ANALYSIS OF A HIGH-SPEED VACUUM SYSTEM

by

Ben Segal

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ABSTRACT

A study has been made of a vacuum system designed around a four-inch oil diffusion pump, which had been constructed at this Laboratory. Measurements made before this project began had seemed to indicate that the pump was not operating satisfactorily. From the results shown here, it appears that this conclusion must have been due to the manner in which the system had been operated and not to the behaviour of the pump itself. Under optimum conditions the behaviour of the pump compared favourably with that of its commercial prototype.

Three types of refrigerated traps were designed and constructed, and their conductances and trapping efficiencies measured. Several oil baffles were also constructed and their conductances measured.

A Pirani gauge circuit was designed and constructed to measure pressures over the range 1 to 100 μ Hg. A "high-speed" ionization gauge was also constructed.

I. INTRODUCTION

Vacuum systems having very low ultimate pressures are often used in this Laboratory for the evacuation of experimental vacuum tubes. While most of these tubes are designed to be sealed-off at a very low pressure, some must of necessity contain sliding or rotating joints or must be readily taken apart to permit changes to be made inside the tube. Such tubes require the use of one or more "O"-ring seals which invariably allow small quantities of air to leak into the vacuum chamber. Tubes of this type must therefore be continuously pumped if a high vacuum is to be maintained, and the pumping system used must be capable of ejecting from the system a large volume of gas per unit time without permitting the pressure to rise excessively.

It was for this reason that several large oil diffusion pumps had been constructed at this Laboratory. These pumps, which were of all metal construction, were similar in design to a commercially available high-speed glass-metal pump. When these metal pumps had first been put into operation and tested, it had appeared that their behaviour was quite unsatisfactory for high-speed applications. There was, however, some doubt as to whether the results obtained due to faulty construction of the pumps or to the manner in which they had been operated.

This project was intended to provide a thorough analysis of these pumps and other components of the complete vacuum system. As part of this project a study was made of the way in which the heating power supplied to the pump, and the backing pressure affect the behaviour of the overall system.

Since no sufficiently accurate gauge was available to measure the pressure at the exhaust end of the metal pump, a Pirani gauge circuit was constructed and calibrated. A "high-speed" ionization gauge was also designed and constructed to overcome the errors introduced when a low-conductance path exists between the gauge and the volume whose pressure is being measured.

In setting up the high vacuum system, considerable attention was paid to the problem of back-diffusion of molecules of pumping fluid from the pump to the vacuum chamber. In a practical application of the system this back-diffusion would not only raise the total pressure in the chamber, but would likely contaminate any sensitive surface which might be present, such as the emitting surface of an oxide cathode. Two steps were taken to prevent these oil molecules from reaching the vacuum chamber. The first step consisted in placing a mechanical baffle at the mouth of the diffusion pump to intercept the oil molecules diffusing back from the pump, and to return the condensed fluid to the pump reservoir. A liquid-nitrogen cooled trap was then placed above the baffle to freeze any molecules which re-evaporated from the surface of the baffle and continued towards the high-vacuum chamber. Several traps and baffles were constructed and their properties studied both for air and for carbon-dioxide which is condensible at liquid-air temperatures.

11. THEORY

Gas Flow at Low Pressures

At a pressure of 10^{-4}mm. Hg. a gas molecule in dry air at 25°C may be expected to travel 500 mm. before it collides with another gas molecule; at lower pressures the mean free path for intermolecular collisions increases inversely as the total pressure. If the dimensions of the vacuum chamber are such that a gas molecule cannot travel more than 50 or 100 mm. without colliding with the walls of the vessel, then under conditions of high vacuum (total gas pressure less than 10^{-4}mm. Hg.) the flow of gas is controlled by the physical dimensions of the vacuum chamber. Each gas molecule, in effect, is "unaware" of the presence of its neighbours and each acts freely and independently of all others. The flow of gases under such conditions is commonly called "free-molecule" flow.

The diagram in fig. 1 represents two vessels of volumes V_1 and V_2, respectively, completely closed except for a tube or aperture which joins them. If the pressures inside these vessels are P_1 and P_2, then a net flow of gas will take place from the vessel of higher pressure, say V_1, to the other. If both P_1 and P_2 are sufficiently small that any flow taking place will be of the "free-molecule" type, then the total quantity of gas travelling from V_1 to V_2 per unit time will be directly proportional to the number of molecules in V_1; and the volume of this gas (measured at pressure P_1) leaving V_1 per unit time will be independent of P_1 and will depend only on the physical dimensions of the path connecting V_1 and V_2. If this volume of gas leaving V_1 per unit time is F, then since F is characteristic of the physical connection only, there must be a similar flow of F (measured at pressure P_2) travelling from V_2 to V_1. The net flow of gas from V_1 to V_2 will then be

$$Q = P_1 F - P_2 F$$

Since the total quantity of gas, Q_1, in V_1 at any instant is

$$Q_1 = P_1 V_1 \quad ,$$

therefore the net flow Q out of V_1 must be

$$Q = -V_1 \frac{\partial P_1}{\partial t}$$

Hence,

$$Q = -V_1 \frac{\partial P_1}{\partial t} = F(P_1 - P_2). \tag{1}$$

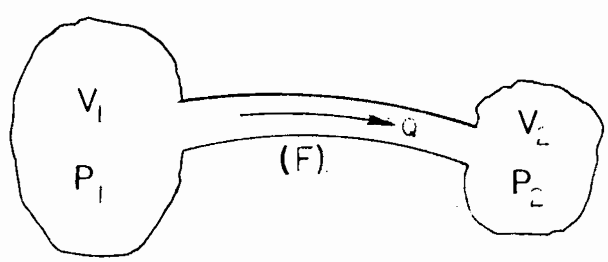


FIG.1. ILLUSTRATING FREE—MOLECULE FLOW FROM ONE VESSEL TO ANOTHER

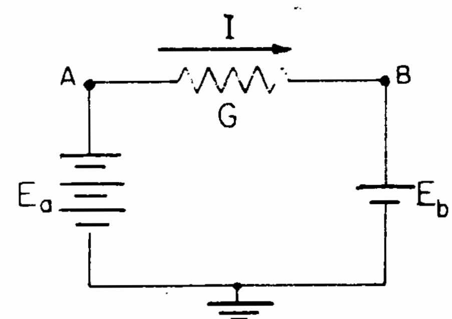


FIG.2. ELECTRICAL ANALOGUE OF FIG.1.

We may compare this gas flow with the flow of electricity in the circuit shown in fig. 2. In this circuit there will be a net current flow from A to B of

$$I = G(E_a - E_b) \; , \tag{2}$$

where E_a and E_b are the relative potentials of points A and B respectively, and G is the electrical conductance between A and B. By comparing equations (1) and (2) we see that the amount of gas, Q, flowing in a tube is analogous to a current flowing in an electrical circuit. Pressure corresponds to an emf. and F is analogous to an electrical conductance. It is because of this latter similarity that F is in fact referred to as the "conductance" of the connecting tube. This analogy is enhanced if we remember that for elements which obey Ohm's law the electrical conductance is a property of the path involved and is independent of the emf. across it.

Theoretical formulae are available (1) for the conductance of apertures or tubes of simple geometrical shape. The Appendix to this thesis also contains charts which are useful for determining conductances.

Vacuum Pump Operation

The purpose of a vacuum pump is to trap as many gas molecules as possible of those which wander into its mouth from the vacuum chamber, and to drive these out of the system. Just as the conductance of a device is a measure of the volume of gas per unit time which will flow from one end to the other, so the speed of a pump is a measure of the volume of gas which leaves the high vacuum chamber per unit time, and is driven by the pump into the exhaust line. Like conductance, the speed of a pump is commonly expressed in litres$\cdot$ sec^{-1}.

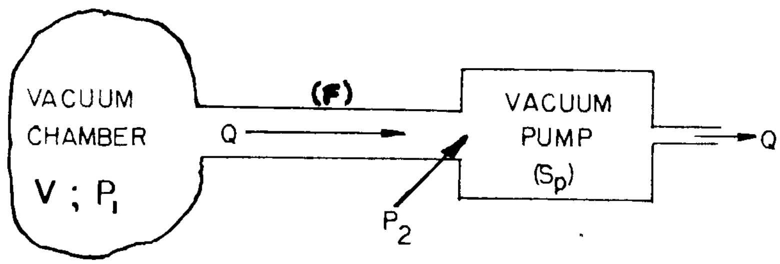


FIG.3. CHAMBER JOINED TO VACUUM PUMP
THROUGH FINITE CONDUCTANCE

In fig. 3, a chamber of volume V at pressure P_1 is connected, by means of a conductance F, to a pump which drives gas from its mouth into the exhaust line at a constant rate S_p litres. sec^{-1}, (as measured at the pressure P_2 existing at the pump mouth). Thus the quantity of gas removed from the system is $Q = S_pP_2$. In the absence of any leaks or evolution of gas from the conductance F, the gas ejected by the pump must come from the chamber V, and so $Q = (P_1 - P_2)F = S_pP_2$. Although the pump drives gas from its mouth at a speed $S_p = \frac{Q}{P_2}$, the rate which gas from V leaves the system is only $S_E = \frac{Q}{P_1}$; thus

$$S_E = \frac{Q}{\frac{Q}{S_p} + P_2}$$

$$= \frac{1}{\frac{1}{S_p} + \frac{1}{F}}$$

or $$\frac{1}{S_E} = \frac{1}{S_p} + \frac{1}{F} \qquad (3)$$

Henceforth, S_p will be called the "intrinsic speed of the pump" and S_E, the "speed of the system" will approach S_p only if the conductance F is very large compared with S_p.

Unfortunately, in any practical vacuum chamber there will be an evolution of gas taking place from the walls of the chamber. Consequently, while gas is "removed" from V at a rate Q, there will be a net removal of (Q-q) where q is the rate of evolution of gas into the chamber (including any leakage). Thus the observed rate of exhaust of V will be $S_x = \frac{Q-q}{P_1}$. As the pressure P_1 drops, the rate Q at which gas is removed will diminish until an "ultimate" pressure P_o is reached such that Q just equals the rate of evolution q. At this pressure we will have

$$Q = S_E P_0 = q,$$

and so

$$P_o = \frac{q}{S_E}$$

Thus,

$$S_x = \frac{Q-q}{P_1} = S_E - \frac{P_o S_E}{P_1}$$

or,

$$\frac{S_x}{S_E} = 1 - \frac{P_o}{P_1}. \tag{4}$$

Although the observed speed of exhaust S_x is a function of the pressure P_1, equation (3) shows that S_E, which represents an intrinsic speed of the system (including F) is independent of the pressure.

Equation (4) is a "universal" curve relating the speed of exhaust with the pressure in the chamber, subject to the parameters S_E and P_o. Fig. 4 shows the form of this curve.

Of the many different types of vacuum pumps available, diffusion pumps (including the so-called condensation pump) are most commonly employed for the production of very low pressures. Fig. 5 illustrates the elements basic to the operation of a diffusion pump.

In this pump, a fluid having suitable properties is boiled and the vapours forced into a long tube by means of a jet at one end. Gas molecules which wander from the vacuum chamber into the vapour stream are carried to the other end of the tube where they are removed by a "backing" pump. The condensed vapours at this end are returned to the fluid boiler. The intrinsic speed of such a pump depends on the design of the pump jets, on the nature of the fluid used and the heat supplied to it, and on the pressure at the exhaust end of the pump. This pressure is generally referred to as the backing pressure.

If it were possible for a diffusion pump to "trap" every gas molecule which wandered into its mouth and to drive these out of the system, then the speed of the system would be limited by the conductance of the path between the chamber and the vacuum pump. The maximum pumping speed which can be obtained from an ideal pump is therefore the value of the conductance of an aperture equal in area to the mouth of the pump. The ratio of the actual intrinsic speed of a pump to the conductance of such an aperture is called the Ho coefficient of the pump. For most diffusion pumps the Ho coefficient lies between 5 and 30%.

