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ANALOG SIMULATION OF A CLOSED LOOP BRAYTON CYCLE TEST FACILITY

by Hugh Klassen

Lewis Research Center

Cleveland, Ohio

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ABSTRACT

The performance of a Brayton cycle test loop, which contained an experimental turbocompressor, was simulated with an analog computer. The working fluid was argon. The results of actual and simulated test runs were compared. The variation of rotative speed with time during injection starts is shown as a function of injection rate, turbine inlet temperature, and loop volume. The operating characteristics of the compressor are shown during injection starts.

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SUMMARY

The performance of a Brayton cycle test loop, using argon as the working fluid, was simulated with an analog computer. The test loop turbomachinery consisted of a turbocompressor designed for a 10-kilowatt-shaft-output powerplant. The simulation was used to predict the performance of the test loop during start and shutdown transients. Comparison of the simulated injection starts with actual injection starts indicated that the real variations of loop operating variables with time agreed well with the computer predictions.

Closed loop injection was both analytically and experimentally a practical method of accelerating the turbocompressor rotor to self-sustaining speeds. For a given injection period, rotative speed increased with injection rate. For a fixed amount of inventory added to the loop, rotative speed first increased and then decreased as the injection rate increased. The turbocompressor could also be brought to design operating conditions from zero rotative speed without surging the compressor at any time.

Heat transfer between the turbine exit gas and the cooler inlet piping had an important effect on actual loop starting characteristics. Elimination of this heat transfer would greatly reduce the rotative speeds obtained during the injection period. Loop volume changes also caused large changes in the rotative speeds obtained during the injection period. Decreasing the volume upstream from the turbine increased rotative speed, while decreasing the volume downstream from the turbine decreased the rotative speed.

INTRODUCTION

The NASA Lewis Research Center is currently conducting an investigation to determine the potential of the Brayton cycle for space power generation. This investigation has included both the study of components and their integration into complete breadboard systems. One such system chosen for study was a two-shaft powerplant with a 10-kilowatt-shaft output and argon as the working fluid. Components were designed and built for this system under contract.

In conjunction with this program, a test loop was built to test both two-shaft and single-shaft systems. The test program using this loop includes both steady-state and transient operation. The transient operation testing includes the study of techniques for start and shutdown.

One proposed method for starting is to evacuate the loop and then inject the system inventory upstream of the heater. Hot gas from the heater passes through the turbine and accelerates the turbocompressor to self-sustaining speed. A check valve prevents back flow into the compressor discharge.

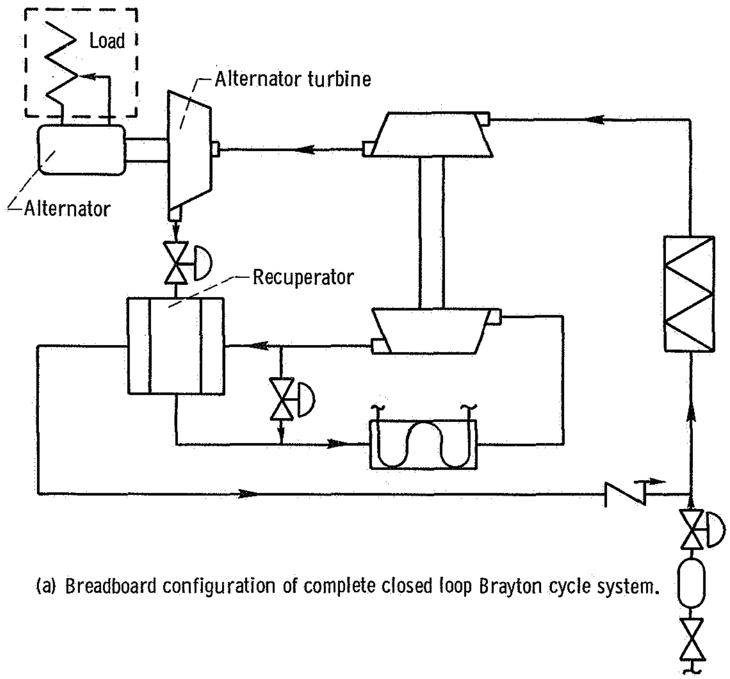
Injection starting may subject turbomachinery to high heat flux, high thrust load, rapid acceleration, and possible compressor surge. An analog simulation offers a means of studying system operating characteristics without hazard to equipment. Time and expense can be saved by experimentally testing only the most promising start techniques, and operation can be studied for conditions that cannot be attained in the test loop. The effect of design changes can be determined without continually making the changes, and results can be obtained that are applicable to space-type test loops and powerplants.

The objectives of this investigation were to make a study of the transient operating characteristics of the ground test facility by means of an analog simulation and to determine its validity by comparing the theoretical transients with those obtained experimentally. The simulation included facility flow conductances, heat transfer, volumes, bearing friction, and turbomachinery operating characteristics.

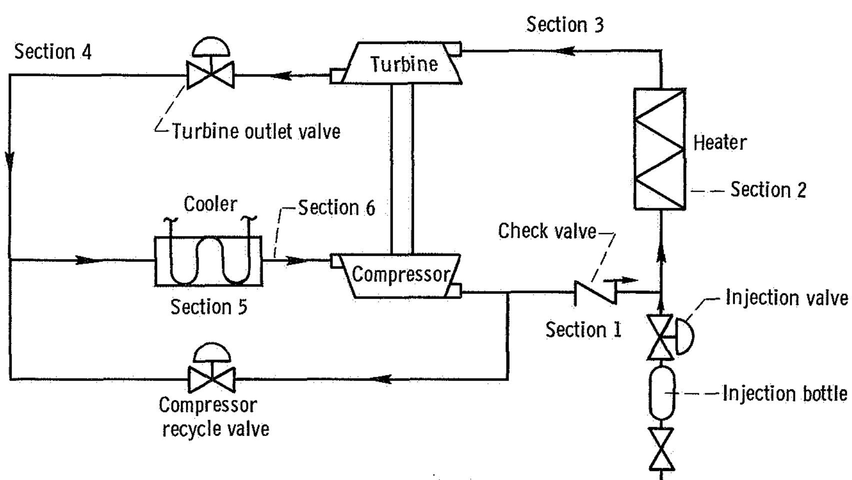
This report presents the following information: a discussion of the events that occur during a closed loop injection start, comparisons of actual and simulated starts and shutdowns, the variation of rotative speed with time as a function of injection rate, turbine inlet temperature, and loop design during the injection period, and compressor operating characteristics following the injection period. All symbols used herein are defined in appendix A.

EXPERIMENTAL TEST LOOP

A schematic diagram of the test loop containing all the components for the two-shaft closed loop Brayton cycle system is presented in figure 1(a). The turbocompressor, turboalternator, and recuperator are experimental flight-type equipment; the heater and cooler are workhorse components intended for ground use only. These components make up the final configuration for testing the two-shaft Brayton cycle system in its breadboard



(a) Breadboard configuration of complete closed loop Brayton cycle system.



(b) Turbocompressor test configuration.

Figure 1. - Test loop flow.

form. The system working fluid is argon. The turbocompressor and turboalternator rotors are mounted on self-acting argon-lubricated bearings.

A schematic diagram of the test loop discussed herein is shown in figure 1(b). This loop is the initial configuration designed to test only the turbocompressor, but since it does not contain the turboalternator, a control valve at the turbine outlet is used to simulate the turboalternator pressure drop and to control the speed. A complete description of this loop, including the instrumentation, is given in reference 1. A schematic cross section of the turbocompressor is shown in figure 2.

