Modelling and Simulation of a Flash Tank Vapour Injection Heat Pump in Several Platforms

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ABSTRACT

In this paper, we model a relatively complex flash tank vapour injection heat pump using the physical modelling software EcosimPro. The heat pump was extensively tested in a previous study and a variety of system parameters were made available for validation. Simulation results are compared against measured data as well as against two other platforms: Dymola and Simulink. Startup and step-change transients have been simulated. The discrepancies between the different models are used to highlight some important aspects in conducting dynamic thermo-fluid simulations. In particular, although qualitatively representative trends may be reproduced, there is an unavoidable need to first obtain experimental measurements to improve the accuracy of the simulated parameters.

1. INTRODUCTION

Dynamic modelling of thermo-fluid systems is of considerable interest from the point of view of design of new plants, or of conducting studies for system optimization. In modelling such systems, engineers tend to favour physics-based equations for component modelling over empirical or semi-empirical equations due to their ability to capture behaviour over the entire domain of operation. Such equations, however, can require parameters that may not be available. Parameters such as detailed geometry, heat transfer coefficients or pressure drop factors, or system parameters such as detailed initial conditions may not be known. In such circumstances, it is instructive to recognise the limits in accuracy that can be obtained while simulating the dynamics of the system. Comparisons made where one set of models have been well-optimised against an experimental system can be especially illuminating. In this study, such a comparison has been made using a relatively complex flash tank vapour injection system.

Vapour injection heat pumps have the benefit of lower discharge temperatures compared to traditional four-component vapour compression systems. They are equipped with a two-stage compressor, where the suction fluid is compressed to an intermediate state and mixed with low temperature vapour taken from the outlet of the condenser. This leads to overall lower discharge temperatures and allows for an increased system capacity, especially at low ambient temperature conditions.

In this study, such a Flash Tank Vapour Injection (FTVI) heat pump has been modelled in the object-oriented physical modelling platform EcosimPro, and the results have been compared against two other modelling platforms: Simulink and Dymola. A system startup in heating mode has been simulated, and upon reaching steady state, the upper stage expansion valve opening is increased in successive steps.

The system was tested in transient operation under ASHRAE's High Temperature 2 test conditions for heating mode as shown in Table 1. The system was turned off from a steady-state operating condition (1 hour of operation) and was then left in off-mode for 24 minutes before being restarted. The system is allowed to reach steady state and run for 2200 seconds. Thereafter, the electronic expansion valve (EEV) opening was increased in discrete, successive steps and data were collected for system parameters under both the startup and step change conditions. Complete details of the testing of the unit may be found in the paper by Xu et al. (2011).

Table 1. ASHRAE High Temperature 2 test conditions.

	Temperature (°C)	Relative Humidity (%)
Indoor Unit (Condenser)	21.1	< 56.42
Outdoor Unit (Evaporator)	8.3	72.9

The operating principle of the cycle is as follows. The indoor unit serves as the condenser in heating mode. It is connected to the upper stage EEV. The flow from the EEV enters the flash tank which acts as a phase separator. The vapour port of the flash tank is connected to the compressor intermediate stage port. This intermediate line contains a 60W heater for superheating the vapour. The liquid port of the vessel is connected to the lower stage thermostatic expansion valve (TXV). The TXV feeds flow into the outdoor unit (evaporator) and controls its superheat. The flow from the evaporator enters the compressor and the cycle is repeated. It is possible to switch the system between heating and cooling modes through the energization of a reversing valve. A schematic of the cycle is given in Figure 1.

