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MASTER

PERFORMANCE OF EXPERIMENTAL JET
PUMPS UNDER NONCAVITATING AND
CAVITATING CONDITIONS

(Title Unclassified)

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Reactor Division

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NONCAVITATING AND CAVITATING CONDITIONS

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PERFORMANCE OF EXPERIMENTAL JET PUMPS UNDER
NONCAVITATING AND CAVITATING CONDITIONSM. E. Lackey R. W. Holcomb
F. E. LynchAbstract

A jet pump with a nozzle-to-throat area ratio of 0.035 was operated in water with several nozzle-stream flows in order to study performance and to observe the effects associated with a change in nozzle-stream flow. The test pump had a nozzle diameter of 0.1324 in. and a throat diameter of 0.710 in. The spacing between the nozzle and the throat was varied between 0.0625 and 1.65 in. Two throat length-to-diameter ratios, 4 and 2, and two diffuser angles, 4° and 5°, were tested.

The results were interpreted in terms of the overall pump performance and as the limitation imposed on the side-stream flow by the static and thermal suppression heads at the throat inlet. In addition to indicating performance, the side-stream flow limitation indicates the type of flow pattern a succeeding jet pump or centrifugal pump will have to accept over a range of operating conditions. To supplement the measured performance results, photographs were taken of the flow at various operating conditions. For noncavitating operation, changes in nozzle-stream flow did not measurably affect the characteristic curve of head versus flow. Under cavitating conditions, both the head and flow characteristics changed as the nozzle-stream flow was varied.

Introduction

The design of the Medium Power Reactor Experiment (MPRE) includes a jet pump to improve the boiler recirculation rates and to provide a convenient method of combining the boiler recycle flow with the boiler through-flow. This jet pump in the reactor core circuit is closely coupled with the vapor separator, expansion tank, and boiler inlet plenum. Since the liquid and the vapor leaving the reactor core are in equilibrium, the liquid entering the jet pump from the vapor separator is saturated. To avoid vapor formation in the throat of the jet pump and deterioration of pump performance, it is necessary to provide cavitation

suppression head by a static head, a centrifugal field in the expansion tank, or subcooling or some combination of these.

To insure satisfactory operation of the jet pump when incorporated in the boiler complex, a full-scale model was tested with water to determine the cavitation suppression head required. The head versus flow characteristics of the pump were investigated to measure the effects of the suction pressure, the amount of subcooling in the nozzle stream, the relative nozzle- and side-stream flows, and the spacing of the nozzle relative to the pump throat. Four versions of the pump were tested to study the effects of the throat length, the diffuser length, and the diffuser discharge angle so that the pump could be made as compact as possible.

Nomenclature

Symbols

- D Distance between nozzle tip and throat entrance, in.
- g_c Dimensional constant, $32.174 \text{ lb}_m \cdot \text{ft} / \text{lb}_f \cdot \text{sec}^2$
- g Local acceleration due to gravity, ft / sec^2
- H Heat = $\frac{Pg}{\rho g_c} + Z + \frac{V^2}{2g}$, ft
- N Ratio of side-stream head rise to nozzle-stream head drop,

$$\frac{H_d - H_s}{H_n - H_d}$$
- P Absolute static pressure, $\text{lb}_f / \text{ft}^2$
- Q Quantity of flow, gpm
- T Temperature, °F
- V Velocity, ft / sec
- Z Elevation, ft
- ϕ Ratio of side-stream flow to nozzle-stream flow, Q_s / Q_n
- ρ Density, $\text{lb}_m / \text{ft}^3$

Subscripts

- c Point at which boiling starts; cavitation
- d Measuring point in discharge line; discharge
- m Value of N or ϕ obtained under noncavitating conditions
- n Measuring point in nozzle supply line; nozzle stream
- s Measuring point in side line; side stream
- v Vapor phase

Test Facility

Figure 1 is a schematic flow diagram of the test facility designed for investigating the operating characteristics of full-scale jet pumps for the MPRE. The main design considerations were:

1. a vacuum-tight system for operation with deaerated water,
2. control of inlet pressure,
3. subcooling for the nozzle stream,
4. adequate flow and pressure capacity to test full-scale models.

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Fig. 1. Flow Diagram of Full-Scale Jet-Pump Test Facility.

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The system consisted of a closed loop in which water was circulated to the nozzle and side port of the jet pump by means of a canned-rotor centrifugal pump and a shaft-sealed centrifugal pump in series with the jet pump. The nozzle and side streams separated at the discharge of the second pump. The nozzle stream passed through a rotameter, a valve, a cooler, and the nozzle. The side stream passed through a rotameter, a heater, a valve, and the bottom of the head tank into the jet-pump inlet.

The nozzle-stream cooler was a double-tube counterflow heat exchanger, with cold water as the heat sink, that was capable of cooling the nozzle stream to about 85°F with a side-stream temperature of 140°F at design flow. The nozzle-stream temperature was measured with a thermocouple just upstream of the nozzle. The nozzle-stream pressure was measured at the same point with a mercury U-tube manometer, which had water over the mercury on one side and air over the mercury open to the atmosphere on the other side.

The side-stream heater was a U-tube and shell heat exchanger with low-pressure steam on the shell side as the heat source. The heater had sufficient capacity to maintain the side-stream temperature at 140°F with the nozzle-stream cooler running at full capacity. The head tank was evacuated with an aspirator driven with cold water. During the cavitation tests, the aspirator was run continuously so that saturation conditions would be maintained at the liquid-vapor interface in the head tank. The amount of suppression head above saturation for the side-stream flow was controlled by the liquid level in the head tank. A sight glass was connected between the side-stream pressure tap and the vapor space in the tank, and another sight glass was connected between the lower portion of the tank and the vapor space to indicate the liquid level. The temperatures were measured with thermocouples in both the vapor and liquid in the tank and in the jet-pump inlet line. The vapor pressure in the tank was measured with a U-tube mercury manometer having one leg vented to the atmosphere. The pressure differential from the discharge to the inlet was measured with a U-tube manometer using water over Meriam No. 3 fluid. The flow in the discharge line passed through a cooler and then to the inlet of the first pump.

Figure 2 is a photograph of the test facility.

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Fig. 2. Full-Scale Jet-Pump Test Facility.

Jet-Pump Test Models

The test models were full-scale versions of the MPRE jet-pump design. The nozzle diameter and throat diameter were sized to produce the design head at design nozzle- and side-stream flows under noncavitating conditions. These dimensions were determined from the results of the experimental work of Cunningham¹ as correlated by LaVerne.² Since the overall length of the jet pump is important to the installation, four models were built and tested. The throat length, diffuser angle, and discharge diameter were varied in these models. One of these was modified by changing the radius of the throat inlet. A sketch of the jet pump giving the dimensions of the models tested is shown in Fig. 3.

The pump models were fabricated from transparent plastic so that the flow could be observed. The nozzle was made of brass and designed so that the location of the nozzle tip could be varied with respect to the throat inlet. Figure 4 is a photograph of the jet-pump parts. The four discharge tubes are shown in Fig. 5, and a photograph of the assembly installed in the experimental facility is shown in Fig. 6.

Fig. 3. Experimental Jet Pump with Variable Nozzle-to-Throat Spacing, D.

Fig. 4. Components of Experimental Jet Pump.

Fig. 5. Throat and Diffuser Sections of Models 1 Through 4.

