ddot{\phi}_1 \dot{\theta}_1 - \dot{\theta}_1^2 \right) \] \hspace{2cm} (2.58)

and for a displacement \(\delta \theta_2\)

Work done = \((- H (h_p + d + z_p) - V_2) \delta \theta_2\) \hspace{2cm} (2.59)

\[ Q_{\theta_2} = -k_1 (2d + h_p) (z - \phi_1 \dot{\theta}_1 - \dot{\theta}_1^2) + c_s (z - \phi_1 \dot{\theta}_1 - \dot{\theta}_1^2) \]

The expressions for \(T, U, D\) and \(Q_{\theta_1}\), may now be substituted in equation (2.44) for each generalised coordinate to give four equations of motion of the system. Since this is a rather lengthy process, an example for the \(z\) direction only is given below

\[
\begin{align*}
(M + m) \dddot{z} &- \frac{m_1}{4} \dddot{\theta}_1 - \frac{m_2}{4} \dddot{\theta}_2 + k_f (z + b \phi) + k_r (z - a \phi) \\
&+ c_f (z + b \phi) + c_r (z - a \phi) = - c_s \ddot{z} + c_s \phi_1 \dot{\theta}_1 + c_s \dot{\theta}_1 \dot{\theta}_2 (2.60)
\end{align*}
\]

Re-arranging gives

\[
(M + m) \dddot{z} - \frac{m_1}{4} \dddot{\theta}_1 - \frac{m_2}{4} \dddot{\theta}_2 \\
+ (c_f + c_r + c_s) \ddot{z} + (b c_f - a c_r - c_s \phi_1) \ddot{\theta}_1 - c_s \dot{\theta}_1 \dot{\theta}_2 \\
+ (k_f + k_r) z + (b k_f - a k_r) \phi = 0 (2.61)
\]

Having obtained the four second order differential equations of motion in this manner, the natural frequencies (eigen values) and mode shapes (eigen vectors) can be calculated.

2.4.3 Calculation of the natural frequencies (eigen values) and mode shapes (eigen vectors)

- The four previously derived equations can be written in matrix form as

\[
\begin{bmatrix} A \end{bmatrix} \dddot{u} + \begin{bmatrix} B \end{bmatrix} \ddot{u} + \begin{bmatrix} C \end{bmatrix} u = 0 \hspace{2cm} (2.62)
\]

where the vector \(u\) represents the coordinates of the system, in this case, \(z, \phi, \theta_1, \) and \(\theta_2\). \([A]\) is the 'mass' matrix, \([B]\) the 'damping' matrix and \([C]\) the 'displacement' matrix.

The computer programme used to solve this problem (127) requires 1st order equations. So the four 2nd order equations must be written as eight 1st order. A new set of subsidiary variables must therefore be defined.

\[
\frac{d}{dt} (u) = \ddot{u} \hspace{2cm} (2.63)
\]
Table 2.1  Natural frequencies (Hz) and approximate percentage critical damping (in brackets) of various tractor, plough and soil force combinations. Arrows indicate coupling between coordinates.
The system equations can now be written

\[
\begin{bmatrix}
A \\
I
\end{bmatrix}
\frac{d}{dt} \begin{bmatrix} \ddot{x} \\ \ddot{\phi} \end{bmatrix} + \begin{bmatrix} B \\
0
\end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{\phi} \end{bmatrix} + \begin{bmatrix} C \\
0
\end{bmatrix} \begin{bmatrix} x \\ \phi \end{bmatrix} = 0
\]  
(2.64)

\[
\begin{bmatrix}
I
\end{bmatrix}
\frac{d}{dt} \begin{bmatrix} \ddot{\phi} \\ \dot{\phi} \end{bmatrix} - \begin{bmatrix} I \\
0
\end{bmatrix} \begin{bmatrix} \dot{\phi} \\ \phi \end{bmatrix} = 0
\]  
(2.65)

where \( I \) is the unit matrix. The equations now only involve \( A \) and \( u \) so that by defining a variable \( v = \begin{bmatrix} \ddot{\phi} \\ \dot{\phi} \end{bmatrix} \), they can be described by one equation

\[
\begin{bmatrix}
A' \\
0
\end{bmatrix}
\frac{d}{dt} \begin{bmatrix} \dot{v} \\ v \end{bmatrix} + \begin{bmatrix} B' \\
0
\end{bmatrix} v = 0
\]  
(2.66)

\([A']\) and \([B']\) are 8 x 8 matrices and \( v \) in this particular case =

\[
\begin{bmatrix}
\dot{\phi} \\
\dot{\phi} \\
\dot{\phi} \\
\dot{\phi}
\end{bmatrix}
\]

The computer programme requires the matrices \([A']\) and \([B']\) as input and they are derived for the tractor and implement problem in Appendix 3.5.I

2.4.4 Results

Table 2.1 is a summary of the natural frequencies and percentage critical damping values for various tractor, plough and soil force combinations. The natural frequency is the imaginary part of the eigen value calculated by the computer programme and the percentage critical damping is approximately the real part of the eigen value divided by the imaginary part. The left hand column indicates the coordinate which had the greatest displacement amplitude at that frequency. At a particular frequency vibration occurs in all coordinate directions and if the vibration amplitudes in two directions are similar, coupling is said to occur between those coordinates. Where significant coupling occurred an arrow marks the other coordinates involved. This information is found from the eigen vectors.

The most important result in this table is the change in damping and natural frequency of tractor pitch between the tractor alone, and when it is ploughing. The percentage critical damping increases from 0.04 to 0.67, which indicates that the vertical soil forces on the implement are exerting a higher damping force on tractor pitch motion and the natural frequency decreases from
4.0 to 1.4 Hz. Thus tractor ride vibration is significantly different when ploughing from when the tractor is alone.

The plough is also affected when attached to the tractor. This can be seen by comparing the results for the plough alone with those when the tractor is ploughing. The natural frequencies of the plough and linkage alone assuming soil forces are acting are 6.4 and 2.9 Hz, with damping ratios of 0.12 and 0.04. When the tractor is ploughing, these become 6.7 and 3.6 with damping ratios of 0.20 and 0.08.

So both tractor and plough behaviour are altered when they are combined i.e., when the tractor is ploughing. This has two important effects. Firstly ride vibration will be reduced when ploughing because tractor pitch and bounce are damped by the vertical force on the plough. Secondly, the damping forces are transmitted by the tractor linkage and since the linkage forces are sensed by the draught control, it will receive spurious signals which are not due to changes in implement draught. Also, for vertical motion, the human body is most sensitive to frequencies of 4 to 8 Hz (124). The natural frequency in pitch for the tractor alone of 4.0 Hz is in this range, whereas when ploughing the frequency of 1.4 Hz is further away, in the range where the body is slightly more tolerant.

2.4.5 Other implements

This analysis shows that a plough significantly affects tractor behaviour and it is likely that other implements will also modify tractor behaviour though perhaps not to such an extent. Cultivators for example have similar soil force characteristics to ploughs except that the first term in the vertical force relationship which was assumed constant for the plough, is significant. Steady state vertical force increases with depth, so it opposes the damping term in the vertical force equation. In fact, if it is greater than the second term, negative damping would result and the system would be unstable. The damping term is proportional to (Draught force \times \text{rate of change of depth}) so, as forward speed increases, the damping term decreases and stability decreases. The draught control response also affects stability, because when the linkage is moved, the vertical force changes, due to the rate of change of depth term. It is possible therefore for the draught control response to excite a natural frequency of the tractor and implement and increase tractor ride vibration.
In general, any implement which is mounted on the tractor linkage will alter the tractor natural frequencies especially in pitch, because the implement weight acts well behind the tractor centre of gravity. Two-wheel trailers which transfer weight on to the tractor drawbar would also be expected to affect tractor ride significantly. There is scope for further work to investigate the effects of implements and trailers on tractor behaviour.

2.4.6 Field measurements

The increased damping of a tractor when it is operating a mounted plough suggests that tractor ride vibration levels when ploughing will be lower than for the tractor alone and a recent survey (123) supports this conclusion. Measurements of driver vibration levels in 71 tractors engaged in a variety of field work showed that ride levels when ploughing tended to be lower than for other operations (fig 2.21). In this survey, the N.I.A.E. ridemeter (124) was used to measure the acceleration levels at the driver's seat. This instrument is frequency-weighted so that its vibration response characteristic closely matches the response of the human body and it provides an average normalised ride level over a test period from 20 s to 5 min.

This survey also indicated that there was some correlation between ride level and plough size. When the ride level measurements of 15 tractors which were ploughing were plotted against implement size, there was a trend of decreasing ride vibration level with increasing implement size (fig 2.22). This would be expected because the vertical damping force on the implement is proportional to draught force, and also the line of action of the force acts further away from the tractor and its movement arm increases as implement size increases.

Since these results were measured over a very wide range of tractors, field surfaces, implements and tractor drivers, they could not be used to draw conclusive evidence for a particular case, such as the decreased ride level when ploughing. Further fieldwork was necessary to investigate the effects of ploughing on tractor ride levels.

Four fields were used for further measurements (Table 2.2). Acceleration levels at the driver's seat in the vertical, lateral and horizontal directions were measured with the N.I.A.E. ridemeter (124). In each field, ride level
Fig 2.21 Vertical ride vibration levels measured at the drivers seat for various farm operations.
Fig 2.22 Decreasing vertical ride vibration level with increasing plough size (results averaged for speeds between 1.3 and 2.3 m/s)
<table>
<thead>
<tr>
<th>Field number</th>
<th>Soil type</th>
<th>Surface</th>
<th>Approximate soil resistance at 230 mm depth N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>clay loam</td>
<td>stubble</td>
<td>0.083</td>
</tr>
<tr>
<td>2.2</td>
<td>clay loam</td>
<td>stubble</td>
<td>0.083</td>
</tr>
<tr>
<td>2.3</td>
<td>sandy loam</td>
<td>stubble, disced 50 mm deep, limed</td>
<td>0.041</td>
</tr>
<tr>
<td>2.4</td>
<td>sandy loam</td>
<td>stubble</td>
<td>0.055</td>
</tr>
</tbody>
</table>

Table 2.2 Fields used for ride level measurements
measurements were made with the tractor ploughing and for the tractor without the plough but running in the furrow at various speeds. The test period for one measurement varied depending on the field width but for each speed at least six measurements were taken and averaged. All measurements were made with a medium power tractor (63 kW pto) and three furrow plough. In field 2.3 measurements were also made with a five furrow plough, both fully mounted and with some weight carried on the depth wheel.

The field measurements were made of the acceleration levels of the driver measured at his seat, and therefore included the effect of the seat suspension. The theoretical analysis did not include the seat suspension but this is not considered to be important for this work since no attempt is being made to predict the actual ride vibration levels which would certainly require information about the seat suspension and also the ground surface. The trends of the effect of the plough on tractor vibration are the important aspects of this work so the inclusion of the seat suspension in the theoretical analysis is not really necessary. Care was however taken to ensure that the seat suspension was not changed during the experiments and the same tractor driver was used throughout.

Tractor vertical vibration levels for fields 2.1 and 2.2 are plotted against forward speed in figs 2.23 and 2.24. The measurements with the tractor alone are at a slightly higher speed than when ploughing because the wheels are not slipping. The results for the two fields are very similar, and at 2.2 m/s, vertical vibration level is decreased by about 50% due to the damping effect of the plough.

Fig 2.25 is a graph of vertical vibration level against speed for field 2.3 where ploughing speeds up to 2.8 m/s were used. The ride level for the tractor alone shows a very different trend to fields 2.1 and 2.2, where the ride level became increasingly high as speed increased. Above about 2.5 m/s, the curve flattens out and ride level remains fairly constant up to 3.1 m/s. The ride level when ploughing is consistently lower, about 25% lower at 2.5 m/s for example, but the curves look as if they will coincide at about 3.1 m/s. The reason for the flattening off of the ride level may be that the surface was fairly loose and deformable because it had been disced. As the tractor crossed the surface, the soil deformation provided a certain amount of damping which
Fig 2.23 Vertical ride vibration levels measured in field 2.1
**Fig 2.24 Vertical ride vibration levels measured in field 2.2**
Fig 2.25 Vertical ride vibration levels measured in field 2.3
had more effect on ride level at higher speed.

The results for field 2.4 shown in fig 2.26, show a similar trend. Although the surface had not been disced, it was fairly soft and deformable and the tractor left obvious wheel ruts where it had travelled. Vertical vibration level in field 2.4 was consistently about 20% lower when ploughing over the speed range from 1.8 to 2.8 m/s.

In field 2.3 measurements were made using a five furrow plough, and the results are compared with those for a three furrow plough in fig 2.27. At 2.2 m/s ride vibration is about 30% lower with the five furrow plough than with the three furrow plough. This is because the vertical damping force on the plough is proportional to draught force, which is greater with the five furrow plough and also its line of action is further back and hence the moment acting on the tractor is greater.

The measurements made with the fully mounted five furrow plough were compared with those made when a depth wheel was attached to the rear of the plough to assist depth control. Some of the plough weight was carried by the wheel and hence not all the vertical force on the plough acted on the tractor. This would be expected to reduce the damping effect of the plough and the results in fig 2.27 shows that it did. The ride level with the fully mounted plough was about 25% lower than when the plough was fitted with a depth wheel at the rear.

So far only vibration in the vertical direction has been discussed. In the longitudinal and lateral directions, no conclusive differences in ride level were found between the tractor alone and when ploughing. In fields 2.2 and 2.4 there was some evidence that the longitudinal vibration might be reduced slightly at the highest speeds (fig 2.28), but more results and preferably at even higher speeds would be necessary to confirm this. Longitudinal vibration may be expected to be reduced because the plough damps tractor pitch motion and since the driver is not seated at the centre of pitch, his motion in the longitudinal direction should be affected.

Some measurements were made with the tractor running on the land out of the furrow and without a plough. The ride levels in all three directions tended to be higher than when the tractor was in the furrow. Not enough results were
Fig 2.26 Vertical ride vibration levels measured in field 2.4
Fig. 2.27 Effect of plough size and depth wheel on vertical ride vibration levels in field 2.3
Fig 2.28 Longitudinal vibration levels measured in fields 2.2 and 2.4.
taken to be conclusive but this trend might be expected since in the furrow two of the tractor wheels travel on a smoother surface than on the land.

2.4.6 Discussion

The main conclusion of this analysis, namely that tractor motion in the vertical direction is significantly affected when ploughing, has several implications for future tractor and implement control designs.

Tractor ride vibration levels increase with speed and investigations are currently being carried out to devise methods of reducing vibration or protecting the driver from the high levels likely in the higher speed operations of future tractors. Considering the amount of work that has been carried out on this and tractor ride in general it is surprising that to date there has been no published work on the effects of implements or trailer on tractor ride. This work suggests that tractor behaviour is modified significantly certainly when ploughing and probably although perhaps to a lesser extent, with other implements.

The damping forces exerted by the plough on the tractor are transmitted by the linkage and so draught controls which sense linkage force receive spurious signals due to the damping forces. These signals cannot be filtered out because they are in the same low frequency range as the signals due to draught changes. Since the damping forces are greater at higher speed due to increased tractor ride vibration, the problem becomes worse as speed increases. This is discussed further in later sections.

2.4.7 Conclusions

1. A theoretical model of a tractor-implement combination showed that when ploughing, the plough exerted considerable damping on the tractor motion.

2. Field measurements confirmed this prediction and tractor vertical ride vibration level measured at the drivers seat was reduced by up to 50% when ploughing heavy land and 30% on light land.

2.5 COMPUTER SIMULATION OF THE FIELD PERFORMANCE OF TRACTOR-IMPLEMENT COMBINATIONS

The theoretical analysis outlines in the previous sections forms the basis of a computer programme to simulate the field performance of tractor-implement combinations. The programme written in C.S.M.P. is shown in Appendix 2.II, together with a specimen input and output. A detailed report (110)
describing how to run the programme has been made available to tractor manufacturers, several of whom now have copies.

2.5.1 General

In the programme the tractor is assumed to follow a surface contour which is read in as a series of elevation points. The coordinates of the surface were measured by Matthews (125) in a field which would typically be cultivated. To simulate fields of varying surface roughness, the same points are used and their vertical coordinates are scaled up or down. This has the effect of altering the height of their power spectral density (p.s.d.) curve but not its shape. This technique is considered valid for a set of similar field surfaces, defined here as those on which primary cultivation would be carried out, because typically the shapes of their p.s.d. curves are similar (125).

By altering the input data to the programme the effect of various parameters can be studied. These include, for example, different implements, methods of mounting, types of control and control parameters. Most of these, however, are discussed in chapters 3 and 4 where measured and predicted field performance are compared.

In this section, the tractor motion in the longitudinal plane will be analysed using the computer simulation. The review of previous work showed that the dynamics of tractor motion had not received much attention. Most attempts to predict tractor performance had assumed steady state conditions, that is that the load acting on the tractor was constant. In fact, in field conditions the load is continuously varying and the effect of the varying load is analysed by comparing steady state and dynamic performance.

2.5.2 Steady state performance

A 52.5 kW tractor is used for these calculations since this size of tractor was used for most of the field measurements described later. The maximum power available at the driving wheels is shown in fig 2.29 for three arbitrary surfaces. The traction characteristics of the tractor tyres on each surface are defined by the slip/coefficient of traction relationship (114). Assuming an infinitely variable gear ratio the equation for output power is (111)

$$\text{Power} = \frac{P T_{\text{max}} \dot{\theta}_{\text{e max}}}{P + RR} \eta (1-S)$$

(2.67)
Fig 2.29  Power, slip and $\cot(1 - \text{slip})$ relationships with coefficient of traction for three field surfaces
where $P$ and $S$ are given by the slip/COT relationship (eqn. 3.10-11),

$T_{\text{max}}$ and $\theta_{\text{e max}}$ are engine torque and speed at maximum power and $\eta$ is the driveline efficiency. For each field condition peak power coincides with maximum tyre efficiency which occurs at a coefficient of traction of between 0.35 in poor conditions and 0.4 in good conditions. This analysis suggests that peak power in this range of field conditions could be obtained within 2-3% by choosing one gear ratio so that the power curve for that gear (dotted line) passes as near as possible to the peak power point.

Peak power is then limited by engine power and if engine power were increased, peak power would occur at the same coefficient of traction but at a higher forward speed.

In a lower gear the tractor is capable of a higher pull, and the maximum tractor power available lies on a curve to the right of the peak power point. In this region maximum power may be limited by wheel slip and is best indicated by plotting COT(1-Slip) against COT. The factor COT(1-Slip) is proportional to tyre output because when multiplied by weight on the rear wheels and theoretical forward speed without slip, it gives the theoretical power available in a particular gear assuming unlimited engine power is available. From fig 2.29 it can be seen that peak tyre output occurs at a coefficient of traction of about 0.6 in good conditions and 0.45 in poor conditions.

The optimum operating point for the tractor is when the rate of work is a maximum. The reason that maximum rate of work does not coincide with maximum efficiency is that there is a penalty of increased draught force at higher speed which means that more energy is expended tilling the same volume of soil. Maximum rate of work occurs at a point somewhere between maximum tyre efficiency and maximum tyre output.

The difference between these two points is more marked in good conditions where maximum efficiency occurs at COT = 0.4 and peak tyre output at COT = 0.6, than in poor conditions where the corresponding figures are 0.35 and 0.45.

Using the relationship for the increase in specific draught of a plough with speed described previously (eqn 2.15) the required tractor power to operate at increasing speed can be calculated. This is shown in fig 2.30, the figures used referring to a three furrow plough in light land. It shows, for example,
Fig 2.30 Effect of forward speed on specific draught, work rate and required tractor power for three furrow plough in light land.
that to increase the typical present ploughing speed of about 1.8 m/s by 50%, requires an increase of 61% in tractor engine power.

2.5.3 Dynamic performance

In practice, the steady draught assumed in the foregoing analysis is impossible and draught fluctuations due either to changes in implement depth as the tractor pitches or variations in soil resistance cause fluctuations in engine speed and forward speed. These fluctuations result in power losses because the tractor is not working at the optimum conditions assumed in the steady state calculation. This is illustrated in fig 2.31 which shows the power available at various stages in the tractor driveline. The light land condition of fig 2.29 is used so the steady state power output curves are identical in the two figures. Losses due to driveline efficiency, rolling resistance and wheelslip all contribute to the fact that only 71% of the engine power is predicted to be available at the tyres assuming steady state conditions. In fact, because of the dynamic losses described above even less power is available as useful work and the dynamic power output lies below the steady state curve as shown. The calculation of the dynamic curve is described below.

To calculate the dynamic performance of a tractor, a set of gear ratios is estimated and read into the programme with all the other data describing the tractor, draught control, implement and field. The dynamic performance in terms of acres cultivated per hour is then calculated in each gear for different implement sizes, and the figures are marked on the steady state power curves in fig 2.32. A 52.5kW tractor operating a plough has been used for this example, so the steps between each figure along the curve for a particular gear corresponds to an extra furrow being added. If a cultivator were used the steps as an extra tine was added would be smaller.

As the implement size is increased for a particular gear, the work rate increases, reaching a maximum before the maximum power in that gear, and then decreasing because the fluctuations about the mean pull slow the tractor down. If the implement size is further increased the tractor stalls due to the load fluctuations, even though under steady state conditions it may be capable of operating at that mean pull.
Fig 2.31 Typical power losses for a 52.5 kW tractor when ploughing light land
Fig 2.32. Calculation of maximum dynamic rate of work on a good surface.
By simulating several gear ratios and implement sizes, the maximum rates of work in each gear can be joined to give a curve of maximum rate of work under dynamic conditions against coefficient of traction. It is shown as a dotted line in fig 2.32 and from this curve for a particular tractor and field condition, the optimum pull, speed and gear ratio can be calculated to coincide with the maximum point on the curve. The power utilisation is only about 91% of the steady state power available and the tractive efficiency is only 65% whereas in steady state conditions it is 71%. This probably explains the lower tractive efficiencies measured for tractors during ground drive comparisons (126) where the draught varied as in normal field conditions, compared with those measured during single wheel tests where a constant draught could be maintained (114).

The same curves are plotted in fig 2.33 for poor surface conditions. Maximum work rate occurs at a COT of about 0.3 and in a higher gear ratio than for the good surface, indicating that large losses due to wheelslip can be expected under dynamic conditions. The tractive efficiency is only 51% compared with its steady state value of 59%.

Unfortunately the procedure used to calculate these curves is rather lengthy and impractical. However, for a typical commercial top link sensing draught control where $D_B = 1800 N$ and $f_{o\text{max}} = 0.3 m/s$, an empirical relationship based on computer results (108) can be defined to calculate the approximate standard deviation of draught and the pull at maximum dynamic work rate.

\[
P(\text{at max dynamic work rate}) = P(\text{at max steady state work rate}) - 1.5 \text{ SD}
\]

where standard deviation of draught,

\[
SD = \frac{nH_s}{1550} \left(2.5 + 4.15 \xi^2\right) \tag{2.69}
\]

where $n =$ number of furrows

$H_s =$ specific draught ($N/m^2$)

$\xi =$ forward speed (m/s)

This relationship is only true for average field conditions and has only been validated practically up to about 2.8 m/s (108).
MAXIMUM TRACTIVE POWER

MAXIMUM DYNAMIC RATE OF WORK

DYNAMIC RATE OF WORK (ha/h)

INCREASING PLOUGH SIZE

FIG 2.33 CALCULATION OF MAXIMUM DYNAMIC RATE OF WORK ON A POOR SURFACE
It is only therefore intended as a guide to estimate the difference between predicted steady state and dynamic power and how parameters such as plough size, speed, ballast, gear ratio etc. affect the difference.

2.5.4 Effect of tractor parameters on dynamic performance

The previous analysis raises some interesting points about how various parameters affect tractor dynamic performance.

**Draught control**

Improvements to draught controls which reduce the draught variation result directly in increased work rates because the tractor can be operated nearer to its peak steady state power. The potential of possible improvements and their effect on the draught variation and hence overall performance are discussed in further detail in sections 3 to 5. The individual effect of control parameters on performance is also discussed.

The question of how much implement controls should be improved is largely an economic question and this will also be discussed later after the various controls have been analysed. Very simply, there is a choice of strategy of whether to improve implement control to enable the tractor to operate at maximum efficiency or whether to merely install more engine power and ballast to overcome the draught fluctuations and accept a decreased tractive efficiency. The latter method is introduced in the next section.

**Tractor power to weight ratio**

For steady state conditions the optimum power to weight ratio can be calculated from engine power $x \eta_t$ $x \eta = \text{COT} \times W_R \times \text{no-slip speed} \times (1-S)$ \hspace{1cm} (2.70)

Assuming the following figures for good field conditions:

- maximum tractive efficiency, $\eta_t = 0.75$
- slip at maximum $\eta_t = 0.1$
- COT at maximum $\eta_t = 0.4$
- driveline efficiency $\eta = 0.95$

Required tractor power can be plotted against forward speed (fig 2.34). The upper line is the predicted relationship assuming dynamic conditions using a three furrow plough on light land. In these conditions maximum power is limited by engine power, and as speed increases an increasing amount of reserve engine power is required to overcome the effect of the increased draught fluctuations.
Fig 2.34 Required tractor power to operate a 3 furrow plough in light land.
In poor conditions where power may be limited by wheelslip the decrease in weight/power ratio is not as marked because as well as increased engine power, increased ballast is also required to minimise wheelslip variations caused by draught changes.

As an example of the necessity of considering dynamic behaviour when designing tractor power to weight ratio, consider a possible future high speed tractor. The required steady state power for say a three furrow plough can be calculated from fig. 2.34. But if the calculated dynamic power is measured from the graph, it can be seen that at, say, the maximum useful ploughing speed (with the particular plough) of 3.35 m/s, the predicted dynamic power required is 34% greater than the steady state power.

**Gear ratios**

The range of gear ratios necessary to achieve the maximum dynamic work rate over a range of field conditions can be calculated, although it will also be influenced by the suitability of the ratios for other tasks. Taking the examples in figs. 2.32 and 2.33, the gear ratio for maximum dynamic work rate is 70 (engine rev/wheel rev) for the good surface and 60 for the poor surface. This suggests that two or preferably three ratios would be sufficient for a fairly wide range of fields. The power lost if the gear ratio does not coincide with the peak of the curve depends on how flat the curve is around its peak. In good conditions the curve (fig. 2.32) is flatter than for poor conditions (fig. 2.33) and the gear ratio is therefore less critical.

### 2.6 CONCLUSIONS

**2.6.1** A theory is presented to analyse the longitudinal and vertical motion of a tractor operating cultivation implements.

**2.6.2** A computer programme based on this theory has been developed to predict tractor field performance.
2.6.3 In the longitudinal direction, tractor work rate is limited by the accuracy of the implement control in controlling draught fluctuations. Typically tractive efficiencies may be 6 to 8% lower under dynamic conditions than those calculated assuming a steady load.

2.6.4 In the vertical plane, tractor ride vibration levels were significantly decreased when ploughing compared with the tractor alone. This interaction will affect both operator comfort and implement control performance particularly at higher speed.
3. DRAUGHT CONTROL

Two strategies can be adopted to provide draught control, either by sensing draught force only or by sensing linkage force. Pure draught sensing is the simpler in theoretical terms but usually more difficult to arrange practically. Although it has been proposed by previous workers (78,82) as an alternative to linkage force sensing, pure draught control is not available commercially. On the other hand, linkage force sensing is available on nearly all modern tractors, either by sensing the top or lower link force. A theoretical analysis of both types of control enables their predicted performance to be compared. Experiments to compare performance in field conditions using an experimental electro-hydraulic control are also described.

In order to understand the performance of present draught controls, a rig was designed to measure the basic control parameters of commercial linkage force sensing controls. The results for 14 tractors measured on this rig are compared and discussed.

Finally, a simple method of improving the performance of present top link sensing controls without major modifications to the tractor is proposed and analysed. A damper fitted to the top link sensing unit to attenuate higher frequency error signals is the result of this analysis and field and laboratory experiments to verify its predicted performance are described.

3.1 TRACTOR HYDRAULIC LIFT

In the analysis of various types of controls the tractor hydraulic lift is often part of the control circuit. Firstly, therefore an analysis of the tractor hydraulic lift is described and its transfer functions calculated so that it can be included directly in the later analyses of different controls.

Fig 3.1 is a diagram of a typical top link sensing draught control. The top link is restrained by a spring so that force in the top link is proportional to displacement of the spool valve. For a compressive force in the top link, movement of the spool valve allows oil under
draught control lever in driver's cab to vary set top link force

draught control lever in driver's cab to vary set top link force

Fig. 3.1 Diagram of typical top link sensing draught control
pressure from the pump to flow into the lift cylinder to raise the linkage. Similarly for a tensile force in the top link, movement of the spool valve allows oil to flow out of the lift cylinder to the reservoir. The oil in the lift cylinder is under pressure because the linkage is supporting the weight of the implement.

The pumps fitted to the majority of British tractors are the constant displacement type, which have a flow rate proportional to rotational speed, in this case proportional to engine speed. When the valve is in its middle or rest position, all the pump flow is directed through the valve to the reservoir. This type of valve is called an open centre valve. An alternative system which is becoming more common on large American tractors is a variable displacement, constant pressure pump and a closed centre valve. When this type of valve is in its centre position there is no flow through it, so the pump output is held at maximum pressure and zero flow rate. The response tends to be faster than for open centre valves because pressure is already built up before the valve moves. For the open centre valve pressure only starts to build up when the pump flow is directed to the lift cylinder. The other advantage of the variable flow pump is that its power requirement matches the power required from the hydraulic components whereas the power requirement of the constant displacement pump is constant and power is wasted if no hydraulic components are being used. The only disadvantage of the variable flow, closed centre system is that it is more expensive.

The analysis here refers to the constant displacement, open centre hydraulic control since they are fitted to the majority of British tractors.

The transfer function for the spool valve and sensing unit varies for each tractor but can typically be described by the diagram shown in the first block of fig 3.2. The input to the spool valve is an error signal which is the difference between the actual top link force and the specified value which is set by adjusting the draught control lever in the driver's cab. The output is the amount of spool
Fig 3.2 Open loop control circuit for typical top link sensing control.
valve opening, \( x_g \), which on the majority of draught controls provides and on-off response when lifting and a proportional response when lowering. When the spool valve has moved there is a delay before the linkage is moved caused by the build up of oil pressure and oil compressibility or compliance in the linkage. Using Laplace transform notation, the effect of the delay time can be described by

\[
e^{-t_d s} \over 1 + sT_d
\]

which is a pure time delay, \( t_d \) seconds and a first order lag of time constant \( T_d \) seconds.

The overall gain of the control, \( k_s \), relates the lower link ends to the spool valve opening. It therefore depends on spool valve displacement, oil flow and linkage geometry. Because the linkage can be assumed to be parallel over the range necessary to control an implement, a movement, \( y_1 \), at the lower link ends is equal to the change in implement depth, \( y_o \). Hence

\[
\dot{y}_o = \dot{y}_1 = k_s x_s
\]

The open loop control circuit can now be drawn and is shown in full in fig 3.2. Typical values of the parameters shown in this diagram are quoted later in section 3.4 for a wide range of tractors.

3.2 PURE DRAUGHT CONTROL

This method of control is not available commercially although it is perhaps the most logical control since the parameter to be controlled, draught force, is the only one which is sensed.

The control circuit (fig 3.3) is basically a closed loop version of the tractor hydraulic lift circuit shown in fig 3.2. The assumption is made that the draught force is proportional to depth over a small range,

\[
H = k_f y_o
\]

The control is non-linear because the spool valve has deadband and saturation features. There are in general four methods for examining non-linear controls,

a) phase-plane,
b) describing function,
c) matrix techniques,
d) simulation by analogue or digital computer,

although the phase-plane is only really a special case of the state
variable approach used in the matrix technique for up to a second order
system. These methods are briefly reviewed in Appendix 3.1 and in the
following sections they are used to analyse the pure draught sensing
control.

3.2.1 Phase-plane

The phase-plane portrait is best plotted in terms of input, $E$, to
the non-linearity. Each region of the non-linearity, e.g. the deadband
proportional and saturated zones, can then be simply marked on the
portrait. It is difficult to analyse the effect of a pure time delay
using this approach because time must be calculated and marked
separately on the portrait, so it is applied below to the control shown
in fig 3.3 with $t_d = 0$ and $T_d = 0.05$ s. Each region is examined
separately.

(a) Saturated

$$E(s) = \frac{2.5 k_s k_x}{s(1+sT_d)}$$

which becomes

$$\ddot{E} + 20\dot{E} = 4.4 \times 10^5 \text{ assuming } H_{set} = \text{constant and}$$

using the values in fig 3.3

Letting $x = E$ and $y = \dot{E}$,

$$\frac{dy}{dt} + 20y = 4.4 \times 10^5$$

and dividing by $y = \frac{dx}{dt}$

$$\frac{dy}{dx} + 20 = \frac{4.4 \times 10^5}{y}$$

Letting the gradient $\frac{dy}{dx} = N$, equation (3.7) becomes

$$y = \frac{4.4 \times 10^5}{N + 20}$$

Putting various values of $N$ into equation (3.8) gives the isoclinal
equations which are then drawn in the saturated region of the portrait,
and the gradient, $N$, indicated at points along the lines (fig 3.4).
Fig 3.4 Phase plane portrait for pure draught sensing with proportional control calculated by isoclinal method.
(b) Proportional

\[ \mathcal{E}(s) = \frac{-k_p (\xi - 450) k_s k_e}{s (1 + s T)} \quad (3.9) \]

\[ \ddot{\xi} + 20 \dot{\xi} = -984 \xi + 4.4 \times 10^5 \quad (3.10) \]

Again letting \( x = \xi, y = \dot{\xi} \) and \( N = \frac{dy}{dx} \), equation (3.10) becomes,

\[ y = -984x + 4.4 \times 10^5 \]

Putting various values of \( N \) into equation (3.11) gives the isoclinal equation for the proportional region which are straight lines involving both \( x \) and \( y \).

