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# HIGH PRESSURE PUMPS FOR ROCKET MOTORS

by

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# High Pressure Pumps for Rocket Motors

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## SUMMARY

The special requirements which have to be met by rocket motor fuel pumps are described, and a comparative study is made of the suitability of all the important types of pumps for use with rocket motors.

A simplified type of open impeller centrifugal pump which has been operated successfully in some short and medium range rockets is recommended as the most suitable type at the present state of development. The theoretical and constructional details of this pump will be given later elsewhere.


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1. Fuel pumps
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I. Barske, U.M.
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1 Introduction

The simplest method of expelling the propellants in liquid propellant rocket motors is to pressurize the propellant tanks by a supply of compressed gas. Above a certain size of motor, however, design requirements for minimum weight can be satisfied better by a system embodying unpressurized tanks and propellant pumps driven by a prime mover of suitable construction. Besides the advantage of lower weight, unpressurized tanks can be fitted more easily into the general lay out of a projectile, as they need not be of the cylindrical or spherical shape dictated by considerations of strength. For a piloted aircraft propelled or assisted by a liquid propellant rocket motor non-pressurized tanks are obviously preferable for reasons of safety.

There is no doubt, however, that this most valuable decrease of weight must be paid for, from the point of view of operation and manufacture, by the increased complication of the fuel feeding system. Normally a fuel pump, an oxidant pump, a prime mover, a power supply for the prime mover, a starting device and a number of valves to control the system are needed. All these components should be of the simplest possible construction, as in many cases the rocket motor will be expendable.

In this note a number of designs which can be considered for use in rocket motors will be critically examined, and the most suitable design will be described and investigated in detail.

A rocket motor pump has to meet the following requirements:-

(a) High output

(b) Complete safety of operation

(c) Minimum weight and dimensions

(d) Simplicity

(e) Low manufacturing cost

In addition to meeting these specific requirements, it should be realised that the operating conditions include nearly every difficulty that can be encountered in pump practice, such as:-

(a) The liquids to be pumped, particularly the oxidants, are highly corrosive.

(b) Their viscosities differ widely and many of them have bad lubricating properties.

(c) The temperatures of some liquids may be close to their boiling points.

(d) Particularly efficacious shaft seals are required to prevent leakage of fuel and oxidant from mixing.

(e) The pumps may have to undergo rapid and large changes of temperature.

(f) The pumps must start rapidly, be easily controlled and withstand rough handling.

It is clear that it is most difficult to find or to develop a type of pump to fulfil all these conditions. The problem, however, is mitigated by the fact that the operating time does not generally exceed a few minutes, and that an efficiency somewhat lower than that of a first class commercial pump will suffice. The pump will be driven by a high speed single stage impulse turbine, and it is, therefore, desirable to select a type of pump which can be coupled directly to it. This will have the accompanying advantage of reducing the size and weight of the pump itself. The pumps which will be considered fall into two main groups, positive displacement pumps and centrifugal pumps; the former are considered first.

## 2 Positive displacement pumps

### 2.1 General features

In positive displacement pumps pressure is generated by a solid body movable in the pump casing and acting directly upon the liquid. Relatively small mechanical and hydraulic friction losses are involved in this process, and provided that the movable body - the piston - is very well sealed against the casing, it follows that this type of pump is, in principle, suitable for generating a high pressure with good efficiency. Generally, a soft or elastic packing would not be suitable for sealing the piston against the high pressure occurring in rocket motor pumps, and a good seal can only be ensured by a good fit between piston and pump casing; the clearance permissible can be increased in proportion to the viscosity of the liquid pumped.

The fine finish required for the working faces and for the small clearances needed between them are dominant features from which are derived the characteristic properties of positive displacement pumps, namely:-

(a) High cost of manufacture

(b) Sensitivity to distortion whether caused by mechanical forces or unequal thermal expansion

(c) Ability to handle clean liquids only

(d) Inability to handle extremely non-viscous liquids

Most of the propellants used in rocket motors, particularly the oxidants, are insufficiently viscous to ensure lubrication. An oxidant pump, moreover, has to be made of chemically resistant materials, such as stainless steel or pure aluminium, which have very bad sliding properties. Sufficient lubrication to prevent seizure of the moving parts of the pumps cannot be guaranteed by such liquids as H.T.P. nitric acid or liquid oxygen.

All types of positive displacement pumps are self priming. This favourable feature, however, is not so important in rocket motors, as the feed system is generally arranged so that the propellants are fed to the pumps by gravity or by slightly pressurizing the tanks.

### 2.2 Piston pumps

A piston can be fitted into the cylinder in which it slides more accurately than most other combinations of moving and stationary parts, and with relatively moderate manufacturing cost. It is, therefore, preferable to use piston pumps for very high pressures. Their method of operation, based on the reciprocating movement of a single acting piston, involves some complications which are undesirable in rocket motors. To obtain uniform flow of the liquid and reasonably good balance of the reciprocating masses a number of pistons have to be embodied in one pump, but this has the advantage of reducing the forces to be applied, as obviously if one piston were used a very large applied force would be required for producing a high pressure. With this type of pump a mechanical means of converting the rotatory movement of the shaft to the oscillating movement of the piston has to be provided.

For modern pumps the well-known crank mechanism has been replaced by simpler and more compact devices, but it cannot be denied that the whole design is somewhat complicated. The number of movable parts is further increased by the inlet and outlet valves or sliding valve gear required for each piston. This arrangement is simplified in some modern pumps by using a rotating block to contain all the cylinders and pistons which are moved relatively to one stationary inlet passage and one discharge passage common to all cylinders. For this design, however, relatively large plane or cylindrical faces have to be sealed continuously against the full pressure.

The possibility of reducing the weight and dimensions of a pump for a given output by increasing the speed is very limited, because the mean velocity of the pistons, which is a measure of the acceleration imparted to the liquid during each suction stroke, must not exceed a certain value, which decreases as the difference between the operating temperature and the boiling point of the liquid is reduced; consequently high speeds cannot be used and piston pumps cannot be coupled directly to a high speed prime mover. The reduction gear needed for speed conversion represents another increase in cost, weight and space occupied.

2.3 Gear pumps

This type of pump is of very simple design, as shown in Fig.1. The two intermeshing toothed wheels are the only movable parts. As the rotational speed of the gears is constant no masses have to be balanced, the pump runs smoothly and a constant delivery is obtained; at the same time neither valves nor valve gear are needed. This remarkable simplicity would make the gear pump most suitable for rocket motors, but unfortunately the sealing and manufacturing problems are still more difficult than in piston pumps. The suction side and the pressure side have not only to be separated directly from each other by the meshing teeth, but also the end faces and the tips of the teeth of both gears have to fit closely to the casing. This means that all faces of the gears are sealing faces, and at least one shaft seal is required in addition to this. The various possibilities of leakage are clearly shown in Fig.1.

The speed which can be applied to gear pumps is limited because a good volumetric efficiency cannot be obtained beyond a certain peripheral velocity of the teeth. At high speeds, both the lack of time and the centrifugal action of the teeth will prevent the liquid from filling the spaces of the gears sufficiently. In addition to this the danger of seizure increases considerably at high speeds. Thus the same undesirable influence of speed limitation on weight, dimensions and method of drive as discussed in the section on piston pumps applies to gear pumps.

2.4 Helical or screw pumps

The longitudinal section of a screw pump is shown in Fig.2. The general arrangement is similar to a gear pump as only two moving parts rotating at a constant speed are included in this design. The problems of sealing, and consequently of manufacture, are still more difficult than those of gear pumps. Both screws are in contact with each other along their full lengths, which means that the core faces of one helix have to be in mesh with the external faces of the other one, and vice versa; furthermore all helical faces of both helices are in contact with each other along the plane of meshing, the external faces have to seal against the casing, and the two balancing pistons for eliminating end thrusts have to seal against full pressure. The correct mutual location of the two screws is ensured by two gear wheels, which are integral with the shafts and have to mesh together extremely accurately. It is obvious that the manufacture of all components is extremely difficult and only very few manufacturers produce this type of pump for high pressures. Since it is practically impossible to obtain high accuracy and hence a close fit, the screws tend to operate as labyrinths, and a larger number of threads, i.e. long screws, are required for high pressures. Such pumps, therefore, must have a considerable axial length and a correspondingly greater weight.

The conditions of liquid flow are more favourable than in gear pumps. The liquid passes through the pump in the axial direction and no centrifugal action of the screws will occur to prevent the liquid from filling the threads. Hence, from this point of view, this type of pump should be more suitable for high speeds than gear pumps. It is not known, however, whether screw pumps have ever been operated as liquid pumps at more than 2,900 r.p.m.

### 2.5 Another type of twin rotor pump

Fig.3 shows the longitudinal and cross sections of a type of rotary high pressure displacement pump which includes two rotors. The action of the rotors is illustrated diagrammatically in Fig.4, and it will be understood easily that the liquid is delivered by only one rotor, disc A, and the displacing members B. The rotor C merely acts as a sealing member, and D is a static part fixed to the casing. The pumping rotor AB is fixed to the driving shaft whereas the shaft of the sealing rotor C is driven by a pair of toothed wheels (Fig.3). In this type of pump, which is being manufactured to various designs, the suction side and the pressure side are continuously separated by area seals whereas only a line contact is obtained between the teeth of a gear pump or the screws of a screw pump. The machining requirements are the same as for gear pumps, since almost all the faces of the rotors are sealing faces.

These pumps are operated at speeds up to 3,000 r.p.m. At high speeds the volumetric efficiency would probably be decreased by the centrifugal effect of the rotors, as in gear pumps.

### 2.6 Vane pumps

The different designs of vane pumps are all based on the arrangement diagrammatically shown by the cross section (Fig.5). Only one rotary member is included in this pump. The design is more compact and the machining of the casing is simpler than that of the twin rotor pumps previously described. The number of movable parts, however, is increased and the total sealing face is probably not less than in a corresponding twin rotor pump, both end faces of the rotor and nearly all faces of the vanes being sealing areas.

As in gear pumps the bearings are highly loaded because the full liquid pressure bears upon one side of the rotor. Still more disadvantageous is the unbalanced liquid pressure bearing upon the vanes as they pass through the positions separating the pressure side from the suction side. Much friction and corresponding wear of the sealing faces is caused by this load which increases in direct proportion to the pressure. Additional friction obtains at higher speed since the centrifugal forces of the sliding vanes act directly upon the cylindrical face of the casing or particular guiding faces in some designs of vane pumps. At the same time high speeds cannot be applied since the centrifugal effect of the vanes will prevent the chambers from being filled correctly, as in gear pumps.

3 Centrifugal pumps

3.1 General features

In contrast to positive displacement pumps which have a direct action, centrifugal pumps generate pressure indirectly. The static pressure is merely a consequence of the kinetic energy imparted to the liquid by the vanes of the rotary impeller. In other words the generation of static pressure is necessarily dependent on the velocity of the liquid. Velocity, however, is always affected by the energy losses due to turbulence and friction. Another unfavourable feature of this "indirect" process is that the velocity of the liquid leaving the impeller vanes is generally higher than that required for discharge. Different designs of diffusers are provided for decelerating the liquid, but owing to turbulence and friction only a certain fraction of the kinetic energy of the liquid can be recovered for conversion into static pressure by this deceleration. In consequence of these inevitable losses the "indirect" or dynamic generation of pressure must be less efficient than the "direct" or static method.

This operating process, though complicated and imperfect is, however, compensated by the simple construction of the pump which includes only one rotary member. The sealing problems, which will be more closely considered later, are not so difficult as in displacement pumps.

From the above reasoning it can be concluded that the best efficiencies will be obtained at low speeds and correspondingly low pressures. On the other hand large rates of flow can be handled easily because of the uniform flow of the liquid through the passages of the pumps.

