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A COMPARATIVE STUDY OF PUMPING SYSTEMS FOR A CHEMICAL PROPULSION ALTITUDE TEST STAND

BY

ALBERT A. YETMAN

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A COMPARATIVE STUDY OF PUMPING SYSTEMS FOR A CHEMICAL PROPULSION ALTITUDE TEST STAND

Prepared by:

Albert A. Yetman

Albert A. Yetman

Approved by:

Daniel J. Grant

Daniel J. Grant

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I. INTRODUCTION

This report describes the results of a feasibility study conducted to determine the size, type and comparative cost of an altitude facility for testing small chemical rocket engine components and systems. The study was intended to determine gross overall requirements and to select a suitable conceptual plan for a detailed design. The final design will be made by the contractor selected to provide A/E and construction and erection services for the integrated Chemical Propulsion Research Facility.

The facility will be located at the Goddard Propulsion Research Test Site located on Telegraph Road in Greenbelt, Maryland.

The major design requirement for the facility would be to maintain a 200,000 foot altitude condition while pumping exhaust products, having a molecular weight of 22.0 and a temperature of 5000°F, at a rate of 0.08 lbs. per second.

In this report, three methods of achieving the above conditions were considered: a mechanical pump system, a boiler steam ejector system, and a Hyprox steam generator ejector system. In addition, a discussion of the economic feasibility of using Rental Facilities is included.

II. CONCLUSIONS

The following conclusions are based on pumping exhaust products having a molecular weight of 22.0 and a temperature of 5000°F at a rate of 0.08 lbs per second or its equivalent of 70°F dry air at a rate of 720 pph. (The conversion from rocket engine exhaust gas to equivalent 70°F dry air is given in Appendix A).

(1) A Mechanical Pumping System is:

- (a) suitable for attaining the 100,000 foot altitude condition
- (b) impractical for the 200,000 foot condition because of the large pumping requirement in CFM

(2) A multistage steam ejector condensing system:

- (a) can achieve the 100,000 and 200,000 foot altitude conditions
- (b) must contain the following:
  - (I) a 30,000 pph capacity boiler
  - (II) an interstage barometric condenser system supplied with cooling water at an approximate rate of 3,600 gpm
  - (III) a four or five stage ejector system (the five stage system allows a blank-off, i.e., zero exhaust flow, near the 10^{-3} TORR vacuum condition)

(3) A Rental Facility:

- (a) is too costly
- (b) provides too many problems to be considered for a permanent or long range operation.

III. RECOMMENDATIONS

(1) Available literature and estimates of requirements indicate that a four stage barometric condensing system will simulate a 200,000 foot altitude environment. However, since the performance of the four stage system is marginal at this altitude it is recommended that a five stage system be designed to assure performance requirements are met.

(2) The complete project, including design, fabrication and erection of the facility, should be accomplished by one contractor who will be required to guarantee the performance of all mechanical and electrical systems.

IV. PUMPING SYSTEM

(a) General Discussion

GSFC has had, and will continue to have requirements for small thrust chemical rocket engine systems for satellite applications. In spite of this mission, the development, monitoring, and testing of small thrusters and thruster systems has been the exclusive domain of the propulsion subcontractor and, in many cases, his subcontractor. This situation is a direct result of a lack of government test facilities for accomplishing work in the above mentioned area.

The proposed rocket engine test facility to be constructed at the Goddard Propulsion Research Test Site would fill this void. The controlled environment it will produce will allow applied research in thermodynamics, fluid dynamics, and gas dynamics of rocket engine systems and components. The facility will enable determination of operational characteristics of state-of-the-art hardware. It will be an indispensable tool for the development testing of new systems which incorporate concepts promising to future space applications.

The study was directed at satisfying the requirements outlined in figure (1). The numbers indicate the maximum range of vacuum conditions that the system will be expected to maintain at the rocket engine nozzle exit plane while removing the corresponding engine exhaust weight flow. In order to achieve the foregoing conditions, a continuous type pumping system must be employed. Two systems which meet this criteria are the mechanical pump and the steam ejector systems. Both have been used successfully to achieve altitude simulation in a large number of turbojet and rocket engines altitude installations throughout the United States.

The various systems considered, including their method of operation, design requirements, and cost, are presented in the following pages.

(b) Mechanical Pumping System

A mechanical pumping system, such as that manufactured by the Kinney Vacuum Division, combines positive displacement lobe-type rotary booster pumps with rotary piston high vacuum second stage pumps. The second stage pumps the system from atmospheric pressure down to the cut-in pressure of the booster pump. At the cut-in pressure, which is in the range of 10^{-1} to 10^{-2} TORR, a switch automatically starts the booster pump.

Both the booster and second stage pumps are lubricated with a special protective vacuum oil. The oil prevents gas leakage and protects the parts from corrosive materials such as the nitric acid, formed by moisture in the system and the commonly used oxidizer, nitrogen tetroxide.

The advantages of this type of unit are ease of operation, fast start up time and relatively little maintenance.

The major disadvantage of the mechanical pump system is the rapid loss in volumetric pumping efficiency below a certain vacuum level. This is illustrated in figure 2 which shows a pumping speed curve for a typical pump system. This figure indicates the inherent disadvantage of such systems in that, as the pump inlet pressure is decreased, the speed at which such pumps are capable of operating also decreases, instead of increasing as required.

The inadequacy of such pumps for testing at conditions simulating very high altitudes is illustrated by the following example:

Consider the pumping speed required to simulate a 200,000 foot altitude, with a small rocket engine exhausting gas products with a molecular weight of 22 at a rate of 0.08 lbs. per second.

Assume the gases are cooled to $600^{\circ}R$ prior to entering the pumps.

Assume that there are no losses in the system and that the pressure at the pump inlet is the same as that in the test chamber.

Then, by the equation of state,

$$V = \frac{RT}{p}$$

where

$$p = 0.17 \text{ TORR}$$
$$R = 1545/22 \text{ ft-lb/lb-}^{\circ}R$$
$$T = 600^{\circ}R$$
$$V = 89{,}000 \text{ ft}^3/\text{lb.}$$

The pumping speed is the product of the specific volume of the gases and the weight flow.

$$S = 89{,}000 \text{ ft}^3/\text{lb} \times 0.108 \text{ lb/sec} \times 60 \text{ sec/min}$$
$$S = 427{,}000 \text{ CFM}$$

This exceedingly high pumping speed requirement makes the mechanical pumping system impractical for the 200,000 foot condition.

However, by making use of pressure recovery methods such as a supersonic diffuser located downstream of the rocket engine cone, as shown schematically in figure 3, the pumping speed can be decreased by as much as a factor of ten.

The function of the diffuser is to augment the mechanical pumps by utilizing a system of shock waves to convert the energy of the high velocity exhaust flow to low velocity high pressure flow. The high pressure exhaust flow is then passed through a heat exchanger for cooling prior to entering the mechanical pumps.

The pressure rise available across a diffuser depends upon many variables such as rocket exit plane Mach number, geometry, and exhaust products. Due to the complicated shock interactions resulting in the diffuser, the design is largely empirical. However, in order to determine the relative merits of a mechanical pumping system, it was assumed that such a diffuser could be properly designed for a pressure ratio of ten. This allows computing a pumping capacity requirement which provides a base for the analysis of the economics of a suitable pumping system.

Mechanical Pump Cost Estimate

As pointed out previously, the use of a mechanical pump system to simulate an altitude of 200,000 feet is feasible only if an exhaust diffuser is used as a means of pressure recovery in order to decrease the pumping speed required. A cost analysis of this type of system follows. A pressure rise of ten across the diffuser was assumed which reduces the pumping requirement to 43,000 FM. From reference 10, the pump costs are figured as 4.00/CFM for the blowers and 10.00/CFM for mechanical rotary pumps. The system under consideration would require five Kinney KMBV-11000/KD-850 type pumps, amounting to a cost of $240,000. In addition, a stainless steel test chamber, heat exchanger, exhaust ducting, plumbing, miscellaneous equipment and labor is required. Reference 11 indicates that for similar but physically smaller and less complex equipment, the cost would be between $50,000 and $60,000. Therefore, to account for the additional size and complexity, and to be conservative, a cost of $90,000 will be added to the pumps. The total cost would be $330,000.

This would be the breakpoint, because the number, size and cost of the pump become exorbitant if higher altitudes or greater exhaust flows are considered. This factor alone eliminates facility expansion as a future consideration.

Mechanical Pump Operating Costs

The mechanical pumps require only periodic maintenance and oil changes. The total oil capacity of the five pumps is 205 gallons. On the basis of $1.00/gallon of oil and one oil change per month, the yearly cost would be $2,460.

The electrical costs are based on the total horsepower of the pumps of 400, 30 operating hours per year and $0.03 per KW hr. for electricity. The electrical costs would be approximately $270/year.

