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**CONTROL CHARACTERISTICS OF RANKINE CYCLE SYSTEMS
WHICH SIMULATE THE MPRE FLUID SYSTEM**

(Title Unclassified)

M. M. Yarosh

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ORNL-TM-1370

Contract No. W-7405-eng-26

REACTOR DIVISION

CONTROL CHARACTERISTICS OF RANKINE CYCLE SYSTEMS
WHICH SIMULATE THE MPRE FLUID SYSTEM

M. M. Yarosh

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CONTROL CHARACTERISTICS OF RANKINE CYCLE SYSTEMS
WHICH SIMULATE THE MPRE FLUID SYSTEM

M. M. Yarosh

ABSTRACT

In conjunction with the Medium Power Reactor Experiment Program at ORNL, studies were conducted on two single-loop Rankine cycle systems. These systems were controlled using the same design concept which has been proposed for control of the MPRE fluid system. Experiments were run with water and potassium test facilities which were reduced scale electrically heated mockups of the proposed reactor and power recovery system. Results from these studies demonstrate that with proper design, the distribution of liquid inventory within the system can be controlled by the degree of cavitation occurring in the turbine-driven centrifugal boiler feed pump. Boiler power changes were made over a wide range and no operator action was required. The changes in fluid flow requirements were accommodated by a variation in the degree of pump cavitation occurring. Step changes in power were imposed on the boiler and there was no indication of instabilities in the fluid system.

INTRODUCTION

The development of a single-loop boiling potassium Rankine cycle reactor system for space electric power production has, for the past several years, been the subject of a small but intensive effort on the part of staff members at the Oak Ridge National Laboratory. In planning the program investigations, it was recognized from the start that a significant effort would be required to develop and prove out a control which would permit the system to accommodate to changes in load. Analytical studies including both analog and digital computer work have been made of the relations between the principal parameters in the Rankine cycle system and this work has been reported.^{1,2}

The experimental studies on the behavior of the control system were conducted using a number of multi-purpose system test rigs designed to

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simulate the system by using electric heater rods in place of fuel elements. The systems were built to achieve several program objectives, including the determination of the basic feasibility of the proposed system control scheme, the behavior of individual components, and the stability and control behavior characteristics of the system as a whole. The early test systems were designed to operate with water because analysis indicated that water systems would have the same fluid flow characteristics as potassium systems. Water systems could be constructed more quickly and at a lower cost than liquid metal systems and with transparent parts, direct observation of the two-phase flow phenomena would be possible. Later in the program, potassium systems based on the water prototypes were constructed and the early tests conducted in water were repeated in potassium.

This report covers the experimental control studies conducted on two of the systems referred to above: The 9l-Rod Intermediate Water System (IWS) and the 9l-Rod Intermediate Potassium System (IPS). Additional work, principally on jet pumps and turbine pumps, has been conducted using other test facilities, and this work is discussed as necessary for the understanding of system operation. A comprehensive topical report covering the design, construction, and operation of the IWS is in preparation and a similar report on the IPS is planned.

THE CAVITATION CONTROL CONCEPT

One of the principal differences which distinguish a power conversion system designed for unattended operation from more conventional systems is the absolute necessity for exceptionally high system reliability. This often specified requirement in reactor systems is for the case of unattended operation essential to the success of the entire mission. If the system is to operate unattended, or under conditions which severely restrict the maintenance which can be performed, it profoundly affects the system design employed. The relative reliability of various systems proposed for space application has been examined by Fraas and Michel.³ Earlier studies had revealed that high reliability in conventional power conversion systems is often achieved through the use of 'on stream' or preventive maintenance.⁴

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Space electric power systems must depend on the initial design rather than preventive maintenance to achieve a high probability for satisfactory operation.

Studies which have been made on the reliability of nuclear power plants have shown that the process control instrumentation and electronic equipment, both nuclear and non-nuclear, account for over 50 percent of all plant malfunctions.⁵ Experience such as this favors the selection of a system that requires a minimum of control devices for satisfactory system operation.

The control concept proposed for the MPRE depends on matching the characteristics of the system and the various components in such a way that there is no need to operate any throttle valves over a wide range of system power outputs. Because all the vapor passes through critical pressure drops before condensation, the boiler exit pressure is almost a linear function of boiler power over the operating power range for the system. To accomplish the control objectives it is essential that the inventory distribution within the system be regulated in order to insure that adequate liquid is available to the boiler under all power conditions. The inventory distribution in the MPRE system will be controlled by a variation in the degree of cavitation occurring in the turbine pump unit which recirculates the liquid inventory. The control studies covered in this report present the experimental results obtained on inventory control and distribution in the test systems. The mechanism by which this control is achieved is explained in the following section.

Pump Cavitation Control

The concept of controlling inventory by varying the degree of cavitation in a pump has been employed in conventional power plants for some time. A technique called 'suction submergence control' allows the liquid level in the 'hot well' of a condenser in a power plant system to be regulated by the degree of cavitation in the condensate pumps.⁶ The degree of cavitation of the pump, and hence its capacity, is a function of the 'net positive suction head' (NPSH) on the pump. The value of the

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NPSH is in turn a function of the liquid inventory in the 'hot well' and thus the pump capacity is made a function of the system inventory distribution.

In the MPRE Program, the turbine driven pump will have performance and cavitation characteristics somewhat similar to the pump curves shown in Figs. 1 and 2. These curves were prepared from test data obtained for the pump used in the IWS.

The cavitation and performance data from Figs. 1 and 2 could be represented as shown by the typical pump curves depicted in Fig. 3. Also shown on this graph are two typical resistance curves for an assumed system. In steady state system operation, the pump head must match the system resistance. This is represented by operation at point Y on the graph for a system resistance represented by line 1. The pump speed is assumed to be 8000 rpm, and the pump is not in cavitation. With a change in power in the boiler of the assumed system, a change in pump flow and head is required to establish a new equilibrium condition, since the pump head and flow must match the system resistance at the new flow. The total inventory in the system is assumed to be constant.

Consider, for example, a reduction in power in the system such that the pump flow required is reduced from that at point Y to that at point X on Fig. 3. For equilibrium conditions, the new pump head must match the system resistance at point X. For this to occur without pump cavitation, the pump speed must be adjusted to give the curve n-n' on the graph or if the pump speed is held constant, an adjustment could be made in the system resistance. In this case equilibrium would be established with the new system resistance (line 2) at point Z on the graph. Either adjustment however would require instrumentation to sense the required change and controls and/or valves to effect the required change.

In the system control concept employing pump cavitation, a power change would be controlled in the following manner. Assume again the system to be operating at Y under steady equilibrium conditions with a fixed system resistance. A reduction in boiler power results in less vapor leaving the boiler and thus initially, since the pump head is unchanged, there is a tendency for inventory to shift to the boiler and hence deplete that available in the condenser. Refer to Fig. 4, a simplified flow

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Fig. 1. Aeronutronic Mark I Turbine Pump - Pump Head vs Pump Flow at Various Speeds.

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Fig. 2. Aeronutronic Mark I Turbine Pump - Typical Cavitation Test Results.

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Fig. 3. System Control by Pump Cavitation.

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Fig. 4. Flow Diagram for Pump Cavitation Control.

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diagram of such a system. In normal operation, two liquid-vapor interface points exist in the system; one in the boiler and one at the discharge of the condenser. The system can be designed so that minor changes in liquid inventory distribution will cause a substantial change in the NPSH on the turbine driven boiler feed pump. Thus the minor shift in inventory to the boiler during the power change under consideration will reduce the NPSH on the feed pump and cause it to pass into cavitation. This change is represented by the change in NPSH as one moves between points Y and X on Fig. 3 (NPSH - A and B on the graph). As the pump performance moves toward point X along the system resistance line, the liquid inventory will continue to transfer toward the boiler until the pump flow matches the lower boiler vapor flow at X. The initial mismatch results in inventory transfer which will reduce the NPSH on the pump. The ensuing cavitation will degrade the head rise across the pump until at point X the pump performance and system requirements are in equilibrium. At this point inventory transfer will cease. Note that the pump speed was not assumed to change, and thus the line n-n' does not indicate the pump speed since the pump is no longer operating on the noncavitating pump performance curve. Additional power changes will cause further shifts in system inventory and further changes in the degree of turbine pump cavitation.

In the simplified system described here, it is the minor shift in system inventory which changes the NPSH on the pump. This, in turn, affects the degree of pump cavitation until the pump head rise matches the new head required. Any tendency to transfer more liquid inventory to the boiler would degrade the pump head rise below that required and the transfer of liquid inventory would reverse because the pump head would be insufficient to supply the flow necessary to match the net effluent from the boiler.

Liquid Inventory Distribution and the Effect of Condenser Jet Pump Cavitation on System Control

The liquid inventory distribution may affect the NPSH in several different ways. In the test systems constructed and operated under the MPRE program, the operating liquid inventory is adjusted so that at full power

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conditions, an adequate liquid level is maintained in the expansion tank above the boiler when a small mass fraction of vapor is flowing through the condensate lines leading to the condenser scavenging jet pump. A shift in system inventory will change the vapor carry-over to line i-j, Fig. 4, and thus the pressure drop in this line changes, causing a change in the jet pump sideport suction pressure. By introducing the condenser jet pump into the circuit ahead of the turbine pump, the sensitivity of the system to an inventory change is increased. This can be seen by an examination of Fig. 5 which represents typical performance curves for such a jet pump.⁷

The pressure ratio N is defined as follows:

$$N = \frac{P_d - P_{sp}}{P_n - P_d}$$

where

P_d = pressure at jet pump discharge,

P_{sp} = pressure at jet pump sideport,

P_n = pressure at jet pump nozzle.

