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IMPROVEMENTS IN RATING ROTARY SCREW COMPRESSORS

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The last report given to Purdue University dealt with the gas slippage that occurs within the casing of a DRY rotary compressor and the qualifications for an oil FLOODED rotary compressor. (References 1, 2 and 3.)

The performance data for the flooded machine was taken from T. G. Krulick's thesis, (Reference 3), prepared at Pennsylvania State University. The data used for the DRY type compressor was developed from SLIDING-VANE type machines. (Reference 2). This DRY unit has an eccentric rotor, 10.62 inches in diameter and 24.5 inches long. It rotates 1000 RPM and has a displacement of 743 cfm.

The Penn State test machine was a rotary screw type compressor which was supplied 21.1 gpm of seal and lube oil. The compressor had a displacement of 650 cfm. A generalized equation for selecting the oil flooding rate is:

$$0.03\,Q + 10 \text{ gpm} \qquad \text{Equation 1}$$

The most satisfactory means of controlling the feed rate is to regulate the oil pump discharge pressure at a value about 30 psig in excess of the gas discharge psig. It may be necessary to vary the discharge pressure +/- 5 psi in order to establish a normal flow rate. Greater pressure differentials are likely to provoke severe fluid pounding. The letter Q represents the compressor displacement in cubic feet per minute, cfm. The abbreviation, gpm, represents the oil flow rate in terms of U.S. gallons per minute. (References 1 and 2).

This article deals mainly with the FLOODED rotary screw compressor and its most recent acquisition, the unsymmetrical rotor. The change in the rotor profile reduces the exhaust velocity and diminishes some of the offensive noise. It also increases the capacity by approximately three percent. The symmetrical rotor capacity factor "X" is 0.0625 in the basic displacement equation:

$$Q\text{, displacement} = d\,L\,U\,X\text{, cfm} \qquad \text{Equation 2}$$

Where d is the rotor diameter and L is the rotor length, both in inches. U is average piston speed. In rotary machines it relates to the rotor tip speed, in feet per second. The effect of the unsymmetrical profile raises the constant X to 0.0645.

The main objective of an oil FLOODED rotary screw compressor is to reduce the gas discharge temperature to a value less than 160°F or to limit the observed temperature rise to 100°F. This type of operation extends the volume and pressure capabilities. It also extends the service life of the compressor and its lubricant. Whereas the usual compression limitation for a piston machine is about 8 atm, it may be as much as 20 atm for an oil flooded screw machine. MYCOM anticipates compression ratios of 50 atm with multiple screw compressors running in series.

The rotary screw and the "liquid liner" compressors are the only designs which may be adopted to the flooding technique for curtailing the discharge gas temperature. The valve areas are already overtaxed with fluid flow through a piston compressor. Entrained liquids in the process lines of heavy chemical plants and refineries, that include gas compressors, are a menace to the safety of such plants. Other types of rotary compressors are disqualified by reason of their limited structural pressure, low efficiency and limited differential pressure.

Usually eighty percent of the heat of compression can be absorbed by the extended lube and seal oil system. This system is less costly than an equivalent duty, gas cooling system. Often it is economical to delete the use of gas intercoolers and rely on the oil flooding system for cooling the gas.

When the ratio of compression exceeds five and the composite oil temperature exceeds 200°F, the oil flow rate must be increased to reduce the discharge temperature. Equation 1 usually applies ample coolant.

The "liquid liner" type of compressor is the closest approach to the flooded screw compressors. Both types of machines can handle both WET suction and discharge connections. The screw compressor can operate at much greater speeds than the "liquid liner" and most rotary compressors, in the order to 3600 versus 1000 rpm. Speeds of 400 Hertz are common for rotary screw units in aerospace service. Catalog data on the "liquid liner" compressor gives an average efficiency of 50 percent for 100 psig air service. The equivalent service for an unsymmetrical two-stage screw compressor as tested by K. Kasahara, et al. (Reference 4) in Tokyo gave a dynamic efficiency of 94 percent at 11 R and 92 percent at 14 R and 17R.

It is a precarious venture to make categorical evaluations relative to the efficiency of one type of compressor in preference to another, based on spot performance and empirical data. The compression efficiency is a function of numerous factors which contribute to the dynamic losses of entrance and exit from the compressor. If a piston machine is involved, the type of valve is pertinent, the free lift area and its ratio to the piston area are all significant factors. The rpm count, the piston stroke and the mol weight of the gas are also important. These variables are all put together in the CALCULATION SHEET given on page 123 in Reference 1.

MYCOM was fortunate in securing several high compression, (20 atm.) cryogenic applications which involved both piston and screw compressors in both single and two-stage configurations. Another fortunate coincidence was that the principal units were designed for and tested with helium gas. The test gases also included the use of R-12, R-22, ammonia and air. The slippage calculations and other techniques that involve the mol weight of the gases were demonstrated to be valid as applied in References 1, 2 and 3. The volumetric efficiency for a DRY screw compressor includes , a) the suction throttle effect, b) the clearance expansion effect, and c) the slippage losses. The FLOODED compressor is only subject to the clearance expansion effect (b). This follows the thermodynamic equation "EV" given below:

$$EV = 100 - C * R\uparrow (1/k) \qquad \text{Equation 3}$$

See Page 115, Reference 1. Where "C" is the percentage of clearance gas volume that is occupied in the cylinder when the piston or rotor are in a zero sweep position. The term "R" is the ratio of compression or the number of atmospheres (atm) of compression. The vertical arrow indicates that "R" is raised to the (1/k) power. The letter "k" is the ratio of the gas specific heat at constant pressure divided by gas specific heat at constant volume. The "volumetric efficiency" is the ratio of the actual gas handled to the full displacement capacity.

The "compressor efficiency" includes the frictional resistance involved in charging and exhausting the respective gas streams. The term "compression efficiency" has never had a specific definition. The manufacturers guaranteed power charts have been the prime documents used to rate gas compressors. This reference data is empirical. It only has an arbitrary value.

