Title

RELATIONSHIP OF DIFFUSION-PUMP PERFORMANCE TO THE THERMODYNAMICS OF THE PUMPING FLUID

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Publication Date
1959-11-17

UCRL 8970

UNIVERSITY OF CALIFORNIA

Ernest O. Lawrence Radiation Laboratory

BERKELEY, CALIFORNIA

For pub. Proceedings of Symposium in Vacuum Techniques - Phila. Oct. 7-9-1959.

UNIVERSITY OF CALIFORNIA

Lawrence Radiation Laboratory
Berkeley, California

Contract No. W-7405-eng-48

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ABSTRACT

Theoretical consideration of the thermodynamics of the processes occurring in the pump fluid of a diffusion pump has contributed to the understanding of pump operation. Many of the anomalous results observed by experimenters in this field can be clarified when analyzed from the thermodynamic point of view.

Preliminary experimental data have confirmed the validity of the theoretical reasoning. Application of the fundamental thermodynamic principles to typical diffusion pumps has markedly improved performance with regard to pumping speed, ultimate base pressure, and forepressure tolerance. Detailed experimental results are presented.

Promising possibilities for extension of this work and further improvement in pump design and performance are discussed.

RELATIONSHIP OF DIFFUSION-PUMP PERFORMANCE TO THE THERMODYNAMICS OF THE PUMPING FLUID*

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INTRODUCTION

Much of the research effort in the past few years on high vacuum pumps has been directed toward the perfection of ionization and sorption pumps of various kinds. This effort has met with a degree of success, especially for application on low throughput, baked or very clean vacuum systems. For high-capacity, continuous duty, the diffusion pump continues to be the "work horse" in the high-vacuum field. This paper suggests approaches toward achieving more of the potential performance of the diffusion pump. The following discussion of fundamental thermodynamic principles as applied to diffusion pumps, and the outline of preliminary experimental results illustrate some of the obvious lines of attack.

THEORETICAL BACKGROUND

Figure 1 is typical of the familiar schematic drawings appearing in the vacuum literature to illustrate the functioning of a diffusion pump. This figure is also appropriate here, but a departure from the usual explanation is in order. The boiler filled with pump fluid is heated from below, and the fluid vapor streams upward in the chimney of the nozzle structure. If the nozzles were all closed off and the structure appropriately insulated, it would be possible to approach the equilibrium vapor pressure of the pump fluid in the enclosure, corresponding to the boiler fluid temperature. This situation is never approached in actuality because of the nozzle openings and the loss of heat from the nozzle structure to the cold wall of the surrounding barrel.

The temperature of the jet structure, as a result of cooling by heat transfer to the barrel, has been measured by Bush[1] and Lange[2] on a 32-in. oil pump and a 4-in. oil pump, respectively. The temperatures indicated in Fig. 1 show the distribution obtained in normal operation of a 6-in. oil pump used in the course of this investigation.

*Work performed under the auspices of the U.S. Atomic Energy Commission.

The large temperature differences measured between the boiler and portions of the jet structure are significant from a thermodynamic point of view. Figure 2 illustrates the form of the liquid-vapor portion of a temperature-enthalpy diagram for a pump fluid. Point 1 indicates saturated vapor at boiler temperature, T_1, as exists at the boiler liquid-vapor interface. As the saturated vapor flows up through the cooler stacks, it will decrease in enthalpy and some of it must condense. There will also be some pressure drop and cooling of the vapor up the stack. This partial condensation and cooling process is indicated by the dotted line ending at state point 2, which represents the vapor condition just preceding the nozzle orifice. The wet vapor now contains condensed droplets and will stream out of the nozzle openings as a two-phase mixture. The condensed droplets have little directed velocity and will contribute drag forces on the higher velocity vapor as well as provide nuclei for further condensation of the vapor as it cools on expansion.

The decelerating effect of the condensed droplets on the vapor stream is probably not the most important facet of their presence. More important, the drops accumulate on the lip of the top nozzle cap and reevaporate upward (thus forming an excellent source of backstreaming) or drip off onto the hot top surface of the next lower nozzle and flash off as vapor, thus contributing to the back pressure against which the top vapor jet must work. Most important, their presence is symptomatic of the wide deviation of the vapor in the jet from the dry (saturation) condition.

An obvious physical alteration to avoid the presence of the two-phase, low-energy mixture is the inclusion of heating elements in the jet structure. This would not only prevent the loss of heat from the vapor but provide sufficient additional energy to superheat the vapor. The resulting vapor state just preceding the nozzle orifice would correspond to point 3 in Fig. 2. The dotted line between 1 and 3 represents the superheating process as the vapor flows up the stack. The optimum amount of superheat will be determined by the chemical stability of the vapor and the velocity and mass flow desired in the vapor jet. Certainly sufficient superheat is desirable to ensure dry vapor throughout the subsequent expansion through the nozzle into a supersonic jet.

The preceding argument has been advanced on the assumption that sufficient vapor will be generated by the diffusion-pump boiler to supply the requirements of the nozzles. Further consideration of the thermodynamics of the processes occurring in the boiler fluid indicates that this is generally not the case, and the classical design of diffusion-pump boilers is an inherent limitation in pump performance. Thus, to achieve optimum operation, boiler design must be considered as well as vapor heating.

