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Research Paper

# A semi-empirical method for assessing the performance of an open-drive screw refrigeration compressor

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## HIGHLIGHTS

- An original semi-empirical model for open-drive screw compressors is presented.
- Ignoring the presence of the lubricating oil does not significantly affect model results.
- The eight model parameters are identified based on eight properly chosen data points.
- Mass flow rate, shaft power and exhaust temperature are appreciably simulated.
- Actual under-estimations are highlighted for few cases with limit pressure ratios.

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## ABSTRACT

A semi-empirical modeling procedure for twin-screw compressor performance simulation is proposed in this paper. In detail, the compression process is split into a number of stages, taking into account fluid leakages, heat transfers as well as heat and power losses. Compared to the simplified models reported in technical literature for other positive-displacement units, a different formulation for ambient heat loss is proposed, while mechanical power losses are modeled according to torque losses depending on both load and viscous friction effects. The proposed model applies to open-drive compressors, so three specific units for refrigeration applications with medium-high evaporation temperature are selected as the test cases for model tuning and validation, based on manufacturer's catalog data. After the identification of eight parameters, the model is able to compute the mass flow rate, the shaft power and the fluid discharge temperature, as well as volumetric and compressor efficiencies and ambient heat losses.

Contrary to geometric models, the semi-empirical procedure proposed in this study does not take the presence of lubricating oil into account, as (i) oil injection with no external cooling results in a thermal neutral process; (ii) the compression work for the oil is negligible in comparison with the one required by the refrigerant fluid; and (iii) even injecting oil at a lower temperature, no considerable saving in refrigerant specific work can be achieved. Actually, this hypothesis does not significantly affect the output of the model. As a matter of fact, referring to the first test case with the larger displacement volume and to the identification of the eight model parameters based on eight convenient data points, the mass flow rate delivered by the compressor is predicted with a maximum error up to 2.44%, and the shaft power is predicted with an even limited error, except for few points with the lowest and highest pressure ratios, where the shaft power is under-estimated by 5%. Moreover, fluid temperature at the compressor exhaust is calculated with an error within ±1 K for pressure ratios up to 7.5. Similarly interesting results are achieved for the other two test case compressors, with absolute values of the errors in mass flow rate and shaft power predictions always less than 3% and 5% respectively.

Finally, further considerations on the actual possibility of generalizing the model to semi-hermetic units are suggested.



## 1. Introduction

The twin-screw compressor is a rotary positive-displacement machine with built-in volume ratio, no suction or discharge valves, no clearance volume, but with a certain amount of internal leakage. These features make the twin-screw compressor behave differently from other positive-displacement machines, like the reciprocating compressor. Usually, in screw compressors, oil is

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injected into the gas to be compressed, is discharged from the compressor in a mixture with the gas and is finally removed from the gas before it leaves the compressor set. The addition of oil increases the delivery pressure as well as the pressure ratio capability, simplifies the compressor design, provides temperature control and reduces both compressor speeds and noise levels. Thus, the use of such compressors during the past decades has been continuing mainly due to the development of specialized machine tools capable of maintaining linear tolerances of the order of 5 μm or less through high accuracy profile milling or grinding at an affordable cost [1].

### 1.1. Common approach in screw compressor simulation

The mathematical modeling of screw compressor processes began more than thirty years ago with the publication of several pioneering papers on this topic, mainly at Purdue compressor conferences. An interesting review of methods and procedures on modern screw compressor practice was presented by Stošić et al. [2]. In particular, the deterministic approach in screw machine simulation is very common, as the geometrical modeling is essential for the investigation of thermodynamic states, fluid leakages, forces, heat transfers, mechanical losses, etc. Dimensions related to screw geometry, chamber volumes and leakage areas are necessary for an accurate mathematical model, either one or multidimensional, so the application of CFD techniques in designing screw machines is playing a significant role, as highlighted by Kovacevic [3] and, more recently, by Rane et al. [4] and Stošić [5].