Fig. 7 is a photograph of the high-speed metal pump and fig. 6 is a sketch of the jet assembly used. The two lower jets operate

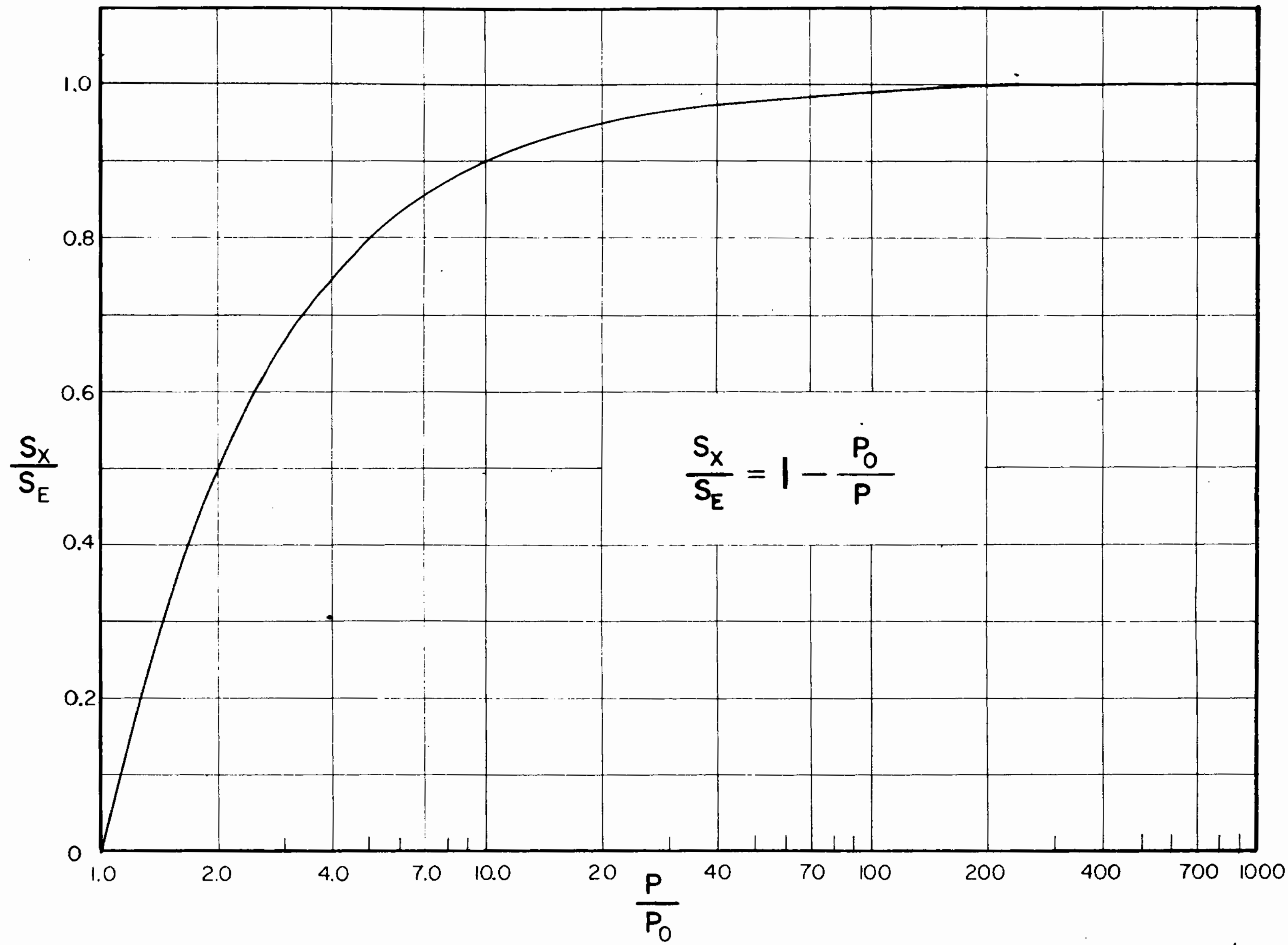


FIG. 4. UNIVERSAL SPEED – PRESSURE RELATIONSHIP

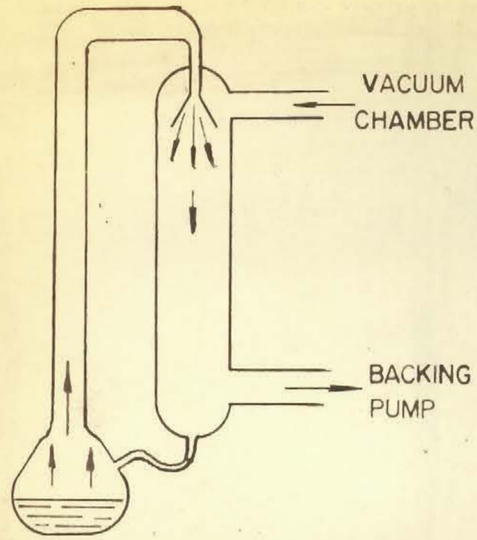


FIG.5, BASIC DIFFUSION PUMP

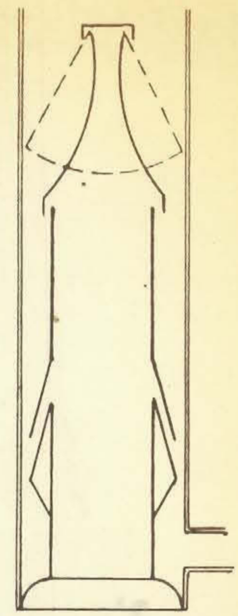


FIG.6 SKETCH OF JET ASSEMBLY USED IN METAL PUMP

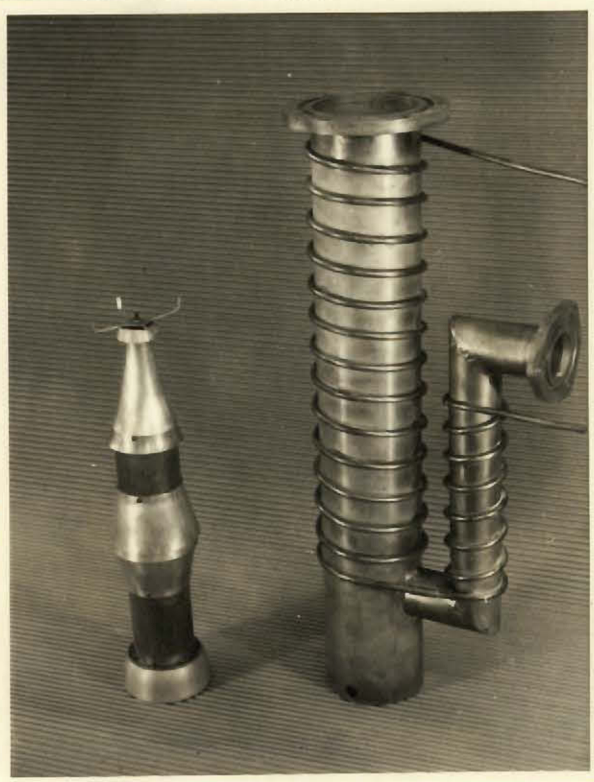


Figure 7 - Photograph of metal pump used during this project, showing the spun aluminum jets.

in a straightforward manner; the upper jet, which is known as an Embree jet [2], has been designed to reduce backstreaming of hot oil molecules and to provide a greater Ho coefficient than is obtained with most other designs. The operation of this jet is adequately described in the reference quoted and attention is here drawn only to the following statement: "The collecting zone with the Embree jet is not measured by the annulus between the jet gap and the walls but by the area of the cone indicated by the dotted line in fig. 6. The speed of the pump is thus the summation of the mechanical admittance to the jet region and the diffuse admittance through the cones of actuating vapours". By use of this design, pumps can be constructed which have Ho coefficients of more than 30%, compared with the more usual 10-20% in other designs.

For a pump of given jet design we may expect to find an optimum heater power depending on the type of fluid used and the heat transfer properties from the heating element to the pump boiler. If too little power is supplied to the pump, then there will not be sufficient vapour produced to give efficient pumping action. Overheating, however, is more serious than insufficient heating. Although increasing the heat supplied to the boiler increases the amount of vapour generated and thus tends to raise the intrinsic speed of the pump, overheating encourages fluid breakdown with a consequent reduction in pumping efficiency. Several workers have noted that Octoil-S (Di-2-ethyl-hexyl sebacate), which was the pumping fluid used during this project, and other esthers of sebacic acid will decompose catalytically on aluminum surfaces at high temperatures. Since the jets of the metal pump were constructed of spun aluminum, one might expect to have severe fluid breakdown with this pump unless the heat is carefully restricted. The end products of such decomposition consist of a dark tarry residue and of volatile fractions which readily diffuse back to the vacuum chamber unless a refrigerated trap is used to condense them. The region of optimum heater power may thus be considered as that giving the lowest ultimate pressure in the absence of a cold trap.

The backing pressure maintained is an important parameter because there is a finite probability that a gas molecule may diffuse backwards from the exhaust end, through the jet stream and up to the mouth of the pump. The rate at which gas will diffuse back in this manner depends largely on the pump design and on the backing pressure.

Mechanical Oil Baffle

An oil baffle consists of some form of obstruction placed at the mouth of the pump so that an oil vapour molecule must collide with it at least once before it can pass from the pump to the vacuum chamber. Since most of these colliding molecules will condense on the surface of the baffle, it is important that the baffle be so constructed that the condensate will return to the pump boiler. Furthermore, it is desirable that the baffle be so designed that the condensed fluid does not drip directly onto the hot jet assembly from which it would quickly re-evaporate.

The use of a baffle does not completely solve the problem of oil backstreaming since some of the condensed fluid evaporates from the baffle and continues back towards the vacuum chamber. Hickman (2) states that "backstreaming may be cured by a mechanical baffle, but the humidity [in the vacuum chamber] can be decreased only by lowering the temperature of the pump intake below that of the evacuated chamber".

It is evident that since an efficient baffle must obstruct the flow of oil molecules travelling back from the pump, it will also impede the diffusionn of gas from the vacuum chamber to the pump. In the design of a high-speed pumping system, it is of considerable importance, in view of equation 3, that the conductance of the baffle be as high as possible. The usual configuration for an oil baffle is a series of rings as shown in figure 8. It can readily be shown that the conductance of such a baffle is a maximum for the case of only two rings having equal areas. In computing these

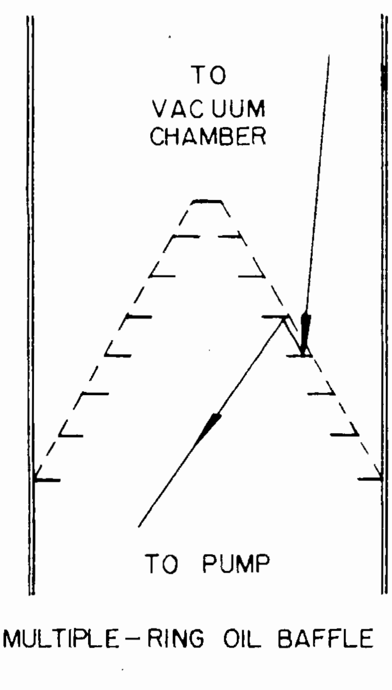


FIG. 8. MULTIPLE-RING OIL BAFFLE

conductances, no account is taken of the fact that a gas molecule may pass through the baffle by means of a series of reflections from one ring to another, as shown in the sketch; therefore, we may expect the actual conductance of a baffle to be somewhat greater than calculated.

Refrigerated Vapour Traps

To prevent any vapour molecules which may evaporate from the surface of the baffle from reaching the high vacuum chamber, it is customary to insert a cold trap between the baffle and the chamber. A satisfactory cold trap consists of a high conductance section joining the vacuum chamber to the pump, and suitably constructed so that it may be kept refrigerated. The usual refrigerant for this purpose is liquid nitrogen which keeps the walls of the cold trap at about -196°C. This ensures that any vapour molecules colliding with the walls will be frozen onto the surface. In a relatively clean leak-free system, where the rate of evolution of gas in the chamber is low, the ultimate pressure is determined mainly by the fluid vapours in the chamber. The insertion of an efficient cold trap will reduce the ultimate pressure to the point where it is limited by the rate of evolution. This may reduce the ultimate pressure by a factor of 10 to 10^3 or more if the chamber is very clean.

