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SMALL DISPLACEMENT, LONG LIFE ON-ORBIT COMPRESSOR DESIGN AND FABRICATION*

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ABSTRACT

The focus of this project, entitled "On-Orbit Compressor Technology Program," is the generation of technology and fabrication of prototype hardware applicable to seven Space Station compressor system applications. The compressors are of the single acting reciprocating piston type and, in general, may be termed miniature in size compared with normal commercially available equipment. The initial technology development is focused on improved valve designs, and the control of pulsations and heating effects in order to increase compressor efficiency and reduce cycle temperatures, thus permitting significantly increased stage pressure ratios. The initial test compressor has been successfully operated at pressure ratios of up to 50:1, and this significant extension of allowable pressure ratio will result in a reduction of the number of required stages and, hence, total hardware thereby reducing system weight and volume. These experiments have also identified the need to employ low shaft speeds, on the order of 250 to 500 rpm, to enhance heat transfer and increase life.

The prototype compressor currently being designed, is to be driven by a low-speed brushless DC motor sealed in a case common to the compressor drive mechanism case. The compressor and motor case will communicate with stage suction pressure so that any minor gas leakage past the piston rings will be returned to the suction. Emphasis in this prototype design is being placed on simplicity, durability, commonality of components, and high efficiency.

INTRODUCTION

The objective of this project is the generation of compressor technology applicable to the Space Station Fluid Management System (FMS), Space Station Propulsion System, and related on-orbit fluid transfer systems. The approach is to perform initial research on a laboratory test compressor (LTC), to then develop a prototype for one specific application, and utilize the results to develop conceptual designs for six additional applications. The prototype development will consist of a detailed design for the specified conditions, followed by fabrication, and testing of the prototype. Both the prototype design and the general conceptual design will be updated based on the test results. The primary emphasis is to develop basic compressor technology (designs, materials, and manufacturing techniques) in a time frame consistent with the support of the Space Station fluid systems development. Design considerations include: (1) commonality, i.e., interchangeability of common hardware assemblies, (2) maximization of service life, and (3) ease of maintenance.

COMPRESSOR APPLICATIONS AND REQUIREMENTS

The compressor requirements for the seven applications cover a wide range of gas properties, pressure, temperature, flow, and duty cycle. These requirements are summarized in Table I. Significant problems and challenges are presented by these requirements, and no existing hardware is currently available to meet the specifications. As an example, the Type I application requires compression of a reducing gas mixture consisting largely of hydrogen, carbon dioxide, and nitrogen from 0.12 MPa (18 psia) up to 8.3 MPa (1200 psia), a total pressure ratio of 67:1, on a continuous basis, with a 10,000 hour operating life goal. Employing a conventional approach, this would be accomplished by reciprocating compressors using three or four stages driven at relatively high speeds, resulting in very small or miniature cylinders. An early selection for the fourth stage for the Type I application was a cylinder with a 0.0095 meter (0.375 inch) bore, a 0.0022 meter (0.086 inch) stroke driven at up to 3300 rpm. The problems posed by such miniature cylinders, valves, and other compressor components were apparent early on, but recent developments have allowed a reversal toward more acceptably sized cylinders (larger) driven at low speeds.

The Type VII application presents a unique challenge since the compression of helium saturated with hydrazene, having a maximum safety limit of 71.1°C (160°F), is required from an initial pressure of 2.76 MPa (400 psia) up to 31 MPa (4500 psia). The problem here, based on conventional compression technology, is to achieve this total pressure ratio (11.25:1) employing a reasonable number of stages without ever exceeding 71.1°C (160°F) at any point of the compression cycle. Based on the starting temperature of 51.7°C (125°F), and assuming isentropic compression with no precooling, the first stage pressure ratio could not exceed 1.156:1, hence, many stages would be required. The thermal problem for this application is further compounded by the lack of a liquid cooled heat sink (ammonia buss duct) which is available for most other applications.

SUMMARY OF TECHNICAL CHALLENGES

We believe that the major challenge of this project is to achieve enhanced and controlled heat transfer through the compressor cycle in order to: (1) achieve high efficiency, (2) limit cycle and discharge temperatures, and (3) reduce the required number of compression stages. Control of cycle temperature is also important with respect to component life, control of liquids and, particularly, to the Type VII application, to stay below the hydrazene safety temperature limit.

*This work is being performed under NASA-JSC Contract NAS 9-18-051 with Southwest Research Institute, San Antonio, Texas, SwRI Project No. 04-2529. Approved for public release, distribution is unlimited.

TABLE I. COMPRESSOR APPLICATION REQUIREMENTS (1)

