FLOWRATE DEPENDENT HEAT-PUMP MODEL

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Abstract: The EU's energy efficiency strategy lead to new requirements for space-heaters. The assessment for electrically driven heat pumps is based on standard EN 14825. Required data may be derived from established rating measurements according to EN 14511. Based on such a calculation, a refined model has been developed and successfully validated for an air-to-water heat-pump. The algorithm is based on Carnot's law at the refrigerant cycle. By a linear interpolation of exergetic efficiency, operating points that fall in between rated points are made available to the calculation. The algorithm also incorporates a functionality that estimates the behaviour of the heat exchangers. These refinements to existing models become important as rating data is usually available for operating conditions that are different from real-world conditions, where temperature spread at the sink side interacts with the load. The model is based on widely available data and can therefore be applied to many heat-pumps. As it originally has been developed for the use in standard calculations, it's one of its strength that results of such assessments and simulations can be compared.

Key Words: heat-pump, model, simulation

1 INTRODUCTION

As heat-pumps operate under constantly changing conditions with varying efficiencies, their assessment is a challenging task. For a good estimation of yearly energy consumption, an accurate modeling of all operating points is necessary. This study presents such a model which was implemented for the Matlab-Simulink® simulation environment. As air-source heat pumps are typically the most exposed to changing source temperatures, the model has been developed keeping these units in mind.

2 BASIC ASSUMPTIONS AND MODELLING

Available heat pump data shows that there is no strong dependency between heating capacity and flow temperature, as long as conditions on the source side are kept identical. Table 1 shows exemplary data of an air-to-water heat-pump. As can be seen, heating capacity reduction of the unit is well below 5 % if flow temperatures increases from 35 ^{0}C to 55 ^{0}C.

Table 1: Heating capacity vs. flow temperature of an air-to-water heat-pump. Data by courtesy of Vaillant Deutschland GmbH & Co. KG (The unit does not necessarily correspond to a series production type)

		A-7	A2	A7	A12
W35	heating capacity	7.32	9.04	10.21	11.18
W55	heating capacity	7.05	8.78	9.98	10.98

This leads to the assumption, that a simple linear interpolation is more than accurate for the determination of heating capacity. Unfortunately, while rated data is based on outlet temperatures, usually return temperatures from the heating systems are given in a real-world application. This is important as today's heat-pump technology will lead to an over temperature of the outlet when demand is low, that is at higher outdoor air-temperatures when heating capacity increases. To correctly model this effect, outlet temperature and heating capacity are calculated in an iterative process based on the given inlet temperature and massflow:

$$T_{sink,outlet} = T_{sink,inlet} + \frac{P_{heat}(T_{sink,outlet})}{\dot{m}_{sink} \cdot c} \quad (1)$$

Once sink and –given– source-temperatures are known, the COP can be found in a look-up table of rating points using linear interpolation for intermediate temperature values.
While this would already deliver in fairly good results, it can be further improved by taking some more care of the electric power consumption, which is much more influenced by outlet temperature (again considering equal source conditions). Comparisons between calculated and measured data show, that the improvement becomes evident especially at higher flow temperatures an low loads. Instead of simply looking up electrical power consumption, this value can be evaluated via a (linear) interpolation/extrapolation of the Carnot-efficiency η_{HP} from the nearest known values (For best results it's important to use rating data close to the operating limits):

$$P_{el} = \frac{P_{heat}}{COP_{carnot} \cdot \eta_{HP}} \quad (2)$$

The perfect (maximum) efficiency COP_{carnot} of a heat pump is described by Carnot's law and is depending only on temperature levels of the cold and hot temperature reservoirs. The Carnot process corresponds best to the refrigerant cycle of a real heat-pump, so temperatures at the refrigerant level are required. This introduces more variables in the calculation process: Mean temperatures at the evaporator and condenser (that is both, inlet and outlet temperatures) and temperature drop across each of these heat exchangers. Using heating capacity, outlet temperature at the condenser, inlet temperature at the evaporator and corresponding flow rates as input values –given or calculated according to the section above– the temperatures at the refrigerant cycle can be calculated. Looking at the sink side:

$$T_{sink,refrigerant} = \frac{T_{sink,inlet} + T_{sink,outlet}}{2} - \Delta T_{cond} \quad (3)$$

While the mean temperature can easily be calculated, temperature drops which are depending on the actual heat-exchanger design are usually considered to be constant for the entire operating range. The model described here estimates actual temperature drop by assuming that this value is proportional to the power transmission at the actual operating point:

$$\Delta T_{cond} = \Delta T_{cond,rated} \cdot \frac{P_{heat,actual}}{P_{heat,rated}} \quad (4)$$

While shown here for the hot (sink) side of the heat-pump, the equations above can be applied to the source side by just using corresponding values. Finally the Carnot-COP at the actual operating point can be evaluated:

$$COP_{carnot} = \frac{T_{sink,refrigerant}}{T_{sink,refrigerant} - T_{source,refrigerant}} \quad (5)$$

Looking at the formulas above, it can be seen that another iteration process –at the cold side– is required to find outlet temperature and cooling capacity, which is implicitly required in the Carnot COP calculation.

