Transient Organic Rankine cycle Modelling for Waste Heat Recovery on a Truck

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Abstract:

The Organic Rankine Cycle is showing promising results for waste heat recovery on long haul truck applications. This technology could further increase the efficiency of current truck powertrains.

In such a context, the dynamic simulation of a Rankine cycle is found to be very important to study its starting and shutting down phases, control strategies and their limits. Such studies are not always easy to perform on a test bench.

This paper deals with the dynamic simulation of a Rankine cycle done under a one dimension commercial fluid dynamic simulation tool (GT-Power). After a brief summary of the component modelling, the paper focuses on the starting and initialization of the model as well as the strategies applied to make the simulation converge. Tank sizing and temperature limitations are addressed to illustrate the use of the model.

Keywords:

Rankine cycle, waste heat recovery, transient modelling, truck.

1. Introduction

Today, diesel engines used in commercial trucks for long haulage rolling distances have achieved an important efficiency while respecting very low pollutant emissions. A high amount of energy is still rejected to the ambient either directly through exhaust gases or indirectly in the radiator or the charge air cooler placed in front of the vehicle. Due to the increase in oil prices but also possible environmental constraints in carbon dioxide emissions **[1]** waste heat recovery techniques (WHR) appear to be very attractive and potentially viable economically. Among all possibilities, the Rankine cycle system has already been installed on mobile applications such as trucks **[2-3]** and cars **[4]**. It is even becoming of high interest and known as a potential solution to further improve truck efficiencies in the United-states for future carbon dioxide regulations **[5]**. A first assessment of the Rankine cycle on trucks **[6]** has already demonstrated that main limitations for such a system are the limited cooling margin, the expansion machine but also an adequate control to maximize at each instant the recovered power. In such a context, the transient simulation of a Rankine cycle system appears as very interesting to improve the knowledge of the system behaviour under truck constraints.

This paper will briefly remind the Rankine cycle architecture simulated under a one dimension commercial tool widely used within the automotive industry (GT-Power). Each component modelling will be reviewed. Most of them are based on implemented component models available in the GT-Power two-phase systems library. The condenser model has already been introduced in a previous paper as an example **[6]** and will be reminded.

Simulations need a proper initialization to converge each time to the right value and behave correctly. As a low number of papers really deal with initial conditions used in transient Rankine cycle system simulations, the second section will explain a possible solution. The focus is done on the starting phase with associated temperature limitation in the last section, the evolution of the tank liquid level. Finally, steps are done on three indentified actuators for the Rankine cycle system. Their analysis gives first ideas on possible control solutions and time transients.

The good understanding of the simulation tool and behaviour is necessary to carry out transient studies as shown in this paper but also to establish adequate control strategies by software in the loop (SIL) approaches. Furthermore, making transient simulation studies enable to predict the behaviour of a system concept before the experiments. It is also a way to limit costs by studying various Rankine cycle designs before making experiments. Transient modelling evaluations and predictions are of high interest to test strategies and better understand critical boundary limits: the maximum fluid temperature, the reservoir emptiness, the maximum evaporative pressure are examples.

2. Component modelling

2.1. Introduction

Rankine cycle modelling has already been reported in the literature. Several approaches are usual in the literature. First, thermodynamic modelling evaluations consisting in making very simple 0D models are very adequate for first assessments of the system. Several authors have evaluated various adequate working fluids **[7-10]** according to the temperature level of heat sources recovered. Those studies are also adequate for a first design of the system components as well as to understand the impact of the Rankine cycle closely coupled to the expander. For a further and deeper evaluation of transients for control studies, a one dimension approach is considered as it integrates a better component sizing but also time constraints (temperature overshoot, speed ramps). Main simulations software already reported in the literature are Modelica/Dynamola **[11-12]**, Matlab-Simulink where a dedicated library has been developed by Vaja **[13]** for intern combustion engines and Rankine cycles.

