DOE/NV/11982--T1

Arthur D Little

Final Report

Advanced Fuel Cells for Transportation Applications

(Development of Compressor/Expander for Fuel Cells in Transportation Applications)

Report to
U.S. Department of Energy

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Contract Number
DE-AC08-96NV11982

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Reference 58473

DISCLAIMER

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1. Executive Summary

This document summarizes the work performed under the Department of Energy (DOE) contract number DE-AC08-96NV11982 by Arthur D. Little, Inc. This Research and Development (R&D) contract was directed at developing an advanced technology compressor/expander for supplying compressed air to Proton Exchange Membrane (PEM) fuel cells in transportation applications.

The objective of this project was to develop a low-cost high-efficiency long-life lubrication-free integrated compressor/expander utilizing scroll technology. The goal of this compressor/expander was to be capable of providing compressed air over the flow and pressure ranges required for the operation of 50 kW PEM fuel cells in transportation applications. The desired ranges of flow, pressure, and other performance parameters were outlined in a set of guidelines provided by DOE. The project consisted of the design, fabrication, and test of a prototype compressor/expander module.

The scroll CEM development program summarized in this report has been very successful, demonstrating that scroll technology is a leading candidate for automotive fuel cell compressor/expanders. The objectives of the program are listed below, followed by a description of the program accomplishments:

- **Develop an integrated scroll CEM** - a fully-integrated, low-cost, high-efficiency scroll CEM was developed under this program. Although the unit has not been operated for sufficient hours to accurately assess reliability and durability, the unit was run successfully without major incident throughout the test program.
- **Demonstrate efficiency and capacity goals** - The test program demonstrated the high-efficiency characteristics of scroll in this application. While the full flow capacity goal was not demonstrated during the test program because of drive torque considerations, this is not a limitation of scroll technology.
- **Demonstrate manufacturability and cost goals** - Manufacturability was demonstrated during fabrication of the scroll CEM, which was accomplished primarily on a three axis CNC milling machine. The manufacturing cost estimate concluded that production cost goals can be achieved.
- **Evaluate operating envelope** - The CEM has been tested at Arthur D. Little over a wide range of conditions.

In summary, while the scroll CEM program did not demonstrate a level of performance as high as the DOE guidelines in all cases, it did meet the overriding objectives of the program. A fully-integrated, low-cost CEM was developed that demonstrated high efficiency and reliable operation throughout the test program. The performance "bar" was set very high on this program, and while the scroll did not exceed the bar, it came very close. Future development of the scroll CEM will undoubtedly improve performance to the level where it can meet these performance goals.

2. Introduction

2.1 Scroll Compressor/Expander Program

This document summarizes the work performed under the Department of Energy (DOE) contract number DE-AC08-96NV11982 by Arthur D. Little, Inc. This Research and Development (R&D) contract was directed at developing an advanced technology compressor/expander for supplying compressed air to Proton Exchange Membrane (PEM) fuel cells in transportation applications.

The objective of this project was to develop a low-cost high-efficiency long-life lubrication-free integrated compressor/expander utilizing scroll technology. The goal of this compressor/expander was to be capable of providing compressed air over the flow and pressure ranges required for the operation of 50 kW PEM fuel cells in transportation applications. The performance guidelines for the scroll CEM are outlined in Table 2-1. Use of an integrated expander allows recovery of energy from the pressurized air exhaust stream, to enhance the overall efficiency of the pressurized air supply system.

Table 2-1: DOE Performance Guidelines

Parameters		DOE Guidelines	
		Compressor	Expander
Flow Rate @ Max Power			
Dry Air	(g/sec)	64 to 76	56 to 70
	(kg/hr)	230 to 273	203 to 254
Water Vapor	(g/sec)	0 to 4	9 to 16
	(kg/hr)	0 to 11	31 to 55
Inlet Pressure	(atm)	1.0	2.8
Outlet Pressure	(atm)	3.2	1.0
Inlet Temperature			
Design Point:	(°C)	20 to 25	118 to 150
	(°F)	68 to 77	244 to 302
Extreme:	(°C)	-40 to 60	65 to 150
	(°F)	-40 to 140	149 to 302
Max. Shaft Power	(kW)	12.6	8.3
Turndown Ratio		10:1	10:1
Contamination		Oil-free <100 ppm	
Efficiency vs Flow & PR			
100% flow	3.2 PR	75%	90%
80% flow	3.2 PR	80%	90%
60% flow	2.7 PR	75%	86%
40% flow	2.1 PR	70%	82%
20% flow	1.6 PR	65%	80%
10% flow	1.3 PR	50%	75%
Volume*	liters	4 total	
Weight*	kg	3 total	
Production Unit Cost* @ 100,000 units/yr		$200 total	
Start-up Response		<5 sec to 90% max. rpm	
Transient Response		<4 sec for 20 to 90% max. flow	
Noise		<80 dB	

*These values do not include heat exchangers or motors/controllers, however sizing traded-off studies should target minimization of the overall air supply subsystem

The project consisted of the design, fabrication, and test of a prototype compressor/expander module, including the following tasks:

- Design and analyses to establish suitable component designs
- Component evaluations to establish refined component designs and the design for the prototype integrated compressor/expander
- Prototype compressor/expander fabrication and qualification testing
- Progressive hardware modifications and upgrades based on test results.

2.2 Need for Compressor/Expander

All Proton Exchange Membrane Fuel Cell (PEMFC) systems currently under development for vehicular applications operate under pressurized conditions, usually about 3 atmospheres. This pressure has been selected to reduce stack/fuel processor subsystem volume, improve stack performance, and facilitate water management. However, achieving high levels of system performance at those pressures requires development of high efficiency compressors, as well as high efficiency expanders to recover energy from the fuel cell exhaust streams. The design and operation of these subsystems is complicated by their need for simultaneously achieving high full and part load efficiency, operation without lubrication in the working spaces, operation over a 10:1 flow range, and compact size. No commercially available equipment meets this combination of requirements, particularly at the pressure ratio of 3:1 associated with current design strategies.

Compressor/expander systems could be the defining element when fuel cell power systems are compared with other options for advanced vehicle power systems, as they may impact system efficiency by 5-7%, weight by up to 1-2 kg/kW, and cost by $5-10/kW. Unless all elements are simultaneously optimized at the appropriate pressure level, fuel cell competitiveness will suffer.

2.3 Scroll Technology

Arthur D. Little has been involved in the development of scroll compressor and expander technologies for over 20 years, and compressors in general for over 30 years. Arthur D. Little is the leading independent research organization worldwide in the development of high efficiency scroll technology. Our preliminary analysis of a baseline PEMFC system with a compressor/expander operating at 3 atm. pressurization, with air flow requirements of about 100-125 SCFM, shows that scroll technology has compelling advantages. This view is consistent with current trends in both automotive and stationary air conditioning compressors, where scroll is rapidly increasing its market penetration in the 3 to 10 kW power range. Scroll is also the technology of choice for General Motors Corporation's methanol fuel cell system conceptual design. The advantages of the scroll in this application include:

- High efficiency at both design conditions and part-load operation.
- Ability to operate without lubrication in the working volumes, with little loss in efficiency and with acceptable life characteristics. This is demonstrated by the commercially available air compressor equipment manufactured by Iwata Corporation, (sold by Powerex) under license from Arthur D. Little.
- Capability to operate efficiently over a wide range of pressure ratios, including the relatively high pressure ratios of 3:1 of the baseline design.
- Efficient use of space combined with high speed operation, resulting in compact configurations.
- Demonstrated adaptability, allowing operation in both compression and expansion modes using the same basic mechanical configurations.
- Accommodation of condensate in the gas stream.

No other existing technology has the ability to simultaneously meet the requirements discussed above. Consequently, a scroll integrated compressor/expander/motor assembly shows great promise in meeting all the requirements for future transportation Proton Exchange Membrane Fuel Cells (PEMFC).

2.4 Scroll CEM Operation

The purpose of this R&D effort was to develop a dry (lubrication free) scroll compressor/expander which is capable of providing compressed air to the fuel cell system and recovering energy from the fuel cell exhaust gases. The compressor and expander both utilize scroll technology, in which two spiral-shaped sheets, or "scrolls", are interleaved to provide chambers enclosing a volume of gas. As shown in Figure 2-1, as the two scrolls orbit relative to each other the chambers spiral inward, contracting (or expanding) as the orbit progresses. These chambers provide the compression (or expansion) of the gas. The power extracted from the hot gas by the expander provides some but not all of the power necessary to drive the compressor. A variable speed D.C. motor is coupled to the compressor/expander assembly to provide compressor power needs not met by the expander, for system startup functions, and for capacity control during part-load operation. At this phase of the development, the goal of the integrated compressor/expander/motor (CEM) was to operate at a maximum speed of 5,000 rpm to minimize size and weight.

The CEM design was based on development work undertaken by Arthur D. Little and its licensees over a period of more than twenty years. Scroll technology has been incorporated into millions of compressors sold by Arthur D. Little licensees throughout the world. Recent developments have addressed dry air compressor applications and high temperature scroll expanders for use in small Brayton cycle engines, which are directly relevant to the specialized needs of the fuel cell application.

GENERAL

A compressor has two scrolls. The top scroll is fixed and the bottom scroll orbits. Each scroll has walls in a spiral shape that intermesh.

INLET—FIRST ORBIT

As the bottom scroll orbits, two air pockets are formed and enclosed.

COMPRESSION—SECOND ORBIT

The air is compressed as the volume is reduced closer to the center of the scroll.

DISCHARGE — THIRD ORBIT

The air is compressed further and discharged through a small port in the center of the fixed scroll.

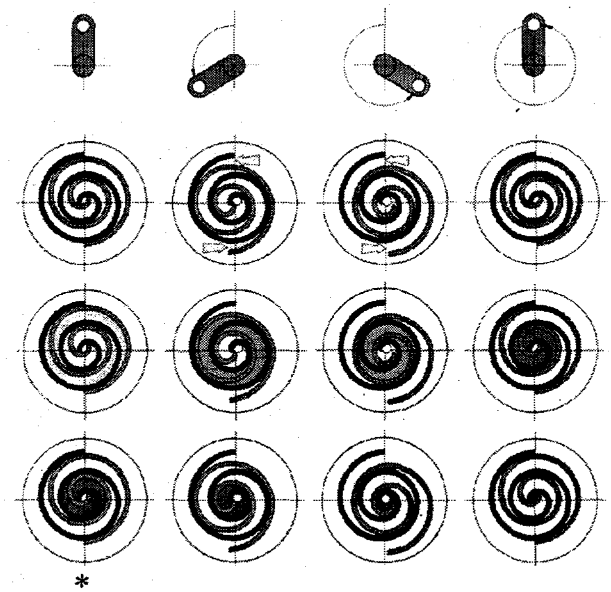


Figure 2-1: Scroll Theory of Operation

3. Program Plan

3.1 Technical Program

As in any research and development effort, there are technical risks involved. Arthur D. Little structured the scroll CEM development program in such a way that these risks were minimized, and so that the program direction could be modified if a given technical path was determined to contain excessive risks. This risk mitigation was accomplished by employing two key program elements.

- Two parallel technology paths were pursued initially, with a downselect to a single path once early test results from experimental hardware were known.
- The scroll hardware was designed and fabricated on a progressive basis, starting with the most technically risky components. Critical component testing enabled an early assessment of these components.

The structure of the early portion of the program is outline in Figure 3-1.

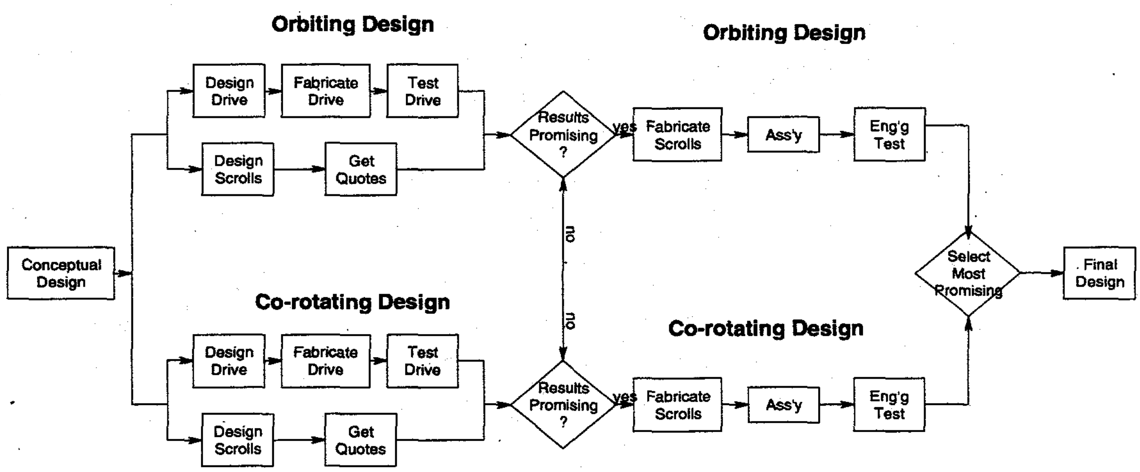


Figure 3-1: Scroll Compressor/Expander Program Flow Diagram

There are two general classes of scroll machinery, orbiting and co-rotating. Each technology has advantages and disadvantages depending upon the application. Both technologies utilize two scrolls, interleaved to provide pockets of contracting (or expanding) volume; the difference is in the motion used to position the scrolls relative to each other. In the orbiting case, one scroll is fixed while the other scroll moves in an orbital motion, accomplishing the compression. In the co-rotating case, both scrolls rotate in the same direction, each rotating on an axis displaced from the other by a distance equal to the orbit radius. Although the motion of the scrolls appears to be rotary to the casual observer, the relative motion between the scrolls is orbital.

The orbiting technology is the more developed of the two, with tens of millions of scroll air conditioning compressors and air compressors in service. The orbiting design is limited in speed, however, due to the large inertial loads generated by the orbiting mass. The co-rotating technology is less developed, although Arthur D. Little has designed air conditioning and other compressors employing this technology, and commercial air and air conditioning compressors are beginning to enter the market. The co-rotating design can potentially operate at greater speeds than the orbiting design, resulting in a much smaller and lighter package. This smaller package would be ideal for the transportation application.

As shown in the program flow diagram , Figure 3-1, Arthur D. Little pursued the orbiting design as the primary technical approach. In parallel with this we also pursued the co-rotating design as an alternate. For both designs, we initially concentrated the design effort on the drive mechanism, the most critical component of the CEM module. If both approaches looked promising after testing of the drive mechanisms, then both approaches would be continued to the next level, which is fabrication and testing of the drive mechanism with scrolls attached. The test results from the scroll/drive testing would then be compared, and a downselect to the most promising approach conducted. The selected technical approach would then be followed to the end of the program.

If testing of either the orbiting or co-rotating drive mechanism indicated potential problems, then the program would follow the path of the most promising approach. In this manner, we would preserve the options for program redirection until testing verified a particular technical approach. As described in detail later in this report, the co-rotating concept was judged to be have a risk level significantly higher than the orbiting design, and as a result the orbiting design path was followed to the conclusion of the program.

3.2 Program Organization

The organization chart for the Scroll CEM program is presented in Figure 3-2. A detailed description of the roles and responsibilities of each team member appears in Table 3-1. As can be seen from the table, the team consisted of experienced mechanical engineers and managers that have previously worked together on many successful scroll development programs in the past. Mr. John McCullough is a former Arthur D. Little employee that has formed a new company, Scroll Corporation. Scroll Corporation is a subcontractor to Arthur D. Little under this effort. Mr. McCullough has worked closely with the team members for many years as an Arthur D. Little employee.

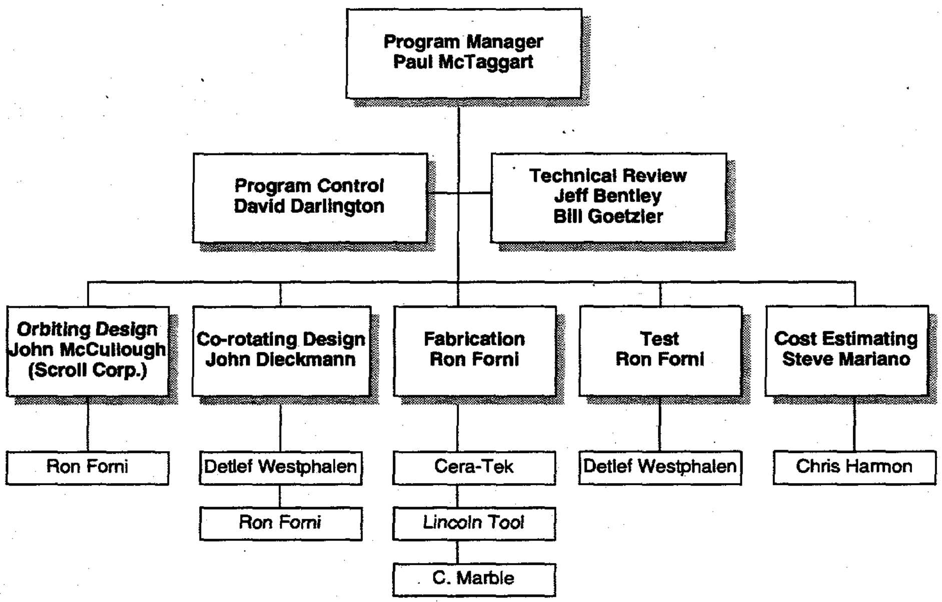


Figure 3-2: Scroll CEM Development Team

3.3 Cost Sharing

The Arthur D. Little team shared the cost of the Scroll CEM program with the Department of Energy, by providing funding for fifteen percent of the cost of the program. Both Arthur D. Little and Scroll Corporation, our largest subcontractor on the program, are participating in the cost by funding fifteen percent of the work that they perform respectively.

Table 3-1: Scroll CEM Program Roles and Responsibilities

Name	Role(s)	Responsibilities
Paul McTaggart	Program Manager System Integration	• Overall responsibility for all aspects of the program • Coordinate all design, fabrication, and test activities • Oversee Scroll system design • Coordinate contract documentation and deliverables
David Darlington	Program Control	• Cost and Schedule tracking and reporting • Tracking of reporting requirements
Jeffrey Bentley	Program Reviewer System Analysis/ Requirements	• Oversee technical, financial, and schedule performance • Provide input to system requirements evaluation • Provide input to CEM operational requirements definition
William Goetzler	Technical Reviewer System Analysis/ Requirements	• Oversee technical performance • Provide input to system requirements evaluation
John Dieckmann	Co-rotating Scroll Design	• Primary responsibility for design of co-rotating Scroll CEM
Ronald Forni	Scroll and System Design and Fabrication System Test & Evaluation	• Thermal and structural analysis including FEA • Assist in Scroll design and performance analysis • Oversee system testing and evaluation
John McCullough (Scroll Corporation)	Scroll and System Design and Fabrication System Test and Evaluation	• Primary responsibility for orbiting design of Scroll CEM • Supervise drawing package production • Supervise fabrication and assembly • Supervise day-to-day testing • Implement design modifications/improvements
Detlef Westphalen	Scroll Analysis and Test	• Scroll system models and efficiency predictions • Assist in co-rotating Scroll design • Test plan development
Steve Mariano	Cost Estimating	• Estimate manufacturing costs of Scroll compressor/expanders

3.4 Program Schedule

The project schedule, showing all tasks and key milestones, is shown in Figure 3-3. The schedule was extended three months from the original schedule in order to continue overlap with a parallel fuel cell development effort, under a no-cost schedule extension granted by DOE. The program was completed on-schedule and on-budget based on this revised schedule.

