Design of Divided Condensers for Desiccant Wheel-Assisted Separate Sensible and Latent Cooling AC Systems

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ABSTRACT

A desiccant wheel-assisted separate sensible and latent cooling air-conditioning system improves the energy efficiency of conventional air-conditioning systems by increasing the evaporating temperature of the refrigerant for sensible cooling. With the addition of a desiccant wheel into the air-conditioning system, the system integration requires a further study to ensure that the desiccant wheel receives sufficient hot air for its regeneration while maximizing the entire air-conditioning system's coefficient of performance. This paper discusses the application of divided condensers which are specifically designed for such systems in order to meet the goals. In-house developed heat exchanger design software was used to evaluate different layouts of divided condensers. The different layouts were compared to each other in terms of outlet air temperature profile. A parametric study was also conducted to explore the possible design with less material cost than a baseline condenser. Finally, the performance of the desiccant wheel-assisted separate sensible and latent cooling air-conditioning systems with various efficiency-enhancing options were simulated under the summer test condition according to the AHRI standard 210/240 for performance rating of unitary air-conditioning and air-source heat pump equipment to investigate its energy saving potential.

1. INTRODUCTION

Researchers have been focusing on energy-efficiency improvement of air-conditioning/heat pump systems for decades. There are various techniques available nowadays to reduce the power consumptions of conventional vapor compression cycles (VCC). Separate sensible and latent cooling (SSLC) is one among such technologies that achieve the reduction of compressor power inputs by decreasing the temperature lift from heat absorption to heat rejection. In the SSLC system, one VCC provides only sensible cooling with a refrigerant evaporating temperature higher than that of a conventional system, and therefore the SSLC system is operated at a reduced temperature lift between the evaporation and condensation processes. A reduced temperature lift leads to a reduced compressor power input. Nevertheless, the increase of evaporating temperature may lead to a loss of dehumidification capability if the refrigerant temperature is above the dew point of the space air. In order to avoid such a loss, the SSLC system uses either a desiccant wheel (DW) or a secondary VCC for latent cooling. The technology has been proven to be successful through experimentation and simulations (Ling *et al.*, 2010, 2011). This paper focuses on applying the SSLC technology to R410A air-conditioning systems. R410A, a commonly used working fluid for the air-conditioning systems, was invented in the early 90's as a mixture of R32 and R125 (50/50 wt.%). As a result of the mandated phase-out for R22 by the Montreal Protocol, R410A emerged in the air-conditioning market as a substitution for R22 owing to its merit of zero ozone depletion potential. This paper discusses the integration of various efficiency-enhancing options into DW-assisted SSLC R410A air-conditioning systems such as a divided heat exchanger (HX), evaporative cooling, and an enthalpy wheel (EW). The divided HX has the benefit of reducing condensing pressure of the system while still maintaining enough hot air supply to the DW. Evaporative cooling can further reduce the condensing temperature but it is only applicable to the SSLC systems on the basis of the divided HX. EW utilizes the cooling from the indoor exhaust air to pre-condition the fresh air and hence reduces the power inputs to the VCC.

2. MODELING APPROACH

CoilDesigner (Jiang *et al.*, 2006), an in-house heat exchanger simulation software package, was utilized to model all the heat exchangers used in the VCC. Engineering Equation Solver (F chart, 2011) gathered the calculation results from CoilDesigner and was used to model other components of the VCC (compressor and expansion device), and EW. The SLHX was also modeled in the CoilDesigner. The compressor was modeled based on three compressor efficiencies: isentropic, volumetric and compressor efficiencies. The three efficiencies were modeled as functions of pressure ratio, degree of superheating, and compressor frequency. The efficiency correlations were obtained by fitting experimental results into polynomial equations. The expansion device was modeled based on an isenthalpic expansion assumption. The EW was modeled using the efficiency-based method. The maximum possible enthalpy difference between two air streams was first calculated, and then the actual enthalpy exchange between the two streams was assumed to be the product of the maximum enthalpy difference and the efficiency. The efficiencies of EW used in the model are 0.82 for sensible heat transfer and 0.65 for latent heat transfer, respectively. The dehumidification performance of the DW was assumed to be a function of DW rotation speed, regeneration temperature, and air velocity through the wheel, as well as the inlet humidity ratio of the regeneration side. The enthalpy of air in the process side of the DW was calculated as a function of DW rotation speed, regeneration temperature, and air velocity. All of the input data described above was obtained from experimental data (Ling *et al.*, 2011). In addition, fan power consumptions were modeled based on two scenarios: one with all AC motor-driven fans and the other with all brushless DC (BLDC) motor-driven fans. The fan efficiency was assumed to be the product of the motor efficiency and the fan blade efficiency.

