Household Dishwasher with a Monovalent Heat Pump System

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Abstract

Appliances like dryers, washing machines or dishwashers add to a household's electricity demand considerably. To reduce their consumption of electricity, these appliances can be equipped with heat pumps for providing the necessary heat. In this paper the integration of a monovalent heat pump system into a dishwasher is theoretically analyzed and the simulation results are compared with those from experimental investigations from a prototype and from dishwashers in which technical measures have been taken to reduce energy demand. To calculate the transient heating process, energy balances as well as the heat transfer kinetics were set up in a mathematical-physical model together with the operating characteristics of a specific heat pump. The simulation program allows forecasting of the non-stationary operational behavior of the washing process and heat pump with a high degree of accuracy. The resulting concept places the evaporator in a latent heat storage filled with water. A coaxial heat exchanger serves as the condenser, a rotary compressor provides compression and a capillary tube is used for expansion. The simulated electricity need for the standardized process is 489 Wh with a washing process time of 128 minutes. Experiments with the dishwasher prototype have shown similar results, where the electrical energy need is 503 Wh and the washing process lasts 126 minutes. Dishwashers with electric resistance heating consume around 1000 Wh and dishwashers with a bivalent heat pump system 910 Wh for a comparable process. Through the use of a monovalent heat pump system, the electrical energy need of dishwashers can be reduced by up to 50%.

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Keywords: appliance, household dishwasher, white goods, monovalent heat pump system, energy efficiency, simulation, prototype

1. Introduction

Various home appliances have heat pumps and refrigeration systems installed in them to provide energy for heating and cooling. Refrigerators and dryers are two classic applications. To power electric home appliances in Switzerland, for example, a total of 5.86 TWh [1] was needed in 2014, a value 12.3% higher than in 2000. Home appliances account for about 10% of the total energy needs for private households in Switzerland (59.0 TWh). The use of heat pumps in home appliances can reduce electrical energy needs markedly.

This paper presents the development of a household dishwasher based on a monovalent heat pump system that requires up to 50% less electrical energy than conventional dishwashers that use standard electrical

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resistance heating. The development involved the creation of a simulation program and building of a prototype to check and validate the pre-calculated characteristic values.

2. State of the art

The task of a dishwasher is to clean dirty dishes. According to the "Sinner circle", the following four factors are needed for achieving satisfactory cleaning results: i. chemistry (in the form of detergents), ii. mechanics, iii. temperature and iv. time [2]. Today's dishwashers featuring electrical resistance heating require on the order of 750-1000 Wh of electricity for one dishwashing process. The optimization trend in recent years has been to reduce the temperature and with it, the energy required for heating the dishwasher, and to extend the length of the dishwasher operation time. A major reason for the longer duration is that the cleaning water dissolves grease more slowly at a lower temperature. Energy needs can be further reduced through additional technical measures. One possibility is to integrate an open sorption process (zeolite) that adsorbs water steam during the drying process and preheats the air used for drying [3]. In the next dishwashing cycle the water is desorbed and the released energy is partially recovered by the hot desorption air in the dishwasher. The total electrical energy needed by the dishwasher for this process is about 800 Wh. That represents a reduction of about 24% compared to the same dishwasher with electrical resistance heating. Another example is the installation of a bivalent heat pump system for heating the water, which is a combination of a heat pump and electrical resistance heating [4]. The heat source used for the heat pump is a latent heat storage with water, which is attached to one side of the housing. A minimum of 910 Wh of electrical energy is required for the standard process [5]. That corresponds to a reduction of 24% compared to a structurally similar dishwasher without a heat pump [4].

3. Requirements and specifications

3.1. General requirements

Different requirements and specifications apply to household dishwashers with a monovalent heat pump system and "open drying process" (*i.e.* the door is opened during the drying process). In conventional dishwashers the door is closed during the drying process (so called "closed drying process"). The following list provides additional general requirements:

- Dishwashing program with a duration of 130 min (excluding drying process)
- Compliance with the standard test pursuant to [5]: The period of time between the start of two dishwashing programs is 24 h.
- Installation limited to a "standard Euro niche" covering an area of 60 cm by 60 cm.
- Common, competitive costs of manufacturing for the additional components

3.2. Concept

Fig. 1 illustrates the concept behind the household dishwasher with a monovalent heat pump system. As the figure shows, a latent heat storage filled with water is employed as the heat source for the heat pump. This storage has high energy storage density due to the phase change between liquid water and ice. Using this approach, the heat pump can provide the entire required process heat monovalently.