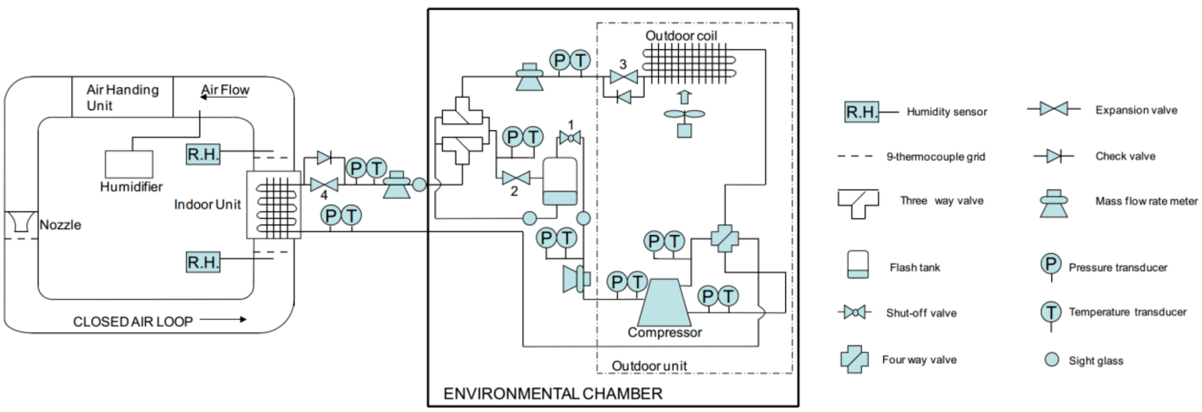

Figure 1. FTVI system schematic

2. MODEL IMPLEMENTATION

The mathematical modelling details have been omitted here since they have been covered in previous articles; the Dymola model is described in Qiao et al.(2015a), while the Simulink and EcosimPro model descriptions may be found in Bhanot et al. (2016, 2018). The Simulink and EcosimPro models were both derived from the Dymola model. Simulink and Dymola have been previously used (Bhanot et al., 2016; Qiao et al., 2015b) to conduct simulations on a residential heat pump unit, and the EcosimPro component library has been compared with the Dymola library in a previous study (Bhanot et al., 2019). The comparison was made for cooling and heating mode simulations of the same heat pump unit. The models showed very similar performance in that study, and thus, the differences observed here may be attributed to differences in the extent to which system-specific details are incorporated in the models rather than inherent shortcomings in any software.

In the Dymola model, several important system parameters have been modelled semi-empirically using curve fits of the system data. In particular, compressor mass flow rate polynomials were developed using measured flow rate, based on the pressure ratios and assuming a polytropic process. The polytropic coefficients were also derived using experimental data. Similarly, modelling the flow through the valves is challenging since the valve geometry information is rarely provided by manufacturers, and the flow coefficient must be fitted to match measured data. The availability of map-based polynomials can greatly facilitate the task of the modeller, although during dynamic operation, the components frequently operate outside of the design conditions for which the polynomials are valid.

Furthermore, the Dymola models benefit in other ways from having been developed in close collaboration with the experimental unit. It was discovered, for instance, that relatively moderate differences in the suction and discharge chamber geometry and heat transfer performance could alter the temperature plots significantly, but real parameters were not available. In addition, the performance of the TXV was also tuned to capture measured trends in Dymola.

Deriving the initial conditions is also a challenging task in dynamic modelling, and significantly influences system behaviour. Not only does it affect the predictions at steady-state operation, but also in the ability of the numerical solver to converge on to a solution efficiently. The charge distribution before system startup is an important consideration but is difficult to establish. Where the system is in single-phase, the refrigerant charge can be easily determined using pressure and temperature measurements. On the other hand, most of the refrigerant charge after shutoff resides in the outdoor unit (the evaporator), and exists in two-phase form, making it impossible to determine the refrigerant density. This charge also cannot be calculated by subtracting total system charge from that found in single phase components. This is because multiple components have two-phase fluid inside. Furthermore, simulations of heat pump systems are rarely able to account for the complete system charge anyway. Often, the models adopt a homogeneous assumption for two-phase fluid, leading to lower system charge. Even when two-phase flow is accounted for using void-fraction based correlations (as for the EcosimPro and Dymola models), the simulated system charge is still significantly lower than actual system charge. Thus, the model initial conditions need to be tuned to obtain the best match to measured data.

3. RESULTS AND DISCUSSION

For the results plots that follow, **solid lines (—)** represent the EcosimPro data, **dot-dash lines (· –)** are the Dymola results, **dashed lines (– –)** are the Simulink results while **dotted lines (··)** represent the measured data.

3.1 Startup Results

The startup dynamic trends are shown in Figure 2 and Figure 3. Just before startup, the majority of the charge resides in the outdoor unit (evaporator). This is seen in Figure 3(b) as the red line. Upon system startup, the compressor scroll-set starts extracting fluid from the low-pressure side. The TXV is closed at the beginning, seen as the purple line in Figure 3(c), due to a lack of outlet superheating. This imbalance between inlet and outlet flow causes a drop in suction pressure as well as suction density.