(c) Deadband

\[ \ddot{\xi} + 20 \dot{\xi} = 0 \quad (3.12) \]

and using the same notation as previously

\[ N = -20 \quad (3.13) \]

In this region therefore the gradient is always -20.

Having constructed the isoclines as shown in fig 3.4, the system response to a particular input can be estimated by sketching the trajectory starting from a particular point, for example, the point \((1125,0)\), which represent a step input of 1125 N. The direction of the trajectory is estimated by the direction of the gradient as its proceeds.

The main use of the portrait is for investigating stability. The response in fig 3.4 to a 1125 N step input is stable because it comes to rest in the deadband. Instability would be indicated by a trajectory which did not come to rest but continued round in a loop. This is called a limit cycle oscillation and is discussed in more detail later.

3.2.2 Describing function

Assuming that an approximate frequency response function for the non-linearity, \( G_N(j\omega) \), is known then the block diagram of the control is,

\[ G(j\omega) = \frac{k_s k_e e^{-j\omega T_d}}{j\omega (1 + j\omega T_d)} \]

The overall transfer function assuming that the ground input disturbances, \( M(j\omega) \) equal zero, is
and stability can be analysed on an inverse Nyquist diagram. The condition is similar to that for a linear system except that the locus of \(1/G(jw)\) must pass to the left of \(G_N(jw)\) rather than the point \((-1,0)\). However \(G_N(jw)\) is amplitude dependent and so the amplitude must be marked along the locus of \(G_N(jw)\). A table of \(-G_N(jw)\) for various common non-linearities, including those analysed here has been compiled by Gibson (117).

The inverse Nyquist diagram can be used to investigate the effect of various parameters on stability and the effect of the delay time will be investigated here. Firstly, consider the proportional control with no time lag, but with a pure time delay. It can be simplified into the form shown in fig 3.5 by incorporating the gain, \(k_s\), into the non-linearity.

\[
\frac{1}{G(jw)} = \frac{jw e^{jwt_d}}{20}
\]  (3.15)

The pure time delay does not affect the gain of \(1/G(jw)\) but it produces a phase shift of \(wt_d\). Having plotted the locus of \(1/G(jw)\) for \(t_d = 0\) (fig 3.6), the locus for other values of \(t_d\) can be plotted by adding a phase shift of \(wt_d\) at each frequency. Increasing \(t_d\) moves the curve further to the left (fig 3.6) and for \(t_d = 0.08\) s, the control is on the verge of instability. For \(t_d = 0.1\) s, it is unstable for an input amplitude \(\frac{E}{DB}\) greater than 1.8 but stable if \(\frac{E}{DB}\) is less than 1.8. Once instability starts, the amplitude of unstable oscillation increases until it reaches 3.8 and then remains at that value in a limit cycle oscillation. The frequency of unstable oscillation, i.e. the frequency at which the \(1/G(jw)\) locus crosses the \(G_N(jw)\) locus, for \(t_d = 0.1\) s is \(w = 15.7\) rad/s (2.5 Hz).

If the proportional control has both a first order lag, \(T_d = 0.05\) s, and a pure time delay then

\[
\frac{1}{G(jw)} = \frac{jw (1 +jwt_d)}{20}
\]  (3.16)
Fig 3.5  Simplified control circuit for describing function calculations
Fig. 3.6 Inverse Nyquist diagram for control circuit in Fig. 3.5 for 3 values of $E_d$. Values of $E$ in brackets.
Fig. 3.7: Inverse Nyquist diagram of pure draught sensing control with $T_d=0.05$ for 3 values of $T$.
\[ \frac{1}{G(j\omega)} = \left(\frac{-\omega^2}{400} + \frac{j\omega}{20}\right)e^{j\omega t_d} \]  

The inverse Nyquist diagram for this case is shown in fig 3.7 for values of \( t_d \) of 0, 0.05 and 0.1 s. Instability occurs for \( t_d = 0.1 \) s if the amplitude \( \frac{E}{DB} \) is greater than 1.6. The frequency of unstable oscillation is about 10.8 rad/s (1.7 Hz).

These examples show that any increase in delay time of either \( t_d \) or \( T_d \) decreases the stability by moving the locus of \( 1/G(j\omega) \) nearer to the \( -G_p(j\omega) \) locus. However most hydraulic controls have a combination of both delays so it is worth examining next the effect of each type together. The minimum value of \( t_d \) measured for a tractor draught control (115) is 0.02 s and if this is combined with a lag of \( T_d = 0.04 \) s say, it can be plotted (fig 3.8) to compare its stability with a control where \( t_d = 0.05 \) s. Stability is improved although neither control is unstable, but if the same comparison is made (fig 3.9) for an on-off control, the advantage of the control where \( t_d = 0.02 \) s and \( T_d = 0.04 \) s is shown clearly because it remains stable whereas the control with \( t_d = 0.05 \) s is unstable.

By repeating the calculations for various combinations of \( t_d \) and \( T_d \), it is possible to plot a graph of \( t_d \) against \( T_d \) to show the range of values for stability. This is shown for both a proportional and on-off control in fig 3.10. The line indicates the values of \( t_d \) and \( T_d \) where the control is on the verge of instability and for any values above the line it is unstable and below the line stable. These curves were calculated using the parameter values shown in fig 3.3 and in simplified form in fig 3.5.

Fig 3.10 shows, as mentioned previously, that any increase either in \( t_d \) or \( T_d \) decreases stability. However, both curves flatten for \( T_d = 0.05 \) s and stability is then dominated by the pure time delay. In general, the value of \( t_d \) is more critical for stability because for the on-off control used in this example, instability is inevitable for \( t_d > 0.049 \) s independent of the value of \( T_d \) and the control is always stable for \( t_d < 0.03 \) s again independent of \( T_d \).
Fig. 3.8 Comparison of pure draught sensing with
(a) pure time delay only, \( t_d = 0.05 \) s
(b) pure and 1st order log, \( t_d = 0.02 \) s, \( T_a = 0.02 \) s

\[
G_{(ao)} = \frac{V_h (1 + 0.04m_0 e^{-0.02z})}{20}
\]

Values of \( c \) are bracketed.

\[
\frac{1}{2} e^{0.5x} = 0.25 \quad x = 1.2 \quad \text{deg}
\]
Fig 3.10 Critical values of $\xi_d$ and $T_d$ for stability of proportional and on-off control shown in Fig 3.3 where $DB = 900$ N and $y_{d_{\max}} = 500$ mm/s
3.2.3 Matrix techniques

State space analysis is a method of writing the system equations in matrix form ready for computer solution. The phase-plane method described earlier is really a particular case in state-space analysis for a second order system and because there are only two state variables, in this case they were \( E \) and \( \dot{E} \), they could be plotted against each other.

The state equation, which is the matrix equation involving all the state variables can quickly be derived for the pure draught control by first writing equations describing the control in fig 3.3, with \( t_d = 0 \).

\[
H = k_1 (y_1 + y_o) \quad (3.18)
\]

\[
\frac{t_d}{d^2 y_o}{dt^2} + \frac{dy_o}{dt} = k_s x_s \quad (3.19)
\]

\[
x_s = 2.5 \quad \text{saturated}
\]

\[
H = k_p (E - 450) \quad \text{proportional}
\]

\[
H = 0 \quad \text{deadband}
\]

\[
E = H_{set} - H \quad (3.21)
\]

Combining 3.18-21 into one equation for say, the proportional region gives,

\[
\frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{dy_o}{dt} = \frac{k_s k_p}{T_d} \left[ H_{set} - k_1 (y_1 + y_o) - 450 \right] \quad (3.22)
\]

\[
\frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{dy_o}{dt} + \frac{k_1 k_s k_p}{T_d} y_o = \frac{k_s k_p H_{set}}{T_d} - \frac{k_s k_p k_1 y_i}{T_d} - \frac{450 k_s k_p}{T_d} \quad (3.23)
\]

Defining the state variables,

\[
x_1 = y_o
\]

\[
x_2 = \dot{y}_o
\]

and letting \( H = H_{set} - 450 \)

\[
\frac{d}{dt} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 & \frac{k_1 k_s k_p}{T_d} \\ \frac{1}{T_d} & -1 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{k_s k_p}{T_d} \end{bmatrix} \begin{bmatrix} H_{set} \\ y_i \end{bmatrix} \quad (3.24)
\]
which is the state equation for the proportional zone. The equations for the other regions will have a similar form but different values in the matrices. Because the matrices vary during the solution depending on which region is applicable at any time, the solution on a digital computer is quite time-consuming and complex. It is not particularly useful for this problem which is only second order and can be analysed more easily on the phase-plane.

3.2.4 Simulation

**Analogue simulation**

The analogue computer circuit and scaling to simulate the pure draught control of fig 3.3 where \( t_d = 0 \) is shown in Appendix 3 II. It is difficult to include the effect of a pure time delay although it is possible to incorporate it using either a tape loop which continuously records the signal and replays it \( t_d \) s later or mathematically using the Padé approximation. Neither method was used because the equipment for the first method was not available and because not enough amplifiers were available on the computer to programme the second method.

Mainly because of these limitations it was decided to concentrate on digital simulation which is discussed in the next section. However, the results from the analogue computer simulation agreed well with the results from the phase-plane analysis. An example is shown in fig 3.11 which is a phase-plane portrait calculated by analogue simulation for the control in fig 3.3 where \( t_d = 0 \) and \( T_d = 0.05 \) s. It can be seen that this compares well with the portrait in fig 3.4 for the same control calculated by the isocline method.

The effect of proportional and on-off control is compared in fig 3.12 where the response to a step input of 1125 N for the same control as above is shown. The proportional response of the spool valve could be simulated using a function generator although some slight rounding occurred at the breakpoints. The on-off response required an extra diode circuit which is shown in Appendix 3 II. The only significant
Fig 3.11  Phase plane portrait for control in fig 3.3 calculated by analogue computer simulation. Curve calculated in fig 3.4 shown for comparison.
Fig 3.12 Response of a pure draught sensing control to a 1125 N step input calculated by analogue computer simulation.
difference shown between the response of proportional and on-off controls to a step input is in the final value of the draught force. The deadband is shown by the dotted lines and the on-off control tends to overshoot more than the proportional control after the boundary of the deadband has been reached. The reason for this is that its correction rate as it enters the deadband is higher than that for the proportional control. This also decreases the stability of the on-off control compared to the proportional control because the response is more likely to overshoot to the other side of the deadband and signal the opposite control movement.

Digital simulation

The C.S.M.P. programme (Continuous System Modelling Program) to simulate the pure draught control of fig 3.3 is shown in Appendix III. None of the restrictions of the previously described methods apply to the digital simulation and for this reason the effects of the various parameters, delay time, deadband, rate of lift on proportional and on-off controls have been calculated and are discussed below.

The only limitation to this method of analysis is the accuracy of the numerical integration incorporated in the programme. This may be a particular problem where the simulation contains discontinuities as it does in this case. Experimentation with several different methods of integration varying in complexity from rectangular to Runge-Kutta and Milne methods showed that the simplest method, rectangular with a very small step size (0.0005 s) was the most accurate. No integration method can be expected to integrate a discontinuous signal perfectly but the error was minimised with this method. The approach used to compare the accuracy of various integration methods was to first plot the signal to be integrated accurately using a graph plotter and calculate the area underneath the line. The results using various integration were then compared with the above figure to compare their accuracy.

3.2.5 Effect of delay time

Using the C.S.M.P. programme, the effect of a pure time delay, \( t_d = 0.05 \) s, and a first order lag, \( T_d = 0.05 \) s, is compared for a
proportional control in fig 3.13 and an on-off control in fig 3.14. Results are plotted on the phase-plane and have unfortunately had to be re-drawn from the original graph plotter plots because of difficulties in ink colour and scaling. The spikes in the trajectory for the on-off control are due to the difficulties in numerical integration discussed previously.

Although the phase-plane portraits have been shown for ease of comparison with the other methods, they have the disadvantage that time is not included directly. Since time must be calculated in the computation, it can however be marked at points along the trajectory, as shown in fig 3.13.

For the on-off control with a pure time delay, $t_d = 0.05$ s, the trajectory continues in a closed loop, indicating a limit cycle oscillation. This confirms the instability predicted in fig 3.10 by the describing function method for the same control parameter values. The frequency of unstable oscillation is 5 Hz, and the amplitude expressed as a percentage of the deadband ($\frac{E}{\Delta E}$) is 1.5. The figures calculated by the describing function method (fig 3.9) are a frequency of 5 Hz and an amplitude of about 1.1. The reason for the difference in the predicted amplitude is due to the approximations in the describing function of the on-off non-linearity in the regions near to its breakpoints.

The response of a control with a mixed delay, $t_d = 0.02$ s and $T_d = 0.4$ s, which was discussed in the previous section on the describing function method, is shown in fig 3.15. The trajectories for both a proportional and an on-off control are shown and the zone boundaries are the same as those in figs 3.13 and 3.14. The advantage of improved stability is shown for the on-off control because the trajectory stops within the deadband rather than continuing in a limit cycle as it did for the pure time delay in fig 3.14.

3.2.6 Effect of deadband and rate of lift

Decreasing the deadband has the effect of increasing the accuracy of the control since the final value is within a smaller range but extra
Fig 3.13: Phase plane portrait to show effect of a) 1st order (lag and b) pure time delay for proportional control
Fig 3.14 Phase plane portrait to show effect of a) 1st order (ag and b) pure time delay for on-off control.
Fig 3.15 Effect of mixed delay, $t_d = 0.02$ s and $\bar{t}_d = 0.04$ s on a proportional and on-off control.
control movements are probably necessary to achieve this. This point is illustrated in fig 3.16 where the response of a proportional and on-off control calculated from the C.S.M.P. programme are plotted for a control deadband of 225 N. An extra correction is needed in each case compared with the 450 N deadband in figs 3.13 and 3.14 before the final value is reached.

Increasing the rate of lift improves performance slightly differently because although the final value is unaltered, the time taken to reach it is reduced. The effect however is not shown clearly on the phase-plane because time is not included.

There is however an important relationship between rate of lift, deadband and time delay in order to maintain stability. For the case of a pure time delay only the conditions for stability can be inferred from the phase-plane diagrams. Instability occurs when the control movement, due to the time delay in switching off after the error signal is within its deadband, is sufficient to cause the error signal to move through the deadband and signal the opposite control. Hence the following condition must be satisfied to maintain stability

\[ DB > \hat{y}_{\text{max}} \cdot k_1 \cdot t_d \]  \hspace{1cm} (3.25a)

for an on-off control and

\[ \left[ DB + \frac{2x_{s_{\text{max}}}}{k_p} \right] > \hat{y}_{\text{max}} \cdot k_1 \cdot t_d \]  \hspace{1cm} (3.25b)

for a proportional control. Eqn 3.25 is verified in fig. 3.17 where the response of a pure draught sensing control with \( t_d = 0.05 \) s is plotted for two rates of lift. Inserting the control parameters shown into eqn 3.25 predicts instability for \( \hat{y}_{\text{max}} = 500 \) mm/s, but stability for \( \hat{y}_{\text{max}} = 375 \) mm/s. The results from the computer simulation agree with this and the limit cycle for the higher rate of lift has a frequency of \( \frac{1}{4t_d} = 5 \) Hz. The amplitude depends on the magnitude of the inequality in eqn 3.25. If this is small, then amplitude is \((\hat{y}_{\text{max}} \cdot k_1 \cdot t_d) N\), and as the inequality increases so too does the limit cycle amplitude.
Fig 3.16 Effect of reducing deadband on a proportional and on-off control, with $T_d = 0.05\, \text{s}$.
Control parameters used:
- \( DB = 900 \text{ N} \)
- \( k_1 = 44 \text{ N/mm} \)
- \( \epsilon_d = 0.05 \text{ s} \)
- On/off response

Max. rate of lift:
- \( \dot{y}_0 \text{ max} = 375 \text{ mm/s} \)

Limit cycle:
- Frequency = 5 Hz
- Amplitude = 1250 N

Fig 3.17: Response of pure draught sensing control for 2 maximum rates of lift with deadband.
No simple expression can be written for the control involving a first-order lag, but stability can easily be checked using either a graph constructed from inverse Nyquist diagrams, such as that shown in Fig 3.10, or a digital computer simulation.

3.3 LINKAGE FORCE SENSING

The most common method of draught control is by sensing the force in either the top or lower links of the tractor three point linkage. Fig 3.18 is a diagram of the control circuit for these controls, the only difference between top and lower link sensing being an extra gain, k₁, at the point shown in the feedback loop.

The equations for $F_T$ and $F_L$ were derived in section 2.3 and are

$$
P_T = \frac{H}{13} \left(15 - \frac{y_o}{4}\right) - \frac{v_{12}}{l_2}
$$

$$
= \frac{k_1 v_o}{13} \left(15 - \frac{y_o}{4}\right) - k_1 \frac{v_o}{4} \frac{d v_o}{dx} \frac{l_2}{l_3}
$$

Hence,

$$
P_T = k_1 y_o - \frac{k_1 v_o^2}{4 l_3} + \frac{2k_1 y_o v_o}{x}
$$

assuming that for a three furrow plough, $l_5/l_3 = 1$ and $l_2/l_3 = 2$.

Ignoring the second order term in $y_o$ which is small for small depth changes,

$$
P_T = k_1 y_o + k_1 k_2 y_o v_o
$$

where $k_2 = \frac{2}{x}$. Also, the lower link force,

$$
P_L = \frac{H}{13} \left(15 - \frac{y_o}{4} + \frac{13}{l_2}\right) - \frac{v_{12}}{l_2}
$$

$$
P_L = 2k_1 y_o + \frac{2k_1 y_o v_o}{x}
$$

$$
P_L = k_1 y_o + k_1 k_2 y_o v_o
$$

where $k_1 = 2k_1$. The only difference between top and lower link sensing is that the gain term for $y_o$ is doubled for lower link sensing. The following analysis therefore, applies to top link sensing since it is the more common control but it can easily be applied to lower link sensing by replacing $k_1$ with $k_1$. 
Fig 3.18: Control circuit for top or lower link sensing control
The methods of analysis used are those described previously in section 3.2. The state equations are derived and it is shown that the control can be analysed on the phase-plane. This however is a lengthy process and since the describing function technique cannot be used because of the two non-linearities, simulation by digital computer is the main method used to investigate the effects of the control parameters.

3.3.1 Matrix techniques

From fig 3.18 the equations for the top link sensing control circuit are

\[ H = k_1 (y_o + y_i) \]  \hspace{1cm} (3.32)

\[ \frac{d^2 y_o}{dt^2} + \frac{d y_o}{dt} = k_p (\xi - 100), \text{ proportional} \]

\[ \frac{d y_i}{dt} = 0, \text{ deadband} \]  \hspace{1cm} (3.34)

\[ \xi = \frac{F_{m}}{F_{set}} - H - H k_2 \frac{d}{dt} (y_1 + y_o) \]

\[ \xi = \frac{F_{m}}{F_{set}} - H - H k_2 \frac{d y_i}{dt} - H k_2 \frac{d y_o}{dt} \]  \hspace{1cm} (3.35)

Combining these for proportional case,

\[ \frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{d y_o}{dt} = \frac{k_p}{T_d} \left[ \xi - H - H k_2 \frac{d y_i}{dt} - H k_2 \frac{d y_o}{dt} - 100 \right] \]  \hspace{1cm} (3.36)

Assuming initially that on level ground \( y_i = \) a constant and \( \frac{d y_i}{dt} = 0 \), then

\[ \frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{d y_o}{dt} = \frac{k k_s}{T_d} \left[ \frac{1}{T_d} (k_2 s_p y_o + k_1 k_2 y_i) - k_1 y_o + \frac{F_{m}}{F_{set}} - k_1 y_i - 100 \right] \]  \hspace{1cm} (3.37)

\[ \frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{d y_o}{dt} \left[ \frac{k k_s s_p}{T_d} y_o + \frac{k_1 k_2 s_p}{T_d} y_i \right] = \frac{k k_s}{T_d} \left( \frac{F_{m}}{F_{set}} - 100 \right) - \frac{k k_s}{T_d} k_1 y_i \]

\[ \frac{d^2 y_o}{dt^2} + \frac{1}{T_d} \frac{d y_o}{dt} \left[ \frac{k k_s s_p}{T_d} y_o + \frac{k_1 k_2 s_p}{T_d} y_i \right] = \frac{k k_s}{T_d} \left( \frac{F_{m}}{F_{set}} - 100 \right) - \frac{k k_s}{T_d} k_1 y_i \]  \hspace{1cm} (3.38)

Letting \( K = \frac{k_1 k_2 k_s}{T_d} \) and \( F_{m}' = F_{m} - 100 \).
\[
\frac{d^2 y}{dt^2} + \frac{dy}{dt} \left( \frac{1}{T_d} + Ky + Ky_1 \right) + \frac{K}{k_2} y = \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} y_i
\]  
(3.39)

Defining the state variables \( x_1 = y_0 \)
\[ x_2 = \dot{y}_0 \]

the STATE EQUATION can be written
\[
\frac{d}{dt} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 1 \\ - \frac{K}{k_2} - \left( \frac{1}{T_d} + Ky + Ky_1 \right) \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} \end{bmatrix} \begin{bmatrix} F_T' \\ y_i \end{bmatrix}
\]  
(3.40)

However, for the reasons given earlier in section 3.2 this is not easy to solve in this form, but having derived the basic equation of the control, it can be solved by a slightly more complex application of the phase plane method.

3.3.2 Phase plane

Letting, \( x = y_0 \) and \( y = \dot{y}_0 \), equation (3.39) can be written,
\[
\frac{dy}{dt} + y \left( \frac{1}{T_d} + Ky_1 + Kx \right) + \frac{K}{k_2} x = \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} y_i
\]  
(3.41)

Divide by \( y = \frac{dx}{dt} \)
\[
\frac{dy}{dx} + \left( \frac{1}{T_d} + Ky_1 + Kx \right) + \frac{K}{k_2} x = \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} y_i
\]  
(3.42)

Let \( N = \frac{dy}{dx} \)
\[
N + \left( \frac{1}{T_d} + Ky_1 + Kx \right) = \frac{1}{y} \left[ \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} y_i - \frac{K}{k_2} x \right]
\]  
(3.43)

\[
y = \frac{\left[ \frac{k_s k_p F_T'}{T_d} - \frac{K}{k_2} y_i - \frac{K}{k_2} x \right]}{N + \frac{1}{T_d} + Ky_1 + Kx}
\]  
(3.44)

which is the ISOCLINAL EQUATION for the PROPORTIONAL zone.

For the SATURATED zone, when \( x_s = 0.1 \)
\[
\frac{T_d d^2 y_0}{dt^2} + \frac{dy_0}{dt} = 0.1 \ k_s
\]  
(3.45)

\[
y = \frac{0.1 k_s}{1 + T_d N}
\]  
(3.46)
For the DEADBAND zone

\[ \frac{d^2y}{dt^2} + \frac{dy}{dt} = 0 \]  

(3.47)

\[ N = \frac{-1}{T_d} \]  

(3.48)

Knowing the isoclinal equations for each zone, it should now be possible to plot \( \dot{y}_o \) against \( y_o \) (i.e. \( y \) against \( x \)). However since it is the \( \dot{y}_o - y_o \) plane rather than the \( \dot{\xi} - \xi \) plane the zones are not easily defined. A relationship between \( \xi \) and \( y_o \) is required and this can be specified for particular input conditions.

Take for example \( y_1 = 0, \frac{dy_1}{dt} = 0, F_{r_{set}} = 1125 \) to examine the system response to a 1125 N step input. Equation 3.35 gives

\[ \dot{\xi} = F_{r_{set}} - k_1y_o - k_1k_2y_o \frac{dy}{dt} \]  

(3.49)

The lines defining the zones on the \( \dot{\xi} - \xi \) plane are

- dead band
- prop. saturated

<table>
<thead>
<tr>
<th>( \xi )</th>
<th>( \dot{\xi} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>900</td>
<td></td>
</tr>
<tr>
<td>-450</td>
<td></td>
</tr>
<tr>
<td>450</td>
<td></td>
</tr>
<tr>
<td>-900</td>
<td></td>
</tr>
</tbody>
</table>

Substituting these four values into equation 3.49 gives

\[ \xi = 450 \quad y = 760 - \frac{11700}{x} \]  

(3.50)

\[ \xi = -450 \quad y = 760 - \frac{27300}{x} \]  

(3.51)

\[ \xi = 900 \quad y = 760 - \frac{3890}{x} \]  

(3.52)

\[ \xi = -900 \quad y = 760 - \frac{35000}{x} \]  

(3.53)

These are plotted on the phase plane in fig 3.19. Using the parameter values shown in fig 3.18 the isoclinal equations corresponding to each of these regions are also plotted and the trajectory of the system response to a 1125 N step input is sketched using the isoclines.

This is a lengthy method of calculating the system response because it has the disadvantage that the lines defining the zone boundaries must be calculated for each input, but since it is impossible to
Figure 3.19 Phase plane portrait calculated for top-link sensing control in fig 3.18.
obtain the equations for $\dot{\varepsilon}$ and $\varepsilon$ to plot on a $\dot{\varepsilon}-\varepsilon$ plane, this difficulty cannot be avoided.

3.3.3 Describing function

The describing function method cannot be applied to this problem because there is more than one non-linearity. One of the assumptions when using describing functions is that if the input to the non-linearity is a sinusoidal function then the output will be a distorted sine wave of the same fundamental frequency. In this case the input to the second non-linearity is already a distorted sine wave from the output of the first non-linearity and so it is impossible to obtain an accurate function to describe its characteristics.

3.3.4 Simulation

Simulation therefore appeared to be the most useful method of analysis and the analogue computer circuit and scaling for a proportional top link sensing control with a 1st order lag only are shown in Appendix 3.II. The analogue computer was not however used much, partly because of lack of availability and partly because insufficient equipment was available to simulate a pure time delay. For these reasons digital simulation was the main method used and the C.S.M.P. programme for a top link sensing control is shown in Appendix 3.III.

As a comparison with the phase-plane method, the phase plane portrait calculated by the C.S.M.P. programme for the control shown in fig 3.18, is shown in fig 3.20. The trajectory for the proportional control agrees well with that previously calculated in fig 3.19 and shows a stable control for the set of control parameters shown in fig 3.18. However the C.S.M.P. programme results are discussed in the following paragraphs on the effect of control parameters on stability.

3.3.5 Effect of parameters on stability

The effect of three types of delay on the response of the control shown in fig 3.18 to a 2250 N step input is shown in fig 3.21. The control is unstable (upper figure) for a pure time delay, $t_d = 0.05$ s, with a limit cycle of 20 Hz and amplitude 1125 N. It is stable (middle
Fig 3.20 Phase plane portrait for top link sensing control in fig 3.18 calculated by C.S.M.P. computer simulation
All parameters except delay as shown in fig 3.18

a) $\tau_d = 0.05 \, s$

b) $T_d = 0.05 \, s$

c) $\tau_d = 0.02 \, s$, $T_d = 0.04 \, s$

Fig 3.21 Effect of delay time on control response to
Taking each of these types of delay in turn, for the first with a pure time delay, an approximate expression can be deduced to predict instability. At the instant when a control movement starts the feedback term due to the vertical force changes by

$$H_k^2 \dot{y}_o$$

The rate of lifting or lowering, $\dot{y}_o$, in this expression would be the maximum rate for an on-off control but proportional to the error signal for a proportional control. If the value of this expression is greater than the deadband then the opposite control will immediately be signalled and the control will be unstable. Therefore, for an on-off control, it will be stable if

$$DB > H_k^2 \dot{y}_o^\text{max}$$  \hspace{1cm} (3.54)

Unlike the stability criterion for pure draught sensing (eqn 3.25) this is independent of the delay time but the limit cycle frequency will be

$$\frac{1}{2\pi t_d} \text{ Hz}$$

and the amplitude

$$k_t \dot{y}_o^\text{max} t_d$$  \hspace{1cm} (3.55)

Results from the computer simulation confirm this analysis. Fig 3.22 shows again the response of the control in fig 3.18 to a 2250 N step input, but for values of $t_d$ of 0.04 s (upper) and 0.08 (lower). For the parameter values shown in fig 3.18 and an initial draught force of 13400 N equation 3.54 predicts instability and the limit cycle amplitudes and frequencies calculated from equation 3.55 agree with those calculated by computer simulation.

Fig 3.23 shows the effect of reducing the rate of lift and increasing the deadband until the control is stable. For $DB = 2700$ N, eqn. 3.55
Fig 3.22 Effect of pure time delay on frequency and amplitude of limit cycle
Fig 3.23 Effect of reducing deadband on top link sensing control where $y_{\text{max}} = 125 \text{ mm/s}$ and $\tau = 0.05 \text{ s}$
predicts a stable control since

\[ 2700 > 13400 \times 1.3 \times 10^{-3} \times 125 \]

i.e. \[ 2700 > 2180 \]

and the computer simulation confirms this. The value of \( t_d \) is 0.05 s so the limit cycles for the other two controls where DB = 900 N and 1800 N have a frequency of 20 Hz and amplitude 280 N. The optimum parameters for a control with a pure time delay can therefore be selected from equation 3.55.

For the control with a first order lag, values of \( T_d \) from 0.02 to 0.1 s have no effect on the control response either for the proportional or on-off case, as shown in fig 3.24. The computer simulation only predicts instability if \( T_d \) is reduced to zero, but as long as \( T_d \) has some value, the lower curve in fig 3.24 shows that the deadband can be reduced to zero without instability. Increasing the rate of lift does not have any effect on performance and the practical explanation of this is that there is an maximum value for the rate of lift beyond which any increase causes an increased vertical force component which opposes the draught force and reduces the error signal. This in turn reduces rate of lift signalled either partially for the proportional control or totally for the on-off control. Once the rate of lift then decreases, the vertical force decreases and so there is an equilibrium point independent of the maximum rate of lift above a certain value.

Fig 3.25 shows the response (curve c) of a control where \( t_d = 0.02 \) s and \( T_d = 0.04 \) s. Increasing rate of lift above this value of 250 m/s for the on-off control causes instability. The approximate optimum responses of all three types of delay discussed can be compared from this figure. An on-off control with DB = 1800 N was used for all three curves shown and the delays and rates of lift are indicated. Taking the time necessary for the draught to reach its deadband as an approximate performance criteria, the mixed delay is superior. The implementation of this type of delay characteristic in the tractor
Fig 3.24 Top link sensing control with 1st order lag, showing that response is unaffected for $T_d$ values of 0.02, 0.04, 0.06, 0.08, 0.1 s.
Fig 3.25 Comparison of the performances of an on-off top link sensing control with three types of delay characteristic.

- a) $T_d = 0.05 \text{s}$, $\dot{y}_{\max} = 100 \text{ mm/s}$
- b) $T_d = 0.05 \text{s}$, $\dot{y}_{\max} = 500 \text{ mm/s}$
- c) $T_d = 0.02 \text{s}$, $\dot{y}_{\max} = 250 \text{ mm/s}$

The diagram shows the force (in kN) over time for each scenario, with the time scale ranging from 0 to 1.0 seconds.
hydraulics may require an extensive re-design. So whilst it is an important factor to bear in mind for new controls, it is probably not economically feasible to change the delay characteristic of existing controls.

3.3.6 Effect of damping

However, some improvement to existing controls may be obtained by damping the spool valve movement (120). This is a relatively cheap and simple addition which has the effect of introducing a first order lag of time constant, $T_d$, into the control circuit in fig 3.18 in front of the non-linearity. The effect of this on the optimum response with a pure time delay previously shown in fig 3.25, is shown in fig 3.26 and its effect on a mixed delay is shown in fig 3.27. In both cases there is an improvement in the time taken for the draught to reach the deadband.

From limited evidence available on the frequency range of the random draught variations when a tractor and implement cross a field surface, it can be concluded that the important frequency range to control is below 4-6 Hz. Fig 3.28 shows the frequency distribution measured (108) for the draught of a plough locked in position control relative to the tractor which was travelling at 1.5 m/s. Higher frequency fluctuations due to vibration, stones and soil shattering effects can cause unnecessary control movements, spool valve movements or wear. The problem of spool valve wear has been mentioned by manufacturers (128, 129) as a cause of deteriorating control performance after a relatively short time compared with the tractor life. Damping the spool valve movement attenuates the high frequency fluctuations and prevents some of these problems.
Fig 3.26 Effect of damping spool valve movement on an on-off top link sensing control with a pure time delay

- a) $t_d = 0.05 \text{ s}, y_0_{max} = 100 \text{ mm/s}$
- b) damper fitted
  - $T_s = 0.04 \text{ s}$
  - $t_d = 0.02 \text{ s}$
  - $y_0 = 250 \text{ mm/s}$

Draught force

13

13.5

12

kN

0.2 0.4 0.6 0.8 1.0

Time s

deadband $d_d = 800 \text{ N}$
Fig. 3.27 Effect of damping spool valve movement on an on-off top link sensing control with a pure time delay, $T_d = 0.02$ s and 1st order lag, $T_d = 0.04$ s.
Fig 3.28 Frequency Distribution of Draught Force with Plough
Locked in Position Control
3.4 A RIG TO MEASURE THE BASIC PARAMETERS OF DRAUGHT CONTROLS

3.4.1 Equipment and procedure

Fig 3.29 is a diagram of a rig used to measure the response characteristics of a tractor draught control. The lower links of the tractor were connected to the pins at A, one on each side of the rig and the top link was replaced by the hydraulic ram B. The lift rods were disconnected from the lower links so that cables could be attached to the lift arms at C. The cables ran via pulleys to an adjustable set of weights D which could be varied to simulate implements of different weight.