An estimated one hundred hours per year of maintenance at $10 per hour would amount to $1000 per year.

The total yearly operating costs would be in the neighborhood of $3730.

(c) Steam Jet Ejector

The steam jet ejector achieves its pumping action by an exchange of energy between the drive media (steam) and the entrained media (rocket engine exhaust). Each ejector stage, as illustrated in figure 4, consists of a diffuser section and a steam nozzle located inside of a suction and mixing chamber.

The steam nozzle, which is the converging-diverging type, discharges an expanding high velocity jet of steam across the suction chamber. The expanding steam jet entrains the rocket exhaust and a turbulent mixing between the two flows takes place. The combined flow then travels to the diffuser where it is converted to high pressure, low velocity flow. The flow is then discharged to either the suction end of the next ejector stage for multistage ejector systems, or to atmosphere in the case of a single stage. Multistage ejector units are used for high vacuum pumping. However, steam requirements and ejector pipe sizes can become very large unless water condensers are placed between the last several ejector stages. The function of this heat exchanger is to condense out the operating steam and any condensibles in the gas flow from the previous stage. The ejector stage immediately following the condenser then has only the non-condensed material to pump.

Condensers used between stages in a steam ejector system are either of the surface or direct contact type. The surface type is basically a shell and tube heat exchanger; whereas, the direct contact condenser utilizes intimate mixing of the coolant and condensible mediums.

The advantages of the surface condenser are the separation of the cooling water from any contaminants in the steam or exhaust products, and the recovery of a large amount of steam heat for re-use. For the relatively short run times of a rocket engine test, recovery of steam heat for re-use would not be worthwhile.

In comparison with the surface condenser, the direct contact condenser requires smaller quantities of cooling water, in addition to lower maintenance and initial purchase costs. The larger passages in the direct-contact condenser prevent clogging and give it the added advantage of not requiring treated or clean water. The water contamination problem is negligible because of the large cooling water to exhaust product ratio.

The direct contact condenser can be gravity pumped by placing it approximately 34 feet above ground level with a leg running into a hot well (figure 5). This system is called a direct contact barometric condenser. The gravity pumping eliminates any chance of flooding the ejectors, which is a possibility with a mechanical pump failure.

In spite of the widespread use of ejectors in the rocket engine and chemical processing industries, there is a lack of design information. This is especially true for multistage steam ejectors-either condensing or non-condensing. The problem is compounded by the fact that the design of a particular unit is based primarily on experience and empirical data. This is due principally to the very complicated and unpredictable flow patterns in the operating system.

In order to obtain estimates for facility requirements and to have a comparison for what design data was available, a simplified gas dynamic and thermodynamic analysis was conducted using conservative factors to account for system losses. The analysis is shown in Appendix A. The most severe case was selected for the study. This corresponds to firing a 25 pound engine with a specific impulse of 300 seconds at the 200,000 foot altitude condition. The results are shown in figure 6.

Based on a 200 psia steam supply, the study indicates that a minimum of four stages will be required to achieve the 200,000 foot condition. This condition can be attained with either the condensing or non-condensing unit. A comparison between the two shows that the non-condensing ejector. system requires almost nineteen times as much steam capacity (28,520 lb/hr versus 537,000 lb/hr) to pump the 720 lbs/hr of equivalent dry air. The large steam requirement for the non-condensing system is tempered somewhat by the 3600 gpm of $55^{o}F$ cooling water needed for the condensing system. With all considerations to purchase cost, operational cost, physical size, and expansion capability, the ejector condenser system is clearly the most feasible and practical.

The foregoing requirements can be compared to data from a reference survey. Both the calculated design points and reference curves are plotted on figure 7. It can be noted that the condensing and noncondensing design points bracket the four and five stage Fondrk curves at the 200,000 foot condition. In comparison with the Fondrk four stage curve, the calculated design point shows a steam requirement of approximately 45 percent less, while requiring practically the same cooling water flow rate. The difference in steam requirements can probably be attributed to the assumptions made in the analysis in this study. Therefore, to ensure an adequate steam flow, a rate of at least 30,000 lbs. per hour of steam at 200 psia will be selected. Further study will indicate whether saturated or supersaturated steam will be required. The four stage condensing ejector system is illustrated in figure 8.

With reference to figure 7, it can be seen that the five stage condensing system requires less steam, and therefore may be more practical for the 200,000 foot condition. An additional advantage of the five stage system is that the blank-off vacuum (maximum altitude at zero exhaust flow) may be in the neighborhood of 10^{-3} TORR. Although this would be a marginal condition, it would allow testing of very low flow engines such as a 10^{-3} lb. thrust subliming solid engine. This would entail pumping about 0.045 lbs/hour of exhaust products, which corresponds to approximately 5,600 cfm of pumping at $70^{o}F$ and 10^{-3} TORR pressure.

The steam requirements to achieve the 100,000 foot altitude can be determined from the Fluidyne curve. A three stage condensing system with a steam output of approximately 8,600 lbs/hr will pump the 720 lbs/hr of dry air. This would be the flow value required if the system were designed specifically for the 100,000 foot condition. In this case the ejector physical dimensions will be sized to the critical 200,000 foot operation. Therefore, because of the lower volume flow at the lower altitude (higher pressure), bleed air and additional steam may be required to achieve stable operation at this condition. In any event, the 30,000 lb/hr of steam and 3600 gpm of cooling water required for the 200,000 foot altitude condition will be sufficient to satisfy the 100,000 foot requirements.

In the final design of the overall system, each stage will be matched to meet a particular design point for stable operation. This match includes not only the geometrical sizing of the stages, but also a minimum steam supply pressure requirement. Therefore, attempting to set an off design point (different altitude condition and/or exhaust flow rate) by lowering the steam supply pressure will result in unstable operation and loss of suction pressure. However, regulation can be attained by either cutting off the steam flow to a selected stage or stages and/or bleeding in a controlled amount of air or nitrogen to make up the difference in exhaust flow.

Multistage Steam Ejector Condensing System Costs

Hardware Cost Estimate

The multistage stage steam ejector system will incorporate as one of its major components, a water tube steam boiler delivering 30,000 pph of dry steam. The A. P. Woodson Company quoted a figure of approximately $1.00/pph of steam, or about $30,000 for a packaged boiler.

The ejector costs can be approximated by extrapolating figure 4 of reference 11. The ejector cost would be $26,000 plus an additional $4000 for the condensers for a total of $30,000.

For a cooling water supply system which will include a 20,000 gallon elevated tank, cooling and holding pond and water treatment provisions, the cost according to our plant engineers would be approximately $20,000.

The total to this point would be $80,000. A like amount of $80,000 can be considered reasonable for equipment installation, facility checkout and acceptance, contractor profit, overhead and any other direct costs.

In addition, a stainless steel test chamber, heat exchanger and various plumbing, miscellaneous equipment and labor is required. The same cost of $90,000 will be used as was used for the mechanical pumping system.

The total facility cost can be expected to be in the neighborhood of $250,000 exclusive of architectural construction costs.

Multistage Steam Ejector Condensing System Operating Costs

To obtain an estimate of facility operation costs, a figure of 30 testing hours per year was determined based on current programs. In the case of the steam boiler, a total of 60 hours per year was considered adequate to account for start up and heat time in addition to actual test time. Using an industry cost figure of $0.01 per pound of steam, the annual fuel and water operating costs for a 30,000 pph boiler would be:

$$.01 \times 60 \times 30{,}000 = \$18{,}000/\text{year}$$

The labor costs would consist of a stationary engineer in addition to some part time help. Therefore, $16,000 is allotted for labor.

The total yearly operating costs would be $34,000.

(d) Hyprox Steam Generator

The Hyprox Steam Generator was also considered as another means of supplying steam to the ejector system. The major difference between this unit and the previously discussed ejector system is in the method of generating steam. The Hyprox unit employs a peroxide and hydrogen fueled steam generator.

The steam is produced in the following manner. The peroxide is decomposed by passing it over a catalyst bed. The reaction forms steam and gaseous oxygen. Hydrogen is then introduced to mix with the oxygen and to form a combustible mixture. The mixture is ignited by the available heat. Water is then added to the burning reactants and is converted to steam by the heat in the chamber.

This system has the advantages of almost instantaneous start up time and very small size in comparison to a steam boiler. The major drawbacks to the Hyprox unit are the very high operating costs, and the need to store large quantities of additional hazardous materials - peroxide and hydrogen - in a limited area.

Hyprox Steam Generator Ejector System Costs

The steam requirements of this system will be considered to be the same as the multistage steam condenser system. This will allow a direct comparison between the two systems for operating costs based on initial capitalization, annual maintenance, and annual fuel costs and amortization over a five year period. The total cost is then $250,000.

