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Review

Mechanical sub-cooling vapor compression systems: Current status and future directions

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ABSTRACT

Using mechanical sub-cooling systems to increase COP of vapor compression cycles is a known method in literature to save energy and increase efficiency. Recently, much progress has been made with respect to investigation into its different aspects that can help to put it into practice. Numerical and experimental works are considered for the purpose of highlighting this progress. These can be categorized as: a) simulation of performance characteristics resulting from different refrigerant combinations in dedicated mechanical sub-cooling systems, b) variation in performance characteristics for a vapor compression cycle using integrated mechanical sub-cooling because of fouling, c) experimental study about consequences of employing a dedicated mechanical subcooling cycle with a simple vapor compression system, and d) experimental investigation about consequences of employing a subcooler in a two-stage refrigeration cycle. Some important results are discussed. Finally, some suggestions are made to provide direction into future research in this area to help put it into practice.

Sous-refroidissement mécanique des systémes de compression de vapeur: situation actuelle et orientations futures

Mots clés : Sous-refroidissement ; Compression de vapeur ; Amélioration de performance ; Systèmes frigorifiques

1. Introduction

Air-conditioning systems, which are often air-cooled, consume a considerable amount of the electricity generated on the planet. Large differences in temperature exist between the condenser and evaporator in these systems. As a result of this, the compressor consumes a greater amount of power and provides a smaller refrigeration effect. The system can be

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Nomenclature

c_p	specific heat capacity (kJ kg^{-1} K^{-1})
$\dot{C}$	rate of thermal capacitance (kW K^{-1})
COP	coefficient of performance (–)
f_h	ratio defined by Eq. (3) (–)
h	specific enthalpy (kJ kg^{-1})
HEICE	Heat Exchanger Inventory Cost Equation
K	total inventory for heat exchangers (kW K^{-1})
FPI	fins per inch (–)
$\dot{m}$	mass flow rate of refrigerant (kg s^{-1})
P	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
R_f	resistance due to fouling (K W^{-1})
SLHX	suction-liquid heat exchanger (–)
T	temperature (°C or K)
UA	overall conductance (kW K^{-1})
$\dot{V}$	volumetric flow rate (m^3 s^{-1})

Greek symbols

α	main cycle evaporator inventory (kW K^{-1})
β_{main}	main cycle condenser inventory (kW K^{-1})
β_{sub}	sub-cooler cycle condenser inventory (kW K^{-1})
η	efficiency (–)
ε	heat exchanger effectiveness (–)
θ	dimensionless subcooler saturation temperature $(=(T_{sc,ev}-T_{m,ev})/(T_{m,cd}-T_{m,ev}))$ (–)

Subscripts

cd	condenser
cl	clean condition
cp	compressor
dl	discharge line
ev	evaporator
fl	fouled condition
in	entering
is	isentropic
m	main cycle
min	minimum
nsc	no sub-cooler cycle
N	normalized
sc	sub-cooler cycle
sl	suction line

damaged under such conditions if the compressor operates for a long time (Abdel-Nabi et al., 1990). In the United States, the contribution of supermarkets to system peaks is considerable (EPRI, 1984). These refrigeration systems are an important sector of energy demand and they represent a large percentage of total energy utilization. Simple vapor compression systems are often used, for example, in cold storage applications and cooling supermarket display cases where evaporator temperatures vary from −40 °C to 7 °C (Zubair, 1990).

Subcooling has been extensively utilized in medium- and low-temperature refrigeration systems (Miller, 1981; Couvillion et al., 1988) wherein a simple vapor-compression refrigeration system is altered to save energy. Subcooling technologies are as follows: a) ambient subcooling b) suction-line heat exchanger usage as heat sink, c) systems with an external heat sink, and d) mechanical subcooling. Each will now be briefly described with focus on mechanical subcooling in detail.

1.1. *Ambient subcooling*

Vapor compression refrigeration systems with ambient subcooling use additional heat exchange surface, which interacts with the ambient, to subcool the liquid refrigerant. There are two ways to provide ambient subcooling (ASHRAE, 1983):

a) The subcooler and condenser are combined in an oversized condenser.
b) A separate heat exchanger (called the subcooler) is used downstream of the condenser.

In practice, the subcooler is often sized, at design conditions, to provide 8 °C of subcooling (EPRI, 1989) because the condensing pressure decreases when additional heat transfer area is added. Therefore, the restriction imposed to the expansion device must be increased to obtain refrigerant subcooling. Such an approach, however, is not always beneficial for the system as, on one hand, the capacity increases, whereas, on the other hand, the amount of refrigerant charge increases as well. The degree of subcooling is restricted by the heat sink temperature. Thorton (1991) modeled ambient subcooling systems and, at design conditions, found the subcooler UA to be about 1/20th of the condenser size. He also showed that ambient subcooling systems surpass fixed and floating head pressure systems due to increased heat rejection to the ambient. This increased heat rejection to the ambient is the reason for the higher COP. It was noted that ambient subcooling with fixed head pressure was especially effective at low ambient temperatures as compared to standard fixed head pressure systems.

1.2. *Subcooling with liquid-suction heat exchanger*

According to ASHRAE (1998), liquid-suction heat exchangers are useful for: a) improving system performance, b) subcooling liquid refrigerant (exiting the condenser) to prevent formation of flash gas at expansion valve inlets, and c) evaporating any remaining liquid in the suction line before entering the compressor.

However, it should be noted that although the subcooling obtained by means of the suction line heat exchanger is always beneficial since the refrigerant charge diminishes but the size effect of increasing the specific volume in the compressor inlet may lead to lower COPs.

Fig. 1 shows a vapor compression refrigeration system with a liquid-suction heat exchanger. In it, the high-temperature liquid exiting the condenser is sub-cooled by an indirect exchange of heat before entering the throttling device. This is the liquid-suction heat exchanger in which the sink is the low-temperature refrigerant vapor exiting the evaporator. It should be noted that it is often the case with the low-temperature side of this heat exchanger that it works as an