The Institution of Mechanical Engineers

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AGRICULTURAL AND ALLIED INDUSTRIAL TRACTORS

A Conference arranged by the Automobile Division of the Institution of Mechanical Engineers, in conjuction with the Institution of Agricultural Engineers

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Paper 1

SPECIAL REQUIREMENTS FOR TRACTORS IN JAPAN

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Walking type tractors have played a leading role in Japanese agriculture for the past 20 years, but owing to a high degree of economic growth in Japan the riding type of tractor is now being extensively introduced. These tractors require special characteristics to suit the agricultural circumstances of Japan. In this paper these requirements are presented and discussed.

INTRODUCTION

JAPAN CONSISTS of four major islands and approximately a hundred minor ones which lie from north to south along the east coast of the Asiatic Continent. The northern part of Japan is located in the sub-frigid zone and the southern part in the sub-tropical. Being in the monsoon area, it has a summer season of high temperature and high humidity, and a cold, dry winter season.

Agricultural land in Japan is divided into two major categories: the 'paddy field', so typical of Oriental agriculture, and the 'upland field'

The paddy field is the basis of rice cultivation. The growing season extends in different districts from late spring to the middle of autumn, and water is kept on the fields almost until the beginning of the harvest season. The fertility of land is preserved by irrigation and by the use of chemical fertilizers, which is one of the essential features. After the rice is harvested, the field is left fallow until the following season in the single cropping districts, but in other districts where the second crops such as winter barley, wheat, and rye follow after rice, the paddy field is drained and utilized like an upland field until the following rice season.

Upland fields are similar to the farmlands of Western countries. Irrigation is sometimes practised on upland fields to moisten the soil, but water is not kept on them to the same extent as on paddy fields. With the exception of paddy rice and the second crops of the paddy, all other crops, including fruit trees and mulberry trees, are cultivated on upland fields.

The total arable land area is 6 113 000 hectares, of which 3 428 000 are paddy fields and 2 685 000 are upland fields. As paddy fields occupy more than half the entire arable land, rice culture is the core of Japanese agriculture and is of unchallenged importance (1)†.

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† *References are given in Appendix 1.1.*

The riding type of tractors can be operated only on fields that are in comparatively good condition, and these are assumed to be about 40 per cent of the total paddy field. The walking type tractors can be operated on 55 per cent of the remainder, leaving 5 per cent which can only be worked by manual or cattle power.

The common farm has an area of approximately 0·5–1 hectare, but the number of farms larger than 1·5 hectares is gradually increasing.

The size of fields is restricted by the convenience of irrigation and drainage, and the average is assumed to be around 10 ares (18 × 54 m). Fields are gradually being consolidated to 20–30 ares to improve the working efficiency of farm machines.

The farm population is decreasing annually at a rate of 2·5 per cent, while the total population is increasing, year by year, the increase being 27 per cent in 1968.

The number of farm households was about 5 350 000 in 1968—a decrease of 1·25 per cent from the previous year. The number of farms that are engaged mainly in agriculture is 3 650 000 (68 per cent), and 700 000 (32 per cent) are engaged mainly in other tasks. Many of the farm workers are old, and it is assumed that in 10 years' time the number of farms will decrease to 2 770 000 owing to a lack of successors.

TRACTOR UTILIZATION

Owing to an exodus of rural labour caused by the rapid economic growth in Japan, 1960 proved to be an opportune time to convert from walking to riding type tractors. Riding tractors were imported from several countries, but tractors suitable for our particular circumstances were requested and domestic tractors of less than 30 hp have been produced to fulfil this requirement.

In 1958 only 2500 riding tractors were in use, but after annual increases at the rate of 150 per cent, 130 000 were in service by 1968.

Table 1.1. Types of operation performed by tractors (2)

Operation	Paddy field		Upland field	
Number of farms using tractors	208 400	(100·0)	178 330	(100·0)
Soil turning (ploughing or rotary tilling)	194 170	(93·2)	161 410	(90·5)
Harrowing	140 000	(67·2)	101 070	(56·7)
Puddling	117 150	(56·2)	—	—
Seeding	920	(0·4)	3020	(1·7)
Inter-cultivating	—	—	9460	(5·3)
Fertilizing	1940	(0·9)	7040	(3·9)
Spraying and dusting	21 370	(10·3)	24 450	(13·7)
Harvesting	4110	(2·0)	32 340	(18·1)
Transporting	35 810	(17·2)	33 130	(18·6)
Others	5490	(2·6)	8900	(5·0)
Stationary tasks: Pumping	820	(0·4)	350	(0·2)
Threshing and hulling	6360	(3·1)	2570	(1·4)
Others	1200	(0·6)	1100	(0·6)

The number of wheel and crawler tractors introduced in 1968 was 41 400, of which 5102 were imported. Out of 40 306 wheel tractors only 5090 were imported, the remainder being produced in Japan.

Eighty-five per cent of wheel tractors were smaller than 30 hp and most of them were domestic, but 82 per cent of tractors larger than 30 hp were imported, as were all tractors larger than 40 hp.

Most tractors were publicly owned in the early stage, but private acquisition has increased year by year and in 1968 reached 85 per cent of the total tractors obtained.

Most of the tractors used at present have a small power rating. The number of tractors in the range of 10–15 hp account for 44 per cent of the total; 16–20 hp, 22 per cent; less than 10 hp, 17 per cent; and only 13 per cent of the tractors are larger than 30 hp.

The percentage of the area tilled by tractors is 6·6 per cent of paddy field, 13·6 per cent of upland field, which is 9·5 per cent of the total field area.

The farm operations done by tractors are shown in Table 1.1 (see also reference (2)). When operating in paddy fields the tractors are primarily used for soil turning (mainly rotary tilling), followed by harrowing, puddling, transporting, etc. In upland fields the order is ploughing, harrowing, transporting, harvesting, etc.

Rotary tilling of paddy fields is usually preferred to ploughing for two main reasons. First, ploughing leaves ridges or ditches which are difficult to eliminate, and these consequently affect the growth of weeds and rice after submergence. Second, the combination of ploughing and harrowing, requiring two processes, is less efficient than rotary tilling.

Most privately owned tractors are used approximately 20–40 days per year. Although this is not counted by the hour, it is considerably less than the co-operatively used tractors, which average 400 hours per year.

The recent trends in production show the predominance of 10–20 hp tractors, but the increase in the number of 20–30 hp tractors being produced is remarkable, and production lines for 30–40 hp tractors are proceeding gradually. However, tractors larger than 40 hp must still be imported, and there is an ever-increasing demand for tractors of this size as improvements are made to farm roads and as rapid progress is made towards consolidating fields, systematizing farm production, and enlarging the size of farms.

DESIGN CONSIDERATIONS

Most of the tractors in Japan are used for rotary tilling and puddling operations on small rice fields, which are wet and soft. In these circumstances the tractors used in this country must possess special characteristics.

Engine

Tractors smaller than 17 hp are fitted with either kerosene or diesel engines of one cylinder. The initial cost of the kerosene engine is about 90 per cent of the diesel, and its fuel consumption is about 1·5 times that of diesel. As the price of kerosene or diesel oil is almost the same, the total operating costs of both types of engine are very similar.

Tractors in the range 17–25 hp are fitted with two-cylinder diesel engines. As the distribution of kerosene into the two cylinders is complicated and expensive, kerosene engines are not used in tractors of this size.

Tractors larger than 25 hp are mounted with three- or four-cylinder diesel engines.