The N.G.P.S.A. issued the 1966 edition of Engineering Data Book which acknowledged the ADIABATIC power as the basic reference value. The 1972 edition reverted the power reference back to status of an insignificant and arbitrary number.

The ADIABATIC horsepower is the basic power reference for all forms of compression. There is no other media that is more efficient. An isothermal operation would require less power, but it does not exist in nature or in fact. All reference data that is taken from thermodynamic tables, Mollier charts and diagrams constitute an isentropic process. Refer to page 115 in Reference 2 for the ADIABATIC head and horsepower evaluations. Refer to pages 20 to 26, Reference 6 for the definition of "Compression Efficiency."

The extended use of screw compressors for cryogenic and heavy chemical processes is contingent upon the effectiveness of the oil separation system. The system shown in Fig. 6 has proven successful for the most critical type of services. The oil separation between the separator No. 2 and No. 4, should reduce the oil content to 1/1400. Adding the charcoal loaded separator No. 5, the oil content should be reduced to 1/3500. Reference 4.

The service target for the helium refrigerators and liquefiers is considered to be 10,000 hours. The screw compressor has achieved this objective. The oil separation facility has only operated for 100 hours of continuous stable performance. The power consumed by the screw compressors during the Tokyo tests was less than that required by the reciprocating units to handle the same load. Our study of these tests gives us the same conclusion.

Footnote: "Liquid Liner" compressors are made by NASH ENGINEERING CORPORATION.

LIST OF REFERENCES

Reference No. 1 — Scheel, L. F.
The Intrinsic Method of Rating Piston Compressors,
Purdue University Conference, July, 1972

Reference No. 2 — Scheel, L. F.
The Sonic Velocity Concept for Rating Volumetric
Efficiency of Rotary Compressors
Purdue University Conference, July, 1972

Reference No. 3 — Scheel, L. F.
Rotary Compressor Rating Methods Updated
Purdue University Conference, July, 1974

Reference No. 4 — Kasahara, K., et al.
New Type Screw Compressor for Helium Liquefiers
and Refrigerators
Suzuki Shokan Co., Ltd., Tokyo

Reference No. 5 — Scheel, L. F.
Gas & Air Compression Machinery
McGraw-Hill Book Co., New York

Reference No. 6 — Scheel, L. F.
Gas Machinery
Gulf Publishing Co., Houston

Reference No. 7 — N.G.P.S.A., Tulsa, Oklahoma
Engineering Data Book, 1966 and 1972 Editions
Compiled by Natural Gas Processors Association

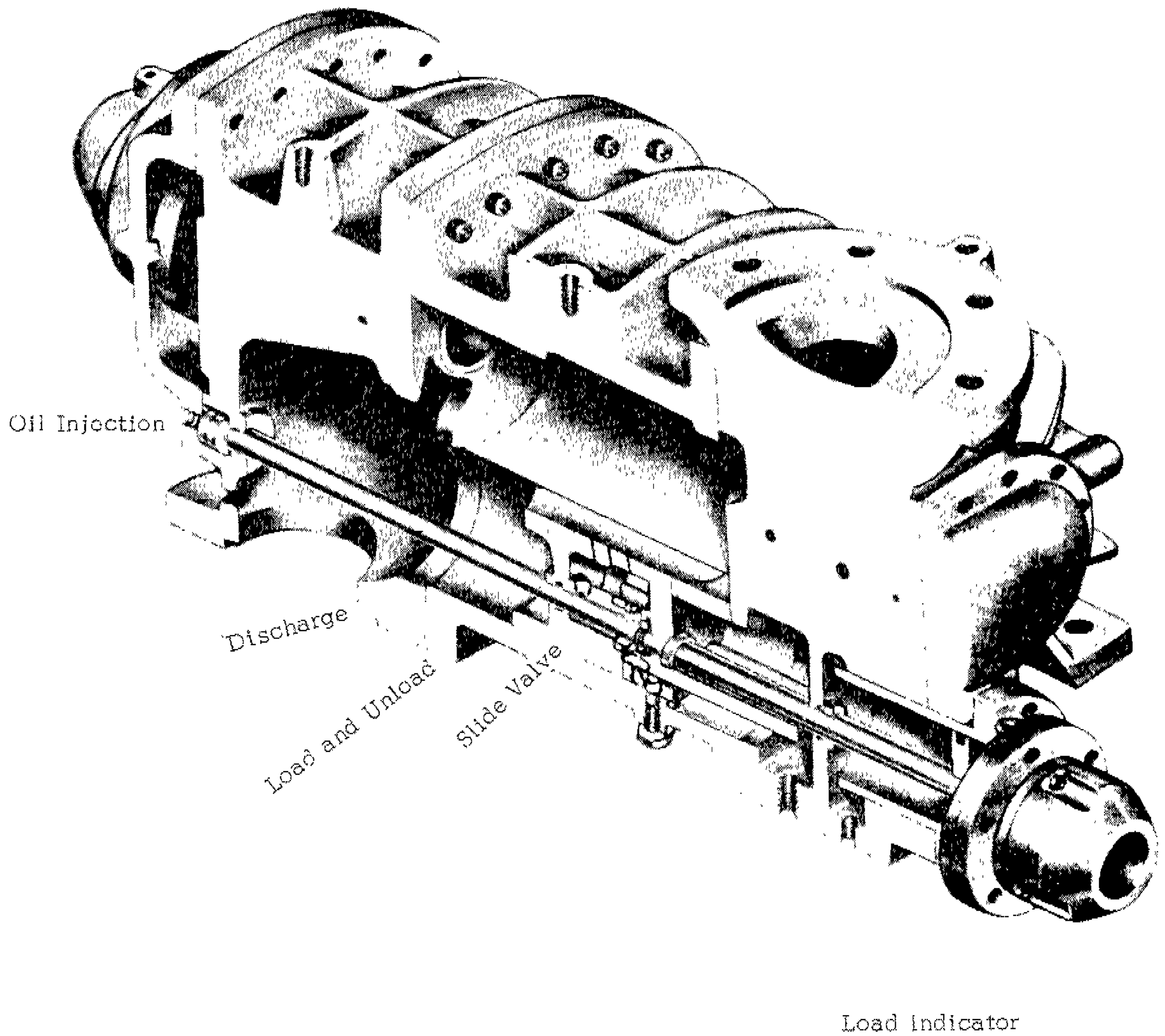


Fig. 1 Illustrates a vertical section of a MYCOM screw compressor with the automatic UNLOADING slide valve assembly in the lower flange and guides. The BUILT-IN discharge port is the void to the left of the UNLOADING piston. The slide valve is moved hydraulically into discharge housing for unloading.

