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Combined effects of fluid selection and flow condensation on ejector operation in an ejector-based chiller

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ABSTRACT

This work investigates the optimization of the coefficient of performance of an ejector-based chiller through changes in the two-phase flow characteristics inside the ejector using wet R134a and dry R245fa fluids. Reducing the superheat at the motive nozzle inlet results in a 12–13% increase in COP with a 14–16 K drop in driving waste heat temperature. The roles of momentum transfer, heat transfer, and two-phase flow on performance are delineated. The change in COP appears to be a combination of the choice of fluid and the effect of phase change on momentum transfer effectiveness. Larger degrees of condensation reduce momentum transfer effectiveness; however, energy savings from reduced motive superheating compensates for the effect of condensation, and causes a net increase in COP. It is recommended that ejector-based chillers be operated such that the motive nozzle inlet is near saturation, and dry fluids like R245fa are used to improve performance.

Effets combinés de la sélection et de la condensation en écoulement d'un fluide sur le fonctionnement d'un éjecteur dans un refroidisseur à base d'éjecteur

Mots clés : Éjecteur ; Condensation ; Tuyère ; Refroidisseur ; Récupération de chaleur perdue ; Froid

1. Introduction

Ejector-based chillers are simple devices that produce cooling from waste heat sources. Because of the absence of a compressor and the ability to use non-toxic refrigerants with low global warming potential, they are safe, low-cost, and low-maintenance chiller systems suitable for off-grid and/or portable applications. The central component in an ejector-based chiller is the ejector component itself. Unlike a mechanical compressor,

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Nomenclature

a	speed of sound [m s^{-1}]
A_c	cross-sectional area [m^2]
A_s	surface area [m^2]
CFD	computational fluid dynamics
COP	coefficient of performance [–]
c_P	specific heat at constant pressure [J kg^{-1} K^{-1}]
E	energy [W]
Gr	Grashof number [–]
h	specific enthalpy [J kg^{-1}]
M	Mach number [–]
$\dot{m}$	mass flow rate [kg s^{-1}]
MER	mass entrainment ratio [–]
P	pressure [Pa]
Pr	Prandtl number [–]
$\dot{Q}$	heat transfer rate [W]
R	gas constant [J kg^{-1} K^{-1}]
s	specific entropy [J kg^{-1} K^{-1}]
$\dot{S}_{gen}$	entropy generation rate [W K^{-1}]
T	static temperature [K]
V	velocity [m s^{-1}]
$\dot{W}$	work [W]
x	quality [–]
Z	Z-factor [–]

Greek symbols

γ	specific heat ratio [–]
μ	dynamic viscosity [N s m^{-2}]
ρ	density [kg m^{-3}]
σ	Stefan–Boltzmann constant [W m^{-2} K^{-4}]
τ	shear stress [N m^{-2}]

Subscripts

amb	ambient
avg	average
ext	external
fg	enthalpy of vaporization
K	boundary
m	motive inlet property
o	ejector outlet property
s	suction inlet property
sat	saturation
sup	superheat
tot	total

an ejector compresses vapor without the use of moving parts and the associated lubricants, bearings, and required maintenance. Furthermore, the absence of moving parts allows for miniaturization of the ejector component to result in small chiller package sizes. The ejector operates on the principle of momentum transfer from a high speed supersonic jet to entrain a second, lower-potential flow to create a pumping effect. The operation of an ejector is shown qualitatively in Fig. 1, where the variation in pressure and velocity is given at the axial positions indicated in the upper ejector schematic. From axial positions $M \rightarrow i \rightarrow ii$, a high-temperature and pressure motive refrigerant flow expands through a converging-diverging nozzle to produce a supersonic jet at the motive nozzle exit. This supersonic flow entrains a suction flow entering from $S \rightarrow ii$. In the ejector mixing section $ii \rightarrow iii \rightarrow iv$, the motive and suction flows interact and mix until the combined flow nominally reaches a supersonic velocity. At iv, the flow must adjust to conditions at the ejector outlet, producing a set of shocks depicted in Fig. 1 as an idealized normal shock at iv, before the beginning of the diffuser at v. From $v \rightarrow vi$, pressure is recovered as the total flow decelerates to a low velocity at the ejector outlet. The desired compression effect produced by the ejector is the rise in pressure from the suction inlet S to the ejector outlet vi.

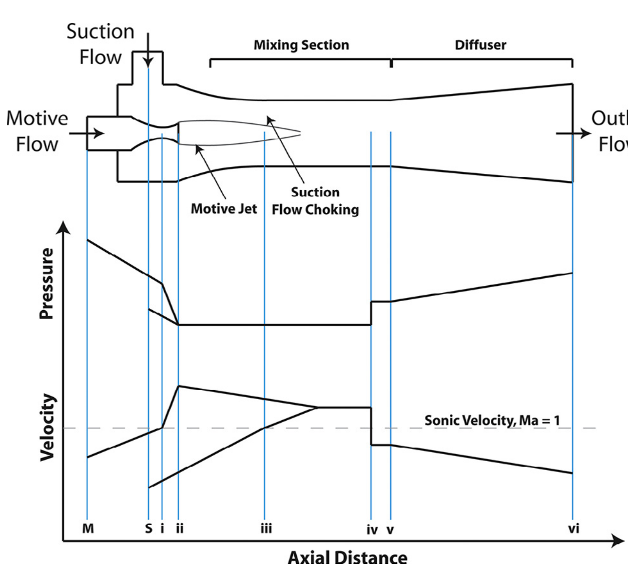

Fig. 1 – Schematic of ejector with corresponding qualitative pressure and velocity profiles for nominal operation, adapted from Srisastra and Aphornratana (2005).

The specific chiller configuration under consideration in this study that uses the ejector is shown in Fig. 2a. An upper loop boils a refrigerant, in this case R134a or R245fa, to the desired state point 1. It then enters the ejector through the motive nozzle where it expands to state 2, defined as the exit of the motive nozzle. It then mixes with the suction flow entering from the bottom loop to state 3, and recovers pressure to the ejector outlet at state 4. This total flow enters the condenser where heat is rejected to the ambient until slightly subcooled conditions are reached (~3 K). At this point, the flow splits, part being pumped through the upper loop to state 8 at the inlet of the boiler, and part expanding through the expansion valve in the bottom loop to the inlet of the evaporator. From states 6 to 7, heat is removed from the conditioned space to produce the desired cooling effect. The flow at state 7 is entrained into the ejector through the suction nozzle to mix with the motive flow from the upper loop.

The exact properties of state point 1 are of great importance in this study. Conventional operation of the cycle dictates that state 1 be superheated by >10 K (typically between 25 and 30 K for gas turbines (ASME, 1992; Boyce, 2006)). This is because in radial and axial expanders, two-phase flow is avoided to