| Application | I | I | II | IV | V | VI | VII |
|---|---|---|---|---|---|---|---|
| System | SS:IWGS-1 | SS:IWGS-2 | SS:INS | SS:PROP-1 | SS:PROP-2 | OSCRS-1 | OSCRS-2 |
| Fluids | [2] | [3] | N2 | H2 | O2 | N2 | He |
| Contaminants | None | [4] | None [5] | Water [6] | Water [6] | None [5] | [7] |
| Pressure: MPa (psia) | | | | | | | |
| Inlet | 0.12 to 0.17 (18 to 25) | 0.07 to 0.10 (10 to 15) | 3.8 to 4.5 (550 to 650) | 0.69 to 2.1 (100 to 300) | 0.69 to 2.1 (100 to 300) | 27.6 to 2.8 (4000 to 400) | 1.7 (250) |
| Outlet | 6.9 to 8.27 (1000 to 1200) | 6.9 to 8.27 (1000 to 1200) | 40.0 to 42.1 (5800 to 6100) | 2.1 to 20.7 (300 to 3000) | 2.1 to 20.7 (300 to 3000) | 41.4 (6000) | 2.8 to 31 (400 to 4500) |
| Temp: °C (°F) | | | | | | | |
| Inlet: | | | | | | | |
| Maximum | 32.2 (90) | 37.8 (100) | 21 (70) | 65.6 (150) | 51.7 (150) | 51.7 (125) | 51.7 (125) [10] |
| Minimum | 15.5 (60) | 26.7 (80) | -95.6 (-140) | 37.8 (100) | 37.8 (100) | 4.4 (40) | 4.4 (40) |
| Flow Rate: Kg/hr (LBm/hr) | | | | | | | |
| Nominal | 0.005 (0.012) | 0.091 (0.20) | 4.90 (10.8) | 0.078 (0.167) | 0.60 (1.33) | [11] | [12] |
| Maximum | 0.014 (0.030) | 0.500 (1.1) | 7.67 (16.9) | 0.14 (0.31) | 1.13 (2.5) | | |
| Operating Life: hr [8] | 10000 | 10000 | 10000 | 25000 | 25000 | 1000 | 1000 |
| Duty Cycle | Continuous | Continuous | 30 hrs/90 days[13] | On: 54 min. Off: 36 min. | On: 54 min. Off: 36 min. | 24 hrs. Continuous Every 3 months | 24 hrs. Continuous Every 3 months |
| Power: KW Peak [9] | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| Line Size: Meters (inch) | | | | | | | |
| Inlet | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.01 (3/8) | 0.01 (3/8) |
| Outlet | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.013 (1/2) | 0.01 (3/8) | 0.01 (3/8) |

Other technical challenges include: (1) design of long life and dynamically stable valves, (2) efficient control of pulsations, (3) meeting the power, weight, and envelope requirements, and (4) achieve the life and maintainability goals. All of this must be accomplished with component commonality as an important consideration. Again, achievement of enhanced and controlled heat transfer appears to be one key to success of the program.

## BACKGROUND DISCUSSION

Work performed to date on the On-Orbit Compressor Technology Program (OOCTP) has resulted in significant progress in achieving efficient, high ratio compression with small units typical of those required in the various applications described above. The following sections present pertinent technical background information and summarize some of the results of the OOCTP laboratory test compressor results to date.

## FACTORS AFFECTING COMPRESSOR PERFORMANCE

Heat Transfer. Heating related phenomena can affect both the volumetric and thermal efficiency of a reciprocating compressor. Two separate mechanisms are involved: one is direct convective transfer of thermal energy to and from the gas and compressor mechanical structure; and the other is an indirect heating mechanism related to the throttling which occurs across the compressor suction valves. In general, improperly controlled heat transfer reduces flow, thermal efficiency, and often increases the gas discharge temperature. With conventional commercial compressors, flow and efficiency reductions easily exceed 10 to 15 percent from thermal causes alone.

Compressor Valves. Valves are the single most failure-prone component on reciprocating compressors used, for example, on gas transmission service, where they must operate continuously, and therefore build up operational stress cycles quickly. Many different types and configurations of compressor valves have been and are being used, including reed, single and multiple poppet, simple plate types, multiple concentric ring, and complex multiple-degree-of-freedom plate types. The required operation of a compressor valve may be likened to an electrical diode to allow flow in only one direction and depended on to quickly open or close in response to a forward or reverse pressure differential. Variance from this ideal behavior often occurs for several reasons, and we must also be concerned about the impact which occurs when the valve strikes the valve guard upon fully opening, and when it strikes the seat upon closure. One common problem with compressor valves is flutter which is an inherent instability caused by excess valve flow capability and/or excess spring loading.

Compressor valves fail for a number of reasons. By failing, they no longer perform the flow check function and thus allow leakage, and this is generally a result of breakage. Another cause of failure is thermally-induced distortion. Some leakage may be acceptable; too much is not. The most common cause of valve failure is dynamic motion inconsistent with that anticipated by the valve designer. For example, in designing the typical plate or ring valve, the designer assumed ideal planar (no cocking) motion with the valve element lifting or closing and striking the guard or seat, respectively, in an ideal fashion. He based his static and dynamic stress loadings on such assumed ideal action. In reality, because of nonsymmetrical springing and/or pulsation-induced torque on the valve, this element often hits first on edge, inducing early failure.

Pulsation Effects. Another potential problem area with compressor valves is dynamic interaction with piping-induced pulsations. Figure 1 shows a sequence of test pressure-volume diagrams for a typical commercial compressor operating with fixed suction and discharge pressures for four separate rotational speeds with the attached piping interactive with the valve dynamics. The suction and discharge event portion of each of these diagrams shows piping pulsation-induced pressure oscillations within the cylinder. These diagrams also reflect the interaction between system-induced pulsations and valve closure variations. The upper left diagram (812 rpm) indicates relatively "clean" performance and timely valve closure. The diagram for 953 rpm, however, now shows the effect of late suction and discharge valve closure. Note the displacement of the expansion and compression lines at the start of each event (indicated on the figure). The pressure-volume diagram for 1100 rpm reflects, like the 812 rpm case, reasonably timely valve closure (the discharge valve closure is slightly late). The 1507 rpm case indicates a very late suction valve closure, but good discharge closure timing. This variation in valve closure is caused by the timing of the piping pulsations relative to the required closure point of the valves. Note for the suction phases of the 812 and 1100 rpm cases that pulsations produce an increasing pressure just before valve closure which, for these cases, occurs in a timely fashion. For the 953 and 1507 rpm cases, the pulsations cause a decreasing pressure prior to valve closure which is now late.