3.2 Centrifugal pumps of conventional design (shrouded impeller pumps)

3.21 Single stage low pressure pumps

The simple construction of a single stage centrifugal pump is illustrated in Fig.6. The impeller is of conventional design, i.e. it consists of two rotating shrouds with the curved blades fixed between them. Both shrouds are sealed against the casing by means of two cylindrical labyrinth seals of equal diameters for preventing the liquid inside the chambers a and b from flowing back to the suction side. This liquid is not under full pump pressure, but under static centrifugal pressure. The kinetic energy of the liquid is partly converted into static pressure in the volute which surrounds the periphery of the impeller and discharges into the pressure line. The impeller shaft extends on both sides through the walls of the casing and is sealed by two glands.

In other designs both bearings are located on the same side of the casing and the impeller is overhung like a fly-wheel; only one shaft seal is required.

The manufacture of this type of pump does not present much difficulty. It is important to have very small clearances in the labyrinth seals, and, therefore, an accurate alignment of the shaft and the impeller relative to the casing is required. It is essential to minimize the deflection of the shaft due to unbalance of the impeller or vibrations. The pump impeller is generally cast and both the inner surfaces of the shrouds and the blade surfaces have to be smooth finished, usually by hand, in order to obtain minimum friction loss.

In commercial and marine practice where long life and high efficiency are most important, the pressures developed by this type of pump are generally limited to 40 or 50 lb/sq in. It is rather exceptional for pressures of 100 lb/sq in. to be reached in one stage because the efficiency decreases owing to both the higher peripheral velocity of the impeller and the increase of internal leakage through the labyrinth seals.

3.22 Multi-stage high pressure pumps

For the discharge pressures of 500 lb/sq in, or higher, which are required for feeding propellants to the combustion chamber of a rocket motor a multi-stage pump would be used if normal practice were followed. This type of pump, an example of which is shown in Fig.7, includes only one rotary member, but the simplicity of the single stage pump is lost. The number of internal seals increases in proportion to the number of impellers, and means have to be provided for balancing the considerable end thrust involved in this design. In the example shown in Fig.7 a balancing disc provided with a peripheral seal is fixed to the shaft. The pressure difference across this seal almost reaches the full discharge pressure, and a considerable internal leakage occurs constantly during operation. This pump has the further disadvantage that the long shaft loaded by the masses of the impellers is susceptible to vibrations and deflections; it is also difficult to maintain the proper alignment of the impellers and the sealing faces relative to the casing. The axial positions of the impellers would be seriously affected by differential thermal expansion of the shaft and the casing which is likely to occur in turbine driven pumps.

In about 1940 some multi-stage high pressure pumps for handling H.T.P., C-fuel and water were built for the Walter submarine boost plants. The duties of the pumps were similar to those of a medium size rocket motor. Their operation as well as their weights and dimensions were not satisfactory, and they were soon replaced by a new type of single stage pump which will be described later.

Another development of multi-stage rocket propellant feed pumps for handling nitric acid and fuel was started in America[1] (1944). Up to seven stages have been used in these pumps; the weights and dimensions are, however, not satisfactory. The operational properties are reported to have met the requirements, but the efficiencies claimed seem to be very optimistic; at any rate these pumps have not been produced on a larger scale for further development of rocket motors.

3.23 Single stage high pressure pumps

Since multi-stage centrifugal pumps have the unsatisfactory properties described in the previous section, it was thought desirable to develop single stage pumps which would be capable of generating the high pressures required in rocket motors.

In theory this can be achieved simply by operating the pump impeller at an appropriately high peripheral velocity, but the hydraulic losses previously mentioned such as internal leakage, turbulence and friction in the passages of impeller and diffuser, and external fluid friction of the impeller shrouds will increase considerably.

Theoretically it is possible to compensate for the influence of the higher pressure difference across the labyrinth seals by reducing the radial clearances but it is found in practice that these clearances should be made wider than usual in order to reduce the increased danger of seizing up at higher speeds. No improvement can be obtained by reducing the diameter of the seals since this dimension is fixed by the area required for the inlet, but an alternative method would be to provide labyrinth seals of greater axial length. This unfortunately is also impracticable because accurate alignment of these extended faces would be more difficult, and the increased weight of the impeller as well as the increased dimensions of the pump would not be desirable; furthermore, the fluid friction losses of the impeller would be increased.

The sealing areas are subjected to considerable wear due to erosion caused by the high rate of flow across the seals, and even very hard materials cannot withstand this action for long, especially if the liquid is not very clean; thus the internal leakage increases constantly during operation.

The main effect of turbulence and fluid friction occurring in the impeller and diffuser passages is to reduce the efficiency of conversion of the high kinetic energy of the liquid into static pressure. The diffuser efficiency is rather low, and the energy losses can be kept within limits only by careful design of the impeller and diffuser vanes and a good hand-finish of the surfaces in contact with liquid flow. The rate of flow, however, is so high that it will very soon destroy the surfaces by erosion, particularly those of the diffuser vanes, and it also increases the danger of cavitation.

In contrast to the losses already considered the external friction loss of the impeller can be extensively reduced by careful designing. This can be shown, without going into physical and mathematical details, by the following treatment:

As mentioned above, a given delivery pressure $p$ lb/sq ft requires a certain peripheral velocity of the impeller $v$ ft/sec according to the equation:

$$p = k_1 \cdot v^2,$$

where $k_1$ is a constant.

On the other hand, the peripheral velocity can be expressed as the product of $r$ the impeller radius in feet and $n$ the number of revolutions per second,

$$v = k_2 \cdot r \cdot n,$$

where $k_2$ is another constant.

Hence

$$p = k_3 \cdot r^2 \cdot n^2 \qquad \text{.....(1)}$$

where $k_3 = k_1 \cdot k_2^2$

For a given fluid, the friction loss $F$ of the impeller also depends on the two factors $r$ and $n$, the approximate relationship being:

$$F = k_4 \cdot r^{4.6} \cdot n^{2.8}$$

where $k_4$ is a constant.

From this equation

$$n^2 = \sqrt[1.4]{\frac{F}{k_4 \cdot r^{4.6}}} .$$

Substituting this in equation (1)

$$p = k_3 \cdot r^2 \cdot \frac{F^{0.715}}{k_4^{0.715} \cdot r^{3.29}} ,$$

and

$$p = k \cdot \frac{F^{0.715}}{r^{1.29}} , \qquad \ldots.(2)$$

where

$$k = k_3 \cdot k_4^{-0.715} .$$

Equations (1) and (2) give the pressure $p$ as a function of $r$ and $n$, and $r$ and $F$, respectively.

In Fig.8 two corresponding families of curves are obtained by plotting $p$ against $r$ for different values of $n$ and $F$ respectively. Since both axes have logarithmic scales these two sets of curves are straight lines. The values for $r$, $p$, $n$ and $F$ are given in inches, lb/sq in, r.p.m. and H.P. respectively since these units are more suitable for practical use.

In order to show the main relationships as clearly as possible equations (1) and (2) are simplified, and the relatively small influence of the inlet diameter on the results is neglected. The following assumptions are also made:-

(a) The impeller is considered as a plane disc, the cylindrical periphery of which has an axial length of 0.1r,

(b) The liquid pumped is assumed to be water,

(c) The pressures $p$ are taken to be 65% of the theoretical pressures throughout in order to allow for the hydraulic losses.

The rates of flow are not considered of importance in this investigation.

Let us consider now a single stage low pressure pump producing a pressure $p = 100$ lb/sq in. at a speed of $n = 3{,}000$ r.p.m. These values are represented in Fig.8 by point A which shows that the impeller radius should be

$$r = 4.25 \text{ inches}$$

and that this impeller would have a friction loss of

$$F = 4.2 \text{ H.P.}$$

According to normal pump practice, a pressure of $p = 500$ lb/sq in, as usually required in rocket motors, could be produced by arranging 5 impellers of this type in series; this five-stage pump would have a friction loss of at least:

$$F = 5 \times 4.2 = 21 \text{ H.P.}$$

As already stated, however, a multi-stage pump would not be desirable, and different methods would have to be tried for obtaining the pressure of 500 lb/sq in, with a single stage impeller.

The first possibility is to run the impeller indicated by point A at a higher speed; this brings us to point B ($r = 4.25$ inches, $p = 500$ lb/sq in) denoting a speed of about 6,700 r.p.m., and an increased friction loss $F = 39$ H.P. which is considerably higher than that of the five-stage pump.

Another possibility would be to keep the speed of the impeller denoted by point A constant ($n = 3,000$ r.p.m.). This leads to point C which proves completely unsuitable since the impeller radius increases to $r = 9.5$ inches, and the friction loss nearly amounts to $F = 170$ H.P.

A single stage impeller with the same friction loss as the five-stage pump mentioned above ($F = 21$ H.P.) is represented by point D. An impeller radius of about 3 inches and a speed of about 9,500 r.p.m. would be required.

The friction loss can be reduced to a still lower figure by moving from point D nearer to the ordinate, for example, as far as point E, which denotes a friction loss equal to that denoted by point A ($F = 4.2$ H.P). For these conditions the impeller radius is reduced to $r = 1.21$ inches, but the speed has increased to $n = 23,500$ r.p.m.

These examples show clearly that the friction losses of single stage high pressure pumps can be kept within reasonable limits by operating at very high speeds. At the same time other important requirements would be satisfied; for instance, dimensions and weight would be small and the pump could be coupled directly to a high speed turbine.

On the other hand, new problems arise due to the effect of the high velocity of the impeller blades on the inlet conditions of the pump. For rotational speeds of the order indicated above, this velocity greatly exceeds those encountered in normal practice, and cavitation is more likely to occur, particularly at the entrance sections of the impeller blades. Furthermore, suitable designs of high speed shaft seals have to be developed, as well as high speed bearings which may have to be considered for the prime mover.

Obviously these difficulties have very much restricted the development of high speed, high pressure pumps in commercial practice. Pressures of about 280 lb/sq.in used in feed pumps of power plants are definitely the maximum reached in one stage. As, however, the operating times of rocket motors are very short and high efficiency is a minor consideration the drawbacks can be accepted in order to obtain the great advantages of simplicity, low weight and small size.

A number of single stage high pressure centrifugal pumps for rocket motors have, in fact, been designed and have operated successfully. Fig.9 shows an example of a turbo-pump unit[2] consisting of a high speed turbine directly coupled to an oxidant pump and a fuel pump, both pumps incorporating all the features of conventional practice. Cavitation, which would seriously affect the safety of operation, has been avoided by arranging two helical boost pumps coaxially with the pump impellers. In other designs slight pressurization of the tanks may be sufficient to prevent cavitation at the pump inlets.

3.3 New designs of single stage high pressure pumps

3.31 Rotating casing pump (scoop wing pump)

Independently of the developments which took place in the specialized industries, in 1937 the author designed a novel type of centrifugal pump, an example of which is shown in Fig.10. The pump impeller is represented by a closed casing provided on one side with a central orifice which is connected with the inlet duct by a seal, e.g. a rubbing ring seal. The opposite side of the casing is fixed to the shaft of the prime mover. Rotational movement is imparted to the liquid mainly by some internal radial vanes, but also by the inner surface of the casing. The pressurized liquid is picked up by the "scoop wing" which is a stationary member of streamlined shape connected to, or integral with, the pressure tube, and projecting radially into the impeller. The liquid enters the internal passage of the scoop wing through a small orifice situated on the leading edge and as close as possible to the outer end. The pressure tube is coaxial with the inlet duct and fixed to the outer stationary casing which surrounds the impeller and is fixed to the casing of the prime mover. For venting purposes the centre part of the impeller can be connected with the open air by means of a small passage which is closed by a shut-off valve during operation. This venting is very important because the pump cannot be operated effectively unless the impeller is completely filled with liquid.

Inside the impeller there is so much space that the velocity of the liquid relative to the inner surface of the impeller is very small, and there are practically no losses in the radial flow of the liquid. The fluid friction occurring between the rotating liquid and the stationary scoop wing is kept within reasonable limits by the streamlined shape and the very smooth surface of the scoop.

