A CRITICAL ANALYSIS OF
THERMAL TRANSPIRATION PUMPING

By

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ABSTRACT

At high Knudsen numbers, flow of a gas may take place as a result of a temperature gradient alone. This phenomenon can form the basis of a pumping device. Operations of the device as a circulation pump and as a vacuum pump are considered. Pumping speed characteristics for single-stage and multi-stage pumps are obtained in terms of dimensionless parameters which eliminate several variables but adequately express the desired quantities - pumping speed and inlet pressure. Information on other vacuum pumping methods is compiled and transformed into dimensionless parameters for comparison with the thermal transpiration pump characteristics. Comparison is done in terms of pumping speed characteristics as well as other features. A preliminary design with idealized conditions is carried out. Difficulties of ideal-operation are pointed out and ways to overcome some are suggested. Certain construction features are discussed. It appears that the thermal transpiration pump is best suited as a circulation or displacement pump, though a 6-stage design operating as a vacuum pump appears feasible and worthy of development.

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LIST OF SYMBOLS

Numbers in brackets refer to page numbers where the symbol is first introduced, or where further explanation or definitions are available.

A	transpiration area per passage (8)
	transpiration area per stage (11)
a	defined quantity (19)
	defined quantity (53)
"a"	defined quantity (48)
b	defined quantity (19)
C	conductance of a passage (31)
D	characteristic dimension of reservoir (39)
d	characteristic dimension of orifice or tube (9)
g	temperature jump distance (52)
K	defined quantity (21)
K_1	defined quantity (21)
K_2	defined quantity (21)
K_n	Knudsen number (2)
L	characteristic dimension (2)
	length of transpiration element (45)
M	molecular weight of gas (D-1)
$\mathring{m}$	mass flow rate (7)
N	total number of stages (17)

n	defined quantity (18)
	defined quantity (38)
	defined quantity (52)
p	pressure (8)
p_{in}	inlet pressure (7)
p_n	pressure in the nth reservoir (18)
p_o	forepressure (14)
p_u	ultimate pressure (19)
p^*	pressure ratio (25)
Q	throughput (7)
R	characteristic gas constant (8)
S_p	pumping speed (7)
S_p^*	defined quantity (25)
T	temperature (8)
T'	modified temperature (53)
T_K	defined quantity (52)
T_r	temperature ratio (25)
T_{re}	defined quantity (54)
T_w	wall temperature (52)
v	specific volume (D-1)
x	number of passages per stage (11)
	degree of ionization (D-1)
y	defined quantity (56)
α	defined quantity (23)
	degree of dissiciation (D-2)
	defined quantity (55)

η	viscosity (38)
θ	time (B-1)
θ_d	characteristic dissociation temperature (D-2)
λ	molecular mean free path (2)
ρ	density (1)

Subscripts

c	cold
h	hot
n	number of stage
N	total number of stages

LIST OF FIGURES

CHAPTER I

INTRODUCTION

1.1 Thermal Transpiration(1,2,3,4,5)

Ordinarily when two bodies of gas, maintained at different thermodynamic conditions, are connected by an orifice or a tube (Fig.1.1), a net gas flow takes place from the high pressure body to the low pressure body until pressures are equalised on the two sides and the flow ceases. Thus finally,

$$p_1 = p_2 \quad \text{or,} \quad p_1/p_2 = 1 \quad \text{and} \quad \rho_1/\rho_2 = T_2/T_1 \tag{1.1}$$

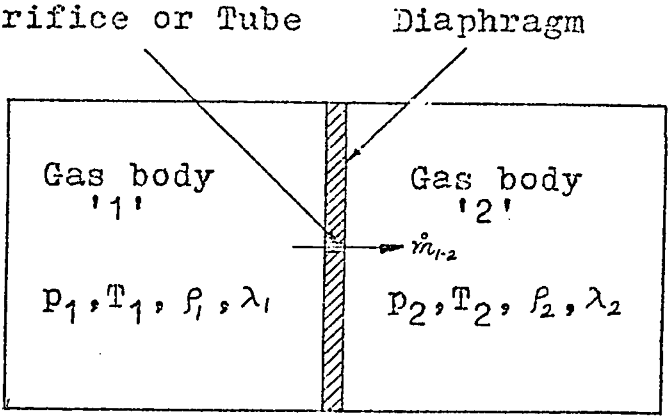

Fig.1.1 Principle of thermal transpiration.

The mean free path of gas molecules at ordinary pressures and temperatures is extremely small (on the order of 10^{-6} in. for air) compared to the characteristic dimension of the orifice or the tube. This allows for the existence of a transitional region near the orifice (or tube), many mean free paths long, so that enough intermolecular collisions take place to produce thermodynamic equilibrium locally and consequently a smooth temperature and density transition from one gas body to the other.

At very low pressures, however, mean-free-paths are much longer and such things as temperature transition can be quite abrupt in terms of molecular collision distances. Indeed, it is possible that molecules may not collide with each other during their entire journey from one end of the passage (orifice or tube) to the other. Under such conditions kinetic theory of gases predicts that flow will cease when[#],

$$p_1/p_2 = (T_1/T_2)^{\frac{1}{2}} \tag{1.2}$$

Thus, when two bodies of a gas are connected by an orifice or a tube whose dimensions are small compared to the molecular mean free path on either side, and in which there exists a temperature gradient between the reservoirs, a net transfer of molecules occurs from the 'cold' side to the 'hot' side building up a pressure difference in accordance with equation (1.2). This phenomenon, resulting in gas flow due to a temperature gradient alone, is called thermal transpiration.

The flow behavior of a gas is often, therefore, highly dependent on its degree of rarefaction. The basic parameter that indicates the degree of rarefaction of a gas is the Knudsen number, K_n, defined as $K_n = \lambda/L$, where λ is the molecular mean free path, and L is the characteristic body dimension of the containing vessel. According to the degree of rarefaction as measured by the Knudsen number, gas dynamic flow regimes may be divided roughly as follows(6),

Continuum flow	$K_n \leq 0.01$
Slip flow	$0.01 < K_n < 0.1$
Transition flow	$0.1 < K_n < 5$
Free-molecule flow	$K_n \geq 5$

Whereas continuum and slip flows may be described by the familiar Navier-Stokes equations of motion, which are derived from continuum mechanics with appropriate boundary conditions; the analysis of transition and free-molecule flows is based on the kinetic theory of gases. In terms of molecular behavior, flow is continuum when intermolecular collisions vastly outnumber molecule-surface collisions. In the other extreme, when the flow is highly rarefied, molecular motion is independent of the presence of other particles and flow changes occur entirely as a result of molecular collisions with bounding surfaces.

1.2 Thermal transpiration pumping(7,8,9,10)

It is obvious that thermal transpiration can form the basis of a pumping device. Also, as this is essentially a low pressure phenomenon, the device would be appropriate as a vacuum pump. The pressure ratio attainable in a single stage, however, is severely restricted due to the limit imposed by realizable temperature ratios. In practice, one side of a stage will probably be at room temperature (say

300°K) and the other heated to some easily obtainable temperature (say 900°K). In that case, the attainable pressure ratio as given by equation (1.2) is,

$$p_2/p_1 = (900/300)^{\frac{1}{2}} = 1.732$$

An alternative approach might be to substantially cool the low-pressure side to near absolute zero by some means, in which case, pressure ratios much larger than the above could be obtained. Nevertheless, such pressure ratios, in themselves, are not too encouraging for the purpose of vacuum pumping - vacuum pumps are often required to achieve pressure ratios of several orders of magnitude. Large pressure ratios, however, are possible with multistaging. The thermal transpiration effect could be cascaded through several stages of the pump. Operating with conditions given in the previous example, a 20 - stage pump would theoretically be capable of producing a pressure ratio of 4 orders of magnitude, which is comparable with those of other vacuum pumping methods. On the basis of these facts it appears desirable to investigate the characteristics of a multi-stage thermal transpiration pump in detail. Subsequent chapters attempt such an analysis.

1.3 State - of - art survey

Historically, the existence of the thermal transpiration phenomenon has been known for almost a century. According to Knudsen$^{(4)}$, it was discovered and first reported in 1879 by O. Reynolds and J.C. Maxwell. Loeb$^{(2)}$ credits Reynolds also for christening it as 'thermal transpiration'.

Pioneering work, especially of an experimental nature, was done by Knudsen between 1907 and 1914. In one of his oft-quoted experiments(11), using an artificially porous body, he was able to achieve $p_2/p_1 = 1.320$. Later, he tried to compound the effect by multistaging. In a 10-stage system a pressure ratio of 10 was obtained(2). This and subsequent attempts at theoretical formulation of the phenomenon applicable to all regions of flow have met with limited success only.

It appears that application of thermal transpiration to practical vacuum pumping was first attempted by Gaede in 1941-42 (cited in ref. (7)). Unlike Knudsen's experiment, Gaede used a wire lattice to provide transpiration elements and achieved somewhat better results. However, he seems to have abandoned further investigation after establishing the operating practicability. Unfortunately, his work is unpublished and has only been briefly discussed in reference 7. A more recent attempt has been made by Schumacher et. al.(8). No theoretical treatment of the parameters involved in their suggested device is offered, though an intuitive explaination of the operation is given. On the experimental side these researchers managed to achieve, within a reasonable pump size, a pumping speed of 2 l/s for helium with forepressures of about 7×10^{-4} torr. Hopfinger and Altman(9) utilized porous ceramics as transpiration elements. They found agreement between their theoretical predictions, made employing the dusty-gas theory of Mason, Evans and Watson(15), and experimental results, thus indicating the possibility of using porous media for gas pumping.

Even though pumps based on thermal transpiration have been constructed and tested, a theoretical investigation of their possible capabilities, practical limitations and comparison with existing pumps is lacking. It is the purpose of this thesis to fill that need.

CHAPTER II

ANALYSIS OF THERMAL TRANSPIRATION PUMPING

2.1 Equations governing gas flow

The operating characteristics of a vacuum pump are generally indicated by its pumping speed* versus inlet pressure curve(16-20). The pumping speed is given by the relation (Fig. 2.1)

$$S_p = Q/p_{in} \tag{2.1}$$

where,

S_p = pumping speed of the system (litres/sec)
Q = throughput of the system (torr-litres/sec)
p_{in} = pressure of gas entering the system (torr)

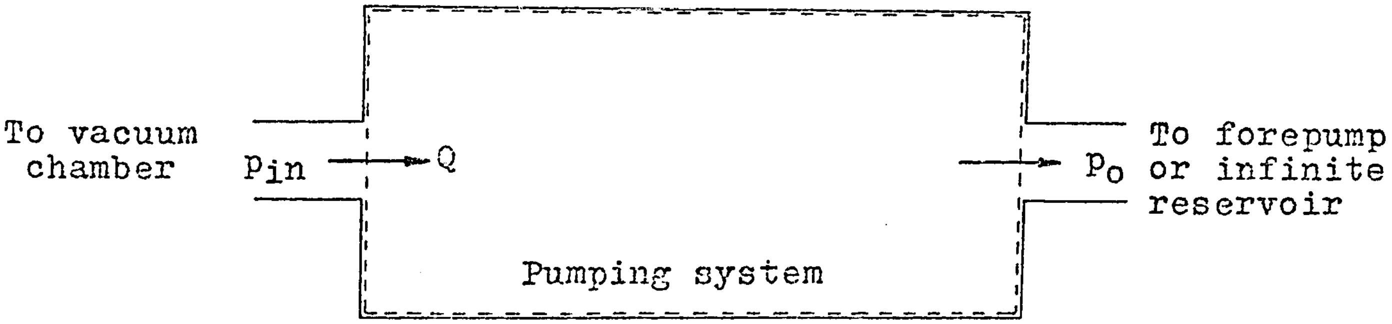

Fig. 2.1 Pumping system identification

The throughput Q is also known as the p-V value of the mass rate of flow into the system. Application of the perfect gas law gives#,

$$Q = \mathring{m}RT \tag{2.2}$$

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* For glossary of vacuum pump terminology, refer to Appendix A.
\# Applicability of the perfect gas law is demonstrated in

where, $\overset{\circ}{m}$ = mass rate of flow (gms/sec)

R = characteristic gas constant (torr-litres/g-mole-^{o}K)

T = absolute temperature of gas (^{o}K)

2.2 Free-molecule flow across an orifice

It can be shown from kinetic theory considerations (1,2,3,5) that for a body of gas contained at pressure p and temperature T, the rate of escape of gas through a small aperture of area A (known as a maxwellian effusive stream) is given by,

$$\overset{\circ}{m} = Ap/(2\pi RT)^{\frac{1}{2}} \qquad (2.3)$$

If the gas on both side of the aperture has molecular mean free paths very large compared to the aperture size, the flow is free molecular and effusion will occur in both directions independently. In addition, if the thermodynamic conditions of the gas on the two sides are different (Fig. 1.1) a net flow of gas will occur according to the relation,

$$\overset{\circ}{m}_{1-2} = A\left[\frac{p_1}{\sqrt{2\pi R T_1}} - \frac{p_2}{\sqrt{2\pi R T_2}}\right]$$

$$= \frac{A}{\sqrt{2\pi R}}\left[\frac{p_1}{\sqrt{T_1}} - \frac{p_2}{\sqrt{T_2}}\right] \qquad (2.4)$$

Implicit in the use of this relation (2.4) are the following assumptions :-

(i) the diameter of the aperture is much smaller than the mean free path of the molecules in either container in order that free-molecule flow exists in the aperture.

(ii) the containers are large enough so that,

  (a) continuum conditions exist in the containers, and

  (b) equilibrium is continuously maintained in both the containers despite loss or gain of molecules as a result of transpiration.

(iii) the gas obeys the perfect gas law.

(iv) the diaphragm containing the aperture is infinitely thin so that $L/d \rightarrow 0$, where L is the 'length' of the aperture.

From equation (2.4) it is apparent that the differential flow will continue until the gas densities on the two sides become so adjusted that the quantity $p/T^{\frac{1}{2}}$ has the same value on either side, that is, until

$$p_1/p_2 = (T_1/T_2)^{\frac{1}{2}} = \rho_2/\rho_1 \qquad (2.5)$$

Another observation that can be made from equation (2.4) is that a net flow of gas will occur even if the pressures are equal in the two containers. A flow then occurs due to the thermal gradient alone. The gas will flow from container '1' to container '2' if $T_1 < T_2$, that is, thermal transpiration flow takes place from the 'cold' to the 'hot' reservoir. A pressure gradient may be superimposed either to reinforce this flow ($p_1 > p_2$) or to retard it ($p_1 < p_2$).

2.3 Pumping speed - single stage pump

It follows from the last section that the system shown in figure 1.1 can be used as a pumping device - either to displace gas at a certain pressure (circulation or displacement pump) or to reduce the pressure of gas inside a fixed volume (vacuum pump). These applications are separately discussed below.

2.3.1 As a circulation pump

If the system is connected between two reservoirs A and B, as shown in figure 2.2, maintained at the same pressure, it will continuously displace the gas from reservoir A to reservoir B.

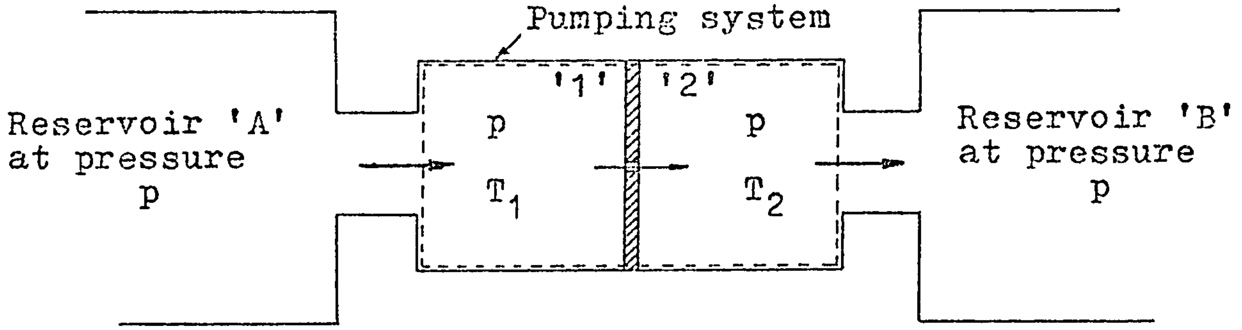


Fig. 2.2 Thermal transpiration circulation pump

Equation (2.4) can be modified for this case to,

$$\mathring{m}_{1\text{-}2} = \frac{A\,p}{\sqrt{2\pi R}}\left[\frac{1}{\sqrt{T_1}} - \frac{1}{\sqrt{T_2}}\right] \tag{2.6}$$

From equations (2.1), (2.2) and (2.6),

$$S_p = \frac{A\,p}{\sqrt{2\pi R}}\left[\frac{1}{\sqrt{T_1}} - \frac{1}{\sqrt{T_2}}\right]\frac{RT}{p_{in}}$$

Recalling from definitions of S_p and Q, in this case

$$T = T_1 \text{ , and } p_{in} = p$$

therefore,

$$S_p = A\sqrt{\frac{RT_1}{2\pi}}\left[1-\sqrt{\frac{T_1}{T_2}}\right] \qquad (2.7)$$

If there are x apertures in the same disphragm and assuming that each of them works independently, the mass rate of flow will be x times that given by equation (2.6). Accordingly, the pumping speed, as given by equation (2.7), will be x times as great, i.e.

$$S_p = xA\sqrt{\frac{RT_1}{2\pi}}\left[1-\sqrt{\frac{T_1}{T_2}}\right]$$

or, replacing xA by $\mathscr{A}$,

$$S_p = \mathscr{A}\sqrt{\frac{RT_1}{2\pi}}\left[1-\sqrt{\frac{T_1}{T_2}}\right] \qquad (2.8)$$

Equation (2.8) predicts the pumping speed of a one-stage thermal transpiration circulation pump. For a fixed geometry ($\mathscr{A}$ = const.) and fixed temperatures T_1 and T_2, the pumping speed of a given gas (R = const.) is constant. Incidentally, this pumping speed is the maximum obtainable with a temperature gradient alone (see Fig. 2.5). It is also seen to be independent of operating pressure, provided of course, that the geometry is such as to ensure the assumptions of section 2 at the operating pressure.

In order to represent equation (2.8) graphically, it is convenient to transform the equation into a non-dimensional form. In Appendix B, appropriate dimensionless quantities involving the parameters in the equation are derived as,

$$S_p^* = S_p/(\mathcal{A}(RT_1)^{\frac{1}{2}})$$

$$T_r = T_2/T_1$$

so that, equation (2.8) may now be written as,

$$S_p^* = \frac{1}{\sqrt{2\pi}}\left[1 - \frac{1}{T_r}\right]$$

Figure (2.3) shows the variation of S_p^* with temperature ratio T_r. It may be observed that the curve is steepest in the region $T_2/T_1 \simeq 1$ and it gradually flattens as T_2/T_1 increases. Thus, the relative gain in pumping speed is greatest when the temperature ratio is greater than but close to unity.

