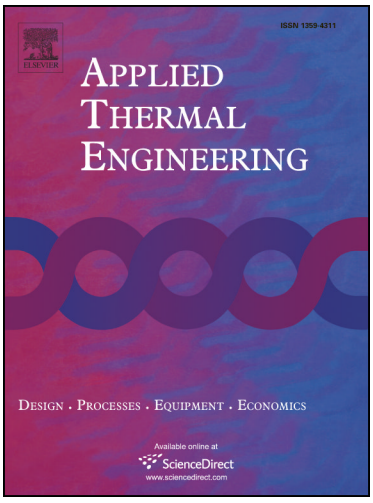

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PII: S1359-4311(18)35100-7
DOI: https://doi.org/10.1016/j.applthermaleng.2018.12.151
Reference: ATE 13158

To appear in: *Applied Thermal Engineering*

Received Date: 17 August 2018
Revised Date: 28 December 2018
Accepted Date: 29 December 2018

Please cite this article as: C. Blackman, K.R. Gluesenkamp, M. Malhotra, Z. Yang, Study of Optimal Sizing for Residential Sorption Heat Pump System, *Applied Thermal Engineering* (2018), doi: https://doi.org/10.1016/j.applthermaleng.2018.12.151

Study of Optimal Sizing for Residential Sorption Heat Pump System

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ABSTRACT

Gas-driven sorption heat pumps (GDSHP) show significant potential to reduce primary energy use, associated emissions and energy costs for space heating and domestic hot water production in residential applications. This study considered a bivalent heating system consisting of a sorption heat pump and a condensing boiler, and focuses on the optimal heating capacity of each of these components relative to each other. Two bivalent systems were considered: one based on a solid chemisorption cycle (GDSHPA), and one based on a resorption cycle (GDSHPB). Simulations of year-round space heating loads for two single family houses, one in New York and the other Minnesota, were carried out and the seasonal gas coefficient of performance (SGCOP) calculated. The sorption heat pump's design heating capacity as a fraction of the bivalent system's total heating capacity was varied from 0 to 100%. Results show that SGCOP was effectively constant for sorption heat pump design capacity greater than 41% of the peak bivalent GDSHPA design capacity in Minnesota, and 32% for GDSHPB. In New York, these values were 42% and 34% for GDSHPA and GDSHPB respectively. Payback period was also evaluated based on postulated sorption heat pump component costs. The fastest payback was achieved with sorption heat pump design capacity between 22–44%.

Keywords: Sorption heat pump, sizing, residential, bivalent

Highlights:

- The optimum design heating capacity of the sorption heat pump ranges between 22–44% of peak capacity.
- Two climates, two energy price scenarios, and two sorption system types were evaluated
- A detailed building model was used to calculate hourly building loads

Nomenclature

ACH	air changes per hour ventilation rate
AHU	air handling unit
CE	combined condenser-evaporator vessel
f	function
f_2	function
GCOP	gas coefficient of performance
GDSHP	gas driven sorption heat pump with boiler and sorption module
GDSHPA	gas driven sorption heat pump with boiler and type "A" chemisorption module
GDSHPB	gas driven sorption heat pump with boiler and type "B" resorption module
HTS	high temperature salt
LTS	low temperature salt
Q	heating capacity of GDSHP system
R	reactor vessel containing low temperature salt (in chemisorption type sorption module)
RA	reactor vessel A containing high temperature salt (in resorption type sorption module)
RB	reactor vessel B containing low temperature salt (in resorption type sorption module)
SCOP	seasonal coefficient of performance
SGCOP	seasonal gas coefficient of performance
SM	sorption module
SMA	sorption module Type A (chemisorption-based)
SMB	sorption module Type B (resorption-based)
T_{out}	outdoor ambient temperature
T_{supply}	temperature heat transfer fluid supplied to air handling unit
ΔT_{lift}	temperature lift of sorption module (supply temperature minus outdoor temperature)
η	thermal efficiency (energy out as fraction of energy into a component)

1. INTRODUCTION

Sorption heat pumps may use the combustion of natural gas as a heat source to drive a sorption cycle. This cycle effectively pumps heat from a low temperature source to a sink of higher temperature allowing for more effective gas utilisation for space and domestic water heating. Cycle heating efficiencies as high as 140% (i.e. COP 1.4) for gas-driven sorption heat pumps (GDSHP) are not uncommon [1].

A bivalent heat pump system consists of a heat pump unit and a supplemental heating source. This study considers a GDSHP integrated with a supplementary condensing boiler. Since, in general, the peak building demand conditions occur for relatively few hours of the year, the more costly sorption module should be dimensioned to avoid inefficient operation at low part loads, whilst cost effectively maximising the seasonal efficiency defined by the seasonal gas coefficient of performance (SGCOP). Additionally, this approach opens the possibility of cost-effective GDSHPs tailor-sized for specific climates.

Based on the working principle, heat pumps can be categorized as vapor-compression (including electric-driven and gas-driven) or vapor-absorption heat pumps [2]. The majority of heat pumps deployed today, and the majority covered in the literature, are of the vapor-compression type. The most common bivalent heat pump system for residential and small commercial applications is the air-to-air vapor compression heat pump with electric resistance as the supplemental heat source, using electricity from the grid. However, other configurations have been investigated. These include various heat pump heat sources (ground-source, solar thermal, and combinations), supplemental heating type (gas-fired condensing boiler, solar thermal), and electricity source (solar PV, fuel cell) [3–5]. A large number of studies are available on vapor compression heat pumps; while vapor-absorption

heat pumps, having low environmental impact, have gained considerable attention in recent years [6].

For lower installed cost, heat pump systems are configured with the heat pump unit (higher cost and higher efficiency) sized for less than peak building loads and the supplemental heating source (typically, lower cost and lower efficiency) sized to meet the peak heating demand. It is widely known that, when sized carefully and strategically operated and controlled, the use of supplemental heating source can also improve annual efficiency and lower annual operating cost. This is because heat pump efficiency can suffer at very low part loads, and using a supplemental heating source avoids over-sizing of the heat pump component [7,8].

Various studies have investigated the sizing and operation strategies for heat pump systems. In these studies, the key sizing and/or operating parameters investigated include: (a) the *capacity of heat pump* with respect to the building load, expressed in relation to the *bivalent temperature* — the ambient temperature at which the heat pump output matches the building load [7–10]; (b) the bivalent operation mode: either *bivalent-parallel* when the heat pump continues to operate in parallel with the auxiliary heat source or *bivalent-alternative* when only the auxiliary heat source would operate at temperatures below the bivalent temperature [10]; (c) the *cut off temperature*—the ambient temperature below which the heat pump unit is switched off, thus only running the auxiliary heat source [10]; (d) heat pump control—on-off versus modulating control [8–11]; (e) sizing of thermal storage [7,11].

Several studies analyzed different heat pump types in different climates with different sets of operating parameters. Most importantly, the system performance is most sensitive to the sizing of the heat pump (thus, the bivalent temperature). Additionally, improved system performance results from bivalent-parallel operation over bivalent-alternative operation [11], and from modulating control over on-off control [8–10]. For larger sized heat pump systems (that is, low bivalent temperatures), the performance improvement from a lower cut-off temperature relative to bivalent temperature is negligible [11]. Similarly, the performance change from varying the size of thermal storage in case of systems with water as the heat sink was marginal to none [7]. The analysis in this study assumes bivalent parallel operation, no cycling losses associated with on-off operation, and no thermal storage (such as a condensing boiler tank).

In most studies that have evaluated the sizing or operation strategy of a specific heat pump system, the evaluation criteria typically include: the annual or seasonal system efficiency, and/or the operating cost (especially for systems using two different fuels). Only a few studies have factored in the installed cost of the system in terms of total cost of ownership [12] and levelized cost of energy [13]. Installed costs of the system are accounted for in studies which compared different types of heat pump systems, for example, Busato et al. [14] compared net present worth and payback period for an electric heat pump, a natural gas engine-driven heat pump, and a natural gas boiler, and Aste et al. [15] compared net present costs for air-to-air heat pumps, air-to-water heat pumps, ground-source heat pumps, and ground-water heat pumps.

Due to the modular nature of the sorption component, which is the costlier component in this system, the sizing of the sorption heat pump system (i.e., the sorption modules vs auxiliary heat source—the condensing boiler) requires the optimization criteria to also account for the capital cost. Therefore, this paper seeks to fill a gap by exploring the optimal sizing of

sorption heat pump and backup heating system, since literature review yielded few studies on residential-scale sorption systems and no sizing studies for sorption heat pumps. This work aims to develop a sizing strategy for bivalent sorption heat pumps considering the similarities to the sizing of bivalent electric heat pumps.