The flow ratio Φ is defined by:

$$\Phi = \frac{\text{Sideport Flow}}{\text{Nozzle Flow}}$$

The cavitation suppression head on the sideport of the jet pump represents the pressure head available to suppress cavitation in the jet pump. This suppression head is influenced by the static elevation existing in the sideport lines to the jet pump, the degree of liquid subcooling, and the pressure drop occurring in these lines. A slight decrease in the sideport cavitation suppression head may cause a large change in the jet pump performance by forcing the jet pump into cavitation. As a result, the head rise across the pump drops abruptly, and this in turn reduces the NPSH available at the boiler feed pump suction. Thus, a very minor shift in

Fig. 5. Effect of Cavitation Suppression Head on Jet Pump Performance.

inventory away from the condenser decreases the liquid inventory in line i-j. This increases the vapor void fraction and hence the line pressure drop, and results in a decrease in cavitation suppression head on the jet pump and a shift in pump operating conditions toward cavitation. It is in this way that the jet pump serves to amplify the effect of a minor shift in system inventory.

In a zero-gravity system where the static head will have no effect, the entire change in cavitation suppression head must result from the pressure changes and subcooling which occur in line i-j. It is important to recognize that this mechanism for controlling the liquid inventory distribution is sensitive to slight liquid-to-vapor ratio changes because of the very large vapor-to-liquid specific volume ratios in the system, and the strong effect of vapor on the two-phase flow pressure drop.

Under normal one-gravity operating conditions, the jet pump will scavenge the liquid from the radiator along with vapor which will be carried into line i-j decreasing the suppression head on the jet pump and resulting in pump cavitation. Any tendency to shift inventory into the condenser will tend to reduce the vapor content in the condensate line i-j. This will act to bring the jet pump out of cavitation, increase the condenser jet pump discharge pressure, and thus the NPSH on the turbine pump. This, in turn, will reduce the degree of cavitation of the turbine pump and hence increase its pumping capacity, thus reversing the inventory shift. The control of the inventory distribution within the system is thus dependent on the performance characteristics of the jet pump-turbine pump combination and not on the operation of sensing and control devices. The variation in pressure rise available from the jet pump alone would be insufficient to provide the system conditions required for the range of power desired. Thus cavitation in the centrifugal pump must exercise the major control action.

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STUDIES ON THE INTERMEDIATE WATER SYSTEM

One of the early test facilities which was designed and constructed for experimental studies in conjunction with the MPRE Program was the IWS. This facility consists of a boiler with 1/2-in.-diam heater elements spaced on 9/16-in. centers in a 91-rod array. The design power is 364 kw. A portion of the exit vapor feeds a turbine driven boiler feed pump. Figure 6 is a schematic of the system showing system components, valves, and primary instrumentation.

The control studies discussed in this report were conducted under Phase II and Phase III operation of the IWS. Currently the system is in Phase IV operation and some preliminary results from this operation are also included. Phase II and part of Phase III operation did not include the valve (V-20) downstream of the turbine. The regenerative feed heater shown downstream of V-4 was not included in the Phase II operation. The phase designations are used to separate, in time, system operation which has been conducted over a period of two years.

System Studies

One of the primary questions to be determined from operation of the test facility was the behavior of the system under conditions where the power to the boiler was varied over an extended operating range. Under these conditions the system was studied both with and without cavitation in the turbine pump. Other studies were made of the transition of the system from one type of operation to the other.

As indicated in previous sections of this report, the term 'cavitation control' is defined as the mode of operation in which the liquid inventory distribution is controlled by the degree of cavitation of the turbine pump. The term 'operator control' is defined as the mode of operation in which system inventory distribution is controlled manually by some operator action, such as adjustment of a valve.

Early tests with the IWS were conducted with the turbine pump operating under noncavitating conditions. This was accomplished by adjustments of the resistance imposed by the boiler feed valve (V-4), the turbine

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Fig. 6. Flow Diagram Intermediate Water System.

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speed as determined by the setting of the turbine throttle valve, and the system inventory. These adjustments were made with the system operating at constant power. The index for determining whether the system was in cavitation control or operator control was the behavior of the liquid level in the expansion tank. Figure 7 is a sketch of the vapor separator-expansion tank region located above the boiler. If the inventory in this region rose too high, it would tend to block the vapor passages. The liquid interface becomes highly turbulent under such conditions and liquid droplets are entrained in the vapor exit stream. A low liquid level, on the other hand, reduces the cavitation suppression head on the sideport of the boiler jet pumps and this, together with vapor bubbles which may enter the line to these jet pumps, tends to induce pump cavitation. Since on cavitation, the head rise across the jet pump drops and, because this pressure rise serves as the driving force for liquid recirculation in the boiler, cavitation of these pumps could lead to an excessively high exit quality and burnout conditions in the boiler. It should be pointed out that no attempt is made in the IWS to take advantage of dynamic forces provided by swirling of the liquid in the expansion tank to increase the cavitation suppression head on the jet pumps. In later system designs, dynamic forces are utilized for this purpose.

Figure 8 is a graph of two sets of data taken when the system was under operator control. For the series incorporating runs 21 through 24, the system was operated from 150 to 250 kw of power in steps of 25 kw, while runs 28 through 31 cover a range of 100 to 300 kw in similar steps. After each power change, it was necessary to adjust the boiler feed valve to maintain the liquid inventory in the boiler expansion tank within acceptable limits. From the graph it is evident that the boiler pressure is very nearly a linear function of the boiler power. The turbine speed and pump pressure rise increase with the boiler power.

The degree of cavitation in the centrifugal pump is determined by measuring the pressure rise across the pump during the test conditions. The degree of cavitation is expressed in terms of a cavitation coefficient as follows:

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Fig. 7. Separator and Expansion Tank Assembly-Intermediate Water System.

Fig. 8. System Performance During Operator Control.

$$N_c = \frac{\Delta P}{\Delta P'}$$

where

ΔP = the measured pressure rise across the pump,

$\Delta P'$ = the pressure rise computed from the pump performance curve at the measured speed and flow,*

N_c = cavitation coefficient.

When the pump is not in cavitation:

$$\Delta P = \Delta P'$$

$$N_c = 1$$

Under cavitating conditions, a degradation of the head rise across the pump occurs and the value of $N_c < 1$.

Included in the curves on Fig. 8 is a plot of the cavitation coefficients computed for this series of runs. Although the system required operator control, the cavitation coefficient was not necessarily at a value of one in these runs. This can be understood from an examination of Fig. 9. This graph represents typical curves for the IWS which are similar to those discussed on Fig. 3. When a mismatch exists between the boiler feed pump pressure rise required to meet the system resistance and that pressure rise produced by a noncavitating pump, the pump cavitation must be sufficient to accommodate the existing difference. As explained earlier, this cavitation is normally accompanied by a minor shift in system inventory which affects the degree of pump cavitation. If the degree of pump cavitation required is large and the expansion tank volume of the water system is insufficient to accommodate the inventory shift accompanying the change in cavitation, then the system will not reach a condition

*See Appendix.

Fig. 9. Pump Performance vs. System Resistance Curves.

of cavitation control before the liquid inventory in the expansion tank exceeds the acceptable operating limits. Thus cavitation of the turbine pump may be occurring as indicated by point A on Fig. 9, but equilibrium conditions will exist only at point B. Before the pump cavitation can proceed to B, the system inventory in the expansion tank may exceed the acceptable limits established. Thus in the IWS studies, even though the turbine pump is cavitating, the system cannot be permitted to proceed to cavitation control.

Initial design considerations with respect to expansion tank volume and operating inventory requirements for a system employing cavitation control must be carefully established so that an adequate expansion tank volume will be available in the system. In the case of the IWS, operation has indicated that the expansion tank volume may be slightly undersized. It should be emphasized, as will be discussed later in this report, that the mass of inventory shifted during a power change is small, and that in properly designed systems having the correct operating inventory, line sizes and system resistance, the system is inherently 'locked' into cavitation control over its entire operating range.

Figure 10 is a chart from an eight-channel, high-speed Sanborn recorder showing the results for test runs 21 to 24 with the system under operator control. The time is in units of minutes. As power in the boiler was increased, boiler pressure followed closely. Note particularly from the boiler flow that an adjustment of the boiler feed valve (V-4) setting was required after each power change. With the system in operator control (requiring adjustment of the boiler feed valve), the flow to the nozzles of the condenser jet pumps varied not only with power level, but with valve (V-4) adjustment. This, in turn, affects the turbine pump suction pressure and it behaved in an erratic manner with respect to pump speed and discharge pressure. The behavior of the jet pumps will be clarified later after consideration of operation under cavitation control.

Operation of the system under conditions of cavitation control was initiated by removing inventory from the system while operating at a fixed power level until it was determined that at that power level the liquid inventory distribution was under control. From this power level a series of tests were conducted by varying only the power to the boiler. No other

Fig. 10. Intermediate Water System-Response to Power Change Under Operator Control.

operator action was taken or required. Data taken at each power level are plotted in Fig. 11 to show the system behavior under cavitation control with conditions otherwise similar to those for the run shown in Fig. 8 with the system under operator control.

A comparison of Figs. 8 and 11 is instructive in understanding the behavior of the system. Figure 12 shows these two sets of data on the same coordinates. At comparable power levels the boiler exit pressure was quite similar. With the system under cavitation control (runs 54-64), the pressure rise across the turbine pump was significantly lower than under operator control, despite a somewhat higher pump speed. The data shown for the turbine pump in Fig. 8 and 11 was superimposed on the set of noncavitating turbine pump performance curves shown in Fig. 13. The measured pump speed in the test is indicated at each point. It is evident from the position of the points that the data from the runs without cavitation (21-24), match the calculated curves quite well. The data for the runs with cavitation (54-58), show substantially higher pump speeds for the head and flow values measured than is obtained from the performance chart.