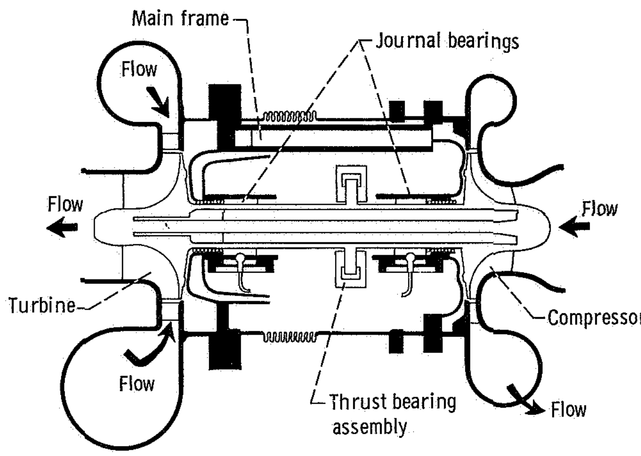


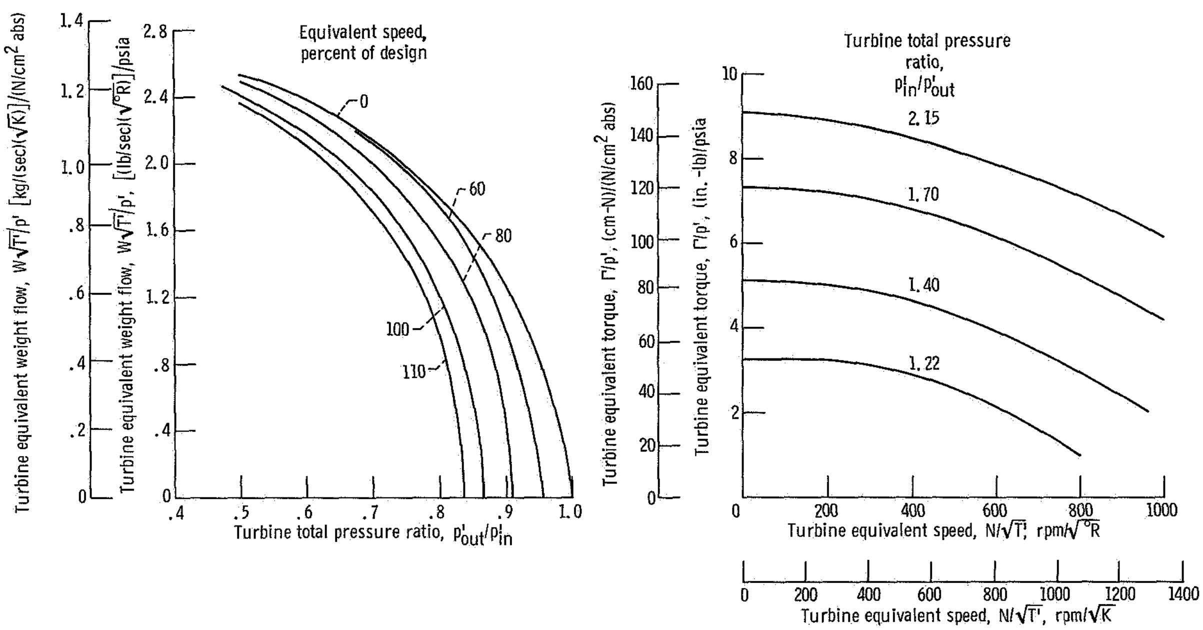
Figure 2. - Turbocompressor package.

The test loop was designed for closed loop injection starts. In this start method, the system inventory of argon is injected into the heater inlet. The hot gas from the heater expands through the turbine and accelerates the turbocompressor rotor to self-sustaining speed. The check valve prevents backflow through the compressor. An injection bottle is provided to simulate the type of start that might be made in a space powerplant. The bottle is charged to the desired pressure level, and the injection valve is then opened to the proper position to provide the required initial flow rate. Constant-rate injection is accomplished by controlling the flow rate with the injection valve. At the beginning of the injection period, the check valve is closed and the compressor recycle valve is partly opened to prevent compressor surge. Inventory is injected until a preset compressor inlet pressure is obtained. When this set point is reached, an automatic control system closes the injection and compressor recycle valves and opens the check valve.

SIMULATION METHOD

A description of the analog computer program used for this simulation is given in appendix B. The program includes the turbomachinery operating maps, loop volumes, piping pressure drops, and piping temperature changes.

The injection procedure used for the simulation is the same as that used during operation of the test loop. Argon is injected into the heater either from the bottle or at a constant rate. When injection commences, the check valve is closed and the compressor recycle valve is partly open. When the compressor inlet pressure reaches the set point, the injection and compressor recycle valves are closed and the check valve is opened.



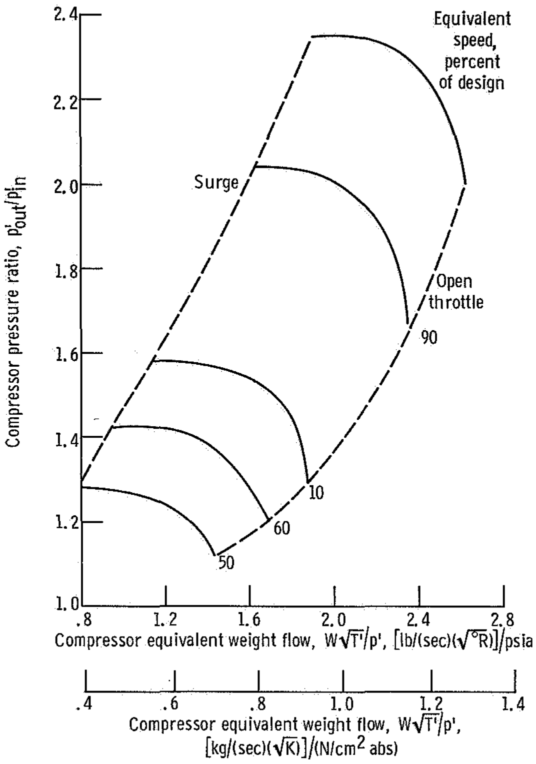
(a) Turbine equivalent weight flow.

(b) Turbine equivalent torque.

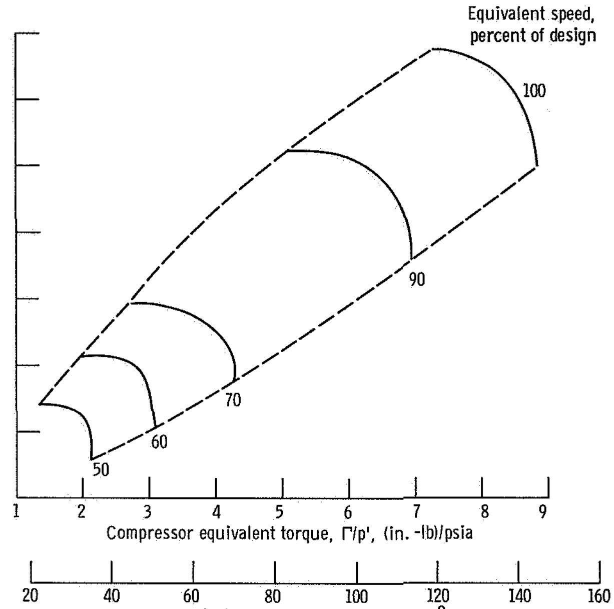
Figure 3. - Turbine and compressor performance maps.

The turbine and compressor performance maps used for the simulation are shown in figure 3. These maps were drawn from test data obtained at the Lewis Research Center. Compressor performance data are given in reference 2 and turbine performance data in reference 3.

Some simplifying assumptions were made to reduce problem complexity and to conserve computer equipment:



(c) Compressor equivalent weight flow.



(d) Compressor equivalent torque.

Figure 3. - Concluded.

1. The heater and cooler discharge temperatures remain constant during injection starts and during the first few seconds of shutdowns. This is approximately true for actual loop operation.

2. During injection starts, before the control valve opens, the compressor operates close to surge. It was assumed that during injection the compressor recycle valve was at a fixed position which approximated surge-line operation. This type of operation was specified because it produces the highest rotative acceleration.

3. The volumes of the compressor, turbine, compressor recycle line, injection line, and the volume between the compressor discharge and the check valve were neglected. These volumes are all small compared to other loop piping sections.

4. Valve operation is assumed to be instantaneous. Comparisons of actual test results with simulated results indicate that this assumption has little effect on the accuracy of start simulations. The effect on emergency shutdown simulations, discussed under Loop shutdown, is much greater.

5. Except for the comparisons of actual and simulated tests, isothermal flow is assumed in the line between the heater and turbine. In the actual test loop, heat transfer in this line causes the turbine inlet temperature to be somewhat lower than the heater outlet temperature.

6. The mean gas temperature in the cooler inlet line is 600° R (333 K). This low temperature is the result of heat transfer between the turbine discharge gas and the cold piping of the cooler inlet line.

7. The initial loop pressure for all starts is 0.75 psia (0.52 N/cm^2 abs). Lower initial pressures result in overloading of some of the computer equipment.

RESULTS AND DISCUSSION

The results obtained from the analog simulation of the Brayton cycle test loop are presented in three sections. The first section discusses the events that occur during an injection start, and the second section compares the actual and simulated injection starts and shutdowns. Rotative speed and turbocompressor pressures, weight flows, and pressure ratios are shown as functions of time. The third section shows the effect of system variables on start characteristics. Rotative speed is shown as a function of time to illustrate the effects of varying injection rates, turbine inlet temperatures, and loop volumes. The effect of eliminating heat transfer in the cooler inlet line is also presented. The compressor weight flow map is used to show the compressor operating characteristics following the injection period.