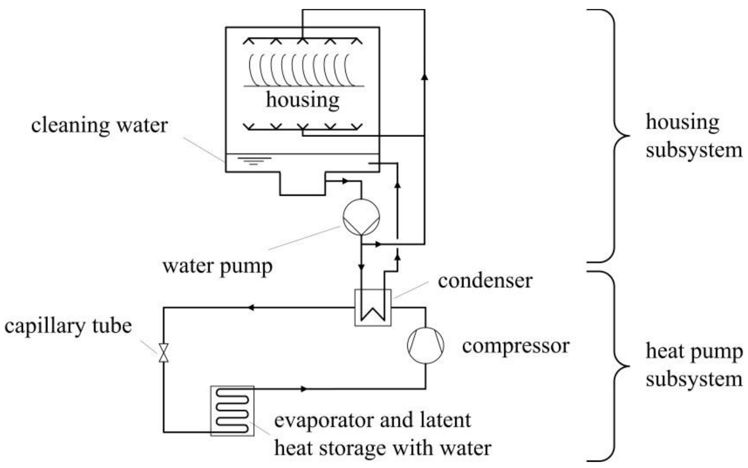

Fig. 1. Concept behind household dishwasher with a monovalent heat pump system

The water pump draws the cleaning water out of the sump of the dishwasher into the water circuit where by it is split into two different lines. In the first line, the cleaning water is pumped through the condenser and heated by the condensing refrigerant before flowing back to the sump. In the second line, the water is pumped to the spray arms in the housing and distributed by means of washing jets, so the dirty dishes are cleaned and heated up.

At the start of the dishwashing process, the evaporator coil in the latent heat storage is immersed in liquid water. As the evaporating refrigerant extracts energy from the water, it is cooled and then partially freezes. Next, the evaporated refrigerant passes through a rotary compressor before going to the condenser. Following the condenser, the pressure level of the refrigerant is lowered to the evaporation pressure using a capillary tube.

The entire heat pump unit with latent heat storage (*i.e.* heat pump subsystem) is placed underneath the housing. With this arrangement, the dishwasher can still be built into a standard Euro niche. Moreover, the latent heat storage is thermally separated from the housing and no heat is exchanged between the housing subsystem and the heat pump subsystem.

3.3. Process requirements

Fig. 2 presents the process requirements for the dishwasher featuring the monovalent heat pump system with an open drying process. For the various cleaning steps, it is necessary to identify the curve of the cleaning water temperature over time, which approximately corresponds to the temperature of the dishes, the housing, and the water masses involved in the different dishwashing baths. At the beginning of the dishwashing process, 4.4 kg of water are filled into the sump and the heat pump warms the housing subsystem (*i.e.* housing, dishes, and washing water, see Fig. 1) from 20°C to 50°C. Then the water is circulated and the dishes are cleaned. The temperature in the dishwasher steadily decreases during this period because of heat lost through the housing and the transient heat conduction in the dishes. When washing is completed, the dirty water is pumped out at a temperature of about 42°C, and the sump is filled with 3.4 kg of fresh water at a temperature of 15°C. The mixing temperature is approximately 36°C. During this intermediate washing step, the dishes are cleaned again to remove any remaining residues. At the end of this cleaning step, the cleaning water is again pumped out.