Figure 14 is a graph comparing the cavitation coefficients from Figs. 8 and 11. During system operation under operator control, the existence and degree of turbine pump cavitation is influenced by the system resistance as determined by the boiler feed valve setting. Thus the cavitation coefficient may have a value of one, or less than one, and is not predictable unless the exact value of the resistance change is known. In the case of system operation under cavitation control, the value of the cavitation coefficient must be less than one and the system behavior is predictable if the system resistance is fixed.

The shape of the cavitation coefficient curve with the system under cavitation control can be understood from an examination of Fig. 9. The degree of turbine pump cavitation required is represented by the difference between the system resistance line which, under cavitation control, is matched by the actual pump pressure rise and the noncavitating pump pressure rise as calculated for the measured pump speed and flow. Note from the curves that at the cross-over points for these two curves, the cavitation coefficient will have a value of one. Between these two points the value of the cavitation coefficient is less than one and must at some

Fig. 11. I.W.S. Performance Under Cavitation Control.

Fig. 12. Comparison of I.W.S. Performance Data - Cavitation vs. Non Cavitation Control.

Fig. 13. Measured I.W.S. Pump Head-Flow Data vs. Non Cavitating Pump Performance Curves.

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Fig. 14. Comparison of Cavitation Coefficients - Control vs. Operator Control.

interim value reach a minimum. The power interval within which the cavitation coefficient value is less than one represents the interval over which cavitation control of the system will be effective. It is evident from Fig. 14 that neither the minimum nor maximum power limit for the system had been reached in the runs shown. As discussed earlier, a further limitation on the power interval might be imposed on the system by insufficient volume in the expansion tank to accommodate shifts in the liquid inventory distribution and by an excessive initial inventory for the system during the particular test series. These limitations can be easily remedied by appropriate design and operating changes.

Subsequent experience with the system and analysis of system operation demonstrates that the proper system inventory should be established at full power conditions. At this power level the system inventory should be the minimum inventory required for system operation under cavitation control. As the power is decreased from the maximum power level, a minor inventory shift to the expansion tank occurs and system control will proceed as described previously. This method of initial inventory adjustment was not used in the runs shown, and as a result the limits on the power interval operating range may have been restricted by improper inventory.

The studies discussed above were conducted under the Phase III operation. The primary changes from Phase II included the installation of a regenerative boiler feed heater in the system, construction of a new boiler having a modified method for spacing the heater elements, and improvements in the instrumentation.*,**

An additional series of tests was conducted with the system under cavitation control. The performance data are shown in Fig. 15. It is evident that, at the 100 kw power level, the cavitation coefficient curve has begun to increase, and this is consistent with the data and discussion given earlier on cavitation control. This series more clearly indicates

* All data bearing a 300 series number was taken during Phase III operation.

** A valve was installed downstream of the turbine and the bearing discharge line was capped off after data 334.

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Fig. 15. System Performance Under Cavitation Control - Phase III.

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the power range over which cavitation control of the system can be effective. The turbine pump pressure rise for this series has been plotted on Fig. 16 together with points for the pump pressure rise at the measured speed and flow as determined from the noncavitating pump performance curves. It can be seen that these data are similar to the typical curves shown in Fig. 9.

The graph of Fig. 15 indicates that the turbine pump pressure rise may fall below the boiler exit pressure at low power levels. It is only necessary that the nozzle feed pressure to the boiler jet pumps remain above boiler pressure to maintain boiler feed flow. At low power levels, little or no pressure rise is required from the boiler jet pumps since the boiler pressure drop is small and gravity recirculation is adequate. The actual boiler nozzle feed pressure may be above the turbine pump discharge pressure because the line losses at low power may be less than the difference in elevation between the turbine pump and boiler jet pump. Slight errors in instrument readings can also reverse the position of these curves.

The behavior of the system during power changes under cavitation control may be seen by an examination of the photograph of the Sanborn chart shown in Fig. 17. For convenience of reproduction the chart has been spliced and this is indicated by the time indications between channels 4 and 5. The Sanborn chart corresponds to data runs 305-309 previously graphed. This test series represents system operation from 300 kw to 100 kw with ramp changes of 50 kw between the data points.

It is evident from the graph that a change in boiler pressure followed the boiler power change within 2 to 4 sec. The turbine pump speed change lagged the power change by approximately 10 sec. The effect of the cavitating operation of the turbine pump can be seen in the channels showing boiler feed flow and pump discharge pressure. The small amplitude rapid fluctuations in discharge pressure are characteristic of cavitation operation in a centrifugal pump. The variation indicated in the feed flow is a result from the effects of the pressure pulse fluctuations at the static taps on the venturi meter. The actual feed flow is essentially constant at a fixed power level. The system response to power changes is seen to be stable over the power range investigated.

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Fig. 16. Effect of Pump Cavitation on Pump ΔP I.W.S.

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Fig. 17. I.W.S. Response to Power Changes Under Cavitation Control.

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The behavior of the system under an imposed step change in power was investigated to determine if a stepped system perturbation would impose instabilities on the system. The test system was operated under cavitation control at 200 kw and the power was stepped up 25 percent to 250 kw. No other operator action was taken. From the Sanborn traces shown in Fig. 18, it is evident that the system operation followed the step change in power and continued in cavitation control at the higher power level. The boiler pressure during this step change began to move almost simultaneously with power and followed the power with a time constant of approximately 40 sec. The time constant is defined as the time required for a variable to undergo approximately 63% of its net change after initiation of a perturbation. Pump speed followed the perturbation with a time constant of approximately 50 sec, while the boiler feed flow time constant was 40 sec. As the system power was raised one would judge from the Sanborn traces of the feed flow and feed pressure that the turbine pump cavitation decreased, and this is consistent with the previously shown shape of the cavitation coefficient curve.

During steady state system operation in cavitation control, minor shifts in inventory may occur; the degree is dependent on system design and where in the pump cavitation regime the system is operating. The sensitivity of the turbine pump to changes in NPSH determine how much of a suction pressure change is required to exercise control. The relative effects of vapor void fraction and liquid static elevation in the discharge lines from the condenser will be reflected in this suction pressure. Figure 19 shows a typical variation of the inventory during steady state operation. The variation in this case amounts to an inventory shift of approximately 2% of the system inventory.

Transition to Cavitation Control and the Role of the Condenser Jet Pump

An increased understanding of the system when it is in cavitation control can be obtained from studies of the system behavior during the transition from operator to cavitation control. A number of studies were conducted on the system during such transitions. Figure 20 is a graph of

Fig. 18. I.W.S. Response of System to Step Change in Power.

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Fig. 19. Expansion Tank Level vs. Time for System in Cavitation Control.

Fig. 20. I.W.S. Expansion Tank Level During Power Transition to Operator Control.

the expansion tank level with the system operating initially under cavitation control. The initial upset of the level at time 1100 occurred when the deaerator in the system was valved out. The system was operating at 200 kw and the power increased to 225 kw at 1245 in an attempt to gradually bring the system out of cavitation control. The cavitation coefficient as calculated increased from 0.94 to 0.97 during this power increase indicating the turbine pump was barely in cavitation. With an increase in power the change in separator level indicates a minor shift of inventory toward the condenser end of the system. At approximately 1420 a further power increase to 250 kw was made, and during the transient a check of the turbine pump indicated that the pump was no longer in cavitation. The effect on the expansion tank liquid level is evident. The system was deliberately forced out of cavitation control by power increases.

Reference to Figs. 9 and 15 will readily explain what had occurred. The noncavitating pump pressure rise curve crosses below the curve representing the system resistance. For the turbine throttle valve setting used in this case, the pump was unable to match the head requirements for the system, and thus the flow delivered to the boiler was less than required. The decrease of the liquid level in the expansion tank is a manifestation of this mismatch. After approximately 40 min, the boiler feed valve was opened to prevent the liquid level in the expansion tank from falling below the acceptable limit. Note that opening the boiler feed valve (V-4) reduces the system resistance which is equivalent to dropping the lower curve in Fig. 9. The turbine pump then automatically goes into cavitation and the system into cavitation control.

The relation between the principal system parameters during the transition between cavitation and operator control was investigated by operating the system under boiler feed valve control at constant power. The valve (V-4) was adjusted to increase the flow to the boiler above that required. Figure 21 is a graph of those system parameters important during the transition. At time 3:00 on the graph, the valve was opened slightly and a rise in separator level began, thus indicating that inventory was being transferred from the condenser end of the system. At time 3:10 a valve was opened and some of the liquid inventory was removed from the system by bleeding from the drain line connected to the boiler bottom

Fig. 21. I.W.S. Parameters During Inventory Adjustment to Cavitation Control.

plenum. At time 3:24 the drain valve was closed and removal of liquid inventory stopped.

Under conditions of operator control the system functions with a high turbine pump suction pressure provided by the noncavitating condenser jet pumps. The condensate lines from the condenser to the sideport of the jet pumps contain glass sections for visual observation of the condensate (see Fig. 6). During operator control in the above run, no vapor voids were observed in these lines. As inventory was transferred from the condenser, voids began to appear in the glass tubes of the condensate lines. This indicated that vapor was being carried out the ends of the condenser tubes to the jet pumps. Any subcooling previously occurring in the condenser was terminated and a rise in fluid temperature at the jet pump sideport can be seen on the graph. The introduction of vapor in this line increases the line pressure drop, and vapor may reach the jet pump sideport. The net result is a reduction in cavitation suppression head on the jet pump and a sharp decrease in the pump pressure rise. This is followed closely by a drop in the turbine pump suction pressure and by time 4:00 on the graph, the system had passed into cavitation.

