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Modeling of scroll compressors – Improvements

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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

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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%.

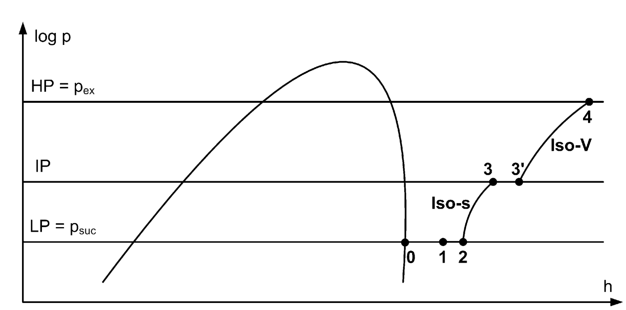

Fig. 1 – Diagram (log p, h) of the compression thermodynamic process.

Table 1 – Constant wall temperature model – Parameters values.

Compressor	Fluid	UA_{suc} [W/K]	V_s/V_{ex} [–]	a [–]	b [–]
# 1 $V_{suc} = 143.678\ cm^3$ $N = 2900$ r/min	R22	60.33	2.187	−0.671	2.329
	R134a	19.01	2.379	−0.777	2.585
	R404A	48.36	2.165	−0.010	0.682
	R407C	42.06	2.219	−0.312	1.464
# 2 $V_{suc} = 172.414\ cm^3$ $N = 2900$ r/min	R22	72.96	2.188	−0.683	2.360
	R134a	22.45	2.380	−0.767	2.560
	R404A	58.14	2.168	−0.004	0.714
	R407C	49.57	2.222	−0.324	1.494
# 3 $V_{suc} = 206.897\ cm^3$ $N = 2900$ r/min	R22	87.09	2.189	−0.677	2.345
	R134a	26.99	2.378	−0.772	2.570
	R404A	69.59	2.166	0.003	0.698
	R407C	58.47	2.220	−0.315	1.471
# 4 $V_{suc} = 37.778\ cm^3$ $N = 3000$ r/min	R22	15.02	1.965	−1.072	2.962
	R134a	3.60	1.977	−0.930	2.480
	R404A	5.00	2.004	−0.404	1.450
	R407C	7.29	2.010	−0.720	2.200
# 5 $V_{suc} = 47.778\ cm^3/$ $N = 3000$ r/min	R22	22.67	2.128	−0.642	2.142
	R134a	5.39	2.302	−0.472	1.766
	R404A	18.97	2.089	−0.320	1.361
	R407C	17.44	2.156	−1.152	3.302
# 6 $V_{suc} = 80.556\ cm^3$ $N = 3000$ r/min	R22	60.35	2.279	0.499	−0.533
	R134a	10.21	2.381	−0.331	1.485
	R404A	19.31	2.160	−0.383	1.524
	R407C	49.82	2.399	−0.391	1.701
# 7 $V_{suc} = 197.556\ cm^3$ $N = 3000$ r/min	R22	131.05	2.937	−0.215	1.422
	R134a	107.02	2.367	−0.134	1.022
	R404A	44.93	2.667	−0.113	0.954
	R407C	146.36	2.799	−0.194	1.273

Table 2 – Constant wall temperature model – Mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate.

