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Techno-Economic Assessment of Heat Recovery in Series Condensers Arrangement: Hot and Humid Regions

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Abstract
A direct expansion (DX) HVAC system is an efficient way to supply fresh and dehumidified air to a built environment. To improve the efficiency of a conventional DX system in hot and humid regions, fresh air dehumidification and conditioning systems with energy recovery measures are the key equipment to reach such a goal. To achieve this goal an integrated system is proposed. The integrated system substitutes the reheat coil with an extra condenser in series arrangement with the main condenser to reheat the supply air while mixing the ventilated air with condenser cooling air. Modeling has broken down into two parts. Psychometric part which has been modeled using EnergyPlus and EES software, and DX system in which data is received via psychometric part, and has been modeled and evaluated by EES software. The case study is located in Bandar-e-Abbas city, quasi-dynamic modeling is to be conducted and results will be analyzed correspondingly. The integrated system's energy saving is 32-48%. Also, system's COP has increased from 1.55 to 3.46 for outside air fraction (ventilation rate) of 0%, and from 2.37 to 3.47 for outside air fraction of 100%. Finally, the payback period starts from 4.035 years for the outside air fraction of 0% and decrease to 2.603 years for the outside air fraction of 100%.

Keywords: Heat Recovery, Exhaust Air, Reheat Coil, Energy Saving

Introduction

Countries' development chiefly depends on energy consumption. With countries being developed, energy consumption has been increasing rapidly (Cuce and Cuce, 2017). Current researches and speculation demonstrate that this increasing fashion tends to carry on (Pérez-Lombard et al., 2008) It is scientifically proven that 20% to 40% of energy use is attributed to building sector (Cuce et al., 2016). A considerable contribution of the consumed energy in this sector is on the account of HVAC systems. With advancing through more development, desire of indoor comfort and hygiene gains proportionately (Jeong and Mumma, 2003). Ventilation is one of the key factors in DX HVAC systems energy consumption. ASHRAE standards have mandated buildings to recirculate and ventilate the indoor air in order to maintain air quality at a constant level in which the ventilation rate is depended on the building application (e.g. 10%-

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30% for residential buildings) (Warden, 2004). Almost 50% of building heat loss (Roulet et al., 2001), and 20% to 40% of the HVAC system energy consumption is allocated to mechanical ventilation in hot and humid regions (Manz and Huber, 2000). DX systems are energy intensive in addition to their coefficient of performance (COP) that deters boldly in hot and humid regions. Also, the increment of ventilation rate makes the energy consumption soars in such HVAC system (Liang et al., 2010). Besides compressor, condenser is another energy intensive component that hot and humid regions deteriorate its efficiency. Fin and tube heat exchangers are widely used in direct expansion systems as their air-cooled condensers which would encounter high flow rates if the ambient temperature rise in hot climates; with high air mass flow rates face velocity increases and as a result air-side pressure drop and fan power increase (Bell and Groll, 2011). Using heat recovery systems are the most adequate scheme to overcome ventilation related issues while retaining the system to perform properly.

Yau (Yau, 2008) proposed a novel system consisted of two heat pipe heat exchangers (HPHX). The first HPHX used to reheat the air after dehumidification and mitigated the reheat coil while the second HPHX recovered the ventilated air heat and pre-cooled the fresh air. The proposed system was able to save 27% of energy. Ramadan et al. (Ramadan et al., 2015) proposed a system that used the condenser heat to preheat the water. The results indicate that the water was preheated up to 43 degrees of centigrade while increasing the cooling load from 3.52 kW to 63. 31 kW by further cooling the condenser. Diao et al. (Diao et al., 2017) experimentally studied heat recovery characteristics of a flat micro-heat pipe array with welded, serrated and staggered fin on its surface for the ventilation. The experiment simulated summer and winter condition which led to 78% and 76.2% of maximum heat recovery efficiency respectively. Naphon (Naphon, 2010) presented a system in which the condenser was coupled to a three-row heat pipe to precool the air before entering the condenser. The proposed system COP increased by 6.4%. Zhang et al. (Zhang et al., 2018) proposed a system in which ventilated air was used to further cool the cycle's refrigerant in an auxiliary condenser and reheated the supply air by using another auxiliary condenser. They managed to reach a COP of 3.3 at an ambient temperature of 35 ℃/28 ℃.

