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Zhiwei Huang
University Of Maryland, United States of America, zwhuang@terpmail.umd.edu

Ransisi Huang
University Of Maryland, United States of America, cchuang5@umd.edu

Jiazhen Ling
University of Maryland, College Park, United States of America, jiazhen@umd.edu

Vikrant Chandramohan Aute
University of Maryland College Park, vikrant@umd.edu

Yunho Hwang
yhhwang@umd.edu

Applicability of A Bifurcated Bare-tube Heat Exchanger in Water-based Hybrid VRF System

Zhiwei Huang, Ransisi Huang, Jiazhen Ling, Vikrant Aute, Yunho Hwang*

Center for Environmental Energy Engineering,
Department of Mechanical Engineering, University of Maryland
4164 Glenn Martin Hall Bldg., College Park, MD 20742, United States
* Corresponding Author: email: yhhwang@umd.edu

ABSTRACT

A water-based hybrid variable refrigerant flow (hVRF) system is a combination of traditional VRF system and water chiller system. Instead of using refrigerants in indoor units, hVRF system utilizes water (or any heat transfer fluid) as working fluid in indoor units. Compared with traditional VRF system, it has the advantages of reduced refrigerant charge, wider selection of refrigerants in terms of flammability and maintenance cost. However, the system COP of hVRF system is slightly lower than conventional VRF system due to single phase heat transfer in indoor coils. Therefore, current study proposes a bifurcated bare-tube heat exchanger (bBTHX) for indoor coils and investigates its applicability in hVRF system to increase the system efficiency. System model was developed for both traditional VRF system and the hVRF system. Results show that the bBTHX has 60% less total pumping power, 65% smaller volume and 70% smaller package- and material-volume than those of traditional fin-and-tube heat exchanger when delivering the same capacity and similar system COP. Simulation results also show that the system charge of hybrid VRF system with R-290 and R-600a are 28% and 27%, respectively lower than that of R-410A hybrid VRF system. Overall, the bBTHX shows a potential applicability as indoor coils for water-based hVRF systems with less refrigerant charge and flexibility of using flammable refrigerants.

1. INTRODUCTION

Variable refrigerant flow (VRF) system is a multiple-unit split type system. VRF system can achieve higher efficiency than traditional central air conditioning unit by modulating the refrigerant flow according to cooling and heating load of individual zone. VRF system with heat recovery function also has the potential of energy saving by internal heat recovery. As shown in Figure 1, VRF system is one type of vapor compression systems. It is usually made of variable capacity compressor, outdoor heat exchanger (could be air-cooled or water-cooled), indoor heat exchanger with variable speed fans, expansion device (usually EEV) and operation mode regulation unit (such as heat recovery unit). Other auxiliary components, such as accumulator and receiver can also be found in VRF systems. One of the drawbacks of VRF system is its high cost, including high upfront system cost due to system complexity and high maintenance cost due to the need of regular leakage check on indoor units inside the building.

Hybrid variable refrigerant flow (hVRF) system is a new type of VRF system that hasn't been widely investigated and applied. Figure 2 represents a typical hVRF system structure. Instead of running refrigerants in indoor units directly like traditional VRF system, the hVRF system runs water (or other type of coolants) in indoor units. The refrigerant exchanges heat with the water loop at a heat exchanger (usually a plate heat exchanger inside the heat recovery unit). The biggest advantage of utilizing the water loop is to eliminate usage of refrigerants inside the building so to reduce the concern of refrigerant flammability, widening the selection of refrigerants. Moreover, the refrigerant charge is lower than that of the traditional VRF system. Additional benefit of water loop is lower system maintenance cost because there is no need to do refrigerant leakage check for indoor units.

Takenake *et al.* (2017) conducted a field test and found though having all the benefits as stated above, the hVRF system has 10% lower COP than conventional VRF system under same operating conditions. This is mainly due to single phase heat transfer at indoor units. Then, they concluded that the indoor coil design is crucial to improve the system efficiency. Fin-and-tube heat exchangers (FTHX) are usually used as indoor coils to exchange heat between coolant and indoor air. However, the relatively low heat transfer coefficient on the water-side makes it a necessity to extend the heat transfer area, resulting in large indoor coil size. For example, the dimension and weight of water coil based indoor unit (250 × 790 × 700, 25kg) is 25% larger than R410A coil based indoor unit (250 × 790 × 700, 25kg) that have the same capacity from one manufacturer. In current study, authors proposed a novel heat exchanger design and applied it as indoor coil in order to enhance single phase heat transfer on water-side to solve the COP degradation problem. Meanwhile, as the heat transfer coefficient increases, the heat transfer area and heat exchanger size could be largely reduced. Additionally, the application of this new heat exchanger also makes it possible for fast onsite indoor unit manufacturing using 3D printing technology which reduces the logistics cost.

