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A general model for two-stage vapor compression heat pump systems

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ABSTRACT

Two-stage vapor compression technology has high potential of performance improvement for cold climate heat pumps, and there are several types of inter-stage configurations that need to be evaluated before making a choice. A general model of these configurations is first derived from a subcooler cycle and then is extended to be capable of evaluating many other inter-stage configurations by employing an “input domain”. The model is solved with a sequential algorithm and an analytical initial solution of the intermediate pressure is presented. After an experimentally validation with additional calculations of the subcooling parameter, the evaporating and condensing pressure, this general model is then used in the performance comparison and analysis of eight different inter-stage configurations. At last, case studies show that, this general model is capable of performing performance comparison among cycles with different types of inter-stage configurations, as well as refrigerant selection and operational analysis.

Un modèle général pour les sytèmes de pompee à chaleur à compression de vapeur biétagée

Mots clés : Pompe à chaleur ; Système biétagé ; Frigorigène ; Injection

1. Introduction

When an air source heat pump (ASHP) runs under low ambient temperature, several problems restrict its application: deteriorated heat output, high discharge temperature and declining coefficient of performance (COP), due to an increase of the pressure ratio. Several concepts have been proposed in the literature to improve the performance of ASHP under low ambient temperature (Bertsch and Groll, 2008; Jin et al., 2008; Jung et al., 2013; Sami and Tulej, 1995; Zehnder, 2004), and one of the most effective concepts is the two-stage compression approach, which has received much attention in recent years due to its reliable and improved performance when temperature differences between the heat sink and heat source are high. Two-stage technology implements the compression process with two lower pressure ratios; as a result, wider operating range could be obtained.

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Nomenclature

a_1–a_4, c_1–c_4	coefficients in the efficiency formulas (–)
COP	coefficient of performance (–)
C	coefficient in relation to frequency (–)
c_p	specific heat at constant pressure (kJ kmol^{-1} K^{-1})
h	specific enthalpy (kJ kg^{-1}))
f	frequency (Hz)
$\dot{m}$	refrigerant mass flow rate (kg s^{-1})
n	polytropic exponent (–)
p	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
R_m	refrigerant mass flow ratio (–)
R_V	theoretical displacement ratio (–)
SH	superheat (°C)
T	temperature (°C)
V	volume (m^3)
$\dot{w}$	work (kW)
w	specific work (kJ kg^{-1})
x	refrigerant quality (–)
ε	heat exchanger subcooling parameter (–)
η	efficiency (–)
ν	specific volume (m^3 kg^{-1})
ρ	density (kg m^{-3})

Subscripts

c	cooling
cy	cylinder
dis	discharge
el	electric
H	high-stage compressor
h	heating
in	input
inj	injection
rat	ratio
L	low-stage compressor
m	intermediate
suc	suction
T	total
vol	volume
th	theoretical
rat	ratio

Superscript

*	parameters at base frequency

Besides, by using inter-stage configurations, such as subcooler or flash tank, performance of the system can be further improved.

Experimental studies were carried out to evaluate the performance of a two-stage compression system under different operating conditions. For example, Tian and Liang (2006) proposed and tested a two-stage compression variable frequency ASHP with a subcooler as the inter-stage configuration and experimental results showed that the heating COP was higher than 2.0, the discharge temperature of the high-stage compressor was below 120 °C when the condensing temperature was 50 °C and the evaporating temperature was −25 °C. Bertsch and Groll (2008) designed, constructed and tested a two-stage heat pump with subcooler for water and heating, which was able to operate at ambient temperatures between −30 °C and 10 °C with supply water temperatures of up to 50 °C. The system could run in single-stage and in two-stage operating mode, and at the same ambient temperature, two-stage mode operation approximately doubles the heating capacity compared to single-stage mode operation. They reported that a heating COP of 2.1 was observed at an ambient temperature of −30 °C. The discharge temperatures of the compressors in two-stage mode stayed below 105 °C at all times. Jin et al. (2008) designed and tested a two-stage heat pump with flash tank. Experimental results showed that the system was capable of providing 50 °C heating hot water when outside temperature was −20 °C, with a heating capacity of 4.71 kW and COP of 1.76, and the discharge temperature of the high-stage compressor was below 100 °C.

There are several inter-stage configurations that can be implemented in a two-stage compression system, as presented in Fig. 1, resulting in different two-stage compression cycles with different performances. Moreover, the theoretical displacement ratio of the low-stage and high-stage compressors is important for compressor selection or optimal operation (Baek et al., 2014; Shapiro and Rohrer, 2006). As a result, an evaluation of these cycles should be performed before making a choice. The evaluation can be implemented with a steady-stage simulation model. For example, Ma et al. (2003) established a thermodynamic model of a subcooler system working with R22. The authors used an empiric factor to reflect the error between practical process and ideal process and assumed an isentropic compression process. Bertsch and Groll (2008) performed simulations for performance comparisons between two-stage cycle with intercooler, two-stage cycle with subcooler and cascade cycle. The simulation model consisted of sub-models for each component of the cycle. These components models were then be arranged and used for all three heat pump cycles, in which way the results were directly comparable. The standard 3rd order polynomial approach according to the ARI standard (ANSI/AHRI Standard 540, 2004) was used for the compressor model. Nikolaidis and Probert (1998) applied the exergy method to investigate the behavior of two-stage compound compression cycle with flash tank; the inter-stage pressure was assumed to be the geometric mean pressure. Torrella et al. (2011) established a general methodology suitable for analyzing any intermediate configuration considered in staged vapor compression refrigeration cycles; seven types of cycles were presented. In their work, the general methodology only depends on two basic parameters related to the degree of subcooling and the degree of desuperheating obtained by the inter-stage configuration. Xu and Ma (2014) established an analytical model of a flash tank cycle working with R410A to obtain an optimum volume ratio of high-pressure cylinder to low-pressure one for the experimental plant; the isentropic efficiency of the compression process was calculated with empirical formula and the volumetric efficiency was not considered. A compressor with volume ratio of 7.5 was finally manufactured based on the calculation results.

Nevertheless, the previous theoretical analysis was always performed for a specified cycle, or between two or three types