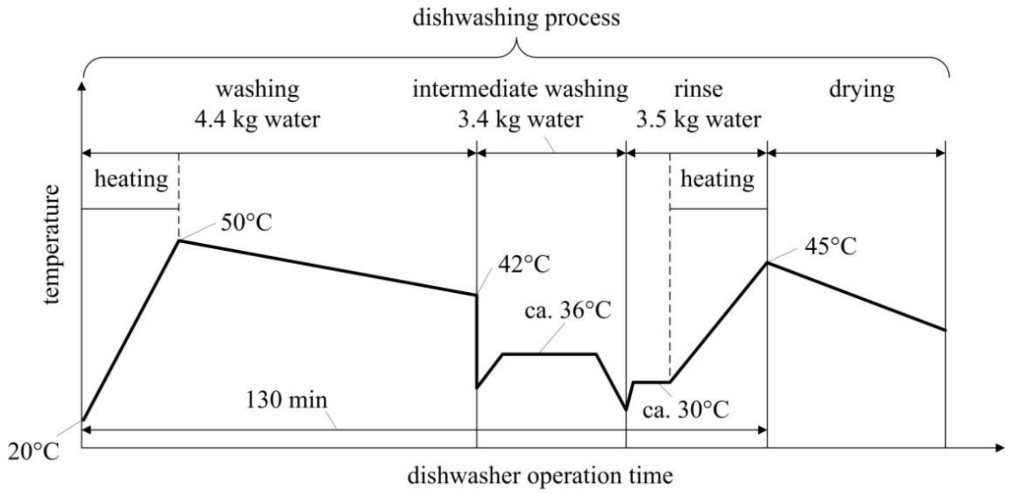

Fig. 2. Process requirements of a dishwasher with a monovalent heat pump system and open drying process (not to scale)

In the final rinse cleaning step, 3.5 kg of fresh water is added and then the housing subsystem is heated by the heat pump to 45°C for drying. The mixing temperature is around 30°C. At the end of this cleaning step, the water is pumped out. During the subsequent drying process, the water remaining on the dishes evaporates and part of the water condenses on the cooler housing walls. The energy necessary for evaporation comes from the dishes, which cools them. After a defined wait period, the dishwasher door opens automatically and room air flows in to help complete the drying of the dishes. The rinse temperature is lower for the open drying process than for closed drying process (45°C instead of 65°C) resulting in a smaller temperature lift and thus greater efficiency.

3.4. Regeneration of the latent heat storage

After the end of the dishwashing process, the ice in the latent heat storage must be thawed and heated up for the next dishwashing process. This regeneration process is accomplished using the surrounding room air and takes around 20 h assuming standard conditions (22°C, 55% RH) [5]. The air cools off in the process due to the melting of ice. However, as the warm dishwasher itself cools to room temperature, it releases thermal energy into the air and warms the surrounding room air. Owing to the supplied electrical compressor energy, the air is supplied with more energy than is extracted by regeneration of the latent heat storage resulting in a net energy input to the room. However, compared to a conventional dishwasher the internal heat load of the room is reduced.

4. Modeling

The basis of the completed simulations is the model of transient behavior for the heat pump process and the dishwashing process itself. Different heat pump configurations were examined and, hence, the costs for prototype development and testing were markedly reduced. Details of the models are presented in the next sections.

4.1. Heat pump process

Fig. 3 presents the heat pump process upon which modeling is based and the associated state values 1-4 in the log *p,h* diagram. The variables used in the following sections refer to this figure. The model assumes that no pressure losses occur in the evaporator and the condenser. The thermo-physical properties for the refrigerant and the cleaning water are taken from [6].

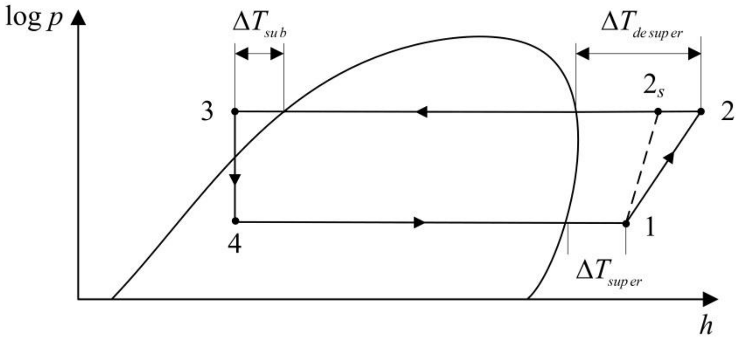

Fig. 3. Schematic heat pump process in the log *p,h* diagram

4.2. Heat pump components

4.2.1. Compressor

The type of compressor used is a positive-displacement constant-speed rotary compressor. The volume flow rate $\dot{V}_{comp}$ that is drawn in depends on the displacement V_{disp}, the rotational speed n_{comp} and the volumetric efficiency λ of the compressor:

$$\dot{V}_{comp} = V_{disp}\, n_{comp}\, \lambda \tag{1}$$

The mass flow rate of refrigerant $\dot{m}_r$ is obtained by multiplying the volume flow rate $\dot{V}_{comp}$ by the refrigerant density ρ_1, which is related to the temperature and pressure at the entrance of the compressor. To calculate the inner compressor power P_i, one assumes adiabatic compression with dissipation, where the refrigerant is considered a perfect gas.