The following assumptions were made for modeling the vapor compression cycle:

- System capacity: 3.5 kW
- Sensible heat factor: 0.714
- Latent capacity: 1.0 kW
- Outdoor/indoor air conditions: 35°C, 44% RH / 27°C, 50% RH
- Air conditions after evaporative cooling (used in Section 4): 24.8°C, 100% RH
- Return air conditions after evaporative cooling (used in Section 4): 19.5°C, 100% RH
- Regeneration temperature: 50°C
- Regeneration air flow rate: 0.15 m^3s^{-1}
- Indoor air flow rate: 0.41 m^3s^{-1}
- Total air flow rate for condenser: 0.42 kgs^{-1} (volume flow rate varied in the condition with or without evaporative cooling)

3. LAYOUT OF DEVIDED HX AND OPTMIZIATION

Since the DW has to be continuously regenerated, it requires hot air discharged from the condenser as the heat source. For the entire system, such requirement leads to a design issue: how to provide enough hot air to the DW while maintaining a high system COP. As the air flow rate through the conventional condenser is usually 2 to 4 times higher than the amount required by the DW, the condenser outlet air temperature is usually not high enough for DW regeneration. Consequently, less air has to be circulated through the condenser than that of the conventional system in order to provide high enough air temperature to the DW. However, it may increase the heat rejection pressure and reduce the system COP (Ling *et al.*, 2011). To address this issue, a divided heat exchanger is hereby proposed as a solution. In a divided heat exchanger, there is one section (the first section) dedicated to supplying DW regeneration air, and the other sections are for the usual heat rejection. The first section produces only the required amount of regeneration air by utilizing high temperature discharge refrigerant from the compressor. The other sections have no requirement of regeneration air production and therefore, the entire condenser can still have the same amount of air and approach temperature as those of the conventional condenser. Moreover, for the heat rejection sections, it is now possible to apply evaporative cooling to further reduce the condensing pressure. While the design of the divided HXs can involve various aspects, this paper deals with two issues: the layout of the divided HXs and the exploration of low cost condenser design. The first section of the divided condenser, which is used to provide hot air for DW regeneration, can be either placed behind the other sections so that air flows in series from other sections to the first section and forms a counter-flow HX configuration with the refrigerant circuit, or it can be placed parallel to the other sections so that the air flows parallel to all the sections. The benefit of the serial layout is that the total height of the HX can be reduced and air can be heated more effectively because it flows through more banks of HX. However, the air-side pressure drop may accordingly be high. The benefit of the parallel layout is to reduce the air side pressure drop. Table 1 shows the details of three HXs layouts for comparison. All three HX layouts have the same tube geometries, fin geometries and tube spacing. Hence, the three HXs have the same primary and secondary heat transfer areas. The only difference among the three is how the tubes and air streams are divided into different sections. Figures 1 to 3 show the air outlet temperature profiles of the three HX layouts. It is clear that the baseline HX's profile has two separate parts of high temperature discharge air, which makes it difficult to collect enough hot air for DW regeneration. In contrast, both divided HX designs make hot air concentrated on the top part of the HX for DW regeneration. From that point of view, both divided HXs are successful. However, the serial layout has less frontal area than the parallel layout, which leads to a higher air velocity. Moreover, the serial design directs part of the airflow through eight banks of tubes, and consequently leads to a much larger air-side pressure drop. The parallel layout has an air-side pressure drop of only 47 Pa, while the serial layout, which is subject to the double penalties of larger air velocity and more banks, has an air-side pressure drop of 164 Pa. Since too much of an air-side pressure drop can consume too much fan power, the serial layout of divided HXs is not preferred.