Another essential and specific feature in the operation of the pump is the conversion of the kinetic energy of the liquid into static pressure by the ram effect occurring at the pick-up orifice of the scoop wing. The efficiency of this pressure conversion is very much better than that of the diffuser passages of any centrifugal pump. At zero flow, for example, this type of pump develops 95% or more of the theoretical pressure corresponding to the peripheral velocity of the liquid. This pressure drops slowly as the liquid starts flowing through the passages of the wing and the pressure pipe.

The manufacture of the pump is cheap and easy owing to its simple design which, in addition, does not include any narrow clearances; no fine finishing of surfaces is required except the outside of the scoop wing which can be easily polished.

The safety of operation is perfect because there is no danger of wear and seizure at high speed and correspondingly high pressure. The seal is always subject to inlet pressure only, which is usually very low. No axial thrust obtains under operating conditions as the hydraulic pressures are balanced.

The above consideration suggests that this type of pump is most suitable for rocket motors, but there is one serious drawback which prevents it from being utilized for this purpose on a larger scale. The dimensions are favourable only for rates of flow of liquid of less than 1 lb/sec. This rate of flow can, however, be doubled by putting two scoop wings, opposite to each other, into one impeller. On the other hand higher rates of flow require an increase in width of the wings, and consequently of the rotating casing which would, therefore, become too bulky.

The pressure which can be generated in this type of single stage pump is limited by the maximum attainable mechanical strength of the hollow impeller since it is subjected to the internal centrifugal pressure of the liquid in addition to its own centrifugal force; this limit is, however, far above the pressures required in present designs of rocket motor pumps.

A design of a rotating casing pump which allows a considerable increase in the rates of flow is shown diagrammatically in Fig.11. The stationary section inside the rotating casing consists of a circular disc which has a smooth surface. Several streamlined pick-up heads of short radial length project beyond the periphery of the disc, and the liquid passes to the central pressure pipe via internal radial passages. The higher rate of flow is obtained from the increased number of pick-up orifices. As in the pump described above the kinetic energy of the rotating liquid has to be converted into static pressure mainly by the ram effect. In addition to this some kinetic energy may be recovered in the radial passages which are shaped like diffusers. The conical side walls of the rotating casing are so arranged that there is a wider space at the periphery for the liquid to pass the pick-up heads. The clearance between both sides of the circular disc and the radial vanes of the impeller is relatively small, but much wider in the region of the streamlined heads. An air bleed ensures that the rotating casing is completely filled with liquid on both sides of the stationary disc.

As is also shown in Fig.11 a very compact unit of pump and prime mover can be designed by providing the outer periphery of the casing of a rotating casing pump with turbine blades. The stationary casing which encloses the turbo-pump unit includes the turbine nozzle or nozzles and the exhaust chamber. A unit of this design can be advantageously used as a high pressure generator for operating auxiliary rocket equipment, such as a hydraulic governing device. Air would be particularly suitable as the driving medium since there would be, in this case, no danger of the liquid inside the pump being heated by the turbine exhaust, and problems of sealing the stationary casing against the rotor would not arise.

No pump of the type shown in Fig.11 has yet been built, but in the development of single stage high pressure pumps it represents the theoretical starting point of a new type which has proved very important and successful. This type will be described in the following section.

3.32 Open impeller pumps

From a comparison of the pump shown in Fig.12 with the preceding design it is obvious that the new pump is a variant of the older design. As can be seen in Fig.12 the pump casing does not rotate as formerly, but remains stationary. The pick-up heads are fixed to the inner periphery of the casing and project radially inwards. The diffuser passages are tangential to the core of the casing which is concentric with the impeller. The diffusers discharge either directly into the pressure pipe or, as shown in this example, into an annular pressure chamber surrounding the casing and connected to the pressure pipe. In place of the internal stationary disc of the earlier design a solid rotary disc is used which is connected to the pump shaft, and, together with the radial vanes attached to it, forms an open pump impeller.

Holes near the impeller hub ensure that the pump casing is filled with liquid on both sides of the impeller disc. The clearances between the edges of the vanes and the casing are the same as those shown in Fig.11.

It is obvious that the conditions of liquid flow and pressure generation are very nearly the same in both pumps. The liquid inside the casing is brought completely into rotation by the impeller vanes, is then picked up at the periphery of the casing, and its kinetic energy is converted into static pressure partly by ram effect and partly by the action of the diffuser. This pump does not include any high pressure seals or small clearances and no end thrust occurs as the pressures on both sides of the impeller are balanced. The fluid friction losses, however, are higher than those in the rotating casing pump because the stationary surfaces in contact with the rotating liquid are larger.

On the other hand there are some advantages in the new design. The inlet can be made as a simple pipe of smaller diameter because the pressure pipe is no longer fitted inside it. Furthermore, this design is more suitable for higher rates of flow as it is easier to increase the number and the size of the diffusers. The pressure that can be developed is practically unlimited as the casing is not subject to centrifugal stresses. The rotating mass is considerably reduced and hence this pump can run rapidly up to speed and there is less danger of critical vibration.

However, an open impeller pump as described above has never been produced. A still simpler design, which ensures the cheapest possible manufacture, was introduced for practical use. As shown in Fig.13 the impeller is an open one, i.e. it does not include a disc-like portion, but the radial vanes, which may be 3 to 5 in number, are fixed directly to the hub. The streamlined pick-up heads are dispensed with and the impeller is accommodated in a smooth circular casing. The axial clearances on either side between the impeller blades and the casing are of the order of 0.020 to 0.030 inches. The diffusers are arranged tangential to the core of the casing, and their number depends upon the delivery required. They are generally of circular cross section, each diffuser consisting of a short cylindrical throat and a tapered section.

Obviously, the simplified diffuser inlets without the streamlined pick-up heads do not enable the ram effect to be utilised so effectively as in the rotating casing pump, or, as could be expected, in the design shown in Fig.12. Nevertheless, the pressures and the overall efficiencies are of the same range as those of single stage conventional type centrifugal pumps designed for the same speed and the same output. An approximate explanation of this fact may be that the losses due to the simple shapes of the blades (entrance shock, lack of recovery of relative inlet velocity) are balanced by absence of the internal leakage which usually occurs in the shrouded impeller pump as indicated above.

Owing to the wide clearances between the impeller and the casing the pump of simplified design is suitable for handling any liquid irrespective of its lubricating properties. If expensive materials, such as stainless steels, have to be used for chemical reasons the small dimensions and the simplicity of the various parts are of particular advantage. All the main parts for production models can be made as rough drop forgings or castings which require relatively little and simple machining.

Years of practical experience have proved the simplified open impeller pump to be unsurpassed for low weight, small dimensions, cheapness of manufacture and reliability at the highest speeds. This pump meets the requirements for rocket motors in an almost ideal way, and in fact, a number of fuel and oxidant pumps of this design[3,4] have been developed and have operated successfully.

The efficiency is probably the only factor in which improvement is desirable, but it has, up to the present, proved satisfactory. Further research and development work will, therefore, be concerned with the possibilities of increasing the efficiency of this type of pump.

3.4 Regenerative pumps

The regenerative pump, an example of which is represented in Fig.14, is also of very simple design, and since in addition it generates a given pressure at a much lower peripheral velocity of the impeller than any other centrifugal pump, it appears to be well suited for use in rocket motors.

The impeller consists mainly of a plane disc the periphery of which is provided on both sides with a number of small radial vanes. It rotates with close clearances between two circular faces of the casing which forms a passage round the periphery of the impeller. This passage is interrupted by a section of the casing through which the blades pass with close clearances on all sides, see Fig.14c, and which separates the suction side of the periphery from the pressure side. The liquid enters the impeller vanes from both sides as indicated and is carried to the pressure side mainly in the direction of the periphery. At this stage the liquid is subjected to centrifugal forces which cause the liquid to move in vortices relative to the blades and the surrounding passage of the casing. Additional energy is imparted to the liquid at every successive cycle (regenerative process) and thus a high delivery pressure is obtained as in the multi-stage centrifugal pump.

In normal practice these pumps handle only small rates of flow up to about 100 gal/min, the pressures amounting to about 200 lb/sq.in., but they are claimed to be suitable for twice this rate of flow and more than twice this pressure which would correspond to the requirements of a medium range rocket. An objection to the use of these pumps is that small clearances would be necessary in order to maintain a reasonable efficiency. The machining of the corresponding faces would not cause many difficulties or much expense, but the faces would be liable to seize during operation, particularly when oxidants are being pumped.

No end thrust is generated in this type of pump, but the pressures acting radially upon the periphery of the impeller are not balanced and, therefore, produce a considerable load on the shaft bearings.

4 Conclusions

A concise summary of the suitability of the pumps considered in this note for use in rocket motors is given in Table I.

Obviously the open impeller centrifugal type of pump can be recommended as the most suitable design for both fuel and oxidant pumps.

In order to secure a highly efficient fuel pump for a small unit it may be thought worth while to use a gear pump driven through a reduction gear. This arrangement should be feasible as fuels generally have some lubricating properties. The characteristics (relationships between pump speed, rate of flow and delivery pressure) of gear pumps and centrifugal pumps, however, are so different that it might be difficult to maintain the desired ratio of the rates of flow of the fuel and oxidant. A more complicated control device would probably be required.

A useful combination would be an open impeller pump for pumping the oxidant and a scoop wing pump for handling the fuel, the rate of flow of which is generally much smaller than that of the oxidant. The fuel pump could probably be made integral with the turbine rotor, as shown in Fig.11, so that this turbo-pump would be an extremely compact unit.

Theoretical and constructional details of the scoop wing pump and the open impeller pump will be given in a subsequent note.

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| Ref.No. | Author | Title, etc. |
|---|---|---|
| 1 | N. van De Verg | Development of Rocket Propellant Feed Pumps for Red Fuming Nitric Acid and Aniline. Progress Report No.1-21. Jet Propulsion Laboratory GALCIT California Institute of Technology, 1944. |
| 2 | A.D. Baxter | Aircraft Rocket Motors. Aircraft Engineering, August 1947. |
| 3 | A.D. Baxter | High Pressure Pumps for Rocket Propellants. R.A.E. Technical Note R.P.D.12, February 1949. |
| 4 | U.M. Barske<br>D.J. Saunders<br>C.G. Saunt | The Beta I Turbo Pump. R.P.D. in draft. |


Attached: R.P. 508, 526-529, 544-552
Table I

Advance Distribution:

M.O.S.
CS(A)
Chief Scientist
P/D.S.R.(A)
ADSR (Records)
DGWRD
GW3 Cmdr. Ashworth 6
D Eng RD
Eng RD6
Chairman ADB
DTRDE
CSXR
DGTD(A)
ERDE Waltham Abbey 6
ADGW (R & D)
TPA3/TIB 180

R.A.E.
Director
DDRAE(W)
Guided Weapons 3
Naval A/C
Mat Dept.
Chem Dept.
Armament Dept.
SME Dept.
Library

TABLE I

Suitability of various types of pumps for use in rocket motors

| Type of Pump / Desirable features of the ideal pump | Positive displacement Pumps | | | | Centrifugal Pumps | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | Shrouded impeller | | Single stage | | |
| | Piston Pump | Gear Pump | Helical Pump | Vane Pump | Multi-stage Pump | Single stage Pump | Scoop wing Pump | Open impeller Pump | Regenerative Pump |
| Suitability for high pressures | + + | + | + | - | + | + | + + | + + | + |
| Suitability for high rates of flow | - | - | - | - | + + | + + | - | + | + |
| Suitability for high speeds | - - | - | + | - | - | + + | + + | + + | + |
| Independence of lubricating properties of liquid pumped | - - | - - | - | - - | + + | + + | + + | + + | + |
| No end thrust | + | + | - | + | - - | + | + | + | + |
| No danger of seizure | - - | - | - | - | + | + | + + | + + | - |
| High efficiency | + + | + + | + | + | + | - | - | - | - |
| No internal seals | - | - | - | - | - | - | + + | + + | - |
| No narrow clearances | - - | - - | - - | - - | - | - | + + | + + | - |
| Minimum size | - | - | - | - | - | - | + + | + + | - |
| Minimum weight | - | - | - | - | - - | + | + | + + | + |
| Simplicity of design | - | + | + | + | - | + | + | + + | + + |
| Cheapness of manufacture | - - | - - | - - | - | - | + | + | + + | + |

The symbols + + and + indicate a greater or smaller degree of approach, respectively, to the desirable feature of the ideal pump. Correspondingly, the symbols - - and - indicate a greater or smaller divergence from the desirable feature of the ideal pump. Thus + +, +, -, - -, display in descending order the correspondence of features of an actual pump with those of the ideal pump.

