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FIRST STAGE MECHANICAL PUMP FOR USE IN A TWO STAGE VACUUM PUMPING SYSTEM

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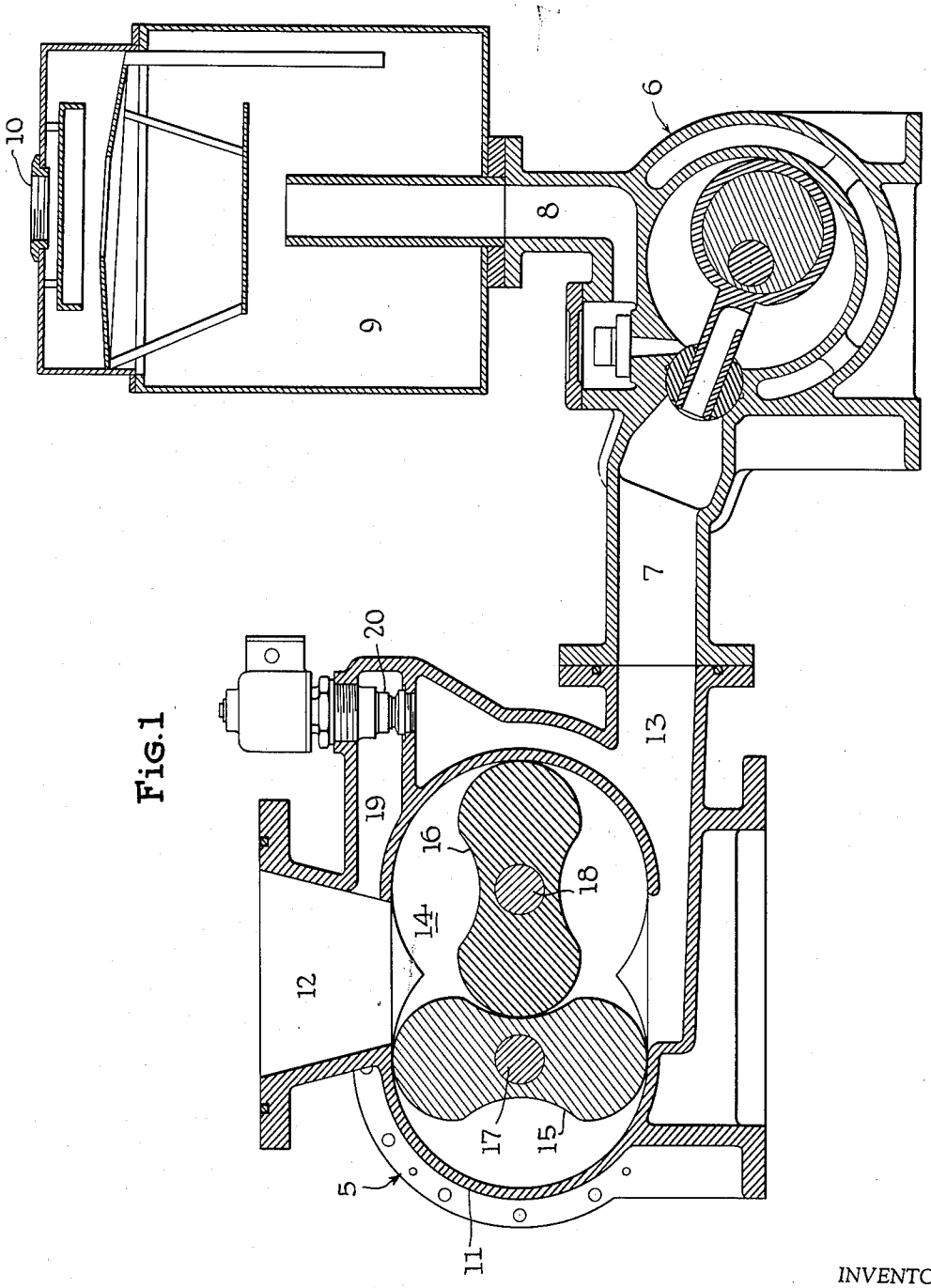


