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Back Flow Effect on Effective Flow Area at Reed Valve Section of a Hermetic Compressor

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Abstract: Valve design is one of the most important aspects of compressor design. In designing the valve system for reciprocating compressors four main features related to the valve performance are sought: Fast response, high mass flow rate, low pressure drop when opened and good backflow blockage when closed. For this reason and in order to obtain an optimum valve system, it is crucial to recognize and predict the phenomena associated with the flow through the valve. The present research is to analyze the effective flow area across the valve section of a hermetic compressor with back flow condition experimentally. Mass flow rate has been an important parameter in analyzing the effective flow area at the valve section. Experimental set up being arranged in order to compute the mass flow rate through the valve section for various valve lifts. Using continuity equation for steady one-dimensional flow, mass flow rate through the valve section is calculated analytically. Effective flow area in terms of lift is obtained experimentally taking the design parameters like flow rate, discharge coefficient, area of cross section and static pressure across the valve section. The effective flow areas are higher at 60% of the valve lift in case of back flow conditions. These experimental results were compared with the analytical values with 5% variation, well with in the accepted limits. These results would be useful in optimizing the performance of reciprocating compressors.

Key words: Effective flow area, valve lift, reciprocating compressor, inlet pressure and back pressure, India

INTRODUCTION

Reed valves of hermetic compressors are operated depending on the pressure difference across the valve section established by the piston motion. The pressure flow field is important for the resultant force, acting on the reed valve, once they are opened. Automatic reed valves have a complex mathematical behavior though they have geometric simplicity which requires simplifying hypotheses. Several studies available in the literature pay little attention to the behavior of the flow but more on the valve dynamics.

Performance and effective valve life requires optimum valve design. Optimum performance is not possible unless modeling of the valve behavior is done. Valves in a compressor can work optimally only in a small range of operating conditions like gas composition, temperature, suction and discharge pressure, etc. When such parameters are altered valve flow area has to be modified for the effective working of compressor and valves.

Predicting the coefficients analytically is necessary in determining the mass flow rate through a refrigeration compressor valving system as reported by Deunie D. Schwerzler. An analytical model of the refrigeration compressor is essential in the refrigeration field to improve its performance by optimization. Therefore, design of valve system has a key role to play in the most critical part in evaluating the performance of the compressor.

Reed valve section: Costagliola (1950) was the first to model the behavior of compressor valves in the full environment of a reciprocating compressor. The work became a milestone for the computation of valve openings. The model has an equation of motion for the valve plate comprising of spring force, gravity force and flow force. The flow force was modeled as the valve pressure difference and the port area.

Generally, reed valve flow is modeled assuming a steady isentropic flow through the port by considering subsonic compressibility factor. Two empirical coefficients had to be found from steady flow experiments. Costagliola (1950) compared this theory with experiments and assumed the discrepancies to be caused

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by leakage and heat transfer. The simplified geometry of the reed type valves in reciprocating compressors is a radial diffuser with axial feeding was carried out by Deschamps *et al.* (1996) in order to model turbulent flow. The refrigerant gas flowing through the valve system is characterized by the Reynolds number based on the feeding orifice diameter d, due to its claimed capability to predict recirculation region and adverse pressure gradient.

Reed valve lift-problem identification: Valve failures are mainly classified into design and operational problems. In evaluating valve failures the factors to be considered are valve lift, plate impact velocity, operating at off-design conditions, valve flutter in the gas stream. By installing redesigned and modified valve the vale failures can be successful resolved. Enhanced valve durability is achieved at the cost of compressor efficiency. Rectification of valve failures depends on apt analysis of valve flow in the compressor by Chaykosky (2002). In evaluating valve failures, valve lift is one of the important factors and it affects compressor performance as per Howes and Long (2001).

Reciprocating compressors in general, small reciprocating compressors in domestic refrigerators, employ fully automatic valves operated by the pressure difference across them. Thus, these reed valves design requires a thorough analysis of the fluid flow through them. The actual dimensions of those valves together with their operation at high frequencies are impediments for experimental determination of pressure and velocity fields.

