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CFDTS DATA ANALYSIS
CY 1965 YEAR END REPORT

JANUARY 1966

CONTRACT SNP-1

NERVA PROGRAM

MASTER

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ROCKET ENGINE OPERATIONS - NUCLEAR

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ABSTRACT

Analyses of data from the first series of the Cold Flow Development Test System experiments were conducted to examine chilldown and startup characteristics of the system. This first series of tests, involving six individual tests, demonstrated the bootstrap ability of the NERVA Engine system under conditions of 50, 60, 70, and 90 psia tank pressure with 14 psia back pressure to the nozzle and turbine. Because the tests are still in progress, a complete evaluation of startup procedures and characteristics could not be made. Rather, the data analyses presented herein concentrated on obtaining information which would aid in upgrading and improving prediction capabilities.

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I. INTRODUCTION

The work reported herein is the result of the data analysis of the Cold Flow Development Test System (CFDTS) tests which were made during the 1965 NERVA Contract Year. Six tests, designated 1.2-02-NNS-001 through 006, were made. These constitute CFDTS Test Series I and represent approximately a third of the total number of tests scheduled to be performed. Since the test program is still in its initial stage, any conclusions reported here will be subject to verification and/or modification as future tests are completed. A final data analysis report for the complete CFDTS Test Series will be issued in Contract Year 1966.

The Cold Flow Development Test System is a full scale NERVA rocket engine test system with an unfueled reactor assembly. It is the first test system to incorporate all major NERVA engine components. Major components of the system are prototype models of the NRX/EST and X-engine components, although there are certain minor components which are unique to the CFDTS (Reference 1). The CFDTS run tank propellant line is much shorter than that of NRX/EST at Test Cell A. This means that tank pressures required for a given performance must be increased to allow for line losses when extrapolating from CFDTS to NRX/EST. There is also a difference in the jacket location on the turbine inlet line, but it is not expected that this difference will significantly influence the overall reliability to NRX/EST. Another significant difference is the turbine. Changes in turbine design to improve its performance have not been incorporated in the first series of tests. The improved turbine, to be used in NRX/EST, was used in the next series of CFDTS tests.

The function of the CFDTS test program is to provide experimental information on transient startup characteristics of the NRX/EXT and XE NERVA engine systems during the chilldown period of engine bootstrap. Using this information, the proper startup procedures and safe operating regimes are to be established. Data will be obtained on chilldown, temperatures, pressures, flow rates, and pressure drops and the effects, at various environmental back pressures, of propellant tank pressure, turbine power control valve setting, and diluent valve

setting. To effect a complete understanding of the system operation, the mathematical model which has been used will be verified or upgraded as required. The test specifications and Test Plan for CFDTs can be found in References 2 and 3, respectively.

In evaluating the data, the guiding principle has been to perform those activities which contribute to the fulfillment of the test objectives. Some conclusions can be drawn quickly. For example, the question of whether a bootstrap startup is possible under given test conditions can be answered after a cursory examination of the data. However, it is desired to obtain information which will aid in the extrapolation from the test condition to other sets of conditions. It is for this extrapolation that data are used to upgrade analytical models of the system and improve prediction capabilities, which is tantamount to increasing the understanding of the system operation. In the analysis of data from this first series of tests, emphasis has been placed on obtaining data which will be used in the future for upgrading analytical models.

Although available engine models and programs have been successfully checked against the NRX-A2 and NRX-A3 (References (4) and (5)), their usefulness was to provide verification of component representation in the power ranged operation. The NRX-A2 and -A3 models were for open loop rather than closed loop engine operation and included a facility, rather than engine, TPA. The turbine drive for NRX-A is independently supplied from an H_2 gas farm and not from the engine itself as in CFDTs. Therefore the NRX-A model did not include the engine turbopump assembly, the hot bleed and diluent lines, and the propellant feed line.

II. SUMMARY AND CONCLUSIONS

The first series of the Cold Flow Development Test System tests provided preliminary information on the chilldown and startup characteristics of the NERVA engine system. Objectives of the tests were accomplished.

Comparison of chilldown runs with wet and dry pump starts shows a substantial energy contribution from the pump inlet line and pump housing. There is some uncertainty concerning the chilldown behavior of the pump inlet line. Judging from the sparse line temperature data available, it appears either that the line support structure is contributing significant energy to the line or that there exists a very low rate of heat transfer in the line.

Results of chilldown calculations using the 364 program did not agree well with measurements. The data indicated where improvements could be made; however, no significant changes will be made to the program until additional fluid temperature measurements become available.

Bootstrap startup was accomplished for tank pressures of 50, 60, 70, and 90 psia with 14 psia back pressure. The increase in tank pressure from 50 to 60 psia showed the greatest influence on overall performance, while further increase produced progressively less effect.

Values of pump efficiencies were estimated, using the system performance data. Since these efficiencies are applicable in the low speed operating region, they provide turbopump information which was not available prior to CFDTIS tests. While the calculated efficiencies appear consistent, their validity cannot be tested until data from the next series of tests become available.

There were indications of temperature asymmetry in the system, but there was insufficient data to establish conclusions concerning the causes of the asymmetry. Two-phase flow does not appear to be the sole cause of the temperature asymmetry experienced in the nozzle tube outlet plenum. The hot bleed port may have contributed to the asymmetry.

Heat transfer coefficients currently used in the Engine System Simulator Program (364) were evaluated through comparison of calculated with measured temperatures for isolated components of the system. It appears that the two-phase and liquid heat transfer correlations need modification, while the gas phase correlation appears to be fairly good.

III. TEST DESCRIPTION

The description of the overall CFDTIS test program was presented in the test plan (Reference 3) and test specifications (Reference 2). The first series of tests did not exactly follow the schedule as given in the test specifications because of an immediate need for bootstrap information to aid in NRX/EST test preparations.

A. TEST OBJECTIVES

General test objectives are described in RN-S-0148. Specific objectives of the first series of tests are given below.

The objectives of tests 001 and 002 were to:

1. Obtain component chilldown information under dry and wet pump start conditions.
2. Checkout the CFDTIS system and test facility.

The objectives of tests 003 through 006 were to:

1. Obtain system bootstrap information at various tank pressures under wet pump start conditions.
2. Obtain component chilldown and component performance information.