$$P_i = \frac{1}{\eta_s}\dot{m}_r\left(h_{2s} - h_1\right) = \frac{1}{\eta_s}\dot{m}_r\, c_{p,g,1}\, T_1\left(\pi^{\frac{\chi-1}{\chi}} - 1\right); \quad \pi = \frac{p_C}{p_E}$$

$$c_{p,g,1} = \text{ specific heat capacity gaseous; } \quad \pi = \text{ pressure ratio} \tag{2}$$

$$p_C = \text{ condensation pressure; } \chi = \text{ isentropic exponent}$$

The electrical compressor power $P_{e,comp}$ can then be calculated by dividing the above figure by the electrical-mechanical efficiency $\eta_{e,m}$, which is assumed to be constant.

The volumetric efficiency λ and the isentropic efficiency η_s describe the operating characteristics of the compressor, whereby both variables depend on the pressure ratio π. For calculation purposes, regressions are made with the following structure, whereby the coefficients are determined experimentally:

$$\lambda = a_\lambda\, \pi + b_\lambda; \qquad \eta_s = a_s\, \pi + b_s \tag{3}$$

4.2.2. Condenser

The condenser used is a horizontally arranged tube-in-tube heat exchanger, whereby the outer tube is made from flexible plastic tubing and the inner tube is made from stainless steel. The cleaning water (index "w") flows in the annular gap and the refrigerant (index "r") inside the inner tube. The materials were selected due to the corrosive nature (acidic or basic) of the cleaning water. The following applies for the heat flow $\dot{Q}_C$ transferred in the condenser:

$$\dot{Q}_C = \dot{m}_r\left(h_2 - h_3\right) = \dot{m}_w\, c_{p,w}\left(T_{C,w,out} - T_{C,w,in}\right) = k_C A_C \Delta T_{m,C} \tag{4}$$

To calculate the logarithmic mean temperature difference $\Delta T_{m,C}$, it is assumed that the temperature of the refrigerant remains constant during heat transfer and corresponds to the condensation temperature T_C (*i.e.*

desuperheating and subcooling are neglected in the model). The modeling is conservative with respect to the determination of the electrical energy needs because this approach leads to higher condensation temperatures than in the real system. For the calculation the temperatures of the cleaning water on input and output ($T_{C,w,in}$ and $T_{C,w,out}$) are needed. Thus, the following applies to the mean temperature difference $\Delta T_{m,C}$:

$$\Delta T_{m,C} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln\left(\Delta T_{max} / \Delta T_{min}\right)}; \quad \Delta T_{max} = T_C - T_{C,w,in}; \quad \Delta T_{min} = T_C - T_{C,w,out} \tag{5}$$

The following applies for the overall heat transfer coefficient k_C in the equation (4) assuming one tube with the reference diameter d_{out} :

$$k_C = \frac{1}{\frac{1}{\bar{\alpha}_r}\frac{d_{out}}{d_{in}} + \frac{d_{out}}{2\lambda_{tube}}\ln\frac{d_{out}}{d_{in}} + \frac{1}{\alpha_w}}$$

$d_{in} =$ inner diameter of the inner tube; $d_{out} =$ outer diameter of the inner tube; (6)

$\lambda_{tube} =$ thermal conductivity of stainless steel tube

The average heat transfer coefficient α_w on the water side is calculated using the method of Petukhov *et al.* [6]. The heat transfer coefficient $\bar{\alpha}_r$ of the refrigerant is averaged for the area of the three sections desuperheating, condensation, and subcooling (compare equation (7)). The individual heat transfer coefficients are calculated for the sections desuperheating and subcooling using the correlation of Gnielinski *et al.* [6] and for the section condensation using the correlation of Haraguchi *et al.* [7].