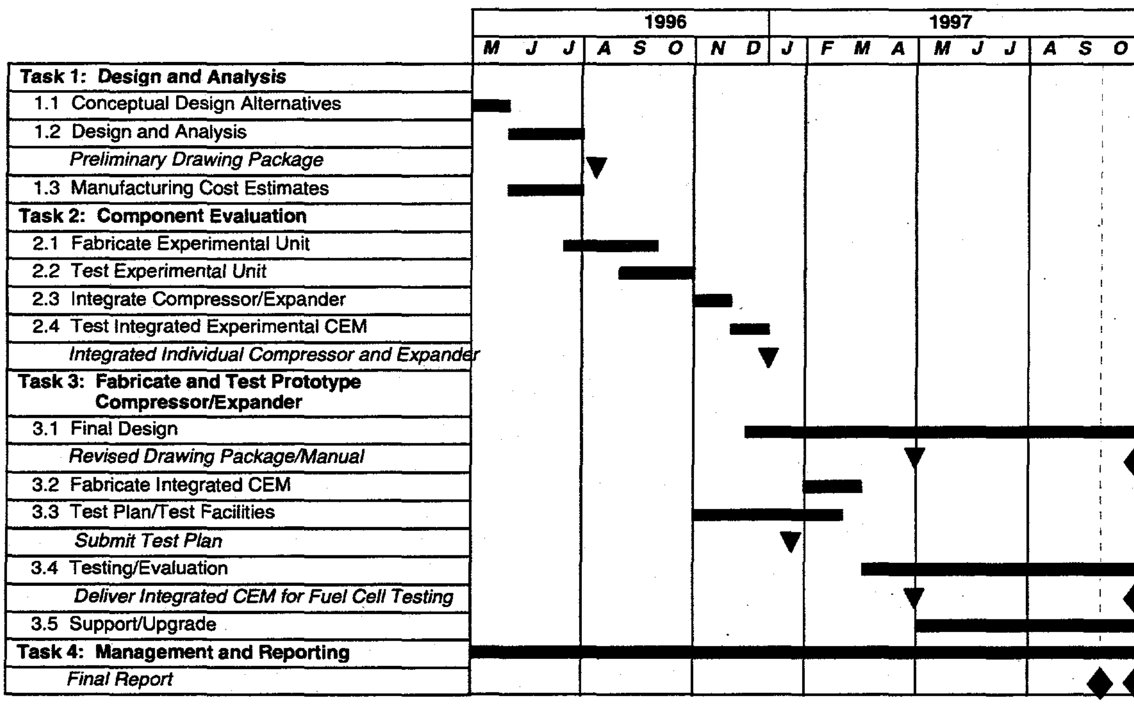


Figure 3-3: Scroll CEM Program Schedule

4. Scroll CEM Conceptual Design

4.1 Generation of Design Concepts

Arthur D. Little's approach to developing design concepts for the scroll compressor/expander was to have J. Dieckmann, R. Forni, and J. McCullough, all of whom are experienced Scroll machinery designers, independently develop conceptual ideas for the Scroll Compressor/Expander. These concepts were developed to the conceptual layout level, and presented to the team on April 18, 1996. Tradeoff studies were conducted by the team on each concept, considering efficiency, size, weight, cost, risk, and other criteria, in order to select the most promising concept or combination of concepts. Once the most promising concepts were identified, more detailed analysis and design of the concepts was begun.

These conceptual designs were grouped into two classes of scroll machinery, orbiting and co-rotating. The orbiting design consists of the compressor and expander scrolls on either side of a compact, centrally-located drive mechanism. This drive mechanism contains bearings and a crank to correctly orient the orbiting scrolls during operation, and a counterweight to balance the inertial forces generated by the orbiting scrolls. A drive belt transmits power from the electric motor to the drive mechanism. Two co-rotating designs were presented, both involving scrolls which both rotate in the same direction, but on axes that are offset by an amount equal to the orbit radius. Although each scroll is rotating, the relative motion between them is orbital. The electric motor is sandwiched between the compressor and expander scrolls, resulting in a compact, integrated package. The two versions of the co-rotating design differ primarily by the drive method and bearing locations.

Tradeoff studies were conducted by the team on each concept, considering efficiency, size, weight, cost, risk, and other criteria, in order to select the most promising concept or combination of concepts. The result of the tradeoff study was that while the orbiting and co-rotating designs each have advantages and disadvantages, there was not a clear-cut choice between the two. The consensus resulting from the meeting was to continue with the conceptual designs of each type of scroll machine, but also conduct some near-term experiments to test the validity of some of the unproven concepts, such as the co-rotating drive mechanism. This early test of critical components is an effective risk management technique Arthur D. Little has utilized on prior programs.

4.2 Co-rotating Design Concepts

The co-rotating scroll compressor/expander design is one in which both of the scrolls rotate synchronously in the same direction. The axis of rotation of each scroll is offset relative to the other by a distance equal to the orbit radius of the orbiting design. The result is that the relative motion between the scrolls is the same as for the orbiting system, even though there is no orbital motion of a single component. Since the motion of both scrolls is rotary, no inertial loads are carried on bearings, and in theory the co-rotating scroll CEM can achieve higher speeds than the orbiting scroll CEM, resulting in a smaller, lighter, and more compact unit. Because of these potential benefits, a co-rotating design was investigated in parallel with the orbiting design as part of the scroll compressor/expander development program.

In all co-rotating scroll designs, the scrolls must be synchronized very closely, otherwise the scrolls would contact each other causing wear or possibly damage. Three concepts for synchronization of the co-rotating design were explored during the program:

- **Universal Coupling** - a shaft with two universal joints, connecting the two scrolls
- **Clutch Teeth** - teeth, similar to gear teeth, that are machined into the scroll involutes or disks, which fix the angular position of the scrolls relative to each other.
- **Metal Bellows** - a bellows arrangement that connects the two scrolls, maintaining torsional rigidity while allowing a radial offset

A sketch of a co-rotating scroll concept with a universal coupling is shown in Figure 4-1 below. The figure shows one half of the device--the other half is a mirror image.

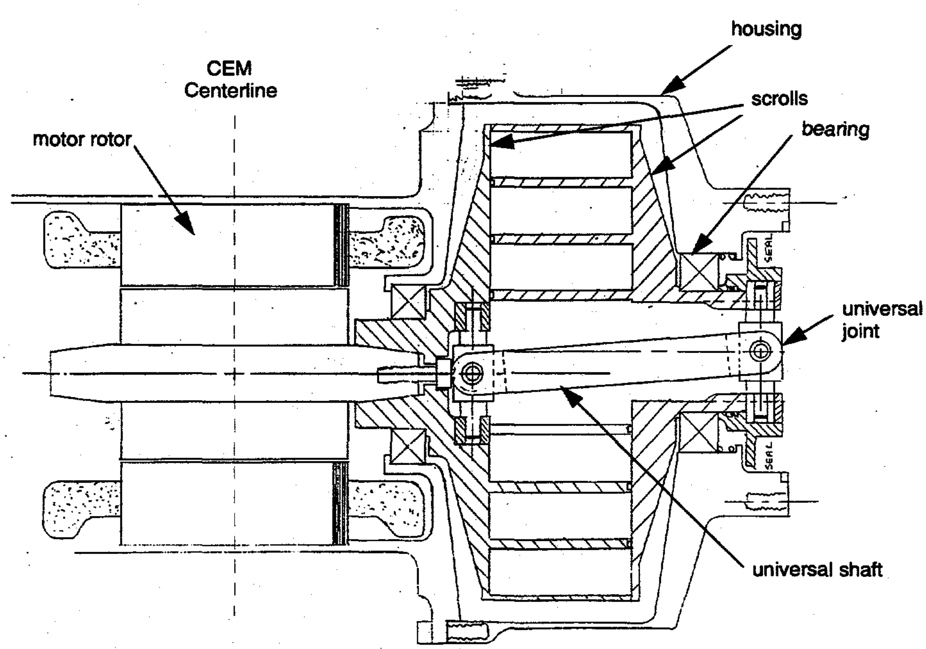


Figure 4-1: Initial Co-rotating Scroll Concept Sketch

The compressor and expander are mounted on opposite ends of a shaft which doubles as the rotor for the intended brushless DC motor. For the compressor, air enters at the outer edge of the scroll and is discharged through the hollow shaft which supports the outer scroll half. The air flow direction is in reverse for the expander. The scroll diameter is about 8 inches, the displacement is about 30 cubic inches, and the design speed is about 7,500 rpm.

The integration of the motor and scrolls reduces motor speed selection flexibility and increases potential for high temperature at the shaft bearings. Despite these limitations, the design saves space, eliminates potential losses associated with a belt or gear drive, and simplifies construction. Neither of the drawbacks was judged to represent an insurmountable problem. More challenging aspects of the design are the drive system and bearing system for the outer scroll halves. Figure 4-1 shows a universal joint drive and a single radial contact bearing. The single radial contact bearing is a simplification: the shaft will have to be cantilevered with two bearings, which will also have to provide some thrust (a significant portion of the outer scroll thrust force can be supported by appropriate sizing of the discharge-end shaft seal).

A second co-rotating scroll CEM concept is shown in Figure 4-2. This concept is based on an integral centrally-located electric motor, similar to the previous concept, but with the scrolls synchronized by "clutch-teeth" similar to gear teeth that are machined into an extension of the outer diameter scroll surfaces. This illustration also shows the full scroll CEM concept, in which the compressor and the expander are located at opposite ends of a centrally located electrical motor, integrally connected to the scrolls.

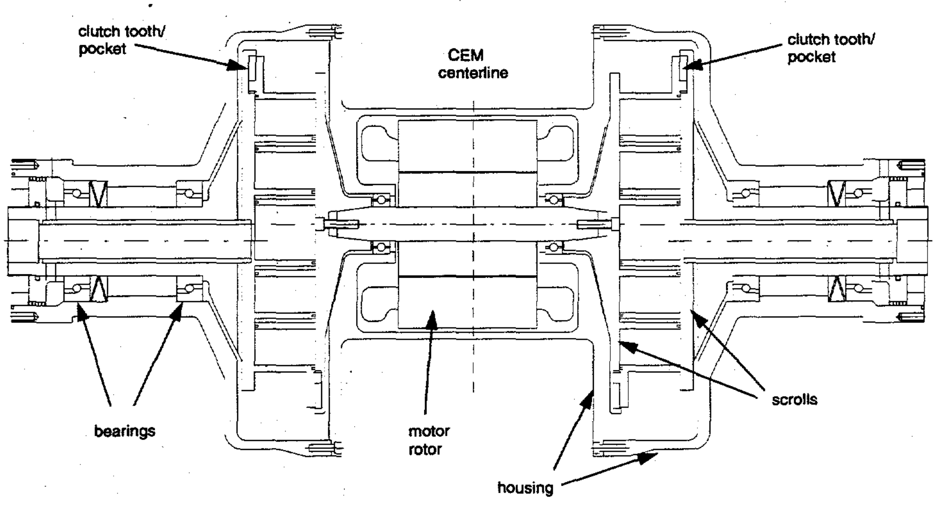


Figure 4-2: Co-Rotating Scroll CEM with Gear Drive

Figure 4-3 shows the details of the clutch teeth drive. A series of teeth on one rotating scroll meshes sequentially with mating pockets on the other rotating scroll. Because of the offset axes of the rotating scrolls, the teeth orbit relative to the pockets, so that the teeth go through one cycle of engagement and disengagement for each revolution of the scroll CEM. At any one point in time, there are several teeth engaged.

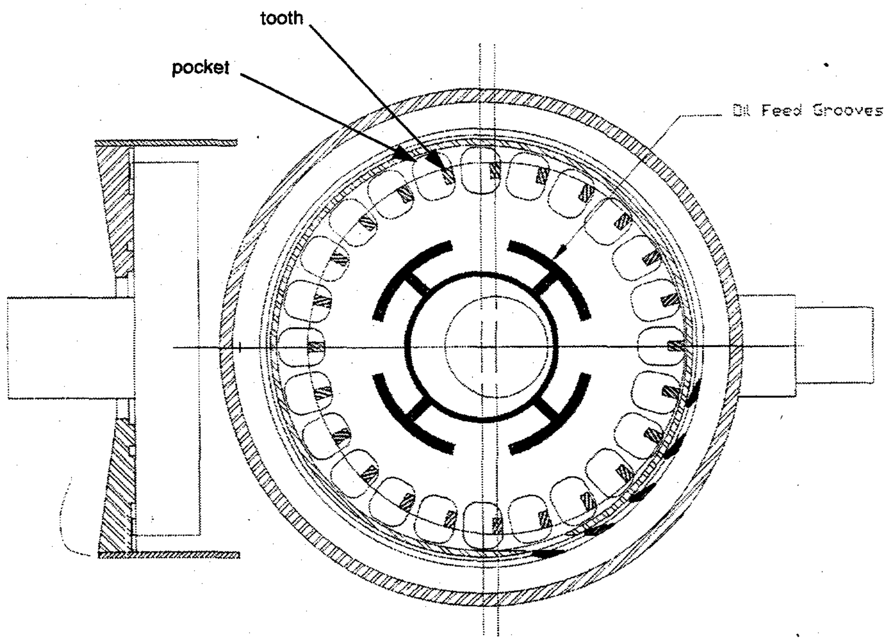


Figure 4-3: Clutch-Tooth Drive Mechanism

A third co-rotating scroll CEM design concept is shown in Figure 4-4. This diagram illustrates one half of a mirror-image CEM that employs a bellows drive concept.

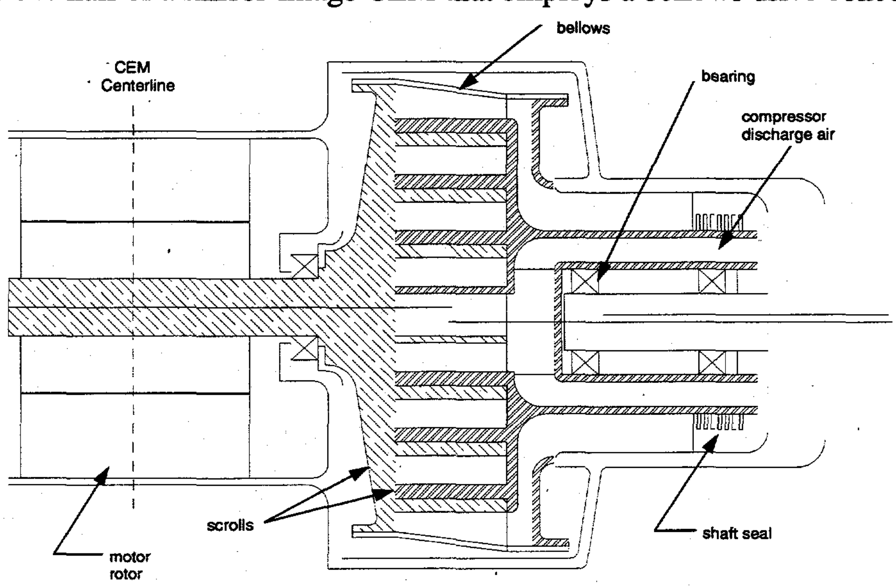


Figure 4-4: Co-Rotating Scroll with Bellows Drive

In this version of the co-rotating scroll CEM design, the left-hand scroll is attached to a solid shaft that is also the rotor of the electric drive motor. The right-hand scroll is attached to a hollow shaft through which the compressed air leaves the scrolls. The gas passage is formed by the area between two concentric tubes that make up the shaft. The bearings are rolling element bearings that are attached to the inside surface of the inner tube of the shaft. In the example shown, synchronicity between the co-rotating scrolls is maintained by a metal bellows that connects the outer circumference of both scrolls. The metal bellows maintains torsional rigidity, but allows radial flexibility to accommodate the offset axes of rotation of the co-rotating scrolls.

4.3 Orbiting Scroll CEM Design Concepts

The initial design concept for the orbiting scroll CEM is illustrated in Figure 4-5. A central drive mechanism serves as the mounting surface for the compressor and expander, which are mounted on opposite sides of the drive, respectively. At the center of the drive is a crankshaft, which is driven by a toothed belt which is in turn driven by an electric motor, providing shaft power to the scroll CEM.

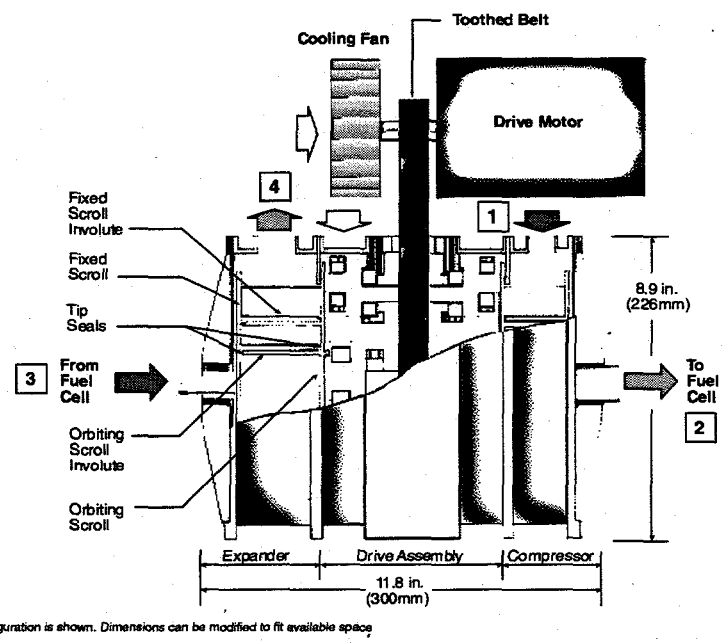


Figure 4-5: Orbital Scroll CEM Design Concept

The crankshaft drives two driveplates, to which the orbiting scrolls are fastened. The driveplates are restrained from rotating about the crankshaft by six rollers, which also act as bearings, taking the radial inertial loads of the CEM. The combination of the driveshaft and rollers constrain the driveplates to an orbital motion. The inertia of the entire orbiting mass is balanced by a counterweight which itself orbits 180 degrees out-of-phase with the orbiting components. The fixed scrolls are mounted to the housing of the drive mechanism, which in turn is mounted to a fixed base.

The inlet air to the compressor enters at the outer diameter of the scrolls, and travels toward the center of the scrolls as it is compressed. The expander gas air flow path is opposite that of the compressor.

5. Critical Component Testing

5.1 Reasons for Critical Component Testing

As discussed previously in this report, Arthur D. Little pursued the orbiting design as the primary technical approach. In parallel with this effort, the co-rotating design was pursued as an alternate. For both designs, we initially concentrated the design effort on the drive mechanism, the most critical component of the CEM module. If both approaches appeared promising after testing of the drive mechanisms, then both approaches would be continued to the next level of development, which is fabrication and testing of the drive mechanism with the scrolls attached. The test results from the scroll/drive testing would then be compared, and a downselect to the most promising approach conducted. The selected technical approach would then be followed to the end of the program.

If testing of either the orbiting or co-rotating drive mechanism indicated potential problems, then the program would follow the path of the most promising approach. In this manner, we preserved the options for program redirection until testing verified a particular technical approach.

5.2 Co-rotating Component Testing

In order to validate the critical design areas of the co-rotating design, several bench-top, breadboard tests were conducted. To accomplish the testing , a test fixture was constructed to simulate the operating conditions of the co-rotating CEM, including the thrust load that would be exerted on the bearings due to the gas pressures inside the scrolls. The test fixture consisted of two shafts, with bearings, offset by a distance equal to the design orbit radius. One shaft was driven by a one horsepower electrical motor. The simulated radial load was supplied manually, using a lever system, which provided a force attempting to force the two scrolls apart radially. The force was measured with a spring scale. This fixture was used to test concepts for both the bearing and timing concepts.