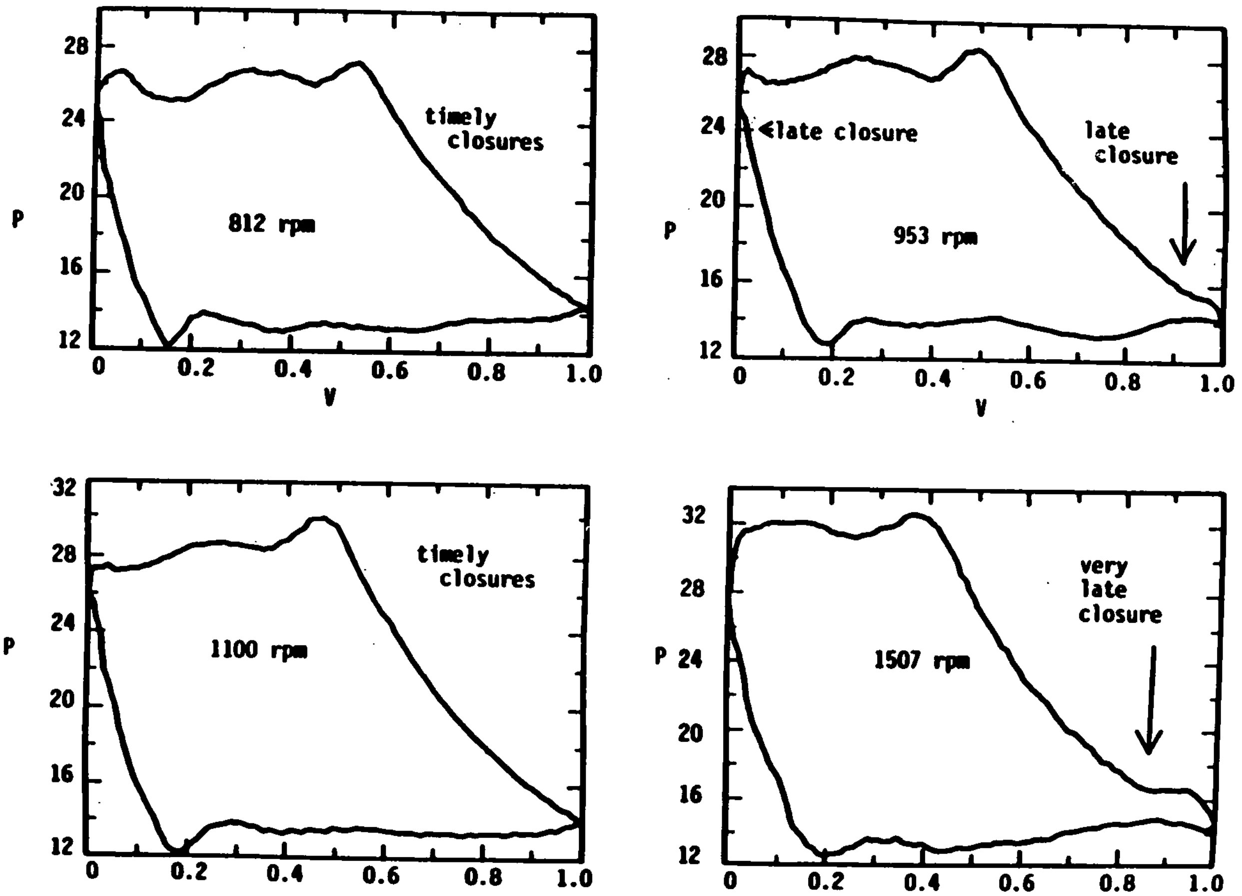



FIGURE 1. SEQUENCE OF TEST PRESSURE-VOLUME DIAGRAMS ILLUSTRATING THE INFLUENCE OF PULSATIONS ON VALVE BEHAVIOR

## LABORATORY COMPRESSOR TEST RESULTS TO DATE

An initial laboratory test compressor was built to obtain valve, pulsation, and heat transfer data pertaining to compressors of a size and operating condition range typical of the On-Orbit requirements. To date, tests have been conducted with air and helium, over a range of speed, pressure, and flow conditions. Cylinder and head cooling has been varied from nonforced, or natural convection condition, to a fully forced cooling condition with cold air as the cooling medium. No tests with liquid cooling have yet been conducted, but are anticipated.

Test data is obtained from the compressor with an analog to digital data acquisition system triggered and phased from shaft rotation angle. Normally, data is obtained for 512 points per one rotation of the shaft to facilitate subsequent Fourier analysis. Various pressure and temperature locations are monitored, along with torque, flow data, etc. This data is written to files, the channels of which may be directly viewed once the acquisition process is complete. Subsequently, the data files are transferred to another computer where detailed analysis takes place.

## USE OF PERFORMANCE SIMULATION AND DIAGNOSTICS

An extremely useful capability termed "quantitative diagnostics"* is being employed for analysis and interpretation of the test data. Simulation programs employing this technique have been of particular benefit in the correlation of OOCTP test results with the simulation results, and subsequent adjustment of the heat transfer models. Certain details in the shape of a test versus simulation pressure versus volume or PV diagram may be brought into agreement by changes in several simulation parameters through a "forcefit" process. The original purpose for which the "forcefit quantitative diagnostics" process was developed was to allow detection and quantification of compressor faults such as valve leaks, ring leaks, etc., based on a file of pressure versus volume and other test data. This "forcefit" process involves, initially, an overlay of a field or test PV card, which may contain faults, and a corresponding predicted PV card, based on assumed compressor geometric parameters, but without faults. From this initial comparison of test and predicted data, certain deviations between the two PV diagrams are used to identify what faults should be implanted in the predictive model, and then a new simulation is performed. This process, performed automatically by the computer code, is continued until the test and predicted PV diagrams are "forced" to agree. The predicted performance data for this faulted simulation may then be used to quantify the test compressor performance and also the cost in wasted energy associated with, for example, a valve leak.

## COMPRESSOR DYNAMIC PERFORMANCE

Figure 2 shows a PV diagram for one of the air tests employing a first design high turbulence suction valve; the corresponding performance and test condition data are noted on the figure. One striking feature of this diagram is the general "cleanness" with virtually no indication of pulsations; also, extraordinarily low valve losses are demonstrated. Comparison with the PV diagrams, shown earlier in Fig. 1, obtained from a commercial compressor under laboratory conditions, dramatically shows the progress made with respect to valves and pulsation control as part of the OOCTP. These diagrams demonstrate, for these particular operating conditions, near ideal control of the valve dynamics and pulsations.