In the design of a cold trap the problem of conductance again arises. In order to freeze oil vapours effectively, the cold trap should be such that every vapour molecule must collide with it at least once before it is able to reach the vacuum chamber. This generally involves a low conductance to gas flow, which is undesirable if the system is to have a high pumping speed. Thus, a compromise must generally be made between high conductance and effective trapping.

Another important factor to consider in the design of a cold trap is the amount of liquid nitrogen (or other refrigerant) which may be "stored" in the trap. It is almost imperative that a good trap should maintain enough liquid nitrogen to operate for at least 10 or 12 hours without any loss in trapping efficiency.

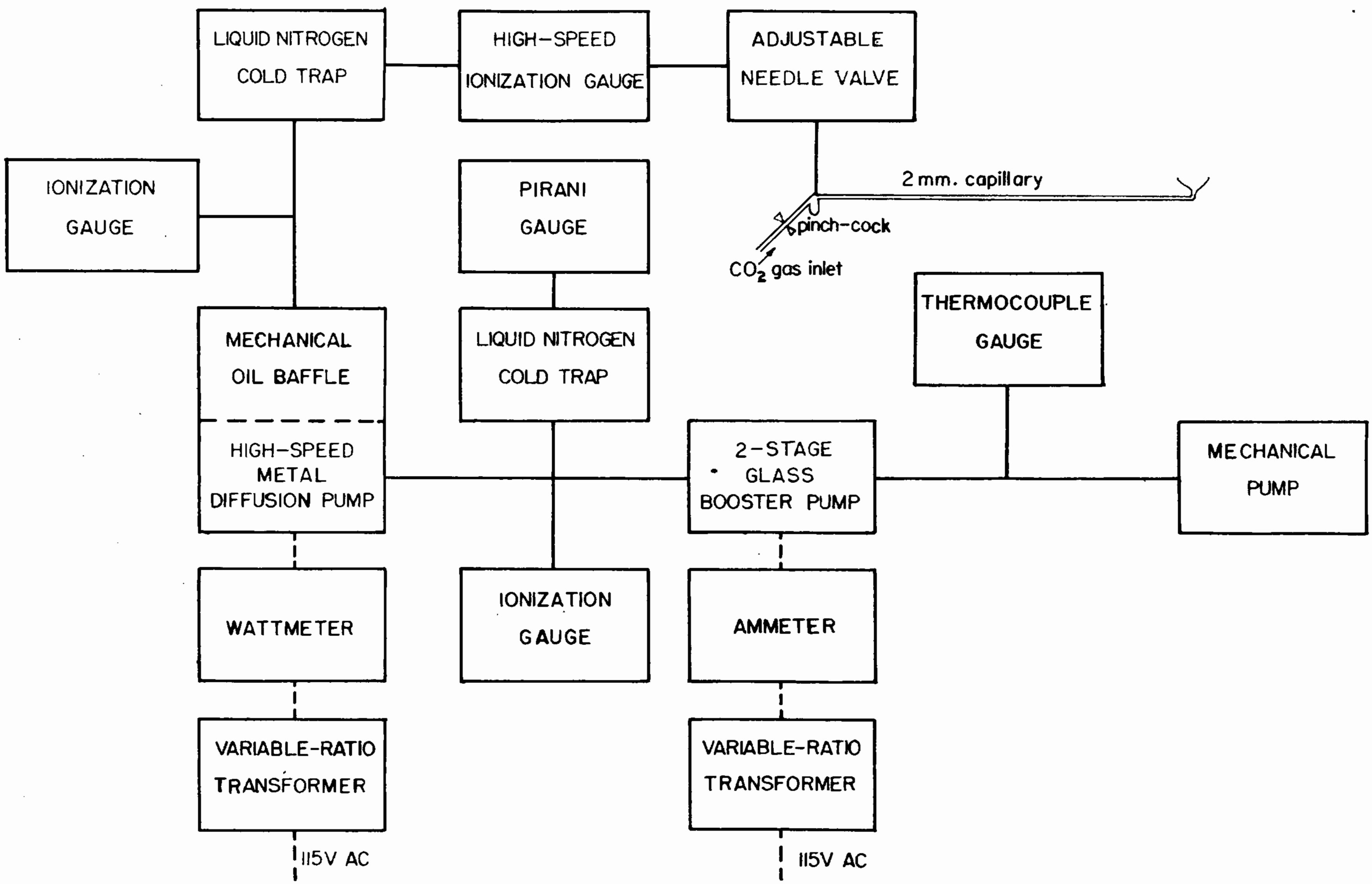


FIG. 9. BLOCK DIAGRAM OF VACUUM SYSTEM

III. APPARATUS AND INSTRUMENTS

The Vacuum System

Fig. 9 shows a block diagram of the vacuum system studied. Basically, the system consisted of the four inch diameter three-stage oil diffusion pump which was to be analyzed in detail, backed by a glass two-stage fractionating oil diffusion pump. The four inch pump used was an all-metal version of a commercial glass-metal pump (Consolidated Vacuum Corp. type GM 220; also Cenco type 93240), while the glass "booster" pump, which was also constructed at this Laboratory, was a copy of a well established commercial design (Consolidated Vacuum Corp. type GF 20 A or Cenco 93230 A). The glass booster pump was capable of providing backing pressures less than 10^{-4} mm. Hg. at the exhaust end of the metal pump. The booster itself was backed by a mechanical rotary pump. This pump, a Welch "Duo Seal" 1400 B, provided pressures ranging from 25 to 200 μ Hg. which were found to be sufficient for safe operation of the booster.

The electrical heaters of both diffusion pumps were connected to variable-ratio transformers. In the case of the metal pump this was done to observe the effect of heater power on the behaviour of the system. In the glass pump this control was used to vary the intrinsic speed, thereby controlling the backing pressure at the exit of the metal pump between 2×10^{-5} mm. and 100 μ Hg.

An Edwards adjustable needle valve (W. Edwards and Co., type LB-1) was used to admit a controlled leak into the high vacuum region. The actual rate of leak was measured by the standard constant-pressure method using a mercury pellet and capillary tube (3). If the cross-sectional area of the capillary tube is A and a mercury pellet is drawn along the tube with a velocity v, then the rate at which air is flowing through the needle valve is,

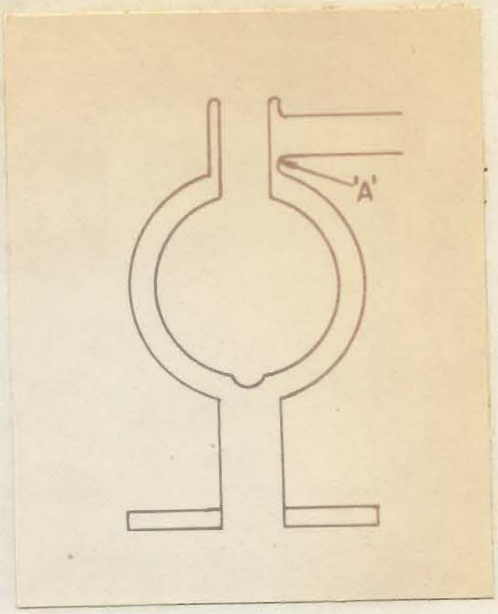
$$Q = BAv$$

where B is the atmospheric pressure. If B is expressed in mm. Hg., A in mm.^{2}, and v in mm. per second, then Q will be in lusecs.(1 lusec = 1 litre$\cdot\mu\cdot$sec^{-1}).

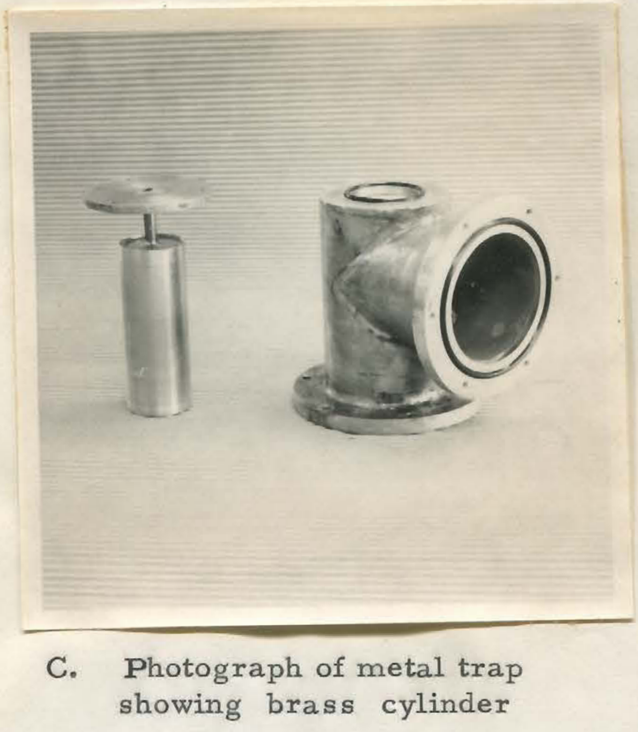
Traps and baffles

Fig. 10 shows the three liquid-nitrogen cold traps examined in the course of this project. The first trap studied was essentially a Dewar vessel consisting of a 2-litre balloon flask concentric with a 3-litre flask. This trap will hereafter be referred to as the "glass trap". The gas flow in this trap took place in the region between the two flasks. There are three distinct sections to consider when analyzing the nature of the gas flow in this trap - there is the neck of the trap consisting of a section of tubing 50 mm. in diameter, the region between the two flasks, and the constriction (marked "A" in fig. 10) where the flasks are connected to the tubing from the vacuum chamber. The theoretical conductances of these sections, based on their dimensions, are 105, 114, and 60 litres$\cdot$ sec^{-1},

A. Photograph of glass trap



B. Sketch of glass trap



C. Photograph of metal trap showing brass cylinder

D. Glass-metal trap

FIG. 10. LIQUID-NITROGEN VAPOUR TRAPS TESTED

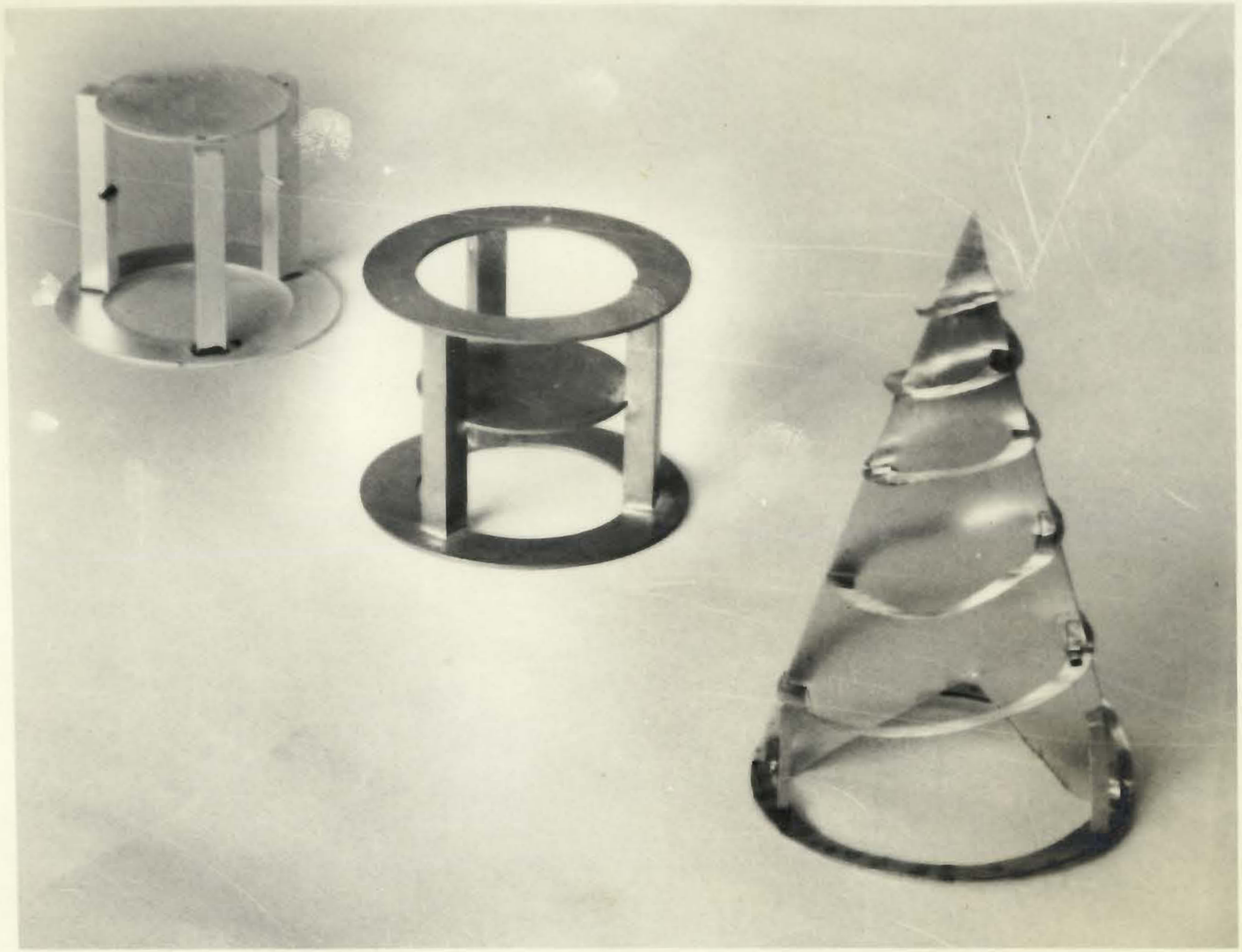


FIG. 11 PHOTOGRAPH OF THE THREE OIL BAFFLES TESTED

respectively. Combining these, the theoretical conductance of the entire trap is 27 litres$\cdot$ sec^{-1}. While this conductance might be sufficient for use in a system designed to be "sealed off", it is hardly sufficient for use in a high-speed dynamic system.