Description of Simulated System Start

A simulated start at the design turbine inlet temperature of 1950° R (1083 K) is presented in figure 4 along with various turbomachinery operating variables shown as functions of time. Injection was stopped when the compressor inlet pressure reached 8.0 psia (5.5 N/cm^2 abs). At the same time, the check valve opened and the compressor recycle valve closed. The other conditions were an injection rate of 0.50 pound per second (0.23 kg/sec), an initial loop pressure of 0.75 psia (0.52 N/cm^2 abs), and a compressor inlet temperature of 536° R (298 K).

The turbine pressure ratio (fig. 4(d)) increased rapidly when injection was first started. The pressure ratio reached a maximum value and then decreased because of increasing turbine back pressure. The turbine equivalent torque Γ/p (fig. 4(g)) and the equivalent weight flow $W\sqrt{T}/p$ (fig. 4(i)) are dependent on the pressure ratio and equivalent speed. Since the rotative speed is only 2200 rpm at 0.55 second, the pressure ratio determines the values of Γ/p and $W\sqrt{T}/p$. These parameters reach maximum values at the same time as the pressure ratio. The turbine weight flow varies directly with both the equivalent weight flow and the turbine inlet pressure. Since the pressure increases throughout the injection period, the turbine weight-flow maximum value occurs

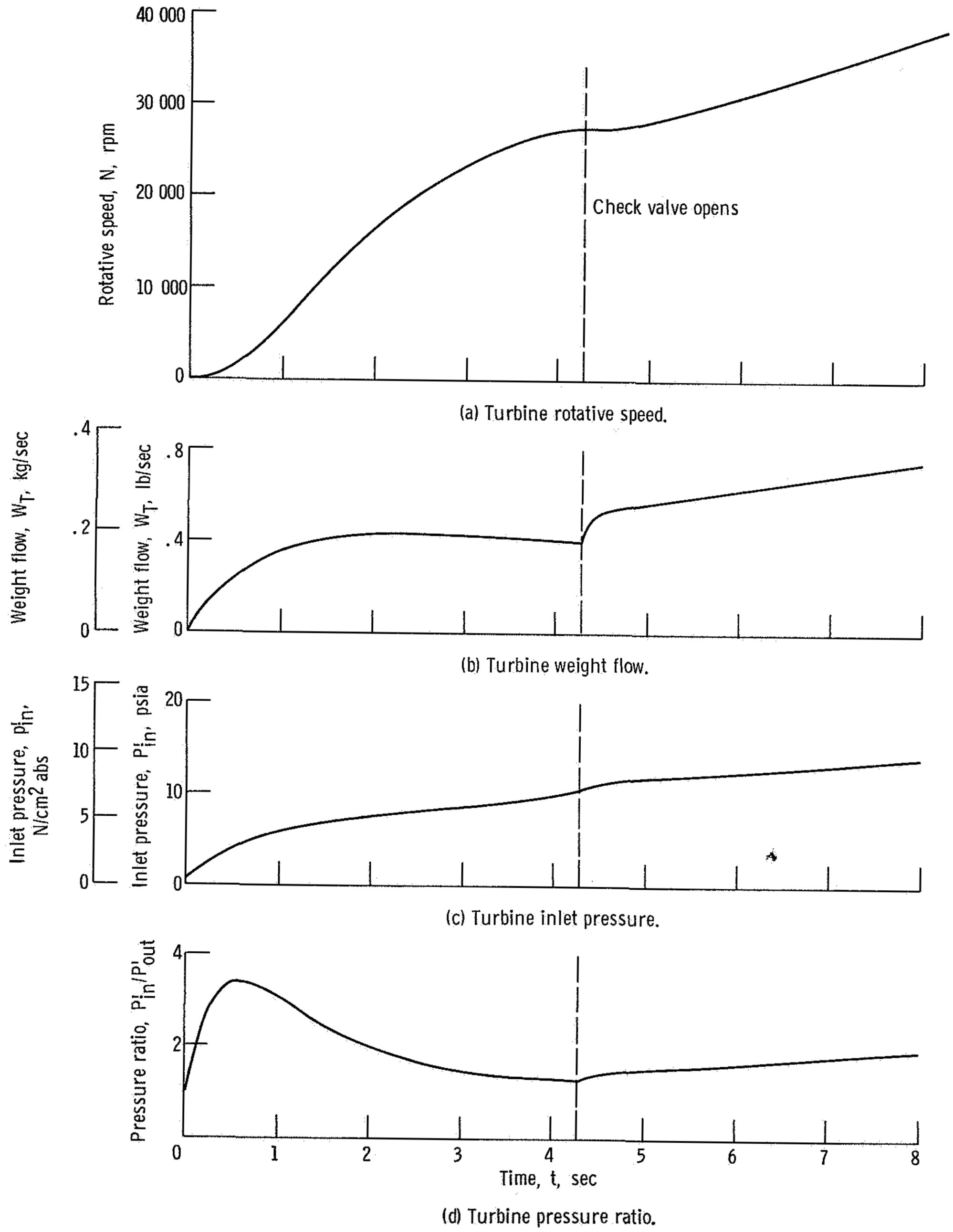
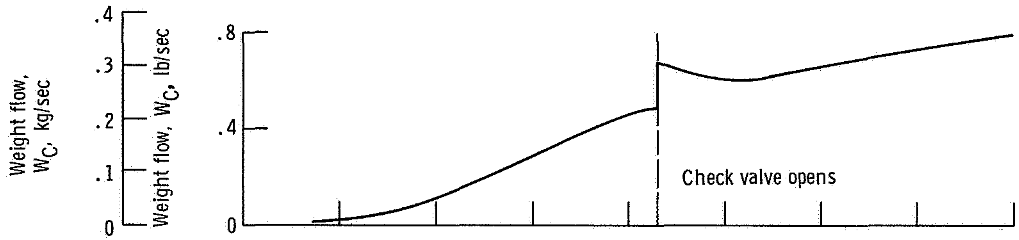
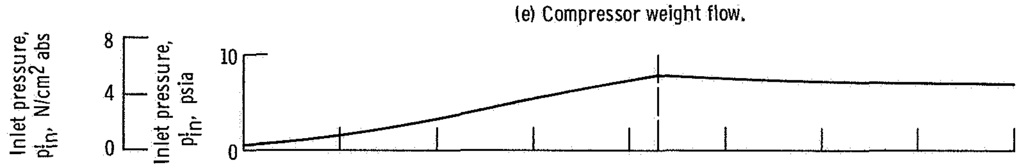


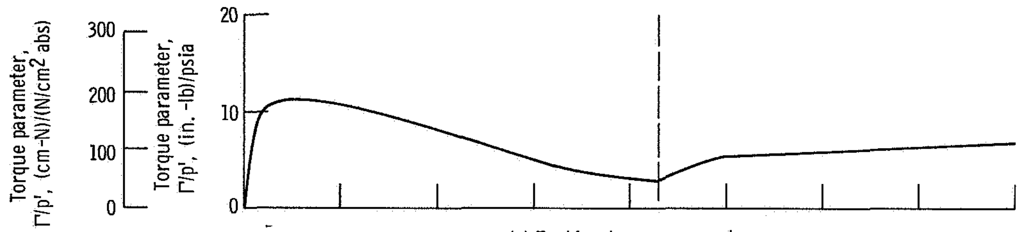
Figure 4. - Predicted startup characteristics of turbomachinery at design turbine inlet temperature of 1950° R (1083 K). Compressor inlet temperature, 536° R (298 K); initial loop pressure, 0.75 psia (0.52 N/cm^2 abs); injection rate, 0.50 pound per second (0.23 kg/sec); injection stopped and check valve opened at compressor inlet pressure of 8.0 psia (5.5 N/cm^2 abs).



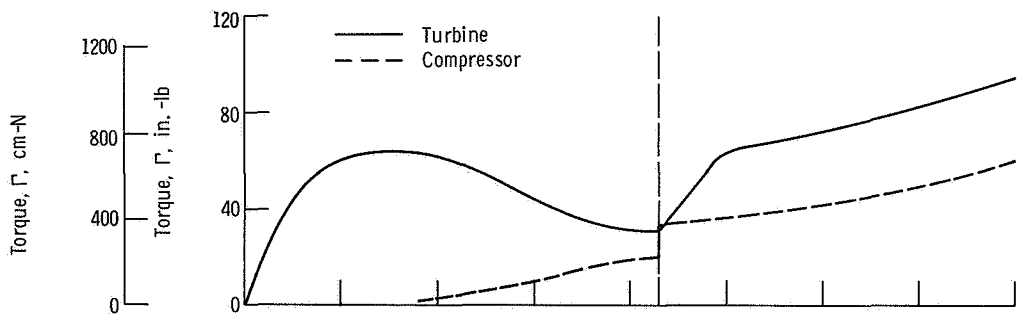
(e) Compressor weight flow.



