A Computer Program for Sizing and Performance Evaluation of Reciprocating Process Gas Compressors

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ABSTRACT
Delivering gas to final header storage tanks or end user locations require the use of pipelines and a means of transporting the gas through the lines. A gas compressor is employed satisfactorily to such needs. Gas compressor types vary (from the dynamic type to the positive displacement type) and which to apply for a particular service requirement will depend on the process conditions. For high pressure delivery, and low gas flow volume process conditions, the positive displacement type of reciprocating compressors is often the choice. Matching the compressor equipment manufacturer's available equipment designs to the various process conditions involves several parameters and requires a detailed selection process conducted by the Engineer. A computer program in Microsoft Excel ™ to handle the selection process is described here, with the equations to develop one.

KEYWORDS COMPRESSORS, COMPRESSOR SELECTION, ENGINEERING WITH MICROSOFT EXCEL, GAS COMPRESSOR SIZING, PERFORMANCE EVALUATION OF COMPRESSORS, RECIPROCATING COMPRESSOR SELECTION.

1. INTRODUCTION
The standard practice in the industry application of compressors is to apply positive displacement Reciprocating type Compressors for low flows usually within limits of about 900 cubic metres per minute (m^3/min). Over and above this, the dynamic design Centrifugal type Compressors is the choice by most equipment owners. Reciprocating Compressor Equipment Manufacturers, design various sizes to meet with the variable process conditions. Designs are of two types - Single acting cylinder or a double acting piston. Single acting units are often applied in low gas power of less than 15 kW [1]. In the dual-acting design, compression occurs either when the piston moves on the inward stroke, toward the crankshaft, or outward stroke, away from the crankshaft. Thus, compression occurs on both sides of the piston. For better operational stability and to minimise dynamic loads in operation, a variant of the double-acting with opposed cylinder-piston connection of equal sizes, in pairs, arranged horizontally and connected to a common shaft called a Tandem cylinder arrangement, allows for load balancing. Such units are usually applied to handle larger gas compression requirements and are made up of two or more stages of compression, with each stage being a cylinder-piston connection.
Equipment Sizing or Selection for a Reciprocating Compressor is on a stage-by-stage evaluation basis. Important Considerations in such analysis as stated by [1, 2] will include amongst others:

a. Temperature rise across a stage
b. Frame or Piston rod loading
c. Total pressure rise across a stage
d. Valve design and losses

2. MATHEMATICAL RELATIONSHIPS FOR RECIPROCATING COMPRESSORS

2.1. Overall Compression ratio

$$r_c = \frac{P_2}{P_1} \tag{1}$$

P_2 = Discharge or Final Delivery Pressure (bar)
P_1 = Suction Pressure (bar)

Knowledge of overall or total compression ratio, r_c, i.e. the ratio of the final delivery pressure to the initial suction pressure, is a guide to first estimation of the number of stages, with corrections being made based on temperature rise limitations. Neerken [1] advises to limit compression ratio for a stage to ≤ 3.5.

2.2. Gas Properties and Specific Heat Ratio
Gas for compression can be a single gas or a mixture made up of several gases. A gas analysis is usually conducted to determine mixture properties. A good program development guide is to have a stored database of critical properties of different gases as shown from the selection taken from [3]. Selection can then be made with a combo button. With knowledge of the percentage by volume of the gases, a mole percentage analysis can be conducted to obtain the final mixture properties. Gas Mixture analysis is handled by the method outlined in section 2.28.
The ratio of specific heats, *k*, is obtained from the table of specific heats at constant pressure taken at two temperature values [3].

$$k = \frac{C_p}{C_v} = \frac{C_{pmix}}{(C_{pmix} - 8.314)} \quad (2)$$

C_p = Specific heat at constant pressure
C_v = Specific heat at constant volume
C_{pmix} = Gas mixture specific heat

