Capacity control of heat pumps

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In Sweden the current trend is to size heat pumps much closer to full coverage of the heating demand than has been the case so far. The driver for this development is both economic and legislative (increasing electricity price, possible power rates and limits in the building code regarding installed electric power for heating). This means that heat pumps will operate at part load almost all the time and that power utilization will be very low. This is particularly the case for new, low-energy buildings and has raised the interest in variable-capacity heat pumps.

General aspects of capacity control

Capacity control, primarily by means of Variable-Speed Drive (VSD) motors, is commonly used to improve the efficiency of operation of systems for heating, cooling and ventilation. However, before going into the specific application of heat pumps it may be pertinent to look at some of the generic aspects.

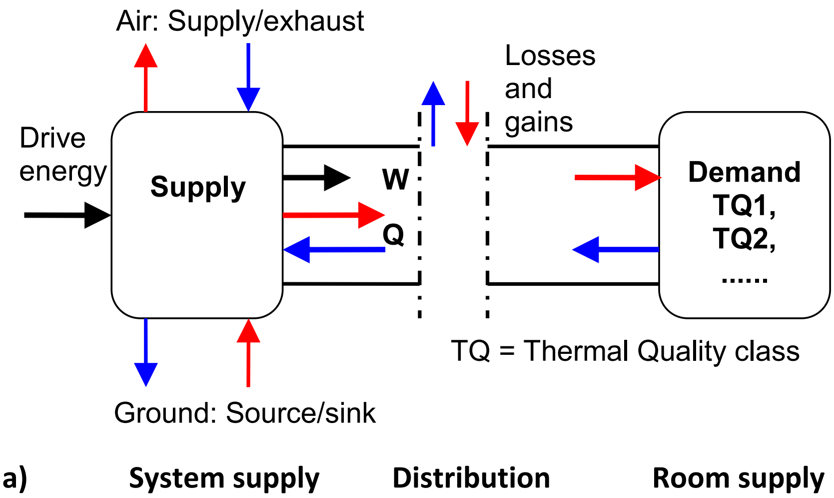
Background

Equipment for capacity control by means of frequency controlled motors for fans, pumps, compressors and other types of motor driven HVAC equipment appeared on the market already in the 1970s for pumps and fans and in the early 1990s for compressors in Japanese residential air conditioners. In those days the technology was considered to be very upmarket and consumers in my own country Sweden would pay up to 700 EUR over the price of a single-speed residential air-to-air heat pump (at that time a high-quality inverter which I bought for laboratory purposes only cost 200-300 EUR). Nowadays this technology is standard and the additional cost quite small. An important driver for the break-through of variable-speed drives in Japan was the availability of surplus capacity from a given size of compressor. It was rather common to have the air-conditioner turned off when no one was at home and to use the surplus capacity to quickly bring the room temperature down. With frequency control this can be achieved by feeding a 50-60 Hz motor with 100-120 Hz. This possibility is still of interest but perhaps not the main driver in the Swedish market. Instead the focus is on saving energy.

Matching of supply and demand

HVAC systems are sized to cover a planned maximum design load, e.g. at the design outdoor temperature for heating or cooling or at the design polluting load for hygienic ventilation. Normal operation is at substantially lower levels, e.g. the ventilation rate in VAV systems rarely exceeds 30% of the design flow. This means that large energy savings are possible for heating, cooling and fan energy by capacity control of the fan motor. Theoretically this means that the fan motor should be optimized for a drive power that is around 30% of the design power in pressure-controlled systems and for only 5% of the design power in future, decentralized systems[12]. Similarly, a heat pump sized for full coverage has 80-90% of the operating hours at a capacity which is less than half the design power.

Demand starts at room level and demand = -load, i.e. a heat load results in a cooling demand. A common experience is that use (= supply) is larger than actual demand. This difference tends to increase when the load factor (i.e. power utilization) goes down. It is more difficult to control a small energy supply with a large power than with a small power (see **Figure 1**).



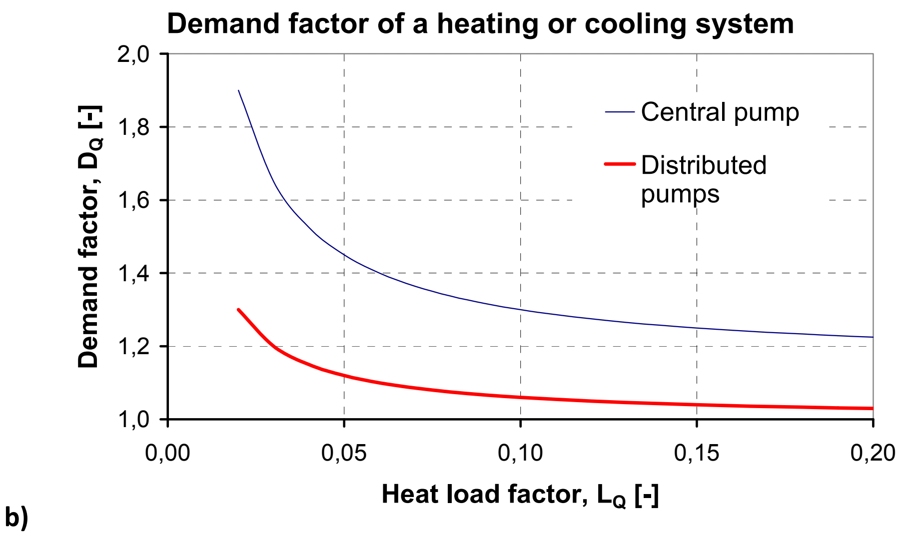


Figure 1. a) System supply, distribution and room supply of an HVAC system. b) Demand factor as a function of load factor. (Fahlén[14]).