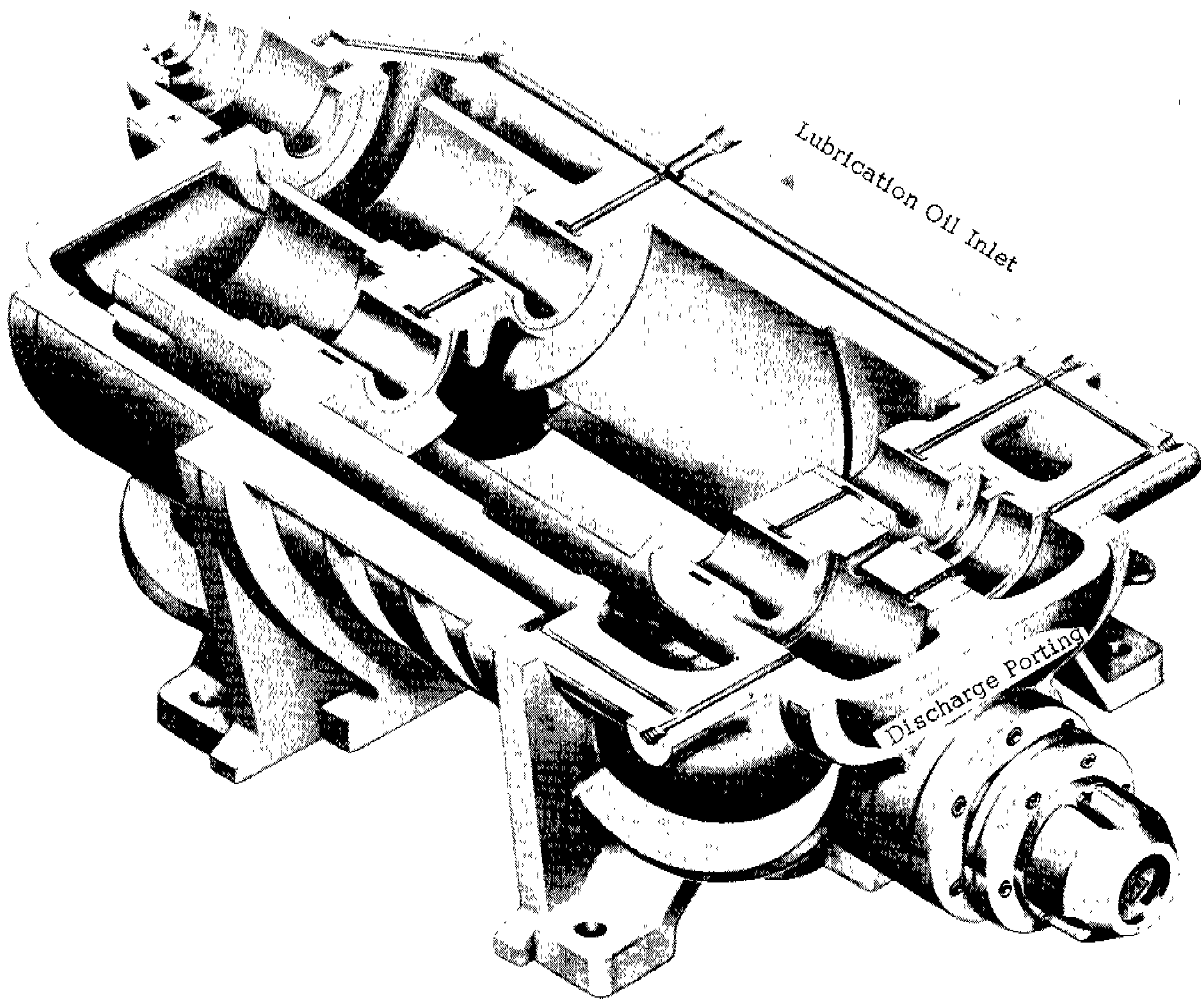


Fig. 2 Illustrates the lower half section of a MYCOM screw without rotors showing lubrication ducts and the BUILT-IN discharge port in the black central spot. The axial discharge port are below the bearings, and radial port is part of slide valve. Shown in fully loaded position.