Dushman has noted that various investigators have found that the vapor-pressure variation with temperature of organic pump fluids is represented very satisfactorily by an equation of the form $\log P = A - \frac{B}{T}$.[3] This relation is a form of the Clausius-Clapeyron equation for change of state. The fact that the pump fluids experimentally satisfy such a relation justifies the application of the basic equation to thermodynamic analysis of conditions in the pump boiler. A more useful form of the equation for this application is

$$\ln \frac{P_2}{P_1} = - \frac{\Delta H_{VAP}}{R} \left(\frac{1}{T_2} - \frac{1}{T_1} \right), \tag{1}$$

where P_2 is the vapor pressure at temperature T_2, P_1 is the vapor pressure at temperature T_1, ΔH_{VAP} is the average heat of vaporization between temperatures T_2 and T_1, and R is the gas constant. A value of the heat of vaporization, ΔH_{VAP}, for a typical pump oil can be calculated from data given in Dushman. Common oils range in value from about 20 kcal/mole to 28 kcal/mole.

For an average value of 25 kcal/mole, Eq. (1) becomes

$$\ln \frac{P_2}{P_1} = - 12,580 \left(\frac{1}{T_2} - \frac{1}{T_1} \right). \tag{2}$$

It is instructive to substitute some values of practical significance into this equation. It is not uncommon practice to vary power input to a diffusion pump in an attempt to optimize performance. In so doing, moderate variations in power produce considerable changes in boiler temperature. As an example, the average boiler temperature of a 6-in. oil pump was increased from 240°F to 247°F as a result of a power increase from 500 to 600 watts. The form of the Clausius-Clapeyron equation indicates that this increase in temperature should correspond to a substantial increase in vapor pressure. The numerical solution for this particular case indicates that the resulting vapor pressure at 600 watts should be 1.55 times as great as the vapor pressure at 500 watts.

Since Langmuir's equation for evaporation rate applies to organic pump fluids, it can be seen that the ratio of evaporation rates, G_2/G_1, at the two temperature conditions should also be

$$\frac{G_2}{G_1} = \frac{P_2}{P_1} \sqrt{\frac{T_1}{T_2}} \approx 1.55. \tag{3}$$

For this latter condition to be true, sufficient additional power would have to be supplied to provide the necessary additional heat of vaporization. If we adopt the conservative assumption that the ratio of useful boiler power to total input power remains the same for the two inputs, the ratio of gross power input would also have to be at least 1.55. The measured value of 1.2 indicates that the evaporation rate and vapor pressure do not increase as predicted by the theory.

The inference to be made is that the additional heat input to the boiler is going into raising the temperature of the oil, rather than increasing the vaporization rate at an essentially constant oil temperature. This indicates that at the higher power input, the boiler does not provide adequate surfaces of the proper geometry to promote the vaporization of the liquid at an increased rate.

Since the generation of vapor in the boiler determines the flow out through the nozzle openings, this departure from theoretical vaporization rates is a very fundamental limitation. To achieve a supersonic jet, sufficient vaporization must occur to provide vapor at sonic velocity in the nozzle throat and to maintain a pressure ratio of greater than 2 to 1 across the nozzle. The vapor jet velocity attained is dependent on the pressure ratio, providing the nozzle exit-section contours do not choke the vapor expansion. For a particular boiler heat input, the evaporation rates of different pump fluids and the resulting stack pressures will be different. Hence, even if the nozzle geometries were calculated on a rational basis for a certain pump fluid, they would not necessarily be a good design for a fluid with different thermodynamic properties. Similarly, performance comparisons of pump fluids run in a pump with a fixed geometry and heat input can not legitimately be used to judge the relative merits of pump fluids. Such data only indicate which pump fluid happens to match the pump geometry and power input best.

The literature on the design of nozzles for diffusion pumps is very sketchy and represents, in general, an entirely empirical approach. Some question might be raised as to the validity of many of the conclusions. In general, the nozzle dimensions have not been related to the thermodynamic properties and vapor state of the pump fluid.

EXPERIMENTAL RESULTS

A series of experiments was run on appropriately modified commercial pumps in order to evaluate the validity of the theoretical reasoning. Figure 3 shows in section the installation of a superheater in the top jet stack of a 6-in., fractionating, oil-diffusion pump. The numbers on the jet structure indicate the location of thermocouples for measurement of stack temperatures.

Figure 4 is a plot of the temperature variation in the jet structure during one complete superheater cooling and heating cycle. The measured pumping speed corresponding to a given temperature condition is plotted adjacent to the temperature curves. The pumping speeds were measured with an oil-burette displacement meter, and the pressures were measured with three untrapped hot-cathode ionization gages. The indicated increase in pumping speed obtained with superheating was reproduced many times. The pumping speed at standard operating conditions is somewhat less than the manufacturer's ratings. This is due to the inherent impedance of the 6-in. glass cross used as a test dome, the use of untrapped ion gages for the pressure measurements, and the usual optimism of vacuum equipment manufacturers.