Only 25 per cent of tractors have air-cooled engines because of their high frequency noise and of users' anxiety for their durability for continuous heavy duty such as rotary tilling.

Most of the combustion chambers in domestic tractors are either vortex flow or precombustion chamber type while the imported tractors are direct injection. As the domestic tractors have a higher rotational speed, this tends to make the weight of engine lighter. The rated speed of the domestic engine is 2200–3000 rev/min, while that of an imported engine is 2000–2500 rev/min; and the horsepower per displacement of the former is 16–26 hp/litre while that of the latter is 12–16 hp/litre (Fig. 1.1) (3).

The elasticity of the engine, expressed by the following equation (by W. Flössel), shows a tendency that the higher the horsepower, the higher the elasticity (Fig. 1.2) (3). This is considered to come from the thermal restriction and lack of power reservation in the smaller engines.

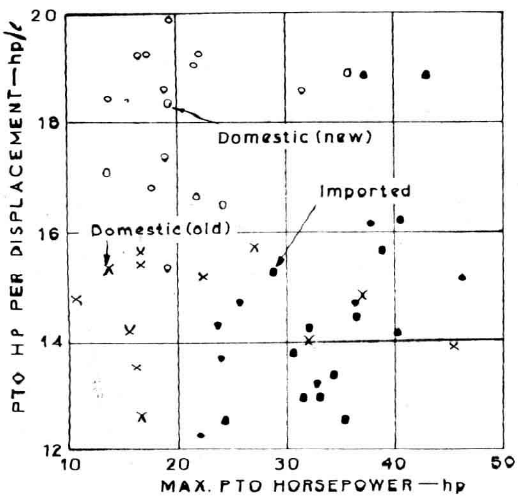

Fig. 1.1. PTO horsepower per displacement of cylinder (3)

$$E = T/t \times n/N \quad . \quad . \quad . \quad (1.1)$$

where E is the elasticity of the engine, T the maximum torque, t the torque at maximum horsepower, n the engine speed at maximum horsepower, and N the engine speed at maximum torque.

Tractor weight

It is needless to say that there is a general tendency that the higher the horsepower the heavier the tractor, but the weight per horsepower of tractor is less when the horsepower is higher. The following linear relationship exists between them:

$$w = 75 - 0{\cdot}7P \quad . \quad . \quad . \quad (1.2)$$

where w is the weight per horsepower (kg/hp) and P is the maximum power-take-off horsepower.

However, to improve the trafficability on soft soil, the recent domestic tractors have a lower weight per horsepower, which is expressed by the following equation:

$$w = 65 - 0{\cdot}7P \quad . \quad . \quad . \quad (1.3)$$

If the front axle load is light, there is a danger that the front wheel may rise. On the other hand, if the front axle load is heavy, the trafficability and steering on soft soil, such as a poorly drained or waterlogged field, will be worse (Fig. 1.3) (4). The average value of front axle load is 38 per cent (34–43 per cent) of the total weight of the tractor alone, and 24 per cent (15–31 per cent) when equipped with a rotary tiller.

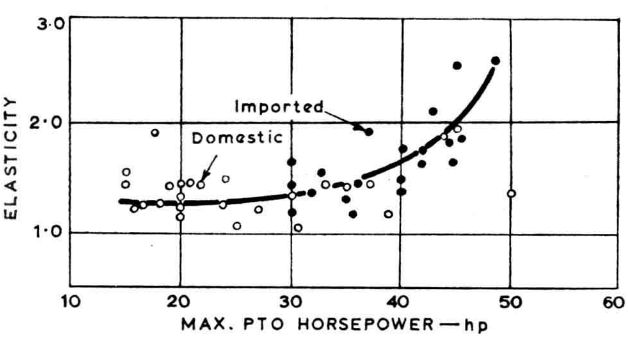

Fig. 1.2. Elasticity of diesel engine (3)

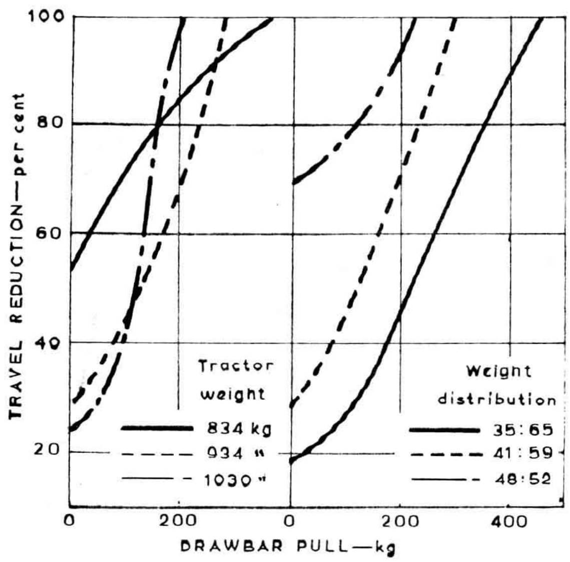

Fig. 1.3. Effect of tractor weight and weight distribution on soft soil (4)

Power-take-off performance

Most imported tractors have only one power-take-off (PTO) speed, but the domestic tractors have usually two speeds, while several are equipped with three speeds. In Japan, the tractors are used mainly for rotary tilling operations which require variable PTO speeds according to the soil conditions and the degree of pulverization required. User demand requires that this gear shift can be easily operated from the driver's seat. Moreover, the tilling parts

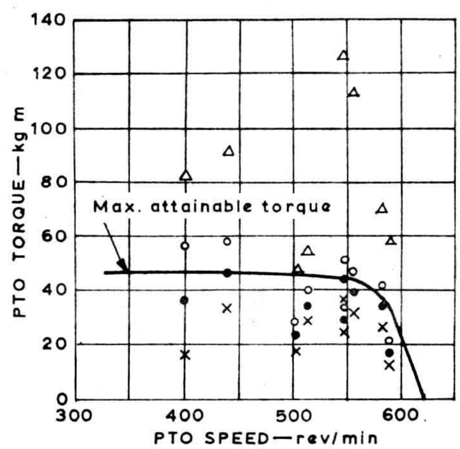

Fig. 1.4. Torque characteristics during rotary tilling operation (5)

must be as light as possible on soft soil for good trafficability. For these reasons many tractors have PTO change gears on the tractor, not on the tiller.

The torque fluctuation during rotary tilling is serious. The respective average values of peak torque and maximum peak torque are about 120 and 300 per cent of the maximum attainable PTO torque in ordinary soil conditions (Fig. 1.4) (5). For the purpose of accelerated durability testing, an apparatus that can apply a fluctuating torque on a PTO shaft was devised at the Institute of Agricultural Machinery (I.A.M.) in consideration of above facts (5).

There is a tendency that the greater the horsepower, the less the specific fuel consumption. Generally, the value is 200–260 g/hp-h at maximum power, and 240–300 g/hp-h at half power.

Travelling performance

The more gear shifts that are available, the easier the selection of a suitable speed for each operation. However, the price increases with the number of gear shifts, and the operation of changing gear becomes complicated. For this reason it is sufficient to have only the minimum number of speeds necessary for the particular operations to be done on each farm.

The majority of the tractors used in Japan have six forward speeds, followed by 8, 5, and 10 in that order. Tractors of less than 30 hp have six speeds, and larger ones have usually eight speeds. Most tractors have two reverse speeds.

The travelling speeds adopted for each operation are as follows: The speed range of 1–2·5 km/h is used for tilling and harrowing; 3·5–4·5 km/h for ploughing; and 8–9 and 14–16 km/h for travelling and transporting.