Figure 22 is a trace from a Sanborn chart for a similar transition to cavitation control by the removal of liquid inventory. At time 1404 the system was operating under manual control and the system resistance adjusted so that liquid inventory was being transferred to the boiler. As inventory was transferred from the condenser, a drop in the turbine pump suction pressure occurred. This forced the turbine pump into cavitation as evidenced by the increase in the turbine pump speed and the decrease in the turbine pump discharge pressure. The evidence of cavitation can be seen in the fluctuation of the pump discharge pressure and the pressure sensor indicating the boiler feed flow. At time 1420 the system was under cavitation control. The initial reduction of the pump speed seen on the graph was later determined to be a result of an increase in end-loading on the pump following the reduction in suction pressure. This increased the thrust bearing drag and slowed the pump down until the reduced hydraulic loading on the pump over-compensated for the increased drag, and the pump speed then increased as expected. The thrust bearing loading was

Fig. 22. I.W.S. Transition to Cavitation Control.

subsequently reduced by machining grooves on the back side of the pump impeller and by improvements in the bearing flow passages.⁸

A profile of pressures throughout the system during the transition runs described above were graphed to determine what pressures change between operator control and cavitation control of the system. Figure 23 is a graph of pressures for the transition shown on Fig. 22 with the system in operator control and cavitation control, respectively. The only significant pressure variation occurs, as expected, in the condenser jet pump to turbine pump circuit.

During a transition to cavitation control as described above, data were recorded on the change in the pressure rise across the condenser jet pump, and these data are plotted in Fig. 24. The pressure rise across the jet pump decreased as shown, and the system passed into cavitation control.

The study of system pressures shown in Fig. 23 was extended to compare system pressures for two separate test runs at approximately equivalent power levels. One set of data is that taken during run 22, a test series with the system under operator control, while the other set of data was for run 56 during a test series under cavitation control. The comparison is shown in Fig. 25. From the graph one can see that there is a substantial difference in the turbine pump discharge pressure. In the fluid circuit leading to the boiler jet pumps through the boiler, turbine, condenser and the condenser jet pumps, the circuit pressures are seen to be essentially identical. Only in the portion of the system between the turbine pump and condenser jet pump does a significant difference exist. In this part of the system, the condenser jet pump nozzle pressure and the condenser jet pump discharge pressure are substantially higher when the system is in operator control. The jet pump pressure rise is seen to be about a factor of two greater for this case than with cavitation control. The effect of this is to keep the turbine pump out of cavitation and permit a 50% greater turbine pump pressure rise.

The jet pump data for the test series from which the above data were taken have been plotted and are shown on Fig. 26. Also shown on this graph is the calculated performance for the radiator jet pump under non-cavitating conditions based on the method of Cunningham.⁹ It can be seen from the graph that the data for the operator control series (21-24) agrees

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Fig. 23. I.W.S. Profile Comparison of Pressures Between Runs 319 and 320.

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Fig. 24. I.W.S. Condenser Jet Pump Performance - Transition to Cavitation Control.

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Fig. 25. I.W.S. Profile Comparison of Pressures Between Cavitation and Non Cavitation.

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Fig. 26. Condenser Jet Pump Performance Phase II I.W.S.

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with the predicted noncavitating performance for the pump. The data for the system under cavitation control shows a substantial degradation in jet pump performance. The reduction of the jet pump head ratio, even at low flow ratios, is believed to be the result of the large vapor or gas voids observed entering the sideport of the jet pump under cavitation conditions. More precise data on the effect of vapor (or gas) ingestion into the sideport of the jet pump is required before this can be confirmed. Under conditions of deep jet pump cavitation the pressure rise across the pump may be negative and the pump then acts as a resistance in the lines from the condensate headers to the turbine pump suction. This was shown by some of the test data in Fig. 5, and by data taken during operation under cavitation control in test series 305-309. Both sets of data indicate that, under conditions of maximum turbine pump cavitation, a net pressure loss can occur across the jet pump. This is shown by the data of Fig. 27. Also indicated in Fig. 27 is the net rise in pressure across the jet pump at the higher power levels as the turbine pump cavitation coefficient increases. The data indicate that, as the power is increased above 250 to 300 kw, the increase in jet pump pressure rise will cause the turbine pump suction pressure to increase and move the pump operating condition out of cavitation.

A graph of the product of the jet pump pressure rise multiplied by the turbine pump pressure rise is an indication of the pressure amplification between conditions of operator and cavitation control. Figure 28 shows that a good correlation of the test data can be obtained for the two modes of operation. The data are indicative of the difference in net energy output from the pumps under the two conditions as a function of the boiler total power output.

An electrical analog simulation of the IWS was constructed and operated using the ORNL analog computer. The results of the latter study have been reported and are in agreement with those test results from the IWS operation.¹ The analog studies indicated that the range of boiler power over which cavitation control of the system can be exercised may be extended by initial adjustments in turbine speed, boiler exit pressure, and liquid inventory in the system. The tests scheduled under Phase IV operation of the IWS are expected to confirm these results.

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Fig. 27. Condenser Jet Pump Performance and System Parameters Under Cavitation Control.

Fig. 28. Pump Pressure Rise Multiple vs. Power - Cavitation vs. Non Cavitation.

CONTROL STUDIES ON THE INTERMEDIATE POTASSIUM SYSTEM

Completion of construction of the Intermediate Potassium System (IPS) enabled the control studies conducted on the IWS to be extended to a similar system using potassium. The design of the IPS was patterned, as far as practicable, on that utilized for the IWS, with modifications indicated by IWS operating experience incorporated where possible. As in its water system counterpart, the IPS used a core consisting of 91 1/2-in.-diam heater rods, but spaced on 5/8 rather than 9/16 in. centers. The increased spacing was partly to facilitate the fabrication of the header sheets and in part a result of uncertainties existing in the final MPRE core configuration at the time of the IPS boiler design. The design power for the system was 364 kw.

Figure 29 is a flow diagram for the system showing the principal instrumentation and indicating the flow passages for liquid and vapor through the system.

The principal difference in the process design from that of the IWS consists in the method for removing heat from the condenser. In the water system the condenser is mounted in a large tank and water circulates past the individual condenser tubes. The potassium system employs an air-cooled heat shroud surrounding the condenser (or radiator) unit. Heat is transmitted from the radiator to the air-cooled shroud, or heat sink, by thermal radiation.

The turbine pump employed in the potassium system was essentially identical in design and performance to that used in the water system with differences only in the materials of construction. Figure 30 shows an isometric sketch of the IPS indicating the lines for liquid and vapor flow. Flowmeters are shown located on the turbine pump bearing feed line, on the main boiler feed line, and on the liquid line to the nozzles of the condenser jet pumps.

The principal objectives for the initial control studies conducted with the IPS were to confirm the control characteristics observed during the IWS operation, and to determine if there were essential differences in any of the characteristics observed. Because the IPS was also to serve as a test bed for the study of component performance and for endurance tests, the control studies were integrated into the other test work.

Fig. 29. Flow Diagram and Instrumentation on Intermediate Potassium System.

Fig. 30. Isometric of I.P.S. Piping and Components.

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One of the primary difficulties in conducting tests during operation of the IPS was the limitation on the range of power over which the system could be operated. The maximum power capability of the system was limited by the reliability of the electric heater elements used in the boiler core. The maximum power achieved in operation of the boiler was approximately 3-kw per rod. As operation continued, it became apparent from ground current leakage measurements in the heater elements that a limitation of between 2 and 2.5-kw per rod was more prudent, and consequently most of the control studies were limited to a maximum boiler power output of approximately 200 kw.

System Studies

Data from tests on the IPS were gathered through an electronic data system and were recorded on magnetic tape. This enabled the gathering of data more quickly than was possible on the water system, and therefore system fluctuations over shorter time periods could be recorded. A complete scan of the system data required approximately 10 sec.

The system normally operated under cavitation control and a series of tests were conducted by varying the power to the boiler with no external operator action taken. Test data were taken during operation over a power range from approximately 70 to 200 kw, and the data are shown in Fig. 31. In this particular series of runs, the power was reduced until the calculated cavitation coefficient approached a value of one. Further power reduction caused the system to move into operator control and boiler feed flow control was required. Examination of the graph indicates that at certain power levels a number of data points were taken. By use of the high speed data scanner, data were taken continuously and the system variations at a fixed power level were recorded. At a power level of approximately 120 kw the cavitation coefficient varied from approximately .68 to .84, while the pump speed varied from approximately 4700 to 5300 rpm. This variation was characteristic of the potassium system when under cavitation control, but the degree of variation appeared to depend on the depth of cavitation, the system inventory, and system resistance parameters.

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Fig. 31. Cavitation Control of I.P.S. to Coefficient of One.

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A comparison of the data shown on Fig. 31 with data from the IWS is shown on Fig. 32 and reveals a strong similarity in the characteristics of the two systems. In the cases compared, the systems were under cavitation control. There was no attempt in this comparison to duplicate conditions as far as the system pressures, pump speeds, or cavitation coefficients were concerned. In the data shown, it is evident that at corresponding power levels the potassium system was operating with the turbine pump cavitating to a much lesser degree than in the water system. For the data shown, the cavitation curve for the water system covers a much wider power range. The shape of the turbine pump speed curve, the pump discharge pressure, and the boiler exit pressure curves are remarkably similar in the two systems.

