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ONDERWERP:
SUBJECT: A CRITERION OF CLASSIFICATION OF PISTON OPERATED MACHINES WITH PARTICULAR REFERENCE TO HYDRAULIC PISTON PUMPS.

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INDEX.

S U M M A R Y.

Piston operated machines have been grouped in classes according to a simple relationship between a certain specific power of the machine and its reduced mass.

The reduced mass implies an operational regime in which the forces of inertia have been reduced to a standard reference value.

Classes have been defined for axial piston pumps of fixed and variable displacement, for radial piston pumps, and for internal combustion engines for aviation.

SCOPE OF THE INVESTIGATION

The direct coupling of a prime mover to a working machine is not always applicable. Some coupling or transmission device is usually required.

Hydraulic devices have become quite popular, but the engineer, faced with the problem of acquiring suitable equipment for a particular operation, may find it rather difficult to choose the best equipment from the large variety of units presently offered.

The speed of the prime mover to which the drive must be coupled, the maximum power of the drive, and its mass or inertia, are the most important points to be considered.

The quantity "mass" is of particular importance in all the problems connected with transportation, where there is a practical limitation to the mass of the drive in relation to the mass of the vehicle and to the power installed.

Figures 5 and 6 consider this aspect for piston operated machines of fixed and variable displacement respectively.

The quantities: power, mass and revolutions, are related in a graphical representation in which account is taken of the products of many manufacturers.

This representation fairly sums up the possibilities and the limitations of the present manufacturing technology of these machines as industrial products.

Besides/...........

Besides being used in the field of power conversion, hydraulically operated piston machines are also used for servo commands. It is well-known that in this kind of application the power must be produced at various speeds, with the possibility of rapid speed inversion.

Consequently, the additional important parameter of "inertial moment" of the rotating masses has to be brought into account.

Figure 7 gives the inertial moments for two series of axial piston pumps as a function of the power developed.

This diagram thus permits a quick estimate of the values of the moment of inertia of axial piston machines.

Alternative solutions employing, for instance, a hydraulic piston motor or an electric d.c. machine can be analysed and compared in respect of moment of inertia.

FUEL RESEARCH INSTITUTE OF SOUTH AFRICA

REPORT NO. 4 OF 1972.

A CRITERION OF CLASSIFICATION OF PISTON OPERATED MACHINES WITH PARTICULAR REFERENCE TO HYDRAULIC PISTON PUMPS.

1. INTRODUCTION.

The importance of hydraulic volumetric machines, i.e. volumetric pumps and motors, and their assembly as hydraulic drives, is of ever increasing importance in the field of power conversion and in the automation of plants.

A prior knowledge of some of the common and typical features of these machines as a 'class' may be very useful to a designer, to a manufacturer, and to the user. This knowledge can readily be gained by an analysis of the external parameters of existing types.

External operational parameters are, for instance, power, fluid pressure, speed and volumetric displacement, which are generally available in any commercial description of the product.

It will be proved in the following discussion that although these machines do not apparently conform to any criterion of classification in practice, they are designed and produced, perhaps unintentionally, according to certain classes.

The same criterion of classification can be extended to other types of piston machines like internal combustion engines, or even to positive displacement machines not making use directly of pistons like vane and gear pumps, provided the external parameters mentioned above can be defined mechanically or hydraulically.

/2. MECHANICAL ....

2. MECHANICAL DESCRIPTION.

Volumetric machines are classified in practice as follows :-

1) Gear and vane pumps and other positive displacement machines not making use of pistons.

2) Radial and axial piston pumps*.

Volumetric machines can, moreover, be of fixed or variable displacement; this is according to whether the volume swept during a revolution is either fixed or adjustable from the outside by means of a setting device.

Volumetric machines of the piston type are often mounted in an assembly of a pump-motor combination forming a hydraulic drive.

Pump and motor can be connected in a drive form either hydraulically (by means of pipes), or hydraulically and mechanically (by means of pipes, shafts and gears).

The first type of drive (hydraulic connection only) is by far the most common. The second type of drive, because of the many possibilities of combination is, at the same time, very flexible in providing a great variety of solutions, but is, of course, of greater complexity. (cf. Ref.1.)

Gear and vane pumps can be applied in the transmission of small powers where the fluid throttling system of control is used. They are preferred for machine tool installations because of their low cost, simplicity and reliability, and have an efficiency lower than in the more elaborate machines of the piston type.

/Radial ....

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* In this text the use of the word 'pump' generally includes the word 'motor'.

Radial and axial piston pumps are the most commonly used components in hydraulic power transmission systems for special cases whenever power is required to be converted at a high rate of efficiency (a problem which was analysed in a previous work, Ref. 2).

Piston pumps generally have an odd number of pistons in order to reduce the periodical variations of the flow (cf. Ref. 3). Use is made of orifices and ports in the distribution of the fluid on account of the inherent simplicity of this system, particularly at high speed.

In Figure 1 a radial piston pump is shown for the purpose of illustration.

It is made of a revolving block (1) bearing some eleven radial cylinders (2) each fitted with a piston (3). Each piston is connected by means of a shank and a pin (4) to a block (5) running on a circular guide carried by a ring (6). The ring can be put off centre relative to the centre of rotation of the block by turning it around its pivoting point. This operation is performed in the case of the figure by means of a control piston (7), acting against a spring (8).

The supply and discharge of the oil to and from the pistons are made through the fixed axle (9), by means of two sets of radial orifices. The pistons move in and out in a radial reciprocating motion by an amount fixed by the eccentricity, fixed in turn by the position of the control piston.

Three fundamental types of axial piston pumps can be discussed, viz:-

(i) The swash plate type (Williams & Jenny) in Figure 2.

/(ii) A type ....

(ii) A type with pistons acting on pivoted pads represented in Figure 3.

(iii) The swingable body type (Thoma) in Figure 4.

The swash plate type of pump bears a shaft (1) carrying a swash plate (2), pivoted at right angles to the shaft so as to form a Cardan joint.

The shaft carries on its extension a piston block (3) with pistons (4) and connecting rods (5), having spherical heads (6) fitting into spherical sockets both on the piston and on the swash plate side. The swash plate can assume various slopes by means of a screw with adjusting knob (7). The oil is supplied through a fixed frontal distributor (8).

The pump with fixed plate and thrust pads is shown schematically in Figure 3.

The shaft (1) is rigidly connected to a piston block (2) carrying a set of pistons (3), which are supported by a fixed plate (4) by means of spherical joints (5) and thrust pads (6). The oil to the pistons is provided through an oil distribution block 7.