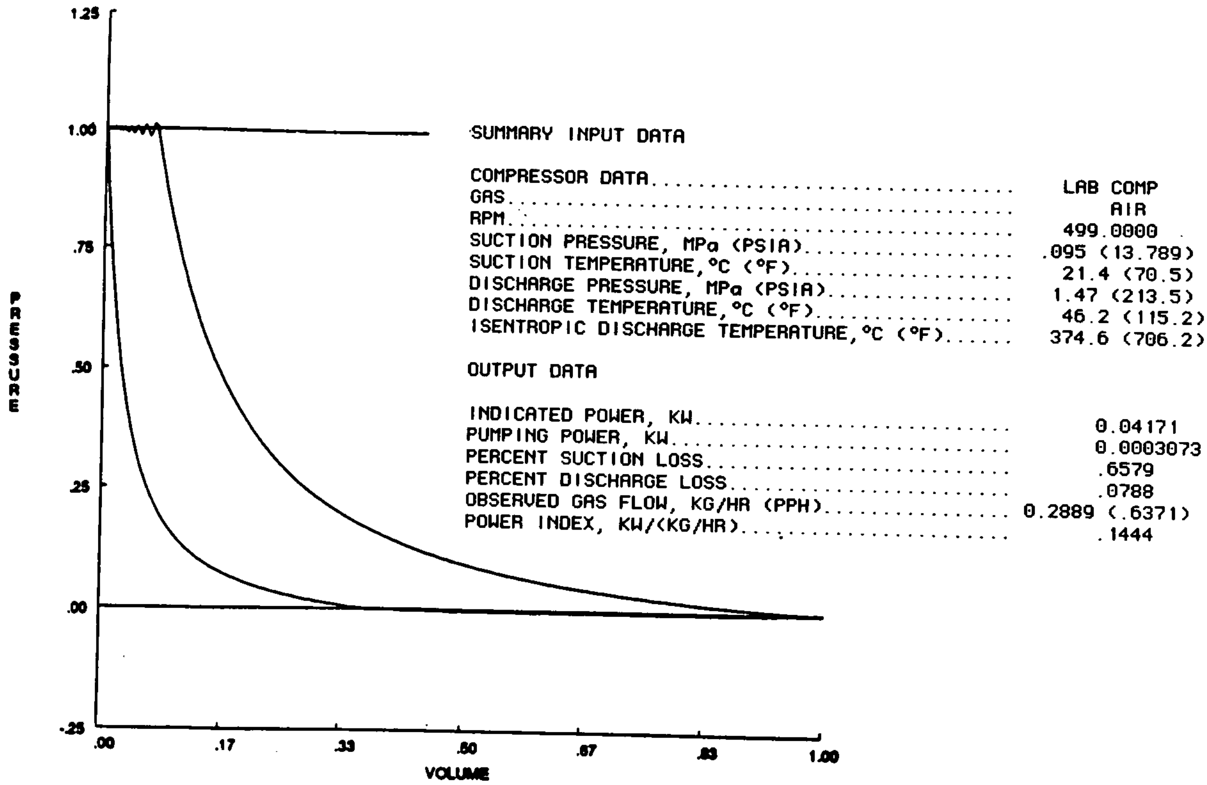



FIGURE 2. EXAMPLE TEST RESULTS FROM LABORATORY COMPRESSOR WITH RANDOM TURBULENCE SUCTION VALVES

*Based on work performed under contract to the Gas Research Institute (Contract No. 5087-271-1485) and the Southern Gas Association.

The power loss associated with the "typical" valve operation, shown in Fig. 1, is quite significant and the unsatisfactory valve dynamics illustrated often leads to shortened valve life. The greatly improved performance demonstrated in Fig. 2 will certainly be conducive to long valve life for the proposed compressors since the dynamic behavior is well controlled. Proper valve and pulsation dynamics alone, however, do not guarantee long valve life.

Thus far, on the OOCTP as demonstrated here, we have been very successful in designing valves which exhibit very low losses with controlled motion conducive to long life. The valves presently employed are of the reed type, chosen because of the simplicity of design and behavior consistent with high reliability. This valve type is the standard for most air conditioning compressor applications which require and have demonstrated long life. During opening and closing, these valves may be easily controlled with respect to the way their motion is "arrested" by the seat and guard.

LABORATORY COMPRESSOR ENHANCED HEAT TRANSFER

One focus of the present OOCTP is to turn the effects of heat transfer to advantage rather than to the detriment of compressor performance. This has been achieved to date with rather dramatic results. Using the latest high swirl inducing suction valve design in the laboratory test compressor, we have demonstrated pressure ratios of over 50:1 with stabilized discharge temperatures below 65.6°C (150°F). This does not mean that such high ratio operation is suggested or will be employed, but it does dramatically provide partial demonstration of the effect of the thermal control thus far achieved.

Normally employed pressure ratios in commercial compressors are generally restricted to the 3:1 to 4:1 range to prevent excessive temperatures. We have largely overcome this limitation through proper thermal control. We feel this approach is a major breakthrough, since the demonstrated efficiency is excellent, and now applications previously thought to require two or three stages may be accomplished in one stage with considerable reduction in required hardware. It must be emphasized that this is a result of employing low-speed, small compressors, low mass flows and, most importantly, a high thermal energy exchange design. Results will tend to be more pronounced for certain applications than for others, and will not necessarily scale up with equal effectiveness for large compressors.

The test data shown in Fig. 3 is for essentially the same pressure ratio and forced air cooling conditions as that in Fig. 2. The difference is in the suction valve configurations used. For the Fig. 2 data set, random turbulence generation suction valves were used, while for the case of Fig. 3, first design high swirl inducing valves were employed. Note the significant decrease in discharge temperature, 46.1 to 33.9°C (115 to 93°F), increase in flow, .289 to .354 kg/hr (.6371 to .7801 pph), and dramatic improvement in efficiency index, .1444 KW/kg/hr to .12097 KW/kg/hr (65.5 to 54.9 watts/pph), attributable to this valve change.