The fractionating jet configuration, with its concentric stacks, entails certain disadvantages for superheating of the vapor. The outer stacks act as radiation shields, and it is thus difficult to heat the whole jet structure evenly from the inside. For this reason a 6-in. nonfractionating pump was modified to include both a central axial superheater and a top jet cap heater for further, more comprehensive experiments. A series of runs was made with the pump under the following conditions of operation:

(a) At normal operating conditions, i.e., at manufacturer's recommended power input with no superheating

(b) With normal power input and 26 watts to the top jet cap heater

(c) With normal power input and 66 watts to the axial superheater

(d) With normal power input and both top jet cap and axial superheater in operation.

Figure 5 summarizes the results of these runs. The improved performance with superheating is clearly indicated. Another way of illustrating the improvement in pumping speed with superheating is shown in Fig. 6. Here the temperature distribution in the pumpjet structure is plotted against time as the superheater is cycled on and off. A constant leak was admitted to the test system during these cycles, and the system pressure varied as indicated in the figure. The reduction in pressure during superheater "on" time, of course, indicates the effect of increased pumping speed on the constant leak. For purposes of comparison, the preceding runs were made with normal power input into the pump. It was observed, however, that satisfactory pump operation could be obtained with the combination of superheating and a lower power input to the boiler than would ordinarily produce pumping.

It is also significant to note in Fig. 5 that a considerable decrease in system base pressure was obtained in the runs with superheating. To confirm that this was a real effect and not due to system conditioning, several runs were made under the no-leak condition to observe the effects on system base pressure as the superheater was cycled on and off. The base pressure decreased during superheater operation by a factor of from 2.5 to 4, depending on the system conditions when the cycling was started. During a typical experimental run, the test-system base pressure dropped from 6×10^{-6} mm Hg at normal pump operation to less than 2×10^{-6} mm Hg with the superheater on. Turning the superheater off caused the pressure to increase to a value slightly less than the initial pressure. A large number of cycles reduced the pressure change to a minimum and apparently constant factor of about 2.5 for this pump and system. It should be noted that in actuality this series of measurements simply shows the modified pump's ability to produce net pumping speed at a lower pressure in a particular system with certain vacuum characteristics.

A series of runs was also made to see if vapor superheating had any effect on the tolerable forepressure* of the pump. Figure 7 is a plot of pump inlet pressure versus the tolerable forepressure under normal pump operating conditions and with the addition of superheating. Figure 7 is not the customary form of presentation of this type of data but is included to emphasize the point that tolerable forepressure is markedly improved by superheating.

In performing the preceding series of experiments, efforts were made to optimize boiler power input relative to superheating settings. This proved to be largely unsuccessful because of the very small variation of vapor generation with power input, as was noted in the theoretical discussion. In addition, at high power inputs considerable instability in pump performance was observed.

It was hoped that visual observation of the boiler in conjunction with temperature and pressure measurements would provide useful information for improving boiler performance. Accordingly, the pump barrel was modified with a glass section and a window was built into the base of the metal jet structure. In addition, provision was made for measuring the boiler oil-pressure head by means of a sight glass and a cathetometer arrangement.

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*Tolerable forepressure is defined as the forepressure which corresponds to an inlet pressure 10% greater than the inlet pressure corresponding to the normal forepressure for a vapor pump at a given throughput.

An array of thermocouples was installed radially across the boiler between the boiler base and the immersion heater to measure temperature distributions at various boiler power levels. Figure 8 shows the radial temperature distribution for power levels of 400, 500, and 600 watts. Runs were made at higher power inputs, but the turbulence resulting from eruptive boiling caused the individual readings to vary so much that they could not be resolved on the temperature-recorder chart. The boiler oil-pressure head was measured at each temperature level. No measurable difference was discernible above 500 watts, although the cathetometer was capable of detecting a least count of 0.1 mm of oil head. This fact further substantiates the contention that the boiler is limited in vaporization rate. At power inputs of 700 watts and above, eruptive boiling occurred. This was intermittent and not particularly violent at its onset, but at the higher power levels of 800 and 900 watts, it became consistently violent and produced instability in the pump operation as well as a much higher base pressure.

The contrast between normal or quiet and eruptive or noisy boiling has been mentioned by Knox.[4] The transition between the two regimes was observed through the boiler windows. The eruptive boiling consisted of a sequence of very rapid events, starting with the formation of large bubbles on the surface of the immersion heaters. These joined into an even larger bubble which expanded up through the liquid oil and exploded with considerable violence at the surface. This was accompanied by a crackling sound and caused a pressure burst in the high-vacuum test dome. Figure 9 is a photograph of the turbulence just as the bubble is breaking through the liquid surface. Some of the obvious means for avoiding this phenomenon and increasing the vaporization rate with quiescent boiling are under investigation.

One further characteristic of diffusion pumps of common interest is the backstreaming rate. Unfortunately, the many star-ups and the cyclic operation of the test pumps prevented any conclusive measurements of the effect of superheating on this rate. Quantitative measurements of backstreaming are also the subject of continuing investigation.

CONCLUSION AND PROSPECTUS

Consideration of typical diffusion pumps indicates that the relation of the thermodynamics of the pumping fluid to the mechanical design of pumps has largely been ignored by diffusion-pump designers. Preliminary experiments confirm the applicability of thermodynamic analysis to diffusion pumps and indicate lines of attack that should be fruitful in achieving improved performance of pumps. The state of the vapor immediately preceding its expansion through the nozzles has been shown to be extremely important in determining pump performance. This state is dependent on the heat transfer to the vapor as it flows up the jet stack and the rate of vapor generation in the pump boiler.