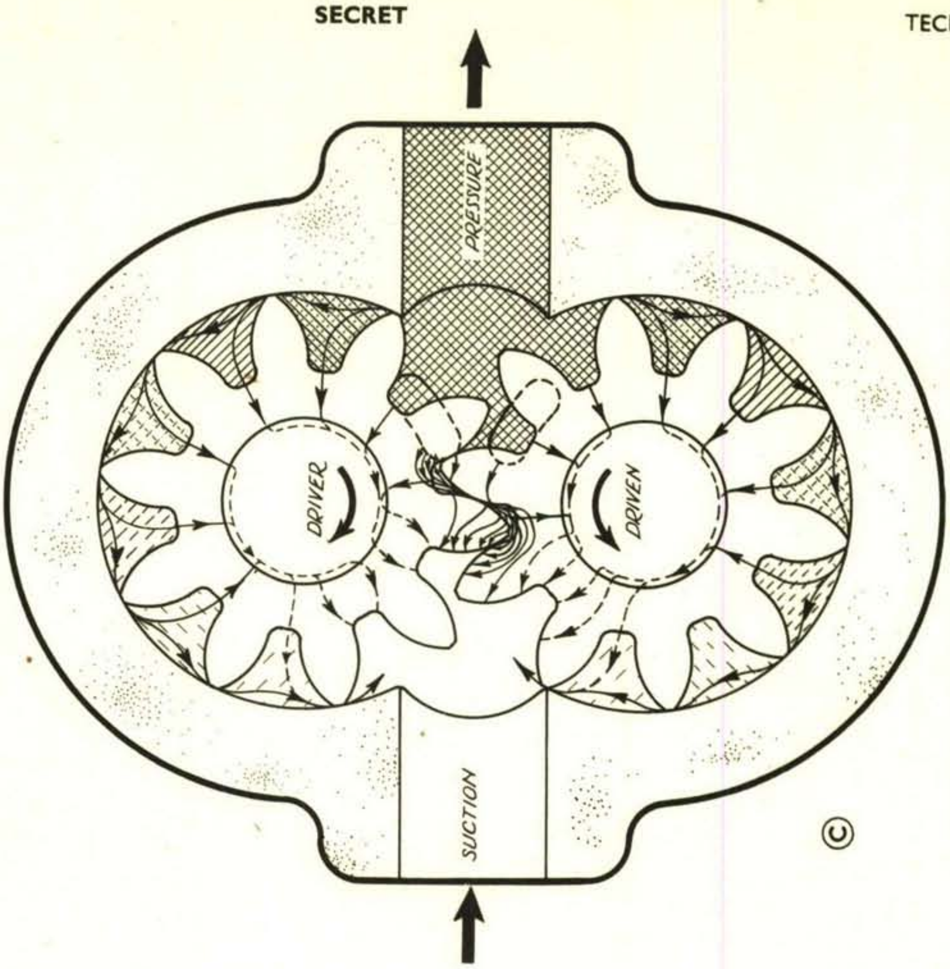
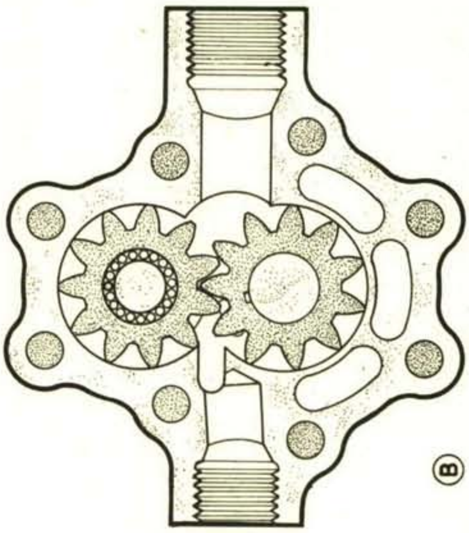