Load matching may be accomplished by two main methods:

1. **Adapt the system to the supply** so that the entire supply capacity can be accepted without noticeable change to the room condition. Examples: Installation of a buffer tank, under-floor heating with a large thermal mass, a large diluting air volume per person etc.
2. **Adapt the supply to the system** so that only the required capacity is supplied: Example: Heat pump with variable-speed drive (VSD) compressor, demand-controlled ventilation (DCV) systems etc.

Both methods[9, 21] have been studied in heat pump related research projects at SP Energy Technology/Chalmers Building Services Engineering within the research programs Climate 21 and Effsys 1. Correctly designed, both methods may save substantial amounts of energy but this article will only cover alternative 2.

Potential advantages with capacity control

Capacity control of heat pumps is interesting for several reasons. The most obvious is that reduced thermal power will lower the temperature differences in the evaporator and the condenser and thus potentially increase the coefficient of performance. Another effect is that the temperature swing in the heating or cooling system will decrease relative to the situation with on-off control (see **Figure 2**). To achieve a given mean temperature of the room heater the mean temperature during actual operation of the heat pump must be higher (lower in the case of cooling).

The example in **Figure 2** yields an operating temperature that exceeds the set value by around 6 K, which reduces *COP* by approximately 15%. The problem is most pronounced in hydronic systems with a small water volume and/or systems with the heat pump directly connected to the heating system. The current praxis, at least in Sweden, is to use the same circulator for the condenser flow and the heating system flow and this means that when thermostatic valves close the condenser flow will drop and the condensing temperature will rise. It is better to separate the heat pump from the heating system and to have separate pumps for the condenser flow and the heating system flow (the heating system flow rate may be quite different from the optimal condenser flow rate).

Other examples of advantages with variable-speed drive capacity control are:

- fewer starts and longer operating times,
- reduced frosting and noise when air is used as heat source,
- possibilities to overrev the compressor and hence increase the capacity and avoid using a supplementary heat source,
- additional degrees of freedom in the control of sanitary water heaters etc.
- reduced latent cooling demand in refrigeration applications.

Many of the advantages are described in detail by Fahlén[5, 7] and Karlsson[21]. By intelligent use of sensors, which are needed anyway for the operating and supervisory control of the heat pump, it is possible to design a self-optimizing control system that controls the compressor capacity and matches this with optimized control of the condenser and evaporator flows. This IRM system may also optimize the defrost control and provide diagnostic FDD information (FDD = Fault Detection and Diagnosis; IRM = Integrated Refrigeration Management[5]).

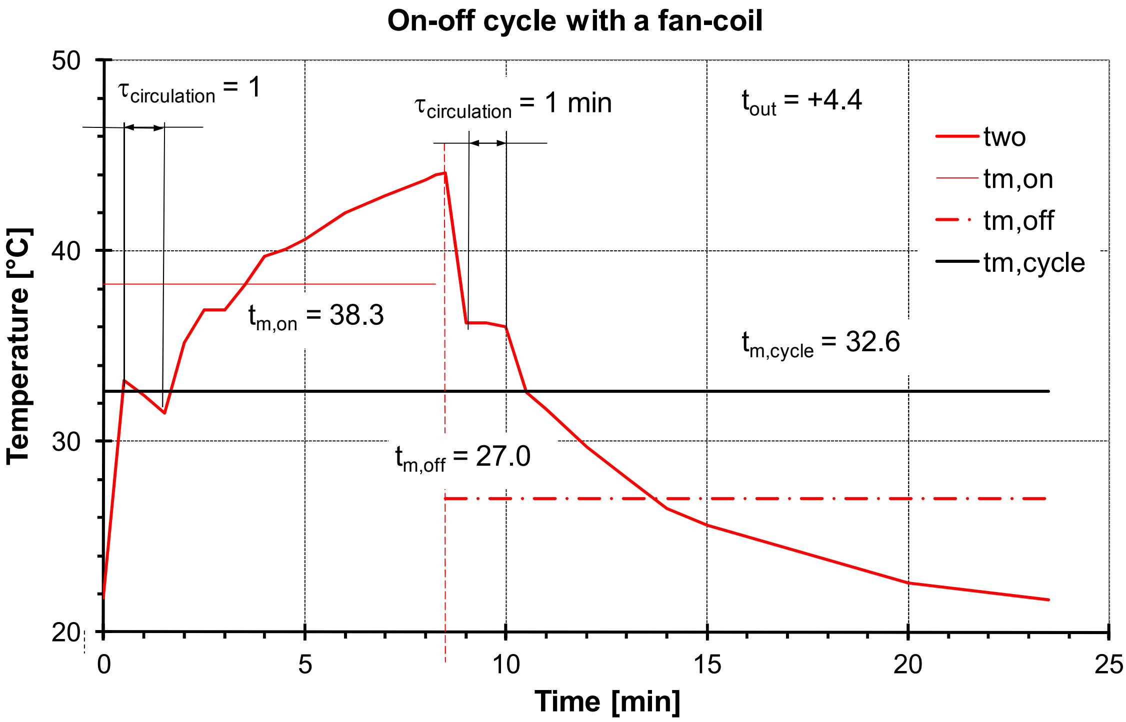


Figure 2. One complete on-off cycle with a heat pump connected to a combined radiator and fan-coil system (Fahlén[9]).

