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by Robert Y. Wong, Robert C. Evans, Donald J. Spackman, and Charles H. Winzig

Lewis Research Center
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ABSTRACT

A Brayton cycle turbocompressor was experimentally investigated to determine whether the gas bearings could withstand the transients associated with a space start using gas injection. Of the two injection methods studied, constant-flow injection and bottle injections, the constant-flow injection method imposed a lower thrust load on the turbocompressor during injection. It was also observed that the turbocompressor unit was able to withstand the transients during start, shutdown, and accidental compressor surge. A compressor-bypass valve proved to be capable of effecting emergency shutdown without causing severe vibrations.

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SUMMARY

As part of a technology program on Brayton cycle space-power systems, a turbocompressor for a reference two-shaft system operating on gas bearings was designed and built, under contract, to operate on argon. An experimental evaluation of the effects on the delivered hardware of transients associated with a simulated space start was made by using gas injection into the test facility heater. Two methods of injection were studied: constant-flow injections and injection flow rates which decay with time (called bottle injection). From a shaft-thrust standpoint, a constant-flow injection was the better of the two gas-injection methods investigated. Furthermore, the delivered turbocompressor was capable of withstanding the transients associated with rapid starts.

Accidental operation through compressor surge from 28 000 rpm occurred without damage to the unit. Also, flow-control valves were used to shut down the turbocompressor during a simulated emergency without causing compressor surge or any other unstable condition.

Strain gages, which were used to monitor bearing load, were found to be inadequate for the temperature range involved. The strain gages prevented operation at design point conditions because they were indicating unsafe pad load. Consequently, an alternate arrangement, including a proximity probe sensitive to bearing mount deflection, was devised for further operation.

INTRODUCTION

The NASA-Lewis Research Center is currently engaged in a technology program to study components for Brayton cycle space-power-generation systems. As part of this program a turbocompressor operating on self-acting gas bearings (with argon) for a reference 10-kilowatt two-shaft system was designed and fabricated under contract. It was then delivered to Lewis for ground test evaluation. A discussion of the reference system may be found in reference 1, and a description of the delivered turbocompressor may be found in reference 2.

A preliminary exploratory study was made of the mechanical and dynamic behavior of the delivered hardware as it was affected by rotor speed and turbine-inlet temperature. Rotor speeds up to the design value of 38 500 rpm and turbine-inlet temperatures up to 1460° R (811° K) were tested. The study showed (1) that the delivered hardware was provided with inadequate instrumentation to indicate journal bearing load, (2) that at 38 500 rpm the package is operating at the threshold of a critical speed, and (3) that there is insufficient thermal isolation between the turbine working fluid and the gas bearings.

The next phase of the turbocompressor evaluation program was to determine the effect on the hardware of transients associated with a space start of a two-shaft Brayton cycle space-power system. Start of a two-shaft system is accomplished by rotating the turbocompressor to a sufficient speed so that the combination of flow from the compressor and heat from the heat source delivers enough energy to the turbine to sustain operation of the compressor. One means of attaining self-sustaining speed without the complication of a starter motor and associated equipment is to evacuate the system and then to inject the system inventory of working fluid in through the heat source and turbine to spin up the turbocompressor to a self-operating speed.

Injection start in this manner subjects the turbocompressor to high-thrust loads and high-internal-temperature gradients which could affect the ability of the gas bearings to perform as designed. Furthermore, a flow-control valve required to direct the injected flow through the heat source and turbine may cause the compressor to surge and result in large rotor vibrations.

The objective of this investigation was to subject the delivered turbocompressor to such transients as may be encountered in a typical space injection start of a two-shaft Brayton cycle space-power system. Observations were made of its mechanical and dynamic reaction to rapid acceleration, high-turbine-pressure ratio, and high-turbine heat flux. In addition, the effect of flow-control valves (as described in the apparatus section) on start and shutdown were studied.

The experimental investigation consisted of injecting argon into the ground-test system at various flow rates up to 5 times system design flow rate and at various turbine-inlet temperatures up to 1660° R (922° K). Observations were made of rotor radial and axial motions together with internal temperatures and journal-bearing loads and thrust-bearing clearances.

TURBOCOMPRESSOR PACKAGE DESCRIPTION

Figure 1 is a cross-sectional view of the turbocompressor package. The radial inflow turbine and centrifugal compressor are mounted on the ends of a common shaft with two journals and a thrust runner between them. The turbine rotor has a tip diameter of 6.02 inches (15.3 cm) at the inlet and 4.29 inches (10.9 cm) at the outlet. There are 11 blades and 11 splitter blades on the rotor. The compressor rotor has a diameter of 3.50 inches (8.89 cm) at the inlet and 6.00 inches (15.25 cm) at the outlet. There are 15 blades on the rotor and 23 blades in the diffuser.

Each of the journal bearings has three self-acting pivoted pads using argon for lubrication. The pivot design includes a fully conforming ball and socket combination. The surfaces of the bearing pad, shaft, pivot ball, and socket are tungsten carbide. The balls for two of the pads are spaced at 120° and are mounted rigidly to the frame, while the ball on the third pad is flexibly mounted on a diaphragm with a nominal spring rate of 4000 pounds per inch (7000 N/cm) and placed 120° from the other two pivots. The diaphragm allows the journal bearing to accomodate some differential thermal growth with sufficient journal-bearing load to maintain stability. The compressor side of the thrust-bearing stator employs the Rayleigh design with eight self-acting stepped pads with argon gas for lubrication. Four orifices are provided on alternate pads for external pres-

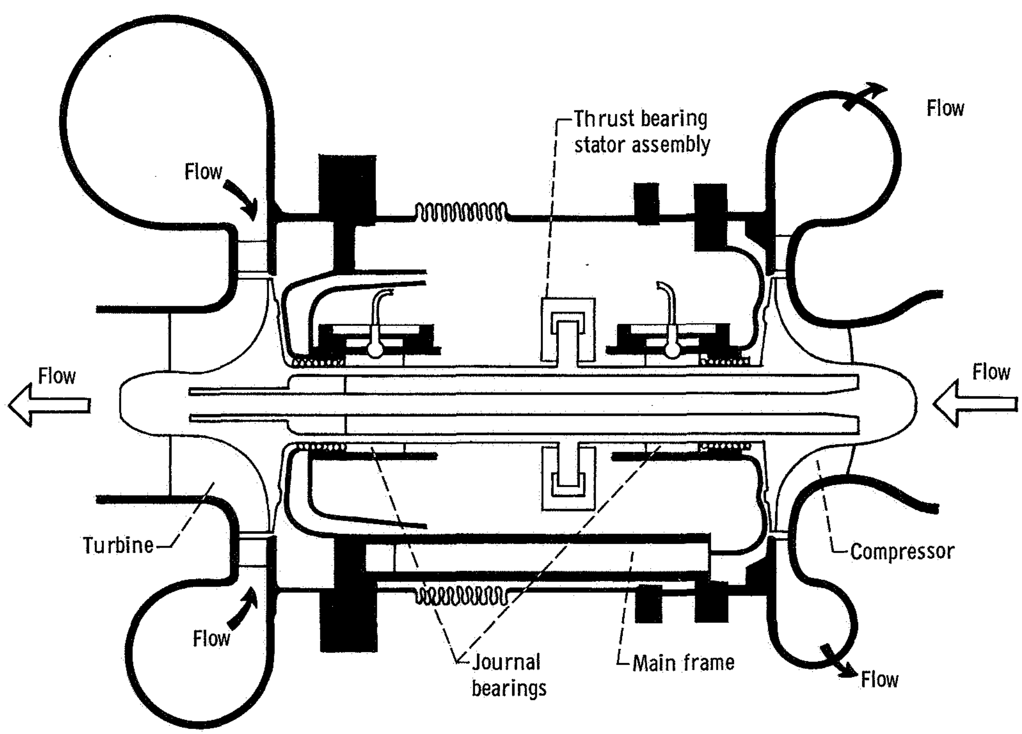


Figure 1. - Turbocompressor package schematic.

surization to maintain lubrication during start and shutdown. The turbine side of the thrust bearing stator is a plain externally pressurized bearing with six orifices. The two thrust stators are rigidly mounted together. The stator assembly is mounted in a gimbal assembly which allows the stators freedom to follow any excursions that the thrust runner may go through as well as changes in mount geometry due to thermal growths.

The turbocompressor was designed to operate at 38 500 rpm with a flow rate of 0.611 pound per second (0.277 kg/sec). The design compressor and turbine total pressure ratios are 2.30 and 1.56, respectively. A more complete description of the turbocompressor may be found in reference 2.

APPARATUS

A schematic layout and a photograph of the test loop used in this investigation are given in figure 2. The test loop as shown has the essential components of an unrecuperated Brayton cycle-space power system with provision for a gas-injection starting system, an inventory-control system, a compressor bypass for emergency shutdown, and a position-control valve to simulate a turboalternator. A photograph of the turbocompressor installation in the loop is shown in figure 3.

Heat Source

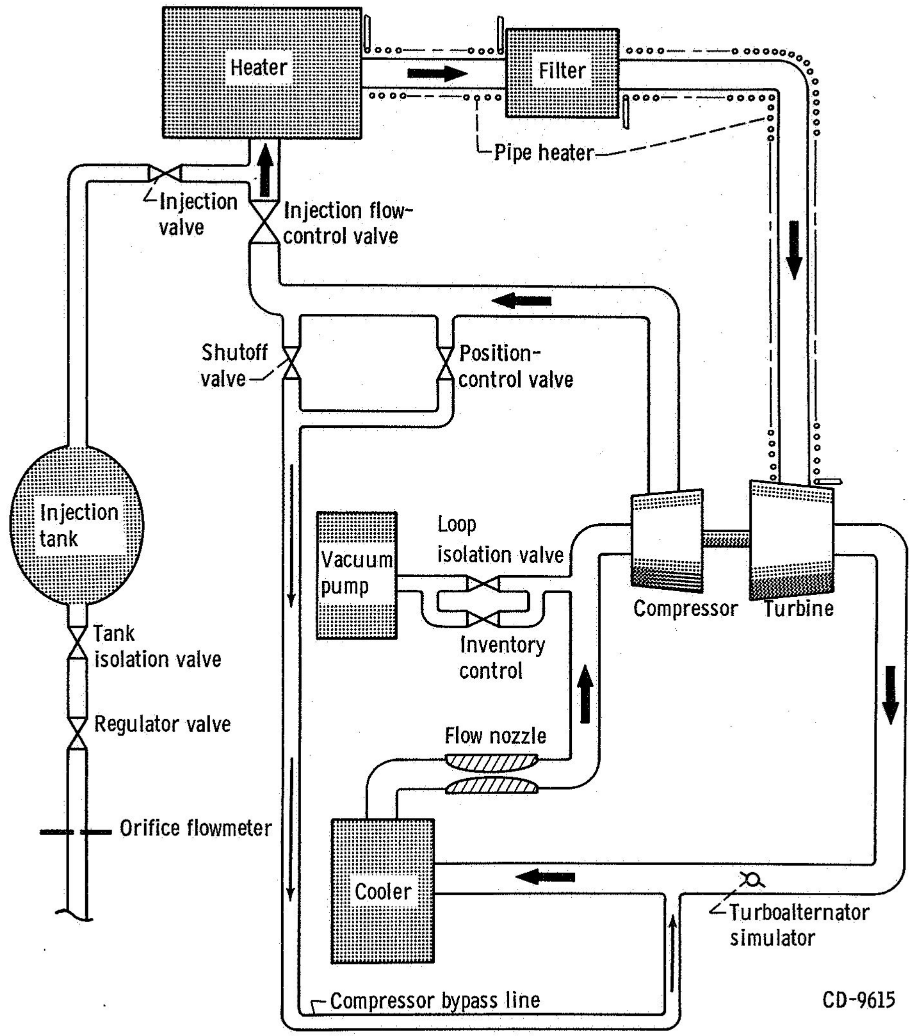
The heat source is simulated with an electric heater which has a 100-kilowatt direct-resistance heating element and has a maximum outlet-temperature capability of 1960° R (1090° K). Instrumentation and controls are provided to enable preheating the heater elements to any desired temperature up to 2160° R (1200° K).