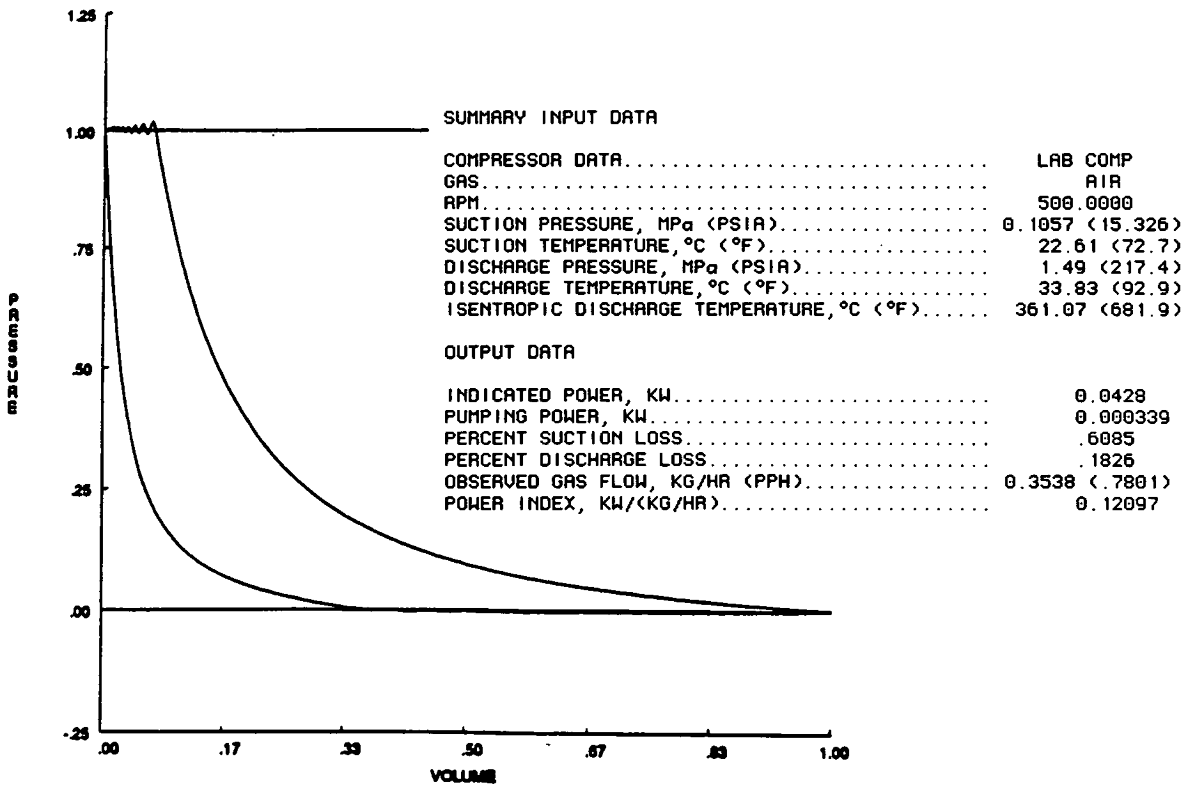
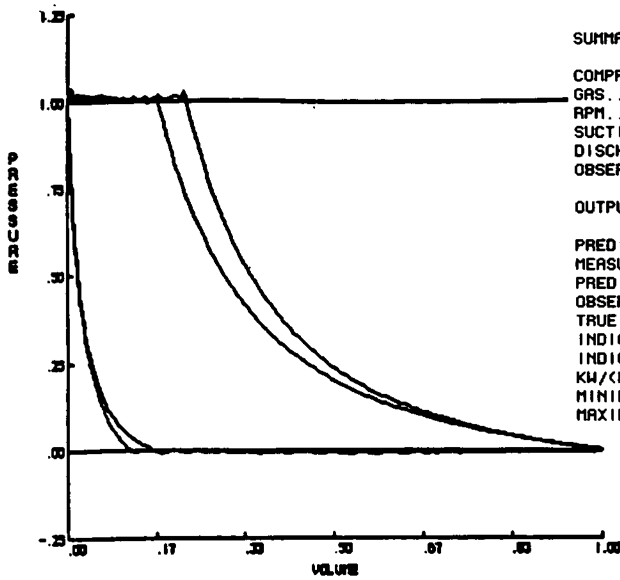



FIGURE 3. EXAMPLE TEST RESULTS FOR CONDITIONS SIMILAR TO FIG. 2, BUT EMPLOYING HIGH SWIRL SUCTION VALVES

Figures 4 and 5 illustrate application of the "forcefit" diagnostics approach to correlation of one file of test data, primarily, to verify the analytical heat transfer model. Figure 4 shows comparison of test and simulation results with previously established (measured) compressor parameters, assuming isentropic compression. The results show both the quantitative data and PV diagram shape to be in considerable disagreement. Figure 5 shows an overlay of PV diagrams and quantitative test data after parametric adjustment to force agreement of test and simulation results.