Methods for capacity control

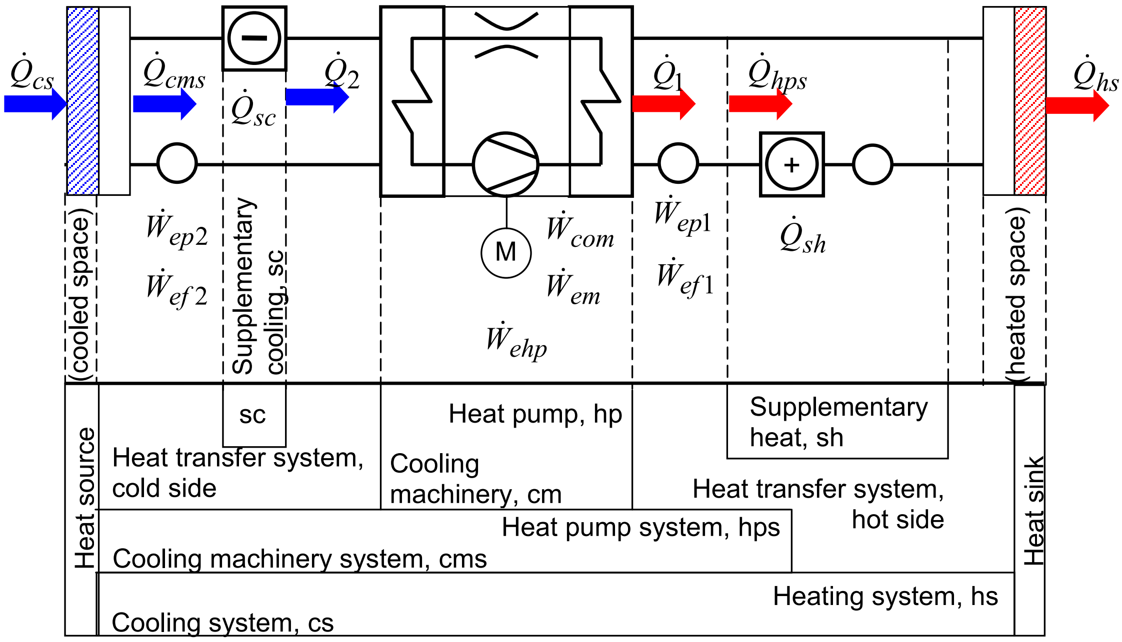
There are many ways of controlling the cooling and heating capacity of a heat pump, mechanical as well as electric. One proven alternative is to split the design capacity between two or more smaller units and sequentially control their start and stop. Already with two power levels and some thermal inertia in the heating or cooling system it is possible to achieve rather a large share of the potential gains of a perfect continuous control. Furthermore, this provides a certain amount of redundancy which may be an advantage in connection with service etc. Naturally, several power steps may be included in the same unit. In older units, various forms of cylinder unloading, suction pressure control etc. were used with the common trait that capacity could be lowered but at the expense of a much reduced coefficient of performance. Currently, the dominating method is continuous capacity control using variable-speed drive of electric motors by means of various types of frequency control. This article will only deal with this type of capacity control.

Variable-speed drive of electric motors is a well-known technology (I have personally designed equipment with VSD operation of pumps and fans since 1977). Less known, at least outside the circle of electric motor designers, is that the matching of motor and controller type is critical for the end result. It is not uncommon to find retrofitted VSD equipment to existing motors that certainly deliver a reduced capacity but without substantial reduction of the drive power.

Motor torque and power requirements

A heat pump system comprises not only a compressor but also flow generators such as pumps and fans. These are required to provide heat transfer flow rates in the evaporator and condenser as well as to transport heat between the heat pump and the cooling or heating load (c.f. the system diagram in **Figure 3**).

The character of the mechanical work that these flow generators have to produce differs somewhat from that of the heat pump compressor and thus provides slightly different operating conditions for the respective motor and motor drive.



$$\dot{W}_{ep2} = \sum_{j=1}^{j=n} \dot{W}_{ep2,j} \qquad COP_{hp} = \frac{\dot{Q}_1}{\dot{W}_{ehp}} \qquad \dot{W}_{ep1} = \sum_{j=1}^{j=n} \dot{W}_{ep1,j}$$

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1 = condenser	hp = heat pump	cm = cooling machine	e = electric
2 = evaporator	hps = heat pump sys.	cms = cooling machinery system	f = fan
	hs = heating system	cs = cooling system	com = compressor
	sh = supplement. heat		m = motor
			p = pump
			p = "parasitic"

Figure 3. System diagram for heat pump systems according to Fahlén[6]. $\dot{Q}_1$ = condenser power, $\dot{Q}_2$ = evaporator power, $\dot{W}_{com}$ = compressor mechanical shaft power, $\dot{W}_e$ = electric power.

Pumps and fans

Heating and cooling systems typically use radial-type flow generators even though there may be axial fans in large systems. Pumps almost without exception use B-wheels (backward-oriented blades) and due to superior efficiency this is becoming standard also for ventilation fans. However, evaporator fans in residential air-conditioning heat pumps are often of the axial type and even cross-flow fans may be used as condenser fans (low efficiency but also low noise generation).