Filter

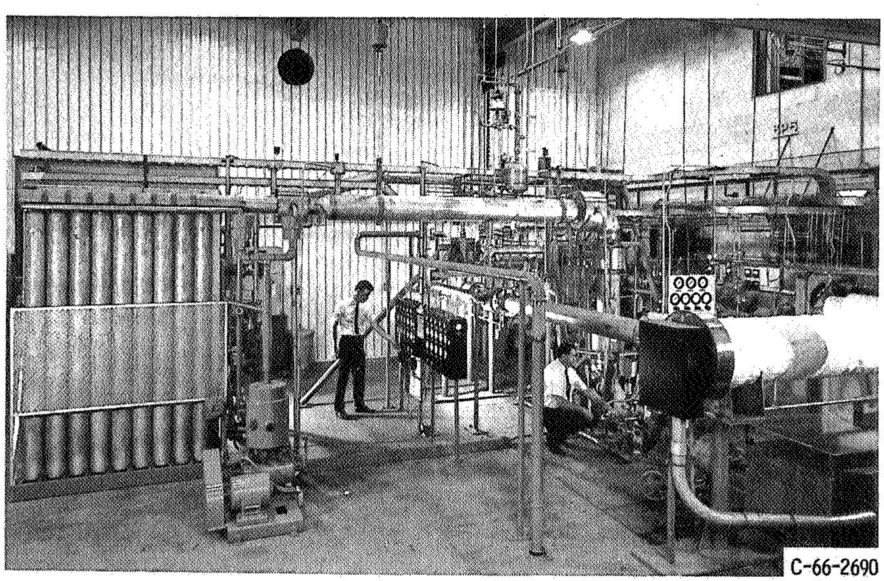
A filter was installed as a precautionary measure to minimize the risk of dirt and ceramic particles from the heater working their way into the bearing and causing a bearing failure. It has an element rating of 2 microns nominal and 10 microns absolute.

Heat Sink

The radiator is simulated with a cooler that has two sections. One section has water



(a) Schematic.



(b) Photograph.

Figure 2. - Test loop.

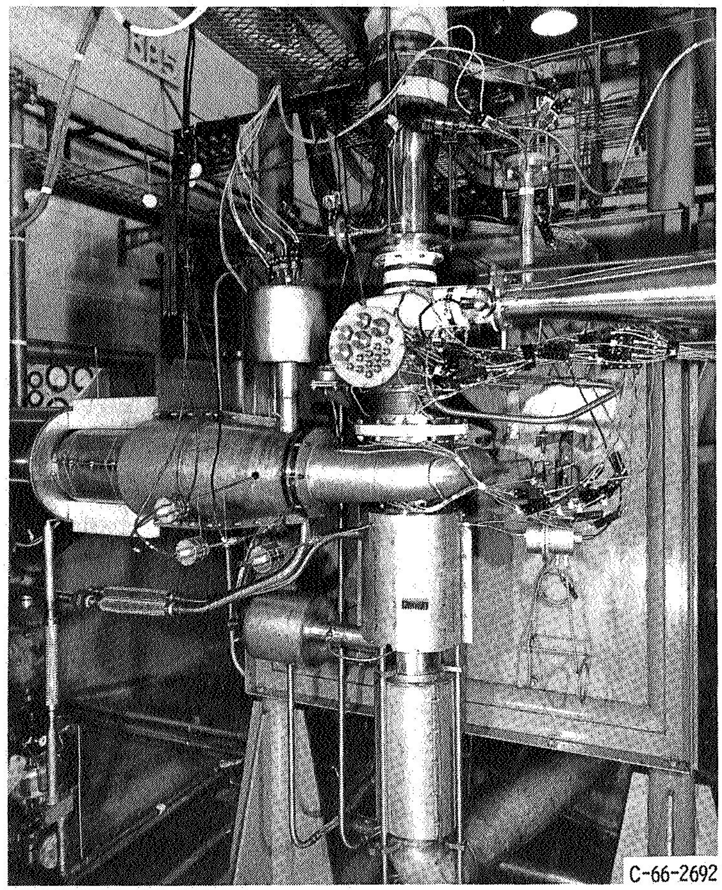


Figure 3. - Turbocompressor installation.

supplied at approximately 540° R (300° K) and the other section is cooled with refrigerator liquid. The cooler has a minimum outlet-gas temperature capability of 460° R (273° K). The temperature of the liquid-cooled section is controlled by mixing the refrigerated liquid with the warm liquid from cooler.

Gas Injection Starting System

Two modes of injection may be selected. One mode is to fill the 350-cubic-inch (5740-cu-cm) injection tank to the desired pressure and then to shut off the isolation valve. The injection valve is then opened to a preselected position and the gas trapped in the injection tank flows into the system. This mode of injection is characterized by a very sharp peak flow rate with a rapid dropoff as the pressure in the tank decays to system pressure. This mode of injection will herein be referred to as a bottle start. The other mode of injection is to leave the isolation valve open and then set the regulator valve to maintain the desired pressure at the inlet to the injection valve. When the injection valve is opened to the preset position, a nominally constant injection flow rate is obtained. This injection sequence is referred to as a constant-flow start. The orifice flowmeter is used to measure the weight flow during a constant flow start. The injection valve is closed by a pressure switch which senses pressure at the compressor inlet.

Trace heaters were installed on the piping between the heater outlet and the turbine inlet because the large mass of metal in the pipes acts as a heat sink during start. The pipe heaters can preheat the pipes to 1260° R (700° K).

The injection flow-control valve (see fig. 2(a)) is a butterfly valve which is closed during injection to direct the injected flow through the heater and turbine. The injection flow-control valve may be opened by the same pressure switch that closes the injection valve, or it may be opened by another pressure switch also located at the compressor inlet. With the second switch, the injection flow-control valve may be opened independently of injection cutoff.

The position-control valve in the compressor-bypass loop may be used to suppress surge during the start period. It may be opened to any desired position between open and closed, and it is closed by the same pressure switch that cuts off injection flow.

The shutoff valve, also in the compressor-bypass loop, is used for an emergency shutdown. It is normally closed and is opened if the speed of the turbine compressor exceeds 40 000 rpm.

The turboalternator simulator is a position control butterfly valve and is used to create the pressure drop designed for the turboalternator.

INSTRUMENTATION

The instrumentation used to measure and record the performance of the turbocompressor is as follows.

Chromel-Alumel thermocouples were used to measure all temperatures in the test. Bare-spike total-temperature rakes were located approximately 2 inches (5 cm) upstream of the turbine and compressor-inlet scroll flanges and 2 inches (5 cm) downstream of the compressor-outlet scroll flange. The bare-spike total-temperature rake for the turbine outlet was located approximately 10 inches (25 cm) downstream of the turbine-exit scroll flange. These thermocouples have a calculated time response of 1.59 seconds for 63 percent of a 1000° R (556° K) step change in temperature level.

Strain-gage pressure transducers were used to measure all pressures. Static-pressure taps and total pressure probes were located 2 inches (5 cm) upstream of the turbine and compressor-inlet scroll flanges and 2 inches (5 cm) downstream of the outlet scroll flanges. The pressure measuring system, including the connecting tubing, has a calculated natural frequency of 45 hertz.

The turbine and compressor journal-bearing loads were measured using bonded-wire strain gages. These gages were welded to the bearing diaphragm in a four-arm-bridge configuration.

The turbocompressor speed was measured with high-temperature magnetic pickups. Six magnetic blocks were imbedded in the shaft for this purpose.

The compressor weight flow was measured by a venturi flowmeter upstream of the compressor inlet. The location of this flowmeter can be seen in figure 2(a). The turbine weight flow was calculated by using the static-pressure at the turbine stator exit and the total-pressure and temperature at the turbine scroll inlet. The injection weight flow was measured by an orifice flowmeter. The location of this flowmeter can be seen in figure 2(a) also.

Shaft motions were measured using capacitive proximity instrumentation, consisting of noncontact capacitance probes and high-frequency oscillator-amplifiers. Six probes were used to measure the shaft motions. Two probes were located near the compressor journal bearing 90° apart and in a radial plane. Two probes were located near the turbine journal bearing. The remaining two probes were mounted on the thrust stators directly opposite each other.

The signals from all capacitance instrumentation were continually monitored on dual beam oscilloscopes and at the same time recorded along with the speed on an FM magnetic tape recorder.

At the start of the injection of argon into the system, temperatures, pressures, speed, and thrust-bearing film thickness were recorded by a high-speed automatic data recorder. This recorder has a capability of recording 30 000 data points per second. For this test, a recording speed of 2500 data points per second was fast enough for the injection transients. Fifty system measurements were used for the transient study. At a recording speed of 2500 data points per second, each system measurement was recorded 50 times per second. The data recorded by the high-speed automatic recorder were processed through a digital computer.

PROCEDURE

Injection Preparation

Prior to injection, the amount the injection valve is to be opened was preselected. In addition, the injection cutoff point and the point for opening the injection flow-control valve were selected, the turboalternator simulator valve was opened, the desired position of the compressor bypass valve was selected, and the heater and cooler were set to the desired temperatures. The sequence used in making an injection start was to place external pressurization on all the gas bearings and then to evacuate the loop to approximately 1 psia (0.7 N/cm^2). Under these conditions the rotor of the turbocompressor would spin to a speed of 1300 rpm in the negative direction. The negative rotation was caused by the external pressurizing gas bleeding out preferentially between the leading edge of the shoes and the shaft and dragging the shaft around. The contractor recommended that the rotor not be subjected to a rapid acceleration when the shaft is rotating in a negative direction. In order to obtain a positive rotation, a small amount of injection flow (which is pumped out immediately) was used to spin the shaft in the positive direction. Then as the effect of external bearing pressurization reduced the positive speed to zero and the system is again evacuated, an injection start sequence would be initiated.

Bottle and Constant-Flow Injections

When using the bottle injection method, the injection tank was filled to the desired pressure, and the tank isolation valve was closed. For the case of the constant-flow start, the regulator valve was set to the desired pressure, and the tank isolation valve was left open. With these conditions satisfied, the loop isolation valve was closed, and injection was made with zero rotor speed. Transient data recordings of all pressures, temperatures, speeds, rotor radial and axial motions, and journal-bearing loads were made.