Proper estimation of the value of the ratio of specific heats, *k*, should be based on average of the values at suction and discharge, k_{ave}. Linear interpolation and extrapolation is applied to obtain the mid-point and out of range value for temperature falling between and outside the ratios of specific heats, *k*, in the database. Thus,

$$k_{ave} = k_0 + \frac{T_{ave} - 0}{100}(k_{100} - k_0) \quad (3)$$

Or

$$k_{ave} = \frac{C_{p-ave}}{C_{p-ave} - 8.314} = \frac{C_{p-0}}{C_{p-0} - 8.314} + \frac{T_{ave} - 0}{100}\left(\frac{C_{p-100}}{C_{p-100} - 8.314} - \frac{C_{p-0}}{C_{p-0} - 8.314}\right) \quad (4)$$

Where, T_{ave} = Average temperature (K)

$$T_{ave} = \frac{T_d + T_s}{2} \quad (5)$$

T_d = Discharge temperature (K)
T_s = Suction temperature (K)

Table 1: Gas .Critical Properties

Chemical Compound	Chemical Formula	Molecular Weight, Mw	Specific of heat ratio, k	Pc (bar)	Tc ($0\,^0$C)	Cp at $0\,^0$C	Cp at $100\,^0$C	Acentric Factor, *w*
Methane	CH4	16.04	1.31	46.4	191.1	34.5	40.13	0.0115
Ethane	C2H5	30.07	1.19	30.07	305.4	49.49	62.14	0.0908
Propane	C3H8	44.097	1.13	42.6	370	68.34	88.68	0.1454
Carbon Dioxide	CO2	44.01	1.3	73.8	304.1	36.04	40.08	0.225
Ethylene	C2H2	28.05	1.24	51.2	283.3	40.9	51.1	0.089

Source: Gresh [3]

2.3. Relative Density and Density of gas

$$\gamma_{sp.gr} = \frac{M_{w-gas}}{M_{w-air}} = \frac{\rho_{gas}}{\rho_{air}} \quad (6)$$

2.4. Loss Factor

Frictional losses in the pistons and rings, piston rod packing, pulsations due to gas surge effects, and oscillations of valves are often accounted for using one of several methods. Scheel [2] suggest that estimates of Pulsation or Pulse Damper Loss be made in line with:

Pulsation or Pulse Damper Loss = 1% of absolute pressure at suction and discharge (7)

Mak [4] has provided a curve fit relationship of the Ludwig loss factor, L_o, curve as a function of the pressure ratio, given by equation (8):

$$L_0 = -0.002188\, r_c^7 + 0.05778\, r_c^6 - 0.6356\, r_c^5 + 3.7711\, r_c^4 - 13.036\, r_c^3 + 26.331\, r_c^2 - 29.019\, r_c + 14.929 \quad (8)$$

The Scheel [2] method is applied in the program.

2.5. Approximate Compression Ratio per Stage

$$r_{c-stg.} = r_c^{\left(1/N_{stg}\right)} \quad (9)$$

Where, $N_{stg.}$ = Number of stages
r_{c-stg} = compression ratio per stage

It was stated earlier that, Single cylinder applications are for low pressure ratios. For high pressure applications, multistage units are the choice to cope with the temperature variations required for efficient performance delivery. The overall compression and hence compression ratio is shared into a number of stages in line with equation (9) to allow for equal work in each stage. Such multistage units often have intercoolers between each stage [1, 5].

2.6. Suction Pressure per Stage

Accounting for the pulsation loss, the corrected suction pressure per stage is:

$$P_s = P_1 - PulsationLoss \tag{10}$$

2.7. Approximate Discharge Pressure per Stage

$$P_{d-stg.} = (r_{c-stg.})(P_s) \tag{11}$$

2.8. Inter-stage Pressure Drop

$$P_{i-drop} \cong 0.1(P_{d-stg.})^{0.7} \tag{12}$$

Gas cooling between stages necessitates, making allowance for the pressure drop in the intercoolers [1]. In selecting compression ratios for a system unit, this is to be considered and adjustments made to computation of the actual discharge pressure in line with equation (12).