In this study an integrated system is proposed in which the ventilated air is mixed with the condenser cooling air to make amends to the condenser efficiency in hot days and decrease its energy consumption. Also, the air that was passing through reheat coil after dehumidification process previously, is now passing through an extra condenser in series arrangement with the main condenser to further cool the refrigerant and results in more energy saving.

Material and Methods

System description

The conventional system is comprised of compressor, mixing box, expansion valve, reheat coil, condenser, evaporator and their respective fans; whereas, the integrated system includes compressor, mixing box, expansion valve, condenser, evaporator, heat exchanger and their respective fans. As it is exhibited in figures 1, the integrated system has a mixing box and a heat exchanger in addition to the conventional system while the reheat coil is eliminated compared to the conventional system. In the integrated system the ventilated air after leaving the conditioned space, enters into the second mixing box where it mixes with the outside air and by decreasing the cooling air temperature passes through the condenser coils. Also, the supply air which was previously heated by the reheat coil after leaving the apparatus, is now heated by the refrigerant that leaves the condenser, in an extra heat exchanger.

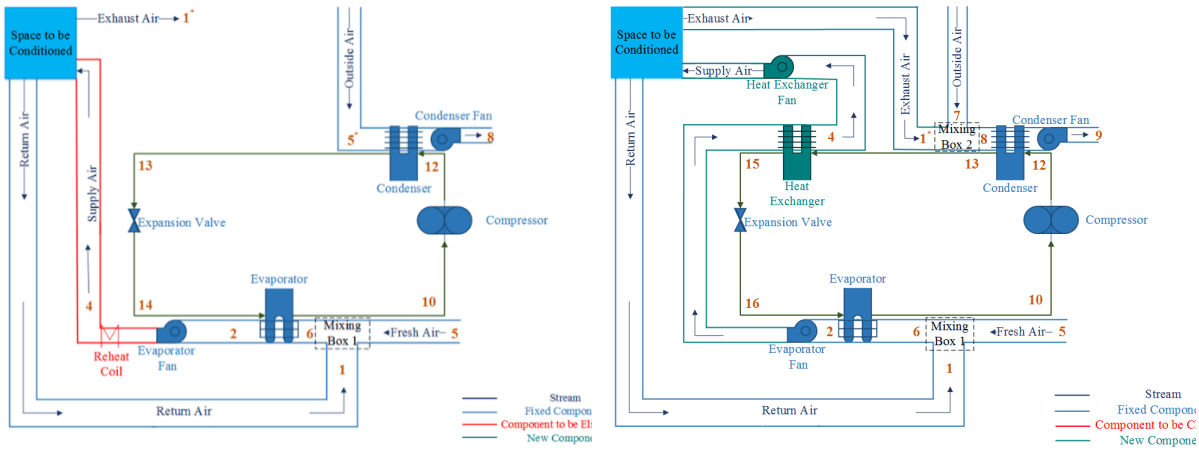

Figure 1. Conventional and integrated system structure

Modeling

The methodology of this study is splitted into two sections. First, sample building is modeled in EnergyPlus software and afterwards the psychrometric process would be modeled. Second, both the conventional and integrated systems would be modeled cell-by-cell. In other words it is fit to say that the psychrometric model connects the building and system models together.

At the first step the sample building would be modeled in EnergyPlus software, through which the historic weather data and the buildings sensible and latent loads are obtained. The sample building is an educational building located in Bandar-e-Abbas (a hot and humid city in Iran). Every building undergoes a specific psychrometric process. Figure 2 shows the sample building's psychrometric process under a typical ventilation rate.

