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DESIGN OF RECIPROCATING SINGLE CYLINDER EXPANDERS FOR STEAM

FINAL REPORT

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Water Programs
Office of Mobile Source Air Pollution Control
Alternative Automotive Power Systems Division
Ann Arbor, Michigan 48105

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1.0 INTRODUCTION

The water base Rankine cycle powerplant with a reciprocating expander has been identified as one of several possible low emission alternatives to the internal combustion engine for automotive application. To achieve the performance necessary to make the automotive steam engine a truly viable alternative requires temperatures and pressures in the order of 1000°F and 1000 psia respectively. At these elevated operating conditions conventional methods of lubricating the upper cylinder walls and inlet valves of a reciprocating expander can pose difficult problems. In an effort to find a solution to the lubrication problem, the Advanced Automotive Power Systems Development Division of the Environmental Protection Agency sponsored a program with the General Electric Company, Space Division, to analytically and experimentally evaluate self-lubricating materials suitable for the automotive steam engine application.

This program included the design of practical single cylinder reciprocating expanders for the purpose of evaluating solid lubricants and other supporting materials for use in a Rankine cycle automotive propulsion system utilizing a water base fluid. This investigation had the following objectives:

- Establish preliminary design of trunk piston and crosshead piston single cylinder expanders.
- Evaluate expander materials technology for higher temperature operation with solid lubricants.
- Evaluate solid and liquid lubricants for reciprocating expanders.
- Build and test one trunk piston expander and one crosshead piston expander at conditions representative of the engine system.

Following the design and fabrication of two single cylinder reciprocating expanders - a crosshead piston expander and a trunk piston expander -

these expanders with different combinations of ring, cylinder and valve stem ring materials were tested at pressures and temperatures up to 1000 psia and 1000°F, respectively. Piston ring materials that were evaluated were graphite impregnated with antimony (Carbone Co. Grade CC5A) and a Cr_3C_2 cermet (Koppers Co. K-1051) coated Inconel X-750. Both of these materials were tested with a hardened Type 440-C stainless steel cylinder liner with no liquid lubrication. In addition to the power piston, the expander inlet steam valve required unlubricated valve stem guides and seals. Seal rings for the valve stem were made of cold worked Haynes alloy No. 25 and some were made of 17-7PH coated with DuPont LPA-101 hard facing. These valve stem rings rubbed against the valve housing which was a cast Ni-Resist Type 3D. The valve and valve stem were nitrided H-11 tool steel.

For the trunk piston expander a rider ring was provided to support the high piston side loads. The rider ring was a split ring design made of Pure Carbon Company graphite Grade P5NR. Also, two oil exclusion rings and a steam seal ring were provided at the bottom of the trunk piston. These were standard rings made of K-35 Ni-Resist, K Iron and K-6E Iron.

The expander crankcase was liquid lubricated in a conventional manner of splash and pressurization. However, since the expander was expected to have hot spot temperatures exceeding 400°F, and since some steam blowby was expected to accumulate in the crankcase, a specially compounded synthetic hydrocarbon oil (Grade XRN-1301-C) supplied by the Mobil Research and Development Corporation was used and evaluated. This lubricant contained oxidation, rust, wear, and foam inhibitors.

In order to get meaningful results on piston ring and cylinder wear and friction, it was necessary to exclude essentially all oil from the power cylinder. In the crosshead expander this was not too difficult following some minor design changes, but the conventional metallic oil exclusion rings for the trunk expander were not very effective in excluding oil from the steam or steam from the crankcase oil. Chevron ring seals made of PTFE impregnated with bronze and MoS_2, mounted in a floating housing to minimize radial loads on the seal, were successful in excluding oil from the steam to less than 4 ppm in the crosshead expander. Water accumulation in the crankcase for the crosshead expander averaged 8.1 m/hr - a possible major leakage path being the inlet valve stem seal. Testing of the trunk piston expander was of short duration (approximately 9.2 hours) and accurate data on the rate of oil migration into the steam or expander condensate was not obtained. However, the rate was significantly higher than for the crosshead expander.

Of the few materials tested, the best results were obtained with graphite (CC5A) rings rubbing against a hardened Type 440-C cylinder liner in the crosshead expander. For the first 200 hours of testing the total radial ring wear was approximately 0.040 inches - giving a wear rate of 0.0002 inch/hour. At 240 hours total wear was sufficient to cause ring breakage. Wear performance of the Cr_3C_2 coated Inconel X-750 rings in a Type 440-C liner was much less impressive. Ring wear was rapid which resulted in ring breakage in less than ten hours of testing.

Even with the graphite rings, ring/liner friction and steam blowby were higher than predicted. At 1000 psia, 1000°F steam inlet conditions theoretically indicated efficiency was calculated to be 85%, at 2000 RPM. Measurements for these conditions gave an engine efficiency of 55%.

Measurement of cylinder pressure as a function of crankangle proved to be a challenging task. Several types and makes of pressure transducers were tried with only limited success. Best results were obtained with a water cooled, inert gas buffered Dynisco Model PT49A pressure transducer.

2.0 ABSTRACT

A reciprocating expander has been identified as one of several possible types of expanders which may be applicable to an automotive Rankine cycle engine. For high engine efficiency, steam pressure and temperature up to 1000 psia and 1000°F, respectively, are necessary. One approach to steam cylinder lubrication is the application of solid lubricants which are capable of withstanding these high pressures and temperatures without excessive friction and wear.

Several solid lubricants and wear resistant materials were tested in a single cylinder steam expander of both the crosshead piston and trunk piston configuration. Also a specially compounded water resistant synthetic hydrocarbon oil was evaluated as a crankcase bearing lubricant. Both the crosshead piston and trunk piston expanders were fabricated and tested over a range of conditions depicted as follows:

Speed Range, 500-2000 RPM
Inlet Steam Temperature, 700-1000°F
Inlet Steam Pressure, 400-1000 psia
Condenser pressure, ∿ 20 psia

The crosshead piston expander was first tested for a total of 242 hours with antimony impregnated carbon-graphite rings (Grade CC5A) rubbing against a hardened Type 440-C stainless steel cylinder liner. Expander performance was generally as predicted by earlier analysis with the exception of higher blowby than anticipated. The carbon-graphite ring wear was excessive for a projected 3000-hour life. A second crosshead piston expander buildup incorporating Cr_3C_2 coated Inconel-750 rings and a Type 440-C liner was tested and after six hours of operation at temperatures and pressures of 1000°F and 700 psia respectively, serious blowby was evident. Inspection of the ring and liner surfaces revealed excessive roughness and the test was terminated.

The trunk piston expander was assembled with Cr_3C_2 coated Inconel X-750 sealing rings, a carbon-graphite (Grade P5NR) rider ring, and cast iron oil exclusion rings at the bottom of the piston. The cylinder liner was hardened Type 440-C stainless steel. Performance of the trunk piston expander was not as good as the crosshead piston expander in terms of shaft horsepower output, brake specific steam consumption, etc. After approximately 9.2 hours of testing, failure of a crankshaft bearing terminated the test. Piston ring and cylinder liner scoring and wear was excessive.

Generally, the test results indicate that solid lubricants are potentially applicable as ring, cylinder, and other components in a high temperature reciprocating expander. Although those few materials tested under this program exhibited relatively short life and no completely satisfactory combination of wear resistant and solid lubricant materials were identified, the very limited number of tests conducted are not sufficient to rule out a non-liquid lubricated reciprocating steam expander. The synthetic hydrocarbon crankcase lubricant performed well through all the tests providing adequate lubrication at bulk oil temperatures up to 230°F.

3.0 CONCLUSIONS AND RECOMMENDATIONS

To some extent it is difficult to separate design from material success or problems. However, from a pure design point of view both the crosshead piston and trunk piston expanders were designed and proved to be sturdy test vehicles for steam expander component testing. The very rugged Waukesha CFR-48 crankcase proved to be reliable during all phases of testing except when a main bearing was starved of oil. (This was no fault of the crankcase.)

In both expander types the heavily loaded valve cam was marginal for long time operation at high speed. Also, the crosshead piston wrist pin was somewhat marginal for long time operation. However, neither component presented a real operational problem following minor corrections. The power piston connecting rod seal for the crosshead expander required early modifications but later proved to be trouble free.

Testing of the trunk expander was short and was terminated following a bearing seizure. However, during less than ten hours of operation it was evident that oil was not being excluded from the steam system and vice versa.

Both expanders were well balanced dynamically as indicated by low vibration. However, expander noise level was considered high - mostly valve gear noise. From a mechanical design point of view the program objectives were generally met with good success. However, testing time was too short to uncover possible long term deficiencies in the designs.

Design improvements can be made to any reciprocating steam expander depending on the ground rules or limitations set forth. For an unlubricated piston and cylinder the problem becomes one of identifying the right materials for sliding components, or of designing away from sliding (non-contacting) components in regions where lubricants are not viable. Studies have shown that a properly designed labyrinth sealed piston, which would not contact the cylinder wall, might give performance as good or better than a piston fitted with graphite rings. When properly supported the labyrinth piston would not be subjected to wear. Such piston designs have been used in steam engines and gas compressors with good success.

Under the test conditions of 1000°F, 1000 psi steam, the carbon-graphite piston rings (Grade CC-5A) rubbing against a hardened Type 440C SS cylinder liner showed far superior performance to the hard-hard materials combination of Cr_3C_2 cermet against Type 440C SS. Although the wear rate of the upper carbon-graphite ring was too high (0.070 inch in 242 hours) for a 3000 hour life, the wear rate of the lower ring appears to be acceptable. It is recommended that further study be carried out to redesign the rings in an attempt to lower the unit loading and to evaluate other carbon-graphite materials that may have improved wear resistance. For example, it is known that carbon-graphites recently developed for the Wankel engine tip seal are compatible in steam and have superior wear resistance to the CC-5A grade. Also, alternate design configurations could greatly improve the performance of a solid lubricated system. For example, making the cylinder of a carbon graphite instead of the piston rings provides much more solid lubricant surface - thus significantly increasing the life of the system.

The performance of the work hardened Haynes alloy No. 25 valve stem rings against the cast Ni-Resist D-3 valve housing was relatively good; the LPA-101 hard facing coating on the rings is not suitable for the steam seal application. Improvement in wear of the housing and steam leakage through the seal may be possible through the use of a wear-in coating to be applied either to the ring surfaces or the ID surface of the Ni-Resist housing. Other material combinations that may provide improved performance are 1) carbon-graphite housing vs ringless stem, 2) Cr_2O_3 coated or solid cast LPA-101 rings vs nitrided steel housing.

The 15% bronze + 5% MoS_2 filled PTFE piston rod seal in the crosshead expander performed flawlessly. No measurable wear was observed and oil leakage into the steam condensate was maintained at values of <4 ppm.

The crankcase lubricant, Mobil XRN-1301C, provided adequate lubrication to all the bearing surfaces and the cam lobe/tappet interface for a period of 187 hours. Water concentration as high as 1.3% and bulk oil temperatures of 250°F did not affect the lubricant and there was no evidence of serious oxidation or degradation of any kind of the lubricant.

4.0 EXPANDER DESIGN

4.1 Expander Design Technology Base

In conformity with contract guidelines, the reciprocating expander configuration was selected to be that of a single acting, uniflow engine. Two single cylinder expanders were to be designed, built, and tested - one a crosshead piston type and the other a trunk piston type. Steam inlet conditions were to be 1000°F, 1000 psia with an exhaust pressure of 20 psia. These expanders were to be representative of one cylinder of a four cylinder engine of approximately 150 hp output, suitable for automotive application. A governing requirement of these designs was that solid lubricants be employed for the piston rings, piston rod packing (where employed), and valve stem seals. Such use of solid lubricants was intended to initiate the development of an engine free of lubricating oil contamination in the steam.

The task of establishing the technology base for the design of the two expanders was implemented by making the following investigations:

1. High speed steam engine valving configurations, and valve gear designs.
2. Wrist pin bearing design suitable for high unidirectional loading.
3. Oil free high temperature piston ring design.
4. Oil free valve stem sealing, friction, and wear.
5. Piston rod sealing.
6. General reciprocating engine mechanical design.

In close coordination with these investigations corresponding materials selection investigations were carried on. These are described in Section 5.0.

Salient results and conclusions of these investigations are briefly discussed below.

4.1.1 High Speed Steam Engine Valving Configurations and Valve Gear Designs

Valving configurations which were identified as being of interest for preliminary design studies include the following:

1. Uniflow engine with spring loaded recompression relief exhaust valve, simple exhaust porting and engine actuated inlet valve, either in the cylinder head or in a side steam inlet passage.

2. "Zero" clearance volume engine with exhaust valve and inlet valve in the cylinder head.

Valve types of interest include: (a) poppet valves of several varieties, (b) sleeve valves, and (c) rotary valves. Sleeve and rotary valves minimize or eliminate the high pressure loading on the actuating mechanism. Rotary valves in particular are of interest because they offer potential capability for a simple variable cut-off mechanism. However, because of the formidable sealing problems of these valve types, it was decided that such an approach involved too much technical risk for the present program. Investigative effort was concentrated on unbalanced and partially balanced cam actuated poppet valves. Principal design problems were identified to be high Hertz stress between cam and cam follower for an unbalanced poppet valve approach, and that of steam leakage for various types of partially balanced poppet valves. It was recognized that valve train accelerations are quite high because the cam shaft must operate at engine speed as opposed to half engine speed for four cycle internal combustion engines. However, it was found that the valve train forces, at least for an unbalanced poppet valve, are dominated by high pressure steam loading forces rather than acceleration forces. Although steam forces produce a high Hertz stress, dynamic design of the valve train is somewhat simplified.

One technique for expander load control for a four cylinder engine is that of inlet valve variable cut-off which must cover a range from near zero steam admission to a maximum value determined by the limit of the steam supply. The status of high speed variable cut-off technology was reviewed, and the following possible alternative design approaches were identified.

1. The Caprotti mechanism, which includes a single poppet valve actuated by a yoke bearing on two variably phased cams. (This mechanism is successfully employed in Skinner steam engines.) For a high speed application, requiring a wide range of cut-off, very high valve train accelerations are involved.

2. A hydraulic valve actuator operating on the principle of the Bosch injector has been successfully employed for achieving variable valve period by the Waukesha Motor Company and others. For a high speed steam expander requiring a camshaft running at full engine speed, however, it appeared doubtful that such a system has a fast enough response. This conclusion was supported by a communication with American Bosch Corporation. However, it was learned that an advanced electro hydraulic variable cut-off design for high speed steam engines is under development by American Bosch for Thermo Electron Corporation.

3. A variable fulcrum type of overhead valve actuation mechanism has been invented and developed by the Ford Motor Company. Preliminary design studies of several forms of this device were made with the cooperation of the Ford Motor Company. Various problems relating to overall expander height, and lubrication of the variable fulcrum mechanism resulted in a decision not to pursue this approach. A further complication is the fact that this approach to variable cut-off requires a mechanism to vary the phase of the cam relative to the crankshaft.

4. A rotary valve mechanism utilizing a rotatable cut-off sleeve between the rotating valve and the valve housing was reviewed. As mentioned above, the formidable sealing problems of this approach required extensive development effort - not consistent with the current contract.

5. Two poppet valves in series, variably phased to provide a change in the period of overlap between opening of each valve have the following advantages:

a. Long cam event angles can be employed. This minimizes the problems of cam design - permitting reasonable acceleration, high lift, and reasonably small cam base circle diameter.

b. Full cut-off range (zero to any desired maximum) can be readily achieved.

Problems are also recognized for this approach as follows:

a. Because of the flow resistance of two valves in series the valves must be larger or employ higher lift than if a single valve is used. Large valves result in high valve train loads unless balanced valves are used. Very high lift tends to require long cam events, large cam base diameter, high cam rubbing velocity and high valve train acceleration.

b. With two series valves, it is difficult to obtain the desired 5% clearance volume in a uniflow type expander.

For two poppet valves in series two alternate cam phasing mechanisms were identified:

a. One involving the use of helical splines, and

b. The other involving a planetary gear train between crankshaft and camshaft with phasing variation achieved by rotating the planet carrier.

After evaluating the above alternate approaches to variable cut-off, it was concluded that the two series poppet valves had the lesser development risks. However, for the single cylinder test engines a single poppet valve with fixed cut-off was chosen for simplicity since the primary objective was to evaluate solid lubricants.

4.1.2 Wrist Pin Bearing Design for High Unidirectional Loading

The high MEP of the steam expander under long cut-off, and the fact that the wrist pin load does not reverse direction (does not facilitate squeeze film lubrication of the wrist pin bearing) required that special attention be given to the wrist pin design. Applicable technology is found in the design of wrist pin bearings for two cycle Diesel engines. Several approaches are as follows:

1. Employment of needle type antifriction bearings.

2. Employment of a palm type bearing which provides for relatively low pressure loading over a large area.

3. Use of a helically grooved bushing in the connecting rod with lubricating oil force fed to the grooves, the wrist pin being fixed in the piston. (Pitch of the grooves must be less than the amplitude of the rocking motion so that the motion drags oil from the grooves over the lands of the bearing.)

4. Use of helically grooved bushings in the piston with the piston pin fixed to the connecting rod.

Consultation with the General Electric Diesel engine design group resulted in the selection of approach No. 4 above. Unit loading in the order of 4000 psi were considered to be safe, since loadings in excess of 5000 psi have been successfully used in two cycle Diesel engines.

4.1.3 Oil Free High Temperature Piston Ring Design

Friction, wear, and sealing effectiveness of oil free piston rings are the most important questions toward which this experimental program is directed. Basic design parameters of interest are the following:

1. Number of rings employed for 1000 psi sealing pressure
2. Width of rings
3. Use of pressure balancing to reduce normal force and resulting friction and wear.
4. Segmentation of rings and sealing between segments
5. Design of springs for preloading segmented rings
6. Provision for positive sealing behind the ring
7. Rubbing velocity.

An extensive survey was made in the literature and by contact with manufacturers and users of carbon and other types of high temperature oil free rings. No directly comparable application was identified. The most pertinent experience was that with high pressure oil free compressor piston rings. Recommendations for number of rings, and width of rings varied widely. Values selected do not represent the most conservative practice, but are the result of judgments which balance design for low friction and wear against the requirements of an acceptably simple and light weight design. In references 1, 2, and 3, some bases for analytical prediction of ring life were found, and the following general design features were applied:

1. For non metallic rings (graphite) two piece segmented construction was applied with a one piece spring backup. For metallic rings (coated) one piece construction was used.

2. Ring widths were approximately .3" for non metallic and .125" for coated metallic rings. Three non metallic compression rings were used for the crosshead design and two for the trunk design (because of smaller available space). For an alternate backup ring design, three coated metallic rings were used for both crosshead and trunk designs.

3. For non metallic rings, in which .060" to .080" radial wear was anticipated, holes were provided in the top edge of the ring groove to assure access of sealing pressure behind the ring, thus preventing ring flutter.

4. Pressure balancing was provided with the non metallic rings to reduce average normal pressure, friction, and wear.

4.1.4 Piston Rod Sealing

In the crosshead piston design, the steam piston rod must seal the condenser steam pressure (20 psia) against atmospheric pressure, and also prevent the transport of oil into the steam. Metal temperature due to steam contact was reasonably low (approximately 300°F). Seal friction, however, may produce local high temperature. Some cooling was provided through the use of a hollow piston rod into which oil was sprayed. Two alternate types of seals were identified for such an application:

1. A Chevron packing made of bronze filled PTFE

2. A segmented carbon ring packing held in place by garter springs.

4.1.5 Oil Free Valve Stem Sealing

Two approaches were selected for valve stem sealing:

1. The use of multiple precision metallic rings for sealing between the valve stem and the valve stem guide housing.

2. The use of carbon ring packing in a packing gland surrounding the valve stem.

Since assurance of success seemed high with (1), based on General Electric steam turbine valve sealing experience, this approach was selected for the test expanders.

4.2 Expander Preliminary Design Studies

Prior to the selection of detail features for the two single cylinder test expanders, studies were conducted in the following areas:

1. General cylinder configuration, cylinder sizing, and expander speed range.
2. Investigation of Ford Motor Company variable cut-off mechanism
3. Cam shaft drive configuration
4. Valve and valve stem seal configuration
5. Wrist pin bearing configuration
6. Piston rod packing configuration
7. Piston ring configuration for crosshead and trunk piston designs.

These studies are shown in Figures 4.2-1 through 4.2-6, and are briefly described below.

4.2.1 General Cylinder Configuration Selection

The principal alternatives with regard to the general cylinder configuration are:

1. A near zero clearance volume, valve-in-head design, with both inlet and exhaust valves located in the cylinder head.

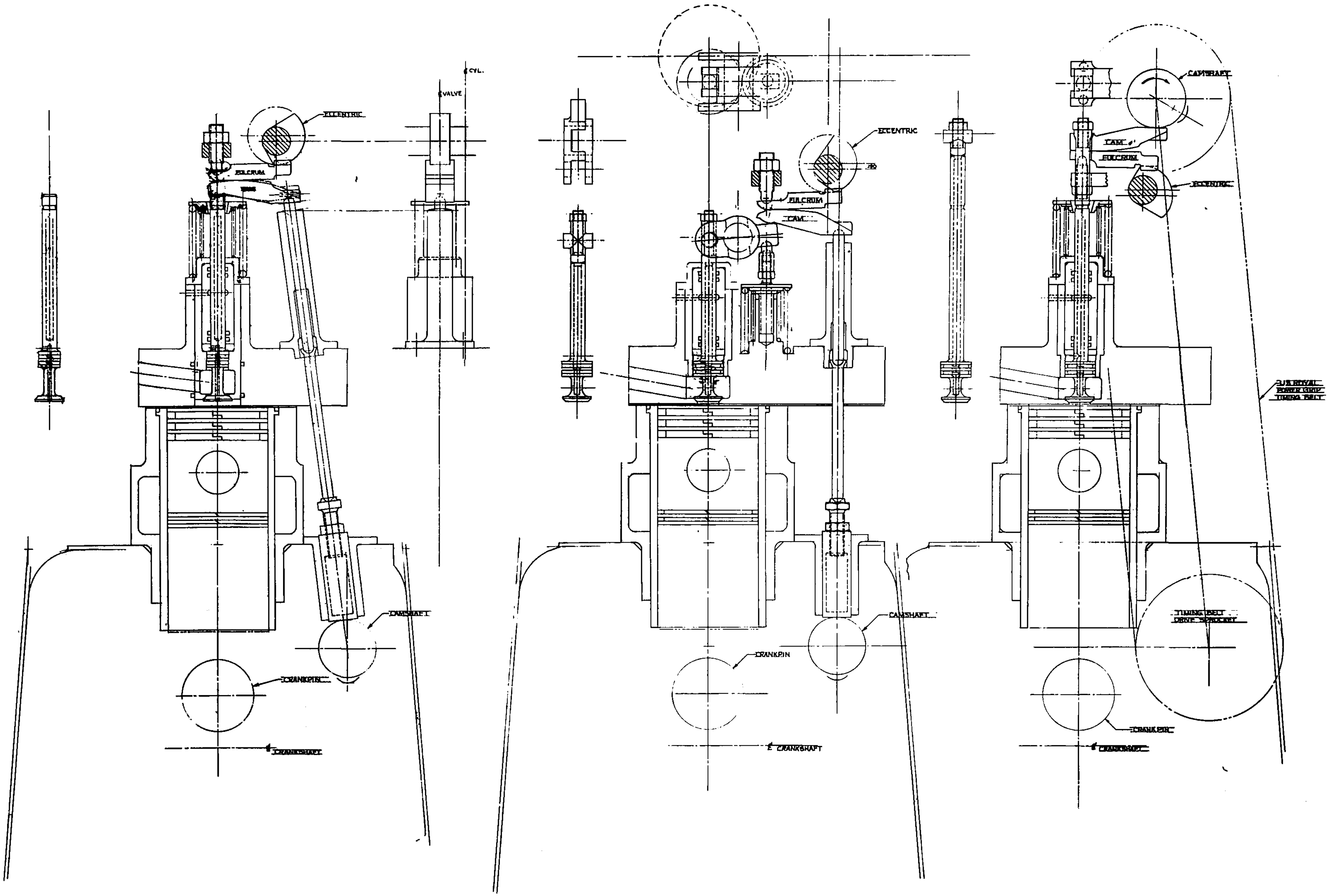


Figure 4.2-1. Valve - Variable Cut Off Mechanism.

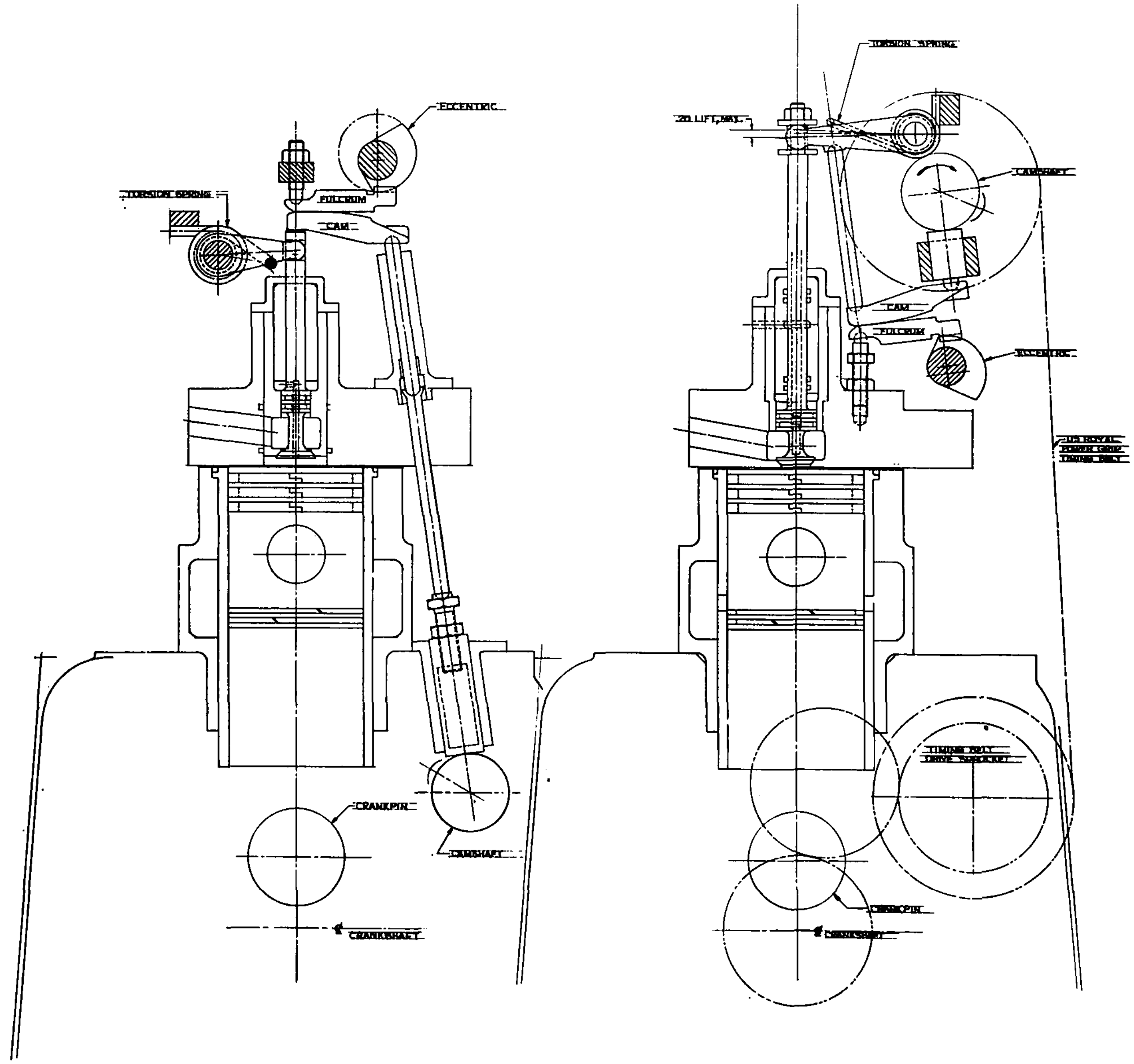


Figure 4.2-2. Valve - Variable Cut Off Mechanism.

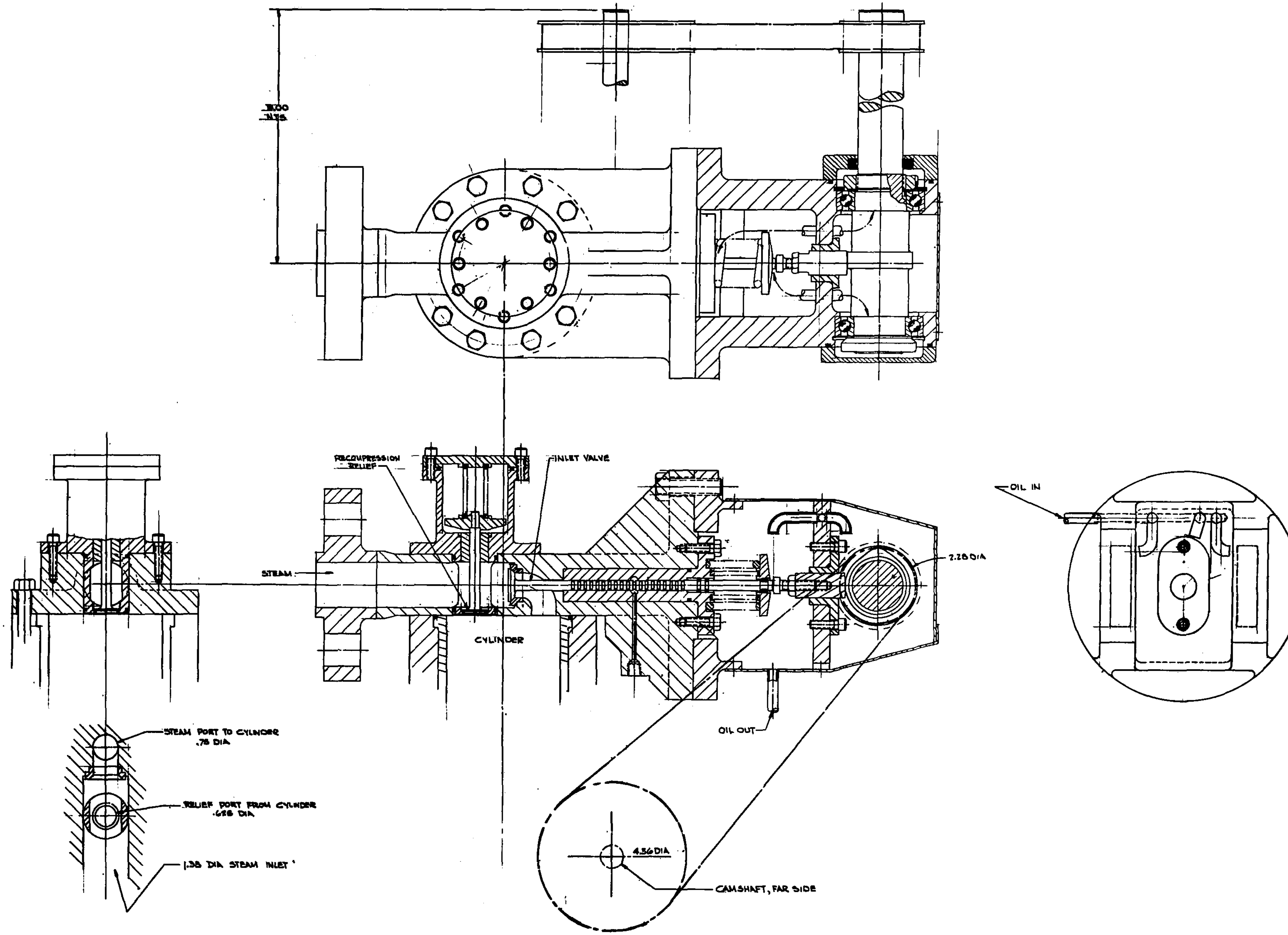


Figure 4.2-3. Cam Shaft Drive, Head, Valve, and Cam Details (GE Dwg. 707E702).

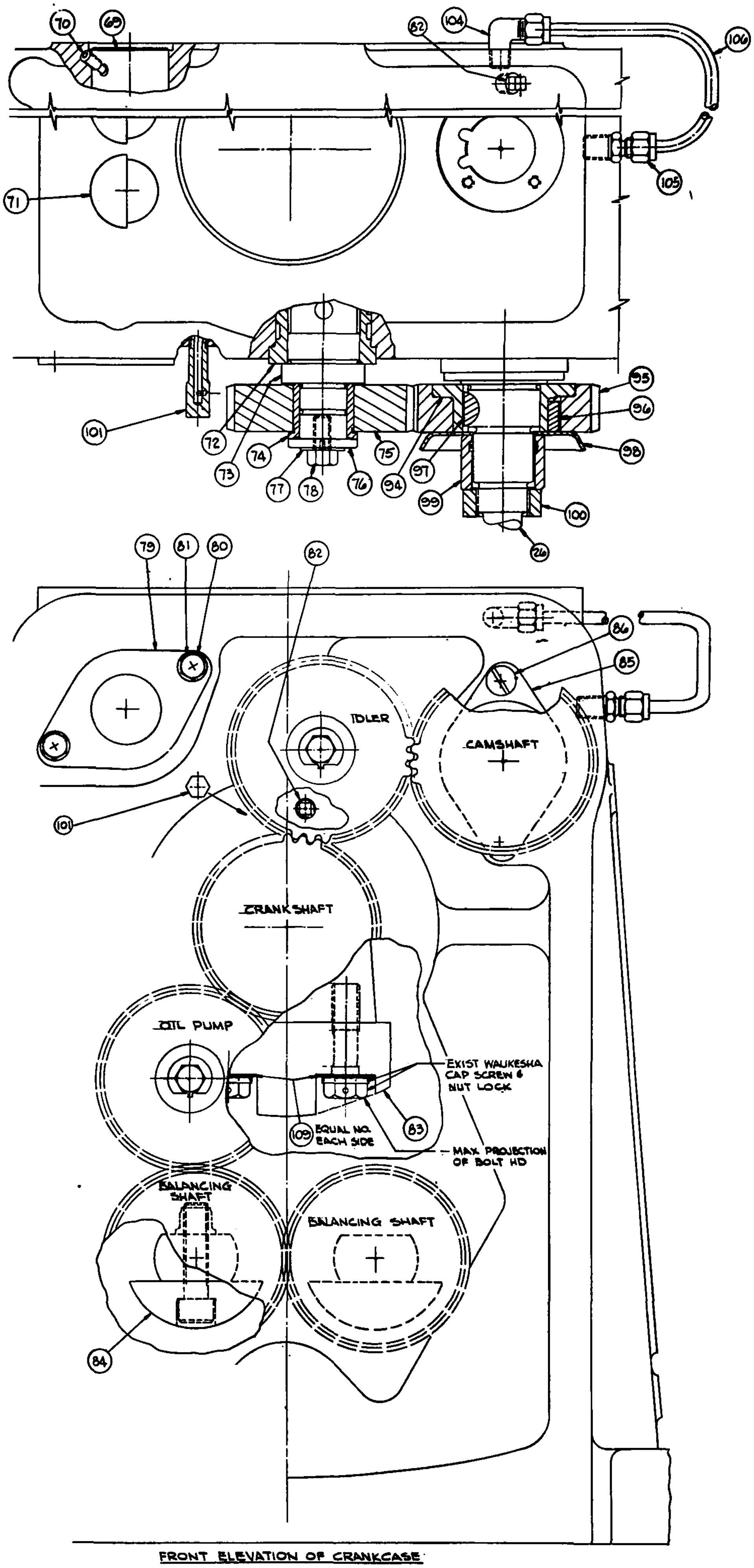


Figure 4.2-4 Cam Shaft Drive, Steam Expander Trunk Type Piston (GE Dwg. 707E719, Sh. 2)

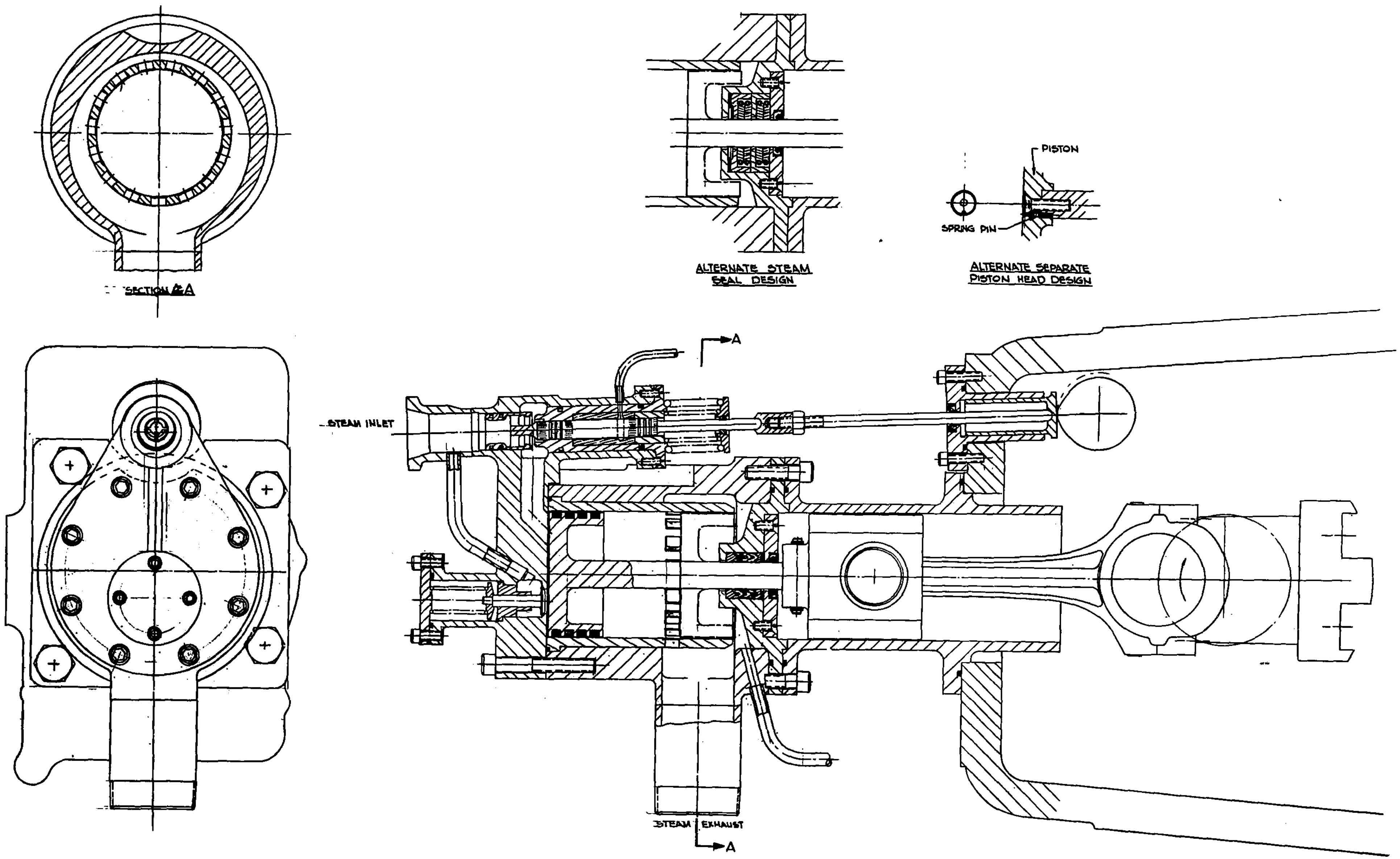


Figure 4.2-5. Piston Ring Packing - Sleeve Valve Steam Expander - Crosshead Type Piston (GE Dwg. 707E705).

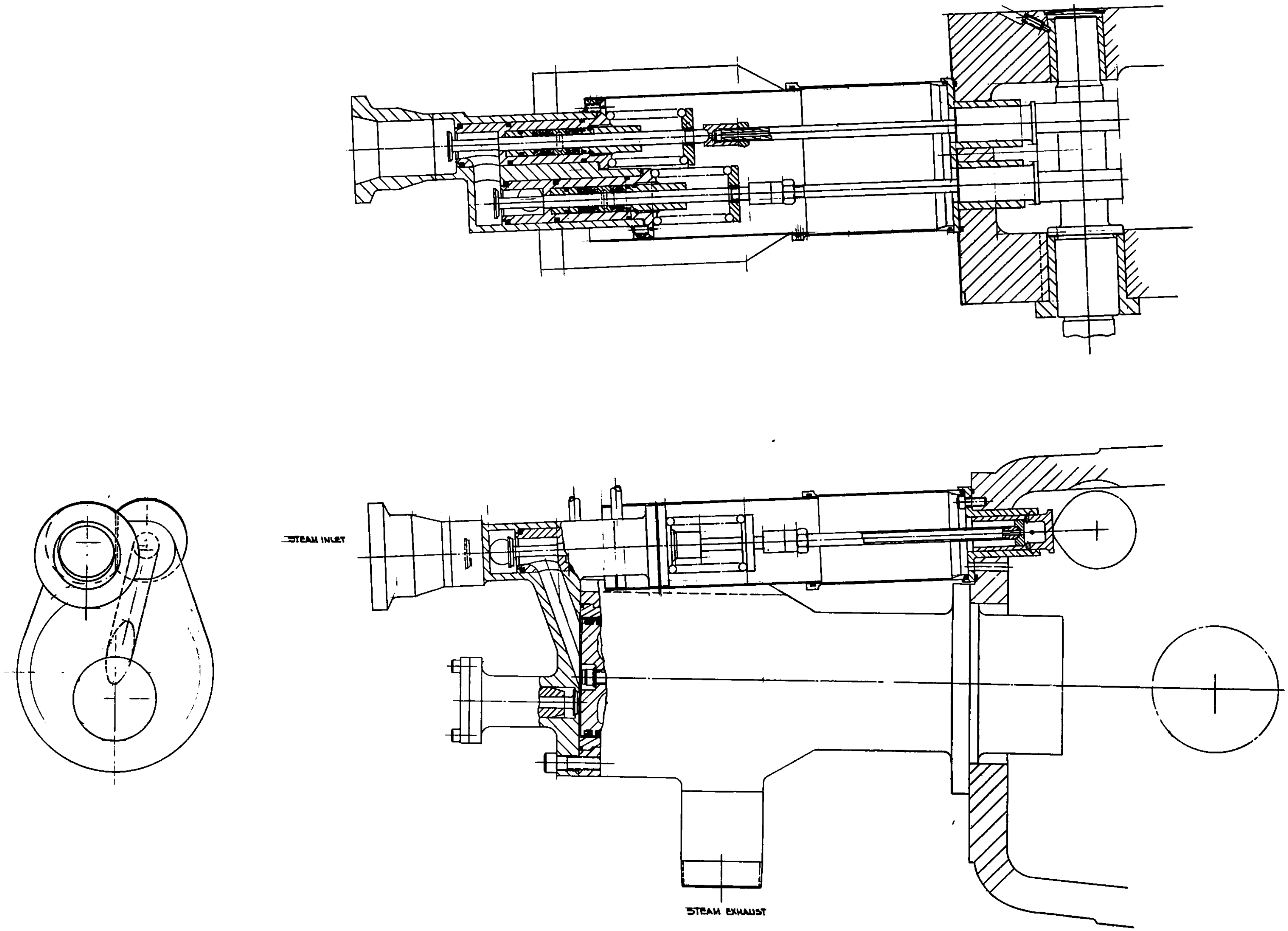


Figure 4.2-6. Valve Sleeve Packing - Cutoff Valve Steam Expander - Crosshead Type Piston (GE Dwg. 707E710).

2. A uniflow engine configuration with exhaust ports opening at approximately 90% of stroke and with an inlet valve located either in the head or in a side inlet passage.

Advantages and disadvantages of these alternatives can be summarized as follows:

Zero Clearance Volume Design Advantages

1. With the exhaust valve open throughout the upward stroke of the piston, work of recompression can be virtually eliminated. This results in a higher MEP and a smaller cylinder displacement for a given power output.

2. Expansion can extend to the end of the downward stroke of the piston - thus increasing the effective expansion ratio. This is a factor favorable to high efficiency.

Disadvantages

1. Cooling of the upper part of the cylinder occurs during steam exhaust. This results in "condensation loss" during the subsequent expansion.

2. High thermal gradients occur in the cylinder head due to the proximity of the inlet and exhaust steam. This is unfavorable from the standpoint of mechanical design reliability.

3. The need for an actuated exhaust valve in addition to an actuated inlet valve results in undesirable complexity.

Uniflow Design Advantages

1. The elimination of cylinder cooling and related losses associated with reverse flow exhaust. With proper design control of the clearance volume, efficiency can be kept equal to that of zero clearance volume design.

2. Cylinder recompression pressure reduces the unbalanced load on the valve train during the inlet valve actuation period.

3. Mechanical simplicity.

Disadvantages

1. Larger cylinder displacement for a given power output and efficiency.

2. Compression relief valve is required to prevent the recompression pressure from exceeding the inlet pressure - thus possibly causing premature opening of the inlet valve and subsequent damage to the valve train.

Largely from considerations of mechanical reliability and simplicity, the uniflow approach was chosen.

Thermodynamic calculations for a 170 IHP, 2500 RPM uniflow engine indicate an adiabatic efficiency of approximately 80% for an inlet pressure and temperature of 1000 psi and 1000°F at an exhaust pressure of 20 psi. This is for a clearance volume of about 5% and exhaust opening at 90% of stroke. A bore of 3-1/2" and a stroke of 3-1/2" with approximately 15% cut-off is required.

4.2.2 Investigation of Ford Motor Company Variable Cut-Off Mechamism

A survey of alternate variable cut-off mechanisms led to the conclusion that it might be feasible to make use of the Ford Motor Company variable cut-off mechanism on the test expanders. Accordingly, a design study of this mechanism was made at the Ford Plant. Results are shown in Figures 4.2-1 and 4.2-2. The essential feature of the mechanism is an overhead rocker arm linkage for which the effective pivot can be varied by the adjustment of an eccentric bearing. This changes the amplitude and event angle of the valve motion. As shown by Figures 4.2-1 and 4.2-2, the mechanism can be arranged to permit either inward or outward opening of the valve. For inward valve operation it is essential that a balanced valve be used to prevent steam inlet pressure from opening the valve. A further potential problem is the need for oil lubrication of the contact interface between the valve actuating rocker and the adjustable pivot. Perhaps the greatest disadvantage of the Ford mechanism for the steam car application is the increased expander height.

4.2.3 Cam Shaft Drive

The principal alternatives considered were the use of an overhead or crankcase mounted camshaft. The former would be belt driven while the latter would be driven from the main crankshaft through a gear train designed for a crankshaft to camshaft speed ratio of unity. These alternate arrangements are illustrated in Figures 4.2-3 and 4.2-4.

4.2.4 Valve and Valve Stem Seal

Considerable attention was given to the selection of the valve configuration, and to the design of the oil free valve stem seal. Valve types investigated included:

1. The double-beat and balance piston poppet valve
2. A ring sealed sleeve type valve
3. The simple unbalanced poppet valve.

The advantage of the balanced valves is the reduced loading on the valve train - specifically upon the cam and cam follower. The disadvantages of balanced valves are complexity and leakage. After detail sizing of a simple unbalanced poppet valve, and following calculation of the cam-tappet Hertz stress the unbalanced poppet valve proved to be quite acceptable.

Alternate types of valve stem seals are illustrated in Figures 4.2-5 and 4.2-6. Metallic piston ring seals and a graphite packing ring seal are shown. The former enjoys a substantial advantage in simplicity. The valve stem seal is divided into two parts with a drain to the condenser between them. This approach minimizes the pressure differential available for forcing steam into the expander crankcase.

4.2.5 Piston Rings

Design considerations relating to the compression rings have been discussed above. Two alternate designs, a graphite ring, and a coated metallic ring were selected - the second design to be used as a backup in the event of failure of the first.

In the trunk piston expander a graphite rider (side load carrying ring), and two oil scraper rings are provided at the bottom of the piston skirt. Piston side load is carried by the rider ring and an oil lubricated land at the extreme bottom of the piston skirt. Thus, side load bearing areas are provided both above and below the wrist pin. Above the oil scraper rings on the trunk piston is a metallic compression ring to seal condenser pressure from the crankcase.

4.3 Expander Description

4.3.1 Crosshead Piston Expander[4]

The crosshead expander shown in Figure 4.3-1 consists of an oil lubricated aluminum crosshead which is connected to the steel power piston by a rigid connecting rod. Isolation of steam and oil is accomplished by a rod seal between the crosshead and the power piston. The rod seal shown is a bronze impregnated Teflon chevron seal mounted in a floating housing. The power piston connecting rod is tubular for reduced weight, and to permit internal oil cooling. An oil jet located in the top of the crosshead wrist pin sprays oil into the power piston connecting rod primarily to remove heat generated by the connecting rod seal.

The wrist pin and sleeve bearing design is quite similar to current 2-cycle Diesel engine practice. Oil is supplied to the sleeve bearing under pressure through the wrist pin journal. The sleeve bearing contains a circumferential groove and closely spaced helical grooves to permit the free flow of oil to all surfaces of the wrist pin.

Figure 4.3-1 shows three relatively wide rings which are pressure balanced segmented graphite rings backed up by an Inconel ring spring. The expander has a removable cylinder liner so that appropriate combinations of piston ring and cylinder materials can be evaluated with minimum difficulty.

The L-head poppet valve configuration insures that lubricating oil is well isolated from the high temperature parts of the expander - thus eliminating the need for thermal insulation to prevent high oil temperatures. Also the orientation of the valve gear minimizes the risk of oil entering the steam system by way of the steam inlet valve.

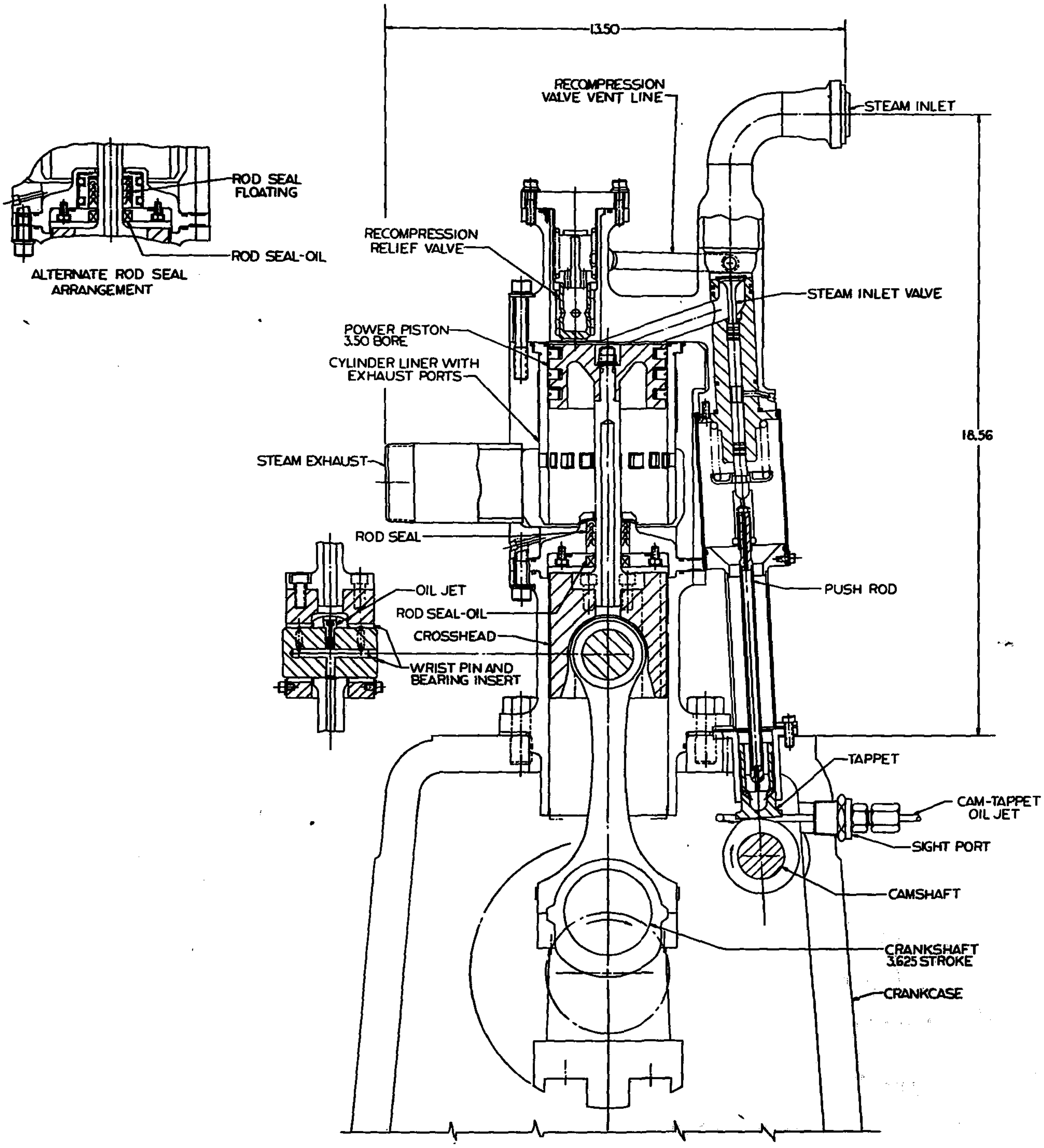


Figure 4.3-1. Crosshead Piston Expander (Dwg. 707E742)

The inlet valve assembly is designed so that the repair of critical components such as valve stem seals and valve seat can be accomplished with minimum rewotk. In addition, different material combinations for the valve and valve seat, and for the valve stem bearing and seals can be evaluated merely by replacing the valve and/or valve housing subassembly without the necessity of remachining major components.

The design of the valve gearing minimizes mass and linkage flexibility - thus providing fast valve response and minimal inertia forces throughout the valve gear system. The valve gearing is oil lubricated except for the top portion of the valve stem which is provided with solid lubrication. The valve cam located within the crankcase receives some oil by splash lubrication, but an oil jet is provided to spray oil directly on the valve cam to insure sufficient lubrication between the cam and cam follower.

A recompression relief valve is provided in the cylinder head. This valve is designed to relieve the cylinder pressure under all conditions of expander operation, including motoring, preventing the steam inlet valve from being forced open due to excessive cylinder pressure.

4.3.1.1 Expander Crankcase

Figure 4.3-2 is a cross section of the Waukesha Motor Company Model CFR-48 crankcase assembly used in the crosshead piston steam expander assembly. Figure 4.3-3 illustrates the basic cam gear arrangement after being modified for a 1 to 1 speed ratio between crankshaft and camshaft.

The CFR-48 crankshaft has a 3.625" stroke, 3.00" diameter main journals and 2.50" diameter crankpin. All journals are nitrided and ground to an 8 RMS finish. Main bearings, bronze sleeve type, are pressure lubricated through oil galleys in the crankcase. Oil is supplied to the crankpin through holes in the crankshaft from the front main bearing. Crankshaft thrust loads are restrained by slotted face type bronze bearings on each side of the left main bearing (refer to Figure 4.3-2). The right end of the shaft is provided with a 16" diameter steel flywheel, and provides for power take-off.

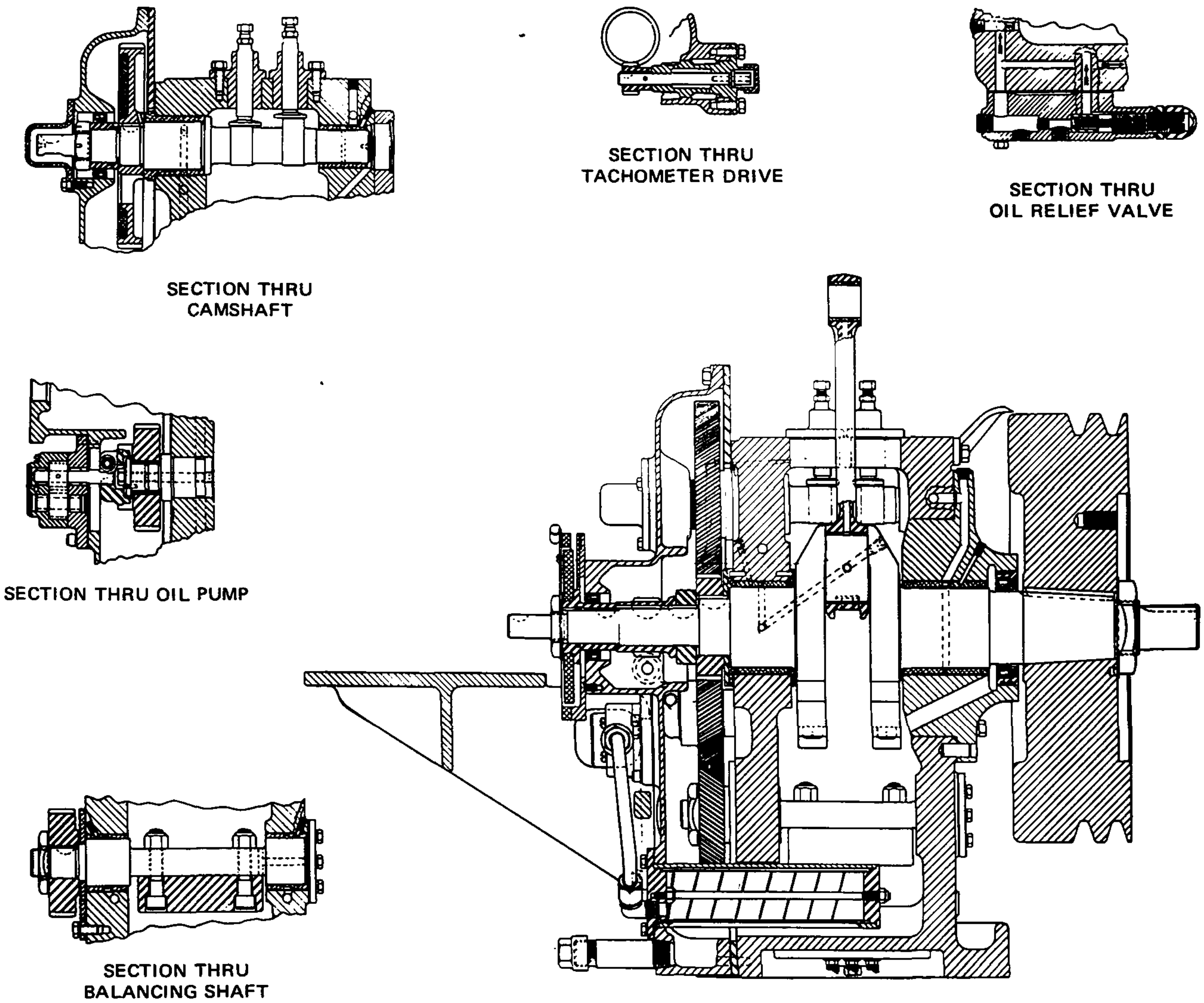


Figure 4.3-2. Waukesha CFR-48 Crankcase Assembly
(Source Reference: Waukesha Motor Company Bulletin 850M).

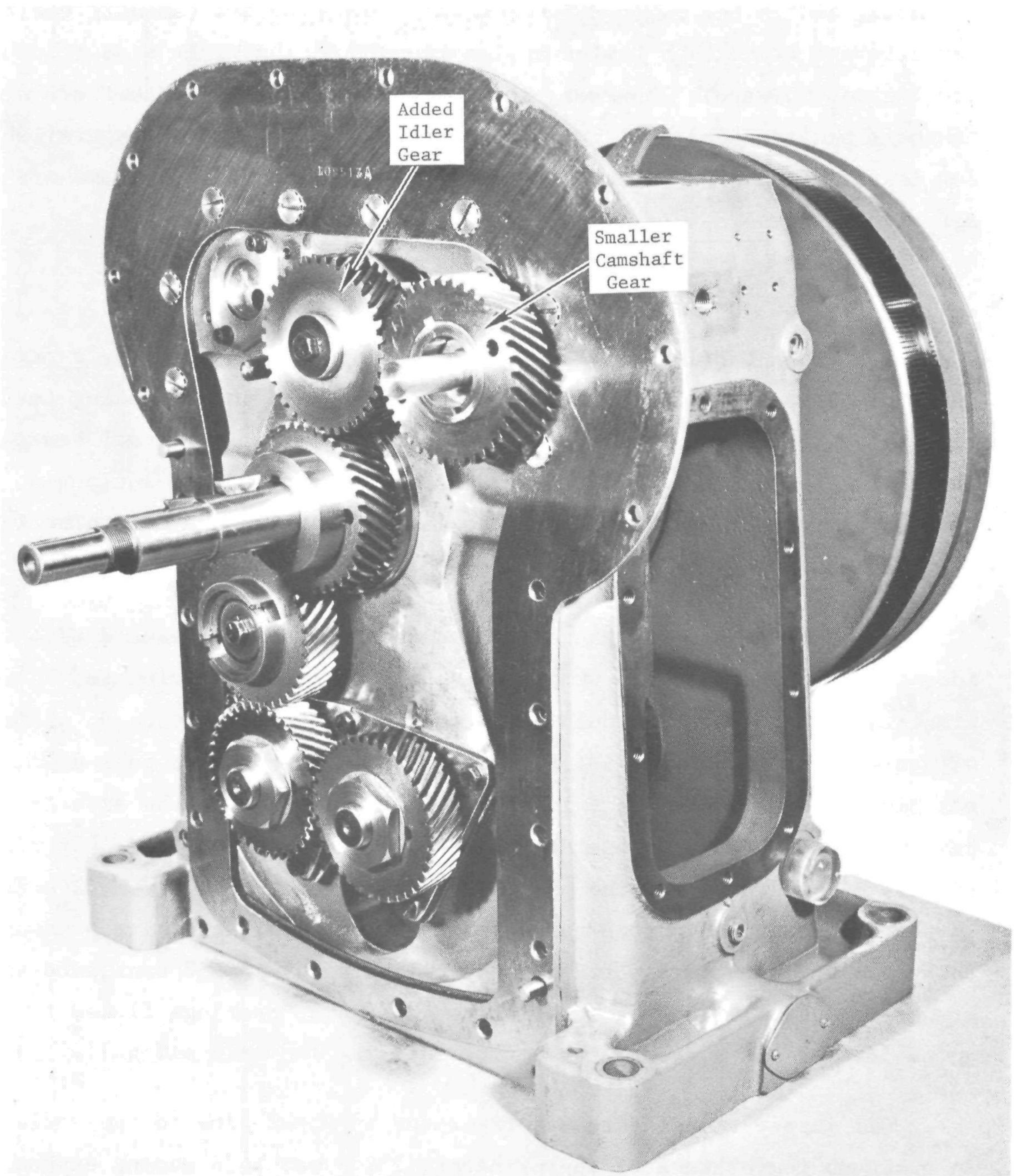


Figure 4.3-3. Waukesha Crankcase Showing Modified Gear Train (P72-2-5B).

A gear case contains the gearing for the camshaft, balance shafts, oil pump, and other external accessories. Counter rotating balance shafts are driven at engine speed through an idler gear. Two camshaft positions are available in the basic CFR crankcase. The standard camshaft position for internal combustion engine application is on the right side (flywheel end looking forward). However, it was more logical to use the left side camshaft position for the steam expanders since more space was available for installation of an idler gear between the crankshaft and camshaft. Refer to Figure 4.3-4 for modification details.

4.3.1.2 Expander Lube System

The basic CFR lubrication system consists of a gear pump, a strainer, and a pressure regulating valve. Oil galleys bored in the casing lead to all bearing bores providing full pressure lubrication to all bearings. The basic lube oil system of the crankcase was modified to include a cartridge type oil filter, a water cooled oil cooler, and a turbine type oil flowmeter.

The lube system for the crosshead and trunk piston expanders are identical. Oil enters the pump through a screen strainer located in the crankcase. The gear type pump is located on the accessory case, and is driven off the end of the idler gear at engine speed. The pump discharge was piped directly to a cartridge type filter mounted on the expander base plate. A pressure gage at the filter inlet was provided. From the filter, oil was piped to the oil relief valve which was bolted directly to the crankcase and ported to the main oil galley. Excessive pressure above the set point, resulted in some oil being by-passed back into the crankcase sump. Oil being supplied to the engine bearings flowed through an oil cooler and a flow meter before entering the main oil galley.