The investigation into the feasibility of a universal joint as a co-rotating drive alternative consisted of several tests and analyses.

- Bench-top testing of a conventional Lovejoy pin-and-block universal joint at speeds up to 5,500 rpm indicated that the initial angular slip of 1.25° did not increase significantly after a number of run hours (at least 12). The test was terminated at this point because of bearing problems rather than U-joint problems. However, this initial angular slip value was judged to be excessive for purposes of a scroll drive.
- Multiple inquiries were made with suppliers in an attempt to find a universal joint with the required compactness and the ability to operate at the desired speeds. A U-joint angle of 6° was specified. None of the suppliers had any suitable products. U-

joints with needle bearings were considered too bulky for our application (none were rated for 7,500 rpm), and pin-and-block U-joints were estimated to wear too quickly at the required speed.

- A PV analysis was done to determine whether a pin-and-block U-joint with reduced-wear surfaces or a clutch-tooth drive would have the required wear life. This analysis indicated the following wear life estimates (assuming maximum angular slip 0.36°; wear factor of 10^{-9} in/hr-psi-fpm):

Universal Joint:	170 hours
Clutch-Tooth:	750 hours

The life may be extendible by a factor of up to 10, depending on the materials chosen. However, the cost and availability of the best materials was determined to make such improvement impractical.

One of the key difficulties of the co-rotating concept is the need for large-diameter bearings for the outer scroll shafts, because the shafts double as compressed air conduits. The initial focus of the component testing was on grease-lubricated bearings to reduce requirements for sealing of the bearings from the compressor inlet air stream. The investigation of rolling element bearings included several elements:

- Initial testing of a low-cost set of sealed, grease lubricated bearings at up to 5,500 rpm was conducted. These tests showed that overheating of the bearings is a problem for readily-obtainable, off-the-shelf bearings at the required speed, even at room temperature ambient conditions.
- Multiple inquiries were made with suppliers to provide appropriately-designed sealed, grease lubricated radial-contact bearings for the application, assuming that the bearings would have to operate at 300°F (the temperature of the gas inside the hollow shaft would be somewhat higher) . The bearings would have to be special-orders with high-temperature greases and heat treated races. None of the suppliers were willing to provide a quote.
- An inquiry was made with Barden, a manufacturer of precision bearings (ABEC-7 or better). Discussions with Barden involved specification of minimum axial and radial deflections (set at 0.001" for the preliminary inquiry) for expected loads, and a reduction in the operating temperature to 200°F. A bearing design using preloaded angular contact bearings was proposed, but cost and delivery time were too high. In any case, the bearings were too large, and at least one more round of bearing selection by Barden would have been required in order to select a bearing set with tolerable deflection and more reasonable size.

Oil film bearings were investigated as an alternative to rolling contact bearings. A spindle for support of the outer scroll halves was designed, including two 2" diameter x

0.75" long journal bearings and a 2" ID x 3" OD thrust bearing. The power loss for this bearing system at the design speed is estimated as 1.2 hp per spindle, assuming that the thrust bearing will carry only one third of the 300 lb. axial load. These high losses, due to the large shaft diameter, would make achieving the system efficiency goals impossible. In addition, the use of oil film bearings increases the sealing requirements of the shaft seals, which prevent penetration of lubricating oil to compressor air inlet.

Conclusions and additional issues regarding the drive mechanisms are listed below.

- Finding a commercially-available U-joint with the required speed capability may be possible with further research, but would require a reduction in the U-joint angle and as a result a product which is larger than desirable. Even with the reduced angle, the wear life of workable U-joints was judged to be insufficient.
- Constructing a U-joint with low-wear materials is also possible. However, the designed U-joint is somewhat longer than desired and has a larger swing diameter than desired (the outer connection of the joint with the shaft would be on the outside end of the shaft, thus increasing the shaft assembly length).
- Extensive testing of any proposed U-joint would be required. This was stressed by the suppliers who indicated they had a product which comes close to our speed requirements.
- The U-joint shaft takes up space in the hollow shaft supporting the outer scroll half, significantly reducing flow area. Increasing the hollow shaft diameter to compensate would exacerbate the bearing problems.
- The clutch teeth drive system has been successfully used for smaller compressors which use oil for lubrication. Adaptation to the air compressor would require low-wear materials, and would have to be tested. Such a design would increase overall system diameter, but would ease space limitations in the hollow shaft supporting the outer scroll half.

5.3 Orbiting Scroll Component Testing

As in the case of the co-rotating design, the design and test of the critical components of the orbital design was conducted in order to provide an early assessment of its viability. This critical component design and test phase was conducted in parallel with the co-rotating effort, with a downselect to the most promising concept after initial test results were complete. The central drive mechanism and its internal components are the most critical parts of the orbiting design, and near-term testing of this component will allow an early indication of any problems that might occur with the design.

5.3.1 Design and Fabrication of Drive Mechanism

Figure 5-1 shows a diagram of the key features of the drive mechanism. At the center of the drive is a crankshaft, which is driven by two toothed belts which are in turn driven by a jackshaft external to the drive mechanism. This jackshaft is driven by an electric motor, which provides shaft power to the scroll CEM. The crankshaft drives two driveplates, to which the orbiting scrolls are fastened. The driveplates are restrained from rotating about the crankshaft by six rollers, which also act as bearings, taking the radial inertial loads of the CEM. The combination of the driveshaft and rollers restrain the driveplates to an orbital motion. The inertia of the entire orbiting mass is balanced by a counterweight which itself orbits 180 degrees out-of-phase with the orbiting components

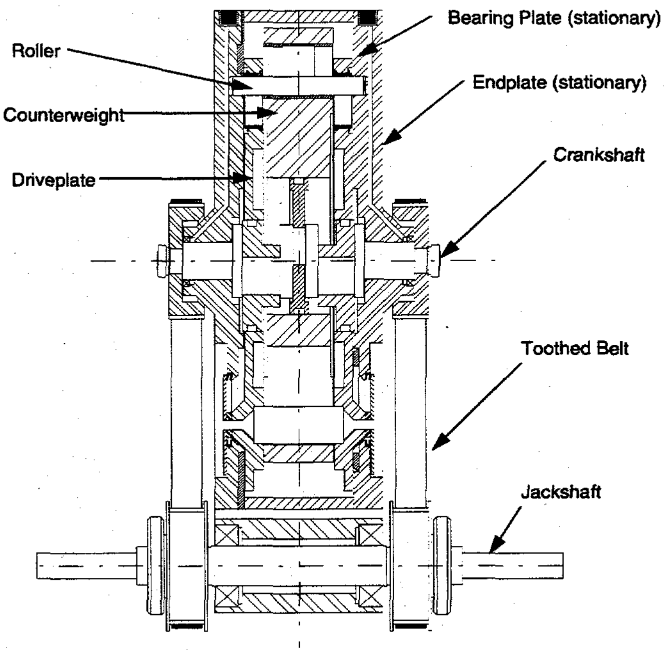


Figure 5-1: Orbiting Scroll CEM Drive Mechanism

The detailed design of the central drive mechanism was conducted and completed, including a complete set of engineering drawings from which the parts were fabricated. Once the components of the drive mechanism were fabricated and delivered, the parts underwent inspection, and then assembly. An oiling system for the drive, which consists of small diameter piping and a small electrically-driven oil pump was designed and fabricated. Assembly of the drive test apparatus, which included a small drive motor, was completed in preparation for the initial operational tests. The drive mechanism was then spun slowly, to determine the frictional load of the drive. Several tests were run with various types of lubricant, to determine the effect of viscosity on the mechanical efficiency of the drive mechanism. The starting torque was measured in each case. The drive unit was then disassembled to determine if there was any contact between the drive components.

5.3.2 Orbiting Scroll Drive Testing

Once the preliminary checkout tests were completed, the drive unit was spun up, initially at low speeds, to subsequently higher speeds. Initially, the drive was spun to approximately 2000 rpm, without any problems. After conducting several additional test runs, a slight tapping, or knocking, sound from within the drive mechanism was noticed. After disassembling the drive, it was found that one of the steel bearing sleeves, which are press-fitted into the aluminum drive plate, had worked its way partially out of the drive plate so that a portion of the sleeve projected slightly beyond the drive plate. This projecting section was contacting the adjacent component of the drive mechanism, causing the noise. A consultation with the machinists indicated that they had machined these parts with a press fit less tight than specified, because they were concerned that the aluminum drive plate might distort. To correct this, the length of the sleeves was shortened slightly, and the sleeves sent out to be plated with an electro-less nickel coating, to build up the outer diameter of the sleeves so that the press fit would be tighter. Once the parts were received from the plater, the machinists reinstalled the sleeves and assembled the drive mechanism. The sleeves were installed with Locktite as an added precaution. A power analyzer was installed so that the power required by the drive motor could be determined, to give an indication of the amount of power that the drive mechanism is absorbing.

Initial testing indicated that the problem had been corrected, and we increased the speed of the mechanism to 3600 rpm, during which no problems were encountered. The test rig was then shut down for inspection, and when restarted exhibited the same tapping sound as before. The mechanism was then disassembled, and upon inspection it was found that several of the steel sleeves had once again worked their way slightly out of the bores. After discussion among the team members, it was decided that a more secure method of retaining the sleeves was required. A new type of sleeve was designed with a shoulder on one side and a retaining clip on the other side, to mechanically hold the sleeve securely in place. The power observed during the test was very close to the drive losses we used in our performance model, validating this part of the model.

Once the redesigned sleeves were received, the parts were installed in the drive plates, and the drive mechanism was assembled. We then resumed testing of the drive mechanism. This testing confirmed that the redesign of the bearing sleeves had corrected the problem with the sleeves working their way out of the drive plate during operation. During testing we noticed that the lubricant we are using inside the drive mechanism (automatic transmission fluid) was leaking from some of the seals at the points at which the orbiting scroll attaches to the drive mechanism. These seals are aluminum disks which have a circular nylon seal at the outer edge. It appears that there is not enough axial force to prevent leakage on several of the seals. We redesigned the seals using spring-loaded Teflon seals, which seal against the inside of the housing. In this design, both the spring in the seal and internal pressure inside the drive mechanism force the seal axially against the housing, providing a leak-free seal.

6. Design of Integrated Scroll CEM

6.1 Downselect to Orbiting Scroll CEM

Based on the results of the breadboard testing and analysis of the co-rotating design concepts, further development of the co-rotating concept was halted. This decision does not mean that construction of a co-rotating compressor is not feasible. Rather, there are a number of design challenges that must be overcome prior to the development of a reliable co-rotating scroll compressor/expander.

- Identification of an appropriate supplier of ball bearings. The bearings must be able to handle the required speeds, provide sufficient stiffness, be of small size, and have reasonable cost.
- Design and selection of the appropriate drive mechanism. Work on designing and testing an optimized drive mechanism would have to be continued. Tradeoffs between U-joint and clutch-tooth drives and among different low-wear materials must be further evaluated.
- Minimizing scroll deflections. Initial investigation indicated that required flank and axial clearance gaps could be large enough to have a significant negative effect on efficiency. Improvement should be possible, but would require significant development testing.

Because of these challenges, the co-rotating scroll concept was determined to have a significant level of technical risk. When the risk levels between the co-rotating and orbital design were compared at this point in the program, the orbital design was judged to be of lower risk, and hence was chosen as the development path for the remainder of the program.

6.2 Orbiting Scroll CEM Design

6.2.1 Overall Configuration

A schematic of the orbiting scroll CEM is shown in Figure 6-1 below. The compressor and expander are driven with a common drive shaft which is powered by an electric motor. Power for the motor will be provided by the fuel cell. The compressor provides combustion air for the fuel cell at the required fuel cell pressure of up to about three atmospheres. The fuel cell exhaust stream is used in the expander to reduce the required compressor power.

In the orbiting scroll CEM, the compressor and expander are both comprised of two scrolls, one of which orbits, the other being stationary. Both orbiting scrolls are connected to the driveshaft through an orbital drive transmission; the compressor orbiting scroll taking power from the drive mechanism, and the expander orbiting scroll providing power. The orbiting scrolls are inside the scroll housings.

The CEM is housed in an aluminum enclosure for noise isolation and for control of cooling air flow. The drive motor is located outside the enclosure. A penetration in the enclosure is provided for the drive shaft.

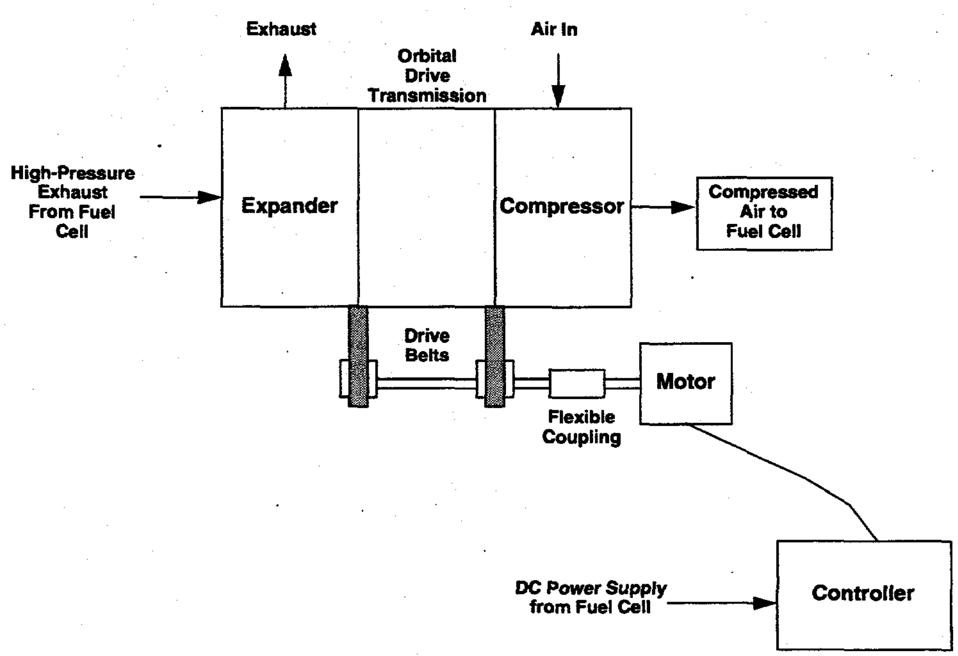


Figure 6-1: Compressor/Expander Module Schematic

Over the course of the design effort, the scroll CEM concept was continuously refined and detailed. The result of this design effort was a complete set of engineering drawings and a bill of materials of sufficient detail to have the components fabricated at a competent machine shop. Figure 6-2 and Figure 6-3 show two views from the assembly drawing. Table 6-1 contains the bill of materials necessary to assemble the scroll CEM.