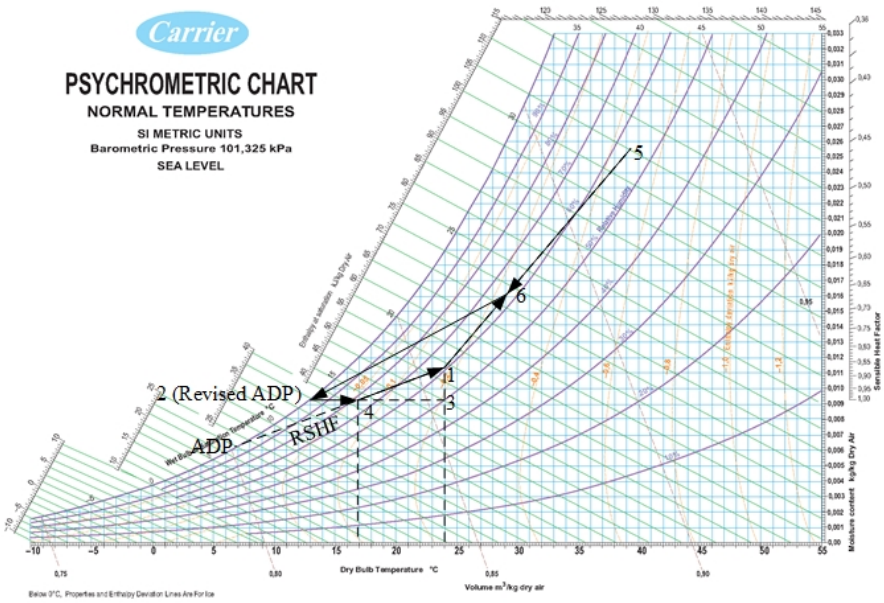

Figure 2. Sample building's psychrometric process

In the last phase, the whole HVAC system is to be modeled cell-by-cell. With attaining the cooling load from the psychrometric process model, the corresponding apparatus temperature, condenser temperature and assuming R22 as the refrigerant, the DX system would be modeled. Suction pressure is set with respect to the lowest apparatus temperature, so that the refrigerant

fully rephrased to superheat vapor and no slugs enter into the compressor to cause damage (Vakiloroaya et al., 2014a); also, the discharge pressure is the subcooled pressure at condenser temperature. It is worth mentioning that the condenser temperature is set to 50 ℃. This study mainly focuses on modeling the heat recovery process, and introduces a methodology to solve the problem of fin and tube heat exchangers implicitly. The heat recovery process is manifested in mixing the ventilated air with condenser cooling air and reheating the supply air by means of an extra heat exchanger. Finned tube heat exchangers are of most common heat exchangers used in an extended variety of applications including HVAC systems. It is mostly used in air cooled condensers in which thermal resistance rely mainly on air-side resistance that covers 90% of overall heat exchanger thermal resistance (Li et al., 2018).

The amount of the heat needed to treat the condenser load is as below:

$$Q_{con} = AU_{con}.LMTD_{con} \tag{1}$$

It is assumed that heat exchanger is counter-flow, also hot and cold streams are refrigerant and air respectively hence logarithmic mean temperature difference is defined as:

$$LMTD_{con} = \frac{(T_{h,in,con} - T_{c,out,con}) - (T_{h,out,con} - T_{c,in,con})}{\ln(T_{h,in,con} - T_{c,out,con} / T_{h,out,con} - T_{c,in,con})} \tag{2}$$

And AU_{con} is the overall heat transfer coefficient and consisted of three resistance terms: air-side, tube wall and refrigerant-side resistance. It is defined as Eq. 3, in which tube wall thermal resistance is negligible (Barbosa et al., 2009):

$$\frac{1}{AU_{con}} = \frac{1}{\eta_{o,con}.\alpha_{a,con}.A_{o,con}} + \frac{1}{\alpha_{r,con}.A_{i,con}} \tag{3}$$

Where $\eta_{o,con}$ is the surface effectiveness and is defined as below

$$\eta_{o,con} = (1 - \frac{A_{f,con}}{A_{o,con}}.(1 - \eta_{f,con})) \tag{4}$$

In which $\eta_{f,con}$ is the fin efficiency that can be calculated by approximation method proposed by Schmidt (Schmidt, 1949):

$$\eta_{f,con} = \frac{\tanh(m_{con}.r_{con}.\phi_{con})}{m_{con}.r_{con}.\phi_{con}} \tag{5}$$

