OPERATION OF STEAM EJECTORS IN SERIES

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SUMMARY

The purpose of this thesis is to investigate the performance, operating characteristics and stability of the vacuum system which has already been built to meet the requirement of the hypersonic wind-tunnel.

For relating the operating variables of steam ejectors, a formula is derived for general use. The formula shows that a linear relation exists between the mass flow of air, the pressure rise in the ejector and the motive steam velocity. Once a multi-stage ejector system is constructed in series, suction effects of each stage are additive. Stable operation can be obtained either by increasing the steam pressure or reducing the mass flow of air. Water requirements for the condensers can be varied, and the increased mass flow of water results in reducing steam consumption. Experimental verification of these relations is included. A formula for calculating efficiency of a multi-stage steam ejector system is also given.

The results of the investigation, based on the experimental data taken from the vacuum system and linked with the general principles of steam ejectors, give a clear understanding of the vacuum system. Optimum operation of the system with respect to the required degree of vacuum and flexibility of capacity is fully discussed.

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TABLE OF NOMENCLATURE

Notations		Units
A	Constant of integration,	cu.ft.
a	Radius of outer boundary of air-flow,	ft.
B	Constant of integration,	cu.ft.
b	Radius of inner boundary of air flow,	ft.
$f(a,b)$	Function defined in 2.2,	$ft.^6$/lb.-sec.
$f'(a,b)$	Function defined in 2.2,	$ft.^5$/lb.-sec.
$g(a,b)$	Function defined in 2.2,	cu.ft.
h	Specific enthalpy,	B.t.u./lb.
k	Ratio of specific heats, C_p/C_v .	
P	Pressure,	psfa.
P_a	Partial pressure of air,	psfa.
P_d	Discharge pressure,	psfa.
P_p	Supporting pressure,	psfa.
P_v	Partial pressure of water vapour,	psfa.
r	Radius,	ft.
t	Temperature,	°F
U	Velocity,	ft./sec.
W	Mass flow,	lb./sec.
W_v/a	Amount of water vapour per one pound of air,	lb./lb.
z	Linear distance along the axis of an ejector,	ft.
γ	Ratio of steam flow, $\frac{W_{sI}}{W_{sII}}$.	
Δ	Denotes increment.	
ΔP	Pressure rise in an ejector,	psfa.
P_{SU}	Suction pressure of an ejector,	psfa.

ΔP_H	Pressure rise in a vacuum pump,	psfa.
μ	Dynamic viscosity,	slugs/ft-sec.
ρ	Density,	lb./ft.^{3}
ϕ	Specific entropy,	B.t.u./lb.-°F.

Subscripts

a	Refers to the air.
s	Refers to the steam.
ϕ	Refers to an isentropic condition.
I	Refers to the first-stage ejector.
II	Refers to the second-stage ejector.

Subscripts not listed above are explained as introduced.

CHAPTER I

INTRODUCTION

1.1 General Remarks

The use of a steam jet ejector for entraining air or gases at sub-atmospheric pressure is increasing rapidly. This is due to several advantages:

a. Lower initial, installation and maintainance costs.

b. No moving parts in itself and reliable in service.

c. Construction readily adopted to special materials for for corrosive or abrasive conditions, and no lubricant or sealing liquid to be affected by gases containing solvents or other contaminants.

On the other hand, a steam ejector is a fixed capacity machine by reason of its construction. An increase or decrease in the quantity of air or gas being handled under constant suction and discharge conditions can not be accomplished in the basic assembly.

Commercial steam ejectors accordingly are arranged in a variety of forms to meet limitations with respect to required degree of compression and flexibility of capacity. Where the required degree of compression is beyond the capacities of a basic single stage assembly, two or more stages are arranged to operate in series, each stage effecting a part of the total compression. Where need for flexibility in capacity exists, two or more ejectors either single or multi-stage as required for compression, can be arranged to operate in parallel so that each set contributes part of the total capacity. Therefore, it is evident that ejectors can be readily arranged in any desired combination to suit the specific requirement.

A combination of ejectors of different individual capacity with which author has been directly connected was to meet the requirement of the evacuator of the hypersonic wind-tunnel in McGill Hypersonic Laboratory (Figures 1 and 2). The actuating fluid is steam which is conveniently generated either from the boiler in the laboratory or from the power house of McGill. The induced fluid for the purpose of testing the ejector set is air at normal room temperature and pressure.

The first stage consists of two ejectors in parallel, one of which is available for flexibility in capacity. Those are succeeded by another one as the second stage. Inter and after condensers of surface type are used and the last stage is supported by two powerful, different individual capacity, mechanical vacuum pumps in parallel (Nash Hytor pumps).

1.2 Purpose and Scope of Investigation

The objective of this thesis is to investigate the performance, operating characteristics and stability of the set-up, and incidently to indicate the possible further improvements which could be made to the said system.

In spite of the success of ejectors operating in multi-stages in practical applications, the data for the performance are meagre and limited to a certain type of practical application. It is therefore logical that before proceeding with the more complicated problems in multi-stage ejectors, a general study of ejector operating characteristics is essential. To accomplish this, the comparison of actual performance with predicated performance has been carried out and is described below.

CHAPTER II

PERFORMANCE AND EFFICIENCY OF STEAM EJECTORS

2.1 Review of Previous Work

Each of the published works on ejectors deals more or less with single stage ejector performance. In the analysis of a complete ejector, it is general to apply the equation of continuity, the principle of momentum, the equation of energy and equation of fluid state to different sections of the ejector. By use of momentum relations, details of the entraining process can be avoided and the results are free from any consideration of effects of viscosity and diffusion. The problem is then to solve the simultaneous equations. However, due primarily to the great number of variables a comprehensive and straightforward expression for general use of performance prediction is not yet possible.

A systematic testing and theoretical study of high-suction ejectors, whose primary flow is supersonic, have been studied more recently (Refs. 9, 11 and 13).

Performance tests on steam ejectors were carried out by Johannesen (Ref. 11). Wet, dry and superheated steam were used for testing with different steam pressures and variations of geometrical shape. The relations between the mass flow of air and the suction pressures were almost a linear function.

A rather thorough investigation of the supersonic ejector has more recently been made by Fabri and Paulon (Ref. 13). The authors present a method of extrapolating classical ejector performance to the supersonic units based on an aerodynamic analysis. The schlieren photographs give qualitative information on the effect of changing the motive fluid pressure. Under the regime of the supersonic flow of the primary fluid, a distinct separation between the primary and secondary fluids are noted. The relation between the mass flow and the suction pressure of the secondary fluid is again found to be linear, and experimental verification of the method is included.

An example of a calculation of a jet pump with a supersonic flow of the primary fluid in a constant area conduit was given by Turner, Adie and Zimmernan (Ref. 8). By use of the charts for the analysis of one-dimensional steady compressible flow, the calculated results were plotted on a graph. From that graph, it was shown that for a fixed condition of the primary flow, the relation of the mass flow of the secondary air to the pressure rise in the jet pump bore a linear function.

2.2 An Approximate Formula for Representing Ejector Performance

The primary fluid flow in a steam ejector is supersonic. By referring back to Reference 13, an assumption is made that the steam and air are only partially mixed and each of them has an individual energy balance. Those air particles in contact with the conduit wall are at rest. The air density in the ejector is low due to the high suction nature. Those air particles in contact with the steam stream have the same speed and temperature as the steam. Due to the low density of the air in the ejector even at high speed, the Reynolds number is relatively small. Then, the air flow in a steam ejector is similar to the laminar flow in an annular space between two concentric tubes.

In case of the one-dimensional steady flow with negligible gravitational effect, the differential equation of the air flow is that (Refs. 14, 15):-

$$\frac{\delta P}{\delta z} = \frac{\mu}{r}\frac{d}{dr}\left(r\frac{dU}{dr}\right) \qquad \text{..........} \quad (1)$$

From Equation (1), the velocity distribution of the air flow is given by:-

$$U = \frac{1}{4\mu}\frac{\delta P}{\delta z}\left(r^2 + A \ln r + B\right) \qquad \text{......} \quad (2)$$

Letting r=a be the outer boundary and r=b at the inner boundary, two conditions are available to determine two constants:

$$U = 0,\ r = a; \qquad U = U_s,\ r = b,$$

where U_s is the velocity of the steam flow.

Evaluating the constants, the velocity distribution of the air is given by:-

$$U = \frac{1}{4\mu}\frac{\delta P}{\delta z}\left(r^2 - a^2 + \frac{a^2-b^2}{\ln\frac{b}{a}}\ln\frac{r}{a}\right) + U_s\,\frac{\ln\frac{r}{a}}{\ln\frac{b}{a}}, \qquad \ldots\ldots\ldots \quad (3)$$

and the mass flow of air W_a by:-

$$W_a = \int_b^a 2\pi\, r\, \rho_a\, U_s\, dr$$

$$= -\rho_a \frac{\pi}{8\mu}\frac{\delta P}{\delta z}\left[a^4 - b^4 + \frac{(a^2-b^2)^2}{\ln\frac{b}{a}}\right]$$

$$+ 2\pi\, \rho_a\, U_s\, \left(\frac{b^2}{2} + \frac{a^2-b^2}{\ln\frac{b}{a}}\right) \qquad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

Let

$$f(a,b) = \frac{\pi}{8\mu}\left[a^4 - b^4 + \frac{(a^2-b^2)^2}{\ln\frac{b}{a}}\right]$$

and

$$g(a,b) = 2\pi\left(\frac{b^2}{2} + \frac{a^2-b^2}{\ln\frac{b}{a}}\right);$$

then Equation (4) becomes:-

$$\frac{W_a}{\rho_a} + \frac{\delta P}{\delta z} f(a,b) = U_s\, g\,(a,b), \qquad \ldots\ldots\ldots\ldots \quad (5)$$

where f(a,b) and g(a,b) are only functions of geometrical parameters which depends on the configuration of the ejector and the flow pattern. In that equation, the δP is the increment of pressure across the infinitesimal distance δz along the axis of the ejector.

From previous works (Refs. 11, 12), it is known that the value of $\frac{\delta P}{\delta z}$ in Equation (5) varies according to the ejector shape and the flow pattern. If an average value of $\frac{\delta P}{\delta z}$ is chosen for an ejector to fit the following relation approximately, that is:-

$$\frac{\delta P}{\delta z} = \frac{\Delta P}{\Delta z} ,$$

where the ΔP is the pressure rise in the ejector from the intake to exit, and Δz is a linear dimension along the axis of the ejector, then Equation (5) may be written into the form:-

$$\frac{W_a}{\rho_a} + \Delta P\, f'(a,b) = U_s\, g(a,b) \quad \ldots\ldots\ldots\ldots \quad (6)$$

2.3 Prediction of the Ejector Performance

By use of Equation (6), the performance of an ejector system can be predicated in the following way:-

(a) For the same steam condition, the mass flow of air through an ejector decreases as the pressure rise in the ejector increases. This is almost a linear function (Figure 3).

(b) The suction pressure of an ejector depends on the steam pressure. Thus the higher the steam pressure, the lower is the suction pressure. This follows since:-

$$U_s \,\alpha\, \left[1 - \left(\frac{P_{sU}}{P_s}\right)^{\frac{k-1}{k}} \right]^{1/2} .$$

Therefore from Equation (6), for a certain value of W_a and ρ_a, the pressure rise ΔP in the ejector is given by:-

$$\Delta P \; \alpha \; \left[1 - \frac{1}{2}\left(\frac{P_{sU}}{P_s}\right)^{\frac{k-1}{k}} \right], \qquad \ldots\ldots\ldots\ldots\ldots \quad (7)$$

where P_{sU} is the suction pressure; P_s is the motive steam pressure and k is the ratio of specific heats of steam.

It can be seen from Equation (7) that by use of steam of higher pressure, the ΔP will be greater, and the vacuum will be better. Curves in Figure 4 are the predicted ones.

(c) An ejector may be followed by another ejector or a vacuum pump which is called the supporting stage of the previous stage. In this case, the supporting pressure of the previous ejector is the suction pressure of its supporting stage. The relation given by Equation (6) is true, whatever the supporting pressure may be. When a multi-stage ejector system operates in series, it can be seen from Equation (6) that the suction effects of each stage are additive.

(d) In case of operation of two steam ejectors in series with a vacuum pump used as an atmospheric stage, the system is so arranged [Figure 5 (a)], that the capacity of each unit is shown as Figure (b). When the system is operating at the mass flow of air W_a', all of the three units are effective. Once the mass flow of air increases to W_a'', the first-stage will be automatically out of action at that instant. If the mass flow of air increases further, say $W_a{}''{}'$, both the first and the second stages shall be no longer effective.

(e) Transverse shocks within the motive fluid may cause instability of a single ejector (Refs. 7, 9). Ejector instability occurs at a certain mass flow of air when the suction pressure merely equals the supporting pressure. It seems that the wave fronts are formed while the pressure rise in the ejector diminishes. The previous discussions [Equations (6) and (7)] show that any decrease in the steam pressure or increase in the supporting pressure will cause a decrease of the pressure rise in the ejector. Although Equation (6) does not deal with the steam flow, it may be used to locate the point of unstable operation at a certain mass flow of air.

2.4 The Single-Stage Ejector Efficiency

The definition of a single-stage ejector efficiency is proposed (Refs. 5, 7) as

$$\text{Ejector Efficiency} = \frac{\text{actual flow ratio}}{\text{isentropic flow ratio}} \quad .. \ (8)$$

When the performance of an ejector is to be studied, it is reasonable to consider the efficiency of the exchange of energy between the motive steam and the induced air. The ejector efficiency is of importance when the ejector is considered from a thermodynamical point of view.

From Figure 6, the least specific energy required to bring air from pressure P_2 to pressure P_3 is $\Delta h_{a\phi}$ and the maximum specific energy that steam can produce between two pressure limits is $\Delta h_{s\phi}$. By the principle of conservation of energy, the fluid flow ratio of the isentropic flow is given by:-

$$\left(\frac{W_a}{W_s}\right)_\phi = \frac{\Delta h_{s\phi}}{\Delta h_{a\phi}}, \qquad \ldots\ldots\ldots\ldots\ldots (9)$$

where $\left(\frac{W_a}{W_s}\right)_\phi$ is the maximum (i.e. isentropic) flow ratio between pressure limits P_1, P_3 and P_2, P_3. This ratio is a logical criterion of ejector performance. By combining Equations (8) and (9), the single-stage ejector efficiency is given by:-

$$\text{Ejector efficiency} = \frac{W_a}{W_s} \Big/ \left(\frac{W_a}{W_s}\right)_\phi$$

$$= \frac{W_a}{W_s} \frac{\Delta h_{a\phi}}{\Delta h_{s\phi}}, \qquad \ldots\ldots\ldots (10)$$

where $\frac{W_a}{W_s}$ is the mass ratio of the air to the steam, both of which are actually measured.

Note that the paths of states of the air and the steam in Figures 6 and 7 are assumed, the actual paths are never known. Both diagrams serve only as an illustration of several steps involved rather than the actual values of energy transmitted form the steam to the air.

2.5 Multi-Stage Ejector Efficiency

Figure 7 shows a two- stage ejector operating in series. The principle of conservation of energy requires that

$$W_a \, \Delta h_{a\phi} = W_{sI} \, (\Delta h_{s\phi})_I + W_{sII} \, (\Delta h_{s\phi})_{II} \quad .. \; (11)$$

Let $\gamma = \dfrac{W_{sI}}{W_{sII}}$

and $W_s = W_{sI} + W_{sII} = (1+\gamma) \, W_{sII}$,

Equation (11) may be written into the form:-

$$\left(\frac{W_a}{W_{sII}}\right)_\phi = \frac{\gamma \, (\Delta h_{s\phi})_I + (\Delta h_{s\phi})_{II}}{\Delta \, h_{a\phi}}$$

and

$$\left(\frac{W_a}{W_s}\right)_\phi = \frac{\gamma \, (\Delta h_{s\phi})_I + (\Delta h_{s\phi})_{II}}{(1+\gamma)(\Delta h_{a\phi})}$$

The ejector efficiency of the two-stage system is given by:-

Ejector efficiency

$$= \frac{W_a \, (1+\gamma)(\Delta h_{a\phi})}{W_s \left[\gamma \, (\Delta h_{s\phi})_I + (\Delta h_{a\phi})_{II} \right]} \quad \ldots\ldots\ldots \; (12)$$

where W_a and W_s are the measured quantities.

2.6 Effect of Performance of the Surface Condenser on the Ejector

In addition to indirect contact heat transfer in a surface condenser, a large amount of air in the condenser hinders the rate of heat transfer. A higher temperature head for better heat transfer is required and the minimum temperature rise of the condensing water is necessary. The air withdrawn from a surface condenser is saturated with water vapour and the amount of water vapour is dependent upon the temperature and absolute pressure. The amount of water vapour required to saturate one pound of dry air is given by:-

$$W_{v/a} = 0.62 \frac{P_v}{P_a} \qquad \text{............... (13)}$$

where 0.62 is the ratio of molecular weights of water vapour to air, P_v and P_a are partial pressures of water vapour and air respectively. The greater load imposed on the second stage ejector by the vapour of saturation coming from the inter-condenser at the higher outlet mixture temperature will decrease the suction pressure of the second stage. Similarly, more compressive work is imposed on the vacuum pumps by the vapour of saturation coming from the after-condenser, which in turn will drop the suction pressure of the vacuum pumps. Therefore the change on the suction curve of the ejector due to reducing the mass flow of condensing water (or using condensing water of high inlet temperature) is the same as due to raising the supporting pressure of the ejector.