The one dimensional tool used in this paper is GT-Power. This software is already well spread out among the automotive industry for engine simulations. It is made of thermal, mechanical, electrical component libraries. The overall system modelling is then possible by placing components linked to their pipes as it would be on a real system. Recently, a two-phase library has been developed for air conditioning and Rankine cycle waste heat recovery systems.

As a model library is implemented, the improve learning curve is more fast than on other computer codes, which is of interest for industries. The main goal is to understand the physical behaviour of the system and improve knowledge on the WHR system. On the contrary, the main drawback is that numerical problems are not always well understood and that the solver is not well known. However, it has already shown enough robustness and still is in a continuous development phase.

The simulations solve the equation of mass continuity, energy and quantity of movement, shown hereunder, in a one dimension by the implicit method. This method is more adapted for low time response simulation systems. The discretization is done with a staggered grid. The two phase state is resolved by a homogeneous model using the averaged property between the liquid and the vapour state.

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m} \tag{1}$$

$$\frac{d(\rho HV)}{dt} = \sum_{boundaries} (\dot{m}H) + V\frac{dp}{dt} - kA_s(T_{fluid} - T_{wall}) \tag{2}$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries} (\dot{m}u) - 4\chi_f \frac{\rho u|u|}{2}\frac{dxA}{\delta_{pipe}} - \chi_p \left(\frac{1}{2}\rho u|u|\right)A}{dx} \tag{3}$$

1.2. Heat exchangers modelling

1.2.1. Evaporator modelling

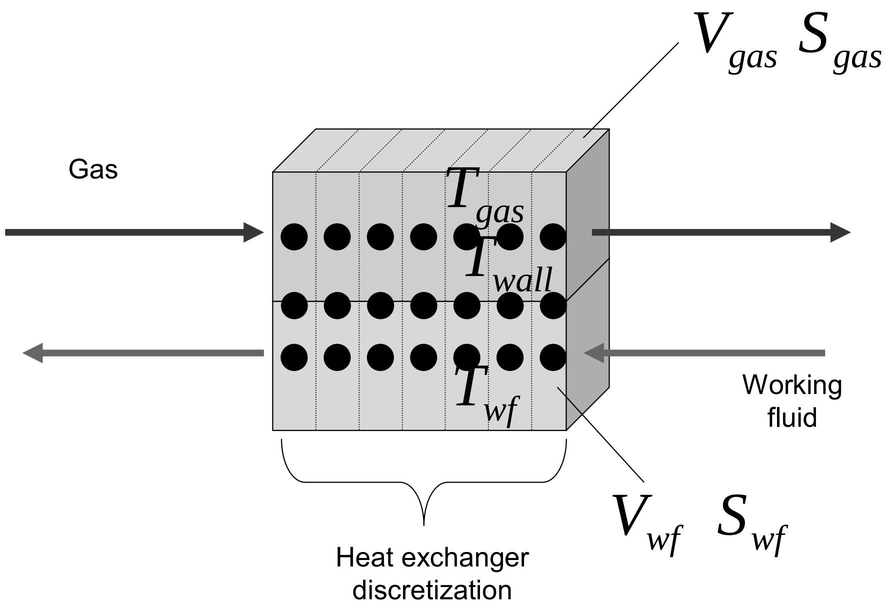


Figure 1 : Equivalent evaporator schematic modelling.

The evaporator recovers heat on various heat sources on the truck either EGR or exhaust gases **[6]**.

This evaporator model uses an already implemented heat exchanger basic model. This basic model can be calibrated against experimental performance data. It consists in a counter-flow heat exchanger discretized along the flow path. The discretization is done every centimetre on the gas pipe side. It is a compromise between the required accuracy and simulation time. It was found that a low amount of discretization could lead to difficulties in the convergence and non physical values. The volume of each fluid side noted V_{gas} and V_{wf} associated to the pipe diameter gives the number of tubes (it is guessed that this number of tubes is calculated by GT-Power). The heat transfer surface area on each fluid side is known from the heat exchanger supplier. Once the number of tube is known, fluid dynamic parameters can be calculated and then heat transfer coefficients. In a detailed model, the fin geometry, the refrigerant flow path, diameters would have been known. As the goal of this model is to be adapted to various geometric type evaporators it was not retained.