The response of system components in the IPS to power changes can be seen from an examination of the Sanborn recorder charts shown in Fig. 33. In this case a ramp change in power from 144 to 125 kw was made over a period of approximately 10 min. As seen from the traces, the system response closely follows the change in boiler power and this is indicative of a tight system coupling. A close comparison of Fig. 33 with Fig. 17 indicates a somewhat faster response of system parameters in the IPS than in the IWS. The design of the potassium system attempted a closer control of line sizes, lengths and system coupling and the somewhat faster response is a result of these efforts.

In order to determine the effect of a change in pump speed on the system parameters, an additional series of tests was conducted across a power range of approximately 110 to 170 kw and these data are shown on Fig. 34. A comparison of the data of Fig. 34 with those of Fig. 31 is shown in Fig. 35 and indicates a change in the position of the cavitation curve. Note that, at the increased pump speed, the power interval over which the cavitation coefficient is less than one is increased and most of this increase results from a reduction in the minimum power at which the cavitation coefficient reaches the value of one. This is in agreement with studies of a similar nature that were carried out on the analog simulation of the IWS.¹ In that study a system was operated under constant conditions except for changes in the pump speed. The results showed

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Fig. 32. Comparison of Cavitation Control Data I.P.S. and I.W.S.

Fig. 33. Response of I.P.S. to Power Change During Cavitation Control.

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Fig. 34. I.P.S. Performance Under Cavitation Control at Reduced Pump Speed.

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Fig. 35. Effect of Turbine Pump Speed on I.P.S. Performance.

that the pump remained in cavitation to lower power levels when the pump speed was increased from the initial reference conditions.

In the data compared above a difference in heat sink temperatures of approximately 50 to 100°F lead to the investigation of the effect of this parameter on system behavior. The test facility was operated under similar power range conditions at two heat sink temperatures. The data of Fig. 36 is a graph of the results. The data shown indicate no pronounced effect of heat sink temperature over the power range studied. The work, however, needs to be extended to cover higher and lower powers to determine if there is an effect on the interval over which the cavitation coefficient is less than one.

Studies of Transition to Operator Control

Several tests were conducted to determine the system behavior in the course of transition between operator and cavitation control conditions. From previous discussion it is apparent that one can shift to operator control of the system by several means. The shape of the cavitation coefficient curve indicates that, if the power can be extended beyond the interval of the range within which the cavitation coefficient is less than one, the system will pass into operator control. A test was conducted with the system operating at a high cavitation coefficient and power was gradually increased until the system gave indications of no longer being under cavitation control. Figure 37 is a graph of the data taken during this test series. Initially, as the power to the system was increased, the system response was normal. Note the drop in expansion tank level indicating a transfer of system inventory to the condenser was occurring. As the cavitation coefficient approached a value of one, a sharp rise in the turbine pump suction pressure occurred, the turbine speed and pump discharge pressure fell off, and the system passed out of cavitation control.

It is apparent from the rapid and significant response of the turbine pump suction pressure to the inventory shift that the jet pump probably came out of cavitation and, in turn, forced the turbine pump out of cavitation. This is confirmed by the Sanborn chart traces obtained in another

Fig. 36. I.P.S. Performance at Two Heat Sink Temperatures.

Fig. 37. Effect of Cavitation Coefficient of One on I.P.S. Performance.

transition study conducted, the results of which are shown in Fig. 38. These traces can be related to the data plotted in Fig. 37. Notice particularly the sudden pressure rise across the jet pump resulting in a rise in pump suction pressure, a drop in turbine pump speed, and a small drop in turbine pump discharge pressure. From time 9:21 on the graph, the flow to the boiler had to be regulated by means of the boiler feed valve. The erratic nature of the flow to the boiler under this regulation indicates that the operator was not able to exercise good control on the boiler feed flow with the type of valve installed. The actual transition to operator control represented by Fig. 38 was accomplished by a gradual throttling of the boiler feed valve and a throttling of the turbine inlet valve. This served to slow the turbine pump and move it in the direction of operator control. It is interesting to note that the transition to cavitation control appears to be dependent on the turbine pump suction pressure and not on the mode by which operator control is achieved.

The importance of the inventory distribution to cavitation operation was examined in the IPS by use of infrared photographs taken of the ends of the radiator tubes during operation under operator control and under cavitation control. The sharp color break evident in Fig. 39 is indicative of a substantial temperature change occurring axially along the length of the tubes. This is a result of liquid potassium accumulating in the condenser tubes because of the shift in system inventory to the condenser. The accumulated liquid becomes subcooled and a color change is seen on the infrared pictures. The system was under operator control. With the system under cavitation control there is no inventory accumulation in the condenser and hence no sharp temperature change occurs along the tubes. Figure 40 is a photograph similar to that taken previously except that in the latter picture the system was under cavitation control. Note from the photograph that both the radiator tubes and the liquid headers probably contain vapor since there is no indication of a drop in temperature which would occur if the passages were filled with liquid.

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Fig. 38. Transition to Operator Control in the I.P.S.

Fig. 39. Infrared Photograph of I.P.S. Radiator Tubes During Operator Control.

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Fig. 40. Infrared Photograph of I.P.S. Radiator Tubes During Cavitation Control.

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A SUMMARY AND CRITIQUE OF THE SYSTEM CONTROL STUDIES

The control studies conducted on the water and potassium test facilities have demonstrated a number of features of fundamental importance to the understanding of the proposed control scheme for the MPRE.

1. When the system is under cavitation control, power input changes to the boiler can be made with no additional control action required for satisfactory system operation.

2. Cavitation control has been demonstrated as effective only when the turbine pump cavitation coefficient remains less than one.

3. The existence of cavitation in the turbine pump does not mean that cavitation control of a system is in effect. It is necessary that the system inventory and line sizes be properly specified so that once the pump is in cavitation, the system can be permitted to proceed into cavitation control while maintaining an acceptable inventory distribution.

4. The cavitation coefficient curve, when plotted against power on the abscissa, has approximately a parabolic shape with the principal axis vertical and the vertex at or near the minimum point. The power interval between the points at which the coefficient has a value of one represents the power range over which the system will remain under cavitation control.

5. Operation of the system at power levels above and below the power interval defined by the cavitation curve results in the system passing into operator control. By adjustment of pump speed, system resistance, or liquid inventory, the system can be made to pass into operator control.

6. The transition to operator control occurs as a result of a substantial increase in jet pump pressure rise which increases the turbine pump suction pressure and drives the turbine pump out of cavitation.

7. The shape and position of the cavitation curve can be influenced by the operating system inventory, the system resistances and the turbine pump speed at defined reference conditions. The design of the turbine pump impeller can also influence this curve.

8. The condenser jet pump acts to amplify the effect on the turbine pump suction pressure of a shift in system inventory. The inventory shift changes the cavitation suppression head on the jet pump resulting in a large change in the jet pump head ratio. During operation under cavitation

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control, the jet pump scavenges almost all of the liquid from the condenser and little head rise occurs across this pump.

9. Ramp changes on the system indicate that there is a close coupling of the system components. Most system time constants are, in general, of the order of a few seconds.

10. Step changes of 20 to 50% in the boiler operating power level in the IWS while under cavitation control have revealed no tendency toward system instabilities.

11. Operation of the IPS facility has demonstrated the similarity of system response and control characteristics to those for the IWS. The savings in cost, time and complexity by conducting preliminary experimental work with water were considerable, and the increase in knowledge gained through visual observation at the transparent sections was invaluable.

Additional work is required to define more explicitly the relations between the parameters affecting the cavitation coefficient curve. It is necessary to determine precisely how to extend the power interval range for cavitation control. Although, as indicated in this report, some studies on this have been conducted, they have not been completed.

Tests for establishing the minimum inventory level at design power are necessary. Under the full-power condition the liquid inventory must be established at a value such that the turbine pump operates under slight cavitation conditions. These tests, and those on the power interval range, will be studied during the Phase IV operation of the water system. Similar studies are scheduled for the IPS after the installation of a new boiler core containing more reliable heater elements. This will permit operation of the IPS to the full power condition.

A study of the system performance indicates that it may be possible to operate over a narrow power range at the extremities of the cavitation curve with control of the liquid inventory distribution exercised by variations in the degree of jet pump cavitation. In such a case the turbine pump could be under noncavitating conditions with the condenser jet pump acting in the control capacity. The total pressure rise available from the jet pump, however, is at most only a few pounds per square inch, and thus its function as a control device is limited to a narrow power interval. At maximum system design power control, exercised by the jet

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pump would eliminate the necessity for turbine pump cavitation at the normal full power conditions.

The importance of vapor bubbles carried out of the condenser into the feeder lines to the condenser jet pump sideports needs further study. There is some meager evidence that, under heavy cavitation, vapor is carried through the jet pumps into the line leading to the turbine pump suction. It is important to know more concerning the liquid vapor distribution in these lines around the condenser jet pump. In this connection there are two difficulties that were encountered which are of importance in the system studies. In the IWS the difficulties of maintaining a system free from air in-leakage have been a persistent problem. The large number of mechanical joints together with thermal cycling and consequent relative movements of system lines have prevented maintaining system tightness once it is achieved. The presence of air in the lines from the condenser to the turbine pump affects the pressure drop in this line, and thus the performance of the turbine pump. Further test work with this facility may require hermetically sealing the system.