$$\bar{\alpha}_r = \frac{1}{A_C}\left(A_{desuper}\,\alpha_{desuper} + A_{cond}\,\alpha_{cond} + A_{sub}\,\alpha_{sub}\right) \tag{7}$$

4.2.3. Evaporator

A copper tube coil immersed in the latent heat storage serves as the evaporator. In the transient calculation, it is assumed that the water in the latent heat storage cools and freezes uniformly. The heat flow $\dot{Q}_E$ being transferred in the evaporator can be calculated as follows:

$$\dot{Q}_E = \dot{m}_r\left(h_1 - h_4\right) = k_E A_E \Delta T_{m,E} \tag{8}$$

The mean temperature difference $\Delta T_{m,E}$ is calculated by taking the difference between the temperature of the water in the latent heat storage $T_{w,lhs}$ and the evaporation temperature T_E . The suction gas superheating ΔT_{super} is not taken into account. The average overall heat transfer coefficient k_E is assumed to be constant and was previously determined experimentally (see section 5.1).

4.2.4. Expansion device

In the heat pump that was built, capillary tubing was used as the expansion device for cost reasons. In the simulation, a thermostatic expansion valve is used for keeping suction gas superheating ΔT_{super} at a constant level (assumption: adiabatic expansion). The temperature at the evaporator outlet T_1 corresponds to the temperature at the compressor inlet and can be calculated using suction gas superheating.

4.3. Transient system equations

4.3.1. Heating of the housing subsystem

The energy change of the housing subsystem $\mathrm{d}E_{hss}$ can be attributed to the condenser heat flow $\dot{Q}_C$ that heats the cleaning water in the condenser. The following applies:

$$\frac{\mathrm{d}E_{hss}}{\mathrm{d}t} = \frac{\left(C_{dw} + m_w\, c_{p,w}\right)\mathrm{d}T_{hss}}{\mathrm{d}t} = \dot{Q}_C\left(t\right) \tag{9}$$

In equation (9), C_{dw} contains the thermal capacity of the dishes and housing in addition to the average heat lost through the housing. This thermal capacity is determined experimentally and is assumed to be constant. Also needed are the water masses m_w of the respective dishwashing bath in the dishwashing process and the specific thermal capacity $c_{p,w}$ of water.

For a time increment Δt, a constant condenser heat flow $\dot{Q}_C^i$ is assumed. This allows the temperature of the housing subsystem T_{hss}^{i+1} to be calculated at the time $i+1$:

$$T_{hss}^{i+1} = T_{hss}^{i} + \frac{\dot{Q}_C^i}{C_{dw} + m_w\, c_{p,w}}\Delta t \tag{10}$$

4.3.2. Cooling of the water and formation of ice in the latent heat storage

The cooling of the water and the formation of ice in the latent heat storage can be attributed to the heat flow $\dot{Q}_E$ discharged by the evaporator. As described in section 4.2.3, a homogeneous distribution of water temperature is assumed. The following applies to the energy change $\mathrm{d}E_{w,lhs}$ of the water:

$$\frac{\mathrm{d}E_{w,lhs}}{\mathrm{d}t} = -\dot{Q}_E\left(t\right); \quad \text{reformulated:} \quad \mathrm{d}E_{w,lhs} = -\dot{Q}_E\left(t\right)\mathrm{d}t \tag{11}$$

The energy change $\mathrm{d}E_{w,lhs}$ during the cooling of the latent heat storage can be converted to a temperature change $\mathrm{d}T_{w,lhs}$:

$$\mathrm{d}E_{w,lhs} = m_{w,lhs}\, c_{p,w}\, \mathrm{d}T_{w,lhs} \tag{12}$$

$m_{w,lhs}$ corresponds to the total water mass. The evaporator heat flow $\dot{Q}_E^i$ is constant for a small time increment Δt. This allows the water temperature $T_{w,lhs}^{i+1}$ to be calculated at the time $i+1$:

$$T_{w,lhs}^{i+1} = T_{w,lhs}^{i} - \frac{\dot{Q}_E^i}{m_{w,lhs}\, c_{p,w}}\Delta t \tag{13}$$