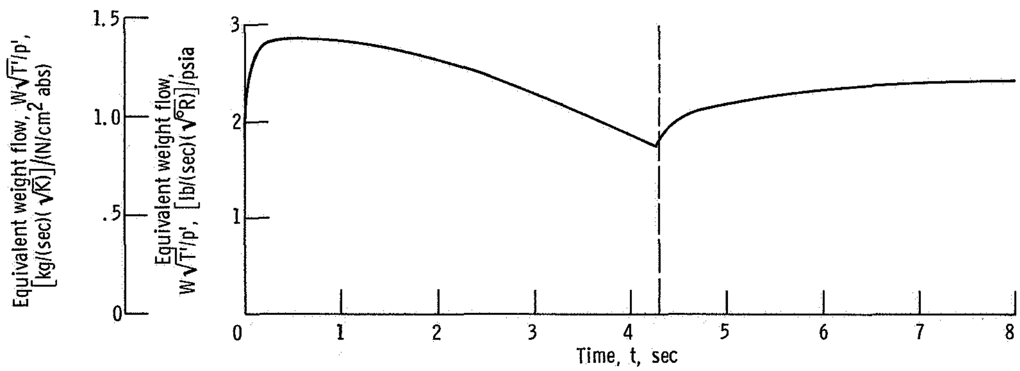
(f) Compressor inlet pressure.



(g) Turbine torque parameter.



(h) Turbine and compressor torque.



(i) Turbine equivalent weight flow.

Figure 4. - Concluded.

later than the equivalent weight-flow maximum value. Turbine torque, which varies directly with the equivalent torque and the turbine inlet pressure, behaves in a similar manner.

Because the compressor is the last piece of equipment in the flow path, the compressor inlet pressure (fig. 4(f)) lags the turbine inlet pressure. For this reason, and because of the low compressor equivalent weight flow at low rotative speeds, compressor weight flow is lower than turbine weight flow during the early part of the injection period.

The compressor torque (fig. 4(h)) increases throughout the injection period because of increasing rotative speed and compressor inlet pressure.

Comparison of Experimental and Simulated Runs

Injection start. - A comparison between an actual and a simulated start during the injection period is presented in figure 5, where rotative speed and several other operating variables are shown as functions of time. The operating conditions were an initial loop pressure of 0.75 psia (0.52 N/cm^2 abs) for the simulation and 1.5 psia (1.0 N/cm^2 abs) for the actual start, an injection rate of 0.47 pound per second (0.21 kg/sec), a heater outlet temperature of 1540^o R (855 K), and a turbine inlet temperature of 1200^o R (667 K). In both starts, injection was stopped at a compressor inlet pressure of 8.1 psia (5.6 N/cm^2 abs).

The simulation injection was stopped at 4.4 seconds. For the actual start, the injection valve started to close at 4.0 seconds. This difference in time resulted partly from the 0.75-psia (0.52-N/cm^2) difference in the initial loop pressure. The good agreement between the actual and simulated turbine and compressor inlet pressures indicated that the assumed loop volumes, temperatures, and weight flows were approximately correct.

At 4.0 seconds, the simulation rotative speed was 16 300 rpm, and the actual rotative speed was 17 500 rpm. This variation was caused by a slight difference between the actual and simulation acceleration rates between 9 000 and 12 000 rpm. Above and below this speed range, the two acceleration rates are in very close agreement. Another comparison between an actual and simulated injection start (not presented herein) showed a similar variation in rotative speed. When injection was stopped, the rotative speed was 14 300 rpm for the simulation and 15 300 rpm for the actual start.

The turbine weight flow computed for the actual start was greater than the simulation weight flow throughout the injection period. At 4.0 seconds, the simulation weight flow was 0.34 pound per second (0.15 kg/sec), whereas the computed weight flow for the actual start was 0.38 pound per second (0.17 kg/sec). This variation may have occurred because the test loop turbine had a greater flow area, possibly the result of hot opera-

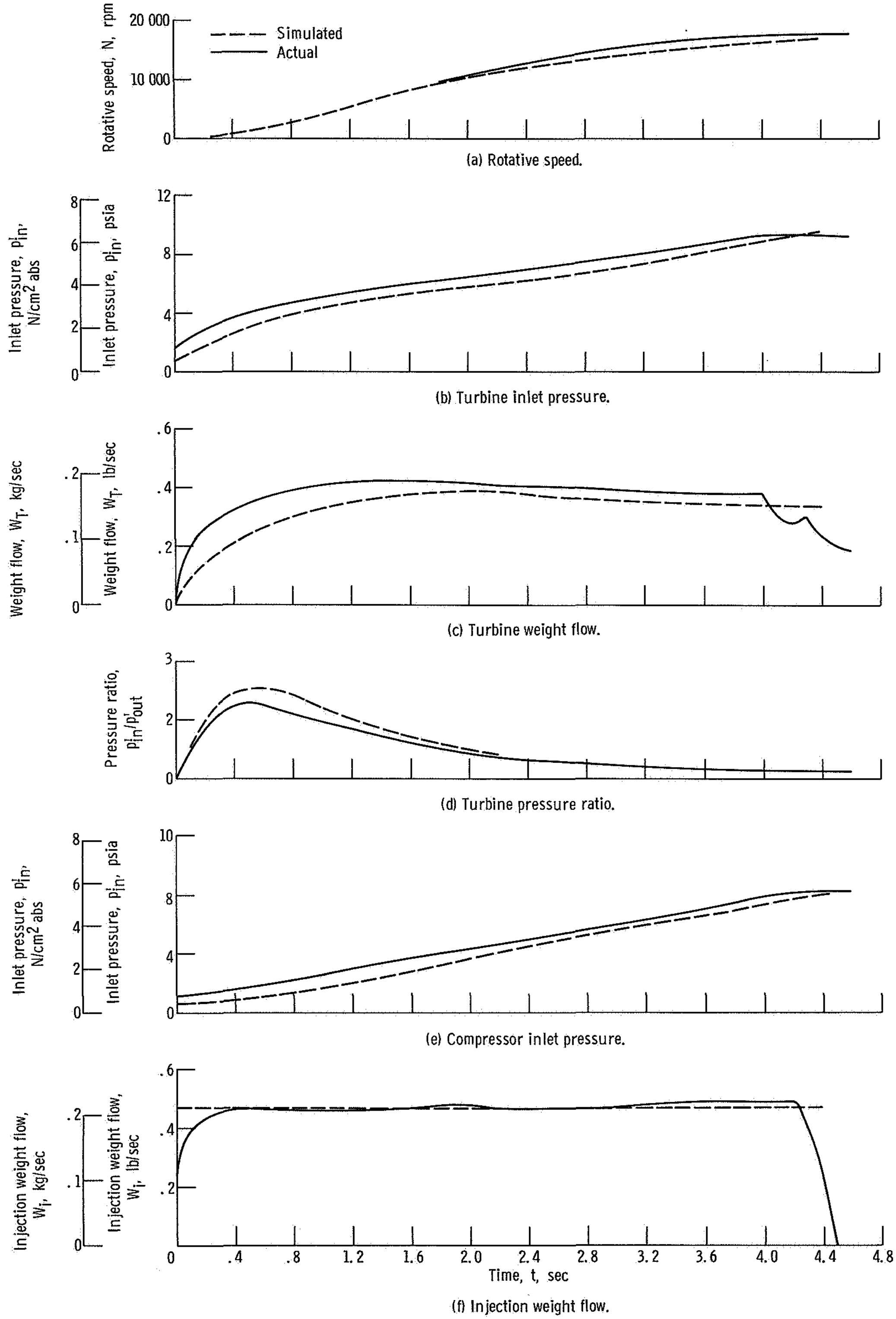


Figure 5. - Comparison of actual and simulated starts showing variation of loop operating variables with time. Turbine inlet temperature, 1200° R (667 K); heater outlet temperature, 1540° R (855 K); injection rate, 0.47 pound per second (0.21 kg/sec); initial loop pressure for simulation start, 0.75 psia (0.52 N/cm² abs); initial loop pressure for actual start, 1.5 psia (1.0 N/cm² abs); injection stopped at compressor inlet pressure of 8.1 psia (5.6 N/cm² abs).