The complex geometry and the strong interaction between the flow and the valve dynamics are obstacles for development of valve models. In this regard, designers and researchers have relied on simplified geometrical and physical models to understand the operation of automatic valves. By approximating the continuous dynamic states with a series of quasi-static stages, the dynamics of the valve displacement can be handled in a satisfactory manner.

Steady state information of the force acting on the valve as a function of the valve opening can be used to compute the valve dynamics step by step to predict flows that include features, such as stagnation and recirculation regions, curvature and adverse pressure gradients. In this present research, the influence of complex geometry and recirculation region is found experimentally.

By comparison the simulation results of the valve opening and the experimental results are in good agreement, although few coefficients were considered as parameters. Hamilton and Schwerzler (1978) realized the importance of predicting valve impact speeds for the estimation of valve lifetime. They correlated the impact speed to the compressor speed, although some discrepancies between theory and experiments occur in the closing times of the valves.

The first time a semi-empirical theory included the 2D motion of the valve plate was given by Arzand-Daurelle *et al.* (1998). This theory analyses the real valve plate motion, valve port size, pressure gradients and impact speeds. Considering valve as a mechanical devise the aspects of fluid dynamics are not primary for valve design in the literature. As on today, in all the papers practical conclusions are made from semi-empirical valve theories and the emphasis was on the valve environment.

Effective flow area: Effective flow area is obtained by considering the restrictions in the valve system as simple orifices in parallel and in series. The concept of incompressible flow was used to resolve the governing equations of the mass flow rate throughout the valve system.

Once the valves are open, the flow dictates the pressure distribution on the valve reed surface and consequently, the resultant force that will govern the valve dynamics and its displacement from the seat with effective flow and force areas being used to evaluate the dynamics and mass flow rate for the suction valve. The experimental set-up based on the operation of the system.

The force needed to cause the valve movement in a Hermetic compressor simulation programs is usually obtained via the effective force area (A_f). A_f is determined from the pressure difference across the valve $(\Delta P)_v$. The effective force area can be understood as a parameter related to how efficiently the pressure difference $(\Delta P)_v$ is used to open the valve. This rise in A_f is an outcome from the substantial decline in the negative pressure levels as the valve lift is increased. The effective flow area (A_v) is an useful parameter in the valve design and is related to the pressure drop through the valve. Given a pressure drop, A_v can yield the mass flow rate across the valve.

Usually in numerical analysis of compressors, two parameters related to the valve efficiency are required: The effective force area (A_f) and the effective flow area (A_v). The force needed to cause the valve movement in compressor simulation is usually obtained via the effective force area (A_f). A_f is determined from the pressure difference across the valve $(\Delta p)_v$. The effective force area can be understood as a parameter related to how efficiently the pressure difference $(\Delta p)_v$ is used to open the valve. The rise in A_f is an outcome from the substantial decline in the negative pressure levels as the valve lift is increased.

The effective flow area (A_v) is useful parameter in the valve design and is related to the pressure drop through the valve. Given a pressure drop, A_v can yield the mass flow rate across the valve. The higher A_v the lesser the loss of energy and therefore, the better the valve performance. As the valve lift is increased the efficiency of the valve in conducting the fluid also increases.

Analytical solution: Using continuity equation for steady and one-dimensional flow, mass flow rate through the valve section is calculated by Soedel (1972) (Fig. 1).

The thermodynamic processes consisting of expansions through the suction and discharge valves result in mass flow into and out of the cylinder control volume through valves. The assumptions made in the derivation of the mass flow equation are as follows:

- One-dimensional isentropic flow. The steady flow equations can be applied to calculate the instant values occurring during unsteady flow
- The up stream conditions can be considered to be stagnation conditions
- Flow coefficient under steady conditions is the same under unsteady or dynamic conditions. The flow coefficients are the same for normal and back flow
- The open valve, no matter what its configuration can be treated instantaneously as a simple orifice of a certain effective cross sectional area

$$\dot{m}_v = A_v P_u \sqrt{\{2k/(k-1)RT_u\}} \sqrt{\{(P_v/P_u)^{2/k} - (P_v/P_u)^{k+1/k}\}} \tag{1}$$

