PRIMARY ENERGY EFFICIENCY ANALYSIS OF DIFFERENT SEPARATE SENSIBLE AND LATENT COOLING TECHNIQUES

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ABSTRACT

Separate Sensible and Latent cooling (SSLC) has been discussed in open literature as means to improve air conditioning system efficiency. The main benefit of SSLC is that it enables heat source optimization for the different forms of loads, sensible vs. latent, and as such maximizes the cycle efficiency. In this paper I use a thermodynamic analysis tool in order to analyse the performance of various SSLC technologies including: multi-evaporators two stage compression system, vapour compression system with heat activated desiccant dehumidification, and integrated vapour compression with desiccant dehumidification. A primary coefficient of performance is defined and used to judge the performance of the different SSLC technologies at the design conditions. Results showed the trade-off in performance for different sensible heat factor and regeneration temperatures.

1. INTRODUCTION

Conventional Air Conditioning (AC) systems rely on cooling the supply air to its dew point temperature ($T_{dp,s}$) and then reheat it before supplying it to the conditioned space as shown in Figure 1 using dashed red lines. This is considered as wasteful use of energy since the vapour compression system (VCC) has to operate at evaporating temperatures less than $T_{dp,s}$ and then reheat it using either internal heat exchangers or additional heat sources to the required conditions (T_s, ϖ_s). A Perfect AC with separate sensible and latent cooling will have a performance as shown in Figure 1 using solid red lines, the return or mixed air is first dehumidified adiabatically and then cooled at constant humidity ratio to the required supply conditions. One of the major challenges with achieving this perfect performance is the isothermal dehumidification process. If dehumidification is achieved using a VCC, air has to be cooled below the dew point of the supply air and then reheated using waste heat from the condenser to the supply air conditions. For solid desiccant dehumidification, the desiccant wheel is heated as water vapour is adsorbed and then further reheated to desorb the moisture content which result in heating while dehumidification of the air (Hwang et al., 2010). Novel liquid desiccant air dehumidification technologies can approach isothermal dehumidification by internally cooling the liquid desiccant as it adsorbs the moisture from the air (Lowenstein, 2008).

Previous research on SSLC has proven that the technology is capable of providing significant energy savings. Ling et al. (2008) presented a theoretical study on using two-stage VCC for SSLC with a modelled COP of 5.04 to 5.38 with a an average of 30% energy savings compared to conventional AC systems. Hwang et al. (2010) experimentally evaluated a SSLC comprising a VCC with solid desiccant regenerated using the VCC waste heat. They used R-744 transcritical cycle and R-410A VCCs in their experimental evaluations and varied the desiccant regeneration temperature between 45.5 and 50.7°C. Their evaluation showed overall COP enhancement between 7 and 34% depending on the regeneration temperature.

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2. SEPARATE SENSIBLE AND LATENT COOLING SYSTEMS

In this paper I compare the performance of 4 different SSLC systems to the baseline AC using VCC. Cycle analysis is made using EES© (Klein, 2013). The baseline VCC is modelled after a 4-component system: evaporator, compressor, condenser, and adiabatic expansion valve. The first SSLC system, SSLC1, is a system based on 2 stage VCC system. The lower evaporating temperature stage is used for the latent cooling while the higher evaporating temperature stage is used for the sensible cooling. The second SSLC system, SSLC2, is a system that uses VCC for sensible cooling only and an isothermal desiccant dehumidification for the latent cooling. For SSLC2, the liquid desiccant regeneration is achieved using VCC waste heat rejected from the condenser. The SSLC3 system is similar to SSLC2 except for heat of regeneration being provided from the desuperheater only – this results in improved operational COP since the sensible cooling VCC condensing temperature is kept constant. Finally, the SSLC4 system is a system that is based on VCC for sensible cooling and heat activated desiccant dehumidification for latent cooling.

