LOW-SOURCE-TEMPERATURE DIFFUSION ABSORPTION REFRIGERATION. PART I: MODELING AND CYCLE ANALYSIS

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ABSTRACT

The diffusion absorption refrigeration (DAR) cycle offers a potentially fully thermally activated cooling technology. However, most implementations operate with high source temperatures, forced liquid cooling, or elevated evaporator temperatures ($\gtrsim 5^\circ\text{C}$). Additionally, few component design resources are available in the literature. In Part I of this investigation, a fully passive DAR design is proposed. Reduced temperature operation is enabled with alternate working fluids ($\text{NH}_3\text{-NaSCN-He}$), a distributed heated bubble-pump generator (BPG), and an enhanced absorber. Detailed models are formulated for the BPG, condenser, evaporator, absorber, and gas circulation loop. These are integrated to yield an overall system model. System behavior is evaluated over a range of operating conditions. With the necessary and reasonably expected component performances, refrigeration COPs of 0.11 – 0.26 can be achieved at design conditions ($T_{\text{amb}} = 24^\circ\text{C}$) with low source temperatures (110 – 130°C) and passive air cooling. In the accompanying paper (Part II), this refrigeration system is experimentally demonstrated, and the proposed models are evaluated.

KEYWORDS

Diffusion absorption refrigeration; absorption; bubble pump; heat and mass transfer

NOMENCLATURE

A	area (m^2)
a	correction factor in model of Liu and Winterton (1991)
AAD	average absolute deviation
CAVP	closest approach vapor pressure (kPa)
CAT	closest approach temperature (K)
C_r	fluid stream thermal capacity rate ratio
C_0	distribution parameter (in bubble rise velocity model)
c_p	specific heat at constant pressure ($\text{kJ kg}^{-1} \text{K}^{-1}$)
D	diameter (m)
D_{ab}	binary diffusion coefficient ($\text{m}^2 \text{s}^{-1}$)
f	friction factor (N m^{-3})
f_{LS}	blending factor (capillary-to-large diameter)
g	gravitational acceleration (9.81 m s^{-2})
H	height (m)
h	convection heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
h'	adjusted convection coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
h^*	mass transfer convection coefficient ($\text{kg m}^{-2} \text{s}^{-1} \text{Pa}^{-1}$)
i	enthalpy (kJ kg^{-1}), or segment number
ID	inner diameter (m)
j	superficial velocity (m s^{-1})
k	thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)
L	length (m)
L^*	dimensionless length (for developing flow)
LMPD	log-mean vapor pressure difference (kPa)
LMTD	log-mean temperature difference (K)
m	mass flow rate (kg s^{-1})
MW	molar mass (kg kmol^{-1})
n	number of measurements, or segment / cell count
NTU	heat exchanger number of transfer units
OD	outer diameter (m)
p	pressure (Pa), or tube pitch
p_v	vapor pressure (kPa)
per	perimeter (m)
Q	heat transfer rate (W)
q	heat flux (W m^{-2})
R	radial dimension (m), or thermal resistance (K W^{-1})
R'	thermal resistance $\times$ unit length (m K W^{-1})
R''	thermal resistance $\times$ unit area ($\text{m}^2 \text{K W}^{-1}$)
R^*	universal gas constant ($8.314 \text{ kJ kmol}^{-1} \text{K}^{-1}$)
$R^{*'}_*$	mass transfer resistance $\times$ unit length ($\text{m}^2 \text{s}^{-1}$, or kmol s kg^{-1})
seg	segment value in axially discretized models
S_r	submergence ratio (reservoir height / bubble pump height)
T	temperature ($^{\circ}\text{C}$)
U	velocity (m s^{-1})
UA	overall heat transfer conductance (W K^{-1})

UA^*	overall mass transfer conductance ($\text{kg Pa}^{-1} \text{s}^{-1}$)
V	volumetric flow rate (ml min^{-1})
WF	wetting factor (liquid film coverage of surface)
x	position vector (m)
x,y	radial, axial coordinates (px, m)
z	axial position (m)

Greek characters

α	void fraction
β	length fraction of unit cell occupied by Taylor bubble
Γ	drift flux parameter in bubble velocity model, or mass flow per unit width in liquid film ($\text{kg m}^{-1} \text{s}^{-1}$)
Δ	change in quantity
δ	thickness (of liquid film)
ε	heat exchanger effectiveness
η	fin efficiency
μ	dynamic viscosity ($\text{kg m}^{-1} \text{s}^{-1}$)
ξ	auxiliary gas (He) mass fraction in gas mixture
ξ^*	auxiliary gas (He) mole fraction in gas mixture
ρ	mass density (kg m^{-3})
ρ^*	molar density (kmol m^{-3})
σ	surface tension (kg s^{-2}), or Stefan-Boltzmann constant ($5.67 \times 10^{-8} \text{W m}^{-2} \text{K}^{-4}$)
τ	shear stress (N m^{-2}), or time scale (s)
ψ	absorbent mass fraction in solution

Subscripts

1, 2, 3...	segment number, or section of condenser
a	annulus
abs	absorber component
acc	fluid acceleration (pressure change)
air	ambient air value
amb	ambient value
aux	auxiliary gas mixture
avg	average value
b	bubble
BPG	bubble-pump generator
C	heat exchanger cool side
Ca	capillary parameter
CF	coupling fluid
cool	cool stream value, or cooling delivery temperature
cond	condenser component
conv	convection heat transfer
cryst	solute crystallization point
CS	absorbent-concentrated solution stream
d	hydrodynamic quantity

DS	absorbent-dilute solution stream
evap	value for evaporation phase change, or evaporator
ext	external value
f	liquid film, or frictional value, final value
fin	fin value
forced	forced flow
G	gas phase
GHX	gas heat exchanger
H	heat exchanger hot side
hs	hydrostatic (pressure change)
i	inner value, or segment index
in	inlet value, or inner fin channel
int	liquid-to-gas interface, or internal value
L	liquid phase
L0	all-liquid-flow value
loss	heat loss (to surroundings)
LS	large-scale channels
LV	liquid-to-vapor change
LW	value from the model of Liu and Winterton (1991)
nat	natural convection value
NB	nucleate boiling value
net	total value
NH ₃	pure ammonia value
NH ₃ -NaSCN	sodium thiocyanate
o	outer value, or overall surface value
out	outlet value, or outer fin value
rad	radiation heat transfer
ref	refrigerant flow value
res	value in liquid reservoir
row	tube row
s	liquid slug
sat	saturation conditions
SHX	solution heat exchanger
sol	solution flow value
source	heat source
sub	subcooled value, or subcooler component
sys	system value (pressure)
trans	Taylor bubble to liquid slug transition
tube	per-tube or tube-wall value
V	vapor phase

Dimensionless parameters

Bo	Bond number $(\rho_L - \rho_G)gD^2/\sigma$
Ca	capillary number $\mu_L j/\sigma$, or $\mu u/\sigma$
COP	coefficient of performance $Q_{\text{evap}}/Q_{\text{BPG}}$

N _f	viscous number $\sqrt{\rho_L (\rho_L - \rho_G) g D^3 / \mu_L^2}$
Nu	Nusselt number hD/k
Pr	Prandtl number $\mu c_p / k$
Re _b	bubble-to-film Reynolds number $\rho_G (U_b + U_f) (D - 2\delta_f) / \mu_G$
Re _f	liquid film Reynolds number $4\rho_L U_f \delta_f / \mu_L$
Re _j	superficial velocity Reynolds number $\rho_L jD / \mu_L$
Re _{j,L}	superficial liquid velocity Reynolds number $\rho_L j_L D / \mu_L$
Sc	Schmidt number $\mu / \rho D_{ab}$
Sh	Sherwood number $h^* D / D_{ab}$

1. INTRODUCTION

1.1 Background

The diffusion absorption refrigeration (DAR) cycle (von Platen and Munters, 1928) has the potential to be implemented as a fully passive refrigeration technology. The technology can deliver cooling using solely thermal input, requiring no electrical input for pumps, compressors, or external fans. Historically, DAR systems have been employed in niche applications such as hotel room refrigerators, for which quiet operation is desirable. In this configuration, electrical resistance heating is employed to drive the cycle. DAR devices have also been commercialized for use in camping/recreational vehicles, wherein heat can be supplied from chemical fuel (e.g., propane). However, both of these applications do not exploit the main advantage of the DAR cycle as one that can be driven by potentially low-exergy sources.

Recently, the DAR cycle has received renewed attention for critical off-grid applications (Acuña *et al.*, 2013; Rodríguez-Muñoz and Belman-Flores, 2014). For example, such systems could be used for medicine and vaccine refrigeration ($T_{\text{cool}} \lesssim 5^{\circ}\text{C}$) in developing countries where electricity or chemical fuel supplies are unavailable or unreliable. In such applications, heat could be supplied with a passive solar thermosyphon or direct solar heating (Dammak *et al.*, 2010). Similarly, this technology could be employed for stationary generator waste-heat activated refrigeration or air-conditioning in military forward operating bases where electricity use must be carefully budgeted.

1.2 DAR cycle description

A schematic of the basic DAR cycle is presented in Figure 1. Because the DAR cycle is intended for operation in a fully passive, thermally activated fashion, mechanical means of circulating working fluid between components (e.g., electrically driven compressors or pumps) should not be employed. Rather, fluid is transported through passive mechanisms. In the bubble-

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4 pump generator (BPG), liquid solution is lifted by the buoyancy of rising desorbed refrigerant-
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6 vapor bubbles. This process establishes the hydrostatic head that drives liquid flow through other
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8 components in the system (e.g., absorber and solution heat exchanger (SHX)).
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12 DAR systems are pressurized with an inert low-density non-condensable, non-absorbable
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14 gas, usually helium or hydrogen. Net gas circulation is effected by the density difference
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16 between warm refrigerant-poor gas mixture rising out of the absorber and cool refrigerant-rich
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18 gas flowing downward from the evaporator. Thus, the cycle can be idealized as two passively
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20 circulating fluid loops (liquid and gas) that interact in the absorber.
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25 The DAR cycle operates at nearly a single total pressure, with only slight (~1%) variations
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27 due to hydrostatic effects. Thus, the conventional throttling mechanism for refrigerant expansion
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29 and cooling employed in vapor-compression and two-pressure absorption systems cannot be
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31 utilized. Rather, the liquid refrigerant stream flowing out of the condenser is contacted with
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33 refrigerant-poor (low vapor pressure) auxiliary gas flowing from the absorber. The vapor
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35 pressure difference between these two streams causes refrigerant evaporation and the cooling
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37 effect. DAR systems have conventionally operated with ammonia refrigerant, water absorbent,
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39 and helium or hydrogen auxiliary gas (Zohar *et al.*, 2005).
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46 In the basic DAR cycle implementation (Figure 1), absorbent-dilute (refrigerant-rich)
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48 solution (DS) flows from the reservoir to the generator (1), where it is heated to desorb
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50 refrigerant vapor. The refrigerant vapor and solution flow up through the bubble-pump generator
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52 (BPG) (2), and are lifted by the buoyancy of the vapor bubbles. Refrigerant vapor separates from
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54 the solution flow at the BPG outlet, and continues to the condenser (4). The refrigerant rejects
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56 heat to the surroundings as it flows through the condenser and exits as subcooled liquid (5). The
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Figure 1 – Schematic of basic DAR cycle configuration

liquid refrigerant partially flashes in the presence of the refrigerant-poor auxiliary gas mixture at the evaporator inlet. The two-phase mixture continues to evaporate and cools the conditioned space as it flows through the evaporator towards the absorber vapor inlet (6). Absorbent-concentrated (refrigerant-poor) solution (CS) from the BPG (3) flows into the absorber solution inlet, and contacts the gas stream from the evaporator in a counter-flow fashion (3 → 1). Refrigerant is absorbed from the gas mixture into the solution, rejecting heat to the surroundings and regenerating both the DS (1) and auxiliary gas streams (7). The DS returns to the reservoir, and the refrigerant-poor gas rises to the evaporator inlet, completing the cycle.

Additional components are employed in practical DAR systems to improve efficiency. These include recuperative heat exchangers to preheat solution flowing to the BPG (SHX) and precool auxiliary gas (GHX) and refrigerant (RPC) flowing to the evaporator. If a volatile absorbent is

employed (e.g., water), then a rectifier would also be employed to purify the refrigerant stream from the BPG. Detailed cycle-level analyses of DAR systems incorporating these components have been developed by Zohar *et al.* (2005, 2007, 2008) and Starace and De Pascalis (2012, 2013).

Commercial DAR systems can operate at refrigeration cooling coefficients of performance ($\text{COP} = Q_{\text{evap}}/Q_{\text{BPG}}$) of 0.2 – 0.3 (Jakob *et al.*, 2008). This value is significantly lower than vapor compression ($\text{COP} \sim 2.0 - 3.0$) or two-pressure absorption systems (0.5 – 1.0), but can be achieved with only thermal input.

1.3 Motivation and objectives

Conventional DAR systems require high source temperatures of 150 – 200°C (Jakob *et al.*, 2008; Srikuhrin *et al.*, 2001) to deliver cooling at temperatures suitable for refrigeration. If this required input temperature can be reduced ($\lesssim 130^\circ\text{C}$), then DAR systems can operate with abundant low-grade thermal sources, such as solar heat or engine waste heat. A number of investigations have been performed to reduce source temperatures through the adoption of alternate refrigerant-absorbent pairs (other than $\text{NH}_3\text{-H}_2\text{O}$). These include R134a-He (Wang *et al.*, 2011), R22-DMAC (Zohar *et al.*, 2009), $\text{NH}_3\text{-LiNO}_3$ (Wang, 2012), and $\text{NH}_3\text{-NaSCN}$ (Acuña *et al.*, 2013). $\text{H}_2\text{O-LiBr}$ has also been considered (Pfaff *et al.*, 1998), but may not be suitable for refrigeration-grade cooling due to the relatively high freezing point of water (0°C).