Fig. 6. Model 2 Installed in the Test Facility.

Experimental Procedure

The head rise across each of the four models was measured for a series of nozzle- and side-stream flows at about 90°F and atmospheric pressure. Under these conditions, no cavitation occurred over the range of flows tested.

Models 1, 1-A, and 2 gave satisfactory performance under noncavitating conditions, and a series of tests was performed on these models to measure the head rise and the maximum side-stream flows at low static suppression heads where cavitation limited the flow through the pump. This was done for different values of nozzle-stream flow for several combinations of static suppression head and subcooling of the nozzle-stream flow.

To obtain the performance under cavitating conditions, the system was first filled with distilled water to the desired liquid level in the head tank, which could be varied from 0 to 30 in. above the jet-pump inlet.

The pumps were started, and the aspirator flow was turned on to remove the air from the system. When the system pressure was pulled down to the saturation pressure of the water, the steam flow to the side-stream heater was turned on to raise the water temperature. The temperature of the side stream was limited to 150°F to protect the plastic parts of the system. When the water temperature reached about 140°F, the nozzle-stream cooling water was turned on to give the desired amount of subcooling, which could be varied from 0 to 58°F. The side-stream heating was adjusted until the side-stream and nozzle-stream temperatures became constant.

When the vapor temperature at the top of the tank agreed with the liquid temperature in the tank and the vapor pressure saturation temperature, the system was assumed to be at equilibrium. The aspirator was run continuously to maintain the vapor-liquid equilibrium in the tank, except for the length of time required to read the vapor pressure.

At equilibrium saturated conditions in the head tank, data were taken at several values of nozzle-stream flow for a series of side-stream flows until the side-stream flow was reached at which cavitation was so severe that the throat and diffuser of the jet pump were filled with vapor bubbles and the head rise across the pump became negative.

Varying the flows through the pump caused the temperatures to change, so the saturation temperature and the amount of nozzle-stream subcooling were not held constant during a series of flows. This variation was small, however, and the aspirator removed vapor at a sufficient rate so that the system readjusted to saturation equilibrium rapidly. At times it was necessary to adjust the side-stream heating to bring the temperature level back to the desired range during the run. The saturation temperature corresponding to the vapor pressure agreed quite well with the vapor temperature as these changes were made.

Flow Measurement

The nozzle- and side-stream flows were measured by means of rotameters. The rotameters had beaded glass tubes and stainless steel floats. They were calibrated at 68°F by weighing the exit water flow. The nozzle-stream rotameter had a full-scale flow of 4.68 gpm, and the side-stream rotameter had a full-scale flow of 18.34 gpm.

Temperature Measurement

The temperatures in the system were measured at the four points indicated in the flow diagram with Chromel-Alumel thermocouples made of 30-gage wire. Beaded junctions were used on the thermocouples in the head tank. The liquid-temperature thermocouple had a bare junction that was beyond the end of a 1/8-in. stainless steel tube in which the wire was run through a two-hole ceramic insulator and sealed with epoxy compound. The vapor-temperature thermocouple was constructed in the same manner except that the bare junction was inside the 1/8-in. tube and small holes were drilled through the tube to provide low vapor flow around the junction.

The nozzle- and side-stream thermocouples had butt-welded junctions joining the bare wire at the end of a two-hole ceramic insulator. The insulators were held and sealed with epoxy compound inside hollow brass screws threaded into the walls of the side port and nozzle sections of the jet pump. The insulator was flush with the inner wall surface, and the junction was against the insulator so that only the junction projected beyond the wall.

The thermocouples terminated in a junction box, and a multipoint recorder and a rotary selector switch were connected in parallel to the terminals of the junction box with copper leads. The recorder was used to provide single- or multipoint monitoring of the temperatures. It had a self-balancing circuit to provide a compensated reference temperature. A potentiometer was connected in series with a cold junction in an ice bath to the common terminal of the rotary switch. The temperature data were taken by using the potentiometer to read the thermocouple outputs. The smallest scale divisions of the potentiometer were 0.005 mv, and a difference of 0.001 mv could be estimated easily.

The four thermocouples were calibrated by comparing the output of each to the output of the cold-junction thermocouple with all of them in the ice bath and by comparing the output of each to the boiling point of water at the pressure read on a mercury barometer with the four thermocouples immersed in a beaker of boiling distilled water. This allowed the thermocouple outputs to be corrected to the standard tables of emf

output for Chromel-Alumel thermocouples. The greatest deviation of the thermocouples from the table was 0.030 mv at the boiling point.

Pressure Measurement

The pressures at the side port, nozzle, and discharge of the jet pump were measured at two 1/16-in.-diam pressure taps 180° apart. The two holes at each location were connected, and the combined pressure was transmitted to one side of a U-tube manometer with clear plastic tubing. The nozzle pressure tap was connected to a 100-in. mercury manometer with water over mercury on the pressure side and air over mercury on the atmospheric side. The pressure differential between the nozzle and discharge was measured on a 100-in. manometer with water over mercury on both sides. The pressure differential between the side port and discharge was measured on a 72-in. manometer with water over Meriam No. 3 fluid on both sides. The smallest division on the scales of these three manometers was 1 mm. The vapor pressure in the head tank was measured on a 40-in. manometer with water vapor over mercury on the vacuum side and air over mercury on the atmospheric side. The smallest division on the scale of this manometer was 0.10 in. The side-port pressure head above the vapor pressure in the tank was read on a glass tube with a scale that had 1-mm divisions.

Results and Discussion

Visual observations and photographs were used for studying cavitation. Photographs of the flow taken over a range of conditions are shown below to aid in visualizing the flow regimes under discussion. The photographs emphasize the effects of nozzle-stream subcooling and nozzle position on cavitation under similar operating conditions — approximately the same nozzle-stream flow and static suppression head. The photographs are close-ups of the nozzle and throat region of the model 2 assembly shown in Fig. 6. Generally the same stages of cavitation were encountered in all of the models tested as the side-stream flow was increased from some relatively low value at which no cavitation could be observed. Several

stages of cavitation in the side stream were identified that can be defined as

1. cavitation inception - infrequent occurrence of vapor bubbles on a solid surface,
2. unsteady cavitation - alternate formation and collapse of vapor bubbles near the diffuser inlet,
3. cavitation breakdown - continuous vapor formation extending into the diffuser.

In addition to the cavitation associated with the side stream, vapor bubbles formed in the nozzle stream just downstream of the nozzle. This condition developed independently of cavitation in the side stream, and the effects on performance apparently were not interrelated. The effects of nozzle-stream and side-stream conditions on performance will be discussed separately. The vapor in the nozzle stream is referred to here as the "nozzle plume," and the formation of the vapor is referred to as "nozzle pluming."

Figure 7 presents photographs indicating the effects of nozzle-stream subcooling on nozzle pluming. The operation represented flow conditions that existed when there was no cavitation induced in the side stream.

Three observations made from these photographs follow.

1. The nozzle plume appears as a solid cone of very fine vapor bubbles originating approximately one to two jet diameters downstream of the nozzle.

2. The volume of the plume is influenced by the magnitude of the thermal suppression head imposed upon it in the form of nozzle-stream subcooling.

3. The reduction in the pump performance appears to be caused by growth in the diameter of the jet stream that obstructs the side-stream flow into the throat, or mixing section, of the jet pump.