Compressor	Fluid	Mean discrepancies [%]			Maximum discrepancies [%]		
		q_m	P	Φ_C	q_m	P	Φ_C
# 1 $V_{suc} = 143.678\ cm^3$ $N = 2900$ r/min	R22	0.09	1.21	1.71	5.61	1.65	5.53
	R134a	1.70	1.12	1.72	5.56	2.75	4.42
	R404A	1.57	0.19	2.43	3.64	0.43	3.84
	R407C	2.31	0.46	2.40	6.16	1.65	4.43
# 2 $V_{suc} = 172.414\ cm^3$ $N = 2900$ r/min	R22	2.16	0.59	3.32	5.45	1.63	5.53
	R134a	1.74	1.09	1.71	5.63	2.68	4.47
	R404A	1.59	0.18	2.44	3.63	0.44	3.84
	R407C	2.29	0.47	2.40	6.04	1.71	4.42
# 3 $V_{suc} = 206.897\ cm^3$ $N = 2900$ r/min	R22	2.18	0.59	3.32	5.58	1.71	5.54
	R134a	1.74	1.12	1.71	5.48	2.78	4.47
	R404A	1.59	0.16	2.44	3.67	0.39	3.84
	R407C	2.31	0.46	2.39	6.17	1.70	4.42
# 4 $V_{suc} = 37.778\ cm^3$ $N = 3000$ r/min	R22	2.28	1.06	3.69	6.91	4.88	5.85
	R134a	2.22	0.85	1.20	9.48	3.59	5.94
	R404A	3.59	1.01	3.15	8.70	5.34	13.31
	R407C	3.17	0.75	6.69	8.19	3.84	10.74
# 5 $V_{suc} = 47.778\ cm^3$ $N = 3000$ r/min	R22	3.15	2.26	4.65	17.09	19.91	9.56
	R134a	1.47	0.85	1.78	4.31	5.35	3.55
	R404A	2.95	1.02	3.51	9.98	4.96	6.86
	R407C	1.97	1.21	4.13	6.56	4.51	5.58
# 6 $V_{suc} = 80.556\ cm^3$ $N = 3000$ r/min	R22	3.59	1.34	8.40	20.73	9.90	24.05
	R134a	1.45	0.90	2.21	4.58	5.26	4.61
	R404A	2.96	1.41	2.54	11.2	7.40	9.32
	R407C	3.12	1.09	8.16	11.06	5.88	16.38
# 7 $V_{suc} = 197.556\ cm^3$ $N = 3000$ r/min	R22	2.78	1.24	5.54	5.77	5.52	11.37
	R134a	2.32	0.53	7.55	5.61	2.35	10.87
	R404A	2.95	1.49	2.81	6.76	5.08	7.29
	R407C	2.95	1.08	7.14	6.68	4.76	13.86

Table 3 – Variable wall temperature model – Parameters values.

Compressor	Fluid	UA_{suc} [W/K]	V_s/V_{ex} [–]	a [–]	b [–]	A [K]	B [–]	C [–]
# 1 $V_{suc} = 143.678$ cm^3 $N = 2900$ r/min	R22	$T_w > 700$ °C						
	R134a	12.77	2.406	−0.762	2.569	−162.3	−1.485	2.886
	R404A	42.05	2.176	−0.015	0.737	−153.5	−0.354	1.840
	R407C	21.89	2.239	−0.351	1.564	−518.8	−1.067	3.700
# 2 $V_{suc} = 172.414$ cm^3 $N = 2900$ r/min	R22	$T_w > 700$ °C						
	R134a	10.81	2.407	−0.752	2.545	−364.3	−2.026	4.051
	R404A	45.27	2.180	−0.031	0.773	−212.6	−0.352	2.038
	R407C	24.45	2.243	−0.367	1.602	−581.6	−1.073	3.914
# 3 $V_{suc} = 206.897$ cm^3 $N = 2900$ r/min	R22	8.32	2.202	−0.706	2.426	−3107	0.621	11.09
	R134a	22.63	2.405	−0.756	2.553	−90.8	−1.293	2.466
	R404A	47.68	2.178	−0.023	0.755	−281.6	−0.368	2.286
	R407C	35.22	2.241	−0.358	1.581	−415.2	−0.962	3.373
# 4 $V_{suc} = 37.778$ cm^3 $N = 3000$ r/min	R22	10.30	1.984	−1.071	2.985	158.5	−1.266	1.673
	R134a			Convergence problems				
	R404A	23.61	2.029	−0.379	1.402	186.2	−0.932	1.132
	R407C	8.84	2.030	−0.780	2.343	−214.2	−0.235	1.897
# 5 $V_{suc} = 47.778$ cm^3 $N = 3000$ r/min	R22	7.208	2.151	−0.641	2.156	−72.7	−1.821	3.014
	R134a	15.74	2.342	−0.444	1.718	338.5	−1.198	0.940
	R404A			Convergence problems				
	R407C			Convergence problems				
# 6 $V_{suc} = 80.556$ cm^3 $N = 3000$ r/min	R22			Convergence problems				
	R134a	35.08	2.416	−0.312	1.451	346.9	−1.034	0.767
	R404A	30.76	2.192	−0.416	1.610	19.3	−0.820	1.653
	R407C	27.71	2.430	−0.365	1.646	513.00	−2.121	1.328
# 7 $V_{suc} = 197.556$ cm^3 $N = 3000$ r/min	R22			Convergence problems				
	R134a			Convergence problems				
	R404A			Convergence problems				
	R407C	132.40	2.864	−0.209	1.339	15.2	−0.039	1.018