The pump jet structure should be heated either directly or indirectly to avoid vapor condensation in the stacks. There are advantages in operating with superheated vapor, both to provide a more energetic vapor jet and to avoid the possibility of condensation in the supersonic expansion. Superheating can also be accomplished in a constant-enthalpy throttling process. Thus appropriate orifices in the stacks supplying the various nozzle stages could be used to tailor the state of the vapor supplying each stage. Since the requirements of the various stages are different, it should thus be possible to achieve an optimum over-all pumping efficiency.

There are many possibilities for providing adequate boilers for diffusion pumps if some departure from the simplicity of the classical type can be accepted. The variations in boiler geometry to provide extended surfaces for vaporization as well as nucleation points are unlimited. The inclusion of a superheating stage above the liquid-vapor interface is a straightforward modification. Flash-type boilers represent an interesting possibility.

This paper has no pretensions as a comprehensive investigation. The intent has been to present promising preliminary results in order to stimulate further work by other experimenters.

ACKNOWLEDGMENTS

The author wishes to acknowledge the very valuable contribution of Mr. Patrick Kennedy in assisting with the experimental program.

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3. Saud Dushman, Scientific Foundations of Vacuum Technique (John Wiley and Sons, New York, N. Y. 1949).
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FIGURE LEGENDS

Fig. 1. Cross section of a 6-in. oil-diffusion pumps showing the internal temperature distribution.

Fig. 2. Liquid-vapor region of the temperature-enthalpy diagram for a diffusion-pump fluid.

Fig. 3. Cross section of a 6-in. oil-diffusion pump indicating the superheater installation.

Fig. 4. Jet-structure temperature and pumping speed vs time for a 6-in. modified oil-diffusion pump.

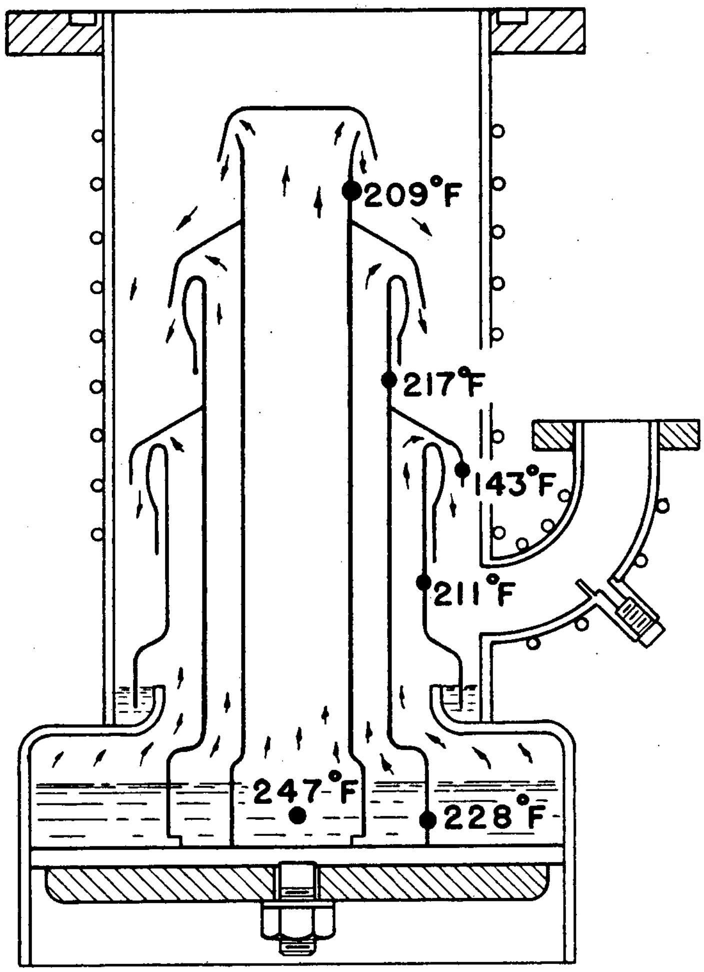
Fig. 5. Pumping-speed curves. Various degrees of superheat on a 6-in. oil-diffusion pump are shown.

Fig. 6. Jet-structure temperature and pump-inlet pressure vs time for a 6-in. modified oil-diffusion pump.

Fig. 7. Pump-inlet pressure vs tolerable forepressure.

Fig. 8. Radial temperature distribution in boiler of 6-in. oil-diffusion pump.

Fig. 9. Bubble formation in eruptive boiling in 6-in. diffusion-pump boiler.