Figure 8 presents photographs of the three types of side-stream cavitation obtained with a nozzle-stream flow of 3 gpm at nozzle-to-throat inlet spacings of 0.375 and 0.75 in. The pump operated with a static suppression head of 17 in. H₂O and 50°F subcooling at the jet inlet.

PHOTO 81433

PHOTO 71562

 $T_s = T_n = 140\text{ }^{\circ}\text{F}, \quad Q_n = 3.0\text{ gpm}, \quad Q_s = 9.7\text{ gpm}, \quad N/N_m = 0.62$

PHOTO 71575

 $T_s = 138\text{ }^{\circ}\text{F}, \quad T_n = 119\text{ }^{\circ}\text{F}, \quad Q_n = 3.0\text{ gpm}, \quad Q_s = 10.8\text{ gpm}, \quad N/N_m = 0.92$

PHOTO 71569

 $T_s = 136\text{ }^{\circ}\text{F}, \quad T_n = 86\text{ }^{\circ}\text{F}, \quad Q_n = 3.0\text{ gpm}, \quad Q_s = 10.7\text{ gpm}, \quad N/N_m = 1.0$

Fig. 7. Nozzle Plume Variation with Nozzle-Stream Subcooling;
 $D = 0.75\text{ in.}, H_s = 17\text{ in. H}_2\text{O}.$

A comparison of the side-stream flow obtained for corresponding states of cavitation indicates that a higher degree of mixing between the nozzle stream and the side stream takes place ahead of the pump throat for the 0.75-in. than for the 0.375-in. nozzle spacing. Observations made from these photographs follow.

1. Incipient cavitation, defined as the infrequent occurrence of vapor bubbles on a solid surface, begins on the surface in the throat

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PHOTO 81434

INCIPIENT CAVITATION
 $Q_s = 10.7$ gpm, $N/N_m = 1.0$

NO SIDE CAVITATION
 $Q_s = 10.7$ gpm, $N/N_m = 1.0$

UNSTEADY CAVITATION
 $Q_s = 10.9$ gpm, $N/N_m = 0.97$

UNSTEADY CAVITATION
 $Q_s = 12.5$ gpm, $N/N_m = 0.95$

CAVITATION BREAKDOWN
 $Q_s = 11.8$ gpm, $N/N_m = 0$

(a) $D = 0.375$

CAVITATION BREAKDOWN
 $Q_s = 12.7$ gpm, $N/N_m = 0$

(b) $D = 0.75$

Fig. 8. Cavitation Flow with Subcooling; $Q_m = 3$ gpm, $H_s = 17$ in.
 H_2O , $T_s = 136^\circ F$, $T_m = 86^\circ F$.

entrance region nearest the side port and progresses down the throat in the form of vapor streamers.

2. Unsteady cavitation, defined as the alternate formation and collapse of vapor bubbles at or near the diffuser inlet, fills the entire

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throat section and causes the vapor to be thrown into the high-pressure region of the diffuser, where it condenses.

3. Cavitation breakdown, defined as continuous vapor formation extending into the diffuser inlet, results when the head loss across the pump caused by the two-phase flow is great enough to prevent the condensation of the vapor.

4. The flow regimes between incipient cavitation and cavitation breakdown were continuous.

Figure 9 presents a photograph of the vapor streamers for a condition representative of what is referred to here as incipient cavitation. This photograph was obtained with no nozzle flow and a 17-in.-H₂O static suppression head. Observations made from this photograph follow.

Fig. 9. Incipient Cavitation of Side Stream.

1. The region on the inside of the throat inlet radius is at a reduced pressure caused by the centrifugal field induced by the side stream entering the throat.

2. The vapor formed at the regions of low pressure stayed close to the wall of the throat until it moved into the diffuser, where it condensed.

Figure 10 presents photographs indicating the effects of retracting the nozzle as far as the test rig would permit upon nozzle pluming and cavitation breakdown. The pump operated both with and without nozzle-stream subcooling at a static suppression head of 17 in. H_2O . Observations made from these photographs follow.

1. The fully retracted nozzle position induced a vortex in the inlet region of the throat. This vortex trapped the vapor formed in the nozzle plume.

PHOTO 81435

PHOTO 71563

$Q_s = 3.6$ gpm, $N/N_m = 0.84$

PHOTO 71572

$Q_s = 3.6$ gpm, $N/N_m = 0.83$

PHOTO 71564

$Q_s = 7.1$ gpm, $N/N_m = 0$

(a) $T_s = T_n = 140$ °F

PHOTO 71573

$Q_s = 10.5$ gpm, $N/N_m = 0$

(b) $T_s = 141$ °F, $T_n = 86$ °F

Fig. 10. Nozzle Plume and Cavitation Variations at Extreme Nozzle Position; $D = 1.65$ in., $Q_m = 3$ gpm, $H_s = 17$ in. H_2O .

2. The vortex extended deep into the diffuser and was not destroyed by cavitation breakdown.

3. Subcooling the nozzle stream suppressed the vortex but did not eliminate it.

Noncavitating Performance

Performance described as noncavitating implies that the nozzle, side, and combined streams were free from vapor either drawn into or produced by the pump. To insure this condition, the five models were tested with cold water at atmospheric pressure. The results of these tests are shown in Figs. 11 through 15. The performance curves for models 1, 1-A, and 2 are similar. The poor performance of model 3 indicates incomplete mixing in the throat prior to diffusion. The loss in head for model 4 was approximately 7% at a ratio of side-stream flow to nozzle-stream flow, ϕ , of 4 in comparison with models 1 and 2. This probably can be accounted for by increased eddy losses in the 10° diffuser. Because of the poor

Fig. 11. Noncavitating Performance Characteristics of the MPRE Boiler Jet Pump - Model 1.

Fig. 12. Noncavitating Performance Characteristics of the MPRE Boiler Jet Pump - Model 1-A.

Fig. 13. Noncavitating Performance Characteristics of the MPRE Boiler Jet Pump - Model 2.

Fig. 14. Noncavitating Performance Characteristics of the MPRE Boiler Jet Pump - Model 3.

Fig. 15. Noncavitating Performance Characteristics of the MPRE Boiler Jet Pump - Model 4.

noncavitating performance, no additional testing was conducted on models 3 and 4.

Cavitating Performance

The cavitating performance was determined by operating the jet pump at a series of successively lower inlet static pressures until the output was some small percentage of its noncavitating value. The inlet static pressure was changed by either or both of the following methods:

1. The operating level in the head tank was adjusted to vary the static suction pressure.
2. The conversion of static pressure to dynamic pressure caused the inlet static pressure to vary as the side-stream flow varied.

Models 1, 1-A, and 2 were tested to determine the cavitation performance of each. Preliminary testing indicated only small differences in the performance of the three models, so model 1-A was not tested further.

Three parameters that could affect the performance during cavitation were (1) nozzle-stream subcooling, (2) suppression head, and (3) nozzle-stream flow.

Figure 16 gives changes in performance at a constant nozzle-stream flow of 3.05 gpm and a suppression head of 17 in. H_2O for values of nozzle-stream subcooling of 0, 21, and 57°F. The shapes of these curves are typical and show the effect of the nozzle-stream subcooling on performance; for example,

1. Increased subcooling of the nozzle stream changes the shape of the performance curve from a gradual deviation from the noncavitating curve to a break and abrupt drop in output at side-stream flows only a little less than that for the flow at breakdown.
2. An increase in the nozzle-stream subcooling results in an increase in the flow at breakdown.