Where:

$$m_{con} = \sqrt{\frac{2.\alpha_{a,con}}{k_{f,con}.t_{f,con}}} \tag{6}$$

$$\phi_{con} = \left[\frac{R_{eq,con}}{r_{con}} - 1\right].\left[1 + 0.35.\ln(\frac{R_{eq,con}}{r_{con}})\right] \tag{7}$$

$$\frac{R_{eq,con}}{r_{con}} = 0.635.\frac{VTS_{con}}{r_{con}}.\left[\frac{VTS_{con}}{2.\left((0.5.VTS_{con})^2 + (0.5.HTS_{con})^2\right)} - 0.3\right]^{0.5} \tag{11}$$

$\alpha_{a,con}$ is the air side heat transfer coefficient and is given in Eq. 12.

$$\alpha_{a,con} = \frac{j_{con}.G_{a,con}.Cp_{a,con}}{\Pr_{a,con}^{0.66}} \tag{12}$$

To obtain air-side heat transfer coefficient, Colburn j-factor is correlated by Wang et al. (Wang and Chi, 2000; Wang et al., 2000) as below:

$$j_{con} = 0.086.\left(\mathrm{Re}_{D,con}{}^{p_1}\right).\left(N_{r,con}{}^{p_2}\right).\left[\frac{P_{f,con}}{D_{c,con}}\right]^{p_3}.\left[\frac{P_{f,con}}{D_{h,con}}\right]^{p_4}.\left[\frac{P_{f,con}}{VTS_{con}}\right]^{-0.93} \tag{13}$$

Where

$$p_1 = -0.361 - \frac{N_{r,con}.0.042}{\ln\left(\mathrm{Re}_{D,con}\right)} + 0.1582.\ln\left[N_{r,con}.\left(\frac{P_{f,con}}{D_{c,con}}\right)^{0.41}\right] \tag{14}$$

$$p_2 = -1.224 - \frac{0.076.\left[\frac{HTS_{con}}{D_{h,con}}\right]^{1.42}}{\ln\left(\mathrm{Re}_{D,con}\right)} \tag{15}$$

$$p_3 = -0.083 + \frac{0.058.N_{r,con}}{\ln(\mathrm{Re}_{D,con})} \tag{16}$$

$$p_4 = -5.735 + 1.211.\ln\left[\frac{\mathrm{Re}_{D,con}}{N_{r,con}}\right] \tag{17}$$

$$\mathrm{Re}_{D,con} = G_{a,con}.\frac{D_{c,con}}{\mu_{con}} \tag{18}$$

$$D_{h,con} = \frac{4.A_{\min,con}.D_{con}}{A_{o,con}} \tag{19}$$

The refrigerant-side heat transfer coefficient is defined as $\alpha_{r,con}$ and evaluated from Gnielinski (Gnielinski, 1976) semi-empirical correlation:

$$\alpha_{r,con} = \frac{k_{r,con}}{D_{i,con}}.\left[\frac{\left(\mathrm{Re}_{i,con} - 1000\right).\mathrm{Pr}_r.\frac{f_{r,con}}{2}}{1 + 12.7\left(\frac{f_{r,con}}{2}\right)^{0.5}\left(\mathrm{Pr}_r{}^{0.66} - 1\right)}\right] \tag{20}$$

Where

$$f_{r,con} = \left(1.58.\ln(\mathrm{Re}_{i,con}) - 3.28\right)^{-2} \tag{21}$$

And $\mathrm{Re}_{i,con}$ is the Reynolds number based in tube inner diameter:

$$\mathrm{Re}_{i,con} = \frac{\rho_{r,con}.v_{r,con}.D_{i,con}}{\mu_{r,con}} \tag{22}$$

After acquiring the mass flow rate of the cooling air, the pressure drop can be calculated and as a result power consumption of the condenser would be yielded. Pressure drop is obtained from Eq. 23 (McQuiston and Parker, 1982).

$$\delta_{p,con} = \frac{G_{a,con}{}^2}{2.\rho_{i,con}}.\left[f_{a,con}.\frac{A_{o,con}}{A_{\min,con}}.\frac{\rho_{i,con}}{\rho_{avg,con}} + \left(1 + \sigma_{con}{}^2\right).\left(\frac{\rho_{i,con}}{\rho_{o,con}} - 1\right)\right] \tag{23}$$