The lowest speed is 1–1·5 km/h in the majority of domestic tractors and 2–2·5 km/h in the imported tractors. The domestic tractors are capable of these low speeds to enable them to perform rotary tilling operations in paddy fields.

Legal restrictions govern the speed at which tractors may travel on the road. The highest speed for the small tractors, where the engine displacement is less than 1500 cm^3, and for which a driver's licence is easily obtained, is limited to 15 km/h. The highest speed for large tractors is legally limited to 50 km/h, although the draft of Safety Standard recommends 25 km/h.

For travelling on soft soils there is a wide variety of subsidiary devices such as girdles, strakes, steel wheels, half tracks, as well as special pneumatic tyres (high lug, wide section). Tractors often travel on the road with these subsidiary devices, and suffer severe shock and vibration. A test rig, consisting of a drum dynamometer (2 m diameter) with projections on the surface of the drums, was constructed at I.A.M. for the purpose of accelerated testing (5).

Many other troubles occur when these subsidiary devices are used on soft fields. When the sticky soil tangles in the cage wheel, the driving force (axle torque divided by wheel radius) attains 2·4 times the total tractor weight (5), and causes breakage of the power transmission system, hub bolts, wheel bosses, etc. Also, turning at the headland with a wide steel wheel can cause damage to the braking mechanism.

A differential lock is essential for travelling on soft soil. Wheel skids should be avoided not only for the trafficability of the tractor but also to ensure that the transplanting machine is running on a smooth hard pan to get a good transplanting result.

Hydraulic system

More than 90 per cent of the imported tractors have an automatic control system, but most of the domestic tractors have a selective system. In rotary tilling operations the hydraulic system is set in the neutral position to prevent the effect of a pitching motion of the tractor on the rotary tiller, and the tilling depth is controlled by gauge wheels attached to the rotary tiller. Therefore, it is not so inconvenient if an automatic control system is not available for the rotary tilling operation. However, for other operations an automatic control system is desirable, and there is now a tendency for domestic tractors to be equipped with such a system.

Waterproof performance

The tractors in Japan are often used on waterlogged or poorly drained fields, and danger can occur if the brake system is not sufficiently waterproof. For example, if muddy water enters the brake system during a puddling operation, the tractor's turning radius will increase as the brake on the inner wheel will not be sufficiently effective. This produces a reduction in the working rate, and can cause the tractor to strike against levees.

The brakes in the majority of domestic tractors are sealed and situated on the shafts in front of rear axles. However, as the drive wheels splash water on to the tractor body, this water can enter any openings and it is therefore necessary to ensure that the water has as few sources of entry as possible.

It is also necessary to ensure that the front axle and rotary shaft bearings are adequately waterproofed. According to the results of a 2-hour test that was performed in the testing pool at I.A.M., water enters the front axle bearings in 22 per cent of tractors and gains entry to the brake system in 7 per cent.

Muddy water enters the engine cylinder through the air cleaner which is in a comparatively low position, and causes wear in the bearings and cylinder. It also gains access to the clutch through holes on the clutch housing, and when the mud dries the clutch facings adhere to each other and are difficult to disengage.

Rotary tilling

The paddy field soil should not be tilled too deeply as this tends to make puddling and transplanting more difficult. The recommended depth is between 13 and 15 cm.

To achieve higher efficiency and better finishing, users demand that the rotary tilling width be wider than the outer width of the tyres. The rotary tilling width is approximately expressed by the following equation:

$$b = 0{\cdot}3\sqrt{P} \quad . \quad . \quad . \quad . \quad (1.4)$$

where b is the rotary tilling width (m) and P is the maximum PTO horsepower.

As a matter of course, the higher the engine horsepower, the higher the rate of work of rotary tilling. The relation between them is hyperbolic and is expressed by the following equation (6).

$$T = \frac{9}{P} + 0{\cdot}1 \quad . \quad . \quad . \quad . \quad (1.5)$$

where T is the time (in hours) required for tilling 10 ares and P is the engine horsepower.

Ridability

Convenience of mounting and dismounting

The steps on imported tractors are about 24 per cent higher than those on domestic tractors, and are unsuited to Japanese physique (see Table 1.2) (7). The optimum height of a step is considered to be 477 mm. Access for mounting and dismounting is too restricted in both domestic and imported tractors.

Table 1.2. Physique of Japanese adult (male, sitting) (7)

Parts	Average, mm	Standard deviation, mm
Height of sitting surface	427	26
Height of head	899	29
Height of eye	788	29
Height of shoulder	601	40
Shoulder to elbow	331	32
Height of elbow	270	24
Thickness of breast	194	17
Length of forearm	391	34
Thickness of abdomen	203	28
Hip to knee	529	25
Height of knee-pan	477	27
Height of upper surface of thigh	550	19
Height of lower surface of thigh	401	—
Hip to rear surface of calf	576	—
Width of shoulder	410	16
Width between elbows	451	40
Width of hip	328	21
Width between knees	399	68
Width of foot	103	6

Seating Dimensions

On imported tractors the height and width of seats are excessive for Japanese operators, while the seating facilities on domestic tractors are usually too small. The optimum height of seat is considered to be 400–430 mm, and the optimum widths at the front and rear of the seat are 400–420 mm and 350–400 mm respectively.

The depth of seat is appropriate (400 mm) in imported tractors, but is too short in about 50 per cent of the domestic tractors.

Adjustability of seat position

The seat on many tractors has no means of adjustment, and of those that are equipped with adjustable devices, the possible range of adjustment is insufficient for comfort (one-eighth to one-ninth of stature, or 180–200 mm).

Height of back rest

The back rests in most tractors are suitably positioned (about 220 mm over the seat surface).

Steering wheel

Diameters and inclinations are generally too large, especially in imported tractors. In most tractors the gripping points are longitudinally so distant and vertically so low that few operators can maintain a comfortable posture (i.e. angle of upper arm 0–25° and angle of elbow 90°, when seated).

Pedal force

In general, too much exertion is required to depress the pedals and they are usually positioned higher than necessary; this is particularly evident in a dual clutch system.

APPENDIX 1.1

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Paper 2

A PROCEDURE FOR PARAMETER OPTIMIZATION WHEN DESIGNING TRACTOR COMPONENTS

J. A. Caywood* H. S. Basrai†

This paper presents a practical computer-aided method called the 'grid and star search technique' for determining the values of adjustable design parameters. The objective is to find that set of values necessary to obtain a solution as nearly optimum as possible. The technique is a systematic search for the best set of parameters in a bounded region of possible answers. A 'performance index' must be defined to determine the merit of the solution after each iterative step in the procedure. The design of a spool type hydraulic relief valve is used as a practical example. The adjustable parameters are spring rate, metering orifice size, damping orifice area, and the volume of oil trapped between the pressure-sensing end of the spool and the damping orifice. The driving function is a step input of rated flow, and the optimum response is one with zero pressure overshoot. When the valve was made and tested, comparison between actual and predicted response was satisfactory.

INTRODUCTION

ALTHOUGH THE DIGITAL COMPUTER has greatly expanded the engineer's ability to analyse his designs, obtaining optimum designs is accomplished primarily through a trial and error technique of successive steps of design and analysis using intuition and experience to determine the correct parameter values. The next evolutionary step in engineering will be synthesis wherein the best possible design is derived from one run of a computer program. Parameter optimization by grid and star search is a step in that direction.