Perhaps the major difficulty in running tests with the IPS was associated with obtaining accurate pressure measurements at the condenser jet pump discharge and the turbine pump suction. These pressures were below 1 psia and the accuracy of the measurements obtained is questionable.¹⁰ Improvements in instrumentation and in measurement techniques have been made since initial operation began, but further developments are needed.

Work to date has concentrated on the demonstration and understanding of cavitation control. The operation of both the IWS and IPS has confirmed that the control concept proposed for the MPRE will regulate the system liquid inventory distribution over a wide range of power. With the initiation of the Phase IV water tests, and the installation in the IPS of a boiler containing improved electrical heater elements, it is hoped that the studies can be extended to cover the zero to full power operating range for both systems. This will necessitate detailed studies on startup techniques and operation at very low power levels. It is hoped that this work can proceed in the near future.

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APPENDIX

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The calculation of the cavitation coefficient was based on equations which represented the performance of the noncavitating pump, and on test data which represented the performance of the pump in actual operation.

The head coefficient ψ_m for the pump is defined by:

$$\psi_m = \frac{P_{TPD} - P_{TPS}}{1.94 \left(\frac{n}{1000} \right)^2} \quad (1)$$

where

P_{TPD} = turbine pump discharge pressure, psia,

P_{TPS} = turbine pump suction pressure, psia

n = pump speed rpm.

The flow ratio Φ for the pump is determined from the measured pump flow by:

$$\Phi = \frac{846 \Phi}{n} \quad (2)$$

where

Φ = pump flow in gpm.

An equation representing the noncavitating pump performance was fitted to the graph of ψ vs Φ and is as follows:²

$$\psi = \frac{1.08 - \Phi}{1.11 - \Phi} \left[0.624 + 0.260 \Phi - 0.231 \Phi^2 \right] \quad (3)$$

For the operating conditions of the pump, the value of ψ_m and Φ can be determined from measured data. The value of ψ in Eq. (3) above is computed using the measured value of Φ .

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The cavitation coefficient is defined by:

$$N_c = \frac{\psi_m}{\psi}$$

The cavitation suppression head on the jet pump sideport was computed as follows:

$$H = 27.7 (P_{CJPSP} - P_{SAT})$$

where

H = cavitation suppression head (in. water),

P_{CJPSP} = pressure at condenser jet pump sideport (psia),

P_{SAT} = saturation pressure at sideport inlet temperature (psia).

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Table A. 1. Turbine and Turbine Pump Circuit Intermediate Water System

Run No.	Power (kw)	T.P. Suction (psia)	T.P. Discharge (psia)	T.P. ΔP (psi)	T.P. Speed (rpm)	Cavitation Coefficient	T.P. NPSH (ft)	Suction Specific Speed	Turbine Inlet Pressure (psia)	Turbine Cavity (psia)	Turbine Discharge (psia)	Pressure Condenser Header (psia)	Bearing Flow (gpm)	Total Jet Pump Discharge Flow (gpm)
20	102.4	1.095	11.353	10.26	3500	.660	1.935	2290	3.055		.599	.098		1.151
21	150.4	3.305	22.353	19.05	4200	.854	6.991	1741	5.020		.845	.288		3.176
22	201.6	3.796	32.353	28.55	4800	1.030	8.029	2134	6.985		1.090	.524		4.498
23	225.3	1.832	38.353	36.52	5425	1.005	3.444	4637	7.967		1.336	.666		4.670
24	250.2	1.340	41.353	40.01	5950	.914	2.255	7297	9.441		1.336	.761		5.092
28	100.5	1.07	16.32	15.26	4000	.742	1.845	3728	4.008		.570	.116		2.178
29	149.1	1.31	26.32	25.01	4800	.860	2.367	4819	6.219		1.061	.211		3.671
30	200.3	1.07	38.32	37.25	5800	.894	1.702	8649	8.920		1.061	.305		4.937
31	175.0	1.31	30.32	29.01	5500	.755	2.295	5884	7.447		1.061	.305		3.979
54	247.7	1.33	29.34	28.01	6250	.559	1.975	7367	8.449		1.326	.609		3.855
55	226.6	1.09	25.34	24.26	6200	.491	1.444	8809	7.712		1.081	.835		3.501
56	197.1	0.84	19.34	18.5	5975	.403	.912	11201	6.484		1.081	.514		3.059
57	176	1.09	15.34	14.26	5650	.349	1.544	6293	5.747		1.081	.467		2.381
58	150.4	1.09	13.34	12.26	5400	.331	1.607	5118	5.010		1.081	.420		1.830
59	150.4	1.10	13.35	12.26	5350	.336	1.659	5058	5.202		.845	.335		1.911
60	176	0.85	15.35	14.50	5500	.373	1.032	8594	5.757		.845	.335		2.560
62	202.2	0.85	21.35	20.50	6000	.443	.934	11380	6.985		1.090	.429		3.249
63	227.2	1.10	25.35	24.26	6400	.461	1.466	9277	7.967		1.090	.524		3.731
64	250.2	1.10	30.35	29.26	6600	.523	1.356	10401	8.950		1.090	.571		3.924
305	302	2.31	35.09	32.78	5600	.813	2.031	5941	6.808	3.321	1.503	1.537	.413	2.845
6	250	2.36	26.79	24.43	5550	.617	2.323	4828	5.384	2.485	1.208	1.348	.339	2.341
7	200	1.51	18.59	17.08	5100	.515	0.699	9169	3.959	1.601	1.110	1.159	.248	1.642
8	150	1.36	11.62	10.26	4000	.502	0.822	5810	2.584	1.208	0.865	0.922	.194	1.378
9	97	1.03	5.73	4.70	2700*	.509	0.502	--	1.601	1.110	0.766	0.686	.074	.756
310	298	1.98	38.14	36.16	5600	.900	1.531	7698	7.049	3.512	1.400	1.392	.413	3.168
11	252	2.31	28.64	26.33	5150	.761	2.547	--	5.575	2.628	1.056	1.250	.355**	2.253
12	201	1.79	19.64	17.85	4800	.606	1.997	3964	3.979	1.646	0.909	0.872	.281	1.644
13	149	2.26	11.66	9.40	3700	.541	2.272	2200	2.726	1.400	1.056	1.203	.153	1.059
318	251.5	3.88	47.64	43.76	5700	1.05	7.715	2084	9.750	5.084	1.056	1.392	.413	2.453
19	251.5	3.73	48.14	44.41	5750	1.04	7.388	2303	9.505	5.084	1.056	1.297	.413	2.808
20	251.5	0.94	48.14	47.20	6600	.845	0.950	11672	9.505	4.593	0.909	0.824	.413	2.481

*Speed from graph (san born)

**Obtained this flow by graph extrapolation from other data

Table A. 2. Data for Condenser Jet Pump to Turbine Pump Circuit Intermediate Water System

Run No.	Power (kw)	P _{CJPSP} (psia)	P _{CJDP} (psia)	P _{CSPNP} (psia)	T _{CJPSP} (°F)	T _{sat} At P _{CSPSP} (°F)	Sideport ΔT Below T _{sat} (°F)	P _{sat} At T _{CJPSP} (psia)	Head Ratio N	Flow Ratio ϕ	P _{CJPSP} - P _{sat} (psi)	Cavitation* Suppression Head in. H ₂ O	ΔP _{CJP} (psi)	Discharge Flow North (gpm)	Discharge Flow South (gpm)	ΔP in Pump Suction Line P _{CJDP} - (T.P.) _{SUC} (psi)
20	102.4	.537	2.46	10.71	52	82	30	.192	.232	-0.034	.345	9.56	1.92	.625	.525	1.365
21	150.4	1.057	5.956	23.162	52	104	52	.192	.285	1.051	.865	23.96	4.90	1.707	1.469	2.651
22	201.6	1.152	8.037	32.162	56	107	51	.222	.285	1.445	.903	25.76	6.89	2.423	2.075	4.241
23	225.3	1.199	6.476	34.662	59	108	49	.247	.187	1.373	.952	26.4	5.28	2.477	2.193	4.644
24	250.2	1.294	6.713	38.162	63	111	48	.285	.172	1.407	1.009	27.9	5.42	2.682	2.410	4.881
25	225.3	1.257	5.672	34.162	60	104	44	.256	.162	1.011	.801	22.2	4.62	2.086	2.014	4.332
28	100.5	.649	2.994	14.132	52	88	36	.192	.210	.418	.457	12.66	2.35	1.067	1.111	1.924
29	149.1	.697	4.460	26.132	57	90	33	.230	.174	1.066	.467	12.94	3.76	1.850	1.821	3.15
30	200.3	.791	5.643	35.632	59	94	35	.247	.162	1.318	.544	15.07	4.85	2.477	2.460	4.573
31	175.0	.886	5.312	22.233	58	98	40	.239	.262	.609	.647	17.92	4.43	2.028	1.951	4.002
54	247.7	1.000	4.527	29.152	72	102	30	.389	.143	1.067	.611	16.93	3.527	.969	1.886	3.197
55	226.6	.953	4.196	25.152	70	100	30	.363	.155	.977	.590	16.34	3.243	1.780	1.721	3.106
56	197.1	.858	3.818	19.152	69	97	28	.351	.193	.862	.507	14.04	2.960	1.549	1.510	2.978
57	176	.858	2.919	16.152	66	97	31	.316	.156	.547	.542	15.01	2.161	1.174	1.207	1.829
58	150.4	.811	3.156	12.661	65	95	30	.306	.247	.131	.505	13.99	2.345	0.800	1.030	2.066
59	150.4	.726	2.692	10.706	63	91	28	.285	.245	.131	.441	12.21	1.97	0.800	1.111	1.592
60	176	.726	3.260	15.662	65	91	26	.306	.204	.498	.420	11.63	2.53	1.174	1.386	2.410
62	202.2	.774	4.017	20.662	68	93	25	.339	.195	.925	.435	12.05	3.24	1.630	1.619	3.167
63	227.2	.868	4.537	25.162	70	97	27	.363	.178	.966	.505	13.99	3.67	1.780	1.951	3.437
64	250.2	.915	5.057	29.662	73	99	26	.402	.168	.982	.513	14.21	4.14	1.910	2.014	3.957
305	302	1.922	2.64	35.682	116	125	9	1.51	.022	1.063	.412	11.33	0.718	1.226	1.619	0.33
6	250	1.686	2.64	26.182	110	120	10	1.27	.041	0.841	.416	11.38	0.954	.955	1.386	0.28
7	200	1.449	1.70	18.682	105	116	11	1.10	.014	0.005	.349	9.63	0.251	.435	1.207	0.19
8	150	1.260	1.41	11.708	100	110	10	0.949	.015	0.267	.311	8.61	0.150	.435	.943	0.05
9	97	1.070	1.41	6.509	94	104	10	0.790	.073	-.522	.260	7.76	0.340	.116	.640	0.38
310	297.6	1.730	2.31	37.226	111	121	10	1.31	.017	1.524	.311	11.57	0.580	1.549	1.619	0.33
11	252.2	1.541	2.50	28.226	106	117	11	1.13	.037	.782	.411	11.25	0.959	.955	1.298	0.19
12	201.0	1.210	1.88	19.226	96	108	12	0.841	.039	.127	.369	10.22	0.670	.533	1.111	0.09
13	149.1	1.541	2.64	11.752	109	117	8	1.24	.120	-.733	.301	8.37	1.099	.116	.943	0.38
318	251.5	1.824	6.33	47.226	73	123	50	0.402	.110	.160	1.422	39.4	4.506	.800	1.653	2.45
19	251.5	1.583	6.04	48.226	73	118	45	0.402	.106	.627	1.186	32.85	4.452	1.121	1.687	2.31
20	251.5	0.878	2.92	48.226	75	97	22	0.430	.045	.468	.448	12.43	1.042	1.012	1.469	1.98

