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ABSTRACT

Flow losses through the valve ports contribute a significant portion to the total energy loss of a high speed compressor. As a result, in the energy-conscious world of today, compressor designers are always busy finding methods and techniques to minimize these losses. It can be argued that the various fundamentally different compressor concepts would have different valve flow losses simply as a result of the difference in the kinematics of their operation, even if other factors including the valve characteristics were the same.

An attempt has been made in this paper to highlight this difference by way of simple mathematical relationship between the kinematic characteristics of a compressor and the corresponding flow losses. Four different kinematic arrangements of the reciprocating as well as the rotary concepts were picked up for the analysis. The results indicate a rather interesting perspective for the compressor designer to keep in mind.

INTRODUCTION

Considerable amount of work has been done on the modeling of self-actuated valves used in the refrigerating and air conditioning compressors (1, 2, 3, 4, 5, 6, 7, 8). It has been shown by the researchers that energy loss through valves consists of several parts, one of which is the kinetic energy loss due to flow restrictions in the gas path while it is flowing through the valve openings and the ports. The intention in this paper is not to explain the mechanics of these losses but simply to demonstrate analytically the difference that various compressor concepts may create in these losses due to the differences in their kinematics alone.

The kinematic arrangements considered for this analysis are: common reciprocating compressor arrangement where the suction and the discharge plenums are located away from the axis of rotation of the shaft (Figure 1a); the well known scotch-yoke arrangement (Figure 1b); another reciprocating compressor arrangement which I call "reverse reciprocating" arrangement for lack of a better term (Figure 1c). The suction and the discharge plenums are located towards the axis of rotation of the shaft in this case. The last one is the rolling piston type rotary compressor arrangement (Figure 1d).

The approach followed in the analysis is straightforward and very simple. A case has been made that the velocity of the gas through the valve openings and the ports depends not only on the effective flow area but also on the rate of increase (or decrease) of the volume of the cylinder space which itself is dependent on the kinematics of the compressor. The effective flow area and the total cylinder displacement are assumed to be the same for all of the four cases. The expressions for volume and pressure drop are non-dimensionalized for convenience in comparing the designs. Thus the only input data that are needed are L/R ratio for recips, R_r/R_c ratio for the rotary, and the volume ratios (or pressure ratios) at which the suction and discharge processes begin.

THEORY

Assumptions

Following assumptions have been made to simplify the analysis without much loss of generality.

1. It is assumed that the effective flow area is the same for each case throughout the cycle of operation. It is recognized that the flow area changes throughout the cycle of operation in reality. However, since the objective here is to compare "only" the effect of kinematics, the assumption is valid.

2. It is assumed that the flow through the valve port and into the cylinder is incompressible throughout the suction and discharge processes. The assumption is valid because the change in pressure (and density) during suction or discharge process is very small and, therefore, it will not introduce any significant degree of error.

3. The effect of gas inertia and pulsations is not considered. It may be argued that kinematics of the compressor may not have any significant effect on gas pulsations.

4. Leakage losses during the suction and discharge processes are neglected. Also, the suction and discharge processes are assumed to start at the same volume ratio for all cases.

5. The speed of rotation is assumed to be constant and the same for all cases.

6. The swept volume is assumed to be the same for all cases. With these assumptions, one can easily develop the expression for the pressure drop at the valve port as follows:

$$\Delta P = \frac{1}{2}\frac{\rho}{g}\left(\frac{Q}{A_e}\right)^2$$

$$Q = \frac{dV}{d\theta}\dot{\theta}$$

Thus,

$$\Delta P = \frac{1}{2}\frac{\rho}{g}\left(\frac{\dot{\theta}}{A_e}\frac{dV}{d\theta}\right)^2$$

To non-dimensionalize, the expression for ΔP is divided by the term:

$$\frac{1}{2}\frac{\rho}{g}\left(\frac{\dot{\theta} V_s}{A_e}\right)^2$$

Thus,

$$\Delta P_n = \left(\frac{1}{V_s}\frac{dV}{d\theta}\right)^2$$

From above expression, it is obvious that the pressure drop at any instant is proportional to the square of the rate at which V, i.e. the cylinder volume, changes with θ at that instant. Since the rate of change of cylinder volume very much depends upon the kinematics of the compressor, it is obvious that the pressure drop for each case will be different. To show the extent to which this difference can occur, following example cases were considered:

<u>Common Reciprocating Compressor</u>:

Referring to Figure 1a, the cylinder volume corresponding to angle θ is given by:

$$V = A_p r\left[1 - \cos\theta + \frac{l}{r} - \frac{l}{r}\sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}\right]$$

The non-dimensionalized volume is found by dividing the above expression by the swept volume.

or $$V_n = \frac{1}{2}\left[1 - \cos\theta + \frac{l}{r} - \frac{l}{r}\sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}\right]$$

The corresponding equation for non-dimensionalized pressure drop is:

$$\Delta P_n = \frac{1}{4}\left[\sin\theta + \frac{r}{l}\frac{\sin\theta\cos\theta}{\sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}}\right]^2$$

It should be realized that in the reciprocating compressor, the reexpansion/suction stroke is from 0 degrees to 180 degrees and the compression/discharge stroke is from 180 degrees to 360 degrees. The actual suction process starts at a point where the volume ratio (ratio of the cylinder volume at a particular time to the total swept volume) is equal to a predetermined value R_s, which is kept same for all cases. During the suction process, the volume ratio is always greater than this assumed value. Similarly, the discharge process starts when the volume ratio is equal to a predetermined value R_d and continues as long as the ratio is less than this value. The values of R_s and R_d depend upon the operating conditions of the compressor and the clearance volume.

Scotch Yoke Arrangement

Referring to Figure 1b, the normalized cylinder volume and pressure drop corresponding to angle θ are given by:

$$V_n = \frac{1}{2}(1-\cos\theta)$$

$$\Delta P_n = \frac{1}{4}\sin^2\theta$$

The angular limits for the calculation of pressure and volume during the suction and discharge processes remain the same as assumed for the reciprocating arrangement.

Reverse Reciprocating Arrangement:

Figure 1c shows this arrangement. The essential difference between this and the regular reciprocating arrangement is in the location of the cylinder volume relative to the motion of the piston. Whereas in the regular recip, the cylinder volume is away from the center of rotation; in the reverse recip it is towards the center. The effect of this change can be seen in the volume/angle plots (Figures 2a and b). It is obvious from these plots that the crank angle to reach a given cylinder volume is different for each design. Consequently, the time available for suction and discharge processes is also different for each design. This translates into different instantaneous flow velocities of gas through suction or discharge ports for each design, and hence the difference in flow losses. The normalized volume and pressure drop equations for the case of reverse recip are as follows:

$$V_n = \frac{1}{2}\left[1-\cos\theta-\frac{\ell}{\gamma}+\frac{\ell}{\gamma}\sqrt{1-\frac{\gamma^2}{\ell^2}\sin^2\theta}\right]$$

$$\Delta P_n = \frac{1}{4}\left[\sin\theta-\frac{\gamma}{\ell}\frac{\sin\theta\cos\theta}{\sqrt{1-\frac{\gamma^2}{\ell^2}\sin^2\theta}}\right]^2$$

The angular limits for the calculation of pressure and volume during the suction and discharge processes remain the same as for the regular recip.

Rolling Piston Rotary:

Figure 1d shows the arrangement of this type of compressor. It is important to realize in this case that suction and compression processes take place simultaneously during one full rotation of the shaft. Thus, it takes 720 degrees of shaft rotation to complete one full cycle of suction and compression. Computation of volume and pressure drops for both suction and discharge processes, however, must be done simultaneously and from 0 to 360 degrees of shaft rotation. Assuming that the effect of vane thickness is negligible, the suction chamber volume equation is (9):

$$V = \frac{1}{2}hR_c^2\left[(1-a^2)\theta-\frac{1}{2}(1-a)^2\sin 2\theta - (1-a)\sin\theta\sqrt{a^2-(1-a)^2\sin^2\theta} - a^2\sin^{-1}\left\{\frac{(1-a)}{a}\sin\theta\right\}\right]$$

From this expression for volume, following normalized equations for volume and pressure drop during suction process can be easily developed:

$$V_n = \frac{\theta}{2\pi} - \frac{(1-a)\sin\theta\cos\theta}{2\pi(1+a)} - \frac{\sin\theta\sqrt{a^2-(1-a)^2\sin^2\theta}}{2\pi(1+a)} - \frac{a^2\sin^{-1}\left\{\left(\frac{1-a}{a}\right)\sin\theta\right\}}{2\pi(1-a^2)}$$

$$\Delta P_n = \frac{1}{\{\pi(1-a^2)\}^2}\left[a^2-a-(1-a)^2\sin^2\theta + (1-a)\cos\theta\sqrt{a^2-(1-a)^2\sin^2\theta}\right]^2$$