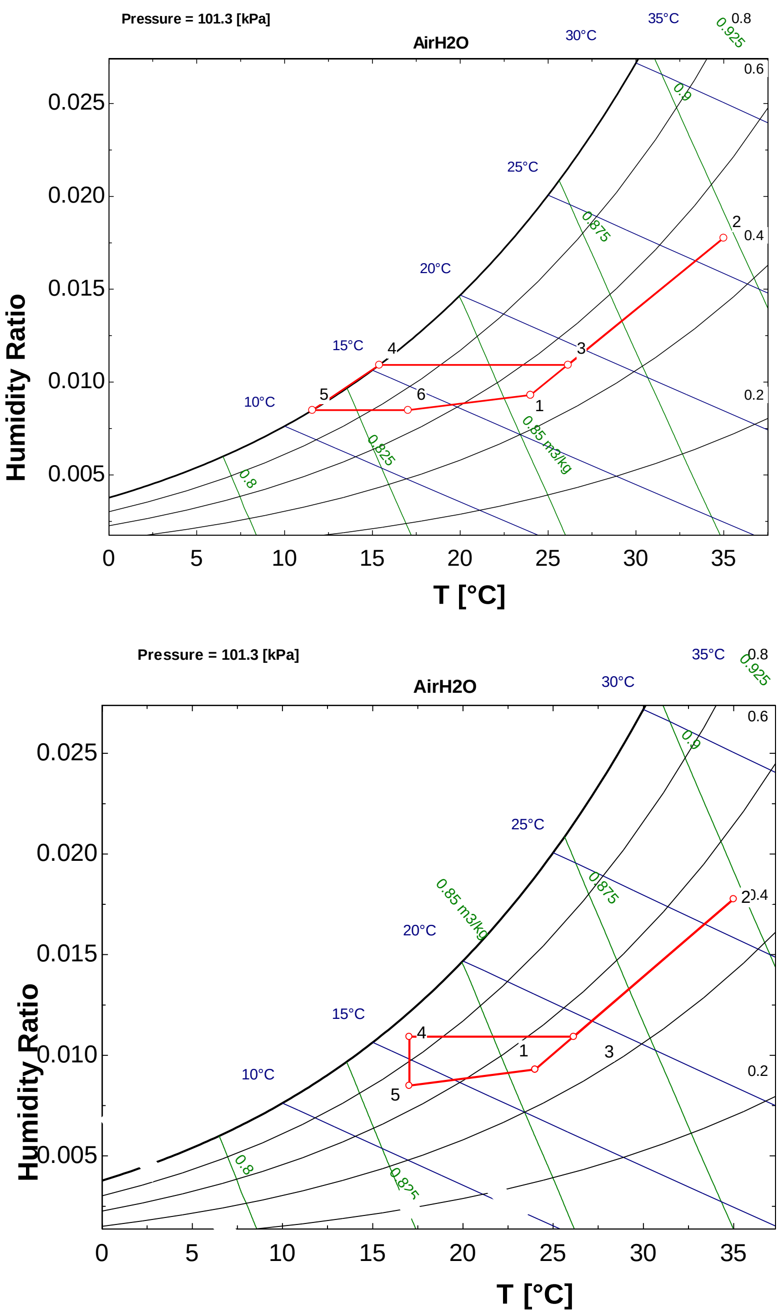

Figure 1. Psychrometric Chart at 1 Atm showing typical room AC (Top) and ideal SSLC (Bottom).

3. SYSTEM MODELING

System modelling is performed in EES using the following simplified cycle assumption:

- Cooling capacity (Q) = 3.517 kW
- Refrigerant for VCC = R-410A
- SHF = 0 to 1.0
- Compressor isentropic efficiency = 70%
- Minimum heat exchanger temperature difference (TTD) = 5 K
- Subcooling = 5 K
- Superheating = 5 K
- Ambient conditions: 35°C, 50% RH, 101.325 kPa
- Room conditions: 24°C, 50% RH, 101.325 kPa
- Desiccant regeneration temperature (T_{regen}) = 40 - 66°C
- Desiccant regeneration coefficient of performance = 0.8 (range from 0.7 to 1.1)