In the present study, we consider a parallel bivalent system, in which both the heat pump and boiler component are capable of simultaneously providing heat at temperatures between the heat pump cut-off temperature and the bivalent temperature. Furthermore, two modular sorption unit types are considered with two different sorption cycle principles. One sorption module (SM) based on solid chemisorption cycle principle (type A) and the other a resorption cycle (type B), both employing ammonia as the working fluid. For this study, simplified performance correlations for each SM type were used to determine the optimum capacity of the SM implemented in a bivalent GDSHP. Optimum capacity was determined based on both cost and performance indicators. Two climate zones in the USA were studied, representative of a cold (Minnesota) and a moderate (New York) climate.

2. Sorption Modules

This study considers a sorption module, defined as a fully modular sorption heat pump device. The sorption module can operate based on an absorption or adsorption cycle principle as described by Blackman et al. [16]. It is modular because it can be combined with a backup boiler, and the sorption heat pump sizing is continuously variable: by virtue of its design, it can be sized and/or coupled together with another sorption module without need for significant re-design. This allows it to be readily designed for any capacity. These modules can be employed for heating and/or cooling purposes and operate at ambient temperatures below 0°C due to the use of ammonia as a refrigerant [17,18].

2.1. Basic Sorption Module Component (Type A)

The sorption component referred to in this work as "Type A" is a sorption heat pump module component comprising two cylindrical vessels, the reactor (R) and the combined condenser-evaporator (CE). The vessels each house a heat exchanger made up of disc-shaped steel plates engineered in such a way as to allow a heat transfer fluid to flow within the discs. The heat exchanger of the reactor provides thermal energy to, or removes thermal energy from, a matrix material infused with a salt (alkali halide) which reversibly absorbs ammonia. The matrix is housed, in good thermal contact, between the heat exchanger discs (see Figure 1). The CE is identical to the R except that the matrix material between the heat exchanger discs does not contain salt. The R and CE vessels have an opening at the top to allow ammonia to flow back and forth via a connecting pipe.

The Type A sorption module (SMA) operates in a batch process with two main modes of operation; absorption and desorption [19]. In absorption mode, the difference in vapor pressure between the salt and the ammonia causes ammonia to evaporate from the CE (operating as an evaporator) and form a salt ammoniate in the reactor. The reaction between salt and ammonia creates heat which is removed via a flow of heat transfer fluid in the heat exchanger discs. This heat is the transferred to the hydronic AHU. Upon completion of absorption (when all the ammonia has reacted with the salt), the SM switches to desorption mode. The desorption mode is characterised by the heating of the reactor via high temperature heat transfer fluid heated by a natural gas burner. The heating of the reactor forces the desorption of ammonia from the salt in the reactor matrix. This desorbed ammonia condenses in the CE where the condensation heat produced in the CE is removed via the heat

transfer fluid flowing in the CE's heat exchanger and this heat is transferred to the hydronic AHU. SMA is characterised by high temperature lift and high heating capacity with moderate coefficient of performance (COP) [17].

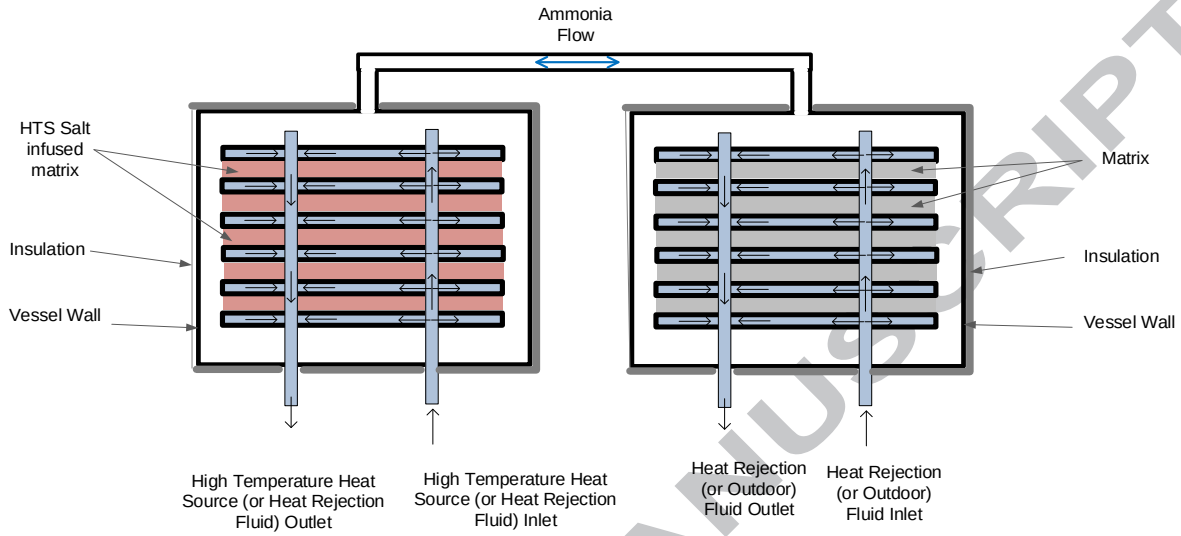

Figure 1 - Schematic diagram of main components of basic sorption module Type A (SMA)

2.2. Resorption Module Component (Type B)

The resorption module component (referred to in this work as Type B) operates under the resorption principle where two different salts (alkali halides) are employed, each having different chemical affinity for ammonia. The salt with the higher affinity is known as the high temperature salt (HTS), and the salt with lower affinity for ammonia is referred to as the low temperature salt (LTS) [20]. In desorption mode, the fluid from the high temperature heat transfer fluid source (from the natural gas burner) drives the flow of ammonia from the HTS which is housed in the primary reactor (R1) to the LTS housed in the secondary reactor (R2). In absorption mode the refrigerant flows from the LTS to the HTS, while the LTS absorbs low temperature (below ambient temperature) thermal energy and expels the ammonia. The ammonia reacts with the HTS, generating heat that is removed with the heat transfer fluid and transferred to the hydronic AHU [21]. SMB is characterised by moderate temperature lift and low heating capacity with high COP [17].

2.3. Heat Pump Sorption Module

To adequately perform as a heat pump unit the sorption module components should be outfitted with hydronic connections, a natural gas burner, pumps and valves. In addition, an outdoor fan coil is used to capture low temperature ambient heat during the absorption mode of operation. Figure 2 shows the sorption module system concept for operation as a heat pump [17].

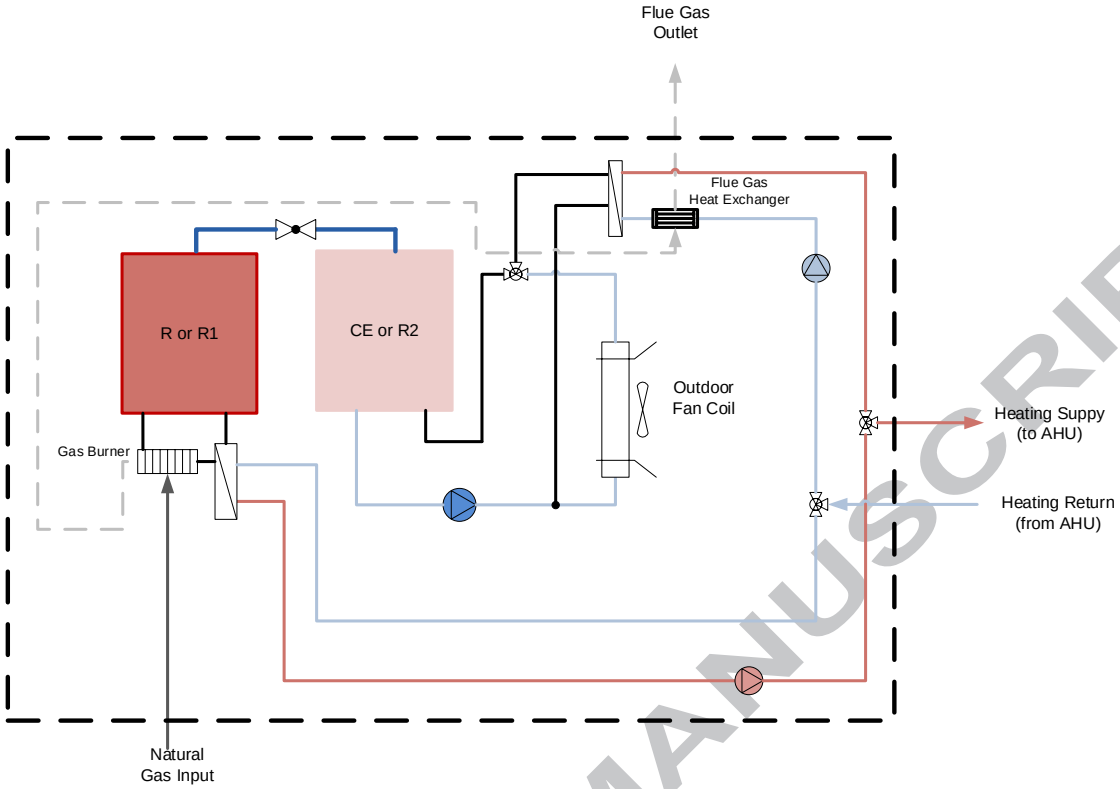

Figure 2 - Schematic drawing of sorption module with complementary components for heat pump operation

3. Gas-Driven Sorption Heat Pump

This work evaluates bivalent GDSHPs comprised of a sorption heat pump module (SM) and a condensing boiler as described in [17]. Two variants of SM are considered to investigate their performance and optimal sizing under different climatic conditions. The bivalent GDSHP operates using the bivalent parallel operation strategy presented in [11]: the SMs provide 100% of the building load when ambient temperatures are at or above the bivalent temperature. At temperatures below the bivalent temperature both the SM and the condensing boiler operate simultaneously to meet the required heating demand. At even lower ambient temperatures, below the cut-off operation temperature for the SM, the SM operates in a non-heat pumping mode in which it operates with condensing boiler efficiency. The bivalent GDSHP unit was considered to be connected to a hydronic air handling unit (AHU) (see Figure 3).