Injections were made into the loop starting with low flow rates and low turbine-inlet temperature. These were gradually increased until a self-sustaining speed was reached.

RESULTS AND DISCUSSION

The objective of this investigation was to subject the delivered turbocompressor to such transients as may be encountered in a space start of a two-shaft Brayton cycle space-power system and to determine the effect of system flow controls, as discussed in the APPARATUS section on start and shutdown. Two modes of injection starts were tried and will be discussed in separate sections. Also, to be discussed is an accidental closing of the turboalternator simulator valve and its effect on the turbocompressor operation. The use a compressor bypass for emergency shutdown, the variation in journal bearing loads as indicated by strain gages, and radial and axial shaft motions observed will also be discussed.

Bottle Injection Start

As discussed in the APPARATUS section, a bottle-injection start is one in which the inventory of gas to be injected is trapped in a fixed volume or bottle and is released into the system from the bottle. To make an injection from this bottle, an injection valve is opened to a preselected fixed position. With a supercritical pressure ratio across the fixed-position valve, the injection flow rate varies with the pressure in the bottle. Thus, the injection flow rate would have a characteristic sharp rise to a peak as the injection valve opens, followed by an exponential variation as the pressure decays within the bottle.

Initial injections were made with flow rates of 0.2 pound per second (0.09 kg/sec) and a turbine-inlet temperature of 540° R (300° K). The injection flow rate and turbine-inlet temperature were then gradually increased to 2.2 pounds per second (1.0 kg/sec) and 1260° R (700° K). At these conditions, the maximum speed obtained was 7800 rpm; however, this was not enough to achieve self-sustaining operation.

The transient data recordings of flow and pressures indicated that the compressor-outlet pressure and compressor weight flow rose to a peak during the injection period. The peak indicated that the compressor flow was about 22 percent of the peak turbine flow. Subsequent flow calibrations of the injection flow-control valve indicated that it leaked in the fully closed position and that flow was going into the compressor in a negative direction. Because the compressor has a much greater change in tangential momentum with negative flow than when flow is in the normal (or positive) direction, it acts as an air brake during the start period. This air brake action caused the compressor to absorb more torque than it would normally. The leakage was also equalizing the pressure around the loop which decreased the pressure ratio attainable across the turbine.

The faulty valve was removed, and the butterfly was repositioned. A calibration before reinstallation showed the remaining leakage flow area to be about 6 percent of turbine nozzle flow area.

For the next series of tests, the peak injection flow rate and turbine-inlet temperature was increased to 3.2 pounds per second (1.45 kg/sec) and 1100° R (611° K), respectively. A maximum change in speed of 12 500 rpm was obtained for this condition. Other tests showed that this speed is sufficient to start the turbocompressor if the turbine-inlet temperature is 1260° R (700° K) or higher. The high-injection flow rate however, caused a high-pressure ratio across the turbine which, in turn, imposed a high-thrust load on the thrust bearings. The minimum thrust-bearing film thickness was 0.0009 inch (0.00023 cm), which corresponds to a thrust load of about 60 pounds (27 N). Also, at this high-injection flow rate, shaft motion monitors showed evidence of mild compressor surge when the compressor-bypass valve was closed.

The amount of gas injected was about 2.5 pounds (1.1 kg) to achieve start conditions at 1260° R (700° K) as compared with 1.3 pounds (0.6 kg) of gas which is the system inventory at design conditions. The amount of gas injected to achieve a start would decrease as the turbine-inlet temperature is increased.

Constant-Flow Injection Start

Because of the high-thrust load imposed on the thrust bearings by the bottle-injection method, constant-flow injections with characteristically less thrust shock were made. Constant-flow injection is accomplished when the injection flow rate is maintained nominally constant during the injection period. Constant-flow injections were started at about 0.3 pound per second (0.1 kg/sec) and turbine-inlet temperature of 530° R (294° K) and increased in steps to about 0.56 pound per second (0.25 kg/sec) and 1100° R (611° K). With these injection conditions and starting from zero speed, a maximum speed of 13 200 rpm was achieved with a total argon injection of about 2.5 pounds (1.1 kg). The minimum thrust-bearing film thickness was 0.00111 inch (0.00282 cm), which corresponds to a thrust load of 40 pounds (180 N). When the injection flow rate was increased to 0.78 pound per second (0.35 kg/sec) a maximum speed of 17 000 rpm was achieved with a turbine-inlet temperature of 1100° R (611° K). The total flow injected into the system was about 2.4 pounds (1.1 kg) of argon. The minimum thrust-bearing film thickness for the higher flow rate was 0.00102 inch (0.00259 cm) which corresponds to a thrust load of 45 pounds (200 N).

Self-Sustaining Starts

Figures 4 to 12 show typical transient recordings of speed, temperature, pressure, and flow rate as functions of time for a successful constant-flow injection start. These recordings, with the exception of injection flow rate, are also representative of bottle starts.

Turbocompressor speed is plotted in figure 4. At about 1.9 seconds from time zero, injection was started. At 2.1 seconds, the speed rose sharply and reached a peak of 13 200 rpm at 5.5 seconds and then gradually dropped off to a minimum of 11 900 rpm at about 30 seconds. Above 30 seconds, the speed gradually increased again.

Injection weight flow, turbine weight flow, and compressor weight flow are shown in figures 5 to 7. At about 1.9 seconds, injection was started with the opening of the injection valve. The injection weight flow increased while the valve was opening. At about 3.5 seconds, the injection valve reached its preselected position and the weight flow was nominally constant at 0.56 pound per second (0.25 kg/sec) for about a 3-second period. At the start of injection cutoff at 4.8 seconds, the injection valve started to close and injection flow dropped to zero at about 6 seconds.

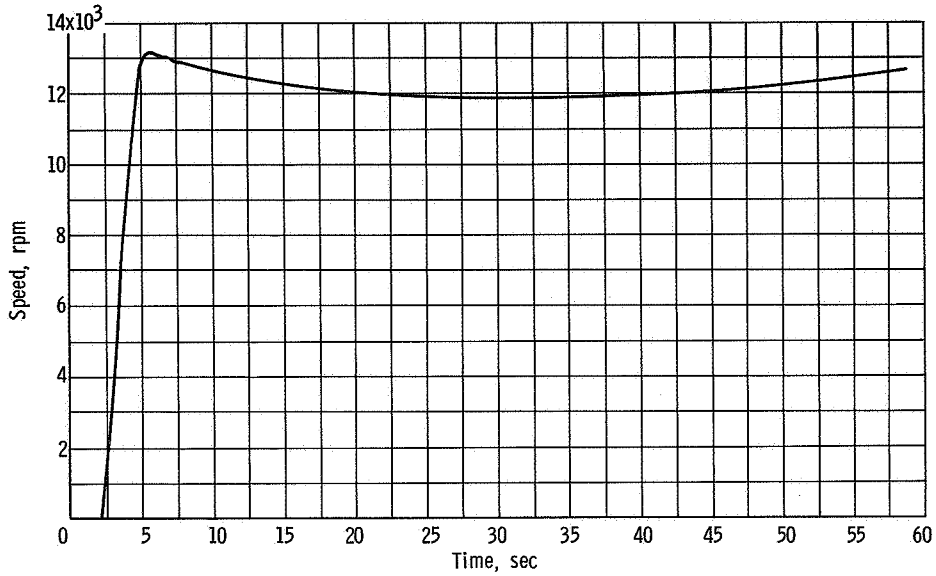


Figure 4. - Speed as function of time for constant-flow injection.

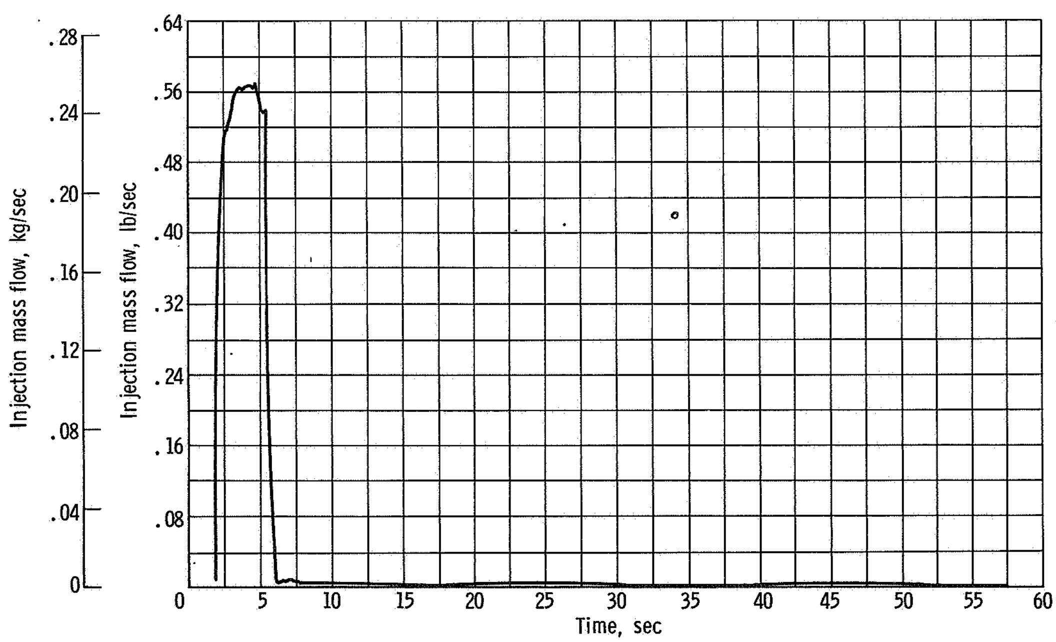


Figure 5. - Injection mass flow as function of time for constant-flow injection.

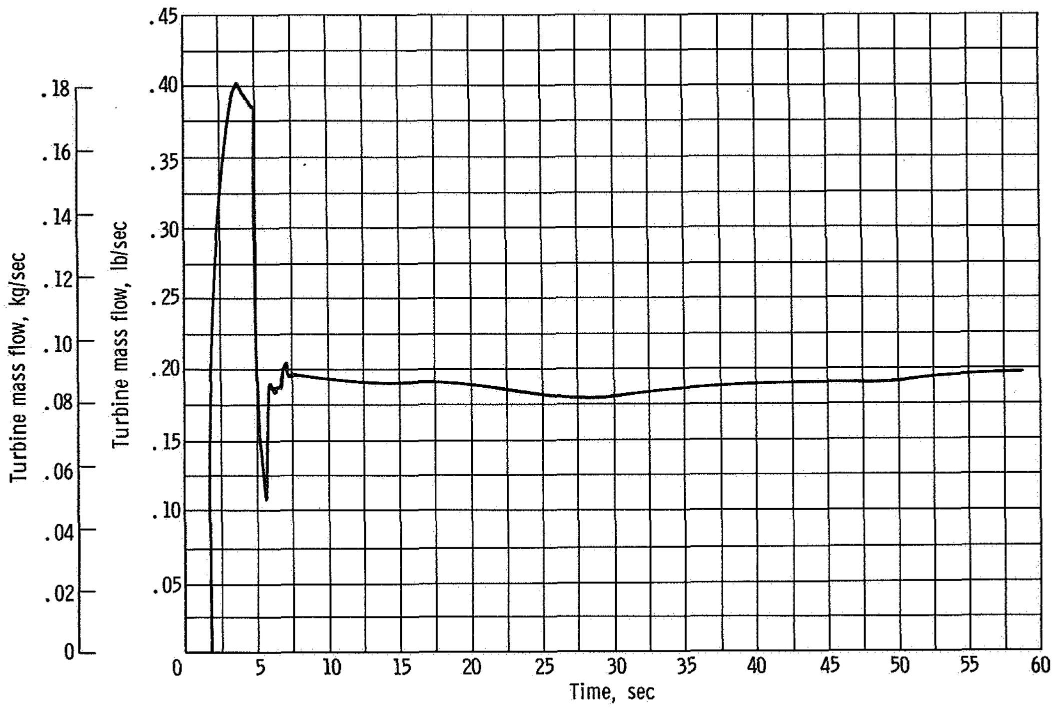


Figure 6. - Turbine mass flow as function of time for constant-flow injection.