Oil supply temperature was regulated by water flow to the cooler. An oil sump temperature of approximately 250°F was held during expander operation. An electric "Calrod" type heater mounted under the crankcase was utilized to preheat the oil prior to startup. A 250°F temperature level vaporized water that entered the sump as a result of steam leakage. The crankcase breather pipe located on the side of the engine was connected to a small water cooled heat exchanger. Water vapor being expelled

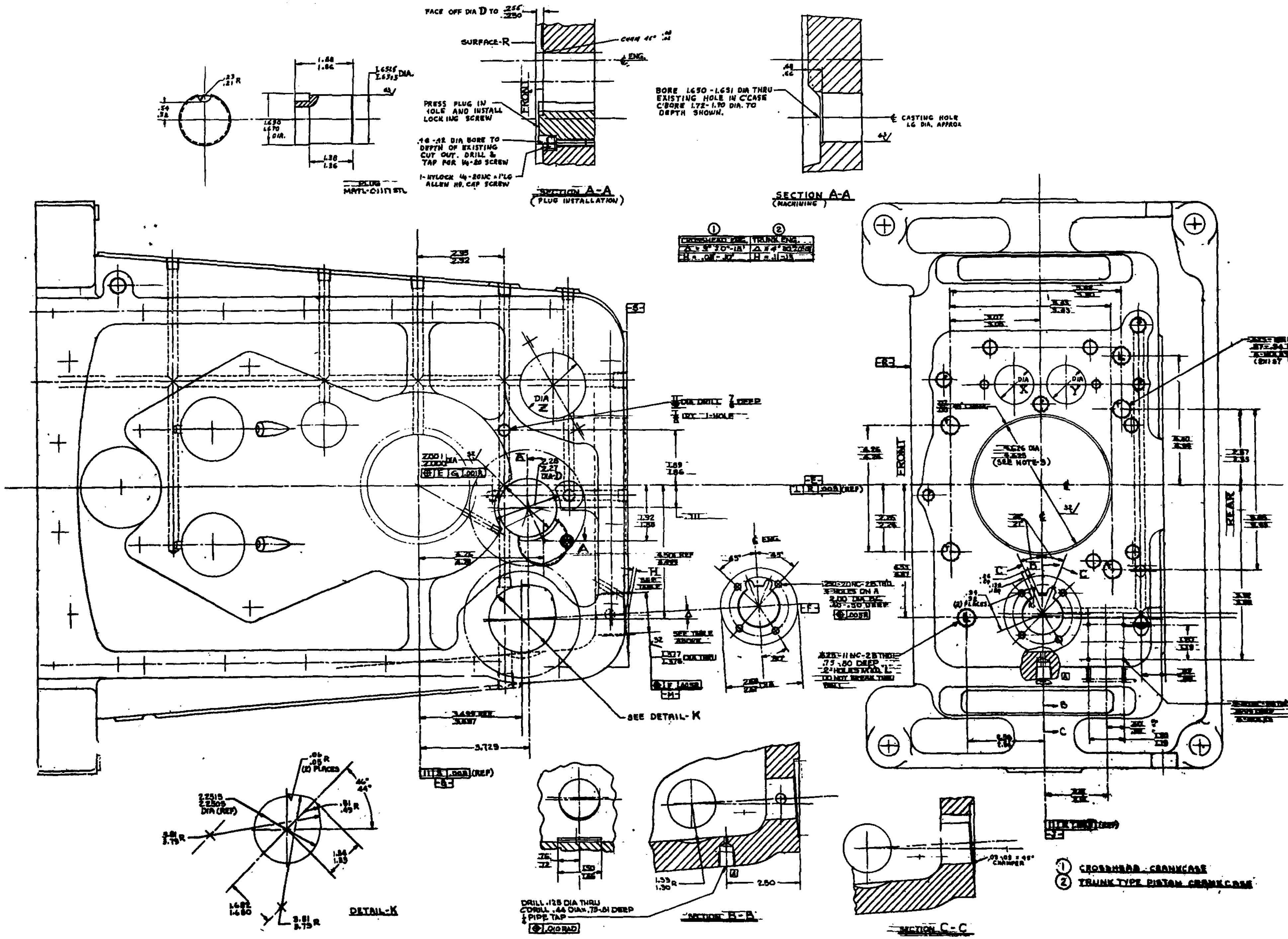


Figure 4.3-4. Steam Expander Crankcase, Modification Details (GE Dwg. 221R901).

from the crankcase passed thorugh the heat exchanger where it was condensed, collected, and measured. This provided an indication of seal leakage rate.

4.3.1.3 Expander Component Hardware

Figures 4.3-5 through 4.3-14 show the major hardware components used in the crosshead expander.

4.3.2 Trunk Piston Expander(4)

Figure 4.3-15 depicts the assembly of the trunk piston expander. Except for the power piston the trunk piston expander is quite similar to the crosshead expander design. For example, there is no difference in basic design of the valve gearing or recompression valve. Also, the connecting rod wrist pin and wrist pin bearing design is essentially the same. Two major differences between the two expanders are as follows: (1) the trunk piston expander is shorter in overall height, and (2) the power piston and connecting rod assemblies are quite dissimilar. However, the reciprocating weights for the two expanders are almost the same - approximately 8.2 pounds. The assembled expander is shown in Figure 4.3-16.

The trunk piston is a two piece design which permits the following:

1. The use of the same wrist pin design as used in the crosshead expander

2. The use of dissimilar materials for the top and bottom portions of the piston for weight reduction

3. Provides for positive oil/steam isolation in the region of the wrist pin while permitting adequate oil lubrication of the wrist pin without subjecting the oil to high temperature

4. Provides for oil cooling of the lower portion of the piston

5. Provides for the interchange of the upper portion of the piston to accommodate alternate sealing ring configuration and materials.

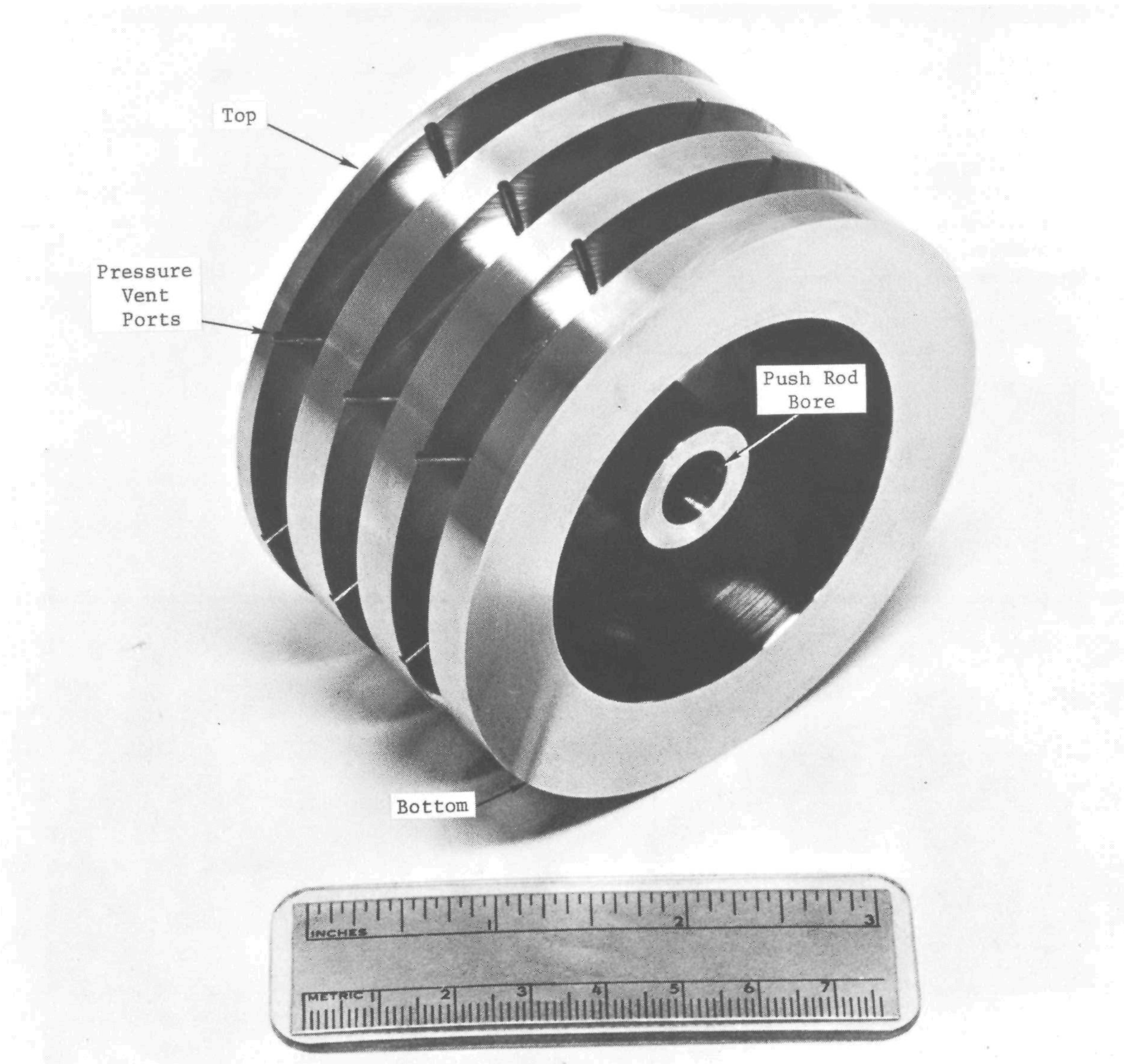


Figure 4.3-5. Power Piston for Carbon-Graphite Rings (P72-3-4U).

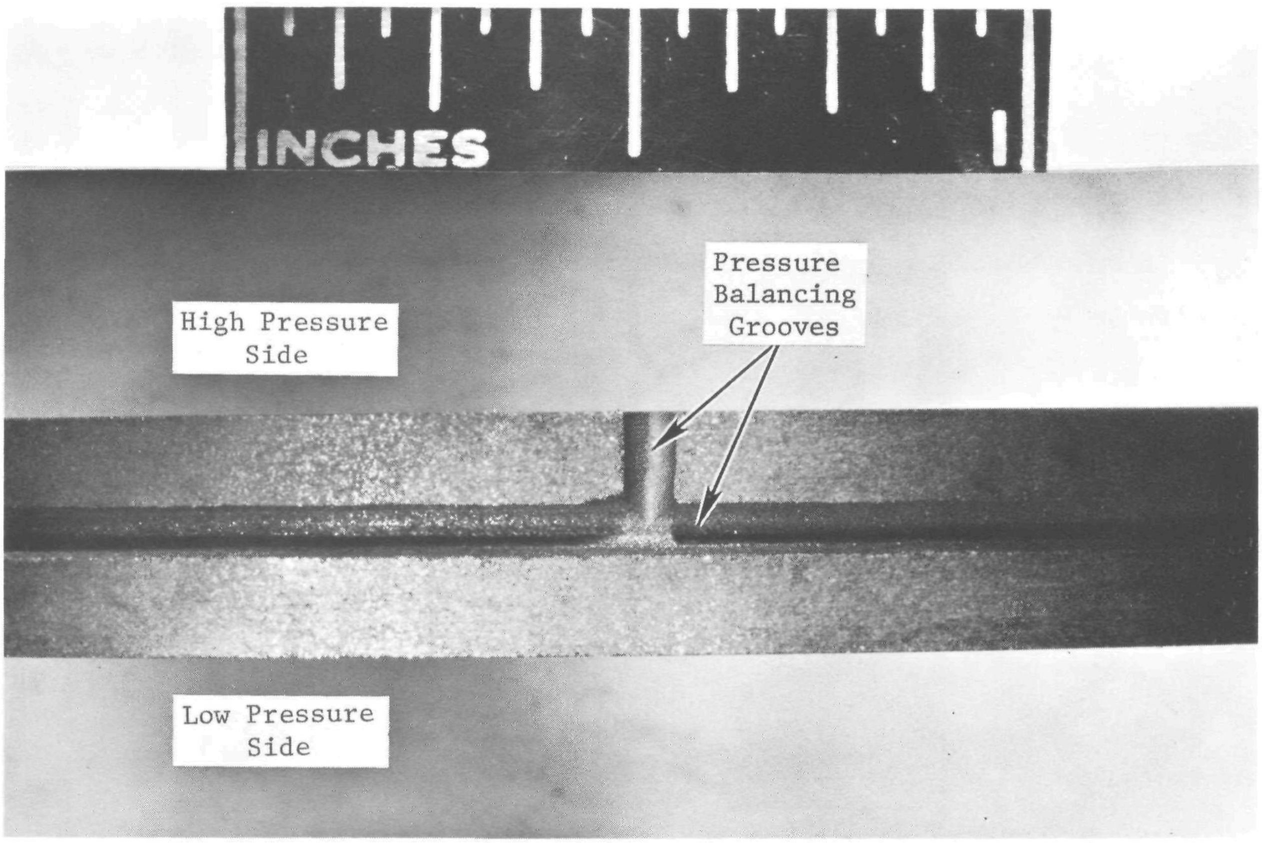


Figure 4.3-6. Pressure Balanced Carbon-Graphite Piston Ring (P72-3-2AD).

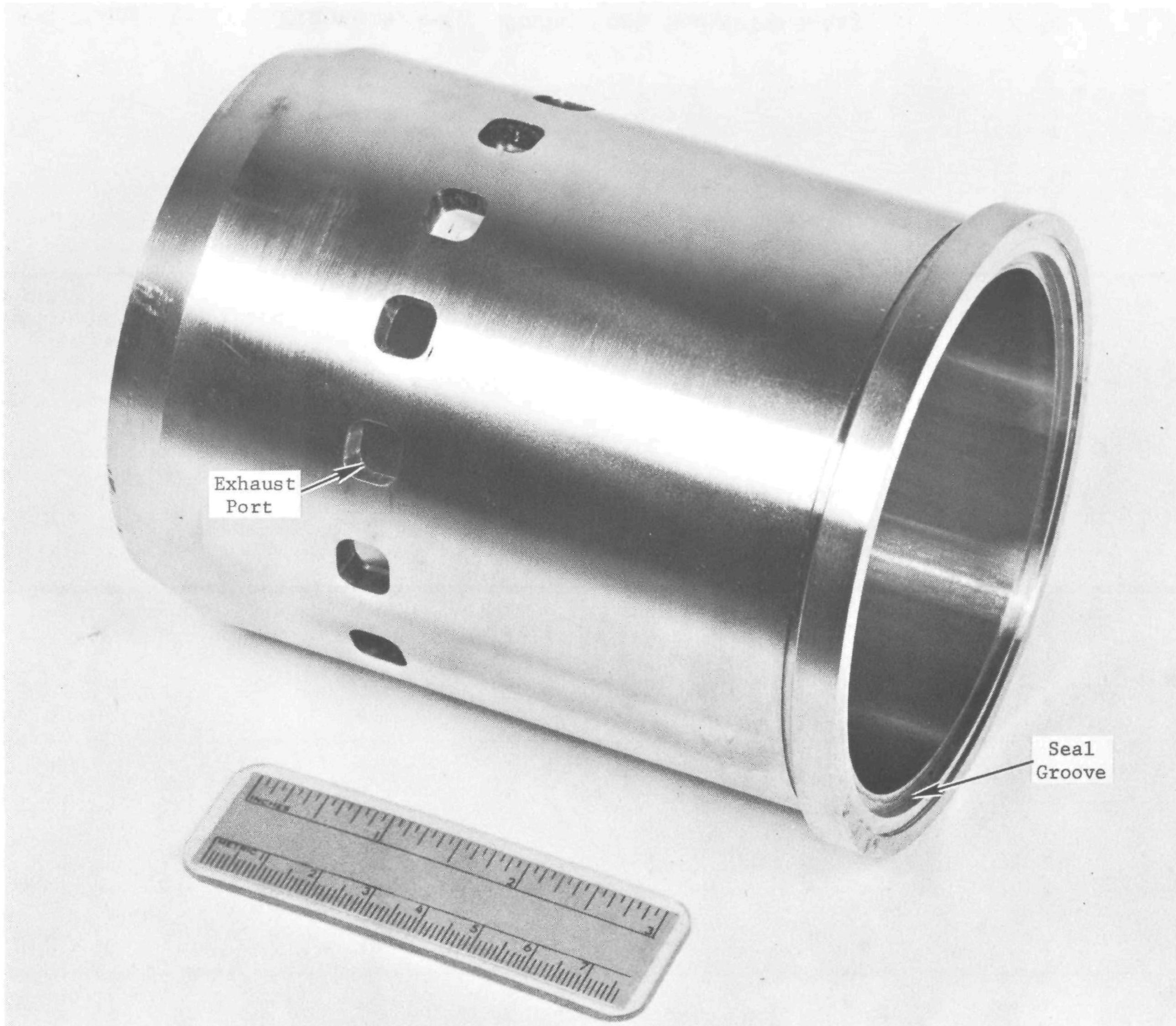


Figure 4.3-7. Cylinder Liner (P72-3-4C).

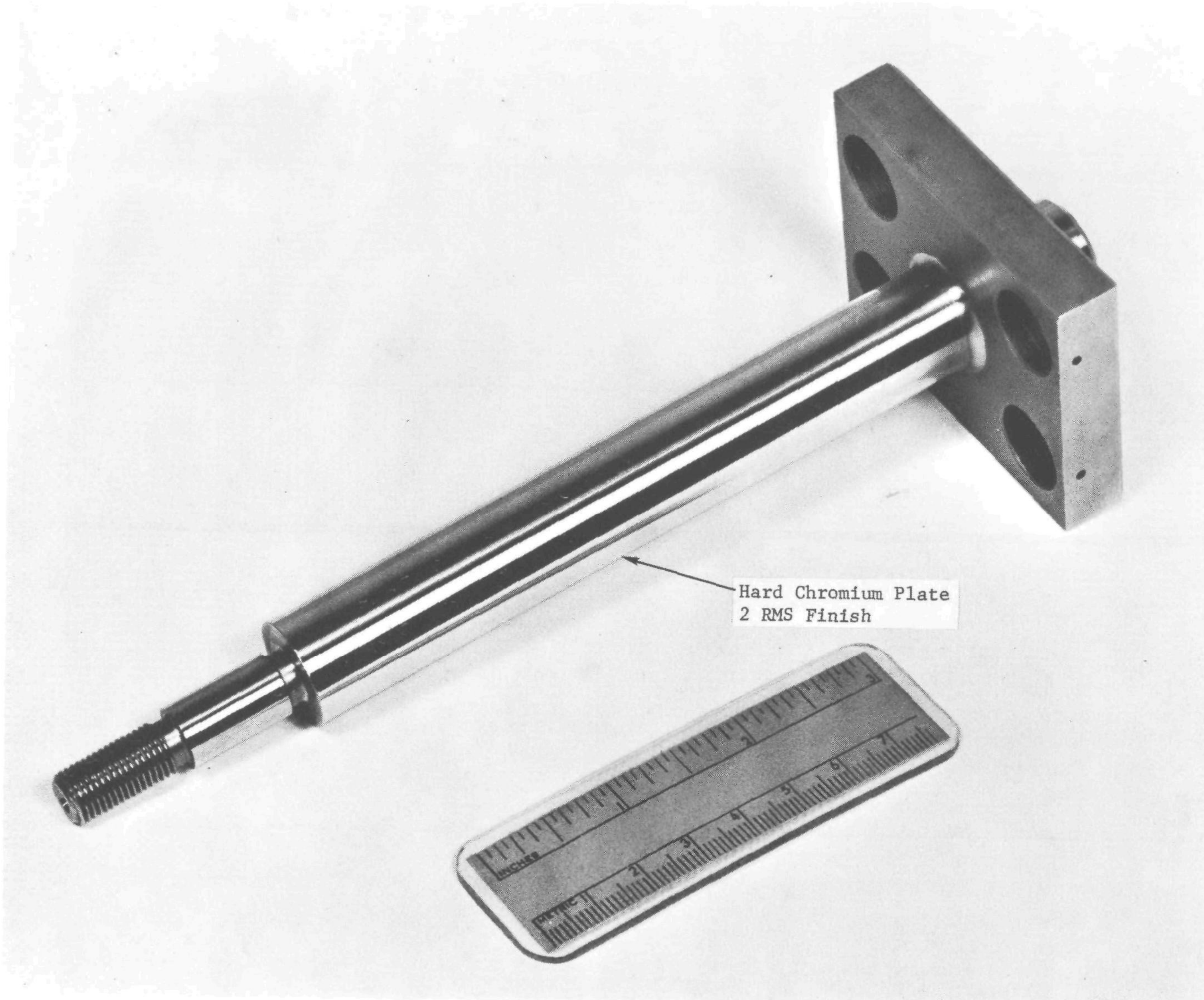


Figure 4.3-8. Power Push Rod (P72-3-4A).

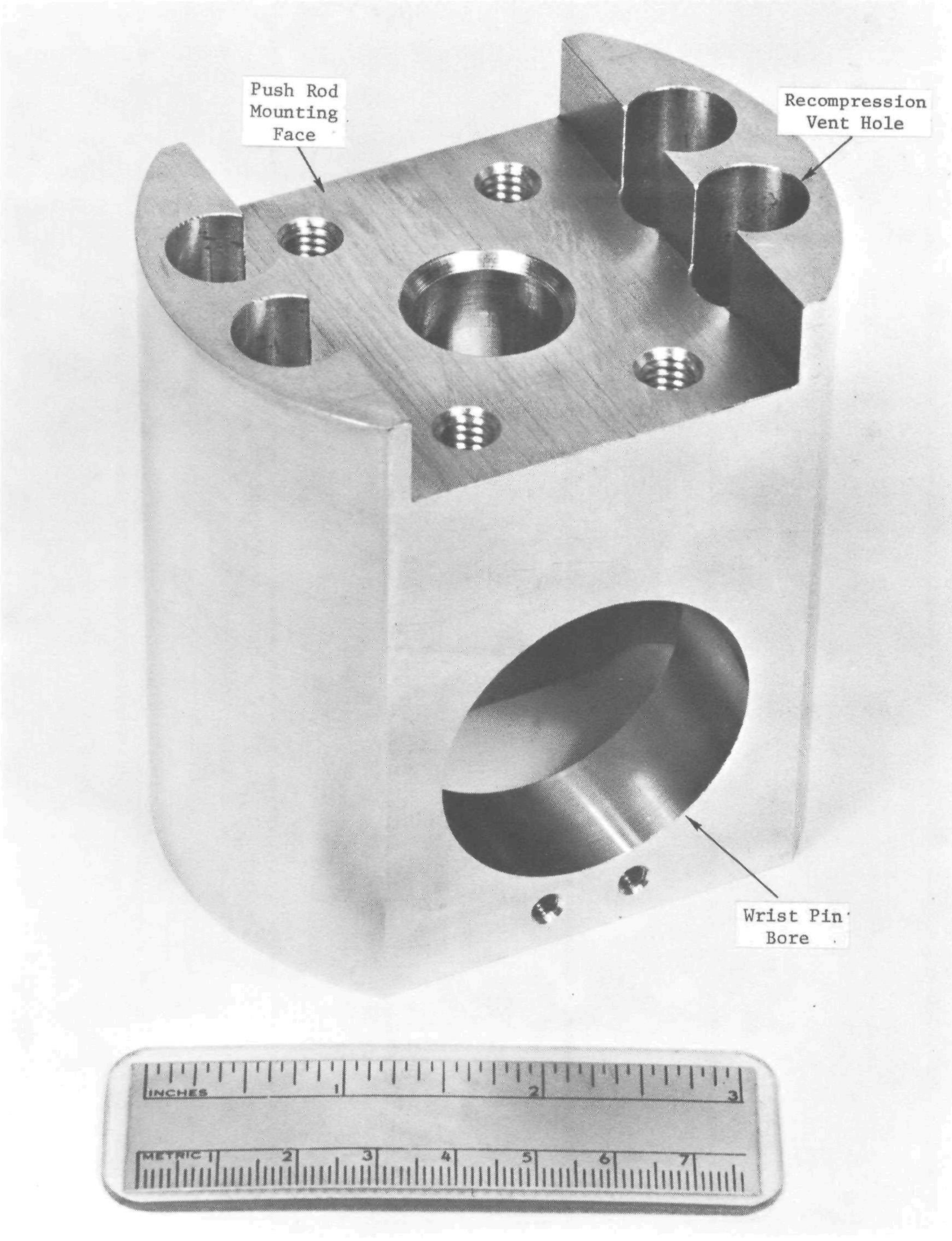


Figure 4.3-9. Aluminum Crosshead (P72-3-4K).

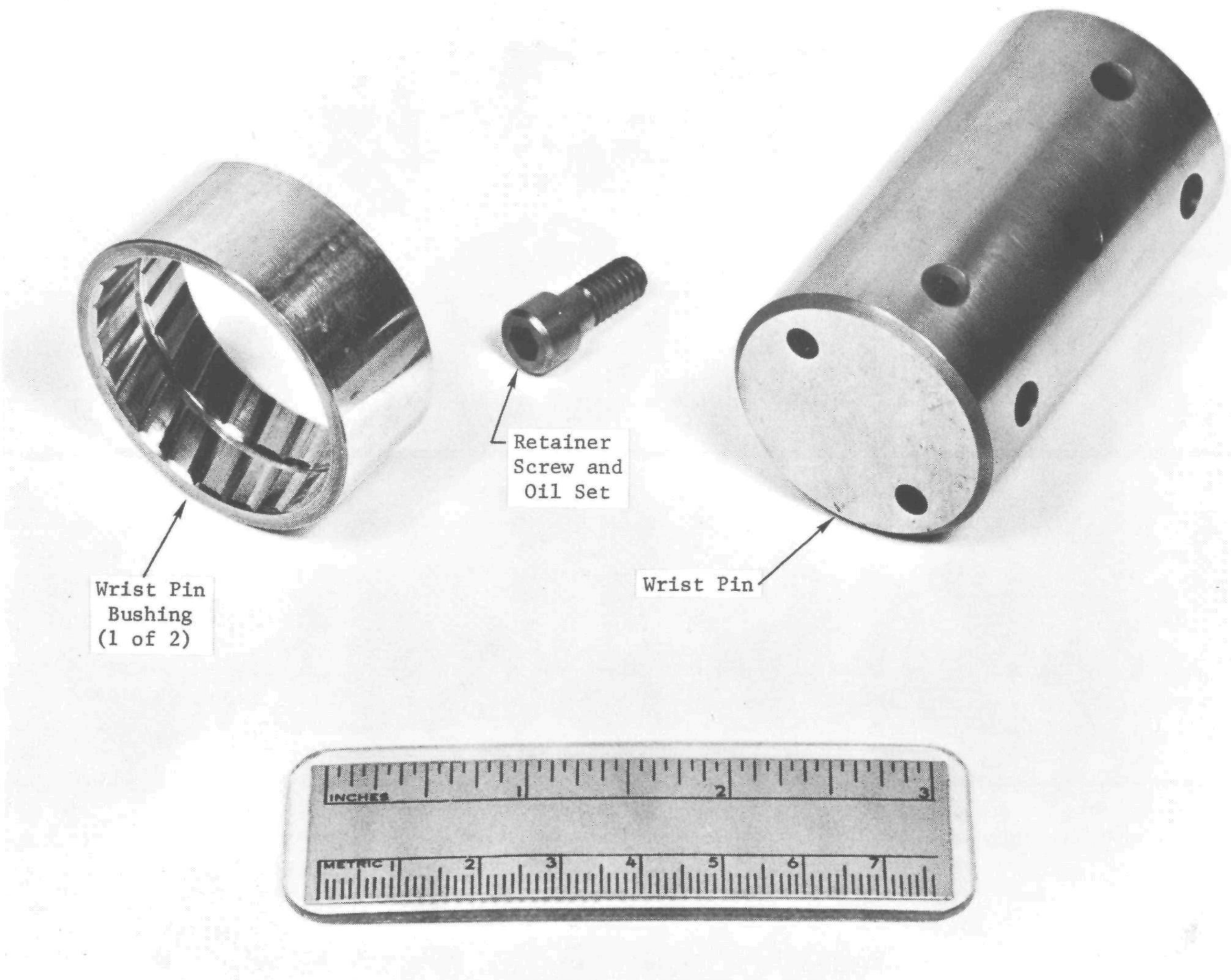


Figure 4.3-10. **Wrist Pin and Bushing (P72-3-4G).**

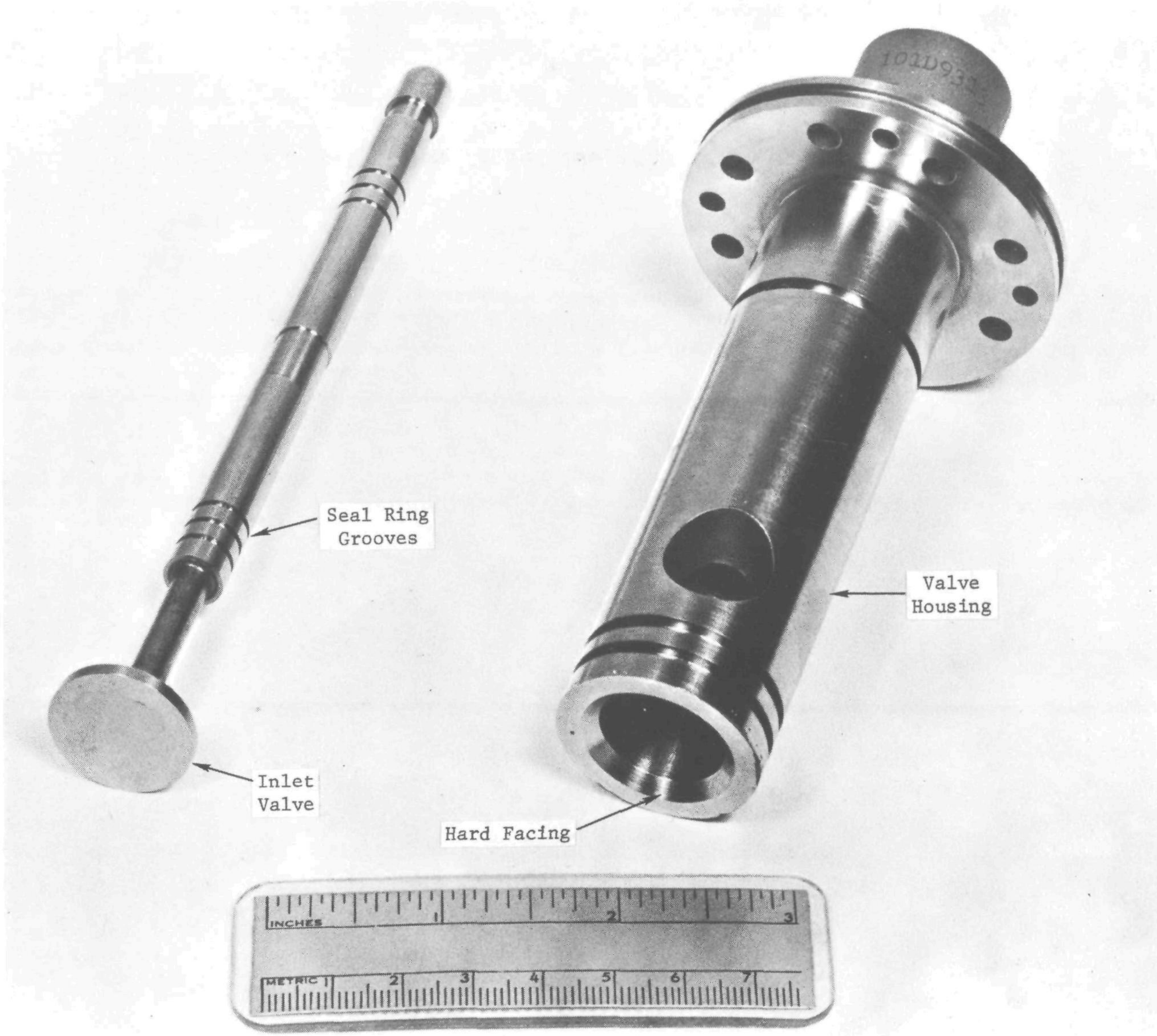


Figure 4.3-11. Steam Inlet Valve Assembly (P72-3-4F).

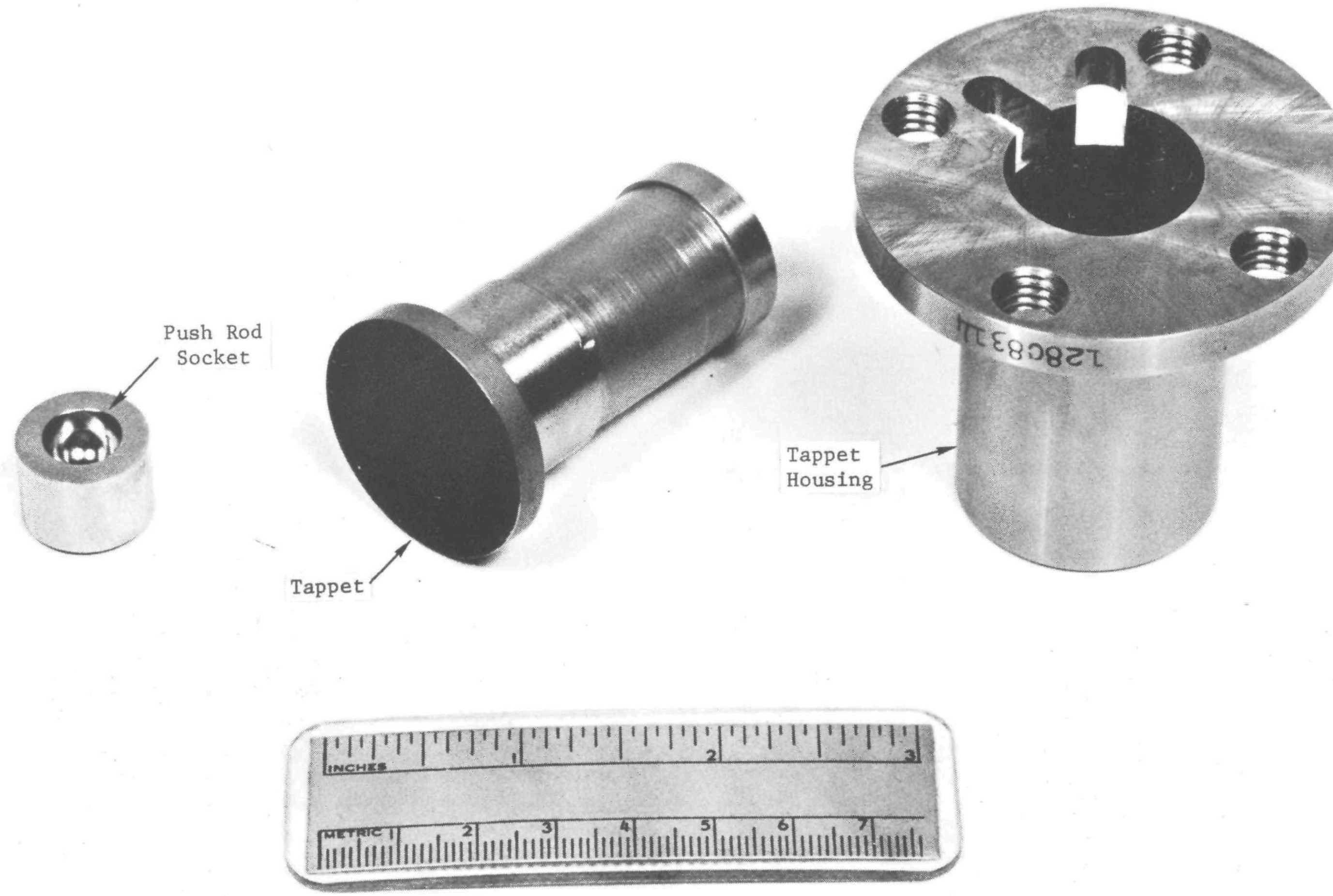


Figure 4.3-12. Cam Tappet Assembly (P72-3-4D).

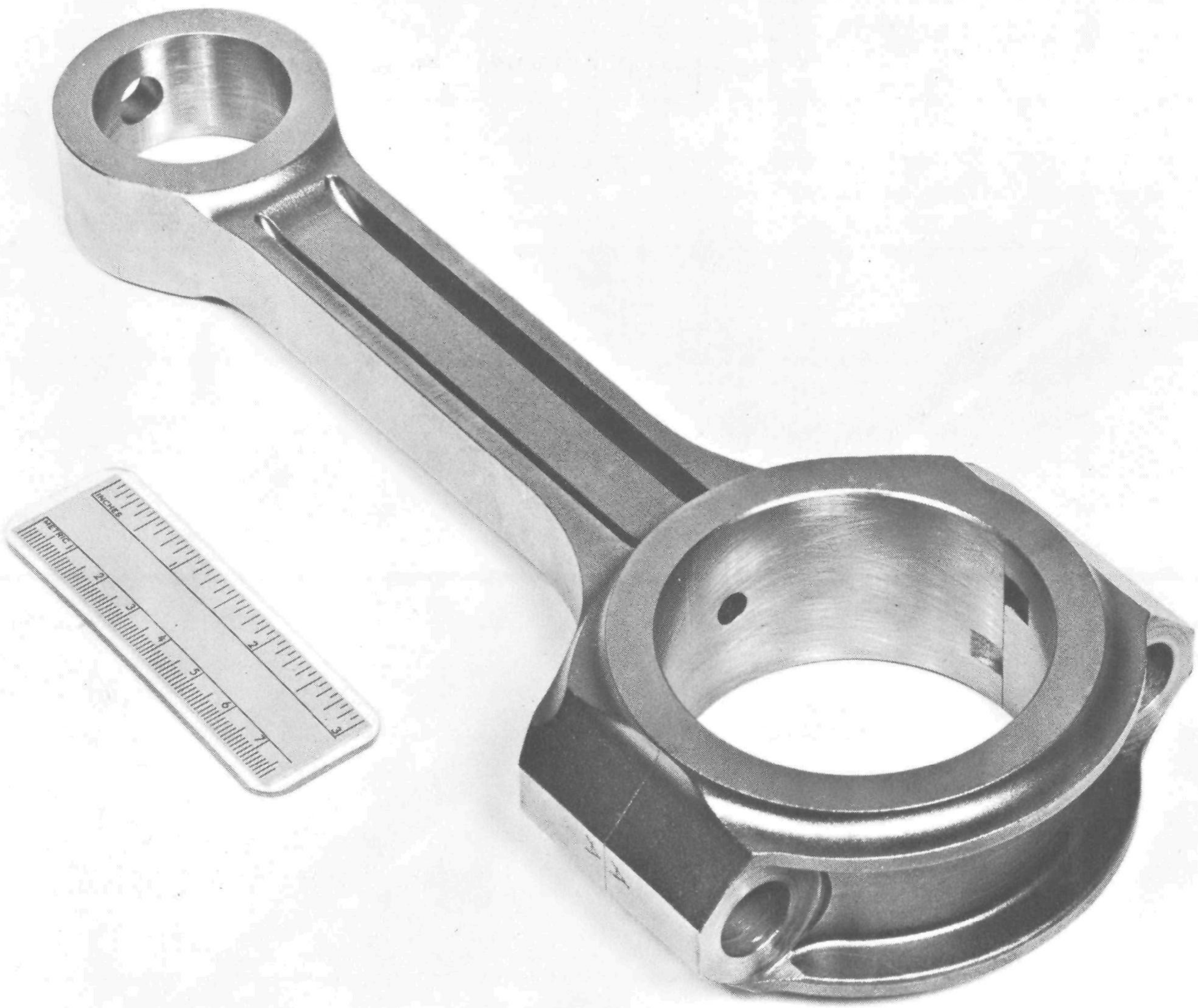


Figure 4.3-13. Connecting Rod (P72-3-4H).

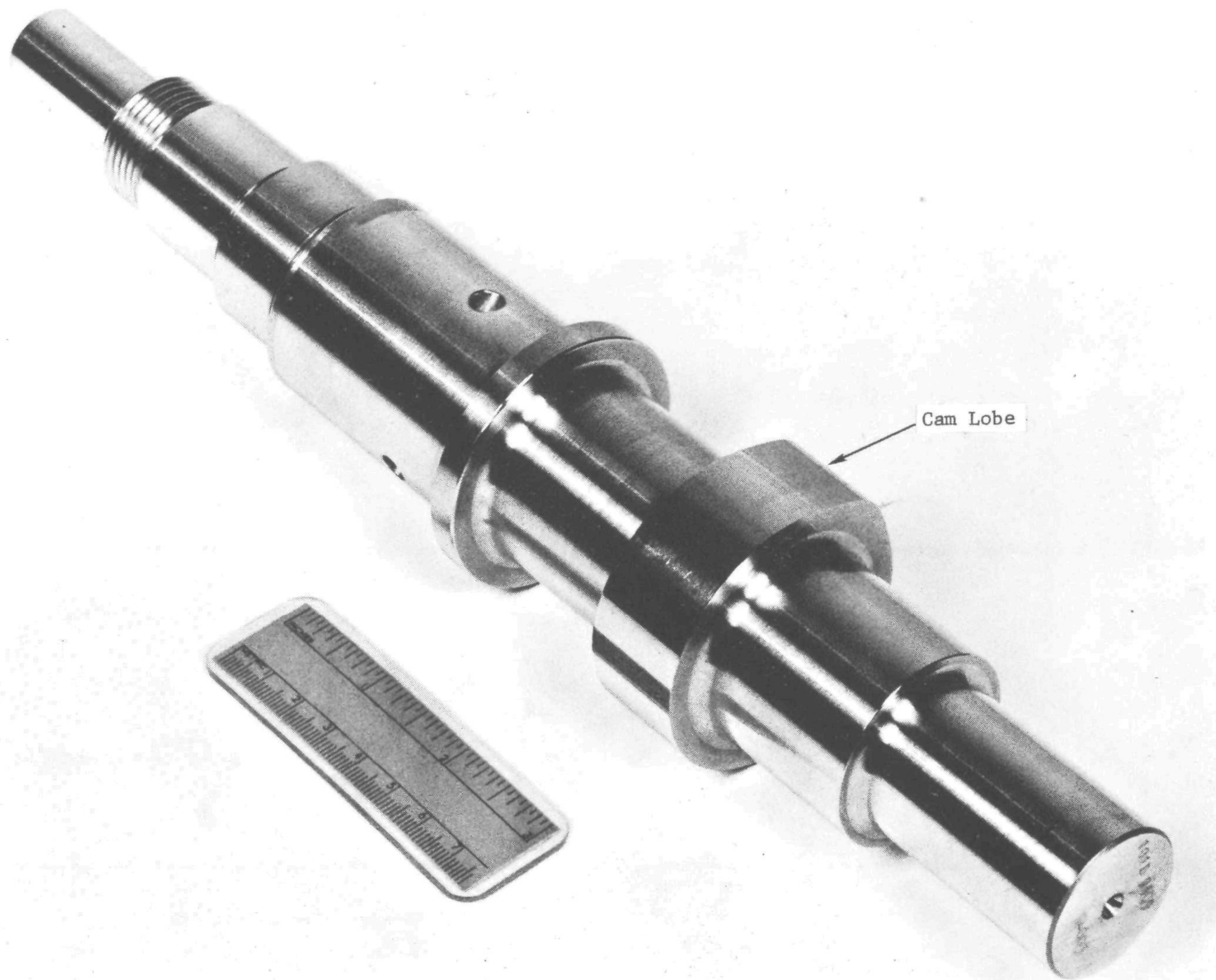


Figure 4.3-14. Camshaft (P72-3-4S).

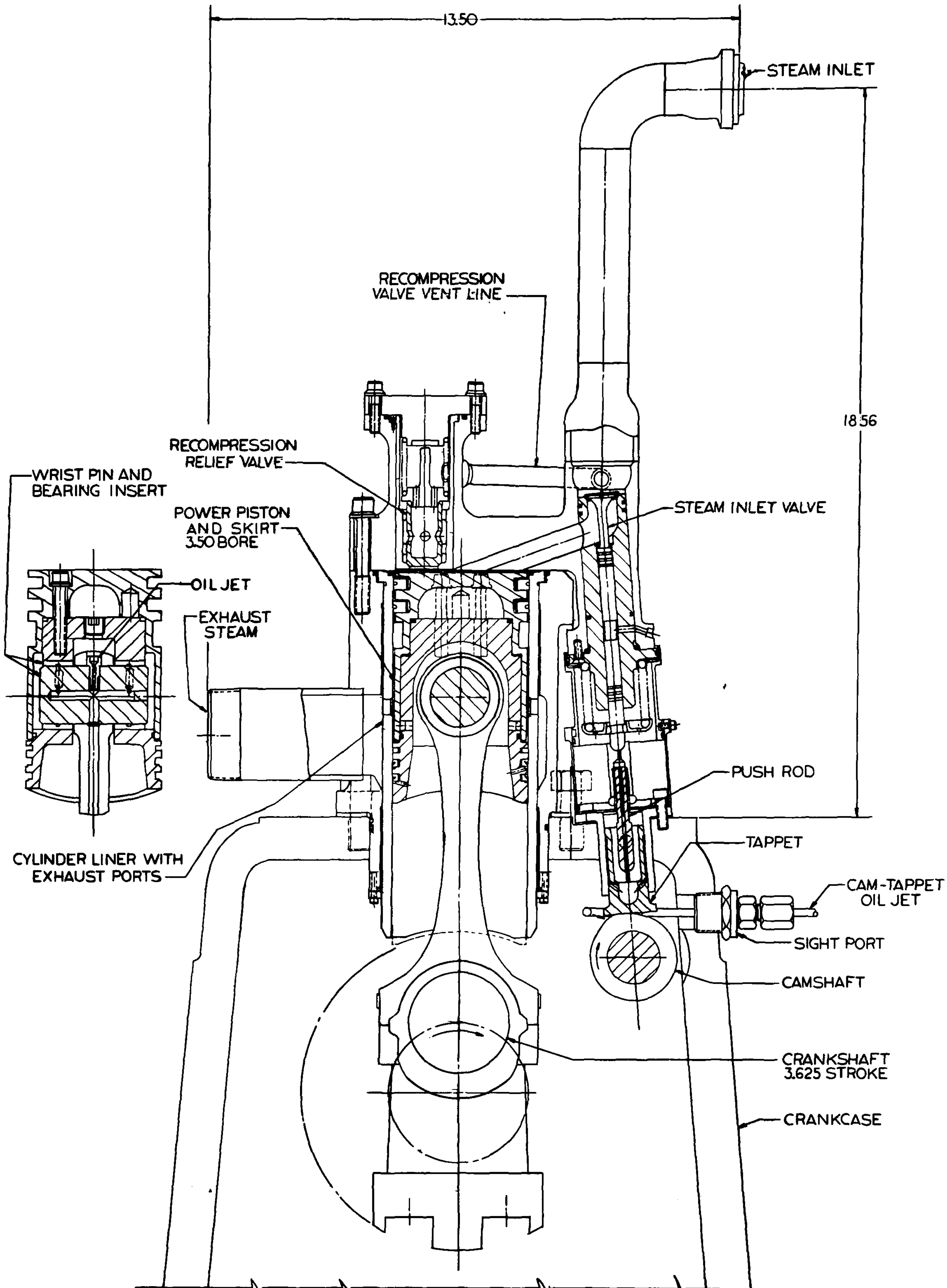


Figure 4.3-15. Assembly of Trunk Piston Expander (Dwg. 707E743).

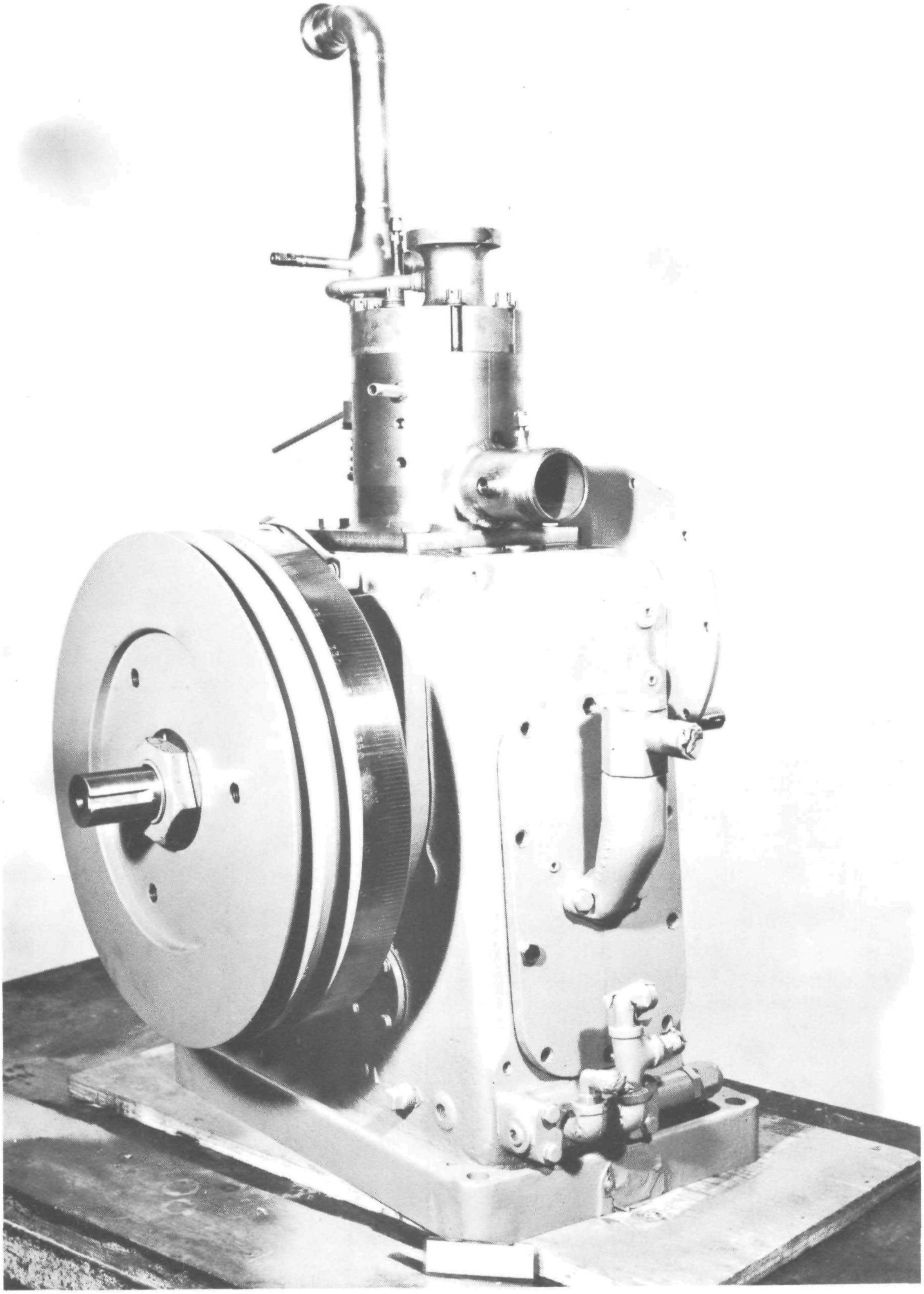


Figure 4.3-16. Assembled Trunk Piston Expander (P72-6-3S)

To keep the trunk piston as short as possible, the basic expander design shown in Figure 4.3-15 shows two non-metallic sealing rings compared with the crosshead expander which contains three non-metallic sealing rings. However, the alternate trunk piston design contains three metallic sealing rings for the same piston length. A wider third ring, called a rider ring, is required in the top portion of the trunk piston to carry piston side loads and is not intended to function as a seal ring.

The trunk piston contains three rings near the bottom of the piston. The top ring of the three is a seal ring to restrict the flow of steam into the expander crankcase. The two bottom rings are oil exclusion rings to prevent oil from entering the steam system. The lower circumferential surface of the trunk piston fuctions as a piston side load support or second rider ring. Since the lower part of the trunk piston is oil lubricated a separate solid lubricant rider ring is not required below the wrist pin.

The trunk piston expander also has a removable cylinder liner so that different ring/liner material combinations can be evaluated with minimum expander rework.

The crankcase assembly including crankshaft, gear train, counter rotating balance shafts, oil pump, etc. are identical to the crosshead expander. Due to the shorter overall cylinder height, the valve axis is inclined at a slightly different angle than the crosshead. The lubrication system for the trunk piston expander is identical to the crosshead expander.

Figures 4.3-17 through 4.3-22 show the major trunk piston expander components which are different from the crosshead expander.

4.4 Expander Performance Analysis

4.4.1 Expander Breathing Analysis

A computer program was written and used to perform expander breathing calculations. The purpose of this analysis was the following:

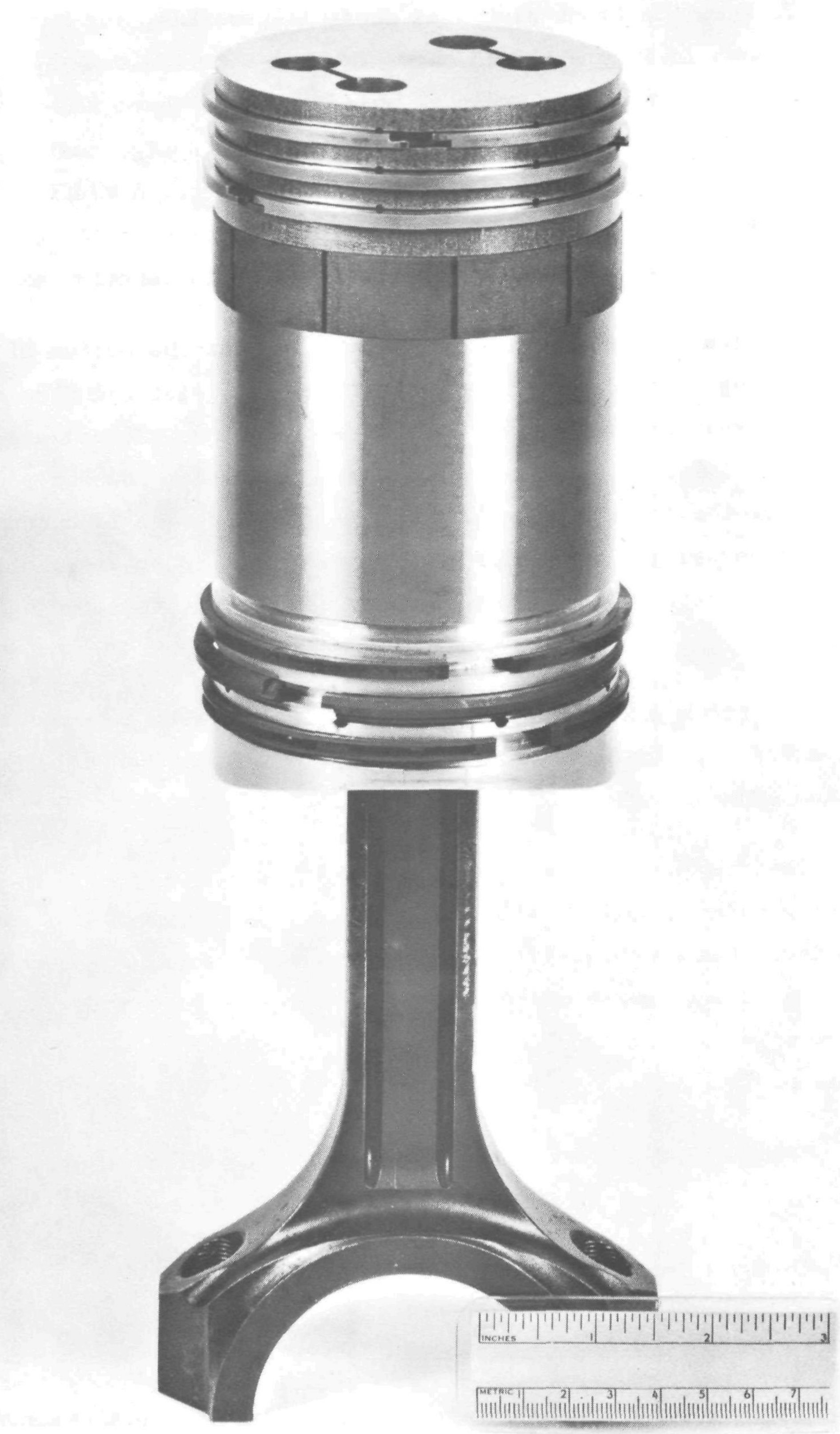


Figure 4.3-17. Trunk Piston and Rod Assembly, Cr_3C_2 Rings (P72-6-3B)

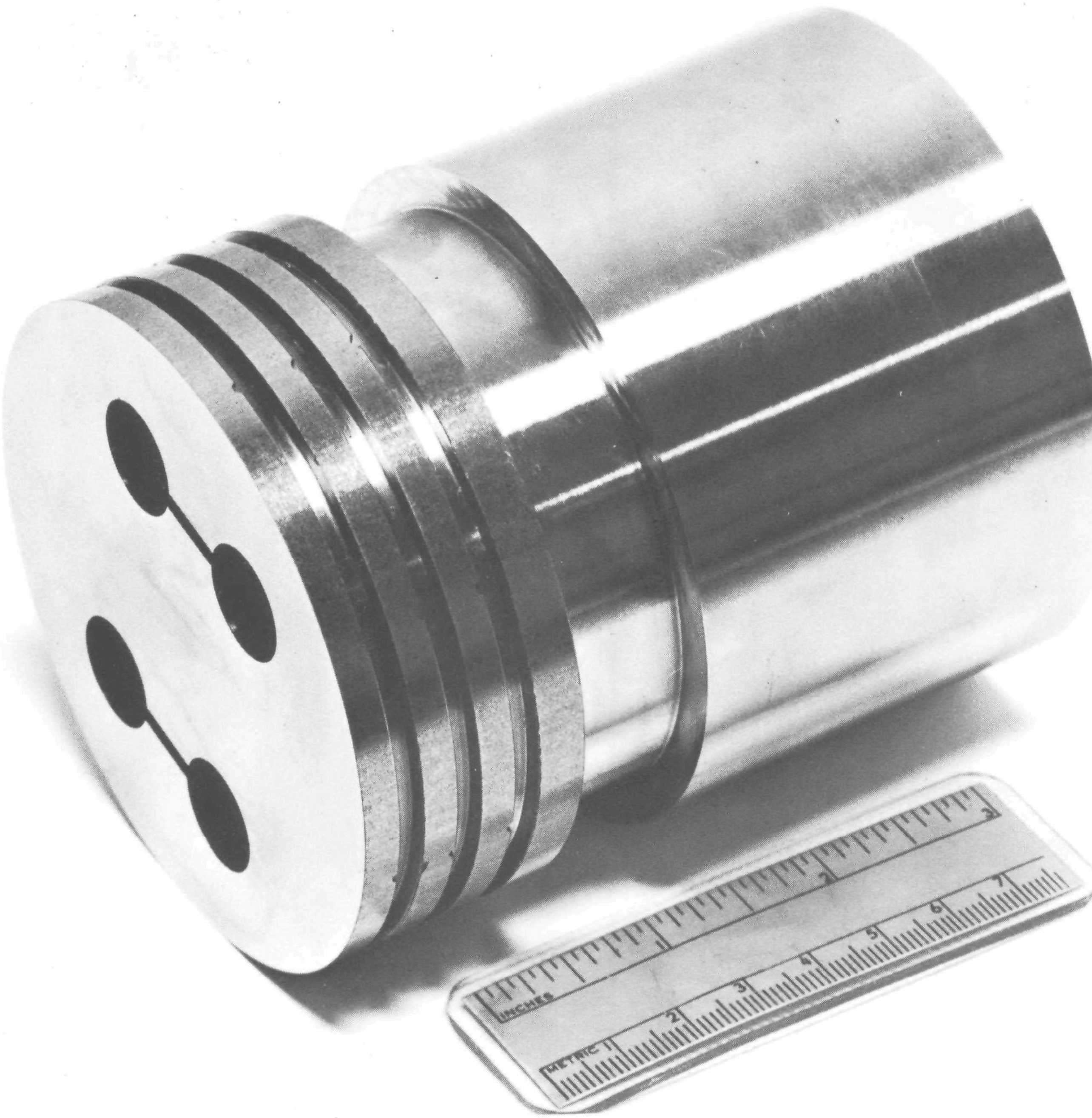


Figure 4.3-18. Piston For Trunk Expander (Upper Portion)
(P72-6-3C)

Figure 4.3-19. Piston for Trunk Expander (Lower Portion)
(P72-5-4H)

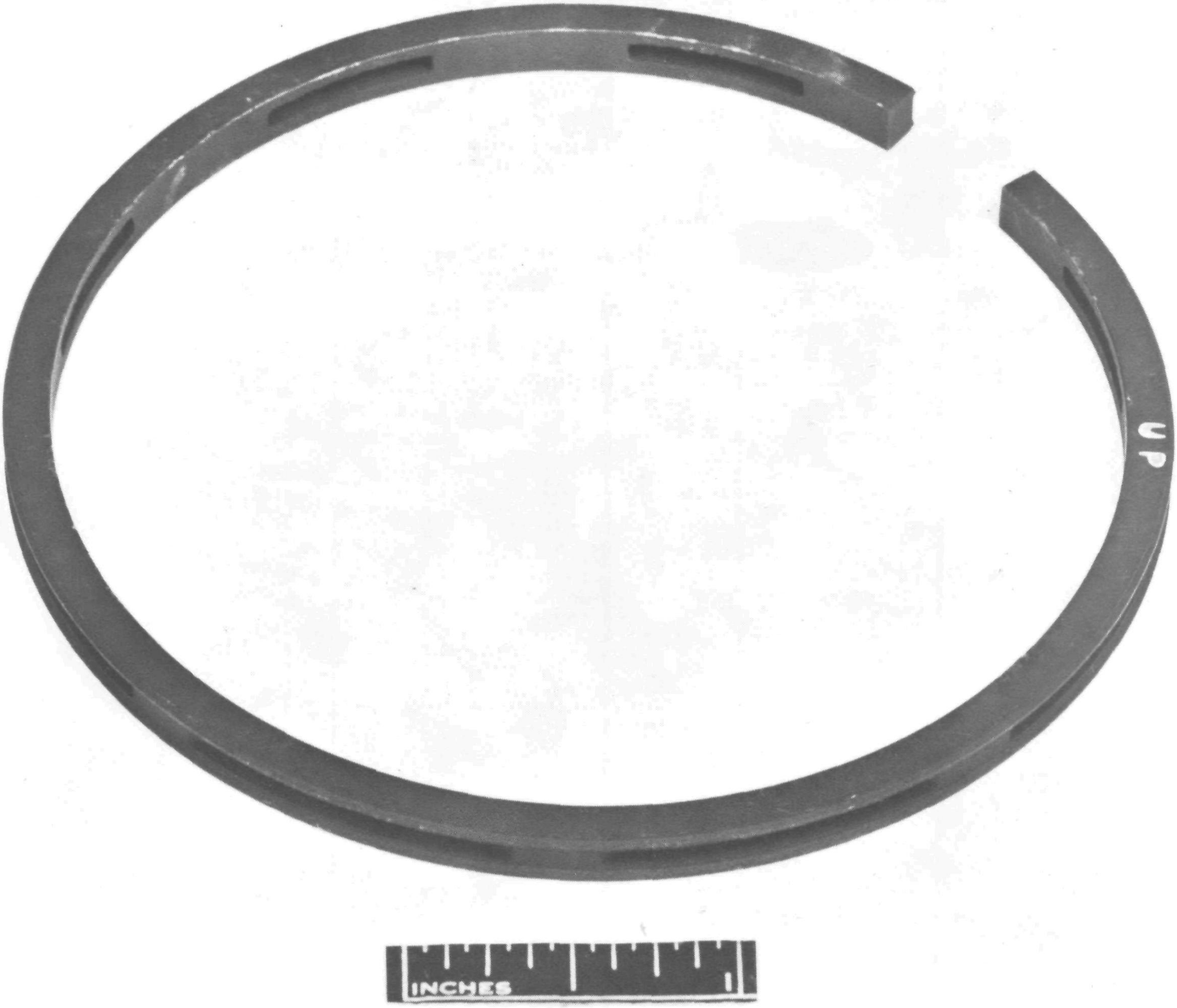


Figure 4.3-20. Trunk Piston Oil Exclusion Ring (P72-5-2B)

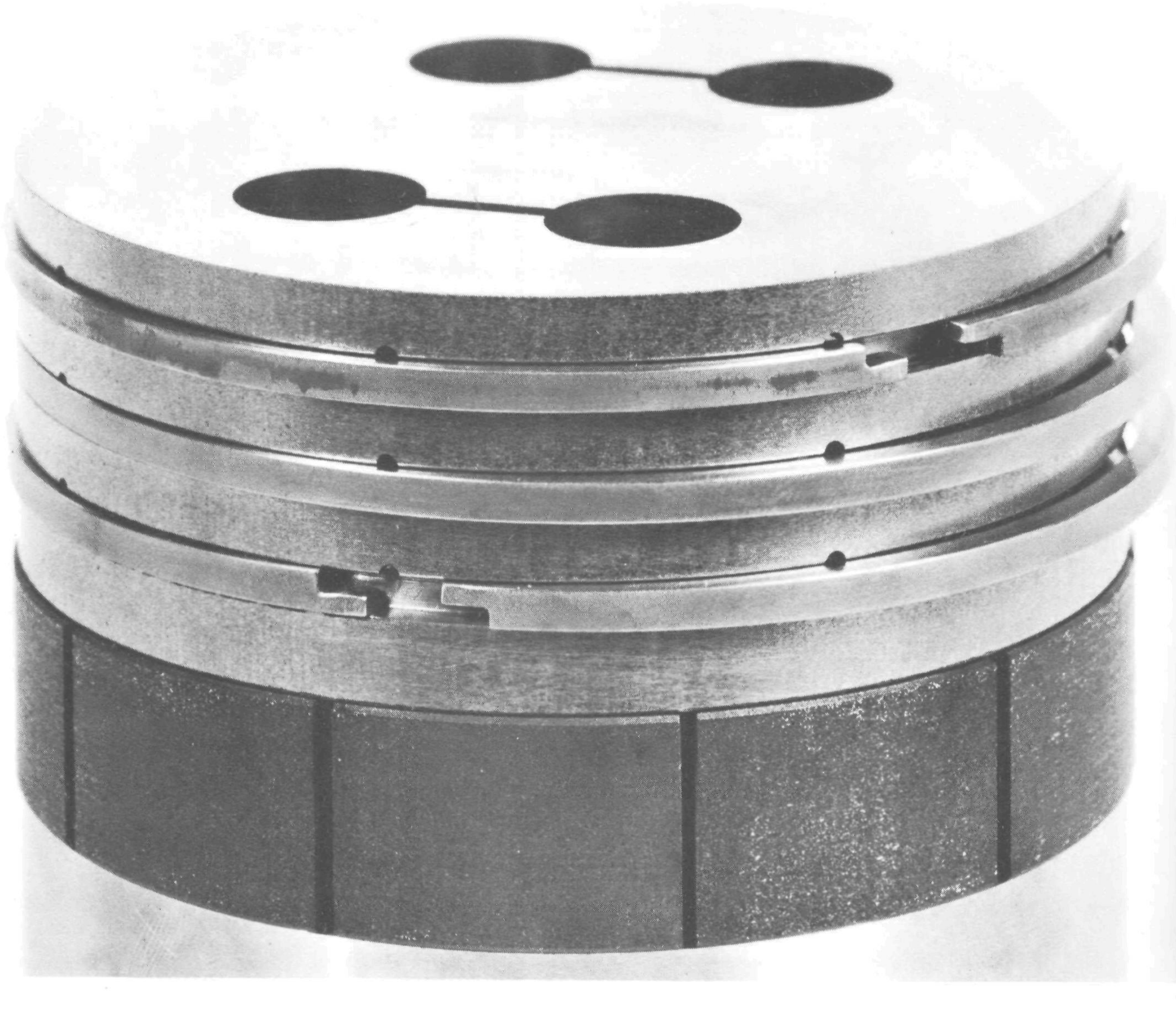


Figure 4.3-21. Trunk Expander Piston Assembly (P72-6-3A)

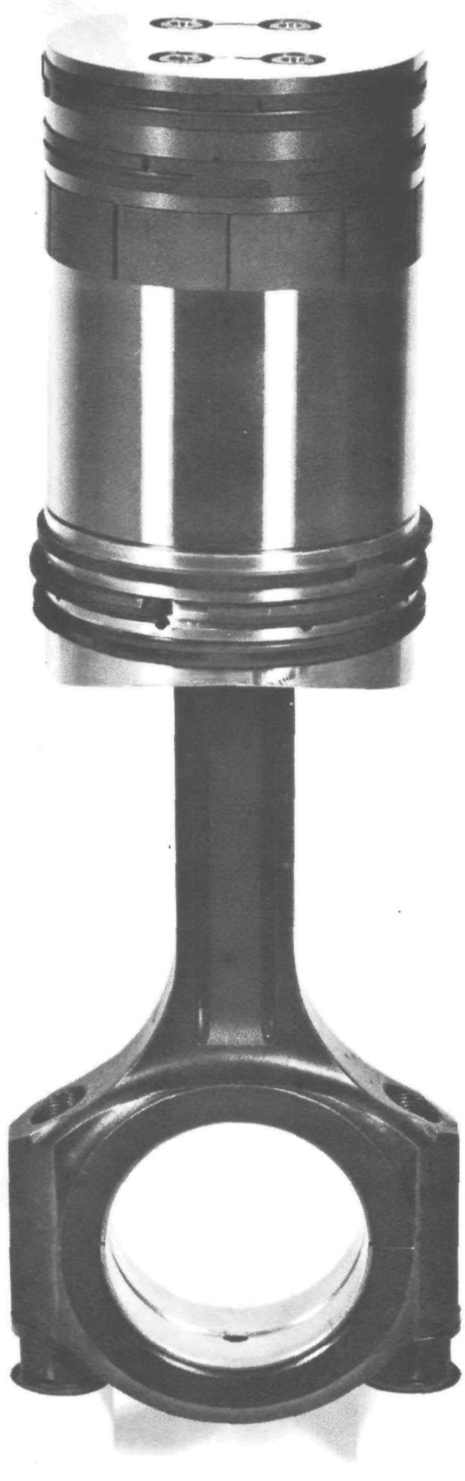


Figure 4.3-22. Trunk Piston and Rod Assembly (Graphite Rings)
(P72-6-3F)

1. To determine the inlet valve diameter, maximum lift, total event angle, cam lead angle, cam base circle radius, and the cam nose radius in such a manner that the steam flow rate is approximately 400 pounds per hour for an indicated horsepower and indicated efficiency near an optimum level.

