Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1992

Low Flow Displacement Compressor: Thermodynamical Process Analysis

M. M. Perevozchikov
Technical University of St. Petersburg; Russia

I. B. Pirumov
Technical University of St. Petersburg; Russia

B. S. Chrustalyov
Technical University of St. Petersburg; Russia

K. S. M. Ignatiev
Technical University of St. Petersburg; Russia

A. Taha
Technical University of St. Petersburg; Russia

LOW FLOW DISPLACEMENT COMPRESSOR: THERMODYNAMICAL PROCESS ANALYSIS

M.M.Perevozchikov, I.B.Pirumov, B.S.Chrustalyov, K.M.Ignatiev, A.Taha

Compresssor Department, Technical University of St.Petersburg. St.Petersburg, Russia

ABSTRACT

The present paper deals with some aspects of reciprocating low flow (V<0.2 m3/min) compressor design for the high pressure gas- or air delivery systems.

We have developed two kinds of mathematical models for calculation the parameters of gas in elements of reciprocating compressor. The major differences of approaches and restrictions for engineering purposes are discussed and illustrated by some results of numeric experiment.

INTRODUCTION

The application of the mathematical models makes it possible to investigate into the thermodynamical and gasdynamical processes in the reciprocating and rotary special-purposed compressors for gas- or air delivery systems.

Fig.1 presents the investigated double-staged reciprocating air compressor with discharge pressure of 65 bar. The main features of this kind of compressors are small dimensions (Dc < 40 mm) highly efficient low volume (Vcool < 0.3 Vst) interstage gas coolers, low number of stages (2-3) with pressure ratio within one stage up to 8 and more, sophisticately profilled flexible plates of the valves, sometimes without restrictors. Below are shown some results of mathematical modelling and the comparison of different kinds of models.

MATHEMATICAL MODELS

1. Model of the first kind.

It is well known that traditional methods of reciprocating compressor design are based on the rough mathematical models that deal with polytropic compression and expansion processes and isobaric suction and discharge processes. The loss in the valves and communicating pipelines are supposed to be constant depending on average gas velocities. The values of gas pressure and temperature are obtained with the help of a number of integral equations, comprising the parameters of a gas in different points of the pressure-volume diagram.

Fig.2 presents superimposed pressure-volume diagrams for the first and the second stages of a reciprocating compressor with a ratio Vcool / Vst1 = 0.3 (the pressure loss at suction and discharge is taken into consideration). The compressor discharge pressure is 65 bar, and the compressible gas is assumed to be perfect. We suppose that the process of heat exchange in the interstage gas cooler takes place only along the 2 - 2' line under constant specific volume until the temperatures of the gas and the cooling media level. In point 1 the gas in the cylinder of the first stage instantly mixes (adiabatically) with a portion of gas left in the cooler and the discharge valve opens.

Fig.3 shows that the discharge process from the first stage and the second stage suction are not the isobaric ($p2 >> p1$) throughout the investigated range of the ratio A. It should also be noted that the temperature T3 much decreases due to the effect of gas expansion and cooling during suction process, and therefore the compressor discharge temperature also decreases. The character of the ratio A influence on the temperatures (T2, T2' and T3) leads to conclusion that interstage gas heat exchanger can work either as gas cooler or as gas heater.

The slight power consumption increase at low values of A is not essential. Thus the design of a special-purposed reciprocating compressor requires taking into consideration the interstage communication volumes as well as other constructive parameters.

Unfortunately this kind of mathematical models does not deal with a number of very important aspects of actual working cycle. For example, the real values of the polytropic numbers could hardly be properly estimated because of the simultaneous effects of mass- and heat transfer during the processes 1-2, 2-2', 2'-3 (fig.2). The valve dynamics (especially when the flutter occures) is also out of consideration, and it is so far impossible to estimate the exact power consumption at the first stage discharge and the second stage suction processes because of considerable varying of gas parameters. It is neccessary to modify the mathematical model to overcome these problems.