For the baseline VCC system, the supplied air conditions varied with SHF resulting in changes in evaporating temperatures and thus observed cycle COP. The evaporating temperature was calculated as shown in equation 1a and the condensing temperature was calculated as shown in equation 1b. A typical P-h diagram with baseline VCC shown in green solid lines is illustrated in Figure 2.

$$T_{evap} = T_{dp,s} - TTD \quad (1a)$$

$$T_{cond} = T_{ambient} + TTD \quad (1b)$$

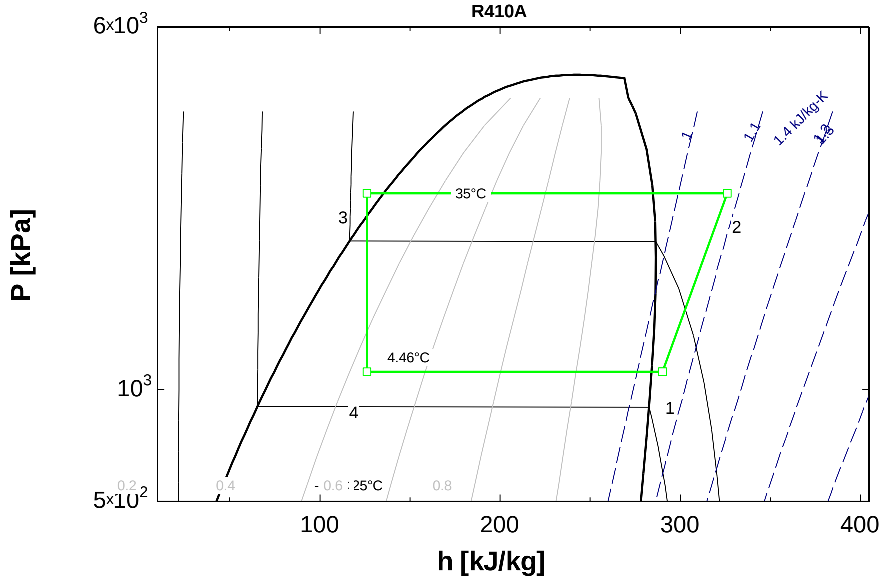

Figure 2. P-h diagram for R-410A with baseline VCC shown in green solid lines.

The SSLC1 is modelled similar to the baseline VCC except for the sensible evaporator temperature which is evaluated as shown in equation 2, and the overall cycle COP is evaluated as shown in equation 3 where the $COP_{baseline}$ is the COP of the baseline VCC.

$$T_{evap,sensible\ SSLC1} = T_s - TTD \quad (2)$$

$$COP_{SSLC1} = SHF \times COP_{sensible\ SSLC1} + (1 - SHF) \times COP_{baseline} \quad (3)$$

For SSLC2, the same model is applied as SSLC1 except that the condensing temperature is also varied as shown in equation 4 below and the COP is calculated as shown in equation 5 since the latent cooling is achieved without expending additional energy.

$$T_{cond,sensible\ SSLC2} = T_{regen} + TTD \qquad (4)$$

$$COP_{SSLC2} = COP_{sensible\ SSLC2} / \ SHF \qquad (5)$$

The SSLC3 system is evaluated similar to SSLC2, the main difference is that the condensing temperature of the VCC is kept constant at *Tambient + TTD*. Finally, the SSLC4 system has 2 components, the VCC – which is simulated similar to the sensible portion of the SSLC2 and a heat activated latent cooling component that was not modelled – but rather analysed. The primary COP of SSLC4 system is evaluated as shown in equation 6. The value of 0.3125 is the typical site-to-source conversion factor in the USA.

$$COP_{SSLC4,\ prim} = SHF \times COP_{sensible\ SSLC1} \times 0.3125 + (1 - SHF) \times COP_{regen} \qquad (3)$$

4. Results

4.1. Overall COP Comparison for Electrically Driven Technologies

Figure 3 below shows the results of comparing the COP of the first 3 SSLC cycles to the baseline. For SSLC2, multiple points exist for the same SHF representing different regenerating temperature options. It is noteworthy to see that all systems approach baseline system COP at higher SHF due to the limited impact of latent cooling on the system efficiency. Furthermore, it is shown that there exists a *Tregen* below which the SSLC2 performs better than SSLC1. Finally, Figure 3 indicates that SSLC2 is only feasible for SHF of 0.5 or higher in order to have enough waste heat from the sensible cooling system to regenerate the liquid desiccant. For SSLC3, the SHF has to be higher than 0.6 and the regeneration temperature has to be 40°C or lower to operate.