2. To insure that the cam and cam follower Hertz stresses, pressure times velocity parameter, and the valve train accelerations are within limits.

3. To obtain predicted engine indicator diagrams for specified test conditions over a range of operating speeds.

The computer program calculates the steam flow into the cylinder which takes place during discrete time intervals of the valve operating cycle, and also calculates the co-occurring changes in cylinder pressure which result from the combined effects of steam inflow and cylinder volume change. The valve and cam geometry are inputs to the program. For portions of the expander cycle during which the valve is closed an expansion or compression process following the equation PV^{γ} = constant was calculated. Flow through the cylinder exhaust ports was calculated in a manner similar to that of steam inflow. The computer program was based upon the assumptions of perfect gas properties for the superheated steam, isentropic flow from the engine inlet plenum to the minimum valve orifice cross section, and upon no recovery of valve orifice velocity head within the cylinder (valve orifice static pressure equals cylinder pressure). An empirical orifice flow coefficient obtained from data in reference 5 was applied. The effective ratio of specific heat, γ, for superheated steam was treated as a variable function of pressure and temperature through the cycle. However, this was found to be an unnecessary refinement, at least insofar as steam flow prediction was concerned. Calculated flow variations corresponding to a change in γ from 1.2 to 1.3 were found to be approximately 5%. This assumed variation in γ exceeds the physical variation which occurs as the steam goes through the expander cycle, and the resulting calculated flow variation was substantially less than the expected flow variation resulting from the effects of piston blowby and cylinder wall heat transfer. The latter effects were neglected in order to finalize the valve and cam design within the short time available, and also because it was felt that an accurate prediction of these effects, particularly that of piston blowby, could not be made prior to testing. An estimated flow margin allowance for these factors was made in the final design selection.

Engine indicator diagrams calculated by the breathing program are shown in Figures 4.4-1, 4.4-2 and 4.4-3. These diagrams all correspond to the selected valve and cam geometry. For operation at 2500 RPM the cam shaft would be shifted so that the inlet valve would close 10° earlier than for operation at 2000 RPM. This shift changes the valve lead angle from 10° to 20° and limits the steam flow to an estimated value which is within the test facility capacity at the higher RPM. Figure 4.4-1 shows the calculated indicator diagram for operation at 2000 RPM, 1000°F inlet temperature, 1000 psia inlet pressure. Valve lead is 10°. Figure 4.4-2 shows the calculated indicator diagram for operation at 2500 RPM with the same inlet conditions as for Figure 4.4-1. The valve lead is 20°. Figure 4.4-3 shows the effect of expander speed on the indicator diagram. As speed is reduced the cylinder pressure drop occurring during the steam admission period becomes less, and the steam flow per revolution increases. This occurs from the fact that the admission time period for the steam flowing through the valve restriction increases with reduction in speed. The steam flow per revolution is reduced by increasing valve lead since the average cylinder pressure during the admission period increases with increasing valve lead (reduced average ΔP across the valve orifice).

Table 4.4-I gives a summary of the analysis of the indicator diagrams shown in Figures 4.4-1 and 4.4-2. This table also indicates the selected geometry of the valve and cam. Table 4.4-II gives a summary of the analysis of the indicator diagrams of Figure 4.4-3.

4.4.2 Steady-State Temperature Distribution

A broad heat transfer analysis was conducted to determine an approximate temperature map for the two single-cylinder steam expanders employing the existing THT-D Transient Heat Transfer computer program. This program is capable of analyzing general three-dimensional heat transfer systems employing a finite difference method with appropriate specified boundary conditions. A variety of modes of heat exchange may be

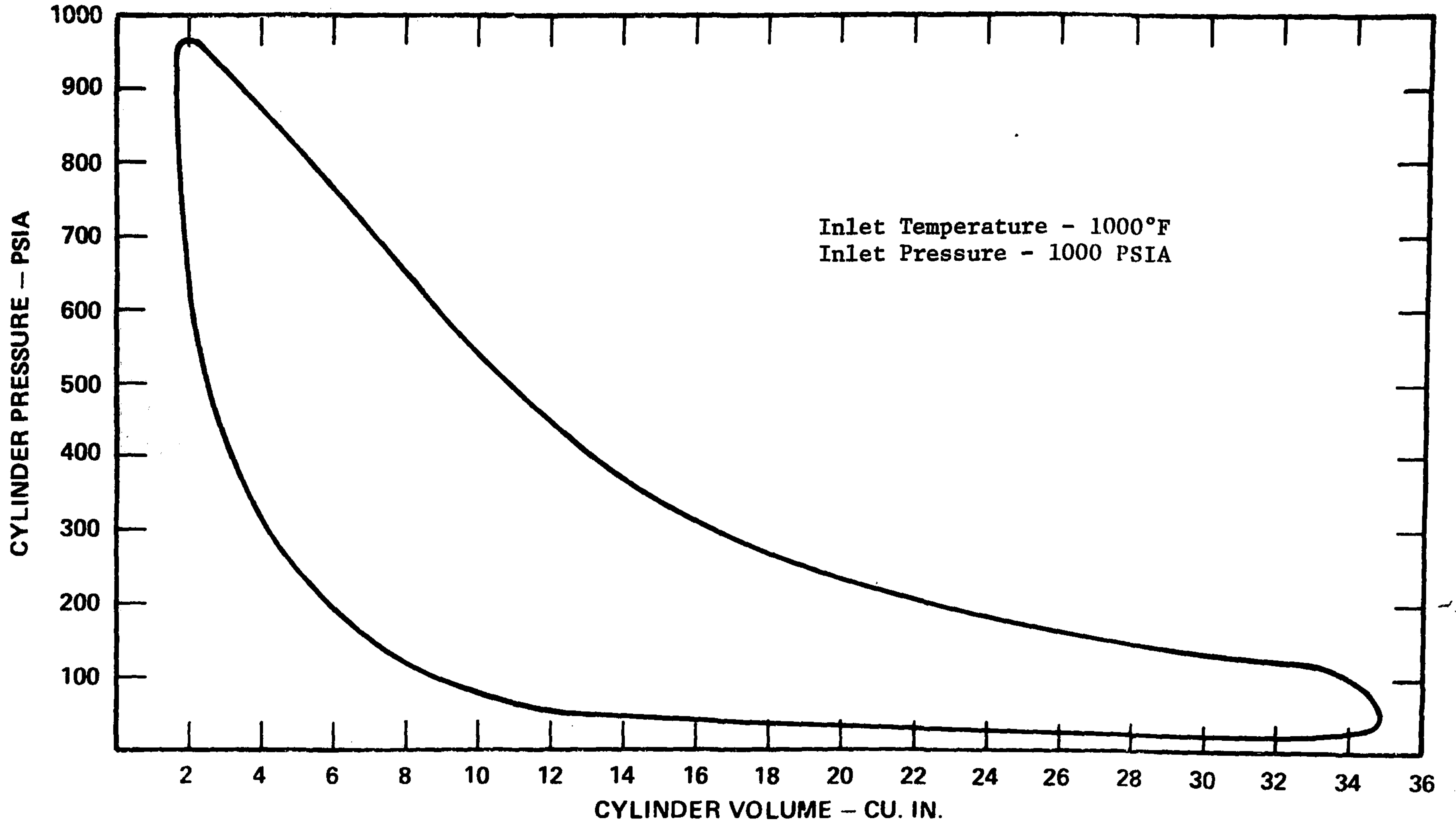


Figure 4.4-1. Expander Indicator Diagram, 2000 RPM, 10° Valve Lead.

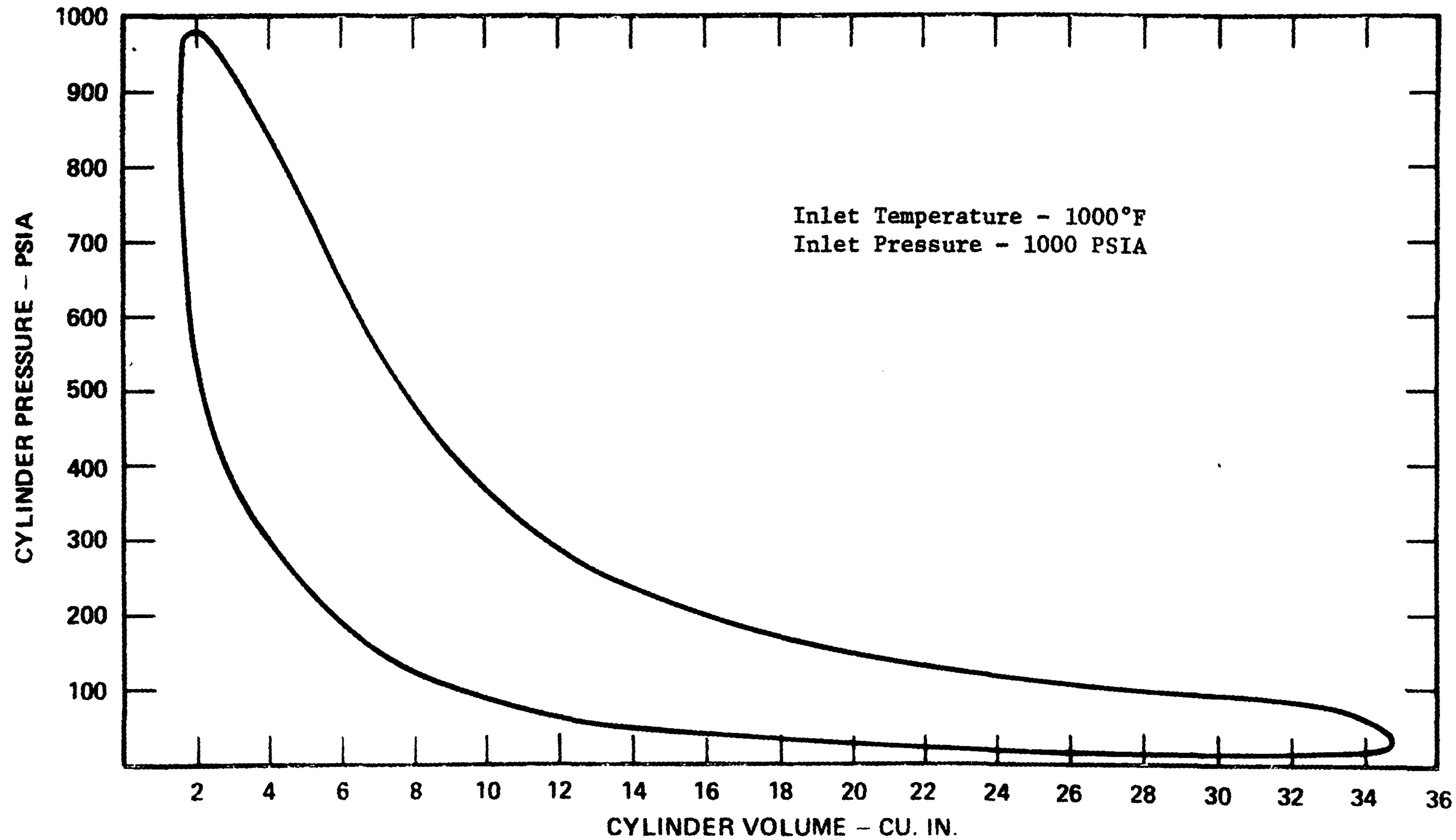


Figure 4.4-2. Expander Indicator Diagram, 2500 RPM, 20° Valve Lead.

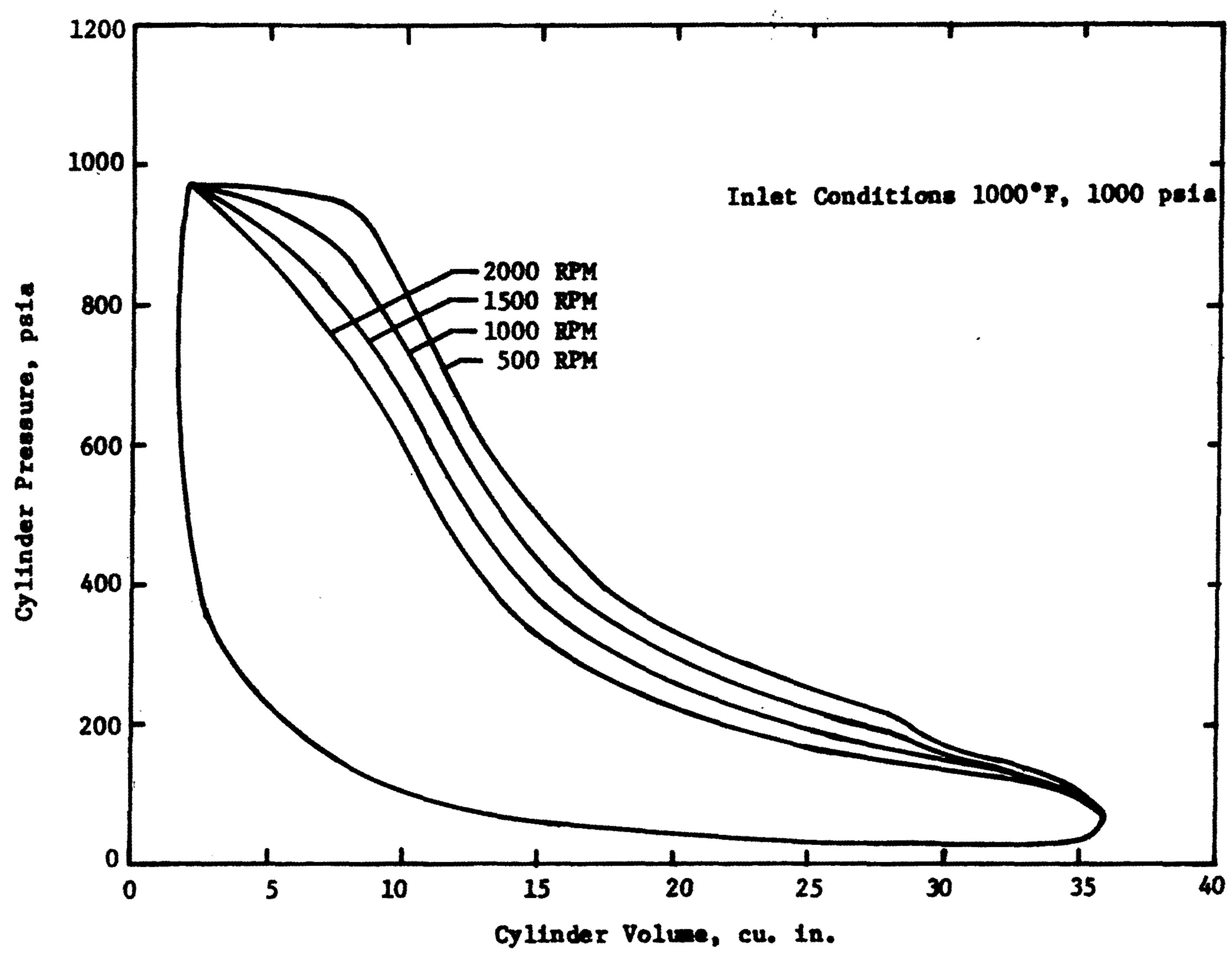


Figure 4.4-3. Theoretical Indicator Diagrams Single Cylinder Expander

TABLE 4.4-I

EXPANDER PARAMETERS FOR INDICATOR DIAGRAMS

Indicator Diagram	Figure 4.4-1	Figure 4.4-2
Valve Event	70°	70°
Valve Lead	10°	20°
Valve Throat Diameter	.75"	.75"
Maximum Lift	.10"	.10"
Cam Base Radius	1.125"	1.125"
Cam Nose Radius	.45"	.45"
RPM	2000	2500
Steam Flow Rate (Before correction for piston blowby and cylinder wall heat transfer)	380 lb/hr	374 lb/hr
Indicated Horsepower	48.6	46.2
Indicated Expander Efficiency	82%	79%

TABLE 4.4-II

EFFECT OF SPEED ON INDICATED EXPANDER PERFORMANCE

(Reference Diagrams of Figure 4.4-3)

Speed	Flow, lb/hr	EMP (psi)	IPH (hp)	Indicated Efficiency
500	139	388	17.1	0.791
1000	255	352	31.0	0.785
1500	319	310	40.9	0.825
2000	380	276	48.6	0.823

analyzed including heat transfer by conduction, convection, and radiation with or without internal heat generation or surface heat flux.

Special attention was given to piston ring, piston and cylinder temperature as a result of steam temperature, frictional heat generation, and the boundary conditions which are expected to exist during expander operation.

Complex geometrical areas in the expanders were replaced with simple geometry to facilitate the node assignment for the computer calculations, without changing the basic design. After dividing the geometry into a series of numbered nodes, measurements were made of full scale drawings and the necessary areas and volumes for each node were calculated and served as input to the computer program. Assumptions and boundary conditions for the computer analysis used were:

1. Due to symmetry, it was assumed that no heat flowed across the centerline or through the circumferential faces.

2. Thermal insulation on the outside of the expander was simulated with an appropriate convection coefficient.

3. The ambient air temperature was taken as 80°F, and the film coefficient for the surface exposed to air was estimated by the appropriate equation for natural convection over a vertical cylindrical surface.

4. Steam temperatures were assumed to be 1000°F in the intake valve region and 540°F in the piston/cylinder region. The exhaust steam temperature used was 250°F.

5. The convective heat transfer coefficient for steam was estimated from conventional equations, such as the Dittus-Boelter equation for flowing fluid. In the region enclosed by the cylinder and the top of the piston, where some steam condensation might occur, the film coefficient was estimated by equations appropriate for film condensation over a flat plate.

6. The cooling oil temperature used was 250°F, and the film coef-

ficients were estimated from equations appropriate for fluid flowing through tubes or over vertical surfaces.

7. All contact coefficients were assumed to be 20,000 Btu/hr-ft^2-°F.

There are five significant places where sliding metal parts generate heat through friction:

1. Graphite ring with cylinder lining (graphite-steel)
2. Crosshead with cylinder (aluminum-steel)
3. Teflon packing with piston rod (teflon-steel)
4. Rider ring with cylinder lining (graphite-steel)
5. Intake valve ring with valve guide (steel-steel).

The heat generated by friction is calculated from:

$$Q_f = \left(\frac{f}{J}\right) \frac{1}{2\pi} \int_0^{2\pi} (F\ u)\ d\theta$$

where f = Sliding coefficient of friction

J = 778 ft-lb/Btu

F = Normal force acting on the sliding surface, a function of θ

u = Sliding velocity, a function of θ

θ = Crank angle

In general the friction heat generated by a variable normal force and a variable sliding velocity is a function of crank angle. For steady state calculations, however, the above equation can be used to calculate the frictional heat, and can be considered as the mean effective frictional heat generated by two sliding parts.

Typical values of the sliding coefficient of friction, f, from the Mechanical Engineers' Handbook - 7th edition, (6) are as follows:

	Dynamic Coefficient of Friction	
Materials	Dry	Grease-Lubricated
1. Steel on steel	0.42	0.03-0.1
2. Steel on graphite	0.21	0.09
3. Teflon on steel	0.04	----

To a first approximation, the frictional heat generated at an expander speed of 2500 RPM are as follows:

Sliding Parts	Total Heat Generated by Friction, Btu/hr
1. Graphite Ring - Cylinder (f = 0.1 and *F = 3 lbf/in of circumference) (59)	305
2. Crosshead - Cylinder (f = 0.07)	1980
3. Teflon packing - piston rod (f = 0.04 and F = 3 lbf/in of circumference)	26.2
4. Rider Ring - Cylinder (f = 0.1)	2840
5. Intake valve - valve guide (f = 0.4 and F = 3 lbf/in of circumference)	170

The total frictional heat was assumed to be distributed evenly between the stationary and the moving parts. For example, for the crosshead-cylinder case above, 990 Btu/hr goes into the crosshead piston and 990 Btu/hr goes into the cylinder.

Steady state temperature distributions were obtained from THT-D program computations, and are shown in Figure 4.4-4 for the crosshead expander and Figure 4.4-5 for the trunk expander. The temperature levels shown are only approximate values since a rigorous thermal analysis was not considered essential. Measured temperature distribution is presented in Section 8.0.

*Ring tension by design. Excludes gas pressure loading.

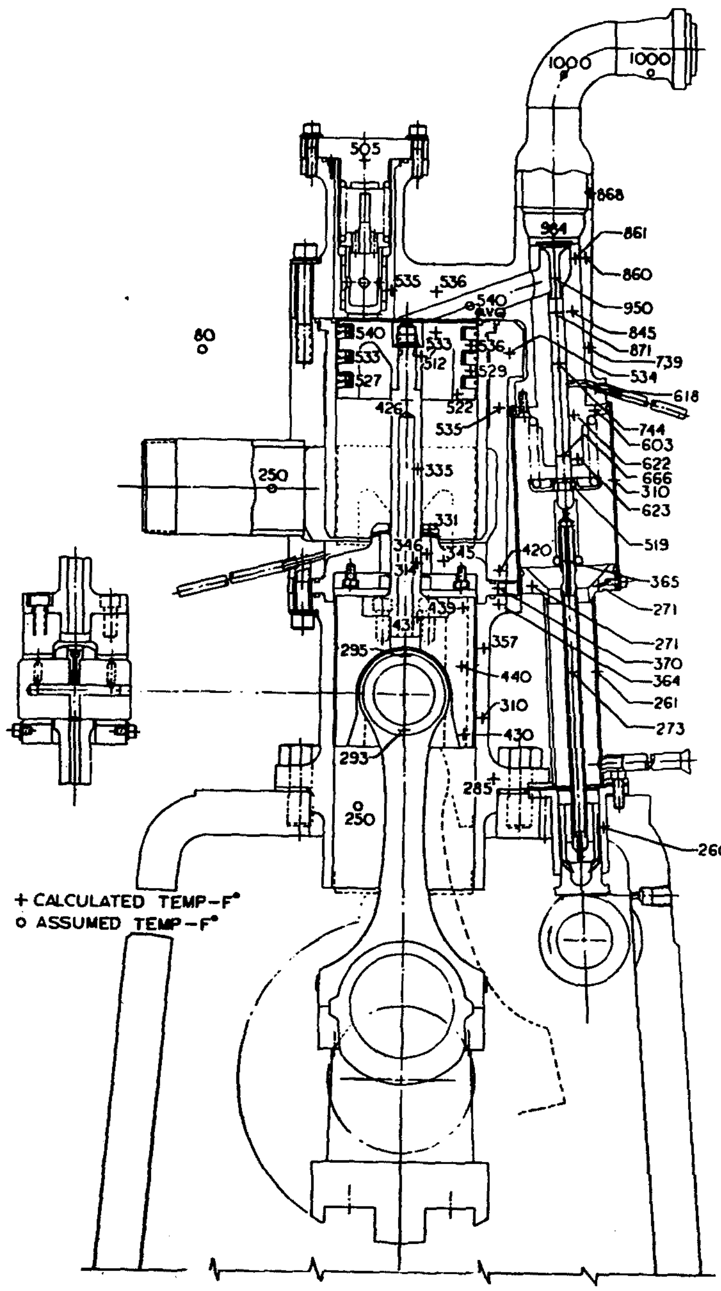


Figure 4.4-4. Calculated Temperature Profile for Crosshead Piston Expander

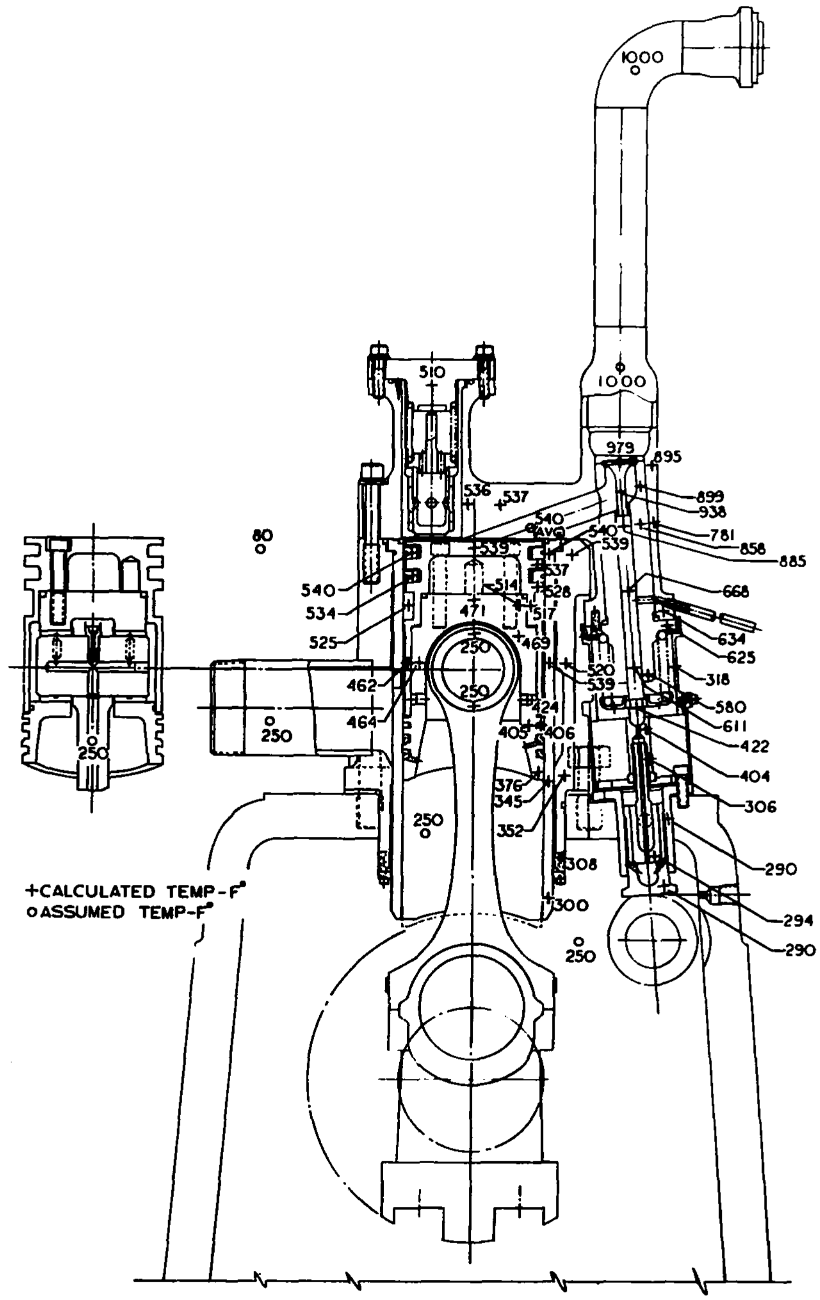


Figure 4.4-5. Calculated Temperature Profile for Trunk Piston Expander

4.4.3 Stress Analysis

4.4.3.1 Crosshead Piston Expander

Before detailed cycle calculations had been made on the crosshead expander, a set of reasonable but conservative estimations of critical loading parameters were made in order to estimate component part stresses. The loading parameters were calculated in sufficient detail to allow their estimation at any crank-angle throughout the cycle, so that they could be combined to determine their worst-case summations.

The piston position, velocity, and acceleration; cylinder swept volume; and inertia force were calculated for 1000 RPM and a 6.146 lb. reciprocating weight which loads the wrist pin. These parameters could be corrected for different rotative speeds and reciprocating weights. The piston pressure forces were calculated using the pressures calculated from an assumed cycle with 15 percent cut-off ($\sim$ 40° ATDC) and isentropic expansion from 40° to 135° (exhaust port opening) according to:

$$PV^{1.28} = 1000\ (2.747)^{1.28} = 3645$$

The pressure remained at the condenser pressure of 20 psia until the exhaust port closed at 225°, after which isentropic compression occurred according to:

$$PV^{1.28} = 20\ (32.535)^{1.28} = 1725$$

This model gave a mean pressure of 290 psi and temperature of 575°F.

The vertical pressure and inertia forces on the wrist pin were combined to give the total vertical forces. Side pressure and inertia forces on the wrist pin and crosshead piston were calculated and combined with the total vertical forces to determine the total wrist pin loads. The inertia forces due to angular acceleration of the connecting rod were included. The combined forces at 2500 RPM varied from 250 - 7800 lb. or 110 - 3470 psi projected pressures on the wrist pin bearing area of 2.25 in^2.

Table 4.4-III and Figure 4.4-6 identify the stresses and factors of safety (FS) at the principal points of interest in the crosshead expander.

4.4.3.2 Trunk Piston Expander

There are virtually no differences between many of the components of the trunk expander and the corresponding ones of the crosshead expander. The principal changes are in the piston assembly and the connecting rod. The same type of cycle modeling assumptions were made in order to predict internal pressures and loads.

Figure 4.4-7 and Table 4.4-IV identify critical stress locations, stresses, and factors of safety. Those which are the same as for the crosshead expander are omitted.

4.4.4 Piston Ring Wear Prediction

4.4.4.1 Crosshead Power Piston Graphite Rings

Zero wear is defined as wear so slight that the surface finish of the wear track is not significantly different from the finish of the virgin surface(2). The wear scar is, then, roughly equal to one-half of the peak-to-peak value of the surface finish.

The engineering model states that the wear between two loaded sliding surfaces can be controlled by limiting the maximum shear stress, τ_{max}, in the contact region. Zero wear can be obtained for a specific number of passes if:

$$\tau_{max} \leq \rho \quad \tau_y \qquad \text{psi}$$

The fractional parameter ρ is dependent upon the number of passes, the materials and lubricant used, and the lubricating conditions, as shown in Table 4.4-V. For 2000 passes, ρ is designated as ρ_R.

TABLE 4.4-III

STRESSES AND FACTORS OF SAFETY AT PRINCIPAL POINTS OF INTEREST IN THE CROSSHEAD EXPANDER

(Figure 4.4-6 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[(a)] Stress, ksi	Factor of Safety, -
A	12.3 ± 12.3	± 42.0	3.41
B	0.9 ± 0.9	± 42.0	47.3
C	3.1 ± 3.1	± 42.0	13.4
E	22.6 ± 13.2	± 47.0	3.57
F	14.2 ± 52.0	± 64.8	1.25
G	20.5 ± 20.5	± 50.0	2.44
H	2.7 ± 0.5	± 4.5	9.09
I	44.7 ± 12.1	± 48.8	4.05
J	2.1 ± 2.1	± 4.5	2.14
K	36.5 ± 24.3	± 61.8	2.55
L	86.0 ± 15.6	± 31.5	2.03
M	17.6	43.0	2.44
N	42.4 ± 42.4	± 75.0	1.77
O	17.7 ± 17.7	± 42.0	2.37
P	78.0 _ 88.1	125.0	1.42[(b)]
Q	12.2 ± 10.3	± 51.5	5.00
R	42.6 ± 7.6	± 40.0	5.26
S	75.2 ± 29.1	± 55.0	1.88
T	30.1 ± 13.7	± 34.0	2.48
U	9.6 ± 0.8	± 50.5	11.63
V	53.0 ± 20.5	± 31.0	1.51
W	57.3 ± 8.2	± 35.0	4.27
X	67.8	81.0	1.19
Y	1.1 ± 1.1	± 18.5	17.1
Z	30.7 ± 27.3	± 44.8	1.64
AA	41.6	84.0	2.02
AB	43.9	120.0	2.82
AC	27.0	32.0	1.19
AD	5.5 ± 5.5	± 16.0	2.94

TABLE 4.4-III (Cont'd.)

STRESSES AND FACTORS OF SAFETY AT PRINCIPAL POINTS OF INTEREST IN THE CROSSHEAD EXPANDER

(Figure 4.4-6 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[(a)] Stress, ksi	Factor of Safety, -
AE	12.3	32.0	2.60
AF	2460 lb	3590 lb	1.46[(c)]
AG	10.8 ± 10.8	± 20.0	1.86
AH	57.7 ± 20.9	± 44.5	2.13
AI	77.2 ± 16.8	± 36.0	2.14
AJ	15.1 ± 15.1	± 20.0	1.33

NOTES: (a) The allowable stresses shown as ± are the allowable alternating stresses when the mean stress if the same as shown in the effective stress tabulation, operating temperature for 1000 hrs. considered.

(b) Spring - Maximum allowable stress is 125.0 ksi for initial stress of 78.0 ksi. Maximum imposed stress is 88.1 ksi.

(c) FS = critical load/applied load.

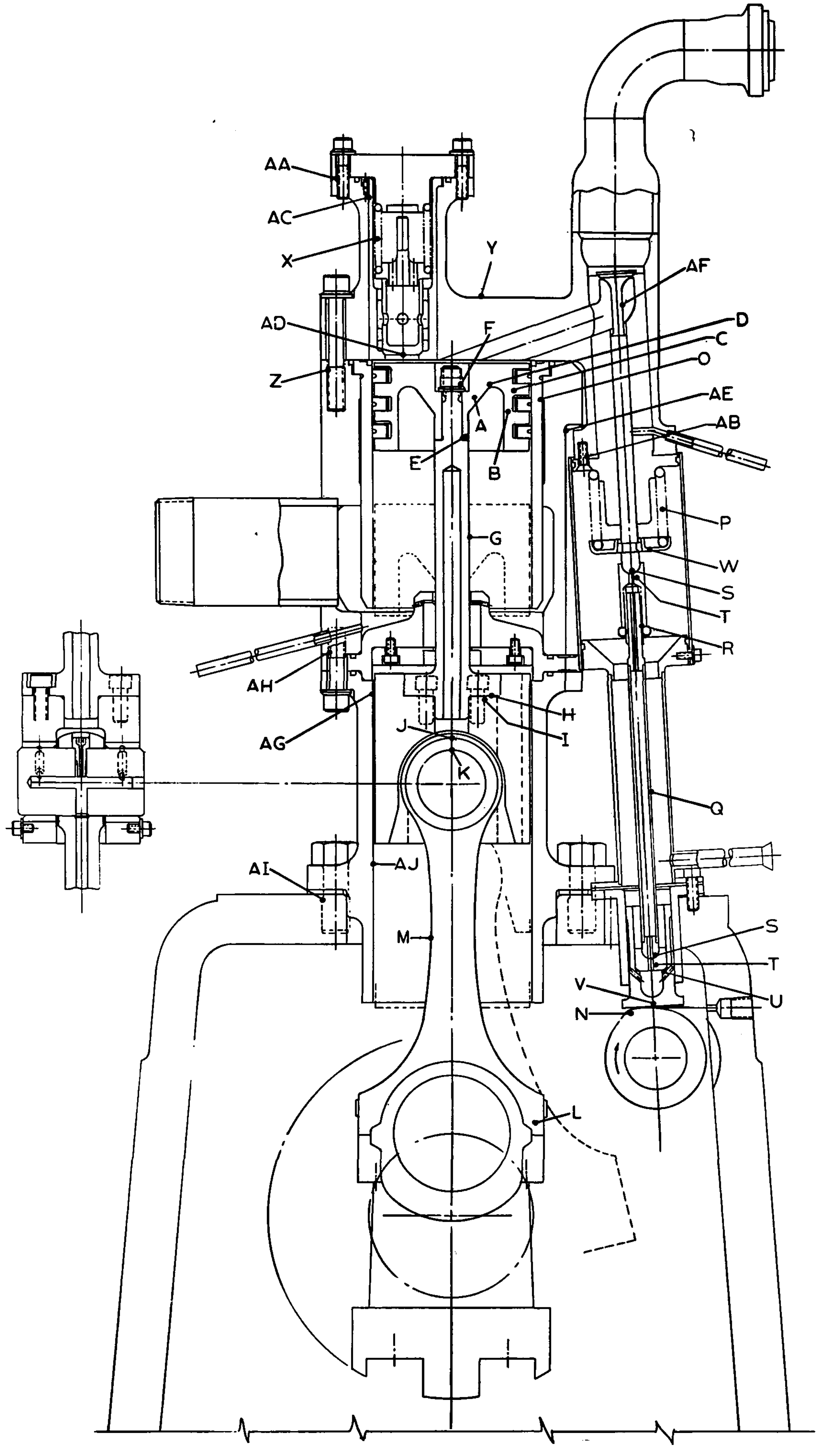


Figure 4.4-6. Location of Stresses, Crosshead Expander (Ref. to Table 4.4-III)

TABLE 4.4-IV

STRESSES AND FACTORS OF SAFETY AT PRINCIPAL POINTS OF INTEREST IN THE TRUNK EXPANDER

(Figure 4.4-7 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[(a)] Stress, ksi	Factor of Safety, -
A	70.2 ± 3.2	± 31.5	9.90
B	10.0 ± 10.0	± 47.5	4.75
C	3.7 ± 3.7	± 47.5	12.9
G	4.9	11.6	2.38
H	4.6 ± 2.7	± 5.0	1.86
I	51.3 ± 22.3	± 47.0	2.10
J	2.0 ± 2.0	± 6.7	3.36
K	34.3 ± 23.5	± 61.8	2.63
L	82.5 ± 6.2	± 33.5	5.41
M	17.6	39.6	2.25
Q	4.2 ± 3.8	± 46.6	12.2
R	71.0 ± 12.2	± 39.3	3.21
S	36.8 ± 36.8	± 44.3	1.20
T	2.9 ± 2.9	± 29.6	10.1
U	4.5 ± 4.5	± 44.3	9.88
V	14.3 ± 14.3	± 31.0	2.17
W	65.6 ± 3.9	± 41.8	10.7

NOTES: (a) The allowable stresses shown as ± are the allowable alternating stresses when the mean stress is the same as shown in the effective stress tabulation, operating temperature for 1000 hr. considered.

(b) Stresses for locations not shown are the same as for the crosshead expander.

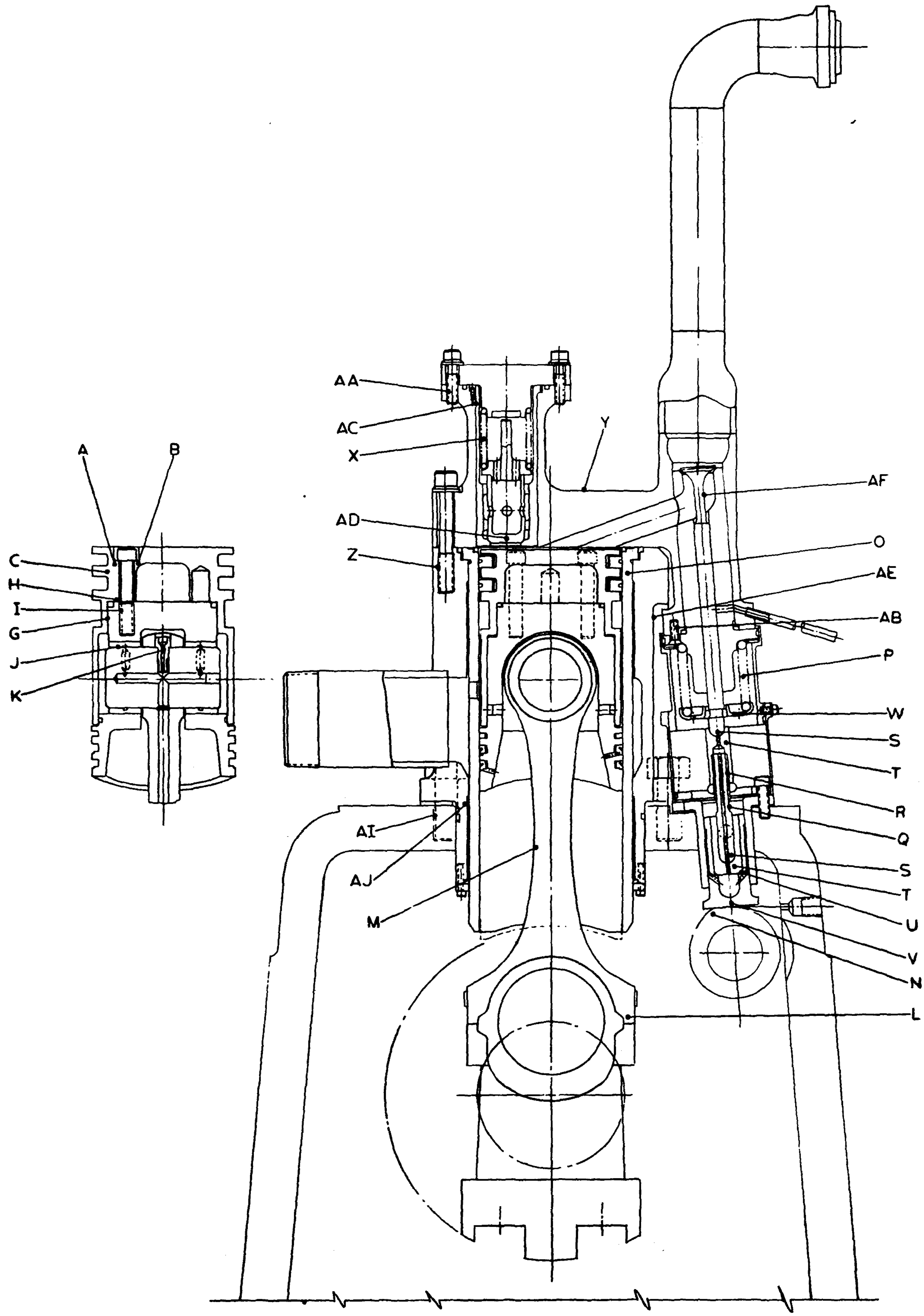


Figure 4.4-7. Critical Stress Locations for Trunk Expander
(Refer to Table 4.4-IV)

TABLE 4.4-V

FRACTIONAL PARAMETER

Type of Lubrication	ρ_R	Material/Lubricant System
Quasi-hydrodynamic	1.00	Fully hydrohynamic lubrication
	0.54	Partial hydrodynamic lubrication
Dry or boundary lubrication	0.54	Low susceptibility to transfer
	0.20	High susceptibility to transfer

Since no published data exists for the exact material combinations to be used in the single cylinder expanders, it is estimated that $\rho_R = 0.20$ for the unlubricated graphite ring in contact with a steel cylinder. This estimate was thought to be conservative, and is the value found for dry 52100 steel vs. 7 stainless steels.[2]

The properties of the piston ring material, U.S. Graphite Co. Grade 107, are shown in Table 4.4-VI. In the absence of specific information concerning its allowable shear yield strength, it is assumed that τ_y = 7500 psi, the RT tensile strength. The ring material is considered so brittle that the yield and fracture strengths are equal. (Another method of estimating τ_y from the microhardness gives a larger allowable τ_y.)

TABLE 4.4-VI

PROPERTIES OF GRADE 107 CARBON GRAPHITE

Composition	-	Carbon graphite impregnated with Sb	
Strength	-	at RT, will increase with temperature	
		Compression	- 38,000 psi
		Transverse fracture	- 10,000 psi
		Tensile	- 7,500 psi
Elastic Modulus	-	$E = 3.5 \times 10^6$ psi	
Hardness	-	Sclersocope	92
		DPH (converted)	865

The pressure, and therefore, frictional loads on the ring/cylinder interface will not be constant throughout the cycle. The pressure differential will decrease toward zero as the steam expands in the cylinder and the exhaust ports are uncovered. Using an idealized representation of the pressure vs. crank angle relationship a mean effective cylinder pressure of 290 psi will occur for the 15% cut-off condition.

The contact load, F' at the ring/cylinder interface will be a function of the spring load, F'_s, the pressure load, F'_p, the frictional load, F'_μ, and the ring geometry as shown by Figure 4.4-8.

Assuming that all radially outward adjustments in ring position which are required for wear compensation are made at bottom dead center, where all forces are zero except for the spring force, the horizontal frictional force is zero during the rest of the cycle. The contact stress equation reduces to:

$$q_o = \frac{F'}{A'} = \frac{4 + 0.0622\ \Delta P - 0.0296\ P_1}{0.2415} \qquad \text{psi}$$

To determine conditions which will ensure zero wear for a desired life of N passes for a given mechanism, the following relation has been derived:

$$\tau_{max} = K\ q_o\ \sqrt{0.5^2 + \mu^2} \leq \left(\frac{2000}{N}\right)^{1/9} \rho_R \tau_y \qquad \text{psi}$$

Solutions are shown in Figure 4.4-9 for the graphite piston rings, for the average value of P_1 = 290 psia, ΔP equal to three fractions of P_1, and coefficients of friction from 0.05 to 0.50 as functions of rotative speed.

By combining equations the fractional area of the piston ring with a stress concentration less than the allowable for zero wear in 1000 hours is determined. These calculations are also plotted in Figure 4.4-9 as fractional area with zero wear. The significance of this figure is to show that, according to analytical predictions, the graphite piston rings should experience only minor wear near the edges under the ideal conditions assumed in the calculations.

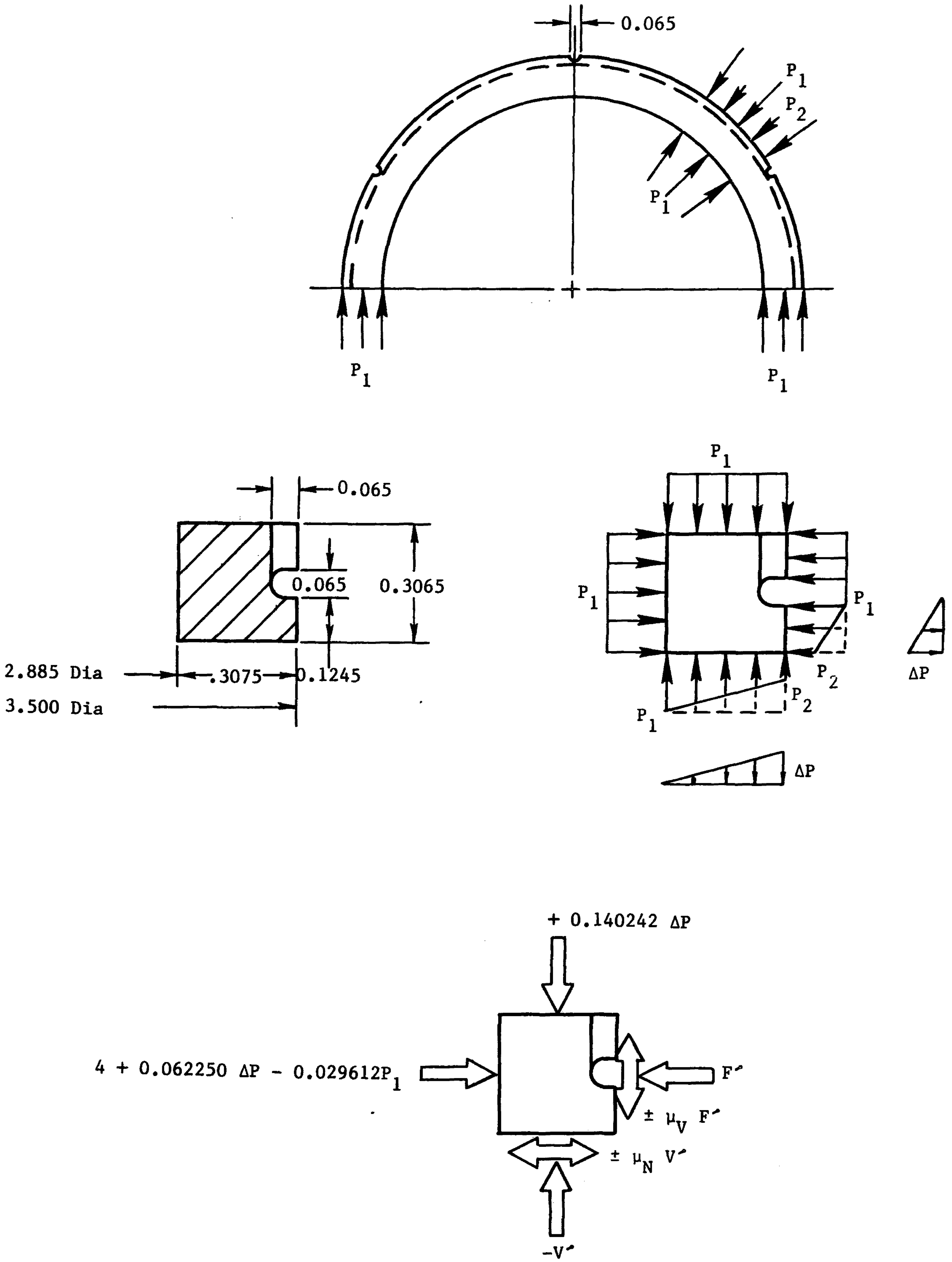


Figure 4.4-8. Graphite Ring Geometry and Load Components (lb/3.5π in.)

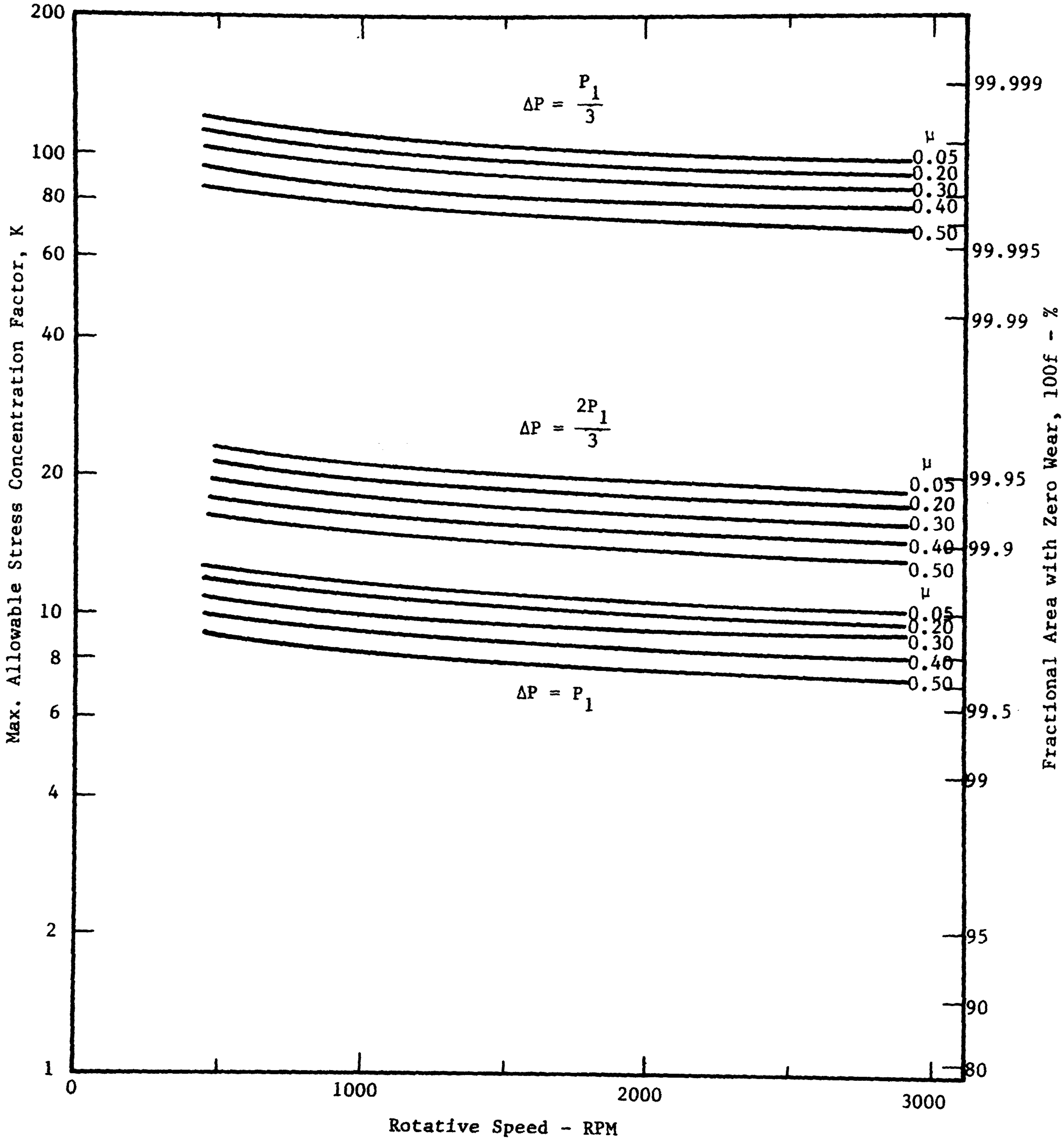


Figure 4.4-9. Maximum Allowable Stress Concentration Factor and Fractional Area with Zero Wear of Graphite Piston Rings in 1000 Hr.
Mean P_1 = 290 psia = upstream pressure on piston ring.
ΔP = pressure drop across piston ring.

This analysis assumes certain ideal conditions which are not expected to exist in the actual single cylinder expander. For example, the analysis makes no provisions for wear due to foreign abrasive particles, wear due to ring misalignment or rocking in the piston ring groove, or wear caused by rough surfaces such as scored cylinder walls.

4.4.4.2 Trunk Expander Piston Rider Ring

In the trunk expander rider ring wear analysis, it was assumed that there were no steam pressure forces acting on the rider ring since the ring was notched on its rubbing face to prevent it from becoming a pressure sealing ring. Therefore, the only load on the rider ring is the side force imposed by steam pressure on top of the piston and the opposing force from the connecting rod. Piston inertia forces are also included. The trunk piston was designed so that the graphite rider ring supported only half of the total side forces. The bottom skirt of the piston supported the other half of the piston total side load.

The cyclic average of all positive side loads on the graphite rider ring is 210 lbs., and the cyclic average of all negative side loads (on the opposite side of the piston) is only 20 lbs. Only the 210 lb. load will be considered in the following discussion. The width (axial length) of the graphite rider ring to be considered is 0.7 inches.

For a ring outside diameter which is less than the cylinder diameter, the Hertz formula(7) for the contact stress is:

$$\text{Max } S_c = q_o = 0.798 \sqrt{\frac{[P]\ \dfrac{D_1 - D_2}{D_1 D_2}}{\left[\dfrac{1-\nu_1^{\,2}}{E_1} + \dfrac{1-\nu_2^{\,2}}{E_2}\right]}}\ ,\ \text{psi} \qquad \text{(a)}$$

where D_1 = Cylinder dia. = 3.5 inches

D_2 = Ring dia., inches

$E_1 = 3.5 \times 10^6\ \text{lb/in}^2$ (graphite) psi

$E_2 = 30 \times 10^6\ \text{lb/in}^2$ (steel) psi

$\nu_1 = \nu_2$ = Poisson's ratio = 0.3

P = load/linear inch = 210 lb/0.7 inch

Equation (a) reduces to:

$$q_o = 13711 \sqrt{\frac{3.5}{D_2} - 1} \text{ , psi} \qquad \text{(b)}$$

For conforming geometries, the maximum shear stress, τ_{max}, is a function of q_o, a corner or edge stress concentration factor K, and the coefficient of friction μ.[2]

$$\tau_{max} = Kq_o \sqrt{0.25 + \mu^2} \leq \left(\frac{2000}{N}\right)^{1/9} \rho_R \tau_y \text{ , psi} \qquad \text{(c)}$$

The latter half of the above expression are the conditions required to ensure zero wear for a life of N passes for a given mechanism with the fractional parameter $\rho_R = 0.20$ and shear yield strength of 7500 psi.

Solutions are shown in Figure 4.4-10 for the graphite rider ring for expander rotative speeds of 500 - 3000 RPM, for coefficients of friction $0.05 \leq \mu \leq 0.50$, and for diametral differences $(D_1-D_2) = \Delta D$ of 0.1, 1, and 10 mils.

The fractional area of the rider ring with a stress concentration factor less than the allowable for zero wear in 1000 hours is shown plotted in Figure 4.4-10.

It may be concluded that the higher contact stresses generated by a large difference in ring and cylinder diameters will cause greater-than-zero wear over a significant fraction of the ring face, while only very small regions near the edges of the ring will have greater-than-zero wear when the diameter difference is small. At least two conditions will tend to reduce an initially large diameter difference: (1) mechanical or thermal loads could spring the ring ends outward and increase the effective diameter of the ring, and (2) the ring will wear toward exactly the same diameter as the cylinder with time. Either should decrease the initial wear rate significantly. Also, as the ring edges wear from sharp corners to circular arcs, stress concentration factors and wear rates should decrease.