The torque requirement of the flow generator depends mainly on the flow character of the external system. Pressure drop has an exponential dependence on flow rate and we have the following relation (fully developed turbulent flow has $n \approx 2$):

$$\Delta p(\dot{V}) = \Delta p_0(\dot{V}_0) \cdot \left(\frac{\dot{V}}{\dot{V}_0} \right)^n \qquad \text{[Pa]} \qquad \text{eq. 1}$$

where $\dot{V}_0$ is a specified reference flow rate (e.g. the design flow rate), $\dot{V}$ is an arbitrary, controlled flow rate and $1.7 < n < 2$ is the flow-related pressure drop exponent. Filters and some types of heat exchanger work in the laminar regime and hence have $n \approx 1$. Torque is proportional to the pressure difference that the machine is working with and thus the flow-related torque may be expressed as:

$$T(\dot{V}) = T_0(\dot{V}_0) \cdot \left(\frac{\dot{V}}{\dot{V}_0} \right)^n \qquad \text{[Nm]} \qquad \text{eq. 2}$$

with $1 < n < 2$ (mostly in the range 1.5 – 2) and T = torque. A standard radial machine generates a flow which is proportional to the rotational speed and hence we have:

$$T(\dot{V}) = T_0(\omega_0) \cdot \left(\frac{\omega}{\omega_0} \right)^n \qquad \text{[Nm]} \qquad \text{eq. 3}$$

with ω as the angular frequency of the rotating part. The relation between angular frequency ω [rad/s] and rotational speed N [r.p.m.] is given by $\omega = 2 \cdot \pi \cdot N / 60$. The corresponding power requirement $\dot{W}_p$ of pumps or fans is given by the product of torque and angular frequency:

$$\dot{W}_p(\omega) = T(\omega) \cdot \omega = T_0(\omega_0) \cdot \left(\frac{\omega}{\omega_0} \right)^{n+1} \qquad \text{[W]} \qquad \text{eq. 4}$$

Finally, the power requirement of the electric motor becomes:

$$\dot{W}_{e,p}(\omega) = \frac{\dot{W}_p(\omega)}{\eta_{e,m}(\omega) \cdot \eta_p(\omega)} = \frac{\dot{W}_{p,0}(\omega_0)}{\eta_{e,m}(\omega) \cdot \eta_p(\omega)} \cdot \left(\frac{\omega}{\omega_0} \right)^{n+1} \qquad \text{[W]} \qquad \text{eq. 5}$$

The relations eq. 3 and eq. 5 show that torque as well as power tend[22, 25] to zero as rotational speed (angular frequency) approaches zero. This is advantageous for the electric motor as it will start unloaded and thus have a very low starting current and this will also be advantageous from a grid perspective. Displacement type compressors, however, will have a very different torque characteristic.

Displacement compressors

Displacement compressors, e.g. piston, rotary piston, vane, Scroll and similar types, have a given transported volume V per revolution. In the case of a heat pump compressor the pressure difference will not primarily depend on the flow rate (compressor speed) but rather on the temperature difference between condenser and evaporator. The saturation pressures of the refrigerant in the condenser and evaporator, p_1 and p_2 respectively, see **Figure 3** for designations, depend directly on the condensing and evaporating temperatures t_1 and t_2. It is the temperature-related difference between condenser and evaporator that decides the torque on the compressor motor.

Another difference between a compressor and a pump or fan is that the fluid no longer may be considered as incompressible (hence the name compressor). Granryd[20] et al provide the following expression for the mechanical shaft power of a displacement compressor:

$$\dot{W}_{com} = \frac{\eta_s}{\eta_{is}} \cdot \frac{\kappa}{\kappa - 1} \cdot p_{in} \cdot \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{\kappa-1}} - 1 \right] \cdot \dot{V}_s \qquad \text{[W]} \quad \text{eq. 6}$$

with the following designations: $\dot{W}_{com}$ = mechanical shaft power, η_{is} = isentropic efficiency, η_s = swept volume efficiency, κ = ratio between the specific heats at constant pressure and constant volume of the refrigerant ($\kappa = c_p / c_v$), $p_{in} \approx p_2$ = inlet pressure of the compressor, $p_{out} \approx p_1$ = outlet pressure of the compressor, V_s = swept volume of the compressor (displacement) and $\dot{V}_s$ = swept volume flow rate of the compressor. By replacing the volume flow rate with the compressor displacement multiplied by the rotational speed (angular frequency) the following relation may be derived:

$$\dot{W}_{com} = \frac{\eta_s}{\eta_{is}} \cdot \frac{\kappa}{\kappa - 1} \cdot p_{in} \cdot \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{\kappa-1}} - 1 \right] \cdot \frac{\omega}{2\pi} \cdot V_s \qquad \text{[W]} \quad \text{eq. 7}$$

with $\dot{V}_s = \frac{\omega}{2\pi} \cdot V_s$ and the rotational speed $N = \frac{60 \cdot \omega}{2 \cdot \pi}$. Hence the torque may be expressed as:

$$T_{com} = \frac{\dot{W}_k}{\omega} \approx \frac{\eta_s}{\eta_{is}} \cdot \frac{\kappa}{\kappa - 1} \cdot p_2 \cdot \left[\left(\frac{p_1}{p_2} \right)^{\frac{\kappa}{\kappa-1}} - 1 \right] \cdot \frac{V_s}{2\pi} \qquad \text{[W]} \quad \text{eq. 8}$$

with $p_{out} \approx p_1$ and $p_{in} \approx p_2$. The equation eq. 8 indicates that contrary to the case with pumps or fans, the torque of a displacement compressor depends not so directly on rotational speed but rather on the pressure ratio $\pi = p_1 / p_2$ (in this simple analysis speed-related frictional forces and flow losses have been neglected). The torque depends also on the absolute level of the evaporator pressure and therefore has a maximum at a certain inlet pressure (varies with the type of refrigerant; **Figure 4** provides an example with R290 propane).