Two other cold traps were designed and built. The first of these was an all-metal trap consisting of a hollow 2-inch diameter brass cylinder, 5 1/4 inches long and suspended within a 4-inch stainless steel "tee" which acted as an extension of the pump wall. In this trap the liquid nitrogen was held in the brass cylinder. Heat conduction from the atmosphere to the cylinder was minimized by the use of a thin Inconel tube supporting the cylinder. The calculated (theoretical) conductance of this trap is about 250 litres$\cdot$ sec^{-1}.

The third trap was designed to have as large a conductance as did the metal trap, but was used with a 2-litre reservoir which resembled the glass trap. The trapping action took place mainly on a 35mm. diameter glass "finger" suspended in the metal tee. We will refer to this trap as the "glass-metal" trap.

Three mechanical oil baffles were constructed and their conductances were compared with the values determined theoretically. These baffles are shown in fig. 11.

Although the two-ring baffle has the maximum theoretical conductance for this type of construction, it is possible for vapour molecules to pass obliquely through the baffle and into the vacuum chamber without suffering a collision. The addition of a third ring to the baffle in the manner seen from fig. 11 overcomes this problem completely. The third baffle is a modified 6-ring construction. Instead of plain rings, the baffle is constructed in the form of a continuous spiral so that condensed fluid is able to flow down the walls of the pump and return to the boiler.

Pressure Gauges

Since the pressure in different parts of the vacuum system were of different orders of magnitude, a variety of instruments was required to conveniently measure the pressures throughout the system. Pressures in the region between the mechanical pump and the glass booster pump were measured with a thermocouple gauge (R. C. A. type 1946). The gauge unit had poor sensitivity and had not been accurately calibrated. Nevertheless, it sufficed to indicate when the pressure was low enough to permit safe use of the oil pumps.

The measurement of backing pressure at the exit of the metal pump required a gauge which would operate accurately and conveniently over the range 100 u to 1 μ Hg. or less. This requirement led to the construction and calibration of a Pirani gauge measuring unit.

Basically, a Pirani gauge is a length of high temperature coefficient resistance wire suitably mounted. If this element is placed in the vacuum chamber and heated slightly by a small d-c current, then its equilibrium temperature (and hence its resistance) will be a function of the pressure. Of course, different gases and vapours have distinctly different thermal conductivities even at the same pressure, so that after calibrating a Pirani gauge circuit for use with a particular gas, care must be taken to ensure that the gauge is not used in a different atmosphere. Certain vapours would also contaminate the gauge resistance element. As a result, the Pirani unit constructed was always used with a small liquid-Nitrogen cold trap which prevented pump oil vapours from reaching the gauge.

If a single Pirani gauge element is used in an unbalanced bridge circuit, the detector voltage will depend on the ambient temperature level so that the useful range of the circuit will be reduced as a result of fluctuations in room temperature. In constructing the present circuit, two reasonably "matched" elements were used in conjugate arms of the bridge circuit to provide a certain degree of temperature compensation.

The circuit, which is shown in fig. 12, was designed around the use of a Sylvania type R1110 resistance element. The compensating element was a Sylvania R111 which had previously been sealed at a very low pressure.

In its initial form, the bridge output voltage showed serious fluctuations due to unequal heating of the measuring unit and the compensating unit by radiation from ionization gauge filaments, incandescent lamps, etc. To overcome this, the elements were enclosed, side by side, within a common copper shielding box. The "zero drift" of the final unit was

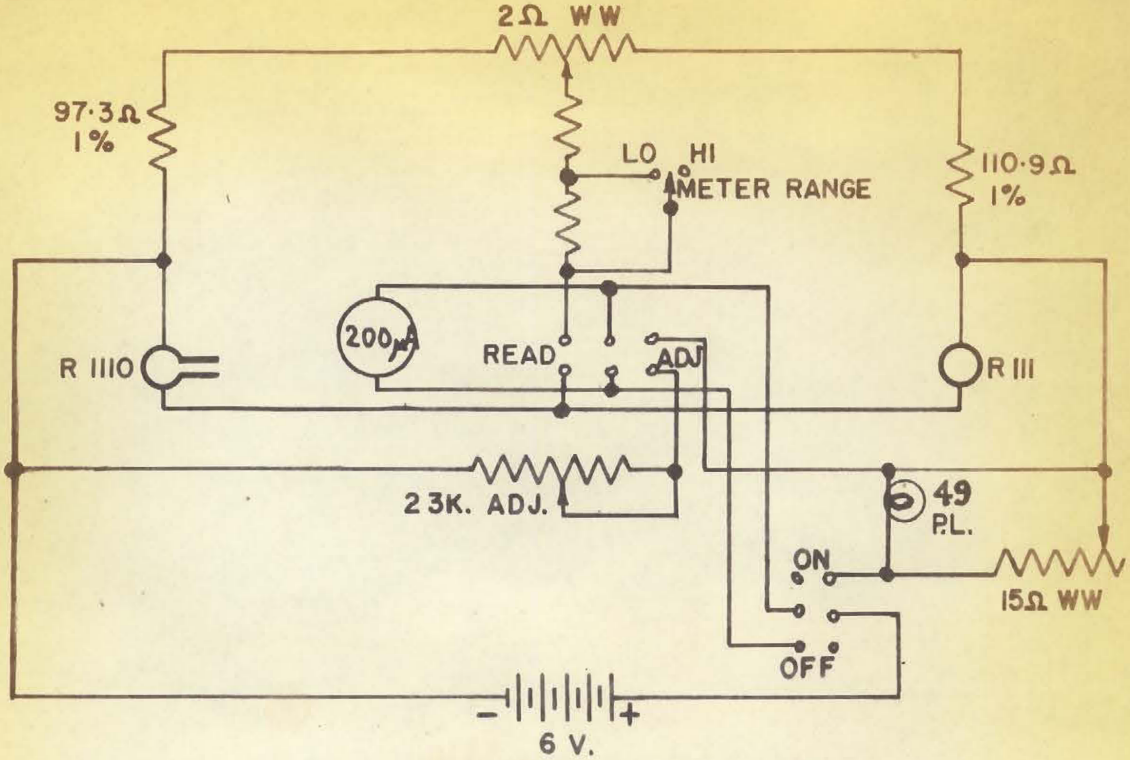


FIG. 12. SCHEMATIC OF PIRANI GAUGE UNIT.

FIG. 13. PHOTOGRAPH OF PIRANI GAUGE UNIT. THE RESISTANCE ELEMENTS ARE CONTAINED IN THE COPPER SHIELDING BOX AT THE LEFT.

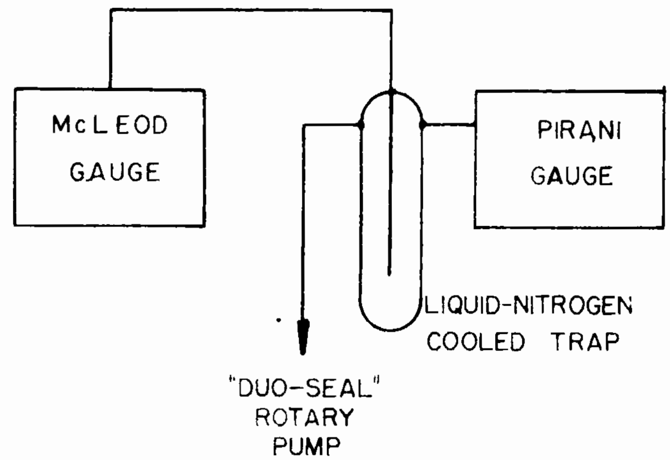


FIG.14. CALIBRATING ARRANGEMENT FOR PIRANI GAUGE

equivalent to about ±1/4 μ Hg. in one hour. Fig. 13 is a photograph of the shielding box containing the two elements, and of the unit housing the remainder of the circuit.

The circuit was calibrated by means of the arrangement shown in fig. 14. A McLeod gauge (Cenco type 94156) was used as the reference gauge over the range 0-30 μ Hg. Above 30 μ an Edwards "Vacustat" was used in place of the larger McLeod gauge. The "Vacustat" calibration had previously been checked against the McLeod gauge in the region below 30 μ Hg. The bridge detector used during the initial calibration was a Simpson 25-microampere meter. The voltage applied across the bridge was 4.06 ± 0.01 volts. The results of the calibration are shown in fig. 15. The gauge calibration proved to be linear over the range 0-120 μ Hg. having a constant sensitivity of 4.8_3 mV per u Hg.

The high-vacuum pressure measurements were made with hot-cathode ionization gauges. These gauges were constructed in the workshops of the Eaton Laboratory and resembled the commercially available Westinghouse WL-5966. The control and measuring circuit used in conjunction with this type of gauge was similar to the circuit given by Dushman[1].

The ionization gauge is generally mounted inside a glass envelope and is joined to the vacuum chamber by means of a short tube whose conductance is typically of the order of 10 litres$\cdot$ sec^{-1}. Blears [4] has pointed out that this type of connection can lead to serious errors due to the pumping action exhibited by the gauge while it is operating. To reduce such errors, a "high-speed" gauge was constructed in which the electrodes of the gauge were entirely contained inside a section of 50 mm. diameter glass tubing which joined the vacuum chamber to the pumping system. A photograph of this gauge is shown in fig. 16 together with a more conventional type of gauge.