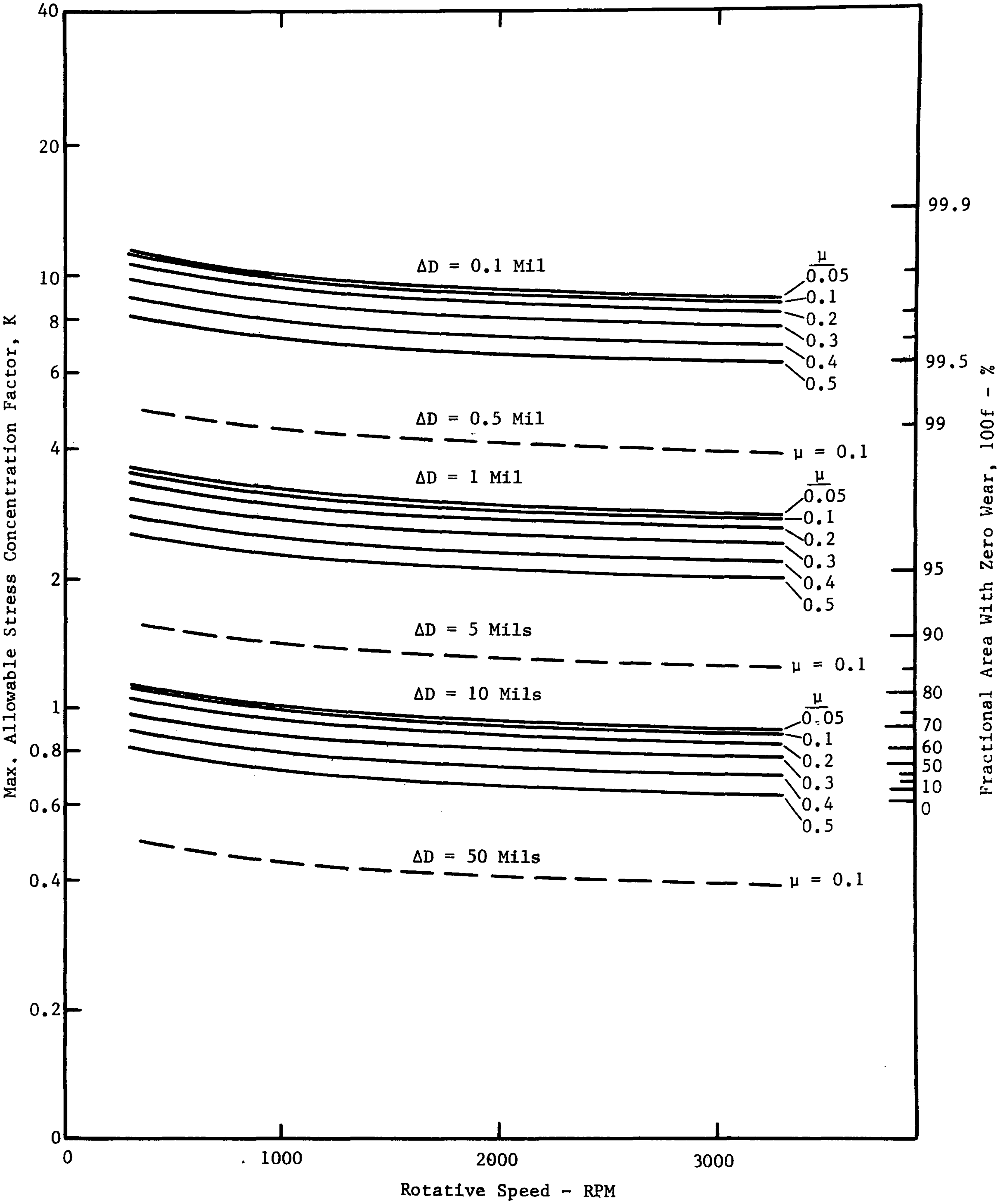


Figure 4.4-10. Maximum Allowable Stress Concentration Factor and Fractional Area With Zero Wear of Graphite Trunk Piston Rider Ring in 1000 Hours.

Using calculated operating temperatures and manufacturing drawing tolerances, the average ΔD was 11 mils for the Type 440C stainless steel cylinder liner and graphite rider ring. Initially, a large wear rate can be expected to occur, but the wear rate should decrease significantly as the wear process laps the ring and cylinder toward the same diameter and the ring edges toward circular arcs.

A satisfactory wear life is predicted by the above simplified and ideal model.

4.4.5 Recompression Valve Dynamics

A finite time increment analysis was made of the recompression valve dynamics. A constant increment of one crank-angle degree was used which gave time increments of 1/3000 sec at 500 RPM to 1/15000 sec at 2500 RPM. The computation method was as follows:

- Assume a condenser pressure and rotative speed, and let inlet pressure = 1000 psia.
- Start calculations at the crank angle at which the calculated cylinder isentropic recompression pressure reached 1200 psia, producing a valve lifting force greater than the 78 lb. recompression spring preload.
- Determine recompression valve acceleration from F = ma, or $a = F/1618.76\ \text{lbm sec}^2/\text{in.}$
- Determine valve initial, average, and final velocities upward for each time period.
- Determine valve initial, average, and final lifts upward for each time period.
- Determine average valve flow area for each time period.
- Determine average valve flow coefficient for each time period.
- Determine average steam flow through valve for each time period.
- Deduct steam flow through valve from cylinder volume at the beginning of the next time period and repeat calculations for the new time period.

- Continue calculations until the inlet valve opened at either $-20°$ for 2500 RPM or $-10°$ for $\leq$ 2000 RPM by cam action (acceptable solution) or until the recompression pressure reached 1670 psia and produced a net lifting force on the inlet valve (unacceptable solution).

Samples of such calculations are shown on Figure 4.4-11. All of the conditions shown except for 1000 RPM, P_c = 100 psia, are acceptable since the inlet valve will open from cam action before 1670 psia recompression pressure is reached.

The results of all calculations are summarized on Figure 4.4-12 which show the allowable variation of condenser pressure with RPM. Less favorable results occur when the inlet pressure is less than 1000 psia.

4.4.6 Crosshead Expander Power Piston Rod Natural Frequency

The transverse bending natural frequency of the crosshead power piston and its rod as a cantilevered beam with the piston mass at its free end was calculated using the formula:[8]

$$\omega^2 = \frac{k}{M+0.23\ m} = \frac{3EI}{L^3\ (M+0.23\ m)} \quad \text{rad}^2/\text{sec}^2$$

where $E = 28 \times 10^6$ psi (rod at 300°F)

$I = 0.01508\ \text{in}^4$ (rod)

$L = 5.75$ in.

$mg = 0.6217$ lb (rod-calculated)

$Mg = 3.043$ lb (piston-measured and calculated)

Then $\omega = 899$ rad/sec, and $f = (\frac{\omega}{2\pi})\ 60 = 8580$ cpm.

Therefore, the rod/piston system was considered satisfactory for the first three harmonics of the maximum operating speed of 2500 RPM, and for higher harmonics for lower operating speeds.

The load deflection characteristics of the piston rod were calculated using the formula for a cantilever beam with an intermediate load:[7]

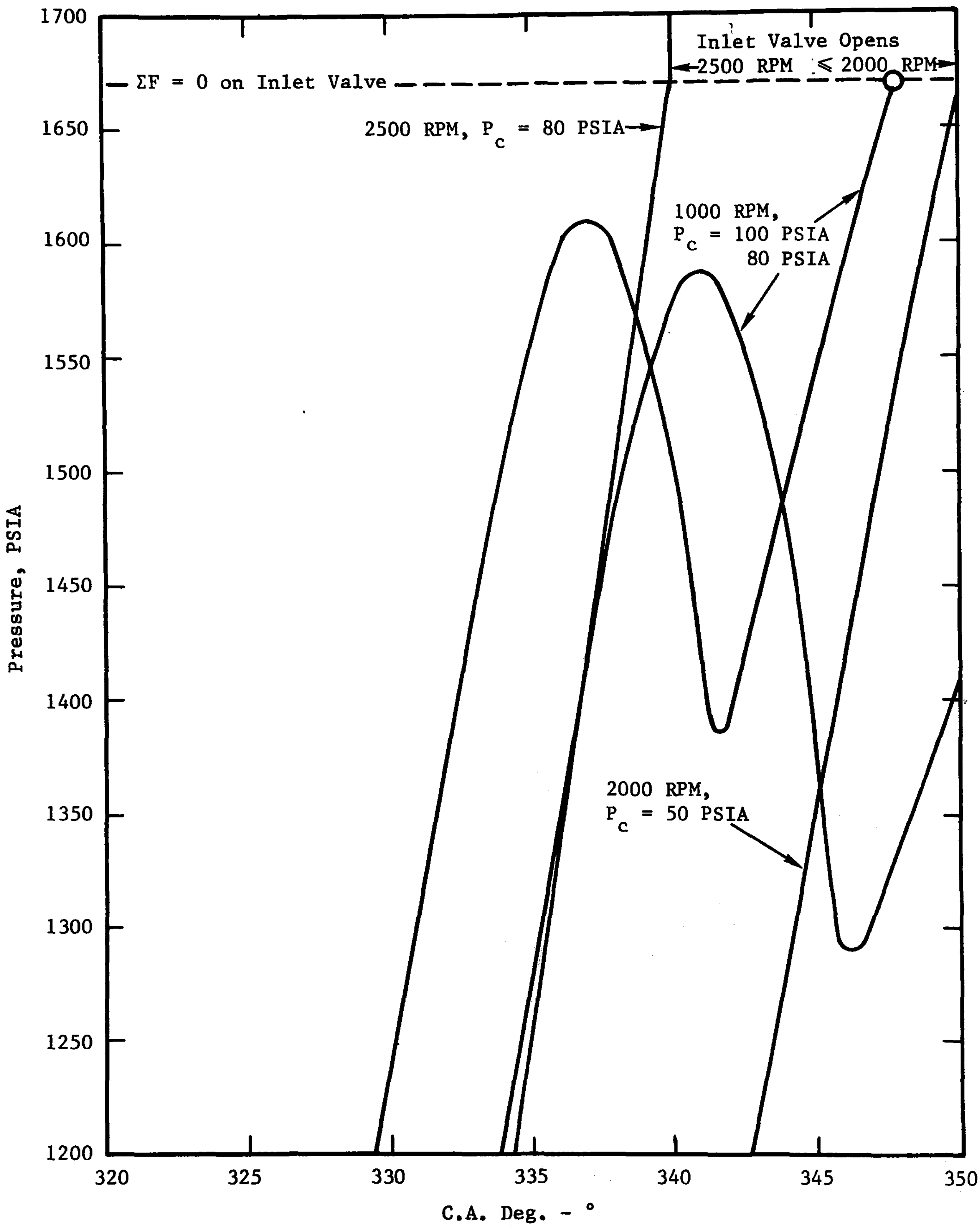


Figure 4.4-11. Examples of Recompression Valve Dynamic Calculations for Inlet Pressure of 1000 PSIA and Different Rotative Speeds and Condenser Pressure, P_c. All Solutions Shown are Acceptable Except for 1000 RPM, Where P_c = 100 PSIA.

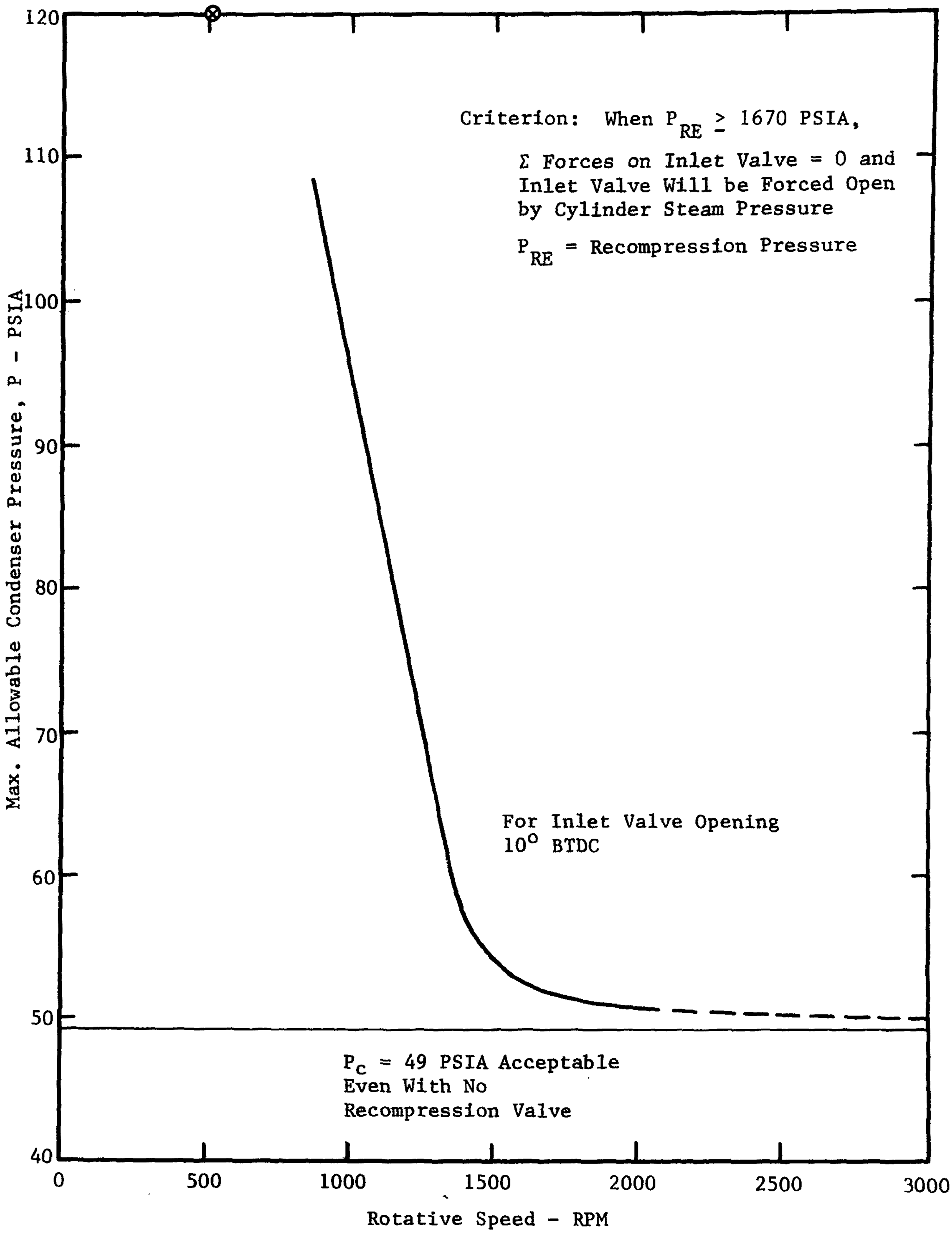


Figure 4.4-12. Maximum Allowable Condenser Pressure for Inlet Pressure of 1000 PSIA and 78 lb. Recompression Valve Spring Preload.

$$y = -\frac{W}{6EI}(3\ a^2\ell - a^3) = -1.924 \times 10^{-4}\ W\ \text{in.}$$

where $a = 5.75$ in.

$\ell = 6.83$ in. (fixed end of rod to top of piston)

W = applied load, lb.

E and I same as previous values

The top of the piston will be deflected 0.010 in. radially by a side load of 52 lb. applied at the free end of the rod (approximately the bottom of the piston).

5.0 EXPANDER MATERIALS

5.1 Materials Technology Base

In recently designed steam reciprocating engines for automotive applications[12], maximum steam temperatures of about 700°F have been utilized. Temperatures have been restricted to this general range by materials limitations, in one form or another. For example, the GM SE-101 engine had a maximum steam temperature of 700°F in recognition of the possible thermal degradation of the organic lubricant used to prevent excessive wear at the steam piston cylinder interface. A design pressure of 800 psia arose from considerations of expander bearing loads. These operating conditions (700°F, 800 psia) might be considered to be current practice in the design of steam engines for automotive applications. Engine thermal efficiency is quite limited, under these conditions, as compared with modern central steam power stations operating at 1000° to 1050°F and 2400-3500 psia. These advanced conditions have been reached by increasing technology over the last 40 years, the same period in which there has been essentially no increase in the cycle conditions for the steam car engine.

The higher temperatures associated with the more efficient automotive steam engine place constraints on materials due to mechanical strength and corrosion considerations. To insure the long life ($\sim$ 3000 hours) required for satisfactory use in the reciprocating steam expanders, the materials must be resistant to oxidation, corrosion by steam and have adequate mechanical properties (yield strength, resistance to creep and cyclic loading, hardness, etc.) at the anticipated operating conditions. Surface properties can be favorably altered by the application of a surface coating; use of surface coatings are considered where appropriate, i.e., wear surfaces. In addition to the constraints imposed on materials by the engine operating conditions, economic constraints can limit the utilization of existing technology in the solution to materials problems.

Cost is a major factor in large quantity production of automotive components. Economy dictates that the lowest cost alloy or material, with properties suitable to the service conditions, be used.

Probably, the most limiting factor in the successful operation of highly efficient steam engines will be lubrication. The dry conditions which superheated steam imposes on a system at the operating conditions can cause severe galling and excessive wear problems in those components whose surfaces are in relative motion, i.e., the cylinder liner/piston ring interface and the inlet valve face/seat and inlet valve stem/guide interfaces. Lubrication of these components are discussed in Section 6.0.

A literature search and industrial survey was made to identify and evaluate the current materials technology in support of the expander designs described in Section 4.0. Personal visits were made to the Ford Motor Company Automotive Research and Engineering Center and the General Electric Company Diesel Engine Department, Large Steam Turbine and Generator Department and Medium Steam Turbine and Generator Department to identify materials used in current reciprocating and steam turbine components. To supplement the information obtained from these visits, library searches were made covering automotive components and oxidation and wear resistant materials for use in dry steam environment through the General Electric Technical Information Center Library and Automatic Information Retrieval System, the SAE and AIME-ASM Transactions and the Engineering Index.

A summary of the results obtained from the technology review follows:

5.1.1 Cylinder Block, Cylinder Head, Intake Manifold, Exhaust Manifold

Experience with materials in large central station steam power station has shown that the useful life of steel is essentially unlimited if the proper steel is selected and the service conditions and water chemistry are properly controlled. Some steels have been in service for forty to fifty years. Invariably when a failure occurs the cause can be attributed to impurities in the steam and the problem usually is solved by correction of the service conditions. Plain carbon and low alloy steels are commonly used for the fabrication of pressure-containing components such as feedwater piping, steam piping, valve bodies, flanges, discs, etc., that are regularly found in steam generating systems. The most commonly used steels are listed in Table 5.1-I together with the appropriate ASTM designations and maximum service temperatures.

The use of these steels up to the maximum temperature listed in Table 5.1-I will vary with the anticipated service conditions and from user to user. For example, one manufacturer limits the use of 1.25% Cr-0.5% Mo steel to 800°F. In actual practice carbon steels have been used successfully in steam service at temperatures to 750°F for 25 to 30 years. Similarly, the 0.5% Mo, the 1.25% Cr-0.5% Mo and the 2.25% Cr - 1.0% Mo alloy steels have proven to be satisfactory for service at temperatures up to 850°, 1000°, and 1050°F, respectively. However, there is a major restriction in the use of 0.5% Mo steel in that it is not recommended for use in welded structures. Welded 0.5% Mo steel has a strong tendency to graphitize in high temperature service. All of these steels can be considered for use in the major static components for automotive steam engines.

In contrast to the high pressure-containing components in steam generating systems, the materials currently being utilized in the production of the cylinder blocks, cylinder heads and intake manifolds of gasoline and diesel fueled internal combustion engines for passenger cars, trucks and locomotives are gray cast iron and cast steel. The most common gray cast iron being used for the production of these components in passenger cars is SAE G4000 (ultimate strength - 40,000 psi). SAE G4000 contains 3 - 3.3% C, 1.8 - 2.1% Si, 0.6 - 0.9% Mn and consists of a lamellar pearlite matrix with Type A flake graphite. For parts subjected to higher pressure or heavy duty engines, SAE G6000 (ultimate strength 60,000 psi) gray cast iron is used. In one large diesel engine the cylinder head is produced from a cast, weldable Mn-Mo steel (0.20% C - 1.35% Mn - 0.25% Mo). The steel casting is normalized at 1650°F (during the hardening of the valve seats) to a hardness of 180 - 225 BHN. This treatment produces an ultimate strength of 90,000 psi, a yield strength of 60,000 psi, tensile elongation of 10% and reduction-in-area of 25%. Special heat resistant parts are frequently made from Ni-Mo alloy steel. Intake manifolds in some internal combustion engines are also being produced from cast aluminum alloy 355

Table 5.1-I

Plain Carbon and Low Alloy Steels Used for Pressure Containing Components in Steam Generating Plants (13)

	Alloy	ASTM Designation	Form	Max. Service Temp., °F
1.	Carbon Steel (0.35% C max)	A216 (Grade WCB)	Casting	850
2.	Carbon Steel (0.25% C max)	A216 (Grade WCA)	Casting	850
3.	Carbon Steel (0.35% C max)	A105 (Grades 1, 11)	Forging	850
4.	Carbon Steel (0.25-35% C max)	A106 (Grades A, B,C)	Pipe	850
5.	Carbon Steel (0.06-0.18% C - 0.35% Si max)	A192	Boiler Tube - High Pressure	850
6.	Carbon Steel (0.27% C max)	A210	Superheater Tube	850
7.	0.25% C - 0.5% Mo	A217 (Grade WC1)	Casting	850
8.	0.25% C - 0.5% Mo	A182 (Grade F1)	Forging	850
9.	0.15% C - 0.5% Mo	A335 (Grade P1)	Pipe	850
10.	0.20% C - 0.5% Mo	A209 (Grade Tia)	Boiler and Superheater Tubes	950
11.	0.15% - 1.0% Cr - 0.5% Mo	A182 (Grade F12)	Forging	1050
12.	0.15% C - 1.0% Cr - 0.5% Mo	A225 (Grade P12)	Pipe	1000
13.	0.20% C - 1.25% Cr - 0.5% Mo	A217 (Grade WC6)	Casting	1050
14.	0.15% C - 1.25% Cr - 0.5% Mo	A182 (Grade F11)	Forging	1050
15.	0.15% C - 1.25% Cr - 0.5% Mo	A225 (Grade P11)	Pipe	1050
16.	0.15% C - 1.25% Cr - 0.5% Mo	A213 (Grade T11)	Boiler & Superheater Tubes	1100
17.	0.18% C - 2.5% Cr - 1.0% Mo	A217 (Grade WC9)	Casting	1050
18.	0.15% C - 2.25% Cr - 1.0% Mo	A182 (Grade F22)	Forging	1050

Table 5.1-I (Cont'd.)

Plain Carbon and Low Alloy Steels Used for Pressure Containing Components in Steam Generating Plants (13)

	Alloy	ASTM Designation	Form	Max. Service Temp., °F
19.	0.15% C - 2.25% Cr - 1.0% Mo	A225 (Grade P22)	Pipe	1050
20.	0.15% C - 2.0% Cr - 0.5% Mo	A213 (Grade T3b)	Boiler & Super-heater Tubes	1200
21.	0.15% C - 2.25% Cr - 1.0% Mo	A213 (Grade T22)	Boiler & Super-heater Tubes	1200
22.	0.15% C - 3.0% Cr - 1.0% Mo	A213 (Grade T21)	Boiler & Super-heater Tubes	1200
23.	0.15% C - 5.0% Cr - 0.5% Mo	A213 (Grade T5)	Boiler & Super-heater Tubes	1200
24.	0.15% C - 9.0% Cr - 1.0% Mo	A213 (Grade T9)	Boiler & Super-heater Tubes	1200
25.	Type 304 SS (18% Cr - 10% Ni)	A213 (Grade TP304)	Boiler & Super-heater Tubes	1500
26.	Type 321 SS (18% Cr - 10% Ni-Ti)	A213 (Grade TP321)	Boiler & Super-heater Tubes	1500

(Al-5% Si - 1.25% Cu - 0.5% Mg - 0.1% Zn - 0.2% Ti - 0.1% Mn - 0.2% Fe).

Materials used for the exhaust manifold in internal combustion engines are subjected to the highest temperatures in the engine and would correspond to the intake manifold of the steam expander. Alloyed cast iron is most commonly used for the exhaust manifold in the internal combustion engine. The SAE G4000d, a Cr-Mo alloyed cast iron and SAE G4000e a Cr-Mo-Ni alloyed cast iron are typical. The range of operating temperatures that have been reported for the higher temperature components of the internal combustion engine that require the use of alloyed gray cast iron are:

Component	Operating Temperature Range, °F
Cylinder Heads	450 - 1000 (locally)
Exhaust Manifolds	300 - 1200
Exhaust Valve Seat	800 - 1300

The alloyed cast irons can be considered for components of the steam expander that operate at lower temperatures and pressures. Where strength is critical and greater reliability (safety) and oxidation resistance is required, as is encountered in high pressure steam systems, nodular cast iron is utilized. Typical nodular cast irons are SAE D5506, D4512 and D7003. SAE D5506 has a yield strength (0.2% offset) of 55,000 psi and a tensile elongation ($\sim$ 2 inches) of 6%; the structure is ferritic-pearlitic. SAE D4512 has a yield strength of 45,000 psi, a tensile elongation of 12% and has a ferritic structure. SAE D7003 has a yield strength of 70,000 psi, a tensile elongation of 3% and a pearlitic structure.

5.1.2 Cylinder Liners/Piston Rings

As stated previously, the high-speed relative motion between the cylinder liner and the power piston rings in combination with the high temperatures and pressures that exist within the cylinder chamber make these components two of the most critical components in the engine. Although the materials currently in use as cylinder liners and piston rings in gasoline and diesel fueled internal combustion engines were reviewed, the technology is not directly applicable to the reciprocating steam expanders being designed to operate with dry solid lubricants. However, cylinder liner/piston ring material combinations used in oil free air compressors are applicable and will be discussed in Section 6.0.

The materials most commonly used for cylinder liners under oil lubricated conditions are alloyed gray cast iron, the Meehanite cast irons, chromium plated gray cast iron and nitrided gray cast iron. Typical of the alloyed gray cast iron is a composition of 3.35% C - 2.15% Si - 0.65% Mn - 0.4% Cr; the liner is hardened by quenching from 1600°F into salt at 475°F followed by cooling in air. The resulting hardness is Rockwell C 45 minimum. The Meehanite cast irons are high strength, fine grained castings combining the properties of cast iron and steel. A commonly used Meehanite, Grade GA, has an ultimate strength of 50,000 psi. Liners that are chromium plated or nitrided are produced from centrifugally cast gray cast iron. The liners are centrifugal cast in order to achieve the bore surface integrity (minimum porosity) that is required for the hard chromium plating and nitriding processes. A typical centrifugally cast gray cast iron used in liners for large diesel engines has an ultimate strength of 35,000 psi and a hardness of $\sim$ 250 BHN.

Gray cast iron also is the most common material used for oil lubricated compression and oil piston rings. Occasionally AISI 1070 and 52100 bearing steels are used. However, the steel segment or steel rail types of oil rings are almost always made of AISI 1070 to 1095 steels. The chromium plating of piston rings has been found to reduce wear of the rings and cylinders by $\sim$ 75%. However, chromium cannot be run against itself so that the rings cannot be chromium plated if the liner is chromium plated. Chromium plating is used primarily on compression rings in heavy duty engines and on oil rings of the segmented type. Since chromium plate reduces the fatigue endurance limit of cast iron, higher strength cast irons must be specified for chromium plating applications. Two examples of cylinder liner/piston ring combinations are shown in Table 5.1-II.

The K-28 nodular cast iron (3.3% C - 2.2% Si - 0.5% Mo - 0.5% Cu - 0.05% Mg) is a centrifugally cast martensitic ductile iron and has a hardness of 372 - 437 BHN. The K-27 ring also is a centrifugally cast martensitic nodular cast iron with a hardness of 250 - 283 BHN. The K-6E ring is a statically cast gray iron (3.7% C - 2.85% Si - 0.45% Mo - 0.3% Cr) consisting of free graphite flake in a fine pearlite matrix and has a

Table 5.1-II

Cylinder Liner/Piston Ring Material Combinations for Heavy Duty Diesel Engines

	Piston Rings	
Cylinder Liner	Type	Material
1. Hard Chromium Plated Centrifugally Cast Gray Cast Iron	Top Compression	K-28 Nodular Cast Iron
	Bottom Seal	K-6E Gray Cast Iron
	Oil	K-Iron
2. Nitrided Centrifugally Cast Gray Iron	Top Compression	Hard Chromium Plated
	Bottom Seal	K-Iron
	Oil	K-Iron

hardness of 230 - 297 BHN. The K-iron (3.7% C - 2.65% Si) is a statically cast unalloyed gray cast iron consisting of free graphite flake in a matrix of pearlite and with an ultimate strength of 30,000 psi.

5.1.3 Piston

The pistons for automotive internal combustion engines are produced primarily from aluminum die castings. The light weight and excellent thermal conductivity of aluminum alloys made them very attractive as the trend in engines went toward higher speeds and higher compression ratios. Most pistons are made from SAE 328 (Al, 11 - 12.5% Si, 0.9% Fe, 1 - 2% Cu, 0.5 - 0.9% Mn, 0.4 - 1.0% Mg, 0.05% Ni, 1.0% Zn, 0.25% Ti) and SAE 332 (Al, 8.5 - 10.5% Si, 1.2% Fe, 2 - 4% Cu, 0.5% Mn, 0.5 - 1.5% Mg, 0.5% Ni, 1.0% Zn, 0.25% Ti). These alloys are usually heat treated (T5) to a hardness of approximately 100 BHN and are tin plated for scuff resistance.

For heavy duty engines, the piston head usually is produced from alloyed (Cr-Mo) cast iron. However, in order to achieve greater reliability, at least one manufacturer has changed to a cast Mn-Mo steel (0.20% C - 1.35% Mn - 0.5% Si - 0.25% Mo) in the production of power piston heads. The Mn-Mo steel has a minimum yield strength of 60,000 psi and a hardness of 180 BHN. In order to reduce weight of the cast steel power piston, the lower portion is constructed from a forged aluminum alloy - usually 4032 alloy (11 - 13.5% Si, 0.8 - 1.3% Mg, 0.5 - 1.3% Cu, 0.5 - 1.3% Ni) in the T-6 condition. The 4032-T6 alloy has a minimum yield strength of 42,000 psi and a hardness of 115 BHN.

One of the major reasons that the high silicon content aluminum alloys are used in the fabrication of aluminum pistons is to minimize the difference in thermal expansion between the cylinder wall material and the piston material. The coefficient of thermal expansion for 4032 alloy between RT and 572°F is reduced to 11.7×10^{-6} in./in./°F from 13.1×10^{-6} in./in./°F for unalloyed aluminum.

5.1.4 Piston Pin

Failure of piston pins rarely occurs. However, 0.001 - 0.002 inch wear of the piston pin results in a noisy engine so that selection of materials to minimize wear is very important. Piston pins are normally produced from AISI 1117 (1.15% Mn), 1016, 5115 or 5120 (0.8% Cr) steel for automotive engines and AISI 8620, 8640 (0.55% Ni - 0.5% Cr - 0.2% Mo) and 5046 (0.4% Cr) alloy steel for heavy duty engines. The low carbon steels are carburized and the higher carbon steels are case hardened by induction to a surface hardness of Rockwell C 60. Usually the surface is polished to a high finish, i.e., on the order of 2 RMS. It has been determined experimentally that the load that the piston pin can withstand without scoring is directly proportional to the surface finish - the better the finish the higher the permissible load. Under severe conditions of loading, the surface of the pin can be roughened by shot peening (followed by lapping to remove the displaced metal) to form depressions in the surface in order to better hold the lubricant.

5.1.5 Piston Pin Bearing

There are a relatively large number of metals and metal alloys that are used in sleeve bearings. The bearing alloys can be grouped in the following classes: tin-base alloys, lead-base alloys, copper-lead alloys, tin-bronze, silver, aluminum alloys, zinc-base alloys, gray cast irons, non-metallic materials (PTFE) and overlays. The proper selection of the bearing material will depend upon the type of load, i.e., steady state or cyclic, amount of load, speed, temperature, corrosive conditions, oil supply, dirt contamination, and shaft hardness.

Type of load - the load carrying capacity of cyclically loaded bearings, as in piston pin bearings, is usually determined by the bearing fatigue strength of the material. This property is of minor importance for bearings under constant load.

Amount of load - cyclic loads up to 1500 psi are considered low and can be supported by ordinary babbitt bearings (tin or lead base alloys). Thin babbitt overlays can operate up to 2500 psi. The limit of alternating loads for the copper-lead bearing material is approximately 4000 psi. A load of 5000 psi can be supported by a trilayer bearing consisting of a mild steel (AISI 1010) back, a 25% Pb-Cu alloy intermediate layer and a plated babbitt overlay. The latter bearing is used in some heavy duty engines.

Speed - babbitt alloys are best suited for high speed operation.

Temperature - strength and corrosion rates are affected by the service temperature. Certain excellant bearing materials, such as babbitts, are restricted by their low melting points. The tin-bronzes such as SAE 62 (86.7% Cu - 10% Sn - 10.3% Pb - 2.0% Zn - 1.0% Ni) and phosphor bronzes such as SAE 64 (78% Cu - 10% Sn - 10% Pb - 0.75% Zn - 0.5% Ni - 0.5% Sb) can be used as bearings at temperatures approaching 600°F. Maximum load for these materials is $\sim$ 4000 psi. Both materials have excellant fatigue properties but poor anti-seizure, conformability and embeddability properties; anti-seizure can be improved by silver plating.

Corrosion Resistance - the tin-base babbitt alloys are the most resistant to the corrosive action of acids that are formed in lubricating oils.

Oil Supply - a common cause of failure in bearings is the loss of the oil supply or breakdown in the oil film. This results in direct contact between the pin and the bearing surfaces and a large increase in friction. When this happens, the anti-seizure characteristics of the bearing are most important. The rating of materials for this property in order of decreasing anti-seizure qualities are: tin babbitt, lead babbitt, aluminum alloys, copper-lead alloys, leaded bronzes, silver and bronze.

Dirt Contamination - in bearing applications where there is a high level of contamination by particulate matter, the bearing material must possess good embeddability. Dirt can be embedded in soft babbitt materials without doing harm.

Shaft Hardness - the harder the bearing, the harder the shaft must be to prevent damage to the shaft. In general applications, soft babbitt bearings can be used satisfactorily with steels as soft as 130 - 165 BHN; for copper lead (to 25% Pb) the minimum hardness of the steel shaft must be 165 - 200 BHN; for aluminum alloys the minimum hardness of the steel shaft must be 200 - 300 BHN; and for leaded bronze the minimum hardness of the steel shaft must be 300 - 400 BHN.

A listing of recommended bearing materials for various bearing loads in internal combustion engines is given in Table 5.1-III. The materials that are currently being used for piston pin bearings in the two leading large diesel engines for locomotives are (1) a leaded tin bronze (80% Cu - 10% Sn - 10% Pb), both as a solid bushing and with a AISI 1010 steel backing (SAE 792) and (2) a tri-metal bearing consisting of a AISI 1010 steel backing, a 0.015 in. thick silver intermediate layer and a 0.0003 in. thick lead overlay. These bearings require a good finish and many axial grooves for pressurized oil lubrication. A crankshaft bearing material that is currently in use in automotive engines is a 0.030 in. thick SAE 780 (8280) aluminum alloy sheet (6% Sn - 1% Cu - 0.5% Ni - 1.5% Si) bonded to a 0.001 in. thick Ni plated AISI 1010 steel backing. A recommendation received from one of the leading aluminum producers also called for the use of a leaded tin bronze bearing for the piston pin bearing and a AISI 1010 steel backed aluminum alloy bearing for the crankshaft.

5.1.6 Connecting Rod

The high alternating loads imposed on the connecting rod in reciprocating engines require the use of relatively high strength steels of high quality. For this reason connecting rods are generally produced from contour forgings made from vacuum degassed ingots. The most common steels used for the production of connecting rods are AISI 1041 for use in automotive engines and AISI 8640 (0.55% Ni - 0.5% Cr - 0.2% Mo), 4140 (0.95% Cr - 0.2% Mo) and 4340 (1.8% Ni - 0.8% Cr - 0.25% Mo) for heavy duty and diesel engines. One automotive engine manufacturer utilizes SAE 80002 malleable iron castings for the production of connecting rods.

5.1.7 Camshaft/Cam/Cam Follower (Tappet)

The cyclic loading and high localized surface stresses that occur as a result of the action between the cam surface and cam follower (tappet) makes the materials selection for these components very important. In some engines, the localized Hertzian stress on the cam surface approaches 200,000 psi. Numerous combinations of camshaft/tappet materials are in common use. Hardenable gray cast iron, such as SAE G4000 is most widely used for automotive camshafts. A typical composition is 3.3% C - 2.25% Si - 0.6% Mn - 0.95% Cr - 0.5% Mo. As cast, the camshaft has a hardness of

Table 5.1-III

Bearing Materials Used in Internal Combustion Engines

Bearing Load, psi	Lowest Cost Material	Alternate Materials	Advantages of Alternate Materials
		(Small and Medium Size Engines)	
1200 max	Pb Babbitt on steel (0.015-0.030 in.)	Zr or cast Al alloy	(a) (b) (c)
		Sn Babbitt on steel (0.015-0.030 in.)	(d)
1200 - 2200	Pb Microbabbitt on steel (0.002-0.005 in.)	Babbitt-impregnated sintered Cu-Ni on steel	(a) (b) (d)
		Babbitt-impregnated sintered bronze on steel	(a) (b) (d)
		Sn Microbabbitt on steel	(d)
		Al alloy on steel or solid	(a) (b) (d)
2200 - 2800	Babbitt-impregnated sintered Cu-Ni on steel	Cu-35% Pb on steel	(a) (b)
	Babbitt-impregnated sintered bronze on steel	Solid Al alloy or Al alloy on steel	(a) (b) (d)
	Cu-25% Pb on steel	Above materials + 0.001 in. Pb alloy overlay	(b) (e)
	Al alloy on steel	Cu-25% Pb on steel + 0.001 in. Pb alloy overlay	(d) (e) (f) (g)
		Al alloy on steel + 0.001 in. Pb alloy overlay	(d) (e) (f) (g)
		(Large Engines)	
1200 max	Pb Babbitt on steel or bronze (0.015-0.030 in.)	Sn Babbitt on steel or bronze	(d)
		Solid Al alloy	(c) (d)

Table 5.1-III (Cont'd.)

Bearing Materials Used in Internal Combustion Engines

Bearing Load, psi	Lowest Cost Material	Alternate Materials	Advantages of Alternate Materials
		(Large Engines)	
1200 - 1500	Pb Babbitt on steel or bronze (0.020 in. max)	Pb Microbabbitt on steel or bronze	(a) (b)
		Aolid Al alloy	(c) (d)
1500 - 2000	Pb Microbabbitt on steel or bronze (0.002-0.005 in.)	Cu-Pb (30-40% Pb) on steel	(a) (b)
		Solid Al alloy or Al alloy on steel	(c) (d)
		Above materials + 0.001 in. Pb alloy overlay	(b) (e)
2000 min.	Solid Al alloy	Cu-25% Pb on steel + 0.0005-0.002 in. Pb alloy overlay	(a) (b) (c)
		Al alloy on steel + 0.0005-0.002 in. Pb alloy overlay	(d) (g)

(a) Yield strength
(b) Fatigue strength
(c) Possible cost savings
(d) Corrosion resistance
(e) Conformability
(f) Embeddability
(g) Inherent lubricity

248 - 311 BHN; the cam surfaces are flame or induction hardened to a Rockwell C hardness of 54 min. For heavy duty engines the camshafts are forged and are usually produced from water quenched carbon steels and low alloy steels of 0.5 - 0.7% C, AISI 4340, or a curburizing grade such as AISI 8620 steel. Heat treatment of the high carbon steels to achieve the necessary hardness of the cam surfaces is accomplished by either conventional through-hardening or case hardening processes using flame or induction. One diesel engine manufacturer uses a forged AISI 1080 shaft which is quenched and tempered to a hardness of 235 BHN. The cam and bearing surfaces are induction case hardened to a hardness of Rockwell C 60 - 65 to a depth of 0.060 in. Care must be taken to insure sufficient depth of case as thin, brittle surface hardened cases tend to spall in service.

Tappets usually fail by scuffing or rapid loss of surface. The most common automotive tappet material is a gray cast iron of a composition similar to 3.2% C - 2.25% Si - 0.8% Mn - 1.1% Cr - 0.6% Mo - 0.55% Ni. The cast iron tappet is hardened by quenching in oil from 1550°F and tempered to a hardness of Rockwell C 55 - 60. Steel tappets also are used and are produced from hardenable steel such as 52100 bearing steel or high-carbon molybdenum steels of the AISI 4000 series or from carburizing grades of steel such as AISI 51200 or 8620.

In some heavy duty engines, where tappet wear has been a severe problem, it was found that the utilization of material containing varying amounts of carbides in the microstructure improved the service life. In one case the successful use of hardened and tempered D-2 tool steel in solving the wear problem was attributed to free carbides in the microstructure. The D-2 alloy has a high carbon content in conjunction with stable carbide formers. The composition is 1.5% C - 12% Cr - 0.8% V - 1.0% Mo. A higher volume carbide content in the microstructure can be obtained with alloys such as Stellite Star J (2.5% C - 32.5% Cr - 17.5% W - Bal Co) and this material at Rockwell C 60 hardness is being used for tappets in some heavy duty engines. Experience has shown that the Star J alloy shows superior performance against a nitrogen rich surface in comparison to a carburized surface. It should be noted that if a nitrided surface is to be used for the cam, a carbonitride process should be utilized in order to obtain the desired case depth to carry the high loads. The core depth of a straight nitrided steel surface is too thin for application on cam or tappet surfaces. A solid tungsten carbide cermet also is being considered for use as a tappet material. In some very heavily loaded cam/tappet designs a roller cam follower is employed. An AISI 8620 steel carburized to a surface hardness of Rockwell C 55 min. and a case depth of 0.060 inch has performed satisfactorily in roller type cam followers. Again a surface finish of 2 - 4 RMS is recommended.

Tappet faces are usually finished to about 6 RMS for the same reasons a good finish is required for the piston pin, i.e., improved load carrying ability. Shot peening of the tappet surface is sometimes done to improve the retention of the lubricant. Coating of the highly finished tappet surface to aid in the wearing-in process and improve frictional characteristics is common practice. An oxide coating (Fe_3O_4) is generally applied to chilled cast iron and a phosphate coating is generally applied to hardened steels or gray cast irons. The phosphates can be manganese phosphate, zinc phosphate or iron-manganese phosphate. Other coatings that have been used are oxidates, manganates, and sulfides.

The following material combinations were considered as candidates for the cam and tappet in the single cylinder engines:

	Cam	Tappet
1.	AISI 8620/carburized (Rc 58 min.)	Chilled Cast Iron (Rc 54) Ferrox Coating (Fe_3O_4)
2.	AISI 8620/carburized (Rc 58 min.)	Carboloy 883 (WC + 6% Co) (R_A 92)
3.	AISI 8620/ carbonitrided (Rc 60-65)	Star J (Rc 61)
4.	Hardenable Cast Iron (Rc 54 min.)	Hardenable Cast Iron (Rc 55-60)/Mn-Fe Phosphate Coating
5.	Hardenable Cast Iron (Rc 54 min.)	D-2 Tool Steel (Rc 58-60)/ Mn-Fe Phosphate Coating

Bearing materials used for camshafts in heavy duty diesel engines are generally cast aluminum alloys. Aluminum alloy 750 (6.25% Sn - 1.0% Ni - 1.0% Cu) is commonly used.

5.1.8 Inlet Valve

Operating conditions for valves in large central steam power stations vary considerably from the operating conditions of valves for automotive and diesel internal combustion engines. The operating conditions for steam valves for central power stations are much less severe than for the valves in internal combustion engines. Actuation of many steam valves is on an infrequent basis so that fatigue and wear of the valve face and seat is not a severe problem. Service temperatures of the steam valve are significantly lower than those for the exhaust valve of an internal combustion engine and they do not fluctuate as they do in the internal combustion engine; thus creep and thermal fatigue problems are less severe. In addition, corrosion as a result of high temperature combustion products is not a problem in steam valves.

In summary, the years of experience in the operation of steam valves or exhaust valves in internal combustion engines are not directly applicable to the selection of materials for inlet steam valves in reciprocating steam expanders. Selection of materials for the various components of the steam inlet will have to be based on the knowledge of the basic properties of the materials with respect to corrosion in steam, mechanical properties and wear behavior under conditions of no lubrication and as supplemented with valve experience in reciprocating internal combustion engines.

Inlet steam valves in large central power steam systems are generally fabricated from a martensitic 12% Cr stainless steel (AISI 410) or Crucible 422 alloy (12% Cr - 1% Mo - 1% W - 0.8% Ni - 0.25% V); the stem and face are nitrided. Both the valve seat and valve guide are made from Stellite 6B. The Stellite 6B on the valve seat usually is applied as a weld overlay on a low alloy steel (2.25% Cr - 1.0% Mo alloy).

The materials in use in valve components for internal combustion engines will vary considerably with the operating temperature of the valve. The lower strength and increased corrosion and wear rates that accompany increased service temperatures necessitates the use of more highly alloyed materials as the temperature of the valve increases. Since the inlet valves in internal combustion engines operate at temperature under 600°F and exhaust valves operate at temperatures as high as 1550°F, it is obvious that the most severe materials problems are associated with the exhaust valve. The development of improved valve materials over the years has been primarily by empirical methods. This stems from the fact that it is not possible to simultaneously reproduce all of the service conditions of the valve by any other means than in an actual engine.

Materials that are recommended for use in inlet and exhaust valves in various types of engines are listed in Table 5.1-IV. The steels in Group A of Table 5.1-IV are used for light duty inlet valves that operate at low temperature or for short times. They also are used for stem materials in two-piece valve construction where they can be welded to higher alloy heads and used for heavier duty inlet and exhaust valves.

Steels in Group B through E are especially made for valve applications with the Group B steels being the least expensive and the Group E steels being the most expensive. The sigma phase forming steels in Group D have good hot hardness and superior resistance to wear than the austenitic steels of Group E. However, they are more brittle and have less resistance to creep than the Group E steels. The nickel-base super alloys listed in Group F are only used where valve temperatures are very high because of their high cost.

Short operating lives of valves can usually be traced to permanent dimensional changes that take place in the valve components during service as a result of inadequate creep strength or poor wear and/or hot corrosion characteristics. The terms used to describe the dimensional changes are:

a. elongation,
b. projection of the stem,
c. face runout, and
d. tip recession.

Table 5.1-IV

Materials Used for Inlet and Exhaust Valves in Internal Combustion Engines

Application	Valve Materials	Approx. Max. Exhaust Valve Temp. °F
A. Intake Valve-Light Duty	Carbon Steel	-
	SAE NV-1 (AISI 1041)	
	(a) SAE NV-2 (AISI 1047)	
	(AISI 1050)	
	Low Alloy Steels	
	SAE NV-4 (AISI 3140)	
	(AISI 4150)	
	SAE NV-6 (AISI 5150)	
	(AISI 6145)	
	SAE NV-5 (AISI 8645)	
B. Intake Valve-Heavy Duty	Martensitic Steels	-
	SAE HNV-2 (Sil F) 4 Si-2.25 Cr (0.4C)	
	4 Si-2.25 Cr 1.5 Ni-0.85 Mo (0.4C)	
C. Intake Valve-Heavy Duty Exhaust Valve-Light Duty	Martensitic Steels	1350
	(b) SAE HNV-3 (Sil 1) 3.25 Si-8.5 Cr (0.45C)	
	2.75 Si-7.5 Cr-1.5 Ni-0.85 Mo (0.3C)	
	(b) SAE HNV-6 (XB) 2.25 Si-20.0 Cr-1.5 Ni (0.8C)	
D. Exhaust Valve-Heavy Duty	Austenitic-Sigma Phase Alloys	1550
	SAE EV-1 (SCR) 23.5 Cr-4.75 Ni-2.75 Mo (0.45C)	
	SAE EV-2 (TXCR) 24.0 Cr-3.75 Ni-1.35 Mo-3.75 Mn (0.4C)	

Table 5.1-IV (Cont'd.)

Materials Used for Inlet and Exhaust Valves in Internal Combustion Engines

Application	Valve Materials	Approx. Max. Exhaust Valve Temp. °F
E. Exhaust Valve-Heavy Duty	Austenitic Steel Alloys	1550
	SAE EV-7 (2155N) 21 Cr-5 Ni-5.5 Mn (0.2C, 0.25N)	
	21 Cr-4 Ni-7 Mn (0.4C, 0.1N, 0.22P)	
	(a) SAE EV-8 (21-4N) 21 Cr-4 Ni-9 Mn (0.4C, 0.4N)	
	SAE EV-5 (Sil 10) 19 Cr-8 Ni-3 Si (0.4C)	
	SAE EV-4 (21-12N) 21 Cr-12 Ni (0.2C, 0.2N)	
	(Cast) 25 Cr-12 Ni (0.2C)	
	SAE EV-9 (TPA) 14 Cr-14 Ni-2.4 W-0.35 Mo (0.45C)	
	(Cast) 15 Cr-15 Ni-3.5 Si-0.4 Mo (1.0C, 0.25 Cu)	
F. Exhaust Valve-Heavy Duty	Austenitic Ni Alloys	1650
	SAE HEV-2 (Inconel M) 16 Cr-Bal Ni-3 Ti-0.5 Al (0.03C)	
	SAE HEV-3 (Inconel X-750) 15 Cr-Bal Ni-1 Cb-2.5 Ti-0.9 Al (0.04C)	
	(c) SAE HEV-5 (Nimonic 80A) 20 Cr-Bal Ni-2.5 Ti-1.2 Al (0.05C)	
	SAE HEV-6 (Nimonic 90) 20 Cr-Bal Ni-18 Co-2.5 Ti-2.2 Al (0.05C)	

(a) Used in current automotive engines

(b) Used in current high performance automotive engines - intake valves

(c) Used in current heavy duty engines

Elongation ("wire drawing") is the permanent increase in length as measured from the valve face to the tip. This causes a decrease in the "lash" (clearance between valve tip and tappet) so that the valve face does not seat properly and results in "blowby" of the exhaust gases. This can be corrected by using a material with greater resistance to creep. Face runout is caused by hot corrosion which also allows leakage of the exhaust gases. Stem projection is similar to elongation in that it reduces "lash"; it is the total elongation as measured through the valve guide and includes any wear of the seat and valve face and elongation of the stem. Tip recession results in an increase "lash" and leads to fracture of the valve. Face runout and stem projection, due to corrosion and wear of the seat and valve face and tip recession, can usually be corrected by appropriate use of valve seat inserts and/or corrosion resistance and hard facing materials. Commonly used insert and facing materials are listed in Table 5.1-V. The Group D and E materials are used most often with the cobalt base alloys, Group E, are preferred for improved resistance to corrosion.

Examples of material combinations used in exhaust valve components for current automotive and large diesel internal combustion engines are given in Table 5.1-VI.

5.2 Expander Materials Selection Study

The structural materials used in the containment of high temperature (1000° - 1050°F) and high pressure (2400-3500 psi) steam as well as materials used for specialized components such as steam valves have been studied in depth for years and their properties are well documented. Similarily, materials in use in heavy duty and high performance reciprocating engine components are well established and most of these materials are adaptable to use in corresponding components of the reciprocating steam expander. For this reason, there was no need to conduct experimental investigations of candidate expander materials for this program.

Table 5.1-V

Materials Used for Valve Inserts and Hard Facing Applications

Material	Nominal Composition, % C	Mn	Si	Cr	Ni	Mo	W	Co	SAE Alloy Designation
A. Cr-Mo and Cr-Mo-W Steels	0.65	0.6	1.00	3.0	-	5.00	-	-	
(Inserts)	(a) 1.00	0.6	2.50	4.0	-	8.50	-	-	
	1.00	0.6	0.25	5.0	-	1.25	-	0.25(V)	
	0.65	0.35	0.25	5.0	-	3.00	3.5	1.00(V)	
	1.35	0.35	0.45	3.5	1.0	6.50	5.5	0.20(Cu)	
B. Cr-Mo Cast Iron	2.50	0.60	2.00	3.0	-	5.00	-	-	
(Inserts)	2.25	1.00	1.00	2.5	-	7.50	-	-	
C. W Steels	0.50	-	4.00	-	-	-	9.5	-	
(Inserts)	0.55	-	0.25	3.5	-	-	13.0	-	HNV-7
D. Ni Alloys	2.00	0.40	0.30	25.0	60.0	-	8.5	-	VF-4 (X-782)
(Facings)	0.20	1.00	1.00	19.5	78.0	-	-	-	VF-1 (80-20 NiCr)
E. Co-Cr-W-Ni Alloys	1.00	-	-	28.0	-	-	4.0	67.0	
(Facings)	(b) 1.25	-	-	28.0	3.0	-	4.5	55.0	VF-2 (Stellite 6)
	1.60	-	-	24.0	24.0	-	12.5	37.0	VF-5 (Stellite F)
	2.50	-	-	30.0	-	-	11.0	52.0	
	(c) 2.40	-	-	29.0	39.0	-	15.0	10.0	VF-3 (Eatonite)

(a) Light duty on large gasoline and diesel engines.

(b) Strongest and toughest of the Co base alloys.

(c) High hot hardness and resistance to shock and pitting.

Table 5.1-VI

Material Combinations Used in Exhaust Valve Components for Internal Combustion Engines

Exhaust Valve Component	Material[a] Light Duty Spark Ignition	Heavy Duty Spark Ignition	Heavy Duty Diesel
Head	SAE EV-8 (21-4N[b]	SAE HEV-5 (Nimonic) 80A[b]	Inconel 751[c]
Face (overlay)	SAE VF-5 (Stellite F)	SAE VF-3 (Eatonite)	SAE VF-2 (Stellite 6)
Stem	SAE EV-8 (21-4N)[b]	SAE HEV-5 (Nimonic) 80A[b]	AISI 3140[c]
Seat (Insert)	(Eatonite or L.E. Jones Alloy)		AISI 410 SS (Re 40-49) Weld overlay
Guide	←(Gray Cast Iron)→		Gray Cast Iron (Type A Flake in Pearlite)

(a) Material compositions are listed in Tables 5.1-IV and 5.1-V

(b) One piece construction.

(c) Two piece construction - flash welded.

However, prior to the selection of the structural materials for the two single cylinder steam expanders, their strength properties and compatibility in a steam-air environment were reviewed. In addition, material and process specifications were established for each component in the steam expanders.

5.2.1 Material Properties

5.2.1.1 Low Carbon and Low Alloy Steels

The use of any one steel for structural applications and containment of high pressure steam primarily depends on the sustained metal temperatures. Low carbon steel is used for steam generation tubes and the low temperature regions of the superheater where the metal temperatures do not exceed 750° - 800°F. As the temperatures in the steam generating system increases, steels of higher alloy content are employed. The C-0.5% Mo steel can be used up to about 850°F; the low chromium-molybdenum steels (1.0 - 2.25% Cr) up to 1000° - 1050°F; the intermediate chromium-molybdenum steels (3 - 9% Cr) up to 1100°F; and the austenitic stainless steels above 1100°F.

The temperature limits imposed on the various steels are the result of loss in elevated temperature strength (yield and creep), structural changes in the microstructure of the steel that may detrimentally affect the properties, i.e., ductility, and oxidation-steam corrosion considerations. The maximum allowable design stresses for the low carbon, low alloy and austenitic stainless steels used in boiler construction as a function of temperature are shown in Figure 5.2-1. The maximum allowable stress is based on the ASME Boiler and Pressure Vessel Code which limits the allowable stress to the level at which the creep rate is 0.01% in 1000 hours at the design metal temperature. Additional restrictions on the allowable stresses are imposed by the code and they are based on the stress-rupture strength of the material as modified by suitable allowances for loss in metal cross-section due to oxidation and corrosion. From Figure 5.2-1, the allowable stress for low carbon steels begins to drop above 700°F and at 850°F is only half the value at room temperature. The maximum allowable stresses for the low alloy Cr-Mo steels are sustained up to about 800°F whereupon they start to drop and reach 50% of the room

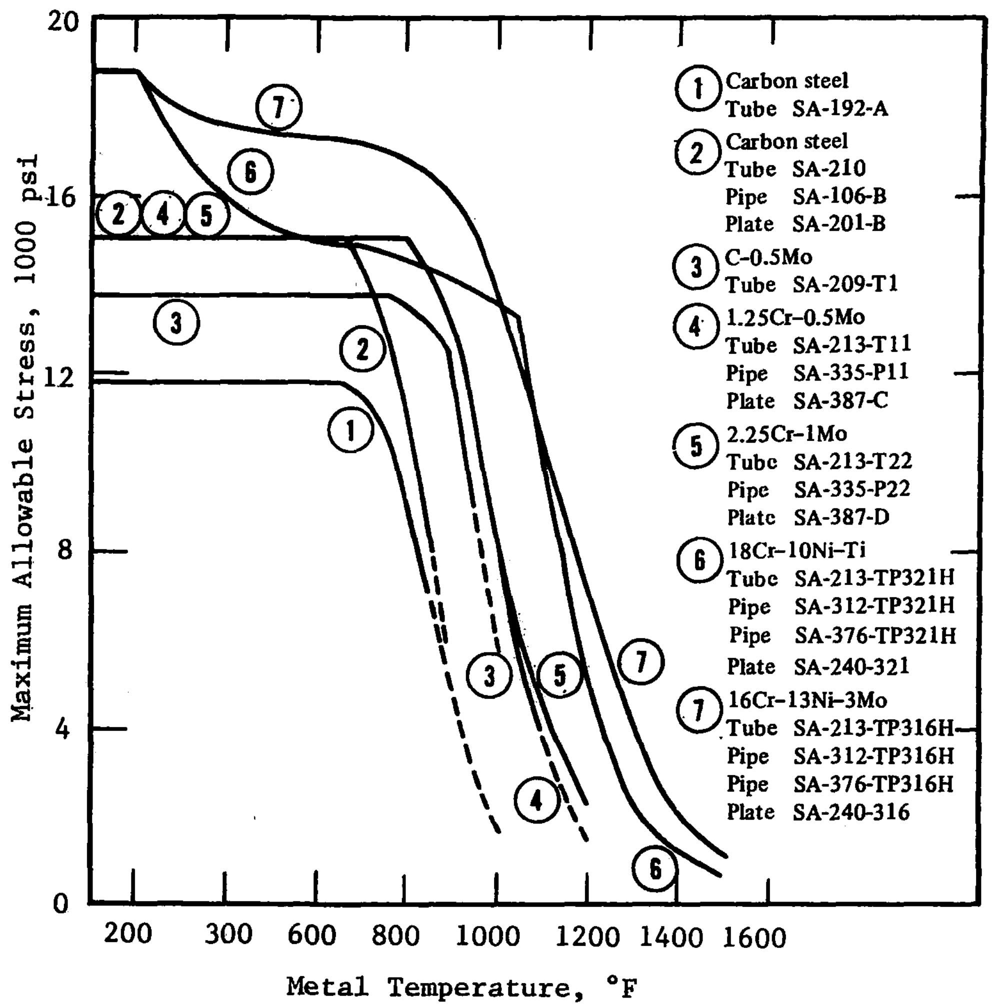


Figure 5.2-1. Maximum Allowable Design Stresses for Low Carbon, Low Alloy and Austenitic Stainless Steels in Large Steam Generating Systems. Reference 14

temperature values at 1000°F. The 50% loss in the maximum allowable design stresses for the austenitic stainless steels occurs at about 1200°F. Of the low alloy, Cr-Mo steels, the 1.25% Cr-0.5% Mo alloy has superior creep rupture properties at temperatures on the order of 1000°F, Table 5.2-I As chromium is added to achieve superior oxidation resistance, the creep-rupture strengths are reduced. Care must be exercised in the fabrication of components from these steels, i.e., amount of cold work, heat treatment, etc., such that grain size and phase morphology are not adversely affected with respect to their influence on strength and ductility.

The thermal stability of the low alloy steels also restricts the temperature at which the steels can be used. Phase changes, such as the graphitization of carbon steels above 750°F and C-Mo steels above 850°F, can result in drastic reduction in strength and ductility of the steel. Chromium additions of more than 0.5% in the steel eliminate this problem.

Compatibility with the surrounding environment is another important consideration in the selection of the structural material. In the steam expander, as is the case for the boiler, the exterior surfaces of the containment materials are subject to oxidation by oxygen in the air and the internal surfaces are oxidized by the oxygen in the steam. Each steel has a threshold temperature above which rapid oxidation or corrosion takes place generally due to the formation of a thick porous scale. As a result the load carrying ability is reduced and, in the case of boiler tubes, restricted flow and reduced heat transfer efficiency occurs. Sufficient oxidation and corrosion resistance can be achieved in the steels by the addition of small amounts of chromium usually in excess of 1%. The chromium in the steel improves the oxidation resistance by promoting the formation of a tightly adhering scale which inhibits further oxidation. The degree to which chromium additions improve the oxidation resistance of low alloy steels in steam can be seen in Figures 5.2-2[14] and 5.2-3[15]. In Figure 5.2-2, the data are plotted as penetration in mils/yr. vs. temperature and in Figure 5.2-3, the data are plotted as penetration in mils as function of time at 1100°F. It is important to point out that these data (Figure 5.2-2) were obtained under actual service conditions from samples periodically cut from a large steam plant (Detroit Edison) and show higher corrosion rates than in steady state laboratory tests by a factor of ∿ 2.

Table 5.2-I

Creep Rupture Properties of Cr-Mo Alloy Steels at 1000°F[15]

Alloy[a]	Stress (ksi) to Produce a Creep Rate of 1% in 10,000 hours	Stress for Rupture, ksi	
		1000 hrs.	10,000 hrs.
1 Cr - 0.5 Mo	11.8	28.4	18.8
1.25 Cr - 0.5 Mo	12.0	27.0	21.0
2.25 Cr - 1.0 Mo	12.0	20.4	15.4
3 Cr - 1.0 Mo	10.5	18.0	14.7
5 Cr - 0.5 Mo	9.0	19.0	14.6
7 Cr - 0.5 Mo	8.0	18.8	14.0
9 Cr - 1.0 Mo	12.0	22.8	19.1

[a] Annealed condition.

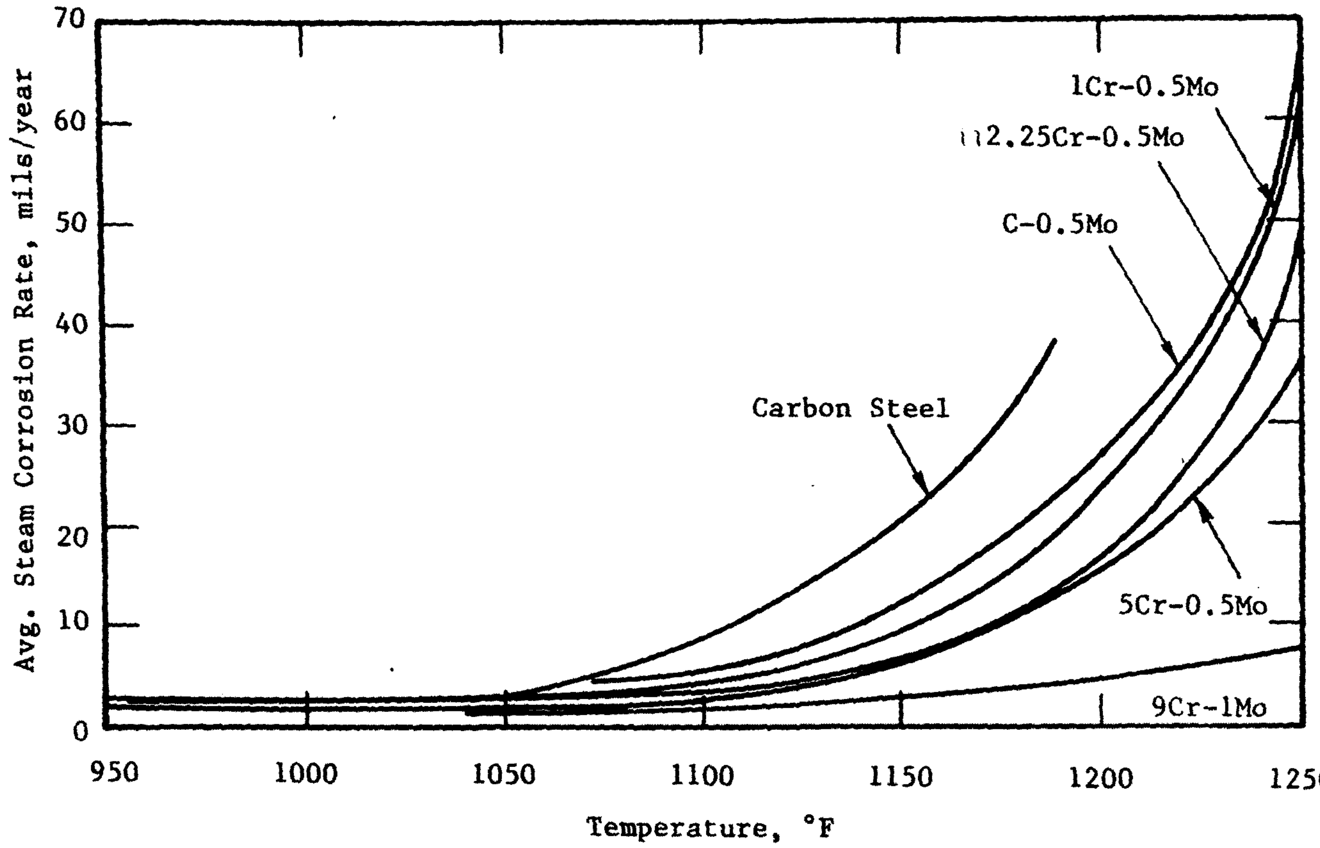


Figure 5.2-2. Effect of Temperature on Long Time Steam Corrosion Rates.