During the compression/discharge process, the expression for the pressure drop will remain the same as for suction, although the volume expression will change as follows:

$$V_n = 1-\frac{\theta}{2\pi}+\frac{(1-a)\sin\theta\cos\theta}{2\pi(1+a)} + \frac{\sin\theta\sqrt{a^2-(1-a)^2\sin^2\theta}}{2\pi(1+a)} + a^2\sin^{-1}\left\{\left(\frac{1-a}{a}\right)\sin\theta\right\}/2\pi(1-a^2)$$

The computations must be done simultaneously for both suction and discharge cycles and should continue from 0 to 360 degrees of shaft rotation.

RESULTS AND DISCUSSION

The normalized volume and pressure drop for the above mentioned four example cases were computed using the following assumed values of the constants:

Suction process volume ratio, $R_s = 0.2$.

Discharge process volume ratio, $R_d = 0.3$.

1/r ratio for recip and reverse recip = 3.5.

Roller/cylinder diameter ratio for rotary compressor, a = 0.85.

The results of pressure and volume computations have been shown in three sets of plots: Figure 2a and 2b show the normalized volume (or percentage displacement) vs. angle plots during reexpansion/suction, and compression/ discharge processes respectively for the four example cases. Figures 3a and 3b show the pressure drop vs. angle plots, and 4a and 4b show the pressure drop vs. volume plots.

It will be noticed that the time (or angle) taken during suction or discharge process in the rotary compressor is much more than the other three cases, thereby resulting in lower pressure drops and consequently very low flow losses. On the other hand, the three reciprocating arrangements are generally close to each other with some small variations, and they all result in higher pressure drops and consequently larger flow losses compared with rotary compressor.

It will also be noticed that during the first 90 degrees of travel the regular recip covers more displacement than the reverse recip or the scotch yoke. A closer look at Figures 2 and 3 will show that the rate of increase of volume (and therefore the pressure drop) is increasing for the regular recip from 0 to 75 degrees, 75 degrees being the point of inflection where the rate of increase of volume is highest. This is also the point of highest pressure drop. The corresponding point during the return compression stroke would fall at 285 degrees. For the reverse recip, the corresponding points are at 105 and 255 degrees. For scotch yoke, these points fall at 90 and 270 degrees, as would be expected. Thus, while the regular recip has higher pressure drops during the early part of the suction process, the reverse recip has higher pressure drops during the later part. This is so because the suction process starts way before the inflection point for both of these cases. The overall effect of this is a higher value of suction flow loss for the reverse recip than the regular recip as well as the scotch yoke as figured out by computing the area under the pressure/volume diagram.

On the other hand the discharge process starts during the later part of the return stroke in either case. The pressure drop is, therefore, higher for regular recip than the reverse recip (or the scotch yoke for that matter) during all of the discharge process. The actual values of the non-dimensional flow losses for each case have been computed by integrating the area under the pressure-volume plots. These values, however, should be multiplied by the appropriate compressor characteristics to find actual energy loss. Such computations were done to get a "feel" for these losses using following assumed compressor characteristics:

<u>Example</u>

$P_s = 1.6$ lbm/ft^3

$P_d = 3.7$ lbm/ft^3

$N = 3500$ rpm

$V_s = 0.001$ ft^3

A_e for suction side = 0.003 ft^2

A_c for discharge side = 0.001 ft^2

Above data is a rough assumption of values for a typical 1 1/4 ton compressor running at ARI conditions to which the assumed values of volume ratio R_s and R_d roughly correspond. The actual power lost in flow losses can be calculated by multiplying the non-dimensional P-V work by the factor $\frac{1}{2}\frac{\rho}{g}\left(\frac{\dot{\theta}}{A_e}\right)^2 V_s^3$ and then converting it into proper units. Thus, following power loss values were calculated for the four example cases:

<u>Reciprocating Compressor:</u>

Suction flow loss	= 4.5	watts
Discharge flow loss	= 26.3	"
Total	= 30.8	"

<u>Scotch Yoke Compressor:</u>

Suction flow loss	= 4.4	watts
Discharge flow loss	= 21.8	"
Total	26.2	"

Reverse Recip:

Suction flow loss	= 4.7	watts
Discharge flow loss	= 19.1	"
Total	= 23.8	"

Rolling Piston Rotary:

Suction flow loss	= 3.0	watts
Discharge flow loss	= 7.2	"
Total	= 10.2	"

CONCLUDING REMARKS

It is interesting to note from the example shown above that although suction flow losses in the three reciprocating type arrangements are very close to each other, the discharge flow losses are significantly different. This is due to (1) higher flow velocity during the discharge process; (2) higher mass density during the discharge process; (3) and more importantly, the fact that throughout the discharge process the pressure drop trend does not change, being located on the same side of the pressure peak all the time. On an overall basis, it is obvious that between the three reciprocating arrangements, the regular recip has the highest loss and the reverse recip the lowest. The rolling piston rotary, of course, has the least loss of all the four. The significance of this interesting difference may not seem important when the actual numbers are compared with the total power input (for instance, in this example the total power input would be about 1500 watts). However, the analysis certainly brings out an interesting perspective for comparing various kinematically different compressor arrangements.

ACKNOWLEDGEMENTS

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NOMENCLATURE

- a Roller to cylinder radius ratio for rotary compressor (R_r/R_c)
- A_e Effective flow area (ft^2)
- A_p Piston area in reciprocating type compressors (ft^2)
- g Gravitational constant (32.17 ft-lbm/lb_f sec^2)
- h Cylinder height in rotary compressor (ft)
- l Connecting rod length (ft)
- N rpm of the motor
- Q Volume velocity (ft^3/sec)
- R_c Cylinder radius for the rotary (ft)
- R_r Roller radius for the rotary (ft)
- R_s Suction process volume ratio
- R_d Discharge process volume ratio
- r Crank radius (ft)
- V Cylinder volume at angle θ (ft^3)
- V_s Total swept volume (ft^3/cycle)
- ΔP Pressure drop (lb_f/ft^2)
- ΔP_n Normalized pressure drop
- ρ Mass density (lbm/ft^3)
- ρ_s Suction gas mass density (lbm/ft^3)
- ρ_d Discharge gas mass density (lbm/ft^3)
- θ Crank angle (radians)
- $\dot{\theta}$ Angular speed of shaft (radians/sec.)

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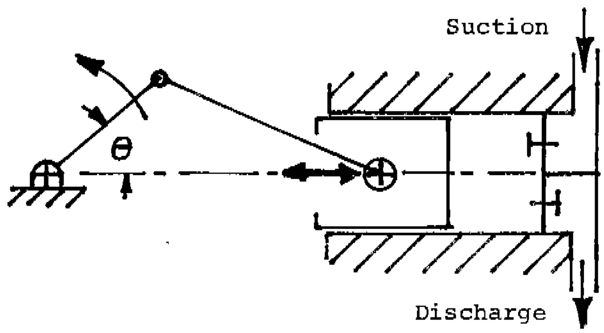

Figure 1a: Reciprocating Compressor

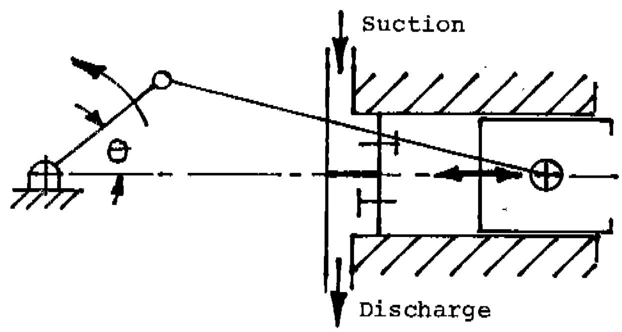

Figure 1c: Reverse Recip.