In order to obtain an appreciation of the magnitude of quantities involved, if the gas is assumed to be air (R = 2152 torr-cm^3/mole-^{o}K) at nearly atmospheric temperature (T_1 = 300^{o}K), and the area of transpiration $\mathcal{A}$ = 25 cm^2, then, at T_r = 10, the pumping speed S_p = 5460 cm^3/sec. It may be noted that nearly half of this pumping speed is realizable even at T_r = 2.5.

Further discussion of pump characteristics is deferred until chapter IV, when the more general vacuum pump is discussed.

2.3.2 As a vacuum pump

If the thermal transpiration system is to be employed for evacuating a chamber, such as shown in figure 2.4, the discharge pressure from the system may be assumed constant

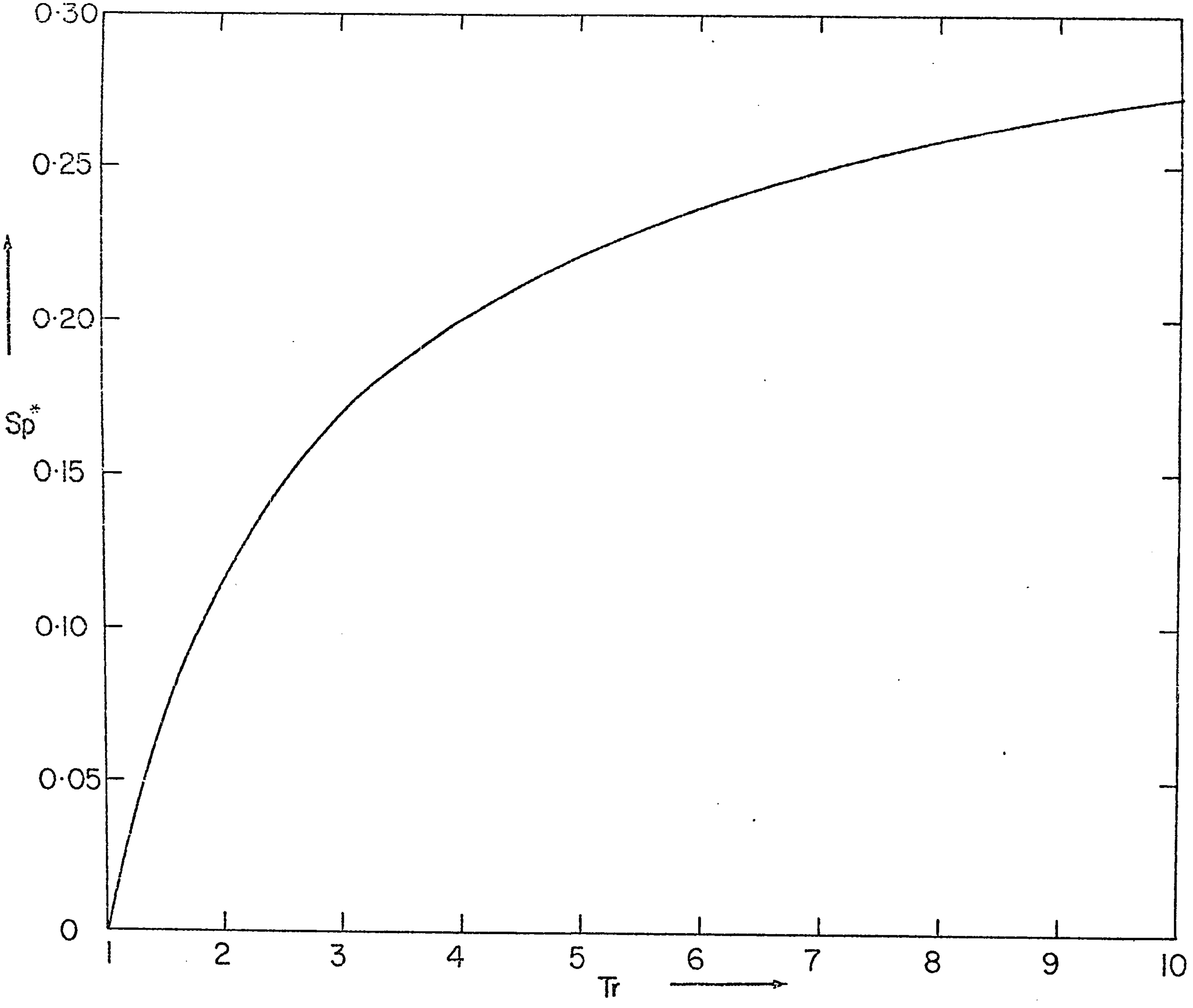


FIG. 2.3 EFFECT OF TEMPERATURE RATIO (Tr) ON PUMPING SPEED (Sp*)

(being provided by a forepump, an infinite reservoir etc.). The pressure in the vacuum chamber, on the other hand, will continuously decrease as long as pumping occurs.

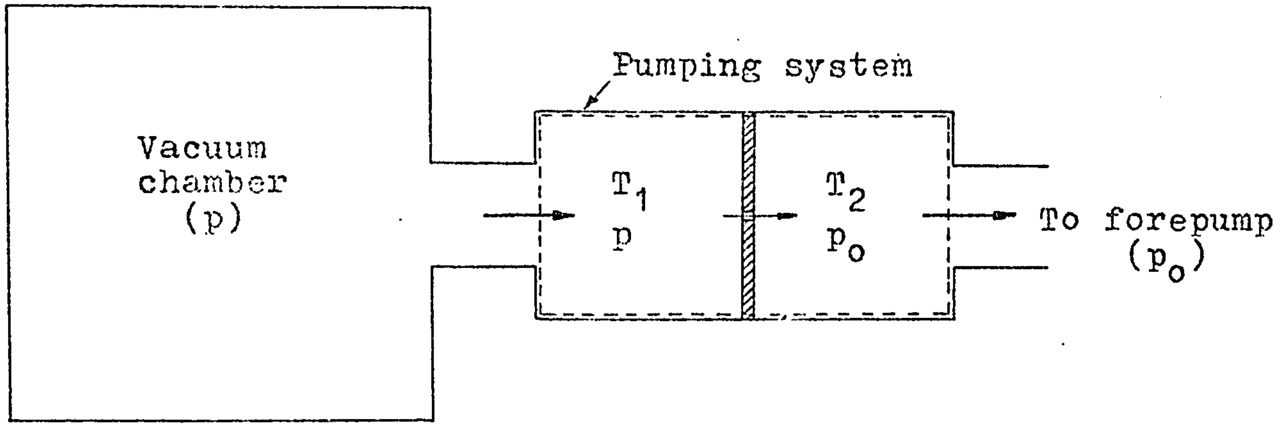


Fig. 2.4 Single-stage thermal trans. vacuum pump

Let,

p_o = constant forepressure

p = pressure in the vacuum chamber at any instant of operation

Initially, $p = p_o$. Pumping occurs until,

$p/p_o = (T_1/T_2)^{\frac{1}{2}}$ or $p = p_o(T_1/T_2)^{\frac{1}{2}}$

From equations (2.4), (2.2) and (2.1),

$$S_p = \frac{A}{\sqrt{2\pi R}}\left[\frac{p}{\sqrt{T_1}} - \frac{p_o}{\sqrt{T_2}}\right]\frac{RT_1}{p} \tag{2.9}$$

As before, for x independently operating apertures,

$$S_p = \mathcal{A}\sqrt{\frac{RT_1}{2\pi}}\left[1 - \frac{p_o}{p}\sqrt{\frac{T_1}{T_2}}\right] \tag{2.10}$$

where,

$p_o \geq p \geq p_o(T_1/T_2)^{\frac{1}{2}}$

Equation (2.10) gives the pumping speed of a single-stage thermal transpiration vacuum pump. As may be seen from this equation, the maximum pumping speed occurs when $p = p_o$, that is, when the pump just starts operating. The pumping speed keeps falling as p decreases, until it finally approaches zero when $p = p_o(T_1/T_2)^{\frac{1}{2}}$.

As before, the pumping speed equation (2.10) may be non-dimensionalized in order to obtain its graphical representation. From Appendix B, the appropriate dimensionless groups are,

$$S_p^* = S_p / \left(\mathcal{A}(RT_1)^{\frac{1}{2}}\right)$$

$$T_r = T_2/T_1$$

$$p^* = p/p_o$$

so that, equation (2.10) may be written as,

$$S_p^* = \frac{1}{\sqrt{2\pi}}\left[1 - p^{*-1}T_r^{-\frac{1}{2}}\right]$$

Pumping speed curves, in terms of non-dimensionalized parameters, are shown in figure 2.5. It may be observed that the circulation pump is just a special application of this pump, that is, operation is at the limiting condition that pressures on either side of the pump are the same.

The pumping speed characteristics of a single-stage pump are similar in nature to those of multi-stage pumps to be discussed in the next section. It appears appropriate, therefore, to postpone the discussion until chapter IV, when the characteristics of multi-stage pumps are discussed.

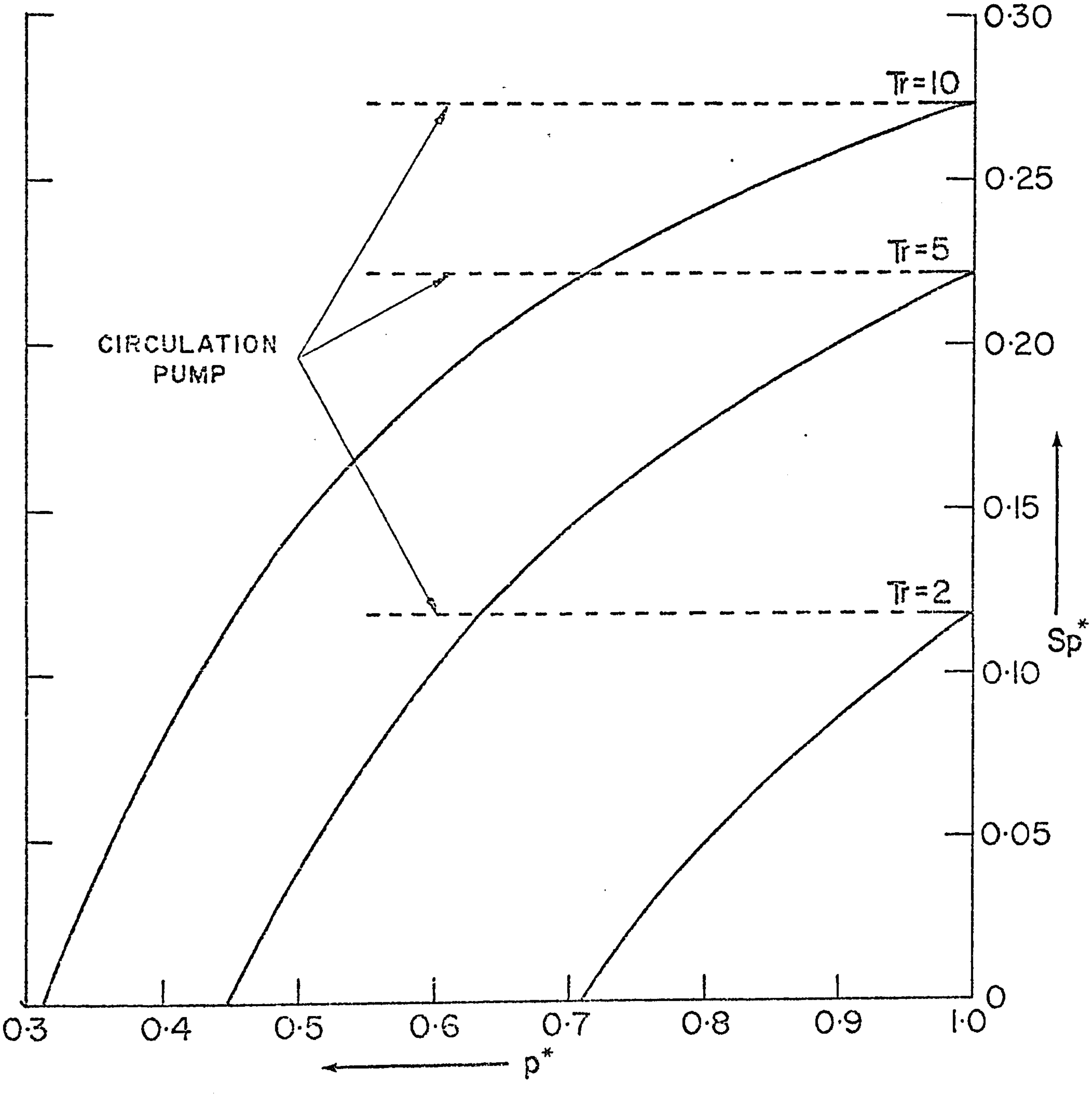


FIG. 2.5 PUMPING SPEED CHARACTERISTICS OF SINGLE-STAGE THERMAL TRANS. VACUUM PUMP

2.4 Multi-stage vacuum pump

It is seen from equation (2.5) that, under the assumptions made therein, the ultimate pressure ratio is entirely a function of the temperature ratio used. In a single-stage pump a temperature ratio of 4 can produce, at the most, a pressure ratio of 2. Vacuum pumps are generally expected to reduce system pressures by several orders of magnitude. Clearly a single-stage thermal transpiration vacuum pump will not be of much use. However, by joining several appropriate single-stage pumps in series, the pumping action can be cascaded to produce sufficiently large pressure ratios. A multi-stage pump having N stages will yield,

$$(p_1/p_2)_N = (T_1/T_2)_1^{\frac{1}{2}} \text{ x } (T_1/T_2)_2^{\frac{1}{2}} \text{ x } \ldots\ldots(T_1/T_2)_N^{\frac{1}{2}}$$

or

$$(p_1/p_2)_N = (T_1/T_2)^{N/2} \qquad (2.11)$$

As an example, for N = 20 and for a moderate temperature ratio of 3,

$$(p_1/p_2)_{20} = (3)^{20/2} \simeq 6 \text{ x } 10^4 \text{ !}$$

2.5 Pumping speed - multi-stage pump

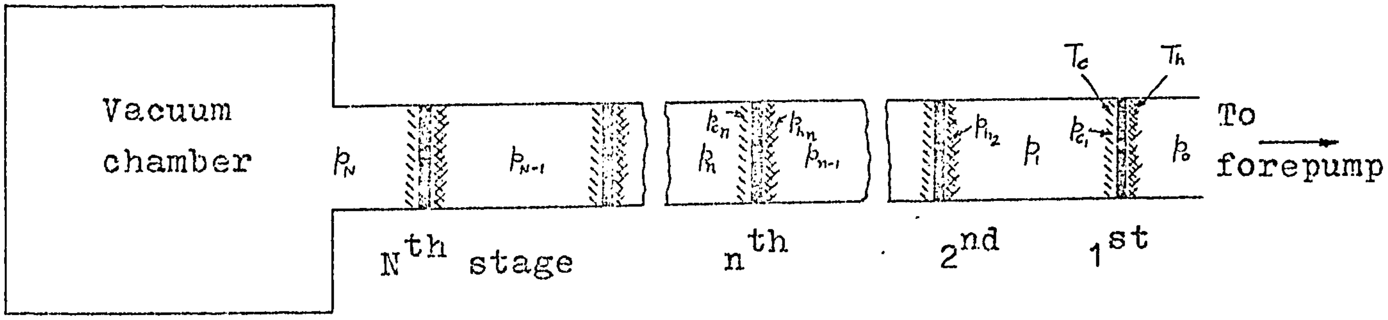


Fig. 2.6 Multi-stage thermal transpiration pump - schematic diagram and terminology

A schematic diagram of an N-stage T.T. pump is shown in figure 2.6. In any given stage, the temperatures in the immediate vicinity of its aperture are T_c and T_h as shown. If allowed to act independently, the stage would have produced pressures p_{c_n} and p_{h_n} respectively on the sides having temperatures T_c and T_h. However, due to the continuum conditions prevailing in the reservoirs, the cold-side pressure of that stage (p_{c_n}) and the hot side pressure of the next stage ($p_{h_{n+1}}$) will immediately equalize, producing a common pressure p_n. For example, p_{c_1} and p_{h_2} (Fig. 2.6) will equalize to give a common pressure p_1. The system shown in figure 2.6 can therefore be represented as in figure 2.7.

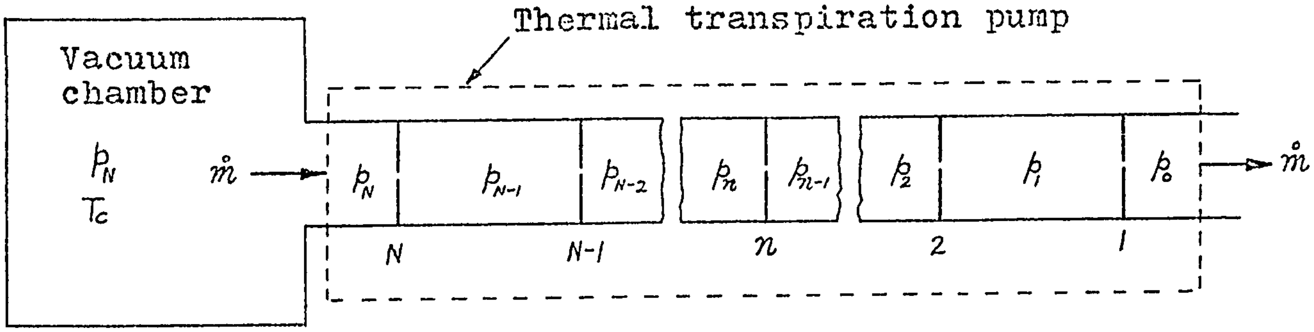


Fig. 2.7 Simplified schematic diagram of multi-stage thermal transpiration pump

In order to predict the pumping speed ~ inlet pressure characteristics of this multi-stage pump, it must be noted that initially (at the start of pump operation),

$$p_N = p_{N-1} = \ldots\ldots\ldots\ldots\ldots p_n = \ldots\ldots p_2 = p_1 = p_0$$

And finally, when transpiration is completed in all stages, in accordance with equation (2.11),

$$p_u/p_o = (T_c/T_h)^{N/2} \tag{2.12}$$

where p_u = ultimate pressure in the N^{th} stage or in the vacuum chamber.

Thus, the instantaneous pressure in the vacuum chamber, p_N varies from an initial value p_o to a final value p_u , as given by equation (2.12).

At any instant of operation, the mass rate of flow of gas in the n^{th} stage is given, as follows from euqation (2.4), by

$$\dot{m}_n = \frac{\mathscr{A}_n}{\sqrt{2\pi R}}\left[\frac{p_n}{\sqrt{T_c}} - \frac{p_{n-1}}{\sqrt{T_h}}\right] \tag{2.13}$$

where $\mathscr{A}_n$ = total x-sectional area of X_n transpiration elements (apertures) each of x-sectional area A_n,

$= X_n A_n$

In order that there be no accumulation or depletion of gas mass within the pumping system, the mass rate of flow in all stages of the system at a given instant of operation should be the same. That is,

$$m_N = m_{N-1} = \ldots\ldots\ldots m_n = \ldots\ldots\ldots m_2 = m_1 = m \tag{2.14}$$

From equations (2.13) and (2.14), and introducing $a = 1/T_c^{\frac{1}{2}}$, $b = 1/T_h^{\frac{1}{2}}$, we can write,

$$\mathscr{A}_N(ap_N - bp_{N-1}) = \mathscr{A}_{N-1}(ap_{N-1} - bp_{N-2}) = \ldots\mathscr{A}_n(ap_n - bbp_{n-1})$$
$$\ldots.\mathscr{A}_2(ap_2 - bp_1) = \mathscr{A}_1(ap_1 - bp_o) \tag{2.15}$$

If $\mathring{m}$ be evaluated from equation (2.15), then, using equations (2.1) and (2.2) the pumping speed S_p can be found.