B. DESCRIPTION OF TESTS IN SERIES 1

Test conditions for the above tests are listed in Table I. The first two runs of the tests were chilldown runs. Run 1 was a dry pump run and run 002 a wet pump run. For a wet pump start, LH_2 is flowed through the inlet line and pump and vented out just downstream of the pump. In wet pump starts, the chilldown

of the pump was considered complete when the pump discharge temperature stabilized at a minimum. This minimum temperature for all the wet pump runs was about 40°R, so that even for the case of the lowest tank pressure used, 50 psia tank pressure (run 005), the state of the fluid at the pump discharge is essentially liquid at the completion of pump chilldown.

For the first series of tests, the ejectors were operating for the purpose of preventing oxygen from backing up in the exhaust ducts.

On bootstrap runs, the test start was initiated through the sequencer which closed the pump discharge bleed valve and opened the pump discharge valve. At the same time, the opening of the turbine power control valve was initiated. The turbine power control valve was ramped to 80° in one second on test number 003 and to 83° in one second on tests 4, 5, and 6.

Further information concerning test description and test procedures are given in Reference 6.

IV. TEST DATA

In this first series of tests, only those basic measurements required for information on the overall systems performance were made. Detailed data on flow distribution, component chilldown, etc., are not available. Redundant measurements were kept to a minimum.

A summary of the data at maximum turbine speed is presented in Table II. Figures 1 through 3 shows important system parameters as a function of time for the various runs.

Figure 1 shows test data for the nozzle chamber fluid temperatures as a function of time. These temperatures give good indications of the rates of system chilldown. It is seen that the rate of chilldown is highly dependent on the tank pressure. In runs 001 and 002, where the tank pressure was 70 psia, the turbine power control valve was closed and the system chilled faster than in run 005, where the tank pressure was 50 psia and the turbine power control valve opened. Thus the use of the turbopump did not compensate for the decrease in tank pressure.

The nozzle chamber pressures shown in Figure 2 also show the significant influence of the tank pressure. As tank pressure is increased, the peak pressure becomes higher and is reached in a shorter period of time.

Measured pump inlet flow rates as shown in Figure 3 indicate increased duration of flow oscillations as the tank pressure is decreased. The maximum flow rate for the 90 psia tank pressure run, run 004, more than double that of the 50 psia tank pressure test, run 005, illustrating the influence of tank pressure in this region.

A. MEASUREMENT SYSTEM ACCURACY

Prior to the actual running of the tests, a study was made to determine the required measurement system accuracies. The required accuracy of each

of each measurement was determined, as well as identifying those parameters whose measurement accuracies would greatly affect the evaluation of the system performance.

The standard deviation of a calculated parameter $U = f(x_1, x_2, \dots, x_n)$ where the x_i 's are measured parameters, is determined from the standard deviation expression,

$$S_u = \left[\sum_{i=1}^n \left(\frac{\partial u}{\partial x_i} \right)^2 S_{x_i}^2 \right]^{1/2}$$

where S_{x_i} is the required system standard deviation of the measurements of the parameter x_i . It is assumed that the measured parameters are statistically independent.

Accuracies corresponding to the assigned range were substituted into the above error expression to yield a standard deviation for the calculated quantity. If the absolute value of this quantity was greater than five percent, greater measurement accuracy was specified.

Critical measurements were identified through the magnitude of the term $\left(\frac{\partial u}{\partial x_i} \right)$ in the error equation.

B. INSTRUMENTATION MALFUNCTIONS

A list of those instruments which malfunctioned is given in Table III. In most cases, there were alternate measurements so that information was not completely lost because of the malfunctions. In run 004, the pump discharge pressure measurement was not obtained, thus requiring an estimate based on the nozzle inlet manifold pressure and on pressure drop in the pump discharge line as established in other runs. In run 001, the Consolidated Systems Corporation (CSC) data system was not turned on until the test has progressed about 15 seconds

Therefore there is some uncertainty in many of the solid temperatures and in the reflector outlet and core inlet fluid temperatures, all of which were recorded on the CSC system.

C. QUALITY OF DATA

In general, pressure data were quite good from the standpoint of consistency. Certain temperature measurements for the same part of the system differed greatly for any given run. These temperature differences may not be caused by measurement errors and will be discussed in subsequent discussions. Meanwhile they will be identified here. Since there were generally only two measurements at any location, no statistical treatment of the data was performed.

Of the two fluid temperature measurements taken in the nozzle tube outlet plenum, one was as much as 80°R lower than the other. This difference occurs in all the tests. Figure 4 shows a typical temperature trace for this location. This discrepancy will be discussed in Section V,C, below.

Reflector outlet plenum temperature measurements were erratic. Of the four thermocouples, only two showed reasonable values. Even these two measurements differed greatly and one of them displayed an oscillation which may have resulted from interference. Typical reflector outlet plenum fluid temperature measurements are shown in Figure 5. THRO-4 does not appear to be valid since it is reading about 40°R too low at FS-1.

There were three nozzle chamber temperature measurements. The one located near the hot bleed port (THNC-2) was up to 60°R lower than the other two, as shown in Figure 6. This difference occurred even on the run where the diluent valve was closed (run 001) so that there is no possibility of diluent flow going into the chamber.

There were two engine inlet flow rate measurements made at the propellant feed line using two flow meters which in general agreed with each other. Other flow rates, such as at the turbine inlet, were calculated based on

pressure and temperature measurements. The turbine flow rate for run 005, where the tank pressure was lower than in other runs, is less accurate than in other runs because the pressure drop measured across the venturi was of the order 1.0 psid whereas the instrument was ranged for 20 psid.

There is some uncertainty concerning the calculated nozzle flow rate as compared to the measured inlet flow, as shown in Figure 7. Using the nozzle chamber temperature and pressure measurements, nozzle chamber flow rates were calculated, assuming choked conditions at the throat. Figure 7 shows measured and calculated flow rates for run 003. The calculated total flow,

$$\left(= \int_{FS-1}^{FS-2} \dot{w} dt \right)$$

based on THNC-3 is approximately 60 pounds, or about 13.5%, less than the measured total flow, with the calculated flow rate lower than the measured throughout the run. If the lower chamber temperature measurement (THNC-2) is used in calculating the flow, the difference in total amount of fluid flow is approximately 30 pounds, with the calculated flow rate lower than the measured during the first two thirds of run duration. Assuming that liquid exists from the pump discharge through the reflector inlet plenum, which has a total volume of approximately 5.5 ft³, the mass storage in the system is approximately 25 pounds. Thus, correcting for mass storage, there is a 35 pound (~ 8%) discrepancy between the measured and calculated total flow using THNC-3, but only a 5 pound difference (~ 1%) when using THNC-2. Therefore, it appears that mass storage effects could possibly be used to reasonably explain the flow values derived using THNC-2, but not when using THNC-3. A combination of mass storage effects and measurement uncertainties could result in the discrepancy.