Table 1: HX layouts of baseline HX and divided HXs in parallel layout and serial layout

HX	Baseline HX	Divided HXs in parallel layout		Divided HXs in serial layout	
Layout	Single condenser	Divided condenser I	Divided condenser II	Divided condenser I	Divided condenser II
Number of tube bank	4	4	4	4	4
Number of circuit	2	2	2	2	2
Number of tube/bank	22	12	10	6	16

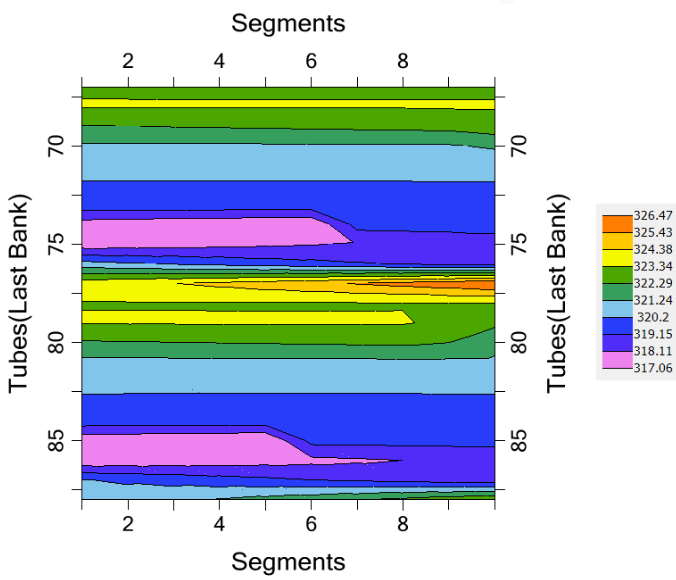


Figure 1: Discharge air temperature profile of the baseline single HX design

The current dimension of the divided condenser is based on the commercial product described in Ling *et al.* (2011). To explore the potential of any material cost savings of the condenser, a parametric study was conducted to investigate different designs of condensers by varying the tube length, tube vertical and horizontal spacings, and fin density. Air-side and refrigerant-side pressure drops are the two constraints which are specified as PD_{air} < 50 Pa and PD_{R410A} < 1°C saturation temperature, respectively. All the feasible designs are plotted in Figure 4 in the form of heat exchanger material cost versus minus condenser capacity. The commercial product aforementioned is marked by a red dot and used as the baseline design. The black arrow indicates the direction of better heat exchanger deigns (Pareto frontier). As shown in the figure, the design at the right corner has the least material (16% less than the baseline design) and the smallest capacity (but only 0.2% less than the baseline design).

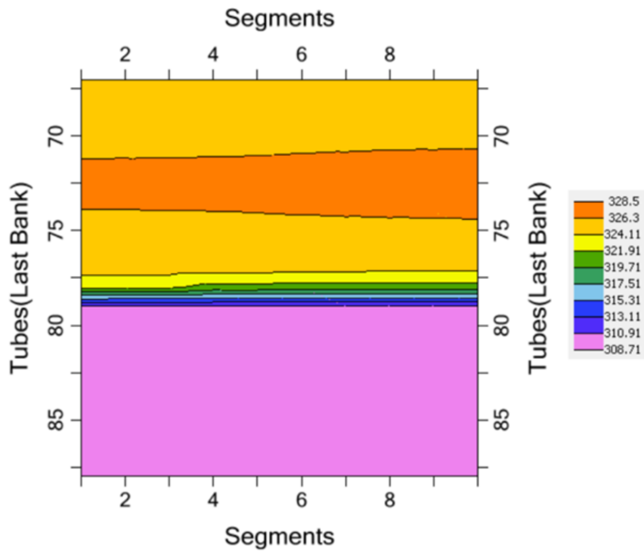


Figure 2: Discharge air temperature profile of the divided HXs in parallel layout