The change in the shape of the performance curve is probably a result of the change in the volume of the nozzle vapor cone. The increase in the breakdown flow is an indication of the extent of the mixing between the nozzle and side streams before they enter the pump throat.

Figure 17 shows the effects of static suppression head on performance at a constant nozzle-stream flow of 3.05 gpm with the nozzle stream

Fig. 16. Comparison of Cavitating and Noncavitating Performance of the MPRE Boiler Jet Pump with Nozzle-Stream Subcooling — Model 1.

Fig. 17. Comparison of Cavitating and Noncavitating Performance of the MPRE Boiler Jet Pump with Three Suppression Heads — Model 1.

subcooled 57°F for values of the side-stream static suppression head of 6, 17, and 28 in. H₂O.

The major effects of the static side-stream suppression head are

1. The shape of the performance curve is practically unchanged as the side-stream static suppression head is varied between 6 and 28 in. H₂O.

2. An increase in the side-stream static suppression head increases the breakdown flow. The shapes of the performance curves indicate that the volume of the nozzle vapor cone was not appreciably changed by varying the side-stream pressure from 6 to 28 in. H₂O. The increase in the breakdown flow with increased suppression head indicates a direct conversion of the static pressure to dynamic pressure.

Figure 18 shows performance curves for nozzle-stream flows of 2.80, 3.05, and 3.28 gpm at a constant nozzle-stream subcooling of 21°F and a static suppression head of 17 in. H₂O. The shape of the performance

Fig. 18. Comparison of Cavitating and Noncavitating Performance of the MPRE Boiler Jet Pump with Three Nozzle-Stream Flows - Model 1.

curve appears to be independent of the nozzle-stream flow. The breakdown flow ratio is approximately inversely proportional to the nozzle-stream flow, which implies that the breakdown flow is independent of the nozzle-stream flow.

A series of runs was made to determine the effect of the nozzle-to-throat spacing on the cavitating and noncavitating performance. (All other tests were performed with a nozzle-to-throat spacing of 0.375 in.) The data from these runs are presented in Figs. 19 through 22.

Fig. 19. Effect of Nozzle Spacing on the Noncavitating Performance of the MPRE Boiler Jet Pump - Model 2.

Fig. 20. Effect of Nozzle Spacing on the Performance of the MPRE Boiler Jet Pump with 0°F Nozzle-Stream Subcooling - Model 2.

Fig. 21. Effect of Nozzle Spacing on the Performance of the MPRE Boiler Jet Pump with 55°F Nozzle-Stream Subcooling - Model 2.

Fig. 22. Effect of Nozzle Spacing on Maximum Side-Stream Flow — Model 2.

Figures 19 through 21 indicate the effect of the nozzle-to-throat spacing on the shape of the performance curves. The performance of the pump was not appreciably changed for nozzle-to-throat spacings between 0.0625 and 0.50 in. Further increases in the spacing up to approximately 1.4 in. resulted in a continuous decrease in the pump output. At the extreme position of 1.65 in., a vortex was induced in the inlet stream that resulted in an abrupt decrease in the pump output.

Figure 22 shows the change in the breakdown flow as the nozzle spacing was changed. The higher breakdown flow obtainable with subcooling

of the nozzle stream indicated an increase in the mixing as the nozzle spacing was increased. The sudden decrease in the breakdown flow at the extreme position indicated that the vortex observed when operating at saturated conditions also existed when operating at atmospheric pressure.

Figure 23 shows the relationship between the breakdown flow and the side-stream suppression head when the pump was operating with no nozzle-stream subcooling for values of the nozzle-stream flow from 0.0 to 3.28 gpm. The equation of the line through these data follows:

$$H_s = 0.158 Q_s^2 . \quad (1)$$

Figure 24 indicates the degree of mixing that took place between the nozzle stream and the side stream before they entered the pump throat. These mixing values were obtained by attributing the excess breakdown flow obtained with nozzle-stream subcooling (as compared with the flow obtained with no nozzle-stream subcooling) to a decrease in the side-stream temperature that resulted from the partial mixing of the nozzle and side streams.

It was assumed that the pump performance away from the breakdown region would be a function of the net suppression head on the nozzle stream. It was further assumed that this head would be most effective in the pump throat. The net suppression head on the nozzle stream was

Fig. 23. Maximum Side-Stream Flow - Models 1 and 2.

determined by adding the inlet suppression head to the nozzle-stream thermal suppression head and subtracting the decrease in static head caused by the side-stream flow. A plot of the ratio of the pump head rise to the noncavitating head rise is shown in Fig. 25. This plot

Fig. 24. Nozzle Subcooling Effect on the Temperature of the Side-Stream Flow — Models 1 and 2.

Fig. 25. Effect of Nozzle Suppression Head on MPRE Boiler Jet-Pump Performance — Models 1 and 2.

indicates that the performance of the pump forms two distinct patterns below 40 in. of nozzle-stream suppression head, one with no nozzle-stream subcooling and one with nozzle-stream cooling. The different patterns probably result from the greater change in the size of the nozzle plume with subcooling.

Summary of Results

The experimental data obtained from a jet pump with a nozzle-to-throat area ratio of 0.035 indicated the following:

Noncavitating Conditions

1. The characteristic curve relating the head and flow parameters was the same for all nozzle-stream flows.
2. The pump performance deteriorated as the length-to-diameter ratio of the throat was decreased from 4 to 2.
3. The 5° diffuser gave approximately 7% less head than the 4° diffuser when operated at the design ratio of side-stream flow to nozzle-stream flow, ϕ , of 4.
4. The variation of the diameter of the diffuser discharge from 1.5 to 1.75 in. resulted in no apparent change in pump performance.

Cavitating Conditions

1. Vapor formation in the nozzle stream resulted in a decrease in the pump head.
2. Subcooling the nozzle stream decreased vapor formation in the nozzle stream and increased the flow at which cavitation caused a loss in the pump pressure rise.
3. The maximum or breakdown side-stream flow was directly related to the net suppression head applied to the jet pump inlet.
4. Nozzle-stream subcooling increased the effective cavitation suppression head. This increase evidently stemmed from the partial mixing of the nozzle and side streams, which reduced the temperature of the side stream before it reached the throat.

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APPENDICES

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Appendix A

SAMPLE CALCULATION

A sample calculation is given below to illustrate the method of constructing the performance curves contained in this report. The pump performance was evaluated from the energies of the fluid streams entering and leaving the pump and the flows of each stream.

