Diesel Engine Waste-Heat Driven Ammonia-Water Absorption System for Space-Conditioning Applications

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ABSTRACT

This paper presents the investigation of a single-effect ammonia-water absorption system driven by heat rejected from a diesel engine. The waste heat is recovered using an exhaust gas heat exchanger and delivered to the desorber by a heat transfer fluid loop. The absorber and condenser are hydronically coupled in parallel to an ambient heat exchanger for heat rejection. The evaporator provides chilled water for space-conditioning. A thermodynamic model is developed for a baseline cooling capacity of 2 kW and a detailed parametric study of the optimized system for both cooling and heating mode operation is conducted over a range of operating conditions. These parametric investigations show that degradation of system performance can be limited, and improved coefficients of performance achieved, by adjusting the coupling fluid temperature as the ambient temperature varies. With the varying return temperature, the system is able to provide the 2 kW design cooling capacity for the entire ambient temperature range investigated using heat that would normally be wasted by direct rejection to the environment.

1. INTRODUCTION

Thermally activated space-conditioning systems have several benefits. These include reduction in electrical demand during peak utility hours, reduction in operational costs, and low environmental impact due to the use of benign refrigerants. These advantages have led to a renewed interest in this technology for a variety of applications. This paper investigates one such application where an absorption heat pump is implemented to use waste heat from the exhaust gas stream of a diesel engine generator. This system leads to better utilization of source energy and is not location limited. It could be operable in remote locations, disaster relief situations, or anywhere a diesel generator is in operation. In all instances, the implementation of a waste heat recovery system would reduce the electrical demand for space conditioning.

System level modeling has been performed by various investigators for a range of capacities and configurations. It has been well established that increased system complexity will lead to improved coefficients of performance (COPs) (Cheung et al., 1996; Engler et al., 1997; Wu and Eames, 2000; Srikhiriin et al., 2001; Garimella, 2003). Engler et al. (1997) showed that maximizing heat recovery within an absorption system leads to improved system COPs. A basic single-effect cycle, a single-effect cycle with a refrigerant precooler, multiple absorber heat exchange cycles, and several Generator-Absorber Heat Exchange (GAX) cycles were investigated in this study, with COPs ranging from 0.5 for the simplest cycle to 1.08 for the most complex.

Garimella et al. (1996) investigated a GAX cycle for space-conditioning applications. Their model showed the system achieving a cooling COP of 0.925 at 35°C and a heating COP of 1.51 at 8.3°C. Double-effect and triple-
effect cycles have also been investigated and have shown high system COPs. DeVault and Marsala (1990) reviewed single-, double-, and triple-effect cycles with a focus on the development of triple-effect systems. They found that cooling COP rises from 0.77 for a single-effect cycle to 1.2 for a double-effect cycle, and to 1.41 for a triple-effect cycle at specified baseline conditions. In addition they highlight several attributes of the triple-effect cycle. These include lower operating pressures, reduced pumping power and potentially lower manufacturing costs compared to those of double-effect cycles. Grossman et al. (1994) investigated several different triple-effect configurations, including the three-condenser three-desorber design (parallel and series,) the double condenser coupled system (parallel and series,) the alternate double condenser coupled and the dual loop triple-effect cycles. COPs ranged from 1.272 to 1.729 at design conditions. They found that the double condenser coupled systems achieved higher COPs than corresponding three-condenser three-desorber systems and required no additional components, only an adjusted plumbing arrangement. Garimella et al. (1997) investigated the use of triple-effect cycles for a light commercial (35.2 kW) space-conditioning application. Their model accounted for combustion efficiencies, and heat exchange between the working fluid and the heat sources and sinks. Gas input based COPs were determined to be 0.78 and 1.4 for cooling and heating modes at baseline conditions, respectively. The study also showed that the performance of the triple-effect cycles degrades to that of a single-effect cycle at ambient temperatures above 46°C or below 5.6°C.