The pump with swingable body (Thoma) is shown in Figure 4. In this pump the piston carrying block and its case can be tilted as one body relative to the shaft, making it possible to adjust the displacement of the pistons and to select the sense of rotation.

The pump is made of a shaft (1) carrying a hemispherical flange (2) bearing a central socket(3), and some peripheral sockets (4), one for each piston. A piston block (5) carries the pistons (6) with connecting rods (7).

/A central ....

A central transmission (8) guides the piston block in its rotation relative to the inner case (9). The inner case bears in front an oil distribution block (10) which is fixed.

The whole mechanical assembly is enclosed in an outer casing, with tilting mechanism, not shown in the sectional view of the figure.

The following properties are typical of the machines described :-

Radial piston pumps are mostly used for power application where large torques at low or moderate speed are required (e.g. for control of cranes, ships rudders, winches etc.) In fact, the radial distribution of the pistons implies a cylinder block with a moment of inertia larger than the corresponding axial types. Consequently these machines are rather slow towards rapid changes in speed and speed inversion if driven at high revolutions.

During rotation the pistons move in and out radially, sliding on a cylindrical surface, where the various points are at different peripheral velocity.

The piston masses must be accelerated to the local peripheral velocity existing at that certain radial distance. A complementary acceleration is thereby imparted onto the piston (Coriolis acceleration), which produces forces appearing as side thrusts directed in the sense of the peripheral velocity of the piston and opposite to it.

The constructive lay-out of radial piston pumps appears favourable with regard to the oil distribution and to the volumetric efficiency. In fact, the distribution of the oil takes place near the centre of rotation of the cylinder block, where the peripheral velocity is small, and with it the loss in kinetic energy.

/The ....

The sealing between pressure and suction side is more favourable in the "hole-shaft" arrangement used, than between flat surfaces, because of the more accurate tolerances possible for this fit. Moreover, the presence of an oil film insures good lubrication as in any journal bearing.

Radial pumps have, therefore, a good volumetric efficiency which can be as high as 98% even at oil pressures of 400 bar*.

Axial piston pumps are the most widely used type, both in the power transmission and in the servo-control fields. They have a very favourable power-inertia ratio, mainly due to the fact that the mass of the piston block is placed alongside the axis of rotation. Their pistons undergo a constant radial centrifugal force, which must be balanced by an oil film pressure in order to avoid rubbing contact between contiguous metal surfaces.

The existence of an active oil film at sufficiently high pressure imposes a limitation on the centrifugal force of the masses, i.e. on the revolutions of the machine.

One can conclude by saying that axial piston pumps have a more favourable power-weight and power-volume ratio than the corresponding radial types. Consequently, they are operated at higher revolutions, but generally at lower pressure than the radial ones.

One should also bear in mind the following important aspects relative to the use, life and quality of piston pumps.

/Whenever ....

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* 1 bar is equavalent to about 1 kg/cm^2 in the technical system.

Whenever the maximum operating power of a piston pump is exceeded e.g. by an increase of the operation pressure, its life span is considerably shortened.

An increase in volumetric efficiency beyond certain practical limits considerably increases the manufacturing costs, and ultimately becomes uneconomical.

The 'quality' of a certain machine should be looked upon as a balanced compromise between efficiency, length of life and manufacturing costs at which a stated power is produced.

3. A REVIEW OF SOME EXISTING TYPES OF PISTON PUMPS.

In the attached Table No. 1 a number of volumetric piston pumps are described by means of their external parameters, i.e. power, speed, volumetric displacement pressure and mass.

The data in the table were collected from a survey of commercial pamphlets of various manufacturers mainly from Western Europe and the United States of America.

The table covers standard rather than special products, i.e. units which are used in typical industrial applications, which were made to comply with certain criteria of reliability, satisfactory service and commercial competitiveness.

The products surveyed have been listed in groups, each group relating to a specific manufacturer. Whenever applicable, each group was divided into two subgroups - one for fixed, the other for variable displacement pumps.

The axial piston pumps have been described first and the radial piston pumps last.

The values for power, pressure and speed refer to maximum continuous values in output.

The survey also reports characteristic parameters of some internal combustion engines (Table No. 2) used in aeroplanes as a useful means of comparison between two different classes of piston machines (see later development.)

The surveyed types of internal combustion engines cover units manufactured before 1942.

4. AN ANALYSIS BASED ON EXTERNAL PARAMETERS.

A criterion for the classification of positive displacement machines, and in particular for piston pumps, can be based on the concepts of power developed and machine mass required to produce this power.

Because power can be generated at different piston speeds, the forces of inertia of the reciprocating masses must also be taken into consideration for a true comparison.

Machines operating at different speeds can be reduced to a condition of standard speed, at which dynamic similarity is attained. Clearly no classification based on a "characteristic speed" (as used for centrifugal pumps and turbines) is possible for volumetric machines, because in these the kinetic energy of the fluid is completely wasted. In other words, volumetric machines are all "similar" to one another in this negative aspect of their performance.

If W (watt) is the power, U $(\frac{m^3}{rev})$ the displacement of a piston in one revolution, N the number of pistons, P $(\frac{N}{m^2})$ the fluid pressure, T (Nxm) the torque, V $(\frac{m^3}{rev})$ the total displacement and n $(\frac{rev}{s})$ the speed, one can write the following expressions :-

/For ....

For the power $W = PUNn = PVn$ .... (1)

For the torque $T = \frac{PUN}{2\pi}$ .... (2)

For the total displacement $V = NU$ .... (3)

A factor expressing the efficiency of the machine does not appear in these relationships because power torque and displacement are considered as net output*.

Introducing the piston stroke length l_o (m), the piston diameter d (m) and their ratio λ, in equation (1), one gets :

$$\lambda = \frac{l_o}{d} \qquad \text{..........} \quad (4)$$

and for the piston displacement

$$U = \frac{\pi d^2 l_o}{4} = \frac{\pi\lambda}{4} d^3$$

Equation (1) can also be written in the following form :

$$W = P \frac{\pi\lambda}{4} d^3 Nn \qquad \text{..........} \quad (1\ \text{bis})$$

Assuming that the pistons reciprocate in an harmonic motion, in which the axial displacement is l, then :

$$l = l_o \sin \omega t$$

where $\omega = 2\pi n$ is the angular velocity and n the speed of the machine.

The second derivative versus time, of the previous expression describes the acceleration; this multiplied by the mass m of the reciprocating parts indicates the force of inertia F, i.e.

$$F = -\omega^2 m\, d^3 \sin \omega t$$

/On ....