Parameter optimization is a formal procedure used to adjust the design parameters of a physical system to obtain specified characteristics. It is applicable to almost any design problem where several parameters can be adjusted to obtain the best solution. The technique applies equally well to multiple bar linkages, hydraulic valves, machine element stress, sensitivity of systems to manufacturing tolerances on their components, cost analysis, system identification, etc. The particular technique presented here is a systematic search for the best set of parameters within a bounded region of possible answers, and is called a 'grid and star search' (1)‡.

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‡ *References are given in Appendix 2.1.*

The theory of the grid and star search is presented first in general terms—implying that it can be applied to any design problem. The theory is followed by a practical example wherein a tractor hydraulic relief valve of optimum performance is synthesized.

Notation

- A Cross-sectional area of spool, in^2.
- A_d Area of damping orifice, in^2.
- A_0 Area of metering orifice, in^2.
- C_d Coefficient of discharge.
- d Diameter of spool, in.
- F_0 Initial spring compression load, lb.
- h Clearance between bore and spool, in.
- k Spring rate, lb/in.
- l Length of spool land, in.
- L Effective damping length, in.
- m Effective mass of spool, ball, disc, and spring, in slugs.
- p Pressure in end chamber, lb/in^2.
- p_s Supply pressure, lb/in^2.
- Q_s Supply flow, in^3/s.
- t Time.
- V Volume of oil upstream of valve, in^3.
- V_0 Initial volume of oil in end chamber, in^3.
- W Width of metering orifice, in.
- x Spool displacement, in.
- β Compressibility of oil, lb/in^2.
- ν Viscosity of oil, lb s/in^2.
- ρ Density of oil, lb/in^3.

THEORY OF THE GRID SEARCH

The grid and star search routine is a combination of two techniques—a grid search and a star search. The basic grid search described below is self-sufficient but can be improved by supplementing it with the star search (2). Discussions of the star search and the benefits of alternating between a star and grid search appear in the next section.

In the grid search a region is defined which has, as its sides, dimensions representing the allowable range of the adjustable design parameters. This region can have any number of sides, corresponding to the number of adjustable parameters, thus forming a 'hypervolume'. The objective of the search is to locate a point within this region which represents the optimum combination of parameters as defined by given criteria.

The grid search is not a random approach but a systematic method of eliminating regions which give poor answers, and of converging on the point which gives the best answer. Basically the idea is to move from one point to another, always in a direction which keeps improving the suitability of the solution to the design problem. To force the searching point to move a shorter distance each time thus ensuring convergence, the volume is reduced to some fraction of its original size after each move. Convergence on the best answer is obtained when the volume has been shrunk to a size where subsequent changes in the parameter values at each step are sufficiently small.

Consider the problem of finding the values of m design parameters, $P_1, P_2, P_3, \ldots, P_m$, such that the difference between the desired and actual behaviour is minimized. This difference is characterized by a scalar quantity called the 'performance index' (PI). Since visualization of movement within the many-sided hypervolume previously defined is impossible, illustration of the technique will be done with only two parameters—P_1 and P_2—while keeping in mind that the theory can be generalized to m parameters.

Since this sequential search technique cannot be conducted in an area having sides of infinite length, upper and lower bounds must be set for each parameter. The bounds can usually be defined from practical design limits or by sample calculations. Each parameter is then normalized so that its lower boundary corresponds to the value 0 and its upper boundary corresponds to the value 1 on a new hypothetical unitary scale. For example, assume that a parameter has a lower boundary of a and an upper boundary of b. Any point Y between the two boundaries would be normalized on the unitary scale as follows:

$$Y' = (Y - a)/(b - a) \quad . \quad . \quad . \quad (2.1)$$

where Y' is the normalized value of Y.

Example: Let $a = 1$, $b = 7$, and $Y = 4$; then

$$Y' = \frac{4-1}{7-1} = \frac{1}{2} \quad . \quad . \quad . \quad (2.2)$$

The next step is to form a grid by dividing each side into thirds, as in Fig. 2.1, and to find the mid-point of the region.

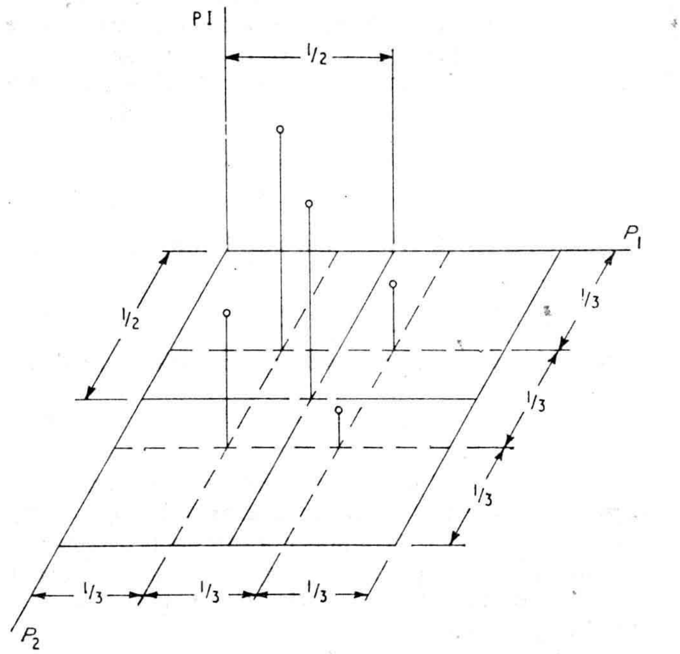

Fig. 2.1. Parameter search grid

Performance indexes are calculated from the values of the parameters given by the nodes of the 1–3 grid lines and by the mid-point. The node or mid-point yielding the best PI becomes the mid-point of a new grid of reduced area, and the process is repeated. This iterative procedure is continued until the region of uncertainty (that region which has not yet been searched) has been reduced so that further reduction results in only minute changes in the parameter values and/or in the PI.

The reduction of the region of uncertainty accomplished at each step must be between 2–3 and 1. If it is less than 2–3, the boundaries of the region of uncertainty will shrink faster than the searching point can move, and some areas will never be searched. If it is equal to 1, of course, no reduction in area will occur. Experience has shown that a reduction (r) of 3–4 is a good compromise, giving fairly rapid convergence while also allowing the technique to recover from 'ill-directed' moves. Ill-directed moves can result from a complex equation for the PI containing discontinuities.

Note that the number of evaluations of the PI required in the first step was $2^m + 1$, and the size of the region of uncertainty was thus reduced by r. The next iterative step

Table 2.1. Effort expended in grid search

n Number of evaluations of the performance index	f Functional reduction of area of uncertainty
$2^m + 1$	r
$2(2^m) + 1$	r^2
$3(2^m) + 1$	r^3
$\vdots$	$\vdots$
$b(2^m) + 1$	r^b

will require 2^m more PI evaluations, resulting in an area reduction of r^2. Table 2.1 shows how this reasoning can be pursued to determine how many PI evaluations are required to achieve a certain fractional reduction in the area of uncertainty. From Table 2.1 it can be seen that:

$$f = r^b \qquad (2.3)$$

$$n = b(2^m) + 1 \qquad (2.4)$$

Solving equation (2.4) for b and substituting into equation (2.3):

$$f = r^{(n+1)/2^m} \qquad (2.5)$$

From which the number n of PI evaluations required is:

$$n = 1 + \frac{2^m \ln f}{\ln r} \qquad (2.6)$$

where f is the final fractional reduction of region of uncertainty, r the reduction factor used at each iterative step $(2–3 < r < 1)$, and m the number of adjustable parameters.