This equation is valid for un-choked, sub-critical flow or $P_V/P_U > P_{crit}/P_U$ (Soedel, 1972). For critical flow, $P_{crit}/P_U = (2/k+1)^{k/k-1} = r_c$ with an assumption $P_d = P_v$. For normal un-choked conditions with:

$$r = P_d/P_u, \dot{m}_v = A_v P_u \sqrt{\{2k/(k-1)RT_u\}} \sqrt{\{r^{2/k} - r^{k+1/k}\}} \tag{2}$$

The pressure above the orifice (P_{or}), the pressure differential across the orifice Δp_{or}, the temperature of the air above the valve (T_u) and the temperature of the air above the orifice (T_{or}) are functions of valve lift and the differential pressure across the valve. The mass flow rate through the orifice can be written as:

$$\dot{m}_{or} = K_{or} A_{or} Y_{or} \sqrt{\{2P_{or}\Delta P_{or}/RT_{or}\}} \tag{3}$$

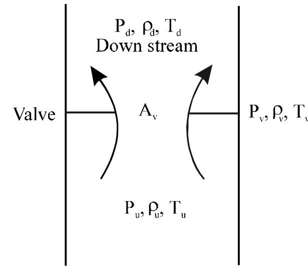

Fig. 1: Valve section

Table 1: Variation of flow coefficients for orifice plates with flange taps

Variables	β	Re: 10^4	Re: 10^5	Re: 10^6
Flange taps (cm)				
4	0.1	0.6134	0.6046	0.6036
	0.3	0.6150	0.6018	0.6003
	0.5	0.6521	0.6275	0.6247
	0.7	0.7630	0.7025	0.6959
8	0.1	0.6118	0.5971	0.5955
	0.3	0.6212	0.6023	0.6002
	0.5	0.6631	0.6271	0.6231
	0.7	0.7985	0.7001	0.6892
Pipe taps (cm)	β	Re:10^4	Re:10^5	Re:10^7
4	0.1	0.6172	0.6070	0.6057
	0.3	0.6565	0.6379	0.6358
	0.5	1.7724	0.7347	0.7306
	0.7	1.0881	0.9868	0.9757
8	0.1	0.6189	0.6027	0.6010
	0.3	0.6591	0.6372	0.6347
	0.5	0.7815	0.7357	0.7305
	0.7	1.1344	0.9887	0.9725

Where, $Y_{or} = 1\text{-}(0.41+0.35\ \beta^4)\ \Delta p_{or}/KP_{or}$. Flow coefficient ($K_{or}$) is found from a Table 1 of flow coefficients versus Reynolds number for fixed β value. The mass flow rates are used to calculate the effective flow area coefficients for the valves. By the continuity equation for steady, one-dimensional flow $\dot{m}_{or} = \dot{m}_v$:

$$K_{or} A_{or} Y_{or} \sqrt{\{2\,P_{or}\Delta P_{or}/RT_{or}\}} = P_u A_v \sqrt{\{2k/(k-1)RT_u\}} \sqrt{\{r^{2/k} - r^{k+1/k}\}} \tag{4}$$

$$A_v = \frac{K_{or} A_{or} Y_{or} \sqrt{\{P_{or}\Delta P_{or}^{(k-1/k)} T_u / T_{or}\}}}{P_u \sqrt{\{r^{2/k} - r^{k+1/k}\}}} \tag{5}$$

Flow coefficients: Generally, the gas flow through a valve is assumed to be a flow through an orifice for a compressible gas is given by Arzand-Daurelle *et al.* (1998) for a perfect gas:

$$m = P_u.A_v.\sqrt{\frac{2\gamma}{(\gamma-1).r.T_{up}}} \sqrt{\frac{2}{{}^{\tau}\gamma^{-\tau}} \frac{\gamma+1}{\gamma}} \tag{6}$$

The basic assumption behind the model was to consider the refrigerant flow through the compressor, except for the valves and inside the cylinder, as a one dimensional steady state current. In this way, it is possible to establish a steady state thermal balance to compute the temperatures and the heat and work flow rates for each component and for the overall system.