2. Mathematical model of the second kind

The flow passage of reciprocating compressor is supposed to consist of a series of vessels of either constant or variable volume, pipelines and other communication units such as self-acting valves, control valves and safety valves that could be regarded as concentrated aerodynamic resistors. The gasdynamics equations in partial derivations and thermodynamics equations in differential form should be applied to describe the unsteady flow.

On the basis of our experimental and theoretical researches we came to the conclusion that under certain conditions (cycle frequency less than 25-30 Hz, and no resonant pressure and velocity pulsations in the pipelines) the working processes can be regarded as the quasi-static ones.

Assuming this character of processes we have developed the mathematical model of the second kind. The parameters of gas in vessels are obtained by solving the thermodynamic differential equations, while gas velocities in elements of communications are determined by pressure differences using the steady flow equations with experimentalloss coefficients. The mathematical model deals with each element of the flow passage (block) in particular. Gas leakages through piston seals and valves are omitted. The

compressible gas is perfect. To each block the followig equations can be applied:
Energy equation

$$\frac{dU_j}{d\varphi} = \frac{\alpha}{\omega} F_j (T_{wj} - T_j) + \frac{i_{j-1} m_{j-1}}{\omega} - \frac{i_{j-1} m_{j-1}}{\omega} - P \frac{dV_j}{d\varphi} \tag{1}$$

Mass flow equation

$$\frac{dM_j}{d\varphi} = (m_{j-1} - m_j) \frac{1}{\omega} \tag{2}$$

Gas density is obtained from

$$\rho_j = \frac{M_j}{V_j} \tag{3}$$

Caloric equation

$$f_1(P, T, U) = 0, \; T_j = \frac{U_j}{c_v} \tag{4}$$

Gas state equation

$$f_2(P, \rho, T) = 0, \; P_j = \rho_j R T_j \tag{5}$$

Enthalpy is obtained from

$$i_j = c_p T_j \tag{6}$$

Heat transfer coefficient α and wall temperature Tw are independent of time and estimated either by experimental data or by generally known methods using average mass flow through the gas cooler.

The mass flow through the j-th element of communications is obtained by following equations:
if $P_j > P_{j+1}$

$$m_j = \mu_j \rho_{j+1} F_j \sqrt{\frac{2(P_j - P_{j+1})}{\rho_j}} \tag{7}$$

or

$$m_j = \mu_j \rho_{j+1} F_j \sqrt{\frac{2k}{k-1} R T_j \left(1 - \left(\frac{P_{j+1}}{P_j}\right)^{\frac{k-1}{k}}\right)} \tag{8}$$

if $P_j < P_{j+1}$

$$m_j = 0 \tag{9}$$

The passage area of a valve at any time is defined by differential equation of valve dynamics. We have found out that the majority of valves can be regarded as single-mass system, and the corresponding equation (for discharge valve) is as follows:

$$F_j = l_j h_{j\,max} \chi_j \tag{10}$$

$$\frac{d^2 \chi_j}{d\varphi^2} = B_j \xi_p (P_j - P_{j+1}) - \nu_j^2 (\chi_j + \chi_{0j}) - \eta \frac{d\chi_j}{d\varphi} \tag{11}$$

It should be noted that all the neccessary integral characteristics of compressor (power consumption, temperature, flow rate etc.) are obtained as well.

RESULTS

Fig.4. presents some results of numeric experiment variants. They demonstrate the possibilities of the question. First, the unusual perfomance of the second stage discharge valve was observed. It closes prematurely and opens soon afterwards for very short phase of dischaarge due to the influence of valve chamber volume. Secondly, the simultaneons effects of unsteady heat- and mass transfer in outlet gas cooler cause under certain conditions the reverse flow in the discharge pipeline. The analysis of interstage cooler volume influence on the perfomance of compressor confirms in general the data shown on fig.3.

CONCLUSIONS

It was proved that the proper design of a reciprocating compressor can be carried out only with help of detailed mathematical model of the second kind.

The considerable decrease of compressor discharge temperature can be achieved by proper selecton of the interstage gas cooler volume.