As soon as the water reaches the freezing point, the phase change begins from a liquid to a solid state. The temperature remains constant during this process. The energy change $\mathrm{d}E_{w,lhs}$ during freezing can be expressed as an increase in the mass $\mathrm{d}m_{ice}$ of the ice:

$$\mathrm{d}E_{w,lhs} = -\mathrm{d}m_{ice}\, \Delta h_{fus} \tag{14}$$

whereby Δh_{fus} corresponds to the enthalpy of fusion for pure water. For an evaporator heat flow $\dot{Q}_E^i$ that is constant during a time increment Δt, the ice formed m_{ice}^{i+1} in the time increment $i+1$ can now be determined:

$$m_{ice}^{i+1} = m_{ice}^{i} + \Delta m_{ice}^{i+1} = m_{ice}^{i} - \frac{\left(-\dot{Q}_E^i\right)}{\Delta h_{fus}}\Delta t = m_{ice}^{i} + \frac{\dot{Q}_E^i}{\Delta h_{fus}}\Delta t \tag{15}$$

4.3.3. Electrical energy needs

The electrical energy needs $E_{e,tot}$ of a dishwasher with a monovalent heat pump system comprise the following two parts:

$$E_{e,tot} = \sum_{i=1}^{i_{max}}\left(P_{e,comp}^{i} + P_{e,au}^{i}\right)\Delta t \tag{16}$$

The electrical power $P_{e,au}$ of the auxiliary units (*i.e.* water pump, see Fig. 1) is assumed to be constant across the entire dishwashing process.

5. Simulations

This section presents the boundary conditions and results of simulations for a standard process [5]. The transient calculations were conducted using a simulation program developed in MATLAB®. This program was used for conducting a parameter study and process optimization for the requirements and specifications from section 3.

5.1. Parameters and boundary conditions

The simulation for the heat pump components was based on the following parameters in Table 1.

Table 1. Details and parameters of the components

rotary compressor	
manufacturer	Rechi Precision CO. Ltd.
type	39E113A
refrigerant	R134a
displacement	11.4 cm^3/rev
rotational speed	2900 rpm
electrical-mechanical efficiency	92%

condenser	
length	2.0 m
inner diameter of outer tube	20.0 mm
inner diameter of inner tube	8.5 mm
wall thickness of inner tube	0.51 mm
thermal conductivity of inner tube	20 W/(m K)
water volume flow rate	10.0 ltr/min

latent heat storage and evaporator	
water mass in latent heat storage	5.5 kg
length of copper tube	10.0 m
inner diameter of copper tube	10.0 mm
wall thickness of copper tube	1.0 mm
overall heat transfer coefficient	150 W/(m^2 K)

The following list contains additional boundary conditions for the simulation:

- Thermal capacity C_{dw} = 44.9 kJ/K
- Suction gas superheating ΔT_{super} = 5 K
- Subcooling ΔT_{sub} = 1 K
- Electrical power input $P_{e,au}$ for auxiliary units = 60 W
- Enthalpy of fusion Δh_{fus} water to ice = 333 kJ/kg
- 13 Standard place settings [5]
- Necessary temperatures and masses for dishwashing baths as indicated in Fig. 2
- Time increment Δt = 5 s

5.2. Results

5.2.1. Temperature curves

Fig. 4 presents the simulation results for the condensation, evaporation, cleaning water and the water in the latent heat storage temperatures over time. At the beginning of washing, the housing subsystem is heated from 20°C to 50°C in 31.0 min, whereby the condensation temperature rises steadily from 41.6°C to 62.5°C. Afterwards the dishwasher slowly cools down to 42°C. Next, the washing water is pumped out and then the sump is filled with the intermediate washing water, whereby a mixing temperature of 35.6°C ensues. The water is then pumped out again after 10 min resulting in a temperature of 30.5°C due to the mixing in of the rinse water. During rinsing, the housing subsystem heats up to 45°C in 16.4 min and the condensation temperature rises from 43.8°C to 57.6°C in the meantime. The entire process except the drying time results in a dishwashing process with a duration of 127.5 min.