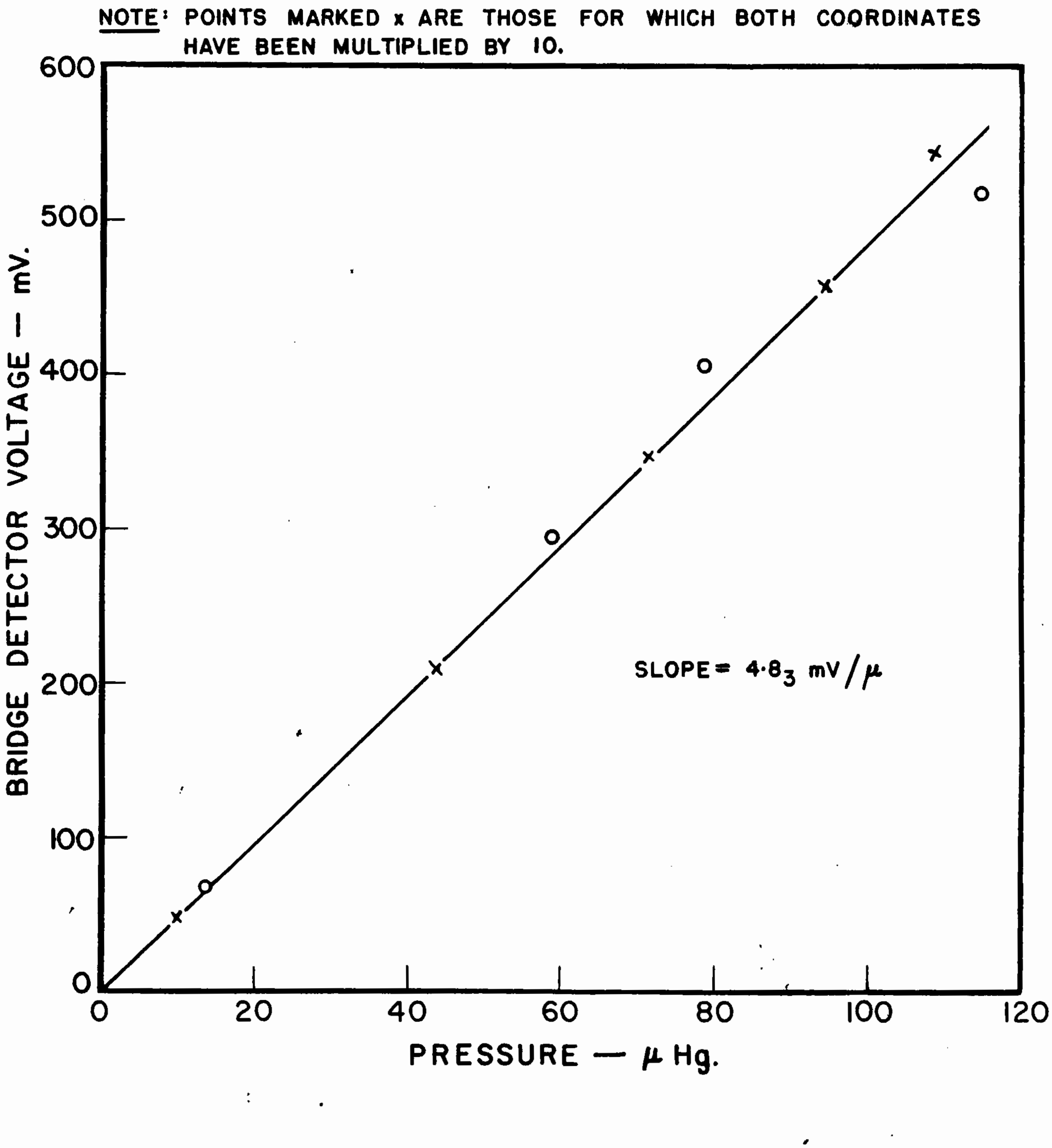


FIG. 15. CALIBRATION OF PIRANI GAUGE CIRCUIT.

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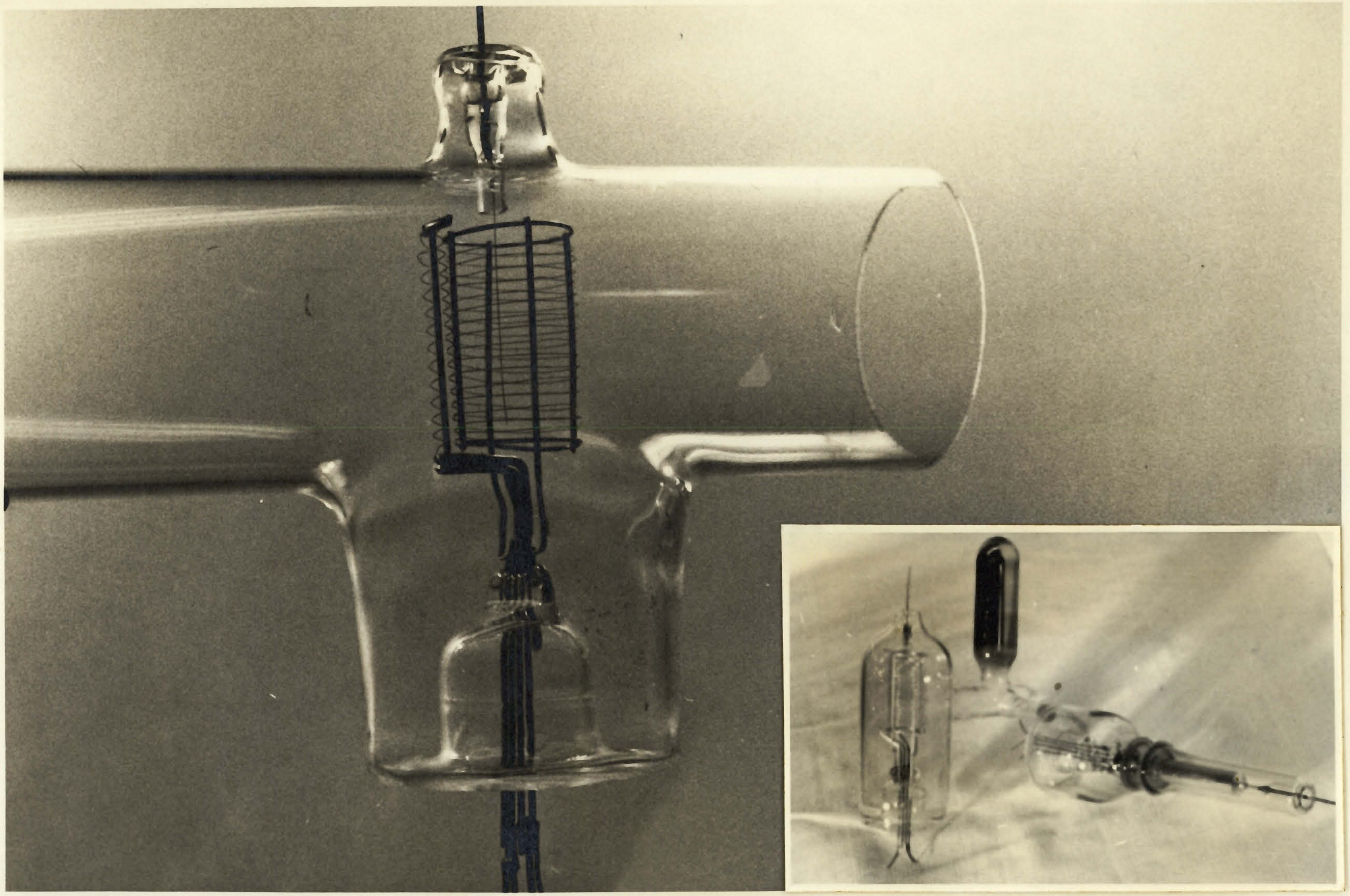

FIG. 16 A PHOTOGRAPH OF THE HIGH-SPEED IONIZATION GAUGE WITH INSERT SHOWING A MORE CONVENTIONAL MOUNTING

IV. EXPERIMENTAL RESULTS

Behaviour of Metal Pump

Improper design of the jets in a diffusion pump, or of the clearance between the jets and the walls of the pump, can sometimes result in excessive back-diffusion of gas from the exhaust end to the mouth of the pump [2]. If this is the case, then the behaviour of the pump will depend strongly on the backing pressure. To investigate the behaviour of the metal pump as a function of backing pressure, the heater current to the booster pump was adjusted until the resulting pumping speed gave the desired backing pressure. By this means it was possible to supply any desired backing pressure down to about 2×10^{-5} mm. Hg. when no air was being admitted into the vacuum chamber. Backing pressures as low as 10^{-4} mm. were usually attainable when the amount of gas admitted was not too great.

Fig. 17 shows the variation in "ultimate" pressure at the high-vacuum end of the system as a function of the power supplied to the metal-pump heater without the use of a cold trap. Different curves are drawn, showing the effect of backing pressure on the ultimate pressure. It can be seen that for backing pressures of about 1 μ Hg. or more, the rate of back-diffusion is so high that the pump cannot effectively overcome this for heater powers up to 400 watts. With a backing pressure of 10^{-4} mm. Hg. the back-diffusion is greatly reduced and is small compared with the pumping speed for a heater power of about 280 watts. This is characterized by a region of fairly constant "ultimate" pressure. A shallow minimum in the ultimate pressure curve is reached in the neighbourhood of 310 watts heater power, and the "ultimate" rises for heater powers greater than about 360 watts, probably due to the increased rate at which the pumping fluid is cracked. After operating the pump for about 30 hours at 400 watts, it was found that as a result of severe oil decomposition, an extra 20 watts of power was needed to approximately reproduce the curves in fig. 20. The fluid was examined at this stage and was found to be extremely dark and opaque, and the aluminum jets were coated with a tarry deposit.

Fig. 18 shows the ultimate pressure obtained at different heating power levels as a function of the backing pressure. Fig. 19 shows more completely the effect of backing pressure on the fine pressure.

The effect of backing pressure on the speed of the pump may be seen from fig. 20. In taking these speed measurements the rate at which gas was admitted into the system was generally so great that the fine pressure was large compared to the ultimate pressure. For those cases where these pressures were of the same order, a correction was applied as indicated by equation (4). We see that for backing pressures greater than 1 μ Hg., the effect of backing pressure on the speed of the

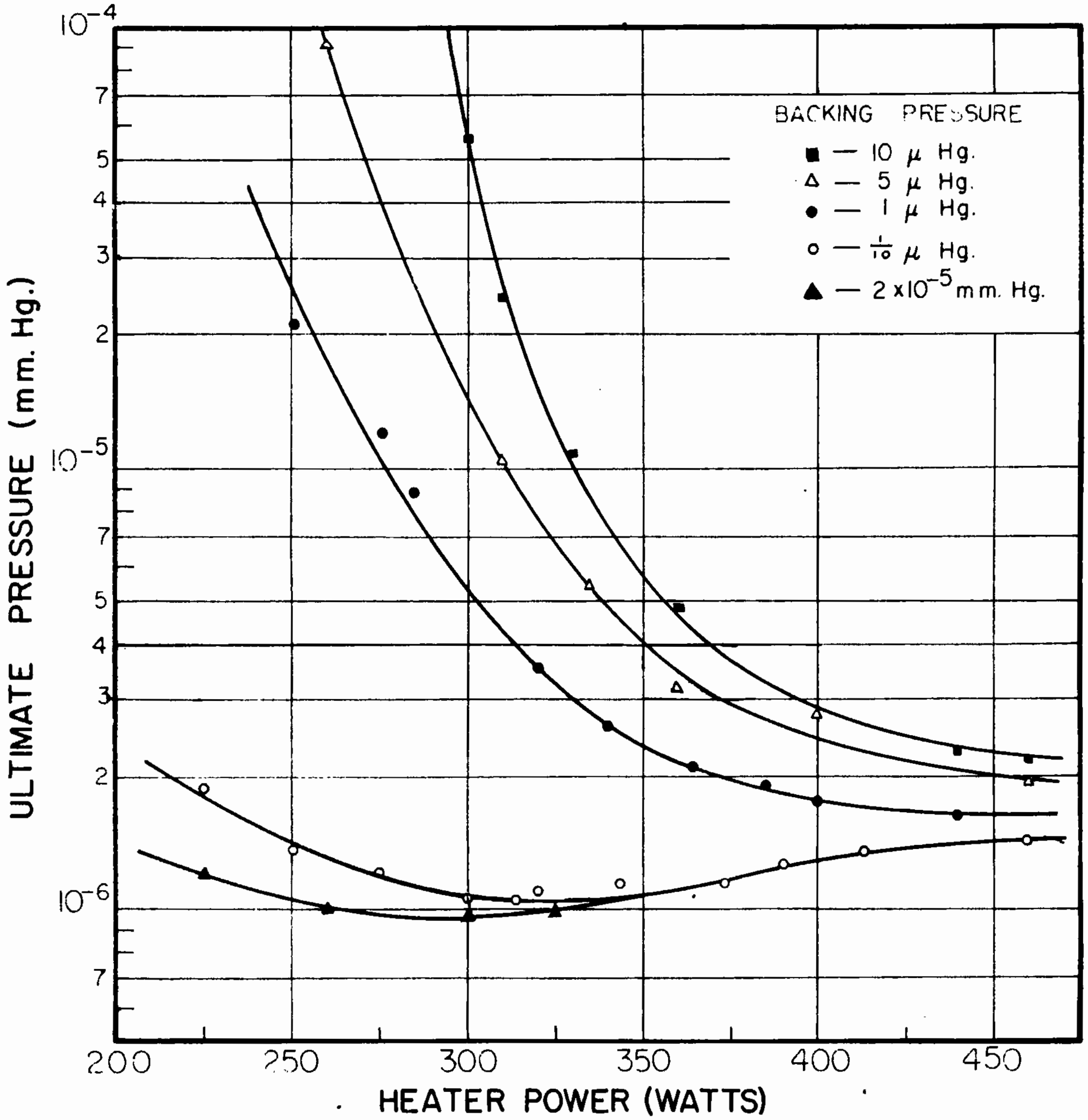

FIG. 17. ULTIMATE PRESSURE vs. METAL PUMP HEATER POWER.