The increased COPs reported in each investigation above come at the cost of intricate control systems and additional components. This leads to an increased system foot print and higher capital investment. For the application in this study, the system needs to be mobile and operate with limited maintenance, in addition to requiring a low initial investment. For these reasons, system complexity was minimized with the selection of a single-effect absorption system with internal heat recovery through the use of a refrigerant precooler and solution cooled rectifier. A system of 2 kW cooling capacity is developed and investigated. The 2 kW capacity was selected assuming that the system would be coupled with an 8 kW generator where approximately 15% of the 21.3 kW energy input would be rejected as heat in the exhaust stream. Such small capacities have not received attention in previous investigations in the literature, even though there are numerous waste heat recovery opportunities at such small capacities. The lack of cost-effective compact components is one of the reasons why small capacity and residential/light commercial systems have not been implemented widely (Garimella, 2003). However, over the past several years, progress has been made in the development of compact components with high heat and mass transfer rates making small capacity systems more viable.

Garimella and co-workers have explored and developed a variety of compact heat exchanger geometries for use in absorption systems (Garimella, 2000; Meacham and Garimella, 2002, 2003, 2004; Determan and Garimella, 2011; Garimella et al., 2011; Nagavarapu and Garimella, 2011; Determan and Garimella, 2012). They have focused on the design, fabrication and demonstration of compact absorber and desorber components for falling-film configurations and have also developed an entire monolithic miniaturized absorption system with microchannel convective flow components. In the falling-film configuration, microchannel tube arrays, in alternating transversely perpendicular orientations, are stacked in a vertical column. Ammonia-water solution flows in falling-film/droplet mode on the outside of the tubes with vapor rising upwards in counterflow with the solution, while the coupling fluid flows through the microchannels. In the convective flow configuration, heat and mass exchangers consist of an array of alternating sheets with integral microscale features that form all system heat and mass exchange components in each pair of sheets in modular fashion. The microchannel features facilitate the flow of ammonia-water solution and coupling fluid while enhancing heat and mass transfer rates. Experimental studies performed on both flow configurations showed that the designs are versatile, modular and scalable, and achieved significant compactness over conventional designs and other configurations proposed in the literature. For both designs, components can be fabricated to function as the absorber, desorber, condenser, evaporator and rectifier of the system. In addition, minor changes to the tube diameter, length, and pitch in the microchannel falling-film configuration, and to the microscale features, dimensions and number of sheet pairs in the convective configuration enable ready modifications to system cooling and heating capacities.

Hu and Chao (2008b, a) have also investigated the use of microchannel technology for absorption applications. They used etched silicon wafers that function as the condenser, evaporator and expansion channel of a small scale lithium bromide-water system. The driving application for this work is microclimate control, particularly for use in hazardous environments. Pence (2010) investigated the potential for fractal flow networks to operate as the desorber of an absorption system.
The modeling investigations discussed above focused on the development of absorption systems for a range of complexities at light commercial and larger capacities. Several of the studies focused on achieving small improvements in COP, albeit with significant increases in the number of components and the associated complexity. Such systems are not particularly suitable for the small cooling capacities of interest here, and the high capital cost of complex multiple-effect systems hinders widespread adoption. Therefore, the present study focuses on small capacity systems using a simple single-effect cycle, but with the potential for a small footprint and low capital costs.

2. SYSTEM DESCRIPTION

A schematic of the diesel engine waste heat driven single-effect ammonia-water absorption heat pump investigated in the present study for cooling mode operation is shown in Figure 1. The system consists of the main absorption loop and auxiliary coupling loops for the absorber, condenser and desorber. It is assumed the evaporator is connected to an auxiliary loop that supplies the space-cooling load. Ammonia-water is selected as the working fluid for this investigation because the Lithium Bromide-Water pair is limited by the freezing point of water, which does not allow heating mode operation. Lithium Bromide-Water is also prone to crystallization at high temperatures. In addition, the high specific volume of water vapor at evaporator conditions leads to excessive pressure drops, effectively ruling it out for compact systems.

2.1 Ammonia-Water Loop

Referring to Figure 1, concentrated solution reaches saturation in the absorber at (1). The solution is then subcooled before exiting the absorber at (2). The solution is then pumped to the high-side system pressure across the pump, from (3) to (4). The rectifier is cooled by this concentrated solution stream, states (5) to (6). The solution is then recuperatively heated in the solution heat exchanger, (7) to (8), before mixing with the rectifier reflux at (9). Concentrated solution enters the desorber where the generated vapor flows counter-current to the falling solution. Dilute solution exits the desorber at (11) while the generated vapor exits at (12). The dilute solution rejects heat to the concentrated solution stream across the solution heat exchanger, (13) to (14), before flowing across an expansion device and mixing with refrigerant vapor in the absorber inlet at (28).