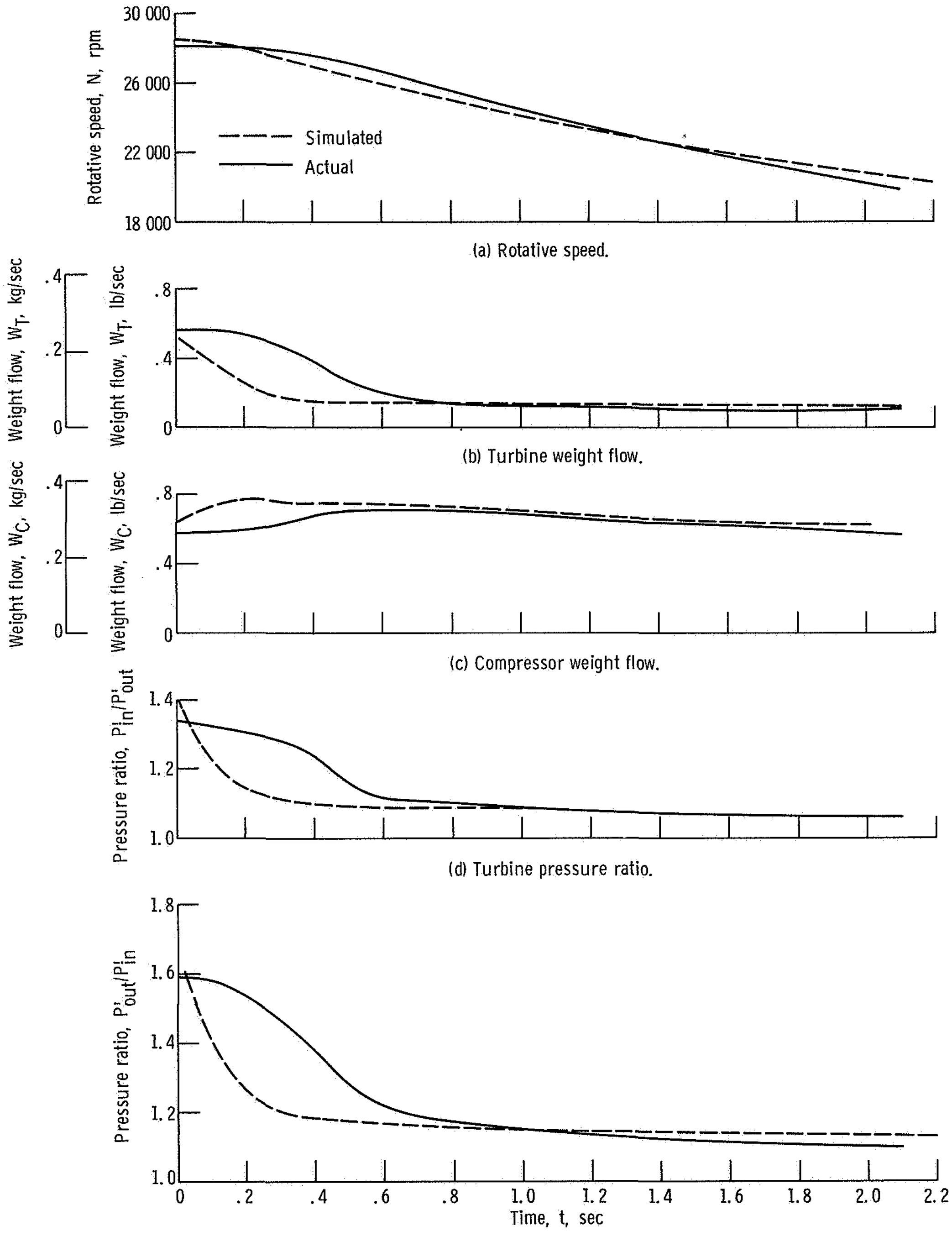


Figure 6. - Comparison of actual and simulated shutdown showing variation of loop variables with time. Compressor recycle valve opened at 28 000 rpm; turbine inlet temperature, 1600° R (889 K); heater outlet temperature, 1930° R (1072 K); initial turbine inlet pressure, 12.8 psia (8.83 N/cm^2 abs).

tion, than the turbine that was tested to obtain the performance maps. During the early part of the injection period, the turbine pressure ratio of the simulation was higher than that of the actual start, which might have been caused by the higher initial pressure in the loop and the time required to establish full injection flow.

Loop shutdown. - A comparison between an actual and a simulated loop shutdown is presented in figure 6, which shows the variation with time of rotative speed, compressor and turbine weight flows, and compressor and turbine pressure ratios. Before the shutdown, the speed was controlled at approximately 28 000 rpm with the turbine-discharge valve. The shutdown was accomplished by opening the compressor recycle valve. Initial conditions included an initial rotative speed of 28 000 rpm for the actual shutdown and 28 500 rpm for the simulation, a heater outlet temperature of 1930° R (1072 K), a turbine inlet temperature of 1600° R (889 K), and a turbine inlet pressure of 12.8 psia (8.83 N/cm^2 abs).

At 2.1 seconds, the actual rotative speed had decreased from 28 000 to 19 800 rpm. During the same period, the simulation rotative speed decreased from 28 500 to 20 500 rpm. Initially, the simulation weight flows and pressure ratios decreased faster than the actual variables. This lag occurred because of the time required for the compressor recycle valve to operate. After 0.7 second, the actual pressures and weight flows agreed fairly well with the simulation values.

Effect of System Variables on Start Characteristics From Simulation

The variation of rotative speed with time and injection rate during the injection period is shown in figure 7 for a design turbine inlet temperature of 1950° R (1083 K), a loop initial pressure of 0.75 psia (0.52 N/cm^2 abs), and a compressor inlet temperature of 536° R (298 K).

The rotative speed at the end of any given injection period increased with the injection rate. At 2.0 seconds, rotative speed increased from 7300 to 24 700 rpm as the injection rate increased from 0.25 to 0.75 pound per second (0.11 to 0.34 kg/sec). The rotative speed increases with injection rate for any given time because both turbine pressure ratio and turbine inlet pressure also increase with injection rate.

An example of the variation of rotative speed with injection rate and time when injection was from the bottle is shown in figure 7. The bottle, which has a volume of 0.72 cubic foot (0.020 cu m) was charged to its maximum pressure of 600 psia (414 N/cm^2 abs). The initial injection rate was 0.75 pound per second (0.34 kg/sec). The curve of rotative speed as a function of time for bottle injection shows that rotative speed reaches a maximum value and then decreases. This decrease occurred because the injection rate decreased to a point where bearing friction torque and compressor torque

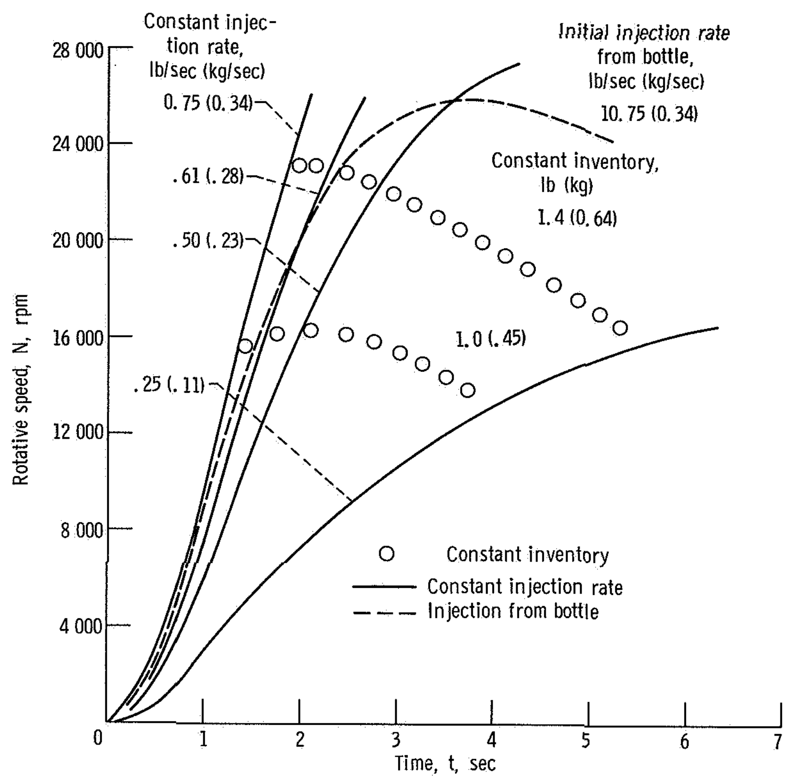


Figure 7. - Variation of rotative speed with time and injection weight flow. Turbine inlet temperature, 1950° R (1083 K); compressor inlet temperature, 536° R (298 K); initial loop pressure, 0.75 psia (0.52 N/cm^2 abs).

exceeded turbine torque. The injection rate from the bottle is proportional to the bottle pressure.