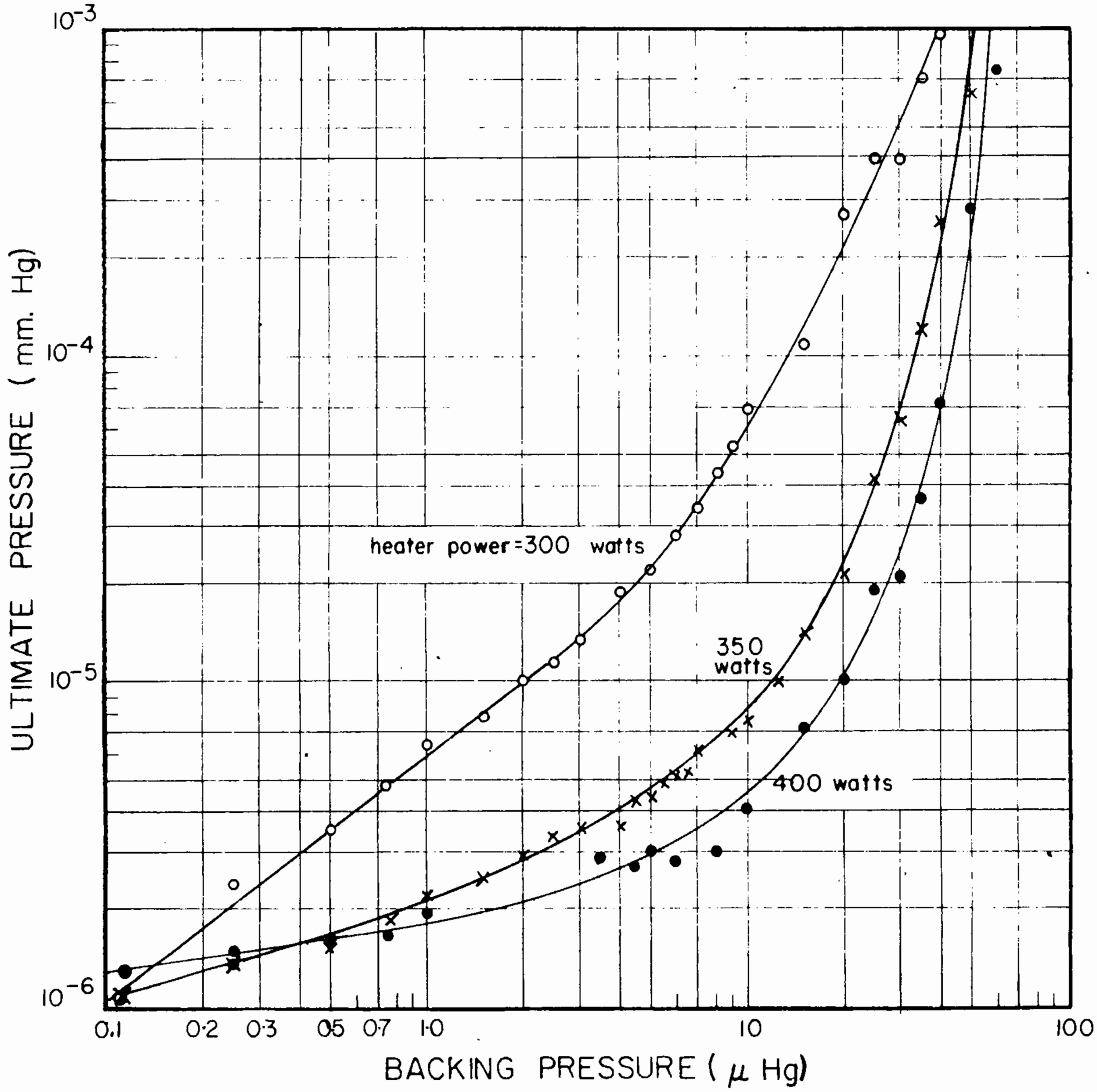

FIG. 18. EFFECT OF BACKING PRESSURE ON ULTIMATE VACUUM

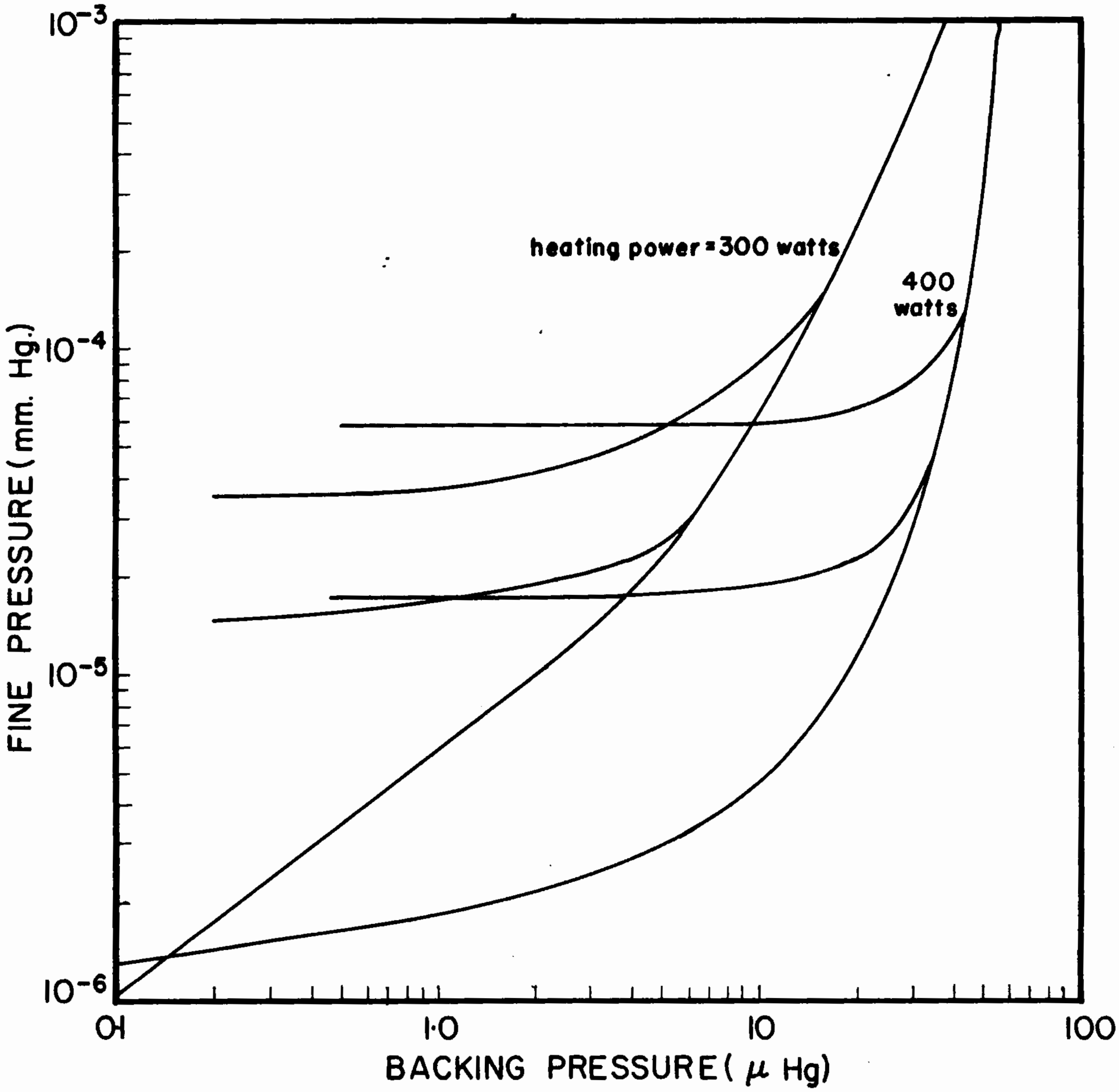

FIG. 19. EFFECT OF BACKING PRESSURE ON FINE PRESSURE AT DIFFERENT POWER LEVELS FOR SEVERAL ARBITRARY VALUES OF GAS LEAKAGE INTO SYSTEM. EXPERIMENTAL POINTS HAVE BEEN OMITTED FOR THE SAKE OF CLARITY.

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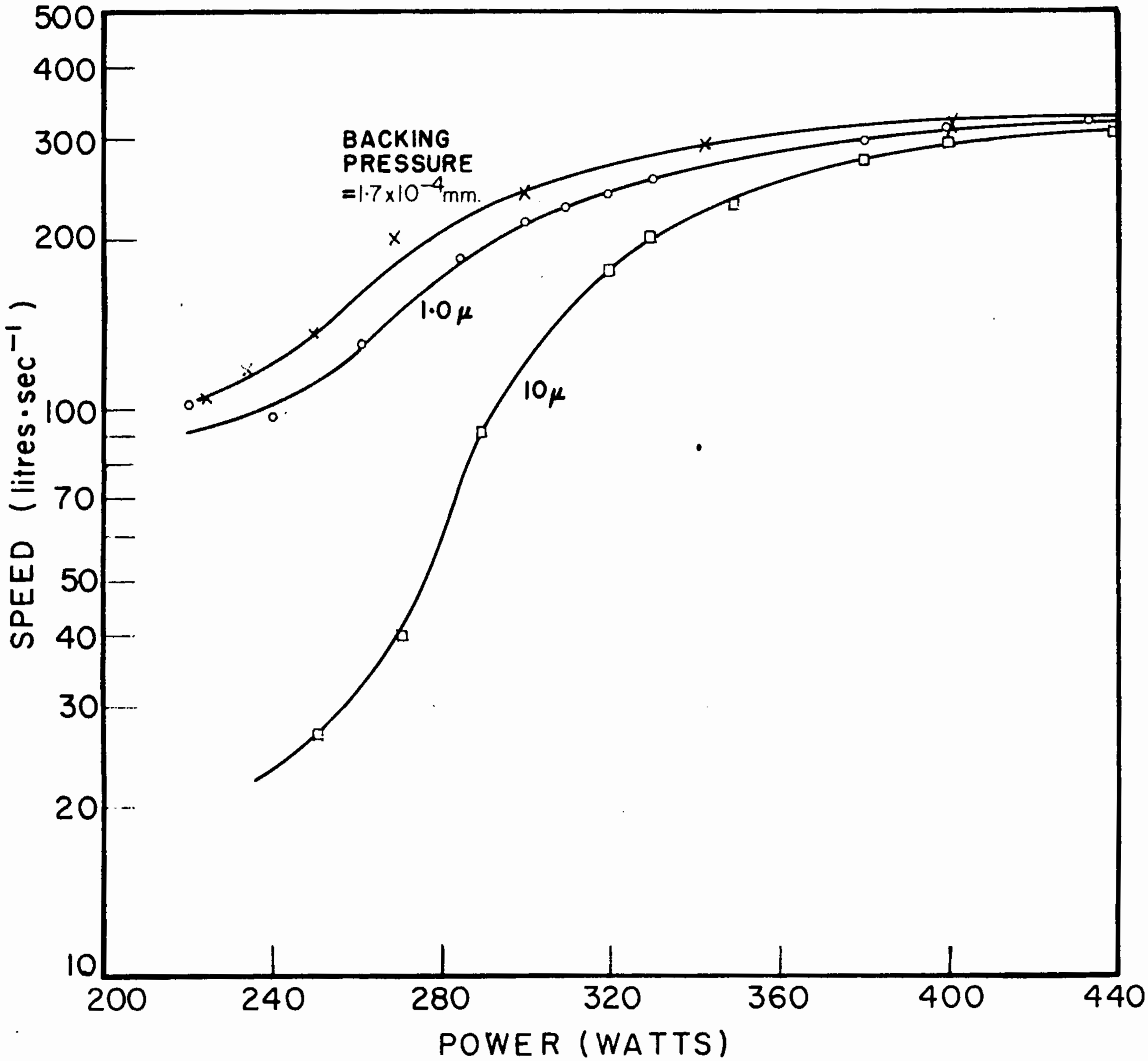

FIG. 20. INTRINSIC SPEED OF METAL PUMP AS A FUNCTION OF HEATING POWER.

 follows page 35.

pump is marked, especially when heater powers less than about 330 watts are used.

