2008

Reciprocating Compressor Program

Christophe Ancel
Danfoss CC

Pierre Ginies
Danfoss CC

RECIPROCATING COMPRESSOR PROGRAM

Christophe ANCEL[1], Pierre GINIES[2]

[1]DANFOSS CC, Technology Department
Trévoux, France
(Phone 04 74 00 96 14, Fax 04 74 00 95 97, c.ancel@danfoss.com)

[2]DANFOSS CC, Technology Department
Trévoux, France
(Phone 04 74 00 95 57, Fax 04 74 00 95 97, p.ginies@danfoss.com)

ABSTRACT

To develop new reciprocating compressors for air conditioning or refrigeration a new Fortran program was built in order to compare different technical solutions. This program is adapted to any refrigerant cycle.
The data include the refrigerant characteristics, the oil viscosity, the time step, the bore and stroke ratio, the dead volume, the number of sealing rings, the friction coefficients, the rotation speed, the parts inertia, the spring and valve characteristics, the shaft and bearing dimensions.
The results cover the volumetric efficiency, the isentropic efficiency, the bearing loads, the friction losses and the oil film thickness for the bearings.
The compressor is considered in terms of sub assemblies with a hierarchy. The first basic elements are the different sealing rings which are fully described. Each cylinder is associated with a set of sealing rings, an axis, a connecting rod, whose mechanical characteristics are stored. The program processes each cylinder separately and takes into account the positions of the cylinders on the crankshaft and the eccentric orientation. Finally, the loads on the main bearings are determined knowing the position of the main bearings of the shaft.
The valve stiffness and the thermal exchanges with the cylinder can be added to the parameters. The computation is possible for any number of cylinders and any number of stages of compression, provided that the pistons are connected to the same shaft.
The program was based on a perfect gas model to obtain the efforts on the piston, from which the loads on all bearings were derived.
The Fortran language has good portability and allows a subroutine architecture, which is useful to modify the program or to add new features. The data can be written into an Excel file and the results can be loaded into another Excel file. This program was used to select design criteria and to make decisions. The product development time was reduced and many lab tests were avoided.

1. INTRODUCTION

The program of Corberan, Gonzalvez, Urchueguia, Calas (2000) was tested and gave good results. The output of this program is geared on the gas characteristics in each component of the compressor, this is why another program was written in order to complete the information in terms of forces and inertia.

The basic assumptions for this new program are :
- perfect gas model
- constant motor speed
- no unbalance
- compression with heat exchange

The following components are modelized :
- multi stage compression
- crankshaft, motor and journal bearings
- pistons, axis and connecting rods
- valves and sealing rings

The data are :
- fluid characteristics
- operating point
- geometrical data and mass properties of the components

The results include :
- the forces on piston
- the forces on crankshaft
- the bearings operating conditions (specific pressure, oil film thickness, energy dissipation)
- the internal leaks values
- the piston and sealing ring friction dissipation
- the volumetric efficiency
- the torque on crankshaft
- the isentropic efficiency
- the compressor absorbed power

The program was written to process the above data in order to obtain the above results. The architecture and the language of the program must be compatible with the needs in terms of evolution, this is why the Fortran language was chosen. The program can be divided into major functions, which regard several fields : mechanics, thermodynamics, fluids mechanics. Thus the first step was to write a physical model for each function of the program. The second step was to define the strategy to solve the coupling of the equations in relation to the time step. The third step was to define a validation method of the program.

The detailed data and results are shown hereafter.

2. DATA

The compressor can have several stages, several cylinders and each cylinder can have several rings. Hence, the lower sub-assembly is defined as a piston with a set of rings, the medium sub-assembly is a set of cylinders belonging to the same stage and the highest sub-assembly is a set of stages on a shaft which defines a compressor. To each type of assembly corresponds a table like the one hereafter. The data can be written in an Excel file, which can be converted into a text file.