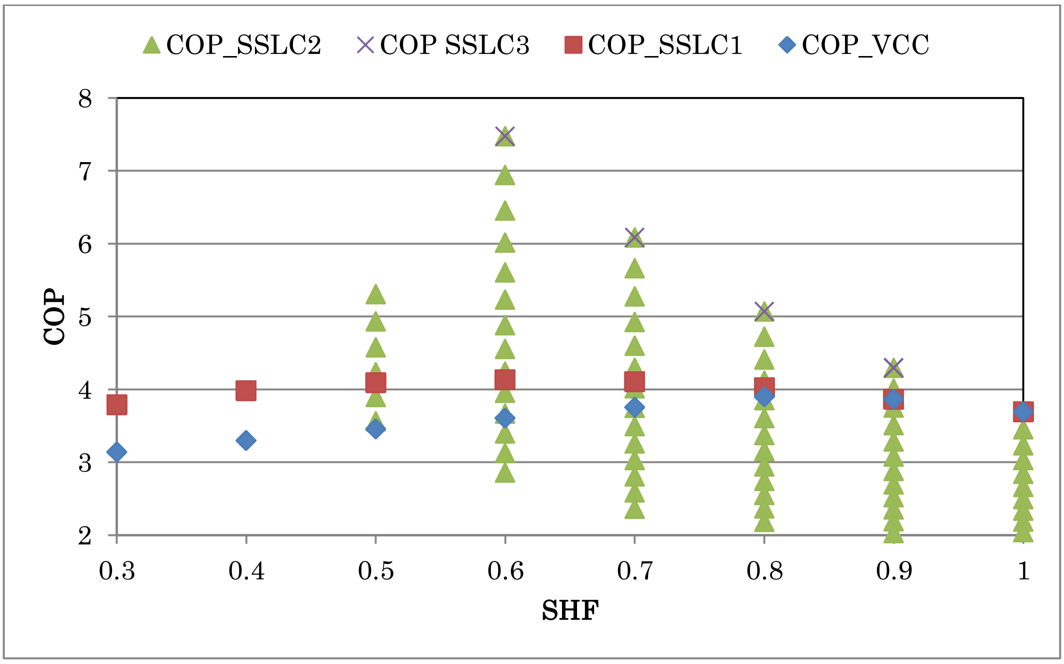

Figure 3. COP of electrically driven SSLC cycles compared to the baseline for varying SHF.

4.2. Primary COP Comparison

In order to compare SSLC4 with the remainder of the systems, we should consider the primary COP by multiplying the site COP by the site-to-source conversion factor. In the case of natural gas, the site-to-source convergence factor is 1.0 while the site to source for electricity is choses to be 0.3125 which is the average for the USA. Figure 4 shows the overall comparison. As shown in this figure, the primary COP of the SSLC4 is well below the COP of other SSLC systems. This is mainly due to the use of additional energy for regeneration of the desiccant system used for the latent load management. The relative performance between the SSLC1 and SSLC4 can be quite different if the desiccant dehumidification used is more efficient. Also, it is noteworthy to see that only SSLC1 and SSLC4 can meet the load at lower SHF. As the SHF approaches 0, the primary COP of SSLC1 approaches that of the VCC performance; there will be no difference between the 2, both of them operating at the same conditions.