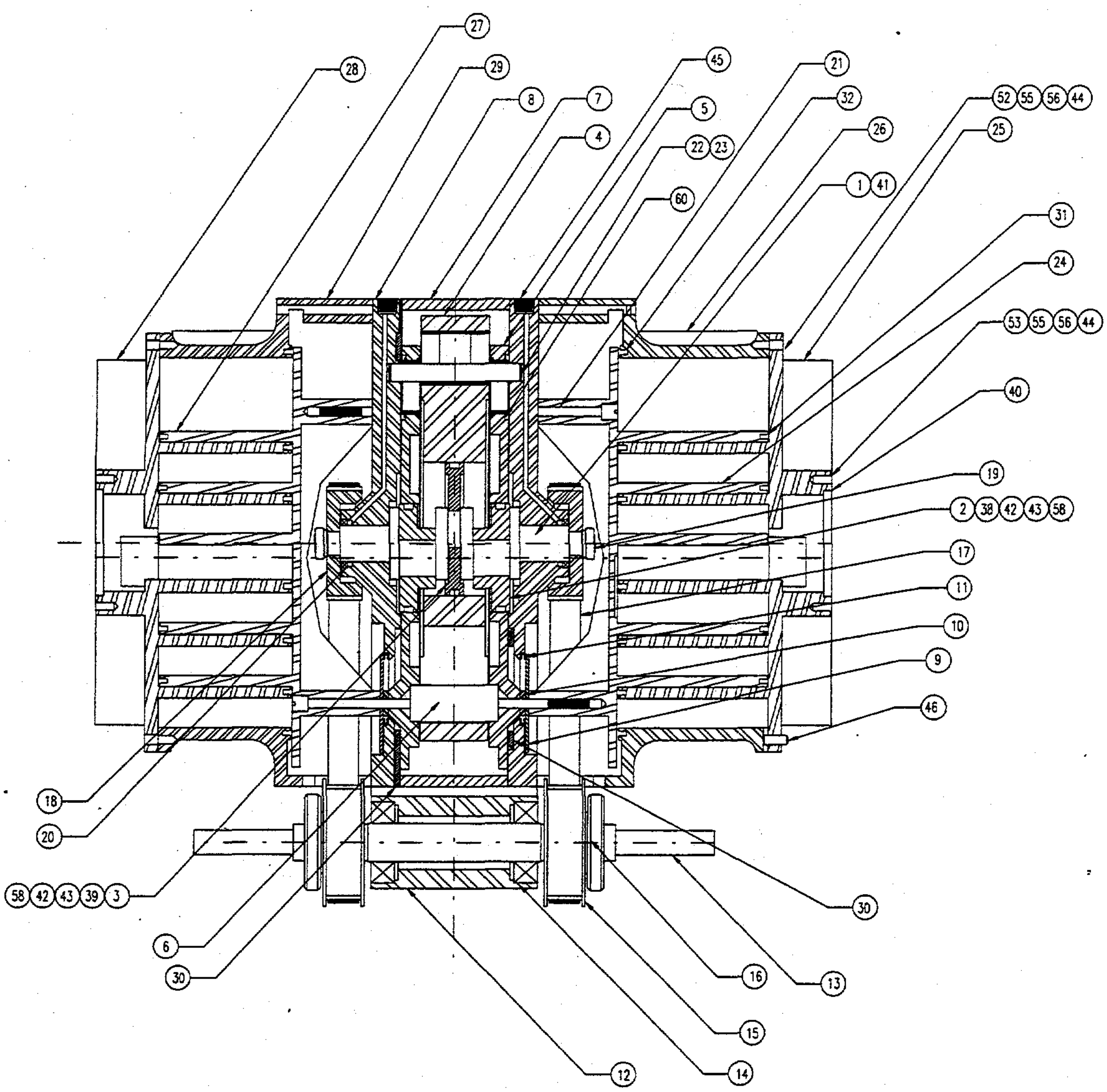


Figure 6-2: Integrated Scroll CEM Assembly- Top View

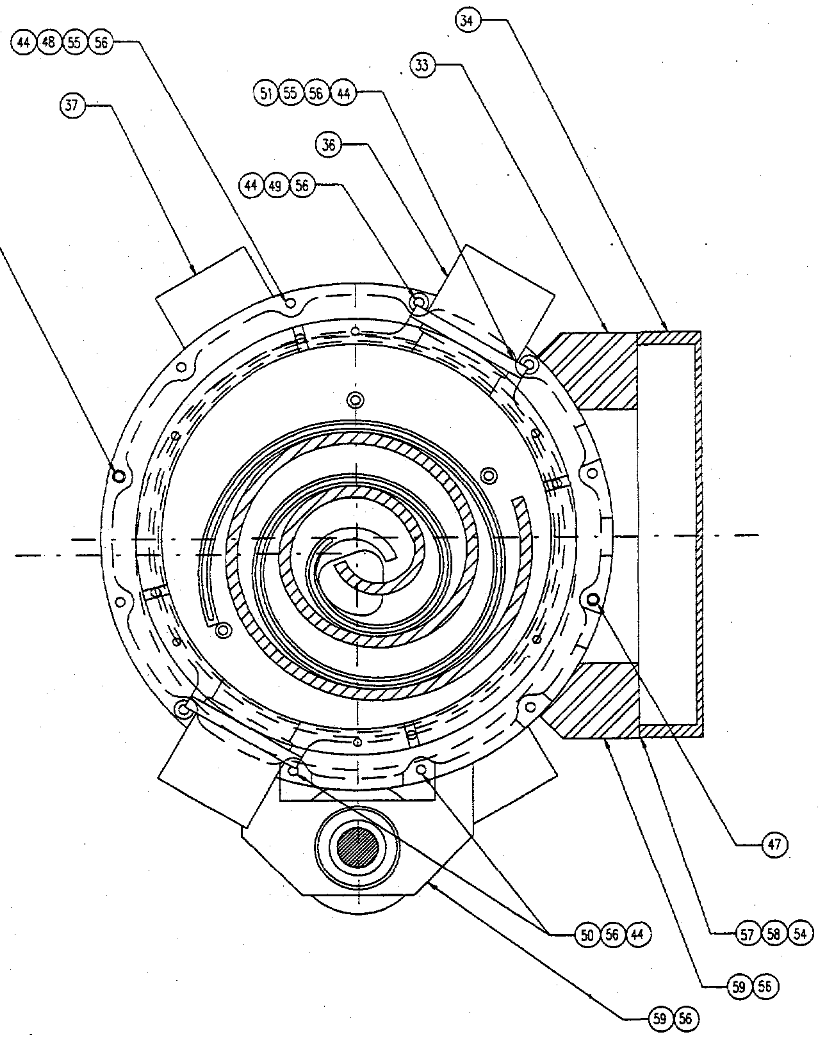


Figure 6-3: Integrated Scroll CEM Assembly- Side View

Table 6-1: Bill of Materials for Scroll CEM

No.	Quantity	Part No.	Description
1	1	8016-048	CRANK SHAFT
2	2	8016-046	DRIVE FRAME BEARING WEB
3	1	8016-046	COUNTER WEIGHT BEARING WEB
4	1	8016-039	COUNTER WEIGHT
5	2	8016-044	DRIVE FRAME
6	6	8016-041	SUPPORT TUBE
7	1	8016-042	HOUSING BARREL
8	2	8016-056	HOUSING DISC (MARK 2)
9	12	8016-040	ORBITAL SEAL PLATE
10	12	8016-040	ORBITAL SEAL SPACER
11	12	S14LB1.5GFP	ORBITAL SEAL (BAL SEAL)
12	1	8016-047	COUNTER SHAFT HOUSING
13	1	8016-047	COUNTER SHAFT
14	2	SKF 60042Z	COUNTER SHAFT BEARINGS
15	2	P38-5M-15	COUNTER SHAFT PULLEYS (GATES)
16	2	JA 3/4	COUNTER SHAFT HUBS (BROWNING)
17	2	500-5M-15	TIMING BELTS (GATES HTD)
18	2	P38-5M-15	CRANK SHAFT PULLEYS (GATES)
19	2	3/8 x 24	CRANK SHAFT NUT
20	2	71 X 7027	CRANK SHAFT OIL SEAL (GARLOCK)
21	6	8016-041	TIE RODS, SCROLL
22	6	8016-043	ROLLERS
23	24	8016-043	ROLLER SHIM
24	1	8016-052	COMPRESSOR ORBITING SCROLL
25	1	8016-054	COMPRESSOR FIXED SCROLL
26	1	8016-050	COMPRESSOR STAND OFF
27	1	8016-051	EXPANDER ORBITING SCROLL
28	1	8016-055	EXPANDER FIXED SCROLL
29	1	8016-053	EXPANDER STAND OFF
30	1 6		THRUST BEARING (MARK 2) THRUST BEARING (MARK 1)
31	4	.095 x .110	TIP SEAL
32	2	.095 x .110	PERIPHERAL SEAL
33	1	8016-045	UPPER OIL SUMP
34	1	8016-045	LOWER OIL SUMP
35	2		MOUNTING BRACKETS
36	2		COMPRESSOR HOSE ADAPTERS
37	2		EXPANDER HOSE ADAPTERS
38	2	# 226	DRIVE O-RING (VITON)
39	1	# 250	COUNTER WEIGHT O-RING (VITON)
40	2	# 136	PORT O-RING
41	2	1/8 X .375	PIN, CRANK SHAFT
42	4	4-40 x .75	SOCKET HEAD BOLT
43	4	5/0 x .75	TAPER PIN
44	64	10-32 x.29	HELICOIL INSERTS
45	22	1/8 NPT	PIPE PLUGS
46	8	3/16 x .5	DOWEL PINS

No.	Quantity	Part No.	Description
47	4	.25 x 1.0"	HOLLOW DOWEL PIN
48	14	10-32 x 3.0	SOCKET HEAD BOLT S.S.
49	6	10-32 x 2.5	SOCKET HEAD BOLT S.S.
50	4	10-32 x .75	SOCKET HEAD BOLT S.S.
51	16	10-32 x .5	SOCKET HEAD BOLT S.S.
52	12	10-32 x .62	SOCKET HEAD BOLT S.S.
53	12	10-32 x 1.3	SOCKET HEAD BOLT S.S.
54	20	4-40 x .5	SOCKET HEAD BOLT S.S.
55	54	10-32	FLAT WASHER BOLT S.S.
56	74	10-32	LOCK WASHER S.S.
57	20	4-40	FLAT WASHER S.S.
58	24	4-40	LOCK WASHER S.S.
59	10	10-32 x ?	SOCKET HEAD S.S.
60	12	VS-106-502	E-RING (SMALLEY)

6.2.2 Lubrication System

Lubrication for the CEM bearing surfaces is provided with a pressurized lubrication system, which is depicted in Figure 6-4 below. For prototype development the oil pump is driven by a 3600-rpm motor, powered by 100 W of 115-Volt 60-Hz single-phase power. The system uses Mercon III transmission fluid. The system is designed to operate with 30 to 50 psig oil pressure.

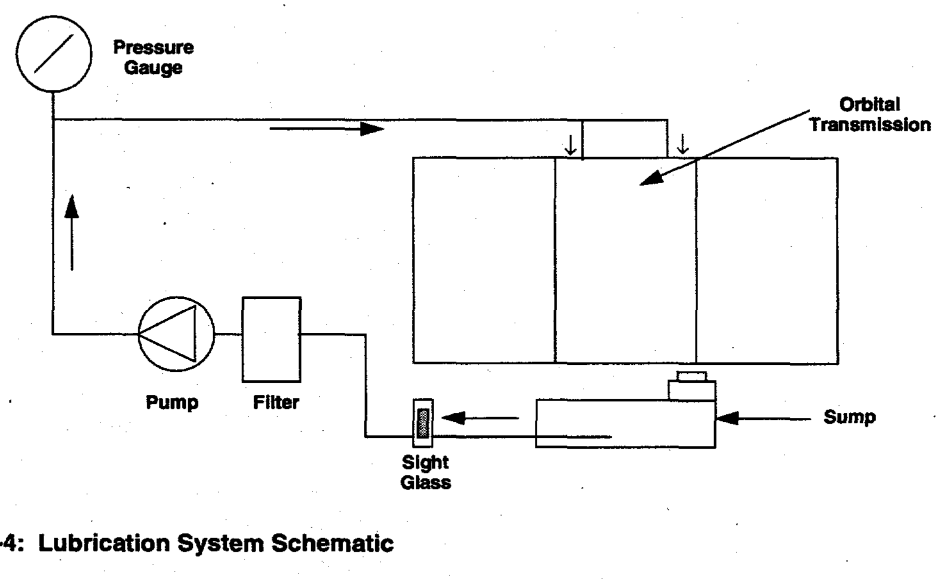


Figure 6-4: Lubrication System Schematic

6.2.3 Cooling System

Cooling of the external surfaces of the CEM is provided with ambient air which is drawn through the aluminum enclosure which surrounds the compressor/expander assembly. For prototype development an axial cooling air fan is powered by a single-phase motor drawing 50 W of 115-volt 60-Hz power. Cooling air flow rate is about 500 cfm. Operation of the system without cooling air flow will result in poor performance and may lead to damage of the machine.

6.2.4 Motor/Controller

The CEM is powered by a brushless DC motor. A controller is used for regulation of motor speed and power. The motor design specifications are shown in Table 6-2 below. Motor input power is assumed to vary as shown in the table due to fuel cell system characteristics.

Table 6-2: Motor Design Specifications

Speed (rpm)	Power (hp)	Torque (in-lb.)	Voltage
960	0.3	20	195
1620	1.1	43	195
2820	3.0	67	177
3960	5.9	94	160
5100	9.4	116	142
6180	11.3	115	125

The circuit schematic for the motor/controller system is shown in below in Figure 6-5.

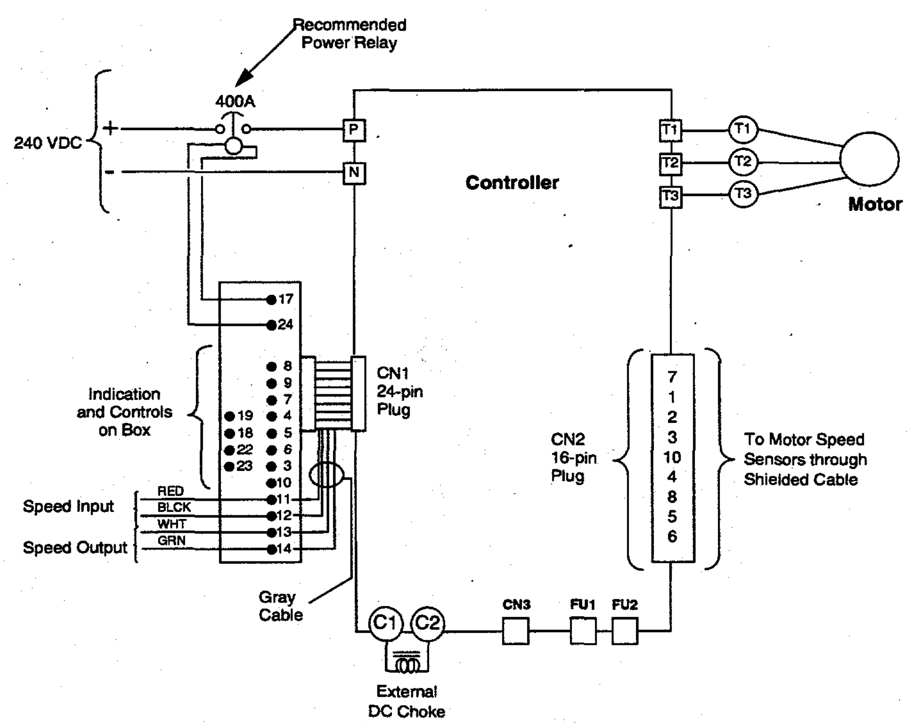


Figure 6-5: Motor/Controller Wiring Schematic

6.2.5 Orbital Drive Transmission

The orbital drive transmission converts the rotary drive shaft motion into the orbital motion required for orbiting scroll operation. The transmission system includes a counterweight which compensates for the inertial loads of the orbiting scrolls. The transmission also provides for properly-aligned axial and radial positioning of the orbiting scrolls.

6.2.6 Compressor and Expander

Both the compressor and expander have one orbiting scroll and one fixed scroll. The compressor's orbiting scroll is powered by the orbital drive transmission discussed above. The expander's orbiting scroll provides power to the transmission. The compressor inlet displacement is 27 cubic inches. The expander inlet displacement is 13.15 cubic inches. The built-in volume ratio of both components is 1.8.

6.3 Orbiting Scroll CEM Fabrication

Design of the scroll components was conducted in parallel with the testing of the orbital drive mechanism. Once the design was completed, engineering drawings of the scroll compressor and expander parts were submitted to several machine shops for quotation. We received quotations from two vendors, and released purchase orders for the scroll parts.

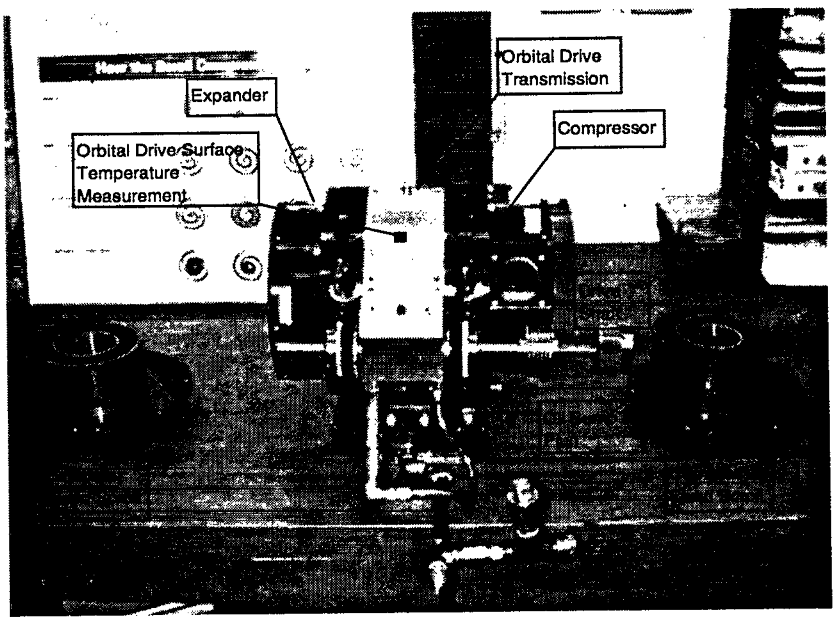


Figure 6-6: Compressor, Expander, and Orbiting Drive Transmission

Ron Forni worked closely with the machine shop in developing the programming for the CNC machines, so that they could cut the scroll involute profile into the front side of each scroll part. After the programming was completed, the machining of the scroll involutes was conducted. After the machining was complete, the parts were inspected, and sent out for Teflon hardcoating. Once all of the remaining components for the scroll compressor/expander were received, the parts underwent inspection at the machine shop. Once it was determined that the parts met the dimensional specifications of the drawings, the parts were trial assembled to the drive mechanism. The scroll assembly was then rotated by hand, to verify that the parts worked together as intended, with no contact between the scrolls.

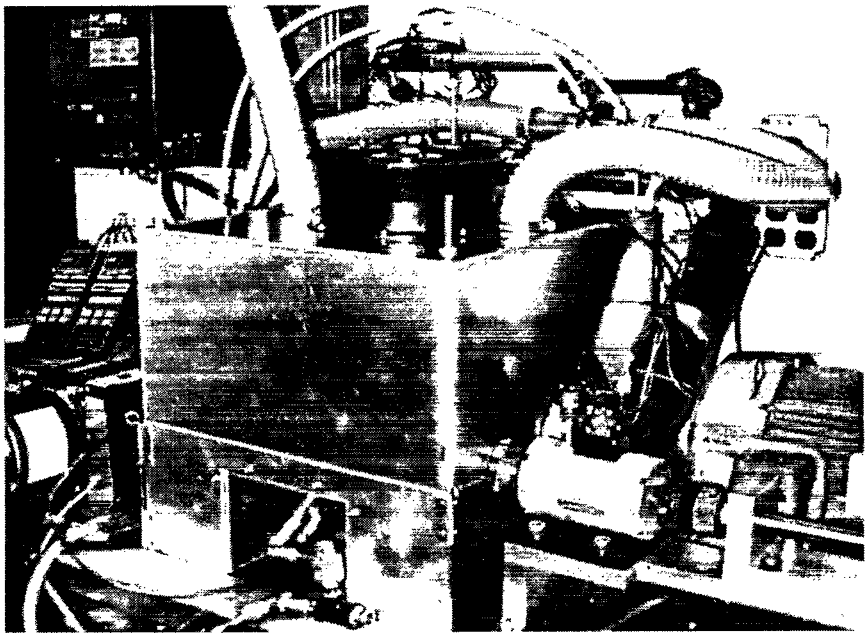


Figure 6-7: Compressor/Expander Module Assembly

6.4 Orbiting Scroll CEM Checkout Testing

The checkout procedure began by spinning the scroll compressor/expander assembly with the motor, starting at low speeds. The speed of the scrolls was incrementally increased, with periodic teardowns during which the parts were inspected for contact or wear. The orbiting scrolls were initially coated with a blue compound (Dykem) to detect contact if it occurred. As the test speeds were increased , the scrolls were coated with a molybdenum disulfide paint (Lubriplate) to prevent galling of the aluminum in case of contact. The checkout testing was done initially without the tip seals, to provide greater axial clearance should any contact between scrolls occur. The assembly was successfully run at speeds up to approximately 1800 rpm, without any noticeable problems.

The next step in the checkout procedure was to install the tip seals, and repeat the checkout runs. The scroll compressor/expander ran with the tip seals, without exhibiting any problems. Although not instrumented at this point, the scroll compressor/expander appeared to be accomplishing the compression and expansion process as designed. After running checkout tests at several different speed levels the scroll compressor/expander was disassembled for inspection. Upon disassembly it was found that the scrolls exhibited a minor amount of rubbing, indicating that the clearances were too tight in that area. This rubbing was determined to be due to a machining variation in the scroll housing disks. The clearance was adjusted using shims to correct the variation, and the testing was resumed. The scroll compressor/expander was installed in the test loop using preliminary instrumentation, so that performance could be measured. The measured performance indicated a low volumetric efficiency, which is consistent with the fact that the scroll was running with greater than design clearances initially.

In order to decrease the clearances, and as a result improve the volumetric efficiency, the scrolls were coated with a graphite lubricant, and the scrolls adjusted to reduce the clearance. The purpose of this procedure was to allow the scrolls to have slight contact with each other, in an attempt to allow the scrolls to "wear in" and as a result compensate for slight variations in machining tolerances. The graphite lubricant prevents galling of the aluminum scrolls as they contact each other. The graphite also has the effect of decreasing the radial clearance, by adding to the thickness of the scrolls, as a result improving the volumetric efficiency. The result of the testing indicated a dramatic reduction in leakage, resulting in a volumetric efficiency measurement of 96% at a speed of 1800 rpm. Preliminary performance tests indicated a measured volumetric efficiency substantially higher than the predicted efficiency. While this preliminary volumetric efficiency data was encouraging, at this point in the test program there was not sufficient instrumentation to measure adiabatic efficiency.

7. Manufacturing Cost Analysis

7.1 Background

The scroll compressor-expander module (scroll CEM) will be used with automotive fuel cells. To evaluate the consistency of the scroll CEM design with competitive automobile component cost requirements, cost of the scroll CEM in automotive production volumes was estimated. The manufacturing cost analysis was initiated during July 1996 on the scroll CEM design, exclusive of the prime-mover. It is important to note that the design has since been advanced and modified, but the basic components of the preliminary design (as of September 1996) have remained constant.

7.2 Objectives

A cost analysis was performed to estimate the direct manufacturing cost per scroll CEM in annual production volumes of 100,000. Part fabrication processes or sources and an assembly process were to be identified based on the steps used to assemble the prototype module. Secondly, the approximate weight of the scroll CEM (less the prime-mover) was calculated.

7.3 Approach

Arthur D. Little has developed a bottoms-up methodology for manufacturing cost analysis, which was applied to the scroll CEM. Table 7-1 summarizes the major steps of this methodology.