The compressor was subdivided into six parts, i.e., shell, compressor body, suction muffler, suction chamber, discharge chamber, discharge line and for each of them; the mass and energy balances were established. The main irreversibility inside the compressor (electric energy conversion losses, friction losses) was taken into account by suitable electrical and mechanical efficiencies.

In Cavallini *et al*. (1996), it is possible to find a detailed description of the governing equations for each compressor component. The compression cycle and the refrigerant flow through the valves and the valve dynamics were analyzed by an unsteady state approach. This allows the direct computation of the heat and work flow rates exchanged together with the mass flow rate processed, as well as the behavior of the main characteristic parameters during the compression cycle. The behavior of the refrigerant mass m inside the cylinder was derived from the following mass balance:

$$dm/dt = m_s(t) - m_d(t) \tag{7}$$

The effective flow area was correlated to the valve plate lift (W) by the discharge coefficient C_D:

$$A_v = \Pi d C_D W \tag{8}$$

Experimental setup and procedure: A fixture is built as shown in Fig. 2 to hold the valve system of the compressor. The fixture is arranged so that the valve reeds can be held at any desired height parallel to the valve seat. The fixture is attached to the end of an orifice flow meter in order to measure the gas flow through the valve system being tested. A pressure gauge is used to measure the pressure before the orifice P_{or} and the pressure differential across the orifice $(\Delta P)_{or}$ is found by using a water manometer. Another manometer is used to measure the pressure differential across the compressor valve $(\Delta P)_v$ installed at the valve test section. Pressure regulators in the gas supply line are adjusted to provide enough flow to the test section to create the desired maximum pressure differential across the compressor valve while the throttling valve in the gas supply line is fully opened as shown in Fig. 3.

The temperatures of the gas are not varied much with time and flow rate. Therefore, the temperatures are measured only once during the experiment. Knowing the pressure before the orifice, the pressure differential across the orifice and the temperatures of the gas flowing through the sections the mass flow rate through the valve is calculated as a function of the valve height and the pressure differential across the compressor valve. The effective flow area is measured both for normal flow and back flow conditions.

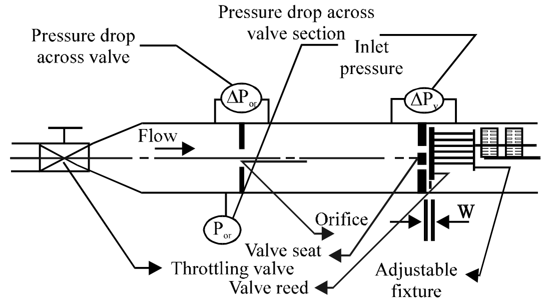

Fig. 2: Effective flow area set up

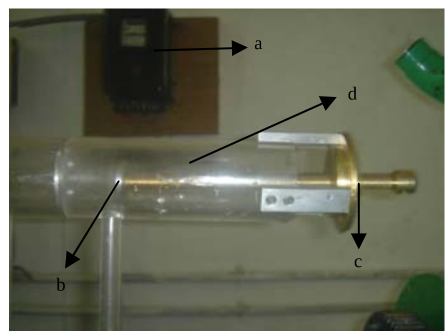

Fig. 3: Adjustable fixture: a) Energy mater; b) Valve section; c) Adjustable fixture for back flow provision; d) Acrylic pipe

RESULTS AND DISCUSSION

Experiments were conducted and effective flow areas were calculated with respect to the rise in valve lifts, for all the inlet pressures (0.05, 0.06, 0.07, 0.08, 0.09 and 0.1 bar). The analytical and experimental values in case of no back flow condition and the analytical and experimental values in case of with back flow condition are compared and the percentage of error is found at all the values of valve lifts (W).

