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Computer Simulation of the Oil Injected Twin Screw Compressor

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COMPUTER SIMULATION OF THE OIL INJECTED TWIN SCREW COMPRESSOR.

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ABSTRACT

In this paper a physical model for prediction of oil-flooded twin screw compressor performance is presented and discussed.

Calculation results are presented.

INTRODUCTION

At the 1982 PURDUE Compressor Technology Conference a short survey of the possibilities and advantages of this physical model was reported.

The following paper contains more detailed information about the model than was used for the calculations presented at that time.

The model has shown itself to be very useful for a deeper understanding of the twin screw compressor.

An automatic generation of all appropiate geometrical data is coupled to this physical model, which makes it easier to calculate the influences of changes in housing and rotor shape parameters.

The presented calculations in this paper are done in this way.

THEORY FOR OIL INJECTED TWIN SCREW COMPRESSORS.

Following is a summary of the assumptions, that has shown to give the best agreement with laboratory measurements.

The main problem in the understanding and simulation of an oil-injected screw compressor is the behaviour of the internal mass and heat transport of the gas/oil mixture.

However, as there are different leakage paths located within the compressor, the behaviour of the gas/oil mixture in and after these paths is very important for twin screw compressor calculations.

By comparison with laboratory tests the following assumptions for the different types of leakage paths have been shown to be the most appropriate:

1. The gas/oil mixture in all leakage paths is homogeneous.
2. The gas/oil mixture ratio is the same in all leakage paths and equal to the mixture ratio in the discharge port.
3. The heat transfer between gas and oil in the control volume (compressor thread) is treated in two different ways:

a) Leakage over the mesh and the discharge end clearance

Because of the high pressure difference and rotor rotation it is assumed that the oil leakage into the control volume is atomized. This assumption leads to a fully developed heat transfer between the oil and the gas.

b) Leakage over rotor tips and through the blowhole

Due to the centrifugal forces it is assumed that the oil that has leaked into the control volume is located against the housing. This means that only the gas that has leaked into the control volume mixes with the gas already in the control volume.

4. It is assumed that the gas will pass through the outlet port with constant enthalpy. In the case where the outlet port opens before the pressure in the control volume has reached the outlet pressure value, it is assumed that the reverse flowing gas has the same temperature as the gas in the control volume.

It is also assumed that the gas/oil mixture is homogeneous during the passage through the outlet port.

5. The cooling of the gas due to heat transfer during the compression phase is calculated by using an average polytropic exponent.

The formulas for describing the condition in a compressor thread can now be written as follows:

a) Continuity equations

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_d - (\dot{m}_{tip} + \dot{m}_{bh} + \dot{m}_{mesh} + \dot{m}_{de}) + \dot{m}_{evap}$$

$$\frac{dm_{oil}}{dt} = \dot{m}_{oil_{in}} - \dot{m}_{oil_d} - (\dot{m}_{oil_{tip}} + \dot{m}_{oil_{bh}} + \dot{m}_{oil_{mesh}} + \dot{m}_{oil_{de}})$$

After integration of the last equation the real thread volume will be

$$V = V_{geom} - \frac{m_{oil}}{\varrho_{oil}}$$

b). The pressure is calculated according to a polytropic process. (see above assumption 5)

$$p = const \times \varrho^n$$

, where the use of n instead of γ will describe the heat transfer between the gas and the surroundings.

c) Temperature calculation.

From the first law of thermodynamics the following formula for the gas temperature rise in the control volume can be derived

$$\frac{dT}{dt} = \frac{1}{\left(\frac{\partial e}{\partial T}\right)_v} \cdot \frac{1}{m}\left[\frac{dH}{dt} - \left\{\left(\frac{\partial e}{\partial v}\right)_T + p\right\} \cdot \left\{\frac{dV}{dt} - v\sum\left(\frac{\partial m}{\partial t}\right)_{in} + v\sum\left(\frac{dm}{dt}\right)_{out}\right\} + \sum\left(h \cdot \frac{dm}{dt}\right)_{in} - h\sum\left(\frac{dm}{dt}\right)_{in}\right]$$

When calculating dry compressors, this equation can be simplified by assuming ideal gas and polytropic process, which gives:

$$\frac{dT}{dt} = -\frac{1}{m \cdot c_v} \cdot p \frac{dV}{dt} - \frac{T}{m}(n-1) \cdot$$

$$\cdot \sum \left(\frac{dm}{dt}\right)_{out} + \frac{1}{m} \sum \left[\frac{dm}{dt}(n \cdot T_{in} - T)\right]_{in}$$

According to the above mentioned assumptions 1, 2 and 3 it can be understood that this equation must be modified before it is able to calculate the gas temperature rise in the control volume of an oil-injected twin screw compressor.