This work evaluates bivalent GDSHPs comprised of a sorption heat pump module (SM) and a condensing boiler as described in [17]. Two variants of SM are considered to investigate their performance and optimal sizing under different climatic conditions. The bivalent GDSHP operates using the bivalent parallel operation strategy presented in [11]: the SMs provide 100% of the building load when ambient temperatures are at or above the bivalent temperature. At temperatures below the bivalent temperature both the SM and the condensing boiler operate simultaneously to meet the required heating demand. At even lower ambient temperatures, below the cut-off operation temperature for the SM, the SM operates in a non-heat pumping mode in which it operates with condensing boiler efficiency. The bivalent GDSHP unit was considered to be connected to a hydronic air handling unit (AHU) (see Figure 3).

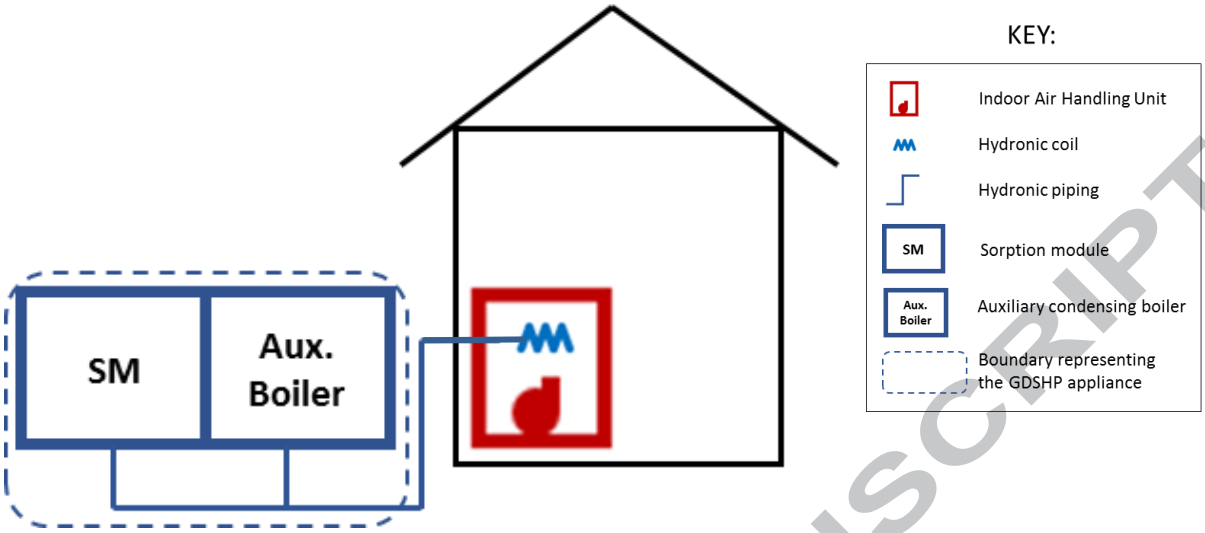

Figure 3 – Schematic of bivalent gas-driven sorption heat pump concept in a house with centrally ducted heating.

A Type A gas-driven sorption heat pump (GDSHPA) consists of the SMA coupled with an auxiliary condensing boiler. Similarly, the Type B gas-driven sorption heat pump (GDSHPB) consists of the SMB coupled with an auxiliary condensing boiler.

3.1. Auxiliary Condensing Boiler

The GDSHPA and GDSHPB systems include an auxiliary condensing boiler that complements the SM when the SM alone cannot meet the required heating demand. Well-dimensioned natural gas-fired condensing boilers typically have efficiencies between 90% and 96% [22], [23], [24]. For the purposes of this study, a generic condensing boiler with a constant efficiency (η_{boiler}) of 92% was assumed for both GDSHPA and GDSHPB.

4. METHODOLOGY

To investigate the impact of the SM capacity sizing for different SM types in different climatic conditions, space heating load simulations for two single family houses were carried out. These hourly building space heating load data were generated via simulations with a 2006 International Energy Conservation Code (IECC) [25] compliant EnergyPlus residential prototype model for New York and Minnesota. The prototype building models were representative of typical construction in these regions. The SM size was defined by the normalised parameter "Design capacity ratio of SM". Where:

$$Design\ capacity\ ratio\ of\ SM = \frac{design\ heating\ capacity\ of\ SM}{peak\ heating\ capacity\ of\ system} \quad (1)$$

From the simulated space heating demand of the building, the hourly outdoor temperatures and the required heating supply temperatures were determined.

Specific heating capacity of each SM type at different outdoor temperatures was employed in the sizing of SM module, while the required auxiliary condensing boiler capacity and thus

overall system size, were calculated. The seasonal gas coefficient of performance (SGCOP) for each GDSHP system type was also calculated.

Evaluations were done for GDSHPA and GDSHPB for 4 scenarios: 1 – low fuel prices and cold climate, 2 – low fuel prices and moderate climate, 3 – high fuel prices and cold climate, 4 – high fuel prices and moderate climate.

To investigate the applicability of each bivalent GDSHP system variant, the systems were evaluated using two metrics: seasonal coefficient of performance and simple payback period for varying sizes of the sorption module (SM). The analysis was conducted for a given design capacity ratio of each SM type. From this design capacity ratio, SGCOP, operating costs and total system costs, were calculated. For the operating cost, the fraction of heating provided by SM and the load provided by auxiliary condensing boiler were calculated for each hour. The system cost was calculated based on the design capacity of the SM and the capacity of the auxiliary condensing boiler. This calculation process was repeated for various SM sizes.

4.1. Simulation of Building and Climate

In order to account for the influence of different climates on the energy performance of the system, two climate locations are considered:

- New York City, NY (40.7°N, 73.8°W), representative of mixed-humid climate characterized by $CDD_{10°C} \leq 2500$ and $HDD_{18°C} \leq 3000$, and
- Minneapolis, MN (44.9°N, 93.2°W), representative of a cold-humid climate characterized by $4000 < HDD_{18°C} \leq 5000$; according to ASHRAE Standard 169-2006 [26].

Table 1 shows the climate characteristics of these locations including the heating design conditions. The outdoor air temperature bin trend for these locations are plotted in Figure 4 based on the Typical Meteorological Year, version 3 (TMY3) weather data [27].