MT44_A.txt 09-03-2007

BDF	FLUID	ICMOT	RPM	NETAG	NLIET	NP	K	KIS	CV	R
0	410	0	3500	1	0	2	1,34	0,58	0	94,5

VISC	PIN	TIN	CRITCV	NPT_TR	NPT_MAX	NPT_L	NPT_SAUT
8	9,95	25	100	360	720	5	50

NPALI	LM	DM	CRM	ZM	LOP	DOP	CROP	ZOP
0	51	25,465	0,0155	140	22	25,465	0,015	0

ILETAM(IETP	PE(IETP)	TE(IETP)	LETAV(IETP)	PC(IETP)	TC(IETP)
0	9,95	25	0	33,84	104

CYLINDER 1

IETP(IP)	TYPP(IP)	NVOLX(IP)	NSEG(IP)	PHI(IP)
1	-1	2	1	0

DIAMP(IP)	DIAMC(IP)	VMORT(IP)	LB(IP)	EXB(IP)
47,543	47,625	0,0000010000	89,64	10,695

CLPAM(IP)	SAM(IP)	KAM(IP)	CLPAV(IP)	SAV(IP)	KAV(IP)
0	0,00028903	0,06	0	0,00022227	0,07

QTHERM(IP)	XAVT(IP)	LPIST(IP)	FMINP(IP)	FMAXP(IP)
0	14,2	28	0,03	0,03

MPIST(IP)	ZPIST(IP)	ACOP(IP)	DVILCH(IP)
0,092	55	0	0

Table 1 : set of data for a compressor (highest sub-assembly)

3. RESULTS

3.1 Results for the whole compressor

The forces on the crankshaft bearings are summarized. The associated oil film thickness and dissipation are released as instantaneous and average values. The sum of the piston torques and of the friction torque created by the crankshaft bearings leads to the mechanical torque. The power absorbed by the motor corresponds to this mechanical torque.

P1 V1.6 MT44_A.tGEN.res		PROCESSED ON 17-03-2008 AT 17:01									
degr	s	J	N	N	N	MPa	µm	W	W	N	N
TETAT(1,IT)	TPS(IT)	CVMX	FMX(IT)	FMY(IT)	FM(IT)	PPJBM(IT)	HJBM(IT)	LJBM(IT)	LJBMM(IT)	FOPX(IT)	FOPY(IT)
0.	0.000000	66.6477	0.	0.	0.	0.000	0.000	0.000	0.000	0.	0.
5.	0.000238	66.6477	133.	15.	133.	0.103	13.950	16.772	0.000	183.	12.
10.	0.000476	66.6477	132.	27.	134.	0.103	13.950	16.772	0.000	181.	22.
15.	0.000714	66.6477	130.	38.	135.	0.104	13.950	16.772	0.000	178.	31.
20.	0.000952	66.6477	127.	48.	135.	0.104	13.950	16.772	0.000	173.	40.
25.	0.001190	66.6477	122.	58.	135.	0.104	13.950	16.772	0.000	166.	48.
30.	0.001429	66.6477	117.	68.	135.	0.104	13.950	16.772	0.000	158.	57.
35.	0.001667	66.6477	110.	76.	134.	0.103	13.950	16.772	0.000	149.	64.
40.	0.001905	66.6477	103.	85.	133.	0.103	13.950	16.772	0.000	139.	71.
45.	0.002143	66.6477	94.	92.	132.	0.102	13.950	16.772	0.000	127.	78.
50.	0.002381	66.6477	85.	99.	131.	0.101	13.950	16.772	0.000	114.	85.
55.	0.002619	66.6477	75.	105.	130.	0.100	13.950	16.772	0.000	101.	90.
60.	0.002857	66.6477	65.	111.	129.	0.099	13.950	16.772	0.000	87.	96.
65.	0.003095	66.6477	54.	116.	128.	0.099	13.950	16.772	0.000	72.	100.
70.	0.003333	66.6477	43.	120.	128.	0.098	13.950	16.772	0.000	57.	105.
75.	0.003571	66.6477	32.	123.	127.	0.098	13.950	16.772	0.000	42.	108.
80.	0.003810	66.6477	20.	126.	128.	0.098	13.950	16.772	0.000	27.	111.
85.	0.004048	66.6477	9.	128.	128.	0.099	13.950	16.772	0.000	12.	113.
90.	0.004286	66.6477	-2.	129.	129.	0.099	13.950	16.772	0.000	-3.	115.

Table 2 : set of results for a compressor (highest sub-assembly)

3.2 Results for each piston

The result table gives the figures in relation to the time and angular steps. The pressure, volume, temperature and mass of gas inside the cylinder are given. Then, the forces on the piston are detailed. The forces on the sleeve and the inertia terms appear. Thus all forces on the connecting rod and on the crankshaft are summarized. Finally, the associated bearing loads, oil film thickness and dissipation are released as instantaneous and average values. The volumetric efficiency of each piston is computed as the ratio of the computed mass flow divided by the theoretical mass flow following the cylinder swept volume. The isentropic efficiency is calculated as the ratio of the ideal compression work divided by the real work based on the crankshaft torque generated by each piston.