3 IMPLEMENTATION

The model has been implemented in Matlab-Simulink® environment and follows straightly the description from section 2. At each simulation time-step a few iterations are required (which could be defused by limiting the number of iterations as they add somewhat to the time-consumption of the whole simulation).
The following, essential parameters are required for the full description of a heat-pump in the described model:

- Heating capacity (P_{heat}) and electrical power consumption (P_{el}) according to EN 14511 at each rated point, evaluated for two different outlet temperatures (preferentially near the operation limits, e.g. at W35 and W 55).
- Temperature drop across the evaporator ($\Delta T_{evap,rated}$) and condenser ($\Delta T_{cond,rated}$) at specified source/sink temperatures
- Flow rates at sink ($\dot{m}_{sink}$) and source ($\dot{m}_{source}$) side according to EN 14511 rating conditions

A thermal capacity of the exchangers has been implemented to account for dynamic effects. Further, losses to the ambient are considered by a heat loss coefficient U_A. Both values are required, but can be set to zero for reasons of simplicity:

- Heat capacity of condenser (C_{cond}) and evaporator (C_{evap})
- Heat loss coefficient to ambient (U_A)

Input values that are required during runtime of the simulation are:

- Actual massflow at sink side
- Actual inlet temperature at sink side
- Actual massflow at source side
- Actual inlet temperature at sink side
- Ambient temperature at the place where the unit is installed

Figure 1 below shows simulation results of the air-to-water heat-pump that has been used for validation of the model. The parameter $\Delta T_{cond,rated}$ was chosen to be 5 ºC (water sink) at A7/W35, while for the evaporator $\Delta T_{evap,rated}$ was assumed to be 10 ºC (air source) at A7/W35. While the simulation has been executed for different massflow rates on the sink side, all other input values except inlet temperature were kept constant. The figure shows the step response for a change of the inlet temperature (return from the heating system) fed to the heat-pump at t = 7200 s. As can be seen, COP and outlet temperature are both depending on actual massflow, as is true for ‘real’ heat-pumps. The depicted step in outlet temperature also shows the thermal inertia simulated by a thermal capacity of some tens of kJ/K for both, evaporator and condenser.

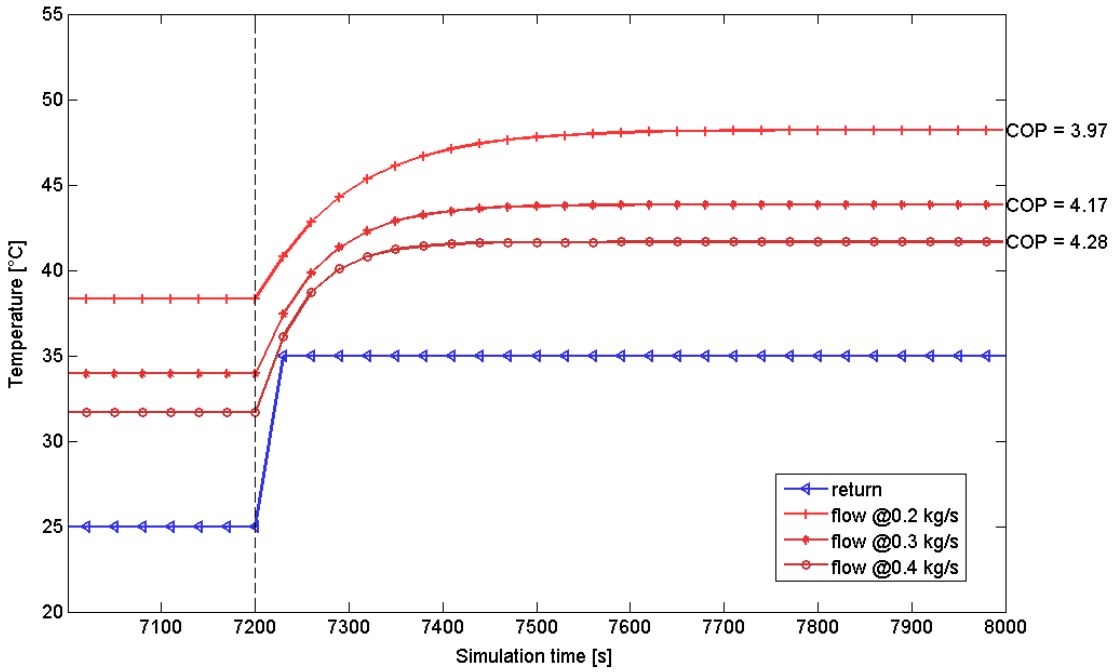


Figure 1: Outlet temperature and COP for a given step of the inlet temperature (return). Dependency on massflow at A12 condition.