DIAGRAM SHOWING THE INTERNAL LEAKAGE PATHS IN A GEAR PUMP AND THE SPREAD OF PRESSURE AROUND THE PERIPHERY OF THE GEARS



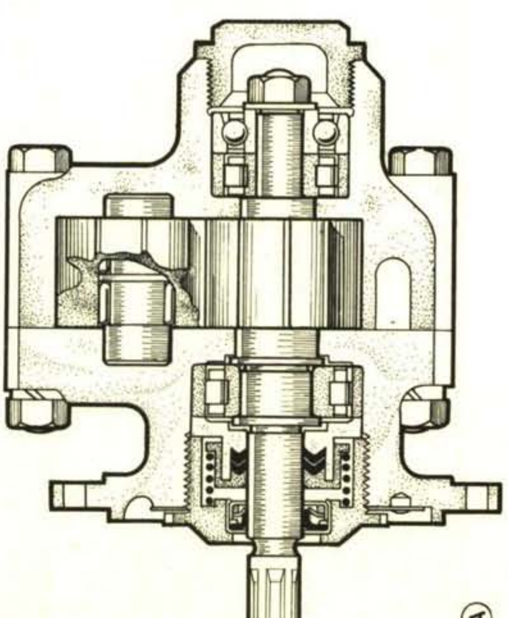




FIG. 1. MEDIUM PRESSURE GEAR PUMP

FIG.2

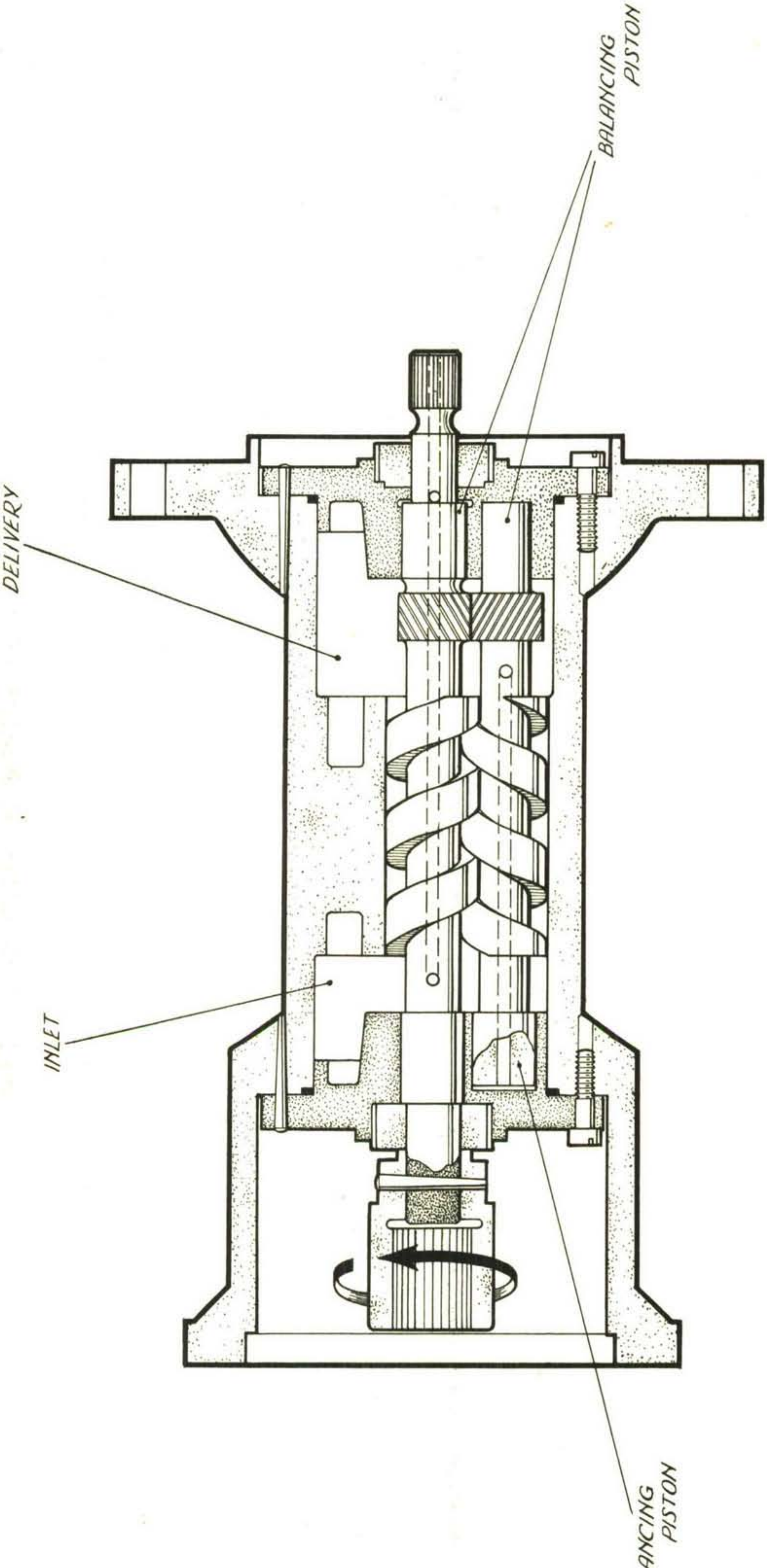



FIG.2. CROSS SECTION THROUGH GERMAN SCREW PUMP

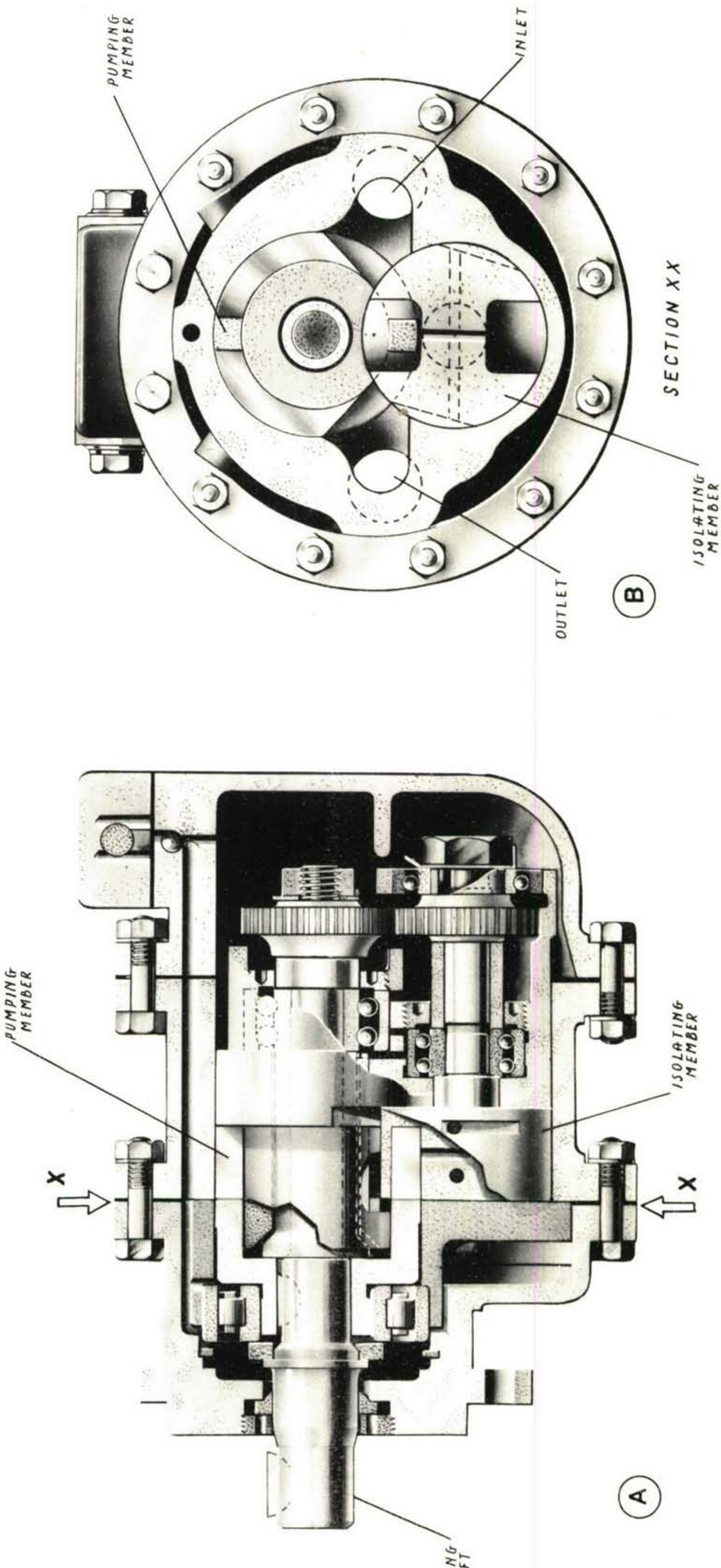
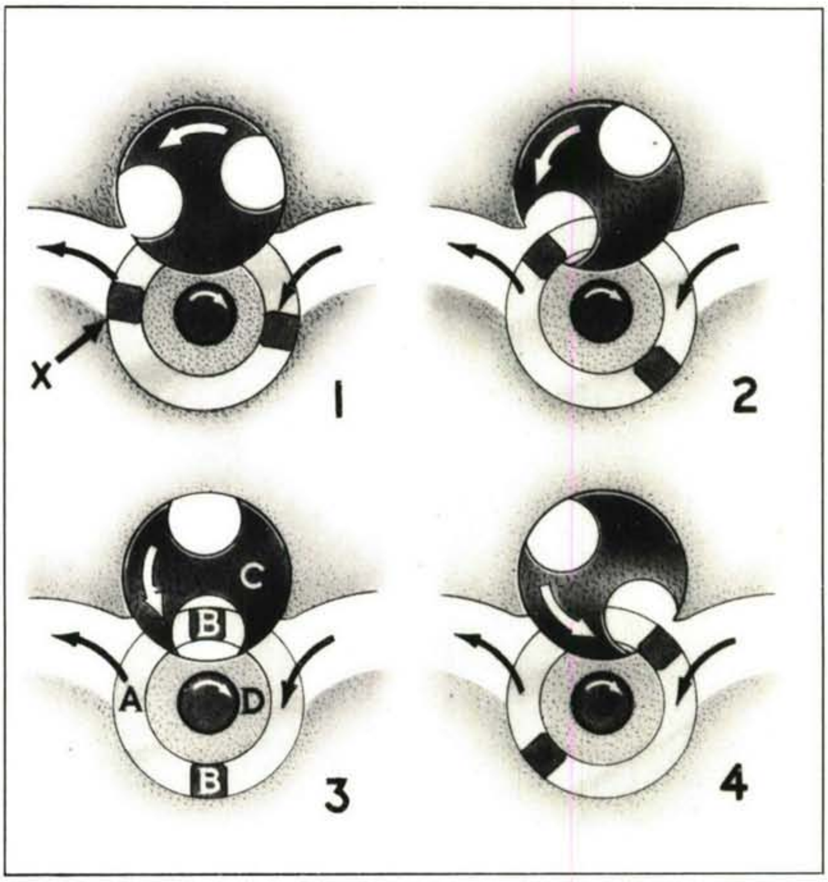



FIG.3. TWIN ROTOR DISPLACEMENT PUMP
(BY KIND PERMISSION OF KEELAVITE ROTARY PUMPS & MOTORS LTD.)




A. ROTOR DISC & ANNULAR SPACE
B. PUMPING BLADES (DISPLACING MEMBERS)
C. ISOLATING MEMBER
D. STATIONARY CORE

1. The blade (b) which has reached position (x) has just finished pumping. The other blade has begun to pump.

2. The blade is entering the recess in the isolating member (c).

3. The blade is now half-way through the isolating member (c) note that the chord of the recess in the stationary core (d) is longer than the distance across the edges of the gap in the isolating member (c) which provides an area seal.

4. The blade is leaving the recess in the isolating member.

Note that the moment a blade begins to pump until it reaches position (x) it is displacing liquid at a constant rate; immediately this position is passed, the other blade is able to deliver at the same rate, and the first blade passes through positions 2 and 3 merely displacing its own volume of liquid out of the gap in the isolating member, thereafter continuing as position (4).