It is expected that additional alternate measurements in future tests, particularly measurements of temperature will yield more information on causes of many of the measurement discrepancies.

V. DATA INTERPRETATION

Interpretation of data requires the explanation of and extraction of information concerning the behavior of a process or phenomenon as indicated by the data. In the following sections, certain salient features displayed by the data from the first series of CFDS tests are examined in light of test objectives in an attempt to provide a basis for improving existing prediction capabilities.

A. FULFILLMENT OF TEST OBJECTIVES

The basic objectives of the first series of tests as outlined in Section III,A, were accomplished. Chillydown information was obtained in the first two tests and preliminary information on bootstrap startup under various tank pressures was supplied by the four bootstrap tests.

1 System Chillydown

Both chillydown runs were with 70 psia tank pressure and a 14 psia nozzle and turbine back pressure. The first run was a dry pump start with the pump inlet line and pump starting at ambient temperatures while in the second run the line and pump were chilled down prior to opening the pump discharge valve. As a result of the added energy from the pump inlet line and pump housing, the fluid density decreased, causing greater pressure drop in the system and lower flow rates, as shown in Figure 8.

It had not been anticipated that the pump inlet line and turbo-pump assembly would supply enough energy to alter the system's performance to the degree obtained. However upon closer examination of the pump inlet line, it appears that the line can contribute a significant amount of heat, particularly if there is conduction of heat from the support structure to the line. Inspection of the wall temperature measurements (Table 4) for the inlet line reveals that it is highly probable that such heat conduction took place. Thermocouples TPIL-1 and TPIL-4 were located at the sections where the support structure webbing:

attach onto the line (Figure 9). The low rate of cooling, as indicated by the relatively high temperatures existing at FS-2, is probably a result of the increased wall thickness (0.94 in.) at these sections and heat conduction from the support structure. The other two thermocouples, TPIL-2 and TPIL-3, were located in the thin sections (0.25-in. thick) of the line. Measurements from TPIL-2 were all invalid because of an instrument malfunction except for run 004. But even in that case, it is doubtful whether the measurement is valid, since at time 0, it indicated a temperature 25°F below that indicated by the other thermocouples. According to the measurements on TPIL-3, the thin sections of the inlet line cooled at a rate similar to that of the turbopump housing and pump discharge line.

Because of the limited inlet line wall temperature data available, it may not be valid to separate the measurements and draw inferences therefrom. Estimates of heat transfer coefficients based on the average of all four measurements (Table VI) indicate the possible existence of film boiling phenomenon in the inlet line. A special discussion of this heat transfer condition is included in Reference 14. Additional wall temperature measurements will help to clarify the situation.

Chiltdown data from the CFDIS were used to check the analytical model used in the 364 program (Reference 7). Although the basic engine model has been checked to some extent with data from NRX-A tests (References 4 and 5), there has not been an opportunity to check a complete chiltdown run. In the present instance, the wet pump start run will be used because of the uncertainty concerning the actual behavior of the pump inlet line chiltdown as discussed above. The inlet line is being modeled separately to determine the best way to represent it.

Calculations of the system parameters are shown in Figures 10 and 11. Comparison of the calculations with measurements indicate that there is fairly good temperature agreement except at the nozzle tube outlet plenum and in all measurements near the end of the run. Nozzle tube outlet plenum temperature

as calculated is low, but because of the difference between the two measurements, the calculated temperature still falls within the 95% confidence band of the measurements. Therefore, no attempt was made to improve the agreement until more data become available from future tests. Measurements toward the end of the run are questionable, particularly that of the reflector outlet, because it is not physically possible for the chamber reflector outlet and core temperatures to cross in the manner indicated by measurements.

Pressure calculations shown in Figure 11 in general did not show good agreement with measurements, although the overall pressure drop as calculated was fairly close to measurements as shown in Figure 12. The pressure drop across the nozzle coolant tubes was too low, which may be a result of errors in temperature calculations. Until more definite conclusions can be reached concerning temperature measurements and calculations, no adjustments of pressure losses will be made. The reluctance to make changes in the analytical model is due in part to the uncertainties in the measurements and also to the fact that earlier checks with NRX-A test data showed good agreement between calculations and measurements. While it is true that conditions in the NRX-A tests were not the same as in the CFDS, the earlier checks nevertheless provide good basis for confidence in the analytical description of the engine system. The discrepancies encountered in the present comparison point out areas where further investigation is required. Such investigation requires additional measurements more fully to characterize the temperatures in question, and it is expected that these measurements will be provided by the next series of tests.

2. Bootstrap Startup

In all of the bootstrap runs, startup was successful; that is, the engine system had sufficient thermal energy to initiate and maintain operation of the turbopump to the extent that addition of reactor power will sustain the operation. Data pertinent to the bootstrap operations are shown in Figures 13 and 14. Figure 11 shows the turbine speed as a function of time. The effect of the tank pressure is most pronounced between 50 and 60 psia. The speed nearly doubled in going from 50 to 60 psia tank pressure. Note that in all cases the turbine started to turn within one second after FS-1.

Pump pressure rise is shown in Figure 14, which indicates that the change in tank pressure from 50 to 60 psia produces the greatest change in pressure rise. Also for high tank pressures, a pressure rise across the pump occurs sooner.

The effect of tank pressure on various performance parameters are shown in Figure 15. In the range of tank pressures tested, the increase from 50 psia to 60 psia tank pressure showed the greatest influence on the performance, while further increases produce progressively less effect. Thus it appears that the region of optimization with respect to tank pressures is in the 50 - 70 psia range for a 14 psia back pressure. Data for other back pressures will be obtained in future tests.

No comparisons of calculated bootstrap performance with the measurements were made because the turbopump performance characteristics which were available prior to the CFDTs tests were applicable only in high speed regions. In the range of speeds encountered in the CFDTs, there was too much uncertainty in the available turbopump data to allow meaningful comparisons.

B. EVALUATION OF TURBOPUMP ASSEMBLY PERFORMANCE

The performance of the CFDTs is highly dependent upon the performance of the turbopump. The ability to predict CFDTs performance with certainty requires sufficient information on turbopump operating characteristics, particularly in the low speed regime. Existing low speed turbopump data is sparse and uncertain. It was felt that the CFDTs tests would provide additional information to aid in describing turbopump performance, at the same time being mindful of the fact that the CFDTs was not intended to be a turbopump test. However, the CFDTs does offer the advantage of having instrumentation ranged for low speed operation, which is necessary in view of inherent experimental difficulties with low speed tests.