2.9. Actual Discharge Pressure

Actual discharge pressure will require that the inter-stage pressure drop be accounted for by adjusting the approximate discharge pressure which is a function of the adjusted suction pressure due to the pulsation losses, and the approximate stage compression ratio. This is given by equation (13):

$$P_{d-actual} = P_{d-stg.} + P_{i-drop} \tag{13}$$

2.10. Actual Compression Ratio per Stage

$$r_{c-actual} = \frac{P_{d-actual}}{P_s} \tag{14}$$

Optimum gas power is achieved when compression ratios in each of the stages are equal for multistage units. However, limits on the gas power capacity of individual cylinders, makes it impossible to obtain equal and hence, balanced compression ratios [6].

2.11. Suction Valve Loss Experienced in Filling Cylinder

$$\theta_s = (aU)^2 \left[\frac{M_w}{10^4 T_1} \right] \tag{15}$$

Where,

θ_s = Suction valve loss
a = Piston/valve area ratio
U = Average piston speed in metre per second (m/s)
M_w = Gas Molecular weight
T_1 = Suction temperature in Kelvin (K)

Scheel [2] gives Piston/Valve area ratio values between 8 and 12 for modern compressors. The Piston/valve area ratio is dependent on the pressures to be handled and valve opening lift, with higher pressures requiring lower valve lifts. Equally, lower Piston/valve area ratio requires a high lift valve. For good design practice [2] suggests an average value of 10. For air and lighter gases, a piston/valve area ratio of 13 is recommended.

2.12. Required Mean Pressure to Exhaust the Cylinder Displacement

$$\theta_d = \frac{\theta_s}{r_c^{(k-1/k)}} \tag{16}$$

2.13. Average Piston Speed

$$U = 2NL \tag{17}$$

U = Piston speed in metre per minute (m/min)
L = Piston Stroke, in metre (m)
N = Design rotating Speed, in RPM

2.14. Intrinsic Correction Factor

$$B = \frac{(1+\theta_d)}{(1-\theta_s)} \tag{18}$$

The intrinsic correction factor, *B*, extends the normal compression ratio, r_c, to represent the actual effective, compression ratio, $r_{c-actual}$, within the cylinder [2].

2.15. Compression Efficiency

This is the ratio of adiabatic efficiency to mechanical efficiency and given by equation (19)

$$\eta_c = \frac{\left[r_c^{\left(k-1/k\right)} - 1\right]}{\left[Br_c^{\left(k-1/k\right)} - 1\right]} \qquad (19)$$

2.16. Suction Inlet Capacity to the Stage

$$Q_s = \frac{Q_{std.}}{60\,x24} \times \frac{P_{std.}}{P_s} \times \frac{T_s}{T_{std.}} \times \frac{Z_s}{1.0} \qquad (20)$$

Standard pressure and temperature have been set by the international committee on weights and measures [7]. For metric standards, the following values apply:

P_{std}. = atmospheric pressure ≈ 1.01325 bar
T_{std}. = standard temperature = 293.15 K (20 deg C)

These values are applied in the program.
This is also the standard adopted by the American Society of Mechanical Engineers (ASME) Power Test Codes Committee [7].
An alternate value adopted by other Engineering standardisation bodies such as the American Gas Association (AGA) is 288.15 K (15 deg C) [7].

Q_{std}. = Inlet Capacity volume at *standard condition (scmd- standard cubic metre per day or m^3/day)*
Q_s = Inlet Capacity volume at *inlet condition (m^3/min)*
P_s = inlet pressure (bar)
T_s = inlet temperature (K)
Z_s = Compressibility at inlet

2.17. Mass Flow

The mass flow rate or weight flow of the gas is defined by the equation (21):

$$M = \frac{Q}{v} = \rho_{gas} Q \qquad (21)$$

Where,
M = Mass flow rate of gas (kg/min)
Q = Volume flow rate (m^3/min)
v = Specific volume of gas (m^3/kg) = $1/\rho_{gas}$

2.18. Specific Volume

$$v = \frac{ZRT}{P} \qquad (22)$$

Where,
Z = Gas Compressibility
R = Gas Constant = $8314/M_w$
M_w = Gas Molecular Weight

2.19. Adiabatic Head of Compression

$$H_{ad} = Z_{av} RT \left(\frac{k}{k-1}\right)_1 \left[r_c^{\left(k-1/k\right)} - 1\right] \qquad (23)$$

H_{ad} = adiabatic head of compression (kN.m/kg)
Z_{av} = average gas compressibility at suction and discharge
Note that the adiabatic head equation (23) is also applied on a stage-by-stage basis.