Fig. 1

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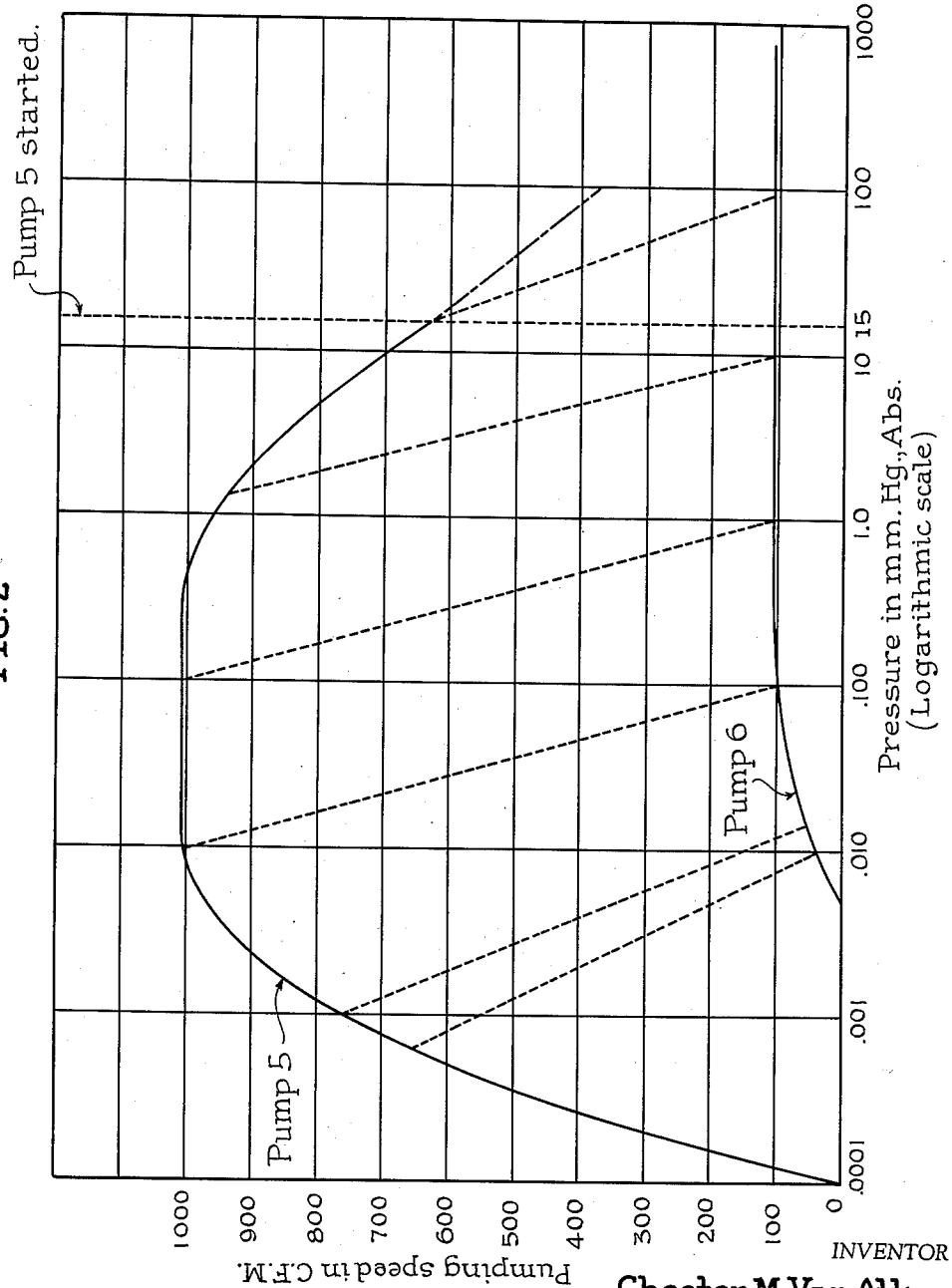
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Fig. 2



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FIRST STAGE MECHANICAL PUMP FOR USE IN A TWO STAGE VACUUM PUMPING SYSTEM

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3 Claims. (Cl. 230—158)

This invention relates to multi-stage high vacuum pumping systems and particularly to mechanical pumps for use in conjunction with exhausting means which are operative to establish a predetermined low absolute pressure at their outlets.

In my Letters Patent No. 2,492,075, dated December 20, 1949, there is disclosed and claimed a multi-stage high vacuum pumping system in which a first pump of the unsealed, rotary, positive displacement type is backed by a rotary piston, oil sealed, positive displacement pump. Experience with the pumping system described in the above patent has served to enhance the understanding of the relationship between desirable operating pressures and the permissible clearances.