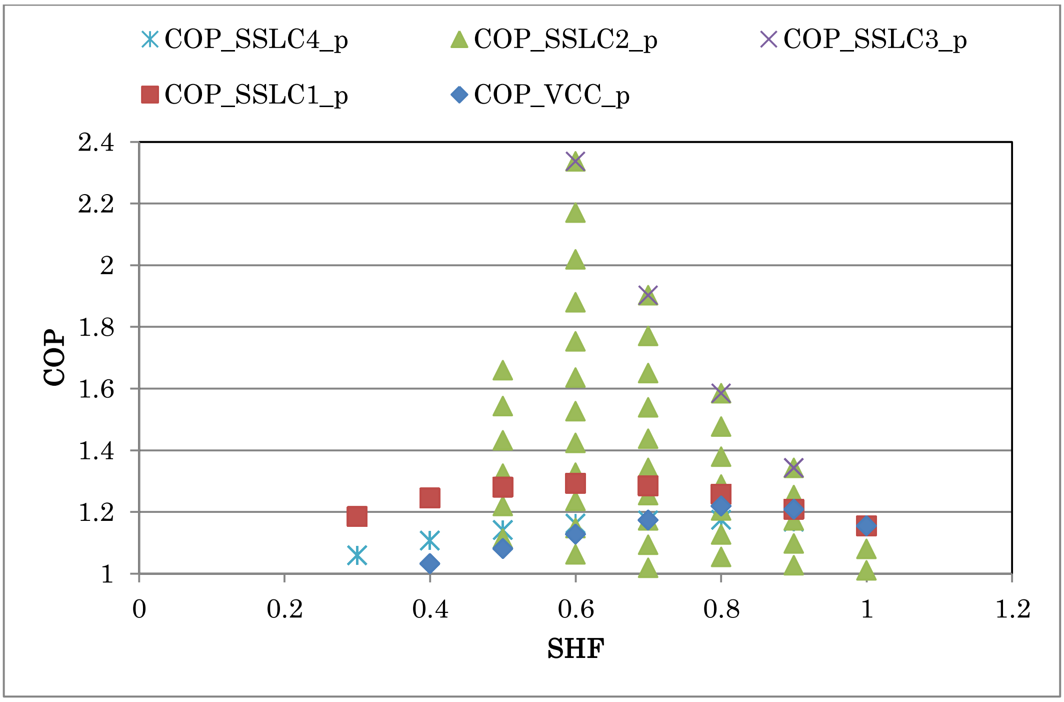

Figure 4. Primary COP of all SSLC cycles compared to the baseline for varying SHF.

4.3. Parametric Analyses

In this section, 3 parametric analyses were investigated, first the impact of compressor isentropic efficiency on the SSLC1 COP is shown in Figure 5. As shown in Figure 5 below, the COP of the SSLC1 system increases as the isentropic efficiency of the VCC compresses increases and as the SHF also increases. Maximum COP of SSLC1 system, 4.65, in this parametric study was achieved at SHF of 0.75 and isentropic efficiency of 80%

Figure 6 below, shows the impact of T_{regen} and *SHF* on the COP of SSLC2 system. As shown in Figure 6, lower T_{regen} results in higher COP and the maximum COP is achieved at a *SHF* of 0.6 with T_{regen} of 40°C