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* All the quantities listed in Tables 1 and 2 satisfy the previously written relationships (1) (2) (3)

On the assumption that m increases with the cube of the piston diameter d, one can express m as follows :

$$m = K_m d^3 \qquad \text{..........} \ (5)$$

where K_m $(\frac{kg}{m^3})$ is a parameter taking into account the density of the metal and the geometry of the linkage.

The maximum value of the force of inertia reached in a cycle is :

$$F = -\lambda_o\, \omega^2 K_m d^3 \qquad \text{..........} \ (6)$$

Introducing the concept of piston "inertial pressure" P_i $(\frac{N}{m^2})$, the ratio between the force of inertia and piston surface will be expressed as follows :

$$P_i = \frac{F}{\frac{\pi d^2}{4}} \qquad \text{..........} \ (7)$$

The combination of equations (4), (5) and (6) yields for P_i

$$P_i = \frac{-4 \quad \lambda \omega^2 K_m d^2}{\pi} \qquad \text{..........} \ (8)$$

Substituting ω with $2\,\pi n$ and neglecting the minus sign (equation 8) yields :

$$d = \frac{1}{4} \left(\frac{P_i}{K_m \lambda \pi}\right)^{\frac{1}{2}} \cdot n \qquad \text{..........} \ (9)$$

Elimination of d between equations (1 bis) and (9), and the grouping of various parameters into a symbol A (the internal parameter of inertia) yields :

$$W = \frac{AP}{n^2} \qquad \text{..........} \ (10)$$

/where ....

where $$A = \frac{1}{2,56} \; \frac{1}{\pi^{\frac{1}{2}}} \; \left(\frac{1}{\lambda}\right)^{\frac{1}{2}} \left(\frac{P_i}{K_m}\right)^{\frac{3}{2}} N \; \left(\frac{m}{s}\right)^3 \qquad \text{.......... (11)}$$

Introducing a factor α (internal factor of inertia), i.e. writing $A = \alpha A_o$ and taking into consideration equation (1) one gets :

$$Vn^3 = A = \alpha A_o \qquad \text{.......... (10 bis)}$$

Pumps, which operate in conditions of dynamical similarity are characterised by the same value of A, in particular for $\alpha = 1$ by the value A_o (where A_o is the internal parameter of inertia in standard reference conditions).

Let the unitary value of 1 $\left(\frac{m^3}{s^3}\right)$ be assigned to A_o.

This value then corresponds to a standard machine which operates at a unitary pressure of $P = 1 \left(\frac{N}{m^2}\right)$, at a unitary speed of $n = 1 \left(\frac{Rev}{s}\right)$, and develops a unitary power of W = 1 (watt).

According to equations (10) and (10 bis), this machine requires a total displacement of $V = 1 \left(\frac{m^3}{rev}\right)$.

Considering any other pump with a value $\alpha \neq 1$, it can be reduced immediately to standard conditions through the following manipulation of the equation (10 bis) :

$$\frac{V n^3}{\alpha} = \alpha^{\frac{1}{2}} V \frac{n^3}{\alpha^{\frac{3}{2}}} = A_o \qquad \text{.......... (10 ter)}$$

where $A_o = 1 \left(\frac{m}{s}\right)^3$

/This ....

This means that the dimensions of the pump must be increased with geometrical similarity until the displacement has reached the value $\alpha^{\frac{1}{2}}V$, and the speed reduced to $\frac{n}{\alpha^{\frac{1}{2}}}$ revolutions per second.

This transformation also implies that the condition of the total power produced does not vary.

Consequently equation (1) can be rewritten in the following manner :

$$\frac{W}{P} = Vn = \frac{\alpha^{\frac{1}{2}}Vn}{\alpha^{\frac{1}{2}}} \qquad \text{.......... (10ter)}$$

The ratio $W^* = \frac{W}{P}$ is herein defined as the "specific power" of the machine.

Let it be assumed that in the process of variation of the geometrical dimensions of the machine the new mass M*, herein called the "reduced mass", is proportional to the new value of the displacement $\alpha^{\frac{1}{2}}V$, which is a reasonable assumption.

The reduced mass M* is then related to the actual mass M by the relationship :

$$M^* = \alpha^{\frac{1}{2}}M \qquad \text{.......... (12)}$$

A graphical representation of the specific power $W^* = \frac{W}{P}$, plotted against the reduced mass $M^* = M \ \alpha^{\frac{1}{2}}$, is shown in Figure 5 for fixed displacement, and in Figure 6 for variable displacement pumps.

The corresponding numerical values have been grouped in Table 1.

In Figure 5 the values of specific power and mass for the internal combustion engines of Table 2 are also shown.

/Comparison ....

Comparison of the upper and lower diagrams of Figure 5 proves that for the same reduced mass an internal combustion engine generated a specific power which is about four and a half times greater than the equivalent of an oil piston pump of the axial type.

This interesting experimental finding can be explained as follows :

An internal combustion engine operates its pistons inside a stationary block. The side thrust of the pistons on the sliding surfaces is therefore small because they only reciprocate without revolving. Piston pumps on the contrary reciprocate their pistons inside a rotary block.

The rotation of the block produces large centrifugal forces onto the pistons, for which hydraulic support by an oil film must always be insured. Consequently an increase in displacement (i.e. in the dimensions of the rotary block) must comply with a decrease in rotational speed in order to keep the centrifugal forces within safe limits.

The characteristics of internal combustion engines, fixed displacement and variable displacement pumps with axial pistons, and variable displacement pumps with radial pistons are shown in Figures 5 and 6. They are plotted in groups, each of them corresponding to a certain class of machine characterised by an equation of the form :

$$W^* = KM^{*\frac{3}{4}} \qquad \text{..........} \quad (13)$$

Values of K are: $K = 4{,}5 \times 10^{-4}$; 1×10^{-4}; $0{,}8 \times 10^{-4}$; $0{,}3 \times 10^{-4}$ $(\frac{m^3}{skg})$, respectively.

/Whenever ....

Whenever a piston pump is used in the control of power involving speed inversion, the moment of inertia of its rotating masses is of great importance.

It is well known that the moment of inertia of piston pumps is very favourable, i.e. small, when compared with the moment of inertia of d.c. electrical machines of the same power.

The moments of inertia of two series of commercial pumps (manufacturers A and B) were available in the present survey and have been plotted as a power/moment of inertia representation in Figure 7.

5. CONCLUSIONS.

From an analysis of the external parameters of hydraulic piston pumps a standard classification was formulated on the concepts of specific power and reduced machine mass.

As external parameters one defined those of power, pressure, speed and displacement relative to a certain piston machine.

By conviently reducing the speed of a unit, the forces of inertia of the reciprocating parts were reduced to certain standard values, used as reference.