In order to solve equations (2.15), the number of independent variables must be reduced to (N - 1), since only (N - 1) independent equations are available from equation (2.15).

In any given situation T_c, T_h and therefore a, b will be fixed. p_o will be fixed by an infinite reservoir or a forepump, and in accordance with equation (2.12), the value of p_N will vary only from p_o to p_u. Transpiration areas $\mathcal{A}_1$, $\mathcal{A}_2$, ...$\mathcal{A}_N$, must also be fixed so that the only independent variables left are p_1, p_2, ...p_{N-2}, p_{N-1}, that are (N-1) in number.

One solution of equations (2.15) that may be of practical interest is for the special case of transpiration areas all equal, that is,

$$\mathcal{A}_1 = \mathcal{A}_2 = \ldots\ldots\ldots \mathcal{A}_n = \ldots\ldots\ldots \mathcal{A}_{N-1} = \mathcal{A}_N \qquad (2.16)$$

In this case, equations (2.15) reduce to,

$$(ap_N - bp_{N-1}) = (ap_{N-1} - bp_{N-2}) = \ldots\ldots\ldots$$
$$\ldots(ap_n - bp_{n-1}) = \ldots\ldots(ap_2 - bp_1) = (ap_1 - bp_o) \qquad (2.17)$$

Equations (2.17) represent a system of (N-1) independent equations and, with the assumptions made above, contain only (N-1) variables in the form of pressures p_1, p_2, ...p_{N-2}, p_{N-1}. Further, since all equations are linear, a direct solution is possible as follows:

Let,

$$(ap_N - bp_{N-1}) = (ap_{N-1} - bp_{N-2}) = \dots\dots\dots$$
$$\dots = (ap_n - bp_{n-1}) = \dots\dots\dots (ap_2 - bp_1) = (ap_1 - bp_o) = K$$

then, $ap_1 - bp_o = K$

or, $p_1 = K/a + bp_o/a$

$$= K_1 + K_2 p_o \qquad (2.18)$$

where, $K_1 = K/a$, and $K_2 = b/a$

also, $ap_2 - bp_1 = K$

or, $p_2 = K/a + bp_1/a$

$$= K_1 + K_2 p_1 \qquad (2.19)$$

From equations (2.18) and (2.19), a generalized expression for p may be written as

$$p_{i+1} = K_2 p_i + K_1 \qquad (2.20)$$

likewise, $p_{i+2} = K_2 p_{i+1} + K_1$

substituting from equation (2.20),

$$p_{i+2} = K_2(K_2 p_i + K_1) + K_1$$
$$= K_2{}^2 p_i + K_2 K_1 + K_1 \qquad (2.21)$$

also, $p_{i+3} = K_2 p_{i+2} + K_1$

substituting from equation (2.21),

$$p_{i+3} = K_2(K_2{}^2 p_i + K_2 K_1 + K_1) + K_1$$
$$= K_2{}^3 p_i + K_2{}^2 K_1 + K_2 K_1 + K_1 \qquad (2.22)$$

From equations (2.20), (2.21) and (2.22), a generalization can be obtained,

$$p_{i+N} = K_2^N p_i + K_2^{N-1} K_1 + K_2^{N-2} K_1 + \dots\dots\dots$$

$$\dots\dots + K_2 K_1 + K_1$$

$$= K_2^N p_i + K_1 (K_2^{N-1} + K_2^{N-2} + \dots\dots\dots + K_2 + 1) \tag{2.23}$$

For $i = 0$, equation (2.23) can be written as,

$$p_N = K_2^N p_o + K_1 (K_2^{N-1} + K_2^{N-2} + \dots\dots\dots + K_2 + 1)$$

or, after substituting for K_2 and K_1,

$$p_N = \left(\frac{b}{a}\right)^N + \frac{K}{a} \sum_{i=0}^{N-1} \left(\frac{b}{a}\right)^i$$

which, on transposing and rearranging the terms, gives,

$$K = \frac{a\left[p_N - \left(\frac{b}{a}\right)^N \cdot p_o\right]}{\sum_{i=0}^{N-1} \left(\frac{b}{a}\right)^i}$$

$$= \frac{a^N p_N - b^N p_o}{\sum_{i=0}^{N-1} a^{N-1-i} b^i} \tag{2.24}$$

Under the assumptions (2.14) and (2.16), equation (2.13) can be written as,

$$\mathring{m} = \mathcal{A} K / (2\pi R)^{\frac{1}{2}}$$

Using equations (2.1) and (2.2), the pumping speed is given by,

$$S_p = \mathring{m}RT/p_{in}$$

For the system under consideration (Fig. 2.6),

$$T = T_c, \quad \text{and} \quad p_{in} = p_N$$

and therefore,

$$\begin{aligned} S_p &= \mathring{m}RT_c/p_N \\ &= RT_c\mathcal{A}K/(p_N(2\pi R)^{\frac{1}{2}}) \end{aligned}$$

substituting for K from equation (2.24),

$$\begin{aligned} S_p &= \frac{RT_c}{p_N}\frac{\mathcal{A}}{\sqrt{2\pi R}}\left[\frac{a^N p_N - b^N p_0}{\sum\limits_{i=0}^{N-1} a^{N-1-i} b^i}\right] \\ &= \frac{\mathcal{A}RT_c}{\sqrt{2\pi R}}\left[\frac{a^N}{\sum\limits_{i=0}^{N-1} a^{N-1-i} b^i} - \frac{b^N}{\sum\limits_{i=0}^{N-1} a^{N-1-i} b^i}\left(\frac{p_0}{p_N}\right)\right] \\ &= \frac{\mathcal{A}RT_c}{\sqrt{2\pi R}}\left[\alpha_N - \beta_N\left(\frac{p_0}{p_N}\right)\right] \qquad (2.25) \end{aligned}$$

where,

$$\alpha_N = \frac{a^N}{\sum\limits_{i=0}^{N-1} a^{N-1-i} b^i} \quad , \quad \beta_N = \frac{b^N}{\sum\limits_{i=0}^{N-1} a^{N-1-i} b^i}$$

Equation (2.25) gives the pumping speed of an N-stage ideal thermal transpiration pump. Over the applicable range of inlet pressure p_N $(p_0 \geq p_N \geq p_u)$, pumping speed may be calculated and thus, $S_p \sim p_N$ characteristics of the pump may be determined. This is undertaken in the next chapter.

CHAPTER III

PERFORMANCE CHARACTERISTICS AND COMPARISON

Often, the most important characteristic of a vacuum pump is its pumping speed versus inlet pressure curve(13, 14, 16-20), although, in a particular application, overriding concern may shift to some other factors such as cost of production and maintenance, ease of operation, contamination of gas, movement of pump parts etc.

It is imperative for a fair comparison, therefore, to compare not only the pumping speed characteristics but also other features of the pumping systems. In this chapter, characteristics of a thermal transpiration pump are obtained in terms of dimensionless parameters; characteristics of other vacuum pumps are reduced to the same parameters and compared; and lastly, particular features of a thermal transpiration pump are compared with those of other pumps.

3.1 Characteristics of a thermal transpiration pump

Although it is possible at this stage to obtain a S_p versus p_N curve for a particular choice of the remaining parameters in equation (2.25), the curve thus obtained may not be of much use in that it will not reveal the effect of these parameters on S_p. Due to the large number of independent variables involved, it would be desirable to reduce the equation into a form that involves only dimensionless quantities. Of course, these dimensionless quantities or groups should be meaningful in that they should reflect changes in parameters of interest. The non-dimensionalization of equation (2.25) is carried out in Appendix B, and is shown to be,

$$S_p^* = \frac{1}{\sqrt{2\pi}}\left[\frac{1}{\sum_{i=0}^{N-1}(T_r)^{-\frac{i}{2}}} - \frac{1}{\sum_{i=0}^{N-1}(T_r)^{\frac{N-i}{2}}}\,\frac{1}{p^*}\right] \qquad (3.1)$$

where,

$S_p^* = S_p/\mathcal{A}(RT_c)^{\frac{1}{2}}$

$T_r = T_h/T_c$

$p^* = p_N/p_o$

N = number of stages

Equation (3.1) contains 4 independent, dimensionless groups. Due to the choice of parameters in these groups, S_p^* is a measure of the pumping speed and p^* is a measure of the vacuum chamber pressure. A S_p^* versus p^* curve will therefore reflect the usual pumping speed curve associated with any vacuum pump.

The dimensionless pumping speed S_p^* is plotted against the pressure ratio p^* for various values of N at different temperature ratios (T_r) in figures 3.1(a) and 3.1(b).

The number of stages is varied from 1 through 21 in steps of 5. The range of temperature ratios chosen is from 1.5 to 3.5, in steps of 0.5. The following observations can be made from these curves :-

(1) Maximum pumping speed does not depend on the number of stages - it is a function only of the temperature ratio.

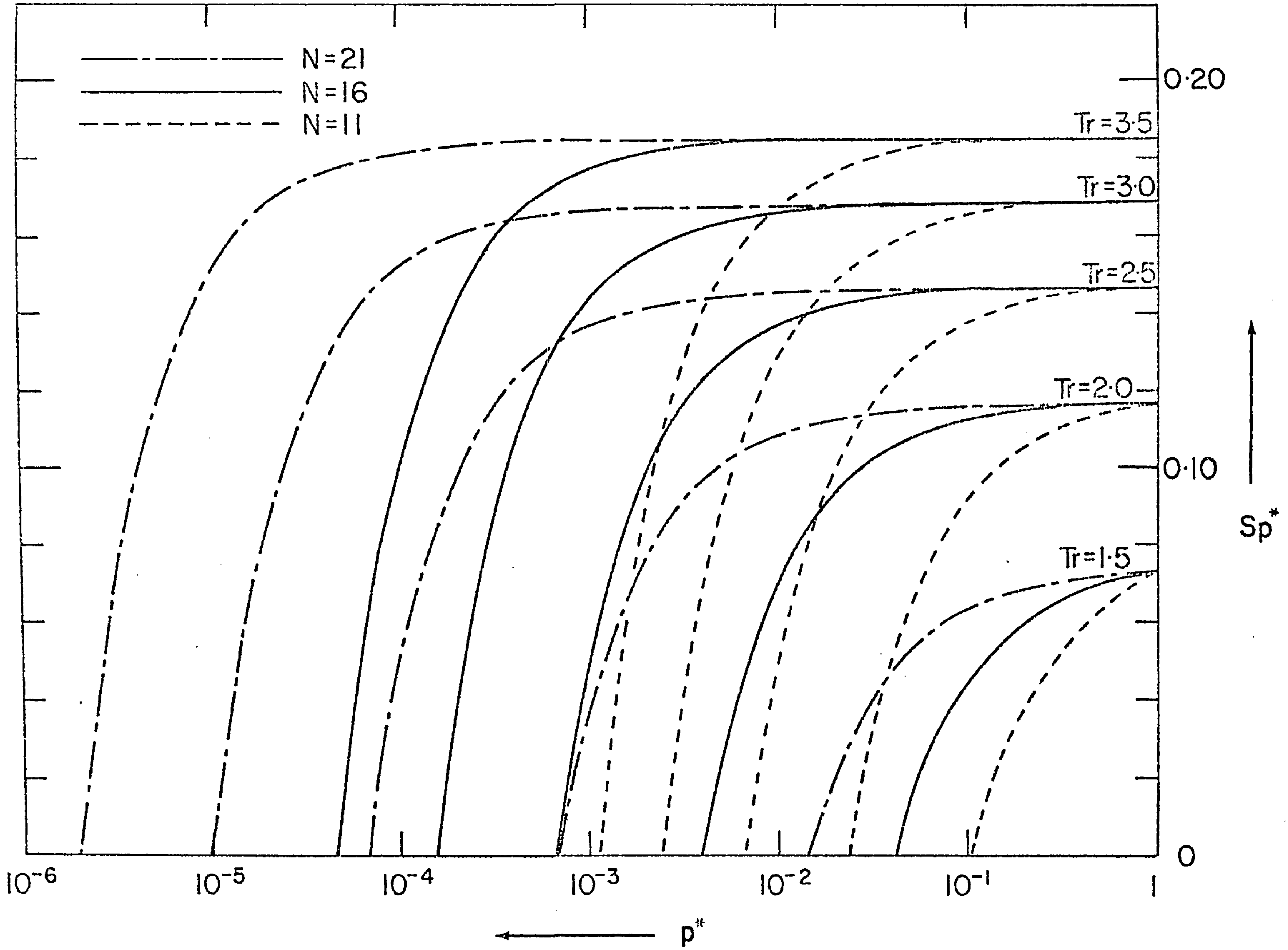


FIG. 3.1a CHARACTERISTICS OF MULTI-STAGE THERMAL TRANSPIRATION PUMP (N=21,16,11)

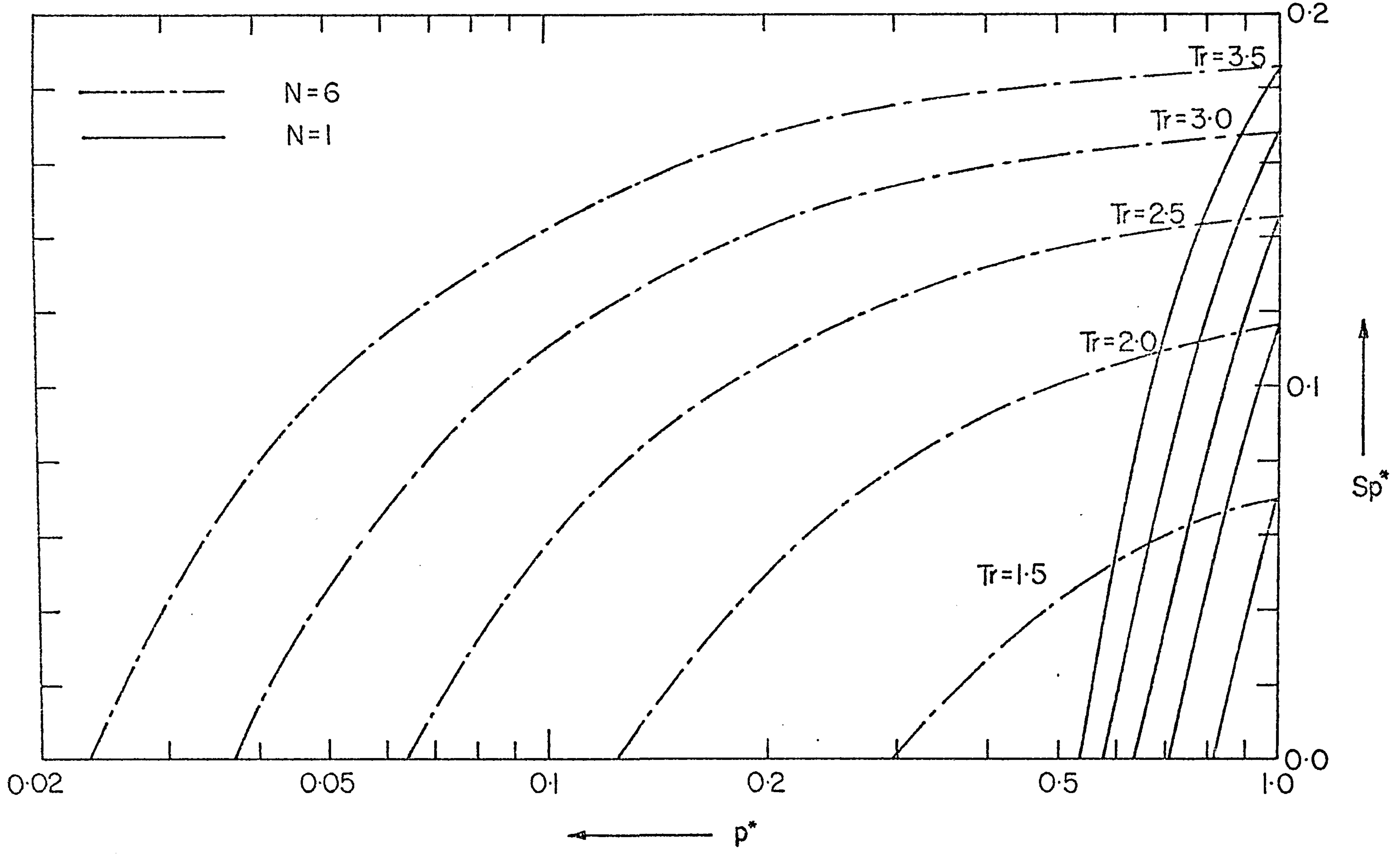


FIG. 3.1b CHARACTERISTICS OF MULTI STAGE THERMAL TRANSPIRATION PUMP (N=6,1)

(2) Pressure ratio is a function of both, the number of stages and the temperature ratio.

(3) For any given number of stages, a greater temperature ratio results in greater pumping speed and pressure ratio. The relative gains in pumping speed and pressure ratio, however, decrease as the temperature ratio increases. That is to say that the transpiration effect is maximum for temperature ratios greater than but close to unity.

3.2 Characteristics of alternative vacuum pumping methods

Various types of available vacuum pumps, catagorized according to their principle of operation, are listed in table 3.1. The table also gives usual pressure ranges of operation and pumping speeds. It should be noted that the figures quoted therein are only approximate. Figure (3.2) shows the advertized pumping speed curves for some commercially manufactured pumps. The data has been taken from the catalogs of several prominent pump manufacturers(17, 18), and reduced, as required, to appropriate units for comparison. Approximate overall dimensions are also included since they are necessary for comparison in the following chapter.