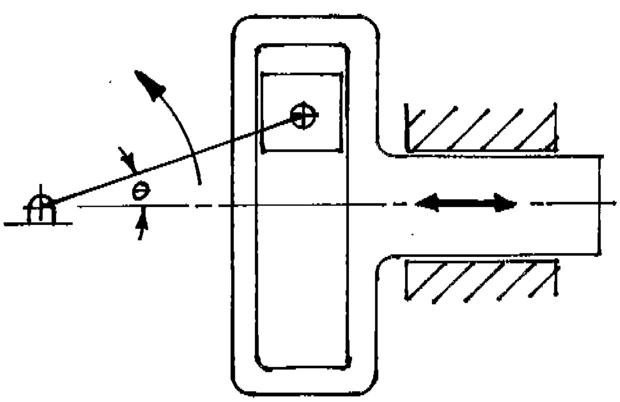

Figure 1b: Scotch Yoke

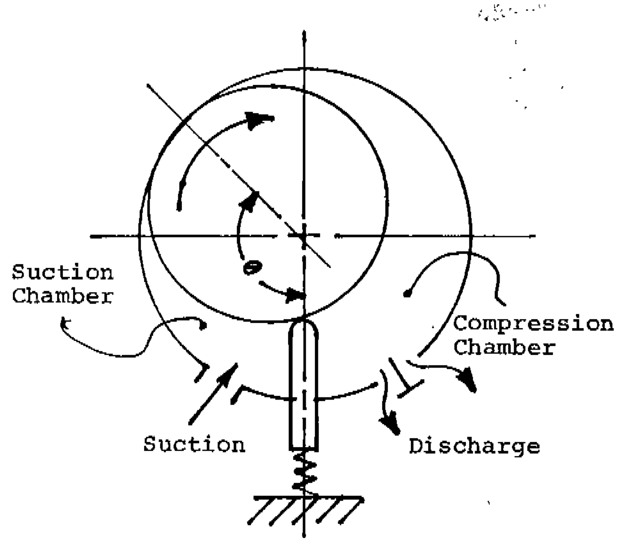

Figure 1d: Rolling Piston Rotary

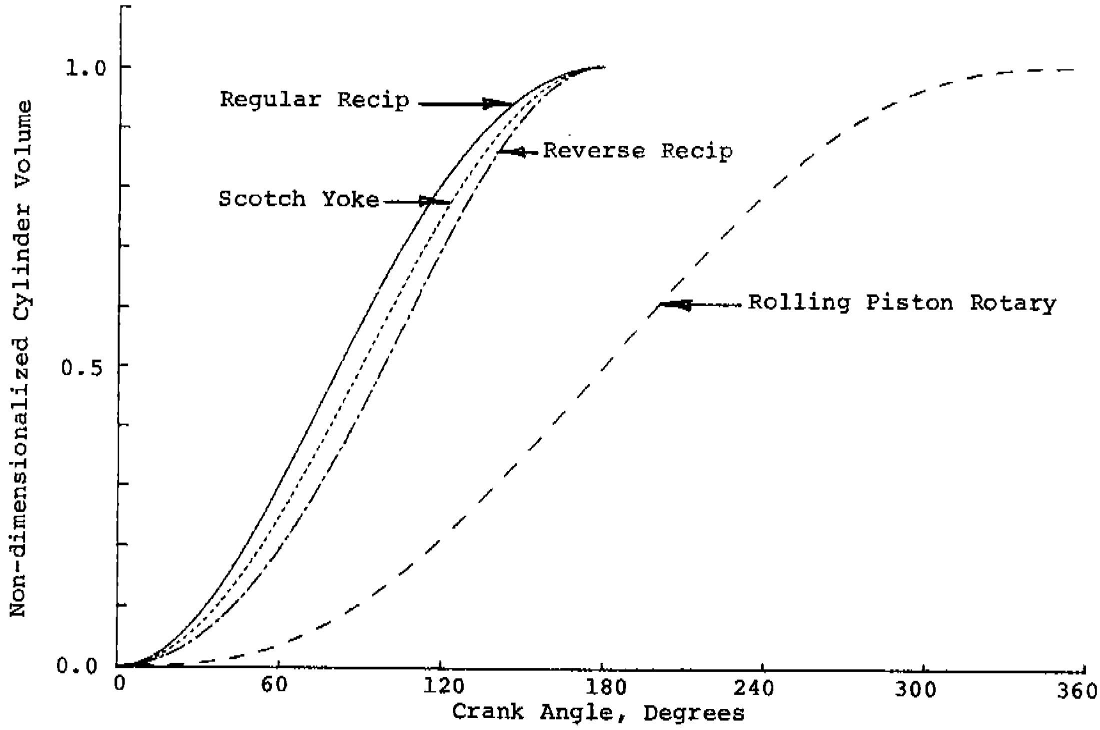