The displacement of each unit was then increased in an inverse proportion of the reduction in speed, so as to keep the initial power constant.

The reduced mass of the unit was calculated by multiplying the actual mass by the ratio between the new and the old displacements.

The specific power and the ratio between the power in output and fluid pressure was then defined.

/The ....

The various types of piston pumps were divided into the following groups, each group corresponding to a certain class, and each class characterised by a certain relationship between specific power W*and reduced mass M of the type :

$$W^* = KM^{*\frac{3}{4}}$$

With K a constant typical of the class as listed under :

(1) Axial piston pumps :

(1a) of fixed displacement ($K = 1 \times 10^{-4}\ (\frac{m^3}{skg})$)

(1b) of variable displacement ($K = 0{,}8 \times 10^{-4}(\frac{m^3}{skg})$)

(2) Radial piston pumps of variable displacement ($K = 0{,}3 \times 10^{-4}\ (\frac{m^3}{skg})$).

A group of internal combustion engines for aeroplanes was also surveyed for the purpose of useful comparison and their values of specific power and reduced mass included in the survey as a separate group (i.e.) :

(3) Four stroke internal combustion engines ($K = 4{,}5 \times 10^{-4}(\frac{m^3}{skg})$).

Finally the moments of inertia of the rotating masses were considered in relation to the machine power for two pump series of the axial piston types.

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PRETORIA.
6th January, 1972.
/KW

6. LIST OF SYMBOLS. (I.S.A. Unit used)

d	piston diameter	(m)
l_o	piston stroke	(m)
m	mass of reciprocating parts	(kg)
n	speed	$(\frac{rev}{s})$
A', A_o	Internal parameter of inertia and standard internal parameter of inertia, respectively	$(\frac{m}{s})^3$
F	force	(N)
K	parameter	$(\frac{m^3}{kg})$
k_m	parameter referred to the mass m of the reciprocating parts	$(\frac{Kg}{m^3})$
M, M^*	Mass of the machine and reduced mass of the machine respectively	(kg)
N	number of pistons	
J	moment of inertia	$(kg \times m^2)$
P	pressure	$(\frac{N}{m}\ 2)$
P_i	piston inertial pressure	$(\frac{N}{m^2})$
T	torque	$(N \times m)$
U	displacement per piston per revolution	$(\frac{m^3}{rev})$
V	total displacement per revolution	$(\frac{m^3}{rev})$
W	power	(watt)
W^*	specific power	$(\frac{watt}{\frac{N}{m^2}})$
α	internal factor of inertia	
λ	stroke - piston diameter ratio	
ω	angular velocity	$(\frac{rad}{s})$

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APPENDIX I (A WORKED EXAMPLE)

The following example of the purpose of the classification will illustrate the use of figures 5 and 6 and table 1.

In any preliminary investigation of the application of a hydraulic drive, two aspects have to be considered.

One of these is the coupling speed of the drive to its prime mover.

Prime mover and drive must be coupled in such a way that they operate under the best conditions of joint efficiency at both total and partial load (see reference 2).

The second aspect is the determination of the mass of the drive.

This aspect is of importance in applications to automotive and particulary to airborne vehicles.

The diagrams of Fig. 5 have proved that in the power range greater than 40 kwatt the weight of the drive generally exceeds the mass of the internal combustion engine of the same power.

Consequently, its application is possible only whenever mass limitations are not too restrictive.

In the range of small powers (say, less than 10 kwatt) for the same value of α , i.e. of the pressure of inertia on the reciprocating masses, the displacement V decreases with the power and consequently the revolutions n must increase.

/This is .....

This is in agreement with equation (10 bis) rewritten as:

$$\alpha = \frac{Vn^3}{A_o}$$

$$(A_o = 1\ (\frac{m}{s})^3)$$

Consequently, for α constant the drive becomes very compact (reduction of V) and very fast (increase of n) only in the range of small powers.

In this range it is very suitable to be connected to fast revolving shafts, and is more favourable in regard to lightness than the hydrokinetic drive.

In fact, the hydrokinetic drive, made of pump-turbine hollow impellers, has the section of the oil passages so reduced below a certain power, that its fillings volume becomes very small, and the weight-power ratio unfavourable.

Let us consider now a traction automotive vehicle, which is to be equipped with a hydraulic drive.

The drive is to be chosen from standard products on the market and preliminary information about its mass is required.

The prime mover is an internal combustion engine having a maximum power of 55×10^3 watt at 2 200 RPM, maximum torque of 306 Nxm at 1 200 RPM.

The characteristic of the I.C. engine is such that the torque is practically constant between 1200 and 2000 RPM.

The I.C. engine develops at 2000 RPM the power of 50×10^3 Watt.

In a preliminary study of the drive it will be assumed that its maximum efficiency at full displacement is $\eta = 0,90$, and its maximum continuous power equal

/to ....

to 50 x 10^3 Watt at 2000 RPM.

Two solutions will be considered.

a) Drive operated at an oil pressure of 100 bars, i.e. $P=10^7 \frac{N}{m^2}$

b) Drive operated at an oil pressure of 200 bars, i.e. $P=2\times10^7 \frac{N}{m^2}$

Moreover $\quad n = \frac{2\ 000}{60} = 33{,}3 \ \frac{rev}{s}$

From equation (10 ter)

Case (a) $\quad V = \frac{W}{Pn} = \frac{50 \times 10^3}{10^7 \times 33{,}3} = 1{,}5 \times 10^{-4} \ (\frac{m^3}{rev})$

Case (b) $\quad V = \frac{W}{Pn} = \frac{50 \times 10^3}{2\times10^7 \times 33{,}3} = 0{,}75 \times 10^{-4} \ (\frac{m^3}{rev})$

Introducing the constant $A_o = 1(\frac{m}{s})^3$ one gets

$$\alpha = \frac{Vn^3}{A_o} \quad \text{and} \quad W^* = \frac{W}{P}$$

in case (a)

$$\alpha = \frac{1{,}5 \times 10^4 \times 33{,}3^{-3}}{1} = 5{,}50$$

$$W^* = \frac{50 \times 10^3}{107} = 0{,}005 \quad (\frac{Watt}{\frac{N}{m^2}})$$

in case (b)

$$\alpha = \frac{0{,}75 \times 10^4 \times 33{,}3^{-2}}{1} = 2{,}75$$

$$W^* = \frac{50 \times 10^3}{2 \times 10^7} = 0{,}0025 \ (\frac{Watt}{\frac{N}{m^2}})$$