Vapor that exits the desorber flows to the rectifier, (15), where water is selectively condensed to increase refrigerant vapor purity. The liquid reflux exits the rectifier at (16) and mixes with the concentrated solution stream at (9). Refrigerant vapor exits the rectifier at (17) and flows into the condenser at (18). The refrigerant is condensed to a saturated liquid at (19) and exits as a slightly subcooled liquid at (20). The refrigerant is further subcooled in the refrigerant precooler, (21) to (22), before flowing through an expansion device. Cooling is provided by the refrigerant stream in the evaporator as it evaporates from (23) to (24). Upon exiting the evaporator, the refrigerant recuperatively cools the same stream exiting the condenser in the refrigerant precooler, (25) to (26). The refrigerant then enters the absorber at (27) and mixes with the dilute solution at (28). The refrigerant is absorbed in the dilute solution stream to complete the cycle.

2.2 Auxiliary Loops

The absorber and condenser are coupled hydronically in parallel to an ambient heat exchanger. The ambient air is heated across this heat exchanger from (37) to (38) while the coupling fluid is cooled from (35) to (36). At the outlet of the ambient sink, the hydronic fluid is split with half of it flowing through the absorber, (31) to (32) and the other half flowing through the condenser, (33) to (34). A parallel configuration was selected because it limits the ammonia-water high-side operating pressures while still allowing for high system performance.

A heat transfer fluid loop couples the desorber with the exhaust gas stream. The gas rejects heat from (43) to (44) in an exhaust gas heat exchanger and heats the heat transfer fluid, (39) to (40). The heat transfer fluid (Paratherm NF) rejects heat in the desorber, (41) to (42), and allows for the desorption of refrigerant from the ammonia-water solution. In the evaporator, the coupling fluid is cooled from (29) to (30), and is then used to provide space conditioning. In the heating mode, the auxiliary loop configuration is modified to allow for the absorber and condenser to provide space conditioning (Figure 2), while the evaporator is connected to the ambient heat exchanger. During heating mode operation, heat is pumped from a low temperature at the ambient heat exchanger to a high temperature at the exit of the condenser and absorber to supply the required heating load in the conditioned space.
3. MODELING APPROACH

The system described above was modeled using the Engineering Equation Solver (EES) (Klein, 2010) platform. Mass, species, and energy conservation equations were used to analyze each component in the system. As ammonia-water is a binary mixture, three independent properties were required to establish each state point. Heat transfer resistances were taken into account with the specification of overall heat transfer conductance $U_A$s for each heat exchanger. Baseline values for the $U_A$s for each component were initially calculated by using reasonable assumptions for the closest approach temperature (CAT) or heat exchanger effectiveness for each component. The resulting $U_A$ values were then used as specifications for the system model. After analyzing the baseline system, parametric analyses were conducted to maximize the system COP. System response to changes in $U_A$ values and other key inputs was assessed to achieve progressive improvements in COP. Each parameter was varied ±15% while the remaining inputs were held constant. Plots of system response to variations in each parameter were used to select the final $U_A$s and other key parameter values. The set of parameter values selected based on these analyses were then used to understand the effect of operating conditions and other settings such as solution and hydronic fluid flow rates on system performance. Because $U_A$ values are representative of component sizes, the $U_A$s were used as indicators of capital cost, and components with low sensitivities were assigned relatively lower values while still not adversely affecting system performance.

For the baseline system, the ambient, exhaust inlet, and absorber and condenser or evaporator coupling fluid inlet temperatures were set depending on the mode of operation. The coupling fluid flow rates and the concentrated solution flow rate were also set at predetermined values to represent the operation of an actual system. Other flow rates such as the dilute solution and refrigerant flow rates were computed from the system model based on these specifications and the respective mass, species, and energy balances.

The inlet temperature and flow rate of the coupling fluid for the evaporator were set at 12.8°C and 0.0983 kg s$^{-1}$, respectively. A 1.5°C glide was assumed on the refrigerant side – lower glides lead to substantially incomplete evaporation and cooling duties, while higher glides result in temperature pinches with the coupling fluid that also

![Figure 1: System schematic (cooling mode)](image1)
![Figure 2: System schematic (heating mode)](image2)
limit cooling capacity or raise the temperature at which cooling can be delivered. An ambient air temperature of 35°C and an air flow rate of 0.4685 kg s⁻¹ were selected. The exhaust gas entered the auxiliary heat exchanger at a temperature of 398°C and flow rate of 0.05 kg s⁻¹, and exited above the flue gas water fraction condensation temperature. A summary of these parameters is presented in Table 1.