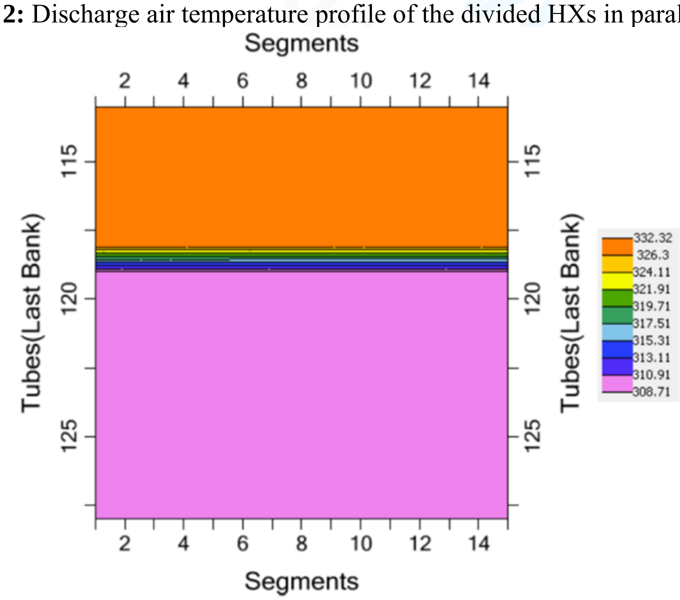


Figure 3: Discharge air temperature profile of the divided HXs in serial layout

The cost savings come from 12% of tube length reduction and 22% of the fin density reduction compared to the baseline design. On the other hand, the design on the top left indicates the largest capacity condenser (0.06% larger than the baseline) at the highest cost (93% higher than the baseline). The largest capacity design requires 12.5% longer tube, 36% larger tube vertical spacing, 26% larger tube horizontal spacing and 27.5% larger fin density than those of the baseline heat exchanger. After reviewing the two extreme design points, it is conclude that the baseline HX is oversized; and the least cost design obtained from the parametric study provides a similar capacity but saves 16% material cost.

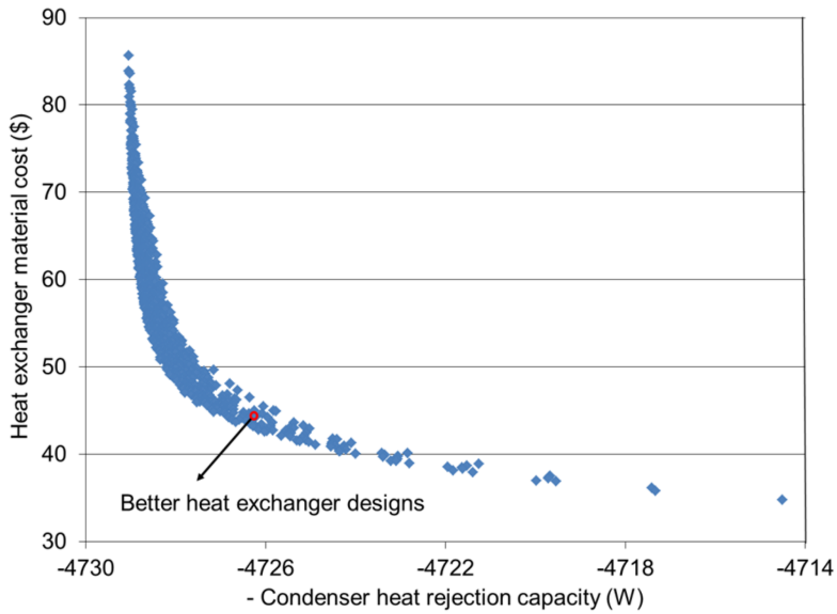
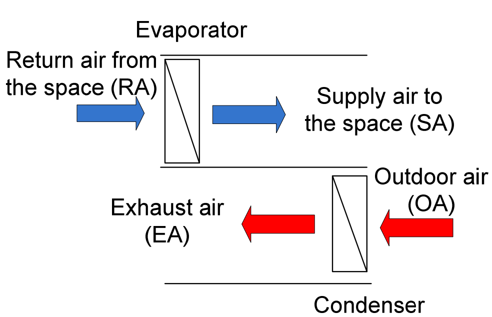