Figure 2a: Volume vs. Crank Angle Plots for Suction Process

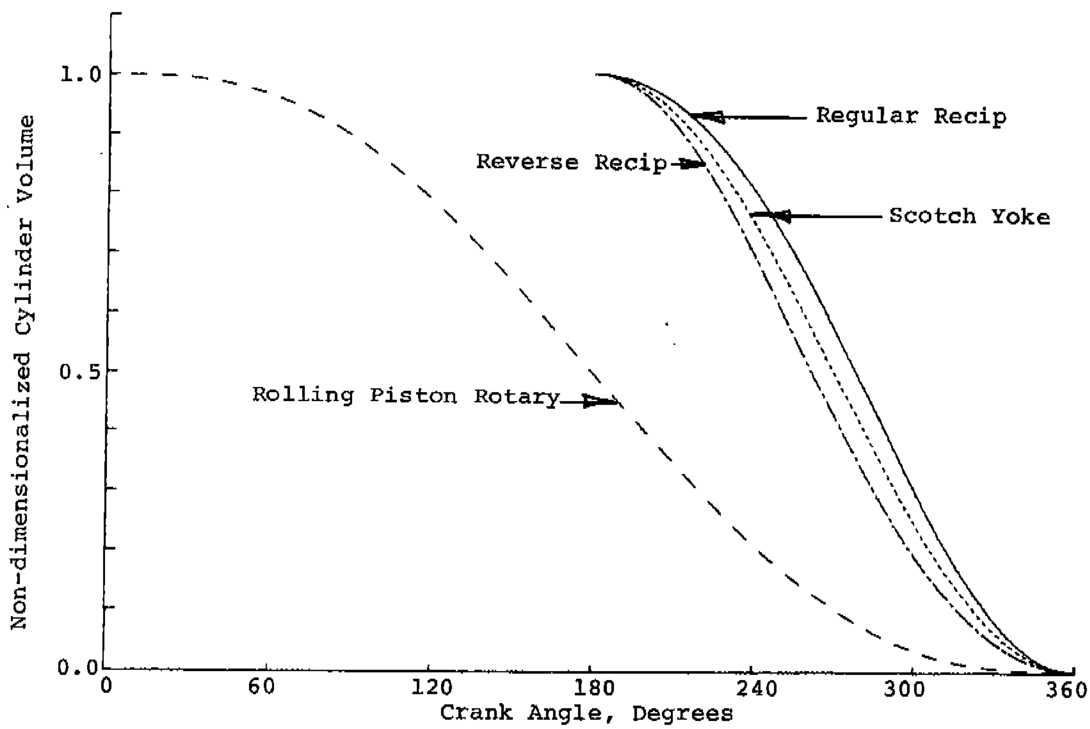

Figure 2b: Volume vs. Crank Angle Plots for Discharge Process

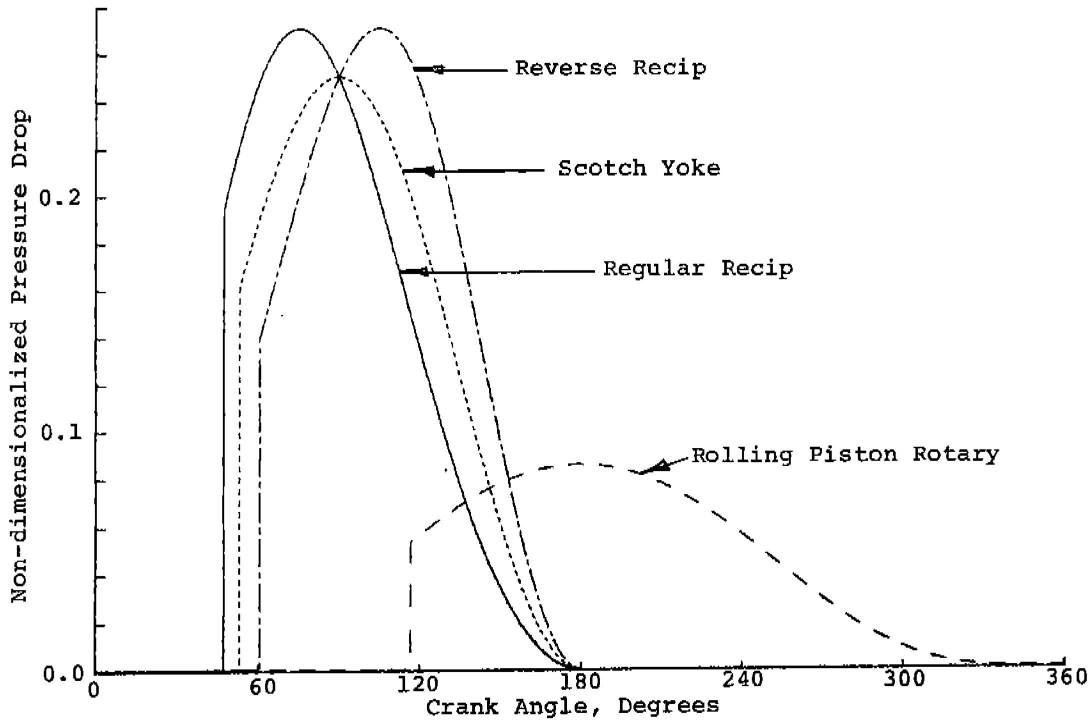