1. Turbine Performance

Because instrumentation ports were not provided in the turbine exhaust collector, no turbine exhaust measurements were taken during the first series of tests. Therefore, no information was obtained.

Information concerning the turbine impedance to flow can be obtained from the measurements of turbine flow rate, turbine inlet pressure and temperature, and turbine back pressure. In previous turbine performance calculations (Reference 8), the flow parameter when the turbine is unchoked was approximated as a linear function of inlet to back pressure ratio. The turbine was assumed choked when the pressure ratio reached or exceeded a value of two. That is, the turbine flow was taken as

$$w_t = \frac{\alpha P_{t_i}}{\sqrt{T_{t_i}}}$$

where

w_t	=	turbine flow rate
P_{t_i}	=	turbine inlet total pressure
T_{t_i}	=	turbine inlet total temperature
α	=	flow parameter

The coefficient was given the form,

$$\alpha = \begin{cases} 4.75 \left(P_t / P_{atm}^{-1} \right) & \text{for } 1 \leq P_{t_i} / P_{atm} \leq 2 \\ 4.75 & \text{for } P_{t_i} / P_{atm} > 2 \end{cases}$$

where, P_{atm} is the turbine back pressure

From turbine inlet and flow measurements of CFD'S bootstrap runs, values of flow parameter were calculated as a function of the pressure ratio, P_{t_i} / P_{atm} . Results are shown in Figure 16. It is seen that the linear approximation

for unchoked flow conditions is a poor one. A curve was visually fitted to the points resulting in the following function:

$$\alpha = 0.475 \left[1 - \text{EXP} \left(-3.7 \left(P_{t_i} / P_{\text{atm}} - 1 \right) \right) \right]$$

The above exponential function should be used instead of the linear approximation.

It should be pointed out that the above results are applicable only to the turbine used in this first series of tests. Since a new turbine inlet manifold is used in the second series, the numerical values of the flow parameter will be changed, although the form of the dependence on pressure ratio should be similar to that obtained here.

2. Pump Performance

Evaluating turbopump performance requires knowing how much energy was extracted from the working fluid by the turbine and how much of this energy was transferred to the fluid being pumped. Since there is no direct way to measure energy content of the fluid, it must be deduced from measurements of state variables. In the low speed operating region, not only is the energy content of the liquid hydrogen a strong function of temperature, but the changes in energy content are very small. As a result, the energy content determination depends upon the use of a small difference of two measured temperatures which makes the accuracy of the measurements critical.

The experimental difficulties discussed above clearly manifested themselves in the pump data. The difference between pump inlet and pump discharge temperature is of the order of one to two degrees Rankine. Thus, even an error of 0.3°R (systems accuracy of these temperature measurements) results in as much as 30% uncertainty in the temperature difference. Because the pressure difference

across the pump, which involves a lower degree of uncertainty than in the temperature difference, is also used in the calculation of pump efficiency, the 30% uncertainty in temperature difference will result in an uncertainty in the efficiency that is less than 30%. However, the uncertainty is high enough so that attempts to correlate the pump efficiency as calculated from pump inlet and outlet measurements did not yield any meaningful results.

Since the CFDTS is not an isolated turbopump test, it can provide additional data from which pump performance characteristics might be extracted. That is, one can use the performance of the entire system to determine what pump characteristics must exist to yield that performance. However, one must be sure that other components of the system are well described and understood to avoid erroneously attributing effects of these components to the pump. This approach has been taken for CFDTS and preliminary results are promising.

The Transient Engine Simulation Program (Reference 9) was used to calculate pump efficiencies required to yield the measured flow, pressure, temperature, and turbine speed performances. Results of the calculations for each of the bootstrap runs were then combined to arrive at a set of curves for pump efficiency as a function of speed and flow rate. Results are shown in Figure 17. There were some scatter in the calculated points and the curves were fitted through the average values. Data from component tests agree well with the CFDTS data at high speeds.

The turbine data used in the above efficiency calculations were the result of extrapolating from high speed turbine test data to low speed ranges. Because of the extrapolation there are some uncertainties involved, but there are two points which justify confidence in the extrapolated turbine data. First, the values of the pump efficiencies obtained were reasonable and consistent, as will be demonstrated below. The second point is that in the very low speed range (say below 1000 RPM), the acceleration of the turbopump as calculated agreed reasonably well with measurements. In this low speed region, it may be

correctly assumed that the pump is doing very little work on the fluid so that the pump efficiency has little influence on the calculations. The turbine efficiency becomes the dominant parameter and agreement between calculations and measurements in this region attests to the validity of the values of turbine efficiency used.

It might also be pointed out that even if there is a large uncertainty in the turbine data, the pump efficiencies obtained are still applicable if the same turbine is used, provided there is internal consistency in the pump efficiencies. Consistency here implies that for a given speed and flow rate, the pump efficiency does not vary for different runs.

That the calculated pump efficiencies are consistent is shown by the good agreement between calculated performance and measurements as shown in Figures 18 and 19. Since the pump efficiencies used for these calculations were from composite data as represented by the efficiency curves, good agreement between calculations and measurements for any particular run can only result from consistent pump efficiency data.

Attempts were made to correlate pressure rise data. A correlation was obtained with the parameters $\frac{\Delta P}{\dot{w}^2}$ vs $N/\dot{w}$, where ΔP is the pressure rise, psi, $\dot{w}$ is the flow rate, lbm/sec, and N is the rotational speed, rpm. Using data from run 003 and for which $N/\dot{w} \geq 120$ rpm-sec/lbm, the correlation obtained was (Figure 20):

$$\frac{\Delta P}{\dot{w}} = -0.047 + 1.47 \times 10^{-4} N/\dot{w} + 1.97 \times 10^{-6} (N/\dot{w})^2$$

Note that for this case, the high speed data from component testing compares well with the CFDS data. When data from run 005 was combined with that from run 003, and also including data for which $N/\dot{w} < 120$ rpm-sec/lbm, the correlation becomes less accurate, as shown in Figure 21. The correlation here was:

$$\frac{\Delta P}{\dot{w}} = -0.094 + 2.60 \times 10^{-4} N/\dot{w} + 2.18 \times 10^{-6} (N/\dot{w})^2$$

The large scatter in the low speed (or high flow rate) region, $N/\dot{w} < 150 \text{ rpm-sec/lbm}$ may be a result of neglecting suction heat effects. These effects were neglected in order to arrive at a more usable correlation, particularly for analog computations.

