Air source heat pumps for cold climate applications

– Recent U. S. R&D results from IEA HPP Annex 41

Air source heat pumps are easily applied to buildings almost anywhere. They are widespread in milder climate regions but their use in cold regions is hampered due to low heating efficiency and capacity at cold outdoor temperatures. This article describes selected R&D activities aimed at improving their cold weather performance.

Keywords: Air source heat pump; cold climate heat pumps; vapour compression; economizer cycles; parallel compressor cycles; IEA HPP Annex 41.

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The IEA Heat Pump Programme (IEA HPP) is a non-profit organisation with 15 member countries – Austria, Canada, Denmark, Finland, France, Italy, Germany, Japan, the Netherlands, Norway, South Korea, Sweden, Switzerland, United Kingdom and the United States http://www.heatpumpcentre.org/en/aboutHPP/Sidor/default.aspx. The Programme carries out a strategy to accelerate the use of heat pumps in all applications where they can reduce energy consumption for the benefit of the environment. It strives to achieve widespread deployment of appropriate high quality heat pumping technologies to obtain energy conservation and environmental benefits from these technologies. Under the management of an Executive Committee the member countries cooperate in projects (called Annexes) in the field of heat pumps and related heat pumping technologies such as air conditioning, refrigeration and working fluids (refrigerants). Over 40 Annexes have been or are being conducted under the Programme.

In 2012 the IEA HPP established Annex 41 to investigate technology solutions to improve performance of heat pumps for cold climates. Four IEA HPP member countries are participating in the Annex – Austria, Canada, Japan, and the United States (U.S.). The principal focus of Annex 41 is on electrically driven air-source heat pumps (ASHP) since that system type has the biggest performance challenges given its inherent efficiency and capacity issues at cold outdoor temperatures. Availability of ASHPs with improved low ambient performance would help bring about a much stronger heat pump market presence in cold areas which today rely predominantly on fossil fuel furnace heating systems. A primary technical objective of the Annex is to define pathways to enable ASHPs to achieve an "in field" heating seasonal performance factor or SPF_h ≥2.63 W/W (HSPF ≥9.0 Btu/Wh), the minimum level necessary in order to gain recognition as a renewable technology in the EU.

ASHPs based on the simple vapour compression cycle suffer both heating capacity (output) and efficiency (coefficient of performance or COP) degradation as the outdoor ambient temperature drops. At the same time, building heat demand is increasing so ASHPs require a supplemental heating source – usually direct electric resistance heating elements – to meet the building heat demand causing lower seasonal performance in areas that experience large numbers of hours at cold temperatures (loosely defined as ≤–7°C for purposes of Annex 41). Thus, the primary criterion for advanced ASHPs to achieve higher SPFh in cold climates is to achieve higher heating capacities at low ambients.

This article describes some of the recent R&D results from the U.S. for two advanced ASHP approaches; a two-stage vapor compression cycle with economizer and a single-stage cycle with two compressors in parallel. The capacity target for US R&D efforts is specified in **Table 1** below.

Two-stage economizer system

Research at Purdue University's Herrick Labs identified a number of cycle concepts that could be useful for ASHPs in cold climate applications (Bertsch et al., 2006). Of these, three were seen to have the highest relative efficiency and relative heat output with low or acceptable discharge temperatures and were selected for detailed

Table 1. U. S. cold climate heat pump performance targets.

Outdoor temperature	Heating capacity
8.3°C (47°F)	9 – 21 kW (2.5 – 6 tons), nominal rating
−25°C (−13°F)	≤25% reduction from nominal

comparison – the two-stage using an intercooler, the two-stage using an economizer and the cascade cycle. **Figure 1** illustrate the three cycles.

System simulation models were created for each of the three technologies to simulate the heating capacity and performance for comparison. The heating supply temperature for each was fixed to 50°C (122°F) to match the heating supply air delivery temperature of a baseline gas-fired furnace per requirements of the project sponsor. All three cycles showed COPs above 2 at temperatures below 0°C (32°F) with the cascade cycle having slightly higher COPs than the others. All three cycles had similar capacities at the low ambient temperatures. The only noticeable difference between these technologies is at the warmer ambient temperatures. The cascade cycle COP at temperatures above 0°C is considerably lower than that of the economizer and intercooler cycles, and this is most likely due to the heating capacity sizing selected for the high stage cycle. Bertsch et al. assumed the cascade cycle has an additional smaller outdoor heat exchanger to allow for the high side cycle to operate without the low side