Figure 3a: Pressure Drop vs. Crank Angle Plots for Suction Process

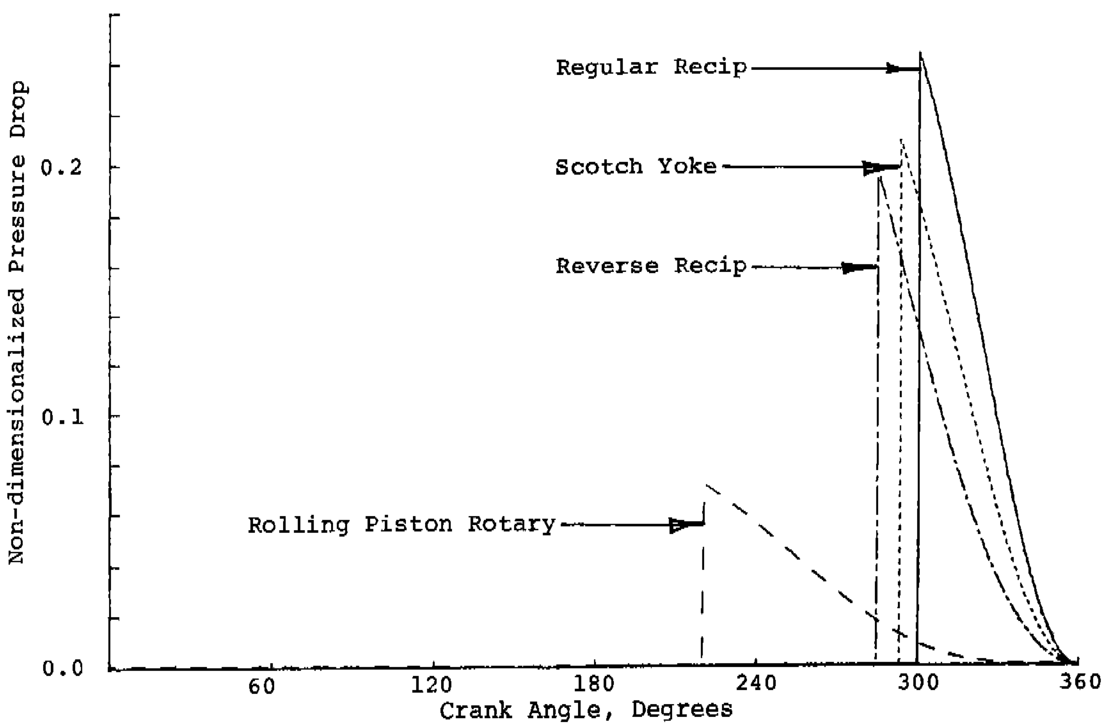

Figure 3b: Pressure Drop vs. Crank Angle Plots for Discharge Process