In order to compare the characteristics in figure 3.2 with those of a thermal transpiration pump, it is essential that they also be drawn in terms of dimensionless parameters, which have been defined as,

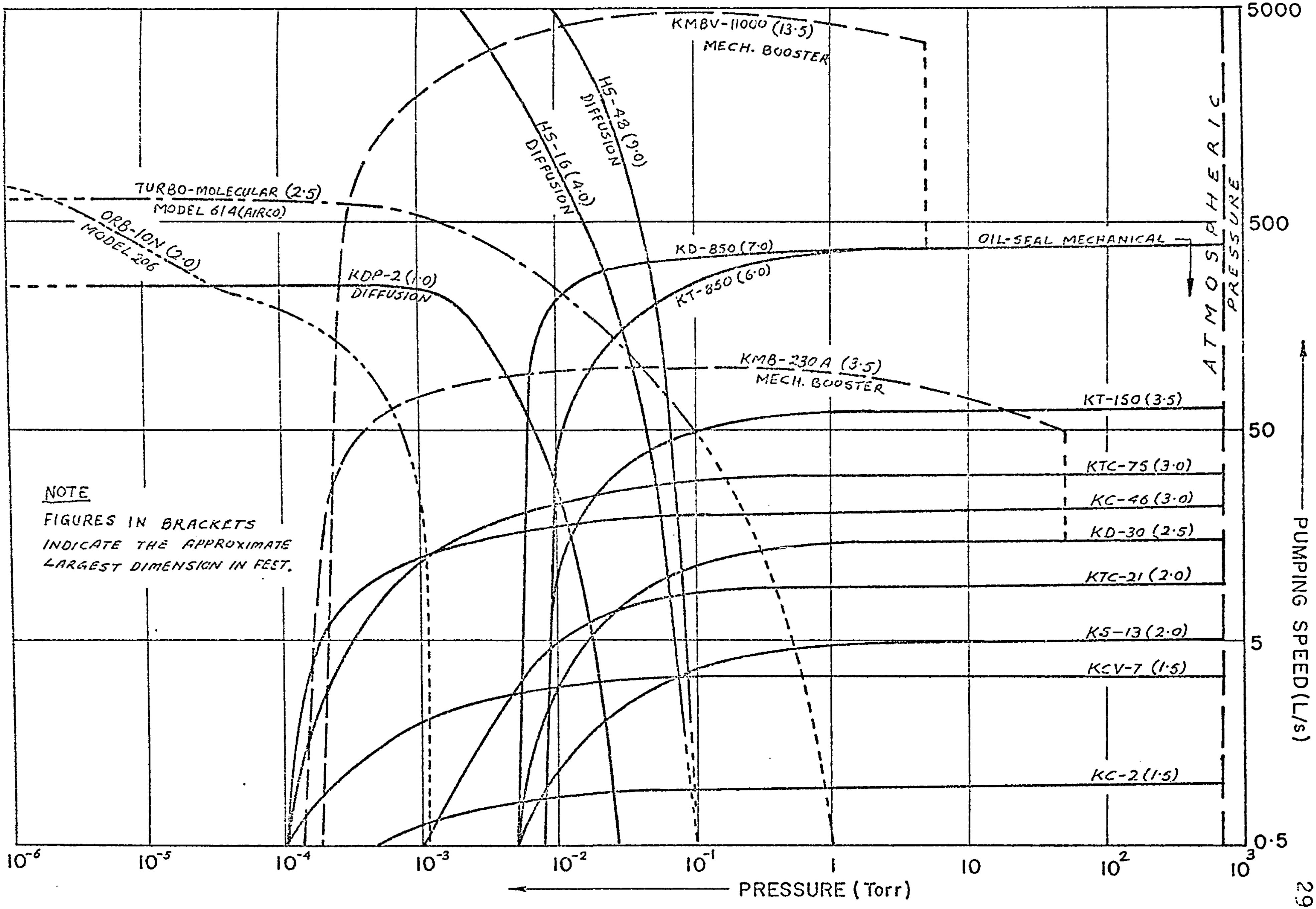


FIG. 3.2 CHARACTERISTICS OF ALTERNATIVE VACUUM PUMPING SYSTEMS

Table 3.1 VACUUM PUMPS*

CATAGORY	CAPACITY (l/sec)	OPERATING PRESSURE RANGE (torr)
1. WATER JET PUMP	up to 500	760 to 20
2. WATER RING PUMP	1 to 2500	760 to 25
3. STEAM EJECTOR	up to 100000	760 to 0.03
4. OIL-SEALED ROTARY PUMPS	0.25 to 500	760 to 10^{-4}
5. MECHANICAL BOOSTER	50 to 35000	10 to 10^{-3}
6. MOLECULAR PUMP	up to 10000	10^{-1} to 10^{-10}
7. VAPOUR DIFFUSION PUMP	up to 95000	10^{-2} to 10^{-9}
8. OIL VAPOUR BOOSTER PUMP	up to 23000	1 to 10^{-4}
9. SPUTTER-ION PUMP	1 to 7000	10^{-2} to 10^{-11}
10. RADIAL FIELD PUMP	400 to 800	10^{-4} to 10^{-11}
11. TITANIUM SUBLIMATION PUMP	up to thousands	10^{-3} to 10^{-11}
12. SORPTION PUMP	depends on sorbent	760 to 10^{-10}
13. CRYOPUMP	up to millions	10^{-3} to 10^{-10}

* compiled from reference (19).

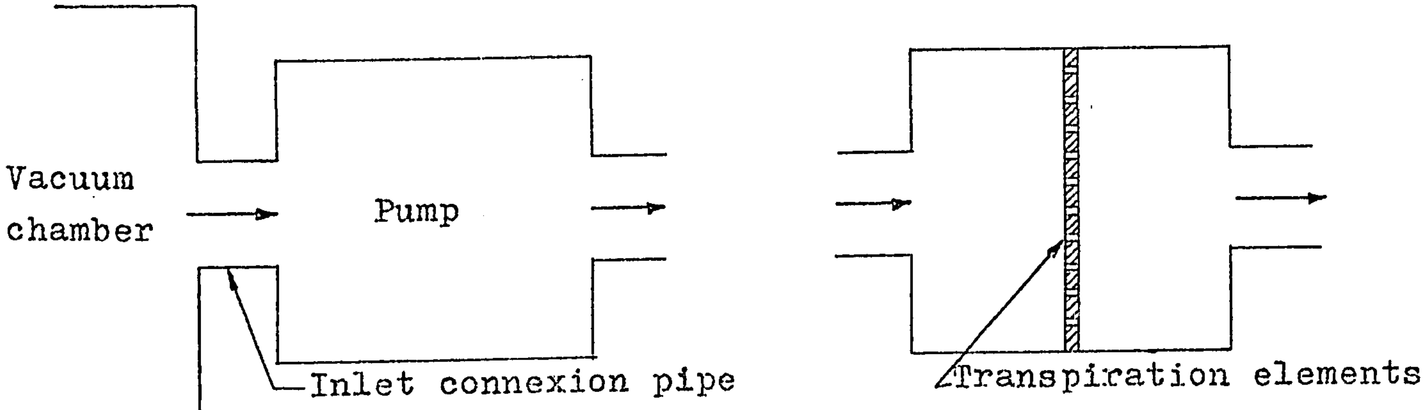
$$S_p^{*} = S_p / (RT_c)^{\frac{1}{2}} \tag{3.2}$$

and $$p^{*} = p_N / p_o \tag{3.3}$$

For the thermal transpiration pump, equation (2.25) gives,

$$S_p \propto \mathcal{A}$$

and therefore, $\mathcal{A}$ directly controls S_p. In all other pumps, there must be some critical area which controls the rate of flow. Unfortunately, no data is published by pump manufacturers regarding this critical area. However, a first approximation may be made by selecting the inlet connexion area for this purpose as explained below.



(a) Conventional pump (b) T.T. Pump

Fig. 3.3 Inlet-connexion area vs. transpiration area

The conductance of an aperture or a tube at low pressures is directly proportional to its area (Ref. 14, p. 47-49). Also, for m apertures or tubes in parallel (Fig. 3.3(b)), the combined conductance is given as[14] mC, where C is the conductance of each element. Thus, the transpiration area $\mathcal{A}$ is proportional to the combined conductance of pumping elements. Conductance everywhere else in the pumping system should theoretically be equal to and in practice somewhat more than this controlling conductance which is nothing but the pumping speed itself. It follows that the inlet-connexion area of the thermal transpiration pump must approximately be equal to $\mathcal{A}$ and therefore, choosing the inlet-connexion area in the case of other pumps as a comparable quantity to $\mathcal{A}$ is justified. It may be noted here that these areas are fair indicators of overall size of the pumps belonging to the same type but may not be related to those of other types.

The data of figure 3.2 has been replotted in terms of dimensionless quantities S_p^* and p^* in figure 3.4, as all the quantities in equations (3.2) and (3.3) are known.

Two significant observations can be made from such a plot :-

(i) the pressure ratio that a pump is capable of producing, and

(ii) the relative magnitude of its pumping speed.

It is seen from figure 3.4 that whereas the thermal transpiration pump covers a comparable range of pressure ratios, its pumping speed (as reflected in S_p^*) is rather low. This fact may not, however, be too significant, since a low pumping speed only means a longer time required for pumpdown.

Besides pumping speed and pressure ratio, another

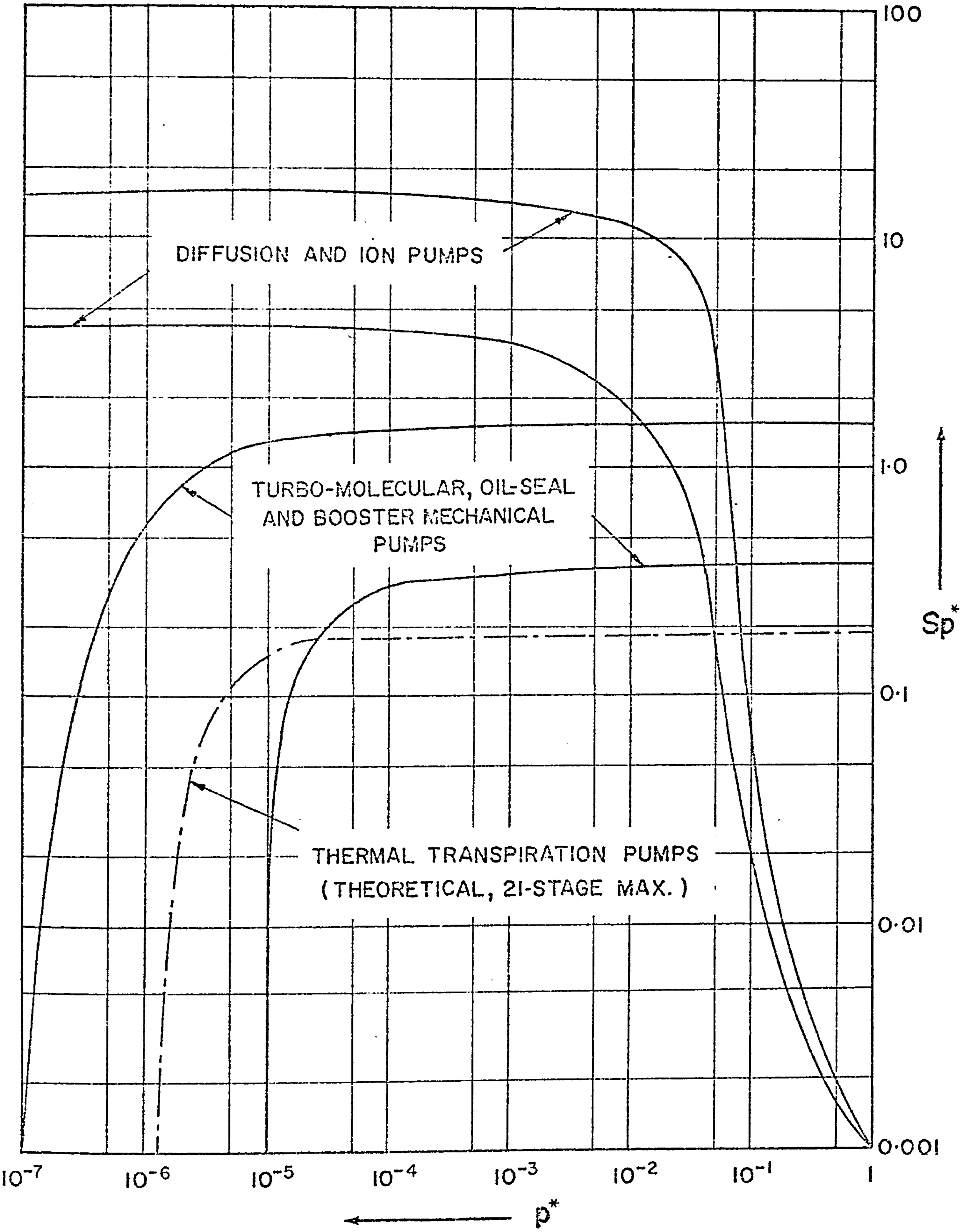


FIG. 3.4 PERFORMANCE COMPARISON OF VACUUM PUMPS

quantity of significance is the ultimate pressure. A comparison of ultimate pressures obtainable with various types of pumps is not evident from figure 3.2 but can be easily obtained by using the values of p_o given therein. For example, oil-sealed machanical pumps, for which p^* varies from 10^{-5} to 10^{-7} and which have $p_o = 760$ torr, will produce ultimate pressures from 10^{-2} to 10^{-4} torr. Likewise, the diffusion pumps and turbo-molecular pumps go down to 10^{-8} torr, and orb-ion pumps down to 10^{-10} torr. Theoretically these pumps are capable of going down to infinitesimally small pressures and, with proper design, so also is the thermal transpiration pump. However, backstreaming, which limits ultimate pressure in these pumps, is not to be expected in a thermal transpiration pump and it is, therefore, capable of producing extremely small ultimate pressures. Unfortunately, physical constraints, mainly on pump size, become prohibitive in going to extremely low pressures. This subject is discussed further in the next chapter in which practical limitattions to the design of a thermal transpiration pump are considered.

It is interesting to note before closing that vacuum pump performance comparison through dimensionless parameters is, to the author's knowledge, a new approach. S_p^* includes not only the pumping speed but also the size of pump (through $\mathcal{A}$), the gas pumped (through R), and the inlet temperature of gas (through T_c). p^* gives the pressure ratio, again a better indicator of pump performance.

3.3 Comparison of other features

A brief description of various types of pumps is given in appendix C. The following observations can be made :

(1) All mechanical pumps, including the molecular drag pump, require many moving parts. This involves noise, dirt, vibration and lubrication problems. The thermal transpiration pump is completely free of these problems.

(2) All mechanical and vapour diffusion pumps leave contamination in the vacua produced. Other pumps that claim to produce contamination free vacua have other problems. Ion pumps, for example, have very low pumping speeds for inert gases, and turbo-molecular pumps are not suitable for pumping lighter gases. The thermal transpiration pump is well suited for inert and light gases.

(3) When pumped by ion (or getter-ion) pumps, the gas undergoes changes in electrical properties. This may be of importance in plasma physics studies. The thermal transpiration pump does not change electrical properties.

(4) The energy requirement for a thermal transpiration pump is in the form of heat (having higher availability) and therefore, the pump could be very economical to operate. The use of heat

also opens up interesting possibilities such as the use of solar radiation, especially in space.

(5) In principle, the pump can operate at <u>any</u> pressure - high or low. The limitation is imposed only by the ability to produce proper-sized orifices.

In conclusion, the thermal transpiration pump, despite its relatively low pumping speed, stands favorably among others. The following chapter deals with practical design aspects relating to operation and construction.

CHAPTER IV

A TENTATIVE DESIGN OF THE IDEAL PUMP

Up to this point attention has been paid only to the feasibility and desirability of a thermal transpiration pump. This chapter undertakes to investigate the preliminary considerations involved in realizing a practical device.

Recalling equation (2.25), it can be noted that α_N and β_N are functions only;of the parameters a, b and N, all of which are fixed. R may be fixed by selecting a particular gas, and since air happens to be the most frequently pumped gas dealt with in vacuum systems, it is chosen here. Gases to be pumped from vacuum chambers will ordinarily be at room temperature and therefore, the choice of $T_c = 300^{\circ}K$ is reasonable. The selection of values for p_o and $\mathscr{A}$, appropriate to achieving maximum utilization of thermal transpiration effect, requires extensive investigation, which follows.

It has been shown(13) that the mean free path, λ, in a gas may be given by

$$\lambda = 8.589\eta(T/M)^{\frac{1}{2}}/p \quad \text{cm.} \tag{4.1}$$

where, p = pressure of the gas (torr)

η = coefficient of viscosity (poise)

T = temperature ($^{\circ}K$)

M = molecular weight

It is also known that the viscosity of a gas is independent of its pressure(1,2,13). The variation of viscosity with temperature is not yet exactly known. However, as shown by Kennard(1) and Dushman(13), a very good approximation to the relationship is given by,

$$\eta \propto T^{n} \qquad (n > \tfrac{1}{2}) \tag{4.2}$$

where the coefficient n depends primarily on the gas type, and to some extent, on the temperature itself. The values of n for various gases over the applicable range of temperatures are given in reference (1). For air, n=0.79.

Equations (4.1) and (4.2) indicate that,

$$\lambda \propto 1/p \tag{4.3a}$$

and,

$$\lambda \propto T^{n + \frac{1}{2}} \tag{4.3b}$$

Fig. 4.1 shows the variation of K_n ($K_n = \lambda/d$) with pressure and temperature.

From equations 4.3a and 4.3b, it can be inferred that smallest values of λ will occur at points in a system where the pressure is maximum and the temperature minimum. For the system under consideration,

$$\left[\lambda\right]_{\text{min.}} = \left[\lambda\right]_{\substack{p=p_o \\ T=T_c}} \tag{4.4}$$

Likewise, the largest value of λ will occur when the pressure is minimum (p_u) and the temperature maximum (T_h), so that

$$\left[\lambda\right]_{\text{max.}} = \left[\lambda\right]_{\substack{p=p_u \\ T=T_h}} \tag{4.5}$$

Conditions (4.4) and (4.5) determine the smallest and the largest values of the mean free path that the pumping system will encounter during its operation.

To ensure free-molecule flow in an orifice of diameter d, it was indicated in chapter II that $K_n \geq 5$ should be maintained. Therefore,

$$\lambda/d \geq 5$$

or, $$d \leq \lambda/5 \tag{4.6}$$

On the other hand, to ensure continuum flow in the reservoirs, $K_n \leq 0.01$ should be maintained. Therefore,

$$\lambda/D \leqslant 0.01$$

or, $$D \geqslant \lambda/0.01 \tag{4.7}$$

where D is a characteristic dimension of the reservoir.

Condition (4.6) dictates that d be smaller than $\lambda/5$ always, and condition (4.4) gives the minimum value of λ which is dependent on the forepressure p_o. In an actual pump, it may not be possible to construct the orifice diameter(d) smaller than a certain minimum. Assuming that minimum value of d to be 0.1 cm., a reference to Fig. 4.1 reveals that free-molecule flow in the orifice is realizable if $p \leqslant 10^{-2}$ torr. This means that p_o may not exceed 10^{-2} torr under the above assumption.

It is apparent that the highest pressure is always encountered by the first stage, and that the subsequent stages, though initially at the same pressure, encounter gradually decreasing pressures. The dimension of the orifice may therefore be progressively increased in the subsequent stages as the pumping proceeds. However, this concept may be useful only when variable-area orifices are used.