The suction chamber of the compressor is modelled as a homogeneous phase separator. Initially, due to the lack of superheating in the evaporator, two-phase fluid floods into the suction chamber. This flooding is over-predicted in the EcosimPro and Simulink model as seen in Figure 3(b), where the yellow line represents the Pipes and other components, including the suction chamber. Discrepancies in the flow rate of the scroll-set, combined with inaccurate initial refrigerant charge distribution may explain this differing behaviour. The suction superheat is also predicted lower in EcosimPro than the measured value. This directly leads to the lower suction temperature prediction, since the suction pressure itself is otherwise well predicted. The lower superheat, combined with the presence of saturated rather than superheated vapour in the suction chamber, are the cause for the lower discharge temperature observed in the EcosimPro simulations. The availability of detailed discharge chamber geometry might also have led to better predictions of the compressor outlet temperature. The EcosimPro model overpredicts the condenser subcooling by around 2 K. This is due to the lower temperature difference between the hot fluid from the compressor and the air. The suction superheat is well predicted, since the heating of the suction line is constant at 60W throughout the cycle due to the injection line heater.

The mass flow rate transients are shown only for the first 250 seconds, since beyond that the mass flow rates are effectively in steady state. It is seen that the EEV flow rate is predicted relatively accurately by EcosimPro, although the TXV starts to open much later and the opening is lower than Dymola. This trend is also seen in the suction superheat plot, Figure 2(c). Details of the flow coefficient through the TXV, if measured, would help alleviate this discrepancy.

Under steady state conditions, the EcosimPro simulations predict a lower charge in the evaporator. This leads to lower capacity predictions (both in EcosimPro and Simulink) compared to measured data. The Dymola charge distribution is expected to be closer to reality, although measured data for charge distribution cannot be obtained experimentally.

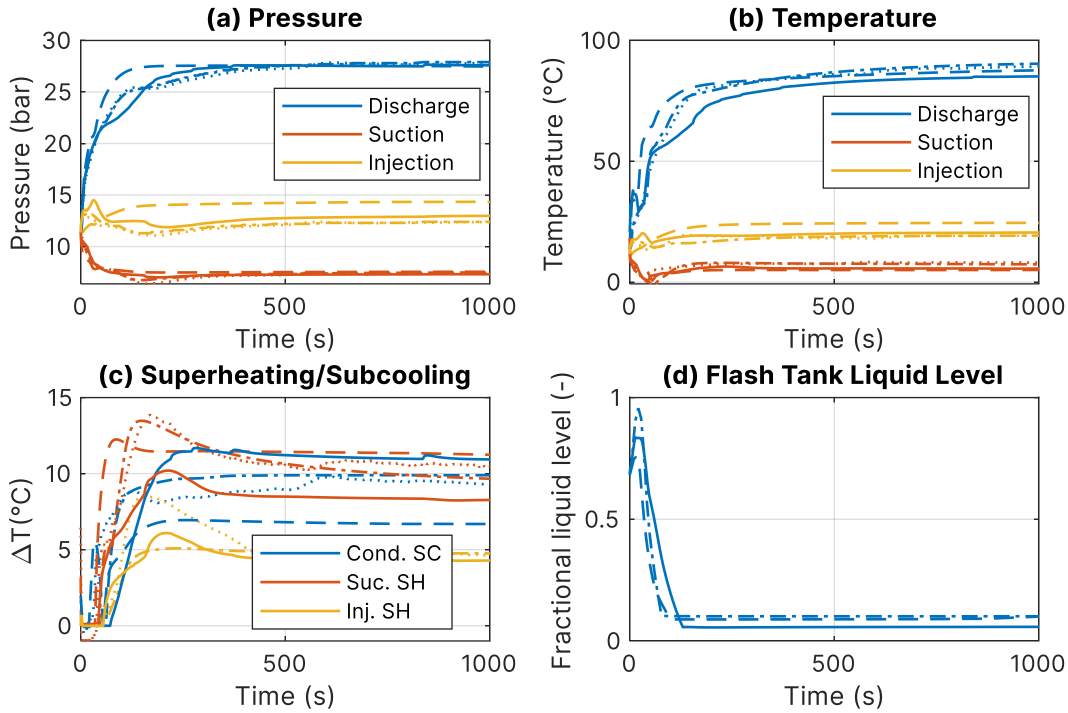

Figure 2. *Startup Transients 1.* ***Dotted lines*** *are measured data,* ***dot-dash lines*** *are Dymola,* ***dashed lines*** *are Simulink data and* ***solid lines*** *are EcosimPro data.*