3.3 Results for each ring

The upstream and downstream pressures are given. Then the mass flow around the ring is detailed with instantaneous and average values.

To process the above data in order to the above results, the program was designed with the following principles.

4. PROGRAM ARCHITECTURE

4.1 Compressor architecture

The architecture of the program is directly connected to the architecture of the compressor.

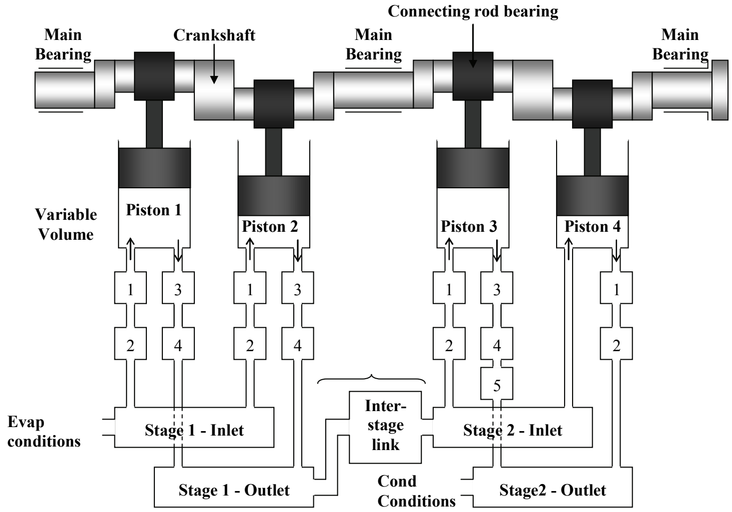

Figure 1 : compressor architecture

4.2 Cylinder

The number of cylinders can be chosen for each stage of compression. The cylinder can include a dead volume. The cylinder wall can experience some heat transfer which can be modelized with a table of the heat exchanged versus the angle of crankshaft over one turn. This table is of the same type as the general Excel data table above and can be prepared using formulas dealing with heat transfer in cylinders.

The valve dynamics is not studied, but the spring loading is taken into account as the minimum pressure difference to open or to close valves. Once a valve is opened, it is assumed to release the flow with a full area.

4.3 Piston

The piston is described with its axis, inertia and sealing rings. There is a special possibility to offset the cylinder axis and the shaft axis with a DVILCH value. This reduces the friction force on the cylinder sleeve during the compression phase, because it is possible to reduce the inclination between the main axis of the connecting rod and the axis of the sleeve.

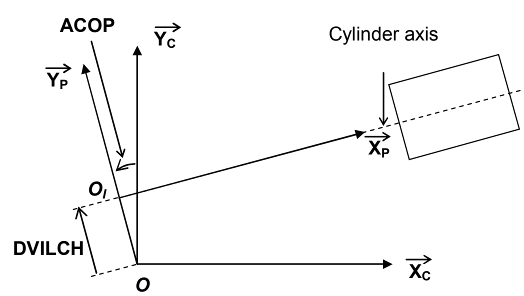

Figure 2 : local coordinate system for piston

4.4 Sealing rings

The number of sealing rings can be chosen for each piston. The loading of the rings consists of a constant elastic loading and of a gas loading due to the pressure difference on the ring, which contributes to mate the ring onto the cylinder wall. As a consequence the cylinder sleeve withstands friction forces.

4.5 Connecting rod

The connecting rod is described as a mechanical part with a set of dimensions. The program computes the inertia characteristics of the connecting rod in order to use them in the equations.