Hyprox Steam Generator Operating Costs

The exact amount of steam required for this system would have to be determined by the manufacturer because he supplies the entire steam generator and ejector system. However, an estimate of the steam costs can be made based on 30,000 pph of steam flow and 30 hours of operation per year. Allowing 20 percent additional time for start up, shut down and other contingencies the total yearly usage would be 36 hours. According to Reference 12, a cost of $.08 per pound of steam, the yearly cost would then be:

$$\text{cost} = \$0.08/\text{lb} \times 36 \text{ hrs/year} \times 30{,}000 \text{ lb/hr} = \$86{,}300/\text{year}$$

In addition, six catalyst beds at $300 each would be required, for a total of $1,800. The unit would require relatively little manpower for maintenance and operation. Therefore, 100 man hours per year at $10/hour will be considered adequate. This would amount to $1000/year.

The total operating costs per year would be $89,100.

(e) Rental Facilities

At the present time, because of the lack of facilities, it is necessary for GSFC to rent off-site facilities from a private company to conduct high altitude rocket engine test work. This cost amounts to $125,000 per year.

This figure is approximately 35 percent greater than the yearly average cost (facility and operating costs) of a steam boiler system amortized over a five year period.

The problems associated with a rental facility are not confined to costs alone. Geographic location, project priority established by the rental facility, and control over the quality of acquired data are of major concern. Changes in the test plan due to prior test results are not easily incorporated. This lack of flexibility is usually due to contractual complications. Added to the foregoing is the time consumed in contract negotiations before a test can be started, and the compromising of optimum data to avoid re-negotiation.

All these factors point up the advisability for a GSFC altitude test facility.

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VI. APPENDIX A

List of Symbols

F = thrust force lbs.

I_{sp} = specific impulse lbF-sec/lbm

$\dot{W}$ = weight flow lbs/unit time

ρ = density lbs/ft^3

V = specific volume ft^3/lb

A = area $length^2$

P = pressure lbs/in^2

R = universal gas constant ft-lb/mole ^oR

T = temperature ^oR

v = velocity ft/sec

M = mach number

k = ratio of specific heats

C_p = specific heat at constant pressure BTU/lb ^oR

H = enthalpy BTU/lb

ΔH_s = isentropic enthalpy drop of steam across the diffuser BTU/lb

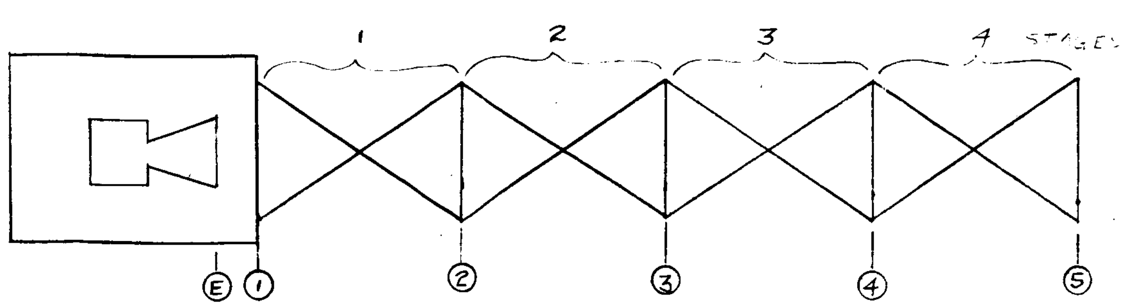
ΔH = isentropic enthalpy rise of the dry air across the diffuser BTU/lb

ΔH_L = enthalpy rise of the vapor load calculated by $\int p\,dv$ BTU/lb

Δq = heat load BTU/hr

N = efficiency

Subscripts refer to the stations shown on the following schematic:



Subscripts

c = steam chamber

t = total or stagnation

s = static condition

x = ejector suction chamber

y = diffuser exit plane

i = in

o = out

p = propellant

A = air

ANALYSIS

Pumping Load

The first step is to determine the rocket engine exhaust weight flow which the ejector system must pump. Using the following values,

$$F = 25.0 \text{ lbs.}$$

$$I_{sp} = 300 \text{ seconds}$$

The propellant weight flow is

$$\frac{F}{I_{sp}} = 0.0833 \text{ lbs/second}$$

The ejector industry commonly uses pounds of dry air per second at 70°F as their standard load flow parameter. Therefore the propellant flow can be converted to an equivalent dry air flow as follows:

Assume that both flows have the same total pressure (Rocket Chamber pressure) and the same static pressure (simulated altitude). The pumping speed is the same for both the Rocket exhaust and the equivalent dry air flows. Therefore

$$\dot{W}_A V_A = \dot{W}_p V_p$$

or

$$\dot{W}_A = \frac{\dot{W}_p V_p}{V_A}$$

$$\dot{W}_A = \frac{\dot{W}_p R_p T_p / P_p}{R_A T_A / P_A}$$

since

$$P_A = P_p$$

$$\dot{W}_A = \frac{\dot{W}_p R_p T_p}{R_A T_A}$$

where

$$R_p = \frac{1545}{22} \frac{\text{ft-lb}}{\text{lb } {}^{o}\text{R}} \; ; \qquad R_A = \frac{1545}{28} \frac{\text{ft-lb}}{\text{lb } {}^{o}\text{R}}$$

$$T_p = 1000^{o}\text{R} \qquad T_A = 530^{o}\text{R}$$

then

$$\dot{W}_A = 720 \text{ lbs/hr}$$

Ejector Stages Required - 200,000 Feet

Reference 2 indicates that for determining the minimum number of stages required to achieve a desired vacuum, an approximate suction pressure ratio of ten to one per stage can be assumed. Therefore, the first stage must provide a suction pressure of 0.17 TORR, the 200,000 feet altitude condition. Subsequent stages must provide suction pressures of:

second stage = 10 x 0.17 = 1.70 TORR

third stage = 10 x 1.70 = 17.0 TORR

fourth stage = 10 x 17.0 = 170.0 TORR

The fourth stage must compress the load from a suction pressure of 170 TORR to 760 TORR, atmospheric pressure. This requires a compression ratio of 4.47. Four stages will thus achieve the 200,000 feet altitude.

Stage Fluid Dynamic Parameters

The first stage steam nozzle exit conditions can be determined using the isentropic equations and the conditions shown in Table 1.

$$T_E = T_c \, (P_x/P_c)^{\frac{k - 1}{R}} = 62.43^{o}R \qquad (1)$$

The steam exit velocity is given by

$$\mu_E = \left\{ \frac{2 \; g \; k}{(R - 1)} RT_c \left[1 - (P_x/P_c)^{\frac{k - 1}{R}} \right] \right\}^{\frac{1}{2}} = 4314 \text{ fps} \qquad (2)$$

Using (1) and (2) the exit Mach number can be determined from

$$M_E = \frac{\mu_E}{(k \; g \; RT_E)^{\frac{1}{2}}} = 9.08 \qquad (3)$$

The total pressure ratio across a normal shock can be determined from the following normal shock relation:

$$P_{ox}/P_{oy} = \left[1 + \left(\frac{2 \; R}{R + 1}\right)(M_E^2 - 1) \right]^{\frac{1}{R-1}} \left[\frac{(R - 1) \; M_E^2 + 2}{(R + 1) \; M_E^2} \right]^{\frac{R}{R-1}} \qquad (4)$$

$$P_{ox}/P_{oy} = \frac{200 \text{ psia}}{P_{oy}}$$

$$P_{oy} = 0.311 \text{ psia}$$

Using the same four isentropic relationships, the fluid dynamic parameters were calculated for the following three stages. The values are tabulated in Table 1.

Steam Nozzle Fluid Dynamic Parameters

TABLE # 1

Stage	1	2	3	4
k	1.310	1.310	1.310	1.310
P_c PSIA	200	200	200	200
T_c ^oR	860	860	860	860
P_x PSIA	0.0034	0.034	0.34	3.4
T_E ^oR	62.43	109.5	188.7	325.4
u_E ft/sec	4314	4185	3958	3532
M_E	9.08	6.65	4.79	3.26
P_{oy} PSIA	0.311	1.80	9.43	44.13
$N = \frac{P_{x_{\text{following stage}}}}{P_{oy}}$	0.110	0.189	0.360	0.333*

$^*P_{x_{\text{following stage}}}$ = 14.7 PSIA for the last stage

Steam and Water Requirements for 4 Stage System

Conditions & Assumptions

The rocket engine exhaust will be converted to 70°F equivalent dry air.

The pressure in the suction chamber of each stage will be as indicated in Table 1.

Saturated steam at 200 psi will be supplied as the motive drive media.

The stage efficiency will be the ratio of the suction pressure of the following stage (assumed pressure) to the total pressure calculated by the normal shock equations times an assumed value of 0.60 which accounts for the mixing efficiency of the motive steam and the load to be pumped.