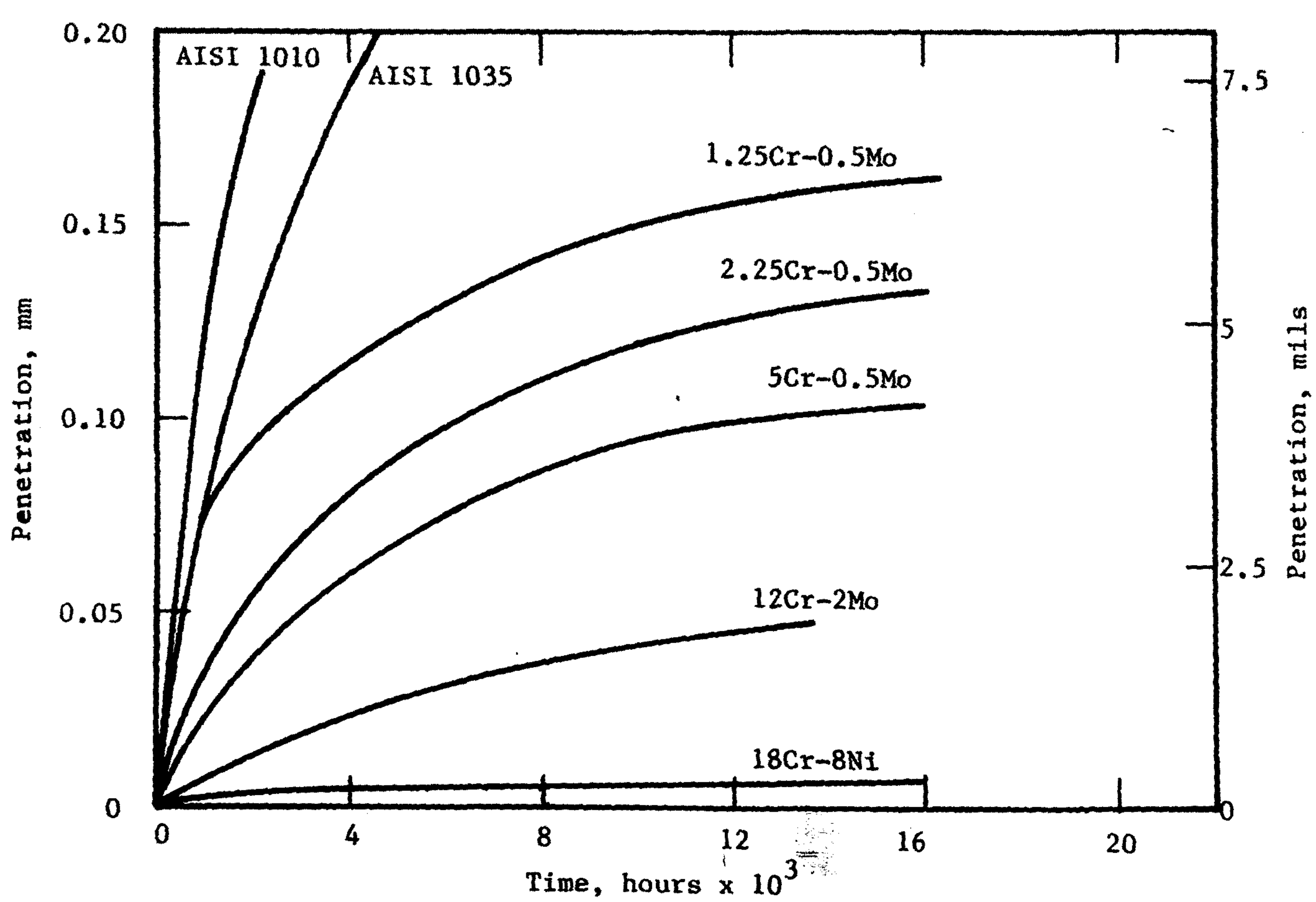


Figure 5.2-3. Effect of Long Time Exposure to Steam at 1100°F.

Similar long time oxidation-corrosion tests in full size commercial tubing (2 in. OD x 0.5 in. wall) were conducted at the Philip Sporn plant of the American Electric Power Company. In these tests, scaling data were obtained on the internal and external surfaces after varying periods of exposure at 1100°, 1200°, 1350°, 1500°F and 2000 psi steam pressure. Data for the 1100°F exposures are given in Table 5.2-II. There is little difference in scale thickness between the two surfaces[17].

5.2.1.2 Medium Carbon Low Alloy Steels

A number of dynamic components in the steam expander require the use of high quality relatively high strength materials. For these components, i.e., power piston head, piston rod, connecting rod and bolts, the following medium-carbon, low alloy steels were reviewed:

	Nominal Composition, %				
Alloy	C	Ni	Cr	Mo	V
AISI 4140	0.4	-	1.0	0.2	-
AISI 4340	0.4	1.8	0.3	0.25	-
17-22-A	0.45	-	1.0	0.55	0.3
H-11	0.35	-	5.0	1.5	0.4

The 0.2% yield strength of these alloys are compared in Figure 5.2-4. H-11 steel has an exceptional combination of high strength and toughness and is used for critical and highly stressed components. When the alternating stress endurance strength is the limiting criteria, vacuum melted grades of the steels are specified. For example, the endurance limits for vacuum melted and air melted AISI 4340 alloy steel heat treated to a strength level of 200,000 psi are 105,000 psi and 90,000 psi, respectively, at room temperature.

The oxidation-steam corrosion characteristics of the medium-carbon, low alloy Cr-Mo-V steels are similar to the low-carbon, Cr-Mo alloy steels. Data for a Cr-Mo-V steel are given in Table 5.2-II.

5.2.1.3 Cast Iron

For reasons of cost and in certain wear applications, cast iron should be specified wherever possible. Cast iron grades usually are

Table 5.2-II

Oxidation-Corrosion of Cr-Mo Steels at 1100°F[a]

Alloy[b]	Exposure time, months	Scale Thickness, mils Steam Side	Air Side
Cr-Mo-V[c]	6	←7.2 total→	
	12	3.3	4.0
	18	4.0	5.0
2.25 Cr - 1.0 Mo	6	2.9	2.9
	12	3.6	3.4
	18	4.1	4.6
	24	5.5	5.7
3 Cr - 1 Mo	6	2.5	2.5
	12	4.8	3.4
	18	4.1	4.2
5 Cr - 0.5 Mo	6	2.8	3.2
	12	3.9	3.5
	18	3.5	2.8
	24	5.5	4.6
9 Cr - 1 Mo	6	2.4	2.1
	12	3.4	3.7
	18	3.8	4.0
	24	5.0	5.4

[a] Steam pressure 2000 psi.

[b] 2 in. OD x 0.5 in. wall tube.

[c] Cr-Mo-V Steel, 0.11 C - 1.0 Cr - 0.86 Mo - 0.22 V.

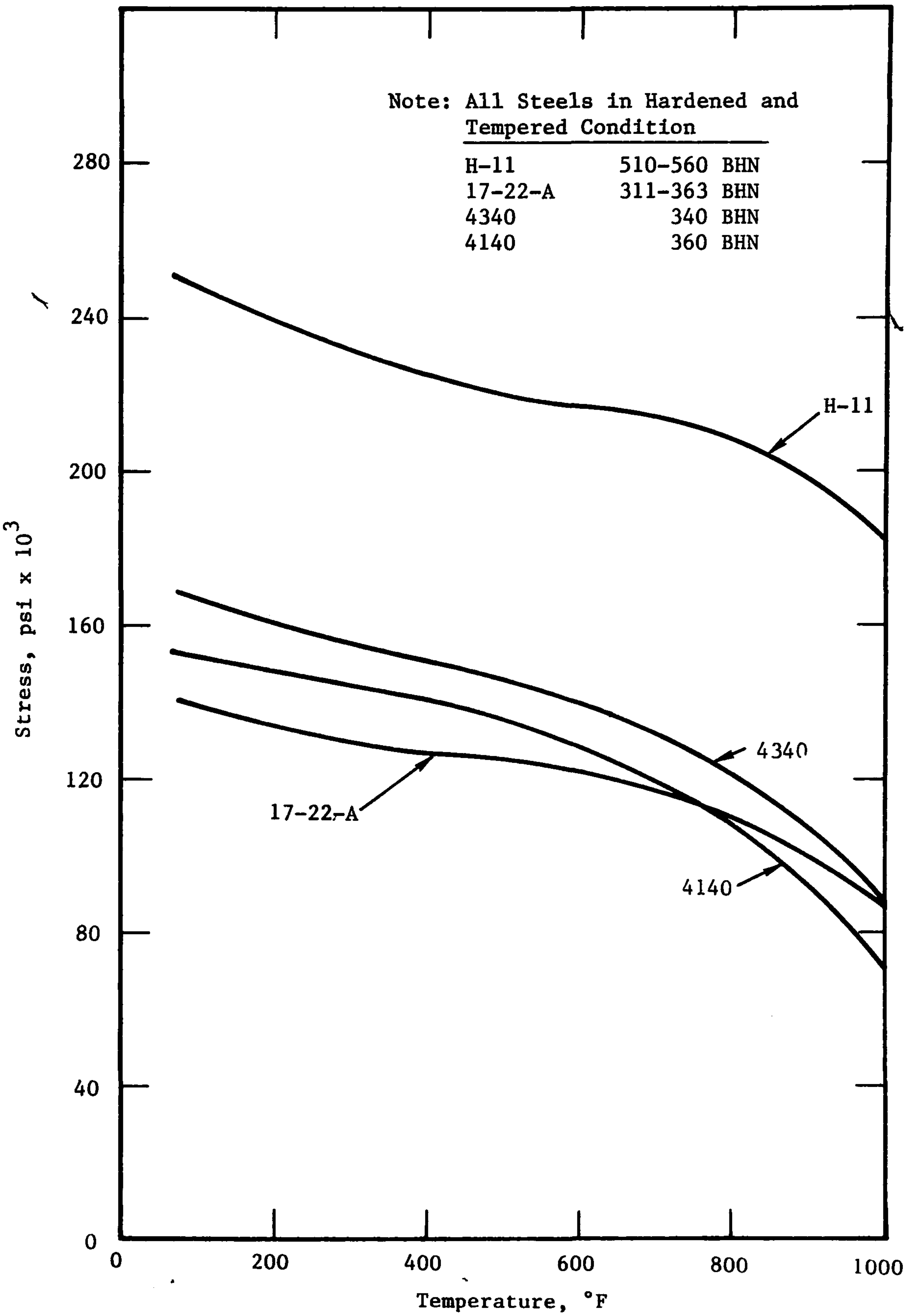


Figure 5.2-4. 0.2% Yield Strength Medium Carbon Low Alloy Steels.

specified to a specific strength level rather than to a chemical composition, i.e., SAE G4000, and numerous grades of cast iron are available. The ultimate strengths and endurance limits of gray cast iron generally are not affected by temperature until above 800°F. For example, a 2.84C - 1.5 Si gray cast iron with an ultimate strength of 48,400 psi has a fatigue endurance limit of 20,000 psi. With the exception of a slight dip in strength levels at the intermediate temperatures, to 42,000 psi ultimate strength and 18,000 psi endurance limit, the initial strength levels are retained to a temperature of approximately 800°F after which they decrease to corresponding levels of 35,000 psi and 14,000 psi. Notched fatigue strength usually are within 10% - 20% of the unnotched strength level.

Although improvements in the high temperature strength of gray cast irons are possible by alloying with molybednum, where reliability is required in critical components of high pressure systems, the use of nodular cast iron is preferred over gray cast iron. Nodular cast iron can be made to significantly higher strength levels than gray cast iron in addition to being able to deform plastically. Measurable tensile elongation values of 3 - 30% are possible with the nodular irons. They also have good resistance to mechanical shock. Typical room temperature properties for a 80-60-03 Pearlitic Grade of nodular iron are as follows:

			Endurnace Limit, ksi[a]	
Ultimate, psi	Yield, psi	Elong., %	Unnotched	Notched
95-130	55-80	3-9	40	24

[a] For casting with 100 psi ultimate strength.

One of the major concerns in the use of cast irons in steam (or air) at elevated temperatures is their dimensional instability due to oxidation. This growth in dimensions can be significantly large in gray cast iron because of the continuous network of graphite. However, additions of chromium and the use of nodular cast iron or, for more severe conditions of corrosion, the use of the Ni-Resist cast irons can eliminate or minimize the dimensional changes due to oxidation. Data on dimensional growth for various cast irons as a result of exposure to steam at 900°F are given in Table 5.2-III.

Table 5.2-III

Dimensional Growth of Cast Irons in High Temperature Steam[18,19]

Cast Iron	Growth at 900°F, in./in. 500 Hrs.	1000 Hrs.	2500 Hrs.
Gray Cast Iron	0.0023	0.0052	0.014
Gray Cast Iron + 0.46% Cr	-	-	0.001[a]
Gray Cast Iron + 0.6% Cr	-	-	0.0005[a]
Ni-Resist Type 2[b]	0.0005	0.0010	0.0015
Ni-Resist Type 3[b]	0.0003	0.00045	0.00048
Ni-Resist Type 2D[b] (Nodular)	0.0003	0.0005	0.0005
Ni-Resist Type 3D[b] (Nodular)	0.0003	0.0000	0.0000

[a] 1000°F/2000 hours.

[b] Ni-Resist Type 2, 2D: 0.0C max-2.25 Si-20 Ni-2.1 Cr.
Ni-Resist Type 3, 3D: 2.6C max-2.1 Si-30 Ni-3.0 Cr.

5.2.1.4 Aluminum Alloys

The use of aluminum alloys for dynamic components and cylinder walls in internal combustion engines is highly desirable because of the low density and high thermal conductivity of aluminum alloys. However, the calculated temperatures in the crown of the power piston head, particularly for the crosshead piston expander, and the calculated stresses in the connecting rods and the piston rod of the crosshead piston expander precludes the use of aluminum alloys for these applications. At this stage in the expander designs, only the skirt portion of the trunk piston and the crosshead piston itself warrant consideration. For these applications, it is desirable to utilize alloys with as low a coefficient of thermal expansion as possible that is consistent with suitable mechanical properties.

As previously stated, the following alloys (high silicon contents) are used in piston heads and skirts in automotive spark ignition and large diesel engines: SAE 328, SAE 332 and 4032. The nominal compositions and the coefficients of thermal expansion are given in Table 5.2-IV. Two other high strength forging alloys (2014 and 2219) that have been recommended for high temperature engine components also are listed. For comparative purposes, the room temperature minimum tensile properties of these alloys are given in Table 5.2-V.

Typical properties of the forged 4032 alloy that was selected for the piston components in this program together with typical properties for 2014 and 2219 alloys are listed in Table 5.2-VI. From these data it is apparent that the 4032 and 2014 alloys have poor thermal stability at the higher temperatures in contrast to the excellant thermal stability of 2219 alloy. Significant degradation in the tensile properties is not observed in the 2219 alloy until a temperature in excess of 600°F is reached or after very long time exposures at 600°F (10,000 hrs.).

Careful consideration must be given to the use of aluminum alloys in steam. Although aluminum alloys should perform satisfactorily in dry steam at elevated temperatures, i.e., 600° - 700°F, erosion may be a problem in wet steam. Further, it is known that conventional commercial aluminum alloys will corrode very rapidly in high purity water at temperatures over 400°F. However, aluminum alloys have been used as cladding

Table 5.2-IV

Nominal Compositions of Aluminum Alloys

Alloy	Nominal Composition, %									Coef. Expansion
	Si	Fe	Cu	Mn	Mg	Ni	Zn	Ti	Other	in./in./°F x 10^{-6}
SAE 328[a]	11.75	0.9	1.5	0.7	0.7	0.0005	1.0	0.25	-	11.5 (68-392°F)
SAE 332[a]	9.5	1.2	3.0	0.5	1.0	0.5	1.0	0.25	-	12.0 (68-392°F)
4032[b]	12.25	1.0[c]	0.9	-	1.1	0.9	0.25[c]	-	0.1 Cr[c]	11.7 (68-572°F)
2014[b]	0.8	-	4.4	0.8	0.4	-	-	-	-	13.6 (68-572°F)
2219[b]	0.2[c]	0.3[c]	6.3	0.3	0.02[c]	-	0.1[c]	0.06	0.1 V 0.18 Zr	13.6 (68-572°F)

[a] Casting - automotive engines.

[b] Forging - large diesel engines.

[c] Maximum.

Table 5.2-V

Room Temperature Tensile Properties of High Silicon Aluminum Alloys

Alloy	Condition	Temper	Minimum Tensile Properties Ultimate, psi	0.2% Yield, psi	Elong., %
SAE 328	Cast	T5	32	26	low
		T65	42	37	low
SAE 332	Cast	T5	31	-	low
4032	Forged	T6	52	42	5
2014	Forged	T6	65	55	10

Table 5.2-VI

Typical Mechanical Properties for Aluminum Alloy Forgings
(4032, 2014 and 2219 in T6 Condition)

Alloy	Test Temp., °F	Exposure Time Prior to Test, hours	Tensile Properties			Young's Modulus ksi x 10^3	Fatigue Properties	
			Ult., ksi	Yield, ksi	Elong., %(4D)		Cycles	Stress, ksi
4032	75	-	55	46	9	11.3	1×10^8	18.0
							5×10^8	16.5
	300	1/2	46	41	9	10.5	1×10^8	13.0
		1000	44	40	9		5×10^8	11.5
	500	1/2	23	22	12	9.3	1×10^8	5.5
		1000	10	7	45		5×10^8	5.0
2014	75	-	70	63	15	10.5	-	-
	300	1/2	51	49	14	9.6	-	-
		1000	34	30	20	9.6	-	-
	500	1/2	25	24	18	8.5	-	-
		1000	11	9.5	43	8.5	-	-
2219	75	-	64	45	10	10.6	1×10^8	16.5
							5×10^8	15.0
	300	1/2	51	39	20	9.9	-	-
		1000	49	37	20	9.0	-	-
	500	1/2	30	22	25			
		1000	30	22	25	8.6	-	-
	600	1/2	22	16	26			
		1000	18	14	28	7.6	-	-
	700	1/2	12	10	30			
		1000	5	4.5	80	6.4	-	-

materials in pressurized water reactors operating in excess of 300°F. In fact, an Al-Ni-Fe alloy (X8001) has been developed specifically for this purpose and has shown outstanding resistance to corrosion in high purity water at elevated temperatures. Corrosion tests on the X8001 alloy containing 0.5% Fe and 1.0% Ni in high purity water exhibited the following corrosion rates:

Exposure[a] Temperature, °F	Corrosion Rate, mils/yr
550	1.7
600	3.1
680	8.0

[a] Test duration - approximately 1500 hours.

5.2.1.5 Case Hardened Low Alloy Steels

Dynamic components that are relatively low stressed but are subject to wear in service require that their surfaces be hardened. An excellent example is the camshaft where cyclic Hertzian stresses on the surface approach 200,000 psi. In these applications, induction hardening of high carbon steels (AISI 1080), carburizing or nitriding of low-medium carbon low alloy steels (AISI 8620, 4340) are employed. Since fatigue fractures generally initiate at the surface, it is extremely important that the surfaces of case hardened components be free of defects or irregularities that will provide stress-risers. Assuming defect free surfaces, the surface hardening treatments will substantially increase the fatigue endurance limit, partly because of the induced compressive stresses in the surface. As an example, where the fatigue endurance limit of heat treated AISI 4340 is approximately 75,000 - 80,000 psi, shot peening the surface will increase the endurance limit to approximately 90,000 - 100,000 psi and nitriding the surface will increase the endurance limit to 120,000 to 135,000 psi. In a crankshaft application, designed for a minimum 115,000 psi tensile strength, a AISI 4140 steel heat treated to a 85,000 psi tensile strength and nitrided on the wear surfaces, provided the necessary wear and fatigue resistance.

5.3 Materials Recommendations

The materials recommendations for the single cylinder steam expanders were based on the materials technology survey (Section 5.1) and the materials selection study (Section 5.2). In general, considerable conservatism was exercised in the selection of materials for the engines in this program. The conservatism in materials selection was based on the fact that the engines were to be used as test vehicles. The selection of materials was further compromised by the fact that only one or at most two components of any design were required. For this reason no castings were utilized for major components of the expander other than the valve guide and tappet. A listing of the materials selected for each component in the single cylinder steam expanders together with the recommended materials specification and heat treatment condition are given in Table 5.2-VII. These materials were selected on the basis of a high probability of success rather than cost. Future studies would be useful, directed toward materials optimization with respect to cost. For example, the addition of approximately 1% Cr + 0.20% Mo (AISI 4140) to AISI 1045 steel represents an increase in price of the finished mill product to about $0.05/lb for quantities in excess of 10,000 pounds. In the final selection of materials, mechanical properties and environmental compatibility must be balanced against cost. Materials must be used up to the limit of their acceptable life before specifying a more costly material.

Table 5.2-VII

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
1. Inlet Manifold Cylinder Head Valve Body	2.25% Cr-1% Mo Steel	ASTM A387 Grade D, Plate ASTM A335 Grade P22, Pipe	Normalize 1700°F, Temper 1250°F
2. Exhaust Manifold	1.0% Cr-0.5% Mo Steel	ASTM A387 Grade B, Plate ASTM A335 Grade P12, Pipe	Normalize 1700°F, Temper 1250°F
3. Cylinder Block	1.25% Cr-0.5% Mo Steel	ASTM A387 Grade C, Plate ASTM A335 Grade P11, Pipe	Normalize 1700°F, Temper 1250°F
4. Cylinder Block-Lower (Crosshead Piston Expander Only)	AISI 304 SS/Gas Nitride Bore	ASTM A240, Plate ASTM A376, Pipe GE/P11BYA11, Nitride	Anneal 1900°F/Rapid Cool
5. Cylinder Liner (A)	17-22-A/Hard Cr Plate Bore	AMS 6304C/GE B5F5, Forging GE/P16DYA10, Chromium Plate	Normalize 1750°F/1-1/2 Hrs. Mar Quench 650°F/10-15 min. Temper 318 - 363 BHN
(B)	AISI 440C	AMS 5630C, Forging	Austenitize 1900°F/Oil Quench Double Temper 675°F/4 Hrs. Hardness Rc 52-56
6. Piston Compression Rings (A)	Sb Impregnated Carbon-Graphite	See Section 6.0 of Report	
(B)	Koppers K-1051/over Inconel X-750		
(Trunk Only) Rider Ring	Carbon-Graphite	-	-
Lower Seal Ring	Koppers K-35 Ni-Resist	-	-
Scraper Seal Ring	Koppers K-Iron	-	-
Oil Cutter Ring	Koppers K-6E	-	-

Table 5.2-VII(Cont'd)

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
7. Power Piston	17-22-A (1% Cr-0.55% Mo-0.3% V)	AMS 6304C/GE B5F5, Forging	Normalize 1750°F/1-1/2 Hrs. Mar Quench 650°F/10-15 min. Temper 311-363 BHN
8. Piston Rod (Crosshead Only)	H-11 Steel	AMS 6487C, Bars & Forgings GE/P16DYA10, Chromium Plate	Austenitize 1850°F/Air Cool Double Temper 1000°F/2 Hrs. Hardness Rc 52-55
9. Rod Seal (Crosshead Only)	PTFE-Bronze, MoS_2 Filled	-	-
10. Crosshead Piston	Al 4032-T6 Alloy	ASTM B247, Forging	Solution Treat 955 ± 15°F/2 Hrs. Water Quench 150-212°F Age 340°F/10 Hrs.
11. Connecting Rod	AISI 4140	AMS 6382G, Forging	Austenitize 1550°F/Oil Quench Double Temper 1000°F/2 Hrs. Hardness 320-360 BHN
12. Piston Pin	AISI 8620/Carburized	AMS 6276C, Bars & Forging GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in. 2 RMS Finish
13. Piston Pin Bushing Bushing	80 Cu-10 Sn-10 Pb/Mild Steel Backing	SAE 792	Stabilize 1000°F/4 Hrs.
14. Camshaft/Cam, Bearing Surfaces	AISI 8620 Carburized	AMS 6276C, Bars & Forgings GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in.
15. Tappet	Chilled Cast Iron/Fe_3O_4 Antichafing Coating	Eaton EMS 91 or EMS 80	-
16. Tappet Housing	Nitralloy 135 Mod/ Gas Nitride Bore	ASTM A355 Class A, Bar Nitride - Drawing Control	Austenitize 1750°F/Oil Quench Temper 1100°F/2 Hrs. Core Hardness 330-370 BHN Case Hardness RC 65 (converted) Case Depth 0.015-0.020 in.

Table 5.2-VII (Cont'd)

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
17. Push Rod	AISI 8620/Carburize	AMS 6276C, Bars & Forgings GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in.
18. Valve Head.Facing	H-11 Steel/Stellite 6	AMS 6487C, Bars & Forgings ASTM A399 Type R Co-Cr-A Hard Surface Rod	Austenitize 1850°F/Air Cool Double Temper 1000°F/2 Hrs. Hardness Rc 52-55 Coating Thickness - 0.125 in.
Valve Stem	H-11 Steel/Gas Nitride	Nitride - Drawing Control	Case Hardness Rc 65 (Converted) Core Depth 0.010-0.015 in.
19. Valve Seat (Facing)	Stellite 1	ASTM A399 Type R Co-Cr-C Hard Facing Rod	Coating Depth 0.125 in. Coating Hardness Rc 48 min.
20. Valve Guide	Ni-Resist D-3 (Nodular)	ASTM A439, Ductile Iron Casting	Stabilize 1600°F/2 Hrs. Furnace Cool to 1000°F @ 100°F/Hr. Hold 1000°F/1 Hr. Slow Cool
21. Valve Rings	H.S.25/LPA-101 Coat	AMS 5759 Coating - Drawing Control	Cold Worked & Aged @ 1100°F/5 Hr. Hardness Rc 45-50 Coating Thickness 0.003 in.
22. High Temp. Static Seals	H.S.25	AMS 5759	Material Cold Reduced to Rc 45-50

6.0 L U B R I C A T I O N

6.1 Lubrication Technology Base - Solid Lubricants

One of the most critical problems associated with the design of an efficient steam engine is that of lubrication within the cylinders and the inlet steam valve. In order to fully utilize the potential thermal efficiency of the steam cycle, it is necessary to operate the expander with high inlet steam temperature. With increased steam temperature, there is a corresponding increase in temperature of the materials within the cylinders and valves. The vast majority of steam expander designs, from the early Stanleys to the most modern engines, obtain upper cylinder lubrication by means of hydrocarbon lubricants pumped in with the steam. Due to thermal degradation, oxidation, and sludging of these lubricants, their use generally limits steam temperatures to about 750°F. In recognition of this limitation, the General Motors SE-101 engine was limited to a maximum steam temperature of 700°F.[12]

The Pritchard engine[20] is lubricated in a similar way. A slow speed mechanical pump forces small quantities of oil into the steam to act as an upper cylinder lubricant. Approximately one pint per 500 miles is used in this fashion. After 10,000 miles of operation with a closed steam cycle, 20 pints (2.5 gallons) of oil will have been pumped into the steam system. A number of other more recent steam engine designs also have expanders with the lubricant added to the working fluid side of the engine.

Even though lubrication technology for steam engines has been established for large, low-speed engines operating at moderate temperatures and pressures, an adequate lubrication system has not been identified for highly efficient steam engines that operate at high inlet steam temperatures and at high inlet pressures. There are two approaches which appear most feasible for lubrication within the cylinders of high temperature, high pressure steam engines: (a) to inject water-miscible fluids, which have been extensively developed for cutting and grinding applications, into the steam; or (b) to use solid lubricants which have been developed for extreme condition applications. With the injection of a water-miscible fluid, it would be necessary to add a separating system to separate the lubricant from the water to avoid fouling of the steam generator tubes; also, this approach would result in a high consumption of water-miscible fluid. For this reason, use of solid lubricants appears to be the most feasible method of lubricating the cylinder walls in order to permit the utilization of higher steam temperatures.

A literature search and industrial survey was made to identify and evaluate the current technology in solid lubricants and wear resistant materials. Searches were made through the following resources: General Electric Company Automatic Retrieval System, SAE Transactions, Engineering Index, NASA Scientific and Technical Information, AGARD and the Defense Documentation Center. The industrial survey was made to determine those solid lubricants that either are in commercial production or in the stage of advanced development.

6.1.1 Criteria for Solid Lubrication

The most important function of any lubricant is to keep the rubbing surfaces separated, both to prevent wear of the moving parts and to minimize the high coefficients of friction associated with metal-to-metal contact. A solid lubricant keeps the moving surfaces separated solely through its own strength. However, to do so, the solid lubricant must remain in the clearance between the moving surfaces for the entire design life of the engine. It follows then, that in order for a material to be an effective solid lubricant, it must possess most or all of the following properties and characteristics depending upon the form of the lubricant:

1. Good bonding properties or high tendency to adhere to rubbing surfaces
2. Low shear strength
3. High compressive strength

4. Laminar crystal structure
5. High bulk fracture strength
6. Good thermal stability
7. Chemical compatibility with surrounding environment (including structural materials)
8. Low wear
9. Good resistance to thermal shock
10. Good fabricability.

Although all of the above listed properties are important with respect to achieving satisfactory performance by solid lubrication, the ability of the lubricant to adhere to the rubbing surfaces is absolutely essential in order to protect the surfaces and prevent scoring, galling, gross welding and resultant metal transfer. This corresponds to wetting in liquid lubricated systems. For self-lubricating composites, it is necessary that the lubricant be transferred to the mating surface to form an adherent thin film. For example, the ability to form a transfer film is what makes graphite an excellant lubricant.

Low shear strength is important from the standpoint of achieving a low coefficient of friction. The coefficient of friction for solid lubricants in general is high in comparison to oils, as shown in the following tabulation:

Lubricant	Typical Coefficient of Friction
Oil	0.001
Solid Lubricant-Bonded	0.01
Solid Lubricant-Composite	0.1
Metal-Metal (Steel)	Approaching 1.0

From these data it can be seen that it is desirable to utilize materials with as low a shear strength as possible.

A high compressive strength is desirable from the standpoint that the higher the compressive strength of the lubricant, the higher is the load that can be applied before metal-to-metal contact is made. In effect, the lubricant should have a low shear strength/compressive strength ratio for best performance.

The laminar crystal structure of the lubricant is associated with low shear strengths and low coefficients of friction. Although effective lubrication can be achieved with solids that do not have a laminar structure, the most effective solid lubricants are those that do have the laminar structure and have good surface adherence properties. The laminar structure consists of alternating layers of atoms in which there are strong bonds between atoms within the layer (covalant or ionic forces) and weak bonds between atoms in adjacent layers (Van der Waals' forces). The Van der Waals' forces are easily broken in the "layer-lattice" structure resulting in the successive atomic layers readily sliding over each other such as basal plane slip in the hexagonal structure of graphite.

Solid lubricants that have a laminar crystal structure are highly anisotropic with respect to mechanical properties. In cases where the lubricant is fabricated into a self-lubricating composite, the lubricant matrix must have sufficient strength to withstand high pressure differentials and frictional drag loads at high temperatures without changing shape or fracturing. To endure the start-up cycle of a steam engine, the self lubricating composite also must have adequate resistance to thermal shock.

Thermal stability of the lubricants is required to assure acceptable performance over the entire temperature range and throughout the design life of the engine. Since the total surface temperature of the lubricant includes the heat of the environment and the heat of friction, the influence of both must be considered. Thermal stability includes resistance to thermal decomposition and crystallographic changes as well as being resistant to oxidation in air and steam. Crystallographic changes will result in changes in frictional characteristics. However, the effects of surface heat are not always detrimental since some solid lubricants are designed to work as a liquid phase.

Another important consideration is compatibility with all engine materials and small quantities of organic lubricants. Consideration of free energy exchange between steam and the candidate materials indicates that a large number of candidate solid lubricants are available for evaluation. However, free energy exchange is somewhat complicated by the addition of cylinder and piston materials.

Finally, an optimum solid lubricant is one that has low wear and the ability to replenish the transfer film such that long life is obtained. Also, for self-lubricating applications, the solid lubricant must be readily fabricable into usable shapes.

6.1.2 Forms and Types of Solid Lubricants

The lubricating solids can be used in several different forms: (a) as loose powder dusted or rubbed on the mating surfaces or supplied to the parts in a controlled stream of a carrier gas; (b) as bonded materials attached to the surfaces with an adhesive binder; (c) as self-lubricating bearing materials where the lubricating solid is dispersed throughout a metal or plastic bearing or where the lubricating solid forms the matrix of a fabricated bearings, and (d) as additives to oils and greases to replace the chemical extreme pressure (EP) additives normally used. The latter application of solid lubricants is not applicable to the lubrication of cylinder walls and will not be discussed further. Of the remaining three forms in which a solid lubricants can be applied, each form has its own advantages and disadvantages.

Loose powders, which have the advantage of minimizing fabrication or assembly problems, have the disadvantage of providing a very thin film of limited life; the film either has to be replenished by external means in order to achieve the desired life or be used only to assist in the wear-in period.

Bonded solid film lubricants generally provide longer life than powders with the performance of the bonded solid film varying with the specific lubricant, the bonding agent and the method of application. Some dry films depend solely on small particle size for bonding. The lubricant particles are held together only by the attraction arising from the extremely fine particle size and there is little bonding to the substrate. A limited amount of dispersing agent is sufficient to bring about bonding. Particle-bonded dry films can be dispensed from aerosol containers or dispersed in water, organic solvent or other volatile carrier. Most of the particle-bonded lubricants are air-drying and require no baking. However, the particle-bonded lubricants have the same problem as the loose powders in that they have very limited life unless they can be replenished. An organic resin (phenolic, epoxy, polyimide) may be added to increase binding between particles and bonding to the substrate. The resin produces an adherent paint-like film that offers the best wear life and widest range of use. The lubricant particles also can be bonded together by a silicone, a suitable ceramic or metal salt (sodium silicate, aluminum phosphate) that is hardened upon removal of the solvent. This type of film can be very hard and is more temperature-resistant than the organic resin bonded films. The type of binder that is used depends upon the adhesion required, the anticipated service temperature and environment, and the desired wear life. This type of bonded film can be applied by spraying, dipping or brushing. Newer application techniques include electrophoretic deposition, plasma spraying and sputtering.

One of the major problems associated in the bonded films is that the lubricant is usually limited by the binder itself. For example, the resin bonded solid lubricants are limited to use temperatures of up to about 400°F; silicones and silicates have permitted service temperatures to increase to about 600°F at the expense of higher friction and lower wear life. The organic resin and metal salt bonded solid lubricants also have a limited wear life that can be attributed to the very small volume of lubricant available. This becomes evident when one considers that the film is only 0.0002 - 0.0005 in. thick.

Because of the relatively short life of the bonded solid lubricants, self-lubricating components containing solid lubricants have been developed for longer life bearing and seal applications in order to take full advantage of the good lubricating qualities of solid lubricants. The solid lubricants are incorporated into the self-lubricating composites by impregnation, sintering, and various forms of high-temperature, high-pressure compaction. Self-lubricating bearings generally fall into two major classes: (a) solid lubricant matrix utilizing graphite or the lubricant plastics such as nylon, polytetrafluoroethylene (PTFE), etc., with reinforcing materials as fiberglass, graphite, molybdenum disulfide, and soft metals; and (b) metal matrix in which lubricating solids such as graphite, PTFE, metal disulfides, selenides, or tellurides and low melting point metals are mixed with metal/alloy powders of copper, iron, nickel, molybdenum, etc., and sintered under pressure or impregnated into a porous metal matrix skeleton.

In the application of solid lubrication for cylinder walls in high performance reciprocating steam expanders where steam temperatures approach 1000°F, the peak steam pressure is 1000 psi and peak piston ring side loads are on the order of 200 - 300 psi at an average linear velocity of 1200 -1600 feet/minute, it was assumed that self-lubricating composites would be required to achieve the necessary life. Therefore, in the survey of solid lubricants, emphasis was placed on the self-lubricating solids or composites. Bonded solid film lubricants were considered primarily as an auxiliary lubricant during the wear-in process or as an aid in establishing a transfer film. The following materials were reviewed as lubricants to minimize wear in the cylinder wall/power piston interface and valve stem/guide interface:

Self-Lubricating Solids

- Carbon-graphites
- Polytetrafluoroethylene (PTFE)
- Sulfides/Selenides
- Metals
- Porous metal composites
- Hard surfacing materials

Bonded Solid Films

- Sulfides/Selenides
- PTFE
- Graphite
- Soft Oxides

6.1.2.1 Self-Lubricating Solids

6.1.2.1.1 Carbon-Graphites

Graphite was found to show promise of being a very effective solid lubricant for steam engine applications. Graphite has been used as a lubricant for many years and is used extensively for brushes in electric motors. In fact, it was excessive wear of motor brushes in high altitude aircraft that led to the detailed study of the mechanism of graphite lubrication. In a detailed study of this problem by the General Electric Company[21,22,23] it was discovered that the low friction and wear normally exhibited by graphite is due not to any lubricant quality inherent in the graphite, but to adsorption on the surface of substances from the atmospheric environment. The laminar structure of graphite undoubtedly is essential, but alone it is not sufficient. It was found that water vapor, adsorbed on the surface, caused reduction in friction and wear rate. The minimum pressure of water vapor that is necessary to achieve minimum wear rates is on the order of 3 - 5 torr. Oxygen showed a similar effect but at much higher pressures, i.e., 300 - 500 torr. It has been proposed that only the edges of the graphite crystals need be covered with the adsorbed film to achieve the low friction and wear rates by permitting the laminar layers of the hexagonal lattice to slip. In the absence of water vapor or oxygen the edges of the laminar layers of atoms are locked by free radicals formed by the evaporation of volatile oxides of carbon. Graphite then is expected to maintain good lubricity in the cylinder wall because of the presence of high pressure steam.

In extremely high temperature applications, graphite tends to oxidize and its uses are restricted to neutral or reducing atmospheres. However, graphite has been successfully used as seals for steam turbine driven electrical generating equipment which tends to verify its stability in high temperature steam. The threshold oxidation temperature of graphite in steam has been listed as about 1300°F[24] and good resistance to oxidation by steam is expected at the maximum proposed cycle temperature of 900°F. This may be further verified by examination of the free energy changes for the two oxidation reactions:

$$C + H_2O = CO + H_2$$

and

$$C + 2H_2O = CO_2 + 2H_2$$

The free energies of these reactions are plotted against temperature in Figure 6.1-1. As may be seen on this plot, the driving force for the reactions (negative free energy) does not become appreciable until temperatures greater than about 1200°F. This is in essential agreement with the temperature limit cited above. In air, graphite cannot be used

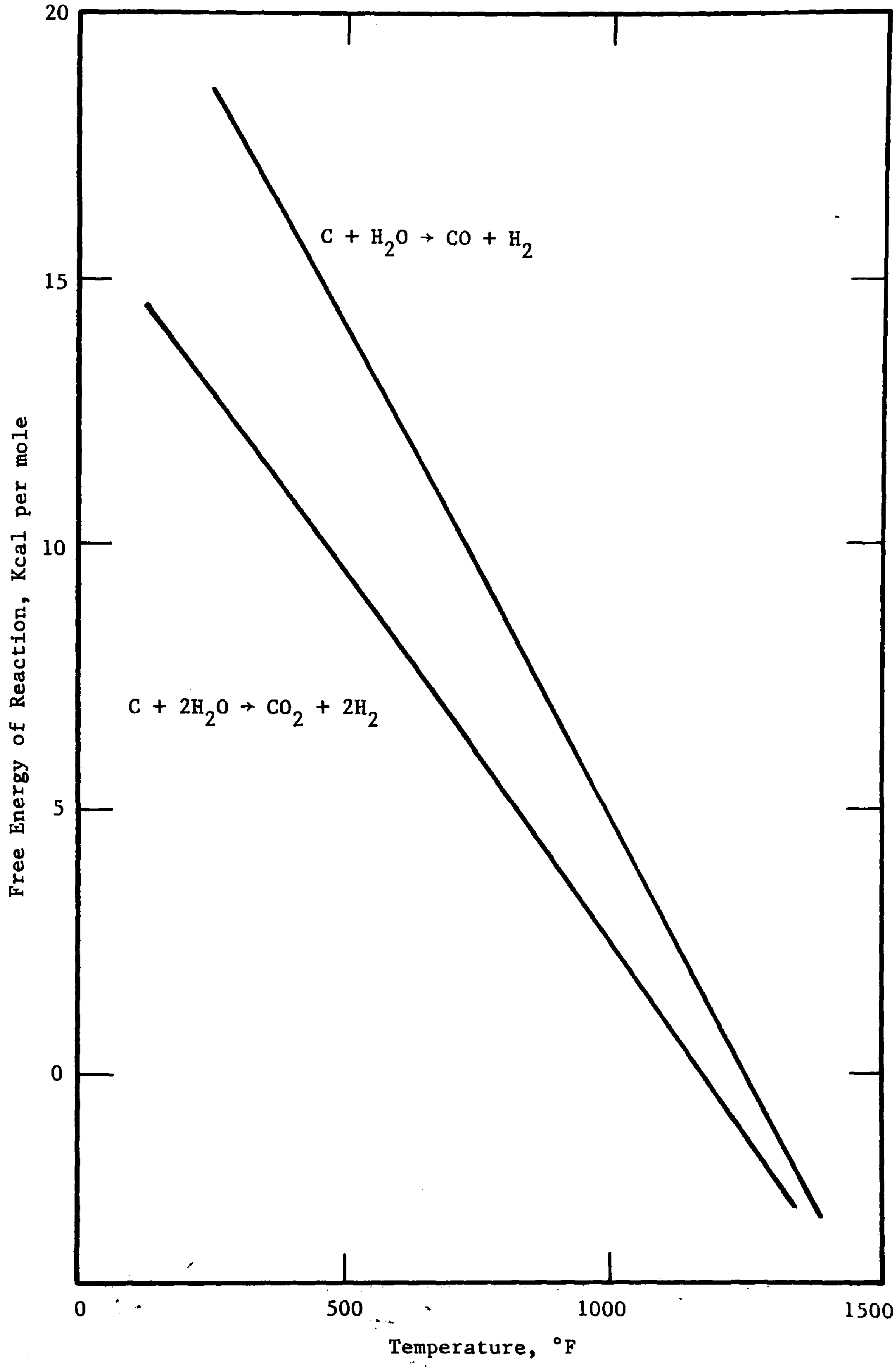


Figure 6.1-1. Free Energy of Oxidation Reactions Involving Graphite and Steam.

much above 750°F without severe oxidation reactions.

Even though graphite displays excellent properties from the standpoint of thermal stability, thermal conductivity, low elastic modulus, low shear strength and thermal shock resistance, it has poor strength properties in tension. Higher strengths are possible with the carbon-graphite grades of which there are an infinite number to choose; lower wear rates also are possible with the carbon-graphite materials. These materials are available commercially and, although it is an oversimplification, are produced in four general classes: (a) straight carbon-graphites with varying carbon/graphite ratios, (b) resin impregnated, (c) metal impregnated (Cu, Cu-Pb, Cu-Sn, Pb, Ag, Babbitt) and (d) high temperature oxidation inhibited grades. The maximum service temperatures for these materials in air are reported to be up to 500°F for the resin impregnated grades, 700°F for most of the straight carbon-graphites and metal impregnated grades, and 1000° - 1200°F for the high temperature grades. Properties of candidate carbon-graphite grades that are believed to be suitable for application as piston compression rings in a steam environment are listed in Table 6.1-I. Care must be exercised in the use of the oxidation-inhibited, high temperature grades and some of the resin impregnated grades in steam and water because of the tendency for some of the resins and oxidation inhibitors to exude to varying degrees in these environments. Consideration also must be given to the compatibility of graphite with metals used for the impregnation process as well as mating metals/alloys that are in contact with the graphite in service. Studies by McGee[(25)] at General Electric have shown that metal oxides that can be reduced to lower oxides or to the metallic state by graphite in the presence of oxygen appear to function as catalysts in the oxidation of the graphite. The relative activity of various metal oxides with respect to their effect on the oxidation of graphite is shown in Table 6.1-II and is based on the change in ignition temperature of pure polycrystalline graphite. The data show that lead oxide is particularily effective in lowering the ignition temperature of graphite.

Although there are numerous influencing factors, the coefficient of friction for the various grades of carbon-graphite will generally range between 0.1 to 0.25. Wear rates also will vary significantly with

Table 6.1-I

Properties of Carbon-Graphite Grades

Vendor	Grade	Type[a]	Hardness Shore (Scleroscope)	Compr. Str., ksi	Transv. Fracture Str., ksi	Tensile Str., ksi	E 10^6 psi	Coef. Th. Exp., in/in/°F $\times 10^{-6}$ (RT-500°F)	Th. Cond. Btu/Ft/Hr °F	Apparent Density gm/cc	Porosity Vol. %	Max. Service Temp., Air
US Graphite Div.	86	CG-R	100	36	12.5	9.5	3.0	2.3	7.6	1.90	0.5	500
	110	CG	92	32	10.0	8.5	3.3	2.3	8.0	1.90	1.0	700
	2980	CG-HT	67	15	4.5	4.0	1.8	2.3	30.0	1.85	4.0	1000
	102	CG-Cu	88	38	9.5	7.5	2.6	3.1	8.7	2.35	4.0	700
	103	CG-Ag	89	35	10.5	7.5	2.8	3.1	9.3	2.70	3.0	700
	107	CG-Sb	92	38	10.0	7.5	3.5	2.4	7.0	2.20	1.0	700
Pure Carbon Co.	P5NR	CG	80	30	8.5	7.0	2.7	2.2	-	1.70	20.0	500
	P03	G	75	20	8.0	5.5	1.7	1.9[b]	-	1.82	10.0	900
	P658RC	CG	90	38	11.0	8.0	3.1	2.2	-	1.80	2.0	500
	X3310	CG-HT	85	29	9.4	8.5	2.6	2.3[c]	-	2.05	0.4	1000
Graphite-Metallizing Corp.	Bronze Graphalloy	CG-Cu-Sn	-	~25	-	~8.0	-	2.3	-	-	-	750
UCC	CDJN	CG-HT	105	36	8.8	7.0	3.2	2.3	4.4	1.76	-	875
	CJP	CG-HT	65	26	8.0	7.0	2.1	2.3	13.3	1.77	-	1100
Carbone Corp.	5890	G										
	5890	G-HT										
	JP-500	CG										

(a) G Graphite; CG Carbon-Graphite; CG-R Resin Impregnated; CG-HT High temperature/oxidation inhibited; CG-Sb, Cu, Ag, Cu-Sn Metal Impregnated

(b) RT to 900°F

(c) RT to 1000°F

Table 6.1-II

Catalytic Activity of Oxides in Graphite Oxidation

Catalyst (acetate or oxide)	w/o as Metal	Ignition Temp., °C
Pb	0.15	384 (738°F)
V	0.20	490
Mn	0.45	523
Co	0.33	525
Cr	0.95	540
Cu	0.20	570
Mo	0.15	572
Ag	0.16	585
Cd	0.21	590
Fe	0.13	593
Pt	0.03	602
Ni	0.45	613
Ir	0.40	638
Rh	0.20	622
Ru	0.30	640
Pd	0.30	659
Ce	0.72	692
Zn	50.00	700
W	0.02	718
Hg	0.10	720
Sn	0.10	738
Uncatalyzed		740 (1364°F)

the grade of carbon-graphite. Overall, the metal impregnated carbon-graphite appears to be the most suitable for use as compression piston rings in the steam expander. Data indicate that after an initial high wear rate during the wear-in period, the metal impregnated carbon-graphites have superior wear rates for long time service; however, the coefficients of friction for the metal impregnated carbon-graphites generally are higher than the other grades. In addition, the metal impregnated carbon-graphites have low porosities and permeabilities, are strong and are compatible with steam. With the exception of their tendency to exude in the presence of water, some of the oxidation inhibited, high temperature grades also are attractive. They have good friction and wear characteristics and low values of porosity and permeability.

Additional possibilities involve composites of graphite with relatively soft oxides. A cadmium oxide-graphite mixture has friction coefficients in the order of 0.1 or less over most of the temperature range from 100 to 1000°F [26]. The cadmium oxide is presumed to improve the adherence of graphite to the surface and, hence, to improve the lubricity effectiveness.

Materials that would be satisfactory as mating materials in contact with the carbon-graphites under dynamic conditions are AISI 440C or hard chromium plate with the latter being limited to service temperatures of less than 700°F. A high hardness is desirable, i.e., greater than Rockwell C 45, and the surface finish should be smooth but not too smooth so as not to be able to hold the transfer film. A finish on the order of 8 - 12 RMS appears satisfactory.

6.1.2.1.2 Polytetrafluoroethylene (PTFE)

Plastic bearings made from polytetrafluoroethylene (PTFE) are available for nearly every type of application. They have been used successfully as inserts to plain bearings, as reinforced thin sheets on plain spherical bearings, and as the retainer material for ball bearings. PTFE is one of those materials that does not have a laminar layer-lattice structure and yet has an exceptionally low coefficient of friction. The low friction has been attributed to its low surface energy. This results in very weak adhesion and shearing takes place primarily at the interface rather than in the bulk material [27]. It also has the necessary characteristic of establishing an adherent transfer film on most rubbing metal surfaces. The major disadvantage of PTFE is its low compressive strength which results in cold flowing under relatively low loads and limits its use to temperatures on the order of 400 - 450°F (it also begins to thermally decompose at about 540°F). Other disadvantages of PTFE are its poor thermal conductivity and high coefficient of thermal expansion.

To improve the flow characteristics of PTFE, PTFE composites have been developed. The three most common materials that have been used to reinforce the PTFE are bronze, glass and carbon-graphite. In one recent symposium of PTFE seals for reciprocating air compressors[28], it was generally considered that no one filler material has demonstrated superior performance over the others and that lack of consistency in the composites was a major factor in the erratic performance. Three typical compositions that are in use for seals in reciprocating compressors are:

1. PTFE + (15% Bronze + 5% MoS_2)
2. PTFE + 20% Glass
3. PTFE + 10% Carbon

   (% by volume)

G. L. Griffin[29] has suggested that non-uniformities in the structure as a result of molding practice of filled polymers could account for differences in performance. In particular, the surface of molded composites is usually deficient in filler material and this could result in varying friction and wear behavior.

Other developments to improve the poor creep properties of PTFE are the use of fibers of PTFE. Compacts can be made from PTFE fibers which have 25 times the tensile strength of conventional PTFE compacts made from powders.

Halliwell of the U.S. Navy Marine Engineering Laboratory has conducted tests with PTFE reinforced with metallic filament windings[30] in an attempt to develop a piston seal for high-pressure air compressors. The filament winding approach has resulted in a superior reinforced matrix as compared to randomly dispersed particles and fibers previously reported. This new approach to the problem has resulted in extended compressor life with low rates of wear and leakage. Halliwell found that reliable compressor operation could be obtained at 5000 psig for periods beyond 1000 hours using such seals in lieu of conventional split rings. The operational life of PTFE rings in a reciprocating steam expander will be affected by the quality (dryness) of the steam. Wear rates of PTFE decrease with dryness until a dewpoint of $\sim -40°F$ is reached whereupon the wear rate increases drastically. However, small amounts of condensed water will cause an increase in wear rates and should be avoided.

Polyimides are the next generation of lubricating plastics pushing the use temperature from the 400° - 450°F for PTFE to 500° to 600°F. The polyimides also have superior friction and wear properties. Unfortunately the polyimides cannot be used in high temperature (> 212°F) steam or water because they will hydrolize.

6.1.2.1.3 <u>Metals</u>

In some applications, metals are used on bearing surfaces to achieve improved friction and wear characteristics. The metals (and alloys) used in these applications can be categorized in three classes: (a) classical layer-lattice crystal, (b) low shear strength and (c) soft oxide formers. Buckley and Johnson[31] investigated the influence of crystal structure of metals on their friction and wear behavior and found that relatively low coefficients of friction could be obtained for cobalt in which the hexagonal crystal structure is stabilized. A coefficient of friction of 0.3 was measured for an oxide-free 25% Mo-Co alloy at speeds of 2000 ft/min (750°F; pressure of 10^{-9} torr). The coefficient of friction of cubic cobalt under similar conditions is on the order of 0.7.

A number of metals that do not have the layer lattice structure are still widely used in bearing applications. These metals are listed in Table 6.1-III; they are soft and shear easily[32]. For example, the metals lead, tin, indium and silver are used in many common plain bearings, i.e., piston pin bearing. However, the coefficients of friction of these metals below their melting points are relatively high (> 0.3). When used above their melting points (liquid film) friction coefficients are considerably lower, on the order of 0.17 for lead.

Table 6.1-III

Soft Metals Used in Bearings

Metal	Moh hardness	Melting point, °C
Indium	1	155
Thallium	1.2	304
Lead	1.5	328
Tin	1.8	232
Cadmium	2	321
Gold	2.5	1063
Silver	2.5-3	961
Platinum	4.3	1755
Rhodium	4.5-5	1955

A commercial material incorporating a soft, low shear strength metal for lubrication is the Bishiralloys[33]. These alloys are produced by pressing and sintering alloy powders, followed by impregnation with lead and finally rolling to size. The compositions and hardnesses of these materials are given in Table 6.1-IV. Alloy C is reported to have the best wear resistance; the coefficient of friction is on the order of 0.2. A continuous film of lead is maintained on the surface by flow of the lead in the pores of the alloy. However, because of the low melting point of lead (621°F) the maximum ambient service temperature for the Bishiralloys is about 575°F.

The metal gallium is a liquid at 86°F and effective boundry lubrication is obtained with gallium rich films on AISI 440C stainless steel. The coefficient of friction of gallium coated AISI 440C vs. AISI 440C at 500°F in air was measured to be 0.104 (sliding speed 390 ft/min, load 1000 grams)[34].

Oxidation of metals will result in significant reductions in the coefficients of friction and protection of the metal surfaces[35]. The transition temperatures for improved friction and wear behavior were determined for iron, copper, nickel, molybdenum and chromium to be 100° - 200°F, 400° - 500°F, 1200° - 1400°F, 800° - 900°F, 800° - 1100°F, respectively. Coefficient of friction values of 0.2 - 0.3 were obtained

Table 6.1-IV

Chemical Composition and Hardness of BISHIRALLOY

	Basic Composition of BISHIRALLOY Matrix Wt. %						wt % Range of Impregnated Lead in BISHIRALLOY	Hardness Brinell
	Fe	Cu	Ni	Mo	Cr	C		
BISHIRALLOY-A	Bal	5.0	--	--	--	less than 0.2	15 ∿ 30	75 - 95
BISHIRALLOY-B	Bal	2.5	1.5	2.5	--	less than 0.2	15 ∿ 30	105 - 135
BISHIRALLOY-C	Bal	2.5	1.0	1.0	5.0	less than 0.2	15 ∿ 30	65 - 95

upon the formation of the oxides in comparison to values of 0.5 - 1.0 for clean surfaces at lower temperatures. The formation of so-called soft oxides MoO_3, WO_3, Cu_2O, ZnO, Co_2O_3, PbO, CdO were the most effective in preventing surface damage. The lowest friction coefficient was obtained with PbO.

A commercial product is available which reportedly provides low friction and wear characteristics as the result of the properties of its oxide surface film. The material is Clevite 300, an iron base alloy containing cobalt and molybdenum. The alloy is produced by powder metallurgy techniques and its properties are listed in Table 6.1-V.

6.1.2.1.4 Sulfides/Selenides

The sulfides and selenides of molybdenum, tungsten, tantalum and columbium have the layer-lattice structure similar to graphite and have extremely low shear strengths. Further, they do not require water vapor or oxygen for their lubricating ability. Another difference between the sulfides or selenides and graphite is that the weak easily sheared bonds occur only between every third layer of atoms instead of every layer as is the case for graphite. In MoS_2, for example, each layer of molybdenum atoms is strongly bonded to the adjacent layers of sulfur atoms on each side but the layer of sulfur atoms is weakly bonded to the sulfur atoms in the next adjacent layer so that shear occurs through the weak sulfur-sulfur bonds. Recent studies on the lubricating mechanism of MoS_2 were reported at the solid lubrication conference in Denver[29].

Since MoS_2 occurs in nature, it is considerably less expensive than the other sulfides and the selenides or tellirides which are synthetically produced. For this reason most of the common solid lubricant composites based on this group are made from MoS_2. The effectiveness of MoS_2 in air is limited to about 750°F where oxidation begins to effect its lubricity. The oxidation products of MoS_2 are MoO_3 and a sulfur compound which can result in abrasion and corrosion and an increase in the coefficient of friction at the lower temperatures. (It should be noted that at temperatures above 1300°F - 1400°F, MoO_3 can be an effective lubricant; the melting point of MoO_3 is 1463°F).

Table 6.1-V

Properties of Clevite 300

MECHANICAL PROPERTIES

Temp. (F)	Yield Strength, psi	Ultimate Strength, psi	Elongation, %	Elastic Modulus psi x 10^6	Compression Strength, psi	Hardness RA
75	70,500	106,000	-	28.4	210,000	66 (31 Rc)
600	-	107,000	2.8	25.8	-	57
900	62,250	95,800	2.7	24.4	-	-
1200	48,000	71,000	6.6	22.5	-	35

FRICTION AND WEAR PROPERTIES

(Based on face-seal tests of Clevite 300 against itself at 18 psi load and 150 fps surface speed, in air.)

Temp., °F	Coefficient of Friction, "f"	Wear Rate, in/hr
500	0.072	0.00073
750	0.063	0.00062
1000	0.052	0.00035
1200	0.047	0.00030

Composites based on the solid MoS_2 lubricant have been developed by Boeing [36] and described by Devine[37]. The composites developed by Boeing were quite brittle and even the most successful composite, 90% MoS_2-8% Fe-2%-Pt, would have strength problems because of the brittle iron sulfides that form at the sintering temperatures. Piston rings have been produced from iron bonded MoS_2 and evaluated by Midwest Research[38]. The solid composite called "Navy Lube"[37] is a MoS_2-Graphite-Sodium Silicate ($NaOSiO_2$) and is not only weak but the $NaOSiO_2$ is hygroscopic and would not be suitable for use in the steam environment.

Another series of composites based on MoS_2 and WS_2 are described by Hopkins[39]. Three of these composites have shown low wear rates and low friction in air. The compositions of these materials are:

1. 53% WS_2 - 12% Co - 35% Ag
2. 80% MoS_2 - 20% Ta
3. 50% MoS_2 - 38% Ta - 12% Fe

Overall, the use of solid lubricant composites with MoS_2, or other disulfides or diselenides, as the base do not appear attractive for use as piston rings in steam expanders primarily because of their low strengths and fragile nature.

6.1.2.1.5 Porous Metal Composites

In order to utilize the good lubricating qualities of some of the better solid lubricants that are too weak to be used as the base of self-lubricating composites, porous metal composites have been developed. These composites are generally produced by powder metallurgy techniques in which the metal or metal alloy and the lubricant are blended, pressed and sintered to form a metal matrix throughout which are discrete pockets of solid lubricant. In one investigation, it was found that a minimum concentration of 5% of the solid lubricant was required to achieve a transfer film and thus low wear rates. Friction coefficients of 0.2 were measured [40]. On the other hand, too high a concentration of the lubricant results in a weak structure. Examples of this type of porous metal composite are: (a) 5-15% MoS_2 in a 95% Ag - 5% Cu matrix, (b) 5-40% MoS_2 in a nickel or 80% Ni - 20% Cr matrix, (c) the commercial Molalloy - MoS_2 in a refractory metal (Ta/Mo) matrix.

Another method of producing the porous metal composites is to prepare a metal skeleton with controlled porosity with respect to volume and distribution and impregnate the pores with the lubricant by application of pressure and heat. Metal skeletons with up to 65% void volume have been produced with Inconel 600, Inconel X-750, Hastelloy X, Nichrome V alloys in which PTFE, graphite, MoS_2 have been impregnated. A Inconel 600 (60 - 65% voids) impregnated with PTFE has performed well in seal applications at temperatures up to 500°F.

The metal fluorides have shown promise for use as lubricants for high temperature application in air. Sliney at NASA Lewis Research Center has investigated the use of fluorides for a number of years [41,42,43]. Coefficients of friction of self-lubricating porous metal composites (40% void volume in Inconel X-750), vacuum-impregnated with fused fluoride eutectic of 62% BaF_2 - 38% CaF_2 were measured to be less than 0.1 at 1000°F and at a sliding velocity of 2000 ft/min while under load. The coefficient of friction increases as the temperature and speed decrease; at 500°F and a speed of 1000 ft/min the coefficient of friction is approximately 0.3. More recent studies have shown that coefficients of friction of 0.2 ± 0.05 could be obtained at speeds as low as 500 ft/min over the temperature range of RT to 1700°F. These data were obtained with a porous nickel alloy impregnated with a high temperature enamel (NBS-418) and an overlay of the BaF_2 - CaF_2 eutectic.

Finally, this approach is compatible with classical piston ring materials technology where gray cast irons are used because of the excellent lubricity imparted by flakes of graphite. Similarly, the nodular Ni-Resist alloys of 20 - 30% nickel content with its spheroids of graphite has been shown to have good friction and wear characteristics in conjunction with good corrosion resistance in superheated steam. It is believed that the use of strong porous metal composites impregnated with a solid lubricant of low shear strength and ability to establish a transfer film is one of the most attractive materials for application as piston rings in reciprocating steam expanders.

6.1.2.1.6 Hard Surfacing Materials

In applying the adhesion theory in friction and wear, relatively low coefficients of friction are achieved between two sliding hard materials that have no affinity for each other, i.e., materials with little or no mutual solubility or tendency to form intermetallic compounds. The hard, wear resistant surfacing materials fall into this class of materials and warrant investigation.

Plasma sprayed coatings are of interest because they can be readily applied. Koppers has plasma sprayed rings with their K-1051 coating (metal bonded Cr_3C_2) for use in lubricated, reciprocating steam expanders running against nitrided steel or AISI 440C stainless steel which was heat treated to a hardness of Rockwell C 50-55. Since the hard metal carbides are metallic in nature with respect to their atomic bonding, a more suitable materials combination with respect to "alloying tendency" might be an oxide coated cylinder liner running against a carbide coated ring. Possible combinations are Union Carbide Corp. LC-19 (Cr_2O_3 + Al_2O_3) sprayed on the cylinder wall and either LW-1 (Cobalt bonded WC) or LC-1 (Nichrome bonded Cr_3C_2) sprayed on the ring surface. In dry rubbing tests between LC-1 (Cr_3C_2-15% Ni-Cr) and LA-2(Al_2O_3), coefficients of friction of 0.15 - 0.27 were measured in the temperature range of RT to 1400°F.

E.I. du Pont de Nemours and Co. have recently made available a series of anti-friction and anti-wear LP alloys offering outstanding advantages. Metallurgically, the LP Alloys consist of hard grains of an intermetallic compound with a Laves Phase structure dispersed in a softer matrix, that provides good embeddability characteristics in contrast to other hard facing materials. Currently preferred compositions contain Laves-phase hexagonal close-packed intermetallic compounds of cobalt, molybdenum and silicon (Co_3Mo_2Si and CoMoSi) in a cobalt-rich matrix. The Laves phase, depending upon its composition has a micro-hardness of 1000 to 1500 DPH and the matrix has a hardness of 200 to 800 DPH. Overall, the bulk hardness ranges from R_c 30 to R_c 60. Tests have shown that, under extreme loading and boundary lubrication conditions, LP Alloys exhibit very little wear.

LP Alloys are produced as medium and fine powders for plasma spraying and powder metallurgy applications; they also can be cast as semi-finished parts. The compositions (in wt. %) of LP Alloys are shown in Table 6.1-VI.