This modification will be as follows:

$$\frac{dT}{dt} = -\frac{1}{m \cdot c_v} \cdot p \frac{d\left(V_{geom} - \frac{m_{oil}}{\varrho_{oil}}\right)}{dt} - \frac{T}{m}(n-1) \cdot$$

$$\cdot \sum \left(\frac{dm}{dt}\right)_{out} +$$

$$+ \underbrace{\frac{1}{m} \sum \left[\frac{dm}{dt}(n \cdot T_{in} - T)\right]_{in}} +$$

Inlet flow + incoming leakage over rotor tips and through blowhole (see assumption 3 b).

$$+ \underbrace{\frac{1}{m} \sum \left[\frac{1-x}{x} \cdot \frac{dm}{dt} \left(\frac{c_{oil}}{c_p} \cdot n \cdot T_{in} - T\right)\right]_{in}} +$$

incoming leakage through mesh and at the discharge end

$$+ \underbrace{\frac{1}{m} \sum \left[\frac{dm}{dt}(n \cdot T^*_{oil} - T)\right]_{in}}_{\text{evaporation}}$$

, where T^*_{oil} is the calculated oil temperature after taking into consideration that the evaporation has cooled the injected oil.

d) Leakage in leakage paths.

Since the leakage in an oil-injected compressor is effected by the oil viscosity it is necessary to have a theory which includes this effect.

Because the leakage paths in a twin screw compressor have a complicated geometry, the model has to take the shape of the leakage path and its influence on the leakage flow into consideration.

Therefore the leakage model is set using finite elements shown in fig. 1.

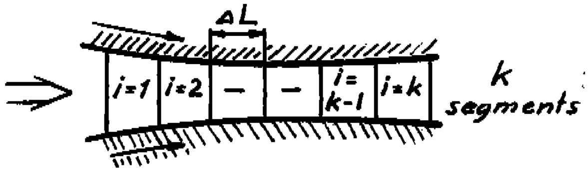


Fig. 1 Leakage path

The pressure drop due to friction is formulated as follows

$$\Delta p_i = f \cdot \left(\frac{\Delta L}{d_{h_i}}\right) \cdot \frac{\varrho_{mix} \cdot u^2_{mix}}{2}$$

where $f = f(Re)$

and $Re = \text{Reynolds number} =$

$$= \frac{\varrho_{mix} \cdot u_{mix} \cdot d_{h_i}}{\eta_{mix}}$$

and $u_{mix} = \dfrac{(\dot{m} + \dot{m}_{oil})}{\varrho_{mix} \cdot A_i}$

ϱ_{mix} is obtained from

$$\frac{1}{\varrho_{mix}} = \frac{x}{\varrho} + \frac{(1-x)}{\varrho_{oil}}$$

The value of η_{mix} is very difficult to find out in a scientific way.

It is found from comparison with experimental data that the following model seems to work well:

$$\eta_{mix} = x \cdot \eta + (1-x) \cdot \eta_{oil}$$

Then the total pressure drop will be:

$$\Delta p_{tot} = K \cdot \frac{\rho_{mix} \cdot u_{max}^2}{2} + \sum_{i=1}^{k} \Delta p_i$$

where K is a constant describing entrance and exit losses.

u_{max} is the velocity of the gas-oil mixture in the smallest section of the leakage path.

From the above equations the leakage due to pressure difference across the leakage paths can be calculated.

However, in practice the 'channel' walls are moving, so therefore it is also necessary to add the leakage depending on this, by using the following formula:

$$\dot{m}_w = \bar{u}_w \cdot \rho_{mix} \cdot A_L$$

, where $\bar{u}_w$ is the average 'channel wall velocity'.

Thus the total leakage is:

$$\dot{m}_L = \dot{m}_{\Delta p} + \dot{m}_w$$

The temperature of the gas and the oil coming out from the leakage paths are, according to assumptions 1 and 2, calculated as follows:

$$T_{mix} = \frac{1}{c_{mix}} \left[x \cdot T \cdot c_p + (1-x) \cdot T_{oil} \cdot c_{oil} \right]$$

$$\text{where} \quad c_{mix} = x \cdot c_p + (1-x) \cdot c_{oil}$$

The enthalpy coming out from the leakage path is then

$$h_{mix} = c_{mix} \cdot T_{mix}$$

e) Filling the compressor thread through inlet port

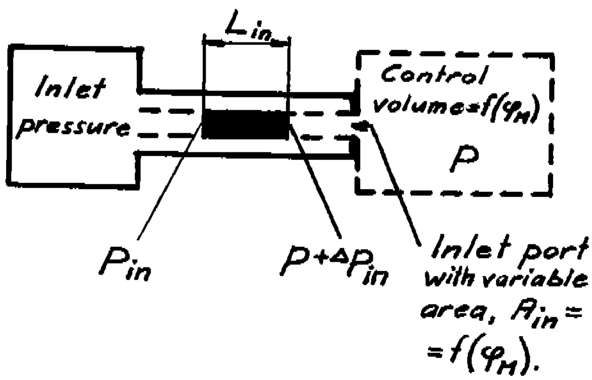


Fig. 2 Inlet modeling

Pressure drop across inlet $= \Delta p_{in} = \frac{1}{c_{in}^2} \cdot \frac{\rho \cdot u_{in}^2}{2}$