The energy which must be transferred to compress the steam load from the previous stage, will be assumed to be the work of compression of a closed system.

Cooling water at 55°F will be provided to the heat exchangers.

The steam requirements will be calculated from the following equations:

$$\dot{\omega}_s = \frac{\dot{\omega}_{air}\,(\Delta H) + \dot{\omega}_L\,(\Delta H_L)}{N\,(\Delta H_s)}$$

The stations will be as denoted in the list of symbols.

Work of Compression

If the steam load from the previous stage is not condensed out of the system, it must be compressed by the next stage ejector. The work required can be determined from the following equations:

$$\int Pdv = \frac{P_2 V_2 - P_1 V_1}{1 - k}$$

Second Stage Work

The second stage must compress the steam load from the first stage from 0.034 psi to 0.340 psi with the following conditions:

$$T = 860^{o}R$$
$$R = 1545/18 \quad = 85.8 \frac{\text{ft-lb}}{\text{lb } ^{o}R}$$
$$\gamma = 1.31$$

therefore from the equation of state

$$V_1 = \frac{RT}{P_1} = 15{,}060 \text{ ft}^3/\text{lb}$$

then

$$V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{\frac{1}{k}} = 2603 \text{ ft}^3/\text{lb}$$

The work then is

$$\int Pdv = - 173{,}900 \text{ ft-lb}$$

The negative sign indicates that the work is to be done on the system.

The above converts to

$$\int Pdv = - 223.5 \text{ BTU/lb}$$

By a similar analysis the third and fourth stage work can be determined. The results are tabulated in Table 2.

TABLE # 2

Work of Compression

	2nd Stage	3rd Stage	4th Stage
$T\ {}^{o}R$	860	860	860
$R\ \frac{ft\text{-}lb}{lb\ {}^{o}R}$	85.8	85.8	85.8
k	1.31	1.31	1.31
$V_1\ ft^3/lb$	15,060	1,506	150.6
$V_2\ ft^3/lb$	2,603	260.3	49.25
$\int Pdv_{ft\text{-}lb}$	173,900	173,900	148,500
$\int Pdv_{BTU/lb}$	223.5	223.5	191.0

Non-Condensing System

First Stage Steam Requirement

The first stage must exhaust the equivalent dry air flow of 720 lbs. per hour at 3.4×10^{-3} psi and compress it to 3.4×10^{-2} psi across the diffuser. Therefore

$$P_2/P_1 = 10$$

and from table 1, N = 0.110

From reference (7) the enthalpy of steam is:

Steam ΔH_s

$H_{s1} = 1198.7$ Saturation at 200 psi

$H_{s2} = 740.0$ ice vapor at 1.7 TORR (0.034 psi)

$\Delta H_s = 458.7$ BTU/lb

The dry air enthalpies can be determined using data from reference 8.

Dry Air $\Delta \dot{H}$

$$T_1 = 70^{o}F = 530^{o}R$$

$$C_p = 0.240 \text{ BTU/lb - }^{o}R$$

$$H_1 = C_p T_1 = (0.240)(530) = 127.1 \text{ BTU/lb}$$

$$K = 1.397$$

$$T_2 = T_1 (10)^{K-1/K} = (530)(10)^{0.2843} = (530)(1.928)$$

$$= 1021\ ^{o}R$$

$$C_p = 0.248 \text{ BTU/lb-}^{o}R$$

$$H_2 = C_p T_2 = (0.248)(1021) = 253.1$$

$$\Delta H = H_2 - H_1 = 253.1 - 127.1 - 126 \text{ BTU/lb}$$

$$\dot{\omega}_s = \frac{\dot{\omega}_{air} (\Delta H)}{0.6 \text{ N } (\Delta H_s)} = \frac{(720)(126)}{(0.6)(0.110)(458.7)} = 3000 \text{ lb/hr}$$

Second Stage Steam Requirements

The second stage must pump the equivalent dry air flow, plus the first stage operating steam load from 3.4×10^{-2} psi to 3.4×10^{-1} psi. Therefore

$$P_3/P_2 = 10$$

$$N = 0.189 \text{ from Table 1}$$

Required Steam ΔH_s

$$H_{s_2} = 1198.7$$

$$H_{s_3} = 815.0$$

$$\Delta H_s = 383.7$$

Dry Air ΔH

$$H_2 = 253.1 : T_2 = 1021\ ^{o}R \quad k = 1.383$$

$$T_3 = T_2\ (10)^{\frac{k-1}{k}} = (1021)(10)^{0.2768}$$

$$T_3 = 1932\ ^{o}R$$

$$C_p = 0.276$$

$$H_3 = C_p\ T_3 = (0.276)(1932) = 533 \text{ BTU/lb}$$

$$\Delta H = H_3 - H_2 = 533 - 253 = 280 \text{ BTU/lb}$$

The steam load from the first stage must be compressed to 3.4×10^{-1} psi. From Table 2, the work of compression is 223.5 BTU/lb. The steam load from the first stage is 1950 lbs/hr. Substituting the above values, the steam requirement becomes.

$$\dot{\omega}_s = \frac{\dot{\omega}_{air}\ (\Delta H) + \dot{\omega}_L\ (\Delta H_L)}{0.6\ N\ (\Delta H_s)}$$

$$= \frac{(720)(280) + (3000)(223.5)}{(0.6)(0.189)(383.7)}$$

$$\dot{\omega}_s = 20{,}000 \text{ lb/hr}$$

Third Stage Steam Requirements

The third stage must pump the equivalent dry air flow, plus the first and second stage operating steam load from 3.4×10^{-1} psi to 3.4 psi, with $P_4/P_3 = 10$ and

$$N = 0.360$$

Steam ΔH_S

$$H_{s_3} = 1198.7$$

$$H_{s_4} = 925.0 \qquad \text{at 170 Torr} = 3.4 \text{ psi}$$

$$\Delta H_S = 273.7 \qquad \text{BTU/lb}$$

Dry Air ΔH

$$H_{s_3} = 533 \qquad T_3 \; 1932 \; ^{\circ}R$$

$$K = 1.333$$

$$T_4 = T_3 \, (10)^{\frac{k-1}{k}} = (1932)(10)^{0.250}$$

$$= (1932)(1.777)$$

$$T_4 = 3433 \; ^{\circ}R$$

$$C_p = 0.297$$

$$H_4 = C_p \, T_4 = (0.297)(3433) = 1038$$

$$\Delta H = H_4 - H_3 = 1038 - 533 = 505 \text{ BTU/lb}$$

The steam load is then:

$$\dot{\omega}_S = \frac{\dot{\omega}_{air} \, (\Delta H) + L \, (\Delta H_2)}{0.6 \, N \, (\Delta H_S)} = \frac{720 \, (505) + 223.5 \, (20{,}000)}{(0.6)(0.360)(273.7)}$$

$$\dot{\omega}_S = 82{,}000 \text{ lb/hr.}$$

Fourth Stage Non-Condensing

The fourth stage must pump the equivalent dry air flow plus the first, second and third stage operating steam load from 3.4 psi to atmosphere. Therefore

$$P_5/P_4 = 4.47 \text{ and } N = 0.333$$

Steam ΔH_s

$$H_{s4} = 1198.7$$

$$H_{s5} = 1010.0 \qquad @760 \text{ TORR or } 14.7 \text{ psi}$$

$$\Delta H_s = 188.7$$

Dry Air ΔH

$$H_4 = 1038 \;;\; T_4 = 3433 \quad k = 1.30$$

$$T_5 = T_4 \,(4.47)^{\frac{k-1}{k}} = (3433)(4.47)^{0.2307)}$$

$$T_5 = 4850\ ^{o}R$$

$$C_p = 0.3066$$

$$H_5 = 4850\ ^{o}R$$

$$C_p = 0.3066$$

$$H_5 = C_p\, T_5 = (0.3066)(4850) = 1488$$

$$\Delta H = H_5 - H_4 = 1488 - 1038 = 450 \text{ BTU/lb}$$

The steam load is then calculated as

$$\dot{\omega}_s = \frac{\dot{\omega}_{air}\,(\Delta H) + L\,(\Delta H_L)}{0.6\, N\,(\Delta H_s)}$$

$$= \frac{720\,(450) + 191\,(82{,}000)}{0.6\,(0.333)(188.7)}$$

$$\dot{\omega}_s = 422{,}000 \text{ lb/hr}$$

The total steam load for the 4 non-condensing stages is 537,000 lb/hr.

Condensing System

Third Stage Steam and Cooling Water Requirements

Reference (9) shows that at the interstage pressure between the second and third stage, the cooling water will vaporize at a water temperature above 69°F. Therefore cooling water at 55°F will be supplied to the barometric condenser placed between the second and third stages.