Table 4 – Variable wall temperature model – Mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate.

Compressor	Fluid	Mean discrepancies [%]			Maximum discrepancies [%]		
		q_m	P	Φ_C	q_m	P	Φ_C
# 1 $V_{suc} = 143.678$ cm^3 $N = 2900$ r/min	R22	$T_w > 700$ °C					
	R134a	0.09	1.21	1.71	0.26	3.05	3.26
	R404A	0.07	0.16	2.47	0.18	0.37	4.72
	R407C	0.14	0.45	2.71	0.45	1.55	4.93
# 2 $V_{suc} = 172.414$ cm^3 $N = 2900$ r/min	R22	$T_w > 700$ °C					
	R134a	0.10	1.16	1.69	0.20	2.97	3.25
	R404A	0.07	0.14	2.48	0.19	0.37	4.73
	R407C	0.15	0.45	2.71	0.32	1.60	4.93
# 3 $V_{suc} = 206.897$ cm^3 $N = 2900$ r/min	R22	0.18	0.58	3.49	0.50	1.54	5.69
	R134a	0.07	1.19	1.70	0.15	3.15	3.26
	R404A	0.06	0.13	2.47	0.24	0.32	4.71
	R407C	0.15	0.45	2.71	0.29	1.59	4.93
# 4 $V_{suc} = 37.778$ cm^3 $N = 3000$ r/min	R22	0.54	1.07	3.94	2.12	4.83	5.82
	R134a			Convergence problems			
	R404A	0.81	1.12	3.33	3.19	5.16	11.85
	R407C	1.50	0.77	7.10	4.20	3.25	10.80
# 5 $V_{suc} = 47.778$ cm^3 $N = 3000$ r/min	R22	1.66	2.31	4.93	14.80	19.89	9.49
	R134a	0.41	0.89	2.09	1.38	5.60	4.02
	R404A			Convergence problems			
	R407C			Convergence problems			
# 6 $V_{suc} = 80.556$ cm^3 $N = 3000$ r/min	R22			Convergence problems			
	R134a	0.44	0.94	2.56	2.25	5.50	5.06
	R404A	0.84	1.42	2.75	3.26	7.21	8.74
	R407C	1.05	1.17	8.58	2.98	6.04	16.99
# 7 $V_{suc} = 197.556$ cm^3 $N = 3000$ r/min	R22			Convergence problems			
	R134a			Convergence problems			
	R404A			Convergence problems			
	R407C	0.67	1.16	7.37	2.30	5.11	15.84

Table 5 – Variable wall temperature model – Parameters values (all fluids).