Table 1 - Climate characteristics of locations used in heat load simulations

	New York City, NY	**Minneapolis, Minnesota**
ASHRAE climate zone	4A (Mixed-Humid)	6A (Cold-Humid)
TMY3 degree-days	1984 $CDD_{10°C}$ 2627 $HDD_{18°C}$	1532 $CDD_{10°C}$ 4202 $HDD_{18°C}$
99.6% heating dry-bulb temperature	-10.7°C	-25.2°C
Cities with similar climate zone	Paris, France Seoul, South Korea	Oslo, Norway Moscow, Russia

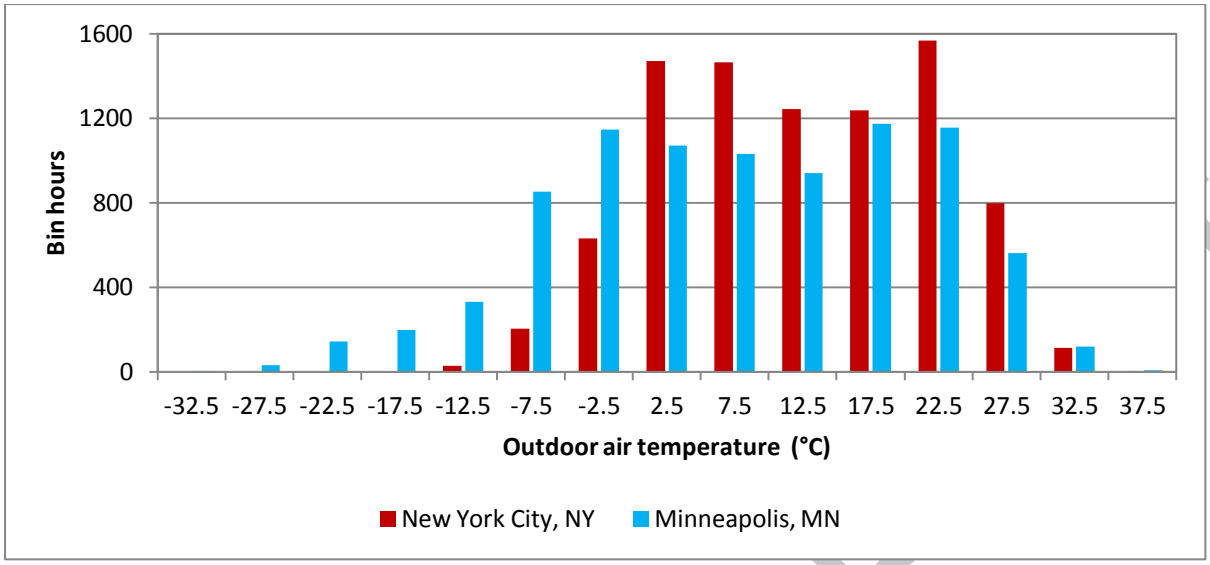

Figure 4 - Outdoor air temperature bins for New York City, NY and Minneapolis, MN

To model residential space heating loads for a typical existing house in these locations, the 2006 IECC version of the publicly available residential prototype building models developed by Pacific Northwest National Laboratory (PNNL) were used for the analysis. These models are used extensively for developing and evaluating residential energy codes in the US and are available for all climate zones in all states in the US, different foundation types and heating systems, and successive versions of IECC [28]. The model represents a two-story, 223 m^2 (2400 ft^2) single-family detached house with a vented attic and a crawlspace. The house has wood-frame construction with insulation properties as required by 2006 IECC for the respective climate zones. Table 2 lists the general characteristics and climate-specific requirements of the building. The space heating loads were modelled in EnergyPlus 8.4.0 using TMY3 climatic data for New York City and Minneapolis, Minnesota. Table 3 shows the building peak and annual space heating loads.

Table 2 - Summary of the general and climate-specific characteristics of the building

General building characteristics		
Building Type	Single-family detached house	
Construction	Wood frame construction, vented attic, crawlspace foundation	
Conditioned floor area	223 m^2	
Conditioned volume	476 m^3	
Gross wall area	221.2 m^2	
Window-to-wall ratio	15%	
Internal heat loads	Lighting: 2.57 W/m^2, Plug loads: 6.53 W/m^2, Occupants: 3	
Thermostat set points	Heating: 22.2°C, Cooling: 23.9°C	
Climate-specific thermal properties of building	**New York City, NY**	**Minneapolis, MN**
Wall U-factor	0.495 W m^{-2} K^{-1}	0.356 W m^{-2} K^{-1}
Ceiling U-factor	0.186 W m^{-2} K^{-1}	0.186 W m^{-2} K^{-1}
Floor U-factor	0.277 W m^{-2} K^{-1}	0.180 W m^{-2} K^{-1}
Window U-factor	2.273 W m^{-2} K^{-1}	1.988 W m^{-2} K^{-1}
Window SHGC	0.394	0.394
Infiltration rate	0.721 ACH	0.162 ACH

Table 3 – Energy simulation results

	New York City, NY	Minneapolis, MN
Peak space heating load	13.4 kW	10.3 kW
Annual space heating load	20898 kWh	16435 kWh

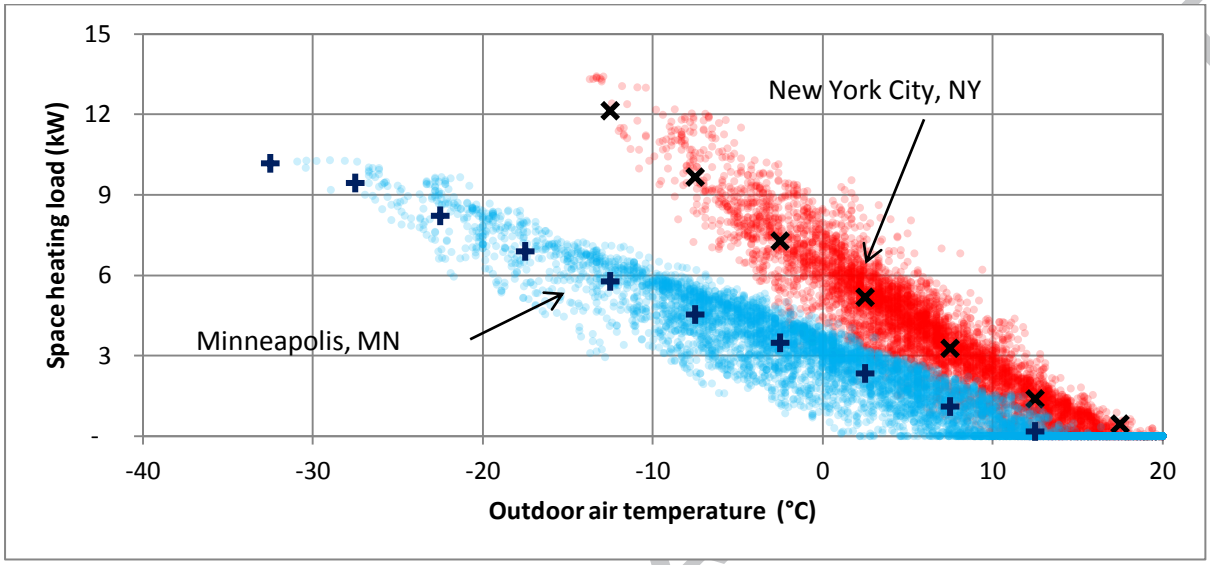

Figure 5. Hourly and bin-average space heating loads for Minneapolis, MN and New York City, NY.

For each hour with space heating load, the required heating fluid supply temperature (T_{supply}) to the AHU is calculated according to an outdoor temperature reset schedule. Where:

$$T_{supply} = linear\ interpoloation \begin{vmatrix} 57.2℃ @ -23.3℃\ T_{out} \\ 37.8℃ @ +15.6℃\ T_{out} \end{vmatrix} \quad (2)$$

4.2. Sorption Module Performance

In this work, we adopt experimental measurements of SMs performance as carried out in [17]. Experimental evaluations as described in [17] were conducted for two sorption module prototypes; Type A and Type B, where one cycle constituted a desorption phase immediately followed by an absorption phase. Each sorption module prototype was connected to two hydronic circuits, heated by steam generated from an electric heater. Several test sequences were carried out comprising at least 40 cycles for each sorption module prototype [17]. The average heating capacity and COP of the sorption modules were measured at various temperature lifts. Temperature lift was defined as the temperature difference between the heat delivery temperature and heat transfer fluid leaving the CE or RB during absorption mode [17].

It was observed that for both SM prototypes type A and type B, the average heating capacity was inversely proportional to the temperature lift. Given that the sorption modules are designed to be totally modular, the number of heat exchanger discs, that is, the heat transfer area of the heat exchangers, determines the heating capacity of the module [17]. Thus, SM heating capacity is given in units of power per square metre of heat transfer area (i.e. as a heat flux). The sorption module capacity was linear with respect to temperature lift (Equation

3a for SMA; Equation 3b SMB), and the SM COP was second-order with respect to temperature lift (Equation 4a for SMA; Equation 4b for SMB).

$$Q_A = -46.95 \times \Delta T_{lift} + 5629.7 \ [\mathrm{W/m^2}] \quad (3a)$$
$$Q_B = -38.29 \times \Delta T_{lift} + 3018.9 \ [\mathrm{W/m^2}] \quad (3b)$$

$$COP_A = -0.0001(\Delta T_{lift})^2 + 0.0088(\Delta T_{lift}) + 1.1596 \quad (4a)$$
$$COP_B = -0.0004(\Delta T_{lift})^2 + 0.0268(\Delta T_{lift}) + 0.9817 \quad (4b)$$

In [17], the experimental evaluations of the SMs were carried out with fixed cycle times of 75 minutes which allowed for full desorption and absorption under all conditions tested. Additionally, since Equations 3a and 3b only account for the temperature lift across the SMs, for the total temperature lift of the GDSHP appliance (including the various heat exchangers), in this work an additional 15°C 'penalty' is applied. Therefore, for the GDSHPA and GDSHPB systems:

$$\Delta T_{lift,required} = (T_{supply} - T_{out}) + 15℃ \quad (3)$$

Figure 6 shows the supply temperature and the required temperature lift ($\Delta T_{lift,required}$) corresponding to the outdoor air temperature. As reported in [17] the maximum temperature lift of the prototype A and prototype B were 80°C and 60°C respectively. Considering the required temperature lift, Table 4 summarises the cut-off temperatures of the SMs, that is, the temperature below which the SMs run in non-heat pumping mode.