Momentum equation gives:

$$\frac{d}{dt}\left(\rho_{in} \cdot L_{in} \cdot A_{in} \cdot u_{in}\right) = \left(p_{in} - \Delta p_{in} - p\right) \cdot A_{in}$$

f) Discharge of the compressor thread through outlet port.

The same principle as above, but in this case the pressure drop across the outlet is

$$\Delta P_d = \frac{1}{C_d^2} \cdot \frac{\rho_d \cdot u_d^2}{2}$$

where ρ_d and u_d are the average density and outlet velocity for a mixture of gas and oil.

Equations expressing these parameters are:

$$u_d = \frac{(\dot{m}_d + \dot{m}_{oil_d})}{\rho_d \cdot A_d}$$

The density ρ_d is calculated as follows:

$$\frac{1}{\rho_d} = \frac{x}{\rho} + \frac{(1-x)}{\rho_{oil}}$$

where

$$x = \frac{\dot{m}_d}{\dot{m}_d + \dot{m}_{oil_d}}$$

It is assumed that the flow through the outlet port and in the outlet pipe takes place with constant enthalpy which leads to:

$$h_d = c_p \cdot T$$

The 'effective area' flow coefficient C_d is dependent on such factors as rotor geometry and diameter, rotor speed, working fluid etc.

CALCULATION OF GEOMETRICAL DATA:

All geometrical data needed for a performance - calculation are generated by preprograms.

These programs are:

- Calculations of length of sealing line between the rotors (mesh and length of sealing lines between male and female tips and the housing):
- Calculation of blow hole area.
- Calculation of volume curve.
- Calculation of axial and radial inlet port areas.
- Calculation of axial and radial outlet port areas.
- Calculation of shape of leakage paths.

All these parameters are expressed as a function of the male rotor angle.

By having this information it is possible to compute the mass and enthalpy leakage and the flow losses as a function of the male rotor angle when the compressor is operating at a steady state speed.

The coupling of the main program and the geometrical data is described in fig. 3.

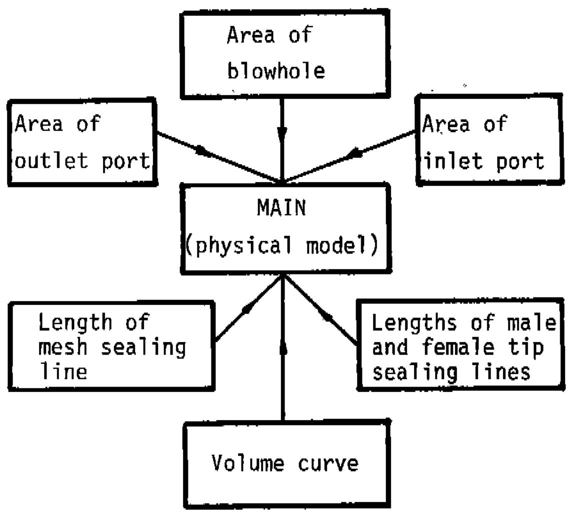


Fig. 3
Coupling of MAIN program and geometrical data

SOME CALCULATION RESULTS:

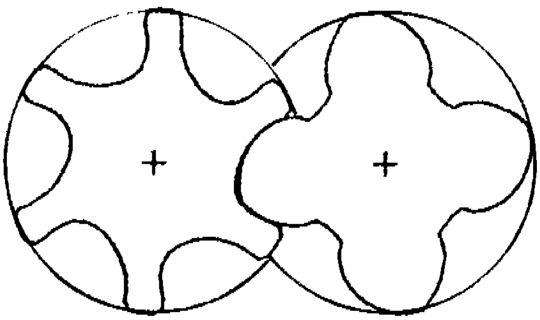
An investigation was made with the computer program concerning the question of what would be the result if the lobe combination was changed.

The two different calculation cases are presented in fig. 4.

The results are shown in table 1.