In conventional BPG designs, heat is transferred to the working fluid through a small region near the base of the component. This *spot-heated* configuration limits heat transfer area, necessitating high source temperatures for given heating rates (i.e., $T_{\text{source}} - T_{\text{BPG}} \propto Q_{\text{BPG}}/A$). Alternate *distributed-heated* BPG designs have been proposed in recent investigations (Dammak *et al.*, 2010; Jakob *et al.*, 2008; Rattner and Garimella, 2015a), which provide larger heat transfer

area, permitting operation with reduced source temperatures. However, this distributed-heated configuration is significantly more challenging to design and analyze.

Existing DAR device designs may not be suitable for operation with alternate working fluids at significantly reduced source temperatures. In previous experimental investigations, system performance has been ensured with high-flux liquid condenser and absorber cooling (Jakob *et al.*, 2008; Srihirin and Aphornratana, 2002) or forced internal fluid circulation (Wang, 2012). While these approaches have enabled proof-of-concept demonstrations, fully passive, low-source-temperature operation is necessary for target applications. Therefore, detailed component- and system-level models must be formulated to guide DAR development efforts under more constrained operating conditions. Such models will also be valuable for refinement of conventional DAR equipment. In the following section, reviews of prior work on low-source temperature DAR design and DAR modeling are presented.

1.4 Prior work

1.4.1 Low-source-temperature DAR systems

A number of detailed investigations have been performed on commercially available DAR implementations with $\text{NH}_3\text{-H}_2\text{O-H}_2/\text{He}$ working fluids (Chen *et al.*, 1996; Srihirin and Aphornratana, 2002; Zohar *et al.*, 2005). Such systems can operate fully passively with high BPG temperatures, and deliver a wide range of cooling COPs ($T_{\text{source}} = 140 - 200^\circ\text{C}$ and $\text{COP} = 0.05 - 0.20$ (Srihirin *et al.*, 2001); $T_{\text{source}} = 160 - 230^\circ\text{C}$ and $\text{COP} = 0.20 - 0.30$ (Jakob *et al.*, 2008)). Low-source-temperature DAR systems have also been experimentally demonstrated in recent years. Jakob *et al.* (2008) developed a DAR system with high efficiency (up to $\text{COP} = 0.48$), high capacity ($Q_{\text{evap}} = 2.5$ kW cooling), and relatively low source temperature ($T_{\text{source}} = 130^\circ\text{C}$) with $\text{NH}_3\text{-H}_2\text{O-H}_2$ working fluid. However, this system was only designed for chiller and A/C applications ($T_{\text{evap}} = 6^\circ\text{C}$), and employed forced liquid cooling for the evaporator and

Figure 2 – Map of demonstrated DAR operating condition windows

condenser. Ben Ezzine *et al.* (2010b) demonstrated a fully passive (air-cooled) experimental DAR system with source temperatures of 125 – 150°C. They employed a new hydrocarbon working fluid mixture ($\text{C}_4\text{H}_{10}\text{-C}_9\text{H}_{20}\text{-He}$), which permitted use of low-cost copper heat and mass exchangers. Evaporator temperatures of -10°C were reported, but cooling performance was only rated for delivery temperatures of 9°C, with COPs of 0.10 – 0.14.

Wang (2012) demonstrated a low-source temperature ($T_{\text{source}} = 93^\circ\text{C}$) DAR-based cooling system with $\text{NH}_3\text{-LiNO}_3\text{-He}$ as the working fluid. Evaporator temperatures of -13°C were reported (COP = 0.16). However, the system employed both forced liquid external condenser and absorber cooling and mechanical internal fluid circulation. While these experimental systems represent significant advances in the field, fully passive, low-source-temperature ($\lesssim 130^\circ\text{C}$) cooling at refrigeration temperatures ($T_{\text{cool}} \lesssim 5^\circ\text{C}$) has not yet been achieved. The current range of demonstrated DAR operating conditions is summarized in Figure 2.

A number of cycle-level modeling studies have also been performed focusing on low-source-temperature DAR systems. Zohar *et al.* (2009) investigated a number of organic refrigerant-DMAC absorbent pairs. Operation at source temperatures of 135 – 150°C was predicted for R22-DMAC and R32-DMAC. Ben Ezzine *et al.* (2010a) modeled a DAR system with the R124-DMAC refrigerant-absorbent pair. They applied representative state point temperatures for both air and water cooled condenser and absorber components. Evaporator temperatures of -20 – 0°C were predicted for $T_{BPG} = 120^\circ\text{C}$ in the air-cooled implementation. Wang *et al.* (2011) proposed an implementation with a binary organic refrigerant (R23/R134a) and DMF absorbent. Very low cooling temperatures ($T_{cool} = -40^\circ\text{C}$) were predicted with heat source temperatures of 110 – 160°C.

Acuña *et al.* (2013) compared DAR system operation with $\text{NH}_3\text{-H}_2\text{O}$, $\text{NH}_3\text{-LiNO}_3$, and $\text{NH}_3\text{-NaSCN}$ refrigerant-absorbent pairs. Operating parameters for the cycle model were based on experimental results from the study of Jakob *et al.* (2008). The $\text{NH}_3\text{-LiNO}_3$ and $\text{NH}_3\text{-NaSCN}$ pairs were predicted to significantly outperform $\text{NH}_3\text{-H}_2\text{O}$ at low source temperatures (120°C). A number of other DAR cycle modeling efforts were reviewed by Rodríguez-Muñoz and Belman-Flores (2014).

Overall, these investigations highlight a number of promising cycle configurations and alternate working fluids that could enable fully passive low-source-temperature refrigeration. However, almost all have employed somewhat simplified closure relations (e.g., assumed state points, approach temperatures, and flow rates). Therefore, detailed component-level design models are needed before the proposed systems can be realized.

1.4.2 DAR modeling and design studies

A number of experimental and cycle level modeling studies have been performed for DAR systems. However, few detailed, experimentally validated component- and system-level design models have been reported.

Srikhirin and Aphornratana (2002) developed a partial DAR cycle model that incorporated selected state values from an experimental facility. This approach enabled rating of individual components, but not prediction of overall system performance. BPG flow rates were predicted using a curve fit to experimental (air-water) component measurements. Zohar *et al.* (2005, 2007, 2008) published a series of complete thermodynamic system level DAR models that assumed certain state point values and temperature changes, effectively setting the fluid flow rates. Chaouachi and Gabsi (2007) developed a DAR system model that incorporated a bubble pump model to determine the refrigerant and solution flow rates, but employed a number of simplifying assumptions such as perfect rectification, complete vapor absorption, and isothermal cooling. Rattner and Garimella (2011) developed a cycle level model that incorporated mechanistic BPG and gas loop passive circulation descriptions. Component heat-transfer rates were predicted using assumed overall conductance (UA) values rather than specified approach temperatures. This permitted more realistic analysis of a specific physical system at varying operating conditions. Starace and De Pascalis (2012, 2013) refined the models of Zohar *et al.* (2005, 2007, 2008), and accounted for mixture effects in the rectifier and incorporated a mechanistic BPG model. Such cycle-level modeling efforts provide starting points toward practical system design. However, all DAR components operate passively, with many incorporating complex two-phase heat and mass transfer phenomena. Detailed component-level models are therefore needed to realize target UA values and heat transfer rates.

The BPG component has received the most attention in the literature. A number of investigations have been performed focusing on the spot-heated configuration (adiabatic bulk flow) with air-water flow (de Cachard and Delhay, 1996; Delano, 1998; Reinemann *et al.*, 1990; Schaefer, 2000; White, 2001). Pfaff *et al.* (1998) conducted an experimental investigation of spot-heated bubble pumps with water-lithium bromide working fluid. Koyfman *et al.* (2003) performed an experimental study of spot-heated R22 bubble-pumps. van der Walt (2012) performed a BPG modeling study, and incorporated flow-boiling heat transfer correlations to predict the developing flow in the short generator region.

A number of recent BPG investigations have focused on the distributed-heated configuration, which could enable operation at lower source temperatures. Jakob *et al.* (2008) designed and fabricated a vertical shell-and-tube coupling-fluid-heated BPG for a residential-scale DAR air-conditioning system. However, no component models were reported. Dammak *et al.* (2010) performed a modeling study of an inclined full-length heated BPG that could be integrated into a flat-plate solar collector. Benhimidene *et al.* (2011) also modeled a full-length uniformly heated BPG. Rattner and Garimella (2015a) formulated a model for a full-length coupling-fluid heated BPG, and experimentally validated it with an ambient pressure steam-water based system. Garma *et al.* (2014) performed an Eulerian-Eulerian two-phase simulation of BPG flow. Jo *et al.* (2014) also performed simulations of full-length uniformly heated BPGs. While a number of distributed-heated BPG studies have been performed, experimental validation of models at realistic DAR system operating conditions has not yet been reported.

No detailed heat and mass transfer models have been reported for the condenser, evaporator, gas heat exchanger (GHX), solution heat exchanger (SHX), solution subcooler, and passively circulating gas loop. Only one model has been proposed for the DAR absorber (Rattner and

Garimella, 2012). While some experimental studies provide implementation details (Ben Ezzine *et al.*, 2010b; Jakob *et al.*, 2008; Srihirin and Aphornratana, 2002; Wang, 2012), limited resources are available for actual device design. Additionally, all flows in DAR systems are passively activated. Thus, component performances are tightly coupled, and system design must be performed in an integrated fashion.

1.5 Present investigation

Results from previous investigations of DAR technology indicate that low-source-temperature ($T_{\text{source}} \lesssim 130^{\circ}\text{C}$), fully passive, and refrigeration-grade cooling ($T_{\text{evap}} \lesssim 5^{\circ}\text{C}$) can be achieved. Such performance goals necessitate the use of alternate working fluids (other than $\text{NH}_3\text{-H}_2\text{O-H}_2/\text{He}$), improved and accurate distributed-heated BPG designs, and significant reconfiguration of other cycle components.

In this investigation, analyses are performed to develop a small capacity (10 – 30 W cooling), fully passive DAR refrigerator with $\text{NH}_3\text{-NaSCN-He}$ as the working fluid mixture. This system is designed to operate in an environment with a low ambient air velocity ($U_{\text{amb}} = 0.12 \text{ m s}^{-1}$ assumed here), such as an indoor setting with background ventilation or outdoors with a breeze. Detailed, first-principles-based, coupled fluid flow and heat and mass transfer models are formulated to guide design of the more complex system components: the BPG, condenser, evaporator, absorber, and passive circulation gas loop. The simpler single-phase heat exchangers (GHX, SHX, and solution subcooler) are not considered here for brevity. Results from these detailed analyses inform simplified models (e.g., UA-LMTD descriptions), which are incorporated into a complete cycle model. This cycle model is employed to predict system-level performance over varying operating conditions. In the companion paper (Part II), these models are applied to develop a complete experimental DAR system. Experimental results are used to

Figure 3 – a. Schematic of bubble-pump generator design, b. Segmented model configurations and discretization approach

assess model accuracy and identify future research needs and paths to DAR technology refinement.

2. COMPONENT-LEVEL MODELING AND RESULTS

2.1 Bubble-pump generator

The bubble-pump generator (BPG) design considered here (Figure 3a) is similar to the configuration proposed by Rattner and Garimella (2015a). Two-phase working fluid flows upward in a narrow vertical tube as vapor is desorbed. Heat is supplied from a co-flow mineral-oil-based coupling fluid in an outer annular channel (Paratherm NF heat transfer fluid (Paratherm Corporation, 2013)). The inner tube terminates in a settling tank to separate the liquid solution and vapor refrigerant streams. This configuration was shown to achieve stable liquid pumping at low coupling-fluid inlet temperatures ($T_{CF,in} - T_{sat} \sim 10$ K for ambient pressure steam-water flow in Rattner and Garimella (2015a)), enabling DAR system operation with low-grade thermal sources. Additionally, low submergence ratio (the ratio of solution reservoir level to BPG height, $S_r = H_{res}/H_{BPG}$) operation was demonstrated ($S_r = 0.2 - 0.4$), enabling compact system

configurations. A model is developed here, accounting for the coupled two-phase flow, flow boiling heat transfer, and mixture effects in the BPG. This is an expanded formulation based on the pure-fluid BPG model proposed by Rattner and Garimella (2015a). Fluid material property formulations are described in Appendix A.