Figure 7 shows that for a fixed amount of inventory, rotative speed does not necessarily increase with injection rate. For an inventory of 1.0 pound, (0.45 kg), rotative speed reaches a maximum value of 16 300 rpm with an injection rate of 0.5 pound per second (0.23 kg/sec). For a design inventory of 1.4 pounds (0.64 kg), rotative speed reaches a maximum value of 23 200 rpm at an injection rate of about 0.70 pound per second (0.32 kg/sec).

At a design turbine inlet temperature of 1950° R (1083 K), no difficulty was encountered in reaching self-sustaining speed. The estimated self-sustaining speed for this temperature is approximately 8000 rpm. At the lowest injection rate of 0.25 pound per second (0.11 kg/sec), 8000 rpm is reached in 2.2 seconds. If design inventory is injected, all the flow rates shown in figure 7 produce rotative speeds far above the self-sustaining speed.

Turbine inlet temperature. - The variation of rotative speed with time and turbine inlet temperature during the injection period is shown in figure 8, which also includes curves of constant turbine pressure ratio. The conditions for these simulated runs were

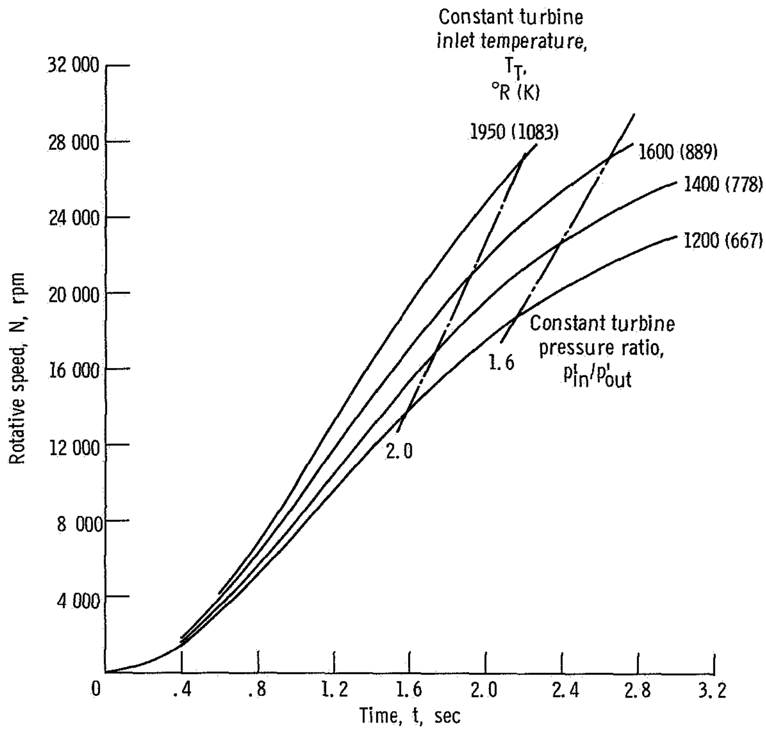


Figure 8. - Variation of rotative speed with time and turbine inlet temperature. Compressor inlet temperature, 536° R (298 K); injection rate, 0.75 pound per second (0.34 kg/sec); initial loop pressure, 0.75 psia (0.52 N/cm^2 abs).

a loop initial pressure of 0.75 psia (0.52 N/cm^2 abs), an injection rate of 0.75 pound per second (0.34 kg/sec), and a compressor inlet temperature of 536^o R (298 K).

Figure 8 shows that for any injection period or for any amount of inventory, rotative speed increases with turbine inlet temperature. At 1.87 seconds, corresponding to a design inventory of 1.4 pounds (0.64 kg), rotative speed increased from 16 600 to 23 300 rpm as the turbine inlet temperature was increased from 1200 to 1950^o R (667 to 1083 K). No difficulty was encountered in reaching self-sustaining speed even when the turbine inlet temperature was reduced from the design value of 1950^o R (1083 K) to 1200^o R (667 K). The estimated self-sustaining speed for 1200^o R (667 K) is 12 000 rpm. This speed was attained in 1.5 seconds with 1.12 pounds (0.508 kg) of inventory.

For a given injection period and injection rate, the rotative speed increases with turbine inlet temperature because of correspondingly higher values for turbine inlet pressure and pressure ratio, which produce higher torque. Curves of constant pressure ratio for values of 1.6 and 2.0 are shown in figure 8. These curves apply to the period after the pressure ratios have passed through their maximum values and are decreasing.

Predicted effects of design changes. - The test loop has a greater volume than that of a flight-type loop. The greater volume is attributed to longer piping runs and to the high volumes of the workhorse heater and cooler. In a flight-type loop or space powerplant, weight and size would be minimized with short piping runs and compact, low-volume heat-transfer equipment.

In the discussion of the simulation method, it was pointed out that heat transfer in the cooler inlet line reduces the average gas temperature in this line to approximately 600^{o} R (333 K). Without this heat transfer, the temperature in this line would be equal to the turbine discharge temperature. The cooler mean temperature would also be greater because of the higher cooler inlet temperature. This heat transfer has an important effect on loop starting characteristics: If the effect of piping and equipment pressure drops are neglected, for a given amount of inventory downstream from the turbine, the turbine-discharge pressure will be proportional to the mean temperature:

$$\text{Pressure} \sim \frac{(\text{Mass of inventory}) \times T_{ave}}{\text{Volume}}$$

Therefore, the turbine pressure ratio will increase as the mean temperature downstream from the turbine is decreased. In a flight-type test loop, short piping sections would minimize heat transfer, and the gas temperature in the cooler inlet line would be almost equal to the turbine-discharge temperature.

The effect of reducing the loop volume is shown in figure 9, for which the inlet volume is defined as the volume between the check valve and the turbine inlet in the direction of flow. Exit volume is defined as the volume between the turbine discharge and the check valve in the direction of flow. For a loop with no heat transfer in the cooler inlet line, rotative speed is shown as a function of time, with the inlet and exit volumes as independent variables and the following operating conditions: turbine inlet temperature, 1950^{o} R (1083 K); initial loop pressure, 0.75 psia (0.34 N/cm^2 abs); injection rate, 0.5 pound per second (0.227 kg/sec); compressor inlet temperature, 536^{o} R (298 K).

With the design exit volume of 30 cubic feet (0.85 cu m) and the design inventory of 1.4 pounds (0.64 kg) injected, the rotative speed increased from 13 800 to 16 200 rpm as the inlet volume was decreased from 16 to 4.1 cubic feet (0.45 to 0.12 cu m). Decreasing the inlet volume decreases the amount of inventory retained upstream; consequently, the turbine inlet pressures, pressure ratios, torques, and weight flows are increased.

With the design inlet volume of 16 cubic feet (0.45 cu m) and the design inventory injected, the rotative speed increased from 2 400 to 13 800 rpm as the exit volume was increased from 7.5 to 30 cubic feet (0.21 to 0.85 cu m). When the exit volume is decreased, the turbine-discharge pressure increases more rapidly, which reduces the turbine pressure ratio and torque. In addition, the decrease in volume shortens the injection period required to inject the design inventory. The estimated self-sustaining speed of