Using the parameters $\Delta H/N^2$ vs Q/N , where ΔH is the head rise, ft, and Q is the volumetric flow rate, gal/min, a correlation using data from all the bootstrap runs was obtained as:

$$\frac{\Delta H}{N^2} = 0.752 + 0.513 \frac{Q}{N} - 0.687 \left(\frac{Q}{N} \right)^2 - 0.251 \times 10^{-4} S$$

The suction specific speed, S_s , is included in the correlation to better fit the low speed data.

C. TEMPERATURE ASYMMETRY

In this first series of tests, detailed data on flow distribution were not sought. Measurements of fluid properties in the various plena of the system were kept to a minimum, in many cases only two measurements of temperature were made. Therefore it is not expected that flow asymmetries based on temperature measurements can be ascertained, and in cases of suspected asymmetry, definite conclusions concerning the causes of asymmetry cannot be reached until additional data become available. However, even with the limited data available some indications of asymmetry occurred, as discussed below.

In the nozzle tube outlet plenum, as pointed out in Section IV-C, a difference of up to 80°R was noted in the two temperature measurements (Figure 4). Since the two measurements show good agreement in the beginning of the test and during the two phase and liquid flow portions of the test, there is reason to believe that the resistance thermometers are reading correctly. The locations of the two measurements, THWIC-2 and THWIC-3, are at angles of 165° and 75° respectively. The higher reading was from THWIC-2, which was located almost directly in line with the hot bleed port ($\theta = 168.5^\circ$). A higher temperature

reading at this location could result from a lower flow rate in the coolant tube around the bleed port caused by the added impedance of the bleed port. Preliminary studies of flow distribution in the nozzle coolant tubes (Reference 10), using water, have indicated at least a 9% lower flow rate in the tubes going through the bleed port manifold. Such a distribution of flow can contribute to temperature asymmetries in both the H_2 and the tube walls.

Two phase flow phenomena can also cause flow asymmetry as experienced in NRX-A tests (Reference 4), but in the present case the outlet plenum temperatures in question showed differences even before a two phase condition existed anywhere downstream of the nozzle inlet line.

Fluid temperature measurements at the reflector outlet showed considerable variations. However, there is some doubt as to the accuracy of these measurements. At most, only three thermocouples (THRO-2, -7, and -9) showed reasonable initial values of temperature ($520^\circ R$), with THRO-9 showing oscillations. In the latter half of each run, the measurements indicated a higher temperature in the reflector outlet than in the core inlet and nozzle chamber. Because of the questionable accuracies of these measurements, no meaningful information concerning asymmetry can be obtained from them.

Some information on temperature asymmetry can be inferred from outer reflector material temperature measurements. Figures 22 and 23 show outer reflector material temperatures for chilldown run 002 and bootstrap run 003. There are two measurements at each of the indicated stations. At Stations 2 and 34, the two measurements are located at 45° and 255° respectively. At Station 18 and 50, they are located at 165° and 255° , respectively. In all cases the lower temperature measurement is at 255° . It is seen that as the reflector outlet is approached, the difference in temperature between the two measurements at a station is decreased, and in the chilldown case, this difference disappears at Station 2 and 18. However, it seems that the difference in temperatures disappear only when the material is not chilled. In run 003 (Figure 23), as the reflector chills, even Stations 2 and 18 show temperature differences increasing with time.

This increase in the difference with time is expected during the chilldown since the location with the lower temperature would have a lower resistance to flow and consequently the flow at that location would be higher than in the higher temperature location, resulting in an even higher rate of chilldown. Eventually the temperature differences must disappear as the system is completely chilled, but the tests are generally terminated before such a condition is reached. Thus, it appears that the temperature asymmetry will probably penetrate farther downstream in the reflector as higher flow rates are encountered.

The shield and core inlet temperature measurements were all located at approximately the same angle ($\sim 120^\circ$) so that asymmetry information is not available. These measurements show a temperature variation in the radial direction, with temperatures increasing towards the center of the engine. This radial distribution probably results from peripheral cooling by fluid in the reflector and shield first pass.

The cause of the temperature differences in the nozzle chamber is not apparent at this time. It is hoped that additional measurements in the next series of tests will help clarify the situation.

D. HEAT TRANSFER CORRELATIONS

Accurate analytical description of the NERVA engine systems depends heavily on reliable convective heat transfer correlations. While studies have been made of convective heat transfer in hydrogen systems (See Reference 4), in many cases these are studies of idealized laboratory systems. Hence, any correlations derived from these studies can only be evaluated using data from actual engine system tests. The NRX-A series of tests has provided information on gas phase heat transfer. However, there has been little data on two phase and liquid phase heat transfer coefficients. Data from this first series of tests has been used to evaluate the heat transfer correlations currently used in calculations.

1. Use of Engine System Simulator Program (364)

In order to evaluate a particular heat transfer correlation, the heat transfer effects must be isolated from other influences of the engine system. Because of the complexity of the total engine system, it would be difficult to discriminate among various heat transfer regimes. Therefore, it would be best to examine only the various components of the system to evaluate heat transfer calculations. The 364 Program is a generalized heat transfer and fluid flow program which allows simulation of any component of the engine system independent of the remainder of the system. This feature of the program will be used here to examine two phase heat transfer in the pump discharge line and gas phase heat transfer in the reflector.

2. Two Phase Heat Transfer Correlations

The correlation currently used is that of Hendricks, Graham, Hsu, and Medeiros (Reference 11), where the ratio of the actual heat transfer coefficient, h_{act} , to the theoretical, h_{th} , is expressed in terms of the Martinelli parameter X . This correlation is:

$$\frac{h_{th}}{h_{act}} = 0.611 + 1.93 X$$

and

$$X = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\mu_{sl}}{\mu_{sv}} \right)^{0.1} \left(\frac{v_{sl}}{v_{sv}} \right)^{0.5}$$

It is seen that the Martinelli parameter is dependent upon the quality (x) of the fluid and viscosity (μ), and specific volume (v) of the saturated liquid and saturated vapor. The theoretical coefficient is formulated from the gas phase correlations. This theoretical value is*

$$h_{th} = \frac{0.023 k}{De} Re^{0.8} Pr^{0.4}$$

*As used in this correlation, Re is defined by $Re = \frac{\rho_m U_{av} D_e}{\mu_f}$

Where ρ_m is the mean density of the film, U_{av} is the average velocity of the mixture of liquid and gaseous phase, μ_f is viscosity at gaseous film conditions.