The pressure drop across the valve plate is compared to pressure drop across the orifice plate for all the inlet pressures and for the entire valve lifts. Pressure drop across the valve section both in case of no back flow condition and with back flow condition were studied. Influence of pressure drop across the mass flow rate was compared for both the cases. Relation between volumetric efficiency and the back flow condition was also established in this study.

The effective flow area is increased by increasing the valve lift. The maximum effective flow area is occurring at valve lift, corresponding to W = 0.006 m for all the inlet pressures (P_{or}). Effective flow area, A_v for different values of valve lifts (W = 2-7 mm) and inlet pressures (P_{or} = 0.05-0.10 bars) when compared to the experimental data showed that the experimental set up can provide a good prediction of the important parameters required for valve design and analysis. The maximum discrepancy was found with 5% which was very much with in the acceptable limits.

The effective flow areas are calculated with respect to the rise in valve lifts, for all the inlet pressures (0.05-0.1 bar) where the coefficient of discharge is proportional to pressure drop across the valve plate. As the valve lift increases from 0.002-0.007 m, the effective flow area increases as well for all the in let pressures (0.05-0.1 bar). For all the valve plate lifts, below 0.004 m, the rise in effective flow area is almost linear at all the values of inlet pressures (0.05-0.1 bar) as shown in Fig. 4-7.

Non linear variation in effective flow area is observed, as the valve lift increases beyond 0.004 m which is a due to valve stiffness and valve flutter. When analytically calculated values are compared with the experimental values, the percentage of error as an average, found to be around 7% which was very much with in the limits as evident by comparing Fig. 4 and 5.

In all these results by comparing analytical and experimental values the percentage of error is more at the lower valve plate lifts, i.e, during the linear variations of effective flow area with respect to the valve lift and it decreases as valve lift rises. This is due to the valve stiffness which is more predominant especially at lower valve plate lifts where as valve flutter is more predominant at higher valve lifts.

Explanation about the valve flutter and stiffness mentioned by Soedel (1972), also supports the findings of the present research. Maximum effective flow area occurred almost at the same point, i.e, valve lift for all the inlet pressures which is independent of inlet pressures in the range up to 0.1 bar. Beyond this point, the value of effective flow area in fact gets decreased in spite of the rise in valve lift due to valve stiffness and back flow which was also established by the analysis carried out by Soedel (1972) and Deschamps *et al.* (1996).

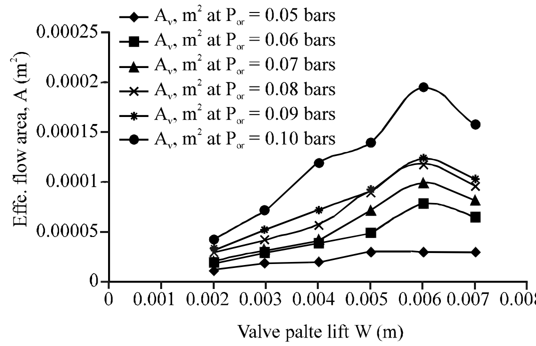

Fig. 4: Variation of effective flow area (A_v) for various valve lifts (W) and for different inlet pressures as per experimental analysis

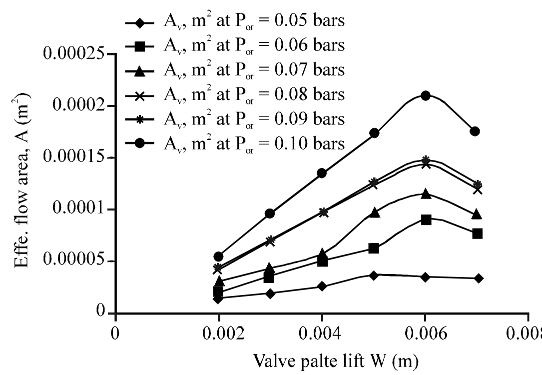

Fig. 5: Variation of effective flow area (A_v) for various valve lifts (W) and for different inlet pressures as per numerical analysis