MU-18016

Fig. 1

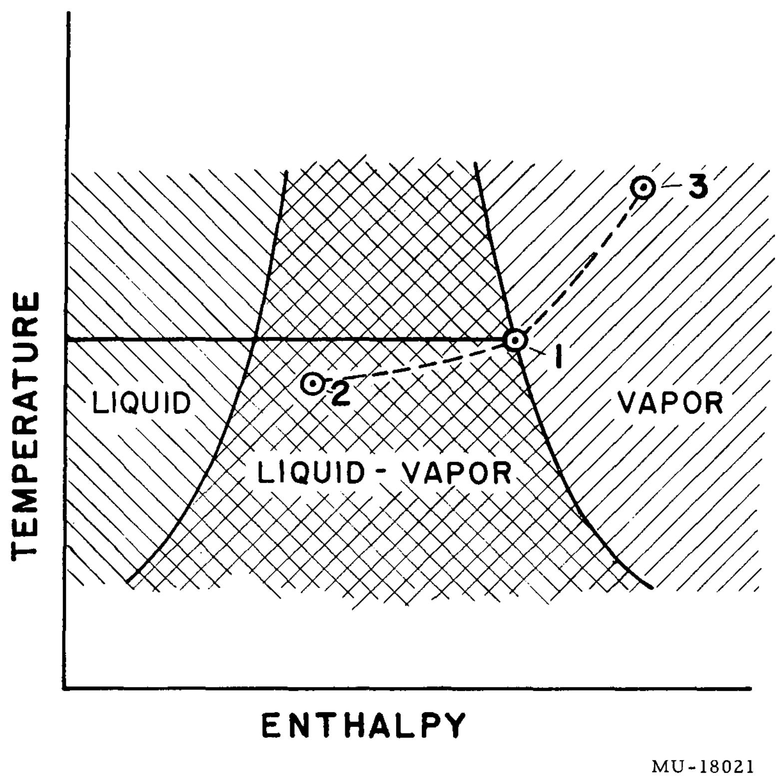

TEMPERATURE
3
1
2
LIQUID
VAPOR
LIQUID - VAPOR
ENTHALPY
MU-18021