The cooling water required can be calculated from

$$\dot{\omega}_{H_2o} = \frac{\Delta q}{C_{p_\omega} \quad T_\omega}$$

where

$$\Delta T = 69^oF - 55^oF = 14^oF \qquad ; \qquad C_{p\omega} = 1.0 \text{ BTU/lb } ^oR$$

$$\Delta q = (q_{A3} - q_{A4}) + (q_{L3} - q_{L4}) = \text{heat to be}$$

removed from the steam and dry air. The above equation may also be written

$$\Delta q = \dot{\omega}_A \ (H_{A3} - H_{A4}) + \dot{\omega}_L \ (H_{L3} - H_{L4})$$

where

H_{A3} = 533 BTU/lb from previous stage

H_{A4} = 126 BTU/lb @ 69°F

H_{L3} = 1000 BTU/lb (assumed enthalpy entering condenser)

H_{L4} = 33 BTU/lb saturated water @ 69°F

$\dot{\omega}_A$ = 720 lb/hr

$\dot{\omega}_L$ = total steam load from first ans second stages

$\therefore \Delta q$ = 22,493,000 BTU/hr

substituting, the cooling water required is

$\dot{\omega}_{H_2o}$ = 3350 gpm

Steam Requirement

The steam requirement for the third stage is then based on pumping the 720 lbs. of equivalent dry air, plus the vapor which saturates the dry air at 69°F. From references 6, the vapor load amounts to 8.0 lbs. of water per lb. of dry air or 5750 lbs. of water per hour.

The following H values for the third stage are calculated using the same procedures as for the previous stages.

Steam ΔH_S

$$H_{s3} = 1198.7$$

$$H_{s4} = 925.0 \quad @ \quad 170 \text{ TORR} = 3.4 \text{ psi}$$

$$\Delta H_s = 273.7 \text{ BTU/lb}$$

Dry Air ΔH

$$H_3 = 126 \text{ BTU/lb} \quad ; \quad T_3 = 525^{\circ}R$$

$$k = 1.40$$

$$T_4 = T_3\,(10)^{\frac{k-1}{k}} = (525)(10)^{0.2855}$$

$$T_4 = 1021^{\circ}R$$

$$C_p = 0.248$$

$$H_4 = (0.248)(1021) = 251 \text{ BTU/lb}$$

$$\Delta H = H_4 - H_3 = 125 \text{ BTU/lb}$$

Saturated Vapor Load H_L

$$V_3 = RT/P = \frac{85.8\ (525)}{(.340)(144)} = 920 \text{ ft}^3$$

$$V_4 = V_3(P_3/P_4)^{\frac{1}{K3}} = (920)(0.1)^{0.715}$$

$$V_4 = 177 \text{ ft}^3$$

$$\Delta H_{VL} = \int P\,dV = \frac{P_4\ V_4 - P_3\ V_3}{1 - k} = \frac{(177)(3.40) - (0.34)(920)}{1 - 1.40}$$

$$= -\ 725 \text{ in-lb}$$

$$= -\ 11.18 \text{ BTU/lb}$$

The third stage steam requirement then is;

$$\dot{\omega}_s = \frac{\dot{\omega}_{air}\ (\Delta H) + \dot{\omega}_{VL}\ (\Delta H_{VL})}{0.6\ N\ (\Delta H_s)}$$

$$\dot{\omega}_s = \frac{(720)(125) + 8\ (720)(11.18)}{(016)(0.360)(273.7)} = 2{,}610 \text{ lb/hr}$$

3-stage total = 25,610 lbs/hr

Fourth Stage Steam and Cooling Water Requirements

The interstage pressure between the third and fourth stages allows the introduction of cooling water below 147°F. Since the saturated vapor load to the fourth stage can be lowered by limiting the cooling water temperature rise, the cooling water leaving the condenser will be 125°F. The reduction in vapor load to the fourth stage will reduce the steam requirements. Cooling water will be supplied at 55°F to the barometric condenser placed between the third and fourth stages.

The cooling water required is

$$\dot{\omega}_{H_2O} = \frac{\Delta q}{C_P \ \Delta T_w}$$

where

$$\Delta T_w = 125 - 55^\circ F = 70^\circ F$$

$$C_{P_w} = 1.0 \text{ BTU/lb } ^\circ R$$

$$\Delta q = \dot{\omega}_A (H_{A4} - H_{A5}) + \dot{\omega}_L (H_{L4} - H_{L5})$$

where

$$H_{A_i} = 251 \text{ BTU/lb from previous stage}$$

$$H_{A_o} = 140 \text{ BTU/lb @ } 125^\circ F$$

$$H_{L_1} = 1000 \text{ BTU/lb (assumed enthalpy entering diffuser)}$$

$$H_{L_o} = 114 \text{ BTU/lb saturated liquid at } 125^\circ F$$

$$\dot{\omega}_A = 720 \text{ lbs/hr}$$

$$\dot{\omega}_L = \text{Steam and vapor load from third stage} = 2610 + 5750 = 8,360 \text{ lbs/hr}$$

$$\therefore \Delta q = 4,881,500 \text{ BTU/hr}$$

substituting

$$\dot{\omega}_{H_2O} = 220 \text{ gpm}$$

The steam requirement for the fourth stage is based on pumping 720 lbs/hr of equivalent dry air, plus the vapor which saturates the dry air at 125°F. From reference 6 this amounts to 1.0 lbs. of water per lb. of dry air or 720 lbs. of water per hour.

The steam load is calculated as follows using the same method as the previous stage.

Steam ΔH_s

$$H_{s4} = 1198.7$$

$$H_{s5} = 1010.0 \quad @ \quad 14.7 \text{ psi}$$

$$\Delta H_s = 188.7 \text{ BTU/lb}$$

Dry Air ΔH

$H_4 = 144$ BTU/lb ; $T_4 = 602\ ^oR$; $k_4 = 1.40$

$P_5 = 14.7$ psi ; $P_4 = 3.4$ psi

$$T_5 = (602)\left(\frac{14.7}{3.4}\right)^{\frac{k-1}{k}} = (602)(4.32)^{0.2855}$$

$$T_5 = 914\ ^oR$$

$$C_P = 0.246$$

$$H_5 = C_P\ T_5 = (0.246)(914) = 225 \text{ BTU/lb}$$

$$\therefore\ \Delta H = H_5 = H_4 = 81 \text{ BTU/lb}$$

Saturated Vapor Load

$$V_4 = \frac{RT_4}{P_4} = \frac{(85.8)(602)}{(3.40)(144)} = 1056 \text{ ft}^3$$

$$V_5 = V_4 \left(\frac{P_4}{P_5}\right)^{\frac{1}{k_4}} = (1056)(0.2314)^{0.715}$$

$$V_5 = 371.3 \text{ ft}^3$$

$$H_{VL} = \int PdV = \frac{P_5\ V_5 - P_4\ V_4}{1 - k_4} = \frac{(14.7)(371.3) - (3.4)(1056)}{1 - 1.40}$$

$$H_{VL} = -4650 \text{ in-lb}$$

$$H_{VL} = -71.7 \text{ BTU/lb}$$

The steam required for the fourth stage is:

$$\dot{\omega}_s = \frac{\dot{\omega}_{air}\ (\Delta H) + \dot{\omega}_{VL}\ (\Delta H_{VL})}{0.6\ N\ (\Delta H_s)}$$