Film temperatures are used to evaluate thermal conductivity, k , Reynold's number, Re , and Prandtl Number, Pr . The hydraulic diameter is denoted by De .

Calculated pump discharge line wall temperatures are shown in Figure 24 along with the measured values for run 002. The location of TPD-1 is near the pump outlet where the wall is chilled down along with the pump prior to start of the run, which explains the constant 50°R measurement at this location. TPD-2 was located about 100 in. from the pump discharge and TPD-4 was located at the 90° bend near the nozzle inlet manifold. The fact that TPD-4 reads lower temperatures than TPD-2 indicates a higher rate of heat transfer at the bend caused by impingement of fluid. Calculated temperatures are for a location near TPD-2.

The calculated temperature shows a sharp break in the slope after about 5 seconds. This break results from the change from two phase to liquid correlation, indicating that the calculated liquid heat transfer coefficient is higher than that for two phase. The liquid correlation used is

$$h = \frac{0.023 k}{De} Re^{0.8} Pr^{0.4}$$

This liquid correlation is admittedly an over-simplified one in that it does not correct for the effects of the wall to bulk temperature ratio. Since the correlation was already in the computer program, it was not changed for this initial analysis. In future analyses, more appropriate correlations will be used.

It is evident from the data shown in Figure 24 that there is a time delay of about 2.5 seconds after FS-1 before the outer wall begins to chill down. In the calculations, this time delay did not show up because the nodal network in the model representation was not fine enough to be entirely representative of a continuous heat conduction path. Therefore the calculated temperatures shown in Figure 24 have been shifted to the right about 2.5 seconds to give a more meaningful comparison. Since, in evaluating heat transfer, it is the rate of

change of temperature that is important, the shift in the calculated temperature curves should not alter the conclusions drawn.

That the heat transfer coefficient is proportional to the slope of the temperature-time curve can be demonstrated through an energy balance for the pipe wall. The energy balance results in the following expression for the convective heat transfer coefficient.

$$h = - \frac{W_p C_p}{A} \frac{d \ln \theta}{dt}$$

where

W_p = weight of line

C_p = heat capacity of wall material

A = heat transfer surface area

$\theta = \frac{T - T_b}{T_o - T_b}$ temperature ratio

T = wall temperature

T_b = fluid temperature

T_o = initial wall temperature

An estimate of the heat transfer coefficient based on the measured wall temperature was made using the above equation and the following approximation:

- a. The outer wall temperature as measured is used as wall temperature, i.e., assumption of no radial temperature gradient
- b. Assume no axial temperature gradient in the portion of pipe that the measurement applies.
- c. Constant fluid temperature.
- d. Constant wall properties.

Measurements from TPDL-2 was used along with the following values:

$$\begin{aligned}W_p &= 79 \text{ lb} \\C_p &= 0.11 \frac{\text{BTU}}{\text{lb-}^\circ\text{R}} \\A &= 2666 \text{ in}^2 \\T_b &= 45^\circ\text{R} \\T_o &= 520^\circ\text{R}\end{aligned}$$

A plot of the data for the log of temperature ratio, $\ln \theta$, versus time (Figure 25) shows that the slope is constant, which means that the heat transfer coefficient was approximately constant throughout the run. The slope obtained from the plot is -0.056, which results in a heat transfer coefficient of $1.8 \times 10^{-4} \text{ BTU/in}^2\text{-sec-}^\circ\text{R}$. A value of about $0.93 \times 10^{-4} \text{ BTU/in}^2\text{-sec-}^\circ\text{R}$ is estimated for the two phase heat transfer coefficient using the Martinelli parameter correlation (dashed line on Figure 24), which is about half that obtained from the data. By fitting a curve to the high data of Hendricks, et. al, the following correlation was obtained and used to calculate a second temperature curve (dotted line on Figure 24):

$$\frac{h_{th}}{h_{act}} = 0.611 + 0.59 X$$

The two phase heat transfer coefficient estimated for this correlation is about $1.95 \times 10^{-4} \text{ BTU/in}^2\text{-sec-}^\circ\text{R}$, which shows a better agreement with that obtained from the measurements of wall temperature.

In the liquid region of the calculations, the heat transfer coefficient is of the order $3.5 \times 10^{-4} \text{ BTU/in}^2\text{-sec-}^\circ\text{R}$, or about twice the magnitude of that estimated from the measurements. It is expected that with a correction for the wall-to-bulk temperature ratio, the agreement will be much improved.

Thus from the above discussion, it may be concluded that the Hendricks, et. al, two phase heat transfer correlation must be modified to yield higher heat transfer.

3. Reflector Gas Phase Heat Transfer Correlations

To check the gas phase heat transfer correlation currently used in the calculations, use will be made of available detailed models of the reflector. An eight channel representation of the reflector has been checked out with NRX-A2 data (Reference 4), and a four channel model has also been prepared and is being used in NRX-A3 data analysis.

The gas phase heat transfer correlation currently used is that of McCarthy and Wolf (Reference 12).

$$Nu_b = 0.025 Re_b^{0.8} Pr_b^{0.4} (T_w/T_b)^{-0.55}$$

where,

Nu_b = Nusselt Number based on bulk fluid temperature

Re_b = Reynolds Number based on bulk fluid temperature

Pr_b = Prandtl Number based on bulk fluid temperature

T_w = Wall temperature

T_b = Bulk fluid temperature

Calculations of reflector outlet fluid temperature were made with both 8 and 4 channel models. Results shown in Figure 26 indicate good agreement between the two models and also between calculations and measurements. There is deviation from the measurement toward the end of the run, but as pointed out earlier, the measurements in this region are questionable. Therefore, the fairly good agreement obtained in this comparison and NRX-A2 comparisons provide reasons to have confidence in the gas phase heat transfer correlation currently in use.

VI. ANALYSIS OF FUTURE DATA

One of the primary objectives of the second series of tests is to establish startup procedures for the X-E and NRX/EST engine systems. These procedures will be determined experimentally by running the tests under conditions duplicating those planned for the X-E and NRX/EST tests. As these tests will be run at varied conditions of tank pressure, back pressure, and TPCV programs, information obtained from the tests will be used to define startup operating regions.

Information obtained from the first series of tests pointed out certain areas where further studies are needed in order to achieve startup prediction capability. Of prime importance is the characterization of turbopump performance. Because of the inherent experimental difficulties on obtaining valid data for low speed turbopump operations, indirect methods must be used to obtain turbopump characteristics. As discussed earlier, one method is to use the entire system behavior to determine the turbopump performance required to yield that behavior. Additional fluid temperature measurements to be made in the next series of tests will allow better characterization of the flow distribution in the system. The additional measurements will also supply a larger sample size upon which the validity of the measurements can be better established. With the validity established, the data can then be used to upgrade analytical models of various parts of the system which will yield more confidence in the pump characteristics obtained from the system's behavior.