The friction factor is correlated by Wang et al. (Wang and Chi, 2000; Wang et al., 2000) as follows:

$$f_{a,con} = 0.0267.\mathrm{Re}_{D,con}{}^{f_1}.\left[\frac{VTS_{con}}{HTS_{con}}\right]^{f_2}.\left[\frac{P_{f,con}}{D_{c,con}}\right]^{f_3} \tag{24}$$

$$f_1 = -0.764 + 0.739.\frac{VTS_{con}}{HTS_{con}} + 0.177.\frac{P_{f,con}}{D_{c,con}} - \frac{0.00758}{N_{r,con}} \tag{25}$$

$$f_2 = -15.689 + \frac{64.012}{\ln\left(\mathrm{Re}_{D,con}\right)} \tag{26}$$

$$f_3 = 1.696 + \frac{15.695}{\ln\left(\mathrm{Re}_{D,con}\right)} \tag{27}$$

Also, power consumption of the condenser fan is given in Eq. 28.

$$P_{fan,con} = \frac{\delta_{p,con}.\dot{V}_{o,con}}{\eta_{fan,con}} \tag{28}$$

Geometrical correlations used for condenser model are as follows (Vakiloroaya et al., 2014a; Vakiloroaya et al. 2014b):

$$A_{o,con} = A_{b,con} + A_{f,con} \tag{29}$$

Where

$$A_{b,con} = \left[\frac{P_{f,con} - t_{f,con}}{P_{f,con}.VTS_{con}}\right].\pi.D_{o,con}.A_{f,conr}.N_{r,con} \tag{30}$$

$$A_{f,con} = \frac{2}{P_{f,con}}.\left[HTS_{con} - \frac{\pi.D_{o,con}^2}{4.VTS}\right].A_{fr,con}.N_{r,con} \tag{31}$$

And

$$A_{i,con} = \left(D_{o,con} - 2.t_{w,con}\right).\pi.N_{t,con}.N_{r,con}.L_{con} \tag{32}$$

$$A_{\min,con} = A_{fr,con} - N_{f,con}.H_{con}.t_{f,con} - N_{t,con}.D_{o,con}.L_{con} + N_{t,con}.D_{o,con}.t_{f,con} \tag{33}$$

Economic analysis:

In this study payback period of heat recovery system, the net present value of recovery system and annual cost saving are to be studied. The term payback period means how long it takes to restore the capital cost spent on a heat recovery unit founds as below (Bejan and Tsatsaronis, 1996):

$$PBP = \frac{CI_{\text{recovery}}}{CE_L - COM_L} \tag{34}$$

Where $CI_{recovery}$ is heat recovery unit investment cost, CE_L and COM_L are levelized annual costs of electricity saving and operation and maintenance cost. Net present value (NPV) is

described as the difference of the amount of cash flowed into the system due to the presence of heat recovery system and can be determined as below (Bejan and Tsatsaronis, 1996):

$$NPV = \sum_{j=1}^{n} \frac{CE_j - COM_j}{(1+i^n)} - CI_{\text{recovery}} \tag{35}$$

Here CE_j and COM_j are the costs of electricity and operation and maintenance for the jth year. The nominal annual escalation rate is assumed to be only a function of the inflation rate. Hence, costs for the jth year can be expressed as Eq. 36 (Bejan and Tsatsaronis, 1996).

$$C_j = (1+r_i)^j \tag{36}$$

Annual levelized cost can also be expressed as (Bejan and Tsatsaronis, 1996):

$$C_L = C_0.CELF \tag{37}$$

In which CELF is found by (Bejan and Tsatsaronis, 1996):

$$CELF = \frac{k(1-k^n)}{(1-k)}.CRF \tag{38}$$

Where:

$$k = \frac{1+r_i}{1+i} \tag{39}$$

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \tag{40}$$

Results and Discussion

Simulation of the building, psychrometric and system model have been conducted quasi-dynamically for 4440 hours from April 21st to October 23rd. the mentioned time is the time that the building HVAC system is turned on, on chilling mode to response to space cooling demand.