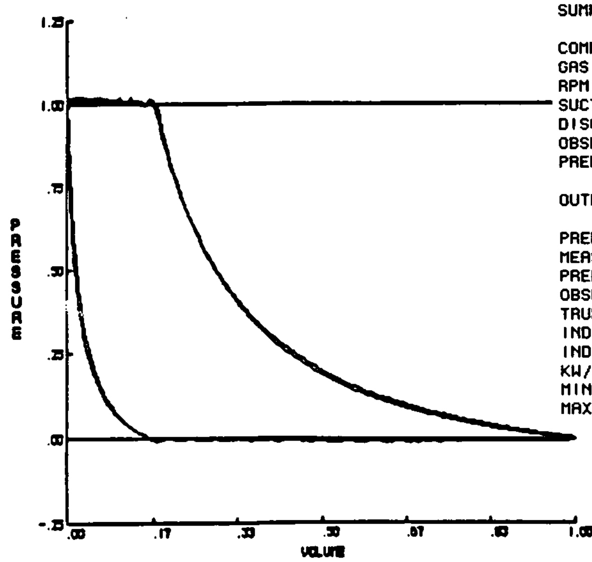

SUMMARY INPUT DATA

| | |
|---|---|
| COMPRESSOR DATA | LAB COMP |
| GAS | AIR |
| RPM | 499.0000 |
| SUCTION PRESSURE, MPa (PSIA) | 0.0932 (13.516) |
| DISCHARGE PRESSURE, MPa (PSIA) | 0.7997 (115.96) |
| OBSERVED SUCTION TEMPERATURE, °C (°F) | 20.89 (69.6) |

OUTPUT DATA

| | |
|---|---|
| PREDICTED GAS FLOW, KG/HR (PPH) | 0.4374 (.9643) |
| MEASURED GAS FLOW, KG/HR (PPH) | 0.4568 (1.007) |
| PREDICTED DISCHARGE TEMPERATURE, °C (°F) | 371.5 (700.69) |
| OBSERVED DISCHARGE TEMPERATURE, °C (°F) | 46.8 (116.3) |
| TRUE VOLUMETRIC EFFICIENCY | .8793 |
| INDICATED THERMAL EFFICIENCY | .9868 |
| INDICATED POWER, KW | 0.04484 |
| KW/(KG/HR) | 0.09816 |
| MINIMUM CYCLE TEMPERATURE, °C (°F) | 12.27 (54.083) |
| MAXIMUM CYCLE TEMPERATURE, °C (°F) | 377.52 (711.53) |

FIGURE 4. EXAMPLE COMPARISON OF TEST AND SIMULATION DATA BEFORE INCLUSION OF HEAT TRANSFER EFFECTS IN SIMULATION




SUMMARY INPUT DATA

| | |
|---|---|
| COMPRESSOR DATA | LAB COMP |
| GAS | AIR |
| RPM | 499.0000 |
| SUCTION PRESSURE, MPa (PSIA) | 0.0932 (13.516) |
| DISCHARGE PRESSURE, MPa (PSIA) | 0.7997 (115.96) |
| OBSERVED SUCTION TEMPERATURE, °C (°F) | 20.89 (69.6) |
| PREDICTED END OF SUCTION TEMPERATURE, °C (°F) | 33.78 (92.8) |

OUTPUT DATA

| | |
|---|---|
| PREDICTED GAS FLOW, KG/HR (PPH) | 0.4538 (1.0006) |
| MEASURED GAS FLOW, KG/HR (PPH) | 0.4572 (1.0077) |
| PREDICTED DISCHARGE TEMPERATURE, °C (°F) | 45.68 (114.23) |
| OBSERVED DISCHARGE TEMPERATURE, °C (°F) | 46.83 (116.3) |
| TRUE VOLUMETRIC EFFICIENCY | .8835 |
| INDICATED THERMAL EFFICIENCY | 1.1340 |
| INDICATED POWER, KW | 0.039211 |
| KW/(KG/HR) | 0.08576 |
| MINIMUM CYCLE TEMPERATURE, °C (°F) | -105.36 (-157.65) |
| MAXIMUM CYCLE TEMPERATURE, °C (°F) | 179.06 ( 354.31) |

FIGURE 5. COMPARISON OF TEST AND SIMULATION DATA OF FIG. 4 AFTER FORCED AGREEMENT WITH HEAT TRANSFER MODEL

A careful comparison of the numerical *before* and *after* results, shown in Figs. 4 and 5, clearly illustrates the effect and importance of enhanced thermal control. Compared with conventional compressors which, as discussed previously, have true volumetric efficiencies (TVE), which are less than the card suction volumetric efficiency or EVS value, we are now achieving TVE's in excess of the EVS value. In addition, the work of compression is greatly reduced because of heat removal during compression. The net result is achievement, in some cases, of thermal efficiencies in excess of that for isentropic compression and with discharge temperatures far lower. This, we feel, is a major breakthrough.

## PROTOTYPE DESIGN FEATURES AND PREDICTED PERFORMANCE

A prototype compressor for the Type II application is being designed, fabricated, and tested as part of the program effort. This compressor will be of the reciprocating piston type. As part of an initial screening study, the liquid energized diaphragm type was evaluated, but discarded because of complexity, excess weight, lubrication and actuation liquid problems, excessive friction, and catastrophic failure issues.

### PROTOTYPE COMPRESSOR TYPE AND ACTUATION MEANS

For the prototype design, two basic piston actuation means have been considered: rotary motor actuation and linear motor actuation. We anticipate taking maximum advantage of thermal transfer for reasons of efficiency and to minimize the number of stages required, as previously discussed. This is best accomplished by employing low drive speeds (e.g., 250 to 500 rpm). With this low-speed approach, the use of linear motors must be ruled out because of excessive weight. We will, therefore, use brushless DC rotary motors to drive the prototype compressor. The use of a low drive speed, in addition to enhancing thermal transfer, also results in a device with a satisfactory (i.e., not too small) size and also enhances component life, which is very important.

We have demonstrated by test and analysis that this approach results in extremely high operational efficiency and low operational temperatures. Very importantly, it also allows use of fewer compression stages, hence, less total hardware. We anticipate the need for only two compression stages for the Type I and Type II applications. For the Type III application, we are anticipating the use of only one stage. The predicted efficiency with this approach, backed by test results, is in excess of the corresponding isentropic compression efficiency, but with easily tolerable temperatures!