Refrigerated Traps

As previously mentioned, three liquid-nitrogen cooled traps were studied. The main factors to consider in evaluating a cold trap are conductance, storage properties for the liquid nitrogen, and its efficiency in trapping condensible vapours. The conductance of a cold trap was measured by the standard constant-pressure method, using two ionization gauges - one on either side of the trap - and a controlled leak from the atmosphere into the vacuum chamber.

An estimate of the trapping efficiency was obtained by admitting carbon dioxide gas into the vacuum chamber at a known rate and observing the pressure rise on the pump side of the refrigerated trap. This rate was then compared with the rate at which air was admitted to give the same pressure rise, suitable corrections being made for the effect of using different gases on the gauge sensitivity. The ratio of the rate at which air was admitted to the rate at which carbon dioxide was admitted, in accordance with the above scheme, was called the "trapping ratio".

Although the "trapping ratio" obtained in this manner was directly proportional to the probability that a molecule of carbon dioxide would be frozen by the trap, the conditions under which these measurements were made were not always consistent with the requirements for free-molecule flow. In an attempt to maintain a pressure of about 5×10^{-5} mm. Hg. at the mouth of the pump while the carbon dioxide was being admitted, the pressure on the side of the trap remote from the pump rose to well above 1 μ so that the flow taking place in this region was primarily viscous or streamlined flow. For this reason, the "trapping ratio" obtained in this manner is not too accurate an indication of the efficiency of a trap as it is actually used in a vacuum system. Table I lists the properties of the three traps tested, and fig. 21 shows the improvement in the ultimate pressure effected by the insertion of a liquid-nitrogen cooled trap.

Mechanical Baffles

Fig. 11 shows the three mechanical oil baffles which were examined during this project. It was found difficult to devise a method for experimentally determining the effectiveness of these baffles in intercepting and returning pumping fluid which diffused back from the pump. All three baffles were constructed on a multiple-ring design which did not permit any molecule to travel vertically through the baffle without suffering a collision. One baffle (the three-ring one) was so constructed that it would intercept all back-streaming vapour molecules regardless of their incident angles; except for this feature, little can be said regarding the relative effectiveness of the baffles.

The conductances of the baffles were measured indirectly. First, the speed of the system was measured with the

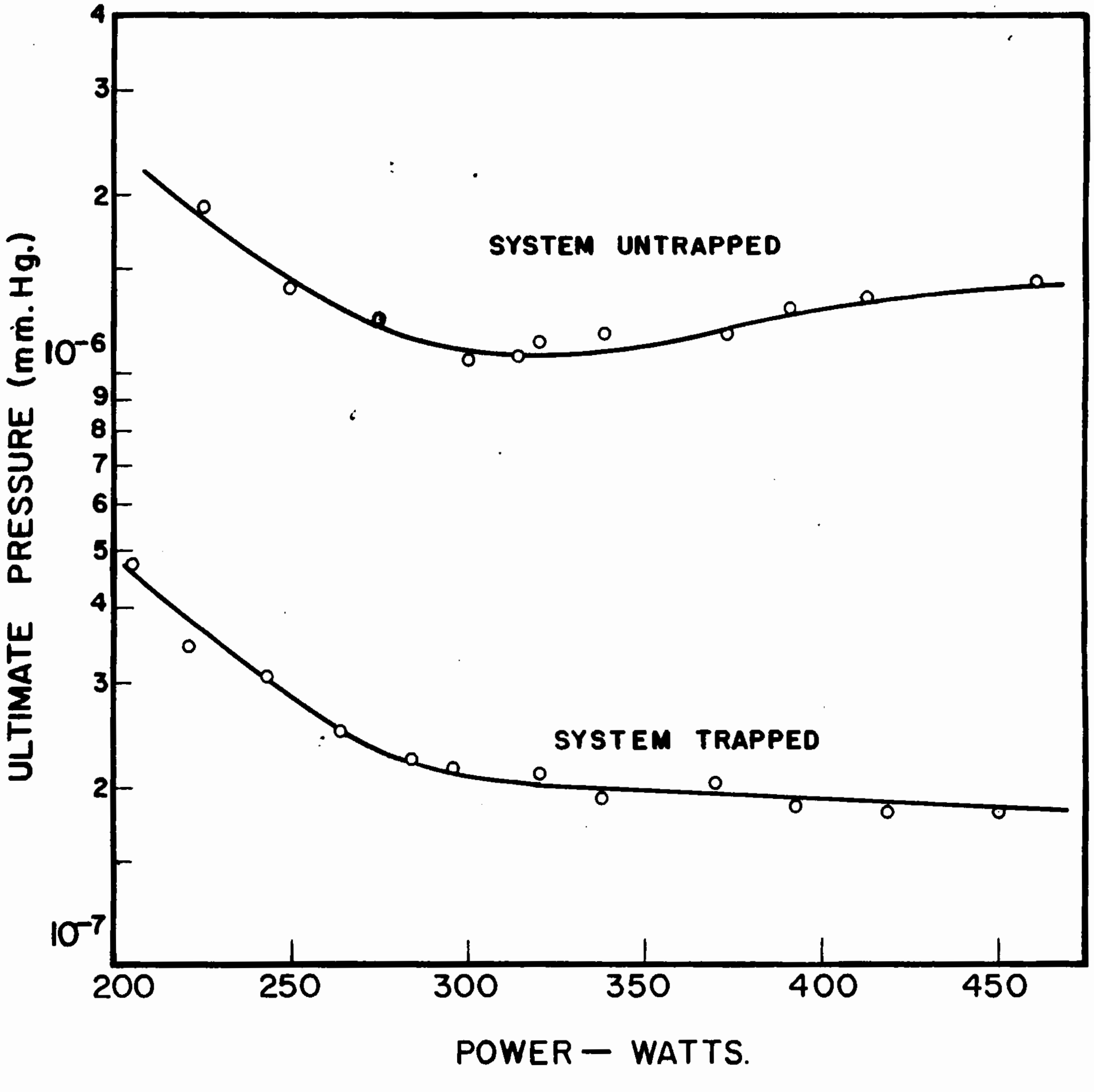

FIG. 21. ULTIMATE PRESSURES OBTAINED WITH AND WITHOUT THE USE OF LIQUID-NITROGEN TRAPPING.

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TABLE I

PROPERTIES OF REFRIGERATED VAPOUR TRAPS

TRAP	THEORETICAL CONDUCTANCE (litres/sec)	MEASURED CONDUCTANCE (litres/sec)	LENGTH OF STORAGE	TRAPPING RATIO
GLASS	27	21.5	$2\frac{1}{2}$ days	$>10^4$
METAL	250	232	$2\frac{1}{2}$ hours	160
GLASS-METAL	333	>300	$13\frac{1}{2}$ hours	17

TABLE II

PROPERTIES OF OIL BAFFLES

BAFFLE	THEORETICAL CONDUCTANCE (litres/sec)	MEASURED CONDUCTANCE (litres/sec)
2-ring	222	>300
3-ring	148	250
6-ring spiral	~ 124	215

appropriate baffle in situ; the speed of the system was then measured with the baffle removed and the conductance calculated from equation (3). The conductance data for the three baffles examined appear in Table II.

Discussion

It is interesting to compare the results shown in fig. 18 with those obtained by Hickman [2]. Hickman carried out a series of investigations on a pump of very similar geometrical design. His pump was of glass outer wall construction with metal jets (Consolidated Vacuum Corp. GM 220 A). Hickman's procedure was essentially as follows. To the oil of the pump (Octoil-S) he added a small quantity of dimethylaminoanthraguinone, a blue-coloured dye. The pump was illuminated from behind so that he could observe the "pressure patterns" in the fluid flow near the jets. These patterns were then photographed for different values of backing pressure with a heater power of 220 watts. Hickman did not record the fine pressure during this experiment but he offers qualitative values for the pressure, based on the photographs. Hickman estimates that there was negligible increase in the fine pressure as long as the top (Embree) jet continued to operate properly. Accordingly, he believes that the backing pressure rose to about 100 μ Hg. before there was any change in fine pressure.

If Hickman's estimates are at all accurate, then it would appear that his glass-wall pump behaved more satisfactorily than the all-metal pump studied here. As can be seen from fig. 19, the metal pump will have completely stopped working before the backing pressure reaches 100 μ Hg. Furthermore, it can be seen that the fine pressure indeed varied with backing pressure for backing pressures as low as 1/10 μ Hg.

While the values of fine pressure estimated by Hickman from his photographs may be somewhat optimistic, it is also possible that the behaviour of the metal pump was inferior to the commercial pump either due to a detrimental effect of the metal pump wall, or to faulty jet design or construction.

Hickman(2) has pointed out several times the lower wall temperatures obtained with a metal construction can have a serious effect on pump operation. He has also observed that in pumps having wide jet clearances, the speed is not independent of backing pressure. This suggests that the jets used here may not have had the proper dimensions as a result of poor design.

Table I shows that although the liquid-nitrogen reservoir used with the glass-metal trap was similar in construction to the all-glass trap, the addition of a glass "finger" extending into the metal tee resulted in greatly reduced storage properties. The trapping "finger" had been silvered on the inner surface to reduce radiation effects. It appears therefore, that silvering both surfaces of the finger and silver plating the inner surface of the tee are necessary if the trap is to "maintain itself" for 2 or 3 days.

The ultimate pressure obtained with the untrapped system was 10^{-6}mm. Hg. which compares favourably with the design estimates based on the commercial prototype. The addition of a refrigerated trap, however, reduced the ultimate pressure only by a factor of five regardless of which trap was actually used. This implies the existence of a "source" of gas. This may have been evolution of gas (outgassing) from the walls of the pump or metal tee, or contaminating vapours of solder, vacuum grease, etc., or due to a very small leak - although all the leaks which could be found were sealed with Glyptol*.

Metals, due to the rough nature of their surfaces, will usually absorb extremely large quantities of gas on exposure to the atmosphere. The metal walls of the pump might therefore be expected to outgas for a long period of time after the system is first evacuated. Although the system was permitted to run for several weeks to allow outgassing effects to diminish, it is possible that evolution of gas from the pump walls was one of the factors which limited the ultimate pressure attained.

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*General Electric G1276 cement.

Summary and Conclusions

In the course of this project, a thorough analysis has been made of a high-speed metal diffusion pump which had been assembled at this Laboratory. The operation of this pump was found to compare favourably with the design estimates based on the behaviour of its commercial prototype. Under optimum conditions, the pump exhibited an intrinsic speed of 200 to 250 litres$\cdot$ sec^{-1}. The area of the mouth of the pump was 76 cm^2 so that the Ho coefficient for this pump was about 25%. Higher pumping speeds were obtained by increasing the heat supplied to the boiler but the pumping fluid was readily cracked under the influence of these elevated temperatures.

The ultimate pressure of 10^{-6} mm. Hg. obtained without the use of a cold trap is in agreement with the figures commonly quoted in the literature for ultimate pressures obtained with non-fractionating pumps employing Octoil-S as the working fluid.

The behaviour of the pump showed a strong dependence on the pressure which was maintained at the exhaust end; however, since a 2-stage diffusion pump was used as a "booster" between the metal pump and the mechanical rotary pump, this dependence did not present a serious practical problem.

A Pirani gauge was constructed to measure pressures in the range 1 to 100 μ Hg. The unit was temperature compensated and had a "zero-drift" equivalent to $\pm$ 1/4 μ in one hour.