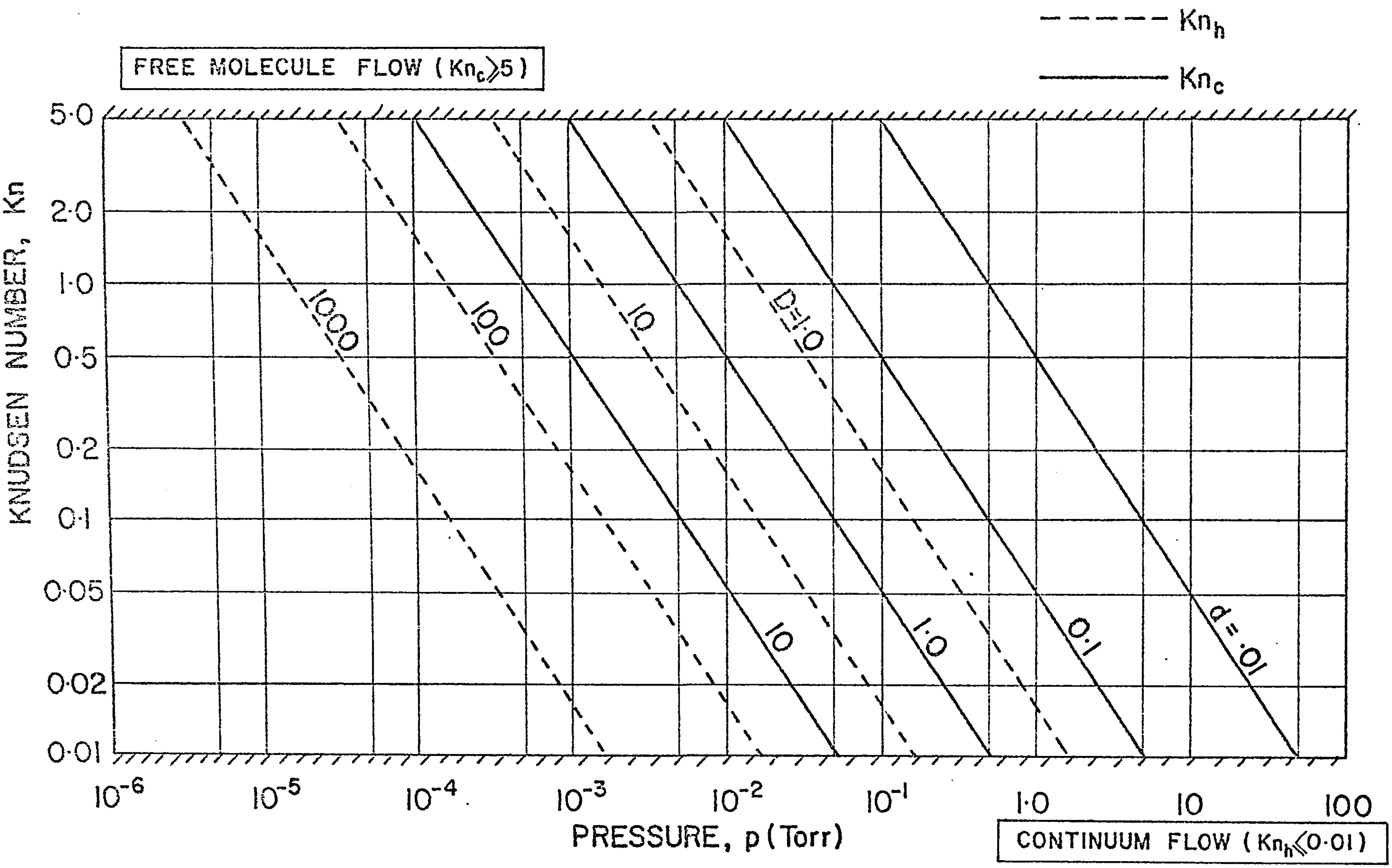


FIG. 4.1 VARIATION OF KNUDSEN NUMBER WITH PRESSURE OF AIR AT $T_h = 900°K$ & $T_c = 300°K$

The characteristic dimension (D) of the reservoirs may be estimated from condition (4.7). In each reservoir, the largest value of λ will occur at the hot end of the lowest pressure stage. For the n^{th} reservoir (n^{th} stage in figure 2.7) equations (4.5) and (4.7) may be modified to give,

$$\left[\lambda_n\right]_{max.} = \left[\lambda\right]_{\substack{p=p_u \\ T=T_h}} \tag{4.8}$$

$$D_n \geqslant \left[\lambda_n\right]_{max.}/0.01 \tag{4.9}$$

where $\left[\lambda_n\right]_{max}$ = maximum value of λ expected in the n^{th} reservoir

and, p_n = minimum pressure expected in the n^{th} reservoir

$= p_o(T_c/T_h)^{n/2}$

From Fig. 4.1,

$$\left[\lambda\right]_{\substack{p=1 \\ T=800}} = 1.584 \text{ cm.} \tag{4.10}$$

Using (4.10) and (4.3a), equation (4.8) becomes,

$$\left[\lambda_n\right]_{max.} = 1.584/p_u$$

or

$$\left[\lambda_n\right]_{max.} = 1.584/p_o(T_c/T_h)^{n/2} \tag{4.11}$$

substituting from (4.11) in (4.9),

$$D_n \geqslant 1/0.01 \left\{1.584/p_o(T_c/T_h)^{n/2}\right\} \tag{4.12}$$

The minimum values of D_n are calculated from (4.12) as given below,

D_1 = 259 cm.
D_2 = 423 cm.
D_3 = 692 cm.
D_4 = 1126 cm.
D_5 = 1839 cm.
D_6 = 3004 cm.
D_7 = 4905 cm.
D_8 = 8010 cm.
D_9 = 1.31×10^4 cm.
D_{10} = 2.14×10^4 cm.
D_{11} = 3.49×10^4 cm.
D_{12} = 5.69×10^4 cm.

$D_{13} = 9.30\text{x}10^4$ $D_{16} = 4.05\text{x}10^5$ $D_{19} = 1.76\text{x}10^6$
$D_{14} = 1.52\text{x}10^5$ $D_{17} = 6.62\text{x}10^5$ $D_{20} = 2.88\text{x}10^6$
$D_{15} = 2.48\text{x}10^5$ $D_{18} = 1.08\text{x}10^6$ $D_{21} = 4.70\text{x}10^6$

The maximum pumping speed of this ideal pump may be found using figure (3.1a) or, calculating from equation (2.8),

$$(S_p^*)_{max} = 0.154651 = (S_p)_{max}/(A(RT_c)^{\frac{1}{2}})$$

since, $A = \pi(0.1)^2/4 \quad \text{cm}^2$
$R = 2152$ torr-cm^3/mole-^{o}K
$T_c = 300\ ^o\text{K}$

therefore,

$$(S_p)_{max} = 0.975946 \quad \text{cm}^3/\text{sec}$$

The ultimate pressure, obtainable with a 21-stage pump, may be found using figure (3.1a) or, from equation (2.12),

$$p_{21}^* = 3.3676 \text{ x } 10^{-5} = p_{21}/p_o$$

therefore,

$$p_{21} = 3.3676 \text{ x } 10^{-7}$$

Thus, the ideal 21-stage thermal transpiration pump, as proposed above, can provide a good pressure reduction(from 5 x 10^{-2} torr to 10^{-7} torr) but, has very small pumping speed (1 cm^3/sec) and requires impractical dimensions. The possibility of increasing the pumping speed by increasing the number of passages per stage is investigated later in section 4.2.4. The overall dimensions of the device may be reduced in one or more of the following ways :-

(i) Shifting the pressure-range: If it is possible to provide orifices of smaller diameter than that assumed above, the operating forepressure may be correspondingly increased and consquently, reservoir dimension (D_n) may be reduced. For example, if $d = 10^{-3}$ cm, the operating forepressure p_o may be increased to 1 torr and therefore, all values of D given above may be reduced by two orders of magnitude. This possibility is further discussed in section 4.2.2.

(ii) Reducing the number of stages : The last stage has the maximum dimensions. Overall pump size may be brought down by reducing the number of stages. Of course, this will lead to a corresponding reduction in pump pressure ratio and hence, also the ultimate pressure obtainable.

(iii) Decreasing the temperature ratio: A smaller temperature ratio will give a smaller pressure ratio, thereby decreasing the values of λ. This is similar to (ii) in effect. However, the rate of increase of λ increases with temperature(1) and therefore, lower temperatures are preferrable. On the other hand, a reduced temperature ratio also results in lower pumping speed (see Fig. 2.3 or Fig. 3.1). It may be advisable therefore, to provide high temperature ratios but low absolute temperatures. For example, if T_c = 76 °K (using liquid nitrogen for cooling, say) and T_h = 500 °K, a temperature ratio of nearly 7 is provided at relatively low absolute temperatures.

(iv) Relaxing the flow conditions : To some extent, the large reservoir dimensions are a result of the severe criteria imposed on the flow conditions, both in the reservoirs and in the orifices. By relaxing these conditions significant reduction in reservoir dimensions can be achieved. This is investigated in section 5.5.

CHAPTER V

DEVIATIONS FROM THE IDEAL CASE

It was pointed out in chapter II that the transpiration equation (2.4) is valid under certain conditions described by the basic assumptions. The same conditions apply in the case of a multi-stage pump. In the previous sections, the difficulties in maintaining some of these conditions have been discussed. It will be the attempt in this section to investigate the effects arising when some of these ideal constraints are relaxed and the benefits to be obtained by a set of compromise design criteria.

5.1 Orifice and tube

In a practical situation any orifice will have some finite length. A true orifice requires that $L/d \rightarrow 0$, but, since d must be kept small by necessity, L/d cannot be made infinitecimally small. This leads to the conclusion that the opening must be treated as a tube, rather than an orifice. This altered picture brings in two more considerations that are discussed below.

Formulae, governing free-molecule flow in tubes, have been known for some time. However, different formulae have been developed for different classification of the tubes (e.g. orifice, short tube or long tube). Recently, following Eckert's suggestions on similarities between the processes of free-molecule gas flow and radiative heat transfer(22), Sparrow, Johnsson and Lundgren presented a unified analysis of mass rate of flow due to thermal transpiration in the free-molecule flow regime applicable for all values of L/d and have given the equation(23),

$$\frac{\mathring{m}}{\frac{\pi}{4}d^2} = \frac{1}{\sqrt{2\pi R}}\left[\frac{p_1}{\sqrt{T_1}} - \frac{p_2}{\sqrt{T_2}}\right]\left[1 - 4\int_0^{L/d} f(x)\,F_{x-1}\,dx\right] \qquad (5.1)$$

Equation (5.1) is similar, as would be expected, to the orifice-equation (2.4), except for the last bracketed term which accounts for the effect of L/d. In effect, the assumptions made in this analysis are also similar to those for the orifice analysis. They are,

1. Free-molecule flow in tube.
2. Pressures and temperatures maintained constant with respect to time in the reservoirs (or quasi-static in case of pump).
3. The surface area of the reservoir walls is much larger than the tube opening.
4. Reflection at the tube wall is diffuse and follows Lambert's Cosine Law.
5. No absorption or adsorption of gas by the tube wall.

The first three assumptions listed above are equivalent to those given in section 2.2. The last two are applicable to tubes only. Figure (5.1) is redrawn from figure (3) of reference (23) and shows the effect of increasing L/d in decreasing mass rate of flow. As compared to the orifice flow, a reduction of almost 50% in the mass

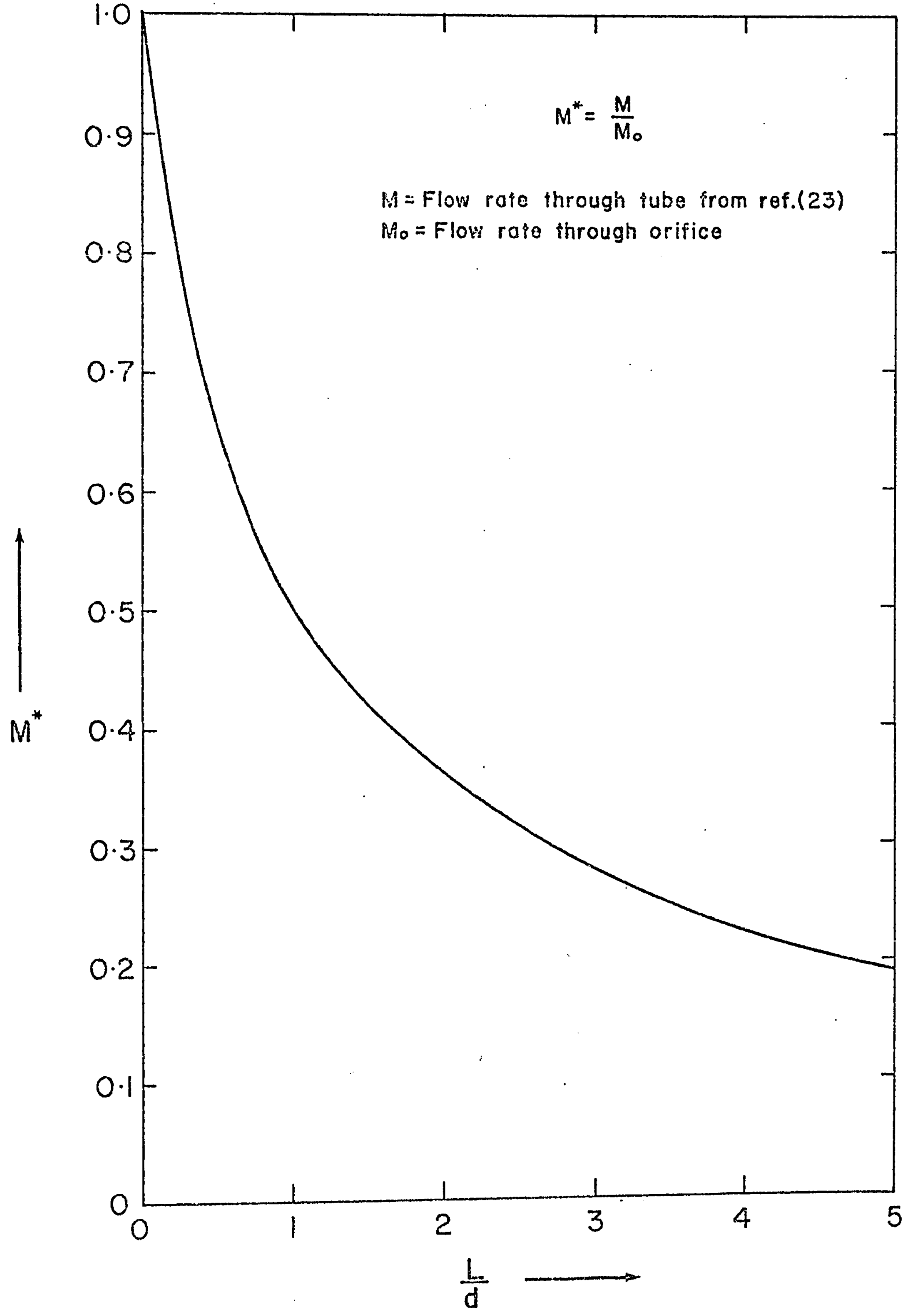


FIG. 5.1 EFFECT OF $\frac{L}{d}$ RATIO ON MASS FLOW RATE

rate of flow is predicted for L/d = 1.

An important effect that is brought into the picture with tubes relates to the degree of surface smoothness. It is assumed in the analysis of Sparrow et. al. that the reflection of molecules at the tube walls is diffuse; that is, Lambert's Cosine Law is obeyed by the reflecting molecules. This assumption has been held from the earliest times in the developement of the kinetic theory of gases (1,2), and in recent times, experiments using extremely accurate molecular-beam scattering techniques(24) tend to confirm it to a large extent. However, the possibility of some specular reflection accompanying the diffuse reflection cannot be completely ruled out. Indeed, it has always been considered the best explaination of the descrepancies occuring between theoretical and experimental results. Hobson & Edmonds(25) found from their experiments that 'smooth' surfaces allow greater mass-flow from cold to hot side than do 'rough' surfaces. In their experiments, the 'rough' surface (corresponding to perfectly diffuse reflection) had to be specially created by acid-leaching of glass tubes, whereas the normal glass tube surface was treated 'smooth' (corresponding to mostly diffuse but some specular reflection). Accordingly, for normal surfaces, instead of equation (2.5) they observed,

$$p_1/p_2 = \text{"a"} \; (T_1/T_2)^{\frac{1}{2}}$$

where "a" was found to have values from 1.1 to 1.3. It may be noted that Hobson(26,27) explored this idea further in an attempt to employ it for pumping purposes, calling it accommodation pumping.

It may be concluded from these observations that the tube should be as short as possible and have a smooth surface, if possible.

5.2 Size of orifice (or tube) and of reservoir

In the tentative design considered in the last section, it was somewhat arbitrarily assumed that the orifice diameter be kept 0.1 cm. The choice is primarily dictated by the following considerations :-

(a) the forepressure at which the pump will operate,

(b) the minimum physically realizable size of the orifice, and

(c) the limits on physical dimensions of the overall system.

It may be noted that (c) is directly related to (a). The higher the operating forepressure, the smaller will be the mean free paths involved and consequently, smaller will be the physical dimensions of the overall system. High forepressures, however, will demand extremely small orifice dimensions. For example, when pumping air from atmosphere (p_o = 760 torr), $\lambda \simeq 6.5 \times 10^{-6}$ cm. and therefore, $d \simeq 10^{-6}$ cm.! Orifices of such dimensions are perhaps not possible and certainly not easy to produce.

It should be mentioned that although artificially produced porous media (like millipore filters, silica gel etc.) may possess pores of comparable dimensions, and that Reynolds actually employed a porous plate in his earliest experiment(15) to demonstrate the phenomenon of thermal transperation, the analysis is both different and difficult. Based on a model proposed by Mason, Evans & Watson(15), the researchers Hanley(28,29), and more recently, Hopfinger and Altman(9) have attempted the solutions of this problem. The model employs what Mason et. al. called "Dusty-gas Theory" and visualizes the porous medium as consisting of large spherical particles, referred to as "dust", which are constrained in space(15). Unlike the kinetic theory analysis, the membrane containing the pores can no longer be treated as a geometric barrier alone since it can also react with the diffusing gas. The gas dissolves in the membrane and the heat of transport is much greater than that predicted by kinetic theory alone - it contains a heat of solution term(28). The phenomenon is called thermo-osmosis and is closely related to thermal transpiration. In terms of thermal transpiration effects, the flow rate is greatly reduced. The reduction in flow rate may also be attributed to large L/d ratios that are always encountered when using porous media. Hopfinger and Altman(9) were able to obtain a maximum flow rate of less than 1 cm^3/sec at a temperature difference of 100°C using porous ceramic of area 38.5 cm^2 and thickness 0.6 cm (effective pore diameter $\simeq 10^{-4}$cm.). The gas used was helium and the pressure on either side of the ceramic was atmospheric. Briefly, it may be concluded that while porous media are an interesting alternative to tubes and orifices, their use poses problems such as extremely low flow rates, complexity of analysis, lack of information(15), possibility of back flow(28) and of course, their inherent suitability to relatively high pressures only.

5.3 Validity of Perfect Gas Law

The combined effect of low pressures and high temperatures may cause the gases to dissociate and/or ionize, thereby making them deviate from the perfect gas law (pV=mRT) appreciably. Besides, in certain application (where even a small back streaming is undesirable) it may be necessary that the gas being pumped is not dissociated or ionized. It was felt necessary, therefore, to investigate whether dissociation and ionization are likely to occur under the expected conditions of pump operation. This investigation is carried out in appendix 'D' and it shows that the effect of these two phenomena, if any, would be negligible.