CHAPTER III

EXPERIMENTAL EQUIPMENT

3.1 Arrangement of Apparatus

The tests were carried out on the vacuum system in the Hypersonic Laboratory of McGill. This unit of equipment was built in 1959-1960 by the hypersonic group. Figure 8 is the schematical drawing of this system, showing the air intake header, one of the two ejectors of the first stage, the ejector of the second stage, the condensers and Hytor vacuum pumps. Figures 2, 9 and 10 are the photographs showing their locations and their actual arrangement.

Motive steam comes from the boiler, is passed through separators, and then through control valves, to the manifolds which lead steam into the three ejectors individually. An overhead air intake header of 8-inch diameter is connected the hypersonic wind tunnel through a large gate valve. When the ejectors were under tests, the gate valve between the wind tunnel and the intake header was shut. The air for the test purpose came from atmosphere through a bypass passage, in which an air flow meter was installed.

City water was used for cooling in the condensers. The flow of water between the two condensers was in series, first it led into the inter-condenser, then to the after-condenser, and finally drained.

The condensate of the inter-condenser was accumulated at the condensate tank, then it was pumped out by a centrifugal pump which was driven by a 1-1/2 H.P. motor. An automatic device, which was governed by the water level in the condensate tank, operated the switch of the motor. Passing through a check valve, the condensate was pumped into a level tank which was opened to the air. For the purpose of filling the pump case for starting and preventing air leakage through the check valve, the water level in the level tank was kept constantly two feet higher than that of the outlet of the condensate tank.

Through the after-condenser, both air, water vapour and condensate were evacuated through the Hytor vacuum pumps which operated in parallel.

3.2 Apparatus List

Items	Description
First Stage Ejector	Elliott 62E Steam-jet Air Ejector, diameter of steam nozzle 0.281 in.
Second Stage Ejector	Elliott 61E Steam-jet Air Ejector, diameter of steam nozzle 0.344 in.
Vacuum Pump No. 1	Nash Hytor Vacuum Pump H-4, 1300 rpm., driven by a 7.5 H.P. motor.
Vacuum Pump No. 2	Nash Hytor Vacuum Pump H-4, 1750 rpm., driven by a 10 H.P. motor.
Inter-Condenser	A 212 sq.ft., triple-path surface type. Fitted with 2 in. pipe for condensing water.
After-Condenser	Wheeler Surface Condenser, 198 sq.ft., double-path. Fitted with 2 in. pipe for condensing water.

3.3 Instrumentation

The locations of pressure taps, elements of thermocouples, bulbs of the resistance thermometer, orifice plates of flow meters and others are shown in Figure 11. The identifications in Figure 11 correspond to that of the instrument list. Figure 12 shows the elevation of the panel board.

3.4 Instrumentation List

	Instrument	Range	Smallest Subdivision
FLOW METERS:			
1. Air flow, 2 in. pipe and 1.461 in. orifice diameter.	H_2O Manometer	36"	0.1"
2. Steam flow, 2 in. pipe and 0.832 in. orifice diameter.	Hg Manometer	18"	0.1"
3. Water flow, 2 in. pipe and 1.600 in. orifice diameter.	Hg Manometer	36"	0.1"
PRESSURES:			
4. Steam main inlet pressure	Borden gauge	0-300 psig	10 psi.
5. First-stage inlet steam pressure.	Borden gauge	0-200 psig	10 psi.
6. Second-stage inlet steam pressure.	Borden gauge	0-200 psig	10 psi.
7. Air inlet pressure	Hg Manometer	32"	0.1"
8. First-stage suction pressure.	Hg Manometer	32"	0.1"

(Pressures - cont'd.)

	Instrument	Range	Smallest Subdivision
9. First-stage discharge pressure	Hg Manometer	32"	0.1"
10. Differential pressure across inter-condenser.	H_2O Manometer	36"	0.1"
11. Second-stage suction pressure	Hg Manometer	32"	0.1"
12. Second-stage discharge pressure	Hg Manometer	32"	0.1"
13. Differential pressure across after-condenser.	H_2O Manometer	32"	0.1"
14. Condensing water pressure between the condensers.	Borden gauge	0-20 psig.	2 psi.
15. Condensing water inlet pressure.	Borden gauge	0-300 psig.	10 psi.
Atmospheric pressure	Hg Barometer (in the laboratory)		0.01"

TEMPERATURES:

	Instrument	Range	Smallest Subdivision
16. First-stage inlet steam temperature. 17. First-stage inlet air temperature. 18. First-stage discharge temperature. 19. Inter-condenser outlet temperature.	Weston Iron-constantan thermocouple with 0° reference junction. A selective switch with 20 sector positions.	0-350 °C	10 °C

(Temperatures - cont'd.)

	Instrument	Range	Smallest Subdivision
20. Second-stage steam temperature. 21. Second-stage inlet air temperature. 22. Second-stage discharge temperature. 23. After-condenser outlet temperature. 24. Throttling steam colorimeter, 3/16" nozzle diameter.	Weston Iron-constantan thermocouple with 0°C reference junction. A selective switch with 20 sector positions.	0-350°C	10°C
25. Inter-condenser condensate temperature. 26. After-condenser condensate temperature. 27. Outlet temperature after vacuum pump no. 1. 28. Outlet temperature after vacuum pump no. 2.	3-wire Bristol resistance thermometer. A selective switch with 12 sector positions.	-30-0-+120°C	2°C
29. Condensing water inlet temperature. 30. Condensing water temperature after inter-condenser. 31. Condensing water outlet temperature.	3-wire Bristol resistance thermometer. A selective switch with 12 sector positions.	-30-0-+120°C	2°C

CHAPTER IV

EXPERIMENTAL PROCEDURE

4.1 Calibration of Meters

Before proceeding with the ejector tests, the instruments were checked and meters calibrated. The thermocouples and thermometers were checked at the boiling point of water under atmospheric pressure. The apparatus contained three sets of flow meters, for air, water and steam. They were all thin plate orifice meters with flange taps. The mass flow formula and coefficients for those meters were taken from Reference 16, and calibration curves were evaluated. Curves of correction factors for both air and steam at temperatures and pressures other than the calibrated ones were plotted (Ref. 17). The flow across the steam nozzle was critical. The throat diameters were measured. By taking the ratio of specific heats of steam 1.33, the amount of steam flow in pounds per hour was calculated from nozzle formula using the observed total pressure and temperature. The readings from steam flow meters were checked against the values calculated fro the nozzles.

4.2 Leakage Check

Before experimental running, efforts to eliminate air leakage were made. By introducing water into the lower part of the vacuum system, leakages at various parts were checked. Compressed air was also used to fill the whole system up to 3 psig. to permit soap bubble leak detection.

4.3 Leakage Tests

The leakage of the system could be measured by evacuating it with the vacuum pumps, then noting the rate of pressure change with all valves closed. Such leakage tests were made before each test. All leaks would be above critical pressure ratio, i.e. passing constant mass flow as long as the final vacuum at the end of test was not less than 15 inches of mercury. The net volume of the system under vacuum was estimated to be 36.7 cubic feet in advance. The drop in vacuum during observation was usually less than 1 inch of mercury per one minute. Therefore the rate of air leakage was less than 6 pounds per hour, which was calculated from a theoretical formula (Ref. 18).

A leakage test was also made after each run to make sure that the rate of leak remained small during the running period. Throughout the tests, a slight change occurred. The reading after running usually had about 0.1 inch of mercury per minute more than that of pre-running. For the two tests, conditions remained the same, except that the temperature of condensate in the condensate tank increased after the running.

4.4 Experimental Running

There were three variables for the ejector test:

(1) Steam pressures: Steam pressures of 70, 100 and 150 psig. were chosen, where the 100 psig. was the design minimum pressure.

(2) Supporting pressures imposed on the ejectors: There were four different supporting pressures obtainable, the atmospheric pressure, the suction pressure produced by the vacuum pump no. 1, that produced by no. 2, and the one produced by the two vacuum pumps in parallel. The first one kept constant and the others were variable depending on the mass flow of air.

(3) Mass flow of air: This, expressed in pounds per hour of air, was the suction capacity of an ejector. The mass flow of air could be varied from no load to maximum, but its maximum value depended on other variables.

First, tests were carried out to determine the performance curves of the Hytor vacuum pumps, individually and then wholly, as a function of the mass flow of air. Suction pressures were observed. Figure 13 was the result.

A series of tests for individual stages of the vacuum system were then performed to determine the effect of (1), (2) and (3) on its performance. The procedure for these experiments all followed a similar pattern. The results of these tests were shown in Figures 14 to 21 inclusive.

Finally an over-all test was carried out at steam pressures of 100 and 150 psig. Figures 22 to 26 were the results.

To avoid any effects from changing conditions of the condensers, the condition of condensers were kept close to constant throughout all the main tests. Condensing water was kept 7,500 GPH and the seasonal variation of inlet condensing water temperature was less than 2°C (3.6°F) over the period of the tests.

The inlet air remained close to room temperature and was at all times the prevailing barometric pressure.

The dryness of the steam was detected by a throttling calorimeter. Throughout the tests, the dryness of steam was practically constant at a value of 97%.

The effect of varying the condenser condition on ejector performance was also investigated by varying the mass flow of condensing water. This led to variation in the effective water temperature. The results of these tests show the effect on ejector performance (Figures 27 to 29 inclusive).

CHAPTER V

DISCUSSION OF RESULTS

Ejector performance is generally plotted as the mass flow of air in lbs. per hr. versus suction pressures in psia. Since an ejector operates on mass flow and momentum transfer principles, capacities are generally based on mass flow rates rather than volumetric units.

The term "supporting pressure" P_p means the back pressure acting on the ejector discharge caused by factors external to the ejector. The term "discharge pressure" P_d is the static pressure which is actually measured at the exit plane of the steam ejector. The discharge temperature t_d is the static temperature which is measured at the same plane. The suction pressure is the static pressure which is actually measured of the inlet section of the air flow of the ejector or the Hytor vacuum pump.

5.1 Suction Curves of Hytor Vacuum Pumps

The relation between the mass flow of air and suction pressures are plotted in Figure 13. These two pumps operate in parallel, drawing air at the same inlet condition and discharge against the same atmospheric pressure. At a certain suction pressure, the over-all mass flow of air must be the sum of the individual ones as exhibited in the same figure.

5.2 The First-Stage Steam Ejector

The results of individual tests of the first-stage steam ejector with four different features of the supporting pressure are shown in Figures 14, 15, 16 and 17. In Figure 14, the supporting pressure P_p is the atmospheric pressure, and in Figures 15, 16 and 17, the supporting pressures are the suction pressures of the Hytor vacuum pumps.

A. Suction curves. In Figures 14, 15, 16 and 17, the experimental tests show that:-

(1) With different motive steam pressure, the mass flow of air and the pressure rise in the ejector has a linear relationship.

(2) A linear relationship is true, whatever the supporting pressure may be.

(3) Once a supporting pressure curve intersects a suction pressure curve, the operation of the ejector becomes unstable which is marked by cyclic variations in suction pressure.

(4) When the steam pressure is 124.6 psia., it causes the ejector to operate unstably at a lower mass flow of air. A stable operation can be obtained by raising steam pressure to 154.6 psia.

The above results can be explained directly from Equation (6) which has been expressed in Chapter II.

Figures 14, 15, 16 and 17 also reveal that the first-stage steam ejector is not made to operate over a great range of mass flow of air at higher supporting pressures. The manufacturer's information reveals that the proper supporting pressure of the first-stage steam ejector is 4 inches of mercury absolute. This follows since the geometrical configuration of the ejector's elements determines its optimum operating pressure which is dependent on the constants f'(a,b) and g(a,b) of Equation (6).

B. Discharge conditions. Throughout all the tests for the steam pressure of 154.6 psia, the discharge pressures and temperatures were observed. They are plotted in Figures 14, 15, 16 and 17.

(1) The discharge pressures When an ejector is operating, its supporting pressure is masked by the discharge pressure at the exit. The pressure which can be actually measured at the exit of the ejector is the discharge pressure. By examining Figures 14, 15, 16, and 17, it is seen that the discharge pressures P_d are always greater than the supporting pressure P_p. As the mass flow of air increases, the departure of P_p from P_d also increases. But contrary to this fact are Figures 14 and 15, where the departure of P_d from P_p is greater at low pressure region. This is due to the higher discharge temperature with low pumping ability of Hytor vacuum pump.

Comparing with other graphs at the same region, it is seen that the steeper the P_p curve, the greater is the departure of P_d and P_p.

(2) The discharge temperature. By examining the variation of the discharge temperatures in Figures 14, 15, 16 and 17, it is found that the discharge temperatures lie around the vicinity of the saturated temperature corresponding to the partial pressures of dry saturated vapour. From the principle of increase of entropy, it is known that the temperature of the air increases and the dryness of the steam improves as they pass through the ejector as shown in Figure 6. It follows that the discharge temperatures depend on the energy balance of the fluids flowing in the ejector.

5.3 The Second-Stage Steam Ejector

Figures 18, 19, 20 and 21 are the results obtained from tests of the second-stage steam ejector at the different supporting pressures. The ejector has been tested with three different motive steam pressures.

A. Suction curves. Evidently, in Figure 18, the relationship between the mass flow of air and the pressure rise in the ejector, which is expressed by Equation (6), is true for all cases. However, in Figures 19, 20 and 21, the suction curves follow the relation of Equation (6) for a portion only, but they become flat at the low pressure region. This is due to choking of the air in the ejector. When the air passage is choked, the mass flow of the air is in a linear proportion to its upstream pressure, which is the suction pressure. Under such a situation, the relation of Equation (6) is no longer applicable, and it is observed that the suction pressure curves with different steam pressures coincide in this region.

B. Ejector efficiencies. The discussions on the discharge pressure and discharge temperature for the first-stage steam ejector are still applicable to the second-stage one. When the inlet and discharge conditions of the air and the steam have been measured, the ejector efficiencies of the second-stage are calculated from Equation (10). Efficiency curves are plotted in Figures 18, 19, 20 and 21 for steam pressure 154.6 psia.

Similar to an efficiency curve of a centrifugal pump, a point of maximum efficiency occurs between the point of maximum pressure rise ΔP in the ejector, and the point of maximum air flow.

5.4 Operation of Two Steam Ejectors in Series

The experimental results of the two steam ejectors operating in series with the different supporting pressures and different motive steam pressures are shown in Figures 22, 23, 24, 25 and 26.

A. Suction curves. The previous discussions on the suction curves for single-stage ejectors are still applicable to the two-stage case. In each of Figures 22, 23, 24 and 25, by choosing two separate points on the suction curve with steam pressure 104.6 psia., the constants of f'(a,b) and g(a,b) of Equation (6) have been determined. An assumption has been made in Chapter II, that the steam and air are only partially mixed. Therefore, the density for the air remains unchanged as the steam condition is shifted from 104.6 psia to 154.6 psia. The predicated curves are drawn as shown in Figures 22, 23, 24 and 25. The agreement between the experimental curves and the predicated ones is quite close.

B. Unstable operation. In Chapter II, Equation (6) shows, when two steam ejectors operate in series, their suction effects are additive. The suction effect of the first-stage ejector will decrease as the air flow increases. Figure 26 clearly shows these facts.

When the mass flow of air increases to a certain extent, the suction curve of the first-stage ejector intersects its supporting pressure curve which is the suction curve of the second-stage ejector at that instant. Operation of the ejector near this region is unstable. It has been noted during the tests that unstable operations are marked by cyclic variations in suction pressures. When the test is over, the observed readings of the over-all suction pressure are plotted onto a curve. By superimposing the curve of the supporting pressure of the first-stage ejector onto the over-all suction curve on the same graph, the exact region of the unstable operation of the first-stage ejector reveals itself. This has been done in Figures 22, 23, 24 and 25.

C. Ejector efficiencies. The previous discussions on the curves of discharge pressure and discharge temperature for single-stage ejectors are still applicable to the two-stage case.

Equation (12) is used to calculate the over-all ejector efficiencies of the two ejectors operating in series. As is shown in Figures 22, 23, 24 and 25, since the change of pressure ΔP in the ejectors becomes dominant, the values of the mass flow of air at the maximum efficiency is close to that where the pressure rise ΔP in the ejectors happens to be maximum.