Compressor	Fluid	UA_{suc} [W/K]	V_s/V_{ex} [–]	a [–]	b [–]	A [K]	B [–]	C [–]
# 1 V_{suc} = 143.678 cm N = 2900 r/min	R22	23.42	2.124	0.098	0.493	−318.3	−0.324	2.372
	R134a	22.87						
	R404A	38.83						
	R407C	28.78						
# 2 V_{suc} = 172.414 cm^3 N = 2900 r/min	R22	29.77	2.124	0.096	0.497	−285.2	−0.312	2.250
	R134a	29.09						
	R404A	49.37						
	R407C	36.62						
# 3 V_{suc} = 206.897 cm^3 N = 2900 r/min	R22	37.59	2.124	0.096	0.497	−264.6	−0.311	2.177
	R134a	36.73						
	R404A	62.29						
	R407C	46.21						
# 4 V_{suc} = 37.778 cm^3 N = 3000 r/min	R134a	18.02	1.911	0.305	0.053	129.7	−0.410	0.902
	R22	14.92						
	R404A	29.94						
	R407C	22.03						
# 5 V_{suc} = 47.778 cm^3 N = 3000 r/min	R134a	32.40	2.177	−0.354	1.500	292.7	−0.930	0.912
	R22	22.74						
	R404A	40.94						
	R407C	34.29						
# 6 V_{suc} = 80.556 cm^3 N = 3000 r/min	R134a	36.26	2.261	−0.081	0.909	357.6	−1.359	1.143
	R22	25.78						
	R404A	31.97						
	R407C	35.39						
# 7 V_{suc} = 197.556 cm^3 N = 3000 r/min	R134a	88.64	2.611	0.104	0.385	−73.7	0.123	1.162
	R22	72.20						
	R404A	82.66						
	R407C	96.05						

Table 6 – Variable wall temperature model – Mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate (all fluids).

Compressor	Fluid	Mean discrepancies [%]			Maximum discrepancies [%]		
		q_m	P	Φ_C	q_m	P	Φ_C
# 1 V_{suc} = 143.678 cm^3 N = 2900 r/min	R22	2.28	1.23	3.14	4.13	3.95	5.81
	R134a	1.70	5.63	1.54	2.54	10.91	5.03
	R404A	0.84	0.42	2.66	1.54	0.67	5.26
	R407C	0.60	2.07	2.66	1.92	3.63	5.26
# 2 V_{suc} = 172.414 cm^3 N = 2900 r/min	R22	2.32	1.25	3.15	4.06	3.97	5.82
	R134a	1.71	5.58	1.54	2.57	10.86	5.07
	R404A	0.82	0.42	2.65	1.46	0.72	5.25
	R407C	0.58	2.07	2.66	1.78	3.56	5.26
# 3 V_{suc} = 206.897 cm^3 N = 2900 r/min	R22	2.30	1.23	3.15	4.14	4.03	5.82
	R134a	1.70	5.59	1.54	2.58	10.94	5.07
	R404A	0.83	0.42	2.65	1.40	0.73	5.27
	R407C	0.58	2.07	2.66	1.83	3.57	5.27
# 4 V_{suc} = 37.778 cm^3 N = 3000 r/min	R134a	1.90	5.96	1.04	11.53	12.90	2.78
	R22	5.38	2.79	3.72	13.53	9.61	5.39
	R404A	2.35	2.83	4.01	4.29	10.87	14.13
	R407C	1.67	2.68	6.66	5.20	6.74	9.79
# 5 V_{suc} = 47.778 cm^3 N = 3000 r/min	R134a	6.31	6.66	2.44	7.93	13.39	4.78
	R22	1.48	1.71	4.60	14.92	19.85	8.99
	R404A	1.99	1.09	9.01	4.73	3.68	12.74
	R407C	0.71	3.07	5.61	1.83	7.91	8.32
# 6 V_{suc} = 80.556 cm^3 N = 3000 r/min	R134a	7.92	5.54	3.45	10.49	10.46	6.10
	R22	1.64	2.01	8.17	8.57	5.60	19.66
	R404A	2.98	1.85	7.84	8.24	6.78	11.19
	R407C	1.75	2.13	8.90	7.07	10.59	17.22
# 7 V_{suc} = 197.556 cm^3 N = 3000 r/min	R134a	1.79	1.67	8.00	2.83	6.02	14.44
	R22	1.65	3.21	3.61	3.79	9.12	10.96
	R404A	2.28	2.62	5.81	6.12	9.39	12.27
	R407C	1.47	2.20	7.27	4.48	5.26	16.03

Table 7 – Variable wall temperature model – Parameters values (two fluids).