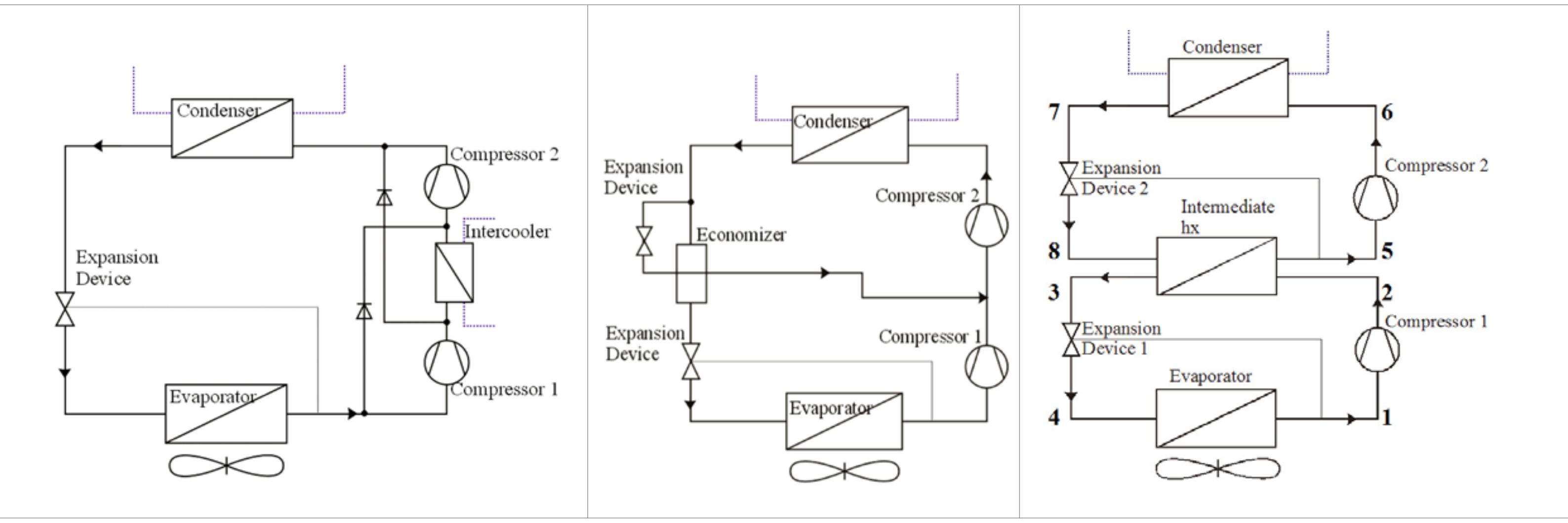

Figure 1. Heat pump cycle schematics – Intercooler (Left) – Economizer (Middle) – Cascade (Right). (Bertsch et al., 2006)