Figure 4: Parametric study of divided condenser designs

4. DW-ASSISTED SSLC SYSTEMS INTEGRATED WITH EVAPORATIVE COOLING AND EW

In this section, several integrated system configurations were selected as candidates for a detailed study. The conventional four-component VCC was considered as the baseline system (Figure 5). Additionally, four SSLC systems were considered. Figure 6 shows components lay out in the air streams of the DW-assisted SSLC system with a single, undivided condenser. Both Figures 7 and 8 describe DW-assisted SSLC systems using divided condensers. The SSLC option 1 is a zero-ventilation system, with the condenser divided into two parts. In this option, the incoming air to the second part of the condenser may or may not flow through evaporative cooling. The SSLC option 2 has the required amount of fresh air mixed with return air from the space, and the condenser is divided into three parts. Each part may face different air conditions, such as the ambient air condition, the ambient air condition after the evaporative cooling process, and the exhaust air from the space after the evaporative cooling process. The SSLC option 3 is the DW-assisted SSLC system with an added EW (Figure 9). The working principle of the EW can be simply described as: hot and humid ambient air flows through the EW, transferring the sensible heat and water vapor to the relatively cold and dry return airstream from the conditioned space. The heat and mass transfer between the two air streams, via the EW, helps recover both the sensible and latent cooling from the space.



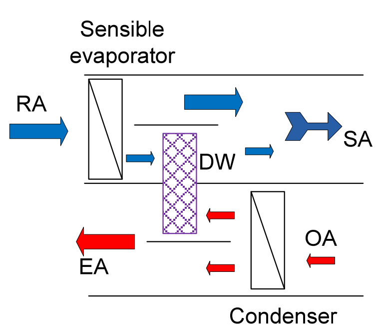


Figure 5: Four component baseline system **Figure 6:** DW-assisted SSLC system with single HX

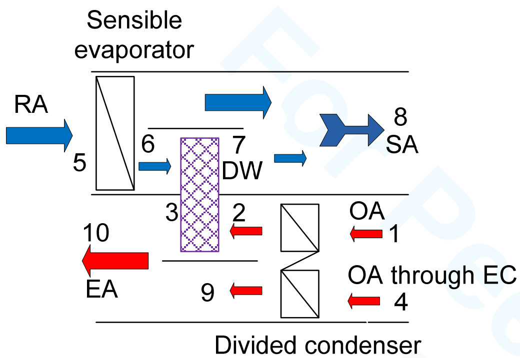


Figure 7: DW-assisted SSLC system (Option 1)

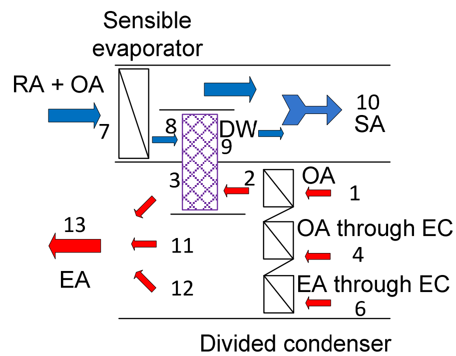


Figure 8: DW-assisted SSLC system (Option 2)

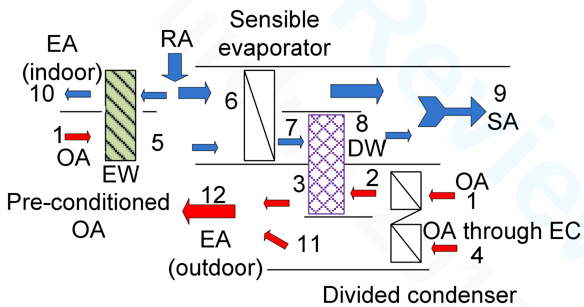


Figure 9: DW-EW-assisted SSLC system (Option 3)

The COPs of different SSLC options are compared with the conventional systems. The system COP is defined as the ratio of the space cooling capacity (3.5 kW) to the total electric power input, including the compressor power input and the power input of all fans. For the baseline systems, the system COP could be considered the same as the COP of the VCC when neglecting the heat load of fans, which is defined as the ratio of the evaporator air-side cooling capacity to the total power input. However, for the DW-assisted SSLC system, the VCC provides extra cooling capacity to compensate the difference between the heat of adsorption and heat of evaporation. It consequently causes a smaller system COP than that of the VCC. In order to search for the optimum operating condition of VCC in which the COP is maximized, a genetic algorithm built in Engineering Equation Solver (Charbonneau and Knapp, 2002) was adapted to maximize the system COP by varying the evaporating and condensing pressures. The optimization problem is formulated as follows:

$$\text{Min } (f) = -\text{ (COP of the system)} \tag{1}$$

$$\text{s.t. system capacity} \geq Q_{req} \tag{2}$$

$$\text{Air discharge temperature off the condenser} \geq 50°\text{C} \tag{3}$$

Equation 1 shows the objective of the optimization followed by two constraints in Equations (2) and (3), which represent the required system cooling capacity should be at least of 3.5 kW and the minimum regeneration temperature should be 50°C, respectively. Figure 10 shows the system COP improvement by applying aforementioned performance-enhancing options, while Table 2 provides the system COP, COP improvement and respective baseline system of each system option. From Figure 10, it can be seen that the improvement resulting from the application of divided condenser (option 1) to the SSLC system with single condenser is 3% and 7% for systems using AC motors and BLDC motors, respectively. The application of evaporative cooling to the lower part of the condenser in the R410A SSLC option 1 system resulted in 6% and 7% COP improvement for systems using AC motors and BLDC motors, respectively. The application of evaporative cooling reduces the actual ambient temperature to the condenser and therefore helps reducing the condensing pressure. The SSLC option 2 divides the condenser into three parts, and the evaporative cooling of return air provides the lowest possible temperature to further cool down the refrigerant. In the SSLC option 2, the COP had 3% and 4% improvement over the SSLC system option 1 with evaporative cooling for systems using AC motors and BLDC motors, respectively. The DW-EW-assisted SSLC system (option 3) demonstrates the highest COP. Application of the EW provides "free" sensible and latent cooling, and reduces the sensible cooling requirement of the VCC by 29% (3.8 kW to 2.7 kW).

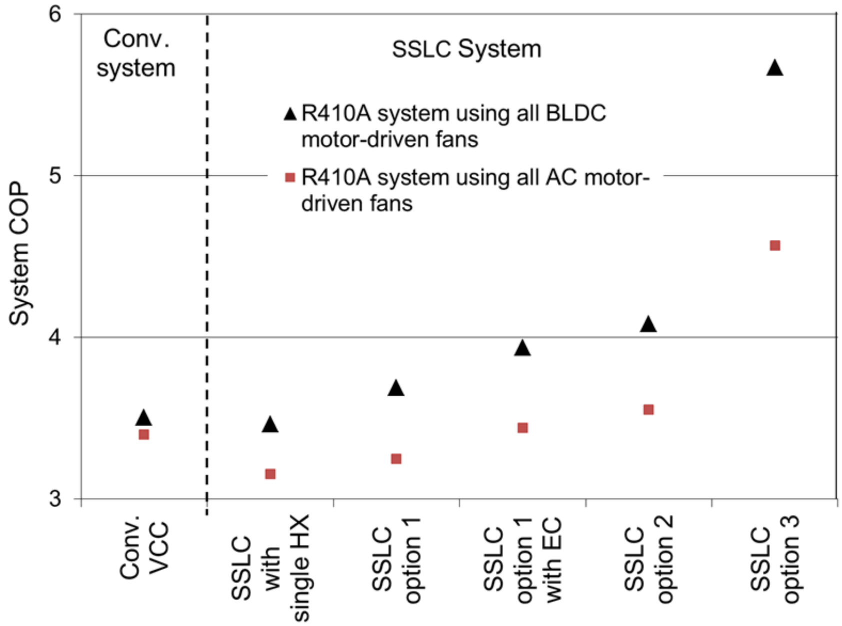


Figure 10: COPs of baseline system and different SSLC options

The heat exchangers were modeled in reduced sizes in proportion to the reduced capacity. The compressor power input was reduced to a minimum with this option. The COP reached 4.6 and 5.7 for systems using AC motors and BLDC motors, respectively, and the improvement over the SSLC system option 1 with evaporative cooling is 33% and 39%, respectively. It is also observed from the simulation results that it is important to apply high efficient motor for the SSLC systems. Because of the reduced air temperature difference across the evaporator, air flow rate through the evaporator has to be increased to match the required cooling capacity. The increased air flow rate can lead to an excessive fan power and an example of it can be found from the COP decrease when the SSLC is applied with single HX. Therefore, BLDC motors are suggested for the SSLC systems. Figure 11 compares the total fan power consumptions for SSLC options investigated. The fan power consumptions of the SSLC system are almost tripled that of the conventional system. The increase is partly due to aforementioned increased air flow rate and moreover it is also due to the increased pressure lift requirement from the DW and EW. However, the cases utilizing BLDC-motor fans save around 50% power consumption compared to the AC-motor fan cases. Such savings are the result of high-efficiency of BLDC motor.