Representative model results are reported for a configuration with inner tube dimensions of $ID_i = 4.9$ mm and $OD_i = 6.4$ mm, and outer tube dimensions of $ID_o = 7.0$ mm and $OD_o = 9.5$ mm (Figure 3a). The full BPG length is 1.74 m, with heat transfer between the working fluid and coupling fluid occurring in the lower 1.60 m. In this case, subcooled dilute solution flows into the BPG inlet at $T_{L,in} = 64.0^\circ\text{C}$ and a NaSCN mass fraction of $\psi_{in} = \psi_{DS} = 0.538$. The inner tube absolute pressure is $p_{sys} = 1.15$ MPa, yielding an inlet saturation temperature of $T_{sat,in} = 73.0^\circ\text{C}$. The heating coupling fluid enters the annular channel at $T_{CF,in} = 120.2^\circ\text{C}$ and a flow rate of $V_{CF} = 175$ ml min^{-1} . Heat loss from the coupling fluid to the surroundings is evaluated assuming an overall thermal resistance of $R_{loss} = 2.0$ K W^{-1} . The driving pressure difference corresponds to a submergence ratio of $S_r = 0.275$ (i.e., $p_{in} - p_{out} = \rho_L g H_{BPG} S_r = 4590$ Pa). The relatively minor variations in hydrostatic pressure along the component are neglected in the evaluation of fluid properties. An ambient temperature of $T_{amb} = 24^\circ\text{C}$ is assumed. This value is lower than that employed in some residential refrigerator rating protocols ($T_{amb} = 32^\circ\text{C}$ (United States Department of Energy, 2012)), but is representative of anticipated operating conditions (i.e., room temperature). Similar ambient temperature conditions have been employed in prior low-source-temperature experimental DAR studies ($27 - 31^\circ\text{C}$ (Jakob *et al.*, 2008), 20°C (Ben Ezzine *et al.*, 2010b), $10 - 28^\circ\text{C}$ (Wang, 2012)). The effect of varying ambient conditions on overall DAR system performance is analyzed in Section 3.

The model is axially segmented, with 20 main diabatic BPG segments (1.60 m total), and a single adiabatic segment (0.14 m) corresponding to the portion of the inner tube in the settling vessel. Consecutive segment lengths increase at a constant ratio (+8.8%, $\Delta z_{20}/\Delta z_1 = 5$) to better resolve sharp variations near the BPG inlet. Flow properties are determined at the end points of each segment, and mass, species, energy, and momentum balances are applied over each segment (Figure 3b).

Mass balances are applied over each segment of the BPG inner tube.

$$m_{L,i} + m_{V,i} = m_{L,i+1} + m_{V,i+1} \quad (1)$$

Liquid and vapor flow rates are predicted to vary from the BPG inlet to outlet from 54.1 – 49.4 g min⁻¹ and 0 – 4.7 g min⁻¹, respectively (55.4 – 50.6 mlmin⁻¹ and 0 – 676 ml min⁻¹). The predicted vapor flow rate peaks at $z = 1.16$ m (4.77 g min⁻¹), and decreases slightly toward the outlet value of 4.70 g min⁻¹ due to heat loss to the surroundings. Liquid and vapor flow rate profiles are presented for the representative case in Figure 4.

NaSCN species balances are applied over each segment of the inner tube.

$$m_{L,i}\psi_i = m_{L,i+1}\psi_{i+1} \quad (2)$$

The predicted solution concentration varies from the dilute solution value of $\psi_{DS} = 0.538$ at the BPG inlet to the concentrated solution value of $\psi_{CS} = 0.589$ at the outlet. This corresponds to an increase in saturation temperature at $p_{sys} = 1.15$ MPa from 73.1 – 87.4°C along the length of the BPG.

Heat transfer from the coupling fluid (CF) annulus is modeled using a correlation for fully developed laminar flow ($Re_{CF} = 50$, $h_{CF} = 720 \text{ W m}^{-2} \text{ K}^{-1}$) (Shah and London, 1978). Radial

Figure 4 – Developing liquid and vapor flow rate profiles in the bubble-pump generator

conduction resistance through the stainless steel inner tube ($k_{\text{tube}} = 14 \text{ W m}^{-1} \text{ K}^{-1}$) is minor ($R_{\text{tube}}/R_{\text{CF}} = 0.04$). Heat loss from the coupling fluid channel to the surroundings is evaluated assuming equal thermal resistance along the component length ($R_{\text{loss},i} = R_{\text{loss}} \times H_{\text{BPG}}/\Delta z_i$). Segmental CF energy balances and inner wall temperatures are evaluated as follows:

$$m_{\text{CF}} c_{p,\text{CF}} (T_{\text{CF},i+1} - T_{\text{CF},i}) = -Q_i - Q_{\text{loss},i} \quad (3)$$

$$Q_i = \frac{T_{\text{CF},i} - T_{\text{tube},i}}{R_{\text{CF},i} + R_{\text{tube},i}} \quad (4)$$

$$Q_{\text{loss},i} = \frac{T_{\text{CF},i} - T_{\text{amb}}}{R_{\text{loss},i}} \quad (5)$$

Here, m_{CF} is the coupling fluid flow rate (147 g min⁻¹), $c_{p,CF}$ is the specific heat (2.18 kJ kg⁻¹ K⁻¹), $T_{tube,i}$ is the inner-tube inner-wall temperature, and T_{amb} is the ambient temperature (24°C). Over the length of the BPG, the CF flow temperature decreases from 120.2°C to 86.7°C. A heat duty of 144.6 W is transferred to the working fluid, while 34.4 W is lost to the surroundings.

The heat transfer and energy balance models for the NH₃-NaSCN mixture must account for the variation of saturation temperature with solution concentration. Additionally, heat loss to the surroundings can cause condensation near the outlet of the BPG. Heat transfer to the working fluid is partitioned between liquid sensible heating ($Q_{L,i}$), vapor generation and sensible heating of the vapor ($Q_{V,i} = Q_i - Q_{L,i}$) as follows.

$$m_{L,i+1}i_{L,i+1} - m_{L,i}i_{L,i} = Q_{L,i} \quad (6)$$

$$m_{V,i+1}i_{V,i+1} - m_{V,i}i_{V,i} = Q_i - Q_{L,i} \quad (7)$$

The heat transfer from the inner tube wall to the liquid and vapor phases is evaluated based on the model of Liu and Winterton (1991), which was validated for subcooled flow boiling. This model applies correction factors to single-phase liquid heat transfer ($h'_{L0} = a_{L0} \times h_{L0}$) and no-flow nucleate boiling ($h'_{NB} = a_{NB} \times h_{NB}$) convection coefficients to account for interactions between these two phenomena. For subcooled boiling, this correlation blends the two heat transfer mechanisms quadratically:

$$Q_i = \left\{ \left[h'_{L0,i} (T_{tube,i} - T_{L,i}) \right]^2 + \left[h'_{NB,i} (T_{tube,i} - T_{sat,i}) \right]^2 \right\}^{1/2} \quad (8)$$

Here, T_L is the liquid bulk temperature. The same quadratic weighting employed in the original correlation is used to partition Q into sensible (Q_L) and phase-change (Q_V) contributions (Equation 9). A limiting factor is applied so that the liquid temperature cannot exceed the

saturation temperature. Nucleate boiling is predicted to be a significant mode of heat transfer in the lower portion of the BPG; $h'_{NB} = 5780 \rightarrow 820 \text{ W m}^{-2} \text{ K}^{-1}$ from $z = 0 \rightarrow 0.72 \text{ m}$, corresponding to 45% $\rightarrow$ 6% of local wall heat flux. For segments with negative heat transfer (condensation, $z > 1.16 \text{ m}$ in this case), the nucleate boiling contribution to Q_i is set to 0 and heat transfer is partitioned between phases as in saturated flow condensation (i.e. $T_L = T_{sat}$). The h'_{L0} portion of the Liu and Winterton (1991) model was employed in segments with condensation for consistency and because the mechanisms of convective flow evaporation and convective flow condensation are similar. Furthermore, the in-tube contribution to heat transfer resistance is minor (h'_{L0} ranges from 2710 – 3350 $\text{W m}^{-2} \text{ K}^{-1}$, $h_{CF} = 720 \text{ W m}^{-2} \text{ K}^{-1}$). Therefore, relatively large uncertainties in phase-change heat transfer coefficients in the upper portion of the BPG with relatively small temperature differences ($T_{CF} - T_{sat}$) would only have minor effects on overall predictions.

$$Q_{L,i} = \begin{cases} \min \left\{ \frac{[h'_{L0,i} (T_{tube,i} - T_{L,i})]^2}{Q_i^2}, m_{L,i+1} i_{L,i+1} - m_{L,i} i_{L,i} \right\} & Q_i \geq 0 \\ m_{L,i+1} i_L (T_{sat,i+1}, \psi_{i+1}) - m_{L,i} i_{L,i} & Q_i < 0 \end{cases} \quad (9)$$

Predicted coupling fluid, liquid, and saturation temperature profiles are presented for the representative case in Figure 5. Near the BPG inlet, the bulk solution temperature (T_L) is significantly below the local saturation temperature $T_{sat} = T_V$ (i.e., subcooled flow boiling). The $\text{NH}_3\text{-NaSCN}$ mixture transitions to saturated flow at $z = 0.15 \text{ m}$. Condensation is predicted to occur from $z = 1.16 - 1.6 \text{ m}$ (the separator portion is assumed adiabatic). The CF temperature approaches that of the working fluid flow around the transition to flow condensation.

Figure 5 – Developing solution, vapor, and coupling fluid temperature profiles along the length of the bubble-pump generator

A momentum balance is applied over each segment of the $\text{NH}_3\text{-NaSCN}$ channel, accounting for hydrostatic pressure drop, fluid acceleration, and dynamic (including frictional) pressure changes.

$$p_i - p_{i+1} = \Delta p_{\text{hs},i} + \Delta p_{\text{d},i} + \Delta p_{\text{acc},i} \quad (10)$$

The hydrostatic contributions (Δp_{hs}) are evaluated based on the average void fraction in each segment.

$$\Delta p_{\text{hs},i} = [\alpha_i \rho_v + (1 - \alpha_i) \rho_L] g \Delta z_i \quad (11)$$

The hydrostatic contributions to pressure drop are found to be dominant in this study accounting for 4540 Pa ($\Delta p_{\text{net}} = 4590$ Pa).

Fluid acceleration pressure changes are evaluated based on the local void fraction and phasic velocities ($U_L = j_L/(1 - \alpha)$, $U_V = j_V/\alpha$), as in Rattner and Garimella (2015a). The net contribution of fluid acceleration to total pressure drop is relatively minor for this case (41 Pa).

A detailed two-phase flow sub-model is employed to determine local void fraction and dynamic pressure changes for local flow rates, channel geometry, and fluid properties. The model accounts for the transition from bubbly to Taylor flow and for the relatively complex flow phenomena that occur in the intermediate Bond number regime ($5 < Bo < 40$, $Bo = 10$ for the representative case). This model is based on the formulation developed by Rattner and Garimella (2015a). The sub-model is described in detail in Appendix B.

For the representative case, the transition from bubbly to Taylor flow is predicted to occur at $z = 0.07$ m. For this particular case, net frictional flow resistance is quite small (13 Pa). At the outlet of the BPG, the Taylor-bubble region frictional pressure gradient is $\nabla p_b = -875 \text{ Pa m}^{-1}$, the liquid slug frictional pressure gradient is $\nabla p_s = 861 \text{ Pa m}^{-1}$, and the bubble-to-slug flow transition pressure gradient is $\nabla p_{\text{trans}} = 618 \text{ Pa m}^{-1}$ (84 Pa per flow transition, 0.14 m long unit cells). At the outlet, the relative bubble length is $\beta = L_b/(L_b + L_s) = 0.853$. Thus the outlet dynamic pressure change gradient is:

$$\begin{aligned} \nabla p_d(z = 1.60 \text{ m}) &= \beta \nabla p_b + (1 - \beta) \nabla p_s + \frac{\Delta p_{\text{trans}}}{L_b + L_s} = \\ &0.853(-875 \text{ Pa m}^{-1}) + (1 - 0.853)(861 \text{ Pa m}^{-1}) + \frac{84 \text{ Pa}}{0.116 \text{ m} + 0.020 \text{ m}} = -1.7 \text{ Pa m}^{-1} \end{aligned} \quad (12)$$

Based on the large magnitudes of each term in Equation 12, the low overall dynamic frictional pressure change magnitude is specific to this particular case. For other flow conditions, ∇p_d may constitute a significant portion of the overall pressure drop.

Figure 6 – Predicted bubble-pump generator solution flow rates at varying operating conditions (for $S_r = 0.275$, $T_{L,in} = 64.0^\circ\text{C}$, $p_{sys} = 1.15\text{ MPa}$, $\psi_{DS} = 0.538$, $T_{amb} = 24.0^\circ\text{C}$)

For the representative case, a driving pressure difference of 4590 Pa is predicted to yield liquid and refrigerant flow rates of 49 g min^{-1} (50.6 ml min^{-1}) and 4.7 g min^{-1} (676 ml min^{-1}), respectively. Assuming a reservoir solution temperature of 35°C^* and dilute solution concentration of 0.538 ($\rho_{L,res} = 960.9\text{ kg m}^{-3}$), this corresponds to a submergence ratio of $S_r = 0.275$. This provides sufficient excess height between the reservoir liquid level and top of the BPG (1.26 m) for the condenser, evaporator, gas heat exchanger, and absorber. By varying the submergence ratio between 0.241 and 0.311, the liquid flow rate can be adjusted from 25 to 70 ml min^{-1} . Thus, the design considered here is expected to provide sufficient control of liquid flow rates for proper absorber operation. Predicted BPG solution flow rates at varying source temperatures (110 – 130°C) and heating rates are presented in Figure 6. Optimal heating rates for

* Typical reservoir temperature measured in the accompanying experimental study (Part II of this work)

maximum liquid pumping can be seen in this figure, as has also been reported in previous studies of conventional spot-heated BPGs (Delano, 1998). At higher heating rates, greater frictional resistance due to increased vapor volumetric flow limits solution pumping. Based on these results, the proposed design can provide sufficient liquid pumping rates for a wide range of DAR system operating conditions.

