1988

# The Analysis of Dimensional Communization Procedures Considering the Efficiency for the Rolling Piston Compressors

Caio Franco da Costa
*EMBRACO - Empress*

# THE ANALYSIS OF DIMENSIONAL COMMONIZATION PROCEDURES CONSIDERING THE EFFICIENCY FOR THE ROLLING PISTON COMPRESSORS

Caio Franco da Costa
EMBRACO - Empresa Brasileira de Compressores S/A
P.O. Box D-27 - 89200 Joinville - SC - Brazil

## ABSTRACT

The present work deals with dimensional commonization procedures and their different effects over the performance of the compressor. The study is made theoretically with the use of a previously validated simulation program at a refrigeration compressor series with capacity range from 650 to 1000 Btu/hr.

Efficiency is analysed through graphs showing the variation of the energy efficiency ratio (EER), capacity and power consumption over the swept volume domain, for each of the commonization alternatives. The results of the energetic and the volumetric efficiency over the swept volume domain are also shown.

Furthermore, in order to get a detailed understanding of the said processes, each specific energy and mass loss causing the capacity and consumption variations are also disclosed in graphs.

## INTRODUCTION

Dimensional commonization procedures are meant primarily as a way to attain process simplification and consequently cost reduction. However such procedures must also be considered in terms of the resulting performance and efficiency of the compressor.

The displacement of a rolling piston compressor is given basically by its three main dimensions: cylinder diameter, cylinder height and roller diameter. Therefore there are three possible commonization procedures for the parts in the same compressor series; obtained by changing singly one dimension keeping the two others constant.

The present work deals, with the commonization procedures and their different effects over the efficiency of the compressor.

## COMMONIZATION PROCEDURES

As mentioned above there are three direct dimensional commonization procedures for any group of pumping kits of rolling piston compressors of the same series. Also a fourth procedure, a derivation, will be examined (see table 1).

The first, and most elementary procedure [≠ H], is to change the height of the cylinder (H) keeping every other dimension constant. The second type of procedure [≠ Ør, = Øe] is to change the roller diameter (Ør), and consequently the eccentricity of the shaft (Ec), keeping constant the cylinder height, cylinder diameter (Øc) and, particulary, the eccentric journal diameter (Øe), hence, shifting also the roller thickness (tr). The third procedure [≠ Ør, ≠ Øe], a variation of the second one, is achieved changing the roller diameter and the shaft eccentricity yet altering also the eccentric journal diameter, so that the roller thickness is kept constant for all kits in the group. The fourth possible commonization [≠ Øc], is to change the cylinder diameter and the eccentricity, keeping constant every other dimension.

The consequences of each of these commonization procedures in the manufacturing process can be easily deduced by those skilled in the art by examining table 2, where the commonization of a group of "n" compressor kits is shwon. As the manufacturing process analysis would be out of the scope of this work, we will not make any further consideration on this matter.

$P_d$ : Compressor discharge pressure
$P_H$ : High pressure
$P_m$ : Pressure of pressure control chamber
$P_s$ : Suction pressure of Compressor
$Q_e$ : Refrigerating capacity
$Q_1$ : In-flow rate to pressure control chamber
$Q_2$ : Out-flow rate from pressure control chamber
$R_3$ : Flow resistance of slider side gap
$T_o$ : Reponse time coefficient of Refrigerating cycle
$T_r$ : Compressor required torque
$V$ : Volume of cylinder chamber
$V_{th}$ : Maximum theoretical volume of cylinder chamber
$X_1$ : Lift of steel ball
$X_3$ : Displacement of Slider
$\varepsilon$ : Compression ratio
$\gamma_H, \gamma_m$ : Gas specific weight
$\mu$ : Friction coefficient
$\kappa$ : Specific heat ratio
$\eta_v$ : Volumetric efficiency of Compressor
$\eta_{v0}$ : Volumetric efficeincy of Compressor at control start

## REFERENCE


1. J. Reichelt, et al., "REFRIGERATING COMPRESSORS FOR AIR CONDITIONING OF PASSENGER CARS." Purdue Compressor Technology Conference, July 1986, pp 537-549
2. T. Maruyama, et al., "CAPACITY CONTROL OF ROTARY TYPE COMPRESSORS FOR AUTOMOTIVE AIR-CONDITIONERS" Purdue Compressor Technology Conference, July 1982, pp 284-291
3. Timothy J. Skinner, et al., "V-5 AUTOMOTIVE VARIABLE DISPLACEMENT AIR CONDITIONING COMPRESSOR" SAE 850040, Feb. 1985
4. Yoji Nishimura, et al., "DEVELOPMENT OF TWO-STAGE VARIABLE DISPLACEMENT COMPRESSOR FOR AUTOMOTIVE AIR CONDITIONER" SAE850039, Feb. 1985
5. H. Holdack-Janssen, et al., "CONTINUOUS AND DISCONTINUOUS CAPACITY CONTROL FOR HIGH SPEED REFRIGERATION COMPRESSORS" Purdue Compressor Technology Conference, July 1986 pp 67-75

Moreover, the refrigerating capacity becomes fixed after the start of capacity control. As for these results, the torque Tr started a drop at N $\geq$ 1800 rpm, and at N = 5000 rpm, in constant to the value of N = 1800 rpm, about a 35% reduction was reached.