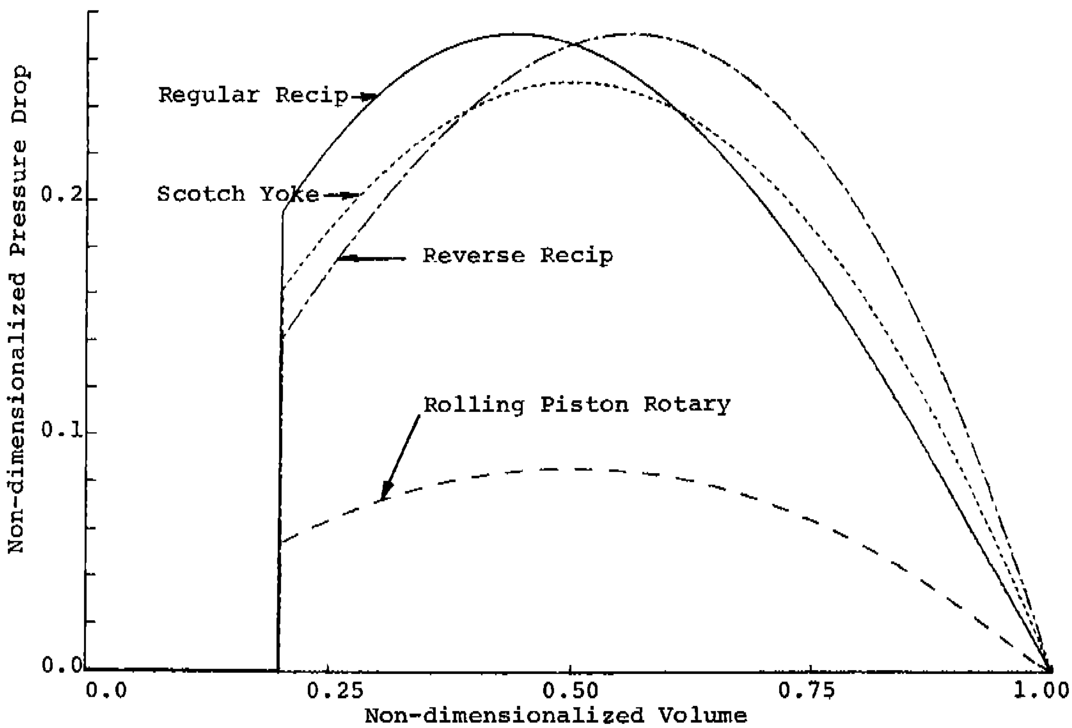

Figure 4a: Pressure Drop vs. Volume Plots for Suction Process

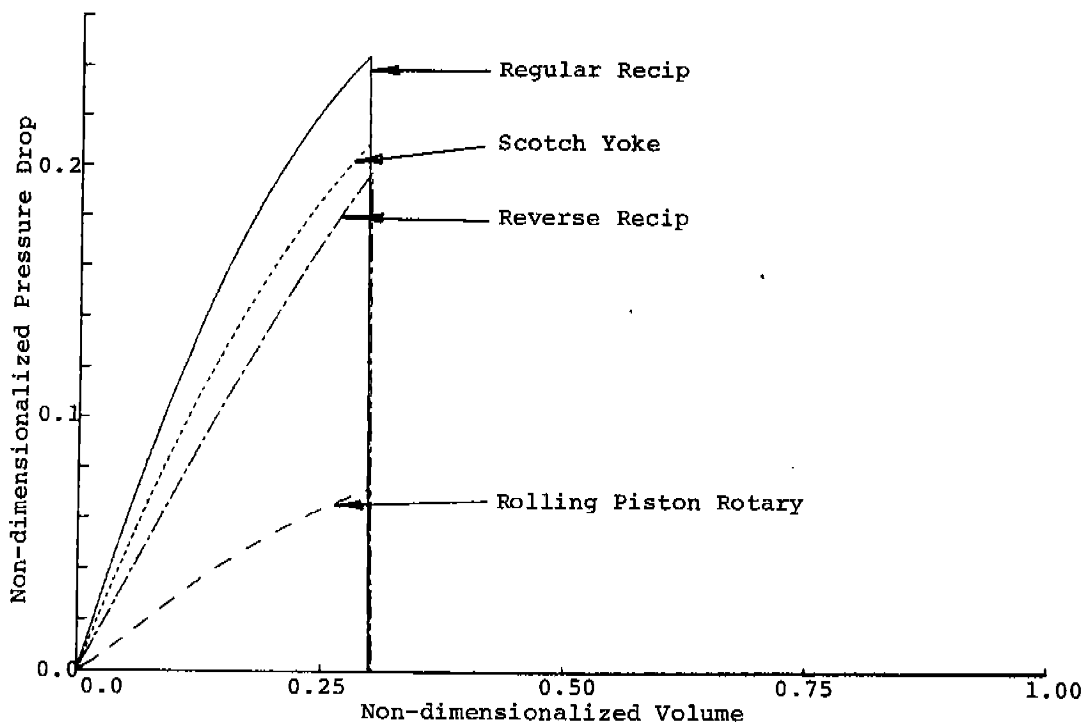

Figure 4b: Pressure Drop vs. Volume Plots for Discharge Process