In the base case for heating mode, the inlet temperature of the coupling fluid for the absorber and condenser was set at 37.8°C, with a total flow rate of 0.09506 kg s⁻¹, and the flow being split evenly between the two components. Ambient air at a temperature of 8.33°C and a flow rate of 0.51284 kg s⁻¹ was used as the low temperature source. A summary of these parameters is presented in Table 2.

4. BASELINE SYSTEM RESULTS

Baseline conditions were established for both cooling and heating mode applications using the inputs and assumptions discussed above. In cooling mode, the concentrated solution flow rate was set at 0.0056 kg s⁻¹, and the dilute solution and refrigerant flow rates were calculated to be 0.00372 and 0.00188 kg s⁻¹, respectively. The concentrated solution, dilute solution, and refrigerant concentrations were 50.05, 24.90 and 99.86%, respectively. The high and low pressures of the system were 1987 and 516 kPa, respectively. The evaporator received chilled fluid (25% propylene-glycol solution) at a temperature of 12.8°C and delivered it at 7.23°C. The low grade waste heat from the absorber and condenser was rejected to a coupling fluid loop and ultimately the surrounding air, which was heated across the ambient heat exchanger from 35°C to 45.79°C. The system provided 2.167 kW of cooling at a coefficient of performance of 0.6948. The cooling COP was computed as follows.

\[
\text{COP}_{\text{Cooling}} = \frac{Q_{\text{Evaporator}}}{Q_{\text{Desorber}}}
\]  

(1)

In the baseline case for the heating mode, the concentrated solution flow rate was set at 0.0056 kg s⁻¹, while the dilute solution and refrigerant flow rates were 0.00387 and 0.00173 kg s⁻¹, respectively. The concentrated solution, dilute solution, and refrigerant concentrations were 46.45, 22.58 and 99.75%, respectively. The system operated at high and low side pressures of 2058 and 445 kPa, respectively. The condenser and absorber received water at 37.8°C and returned it at 51°C. The system provided 5.039 kW of heating at a COP of 1.665. The heating COP was calculated as follows.

\[
\text{COP}_{\text{Heating}} = \frac{Q_{\text{Absorber}} + Q_{\text{Condenser}}}{Q_{\text{Desorber}}}
\]  

(2)

### Table 1: Cooling mode design specifications

<table>
<thead>
<tr>
<th>Component</th>
<th>UA [W K⁻¹]</th>
<th>Coupling Fluid</th>
<th>Duty [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Flow Rate [kg s⁻¹]</td>
<td>Inlet Temperature [°C]</td>
</tr>
<tr>
<td>Absorber</td>
<td>300</td>
<td></td>
<td>0.0635</td>
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<tr>
<td>Ambient Heat Exchanger</td>
<td>2400</td>
<td>Air</td>
<td>0.4685</td>
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<tr>
<td></td>
<td></td>
<td>CF</td>
<td>0.127</td>
</tr>
<tr>
<td>Condenser</td>
<td>350</td>
<td></td>
<td>0.0635</td>
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<tr>
<td>Desorber</td>
<td>80</td>
<td></td>
<td>0.05</td>
</tr>
<tr>
<td>Evaporator</td>
<td>1000</td>
<td></td>
<td>0.0983</td>
</tr>
<tr>
<td>Exhaust Heat Exchanger</td>
<td>31.7</td>
<td></td>
<td>0.05</td>
</tr>
<tr>
<td>Rectifier</td>
<td>11.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refrigerant PreCooler</td>
<td>39</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solution Heat Exchanger</td>
<td>100</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pump</td>
<td>11.24</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Bold values are calculated based on the set parameters.
5. PARAMETRIC ANALYSES