The former pump is of the type comprising a pair of rotors, each provided with portions which fit conjugate portions of the other during rotation thereof and which are enclosed in a housing having an inlet and an outlet. That type of pump is adapted for use at high rotational speeds because its parts are in balance and thus has a large pumping capacity in relation to its physical dimensions. As such a pump is not oil sealed, frictional drag is reduced to a minimum so that power demands, even at high rotational speeds, are not excessive. Because of the clearances between its rotors and between the rotors and their housing, the factor that limits the acceptance of such pumps as the first unit in a high vacuum pumping system is internal leakage. This internal leakage includes back flow or slippage, and also reverse pumping action occasioned by the rotors entrapping and carrying back to the inlet, pockets of gas at outlet pressure. The present invention is directed to the factors which enable such a pump, when used as the first pump in a high vacuum pumping system, to have high pumping efficiency when working against a low absolute pressure established by the exhausting means.

An important measure of the effectiveness of a first pump in a multi-stage high vacuum pumping system is its ability to maintain a high compression ratio against the pressure at its outlet established by the exhausting means. If the first pump meets this criterion, it then retains high pumping efficiency even when the exhausting means has a relatively small displacement in comparison. Such a large compression ratio cannot exist, however, unless the internal leakage rate is made substantially smaller than the ideal pumping capacity or displacement of the pump.

For a more complete understanding of the invention, reference is made to the fact that it is well known that conductance of gases through an aperture reaches a minimum value at absolute pressures below a critical pressure. Below that critical pressure, conductance is independent of the pressure and the flow is diffusive in character. Above that critical pressure, conductance increases proportionately with the pressure and the flow is viscous in nature. For that reason, when an unsealed, rotary, positive displacement pump is used in the range of atmospheric pressures, the internal leakage, due to

slippage, occurring even for the smallest attainable clearances, is so large that the compression ratio is limited. For this reason alone the compression ratio is limited to approximately five. This ratio of 5 to 1 is attainable even with rotors which depart significantly from the ideal rotor form since further impairment of performance due to reverse pumping action is not appreciable. At pressures near or below the critical value, the internal leakage for easily attainable clearances is so small that the reverse pumping action due to these same departures from ideal form would constitute the dominating source of internal leakage and seriously reduce the otherwise attainable high compression ratio.

In the ideal form, the clearances between the rotors and between them and their housing are a near approach to line contact and are constant thus reducing internal leakage sources to a minimum. In forming rotors for pumps of this type, a close approach to the ideal form gives rise to special problems of machining and design and the present invention is concerned with establishing clearance requirements that are attainable in production and will ensure that the rotary, unsealed, positive displacement pumps may have a suitably high compression ratio when a suitable low absolute pressure is maintained at their outlets.

Reference has been made to the fact that below a critical pressure, conductance through an aperture is independent of the pressure at the high pressure side of a slit-shaped aperture. That critical pressure, called the transition pressure, is for air and similar gases, the pressure for which the mean free path of the molecules of the gas, i. e. the average distance a molecule may travel without colliding with another, is approximately equal to one-sixth of the smallest dimension of the cross section of the aperture.

In accordance with the invention, the dimensions of the clearances are determined by the mean free path of the gas molecules at an outlet pressure such that the critical pressure, corresponding to the chosen clearances, lies in the lower portion of the pressure range of high pumping efficiency of the exhausting means but well above the pressure region of low pumping efficiency of said means. With such clearances, slippage is reduced at that and lower absolute pressures to a substantially constant and tolerable minimum.

As has been stated earlier, pumping speed is also lost by internal leakage due to reverse pumping action. In accordance with the invention, reverse pumping action is minimized to an acceptable extent by the fact that irregularities and pockets are eliminated in which gas at discharge pressure could be entrapped and released at the inlet.

These generally stated factors enable rotary, unsealed, positive displacement pumps to be operated with a large compression ratio against the fore pressure of the exhausting means. Thus, high pumping efficiency is ensured even though the exhausting means has small displacement in comparison

In the accompanying drawings, an embodiment of the invention is sufficiently detailed and its performance so illustrated as to enable the generally summarized features to be more fully understood.

In the drawings:

Fig. 1 is a vertical, longitudinal section through a multi-stage high vacuum pumping system including a rotary, unsealed positive displacement pump backed by an oil sealed, rotary piston positive displacement pump, and

Fig. 2 is a graph illustrating performance characteristics of the first pump when operated in conjunction with the second pump.

In the drawings, there is shown a two stage high vacu-

um pumping system with the first and second or backing pumps being generally indicated at 5 and 6, respectively. The vacuum pump 6 is of the rotary piston, oil sealed type and has an inlet 7 and an outlet 8 with the outlet 8 discharging into the oil separating tank 9 having its outlet 10 open to the atmosphere.

The pump 5 is of the rotary, unsealed, positive displacement type and its housing 11 has an inlet 12 and an outlet 13. The outlet 13 and the inlet 7 of the pump 6 are interconnected and establish an interstage chamber. The housing 11 is chambered at 14 to receive the rotors 15 and 16 which are fast on the shafts 17 and 18, respectively, and whose conjugate curved portions fit one another and the housing during operation of the pump 5 in a manner that will presently be described.