D. Optimum operation. It has been mentioned at the beginning of Chapter I that a variety of forms of steam ejectors are used to overcome limitations with respect to the required degree of compression and flexibility. For example, Figure 25, when two ejectors operate in series, in order to avoid unstable operation, the optimum mass flow of air must be always less than 300 lbs. per hr. For the purpose of clarity, Figure 30 is drawn in accordance with Figure 25. The operation must follow line (1) as shown in Figure 30. Once, more air flow is required, the first-stage ejector must be shut down and the second-stage used to operate over the range. The suction curve then follows line (2) and stable operation is maintained. If a better suction pressure is required with further increasing of air flow, it is necessary to operate two ejectors in parallel in the first stage. In such a case, the suction pressure will follow line (3) as shown in the same figure. The third ejector is available for this mode of operation.

5.5 Condensers

The pressure drop across the condensers is negligible (i.e. less than 1 in. H_2O).

Performance curves of the condensers at different flow rates of condensing water are shown in Figures 27 and 28.

A. Vapour content in the exit air. At the given testing conditions, when the flow rate of water is reduced, the pressures in the condensers increase only slightly, while, the exit temperatures of the mixture of air and water vapour increase significantly. The water vapour content in the air at the exit of the condensers is calculated from the pressures and temperatures using Equation (13) and plotted in the same graphs.

B. Water requirements. By reducing the flow rate of condensing water, the drop off in the over-all suction curves is shown in Figure 29. This loss is caused by increasing vapour constant in the exit air thus imposing more load on each supporting stage, which, in turn, increases the supporting pressure. Therefore, the over-all suction curves obtained from less flow rate of water have similar form to that obtained from reducing steam pressure. It is seen in Figure 29 that the effect of reducing the flow rate of water intensifies around the air flow of 300 lbs. per hr. This is caused by the increased load imposed on the second-stage ejector by the increased vapour content coming from the inter-condenser.

C. Series flow of condensing water between condensers.

The vacuum in the inter-condenser is always higher than that of the after-condenser. In accordance with this fact, the exit temperature of the mixture of air and water vapour must be kept lower in the inter-condenser than that of the after-condenser in order to minimize the amount of vapour content in the exit air from the inter-condenser. Therefore, the series flow of condensing water, first to the inter-condenser and then to the after-condenser, is significant for economical considerations.

By comparing the temperature curves between Figures 27 and Figure 28, it is noted that the temperature curves of the after-condenser keep more flat. Moreover, the exit temperatures of the mixture are 14° to 22°F higher than that of the outlet temperature of the outlet temperature of the condensing water. Scale formation in the inter-condenser is suspected. (Recent inspection of the tubes has confirmed this suspicion).

CHAPTER IV

CONCLUSIONS

The following conclusions can be made regarding the performance, operating characteristics and stability of steam ejectors. These conclusions must be interpreted with care when applied to other than the McGill Hypersonic ejector system.

1. Under the same steam pressure when the mass flow of air increases, the pressure rise in the steam ejector decreases. This is a linear function. This relation is true till the choking of air occurs in the ejector. Once the air is choked, the mass flow of the air follows the familiar rule, i.e. the mass flow of air is in a linear proportion to the upstream pressure only.

2. For a certain mass flow of air, the suction pressure of a steam ejector depends on the motive steam pressure. Thus the higher the steam pressure, the lower is the suction pressure (to a limit determined by the geometry).

3. Conversely, increase of steam pressure increases the mass flow of air for a certain pressure rise in the steam ejector.

4. Once, a multi-stage steam ejector is constructed in series, suction effects due to each stage are additive.

5. Once the suction curve of a steam ejector intersects the curve of the supporting pressure, the operation of the steam ejector becomes unstable. Stable operation can then be obtained either by increasing the steam pressure or by reducing the mass flow of air.

6. The discharge pressure of a steam ejector is dependent on the supporting pressure, and it is always higher than the supporting pressure. The more the mass flow of air, the greater is the deviation between them.

7. When condensers are used between stages, for a certain mass flow of air through the ejectors, increased condensing water results in reducing steam consumption. The reverse is also true.

As previously stated the objective of this thesis is to investigate the performance, operating characteristics and stability of the vacuum system which has been set up already in the Hypersonic Laboratory of McGill by the hypersonic group. The above results, based on the experimental tests of the set-up and linked with the steam ejector principles listed above, lead to a complete understanding of the vacuum system. The optimum operation, which can be obtained with conditions of the motive steam and condensing water available at the time, is fully discussed in Section 4, Chapter V.

BIBLIOGRAPHY

1. Watson, F.R.B.
The Production of a Vacuum in an Air Tank by Means of a Steam Jet.
Inst. Mech. Eng., pp. 231-300 (1933).

2. Dawson, R.
Performance of Single-Stage Steam Jet Operated Ejectors.
The Engineer, Vol. 164, pp. 650-652, 680-681 (1937).

3. Work, L.T. and Haedrich, V.W.
Performance of Ejectors as a Function of the Molecular Weights of Vapours.
Ind. and Eng. Chem., Vol. 31, pp. 464-477 (1939).

4. Royds, R. and Johnson, E.
The Fundamental Principles of the Steam Ejector.
Inst. Mech. Eng., Vol. 145, pp. 193-209 (1941) and Vol. 146 pp. 223-235 (1941).

5. Flügel, G.
The Design of Jet Pumps.
N.A.C.A. Tech. Memo. No. 982 (1941).

6. Keenan, J.H. and Newmann, E.P.
A simple Air Ejector.
Trans. A.S.M.E. Vol. 64, pp. A75-81 (1942).

7. Elrod, H.G.
The Theory of Ejectors.
Trans. A.S.M.E. Vol. 67, pp. A170-174 (1945), and Vol. 68 pp. A162-165 (1946).

8. Turner, L.R., Adie, A.T. and Zimmerman, R.H.
Charts for the Analysis of One-Dimensional Steady Compressible Flow.
N.A.C.A. Tech. Note No. 1419 (1948).

9. Keenan, J.H., Newmann E.P. and Lustwerk, F.
An Investigation of Ejector Design by Analysis and Experiment.
Trans. A.S.M.E. Vol. 72, pp. A299-309 (1950).

10. Lane, A.G.
A Study of the Air Ejector.
McGill Master of Engineering Thesis (1950).

11. Johanneson, N.H.
Ejector Theory and Experiments.
Trans. Danish Acad. Techn. Sciences, A.T.S. No. 1 (1951).

12. Tucker, H.J.
Constant Diameter Air Injector.
McGill Master of Engineering Thesis (1955).

13. Fabri, J.F. and Paulon, J.
Theory and Experiments on Supersonic Air-to-Air Ejector.
N.A.C.A. Tech. Memo. No. 1410 (1958).