2.20. Compression Power

$$GP = \frac{MH_{AD}}{\eta_C} \qquad (24)$$

GP = Compression Power or Gas Power (kW)

2.21. Volumetric Efficiency

$$\varepsilon_v = 0.97 - C_c\left[\frac{r_c^{(1/k)} - 1}{\left(Z_2 / Z_1\right)}\right] \qquad (25)$$

Volumetric efficiency, $\varepsilon_{v,}$ defined as the amount of gas volume displaced with each stroke of the piston, decreases with an increase in compression ratio [2], [6].
The retained volume at the end of a piston stroke is the Cylinder clearance, C_c, expressed as a percentage of the swept volume and is usually provided by compressor equipment manufacturers. Scheel [2] gives a minimum clearance value of about 10 %. Suggested value is within the range of 10 % to 15 % for equipment selection purposes [1].

2.22. Cylinder Displacement Required

$$C_{dis} = \frac{Q_s}{\varepsilon_v} \qquad (26)$$

2.23. Cylinder Diameter

Cylinder diameter, *D*, is based on the area of the Head-end.

$$D = \sqrt{\left(\frac{4A_{he}}{\pi}\right)} \qquad (27)$$

2.24. Area of Head-end

$$A_{he} = \frac{\pi D^2}{4} \qquad (28)$$

Where,
A_{he} = Area of Head-end (m^2)
D = cylinder diameter (mm or m)

2.25. Area of Crank-end

$$A_{ce} = A_{he} - \left(\frac{\pi d^2}{4}\right) \qquad (29)$$

Where,
d = piston rod size diameter (mm or m)
A_{ce} = Area of Crank-end (m^2)

2.26. Frame Load in Compression

Frame loads are the maximum permissible forces that can be sustained due to the pressures acting on the piston.

$$F_{LC} = P_d A_{he} - P_s A_{ce} \qquad (30)$$

F_{LC} = frame load in compression (N)

2.27. Frame Load in Tension

$$F_{LT} = P_s A_{he} - P_d A_{ce} \qquad (31)$$

F_{LT} = frame load in tension (N)

2.28. Compressibility Calculation by Redlich-Kwong Equation of State

The calculation of compressibility factor based on the Redlich-Kwong Equation of State (EoS) – Corresponding State Method can be defined by the functional relationship, Z=f(T_r, P_r). Results were within reasonable accuracy. The Redlich-Kwong Corresponding State EoS is of the form given by Edmister [8]:

$$Z^3 - Z^2 + \left[\frac{\Omega_a P_r}{T_r^{2.5}} - \frac{\Omega_b P_r}{T_r}\left(1 + \frac{\Omega_b P_r}{T_r}\right)\right]Z - \frac{\Omega_a \Omega_b P_r^2}{T_r^{3.5}} = 0 \qquad (32)$$

Where,

Ω_a = 0.42748
Ω_b = 0.08664
P_r = Reduced pressure = P/P_c
T_r = Reduced temperature = T/T_c

P_c = Critical Pressure (bar)
T_c = Critical Temperature (K)

2.28.1. Solution to the Redlich-Kwong Cubic Compressibility Equation
A number of methods are available for solving cubic polynomial equations [7, 8]. The method of solution using Microsoft Excel TM is based on the built-in interpolation search solution methods – the Newton method and the Conjugate Gradient method. Noting that the equation (32) is written in a solution form, i.e. the left-hand side equals zero. Using the Solver Add-in option dialog box under the Tools menu, the desired constraints can be set as follows:

Set Target Cell:
Equal To:
Subject to: Guess value:

The Microsoft Excel TM Goal Seek option can also be applied as a solution method, with care being exercised to avoid having circular references – repeated recalculation of particular cell values as input and output.

