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Refrigerant and Scroll Compressor Options for Best Performance of Various European Heat Pump Configurations

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ABSTRACT

A variety of refrigerants is used for today's heat pump systems in Europe. In a theoretical analysis the impact of chosen refrigerant upon performance is discussed from a theoretical and from a compressor viewpoint.

The requirements for the heating system, as well as from the weather conditions are not uniform throughout Europe. Therefore compressor versions have been developed to achieve the required COPs as well as the necessary operating range in different types of applications. Analysis of seasonal performance using statistical weather data for different locations shows how various scroll compressor types can be matched to types of heat pumps to give best performance, considering cost.

In considering heat pump types the requirements for air source can differ from those for ground source, and the heat distribution system also needs to be taken into account. This may be underfloor heating operating at relatively low temperature or radiator systems requiring higher temperature. In both cases, the domestic hot water option may be added.

1. INTRODUCTION

The European emphasis on the reduction of CO2 emissions stimulates an increased use of heat pumps replacing the traditional heating systems for residential houses presently equipped with gas or oil burners. The specifics of this application require some adaptations to the compressors to make the heat pump system competitive. The prime aspects are operating range of the compressor and the compressor coefficient of performance, as this has a direct impact upon the economics. The choice of the refrigerant is also a matter of great importance to the manufacturer. This paper considers the most commonly used refrigerants in Europe in their impact on the performance of both ground source and air source heat pumps (GSHPs and ASHPs) Seasonal COPs, referred to as SCOP or SPF (seasonal performance factor) calculated from compressor selection data, have been used to compare different R407C compressor types.

2. REFRIGERANT CHOICE

2.1 Refrigerant

Table 1 presents the refrigerants in heat pumps mostly discussed nowadays, either in industry or in investigation centres, together with their main characteristics. There are 3 groups:

- Hydrofluorocarbons HFC: mainly R407C R134a, R410A, and R404A,
- Hydrocarbons HC: propane 290 and iso-butane R600a,
- Mineral Substances: Ammonia R717 and Carbon dioxide R744.

Table 1. Refrigerant properties

	R407C	**R134a**	**R410A**	**R404A**	**R290**	**R600a**	**R717**	**R744**
Mixture/Single Substance	Mixture	Single	Mixture	Mixture	Single	Single	Single	Single
GWP(100y) [1]	1520	1300	1725	3260	3	3	~0	1
Toxicity	No	No	No	No	No	No	Yes	No
Flammable	No	No	No	No	Yes	Yes	Slight	No
Boiling Temp, °C [2]	-44.1	-26.1	-51.8	-47	-41.2	-12.2	-33.6	/
Critical Temp, °C	87	101	72	73	97	135	133	31
Condensing Temp @ 26 bar abs, °C	58	80	43	55	70	114	60	-11
Refrigeration Effect @ 0°C, MJ/m3	4.2	2.9	6.9	5.1	3.9	1.5	4.3	23

[1]: Global Warning Potential (100 years)
[2]: at 1 bar absolute

Hydrocarbons and Minerals have a negligible GWP, which makes them attractive for environment reasons. On the other hand, HC are flammable and then present a risk that is not currently accepted for HP applications.

Ammonia R717 is toxic and is prohibited in residential and commercial applications. CO2 is non-flammable, non-toxic and cheap. On the other hand, its physical properties (low critical temperature and high density) make it difficult to apply with existing technologies. Moreover there are serious doubts on its efficiency potential for this application. Keep in mind that COP is mainly proportional to critical temperature (Calm and Domanski 2004). Based on this first analysis, only HFC will be considered in the discussion that follows.

R134a presents the lowest GWP and the highest critical temperature. On the other hand it presents a low refrigerant effect and a high boiling temperature, not ideal for component cost and low temperature applications like air-to-water heat pumps.

R410A has the highest refrigerant effect but the lowest critical temperature, it is currently the most expensive and its higher working pressure requires different component and system design. Its thermodynamic properties limit its use at high-pressure ratios due to high discharge temperatures. On the other hand, it has excellent heat transfer properties and pressure drop effects are less. Both these factors assist in reducing system cost.

R404A is used in refrigeration applications in Europe due to its high density, low boiling point and good behaviour at high-pressure ratio (low discharge temperatures). On the other hand, it has the highest GWP of all HFC considered here and its low critical temperature makes it less attractive for Heat pumps.

Finally R407C is a good compromise. Its glide can be beneficial in counter flow plate heat exchangers resulting in lower heat exchanger temperature differences.