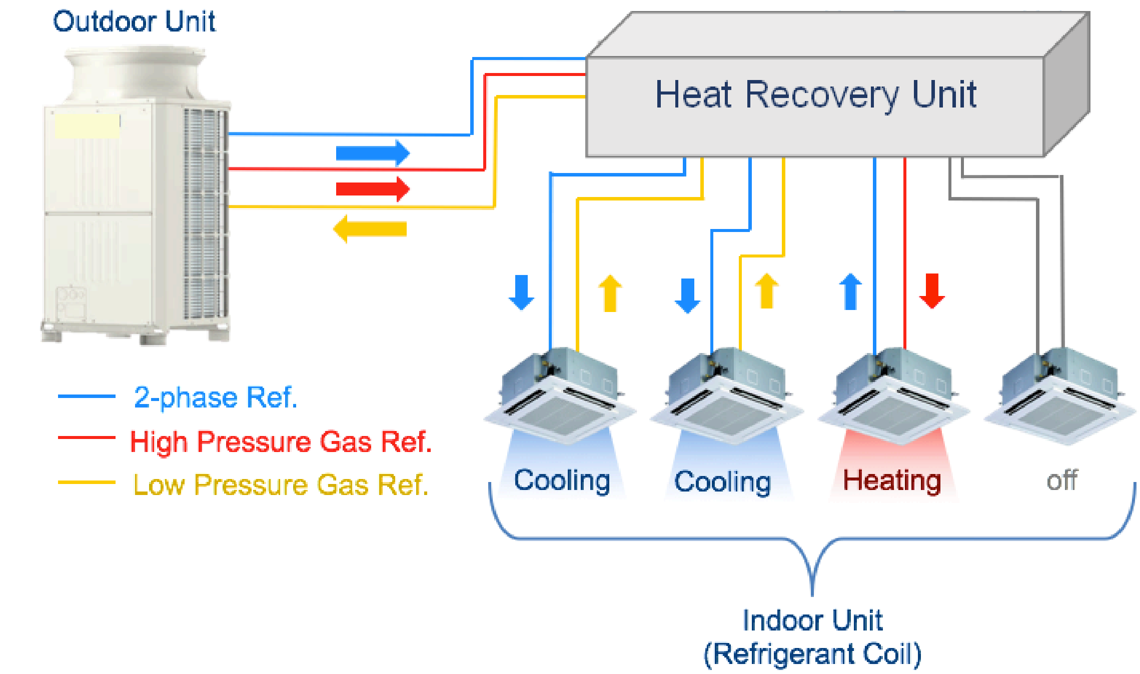

Figure 1: Traditional VRF system (with heat recovery)