2.2 Condenser

The main challenge in designing a DAR condenser is achieving sufficient heat rejection to the surroundings ($Q_{\text{cond}} \sim i_{\text{LV,NH}_3} \times m_{\text{ref}}$) with only passive transport mechanisms: natural convection (air flow driven by the elevated condenser temperature), ambient air circulation convection (low velocity “background” air flow in the environment), and radiation. In conventional DAR systems, a serpentine finned tube condenser is employed (Chen *et al.*, 1996). A large condenser is required in the proposed system design because the operating pressure is relatively low ($p_{\text{sys}} \sim 1.15$ MPa) to accommodate low source temperatures ($T_{\text{BPG,CF,in}} = 110 - 130^\circ\text{C}$). This results in a low driving temperature difference for condenser cooling ($T_{\text{sat,NH}_3}(p_{\text{sys}}) = 29.5^\circ\text{C}$, $T_{\text{amb}} = 24^\circ\text{C}$).

The proposed serpentine finned tube condenser design is presented in Figure 7. The design has 9 rows of stainless steel tubes (ID = 10.9 mm, OD = 12.7 mm, $L_{\text{row}} = 0.51$ m, $L_{\text{tot}} = 4.57$ m). Tube rows are spaced by $p_{\text{row}} = 31.8$ mm. The tubes are enhanced with press-fit smooth rectangular fins with thickness 0.19 mm and height 27.5 mm. A relatively wide fin pitch is selected (4.2 mm, 236 fins m^{-1}). This limits heat transfer area (total external heat transfer area: 1.79 m^2), but is necessary to prevent excessive flow resistance, which can reduce overall cooling performance. The spacing is less than the optimal value predicted with the model of Bar-Cohen and Rohsenow (1984) (7.4 mm at fin temperature $T_f = 29.5$ C). However, the proposed design is

Figure 7 – Schematic and operating conditions for proposed condenser design

intended to take advantage of ambient air circulation in the enclosing ventilation hood, which augments natural-circulation-driven flow through the fin channels. The condenser is inclined slightly downward (towards the refrigerant outlet) so that condensate drains into the evaporator and acts a gas trap to prevent non-condensable helium ingress.

The condenser is modeled with three segments corresponding to the desuperheating, condensing, and subcooling portions of the component. For each section, a thermal resistance network is formed from the ammonia flow to the ambient. Segmental heat transfer is determined using the effectiveness-number of transfer units approach (ε -NTU) for cross-flow heat exchangers. Representative results are reported for operation with a system pressure of $p_{\text{sys}} = 1.15 \text{ MPa}$, refrigerant flow rate of $m_{\text{ref}} = 4.0 \text{ g min}^{-1}$, vapor inlet temperature of $T_1 = 90^\circ\text{C}$, ambient air temperature of 24°C , and an ambient airflow velocity across the fins of $U_{\text{amb}} = 0.12 \text{ ms}^{-1}$. This ambient air-flow is representative of background ventilation in an indoor operating environment or very low wind-speeds in an outdoor application.

In the first section, a vapor-side flow convection coefficient of $h_{\text{int},1} = 23 \text{ W m}^{-2} \text{ K}^{-1}$ is predicted ($\text{Re}_1 = 690$). Conduction through the circular tube wall results in a minimal heat transfer resistance of $0.0016 \text{ m K W}^{-1}$. Ambient circulation through the fin channels (assumed $U_{\text{amb}} = 0.12 \text{ m s}^{-1}$) results in a laminar channel flow heat transfer coefficient of $h_{\text{ext,forced}} = 22 \text{ W m}^{-2} \text{ K}^{-1}$ ($\text{Re}_{\text{ext}} = 55$). At the average fin temperature of $T_{f,1} = 27.5^\circ\text{C}$, natural convection through the fin channels is predicted to be minimal ($h_{\text{ext,nat},1} = 0.9 \text{ W m}^{-2} \text{ K}^{-1}$ using the correlation of Elenbaas (1942)). This result ($h_{\text{ext,conv},1} \sim h_{\text{ext,forced}}$) indicates that, even at very low ambient air-flow rates (0.12 m s^{-1}), ambient air circulation convection is the dominant mode of heat rejection at the low driving temperature difference ($T_{\text{sat,NH}_3}(p_{\text{sys}}) - T_{\text{amb}} = 5.5 \text{ K}$).

Radiation heat transfer is modeled assuming that the rectangular fin array acts as a black body. The total heat transfer rate from the desuperheating section is evaluated from the bounding surface at the average fin temperature ($Q_{\text{rad},1} = 0.26 \text{ W}$). The effective radiation heat transfer coefficient is determined by averaging over the total external finned surface area ($A_{\text{net},1} = 0.073 \text{ m}^2$), yielding $h_{\text{eff,rad},1} = 1.0 \text{ W m}^{-2} \text{ K}^{-1}$.

Finally, the combined convection and radiation external heat transfer coefficient is obtained as: $h_{\text{ext},1} = h_{\text{ext,conv},1} + h_{\text{eff,rad},1} = 22.7 \text{ W m}^{-2} \text{ K}^{-1}$. Assuming a uniform value of $h_{\text{ext},1}$, the total finned surface efficiency is determined to be $\eta_{o,1} = 0.95$. The net lengthwise thermal resistance from the refrigerant to ambient is $R'_{\text{net},1} = 1.41 \text{ mKW}^{-1}$. Finally, the desuperheating section length, heat transfer, and average fin temperature are solved implicitly as follows:

$$Q_1 = m_{\text{ref}} c_{p,\text{NH}_3} \varepsilon_1 (T_1 - T_{\text{amb}}) \quad (13)$$

$$\varepsilon_1 = f \left(\text{NTU}, \frac{m_{\text{ref}} c_{p,\text{NH}_3}}{m_{\text{air}} c_{p,\text{air}}} \right) \quad (14)$$

$$NTU_1 = \frac{L_1 / R'_{\text{net},1}}{m_{\text{ref}} c_{p,\text{NH}_3}} \quad (15)$$

$$Q_1 = m_{\text{ref}} c_{p,\text{NH}_3} \left[T_1 - T_{\text{sat},\text{NH}_3} (p_{\text{sys}}) \right] \quad (16)$$

$$Q_1 = A_{\text{tot},1} h_{\text{ext},1} (T_{f,1} - T_{\text{amb}}) \quad (17)$$

Here, the effectiveness (ε_1) is evaluated using results for cross-flow heat transfer. For the desuperheating heat transfer of 5.9 W, the required heat exchanger length is $L_1 = 0.19$ m.

Similar approaches are employed to model heat transfer in the condensing and subcooling segments of the condenser. In the condensing section, the internal heat transfer coefficient is evaluated using the correlation of Akers *et al.* (1958) for flow in horizontal tubes at an average quality of 0.5 ($h_{\text{int},2} = 770 \text{ W m}^{-2} \text{ K}^{-1}$). The condensing section accounts for the major portion of the component length (4.1 m) and heat transfer ($Q_2 = 76.4 \text{ W}$). Internal convective heat transfer in the final subcooling segment ($L_3 = 0.26$ m) is modeled assuming laminar liquid flow ($h_{\text{int},3} = 160 \text{ W m}^{-2} \text{ K}^{-1}$). The ammonia stream is predicted to nearly approach the ambient temperature ($T_{\text{out}} = 24.1^\circ\text{C}$, $T_{\text{amb}} = 24.0^\circ\text{C}$, $Q_3 = 1.7 \text{ W}$). The overall condenser heat transfer is 84.1 W.

Condenser outlet temperatures for varying refrigerant flow rates and ambient air velocities are presented in Figure 8 for $T_{\text{cond},\text{in}} = 90^\circ\text{C}$ and $p_{\text{sys}} = 1.15 \text{ MPa}$. For the anticipated ranges of refrigerant flow rates ($3 - 5 \text{ g min}^{-1}$), ambient air velocities of only $\sim 0.1 - 0.2 \text{ m s}^{-1}$ are sufficient for complete condensation. Refrigerant outlet temperatures approach T_{amb} at higher ambient air velocities (liquid subcooling). This indicates that the proposed DAR system design can function at low operating pressures ($p_{\text{sys}} = 1.15 \text{ MPa}$) provided that some minimal ambient circulation is present.

Figure 8 – Predicted condenser outlet temperatures for varying refrigerant flow rates and ambient air velocities ($T_{cond,in} = 90^{\circ}\text{C}$, $p_{sys} = 1.15$ MPa)

If no ambient circulation is present ($U_{amb} = 0$) then the system pressure must be increased to 1.58 MPa to fully condense the refrigerant through natural convection and radiation heat transfer ($T_{sat,NH_3}(1.58 \text{ MPa}) = 40.6^{\circ}\text{C}$, $T_{amb} = 24.0^{\circ}\text{C}$). Under such stagnant air-side conditions, DAR system operation may also cause an ambient air temperature increase near the condenser, resulting in even greater required system pressures. For the baseline system operating conditions ($\psi_{DS} = 0.538$), this would necessitate a ~ 13 K increase in the BPG source temperature for equivalent operation ($T_{sat,NH_3-NaSCN}(p = 1.15 \text{ MPa}, \psi=0.538) = 73.0^{\circ}\text{C}$, $T_{sat,NH_3-NaSCN}(p = 1.58 \text{ MPa}, \psi=0.538) = 86.4^{\circ}\text{C}$).

2.3 Evaporator

The evaporator component in the DAR cycle delivers cooling to the conditioned space through evaporative mass transfer of liquid refrigerant to a low-vapor-pressure auxiliary gas stream. This mode of operation differs significantly from conventional vapor-compression and two-pressure absorption system evaporators, in which refrigerant flashes to a two-phase state

Figure 9 – a. Schematic of evaporator design, b. Segmented model discretization scheme

across a high-pressure-drop expansion device and continues to evaporate at the saturation temperature along the length of the component. Thus, DAR evaporator performance may be limited by both heat transfer resistance and interface-to-gas mass transfer resistance. Effective component design must account for both of these factors.

A tube-in-tube evaporator design is considered in this investigation (Figure 9a). Refrigerant (NH_3) and auxiliary gas (NH_3 -He mixture) flow vertically downwards in an inner tube ($ID_i = 17.3$ mm, $OD_i = 19.1$ mm, $L = 0.30$ m). The liquid refrigerant is assumed to flow as a falling film on the tube wall. Coupling fluid (CF) water flows vertically upward (counter-flow) in a narrow outer annulus ($ID_o = 20.9$ mm). In actual applications, the evaporator would be directly air coupled (i.e., installed inside the conditioned space). Forced liquid coupling is employed here to enable more accurate experimental component performance measurement (see Part II). Results are reported for a representative case for which operating and inlet conditions are summarized in Table 1.

At the representative operating conditions, the refrigerant flow rate may be too low to ensure complete liquid film wetting of the inner tube. Similar considerations were highlighted in the

Table 1 – Representative evaporator operating conditions

System pressure	p_{sys}	1.15	MPa
Refrigerant inlet flow rate	$m_{\text{L,in}}$	4.0	g min^{-1}
Refrigerant inlet temperature	$T_{\text{L,in}}$	25.0	$^{\circ}\text{C}$
Auxiliary gas inlet flow rate	$m_{\text{G,in}}$	20.0	g min^{-1}
Auxiliary gas inlet flow temperature	$T_{\text{G,in}}$	21.0	$^{\circ}\text{C}$
Auxiliary gas inlet helium mass fraction	ζ_{G}	0.30	-
Coupling fluid flow rate	m_{CF}	80.0	g min^{-1}
Coupling fluid inlet temperature	$T_{\text{CF,in}}$	5.0	$^{\circ}\text{C}$

design study of Jakob *et al.* (2008). Based on the model of El-Genk and Saber (2001), a minimum refrigerant flow rate of 26.9 gmin^{-1} ($\sim 7 \times m_{\text{L,in}}$) is required to transition from the rivulet to continuous film flow mode (assuming a contact angle of 30°). Thus, a wicking cloth sleeve (representative thickness 0.7 mm) would be installed inside the evaporator to improve film distribution on the tube wall. It is assumed that this sleeve enforces an effective film thickness of $\delta_f = 0.7 \text{ mm}$, rather than the much thinner value that would be predicted with Nusselt film theory ($\delta_f = 0.05 \text{ mm}$ at the inlet).

An axially segmented model ($n_{\text{seg}} = 25$, $\Delta z = 12 \text{ mm}$) is employed to describe the varying properties and transport along the length of the evaporator component. Flow properties are stored at the inlet and outlet nodes of each segment for the gas mixture, liquid refrigerant, and CF flows (Figure 9b). Liquid-to-gas interface properties are evaluated at the center of each segment. Mass, species, and energy balances are applied over each segment as appropriate.

The mass balances for each segment of the refrigerant and gas streams are defined as follows:

$$m_{\text{L},i+1} = m_{\text{L},i} - m_{\text{evap},i} \quad (18)$$

$$m_{G,i+1} = m_{G,i} + m_{\text{evap},i} \quad (19)$$

Here, m_{evap} is the rate of evaporation in each of the 25 segments, which ranges between 0.04 and 0.20 g min⁻¹. The refrigerant liquid flow rate is predicted to decrease from 4.0 to 2.7 g min⁻¹ along the length of the component (32% evaporation).

Species balances are applied over each segment of the gas stream.

$$\xi_i m_{G,i} = \xi_{i+1} m_{G,i+1} \quad (20)$$

With the increasing vapor-phase ammonia concentration along the length of the component, the gas helium mass fraction decreases from 0.30 to 0.28 (helium mole fraction ξ^* decreases from 0.65 to 0.63).