Turbine exhaust measurements will be made in the next series of tests. But it is anticipated that difficulties similar to those involved in the pump efficiency calculations will be encountered in trying to establish turbine efficiency from the turbine measurements.

The use of isolated components in evaluating and investigating certain phenomena has been demonstrated to be feasible. This method of data analysis will be used in future analysis efforts.

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TABLE I

CFDTS TEST CONDITIONS

Test Number Test Data	Measure- ment Symbol	Unit of Measure- ment	001	002	003	004	005	006
			April 15	April 16	April 17	April 19	April 20	April 20
<u>TEST CONDITIONS</u>								
Tank pressure	PFT-HII	psia	68.0	68.7	69.7	89.8	48.2	57.7
Nozzle back-pressure	PNED	psia	13.8	13.8	13.8	13.9	13.8	13.9
Turbine back-pressure	PTEL	psia	14.2	Malfunc- tion	14.2	14.3	14.3	14.3
8 Diluent valve position			closed	open	open	open	open	open
Type of pump start			dry	wet	wet	wet	wet	wet
Chamber temperature at FS-2	THNC	°R	140	140	135	144	133	148
TPCV program			closed	closed	(opened full at test start)			

TABLE II

SYSTEM OPERATING PARAMETERS AT MAXIMUM TURBOPUMP SPEED

No.	Station	Run 003 5280 RPM @ 16 sec			Run 004 5935 RPM @ 9.7 sec		
		Flow lb/sec	Pressure psia	Temp °R	Flow lb/sec	Pressure psia	Temp °R
1	Tank Outlet		70.8	38.2		91.4	39.2
2	Pump Inlet	20.1	68.5	38.4	24.0	87.9	39.1
3	Pump Outlet		123.5	39.6		156	40.1
4	Nozzle Manifold Inlet			46			48
5	Nozzle Outlet		112.2	53		141.4	58
6	Reflector Outlet		107.6	243		134.7	339
7	Shield Outlet		95.3	321		118.1	383
8	Core Outlet		44.2	443		55.2	466
9	Nozzle Chamber		44.2	44.3		55.2	466
10	Diluent Bleed Inlet	.49	87.7		.58	108.9	
11	Diluent Bleed Outlet		74.6	401		930	427
12	Hot Bleed		44.2	443		55.2	466
13	Mixing Ring Inlet		42.1	421		51.2	452
15	TPCV Inlet		41.3	427		50.3	453
16	Turbine Inlet	.97	40.2	429	1.12	49.0	453
17	Turbine Outlet						
18	Turbine Exhaust Nozzle						

No.	Station	Run 005 2180 RPM @ 38.5 sec			Run 006 4222 RPM @ 28.1 sec		
		Flow lb/sec	Pressure psia	Temp °R	Flow lb/sec	Pressure psia	Temp °R
1	Tank Outlet		48.9	38.9		58.4	39.1
2	Pump Inlet	8.7	49.2	39.1	15.1	57.6	39.2
3	Pump Outlet		58.2	39.9		92.4	40.2
4	Nozzle Manifold Inlet		55.0	45		89.9	45
5	Nozzle Outlet		48.9	45		82.7	49
6	Reflector Outlet		46.1	274		77.1	284
7	Shield Outlet		42.5	314		70.7	310
8	Core Outlet		19.8	419		32.4	400
9	Nozzle Chamber		19.8	419		32.4	400
10	Diluent Bleed Inlet	.21	39.1		.36	64.8	
11	Diluent Bleed Outlet		33.5	405		55.6	394
12	Hot Bleed		19.8	419		32.4	400
13	Mixing Ring Inlet		19.2	406		31.1	394
15	TPCV Inlet		19.1	423		30.5	413
16	Turbine Inlet	.29	18.8	427	.67	29.8	416
17	Turbine Outlet						
18	Turbine Exhaust Nozzle						

TABLE III

INSTRUMENTATION DISCREPANCIES - CFDTS TESTS NOS. 1 THROUGH 6

<u>Measurement No.</u>	<u>Parameter</u>	<u>Description</u>	<u>Run Nos.</u>	<u>Comments</u>
7	PPIL-3	Pump Inlet Line	1, 2	Data questionable, reads higher than PRTPL-1, should read lower.
10	PPD-1	Pump Discharge	4	Data invalid. Zero shift of 120 psi prior to test.
16	PNMI-1	Nozzle Manifold Inlet	4	Data invalid, transducer reads 20 psi high at test start and low during run.
21	PNM-4	Nozzle Manifold	4	Transducer malfunctioned, data invalid.
59	PCI-2	Core Inlet	4	Data questionable, reads 5 psi low at test start.
74	PDJ-1	Diluent Jacket	1	Data invalid, transducer inoperative.
92	PTEN-2	Turbine Exhaust Nozzle	1-6	Data invalid.
127	THPD-4	Pump Discharge	1	No data.
177	THCI-7	Core Inlet	1, 2	Data invalid.
187	TCM-5	Core Modules	1, 2	Data questionable, reads 400°R at end of run, inconsistent with surrounding measurements.
189	TCM-7	Core Modules	1-6	Data questionable, reads 400°R at end of run, inconsistent with surrounding measurements.
192	TIR-1	Inner Reflector	1	No data.
193	TIR-2	Inner Reflector	1	No data.
194	TIR-3	Inner Reflector	1	No data.

TABLE III (cont.)

<u>Measurement No.</u>	<u>Parameter</u>	<u>Description</u>	<u>Run Nos.</u>	<u>Comments</u>
195	TIR-4	Inner Reflector	1	No data.
201	TCD-1	Control Drum	2-6	TC inoperative - deleted.
202	TCD-2	Control Drum	2-6	TC inoperative - deleted.
204	TCD-4	Control Drum	2	No data.
206	TCD-6	Control Drum	1-6	TC inoperative - deleted.
216	TOR-4	Outer Reflector	1	No data.
232	THSISP-8	Shield Inlet Second Pass	3,4, 5,6	TC inoperative - deleted.
265	TNWI-1	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
266	TNWI-2	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
267	TNWI-3	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
268	TNWI-4	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
281	TNWI-5	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
282	TNWI-6	Nozzle Wall Inner	1-6	Data invalid, TC junction separated during first test.
299	TPDL-3	Pump Discharge Line	1,2,3	No data.