Figure 6 represents a conceptual drawing of the proposed reciprocating piston compressor and actuation means. The piston is "follower actuated" by bearing against an eccentrically mounted antifriction bearing. The piston is kept in contact with the actuator by a combination of gas and spring forces. The piston is laterally supported by wear rings and, secondarily, by seal rings.

The prototype design shown in Fig. 6, we believe, minimizes the risk of catastrophic failure by using a simple compressor cylinder and drive system design employing largely proven concepts. The major risk items are the piston seal rings and the compressor valves; however, results to date have greatly increased our confidence in the durability of these items. Piston ring seals of the type proposed were used in previous compressor and pump designs and development projects in which we have been involved. One application in this prior work was, in some ways, more severe than for the proposed compressors. Our experience has shown this type seal will not fail suddenly, or in a catastrophic manner. These seals, once broken in, wear very slowly and predictably in the absence of particulate contaminants. Because they are "O" spring energized, these ring seals can accommodate some wear without significant leakage.

We must anticipate some minor, but manageable, leakage with these seal rings at some point in their life, but because the compressor case will communicate with the suction port, any leakage passes back to the stage suction. With this approach, we accept some small but acceptable loss of efficiency because of eventual minor ring seal leakage in order to gain reliability, and to avoid catastrophic failure and increased complexity. To achieve a leak-free compressor design would require use, for example, of a liquid coupled diaphragm design subject to catastrophic fatigue failure, and with greatly increased mechanical complexity, weight ,and friction; thus, we do not feel this is an effective approach.

It is presently anticipated that the actuation bearing ultimately may be unlubricated and constructed of a combination of metallic and ceramic component parts. The primary reason for providing lubrication in an antifriction bearing is to conduct heat away from the balls and races. With the low speeds to be employed, along with the bearing materials tentatively selected, we anticipate that a "no lubrication" bearing concept may work. For the prototype, conventional off-the-shelf bearings will be employed.

A low-speed, high torque, brushless DC rotary motor will provide the power to drive the compressor. The one possible disadvantage of this low-speed, enhanced thermal transfer approach is that the low drive speed may require the use of a speed reduction unit or employment of a special low-speed, high torque, low weight DC motor.

Because the approach chosen results in compressor systems for all three types, which are predicted to be below the weight limit, there is the possibility of providing a redundant compressor module with the follower piston pneumatically retracted and inactive, until needed to replace the other cylinder.

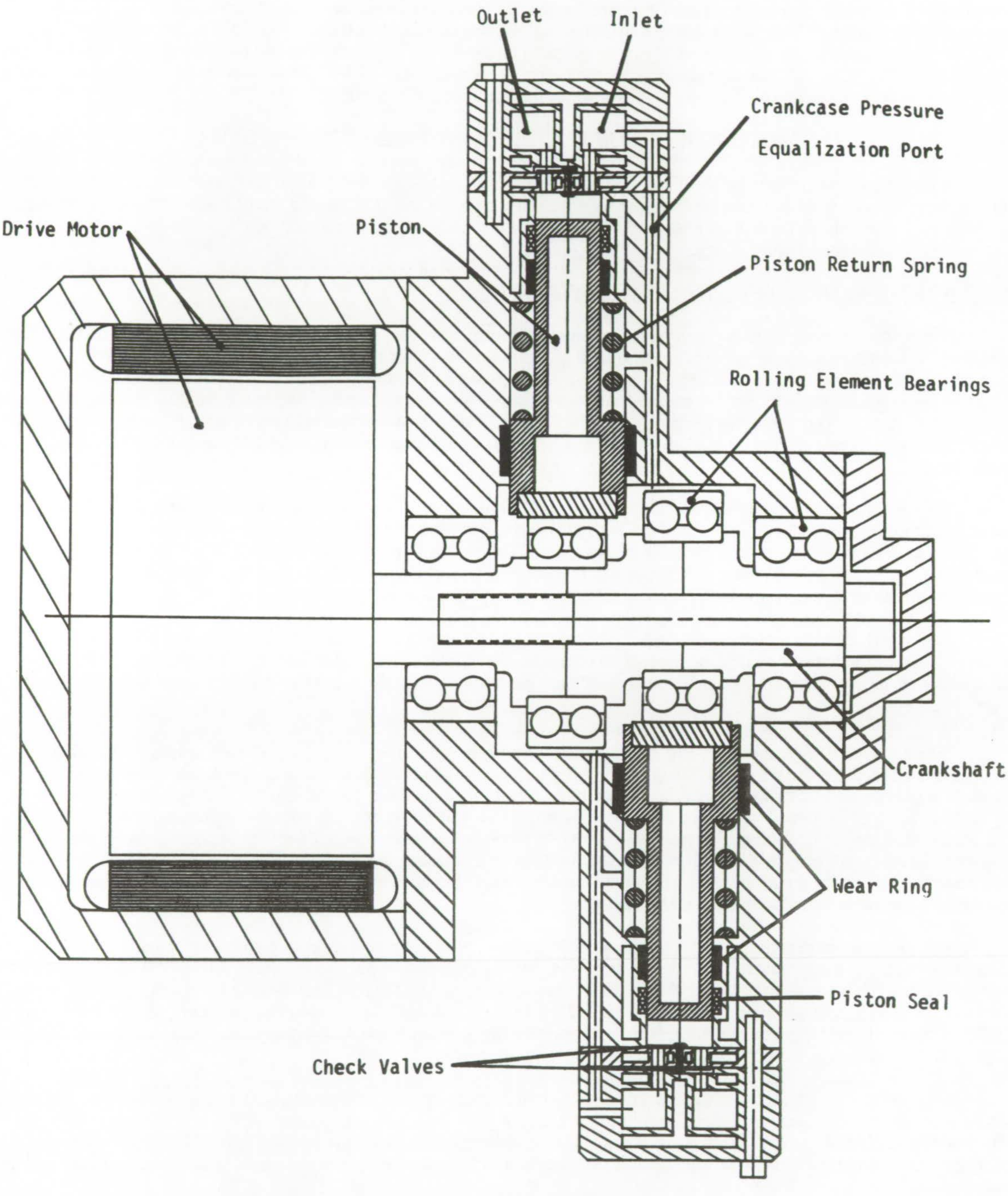



FIGURE 6. CONCEPTUAL ILLUSTRATION OF PROPOSED COMPRESSORS, AND DRIVE MEANS, SHOWN HERE WITH TWO DRIVEN COMPRESSOR CYLINDERS