Thermal resistance from the liquid-bulk to the interface is modeled assuming conduction transport through half the film thickness (similar to the Nusselt assumption): $h_{L,\text{int}} = k_L/(\delta_f/2) = 1510 \text{ W m}^{-2} \text{ K}^{-1}$. The same convection coefficient value is assumed for transport from the film bulk to the tube inner wall ($h_{L,\text{tube}} = 1510 \text{ W m}^{-2} \text{ K}^{-1}$). The effective conductivity of the liquid film may be slightly altered in practice because of the presence of the wicking material in the film. Thermal resistance from the gas phase to the interface is predicted assuming laminar-to-transitional flow ($\text{Re}_G \sim 2100$), resulting in a convective transport coefficient of: $h_{G,\text{int}} = 37 \text{ W m}^{-2} \text{ K}^{-1}$. Thermal conduction resistance through the stainless steel tube wall is minimal: $R''_{\text{tube}} = 6.3 \times 10^{-5} \text{ m}^2 \text{ KW}^{-1}$. Convection heat transfer in the coupling fluid (CF) channel is modeled assuming laminar annular flow ($\text{Re}_{\text{CF}} = 40$): $h_{\text{CF}} = 1970 \text{ W m}^{-2} \text{ K}^{-1}$. The net CF-to-liquid refrigerant resistance is:

$$R'_{CF,L} = \frac{1}{WF} \left\{ \frac{1}{\pi ID_i h_{L,tube}} + \frac{\ln(OD_i/ID_o)}{2\pi k_{tube}} + \frac{1}{\pi OD_i h_{CF}} \right\} = 0.022 \text{ m W K}^{-1} \quad (21)$$

Here, WF is the wetting factor, which represents the portion of the tube perimeter effectively wetted by refrigerant film in each segment. A value of $WF = 1$ is employed in the baseline case, but this parameter can be varied to simulate the effect of incomplete liquid distribution.

Gas phase thermal resistance is greater than that of the refrigerant liquid film or CF. However, this does not directly limit evaporator performance because the cooling heat-transfer path is from the CF to the liquid-gas interface. In cases where incomplete film wetting occurs, direct heat transfer could occur between the gas and CF (through the dry portion of the wall). This secondary transport path is not considered in this model.

The interface energy balance for each segment is defined as:

$$m_{\text{evap},i} i_{LV,NH_3}(T_{\text{int},i}) = [\pi (ID_i - 2\delta_f) \Delta z WF] [h_{L,\text{int}}(T_{L,i+1} - T_{\text{int},i}) + h_{G,\text{int}}(T_{G,i+1} - T_{\text{int},i})] \quad (22)$$

Driving temperature differences for heat transfer in each segment i are evaluated using downwind nodal values (i.e., $T_{L,i+1} - T_{\text{int},i}$, $T_{G,i+1} - T_{\text{int},i}$) for improved numerical stability. Evaporative mass transfer is modeled assuming that liquid flows from the film to interface, and vapor flows from the interface to the gas bulk. Corresponding energy balances are applied for the liquid, gas, and coupling fluid streams.

$$m_{L,i} i_{L,i} - m_{L,i+1} i_{L,i+1} - m_{\text{evap},i} i_{L,\text{int},i} = WF \left\{ \begin{aligned} & [\pi (ID_i - 2\delta_f) \Delta z] [h_{L,\text{int}}(T_{L,i+1} - T_{\text{int},i})] \\ & + \frac{T_{L,i+1} - (T_{CF,i} + T_{CF,i+1})/2}{R'_{CF,L}/\Delta z} \end{aligned} \right\} \quad (23)$$

$$m_{G,i} i_{G,i} - m_{G,i+1} i_{G,i+1} + m_{\text{evap},i} i_{V,\text{int},i} = WF \left\{ [\pi (ID_i - 2\delta_f) \Delta z] [h_{G,\text{int}}(T_{L,i+1} - T_{\text{int},i})] \right\} \quad (24)$$

$$m_{CF} C_{p,CF} (T_{CF,i+1} - T_{CF,i}) = WF \left\{ \frac{(T_{CF,i} + T_{CF,i+1})/2 - T_{L,i+1}}{R'_{CF,L}/\Delta z} \right\} \quad (25)$$

The CF is counter-flow to the refrigerant and gas mix, thus the inlet to CF segment i is node $i+1$. In the discretized energy balance above, no working fluid-to-CF heat transfer is considered in the top segment ($i = 1$). This assumption is made to account for premixing of refrigerant and gas at the evaporator working fluid inlet. As indicated above, this evaporator design model does not account for heat transfer directly from the CF to gas stream (bypassing the refrigerant film) if $WF < 1$. This is expected to only lead to minor changes in evaporator operation because $R''_{G,int} (0.027 \text{ m}^2 \text{ K W}^{-1})$ is large compared to $R''_{int,tube} (0.001 \text{ m}^2 \text{ K W}^{-1})$.

At the developing laminar-to-transitional gas-phase conditions ($Re_G = 2100$, $Sc_G = 0.92$), the Sherwood number is predicted to be $Sh_G = 8.4$. For the NH_3 -He gas mixture, this corresponds to a convective mass transfer coefficient of $h^*_{G,int} = 0.0048 \text{ kg m}^{-2} \text{ s}^{-1} \text{ Pa}^{-1}$ (units of mass flux/vapor pressure difference). The segmental evaporative mass transfer rate is:

$$m_{evap,i} = WF h^*_{G,int} \left(\frac{MW_{NH_3}}{R^* T_{int,i}} \right) \left[\tau (ID_i - 2\delta_f) \Delta z \right] (p_{v,int,i} - p_{v,G,i+1}) \quad (26)$$

The resulting interface vapor pressure ($p_{v,int} = p_{sat,NH_3}(T_{int})$) varies from 573–445 kPa over the length of the component. The gas-phase vapor pressure is determined assuming ideal gas mixing ($p_{v,g} = (1 - \xi^*)p_{sys}$), and varies from 407–431 kPa along the length of the component. Thus, for this operating condition, the gas phase has the capacity to absorb additional refrigerant vapor and permit additional evaporative cooling if the component length is increased.

The evaporator mass transfer performance can be rated with an overall conductance: $UA^*_{evap} = m_{evap}/LMPD_{evap}$. Here, LMPD is the log-mean vapor pressure difference between the liquid

Figure 10 – Developing temperature profiles and evaporated refrigerant fraction in the evaporator for the representative case

refrigerant and gas streams, analogous to the more familiar log-mean temperature difference (LMTD). For the baseline operating conditions $\text{LMPD}_{\text{evap}} = 227 \text{ kPa}$, resulting in a conductance of $\text{UA}^*_{\text{evap}} = 9.4 \times 10^{-11} \text{ kgPa}^{-1} \text{ s}^{-1}$.

The segmental mass transfer equation (26) and energy equations (22 – 25) are solved simultaneously to determine the component performance. The resulting temperature profiles for the gas, interface, refrigerant, and CF are presented in Figure 10 for the representative case. For this case, the refrigerant evaporation proceeds rapidly near the inlet of the component, yielding a minimum liquid temperature of 1.5°C 60 mm from the inlet. The interface temperature tracks that of the liquid refrigerant, reaching a minimum value of 0.9°C 48 mm from the inlet. The liquid refrigerant and interface temperatures increase slightly toward the working-fluid outlet ($z = 0.30 \text{ m}$), where the heating counter-flow CF stream is warmer. The CF temperature approaches the minimum refrigerant temperature relatively closely ($T_{\text{CF,out}} = 2.5^\circ\text{C}$) due to the low interface-

to-CF thermal resistance. In the representative case, 14.1 W of cooling is delivered to the CF stream. Due to relatively poor gas-phase heat transfer, the gas bulk temperature only decreases from 15.0 – 9.8°C along the length of the component. Thus, even the relatively high gas flow rate ($m_{G,in}/m_{L,in} = 5$) only effects a minor parasitic cooling load ($m_{G,out} \times c_{p,G} \times (T_{G,in} - T_{G,out}) = 6.1$ W). In contrast, if the component were configured such that $T_{G,out} = T_{int,out}$, the gas cooling load would be 14.9 W. As such, DAR evaporators should be designed so that interface-to-CF (or conditioned space) heat transfer resistance is significantly less than interface-to-gas resistance.

A sensitivity study was conducted to identify the key factors that affect DAR evaporator performance. For these cases, the CF flow rate was adjusted to maintain the same temperature change along the length of the component ($5 \rightarrow 2.5^\circ\text{C}$). In this analysis, unless noted otherwise, all parameters are kept at the representative case conditions. Cooling capacity is particularly sensitive to the assumed wetting factor (WF). Varying WF from 1.00 – 0.53 results in decreasing cooling capacity, from 14.1 – 4.2 W (about 1 W loss per 0.05 reduction in WF). Furthermore, in actual implementations, film dryout may permit heat transfer from the warm gas to the tube, resulting in local CF heating.

Increasing the evaporator length from 0.15 to 1.0 m increases the evaporator cooling capacity from 5.8 to 37.2 W. For the longest case ($L = 1.0$ m), 64% of the refrigerant is evaporated, yielding significantly improved performance compared with the baseline case (32% evaporation).

Based on this result, the proposed evaporator design may be undersized. However, in practice, the evaporator length must be balanced with that of the gas heat exchanger (GHX), which fits between the bottom of the evaporator and top of the absorber. It is also important to consider gas-loop frictional pressure drop changes due to modifications to the evaporator and GHX

geometries. Furthermore, high evaporation rates may lead to film dryout and heating of the CF by the gas stream, as previously discussed.

Increasing the gas flow rate from $m_{G,in} = 6$ to 100 g min^{-1} increases cooling capacity from 5.8 to 22.0 W. However, only minor performance gains are predicted for $m_{G,in} > 50 \text{ g min}^{-1}$. Beyond this point, the evaporation process is not mass transfer limited and parasitic gas-phase cooling accounts for a greater portion of the evaporator capacity. The gas circulation is effected passively through natural convection. Therefore, greater flow rates may not be possible without excessive component and plumbing diameters to reduce frictional resistance. Additionally, the required GHX size may be impractical at high flow rates.

The evaporator performance is particularly sensitive to the gas phase inlet concentration ($\xi_{G,in}$). Increasing the inlet concentration from 0.285 to 0.400 ($p_v = 427 \rightarrow 300 \text{ kPa}$) increases the cooling capacity from 8.2 to 42.3 W. The highest concentration corresponds to a refrigerant evaporation fraction of 70%. This result suggests that the proposed evaporator design may, in fact, be adequately sized if the absorber is sufficiently effective. At 35°C and the baseline BPG CS concentration ($\psi_{CS} = 0.589$, Section 2.1), the solution vapor pressure is 269 kPa. Thus, if the absorber can achieve a gas-to-solution closest approach vapor pressure of $CAVP_{abs} \sim 30 \text{ kPa}$, then high evaporator performance (70% evaporation) can be achieved with the relatively small ($L = 0.3 \text{ m}$) proposed design.

2.4 Absorber

The function of the DAR absorber is to absorb vapor refrigerant from the passively circulating gas stream. This regenerates the refrigerant capacity of the auxiliary gas, enabling evaporative cooling in the evaporator. All passively cooled DAR systems reported in the literature have employed single serpentine tube absorbers (Figure 11a). In this design, liquid and

Figure 11 – a. Conventional serpentine-tube DAR absorber configuration, b. Proposed enhanced absorber design

vapor contact in a counter-flow fashion through a long coiled tube. This approach ensures effective liquid-vapor distribution as there is only a single fluid flow path. However, it results in limited interfacial area with internal stratified or rivulet flow and high pressure drops due to the long channels.

In this investigation, an alternate absorber design is proposed to address these limitations (Figure 11b). In this configuration, rising gas and downward flowing liquid contact in a counter-flow fashion in multiple parallel absorber tubes. This enables greater interface area – potentially up to the entire internal tube areas if film wetting is uniform. The multiple parallel short flow paths lead to reduced gas-phase pressure drop. Additionally, the number of absorber tubes can be increased for larger capacity implementations. However, solution and gas maldistribution can limit performance of a multi-tube absorber, and is analyzed later in this section. In the accompanying experimental study (Part II), a “feed-wire” distributor design is employed to wick solution into the absorber tubes from a shared reservoir, reducing the potential for

Figure 12 – Absorber model schematic and inlet conditions

maldistribution. Individual tubes are longitudinally finned in the same fashion as the solution subcooler, improving external heat transfer.

A representative design is considered here with ten parallel vertical absorber tubes ($ID = 17.3$ mm, $OD = 19.1$ mm, $L = 0.35$ m). Each tube is wrapped with 12 $H_{fin} = 25$ mm tall, $\delta_{fin} = 0.35$ mm thick, smooth corrugated aluminum fins (total external heat transfer area: 2.33 m^2). To improve internal film wetting, a helical wire insert would be installed in the tubes. This would enforce a film thickness similar to the wire diameter ($\delta_f = 0.6$ mm assumed here). Results are reported for a representative case with $p_{sys} = 1.15$ MPa, $T_{amb} = 25^\circ\text{C}$, solution inlet conditions of: $m_{L,in} = 50 \text{ g min}^{-1}$, $\psi_{L,in} = 0.581$, $T_{L,in} = 35^\circ\text{C}$, and gas inlet conditions of: $m_{G,in} = 22 \text{ g min}^{-1}$, $\xi_{G,in} = 0.28$, $T_{G,in} = 20^\circ\text{C}$. The model is formulated in a segmented fashion ($N_{seg} = 25$, equal length segments), with liquid- and gas-phase state points specified at the ends of each segment ($i = 1$ to $N_{segs}+1$) and interface conditions specified at the center of each segment ($i = 1$ to N_{segs}) (Figure 12).

