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The purpose of this paper is to negate the implications that ROTARY compressors relate to dynamic behavior and Balje' performance charts, as proposed in the 1972 Purdue Proceedings, Ref 1. The rotor tip speed is in no way related to the dynamic head of rotary compressor The head is dependent upon a displacement closure and the SLIPPAGE. The displacement is a function of the rotor geometry, the slippage, the rotor clearances and the other features which are evaluated by the equation given herein. The two fallacious statements are found midway in the RH column of page 116 and midway in the LH column of page 117. Ref 1.

The principal difficulty of the helical and axial screw machines has been its capacity deficiency, by about 15%. This deficiency is believed to be the result of internal by-passing. This recycling creates a greater power burden and reduces the net capacity. The volumetric efficiency developed from Equation E3 for small (2 to 7%) clearances does not satisfy the catalogs volumetric efficiencies. See Fig 10, Ref 1. It contributes in part, but it is more related to the sonic velocity SLIPPAGE. When the expansive function is coupled with the SLIPPAGE, a rational capacity is produced. Equation "S" itemizes the factors which affect the internal by-passing between the pressurized rotor zone and the casing.

$$S = 35\ A\ (k\ T/m)^{0.5}\ R^{\sigma}$$

The constant "35" relates the equation variables to the by-pass volume in terms of cubic feet per minute (cfm). "A" represents the aperture area in square inches, through which the compressed gas returns to the suction chamber. The equations for sizing this area for each type of rotary compressor are described below. The radicals $(k\ T/m)^{0.5}$ and R^{σ} evaluate the intensity of the sonic velocity and the static pressure which affect the velocity of SLIPPAGE.

The escape area is dependent upon a common definition in order to obtain consistent and accurate results.

The escape area for a sliding-vane eccentric-rotor compressor is described by Equation "A1". Herein "GC" represents the circumferential gap between the sliding vane and the casing liner. This value and the fixed axial clearance "GA" is usually set at 0.005 inch. The "GA" clearance also includes the thermal expansion factor of 0.020 and which may be as small as 0.012 inch, depending upon the rotor length and operating temperature. The small letters (d and d') represent the male and female rotor diameter in inches. The figure, #c, represents the number of peripheral cells. The number usually is even and satisfies a 4 inch peripheral arc equation:

$$\#c = \pi d/4\ (+/-)$$

The by-pass aperture for an eccentric rotor machine with sliding-vanes is:

$$A1 = GC^{*}\ L + GA^{*}\ d\pi/\#c, \text{ square inches}$$

A2 represents the aperture area for the twin screw, helical and axial models. The area includes 75% of the combined periphery at a nominal radial GC clearance gap of 0.005 inch. The axial clearances are the same 0.005 and 0.020 inch.

$$A2 = 3.142\ (0.75)\ (d + d')\ GA = \\ = 0.06\ (d + d'), \text{ square inches.}$$

The by-passing aperture for a twin straight lobe, rotary compressor is:

$$A3 = 3\ GA^{*}\ L + 0.2\ d, \text{ square inches.}$$

Where GA is the axial clearance. Usually 0.005 is adequate on the fixed or driving end and 0.020 inch is sufficient for the out-board end which provides for the thermal expansion of the rotor. The constant "0.2" includes a radial clearance of 0.020 inch plus other factors.

SLIPPAGE, Sonic vs Thermal Check

The percent of SLIPPAGE can be checked by the amount of super-heating experienced over the normal adiabatic temperature rise. The equation for a thermal balance follows:

$$T6 = (1 - Y)\ T1 + Y^{*}\ T6\ R^{\sigma} \text{ and}$$
$$Y = (T6 - T2)/(T6 - T1)\ R^{\sigma}$$

Where T2 is the normal adiabatic discharge temperature. T6 is the actual discharge temperature which reflects the amount of suction warm-up experienced from the SLIPPAGE by-pass. R^{σ} is the Ratio of Compression, raised to the exponent "σ", which is $(k - 1)/k$ and "Y" is the decimal percentage of SLIPPAGE. Where the history of such discharge temperatures are sparse, the proximate discharge temperature may be established by applying an exponent multiplier of 1.50. For example, where the air exponent is $(1.4 - 1.0)/1.4 = 0.286$, the corrected value would be 0.286 times 1.50 = 0.430. This is the estimated thermal exponent. This procedure should hold for all so-called DRY operations of the helical-screw design, where $T6 \leqslant 1060$ R, $k < 1.5$ & > 1.15 and $R \leq 4$ for diatomic and triatomic gases.