A 2.5 kW electric motor E was used to drive a hydraulic pump F, which supplied oil under pressure to the double-acting ram B, via a solenoid valve G and a reducing valve H. An oil cooler J was included in the circuit before the oil returned to the reservoir K.

The pressure gauge L was calibrated to represent the load in the top link so that to apply a desired load as step input the following procedure was adopted:–

(i) The oil supply to the solenoid valve was cut off by closing the valve M;

(ii) The desired load was set using the reducing valve;

(iii) The valve M was opened, the electrical switch to the solenoid valve operated, and pressure in the ram built up immediately.

The hydraulic ram was double-acting so that any load from 11 kN compression to 6.7 kN tension could be applied.

If the tractor had bottom link sensing, a force applied at the top link would be reacted by an equal and opposite force along the lower links so the force sensed by the control would be identical to that in the top link.

A load cell, N, in the top link measured the force applied by the ram and a rotary potentiometer mounted on the pivot of the lift arms recorded their rotation which was later converted into movement at the
Fig 3.29 Draught Control Response Rig
<table>
<thead>
<tr>
<th>Tractor</th>
<th>Power Class</th>
<th>Minimum deadband N</th>
<th>Maximum rate of lift (at rated engine speed) m/s</th>
<th>Lifting sensitivity m/s N</th>
<th>Maximum rate of lowering m/s</th>
<th>Lowering sensitivity m/s N</th>
<th>Delay lifting s</th>
<th>Delay lowering s</th>
<th>Variable rate of lifting or lowering</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>under 38 kW</td>
<td>450</td>
<td>0.41</td>
<td>*</td>
<td>0.53</td>
<td>0.11</td>
<td>0.05</td>
<td>0.08</td>
<td>N.A.</td>
<td>15 year old tractor</td>
</tr>
<tr>
<td>b</td>
<td>under 38 kW</td>
<td>1540</td>
<td>0.60</td>
<td>*</td>
<td>0.25</td>
<td>0.10</td>
<td>N.A.</td>
<td>N.A.</td>
<td>N.A.</td>
<td>lift</td>
</tr>
<tr>
<td>c</td>
<td>under 38 kW</td>
<td>2340</td>
<td>0.26</td>
<td>~ 0.11</td>
<td>0.48</td>
<td>0.18</td>
<td>0.08</td>
<td>0.08</td>
<td>lower</td>
<td></td>
</tr>
<tr>
<td>d</td>
<td>under 38 kW</td>
<td>1560</td>
<td>0.23</td>
<td>~ 0.11</td>
<td>0.44</td>
<td>0.11</td>
<td>0.12</td>
<td>0.06</td>
<td>lower</td>
<td></td>
</tr>
<tr>
<td>e</td>
<td>38-50 kW</td>
<td>1560</td>
<td>0.15</td>
<td>*</td>
<td>0.19</td>
<td>0.18</td>
<td>N.A.</td>
<td>N.A.</td>
<td>lower</td>
<td></td>
</tr>
<tr>
<td>f</td>
<td>38-50 kW</td>
<td>1910</td>
<td>0.23</td>
<td>*</td>
<td>0.51</td>
<td>0.06</td>
<td>0.03</td>
<td>inst.</td>
<td>inst.</td>
<td>lift</td>
</tr>
<tr>
<td>g</td>
<td>38-50 kW</td>
<td>1900</td>
<td>0.28</td>
<td>~ 0.06</td>
<td>0.44</td>
<td>0.11</td>
<td>0.10</td>
<td>0.02</td>
<td>lower</td>
<td></td>
</tr>
<tr>
<td>h</td>
<td>38-50 kW</td>
<td>1470</td>
<td>0.22</td>
<td>*</td>
<td>0.19</td>
<td>0.10</td>
<td>0.07</td>
<td>inst.</td>
<td>inst.</td>
<td>lower</td>
</tr>
<tr>
<td>i</td>
<td>38-50 kW</td>
<td>1230</td>
<td>0.31</td>
<td>*</td>
<td>0.26</td>
<td>0.11</td>
<td>0.06</td>
<td>0.08</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>j</td>
<td>50-70 kW</td>
<td>1110</td>
<td>0.25</td>
<td>*</td>
<td>0.33</td>
<td>0.12</td>
<td>0.20</td>
<td>0.08</td>
<td>lower/lift</td>
<td></td>
</tr>
<tr>
<td>k</td>
<td>50-70 kW</td>
<td>990</td>
<td>0.21</td>
<td>*</td>
<td>0.51</td>
<td>0.06</td>
<td>0.05</td>
<td>inst.</td>
<td>inst.</td>
<td>lift</td>
</tr>
<tr>
<td>l</td>
<td>50-70 kW</td>
<td>2000</td>
<td>0.18</td>
<td>*</td>
<td>0.30</td>
<td>0.06</td>
<td>0.02</td>
<td>0.04</td>
<td>lift</td>
<td></td>
</tr>
<tr>
<td>m</td>
<td>50-70 kW</td>
<td>1030</td>
<td>0.31</td>
<td>~ 0.14</td>
<td>0.62</td>
<td>0.28</td>
<td>0.07</td>
<td>0.05</td>
<td>lower</td>
<td></td>
</tr>
<tr>
<td>n</td>
<td>50-70 kW</td>
<td>1560</td>
<td>0.18</td>
<td>*</td>
<td>0.19</td>
<td>0.12</td>
<td>0.09</td>
<td>inst.</td>
<td>inst.</td>
<td>lower</td>
</tr>
</tbody>
</table>

N.A. = not available

Table 7.1: Summary of control response of 16 tractors.
ends of the lower links.

The tractor engine was maintained at its rated speed for the tests and the hydraulic oil allowed to warm up before any readings were taken. Since the work load on the hydraulics was small, the oil normally reached a maximum of 25-30°C and remained at this throughout the tests.

To measure the control deadband, the top link force was varied above and below a set-point until the lift arms just began to raise or lower. This was repeated with the draught control lever at different set-points to represent different implements. The difference between the force necessary to cause lifting and that necessary to cause lowering was the deadband i.e. the maximum amount that the error signal could change before a response occurred.

At each set-point both compressive and tensile loads were applied in steps of 1100 N to measure the effect of different levels of error signal on the rate of movement of the lift arms. A solenoid valve ensured that the top link force was applied instantly so the delay time between force application and movement of the lift arms could be accurately measured.

3.4.2 Results and discussion

Table 3.1 is a summary of the results obtained from the rig for 14 tractors divided into the three power classes shown. Fig 3.30 is a diagram of a typical draught control response which defines the measured parameters.

The range of deadbands on modern tractors was from 920 to 2340 N, the value of 450N for the 15 year old tractor (a) was probably due to wear. The maximum rates of lift varied considerably, from 0.15 to 0.60 m/s and five of the tractors had a control to vary the rate of lift. All except four of the tractors had on-off control when lifting and proportional control when lowering. The delay time was the most difficult parameter to measure because the recordings were on a paper chart and the accuracy with which the times could be measured was no
Fig 3.30 Typical draught control response to define parameters quoted in Table 3.1.
better than 0.005 s. Another factor which may have influenced the delay time was that it was impossible to obtain a perfect step input. The build up of pressure in the ram took a finite time and also the ram usually had to move a small distance as the free play in the tractor linkage which was connected to the rig was taken up. Nevertheless, conditions were similar for all tractors and so the figures give a good comparison between them.

Another test which was carried out using the rig demonstrated that reasonable accuracy was being obtained. A tractor whose draught control characteristics could be changed by altering one of the hydraulic valves was lent by a manufacturer. The effect of the valves had been tested previously in field conditions. The results are shown in Table 3.2 and it can be seen that the valve size had a significant effect on both deadband and delay time.

Using the instability criterion developed previously (eqn 3.55), the optimum valve size can be calculated. From Fig 2.11 ploughing a medium/heavy soil at 270 mm gives a set force in the top link of 2250 N and a draught force of 12300 N. The rate of lift, \( \dot{y}_0 \) from Table 3.2 is 0.22 m/s and a reasonable forward speed to assume is 1.8 m/s. With the linkage on this tractor set for ploughing the ratio of top link force to force on the sensing element is 2:1, so inserting these figures into equation 3.55 gives a critical deadband of

\[
\frac{2 \times 12300 \times 0.22}{1.8 \times 2} = 1500 \text{ N}
\]

From Table 3.2 in the column marked set 2250 N, the critical deadband figure lies between 1460 N for valve 59 and 1530 for valve 1/32. Instability will not therefore occur if valve 1/32 is used but will occur if valve 59 is used since the deadband is smaller than the critical value. This predicted optimum valve size agreed with the manufacturers results from field experiments in similar soil conditions (129).

3.5 EXPERIMENTAL IMPLEMENT CONTROL

In this section the experimental implement control, built to measure the effect of control parameters on field performance, is
<table>
<thead>
<tr>
<th>Value</th>
<th>Relative area of orifice</th>
<th>Deadbands N</th>
<th>Delay</th>
<th>Maximum rate of lift m/s</th>
<th>Maximum rate of lower m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Set 0</td>
<td>Set 2250N</td>
<td>Set 4500N</td>
<td>Set 6750N</td>
</tr>
<tr>
<td>Production</td>
<td></td>
<td>1220</td>
<td>1830</td>
<td>2240</td>
<td>2420</td>
</tr>
<tr>
<td>75</td>
<td>1.0</td>
<td>760</td>
<td>1800</td>
<td>2110</td>
<td>2700</td>
</tr>
<tr>
<td>1/32</td>
<td>2.2</td>
<td>630</td>
<td>1530</td>
<td>1490</td>
<td>1830</td>
</tr>
<tr>
<td>59</td>
<td>3.8</td>
<td>780</td>
<td>1460</td>
<td>1600</td>
<td>1600</td>
</tr>
<tr>
<td>53</td>
<td>8.0</td>
<td>840</td>
<td>1040</td>
<td>1280</td>
<td>1560</td>
</tr>
<tr>
<td>49</td>
<td>12.0</td>
<td>750</td>
<td>840</td>
<td>940</td>
<td>870</td>
</tr>
<tr>
<td>44*</td>
<td>16.0</td>
<td>650</td>
<td>1090</td>
<td>1360</td>
<td>1750</td>
</tr>
</tbody>
</table>

* carried out at engine speed of 1000 rev/min to achieve reasonable rate of lowering all others carried out at rated speed of 1800 rev/min.

Table 3.2 Effect of varying valve size on a particular tractor
described. Some preliminary measurements of the time delays at various points in the hydraulics circuit are also described. These were carried out because the theoretical analysis in sections 3.2 and 3.3 had shown the importance of the delay time, and it was necessary to measure it accurately.

3.5.1 Equipment

The experimental implement control had been built for the N.I.A.E. project on implement controls and was completed but not developed before the work for this thesis began. Basically it was an electro-hydraulic control which was fitted to a 52 kW tractor. A strain gauged dynamometer linkage (119), fitted between tractor and plough, sensed draught and vertical force on the implement and a rotary potentiometer mounted on the cross shaft sensed lift arm rotation. Electrical signals proportional to draught, vertical force and lift arm position were fed to an electronic control box (106) where they could be individually selected or mixed and then amplified. The combined error output then controlled the operation of an electro-hydraulic valve which in turn controlled the flow of oil from the tractor external hydraulics to rams which lifted or lowered the linkage. Originally a proportional valve was fitted but this proved impractical because it required an extremely high level of oil filtration to operate without clogging and it was replaced by an on-off type solenoid valve. The experimental control is described in detail elsewhere (106) and its operation is discussed further in later sections describing field measurements.

3.5.2 Measurement of time delays

For these experiments, dummy signals were switched into the circuit at the control box and the other inputs disconnected so that the moment of application of an error signal could be accurately controlled. Four parameters were monitored and recorded on a chart recorder;

1. input signal,
2. movement of electro-hydraulic valve,
3. oil pressure, (a) immediately downstream of valve,  
   (b) at hydraulic rams,  
4. movement of lift arms.  
   A linear potentiometer was mounted in contact with the spool  
   of the electro-hydraulic valve so that displacement either side of  
   the mean position could be measured. Oil pressure was measured by  
   connecting a pressure transducer at the appropriate point in the  
   hydraulic circuit. Movement of the lift arms was measured by a  
   rotary potentiometer on the lift shaft. Fig 3.31 shows the relative  
   positions of each monitoring point in the circuit.  
   A three furrow plough was held on the linkage while dummy  
   signals initiating the following four control movements were switched  
   into the circuit;  
   (a) starting lifting;  
   (b) stopping lifting;  
   (c) starting lowering;  
   (d) stopping lowering;  
   For (a) and (c) the plough was initially stationary whereas  
   for (b) and (d) the plough was initially lifting and lowering  
   respectively. The four signals were repeated three times and average  
   values for the delays calculated from the chart recordings. Only  
   one pressure transducer was available so it was fitted at point B.  
   (Fig 3.31) initially and the experiment repeated with it fitted at  
   C.  
   Further tests  
   It was noticed during the tests that the linkage behaved as if  
   it were supported by a spring. This was shown most clearly by a  
   recorded trace of lift arm rotation when a 'stop lowering' signal  
   was given, as shown in Fig. 3.32.  
   The load/deflection relationship for the hydraulic lift was  
   measured by loading the ends of the lower links in 2250 N steps up  
   to 22500 N and measuring the deflection relative to the tractor.
Fig 3.31 Block-Diagram of Electro-Hydraulic Circuit Showing Monitoring Positions

A - Solenoid valve movement
B - Oil pressure immediately downstream of valve
C - Oil pressure at rams
D - Movement of linkage
Fig 3.32 Recorded trace of lift arm rotation when a dummy 'stop dropping' signal given.
This procedure was repeated with the load decreasing to check for oil leakage. For comparison, the load/deflection relationship was measured for a commercial draught control whose delay time was already known.

3.5.3 Results

Delay times

Table 3.3 shows the delay times between the signal and a displacement or pressure change occurring at the particular point in the circuit (Fig 3.31).

The figures in column D are the overall delay times between the error signal and lift arm movement. Averaging the two columns marked D gives overall delay times of:

1. Starting lifting 0.058 s
2. Stopping lifting 0.036 s
3. Starting dropping 0.027 s
4. Stopping dropping 0.071 s

The scatter of experimental results was up to 20% either side of the average figure but that the averages were sufficiently repeatable is shown by a comparison of the two columns marked D. The delays for stopping lifting and starting dropping are approximately half the values obtained for the other two, because the forces applied to the hydraulic lift are much smaller.

The delay times in Table 3.3 were recorded with the plough suspended above the ground but they should not be significantly different when the plough is in the ground, with one exception. The delay for stopping lowering depends on the rate of lowering of the plough which is smaller when the plough is in the ground and consequently the delay will be smaller.

Compliance

Both load/deflection characteristics, shown in Fig 3.33 were approximately straight lines after an initial curve up to about 4500 N. Slight leakage or hysteresis occurred in each case and
### Table 3.3

Delay times at four points in the control circuit

1. **Pressure Transducer at B**

<table>
<thead>
<tr>
<th>Type of Signal</th>
<th>On/Off</th>
<th>Delay Times, s</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>A</td>
</tr>
<tr>
<td>Lift</td>
<td>On</td>
<td>0.015</td>
</tr>
<tr>
<td>Lift</td>
<td>Off</td>
<td>0.012</td>
</tr>
<tr>
<td>Lower</td>
<td>On</td>
<td>0.016</td>
</tr>
<tr>
<td>Lower</td>
<td>Off</td>
<td>0.022</td>
</tr>
</tbody>
</table>

2. **Pressure Transducer at C**

<table>
<thead>
<tr>
<th>Type of Signal</th>
<th>On/Off</th>
<th>Delay Times, s</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>A</td>
</tr>
<tr>
<td>Lift</td>
<td>On</td>
<td>0.014</td>
</tr>
<tr>
<td>Lift</td>
<td>Off</td>
<td>0.016</td>
</tr>
<tr>
<td>Lower</td>
<td>On</td>
<td>0.020</td>
</tr>
<tr>
<td>Lower</td>
<td>Off</td>
<td>0.017</td>
</tr>
</tbody>
</table>
Fig. 3.33 Load/deflection curves for N.I.A.E. experimental control and a commercial control measured at lower link ends.
when the load was removed there was a difference of approximately 7.5 mm from the initial zero reading. The commercial control was about two and a half times stiffer than the N.I.A.E. control, the respective stiffness being 675 KN/mm and 228 KN/mm.

A simplified analysis of the effect of compliance in the linkage is given in Appendix 3.IV and it is shown that compliance causes a delay time of about 0.05 s for the experimental control and 0.03 s for the commercial control.

3.5.4 Discussion

Comparison of measured and theoretical delays

For the N.I.A.E. control, the theoretical delay due to compliance in the hydraulic lift is about 0.05 s. Adding the average delay of 0.016 s before the electro-hydraulic valve operates gives an overall delay of about 0.066 s, which compares well with the average measured value of 0.058 s.

The overall delay for the commercial control has been measured previously (tractor k in Table 3.1) though with less accuracy than in the present work, and was found to be approximately 0.05 s. The theoretical delay due to compliance is about 0.03 s but presumably there is also some delay for the hydraulic valve to operate which should be added to this figure.

From these two results it is reasonable to assume that the overall delay for draught controls has two components;

1. a small delay for the hydraulic valve of electro-hydraulic valve to operate,

2. a longer delay due to compliance in the hydraulic lift.

Some delay however small will occur for any valve controlling the flow of oil so it is unlikely that the value of approximately 0.01 s can be improved upon significantly. It should be possible however to reduce the delay due to compliance, which is of the order of 0.03 - 0.05 s.
Compliance

Two sources of compliance on the N.I.A.E. control are:-
1. small amounts of air trapped in the hydraulic circuit,
2. expansion of the flexible hydraulic pipes.

These sources probably account for the greater part of the compliance of the N.I.A.E. control compared with the other control measured. Slight aeration of the oil could be cured by inserting a bleed valve at the highest point in the circuit but replacing the flexible pipes would involve a re-design of the control. Metal pipes were used initially but were found to be unreliable because of vibration caused by sudden pressure changes.

It should be possible to reduce the compliance even further until the only source is the compressibility of the hydraulic oil which would give a hydraulic lift stiffness of $3.4 \times 10^6 \text{ N/mm}$ at the ends of the lower links (Appendix 3.IV). Using the previous theoretical analysis and assuming the same damping ratio gives a delay time of 0.02 s due to compressibility of oil. This was calculated for the N.I.A.E. control which has a greater volume oil to compress and hence more compliance than most commercial controls, so the delay time for commercial controls due to compressibility may be slightly less than 0.02 s.

Scope for improvements

Apart from aeration of the oil and expansion of flexible piping which are restricted to the N.I.A.E. control, other possible causes of compliance are expansion of pipes, oilways and seals. Clearly the volume of oil under pressure should be kept to a minimum. Mechanical components such as the lift shaft, arms and rods are already sufficiently strong and it is unlikely that they contribute significantly to the compliance.

3.6 Field Measurements with Experimental Implement Control - Sinusoidal Draught Variation

3.6.1 Equipment and procedure

The experimental implement control used for these measurements was described in section 3.5. The signal which controlled the
<table>
<thead>
<tr>
<th>Field number</th>
<th>Surface</th>
<th>Mean draught for 3 furrow plough at 230 mm depth N</th>
<th>Specific Soil resistance at 230 mm depth N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Rotary cultivated after brussel sprouts</td>
<td>9600</td>
<td>0.039</td>
</tr>
<tr>
<td>3.2</td>
<td>Stubble</td>
<td>10500</td>
<td>0.043</td>
</tr>
<tr>
<td>3.3</td>
<td>Rotary cultivated stubble</td>
<td>10100</td>
<td>0.041</td>
</tr>
<tr>
<td>3.4</td>
<td>Rotary cultivated after brocoli</td>
<td>8500</td>
<td>0.035</td>
</tr>
<tr>
<td>3.5</td>
<td>Rotary cultivated after cabbages</td>
<td>9500</td>
<td>0.039</td>
</tr>
</tbody>
</table>

Table 3.4 Fields used for measurements with experimental control and tractor fitted with eccentric wheels
electro-hydraulic valves could be altered by varying the inputs to
the electronic control box. In this work, pure draught sensing was
arranged by selecting the input from the draught force only and top
link sensing by selecting the draught and vertical force in the
correct proportion (given in eqn 2.20). The tractor was originally
fitted with a pivoted wheel mounted under the belly of the tractor
to sense tractor pitch which could then be used as an input to the
control. However, the wheel was found to be too sensitive to ground
irregularities and not therefore used.

The input draught disturbance was a sinusoidal draught variation
obtained by setting the tractor wheels eccentric by 50 mm. This
produced an implement depth variation of ±90 mm. A 52 p.t.o. kW
tractor and three furrow mounted plough were used for all the experi-
ments.

Measurements were made in the five fields described in Table 3.4.
They were all light soils because it was necessary that the wide
depth variation did not cause tractor stall. The effect of the
following variables was investigated:
a) sensing mode, i.e. pure draught or simulated top link
sensing,
b) width of deadband,
c) rate of lift,
d) forward speed.

Draught force was measured with the three-point linkage
dynamometer (119) and recorded on an oscillograph. The draught
variation measured was the amplitude of the resulting recording for
a particular set of control parameters.

3.6.2 Results and discussion

The measured draught variations in each field for various
control settings and speeds are shown in Table 3.5. The results
are similar for all fields although fields 2 and 3 which were slightly
heavier produced higher draught variations. If the results are
<table>
<thead>
<tr>
<th>Mode</th>
<th>Deadband N</th>
<th>Rate of lift</th>
<th>Forward speed m/s</th>
<th>Draft variation in field number</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>3.1</td>
<td>3.2</td>
</tr>
<tr>
<td>Pure draught sensing</td>
<td>1800</td>
<td>270</td>
<td>1.35</td>
<td>2620</td>
</tr>
<tr>
<td></td>
<td>3600</td>
<td>270</td>
<td>1.35</td>
<td>4000</td>
</tr>
<tr>
<td></td>
<td>5400</td>
<td>270</td>
<td>1.35</td>
<td>4090</td>
</tr>
<tr>
<td></td>
<td>1800</td>
<td>180</td>
<td>1.35</td>
<td>5070</td>
</tr>
<tr>
<td></td>
<td>1800</td>
<td>90</td>
<td>1.35</td>
<td>4720</td>
</tr>
<tr>
<td>Simulated link sensing</td>
<td>3600</td>
<td>270</td>
<td>1.35</td>
<td>4720</td>
</tr>
<tr>
<td></td>
<td>5400</td>
<td>270</td>
<td>1.35</td>
<td>5070</td>
</tr>
<tr>
<td></td>
<td>7200</td>
<td>270</td>
<td>1.35</td>
<td>4720</td>
</tr>
<tr>
<td></td>
<td>3600</td>
<td>180</td>
<td>1.35</td>
<td>4720</td>
</tr>
<tr>
<td></td>
<td>3600</td>
<td>90</td>
<td>1.35</td>
<td>4720</td>
</tr>
</tbody>
</table>

Table 3.5 Measured draught variations for experimental control fitted to tractor with eccentric wheels.
averaged over the five fields then three trends are clear; draught
variation increased with forward speed but decreased with a reduction
in deadband or increase in rate of lift.

The minimum deadband and maximum rate of lift chosen for each
sensing mode was limited by instability. It was possible however, as
predicted in sections 3.2 and 3.3, to use a small deadband for pure
draught sensing than top link sensing before instability occurred.
At the same deadband, top link sensing was better than pure draught
sensing, but when the minimum deadband possible before instability
was used for each control, then performance was similar.

3.6.3 Comparison of measured and predicted results

Theoretical predictions of the draught variations in each field
obtained by running the computer programme simulation described in
section 2.5 are shown in figs 3.34 to 3.39 together with the measured
results.

Fig. 3.34 shows the effect of speed for the pure draught sensing
mode. The separate lines are the theoretical predictions for each
field though there is little difference between them. The measured
points are the average of the 3600 and 5400 N deadbands at the highest
rate of lift. Similar results for the simulated top link sensing mode
are shown in fig. 3.35. In both cases the measured and predicted
trends are similar despite some experimental scatter. The field
condition had more effect on the top link sensing mode and this was
correctly predicted suggesting that the assumptions made in the simulation
were valid.

Varying the deadband had a marked effect on both controls as shown
in figs. 3.36 and 3.37. Again the predicted results were similar to
the experimental points although accuracy was better for the pure
draught sensing.

The effect of varying rate of lift is shown for each control in
figs. 3.38 and 3.39. Agreement is satisfactory for pure draught sensing
where the heavier soil in fields 2 and 3 has produced a noticeable
Fig 3.34 Increase in draught variation with speed for pure draught

Sensing corner

Forward speed m/s 2.2 2.0 1.8 1.6 1.4

Draught variation

Points - measured

Lines - predicted
Fig 3.35 Increase in draught variation with speed for top link sensing control
Fig 3.36 Increase in draught variation with deadband for top link sensing control.
Fig 3.37 Increase in draught variation with deadband for pure draught sensing control.
Fig 3.38 Decrease in draught variation with rate of lift for top link sensing control.
increase in draught variations at the lower rates of lift. For top link sensing the experimental results contain a lot of scatter and comparison is difficult. The results are averaged over the three speeds and the computer predicted engine stall in fields 3.2 and 3.3 in 5th gear for the lowest rate of lift. This occurred in field 3.3 during field-work but not in field 3.2 although the engine did slow down.

3.6.4 Conclusions

In general, agreement between measured and predicted results was satisfactory and the effect of a) sensing mode, b) speed, c) deadband and d) rate of lift on control performance could be predicted. Agreement was better for pure draught sensing than for top link sensing. This was probably because of the added complexity of predicting the vertical soil forces. There were no clear differences in performance between each sensing mode; pure draught sensing allowed the use of a smaller deadband before instability occurred but performance was only similar to the top link sensing control with a larger deadband.

3.7 FIELD MEASUREMENTS WITH EXPERIMENTAL IMPLEMENT CONTROL – RANDOM DRAUGHT VARIATIONS

3.7.1 Equipment and procedure

For these experiments the experimental control was again used fitted to a 52 p.t.o. kW tractor but three implements were used; a) three furrow mounted plough, b) seven rigid tine cultivator, c) three-furrow semi-mounted plough.

Naturally occurring field surfaces were used to provide random draught variation and the same parameters as before were varied. Measurements were made in the eleven field conditions described in Table 3.6, which have been classified into four groups. Each test run for particular control setting and implement lasted for approximately one minute during which the draught force was recorded on magnetic tape. The recordings were later analysed on a PDP8E computer by sampling the draught force every 100 ms and calculating the mean and standard deviation. In each field recordings were also taken at three
<table>
<thead>
<tr>
<th>Field No.</th>
<th>Group</th>
<th>Implement</th>
<th>Field description and comments</th>
<th>Mean draught (N)</th>
<th>Specific soil resistance (N/mm² at 228 mm depth)</th>
<th>Standard deviation of draught force in position control (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.6</td>
<td></td>
<td>Mounted</td>
<td>3 furrows - medium loam with stones - stubble</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.7</td>
<td>I</td>
<td>Mounted plough</td>
<td>3 furrows - medium loam - stubble</td>
<td>13150</td>
<td>0.053</td>
<td>1970</td>
</tr>
<tr>
<td>3.8</td>
<td></td>
<td></td>
<td>3 furrows - light/medium loam - stubble</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.9</td>
<td></td>
<td></td>
<td>2 furrows - heavy loam - uneven surface</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.10</td>
<td>II</td>
<td>Mounted plough</td>
<td>2 furrows - heavy loam - stubble</td>
<td>13300</td>
<td>0.069</td>
<td>1910</td>
</tr>
<tr>
<td>3.11</td>
<td></td>
<td></td>
<td>3 furrows - ploughed shallower - med/ heavy loam</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.12</td>
<td>III</td>
<td>Cultivator</td>
<td>Sandy loam - stubble</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.13</td>
<td></td>
<td></td>
<td>Sandy/medium loam - stubble</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.14</td>
<td></td>
<td></td>
<td>Sandy/medium loam - stubble</td>
<td>13150</td>
<td>*</td>
<td>1710</td>
</tr>
<tr>
<td>3.15</td>
<td>IV</td>
<td>Semi-mounted</td>
<td>Clay - stubble</td>
<td>13100</td>
<td>0.053</td>
<td>2000</td>
</tr>
<tr>
<td>3.16</td>
<td></td>
<td>plough</td>
<td>Peaty - uneven rotary cultivated surface</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* Not applicable

Table 3.6 Summary of field conditions for measurements with random draught variations.
<table>
<thead>
<tr>
<th>Sensing mode</th>
<th>Deadband (N)</th>
<th>Forward speed (m/s)</th>
<th>Rate of lift (mm/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>140</td>
<td>220</td>
</tr>
<tr>
<td>Pure draught</td>
<td>/</td>
<td>1.35</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>C</td>
<td>HCL</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td>Simulated top or lower link</td>
<td>1.35</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>C</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.35</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>A</td>
<td>A</td>
</tr>
</tbody>
</table>

Table 3.7 Summary of predicted unstable control settings

L = Unstable with mounted plough - light soil  C = Unstable with cultivator
H = Unstable with mounted plough - heavy soil  A = All stable

Where instability is predicted for an implement, the relevant symbol is marked against the control settings. If only one symbol is marked at the control settings then the others are stable.
<table>
<thead>
<tr>
<th>Group</th>
<th>Forward speed (m/s)</th>
<th>Rate of lift (mm/s)</th>
<th>Deadband (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.35</td>
<td>1.80</td>
<td>2.25</td>
</tr>
<tr>
<td>I</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mounted plough</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Light soil</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Predicted ratio</td>
<td>.69</td>
<td>.75</td>
<td>.83</td>
</tr>
<tr>
<td>Measured ratio</td>
<td>.71</td>
<td>.79</td>
<td>.79</td>
</tr>
<tr>
<td>II</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mounted plough</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heavy soil</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Predicted ratio</td>
<td>.68</td>
<td>.73</td>
<td>.85</td>
</tr>
<tr>
<td>Measured ratio</td>
<td>.69</td>
<td>.73</td>
<td>.79</td>
</tr>
<tr>
<td>III</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cultivator</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Predicted ratio</td>
<td>.68</td>
<td>.78</td>
<td>.90</td>
</tr>
<tr>
<td>Measured ratio</td>
<td>.65</td>
<td>.77</td>
<td>.82</td>
</tr>
<tr>
<td>IV</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Semi-mounted plough</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Predicted ratio</td>
<td>.85</td>
<td>.90</td>
<td>.97</td>
</tr>
<tr>
<td>Measured ratio</td>
<td>.75</td>
<td>.82</td>
<td>1.13</td>
</tr>
</tbody>
</table>

Table 3.8 Comparison of measured and predicted standard deviation ratios - Pure draught sensing
different speeds with the implement locked in position relative to the tractor (position control), and the average used as a measure of the variability of the field surface. The ratio of standard deviation at a particular control setting to standard deviation in position control was used as a measure of performance.

Speeds above 2.25 m/s required an additional tractor to provide adequate power and were only used in fields 3.8 and 3.11.

The control settings at which instability was predicted are shown in Table 3.7. The high rate of penetration of the cultivator decreases stability in both sensing modes. Higher values of deadband are shown for the top or lower link sensing mode because this mode is inherently less stable than pure draught sensing.

3.7.2 Results and discussion

A summary of the results for the two control sensing modes is shown in Tables 3.8 and 3.9. The criterion used as an overall measure of performance was the ratio.

\[
\text{Standard deviation in particular control setting} \quad \text{Standard deviation in position control setting}
\]

The results for varying forward speeds in columns 4-6 are the averages of the results for the two smaller deadbands at the maximum rate of lift. The results for varying rates of lifts in columns 7-9 are the averages of the three speeds at the smallest deadband and the results for varying deadbands in columns 10-12 are the averages of the three speeds at the maximum rate of lift.