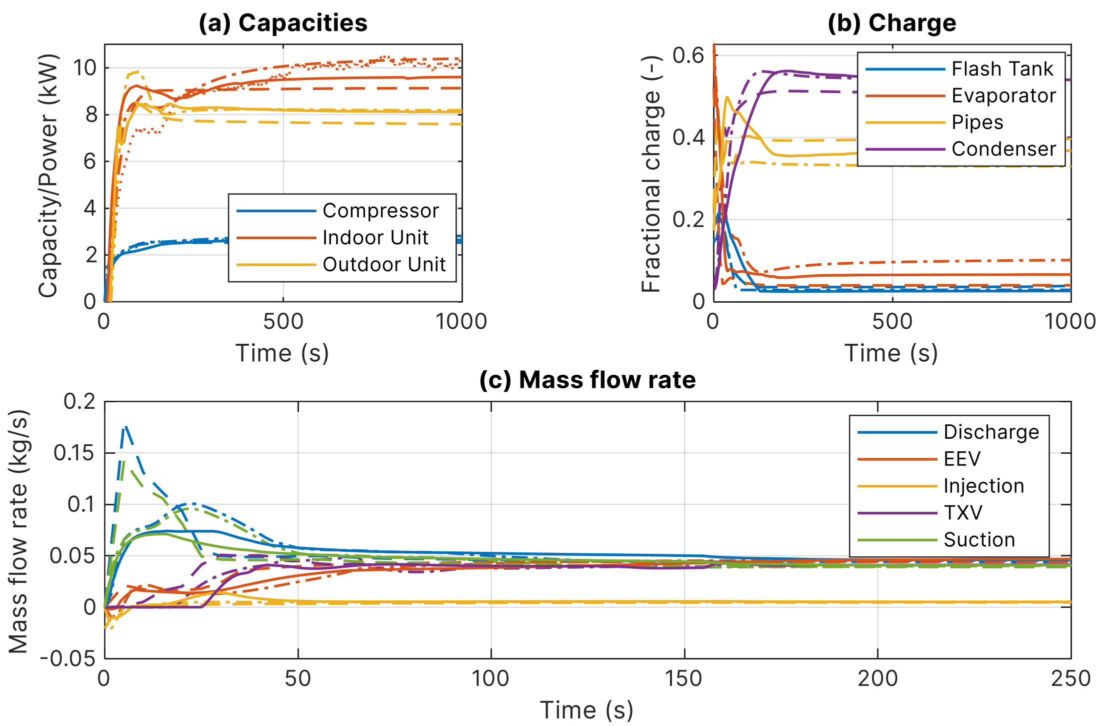

Figure 3*. Startup Transients 2.* ***Dotted lines*** *are measured data,* ***dot-dash lines*** *are Dymola,* ***dashed lines*** *are Simulink data and* ***solid lines*** *are EcosimPro data. Only 250 seconds shown for the mass flow rate*

The Simulink models consistently give worse predictions than both EcosimPro and Dymola. This can be explained through two factors. The first factor is the homogeneous two-phase flow assumption adopted in the Simulink models, as opposed to the slip-ratio based void fraction models in the other two platforms. Furthermore, the piping volumes were not accounted for thoroughly. Specifically, portions of the liquid line and the vapour line were present in both the indoor temperature and outdoor temperature regions in the real system, whereas in Simulink these pipes were treated as lumped. This assumption was adopted since the models were not robust enough to simulate with greater level of detail in acceptable timeframes. Nevertheless, these discrepancies highlight both that a greater scope exists to more accurately model heat pump transients by properly accounting for two phase flow, but also that, despite removing these shortcomings, the obtainment of very high accuracy might still elude the modeller.