Thus it can be seen that convergence to a finite area is assured and the number of evaluations required for convergence can be calculated.

Example: For $m = 4$,

$r = 0{\cdot}75 \qquad f = 0{\cdot}01 \qquad n = 257$

GRID AND STAR SEARCH

The grid search routine can be criticized for searching only on the diagonals of the grid.

Fig. 2.2 shows the search pattern after four iterations, assuming the best PI remains at the original mid-point. If the surface described by the PI exhibits an extremum (either a minimum or a maximum) along an orthogonal

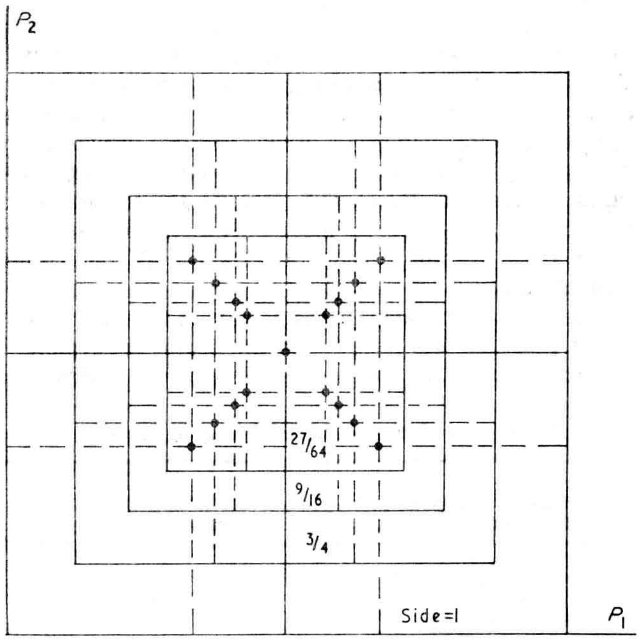

Fig. 2.2. Grid search pattern after four iterations

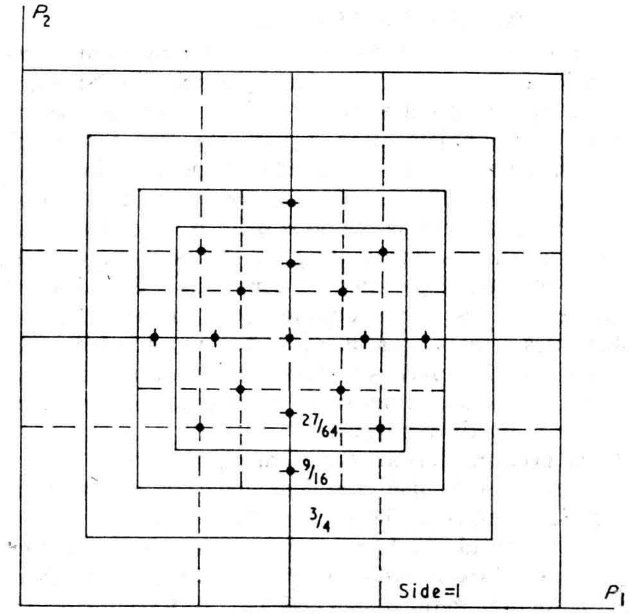

Fig. 2.3. Grid and star search pattern after four iterations

axis through the mid-point, the extremum would remain undetected.

This fault can be overcome by alternating between a grid and a star search. The star search is accomplished by selecting points along orthogonal axes through the mid-point, at a distance of plus and minus one-third the side length from the mid-point. Fig. 2.3 shows the search pattern after four iterations of a grid and star search routine, again assuming that the best PI remains at the original mid-point.

Table 2.2. Effort expended in grid and star search

Search technique	n Number of evaluations of the performance index	f Fractional reduction of area of uncertainty
Grid	$2^m + 1$	r
Star	$2^m + 2m + 1$	r^2
Grid	$2(2^m) + 2m + 1$	r^3
Star	$2(2^m) + 2(2m) + 1$	r^4
Grid	$3(2^m) + 2(2m) + 1$	r^5
Star	$3(2^m) + 3(2m) + 1$	r^6
Grid	$4(2^m) + 3(2m) + 1$	r^7
Star	$4(2^m) + 4(2m) + 1$	r^8
⋮	⋮	⋮
Grid	$\frac{b}{2}(2^m) + \left(\frac{b}{2} - 1\right)(2m) + 1$	r^{b-1}
Star	$\frac{b}{2}(2^m) + \frac{b}{2}(2m) + 1$	r^b

From Table 2.2 it can be seen that:

$$f = r^b \qquad (2.7)$$

$$n = \frac{b}{2}(2^m) + \frac{b}{2}(2m) + 1 \qquad (2.8)$$

Solving equation (2.8) for b and substituting into equation (2.7):

$$f = r^{(n-1)/(2^{m-1}+m)} \quad . \quad . \quad . \quad (2.9)$$

from which the number n of PI evaluations required is:

$$n = 1 + (2^{m+1} + m)\frac{\ln f}{\ln r} \quad . \quad . \quad (2.10)$$

which is more economical of PI evaluations than the grid search alone, when the number of adjustable parameters is three or more.

Example: For $m = 4$,

$$r = 0{\cdot}75 \qquad f = 0{\cdot}01 \qquad n = 193$$

DISCUSSION OF THE PERFORMANCE INDEX

The grid and star search is applicable to any parameter optimization problem. The PI can be based on minimum cost, maximum stress, minimum time, minimum sensitivity to manufacturing or environmental tolerances, etc., and even combinations of the above.

One of the more important uses of the grid and star search is the identification of the unknown parameters of system mathematical models. First, the driving function and response of the real system are simultaneously recorded as functions of time. The driving function is then used as the input to the mathematical model and the simulated response is compared with the real response. The grid and star search can be used to select the parameters which most nearly force the set of differential equations to simulate the real response. The accuracy is limited only by the engineer's ability to determine the types of terms required in the differential equations, and the number of parameters he allows to 'float'. No restrictions are placed on linearity or continuity of the system equations.

The most common PI used in identification or synthesis of dynamic systems, four bar links, etc., is the integral square error.

Example:

$$\mathrm{PI} = \int_{t_1}^{t_2} (x - x^*)^2 \, \mathrm{d}t \quad . \quad . \quad (2.11)$$

where t_1 is the initial time, t_2 the final time, x the displacement calculated at time t during numerical integration, and x^* the displacement of the desired trajectory at time t.

Assume that in addition to following a desired trajectory, the designer wants the system to be insensitive to changes in oil viscosity. The PI might be calculated as follows:

$$\mathrm{PI} = \int_{t_1}^{t_2} \left[W_1(x - x^*)^2 + W_2 \frac{\partial x}{\partial v} \right] \mathrm{d}t \quad (2.12)$$

where W_1 and W_2 are weighting factors ($W_1 + W_2 = 1$), and v is the oil viscosity.

The grid and star search routine has one weakness common to all parameter optimization techniques presently in use—it can converge on a local extremum. Fig. 2.4 illustrates how an erroneous convergence on a local maximum could occur. It is a simple matter, however, to determine whether this has happened by moving the boundaries around a few times, checking for a better PI.