Table 2: Detailed system COP, COP improvement over respective baseline systems

System option	System COP (AC/BLDC motors)	COP improvement [%] (AC/BLDC motors)	Respective baseline system
Conventional system	3.40/3.50	N/A	N/A
SSLC with single HX	3.15/3.46	-7/-1	Conventional system
SSLC option 1	3.25/3.69	3/7	SSLC with single HX
SSLC option 1 w evaporative cooling	3.44/3.94	6/7	SSLC option 1
SSLC option 2	3.55/4.08	3/4	SSLC option 1 w evaporative cooling
SSLC option 3	4.57/5.67	33/39	SSLC option 1 w evaporative cooling

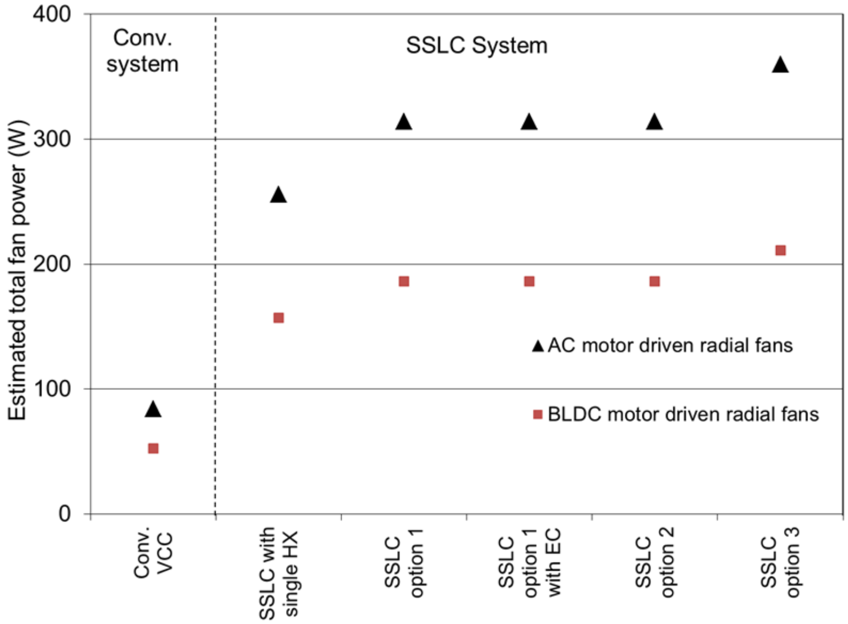


Figure 11: Fan power estimation for different systems

5. CONCLUSIONS

In this paper, different efficiency-enhancing options for the DW-assisted SSLC R410A air-conditioning system are discussed. The design of divided heat exchanger has been investigated in terms of heat exchanger layout and the potential of heat exchanger material cost savings. It is concluded that the parallel layout of the divided heat exchanger is suitable for the SSLC system and through optimization, it is possible to reduce the current condenser material cost by 16%. Other options such as evaporative cooling and enthalpy wheel can further improve the system performance up to 7% and 39%, respectively. To better demonstrate the energy savings of the SSLC system, high-efficiency BLDC motors are recommended for the system.

NOMENCLATURE

AC	alternating current	(-)
BLDC	brushless direct current	(-)
COP	coefficient of performance	(-)
DW	desiccant wheel	(-)
EA	exhaust air	(-)
EW	enthalpy wheel	(-)
HX	heat exchanger	(-)
OA	outdoor air	(-)
PD	pressure drop	(-)
Q	capacity	kW
SA	supply air	(-)
SSLC	separate sensible and latent cooling	(-)
VCC	Vapor compression cycle	(-)

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