Figures 1 and 2 show a comparison of refrigeration COPs for the HFCs considered here. The heating COP, defined with reference to the condenser enthalpy difference is equal to refrigeration COP + 1. Figure 1 shows the influence of the condensing temperature on the COP at constant evaporating temperature (-5°C). This represents typically Brine to Water heat pumps. Figure 2 is aimed at showing what would happen in the case of an Air-to-Water heat pump at variable evaporating temperature.

The conclusions are about the same for both conditions, Compared to R407C at up to 50°C condensing temperature, R134a is up to 5% higher efficiency while R410A is a few percent lower and R404A close to 10% lower.

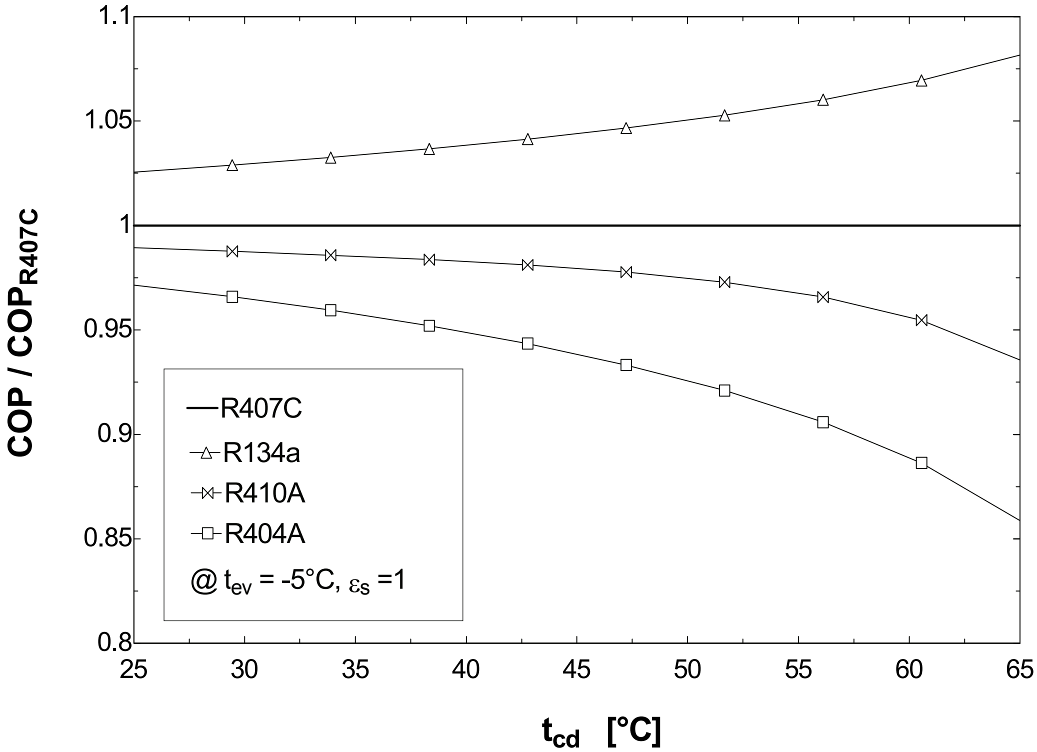

Figure 1. Refrigeration COP for different condensing temperatures

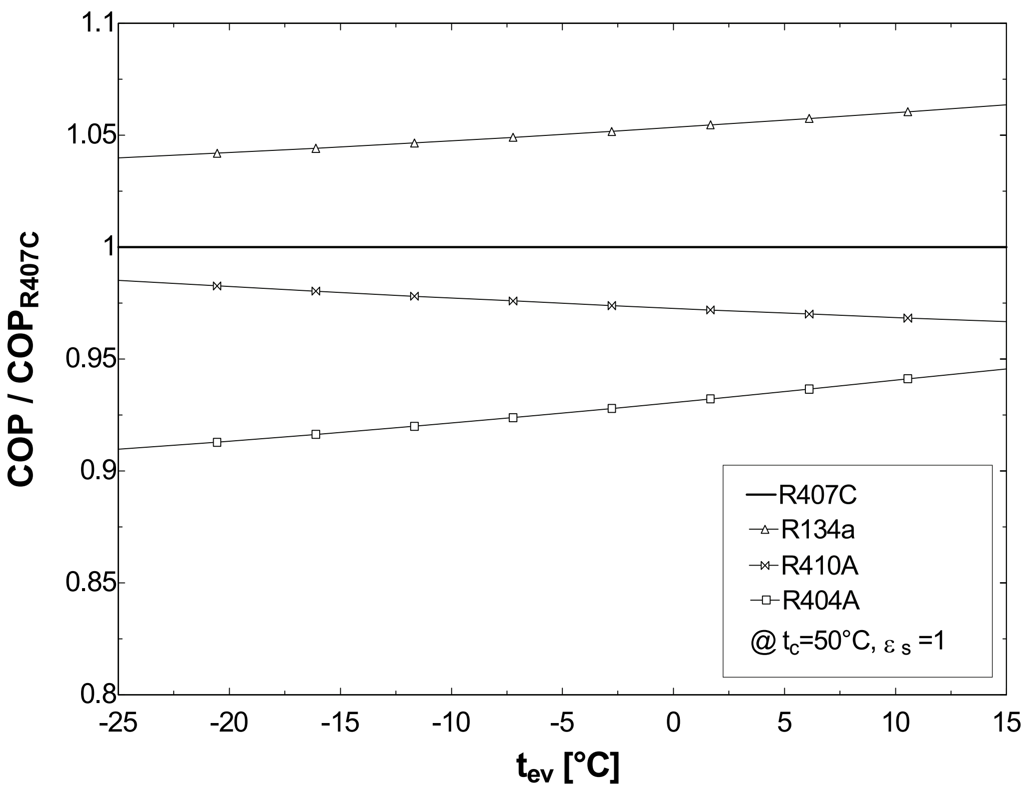

Figure 2. Refrigeration COP at different evaporating temperatures

Figure 3 shows the isentropic discharge temperature comparison. This does not give the good estimate of the real discharge temperature but gives the trends among the refrigerant options. It can be seen that R404A and R134a are “low discharge temperature” refrigerants, as it is well known, on the other hand R410A gives the highest discharge temperature of all. High discharge temperature means a restricted operation envelope, which can imply higher running costs due to the need of direct heating.