Fig. 2

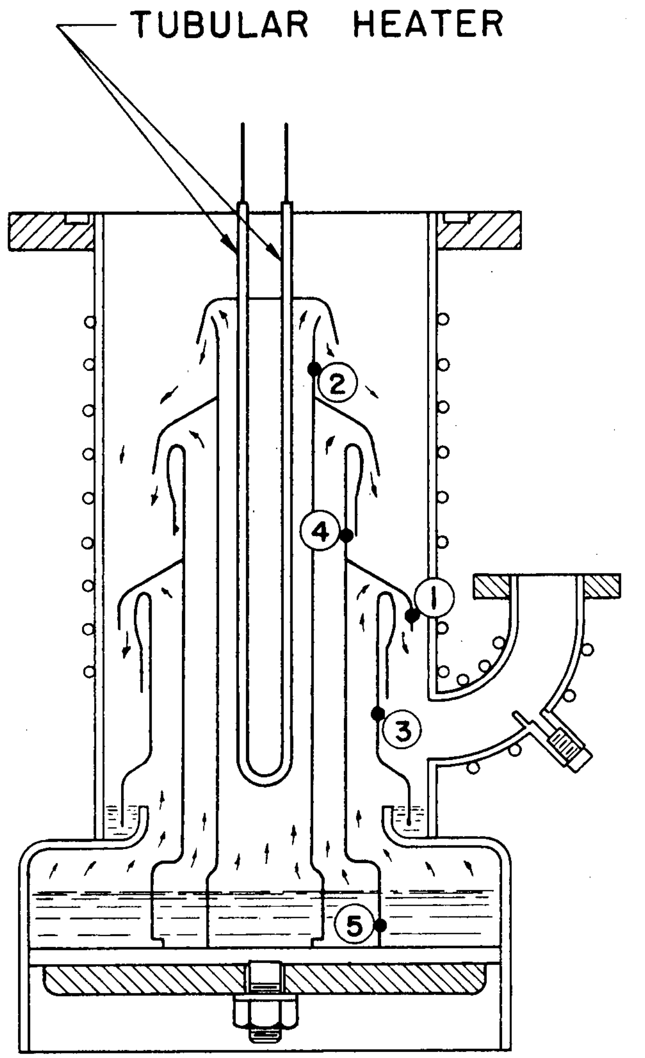
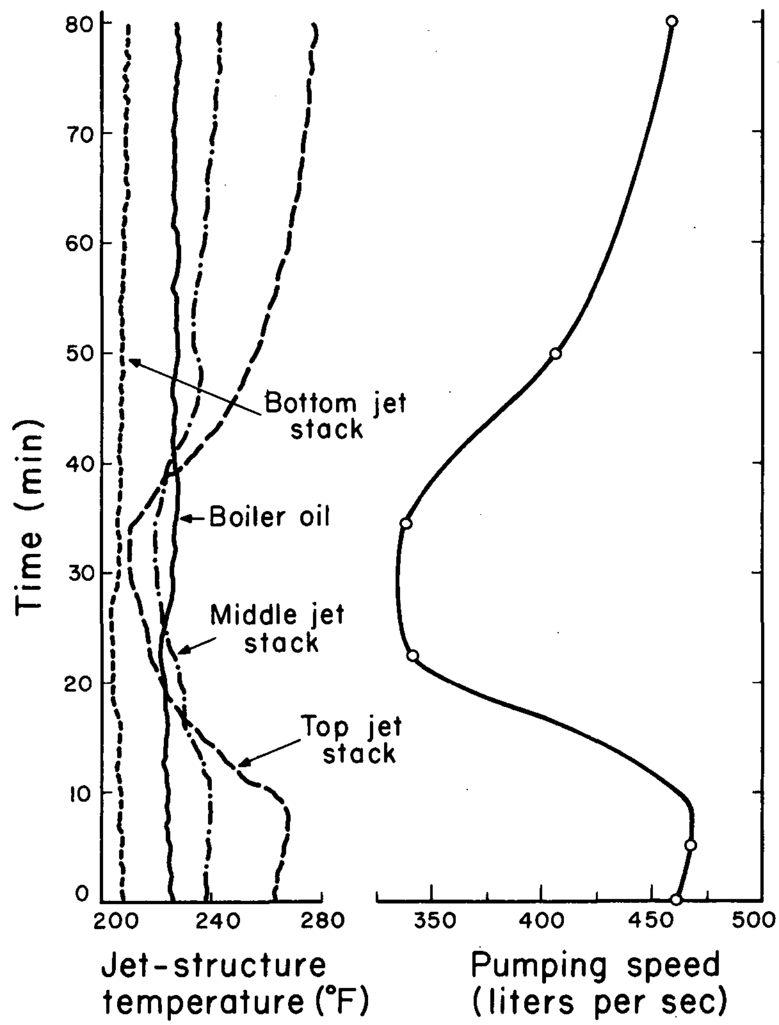
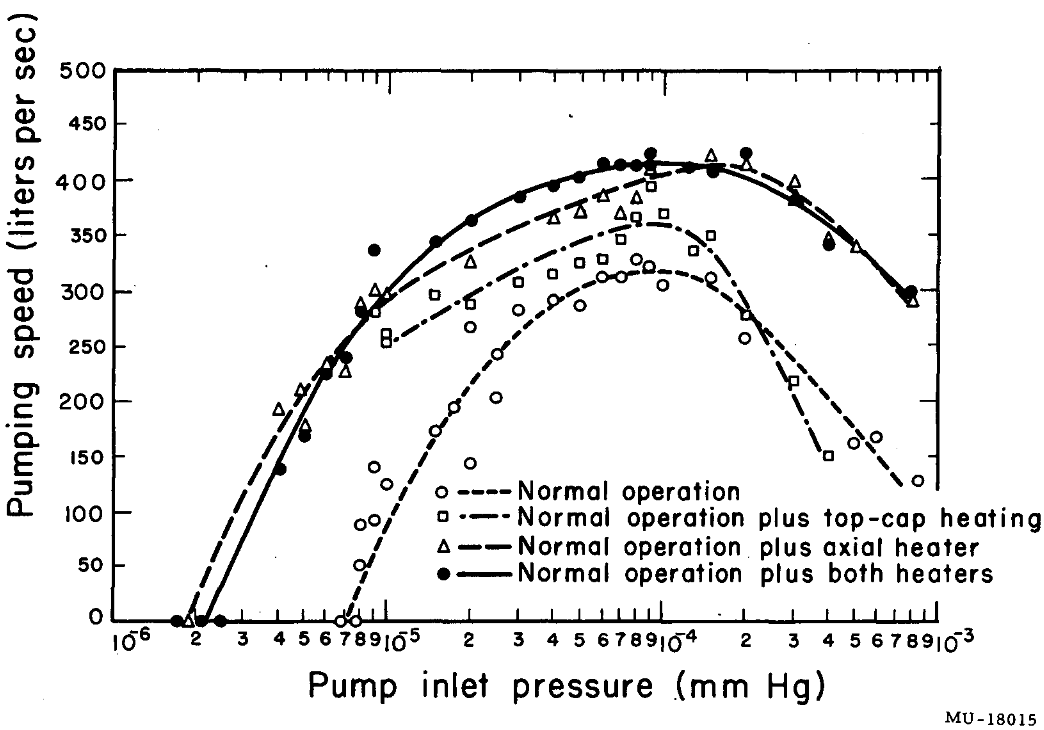