System performance for cooling and heating modes was investigated for a range of ambient conditions. Cooling mode performance was investigated for an ambient temperature range of 20 to 49°C, while heating mode performance was investigated from -19 to 10°C. In addition, system response to changes in chilled and heated water temperatures was investigated. For each mode, a standard and an adjusted case was investigated. In the standard cases, only the ambient temperature was varied. In the adjusted cooling mode case, above an ambient temperature of 35°C, a 1°C increase in the chilled water inlet temperature was made for every 1°C ambient temperature increase to limit cooling capacity decrease at the cost of a slight increase in chilled water temperature. Similarly, in the adjusted heating mode case, a 1°C decrease in heated water inlet temperature was made for every 5°C ambient temperature decrease. This temperature adjustment in the heating mode was made over the entire ambient temperature range, not just beyond a threshold ambient temperature. It should be noted that these adjustments move the cooling and heating delivery temperatures away from typical standards but were deemed acceptable based on the ambient conditions and improvements in system level heating and cooling loads and COPs. Figure 3 shows how the variation in chilled water temperature at the evaporator inlet and outlet with changes in the ambient temperature for both cooling mode cases. For the standard case, the temperature difference between the inlet and outlet decreases as the ambient temperature increases, indicating a reduction in system capacity. At an ambient temperature of 35°C, the temperature difference is 5.6°C, while at 48.9°C, it is 4.4°C. In the adjusted case, the chilled water outlet temperature variation is almost parallel to the chilled water inlet temperature variation. In this case, the temperature difference at 48.9°C is 5.3°C. Figure 4 shows the heated coupling fluid inlet and outlet temperature variations in the heating mode. Both cases show a reduction in temperature difference between the inlet and outlet indicating a decline in system performance. The adjusted case maintains a higher temperature difference, showing that degradation of performance is limited.

5.1 Cooling Mode Results

Figure 5 shows a plot of the coefficients of performance for both cooling mode cases. It can be seen that both cases follow the same COP trend up to an ambient temperature of 35°C. At temperatures greater than 35°C, the unadjusted system continues to show a decrease in COP with increased ambient temperature. The increased ambient temperature decreases the ability of the system to reject heat; therefore, to reject the same amount of heat at a higher ambient, the system must operate at higher pressures. In both cases, the low side pressure is set by the temperature of absorption. As the coupling fluid inlet is held constant in the evaporator, the low side pressure cannot increase as needed for the same concentration. A reduction in the ammonia fraction in the concentrated solution is required. For the unadjusted case, at an ambient of 48.9°C, the concentrated solution concentration is reduced to 43%. Lower concentrations allow for absorption to occur at higher temperatures while maintaining the desired low side pressure and evaporator temperature. However, the lower concentration results in a reduction in driving temperature difference at the desorber and in the refrigerant vapor generated, which decreases cooling capacity. The adjusted system where the chilled water temperature is allowed to rise does not experience the same reduction in

<table>
<thead>
<tr>
<th>Component</th>
<th>UA [W K⁻¹]</th>
<th>Flow Rate [kg s⁻¹]</th>
<th>Inlet Temperature [°C]</th>
<th>Duty [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absorber</td>
<td>300</td>
<td>0.04753</td>
<td>37.8</td>
<td>3,110</td>
</tr>
<tr>
<td>Ambient Heat Exchanger</td>
<td>2400</td>
<td>Air</td>
<td>0.51284</td>
<td>8.33</td>
</tr>
<tr>
<td>Condenser</td>
<td>350</td>
<td>0.04753</td>
<td>37.8</td>
<td>1,929</td>
</tr>
<tr>
<td>Desorber</td>
<td>80</td>
<td>0.05</td>
<td>179.6</td>
<td>3,026</td>
</tr>
<tr>
<td>Evaporator</td>
<td>1000</td>
<td>0.12921</td>
<td>7.59</td>
<td>1,997</td>
</tr>
<tr>
<td>Exhaust Heat Exchanger</td>
<td>31.7</td>
<td>0.05</td>
<td>156.4</td>
<td>3,026</td>
</tr>
<tr>
<td>Rectifier</td>
<td>11.5</td>
<td>395</td>
<td></td>
<td>245</td>
</tr>
<tr>
<td>Refrigerant PreCooler</td>
<td>39</td>
<td>245</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solution Heat Exchanger</td>
<td>100</td>
<td>1,646</td>
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</tr>
<tr>
<td>Pump</td>
<td></td>
<td>12.12</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Bold values are calculated based on the set parameters.
performance. The increased chilled water temperature allows the system to reject additional heat to the ambient when compared to the unadjusted system. This is because the low side pressure is able to rise, allowing the absorber to operate at higher capacities and concentrations. In this case, at 48.9°C, the concentrated solution concentration is 52%. This higher concentration enables an increased refrigerant generation. For the adjusted case, system COP is maintained around 0.69 as the chilled water temperature is increased above 35°C. In the unadjusted case, system COP decreases to 0.60 at the highest ambient temperature.