4 VALIDATION

For the validation process, EN 14511:2013 and EN 14825:2013 measurement data of an air-to-water heat-pump has been used (Table 2 and Table 3). Again, $\Delta T_{cond,rated}$ was chosen to be 5 ºC while $\Delta T_{evap,rated}$ = 10 ºC, both at A7/W35 rating conditions. Massflow at each measured operating point is known and has been set accordingly in the simulation.

Table 2: Comparison of measured and simulated performance data for average climate, low temperature application acc. EN 14825:2013 (0.5 kg/s massflow)

quantity	origin of data	unit	rating condition			
			A-7	A2	A7	A12
inlet temp.	measurement/ simulation	ºC	30.6	27.8	25.7	23.5
outlet temp.	measurement	ºC	34.0	32.1	30.5	28.8
outlet temp.	simulation	ºC	34.1	32.1	30.6	28.9
heating capacity	measurement	kW	7.04	8.98	10.03	11.16
heating capacity	simulation	kW	7.34	9.08	10.26	11.24
el. Power	measurement	kW	2.35	2.28	2.11	2.01
el. Power	simulation	kW	2.36	2.29	2.09	1.98
COP	deviation from measurement	%	+3.7	+0.8	+3.6	+2.2

Table 3: Comparison of measured and simulated performance data for average climate, high temperature application acc. EN 14825:2013 (0.3 kg/s massflow)

quantity	origin of data	unit	rating condition			
			A-7	A2	A7	A12
inlet temp.	measurement/ simulation	ºC	46.1	38.2	33.5	29.0
outlet temp.	measurement	ºC	52	45.4	41.7	37.9
outlet temp.	simulation	ºC	51.8	45,3	41.6	37.9
heating capacity	measurement	kW	7.24	8.99	10.3	11.25
heating capacity	simulation	kW	7.09	8.90	10.14	11.15
el. Power	measurement	kW	3.08	2.78	2.52	2.35
el. Power	simulation	kW	3.06	2.83	2.56	2.36
COP	deviation from measurement	%	-1.3	-2.9	-3.2	-1.5

The maximum deviation of the COP is below 4 %. Thus, the simulation shows good agreement with measured values, especially when keeping in mind that measurement uncertainty according to EN 14511 is 6 % for COP and component tolerances may even be higher. The inclusion of power dependent temperature drop leads to somewhat better results at higher flow temperatures, but does show nearly no difference at low temperatures. Air-temperature drop at evaporator was assumed to be at a constant level of 5 K.

4 IMPACT ON YEARLY SIMULATIONS

The following section gives some numbers of how the described model influences simulated yearly energy consumption of a small single family house located in Bern / Switzerland. The building model has been described as 'SFH45' in Subtask C of IEA SHC Task 44 / HPP Annex 38 (Dott, 2013). Modelling was done in Matlab-Simulink® using the CARNOT blockset which has already implemented an empirical heat-pump model (Schwamberger, 1991). The newly developed model can be regarded as an enhancement of the one implemented in CARNOT. The simulations carried out in in this section shall show if the semi-empirical model presented here shows any advantage over the model implemented in CARNOT. The simulated air-to-water heat-pump is the type used for validation purposes (see section 3). Setpoint for sanitary water is 50 ºC in the 300l storage tank, which led to tap temperatures over 45 ºC for more than 80 % of the drawing time. The room temperature is controlled by a simple on/off mechanism which activates (deactivates) the heat-pump and heating-distribution system when the temperature falls below (rises above) 20 ºC. Heat transmission to the rooms is accomplished by means of a floor emission system without an additional heat storage. Flow temperatures of the heating system are always below 34 ºC. This control strategy holds room temperatures very tight, mostly in a range between 19 ºC and 22 ºC (over 96 % of the time, evaluated during winter season). This setup has been chosen as a direct comparison between the two heat-pump models show that the CARNOT implemented version exhibits slightly lower COP values at lower outlet temperatures. Table 4 shows the comparative results of the newly developed model presented here and the standard CARNOT library model in a yearly simulation. The evaluation of the generator SPF (SPF_{gen}) includes just the heat-pump itself, without any additional circulation pumps or heat losses.

The same applies to the electricity consumption E_{HP}. For both numbers, standby electricity consumption has been neglected.

Table 4: Comparison of measured and simulated performance data for average climate, low temperature application acc. EN 14825:2013 (0.5 kg/s massflow)

	SPF_{gen} [-]	E_{HP} [kWh]
CARNOT library model	3.70	2372
new model	3.69	2379

As deviations between the two models usually stay below 3 % when comparing single COP values at steady state, it is no surprise that yearly simulations show merely any difference in outcome. The model described here gives some more headroom to fit the model to actual measured data, but finally shows no advantages in the application presented here.

5 REFERENCES

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