(1) Mounted plough

Instability occurred when ploughing light soil for both pure draught and simulated top link sensing causing high standard deviation ratios for the smallest deadband and highest rate of lift. In the heavy soil conditions, instability occurred much less frequently with the same deadband and rate of lift probably because of the lower rate of penetration and smaller vertical forces. Otherwise no clear differences in performance were shown between the two soil types or the two sensing modes.
<table>
<thead>
<tr>
<th>Group</th>
<th>Mounted plough</th>
<th>Measured ratio</th>
<th>Predicted ratio</th>
<th>Forward speed (m/s)</th>
<th>Rate of lift (mm/s)</th>
<th>Deadband (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Light soil</td>
<td>.79</td>
<td>.70</td>
<td>1.35</td>
<td>270</td>
<td>.77</td>
</tr>
<tr>
<td></td>
<td></td>
<td>.79</td>
<td>.76</td>
<td>1.80</td>
<td>220</td>
<td>.72</td>
</tr>
<tr>
<td></td>
<td></td>
<td>.80</td>
<td>.84</td>
<td>2.25</td>
<td>140</td>
<td>.74</td>
</tr>
<tr>
<td>I</td>
<td>Mounted plough</td>
<td>.77</td>
<td>.74</td>
<td></td>
<td></td>
<td>.77</td>
</tr>
<tr>
<td></td>
<td>Light soil</td>
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<td>.60</td>
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<td></td>
<td>.76</td>
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<td></td>
<td></td>
<td>.76</td>
<td>.60</td>
<td></td>
<td></td>
<td>.86</td>
</tr>
<tr>
<td>II</td>
<td>Mounted plough</td>
<td>.70</td>
<td>.74</td>
<td></td>
<td></td>
<td>.74</td>
</tr>
<tr>
<td></td>
<td>Heavy soil</td>
<td>.70</td>
<td>.61</td>
<td></td>
<td></td>
<td>.60</td>
</tr>
<tr>
<td></td>
<td></td>
<td>.71</td>
<td>.61</td>
<td></td>
<td></td>
<td>.86</td>
</tr>
<tr>
<td>III</td>
<td>Cultivator</td>
<td>.70</td>
<td>.73</td>
<td></td>
<td></td>
<td>.75</td>
</tr>
<tr>
<td></td>
<td></td>
<td>.76</td>
<td>.71</td>
<td></td>
<td></td>
<td>.81</td>
</tr>
<tr>
<td></td>
<td>Predicted ratio</td>
<td>.75</td>
<td>.88</td>
<td></td>
<td></td>
<td>.88</td>
</tr>
<tr>
<td></td>
<td></td>
<td>.74</td>
<td>.90</td>
<td></td>
<td></td>
<td>.88</td>
</tr>
</tbody>
</table>

Table 3.9 Comparison of measured and predicted standard deviation ratios - simulated top link sensing.
(ii) Cultivator

Better results were obtained with the cultivator for pure draught sensing than for simulated top link sensing. In the latter case the vertical force which increased with depth caused a force in the top link opposing the force due to the draught force with a consequent deterioration in performance. Also, instability occurred frequently in the simulated top link sensing mode causing high values of standard deviation ratio with a deadband of 3600 N and a rate of lift of 270 mm/s.

(iii) Semi-mounted plough

Because a semi-mounted plough only normally transmits horizontal forces to the sensing element of a control, results are only shown for pure draught sensing. The results showed that semi-mounted ploughing was unsuitable for the three furrow plough on uneven surfaces, the standard deviation ratio exceeding unity several times. The reasons were that large changes in inclination were necessary to alter the draught force significantly and at negative inclinations (i.e. the front share is shallower than the rear share) the rate of penetration was low so that the plough took too long to return to its correct depth. For the semi-mounted arrangement to be suitable for short implements large changes in inclination must not be necessary and this work with a three furrow plough showed that it would only be satisfactory for relatively even surfaces and consistent soil conditions.

Semi-mounted ploughing has advantages over mounted ploughing when long implements are used because the vertical force on the implement is not sensed. As implements become longer the distance between the line of action of the vertical force and the lower link ends increases until the moment of the vertical force about the lower link ends is greater than the moment of the draught force. The force in the top link is then dominated by the vertical force and top link sensing control is unsatisfactory. The vertical force is not sensed if a semi-mounted plough is used with lower link sensing control and the instability criterion is similar to that for pure draught sensing.
3.7.3 Comparison of measured and predicted results

Agreement between measured and predicted results was generally satisfactory and the predominant trends of the measured results were simulated with acceptable accuracy although there were some discrepancies in individual results. For instance, semi-mounted ploughing performance was correctly predicted as unsatisfactory but the computer predictions were not as bad as the measured performance. The reason was probably that the soil resistance was assumed constant in the computer simulation but varied slightly in the field. If the control responded to these variations by altering the plough inclination then the rate of penetration, which was very low for zero or negative inclinations, decreased and hence the performance deteriorated.

For both sensing modes at a rate of lift of 140 mm/s computer predictions were up to 25% worse than the measured values. The reason for this could not be found.

With a fully mounted plough in light soil in the pure draught sensing mode, the computer simulation did not predict instability at the high sensitivities and consequently underestimated the standard deviation ratio by 15%. The factors affecting instability are delay time, rate of lifting or lowering and the rate of change of draught with depth. The error was probably caused by the rate of change of draught with depth transiently exceeding its steady state value, as indicated by draught force recordings.

At forward speeds above 2.25 m/s performance deteriorated rapidly, the standard deviation of draught force doubling for an increase in speed from 1.35 to 2.8 m/s. The corresponding standard deviations of depth are 12.7 mm at 1.35 m/s and 33 mm at 2.8 m/s, the latter variation resulting in unburied trash and subsoil being brought to the surface.

Reasonable agreement was obtained between measured and predicted values at higher speeds as shown in Fig. 3.40. The results shown are the averages for the results at the two smallest deadbands and two
Fig 3.40 Increase in draught variation with forward speed
highest rates of lift. At 2.8 m/s the predicted standard deviation was 9% lower than the measured value and is thought to be due to tractor bouncing causing greater movement of the plough which was not included in the simulation.

3.7.4 Control stability

Figs 3.41 and 3.42 are typical recordings taken during fieldwork, showing unstable operation. For the pure draught control sensing control (fig 3.41), the following control parameters were used;

\[ DB = 900 \text{ N} \]
\[ \dot{y}_0 = 270 \text{ mm/s} \]
\[ k_1 = 60 \text{ N/mm at 230 mm depth approx.} \]

Inserting these figures in eqn 3.25 and using the measured value of \( t_d \) of 0.057 s for starting lifting, gives

\[ 900 < 60 \times 0.057 \times 270 \]
\[ 900 < 920 \]

which predicts instability of frequency \( \left( \frac{1}{4t_d} \right) \text{Hz} = 4.4 \text{ Hz.} \)

For the top link sensing control, the following parameters were used;

\[ DB = 3600 \text{ N} \]
\[ \dot{y}_0 = 270 \text{ mm/s} \]
\[ k_2 = \frac{2}{2/1350} = \frac{2}{1350} \text{ s/mm} \]
\[ H_{\text{mean}} = 13.5 \text{ kN} \]

substituting in eqn. 3.55 gives

\[ 3600 < \frac{2}{1350} \times 13500 \times 270 \]
\[ 3600 < 5400 \]

which predicts instability, with limit cycle frequency equal to \( \left( \frac{1}{4t_d} \right) \text{Hz}, \)
i.e. 8.8 Hz. This agrees reasonably well with the measured value of 9.5 Hz.

3.7.5 Conclusions

1. For the mounted plough, the effects of control parameters on field performance were similar to those found previously using a
Fig 3.41 Draught and vertical force recordings for pure draught sensing control showing unstable operation (limit cycle frequency = 4.5 Hz)
Fig 3.42 Draught and vertical force recordings for top link sensing control showing unstable operation (unit: cycle frequency: 9.5 Hz)
sinusoidal draught variation.

2. For the cultivator, control was generally more difficult than for the plough mainly because of the different vertical force characteristics.

3. Semi-mounting the three furrow plough was not successful in the field conditions used for measurements. The changes in inclination of the plough during correction signals were too great.

4. Speeds up to 2.8 m/s were used for mounted ploughing and both pure draught and top link sensing control performance were found to deteriorate rapidly above the present typical maximum ploughing speed of 2 m/s.

3.8 DAMPING TOP LINK MOVEMENT TO IMPROVE CONTROL PERFORMANCE

3.8.1 Introduction

One method of improving draught control performance without major modification to existing designs is to damp the top link movement. This should have the advantages of:

1) improving stability;

2) preventing unnecessary control signals when the top link force changes due to shock load or vibrations rather than changes in draught force;

3) reducing unnecessary movement of the spool valve due to high frequency force fluctuations and thus reducing wear.

The theory behind the improvement to stability was discussed in section 3.3 and it was suggested that force fluctuations below about 6 Hz should be attenuated.

The two main limitations to mounting a damper at the top link sensing unit are the lack of available space and the small amount of movement over which the damper must operate. Originally it was proposed to use a piston damper but the size of damper which would provide a sufficient damping force was too large to be mounted near the top link. Hence two rotary dampers were used and a linkage was designed to transmit the top link movement to the dampers which could then be
positioned just above the sensing unit. The detail design and equations to predict the response time are shown in Appendix 3.V. A lever ratio of 3.5:1 was incorporated in the linkage to increase the ratio of damper to top link movement, because the maximum likely movement of the top link sensing element when ploughing was estimated as only 6-7 mm.

3.8.2 Laboratory measurements

The damper was fitted to a tractor whose draught control characteristics could be varied. An experimental valve provided by a manufacturer (129) enabled the response rate of the control to be increased enough to cause instability. Measurements were therefore made of the effect of the damper on the standard control alone and with the experimental valve fitted (referred to in the following text as standard and experimental control respectively). The reason for including the experimental control was to measure the effect of the damper in improving control stability. Measurements were made for each control with the dampers at their maximum and minimum settings and with no damping at all.

The measured delay times are plotted against the magnitude of the step force for the standard control in Fig. 3.43 for the experimental control in Fig. 3.44. The predicted values, calculated as shown in Appendix 3V are also plotted. The amount of damping also had an effect on the control deadband as shown in Table 3.10.

<table>
<thead>
<tr>
<th>Control with experimental valve</th>
<th>Damping</th>
<th>Deadband, N</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zero</td>
<td>660</td>
<td></td>
</tr>
<tr>
<td>min.</td>
<td>990</td>
<td></td>
</tr>
<tr>
<td>max.</td>
<td>1520</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.10
Deadbands measured for various amounts of damping
Fig 3.43 Measured and predicted delays for standard control
Fig 3.44 Measured and predicted delays for experimental control
Theoretically, damping the top link movement should not have any effect on the deadband but in practice, as can be seen from Table 3.10, it increased the deadband by up to three times for the standard control. The reason for this is probably the internal friction in the damper. The deadband was measured by increasing the top link force slowly until the lift arms began to lift and then decreasing the force until they lowered. If the dampers needed some force to overcome the internal static friction then this force would increase the deadband. In field operation the deadband may not be as large because the damper and top link sensing unit are moving continuously.

Agreement between predicted and measured delays was better for the experimental control than for the standard control. The measurement techniques used in both cases were similar but it may be possible that insufficient care was taken to ensure that the initial force was zero before a step force was applied. If the initial force was not zero then measured delay times would be longer than those predicted as is the case for some of the results for the standard control.

3.8.3 Field measurements

Measurements were made of the effect of the damper on the same tractor as used for the laboratory measurements, again using the standard and experimental control.

Two fields were used for measurements, both sandy loam, and because the soil resistance and surface roughness were similar, the results have been averaged. A three furrow plough was used and measurements were made at three forward speeds with and without the damper operating. Draught force was recorded and later analysed to calculate the standard deviation which was used as a measure of how effective the control was in limiting draught fluctuations. Runs were also carried out in position control and the standard deviation of draught in this control with the plough rigidly attached to the tractor was a measure of the draught variation with no control. The ratio of

\[
\frac{\text{Standard deviation}}{\text{Standard deviation in position control}}
\]
is plotted against forward speed for the standard control in fig. 3.46 and for the experimental control in fig. 3.47.

3.8.4 Field results

The results for both controls, although they contain some experimental scatter, show a similar trend. Damping results in an inferior performance at low speed but above about 2.3 m/s it appears to improve performance considerably. This deterioration of low speed performance was expected for the standard control because the main effect of the damping is to increase the delay time. It was not expected for the experimental control because the damping should have increased stability and improved performance. It would have been useful to make measurements at a higher speed to confirm these trends but the tractor did not have sufficient power to operate at more than 2.3 m/s.

3.8.5 Discussion

The most interesting result is when the standard undamped control is compared with the experimental damped control (fig. 3.48). One of the claimed advantages of damping was that it should enable a higher rate of lift or smaller deadband to be used. The experimental control had a 15% smaller deadband than the standard control and should therefore have achieved better control if the damping did prevent instability. In fact, the improvement is shown clearly above about 2.1 m/s. This speed range may be important in future if higher cultivation speeds are used and improvement to controls will certainly be needed if tractors are to be operated efficiently at these speeds. The rapid deterioration in the performance of existing controls at high speed has been shown previously (fig. 3.40) and is also shown by the results for the undamped standard and experimental control.

During field measurements the experimental control was not completely unstable as expected, although it tended to induce an excessive number of response movements and make the driver uncomfortable. To the tractor operator or an observer the damper had no noticeable effect in reducing this excessive movement and, although the measurements made showed
Effect of damping on field performance of standard (fig 3.46) and experimental (fig 3.47) controls.
Fig 3.48 Averaged field results to compare effect of speed on undamped standard control and damped experimental control.
that it did have some effect, it was not as much as intended. This is probably due to two factors, backlash in the linkage and insufficient damper stroke. Although care had been taken at the design stage, there was some free play in the linkage which decreased the small percentage of the maximum damper stroke being used.

3.8.6 Comparison of measured and predicted performance

The computer programme for simulating the field performance of a tractor and implement was used to predict the performance of the undamped standard control and the damped experimental control. The simulated field condition was similar to that used for field measurements, the results are shown in Fig 3.49 along with the points measured in the field. The predicted performance of the damped experimental control was consistently better than the standard control and the improvement increased with speed. Above 2.1 m/s the trend was very similar to that measured in the field. The simulated damping made the control response similar to a first order lag of time constant 0.04 s, to which must be added the inherent 0.02 s pure delay in the experimental control. The pure time delay of the standard control was 0.05 s. The experimental deadband was 660 N, 15% smaller than for the standard control.

Despite some shortcomings in the design of the damper and linkage it achieved part of the damping necessary to improve draught control performance. The major constraint on the design is the small movement of the top link which has to be damped. The design is further complicated by the difficulties in mounting a damper at the existing draught control sensing unit because of the lack of space and suitable attachment points. The best design would be one in which the damper was integral with the sensing unit and would have to be incorporated in the original design. It would then be possible to provide a bigger damper which moved more oil over small displacements. The other alternative, rather than damping the top link which is subjected to large forces and therefore requires high damping forces, is to damp the spool valve movement.
Fig 3.49 Comparison of field measurements (points) and computer predictions (lines) for standard undamped and experimental damped controls
A smaller, less expensive damper can then be used which may be difficult
to build into existing tractors without redesigning the hydraulic
block. If further work on damping the top link or spool movement were
envisaged it would therefore be necessary to work in conjunction
with a tractor manufacturer.

3.8.7 Conclusions

1. In practice, damping the top link movement did not result in as
much improvement in control performance as predicted, probably
because of backlash in the linkage which although small had a
significant effect on the small top link movements involved.
The damper stroke calculated in the original design was probably
also insufficient. It tended however to show most improvement
at higher speeds above 2.3 m/s where improvements to existing
controls are most important.

2. The optimum position of the damper would be integral with the top
link sensing unit so that the volume of oil displaced for a given
top link movement could be maximised.

3. Alternatively, a more economical design may be to damp the spool
valve movement by including a damper in the hydraulic block.

4. Further work involving either 2) or 3) would entail considerable
modifications to the tractor and would best be carried out in
conjunction with a manufacturer.
4. TORQUE SENSING CONTROL

4.1 INTRODUCTION

This method of control has recently been made available by one manufacturer as a supplement to the existing top link sensing controls on their range of tractors. Essentially, a signal proportional to driveline torque is used to control the linkage position and hence implement depth.

As for the previous control, a theoretical analysis is developed to predict the control response under dynamic conditions. Laboratory measurements of response rates measured on a track treadmill to simulate dynamic conditions are described. Finally, the response of the driveline torque sensing control is compared with top link sensing controls in field conditions and the relative advantages discussed.

4.1.1 Description of torque sensing control

Driveline torque is sensed at the input to the differential by a special coupler. The coupler is displaced axially when torque is applied due to the movement of ball bearings in ramps. The axial displacement operates a hydraulic valve which in turn controls lift arm movement. If the draught force increases, driveline torque increases and the coupler moves apart. This causes a lift signal; the draught force decreases and torque is restored to its original value.

Torque sensing differs from linkage force sensing in that it requires an additional feedback signal to prevent over-correction. This is achieved by a cam on the lift arms which reduces the error signal from the torque sensor when the arms rotate. McKeon (16) refers to this position feedback as a 'cut-off' signal. It is necessary because of the time delay between lift arm movement and a change in torque. He argues that it was incorporated to provide a similar 'cut-off' signal to that inherent in linkage force sensing controls. This occurs for instance in top link sensing controls; when lifting starts, the compressive force in the top link decreases because the vertical force on the implement has increased and this reduces or 'cuts-off' the compressive force causing the lift signal originally.

4.2 THEORETICAL ANALYSIS
4.2.1 Control circuit

A simplified block diagram of torque sensing control is shown in fig 4.1. The engine governor characteristics are not included and it is assumed that the engine torque/speed relationship is described by the function shown.

The set torque, $T_{set}$, is adjusted by a lever in the operator's cab. The difference between the set torque and the actual driveline torque causes a displacement of the sensing coupler which moves the hydraulic spool valve. The response of the spool valve in lifting or lowering the tractor linkage is shown in the next block and then there is a delay, $t_d$, before the lift arms actually move due to a small oil flow as the valve moves and pressure build-up caused by oil compressibility when the linkage moves. When implement depth alters and the draught force, $H$, changes, feedback from the lift arm movement causes a movement of the spool valve back to its original position, thus cutting off the control signal. The control incorporates a device for varying the sensitivity, which effectively alters the linkage ratio transmitting both the torque sensor and the feedback 'cut-off' signals to the spool valve. There are three sensitivity settings provided and their effect on the control circuit shown in fig 4.1 is to alter the values of $k_y$ and $k_g$ simultaneously. The equation governing engine and driveline response can be found by examining the torque balance at the engine flywheel.

$$ I_e \ddot{\theta}_e = T_e - (H+RR) \cdot \frac{g}{g_1} \tag{4.1} $$

where engine torque $T_e$ is the function of engine speed shown. Driveline torque is the engine torque multiplied by the gear ratio, $g_1$.

4.2.2 Simulation

The response of the torque sensing control to a step change in draught force of 2250 N about a mean draught of 13400 N is shown in fig 4.2. This was calculated using the C.S.M.P. programme shown in Appendix 4.1. The advantage of the proportional control is shown clearly for the torque sensing control because it incorporates a position feedback term. The on-off control on the other hand, oscillates as each control movement overshoots, engine torque gradually changes and the opposite control is signalled. This performance is in contrast to the pure draught sensing control which showed little difference between proportional and on-off response.
Fig 4.1 - Control circuit for torque sensing control
Fig 4.2 Response of torque sensing control shown in Fig 3.37 to a 2250 N step change in draught.
4.2.3 Stability

The stability of this control is governed by the value of \( k \) which is the rate of change of draught with depth \( \frac{dH}{dy_o} \) for a particular implement and field condition. If the minimum control movement is sufficient to change the draught force by a greater amount than the deadband, then the control will oscillate between lifting and lowering. The control is unstable if:

\[
\frac{v_{\text{min}}}{dy_o} \frac{dH}{dy_o} > DB
\]

assuming that the average depth of a mounted plough is directly proportional to movement at the lower link ends. The critical values of \( \frac{dH}{dy_o} \) for instability are

- Setting A: 136 N/mm
- Setting B: 210 N/mm
- Setting C: 294 N/mm

Since the rate of change of draught with depth usually increases with depth, stability also depends on ploughing depth. Using these figures, a three furrow plough on heavy land at a typical ploughing depth (200 - 230 mm) would be unstable at setting A but stable at setting C. On light land it would be stable at all three settings.

Because the rate of change of draught with depth is directly proportional to the size of the implement, large implements tend to be less stable. This however is not a problem, as long as the necessary sensitivity settings are provided because instability can always be avoided by reducing the minimum control movement. The draught variation will not be increased by this because larger implements require a relatively smaller movement to change the draught force by a given amount.

4.3 Laboratory Measurements of Response Characteristics

4.3.1 Equipment and Procedure

The tractor was mounted on the N.I.A.E. treadmill (fig 4.3) which has been described previously (66). The tractor rear wheels drove a rolling road which in turn drove a variable delivery pump. The hydraulic oil from the pump passed through a check valve, a remote controlled relief valve and a cooler before returning to the pump to complete the loading circuit. By varying the pump
Restraining frame fixed to ground

Rotary potentiometer on cross shaft

Load cell

Weight on lower link ends

Rolling road

Fig 4.3 Tractor Equipped for Measurements on Treadmill
delivery and pressure, the load could be varied from 5000 N to 36000 N.

A link attached to the drawbar restrained the tractor and a load cell in the link measured draught force. A 100kg weight was attached to the tractor lower links to ensure that they would lower when the control valve opened. Their movement was measured by a rotary potentiometer on the end of the cross shaft. The draught force and lift arm movement were recorded on a chart recorder. The recording of lift arm movement was later calibrated to measure implement depth.

Third and fourth gears were used since these would normally be used for cultivation but the results were the same for both gears. Before measurements were taken the tractor was warmed up to ensure that the oil had reached a typical working temperature.

To measure the control deadband i.e. the minimum change in draught force necessary to cause a control movement, the draught was increased until the lift arm rose and then decreased gradually until they started lowering. The difference in draught was then the deadband.

Since the control included some position feedback there was a relationship between draught and lift arm position for a given control set point. To measure this relationship the draught was increased gradually over the range 5000 - 3600 N. This caused successive lifting movements so that lower link position could be plotted against change in draught.

Finally, measurements were made of the control response to sudden draught changes. The draught was increased as quickly as permitted by the treadmill in steps of 4000, 8000, 12000 and 16000 N. Delay time, i.e. the time between the draught force exceeding the deadband and the start of lift arm movement, was measured. The rate of lifting was also measured but the rates of lowering are not quoted since these are affected by the load on the lift arms which for this experiment was less than for a typical implement. The control was provided with three sensitivity settings, which effectively varied the linkage ratios transmitting both the torque sensor and cut-off signals. Thus the most sensitive setting A produces the largest lift increment before the signal is cut-off and setting C the smallest. Measurements were made with the control at each setting.
4.3.2 Results

Two typical recordings are shown in fig 4.4. The top one shows the draught force increasing until the lift arms start to rise and then decreasing until they lower. The difference between the draught measurements at these two points was the deadband. The lower recording shows a rapid increase in the draught force and the measurement of the time delay between the draught force exceeding its deadband and the start of lift arm movement. Table 4.1 is a summary of these results at each control setting.

Table 4.1 Measured parameters for torque sensing control

<table>
<thead>
<tr>
<th>Setting</th>
<th>Deadband (DB)</th>
<th>Delay time</th>
<th>Minimum movement (measured at lower link end)</th>
<th>Sensitivity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N</td>
<td>s</td>
<td>y_{min} mm</td>
<td>N/mm</td>
</tr>
<tr>
<td>A</td>
<td>1550</td>
<td>0.08</td>
<td>11.4</td>
<td>82</td>
</tr>
<tr>
<td>B</td>
<td>1900</td>
<td>0.09</td>
<td>9.4</td>
<td>133</td>
</tr>
<tr>
<td>C</td>
<td>2380</td>
<td>0.12</td>
<td>8.1</td>
<td>233</td>
</tr>
</tbody>
</table>

The rate of lifting at maximum engine speed (2400 rev/min) was 185 mm/s measured at the lower link ends. The column marked 'minimum movement' refers to the minimum movement of the lower links before the position feedback operated and cut off the lift signal. Fig 4.5 shows the stepped movement of the lower links as the draught force was increased.

The delay time has two components, a delay between a change in draught force and a change in torque due to the inertias of the moving parts, and a delay between displacement of the coupler and lift arm movement. The latter delay is due to a combination of oil flow as the spool valve moves and oil compressibility. The delay measured on the treadmill includes both components but its accuracy may have been reduced for two reasons. Firstly, the response of the treadmill did not provide a step increase in draught as shown by the lower recording in fig 4.4, the rate of change of draught was about 72000 N/s.
Fig 4.4 Typical recordings of draught and lift arm movement
Fig 4.5 Movement of Lower Links With Increase in Draught Force
This is likely to cause the delay to be underestimated because the torque starts changing before the draught has exceeded its deadband. Secondly, since the tractor was not moving forward, any effect due to tractor inertia was not included. This would tend to increase the time delay in the field.

4.3.3 Discussion

The control parameters of the torque sensing control were similar to those measured for linkage force sensing controls. The rates of lifting and lowering were similar and the deadband of 1550 – 2380 N was comparable to deadbands of between 900 and 2000 N measured for conventional controls. The delay time of 0.08 – 0.12 s was in the range measured for conventional controls of between 0.03 and 0.12 s (Table 3.1).

The only published work on torque sensing control is by McKeon (16) who describes a static measurement of the control parameters. The tractor rear wheels without tyre were restrained and load applied by a hydraulic cylinder to the tractor drawbar. When the engine was prevented from rotating a known torque was applied to the driveline. The deadband measured by this method was 6700 N which was due to

(1) system hysteresis,
(2) coupling preload,
(3) test fixture losses.

The gradient of change in draught against lower link movement was between 42 and 350 N/mm for different prototype sensitivities. McKeon does however, point out that the results were of limited value because they were based on static conditions. This conclusion is borne out by the fact that the static deadband was three times larger than the dynamic deadband measured in this work.

4.4 FIELD PERFORMANCE OF TORQUE SENSING CONTROL

4.4.1 Equipment and Procedure

The fields used for measurements are described in Table 4.2. They have been divided into groups of similar specific soil resistance and defined as light, medium or heavy land, although in more general terms they would probably all be described as medium. A 63 pto kW tractor was used for all measurements with either a two or three furrow mouldboard plough. The plough size was chosen to allow the maximum range of forward speeds and for each speed,
<table>
<thead>
<tr>
<th>Field number</th>
<th>Soil type and surface condition</th>
<th>Plough size used</th>
<th>Approximate specific resistance N/mm²</th>
<th>Group</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1</td>
<td>Clay loam, grassland, moist</td>
<td>3F</td>
<td>89600</td>
<td>Heavy</td>
</tr>
<tr>
<td>4.2</td>
<td>Sandy, rotary cultivated after cabbage</td>
<td>3F</td>
<td>56500</td>
<td>Light</td>
</tr>
<tr>
<td>4.3</td>
<td>Sandy, barley stubble</td>
<td>3F</td>
<td>59200</td>
<td>Light</td>
</tr>
<tr>
<td>4.4</td>
<td>Clay loam, Barley stubble</td>
<td>2F</td>
<td>95100</td>
<td>Heavy</td>
</tr>
<tr>
<td>4.5</td>
<td>Clay loam with gravel, barley stubble</td>
<td>2F</td>
<td>75800</td>
<td>Medium</td>
</tr>
<tr>
<td>4.6</td>
<td>Clay, loam, barley stubble</td>
<td>3F</td>
<td>66100</td>
<td>Medium</td>
</tr>
<tr>
<td>4.7</td>
<td>Sandy loam, barley stubble</td>
<td>3F</td>
<td>58600</td>
<td>Light</td>
</tr>
<tr>
<td>4.8</td>
<td>Clay loam, after sugar beet</td>
<td>3F</td>
<td>81300</td>
<td>Heavy</td>
</tr>
<tr>
<td>4.9</td>
<td>Light loam over chalk, overgrown stubble</td>
<td>3F</td>
<td>58100</td>
<td>Light</td>
</tr>
</tbody>
</table>

Table 4.2 Fields used for measurement of torque sensing control
draught force was recorded in each of the following controls: 1) torque sensing at each sensitivity, 2) top link sensing, 3) position sensing. Each run for a particular control and forward speed was repeated at least once and usually twice and the results averaged. The draught force recordings were later analysed by sampling every 100 ms and calculating the mean and standard deviation. The standard deviation in position control, i.e. with the implement fixed in position relative to the tractor, was a measure of the unevenness of the field surface and the effectiveness of a particular control was measured by comparing its standard deviation to that in position control. Thus the results quoted refer to 'Standard deviation ratio', which is defined as

\[
\text{Standard deviation in particular control} / \text{Standard deviation in position control}
\]

4.4.2 Results and Discussion

Fig 4.6 is a graph of standard deviation ratio against forward speed for three soil types. When the results are averaged for all speeds, torque sensing gave a 7% higher standard deviation than top link sensing in light land, 9% higher in medium land and 16% higher in heavy land. For all soil types with the exception of light land at 2.8 m/s, the lowest sensitivity setting, that is the one which produced the smallest lift increment, gave the best results. This suggests that especially in heavy land, an even smaller lift increment may change the draught sufficiently and improve performance.

Torque sensing is more sensitive to soil type than top link sensing and performance deteriorates more in heavier soils. This can be seen in fig 4.7; the experimental points for heavy soil are fairly close to the others for top link sensing but considerably worse for torque sensing at all three settings. This again indicates that a lower sensitivity setting may be advantageous.

Comparing the relationships of torque sensing and top link sensing with speed, it can be seen that both controls deteriorate at a similar rate. In light land however, the results are not typical of results obtained previously (fig 3.40) which showed an increasing standard deviation ratio with speed. These results, except for sensitivity setting C show a tendency toward flattening off above about 2.25 m/s. This may be because light land tends to have a fairly deformable surface and measurements have shown that, on deformable surfaces, tractor ride vibration levels tend not to increase above about 2.25 m/s.
Fig 4.6 Increase in measured standard deviation ratios for different controls in 3 field conditions.
The reason probably is that the soil deformation damps the tractor motion as it travels faster. Implement depth variation is caused mainly by tractor pitch and bounce, so if these tend not to increase above a certain speed, then draught variation should not increase.

A disadvantage of torque sensing which was shown during fieldwork, although it occurred very infrequently, was its response when the available traction changed suddenly which may occur, for example, near the headland. When the tractor wheels reached a patch of soft slippery ground, the driveline torque decreased and caused the implement to lower, which increased the draught force and eventually the tractor stalled. Linkage force sensing controls do not have this disadvantage because the tractor wheels sink due to the increased wheelslip and a compressive force in the top link and force in the lower links will result and cause the implement to be lifted. It is not however likely to be a problem on most field surfaces used for cultivations but it may be important where traction is very poor and variable.

4.4.3 Comparison of predicted and measured results

A comparison of predicted and measured results for each control and field condition at various speeds is shown in fig 4.7. Reasonable agreement was found for all conditions with the possible exception of light land. The computer simulation predicted a continually increasing standard deviation ratio with speed whereas the field results except for setting C showed a tendency toward flattening off above 2.25 m/s. This tendency could be due to the surface deformability, discussed in the previous section in which case the high ratio for setting C appears to be out of place. A possible reason for the high value is that the sensitivity, that is, the minimum control movement, was too high for light land and its effect was greater at high speed. The predicted results were slightly better at setting B than setting C in light land for this reason.

When the field results are averaged for all field conditions and speeds, torque sensing results in a 10.7% higher standard deviation than top link sensing. The predicted value was slightly under-estimated at 9% but considering the experimental error that is inevitable with field
Fig. 4.7 Comparison of measured (points) and predicted (dotted lines) standard deviation ratios for each control in 3 field conditions.

Key to Experimental Points:
- • light load
- • medium load
- • heavy load
measurements, this indicates that the computer programme provides a realistic
simulation of field conditions.

4.4.4 Conclusions

1. The field performance of torque sensing control was inferior to
that of top link sensing control. On average, the standard
deviation of draught for torque sensing control was 10.7%
higher than for top link sensing control.

2. A lower sensitivity setting for the torque sensing control may be
an improvement, particularly in heavy field conditions.

3. Computer predicted results agreed well with measured results over
the range of conditions and speeds used.
5. DEPTH CONTROL

5.1 INTRODUCTION

The traditional method of providing depth control of an implement is by attaching depth wheels rigidly to the implement (fig 2.18). No control system is required; variation in vertical soil force on the implement alter the soil reaction force at the depth wheels and changes in soil resistance alter the tractive effort required from the tractor. Assuming that the wheels follow the surface contour, the only variations in depth are due to deflection of the tyre fitted to the wheel unless a rigid wheel or skid is used, sinkage of the wheel or movements of the wheel over irregularities such as stones or clods. The main disadvantage of this method is that weight is transferred to the tractor.

5.1.1 Pivoted depth wheel

One method devised to provide depth control while retaining weight transfer is a pivoted depth wheel (fig 5.1) which operates the existing tractor hydraulic lift to lift or lower the implement in response to depth error signals. From fig 5.1 it can be seen that if the implement is too deep the wheel moves upwards and the pivoted linkage exerts a compressive force on the top link sensing unit which signals a lift movement. If the wheel moves downwards, a tensile force is produced to signal a lowering movement.

There are two important constraints in the design of this pivoted depth wheel, namely that

a) the control system must be stable when operating with any of the range of top link sensing controls,

and b) the forces exerted by the wheel and linkage on the sensing unit must be large enough to provide both lifting and lowering control signals.

Also, the draught variation allowed by the control should be no greater than ± 10% of the mean depth, which was shown by surveys (1) to be acceptable to operators.

5.2 THEORETICAL ANALYSIS

5.2.1 Equations of motion

To analyse whether the above ideals can be achieved without instability, the assumption is made that the control can be treated as a planar, three
Fig 5.1  Pivotted depth wheel fitted to a cultivator
parallel linkage mechanism. Movement of the links does not therefore alter the angle, \( \theta_w \).