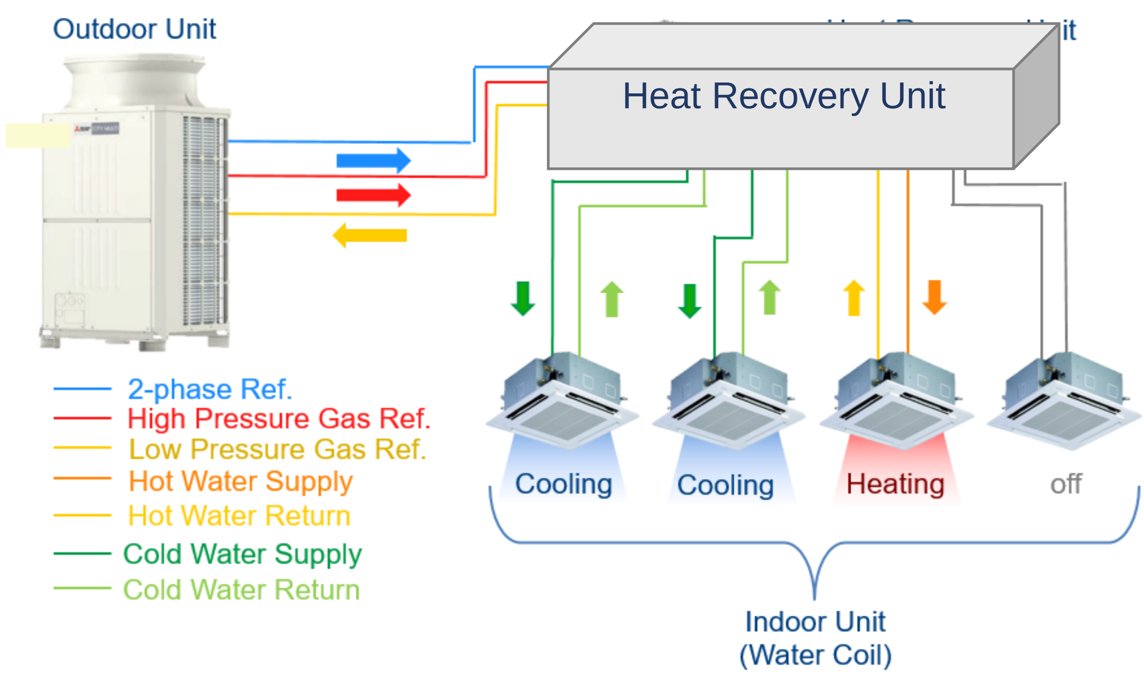

Figure 2: Hybrid VRF system

2. NOVEL HEAT EXCHANGER CONFIGURATION

Huang et al. proposed a finless bifurcated bare tube heat exchanger that was inspired by nature (2016b, 2017b). The two key design points of this new heat exchanger are: (1) finless design with small diameter tube and (2) bifurcated tube. Finless designs using bare tubes with hydraulic diameter less than 1 mm can exceed the air-side heat transfer performance of conventional heat exchangers by 200% with air velocity range of 1~3 m/s (Paitoonsurikarn *et al.*, 2000; Bacellar *et al.* ,2016; Shabtay *et al.*, 2016; and Huang *et al.*, 2018). This is the motivation of eliminating fins in current design. The reason of using bifurcated tubes is that the addition of bifurcation enhances air-side heat transfer by creating 3D flow and improves water-side heat transfer by boundary layer interruption and redevelopment. Numerical air-side parametric study shows bifurcated bare tube heat exchanger has 15% higher air-side heat transfer coefficient and 4%~12% lower air-side pressure drop than those of baseline bare tube heat exchanger with the same outer diameter (0.8 mm), frontal area, volume, and air velocity (3.5~5 m/s) (Huang et al., 2017a, 2017b).

This novel heat exchanger consists of two levels of tubes: the main tubes and the branch tubes as shown in Figure 3. Main tubes are all vertical tubes of which outer diameters are noted as D_1 while branch tubes as D_2. Longitudinal tube pitch (P_l) is defined as the center distance of two adjacent main tubes. The transversal tube pitch (P_t) is then defined as the center distance of two adjacent layers, as shown in Figure 4. Bifurcation angle (θ) is the angle between branching tube and the center line. Both air- and water-side thermal and hydraulic performances are simulated using ANSYS® Workbench™ 18.0. Detailed simulation and analysis could be found in Huang *et al.* (2017a). Airside CFD model was validated against experimental data measured from a 3D printed sample, as shown in Figure 5. Due to leakage issue, this sample was only tested to validate the airside pressure drops for now and a new leak tight prototype is on the way to validate heat transfer. Experimental validation shows a good agreement of less than 3% after applying a correction factor of 0.51, as shown in Figure 6. The potential reasons for deviation include but are not limited to experimental uncertainties, model uncertainties and manufacturing uncertainties.