14. Streeter, V.L.
Fluid Dynamics.
McGraw-Hill Book Co., (1948).

15. Pei, S.I.
Viscous Flow Theory - Laminar Flow.
D. Van Nostrand Co., (1956).

16. A.S.M.E. Research Publication: Fluid Meters, Part 1, 4th Edition.

17. Zsombor-Murray, P.
Orifice Design.
McGill Mech. Engng. Res. Lab. Rept. SCS 11 (1959).

18. Heat Exchange Institute: Standards of Heat Exchange Institute, Steam Jet Ejectors, 3rd Edition, (1956).

19. Keenan, J.H. and Keyes, F.G.
Thermodynamic Properties of Steam.
John Wiley and Sons, Inc., (1957).

20. Keenan, J.H. and Keyes, F.G.
Gas Tables.
John Wiley and Sons, Inc., (1957).

21. Shapiro, A.H.
The Dynamics and Thermodynamics of Compressible Fluid Flow, Vol. 1.
The Ronald Press Co., (1954).

22. Binder, R.C.
Advanced Fluid Dynamics and Fluid Machinery.
Prentice-Hall, Inc., (1957).

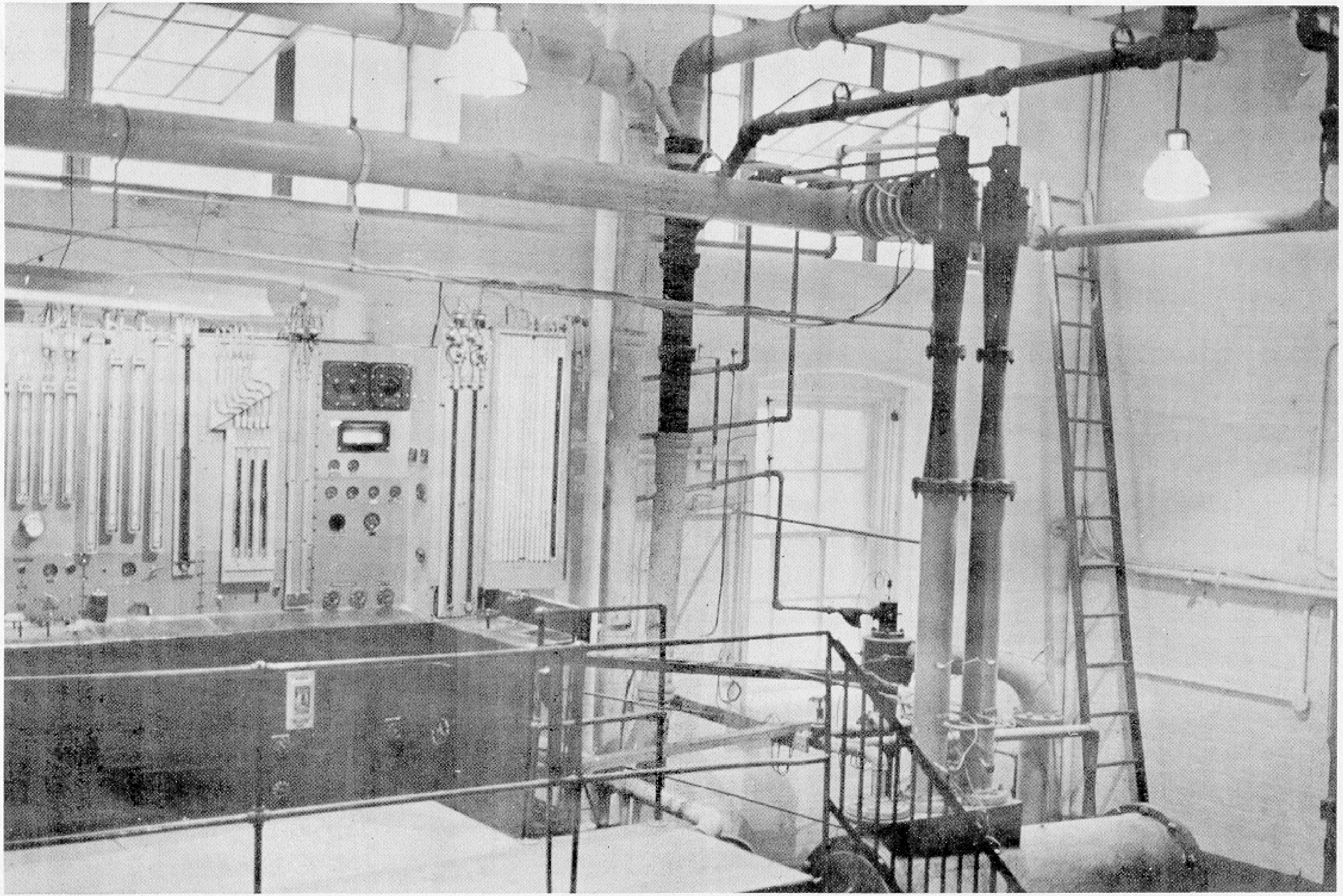


FIGURE 1. General View

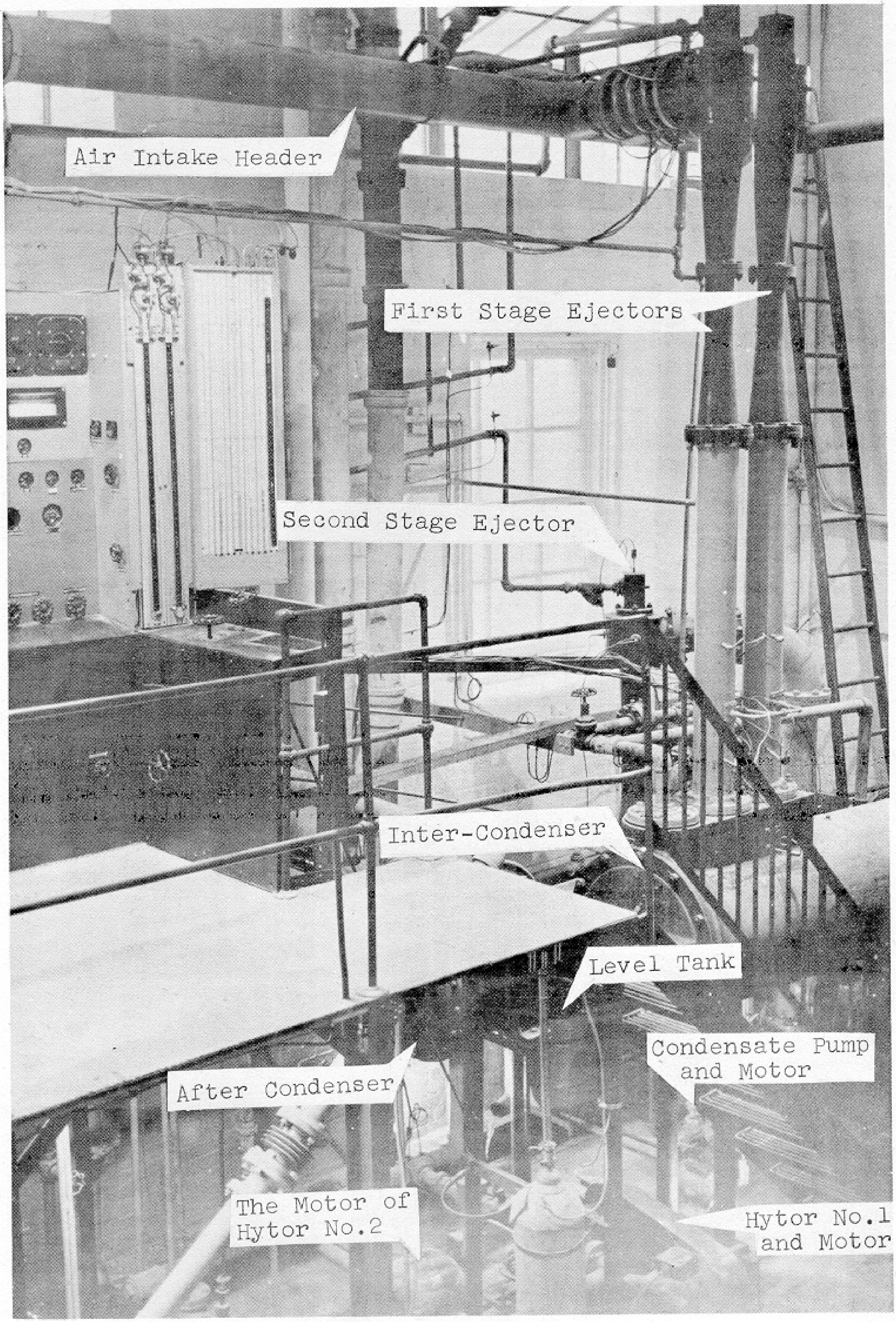


FIGURE 2. Vacuum System

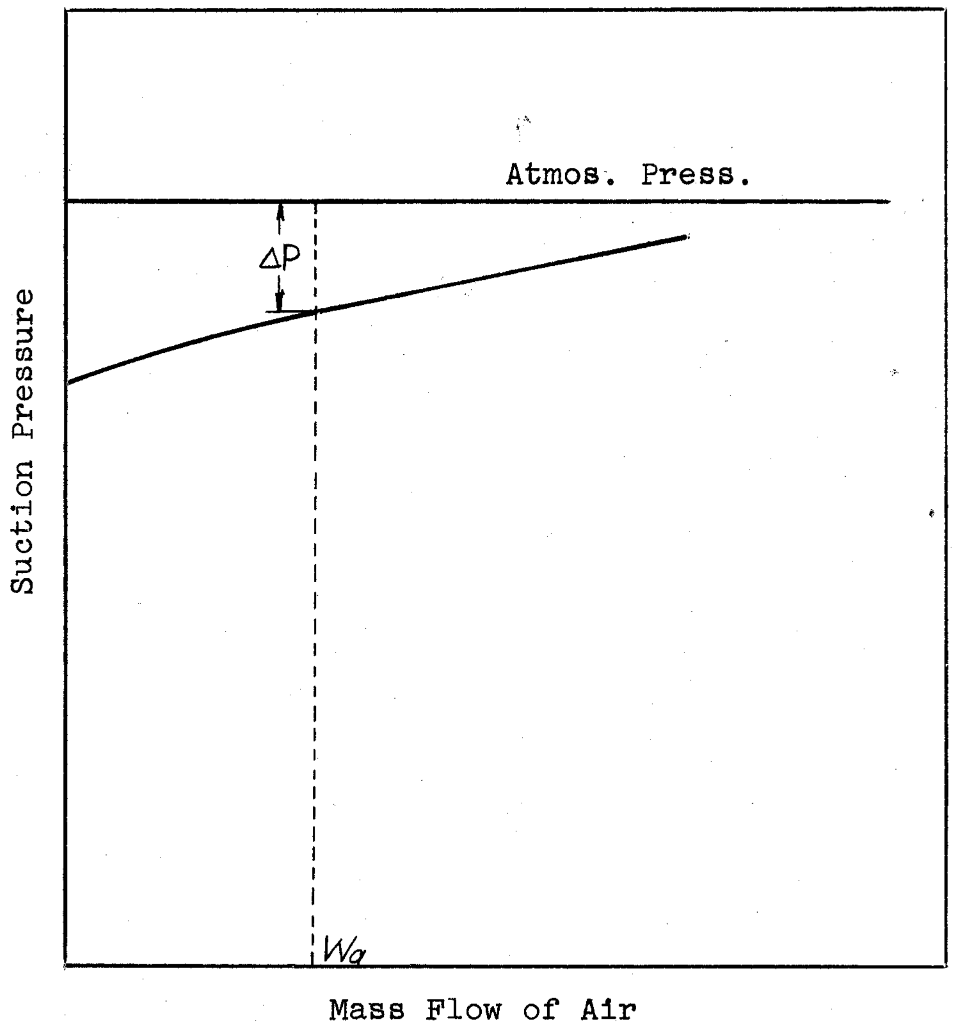


FIGURE 3. Relation Between Suction Pressures and Mass Flow of Air

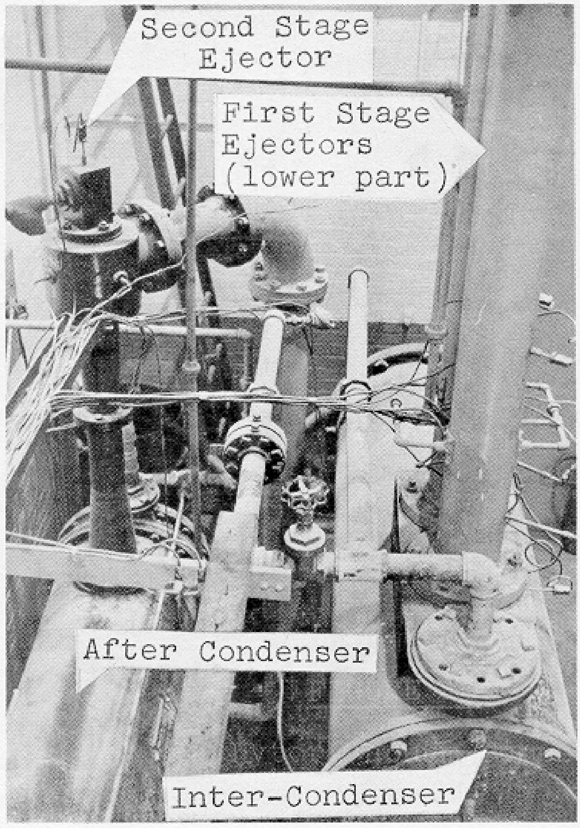


FIGURE 9.

Photograph of the Two-Stage Steam Ejector showing the Actual Arrangement.

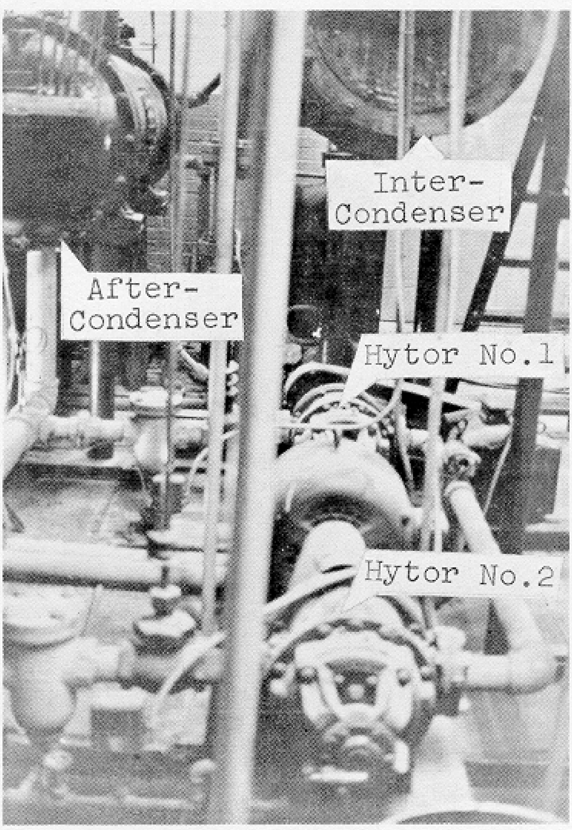


FIGURE 10

Hytor Vacuum Pumps

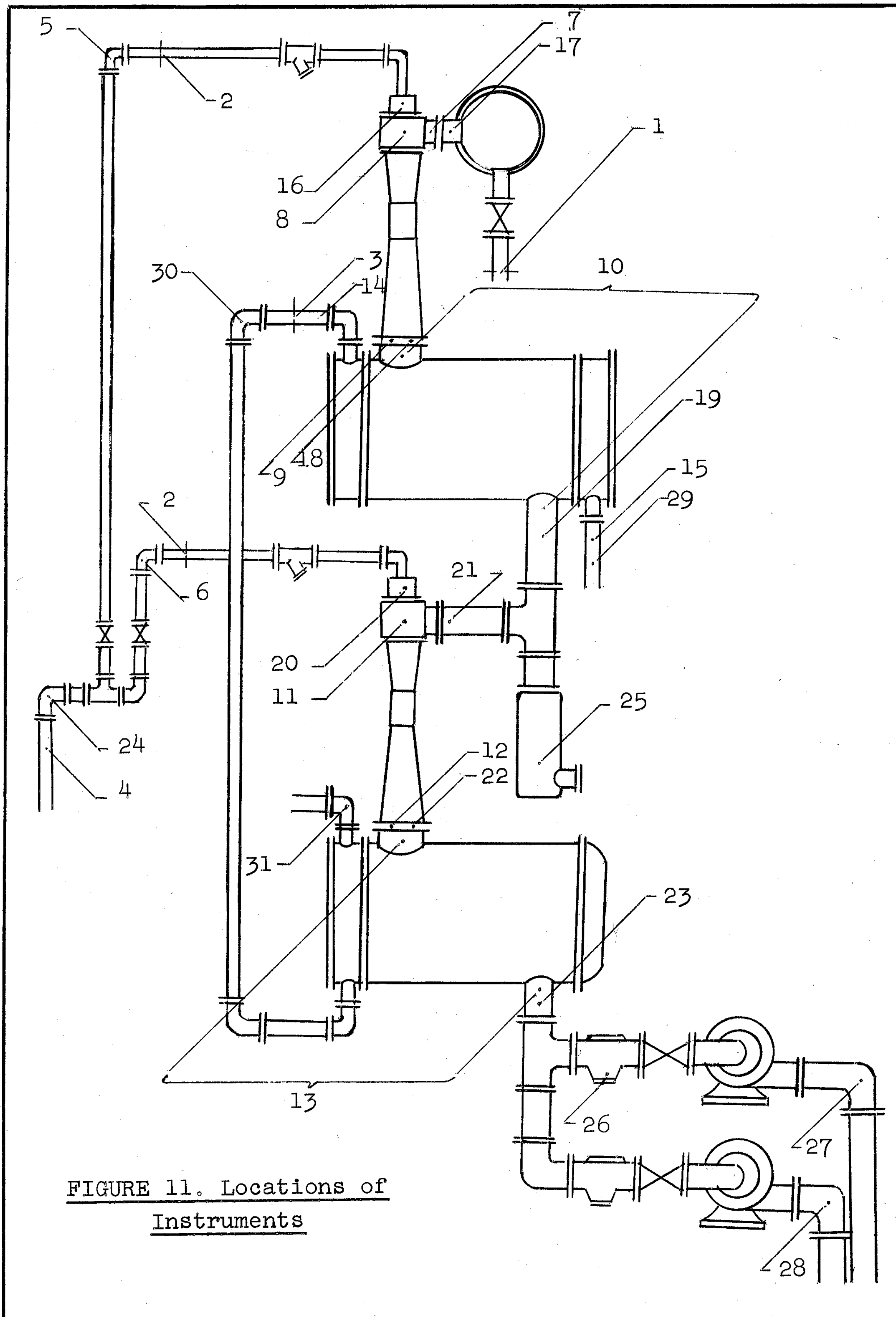
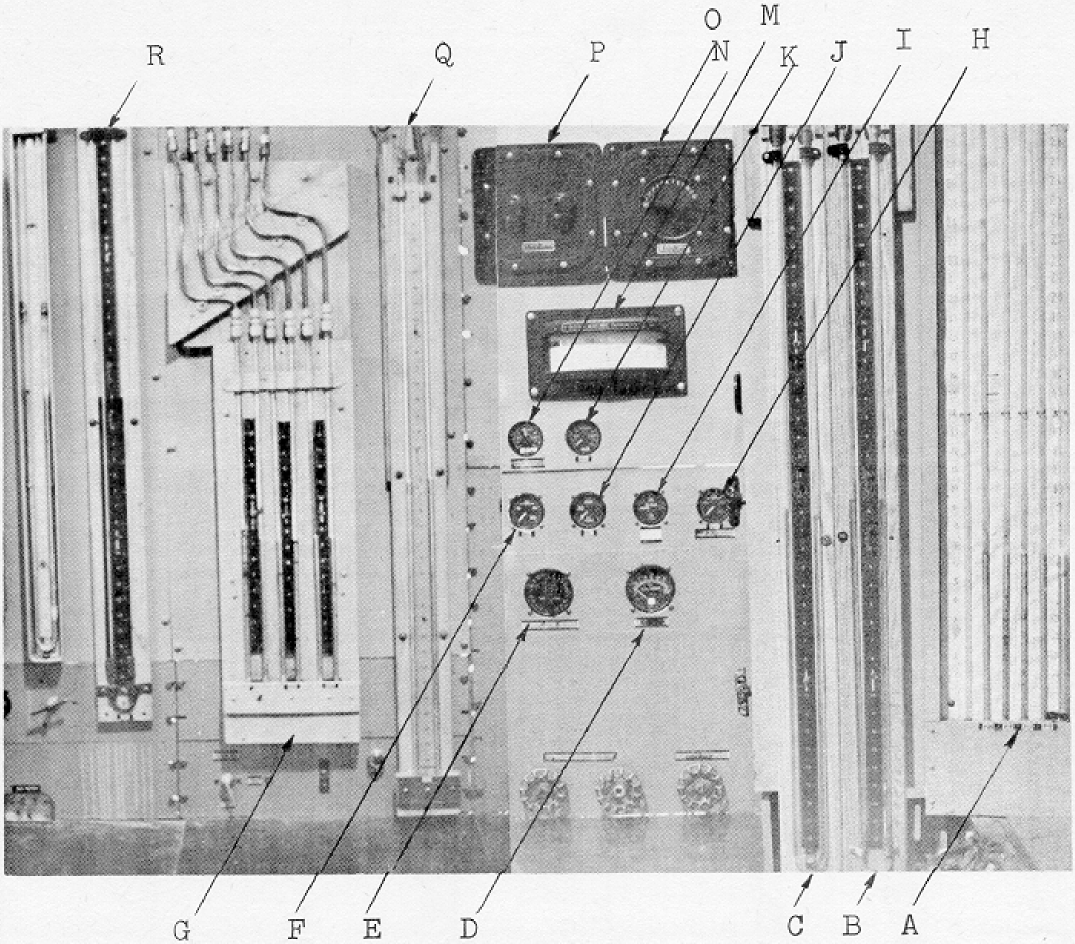


FIGURE 11. Locations of Instruments



A - Mercury manometer set.
B - Differential manometer of inter condenser.
C - Differential manometer of after condenser.
D - Dial of thermocouples
E - Selective switch of thermocouple.
F - Borden gauge of 1st-stage steam.
G - Steam flow meters
H - Borden gauge of 2nd-stage steam.
I - Borden gauge of condensing water between condensers.
J - Borden gauge of 1st-stage steam.
K - Borden gauge of inlet water.
M - Dial of resistance thermometer.
N - Borden gauge at steam main.
O - Selective switch of thermometer.
P - Standardizing box of resistance thermometer.
Q - Flow meter of condensing water.
R - Flow meter of air.