Introducing a reference value for the torque, $T_{com,0}$, at the design values for the condensing and evaporating pressures $p_{1,0}$ and $p_{2,0}$, it is possible to express the torque at an arbitrary operating condition as:

$$T_{com} = T_{com,0} \cdot \frac{p_2}{p_{2,0}} \cdot \frac{\left[\left(\pi(p_1, p_2) \right)^k - 1 \right]}{\left[\left(\pi_0(p_{1,0}, p_{2,0}) \right)^k - 1 \right]} \qquad \text{[Nm]} \quad \text{eq. 9}$$

with $T_{com,0} = \dfrac{\eta_s}{\eta_{is}} \cdot k \cdot p_{2,0} \cdot \left[\left(\dfrac{p_{1,0}}{p_{2,0}}\right)^k - 1\right] \cdot \dfrac{V_s}{2\pi}$ och $k = \dfrac{\kappa}{\kappa - 1}$.

Depending on the operating conditions the compressor torque may vary from zero, with pressure-equalized start or the same temperature indoors and outdoors, to varying maximum torque in relation to pressure according to eq. 9 and **Figure 4**. The power requirement of the compressor is obtained by multiplying the torque and the angular frequency (rotational speed).

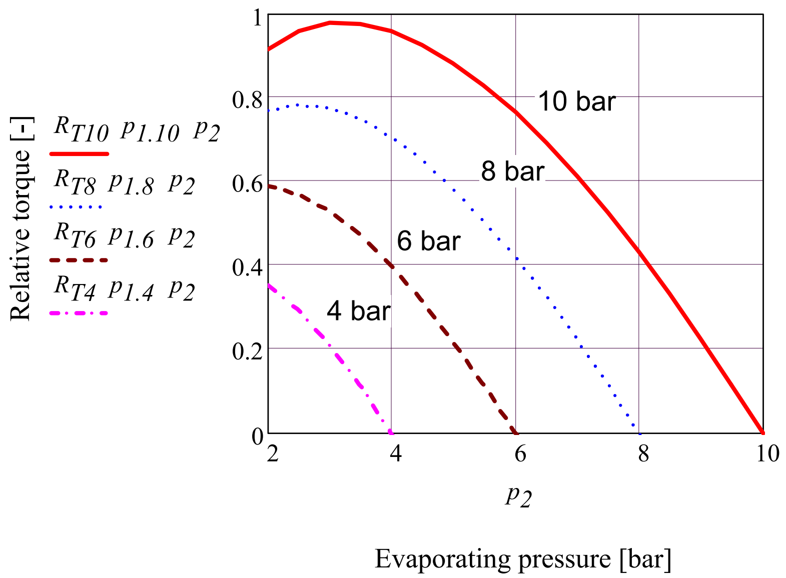

Figure 4. Relative torque $T = T_{com} / T_{com,0}$ as function of evaporating pressure p_2 with $p_1 = p_{1,0} = 10$ bar, $p_{2,0} = 2$ bar (based on an example according to Fahlén with an air-to-air heat pump and propane, R290, as refrigerant: $t_{1,0} = 29$ °C, $p_{1,0} = 10.5$ bar and $t_{2,0} = -3$ °C, $p_{2,0} = 4.3$ bar).

Differentiation of eq. 9 with respect to p_2 indicates that the torque has a maximum, which is also visible in the diagram. The maximum value appears at the suction pressure:

$$p_2 = p_1 \cdot (1-k)^{\frac{1}{k}} \text{ with } k = \frac{\kappa}{\kappa - 1} \qquad \text{[bar]} \quad \text{eq. 10}$$

With eq. 9 as the starting point of a sensitivity analysis (logarithmic differentiation) it is possible to see how the changes Δp_1 och Δp_2 of the condensing and evaporating temperatures affect the motor torque:

$$\frac{\Delta T_k}{T_k} = (k-1) \cdot \pi^{k-2} \cdot \frac{\Delta p_1}{p_1} + \left(1 - (k-1) \cdot \pi^{k-2} \cdot \frac{p_1}{p_2}\right) \cdot \frac{\Delta p_2}{p_2} \qquad \text{[-]} \quad \text{eq. 11}$$

The pressures can easily be converted to temperatures by means of the Clausius-Clapeyron relation (Fahlén[15]).

Electric motors

The low efficiency of electric motors that appear in residential units has been an obstacle for achieving high seasonal performance factors. When I started evaluating residential heat pumps in the early 1980s I first thought that something was wrong with the measurements but the pumps and fans had measly total efficiencies of around 3-4%, largely depending on the motors. Also the compressor motors had relatively low efficiencies. This was confirmed during discussions I had in Japan with the R&D departments of several major manufacturers in connection with my writing the competition specifications of the Nordic heat pump competition in 1993. Better electric motors were high on the agenda.

One reason for the low efficiency of ordinary asynchronous motors is that they will always draw a certain current for the excitation the magnetic windings. No energy for useful work will be taken from the magnetic field and the magnetizing power will thus be a total loss. In particular during part-load operation this becomes significant; the magnetizing power remains more or less unaffected whereas the useful power is reduced. These effects are relatively more important for small motors than for large motors and thus the introduction of permanent magnet motors has had a tremendous impact for the development of more efficient small fans, pumps and compressors (the permanent magnets draw no current at all for magnetization).

There are many alternative configurations for motors and motor drives but two alternatives are of special interest for small motors, the DC motor with permanent magnets and electronic commutation (EC = electronically commutated or BLDC = brushless direct current motor) and the synchronous AC motor with permanent magnets (PMSM = permanent magnet synchronous motor). They appear similar in their construction but the BLDC is fed by a rectangular-pulse current (pulsed DC) and the PMSM is fed by a sinusoidally varying current.