During the injection period, the turbine weight flow (as shown in fig. 6) rose to a peak of 0.40 pound per second (0.17 kg/sec) at 3.8 seconds, gradually dropped off to 0.39 pound per second (0.18 kg/sec) at 4.9 seconds, and then dropped sharply to about 0.11 pound per second (0.050 kg/sec) as injection was cut off. At approximately the same time that injection was cut off, the injection flow-control valve was also opened. As the flow was circulated around the loop by the compressor, the turbine flow rose sharply from the minimum of 0.11 pound per second (0.050 kg/sec) at 5.8 seconds to about 0.19 pound per second (0.090 kg/sec). The gradual drop in flow rate after 7.5 seconds was caused by the gradual drop in turbocompressor speed (fig. 4). Above 30 seconds, the weight flow gradually increased as the speed increased.

The compressor weight flow (fig. 7) rose to a peak of 0.03 pound per second (0.13 kg/sec) at 3.5 seconds. This peak was about 7 percent of turbine flow. The flow rate then fell off to about zero at 4 seconds. This evidently was caused by the compressor rotation stopping the injection flow-control valve leakage from passing backward through the compressor. Then, the flow rose sharply to 0.09 pound per second (0.054 kg/sec) at 5.0 seconds. This rise was probably caused by the compressor starting to pump flow into the volume between the compressor and the injection flow-control valve. Another sharp drop and cycling in the flow rate followed before it rose to about 0.19 pound per second (0.10 kg/sec) at 5.8 seconds. The erratic flow behavior between 5.0 and 5.8 seconds was probably caused by an interaction between the opening of the in-

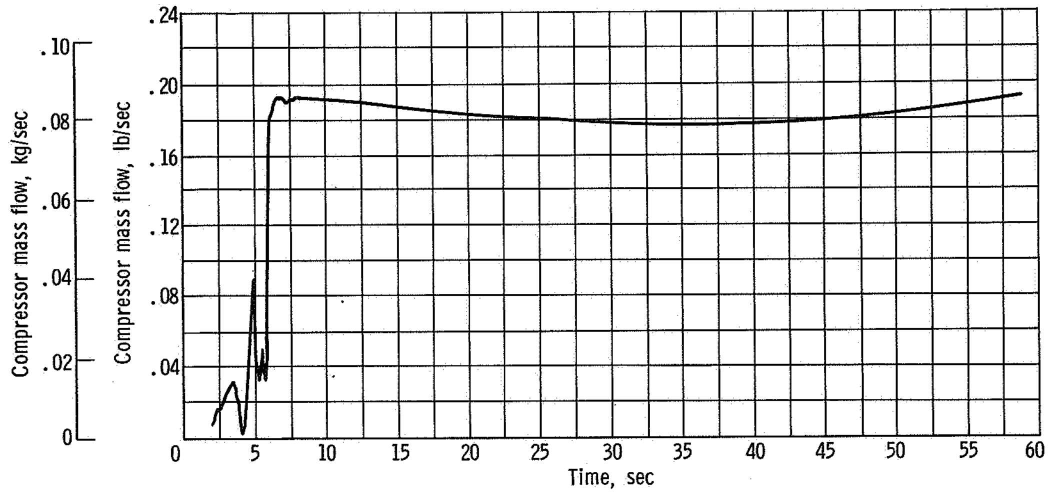


Figure 7. - Compressor mass flow as function of time for constant-flow injection.

jection flow-control valve and the compressor. It is possible that a slightly early opening of the injection flow-control valve could bleed the high-pressure argon from the volume between the injection flow-control valve and the turbine toward the compressor. The flow bleeding toward the compressor would tend to retard compressor flow, and possibly surge the compressor, and thus result in the erratic flow behavior. The gradual decrease in flow rate to about 0.176 pound per second (0.093 kg/sec) from 1.9 pound per second (0.10 kg/sec), which is followed by gradual rise, is the result of the speed variation discussed previously.

Inlet and outlet total temperature for the turbine and compressor are shown in figure 8. At the start of injection (1.9 sec), the turbine-inlet temperature was 790° R (440° K), which rapidly rose to 1110° R (617° K) at injection cutoff (6 sec). At injection cutoff there was insufficient temperature to sustain operation of the turbocompressor; therefore there was a gradual drop in speed (fig. 4). Flow was being pumped around the loop, however, and the turbine inlet temperature continued to rise. At about 30 seconds after time zero, the work output of the turbine equalled the work required by the compressor and the turbocompressor became self-sustaining. Further increase in temperature above 1260° R (700° K) resulted in an acceleration of the turbocompressor. Turbine-outlet temperature had a trend similar to that of the inlet temperature.

The compressor-inlet and outlet temperatures were 540° R (300° K) at the start of injection and rose to peaks of 600° R (333° K) and 590° R (328° K), respectively. The inlet temperature rose to a 10° R (5.6° K) higher peak because of the leakage flow going backwards through the compressor and the energy input to the flow by the compressor wheel. After proper flow was established by the compressor, the compressor-inlet temperature was 540° R (300° K), and the outlet temperature was 560° R (311° K). It is also noted that there was a slight variation in outlet temperature which was caused by the drop

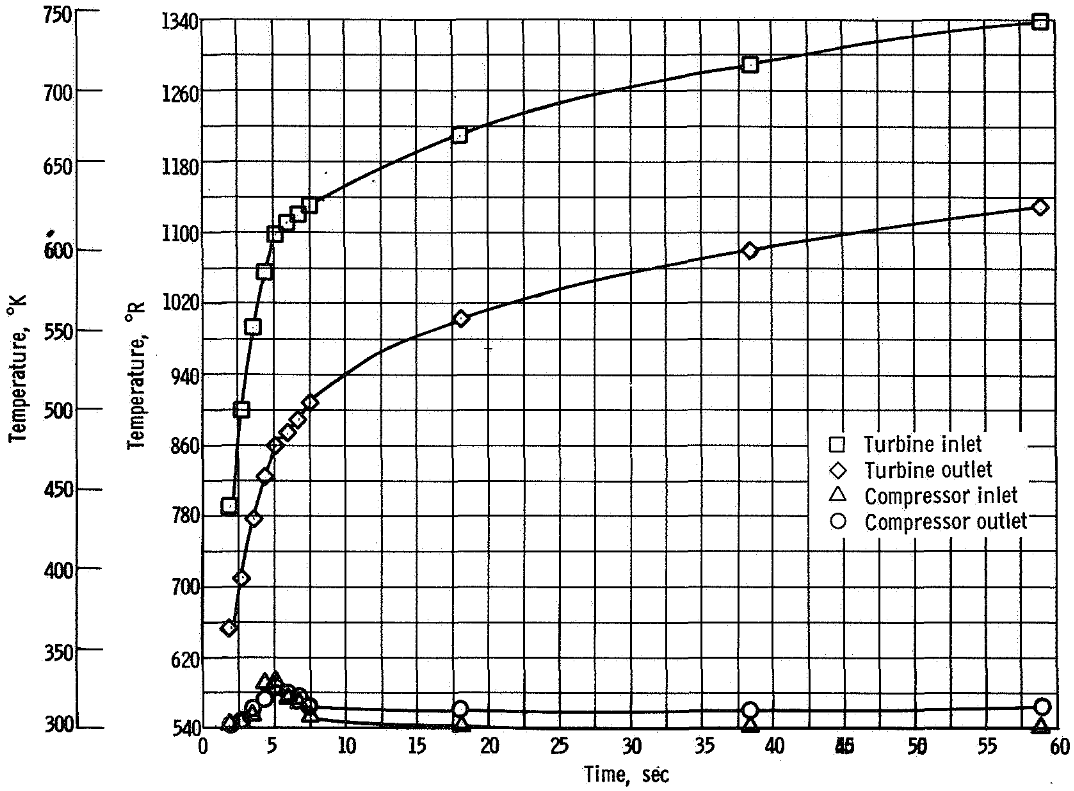


Figure 8. - Inlet and outlet temperatures as functions of time for turbocompressor during constant-flow injection.

in speed when the turbocompressor was not quite self-sustaining.

The inlet and outlet total pressures for the turbine and compressor are plotted in figure 9. It can be seen that, at the start of injection, the loop was evacuated to 1.25 psia (0.852 N/cm^2 abs). As injection was started, the pressure in the loop rose rapidly with the turbine-inlet pressure rising more rapidly than the other pressures. At 4.9 seconds, the compressor outlet pressure began to rise very sharply. At 5.1 seconds, the compressor outlet pressure crossed the turbine-inlet pressure cruve. From 5 to 5.1 seconds, the turbine-inlet pressure was about constant. This indicates that injection was being cut off and that the injection flow-control valve was opening. At completion of the injection (6 sec), the compressor outlet pressure peaked at 8.75 psia (6.02 N/cm^2 abs), and the turbine inlet peaked at 8.60 psia (5.92 N/cm^2 abs). The turbine-outlet and compressor-inlet pressure peaked at 8.0 psia (5.02 N/cm^2 abs) and 8.1 psia (5.08 N/cm^2 abs), respectively. After the peak pressure there was a slight drop in pressure due to an adjustment of the inventory by the inventory control. Then, the general level of pressure rose in the loop as more heat was put into the gas by the heater. The bearing cooling flows were then turned on. At about 50 seconds, the compressor-outlet and turbine-inlet pressure started a gradual rise. This rise was caused by the gradual rise