Case (a):
From Figure 5, the diagram of axial piston pumps with fixed displacement yields:

$$W^* = \frac{W}{P} = 0{,}005 \quad \left(\frac{\text{Watt}}{\frac{N}{m^2}}\right)$$

$$M^* = 180 \quad (\text{kg})$$

$$M_1 = \frac{M^*}{\alpha^{\frac{1}{2}}} = \frac{180}{2{,}35} = 76{,}5 \ (\text{kg})$$

Analogously, from Figure 6

$$W^* = \frac{W}{P} = 0{,}005 \left(\frac{\text{Watt}}{\frac{N}{m^2}}\right)$$

$$M^* = 275 \ (\text{kg})$$

$$M_2 = \frac{M^*}{\alpha^{\frac{1}{2}}} = \frac{275}{2{,}35} = 117 \ (\text{kg})$$

The total mass of the drive becomes

$$M_1 + M_2 = 117 + 76{,}5 = 193{,}5 \text{ kg}$$

Case (b):
From Figure 5, the diagram of axial piston pumps with fixed displacement yields:

$$W^* = \frac{W}{P} = 0{,}0025 \quad \left(\frac{\text{Watt}}{\frac{N}{m^2}}\right)$$

$$M^* = 70 \ (\text{kg})$$

$$M_1 = \frac{M^*}{\alpha^{\frac{1}{2}}} = \frac{70}{1{,}65} = 42 \quad (\text{kg})$$

analogously from Figure 6

$$W^* = \frac{W}{P} = 0{,}0025 \quad \left(\frac{\text{Watt}}{\frac{N}{m^2}}\right)$$

$$M^* = 100 \quad (\text{kg})$$

$$M_2 = \frac{M^*}{\alpha^{\frac{1}{2}}} = \frac{100}{1{,}675} = 60 \ (\text{kg})$$

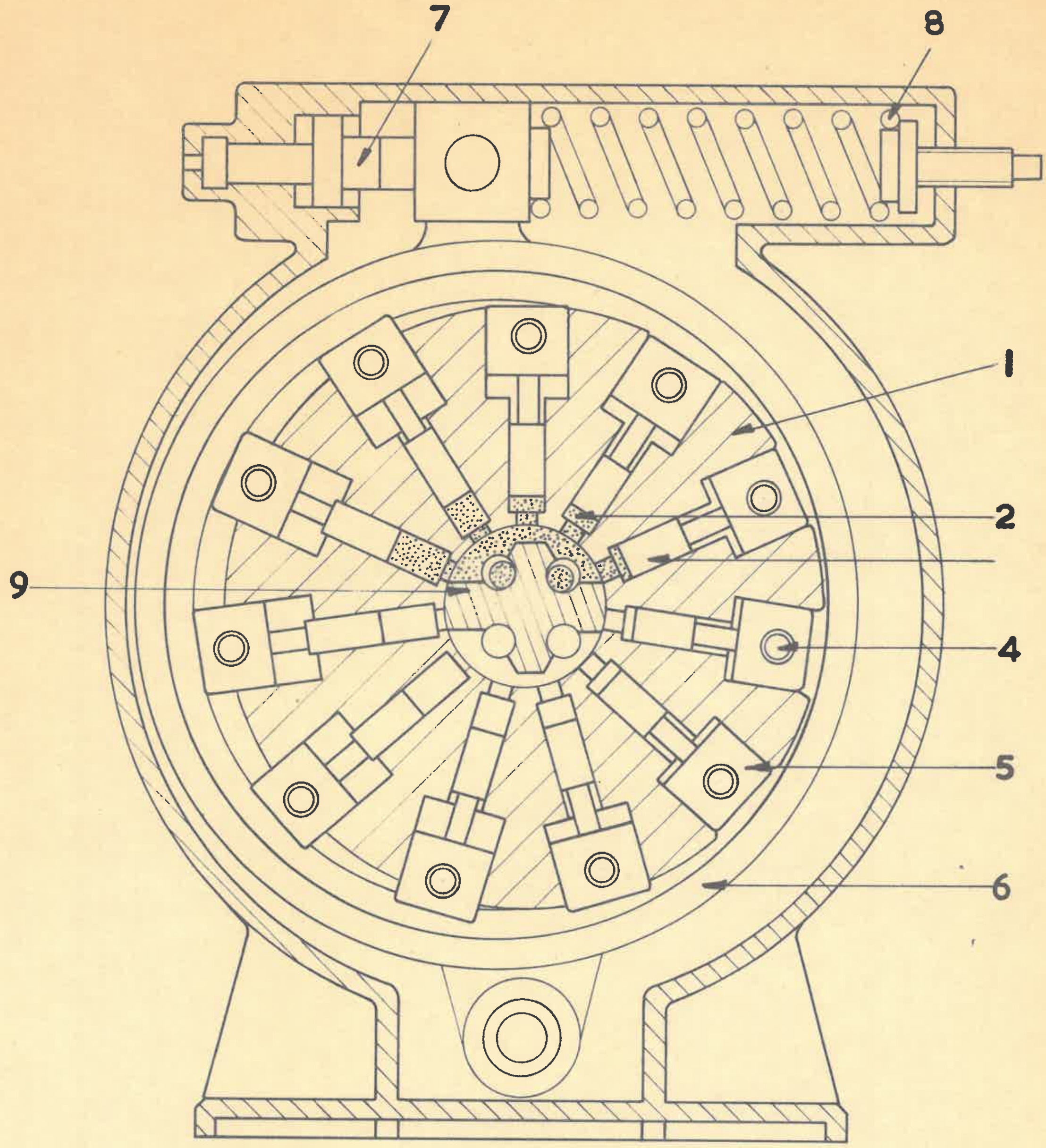
The total mass of the drive becomes

$$M = 42 + 60 = 102 \quad (\text{kg})$$

In describing the previous solution, one has tacitly assumed an arrangement made of a hydraulic pump with variable displacement coupled to the I.C. engine and that of a hydraulic motor displacement coupled to the differential of the vehicle.

Other solutions could have been considered, e.g. a hydraulic pump of variable displacement, driving two hydraulic motors coupled to the traction wheels of the vehicle, without differential, etc.