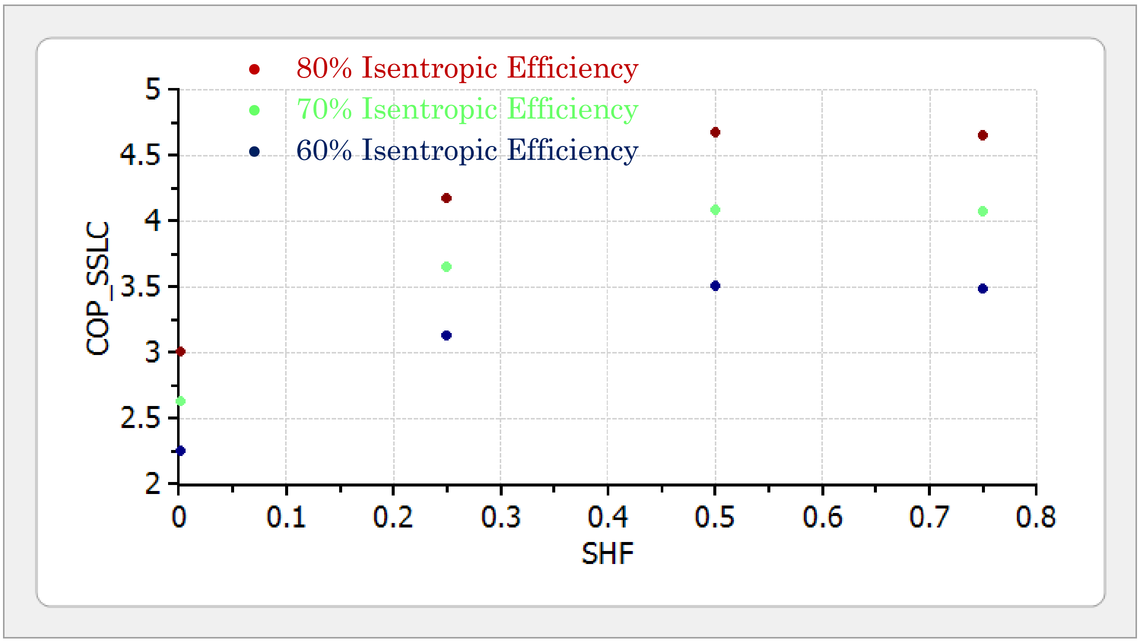

Figure 5. COP of the SSLC1 system as a function of *SHF* and compressor isentropic efficiency.

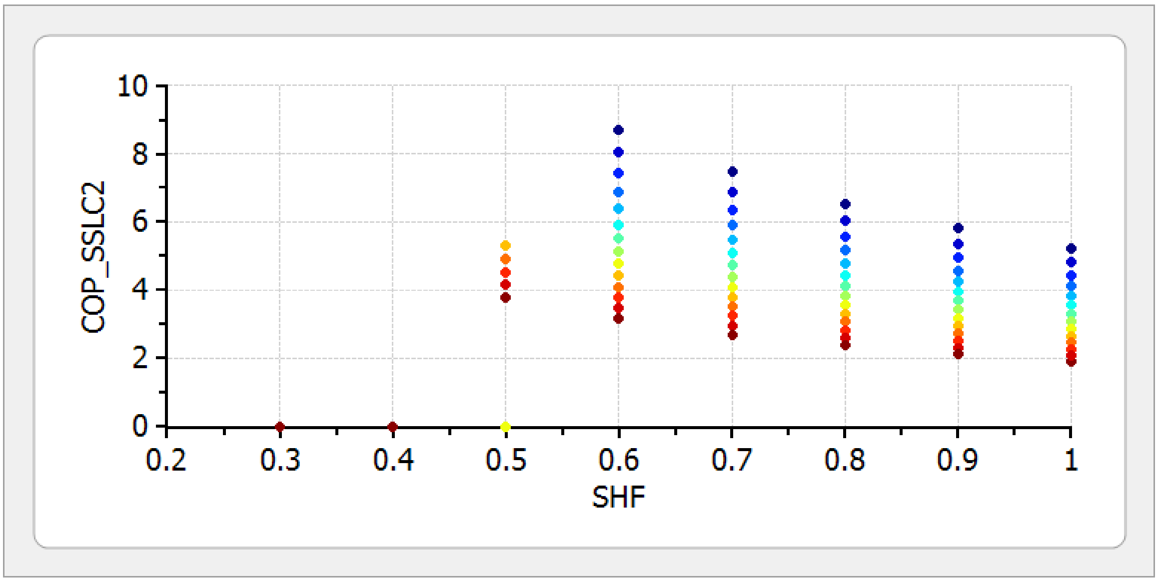

Figure 6. COP of the SSLC2 cycle as a function of *SHF* and *Tregen*.