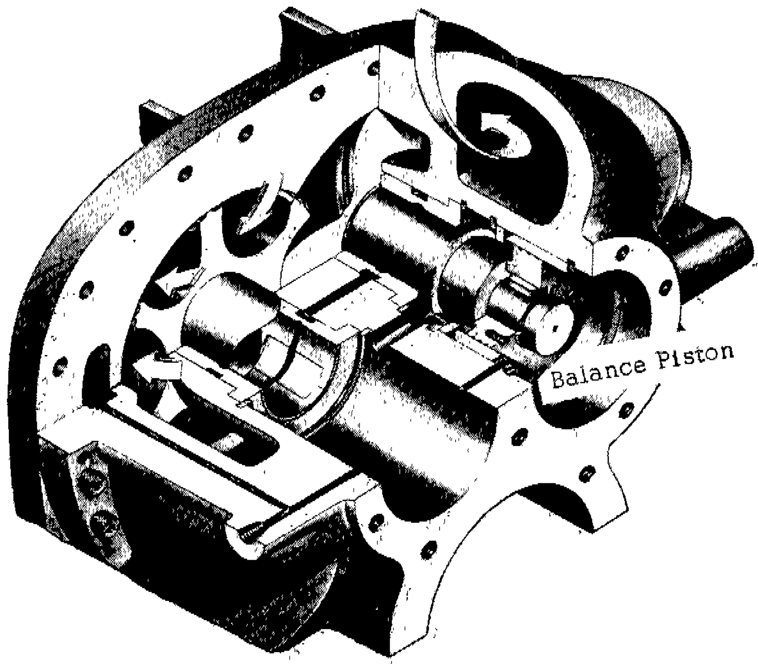


Fig. 3 Shows typical MYCOM suction manifold gas flow path, also balance piston assembly provided to counteract thrust force.

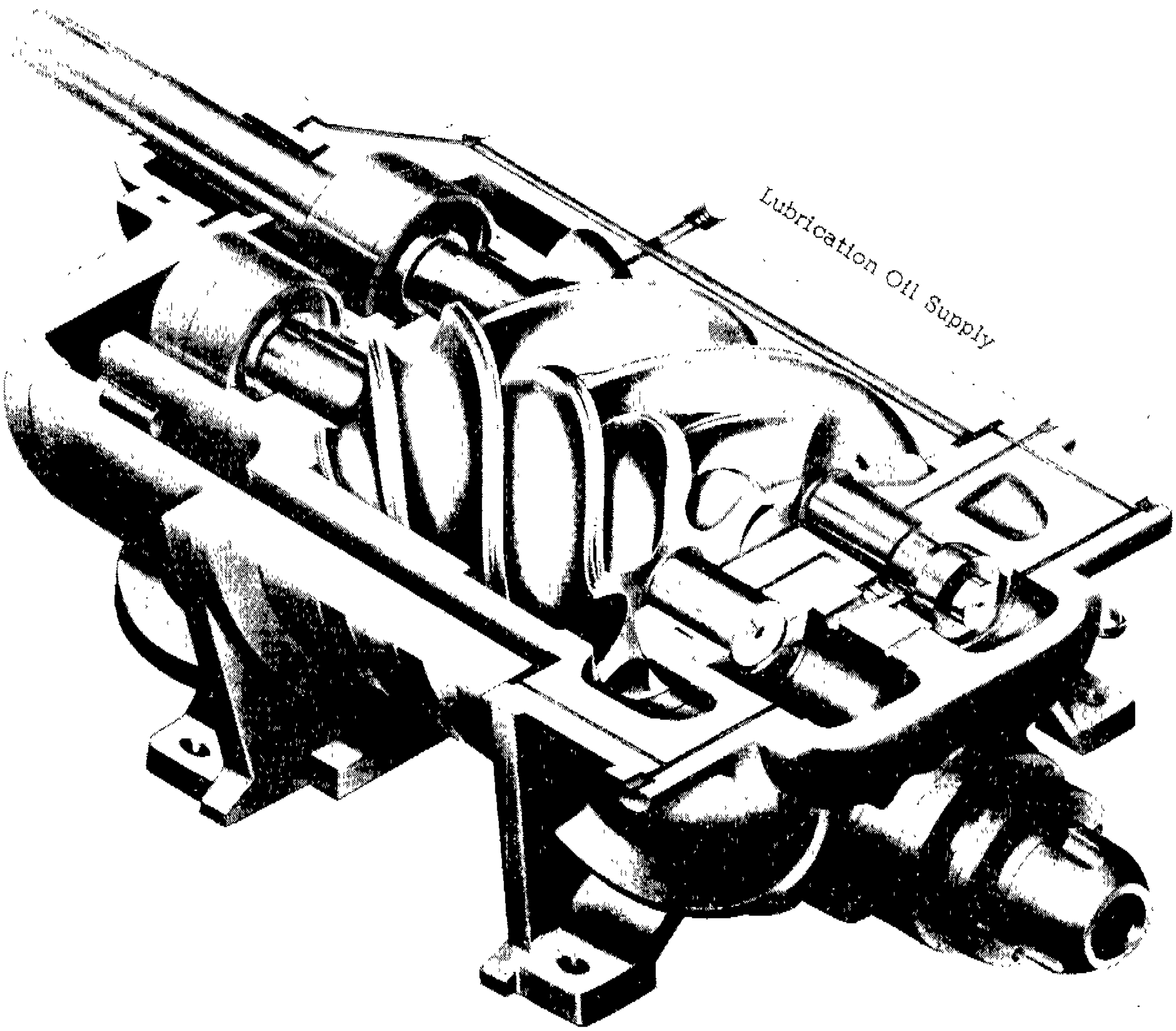


Fig. 4 Shows the MYCOM screw lower half casing with the rotors in place. The lubrication ducts are integral with compressor structure, requiring only one external oil supply connection.