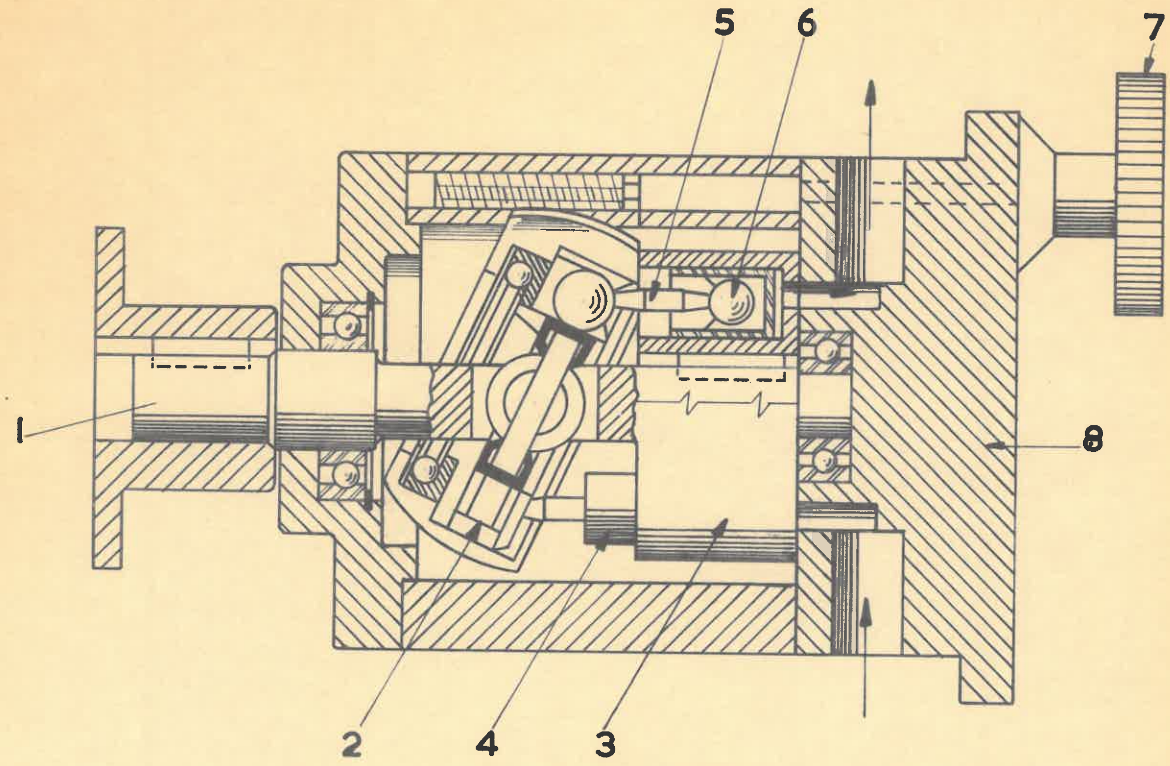
1	REVOLVING BLOCK	6	RING
2	RADIAL CYLINDER	7	CONTROL PISTON
3	PISTON	8	SPRING
4	PIN	9	FIXED AXLE
5	BLOCK		



RADIAL PISTON PUMP

FIG. 1

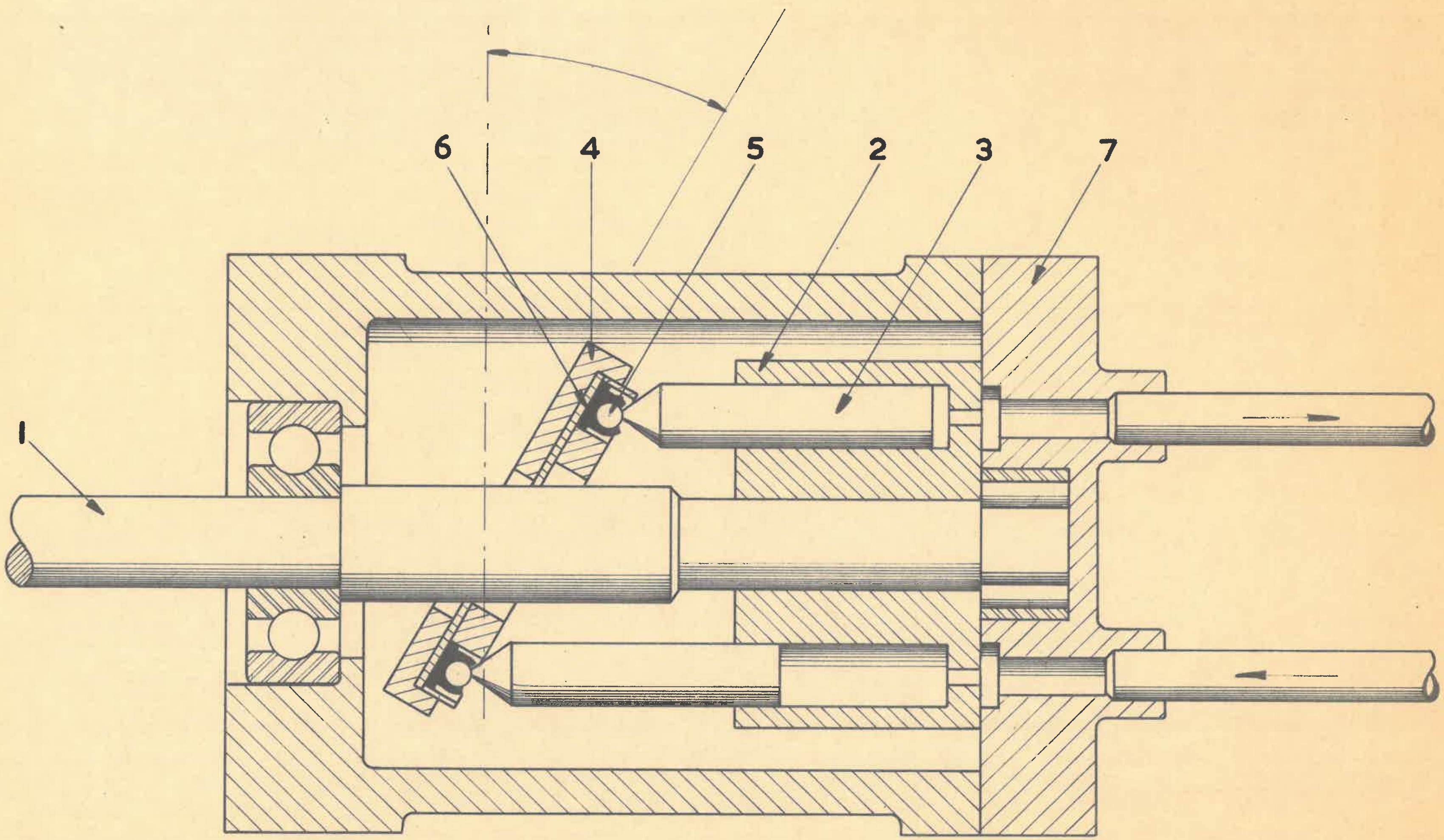
1	SHAFT	5	CONNECTING ROD
2	SWASH PLATE	6	SPHERICAL HEAD
3	PISTON BLOCK	7	ADJUSTING KNOB
4	PISTONS	8	DISTRIBUTOR