From fig 5.1, the implement depth, \( y_o \) is given by

\[
y_o = d_5 - r_w - (y_c + y_o') + y_d + \pi_3
\]

Dealing with each of these terms in turn, \( d_5 \), \( r_w \) and \( y_c \) are constants but \( y_c \) is a variable describing displacement of the wheel relative to the cultivator frame. It is proportional to displacement, \( y_t \), of the top link sensing spring, where

\[
y_t = \frac{F_t}{K_t}
\]

A displacement, \( y_t \), causes a change in the angle \( \theta_w \). where \( \theta_w \) is considered to be a mean angle, \( \theta_w' \), and a small variation in angle, \( \theta_d' \),

\[
\theta_w = \theta_w' + \theta_d'
\]

The variation \( \theta_d' \) is small compared to \( \theta_w \) because the expected displacement of \( y_t \) is approximately equal to the deadband of the spool valve. The deadband, DB, is usually expressed in terms of the minimum change in force before the control operates, a typical value being 1800 N. Taking a typical spring stiffness, \( k_t = 2.5 \times 10^6 \text{ N/m} \), then

\[
y_t = \frac{1800}{2.5 \times 10^6} = 0.72 \text{ mm}
\]

and the angle, \( \theta_d' \) is approximately

\[
\theta_d' = \frac{y_t}{d_1} = \frac{0.72}{200} \text{ rad} = 0.0036 \text{ rad} = 0.2^\circ
\]

where \( d_1 = 200 \text{ mm} \). For small angles of this order

\[
\theta_d' = \frac{F_t}{K_t d_1}
\]

Displacement of the centre of the wheel is approximately, \( \theta_d' d_2 \) and vertical displacement, \( y_c \), given by

\[
y_c = \left[ d_2 \cos \theta_w' - d_2 \cos (\theta_w' + \theta_d') \right]
\]

\[
y_c = - \theta_d' d_2 \sin \theta_w'
\]

assuming \( \cos \theta_d' = 1 \), and \( \sin \theta_d' = \theta_d' \) for small angles. Hence,
\[ y_c = - \frac{F_r \frac{d_2}{K_w} \sin \theta_w}{d_1} \]  
\[ (5.6) \]

The tyre deflection, \( y_d \), is proportional to the vertical load on the tyre,
\[ y_d = \frac{P_r}{k_w} \]  
\[ (5.7) \]

where \( k_w \) is the tyre stiffness, which is assumed to be constant over the force range necessary to provide a control signal to operate the hydraulics lift. This assumption is justified in fig 5.2 which shows the load/deflection characteristic measured for a 6.00 x 9 tyre on a concrete surface. The operating load range will be that necessary to exceed the control deadband, \( DB \). Taking typical figures of \( DB = 1800 \) N and \( \frac{d_1}{d_2} \sin \theta_w = 0.2 \)

the load variation is \((1800 \times 0.2) = 360 \) N about a mean approximately equal to the load due to the wheel weight,
\[ \frac{W \frac{d_1}{d_2} \sin \theta_w}{d_2 \sin \theta_w} \]

Again taking typical figures of \( W_w = 500 \) N and \( \frac{d_1}{d_2} = 0.9 \), the mean load is \( 450 \) N so the expected range of \( F_r \) is
\[ (450 \pm 360 \ N) \text{, i.e. 270 to 630 N}. \] This is within the linear region for \( k_w \) in fig 5.2.

If the tyre stiffness is linear over this range then the ground pressure, \( p_g \), under the tyre can be assumed constant for load changes of this order near to the zero load condition (116). The relationship between ground pressure and tyre sinkage is given by Bekker (116)
\[ z_s = \left[ \frac{p_g}{k_c + k_f} \right]^{1/n_s} = \left[ \frac{p_g}{k} \right]^{1/n_s} \]  
\[ (5.8) \]

and so for a tyre of width, \( 1 \), in a particular soil condition defined by \( k_c \) and \( k_f \), sinkage is constant for a constant ground pressure.

Gathering the constant terms in equation (5.1), it can be re-written,
\[ y_o = (d_5 - r_w - y'_o + z_s) - y_c + y_d \]  
\[ (5.9) \]
\[ y_o = \lambda - y_c + y_d \]  
\[ (5.10) \]

Taking moments about the pivot point gives and equation for \( F_t \) in terms of the variables \( F_r, R_w, y_c, y_d \) and \( \theta_w \) which simplifies to give a relationship.
Fig 5.2 Load/deflection characteristic for a 6.00 x 9 tyre
between $F_T$ and $y_o$.

$$F_T \cdot d_1 = F_r \cdot d_2 \sin \theta_w + R_w \left( d_2 \cos \theta_w + r_w - y_d \right) - W \cdot d_4 \sin \theta_w \quad (5.11)$$

But from equation (5.7),

$$F_r = k_w \cdot y_d \quad (5.12)$$

The rolling resistance, $R_w$, is proportional to load on the tyre, over a small range (116) of the order calculated previously,

$$R_w = c \cdot F_r \quad (5.13)$$

where, $c$ is the coefficient of rolling resistance which varies with the particular tyre and soil condition, typical values for agricultural conditions, being between 0.02 and 0.12 (114) Substituting equation (5.12),

$$R_w = c \cdot k_w \cdot y_d \quad (5.14)$$

Substituting (5.12) and (5.14) into (5.11) gives

$$F_T \cdot d_1 = k_w \cdot y_d \left( d_2 \sin \theta_w + c \cdot d_2 \cos \theta_w + c \cdot r_w - c \cdot y_d \right) - W \cdot d_4 \sin \theta_w \quad (5.15)$$

Neglecting the second order term in the small deflection, $y_d$, and assuming that $\cos \theta'_w = \cos \theta_w$ and $\sin \theta'_w = \sin \theta_w$ since the typical value of $\theta'_w$ of 0.0036 rad is small compared to $\theta'_w$, which is typically 0.65 rad, and substituting for $y_d$ from equation (5.10),

$$F_T \cdot d_1 = k_w \left( y_o + y_c - \lambda_1 \right) \left( d_2 \sin \theta'_w + c \cdot d_2 \cos \theta'_w + c \cdot r_w \right) - W \cdot d_4 \sin \theta'_w \quad (5.16)$$

Again, substituting for $y_c$ from equation (5.6),

$$F_T \cdot d_1 = k_w \left( y_o - F_T \cdot d_2 \sin \theta'_w - \lambda_1 \right) \left( d_2 \sin \theta'_w + c \cdot d_2 \cos \theta'_w + c \cdot r_w \right) - W \cdot d_4 \sin \theta'_w \quad (5.17)$$

Letting

$$\lambda_2 = d_2 \sin \theta'_w + c \cdot d_2 \cos \theta'_w + c \cdot r_w \quad (5.18)$$

and collecting the $F_T$ terms,

$$F_T \left( d_1 + k_w \lambda_2 \cdot d_2 \sin \theta'_w \right) = k_w \lambda_2 \left( y_o - \lambda_1 \right) - W \cdot d_4 \sin \theta'_w \quad (5.19)$$

Letting

$$\lambda_3 = d_1 + k_w \lambda_2 \cdot d_2 \sin \theta'_w \quad (5.20)$$

then

$$F_T = \left[ \frac{k_w \lambda_2}{\lambda_3} \right] y_o - \left[ \frac{k_w \lambda_2 \lambda_1}{\lambda_3} + \frac{W \cdot d_4 \sin \theta'_w}{\lambda_3} \right] \quad (5.21)$$
\[ F_T = D_1 y_0 - D_2 \]  \hspace{1cm} (5.22)

where \( D_1 \) and \( D_2 \) are the constants shown in square brackets in equation (5.21).

Equations (5.7 - 22) are only true if the wheel remains in contact with the ground. If the wheel leaves the ground, then \( F_x = R_w = 0 \) and taking moments about the pivot point,

\[ F_T = - W_4 \frac{A}{d_1} \sin \theta_w' \]  \hspace{1cm} (5.23)

5.2.2 Control circuit

Using equation (5.22) and the circuit for the tractor draught control shown in fig 3.2, the depth control circuit can be drawn and is shown in fig 5.3. Since the wheel and linkage are designed to operate with existing tractor hydraulics, it must be stable with all of them. An on-off control with a pure time delay, \( t_d' \), is inherently the least stable of the available controls, but has been shown to be stable if (107)

\[ DB > \frac{dy_0}{dt} t_d k_1 \]  \hspace{1cm} (5.24)

where \( k_1 \) is the ratio of sensed force to change of implement depth. In this case \( k_1 = D_1 \), and taking typical values of \( DB = 1800 \text{ N} \), \( \frac{dy_0}{dt} = 200 \text{ mm/s} \) and \( t_d = 0.05 \text{ s} \), then for stability,

\[ D_1 < 180 \text{ N/mm} \]  \hspace{1cm} (5.25)

For the following design parameter values,

\[ D_1 = 154 \text{ N/mm} \]

and the control is therefore stable.

\[ d_1 = 0.2 \text{ m} \]
\[ d_2 = 1 \text{ m} \]
\[ d_3 = 1.24 \text{ m} \]
\[ d_4 = 0.9 \text{ m} \]
\[ d_5 = 0.68 \text{ m} \]
\[ \theta_w' = 0.65 \text{ rad} \]
\[ y_c' = 0.22 \text{ m} \text{ (for set depth of 0.24 m)} \]
\[ r_w = 0.22 \text{ m} \]
\[ c = 0.1 \]
Fig. 5.3 Control circuit for depth control with pivoted depth wheel.

Set depth error

Output

Gain

Integral

Proportional

Error

Input disturbance
Another design requirement is that a lowering signal must be given if the
wheel leaves the ground. Under these conditions, $F_T$ given by equation
(5.23) must be greater than the (deadband/2) assuming that when the wheel
rests on the ground, the value of $F_T$ corresponds to the middle of the deadband,

$$\left| \frac{-W_d \sin \theta}{d} \right| > \frac{DB}{2}$$

(5.26)

In this case

$$1370 \text{ N} > 900 \text{ N}$$

and so the requirement is met.

The nominal deadband in terms of the depth variation, $DV$, allowed before
a control movement is signalled is given by the change in $y_c$ and $y_d$ when $F_T$
changes from $F_{T \text{set}}$ to $(F_{T \text{set}} + DB)$ where $F_{T \text{set}}$ is the set top link force which
in practice would ideally be zero to maximise the weight transfer to the
tractor. Substituting these two forces into equation (5.21) gives a depth
deadband, $DV$, of 10.1 mm.

5.3 FIELD PERFORMANCE OF A PIVOTED DEPTH WHEEL OPERATING THE TRACTOR

5.3.1 Equipment and Procedure

Five fields were used for measurements to compare the performance of the
depth wheel with draught, torque sensing and position control. All measurements
were made with a 63 pto kW tractor and 5 or 7 rigid tine cultivator. The fields
and their surface conditions are described in Table 5.1. Depth was measured
with a pivoted wheel attached to the implement frame. A rotary potentiometer
mounted at the pivot of the wheel produced an electrical signal proportional
to depth which was recorded on an FM tape recorder. The wheel was positioned
near the implement centreline and in front of the tines so that it ran on
undisturbed ground.
<table>
<thead>
<tr>
<th>Field Number</th>
<th>Location</th>
<th>Surface Condition</th>
<th>Number of Times</th>
<th>Soil Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>Meppershall</td>
<td>Rotary cultivated after brussel sprouts</td>
<td>7</td>
<td>light loam</td>
</tr>
<tr>
<td>5.2</td>
<td>Langford</td>
<td>Burnt barley stubble</td>
<td>5</td>
<td>loam with gravel</td>
</tr>
<tr>
<td>5.3</td>
<td>Wrest Park</td>
<td>Barley stubble</td>
<td>5</td>
<td>clay loam</td>
</tr>
<tr>
<td>5.4</td>
<td>Wrest Park</td>
<td>Bean stubble</td>
<td>5</td>
<td>clay loam</td>
</tr>
<tr>
<td>5.5</td>
<td>Wrest Park</td>
<td>Overgrown bean stubble</td>
<td>5</td>
<td>clay loam</td>
</tr>
</tbody>
</table>

Table 5.1 Fields used for measurements with pivoted depth wheel

<table>
<thead>
<tr>
<th>Field Number</th>
<th>Forward speed m/s</th>
<th>Depth Control</th>
<th>Standard deviation of depth mm</th>
<th>Torque Sensing Control</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>1.35</td>
<td>28</td>
<td>30</td>
<td>35</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>30</td>
<td>29</td>
<td>36</td>
</tr>
<tr>
<td>5.2</td>
<td>1.35</td>
<td>29</td>
<td>26</td>
<td>32</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>34</td>
<td>39</td>
<td>45</td>
</tr>
<tr>
<td>5.3</td>
<td>1.35</td>
<td>29</td>
<td>30</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>36</td>
<td>41</td>
<td>38</td>
</tr>
<tr>
<td>5.4</td>
<td>1.35</td>
<td>24</td>
<td>23</td>
<td>23</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>30</td>
<td>28</td>
<td>31</td>
</tr>
<tr>
<td>5.5</td>
<td>1.35</td>
<td>26</td>
<td>25</td>
<td>27</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>33</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>Average</td>
<td>1.35</td>
<td>27</td>
<td>27</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>1.80</td>
<td>32</td>
<td>33</td>
<td>36</td>
</tr>
</tbody>
</table>

Table 5.2 Standard deviation of depth for various controls measured in five fields.
In each field, one run with a particular control operating lasted for approximately one minute. Each control was recorded at two speeds and two or three repeats of each run were made depending on the land available. It would have been preferable to have included some runs at higher speed but the dry conditions made the specific soil resistance so high that the tractor did not have enough power to work faster than 1.8m/s.

5.3.2 Results

The records were analysed using a time domain analyser to calculate the mean and standard deviation. The results for standard deviation of depth in each field at two forward speeds are shown in Table 5.2. The averages for each control in all fields are plotted in fig 5.4. The relative performance of each control tended to vary depending on the field condition. The average trends however are shown quite clearly in fig 5.4. Draught and depth control provided very similar control of depth and were both better than torque sensing control.

The most important result was that implement depth control with the depth wheel and linkage was no better than with draught control. This was probably because the specific soil resistance was more constant than had been expected. However there would have been no way of knowing how much the soil resistance did vary except by measuring the variations in force necessary to pull a trailed implement. It is possible that the extremely dry conditions before the field measurements were made had hardened all the soil and tended to reduce variations in soil resistance.

It was not possible to make objective measurements of control performance over cultivated land because of difficulties in measuring depth. The depth wheel was however used over cultivated ground with many large, hard clods on the surface and appeared to operate satisfactorily, though no better than draught control. However when some runs were made from uncultivated land across a 12 m strip of cultivated land the tractor stalled in draught control but kept going with the depth wheel operating. In draught control the change in soil forces made the control signal lower the implement too much and caused tractor stall. This would indicate that the depth wheel has an advantage in fields where soil resistance varies considerably.
Fig 5.4 Depth variation for various controls at two speeds - averaged values for all fields.
The values obtained for standard deviation of depth varied between 24 and 40 mm for depth and draught control over the limited range of field conditions. This compares with values of between 18 and 56 mm found previously for cultivations over a wider range of conditions, with an average value of 36 mm.

5.3.3. Discussion and conclusions

The field results showed that the pivoted depth wheel on a cultivator provided similar control of depth to that obtained with draught control. Its principle of operation has therefore been proven in that it produces signals which are proportional to implement depth and which operate the tractor hydraulics to provide a control movement response.

In field conditions where the specific soil resistance changed suddenly because part of it had been cultivated, the pivoted depth wheel had advantages over draught control in preventing tractor stall. This should be beneficial in fields where soil resistance changes due, for example, to a change from clay to loam and the farmer wants to cultivate it to the same depth. The advantage of the pivoted depth wheel over fixed depth wheels in this case is that weight transfer is not lost and maximum tractive effort can be obtained. The fact that the pivoted depth wheel did not provide consistently less depth variation than draught control except in field 28 may have been due to the unusual field conditions. The lack of variation in soil resistance may have been the result of a dry spell of weather which made all the soil very hard. It would obviously be useful to make further measurements over a wider range of conditions.

The pivoted depth wheel however has a drawback in that it requires an additional linkage connected to the tractor. This means that the tractor must have an additional fixed mounting point, usually a bracket bolted underneath the sensing unit. An extra bracket must also be attached to the implement to accommodate either the pivot point for the wheel of the fixed point to which the lower top link is connected. Once all linkages are connected however, the implement depth can be altered easily by adjusting the upper top link length.

The design of the wheel presents no major difficulties, although it will be integral with the implement and not therefore interchangeable. The main
constraints are in providing enough unrestricted movement of the wheel to allow a range of depths to be set and to allow the implement to be lifted out of work. Because of the extra link the wheel moves when the implement is lifted so it must not contact the frame before the implement is lifted out of work. Although no trouble with the wheel blocking up was experienced it may be a problem where the tines are more closely spaced and adequate clearance must be left between tines and wheel to prevent this.

It may be possible to adapt the idea of a pivoted depth wheel to operate with a plough. In a preliminary design at N.I.A.E. the wheel runs on the uncultivated land alongside the plough and a torque tube transmits the signal to an additional top link in the same way as for the cultivator. It is slightly more complicated because the torque tube needs supporting at both ends, and not therefore as appealing as the simpler design for the cultivator.
6. DISCUSSION

As stated previously, the objects of this work were firstly, to understand the operation of present implement controls and their effect on tractor performance and secondly, to suggest improvements or new controls for future tractors. The discussion of the extent to which these objects have been achieved naturally falls into three sections, the first of which deals with the performance of present controls, the second with their optimisation and the third with the likely requirements of future controls and how those requirements might be met.

6.1 PERFORMANCE OF PRESENT CONTROLS

The performance of present controls has been established by a combination of laboratory and field measurements, as well as theoretical analysis.

6.1.1 Laboratory measurements

The laboratory measurements provided a data base of the control parameters of commercial controls. The results for one lower link and 13 top link sensing controls showed that there were significant differences between the main control parameters, i.e. deadband, rate of lift and delay time. The deadband and rate of lift were well defined parameters of hydraulic controls and could, if necessary, be altered by fairly simple design modifications. The delay time, however, was less understood and further experiments were carried out to investigate it. The experimental control, built to measure control performance in the field, was used for these measurements. The delay was shown to have two components, a delay for the electro-hydraulic valve to move and a delay in building up oil pressure due to linkage compliance and oil compressibility. The result was a compound delay consisting of a pure time delay and a first order lag. It was also concluded from this work and previous laboratory measurements that the delay in commercial controls was due to similar factors. But the effective stiffness measured at the lower link ends was higher for commercial controls and the measured delays could be approximated to
pure time delays.

The measurements of the other commercial control available, the torque sensing control, also provided a data base for later field measurements and theoretical analysis. The treadmill used for this work enabled measurements to be made under dynamic conditions. The results were significantly different from previous static measurements (16) but parameter values were similar to those measured for top or lower link sensing controls.

6.1.2 Field measurements

From the two series of measurements with the experimental implement control, three approximate trends were shown (figs 3.34 - 3.40); draught variation,

1) increased linearly with deadband,
2) decreased linearly as rate of lift increased,
3) increase as a function of the square of forward speed.

These results suggested that performance could be improved simply by increasing rate of lift, or decreasing deadband. But in fact, as the theoretical analysis had shown, the scope for improvement was limited by instability.

Reasonable agreement was found between the predicted and measured parameter values to cause instability. Frequencies of unstable oscillation were in the range expected but measured amplitudes tended to be higher, probably because small amounts of free-play in the linkages which caused the strain gauges to measure high shock loads as play was taken up.

Cultivators were found to be more difficult to control than ploughs, a problem which has previously been noticed by farmers (1). The reason for this is the difference in the steady state vertical force/depth relationships for a plough and a cultivator. Around the working depth of a plough, the steady state vertical force is either fairly constant or decreases slightly as depth increases, whereas in the case of a cultivator it always increases. For the cultivator,
therefore, the component of vertical force in the top link tends to
cancel out the draught force component which should be providing the
controlling signal. A bigger change in depth is necessary to provide
a similar change in top link force and hence the control allows a
wider depth variation.

A semi-mounted plough, with pure draught sensing allowed more
draught variation than the fully mounted plough (Table 3.8). This was
due to the fact that to make significant changes in draught, the front
of the plough had to be moved excessively. Thus, plough inclination
changed significantly and penetration was difficult, particularly when
the inclination was negative.

In the third series of field work, the performance of a torque
sensing control was compared with conventional top link sensing.
Although top link sensing provided slightly better control of a mounted,
two or three furrow plough, torque sensing showed that it may have
other advantages. For example, it could also be used with semi-mounted
or trailed implements and should, therefore, be more applicable to
larger tractors, which tend to use larger semi-mounted or trailed
implements. This is discussed in more detail later, but for the range
of ploughs designed to match present tractors of around 50 kW, torque
sensing showed no advantages over top link sensing.

6.2 IMPROVEMENT OF IMPLEMENT CONTROLS

Having established the performance of present controls, the question
of how they can be improved arises. It will be assumed that there
will continue to be a demand for both depth and draught control, and
there is therefore, a sub-section on each. The debate, however, of
whether the ultimate aim of implement should be to control draught
or depth is likely to continue for some years, and will probably be
influenced by future cultivation trends.

6.2.1 Draught control

Effect of type of sensing

Previous work (65) had shown that there was no significant
difference in performance between top and lower link sensing controls
which had similar control parameters. The theoretical analysis here showed that no significant difference would be expected, the only advantage of lower link sensing being that it could be used with semi-mounted implements.

Pure draught sensing was found to provide similar performance to top link sensing in field conditions. Torque sensing was found to be consistently inferior to top link sensing but in most fields, the difference was small and would probably not be noticed by an operator. The conclusion can be drawn therefore, that type of sensing had little effect on present performance. It does however have a significant effect on how much the control can be improved.

The inherent disadvantage of top or lower link sensing is that the sensed signal has components not only due to the draught force but also due to the vertical force on the implement. It is this factor which limits the scope for improvement by inducing instability. Torque sensing has the inherent disadvantage of an added delay time between a draught change and a change in driveline torque due to the tractor and driveline inertia.

Position feedback is necessary to prevent over-correction but stability can always be maintained by varying the feedback gain. Computer simulations showed that sensing other related parameters e.g. torque elsewhere in the driveline, engine torque, or engine speed, would provide similar performance. Engine speed sensing was investigated for a manufacturer (81) interested in patenting the idea. In the computer model used, engine speed was sensed by measuring governor rack displacement and predicted performance was similar to that for torque sensing.

Wheelslip sensing is another possibility, although a reliable, inexpensive method of measuring wheelslip has yet to be found. It would probably be necessary to combine the wheelslip signal with a draught or depth signal to prevent the control being too sensitive to wheelslip variations due to changes in the field surfaces rather
than draught force. A control which maintained a constant depth or draught but had an over-ride which lifted the implement when the wheelslip exceeded a critical value, may have some applications.

Pure draught sensing does not have any of the inherent limitations of the above-mentioned controls. It has, therefore, the most potential for improvement and could be implemented simply by adding the signals from both a top and lower link sensing unit.

**Effect of control parameters on stability**

For top or lower link sensing, equation (3.55) is an approximate expression for control stability. Parameters, such as rate of lift or deadband can be selected using this equation to predict stability. But the values of these parameters vary with field conditions, because the equation contains the mean draught, \( H \), the forward speed, \( x \), and the ratio \( \frac{1}{2} \) which is proportional to plough length. In order to improve performance over a wide range of field conditions, rate of lift and deadband should be capable of being varied either manually or automatically. For example, stability improves with forward speed and so the deadband could be automatically decreased as a higher gear was engaged or it could be manually adjusted on a scale proportional to \( 1/x \) speed. Rate of lift is more difficult to vary, but if variable displacement hydraulic pumps become more common rather than the constant displacement types fitted at present, it would be fairly simple.

Stability decreases as plough size increases because the ratio \( \frac{1}{2} \) increases proportionally to plough length. Since it is not practical to automatically sense the length of plough, manual adjustment of rate of lift or deadband would appear to be the most flexible arrangement. In field conditions, it would be possible for the operator to adjust these parameters to ensure that control stability was maintained throughout the field. On specialised tractors, always used for ploughing or cultivating say, it may be possible to sense implement size by measuring lift cylinder pressure when the implement is raised.
Torque sensing control has the advantage that stability is independent of speed and can always be maintained even for long implements by reducing the feedback gain. However, this may cause other problems because the lift increment, $y_{\text{min}}$, is reduced. On rough surfaces, where large depth variations occur, particularly for long implements, the lift increment may not allow sufficient movement of the implement relative to the tractor. On smooth surfaces, the required linkage movements are small because the tractor pitches less. To improve torque sensing control, a variable feedback gain should be incorporated rather than several discrete settings.

Equation (3.25) can be used to predict the stability of a pure draught sensing control with a pure time delay. This is dependent on the rate of change of draught with depth, $k_1$, which depends on soil resistance and implement size. Rate of lift or deadband should be variable, therefore, so that the optimum value for a particular field and implement size could be chosen. However the most potential for improvement lies in reducing the delay time. It has been shown that $t_d$ can be reduced to 0.02 s in commercial controls, and it may be possible to reduce it to negligible proportions by careful design.

The most practical arrangement of pure draught sensing control, which would not entail major modification to present designs is to incorporate both a top and lower link sensing unit and add the signals from each. The vertical force is eliminated because it has equal components in each link but they act in opposite directions.

6.2.2 Depth control

The method of achieving depth control of an implement, while retaining weight transfer, was analysed in section 5. The wheel and linkage were designed according to the stability criteria developed; a method found to be successful since stability was maintained even on the hardest surfaces used for field measurements.

The limited range of conditions used for field measurements restricts the conclusions which can be drawn about the performance of
the wheel. Basically, it operated in the manner in which it was
designed and provided depth control similar to that provided by the
tractor top link sensing draught control. On ground where the soil
resistance does not vary significantly then this performance would be
expected, since both controls are limited by the tractor hydraulic
response. The depth wheel should show an improvement in conditions
where the soil resistance varies. Here the draught control will allow
depth variations but the depth wheel will maintain its control of
depth. Its potential in these conditions was indicated by the ad hoc
tests in the field described previously where the soil resistance was
artificially altered by disturbing the soil.

The design of the wheel presents some problems but these would
be easier to overcome if the wheel were designed integrally with the
implement. Traditional cultivation practice favours an arrow formation
of the tines, to break the soil up gradually. The wheel must run on
undisturbed soil, so if it is in the middle of the cultivator it does
not permit the first tine of the 'arrow' to be used. To overcome
this, either the wheel must be as far forward as possible and the
arrow formation behind it, or two wheels could be used positioned
well to either side of the first tine. This latter method may also
have the advantage of maintaining the implement more closely horizontal
to the ground surface and prevent depth variations due to roll of the
implement. Another problem which may be encountered if the wheel is
placed between two tines is that blockages are more likely. No
blocking was encountered during field work but the fields used did
not contain much surface vegetation. The double wheel approach may
also overcome any likelihood of blockages.

The main disadvantage of the pivoted depth wheel is that an extra
connection between tractor and implement is required. This also
necessitates an extra mounting point on the tractor and increases
the complexity of coupling the tractor and implement.
6.3 **FUTURE IMPLEMENT CONTROL REQUIREMENTS**

Future trends in implement control depend on future tractor designs. It is uncertain how tractors will be improved in future, so some assumptions will be made here.

The arguments will be restricted to the fairly short term future, say the next decade because attempts to predict further than this are pure speculation. It will be assumed that cultivation will remain one of the most important functions of a tractor, and one which requires high power levels. Also, the gradual replacement of tractors with more specialised self-propelled machinery will be assumed to continue. An example of this trend is for grain harvesting, which has seen an almost universal switch to combine harvesters from tractor drawn implements. Thus the concept of a specialised cultivating machine which would make the use of more sophisticated controls economic, must be considered as a likely machine of the future.

The objective of improvements to any cultivation machines or tractor/implement combinations must be to improve the economy of soil tillage. This can be achieved either directly, by reduced costs per hectare, or indirectly, by higher work rates and improved timeliness.

There are three probable trends toward achieving this broad objective:

a) increased implement width,

b) increased forward speed,

c) increased use of p.t.o. driven implements.

Increasing implement width involves a simple scaling up of present tractor/implement combinations, that is, tractor engine power is increased and the entire combination is increased in size to maintain present levels of power to weight ratio. Increasing forward speed only involves increasing the tractor engine power, and is, therefore, much cheaper. The increased power is used with existing equipment operating at higher speed. Increased use of p.t.o. driven implements is somewhat outside the scope of this thesis but will be discussed briefly because it may be found necessary to control the depth of
rotary cultivators to prevent excessive torque fluctuations or engine stall. If this is the case then much of the work in this thesis will be relevant to the development of such a control.

Each of these approaches will be discussed in the following sections, particularly in relation to the requirements they impose on the implement control.

6.3.1 Increased implement width

No new technology is required to scale up a tractor and implement to work at present speeds. Draught control performance deteriorates with implement size because a given depth change causes a bigger change in draught but the heavier, more powerful tractor is able to cope with greater fluctuations, so that tractive efficiency does not decrease. But if implement length as well as width is increased, as is necessary for a plough for example, other problems arise. Changes in tractor pitch cause large depth variations, particularly at the rear of the plough, and it becomes necessary to allow the angle between tractor and implement to vary by semi-mounting or trailing the implement. In addition, there is also the problem that larger implements cannot follow short wavelength undulations.

To overcome these problems, more depth controls and flexible members in the implement frame are likely to be used. The idea of controlling top link length to allow the angle between tractor and implement to vary will become more important. The disadvantages that accompany these trends are decreased manoeuvrability and increased turning circle for semi-mounted and trailed implements. And for depth controlled implements, tractive efficiency usually decreases due to changes in draught when the soil resistance varies. In order to quantify the loss of efficiency it is necessary to know the likely variations in soil resistance and, although some limited data exists for the U.S.S.R. (62) there is none available for European conditions. The collection of a range of this data would be a worthwhile project.

If long mounted implements are used, with their advantage of manoeuvrability then the performance of top or lower link sensing
controls is inadequate. Stability is the main problem because the linkage force is dominated by the vertical force component, which increases as the moment arm of the vertical force moves further back. There are, however, other adequate controls such as pure draught or torque sensing. Pure draught control could be arranged in many ways if electro-hydraulic components were used but a more practical method, as mentioned previously, would be to add the signals from both a top and lower link sensing unit and thus eliminate the vertical force components. Torque sensing was also shown to provide adequate control by careful selection of the feedback gain and was better than linkage force sensing for large ploughs. Other controls, such as engine speed, torque or wheelslip sensing probably have similar potential.

6.3.2 Increased forward speed

This is not such a simple solution as increased implement width, because new technology is probably required to operate at higher speeds. There are three obvious problem areas:

a) implement control,
b) tractor ride and operator comfort,
c) implement design.

All these are inter-related and must be improved simultaneously before higher speeds are feasible.

Implement control

Draught fluctuations increase with speed because increased tractor pitch causes increased depth variation of the implement. One of two strategies can be adopted, either draught controls can be improved or excess tractor engine power and possibly extra ballast can be provided to overcome the effect of the fluctuations. The approximate amount of reserve engine power required to overcome draught fluctuations at higher speed was derived in section 3.5, but here the discussion will concentrate on potential improvement to implement controls, since present efficiency levels need not be sacrificed using this approach.
The long term future for top and lower link sensing controls is bleak. The stability problem inherent in these controls due to the mixture of draught and vertical force components has been discussed in detail in section 3.3. But there are some short term improvements which could be made. Equations have been presented to predict stability to assist in the selection of parameter values and these could be applied for a wide range of implements and field conditions by incorporating manual or automatic adjustment of parameters as discussed in section 6.2.1.

Damping the spool valve movement was shown both theoretically and from some limited practical evidence to offer some improvement, particularly at higher speeds. This would ideally be incorporated in the original hydraulic design because of difficulties in later modifications. Since the important frequency range of draught fluctuations to control is below about 6 Hz, the damper can be tuned to only attenuate higher frequencies, which may cause unnecessary control movement, spool valve movement and wear.

Another inherent problem for top or lower link sensing controls at high speeds are the linkage force fluctuations due to the damping effect of a plough on the tractor. Since tractor ride vibration increases with speed, so do these force fluctuations. And they cannot be filtered out because they are in the same frequency range, 1-6 Hz, as the force fluctuations due to draught changes. To overcome this problem either pure draught or some other parameter which does not involve vertical force must be sensed instead of top or lower link force. The problem is greatest for ploughs because they cause higher vertical damping forces than other implements.

Pure draught sensing, by adding the top and lower link signals, would not be affected by these spurious force fluctuations and it would permit the beneficial effect of increased damping of tractor ride to be retained without detrimental effect on the implement control. This appears to be one of the most promising methods of control for
higher speed because it is not difficult to design, does not require major modifications to present tractor designs and by careful design of the hydraulic delay, it can provide good performance. Of course, if specialised cultivation tractors become economically feasible, then electro-hydraulic controls may be worthwhile and draught could be sensed by strain gauges. Another advantage of electro-hydraulic controls is that it is much easier to transmit the operator's control via electric cable into his cab than by mechanical means. As cabs become more remote and insulated from the tractor this may be a very important feature.

Driveline torque sensing or a control which senses other similar parameters, as mentioned previously, does not appear to be the answer to the control problem at high speed. Field performance was consistently slightly inferior to top link sensing and showed a similar deterioration with speed.