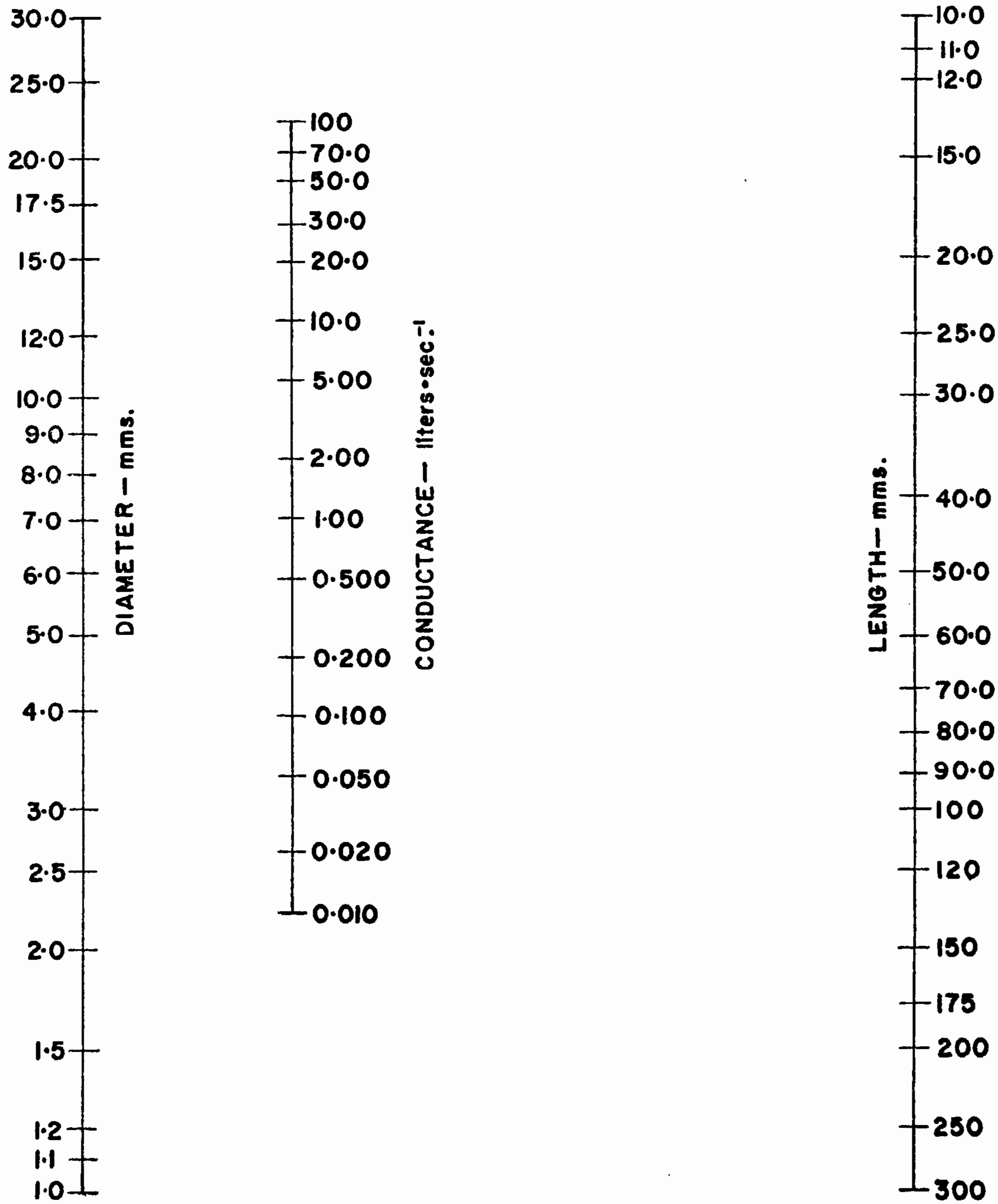
Three refrigerated vapour traps and three mechanical oil baffles were also constructed as part of the vacuum system. In designing the traps it was found that a compromise was necessary between high conductance and efficient trapping. This was due to the fact that the conductance was increased by reducing the area of the trapping surface rather than by increasing the outer trap-wall dimensions. In each case, the ultimate pressure with a refrigerated trap was 2×10^{-7} mm. Hg. This indicates that the ultimate pressure was probably due to the vapour pressure of "O"-ring grease or other contaminant present in the vacuum chamber.

For the baffles tested, it was verified that the greatest conductance occurs when only two rings having equal areas are used, although the actual conductances of the baffles were found to be somewhat greater than calculated, due to the finite spacing between the rings.

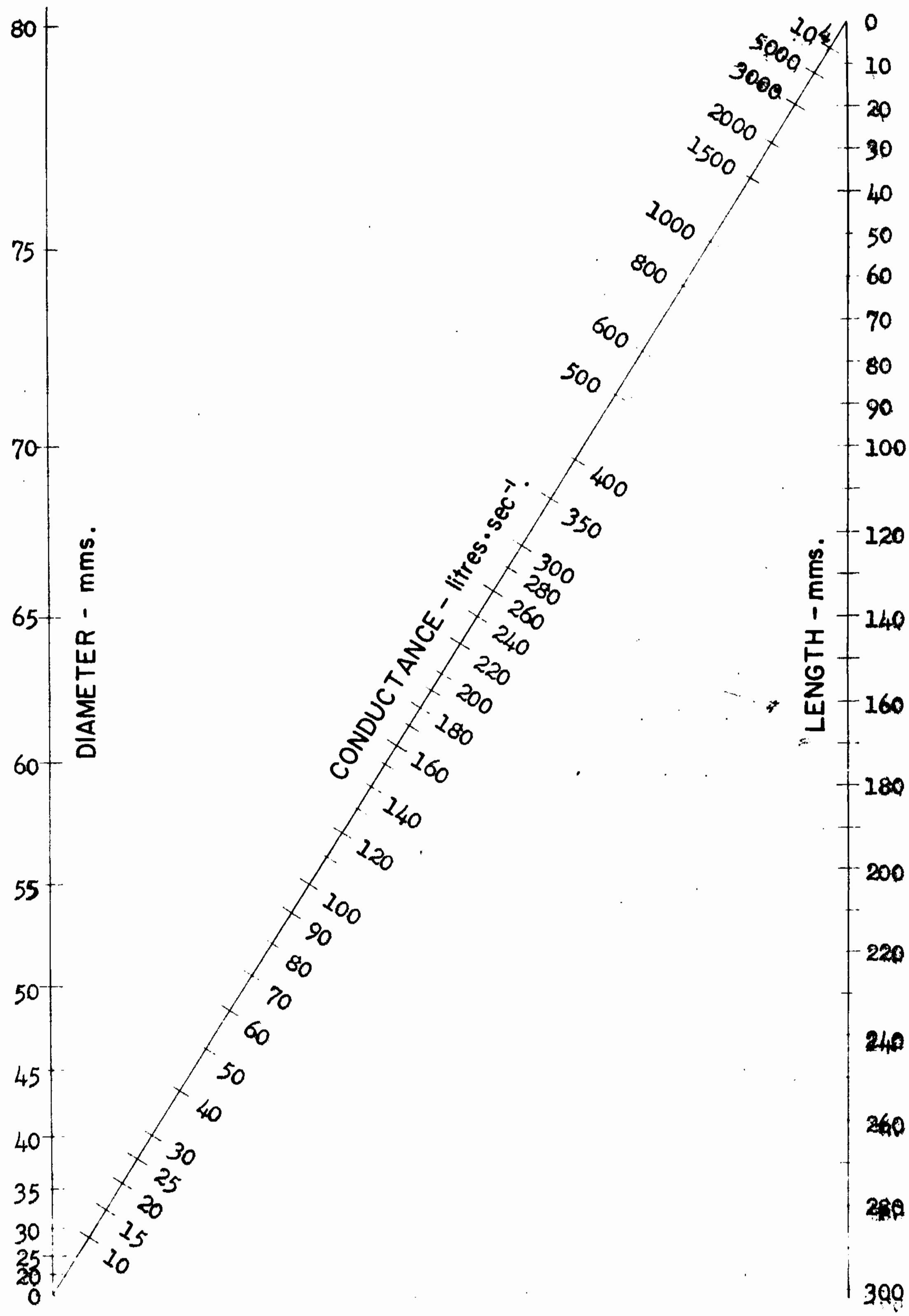
APPENDIX

In this section, several charts are shown giving the conductances of certain geometrical shapes under conditions of free-molecule flow. The conductances have been calculated from the theoretical relations given by Dushman[1] and are valid for dry air at 25°C.

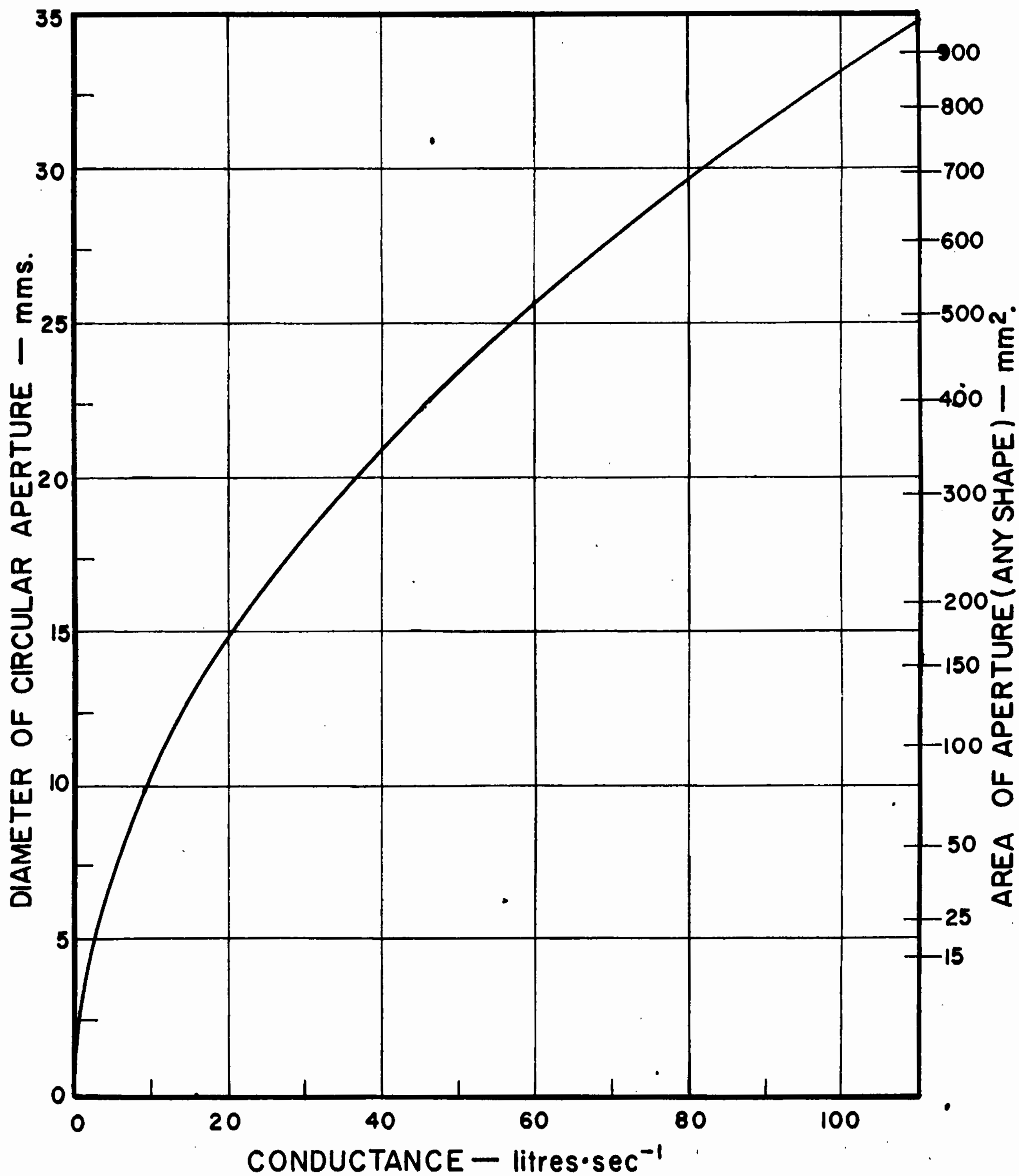
CONDUCTANCE OF CYLINDRICAL TUBING
HAVING DIAMETER LESS THAN 30 mm.



CONDUCTANCE OF CYLINDRICAL TUBING HAVING DIAMETER LESS THAN 80 mm.



THEORETICAL CONDUCTANCE OF AN APERTURE



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