AXIAL PISTON PUMP (WILLIAMS & JANNEY) TYPE

FIG. 2

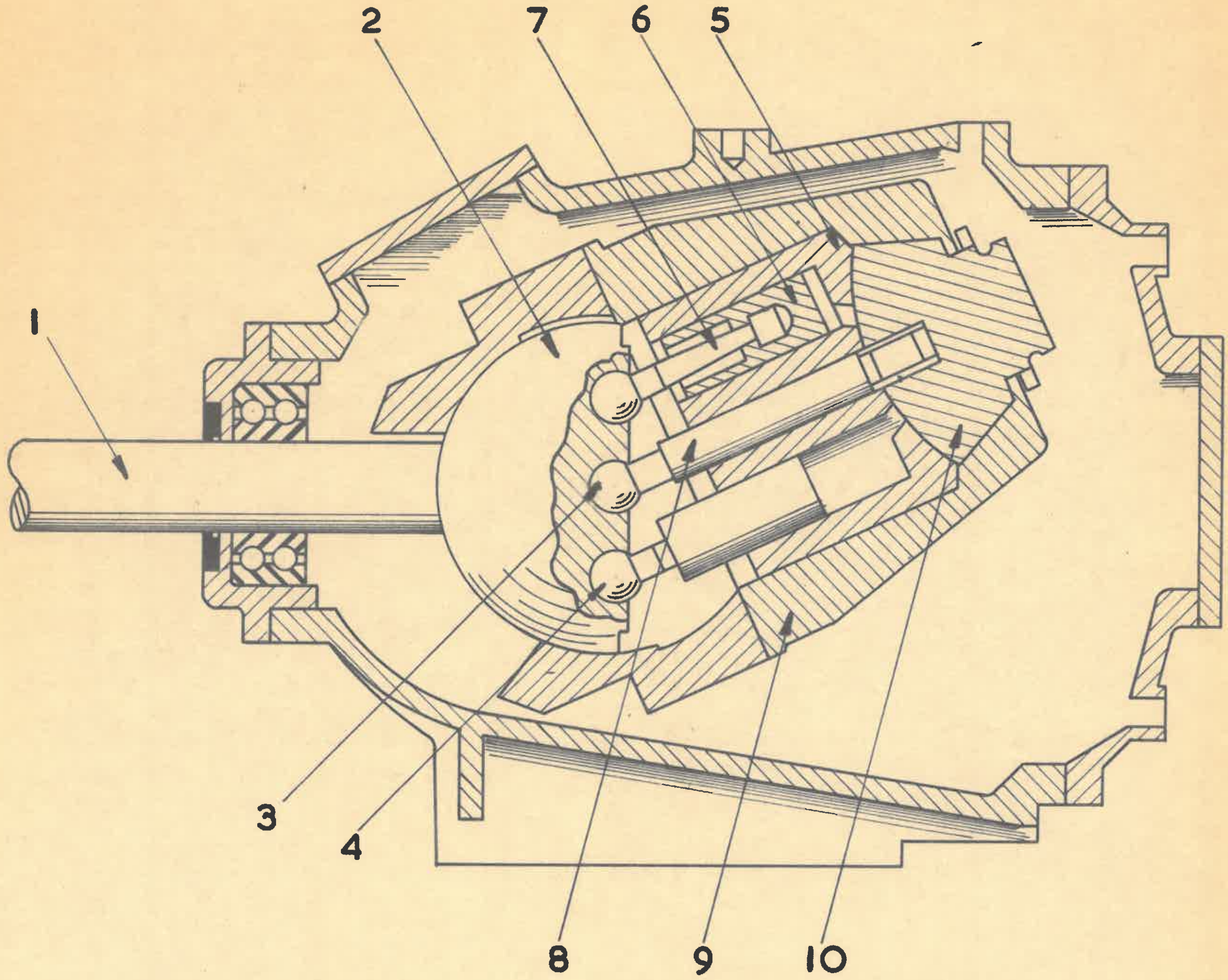
1 SHAFT
2 PISTON BLOCK
3 PISTON
4 FIXED PLATE
5 SPHERICAL JOINT
6 THRUST PAD
7 OIL DISTRIBUTION BLOCK