5.4 Number of orifices per stage

The pumping speeds obtainable with a single-orifice pump are extremely small, as was shown in chapter IV. A natural suggestion to improve them is to employ several orifices. This, however, causes certain deviations from the assumed conditions of the earlier analysis. Of primary significance, the gas entering the orifice may not be at the temperature of the bulk of the gas in the reservoir, and in fact, may not be in equilibrium and thus representable by a maxwellian distribution.

If the amount of gas transpiring through a diaphragm into a reservoir is increased substantially, it will cause a non-uniform temperature distribution in the gas body adjoining the diaphragm. In addition, the temperature of the diaphragm, which will likely be different from the bulk gas temperature in either reservoir, will influence the gas temperature in the region immediately next to it.

Poisson showed long ago that at a wall bounding an unequally heated gas at low pressures, there exists a discontinuity of temperature (loq. cit. ref. 1, p-311). The discontinuity, known as temperature jump, may be expressed by the relation,

$$T_k - T_w = g\ \partial T/\partial n$$

where T_w is the wall temperature and T_k is what the temperature of the gas would be if the temperature gradient along the outward drawn normal to the wall, $\partial T/\partial n$, be continued without change right up to the wall. The quantity g represents a characteristic distance in the gas and is called the temperature jump distance. Experiments carried out by Smoluchowski(1) show that $g = 2.7\lambda$ for air.

To incorporate the above deviations from the ideal model in an exact manner, one should determine the variations in the distribution function near the separating boundary.

The task is extremely complicated, and in any case, beyond the scope of this work. An approximate theoretical development is described below.

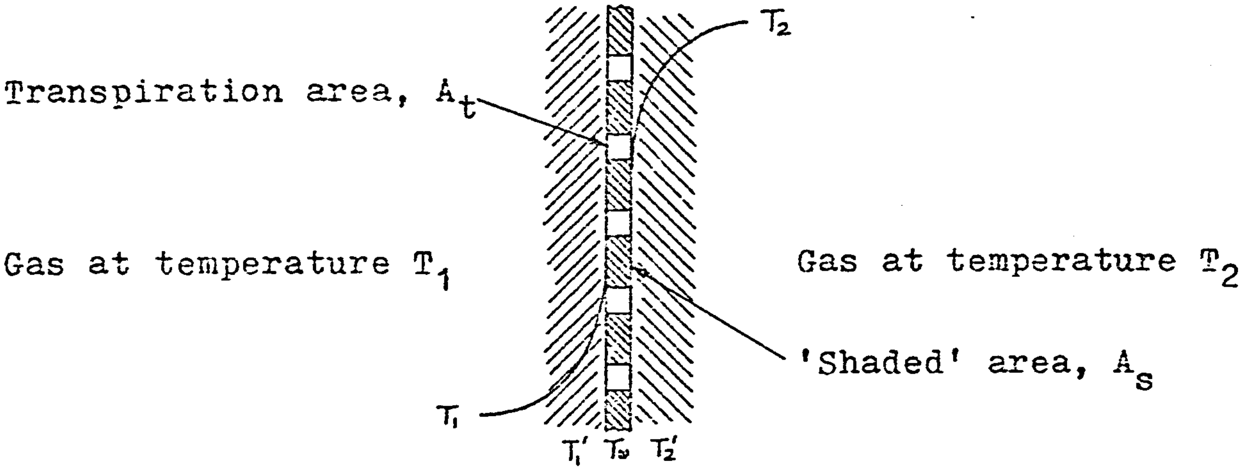


Fig. 5.2 Effect of multi-passage transpiration

Let a be the fraction that represents the ratio of the total transpiration area to the total stage x-section area ($a = A_t/(A_t + A_s)$). Obviously, the fraction of 'shaded' area would be $(1 - a)$.

The following assumptions are made in this analysis:-

(1) T_1- molecules travelling towards the wall acquire, after hitting the T_2'- molecules coming out of the transpiration area and molecules reflected from the wall, a new temperature T_1'. Likewise, T_2- molecules on the other side acquire temperature T_2'. Thus, in the immediate vicinity of the wall the gas bodies are at temperatures T_1' and T_2', rather than at T_1 and T_2 respectively.

(2) The wall temperature T_w is the arithmatic mean of T_1' and T_2'.

(3) Molecules reemitting from the wall, as well as those transpiring from the other side, possess energies described by the maxwellian distribution function.

(4) Energy carried by a molecule can be represented by the temperature of the ensemble of gas molecules to which it belongs.

(5) The number density of the molecules is approximately the same on the two sides of the wall. This means that the number of molecules coming out of the orifices or from the 'shaded' area is proportional to the respective areas.

(6) The redistribution of energy produces molecules having the arithmatic mean of the energies associated with the colliding molecules.

Considering an energy balance at a plane near the wall (the dotted line left of the wall in fig. 5.2),

(i) Energy associated with the T_2'- molecules transpiring from the other side = $\overleftarrow{aT_2'}$

(ii) Energy associated with the molecules reflecting from the wall = $\overleftarrow{(1 - a)T_{re_1}}$, where T_{re_1} is the temperature of the molecules reflected into the reservoir '1' after partial or complete energy accommodation at the wall.

(iii) Energy flux from the gas bulk = $\overrightarrow{T_1}$ (since area is unity)

Thus,

total energy flux from right = $aT_2' + (1-a)T_{re_1}$

total energy flux from left = T_1

These two fluxes should give, after redistribution of energy, the T_1'-molecules, so that,

$$\frac{aT_2' + (1-a)T_{re_1} + T_1}{2} = T_1'$$

or, $$aT_2' + (1-a)T_{re_1} + T_1 = 2T_1' \tag{5.2}$$

The energy accommodation coefficient, α, is defined as (see for example ref.(1), p-312),

$$\alpha = (T_i - T_{re})/(T_i - T_w)$$

where T_i is the temperature of the incident molecules. Thus,

$$T_{re} = (1-\alpha)T_i + \alpha T_w$$

or, $$T_{re_1} = (1-\alpha)T_1' + \alpha T_w \tag{5.3}$$

Substituting from (5.3) into (5.2),

$$aT_2' + (1-a)\Big[(1-\alpha)T_1' + \alpha T_w\Big] + T_1 = 2T_1' \tag{5.4}$$

The wall temperature, T_w, is given by,

$$T_w = (T_1' + T_2')/2 \tag{5.5}$$

Substituting from (5.5) into (5.4), and rearranging the terms,

$$T_1'\left\{4 - 2(1 - a)(1 - \alpha) - \alpha(1 - a)\right\} - T_2'\left\{2a + \alpha(1 - a)\right\} = 2T_1$$

or, $$T_1'\{2 + 2a + \alpha(1 - a)\} - T_2'\{2a + \alpha(1 - a)\} = 2T_1$$

Introducing $y = \{2a + \alpha(1 - a)\}$,

$$(y + 2)\, T_1' - yT_2' = 2T_1 \qquad (5.6)$$

By analogy, an energy balance on the right-hand side of the wall gives,

$$(y + 2)T_2' - yT_1' = 2T_2 \qquad (5.7)$$

Equations (5.6) and (5.7) may be solved simultaneously for T_1' and T_2' in terms of T_1, T_2, and y, giving

$$T_1' = \{(y + 2)T_1 + yT_2\}/2(y + 1) \qquad (5.8)$$

and, $$T_2' = \{(y + 2)T_2 + yT_1\}/2(y + 1) \qquad (5.9)$$

The modified temperature ratio for thermal transpiration, T_r', is

$$\begin{aligned} T_r' &= T_2'/T_1' \\ &= \frac{(y+2)T_2 + yT_1}{(y+2)T_1 + yT_2} \\ &= \frac{(y+2)T_r + y}{(y+2) + yT_r} \\ &= \frac{yT_r + y + 2}{yT_r + y + 2} + \frac{2(T_r - 1)}{y(T_r + 1) + 2} \end{aligned}$$

or, $$T_r' = 1 + \frac{2(T_r - 1)}{y(T_r + 1) + 2}$$

Substituting for y,

$$T_r' = 1 + \frac{2(T_r - 1)}{\{2a + \alpha(1 - a)\}\{T_r + 1\} + 2} \qquad (5.10)$$

Equation (5.10) is plotted for various values of T_r, a, and α in figure (5.3). Three representative values of both α and a are chosen. Also, a line representing the ideal situation (one small orifice, no accommodation) is shown. The trend shown by the curves is in agreement with expectations. At lower values of a the modified temperature ratios are closer to the ideal limit than those for higher values, indicating that to preserve the applied temperature ratio, the interference in the reservoirs from the transpiring molecules should be minimized. These curves also indicate that the modified temperature ratios are higher for low accommodation coefficients. For $\alpha = 0$ and $a = 0.05$, the value of T_r' is only slightly smaller than the corresponding value of T_r.

In choosing an appropriate value of α, it should be noted that very little reliable data is available on accommodation coefficients. Some experimentors have measured values anywhere from near zero to near unity, depending on the nature and state of the gas, wall conditions, measurement technique etc. For this application, it is certainly desirable if possible to choose the diaphragm material such that it has very low energy accommodation. As regards a, the choice also depends on several other factors. Orifice size will be determined by pumping pressure in a stage. Construction difficulties may also impose limitations on the size as well as the number of orifices possible.

The above considerations suggest that $0.1 \leqslant a \leqslant 0.2$.

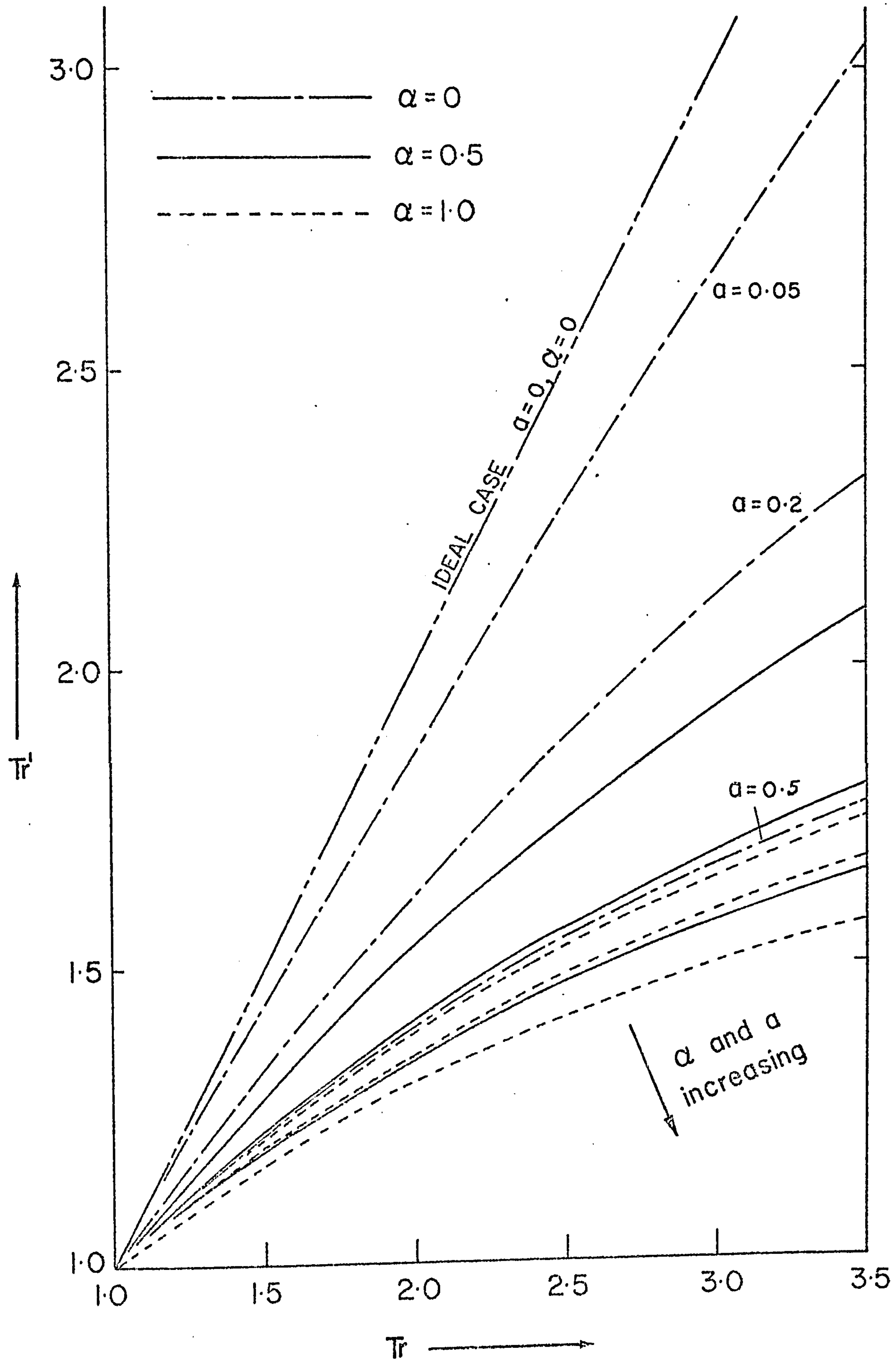


FIG. 5.3 EFFECT OF TRANSPIRATION AREA AND ENERGY ACCOMMODATION ON TEMPERATURE RATIO

Selecting $a = 0.2$, the maximum allowable number of passages may be calculated from,

$$x_{\max} = \frac{\text{Total transpiration area}}{\text{Transpiration area per passage}}$$

$$= \frac{\left(\frac{\pi}{4}D^2\right)a}{\frac{\pi}{4}d^2}$$

$$= \left(\frac{D}{d}\right)^2 0.2$$

5.5 Relaxation of flow conditions

It was observed in section 4.1 that extremely large-sized reservoirs on one hand, and extremely small-diameter orifices on the other, are required if conditions of continuum and free-molecule flows, as dictated by equations (4.7) and (4.6) respectively, are to be maintained strictly. One method of circumventing these difficulties lies in 'relaxing' the flow-condition requirements.

Since the transpiration effect is maximum for free-molecule flow and nil for continuum flow, it would be expected that part of the effect may be realized for intermediate flow regimes. Indeed, Knudsen(4) attempted to formulate a unified analysis for the entire range of flow. His, and subsequent attempts have resulted only in semi-empirical formulae which are restricted in applicability. A generalized theoretical analysis, especially for the transition regime, is lacking till now. It is apparent, however, from the experimental measurements on thermal transpiration by several researchers(12,25, 31, 32, 33) that the transpiration effect is mainly confined to the range $0.1 \leqslant K_{n_c} \leqslant 1.0$. Specifically, the percentages of effect realized (in terms of ratio p_c/p_h) at these two values of K_n in two fairly recent researches are given in the table below.

REF. NO.	TEMPERA RATIO. T_r	THEO. LIMIT $\frac{p_c}{p_h} = \sqrt{\frac{T_c}{T_h}}$	% EFFECT REALIZED AT		REMARKS
			$K_{n_c} \simeq 1{\cdot}0$	$K_{n_h} \simeq 0{\cdot}1$	
(33)	3.0	0.576	69.2	3.51	Long tube, Air
(26)	3.81	0.512	97.3	1.52	Orifice, Helium

It is seen for example, that for $K_{n_c} = 1.0$, about 70% of the theoretical orifice limit of thermal transpiration effect is realized, and for $K_{n_h} = 0.1$, about 3.5% is realized (for $T_r = 3.0$). In the proposed pump if the former condition is maintained in the tubes and the latter in the reservoirs, a net effect of almost 70% of the orifice limit may still be obtained. At the same time, the reservoir dimension may be reduced 10 times and the orifice dimensions may be increased 5 times. This permits a substancial reduction in the overall size of the pump, though at the cost of pumping speed (smaller than before) and ultimate pressure (higher than before).

It would be desirable to predict the pumping speed curves for the above case (i.e. transition flow). This, unfortunately, cannot be done. As was pointed out in chapter I, molecule behaviour in the transition regime is not fully understood. However, if the experimental results be any guide, a pressure ratio equal to 70% of the theoretical limit may be expected. The pumping speed may start at 70% of the theoretical maximum, falling off to zero at the ultimate pressure.

Transpiration phenomenon in the slip flow regime is much better understood(1,2). Using the concept of 'slip distance', Kennard(1) has shown that for a mean pressure of 0.5 torr, a temperature difference of 100°C can cause a pressure difference of about 5% of mean pressure. Williams(30) has analysed the problem essentially in the same manner but with a slightly different technique and compared his results with the experimental results of Arny & Bailey(32,33), Hobson(25) etc. However, the magnitude of the transpiration effect in the slip flow regime is quite small from the view point of its usage for pumping.

Some or all of the above factors should be incorporated in the design criteria for a practical thermal transpiration pumping device in order to achieve optimum pumping speed and pressure ratio within a reasonable size.

CHAPTER VI

CONSIDERATIONS AND SUGGESTIONS FOR A PRACTICAL DEVICE

This chapter will attempt to outline the considerations involved in designing a practical thermal transpiration pumping device by synthesising the information contained in the previous chapters. Design concepts, based on this synthesis, will be discussed and suggestions for further work will be pointed out.

The feasibility and desirability of a thermal transpiration pumping device have been established in chapters II & III. The tentative design in chapter IV, however, revealed a major problem - the problem of its size - especially when attempting any significant pressure reduction. In addition, the expected pumping speeds tend to be small in comparison to those of conventional pumping devices. Effects of the deviations from ideal-case conditions were discussed in chapter V, and they suggested a promise of improvement in terms of size reduction at the sacrifice of some performance. Incorporating these deviations to a desirable, allowable or unavoidable extent will depend on the specific pumping requirements (such as the maximum pumping speed, maximum pressure ratio, forepressure etc.). One thing, however, is certain - the constraint on overall physical dimensions of the device. In the following design it is assumed that the overall dimensions cannot exceed 100 cm x 30 cm dia and the geometrical configuration is as shown in fig. 2.7.

6.1 Considerations

6.1.1 Ideal operation : Under the restrictions imposed on minimum orifice size and maximum overall dimensions, not even one-stage of ideal operation is possible within these geometrical constraints. If orifices of smaller diameter (say, 0.01 cm) are realizable, the forepressure may be increased (to 10^{-1} torr) and consequently, the reservoir dimensions can be decreased (by an order of magnitude). In that case, a two-stage ideal operation is possible. The maximum pumping speed is nearly 1 cm^3/sec and the pressure ratio is nearly 2.67.

6.1.2 Non-ideal operation : The following considerations evolve with regard to non-ideal operation:-

1. Flow conditions
   - (a) For orifice (or tube): As explained in section (5.5), the condition $K_{n_c} = 1.0$ produces almost 70% of the full transpiration effect, at the same time allowing the use of higher forepressures for a given orifice diameter. This ultimately helps to reduce the overall size of the device and should therefore be adopted.
   - (b) For reservoirs: The best compromise between Knudsen number and reservoir dimensions is at $K_{n_h} = 0.1$, for which losses due to transpiration in the reverse direction inside

the reservoirs is only about 4%, but the requirement of reservoir size is reduced by an order of magnitude.