*This does not include dynamic head effect on sideport pressure

CJPSP = Condenser jet pump sideport pressure

CJDP = Condenser jet pump discharge pressure

CJPNP = Condenser jet pump nozzle pressure

Table B. Dextir Data From IPS System

Run			Boiler Power (kw)	Turbine Speed (rpm)	Separator Level (in.)	Boiler Exit (psia)	Turbine Pump Suction (psia)	Turbine Pump Disch. (psia)	Turbine ΔP (psi)	Cond. Jet Pump ΔP (psi)	Boiler Feed Flow (gpm)	Cond. Jet Pump Disch. North (gpm)	Cond. Jet Pump Nozzle Flow (gpm)	Bearing Flow (gpm)	Head Coef. Ratio	Condensate Jet Pump		Radiator Inlet (psia)	Condensate Temp. (°F)
Day	Hour	Min.														Head Ratio	Flow Ratio		
																N	ϕ		
244	14	19	105.7	4771	3.14	10.03	-263	14.1	1.985	.091	1.072	1.173	.795	.478	.669	.006	1.949	.003	827.1
244	15	37	125.3	5195	3.23	11.89	-189	18.4	2.393	.090	1.214	1.319	.878	.545	.738	.005	2.003	.106	865.6
244	20	00	124.9	5203	3.19	11.97	-141	18.0	2.417	.087	1.240	1.335	.881	.549	.718	.005	2.029	.162	867.3
245	0	23	121.7	5112	3.10	12.08	-049	17.3	2.353	.090	1.232	1.318	.868	.537	.708	.005	2.039	.065	851.3
	0	23	121.4	4833	3.1	11.69	-065	17.2	2.325	.090	1.250	1.330	.881	.528	.789	.005	2.018	.069	852.1
	0	23	121.1	5206	3.0	11.71	-054	17.4	2.346	.087	1.232	1.325	.875	.543	.689	.005	2.029	.067	851.3
	0	23	121.3	4669	3.0	11.75	-060	17.2	2.341	.087	1.256	1.330	.883	.521	.844	.005	2.014	.065	851.3
	0	26	121.9	4918	3.12	11.75	-055	17.3	2.331	.091	1.210	1.229	.864	.523	.766	.005	2.006	.072	852.1
	0	26	121.8	5262	3.14	11.75	-063	17.6	2.342	.086	1.219	1.309	.868	.532	.682	.005	2.018	.067	852.1
	3	15	123.0	4913	3.10	11.80	-064	17.3	2.343	.081	1.295	1.362	.892	.537	.767	.005	2.053	.064	850.8
	7	55	125.0	4878	3.14	12.17	-061	18.0	2.371	.089	1.308	1.376	.895	.549	.812	.005	2.075	.089	858.0
	9	41	142.9	5929	3.37	14.08	.063	22.3	2.708	.033	1.481	1.549	.992	.626	.680	.001	2.124	.204	901.9
	11	17	167.8	6117	3.23	16.60	.209	29.3	3.031	-.186	1.729	1.763	1.104	.693	.841	-.006	2.194	.339	927.2
	12	47	166.9	6082	3.39	16.18	.100	28.15	3.062	-.098	1.683	1.728	1.095	.678	.826	-.003	2.157	.279	969.5
	16	7	188.3	6538	3.05	18.12	.078	34.23	3.451	-.266	1.940	1.940	1.209	.730	.866	-.008	2.208	.339	939.1
	19	0	193.1	6611	3.19	18.49	.124	35.90	3.533	-.279	1.994	1.967	1.228	.713	.888	-.008	2.205	.370	946.7
246	3	3	192.8	6576	3.28	18.61	.231	35.22	3.569	-.279	1.972	1.949	1.214	.712	.879	-.008	2.210	.446	955.1
246	8	13	192.3	6625	3.03	18.79	.263	35.65	3.546	-.278	1.975	1.949	1.212	.711	.876	-.008	2.216	.430	956.0
	11	36	134.6	5827	3.12	12.77	-.182	20.04	2.524	.087	1.357	1.434	.926	.585	.636	.004	2.098	.061	858.9
	11	48	134.4	5805	3.05	12.73	-.186	20.38	2.609	.096	1.370	1.446	.932	.590	.652	.005	2.105	.078	859.3
	13	49	98.4	4418	3.01	9.33	-.308	12.82	1.884	.073	.970	1.080	.748	.443	.709	.006	1.888	-.059	780.3
	15	24	79.5	3282	3.14	7.22	-.336	9.53	1.445	.060	.791	.871	.649	.302	.960	.006	1.684	-.066	730.0
	19	1	167.5	6119	3.50	15.83	-.203	28.45	3.076	-.046	1.695	1.737	1.099	.681	.825	-.002	2.162	.245	915.4
247	3	3	170.7	6173	3.19	16.43	-.074	29.03	3.168	.024	1.746	1.777	1.114	.695	.824	-.002	2.192	.265	917.1
	19	38	169.6	6128	3.32	16.04	-.279	28.72	3.119	-.063	1.735	1.767	1.108	.690	.833	-.002	2.189	.221	913.7
	19	38	169.1	6118	3.16	16.03	-.275	28.95	3.121	-.059	1.715	1.746	1.096	.682	.842	-.002	2.187	.214	911.2
	19	39	168.3	6131	3.21	16.06	-.275	28.70	3.106	-.071	1.735	1.765	1.108	.686	.831	-.002	2.185	.221	911.2
	19	39	168.2	6113	3.44	16.04	-.275	28.85	3.103	-.064	1.719	1.755	1.105	.685	.841	-.002	2.175	.233	919.6
	20	02	169.8	6146	3.16	16.11	-.264	28.95	3.167	-.049	1.739	1.773	1.110	.697	.833	-.002	2.195	.293	904.9
248	3	12	165.4	6133	3.25	15.79	-.136	28.08	3.117	.008	1.730	1.752	1.091	.684	.810	.000	2.213	.178	918.0
248	18	59	169.1	6173	3.41	15.96	-.270	28.28	3.113	-.027	1.785	1.804	1.117	.706	.808	-.001	2.230	.192	910.8
249	4	12	167.2	6189	3.39	15.94	-.188	27.86	3.103	-.000	1.728	1.259	1.101	.688	.788	-.000	2.194	.168	904.4
249	12	18	169.1	6183	3.12	15.91	-.253	28.6	3.137	-.016	1.623	1.684	1.071	.674	.813	-.001	2.114	.183	908.6
249	19	0	169.1	6182	3.41	15.84	-.398	28.62	3.117	-.053	1.675	1.728	1.099	.682	.820	-.002	2.144	.188	917.5
	4	9	165.4	6136	3.10	15.70	-.267	27.69	3.085	.013	1.643	1.692	1.071	.671	.799	.000	2.159	.150	902.7
250	8	14	165.9	6112	3.19	15.91	-.153	27.78	3.094	-.020	1.663	1.707	1.078	.672	.806	-.006	2.167	.158	912.4
	10	16	98.6	4047	3.05	9.13	-.383	12.92	1.856	.057	1.013	1.105	.756	.442	.855	.004	1.924	-.095	778.2
	10	54	80.16	3749	3.10	7.42	-.379	9.56	1.473	.052	.795	.884	.647	.325	.741	.005	1.732	-.101	738.1
	13	5	70.88	3103	3.25	6.46	-.453	8.25	1.313	.044	.749	.810	.615	.257	.944	.005	1.637	-.106	724.5
	13	7	71.2	2920	2.87	6.50	-.449	8.14	1.309	.052	.736	.806	.507	.170	1.076	.006	.603	-.106	724.9
	13	19	76.9	3016	3.07	6.97	-.465	8.68	1.425	.049	.579	.695	.586	.225	1.053	.005	1.373	-.106	732.6
	13	19	76.8	3027	3.10	6.93	-.461	8.90	1.408	.050	.610	.717	.600	.225	1.070	.005	1.392	-.106	733.0
	13	19	76.8	3009	3.03	6.90	-.461	8.79	1.423	.046	.598	.702	.584	.222	1.071	.005	1.405	-.106	732.6
	13	19	76.3	3142	2.96	6.90	-.464	8.57	1.402	.046	.648	.742	.607	.229	.958	.005	1.443	-.105	732.2
	13	20	76.2	3225	3.03	6.97	-.464	8.60	1.421	.049	.827	.864	.646	.255	.911	.005	1.675	-.106	732.2