Table 7-1: Manufacturing Cost Analysis Tasks

Task Number	Description
Parts List and Weight	
1	Define a parts list
2	Weigh or estimate weight of each part
3	Identify materials, material volumes, tolerances and manufacturing processes for each part
4	Characterize purchased part and machined parts
Purchased Parts Cost	
5	Obtain quotes and prices for purchased parts from vendors
6	Calculate total purchased part cost
Machined Parts Cost	
7	Estimate stock material volume and cost per part
8	Identify machining steps needed for individual machined parts
9	Estimate machining times for machined parts
10	Obtain and compare quotes for machined parts
11	Calculate machined part costs
Assembly Tree and Cost	
12	Develop part-by-part "assembly tree"
13	Estimate assembly times
14	Calculate total assembly cost

The tasks identified in Table 7-1 were typically followed chronologically as presented, however in several instances, tasks were executed simultaneously. The final step involved summing the total direct cost estimate including materials, purchased parts, and direct labor for part fabrication and assembly.

Again, the dimensions and materials from the drawings provided in September 1996 have been taken literally for the manufacturing cost estimate, except that cast iron was substituted for bronze in the counterweight. The cost estimate includes the countershaft assembly, main housing assembly, orbiting scrolls, fixed scrolls and standoff collars, and does *not* include the oil sump, oil pump, prime-mover, cooling fan/housing, and sound muffling system.

7.4 Results

The total weight of the scroll CEM (125 parts) is approximately 56 pounds per module and the direct manufacturing cost is projected to be $170 - $180 per module. This manufacturing cost estimate assumes an annual production volume of 100,000 modules. A detailed summary of the results can be found in the Appendix . Figure 7-2 illustrates the relative breakdown of the manufacturing cost components.

If the capital investment for machining and process equipment is included the manufacturing cost would rise to $280-$300 per module. This assumes a 250% fixed cost burden rate (includes investment amortization and all overhead) estimated by Arthur D. Little.

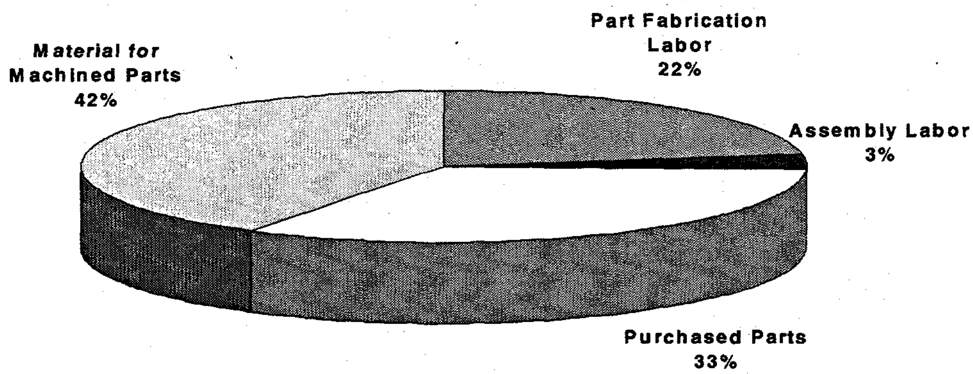


Figure 7-2: Components of Total Manufacturing Cost Estimate

The assembly tree for the scroll CEM is outlined in the Appendix. The assembly tree is to be read starting from the lower right hand corner towards the upper left hand corner.

Engineering judgment was used to determine which parts would be machined in-house and which would be purchased from suppliers. The fabrication and assembly sequence are one feasible path to manufacturing the scroll CEM, but we would expect that with a more detailed manufacturing study, the processes could be improved upon, with other client cost reduction.

8. Performance Testing

8.1 Introduction

Once checkout testing of the scroll CEM was completed, the formal test program was begun. The system performance testing included:

- Measurement of drive system losses
- Measurement of integrated system performance on the fully-instrumented compressor/expander system
- Supporting analysis to estimate individual component performance criteria

Separate measurement of individual component performance was not done due to the integrated system design, which requires both components to be operating simultaneously. Detailed measurement of the integrated system and subsequent analysis was used to estimate individual component performance.

8.2 Orbiting Scroll CEM Test Apparatus

The design and fabrication of the test apparatus, used to test the performance of the integrated scroll compressor/expander, was conducted in parallel with the fabrication of the scroll CEM. Figure 8-1 presents a schematic diagram of the test facility used during the performance tests. The test apparatus includes a drive motor to provide shaft power to the scroll CEM, as well as an electric heater to simulate the heated air from the fuel cell that enters the expander. Instrumentation included flowmeters, thermocouples, pressure gauges, and a torquemeter with digital readout to determine the motor shaft power. Flow rates were measured separately for the compressor and expander flows, and the temperatures and pressures of the inlet and exit flows were measured.

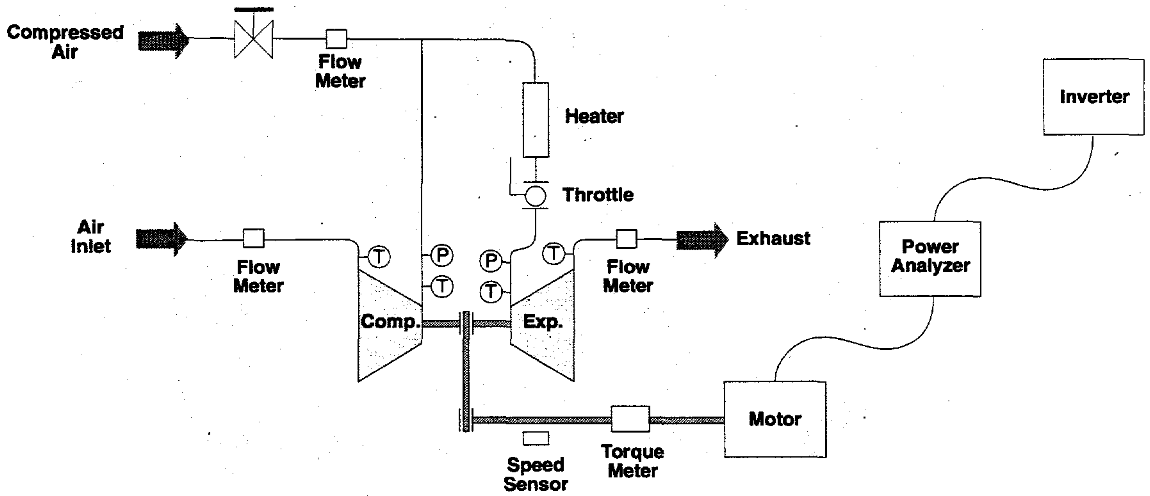


Figure 8-1: Performance Test Schematic

The measured parameters and instrumentation used are summarized in Table 8-1 below.

Table 8-1: Measured Parameters and Instrumentation

Measurement Type	Location	Instrumentation
Pressures	Compressor Exit	Pressure Gauge
	Expander Inlet	Pressure Gauge
Temperatures	Compressor Inlet	Thermocouple
	Compressor Exit	Thermocouple
	Expander Inlet	Thermocouple
	Expander Exit	Thermocouple
Flow	Compressor	Pitot Probe Meter
	Expander	Pitot Probe Meter
	Added Flow	Rotameter
Torque	System Drive Shaft	In-Line Shaft Torque Meter
Shaft Speed	System Drive Shaft	Strobe Light
Noise		Noise Meter

After completion of the design effort, the necessary parts were ordered, the components were received, and assembly completed. The completed test apparatus is shown in Figure 8-2. The test apparatus allowed us to perform full performance measurements on the scroll compressor/expander, including measurement of the adiabatic efficiency. After completion of the fabrication and checkout of the test apparatus, we began the "official" portion of the test program, increasing in speed increments as outlined in the test plan.

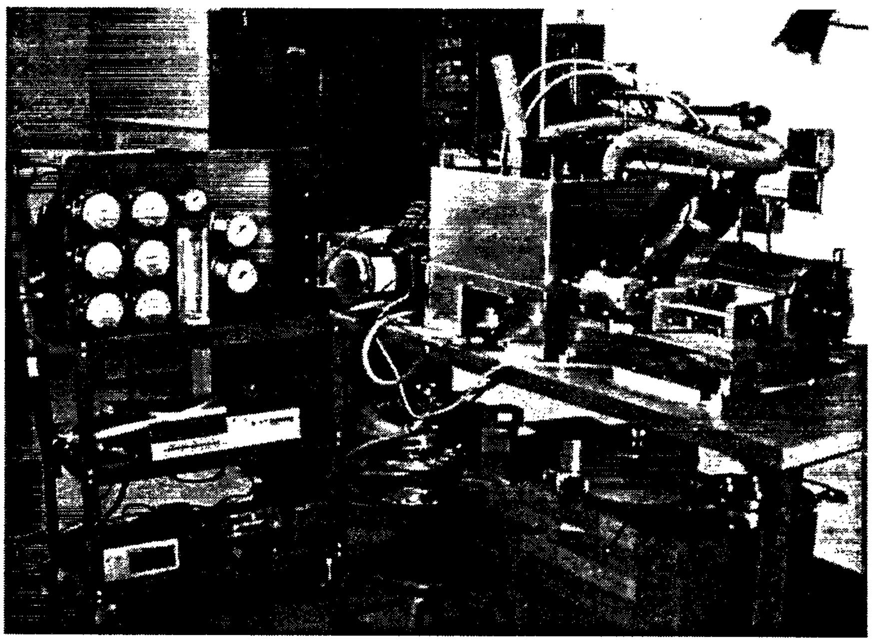


Figure 8-2: Scroll CEM Test Apparatus

8.3 Test Conditions

Table 8-2 below shows the test conditions which were run for performance testing of the integrated system.

Table 8-2: System Test Conditions

	Compressor			Expander	
Flow Rate	Flow Rate[1] (kg/hr)	Inlet Temp[2] (F)	Outlet Pressure Range (psig)	Inlet Temperature Range (F)	Fuel Cell Pressure Drop Range (psi)
10%	5	70	0-16	94	0-0.6
20%	10	70	5-16	120	0-1.2
40%	20	70	8-25	170-200	0-1.7
60%	30	70	16-33	220-250	0-3.4
80%	40	70	25-33	250-275	0-4.5
100%	43	70	32	255	0

[1]+/-5% [2]+/-10°F

8.4 Performance Testing

After completion of the scroll CEM checkout testing, performance testing began with a test of the system drive losses. This was done with the compressor and expander outer (fixed) scrolls removed. The scroll assembly was run at incrementally higher speeds, beginning the performance testing portion of the test program. The unit was successfully run at various speed points up to 60% speed. At that point, the performance data indicated that the scroll was operating as predicted, with the drive torque exhibiting a "leveling-off" trend as expected.

The speed of the scroll CEM was then increased to 70%. At this condition a fairly sharp increase in the drive torque reading was noticed. This was unusual, since a continuing of the leveling-off trend that was encountered during testing up to the 60% speed point was expected. A sharp increase in drive torque generally indicates that some contact is occurring within the unit, causing an increase in friction which subsequently causes an increase in the measured drive torque. This contact could potentially be between the scrolls, between the scrolls and the housing, or inside the drive mechanism.

Because of this possible contact, it was decided to temporarily discontinue testing and to disassemble the unit to investigate the source of the increased torque. A teardown and inspection of the scroll compressor/expander was then conducted. Upon disassembly, it was noticed that the drive plates exhibited some wear on the aluminum surfaces, varying from 0.001 to 0.004 inches deep. This appeared to be due to drive plates rubbing on the thrust bearings, resulting from misalignment, which could account for the increase in drive torque readings. The end plates were redesigned, incorporating an improved system of securing the thrust bearings, improving the alignment of the components within the drive mechanism, to correct the wear problem. The new end plates were installed, the unit reassembled, and the test program resumed. After repeating the 70% speed point, the speed of the unit was increased to 80%. The performance data at this speed point looked very good, with the measured drive torque remaining below the predicted level. While running at this 80% point, however, some leakage of oil from the drive mechanism was noticed. This oil was leaking from several of the oil seals around the posts that connect the orbiting scroll to the drive mechanism. Because of this leakage, it was decided to stop testing and investigate the reason for the leakage.

The existing seals were tested using an orbital seal tester and determined to require a higher spring force to prevent the leakage. Several other seal alternatives were also tested to see if there was a better seal design for this application. The result of this testing was that the existing seal design was the best alternative, as long as a higher spring force was used. New seals with the higher spring force were ordered, received, and installed in the unit. The surface of the end plates that the seals act against was also hardened and polished, in an effort to improve sealing.

The scroll compressor/expander unit was reassembled, checked out, and testing was resumed at the 80% speed point. The leakage of oil was reduced due to the new seal design, but leakage still existed. Testing was halted to investigate the reason for the leakage further. After disassembling the unit, some wear on the aluminum thrust surface was noticed. Although the wear was not extreme, is was significant enough to warrant a design modification. A redesigned thrust bearing, constructed from hardened steel, was added to the aluminum to minimize wear.

To address the oil leakage problem, several promising alternatives were identified as a result of testing using the orbital seal tester. One alternative is a seal with double-edge contact (basically two concentric seals) which we incorporated when we reassembled the unit. Another promising concept uses seals on both sides of the orbiting scroll attachment point. The scroll compressor/expander unit was reassembled and testing resumed. The level of oil leakage present was relatively minor, and more of an inconvenience than a serious problem. The oil cannot enter the gas path, so performance of the compressor was not impacted.

After completing the assembly of the unit, the low speed test points were repeated on our test apparatus. During these early test points a modest increase in the drive torque measurement was noticed, when compared to the earlier points prior to installation of the thrust plate. This increase was expected, as the new thrust plate provided an increased surface area over which to spread the thrust load, resulting in increased viscous losses due to shear in the layer of oil between the drive plate and the thrust

plate. This drive torque reading, while increased due to the larger thrust surface, was consistent through speeds up to 80%.
The fourth quarterly program review meeting was held on July 31, 1997 and was attended by representatives from DOE, Arthur D. Little, Delphi- H, and Delphi-E. Performance data from the test program were presented, indicating excellent performance, exceeding initial predictions. This was followed by a demonstration of the scroll CEM running at various speeds on the test apparatus. When the meeting reconvened, it was jointly decided that Arthur D. Little would continue testing during the following week to 90% speed, and discuss the test results with DOE and Delphi via a conference call. During this call it would be decided if it was prudent to go higher in speed.

Testing of the scroll CEM was continued, increasing the speed in small increments up to 85% speed. During this process a marked increase in drive torque was noticed, which began to increase rapidly as we approached 85% speed. This sharp increase in drive torque was an indicator that there was a problem occurring in the unit, most likely due to contact between the scrolls or components in the drive mechanism. The unit was run to 87% speed, and this trend continued, indicating that there was indeed some contact occurring within the unit. At that point it was decided that there would be substantial risk to the unit if testing was continued at higher speeds.

The conference call was held as planned and the results discussed with DOE and Delphi. Arthur D. Little recommended that if the scroll CEM needed to be delivered to Delphi by the end of August, that the unit should have a "redline" of 80% speed (40 g/sec). There was also another issue related to the fuel cell pressure drop between the compressor outlet and expander inlet. The actual fuel cell pressure drop had increased substantially from the original DOE guidelines, of 4 to 13 psi. This increased pressure drop resulted in an increase in the load on the thrust bearing, which if excessive could damage the machine. Arthur D. Little offered to test to several psi higher than the DOE guidelines, but felt it would harm the machine if was run at higher pressure drops than this. Figure 8-3 shows the envelope of pressure drop values that the scroll CEM can safely operate within.

Delphi indicated that they could not do anything to lower the fuel cell pressure drop, and that it might actually be higher than their prediction. As a result, it was decided jointly that since this scroll compressor/expander prototype was not designed to be compatible with the high pressure drop of the current fuel cell design, and it would not make sense to deliver the scroll CEM to Delphi.

Throughout the test program, there was a continuing trend of performance improvements. As is typical of an R&D program of this type, the development process is an iterative one in which the scroll CEM was run, disassembled, design modifications made, and the CEM retested. Figure 8-4 presents a graph of the history of the scroll CEM testing. As can be seen from the figure, this iterative process resulted in an increase in performance throughout the test effort.

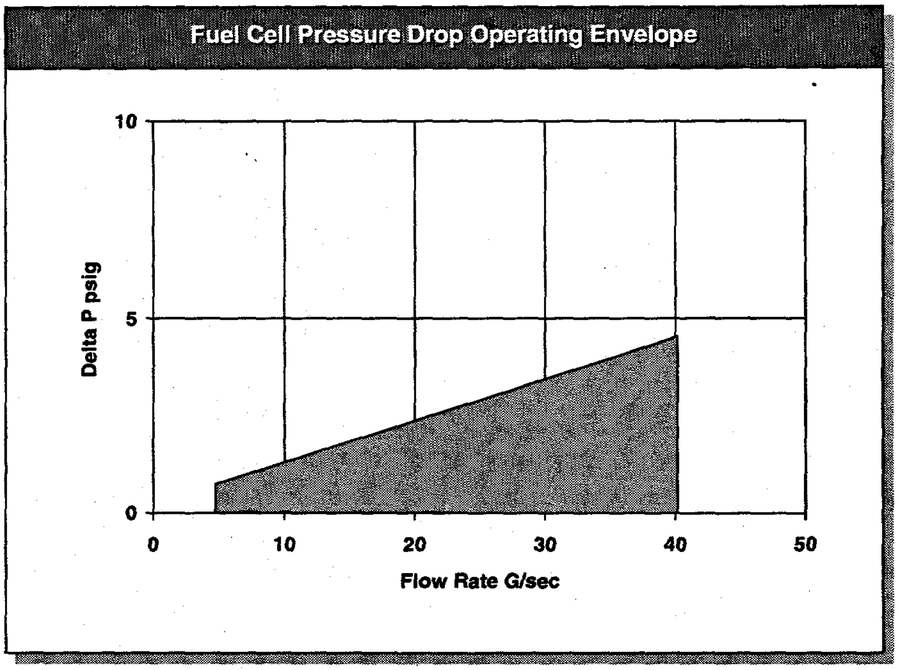


Figure 8-3: Fuel Cell Pressure Envelope

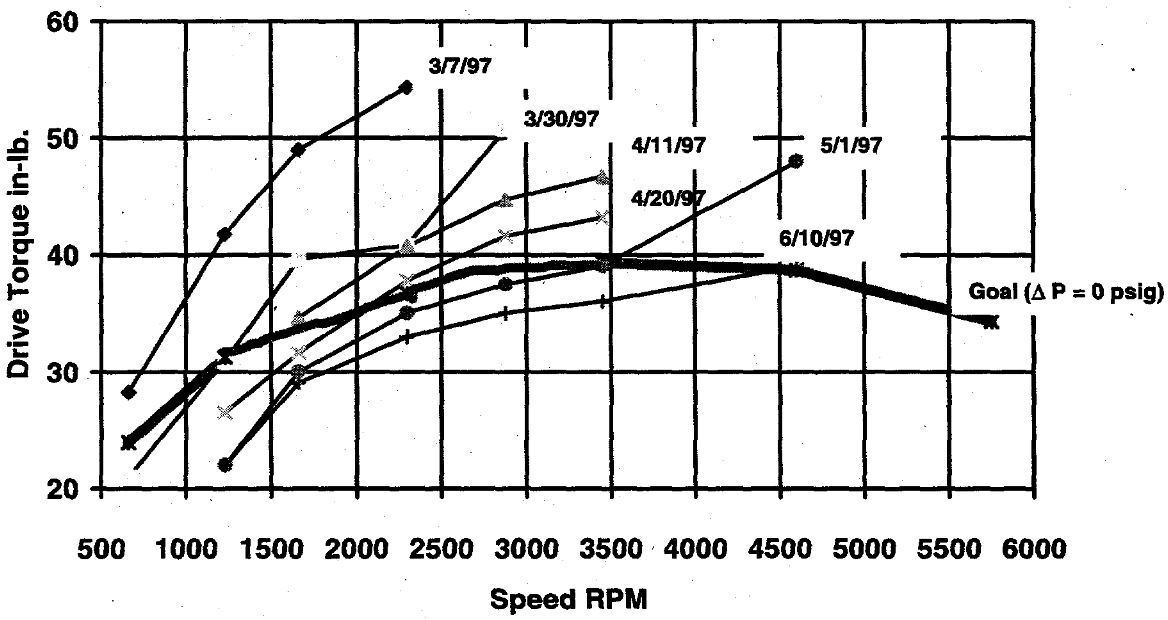


Figure 8-4: Performance History of Scroll CEM - Drive Torque vs. Speed

8.5 Test Data

The test data for the integrated scroll CEM is presented in Tables 8-3 and 8-4 below.