$$= \frac{(720)(81) + (1)(720)(71.7)}{(0.6)(0.333)(188.7)}$$

$$\dot{\omega}_s = 2{,}910 \text{ lb/hr}$$

The four stage steam total is 28,520 lb/hr

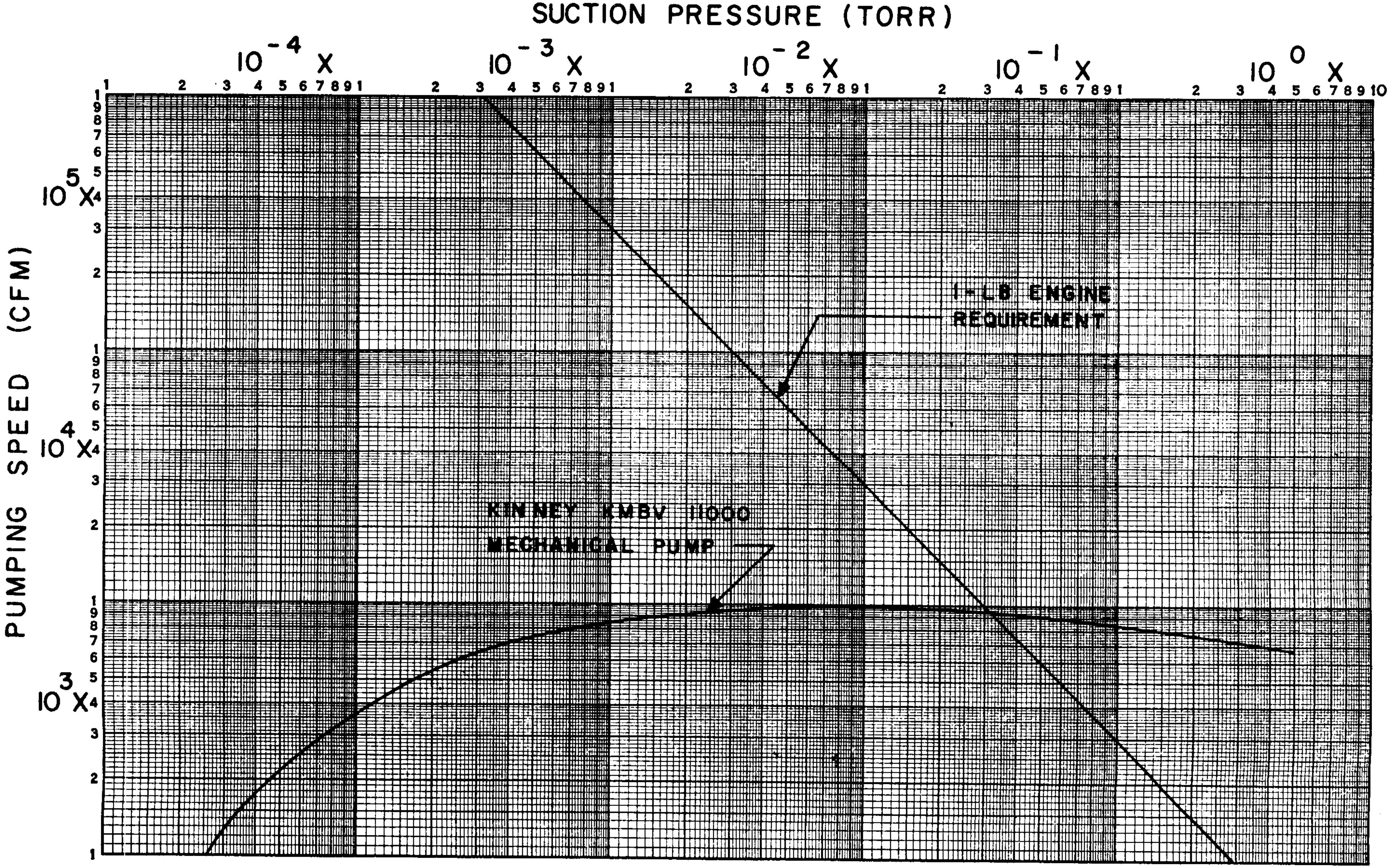
MONOPROPELLANT AND BIPROPELLANT ENGINES
(ASSUMED (I_{SP}) MAX = 300 SECONDS)

THRUST LEVEL (LBS)	WEIGHT FLOW (LBS / SEC)	PRESS.-ALT (TORR)	ALTITUDE (FEET)	TYPE OF TEST(S)
0.5 – 25.	0.0017 – 0.0833	8.0	100,000	PULSE & STEADY STATE
0.5 – 25	0.0017 – 0.0833	0.17	200,000	PULSE & STEADY STATE IGNITION PHENOMENA

SUBLIMING SOLID ENGINES
(ASSUMED I_{SP} = 80 SECONDS)

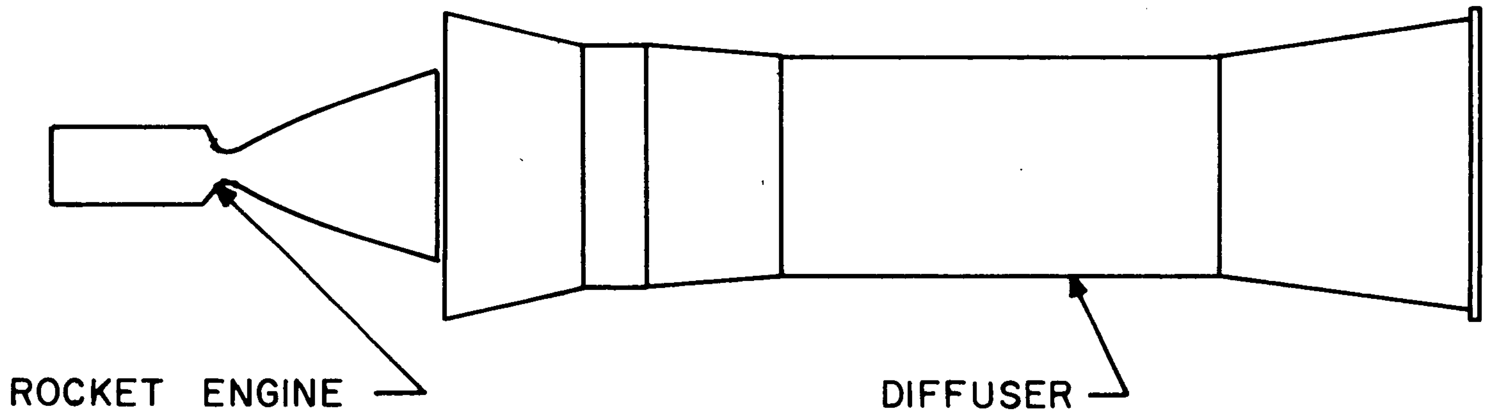
THRUST LEVEL (LBS)	WEIGHT FLOW (LBS / SEC)	PRESS.-ALT (TORR)	ALTITUDE (FEET)	TYPE OF TEST(S)
10^{-3}	1.25×10^{-5}	10^{-3}	$\simeq$ 300,000	IGNITION PHENOMENA PLUME IMPINGEMENT HEAT TRANSFER SUBSYSTEM INTERACTION