From the test results,

$$Q_s = \text{side-stream flow} = 15.1 \text{ gpm}$$

$$Q_n = \text{nozzle-stream flow} = 3.05 \text{ gpm}$$

$$Q_d = \text{discharge flow} = 18.15 \text{ gpm}$$

$$\frac{P_s - P_v}{\rho_s} = \text{side-stream static suppression head} = 26.58 \text{ in., H}_2\text{O}$$

$$\frac{P_d}{\rho_d} - \frac{P_s}{\rho_s} = \text{static head rise in side stream} = 0 \text{ in. (specific gravity} = 1.95)$$

$$\frac{P_n}{\rho_n} - \frac{P_d}{\rho_d} = \text{static head drop in nozzle stream} = 197.9 \text{ in. (specific gravity} = 12.6)$$

$$T_s = \text{side-stream temperature} = 139.1^\circ\text{F}$$

$$T_n = \text{nozzle-stream temperature} = 90.2^\circ\text{F}$$

$$T_v = \text{head-tank-vapor temperature} = 138.4^\circ\text{F}$$

$$P_v = \text{head-tank-vapor pressure} = 5.7 \text{ in. Hg}$$

From the pump dimensions,

$$\text{Area of the side port} = 0.0123 \text{ ft}^2$$

$$\text{Area of the discharge} = 0.0169 \text{ ft}^2$$

$$\text{Area of the nozzle inlet pipe} = 0.0023 \text{ ft}^2$$

The performance calculations were the following:

Total side-stream head

$$\frac{P_s}{\rho_s} + Z_s + \frac{V_s^2}{2g} = H_s \quad (A.1)$$

Total nozzle-stream head

$$\frac{P_n}{\rho_n} + Z_n + \frac{V_n^2}{2g} = H_n . \quad (A.2)$$

Total discharge head

$$\frac{P_d}{\rho_d} + Z_d + \frac{V_d^2}{2g} = H_d . \quad (A.3)$$

Ratio of side-stream pressure rise to nozzle-stream pressure drop

$$\frac{H_d - H_s}{H_n - H_d} = \frac{\left(\frac{P_d}{\rho_d} + Z_d + \frac{V_d^2}{2g}\right) - \left(\frac{P_s}{\rho_s} + Z_s + \frac{V_s^2}{2g}\right)}{\left(\frac{P_n}{\rho_n} + Z_n + \frac{V_n^2}{2g}\right) - \left(\frac{P_d}{\rho_d} + Z_d + \frac{V_d^2}{2g}\right)} = N . \quad (A.4)$$

It was assumed that $\rho_d = \rho_s = \rho_n = 62.3 \text{ lb/ft}^3$, and since $Z_d = Z_s = Z_n$, Eq. (A.4) was written as

$$N = \frac{\frac{(P_d - P_s)}{\rho_s} + \frac{V_d^2}{2g} \left[1 - \left(\frac{V_s}{V_d}\right)^2\right]}{\frac{(P_n - P_d)}{\rho_s} - \frac{V_d^2}{2g} \left[1 - \left(\frac{V_n}{V_d}\right)^2\right]} , \quad (A.5)$$

for which

$$\begin{aligned} N &= \frac{\frac{(0)(2.95)}{12} + \left(\frac{18.15}{7.481 \times 60 \times 0.0169}\right)^2 \frac{1}{64.4} \left[1 - \left(\frac{15.1 \times 0.0169}{18.15 \times 0.0123}\right)^2\right]}{\frac{197.9 \times 12.6}{12} - \left(\frac{18.15}{7.481 \times 60 \times 0.0169}\right)^2 \frac{1}{64.4} \left[1 - \left(\frac{3.05 \times 0.0123}{15.1 \times 0.0023}\right)^2\right]} \\ &= \frac{0 - 0.027}{207.8 + 0.015} = -0.00013 \cong 0 . \end{aligned}$$

Since N is approximately zero, the side-stream flow is considered as the breakdown flow.

The ratio of the side-stream flow to the nozzle-stream flow is

$$\frac{Q_s}{Q_n} = \phi , \quad (\text{A.6})$$

or

$$\frac{15.1}{3.05} = \phi = 4.95 .$$

The relationship obtained between the breakdown flow and the side-stream suppression head for no nozzle-stream subcooling was assumed to apply for operation with nozzle-stream subcooling:

$$\begin{aligned} (H_s)_{\text{required}} &= 0.158 Q_s^2 \\ &= 0.158 (15.1)^2 = 36 \text{ in.} \end{aligned} \quad (\text{A.7})$$

Since the required suppression head must equal the available suppression head,

$$(H_s)_{\text{required}} - (H_s)_{\text{available}} = 0 , \quad (\text{A.8})$$

but

$$(H_s)_{\text{required}} - (H_s)_{\text{available}} = 36 - 26.58 = 9.42 \text{ in.} \neq 0 .$$

Since the required suppression head was greater than the side-stream static head, it was assumed that the side-stream temperature had been lowered by mixing with the nozzle stream. The decrease in temperature was assumed to be sufficient to allow a 9.42-in. decrease in the vapor pressure of the side stream:

$$\text{Vapor pressure at } 139.1^\circ\text{F} = 78.153 \text{ in.}$$

and

$$78.153 \text{ in.} - 9.42 \text{ in.} = 68.733 \text{ in.}$$

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The temperature equivalent to a vapor pressure of 68.733 in. is 134.17°F. The reduction in the temperature of the side-stream is

$$139.1 - 134.17 = 4.93^{\circ}\text{F} .$$

The difference between the side-stream temperature and the mixed-mean discharge temperature is

$$\frac{T_s - T_n}{1 + \phi} = \frac{139.1 - 90.2}{1 + 4.95} = 8.22^{\circ}\text{F} . \quad (\text{A.9})$$

For the conditions given at the beginning of this appendix, the calculated results were

$$N \cong 0 ,$$

$$\phi = 4.95 ,$$

$$T_c = 134.2^{\circ}\text{F} ,$$

$$T_d = 130.9^{\circ}\text{F} .$$

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Appendix B

CALCULATIONS FOR THE MPRE BOILER JET PUMP

The MPRE boiler jet pump was designed³ to operate at a ϕ of 4 and an N of 0.04 with saturated potassium at a 2000°R side-stream temperature and a nozzle-stream temperature of 1910°R. The maximum design flow is 12.43 gpm. A safety factor of 1.1 was allowed in the side-stream flow that gave a required maximum, or breakdown, flow of 13.67 gpm and a ϕ of 4.4.

From Eq. (A.9),

$$\frac{T_s - T_n}{1 + \phi} = \frac{2000 - 1910}{1 + 4.4} = 16.67^\circ\text{R}.$$

The difference between the side-stream temperature and the discharge temperature is 16.67°R.

From Fig. 24, the difference between the side-stream temperature and the cavitation temperature at a side-stream-to-discharge temperature difference of 16.67°R is 4.8°R. At 2000°R, the vapor pressure of potassium is equivalent to approximately 5.6 in. per °R.

From Eq. (A.7),

$$(H_s)_{\text{required}} = 0.158 Q_s^2 = 0.158 (13.67)^2 = 29.52 \text{ in.}$$

The thermal-suppression head available is

$$4.8^\circ\text{R} \times 5.6 \text{ in./}^\circ\text{R} = 26.88 \text{ in.}$$

Since, from Eq. (A.8),

$$(H_s)_{\text{required}} - (H_s)_{\text{available}} = 0,$$

and $(H_s)_{\text{available}}$ is equal to the static plus the thermal-suppression head available, Eq. (A.8) is rewritten

$$(H_s)_{\text{total required}} - (H_s)_{\text{thermal available}} = (H_s)_{\text{static required}}$$

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or

$$29.52 - 26.88 = (H_s)_{\text{static required}} = 2.65 \text{ in.}$$

Approximately 6 in. of head is provided when the expansion tank is empty in the MPRE design. At a flow of approximately 13.7 gpm from the expansion tank to the jet pump inlet, the head loss is estimated to be 3 in., which allows sufficient head for the system to operate with an empty expansion tank.