Compressor #5	Fluid	UA_{suc} [W/K]	V_s/V_{ex} [-]	a [–]	b [–]	A [K]	B [–]	C [–]
$V_{suc} = 47.778$ cm N = 3000 r/min	R22 R134a	34.78 48.14	2.155	−0.081	0.841	415.75	−1.056	0.592

Unfortunately, there can be differences of dozens degrees between the simulated compressor outlet temperature and the value of this temperature given by the manufacturer ("theoretical" value). This value is not given directly in the manufacturer's data but is obtained by calculation using other data such as the subcooling, condensation temperature and the heat flow released at the condenser of a heat pump which uses the compressor. The lack of precision on this temperature can lead to large discrepancies between "theoretical" and simulated heat flow released at the condenser of a heat pump which uses the compressor.

In order to minimize the discrepancies on mass flow rate and obtain better values of the outlet temperature of the compressor and thus, of the heat flow, the model must be improved.

In Duprez et al. (2007), the temperature of the fictitious wall was set to a constant value (50 °C). A way of improving the model consists in using a wall temperature depending on the operating conditions.

3. Wall temperature modification

The model improvement consists in using a fictitious wall temperature linked to the phase change temperatures. Different forms of correlations have been tested (linear, quadratic, cubic,...). Depending on the fluid and the compressor type, the best correlation form was different so the chosen one is a good compromise between simplicity, a reduced number of parameters and the results accuracy. This function linking the phase change temperatures is

$$T_w = A + BT_{evap} + CT_{cond} \quad (1)$$

Three new parameters (A, B and C) are thus added to the model and the refrigerant mass flow rate calculation requires the knowledge of six parameters: the three new ones plus UA_{suc}, the suction volume (V_{suc}) and the compressor rotation speed (N). The last two ones are found in the technical datasheets of the manufacturer.

3.1. Comparison of the different models

The two models (Duprez et al. (2007) and the improved one) have been tested on seven compressors (Bitzer, Copeland).

Table 1 presents parameters values obtained with the constant wall temperature model. Table 2 presents the mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate[3]. In most cases, with this model, the mean discrepancies on mass flow rate and power consumption are lower than 3% while those on heat flow rate are lower than 6%.

[3] The calculations of the refrigerant mass flow rate, power consumption and heat flow rate are presented briefly in the Annex.

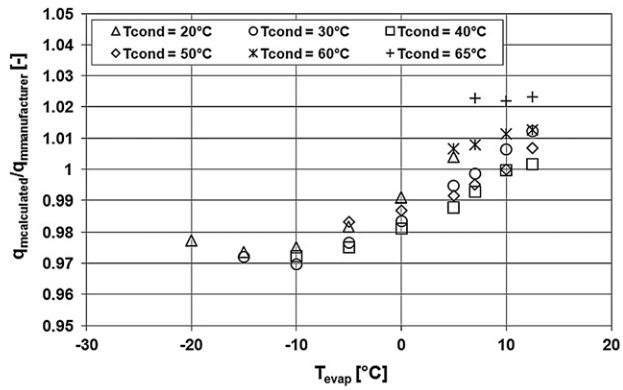

Fig. 2 – $q_{mcalculated}/q_{mmanufacturer}$ versus T_{evap} for compressor #5 with R407C.

Table 3 presents parameters values obtained with the variable wall temperature model. Table 4 presents the mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate. The use of this variable wall temperature decreases the mean discrepancies on mass flow rate (there is hardly any change from the point of view of power consumption and heat flow rate). Sometimes, this model leads to very high wall temperatures or to convergence problems. A way of overcoming those problems is presented hereafter

3.2. Parameters calculation on several fluids

We first began with the determination of parameters fluid by fluid for each compressor. After that we tried to generalize the procedure for one compressor whatever the fluid. In this aim, we made the assumption that the geometric parameter (V_{suc}/V_{ex}) and the five coefficients (two for the pseudo-isentropic efficiency and three for the variable temperature law) are constant for a compressor and that the