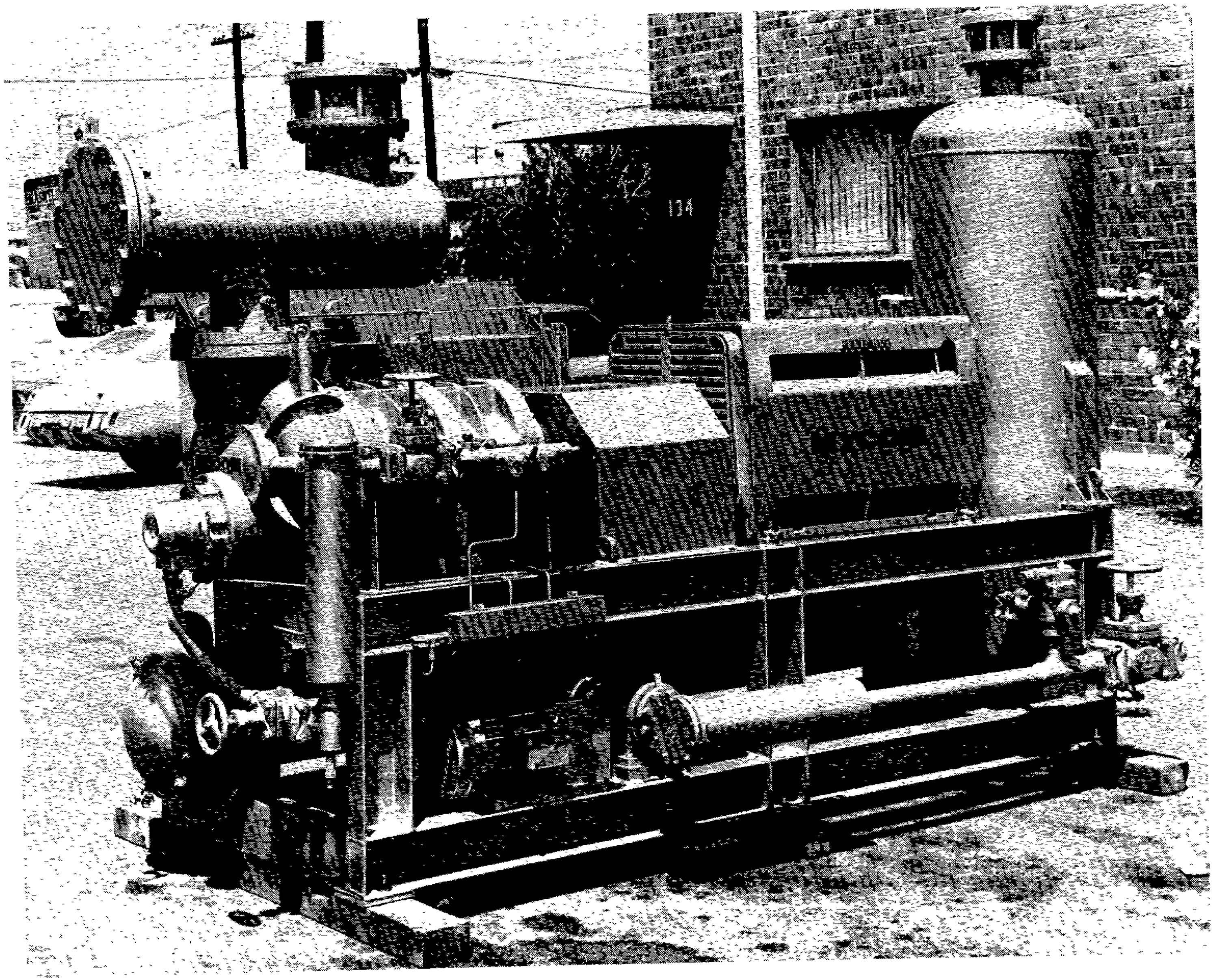


Fig. 5 Shows a typical MYCOM oil FLOODED screw unit with vertical gas-oil separator, oil cooler and filter vessels.

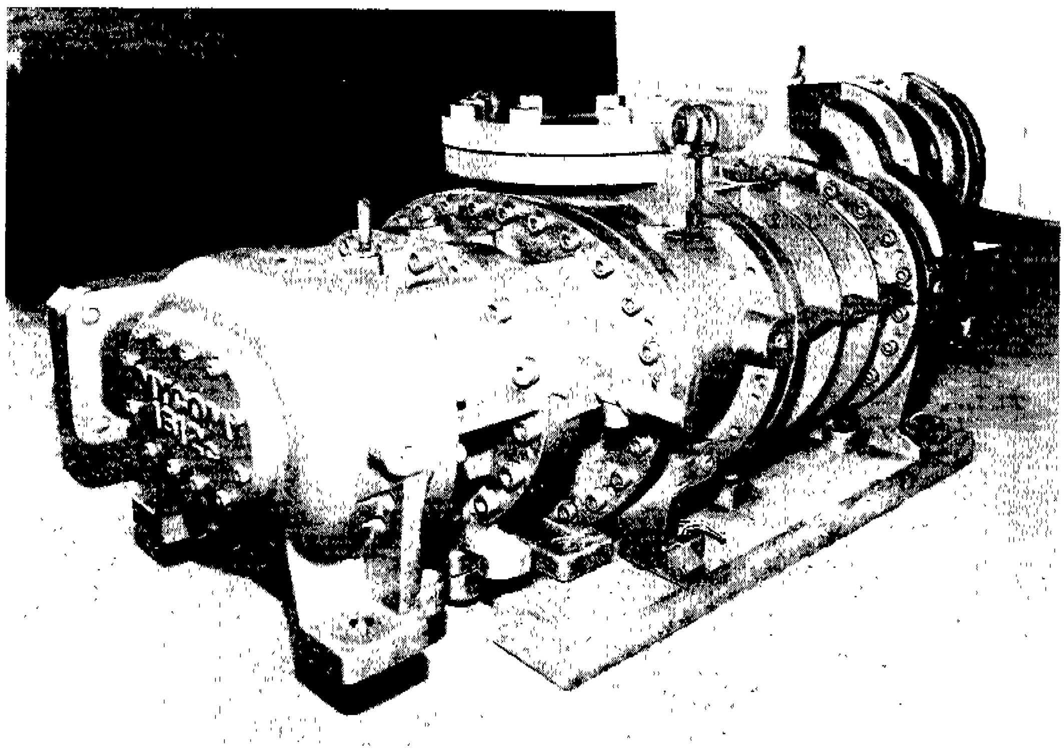


Fig. 6 Illustrates MYCOM two-stage integral screw compressor with single drive shaft, either directly driven by two pole motor, or internally provided with speed up gear for direct drive by four pole motor.