The interstage communication volumes are much fluent on the whole compressor perfomance (especially the valve perfomances) and should not only be taken into consideration but thoroughly calculated.

It was shown that under certain conditions the gas coolers of both stages can work either as gas cooler or gas heater. There is a possibility of reverse flow achieving as much as 20 % of the direct flow maximal speed.

NOMENCLATURE

P	pressure
T	temperature
ρ	density
U	internal energy
M	gas mass
D	cylinder diameter
i	enthalpy
φ	crank angle
F	heat transfer surface
V	volume
m	mass flow
ω	angular speed
α	heat transfer coefficient
R	gas constant
C_v	specific heat at constant volume
C_p	specific heat at constant prassure
l	valve perimeter
μ	valve flow coefficient
ξ	flow force coefficient
B	gas force complex
ν	kinematic viscosity

χ dimensionless valve plate lift
χ_o predeformation of valve spring
h_{max} maximal valve plate lift
η aerodynamic damping coefficient
k isentropic coefficient (C_p / C_v)
A_1 $=V_{cool}$ / V_{st1}
A_2 $=V_{st2}$ / V_{st1}

SUBSCRIPTS

st - stage
cool - cooler
c - cylinder
w - wall

REFERENCE

1. Frencel M.I. "Reciprocating Compressors", Mashinostrojenije, Moscow, 1969.

2. Fotin B.S.,Pirumov I.B."Reciprocating Compressors",Mashinostrojenije, Leningrad,USSR, 1987.

3. Pirumov I.B."Valve of Reciprocating Compressor. Simulation and Optimisation",Transactions of the LPI, Leningrad, USSR, N 384, PP 83-91,1982.

4. Pirumov I.B.,Chrustalyov B.S. "Problems of dynamic valve investigations", CINTICHIMNEFTEMASH, N 306/76, USSR, 1976.

5. Ignatiev K.M., Perevozchikov M.M. "Hardware and Software Development of Experimental Reciprocating Compressor Computer" Proc. of the VI conference of compressor reseach, disign and thechology, Kazan, Russia,1991.

6. Perevozchikov M.M. "CAD for Development Low Flow Reciprocating Compressor", Proc. of the VI conference of compressor reseach, design and thechnology, Kazan, Russia, 1991.

7. W.Soedel. Introduction to Computer Simulation of Positive Displacement Type Compressors. 1972 Shot Course Text, Rey W. Herrick Laboratories.

8. Kakuda M.,Kitora Y.,Hirahara T.,Yamamoto T.,"Investigation of pressure Pulsation in Suction Pipe on Rotary Compressor ", Purdue University, School of Mechanical Engineering, 1988.

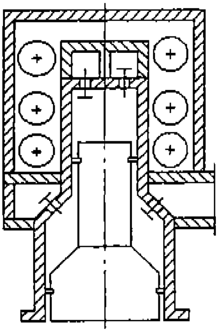


fig.1 The schematic view of the double stage compressor.

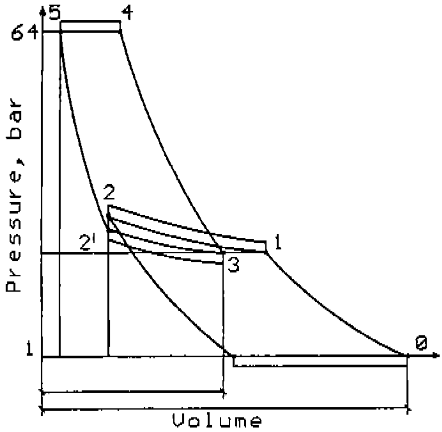


fig.2 Theoretical pressure-volume diagram.