## EXAMPLE PREDICTED PERFORMANCE

Initially, and before test data was obtained from the laboratory compressor, performance predictions were made for all seven applications, based on our understanding of thermal effects at that point in time. The resulting compressor concepts employed pressure ratios not exceeding 4:1 and were proposed to be driven at relatively high speeds. Based on the enhanced thermal effects discussed previously, the compressor concepts for all seven applications are being revised to take advantage of lower drive speeds, larger compressor cylinders (still quite small), and higher operating pressure ratios. For example, the Type I application initially required four compression stages with the fourth stage sized to have a 0.0095 meter (0.375 inch) bore, a 0.0022 meter (0.086 inch) stroke driven at up to 3300 rpm — a very miniature compressor cylinder, indeed! The Type I compressor design is now configured to employ only two stages, with the final stage having a 0.0254 meter (1.0 inch bore), a 0.0076 meter (0.3 inch) stroke, and running at up to 350 rpm.

Table II shows results of predicted performance for Types II and III applications. The Type II application, which is for a mixed waste gas, is compressed from as low as 0.07 MPa (10 psia) up to as high as 8.27 MPa (1200 psia). The original concept was to employ four stages; however, as reflected in Table II, this can be accomplished in two stages. Similarly, the Type III application, originally thought to require two stages can be accomplished, we believe, with just one stage. As reflected in Table II, this can be accomplished with very low discharge temperatures and with thermal efficiencies exceeding 100 percent relative to isentropic compression.

TABLE II. PREDICTED PERFORMANCE FOR TYPES II AND III APPLICATIONS BASED ON ENHANCED HEAT TRANSFER WITH HIGH SWIRL SUCTION VALVES

| Configuration | Gas | RPM | Suction Pressure | Discharge Pressure | Flow | Discharge Temperature | Indicated Power (KW) | Thermal Efficiency re: Isentropic |
|---|---|---|---|---|---|---|---|---|
| Type II-1 | Mix | 635 | 0.069 MPa (10 psia) | 0.828 MPa (120 psia) | 0.4962 kg/hr (1.094 pph) | 22.06°C (71.7°F) | 0.0326 | 119.2 |
| Type II-2 | Mix | 405 | 0.69 MPa (100 psia) | 8.28 MPa (1200 psia) | 1.837 kg/hr (4.05 pph) | 59.06°C (138.3°F) | 0.1199 | 120.0 |
| Type III | N2 | 450 | 3.8 MPa (550 psia) | 421 MPa (6100 psia) | 7.802 kg/hr (17.2 pph) | 86.33°C (187.4°F) | 0.5987 | 115.5 |

Table III shows the predicted power consumption, weight, and required volume for the Types I, II, and III compressor systems, including compressor mechanical losses and drive motor inefficiency, plus requirements of cooling and control systems. In all cases, the estimated power, weight and volume falls below the allowable values.

TABLE III. ESTIMATED POWER CONSUMPTION, WEIGHT, AND VOLUME FOR TYPES I, II, AND III COMPRESSOR SYSTEMS

| | Power | | Weight | | Volume | |
|---|---|---|---|---|---|---|
| | Allowable | Estimated | Allowable | Estimated | Allowable | Estimated |
| Type I System | 1 KW | .221 KW | 36 kg (80 lbs) | 18 kg (40 lbs) | 0.0425 m^3 (1.5 ft^3) | 0.0066 m^3 (.2344 ft^3) |
| Type II System | 1 KW | .348 KW | 36 kg (80 lbs) | 21 kg (47 lbs) | 0.0425 m^3 (1.5 ft^3) | 0.0283 m^3 (less than 1 ft^3) |
| Type III System | 1KW | 1.8 KW | 36 kg (80 lbs) | 26.8 kg (59 lbs) | 0.0425 m^3 (1.5 ft^3) | 0.016 m^3 (.57 ft^3) |

## SUMMARY AND CONCLUSIONS

Heat transfer in existing reciprocating piston compressors generally causes a reduction of overall efficiency because it is not properly controlled. Attempts to achieve near isentropic compression in order to achieve high efficiency leads to excessive and destructive temperatures if high compression ratios are attempted. Restriction of pressure ratios, resulting in an increased number of compression stages, is then required to control discharge temperature.

Based on experimental and analytical results to date, we now know that heat transfer can be controlled during the cycle using unique new techniques, which result in high efficiency with reasonable gas discharge temperatures while employing high pressure ratios, indeed, even very high pressure ratios. One advantage to this approach, in addition to reduced overall power consumption, is the requirement for fewer stages, thus, less hardware. Also, the use of high stage pressure ratios, thus minimizing hardware, could allow the addition of idle, redundant compressors without exceeding the weight limitation. The recent thermal control breakthroughs permit exciting new possibilities for the On-Orbit Compressors because of the high efficiencies achieved with high pressure ratios and low discharge temperatures.

Tests conducted to date with the laboratory compressor have indicated near ideal control of valve dynamics and losses, and pulsations. Since pulsations and valve losses waste power, the present approach will result in higher compressor efficiency. Further, proper control of valve dynamics is essential for long valve life.

The predicted performance, power consumption, total system weight, and volume requirements for the application types discussed herein all fall well within the required specifications.

The prototype compressor and drive system emphasizes simplicity, minimizes the risk of catastrophic failure and, with proper mechanical design, can provide long life. Further, it is anticipated that the design will require little or no periodic maintenance.