Mass balances are applied over each segment of the liquid and gas subdomains.

Figure 13 – Profiles of liquid-phase NaSCN concentration and gas-phase NH₃ vapor pressure

$$m_{L,i+1} - m_{L,i} = m_{abs,i} \quad (27)$$

$$m_{G,i+1} - m_{G,i} = m_{abs,i} \quad (28)$$

For the representative case, the per-tube liquid flow rate increases from 5.00 – 5.19 g min⁻¹ along the length of the absorber. This corresponds to a per-tube gas phase flow reduction from 2.20 – 2.01 g min⁻¹.

Species balances are also applied for the solution NaSCN and gas-phase helium contents.

$$m_{L,i}\psi_i = m_{L,i+1}\psi_{i+1} \quad (29)$$

$$m_{G,i}\xi_i = m_{G,i+1}\xi_{i+1} \quad (30)$$

The resulting solution concentration decreases from the CS value ($\psi_{in} = 0.581$) to the DS value ($\psi_{out} = 0.559$). The gas stream helium mass concentration increases from 0.280 – 0.307. This corresponds to an ammonia vapor pressure reduction from 433–398 kPa. Profiles of solution concentration and gas vapor pressure at bulk and interface values are presented for this representative case in Figure 13.

Sensible heat transfer from the liquid and gas streams to the interface is predicted assuming annular falling film flow. The gas-phase flow is laminar ($Re_G \sim 190$), resulting in a gas-to-interface convection coefficient of $19 \text{ W m}^{-2} \text{ K}^{-1}$. Heat transfer from the liquid bulk to the interface is modeled as conduction through the liquid film ($h_{L,int} = 6870 \text{ W m}^{-2} \text{ K}^{-1}$). This value is also assumed for convection from the liquid bulk to the inner tube wall. As in the evaporator analysis, a wetting factor (WF) correction is implemented to permit modeling of incomplete film coverage of the tube inner surfaces. For the representative case, $WF = 1$ is assumed. Conduction resistance through the tube wall is minimal; $R'_{tube} = 1.0 \times 10^{-3} \text{ m K W}^{-1}$ for each tube.

Natural convection and radiation transport from the fins is evaluated at an average fin surface temperature ($T_{f,avg} = 31.8^\circ\text{C}$). The convection contribution is estimated by weighting the bounding vertical cylinder natural convection coefficient by the total external surface area ($A_{ext,out}$), resulting in $h_{ext,conv} = 1.0 \text{ W m}^{-2} \text{ K}^{-1}$. External radiation fin heat transfer is modeled assuming that the tall fins can be approximated as a black surface at the bounding cylinder, resulting in $h_{ext,rad} = 2.1 \text{ W m}^{-2} \text{ K}^{-1}$. The net external transport coefficient (weighted by the exterior area) is thus $3.1 \text{ W m}^{-2} \text{ K}^{-1}$, resulting in an exterior surface efficiency of 0.98. The net thermal resistance from the liquid bulk to the ambient is $R'_{L,amb} = 0.51 \text{ m K W}^{-1}$. Segmental energy balances for the liquid, interface, and gas are defined as follows.

$$m_{L,i} \dot{i}_{L,i} - m_{L,i+1} \dot{i}_{L,i+1} + m_{abs,i} h_{NH_3,L} (T_{int,i}) = WF \left[\frac{T_{L,i+1} - T_{int,i}}{R'_{L,int} / \Delta z_{seg}} - \frac{(T_{L,i} + T_{L,i+1})/2 - T_{amb}}{R'_{L,amb} / \Delta z_{seg}} \right] \quad (31)$$

$$m_{abs,i} h_{NH_3,LV} (T_{int,i}) = WF \left[\frac{T_{int,i} - T_{L,i+1}}{R'_{L,int} / \Delta z_{seg}} - \frac{T_{int,i} - (T_{G,i} + T_{G,i+1})/2}{R'_{G,int} / \Delta z_{seg}} \right] \quad (32)$$

Figure 14 – Temperature profiles of solution, interface, and gas in the absorber

$$m_{G,i+1}i_{G,i+1} - m_{G,i}i_{G,i} - m_{abs,i}h_{NH_3,V}(T_{int,i}) = WF \frac{(T_{G,i} + T_{G,i+1})/2 - T_{int,i}}{R'_{G,int}/\Delta z_{seg}} \quad (33)$$

This analysis assumes that vapor ammonia (Quality = 1) flows from the gas stream to the interface, and into the solution stream as liquid ammonia (Quality = 0). Sensible heat transfer from the gas-to-interface and liquid-to-ambient are evaluated at average segment temperatures. Downwinding is employed to improve stability for liquid-to-interface heat transfer calculations. The resulting temperature profiles for the liquid, interface, and gas are presented in Figure 14. The liquid bulk temperature effectively matches that of the interface along the absorber length. The liquid-to-ambient driving temperature difference for heat rejection varies from 3.8 – 11 K along the length of the component. The gas temperature approaches that of the interface near the top of the absorber ($\min(T_{int} - T_G) = 1.7$ K). Liquid and gas streams exit the component slightly above the ambient temperature (24°C) at 27.8°C and 31.2°C, respectively.

Finally, a mass transfer analysis is performed for overall model closure. Mass transfer calculations are evaluated in two parts for the gas-to-interface and interface-to-liquid transport. Gas-to-interface mass transfer is evaluated assuming laminar convection ($Sh_{int,G} = 4.3$, $R^{*'}_{G,int} = 1.5 \times 10^9 \text{ s}^{-1}$, units of mass flux/vapor pressure difference). Interface-to-liquid mass transfer is evaluated assuming diffusion through half the film thickness ($R^{*'}_{int,L} = 3.54 \times 10^5 \text{ kmol s kg}^{-1} \text{ m}^{-3}$, units of mass flux/molar density difference). The segmental absorption mass transfer rates are evaluated through implicit solution of the following equations:

$$p_{v,int,i} = p_{v,NH_3-NaSCN}(T_{int,i}, \psi_{int,i}) \quad (34)$$

$$m_{abs,i} = \frac{(p_{v,G,i} + p_{v,G,i+1})/2 - p_{v,int,i}}{R^{*'}_{int,G} / \Delta z_{seg}} \quad (35)$$

$$m_{abs,i} = \frac{\rho_{NH_3,int,i+1}^* - (\rho_{NH_3,L,i}^* + \rho_{NH_3,L,i+1}^*)/2}{R^{*'}_{int,L} / \Delta z_{seg}} \quad (36)$$

The resulting gas, interface, and solution vapor pressure/concentration profiles are presented in Figure 13. Overall, nearly constant driving potentials for mass transfer persist along the length of the component. The absorber closest approach vapor pressure (CAVP = $p_{v,G,out} - p_{v,NH_3NaSCN}(T_{L,in}, \psi_{L,in})$) is 111 kPa. In an ideal absorber, this value would approach 0 kPa. This CAVP can also be interpreted as a solution subcooling requirement for the absorber. With perfect liquid-to-vapor mass transport (CAVP = 0 kPa), the same absorber mass transfer performance could be realized with 14.5 K higher solution/ambient temperatures (i.e., $p_{v,NH_3NaSCN}(T_{L,in}, \psi_{L,in}) = p_{v,NH_3NaSCN}(T_{L,in} - 14.5 \text{ K}, \psi_{L,in}) + 111 \text{ kPa}$). Thus, the proposed absorber can be considered species-transport-limited. Gas-phase mass transfer resistance is relatively insignificant; if the gas-phase diffusivity is doubled, the total absorption rate only increases from 1.94–2.00 g min^{-1}

(+3%). Liquid phase resistance is much more significant. Doubling the liquid-phase diffusivity increases the absorption rate from $1.94 - 2.83 \text{ g min}^{-1}$ (+46%). The absorber performance can be characterized with an overall mass transfer conductance: $UA^*_{\text{abs}} = m_{\text{abs}}/\text{LMPD}_{\text{abs}}$. Here, LMPD is the log-mean vapor pressure difference between the solution and gas streams in the component. For the baseline operating conditions, $\text{LMPD}_{\text{abs}} = 135 \text{ kPa}$, resulting in a conductance of $UA^*_{\text{abs}} = 2.4 \times 10^{-10} \text{ kgPa}^{-1} \text{ s}^{-1}$.

The absorber model predicts a per-tube refrigerant absorption rate of 0.19 g min^{-1} (1.9 g min^{-1} total). This corresponds to a full absorber heat-rejection rate of 42 W . At this absorption rate, a maximum steady-state evaporator cooling capacity of $h_{\text{LV,NH3}} \times m_{\text{abs}} = 41 \text{ W}$ can be achieved (not accounting for parasitic cooling). Absorber and evaporator performance are tightly coupled, and it is difficult to independently identify which is the more significant limitation on cycle performance. In Section 3, a parametric study is performed to quantify the effects of evaporator and absorber performance on overall cycle operation.

A sensitivity analysis was conducted to identify key factors affecting the performance of the proposed absorber design. Reducing the wetting factor (WF – effective fraction of tube area wetted by the liquid film) from 1.0 to 0.2 leads to a nearly linear reduction in absorption capacity from 1.9 to 0.5 g min^{-1} . Thus, implementation of wetting enhancement techniques, such as the proposed helical coil inserts, may be critical to achieving adequate component performance.

Varying the per-tube solution inlet flow rate has minimal effects for $m_{\text{L,in,tube}} = 2.5 - 5.0 \text{ gmin}^{-1}$, resulting in absorption rates of $m_{\text{abs,tube}} = 0.184 - 0.194 \text{ gmin}^{-1}$. However, reduction to $m_{\text{L,in,tube}} = 0.5 \text{ gmin}^{-1}$ significantly limits absorption ($m_{\text{abs,tube}} = 0.106 \text{ gmin}^{-1}$). This analysis indicates that uneven liquid inlet distribution may result in effectively inactive tubes. One

challenge with the proposed absorber design is thus to design a distributor header or drip-tray that delivers similar liquid flow rates to all tubes.

Absorption rates are predicted to vary approximately linearly with solution inlet concentration, increasing from $0.61 - 2.24 \text{ g min}^{-1}$ for $\psi_{L,in} = 0.51 - 0.60$. Thus, slightly improved absorber performance could be achieved by increasing the NaSCN charge in experimental systems. However, it is important to ensure that the crystallization concentration is not exceeded ($\psi_{crsyt}(25^{\circ}\text{C}) = 0.645$). Additionally, increased solution concentrations may necessitate higher source temperatures for BPG operation. These results again highlight the highly coupled nature of DAR systems. Practical factors and effects on all other components must be considered when adjusting design or operating conditions.

2.5 Gas loop passive circulation

The auxiliary gas stream circulates through the gas loop, which is comprised of the evaporator, gas heat exchanger (GHX), and absorber. Flow is driven by hydrostatic pressure differences between the cold sinking leg and warm rising leg. In this analysis, the gas circulation rate is predicted by applying simplified momentum balances over each component in the gas loop. The considered component geometries and operating conditions are the same as those analyzed in the preceding sections. The GHX is modeled as a vertical shell-and-tube heat exchanger. No baffles are employed on the shell side to reduce pressure drop in the passive circulation loop. The cool gas stream flows downwards through seven parallel circular tubes ($ID_i = 10.9 \text{ mm}$, $OD_i = 12.7 \text{ mm}$, $H = 0.45 \text{ m}$). The warm auxiliary gas stream rises (counter-flow) through the shell side of the GHX ($ID_o = 52.5 \text{ mm}$). A summary of the gas loop passive circulation analysis is presented in Figure 15.

Figure 15 – Schematic of gas loop passive circulation analysis

The analysis begins in the evaporator, where cold refrigerant-gas mixture ($m = 18.7 \text{ g min}^{-1}$, $\xi = 0.268$, $T = 3^\circ\text{C}$) flows downwards. The temperature of this mixture varies as it continues through the cold leg of the gas loop. A pressure increase of 30.7 Pa is predicted from the evaporator inlet to the absorber gas inlet, accounting for hydrostatic pressure gain (32.6 Pa), channel flow losses (1.6 Pa), and minor losses due to expansions, contractions, and tube bends (0.3 Pa).

The gas stream is assumed to rise through the absorber at 30°C and intermediate flow rate and concentration ($m = 17.7 \text{ g min}^{-1}$, $\xi = 0.284$), and exit the absorber at the auxiliary gas conditions ($m = 16.7 \text{ g min}^{-1}$, $\xi = 0.300$). Rising flow through the warm-leg components of the gas loop is predicted to result in 29.6 Pa hydrostatic losses, 1.0 Pa frictional channel flow losses, and 0.1 Pa minor losses (30.7 Pa net).

Overall, the proposed model predicts a net hydrostatic pressure difference of 3.0 Pa between the sinking cold leg and rising warm leg of the gas loop. This results in an auxiliary gas flow of 16.7 gmin⁻¹, which causes the balancing frictional flow resistance. Based on this analysis, gas density changes due to temperature as well as helium concentration (ξ_G) both contribute significantly to motive forces in the gas loop. At the considered operating conditions, the average ~10 K difference between the cold and warm legs of the gas loop results in a 4% change in fluid density. Similarly, the helium concentration swing (0.268 – 0.300) yields a further 6% change in fluid density. This gas loop model yields an “order of magnitude” estimate of gas flow rates. While a more detailed analysis could be performed, it is difficult to precisely predict minor losses from tube bends, fittings, heat exchanger headers, etc. *a priori* for an actual DAR system.