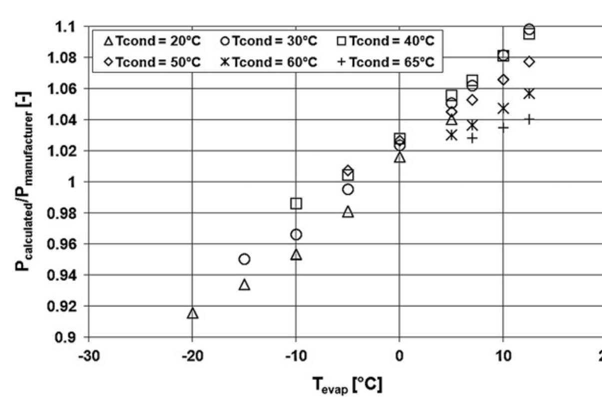

Fig. 3 – $P_{calculated}/P_{manufacturer}$ versus T_{evap} for compressor #5 with R407C.

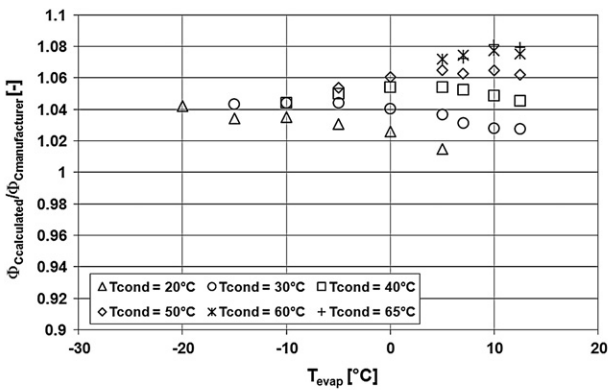

Fig. 4 – $\Phi_{Ccalculated}/\Phi_{Cmanufacturer}$ versus T_{evap} for compressor #5 with R407C.

UA_{suc} depends on the fluid used. Therefore, if the whole set of parameters is calculated for several fluids, a correction must be used to take into account the different fluids in the UA_{suc} values.

A Dittus-Boelter type Eq. (2) is used in order to calculate the heat transfer coefficient, U.

$$Nu = 0.023Re^{0.8}Pr^{0.3} \tag{2}$$

A heat transfer ratio between fluids *i* and *j* is then defined as

$$\frac{U_j}{U_i} = \frac{\frac{\lambda_j}{d}0.023\left(\frac{q_{mj}d\rho_j}{\rho_j S\mu_j}\right)^{0.8}\left(\frac{\mu_j c_{pj}}{\lambda_j}\right)^{0.3}}{\frac{\lambda_i}{d}0.023\left(\frac{q_{mi}d\rho_i}{\rho_i S\mu_i}\right)^{0.8}\left(\frac{\mu_i c_{pi}}{\lambda_i}\right)^{0.3}} \tag{3}$$

The geometric parameters (*d* and *S*) are identical in the two terms of the ratio so that Eq. (4) is obtained.

$$\frac{U_j}{U_i} = \frac{\frac{\lambda_j}{d}\left(\frac{q_{mj}}{\mu_j}\right)^{0.8}\left(\frac{\mu_j c_{pj}}{\lambda_j}\right)^{0.3}}{\frac{\lambda_i}{d}\left(\frac{q_{mi}}{\mu_i}\right)^{0.8}\left(\frac{\mu_i c_{pi}}{\lambda_i}\right)^{0.3}} \tag{4}$$

During parameters determination, a single UA_{suc} is evaluated for one fluid. The UA_{suc} values for the other fluids are obtained by multiplying UA_{suc} by the ratio defined in (4).

Tables 5 and 6 present respectively the parameters values using all the fluids and the mean and maximum discrepancies on mass flow rate, power consumption and heat flow rate for three compressors.

3.3. *Use of global parameters for simulation of other fluids*

If no manufacturer data are available concerning the desired fluid or if the parameter calculation leads to the physical aberrations or convergence problems mentioned above, the improved model can be used with global compressor parameters determined with several other refrigerants.