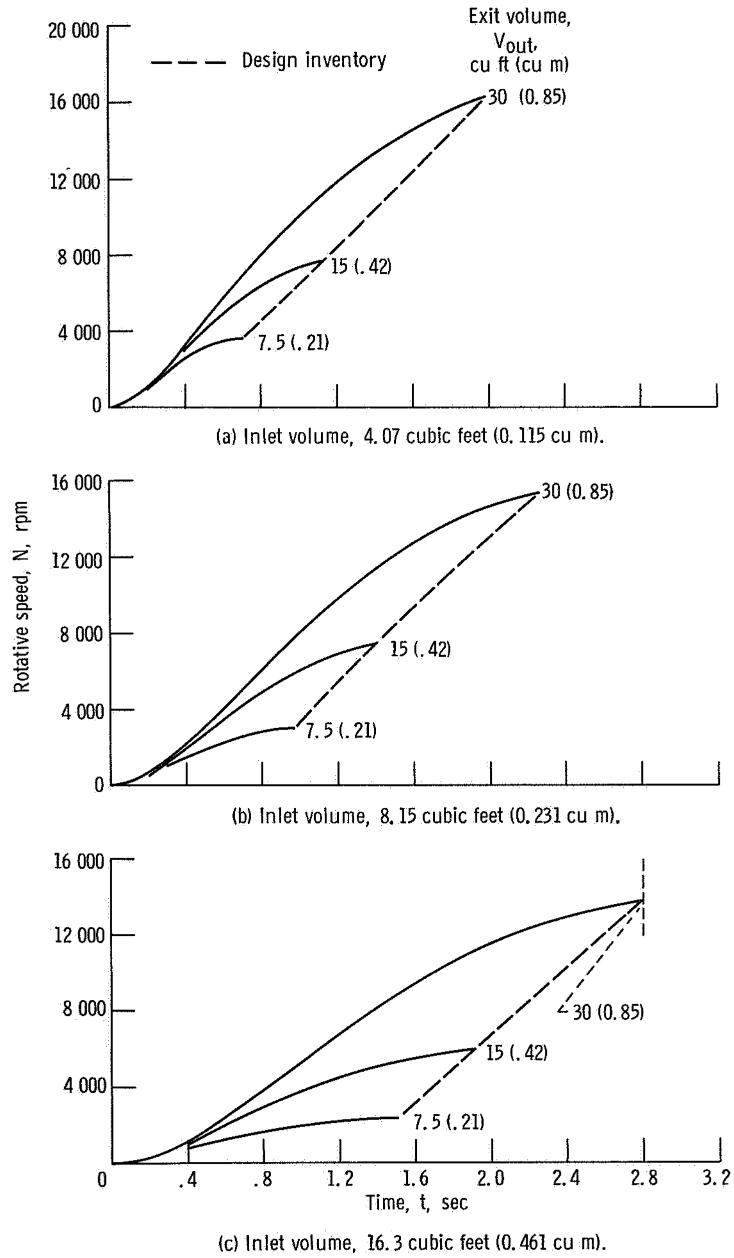


Figure 9. - Variation of rotative speed with time, inlet volume, and exit volume. Turbine inlet temperature, 1950° R (1083 K); compressor inlet temperature, 536° R (298 K); injection rate, 0.5 pound per second (0.227 kg/sec); initial loop pressure, 0.75 psia (0.34 N/cm^2 abs); design inventory, 1.4 pound (0.64 kg).

8000 rpm could not be obtained for exit volumes of 7.5 and 15 cubic feet (0.21 and 0.42 cu m), even by overcharging the loop. If injection starting is used for test loops and powerplants with small exit volumes, open loop starts must be used. In open loop starts, part of the injected gas is vented to a low-pressure area after it passes through the turbine. In figure 10 the increase in rotative speed produced by heat transfer in the

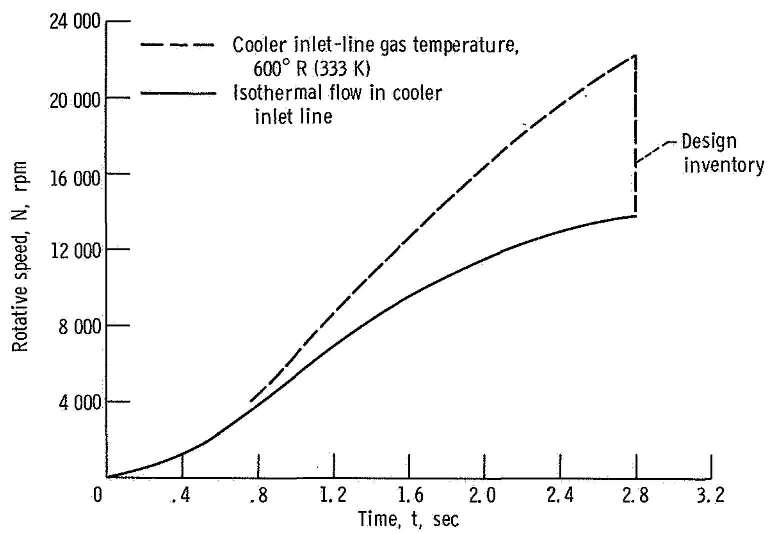


Figure 10. - Variation of rotative speed with time as seen through comparison of injection start for cooler inlet-line gas temperature of 600° R (333 K) with injection start for isothermal flow in cooler inlet line. Turbine inlet temperature, 1950° R (1083 K); compressor inlet temperature, 536° R (298 K); injection rate, 0.5 pound per second (0.23 kg/sec); initial loop pressure, 0.75 psia (0.34 N/cm^2 abs); inlet volume, 16 cubic feet (0.45 cu m); exit volume, 30 cubic feet (0.85 cu m).

cooler inlet line is shown for design inlet and exit volumes of 16 and 30 cubic feet (0.45 and 0.85 cu m), respectively. Conditions are the same as those for figure 9. When design inventory is injected, a rotative speed of 13 800 rpm is obtained for the case with no heat transfer. If the average gas temperature in the line is 600° R (333 K), the rotative speed is 22 200 rpm.

Compressor operating characteristics after injection period. - The compressor should not operate in surge at any time. Experimental results reported in reference 1 show that compressor surge may cause bearing instability. Also, if the compressor enters the surge region after the injection period, it may be impossible to accelerate to design operating conditions.

The compressor operating characteristics after the injection period are shown in figure 11. The operating curves drawn on the compressor equivalent weight flow map represent compressor performance during rotative acceleration. Curves are shown for

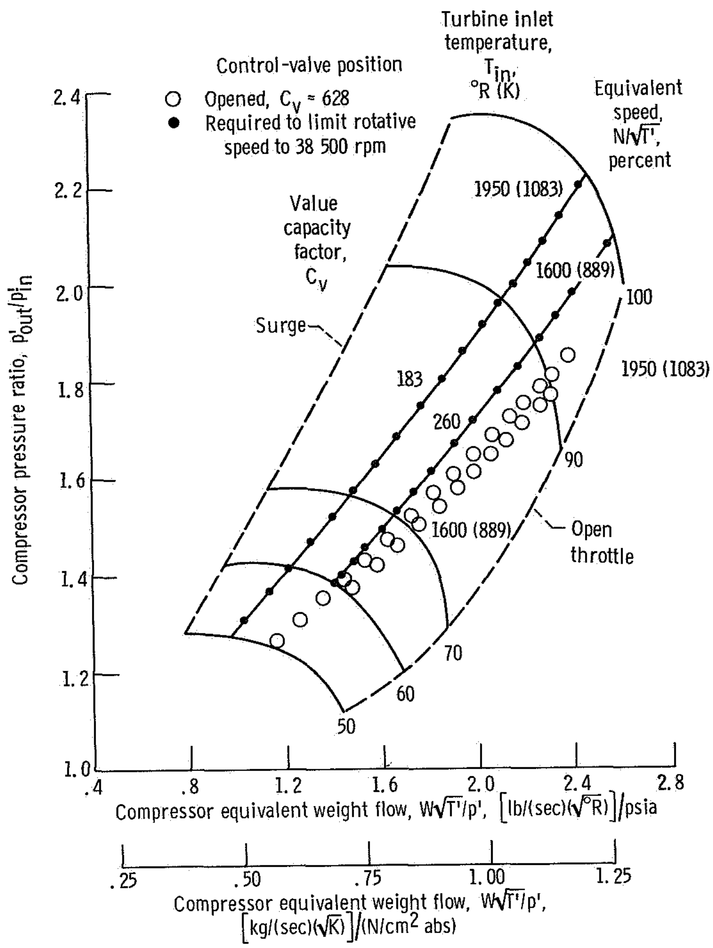


Figure 11. - Compressor equivalent weight-flow map showing compressor operating characteristics after the check valve opens.

turbine inlet temperatures of 1600° and 1950° R (889 and 1083 K) for two turbine-discharge control-valve positions. One pair of curves is for an open control valve and the other pair is for the control-valve position required to limit rotative speed to 38 500 rpm. None of the conditions shown causes the compressor to enter the surge region. As expected, the partially closed control-valve positions caused the compressor to operate closer to surge. The effect was greater at 1950° R (1083 K) than at 1600° R (889 K) because it was necessary to close the valve further to limit rotative speed to 38 500 rpm. With the control valve open, the compressor operates slightly closer to surge at the higher temperature.

Also shown in figure 11 are the control-valve capacity coefficients C_v required to limit rotative speed to 38 500 rpm. These values are 260 for 1600° R (889 K) and 183 for 1950° R (1083 K). For the open control valve, the value of C_v is 628.