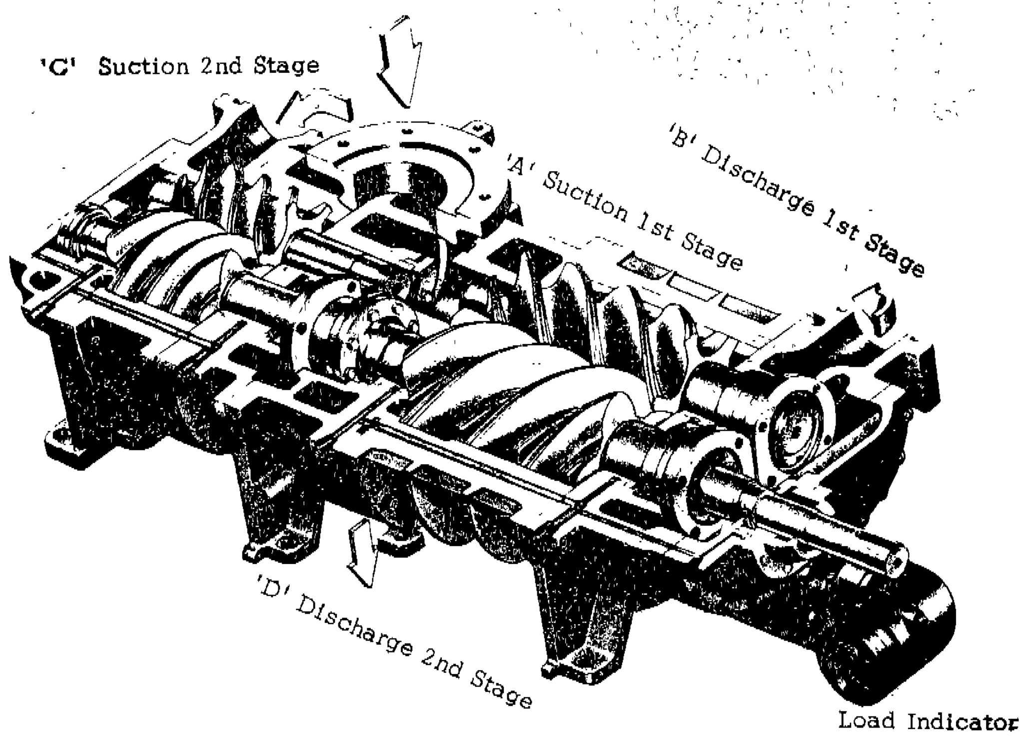


Fig. 7 Shows the horizontal lower casing for a two-stage MYCOM compressor. The male rotor lies on the left side of the casing. The gas flows into first stage at 'A', discharged at 'B', then enters second stage at 'C', discharged at 'D'.

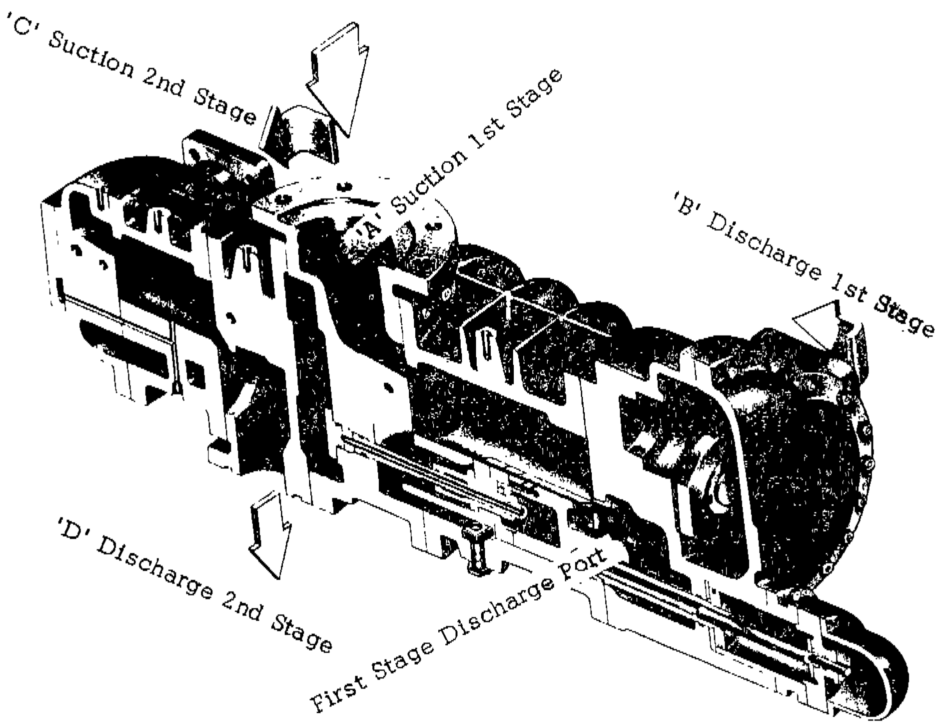


Fig. 8 Illustrates MYCOM two-stage screw vertical section, with rotors removed showing first stage unloader valve, with the BUILT-IN discharge porting and hydraulic control in the lower portion of the casing. First stage is unloaded during start up.

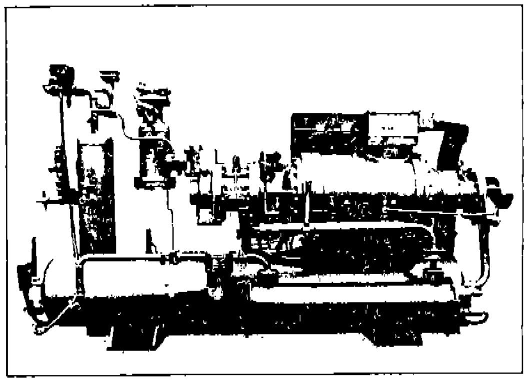
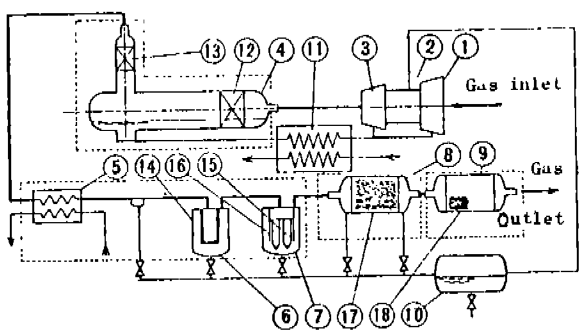


Fig. 9 Shows the MYCOM-two-stage screw unit for HELIUM compression model H-1080FD. Two pairs of rotors are driven by dual shaft semi-hermetic motor.