The startup trends can be further clarified by visualising the system parameters on a schematic diagram. Figure 4 shows the status of the system at time t=30 seconds. It is seen that the flash tank is virtually full at this point, due to the addition of fluid from the condenser adding onto existing liquid. The evaporator outlet vapour quality is seen as still very low. The lack of suction superheat is visible in the pressure-enthalpy diagram. Furthermore, the mass flow rate differences between the inlet and outlet of the evaporator indicate, and likewise for the condenser inlet and outlet, are clearly visible. The low vapour quality in the compressor circle represents the vapour quality in the suction chamber. The suction chamber nearly fills with liquid in the initial part, leading to lower discharge temperatures seen in the EcosimPro model. The visualisation technique is inspired by the work of van Gerner and Braaksma (2016).

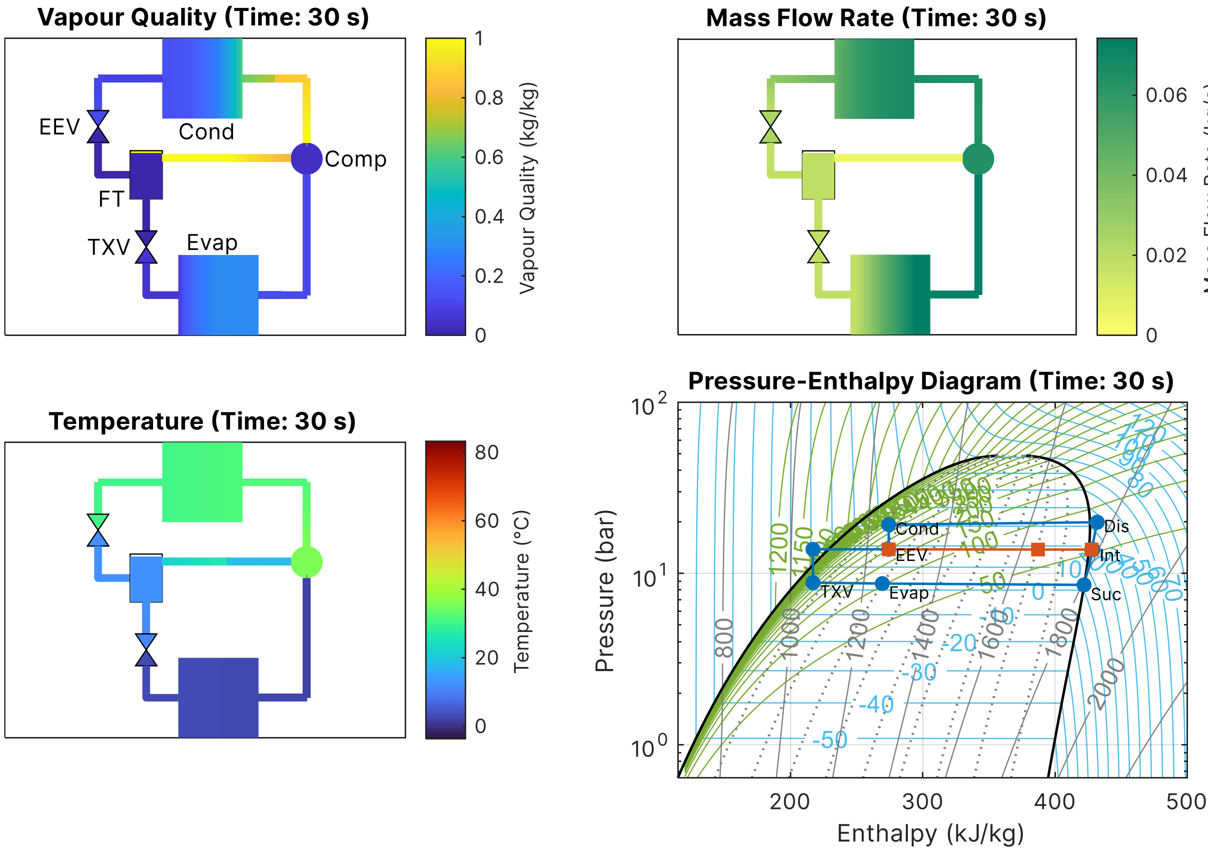

Figure 4. Fluid parameters at time t=30 seconds

3.2 EEV Step Change results

The electronic expansion valve is held fixed at 18% opening for the duration of the startup. Once the system has run for 2200 seconds, the EEV opening is successively changed as shown in Figure 5. Note that, for the Simulink models, the results were only available until 3500 seconds since the simulation had not been conducted beyond that. The results are shown in Figure 6.