As it is shown in figure 3 the integrated scenario experiences a considerable shift in energy consumption which is due to replacing the reheat coil with an extra condenser in series arrangement with the main one. Although excessive energy consumption associated to the heat exchanger fan would be exerted but, in comparison to that of reheat coil is trifling and make the contribution tangible enough. At the very low outside air fraction (OA%) (fresh air fraction) the effect of mixing the ventilated air with the condenser cooling air is not apparent, since there is almost no quota of energy saving but, as the outside air fraction starts to rise the chilled ventilated air mixes with condenser cooling air and by decreasing its temperature compels the condenser to reject heat with less mass flux rate and as a result, lower fan power consumption which is why in higher outside air fraction the integrated system undergoes more energy saving. Figure 4 illustrates the COP of both systems versus outside air fraction. In the conventional system with increasing in outside air fraction, the cooling load increases but, since energy consumption in reheat coil does not increases with outside air fraction, COP of the conventional system increases. In the integrated system there is a shift in the COP that is in wake of eliminating the reheat coil and substituting it with a heat exchanger. The value of integrated system's COP is in compliance with that of the Zhangs' et al. (Zhang et al., 2018) system which recovered the ventilated air heat and omitted the reheat coil in a parallel condenser arrangement. In the integrated system, COP slightly increases with increasing the outside air fraction up to 50%, and with further increasing it to 100% dwindles.

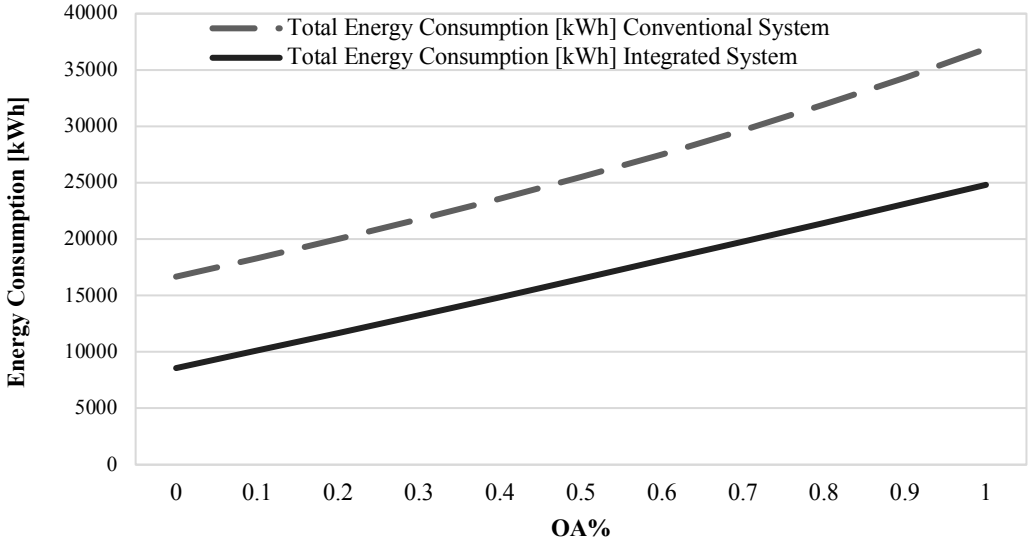

Figure 3. Total annual energy consumption [kWh]