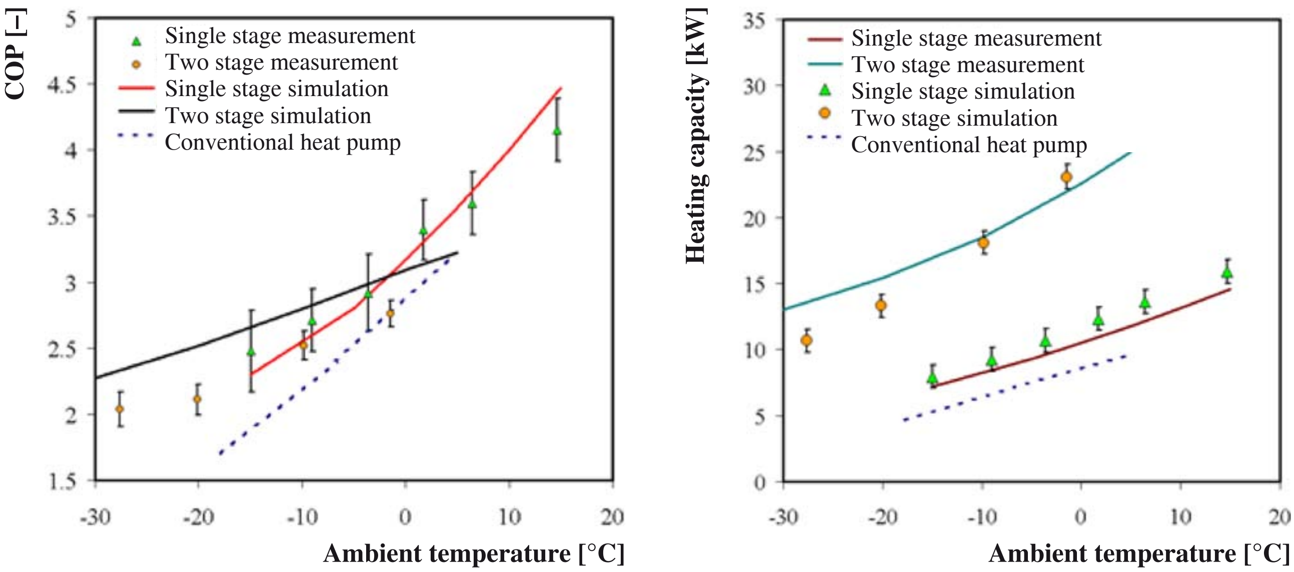

Figure 2. Experimental results of two-stage heat pump compared to simulation results; manufacturer's data was used to indicate performance of the conventional single-stage ASHP. (Bertsch et al. 2008)

cycle. Overall, all three cycles are predicted to be able to adequately satisfy the heating load. The conclusion made from these results and the equipment required (relative cycle complexity) is that the two-stage economizer cycle would be the best choice for an ASHP in colder climates.

A two-stage, economizer system using R-410A was experimentally tested down to ambient temperatures of –30°C (–22°F), achieving a heating capacity and COP of roughly 11 kW and 2.1 respectively (Bertsch et al., 2008). The plots of the experimental results compared to the simulation of the system heating capacity and COP are shown in **Figure 2**. The two-stage heat pump is shown to have much larger heating capacities than a conventional heat pump at low outdoor temperatures. For an outdoor air temperature of about –29°C (~–20°F) the tested capacity was ~11 kW or ~85% of the measured capacity at the U.S. nominal heat pump rating condition of ~8.3°C (47°F) compared to the desired target of 75% (**Table 1**). The test system could be easily built from off-the-shelf components with little modifications, showing promise for being manufacturable with relatively low cost premium compared to conventional ASHPs.

A field test of an advanced two-stage ASHP with economizer VI (vapor-injection) loop designed for cold climate operation has been completed at a U.S. Army base in Indiana (Caskey et al., 2013). The heat pump system is similar to the concept analyzed by Bertsch, et al. (2008). It features a large tandem scroll compressor (two parallel compressors) for low ambient temperature capacity boosting in series with a smaller variable-speed (VS) scroll compressor for cooling operation and moderate ambient heating. The originally installed HVAC system was a natural gas furnace with a split system air conditioner. A side-by-side performance comparison between the originally installed HVAC system and the two pre-commercial heat pump units developed at Purdue University was conducted during the 2012-2013 heating season.

For the monitored period at the Army site the ASHP system achieved about 19% source energy savings vs. the baseline gas furnace system. Using average Indiana residential electricity and gas prices the utility costs for the ASHP and baseline furnace would have been about the same. The ASHP used no electric back up heating during the test period. Furthermore, the installation and maintenance cost and complexity of the heat pump is comparable to those of conventional ASHPs.

Single-stage ASHP with two parallel compressors

Research at the Oak Ridge National Laboratory (ORNL) investigated sixteen different ASHP cycle configurations of which eight were determined to be able to meet the heating capacity degradation target listed in **Table 1**. Both variable-speed (VS) and dual, parallel single-speed compressors (Tandem compressors) were investigated. The VS-based designs offered somewhat greater low-temperature capacity capability while the Tandem approaches were less complex and had almost as much capacity capability. On this basis the Tandem approach appeared to offer the most cost-effective option, and was chosen for laboratory experimental investigation (Shen, et al 2014).