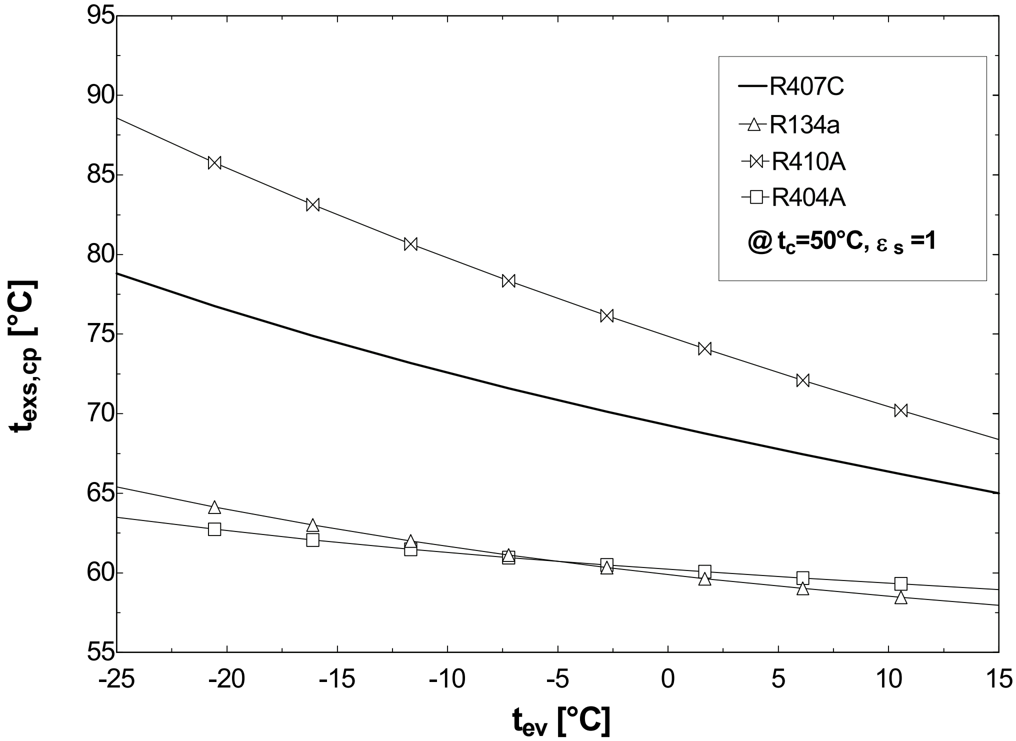

Figure 3. Isentropic compressor discharge temperature

2.2 Compressors

The compressor isentropic efficiency depends on the electromechanical losses (motor and friction) and thermo-fluid losses (mainly throttling and superheating). A compressor is in general optimised for one given refrigerant so this not completely fair to make a general statement on what happens with other refrigerants. Nevertheless the way the different losses vary with different refrigerants are known and some general trends can be given on what would happen in a specifically optimised compressor. Among all the losses mentioned above, the mechanical friction losses are probably the ones mostly variable with the refrigerant choice since its thermodynamic properties will dictate the needed swept volume and hence friction surface and friction pressures. So what can be expected is a lower isentropic efficiency for R134a compared to R407C and a slightly higher for R404A and even more with R410A.

The vapour injected R407C compressor is now starting to become available for R407C. The economiser cycle offers better theoretical COP at all conditions compared to the standard vapour compression cycle. Vapour injection also has the effect of cooling the compressor, thus extending the range of operation. Increased compressor capacity can offset the additional cost of the heat exchanger required for subcooling. Vapour injection is a very attractive option for ASHPs, delivering more capacity at extreme conditions, and ability to operate at lower ambient temperatures than the standard scroll.

2.3 System

Finally, the heat transfer properties can vary quite significantly with the refrigerant type. Higher-pressure refrigerants give better heat transfer properties. Typically R134a will need additional heat transfer surface to keep the same DT compared to R407C while the use of R410A can reduce the DT in the heat exchanger of a few K’s.

This last refrigerant is becoming widely used in air-conditioning applications despite a lower refrigerant COP but due to the compressor and systems benefits mentioned above (Hundy and Pham 2001).

Impact on costs: according to the above analysis, the use of R134a will dictate a larger compressor swept volume (more than 40%), which means an extra cost of about 25% compared to R407C. Also the heat exchangers will need more surface which typically increase its cost of about 10%.

R410A is more expensive, and components such as the compressor, the heat-exchanger and the controls designed for higher pressure and are at present more expensive.