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Appendix C

TYPICAL EXPERIMENTAL DATA

Table 1. Typical Performance Data Obtained for Model 1 with Several Suppression Heads and Amounts of Nozzle-Stream Subcooling

Q_n (gpm)	Q_s (gpm)	P_v (in. Hg)	$\frac{P_s - P_v}{\rho_s}$ (cm, sp gr 1.0)	$P_n - P_d$ (cm, sp gr 12.6)	$P_d - P_s$ (cm, sp gr 1.95)	T_v (°F)	T_s (°F)	T_n (°F)	ϕ	N
3.05	6.2	5.625	44.6	191.5	68.1	138.5	139.7	138.9	2.03	0.0551
	7.12	5.625	44.3	192.0	64.0	138.7	139.8	139.1	2.33	0.0517
	8.0	5.675	43.0	192.2	54.5	138.7	140.0	139.3	2.62	0.0440
	8.9	5.675	42.5	195.3	46.5	138.7	140.0	139.5	2.92	0.0369
	9.78	5.775	42.2	196.0	38.5	138.8	140.1	139.7	3.21	0.0305
	10.48	5.785	41.8	195.0	0	138.8	140.1	139.8	3.44	0
	6.4	5.69	14.7	195.3	0	138.6	140.4	139.6	2.29	0
2.8	6.4	5.69	14.7	167.2	0	138.6	140.4	139.6	2.10	0
3.28	6.4	5.69	14.7	224.8	0	138.6	140.4	139.6	1.95	0
3.05	8.9	6.06	43.0	194.1	65.0	140.2	140.6	118.9	2.92	0.0519
	9.77	6.06	42.5	194.1	61.0	140.3	140.7	119.4	3.20	0.0487
	10.65	6.06	42.0	194.4	56.6	140.0	140.6	119.5	3.49	0.0451
	11.55	6.06	41.0	196.2	52.6	139.6	139.5	119.0	3.79	0.0416
	12.5	6.06	40.1	196.0	0	139.6	139.6	119.1	4.10	0
2.8	8.9	5.92	43.3	166.5	54.0	141.5	142.3	118.2	3.18	0.0503
	9.77	5.97	42.5	168.1	51.0	142.1	142.7	119.5	3.49	0.0470
	10.65	6.17	42.0	167.2	46.0	142.0	142.7	119.7	3.80	0.0427
	11.55	6.17	41.0	168.0	43.0	141.1	140.9	119.3	4.13	0.0397
	12.4	6.18	40.2	168.1	0	141.1	140.9	119.2	4.43	0
3.28	8.9	5.86	43.3	223.8	65.0	136.7	139.1	117.5	2.71	0.0527
	9.77	5.76	42.3	225.8	61.0	138.6	139.5	118.5	2.98	0.0474
	10.65	5.76	42.0	225.3	56.6	138.6	139.6	119.0	3.25	0.0454
	11.55	5.76	41.0	227.2	52.6	138.4	139.0	119.2	3.52	0.0396
	12.4	5.76	40.2	227.0	0	138.4	139.1	119.1	3.78	0
3.05	6.22	5.768	14.9	194.7	73.2	140.2	140.8	87.0	2.04	0.0583
	7.14	5.968	14.3	194.0	70.4	140.1	140.4	87.7	2.34	0.0563
	8.02	5.918	13.5	195.2	68.6	139.6	139.5	88.1	2.63	0.0545
	8.9	5.818	12.8	194.4	64.0	139.0	138.9	88.0	2.92	0.0510
	9.25	5.82	12.6	195.0	0	139.0	139.0	88.1	3.03	0
	7.15	7.00	44.3	191.0	70.9	146.1	147.1	85.5	2.34	0.0576
	8.9	7.00	43.3	191.0	66.0	145.9	146.6	86.4	2.92	0.0536
	10.65	6.55	42.5	192.4	60.0	142.4	143.3	85.3	3.49	0.0483
	10.65	6.35	42.0	191.2	59.2	143.6	144.1	86.3	3.49	0.0480
	11.55	6.35	41.5	193.1	56.6	143.3	144.0	86.6	3.79	0.0454
	12.5	6.40	41.0	191.9	50.6	143.0	143.7	86.7	4.10	0.0409
	12.5	6.75	40.0	191.0	51.0	146.6	146.1	87.3	4.10	0.0414
	12.7	6.40	40.5	193.1	49.6	143.2	143.5	86.9	4.16	0.0398
	12.9	6.42	40.3	193.0	0	143.3	143.4	87.0	4.23	0
	10.65	5.80	71.0	195.8	60.0	139.5	140.3	89.7	3.49	0.0475
	11.55	5.80	70.5	195.8	57.6	139.2	140.0	89.6	3.79	0.0456
	12.5	5.75	69.7	196.8	55.0	139.1	139.6	89.9	4.10	0.0433
	13.45	5.70	68.2	197.4	51.0	138.7	139.5	90.0	4.41	0.0401
	14.45	5.70	67.6	197.8	43.0	138.4	139.1	90.2	4.74	0.0337
	15.1	5.72	67.5	197.9	0	138.4	139.1	90.2	4.95	0

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Table 2. Performance Characteristics of Model 2 for Various
Nozzle Positions at Atmospheric Inlet Pressure
with $Q_n = 3.05$ gpm and $T_s = T_n$

D (in.)	Q_s (gpm)	$P_n - P_d$ (cm, sp gr 12.6)	$P_d - P_s$ (cm, sp gr 1.95)	T_s (°F)	ϕ	N
0.0625	0	191.8	89.1	83.6	0	0.0720
	3.70	192.8	79.7	83.4	1.21	0.0641
	7.15	193.2	69.3	83.4	2.34	0.0556
	8.90	191.8	63.4	83.4	2.92	0.0512
	10.65	197.2	58	83.4	3.49	0.0456
	12.5	193.3	51.8	83.8	4.10	0.0415
	14.45	194.6	44.3	83.7	4.74	0.0353
	16.35	195.8	41.6	83.6	5.36	0.0329
	18.35	196.2	33.0	83.5	6.02	0.0261
0.25	0	192.2	90.2	83.7	0	0.0728
	3.55	190.6	79.6	83.6	1.16	0.0647
	7.15	193.4	71.2	83.6	2.34	0.0571
	8.9	194.4	66.1	83.6	2.92	0.0527
	10.65	195.0	59.8	83.6	3.49	0.0475
	12.5	193.4	51.8	83.4	4.10	0.0415
	14.45	195.0	45.6	83.3	4.74	0.0363
	16.35	196.8	40.0	83.1	5.36	0.0315
	18.35	197.8	33.8	83.1	6.02	0.0265
0.375	0	189.8	90.0	82.9	0	0.0735
	3.75	192.6	80.3	82.9	1.23	0.0646
	7.15	193.6	69.6	83.0	2.34	0.0557
	8.9	194.4	64.6	83.0	2.92	0.0515
	10.65	195.6	58.8	83.0	3.49	0.0466
	12.5	196.6	53.3	83.0	4.10	0.0420
	14.45	195.8	45.6	82.9	4.74	0.0361
	16.35	193.8	39.0	82.9	5.36	0.0312
	18.35	197.2	32.0	82.5	6.02	0.0252
0.75	0	191.8	93.3	83.4	0	0.0754
	3.7	190.2	72.6	82.5	1.21	0.0592
	7.15	193	69.0	82.7	2.34	0.0554
	8.9	194.2	62.3	82.8	2.92	0.0497
	10.65	194.2	55.0	82.6	3.49	0.0439
	12.5	195.4	49.6	82.5	4.10	0.0394
	14.45	197.6	39.0	82.5	4.74	0.0306
	16.35	199.2	36.0	82.3	5.36	0.0280
	18.35	199.2	31.0	82.2	6.02	0.0241
1.125	0	189.8	96.5	83.4	0	0.0788
	3.75	189.2	74.1	82.1	1.23	0.0607
	7.15	193	67.5	82.3	2.34	0.0542
	8.9	192.6	60.0	82.4	2.92	0.0483
	10.65	193.8	52.6	82.4	3.49	0.0421
	12.5	194.4	43.7	82.3	4.10	0.0349
	14.45	197.6	36.3	82.3	4.74	0.0285
	16.35	197.2	31.3	82.2	5.36	0.0246
	18.35	201	26.2	82.1	6.02	0.0202
1.4	0	188.8	103.1	82.2	0	0.0847
	3.55	190.2	78.8	82.1	1.16	0.0642
	3.7	190.2	78.0	82.3	1.21	0.0636
	4.55	190.4	75.6	82.3	1.49	0.0616
	5.35	193.4	61.6	82.4	1.75	0.0494
	7.15	191.6	68	82.4	2.34	0.0550
	8.9	194.2	59.1	82.5	2.92	0.0472
	10.65	196.6	51.2	82.5	3.49	0.0404
	12.5	196.8	43.6	82.4	4.10	0.0343
	14.45	200.6	38.0	82.4	4.74	0.0294
	15.4	198.2	34.0	82.1	5.05	0.0266
	16.35	197.8	0	82.1	5.36	0
1.65	0	188.4	108.4	82.2	0	0.0892
	3.6	189.2	80.0	82.4	1.18	0.0656
	6.95	192.6	51.3	82.4	2.28	0.0413
	8.0	193.0	0	82.4	2.62	0