Table 6.1-VI

Chemical Composition of LP Alloys

	Co	Mo	Si	Cr	Vol. % Laves
LPA 100	35	35	10	--	65
LPA 200	70	28	2	--	20
LPA 300	45	48	7	--	98
LPA 400	62	23	2	8	50

In one application, the sleeve and roller bearings and the piston skirt and rings of a 2-cycle engine were coated with LPA-100 and the engine operated without oil in the fuel for 50 hours with satisfactory results.

Another new material of interest that is available from E.I. du Pont de Nemours is the compound Ni_3B. The material is applied to the surface by electro-chemical methods and has a microhardness of 900 - 1500 knoop (R_c 60-70). Friction data of Ni_3B against hardened steel show a coefficient of friction of 0.1 at 600°F in air, unlubricated.

Other materials in this class that could be considered are: various grades of SiC, Si_3N_4 and TiC in the form of pressed and sintered compacts or applied as coatings.

6.1.2.1.7 Mixed Composites

Multicomponent self-lubricating composites have been developed that contain a metal matrix, a film former, and a load-carrying component [44]. Examples of these types of composites are Ag-PTFE-WSe_2, Cu-PTFE-WSe_2, Ag-Bronze-PTFE-WSe_2, Ag-Hg-PTFE-$MoSe_2$. The latter material has shown excellant friction and wear characteristics at 600°F [45]. The modified PTFE melts at 590°F and is thought to provide the excellent friction and wear characteristics.

Other mixed component self-lubricating composites that have been investigated are: WSe_2-Ga/In [45] and Graphite-WS_2-NaF [46].

6.1.3 Bonded Solid Film Lubricants

The primary disadvantage of the bonded solid film lubricants is that there is little stored lubricant in the film and as a result, service lives are limited. The bonding agents are either organic thermoplastic or thermosetting resins (phenolic, epoxy, polyimide) or inorganic metal salt (sodium silicate) and fused ceramic (aluminum phosphate). Because the bonded films have so little strength, the films are kept as thin as possible, usually on the order of 0.0005 inch or less. For these reasons, the use of bonded films is generally limited to high load/low sliding velocities or low load/moderate velocities. Most commercially available bonded solid film lubricants are limited in temperature by the bonding agent. For example, at temperatures above 400°F, the organic bonding agents tend to thermally decompose causing failure of the film because of lack of adhesion. Commercial products of this type incorporating various solid lubricants are:

Solid Lubricant	Bonding Agent	Service Temp., °F
MoS_2	Epoxy	275 - 300
90% MoS_2-10% Graphite	Phenolic	300
PTFE	Polyimide	450 - 500

A newer bonded film based on the polyimide for use at higher temperatures was reported by Campbell and Hopkins[(47)]. It consists of MoS_2 and Sb_2O_3 dispersed in the polyimide binder (MLR-2) and it shows excellent wear properties. However, it should be pointed out again that the polyimide resins cannot be used in high temperature steam and water.

The metal salt and fused ceramic bonded films were developed for use at temperatures generally above 600°F. Examples of these types of bonded films were reported by Hopkins[(48)]: MLF-5, is a sodium silicate bonded MoS_2-graphite - Au film and MLF-9 is a aluminum phosphate bonded MoS_2-graphite - Bi film. Ceramic bonded MoS_2-graphite films have been used in excess of 700°F in air. As mentioned previously sodium silicate is hygroscopic and cannot be used on steam or water.

The solid lubricants of interest that are being developed at Midwest Research are the FSAL 28 ($AlPO_4$ bonded $Ba_2F + Ca_2F$) and FSAL 29 ($AlPO_4$ bonded $Ba_2F + Ca_2F + Mg_2F$). These lubricants are applied as bonded films and can be sintered at temperatures below the fluoride eutectic temperatures. About 6 vol.% $AlPO_4$ has been found to be the optimum binder content. Coefficients of friction of the FSAL 28 versus Inconel X in air are on the order of 0.1-0.15 at 500°F and 0.10-0.12 at 1000°F. Loads have varied between 6000 - 25,000 psi at speeds of ~180 ft/min.

At the higher temperatures, >1000°F, PbO has been shown to exhibit good friction and wear properties. Because PbO oxidizes to Pb_3O_4 at temperatures below 1000°F resulting in loss in lubricity, additions of SiO_2 are made to PbO to inhibit the oxidation reactions[49]. Very low friction characteristics (0.1 coefficient) have been achieved with 10% SiO_2-PbO bonded film at temperatures of 500° to 1200°F.

Another very promising new material that is being studied is graphite fluoride, $(CF_x)_n$[50]. Coefficients of friction equal or superior to MoS_2 and graphite were reported at temperatures up to 750°F in air. Superior wear properties are attributed to greater adhesive qualities. Although moisture is beneficial, it is not a prerequisite for low shear strength.

The use of bonded solid film lubricants does not appear to be as promising for the lubrication of the cylinder walls in a steam reciprocating steam expander as do the self-lubricating composites. It would appear that if the bonded solid films were to be evaluated for this application, they should be applied to the cylinder wall rather than the piston rings in order to provide a greater amount of available lubricant for increased life.

6.2 Lubrication Technology Base - Liquid Lubricants

Lubrication of the large, slow speed reciprocating steam expander operating under steam at relatively low temperatures and pressures has been well established for years. On the other hand, lubrication of the small, high speed and high performance steam expander operating at high steam temperatures and pressures is considerably more complex and may be more complicated than the lubrication of internal combustion engines. Using conventional techniques of lubricating steam cylinders and valves by injecting lubricants directly on the parts and/or by injecting the lubricants into the inlet steam, two separate lubricating systems have been used, one for lubricating the components in the crankcase and one for lubricating the cylinder walls and valves that are operating in the high temperature steam. This is accomplished through the use of a cross-head piston design; both the General Motors SE-101 and SE-124 steam engines employed this concept. A further complication is the fact that the cylinder walls cannot be cooled without decreasing the efficiency of the engine cycle so that the fluid lubricating the hot cylinder walls and valves will come in contact with metal surfaces that are at temperatures of 600° - 800°F. Thermal decomposition products of the lubricant resulting from being in contact with metals at these temperatures will tend to foul up the steam generator tubes unless the lubricant is separated from the condensate water. Frequent lubricant changes would be expected.

Successful use of solid lubricants to lubricate the cylinder wall/piston ring and valve stem/valve guide interfaces will eliminate or greatly minimize many of the problems encountered with liquid lubricants. In fact, it may be possible to utilize a standard trunk piston design. However, due to the fact that the engine cylinders cannot be cooled, as stated previously, even the bulk temperature of the crankcase lubricant may be exceptionally high in comparison to the crankcase temperature in conventional internal combustion engines. Crankcase lubricant bulk temperature may be as high as 300°F unless a special cooling system is employed to cool the lubricant. Even so, the limiting temperature may be the hot spot temperature that the lubricant will see on the bottom surface of the power piston head of the trunk piston design or the internal surface of the piston rod in the crosshead piston design. The average bulk lubricant temperature and maximum hot spot temperatures for the two engine designs in this program are:

	Max. Lubricant Temp., °F	
Engine Design	Bulk	Hot Spot
Trunk Piston	250	540
Crosshead Piston	250	425

Although these temperatures are not unusually high for some conventional petroleum oils, it may be desirable to select a lubricant capable of operating at higher temperature in order to achieve a longer service life or some specific property. It may be possible to provide a lubricant that only requires changing every two years or even the life of the engine as is the case for oils in large steam turbines. A trade-off can be made between the cost of the lubricant and the service life.

Hydrocarbon oils that are extracted from petroleum sources and synthetic lubricants both were considered for use as the crankcase lubricant in the steam expanders. In the selection of a suitable lubricant, it is necessary to: first, select a suitable base and second to select the proper chemical additives that are compatible with the base and the environment.. To be considered a good lubricant for the intended application, the lubricant must have the following important characteristics:

1. Good oxidation resistance
2. High thermal stability
3. Low volatility
4. Suitable viscosity
5. Low pour point
6. Adequate lubricity
7. High ignition temperature
8. Good hydrolytic stability

The last characteristic is particularly important with respect to steam expander applications in that some contamination of the crankcase with water is highly probable, particularly during start-up and shutdown. For this reason, the lubricant needs better rust protection and much better demulsibility than do current motor oils. Additives similar to those used in hydraulic oils, i.e., calcium sulfonates could be used effectively for rust protection. The partial organic acid esters and phosphorus containing acid esters also are used for this purpose. Rust inhibitors usually have a high polar attraction toward metal surfaces and form a continuous protective film over the surfaces. Care must be exercised in the selection of rust inhibitors to avoid corrosion reactions with nonferrous metals or emulsions with water. Chemical additives are rarely effective in improving the hydrolytic stability of lubricants and are seldom used to improve this property; hydrolytic stability is an inherent quality of the base lubricant. Hydrocarbons have excellent hydrolytic stability and major concern is with the additives and with synthetic lubricants. The straight alkyl-chain compounds generally are more easily attacked by water than the highly branched structures. For example, the esters have long chains and generally have poor hydrolytic stability at temperatures of 400°F and for this reason the esters will not be considered as a base for this application. Hydrolysis and water contamination can result in the following reactions: (a) change in physical properties, (b) generation of sludges and other insoluble compounds, (c) decrease in solubility of essential additives, (d) corrosive attack, (e) release of volatile compounds due to chemical breakdown. Obviously all of these changes are detrimental and must be kept to a minimum. It is expected that a complete reformulation of the crankcase lubricant will be required because most of the extreme pressure agents and inhibitors currently in use have little tolerance to water.

Because the bulk temperature of the crankcase lubricant is expected to be higher than the temperatures in the current internal combustion engines, oxidation stability of the lubricant must be superior to the current premium-grade motor oils. Oxidation of the hydrocarbons results in the formation of oil-soluble acid compounds which increases the viscosity of the oil and can be corrosive; oxidation of the synthetics are more complex depending upon the type of structure, i.e., some form volatile gases and do not change the viscosity. Additives to inhibit oxidation include sulfur and phosphorus compounds and the amine and phenolic compounds.

Other additives which are believed to be required in the lubricant are defoamants and anti-wear additives. The most common defoamant is the organo silicon oxide polymer and only small concentrations are required (1 - 20 ppm) to inhibit foam formation caused by the action of the engine crankshaft. Anti-wear additives are required to minimize friction and wear in highly loaded components such as the cam/tappet interface where Hertzian stresses can be as high as 200,000 psi. Extreme-pressure (EP) additives are used and they react with the rubbing metal surfaces to form a lubricating film which protects the metal surface when the lubricating oil film is lost. The most common EP additives are zinc, phosphorus, sulfur and chlorine compounds.

In the survey for a suitable crankcase lubricant for the steam expanders to be tested in this program, one of the more attractive solutions appeared to be the use of a highly refined turbine oil. Many of these oils are the result of extensive development work at the Pennsylvania State University, where processes have been developed for "super-refining" mineral oils to achieve lubricants which have lower pour points, better viscosity-temperature characteristics and higher thermal and oxidative stability. Within the family of super-refined mineral oils, it is possible to use either paraffinic or aromatic-base oils. There are distinct differences in the physical properties of these two types of hydrocarbons. Because of their intermolecular spacing, the paraffinic oils are more compressible, show less change in viscosity with temperature, have lower densities and lower pour points.

The production of super-refined mineral oils is essentially a clean-up operation where the oils are carefully dewaxed, many of the polar impurities are removed, and selective catalytic hydrogenation is used to minimize unsaturation. This process removes not only undesirable impurities, but also naturally-occurring oxidation inhibitors and polar compounds, which are essential for boundary lubrication. Thus, it is necessary to compound these oils with oxidation inhibitors and anti-wear additives. Since thermal decomposition will also proceed by the formation of free radicals, a small percentage of an aromatic disulphide was found to be effective in protecting the oil. The use of the aromatic disulphide gives an added bonus since it will also act as an anti-wear agent by surface reaction between the metal and the sulphide.

The use of blending techniques opens up many possibilities as far as the control of properties such as viscosity and pour point. A wide range of properties can be obtained by blending suitable base stocks to achieve the bulk viscosity and pour point required for the application.

A common turbine oil that has proven to have satisfactory service (∿ 20 years) at temperatures < 250°F in steam turbines and generators is produced by a number of refining companies and is known as Teresso 65, Industrial Oil 61, Tellus 69 or DTE Extra Heavy. The oil has oxidation and rust inhibitors but no EP anti-wear additive which would have to be formulated. Characteristics of the oil are as follows:

Flash point, deg F, min	420
Viscosity, Saybolt Universal, secs 100 deg F, min	540
max	700
Viscosity index, min	85
Pour point, deg F, max	+25
Neutralization value (total acid number) mg KOH/g, max	0.80
Oxidation stability test, hours, min (a)	1000
Sludge, %, max (b)	0.10
Rust prevention test	Shall Pass

(a) Test carried out to total acid number of 2.0 mg KOH/g.

(b) In oil taken from oxidation test.

The major disadvantages to the use of this oil are the relatively high pour point and the low hot spot temperature capability. Although a modified turbine oil would probably be satisfactory for the crosshead piston design, it may not be satisfactory for the trunk piston design because of the anticipated 540°F hot spot temperature.

The synthetic hydrocarbon lubricants have about a 100°F advantage over the mineral oil based turbine oils and appear to have the necessary properties for the application. For relatively long time service, the following generalized maximum service temperatures can be stated:

Base	Max. Bulk Oil Temp., °F	Max. Hot Spot Temp., °F
Mineral Oil	250° - 275° (possibly 300°F)	∿ 500
Synthetic Hydrocarbon	350° - 375° (possibly 400°F	∿ 600

A special synthetic hydrocarbon formulation (XRN-1301-C) was prepared by Mobil Research and Development Corp. for evaluation in the program. The lubricant has suitable oxidation, rust and foam inhibitors and an anti-wear additive. The properties are given in Table 6.2-I. The lubricant has a low pour point (< -65°F) and a viscosity in the SAE 30 range. However, its high viscosity index (149) puts it in the SAE 40 range at elevated temperature. The lubricant appears to have good antifoam performance and good rust protection and although it forms a small amount of emulsion, no emulsion problem would be expected in the engine.

A general comparison of various other synthetic lubricants is made in Table 6.2-II[51]. Known disadvantages of some of these synthetic lubricants include poor hydrolytic stability of the esters and poor viscosities and high pour points of the polyphenyl ethers (in addition to their high cost). Projected costs of the synthetic hydrocarbons are not expected to exceed current premium grade motor oils.

The ESSO Research and Engineering Co. (Govt. Research Lab) is under contract to Steam Engine Systems Corp. to develop a suitable lubricant for the lubrication of cylinder walls in a reciprocating steam expander operating at 1000°F [52]. In their work they found the inhibited blends of hydrorefined paraffinic distillates and residua to be the most suitable

Table 6.2-I

Properties of Experimental Synthetic Hydrocarbon Lubricant[a]

	XRN 1301 C
Gravity, °API (ASTM D-287)	37.4
Specific Gravity	0.837
Pour Point, °F (ASTM D-97)	<-65
Flash Point, °F (ASTM D-92-1)	460
Viscosity: cs @ 210°F	11.05
SUS @ 210°F	63
cs @ 100°F (ASTM D-445)	74.5
SUS @ 100°F	345
cs @ -40°F	34,000
Viscosity Index	149
Acid Number (ASTM D-664-1)	0.08
Base Number (ASTM D-664-3)	0.11
Foam (ASTM D-892)	
Sequence I Tendency, ml	330
Stability, ml	0
Sequence II Tendencey, ml	40
Stability,	0
Sequence III Tendency, ml	380
Stability, ml	0
Rust Test, 48 Hrs, Distilled Water (ASTM D-665-2)	Pass
48 Hrs, Syn.Sea Water (ASTM D-665-4)	Pass
Emulsion Test (ASTM D-1401)	
Total Emulsion, ml	5-10
Time for 3 ml, Minutes	31-33
Time for Complete Break, Minutes	34-37
Water Trace, ppm	40
Surface Tension, Dynes/cm	30.7
Panel Coker, 24 Hrs @ 600°F, Deposit, mgs	27
Mobil B-10 Catalytic Oxidation, 40 Hrs @ 260°F	
% Viscosity Increase @ 210°F	1
Lead Loss, mg	1.4
Sludge	Nil
Neutralization Number Increase	0.2

Table 6.2-I (Cont'd)

Properties of Experimental Synthetic Hydrocarbon Lubricant[a]

Mobil B-10 Catalytic Oxidation, 40 Hrs @ 325°F	
% Viscosity Increase @ 210°F	17
Lead Loss, mg	17.3
Sludge	Nil
Neutralization Number Increase	2.8
Mobil B-10 Catalytic Oxidation, 72 Hrs @ 325°F	
% Viscosity Increase @ 210°F	230
Lead Loss, mg	225
Sludge	Trace
Neutralization Number Increase	12.4
SAE Wear Test, Steel-on-Steel, 30 Min @ 150 lb Load, 250°F Oil Temperature	
Total Weight Loss	0.018 mg
Almen Load, psi	8000
Mobil Thin Film Oxidation Test -600°F (100 = Clean)	93
Mobil Thin Film Oxidation Test -625°F (100 = Clean)	69

(a) Mobil Research and Development Corporation

Table 6.2-II

Performance of Selected Types of Synthetic Oils[51]

Class of Compound	Probable Maximum Thermal Stability Limit, °F	Resistance to Water Degradation	Extreme Pressure Lubricity
Polyglycols	600	Excellent	Good
Phosphate esters	800	Good	Excellent
Dibasic acid esters	600	Good	Good
Chlorofluorohydrocarbons	600	Excellent	Excellent
Silicones	900	Poor	Poor (Steel)
Silicate esters	800	Poor	Fair
Fluoroesters	600	Good	Poor
Neopentyl polyol esters	600	Good	Good
Polyphenyl ethers	600	--	Good
Silanes	700	--	Poor

of the petroleum base oils. A combination tricresyl phosphate (TCP) and ethyl-702 appears to offer the best anti-wear and anti-oxidant properties in the selected bases. The candidate lubricant (designated 2415-50-1) has a higher viscosity (SUS) rating at the standard temperatures of 100°F and 210°F, 663 and 74 respectively, but a lower viscosity index (102) than the Mobil XRN-1301-C lubricant. However, as discussed previously, there is no requirement to lubricate the cylinder walls or valve parts in this program.

6.3 Lubricant Recommendations

6.3.1 Solid Lubrication

In the steam expander design, there are three areas requiring solid lubrication; these include the inlet valve, the piston/cylinder and the shaft connecting the power piston with the crosshead piston. The inlet valve stem will slide in a guide that is exposed to steam and that is sealed by means of compression rings. The shaft connecting the power piston to the crosshead piston will slide in a seal that is exposed to steam on one side and oil on the other. The most severe lubrication problem is the piston/cylinder wall, and either the power piston rings or the liner can serve as the lubricant.

Although the qualitative correlation of the coefficient of friction and wear rate is generally recognized, the exact correlating function between the two is difficult to obtain. Theoretical relationships may be developed which are useful in ranking material and lubricant combinations. However, the reduction of the theoretical relationships to useful design parameters by testing at the expected service conditions is necessary.

Friction effects arise from the tangential forces transmitted across the interface of contact between two bodies. The wear phenomena consists of the removal of material from the contacting surfaces. Adhesion, the ability of contacting bodies to withstand tensile forces after being pressed together, is the primary interacting phenomena. The concept of surface energy makes it possible to develop parameters for making rudimentary predictions of the performance of specific materials under sliding conditions.

While 90% or more of the resistance to sliding arises from the need to shear strongly adherent surface atoms of the contacting materials, other factors have to be considered. A roughness component arises from the necessity of lifting one surface over the other's asperities, which usually contribute about 0.05 to the friction coefficient. Hydrodynamic lubrication can eliminate this component (liquid phase lubrication). A plowing component is present when a hard, sharp surface digs into a softer surface and produces a groove. For really rough surfaces this term may be large, but otherwise it is usually negligible.

Adhesive wear is best combatted with hard materials with low interaction tendencies (or one member of the pair a non-metal), or by use of two metals with a high tendency for interaction reduced by a good boundary lubricant. The use of hard materials does not in itself produce significantly lower wear rates, but the probable low alloying tendency may produce a factor-of-10 change in the wear rate.

Although adhesive wear (sometimes called galling and scuffing) resulting in welding of the surface asperities is usually associated with metallic materials, the adhesion theory is also applied to solid polymers. The difference between the metals and plastics is that the deformation in plastics may be partly elastic over a wide range of load. Also, the flowing term and surface roughness are of greater importance in plastics. The mechanism of wear with carbon-graphites is primarily abrasion which occurs when asperities of two moving surfaces touch and wear fragments are formed from one or both surfaces. Grooves are generally plowed in the "softer" carbon graphite.

Based on the current technology of solid lubrication and wear resistant materials that was discussed in Section 6.1 and in the preceding paragraphs, a list of candidate material combinations for use in sliding contact was compiled for the piston ring/cylinder application and is tabulated in Table 6.3-I. The primary selections were a metal impregnated carbon-graphite material for the piston rings rubbing against an AISI 440C SS or hard chromium plated steel cylinder liner. The same material combination was selected for the back-up design for the piston rod seal in the crosshead piston design. The primary selection for the rod seal was a 15% bronze + 5% MoS_2 filled PTFE material rubbing against a hard chromium plated H-11 alloy rod.

Table 6.3-I

CANDIDATE PISTON RING/CYLINDER LINER COMBINATIONS

	Ring	Liner
1.	Carbon-Graphite, Sb Metal Impregnated[a]	Hard Cr Plate[b] or AISI 440C SS[a]
2.	LC-1C (Cr_3C_2 + 15% Ni Cr) or K-1051 (Cr_3C_2 Cermet)	Carbon-Graphite, Sb Metal Impregnated
3.	Graphite 5890	Graphite 5890
4.	Hast X Porous Structure (35 v/o) + Graphite Filled	Hard Cr Plate[b]
5.	Hast X Porous Structure(35 v/o) + NBS418 + (BaF_2 + CaF_2) Eutectic Filled	Hard Cr Plate[b
6.	LPA-100 (Laves Phase - 65 v/o) + 20% Ni	LC-19 (Cr_2O_3 + Al_2O_3) or Nitrided Steel or Ni-Resist D3
7.	Ni_3 B	LC-19 (Cr_2O_3 + Al_2O_3) or AISI 440C SS
8.	LW-1 (WC + 9% Co)	LC-19 (Cr_2O_3 + Al_2O_3) or Nitrided Steel or Meehanite
9.	LC-1C (Cr_3C_2 + 15% Ni Cr) or K-1051 (Cr_3C_2 Cermet)	LC-19 (Cr_2O_3 + Al_2O_3)
10.	LSR-1 (TiC) over AISI 440C	LC-19 (Cr_2O_3 + Al_2O_3) or Nitrided Steel
11.	K-35 (Ni-Resist) or H.S.31	LC-19 (Cr_2O_3 + Al_2O_3)
12.	K-35 (Ni-Resist)	Hard Cr Plate + MoS_2 Bonded Film[b]
13.	K-35 (Ni-Resist)	Hast X Porous Structure (35 v/o) + Graphite Filled
14.	LW-1 (WC + 9% Co)	Hast X Porous Structure (35 v/o) + (BaF_2 + CaF_2) Eutectic Filled

(a) Primary recommendation for first engine.

(b) Limited to temperature below 700°; for temperatures in excess of 700°F., one of the following liner coatings can be used:

- LC-19 (Cr_2O_3 + Al_2O_3) or LC-4 (Cr_2O_3)
- LC-K (Cr_3C_2 + 15 Ni Cr)
- Chromize

A hard surfacing material (LPA-100 + 20% Ni) and Ni-Resist (D-3 type) cast iron materials combination was selected for the valve stem ring/valve guide components with cold worked Haynes alloy 25 as a back-up material for the valve stem ring.

6.3.2 Liquid Lubrication

The Mobil synthetic hydrocarbon oil designated XRN 1301-C was selected for use in the first single cylinder steam expander test. This oil was specially formulated by Mobil for use in this program. The synthetic hydrocarbon base was selected because of its excellent thermal stability, temperature-viscosity characteristics and low pour point. Appropriate non-metallic inhibitors have been added to the base to provide the necessary oxidation resistance, rust protection, anti-wear properties and defoamant characteristics. Properties of the oil are shown in Table 6.2-I. In general, the XRN 1301-C oil should have approximately a 100°F advantage in use temperature (bulk oil temperature and hot spot temperature) over the oils based on the petroleum base stocks. Probably the greatest disadvantage of the XRN 1301-C oil over the natural hydrocarbon oils is its poorer demulsibility characteristics. However, it is believed the demulsibility of the XRN 1301-C is satisfactory for use in the steam expander.

One of the major reasons for not selecting the inhibited hydrofined paraffinitic oil (2415-50-1) being developed for SES by ESSO Research and Engineering Company for use in the program was the fact that it was not fully developed. Further changes were anticipated to be made in the anti-rust inhibitor; also, a pour point depressant was to be added later to lower the relatively high pour point of + 20°F. Upon completion and evaluation of the final formulation, the 2415-50-1 type oil should be reconsidered for evaluation as the crankcase lubricant. No existing commercial liquid lubricant was found to satisfy the crankcase lubrication segments of the reciprocating steam expander.

7.0 T E S T F A C I L I T Y

7.1 Facility Description

A schematic representation of the steam expander test facility is shown in Figure 7.1-1.

The steam generator was an electrically heated once-through boiler-superheater which was designed to provide 1000 psi, 1000°F, superheated steam at a rate of 400 lb/hr. Three hundred and eight (308) feet of 3/8" OD x .031" wall, Type 316 stainless steel tubing was coiled on a two (2) ft. diameter with a pitch of 1" to form the once-through boiler and superheater section. Calculated pressure drop through the steam generator at rated output was 80 psi. The coil terminates in a small vapor drum which discharged steam through a 1" schedule 40 pipe to a throttle valve and then to the expander. The steam generator was equipped with a pressure relief valve, and was code stamped in accordance with the ASME Power Boiler Code. Design pressure was 1100 psi, and design temperature was 1200°F. Two separate three phase, saturable core reactors (180 KWe) controlled power to the steam generator coil. Automatic and manual modes of power control were provided.

A by-pass control valve and desuperheater permitted checkout of the test facility prior to installation of the test expander. The by-pass valve and expander throttle valve also permitted inlet steam flow control to the expander with a constant boiler heating rate. This form of control was used during startup and shutdown of the expander.

A General Electric Company Model TLC-65/50 HP, 2500/750 RPM D.C. type dynamometer was used to load (up to 65 HP) the steam expander. It was capable of motoring (operating as a DC motor) up to 50 HP. Speed regulation of ± 1% of rated speed was maintained during both modes of

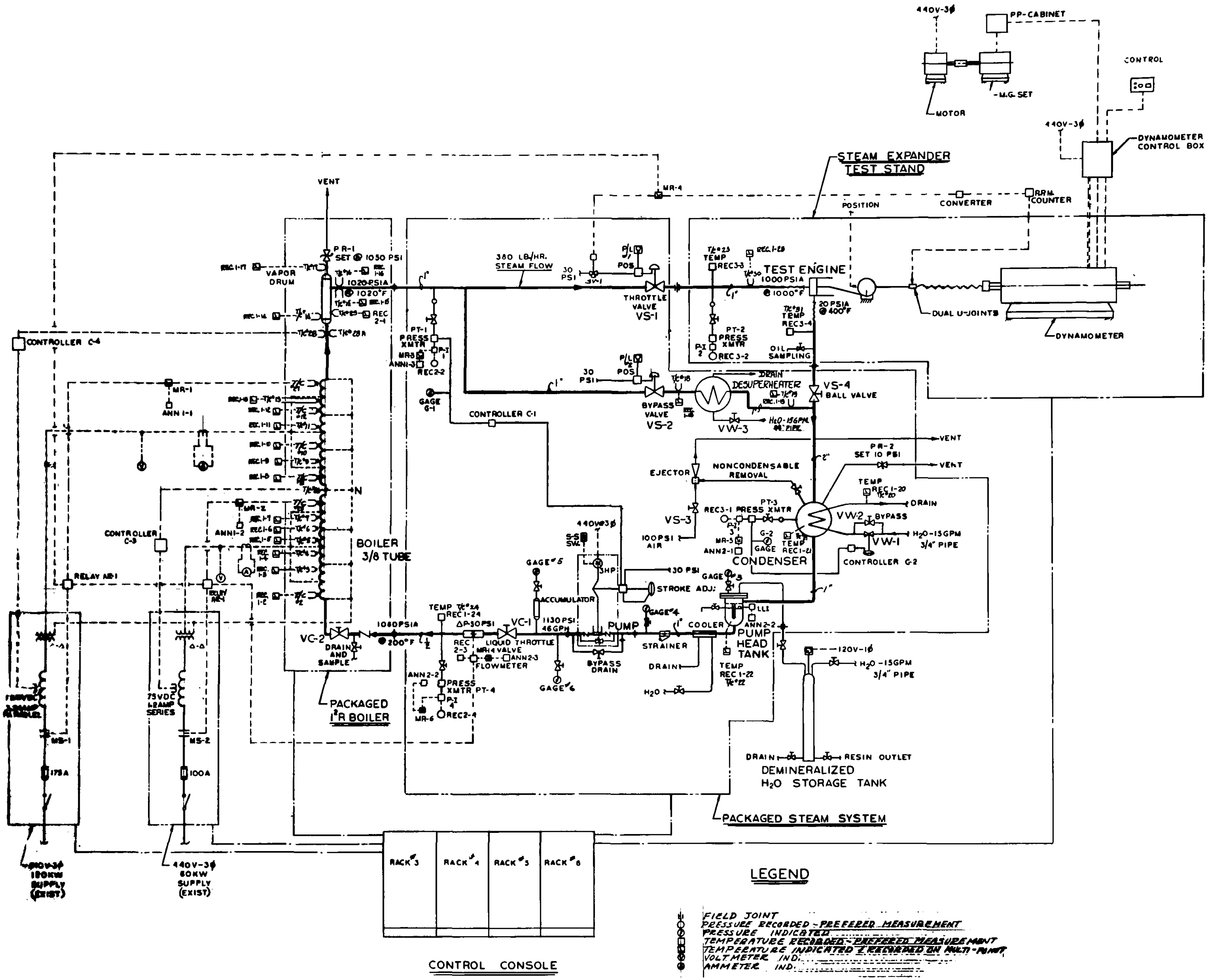


Figure 7.1-1. Steam Expander Test Facility Schematic (GE Dwg. 707E676).

operation. The dynamometer was capable of delivering up to 500 ft.lbs of torque, and was used to rotate the expander during checkout of the expander and during startup on steam operation.

The steam expander was coupled to the dynamometer through a flexible Spicer shaft with universal joints at each end and a splined telescoping center section. A shear pin adapter at the dynamometer shaft connection limited inertial torque loading of the expander by the massive dynamometer in the event of "freeze-up" or other failure which might suddenly stop the expander. The dynamometer was mounted on a separate foundation from the expander, and was isolated from the floor of the test cell for minimum vibration. An electronic (strain gage type load cell) system was used for torque measurement.

The expander exhaust was piped to a water cooled condenser which was capable of 400 lb/hr throughput of steam, and which could maintain 20 psia discharge pressure. Steam discharge from the desuperheater also entered the condenser during by-pass operation.

A water storage tank equipped with a liquid level indicator, filling system and oil removal system received the condensate from the condenser. Demineralized makeup water was supplied to the tank directly from the demineralizer as required during operation. Open loop operation could be conducted by discarding the condensate and by supplying demineralized water for boiler feed. Most of the expander testing was done in that manner.

A plunger type motor driven pump with automatic stroke adjustment supplied the boiler with water. A small heat exchanger ahead of the pump provided cooling to prevent pump cavitation. The pump was capable of providing up to 1.0 gpm at 1200 psi to the boiler. A built-in pressure relief valve prevented an overpressure of the pump. Pump discharge pressure and flowrate were controlled automatically by sensing boiler discharge pressure. Steam flowrate was determined by measuring boiler feed water flow. A turbine type meter was provided for water flow measurement.

A control console located in the control room contained all the operating controls and instrumentation for both the facility and steam expander. All critical parameters were recorded. Protective circuits for overspeed, overtemperature, and overpressure were provided. Temperature, pressure or expander speed above set limits would result in an automatic shutdown of the facility.

Water and steam samples were taken from the test loop for analysis at several locations. An oil/steam sampling valve was provided at the outlet of the expander. An oil/water removal valve was provided near the top of the pump head tank, and a water sampling valve was provided near the inlet to the boiler. All of these valves provided different methods of removing fluid samples to be analyzed for oil or other contaminants.

Figure 7.1-2 shows the I^2R boiler with a portion of its outer insulated casing removed. Figure 7.1-3 shows the high pressure plunger type water pump in the foreground with condenser and water supply tank toward the rear.

Figures 7.1-4 and 7.1-5 show the crosshead expander installed and instrumented ready for testing.

Considerable difficulty was experienced with the Spicer shaft and couplings (reference Figure 7.1-4) between the expander and dynamometer. Several shaft failures were encountered during initial testing of the crosshead expander.

An analytical study of the expander-dynamometer rotating system indicated that the peak torque (for non-resonant condition) on the drive shaft was 219 ft/lb. When assuming that the stiffness or torsional spring constant of the "Spicer" shaft was approximately that of a rigid tube, the calculated natural frequency was above 2800 RPM. However, the exact stiffness of the shaft which included two universal joints was not known, and apparently the stiffness was much lower than the value assumed.

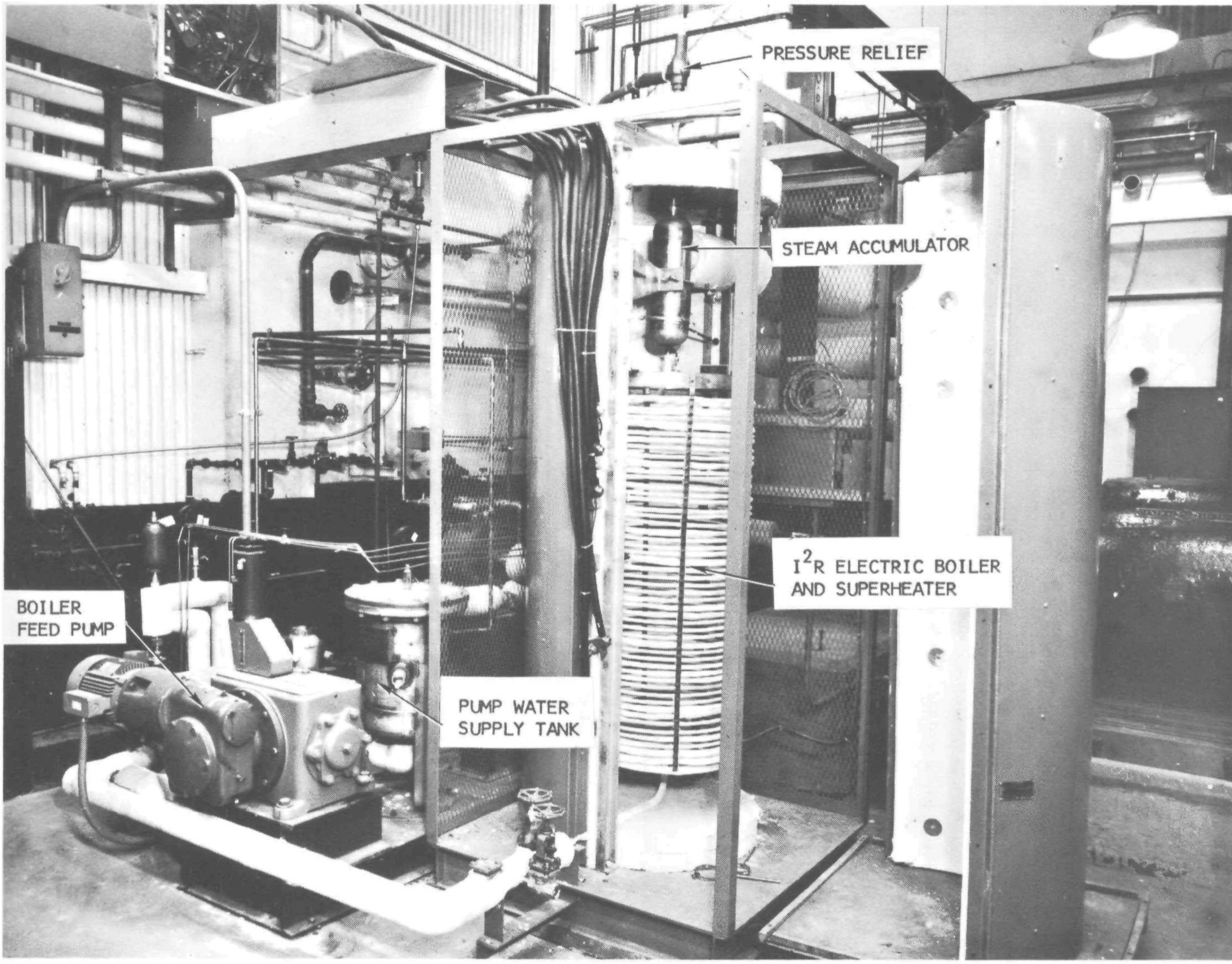


Figure 7.1-2. Single Cylinder Expander Test Facility (P72-2-5F)

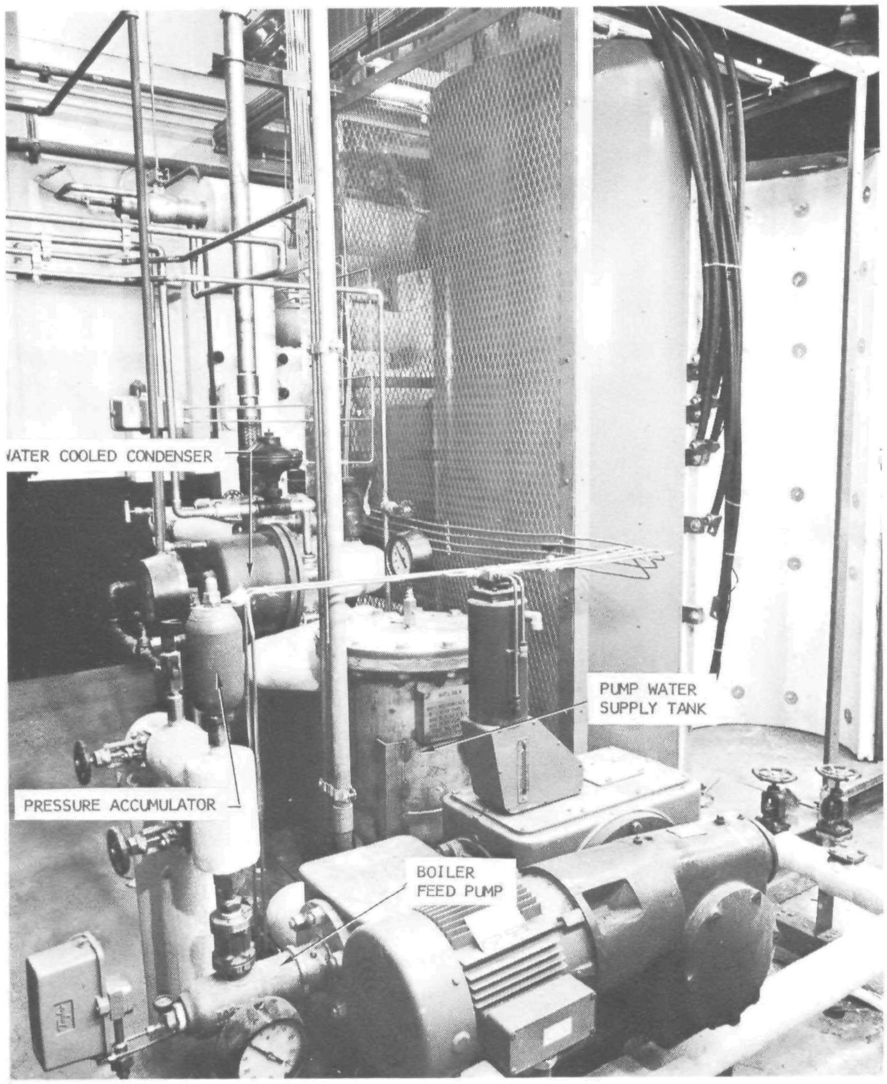


Figure 7.1-3. Single Cylinder Expander Test Facility (72-2-5G)

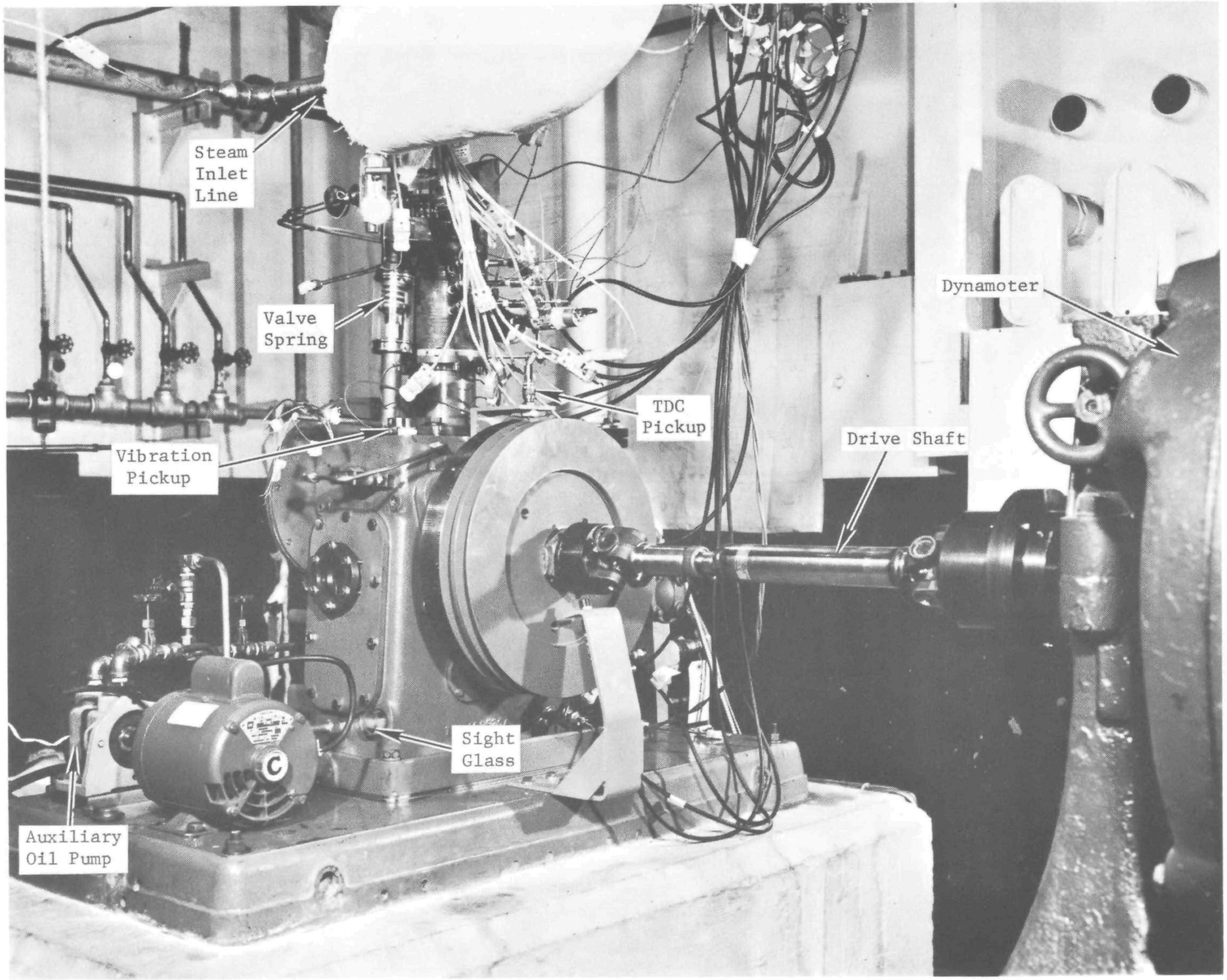


Figure 7.1-4. Installation of Crosshead Expander (P72-4-4M)

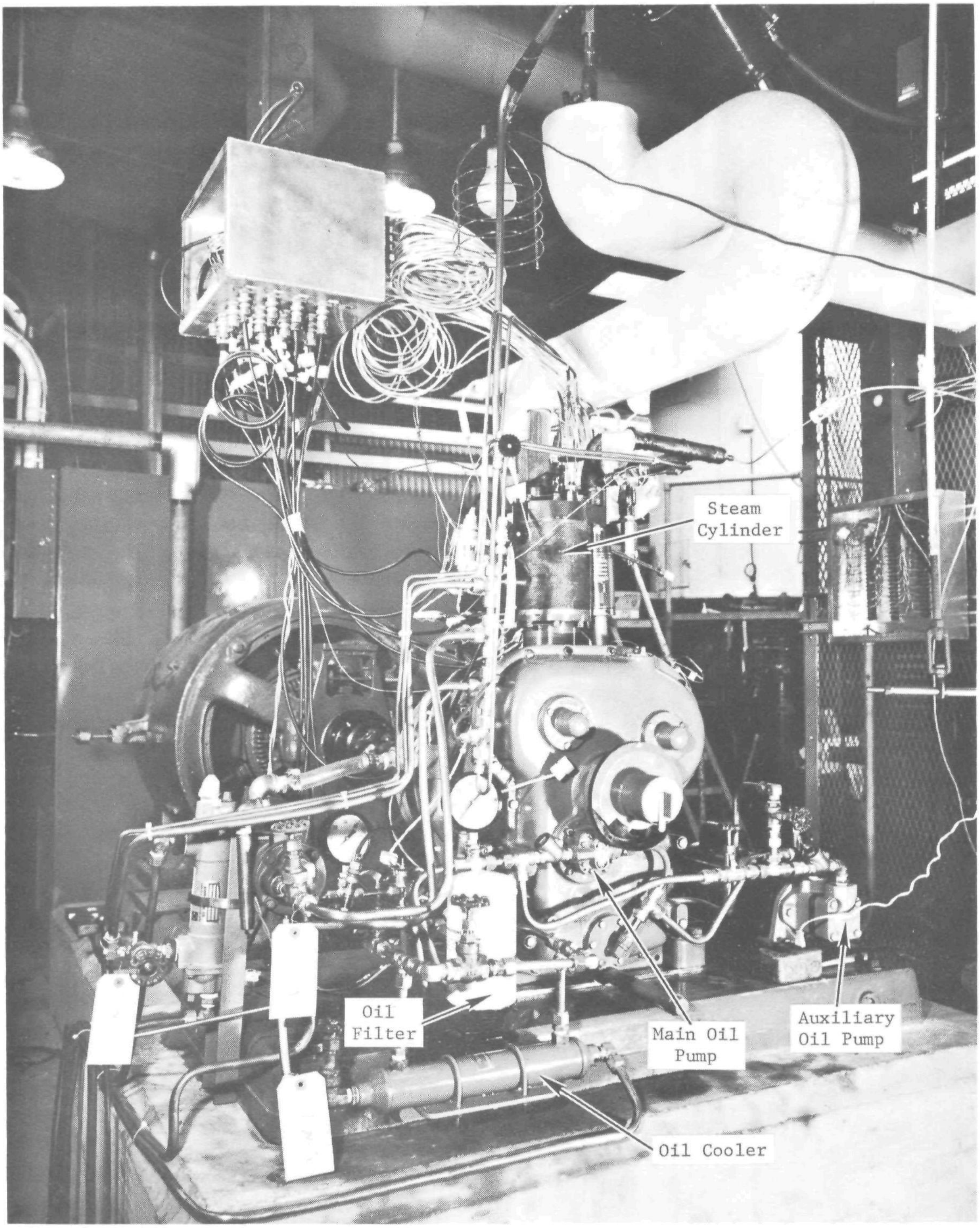


Figure 7.1-5. Installation of Crosshead Expander (P72-4-4N).

Numerous discussions were held with engineering personnel at the Dana Corporation, Waukesha Motor Company, T.B.Woods Company, and others in an attempt to resolve the shaft breakage problem. It was concluded that the rotating mass system had a critical torsional speed within the 0 - 2500 RPM operating range. As a result, a new shaft was designed which included provisions for torsional damping and reduction of shock loading. Also, instrumentation was installed to detect torsional critical speeds. Calculations indicated the torsional natural frequency of the "resilient" shaft to be 610 RPM. Newly installed instrumentation indicated the natural frequency to be in the vicinity of 700 RPM, and therefore operation between 600 and 800 RPM was avoided. Operation at 500 RPM with high inlet steam pressure resulted in excessive temperature rise of the shaft. This was due to oscillations caused by cyclic torque of the expander as it went from power to recompression during each revolution. Therefore, only test points of low cylinder pressure were obtained at 500 RPM.

7.2 Expander Instrumentation

Each expander was well instrumented with 33 thermocouples, 4 pressure transducers, 2 vibration pickups, 2 speed pickups, 2 steam flow sensors, a shaft torque meter, cam push rod force sensor, plus other miscellaneous sensors.

Measurement of steam pressure in the expander cylinder as a function of crank angle was a challenging task. It was generally concluded that the difficulties which were encountered with the cylinder pressure transducer were related to overtemperature or poor temperature distribution throughout the transducer.

Three different types of pressure transducers were tried for cylinder pressure measurement. All three transducers required water cooling. Two of the transducers were strain gage types, and one utilized a piezoelectric crystal. All of the transducers had a temperature limit in the range of 300 - 400°F. If the diaphragm of the transducer exceeded its temperature limit, transducer output response became erratic and sensitivity decreased rapidly. Temperature of the diaphragm for the strain gage transducers was determined from gage resistance measurements during testing. Calibration curves were then used to determine gage sensitivity. Zero shift with temperature change was significant. Therefore, only peak pressure measurements were considered to be reliable. To help reduce the diaphragm temperature, a small amount of argon gas was continuously injected into the vicinity of the gage diaphragm. The argon, a noncondensable gas, reduced the condensation rate of steam on the water cooled diaphragm - thus reducing the temperature of the diaphragm. All cylinder pressure data were obtained by using this technique. The piezoelectric transducer was very temperature sensitive and no valid pressure measurements were obtained with this unit. The best cylinder pressure data was obtained with a Dynisco Model PT49A transducer. This unit contained water cooling passages in the diaphragm.

8.0 TEST RESULTS
CROSSHEAD PISTON EXPANDER

8.1 Component Performance(s)

The crosshead piston expander was first rotated by motoring with the dynamometer on March 30, 1972. The expander was first operated on steam on April 26. Problems arose due to failure of the expander-dynamometer coupling shaft because of shaft torsional vibratory fatigue. A temporary fix allowed the first test point data to be taken on April 27-28, May 2-3, and May 10-11 while awaiting the delivery of a torsionally-damped shaft.

Performance testing of the crosshead expander with graphite rings and with the inlet valve set to open 10° BTDC was completed on June 8. The total operating time of the expander on that date was 87.2 hours. An additional 152.3-hour endurance test at 1000 psia, 1000°F, 1500 RPM was completed on June 16. The total operating time with the original graphite rings was 241.9 hours.

Visual examination and dimensional measurements of the expander components were made at various times during the 87.2 hours of performance and 152.3 hours of endurance testing of the first engine build-up with carbon graphite piston rings. Major inspection of component parts occurred after motoring checkout and after 7.5, 31.3, 52.4, 152.6 and 241.9 hours of test under steam. Following is a summary of the findings of the post test inspections performed on crosshead expanders.

8.1.1 Camshaft/Valve Lifter (Tappet)

The AISI 8620 camshaft exhibited light pitting on the carburized lobe adjacent to the contact area. The noncritical areas of the chill-cast Cr-Mo cast iron tappet also showed evidence of pitting which appears to be caused by a form of pitting corrosion. However, the cam lobe/tappet contact surfaces were highly polished and no evidence of pitting or other forms of damage were observed.

Early in the test program during the checkout of the expander, severe damage to the tappet/cam lobe surfaces was observed after only 2 hours of motoring without steam. In this case the materials combination was a cobalt base cast Stellite tappet against a carburized AISI 8620 cam lobe. However, inadvertently a cast Stellite 6B alloy tappet with a hardness of Rc 38-39 rather than the intended Stellite Star J alloy with a hardness of Rc 60-61 was used. The failed Stellite 6B tappet had the classical appearance of spalling (pitting), i.e., irregular shaped holes apparently caused by rupture of the metal below the surface, and scuffing due to a galling action between the tappet and cam surfaces. Visual examination of the carburized AISI 8620 steel cam surface revealed a buildup of metal on the nose and on each side of the lobe of the cam surface. The metal transfer to the cam surface is believed to be caused by a galling action between the tappet and cam surface as a result of a combination of a high tendency for adhesion between the two surfaces and a low bulk fracture strength of the Stellite 6B.

The chill-cast Cr-Mo cast iron tappet with a steam tempered black oxide (Fe_3O_4) antichafing coating was selected for the second engine buildup in preparation for performance testing. The chill-cast tappet was selected on the basis that experience has shown that the chill-cast iron tappet generally is superior to hardenable cast iron against hardened steel cam surfaces.[53] The major disadvantage of chill-cast cast iron tappets is its greater tendency to failure by spalling (pitting) especially in the presence of certain EP additives in the oil[54,55,56,57]. However, since the tappet material was changed to the chill-cast Cr-Mo cast iron and an auxiliary oil pump was installed to insure pre-startup lubrication no further cam lobe/tappet damage has occurred other than the slight pitting corrosion of non-critical areas.

8.1.2 Inlet Steam Valve/Housing

The H-11 alloy inlet steam valve stem assembly required a light force in order to remove it from the cast Ni-Resist Type D-3 alloy housing. The cold worked (Rc 49) Haynes alloy No. 25 upper stem compression seal rings had worn a shallow groove (~0.002") in the ID surface of the housing; no significant wear of the housing was observed at the location of the bottom seal rings. Possibly the application of a wear-in coating on the Ni-Resist housing (or rings) would reduce the observed wear by providing lubrication until the Ni-Resist surface has a chance to work harden and achieve a harder and more wear resistant surface. Evidence of some pitting also was observed on all components of the housing. However, in spite of the slight pitting and wear in the housing, the steam leakage past the inlet valve stem seal was on the order of only 0.5% of the total steam flow at steam conditions of 1000°F/1000 psi, Figure 8.1-1.

Other components of the inlet steam valve assembly were in excellent condition. The nitrided spherical end of the H-11 alloy push rod and the spherical Stellite 6B seat were unpitted and highly polished. Both the Stellite 1 and Stellite 6 alloy valve seat and valve facing respectively were in good condition and were reuseable. Although the low alloy steel inlet valve spring relaxed approximately 1/8" in the first 52.4 hours of testing, no further problems were encountered with spring relaxation after the addition of oil jets to the housing to improve cooling of the spring.

LPA 101 plasma spray-coated 17-7 PH alloy compression seal rings were incorporated in the inlet steam valve assembly for the first 52.4 hours of engine testing. Difficulties were encountered with the LPA 101 coating during fabrication of the rings and during the period of testing due to frequent chipping of the coating. An increasing rate of steam leakage through the inlet valve stem (Figure 8.1-1) was observed over the initial 52.4 hours of testing as a result of continued chipping of the coating. Upon replacing the LPA 101 coated 17-7 PH rings with the cold worked Haynes Alloy No. 25 rings, no further increase in the leakage rate occurred and after a period of time the leakage rate decreased to the $\sim$ 0.5% value. The LPA type material (laves phase in cobalt matrix) may have potential as a ring material but not as a coating. Possibly a cast ring of solid LPA alloy may warrant evaluation.

8.1.3 Recompression Valve

All the materials of construction of the recompression valve assembly are identical to the materials used for the fabrication of the inlet steam valve with the exception of the spring. After (100 hours) of operation

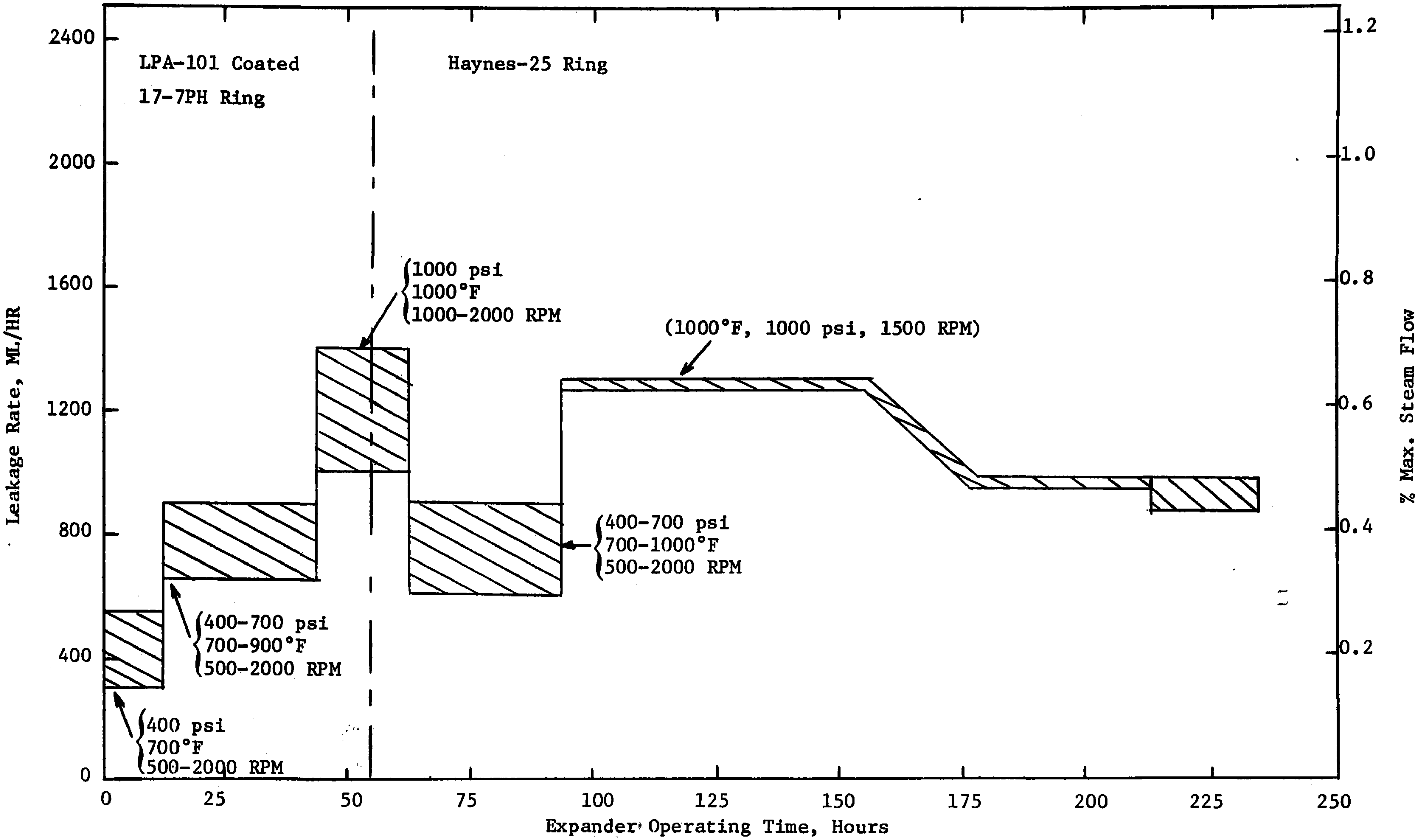


Figure 8.1-1 Inlet Valve Stem Steam Leakage Rate

the low alloy steel recompression valve spring relaxed nearly 0.100" and was replaced with a spring fabricated from Inconel X-750. Although even the Inconel X-750 recompression valve spring had relaxed 0.040" at the end of the final 141.9 hours of test, it still was reuseable.

All other components of the recompression valve were in excellent condition.

8.1.4 Piston Rings/Cylinder Liner

At the end of the 241.9 hours of testing, the Sb impregnated carbon-graphite power piston rings (CC-5A material) exhibited severe wear and damage. Each ring segment was minus at least one lap joint and approximately one half of each ring segment located in the bottom ring groove was missing. The pressure balancing circumferential grooves had all but disappeared from the rubbing surfaces of those segments in the upper ring groove. From probe measurements of the ring groove depth with the rings installed in the expander and actual micrometer measurements at the time the expander was disassembled after 52.4 hours and at the completion of the testing of 241.9 hours, it appears that the wear of the carbon-graphite rings is linear out to about 153 hours. Beyond 153 hours, the wear rate of the top ring increased and the wear rate of the middle and lower rings appeared to decrease. The wear data are shown in Table 8.1-I and Figure 8.1-2. Photographs of the rings after 52.4 and 241.9 hours on test are presented in Figures 8.1-3 and 8.1-4.

The Inconel X-750 ring spring of the power piston bottom ring groove exhibited substantial wear at the end joint. This ring spring has shifted radially outward due to the missing graphite segments and had scored the I.D. of the Type 440C SS cylinder liner, particularly near the exhaust ports. Visual examination of the cylinder liner after 152.6 hours of testing revealed no scratches or damage of any kind.

8.1.5 Power Piston Head

The Cr-Mo-V alloy power piston exhibited little, if any, wear except for light scratches in the lower ring groove surfaces caused by the Inconel X-750 ring spring.

Table 8.1-I

PISTON RING WEAR

(Carbon-Graphite Grade CC5A)

Engine Hours On Steam	Ring Groove Depth, In.[a]	Ring Wear, In.
0	0.071[b]	0
7.5	0.071	0
31.3	0.064	0.007
52.4	0.060[b]	0.011
152.6	Top 0.037	0.034
	Mid 0.039	0.032
	Bot	
241.9	Top 0.000[b]	∿0.071
	Mid 0.028[b]	0.043
	Bot 0.036[b]	0.035

[a] Tolerance ± 0.001 In.

[b] Micrometer Measurement; all other measurements by probe.

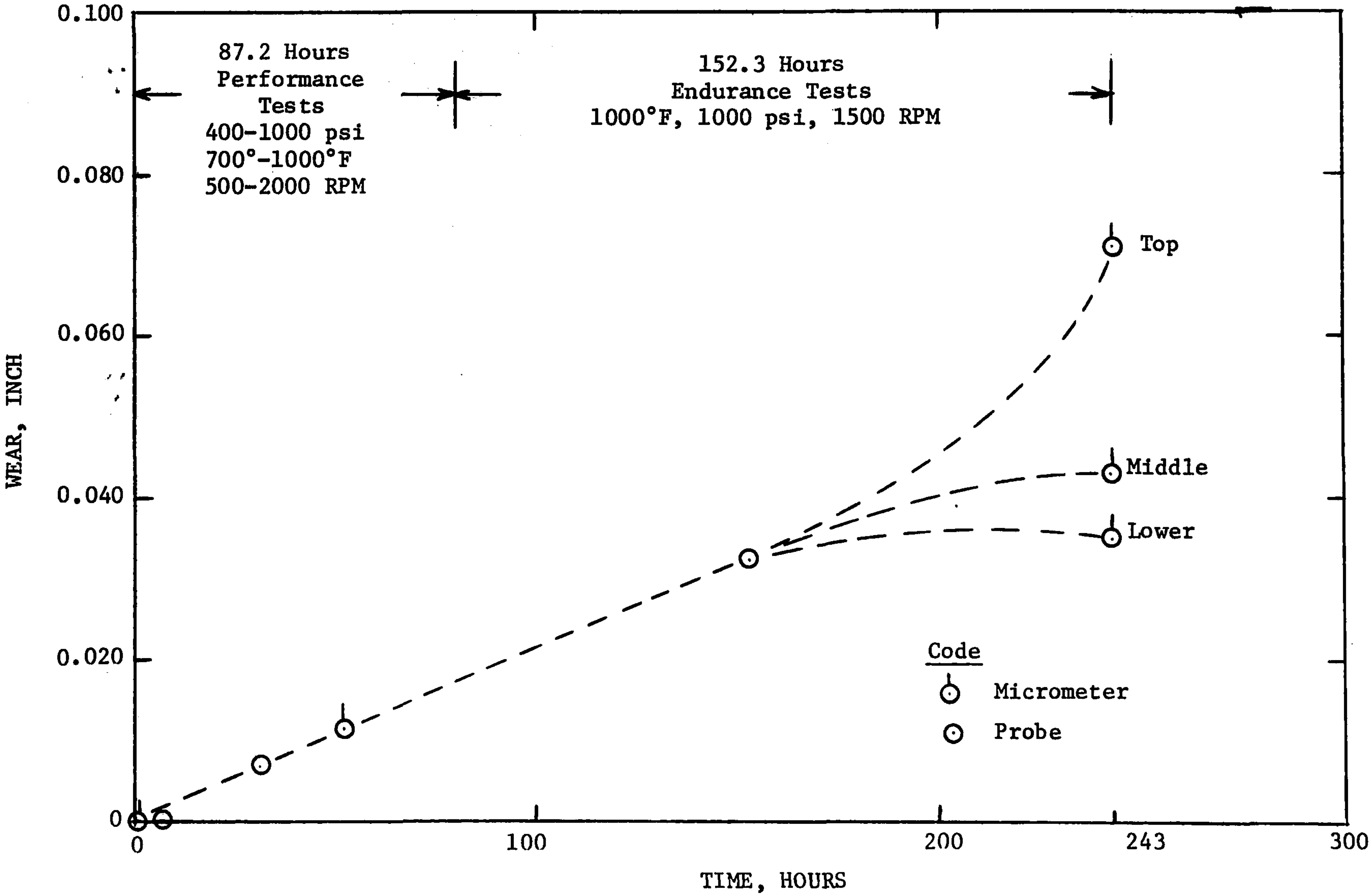


Figure 8.1-2 Piston Ring Wear at Middle of Segments
(Carbon Graphite Grade CC5A).