From this discussion, R407C and R410A become the two short listed candidates based on performance, compactness and price. For special heat pump applications requiring high condensing temperatures if disregarding the price, R134a can be the only remaining choice.

3. SEASONAL PERFORMANCE

3.1 Method

Compressor selection software has been used to derive comparative seasonal performance data as described by Winandy and Hundy, 2007, 2008. This method uses the pre-configured heat pump characteristics of evaporator and condenser temperature differences, refrigerant superheat and sub-cooling. The fan/pump power is added and there is also an allowance for defrost de-rating in the software. Ambient temperature profiles from Meteonorm are used to define the heat load and the evaporating temperature in the case of air source. In this way comparative seasonal COP for various locations, refrigerants and compressor types can be readily investigated without the need for extensive tests. A comparison of performance characteristics with actual rating point data was made in the referenced paper (2008) with good results.

The model can be configured for constant heated water temperatures or for compensated control. For the purpose of this example, an R407C ASHP has been chosen and a comparison is made between a standard heating scroll and a vapour injected equivalent. The heating scroll has a discharge valve that enhances high pressure ratio efficiency, and the vapour injected scroll additionally has an intermediate pressure injection port. The heat pump temperature differences are taken as 7 – 10K (evaporator), 4K (condenser) with zero subcooling and 4K superheat. A 6% defrost de-rating is applied at evaporating temperatures below zero when the air temperature is above zero. Hot water delivery temperature is 55°C.

For compensated control the conditions are the same as above – the only change made is to set the water temperature at 55°C for ambient –10°C and it is then reduced by 0.7K for each degree rise in ambient temperature, to compensate for lower building heat loss.

3.2 Results

The results are summarised in Table 2. The improved SCOP with compensated control results from lower condensing temperatures during most of the season and correspondingly lower compressor run hours.

Table 2 SCOP comparison for heating only, location – Frankfurt

Heat pump compressor	SCOP Constant water temperature 55°C	SCOP Compensated Control with water temperature 55°C at –10°C ambient
Heating Scroll	2.06	2.85
Vapour Injected Scroll	2.69	3.18

The heat pump is sized for 20% excess capacity at +2°C air temp. Below 0°C, capacity becomes inadequate and back up heating required. This is shown as dark shaded in Figure 4.