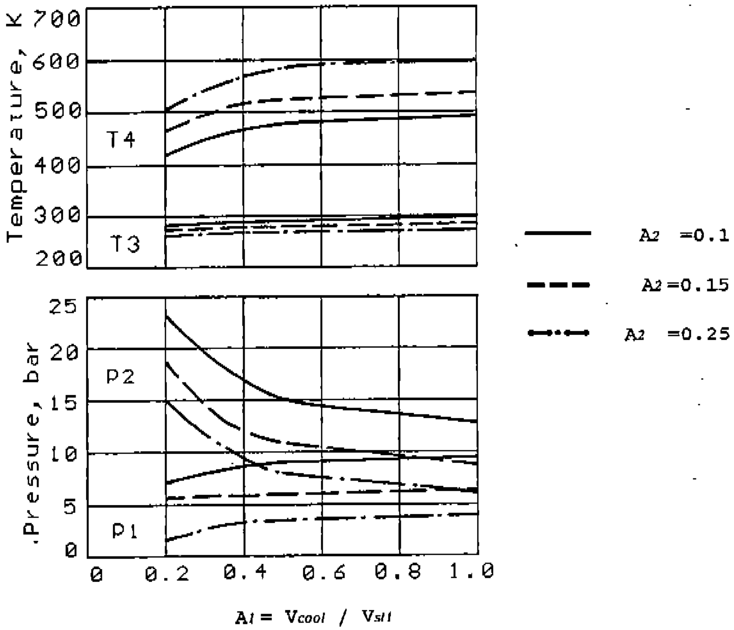
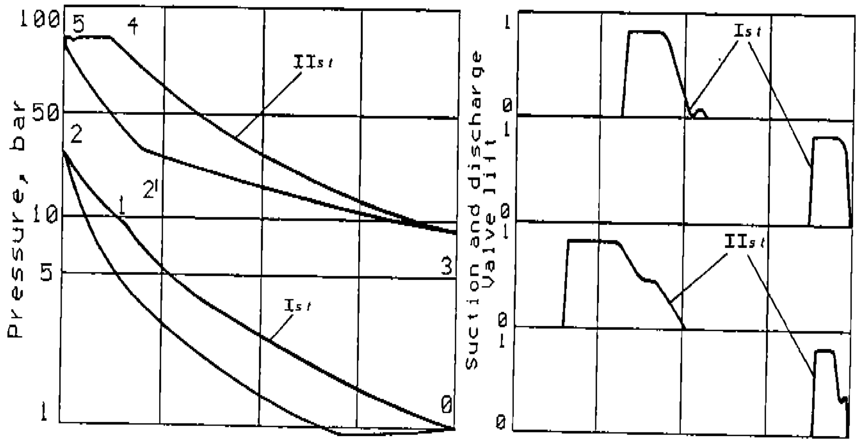
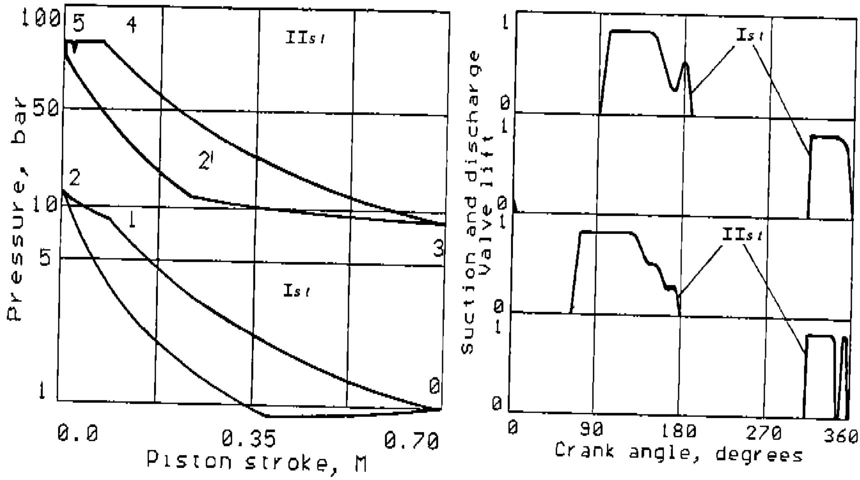


fig.3 Effect of cooler volume on the maximal and minimum pressure and temperature.



$V_{cool} = 0.1\ V_{st1}$



$V_{cool} = 0.3\ V_{st1}$

fig.4 Effect of volume cooler on the working processes.