It can be seen that the changes in the EEV opening do not have a significant impact on the suction pressure. This fact was reported in the experimental measurements, and is captured by all three simulation platforms. The intermediate line sees a steady pressure rise, whereas the discharge pressure drops continuously as we further open the EEV.

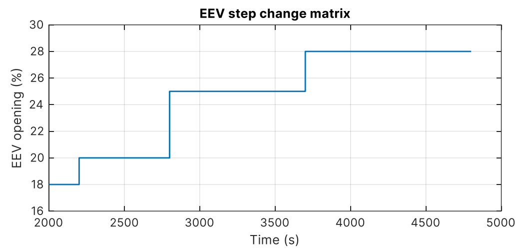

Figure 5. EEV position with time

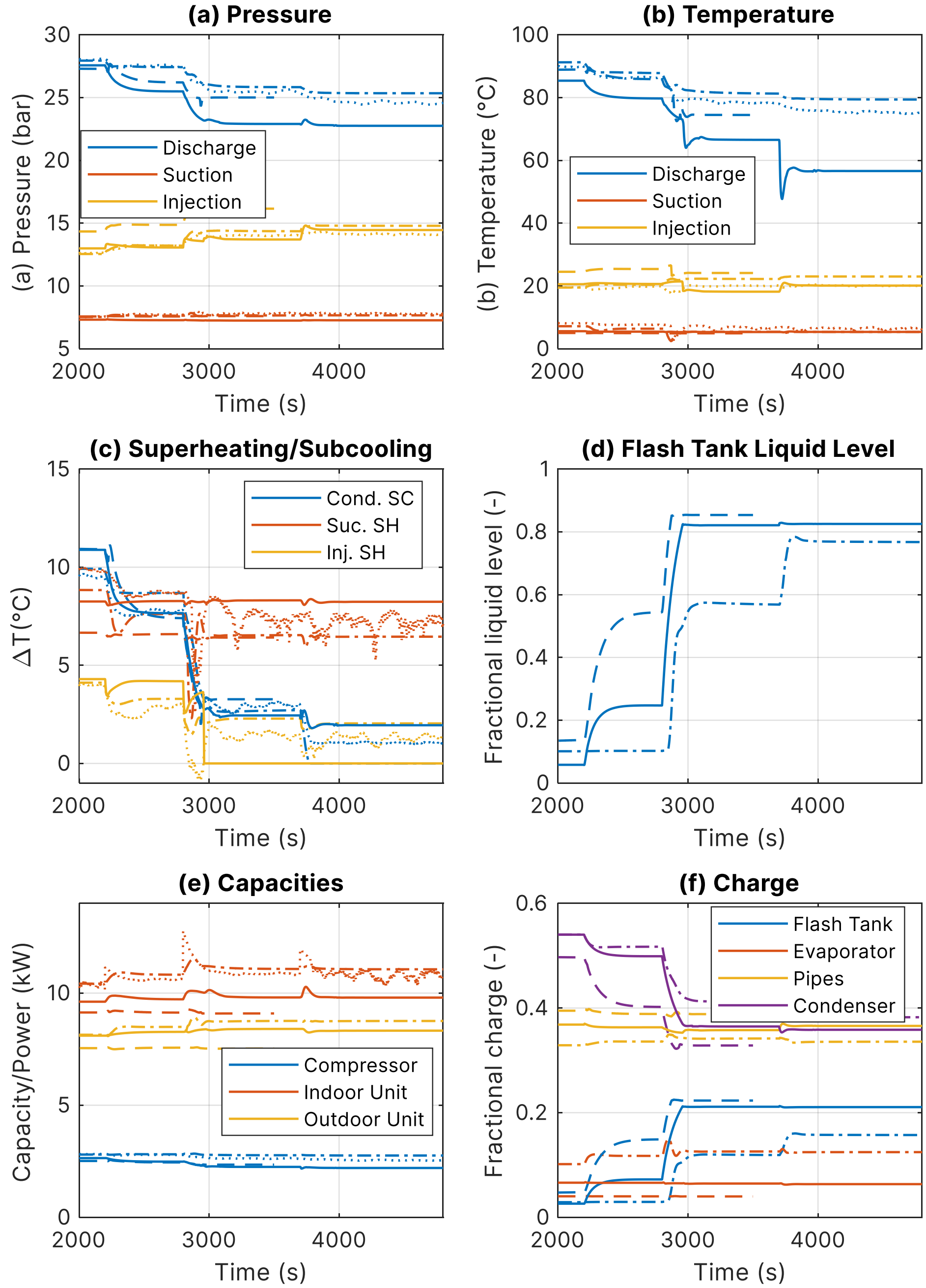

***Figure 6.** EEV step change transients. **Dotted lines** are measured data, **dot-dash lines** are Dymola, **dashed lines** are Simulink data and **solid lines** are EcosimPro data.*

The discharge pressure and temperature are not accurately predicted by the EcosimPro models, although the starting values before the first step change are relatively accurate. The condenser capacity is under-predicted, and the condenser charge is much lower than predicted by the Dymola model. These under-predictions cause a reduced discharge pressure, and thus a reduced discharge temperature in the compressor.

The flash tank liquid level is poorly predicted. In particular, the flash tank begins to fill up when the valve is 20% open, which is not observed in the Dymola models. Furthermore, during the next stage, the flash tank is nearly fully filled. This leads to the lack of any superheating available in the suction side, since liquid is flowing out of the vapour port of the flash tank. These changes may be explained by differences in charge prediction since the excess charge not circulating in the cycle accumulates in the flash tank, which here acts as a liquid receiver. Interestingly, the sudden filling of the accumulator was reported by Qiao et al. in an initial study of the system (2012).

The condenser subcooling was reported by Qiao et al. (2015b) to go to 0 K for the maximum valve opening, although experimental data still shows 1 K of subcooling. The condenser subcooling is relatively well predicted by the EcosimPro model, although the maximum valve opening condition does not show 0 K subcooling.

The capacities and compressor power results show that the general trend of the system is captured. In particular, with a greater opening of the valve, the flow rate through the compressor increases, leading to higher capacities in both the indoor and outdoor units. On the other hand, the compressor power consumption decreases due to the reduced pressure ratio. The EcosimPro model under-predicts the power consumption, however, since the pressure ratio is lower due to the lower discharge pressure.