Table 8-3: Integrated CEM Test Data 1

Test ID	1	2	3	4	5
Flow (% of Design)	83	78	79	64	64
Pressures (psig):					
Fuel Cell Pressure Drop	0.0	4.5	0.0	3.5	0.0
Compressor Exit	32.3	32	32.3	25	25
Expander Inlet	32.3	27.5	32.3	21.5	25
Temperatures (F):					
Compressor Inlet	72	79.8	71.2	74.5	69.7
Compressor Exit	316.8	320.2	312.4	274.2	264.3
Expander Inlet	254.4	275	246.9	249.8	223.1
Expander Exit	87.5	104.1	83.9	100.8	79.3
Flow (g/s):					
Compressor	43.46	38.87	40.77	32.05	33.21
Expander	52.6	44.2	49.9	33.8	38.8
Shaft Torque (in-lb)	44.7	52.9	43.2	48.5	37.9
Shaft Speed (rpm)	4722	4500	4500	3800	3800
Calculated Shaft Power Ws (W)	2491	2809	2294	2175	1699
Noise[1] (dB)		89	89	85	

Table 8-4: Integrated CEM Test Data 2

Test ID	6	7	8	9	10
Flow (% of Design)	40	36	21	20	10
Pressures (psig):					
Fuel Cell Pressure Drop	2.0	0.0	1.0	0.0	0.5
Compressor Exit	16	15	9	9	4.5
Expander Inlet	14	15	8	9	4
Temperatures (F):					
Compressor Inlet	74.7	68	62.8	63.9	76.3
Compressor Exit	219.7	200.1	135.6	142.1	109.3
Expander Inlet	200.1	172.4	121.4	123.4	94.4
Expander Exit	95.7	75.7	70.4	67.1	75.4
Flow (g/s):					
Compressor	20.20	18.23	11.33	10.99	4.92
Expander	21.4	20.7	12.0	13.0	5.9
Shaft Torque (in-lb)	39.5	34	31.3	28	20.4
Shaft Speed (rpm)	2540	2300	1405	1405	792
Calculated Shaft Power Ws (W)	1184	923	519	464	191
Noise[1] (dB)	79.52		75	74.5	69

The system drive loss measurements are tabulated in Table 8-5 below.

Table 8-5: Drive Loss Test Data

Machine Speed (rpm)	Torque (in-lb)	Calculated Power (W)
656	5.5	43
1200	6.5	92
1662	7.7	151
2300	10.0	271
3450	13.0	529
4000	14.6	689
4600	16.2	880

9. Analysis

9.1 Analysis Procedure

Analysis to support the performance testing was done in order to improve understanding of the system performance and to characterize the performance of the individual components (see Figure 9-1 below).

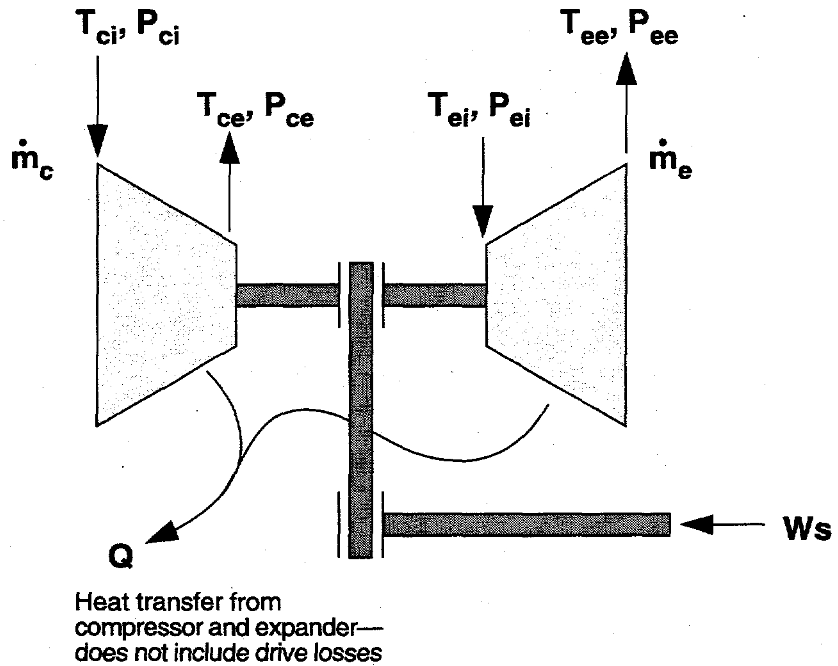


Figure 9-1: Definition of Mass and Energy Flows

The determination of efficiencies of the expander and compressor requires development of estimates for shaft power associated with these components. The shaft power (equation 5, below) has four components, only one of which is measured directly (drive power input). The drive loss is estimated based on a measurement with the machine unloaded (Section 3). The analysis procedure uses this estimate, and also uses a modeled compressor efficiency to allow calculation of all the terms in the shaft power equation. The analysis proceeded as follows:

- Determine Drive loss as a function of speed: A curve fit of the drive loss measurements were used to characterize the drive losses.
  $L_d = f(N)$ (1)
- Some adjustment to experimentally-measured mass flows was done in order to smooth the volumetric efficiency curves. This adjustment eliminated the anomaly of having volumetric efficiency higher than 100% and represents a reduction of

volumetric efficiency in these and other cases. The adjustments were all within expected measurement accuracy.

- Compressor power based on compressor efficiency estimates:

$$W_c = (W_{c,\ ideal})/\eta_c \quad (2)$$

$$W_{cideal}, = \dot{m}_c\,(h_{ce,\ ideal} - h_{ci}) \quad (3)$$

where $h_{ce,ideal}$ is calculated based on assuming adiabatic isentropic compression

$$\text{Loss } L_c = W_c - W_{c,ideal} \quad (4)$$

- Expander Power, efficiency, and loss:

$$W_e = W_c + L_d - W_s \quad (5)$$

$$W_{e,ideal} = \dot{m}_e\left(h_{ei} - h_{ee,ideal}\right) \quad (6)$$

where $h_{ee,ideal}$ is calculated based on assuming adiabatic isentropic expansion.

$$\text{Efficiency } \eta_e = \frac{W_e}{W_{e,ideal}} \quad (7)$$

$$\text{Loss } L_e = W_{e,ideal} - W_e \quad (8)$$

- An adjusted compressor efficiency is calculated to account for the drive loss and parasitic losses (oil pump and cooling fan)

$$\eta_c{}^* = (W_{c,ideal})/(W_c+L_d+W_{pump} + W_{fan}) \quad (9)$$

9.2 Analysis Results

The test data were analyzed using the procedure outlined above, resulting in a performance characterization of the scroll CEM and its components. This performance characterization is presented in Tables 9-2 and 9-3 below. The efficiencies are plotted in Figure 9-2.

Table 9-2: Performance Characterization 1

Test ID	1	2	3	4	5
Flow (% of Design)	83%	78%	79%	64%	65%
Drive Loss (W)	821	765	765	596	596
Compressor					
Adjusted Flow (g/s)	41.5	38.9	39.5	32.1	32.4
Estimated efficiency	80.5%	80.0%	80.0%	79.0%	79.0%
Ideal Shaft Power (W)	4,844	4,578	4,610	3,136	3,137
Shaft Power (W)	6017	5723	5763	3969	3971
Loss (W)	1173	1145	1153	834	834
Adjusted Efficiency	69.9%	69.6%	69.6%	67.3%	67.3%
Volumetric Efficiency	99.6%	99.4%	99.4%	96.1%	96.1%
Expander					
Adjusted flow (g/s)	50.6	43.2	48.7	33.8	38.0
Shaft Power (W)	4348	3679	4234	2391	2868
Ideal Shaft Power (W)	5693	4600	5417	3039	3572
Efficiency	76.4%	80.0%	78.1%	78.7%	80.3%
Volumetric Efficiency	95.4%	93.1%	95.6%	89.0%	90.4%
Loss (W)	1,345	922	1,184	649	704

* Including drive loss

Table 9-3: Performance characterization 2

Test ID	6	7	8	9	10
Flow (% of Design)	40%	36%	20%	20%	10%
Drive Loss (W)	330	285	132	132	41
Compressor					
Adjusted Flow (g/s)	20.2	18.2	10.1	10.0	4.9
Estimated efficiency	77.3%	75.0%	67.0%	67.0%	58.0%
Ideal Shaft Power (W)	1,410	1,195	432	427	117
Shaft Power (W)	1824	1593	645	637	201
Loss (W)	414	398	213	210	84
Adjusted Efficiency	62.8%	60.6%	49.7%	49.6%	34.9%
Volumetric Efficiency	90.6%	89.2%	80.4%	79.4%	71.0%
Expander					
Adjusted flow (g/s)	20.6	20.7	11.3	11.8	5.9
Shaft Power (W)	970	955	257	304	51
Ideal Shaft Power (W)	1321	1325	429	488	121
Efficiency	73.4%	72.0%	60.0%	62.4%	42.2%
Volumetric Efficiency	83.2%	81.3%	75.2%	75.4%	70.7%
Loss (W)	351	371	172	184	70

*Including drive loss

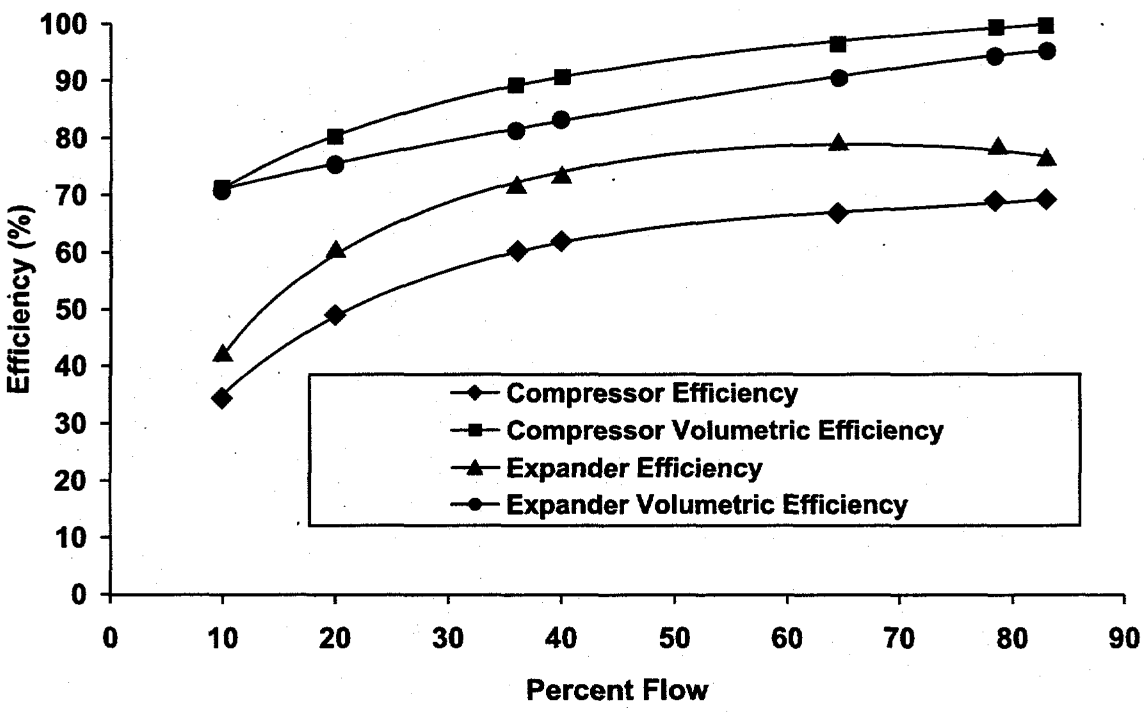


Figure 9-2: Component Efficiencies

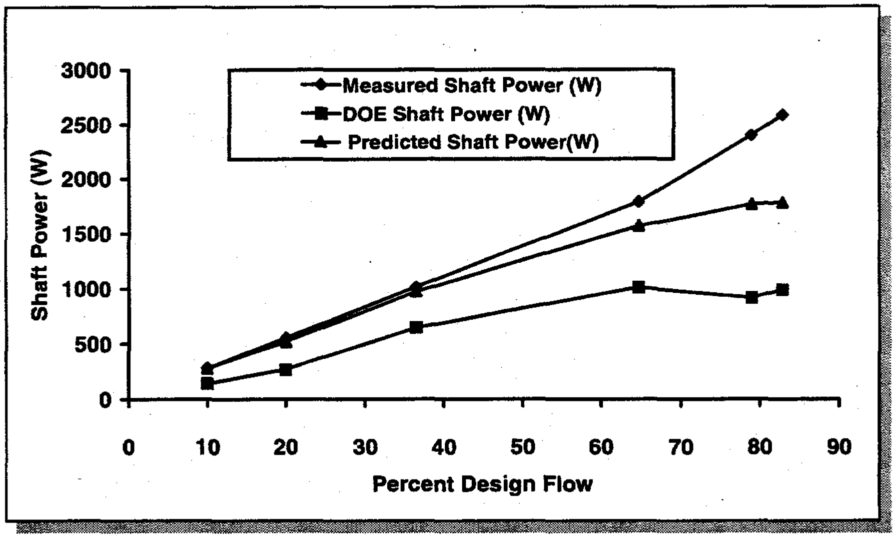


Figure 9-3: Shaft Power

The transient characteristics of the compressor were not tested because laboratory operation was conducted with a motor/invertor system whose rotational inertia and torque characteristics would not necessarily match those of a CEM integrated with a fuel cell. The rotational inertia of the orbiting scroll design is inherently low, because the scrolls and counterweight orbit around a small radius (5/16") rather than rotating. The rotational inertia of the first generation system is estimated to be 1.7 lbm-in^2. This compares with a rotational inertia of 8.3 lbm-in^2, estimated for a permanent-magnet motor sized 10% larger than the estimated 6 hp required for the 50g/s flow rate.

The speed-up curve was determined assuming that a motor with the above characteristics is used. It is assumed that expander inlet temperature is 250F. CEM efficiencies (volumetric and total) beyond the measured flow conditions are assumed to be equal to the values estimated for the 80% flow condition. A steady-state shaft torque requirement for the system was estimated and approximated with a linear curve fit as a function of machine speed. The system equations are:

$$T_m = (I_m + I_{CEM})\omega' + T_{steady}$$

$$T_{steady} = A + B\omega$$

The calculated values are:

T_m: Motor torque (assumed constant): 73 in-lbf

I_m: Motor rotational inertia: 8.3 lbm-in^2

I_{CEM}: CEM rotational inertia: 1.7 lbm-in^2

A: 38 in-lbf

B: 0.054 in-lbf-s

The estimated speedup times are 0.72 sec for 20% to 90% speed, and 0.82 for 0% to 90% speed.

9.3 Comparison With DOE Guidelines

The results of the scroll performance characterizations generated from the test data were compared to the DOE guidelines to provide a comparison between measured performance and the goals. This comparison is presented in Table 9-4. As noted in the table, the scroll CEM was originally intended for integration with the GM/Delphi fuel cell effort. For consistency, the ADL internal airflow target was changed to 50 g/sec early in the program, based on the projected airflow requirements of the GM/Delphi fuel cell. The scroll CEM achieved 42 g/sec during testing. Full airflow was limited by drive torque measurements, which indicated possible contact between internal components of the drive mechanism resulting in halting of the test program. Testing was performed with dry air, and as a result, data for air with water vapor was not obtained. Testing at all inlet and outlet pressure ranges was conducted during the course of testing.

Measured efficiency levels were relatively high, but fell short of the guidelines by 5 - 10 percent, depending on the flow rate. The efficiency values achieved during this effort were felt to have demonstrated the viability of scroll technology in this application and were considered successful at this point in the development of the scroll CEM. Further development of the scroll CEM will undoubtedly result in an increase in the efficiency values, resulting in efficiency reaching or potentially exceeding the levels of the guidelines.

The volume and weight guidelines were exceeded by a large amount. We believe that the guideline values are based on an extrapolation of automotive turbocharger values, and as a result are not appropriate for scroll technology. The production unit cost is projected to be very close to the guidelines, $280 versus a goal of $200. Given that the cost estimate is so close to the guidelines, it is anticipated that the goal can be reached by a modest cost reduction effort.

Startup and transient response estimates indicate a response time well below the guideline values. This highlights a key advantage of orbiting scroll technology. Because of the very small orbit radius of the scroll hardware, the moment of inertia of the scroll CEM is much lower than that for rotating machinery such as co-rotating scroll, screw, and turbomachinery. The noise level recorded was only 9 dB higher than the guidelines, resulting from a modest effort at muffling the scroll CEM hardware. A more concentrated effort during subsequent efforts should allow the noise level guidelines to be achieved.

Table 9-4: Comparison of Measured Performance with DOE Guidelines

Parameters		DOE Guidelines		Scroll CEM Performance	
		Compressor	Expander	Compressor	Expander
Flow Rate @ Max Power					
Dry Air	(g/sec)	64 to 76**	56 to 70	42	51
	(kg/hr)	230 to 273	203 to 254	150	184
Water Vapor	(g/sec)	0 to 4	9 to 16	See Note 1	See Note 1
	(kg/hr)	0 to 11	31 to 55		
Inlet Pressure	(atm)	1.0	2.8	1.0	2.7 to 3.2
Outlet Pressure	(atm)	3.2	1.0	3.2	1.0
Inlet Temperature					
Design Point:	(°C)	20 to 25	118 to 150	15 to 27	34 to 135
	(°F)	68 to 77	244 to 302	60 to 80	94 to 275
Extreme:	(°C)	-40 to 60	65 to 150	See Note 2	See Note 2
	(°F)	-40 to 140	149 to 302		
Max. Shaft Power	(kW)	12.6	8.3	6.0[4]	4.3[4]
Turndown Ratio		10:1	10:1	10:1	10:1
Contamination		Oil-free <100 ppm		<50 ppm[3]	
Efficiency vs. Flow & PR					
100% flow	3.2 PR	75%	90%	---	---
80% flow	3.2 PR	80%	90%	70%	80%
60% flow	2.7 PR	75%	86%	68%	80%
40% flow	2.1 PR	70%	82%	64%	73%
20% flow	1.6 PR	65%	80%	52%	62%
10% flow	1.3 PR	50%	75%	39%	42%
Volume*	liters	4 total		38 liters	(13.2"x11.8"x14
Weight*	kg	3 total		27.8 kg[5]	
Production Unit Cost* @ 100,000 units/yr		$200 total		$280[6]	
Start-up Response		<5 sec to 90% max. rpm		0.82 sec[7]	
Transient Response		<4 sec for 20 to 90% max. flow		0.72 sec[7]	
Noise		<80 dB		89 dB	

*These values do not include heat exchangers or motors/controllers, however sizing traded-off studies should target minimization of the overall air supply subsystem

** ADL internal target for program's first phase was 50 g/s, based on projected GM/Delphi fuel cell requirements.

[1]Testing was done with dry air
[2]Extreme condition testing not performed
[3]Worst case estimate
[4]Estimates based on net measured value of 2.49 kW at maximum tested flow range
[5]Includes only the compressor, expander, and drive
[6]Assumes 250% fixed cost burden rate, excludes oil sump, oil pump, and prime mover
[7]Calculated value, see text.