In the model, heat transfer coefficients are defined by implemented correlations. After several attempts, it turns out that Klimenko correlation **[14]** was the most stable in the Rankine cycle simulations with the fluid used. Gungor and Winterton correlations **[15]** gave high heat transfer coefficient predictions that lead to instabilities. This was observed for the refrigerant equivalent tube diameter identified. Each time the correlation is changed, a new calibration has to be done against the heat exchanger performance supplier data. Those data gives the amount of heat transferred and the outlet evaporator temperatures for various inlet conditions (mass flow rates and temperatures on the gas side and the refrigerant side). For the pure liquid and vapor phase, the well known Dittus-Boelter correlation **[15]** is adopted on the working fluid side.

On the gas side, the coefficient A and exponent b of the Dittus-Boelter correlation given in (4) are adapted as well as heat transfer multipliers on the refrigerant side to match the supplier data.

$$Nu = A\,\mathrm{Re}^{b}\,\mathrm{Pr}^{\frac{1}{3}} \qquad (4)$$

Pressure drops are calibrated with supplier data by an additional component representing pressure losses on the fluid as a function of the fluid state before and after the evaporator. The equation under the form given in (5) was found to match relatively well data from the supplier for first transient estimations in nominal conditions of the Rankine cycle. Parameters of this equation are identified by the fminsearch function (implemented in Matlab) on all operating points with the law presented. Limiters are implemented in the simulation model on those pressure drops when mass flows used are out of the correlation validity range. The pressure drop model will probably be refined in the future as it is better implementing pressure drops that evolve along the evaporator and avoid limiters.

$$\Delta P = C\,\dot{m}^{d}\,\left(\mu_{in}{}^{e} + \mu_{out}{}^{f}\right)\left(\frac{1}{\rho_{in}{}^{g}} + \frac{1}{\rho_{out}{}^{h}}\right) \tag{5}$$

1.2.2. Condenser modelling

The condenser model is based on existing automotive parallel flow (PF) air-conditioning condensers installed on a commercial truck. The condenser model is a parallel flow with louvered fins on the air side. The main assumptions in this model are the followings: refrigerant passes are not taken into account for simplicity, micro-channels used to enhance heat transfer in PF condensers are not modelled, the fluid distribution is considered as uniform and the longitudinal conduction through the wall is not modelled.

The air side heat transfer coefficient is predicted by the Chang and Wang correlations found as a basis in several papers for modelling condensing or louvered fin radiators **[16] [17]**. According to Schwentker et al. **[18]** it is recognized as an accepted standard correlation used in heat exchanger modelling. In the same manner, gas side pressure drops are predicted by means of friction coefficient correlations taken from the literature. In this case Dong et al. correlation has been preferred. The pressure drop and the heat transfer coefficient expressions are then integrated along the condenser in a Fortran user model implemented in GT-Power. The latter user model enables to define the heat flow rate between the air side and the wall for the particular geometry used.

On the working fluid side, for pure liquid and vapour parts the heat transfer coefficient is given by the Dittus-Boelter correlation **[15]** whereas Shah **[15]** is used for the two-phase zone. The latter correlation is found as easier to be implemented in the user model as it does not need boiling flow maps.

1.3. Turbine modelling

A kinetic turbine is modelled according to a component model implemented in GT-Power for turbo-compressors. The turbine is represented by a performance map classically represented by two non dimensional graphs for kinetic turbines: normalized efficiency versus normalized blade speed ratio and normalized mass flow rate versus blade speed ratio.