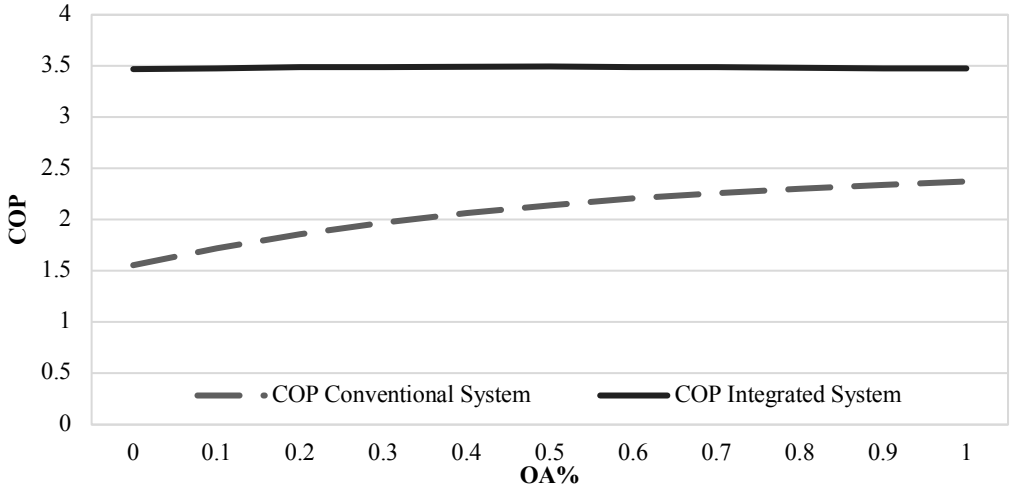

Figure 4. Coefficient of performance

Besides technical feasibility, every novel system should be justified economically; if the heat recovery strategy doesn't payback what it has been invested on it, then it is not feasible and reasonable to be industrialized. The notion that enables the system to be assessed in such a way, is net present value (NPV). Net present value (NPV) is defined as the difference of the amount of cash flowed into the system due to the presence of a heat recovery system. When the NPV reaches to zero, means that the heat recovery system has prospered the entire system as much as the invested capital on the recovery system. As it is evident in figure 5, the payback period of the heat recovery system proposed for the integrated system chiefly depends on the outside air fraction (OA%). In the case where there is no ventilation the payback period is the highest due to the capital invested on the mixing box setup which has become idle. On the other hand, by increasing the outside air fraction the ventilation rate rises; and by rising the ventilation rate the potential of recovering the ventilated air in the condenser-side mixing box magnifies that boots the payback period. The recovery potential rises due to the dependency between ventilation rate and mixing box performance. With higher ventilation rate, more exhausted heat is being recovered through the mixing box. It is worth mentioning that the system's life span is assumed to be 15 years throughout the economic analysis.

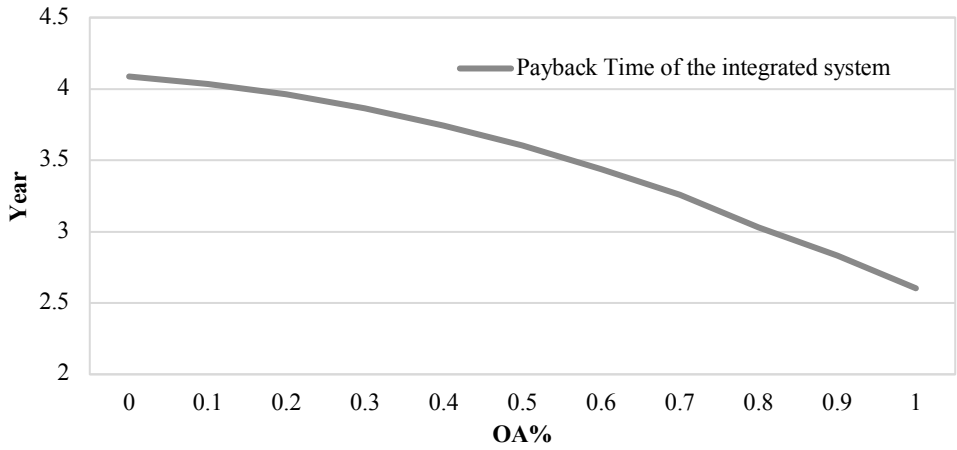

Figure 5. Payback period

Conclusion

In this study the energy, exergy and economic analysis of an integrated DX HVAC system have been assessed. The integrated system substitutes the reheat coil with an extra condenser in series arrangement with the main condenser to reheat the supply air while mixing the ventilated air with condenser cooling air. Assessing the system has resulted in the followings:

- Integrated system is capable of saving energy by 32% up to 48% depending on the outside air fraction.
- For the OA% of 0%, COP increases from 1.55 to 3.46 and for the OA% of 100%, increase from 2.37 to 3.47.
- The payback period of the heat recovery strategy implemented in the integrated system is less than roughly 4 years for any outside air fraction.

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