The housing 11 is also provided with a conduit 19 which by-passes the rotor chambers and is under the control of a valve 20. With the valve 20 open and the pump 5 idle, the second pump 6 may be used by itself or, with the valve 20 closed, both pumps may be used with the second pump 6 then backing the first pump 5.

The pump 6 may be assumed to have a displacement of 125 C. F. M. and good pumping efficiency from atmospheric pressure down to pressures in the order of 0.1 mm. Hg. Below that pressure, efficiency decreases with the pressure so that where pressures in the order of 0.001 mm. Hg or lower are to be maintained, the service of the first pump 5 is required.

As will be apparent, the rotors 15 and 16 are in balance so that they may be driven at high rotational speeds ensuring large pumping capacity for relatively small physical dimensions. The rotors 15 and 16 are, in practice, a practicable approach to the ideal form in that the clearances between them and between them and their housing are closely held so that, during operation, apertures of substantially constant size are maintained with the clearances being not greater than about six times the mean free path of the molecules of the gas being pumped at a predetermined low absolute interstage pressure. Each clearance, in the exemplary embodiment being described, is essentially a long narrow slot whose width may be assumed to be .008 inch, for example, thus to be equal to about six times the mean free path of the molecules of the gas at an interstage pressure, of say 1.5 mm. Hg and it will be apparent that clearances of that order are feasible to maintain in production. With clearances thus related to a predetermined interstage pressure, the leakage remains at a substantially constant and tolerable level at that and lower absolute pressures.

One other important feature of the rotors 15 and 16 is the fact that no pockets or irregularities are provided by which gas at outlet pressure can be entrapped and carried back to the inlet. This is, in part, effected by maintaining the clearances so that the line of minimum clearance between the rotors moves continuously through a cycle as contrasted with the discontinuous motion resulting from the maintenance of minimum clearance only over limited portions of the rotors which would result in the points of near contact being separated by gas carrying pockets. Coupled with that requirement is that of the surface of the rotors being smooth or polished.

The pump 5 may be assumed to have a cylinder length of sixteen inches, a cylinder bore of 9¾ inches with an operating speed of 1750 R. P. M. Its displacement at that speed is, then 1234 C. F. M. The ratio between the displacement of the pumps 5 and 6 is approximately 10 to 1. The radial clearances between its rotors and between them and their housing 11 is in the order of .008 inch. Between the end boundary surfaces of the chamber 14 and the proximate ends of the rotors, the clearances may be and usually are greater because of the length of the leakage paths defined thereby.

Such a high vacuum pumping system is usually initially operated with the valve 20 open and only the pump 6 in service until the pressure in the interstage chamber is suitably reduced. Until the gas flow through the clearances of the pump 5 is diffusive in character, the compressure ratio of the pump 5 is approximately 5 to 1. The interstage pressure at which the pump 5 is put into service is usually a matter of economy of operation, but such pressure is always included in the range of good pumping efficiency of the pump 6.

When the transition pressure in the interstage chamber is reached, the flow of gas through the clearances of the pump 5 becomes diffusive in character and the pump 5 operates with a much higher compression ratio, the actual ratio being dependent on internal leakage. With constant radial clearances in the order of .008 inch and with the elimination of pockets by which a gas at interstage pressure could be carried back to the inlet of the pump 5, leakage is reduced to an acceptably small fraction of the displacement of the pump 5. Thus, with internal leakage effectively reduced without imposing impracticable production requirements, the pump 5 may have a compression ratio in the order of sixty or higher. While acceptable results can be attained with a smaller ratio, the higher the ratio, the better the performance. The pump 5 is, accordingly, well suited for use as the first pump in a high vacuum multi-stage pumping system. This feature has the important additional advantage that, with an oil sealed backing pump, it is attended by a vapor compressor action since the vapor pressure of the sealing oil present in the interstage chamber is reduced by the compression ratio.

A typical pumping speed curve of the pump 5 is shown in Fig. 2. It will be noted that the pump 5 is put into operation when the interstage pressure is in the order of 15 mm. Hg at which pressure the pumping speed of the backing pump 6 is satisfactory as shown by its pumping speed curve. With the clearances which the pump 5 has thus far been defined as having, this interstage pressure is in excess of the transition pressure and, as the flow of gas through the pump 5 is viscous, its compresion ratio is in the order of five until the transition pressure, approximately 1.5 mm. Hg is reached. Thereafter, the flow through the pump is diffusive and the pump 5 operates with a high compression ratio and its pumping speed remains high until the interstage pressure approaches the limiting value, say 0.01 mm. Hg or less, maintainable by the backing pump 6, below which the pumping speed of the pump 5 falls off as the inlet pressure diminishes toward its ultimate blank-off value determined by the compression ratio.