Over the past twenty years I have personally been involved in many various projects with the aim of achieving a better match between actual demand and the supply or removal of air and heat ("the right quality and the right quantity at the right time and the right place"). Since the mean power invariably is much lower than the maximum design power, the motor efficiency at part-load is all important for the seasonal system efficiency. For example, as previously mentioned, the average occupancy in office buildings is only around 30% and hence the typical power requirement of the ventilation fan motor will only be $(0.3)^3 = 0.03 \approx 3\%$ of the maximum power (c.f. eq. 5 and $n = 2$) in systems with control-on-demand ventilation and decentralized fans with direct flow control. In systems with central AHUs and pressure-controlled fan operation, the theoretical drive power only drops linearly or slightly more with variable supply pressure control; regarding the theoretical prerequisites, see reports by Fahlén[12] and Markusson[22].

Corresponding situations exist also for pump and compressor operation. A heat pump sized for full coverage of the heating demand will deliver around 90% of the heat at a mean capacity less than half the design value and a very large part at less than 25% of the maximum motor power (varying conditions at the heat source and heat sink will make the motor power drop relatively more than the reduction in heating capacity).

To reap the full potential benefits of capacity control [12-14] it is necessary to conceive new systems for heating, cooling and ventilation, to develop new components that can cope with large turn-down ratios while maintaining acceptable efficiency and finally to establish smart control strategies for the overall operation of the systems. Therefore, Chalmers/SP have worked systematically with all these aspects since the 1980s and in recent years also made some major advancement in the specific areas of motors and motor drives. I a Ph.D. project with two Ph.D. students we studied HVAC system design as well electric motor development (see

Fahlén[10] 2004). The project has resulted in two theses, one regarding new HVAC system design by Caroline Markusson[22] and one on new motors and drives by Johan Åström[25].

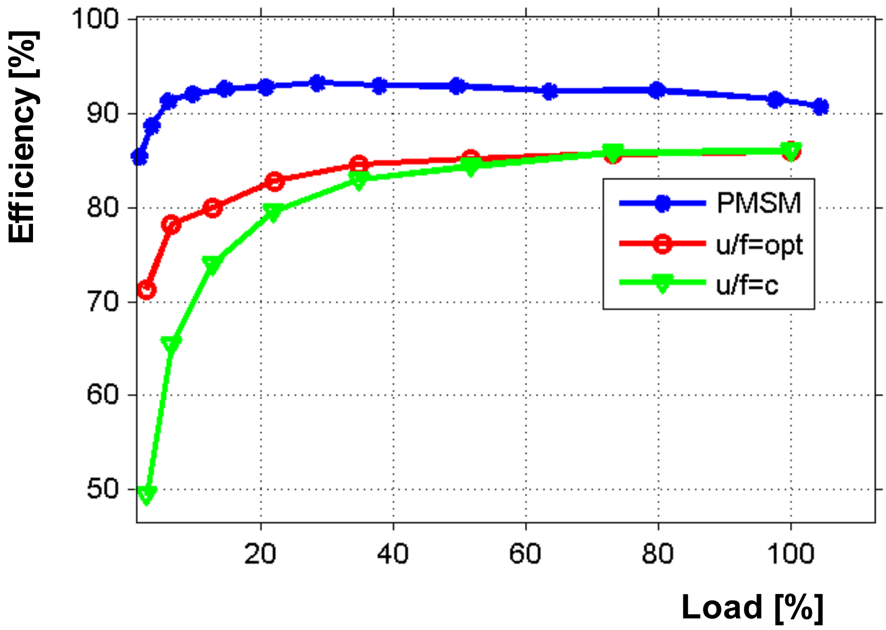

Figure 5. Comparison by Åström[25] of three alternative motor configurations. The two lower represent current practices and the top curve represent a new PMSM motor developed by Åström.

Åström has developed calculation models to customize the combination of motor and drive and in this he has handled rectifier losses, inverter losses and motor losses, not only theoretically but also experimentally. **Figure 5** illustrates a comparison between three alternative motor topologies. The two lower are based on a modern eff1 (IEC2) asynchronous motor with a design power of 4 kW and a quadratic torque characteristic $T = k \cdot \omega^2$. The curve with the lowest efficiency represents a constant ratio between voltage and frequency (u/f = c) and the slightly better curve a variable, optimal ratio (u/f = opt). The highest efficiency is provided by a new motor topology (PMSM) with matching drive and control that was developed within the project.

As shown by the diagram, the new development retains its high efficiency way down in the load range. The difference between a standard, high efficiency, variable-speed asynchronous motor and the new development with optimized control is 7% at 100% load and almost 15% at 20% load (there is also a newly developed BLDC alternative that is even better). This achievement was one of the major objectives of the work since most of the operating hours of HVAC systems will be at very low loads if they have good control-on-demand operation. This will be further accentuated in modern nZEBs (near-zero-energy buildings) where power utilization is substantially lower than in older buildings (energy demand will drop relatively more than will the design power).

Heat pump application example

Figures 6 – 8 below illustrate the importance of matching the heat pump compressor control with adapted distribution powers (”parasitic powers” to pumps and fans) while at the same time maintaining the efficiency of the electric motors. The diagrams derive from development of a rotating air-to-air heat pump (an EU project where Chalmers and SP lead the scientific part; total budget 2.8 MEUR).