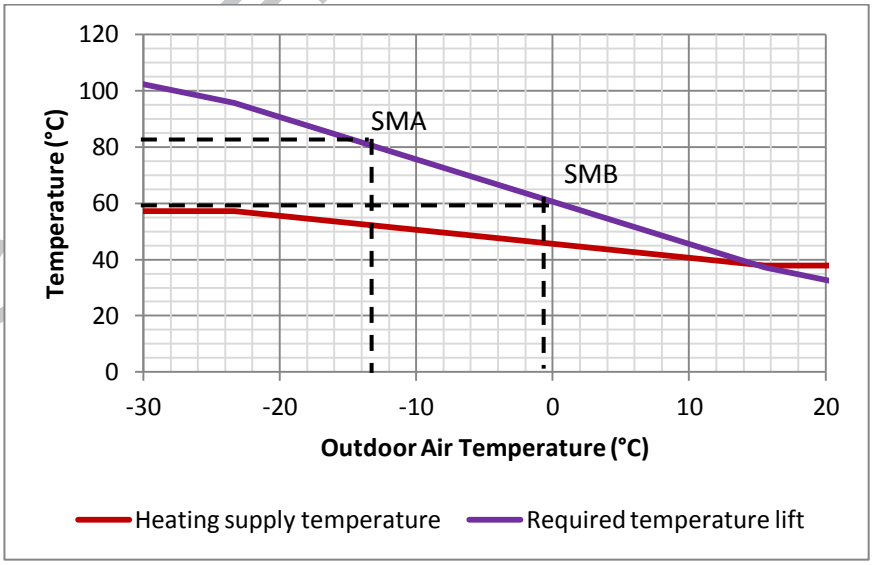

Figure 6. Supply temperature and required temperature lift across the sorption module at varying outdoor air temperature.

Table 4 - Maximum temperature and sorption module cutoff temperature

	Maximum temperature lift	**Sorption module cut-off temperature**
SMA	80°C	-13°C
SMB	60°C	0.4°C

Based on calculations of the required temperature lift for each hour, the heat flux (in kW/m^2) of SM is calculated as:

$$\phi = \frac{Q}{A_{measured}} = \frac{f_1(\Delta T_{lift,required})}{A_{measured}} \quad (4)$$

Then, the capacity of the SM and the fraction of space heating load provided by the SM of given area A [m^2] is calculated as

$$Capacity_{SM}\ (kW) = \frac{\phi \times A}{1000} \quad (5)$$

$$f\dot{Q}_{SM} = \max\left(\frac{Capacity_{SM}}{\dot{Q}}, 1\right) \quad (6)$$

The fraction of space heating load to be provided by the auxiliary condensing boiler is given by:

$$f\dot{Q}_{boiler} = \left(1 - f\dot{Q}_{SM}\right) \quad (7)$$

Thus, the space heating loads met by SM and boiler will be given by equations 10 and 11 respectively:

$$\dot{Q}_{SM} = \dot{Q} * f\dot{Q}_{SM} \quad (8)$$

$$\dot{Q}_{boiler} = \dot{Q} - \dot{Q}_{SM} \quad (9)$$

The natural gas consumption was given by:

$$NGuse = \frac{\dot{Q}_{SM}}{GCOP} + \frac{\dot{Q}_{boiler}}{\eta_{boiler}} \quad (10)$$

In addition to consumption of natural gas, the SMs as well as the auxiliary burner utilise electricity. A condensing boiler used for hydronic heating systems is presented in [29]. For a system configuration similar to that expected for the GDSHP and hydronic AHU, the condensing boiler delivers 68 kWh of thermal energy per kWh of electricity consumed. That is, the condensing boiler had an electrical coefficient of performance (ECOP) of 68. Additionally, for the purposes of this study it was assumed that the ECOP of the sorption module is 25% lower than a standard condensing boiler due to additional electrical components. Therefore, the electricity consumption of the GDSHP was given by:

$$Elec_{use} = \frac{\dot{Q}_{boiler}}{ECOP_{boiler}} + \frac{\dot{Q}_{SM}}{ECOP_{SM}} \quad (11)$$

4.2.1. Calculation of COP

Knowing the required temperature lift for each hour, the COP of SM is calculated as:

$$COP = f_2(\Delta T_{lift,required}) \quad [12]$$

The efficiency of the main burner and post burner (i.e. flue gas) heat recovery of the SM are estimated as $\eta_{post\ burner} = 0.10$ and $\eta_{main\ burner} = 0.84$ [23]. The gas coefficient of performance of SM is calculated as below:

$$GCOP = \eta_{post\ burner} + \eta_{main\ burner} \times COP, if\ \Delta T_{lift,required} \leq \Delta T_{lift,max}\ ; else\ 0 \quad [13]$$

The SGCOP is given by:

$$SGCOP = f\dot{Q}_{SM} * GCOP + f\dot{Q}_{boiler} * \eta_{boiler} \quad (14)$$

4.3. Economic Calculations

The economic analysis was performed using following considerations for energy prices and system costs.

4.3.1. Energy Prices

Two energy price scenarios were evaluated: the low energy price case with \$10/MMBtu for natural gas and \$0.12/kWh for electricity and high energy price case with \$20/MMBtu for natural gas and \$0.22/kWh for electricity. The low and high energy prices are determined from the current US average and highest state average residential energy prices, respectively [30], [31].

Table 5 - Energy prices used the two energy price scenarios; low energy price case and high energy price case.

	Low Energy Price Scenario	**High Energy Price Scenario**
Natural gas	0.034 USD/kWh$_{thermal}$ (10 USD/MMBtu)	0.068 USD/kWh$_{thermal}$ (20 USD/MMBtu)
Electricity	0.12 USD/kWh	0.22 USD/kWh

From natural gas and electricity prices, the annual operating cost of the GDSHP system was calculated by:

$$Annual\ operating\ cost = \sum NGuse * NGprice + \sum Elecuse * Elecprice \quad (15)$$

4.3.1. System Costs and Payback Period

For gas-driven absorption and adsorption heat pumps, the current installed cost vary from €315 (\$335) per kW to €1300 (\$ 1383) per kW according to [32] and [33]. Given that the SM

based bivalent GDSHP is a novel development, cost estimates are employed within the range found in literature. For the SMA, a low installed cost scenario of €315/kW and high cost scenario of €1300/kW were assumed. The SMB, working on the resorption principle, is expected to have a higher cost than the respective SMA due to the lower thermal capacity per unit heat transfer area as well as the increased complexity of the resorption system [17], [34], [35]. Therefore, for SMB, in each scenario the installed cost was considered to be 25% higher than for SMA. For the purposes of this study, an assumed fixed cost of 10% was used to account for electrics, control system and other components that are predetermined irrespective of the SM size. The installed cost of the auxiliary condensing boiler unit integrated with both GDSHPA and GDSHPB was estimated to be $260 per kW heating capacity [36]. Thus:

$$\begin{aligned} &Total\ installed\ cost\ GDSHP\ (\$) \\ &\quad = Fixed\ cost\ (\$) + \left[IncrementalCost_{SM}\left(\frac{\$}{kW}\right) * Size_{SM}\ (kW)\right] \\ &\quad + \left[IncrementalCost_{boiler}\left(\frac{\$}{kW}\right) * Size_{boiler}\ (kW)\right] \end{aligned} \quad (16)$$

The capacity of the sorption module used in the determination of the installed system cost was based on measured capacity and heat transfer area from evaluations carried out in [17], where:

$$Capacity_{SM}\ (kW) = A_{SM} * \frac{Capacity_{SM,measured}}{A_{SM,measured}} \quad (17)$$

The simple payback time of the GDSHP is calculated taking a condensing boiler as the baseline heating system, where:

$$Payback\ period = \frac{Total\ installed\ cost\ GDSHP - Total\ installed\ cost\ condensing\ boiler}{Annual\ operating\ cost\ condensing\ boiler - \ Annual\ operating\ cost\ GDSHP} \quad (18)$$

The methodology used for the full analysis is summarised in Figure 7. Energy and economic calculations were carried out in Microsoft Excel based on exported hourly data from the EnergyPlus simulations and the equations defined in this work.