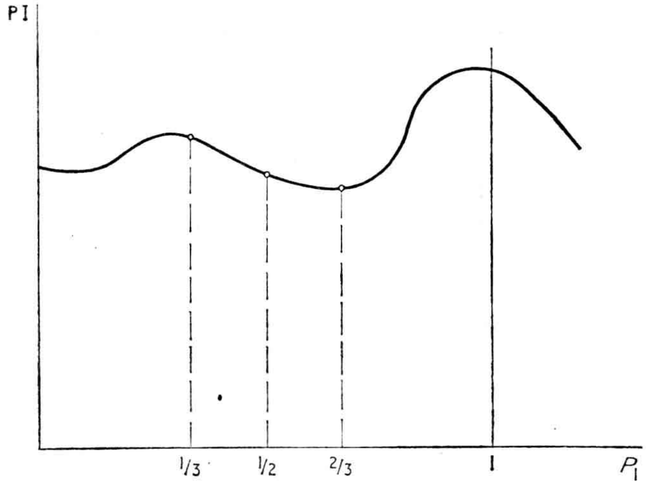

Fig. 2.4. Convergence on a local maximum

APPLICATION TO A TRACTOR HYDRAULIC RELIEF VALVE

To illustrate the application of the grid and star search technique, a hydraulic pressure relief valve was developed and tested in a laboratory to compare the actual response with the computed response. A spool type valve design was selected as opposed to a seating type valve because more accurate mathematical models can be derived for sliding spools (3).

The most important consideration in any optimization problem is the 'desired' behaviour or response of the component being designed. Past experience with relief valves has shown that two characteristics adequately describe their behaviour:

(1) steady-state pressure flow characteristics;
(2) transient response of pressure before it reaches the steady state.

Ideally, the valve should be capable of relieving maximum flow without a pressure rise over the cracking pressure and should have the capacity to relieve a very high rate of pressure rise in the system without overshoot. The desired behaviour can be the ideal behaviour, even although it may be impossible for a practical physical system to meet the stated requirements. The grid and star search technique will give a mathematical 'best possible fit' to the desired behaviour. The quality of the fit is limited only by the number of adjustable parameters and the types of terms they modify.

For the tractor on which the relief valve was to be installed, it would be required to respond to a pressure rise rate of 180 000 lb/in^2/s. It was also determined that a rise of 300 lb/in^2 over the cracking pressure would provide adequate protection. With this information, the desired behaviour can be defined as shown in Figs 2.5 and 2.6.

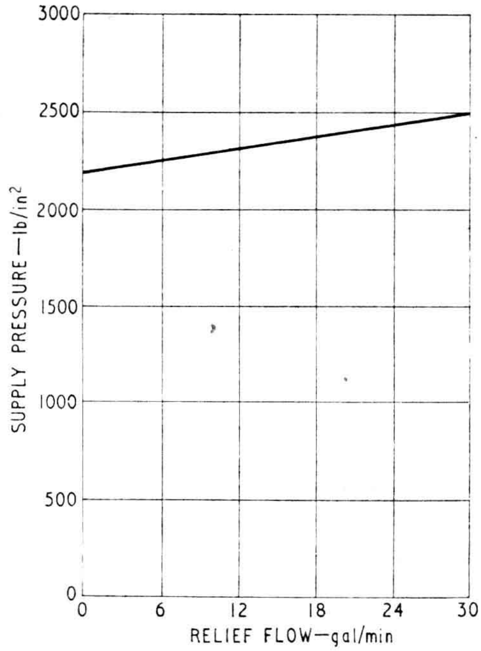

Fig. 2.5. Desired pressure flow characteristics

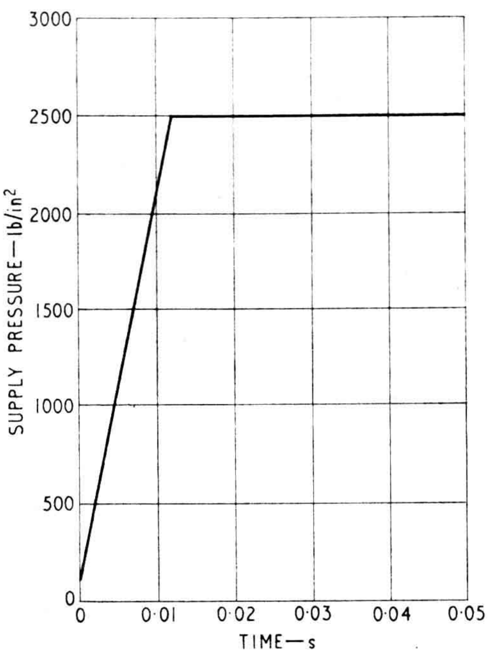

Fig. 2.6. Desired pressure response

The PI was defined as:

$$\mathrm{PI} = \int_{t_0}^{t_f} \{ (p_s^{\star} - p)^2 \,\mathrm{d}t \qquad . \quad . \quad (2.13)$$

where $p^{\star}$ is the desired pressure, p_s the calculated or simulated pressure, t_0 the initial time, and t_f the final time.

For an optimum solution the PI must be minimized.

Mathematical model of the valve

Fig. 2.7 illustrates the preliminary design of the valve used to simulate the performance on the computer. The set of

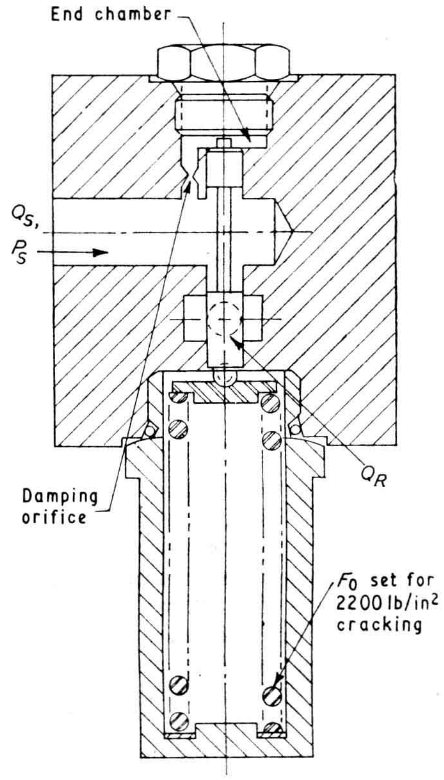

Fig. 2.7. Sketch of prototype

differential equations constituting the mathematical model of the relief valve is as follows:

System pressure dynamics:

$$\frac{\mathrm{d}p_s}{\mathrm{d}t} = \frac{\beta}{V}\left[Q_s - C_d w x\sqrt{\left(\frac{2p_s}{\rho}\right)} - C_d A_d\sqrt{\left\{\frac{2p_s - p}{\rho}\right\}}\right] \qquad . \quad . \quad . \quad (2.14)$$

Pressure sensing chamber dynamics:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\beta C_d A_d}{V_0 + Ax}\left[\sqrt{\left\{\frac{2}{\rho}(p_s - p)\right\}} - \frac{A\,\mathrm{d}x}{\mathrm{d}t}\right] \qquad . \quad . \quad (2.15)$$

Valve spool dynamics:

$$\frac{d^2x}{dt^2} = \frac{1}{m}\left[pA - F_0 - kx - \left(\frac{\pi d l \nu}{h}\right)\frac{\mathrm{d}x}{\mathrm{d}t} - 2C_d A_0 p_s(\cos 69°) - \{C_d W L\sqrt{(2\rho p_s)}\}\frac{\mathrm{d}x}{\mathrm{d}t}\right] \qquad . \quad . \quad . \quad (2.16)$$

Four parameters appeared to be most promising in influencing the behaviour of the system, and were selected for optimizing. The parameters are:

(1) Width of the metering orifice, W
(2) Area of damping orifice, A_d
(3) Spring rate, k
(4) Initial volume in the end chamber, V_0

A trial simulation was performed using the Runge–Kutta technique of numerical integration with some initial 'guesses' of the values of the parameters. W was set at 1·175 in, the maximum circumference of the spool. k was set at 660 lb/in, based on spool travel requirements for relieving 30 gal/min at 2500 lb/in². The values for A_d and V_0 were assumed to be 0·0025 in² and 0·5 in³ respectively. The response obtained with these values is illustrated in Fig. 2.8.