Figure 3 illustrates the cycle concept used for the laboratory prototype test unit. It includes a pair of identical, single speed scroll compressors capable of operating with discharge temperatures of up to 138°C (280°F). A conventional thermal expansion valve (TXV) is used for refrigerant flow control for space cooling mode, and a separate electronic expansion valve (EXV) is used for space heating mode. The EXV targets to control optimum

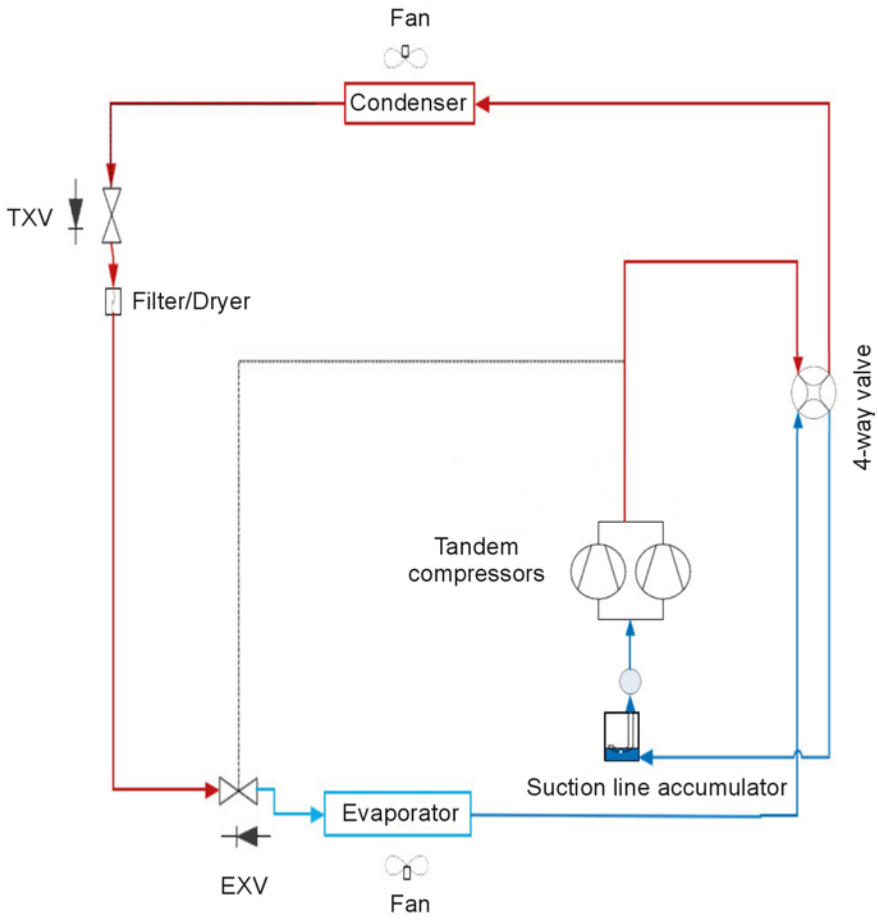

Figure 3. System concept – ASHP with dual, parallel compressors.