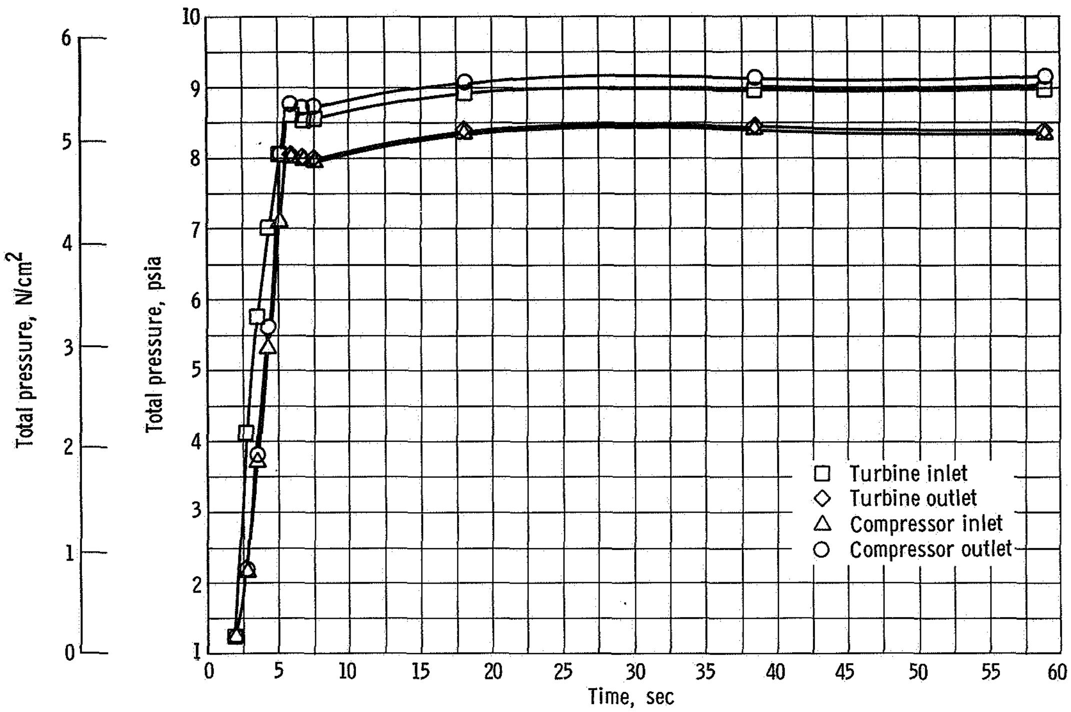


Figure 9. - Inlet and outlet pressures as functions of time for turbocompressor during constant-flow injection.

in turbocompressor speed after it became self-sustaining. At 59 seconds, the pressure at the turbine outlet and compressor inlet appeared to be approaching a constant value. This indicates that the inventory controller had the system inventory under control.

Turbine and compressor aerodynamic speed (shaft speed divided by the quantity: design speed multiplied by the square root of the ratio of component-inlet temperature to design component-inlet temperature) are plotted in figures 10 and 11. It can be seen that the peak turbine aerodynamic speed was 45.1 percent and that it dropped to 38.2 percent at 35 seconds. The minimum did not occur at the minimum shaft speed because the temperature was increasing faster than the speed. The compressor peak aerodynamic speed was 32.8 percent, which dropped to 29 percent at about 30 seconds. This trend was similar to the trend of mechanical speed (fig. 4). The difference in aerodynamic speed between the turbine and compressor is due to the low turbine inlet temperature (far below the design value of 1950° R (1082° K)), while the compressor-inlet temperature was very near the design value of 536° R (298° K).

The variation in turbine blade-jet speed ratio is presented in figure 12. It can be seen that, after injection, the blade-jet speed ratio rose rapidly to a sharp peak of 0.89 at injection cut off. Once flow around the loop was established by the compressor, the turbine was operating at a blade-jet speed ratio of 0.75 and gradually decreased as the speed decreased. After 30 seconds, the blade-jet speed ratio continued to decrease be-

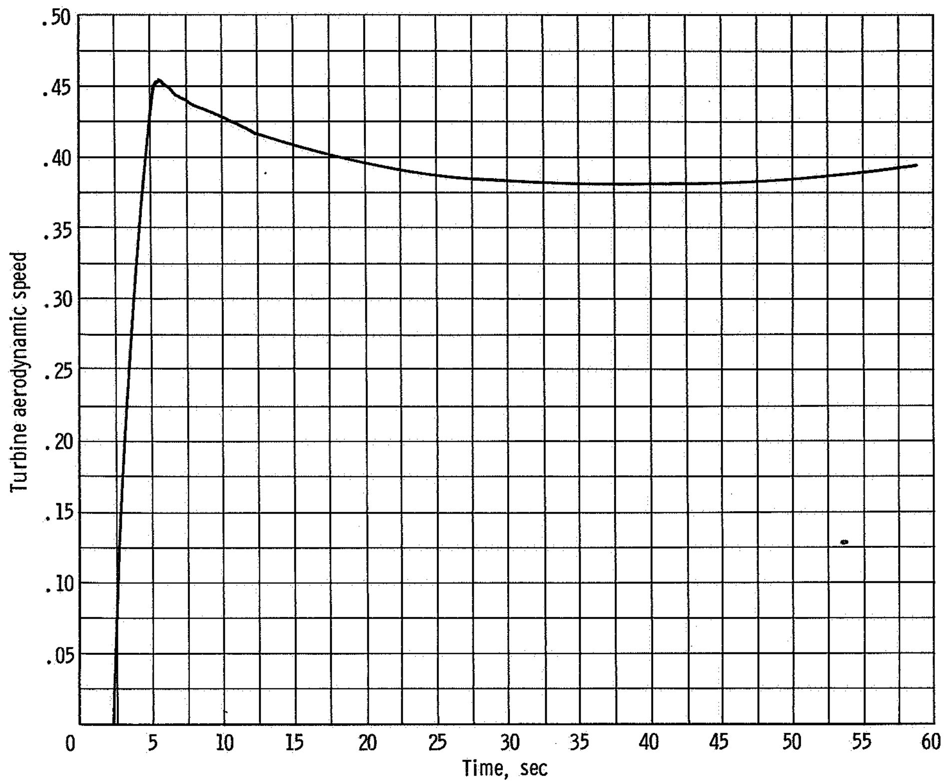


Figure 10. - Turbine aerodynamic speed as function of time for constant-flow injection.

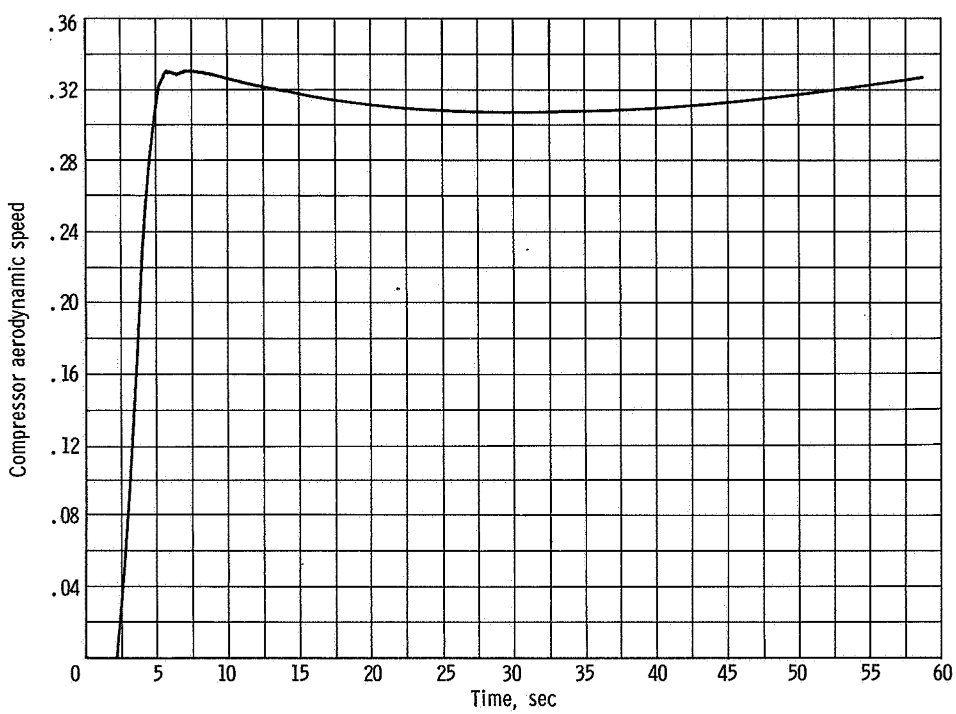


Figure 11. - Compressor aerodynamic speed as function of time for constant-flow injection.

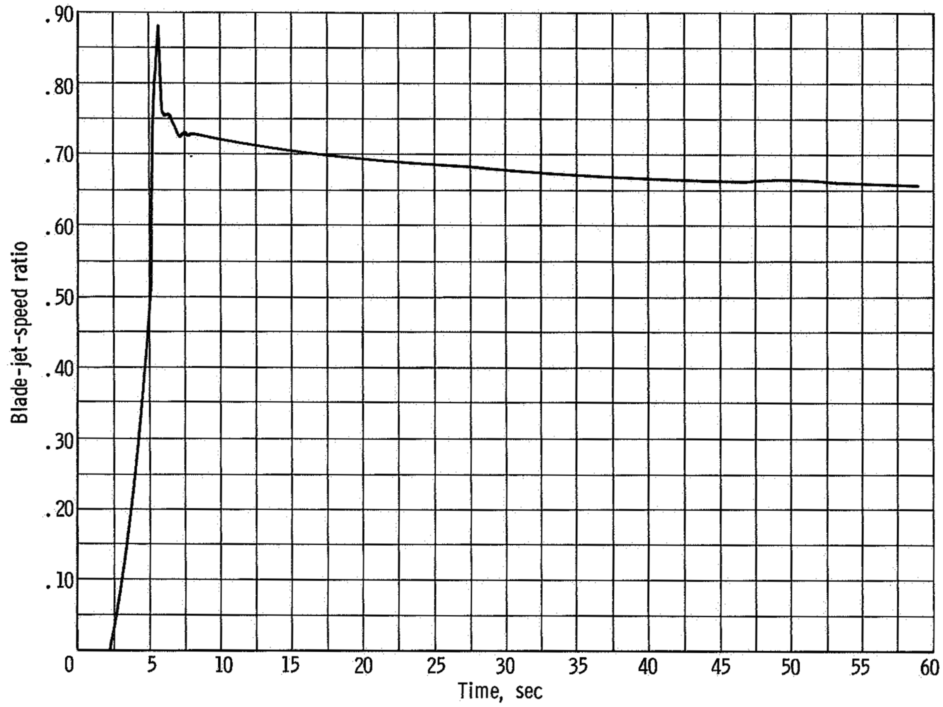


Figure 12. - Turbine blade-jet speed ratio for constant-flow injection.

cause the jet speed made available by increasing compressor pressure ratio and gas temperature, was rising faster than rotor speed. This curve shows, however, that, once flow is established by the compressor, the turbine operates near a blade-jet speed ratio of 0.7.

Effect of Compressor Surge

On one of the runs, the controller for the turboalternator simulator valve malfunctioned. The result was that the turboalternator simulator valve slammed shut, stopped flow around the loop abruptly, and sent the compressor into surge. Pressure pulsations resulting from the surge caused large vibrations in the turbocompressor rotor. The maximum radial excursions of the rotor were observed to be about 0.00083 inch (0.00211 cm) on the turbine end and 0.00079 inch (0.00201 cm) on the compressor end. The maximum axial excursions of the rotor were 0.00074 inch (0.00188 cm). It was fortunate that the turbocompressor was operating at 28 000 rpm or about 73 percent of design speed at the time of the accident. Surge at design speed would have excited much larger motions because the pressure pulsations would have been greater.