The fundamental cause of depth variations is tractor pitch and therefore particularly on rough surfaces, the angle between tractor and implement may need to be controlled or allowed to vary. For this to be successful, it eliminates any weight transfer since any linkage which transferred implement weight to the tractor would also cause implement movement when the tractor pitched. The previous proposal for work on measuring typical variations in specific soil resistance should be extended to include measurements at higher ploughing speeds than used at present, so that the draught variation of high speed depth-controlled implements can be quantified.

**Implement design**

Some implement designs require modification to operate efficiently at higher speed. This is especially true for mouldboard ploughs whose draught increases drastically with speed because a mouldboard can only be designed to operate at maximum efficiency at one particular speed. Above this speed, too much energy is imparted to the soil and it is accelerated unnecessarily and thrown to the side. Designing mouldboards
to impart sufficient energy to just invert the soil slice at higher speeds presents no problem, successful attempts have already been made. But this will inevitably change the draught and vertical force/depth characteristics and also the penetration equation which in turn may have a significant effect on the operation of the implement control.

Cultivators and p.t.o. driven rotary cultivators may require less modification than ploughs to operate at higher speeds. Their power requirement does not increase as rapidly with speed and so their control requirements at higher speeds are likely to be similar to those at present speeds.

**Tractor ride and operator comfort**

Implements can modify the tractor ride behaviour significantly. Fieldwork described in section 2.4 showed that vertical vibration levels measured at the driver's seat when ploughing light land, were up to 30% lower than when the tractor was alone, and 50% lower in heavy land. This has important implications for future implement controls because if the advantage of improved ride is to be retained, the implement must be fully mounted to the tractor. It may also influence future cultivation practice. If operator comfort at high speed is not improved by suspended cabs or tractors, then the improved ride advantage of ploughing over other primary cultivation methods may be an important factor in influencing the choice of method.

There is scope for more research into the interaction of tractor and implement ride dynamics since this work has only investigated the effect of ploughing. The effect of other implements and trailers requires research effort to complement the current work (8) on tractors alone. Longitudinal vibration levels on tractor are as severe as vertical levels in terms of exceeding the tolerances specified for the operator and trailed implements such as heavy discs or cultivators may have a significant effect in the longitudinal direction.
6.3.3 Increased use of p.t.o. driven cultivators

The alternative to using draught implements for soil tillage is to use implements driven by the tractor power take-off (p.t.o.). Rotary cultivators have been used for many years, particularly for preparing the fine soil tilth necessary for a seed-bed. Research into a modified version of this implement, the rotary digger (9), has shown that it can also be used for primary cultivation. It has advantages over draught implements in slippery conditions because the tractor no longer needs to produce high draught forces and overall efficiency is therefore increased.

The depth of rotary diggers is at present controlled simply by depth wheels or skids and this leads to similar problems as those for depth-controlled draught implements. Variations in soil resistance cause variations in the power requirement of the tractor. The tractor engine cannot therefore be operated at peak power because reserve power is required when the torque requirement increases due to soil variation. A possible method of improving work rate by operating the engine nearer its peak power would be to control the torque requirement of the rotary digger by controlling its depth. The quality of work would not be affected significantly by small depth changes.

Work on the torque sensing control indicates a control for rotary cultivators could be achieved by sensing p.t.o. torque, engine torque or engine speed. Position feedback from the linkage position would be required for any of these methods of control.

There is a lack of knowledge of the torque fluctuations of depth-controlled rotary cultivators and measurements of them over a range of field conditions would be useful for two reasons; firstly, for assessing the loss in overall efficiency because the tractor engine cannot be operated at full power and secondly, for comparing the relative efficiencies of p.t.o. and draught cultivations. The work in section 3.5 showed that this comparison is affected not only by mean power requirements but also the amplitude of the fluctuations around the mean.
7. CONCLUSIONS

7.1 GENERAL

7.1.1 A theory has been presented to predict the longitudinal and vertical motion of a tractor/cultivation implement combination in field conditions.

7.1.2 Implement controls have been studied both theoretically and in laboratory and field conditions. Their response and effect on overall tractor motion has been predicted and validated by a wide range of field performance measurements over three cultivation seasons.

7.1.3 Equations for optimising present controls are presented and suggestions for future draught controls are made in the context of possible trends in future tractor design.

7.2 PRESENT IMPLEMENT CONTROLS

7.2.1 Present methods of implement control do not allow the optimum rate of work of the tractor/implement combination, calculated for steady state conditions, to be realised. Dynamic tractive efficiencies may be 4-8% lower on typical field surfaces.

7.2.2 Top or lower link sensing provides better performance than torque sensing for the present size of mounted implements but is less stable for longer implements.

7.2.3 Pure draught control using an experimental electro-hydraulic control gave similar field performance to top link sensing.

7.3 IMPROVEMENT OF PRESENT IMPLEMENT CONTROLS

7.3.1 All the types of implement control analysed were non-linear. The theoretical analyses were used mainly to investigate the effect of control parameters on performance and control stability. For pure draught, top or lower link, driveline torque and depth wheel sensing it was possible to predict instability. In some cases a simplified model of the control was necessary in order to do this.

7.3.2 The following equations summarise the conditions for stability
for the controls examined and can therefore be used when selecting parameter values to avoid instability.

**Pure draught sensing** is stable if

$$DB > k_1 \dot{Y}_o \max t_d$$  \hspace{1cm} (7.1)

for an on-off response and,

$$\left[ DB + 2\dot{Y}_o \max \right] > k_1 \dot{Y}_o \max \frac{t_d}{k_p k_s}$$  \hspace{1cm} (7.2)

for a proportional response. The approximate stability criteria which can be used as a design guide for top or lower link sensing, are that control is stable if

$$DB > k_2 \frac{H \text{mean}}{k_p k_s} \dot{Y}_{o \max}$$  \hspace{1cm} (7.3)

for an on-off response and

$$\left[ DB + 2\dot{Y}_o \max \right] > k_2 \frac{H \text{mean}}{k_p k_s} \dot{Y}_{o \max}$$  \hspace{1cm} (7.4)

for a proportional response. Torque sensing control is stable if

$$\frac{DB}{k_1} > \dot{Y}_{\text{min}}$$  \hspace{1cm} (7.5)

The depth sensing control is stable if

$$DB > D_1 \dot{Y}_o \max t_d$$  \hspace{1cm} (7.6)

for an on-off control and

$$\left[ DB + 2\dot{Y}_o \max \right] > D_1 \dot{Y}_o \max \frac{t_d}{k_p k_s}$$  \hspace{1cm} (7.7)

for a proportional control.
7.4 SUGGESTIONS FOR IMPROVEMENTS AND FUTURE IMPLEMENT CONTROLS

7.4.1 For better depth control, a pivoted depth wheel and linkage operating the tractor hydraulics provided reasonable depth control and maintained weight transfer, over a limited range of field conditions.

7.4.2 For improved draught control, pure draught sensing, probably by adding the top and lower link signals, has most potential if the delay characteristic is modified from a pure time delay to a first order lag. Top or lower link sensing controls can also be improved in this way and some improvement, without major modifications, can be gained by damping the spool valve movement.

7.4.3 Three possible future tractor design trends will influence future implement controls,
   a) increased implement size
   b) increased forward speed
   c) increased use of p.t.o. driven implements. For a) there will be a trend away from mounted implements because of increased depth variations due to tractor pitch and more depth controls will be required. For b), better draught control probably by pure draught sensing will be needed to reduce the increased draught variations due to tractor pitch and bounce. For c) a control which senses p.t.o. torque may be required, in which case much of this work will be relevant in its development.

LIST OF SYMBOLS

\[\begin{align*}
\text{a} & \quad \text{distance from tractor rear wheel to tractor c. of g., m} \\
A & \quad \text{constant in rate of entry equation (maximum plough depth), m} \\
\text{b} & \quad \text{distance from tractor front wheel to tractor c. of g., m} \\
B & \quad \text{coefficient in rate of entry equation, m}^{-1} \\
\text{c} & \quad \text{coefficient of rolling resistance} \\
c_a & \quad \text{damping coefficient of hydraulic lift, Ns/m}
\end{align*}\]
$c_b$ damping coefficient of top link sensing unit, Ns/m
$c_f$ damping coefficient of front tyre, Ns/m
$c_r$ damping coefficient of rear tyre, Ns/m
$c_s$ equivalent damping coefficient of soil, Ns/m
$c$ output from control circuit
$\text{COT}$ coefficient of traction (drawbar pull/weight on rear wheels)
d mean implement depth, m
d$_1 -$ d$_5$ distances defined in fig 3.20, m
$D$ dissipative energy, J
$DB$ control deadband, N
$DV$ deadband in terms of depth, mm
$D_1$, $D_2$ constants in depth wheel equation, N/mm, N
e horizontal distance from tractor rear wheel to lower link ends, mm
$E$ maximum output from non-linear device
$f_1$ function describing steady state relationship of $H$ with depth
$f_2$ function describing steady state relationship of $V$ with depth
$f_2'$ function describing steady state relationship of $V'$ with depth
$f_3$ function describing relationship of torque with engine speed
$f_4$ function describing relationship of slip with COT
$F_r$ soil reaction force for pivoted depth wheel, N
$F_L$ horizontal component of lower link force, N
$F_R$ force in lift rods, N
$F_T$ horizontal component of top link force, N
$g_1$ gear ratio (engine speed/output speed of drive shaft from gearbox
in particular ratio
$G$ overall gear ratio (engine speed/wheel speed)
$G(s), G(j\omega)$ general transfer function for linear part of control
$G_N(s), G_N(j\omega)$ general transfer function for non-linear part of control
$h$ distance defined in fig 3.9, m
$h_d$ height of drawbar above ground, m
$h_g$ height of tractor c. of g. above ground, m
\( h_l \) height of lower link sensing unit above ground, m

\( h_p \) height of lower link ends above ground, m

\( H \) draught force, N

\( H_D \) horizontal drawbar force, N

\( H_s \) specific draught force (draught/unit cross sectional area of disturbed soil) N/m^2

\( I \) inertia of implement about its c. of g., kg m^2

\( I_e \) inertia of engine, driveline and tractor rear wheels referred to engine flywheel, kgm^2

\( I_{eq} \) inertia of engine, driveline and tractor rear wheels referred to the rear axle, kg m^2

\( I_{\theta 2} \) inertia of implement about lower link ends, kg m^2

\( I' \) inertia of implement about tractor c. of g., kg m^2

\( j \) square root of -1

\( J \) inertia of tractor about its c. of g., kg m^2

\( k \) sinkage constant \( = k_c/l + k_p \), N/m

\( k_a \) stiffness of hydraulic lift, N/m

\( k_b \) stiffness of top link sensing unit spring, N/m

\( k_c \) sinkage constant, N/m

\( k_d \) gain of torque sensing coupler, mm/Na

\( k_f \) stiffness of front tyre N/m

\( k_g \) feedback gain (spool valve displacement/lift arm rotation) mm/rad

\( k_l \) extra gain term for lower link sensing

\( k_c \) compliance at lower link ends due to oil compressibility, N/m

\( k_p \) slope of proportional control response, mm/N

\( k_r \) stiffness of rear tyre, N/m

\( k_s \) gain (rate of lift/spool valve movement), s^{-1}

\( k_t \) stiffness of top link sensing unit spring N/mm

\( k_v \) stiffness of depth wheel, N/mm

\( k_s \) sinkage constant, N/m

\( k_l \) gain (lift arm rotation/lower link displacement) rad/mm
$k_1$, slope of draught/depth relationship, N/mm

$k_2$, constant for linkage force sensing control, S/mm

$k_3, k_4$, constants in governor equation

$K$, constant in state equations

$K_1, K_2$, constants in draught/forward speed equations for plough, N/m$^2$, Ns$^2$/m$^4$

$K_3, K_4$, constants in draught/forward speed equations for cultivator, N, Ns/m

$l$, nominal tyre width, mm

$l_{fn}$, distance from depth wheel to nth furrow, m

$l_p$, length of plough (fig 2.15), m

$l_1 - l_6$, $l_6'$, distances defined in fig 2.19, m

$L$, resultant force on implement, N

$m$, mass of implement, kg

$M$, mass of tractor, kg

$n$, number of plough furrows

$n_s$, sinkage exponent

$N$, value of gradient along an isocline

$p_g$, ground pressure of depth wheel, N/m$^2$

$P$, thrust at rear wheels, N

$q_i$, generalised co-ordinate

$Q_{q_i}$, external force acting in $q_i$ co-ordinate direction

$r$, rear wheel radius, m

$r_w$, radius of depth wheel, m

$R$, input to control circuit

$R_w$, rolling resistance of depth wheel, N

$R_{w_1}, R_{w_2}$, rolling resistance of depth wheels on trailed plough, N

$RR$, rolling resistance of tractor wheels, N

$S$, wheel slip

$S_i$, soil support force on implement, N
ST saturated zone boundary

t time, s

t_d pure time delay, s

T kinetic energy, J

T_d time constant of 1st order lag, s

T_e engine torque, Nm

T_set set engine torque, Nm

U potential energy, J

v_p plough resultant velocity, m/s

V total vertical force on implement, N

V vertical force on plough bodies due to forward steady state motion through the soil, N

V_L vertical force at lower link ends, N

V_o total steady state component of vertical force on implement, N

V_w soil/reaction force on depth wheel, N

V_w1, V_w2 soil reaction forces on depth wheels of a trailed implement, N

w tractor wheelbase, m

W tractor weight, kg

W_i implement weight, kg

W_F weight on tractor front wheels, kg

W_R weight on tractor rear wheels, kg

W_w weight of depth wheel and linkage, kg

x, x, x tractor forward displacement, velocity acceleration, m, m/s, m/s^2

x_d spool valve displacement, mm,

x_n nth state variable

x_p implement longitudinal displacement, m

x_r governor rack displacement, m

x_r max, x_r min maximum and minimum governor rack displacements, m

x_r spool valve opening, mm
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x_{s_{\text{max}}}$</td>
<td>maximum spool valve displacement, m</td>
</tr>
<tr>
<td>$y_c$</td>
<td>displacement of pivoted depth wheel relative to implement, m</td>
</tr>
<tr>
<td>$y_c'$</td>
<td>vertical distance from lower link ends to centre of depth wheel, m</td>
</tr>
<tr>
<td>$y_d$</td>
<td>deflection of depth wheel tyre, m</td>
</tr>
<tr>
<td>$y_{dw}$</td>
<td>depth of rear furrow of semi-mounted plough, m</td>
</tr>
<tr>
<td>$y_{r_n}$</td>
<td>depth of nth furrow of semi-mounted plough, m</td>
</tr>
<tr>
<td>$y_i$</td>
<td>depth disturbance, m</td>
</tr>
<tr>
<td>$y_l$</td>
<td>displacement of lower link ends, m</td>
</tr>
<tr>
<td>$y_{\text{min}}$</td>
<td>minimum correction movement of lower link ends for torque sensing control, m</td>
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<tr>
<td>$y_o$</td>
<td>implement depth, m</td>
</tr>
<tr>
<td>$\dot{y}<em>{o</em>{\text{max}}}$</td>
<td>maximum rate of change of implement depth, m/s</td>
</tr>
<tr>
<td>$y_r$</td>
<td>governor setting</td>
</tr>
<tr>
<td>$y_t$</td>
<td>displacement of top link sensing spring, m</td>
</tr>
<tr>
<td>$z, z, z'$</td>
<td>tractor vertical displacement, velocity acceleration, m, m/s, m/s²</td>
</tr>
<tr>
<td>$z_p$</td>
<td>implement vertical displacement, m</td>
</tr>
<tr>
<td>$z_s$</td>
<td>sinkage of depth wheel, m</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>angle of top link to horizontal, rad</td>
</tr>
<tr>
<td>$\beta$</td>
<td>angle of lower links to horizontal, rad</td>
</tr>
<tr>
<td>$\delta$</td>
<td>inclination of plough to horizontal, rad</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>lift arm rotation, rad</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>error signal</td>
</tr>
<tr>
<td>$\eta$</td>
<td>driveline efficiency</td>
</tr>
<tr>
<td>$\eta_t$</td>
<td>tractive efficiency</td>
</tr>
<tr>
<td>$\omega_c$</td>
<td>engine speed, rad/s</td>
</tr>
<tr>
<td>$\omega_{c_{\text{e}}}$</td>
<td>nominal engine speed at governor setting $y_r$, rad/s</td>
</tr>
<tr>
<td>$\omega_{L}$</td>
<td>angle of force $L$ to horizontal, rad</td>
</tr>
<tr>
<td>$\theta_w$</td>
<td>angle between depth wheel and linkage ($= \text{mean } \theta_w + \text{variation } \theta_\delta$) rad</td>
</tr>
</tbody>
</table>
\( \theta_1 \) rotation of lower links, rad

\( \theta_2 \) rotation of implement about lower link ends, rad

\( \lambda_1, \lambda_2, \lambda_3 \) constants in depth wheel equations

\( \nu \) governor damping coefficient

\( \phi \) tractor pitch, rad

\( \omega \) frequency, rad/s

\( \nu_n \) governor natural frequency, rad/s
REFERENCES


2. Howe, S.D. Direct drills. Power farming, Dec 1972 10


5. M.A.F.F. Annual Statistics 197


8. Ride vibration studies. N.I.A.E. project review PR/E/75/1445


10. Bunting, E.V. The Ferguson system - past and present. ASAE paper 65-133


22. GETZLAF, G. Comparative study of the forces acting on standard plough bodies. Grundl. Landtech. 1953 16
24. GETZLAF, G. Changes in forces when turning plough bodies from their normal position. Grundl. Landtech. 1952 3 71
25. KUZENKO, V.V. Modelling of the load dynamics of ploughs during laboratory tests. Trakt. Selkhozmash. 1970 (10) 22
28. KRASTIN, E.N. The use of dimensional analysis for evaluation of draft characteristics of a plow during operation in different condition. Doklady N.I.I.S.P. 1973 8(1) 66
29. REAVES, C.A. Mouldboard plow - design and use for higher horsepower lower weight tractors. SAE paper TI0681 Sept 1971
33. SKALWEIT, H. Measurement of forces on the three point linkage in field tests on ploughs with control systems operated through the hydraulic lift. Grundl. Landtech. 1964 20 53 (N.I.A.E. Trans 272)
34. DAVIDSON, J.B.; FLETCHER, L.J.; COLLINS, E.V. Influence of speed upon draft of a plow. Trans. A.S.A.E. Vol 13 Dec 1919
36. MCKIBBEN, E.G.; REED, I.F. Influence of speed on the performance characteristics of implements. SAE National tractor meeting, Milwaukee Sept 1972
37. GORYACHKIN, V.P. 1929 Sob Sochimenii (collected papers) Vol 2 Kolos 1955
38. KINACEZ, A.A. Draught resistance of chisel ploughs. Sel' Khozmaschine 1957 2 7 (N.I.A.E. Trans 89)
39. MOLLER, R. Draught requirements and working efficiency of rigid and spring cultivator tines. Grundl. Landtech 1959 11 85


SKALWEIT, H. Measurements at two points of forces between tractor and mounted plough. Landtech. Forschung, 1964 11 151

SKALWEIT, H. Determination of forces on tractor and plough with a control system operated via the hydraulic lift. Landtech. Forschung, 1962 12 55

SKALWEIT, H. The ideal and actual in depth control via the hydraulic lift. Grundl Landtech 1963 18 574

SKALWEIT, H. Field measurements on tractors with the three point linkage and control system operated through the hydraulic lift. Landtech Forschung 1954 14 1

TREUGOT, A. Movements and lifting forces in the tractor/three point linkage/implement system. Ber. Landtech. 121 KTL Frankfurt/Main 1968 82


HAIN, K. Kinematics of working depth control of tractor mounted implements. Grundl. Landtech. 1952 3 119


SKALWEIT, H. Forces operating to maintain the working depth of tractor mounted implements. Grundl. Landtech 1952 3 109

TOKAREV, V. Effect of the angle of inclination of the tractor links on plough performance. Tekh. sol. khoz 1970 30 (2) 75

British Leyland Mini tractor


59. KUCHEVSKII, J. Assessment of the evenness of ploughing depth on fields with different profiles. Elektrif sel'khoz 1975 8 53

60. MATSNEV, M. Comparative assessment of evenness of ploughing depth of different types of plough. Tekh. Selkhoz 1972 9 72

61. SEIPERT, A. Evenness of furrow depth when ploughing. Grundl. Landtech. 1962 17 (7) 226


68. HESSE, H. Reasons and scope for further development of plough control systems. LandbForsch., Volkenrode, 1973 23 (1) 78

69. HESSE, H. Limitations to fast hydraulic control systems. Grundl. Landtech 1973 23 (2) 45

70. HESSE, H.; MOLLER, R. An electro-hydraulic depth control which senses two parameters for large mounted ploughs. Parts I and II. Grundl. Landtech 1972 22 (3) 75 and 1972 22 (4) 102

71. ZEVELEV, Z.N. Some questions of the structure and kinematics of mounted systems with an automatic mechanism for maintaining the pre-set cultivation depth. Povysh effektivnosti. ispolz. tekh. v Selkhov. Proizv 54 Gorki 1968 16

72. HOOK, R.W.; MURPHY, K.E. Integral flexible implement for farm tractor features depth control on uneven terrain. SAE 1970 78 (1) 62


76. PISAR, E. Results of tests with a proposed system for receiving the signals for controlling the hydraulic linkage. Acta operativo-economica V, Univ. agric. Nitra III 1971 127


TAPP, G.E. County Commercial Cass Ltd., Private communication

WILSON, R.W. Hydraulic power lift controls and power utilisation for larger tractors. Paper 710685 SAE meeting, Milwaukee, Sept. 1974

HESSE, H.; MOLLER, R. Possible methods of increasing the load on the rear axle Grundl. Landtech. 1969 19 (14) 119

HESSE, H.; MOLLER, R. Investigation of a system for increasing the drive axle load of a farm tractor operating with a semi-mounted plough. Grundl. Landtech. 1974 24 (5) 164


ZOZ, F.M. Predicting tractor field performance. ASAE paper 70-118

GRECHENKO, A. Predicting the performance of wheel tractors in combination with implements J. Agric. Engng. Res. 1968 13 (1) 49

LEVITAMUS, A.D.; KORSUM, N.J. Optimum relationship between engine speed and working width of a machine operating with tractors of various powers. Trakt. Selkhozmash. 1971 (2) 7

ZOZ, F.M. Optimum width and speed for least cast tillage. ASAE paper 73-1528

WISMER, R.D.; LUTH, H.J. Off-road traction prediction for wheeled vehicles. ASAE paper 72-619


Dwyer, M.J.; Pearson, G. A field comparison of the tractive performance of two and four wheel drive tractors J. Agric. Engng. Res. 1976 21(1) 77

Steinkampf, H. Problems in the efficient conversion of engine horsepower to drawbar horsepower in the case of high power tractors. Grundl. Landtech. 1974 24 (1) 14

Caraghiubic, G.; Mihaouzi, I. Investigation of the effectiveness of controlled hydraulic power lifts in increasing tractor adhesion Lucrari, Stuntif, 15 Inst. cercet. Mec. Agric. Bucuresti 337

Foxwell, W.F. Maximising the drawbar horsepower of farm tractors. Paper for discussion, Akron Rubber Group Inc. Akron, Jan. 1975
95. HUNT, DONELL Selecting an economic power level for the big tractor. ASAE paper 71-147
97. COLEMAN, R.N.; WLIKINS, J.D. High speed field tractors - Why? SAE paper 710586
100. PERSHING Simulating tractive performance SAE paper 710525
101a. PERSHING, A computational scheme for matching required and available power in vehicle simulation. SAE Miss. Val. Sect. Apr. 1971
102. SMITH, D.W., YOERGER, R. Variations in the forward motion of farm tractors Trans. ASAE 1975 18 (3) 401
103. Investigation of the performance of tractor draught controls. NIAE project paper T/TDPD/68/1253
104. N.I.A.E. Project review PR/T/72/1410
105. N.I.A.E. Project review PR/T/73/1410
110. CROLLA, D.A. A computer programme for simulating the performance of a tractor and mounted cultivation implement under varying load conditions. N.I.A.E. Note 62/1410


128. Wigginton, R. British Leyland Tractors, Private communication

129. Hull, C. David Brown Tractors. Private communication

130. IBM System/360 continuous system modelling program (360A-CX-16X). Users' manual

131. Mechanical World Year Book 1970
APPENDIX 2.1  

Values of parameters and equations of motion used for tractor/implement ride model

Values of parameters

The figures used are from a ballasted 63 p.t.o. kW tractor and three furrow plough.

- Tractor mass
- Tractor moment of inertia
- Plough mass
- Plough moment of inertia
- Front tyre stiffness
- Front tyre damping
- Rear tyre stiffness
- Rear tyre damping
- Hydraulic stiffness
- Hydraulic damping
- Top link sensing unit stiffness
- Top link sensing unit damping
- Distances (see Fig. 2.19)
- Equivalent soil stiffness
- Equivalent soil damping

Equations of motion

The four second order equations of motion can be written

\[ a_1 \ddot{z} + a_2 \dot{\theta} + a_3 \dot{\theta}_1 + a_4 \dot{\theta} + a_5 \ddot{\theta} + a_6 \dot{\theta}_1 + a_7 \dot{\theta}_2 + a_8 \dot{\theta}_3 + a_9 \dot{\theta}_4 + a_{10} \ddot{\theta} = 0 \]
\[
\begin{align*}
&b_1 z + b_2^* + b_3 \theta_1 + b_4 \theta_2 + b_5^* + b_6^* + b_7 \dot{\theta}_1 + b_8 \dot{\theta}_2 + b_9 z + b_{10}^* \\
&+ b_{11} \theta_1 + b_{12} \theta_2 = 0 \\
&c_1 z + c_2^* + c_3 \theta_1 + c_4 \theta_2 + c_5^* + c_6^* + c_7 \dot{\theta}_1 + c_8 \dot{\theta}_2 + c_9 z + c_{10} \theta_2 = 0 \\
d_1 z + d_2^* + d_3 \theta_1 + d_4 \theta_2 + d_5^* + d_6^* + d_7 \dot{\theta}_1 + d_8 \dot{\theta}_2 + d_9 z + d_{10}^* \\
+ d_{11} \theta_1 + d_{12} \theta_2 = 0
\end{align*}
\]

where \( a_1 = M + m \)

\[
\begin{align*}
a_2 &= -\frac{m l_4}{2} \\
a_3 &= -\frac{m l_1}{2} \\
a_4 &= -\frac{m l_2}{2} \\
a_5 &= c_f + c_r + c_s \\
a_6 &= b_0 - a c_r - c_s l_4 \\
a_7 &= -c_s l_1 \\
a_8 &= -c_s l_2 \\
a_9 &= k_f + k_r \\
a_{10} &= b k_f - c k_r \\
b_1 &= -\frac{m l_4}{2} \\
b_2 &= J + I \rho + m l_4^2 \\
b_3 &= m l_1 l_4 \\
b_4 &= m l_2 l_4 \\
b_5 &= b c_f - a c_r - c_s l_4 \\
b_6 &= b^2 c_f + a^2 c_r + c_s l_4^2 \\
b_7 &= c_s l_1 l_4 \\
b_8 &= c_s l_2 l_4 \\
b_9 &= b k_f - a k_f + k_r (d + h_g) \\
b_{10} &= b^2 k_f - a^2 k_r - k_r l_4 (2d + h_g) \\
b_{11} &= a k_f l_1 (2d + h_g)
\end{align*}
\]
\[ b_{12} = -k_1 l_2 (2d + h_p) \]
\[ c_1 = -m l_1 \]
\[ c_2 = m l_1 l_4 \]
\[ c_3 = m l_1 \]
\[ c_4 = m l_1 l_2 \]
\[ c_5 = -c_4 l_1 \]
\[ c_6 = c_5 l_1 l_4 \]
\[ c_7 = \frac{c_5}{4} l_1^2 + c_6 l_1^2 \]
\[ c_8 = c_6 l_1 l_2 \]
\[ c_9 = \frac{k_1 l_1^2}{4} \]
\[ d_1 = -m l_2 \]
\[ d_2 = m l_2 l_4 \]
\[ d_3 = m l_1 l_2 \]
\[ d_4 = I_{\theta_2} + m l_2^2 \]
\[ d_5 = -c_8 l_2 l_4 \]
\[ d_6 = c_8 l_2 l_4 \]
\[ d_7 = c_8 l_2 l_1 \]
\[ d_8 = c_6 l_3^2 + c_5 l_2 \]
\[ d_9 = k_1 (d + h_p) \]
\[ d_{10} = -k_1 (dl_2 + dl_4 + h_p l_4) \]
\[ d_{11} = -k_1 l_1 (d + h_p) \]
\[ d_{12} = -k_1 l_3^2 - k_1 l_2 (2d + h_p) \]

Combining these four equations with the four first order equations gives the matrix equation:

\[
\begin{bmatrix}
\Lambda \\
\frac{d}{dt} (\mathbf{y})
\end{bmatrix} + \begin{bmatrix}
B^t
\end{bmatrix} (\mathbf{y}) = 0
\]
Where $\mathbf{v} = \begin{bmatrix} z \\ \dot{\theta}_1 \\ \dot{\theta}_2 \\ z' \\ \dot{\theta}_1 \\ \dot{\theta}_2 \\ \beta \\ z' \\ \beta \\ \theta_1 \\ \theta_2 \end{bmatrix}$

$A' = \begin{bmatrix} a_1 & a_2 & a_3 & a_4 & 0 & 0 & 0 & 0 \\ b_1 & b_2 & b_3 & b_4 & 0 & 0 & 0 & 0 \\ c_1 & c_2 & c_3 & c_4 & 0 & 0 & 0 & 0 \\ d_1 & d_2 & d_3 & d_4 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}$

$B' = \begin{bmatrix} a_5 & a_6 & a_7 & a_8 & a_9 & a_{10} & 0 & 0 \\ b_5 & b_6 & b_7 & b_8 & b_9 & b_{10} & b_{11} & b_{12} \\ c_5 & c_6 & c_7 & c_8 & 0 & 0 & a_9 & 0 \\ d_5 & d_6 & d_7 & d_8 & d_9 & d_{10} & d_{11} & d_{12} \\ -1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -1 & 0 & 0 & 0 & 0 \end{bmatrix}$
APPENDIX II

A computer programme for simulating the performance of a tractor and mounted cultivation implement in field conditions

1. INTRODUCTION

This section is based on a report (110) prepared for tractor manufacturers who were interested in running this programme. It contains a brief description of the programme, how to run it and some typical input data for a medium power tractor operating a plough or cultivator. The theory on which the programme is based and the fieldwork carried out to validate it has been described already in sections 3 and 5.

The programme and report were written originally in Imperial units and are shown here unchanged.

2. The computer programme

The computer programme is given in full in Appendix IIa

2.1 Input data

This section describes what input data the programme requires and how to read it into the programme. A range of typical data is quoted so that you can either use this or substitute your own data. If in doubt follow the format used in the example programme in Appendix IIa.

2.1.1 Tractor

Computer variable

TORQSP Function describing the torque v engine speed curve of the tractor engine. It is read in as a series of points of (engine speed-rev/min, torque-lb ft). A typical relationship is shown in Fig. 2.35

GEARS Table containing the overall gear ratios i.e. engine revolutions axle revolutions for all the tractor gears. Reading in all the gear ratios initially makes it easier to change gear for subsequent runs.

MXENSP Maximum engine speed (rev/min). This is used as the initial condition of the engine speed

GEAR Gear number for this run

WT Tractor weight ballasted for the cultivation operation (lb).
<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>WF</td>
<td>Distance from the front wheel to the tractor centre of gravity (in)</td>
</tr>
<tr>
<td>R</td>
<td>Rear wheel rolling radius (in)</td>
</tr>
<tr>
<td>RP</td>
<td>Front wheel rolling radius (in)</td>
</tr>
<tr>
<td>IWF</td>
<td>Rotational Inertia of front wheel (lb-ft-s²)</td>
</tr>
<tr>
<td>IWR</td>
<td>Rotational Inertia of rear wheel (lb-ft-s²)</td>
</tr>
<tr>
<td>IE</td>
<td>Rotational Inertia of engine and transmission referred to the engine flywheel (lb-ft-s²)</td>
</tr>
<tr>
<td>W</td>
<td>Wheelbase (in)</td>
</tr>
<tr>
<td>EFFCY</td>
<td>Driveline efficiency</td>
</tr>
</tbody>
</table>

The information for TORQSP and GEARs can be obtained from a standard O.E.C.D. test report (113). The inertias are probably the most difficult data to obtain, but if they are not available, they can be calculated for the front and rear wheels making the assumptions that the wheels can be replaced by a disc of equivalent weight and the tyres by toroidal rings.

### 2.1.2 Draught control

Fig. 2.36 is an example of a typical draught control response having on-off control when lifting and proportional control when lowering. All the information on this graph is required, plus the delay times when lifting and lowering. The variable MODE can be set so that pure draught sensing, top link sensing or lower link sensing control can be specified. For pure draught sensing control only the draught force is sensed.