2. Orifice diameter & operating forepressure

The considerations are similar to those of ideal operation; that is, if the minimum orifice diameter is 0.01 cm (rather than 0.1 cm), a reduction of reservoir dimensions by an order of magnitude will result.

3. Orifice or tube

If tubes are used, the ratio L/d should be as small as possible for greater pumping speed. Since d itself must be small, it may not be possible to keep the ratio L/d smaller than 1. This will reduce the pumping speed to about 50% of the ideal case, but the pressure ratio will not be affected.

4. Number of orifices per stage

Following the scheme suggested in section 5.4, the maximum number of orifices is limited to,

$$x_{max} = 0.2\ (D/d)^2$$

where

D = Diameter of the diaphragm containing the orifices.

d = Diameter of the orifices.

Incorporating the above considerations, let $d = 0.01$ cm, and $p_0 = 10^{-1}$ torr, so that $K_{n_c} \gg 5$ in the passages. Also let $L/d = 1$ and $K_{n_h} \ll 0.1$. Then, referring to dimensions of the reservoirs calculated in chapter IV, all may be reduced by two orders of magnitude, so that 6-stage operation is allowable within the overall size limits ($D_6 \simeq 30$ cm, the maximum allowable). The overall length of this pump would be approximately 75 cm. Since the transpiration is almost ideal (except a slight back flow in the reservoirs), the pressure ratio may be expected to be around 20 so that the ultimate pressure may be about 5×10^{-3} torr. The dimensionless pumping speed curve is shown in fig. 3.1a, from which,

$$(S_p^*)_{max} = 0.154651$$

Since, $A = (0.01)^2/4 \ \text{cm}^2$

$R = 2152$ torr-cm^3/mol-$^\circ$K

$T_c = 300^\circ$K

the maximum pumping speed per passage may be given as,

$$(S_p)_{max} = 0.00975946 \ \text{cm}^3/\text{sec}$$

The maximum number of passages, as discussed above, is given by,

$$\varkappa_{max} = 1.767 \times 10^6$$

Correspondingly, the maximum pumping speed of the pump is

$$S_{p\ max} = 0.00975946 \times 1.767 \times 10^6 = 17.2 \text{ litres/sec.}$$

The pumping speed will be reduced if tubes, rather than orifices, are used. For L/d = 1, the reduction in flow is approximately 50%, so that the expected maximum pumping speed of the pump would be,

$$(S_{p\ max})_{L/d\ =1.0} \simeq 8.6 \text{ litres/sec.}$$

The above design promises to give reasonably good pumping speed and pressure ratio. The overall dimensions of the pump are also within the specified limits. However, the main design problem has now shifted from that of extremely large pump size to that of an extremely large number of passages required for a good pumping speed. A suggestion is given in the next section which may alleviate this problem.

6.2 Suggestions for construction

The suggestions given below concern the two most important problems relating to the conceptual design of the proposed thermal transpiration pumping device. One was shown to be , in the last section, the excessively large number of small-diameter passages, and the other is the problem of heating and cooling the gas close to the diaphragm.

6.2.1 Use of a narrow slit : For a reasonably good pumping speed, the transpiration area must be large. For a fixed overall-size of the pump, this may be achieved by either increasing the number of passages or by increasing the transpiration area per passage. The latter is fixed by the operating pressure which, in turn, is fixed from overall size considerations. Any reasonable number of tubes or orifices is unable to provide even moderate pumping speeds. It is therefore suggested that a narrow slit, rather than orifices or tubes, should be used as the transpiration element. The slit-width will serve as the characteristic dimension instead of the diameter of the orifice. Even a very narrow slit may provide large transpiration area, especially when situated at a large radius. Of course, several slits may be used to increase the transpiration area still further. Fig. 6.1 illustrates the idea.

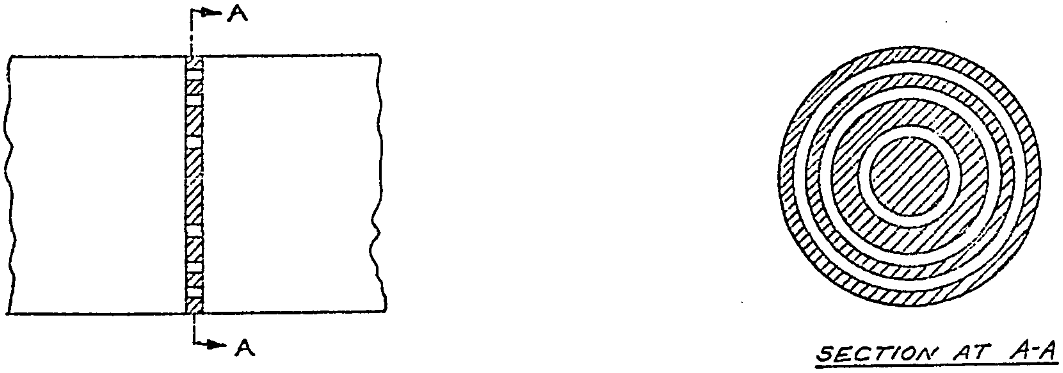
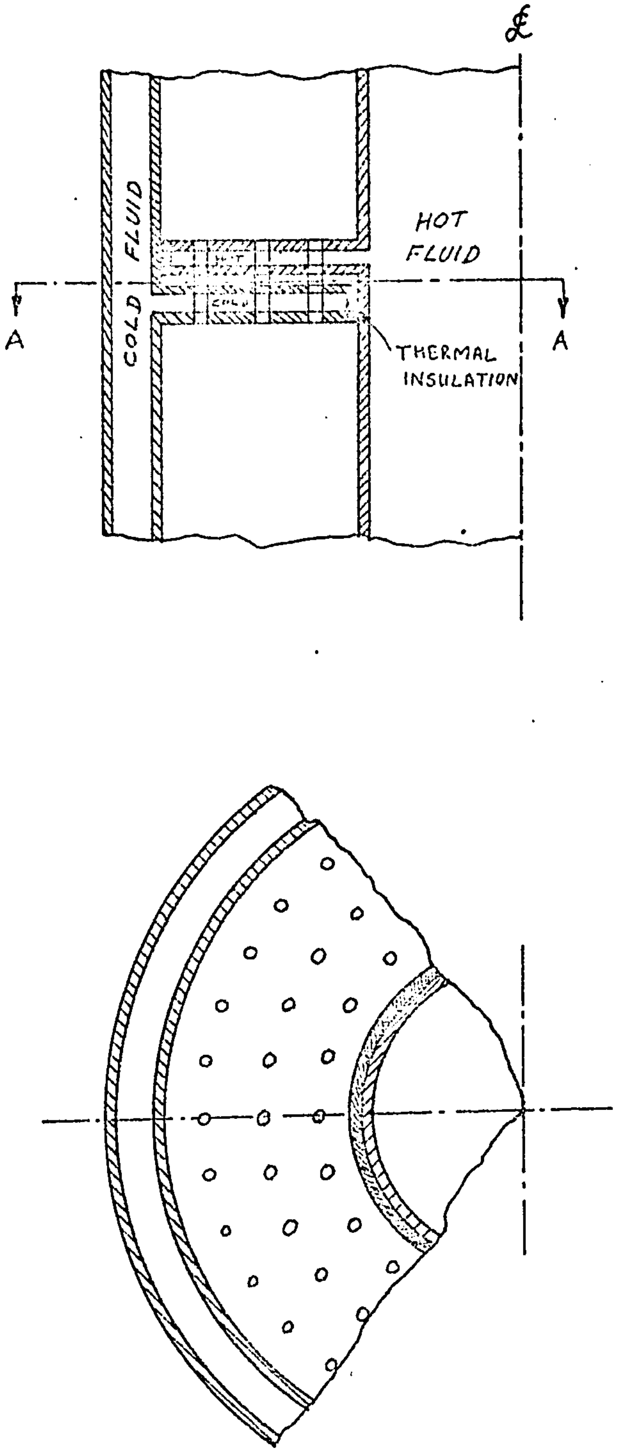


Fig. 6.1 Use of narrow slits for transpiration

In deriving thermal transpiration relations for finite length slits the flow through them at large radii may perhaps be considered as that between two parallel plates. The problem may be more complex for slits situated at smaller radii.

6.2.2 <u>Method of heating and cooling the gas</u> : The following methods are suggested:-

(a) Liquid circulation method A conceptual design is shown in figure 6.2. The heat transfer



PARTIAL SECTION AT A-A

FIG. 6.2 FLUID CIRCULATION METHOD FOR HEATING GAS characteristic of the containing material is required to be good in order to heat up or cool down the molecules as they enter the passage. An obvious disadvantage of such an intricate design is that the passage diameter cannot be made very small.

(b) Hot filament method By placing an obstruction, maintained at a desired temperature, in the path of the gas in the reservoir, the molecules upon striking it tend to acquire its temperature. Such heated obstacles may consist of baffles, wire gauze or the like.

(c) Electrodeless electrothermal heating Electrodeless discharges may provide an interesting method for heating the gas in a thermal transpiration pump due to the fact that the heating apparatus is then entirely outside the pump body.
In the kilocycle range of power supply frequency, the discharge is usually established by electrodes or coils physically insulated from the gas, but in close proximity to it to maximize electromagnetic coupling. At higher frequencies such as in the microwave range, 10^9 to 10^{11} cps, a discharge may be excited by placing the gas at the focus of a horn antenna. If an intense laser source is available, the same interaction

can be obtained even at optical frequencies(35). There are two problems associated with this method. One is that the gas expelled by the pump will be ionized which may not be desirable in certain applications. The other problem is that these discharges impart energy principally to the light-weight electrons which, in turn, must collide with the much heavier ions and molecules in order to transfer the energy. This energy transfer is inefficient due to the large difference in the masses of electrons and the heavy particles. Judicious use of magnetic fields can enhance the heating process by confining the energetic electrons to the heating zone for longer periods.

Of course , the suggestions given above are merely conjectural and require careful investigation before being applied to an actual device.

CHAPTER VII

CONCLUSIONS

The conclusions drawn from this study of thermal transpiration pumping may be summarized as follows:-

(a) Fundamental considerations

1. Except at a pressure ratio of unity, the non-dimensionalized pumping speed increases with temperature ratio as well as the number of stages.

2. The maximum non-dimensionalized pumping speed (occuring at $p_N = p_o$) is independent of the number of stages but increases with increased temperature ratio.

3. The pumping speed decreases as the pressure ratio decreases. When the pumping speed falls to zero, the pressure ratio becomes the ultimate pressure ratio (p_u/p_o).

4. The pumping speed is directly proportional to the area of transpiration.

5. The pumping speed, being inversely proportional to the square-root of the molecular weight of a gas, is greater for lighter gases like hydrogen and helium.

6. The thermal transpiration pump principle can be applied for operation at any pressure and to produce any ultimate pressure.

(b) Practical considerations

1. The thermal transpiration pump is best suited as a circulation or displacement pump in which only single-stage operation is required and the pumping speed is maximum.

2. The relative increase in the pumping speed is maximum for temperature ratios near unity. About 50% of the theoretical limit is obtainable at $T_r = 4$ and almost 70% at $T_r = 10$. For this reason, it may not be profitable to provide temperature ratios greater than 10.

3. The pressure ratio achievable in a single stage is limited by the temperature ratio that may be applied. Temperature ratios cannot be made too large in practice. For the temperature ratio quoted in (2) above, the pressure ratio per stage is restricted to about 3.

4. The increase in pump dimensions required is almost inversely proportional to the decrease in operating pressures. As a result, the system tends to be excessively large when operating at forepressures lower than about 10^{-2} torr.

5. Although several stages may be coupled in tandem (multistaging) to produce greater overall pressure ratios, the problem of large size mentioned in (4)

above is correspondingly greater.

6. Substantial reduction in pump-size requirements may be obtained by operating the pump at non-ideal flow conditions (non-free-molecular in orifices and non-continuum in reservoirs), though it may also mean some sacrifice of pump performance in terms of its pumping speed and pressure ratio.

7. For pump operation at ideal flow conditions, which allows only a single transpiration passage, the pumping speeds are extremely small.

8. The transpiration area, and therefore the pumping speed, may be increased by providing a number of passages instead of just one. This, however, introduces a loss in temperature ratio. An approximate analysis shows that this loss is not significant for $a < 0.01$ where a is the ratio of the transpiration area to the total diaphragm area.

9. The diaphragm temperature may be different from the temperature of the gas adjacent to it, and for this reason, energy accommodation at the diaphragm becomes important in determining the 'effective temperature ratio'. For better preservation of the applied temperature ratio this energy accommodation should be minimized by selecting a suitable material.

10. An orifice is always preferrable to a tube as the

transpiration element. However, if the use of a tube (or a similar finite length element such as a slit) becomes inevitable, the ratio L/d should be kept as small as possible.

11. If using a finite length transpiration element (like tube or slit), the surface of the passage should be as smooth as possible. This is to produce maximum specular reflection of the impinging molecules on this surface. Specular reflection which conserves the tangential component of momentum, is believed to ultimately increase the transpiration effect above that produce by simple diffuse reflection theory.

The development of the thermal transpiration pump is highly desirable in view of its potential to produce absolutely clean vacua and its ability to deal with light and inert gases more effectively than any other method. The final conclusion may be summed up as a suggestion to design in detail, construct and test a 6-stage thermal transpiration pump based on information contained in this thesis.

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APPENDIX A

TERMINOLOGY ASSOCIATED WITH VACUUM PUMPING

The glossary of terms given below is compiled from reference (36) in alphabatical order. Terms marked (*) are recommended, and those marked (#), though in current usage, are condemned by the American Vacuum Society.

ACCOMMODATION COEFFICIENT* - The ratio of the energy actually transferred between impinging gas molecules and a surface and the energy which would be theoretically transferred if the impinging molecules reached complete thermal equilibrium with the surface.*

BACK STREAMING* - The direct flight of vapor molecules by scattering from the hot vapor jet or evaporation from hot nozzle parts in the direction of the mouth or intake port of a vapor pump.*

CONDUCTANCE* - The throughput under steady state conservative conditions divided by the measured difference in pressure between two specified cross sections inside a pumping system.*

FOREPRESSURE* - The total pressure on the outlet side of a pump measured near the outlet port. Sometimes called the <u>back pressure</u># or <u>backing pressure</u>. Also known as <u>outlet pressure</u>, <u>exhaust pressure</u>, or <u>discharge pressure</u>.

FORE PUMP* - The pump which produces the necessary fore vacuum for a pump which is incapable of discharging gases at atmospheric pressure. Sometimes called the <u>backing pump</u>.

FREE AIR DISPLACEMENT* - The volume of air passed per unit time through a mechanical pump when the pressure on the intake and exhaust sides is equal to atmospheric pressure. Also called <u>free air capacity</u>.

GAS - Gas is defined as the state of matter in which the molecules are free to occupy any space within an enclosure*. In vacuum technology the word gas has been loosely applied to the non-condensable gas and vapor within a vacuum system.

INLET PRESSURE* - In connection with performance data on pumps, when not otherwise specified, <u>inlet pressure</u> refers to the "total static pressure" measured in a standard testing chamber by a vacuum gauge located near the inlet port. Sometimes referred to as <u>intake pressure</u>, <u>fine pressure</u>#, <u>head pressure</u>.

LEAK RATE* - The quantity of gas in pressure-volume units at room temperature flowing into the system or through the pump from an external source in unit time.*

LOAD* - The quantity of gas (not including pump fluid vapor) in mass units flowing across the inlet port of a pump in unit time.* Sometimes called <u>capacity</u>#. Typical units are pounds per hour or grams per hour.

MEAN FREE PATH (of any particle)* - The average distance that a particle travels between successive collisions with the other particles of an ensemble.* In vacuum technology the ensemble of particles of interest comprises only the molecules in the gas phase.

MOLECULAR EFFUSION* - The passage of gas through a single opening in a plane wall of negligible thickness where the largest dimension of the hole is smaller than the mean free path.*

PERMEABILITY COEFFICIENT* - The steady-state rate of flow of gas through unit area and thickness of a solid barrier per unit pressure differential at a given temperature. Usually expressed in cubic centimeters (S.T.P.) per second, per square centimeter of cross-section, per millimeter of thickness, per centimeter of mercury pressure drop across the barrier.* Also called simply permeability.

PERMEATION - As applied to gas flow through solids permeation is the passage of gas into, through, and out of a solid barrier having no holes large enough to permit more than a small fraction of the gas to pass through any one hole. The process always involves diffusion through the solid and may involve various surface phenomena, such as sorption, dissociation, migration, and desorption of the gas molecules.

PUMPING SPEED* - The speed of a pump for a given gas is the ratio of the throughput of that gas to the partial pressure of that gas at a specified point near the mouth (or inlet port) of a pump.*

THROUGHPUT* - The quantity of gas in pressure-volume units at a specified temperature flowing per unit time across a specified open cross section of a pump or pipe line.*

TORR* - Suggested international standard term to replace the English term mm. of Hg. The Torr is defined as 1/760 of a standard atmosphere or 1,013,250/760 dynes per square cm. This is equivalent to defining the Torr as 1333.22 microbars and differs by only one part in seven million from the international Standard millimeter of mercury.

ULTIMATE PRESSURE* - The limiting pressure approached in the vacuum system after sufficient pumping time to establish that further reductions in pressure will be negligible. Sometimes called the ultimate vacuum#. The terms blank-off pressure or base pressure are also sometimes used in referring to a pump under test.