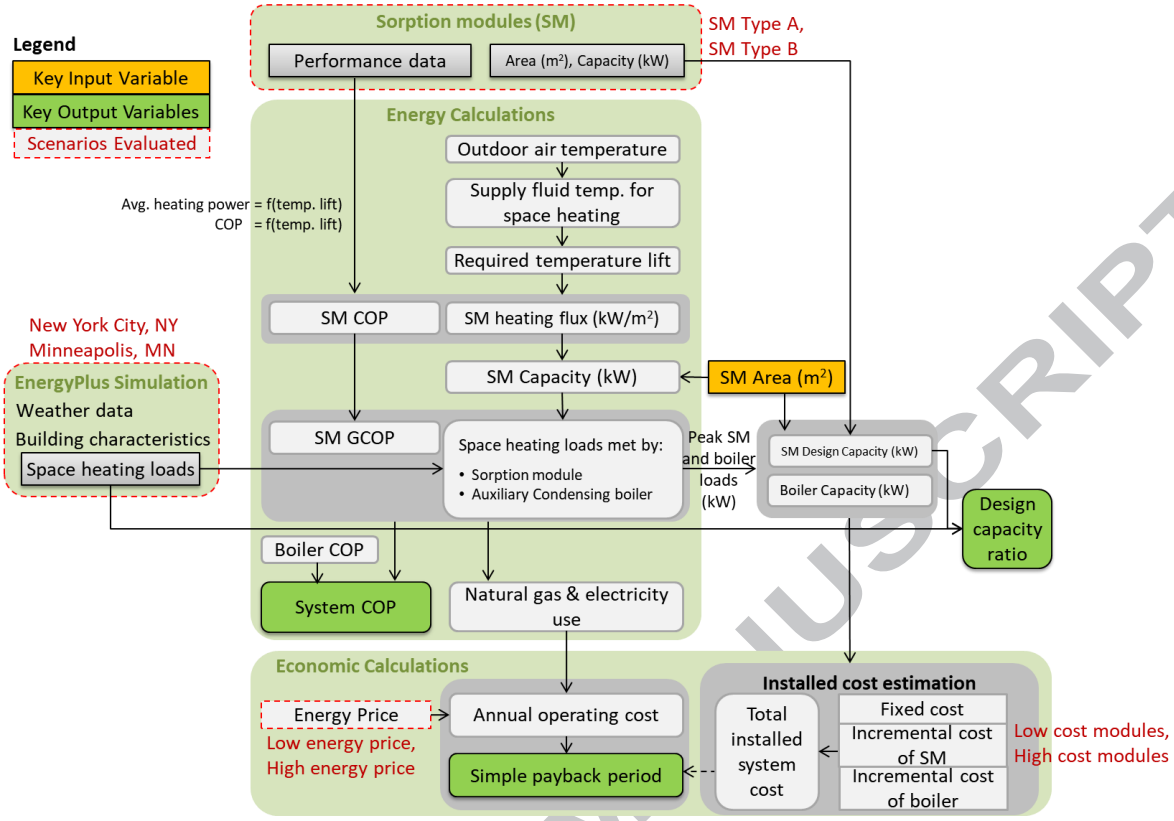

Figure 7 – Analysis methodology

5. RESULTS AND DISCUSSION

5.1. Influence of Design Capacity Ratio on SGCOP

Analyses were carried out to determine the system SGCOP for various design capacity ratios of the SMs for GDSHPA and GDSHPB and the results shown in Figure 8. In the case of New York City, the GDSHPB exhibits higher SGCOP than GDSHPA this is attributed to the higher COP and thus GCOP of the SMB. However, in the case of Minneapolis, GDSHPA marginally outperforms GDSHPB at higher design capacity ratios. This observation is due to the fact that the Minneapolis climate is characterised by a higher proportion of the heating demand occurring at low ambient temperatures. The GDSHPA's lower SM cut-off temperature allows it to run as a heat pump at lower outdoor temperatures and thus with a higher GCOP.

Although SGCOP increases with increasing SM design capacity ratio for both GDSHPA and GDSHPB, little improvement in SGCOP was observed at high design capacity ratios. In New York city SM design capacity ratios greater than 0.40 and in Minneapolis ratios greater than 0.30, for both GDSHPA and GDSHPB, show only a very small increase in SGCOP.

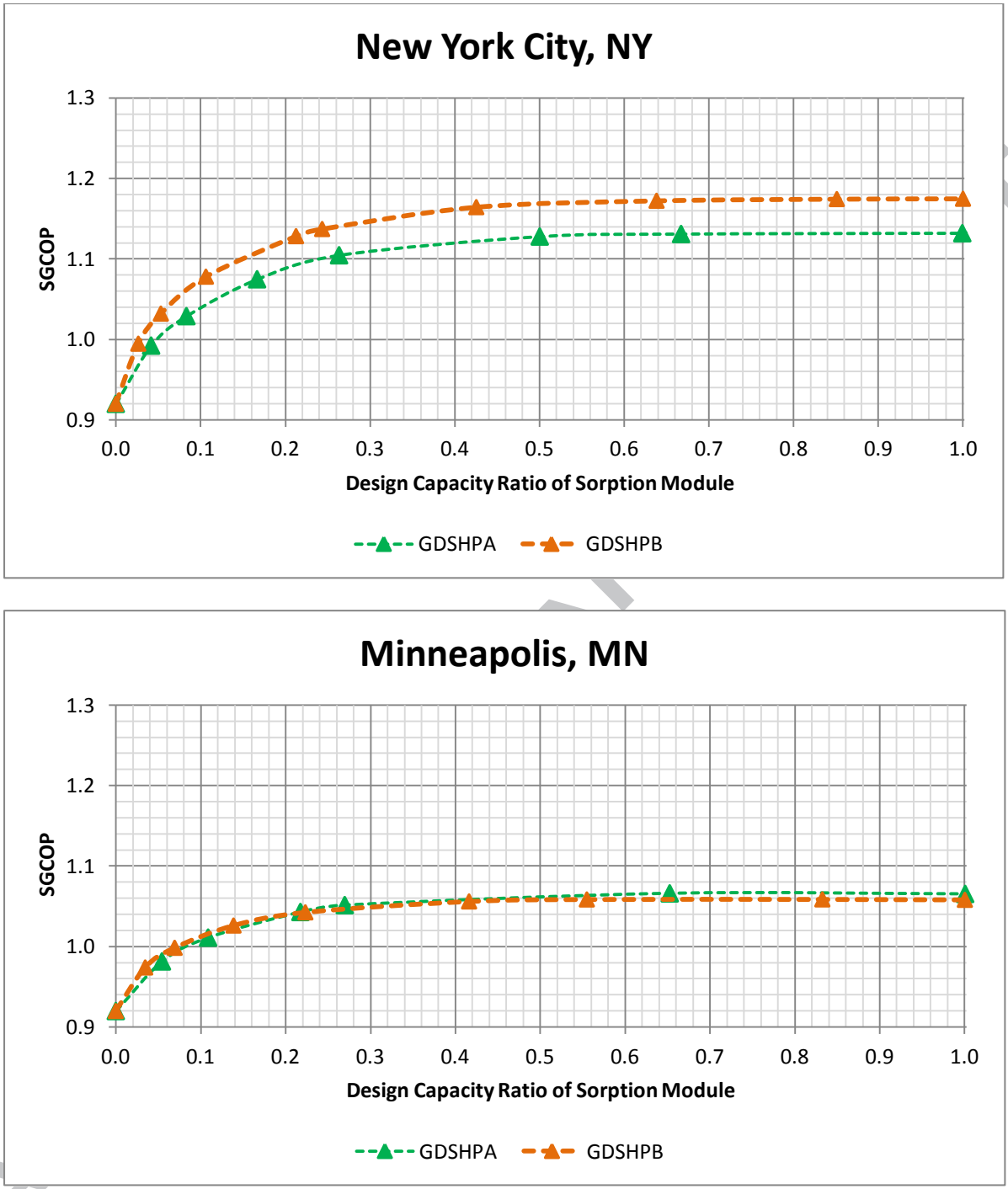

Figure 8 – System SGCOP versus SM design heating capacity ratio for New York City and Minneapolis, MN.