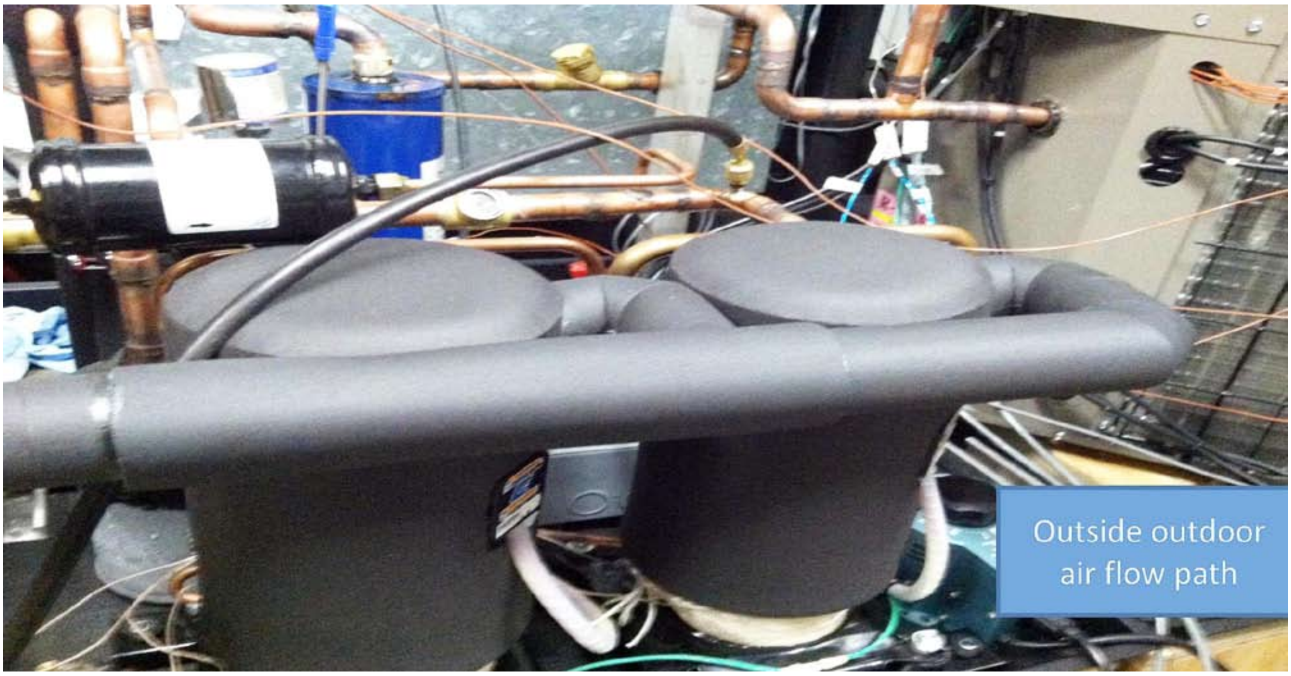

Figure 4. Insulation on test system compressors.

discharge temperature as a function of ambient temperature, rather than the evaporator exit superheat degree. The compressors were insulated and placed outside the outdoor air flow stream to minimize the compressor shell heat losses (**Figure 4**) – testing data confirmed that the insulation layer reduced the compressor heat loss by 50%, and boosts the capacity and COP at –13°F more than 5%.

Figures 5 and 6 below show the capacity and COP vs. ambient temperature, respectively, from the lab test results. This system met the heating capacity target, achieving 77% of the nominal rated capacity at –25°C (–13°F). **Figure 6** indicates that it maintains reasonably good COPs (~2.0) at –25°C as well.

Concluding remarks

The two advanced ASHP system concepts discussed above are both able to achieve the heating capacity improvement target noted in **Table 1** – ≤25% degradation of heating capacity at –25°C (–13°F) vs. the rated capacity at 8.3°C (47°F). But this capacity improvement comes at the cost of increased system complexity which will result in increased cost compared to conventional ASHPs. It is possible that either of these approaches could result in a technically feasible ASHP that meets the Annex 41 SPF_h goal (≥2.63) and may approach the SPF_h levels of GSHPs, gas-engine-driven ASHPs, gas absorption ASHPs, or other residential HVAC systems in cold climate locations. Whether this level of efficiency can be achieved at lower installed costs than the aforementioned alternative systems remains to be seen. ■

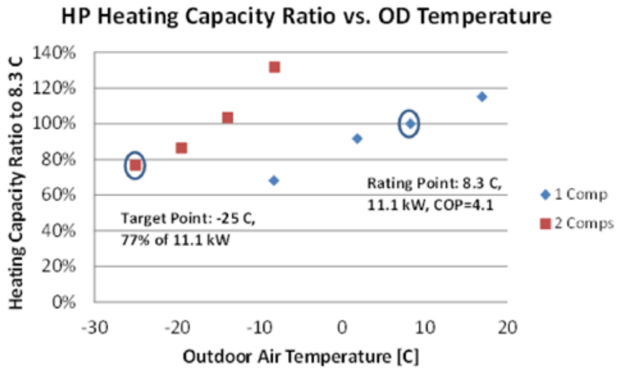

Figure 5. Lab prototype heating capacity ratio (relative to capacity at the nominal 8.3°C (47°F) rating point, with one compressor).

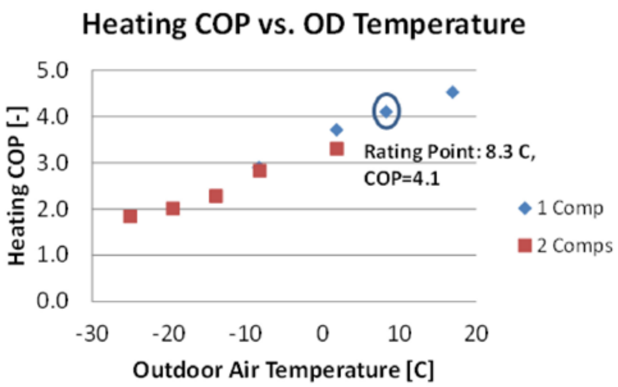

Figure 6. Lab prototype heating COP.

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