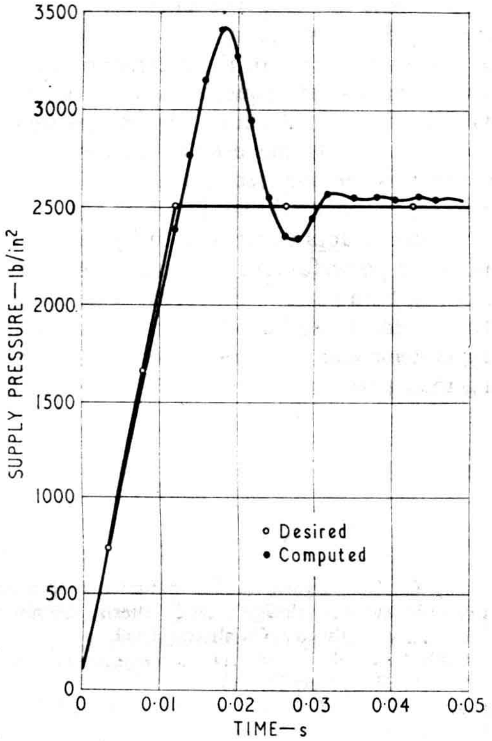

Fig. 2.8. Trial response using initial guessed parameters

The grid and star search technique was next utilized to determine the values of the parameters that would allow the system to respond with the desired characteristics. The region of uncertainty was first established by selecting

Table 2.3

Parameter	Maximum	Minimum
Damping orifice, A_d, in²	0·01	0·0005
End chamber volume, V_0, in³	5	0·5
Spring rate, k, lb/in	800	400
Orifice width, W, in	1·175	0·75

the limits on the parameters. The minimum PI calculated by the computer was at a point where the values of the parameters were as follows:

$$A_d = 0{\cdot}0048 \text{ in}^2$$
$$V_0 = 0{\cdot}975 \text{ in}^3$$
$$k = 531{\cdot}69 \text{ lb/in}$$
$$W = 1{\cdot}175 \text{ in}$$

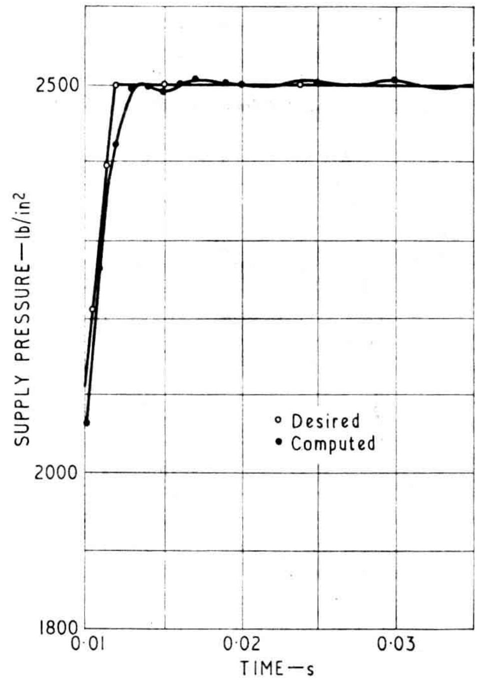

Fig. 2.9. Pressure response with optimum parameters

Note that the orifice width converged to its upper boundary. However, since the spool diameter was fixed, the orifice width could not be enlarged.

Fig. 2.9 represents the simulated pressure response with the above values of the parameters, and Fig. 2.10 represents the steady-state pressure flow characteristics.

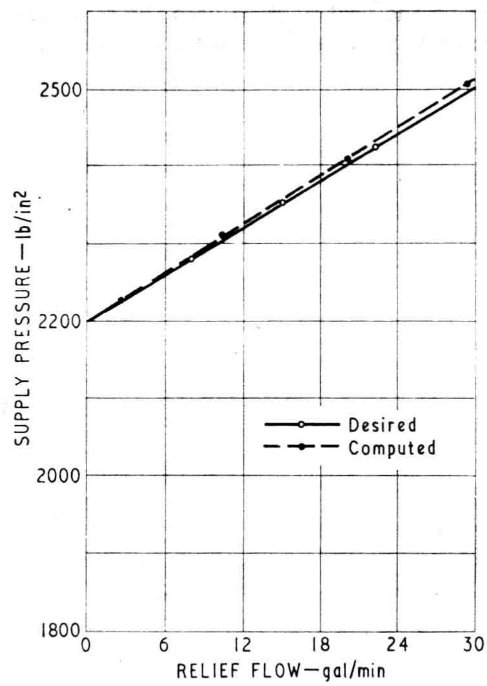

Fig. 2.10. Pressure flow characteristics

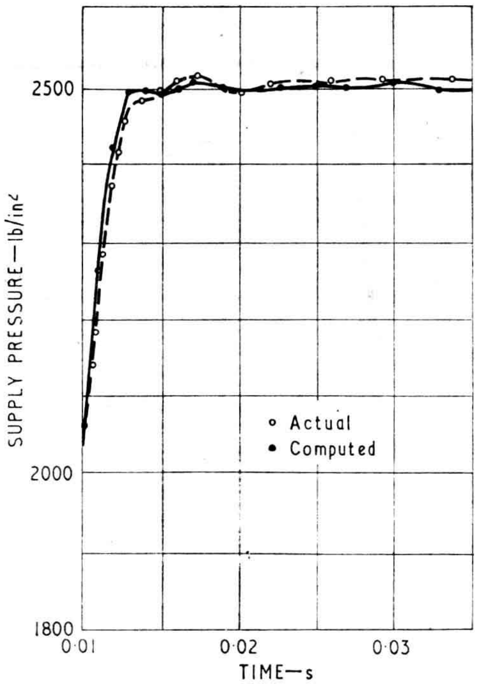

Fig. 2.11. Comparison of measured and computed responses

The valve was constructed using the above parameters and tested in the laboratory. The actual response was recorded on the oscillograph. Comparison of the simulated and actual pressure response is illustrated in Fig. 2.11.

CONCLUSIONS

Comparing the trial solution with the optimum solution shows that the grid and star search technique saved considerable man hours and expense in the preliminary design of the relief valve on the computer. The initial guesses for parameter values gave poor results, and it is not readily apparent, from studying the system equations, how the parameters should be changed.

The advent of the computer allowed the engineer to do his trial and error on paper instead of in actual hardware, resulting in considerable savings of time and effort. Optimization routines can now eliminate trial and error, even on paper, allowing the engineer to solve design problems by direct synthesis.

The main advantages of the technique presented here over other available techniques are mathematical simplicity, adaptability to both algebraic and differential systems, and ability to rapidly converge to the optimum solution. Moreover, it is the only technique available that permits the engineer to specify design limits on the parameters.