**FIG.4. DIAGRAM OF OPERATION OF TWIN ROTOR PUMP**
**(BY KIND PERMISSION OF KEELAVITE ROTARY PUMPS & MOTORS LTD.)**

FIG.5

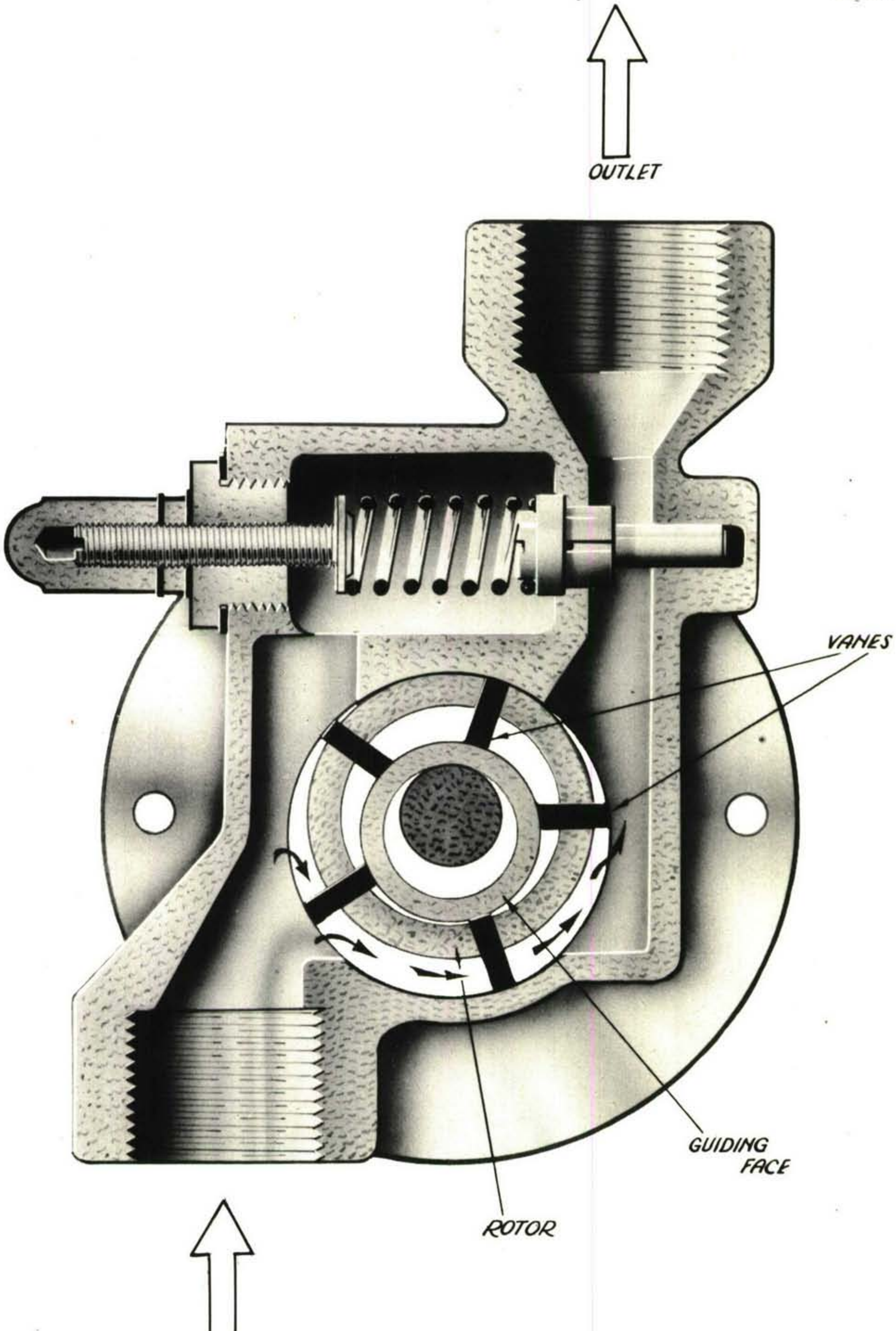



FIG.5. CROSS SECTION OF A VANE PUMP

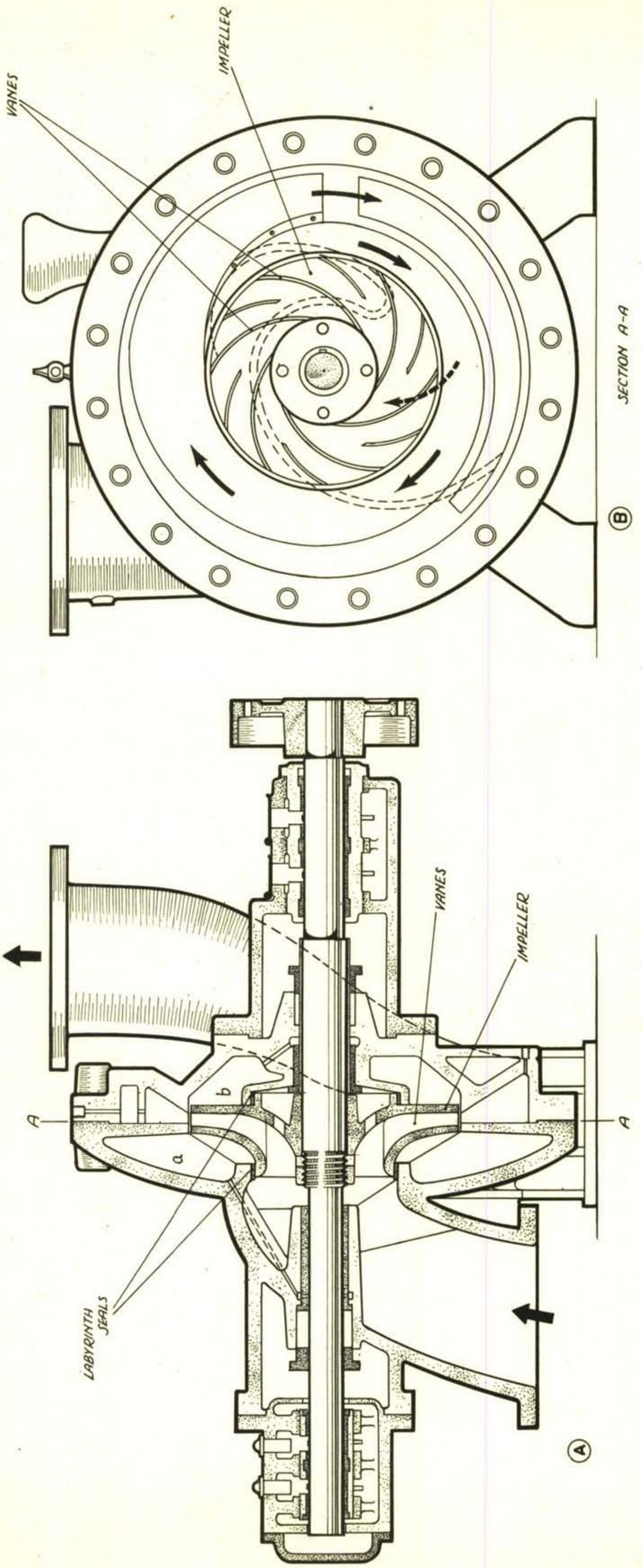



FIG.6. CONVENTIONAL SINGLE STAGE CENTRIFUGAL PUMP

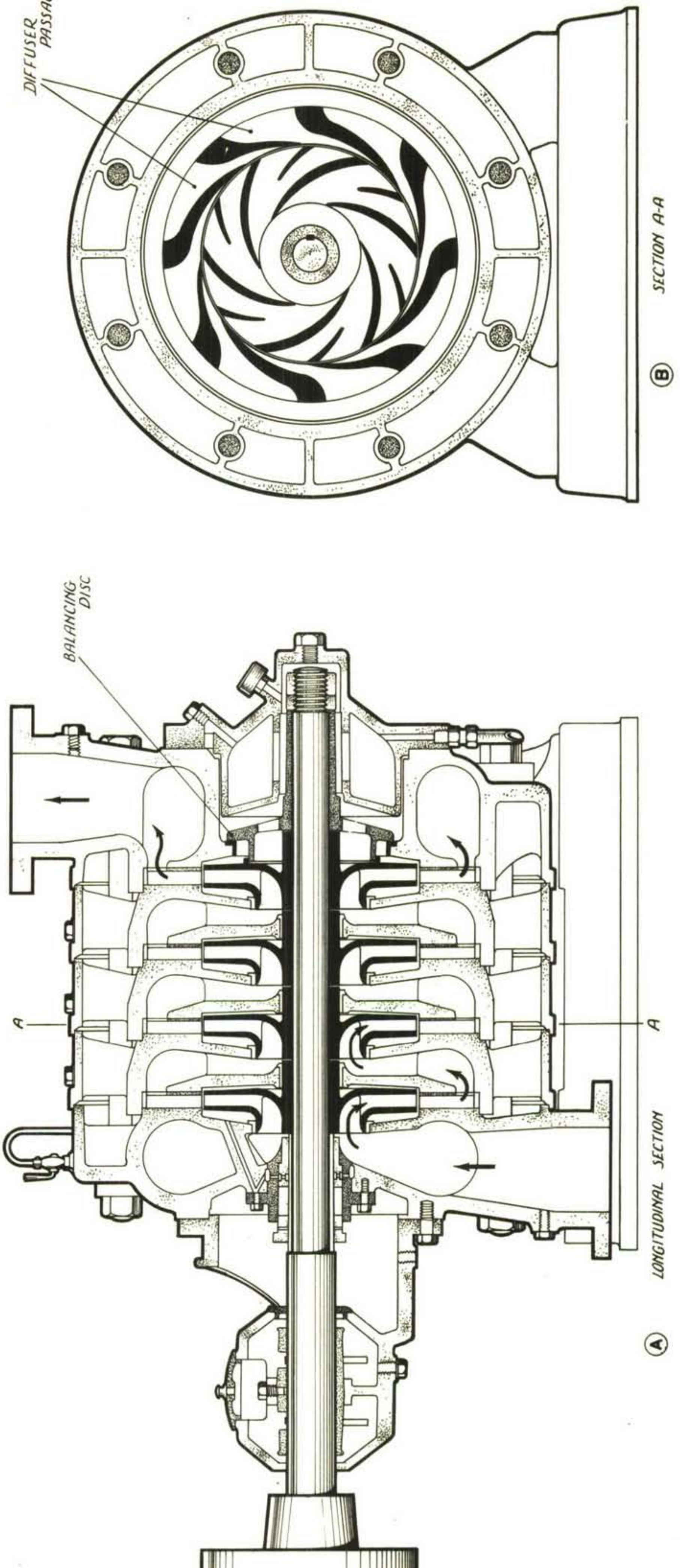



FIG.7. MULTI-STAGE CENTRIFUGAL PUMP

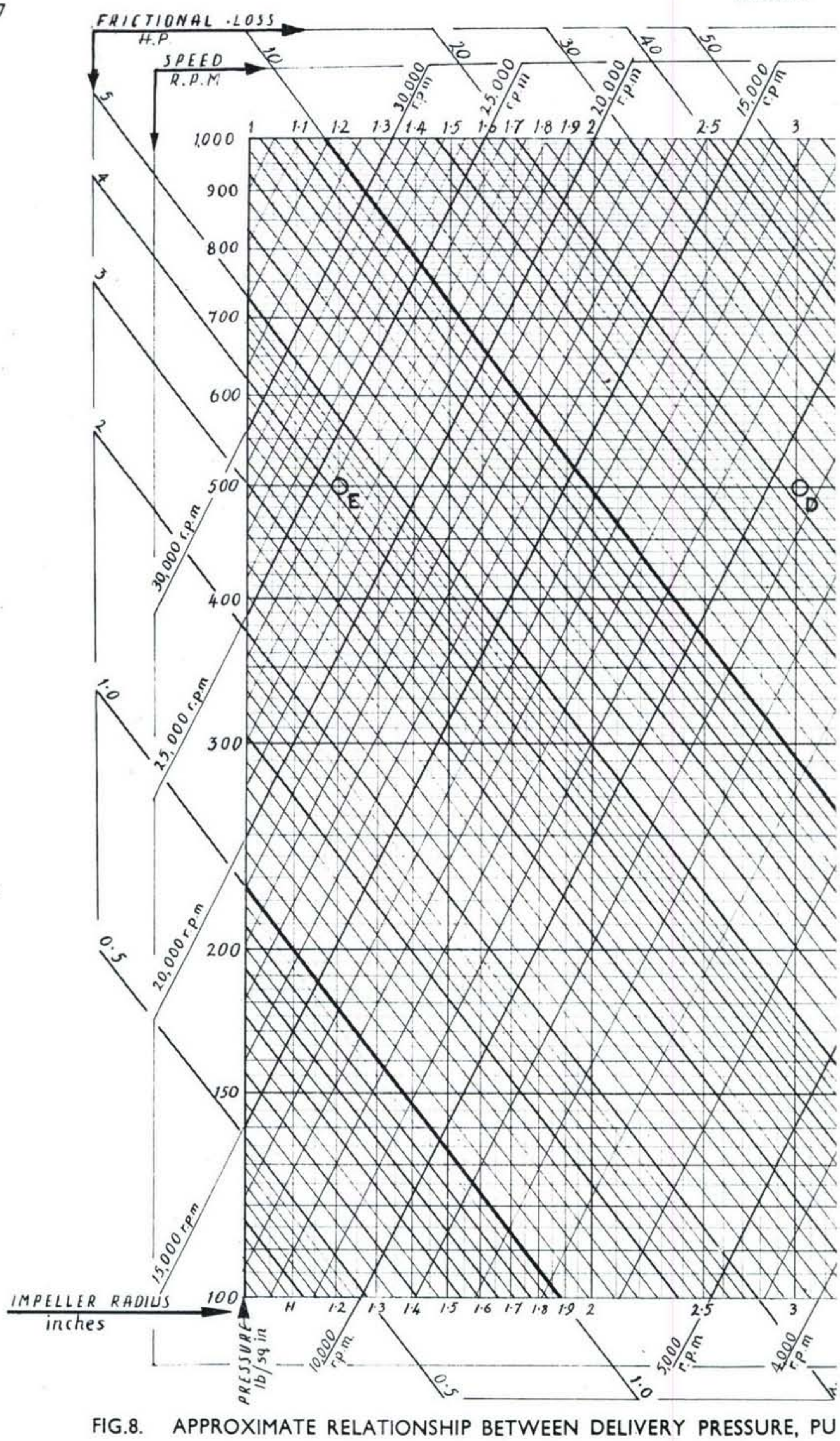
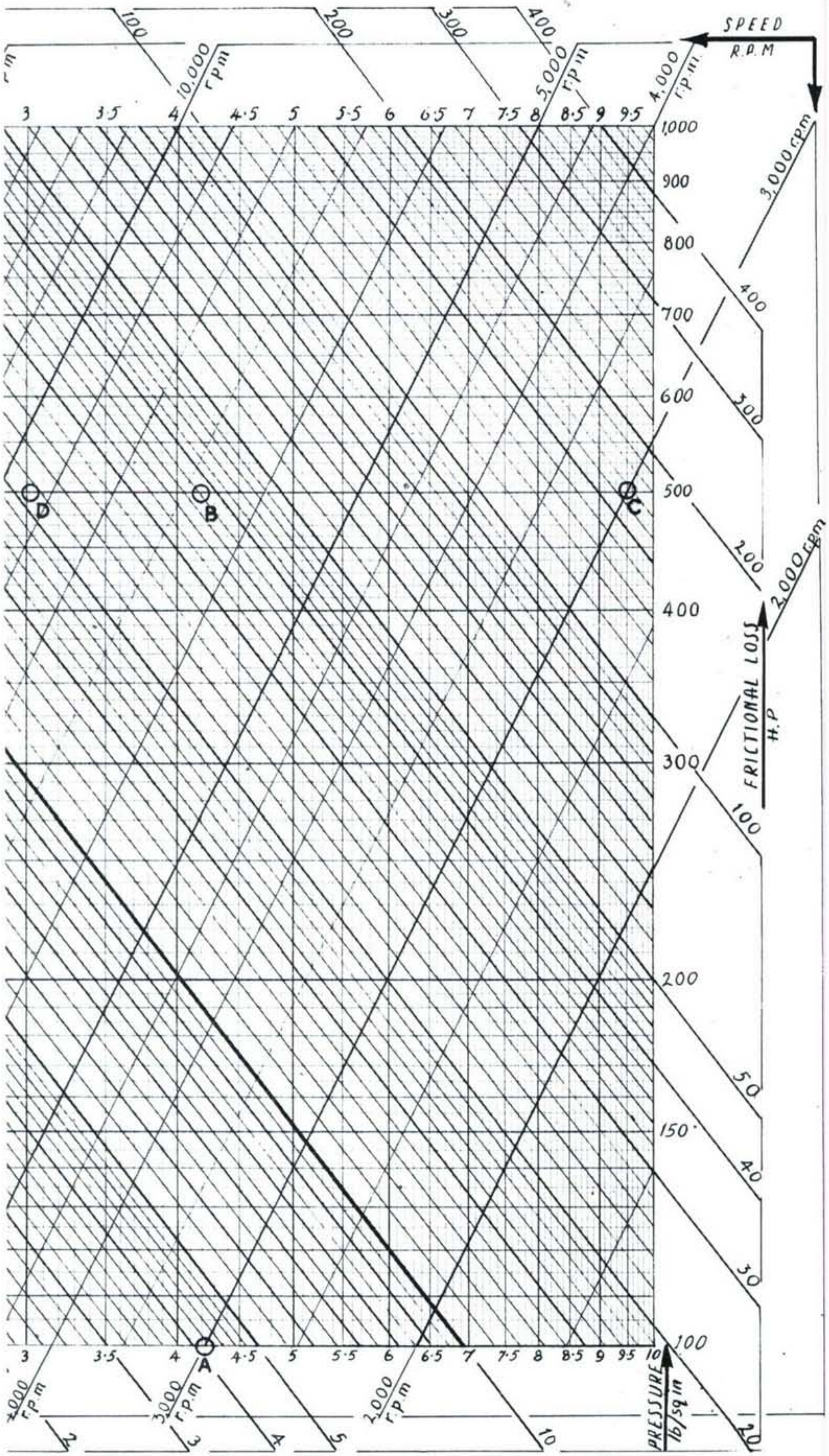