3. FULL SYSTEM ANALYSIS

A complete cycle level model is formulated to analyze the DAR system for low-source-temperature operation. Results from the individual component level analyses (Section 2) are incorporated for closure. Mass, species, and energy balances are applied to each component to obtain the state points indicated in Figure 16. UA-LMTD models are formulated for the single-phase heat exchanger components (solution heat exchanger (SHX), solution subcooler, gas heat exchanger (GHX)) as follows.

The SHX is assumed to be a tube-in-tube counter-flow heat exchanger, with the warm stream (H) flowing in the inner channel and the cool stream (C) flowing through the outer channel. The heat transfer rate from the warm stream to the cool stream is:

$$Q_{SHX} = UA_{SHX} \frac{(T_{SHX,H,in} - T_{SHX,C,out}) - (T_{SHX,H,out} - T_{SHX,C,in})}{\ln \left(\frac{T_{SHX,H,in} - T_{SHX,C,out}}{T_{SHX,H,out} - T_{SHX,C,in}} \right)} \quad (37)$$

Figure 16 – Schematic of full DAR cycle model. All reported heat transfer rates are overall component values.

Additionally, the outer channel flow loses heat to the ambient as:

$$Q_{SHX, Loss} = UA_{SHX, Loss} \frac{(T_{SHX, C, in} - T_{amb}) - (T_{SHX, C, out} - T_{amb})}{\ln \left(\frac{T_{SHX, C, in} - T_{amb}}{T_{SHX, C, out} - T_{amb}} \right)} \quad (38)$$

The solution subcooler precools the warm solution stream flowing from the SHX outlet to the absorber inlet by rejecting heat to the ambient. It is modeled as:

$$Q_{sub} = UA_{sub} \frac{(T_{sub, in} - T_{amb}) - (T_{sub, out} - T_{amb})}{\ln \left(\frac{T_{sub, in} - T_{amb}}{T_{sub, out} - T_{amb}} \right)} \quad (39)$$

The gas heat exchanger (GHX) is modeled as a counter-flow heat exchanger. The evaporator working-fluid outlet stream (C) may have the two phases exiting in inequilibrium with each other (see Section 2.3). However, for simplicity, heat transfer is modeled assuming no phase change, that the driving temperature differences are between the two gas streams (C and H), and that the cool stream (C) exits in thermal equilibrium ($T_{L,C,out} = T_{G,C,out}$).

$$Q_{GHX} = UA_{GHX} \frac{(T_{GHX,H,in} - T_{GHX,C,out}) - (T_{GHX,H,out} - T_{GHX,C,in})}{\ln \left(\frac{T_{GHX,H,in} - T_{GHX,C,out}}{T_{GHX,H,out} - T_{GHX,C,in}} \right)} \quad (40)$$

Additionally, the following closure models and parameters are applied:

- The condenser closest approach temperature is: $CAT_{cond} = T_{cond,out} - T_{amb} = 2 \text{ K}$
- The evaporator-to-CF stream (T_{cool}) closest approach temperature is: $CAT_{evap} = T_{cool} - T_{evap,out} = 2 \text{ K}$
- Unevaporated refrigerant liquid from the evaporator flows through the GHX without additional evaporation. The stream mixes adiabatically with the absorber solution outlet stream
- The evaporator mass transfer conductance is: $UA^*_{evap} = 0.8 \times 10^{-10} \text{ kg Pa}^{-1} \text{ s}^{-1}$
- The gas flow exiting the evaporator is warmer than the outlet liquid refrigerant stream by a $CAT_{evap,G,L} = 5 \text{ K}$. This mixture is assumed to reach thermal equilibrium at the SHX outlet.
- The gas heat exchanger (GHX) overall conductance is: $UA_{GHX} = 2.0 \text{ W K}^{-1}$
- The solution heat exchanger (SHX) overall conductance is: $UA_{SHX} = 5.0 \text{ W K}^{-1}$
- The SHX heat loss conductance is: $UA_{SHX,amb} = 0.5 \text{ W K}^{-1}$

- The solution subcooler overall conductance is: $UA_{\text{sub}} = 1.6 \text{ WK}^{-1}$
- Absorber gas and liquid outlet closest approach temperatures are: $CAT_{\text{abs}} = T_{\text{abs,G,out}} - T_{\text{amb}} = T_{\text{abs,L,out}} - T_{\text{amb}} = 5 \text{ K}$
- The absorber mass transfer conductance is: $UA^*_{\text{abs}} = 1.6 \times 10^{-10} \text{ kg Pa}^{-1} \text{ s}^{-1}$

Representative cycle-level model results are presented in Figure 16 for operating conditions of: $\psi_{\text{DS}} = 0.538$, $p_{\text{sys}} = 1.15 \text{ MPa}$, $Q_{\text{BPG}} = 115 \text{ W}$, $T_{\text{cool}} = 3^\circ\text{C}$, and $T_{\text{amb}} = 24^\circ\text{C}$. For these conditions, the refrigerant generation rate is 3.9 g min^{-1} . Concentrated solution and auxiliary gas flow rates are assumed to be 39.0 g min^{-1} ($10 \times m_{\text{ref}}$) and 19.5 g min^{-1} ($5 \times m_{\text{ref}}$), respectively. These values are representative of results from the first-principles based BPG and gas loop models formulated in Section 2.

At these conditions, the evaporator delivers 13.1 W of cooling to the coupling fluid at $T_{\text{cool}} = 3^\circ\text{C}$ ($\text{COP} = Q_{\text{evap}}/Q_{\text{BPG}} = 0.114$), which is sufficient for preservation of many temperature sensitive pharmaceuticals and foodstuffs ($\sim 5^\circ\text{C}$ required). The concentrated solution (CS) concentration is $\psi_{\text{CS}} = 0.592$, which is well below the crystallization point at the ambient temperature ($\psi_{\text{cryst}}(24^\circ\text{C}) = 0.646$). Thus, the system operates within working-fluid constraints. The BPG outlet temperature is predicted to be 88.2°C , which is comparable to results predicted in Section 2.1 for BPG coupling fluid inlet temperatures of $110 - 130^\circ\text{C}$. Thus, the proposed DAR system can deliver cooling suitable for refrigeration with low thermal input temperatures, meeting the proposed design objectives.

Figure 17 – Predicted DAR system performance at varying **a.** cooling delivery temperature (T_{cool}) and **b.** ambient temperature (T_{amb})

A sensitivity study was conducted to assess the performance of the investigated DAR system at varying operating conditions. For this analysis, baseline conditions and closure models (summarized above) are employed except where specified otherwise. The predicted system COP increases with cold delivery temperature, reaching 0.282 at representative A/C conditions ($T_{cool} = 15$ °C) (Figure 17a). The cooling delivery temperature can also be reduced slightly below the (water) freezing point, with a COP of 0.034 at $T_{cool} = -2$ °C. System performance decreases with increasing ambient temperature (Figure 17b). For these cases, the system pressure is varied such that $T_{sat,NH3}(p_{sys}) - T_{amb} = 5.5$ K (to ensure proper condenser operation). A COP of 0.054 can be maintained at $T_{amb} = 30$ °C. However, the BPG source temperature may also have to be increased ($T_{BPG,out} = 95.8$ °C) to ensure proper operation at these conditions.

In the preceding component-level analyses, transport rates in the evaporator and absorber components were highlighted as potential performance limitations. The two component mass transfer conductances (UA^*_{evap} , UA^*_{abs}) were thus varied to determine effects on overall system performance (Figure 18). Cooling capacity is predicted to sharply decrease with UA^*_{evap} . COP decreases to <0.05 for UA^*_{evap} values below $\sim 0.5 \times 10^{-10}$ kg Pa⁻¹ s⁻¹. Component performance

Figure 18 – Effects of absorber and evaporator mass transfer conductances on overall DAR system performance

tends to vary more smoothly with UA^*_{abs} values, indicating that evaporator performance may be a more severe limitation near baseline system conditions. If UA^*_{evap} and UA^*_{abs} can be increased to 1.6 and $2.4 \times 10^{-10} \text{ kg Pa}^{-1} \text{ s}^{-1}$, respectively, then the predicted system COP reaches 0.256. Thus, future system design investigations should focus on mass transfer enhancement of the evaporator and absorber.

The effects of relative auxiliary gas (m_{aux}/m_{ref}) and solution flow rates (m_{CS}/m_{ref}) are summarized in (Figure 19). An optimal auxiliary gas-to-refrigerant refrigerant flow rate ratio of 1.85 is identified for the baseline conditions ($COP = 0.162$). At higher m_{aux} , the parasitic gas cooling load in the evaporator limits overall performance. At lower flow rates, the gas stream becomes saturated with vapor refrigerant, limiting evaporation. DAR system performance is predicted to decrease with greater relative solution flow rates (m_{CS}/m_{ref}). This occurs because a

Figure 19 – Predicted DAR system performance at varying relative (a) auxiliary gas and (b) solution flow rates

larger portion of input heat (Q_{BPG}) is used for sensible solution heating, and the driving vapor pressure difference is reduced in the absorber. For the representative operating conditions, solution flow rates cannot be reduced below $\sim 5 \times m_{ref}$ due to NaSCN crystallization concerns. Additionally, absorber solution distribution and film wetting may be poor at low flow rates, limiting practical component performance.

4. CONCLUSIONS

In this investigation, detailed component models were formulated for all cycle components in a low-source-temperature ($T_{source} \leq 130^\circ\text{C}$), passively cooled DAR refrigeration ($T_{cool} \sim 5^\circ\text{C}$) system with NH_3 -NaSCN-He working fluids. This study presented the first detailed design models for the DAR coupling-fluid heat bubble-pump generator (with actual working fluids), condenser, evaporator, and passive circulation gas loop. A new, enhanced absorber design that addresses major limitations of conventional implementations was analyzed. Simplified descriptions of the component sub-models were integrated into a complete system-level model, which was applied to analyze performance over varying operating conditions.

The investigated DAR system was predicted to achieve a COP of 0.11 at baseline conditions, assuming complete film wetting in the evaporator and absorber. Parametric studies suggested that much higher COPs could be achieved at elevated cooling temperatures (~ 0.28 for A/C applications). Bubble-pump generator model results indicated that stable operation could be achieved with thermal input at $110 - 130^{\circ}\text{C}$, which could be delivered from low-grade sources such as solar collectors or engine exhaust. Low-velocity ambient air circulation ($\sim 0.1 \text{ m s}^{-1}$) was predicted to be necessary for proper condenser operation at design conditions. Significant system modifications or higher source temperatures may be required if no such flow is present.

An improved COP of 0.26 was predicted if evaporator and absorber mass transport conductances were significantly increased. These two parameters (UA^*_{evap} and UA^*_{abs}) were found to be the key factors limiting system performance. However, enhanced inter-phase transport may be difficult to achieve with the limited pressure drop budgets available in passive DAR systems. Incomplete film wetting or liquid distribution can dramatically decrease absorber and evaporator capacities. In practice, these may be more severe concerns at reduced solution and refrigerant flow rates. Thus, future investigations should focus on enhancement of these two components.

One limitation of the proposed DAR design is the need for relatively low ambient temperatures (here $T_{\text{amb}} = 24^{\circ}\text{C}$) to ensure proper operation of the passively cooled condenser and absorber. If higher source temperatures are available, then the system pressure can be increased, enabling heat rejection to higher ambient temperatures and an increased driving vapor pressure difference in the absorber.

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4 This investigation highlighted the strongly coupled nature of DAR cycle components. For
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6 example, reduced BPG solution pumping rates could enable increased refrigerant generation
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8 (reduced parasitic solution heating load). However, this could lead to poor liquid distribution and
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10 film wetting in the absorber, limiting cooling capacity. Similarly, the passive gas loop circulation
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12 analysis (Section 2.5) demonstrated how the behavior of the evaporator, gas heat exchanger,
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14 absorber, and plumbing connections affect gas circulation rates. Modifications to any individual
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16 component can thus dramatically affect overall system performance. By providing a set of
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18 detailed component level models, this investigation enables integrated system design, which
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20 accounts for such effects. Validation of this model was not discussed in this paper. In the
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22 accompanying Part II of this work, these models are assessed and validated with results from an
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24 experimental investigation of a DAR system.
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APPENDIX A: FLUID THERMODYNAMIC AND TRANSPORT PROPERTIES

Thermodynamic and transport properties for the ammonia-sodium thiocyanate ($\text{NH}_3\text{-NaSCN}$) mixture are determined using correlations from Infante Ferreira (1984). That study proposed correlations for the crystallization curve (ψ_{cryst}), vapor-pressure (p_v), liquid-phase enthalpy (i_L), liquid density (ρ_L), liquid viscosity (μ_L), and thermal conductivity (k_L). The correlation for liquid viscosity was fit to data for $-10^\circ\text{C} < T < 60^\circ\text{C}$. At higher temperatures (as in the BPG), the viscosity correlation predicts negative values. Thus a limiting condition was applied so that μ_L is not less than the pure ammonia value (Fenghour *et al.*, 1995). No surface tension data are available for the $\text{NH}_3\text{-NaSCN}$ mixture. Thus, values for the similar mixture $\text{NH}_3\text{-LiNO}_3$ are employed as an approximation (Zacarias *et al.*, 2013). Similarly, no binary diffusion data are available for the mixture ($D_{\text{ab,L}}$); therefore, the generalized Stokes-Einstein approach is employed for predicting this property.

Pure ammonia fluid properties (as in the BPG vapor phase or condenser) are evaluated using the model of Tillner-Roth *et al.* (1993) and transport property correlations included in the Engineering Equation Solver software (Klein, 2014). $\text{NH}_3\text{-He}$ mixture gas phase thermodynamic and transport properties (i_G , ρ_G , k_G , μ_G , $D_{\text{ab,G}}$) are evaluated by applying ideal gas mixing rules.