It was fortunate, also, that the gas bearings were still operating with external pressurization since the load capacity is much greater and since insufficient self-acting pressure would have been generated below 20 000 rpm. It was further observed that below 15 000 rpm the rotor excursions caused by surge were very mild.

The maximum rotor excursion due to surge can be seen in figure 13 which is a photograph of an oscilloscope trace of the turbine-end and compressor-end shaft orbits indicated by radial probes. Rotor excursions resulting from compressor surge occurred at various speeds during the rapid shutdown but the rotor excursions at 27 000 rpm were the most severe. At this speed the normal orbit diameter associated with mass eccentricity

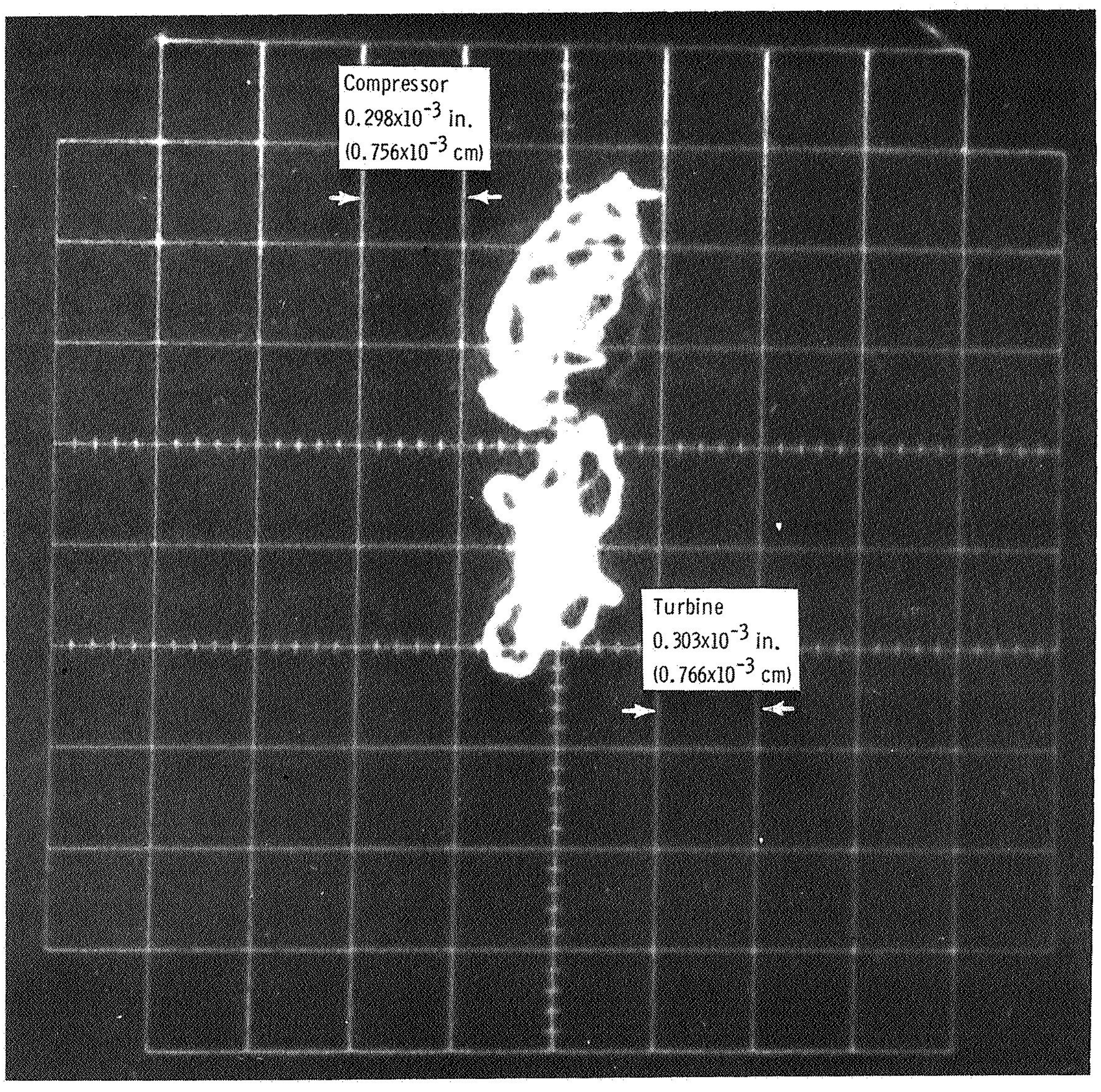


Figure 13. - Turbine and compressor orbits during surge (27 000 rpm).

for the compressor end is 0.000179 inch (0.000455 cm), while the turbine end is 0.00034 inch (0.000864 cm).

Turbocompressor Shutdown by Use of Compressor Bypass

The malfunctioning of the turboalternator simulator controller demonstrated the danger of closing that valve for emergency shutdown. An alternate method of emergency shutdown is to open the compressor bypass valve. This would move compressor operation toward maximum flow and increase compressor torque requirements, while the turbine pressure ratio would move toward 1.0 and, thus, quickly reduce turbine torque. The net torque would become negative and the turbocompressor would decelerate.

In order to determine how quickly an emergency shutdown could be accomplished by use of compressor bypass, an emergency shutdown was simulated and transient data recordings were made of speed and turbocompressor inlet and outlet pressure and temperature.

In the simulated emergency shutdown, the system conditions were the same as those conditions in the system when the turboalternator simulator valve malfunction occurred. The bypass valve was then opened and transient data recordings were made of speed, pressures, temperatures, and shaft motions as the turbocompressor slowed down in speed.

The transient data showed that this method of stopping the rotation of the turbocompressor was quick and safe. Figure 14 shows a curve of the speed from the time just be-

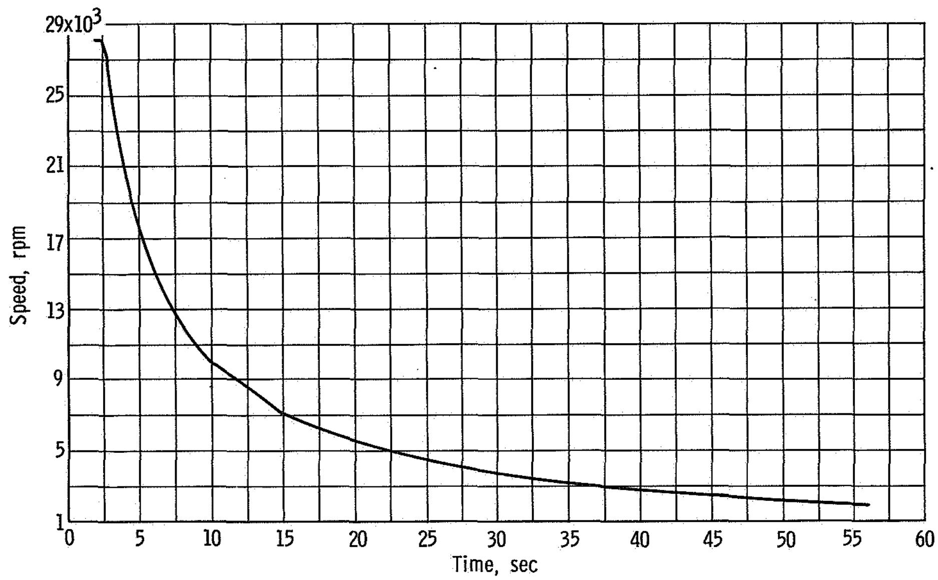


Figure 14. - Speed as function of time for simulated emergency shutdown using compressor bypass line.

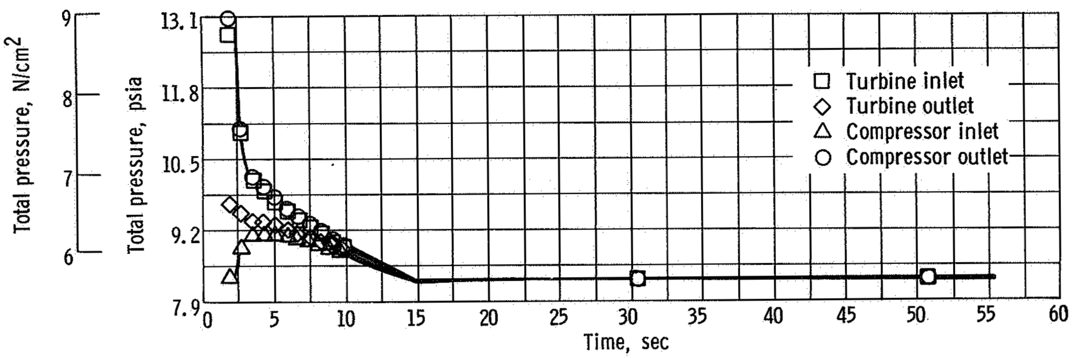


Figure 15. - Inlet and outlet pressure as function of time for simulated emergency shutdown using compressor bypass loop.

fore the bypass valve was opened until the time the speed settled out at a preselected speed of approximately 2000 rpm. The speed dropped from 28 000 to 15 000 rpm in 3.65 seconds and down to 7100 rpm in 12.5 seconds.

Figure 15 shows curves of total pressure for the turbine inlet and outlet and the compressor inlet and outlet. From these curves it can be seen that surge pressure pulsation did not occur. And after a period of approximately 12.5 seconds, all pressures were equal. The shaft orbit traces showed no noticeable vibrations.

Turbocompressor Shaft Motions

Relative radial motions between the rotor and the frame, and the relative axial motions between the thrust runner and stator are of vital importance to the operation of the turbocompressor. These motions were monitored and recorded during testing of the turbocompressor package. Figure 16 shows some typical radial motions as observed over the speed range investigated during a relatively slow shutdown.

Because the maximum peaks of the shaft motions vary in speed and magnitude from test to test, only the envelopes of these motions are of interest. These plots are shown in figures 17 and 18 for the turbine-end and compressor-end radial motions, respectively. The axial motions are shown in figure 19. All motions are for external pressurization only.