FIG.8. APPROXIMATE RELATIONSHIP BETWEEN DELIVERY PRESSURE, PU

FIG.8




ɌE, PUMP SPEED, IMPELLER RADIUS AND HYDRAULIC FRICTION LOSSES

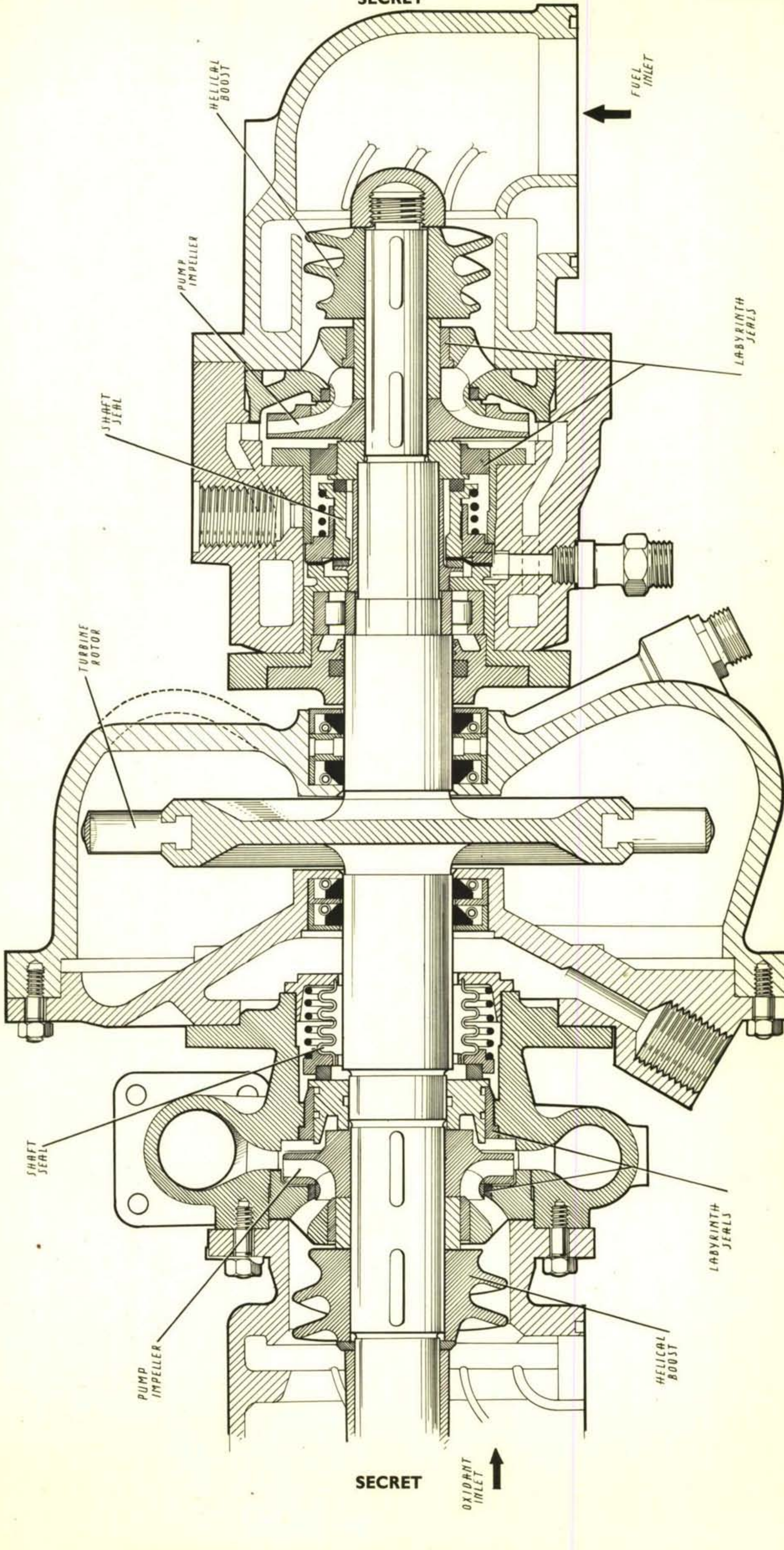
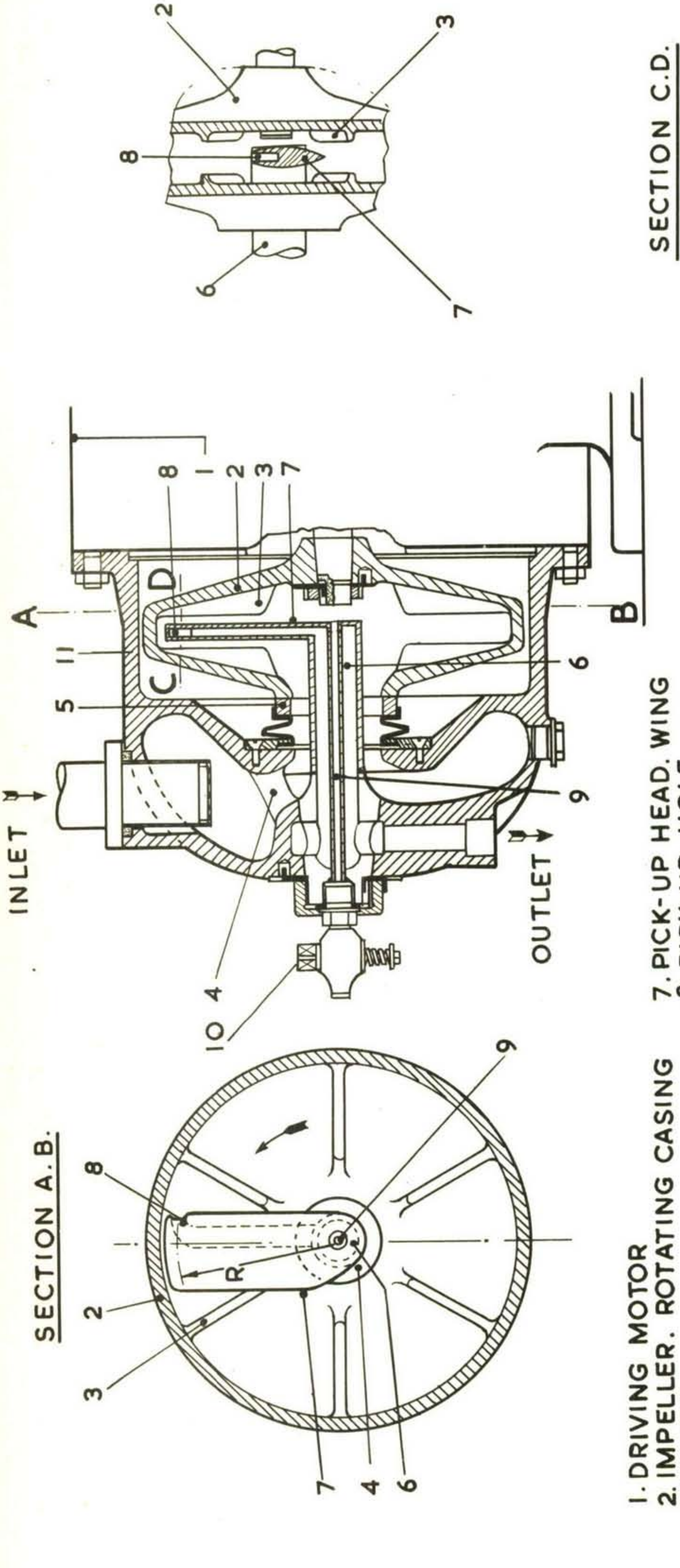



FIG.9. ROCKET MOTOR TURBO-PUMP SET

FIG. 10




1. DRIVING MOTOR
2. IMPELLER. ROTATING CASING
3. VANES
4. ENTRY
5. RUBBING RING SEAL
6. PRESSURE TUBE
7. PICK-UP HEAD. WING
8. PICK-UP HOLE
9. VENTING PIPE
10. VENTING COCK
11. OUTER CASING. STATIONARY