10. Remaining Scroll CEM Tasks

During the last few months of the program, efforts were focused on addressing the remaining technical issues related to this phase of development of the scroll compressor/expander. These remaining areas are:

- Understanding the reasons for the high drive torque measurements
- Solving the oil leakage between the orbiting scroll and the end plates
- Investigating the use of a shaft-driven oil pump.

In an effort to understand the high drive torque measurements, an accelerometer and an FFT spectrum analyzer were used to measure the imbalance present in the CEM. During this investigation it was found that the balance of the machine was off resulting from a difference in weight of 62 grams between the orbiting scrolls of the compressor and expander. This level of imbalance is significant, and may be a contributor to the high torque readings during testing. The next phase of the scroll CEM development will include precision balancing of the scroll and drive hardware.

John McCullough focused his effort on solving the oil leakage problem, resulting in two versions of a novel non-contacting seal that appear promising. Both seal types have been tested on the orbital seal test apparatus and appear to work well.

A DC powered fan motor was ordered and installed to replace the AC motor previously used for the cooling fan. An investigation of the use of a shaft-driven oil pump to supply lubrication to the drive mechanism was also conducted. A variable speed DC motor was installed on the pump, and the speed of the pump varied during operation of the scroll CEM, while the oil supply to the drive was closely monitored. During testing it was found that the oil pump was very sensitive to the motor speed, to the point where it appeared to be unstable. The oil pressure would jump from 10 to 40 psi with a very minor increase in pump speed. The conclusion from this testing is that the type of pump used in our test apparatus is not compatible with direct drive from the scroll motor shaft.

The process of updating the drawing package to match the latest design changes of the prototype was completed. This drawing package includes assembly level drawings of the scroll CEM, a complete bill of materials, and detailed drawings of the test apparatus.

11. Conclusion

In conclusion, the scroll CEM development program summarized in this report has been highly successful. The objectives of the program are listed below, followed by a description of the accomplishments of the scroll CEM program:

- **Develop an integrated scroll CEM** - a fully-integrated, low-cost, high-efficiency scroll CEM was developed under this program. Although the unit has not been operated for sufficient hours to accurately assess reliability and durability, the unit was run successfully without major incident throughout the test program.

- **Demonstrate efficiency and capacity goals** - The test program demonstrated the high-efficiency characteristics of scroll in this application. While the peak efficiency achieved was less than the DOE guidelines, the efficiency is still relatively high and remains high even at low flows. This characteristic is very different from turbomachinery, which typically drops precipitously as flow is reduced. The full capacity goal was not demonstrated during the test program because the drive torque characteristic limited the maximum speed of the scroll CEM. This is not a limitation of scroll technology, however, because capacity can easily be increased at the same speed by increasing the displacement of the compressor by changing the design parameters, or even using two compressors in parallel for high flow capacity.

- **Demonstrate manufacturability and cost goals -** Manufacturability was demonstrated during fabrication of the scroll CEM, which was accomplished primarily on a three axis CNC milling machine. The manufacturing cost estimate concluded that production cost goals can be achieved.

- **Evaluate operating envelope -** The CEM has been tested at Arthur D. Little over a range of conditions. The tested range of machine speed and compressor discharge pressure is shown in Figure 11-1.

In summary, while the scroll CEM program did not demonstrate a level of performance as high as the DOE guidelines in all cases, it did meet the overriding objectives of the program. A fully-integrated, low-cost CEM was developed that demonstrated high efficiency and reliable operation throughout the test program. The performance "bar" was set very high on this program, and while the scroll did not exceed the bar, it came very close. Future development of the scroll CEM will undoubtedly improve performance to the level where it can meet these performance goals.

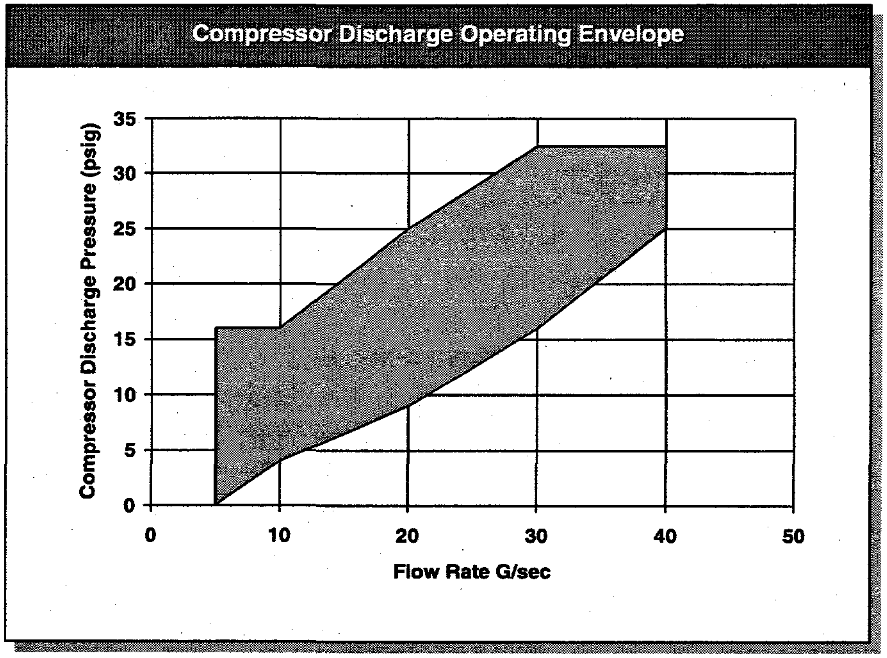
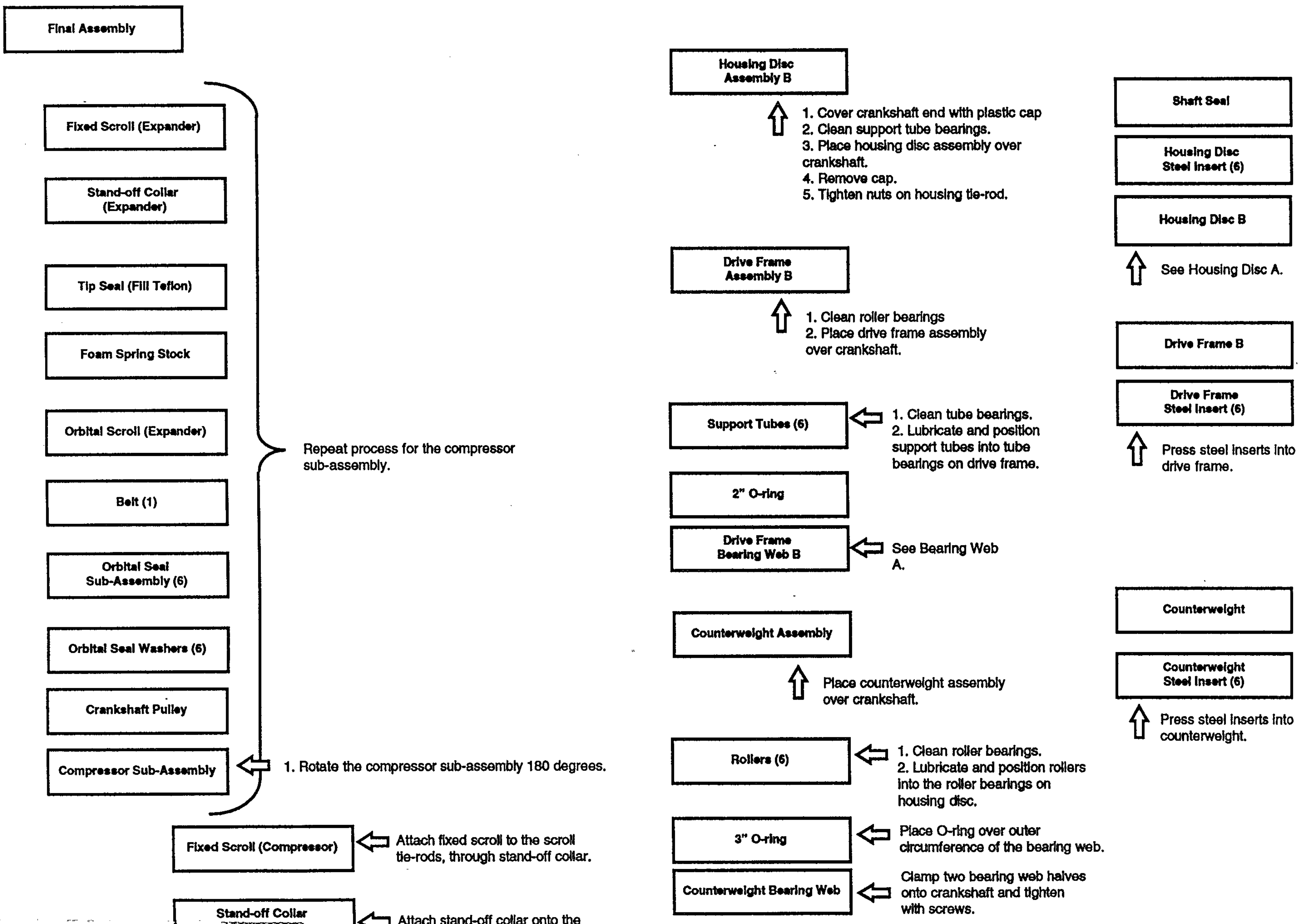
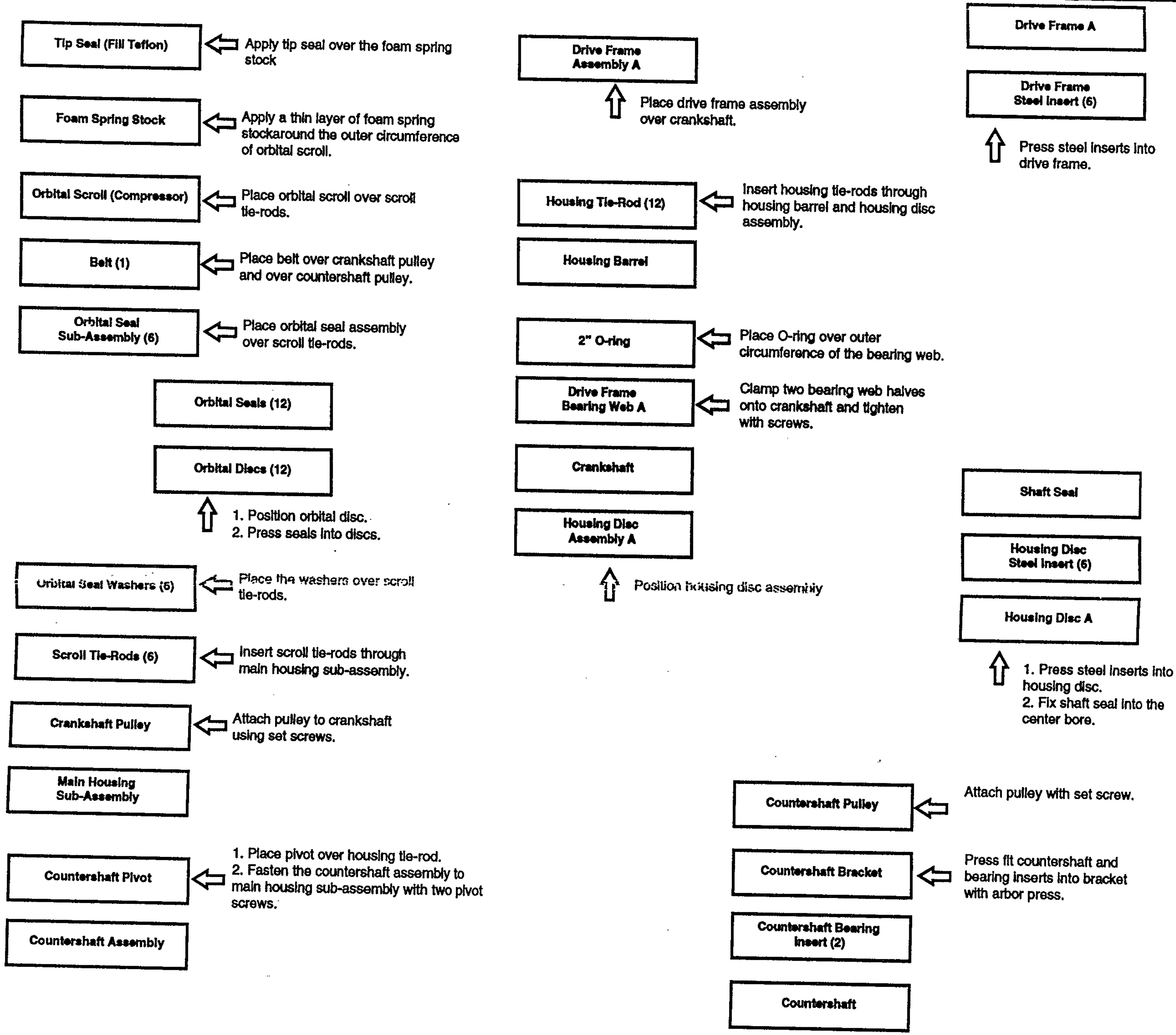


Figure 11-1: CEM Operating Envelope

Appendix

Scroll Compressor Manufacturing Cost Estimate   Assembly Tree





* oil sump and oil pump not included

scroltre.ppt58473CH101096

Drawing #	Part Name	Material	Form	# of Parts	Minimum Tolerance	OD/Width	Thickness	Length	Volume	Weight (lbs) Steel	Al	Brass	Other	TOTAL
400-12	**Countershaft**	Steel	rod	1	0.001	0.79		10.500	9	2.529				2.529
-12	**Countershaft Bracket**	6061-T6 Aluminum	casting	1	0.002		5.250	3.360	14		1.41			1.411
-12	**Countershaft Pivot**	6061-T6 Aluminum	plate	1			0.150	2.000			0.03			0.034
-08	**Housing Disc**	6061-T6 Aluminum	plate	2	0.001	10.00	0.600		41.35		4.14			8.270
-08	**Housing Disc Steel Inserts**	Hardened alloy steel	plate	12	0.002	2.71	0.255		0.96	0.271				3.246
-10	**Orbital Disc**	6061-T6 Aluminum	sheet	12	0.001	2.00	0.600				0.004			0.053
-10	**Orbital Seal**	Nylon		12	0.002	1.92	0.100							0.000
-10	**Orbital Seal Washer**	6061-T6 Aluminum	sheet	12	0.001	0.51	0.090							0.000
-05	**Drive Frame**	6061-T6 Aluminum	casting	2	0.002	9.50	0.375				2.40			4.800
-05	**Drive Frame Steel Inserts**	Hardened alloy steel	tube	12					0.08	0.022				0.259
-04	**Couterweight**	Cast Iron	casting	1	0.002	8.72	1.375						10.686	10.686
-04	**Couterweight Steel Inserts**	Hardened alloy steel (Rc58	tube	6	0.001	1.10	1.375		0.28	0.079				0.476
-01	**Crankshaft**	1018 Steel	rod	1	0.001 (dia)	1.48		5.660	3.14	0.889				0.889
-02	**Drive Frame Bearing Web**	6061-T6 Aluminum	rod	2	0.001 (dia)	2.23	0.375				0.130			0.260
-07	**Housing Barrel**	6061-T6 Aluminum	casting	1	0.001	10.00	0.450	2.185	15.10		1.464			1.464
-11	**Tie-Rod (housing)**	Low Carbon Steel	rod	12	0.030	0.19		7.750		0.033				0.397
-03	**Counterweight Bearing Web**	Phos. Bronze	rod	1	0.001 (dia)	2.73	0.375					0.366		0.366
-14	**Roller**	Hardened alloy steel	rod	6	0.005	0.38		2.460		0.082				0.489
-06	**Support Tube**	6061-T6 Aluminum	tube	6	0.001	0.75		1.785			0.024			0.146
-13	**Tie-Rod (scroll)**	Low Carbon Steel	rod	6	0.030	0.19		4.500		0.055				0.331
PP	**Pulleys**			4		2.00	0.75				0.229			0.914
PP	**Belt**	Rubber		2									0.068	0.137
PP	**3" O-ring**	Rubber		1									0.007	0.007
PP	**2" O-ring**	Rubber		2									0.007	0.013
-16	**Orbital Scroll (Compressor)**	6061-T6 Aluminum	cast	1	0.001	8.60	5.00				3.600			3.600
-18	**Fixed Scroll (Compressor)**	6061-T6 Aluminum	cast	1	0.001	8.60	5.00				3.600			3.600
-17	**Standoff Collar (Compressor)**	6061-T6 Aluminum	cast	1	0.003	10.00	4.66		23		2.26			2.264
PP	**Foam Spring Stock**	foam		2									0.001	0.002
PP	**Tip Seal**	fill teflon		2									0.001	0.002
-19	**Orbital Scroll (Expander)**	6061-T6 Aluminum	cast	1	0.001	8.60	5.00				3.600			3.600
-21	**Fixed Scroll (Expander)**	6061-T6 Aluminum	cast	1	0.001	8.60	5.00				3.600			3.600
-20	**Standoff Collar (Expander)**	6061-T6 Aluminum	cast	1	0.003	10.00	4.66		23		2.26			2.264
	TOTALS			128										56
Production Volume	100,000													
Labor Cost $/hr	$ 44													
Burden Rate	2.5													