FIGURE 12. Photograph of Rig Showing the Elevation of the Panel Board

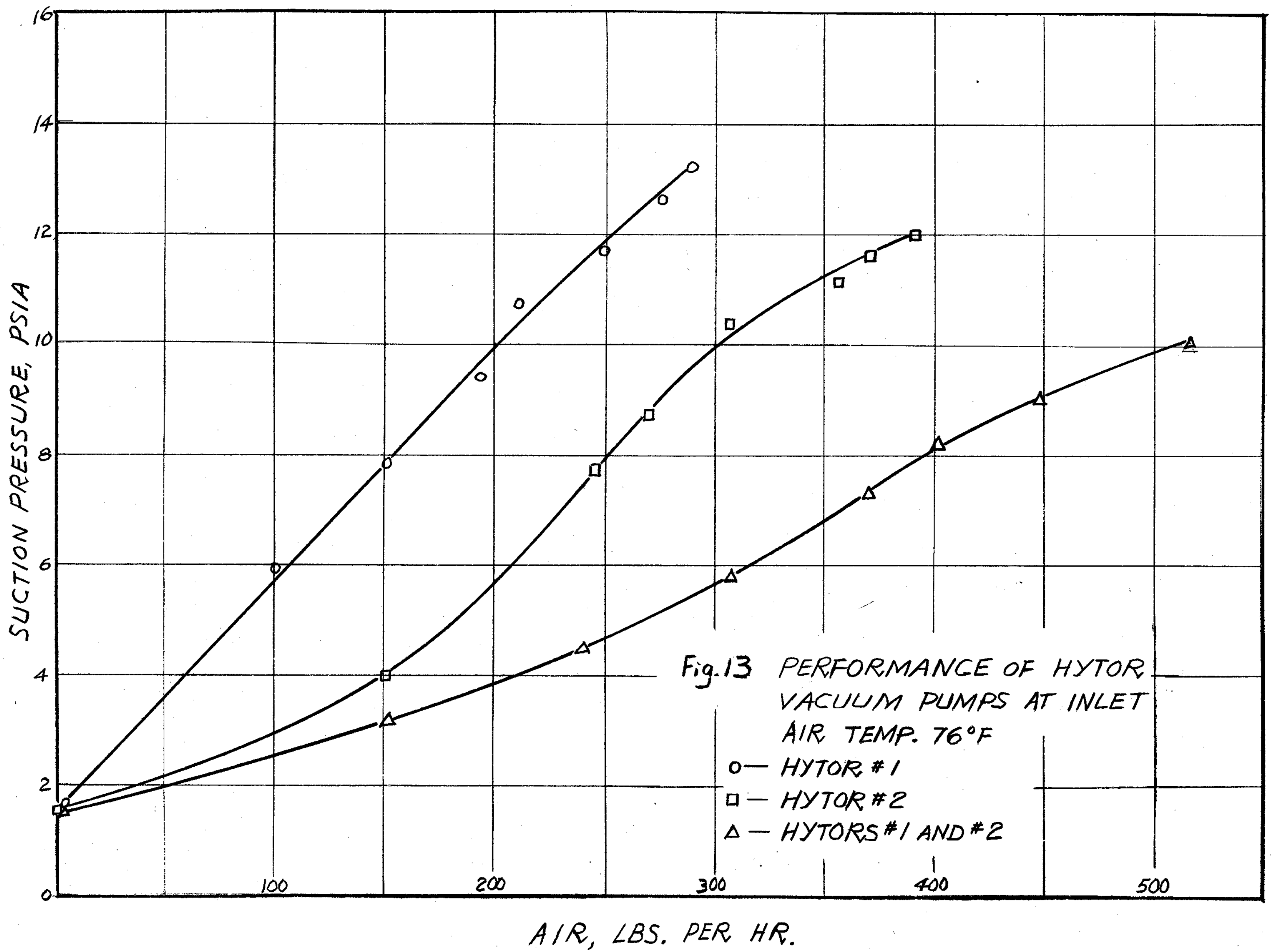


Fig. 13 PERFORMANCE OF HYTOR VACUUM PUMPS AT INLET AIR TEMP. 76°F

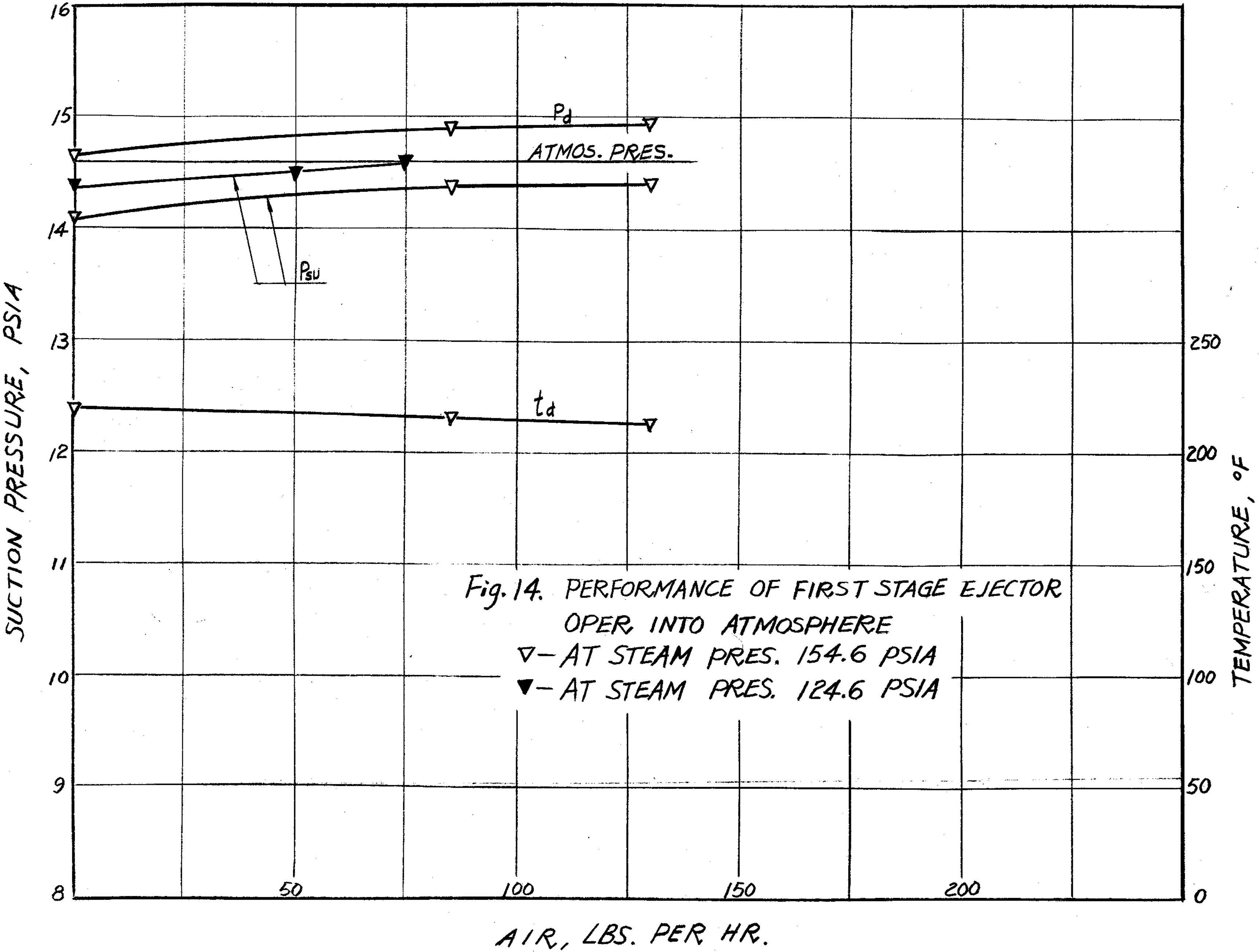


Fig. 14. PERFORMANCE OF FIRST STAGE EJECTOR OPER. INTO ATMOSPHERE
▽ – AT STEAM PRES. 154.6 PSIA
▼ – AT STEAM PRES. 124.6 PSIA

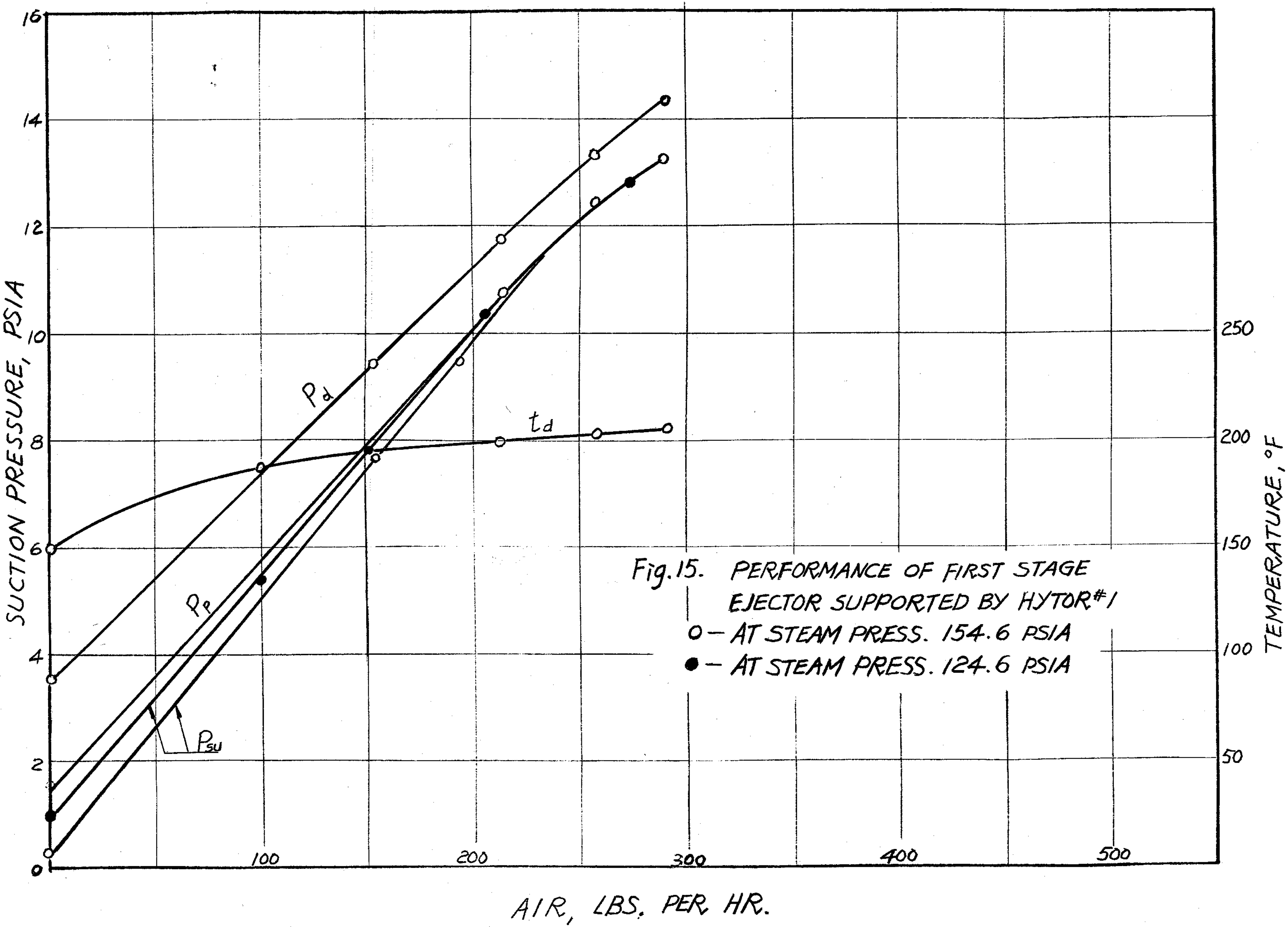


Fig. 15. PERFORMANCE OF FIRST STAGE EJECTOR SUPPORTED BY HYTOR #1

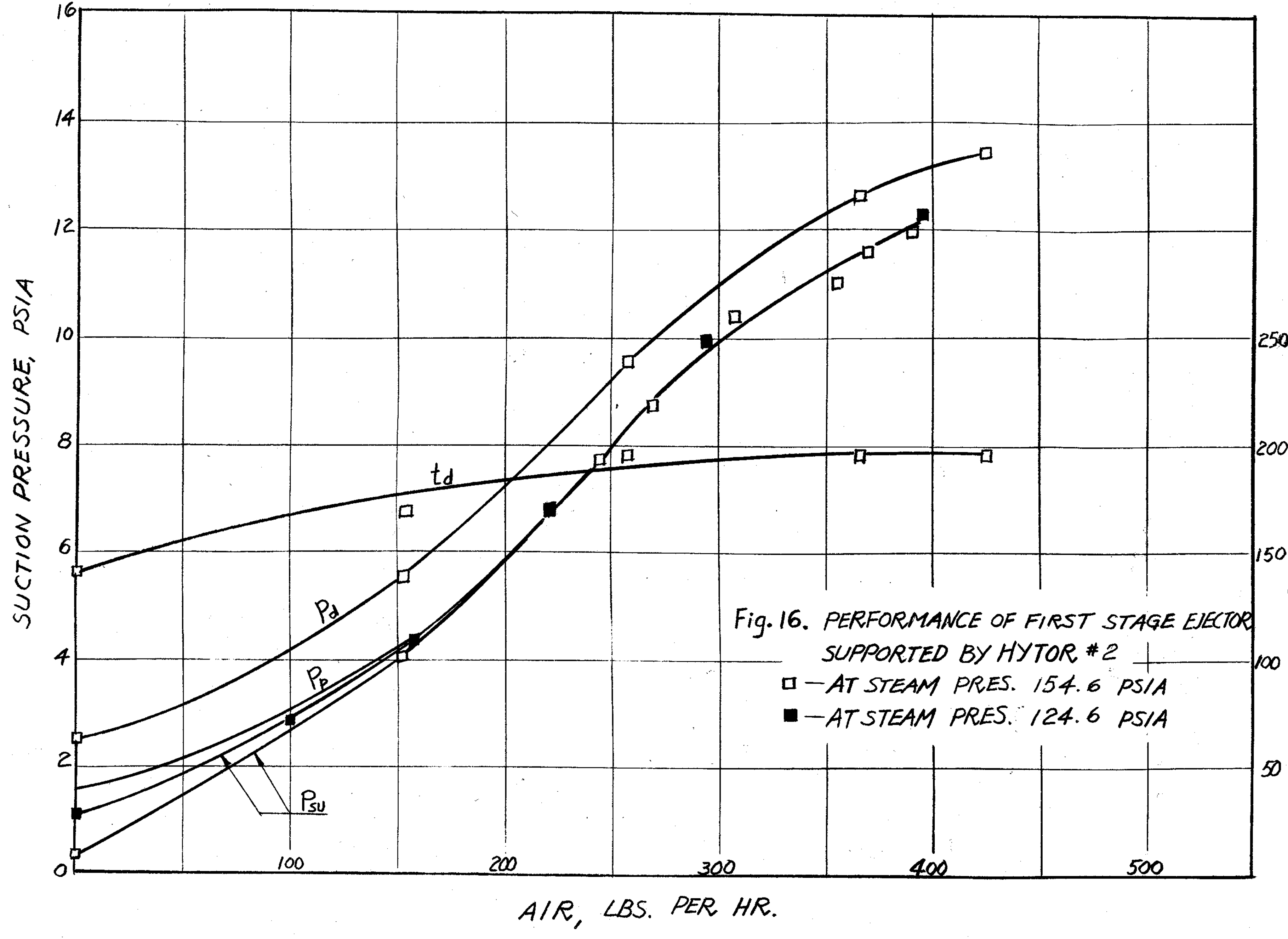


Fig. 16. PERFORMANCE OF FIRST STAGE EJECTOR SUPPORTED BY HYTOR #2

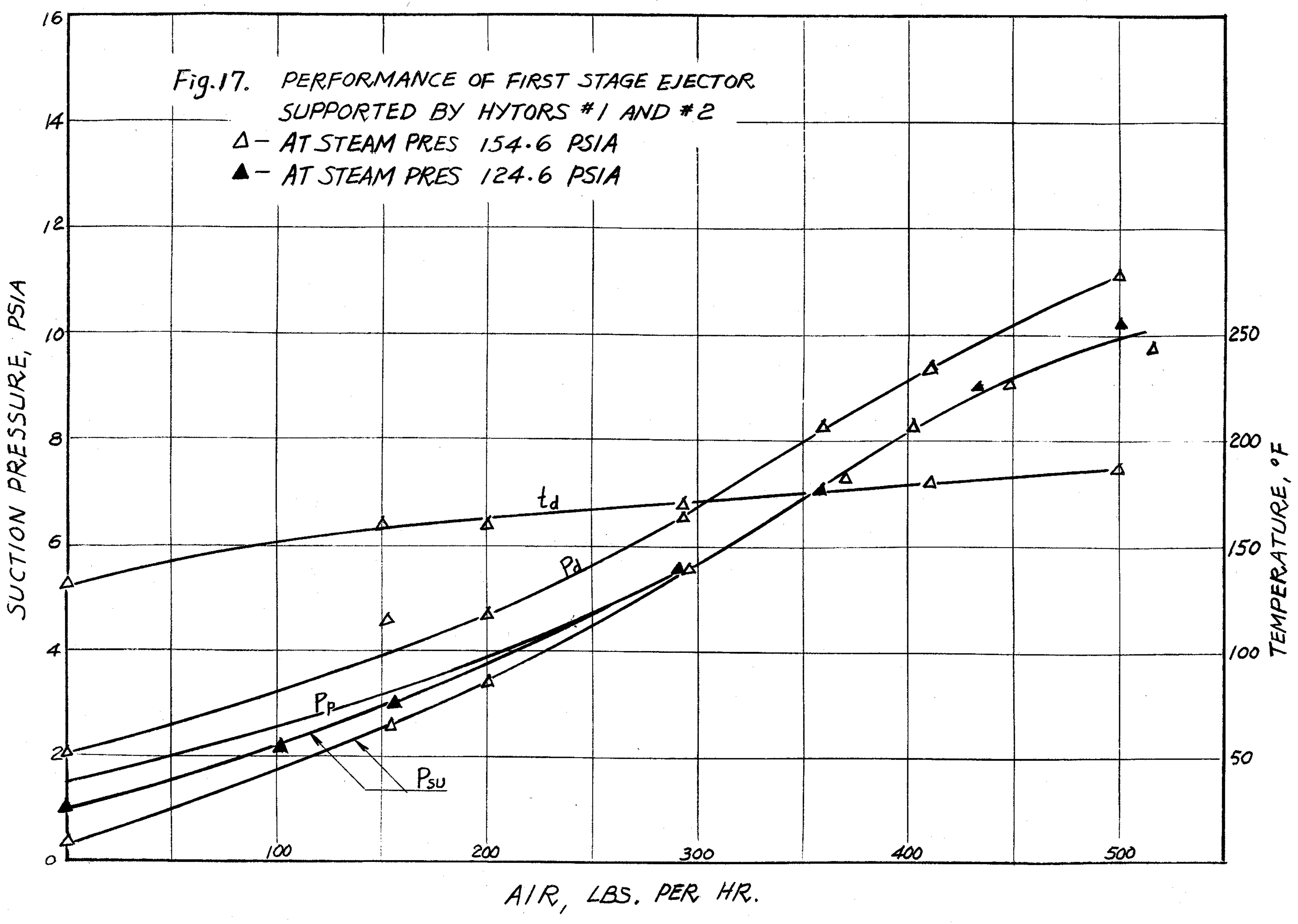


Fig. 17. PERFORMANCE OF FIRST STAGE EJECTOR SUPPORTED BY HYTORS #1 AND #2
Δ – AT STEAM PRES 154.6 PSIA
▲ – AT STEAM PRES 124.6 PSIA
SUCTION PRESSURE, PSIA
TEMPERATURE, °F
AIR, LBS. PER HR.
t_d
P_d
P_p
P_{su}
0
2
4
6
8
10
12
14
16
100
200
300
400
500
50
100
150
200
250