When the Lube Oil rate is in the proximity of 3 gallons (+/- 33%) per 100 cubic foot of displacement, the machine is said to be operating in an OIL FLOODED condition. This operation differs from the DRY condition in that the emanating noise level is diminished, substantial heat is removed and the discharge temperature is reduced. The normal DRY clearances are sealed, the capacity is improved and the required power is reduced. These improvements make it possible to approach a single-stage Ratio of Compression of ten (R = 10). The disadvantage of FLOODING is the nuisance of separating the oil from the compressed gas.

The temperature rise of the FLOOD mixture can be proximated by applying a multiplier of 0.4 to the adiabatic exponent. For example: where σ is 0.286, use (0.286 times 0.4) = 0.115 = σ'. Presume R = 2.37, $R^{\sigma'} = 1.105$ and $T6 = T1^{*}\ R^{\sigma} = 545\ (1.105) = 603$ R or 143 F. The probable discharge temperature for a DRY cycle operation is, $T6^{*}\ R^{3\sigma/2} = 545\ (1.460) = 798$ R or 388 F.

PENN STATE DRY ROTOR TEST DATA

The DRY test, flow-rate of 374 cfm compares favorably with the calculated flow-rate of 367 or 2% low. The Test discharge temperature of 338 F checks the calculated value of 338 F. The metered horsepower rating indicates a motor efficiency of 90.5%. This is a plausible value. See Table 1, Column B.

It is presumed that the clearance gas expansion realizes its share of power recovery for the DRY cycle. It is further presumed and confirmed by test analysis that there is no trapped clearance gas to expand and subsequently no power recovery for the OIL FLOOD cases.

The expansion volumetric efficiency for the 7% rotor/casing clearance, is 94%. The SLIPPAGE volumetric efficiency is 62.5%. The composite volumetric efficiency is 58.7% at 2.37 R. The SLIPPAGE volume which escapes the confined discharge area, goes into the suction chamber where it is recycled and warms the suction temperature of the fresh charge. Additional power is required for both of these recycling operations. Equation T6 is not applicable to OIL FLOOD operations.

The entrance and exit losses were each assumed to consume 4 velocity-heads of resistance as related to the pitch line speed, making the dynamic efficiency (ED) equal to 93.3 and 96.8% respectively.

OIL FLOOD TESTS

Another identification of OIL FLOOD conditions is when the oil rate is in the proximity of 5 parts per thousand of compressor displacement. The Oil Flood tests are shown on Table I, Column M, N and O for discharge pressures of 90, 100 and 110 psig, respectively. The test, gas flow-rate was 540 cfm versus a calc flow of 510 cfm. A difference of 0.001 inch in measuring the radial clearance could account for this 5.5% discrepancy. The system design should provide for an oil circulation within the parameter described herein and more specifically capable of absorbing 88% of the total compression enthalpy within a temperature differential of 60 degrees F.

The quantity of oil required for these test cases was 21.1 gpm for a 650 cfm displacement machine. The remaining 12% of the compression energy can account for raising the gas temperature about 80 degrees F above the gas inlet temperature. The hypothetical SLIPPAGE for the case of the Flooded Tests is 352 cfm. Only 40% of this likely SLIPPAGE or 140 cfm escaped from discharge pressure zone and entered the suction chamber, by reason of the resistance of the quench oil circulation. The overall effect of the OIL FLOOD is to improve the machine capacity and reduce the power requirement.