Figure 6 presents a comparison of three alternative motor topologies that were used to simulate the coefficients of performance of the heat pump (COP_{hp}) and the heat pump system (COP_{hps}) respectively. **Figure 7** shows how the coefficient of performance of the heat pump system varies as function of the fractional capacity ($0 \le f \le 1$) at the operating condition +7 °C outdoor temperature and +20 °C indoor temperature. In this example the fans are uncontrolled with the implication that their powers become relatively more important in relation to the output of the heat pump. This also implies that with a sinking efficiency of the compressor motor the *COP* will not improve by much in the base case. The modern motor and our new concept, however, will provide substantial improvements.

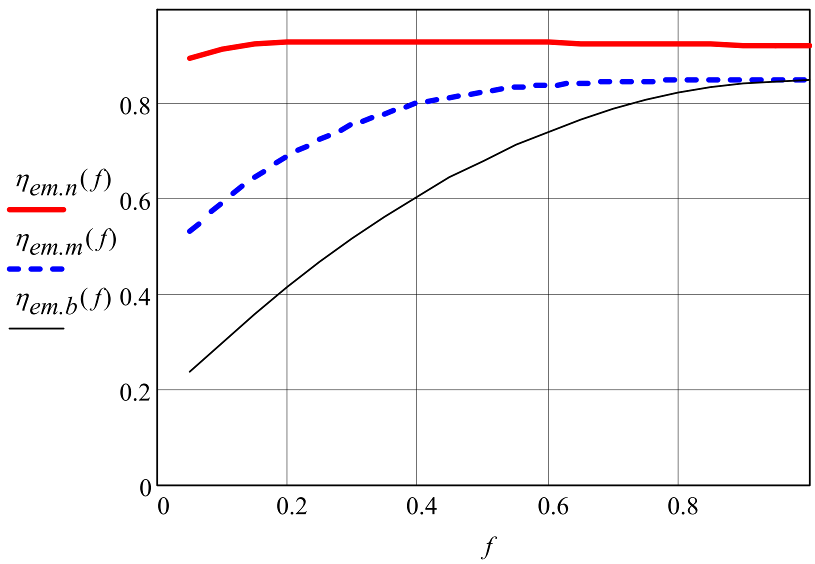

Figure 6. Motor efficiency (η_{em}) as a function of the fractional capacity (f). Index *b* represents a base case with today's standard practice, *m* corresponds to modern, state-of-the-art equipment and *n* is our newly developed concept (motor power around 1 kW).

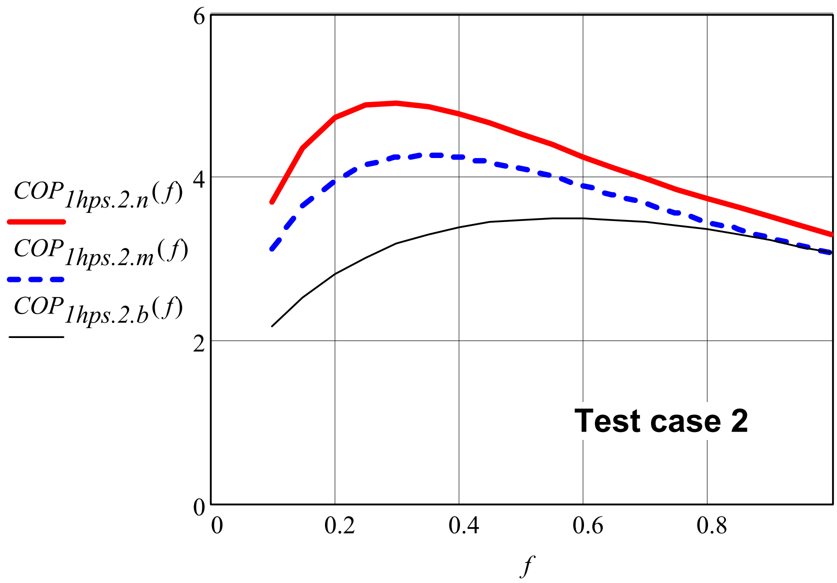

Figure 7. The coefficient of performance of the heat pump system (COP_{hps}) as function of the fractional capacity ($0 \le f \le 1$) for three alternative motor topologies. Powers to the evaporator and condenser fans are constant.

Figure 8 provides the same comparison as **Figure 7** with the difference that also the powers of the evaporator and the condenser fans are reduced as the compressor power goes down. This also means that the air flow rates will drop in relation to the case of **Figure 7** and this will make the condensing temperature rise and the evaporating temperature drop compared to **Figure 7**. The reduction of fan powers, however, will be larger than the increase in drive power to the compressor and the end result is substantially improved part-load values for COP_{hps} in **Figure 8**. With the new motor technology, *COP* continues to increase all the way down to $f = 0.10$ and will then be an impressive 7.4!

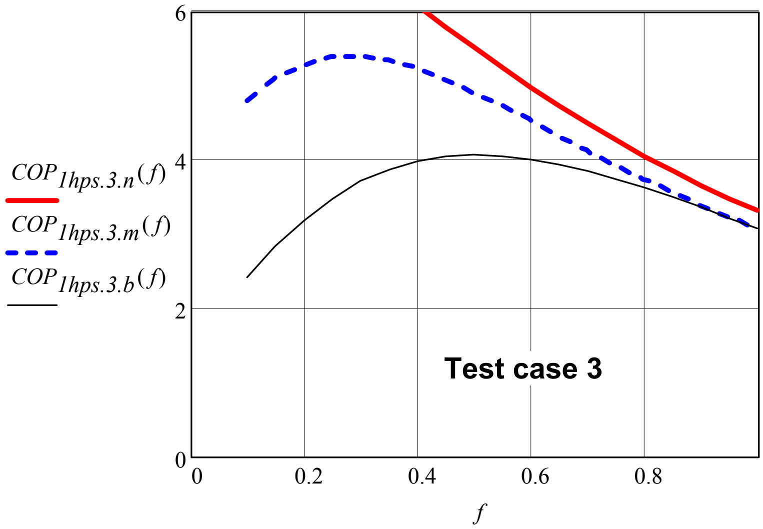

Figure 8. Coefficient of performance of the heat pump system (COP_{hps}) as function of the fractional capacity ($0 \le f \le 1$) for three motor topologies. Evaporator and condenser fan powers vary in relation to the relative thermal capacity.