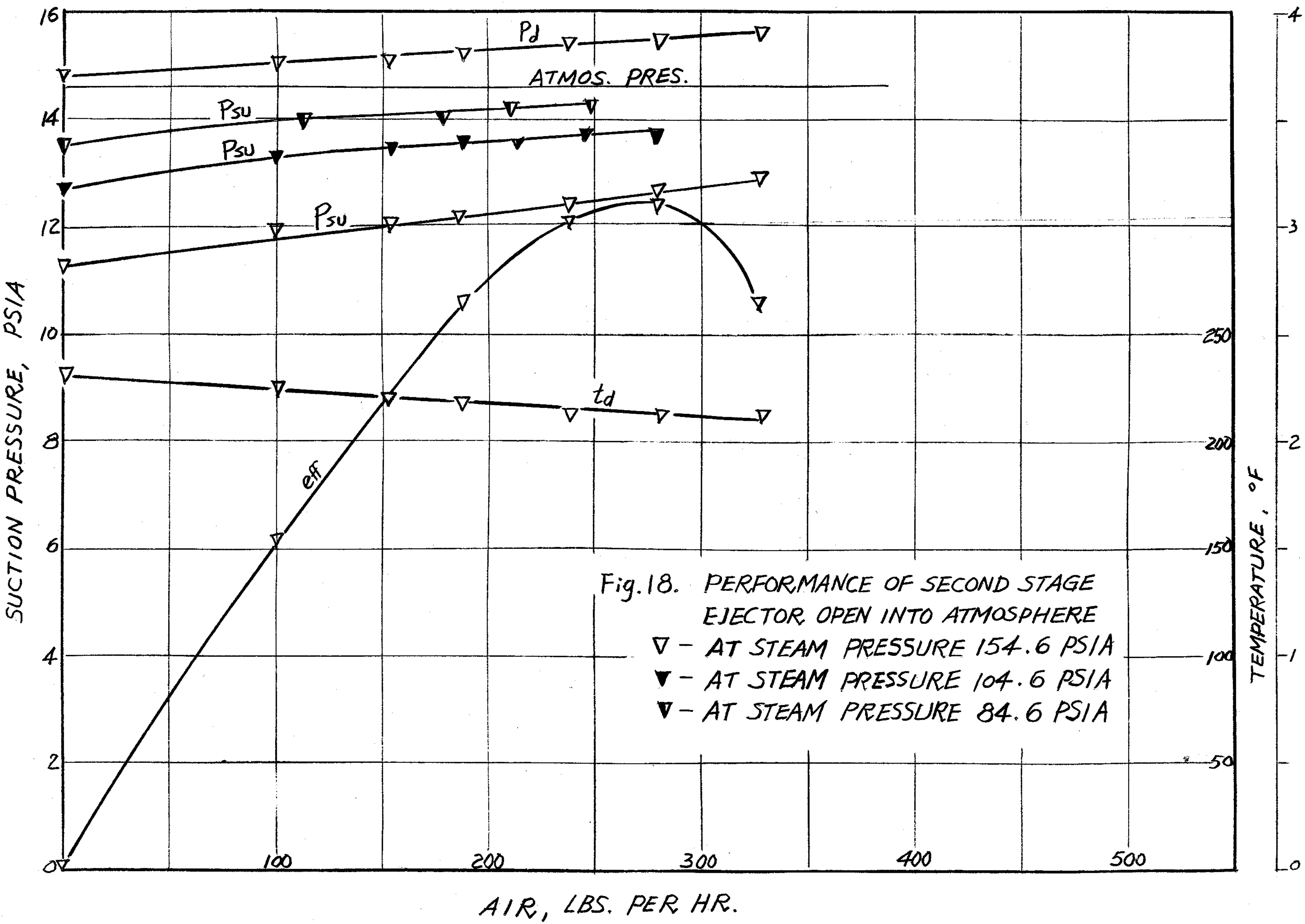
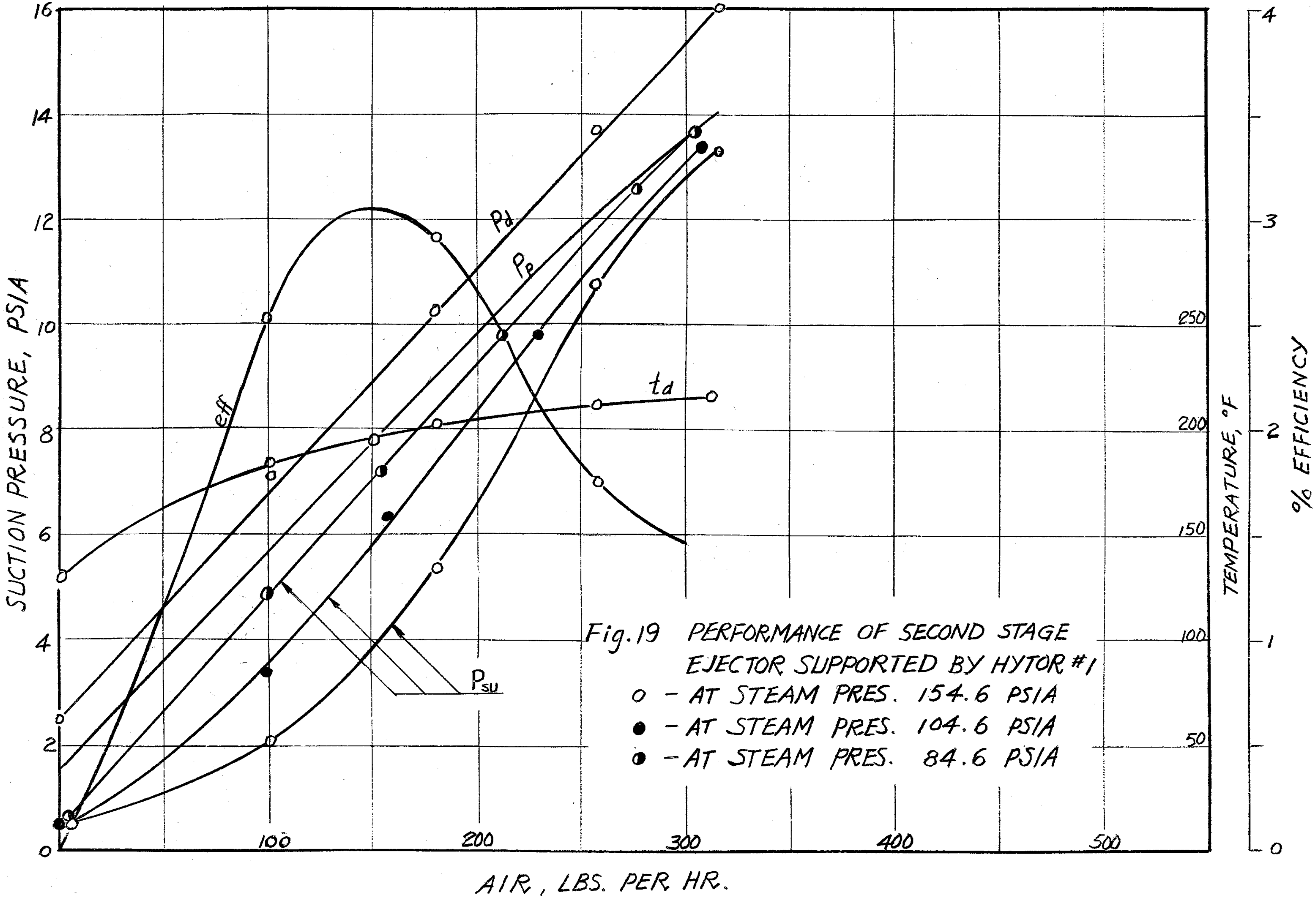


Fig. 18. PERFORMANCE OF SECOND STAGE EJECTOR OPEN INTO ATMOSPHERE



SUCTION PRESSURE, PSIA
AIR, LBS. PER HR.
TEMPERATURE, °F
% EFFICIENCY
eff
Pd
Pp
td
Psu
0
2
4
6
8
10
12
14
16
100
200
300
400
500
50
100
150
200
250
1
2
3
4
Fig. 19 PERFORMANCE OF SECOND STAGE EJECTOR SUPPORTED BY HYTOR #1
○ - AT STEAM PRES. 154.6 PSIA
● - AT STEAM PRES. 104.6 PSIA
◐ - AT STEAM PRES. 84.6 PSIA

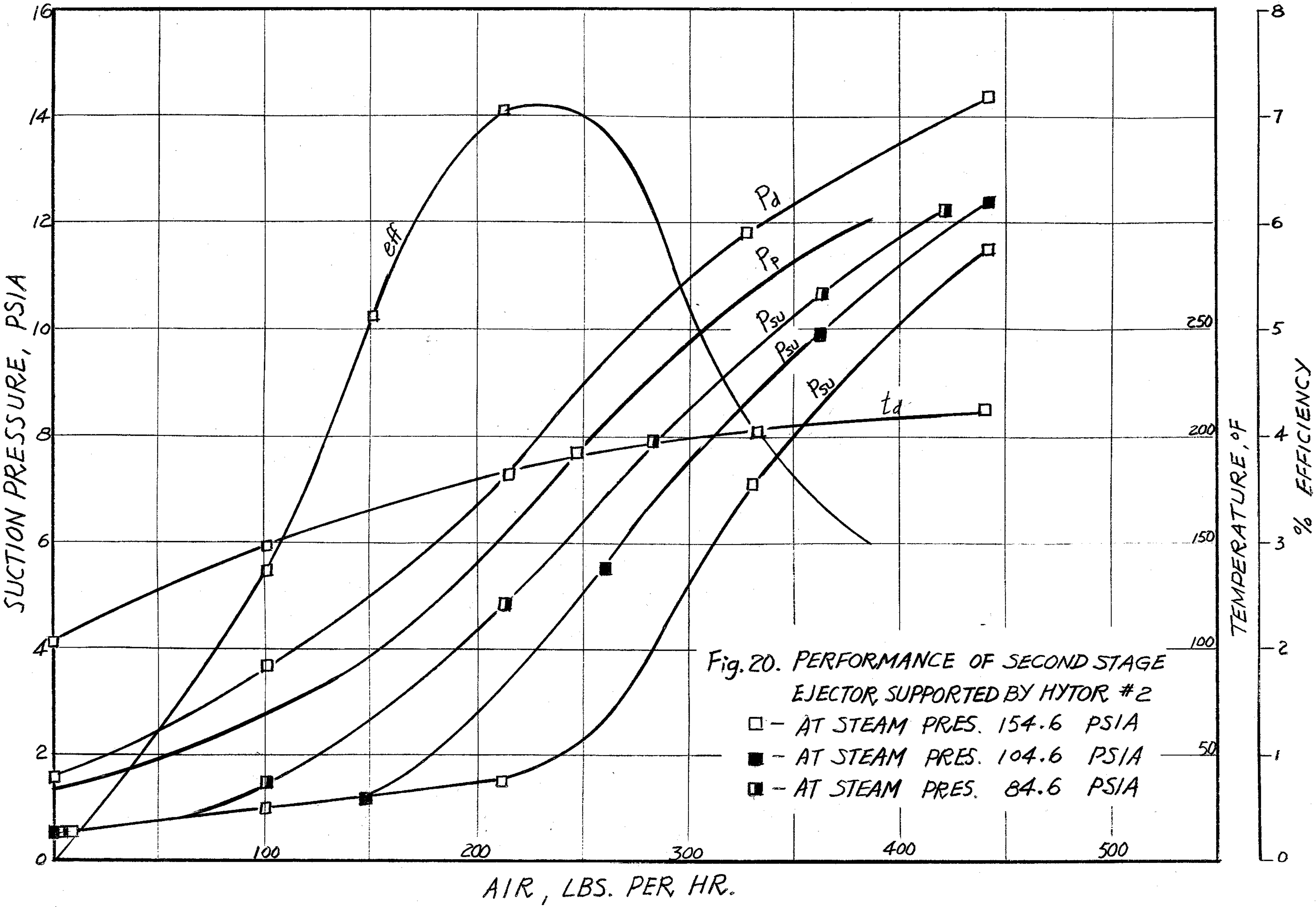


Fig. 20. PERFORMANCE OF SECOND STAGE EJECTOR SUPPORTED BY HYTOR #2

□ – AT STEAM PRES. 154.6 PSIA

■ – AT STEAM PRES. 104.6 PSIA

◧ – AT STEAM PRES. 84.6 PSIA

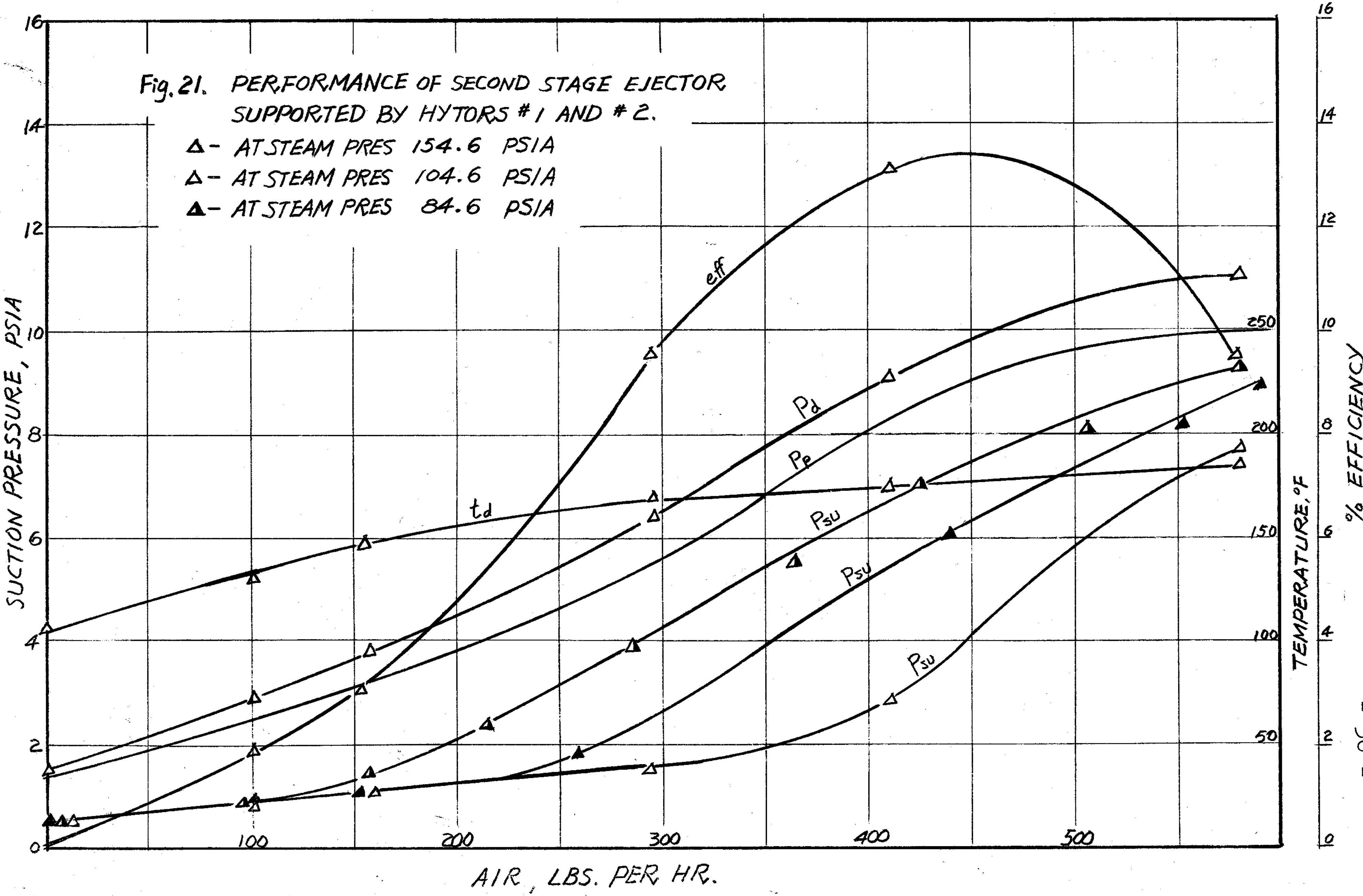


Fig. 21. PERFORMANCE OF SECOND STAGE EJECTOR SUPPORTED BY HYTORS # 1 AND # 2.