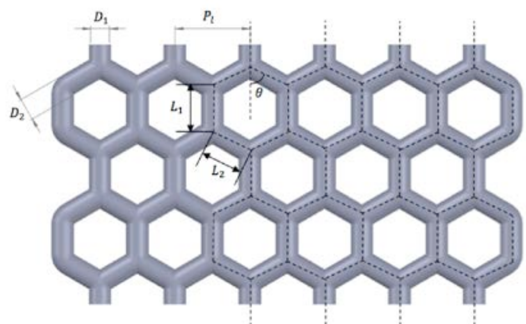

Figure 3: Bifurcated tube structure

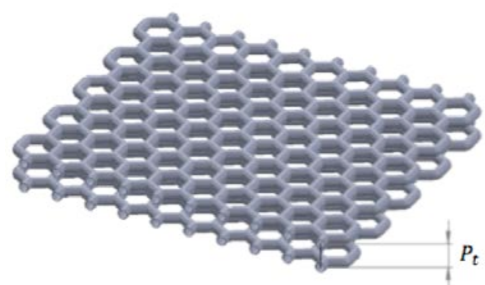

Figure 4: BTHX staggered pattern

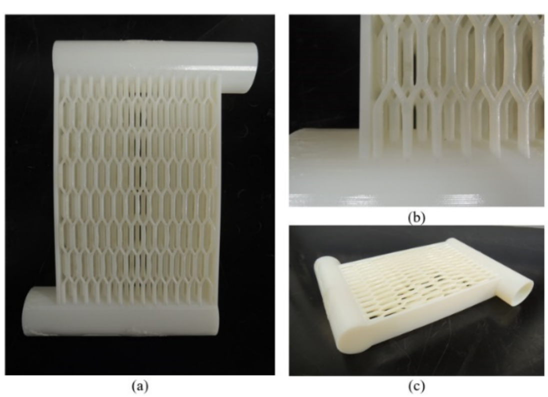

Figure 5: bBTHX sample

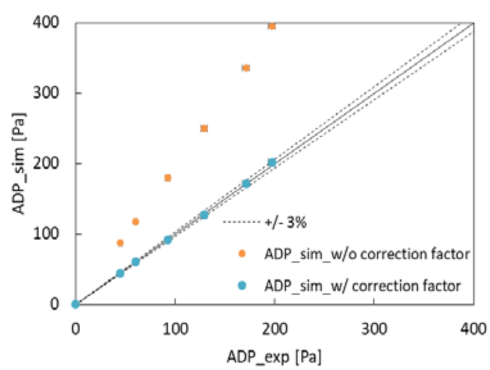

Figure 6: Experimental validation for bBTHX airside pressure drop

3. SYSTEM DESIGN AND SIMULTATION

To evaluate the applicability of the novel heat exchanger, we need to design and evaluate the performance of hVRF system against traditional VRF system. Five different systems were designed and numerically simulated using a component-based steady-state vapor compression system solver (Winkler *et al.*, 2008, Beshr *et al*. 2016). The system configuration, indoor coil type, refrigerant type and design purpose are summarized in Table 1. The refrigerant investigated are R410A, R290 (Propane) and R600a (Isobutane). R410A is a widely used refrigerant in market while R290 and R600a are flammable refrigerants. Two types of indoor units were evaluated, one is a 5 mm slit fin-and-tube HX and the other is bBTHX. Details are discussed in next chapter.

Table 1: Different system design for evaluation

System No.	Type	Ref.	Indoor Coil Type	Design Purpose
1	VRF	R410A	5 mm slit fin-tube HX	Baseline
2	hVRF	R410A	5 mm slit fin-tube HX	To compare hVRF and VRF system (compare 2 with 1)
3	hVRF	R410A	bBTHX	To compare different indoor coil designs (compare 3 with 2)
4	hVRF	R290	bBTHX	To compare different refrigerants (compare 4, 5 with 3)
5	hVRF	R600a	bBTHX	

Traditional VRF system is evaluated using R410A as refrigerant. The piping design restrictions are from VRF system data book in industry. The schematic of traditional VRF system is shown in Figure 7. Based on AHRI VRF testing standard (AHRI, 2010), the minimum indoor unit quantity is two, thus in current study, two indoor units are designed. Length is 110 m for refrigerant pipe A and 60 m for pipe B. Diameter of pipe A is 9.5 mm and diameter of pipe B is 16 mm. Both indoor and outdoor units are traditional fin-and-tube heat exchangers. The piping design restriction of HVRF system is also from VRF system data book in industry. The schematic of traditional VRF system is shown in Figure 8. Two indoor units are designed with the height of 15 m (49 feet). Pipe A is refrigerant loop and the length is 110 m. Pipe B is water pipe and the length is 60 m and the heights of two indoor units are both 15 m. The diameter of pipe B is 22 mm. Outdoor unit is the same traditional fin-and-tube heat exchanger as what was used for baseline. Indoor unit is bBTHX. Outdoor unit and indoor unit exchange heat through a plate heat exchanger.