The compression ratio of the pump 5, when the flow of gas is diffusive, is dependent on the extent to which leakage has been minimized and is several times the ratio existing when the gas flow is viscous. The higher the compression ratio, the better, and ratios in the order of 40–60 to one or better enable excellent results to be attained.

While the operation of the pump 5 may be started at a higher interstage pressure than above indicated, there is little advantage in so doing because of the power consumed. The pump 5 could also be placed in service at a somewhat lower interstage pressure, but it is preferred that there be a substantial overlap between its operation and the lower limits of the range of good pumping speed of the pump 6.

With a pump 5 of the dimensions stated, an improvement only in the order of three percent in maximum pumping speed is noted with radial clearances reduced to .004 inch. Coupled with this small improvement is the fact that any gear backlash in the drive of the pump 5 becomes objectionable with such small clearances. A substantial reduction in maximum pumping speed, say 14%, results when the radial clearances are increased to .016 inch.

It has been pointed out that the ratio in displacement between the pumps 5 and 6 is approximately 10 to 1. If

this ratio is varied by utilizing a backing pump of one-half the displacement assumed for the pump 6, there is a substantial loss in the maximum pumping speed while a modest gain results in the use of a backing pump of twice that displacement. This gain is not proportional and is achieved at a substantial increase in equipment cost and size. It should be noted, however, that the inlet pressures at which the performance of the pump 5 falls off improves as the displacement of the backing pump is increased so that for some uses, backing pumps of larger capacity may be justified. By increasing the rotational speed of the pump 5 to provide a ratio between the displacement of the pumps 5 and 6 in the order of 20 to 1, the pumping speed of the pump 5 may be increased by about 70% without significant increase in the cost and with no increase in the dimensions of the equipment.

From the foregoing, it will be apparent that the invention establishes the requirements of pumps such as the pump 5 for use in multi-stage vacuum pumping systems as being that the clearances between the rotors and between each rotor and the housing must be constant by which is meant (1) the radial clearances must not exceed about six times the mean free path of the molecules of the gas being pumped at a pressure against which, as its backing pressure, the pump is required to maintain approximately its maximum pumping speed, and (2) that the line of minimum clearance between the rotors must move continuously through a cycle as contrasted with the discontinuous motion resulting from the maintenance of minimum clearances only over limited portions of the rotors resulting in the lines of near contact being separated by gas carrying pockets.

What I therefore claim and desire to secure by Letters Patent is:

1. A mechanical vacuum pump for use in a gas pumping system wherein exhausting means provide a low pressure at the outlet of said mechanical pump, said pump comprising a pair of rotors, and a housing for said rotors having a gas receiving inlet and an outlet to be disposed in communication with the exhausting means, there being a constant clearance between each end surface of said housing and the end surfaces of said rotors proximate thereto, and said rotors including conjugate curved portions which fit said housing and one another during their rotation to provide continuous zones of constant clearance, each clearance being approximately equal to six

times the mean free path of the molecules of the gas to be pumped at the maximum outlet pressure against which said pump maintains its maximum pumping speed.

2. In a multi-stage high vacuum pumping system, a mechanical vacuum pump comprising a pair of rotors, and a housing for said rotors having a gas receiving inlet and an outlet, there being a constant clearance between each end surface of said housing and the end surfaces of said rotors proximate thereto, and said rotors including conjugate curved portions which fit said housing and one another during their rotation to provide continuous zones of constant radial clearance, and exhausting means having an inlet operatively connected to the outlet of said pump, each radial clearance of said pump being approximately equal to six times the mean free path of the molecules of the gas being pumped at the interstage pressure against which said pump maintains its maximum pumping speed.

3. In a multi-stage, high vacuum pumping system, a mechanical vacuum pump comprising a pair of rotors, and a housing for said rotors having a gas receiving inlet and an outlet, there being a constant clearance between each end surface of said housing and the end surfaces of said rotors proximate thereto, and said rotors including conjugate curved portions which fit said housing and one another during their rotation to provide continuous zones of constant radial clearance, and exhausting means having an inlet in communication with the outlet of said pump, each radial clearance of said pump being equal to six times the mean free path of the molecules of the gas being pumped at the interstage pressure against which said pump is to maintain its maximum pumping speed, the displacement of the first pump relative to that of the exhausting means being in the range of 5 to 20 to one.

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