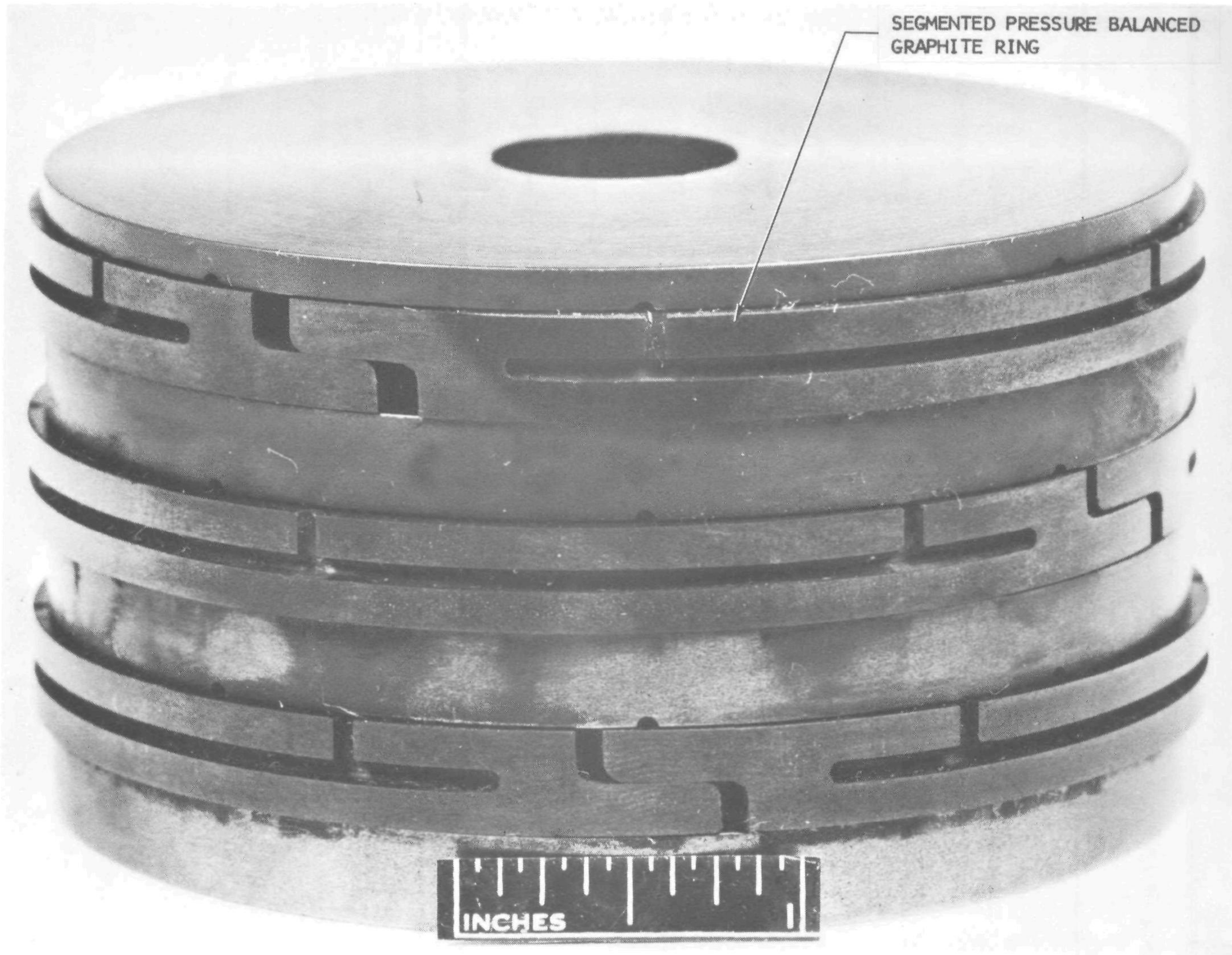


Figure 8.1-3. Crosshead Expander Carbon-Graphite Rings (CC-5A) after 52.4 hours on test (P72-5-4A)

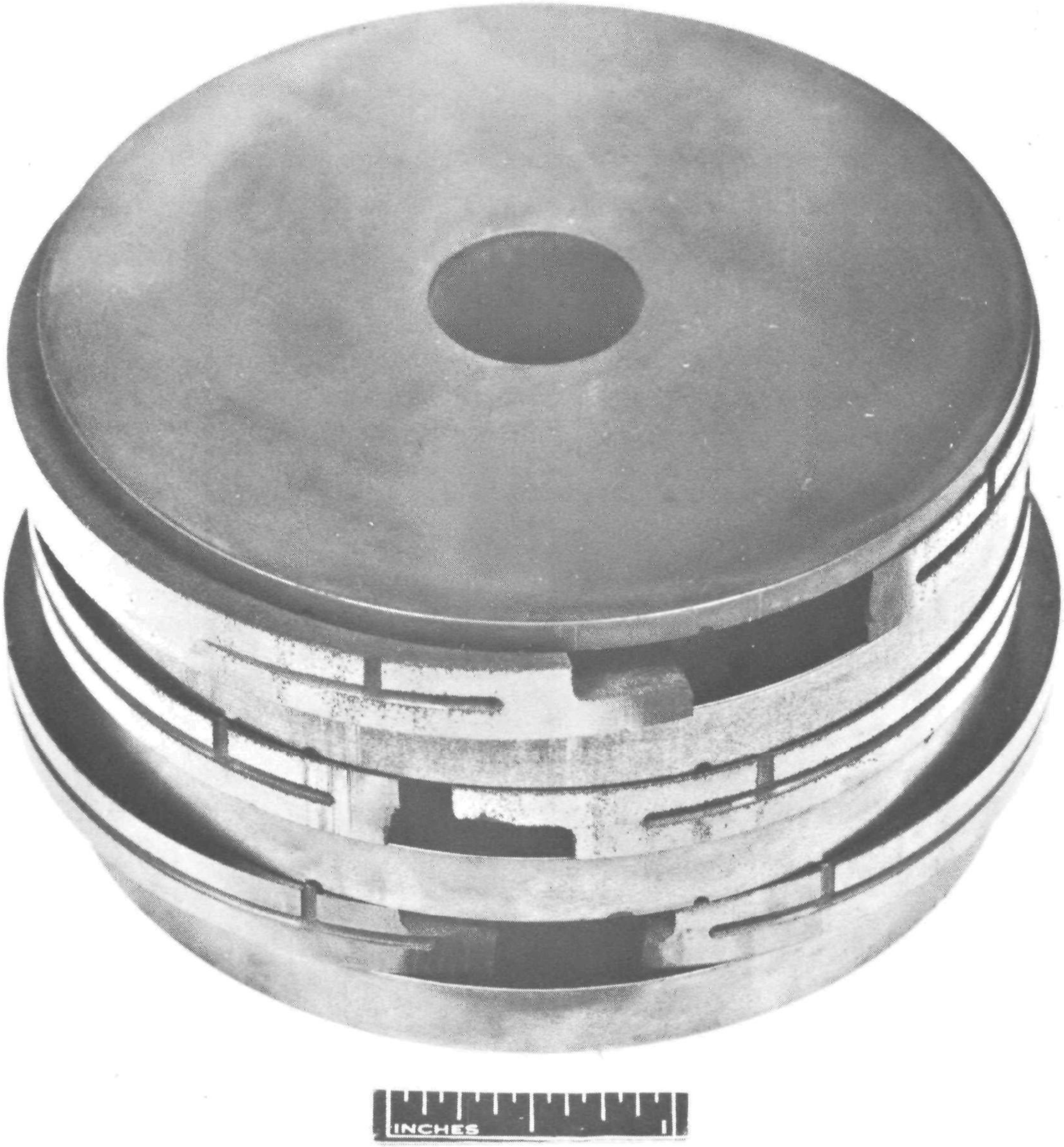


Figure 8.1-4. Crosshead Expander Carbon-Graphite Rings (CC-5A) after 241.9 hours on Test (P72-6-3L)

8.1.6 Power Piston Rod Seal

The power piston push rod chevron seal was in excellent condition. The seal was fabricated from 15% bronze + 5% MoS_2 filled PTFE and inspection of the seal after completion of the 241.9 hours of test revealed no measurable wear. The mating surface was a hard chromium plated H-11 steel.

During the initial stages of checkout testing without steam, traces of oil entered the expander steam chamber through piston rod seals. It was concluded that the oil was bypassing the seal by a hydrodynamic pumping action. By modifying the mounting of the chevron seal to take advantage of the hydrodynamic action of the seal, the problem was solved. The chevron seal was installed with a floating housing. An additional wave spring was added to the seal assembly and all wave springs were placed on the oil side of the seal. With this modification to the seal, oil leakage into the steam expander condensate was maintained at less than 4 parts per million as shown by Figure 8.1-5.

Leakage of water past the piston rod seal into the crankcase also was monitored at intervals during the performance and endurance testing. Total water leakage was determined by 1) analyzing the % water content in the oil (emulsion), 2) measuring the quantity of water (demulsified, condensed steam) in ml that accumulated in the lower section of the crankcase, and 3) measuring the quantity of water that boiled off through the crankcase vent. These data are given in Table 8.1-II. Examination of the data show that relatively little water has leaked into the crankcase. The water content of the oil after the final 89.3 hours endurance test analyzed only 0.16%; however, even the $\sim$1.0% water content in the oil after the first 63.5 hours of endurance testing would not be expected to be detrimental to the lubricating properties of the oil.

8.1.7 Crosshead Piston

Both the aluminum alloy crosshead piston and the mating nitrided Type 304SS lower cylinder wall exhibited little, if any, wear. Only very shallow pitting was observed on the lower position of the nitrided ID surface of the cylinder.

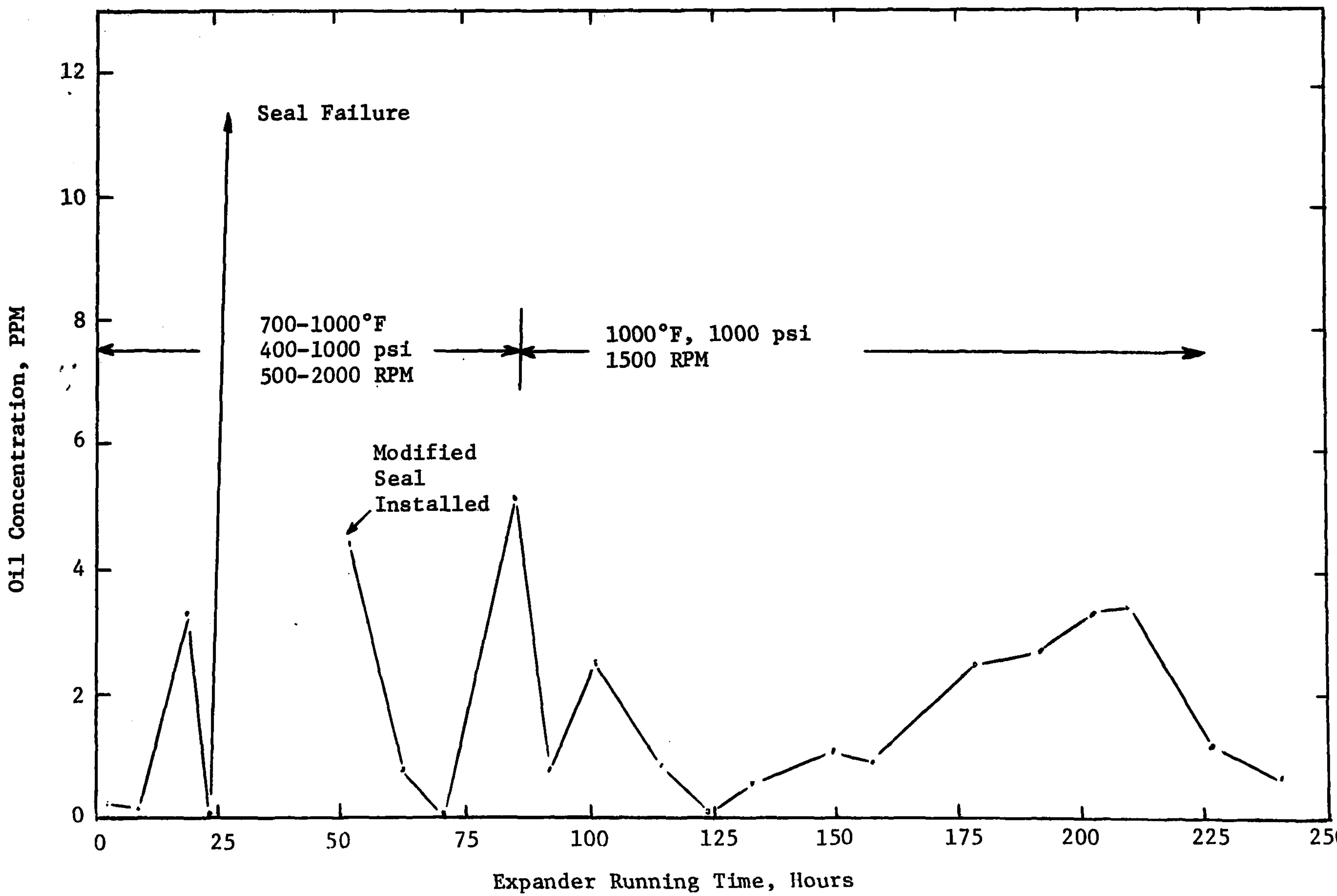


Figure 8.1-5 Oil Concentration in Steam Condensate.

Table 8.1-II

Water Leakage into Crankcase

Total Engine Hours on Steam	Water Content Crankcase Oil, %	Water Drained from Crankcase before Analytical Sample ml	Vapor Boil-Off Crankcase Vent ml/hr
0	0.0008 (8ppm)	-	-
12.9	0.028	-	nil
31.3[a]	0.60	5	4.5
152.6[b]	1.00 (1 ml)	113	6.3
241.9[c]	0.16	50	1.1

[a] Piston rod seal failure; crankcase oil drained 2050g oil, 12.3g water.

[b] 63.5 hours on endurance test, all water drained prior to start of 63.5-hr run.

[c] 89.3 hours on endurance run, all water drained prior to start of 89.3-hr run.

8.1.8 Wrist Pin Bushing

The diametral clearance of the wrist pin/bushing assembly increased approximately 0.004". The high load areas of the wrist pin exhibited burnished markings but to no discernable depth.

8.1.9 Other Components

All other components of the expander were in excellent condition and reuseable for further testing.

8.1.10 Crankcase Lubricant

A sample of the XRN 1301C oil which had accumulated 187.7 hours of operation was drained from the crankcase after the completion of the 152.3-hour endurance test and sent to Mobil Research and Development Corporation for analysis. The results of the analyses are presented in Table 8.1-III. The data show relatively little change of the used oil from new oil and no serious oxidation or degradation. The slight viscosity increase is possibly due to the significantly higher water content of the used oil. The increase in neutralization number reflects some slight oil oxidation that was confirmed by differential infrared analysis. The ash content is very low and shows expected traces of iron wear metal and possibly dirt as indicated by the silicon content. Overall, the XRN-1301C oil performed well and provided satisfactory lubrication with little evidence of degradation over 187.7 hours of operation.

Buildup of the crosshead expander with Cr_3C_2 coated Inconel-X piston rings was completed on June 23. The expander was run for about one hour at 400 psig and 700° F inlet steam conditions on June 23 for preliminary checkout. No difficulty was encountered during the checkout test, and visual inspection of the piston rings by observation through the exhaust port revealed no ring or cylinder liner damage.

Performance testing of the crosshead expander with the Inconel-X rings began on June 26, with initial performance being satisfactory. However, after approximately six hours, expander performance began to decay, as indicated by poor cylinder recompression and increased steam consumption. The expander was shutdown and the piston rings were inspected by viewing

<u>Table 8.1-III</u>

Change in Properties of XRN-1301C Oil
After 187.7 Hours of Engine Operation

Property	New	Used
Viscosity @ 100 cs	75.45	80.20
@ 210 cs	11.07	11.60
Viscosity Index	147	147
Neutralization Number	0.06	0.5
Water Content, ppm	38	431
Ash, wt %	0.001	0.006
Metal Contents, ppm by Emission Spectrograph		
Al	-[a]	0
Cr	-	0
Cu	-	0
Fe	-	78[b]
Mg	-	0
Ba	-	0
Si	-	4

Differential infrared analysis of new oil vs. used oil shows loss of abcut 10 - 20% of antioxidant and slight oil oxidation.

---

[a] Insufficient ash for determination

[b] Computer extrapolation value might be slightly high

through the exhaust port. Extensive ring wear and cylinder wall roughness were observed. Upon disassembly, the Cr_3C_2 coated Inconel-X rings were found to be worn approximately 60 mils which included the 3 to 6 mil Cr_3C_2 outer coating. The hardened Type 440-C liner contained a number of axial grooves in the order of 1 to 3 mils deep, and there were also axial streaks of metal buildup on the cylinder wall 2 to 3 mils thick. The stepcut tabs on all rings were broken off, and 20 to 30% of each ring was missing as shown by Figure 8.1-6. All other expander components were virtually undamaged.

Oil leakage into the steam condensate was high initially, i.e., 55 ppm after 3.2 hours; however the oil concentration rapidly decayed to 25.2 ppm after 4.2 hours and 4.0 ppm after 5.4 hours. It is assumed that some oil had been trapped in the system prior to engine start-up which resulted in the high initial oil content in the condensate. The piston rod seal was functioning as expected and further testing would have resulted in oil concentration of <4.0 ppm.

8.2 Thermodynamic Performance

8.2.1 Thermodynamic Performance - Graphite CC-5A Piston Rings

Test results for the crosshead expander are presented in Table 8.2-I.

Figure 8.2-1 shows the increase in brake horsepower with speed and with increasing capacity of the steam to perform work as pressure increases. More efficient breathing and lower friction power losses at low speeds cause the deviations from straight lines. The effects of greater blowby at lower speeds and higher pressures are also present. A reference line of theoretical indicated power at 1000 psia and 1000°F is shown (assumes no blowby). The brake data compare favorably with the prediction, since the brake power would be expected to be less than the indicated power due to blowby and friction losses.

The brake mean effective pressure is shown in Figure 8.2-2 as a function of speed and inlet steam conditions. It reflects the same effects which the power curves show. The theoretical mean effective pressure at 1000 psia and 1000°F is shown as a dashed line. The difference between the

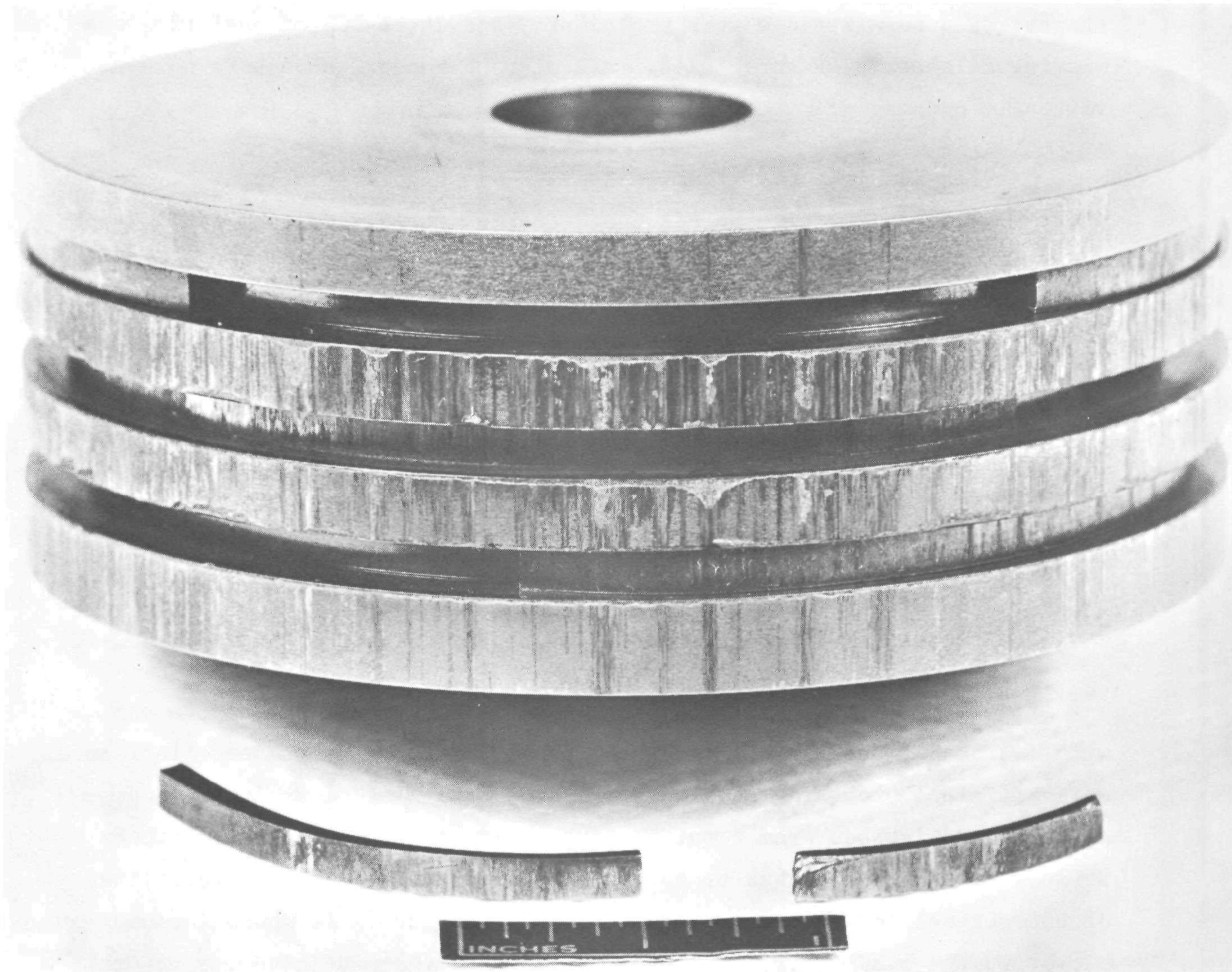


Figure 8.1-6. Crosshead Expander Cr_3C_2 Coated Inconel-X Rings After Approximately 10 Hours of Testing (P72-6-3N)

Table 8.2-I

Crosshead Expander Test Data
May 10 - June 7, 1972

Test Point	Brake Torque lb-in	BMEP psi	Speed rpm	Brake H.P.	Cond. Press. psia	Inlet Press. psig	Inlet Temp. °F	Steam Rate lb/hr	Specific Rate lb/hp-hr	Operating Time hr
101E	636	115	475	4.79	21.0	380	604	92	19.3	
101G	509	92	500	4.04	19.8	403	700	110	27.2	
102E	530	95	850	7.15	22.0	400	640	102	14.3	
102G	551	99	1000	9.15	20.4	435	700	-	-	
102H	509	92	1000	8.08	19.5	408	694	150	18.6	
103H	403	73	1500	9.59	19.5	405	700	170	17.7	
104H	288	52	1995	9.13	19.5	412	706	167	18.3	
105E	636	115	546	5.51	22.0	400	634	74	13.3	
105H	594	107	500	4.71	19.5	405	694	91	19.4	
106E	1356	244	590	12.69	21.6	700	682	176	13.9	
106H	1272	229	500	10.09	19.8	700	712	175	17.3	
107H	1060	191	1008	16.95	19.8	700	688	280	16.5	
108H	933	168	1495	22.12	20.0	698	688	340	15.4	
117H	594	107	1001	9.43	19.7	440	1012	105	11.1	
119H	254	46	1992	8.04	19.6	400	988	127	15.9	
122H	1102	199	999	17.47	19.6	705	988	187	10.7	
123H	933	168	1500	22.20	19.9	700	988	235	10.6	
124H	721	130	1995	22.82	20.0	690	1000	282	12.4	
127F	1590	286	1000	25.23	23.0	912	1020	415	16.4	
127H	1823	328	1001	28.95	19.5	1020	1010	340	11.7	
128H	1442	260	1500	34.31	19.5	990	1008	402	11.7	
129G	1102	199	2000	34.97	23.7	950	940	415	11.9	
129H	1208	218	2000	38.35	20.0	990	1035	430	11.2	
129HA	1187	214	2000	37.67	19.8	1000	1040	432	11.5	87.2

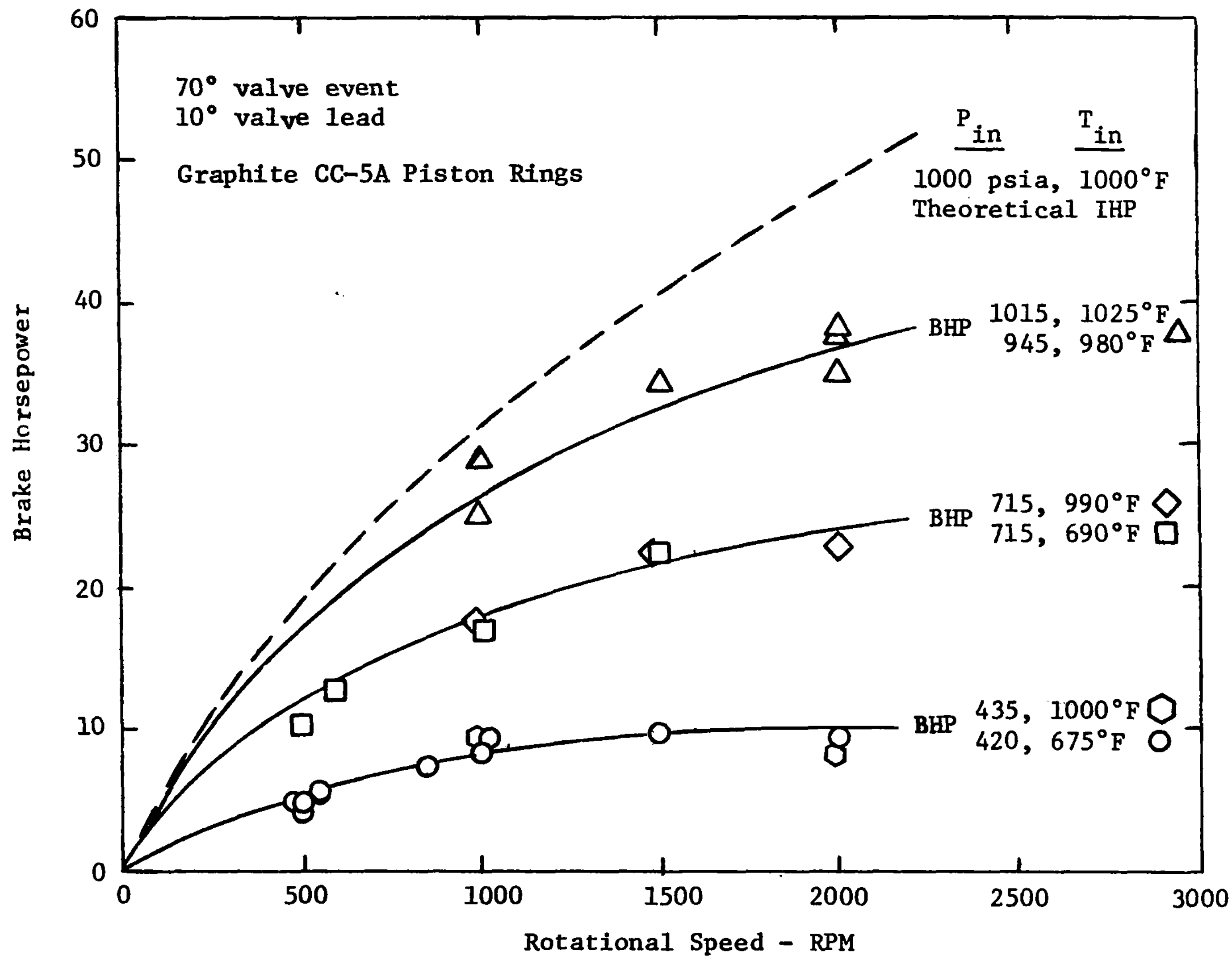


Figure 8.2-1. Crosshead Expander Performance

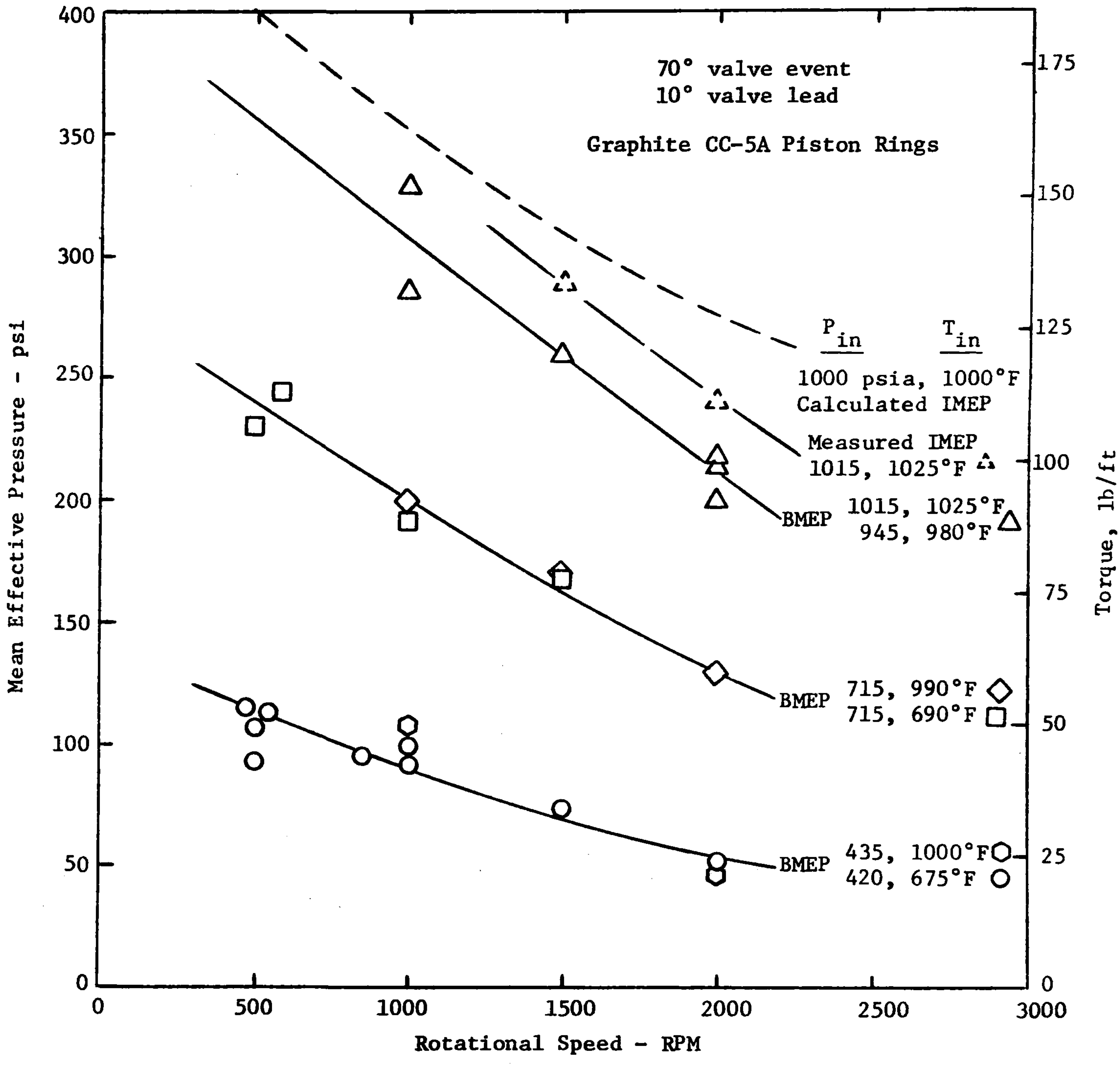


Figure 8.2-2. Crosshead Expander Performance

measured and theoretical IMEP is chargeable to the theoretical assumption of no blowby. The difference amounts to about 6 hp or 13 percent.

The difference between the IMEP and BMEP curves at 1015 psia and 1025°F represent frictional power losses of 3.9 hp at 1500 rpm and 4.7 hp at 2000 rpm. These losses are almost exactly what is expected from two-stroke diesel engines operating at the same IMEP[10]. The corresponding mechanical efficiencies are 0.90 and 0.89.

The specific steam consumption is shown in Figure 8.2-3 as a function of speed and inlet conditions. The influence of speed is strongest at 500 rpm, but the SSC is practically invariant at the higher speeds. A strong influence of the steam enthalpy is also seen. The influence of other parameters upon the brake SSC curves is not apparent. Friction and blowby would contribute to the difference between the 1005 psia, 1005°F curve and the theoretical indicated SSC.

The engine efficiency as a function of speed and inlet conditions is shown in Figure 8.2-4. The definition of engine efficiency as shown by the figure is:

$$\eta_e = \frac{Q_{out}}{Q_{in}} = \frac{BHP}{w\Delta h}$$

The theoretical indicated efficiency is shown for reference.

One cylinder pressure-volume plot is shown by Figure 8.2-5. Since this curve is drawn from a small oscilloscope photograph, such as shown by Figure 8.2-6, absolute accuracy is not expected, but the correlation between predicted and actual pressures appear good. The recompression curves match very closely until shortly before TDC. The match is also quite good from 7 in^3 (approximate point of inlet valve closure) through expansion. The variation through this interval is probably primarily due to blowby.

Following completion of approximately 87 hours of performance testing, the crosshead expander was held at 1500 rpm with 1000°F and 1000 psia steam inlet conditions for an additional 150 hours. Figure 8.2-7 shows the decay of shaft horsepower, and Figure 8.2-8 shows a steady

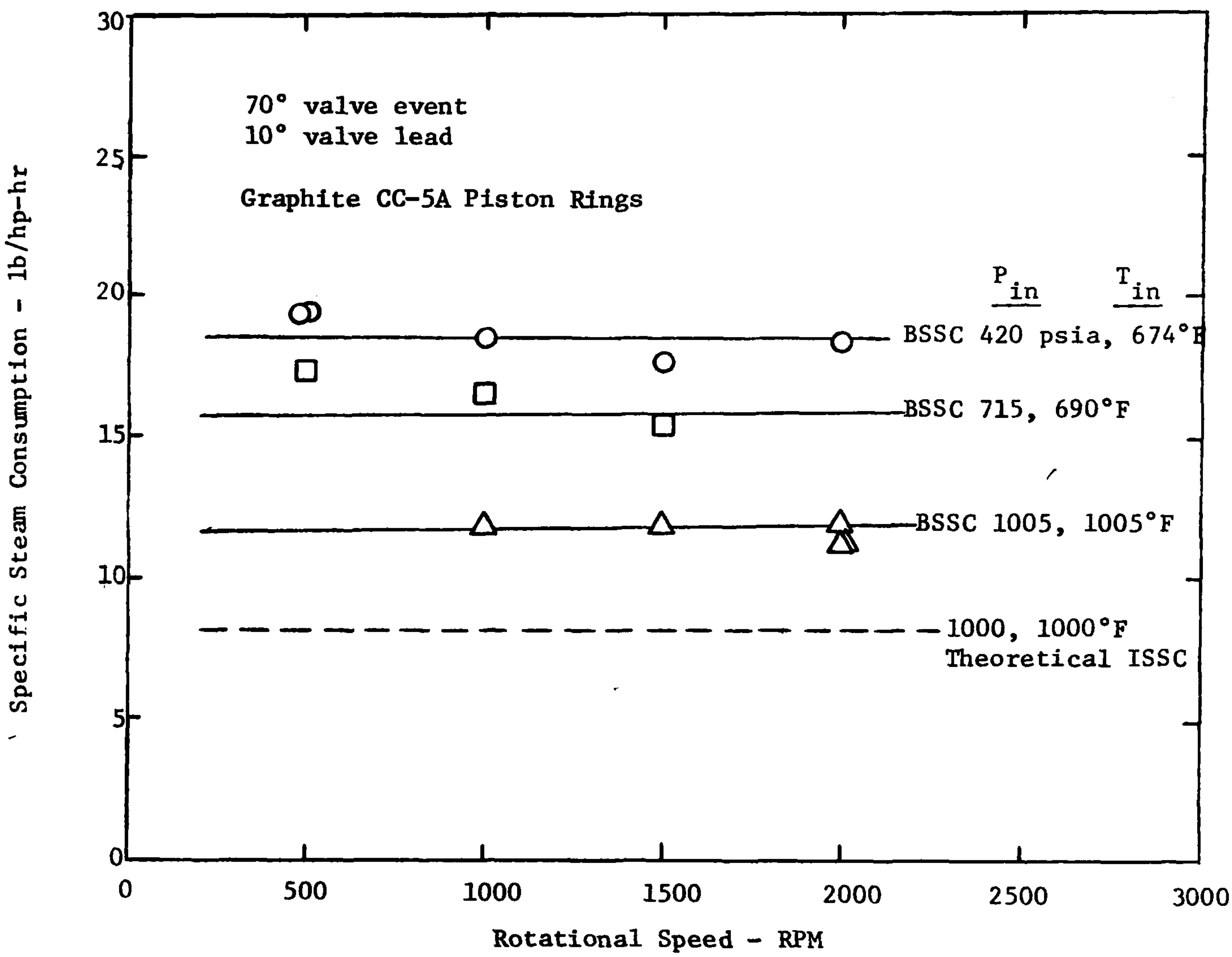


Figure 8.2-3. Crosshead Expander Performance

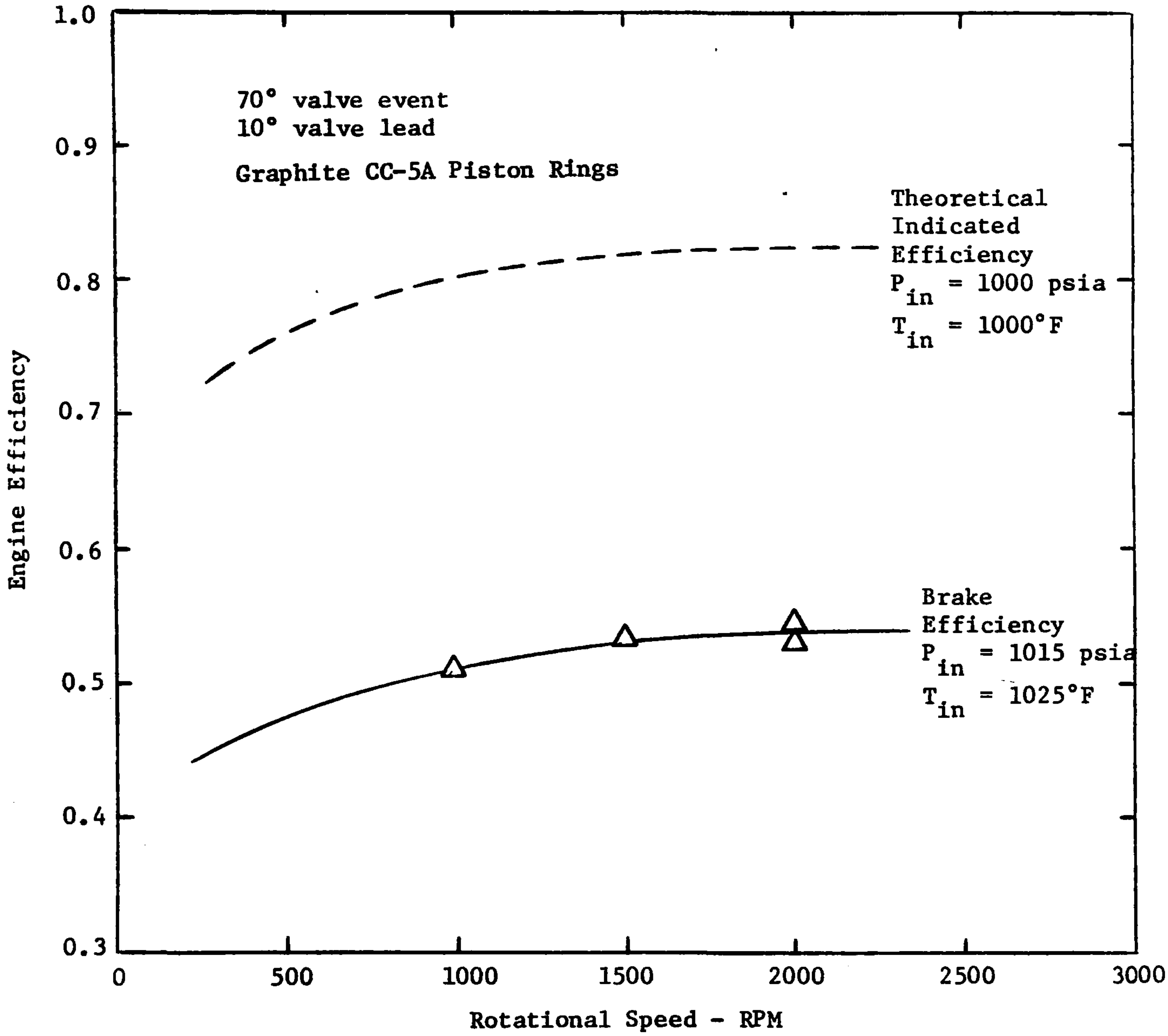


Figure 8.2-4. Crosshead Expander Performance

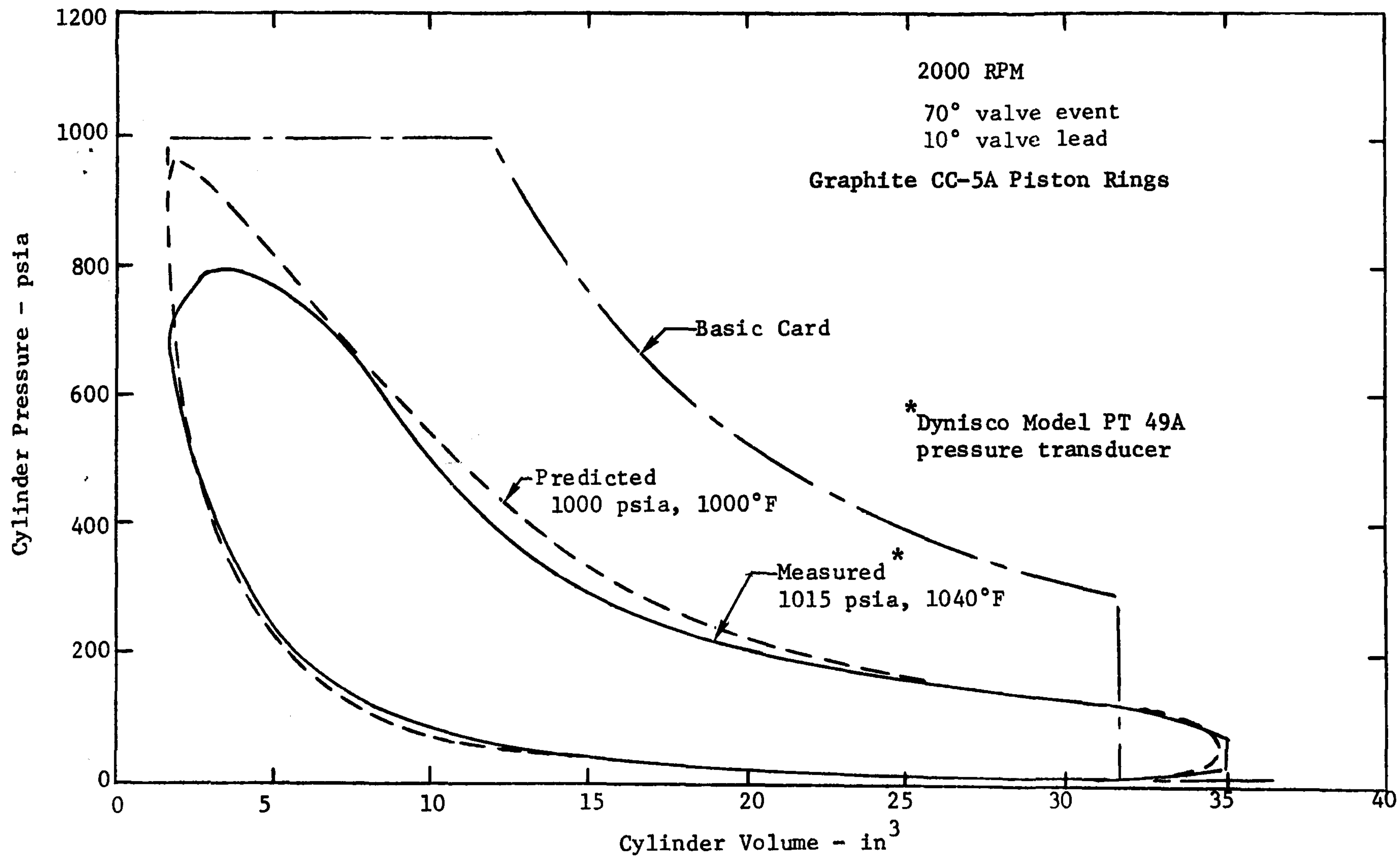
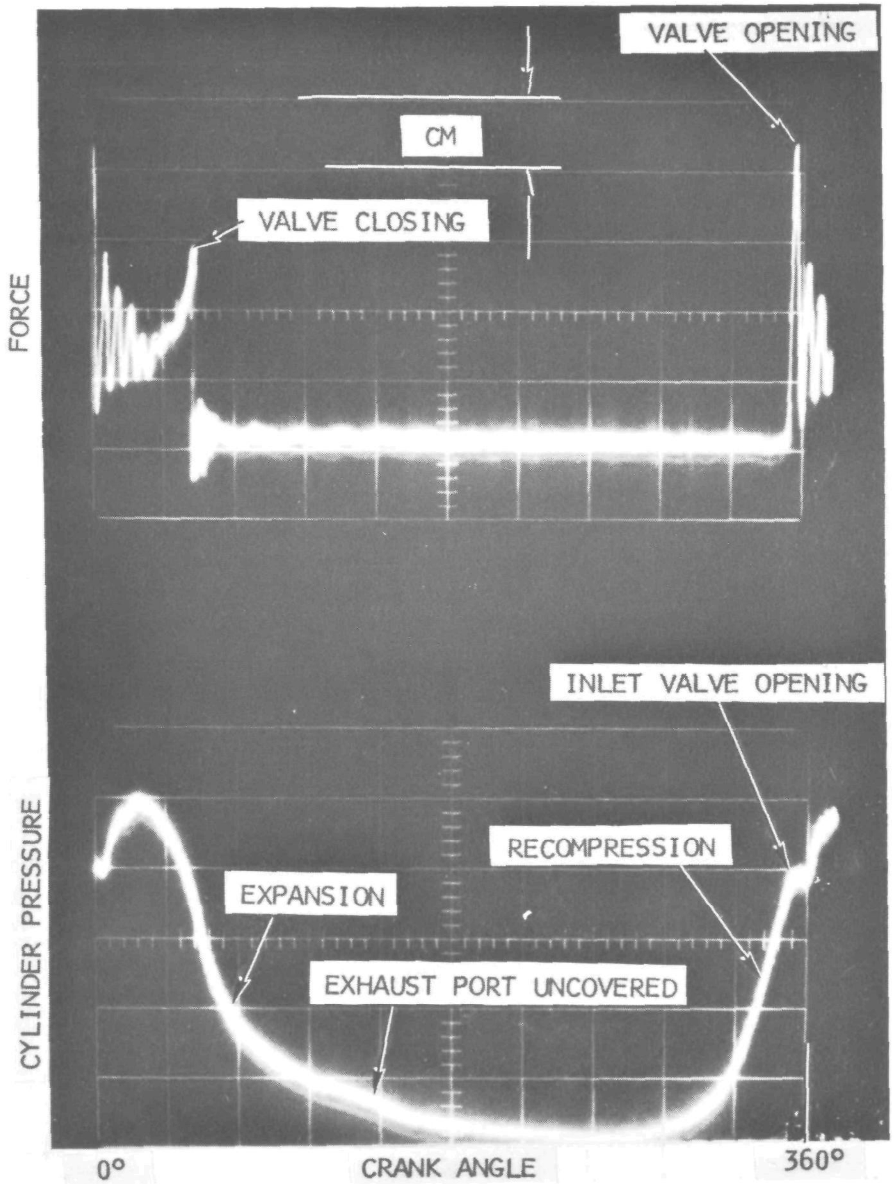


Figure 8.2-5. Crosshead Expander Typical P-V Diagram



<u>INLET VALVE PUSH ROD FORCE</u>

EXPANDER SPEED - 2000 RPM
INLET PRESS. - 1000 PSIG
INLET TEMP. - 1000°F
VALVE EVENT - 70°
VALVE OPENS - 10° BTDC
FORCE - 215 LB/CM

<u>CYLINDER PRESSURE</u>

EXPANDER SPEED - 2000 RPM
INLET PRESS. - 1000 PSIG
INLET TEMP. - 1000°F
PRESSURE - 155 PSI/CM

Figure 8.2-6. Oscilloscope Photos

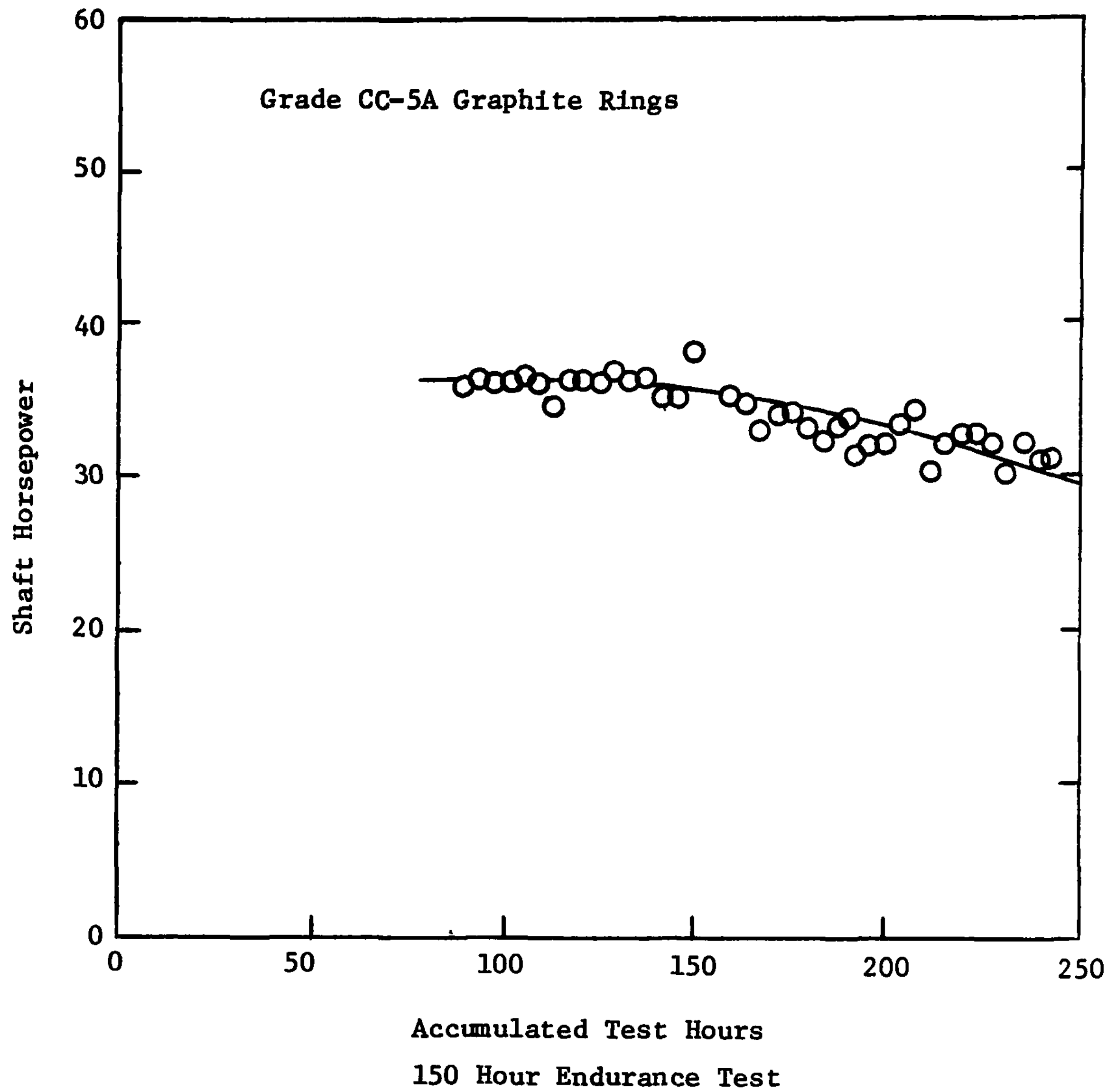


Figure 8.2-7. Single Cylinder Crosshead Expander - 1500 RPM

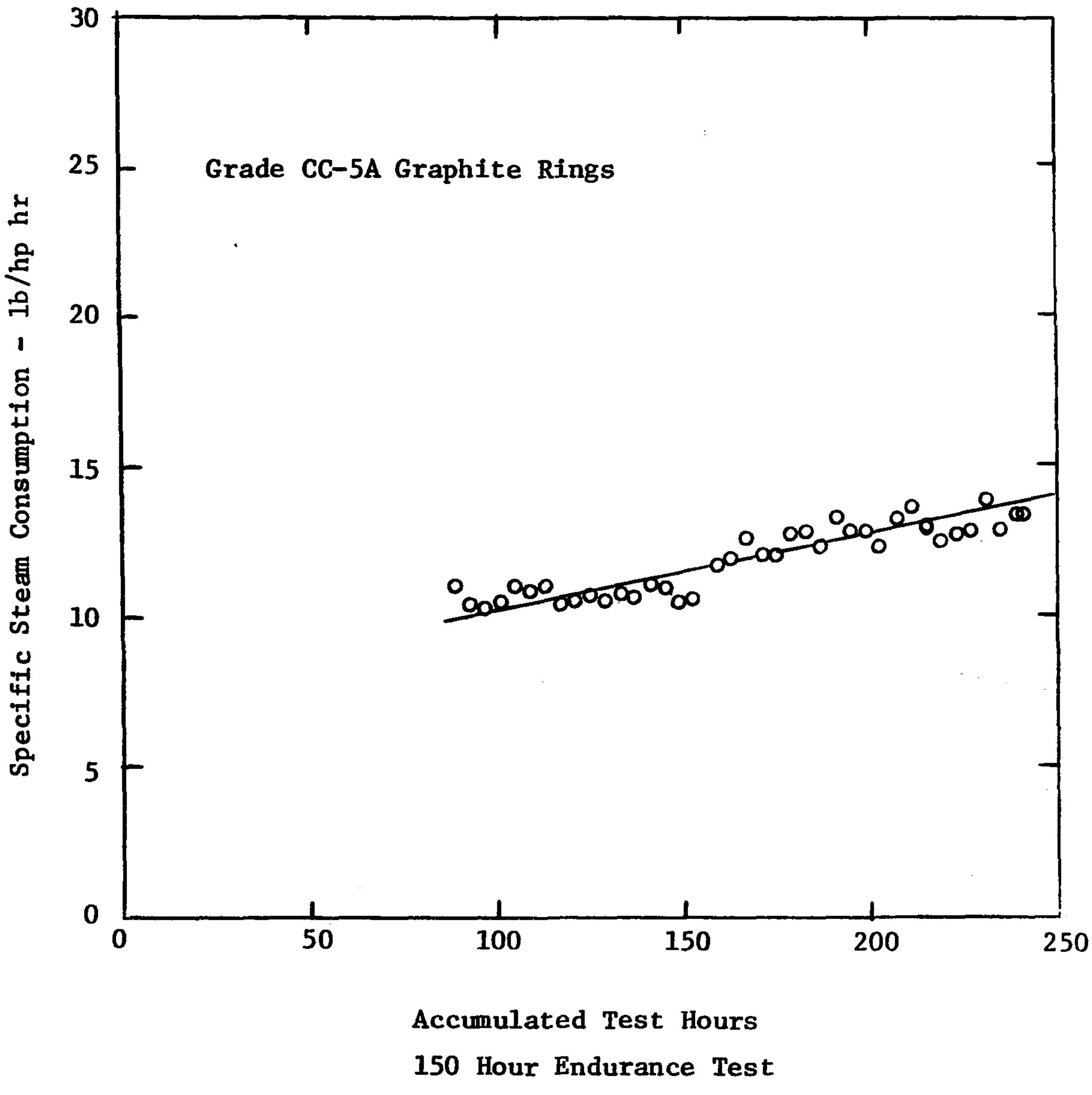


Figure 8.2-8. Single Cylinder Crosshead Expander - 1500 RPM

increase in SSC with time. Both figures reflect an increase in steam blowby as ring wear progressed. Figure 8.2-9 indicates a decline of engine efficiency, but there is a strong indication that the decline is asymptotic. This would be expected since the expander would continue to produce power with no rings.

All test data shown by Figures 8.2-1 through 8.2-9 were taken from the single cylinder crosshead expander which contained antimony impregnated graphite (grade CC-5A) piston rings and a Type 440C stainless steel cylinder liner. Figure 8.2-10 shows the measured temperature distribution for the crosshead expander.

8.2.2 Thermodynamic Performance - Cr_3C_2 Coated Inconel X-750 Piston Rings

The most meaningful comparison of the performance of the crosshead expander with graphite and with Cr_3C_2 coated Inconel X-750 power piston rings is the specific steam consumption (SSC) of each assembly. Figure 8.2-11 shows that the Cr_3C_2 coated rings operating at an average inlet condition of 396 psia, 687°F had a SSC which was 17 percent higher at 500 rpm to 4 percent higher at 2000 rpm than that of the graphite rings operating at an average inlet condition of 420 psia, 675°F. The Cr_3C_2 coated rings operating at 695 psia, 702°F had a SSC which was approximately 16 percent higher than that of the graphite rings operating at 715 psia, 690°F from 1000 - 2000 rpm.

The greater SSC of the Cr_3C_2 coated rings could be caused by increased steam blowby past the rings and/or by greater friction power losses. The brake horsepower (BHP) and brake mean effective pressure (BMEP) were greater for the Cr_3C_2 coated rings than they were for the graphite rings, as shown by Figures 8.2-12 and 8.2-13.

The greater SSC of the Cr_3C_2 coated rings despite higher BHP and BMEP appears to be due to higher ring friction as indicated by Figures 8.2-14 and 8.2-15 which compare pressure-volume for similar test conditions for the two ring types. These figures fail to show any strong evidence of greater blowby for the Cr_3C_2 coated rings since the expansion and recompression portions of the diagrams are very similar to those for the graphite rings.