GSFC
ROCKET ENGINE ALTITUDE FACILITY
REQUIREMENTS
FIGURE 1



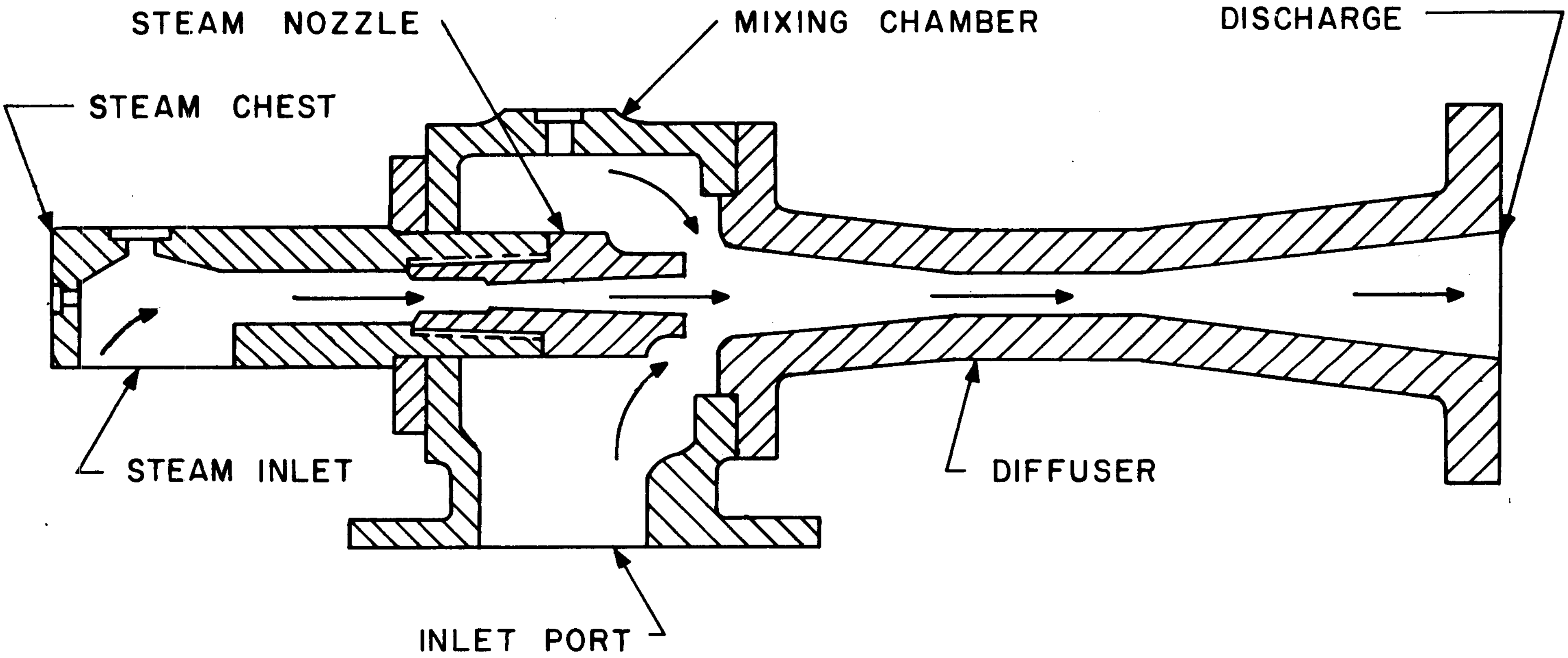
PUMPING SPEED CURVES

FIGURE 2

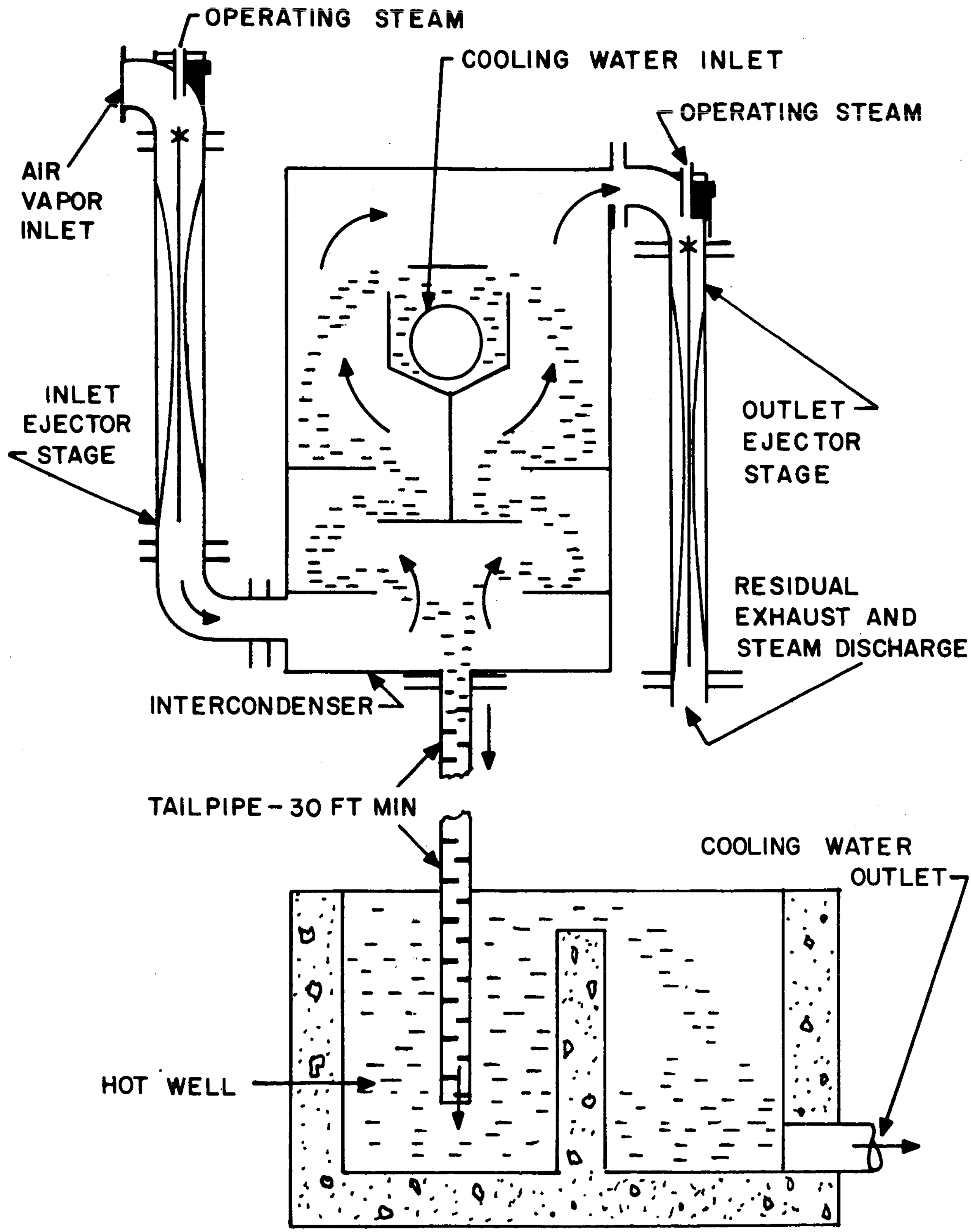


EXHAUST DIFFUSER FOR PRESSURE RECOVERY

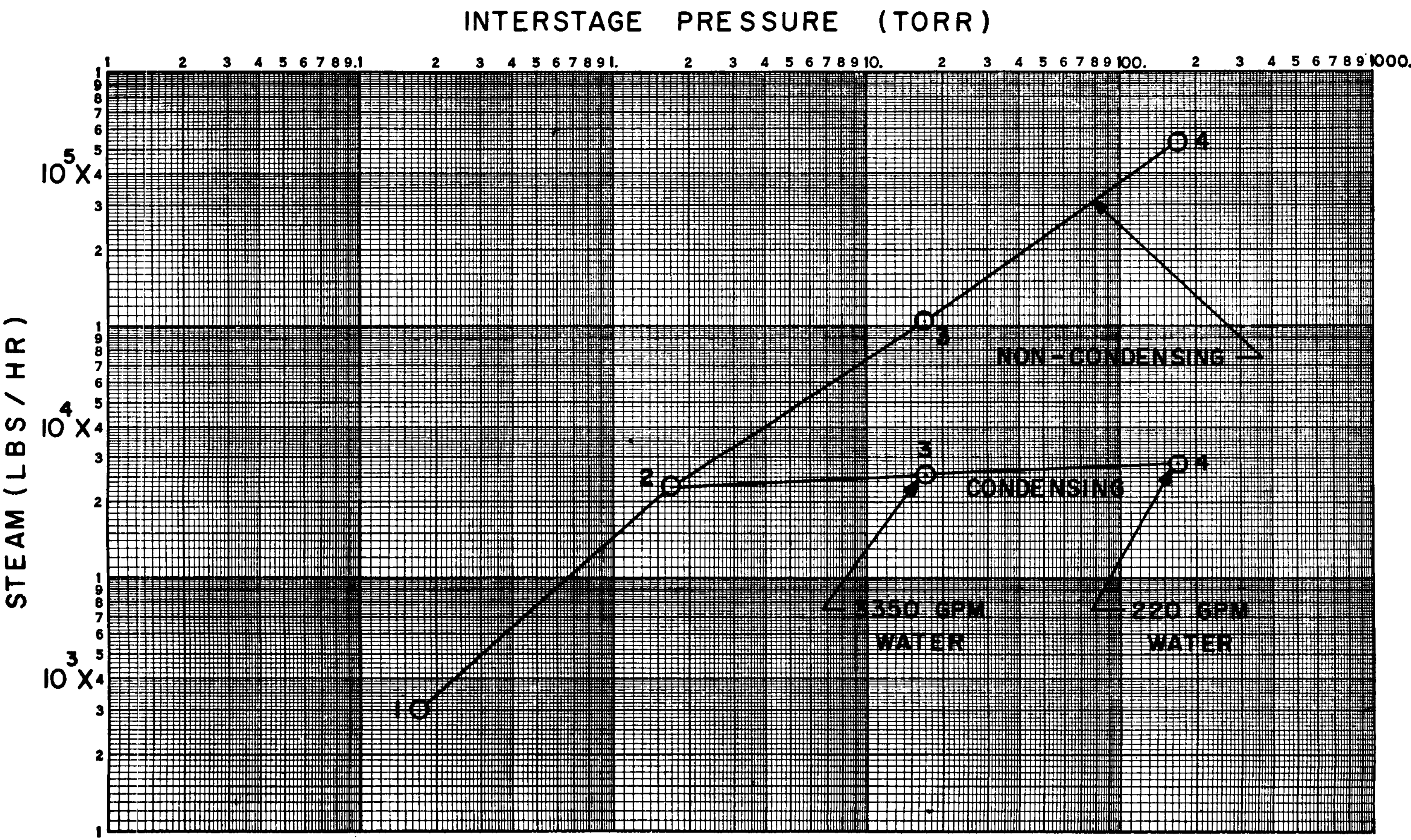
FIGURE 3



CROSS SECTION OF A TYPICAL STEAM EJECTOR
FIGURE 4

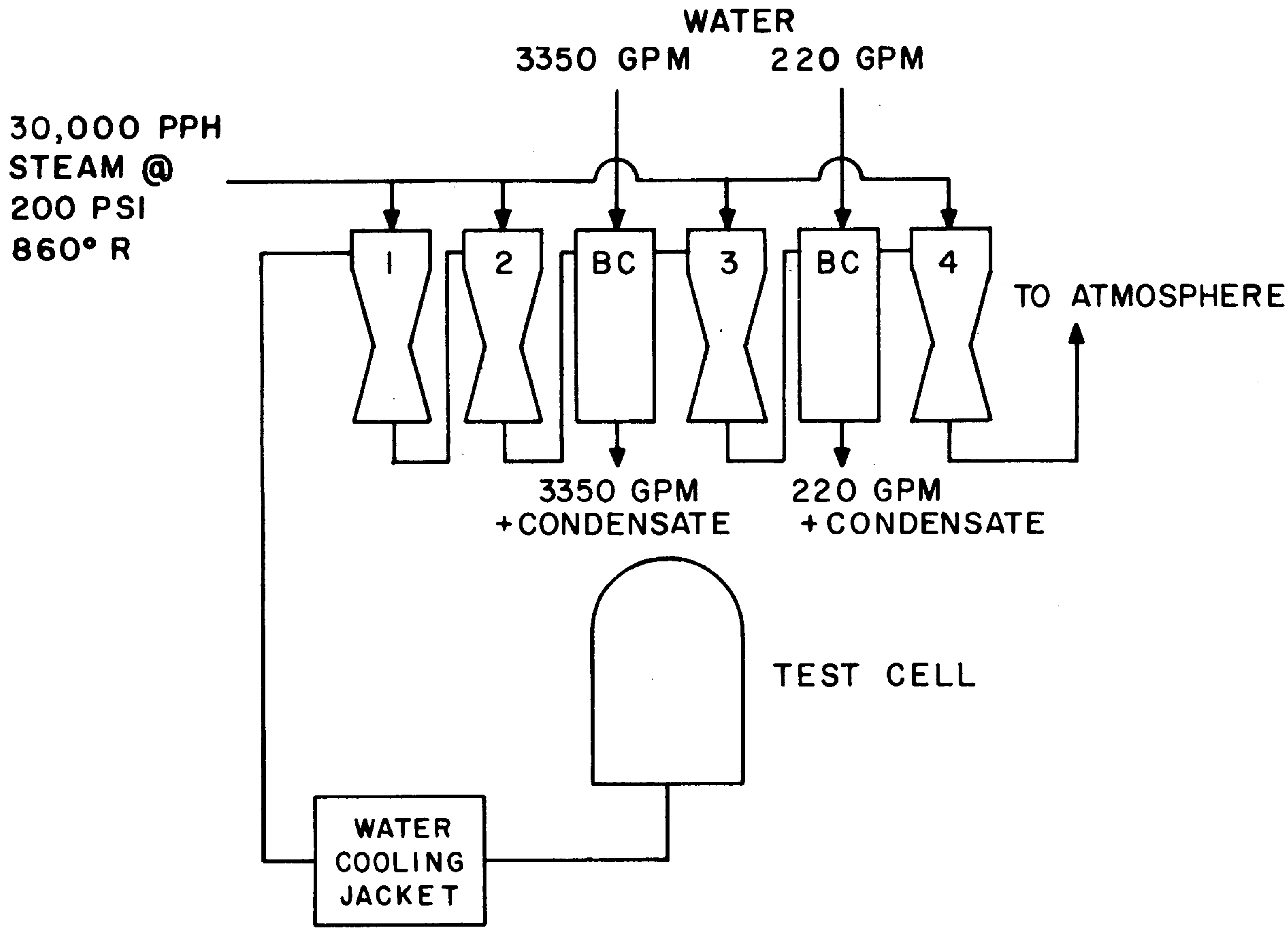


BAROMETRIC CONDENSER
FIGURE 5



4-STAGE EJECTOR SYSTEMS

FIGURE 6



FOUR STAGE STEAM EJECTOR CONDENSING SYSTEM
FIGURE 8

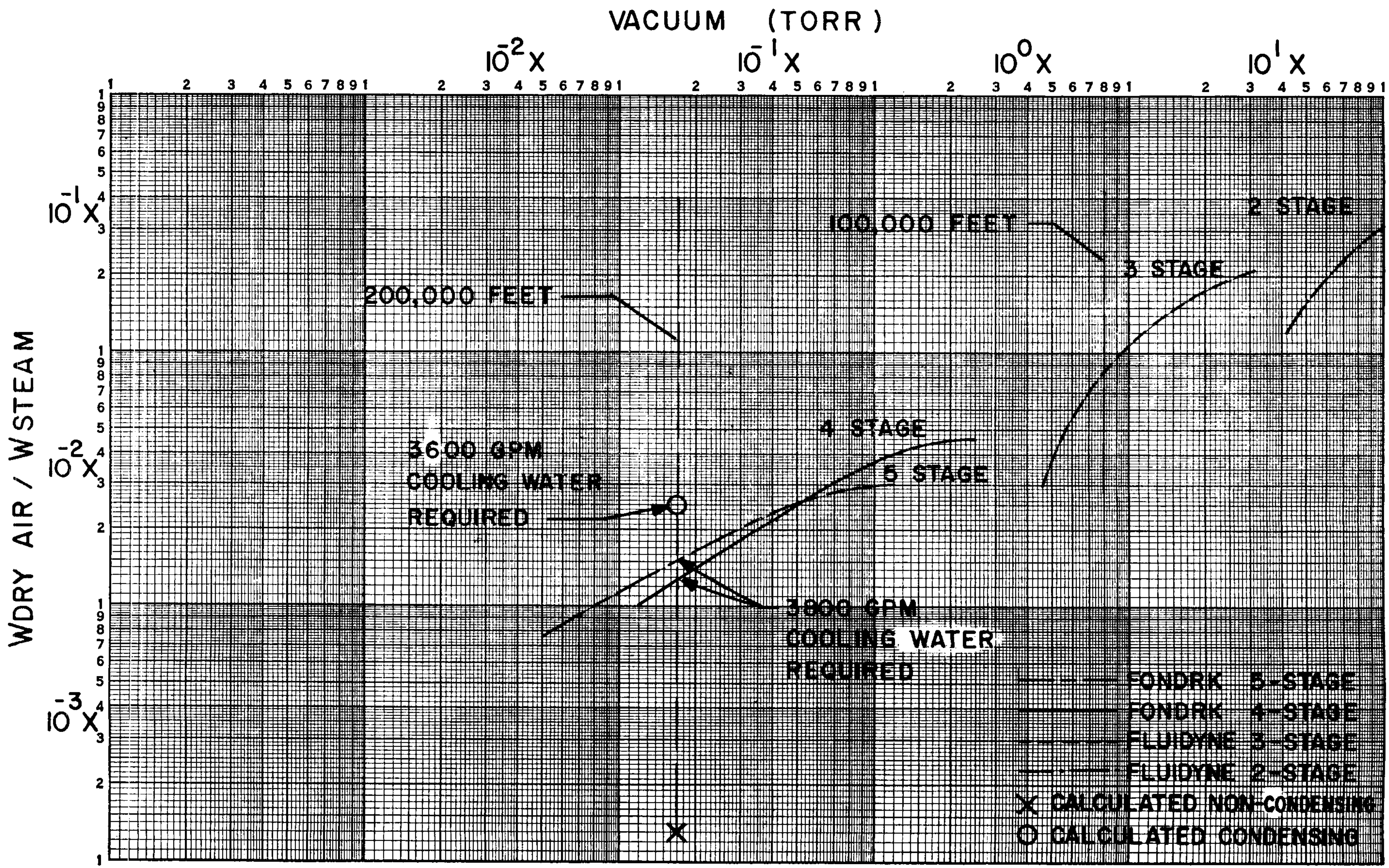

VACUUM (TORR)
WDRY AIR / WSTEAM
200,000 FEET
100,000 FEET
2 STAGE
3 STAGE
4 STAGE
5 STAGE
3600 GPM
COOLING WATER
REQUIRED
3800 GPM
COOLING WATER
REQUIRED
FONDRK 5-STAGE
FONDRK 4-STAGE
FLUIDYNE 3-STAGE
FLUIDYNE 2-STAGE
X CALCULATED NON-CONDENSING
O CALCULATED CONDENSING

FIGURE 7

	STEAM BOILER CONDENSING SYSTEM	HYPROX STEAM SYSTEM*	MECHANICAL PUMP SYSTEM	RENTAL FACILITIES
TOTAL FACILITY COST	250,000	250,000	330,000	
5 YEAR OPERATING COST	170,000	445,500	18,650	
YEARLY AVERAGE COSTS (5 YRS)	84,000	139,100	69,730	125,000
REMARKS	ALLOWS FOR EXPANSION	ALLOWS FOR EXPANSION. REQUIRES STORAGE OF LARGE QUANTS. OF HAZARDOUS MATERIALS.	EXPANSION NOT PRACTICAL – PUMP CONTAMINATION A PROBLEM	POOR CONTROL OF DATA QUALITY & TEST FLEXIBILITY

* THIS IS NOT THE MANUFACTURER'S ESTIMATE BUT AN ESTIMATE MADE FOR COMPARISON PURPOSES

SUMMARY COST SHEET FOR GSFC SMALL THRUSTER FACILITY

FIGURE 9