5.2. Influence of Design Capacity Ratio on Payback Period

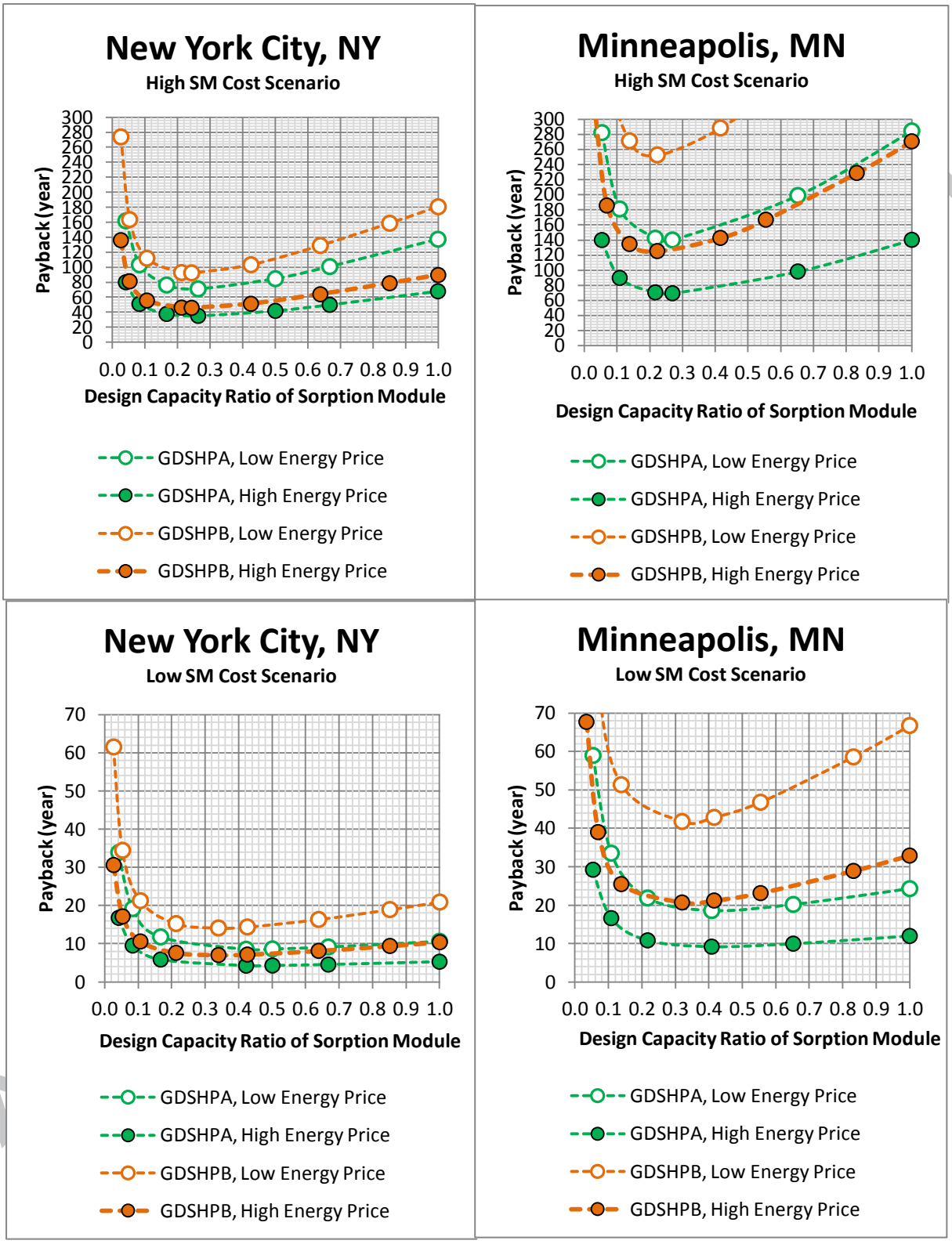

Figure 9 - Simple payback period versus SM design heating capacity ratio for New York City and Minneapolis, MN for high (upper graphs) and low (lower graphs) sorption module cost scenarios.

Interestingly, for all scenarios, lower energy savings and thus longer payback periods are observed for the GDSHPB compared to GDSHPA in both climates. This can be attributed to the higher expected cost of the SMB. Additionally, despite SMB having higher COPs than SMA at ambient temperatures above 5°C, the GDSHPB had a lower SGCOP due to its limited range of operation. The limited running of the SMB in heat pump mode is even more pronounced for Minneapolis due to the larger portion of hours with low outdoor temperatures.

As might be expected, for both GDSHPA and GDSHPB the higher energy price scenario gives lower payback time since energy cost savings are higher. Conversely, the lower the cost of the SM the lower the overall installed cost and the shorter the payback period.

5.1. Optimum Sorption Module Design Capacity Ratio

For each scenario, that is, for each GDSHP type, SM cost and location, there was an optimum SM design capacity ratio which provides the minimum payback period. These optima are summarised in Table 6.

Table 6 - Optimum sorption module design capacity ratio for each climate

Climate/ Location	System Type	Low SM Cost Scenario		High SM Cost Scenario	
		Optimum SM design capacity ratio	Energy Savings	Optimum SM design capacity ratio	Energy Savings
New York	GDSHPA	0.42	14.4%	0.26	11.8%
	GDSHPB	0.34	13.0%	0.24	11.1%
Minnesota	GDSHPA	0.41	8.1%	0.27	7.0%
	GDSHPB	0.32	5.1%	0.22	4.5%

In all scenarios the optimum design capacity of the SM was below 50% of the peak capacity of the GDSHP, with design capacity ratio of the SMs ranging from 0.22 to 0.42. Energy savings for New York City were between 11.1% and 14.4% compared to a condensing boiler, while in the case of Minneapolis savings ranged from 4.5% to 8.1%. The milder New York climate coupled with higher heating energy demand compared to Minneapolis means that the SM module operates in heat pump mode for more hours of the year producing higher energy savings and thus energy cost savings.

Additionally, a link is observed between SM cost and energy savings: the lower the cost of the SM, the more cost effective it is to incorporate a higher capacity SM, leading to higher energy savings.

Payback times at optimum SM design capacity ratios varied between 4.2 and 92 years in the case of New York City. In the case of Minneapolis payback times were longer ranging between 9.2 years and 252 years.

For each location further analyses were carried at the optimum SM design capacity ratio under the most favourable scenarios, i.e. GDSHPA, high energy prices and low SM cost. Analyses yielded the following:

- In New York the GDSHPA with a SM design capacity ratio of 0.42 gave rise to energy savings of 14.4% and a payback time of 4.2 years compared to a standard condensing boiler. The SMA covered over the course of a year 86.6% of the space heating load.
- In Minnesota with a SM design capacity ratio of 0.41 the GDSHPA provided 8.1% energy savings over a baseline condensing boiler. That provided a payback time of 9.2 years whilst covering 74.1% of the space heating load.

Figure 10, shows the results of the operation characteristics from annual hourly simulations carried out for each location under the best scenario. By inspection of Figure 10, for the building in New York City, the bivalent temperature is 2 to 6°C. This is the range of ambient temperature at which the SM is no longer able to meet the entire building load. It varies across a range due to variations in solar insolation and internal loads. For the building in Minneapolis, the bivalent temperature is -8 to -1°C. In both locations, the cutoff temperature is lower than the coldest temperature observed, leading to parallel bivalent operation at all temperatures below the bivalent temperature.

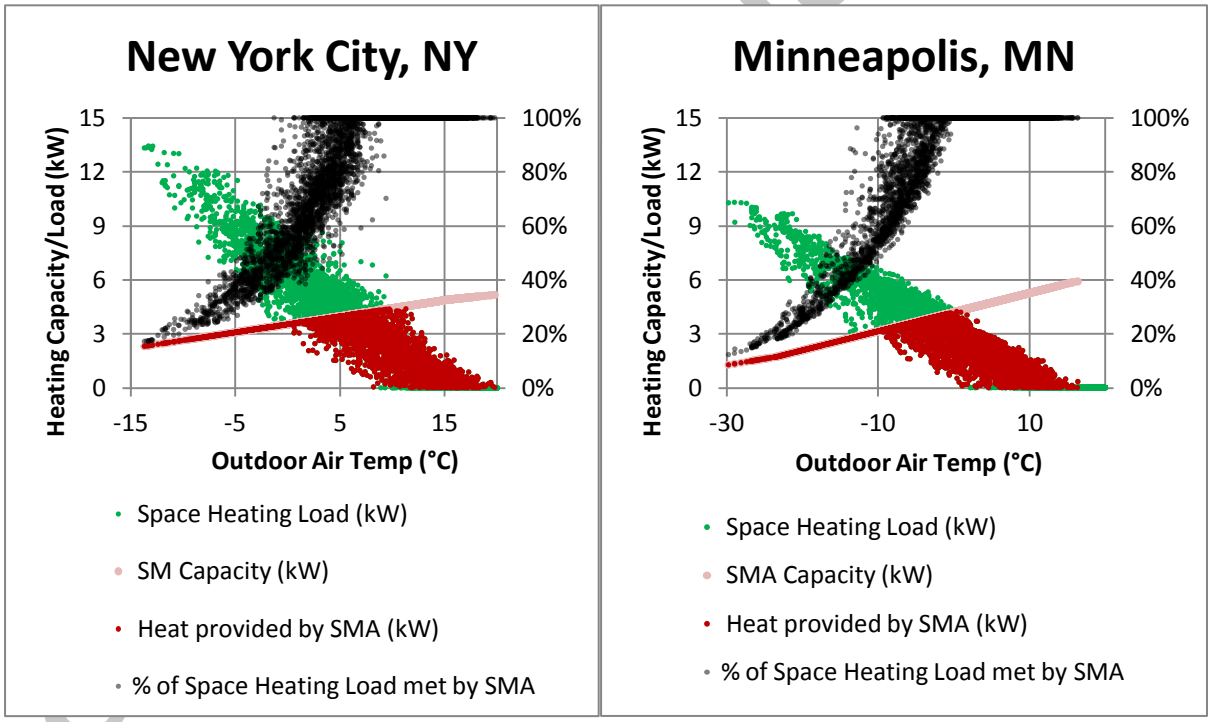

Figure 10 – Annual simulation hourly results at optimum SM design capacity ratio for New York City, NY (left) and Minneapolis, MN (right).