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Table 3. Performance Characteristics of Model 2 for Various Nozzle Positions at Saturation Inlet Pressure with $Q_n = 3.05$ gpm and $T_s \cong T_n$

D (in.)	Q_s (gpm)	P_v (in. Hg)	$\frac{P_s - P_v}{\rho_s}$ (cm, sp gr 1.0)	$P_n - P_d$ (cm, sp gr 12.6)	$P_d - P_s$ (cm, sp gr 1.95)	T_v (°F)	T_s (°F)	T_n (°F)	ϕ	N
0.0625	0	6.24	74.9	188.2	80.6	141.6	144.8	148	0	0.0664
	3.6	6.16	74.5	188.4	75.7	141.9	142.9	146.6	1.18	0.0623
	7.15	6.18	73.5	189.7	64.8	142.6	144.1	144.1	2.34	0.0530
	10.65	6.25	71.2	192.7	42.3	143.1	143.9	143.3	3.49	0.0340
	11.55	6.3	69.5	192.9	34	143.1	143.8	143.3	3.79	0.0273
	12.5	6.38	69.0	196.3	25	143.1	144.2	143.7	4.10	0.0197
	12.9	6.37	68.8	196.1	0	143.1	144.2	143.7	4.23	0
0.25	0	6.21	74.5	188.3	86.1	143.3	143.3	144.5	0	0.0709
	3.6	6.30	74.3	187.9	77.6	142.9	143.5	144.7	1.18	0.0640
	7.15	6.38	73.5	189.3	68.6	143.2	144.0	144.1	2.34	0.0562
	10.65	6.28	70.8	192.3	48.7	143.3	144.2	143.9	3.49	0.0393
	11.55	6.38	70.0	192.9	41.6	143.5	144.3	143.9	3.79	0.0334
	12.5	6.46	69.0	193.7	33.0	143.5	144.7	144.5	4.10	0.0264
	12.95	6.45	68.5	194	0	143.5	144.7	144.5	4.25	0
0.375	0	6.33	74.5	185.7	86.6	143.8	143.7	145.2	0	0.0723
	3.6	6.33	74.0	187.7	77.5	143.3	143.7	145.5	1.18	0.0640
	7.15	6.48	73.2	188.5	67.2	143.8	144.5	144.5	2.34	0.0553
	10.65	6.43	70.8	193.3	48.6	144.0	144.8	144.3	3.49	0.0390
	11.55	6.43	70.0	192.9	42.6	144.3	144.9	144.2	3.79	0.0342
	12.5	6.43	69	196.3	36.0	144.2	145.1	144.5	4.10	0.0284
	13.05	6.43	68.4	195.0	0	144.2	145.1	144.5	4.28	0
0.75	0	6.43	74.5	185.7	89.5	144.3	144.6	145.1	0	0.0747
	3.6	6.43	74.0	185.3	70.6	143.9	144.7	145.6	1.18	0.0591
	7.15	6.58	73.0	188.1	67.8	144.5	145.1	145.2	2.34	0.0559
	10.65	6.58	70.5	192.5	48.0	144.7	145.5	145.0	3.49	0.0387
	11.55	6.64	69.8	193.3	44.0	144.8	145.5	145.1	3.79	0.0353
	12.5	6.68	69.0	191.5	35.0	145	145.7	145.1	4.10	0.0283
	12.95	6.67	68.4	192	0	145	145.7	145.1	4.25	0
1.125	0	6.43	74.5	185.3	92.4	145	145.6	146.5	0	0.0773
	3.5	6.48	73.3	187.7	74.0	144.5	145.4	146.5	1.15	0.0611
	7.15	6.63	73.0	191.9	68.8	145.2	145.8	146.0	2.34	0.0556

Table 3 (Continued)

D (in.)	Q _s (gpm)	P _v (in. Hg)	$\frac{P_s - P_v}{\rho_s}$ (cm, sp gr 1.0)	P _n - P _d (cm, sp gr 12.6)	P _d - P _s (cm, sp gr 1.95)	T _v (°F)	T _s (°F)	T _n (°F)	φ	N
1.125	10.65	6.66	70.5	192.9	53.3	145.4	145.9	145.7	3.49	0.0428
	11.55	6.68	69.8	191.9	48.8	145.5	146.2	145.8	3.79	0.0394
	12.6	6.69	69.0	192.0	0	145.5	146.2	145.8	4.13	0
1.4	0	6.58	74.5	184.5	95.5	145.2	148	148.5	0	0.0803
	3.6	6.58	73.5	186.5	69.0	145	146.1	147.8	1.18	0.0574
	7.15	6.78	72.8	188.9	66.0	145.8	146.4	146.8	2.34	0.0542
	10.65	6.88	70.5	189.9	50.0	145.9	146.7	146.4	3.49	0.0408
	11.55	6.84	70.0	188.3	44.5	146.3	146.8	146.5	3.79	0.0366
	12.9	6.85	68.5	188.5	0	146.3	146.8	146.5	4.23	0
1.65	0	6.78	74.3	183.3	99.1	146.5	146.5	147.5	0	0.0838
	3.6	6.88	73.0	186.1	64.3	145.7	146.4	147.8	1.18	0.0536
	7.15	6.88	73.0	186.5	41.8	146.5	147.0	147.5	2.34	0.0347
	8.0	6.95	72.5	186.1	35.0	146.6	147.3	147.2	2.62	0.0292
	8.35	6.94	72	186.2	0	146.6	147.3	147.2	2.74	0