AXIAL PISTON PUMP WITH FIXED PLATE AND THRUST PADS

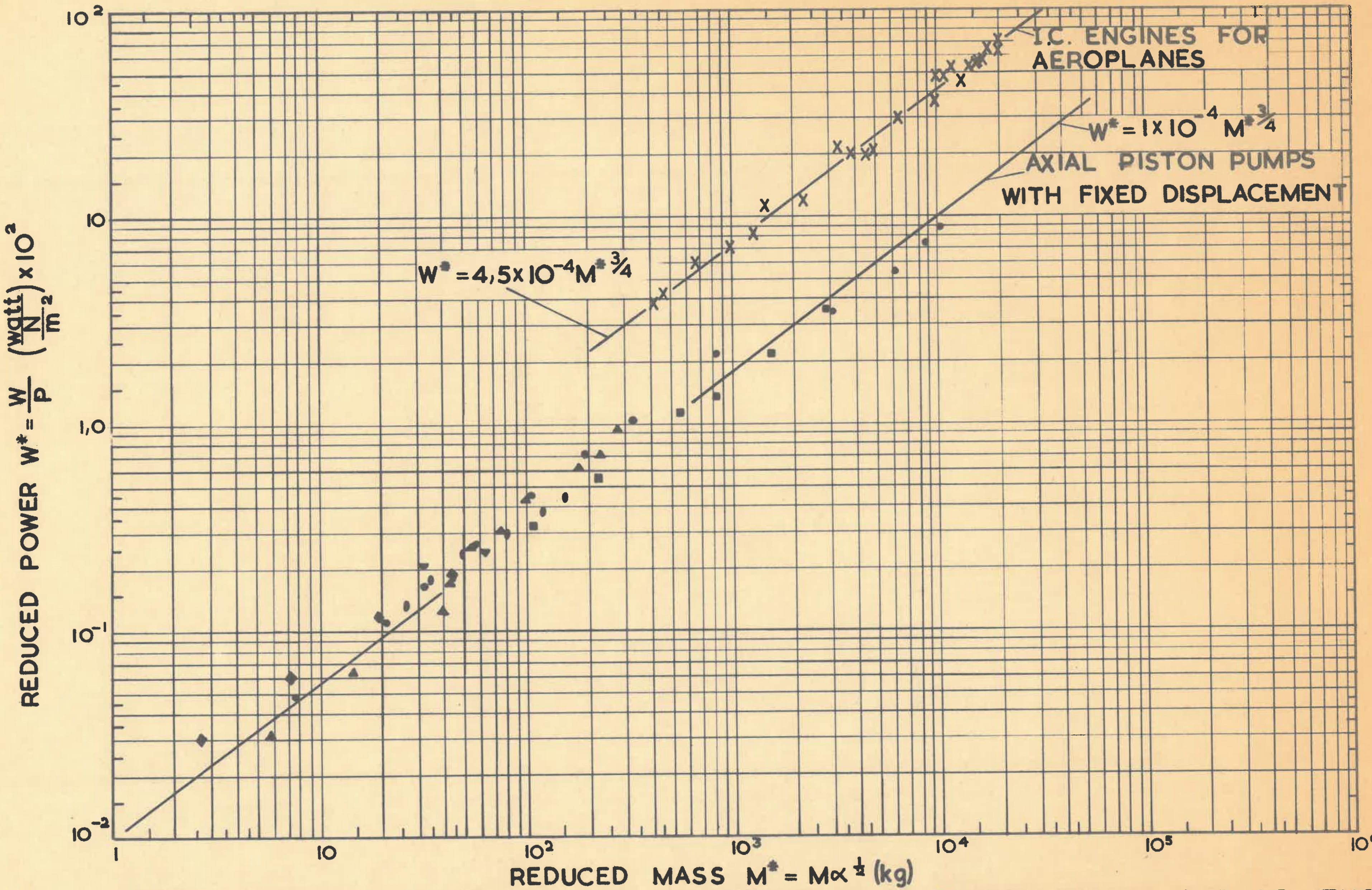
FIG. 3

1	SHAFT	6	PISTON
2	SEMISPHERICAL FLANGE	7	CONNECTING ROD
3	CENTRAL SOCKET	8	CENTRAL TRUNNION
4	SIDE SOCKET	9	INNER CASE
5	PISTON BLOCK	10	DISTRIBUTION BLOCK



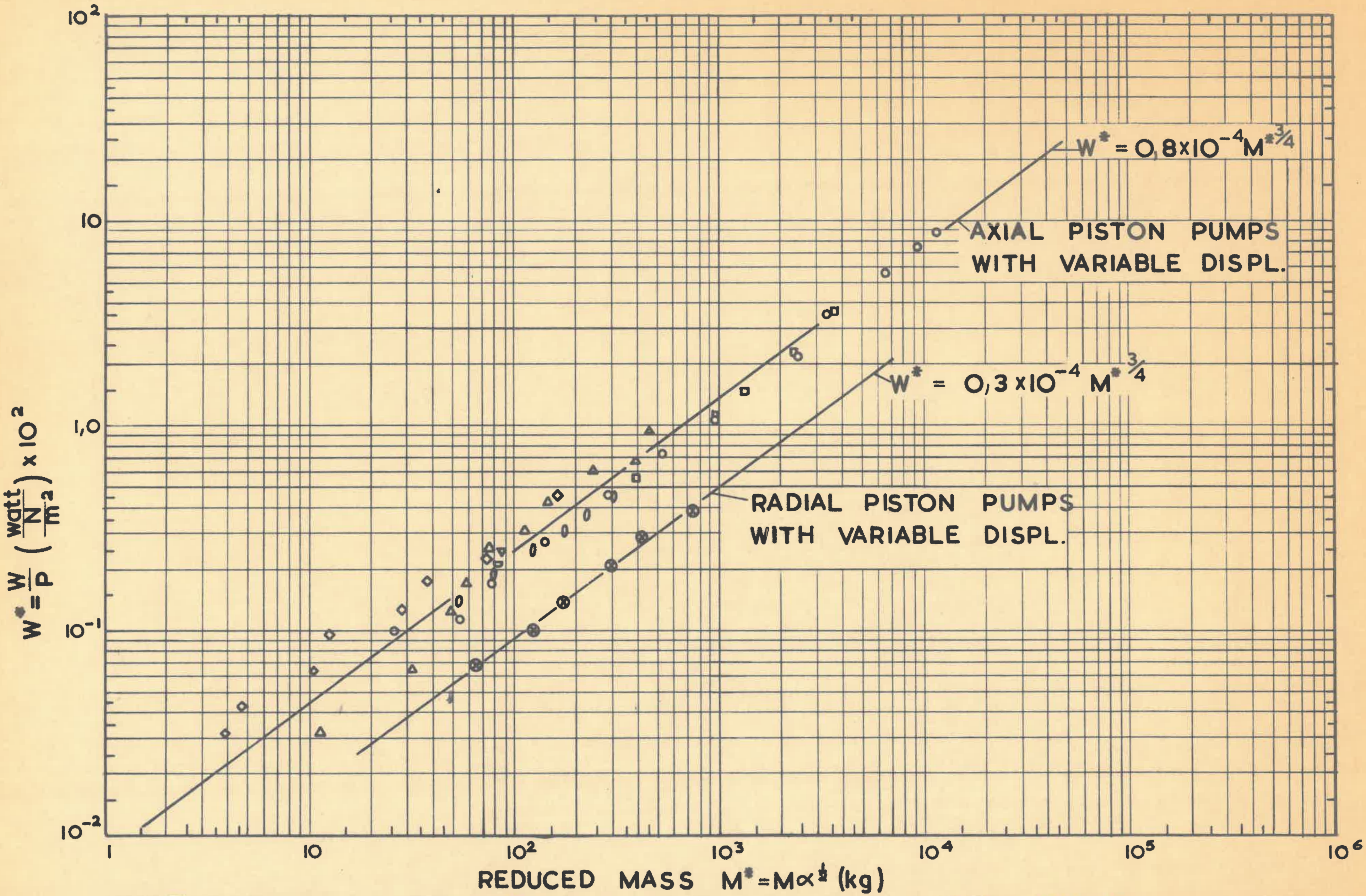
AXIAL PISTON PUMP WITH SWINGABLE BODY (THOMA TYPE)

FIG. 4



CHARACTERISTICS OF FIXED DISPLACEMENT AXIAL PISTON PUMPS AND I.C. ENGINE PLOTTED IN A REDUCED MASS REDUCED POWER REPRESENTATION

FIG. 5



CHARACTERISTICS OF VARIABLE DISPLACEMENT PISTON PUMPS PLOTTED IN A REDUCED MASS REDUCED POWER REPRESENTATION.

FIG 6