The cooling load provided by both systems follow a trend similar to that of the COPs (Fig. 6.) as expected. For the adjusted case, the system is able to maintain cooling loads above 2.0 kW for the entire range of ambient temperatures investigated, satisfying the goal of providing 2.0 kW of cooling from waste heat across the ambient temperature range of interest. The absorber and condenser loads remain fairly constant for the adjusted case above 35°C but decrease continuously for the unadjusted case. The higher chilled water temperature and the somewhat higher resulting space-conditioning temperature are the penalties for this improved system capacity.

This analysis showed that the high-to-low side pressure difference is reduced when the chilled water temperature is allowed to rise with increasing ambient temperature (Fig 7.) At 35°C, the differential pressure for both cases is 1472 kPa. However, at 48.9°C, the differential pressure for the unadjusted case is 2109 kPa, but only 1933 kPa for the adjusted case. The increased pressure difference affects both the high and low side components. An elevated high-side pressure can reduce the vapor generated in the desorber. Higher low-side pressures can limit evaporator performance and solution concentration.
5.2 Heating Mode Results

Figures 8 and 9 show plots of the heating mode COPs and duties for the two cases investigated, respectively. System performance decreases for both cases but to different extents. The decreasing ambient temperature limits the amount of ambient heat that can be extracted from the ambient, which leads to the decreasing COPs. Delivering heat at the specified temperature limits the heat duty that can be supplied by the evaporator. The low temperature in the evaporator has the potential to decrease heated water delivery temperatures if not restricted. The system restricts refrigerant flow to the evaporator by decreasing the solution concentration. As the solution concentration is decreased, vapor generation in the desorber decreases. At an ambient of -5.6°C, the concentrated solution concentrations for the unadjusted and adjusted cases are 0.374 and 0.387, respectively. For the adjusted case, allowing a somewhat lower coupling fluid inlet temperature to the absorber and condenser improves performance because it decreases low-side and high-side pressures, providing an increased driving temperature difference at the evaporator and in the generated refrigerant vapor flow rate. The tradeoff for this increased duty is the reduced coupling fluid temperature, which leads to a modest decrease in the conditioned space temperature.

System pressure variations for the two heating mode cases are presented in Figure 10. Adjusting the coupling fluid temperature leads to reduced high side pressure and thus, the low-to-high side pressure difference. For both cases, low side pressure decreases with a decrease in ambient temperature. At lower ambient temperatures, less refrigerant is generated; this decreases the condenser duty and limits the coupling fluid temperature rise across the component. As a result, the condensation temperature decreases, lowering the high-side pressure. The absorber duty is also decreased due to a lower refrigerant flow rate, which in turn lowers the working fluid temperature in the absorber and the low-side pressure.

Figure 7: System pressure for cooling mode cases

Figure 8: Heating mode COPs

Figure 9: Heating mode loads
6. CONCLUSIONS

A detailed investigation of a small capacity diesel engine waste heat driven ammonia-water absorption system was performed for both cooling and heating mode applications. The primary components were hydronically coupled, leading to a compact package using microscale heat and mass exchangers (Determan and Garimella, 2012) with versatile implementation and installation possibilities. The baseline system can achieve a cooling $COP$ of 0.695 at an ambient temperature of 35°C and a heating mode $COP$ of 1.66 at an ambient temperature of 8.33°C. System performance was analyzed over a wide range of ambient temperatures, with different set point adjustments designed to improve cooling and heating loads, which led to high $COP$s in both modes, even at extreme ambient temperature conditions. Degradation of system performance is mitigated by adjusting water delivery temperatures, thus maintaining high cooling or heating loads even at extreme ambient temperatures. The tradeoff of achieving somewhat higher delivered temperatures to the conditioned space in the cooling mode, and somewhat lower delivered air temperatures to the conditioned space in the heating mode is deemed acceptable in the interest of maintaining high space-conditioning duties and coefficients of performance. The system under consideration here represents one of the first small-capacity options available for waste heat recovery with high performance over a wide range of operating conditions, and a potential for a compact envelope and low capital costs.

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