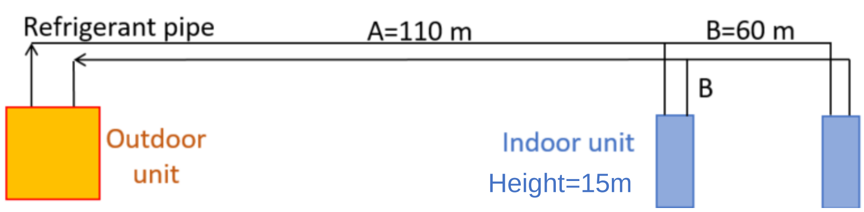

Figure 7: VRF system schematic

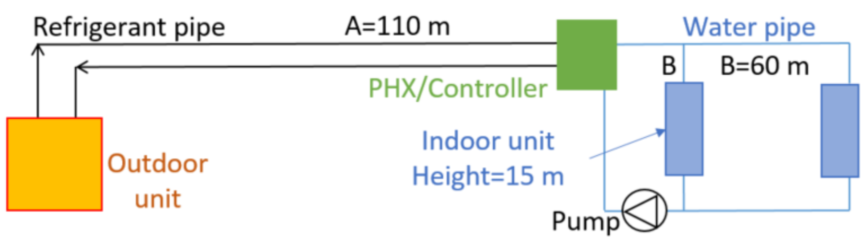

Figure 8: hVRF system schematic

For both systems, the simulation condition is dry test condition for air condition mode from AHRI standard (AHRI, 2008). Ambient dry/wet bulb temperature is 35/23.9°C and indoor air dry/wet bulb temperature is 26.7/19.4°C. Convergence criteria for high pressure side is subcooling temperature equals 5.5 K and convergence criteria for low pressure side is suction superheat temperature equals 5 K. Power input to the fan motors are 36 W and 18 W for 1 m^3/s airflow rate for evaporator and condenser, respectively. The air flow rate for evaporator is 0.283 m^3/s each, and the air flow rate for condenser is 0.693 m^3/s total.

3. INDOOR UNIT COIL DESIGN OPTIMIZATION

Indoor unit of hVRF system is a fan coil unit which consists of air-to-water heat exchanger, fan and motor, drain pan and drainage pump. Different indoor units can be used for hVRF system, such as ducted or ceiling fan coil unit, floor mounted fan coil unit and cassettes fan coil unit. Among these three types, ducted or ceiling fan coil unit is the most compact design thus was selected as the indoor unit type for current study. The fin-and-tube heat exchanger is placed diagonally inside the indoor coil unit. The frontal area (H x W) limit for the heat exchanger is approximately 250 x 900 mm (based on one commercial model). Here we design two heat exchangers, one is a 5 mm slit FTHX with enhanced micro fin tube on waterside and the other is bBTHX. Here are the assumptions and design conditions:

- Inlet air and water temperatures are 26.7 and 7°C.
- Inlet air flow rate is 0.283 m^3/s, and inlet water flow rate is 0.163 kg/s.
- Design capacity is 3,413 W.
- Dimension limitation is 250 x 900 x 100 mm.

3.1 Fin-and-Tube Heat Exchanger Design

The 5 mm slit fin-and-tube heat exchanger was designed using correlations developed based on experimental data at the University of Maryland. The heat exchanger specifications are shown in Table 2. Note fan power is the product of airside pressure drop and air volume flow rate over efficiency.

Table 2: Design of indoor unit using 5 mm slit fin-and-tube HX

Tube per bank	-	42	Water volume	L	0.174	fin effectiveness		0.796
Tube bank #	-	1	Volume	cm^3	2513.7	AHTC	W/m^2K	121
Circuit #	-	7	Fin material volume	cm^3	218	WHTC	W/m^2K	10521
Tube length	m	0.25	Capacity	W	3464	Tube material volume	cm^3	31
Tube OD	mm	5	T_water_out	K	285.2	Total material volume	cm^3	249
Tube ID	mm	4.6	T_air_out	K	289.3	Total power (100% efficiency)	W	7.53
Tube spacing	mm	21	ADP	Pa	11.79	Fan efficiency	-	0.6
FPI	-	22	WDP	kPa	25.75	Fan Power	W	5.5
H × W × D	mm	250 x 882 x 11.4	AHTA	m^2	4.14			
A_fr	m^2	0.221	WHTA	m^2	0.15			

