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Transient model of a geothermal heat pump in cycling conditions – Part A: The model

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ABSTRACT

A transient model of a water-to-air geothermal heat pump subjected to cycling operation is proposed in this paper. A distributed approach is used where each component is modeled individually and then linked into a so-called global model. The model can predict the state of the refrigerant throughout the circuit during normal operation and at compressor shut down where there may still be some small refrigerant flow. The governing equations of refrigerant and secondary fluid flows, along with tube wall temperatures evolutions, are solved using the finite volume method. The originality of the proposed model lies in its distributed approach which allows to capture both the on-cycle and the off-cycle, with fan on or off at compressor shut-down. Results presented in a companion paper show that the model is able to predict with relatively good agreement the measured transient behavior of a geothermal heat pump.

Modèle transitoire d'une pompe à chaleur géothermique en régime cyclique – Partie A : le modèle

Mots clés : Régime transitoire ; Pompe à chaleur ; Géothermique ; Écoulement diphasique ; Frigorigène ; Modèle des volumes finis

1. Introduction

Geothermal heat pumps have relatively high coefficient of performance (COP) in both heating and cooling and are among the most energy efficient and environmentally friendly heating and cooling systems. However, as noted by Votsis et al. (1992) and Katipamula and O'Neal (1991), performance may decrease at reduced loads when heat pumps are cycling. This paper examines these cycling effects using a distributed model approach to fully capture the transients affecting the heat pump in its operation. The results of the experimental validation of the model are presented in a companion paper (Ndiaye and Bernier, 2012).

To quantify the performance of a heat pump under cycling conditions, a dynamic model is necessary. Dynamic models of vapor compression systems (such as heat pumps or

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Nomenclature

A	area (m^2)
Bo	boiling number
C	coefficient
C	compressor clearance ratio
CH	heat loss characteristic parameter
CP	pressure drop characteristic parameter
Cp	specific heat capacity ($J\ kg^{-1}\ K^{-1}$)
d	condensation length (m)
D	diameter (m)
e	flute depth (m)
e^*	non-dimensional flute depth
f	friction factor
F	friction force per unit volume ($N\ m^{-3}$)
F	force (N)
g	gravitational acceleration ($m\ s^{-2}$)
h	enthalpy ($J\ kg^{-1}$)
h	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
K	constant
L	length (m)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
MCP	calorific capacity ($J\ K^{-1}$)
n	polytropic exponent
N	frequency (Hz)
Nu	Nusselt number
p	flute pitch (m)
p^*	non-dimensional flute pitch
P	perimeter (m)
P	pressure (Pa)
Pr	Prandtl number
Q	heat flux per unit volume ($W\ m^{-3}$)
$\dot{Q}$	heat flux (W)
r^*	diameter ratio
Ra	Rayleigh number
Re	Reynolds number
t	time (s)
T	temperature (°C)
u	velocity ($m\ s^{-1}$)
UA	thermal conductance ($W\ K^{-1}$)
v	specific volume ($m^3\ kg^{-1}$)
v	secondary fluid velocity ($m\ s^{-1}$)
V	volume (m^3)
w	specific theoretical work ($J\ kg^{-1}$)
$\dot{W}$	electrical power (W)
x	direction of air flow in plenum
x	vapor quality
X_{tt}	Martinelli parameter
z	direction along the tube (m)

Greek letters

α	flute helix angle (°)
α	void fraction
β	thermal expansion coefficient (K^{-1})
∂	partial differential
Δ	differential
ϕ^2_{fo}	two-phase frictional multiplier
μ	viscosity ($kg\ m^{-1}\ s^{-1}$)
ρ	density ($kg\ m^{-3}$)
θ^*	non-dimensional flute angle

Subscripts

a	air in the plenum
ann	annular
b	bulb
bc	bulb content
bw	bulb wall
c	compressor
cd	condenser
ch	compressor crankcase heater
d	diaphragm
D	compressor discharge side
dis	compressor discharge tubing
e	external
en	entrance
env	environment
ev	evaporator
ex	exit
f	free volume
f	friction
h	hydraulic
i	internal
l	liquid phase
M	thermal mass
P	perimeter
P	plenum
r	refrigerant
S	compressor suction side
S	spring
SS	static superheat
suc	suction
sw	swept
t	tube
v	vapor phase
v, vol	volume
w	tube wall

refrigerators) may be subdivided into three categories: (i) empirical models, (ii) lumped-parameter models, and (iii) distributed models.

Empirical models are based on one (Goldschmidt et al., 1980; Katipamula and O'Neal, 1991) or two time constants (Mulroy and Didion, 1985; Tree and Weiss, 1986; Votsis et al., 1992) or on other correlations. These models rely on experimental data or on data issued from a more elaborate model.

Lumped-parameter and distributed models are based on the governing equations describing the physics of the studied phenomenon. For these two categories, the modeling essentially consists in regrouping the models of the main components (evaporator, condenser, compressor, expansion valve). Lumped-parameter and distributed models differ in their treatment of the evaporator and condenser.

Distributed models have been the subject of several investigations by McArthur and Grald (1987), Vidmar and