TABLE III (cont.)

<u>Measurement No.</u>	<u>Parameter</u>	<u>Description</u>	<u>Run Nos.</u>	<u>Comments</u>
306	TPIL-2	Pump Inlet Line	1-6	No data, calibration good. Cause indeterminate.
307	TPIL-3	Pump Inlet Line	1-2	Data questionable, ambient indication only, cause indeterminate.
308	TTPH-1	TPA Housing	2, 3	No reading.
310	TTPH-3	TPA Housing	1-6	Data questionable.
311	TPIL-4	Pump Inlet Line	1-6	Data invalid, RTZ overscale on pre-post calibrations. Cause indeterminate.
312	TNM-1	Nozzle Manifold	1, 2	Reading invalid.
402	PFTH 11	Propellant Fuel Tank	2, 5	Data questionable, reading 6 to 8 psi lower than other measurements.
421	PTEL-1	Turbine Exhaust Line	1-6	Data questionable.
422	PTEL-2	Turbine Exhaust Line	2	Data questionable.
436	THNED-1	Nozzle Exhaust Duct	1-6	Invalid data.

TABLE IV

MATERIAL TEMPERATURE DATA - CFDTS

Time	Turbopump Inlet Line				Turbopump Housing		Turbopump Discharge Line			
	<u>TPIL-1</u>	<u>TPIL-2</u>	<u>TPIL-3</u>	<u>TPIL-4</u>	<u>TTPH-1</u>	<u>TTPH-2</u>	<u>TPDL-1</u>	<u>TPDL-2</u>	<u>TPDL-3</u>	<u>TPDL-4</u>
(Run No. 1, FS 1 42 sec, FS-2 97 sec)										
0	513	377*	-	513	-	514	95	125	-	218
FS-1	513	349	-	-	-	514	-	-	-	-
FS-2	466	516	-	-	-	59	-	-	-	-
(Run No. 2, FS 1 178 sec, FS-2 203 sec)										
0	515	-	-	517	522	522	518	516	-	513
FS-1	364	-	-	319	81	54	49	516	-	514
FS-2	312	-	-	505	15	54	47	59	-	69
(Run No. 3, FS 1 167 sec, FS-2 196 sec)										
FS-1	358	-	-	363	-	53	-	514	-	515
FS-2	324	-	-	326	-	45	-	123	-	77
(Run No. 4, FS 1 163.6 sec, FS-2 186.4 sec)										
0	525	500*	521	530	530	531	526	528	528	528
FS-1	328	365	56	343	61	45	52	530	530	529
FS-2	209	345	45	514	6	46	42	88	121	77
(Run No. 5, FS-1 260.3 sec, FS-2 322.3 sec)										
0	521	-	520	-	522	520	523	527	527	518
FS-1	286	-	51	-	72	52	50	515	517	518
FS-2	241	-	56	-	69	50	52	54	72	64
(Run No. 6, FS-1 195.7 sec, FS-2 240.6 sec)										
0	508	-	506*	-	507	506	498	518	518	521
FS-1	316	-	49	-	72	55	48	522	518	521
FS-2	276	-	54	-	72	54	51	58	91	55

*Data Questionable

TABLE V

HEAT TRANSFER COEFFICIENT ESTIMATES FOR PUMP INLET LINE

<u>Run No.</u>	<u>Tank Pressure</u> psia	$\bar{h}_1 \times 10^4$ BTU/in ² sec-°R	$\bar{h}_2 \times 10^4$ BTU/in ² sec-°R	Film Boiling $h_f \times 10^4$ Ref. 0/3 BTU/in ² Sec-°R
2	70	0.21	0.35	0.31
3	70	0.21	0.37	0.31
4	90	0.25	0.33	0.45
5	50	0.22	0.28	1.13
6	60	0.23	0.30	0.61

NOTE:

$$h = \frac{W C_p}{A \Delta t} \ln \left[\frac{(T_{w1} - T_b)}{(T_{w2} - T_b)} \right]$$

h_1, h_2 are calculated for time 0 to FS-1 and FS-1 to FS-2 respectively.

T_{w1}, T_{w2} are measured wall temperatures at beginning and end of time interval.

T_b is average fluid temperature.

C_p is average heat capacity of wall.

w is weight of line.

A is heat transfer area.

Δt is the time interval.

CFDTS NOZZLE CHAMBER TEMPERATURES

Figure 2

Figure 3

Figure 4

Figure 5

Figure 6

Figure 7

RUN 001 - DRY PUMP START
RUN 002 - WET PUMP START
TANK PRESSURE = 70 PSIA
BACK PRESSURE = 14 PSIA

Figure 8

0 = TC/CA EXTERNALLY MOUNTED

SKETCH OF PUMP INLET LINE & STRUCTURE

CFDTS COOLANT FLUID TEMPERATURES RUN 002

CFDTS COOLANT FLUID PRESSURES AND WEIGHT FLOW RATE RUN 002

CODE:

—— MEASURED

--- CALCULATED

PRESSURE DROP ACROSS VARIOUS COMPONENTS
CFDTS RUN 002

Figure 12

Figure 13

PUMP PRESSURE RISE VS TIME CFDT'S BOOTSTRAP RUNS

_____ PUMP PRESSURE RISE
 CHAMBER PRESSURE
 - - - - - PUMP FLOW RATE
 - - - - - TURBINE SPEED
 - - - - - TIME TO PEAK

VALUES OF PERFORMANCE PARAMETERS AT MAXIMUM SPEED
 FOR VARIOUS TANK PRESSURES

TURBINE FLOW COEFFICIENT OBTAINED FROM CFDTS DATA

MK 3 - MOD 4 TURBOPUMP
PUMP EFFICIENCY VS ROTATIONAL SPEED/PUMP FLOW RATE

COMPARISON OF CALCULATED AND MEASURED PUMP SPEED

COMPARISON OF CALCULATED AND MEASURED PUMP FLOW

PUMP PRESSURE RISE CORRELATION CFDTs HIGH SPEED DATA

PUMP PRESSURE RISE CORRELATION CFDTs HIGH AND LOW SPEED DATA

Figure 22

OUTER REFLECTOR TEMPERATURES CFDTS RUN 003

Figure 23

Figure 24

$$h = \frac{W_p C_p}{A} \frac{d \ln \theta}{dt}$$

ESTIMATION OF HEAT TRANSFER COEFFICIENT

COMPARISON OF CALCULATED AND MEASURED
REFLECTOR OUTLET PLENUM FLUID TEMPERATURE RUN 003