From figure 17 it can be seen that the turbine-end journal maximum excursions increased rapidly to a value of 0.00075 inch (0.0019 cm) at 5000 rpm. At this point the turbocompressor was at the first rigid-body critical speed until it reached 6000 rpm. Then the effects of the second rigid-body critical speed became prominant, and the excursion went up to 0.00155 inch (0.00390 cm) between 8000 and 9000 rpm. Once through this critical speed, the excursions dropped rapidly as the speed increase. Finally, at

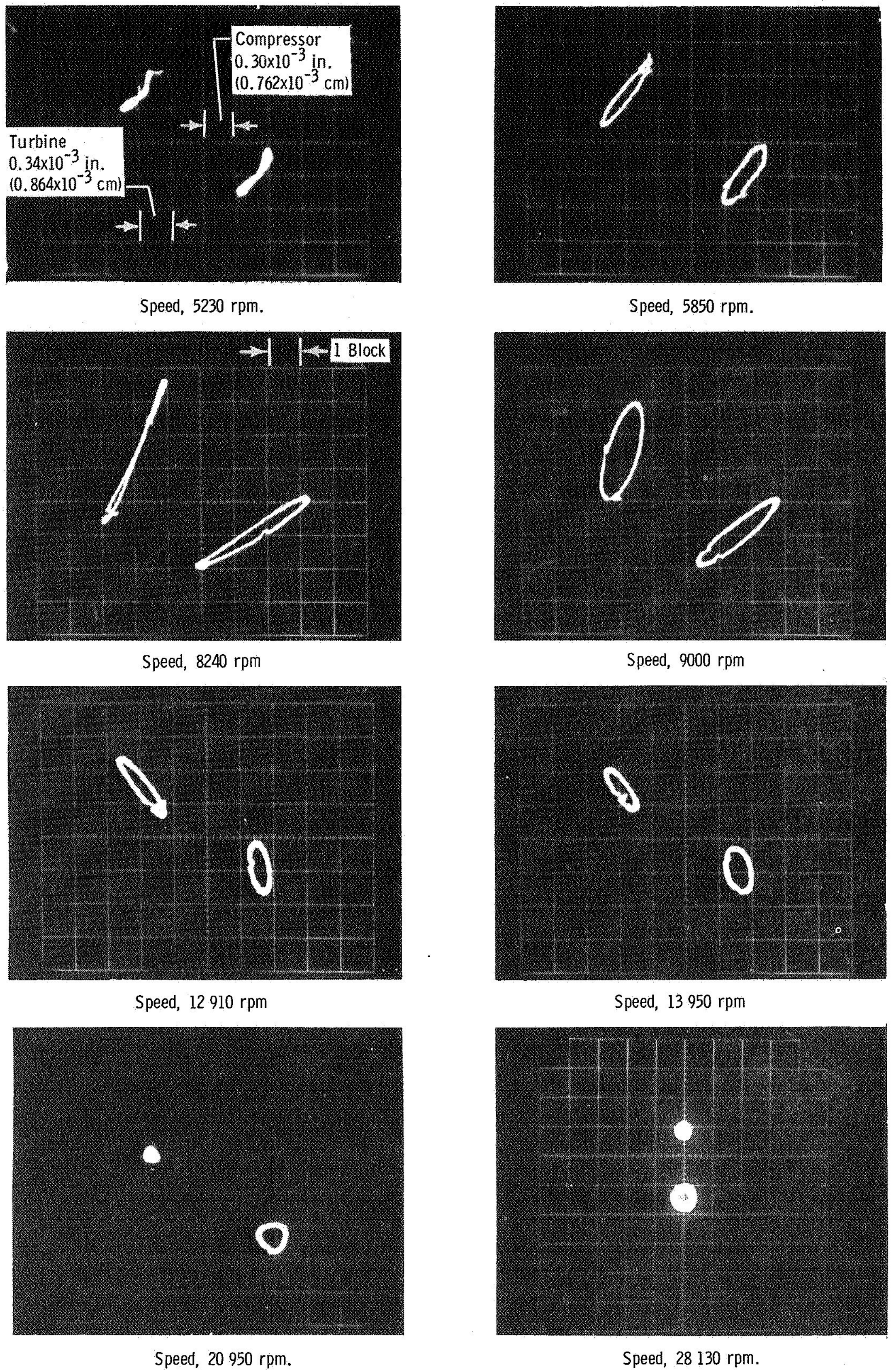


Figure 16. - Typical shaft orbits of turbo compressor.

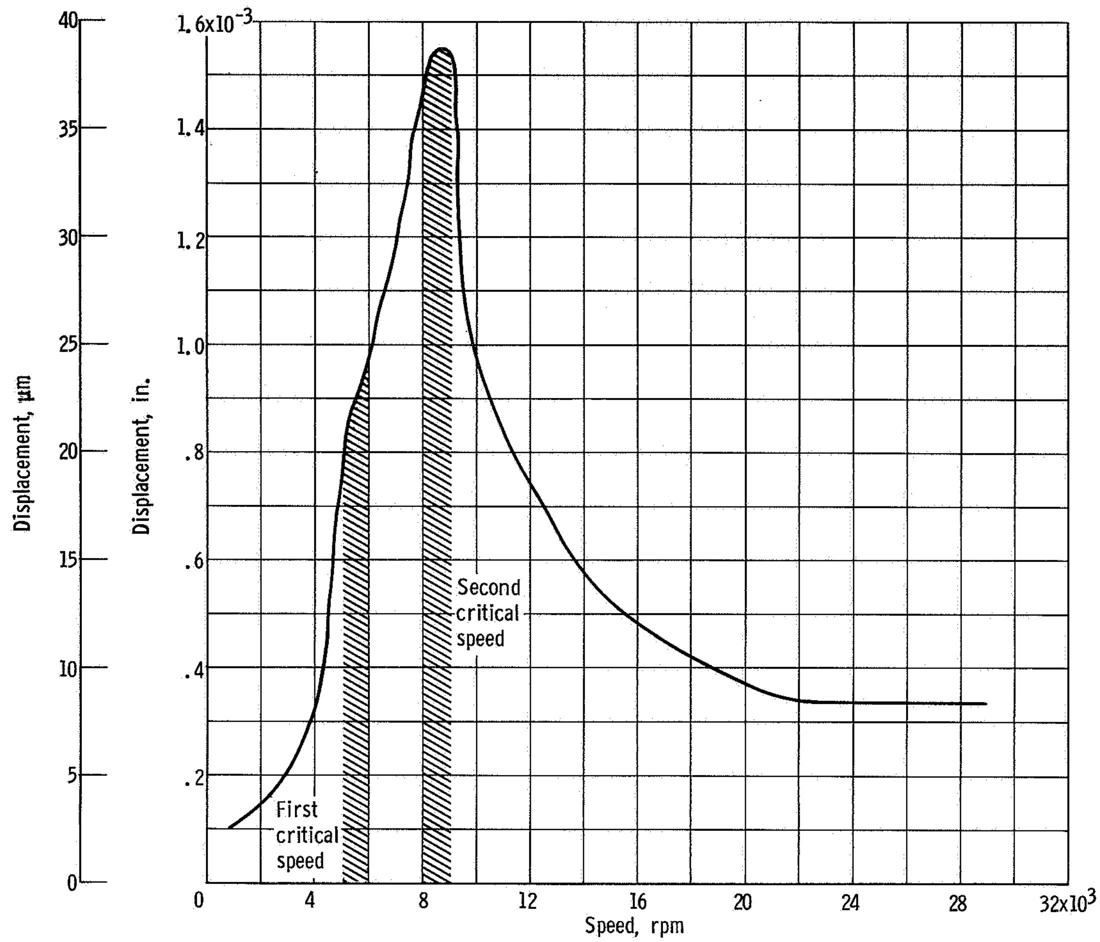


Figure 17. - Envelope of turbine-end journal radial displacement as function of speed for all tests.

22 000 rpm, the motions began to hold steady at 0.00034 inch (0.00086 cm) as the speed was increased to 29 000 rpm.

The maximum excursions of the compressor-end journal as shown in figure 18 indicate that the maximum shaft excursions increased as the speed was increased above 2000 rpm. The maximum displacement reached a peak of 0.00148 inch (0.00376 cm) at about 8000 rpm. As the speed is increased further, the excursions decreased to a value of 0.00020 inch (0.00050 cm) at 21 000 rpm. From this point, an increase in speed to 29 000 rpm had no effect on the maximum excursion of the compressor end journal.

Further examination of the phase relation between the motions of each end showed that the 5000- to 6000-rpm critical speed was a conical mode, while the 8000- to 9000-rpm critical speed was a cylindrical mode. These rigid-body critical speeds are shown as crosshatched areas in figures 17 and 18.

It should be noted at this point that the effect of rapid injection limits the maximum excursions to somewhat lower values than those shown on the plots. The plots were obtained from data taken during much slower shutdowns. Data taken during injections in-

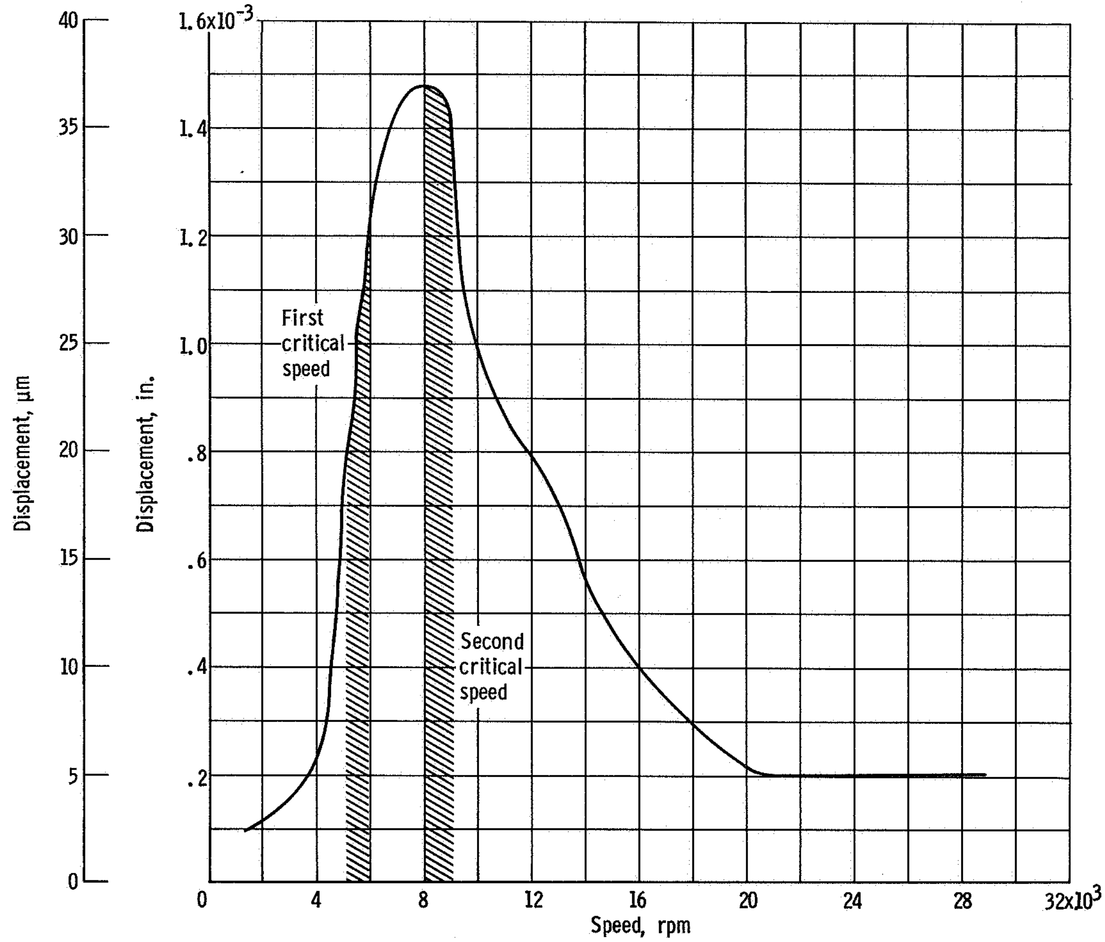


Figure 18. - Envelope of compressor-end journal radial displacement as function of speed for all tests.

dicate that the turbine-end journal motions reached only 65 percent of the amplitude obtained during shutdown. However, the compressor end maximum excursion reached 85 percent of the shutdown amplitude at the first and second critical speeds.