Table B. (continued). Dexter Data From IPS System

Run			Boiler Power (kw)	Turbine Speed (rpm)	Separator Level (in.)	Boiler Exit (psia)	Turbine Pump Suction (psia)	Turbine Pump Disch. (psia)	Turbine ΔP (psi)	Cond. Jet Pump ΔP (psi)	Boiler Feed Flow (gpm)	Cond. Jet Pump Disch. North (gpm)	Cond. Jet Pump Nozzle Flow (gpm)	Bearing Flow (gpm)	Head Coef. Ratio	Condensate Jet Pump		Radiator Inlet (psia)	Condensate Temp. (°F)
Day	Hour	Min.														Head Ratio H	Flow Ratio F		
250	13	20	75.5	3253	3.05	6.94	-465	8.74	1.417	.048	.905	.920	.669	.266	.909	.005	1.751	-.107	733.4
	13	20	75.2	3337	3.10	6.91	-464	8.74	1.419	.048	.919	.926	.662	.270	.864	.005	1.795	-.107	733.0
	13	20	74.2	3389	3.14	6.85	-469	8.81	1.406	.046	.950	.953	.672	.283	.844	.005	1.835	-.107	733.0
	13	20	73.3	3349	3.21	6.83	-471	8.56	1.391	.047	.956	.964	.676	.297	.842	.005	1.853	-.105	733.0
	13	21	72.5	3292	3.30	6.76	-467	8.64	1.363	.048	.808	.861	.637	.278	.878	.005	1.706	-.105	733.4
	13	21	72.2	3216	3.32	6.68	-469	9.03	1.363	.043	.788	.840	.631	.261	.960	.005	1.662	-.106	733.0
	13	21	71.7	3307	3.39	6.81	-469	8.90	1.361	.046	.767	.825	.624	.259	.896	.005	1.643	-.106	733.0
	13	21	70.1	3251	3.39	6.70	-469	9.01	1.338	.050	.778	.830	.624	.258	.939	.005	1.659	-.106	733.4
	13	21	69.9	3093	3.41	6.68	-469	8.82	1.325	.047	.739	.801	.611	.252	1.015	.005	1.622	-.106	733.4
	13	21	69.6	3087	3.46	6.61	-472	8.85	1.316	.046	.663	.745	.591	.236	1.023	.005	1.522	-.107	733.0
	13	21	69.3	2946	3.50	6.60	-472	8.69	1.312	.049	.399	.572	.546	.199	1.112	.005	1.096	-.104	733.0
	13	22	70.7	2653	3.23	6.60	-469	8.89	1.325	.045	-.100	.287	.501	.172	1.43	.005	.144	-.105	733.0
	13	22	73.8	2881	2.96	6.66	-469	8.60	1.367	.056	.242	.474	.524	.182	1.16	.006	.810	-.105	732.6
	13	22	76.3	2914	2.69	6.81	-487	8.61	1.397	.052	.607	.707	.595	.212	1.12	.006	1.377	-.105	732.2
	15	37	170	6183	3.37	16.15	-.345	28.81	3.121	-.076	1.772	1.799	1.121	.705	.823	-.003	2.210	-.185	920.5
	19	59	173.6	6231	3.39	16.28	-.317	29.52	3.206	-.139	1.760	1.791	1.116	.706	.831	-.005	2.210	-.219	930.6
251	3	9	173.4	6213	3.23	16.68	-.134	30.03	3.202	-.098	1.773	1.805	1.126	.711	.845	-.003	2.205	-.227	926.4
251	20	27	172.0	6197	3.28	16.25	-.382	29.42	3.169	-.092	1.737	1.773	1.112	.698	.838	-.003	2.190	-.195	923.0
252	3	3	171.4	6201	3.41	16.45	-.227	29.74	3.212	-.071	1.773	1.771	1.112	.697	.841	-.002	2.185	-.237	920.5
257	8	00	170.2	6516	2.91	16.20	.311	29.27	3.33	.002	1.728	1.776	1.115	.708	.732	.000	2.185	-.126	760.7
	9	35	168.8	6343	3.14	15.98	.317	28.63	3.30	-.004	1.704	1.756	1.101	.707	.752	-.000	2.191	-.135	713.4
	10	31	168.3	6315	3.01	15.77	.409	29.07	3.29	-.003	1.669	1.740	1.110	.701	.771	-.000	2.135	-.140	709.9
	11	24	167.2	6073	2.73	15.77	.437	28.65	3.12	-.003	1.797	1.798	1.145	.655	.823	-.000	2.142	-.136	705.7
	13	5	167.5	5896	2.69	15.95	-.493	28.95	3.01	-.010	1.696	1.724	1.111	.640	.880	-.000	2.102	-.140	679.9
	13	48	168.3	5944	2.60	15.93	-.513	29.03	3.01	-.017	1.721	1.745	1.112	.657	.866	-.001	2.138	-.140	669.6
	14	9	167.2	5732	2.60	16.02	.525	29.07	2.78	-.009	1.708	1.709	1.115	.595	.935	-.000	2.065	-.140	677.8
	14	24	167.2	5629	2.51	16.13	.533	29.43	2.70	-.010	1.701	1.695	1.118	.570	.982	-.000	2.030	-.143	679.9
	14	36	168.2	5480	2.46	16.19	.533	29.32	2.61	-.012	1.714	1.686	1.118	.539	1.03	-.000	2.015	-.141	681.7
	14	49	167.8	6345	2.46	15.10	.572	28.63	3.28	-.023	1.729	1.753	1.068	.708	.75	-.001	2.283	-.138	658.5
340	13	33	152.5	4953	2.80	14.66	.735	20.88	2.11	.234	1.505	.534	-.917	.480	.901	.012	-2.165	.748	836.0
	14	3	162.0	5052	2.59	15.55	.726	23.07	2.24	.159	1.603	.591	-.907	.487	.959	.007	-2.305	.870	850.4
	14	22	169.8	5201	2.47	16.31	.722	24.68	2.30	.134	1.681	.641	-.907	.507	.970	.006	-2.413	.971	861.8
	14	43	170.6	5141	2.49	16.37	.844	24.93	2.99	.211	1.689	.660	-.877	.507	.997	.009	-2.504	1.00	864.0
	15	9	178.7	4753	2.00	17.22	2.609	24.46	2.33	2.30	1.909	.806	-.829	.532	1.047	.105	-2.944	1.182	870.3
341	9	24	169.6	5230	2.66	16.15	.878	24.30	2.32	.081	1.670	.619	-.932	.499	.940	.003	-2.328	1.00	853.1
	9	56	169.6	5743	2.74	16.55	.877	24.72	2.57	.048	1.670	.700	-.867	.596	.792	.002	-2.615	.961	859.7
	10	25	181.3	5971	2.68	17.77	.859	27.22	2.77	-.077	1.807	.786	-.842	.607	.810	-.003	-2.853	1.163	876.6
	12	24	141.0	4911	2.61	13.85	.842	17.53	2.25	.220	1.377	.497	-.886	.503	.758	.013	-2.121	.612	816.0
	12	43	134.9	4845	2.60	13.35	.821	17.38	2.14	.234	1.318	.462	-.906	.512	.775	.014	-2.021	.590	812.6
	13	17	121.8	4350	2.73	11.92	.840	12.45	1.97	.176	1.071	.322	-.909	.483	.677	.015	-1.709	.423	783.3
	13	18	121.9	4478	2.72	11.96	.845	16.94	1.96	.155	1.446	.584	-.710	.432	.866	.010	-2.644	.446	782.8
	13	18	121.8	4618	2.71	11.92	.830	17.26	1.97	.196	1.415	.595	-.678	.453	.831	.012	-2.755	.417	782.8
	13	18	121.9	4593	2.89	11.98	.829	13.75	1.96	.187	.337	.026	-.711	.426	.702	.014	-1.073	.426	783.3
	13	57	111.5	3962	2.63	10.90	.791	12.91	1.78	.164	1.037	.227	-.961	.379	.855	.014	-1.473	.330	759.4
	13	57	111.8	3987	2.55	10.91	.798	13.52	1.78	.149	1.154	.406	-.725	.382	.689	.012	-2.120	.362	759.9
	13	57	111.7	4198	2.48	10.84	.805	14.87	1.80	.149	1.291	.490	-.711	.400	.861	.011	-2.379	.340	754.4
	13	57	111.5	4242	2.68	10.96	.774	15.30	1.79	.177	1.317	.499	-.720	.401	.871	.012	-2.385	.363	759.0
	15	15	155.4	5388	3.02	15.13	.793	21.54	2.43	.148	1.558	.627	-.872	.568	.782	.007	-2.437	.770	841.9

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