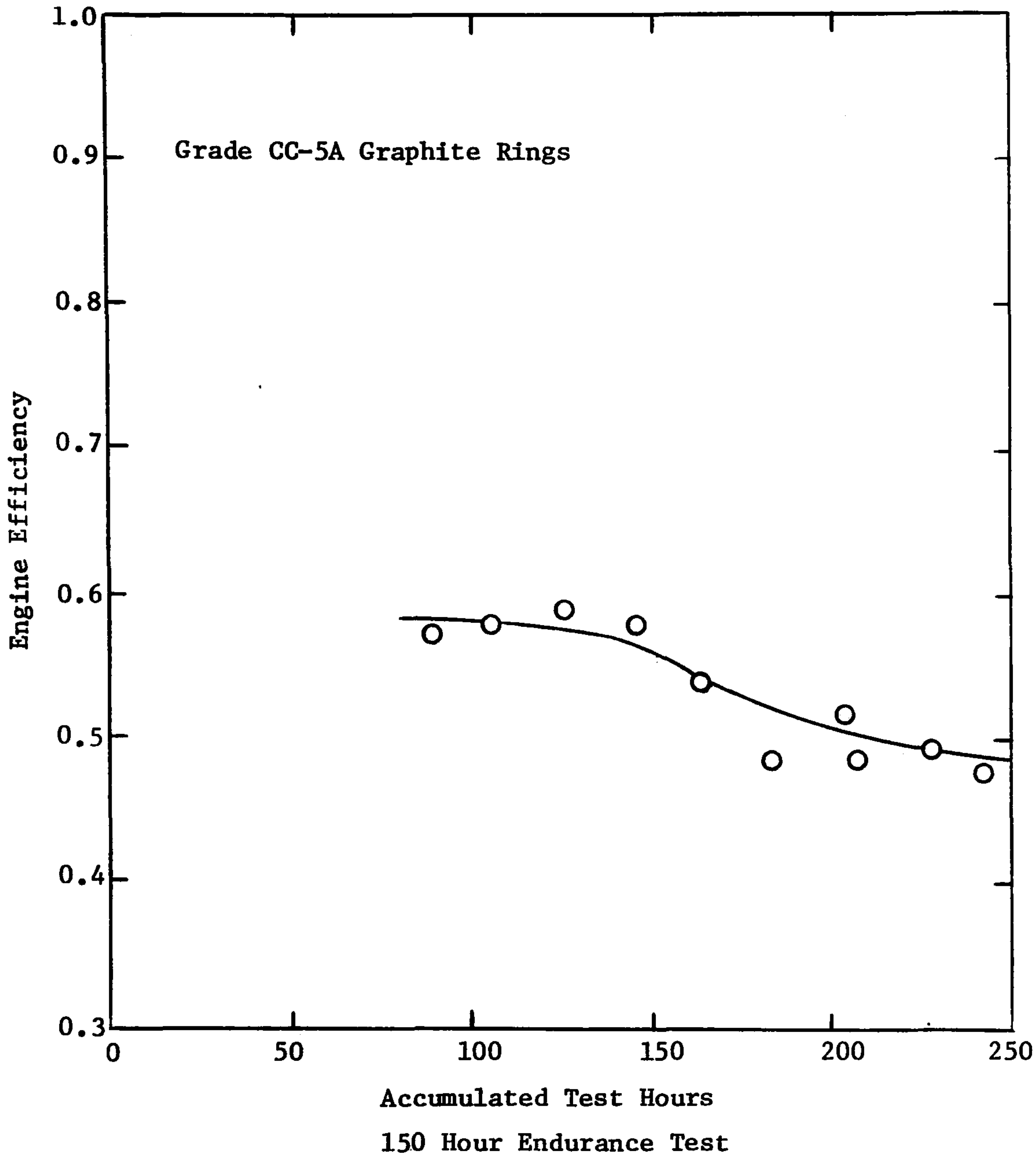


Figure 8.2-9. Single Cylinder Crosshead Expander - 1500 RPM

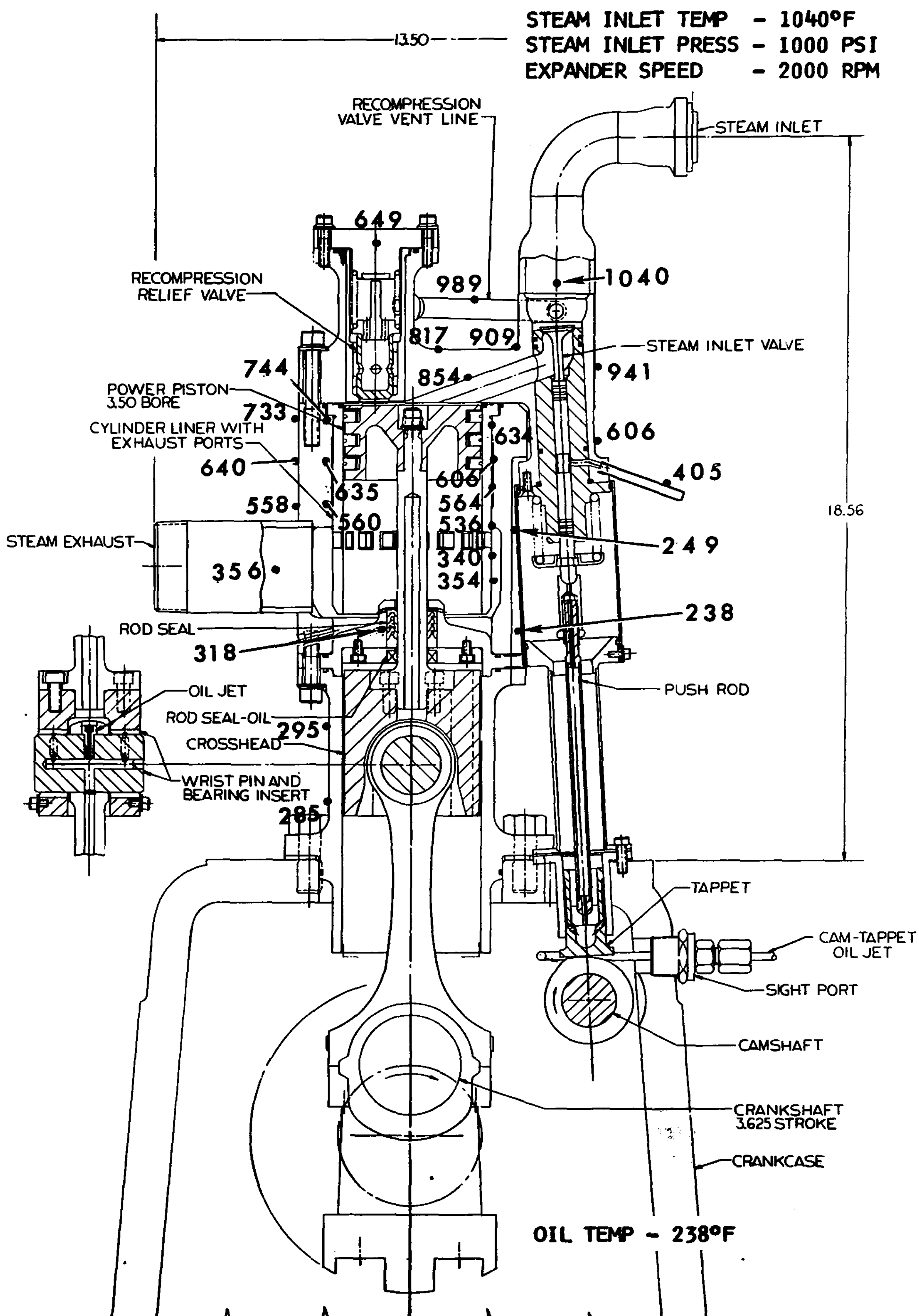


Figure 8.2-10. Measured Temperature Distribution of Crosshead Expander

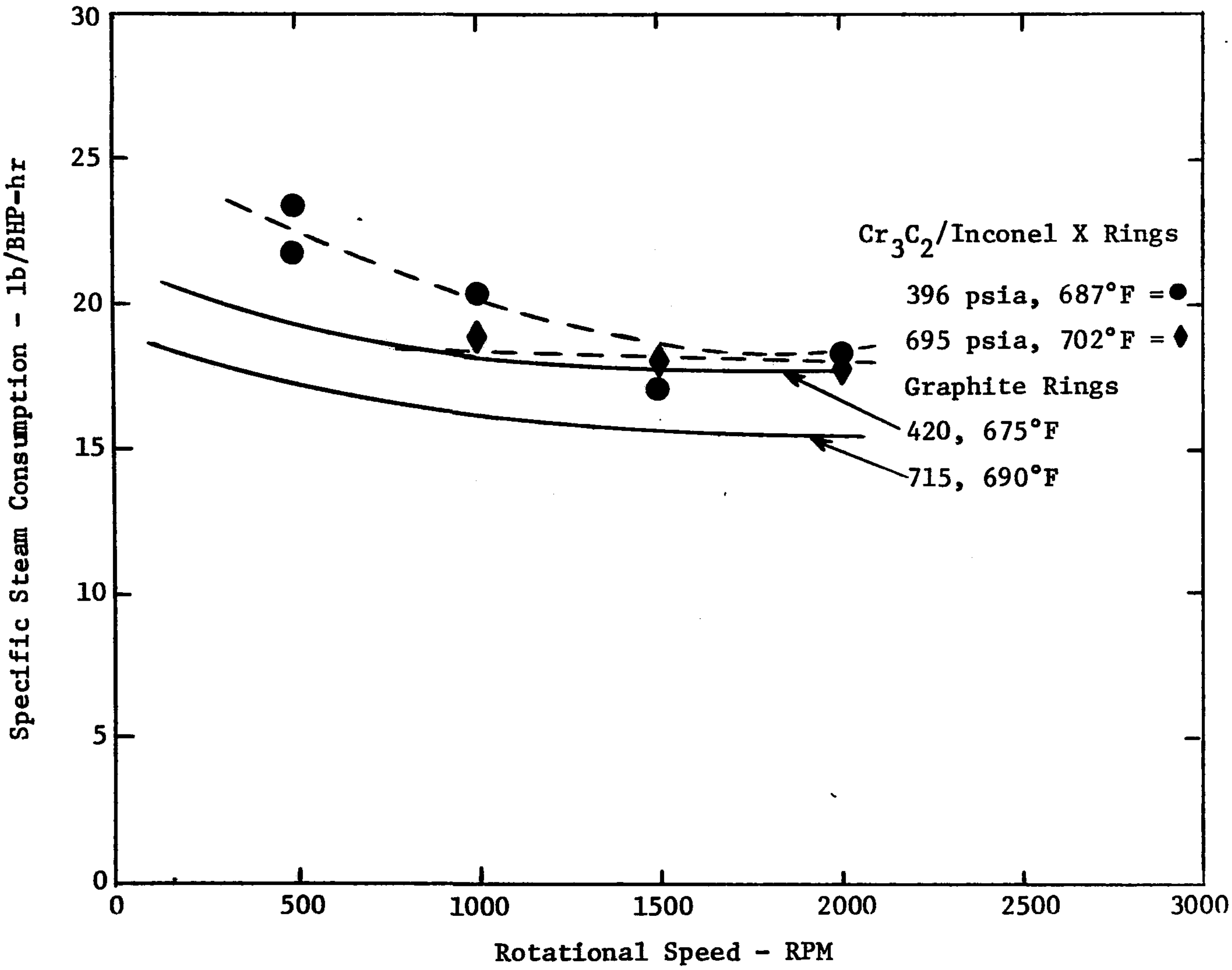


Figure 8.2-11. Comparison of Crosshead Expander Specific Steam Consumption with Graphite and with Cr_3C_2 Coated Inconel-X Piston Rings

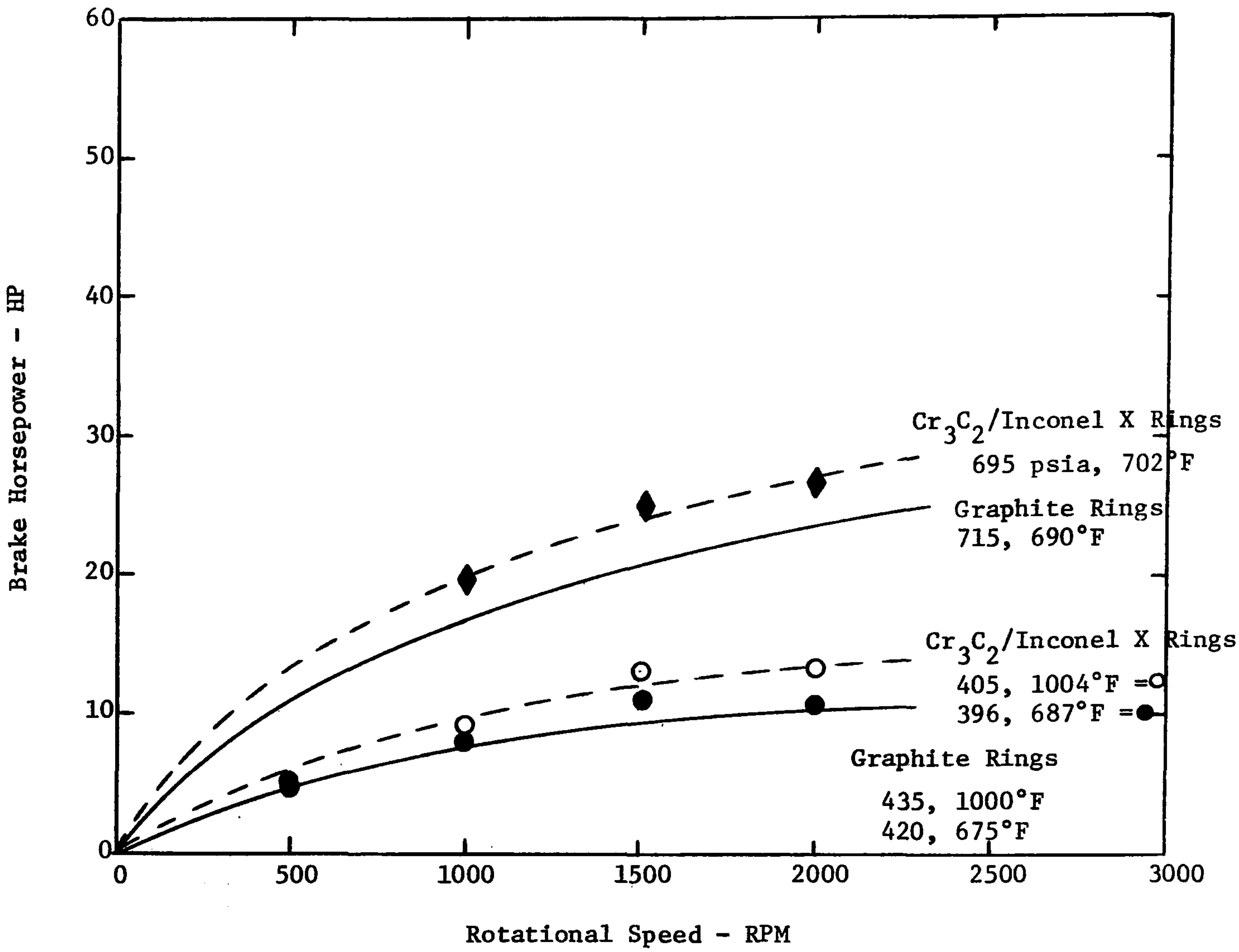


Figure 8.2-12. Comparison of Crosshead Expander Power Production with Graphite and with Cr_3C_2 Coated Inconel-X Piston Rings

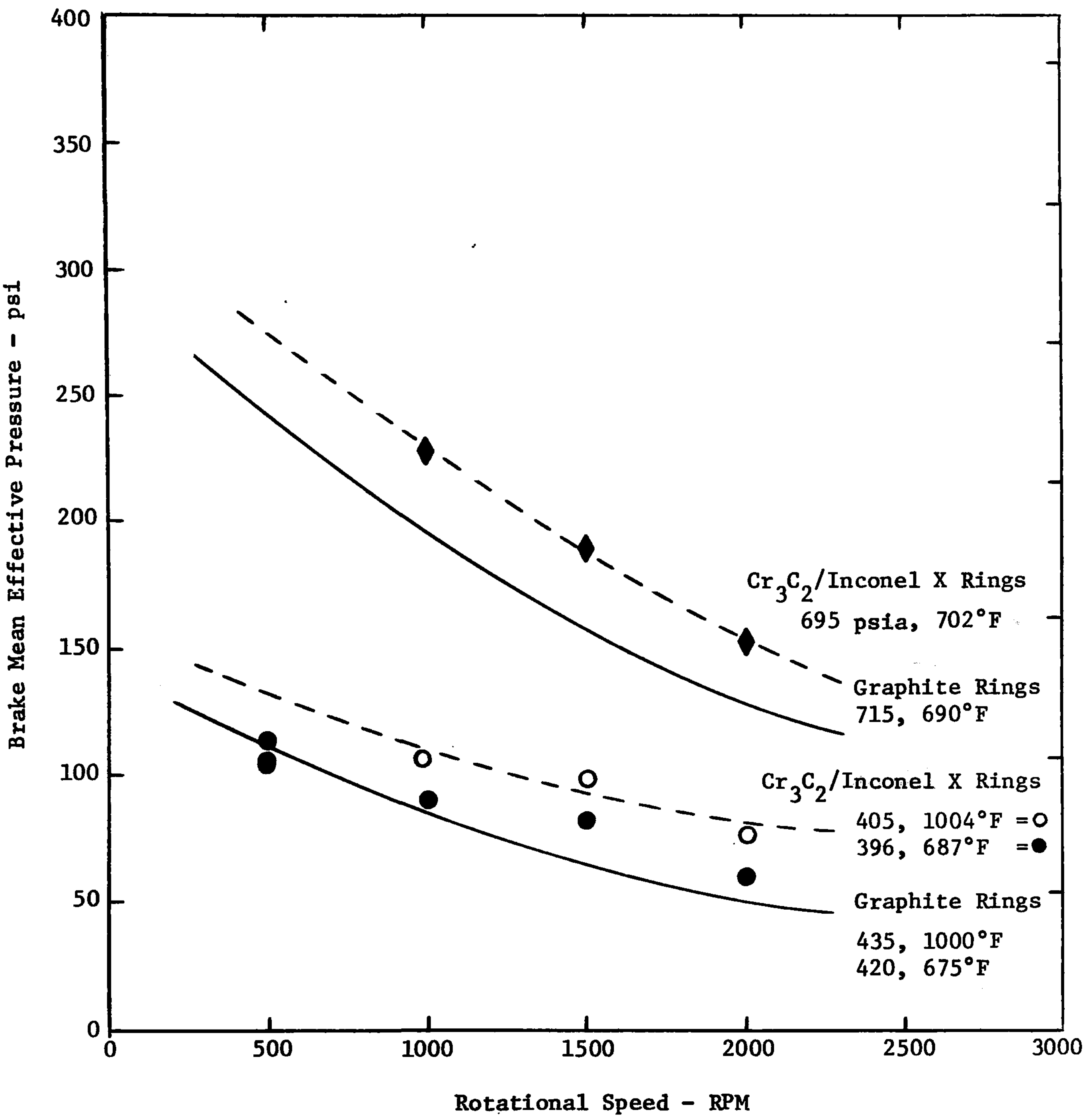


Figure 8.2-13. Comparison of Crosshead Expander BMEP with Graphite and with Cr_3C_2 Coated Inconel-X Piston Rings

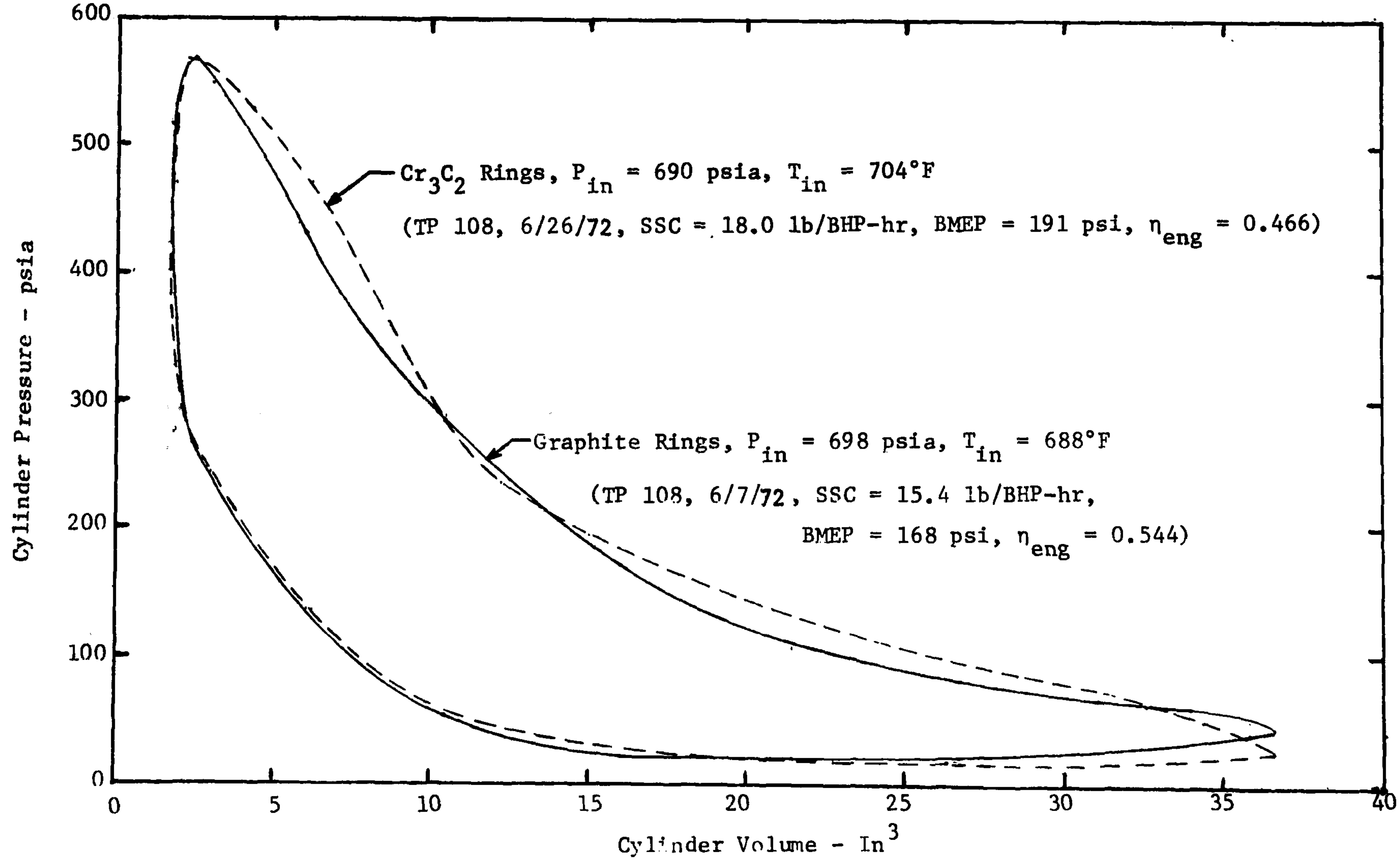


Figure 8.2-14. Comparison of Crosshead Expander P-V Diagrams with Graphite and with Cr_3C_2 Coated Piston Rings at 1500 RPM and Similar Steam Inlet Conditions

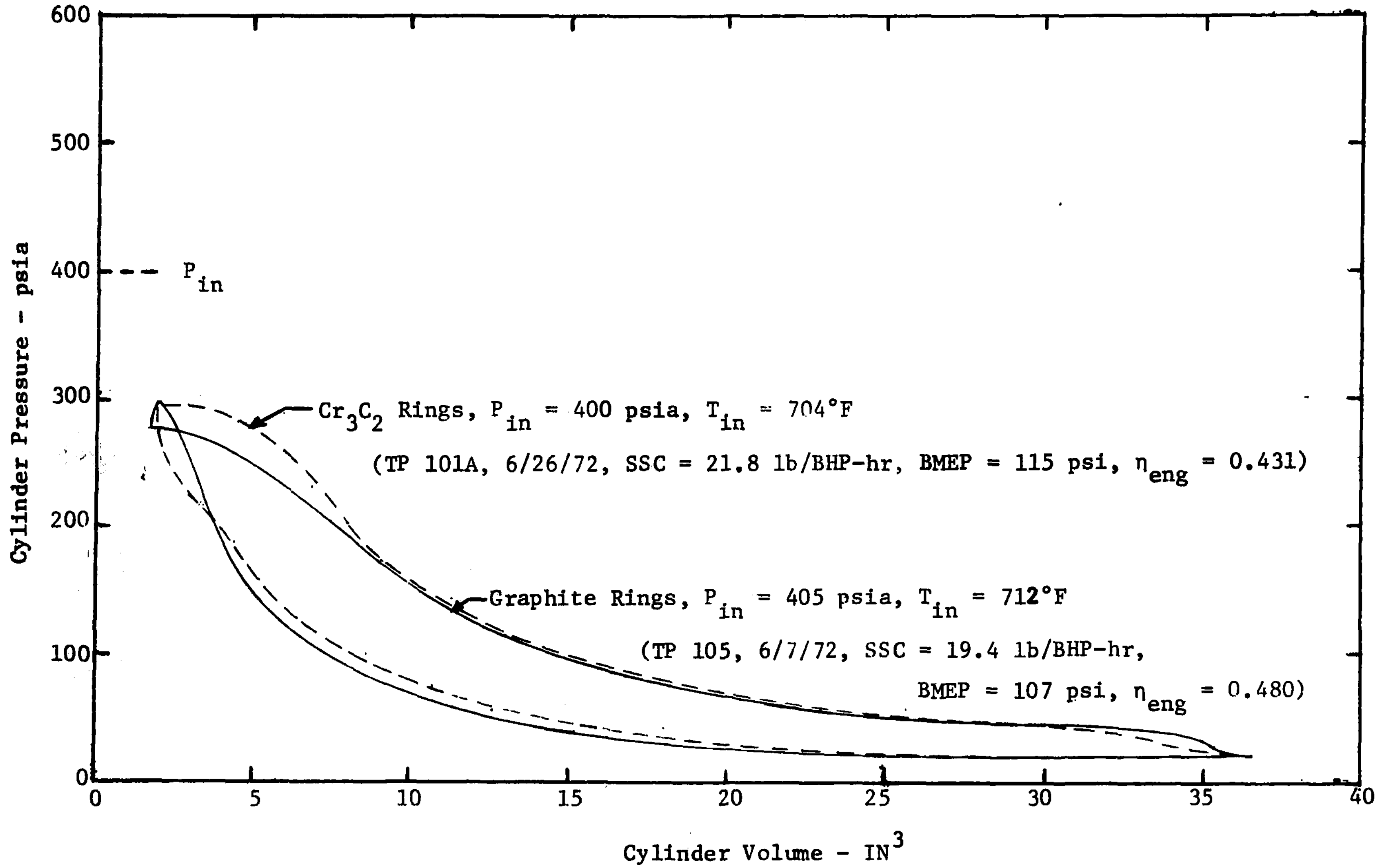


Figure 8.2-15. Comparison of Crosshead Expander P-V Diagrams with Graphite and with Cr_3C_2 Coated Piston Rings at 500 RPM and Similar Inlet Steam Conditions

The areas within the P-V diagrams are very nearly the same for each type of ring - thus the expander indicated horsepower would be very similar. Also measured engine efficiencies ($BHP/w\Delta h_{isentropic}$) are lower for the Cr_3C_2 coated rings. It is therefore concluded that the principal cause of the increased SSC of the crosshead expander with Cr_3C_2 coated rings was due to higher ring friction.

9.0 TEST RESULTS - TRUNK PISTON EXPANDER

9.1 Component Performance[11]

The trunk piston expander was assembled with Cr_3C_2 coated Inconel X-750 piston rings and with a carbon-graphite (grade P5NR) piston rider ring. Oil exclusion rings in the aluminum piston skirt were made of Type K6E and Type K (cast iron).

A 1.5 hour motoring checkout test was conducted on July 6. The speed was taken to 2000 RPM for approximately 15 minutes. The expander performance during checkout was quite satisfactory with the exception of the presence of more oil than anticipated in the cylinder. However, a condenser pressure of 20 psia during steam testing was expected to correct this.

Performance testing began July 7. Seven test points (6.75 hrs. testing time) were taken before the expander was shut down for inspection. Borescope inspection of the piston and cylinder liner revealed minor scoring of the liner but the compression rings appeared to be in good condition. The oil exclusion rings and aluminum piston skirt were not visible for inspection. A small amount of water (< 50 ml) was drained from the crankcase during this shutdown. Valve stem leakage during the 6.75 hour test period was approximately 940 ml/hr or approximately 0.5% of total steam flow. An analysis of the steam condensate from the condenser for oil after 3.4 hours of testing indicated an oil concentration of 192 ppm. The high oil concentration in the condensate probably due to residual oil being in the system or a result of the high speed motoring checkout test. Because of the short duration of the test it was not possible to compare the relative oil leakage between the crosshead piston and the trunk piston.

The expander was restarted but 2.5 hours later the recorded speed trace indicated a drop in speed from 2000 RPM to 0 RPM in approximately 4 seconds. An attempt to rotate the expander was unsuccessful. The expander was removed from the test facility and dismantled.

Examination of the expander during and after disassembly revealed the following:

1. The cylinder liner was scored in the area contacted by compression rings as shown by Figure 9.1-1. Oil was present in the piston ring grooves and on the cylinder wall at shutdown.

2. The aluminum piston skirt was badly worn and scored as shown in Figure 9.1-2. It appeared that the piston skirt to liner clearance was insufficient at sometime during testing. Review of temperature recorder charts revealed that the crankcase oil temperature inadvertently exceeded the maximum allowable by about 50°F. The high oil temperature resulted in excessive thermal expansion of the aluminum skirt - thus causing interference with the cylinder liner.

3. The rear main crankshaft bearing was seized to the crankshaft. Particles of aluminum were present in the main bearing oil ports.

4. The oil filter appeared completely plugged with a dark slurry of what appeared to be carburized oil. Filter plugging results in the opening a filter bypass if the filter pressure drop exceeds 12 psi. The bypass could allow oil with aluminum chips from the piston skirt to enter the main crankcase oil gallies and reach the rear main bearing - thus restricting oil flow to the bearing. No damage to other bearings, gears, etc. was found. Emission spectrographic chemical analysis of the dark deposit that had plugged the filter shows high concentrations of iron, copper lead, zinc and chromium, plus trace amounts of tin, aluminum and nickel that would be expected to come from wear of engine components. A high concentration of sodium indicates the possibility of contamination with the treated water. This combination of oil, water, metal fines, plus the

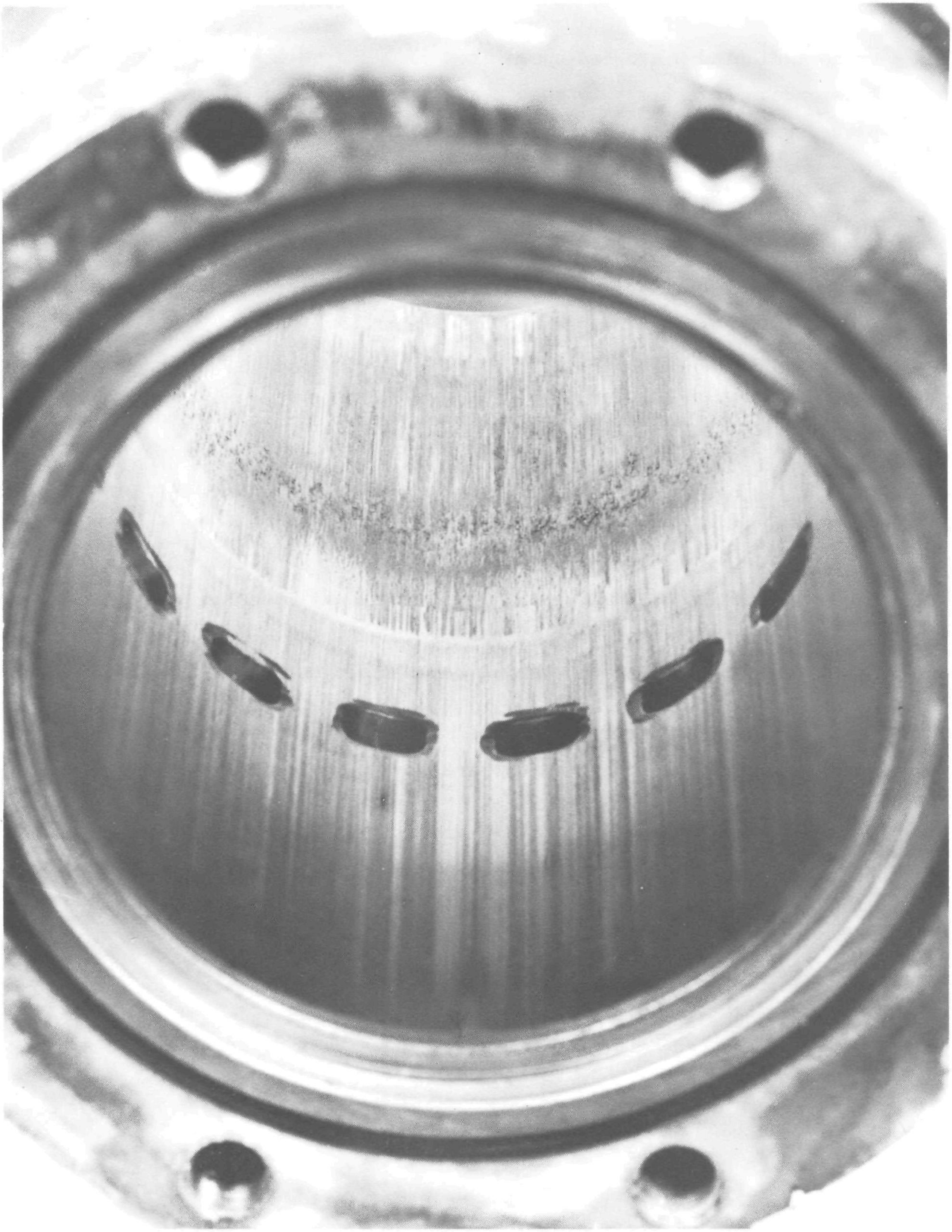


Figure 9.1-1. Trunk Piston Liner Following 9.2 Hours Test (P72-6-3Z)

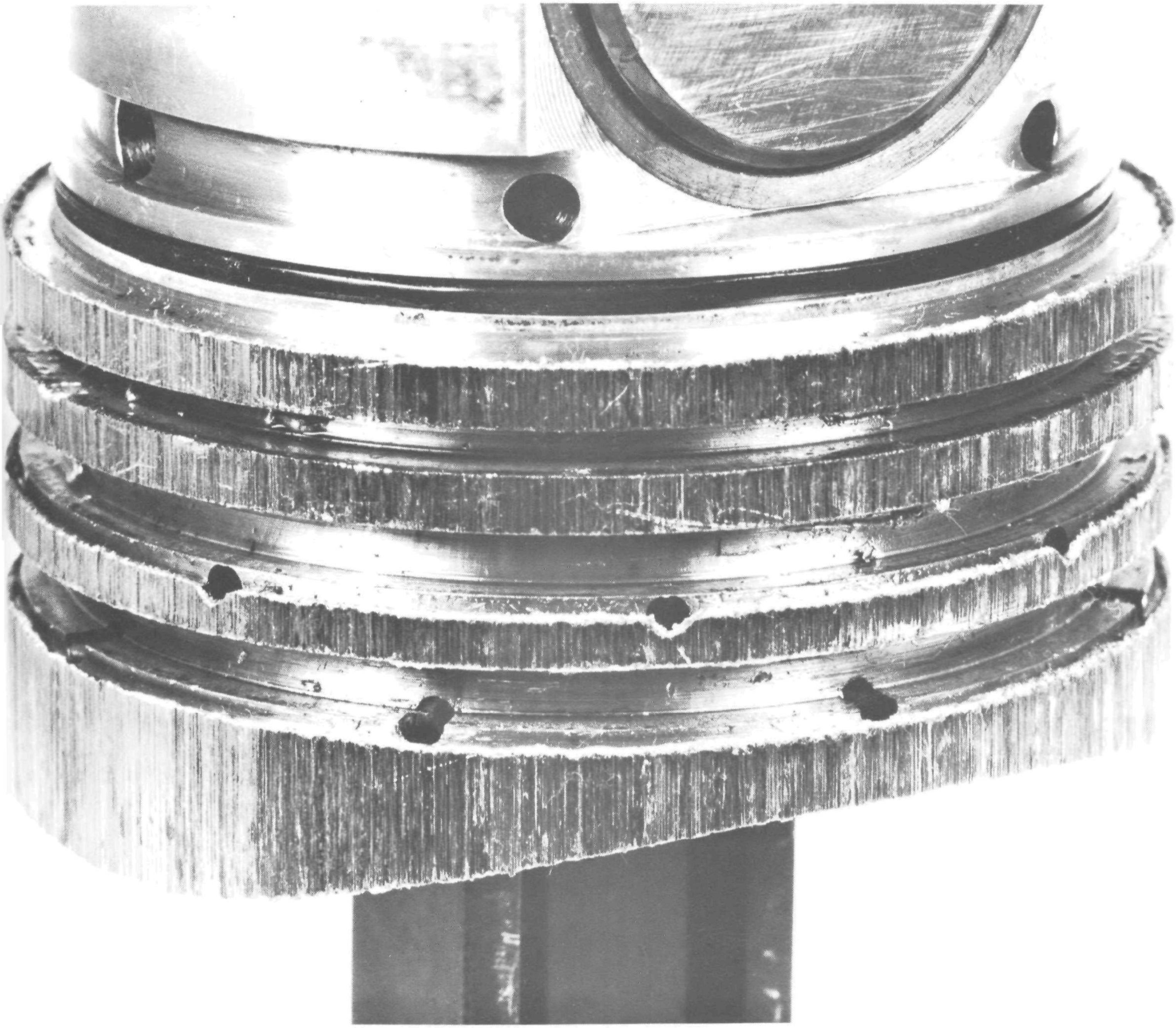


Figure 9.1-2. Trunk Piston Skirt Following 9.2 Hours Test (P72-6-3W)

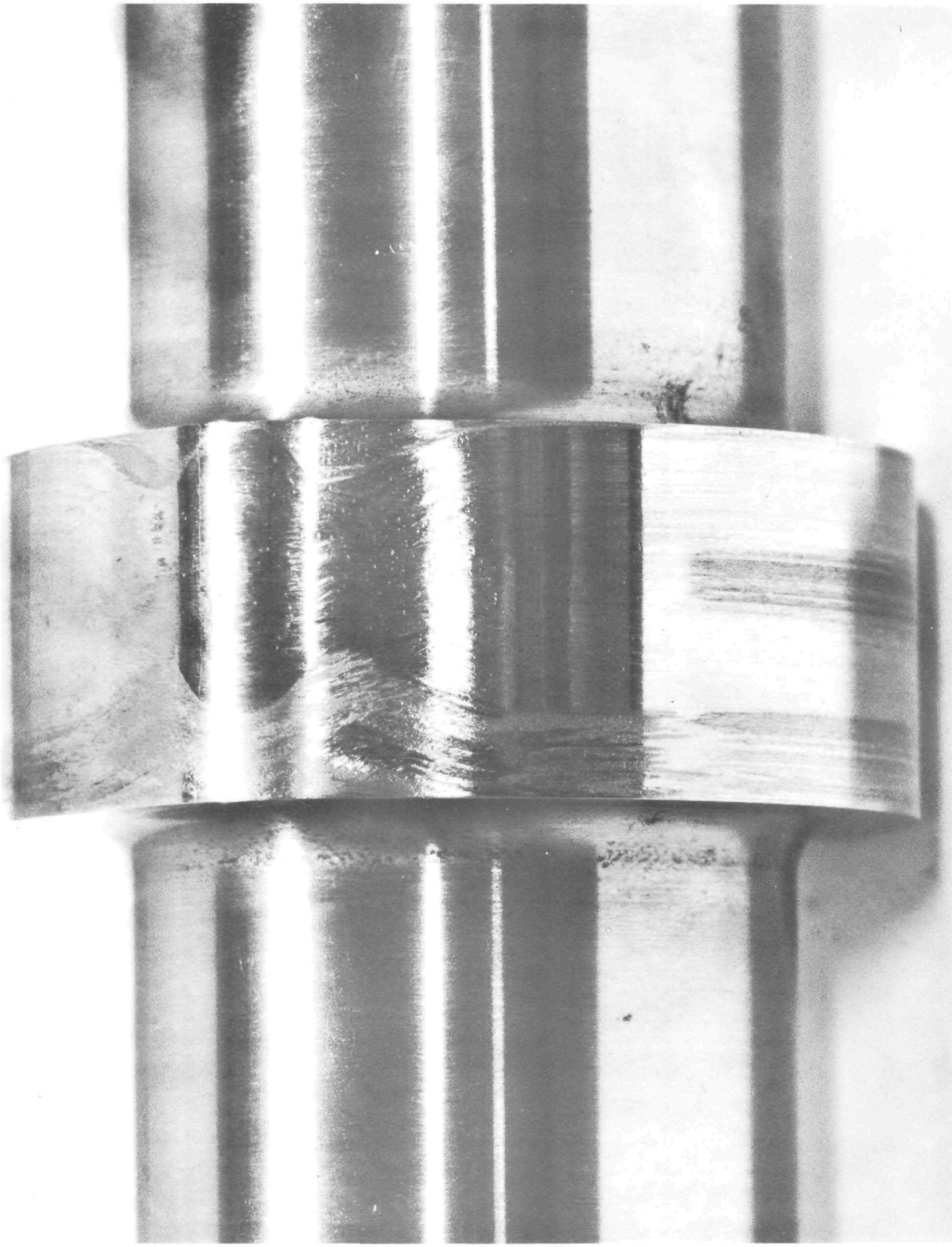


Figure 9.1-3. Cam Lobe From Trunk Piston Expander (P72-6-3U)

presence of sodium ions at high temperatures, could react to form the grease-like deposit. Analysis of oil from the crankcase for water content indicated a concentration of 1.3% in the first 500 ml sample and 0.13% in the second 500 ml sample.

5. The camshaft lobe, Figure 9.1-3, and cam tappet appeared worn or burnished over the contact areas.

Just prior to bearing seizure, a large amount of condensate and oil mixture was observed coming from the crankcase vent. Bearing failure and excessive cam wear may have resulted from poor lubrication: (1) excessive oil temperature, (2) too much water in the oil, (3) restricted oil flow, and (4) a combination of the above three.

9.2 Thermodynamic Performance

The brake mean effective pressure (BMEP) of the trunk expander is compared on Figure 9.2-1 to those of the crosshead expander with both carbon-graphite (Grade CC5A) rings and a Type 440C stainless steel liner, and Koppers K1051 (Cr_3C_2 coated Inconel X-750) rings and a Type 440C stainless steel liner. Three sets of inlet steam conditions are shown.

The first test series (400 psia, 712°F) of the trunk expander shows BMEP which are initially considerably lower than those of the crosshead expander, but which later in time are only slightly lower. A strong influence of "wear-in" time is suggested. Both ring friction and leakage losses are involved, but insufficient data is available to determine which is the more influential.

The second test series (417 psia, 998°F) shows BMEP which are somewhat lower than those of the crosshead expander but which decrease with increasing speed as do the crosshead data. This suggests that "wear-in" has been completed, but that a higher friction coefficient exists.

The third test series (690 psia, 703°F) shows its first point having a BMEP very close to that of the crosshead expander with graphite rings, with its final point very close to that of the crosshead expander with Cr_3C_2 coated Inconel rings. Similar conclusions relating to "wear-in" as a function of time may be drawn from the brake horsepower versus speed curves of Figure 9.2-2 and from the brake specific steam consumption curves of Figures 9.2-3 through 9.2-5.

The change of brake specific steam consumption (BSSC) with time is more clearly shown in Figure 9.2-6 for the trunk and crosshead expanders with Cr_3C_2 coated rings versus Type 440C liners. In any pressure-temperature sequence of testing, speed was varied from 500 to 2000 RPM. Therefore, speed effect could distort the performance versus time trend. However, it appears from Figure 9.2-6 that testing time had a stronger influence on improving BSSC (up to the time of ring or expander failure) than operating speed, pressure, or temperature.

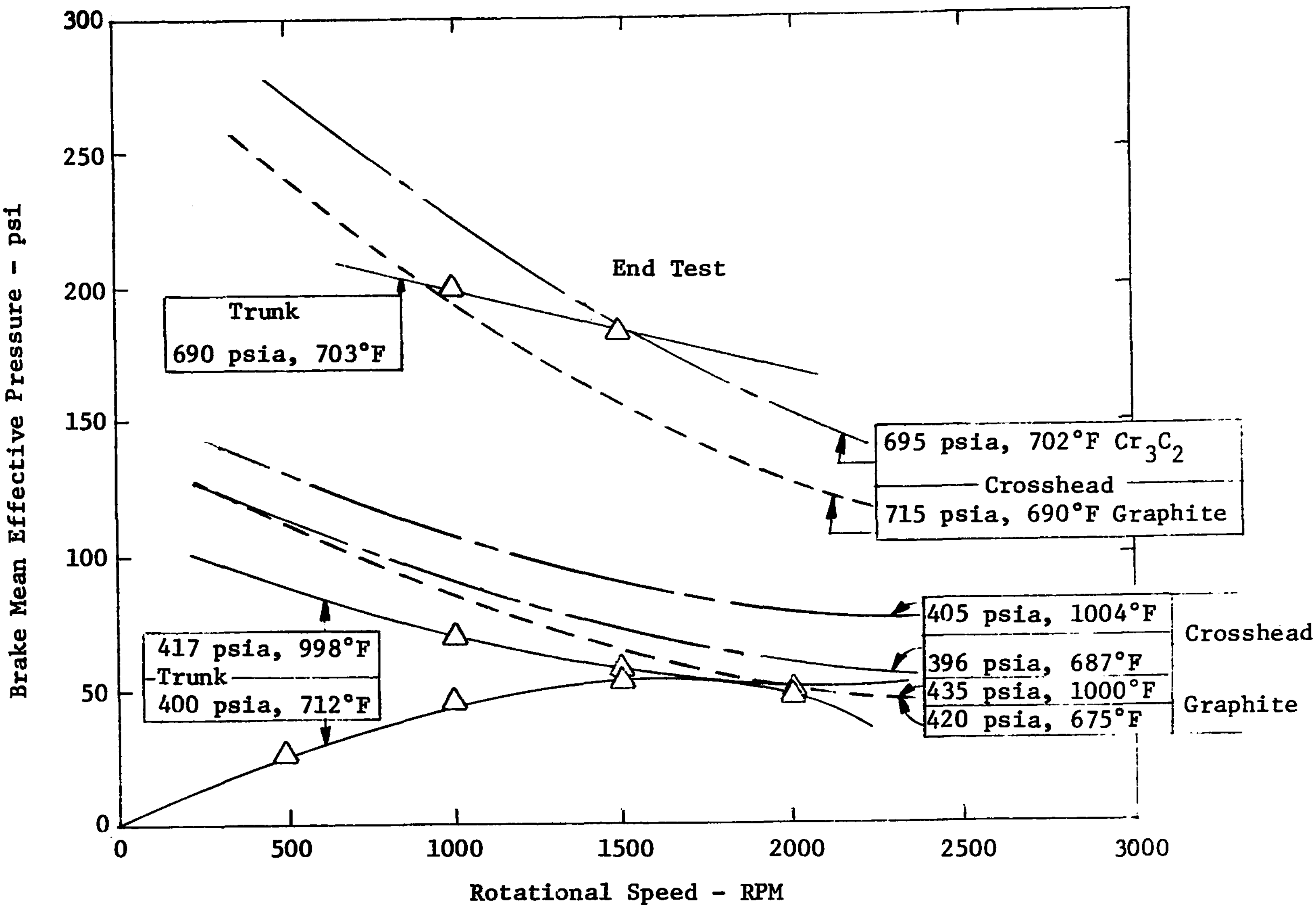


Figure 9.2-1 Trunk and Crosshead Expander Brake Mean Effective Pressure As A Function of Rotational Speed, Inlet Steam Conditions, and Piston Ring/Cylinder Liner Materials

Expander	Piston Ring Material	Cylinder Liner Material
Trunk Crosshead	Cr_3C_2-Coated Inconel X-750	Type 440C stainless steel
Crosshead	Carbon-Graphite Grade CC5A	Type 440C stainless steel

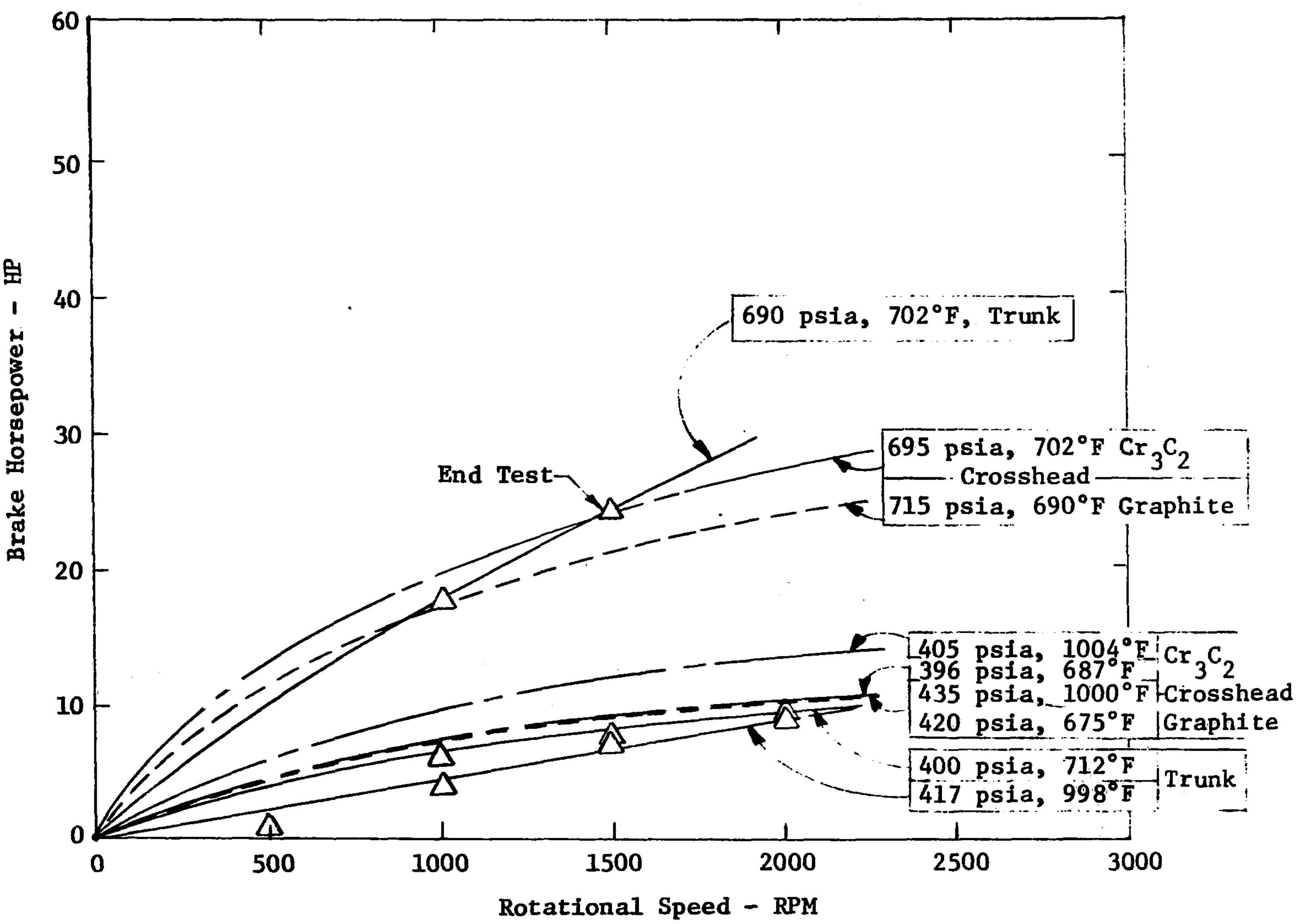


Figure 9.2-2 Trunk and Crosshead Expander Brake Horsepower As A Function of Rotational Speed, Inlet Steam Conditions, and Piston Ring Material

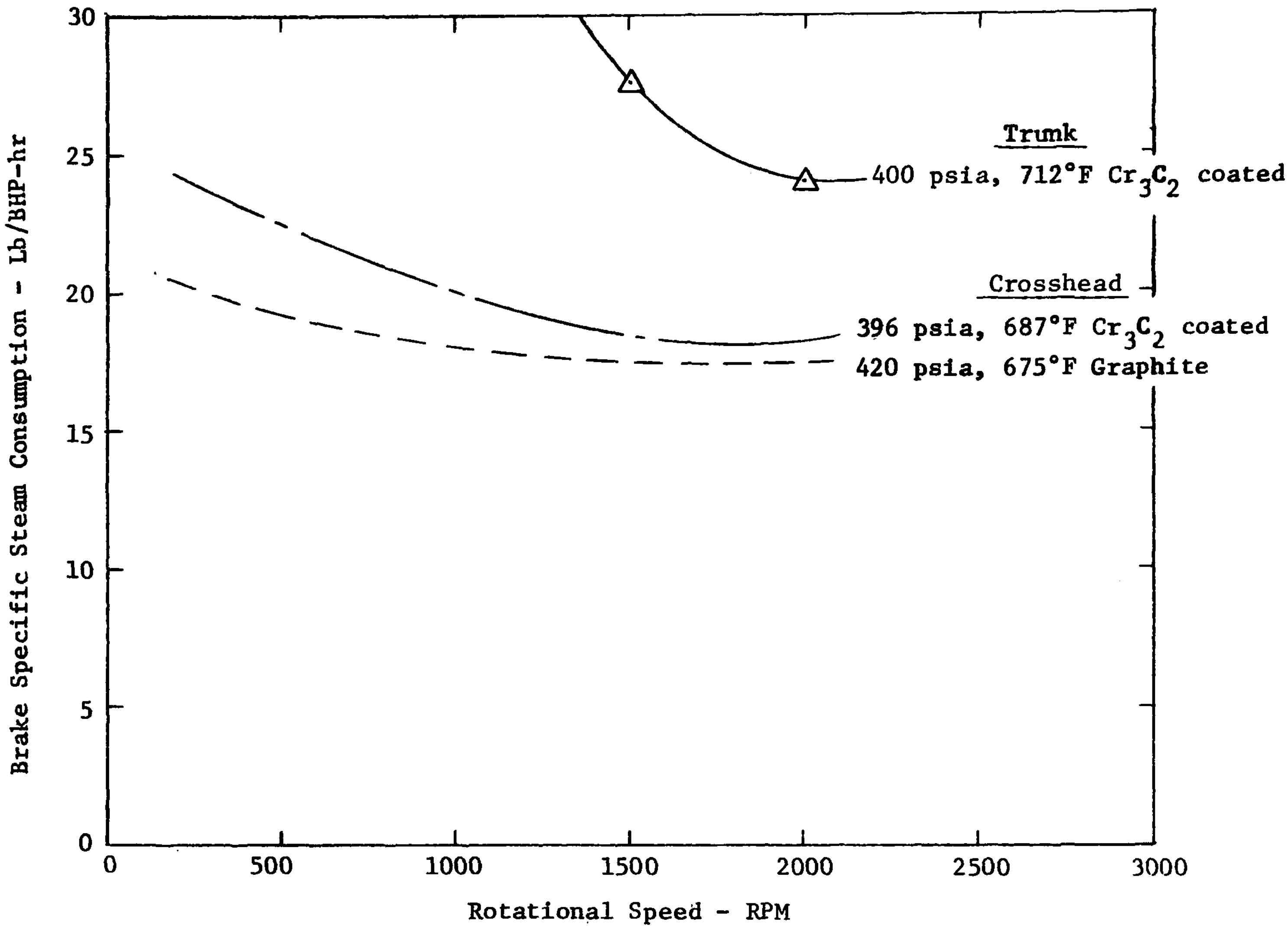


Figure 9.2-3 Trunk and Crosshead Expander Brake Specific Steam Consumption As A Function of Rotational Speed and Piston Ring Material for Steam Inlet Conditions of Approximately 400 psia, 700°F

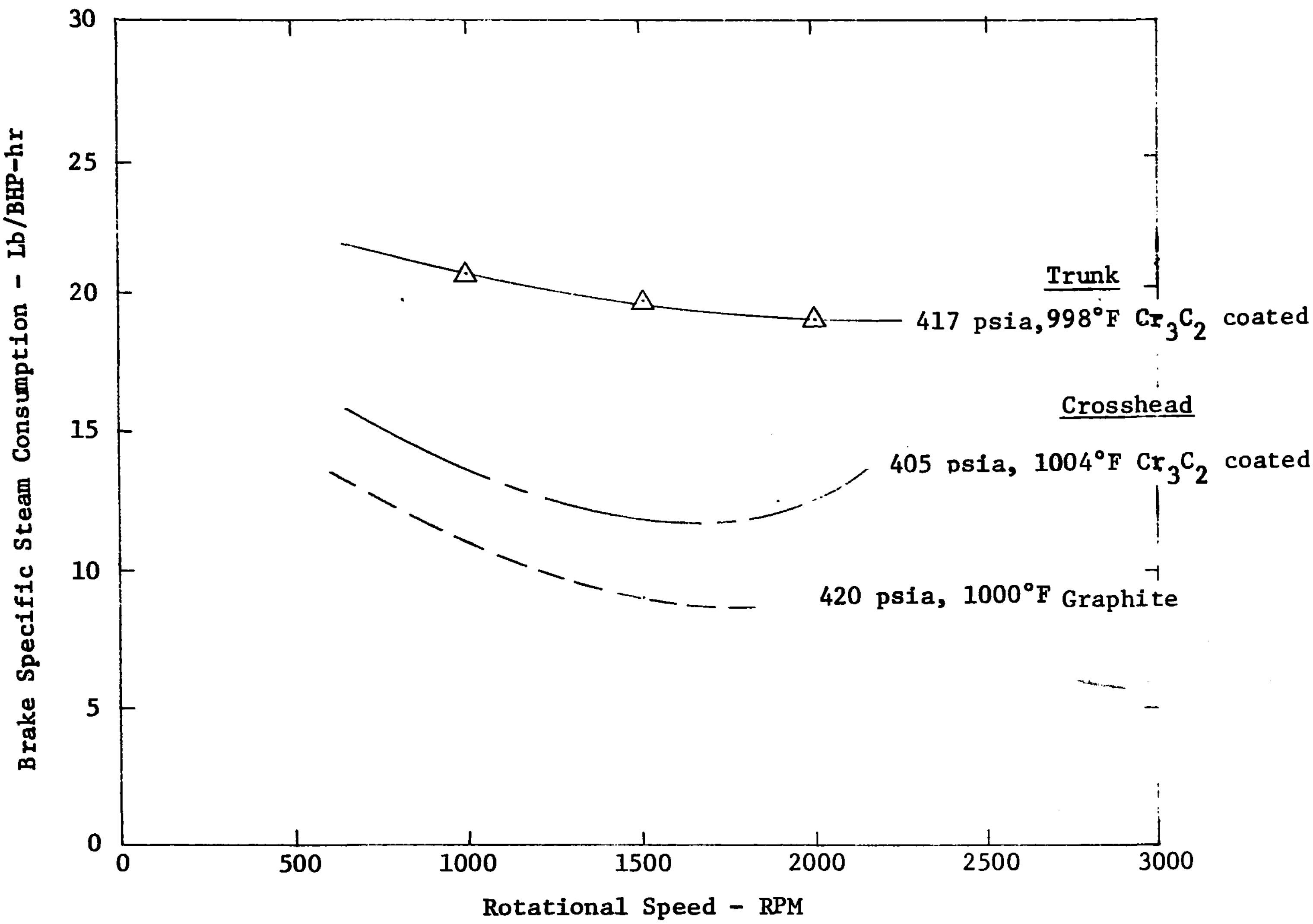


Figure 9.2-4 Trunk and Crosshead Expander Brake Specific Steam Consumption As A Function of Rotational Speed and Piston Ring Material for Steam Inlet Conditions of Approximately 400 psia, 1000°F

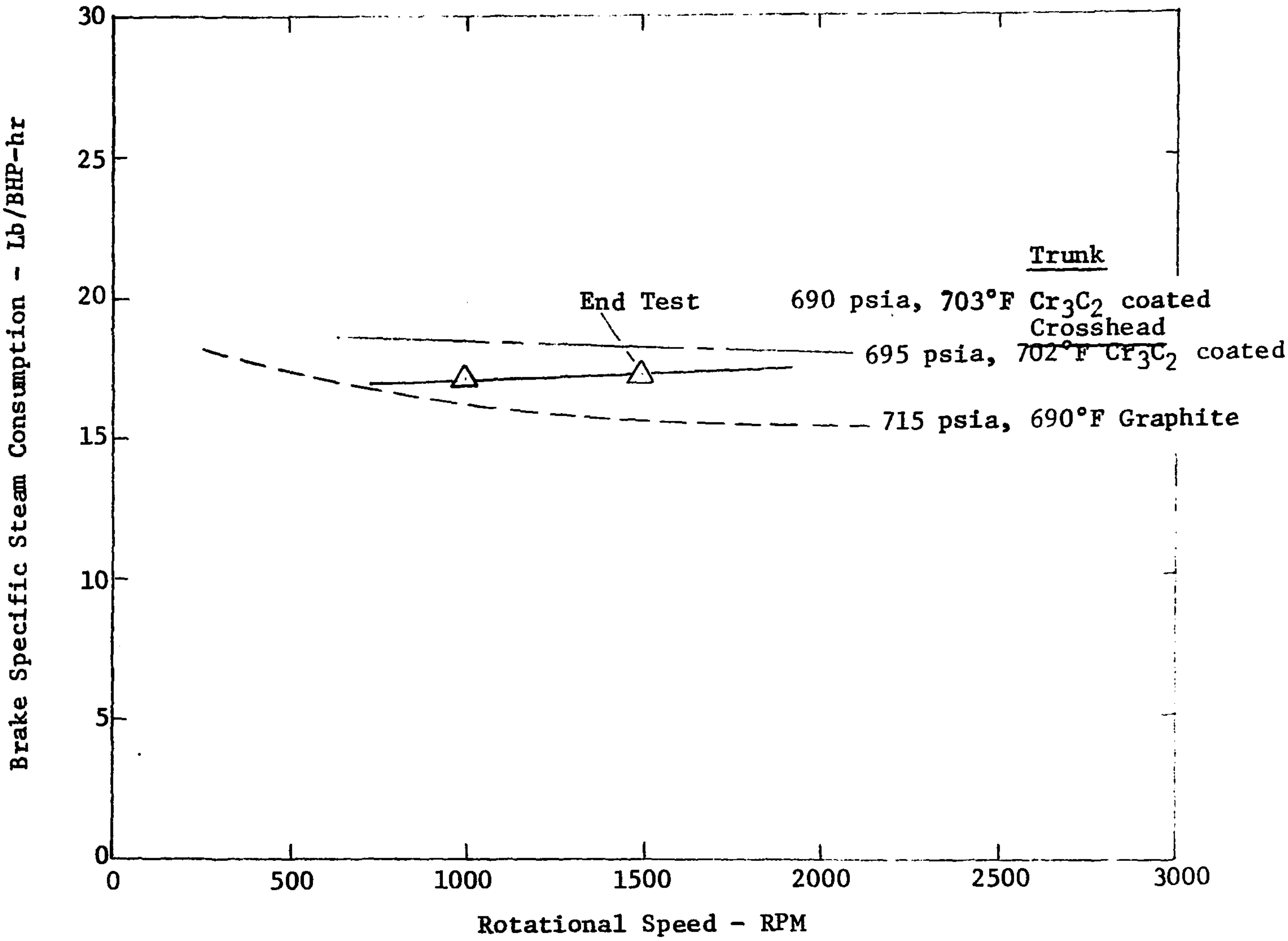


Figure 9.2-5 Trunk and Crosshead Expander Brake Specific Steam Consumption As A Function of Rotational Speed and Piston Ring Material for Steam Inlet Conditions of Approximately 700 psia, 700°F

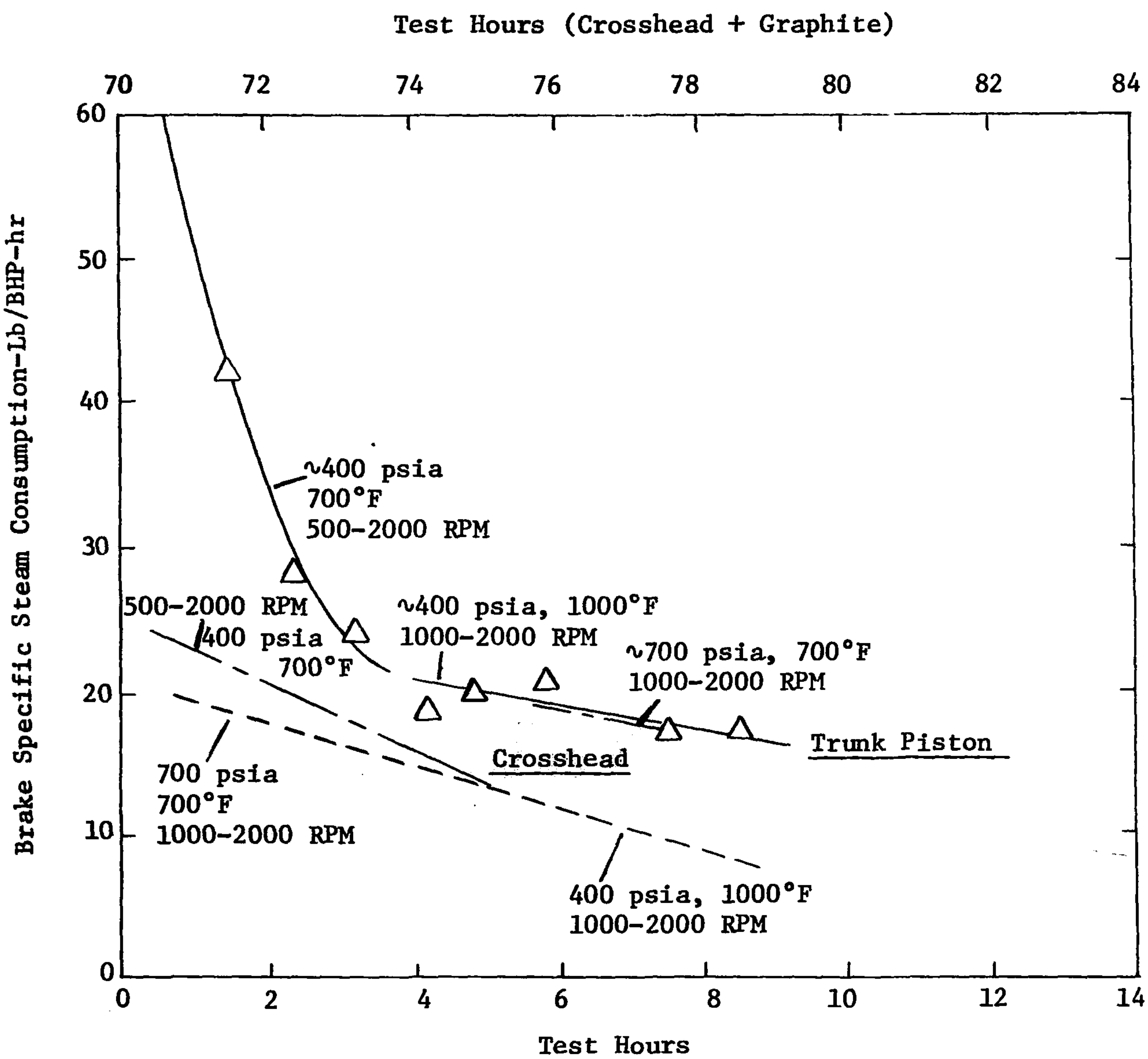


Figure 9.2-6 Crosshead and Trunk Expander Brake Specific Steam Consumption As A Function of Testing Time. Piston Ring Material was Cr_3C_2 Coated Inconel X-750 and Cylinder Material was Type 440C Stainless Steel

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