Drawing #	Part Name	Starting Weight (lbs)	Finish Weight (lbs)	Total Volume (lbs/year)	Material Price ($/lb)	Material Scrap	Material Cost per part	Total Cost $	Purchased Parts	Avail. & Price*
400-12	Countershaft	1.45	2.53	252858	$ 0.79	0.57	$ 1.14	$ 1.14		pp
-12	Countershaft Bracket	1.6	1.41	141123	$ 1.40	1.13	$ 2.24	$ 2.24		pp
-12	Countershaft Pivot	0.034	0.03	3400	$ 1.45	1.00	$ 0.05	$ 0.05	$ 19.25	pp
-08	Housing Disc	4.71	4.14	413522	$ 1.45	1.14	$ 6.83	$ 13.66		pp
-08	Housing Disc Steel Inserts	0.34	0.27	27052	$ 0.40	1.26	$ 0.14	$ 1.63		
-10	Orbital Disc	0.024	0.00	441	$ 1.34	5.44	$ 0.03	$ 0.39		
-10	Orbital Seal		0.00	0				$ -		
-10	Orbital Seal Washer		0.00	0	$ 1.34			$ -	$ 19.00	
-05	Drive Frame	3	2.40	240000	$ 1.45	1.25	$ 4.35	$ 8.70		pp
-05	Drive Frame Steel Inserts	0.022	0.02	2162	$ 0.40	1.00	$ 0.01	$ 0.10		
-04	Couterweight	26.38	10.69	1066600	$ 0.30	2.47	$ 7.91	$ 7.91		pp*
-04	Couterweight Steel Inserts	0.1	0.08	7927	$ 0.40	1.26	$ 0.04	$ 0.24		
-01	Crankshaft	3	0.89	88923	$ 0.79	3.37	$ 2.36	$ 2.36		pp**
-02	Drive Frame Bearing Web	0.288	0.13	13007	$ 1.30	2.21	$ 0.37	$ 0.75	$ 38.60	
-07	Housing Barrel	1.725	1.46	146444	$ 1.40	1.18	$ 2.42	$ 2.42	$ 12.80	pp*
-11	Tie-Rod (housing)	0.033	0.03	3307	$ 0.56		$ 0.39		$ 4.65	
-03	Counterweight Bearing Web	0.703	0.37	36596	$ 0.85	1.92	$ 0.60	$ 0.60	$ 22.70	*
-14	Roller	0.082	0.08	8157	$ 0.40	1.00	$ 0.03	$ 0.20		pp**
-06	Support Tube	0.024	0.02	2425	$ 3.30	1.00	$ 0.08	$ 0.48		*
-13	Tie-Rod (scroll)	0.055	0.06	5511	$ 0.56		$ 0.23		$ 1.38	
PP	Pulleys		0.23				$ 6.32		$ 25.28	
PP	Belt		0.07	6834			$ 1.99		$ 3.98	
PP	3" O-ring		0.01				$ 0.60		$ 0.60	
PP	2" O-ring		0.01				$ 0.50		$ 1.00	
-16	Orbital Scroll (Compressor)	3.7	3.60		$ 1.40	1.03	$ 5.18	$ 5.18		
-18	Fixed Scroll (Compressor)	3.7	3.60		$ 1.40	1.03	$ 5.18	$ 5.18		
-17	Standoff Collar (Compressor)	2.53	2.26		$ 1.40	1.12	$ 3.54	$ 3.54		
PP	Foam Spring Stock		0.00			-	$ -	$ -	$ 0.50	
PP	Tip Seal		0.00			-	$ -	$ -	$ 1.00	
-19	Orbital Scroll (Expander)	3.7	3.6		$ 1.40	1.03	$ 5.18	$ 5.18		
-21	Fixed Scroll (Expander)	3.7	3.6		$ 1.40	1.03	$ 5.18	$ 5.18		
-20	Standoff Collar (Expander)	2.53	2.26		$ 1.40	1.12	$ 3.54	$ 3.54		
	TOTALS							$ 70.67	$ 57.39	
Production Volume	100,000									
Labor Cost $/hr	$ 44									
Burden Rate	2.5									

Arthur D Little

Drawing #	Part Name	Machining Process Times, seconds per part													
		CNC Machine Times												Case Hardening	Finish Grinding
400-12	Countershaft												400	15	240
-12	Countershaft Bracket									410					
-12	Countershaft Pivot										8				
-08	Housing Disc						613								
-08	Housing Disc Steel Inserts								535.5						
-10	Orbital Disc												10		
-10	Orbital Seal														
-10	Orbital Seal Washer												10		
-05	Drive Frame				297										
-05	Drive Frame Steel Inserts								225						
-04	Couterweight			623											
-04	Couterweight Steel Inserts								75						
-01	Crankshaft												800	15	360
-02	Drive Frame Bearing Web	170													
-07	Housing Barrel					797									
-11	Tie-Rod (housing)														
-03	Counterweight Bearing Web		226												
-14	Roller												120	15	30
-06	Support Tube							50							
-13	Tie-Rod (scroll)														
PP	Pulleys														
PP	Belt														
PP	3" O-ring														
PP	2" O-ring														
-16	Orbital Scroll (Compressor)											671			
-18	Fixed Scroll (Compressor)											671			
-17	Standoff Collar (Compressor)														
PP	Foam Spring Stock														
PP	Tip Seal														
-19	Orbital Scroll (Expander)														
-21	Fixed Scroll (Expander)											671			
-20	Standoff Collar (Expander)											671			
	TOTALS	170	226	623	297	797	613	50	836	410	8	2684	1340	45	630
Production Volume	100,000														
Labor Cost $/hr	$ 44														
Burden Rate	2.5														

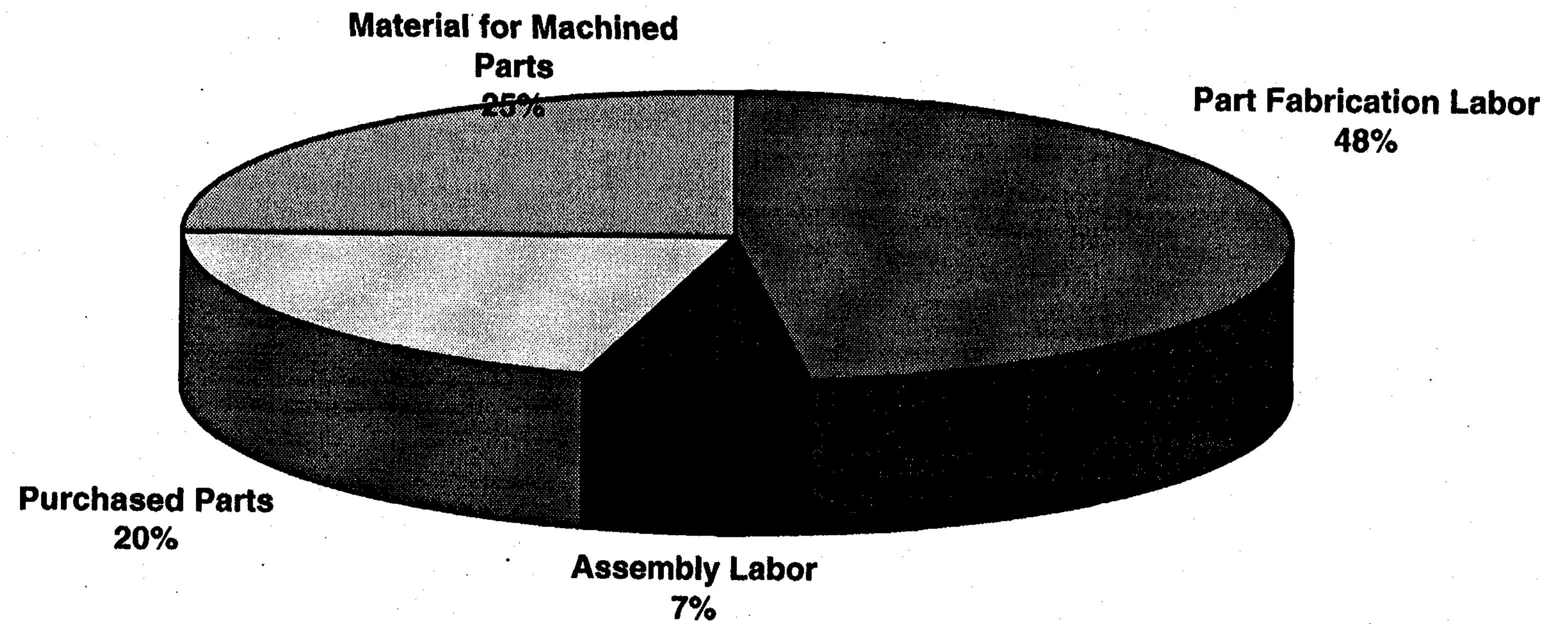
Drawing #	Part Name	Machining: Total Machining Time, sec	Machining: Total Machining Cost $	Prototype: Costs	Prototype: ADL Purchase Order #	
400-12	Countershaft	655	$ 7.94	0.00		
-12	Countershaft Bracket	410	$ 4.97	725.00	611406	
-12	Countershaft Pivot	8	$ 0.10	0.00		
-08	Housing Disc	1226	$ 14.86	0.00	209861	
-08	Housing Disc Steel Inserts	536	$ 6.49	0.00		
-10	Orbital Disc	120	$ 1.45	525.00	611406	
-10	Orbital Seal	0	$ -	0.00		
-10	Orbital Seal Washer	120	$ 1.45	0.00		
-05	Drive Frame	593	$ 7.19	1860.45	210279	
-05	Drive Frame Steel Inserts	225	$ 2.73	0.00		
-04	Couterweight	623	$ 7.55	10576.00	209861	
-04	Couterweight Steel Inserts	75	$ 0.91	0.00		
-01	Crankshaft	1175	$ 14.25	1350.00	611406	
-02	Drive Frame Bearing Web	340	$ 4.12	490.00	611406	
-07	Housing Barrel	797	$ 9.66	0.00	209861	
-11	Tie-Rod (housing)	0	$ -	240.00	611406	
-03	Counterweight Bearing Web	226	$ 2.74	295.00	611406	
-14	Roller	990	$ 12.00	135.00	611406	
-06	Support Tube	300	$ 3.64	210.00		
-13	Tie-Rod (scroll)	0	$ -	135.00	611406	
PP	Pulleys	0	$ -	30.00		
PP	Belt	0	$ -	10.00		
PP	3" O-ring	0	$ -	48.00	503885	
PP	2" O-ring	0	$ -	0.00		
-16	Orbital Scroll (Compressor)	671	$ 8.13	3500.00	209861	
-18	Fixed Scroll (Compressor)	671	$ 8.13	3500.00		
-17	Standoff Collar (Compressor)	0	$ -	3500.00		
PP	Foam Spring Stock	0	$ -	20.00		
PP	Tip Seal	0	$ -	20.00		
-19	Orbital Scroll (Expander)	0	$ -	3500.00		
-21	Fixed Scroll (Expander)	671	$ 8.13	3500.00		
-20	Standoff Collar (Expander)	671	$ 8.13	3500.00		
		0	$ -			
		0	$ -	6427.37	611662	
	TOTALS	11102	$ 134.59	$ 43,372		
Production Volume	100,000					
Labor Cost $/hr	$ 44				Supplier Markup on VC	
Burden Rate	2.5					

Machining Operations				Percent of ops time	
			< 25 operations	60%	
			> 25 operations	40%	
			Time (min)		
Part	**Operation**	**Operations**	**Total Process Time**	**Setup/travel time**	**Total (sec)**
Housing Disc	Drill 2.5" thru holes	6	0.6		36
	Drill & Tap 1/8 NPT	5	0.525		31.5
	Drill cntr hole & counterbore both ends	1	0.12		7.2
	Drill 0.189 thru holes	12	0.52		31.2
	Mill surfaces	2	0.56		33.6
	Slot Mill edge channel	1	1		60
	Slot Mill cntr channel	1	0.76		45.6
		28	4.085	6.1275	612.75
Housing Barrel	Drill 0.189 thru holes	12	1.83		109.8
	Slot Mill	12	0.79		47.4
	Form Mill web	12	0.79		47.4
	Side Mill outer face	1	1.9		114
		37	5.31	7.965	796.5
Counter Weight	Drill 0.975" holes	6	0.74		44.4
	Drill 0.5" holes	6	0.74		44.4
	Drill cntr hole	1	0.234		14.04
	Form Mill edge	12	1.32		79.2
	End mill both faces	2	1.12		67.2
		27	4.154	6.231	623.1
Drive Frame Bearing Web	Drill cntr hole	1	0.05		3
	Drill & Ream for 5/0 taper pin	2	0.33		19.8
	Drill .188D spotface	2	0.08		4.8
	Drill .120 to split	2	0.2		12
	Tap 4-40 beyond split	2	0.12		7.2
	End Mill to 1.25" depth & surface opp. sid	2	0.48		28.8
	Slot Mill 0.1875" cntr channel	1	0.24		14.4
	Cut in half	1	0.2		12
		13	1.7	1.133	170
Counterweight Bearing Web	Drill cntr hole	1	0.05		3
	Drill & Ream for 5/0 taper pin	2	0.33		19.8
	Drill .188D spotface	2	0.08		4.8
	Drill .120 to split	2	0.2		12
	Tap 4-40 beyond split	2	0.12		7.2
	End Mill to .125" depth both surfaces	2	0.64		38.4
	Slot Mill 0.1875" cntr channel	1	0.64		38.4
	Cut in half	1	0.2		12
		13	2.26	1.506666667	226
Drive Frame	Drill 1.10" holes	6	0.225		13.5
	Drill 0.25" holes	6	0.225		13.5
	Counter sink & counter bore 0.25" holes	6	0.9		54
	Drill 2.24" cntr hole	1	0.056		3.36
	Slot Mill 1.25" x 2" deep groove(inner face	1	0.28		16.8
	End Mill both faces	2	1		60
	End mill edges from both sides	2	0.28		16.8
		24	2.966	1.977333333	296.6
Counter Shaft Assembly					
Bracket	Drill 1.125" hole	1	0.3		18
	Counter bore both ends of 1.125" hole	2	1		60
	Drill & tap 1/4-28	1	0.1		6
	Drill & tap 6-32	4	0.4		24
	Saw Mill both sides	2	1.98		118.8
	Face Mill both Ends	2	0.32		19.2
		12	4.1	2.733333333	410
Elbow	Drill & tap 6-32	2	0.05		3
	Drill 0.189 thru hole	1	0.01		0.6
	Side mill web	1	0.02		1.2
		4	0.08	0.053333333	8
Scrolls	Side mill inner involute surface	2	2.17		129.9
	Side mill outer involute surface	2	2.50		150.2
	End mill tip seal groove	1	1.20		72.0
	Drill 1.88" deep holes	6	0.84		50.4
		11	6.71	4.472902672	670.9
Standoff	Drill & tap 10-32	6	0.15		
	Drill .002 holes	12	1.50		
	Bore 0.505 side wall through holes	10	1.50		

Machining Operations				Percent of ops time			
			< 25 operations	60%			
			> 25 operations	40%			
			Time (min)				
Part	**Operation**	**Operations**	**Total Process Time**	**Setup/travel time**	**Total (sec)**		
	End mill 1.75 dia through holes	2	0.30				
	Face mill 0.52 rad corners	8	0.16				
		38	3.61	5.415000000	541.5		
Support tubes	Cut sections	6	0.14		8.4		
	Debur	12	0.36		21.6		
		18	0.5	0.333333333	50.0		
Drive Frame Steel Inserts	Cut sections	12	0.4				
	Debur	24	1.1				
		36	1.5	2.250000000	225.0		
Couterweight Steel Inserts	Cut sections	6	0.2				
	Debur	12	0.55				
		18	0.75	0.5	75.0		
Housing Disc Steel Inserts	Cut sections	6	0.2				
	Debur	12	0.55				
	End Mill	6	1.62				
	Angle Cut	6	1.2				
		30	3.57	5.355	535.5		

		Time per part (sec)		Total Time per unit (sec)			Sub-Assembly Times	
# of *repetitions*		Mechanical Assembly	Cleaning	Mechanical Assembly	Cleaning			
	Countershaft Assembly							
1	position bearing(2), countershaft & bracket	20		20	0			
1	press	10		10	0			
1	position pulley	10		10	0			
1	set pulley in place with screw	45		45	0			
				Countershaft Assembly			**85**	
	Housing Disc Assembly A			0	0			
1	position	10		10	0			
6	press inserts into housing disc	5		30	0			
1	apply shaft seal	5		5	0			
	Orbital Seals			0	0			
1	position	10		10	0			
12	press seals into orbital discs	5		60	0			
	Drive Frame Assembly A			0	0			
1	position	10		10	0			
6	press inserts into drive frame	5		30	0			
	Counterweight Assembly			0	0			
1	position	10		10	0			
6	press inserts into counterweight	5		30	0			
	Drive Frame Assembly B			0	0			
1	position	20		20	0			
6	press inserts into drive frame	5		30	0			
	Housing Disc Assembly B			0	0			
1	position	20		20	0			
6	press inserts into housing disc	5		30	0			
1	apply shaft seal	5		5	0			
				Pre-Assembly			**300**	
	Main Housing Sub-Assembly			0	0			
1	position housing disc assembly A	5		5	0			
1	*position crankshaft*	5		5	0			
1	clamp bearing web halves	5		5	0			
1	tighten with screws	10		10	0			
1	fit 2" O-ring	5		5	0			
1	position housing barrel	5		5	0			
12	insert housing tie-rods	5		60	0			
1	position drive frame assembly	5		5	0			
1	clamp bearing web halves	5		5	0			
1	tighten with screws	10		10	0			
1	fit 3" O-ring	5		5	0			
6	clean roller bearings		4	0	24			
6	lubricate rollers		4	0	24			
6	position rollers	5		30	0			
1	position counterweight assembly	5		5	0			
1	clamp bearing web halves	5		5	0			
1	tighten with screws	10		10	0			
1	fit 2" O-ring	5		5	0			
6	clean tube bearings		4	0	24			
6	lubricate support tubes		4	0	24			
6	insert support tubes	5		30	0			
6	clean roller bearings		4	0	24			
6	lubricate rollers		4	0	24			
1	position drive frame assembly	5		5	0			
6	clean tube bearings		4	0	24			
6	lubricate support tubes			0	0			
1	position housing disc	5		5	0			
11	tighten nuts over housing tie-rods	5		55	0			
				Main Housing Sub-Assembly			**438**	
	Compressor Sub-Assembly			0	0			
1	place pivot over housing tie-rod	2		2	0			
2	fasten countershaft assembly	10		20	0			
1	to the main housing assembly			0	0			

		Time per part (sec)		Total Time per unit (sec)			Sub-Assembly Times	
# of repetitions		Mechanical Assembly	Cleaning	Mechanical Assembly	Cleaning			
1	position pulley	10		10	0			
1	set pulley in place with screws	10		10	0			
1	adjust tension	45		45	0			
6	insert scroll tie-rods	5		30	0			
6	position washers over scroll tie-rods	3		18	0			
6	position orbital seals over tie-rods	3		18	0			
1	place belt around pulley and adjust	60		60	0			
1	position orbital scroll (compressor)	17		17	0			
1	apply foam spring stock	10		10	0			
1	apply tip seal over foam	10		10	0			
1	position standoff collar	10		10	0			
12	attach standoff collar	5		60	0			
1	position fixed scroll (compressor)	10		10	0			
6	attach fixed scroll (compressor)	5		30	0			
				Compressor Sub-Assembly			**360**	
1	rotate compressor sub-assembly	10		10	0		10	
1	repeat steps for compressor sub-assembly						**360**	
				Final Assembly			**1553**	secs
							25.9	min
				Labor Cost ($/hour)			$ 43.65	
				Final Assembly Cost			$ 18.83	


Material for Machined Parts
25%
Part Fabrication Labor
48%
Purchased Parts
20%
Assembly Labor
7%

Weight (lbs)		56	
		cost/unit	percent
Part Fabrication Labor	$	134.59	48%
Assembly Labor	$	18.83	7%
Purchased Parts	$	57.39	20%
rial for Machined Parts	$	70.67	25%
TOTAL	**$**	**281.48**	

Point	**X**	**Y**						**Diameter**
1	18.27	0.51		outer 1	5.591	2.186		6.00
2	18.07	0.42		outer 2	4.611	1.856		4.97
3	17.29	0.18		outer 3	4.224	1.563		4.50
4	17.06	0.08		outer 4	2.645	1.104		2.87
5	16.28	-0.163		outer 5	1.47	0.583		1.58
6	16.07	-0.263						
7	14.81	-0.746						
8	14.616	-0.833		inner 1	5.192	1.985		5.56
9	13.846	-1.143		inner 2	4.182	1.645		4.49
10	13.635	-1.267		inner 3	3.214	1.223		3.44
11	12.878	-1.565		inner 4	2.224	0.88		2.39
12	12.679	-1.676		inner 5	1.26	0.483		1.35
* Length of path that the tool has to travel along the inner and outer surface of the involute.								
** Involute distance was approximated as the total of 5 semi-circles.								
Machinery Cost								
Dm	US$							
750,000	$ 492,975							
900,000	$ 591,570							

	Semi-Circle	
	9.429734745	
	7.807672651	
	7.074714771	
	4.502144791	
	2.484038955	
TOTAL	**31.30**	inches
	8.731294944	
	7.059003674	
	5.401695419	
	3.756987559	
	2.119637592	
TOTAL	**27.07**	inches