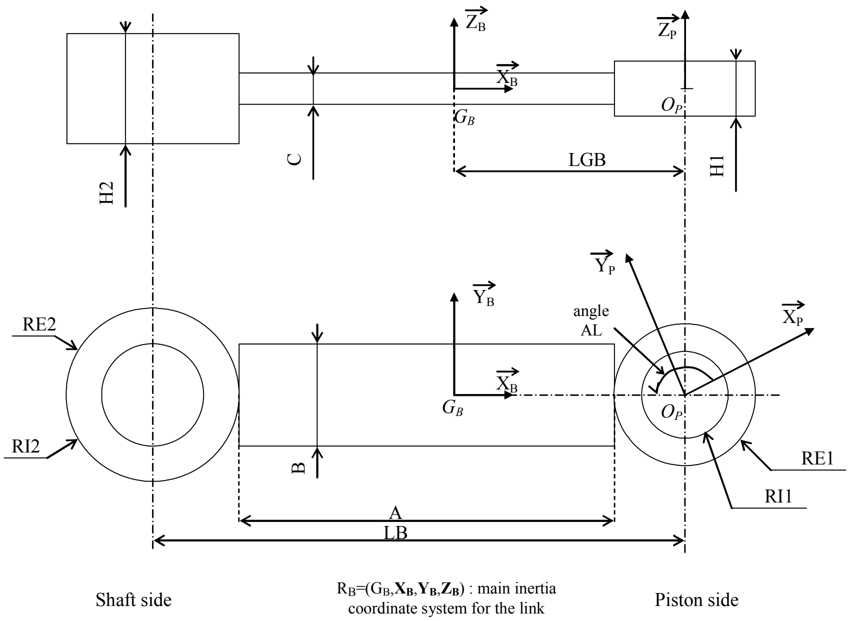

Figure 3 : connecting rod description

Once the architecture of the program is set, then the solving procedure is to be found.

5. SOLVING

5.1 Strategy

As the time is the leading parameter of the study, it is necessary to define a kinematic law in relation to the shaft rotation. We assumed a constant motor speed and an average efficiency. A time step is chosen in the program data. This defines the angular step. The phase shift must be taken into account for each piston.

The features required to represent a compressor includes many phenomena. Among these are a number of central phenomena like the suction and the compression. Mechanical parts are in equilibrium with the gas pressure and this generates forces on the piston. The forces on the piston must be compensated by forces on the driving parts of the compressor in order to secure the transmission of the energy from the motor o the gas.

A general iterative process is defined for each angular step. The leading parameter is the volume V inside the cylinder. The conditions p, T in each cylinder are calculated in relation to V. There are mass flows between the cylinder and the low pressure side, as well as between the cylinder and the high pressure side. The pressure inside the cylinder is compared to the pressure upstream the suction valve and downstream the discharge valve, in order to set the valves status (open or closed). The mass flow through the valves is computed. This mass flow depends on the opening of valves and on the ratio between the upstream and downstream pressures. The leak flow through the sealing rings is calculated as well as the head losses through the valves. The new mass of gas inside the cylinder is obtained from these mass flows. The new p,T conditions are adjusted.

A heat transfer curve with the cylinder wall can be implemented and will modify the gas characteristics inside the cylinder. The curve can be computed with classical formulas as an assumed set of data and can be adjusted following a first run of the program.

5.2 Test and validation

The program was tested on the current MANEUROP compressor range and gave good results compared to lab measurements.

The details of the modelization appear hereafter.

6. MODELIZATION

6.1 Kinematics

Notation :

O : axis of the shaft
A : axis of the connecting rod-shaft bearing
B : axis of the piston bearing
OA = EXB : eccentricity
AB = LB : length of the connecting rod
ACOP : angle for the orientation of each cylinder axis
DE : angle piston - shaft eccentric pin
VIT : rotation speed of the shaft
AL : angle between the piston and the connecting rod
XP = $O_I B$: piston coordinate
$VP = \dot{X}P$: piston speed
$GP = \ddot{X}P$: piston acceleration
XA, YA : coordinates of A in the system (O_I, $\mathbf{X_P}$, $\mathbf{Y_P}$)

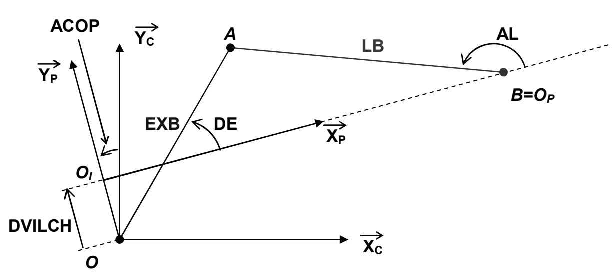

Figure 4 : description of the variables

$$\boxed{XA = EXB \cdot \cos DE} \tag{1}$$
$$\boxed{YA = EXB \cdot \sin DE - DVILCH} \tag{2}$$
$$\sin AL = \frac{YA}{LB} \tag{3}$$
$$\cos AL = -\sqrt{1-\left(\frac{YA}{LB}\right)^2} \tag{4}$$
$$\boxed{XP = XA - LB \cdot \cos AL} \tag{5}$$
$$\boxed{XPMAX = \sqrt{(EXB + LB)^2 - DVILCH^2}} \tag{6}$$