**Computer variable**

- **DB**: Deadband (lb)
- **MAXROL**: Maximum rate of lifting (in/s)
- **MAXRLW**: Maximum rate of lowering (in/s)
- **PROPUP**: Slope of proportional response when lifting (in \( s^{-1} \text{ lb}^{-1} \))
  - Set to 10.0 for on-off control
- **PROPDN**: Slope of proportional response when lowering (in \( s^{-1} \text{ lb}^{-1} \))
- **DELUP**: Delay when lifting (s)
- **DELDN**: Delay when lowering (s)
Fig 2.36 Typical draught control response
MODE

Set to 0. for pure draught sensing
Set to 1. for simulated lower link sensing
Set to 2. for simulated top link sensing

A summary of the draught control parameters measured on the N.I.A.E. rig (115) is shown in Table 3.1. There are two points to note about these results. The deadband quoted is the maximum deadband obtainable. Most manufacturers offer a variety of top link positions and the most sensitive one was always used in these tests. This may not be the position recommended for ploughing but the effective deadband for any top link position can be found knowing its lever ratio. Also the rates of lowering quoted were all measured with the same weight (1000 lb) acting on the ends of the lifting arms. For tractors not fitted with a flow control for lowering, the rate of lowering will vary with the weight acting on the lift arms i.e. implement weight. Fortunately most tractors have a flow control and as long as there is some weight acting on the lift arms the rate of lowering is independent of the weight. If this is not the case, the relationship must be measured and incorporated in the programme as a relationship between ROLOW and VERF.

2.1.3 Implement

Computer variable

HDC Function describing draught v implement depth curve. Read in as a series of points of (depth-in, draught-lb)

VDC Function describing vertical force v implement depth curve. Read in as a series of points of (depth-in, vertical force-lb)

L Distance from tractor front wheel to line of action of vertical force on implement, normally assumed to be halfway along the length of implement in contact with the soil (in)

LG Distance from tractor front wheel to centre of gravity of implement (in)

PLWT Implement weight (lb)

A Constant in plough rate of entry equation (in)

B Constant in plough rate of entry equation (in)
Constant in cultivator rate of entry equation

N.B. If plough is simulated set \( C = 0 \), and if cultivate is simulated set \( A = B = 0 \).

SETD Set implement depth (in)

The entry of a plough into the ground can be described by the equation:

\[
y_0 = A(1 - e^{-Bx})
\]

where \( y_0 \) = plough depth
\( x \) = distance along the ground

\( A, B \) = constants

The constants \( A \) and \( B \) which are read into the programme depend on the soil characteristics and the tractor linkage. A small FORTRAN programme shown in Appendix IIc is used to calculate \( A \) and \( B \). The input data required is

1. a title of up to 60 characters
2. The integer number of points on the draught v depth curve.
3. the points on the draught v depth curve in free format, 10 numbers to a line.
4. the integer number of points on the vertical force v depth curve,
5. the points on the vertical force v depth curve,
6. the co-ordinate points of the linkage mounting points on the tractor, which are now given in tractor O.E.C.D. test reports (113), \((t_x, t_y)\) (See Fig 2.37)
7. the lower link length \( l \) and the set depth of the plough (+ SETD)

The programme then calculates the top link length required so that the plough is level at the set depth. It then calculates the maximum depth reached by the plough, which occurs when the resultant force on the plough passes through the instantaneous centre of rotation. The maximum depth is the constant \( A \).

The equation of the entry of the plough can be differentiated and
Fig 2.37 Linkage Data for Programme to Calculate Rate of Entry Equation
re-arranged to give the rate of entry in terms of the depth. 

\[
\frac{dy_o}{dx} = B(A-y_o) \tag{2.23}
\]

The initial rate of entry of the plough, i.e. when \( y_o = 0 \), is a constant, so knowing this and \( A, B \) can be found.

One particular assumption is made in this programme. The relationship of draught and vertical force with depth were measured with one particular linkage, and it is assumed that the forces are the same for other linkages. This is not strictly true because the variation in inclinations of the plough caused by different linkages alter the forces slightly. However the effect is small and a change in inclination either increases both \( H \) and \( V \) or decreases them both. Hence the angle of the resultant force, which is important when calculating the maximum depth is altered very little.

For the cultivator, the rate of entry into the ground is simply a constant, and has been found to be independent of soil type over a wide range of soils (108), and not significantly affected by the tractor linkage.

Typical data is available to simulate a three furrow mounted plough and a seven rigid tine cultivator. They can be simulated in three field conditions, shown in Table 2.3 (plough) and Table 2.4 (cultivator). Semi-mounted implements have also been simulated with the programme but it involves alterations to the programme. If you want to simulate semi-mounted implements, please contact the author.

The curves of draught and vertical force against implement depth are shown in Fig 2.38 and 2.39. They were all measured at one speed, 1.3 m/s (3 mile/h), but in fact draught increases with speed. So the programme, having read the draught for a particular depth from the input curve, then alters it to its value at the true tractor forward speed.

Specific draught can be described by the following equation (89)

\[
\text{Specific draught} = K1 + K2 \text{ (Speed)}^2
\tag{2.15}
\]

where

\[
K2 = 0.01 K1 + 0.04 \tag{2.16}
\]

Knowing the draught, depth and width of ploughing at 1.3 m/s (3 mile/h), \( K1 \) and \( K2 \) can be calculated from these equations, and hence the draught at
TABLE 2.3

Input data for three furrow mouldboard plough in three field conditions

<table>
<thead>
<tr>
<th>Field type</th>
<th>Heavy</th>
<th>Medium</th>
<th>Light</th>
</tr>
</thead>
<tbody>
<tr>
<td>Function</td>
<td>HDC</td>
<td>VDC</td>
<td>HDC</td>
</tr>
<tr>
<td>Coordinate points</td>
<td>HDC</td>
<td>VDC</td>
<td>HDC</td>
</tr>
<tr>
<td>(0., 0.)</td>
<td>(0., 1200.)</td>
<td>(0., 0.)</td>
<td>(0., 1200.)</td>
</tr>
<tr>
<td>(2., 760.)</td>
<td>(2., 1310.)</td>
<td>(2., 570.)</td>
<td>(2., 1300.)</td>
</tr>
<tr>
<td>(4., 1520.)</td>
<td>(4., 1460.)</td>
<td>(4., 1130.)</td>
<td>(4., 1370.)</td>
</tr>
<tr>
<td>(5., 1950.)</td>
<td>(6., 1520.)</td>
<td>(6., 1730.)</td>
<td>(6., 1380.)</td>
</tr>
<tr>
<td>(6., 2450.)</td>
<td>(8., 1500.)</td>
<td>(7., 2100.)</td>
<td>(8., 1310.)</td>
</tr>
<tr>
<td>(7., 3150.)</td>
<td>(9., 1440.)</td>
<td>(8., 2550.)</td>
<td>(9., 1230.)</td>
</tr>
<tr>
<td>(8., 4500.)</td>
<td>(10., 1320.)</td>
<td>(9., 3300.)</td>
<td>(10., 1100.)</td>
</tr>
<tr>
<td>(9., 6000.)</td>
<td>(11., 1130.)</td>
<td>(9., 4500.)</td>
<td>(11., 940.)</td>
</tr>
<tr>
<td>(12., 750.)</td>
<td>(12., 700.)</td>
<td>(12., 700.)</td>
<td>(12., 620.)</td>
</tr>
</tbody>
</table>


Constant A: 7.9, 9.1, 11.5

Constant B: 0.057, 0.049, 0.039
Fig 2.38 Curves of Horizontal and Vertical Force Against Depth for a Three Furrow Plough in Three Soil Conditions.
<table>
<thead>
<tr>
<th>Field type</th>
<th>Heavy</th>
<th>Medium</th>
<th>Light</th>
</tr>
</thead>
<tbody>
<tr>
<td>Function</td>
<td>HDC</td>
<td>VDC</td>
<td>HDC</td>
</tr>
<tr>
<td>Coordinate points</td>
<td>(0., 0.)</td>
<td>(0., 1250.)</td>
<td>(0., 0.)</td>
</tr>
<tr>
<td></td>
<td>(2., 800.)</td>
<td>(4., 1250.)</td>
<td>(2., 650.)</td>
</tr>
<tr>
<td></td>
<td>(4., 1750.)</td>
<td>(6., 1400.)</td>
<td>(4., 1350.)</td>
</tr>
<tr>
<td></td>
<td>(6., 3000.)</td>
<td>(8., 1720.)</td>
<td>(6., 2160.)</td>
</tr>
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<td></td>
<td>(7., 3850.)</td>
<td>(9.2, 2000.)</td>
<td>(8., 3100.)</td>
</tr>
<tr>
<td></td>
<td>(7.5, 4500.)</td>
<td>(9.2, 2000.)</td>
<td>(9., 3720.)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(9.6, 4500.)</td>
</tr>
<tr>
<td></td>
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</tr>
<tr>
<td></td>
<td></td>
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</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Implement weight PLWT</td>
<td>1250</td>
<td>1250</td>
<td>1250</td>
</tr>
<tr>
<td>Constant C</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
</tr>
</tbody>
</table>
Fig 2.39 Curves of Horizontal and Vertical Force Against Depth for a Seven-Tine Cultivator in Three Soil Conditions
the true tractor forward speed.

2.1.4 Field

Computer variable

SLIPUL Function describing slip v. traction coefficient curve for a particular field and tyre. Read in as a series of points of (traction coefficient, slip (%))

RRC Rolling resistance

SC Field surface roughness factor

Table 2.5 gives typical slip against traction coefficient curves and rolling resistance coefficients for three field conditions. These figures were obtained with a ballasted Ford 5000 tractor fitted with 13.6 x 38 tyres, and they were plotted in Fig. 2.40. Values of the field surface roughness factor, SC, were obtained by measuring the draught variation when the implement was locked in position relative to the tractor (position control). To simulate a field surface, the co-ordinate points of the surface are multiplied by the scale factor, which is varied until the predicted draught variation equals the measured draught variation. The simulated field surface is then a good representation of the actual field surface. Typical values for the scale factor are:

- rough surface 0.35
- medium surface 0.25
- smooth surface 0.2

If you want to predict the effect of different tyre sizes in various field conditions this can be done using data from the 'Handbook of Agricultural Tyre Performance' published by the N.I.A.E. (114).

This handbook contains tables of the performance of a range of tractor tyres at various inflation pressures and loads. The rolling resistance values are quoted, so by dividing this value by the load on the tyre, the rear wheel rolling resistance coefficient can be obtained. No information on front wheel rolling resistance is available, although work on this is in progress at N.I.A.E. so the best approximation is to assume that the front wheel rolling resistance...
<table>
<thead>
<tr>
<th>Traction conditions</th>
<th>Good</th>
<th>Average</th>
<th>Poor</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(0., 0.)</td>
<td>(0., 0.)</td>
<td>(0., 0.)</td>
</tr>
<tr>
<td></td>
<td>(0.1, 2.)</td>
<td>(0.1, 2.)</td>
<td>(0.1, 2.)</td>
</tr>
<tr>
<td></td>
<td>(0.2, 4.)</td>
<td>(0.2, 4.)</td>
<td>(0.2, 4.)</td>
</tr>
<tr>
<td></td>
<td>(0.3, 7.)</td>
<td>(0.3, 7.)</td>
<td>(0.3, 8.)</td>
</tr>
<tr>
<td></td>
<td>(0.4, 10.)</td>
<td>(0.4, 12.)</td>
<td>(0.4, 17.)</td>
</tr>
<tr>
<td></td>
<td>(0.5, 14.)</td>
<td>(0.5, 22.)</td>
<td>(0.5, 37.)</td>
</tr>
<tr>
<td></td>
<td>(0.6, 21.)</td>
<td>(0.6, 37.)</td>
<td>(0.59, 100.)</td>
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<td></td>
<td>(0.7, 32.)</td>
<td>(0.7, 70.)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(0.8, 54.)</td>
<td>(0.74, 100.)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(0.87, 100.)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rolling resistance coefficient RRC</td>
<td>0.07</td>
<td>0.10</td>
<td>0.12</td>
</tr>
</tbody>
</table>
Fig 2.40 Typical Curves of Slip vs Traction Coefficient for Three Surface Traction Condition
coefficient is the same as that of the rear wheel. This value is read into the programme as RRC and when multiplied by the total tractor and implement weight plus the vertical soil force, it gives the total rolling resistance of the tractor.

To obtain the slip v traction coefficient curve from the handbook is slightly more complicated. Firstly we assume that the curve can be described by the following equation,

\[
\text{Slip} = \frac{1}{k} \log_e \frac{c}{c-TC}
\]

where c and k are constants. The handbook quotes two points on this curve, so by substituting these points back in the equation we can find c and k and hence the equation.

To obtain the slip v traction coefficient of a particular tyre at a load and inflation pressure, the following information must be read from the handbook.

- load on tyre \( WT_1 \)
- pull at 20% slip \( PULL_2 \)
- pull at maximum efficiency \( PULL_1 \)
- slip at maximum efficiency \( SLIP_1 \)

A simple iterative procedure is necessary to find the constants c and k and a FORTRAN programme has been written to do this. It is shown in Appendix IID along with sample data and output. The input data consists of:

1) the integer number of the data combinations following,

2) a title of up to 60 characters, describing the tyre etc.,

for each tyre data combination

3) the values of \( WT_1, PULL_2, PULL_1, SLIP_1 \) from the handbook, in free format i.e. each value separated from the next by a space.

The output is ten points of \((TC, SLIP)\) on the curve, ready to read in to the main programme. There is one limitation to this programme. If the two points are very close, the equation cannot be found, i.e. if the slip at
maximum efficiency is 19, 20 or 21, the programme cannot be used and some alternative source of slip/pull data must be used.

2.2 Running the programme

A copy of the complete programme is shown in Appendix IIa. Having made a copy of the programme and replaced the example data with your own data, the programme is ready for use. It can be run at any computer centre which has an implementation of the I.B.M. language (S/360 CSMP) (130). Any centre with I.B.M. S/360 should have this language available.

If you wish to run the programme several times, changing part of the input data each time, there are several ways you can do this.

a) To vary a PARAMETER

If for instance you want three runs in three gears, the following statement in the input data would achieve this

```
PARAMETER GEAR = (3, 4, 5)
```

Only one statement of this kind can be used for one sequence of runs.

b) To vary more than one PARAMETER

If for instance, you want to change the gear, and increase the dead-band and rate of lift for a second run, the end of the programme must be modified to

```
END
PARAMETER GEAR = 5, DB = 800., MAXROL = 16.
END
STOP
```

If desired, several PARAMETER and END combinations can be inserted to provide subsequent runs. Once a PARAMETER has been changed, it remains changed for subsequent runs.

c) To vary a FUNCTION

If, for instance, you want to change the horizontal and vertical force against depth curves to simulate a different field for the second run, the following statements are required at the end of the programme.
END
OVERLAY HDC = (0., 0.) .. etc
OVERLAY VDC = (0., 1200.) ... etc.
END
STOP

Several OVERLAY and END combinations can be used, and PARAMETER statements can be changed at the same time if required. The only limitation is that the number of \((x,y)\) points in the OVERLAY statement must not be greater than that in the original FUNCTION statement.

For more detailed information, consult the I.B.M. User's Manual (130).

2.3 Output

The programme simulates one minute of real time or 300 ft. of ploughing whichever is the shorter. These can be altered if required using the FINTIM statement shown below. If the tractor engine stalls or the wheelslip reaches 100\% during this time, the programme stops and prints out a message giving the reason.

A typical output is shown in Appendix IIb. The draught control parameters for the run are printed, with the average speed and slip. The draught force is sampled every .01 s during the run and the standard deviation and mean calculated and printed. The standard deviation is used as a measure of the effectiveness of a particular draught control.

Alternatively parameters can be printed at given intervals in the simulation. The interval is controlled by a PRDEL command in the TIMER statement.

e.g. PRINT HORF., VERF

    TIMER FINTIM = 60., DELT = .01, PRDEL = .1

These statements would cause HORF and VERF to be printed against TIME, every 0.1 s. Or any parameter can be plotted against time, the interval being controlled by OUTDEL.

e.g. PRTPLT HORF

    TIMER FINTIM = 60., DELT = .01, OUTDEL = .1
These statements would cause HORF to be plotted against TIME at intervals of 0.1 s. Examples of these outputs are shown in Appendix IIc. Please note that care should be taken when using these statements not to ask for too much information. For instance the above commands both print out 600 lines. If you just want to check some variables then reduce the FINTIM to ensure you do not get too much unnecessary information.

3. Suggestions and conclusions

Only a brief outline of the CSMP input and output statements used in this programme has been given here. For more detail or to clarify any of the points, you must consult the CSMP manual. If you have any queries about the programme described here or if you want to adapt it for another purpose, please contact the author.

Although the programme was originally developed to study draught control performance in field conditions, it has scope for a variety of other uses. For example, although steady state performance of tractors can be predicted, under dynamic conditions the draught force fluctuates and the tractive performance will be lower than that predicted from steady state considerations. The programme could be used to investigate the effects of draught variations on for example, forward speed, slip or engine speed. For instance, limits of tolerance of draught variation to maintain maximum forward speed could be calculated.

The effect of ballasting the tractor can be investigated by varying the input data relating to the tractor weight and centre of gravity position. Tyre performance can be investigated at the same time. Different tyre sizes and inflation pressures can be simulated by varying the slip v traction coefficient curve and rolling resistance and rolling radius values.

For a particular field condition there is an optimum combination of implement size and tractor gear ratio to achieve the maximum rate of work. The programme can be used to find this optimum combination. The gear can easily be changed, by changing the parameter GEAR, and to alter the implement size, it is sufficiently accurate to scale up or down the figures of draught and vertical force against depth given in the examples. For instance a four
The furrow plough can be approximated by multiplying the figures given for a three-furrow plough by $4/3$. Besides these two curves, the parameters $L$, $L_C$, and $PLWT$ must be altered. The constant $A$ and $B$ must be calculated for the new draught and vertical force relationships. For the cultivator, $C$ is independent of the number of tines.

The programme can be used to test new types of implement control. This requires an alteration to the programme to make the variable $A_1$ dependent on whatever parameter is sensed, rather than on the signal from the horizontal and vertical forces as at present. Two types of sensing that have been investigated already are driveline torque sensing and engine speed sensing. Both these controls however require additional feedback from the lift arm position to prevent over-correction and additional programming is required to simulate this. There is scope to test other types of control sensing in the programme, or more complicated arrangements which sense more than one parameter.

Various linkage configurations can also be simulated using the programme. First the Fortran programme must be used to calculate the constants in the equation for the rate of entry of the plough. Then these constants can be used in the main programme to find the effect of the linkage on the draught control response.

These are just some examples of the possible uses for the programme. They are not intended to form a comprehensive list since there are many other aspects of tractor and implement performance which could be studied by making the necessary parameter changes.
APPENDIX IIa

The C.S.M.P. Programme

TITLE TEST PROGRAMME USING FORD 5000 WITH 3 FURROW PLough

* READ IN TRACTOR PARAMETERS

  * TORUSP - FUNCTION DESCRIBING TORQUE Vs. ENGINE SPEED CURVE.
  * GEAR - TABLE CONTAINING OVERALL GEAR RATIOS FROM GEAR 1 TO MAX
  * UXENSP - MAX. ENGINE SPEED (REV/IN)
  * WT - TRACTOR WEIGHT (LB)
  * WF - DIST. FROM FRONT WHEEL TO TRACTOR C.G. (IN)
  * W - WHEELBASE (IN)
  * R - REAR WHEEL ROLLING RADIUS (IN)
  * RF - FRONT WHEEL ROLLING RADIUS (IN)
  * EFFCY - DRIVELINE EFFICIENCY (%)
  * IWF - INERTIA OF FRONT WHEEL (LB.FT.SEC.SEC)
  * IWR - INERTIA OF REAR WHEEL
  * IE - INERTIA OF ENGINE

FUNCTION TORUSP=(1200, 150), (1400, 132), (1600, 183), ...
(1800, 173), (2000, 178), (2200, 163), (2250, 160), ... 
(2300, 155), (2500, 150)

TABLE GEARs(1-3)=15.5, 173.5, 69.5, 72.8, 60.4, 48.3, 40.2, 20.5
PARAMETER GEAR=(3, 4.5), UXENSP=2380, WT=8570.
PARAMETER WF=50, W=82, R=27.5, RF=11.3, EFFCY=95.
PARAMETER IWF=6.6, IWR=14.2, IE=.8

* READ IN DRAUGHT CONTROL PARAMETERS

  * DB = DEADBEARD (LB)
* MAXROL = MAX. RATE OF LIFTING (IN/SEC)
* MAXRLW = MAX. RATE OF LOWERING (IN/SEC)
* PROPUP = SLOPE OF PROPORTIONAL RESPONSE LIFTING (IN/SEC/LB)
* PROPDN = SLOPE OF PROPORTIONAL RESPONSE LOWERING (IN/SEC/LB)
* DELUP = DELAY LIFTING (SEC)
* DELDN = DELAY LOWERING (SEC)
* MODE = SET = 0 FOR FORCE BRAUGHT SENSING CONTROL
* = 1 FOR LOWER LINK SENSING
* = 2 FOR TOP LINK SENSING CONTROL

PARAMETER BD=400., MAXROL=12., MAXRLW=12.
PARAMETER PROPUP=16., PROPDN=.11, DELUP=.05, DELDN=.05
PARAMETER MODE=2.

* READ IN IMPLEMENT PARAMETERS

HDC = FUNCTION DESCRIBING HORIZONTAL FORCE(DRAUGHT) V. DEPTH CURVE FOR A PARTICULAR FIELD. READ IN AS A SERIES OF POINTS OF (DEPTH-IN, DRAUGHT-LB)
VDC = FUNCTION DESCRIBING VERTICAL FORCE V. DEPTH READ IN AS A SERIES OF POINTS OF (DEPTH-IN, VERTICAL FORCE-LB)
PLWT = IMPLEMENT WEIGHT (LB)
L = DIST FROM FRONT WHEEL TO LINE OF ACTION OF V
LG = DIST FROM FRONT WHEEL TO IMPLEMENT CG
A = CONSTANT IN PLUGH RATE OF ENTRY EQUATION (IN)
B = " " (1/IN)
C = CONSTANT IN CULTIVATOR RATE OF ENTRY EQUATION
   (N.B. IF PLUGH IS SIMULATED, SET C=0.
   IF CULTIVATOR IS SIMULATED, SET A=B=0.)
SETD = SET IMPLEMENT DEPTH (IN)

FUNCTION HDC=(0.,.0.), (2., .570.), (4., 1.130.), (6., 1.730.), . . .
(7., 2.100.), (9., 2.550.), (9., 3.300.), (9., 4.500.)
FUNCTION VDC=(0., 1200.),(2., 1300.),(4., 1370.),(6., 1380.), . . .
(8., 1310.),(9., 1320.),(10., 1100.),(11., 940.),(12., 790.)
PARAMETER SETD=7.5, A=9.1, B= 6.049, C= 0., PLWT=1200.
PARAMETER L=136.,LG=126.

* READ IN FIELD PARAMETERS

* SLIPUL -FUNCTION DESCRIBING SLIP V. TRACTION COEFF.
  IN A PARTICULAR FIELD. READ IN AS A SERIES OF
  POINTS OF (TRACTION COEFF.,SLIP)

* RRC -ROLLING RESISTANCE COEFF.

* SC -SURFACE PLOWNESS FACTOR

FUNCTION SLIPUL=(0.,0.),(.1,.2.),(.2,4.),(.3,7.),(.4,10.),...
(.5,14.),(.6,21.),(.7,22.),(.8,54.),(.9,700.)

PARAMETER SC=.2.,RRC=.7.

* MAIN PROGRAMME

* CONSTANT G=32.2

FIXED K,L1,L2,L3,J,HISTO,GEAR,IDEUP,IDELDN.
STORAGE BUMPS(300),HISTC(20),JD(12),GEARS(12)
INITIAL

HOSORT
IT(BUMPS(1),HISTO) GO TO 5
2 READ(S,100)BUMPS
150 FORMAT(10F6.0)
3 CONTINUE
D=DD/2.
IDEUP=DELUP*101.
IDELDN=DELDN*101.
GR=GEARS(GEAR)
MENSP=MXENSP+1.046
G=GEAR

DYNAMIC
Z1=DIST+2.+1.
Z2=(DIST+(L-L2)/12.)*2.+1.
Z3=(DIST+(L/12.))*2.+1.
L1=Z1
L2=Z2
L3=Z3
\[ X_1 = L_1 \]
\[ X_2 = L_2 \]
\[ X_3 = L_3 \]
\[ Y_1 = (\text{BUHPS}(L_1) + (\text{BUHPS}(L_1) - \text{BUHPS}(L_1)) 
\times (Z_1 - X_1)) \times 12 \times SC \]
\[ Y_2 = (\text{BUHPS}(L_2) + (\text{BUHPS}(L_2) - \text{BUHPS}(L_2)) 
\times (Z_2 - X_2)) \times 12 \times SC \]
\[ Y_3 = (\text{BUHPS}(L_3) + (\text{BUHPS}(L_3) - \text{BUHPS}(L_3)) 
\times (Z_3 - X_3)) \times 12 \times SC \]
\[ Y_4 = Y_2 - (Y_3 - Y_2) \times (L_0) / W \]
\[ \text{DYDT} = \text{DERIV}(0 .. Y_4) \]
\[ YC = \text{INTGRL}(SETD, \text{DYCDT}) \]
\[ YCACY = YC - YCADJ \]
\[ YCADJ = \text{INTGRL}(0 .. Y_3, \text{DYCDT}) \]
\[ G_1 = (\text{ROLEH} + \text{YACDT}) \]
\[ \text{ALTER1} = \text{FCNSN}(G_1, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{DYCDT} = \text{ALTER1} + \text{ALTER2} \]
\[ \text{DYCDT} = \text{FCNSN}(YCADJ, \text{ROLEH1, ROLEH2, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3}) \]
\[ \text{ALTER2} = \text{FCNSN}(YCADJ, 0, 0, 0, 0, 0, 0, 0, 0, 0) \]
\[ F_1 = \text{FCNSN}(\text{DYCDT, ROLEH1, ROLEH2, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3, ROLEH3}) \]
\[ Y_0 = Y_1 - Y_4 + YCACY \]
\[ Y = \text{AFGEN}(HOG, Y_0) \]
\[ C_1 = (W / 42, / Y_0 - 30) / 1, 0, 0 \]
\[ C_2 = 0, 1, C_1 + Y_4 \]
\[ \text{HORF} = (C_1 + C_2) \times (\text{SPEED} \times 30 / 32, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{VERF} = \text{AFGEN}(6, 9, 0, 0, 0, 0, 0, 0, 0) \]
\[ V_1 = \text{HORF} \times \text{DYCDT} / (\text{SPEED} \times 12, 0, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{SPED} = \text{FCNSN}(\text{SPEED} - 3, 1, 1, 1, 1, 1, 1, 1, 1, 1) \]
\[ Y = \text{FCNSN}(\text{SPEED} - 3, \text{VERF}, \text{VERF}, \text{VERF}, \text{VERF} - V_1) \]
\[ \text{VERF} = \text{FCNSN}(YCADJ, W, 0, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{DYCDT} = \text{DERIV}(0, 0, Y_0) \]
\[ \text{DIST} = \text{INTGRL}(0, \text{SPEED}) \]
\[ \text{ENAC} = (\text{TORQ} - \text{RENT}) / 100 \]
\[ \text{TC} = \text{DELAY}(1, 0, 1, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{SLIP1} = \text{AFGEN(SLIP1, SLIP1 + 1, 0, 0, 0, 0, 0, 0, 0, 0)} \]
\[ \text{SLIP} = \text{SLIP1} / 100 \]
\[ \text{DRAG} = \text{HORF} \times \text{RAC} \times (1, 1, 1, 1, 1, 1, 1, 1, 1, 1) \]
\[ \text{WR} = (W \times \text{HORF} \times \text{LMT} + 0, 0, 0, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{SPFED} = \text{NSP} \times (1, -\text{SLIP}) \]
\[ \text{TCDT} = \text{HORF} / \text{R} \]
\[ \text{NSP} = \text{NSP} \times \text{R} / \text{GR} / 12 \]
\[ \text{ENSP} = \text{INTGRL}(\text{NSP}, \text{NSP} + 1, 0, 0, 0, 0, 0, 0, 0, 0) \]
\[ \text{ENREV} = \text{NSP} \times \text{R} / 100 \]
\[ \text{TORQ} = \text{AFGEN}(\text{TORQ}, \text{NSP}, \text{REV}) \]
\[
\begin{align*}
R_	ext{EQ} &= \text{DRAG} \times \left( \text{CL} \times \text{EFFC} \times \text{L} \right) \\
I_{HQ} &= \text{MV} \times I_{EF} + \text{P} + \text{P} + \text{P} + \text{P} \times \text{P} \\
R_{HF} &= 2 \times I_{HF} \times \text{SPEED} \times \text{SPEED} + 144 \times (\text{RF} + \text{ENSP}) + 2 \\
\text{MV} &= (\text{NT} + \text{PLMT}) \times \text{SPEED} \times \text{SPEED} / (\text{G} + \text{ENSP} + \text{ENSP}) \\
\text{ROLOW} &= \text{FCHS} \times (\text{C} \times \text{AB} \times \text{CC}) \\
\text{AB} &= 8 \times (\text{A} \times \text{Y}) \times \text{SPEED} + 12 \\
\text{CC} &= \text{C} \times \text{SPEED} + 12 \\
\text{ROLIF} &= (\text{MAXROL} \times \text{MAXSP}) + \text{C} \times \text{REV} \\
\text{SET} &= \text{HSET} - \text{MODE} \times \text{VSET} \\
\text{HSET} &= \text{AFCEN} (\text{HOC, SETD}) \\
\text{VSET} &= \text{AFCEN} (\text{YDC, SETD}) \\
\text{SIGNAL} &= (\text{HORF} + \text{HOVE} \times \text{VERF}) - \text{SET} \\
\text{AI} &= \text{DEADS} \times (\text{D}, \text{D}, \text{SIGNAL}) \\
\text{ROLF} &= -\text{PROPUP} \times \text{A1} \\
\text{ROLF} &= -\text{PROPUP} \times \text{A1} \\
\text{ROLP} &= -\text{LIM} \times (\text{ROLF}, \text{C}, \text{U} \times \text{U}) \\
\text{ROLP} &= -\text{LIM} \times (\text{SAXL}, \text{R}, \text{U} \times \text{U}) \\
\text{PROCEDURE VALVE} &= \text{CONS}(\text{A1}) \\
\text{IF}(\text{KEEP, NE} = 1) &= \text{GO TO 220} \\
\text{IF}(\text{A1}23, 22, 21) &= \text{GO TO 210} \\
\text{IF}(\text{D} \times (\text{J}) \times \text{LEN}) &= \text{GO TO 2} \\
\text{6 VALVE} &= 1 \\
\text{GO TO 210} \\
\text{23 DO 4 J:} &= \text{IDOUP} \\
\text{5 IF}(\text{D} \times (\text{DD}) = \text{GE} . \text{U}) &= \text{GO TO 22} \\
\text{7 VALVE} &= 1 \\
\text{GO TO 210} \\
\text{22 IF}(\text{IF} \times \text{VALVE} = \text{EQ} . 1 \text{AND} . \text{DD} \times (\text{IDOUP}) \times \text{GT} . \text{U}) &= \text{GO TO 6} \\
\text{IF}(\text{VALVE} = \text{EQ} . 1 \text{AND} . \text{DD} \times (\text{IDOUP}) \times \text{LT} \text{U}) &= \text{GO TO 7} \\
\text{VALVE} &= 0 \\
\text{210 CONTINUE} \\
\text{DO 20 J = 1, 11} \\
\text{K} &= 13 - \text{J} \\
\text{20 DD} \times (\text{K}) &= \text{D} \times (\text{K} - 1) \\
\text{DD} \times (1) &= \text{A1} \\
\text{220 CONTINUE} \\
\text{ENDD} \\
\text{PROCEDURE J, SUM, SUM, SUM} &= \text{LIF} \times (\text{HORF, SLIP}) \\
\text{IF}(\text{TIME, NE} \times 0, 0) &= \text{GO TO 230} \\
\text{CT} &= 0.0
\end{align*}
\]
SUMX=0.0
SUMX2=0.0
SUMSLP=0.
DO 9 J=1,20
9 HISTO(J)=J
GO TO 777
888 J=HURF/250.+1.
HISTO(J)=HISTO(J)+1
SUMX=SUMX+HURF
SUMX2=SUMX2+HURF**2
SUMSLP=SLIP+SUMSLP
CT=CT+1.
777 CONTINUE
ENDPRO
TERMINAL
IF(ENSPE. LE. 0.)WRITE(6,111)
IF(SLIP. GE. .95.)WRITE(6,112)
111 FORMAT(' TRACTOR STALL DUE TO ENGINE STALL')
112 FORMAT(' TRACTOR STALL -100% WHEELSLIP')
SD=SRT((SUMX2-SUMX*SUMX/CT)/(CT-1.))
AV=SUMX/CT
AVSLP=SUMSLP/CT+100.
AVHP=AV*AVSLP/550.
ACRE=42.*AVHP+360.j./108.(/140.
WRITE(6,101)GEAR,SETO
IF(MODE.EQ.0.)GO TO 10
WRITE(6,102)
GO TO 11
10 WRITE (6,103)
11 CONTINUE
WRITE(6,104)DB.DELUP
WRITE(6,105)XAPOL
WRITE(6,106)DIST
WRITE(6,107)SD
WRITE(6,108)AV
WRITE(6,109)
WRITE(6,110)AVSLP. AVSP, ACRF
101 FORMAT(1HO,'GEAR',.9X,11. 'SET DEPTH',.F4.1. 'INS'))
102 FORMAT('MODE',.9X,'TOP LINK SENSING')
103 FORMAT('MODE',.9X,'PUPE DRAINAGE SENSING')
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END DATA
STOP
APPENDIX IIb

This output was obtained by running the programme and input data in

APPENDIX IIa.