The relative axial motion between thrust runner and stator (as shown in fig. 19) indicates a rise in displacement as the speed is increased above 1000 rpm and reached a peak of 0.00127 inch (0.00323 cm) at 8000 rpm. From 8000 rpm, as the speed increased to 18 000 rpm the excursion dropped to 0.00029 inch (0.00074 cm) and remained constant up to 28 600 rpm.

Here, again, it should be noted that the maximum excursions during injection were somewhat less than the excursions for shutdown when going through the first and second critical speeds. In this case, the excursions during injection were only 59 percent of the slow shutdown values at the first and second critical speeds only.

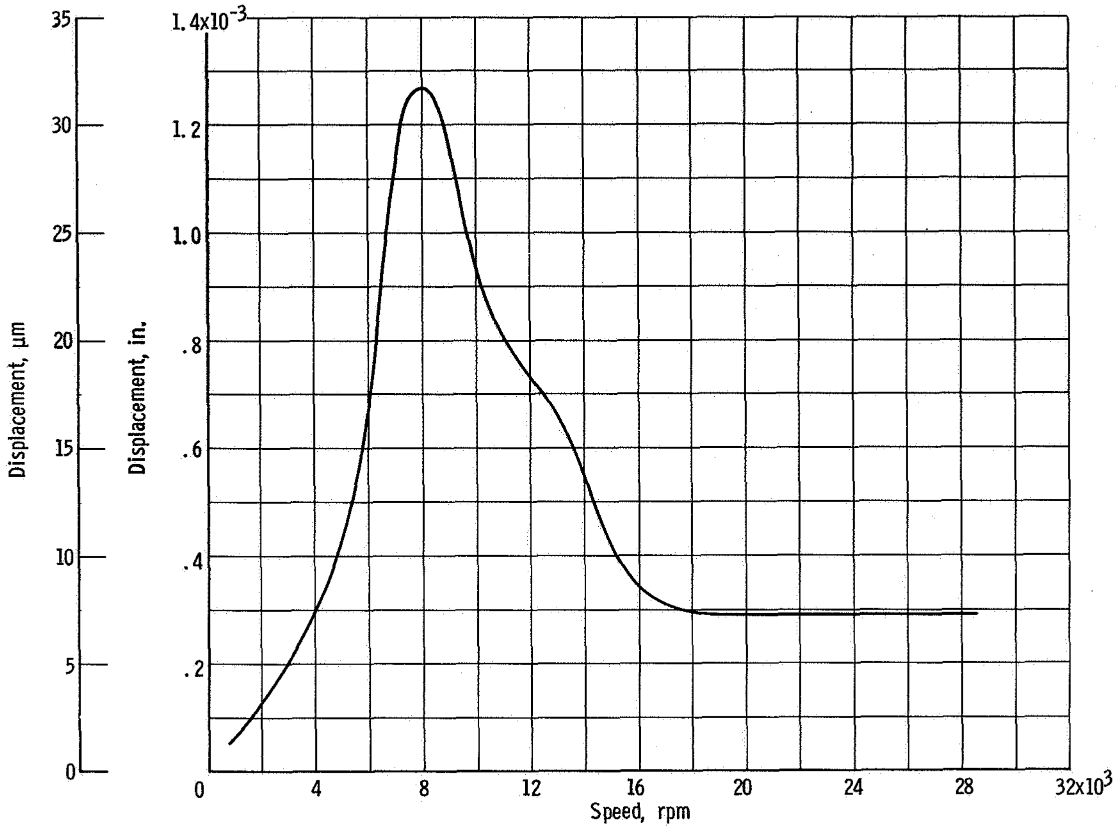


Figure 19. - Envelope of axial thrust bearing axial motion as function of speed for all tests.

Journal Bearing Load

The gas bearings used for the turbine and compressor journals are dependent on pad load for maintaining stable operation. Work done by the contractor indicated that, with a load of about 4 pounds (18 N), the operation of the bearings would be at the threshold of an instability called whirl.

The bearings were preloaded at assembly utilizing the deflection of the diaphragms to provide the required load. Strain gages were provided on the diaphragms to indicate the amount of load on the bearings. As observed in preliminary tests, the strain gages were very sensitive to temperature gradients. Attempts were made to calibrate the strain gages for temperature changes while they were installed in the package. This calibration showed a very sharp downward shift in zero as the temperature was applied. The conservative approach to the calibration was followed by assuming that the strain gages were indicating load values which were lower than the actual load.

As a safety precaution for operating the turbocompressor, a journal bearing indicated load minimum was set at 8 pounds (36 N). If either journal bearing indicated that the load dropped below the minimum set value, the system was shut down. Eight pounds

(36 N) was chosen in order to give enough time to slowly drop the speed below the critical speeds before the journal bearings became completely unloaded.

Figure 20 is representative of the test runs. From this figure, it can be seen that, after the initial increase in load due to increased speed, the indicated loads on both the turbine and compressor journal bearings were decreasing rapidly with time. It was impossible to operate the turbocompressor longer than 30 minutes after injection before the indicated load on the turbine journal bearing was below the minimum 8 pound (36 N) safety limit.

With a turbine inlet temperature of 530° R (294° K), a shaft speed of zero rpm, and all bearings pressurized at normal pressure, the indicated load on the compressor journal bearing is 15.5 pounds (69.0 N) and the turbine journal bearing 15.0 pounds (65.8 N). These loads decreased as the turbocompressor package and connecting piping increased

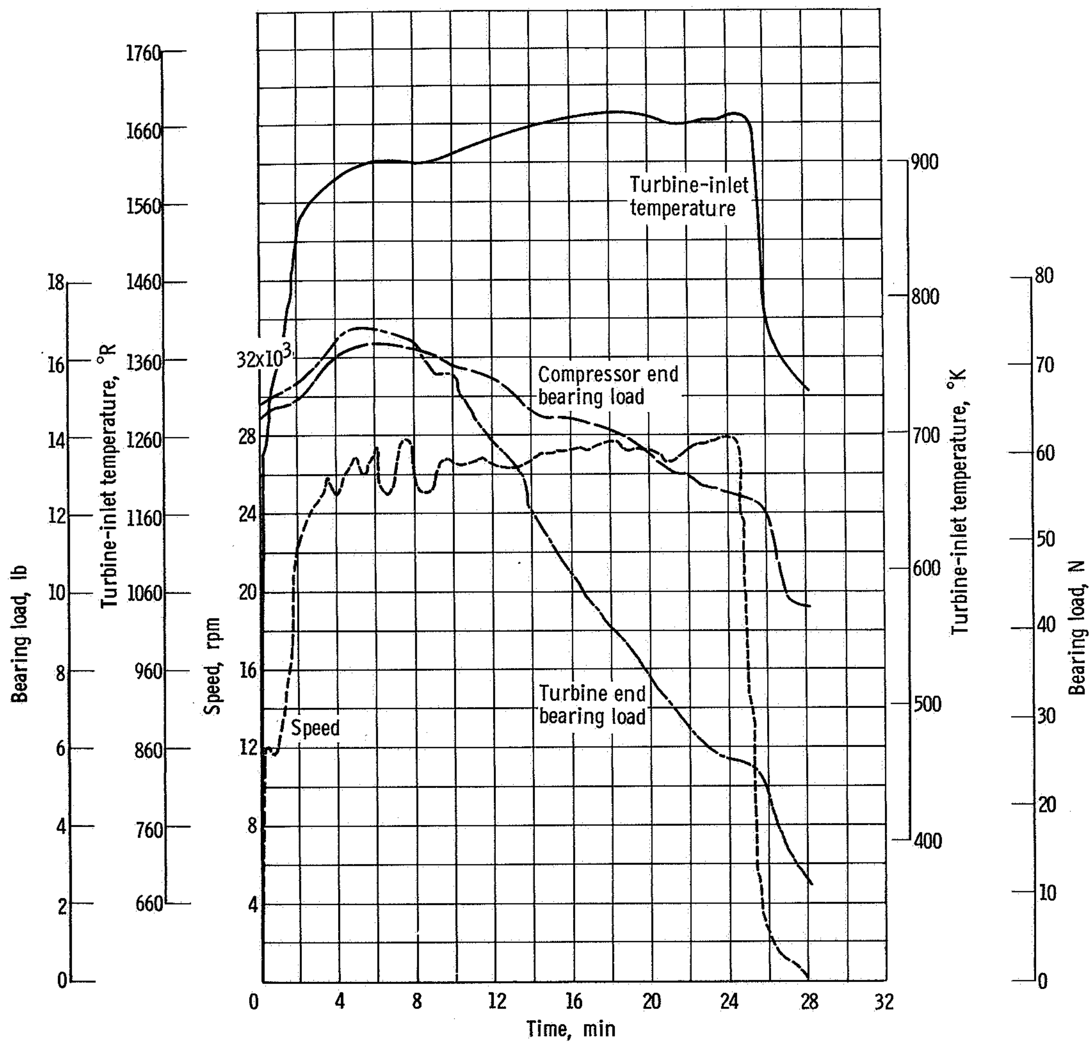


Figure 20. - Speed, turbine-inlet temperature, and bearing loads as functions of time for complete test.

in temperature before injection. The difference between the loads in figure 20 and those stated above are due to the difference in package and inlet temperatures at the time of injection.

Due to the sensitivity of the strain gages to elevated temperatures and thermal gradients, another means of determining bearing load will be necessary. For future testing of the turbocompressor it was decided to install a proximity probe to monitor diaphragm deflection and also to incorporate a means of pneumatically varying the load on the diaphragms and thus allow for longer periods of running time.

SUMMARY OF RESULTS

The objective of the investigation reported herein was to subject the delivered turbocompressor to such transients as may be encountered in a space start of a two-shaft system and to observe reaction to these transients. The results of this investigation may be summarized as follows:

1. The turbocompressor could be started by gas injection. The better of two methods attempted was found to be a constant-flow injection from a shaft-thrust standpoint. The quantity of gas injected to achieve a start was about 2.5 pounds (1.1 kg) at rates from 0.58 to 0.78 pound per second (0.26 to 0.35 kg/sec). The faster rate gave a higher turbocompressor speed at the end of injection.

2. The delivered turbocompressor with the flex-mounted pivot was capable of withstanding the transients associated with rapid starts and shutdowns. Radial and axial excursions of 0.0015 inch (0.0038 cm) and 0.0012 inch (0.0031 cm), respectively, were observed without shaft contact with the bearings during numerous starts and shutdowns.

3. An accidental closing of the turboalternator simulator valve sent the compressor into surge at a speed of 27 000 rpm. The resulting rotor vibrations were large in both axial and radial directions but were not severe enough to cause damage to the turbocompressor unit because it was operating sufficiently far below design speed and the bearings were externally pressurized.

4. Emergency shutdown of the turbocompressor by use of a compressor bypass loop was accomplished rapidly without inducing compressor surge or any other condition that might damage the unit.

5. The strain gages were highly sensitive to elevated temperatures and therefore inadequate for use as the diaphragm load indicators.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 12, 1968,
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