Consequently, with a system employing a capacity control compressor there is no attachment and detachment of the electromagnetic cluth in the range of capacity control, so a stable value with no fluctuation is maintained, thus yielding comfortable temperature regulation and driving feeling (See Fig. 12)

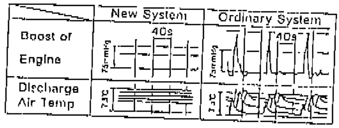


Fig. 12 Comparison of fluctuations

## CONCLUSIONS

In the development a vane rotary type capacity control compressor equipped with a capacity control mechanism which engaged the method of automatically delaying the ending position of suction process, we studied dynamic behavior of capacity control mechanism and came to the following conclusions.

1. The control mechanism using a arc-shaped slider is simple, with few additional parts and enables capacity control over the wide range from 10 to 100%.
2. At the refrigerating cycle employed the capacity control compressor, with pressure control valve which stabilize suction pressure constant, refrigerating capacity becomes constant after the start of capacity control, and reduction of compressor torque is reached. Consequently yielding comfortable temperature regulation and driving feeling with no fluctuation in the range of capacity control.
3. Through analysis of capacity control compressor, results of theoretical analysis tend to agree with experimantal one and the methods we employed can establish outstanding control responce and stability characteristics.

## NOMENCLATURE

$A_1$ : Pressure reception area of Diaphragm
$A_2$ : Pressure reception area of Steal ball
$A_3$ : Pressure reception area of Slider
$C$ : Flow rate coefficient
$d_2$ : Diameter of outlet hole from pressure control valve
$d_3$ : Diameter of outlet hole from pressure control chamber
$Fi$ : Sum of initial spring strengths of spring (1) and (2)
$F_3i$ : Initial spring strength of spring (3)
$F_3$ : Side pressure applied to slider
$g$ : Gravitational acceleration
$Ko$ : Gain constant of Refrigerating cycle
$k_1$ : Spring constant of spring (1)
$k_2$ : Spring constant of spring (2)
$k_3$ : Spring constant of spring (3)
$M_3$ : Mass of Slider
$N$ : Compressor rotation speed
$No$ : Compressor rotation speed at control start
$Pa$ : Atmospheric pressure
$Pc$ : Critical pressure ratio

Fig. 9 shows variations due to the degree of openness of return ports for cylinder internal pressure of the capacity control compressor. In the diagram, there is a comparison in the return port full closed state with the PV diagram, and as the return ports gradually open, the substantial compression start point gradually moves, as can be seen by the thinning of the PV diagram.

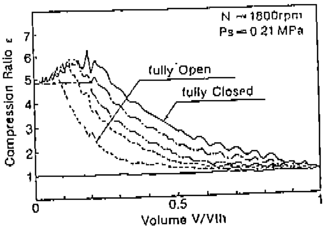


Fig. 9 Variations of PV diagram

Fig. 10 shows calorimeter characteristics using an suction pressure stabilized control valve. In this experiment, at 1000 rpm suction pressure was set to 0.16MPa with an expansion valve, and compressor speed was gradually increased in that state. Discharge pressure Pd was held constantly fixed at 1 MPa. The result was that suction pressure Ps became constant at 0.13MPa with N greater than or equal to 1200rpm at a volumetric efficiency $\eta_v$ given as follows:

$$\eta_v = \eta_{v0} \times N_0/N \quad (Qe \propto \eta_v \cdot N = \text{const.})$$

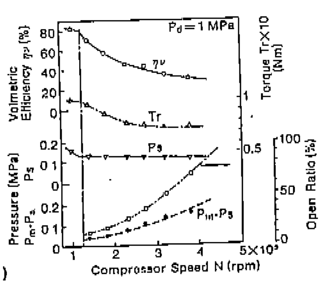


Fig. 10 Calorimeter characteristics

Accompanying this was a similar trend of reduction in torque Tr. At this time, the return port degree of openness was different from the volumetric efficiency $\eta_v$ reduction curve to raise speed, and clearly increased in direct proportion to the rotation speed.

## PERFORMANCE OF REFRIGERATING CYCLE

Finally we introduce in Fig. 11 an example of refrigerating cycle characteristics using the aforementioned capacity control compressor. Accompanying increases in compressor speed, suction pressure Ps gradually decreased, and at N = 1800 rpm the setting suction pressure Ps = 0.15MPa was reached. And, N > 1800 rpm suction pressure is fixed to constant by capacity control. In this case, discharge pressure Pa is shown by the rising curve up to 1800 rpm, but subsequently the circulation rate of refrigerant becomes constant, so the curve changes to dropping.

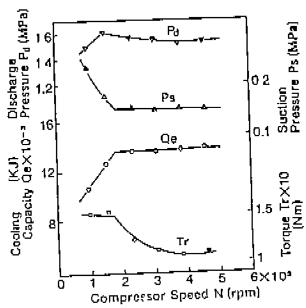


Fig. 11 Performance of Refrigerating cycle