VACUUM - In vacuum technology the term vacuum refers to a given space filled with gas at pressures below atmospheric pressure. The following classification of degrees of vacuum has been proposed:

Condition	Pressure range
(a) Low vacuum:	760 to 25 Torr
(b) Medium vacuum:	25 to 10^{-3} Torr
(c) High vacuum:	10^{-3} to 10^{-6} Torr
(d) Very high vacuum:	10^{-6} to 10^{-9} Torr
(e) Ultra-high vacuum:	10^{-9} Torr and below*

APPENDIX B

NON-DIMENSIONALIZATION OF PUMPING SPEED EQUATION

In chapter II the equation for pumping speed is shown to be

$$S_p = \frac{\mathcal{A} R T_c}{\sqrt{2\pi R}} \left\{ \alpha_N - \beta_N \left(\frac{p_o}{p_N} \right) \right\} \qquad (2.25)$$

where,

and

$$\alpha_N = \frac{a^N}{\sum_{i=0}^{N-1} a^{N-1-i} b^i} , \qquad \beta_N = \frac{b^N}{\sum_{i=0}^{N-1} a^{N-1-i} b^i}$$

Designating the fundamental quantities as,

L → length

θ → time

T → temperature

M → mass

the independent variables involved in (2.25) have the following dimensions,

$S_p \rightarrow L^3\theta^{-1}$
$\mathcal{A} \rightarrow L^2$

$R \rightarrow L^2\theta^{-2}T^{-1}$

$T_c \rightarrow T$

$p_o \rightarrow ML^{-1}\theta^{-2}$

$p_N \rightarrow ML^{-1}\theta^{-2}$

Variables α_N and β_N are dependent on a, b and N. Since, a $(=T_c^{-\frac{1}{2}})$ has already been included above, only b $(=T_h^{-\frac{1}{2}})$ and N are the additional independent variables,

$T_h \rightarrow T$

and, $N \rightarrow N$

Thus, there are 8 independent variables. Since we have assumed 4 fundamental quantities, the number of dimensionless groups to be formed is 4 (see reference 21). Further, S_p and p_N must appear in different groups in order to obtain $S_p \sim p_N$ curves. With this in view, the following four groups are formed,

$$p_N/p_o = p^*$$

$$T_h/T_c = T_r$$

$$N = N$$

and, $$S_p/(\mathcal{A}(RT_c)^{\frac{1}{2}}) = S_p^*$$

Equation 2.25 can be rewritten as,

$$\frac{S_p}{\mathcal{A}\sqrt{RT_c}} = \frac{\sqrt{T_c}}{\sqrt{2\pi}}\left[\frac{a^N}{\sum_{i=0}^{N-1} a^{N-1-i} b^i} - \frac{b^N}{\sum_{i=0}^{N-1} a^{N-1-i} b^i} \cdot \frac{1}{p_N/p_o}\right]$$

Dividing numerator and denominator of r.h.s. by a^{N-1},

$$S_p^* = \frac{\sqrt{T_c}}{\sqrt{2\pi}}\left[\frac{a^{N-(N-1)}}{\sum_{i=0}^{N-1} a^{N-1-i-(N-1)} b^i} - \frac{b^N a^{-(N-1)}}{\sum_{i=0}^{N-1} a^{N-1-i-(N-1)} b^i} \cdot \frac{1}{p^*}\right]$$

$$= \frac{\sqrt{T_c}}{\sqrt{2\pi}}\left[\frac{a}{\sum_{i=0}^{N-1}\left(\frac{b}{a}\right)^i} - \frac{b^N \cdot a^{-N} \cdot a}{\sum_{i=0}^{N-1}\left(\frac{b}{a}\right)^i} \frac{1}{p^*}\right]$$

$$= \frac{\sqrt{T_c}}{\sqrt{2\pi}}\left[\frac{a}{\sum_{i=0}^{N-1}\left(\frac{b}{a}\right)^i} - \frac{a}{\sum_{i=0}^{N-1}\left(\frac{b}{a}\right)^i}\left(\frac{b}{a}\right)^N \frac{1}{p^*}\right]$$

Since, $a = 1/(T_c)^{\frac{1}{2}}$, and $b/a = (T_h/T_c)^{-\frac{1}{2}} = T_r^{-\frac{1}{2}}$, we get,

$$S_p^* = \frac{1}{\sqrt{2\pi}}\left[\frac{1}{\sum_{i=0}^{N-1} T_r^{-\frac{i}{2}}} - \frac{1}{\sum_{i=0}^{N-1} T^{-\frac{i}{2}}}\left(T_r\right)^{-\frac{N}{2}}\frac{1}{p^*}\right]$$

or,

$$S_p^* = \frac{1}{\sqrt{2\pi}}\left[\frac{1}{\sum_{i=0}^{N-1} T_r^{-\frac{i}{2}}} - \frac{1}{\sum_{i=0}^{N-1} T_r^{\frac{N-i}{2}}}\;\frac{1}{p^*}\right]$$

APPENDIX C

BRIEF DESCRIPTION OF VARIOUS TYPES OF PUMPS

The following description corresponds to the pump types listed in table 3.1 and is compiled from references 13, 14, 16, 19 and 20.

1. Water jet pump

A high pressure jet entrains gas entering through a side arm and ejects it to atmosphere.

Characteristics:

(i) No moving parts.

(ii) Can be used to pump certain corrosive vapors and dusts.

(iii) Can back mercury ejector pumps to give a system free from organic contamination,

(iv) Suitable for small applications only.

2. Water ring pump

Consists of an eccentrically mounted multi-vane impeller, which sets up a water ring against the stator wall. Clearance between the impeller base circle and the water ring provides the compression space.

Characteristics:

(i) Back leakage of gas through clearance.

(ii) High power consumption and prone to frost damage.

(iii) Somewhat lower ultimate pressures possible by using cooled circulated oil. instead of water.

3. Steam ejector

Dry steam at 100-200 psi produces a high velocity jet which entrains gas from the system.

Characteristics:

(i) Design is inflexible; only meets specified steady conditions.

(ii) Compression ratio of about 7 per stage; higher compression ratios and lower ultimate pressures possible by coupling in series.

(iii) Particular problem in operation is suck-back due to several factors like nozzle corrosion, drop in steam pressure etc.

4. Oil-sealed rotary pump

There are three basic designs in all of which the oil functions as a lubricant as well as a sealant between the moving parts of the pump :-

(a) The two-bladed pump where gas is trapped between the blades and swept out through the exhaust valve.

(b) The single-bladed pump where the single blade is slotted into the stator, and the gas trapped between the blade and rotor is swept out through the exhaust valve.

(c) The rotary piston pump which is similar to the single-bladed pump, but where the single blade is part of a sleeve which fits around the rotor.

The blade is hollow and acts as an inlet valve closing off the pump from the system when the rotor is at top centre.

Characteristics:

(i) The ultimate pressures attainable are limited by leakage between the high and low pressure sides of the pump (due mainly to carry-over of gases and vapors dissolved in the oil which flash off when exposed to the low inlet pressures) and decomposition of oil exposed to high temperature spots generated by friction.

(ii) Gas ballast may be used in reducing the extent of water vapor contamination of the oil. However, then the attainable ultimate pressure rises by nearly an order of magnitude or more.

(iii) There are severe restrictions on the nature and quality of oil to be used.

(iv) Oil suck-back into the system can occur if the pump is stopped under vacuum.

5. Mechanical booster

Consists of two dumb-bell shaped impellers which are synchronized by external gears and rotate in opposite directions wittin a stator.

Characteristics:

(i) Impellers run dry - no oil contamination.

(ii) Large back leakage occurs through clearances.

(iii) Mainly useful in boosting up the compression ratios of other pumps.

(iv) Overheating may cause loss of clearances resulting in seizure.

(v) The pump is sensitive to the contours of impellers and stator and to back-lash in gears and timing mechanism.

6. <u>Molecular pump</u>

A rotor, revolving at very high angular speeds within a stator, imparts momentum to gas molecules, and therefore, the molecules move along the small rotor-to-stator clearances. There are two types :-

(a) <u>Molecular drag pumps</u> have a finned rotor rotating in a slotted stator, or a plain disc rotor rotating between two stator plates with spiral grooves. The clearance is required to be very small.

(b) <u>Turbo-molecular pumps</u> are made with radial slots in both rotor and stator fins. They are preferred design due to large clearances.

Characteristics:

(i) Extremely high rotational speeds are necessary for reasonable compression ratio per stage and pumping speed.

(ii) Light gases (He, H_2 etc.) are almost impossible to expell completely because of back diffusion.

(iii) Vapor contamination is almost nil.

(iv) Pumping speed is dependent upon the gas viscosity and therefore, is maximum for hydrogen. However, this very factor(viscosity) is also responsible for back-diffusion of a gas. Thus, despite its high pumping speed, the ultimate pressure of hydrogen is limited.

7. Vapor diffusion pumps

High velocity vapor streams emerge from the jets to impinge and condense on the pump body, a portion of the gas being trapped, compressed and transferred to the next stage until removed by the forepump.

Characteristics:

(i) Operation at too high an inlet or fore pressure causes the jets to break up and allows vapor to migrate freely towards the vacuum system.

(ii) If an oil-sealed rotary pump is used for backing, it may be necessary to provide backing line traps (liquid N_2) to suppress contamination.

(iii) Forepressure is generally required to be not more than 10^{-2} torr; less for lighter gases like hydrogen and helium.

(iv) Pumping speed is dependent on the molecular weight of the gas; greater for lighter gases.

(v) Variation of heater input can cause large variations in pumping speeds.

(vi) Pumping speed remains constant well below the ultimate pressure limitation imposed by the pump fluid. For instance, the air speed of an untrapped mercury pump remains constant at least to 10^{-9} torr (partial air pressure) even through the total ultimate pressure is limited to 10^{-3} torr by mercury vapor.

(vii) Backstreaming of vapor is a serious problem unless a cooled baffle is used.

8. Oil vapor booster pump

Similar to the vapor diffusion pump; differs primarily with regard to boiler pressure which is normally an order of magnitude higher.

Characteristics:

(i) Used for high-speed duties in the pressure range 10^{-1} to 10^{-3} torr, where rotary pumps are at their limit and diffusion pumps unstable.

(ii) Particularly suitable for dirty and mainly hydrogen loads.

9. Sputter-ion pump

Pumping mechanism consists of three modes :-

(a) Active gases N_2, O_2, H_2, H_2O, CO, CO_2, NH_3 etc - Gettering.

(b) Inert gases He, Ne, Xe, A, Kr etc. - ion burial and covering.

(c) Light gases H_2, He - ion burial and diffusion into cathode.

Characteristics:

- (i) Acts as its own pressure gauge and leak detector.
- (ii) Although no organic contamination is caused by the pump itself, the forepump used may still contaminate the vacua.
- (iii) Cannot sustain high throughputs.
- (iv) Starting period relatively long.
- (v) Stray magnetic fields or unconfined discharges may be present.
- (vi) Pumping characteristics are selective (to certain gas and other conditions - e.g. argon instability).
- (vii) The outcoming gas from the pump is ionized which may not be desirable for some applications.

10. Radial field pump

The pumping mechanism is similar to that of the sputter-ion pump, but the supply of electrons is independent of pressure and comes from a hot filament. Consequently, start-up conditions are more critical than those encountered with sputter-ion pumps, but it eliminates the possibility of stray fields.

11. Titanium sublimation pump

Titanium is evaporated from a tungsten filament overwound with titanium wire onto the vacuum chamber wall. Active gases are pumped by chemical combination, but there is no pumping speed for inert gas or saturated hydrocarbons so that sublimation pumps are always used in conjunction with diffusion or sputter-ion pumps.

Characteristics:

(i) One gram of sublimed titanium (M = 48) is capable of removing 1/96 mole, or 130 torr-litres of a diatomic gas. Some typical pumping speeds are shown in table C-1 below.(19)

TABLE C-1

Pumping speed 1 $s^{-1}cm^{-2}$ of titanium film at 293°K and 77°K for various gases.

Gas:	H_2	N_2	O_2	CO	CO_2	H_2O	CH_4	A	He
293°K film	2.3	2.3	1.55	4.6	3.9	3.1	nil	nil	nil
77°K film	6.2	6.2	-	11	9.3	14	nil	nil	nil

(ii) At low pressures the film is deposited faster than it is being consumed allowing deposition to be carried out at intervals (conductance - limited condition of operation).

(iii) Provides high pumping speeds for <u>active gases</u> but no pumping action for inert gases.

(iv) Free from organic contamination - in itself.

(v) The process volume needs to be shielded against titanium.

(vi) The filaments need to be thoroughly degassed.

12. Sorption Pump

Gases are trapped on the surface of a sorbent material cooled to liquid nitrogen temperature and subsequently released when the sorbent material returns to ambient temperature. The sorbent material usually is either activated charcoal or a molecular sieve.

Characteristics:

(i) Ultimate pressure is dominated by the inert gases.

(ii) Water vapor is strongly sorbed and is not released when the sorb returns to room temperature. Baking is therefore necessary.

(iii) Release of desorbed gases may not be perfect.

(iv) Pre-chilling the pump, while permitting a shorter actual pumping cycle, takes a long time to be carried out.

(v) Ideally suited as the roughing pump on Sputter-ion pump.

(vi) Can only be used on a cycle basis.

(vii) Has negligible pumping speed for He, H_2, and Ne.

13. Cryopump

Gases condense on a cryosurface maintained at low temperatures within a vacuum enclosure.

To a first approximation the temperature of the cryosurface determines the ultimate pressure attainable and the refrigerative power determines the pumping speed.

Characteristics:

(i) It is not feasible to pump He by cryopumping since it has the least saturation temperature.

(ii) If N_2 be used as refrigerant (as is usual), a small diffusion or sputter-ion pump is needed to handle He, H_2 and Ne.

(iii) If He be used as refrigerant, all species except He may be pumped; thermal shielding however is important and difficult.

(iv) Refrigeration required is small but expensive.

(v) Cryosorption may also remove He, H_2, Ne to some extent.

APPENDIX D

APPLICABILITY OF THE PERFECT GAS LAW

Perfect gas law

A thermal equation of state of the form $p = p(v,T)$ connects the three thermodynamic variables p, T, and $v = 1/\rho$. Most real gases in thermal equilibrium at moderate pressures and temperatures obey very closely the so-called 'perfect equation of state' or 'perfect gas law'.

$$pv = \mathcal{R}T/M = RT \quad \text{or} \quad p = \rho RT \quad \text{(thermally perfect)}$$

Imperfection at very low temperatures and high pressures is caused by the Van der Waals forces which gradually become more important as the intermolecular distances decrease. Another type of imperfection, in the other extreme, may set in at sufficiently high temperatures and low pressures as a result of dissociation and ionization. These processes increase the number of particles per unit volume. Considering dissociation, if the mixture of dissociated and undissociated particles is assumed to be a mixture of two thermally perfect gases, then Dalton's law of partial pressures applies,

$$pv/RT = 1 + \alpha$$

If ionization also takes place, then for a single, diatomic gas,

$$pv/RT = (1 + \alpha)(1 + x)$$

where α and x are degrees of dissociation and ionization respectively.

Estimate of imperfections in the thermal trans. pump

Under the anticipated conditions of operation of the thermal trans. pump, the intermolecular distances will always be sufficiently large to avoid Van der Waals forces becoming important. This can be seen from the critical data for various gases given on p. 210 of reference (1).

However, the other type of imperfection - that caused by dissociation and/or ionization - may occur in the gas, as both these processes are favoured by low pressures and high temperatures. An expression for α in terms of p and T is given by$^{(34)}$,

$$\alpha = \left[A(p,T) \frac{p}{T^{5/2}} e^{\frac{\theta_d}{T}} + 1 \right]^{-\frac{1}{2}}$$

where θ_d is the characteristic dissociation temperature of the gas.

If the ratio $A(p,T)/T^{5/2}$ is the same for several diatomic gases (and this is generally true), a single plot of α vs. T/θ_d is valid for all of them. Figure D-1 (redrawn from reference (34)) shows such a plot. It is seen that for pressures down to 10^{-5} torr, significant dissociation is not likely to occur for $T/\theta_d \leqslant 0.02$ or, for $T \leqslant 1200^{\circ}$K for oxygen ($\theta_d = 59{,}000^{\circ}$K).

Analysis for ionization follows similar lines. For polyatomic gases, ionization can occur only after appreciable dissociation and may therefore, be ruled out. The ionization

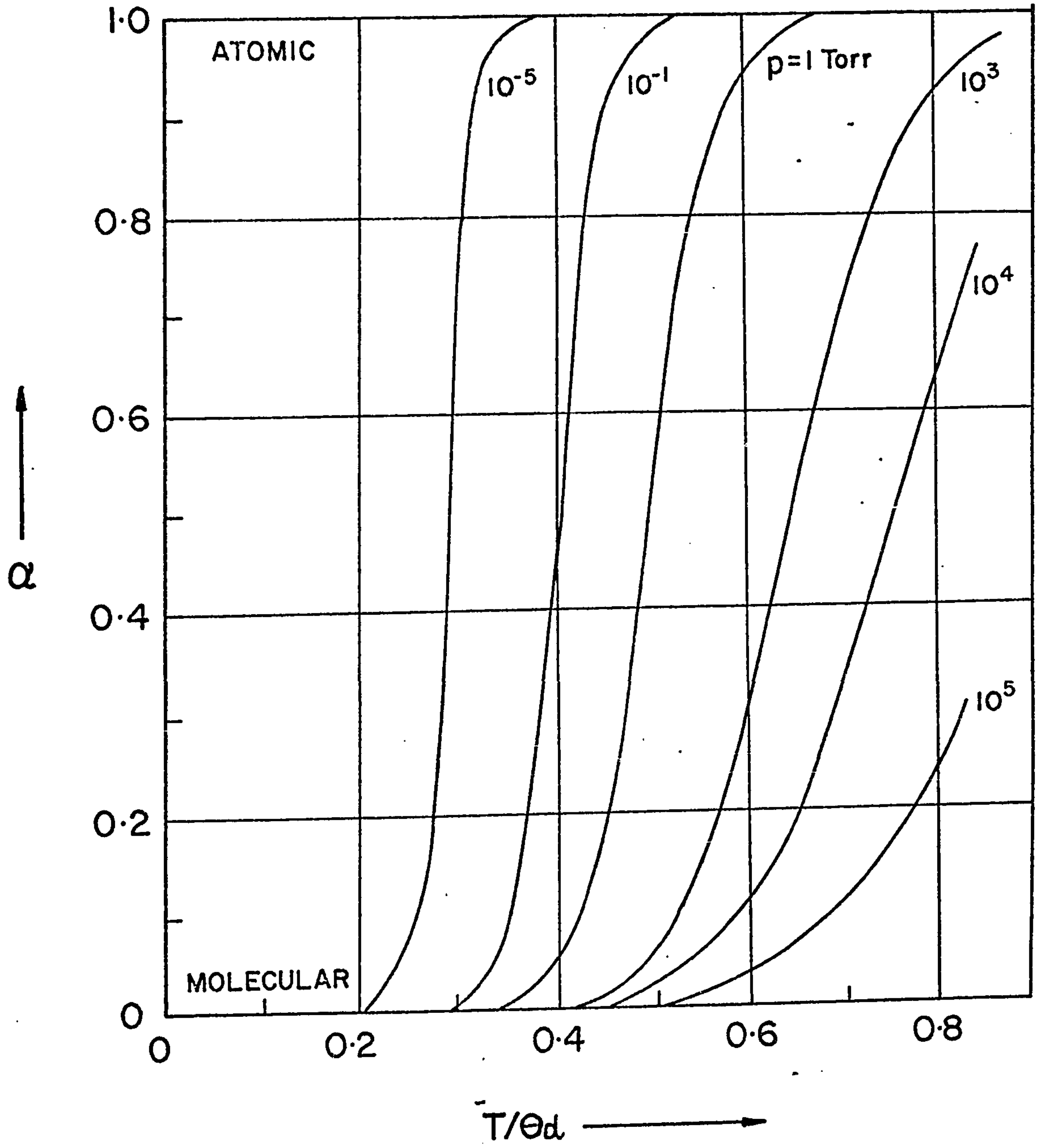

FIG. D-1 VARIATION OF DEGREE OF DISSOCIATION WITH TEMPERATURE AND PRESSURE temperatures for monoatomic gases are much higher than dissociation temperatures for diatomic gases (see reference 34). For example, argon (θ_i= 183,000°K) requires temperatures around 3600°K to ionize at a pressure of 10^{-5} torr.

It may be concluded that the possibility of dissociation or ionization of the gas being pumped is almost negligible. The perfect gas law may, therefore, be applied in this analysis without any modifications for imperfections.

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