The variable extension 1P stands for the speed (first derivative).

$$\boxed{DE1P = VIT} \tag{7}$$
$$\boxed{XA1P = -EXB \cdot DE1P \cdot \sin DE} \tag{8}$$
$$\boxed{YA1P = EXB \cdot DE1P \cdot \cos DE} \tag{9}$$
$$\boxed{AL1P = \frac{YA1P}{LB \cdot \cos AL}} \tag{10}$$
$$\boxed{VP = XA1P + LB \cdot AL1P \cdot \sin AL} \tag{11}$$

The variable extension 2P stands for the acceleration (second derivative).

$$\boxed{XA2P = -EXB \cdot DE1P^2 \cdot \cos DE} \tag{12}$$
$$\boxed{YA2P = -EXB \cdot DE1P^2 \cdot \sin DE} \tag{13}$$
$$\boxed{AL2P = \frac{YA2P}{LB} + AL1P^2 \cdot \sin AL} \tag{14}$$
$$\boxed{GP = XA2P + LB \cdot \left(AL2P \cdot \sin AL + AL1P^2 \cdot \cos AL\right)} \tag{15}$$

6.2 Dynamics

Writing the equilibrium of forces for the connecting rod gives the forces applied on it using the dynamic momentum KOP.

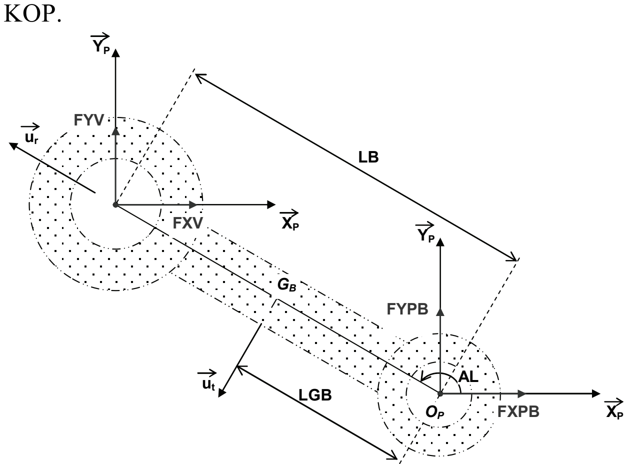

Figure 5: description of the forces

$$\boxed{FYB = \frac{\frac{KOP}{LB} + MB \cdot (GX \cdot \sin AL - GY \cdot \cos AL) - FXP \cdot \sin AL}{\cos AL}} \tag{16}$$

$$\boxed{FXB = -FXP} \tag{17}$$

Notation :
MB : mass of the connecting rod
CB : momentum of inertia of the connected rod computed at its center of gravity
LGB : length AB of the connecting rod

$$\boxed{KOP = MB \cdot LGB \cdot (AL2P \cdot LGB - GP \cdot \sin AL) + CB \cdot AL2P} \tag{18}$$

Finally the values of the forces in the local system of coordinates associated with each piston can be written in the global system of coordinate associated with the compressor frame.

7. CONCLUSION

This program is based on a full mechanical model which computes all internal forces from the piston gas force. The strategy is a classical finite differences method with a time step. The program is versatile with multiple possibilities of design using multi level sub-assemblies.

The program is useful for parametric studies regarding :

- the scale effect in relation to the leaks and the number of sealing rings taking into account torque variation. This helps to choose the number of cylinders.

- the bore / stroke ratio and the ratio of the valve areas (suction area / discharge area).

- the parts loading. This is useful to define the bearing and shaft dimensions.

- the flow pulsation. This is interesting to design the suction and discharge volumes and the mufflers.

REFERENCE

Corberan J., Gonzalvez J., Urchueguia J., Calas A., 2000, Modeling of Refrigeration Piston Compressors, *International Compressor Engineering Conference at Purdue,* Proceedings Vol II, p 571-578