### 1.2. Objective of the paper

The current study aims at suggesting a semi-empirical procedure for assessing the performance of an open-drive screw compressor. A semi-empirical model consists of a series of thermodynamic equations deduced from mass, energy and momentum conservations, where critical parameters are determined according to real performance data. As a matter of fact, both empirical and semi-empirical models are often applied to describe the thermodynamic behavior of positive-displacement compressors, due to the complex geometry and internal transport mechanisms, as proposed, among many authors, by Winandy et al. [6] or by Byrne et al. [7] for a scroll compressor, and by Duprez et al. [8] for both reciprocating and scroll architectures. Thus, this study is novel as no previous literature work, adopting a semi-empirical approach, focused on screw compressors. In addition, improvements over similar models reported in technical literature for other positive-displacement units are provided. In detail, a more precise formulation for ambient heat loss is proposed, while mechanical power losses are modeled according to torque losses depending on both load and viscous friction effects instead of introducing loss terms without justification.

As described in the next sections, the proposed method can be effectively applied after having identified the eight model parameters based on eight convenient data points, differently from other models for positive-displacement units where lots of data were used for tuning. In detail, three cases referring to different displacements are considered to support the procedure, with an in-depth analysis and considerations on efficiency figures and heat losses. An interesting conclusion on the built-in volume ratio is also reported, based on considerations concerning the parameters calculated for the three test case compressors.

## 2. Screw compressor modeling

The work formerly presented by Winandy et al. [6] for a scroll-type machine is taken as the starting point for the modeling here

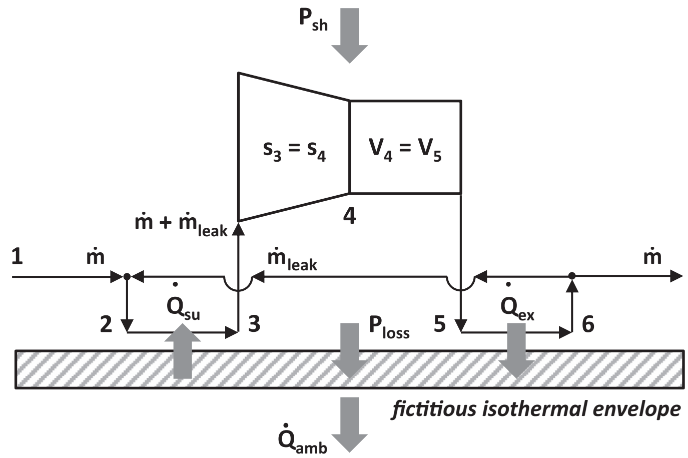


**Fig. 1.** Conceptual scheme of the model for an open-drive screw compressor.

proposed.[1] The description of the fluid compression process, along with the related equations, is included in this section. Some assumptions are preliminarily introduced:

- the kinetic energy of the fluid is neglected in comparison with its internal energy,
- the fluid experiences no pressure drops at both the suction and discharge ports of the compressor,
- fluid leakages through the clearances are assumed to be adiabatic,
- the working fluid only consists of the refrigerant, even though a mixture of oil and refrigerant vapor is actually delivered by the compressor.

The last assumption may be justified with reference to the compressor as an open system, where:

- oil separation at the compressor exhaust and its re-injection with no external cooling result in a thermal neutral process,
- the compression work for the oil can be neglected in comparison with the one required by the refrigerant fluid, as the oil is an almost incompressible fluid, with specific volume really lower than refrigerant fluid vapors,
- even injecting oil at actually lower temperatures, no considerable saving in refrigerant specific work can be achieved, as experimentally highlighted by De Paepe et al. [9].

Fig. 1 shows the schematic processing of the working fluid as it flows through the compressor in a number of hypothetical stages.

The first stage ($1 \rightarrow 2$) considers that the mass flow rate entering the compressor ($\dot{m}$) and the leakage flow rate ($\dot{m}_{leak}$) mix altogether, resulting in a slight increase in fluid specific enthalpy ($h_2 > h_1$):

$$\dot{m} \cdot h_1 + \dot{m}_{leak} \cdot h_6 = (\dot{m} + \dot{m}_{leak}) \cdot h_2 \quad (1)$$

This process is supposedly adiabatic.

The second stage ($2 \rightarrow 3$) is related to the heat transfer between the fluid entering the variable volume chambers and a fictitious isothermal envelope [6,10], i.e. a lumped variable introduced to simulate

[1]Although Winandy et al. [6] proposed their model for a scroll compressor working with HCFC-22, i.e. a phased-out fluid based on the EU regulation 2037/2000, the model has been continued to be used successfully, as demonstrated by Dardenne et al. [27] for a positive-displacement unit working with HFC-410A.