CASE 1

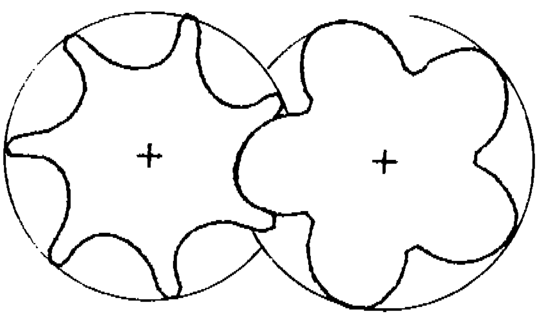
4+6 comb., D_M = 300 mm, D_F = 300 mm



Wrap angle (male) = 300^0

CASE 2

5+7 comb., D_M = 297.1 mm, D_F = 289.4 mm



Wrap angle (male) = 350^0

Fig. 4 Pair of rotors with different lobe combination.

OPERATING CONDITIONS

Gas = R22

Male rotor speed	= 3000 rpm
Evaporation temperature	= 0^0C
Inlet "	= 10^0C
Inlet pressure	= 0.498 Mpa
Condensing temperature	= 40^0C
Outlet pressure	= 1.534 Mpa

Table 1

Radial built in volume ratio = 3.5
Axial " " " " = 4.7

	η_{vol}	η_{ad}
4+6 comb.	0.94	0.69
5+7 comb.	0.96	0.78

Radial built in volume ratio = 2.6
Axial " " " " = 4.7

	η_{vol}	η_{ad}
4+6 comb.	0.94	0.74
5+7 comb.	0.96	0.82

NOMENCLATURE

A_{in} inlet port area
A_L leakage path area
A_i leakage path segment area
c_v specific heat of gas at constant volume
c_p specific heat of gas at constant pressure
c_{oil} specific heat of oil
c_{mix} specific heat of gas-oil mixture
c_{in} flow contraction coefficient at inlet
c_d flow contraction coefficient at outlet
γ adiabatic exponent
D_M male rotor diameter
D_F female rotor diameter
d_h hydraulic diameter of leakage path
e specific internal energy of gas
f friction factor
η dynamic viscosity of gas
η_{oil} dynamic viscosity of oil
η_{mix} dynamic viscosity of gas-oil mixture
η_{vol} volumetric efficiency
η_{ad} adiabatic efficiency
H heat flow into control volume
h_d specific enthalpy of gas at outlet
h_{mix} specific enthalpy of gas-oil mixture
K constant describing entrance and exit losses
k number of segments in leakage path
L_{in} 'inertia' length at inlet
ΔL length of leakage path segment
m mass of gas in control volume
m_{oil} mass of oil in control volume
$\dot{m}_{in}$ mass flow of gas through inlet port
$\dot{m}_{oil_{in}}$ mass flow of oil through inlet port
$\dot{m}_d$ mass flow of gas through outlet port
$\dot{m}_{oil_d}$ mass flow of oil through outlet port
$\dot{m}_{evap}$ mass flow of gas caused by evaporation
$\dot{m}_{bh}$ leakage of gas through blowhole
$\dot{m}_{oil_{bh}}$ leakage of oil through blowhole
$\dot{m}_{de}$ leakage of gas over discharge end
$\dot{m}_{oil_{de}}$ leakage of oil over discharge end
$\dot{m}_{mesh}$ leakage of gas through mesh
$\dot{m}_{oil_{mesh}}$ leakage of oil through mesh
$\dot{m}_{tip}$ leakage of gas over rotor tips
$\dot{m}_{oil_{tip}}$ leakage of oil over rotor tips
$\dot{m}_{\Delta p}$ leakage caused by pressure difference
$\dot{m}_W$ leakage caused by moving walls
$\dot{m}_L$ total leakage through leakage path
n polytropic exponent
p pressure in control volume
p_{in} pressure at compressor inlet
Δp_{in} pressure drop at compressor inlet
Δp_d pressure drop at compressor outlet
Δp_i pressure drop over leakage path segment
Δp_{tot} total pressure drop over leakage path segment
Re Reynolds number
ρ density of gas
ρ_{oil} density of oil
ρ_{mix} density of gas-oil mixture
ρ_{in} density of gas at inlet
ρ_d density of gas-oil mixture at outlet
t time
T temperature of gas in control volume
T_{in} temperature of gas at inlet
T_{mix} temperature of gas-oil mixture
T_{oil} temperature of oil
T^*_{oil} temperature of evaporated gas

u_{in} gas velocity at inlet

u_d gas-oil mixture velocity at outlet

u_{mix} gas-oil mixture velocity in leakage path segment

u_{max} maximum velocity of gas-oil mixture in leakage path

$\overline{u}_w$ average wall velocity of leakage path

v specific volume

V control volume

V_{geom} gemetrical thread volume

φ_M position of male rotor angle

x gas-oil mixture ratio

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