3.2 bBTHX Design Optimization

The bBTHX indoor coil was optimized using multi-scale assisted optimization methodology (Abdelaziz, 2009), which is a methodology for innovative heat exchanger design optimization. It enables efficient integration of the enhanced HX segment performance prediction using CFD simulations with overall HX performance prediction using segmented ε-NTU method, which provides significant computational savings. Main steps are: (1) parameterize the new geometry; (2) run parallel parameterized CFD (an automatic approach to CFD simulations) for Design of Experiment (DoE) space; (3) generate meta-model for selected parameters, such as j and f factors (these meta-model functions as heat transfer and pressure drop correlations); (4) solve heat exchanger performance using ε-NTU method; and (5) optimize design using multi-objective optimization algorithm. The accuracy of meta-models was evaluated using the Meta model Acceptability Score (MAS) (Hamad, 2006). Table 3 summarizes the specifications of air- and water-side models.

Table 3: Summary of air- and water-side meta-models

	Air-side		Water-side	
Input parameters	N_r, D_1, P_l / D_1, P_t / D_1, LR, θ, V_a		D_1, P_l / D_1, LR, θ, V_w	
Output parameters	j, f		Nu, WDP	
Meta-model samples	783		258	
Random points for verification	50		25	
Metamodel acceptability	j	f	Nu	WDP
Score	100% within 15%	100% within 15%	100% within 8%	98% within 10%

The design optimization problem is described as below. Two objectives are minimization of total power and heat exchanger volume. The constraints include (1) total heat exchanger capacity should be similar or larger than baseline; (2) total power is 30% less than baseline; (3) total heat exchanger volume is 30% less than baseline; (4) aspect ratio (AR) is similar to baseline; and (5) frontal area is similar to baseline. Note that the power means the sum of product of pressure drop and volume flow rate of water and air and the efficiency is assumed to be one.

$$
\begin{aligned}
&\text{Optimization:}\\
&\quad \min Power\\
&\quad \min V_{HX}\\
&s.t.\\
&\quad 0.9 \cdot \dot{Q}_{baseline} \le \dot{Q} \le 1.1 \cdot \dot{Q}_{baseline}\\
&\quad Power \le 0.7 \cdot Power_{baseline}\\
&\quad V_{HX} \le 0.7 \cdot V_{HX\_baseline}\\
&\quad 0.9 \cdot AR_{baseline} \le AR \le 1.1 \cdot AR_{baseline}\\
&\quad 0.9 \cdot A_{fr,baseline} \le A_{fr} \le 1.1 \cdot A_{fr,baseline}
\end{aligned}
$$

The optimization results are shown in Figure 9. Along the Pareto fronts from left to right, the volume increases while power decreases. The increase in tube outer diameter is the main reason for volume increase. Transverse pitch to outer diameter ratio (Pt/OD) reaches the minimum limit (1.5). Pl/OD has a negative relationship with LR: as Pl/OD increases, LR (=L_1/L_2) tends to decrease. This is because larger Pl/OD results in lower air-side heat transfer coefficient, thus LR needs to be smaller to increase the heat transfer coefficient, and vice versa.

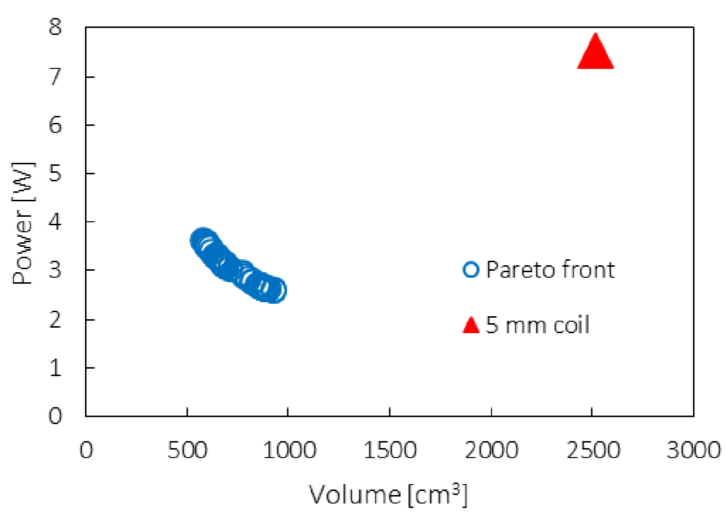

Figure 9: Optimization results of bBTHX indoor unit coil

Table 4 compares one selected optimal design with baseline 5 mm coil. When the two distinctive designs are compared, the advantages of applying bBTHX as an indoor coil for the hVRF system are clear. The CFD verification of the design point is summarized in Table 5. The summary and reasons causing the differences in each parameter are:

- Higher air-side heat transfer coefficient: bBTHX has 117% higher air-side heat transfer coefficient than baseline. Mechanisms of heat transfer enhancement include larger mass flux, branch tube with smaller diameter and 3D flow caused by the addition of bifurcation (details are in Huang *et al.*, 2017a).
- Lower air-side pressure drop: bBTHX has 61% lower air-side pressure drop than FTHX. The addition of bifurcation causes flow bypass and lower flow rate at bare tube region (details are Huang *et al.*, 2017a). However, the most important reason for a lower pressure drop is the reduction in depth. bBTHX's depth is only 3.9 mm while that of FTHX is 11.4 mm.
- Slightly lower water-side heat transfer coefficient: mechanisms including smaller OD branch tube, boundary layer redevelopment and flow separation at the bifurcation of bBTHX enhance the water-side heat transfer coefficient. It has much higher water-side heat transfer coefficient than bare round tube heat exchanger with the same diameter. However, in current study, the water-side heat transfer coefficient is slightly lower than baseline because the 4 mm FTHX's tube is a microfin enhanced tube. This indicates that bifurcation structure could be utilized as one of the method to enhance tube side single phase heat transfer.
- Lower water-side pressure drop: water-side pressure drop of bBTHX is 63% lower than baseline. The reason is that it has more tube numbers than FTHX, resulting in lower water velocity. This means that using bifurcation would result in much lower pressure drop and slightly lower heat transfer than using microfins to enhance water-side heat transfer.
- Lower total pumping power: bBTHX has 65% lower total pumping power.
- Smaller volume and material volume: bBTHX has 75% less total material volume and 65% less envelope volume than the baseline.

Table 4: Design of indoor unit B – bBTHX

Geometric parameter			Model result			% difference compared with FTHX
Tube per bank	-	548	Capacity	W	3,449	-0.4%
Tube bank #	-	5	T_water_out	K	285.2	0.0%
Tube length	m	0.25	T_air_out	K	289.6	0.1%
Tube OD	mm	0.56	ADP	Pa	4.62	**-60.8%**
Tube ID	mm	0.50	WDP	kPa	9.6	**-62.7%**
Pl/OD	-	2.95	AHTA	m^2	1.22	**-70.5%**
Pt/OD	-	1.51	WHTA	m^2	0.98	**553.3%**
LR		5.48	AHTC	W/m^2K	262.6	**117.0%**
θ		19.7	WHTC	W/m^2K	8,714.6	**-17.2%**
H × W × D	mm	250 x 900 x 3.9	Total material volume	cm^3	61.2	**-75.4%**
A_fr	m^2	0.225	Water volume	L	0.108	**-37.9%**
			Volume	cm^3	877.5	**-65.1%**
			Total power (100% efficiency)	W	2.66	**-64.7%**
			Fan efficiency	-	0.6	
			Fan Power	W	2.18	**-60.4%**

Table 5: Design point verification against CFD simulation for HVRF indoor coil

Optimization results				CFD results				Percentage deviation			
AHTC [W/m^2K]	ADP [Pa]	WHTC [W/m^2K]	WDP [Pa]	AHTC [W/m^2K]	ADP [Pa]	WHTC [W/m^2K]	WDP [Pa]	AHTC [%]	ADP [%]	WHTC [%]	WDP [%]
262.6	4.6	8,714.6	1,180.1	270.3	4.8	8,284.8	1,121.6	3%	2.9%	-4.9%	-5%

4. SYSTEM PERFORMANCE COMPARISON

Performances of all systems are summarized in Table 6. When the system No. 1 and No. 2 are compared, we can find that the COP decreases from 2.9 to 2.56. This is mainly because of the single-phase heat transfer at indoor unit. However, the advantage of using hybrid system is the reduction of refrigerant charge. The difference between No. 2 and No. 3 is the indoor coil design. After the indoor coil is optimized, the total COP increases by 10%. No. 3, No. 4

and No. 5 are to compare system performance with different refrigerants. Using R290 (System No. 4) and R600a (System No. 5) largely increases the system COP and further reduces refrigerant charge.