1. Low stage compressor	10. Oil sump
2. Hermetic motor	11. Oil cooler
3. High stage compressor	12. Demister
4. No.1 Oil separator	13. Demister
5. After-cooler	14. Sintered alloy
6. No.2 Oil separator	15. Glass cloth
7. No.3 Oil separator	16. Baffle
8. No.4 Oil separator	17. Glass wool
9. No.5 Oil separator	18. Active carbon

Fig. 10., Block diagram of oil separation device,

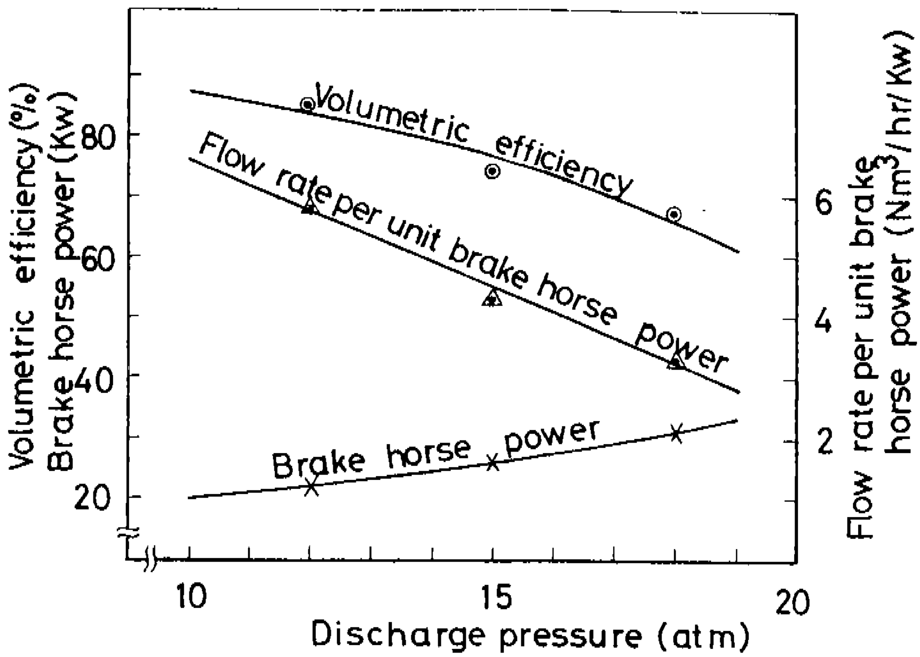
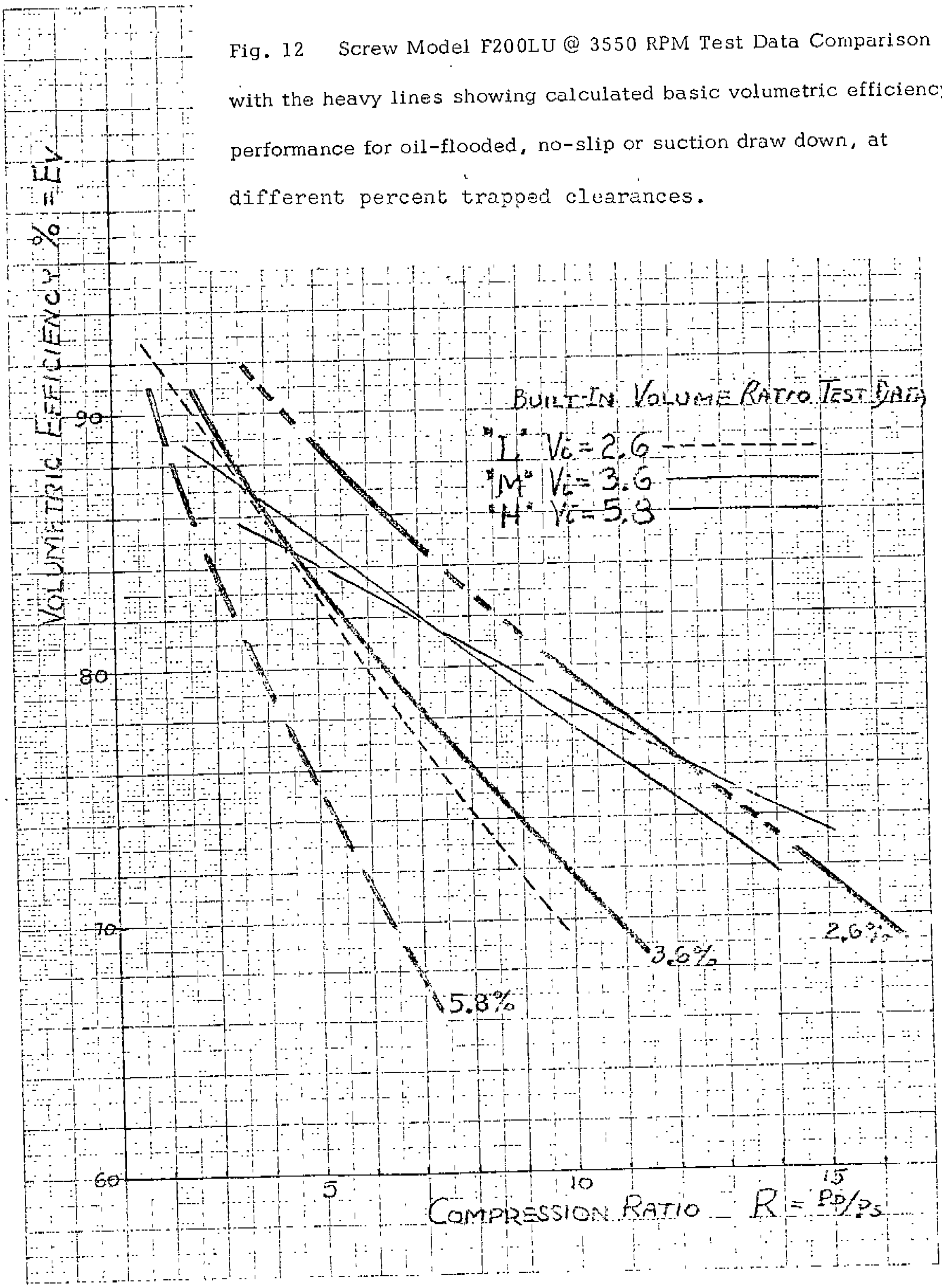