***SIMULATION HALTED***  DIST = 3.0005E 02

GEAR  3
SET DEPTH  7.5 INS
MODE  TOP LINK SENSING
DEADBAND  400.1 LB
DELAY  2.15 SEC
MAX R.O.L.  12.3 INS/SEC

DISTANCE PLOUGHED  300. FT
STANDARD DEVIATION OF DRAUGHT  129. LB
MEAN DRAUGHT  2182. LB
AVERAGE SPEED  AVERAGE SLIP  WORK RATE
(MILE/H) (PERCENT) (ACRES/H)
3.49  6.8  1.04

***SIMULATION HALTED***  DIST = 3.0005E 02

GEAR  4
SET DEPTH  7.5 INS
MODE  TOP LINK SENSING
DEADBAND  400.1 LB
DELAY  2.15 SEC
MAX R.O.L.  12.3 INS/SEC

DISTANCE PLOUGHED  300. FT
STANDARD DEVIATION OF DRAUGHT  2182. LB
MEAN DRAUGHT  2182. LB
AVERAGE SPEED  AVERAGE SLIP  WORK RATE
(MILE/H) (PERCENT) (ACRES/H)
4.84  7.4  1.40

***SIMULATION HALTED***  DIST = 3.0002E 02

GEAR  5
SET DEPTH  7.5 INS
MODE  TOP LINK SENSING
DEADBAND  400.1 LB
DELAY  2.15 SEC
MAX R.O.L.  12.3 INS/SEC

DISTANCE PLOUGHED  300. FT
STANDARD DEVIATION OF DRAUGHT  272. LB
MEAN DRAUGHT  2315. LB
AVERAGE SPEED  AVERAGE SLIP  WORK RATE
(MILE/H) (PERCENT) (ACRES/H)
5.79  7.3  1.67

STOP
APPENDIX II

1) The programme

C. CALCULATION OF CONSTANTS IN RATE OF ENTRY EQUATION

DIMENSION A(2), B(2), C(2), D(2), E(2), F(2), G(2),
* C0(2), O0(2), F0(2), TITLE(18), H(2, 20), V(2, 20)
COMMON H, V, NPTS, NPTSV
DATA PI/3.14159/
COSY(S, T, U) = (S*S + T*T - U*U) / (2*S*T)
ANGLE(R) = ATAN((I + R**2) * R)
TAN(Z) = SIN(Z) / COS(Z)
READ(5, 10) TITLE
READ(5, 10) NPTS, NPTSV
READ(5, 10) (H(I, J), I = 1, 2, J = 1, NPTSV)
READ(5, 10) (V(I, J), I = 1, 2, J = 1, NPTSV)
10 FORMAT(18A4)
11 FORMAT(210)
12 FORMAT(10F0, 0)
WRITE(6, 12) TITLE
READ(S, 11) /1
READ(S, 14) A, F(2)
14 FORMAT(11A4)
12 FORMAT(11A4)
D(2) = 22.7 - F(2)
F(2) = F(2) + 21.2
D(1) = A(1) + SQRT(A(2) - D(2))**2
C(1) = D(1)
BC = HYPOT(B, C)
F(1) = F(1) + 50.
F(2) = F(2)
CF = HYPOT(C, F)
DF = HYPOT(D, F)
COSDF = COSY(CD, DF, CF)
CDF = ANGLE(-COSDF)
CDF = PI - CDF
W = ATAN((A2) - D(2)) / (D(1) - A(1))
999 I = 1, 1000
W = W + 0.001
D(1) = A(1) + AD * COS(W)
D(2) = A(2) - AD * SIN(W)
BD = HYPOT(B, D)
BC = HYPOT(3, C)
COSADB = COSY(AD, BD, AB)
COSDBC = COSY(BD, BC, 21.2)
G1 = TAN(W)
G2 = TAN(ANGLE(COSDBC) - ANGLE(COSADB) - W)
C2 = B(2) - G2 + B(1)
C1 = A(2) - G1 + A(1)
E(1) = (C2 - C1) / (G1 - G2)
E(2) = G1 + E(1) + C1
COSDBC = COSY(3B, CD, BC)
FDV = 3*PI / 2. - W - ANGLE(COSADB) - ANGLE(COSDBC) - CDF
F(1) = D(1) + DF * SIN(FDV)
F(2) = D(2) - DF * COS(FDV)
CALL VALUE(-F(2), HORF, VERF)
FORCE = SQRT(HORF*HORF + VERF*VERF)
COSF = VERF/HORF
C3 = F(2) - COSF * F(1)
TEST = COSF * E(1) + C3
999 IF( TEST.LE.E(2)) GO TO 200  
STOP
200 AA = -F(2)  
BB = 0.45/AA  
WRITE(6,201) AA, BB
201 FORMAT(' CONSTANTS IN RATE OF ENTRY EQUATION, Y=A(1-EXP(-B*X))',/ ' A=',F5.1, ' B=',F5.3)
STOP
END
FUNCTION HYPOT(Y,Z)
DIMENSION Y(2),Z(2)
HYPOT=SQRT((Y(1)-Z(1)**2+(Y(2)-Z(2))**2)
RETURN
END
SUBROUTINE VALUE(X,HX,VX)
DIMENSION H(2,20),V(2,20)
COMMON H,V,NPTSH,NPTS V
DO 99 J=1,NPTSV
IF(H(1,J).GE.X) GO TO 98
99 CONTINUE
98 GRADH=(Z(2,J)-Z(2,J-1))/(H(1,J)-H(1,J-1))
HX=H(2,J-1)+GRADH*(X-H(1,J-1))
DO 97 J=1,NPTSV
IF(V(1,J).GE.X) GO TO 96
97 CONTINUE
96 GRADV=(V(2,J)-V(2,J-1))/(V(1,J)-V(1,J-1))
VX=V(2,J-1)+GRADV*(X-V(1,J-1))
RETURN
END

2) Typical input (See Fig.6)

LEYLAND 270

\begin{tabular}{cccccc}
8 & 9 & 10 & 11 & 12 & 13 \\
0 & 0 & 8 & 2000 & 9 & 2280 \\
11 & 3350 & 11 & 2 & 4500 & 12 & 6000 \\
0 & 1200 & 2 & 1340 & 4 & 1440 \\
9 & 1430 & 10 & 1320 & 11 & 1300 & 12 & 750 \\
8 & 3 & 21 & (t_x,t_y) \\
18.5 & 37.3 & (t_x,t_y) \\
33 & 9 & (t, S_{10}) \\
\end{tabular}

3) Typical output.

LEYLAND 270
CONSTANTS IN RATE OF ENTRY EQUATION, Y=A(1-EXP(-B*X))

A = 11.1 B = 0.041
1) The programme

```
DIMENSION TITLE(15)
INTEGER TITLE
READ(5,10)
10 FORMAT(1X)
DO 6 K=1,10
READ(5,15)
15 FORMAT(15X)
WRITE(6,12)
12 FORMAT(/15X,1S)
READ(5,15)
1 FORMAT(4F9.0)
TC1=PULL1/UT
TC2=PULL2/UT
AN=TC2
DO 2 J=1,200
A=AN
B=LOG(A/(A-TC1))/SLIP1
AN=TC2/(1-EXP(-B+28))
TEST=AN+(1-EXP(-B+SLIP1))
2 IF(CABS(TEST-TC1).LT.0.01)GO TO 4
ADIFF=AN-4
WRITE(6,3)ADIFF,AN,3
3 FORMAT(* INSUFFICIENT CONVERGENCE AFTER 200 ITERATIONS*,
2 DIFFERENCE BETWEEN AN AND A =',F3.4,
1 A=',F3.4,' \ & 3=',F3.4,' AFTER',15,' ITERATIONS')
LL=20.*A
IF(LL.GT.20)LL=20
WRITE(6,23)
DO 21 L=1,LL
X=FLOAT(L)/20.
Y=LOG(A/(A-X))/B
IF(Y.GE.1.0)GO TO 24
XL=X
21 IF(L.LE.LL.AND.FLOAT(L/2).GE.XL/2.)WRITE(6,22)X,Y
24 XL=XL+L
WRITE(6,22)XL,Y
22 FORMAT(* \ & 13=',F3.4,13=',F3.4,1)\ & COORDINATES IN EQUATION \ & \ & SLIP=LOG(C/(C-TC))/K',
23 FORMAT(* \ & 13=',F3.4,13=',F3.4,' \ & \ & \ & \ & \ & AFTER',15,' ITERATIONS')
CONTINUE
STOP
END
```

2) Typical input.

```
16.9/14.30 TIRE, 4410.43.  LOAD, 15 PSI, GOOD CONDITIONS
4410. 3390. 1730. 18.
AVERAGE CONDITIONS
POOR CONDITIONS
4410. 1820. 15. 15.
BAD CONDITIONS
4410. 1370. 13. 15.
```
3) Typical output (See Fig. 2.41):

16.9/16-39 TYPE, 441 A/F. LOAD, 15 PSI, GOOD CONDITIONS

CONSTANTS IN EQUATION  \( \text{SLIP} = \log\left(\frac{C}{(C-TC)}\right)/K \)

\( C = 2.6903 \)  \( K = 0.0163 \)  AFTER 39 ITERATIONS

(TC,SLIP) POINTS

(0.10, 2.3)  
(0.20, 4.7)  
(0.30, 7.3)  
(0.40, 9.9)  
(0.50, 12.6)  
(0.60, 15.5)  
(0.70, 18.5)  
(0.80, 21.7)  
(0.90, 25.0)  
(1.00, 28.5)

AVERAGE CONDITIONS

CONSTANTS IN EQUATION  \( \text{SLIP} = \log\left(\frac{C}{(C-TC)}\right)/K \)

\( C = 0.6207 \)  \( K = 0.0676 \)  AFTER 12 ITERATIONS

(TC,SLIP) POINTS

(0.10, 2.5)  
(0.20, 5.5)  
(0.30, 9.4)  
(0.40, 14.7)  
(0.50, 23.2)  
(0.60, 48.7)  
(0.62, 100.0)

POOR CONDITIONS

CONSTANTS IN EQUATION  \( \text{SLIP} = \log\left(\frac{C}{(C-TC)}\right)/K \)

\( C = 0.6157 \)  \( K = 0.0555 \)  AFTER 19 ITERATIONS

(TC,SLIP) POINTS

(0.10, 3.2)  
(0.20, 7.1)  
(0.30, 12.0)  
(0.40, 18.0)  
(0.50, 30.1)  
(0.60, 66.2)  
(0.61, 100.0)

BAD CONDITIONS

CONSTANTS IN EQUATION  \( \text{SLIP} = \log\left(\frac{C}{(C-TC)}\right)/K \)

\( C = 0.4374 \)  \( K = 0.0617 \)  AFTER 39 ITERATIONS

(TC,SLIP) POINTS

(0.10, 4.2)  
(0.20, 9.9)  
(0.30, 16.7)  
(0.40, 39.7)  
(0.44, 100.0)
Fig 2.41 Predicted Slip v. Cot Curves (Appendix IIa) for 16-9/14-30 Tyre on Four Surfaces
APPENDIX 3.I Review of methods of analysis for non-linear controls

Phase-plane

This is a graphical technique of plotting $\frac{dx}{dt}$ against $x$, where $x$ is some function of time (117). No approximations need be made but the method has several limitations:

a) the linear part of the system must be capable of being described by a 2nd order differential equation,

b) the input may only be in the form of initial conditions, i.e. step and ramp inputs are allowable, but sinusoidal or random inputs are not,

c) it cannot be used for time-varying parameters.

This third limitation means for example that it cannot be used for a pure time delay, whose transfer function is

$$G(s) = e^{-t_d s}$$

The method can however be used for systems where the coefficients are dependent on the amplitude, or a derivative of the amplitude, of the signal at any point.

To consider the control shown in fig. 3.50 for example, the equations of the system are

$$\dot{E} = R - C$$

(3.56)

$$x_s = E \quad \text{Saturated region}$$

(3.57)

$$x_s = k_p \left( E - \frac{DB}{2} \right) \quad \text{Proportional region}$$

$$x_s = 0 \quad \text{Deadband region}$$

(3.58)

A system equation can then be written for each region. In the saturated region, assuming that the input, $R$, is zero initially,

$$\frac{d^2 E}{dt^2} + \frac{1}{T_d} \frac{dE}{dt} = \frac{k_s E}{T_d}$$

(3.59)

Letting $x = E, y = \dot{E}$, this becomes

$$\frac{dy}{dt} + \frac{y}{T_d} = \frac{k_s E}{T_d}$$

(3.60)

Dividing by $\frac{dx}{dt} = y$ and letting $\frac{dy}{dx} = N$

$$N + \frac{1}{T_d} = \frac{k_s E}{T_d y}$$

(3.61)
Fig 3.50 Phase plane portrait for control shown above
which is the ISOCLINAL EQUATION. For constant values of the gradient \( N \), lines can be drawn on a graph of \( y \) against \( x \). In this case for the saturated region, they are horizontal lines. The value of the gradient is marked on each line and by repeating the process for each zone the phase portrait is constructed. Then from a starting point on the portrait, say the point \((0, X)\) to represent a step input of \( X \) the trajectory can be sketched in using the indications of the gradient to estimate its path which will settle at a singular point if the system is stable or continue in a closed loop if it is unstable. This form of instability is a limit cycle oscillation.

**Describing function**

A sinusoidal input to a non-linear device gives a distorted sine wave output which can be described mathematically by a Fourier series. Assuming the fundamental frequency is not altered by the non-linearity, the Fourier series is a transfer function for the non-linearity, describing the ratio of the fundamental component of the output to the input (117).

For example, a sinusoidal input to the proportional control shown below would give the distorted output shown

\[
y = \frac{k_s E}{T_d N + 1}
\]

(3.62)

The limitations of this method are that:

a) the system must be time-invariant, \( G_N(j\omega) \) cannot be a function of time

b) \( G_N(j\omega) \) is the only non-linear element

The second limitation is necessary because the input to the second non-linearity would already be a distorted sine wave. This prevents the application of the describing function technique to the top link
To analyse control stability consider the non-linear control
below, where \( G(j\omega) \) represents the linear portion.

\[
\frac{C(j\omega)}{R(j\omega)} = \frac{G_N(j\omega) G(j\omega)}{1 + G_N(j\omega) G(j\omega)}
\]  
(3.63)

Using a Nyquist diagram, stability is dependent upon the equation

\[ 1 + G_N(j\omega) G(j\omega) = 0 \]  
(3.64)

\[ -G_N(j\omega) = \frac{1}{G(j\omega)} \]  
(3.65)

For stability, the locus of \( \frac{1}{G(j\omega)} \) must pass outside the point given by
\(-G_N(j\omega)\). This is similar to the linear case where \( \frac{1}{G(j\omega)} \) must pass outside the point \((-1,0)\).

One advantage of the describing function method is that the pure
time delay \( e^{-t_d j\omega} \) can be included. Having plotted \( \frac{1}{G(j\omega)} \), its effect
is to alter the phase shift by \( \omega t_d \) but not to alter the gain.

**Matrix techniques**

An nth order system can be described by n system variables (state
variables) in n first order equations. These variables are usually
the output, \( x \), and its derivatives, but need not necessarily be. The
equations arranged in matrix form lend themselves to being solved by
digital computer and the method is therefore very useful for high order
systems. Non-linear effects can be included, effectively by changing
the matrix elements during the computer solution.

Although it is undoubtedly possible to solve non-linear problems
by this method, few attempts have been made because of the difficulties
in computing. The increasing sophistication of digital simulation
languages has probably attracted the non-linear problems which may have
been solved using state variables.

In fact, for the draught control problem which is only a second
order equation, the state variables have already been plotted on the
phase-plane because $\dot{e}$ and $e$ are state variables of the system.

Simulation

(i) Analogue computer

The advantages of an analogue computer for investigating simple control circuits are that it is fairly cheap and simple and the available accuracy of about $\pm 1\%$ is usually sufficient for most physical problems. Also it has the advantage for the operator that because he is involved in setting up and running the simulated problem he tends to understand it better and get a 'feel' for what physical processes are occurring.

The disadvantages of analogue computer simulation are that accuracy tends to deteriorate with problem size and it is difficult or complicated to simulate some non-linearities. Non-continuous functions for example are difficult to simulate accurately without rounding off occurring although sometimes this may not be a disadvantage if the process being simulated is subject to rounding.

(ii) Digital computer

With the increasing number and size of digital computers available, several programmes have been written to simulate the continuous dynamic problems which would have previously been solved on an analogue computer. They are typically very simple to use and the input required is merely the basic problem equations.

The advantages of these digital simulation languages (e.g. CSMP, SLAM, MIDAS, DAS, DYNASAR) are that they are simple to use and accurate for many problems to about $\pm 0.01\%$. Their disadvantages are the high cost and a longer turn around time than for the analogue computer.

The particular language used in this work was CSMP (Continuous System Modelling Program) written by IBM. The reason for using this language was that it was readily available and could be used on the ICL 4/70 computer belonging to the Agricultural Research Council at Rothamsted. The main problem found for the particular simulations in this work were those of attempting to integrate a discontinuous function, say the output from an on-off device. It is easy to specify
an integration which the mathematical integration routines, on which these languages are really based, cannot perform. This may happen because the programming language is simple and no error or warning is given that something may be wrong. The onus for checking these problems is with the operator who must be aware of the limitations of the numerical methods used for integration particularly. The problems can usually be overcome or minimised to acceptable levels careful selection of the step size and integration method used.
APPENDIX 3.II Analogue computer circuits

Pure draught sensing

The circuit is shown in fig 3.51 and it is scaled as follows;
(since the scaling was originally done in Imperial units it is shown here unchanged).

Maximum values of variables

\[ \dot{y}_o = 20 \text{ in/s (} \approx 500 \text{ mm/s)} \]
\[ x_s = 0.1 \text{ in (} \approx 25 \text{ mm)} \]
\[ H = 250 \text{ lb (} \approx 1125 \text{ N)} \]

Integrator 1

\[ \frac{-d}{dt} \left[ \frac{\dot{y}_o}{2} \right] = -20 \left[ \frac{\dot{y}_o}{2} \right] + \frac{200 \times 20}{2 \times 100} \left[ 100 x_s \right] \]

Time scale by \( \frac{1}{\beta} = \frac{1}{10} \)

\[ -0.1 \frac{d}{dt} \left[ \frac{\dot{y}_o}{2} \right] = -(0.2) 10 \left[ \frac{\dot{y}_o}{2} \right] + (0.2) 10 \left[ 100 x_s \right] \]

Integrator 2

\[ \frac{-d}{dt} \left[ \frac{H}{25} \right] = -250 \times 2 \left[ \frac{\dot{y}_o}{2} \right] \]

Time scale

\[ -0.1 \frac{d}{dt} \left[ \frac{H}{25} \right] = -(0.2) 10 \left[ \frac{\dot{y}_o}{2} \right] \]

Top link sensing

The circuit is shown in fig 3.52 and it is scaled as follows;

Maximum values of variables

\[ y_o = 20 \text{ in/s (} \approx 500 \text{ mm/s)} \]
\[ x_s = 0.1 \text{ in (} \approx 25 \text{ mm)} \]
\[ H = 250 \text{ lb (} \approx 1125 \text{ N)} \]
\[ H_{\text{mean}} = 2500 \text{ lb (} \approx 11250 \text{ N)} \]
\[ P = 2082 \text{ lb (} \approx 9300 \text{ N)} \]

Integrator 1

\[ \frac{-d}{dt} \left[ \frac{\dot{y}_o}{2} \right] = -20 \left[ \frac{\dot{y}_o}{2} \right] + \frac{200 \times 20}{2 \times 50} \left[ 50 x_s \right] \]

Time scale by \( \frac{1}{\beta} = \frac{1}{10} \)

\[ -0.1 \frac{d}{dt} \left[ \frac{\dot{y}_o}{2} \right] = -(0.2) 10 \left[ \frac{\dot{y}_o}{2} \right] - (0.4) 10 \left[ 50 x_s \right] \]
Fig 3.51 Analogue computer circuit for proportional control - pure draught sensing
Fig 3.52 Analogue computer circuit for on-off control-top link sensing
Integrator 2
\[-\frac{\text{d}}{\text{dt}} \left[ \frac{H}{25} \right] = -\frac{250 \times 2}{25} \left[ \frac{\dot{y}_0}{2} \right] \]

Time scale
\[-0.1 \frac{\text{d}}{\text{dt}} \left[ \frac{H}{25} \right] = -(0.2) 10 \left[ \frac{\dot{y}_0}{2} \right] \]

Amplifier 3
\[-\left[ 10 k_2 \dot{y}_0 \right] = 0.033 \times 10 \times 2 \left[ \frac{\dot{y}_0}{2} \right] \]
\[-\left[ 10 k_2 \ddot{y}_0 \right] = 0.66 \left[ \frac{\ddot{y}_0}{2} \right] \]

Amplifier 4
\[-\left[ \frac{F_T}{200} \right] = 25 \left[ \frac{H}{25} \right] + 25 \left[ \frac{H k_2 \dot{y}_0}{25} \right] + 2 \times 2500 \times 0.033 \left[ \frac{\dot{y}_0}{2} \right] \]
\[-\left[ \frac{F_T}{200} \right] = 0.125 \left[ \frac{H}{25} \right] + 0.125 \left[ \frac{H k_2 \dot{y}_0}{25} \right] + 0.733 \left[ \frac{\ddot{y}_0}{2} \right] \]

Non-linearity
The proportional control was simulated using the function generating facility on the PACE 20 computer. Some rounding occurred at the breakpoints but it was not considered sufficient to cause any significant inaccuracy since there was probably a small amount in practice. The on-off control could not be simulated on the function generator and the best circuit was found to be that shown in fig 3.53. This produced very good breakpoints and the gradient of the response which should ideally be infinity was about 20 v/v.
Fig 3.53 Circuit used to simulate an on-off response on analogue computer
APPENDIX 3.III  C.S.M.P. computer programmes

(1) Pure draught sensing control

* PURE DRAUGHT SENSING CONTROL
PARAMETER K1=44*KP=(.0056,10.*),DB=900.
PARAMETER KS=900,*XSMAX=2.5
PARAMETER HSET=11250,*YIN=300.
E=HSET-H
D=DB/P.
XS1=DEADSP(-D,D,E)
XS2=KP*XS1
XS=LIMIT(-XSMAX,XSMAX,XS2)
DYCDT1=KS*XS
DYCDT2=DELAY(0.004,DYCDT1)
DYCDT=REALPLC(0.004,DYCDT2)
YC=INT:LC0.0,04.0,YDCDT)
YO=YIN+YC
H=KI*YO
METHOD RECT
PREPARE TIME,H
TIMER' FINTIM=1.,DELT=.001,OUTDEL=.01
END
STOP
ENDJOB

(2) Top link sensing control

* TOP LINK SENSING CONTROL
PARAMETER K1=44*KP=(.0056,10.*),DB=900.
PARAMETER KS=900,*XSMAX=2.5,KP=.0013
PARAMETER FTSET=11250,*YIN=300.
E=FTSET-H-H*DYCDT*K2
D=DB/P.
XS1=DEADSP(-D,D,E)
XS2=KP*XS1
XS=LIMIT(-XSMAX,XSMAX,XS2)
DYCDT1=KS*XS
DYCDT2=DELAY(0.004,DYCDT1)
DYCDT=REALPLC(0.004,DYCDT2)
YC=INT:LC0.0,04.0,YDCDT)
YO=YIN+YC
DYCDT=DELAY0.0,YO)
H=KI*YO
METHOD RECT
PREPARE TIME,H
TIMER' FINTIM=1.,DELT=.001,OUTDEL=.01
END
STOP
ENDJOB
APPENDIX 3.IV Calculation of approximate time delay due to linkage compliance with mounted plough suspended on the linkage.

The linkage is simplified to that shown in fig 3.54 where the plough is assumed to be supported by a spring and damper. Also the following assumptions are made:

a) the spring and damper act at right angles to the lower links,

b) the position of the instantaneous centre of rotation does not change significantly for small movements of the linkage,

c) the effect of the top link sensing spring can be ignored because the plough weight causes sufficient tensile force in the top link for the movement of the sensing unit to have reached its limiting position which is built in as a mechanical stop.

The spring rate \( k_a \) can be calculated from the spring rate measured at the lower link ends (fig 3.33). From fig 3.54,

\[
k_a = 387 \left( \frac{0.88}{0.5} \right)^2
\]

\[
k_a = 681 \text{ N/mm}
\]

When the plough is lifted, a step force is applied to the piston in the hydraulic lift cylinder which has little mass and so its acceleration can be considered infinite. This therefore is equivalent to applying a velocity \( \dot{x}_1 \) as a step function or a displacement \( x_1 \) as a ramp function.

The equation of motion of the plough is

\[
m \ddot{x}_p \left( \frac{6.08}{5.08} \times \frac{0.69}{0.5} \right) = -k_a (x_1 - x_p) - c_a (\dot{x}_1 - \dot{x}_p)
\]

(3.66)

and

\[
x_p = \left( \frac{6.08}{5.08} \times \frac{0.69}{0.5} \right) x_a = 2.09 x_a
\]

(3.67)

so that equation 3.66 can be written

\[
m \ddot{x}_a = -k_a' (x_a - x_1) - c_a' (\dot{x}_a - \dot{x}_1)
\]

(3.69)

where

\[
k_a' = \frac{k_a}{2.09^2} \text{ and } c_a' = \frac{c_a}{2.09^2}
\]

Re-arranging equation 3.69

\[
x_a + \frac{c_a'}{m} \dot{x}_a + \frac{k_a'}{m} x_a = \frac{c_a'}{m} \dot{x}_1 + \frac{k_a'}{m} x_1
\]

(3.70)
All dimensions in metres

Fig 3.54 Linkage and plough dimensions for experimental control when measuring time delays
Taking Laplace transforms and letting natural frequency, \( \omega_n^2 = \frac{k'_a}{m} \)
and \( 2\nu_a \omega_n = \frac{\nu_a}{m} \) where \( \nu_a = \) damping ratio

\[
x_a(s) (s^2 + 2\nu_a \omega_n s + \omega_n^2) = x_i(s) (2\nu_a \omega_n s + \nu_a^2).
\]

For the input \( x_i(s) = \frac{A_0}{s^2} \),

the solution to equation 3.71 is

\[
x_a(t) = \frac{A_0}{w_d} e^{-\nu_a \omega_n t} \sin w_d t + A_o t
\]

where damped natural frequency \( w_d = \omega_n (1 - \nu_a^2) \).

Differentiating equation 3.72 gives

\[
\dot{x}_a(t) = -A_o e^{-\nu_a \omega_n t} \cos w_d t - \frac{\nu_a \omega_n}{w_d} \sin w_d t + A_o
\]

The time for \( x_a(t) \) to reach the input velocity \( A_o \) is given by

\[
A_o = -A_o e^{-\nu_a \omega_n t} \cos w_d t - \frac{\nu_a \omega_n}{w_d} \sin w_d t + A_o
\]

Therefore either \( e^{-\nu_a \omega_n t} = 0 \) giving \( t = \infty \)

or \( \tan w_d t = \frac{w_d}{\nu_a \omega_n} \).

For a plough mass of 545 kg,

\( \omega_n = 23.1 \text{ rad/s} \)

and assuming \( \nu_a = 0.5 \) then

\( w_d = 20 \text{ rad/s} \)

Substituting these into equation 3.76 gives an approximate time delay,

0.05 s, that is the time required for \( x_a \) to reach the input velocity.

Using a similar analysis for the commercial control which has a different hydraulic stiffness gives an approximate delay of 0.03 s.

Calculation of compliance at lower link ends due to oil compressibility

The compressibility of hydraulic oil for the majority of normal applications is 0.5% by volume per 1000 lb/in\(^2\) pressure (131). This figure includes an allowance for expansion of oil-ways and pipes.

The volume of oil in the hydraulic circuit of the experimental implement control between the rams and the valve can be calculated from the following dimensions.
2 cylinders 5" long 1 3/4" Dia
2 pipes 24" long 3/8" Dia
1 pipe 108" long 1/2" Dia
Total volume = \pi(2 \times 5 \times (7/8)^2) + 2 \times 24 \times (3/16)^2 + 108 \times (1/4)^2
= 50.5 \text{ in}^3

The compressibility \( k_o \) of the oil is therefore

\[ k_o = \frac{1000}{252} \left( \frac{\text{lb/in}^2}{\text{in}^3} \right) \]

\[ = \frac{1000}{252} \times \pi \times 2 \times (7/8)^2 \left( \frac{\text{lb/in}^2}{\text{in}^3} \right) / \text{in movement of lift rods} \]

\[ = \frac{1000}{252} \times \pi \times 2 \times (7/8)^2 \times 2 \times (7/8)^2 \text{ lb/in. movement of lift rods} \]

Referring this compressibility to the ends of the lower links gives

\[ k_o = \frac{1000}{252} \times \pi \times 2 \times (7/8)^2 \times 2 \times (7/8)^2 \times \left( \frac{20}{25} \right)^2 \]

\[ = 30,000 \text{ lb/in.} \]

\[ = 5.8 \times 10^6 \text{ N/m} \]
APPENDIX 3.1 Details of top link damper and calculation of the increase in time delay due to damper, when measured on laboratory test rig

Details of top link damper

2 Kinetrol KD A4 DD dampers used, each of which gave a damping coefficient adjustable between 29 and 294 kNms/rad. With an arm length of 50 mm and a lever ratio of 3.5:1 the damping coefficient acting at the top link sensing unit was 278 to 2780 kNms/m.

The mass of the sensing unit was approximately 2.1 kg.

The stiffness of the top link spring was 1885 kN/m.

Response of top link sensing unit to a step force input, \( f_0 \), applied by the test rig

The equation of motion of the spool valve is

\[
\ddot{x}_b + c_b \dot{x}_b + k_b x_b = f_0
\]  

(3.77)

Since \( m_b = 2.1 \) kg

\( c_b = 278 \) kNms/m

and \( k_b = 1885000 \) N/m

the first term in equation 3.77 can be ignored and the solution is then

\[
x_b(t) = \frac{f_0}{k_b} \left(1 - e^{-k_b/c_b t}\right)
\]

(3.78)

Assuming that the initial displacement of the sensing unit is zero, i.e. in the middle of its deadband, then the control delay will be increased by the time taken for the control valve to exceed the deadband. This corresponds to a distance of \( \frac{DB}{2k_b} \). Substituting into the solution of the equation gives

\[
\frac{DB}{2k_b} = \frac{f_0}{k_b} \left(1 - e^{-\frac{k_b t}{c_b}}\right)
\]

(3.79)
Re-arranging to find the time taken to move $\frac{DB}{2k_b}$

$$t_b = \frac{c_b}{k_b} \log_e \left( \frac{f_o}{f_o - DB} \right)$$

(3.80)

To this must be added the delay $t_d$ inherent in the control to give an overall delay time of $(t_b + t_d)$s.

If the initial displacement is not zero, but some value say $\frac{f_i}{k_b}$

then the solution is slightly different

$$x = \frac{f_o}{k_b} (1 - e^{-\frac{k_b t}{c_b}}) - \frac{f_i}{k_b} e^{-\frac{k_b t}{c_b}}$$

(3.81)

and the time taken to reach a displacement $\frac{DB}{2k_a}$ is given by

$$\frac{DB}{2k_b} = \frac{f_o}{k_b} e^{-\frac{k_b t}{c_b}} \left[ \frac{f_o}{k_b} + \frac{f_i}{k_b} \right]$$

(3.82)

$$t_b = \frac{c_b}{k_b} \log_e \left[ \frac{f_o + f_i}{f_o - DB/2} \right]$$

(3.83)
APPENDIX 4.I  C.S.M.F. computer programme for torque sensing control

* TORQUE SENSING CONTROL
PARAMETER TSET=15150, KD=0.0188, K1=4.4, KFKF1=10.04
PARAMETER Hk=3110, T3=99, IEC=13.5, R=0.71
PARAMETER D=10.9, KP=(46, 100), KS=900, XSMAX=2.5
PARAMETER HDIST=2250, YIN=300
XD=KD*(TSET-KFKF1*YO-TD)
XS1=DEADSP(-D, D, XD)
XS2=KP*XS1
XS=LIMIT(-XSMAX, XSMAX, XS2)
DYCDT1=K5*XS
DYCDT=DELAY(50, 0.05, DYCDT1)
YC=INTQRL(0, DYCDT)
YO=YIN+YC
H=MI*YO+HDIST
ENAC=I-(H+RR)*R/3*.95)/IEQ
ENSP=INTQRL(243, 4, ENAC)
ENREV=ENSP/.1046
T=AFGEN(TOROSP, ENREV)
TD=T*3
FUNCTION TOROSP=(1200, 243), (1400, 246), (1600, 247),...
(1800, 240), (2000, 234), (2200, 220), (250, 220),...
(2300, 182), (2380, 0)
TIMER FINIT=2, DELT=.001, OUTDEL=.001, PRDEL=.01
METHOD RECT
PREPARE TIME, H
PRINT XD, DYCDT, H, ENAC, ENSP
END
STOP
ENDJOB