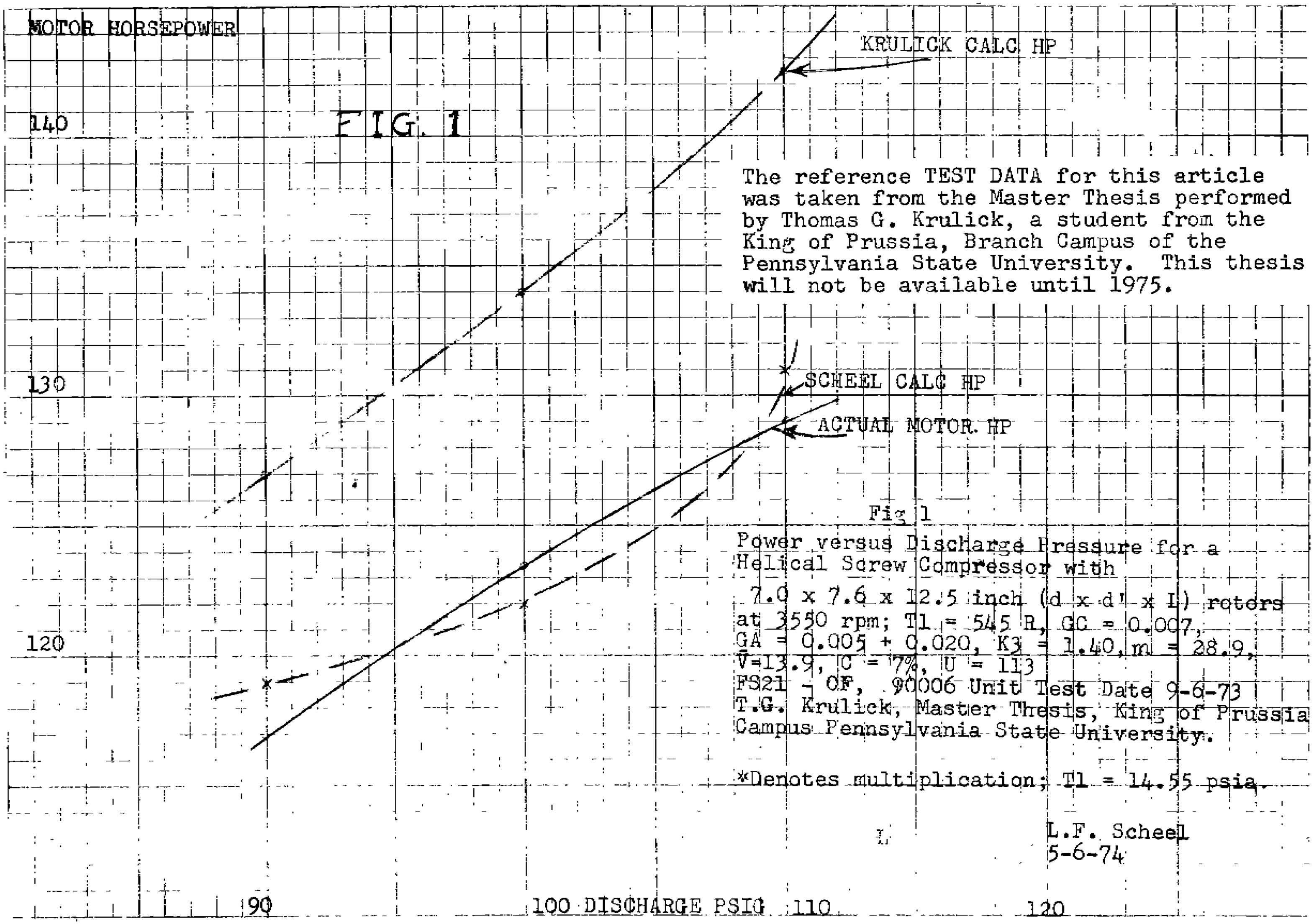
The three calculated power requirements checked the metered power within 3.3%. The average motor efficiency is 95% which is a likely value for 125 hp motor. Fig 1 shows the Bhp required for the 90 to 110 psig array of discharge pressure for the actual tests, the Krulick Thesis calculations and the author's calculated results.

TABLE 1

SUMMARY OF PENN STATE TEST DATA

Test Column	B	M	N	O
Discharge Pressure,	34.6	104.5	114.5	124.5
Adiabatic Head, ft	27,600	77,600	82,200	86,800
Displacement, cfm	650	650	650	650
Metered Cap, cfm	374	510	510	510
Slippage, cfm	244	350	348	357
Slip in Oil Flood, cfm	-	140	140	140
Exp. Vol. Eff. E4	94	100	100	100
Total Vol. Eff. E3	62.5	78.5	78.5	78.5
Calc. Cap.	367	510	510	510
Flow, lb/min.	26.5	37.1	37.1	37.1
Mix Suction °F, T5	167	130	127	133
Psuedo Disc. °F, T2	338	540	570	595
Realistic " °F, T6	338	218	215	222
Slip/Capacity, %, Y	38.4	78.5	78.5	78.5
Adiabatic hp,	22.2	93.2	96.5	98.4
Gas horsepower,	41.3	113.3	115.3	124.5
Motor Meter hp,	45.7	117.0	123.5	129.5
Motor Eff. Gas/Meter	90.5	97.0	93.6	96.0

FIG. 1



The reference TEST DATA for this article was taken from the Master Thesis performed by Thomas G. Krulick, a student from the King of Prussia, Branch Campus of the Pennsylvania State University. This thesis will not be available until 1975.

Fig 1

Power versus Discharge Pressure for a Helical Screw Compressor with 7.0 x 7.6 x 12.5 inch (d x d' x L) rotors at 3550 rpm; T1 = 545 R, GC = 0.007, GA = 0.005 + 0.020, K3 = 1.40, m = 28.9, $\bar{V}$=13.9, C = 7%, U = 113
FS21 - OF, 90006 Unit Test Date 9-6-73
T.G. Krulick, Master Thesis, King of Prussia Campus Pennsylvania State University.

*Denotes multiplication; T1 = 14.55 psia.

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