The water in the latent heat storage is cooled down initially within 10.2 min from 20°C to 0°C and the evaporation temperature decreases from 2.0°C to -9.8°C. Subsequently, the temperature of the water remains constant at the freezing point throughout the entire dishwashing process. A total of 3.35 kg of ice is formed (*i.e.* 5.5 kg of water in the latent heat storage does not freeze completely). After the freezing point is reached, the evaporation temperature rises from -9.8°C to -7.3°C up till the compressor shuts off. During rinsing, the evaporation temperature rises from -10.0°C to -8.0°C. The reasons for this rise are the increasing condensation temperature and the shape of the saturated liquid line in the log *p,h* diagram. With the condensation temperature rising and subcooling remaining constant, the steam quality of the refrigerant in the evaporator increases after expansion. If suction gas superheating is constant, a smaller evaporator heat flow results. This is reflected in a smaller temperature difference as shown in equation (8) and must lead to an increase in the evaporation temperature.

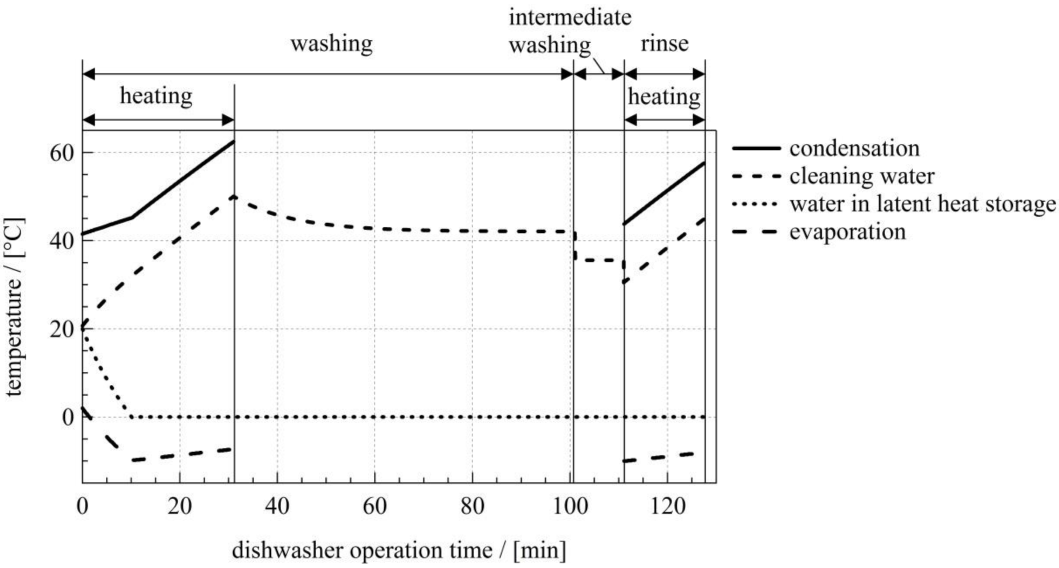

Fig. 4. Simulation results for the condensation, evaporation, cleaning water and water in the latent heat storage temperatures over time

5.2.2. Output curves

Fig. 5 shows the simulation results for the electrical compressor power as well as the heat flows of the evaporator and condenser over time.

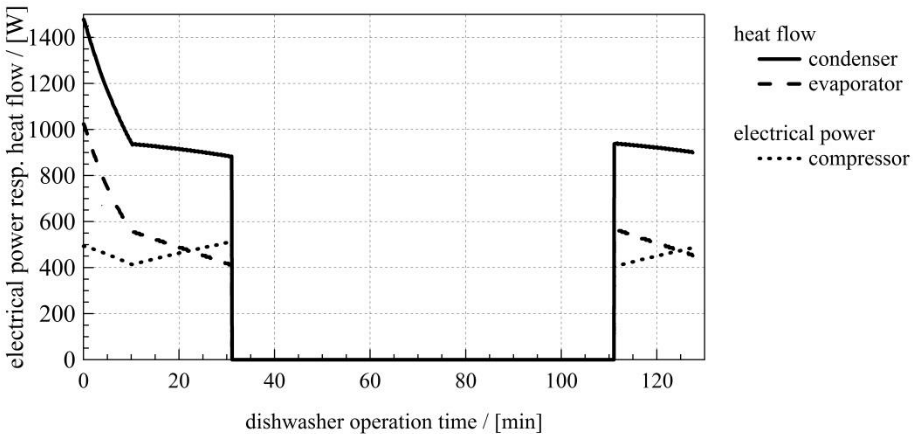

Fig. 5. Simulation results for electrical compressor power as well as the heat flows of the evaporator and condenser over time