2.28.2. Handling Gas Mixture
Gas Mixture analysis with the Redlich-Kwong Equation of State is conducted in line with the relations [8]:

$$\frac{T_{cm}}{P_{cm}} = \sum_{i}^{N} x_i \left(\frac{T_{ci}}{P_{ci}} \right) \tag{33}$$

$$\frac{T_{cm}^{2.5}}{P_{cm}} = \left[\sum_{i}^{N} x_i \left(\frac{T_{ci}^{1.25}}{P_{ci}^{0.5}} \right) \right]^2 \tag{34}$$

$$T_{cm} = \left\{ \frac{\left[\sum_{i}^{N} x_i \left(T_{ci}^{1.25} / P_{ci}^{0.5} \right) \right]}{\sum_{i}^{N} x_i \left(T_{ci} / P_{ci} \right)} \right\}^{2/3} \tag{35}$$

$$P_{cm} = \frac{T_{cm}}{\sum_{i}^{N} x_i \left(T_{ci} / P_{ci} \right)} \tag{36}$$

T_{cm} = Mixture Critical Temperature (K)
P_{cm} = Mixture Critical Pressure (bar)
T_{ci} = Individual gas Critical Temperature (K)
P_{ci} = Individual gas Critical Pressure (bar)
x_i = mol percent or percent by volume of each gas

3. Practical Application Example With Program Input/Output
Estimate the sizing and performance requirements for a new compressor application in which 116808 m^3/day of a gas mixture is to be compressed in a 3-stage Reciprocating Compressor. The given data are:

Initial Suction Pressure = 14.3 bar absolute
Final Discharge Pressure = 130 bar absolute

Table 1: Example stage specification

	1st stage	2nd stage	3rd stage
Cylinder clearance (%)	15	15	15
Piston/Valve area ratio	10	10	10
Suction temperature (deg. C)	38	38	38

Design Requirements:
Initial suction pressure drop = 1%
Final pressure drop = 1%
Number of cylinders per stage = 1

Stroke length = 450 mm
Piston rod size = 12.5 mm
Desired operating speed = 277 rpm
The gas mixture consists of the following composition:

Table 2: A Gas Mixture Composition

Component	mole percent (%)
Methane	76
Ethane	8
Propane	2
n-Butane	1
n-Pentane	1
Air	1
Hydrogen	5
Carbon Monoxide	2
Carbon dioxide	3
Hydrogen Sulphide	1

3 stage appl. metric

Reciprocating Compressor Performance and Sizing Calculations

INPUT DATA:

Inlet capacity	M3/day	116808		
Suction Pressure	bara	14.3		
Suction Temperature	deg C	38	38	38
Discharge Pressure	bara	130		
Cylinder Clearance (specify for all 3-stages)	%	15	15	15
Piston/Valve area ratio		10	10	10
Desired Operating Speed	rpm	277		
Stroke	mm	450		
Piston rod size	mm	12.5		
Number of cylinders per stage		1		
Design Requirements:				
initial suction pressure drop	%	1		
final pressure drop	%	1		

Gas Properties (Critical Data) Specification

gas select buttons

mole %	Gas type	Mw	Pc, Bara	Tc, K
76.00	Methane	16.04	46.4	191.1
8.00	Ethane	30.07	48.8	305.4
2.00	Propane	44.097	42.6	370
1.00	n-Butane	58.124	38	425.2
1.00	n-Pentane	72.15	33.7	470.6
1.00	Air	28.966	37.7	132.6
5.00	Hydrogen	2.016	13	33.3
2.00	Carbon monoxide	28.01	35	132.9
3.00	Carbon dioxide	44.01	73.8	304.1
1.00	Hydrogen sulphide	34.08	90	373.9
	undefined	1	1	1
	undefined	1	1	1
	undefined	1	1	1