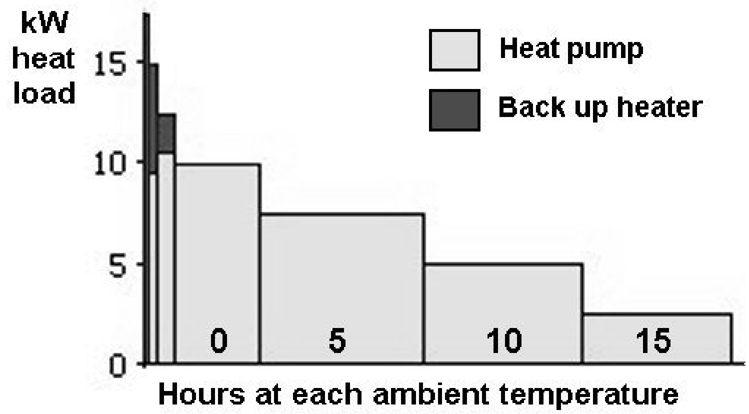

Figure 4 Heat load in each ambient bin. All heat in lowest bin supplied by back up heater

Compensated control has been set to give a maximum water temperature of 55°C at –10°C air temperature. If the air temperature falls below this value the water temperature is correspondingly increased, and there are just 4 hours in the –15°C bin. This demands a condensing temperature of 62.5°C which is outside the compressor operating envelope The calculation has been made with this heat supplied 100% by the back up heater. Other options are possible but the overall result changes very little due to the small number of hours involved.

The envelope of the standard scroll restricts its use at low ambient temperatures in this application. At air temperatures below –5°C water temperature of 55°C cannot be delivered and in this situation the SCOP is again calculated using 100% back up heat for these low ambient bins.

3.3 Addition of Domestic Hot Water (DHW)

DHW can be added in the model as a percentage of the heating load. In the following calculations the total heating load at 2°C is the same as before, except that 20% is taken for DHW supplied at a constant 58°C. The HP is assumed to cycle between the DHW load and the heating load as required, with the condensing temperature adjusted according to the compensated control requirement. This is set up in the same way as before.

The calculation for SCOP now includes the bins for summer period where the load is DHW only. This is shown in Figures 5 and 6. When the air temperature is 25°C and above the operating conditions are outside the envelope and the calculation is made on the assumption of back up heating for DHW at these conditions. Alternative options are pressure reducing valve to enable the compressor operation to continue or solar heat, and both of these options would benefit the SCOP. When the air temperature is –5°C and below back up heating is required for DHW (The compressor hours are prioritised for heating). Back up heating in these bins could be reduced with a larger compressor.

Ambient, °C	- 15.0	- 10.0	- 5.0	0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0	Total
Hours	4	60	208	1003	1960	1906	1802	1250	492	72	3	**6943**
Evaporating Temp, °C	- 22.00	- 17.00	- 12.14	- 7.50	- 2.86	1.79	6.43	11.07	15.71	20.36	25.00	
Heating												
Condensing Temp, °C	62.50	59.00	55.50	52.00	48.50	45.00	41.50	39.00	39.00	39.00	39.00	
Load, kW	13.81	11.83	9.86	7.89	5.92	3.94	1.97	0.00	0.00	0.00	0.00	
Capacity, kW	0.00	9.41	10.43	10.72	12.06	14.79	16.67	0.00	0.00	0.00	0.00	
Input Power inc Fan, kW	0.00	4.24	4.03	3.78	3.63	3.55	3.40	0.00	0.00	0.00	0.00	
Run hours	0.00	60.00	196.57	737.79	961.32	508.42	213.19	0.00	0.00	0.00	0.00	
% Run Time	0.00	100.00	94.51	73.56	49.05	26.67	11.83	0.00	0.00	0.00	0.00	
Capacity x Run Time, kWh	0.00	564.58	2051.11	7912.56	11596.67	7518.11	3553.94	0.00	0.00	0.00	0.00	**33197.0**
Input, kWh	0.00	254.54	791.94	2788.02	3487.27	1805.53	725.27	0.00	0.00	0.00	0.00	**9852.6**
Direct Heating, kWh	55.22	145.42	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	**200.6**
Total Heating Input, kWh	55.22	399.96	791.94	2788.02	3487.27	1805.53	725.27	0.00	0.00	0.00	0.00	**10053.2**
DHW												
Condensing Temp, °C	62.00	62.00	62.00	62.00	62.00	62.00	62.00	62.00	62.00	62.00	62.00	
Load, kW	1.77	1.77	1.77	1.77	1.77	1.77	1.77	1.77	1.77	1.77	1.77	
Capacity, kW	0.00	0.00	10.65	10.91	12.16	14.73	16.39	18.20	0.00	0.00	0.00	
Input Power inc Fan, kW	0.00	0.00	4.60	4.59	4.67	4.85	4.94	5.04	0.00	0.00	0.00	
Run Hours	0.00	0.00	11.43	163.13	285.98	229.60	195.06	121.85	0.00	0.00	0.00	
% Run Time	0.00	0.00	5.49	16.26	14.59	12.05	10.82	9.75	0.00	0.00	0.00	
Capacity x Run Time, kWh	0.00	0.00	121.72	1779.32	3477.04	3381.24	3196.75	2217.50	0.00	0.00	0.00	
Input, kWh	0.00	0.00	52.55	748.26	1335.00	1112.68	963.62	613.97	0.00	0.00	0.00	**4826.1**
Direct heating, kWh	7.10	106.44	1.19	0.00	0.00	0.00	0.00	0.00	872.81	127.73	5.32	**1120.6**
Total DHW Input, kWh	7.10	106.44	53.74	748.26	1335.00	1112.68	963.62	613.97	872.81	127.73	5.32	
Total % Run Time	**0.00**	**100.00**	**100.00**	**89.82**	**63.64**	**38.72**	**22.66**	**9.75**	**0.00**	**0.00**	**0.00**	