For example, Table 7 presents the parameters values obtained for compressor # 5 with two fluids: R22 and R134a (manufacturer's data are available for those two fluids as well as R407C and R404A). Then the behavior of this compressor using R407C is simulated ($UA_{suc,R407C} = 51.71$ W/K; this value is obtained using Eq. (4)). Figs. 2, 3 and 4 present respectively the ratio between the calculated mass flow rates, power consumptions and heat flow rates and the manufacturer values. It can be noticed that the mean discrepancy on mass flow rate is lower than 2% while those on power consumption and heat flow rates are both lower than 5%.

4. Conclusion

The scope of this paper was the presentation of an improved model of scroll compressors.

This model calculates the mass flow rate of refrigerant, the power consumption and the heat flow rate released at the condenser of a heat pump that uses this compressor, from the knowledge of operating conditions and different parameters.

The parameters used in the model are either found in the technical datasheets of the compressors (the suction volume, V_{suc}, and the rotation speed of the compressor, *N*) or are determined by fitting the calculated mass flow rates and power consumptions with the values given by the manufacturers (the global heat transfer coefficient at the suction multiplied by the exchange surface area, UA_{suc}, the ratio between the suction volume and the exhaust volume, V_{suc}/V_{ex}, the two coefficients of the pseudo-isentropic efficiency law, a and b, the three coefficients of the variable wall temperature, A, B and C). Four of these parameters are fitted using mass flow rate values (UA_{suc}, A, B and C) and the last three ones are fitted using power consumption values (V_{suc}/V_{ex}, *a* and *b*).

This improved model has been tested on seven compressors. The mean discrepancy values on mass flow rate, power consumption and heat flow rate are respectively 0.79%, 1.32% and 4.12% (R22), 0.22%, 1.08% and 1.95% (R134a), 0.37%, 0.59% and 2.70% (R404A), 0.61%, 0.74% and 5.20% (R407C).

The whole set of parameters can also be determined by using manufacturer data for several fluids and a correlation for the UA_{suc} parameter because it depends on the fluid nature.

Those general parameters can be used in two important cases: when the fitting procedure on one fluid leads to convergence problems or physical aberrations and when there is no manufacturer data concerning the desired fluid. In those cases, the general parameters are very useful for simulating the behavior of the compressor with the desired fluid.

Annex

The refrigerant mass flow rate and the power consumption are calculated using the relations presented below (Duprez et al., 2007).

$$T_1 = T_{evap} + \Delta T_{sup} \tag{5}$$

The temperature at the end of the isobaric heating process (transformation 1–2), T_2, may be deduced from

$$UA_{suc}\Delta T_{logsuc} = q_m(h_2 - h_1) \tag{6}$$

With the log-mean difference temperature defined by

$$\Delta T_{\rm logsuc} = \frac{(T_{\rm w} - T_1) - (T_{\rm w} - T_2)}{\ln\frac{T_{\rm w}-T_1}{T_{\rm w}-T_2}} \quad (7)$$

In Eq. (6), the refrigerant mass flow rate is calculated by Eq. (8)

$$q_{\rm m} = \frac{1}{v_2} V_{\rm suc} N \quad (8)$$

In which v_2 is the specific volume of refrigerant at point 2 (Fig. 1).

$$P = \frac{p_{\rm m}(h_3 - h_2)}{\eta_{\rm pseudo-iso-s}} + (Hp - IP) q_{\rm m} v_{3'} \quad (9)$$

In which h_2 and h_3 are the refrigerant specific enthalpies at points 2 and 3, $v_{3'}$ is the specific volume of refrigerant at point 3′ and the pseudo-isentropic efficiency is expressed by Eq. (7).

$$\eta_{\rm pseudo-iso-s} = a\frac{\rm IP}{\rm LP} + b \quad (10)$$

The heat flow rate is

$$\Phi_{\rm C} = q_{\rm m}(h_{\rm out,condenser} - h_4) \quad (11)$$

Where the enthalpy at the condenser outlet $h_{\rm out,condenser}$ is

$$h_{\rm out,condenser} = f({\rm HP}, (T_{\rm cond} - \Delta T_{\rm sub})) \quad (12)$$

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