Fig. 3



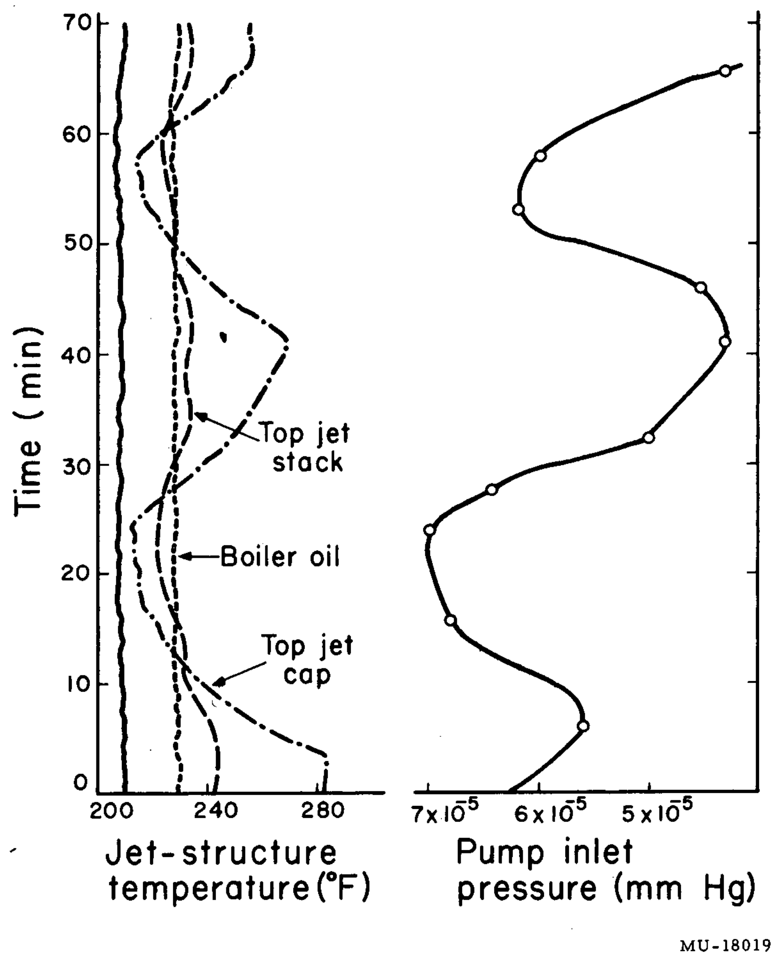
MU-18018

Fig. 4



Pumping speed (liters per sec)
500
450
400
350
300
250
200
150
100
50
0
10^{-6} 2 3 4 5 6 7 8 9 10^{-5} 2 3 4 5 6 7 8 9 10^{-4} 2 3 4 5 6 7 8 9 10^{-3}
Pump inlet pressure (mm Hg)
Normal operation
Normal operation plus top-cap heating
Normal operation plus axial heater
Normal operation plus both heaters
MU-18015

Fig. 5



70
60
50
40
30
20
10
0
Time (min)
200
240
280
Jet-structure temperature (°F)
Top jet stack
Boiler oil
Top jet cap
7x10^-5
6x10^-5
5x10^-5
Pump inlet pressure (mm Hg)
MU-18019

Fig. 6

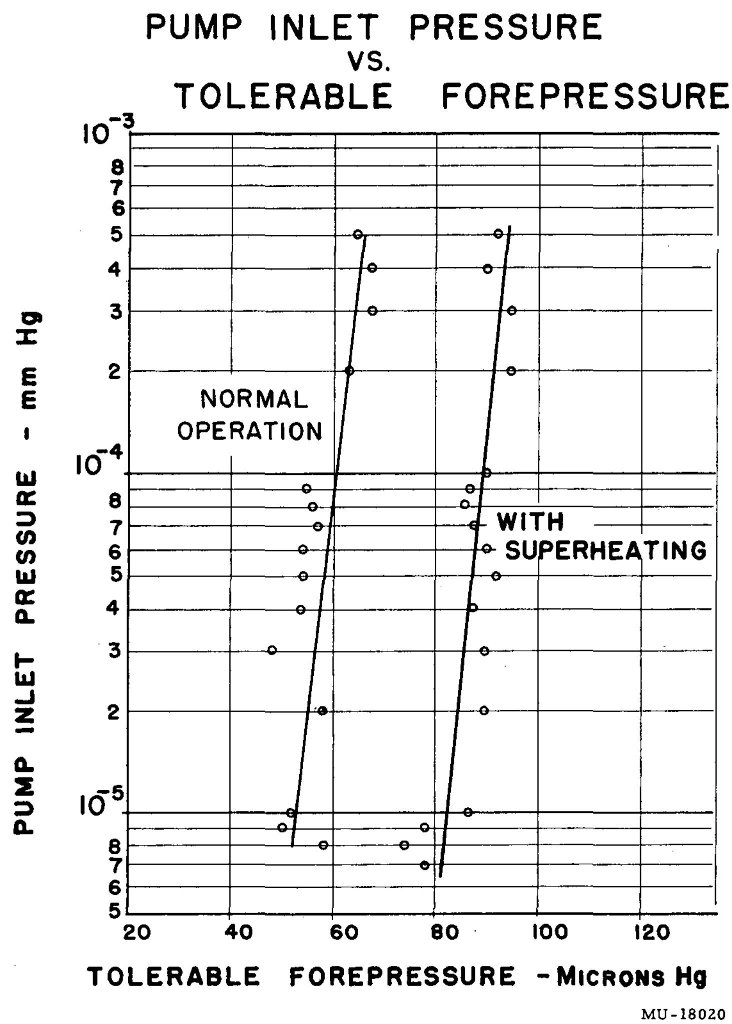

PUMP INLET PRESSURE
VS.
TOLERABLE FOREPRESSURE
PUMP INLET PRESSURE - mm Hg
NORMAL
OPERATION
WITH
SUPERHEATING
TOLERABLE FOREPRESSURE - MICRONS Hg
MU-18020