SUMMARY OF RESULTS

The performance of a Brayton cycle test loop was simulated with an analog computer. The test loop turbomachinery consisted of a turbocompressor designed for a 10-kilowatt-shaft-output powerplant. The working fluid was argon. The results of the simulation were compared with test results and were then used to study the effect of system variables on closed loop injection start characteristics. The following results were obtained from the study:

1. Comparison of the simulated injection starts with actual injection starts indicated that the actual variations of loop operating variables with time agreed well with the computer predictions.

2. Closed loop injection was both analytically and experimentally a practical means of accelerating the turbocompressor rotor to self-sustaining rotative speeds. At the design turbine inlet temperature of 1950° R (1083 K), starts could be obtained with injection flow rates of less than 0.25 pound per second (0.11 kg/sec) without exceeding the design inventory. At a turbine inlet temperature of 1200° R (667 K), starts could be obtained at flow rates of less than 0.75 pound per second (0.34 kg/sec) without exceeding the design inventory.

3. The rotative speed at the end of any given injection period increased with the injection rate.

4. When a fixed amount of inventory was injected into the loop, an injection rate could be found that would produce a maximum rotative speed. The value of the injection rate corresponding to the maximum rotative speed increased with the amount of inventory injected.

5. The turbocompressor could be brought to design operating conditions from zero speed without surging the compressor at any time. Surge does not occur even if the turbine-discharge control valve has been preset to the position required to limit the rotative speed to 38 500 rpm.

6. Heat transfer in the cooler inlet line had an important effect on the loop starting characteristics. Elimination of this heat transfer would cause large decreases in the rotative speeds obtained during the injection period.

7. The test loop volume had an important effect on starting characteristics. Decreases in the volume upstream from the turbine produced higher rotative speeds during the injection period; decreases in the volume downstream from the turbine reduced rotative speeds. Closed loop injection cannot be used to start test loops with relatively small volumes downstream from the turbine.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 30, 1968,
120-27-03-13-22.

APPENDIX A

SYMBOLS

C_v valve capacity factor, $27.2\, W\sqrt{T/(p_{out}\, \Delta p)}$; $30.4\, W\sqrt{T/(p_{out}\, \Delta p)}$

C_p heat capacity at constant pressure, Btu/(lb)(^{o}F); J/(kg)(K)

Δh specific work output, Btu/lb; J/g

I polar moment of inertia, (lb-ft)-sec^2; kg-m^2

N rotative speed, rpm

p absolute pressure, psia; N/cm^2 abs

PR total pressure ratio, ratio of outlet total pressure to inlet total pressure

R gas constant, ft-lb/(lb)(^{o}R); J/(K)(kg)

T absolute temperature, ^{o}R; K

t time, sec

V volume, cu ft; cu m

W weight flow, lb/sec; kg/sec

Γ torque, in. -lb; cm-N

Γ_F friction torque; in. -lb; cm-N

Subscripts:

ave mass-averaged value

C compressor

i injection

T turbine

in conditions at inlet

out conditions at outlet

Superscript:

' absolute total state

APPENDIX B

COMPUTER PROGRAM

Turbocompressor Calculations

The turbine and compressor maps were simulated with empirical algebraic equations. Pressure ratio and equivalent speed were the independent variables used to compute equivalent weight flow and equivalent torque. For the turbine, the equivalent weight flow is

$$\left.\begin{aligned} \frac{W\sqrt{T'}}{P'} &= \left(0.681 - \frac{0.591}{1.133 - PR}\right)\left(\frac{N^2}{T}\times 10^{-6} + 0.75\right) + 2.82 \text{ (English units)} \\ &\text{or} \\ \frac{W\sqrt{T'}}{p'} &= \left(0.334 - \frac{0.290}{1.133 - PR}\right)\left(0.272\frac{N^2}{T}\times 10^{-6} + 0.37\right) + 1.38 \text{ (SI units)} \end{aligned}\right\} \qquad (1)$$

and the equivalent torque is

$$\left.\begin{aligned} \frac{\Gamma}{p'} &= -3.105\frac{N^2}{T}\times 10^{-6} - 16.67\ PR + 17.00 \text{ (English units)} \\ &\text{or} \\ \frac{\Gamma}{p'} &= -28.26\frac{N^2}{T}\times 10\text{-}6 - 273.1\ PR + 278.5 \text{ (SI units)} \end{aligned}\right\} \qquad (2)$$

For the compressor, several sets of maps were used, depending on the operating conditions. Two of these, for equivalent weight flow and equivalent torque, respectively, are

$$\left.\begin{aligned} \frac{W\sqrt{T'}}{p'} &= 1.137\frac{N^2}{T}\times 10^{-6} - 1.15\ PR + 1.90 \text{ (English units)} \\ &\text{or} \\ \frac{W\sqrt{T'}}{p'} &= 0.3907\frac{N^2}{T}\times 10^{-6} - 0.564\ PR + 0.93 \text{ (SI units)} \end{aligned}\right\} \qquad (3)$$

and

$$\frac{\Gamma}{p'} = 4.955\frac{N^2}{T}\times 10^{-6} - 3.90\ PR + 3.126 \text{ (English units)}$$

or $\qquad$ (4)

$$\frac{\Gamma}{p'} = 45.09\frac{N^2}{T}\times 10^{-6}\ \Sigma\ 63.88\ PR + 51.20 \text{ (SI units)}$$

These two equations are used only for operation above 50 percent speed near the open throttle line with the check valve open and the recycle valve closed. During the injection period, the compressor was assumed to operate along the surge line. This assumption is discussed in the section SIMULATION METHOD.

Weight flows are obtained from equations (1) and (3) and from the inlet pressures and temperatures:

$$W = \frac{W\sqrt{T}}{p} \times \frac{p}{\sqrt{T}}$$

Torque is obtained from equations (2) and (4) and from the suction pressure. The pressures are obtained from the piping calculations. Turbine inlet temperature is specified by the programmer. Compressor inlet temperature has a constant value of 536° R (298 K).

Rotative acceleration is directly proportional to the torque applied to the shaft and is inversely proportional to the polar moment of inertia:

$$\frac{dN}{dt} \sim \frac{\Gamma_T - \Gamma_C}{I} \tag{5}$$

Rotative speed is obtained by integrating equation (5).

The changes in total temperature across the compressor and turbine are obtained from the specific work:

$$\Delta T \sim \frac{\Delta h}{C_p} \sim \frac{\Gamma N}{W} \tag{6}$$

The bearing friction torque is assumed to be proportional to the rotative speed:

$$\Gamma_F \sim N \tag{7}$$

This torque is equivalent to a power loss of 300 watts at 38 500 rpm.

Piping Calculations

The loop piping is divided into six sections, each of which is identified by number in figure 1(a). Temperatures and pressures are computed separately for each section.

Piping temperatures. - Isothermal flow is assumed for all sections except the heater and cooler. Section 1 (fig. 1(a), p. 3) is assumed to be at the compressor discharge temperature. Section 3 is at the heater outlet temperature in most cases. The temperature assumptions for section 4 are discussed in the sections SIMULATION METHOD and Predicted effects of design changes. Section 6 is at 536° R (298 K).

The heater (Sec. 2) and cooler (Sec. 5) flows are not isothermal. The mean gas temperatures for these two sections are obtained from a simple arithmetic approximation, which is based on the assumption that the heater and cooler discharge gas temperatures are constant during a simulated test run. The temperatures of the heater and cooler heat-transfer surfaces also remain constant because of thermal inertia. Therefore, for any simulated test run, constant values can be used for the temperature differentials between the discharge gas and the heat-transfer surfaces in the heater and cooler. These constant values are used to compute mean temperatures. The discharge temperature differentials should vary from run to run as the heater outlet temperature changes. In the program, however, constant values are assumed for all conditions. This practice is justified because no significant errors result. For the conditions in this test loop, the computed mean gas temperatures are almost independent of the assumed discharge temperature differentials.

Piping pressures. - The ratio of pressure to temperature is obtained from the perfect gas law

$$\begin{gathered} \frac{p}{T} = \frac{R}{V} \times \text{Mass} \\ \frac{d\left(\frac{p}{T}\right)}{dt} = \frac{R}{V}\left(W_{in} - W_{out}\right) \end{gathered} \tag{8}$$

After the ratio p/T is determined, it is multiplied by the mean temperature to obtain the mean pressure.

The pressure drop across each piping section and the control valve is obtained from the approximation

$$\Delta p \sim \frac{W^2 T_{ave}}{p_{ave}}$$

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