APPENDIX B: TWO-PHASE FLOW MODEL FOR BUBBLE-PUMP GENERATOR

The two-phase flow sub-model employed in the segmented bubble-pump generator (BPG) model (Section 2.1) is described here. This model takes, as inputs, the channel geometry (D), liquid and vapor axial flow rates (volumetric flow rates V_L and V_V , or equivalently, superficial velocities j_L and j_V), and thermophysical fluid properties. It returns the local void fraction (α) and hydrodynamic pressure gradient (∇p_d), yielding closure of the segmented momentum equation (Equation 10). The modeling approach employed here is specific to the two-phase flow characteristics expected in distributed heated BPGs in which the flow pattern transitions from single-phase liquid (inlet), to bubbly flow, to intermediate Bond number Taylor flow ($5 \lesssim Bo \lesssim 40$). The formulation presented here is based on that developed by Rattner and Garimella (2015a), which was validated for ambient-pressure steam-water BPG flow. Representative results are reported in this section for the NH_3 -NaSCN flow case discussed in Section 2.1.

Two-phase flow behavior depends strongly on flow regime. Therefore, the Taitel *et al.* (1980) bubbly-to-Taylor flow regime transition criterion is evaluated for each working fluid channel segment.

$$\text{IsBubbly}_i = \left\{ j_{L,i} - 2.34 j_{V,i} + 1.07 \left[\frac{\sigma g (\rho_L - \rho_V)}{\rho_L^2} \right]^{1/4} > 0 \right\} \quad (\text{A.1})$$

Using this model, the first two segments (67 mm) in the representative case are predicted to exhibit bubbly flow, with the rest in Taylor flow. In these bubbly flow segments, general purpose void fraction (Premoli *et al.*, 1971) and frictional pressure drop correlations (Friedel, 1979) are employed. At the end of the bubbly flow section in the representative case, the void fraction and

frictional pressure gradient reach 0.306 and 60 Pa m⁻¹, respectively. In the remaining segments, a mechanistic Taylor flow model is employed.

The intermediate Bond number Taylor flow model of Rattner and Garimella (2015a) is adopted here with minor modifications, and is summarized as follows. The model idealizes Taylor flow assuming repeating identical *unit cells*, each comprised of a liquid slug and cylindrical Taylor bubble. Continuity in each unit cell requires equal volumetric fluxes through cross-sections of the bubble and liquid slug portions. Assuming uniform velocity profiles in the liquid slug (U_s), Taylor bubble (U_b), and liquid film (U_f):

$$U_s = j = \alpha_b U_b + (1 - \alpha_b) U_f \quad (\text{A.2})$$

Here α_b is the void fraction in the portion of the unit cell from the tail to nose of the Taylor bubble. At the inner-tube outlet in the representative case, $\alpha_b = 0.902$, and the bubble diameter is 4.68 mm ($ID_i = 4.93$ mm). At this same point, the model predicts: $U_s = 0.63$ m s⁻¹, $U_b = 0.77$ m s⁻¹, and $U_f = -0.59$ m s⁻¹ (i.e., downward flow).

Additionally, continuity requires that the upward vapor flow rate in the portion of the unit cell containing the Taylor bubble ($\beta = L_b / (L_s + L_b)$) equals the total vapor flow rate.

$$j_v = \beta \alpha_b U_b \quad (\text{A.3})$$

The total unit-cell void fraction is thus:

$$\alpha = \beta \alpha_b \quad (\text{A.4})$$

In the representative model case, α and β increase along the working-fluid channel and reach outlet values of 0.770 and 0.853, respectively.

The Taylor bubble rise velocity model formulated by Rattner and Garimella (2015b) is adopted here. This model incorporates capillary-scale (Liu *et al.*, 2005) and large Bond number results (Bendiksen, 1985; White and Beardmore, 1962), and was validated for intermediate Bond number flows.

$$U_b = C_0 j + \Gamma \sqrt{gD} \quad (\text{A.5})$$

$$C_0 = f_{LS} C_{0,LS} + (1 - f_{LS}) C_{0,Ca} \quad (\text{A.6})$$

$$f_{LS} = \left(\frac{1}{1 + 4840 \text{Re}_j^{-0.163}} \right)^{0.816/\text{Bo}} \quad (\text{A.7})$$

$$C_{0,LS} = 1.20 + \frac{1.09}{1 + (\text{Re}_j/805)^4} \quad (\text{A.8})$$

$$C_{0,Ca} = \frac{1}{1 - 0.61 \text{Ca}^{0.33}} \quad (\text{A.9})$$

$$\Gamma = \begin{cases} 0.344 \left[1 - \exp\left(\frac{-0.01 N_f}{0.345}\right) \right] \sqrt{1 + \frac{20}{\text{Bo}} - \frac{93.7}{\text{Bo}^2} - \frac{676.5}{\text{Bo}^3} + \frac{2706}{\text{Bo}^4} \frac{1 - 0.96 \exp(-0.0165 \text{Bo})}{1 - 0.52 \exp(-0.0165 \text{Bo})}} & \text{Bo} > 4.55 \\ 0 & \text{Bo} \leq 4.55 \end{cases} \quad (\text{A.10})$$

$$\begin{aligned} \text{Re}_j &= \frac{\rho_L j D}{\mu_L} & \text{Bo} &= \frac{(\rho_L - \rho_v) g D^2}{\sigma} \\ \text{Ca} &= \frac{\mu_L j}{\sigma} & N_f &= \sqrt{\frac{\rho_L (\rho_L - \rho_v) g D^3}{\mu_L^2}} \end{aligned} \quad (\text{A.11})$$

Here Re_j is the liquid slug Reynolds number (42,180 at the outlet), Bo is the Bond number (10.7), Ca is the capillary number (0.0021 at the outlet), and N_f is a measure of relative gravitational-to-viscous forces (14,630). For the representative case, the predicted bubble rise

velocity increases from $0.191 - 0.770 \text{ m s}^{-1}$ over the length of the Taylor-flow portion of the BPG.

In most prior investigations of Taylor flow, liquid films have been modeled as free-falling flows with negligible interfacial stress (Sylvester, 1987; Thulasidas *et al.*, 1995). A refined film-flow model is adopted here, which accounts for the effects of interfacial shear. This correction is expected to be important in this study due to the high volumetric vapor fluxes ($j_V = 0.59 \text{ m s}^{-1}$ at the BPG outlet) and relatively high vapor-to-liquid viscosity ratio ($\mu_V/\mu_L = 0.17$).

$$U_f = \frac{4}{\pi [D^2 - (D - 2\delta_f)^2]} \int_{D/2 - \delta_f}^{D/2} (2\pi r) \left\{ \frac{g(\rho_L - \rho_G)}{4\mu_L} \left[\left(\frac{D}{2} \right)^2 - r^2 + 2 \left(\frac{D}{2} - \delta_f \right)^2 \ln \left(\frac{2r}{D} \right) \right] + \frac{\tau_{\text{int}} (D/2 - \delta_f)}{\mu_L} \ln \left(\frac{2r}{D} \right) \right\} dr \quad (\text{A.12})$$

$$f_b = \frac{4}{\text{Re}_b} \left\{ \frac{3.44}{\sqrt{L_b^*}} + \left[\frac{1.25}{4L_b^*} + 16 - \frac{3.44}{\sqrt{L_b^*}} \right] \sqrt{1 + 0.00021(L_b^*)^{-2}} \right\} \quad (\text{A.13})$$

$$\text{Re}_b = \frac{\rho_G (U_b - U_f) D_b}{\mu_G} \quad L_b^* = \frac{L_b}{\text{Re}_b D_b} \quad (\text{A.14})$$

$$\tau_{\text{int}} = \frac{f_b}{8} \rho_G (U_b - U_f)^2 \quad (\text{A.15})$$

A developing laminar flow model is employed to determine the bubble friction factor (Equations A.13 – A.15(Shah and London, 1978)). The laminar flow assumption is applied because $1680 < \text{Re}_b < 3570$ for this case, which corresponds to laminar-to-low-transitional channel flow. The liquid film is also modeled assuming laminar flow (Equation A.12). In the representative case $2520 < \text{Re}_f < 3910$, which is also in the laminar-to-transitional Reynolds number range ($\text{Re}_f = 4\rho_L U_f \delta_f / \mu_L$, $\text{Re}_{f,\text{trans}} \sim 3000$ (de Cachard and Delhaye, 1996)). The film

thickness (δ_f) and bubble region void fraction ($\alpha_b = (D - 2\delta_f)^2 / D^2$) are determined by substitution of the film mean velocity (Equation A.12) into the continuity constraint (Equation A.2). For the representative case, liquid-film thickness increases from 0.106 – 0.124 mm along the length of the Taylor flow section. The frictional pressure gradient due to film-to-wall shear around the Taylor bubbles is:

$$\tau_{\text{int}}(\pi D_b) - \tau_w(\pi D) = \frac{\pi}{4}(D^2 - D_b^2)(\rho_L - \rho_G)g \quad (\text{A.16})$$

$$\nabla p_{f,b} = \frac{4\tau_w}{D} \quad (\text{A.17})$$

This pressure gradient is negative because it acts in the bulk flow direction (upward), opposing the downward flowing film, and approaches -868 Pa m^{-1} at the BPG outlet.

Average Taylor bubble lengths in each segment are estimated using the correlation proposed by Rattner and Garimella (2015b) for intermediate Bond number flow.

$$\frac{L_b}{D} = 1.813 \left\{ \exp \left[1.815 - 0.866 \left(\frac{U_b}{j} \right)^{2.176} \right] \left(\frac{\alpha}{1-\alpha} \right)^{1.637} + 1 \right\} \quad (\text{A.18})$$

For the representative case, this predicts $L_b = 116 \text{ mm}$ ($24 \times D$) at the BPG outlet. At that point, $\beta = L_b / (L_b + L_s) = 0.853$; therefore, liquid slugs are predicted to only be $L_s = 20 \text{ mm}$ ($4 \times D$).

The liquid slug is usually found to be turbulent in this investigation. In the representative case, $\text{Re}_j = 42,190$ at the BPG outlet. Frictional resistance in the liquid slug is evaluated using the universal (laminar-to-turbulent) channel flow correlation of Churchill (1977).

$$f_s = 8 \left\{ \left(\frac{8}{\text{Re}_j} \right)^{12} + \left[\left(2.457 \ln \left(\frac{1}{(7/\text{Re}_j)^{0.9}} \right) \right)^{16} + \left(\frac{3750}{\text{Re}_j} \right)^{16} \right]^{-1.5} \right\}^{1/12} \quad (\text{A.19})$$

$$\nabla p_{f,s} = f_s \frac{\rho_L j^2}{2D} \quad (\text{A.20})$$

In the representative case, the liquid-slug frictional pressure gradient approaches 860 Pa m^{-1} at the working-fluid channel outlet. However, the net effect of this flow resistance is small, because the liquid slugs only account for a minor portion of the tube length.

The precise mechanisms of pressure loss at Taylor-bubble to liquid-slug transitions are not yet fully characterized for the intermediate Bond number regime. At the capillary-flow limit, surface tension forces are the major component of transitional pressure drop (Δp_{trans}) (Bretherton, 1961; Kreutzer *et al.*, 2005). For large Bond number flows, inertial effects dominate Δp_{trans} (Dukler and Hubbard, 1975; Fukano *et al.*, 1989). This model employs the empirical extended capillary scale transition pressure drop model proposed by Rattner (2015).

$$\Delta p_{\text{trans}} = \frac{1}{2} \frac{\rho_L j^2}{\text{Re}_j^{2/3} \text{Ca}^{1/3}} \left(10.88 + \frac{60}{1 + (\text{Re}_{j,L}/800)^{-2}} \right) \quad (\text{A.21})$$

$$\text{Re}_{j,L} = \frac{\rho_L j_L D}{\mu_L} \quad (\text{A.22})$$

In the representative case, the predicted flow transition pressure drop increases from 18 – 84 Pa over the length of the Taylor flow portion of the BPG. For this case, the total pressure drop contribution of Δp_{trans} is 1205 Pa, which is much greater than the liquid slug frictional contribution (225 Pa), and is of an order similar to the bubble region frictional resistance (-

Figure A.1 Schematic summary of mechanistic Taylor-flow model

1407Pa). Thus, accurate prediction of Δp_{trans} and the frequency of transitional pressure drops (i.e., $L_b + L_s$) are critical for modeling Taylor flow in the intermediate Bond number regime.

The liquid film, liquid slug, and transition pressure drops are combined to yield the hydrodynamic pressure drop in each segment:

$$\frac{\Delta p_{d,i}}{H_{\text{seg},i}} = \beta \nabla p_{f,b,i} + (1 - \beta) \nabla p_{f,s,i} + \frac{\beta}{L_b} \Delta p_{\text{trans}} \quad (\text{A.23})$$

The net hydrodynamic pressure drop gradient magnitude (∇p_d) is relatively small for the representative case: 208 Pa m⁻¹ at the BPG inlet to 4 Pa m⁻¹ at the outlet. As indicated in Section 2.1, this is specific to the representative case, for which relatively large magnitude pressure drops due to liquid film friction, liquid slug friction, and flow transitions cancel out. In other cases, the average ∇p_d magnitude could be much greater. A schematic summary of the developed Taylor flow model is presented in Figure A.1. The development of the pressure and void fraction along the BPG channel is presented in Figure A.2 for the representative case.

Figure A.2 – Pressure and void fraction profiles along the working-fluid channel for the representative BPG model case

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