Fig. 11 Shows typical performance data for the two-stage semi-hermetic MYCOM HELIUM compressor unit.

Fig. 12 Screw Model F200LU @ 3550 RPM Test Data Comparison with the heavy lines showing calculated basic volumetric efficiency performance for oil-flooded, no-slip or suction draw down, at different percent trapped clearances.



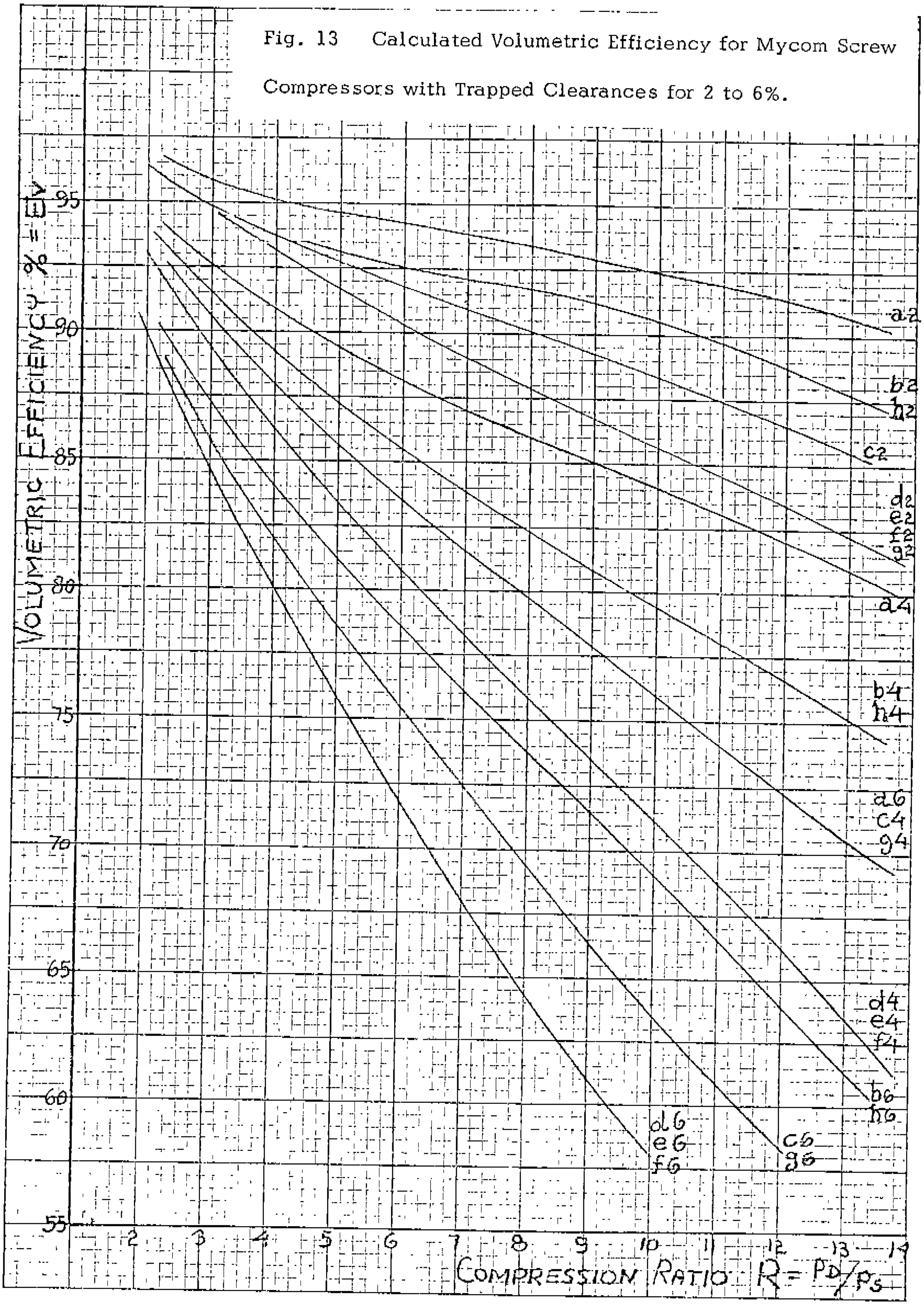


Fig. 13 Calculated Volumetric Efficiency for Mycom Screw Compressors with Trapped Clearances for 2 to 6%.
VOLUMETRIC EFFICIENCY % = EV
95
90
85
80
75
70
65
60
55
2
3
4
5
6
7
8
9
10
11
12
13
14
COMPRESSION RATIO R = PD/PS
a2
b2
h2
c2
d2
e2
f2
g2
a4
b4
h4
a6
c4
g4
d4
e4
f4
b6
h6
d6
e6
f6
c6
g6