Overall, further work is needed to understand the differences between the Dymola model and the EcosimPro results, in particular the differences in the filling level of the flash tank, and the discrepancy in the discharge temperature measurements. Nevertheless, several areas where EcosimPro simulations fall short have been explained through a lack of availability of component parameters and initial conditions. It is worth noting however, that the differences between Simulink and EcosimPro suggest that quite a bit of ground can be made up by including detailed internal volumes, and in forgoing the simplified homogeneous fluid assumption for two-phase flow.

4. CONCLUSIONS

A flash tank vapour injection cycle has been modelled in EcosimPro and the results have been compared against two other platforms: Dymola and Simulink. The models have been compared to measured data and a wide variety of transient parameters have been compared. The EcosimPro model generally does worse at predicting system transients compared to the Dymola model, and the Simulink model is worse still. The discrepancies between the former two are explained through a combination of lack of good initial conditions as well as the lack of model tuning based on measured data.

With access to experimental data before undertaking modelling, the accuracy could potentially be further improved, as evidenced by the accuracy of Dymola models. Nevertheless, several important transients, such as the stability of suction pressure and the reduction of injection superheat and condenser subcooling during EEV opening step changes are qualitatively captured. The difficulty in capturing TXV and compressor performances were in particular found to be significant reasons for the differences in accuracy. Empirical or semi-empirical models are necessary to capture the behaviour of these components, especially in off-design and dynamic conditions.

For modellers, it is thus useful to bear in mind that, in the absence of detailed component information, there are limits to how much accuracy can be obtained in transient simulations of systems. Previous experience with the systems under consideration can be useful in helping to understand shortfalls in simulation results.

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