Table 4. Performance Characteristics of Model 2 for Various Nozzle Positions at Saturation Inlet Pressure with $Q_n = 3.05$ gpm and $T_s - T_n \cong 50^\circ\text{F}$

D (in.)	Q_s (gpm)	P_v (in. Hg)	$\frac{P_s - P_v}{\rho_s}$ (cm, sp gr 1.0)	$P_n - P_d$ (cm, sp gr 12.6)	$P_d - P_s$ (cm, sp gr 1.95)	T_v (°F)	T_s (°F)	T_n (°F)	ϕ	N
0.0625	0	6.05	74.1	183.7	86	142.1	143.2	90.5	0	0.0726
	3.95	6.28	73.5	188.7	77.6	143.1	143.2	90.5	1.30	0.0638
	7.15	6.23	72.5	188.5	67.3	143.2	142.8	86.5	2.34	0.0554
	10.65	5.88	70.0	190.7	54	140.8	141.1	87.8	3.49	0.0439
	11.55	5.68	69.5	189.9	49	141.0	139.6	87.7	3.79	0.0400
	12.5	5.53	68.5	190.3	46.5	138.1	138.4	87.8	4.10	0.0379
	13.45	5.36	68.0	189.9	42.3	137.0	137.1	86.6	4.41	0.0345
	14.1	5.38	67.8	189.8	0	137.0	137.1	86.6	4.62	0
0.25	0	5.30	74	185.7	87.5	137.5	137.5	87.2	0	0.0731
	3.5	5.28	73.5	189.3	79.8	137.4	137.5	87.2	1.15	0.0654
	7.15	5.28	72.8	191.3	72.0	137.2	137.3	85.0	2.34	0.0584
	10.65	5.13	70.0	191.9	58.3	135.9	136.2	86.5	3.49	0.0471
	11.55	5.03	69.0	191.8	52.8	135.0	135.0	86.3	3.79	0.0427
	12.5	4.93	68.5	192.3	50.5	134.3	134.5	86.2	4.10	0.0407
	13.45	4.90	67.5	191.9	46.0	133.7	134.0	85.9	4.41	0.0372
	14.45	4.91	67.0	191.8	0	133.7	134.0	85.9	4.74	0
0.375	0	4.90	74	191.3	91.3	133.4	134.8	82.6	0	0.0740
	3.6	4.88	73.5	187.3	78.6	133.5	134.8	82.6	1.18	0.0651
	7.15	5.03	72.5	190.7	71.4	135.3	136.3	83.9	2.34	0.0580
	10.65	5.18	70.0	190.3	56.0	136	136.7	86.0	3.49	0.0456
	11.55	5.28	69.0	192.7	53.0	136.7	137.4	86.8	3.79	0.0427
	12.5	5.43	68.5	191.9	49.5	137.6	138.0	87.3	4.10	0.0400
	13.45	5.45	68.0	193.3	46.0	138.4	139.2	88.5	4.41	0.0369
	14.65	5.46	66.9	193.2	0	138.4	139.2	88.5	4.80	0
0.75	0	6.05	74	189.7	92.5	141.8	139	84.3	0	0.0756
	3.55	6.03	73.5	187.9	70.3	141.6	138.9	84.4	1.16	0.058
	7.15	6.03	72.5	189.7	70.0	143	143.4	86.3	2.34	0.0572
	10.65	6.23	69.5	189.7	52.5	143.6	143.6	88.6	3.49	0.0429
	11.55	6.23	69.0	192.3	48	143.7	143.7	89.5	3.79	0.0387
	12.5	6.23	68.5	191.3	45.0	143.6	143.6	89.6	4.10	0.0365
	13.45	6.33	67.5	192.3	40.0	143.8	143.8	89.4	4.41	0.0322

Table 4 (Continued)

D (in.)	Q _s (gpm)	P _v (in. Hg)	$\frac{P_s - P_v}{\rho_s}$ (cm, sp gr 1.0)	P _n - P _d (cm, sp gr 12.6)	P _d - P _s (cm, sp gr 1.95)	T _v (°F)	T _s (°F)	T _n (°F)	φ	N
0.75	14.45	6.25	67.0	193.1	38.0	143.7	144.6	90.4	4.74	0.0305
	15.35	6.33	65.5	192.5	34.0	146.0	145.2	91.6	5.03	0.0274
	15.5	6.34	65.5	193.0	0	146.0	145.2	91.6	5.10	0
1.125	0	6.70	74	187.7	97.0	147.0	148.9	85.6	0	0.0801
	3.55	6.73	73.5	183.3	68.5	146.9	148.9	85.5	1.16	0.0579
	7.15	7.03	72.5	190.7	68.0	148	148.3	87.8	2.34	0.0553
	10.65	6.85	70.0	190.9	53.1	147.5	147.8	89.8	3.49	0.0431
	11.55	6.83	69.5	190.9	48.5	147.0	147.5	90.5	3.79	0.0394
	12.5	6.73	68.0	191.5	46.0	146.8	147.0	90.5	4.10	0.0372
	13.45	6.63	67.5	191.7	42.5	146.2	147.0	90.5	4.41	0.0344
	14.45	6.63	67.0	192.5	39.5	145.5	145.8	90.6	4.74	0.0318
	15.35	6.63	66.0	191.7	34.0	146.8	145.9	90.7	5.03	0.0275
	15.95	6.64	65.5	191.8	0	146.8	146.0	90.7	5.23	0
1.4	0	6.05	74	185.7	103.1	143	143.3	82.9	0	0.0861
	3.55	6.03	73.5	189.3	75.5	141.9	140.9	86.0	1.16	0.0618
	7.15	5.68	73.0	189.7	66.1	139.9	139.7	87.0	2.34	0.0540
	10.65	5.58	70.8	193.3	51.6	139.1	139.6	87.1	3.49	0.0414
	11.55	5.58	69.8	192.7	48	138.4	139.1	87.8	3.79	0.0386
	12.5	5.58	69.0	195.5	44.8	138.3	139.1	87.6	4.10	0.0355
	13.45	5.48	68.3	194.9	41.8	138.6	139.6	88.0	4.41	0.0333
	14.45	5.46	68	195.3	37.5	138.5	139.4	87.8	4.74	0.0298
	15.35	5.23	67.5	194.7	32.6	138.5	140.2	87.6	5.03	0.0260
	15.35	6.47	66.0	193.7	33.3	145.0	146.0	90.5	5.03	0.0267
	16.15	5.23	67	194.2	0	139.1	140.3	87.7	5.30	0
1.65	16.35	6.48	65.5	193.5	0	145.0	146.0	90.5	5.36	0
	0	6.40	74.0	184.3	107.1	143.9	144.5	84.4	0	0.0901
	3.55	6.38	73.5	183.7	63.0	143.8	144.4	84.3	1.16	0.0532
	7.15	6.48	72.5	183.7	43.5	143.7	145.0	86.5	2.34	0.0367
	8.9	6.48	71.5	183.1	22.0	144.4	144.8	87.6	2.92	0.0186
	9.8	6.38	70.5	183.3	17.0	144.1	144.6	88.1	3.21	0.0144
	10.65	6.28	70.0	184.9	13.2	143.8	144.3	88.3	3.49	0.0111
	11.55	6.23	69.5	185.0	8.3	143.5	143.9	88.5	3.79	0.0070
	12.15	6.25	69.0	185.5	0	143.6	144.0	88.6	3.98	0

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