Fig. 7

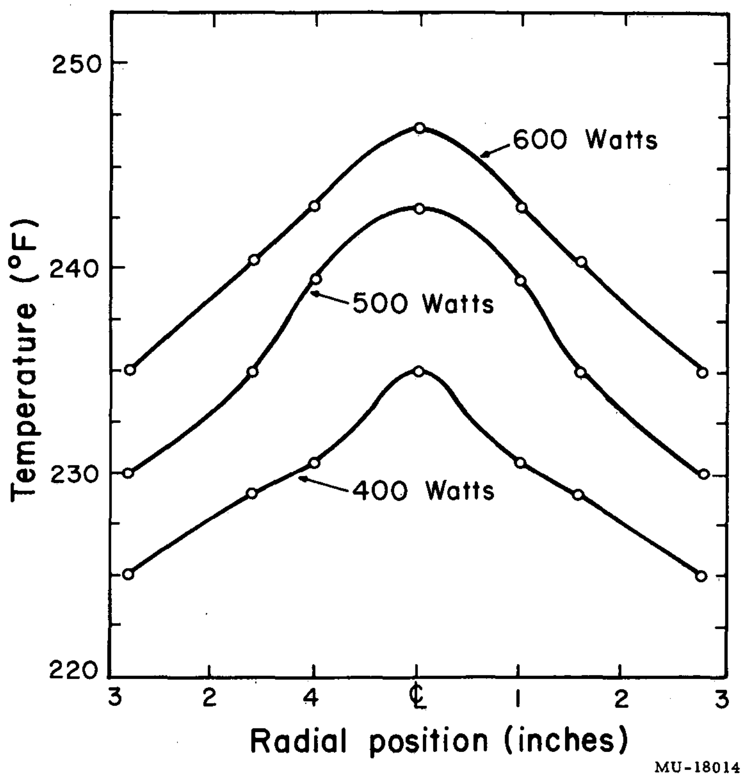

250
240
230
220
Temperature (°F)
600 Watts
500 Watts
400 Watts
3
2
4
℄
1
2
3
Radial position (inches)
MU-18014

Fig. 8

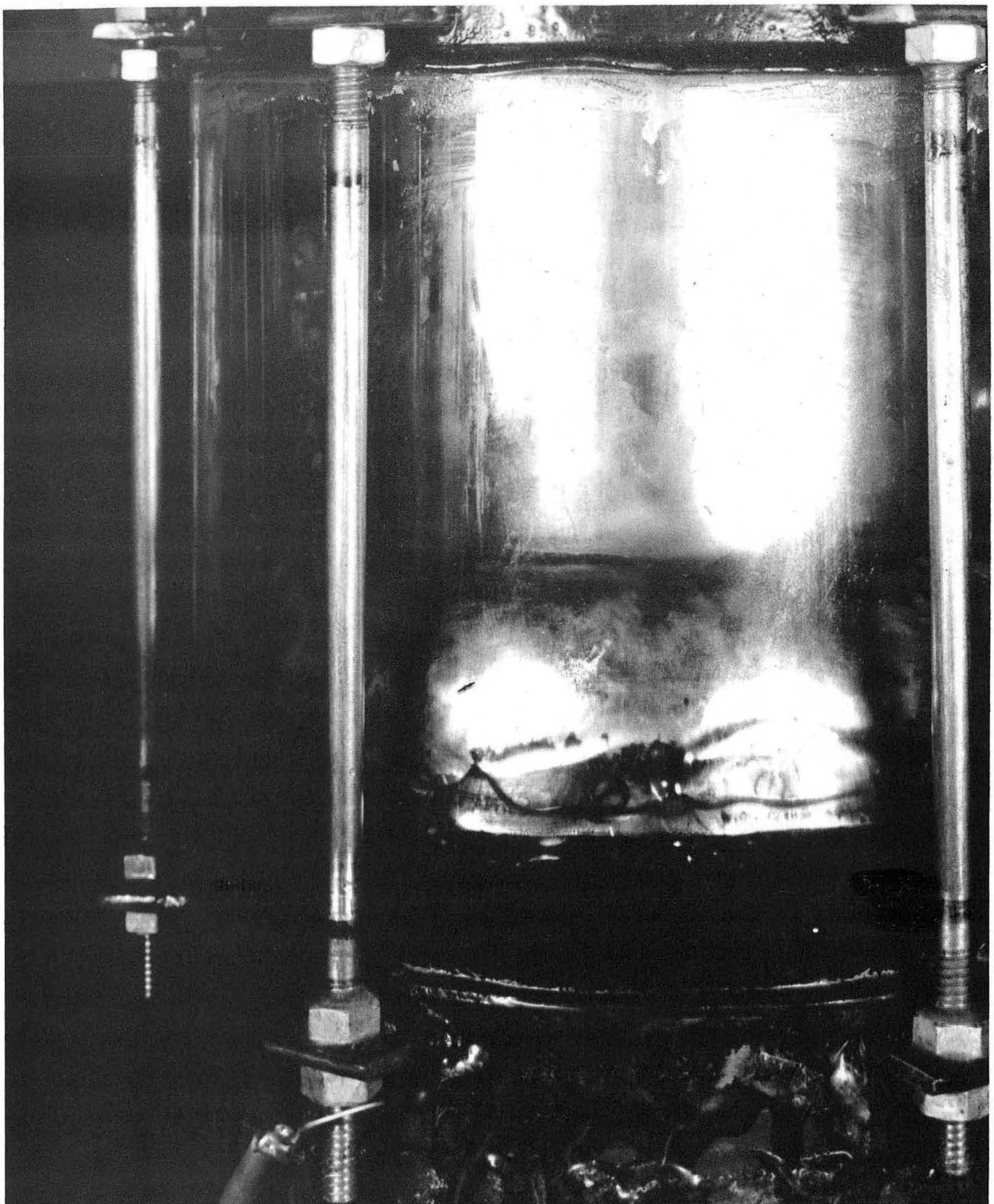


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Fig. 9