SCOP = 3.05

Figure 5 SCOP calculation – Location: Frankfurt, compensated control, with DHW at 58°C, vapour injected scroll

A comparison between three scrolls has been made for these conditions and loads, each HP having approximately the same capacity at 2/55°. The standard air conditioning compressor, and the standard heating compressor are compared with the vapour injected equivalent. These results are summarised in Figure 6.

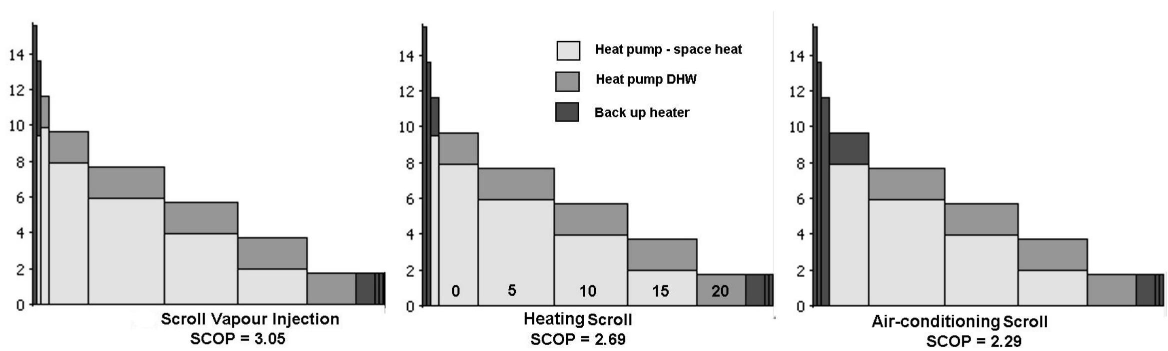

Figure 6 SCOP results comparing standard scroll types and vapour injection type

4. CONCLUSION

In summary R407C has advantages as a refrigerant for heat pump applications, but the climate at the location considered and the choice of the compressor model are influential factors. Among the refrigerants considered the differences are not dramatic and this may explain why essentially all refrigerants are found in existing European heat pumps today and the choice made by each supplier may very well be influenced by considerations other than SPF.

Whilst the air conditioning scroll exhibits high COPs at moderate conditions, its operating range limits its application in air source heat pumps. This is also partially true with heating scroll. The vapour injected scroll presents the best overall performance, and also delivers more capacity at low ambient conditions, reducing the need for back up heating.

One should keep in mind that this SPF calculation was performed for a specific heating system (air-to-water with wall-hung radiators) and other systems are important as well like ground-source to water and low temperature under-floor heating .It becomes clear that there is no easy and straight-forward approach to the choice of refrigerant. The fact that R407C is the predominantly used refrigerant in European heat pumps may be attributed to the fact that this refrigerant has advantages in some heat pump applications but suppliers also make the decisions for their refrigerant of choice base on cost, market region, product proliferation and equipment availability.

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