Table 1 provides an overview of the very large impact that system design, sizing and motor characteristic has on part-load performance.

Table 1. The heat pump system coefficient of performance of alternative system designs at 100, 50 and 20% load (hps = heat pump system; 1-3 are the alternative system options; motor type: b = base, m = modern and n = new design).

Alternative	*COP*	*f* = 1.0	*f* = 0.5	*f* = 0.2
1. Constant evaporating and condensing temperature.	$COP_{1hps.1.b}$	3.1	2.4	1.4
	$COP_{1hps.1.m}$	3.1	2.8	2.2
	$COP_{1hps.1.n}$	3.3	3.2	2.8
2. Constant flow rates, constant fan powers. Decreasing temperature lift.	$COP_{1hps.2.b}$	3.1	3.5	2.8
	$COP_{1hps.2.m}$	3.1	4.1	4.0
	$COP_{1hps.2.n}$	3.3	4.5	4.7
3. Adapted flow rates, decreasing fan powers. Decreasing temperature lift.	$COP_{1hps.3.b}$	3.1	4.1	3.2
	$COP_{1hps.3.m}$	3.1	4.9	5.3
	$COP_{1hps.3.n}$	3.3	5.5	7.1

Research and development

Chalmers and SP have worked in many projects related to capacity control of heat pumps over the years with the following main objectives:

- Demand analysis[11] (temporal and spatial variations of cooling and heating demands)
- Load matching[12, 14] (primarily system aspects and related control strategies; the actual capacity control is trivial),
- Control of auxiliary (parasitic) powers[10, 12, 14],
- Applications: Space heating[9, 11, 21], sanitary water heating[7, 17], comfort cooling[18], supermarket refrigeration,
- Defrosting[5] (effect of area-specific coil load on frosting, control strategy etc.),
- Test methods[8].

In space heating applications the first heat pump category to benefit from developments in capacity control[24] has been the residential air-to-air heat pump. This is not unexpected as the large variation in outdoor temperature results in large excess capacity during spring and autumn if the heat pump is sized for reasonable energy coverage. Air-source heat pumps also experience an additional benefit from substantially reduced frosting as the area-specific cooling load on the outdoor coil is reduced.

The trend in the heat pump market is towards rising energy coverage. Higher electricity prices, the possibility of hourly tariffs and new power rates as well as new requirements in the building code affect the possibility of using electricity for peak heating (the standard alternative so far). Thus it comes natural to size all types of heat pump as close to full coverage as possible by overreving the compressor on the coldest days. This applies also to exhaust-air heat pumps which can be made to cool the exit air to very low temperature by means of variable capacity.

The same advantages as with outdoor air heat pumps can be obtained also in cooling applications[23] for comfort as well as for food storage. Cooling applications also have the additional benefit of avoiding unwanted dehumidification and this substantially reduces electric drive power to the compressor. In cooling of unwrapped food, this also avoids the risk of negative influence of the food quality from dehydration.

In spite of participation by many Swedish heat pump manufacturers in our research projects and the many theoretical advantages which we have high-lighted during these projects, capacity control has not yet experienced the same market impact in ground-source applications as for air-source systems. The reason is mainly the additional cost which is not fully recovered by the efficiency gains. Under par efficiency gains may have real and as well as fictitious causes.

The real ones are related to lack of high-efficiency variable-speed compressors, partly due poor motor efficiency during capacity turn down and partly due non-optimal matching of pump and fan power (air-to-air heat pumps have compressor and motor technologies specifically designed for this application). Professor Granryd showed already in 1974[19] that there is a fairly simple relation between cooling or heating capacity and the optimal power for operating the flow generators at the evaporator and condenser respectively. Fredrik Karlsson[21] has later, in one of our projects, shown that a linear relationship between pump or fan power and the compressor power provides a result which is close to the optimal provided that sizing at full capacity has been made correctly. However, the design value for the parasitic power ratio at maximum capacity is commonly too high in current systems but this is not revealed in testing according to European standards (see below).

The fictitiously missing performance gain relates to the negative influence on technical development caused by European rating standards in the heat pump sector. Standard performance data are given only for the most favourable temperature conditions and with marginal impact of the parasitic powers.

Hence the benefits from capacity control during normal operation will not be displayed. Also, from a systems perspective, it is at least as important to have efficient control of the terminal equipment[12, 16] (air heaters, air coolers, storage water heaters, display cabinets etc.) as to control the heat pump per se.

Early on (from the 1980s and onwards) SP developed test methods for steady-state as well as for transient operation. In order to have a reasonably fair comparison of different heat pump systems, we dealt rather carefully with the parasitic drive powers[1] and conducted diurnal[2] measurements of complete heat pump systems for heating as well as for hot water. Already when the first capacity-controlled air-to-air heat pumps appeared, we were able to test[3, 4] these and the results showed many of the benefits described above. We[8] also made a proposal for a European part-load rating standard around 15 years ago but unfortunately there is still no official European issue that makes it possible to show the full advantage of capacity control. Work has been going on for a long time and is supposed to be completed by now so hopefully the outcome will be worth our while.

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