Initially, the heat flows decline markedly until the freezing point of the water is reached after 10.2 min. The condenser heat flow declines from 1478 W to 935 W and the evaporator heat flow from 1024 W to 554 W. The electrical compressor power falls from 494 W to 414 W. The reason for the decline is the drop in evaporation temperature (see section 5.2.1) or in evaporation pressure, which leads to a lower refrigerant density in the suction line and to a smaller refrigerant mass flow being conveyed. This situation, in turn, results in diminishing heat flows and diminishing electrical compressor power despite the higher condensation temperature. During the remaining heat-up period for washing, the condenser heat flow falls from 935 W to 884 W and the evaporator heat flow from 554 W to 413 W. The electrical compressor power increases from 414 W to 512 W due to the ever larger pressure ratio. During rinsing, the condenser heat flow declines from 938 W to 900 W and the evaporator heat flow from 565 W to 452 W. The electrical compressor power rises at the same time from 405 W to 489 W.

5.2.3. Electrical energy needs

Table 2 shows the calculated electrical energy needs for a standard process. The table is divided into the different cleaning steps, and a distinction is made between the electrical energy needs for the compressor versus those for the auxiliary units.

Table 2. Calculated electrical energy needs for a standard process

	washing		**intermediate washing**	**rinse**	**dishwashing process**
	heating	no heating, only cleaning	no heating, only cleaning	heating, no cleaning	
compressor [Wh]	238.8	0	0	122.8	361.6
auxiliary units [Wh]	31.1	70.0	10.0	16.4	127.5
total [Wh]	269.9	70.0	10.0	139.2	**489.1**

The dishwasher with monovalent heat pump system needs 489.1 Wh of electrical energy for the entire dishwashing process, with 361.6 Wh (73.9%) of that total required for the operation of the compressor.

6. Validation of the simulation results

A prototype of a dishwasher with a monovalent heat pump system complying with the above specifications and design was built and used for experimental testing [8]. Two key characteristic values were determined. The first value was the average electrical energy need for the dishwashing process of 503 Wh. The second value was an average duration of the dishwashing process of 126.0 min (drying time not included).

A comparison of the simulation results and the values determined experimentally indicates the high accuracy of the simulation. There was a difference of 13.9 Wh (2.8%) in electrical energy needs and of 1.5 min (1.2%) in the duration of the dishwashing process.

7. Conclusion

Table 3 compares the reduction of electrical energy needs for the prototype to other types of dishwashers. The results show that the use of a monovalent heat pump system in a dishwasher with an open drying process can reduce the electrical energy needs by up to 50% compared to dishwashers with electrical resistance heating and a closed drying process. The reduction is 37.1% compared to dishwashers with an open adsorption system [3] and 44.7% compared to a dishwasher with a bivalent heat pump system [4]. Compared to structurally similar dishwashers (Adora SL from V-ZUG Ltd.) with electrical resistance heating and open drying process, the reduction is 32.9% A dishwasher with a monovalent heat pump system and open drying process has been available on the market since 2014.

Table 3. Comparison of the electrical energy needs of different dishwashers

dishwashers	source	electrical energy need [Wh]	energy reduction* [%]
standard dishwasher with electrical resistance heating and closed drying process (Adora N)	V-ZUG Ltd.	930	45.9
dishwasher with open adsorption system	[3]	800	37.1
dishwasher with bivalent heat pump system	[4]	910	44.7
standard dishwasher with electrical resistance heating and open drying process (Adora SL)	V-ZUG Ltd.	750	32.9
prototype dishwasher with monovalent heat pump system and open drying process (identical Adora SL)	experiment	503	0

* The energy reduction expresses how much lower in percent the energy needs are for the prototype compared to other dishwashers.

The characteristic key values were calculated in advance using highly precise transient simulations. These simulations allowed parameter studies to be conducted, leading to improved heat pump design and an overall system optimization. The time and costs for setup and for the experimental investigations of the prototype were thus kept to a minimum.

Acknowledgments

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