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Modeling of scroll compressors – Improvements

*Marie-Eve Duprez**, *Eric Dumont* [1], *Marc Frère* [2]

Thermodynamics Department, Université de Mons – Faculté Polytechnique, 31 bd Dolez, 7000 Mons, Belgium

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ABSTRACT

This paper presents an improvement of the scroll compressors model previously published by Duprez et al. (2007). This improved model allows the calculation of refrigerant mass flow rate, power consumption and heat flow rate that would be released at the condenser of a heat pump equipped with the compressor, from the knowledge of operating conditions and parameters. Both basic and improved models have been tested on scroll compressors using different refrigerants. This study has been limited to compressors with a maximum electrical power of 14 kW and for evaporation temperatures ranging from −40 to 15 °C and condensation temperatures from 10 to 75 °C. The average discrepancies on mass flow rate, power consumption and heat flow rate are respectively 0.50%, 0.93% and 3.49%. Using a global parameter determination (based on several refrigerants data), this model can predict the behavior of a compressor with another fluid for which no manufacturer data are available.

Modélisation des compresseurs à spirale - améliorations

Mots clés : Compresseur à spirale ; Pompe à chaleur ; Synthèse ; Amélioration ; Modélisation ; Débit ; Consommation d'énergie ; Transfert de chaleur

1. Introduction

Due to the pressure of several laws concerning the limitation of green house gases emission (Kyoto protocol) and in order to save the Earth Planet, the house heating world is changing.

With the objective of decreasing the primary energy consumption in the dwelling sector, the insulation level of new houses becomes better and the heat pump market is expanding. The average increase of heat pump sales between the years 2005 and 2006 in Europe is 52% (EHPA, 2008). Those

* *Corresponding author*. Tel.: +32 65 37 42 08; fax: +32 65 37 42 09.
E-mail addresses: marie-eve.duprez@umons.ac.be (M.-E. Duprez), eric.dumont@umons.ac.be (E. Dumont), marc.frere@umons.ac.be (M. Frère).
[1] Tel.: +32 65 37 42 04; fax: +32 65 37 42 09.
[2] Tel.: +32 65 37 42 06; fax: +32 65 37 42 09.
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Nomenclature

A	Heat exchanger surface area [m^2]
A	T_w correlation parameter [K]
a	$\eta_{pseudo\text{-}iso\text{-}s}$ correlation parameter [–]
B	T_w correlation parameter [–]
b	$\eta_{pseudo\text{-}iso\text{-}s}$ correlation parameter [–]
C	T_w correlation parameter [–]
c_p	specific heat at constant pressure [J kg^{-1} K^{-1}]
d	Pipe diameter [m]
h	Specific enthalpy [J kg^{-1}]
HP	High pressure [Pa]
IP	Intermediate pressure [Pa]
LP	Low pressure [Pa]
N	Rotation speed [r min^{-1}]
Nu	Nüsselt number (U d λ^{-1}) [–]
P	Power consumption [W]
Pr	Prandtl number (μ c_p λ^{-1}) [–]
q_m	Mass flow rate [kg s^{-1}]
Re	Reynolds number (u d ρ μ^{-1}) [–]
S	Surface [m^2]
T	Temperature [K]
U	Heat transfer coefficient [W m^{-2} K^{-1}]
u	Fluid speed [m s^{-1}]
v	Specific volume [m^3 kg^{-1}]
V	Volume [m^3]
ΔT_{sub}	Condenser subcooling [K]
ΔT_{sup}	Evaporator superheating [K]
$\Delta T_{log\ suc}$	Log-mean difference temperature [K]
Φ_C	Heat flow at the condenser [W]
λ	Thermal conductibility [W m^{-1} K^{-1}]
η	Efficiency [–]
μ	Dynamic viscosity [Pa s]
ρ	Density [kg m^{-3}]

Subscripts

cond	Condensation
evap	Evaporation
ex	Exhaust
out	condenser At the outlet of the condenser
pseudo-iso-s	Pseudo-isentropic
suc	Suction
w	Wall

statistics concern the following countries: Austria, Czech Republic, Estonia, Finland, France, Germany, Netherlands, Norway, Sweden and Switzerland. The most important increases are noticed for French and German markets (respectively 144 and 103%).

It would then be useful to obtain a simple calculation tool able to simulate the behavior of a heat pump integrated in a dwelling.

Before modeling a complete system, it is necessary to have the simplest and the most accurate models of its different components. The major component of the heat pump to be modeled is the compressor.

Duprez et al. (2007) have presented a brief review of different models. In some of them the compressor is divided in several chambers in which the compression process is simulated for each gas pocket. They require the knowledge of parameters quite difficult to obtain (scrolls dimensions, pocket perimeters and volumes). The Winandy et al.'s (2002) model is exclusively thermodynamic and is the basis of the model developed by Duprez et al. (2007). An improved version of this model is presented in this paper.

2. Scroll compressors model

Duprez et al. (2007) have presented a way of modeling scroll compressors based on the model developed by Winandy et al. (2002). In this modified model, the authors have divided the compression process in three steps (Fig. 1).

There is first an isobaric heating up in the suction pipe due to a heat transfer with a fictitious wall at temperature T_w (1–2). Then the gas is compressed until the volume created by the scrolls matches the exhaust volume, V_{ex}. This compression is assumed isentropic (2–3) and the pressure reached at point 3 is called "intermediate pressure, IP" (it can be lower or greater than the high pressure). The introduction of a pseudo-isentropic efficiency leads to point 3′. Finally, the end of the compression happens at V_{ex} (constant volume) by mass accumulation or expansion in the exhaust chamber until the pressure is equal to the exhaust one (3′–4).

The data of this model are: the evaporation temperature (T_{evap}), the condensation temperature (T_{cond}) and the temperature at the compressor inlet (T_1) or the superheating (ΔT_{sup}).

The parameters of the model are: the temperature of the fictitious wall (T_w), the heat transfer coefficient multiplied by the heat transfer surface area during the isobaric process in the suction line (UA_{suc}), the suction volume (V_{suc}), the rotation speed of the compressor (N), the ratio between the suction and the exhaust volumes (V_{suc}/V_{ex}), the pseudo-isentropic efficiency ($\eta_{pseudo\text{-}iso\text{-}s}$, which is linearly linked to the compression ratio IP/LP). The details of this model are presented in Duprez et al. (2007).

The model has been tested on several compressors. The mean discrepancies on mass flow rate and power consumption are lower than 3%.

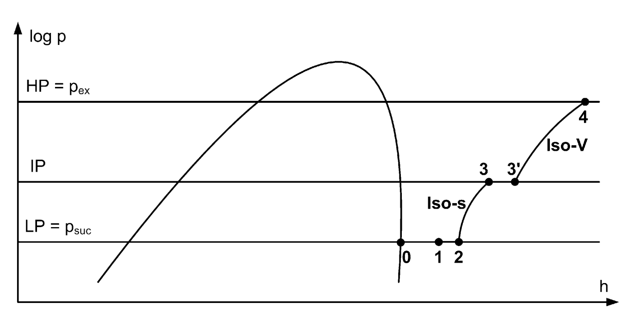

Fig. 1 – Diagram (log p, h) of the compression thermodynamic process.