Figure 7 shows the operating domain of the SSLC2 system inside the brown line and the operating domain in which the performance exceeds that of the SSLC1 system inside the green line. The operating domain of SSLC2 shows a trade-off a linear behaviour between SHF of 0.5 and 0.6 as the regeneration temperature drops from 56 to 40°C. As shown in Figure 7, SSLC2 is only advantageous at lower SHF and when desiccant systems with lower regeneration temperature are available.

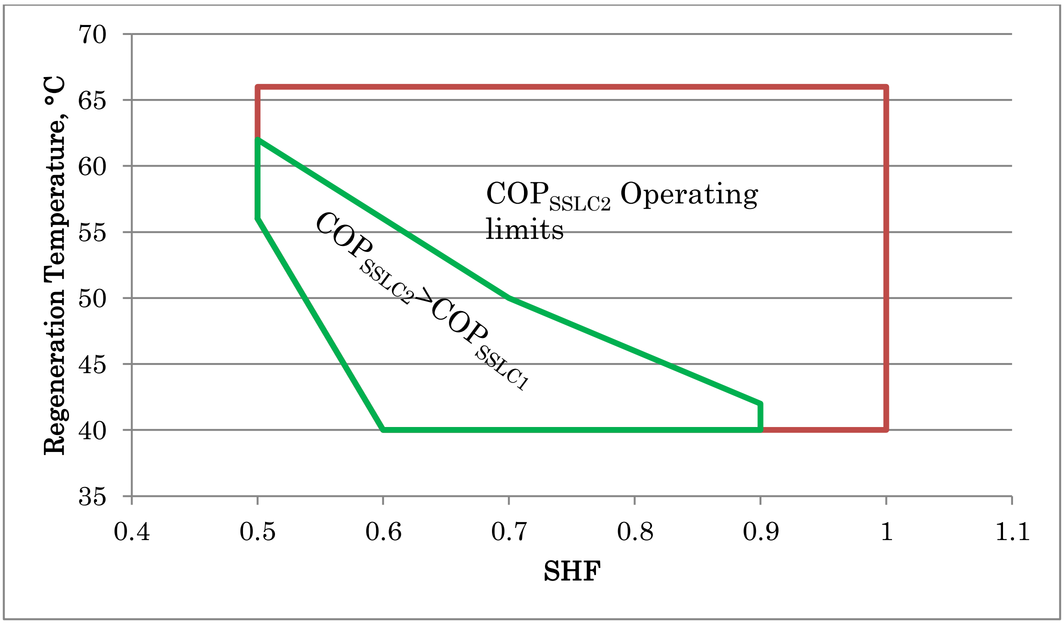

Figure 7. Operating domain for SSLC2 identifying regions where it is superior to the SSLC1.

5. CONCLUSIONS

Simplified thermodynamic cycle analyses for different separate sensible and latent cooling systems is developed and studied to identify most beneficial system performance. A liquid desiccant system with lowest possible regeneration temperature operating using the condenser waste heat of the sensible cooling sub-system is shown to provide the best performance but with limited operating conditions due to limited availability of waste heat from the condenser. The primary COP analyses show that separate sensible and cooling technology using VCC only is superior at all design conditions and that the performance of liquid desiccant systems utilizing condenser waste heat provide the best performance. In order to expand the operating domain of the most efficient system, supplemental heating can be provided to the liquid desiccant latent cooling as the operating conditions require.

6. SYMBOLS AND ACRONYMS

AC	Air Conditioning
COP	Coefficient of Performance
EES	Engineering Equation Solver
h	Enthalpy, kJ/kg
P	Pressure, kPa
Q	Capacity, kW
RH	Relative Humidity, %
SHF	Sensible heat factor
SSLC	Separate Sensible and Latent Cooling
T	Temperature, °C
TTD	Terminal Temperature Difference, K
VCC	Vapor Compression Cycle
ω	Absolute humidity, kg/kgdry air

Subscripts:

cond	Condenser
dp	Dew Point
evap	Evaporator
regen	Regeneration
s	Supply

7. ACKNOWLEDGMENT

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8. REFERENCES

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