FIG. 10. SMALL FLOW CENTRIFUGAL PUMP

FIG. 11

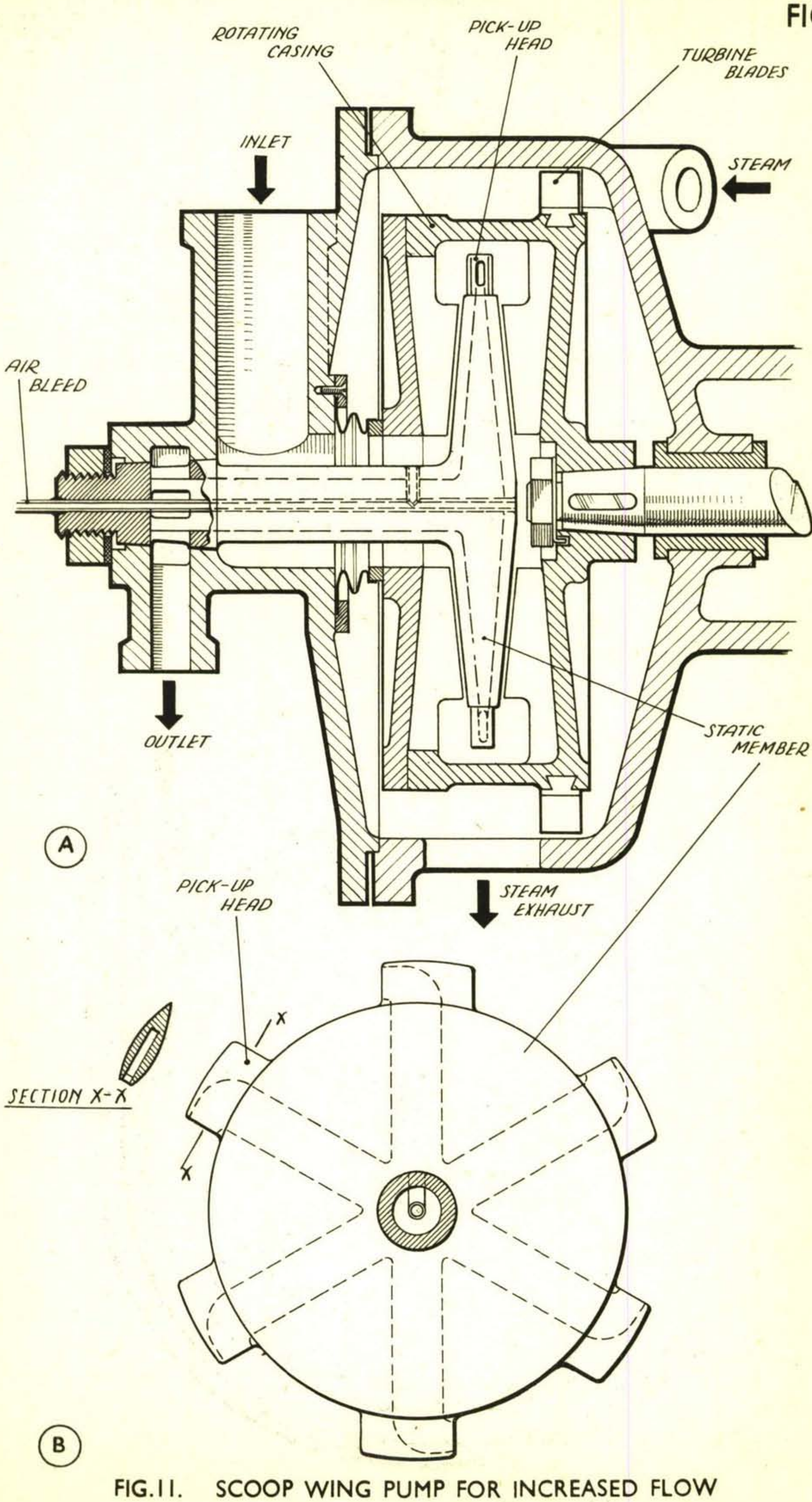



FIG. 11. SCOOP WING PUMP FOR INCREASED FLOW

FIG. 12

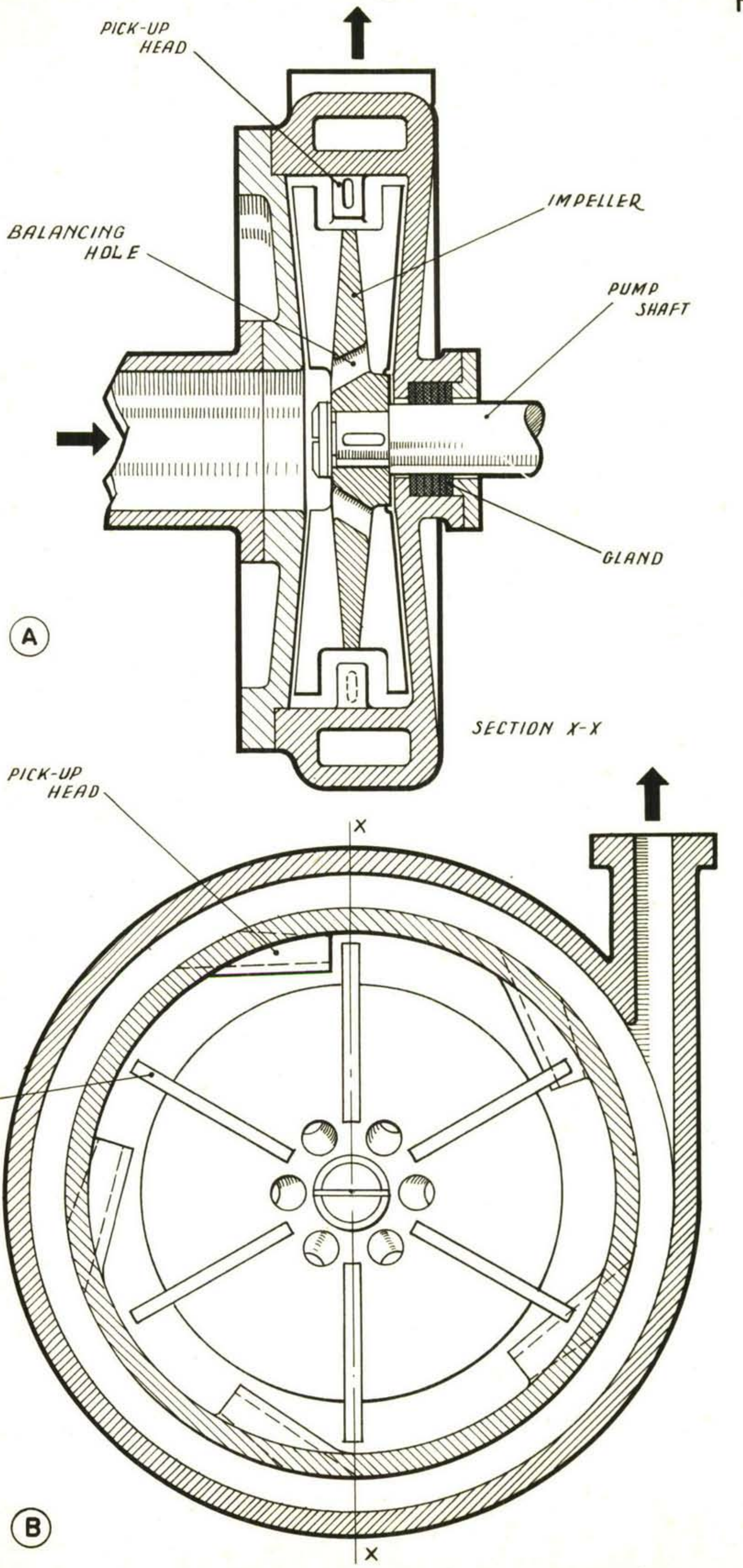



FIG. 12. OPEN IMPELLER PUMP (INITIAL DESIGN)

FIG. 13

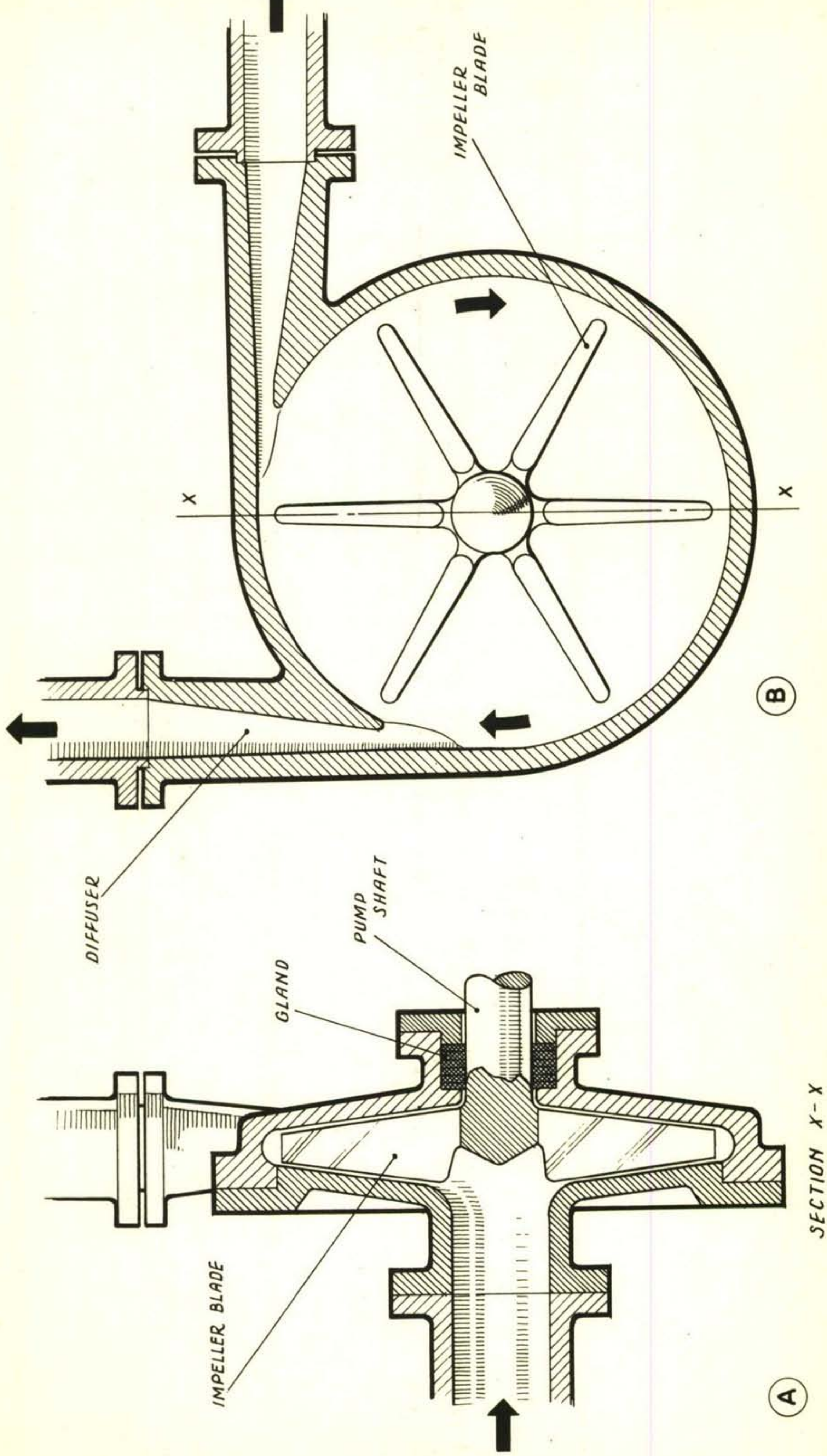



FIG. 13. OPEN IMPELLER PUMP (SIMPLIFIED DESIGN)

FIG.14

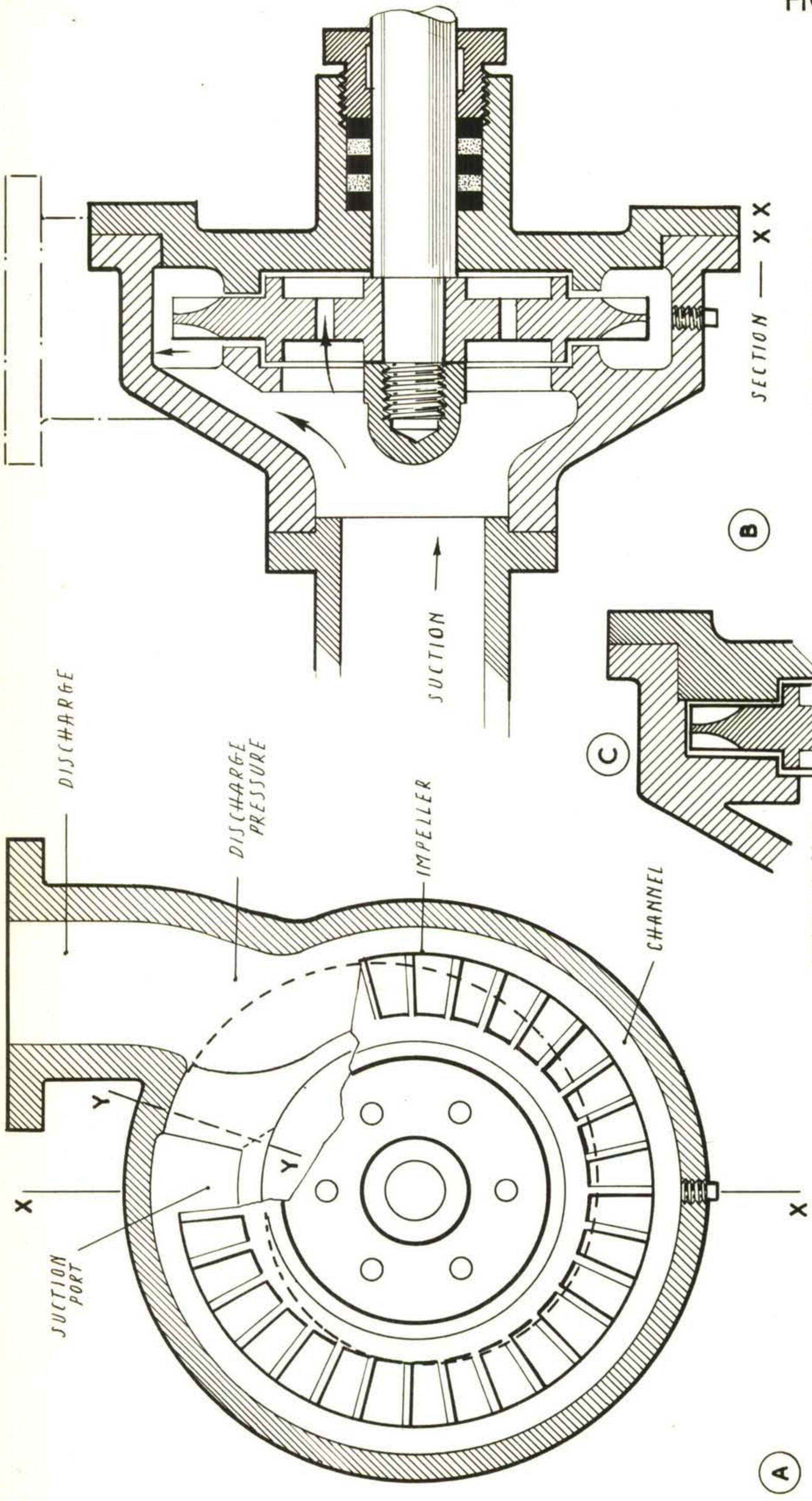



FIG.14. REGENERATIVE PUMP

~~SECRET~~

UNCLASSIFIED

UNCLASSIFIED

~~SECRET~~