With a broad application capability, the optimization technique is a powerful tool for synthesis and of kinematic mechanisms, fluid power systems, minimum cost problems gear trains, etc., as well as identification of parameters in existing systems and components for the purpose of improving their performance.

APPENDIX 2.1

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Paper 3

THE PATTERN OF TRACTOR DEVELOPMENT IN GERMANY

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This paper reviews the types of tractor in use in West Germany and discusses the sales of new tractors over the last few years. It is shown that the interest in all-wheel drive tractors is steadily increasing owing to their higher power rating and improved tractive ability. The paper also comments on tractor equipment and design in relation to the proposed E.E.C. regulations for safety and performance, and concludes by summarizing some of the problems of tractor development.

INTRODUCTION

ACCORDING TO THE SO-CALLED "Green Report", published annually by the West German Government, there are about 1 344 000 agricultural holdings in West Germany; about 480 000 of these are full-time farms, i.e. the income is derived exclusively from agricultural activities, whilst the rest are part-time farms whose owners depend on employment as craftsmen or in industry and/or commerce for supplementary earnings. These part-time farmers have often acquired a high degree of mechanization out of income derived from their non-agricultural activities.

Official statistics for motor vehicles show that about 1·3 million tractors are used in German agriculture. Although this figure tallies quite well with the number of agricultural holdings, it does not show that many farmers keep several tractors, amongst them old models, for specific purposes simply because they cannot be sold advantageously.

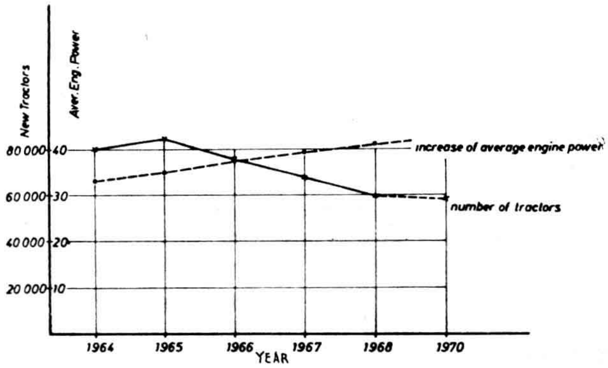

Fig. 3.1. Engine power and number of tractors sold

SALE OF TRACTORS

While the total number of agricultural tractors increased very rapidly between 1950 and 1965, it has remained almost constant over the last three-year period. These trends seem to be confirmed by the number of new tractors registered annually; new registrations have, in fact, declined since 1965 and only amount to about 60 000 p.a. at the present time (Fig. 3.1). The same illustration also shows that the average tractor engine power has increased over the years and now amounts to about 42 hp metric or 41·4 b.h.p. The distribution into groups according to engine power, is illustrated in Fig. 3.2, which is based on returns for 1969 available at present. Because the basic statistical grouping according to engine power was

Fig. 3.2. Tractors sold

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changed from June 1969, the columns marked 5M are based on the returns for only five months, whilst the others are based on a 10-month period. Highest sales were made in the 24–50-hp engine size range; 15 per cent of all tractors sold had engines giving more than 60 hp (59·2 b.h.p.).

TRACTOR TYPES AND CHARACTERISTICS

According to the periodical *Landtechnik*, 233 models of tractor with engines ranging from 7 to 165 hp were available on the market to satisfy the demand in 1969. A pictorial representation (Fig. 3.3) of the tractor types available

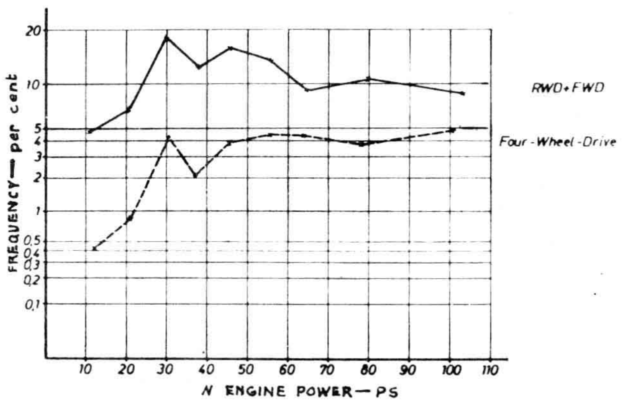

Fig. 3.3. Offered tractors—engine power of tractors

shows that the choice was greatest in the 30–55-hp group. The same illustration also shows that of all the tractors on offer, 67 models or 28·8 per cent were four-wheel drive versions, which implies a growing interest in all-wheel drive.

Most all-wheel drive tractors offered by German manufacturers feature smaller wheels at the front than at the rear. It is said that this gives more favourable turning by means of the steered wheels, and normal transmission systems can be used. Undoubtedly the development of such tractors has also been facilitated by the availability through Zahnradfabrik Friedrichshafen (ZF) of driven front axles. More recently four-wheel drive tractors have become available with equal sized wheels at the front and rear; like tractors for special purposes and the building industry, these machines are fitted with articulated steering (Fig. 3.4 (Deutz 160 06)).

The total number of all-wheel drive tractors includes special vineyard and orchard models. Because these are intended for inter-row cultivation and work on slopes, there is a demand for minimum overall width, good manoeuvrability, and high tractive performance. Track width is less than 1 m (39·4 in), but nevertheless engine output is up to about 40 hp (39·5 b.h.p.), which is frequently to be transmitted through the power take-off (p.t.o.) shaft to rotary cultivation implements. Tractive work on the slope requires good longitudinal stability, to prevent overturning rearwards. All-wheel drive fulfills this requirement since the necessary load on the front axle

Fig. 3.4.

also improves tractive capacity. Higher loads on the front axle also improve steering under these conditions, whilst effective all-wheel braking is ensured downhill when front-wheel drive is engaged. Nevertheless, this type of tractor demands alert, resourceful drivers, since the narrow track width increases the risk of overturning sideways. To minimize this, the centre of gravity of the tractor is kept as low as possible. Because of the narrow track width, implements are frequently attached by means of a swinging frame rather than the three-point linkage, and in this way only suitable special equipment can be used.

If one deducts the number of these special-purpose tractors from the total number of all-wheel drive tractors, the trend illustrated in Fig. 3.5 becomes apparent, which

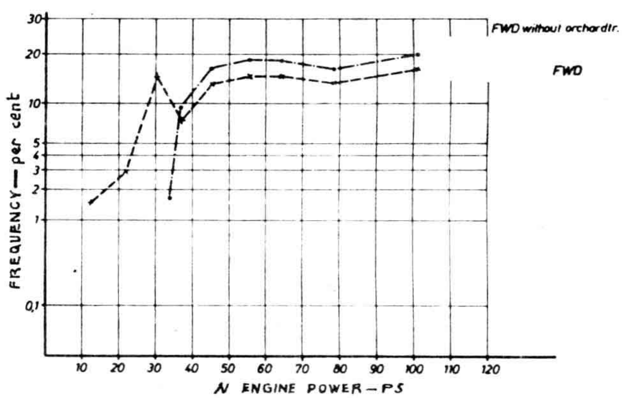

Fig. 3.5. Engine power of tractors

does not include low-powered tractors. Thus 50 per cent of the available all-wheel drive tractors have a mean power output of more than 56 hp, whereas the mean power output of 50 per cent of all tractors amounts to only 40 hp. These figures confirm that all-wheel drive tractors offered for agricultural purposes have more powerful engines, so that the extra power can be converted into traction where climatic and soil conditions demand this.