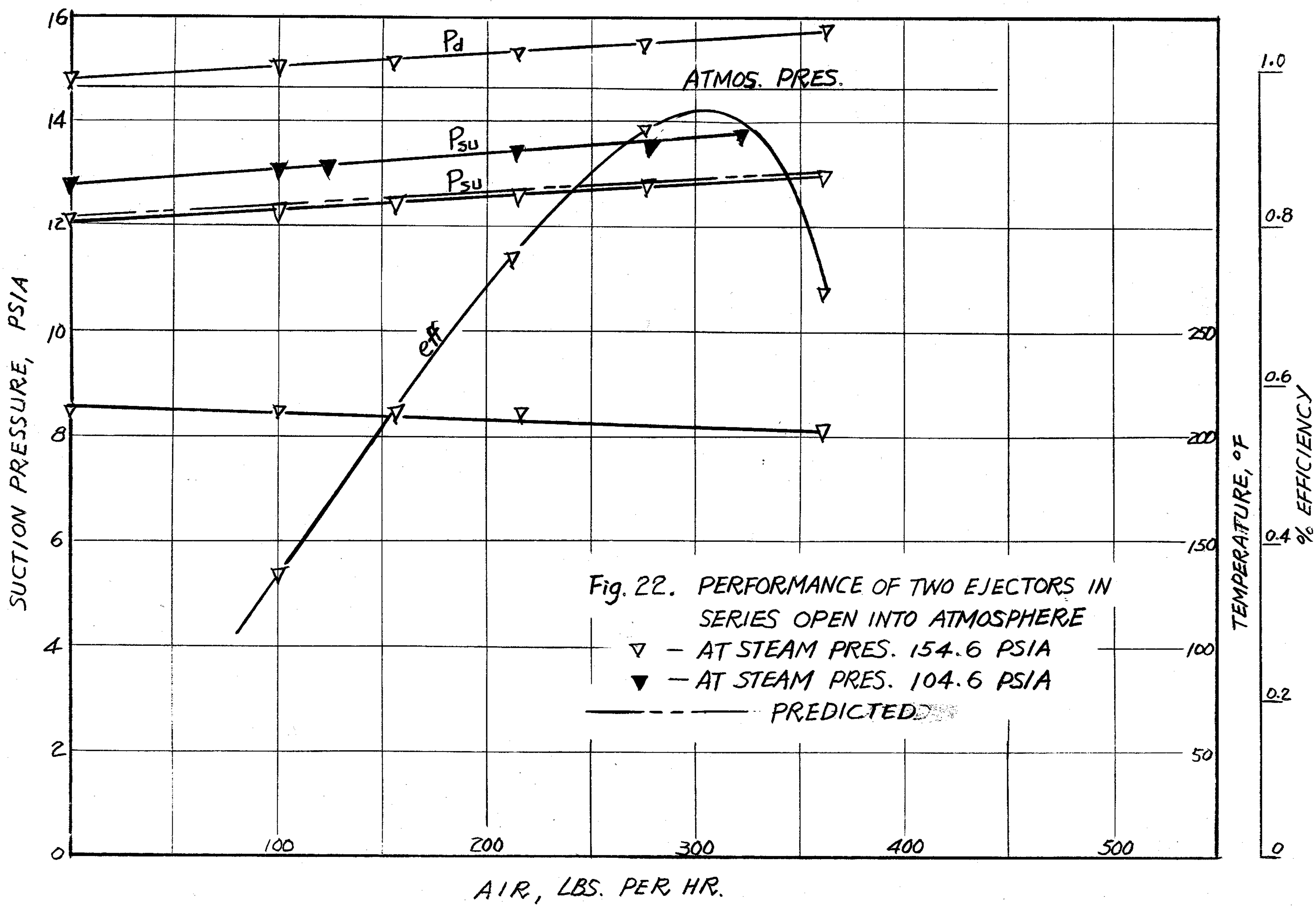


Fig. 22. PERFORMANCE OF TWO EJECTORS IN SERIES OPEN INTO ATMOSPHERE

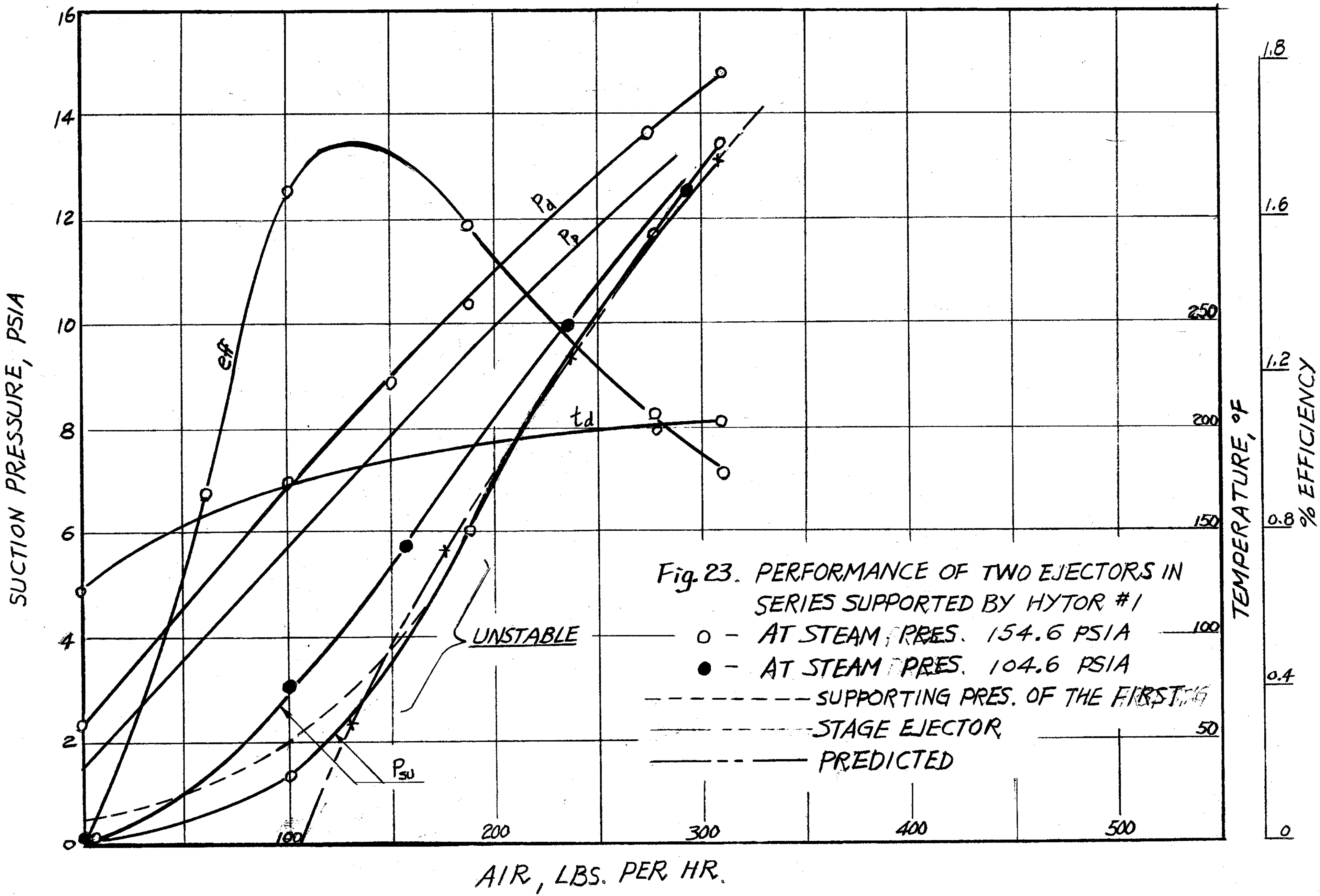


Fig. 23. PERFORMANCE OF TWO EJECTORS IN SERIES SUPPORTED BY HYTOR #1

FIGURE 24. Performance of Two Ejectors in Series Supported by Hytor No. 2.

□ - At Steam Pressure 154.6 psia.

■ - At Steam Pressure 104.6 psia.

--------- Supporting Pressure of the First Stage Ejector.

———— Predicted.

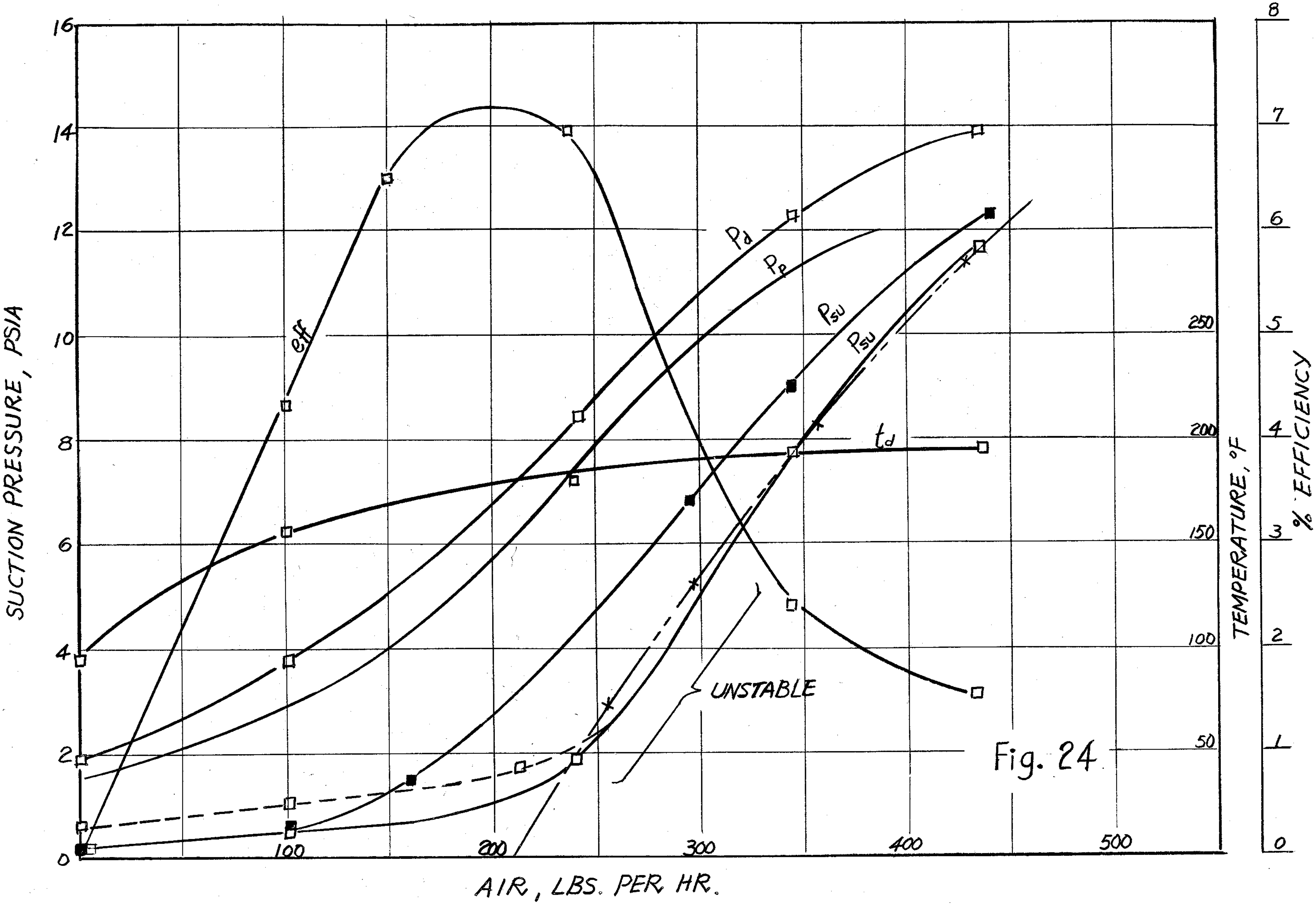

SUCTION PRESSURE, PSIA
0
2
4
6
8
10
12
14
16
100
200
300
400
500
AIR, LBS. PER HR.
TEMPERATURE, °F
50
100
150
200
250
% EFFICIENCY
0
1
2
3
4
5
6
7
8
eff
P_d
P_p
P_{su}
P_{su}
t_d
UNSTABLE

Fig. 24

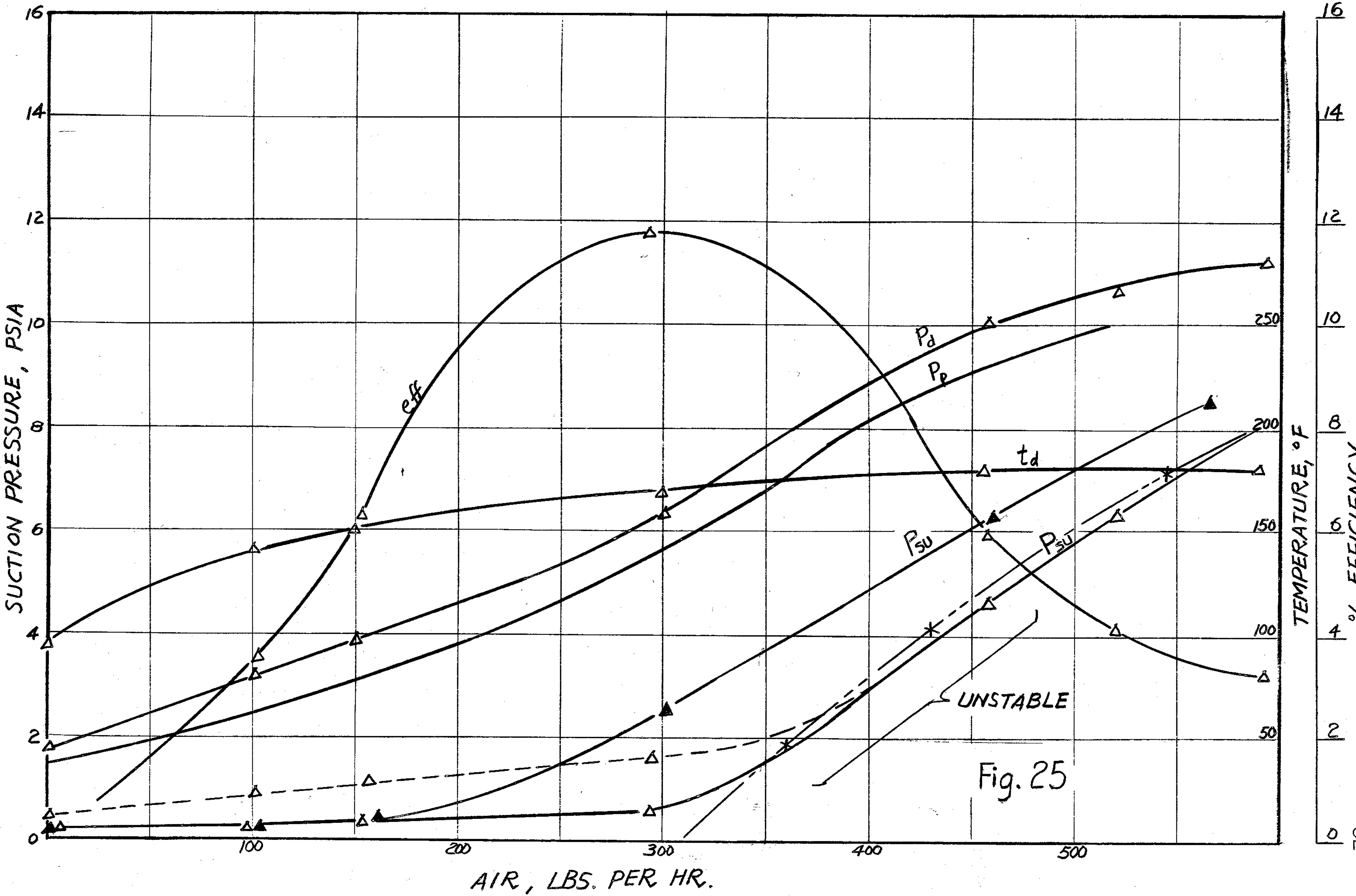
FIGURE 25. Performance of Two Ejectors in Series Supported by Hytors No. 1 and No. 2

△ - At Steam Pressure 154.6 psia.

▲ - At Steam Pressure 104.6 psia.

-------- Supporting Pressure of the First Stage Ejector

— - — Predicted.



SUCTION PRESSURE, PSIA
AIR, LBS. PER HR.
TEMPERATURE, °F
% EFFICIENCY
eff
P_d
P_p
t_d
P_{su}
P_{su}
UNSTABLE
0
2
4
6
8
10
12
14
16
100
200
300
400
500
50
100
150
200
250

Fig. 25

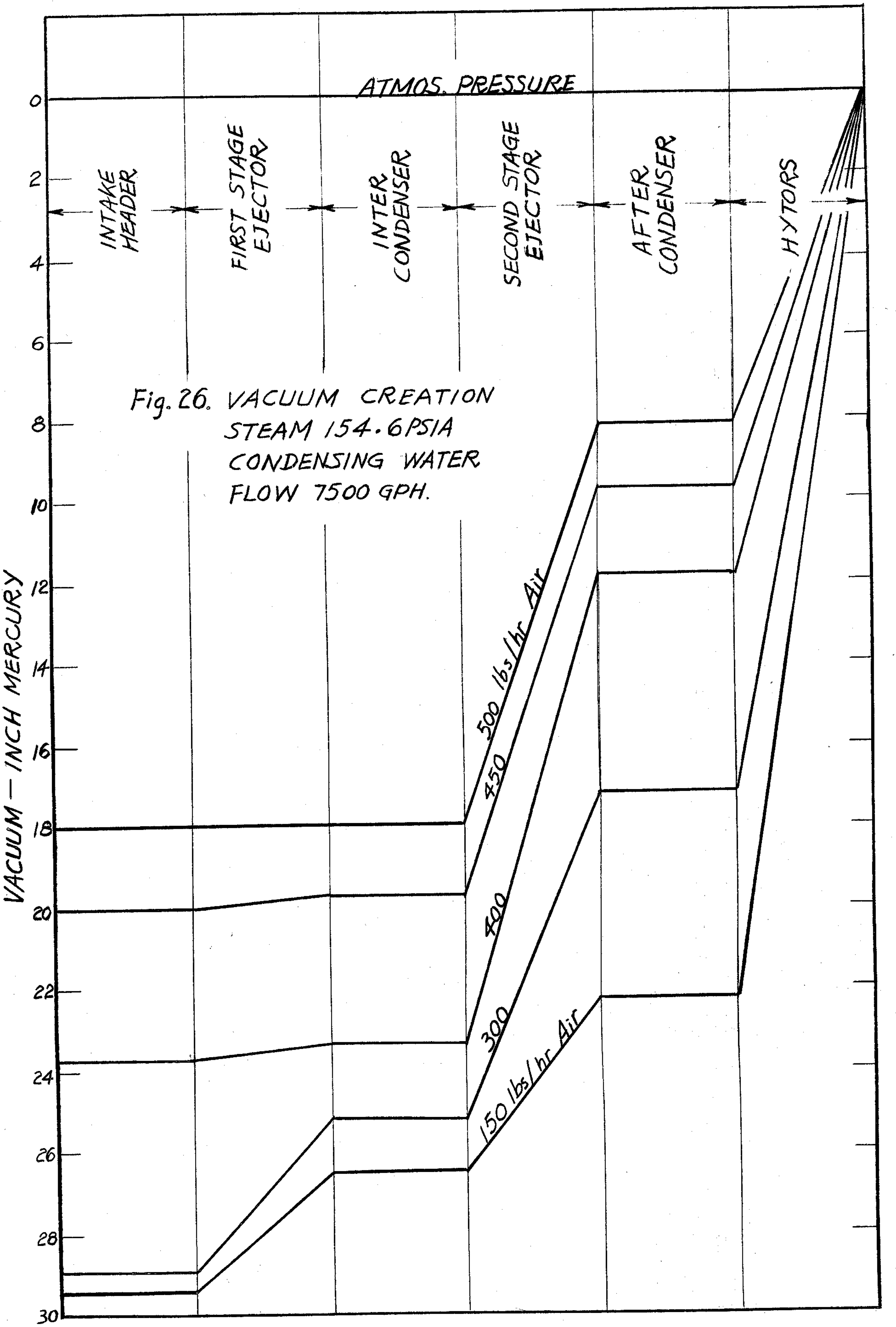


Fig. 26. VACUUM CREATION
STEAM 154.6 PSIA
CONDENSING WATER
FLOW 7500 GPH.