5.2. GDSHP Design Considerations

In the present study of a bivalent GDSHP system there are several optimisation parameters, principally; SM module type, SM design capacity ratio, heat load, climate, energy prices and SM cost.

The SM type has a significant impact on the achievable energy savings within a given climate, for colder climates a SM with a low cut-out temperature is required to maximise energy savings. For both climates SMA proved the superior sorption module type due to its

comparatively low cut-out temperature and lower cost, however, for warmer climates, SMB with its superior COPs at higher outdoor temperatures, might have an advantage.

The optimum design capacity ratio did not vary significantly based on climate for either SM sorption module type suggesting that this parameter is more sensitive to energy and system costs.

An additional consideration is acceptable payback period for a given application. Considerations beyond cost might be applicable, for instance, reduction of emissions or regulations on minimum renewable energy mix in the heating system. Figure 11 shows pareto fronts for New York and Minnesota. Here it is observed that especially in the case of GDSHPB in New York moderately higher SGCOP would be achievable if slightly longer payback periods are allowed.

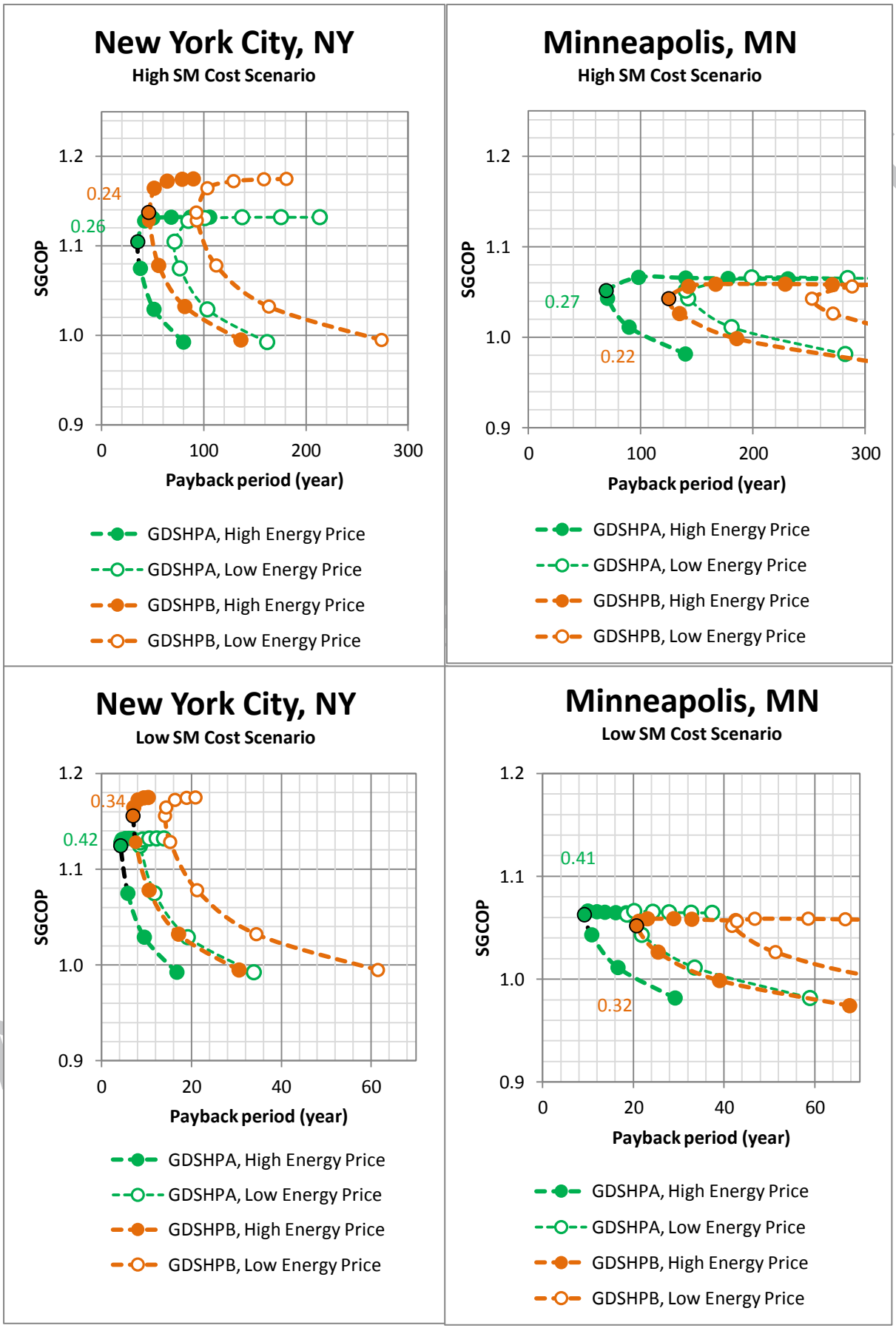

Figure 11 – Seasonal COP versus simple payback period for New York City, NY and Minneapolis, MN.

6. CONCLUSIONS AND PERSPECTIVES

In this study a method for the optimum sizing of a bivalent gas-driven sorption heat pump (GDSHP) incorporating a sorption module (SM) and an auxiliary condensing boiler is presented. Two different sorption module types were investigated; one operating under the basic chemisorption cycle (Type A) and the other under the resorption process (Type B).

The GDSHP systems were evaluated based on several scenarios based on SM type (Type A vs. Type B), climate (New York vs. Minnesota), natural gas prices (3.4 US cents/kWh vs. 6.8 US cents/kWh), electricity prices (12 US cents/kWh vs. 22 US cents/kWh) and SM cost ($335/kW vs. $1383/kW).

The results of the evaluation indicated that the sorption module design heating capacity ratio that results in the shortest payback period was 0.42 and 0.34 for GDSHPA and GDSHPB respectively, for systems installed in New York with a sorption module cost of $335/kW. In the case of Minnesota, the corresponding optimum capacity ratios were 0.41 for GDSHPA and 0.32 for GDSHPB.

In New York, a GDSHPA, with SM capacity ratio of 0.42 had expected payback periods of 4.2 years and 8.5 years for the high energy cost and low energy cost scenarios respectively. However, for the GDSHPB system, even at optimum SM capacity ratios, exhibited higher payback periods 7.0 and 14.1 years for the high energy cost and low energy cost scenarios respectively.

The optimum design capacity ratio did not vary significantly based on climate for either SM sorption module type but was more sensitive to energy and system costs. However, the SM type had a significant impact on the achievable energy savings and thus minimum payback times for a given climate. Even though SMB had higher COP than SMA at higher ambient temperatures, its relatively high cut-out temperature, and lower heating capacity at low outdoor temperatures meant it performed less favourably than SMA. Therefore, SMA is more suitable for both cold-humid (Minnesota) and mixed-humid (New York) climates. SMB might however be more applicable to warmer climate zones.

Based on the study carried out, the optimum sizing of the bivalent GDSHP can be determined from studies of cost, performance, and climate. Thus, allowing for an evaluation of the techno-economic feasibility, from a design perspective, of modular residential sorption heat pumps. Optimisation or trade-offs beyond achieving the lowest payback time are also relevant to be explored considering desired savings level, reduction of emissions or other important considerations at the time of design.

ACKNOWLEDGMENTS

This work was sponsored by the U. S. Department of Energy's Building Technologies (BTO) Office under Contract No. DE-AC05-00OR22725 with UT-Battelle, LLC. The authors also

acknowledge Mr. Antonio Bouza, Technology Manager for HVAC&R, Water Heating, and Appliance, U.S. Department of Energy BTO.

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Highlights:

- The optimum design heating capacity of the sorption heat pump ranges between 22–44% of peak capacity.
- Two climates, two energy price scenarios, and two sorption system types were evaluated
- A detailed building model was used to calculate hourly building loads