OUTPUT DATA: Compressor Performance Requirements

Gas Constant, R	428.72
Overall Compression Ratio	9.27
Molecular Weight, Mw	19.39
Critical Pressure,	46.44
Critical Temp.,	207.13

Calculated Output	Units	Stage 1	Stage 2	Stage 3
Ratio of Specific heat, k		1.289	1.289	1.289
Pulse Damper loss	bar	0.143	not defined	1.300
Specific heat exponent ratio		0.224	0.224	0.224
Approximate compression ratio per stage		2.101	2.101	2.101
Approximate Discharge pressure/stage	bara	29.744	62.493	131.300
Suction Pressure per stage	bara	14.157	29.744	62.493
Interstage Pressure drop	bara	1.075	1.808	0.000
Discharge Pressure	bara	30.819	64.301	131.300
Actual compression ratio/stage		2.177	2.162	2.101
Discharge Temperature	deg C	97.355	96.774	94.414
Suction Compressibility factor/stage		0.973	0.946	0.909
Discharge Compressibility factor/stage		0.973	0.959	0.984
Mean Compressibility factor/stage		0.973	0.952	0.947
Inlet Capacity	m3/min	6.103	2.824	1.291
Mass flow	kg/min	66.567	66.567	66.57
Adiabatic Head of compression	kN.m/kg	110342.33	106925.88	102007.84
Compression Power	kW	137.570	132.570	122.774
suction specific volume	m^3/kg	0.09169	0.04243	0.01940
Volumetric Efficiency	%	84.581	84.888	86.217
Compression Efficiency	%	88.896	88.807	88.428
Cylinder displacement required	m^3/min	7.216	3.327	1.498
Average Piston speed	m/min	249.300	249.300	249.300
Cylinder diameter	mm	19.701	13.377	8.976
Area, head-end	mm^2	304.836	140.544	63.276
Area, crank-end	mm^2	182.117	17.825	-59.442
Frame load compression	N	6816.54	8506.89	12022.93
Frame load tension	N	-1297.14	3034.17	11759.09

Figure 1. Microsoft Excel Program Screen Print-out Solution of input/output for the Example

4. Conclusions

The programmed solution method using Microsoft Excel (VBA) for the sizing and performance analysis calculations of reciprocating compressors is greatly simplified in handling the several parameters involved. What-if type of analysis can be conducted by changing certain variables such as suction pressure to fulfil limiting discharge pressure condition. The tedium with certain manual tasks in hand calculation is also eliminated.

5. References

1. R. F. Neerken, Keys to Compressor Selection, *Chemical Engineering*, 1979, pp. 17-33.

2. Scheel, J. F., Gas Machinery, Gulf Publishing, Houston, 1972

3. Gresh, T., Compressor Performance : Selection, Operation and testing of Axial and Centrifugal Compressors, Butterworth-Heinemann, 1991

4. H. Mak, Handheld Calculator Program Helps Size New Reciprocating Compressors, *Oil & Gas Journal*, June 25, 1984, pp. 86-89

5. Brown, R. N., Compressors: Selection and Sizing, Gulf Publishing, 2nd ed., 1997

6. Ludwig, E. E., Applied Process Design for Chemical and Petrochemical Plants, Vol. 3, Gulf Publishing, Houston, 1993

7. Avallone, E. A., ed., Baumeister III, T. ed., Sadagh, A. M. ed., Mark's Standard Handbook of Mechanical Engineers, 11th ed., 90th anniversary ed., McGraw-Hill, 2007, pp. -4:27

8. Edmister, W. C., Applied Hydrocarbon Thermodynamics, Part 1 & 2, 1985

9. Escoe, K., "Mechanical Design of Process Systems", Vol. 1 & 2, 1986

10. Reid, R. C., Sherwood, T. K., The Properties of Gases and Liquids, McGraw Hill, 1966

11. R. C. Reid, Derivative Compressibility factors, *Ind. & Eng. Ch*. Fundamentals, Nov. 1962

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