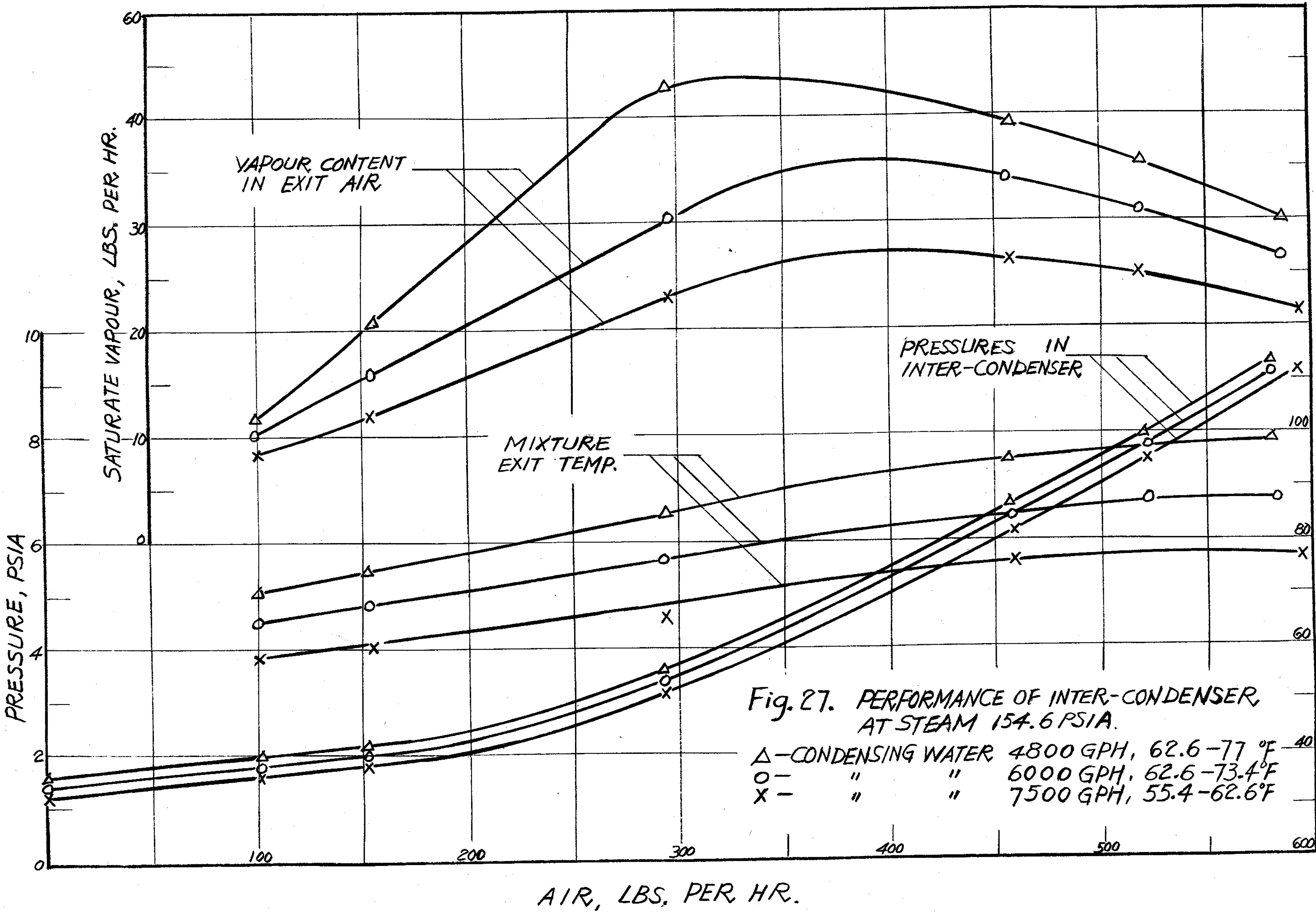


Fig. 27. PERFORMANCE OF INTER-CONDENSER AT STEAM 154.6 PSIA.

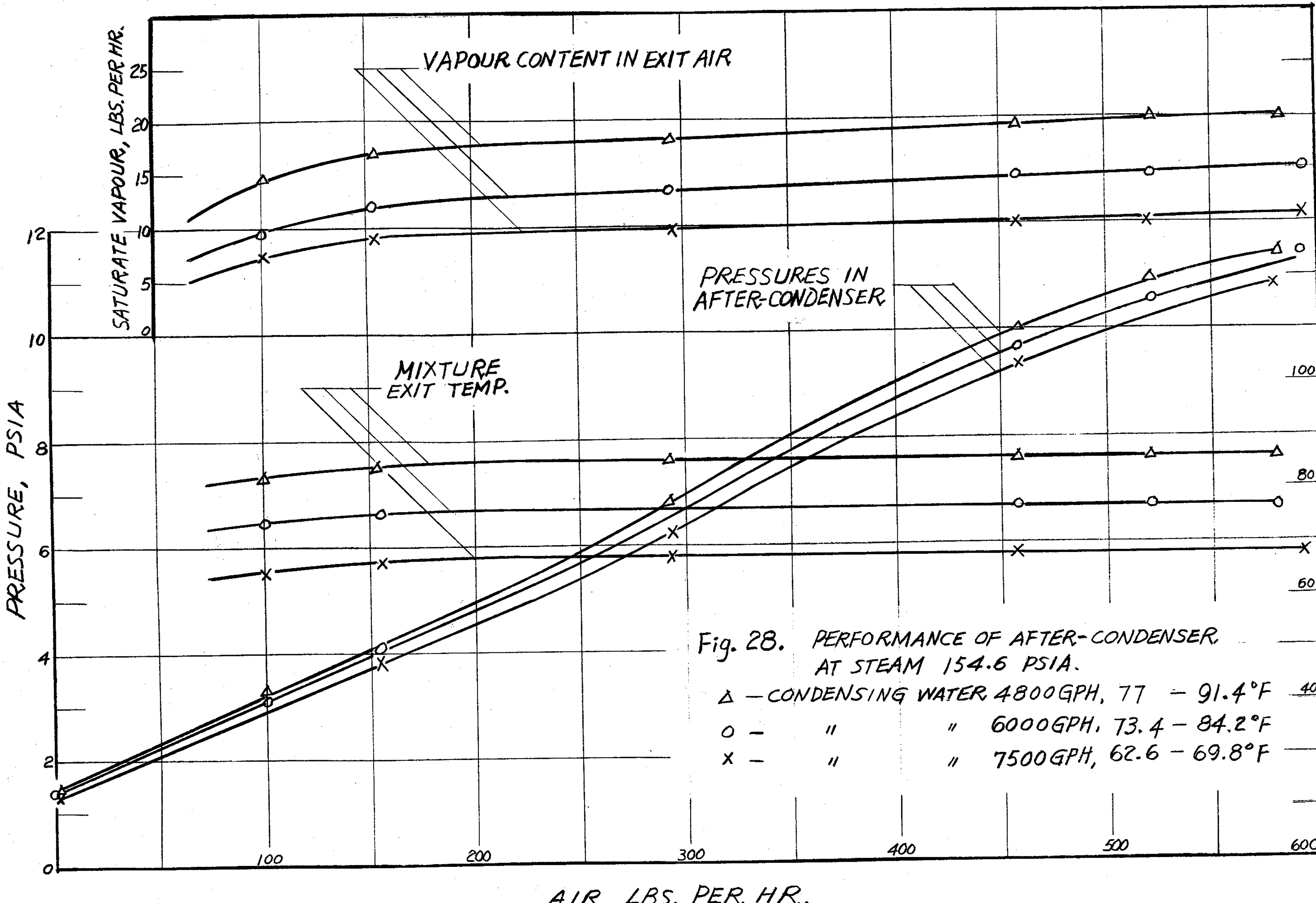


Fig. 28. PERFORMANCE OF AFTER-CONDENSER AT STEAM 154.6 PSIA.

△ — CONDENSING WATER 4800 GPH, 77 – 91.4°F
○ — " " 6000 GPH, 73.4 – 84.2°F
× — " " 7500 GPH, 62.6 – 69.8°F

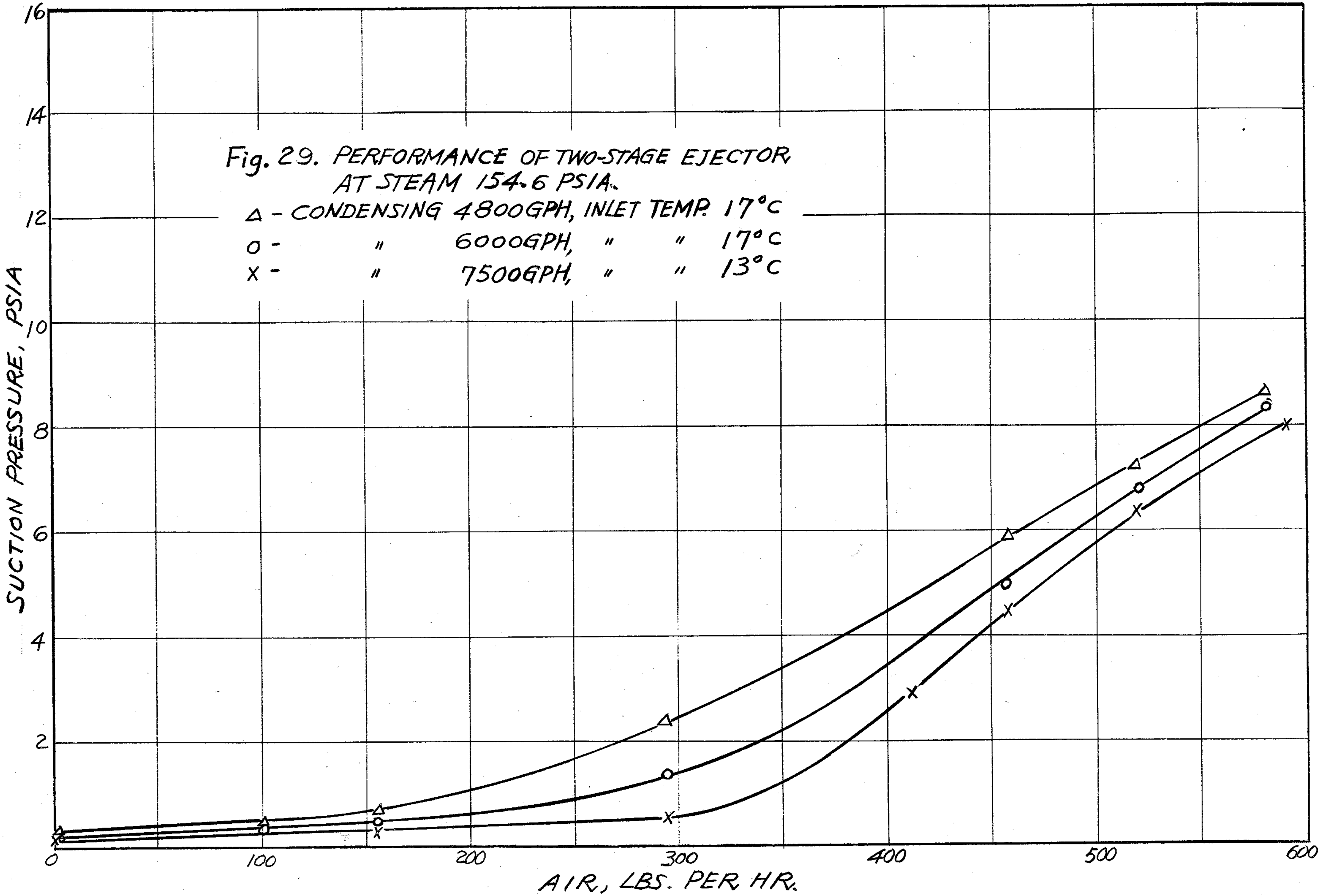
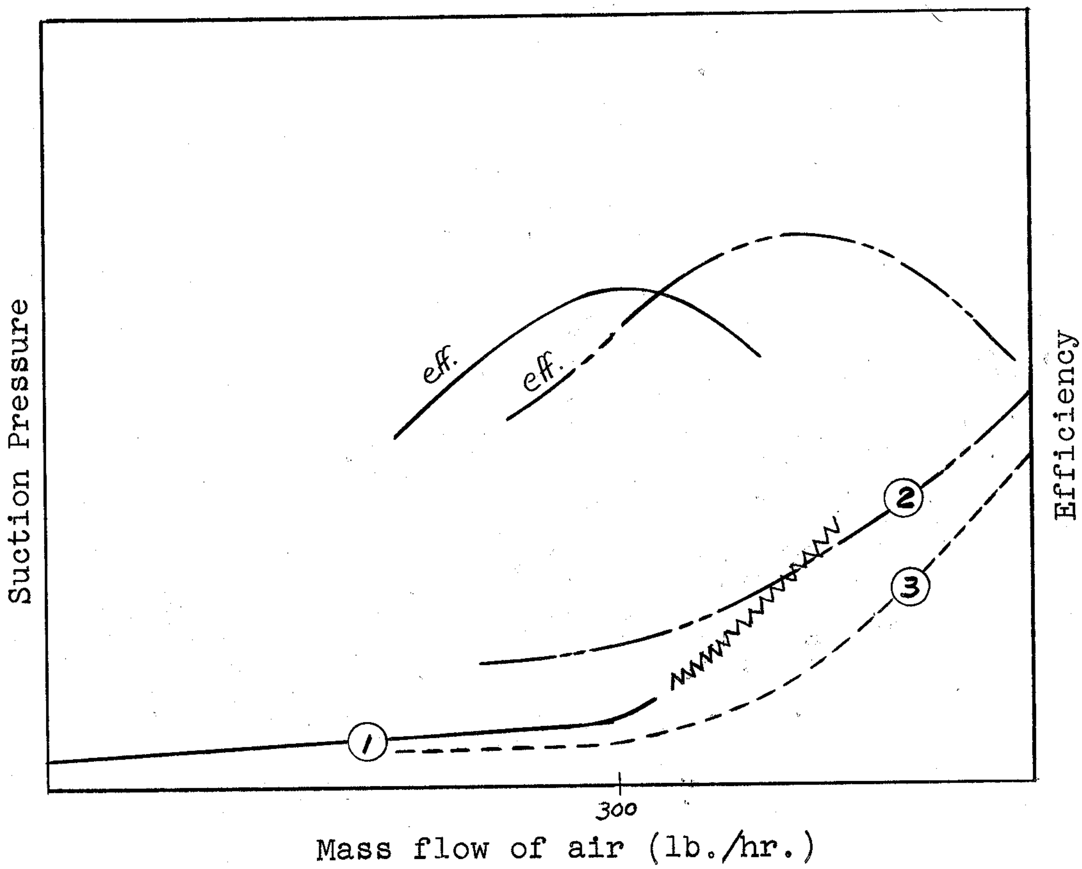


Fig. 29. PERFORMANCE OF TWO-STAGE EJECTOR
AT STEAM 154.6 PSIA.
△ - CONDENSING 4800GPH, INLET TEMP. 17°C
o - " 6000GPH, " " 17°C
x - " 7500GPH, " " 13°C
SUCTION PRESSURE, PSIA
AIR, LBS. PER HR.
0
2
4
6
8
10
12
14
16
0
100
200
300
400
500
600



Notations:

- Stable } 2-Stage, 2 ejectors with Hytors.
- Unstable } 2-Stage, 2 ejectors with Hytors.
- Stable, second-stage with Hytors.
- Stable, 2-stage, 3 ejectors with Hytors.

FIGURE 30. Variety of Optimum Suction Curves of the Vacuum System.