Table 6: System performance comparison

System No.	Refrigerant	COP [-]	Capacity [W]	P_comp [W]	P_con,fan [W]	P_waterloop [W]	P_eva,fan [W]	P_total [W]	Refrigerant charge/change [kg]/[%]
1	R410A	2.9	6,869.0	2,335.0	12.5	-	20.4	2,367.9	17.83
2	R410A	2.56	6,865.4	2,591.4	12.5	65.5	11	2,680.4	10.51 **(-41%)**
3	R410A	2.82	6,859.1	2,353.1	12.5	60.2	4.4	2,430.2	10.51 **(-41%)**
4	R290	3.05	6,869.2	2,173.5	12.5	60.2	4.4	2,250.6	5.66 **(-68%)**
5	R600a	3.2	6,856.4	2,064.8	12.5	60.2	4.4	2,141.8	5.86 **(-67%)**

6. CONCLUSIONS

Variable refrigerant flow (VRF) system can achieve higher efficiency than traditional central air conditioning unit. The drawback of VRF systems is the complex and long refrigerant piping, high initial cost, high refrigerant charge and complexity in maintenance for refrigerant leakage check. A water-based hybrid VRF system is a combination of traditional VRF system and water chiller system. Compared with traditional VRF system, it has the advantages of reduced refrigerant charge, wider selection of refrigerants including flammable ones and lower maintenance cost since there is no concern on indoor refrigerant leakage. However, the hVRF system shows a slightly reduced COP due to low water-side heat transfer coefficient. Therefore, we proposed a bifurcated bare-tube heat exchanger to enhance heat transfer. The key feature of this design is the addition of bifurcation that enables 3D flow mixing on air-side and boundary layer redevelopment on water-side. We simulated five different VRF and hVRF systems using detailed steady state numerical model to study the impact of using hybrid water loop, improved heat exchanger design and flammable refrigerants. All five different VRF systems have similar capacities. The biggest advantage of hVRF system is the refrigerant charge reduction. bBTHX is proved to have the potential to be applied as the indoor unit of VRF system. Optimal design has 60% less total pumping power, 65% smaller volume and 70% smaller package- and material-volume than those of traditional fin-and-tube heat exchanger when delivering the same capacity, and the new design also increases the system COP by 10%. Besides having smaller pressure drop on both air- and water-side, bBTHX also reduces the weight of the coil by reducing material volume and internal water volume. When flammable refrigerants, R290 and R600a, are used, the system COP increases and the charge is reduced. Overall, the bBTHX shows a potential applicability as indoor coils of hVRF systems. More study should be conducted on the manufacturability of this type of heat exchanger and experimental validation.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

A_{fr}	frontal area	(m^2)
$A_{fr\_baseline}$	baseline heat exchanger frontal area	(m^2)
AR	aspect ratio	(–)
$AR_{basline}$	baseline heat exchanger aspect ratio	(–)
bBTHX	bare tube heat exchanger	
COP	coefficient of performance	
D_1	main tube diameter	(mm)
D_2	branch tube diameter	(mm)
EEV	electronic expansion valve	
f	Chilton and Colburn f factor	(–)
FTHX	fin-and-tube heat exchanger	
hVRF	hybrid variable refrigerant flow	
j	Chilton and Colburn j factor	(–)
L_1	main tube length	(mm)
L_2	branch tube length	(mm)

N_r	row number	(–)
Nu	water-side pressure drop	(mm)
OD	outer diameter	(mm)
P	power	(W)
P_l	longitudinal tube pitch	(mm)
P_t	transversal tube pitch	(mm)
$\dot{Q}_{baseline}$	baseline heat exchanger capacity	(W)
$\dot{Q}$	heat exchanger capacity	(W)
V_a	air velocity	(m/s)
V_{HX}	heat exchanger volume	(m^3)
VRF	variable refrigerant flow	
V_w	water velocity	(m/s)
WDP	water-side pressure drop	(Pa)
θ	bifurcation angle	(deg)

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