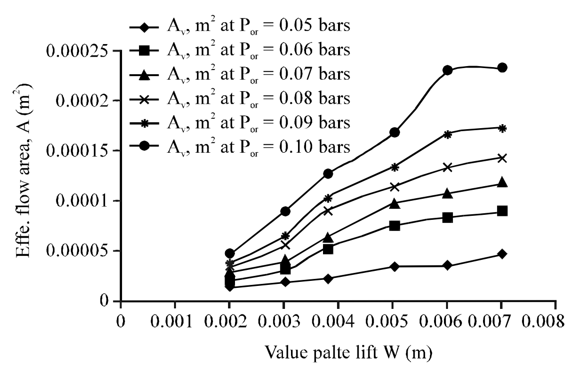

Fig. 6: Variation of effective flow area (A_v) for various valve lifts (W) and for different inlet pressures as per experimental analysis (without backflow condition)

The effect of back flow, valve stiffness and valve flutter were observed and established after the experimentation done with out back flow condition. As shown in Fig. 6-7. Experimental as well as analytical values

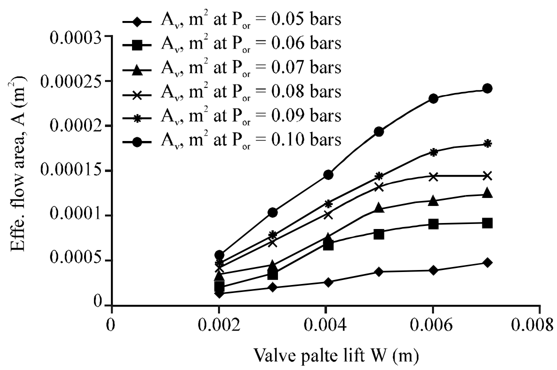

Fig. 7: Variation of effective flow area (A_v) for various valve lifts (W) and for different inlet pressure as per numerical analysis (without backflow condition)

with out back flow condition are more or less similar to the values obtained with back flow condition. In case of with out back flow condition, even beyond the valve lift of 0.006 m, the effective flow area, A_v is increasing unlike in the previous case because of the absence of back flow, in both experimental as well as analytical values.

By comparing the analytical and experimental values in case of no back flow condition and the analytical and experimental values in case of with back flow condition, the percentage of error was less at all the values of valve lifts (W) in case of no back flow condition. In both the cases of back flow and no back flow the percentage of error is more at lower valve lifts due to the influence of valve stiffness. Valve stiffness depends upon the plate thickness and valve material.

CONCLUSION

The results predict the performance of compressor over the entire range of operating conditions and the influence of different design and operating parameters, such as the inlet pressure (P_{or}), valve plate lift (W), on effective flow area (A_v).

Effective flow area (A_v) for different values of valve lifts (W = 2-7 mm) and inlet pressures (P_{or} = 0.05-0.10 bar) when compared to the experimental data, showed that the experimental set up can provide a good prediction of the important parameters required for valve design and analysis. The discrepancy is found around 5% which is very much with in the acceptable limits. A method of calculating the effect of effective flow area (A_v) with respect to valve performance has been established. The effective flow area is increased by increasing the valve lift. The maximum effective flow area is occurring at valve lift, W = 0.006 m (60% of lift) for all inlet pressures, P_{or} of the range 0.05-0.1 bar. Beyond 60% of valve lift the effective flow area does not increase considerably.

NOMENCLATURE

A_v = Effective flow area
K_{or} = Flow coefficient
M = Mach number
P_d = Down stream pressure
P_{or} = Inlet pressures
P_u = Upstream pressure
P_v = Pressure at valve section
R = Gas constant
T = Local temperature
T_u = Upstream temperature
T_d = Down stream temperature
T_u = Stagnation temperature
T_v = Temperature at valve section
d = Valve seat diameter
h = Enthalpy
h_u = Stagnation enthalpy
k = C_p/C_v ratio of specific heats
m_d = Discharge mass flow rate
m_s = Suction mass flow rate
t = Time
v = Velocity
β = Orifice diameter/pipe diameter
ρ_d = Density at down stream
ρ_u = Density at upstream
ρ_v = Density at valve section
τ = Ratio of downstream and upstream pressures

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