The model is based on the identification of coefficients from the well known parabolic form of the reduced efficiency η_{norm} and the reduced blade speed ratio BSR_{norm} respectively defined in (6) and (7).

$$\eta_{norm} = \frac{\eta}{\eta_{peak,eff}} \tag{6}$$

$$BSR_{norm} = \frac{BSR}{BSR_{peak,eff}} \tag{7}$$

The first part of the curve (when $BSR_{norm} < 1$) is given in (8) whereas the second part ($BSR_{norm} > 1$) is shown in (9).

$$\eta_{norm,fit} = \left(1-(1-BSR_{norm,fit})\right)^{i} \tag{8}$$

$$\eta_{norm,fit} = 1-\left(\frac{BSR_{norm,fit}-1}{z_0-1}\right)^{i} \tag{9}$$

A similar fitting method is done by Dinescu et al **[19]** and Vaja **[13]**. They only used the first term with $i = 2$ for the entire curve.

BSR is defined as $BSR = \frac{U}{u_{nozzle,s}}$, where U the blade speed is given by $U = \frac{N\delta_{turb}}{2}$ and $u_{nozzle,s}$ is the isentropic speed achieved by the fluid through an equivalent nozzle defined in (10).

$$u_{nozzle,s} = \left[2c_p T_{in}\left(1-\tau^{\frac{1-\gamma}{\gamma}}\right)\right]^{\frac{1}{2}} \tag{10}$$

According to input conditions in the turbine, the efficiency is therefore directly determined by (8) and (9).

The pressure ratio applied to the fluid as a function of the mass flow rate is directly given by the identification of parameters J and K given by a normalized function (11) fitted from tests data.

$$\dot{m}_{norm,fit} = J + BSR_{norm}{}^{k}\,(1-J) \tag{11}$$

This function gives also the pressure ratio through various turbine rotational speeds. Other approaches are possible like the identification of an equivalent diameter as done in **[13]** and **[19]**. However, the main assumption done in these models is that the mass flow rate is independent from the rotational speed. They are found as more restrictive compared to what is proposed in GT-Power. When considering a volumetric turbine type expander, the expansion machine is simply modelled by an isentropic efficiency and a volumetric efficiency that are considered constant whatever the rotational speed. They are respectively defined in (12) and (13). It is also possible to define a proper law identified from experiments. In that case, for example for a scroll expansion machine, Quoilin et al. **[20]** proposed a polynomial law to model scroll expansion machines efficiency.

$$\eta_{vol} = \frac{\dot{V}_{in}}{D_{\exp}\,\frac{N_{\exp}}{60}} \tag{12}$$

$$\eta_{is} = \frac{h_{in,\exp} - h_{out,\exp}}{h_{in,\exp} - h_{out,\exp,is}} \tag{13}$$

1.5. Other components

A simple simulation model is used to model the reciprocating pump. The rotational speed is imposed and fixes the volumetric flow rate through the equation (14).

$$\dot{m}_{pump} = \rho_{pump,in}\,\frac{N_{pump}}{60}\,D_{pump} \tag{14}$$

The working fluid reservoir is simply modelled by a volume. Other components such as by-pass valves are requested on the evaporator side but also on the refrigerant side. Each three-way valve is modelled by two orifices restrictions placed on each pipe. Those orifice restrictions are opened and closed on a reverse manner to direct the working fluid (WF) either on the main line or on the by-pass line. The mass flow rate going through the orifice diameter is given by the well known nozzle equation not reminded here.

2. Simulation strategy

2.1. Initial conditions

The strategy adopted consists in starting the simulation as the real system would behave in reality. It is important to well know starting cycle conditions to further make sensitivity analyses on the cycle. Several attempts where done to directly start the simulation from known thermodynamic conditions without success. For that purpose the convergence needs to be driven from the initial state defined hereunder to the steady state operating point.

When the Rankine cycle system is off, the liquid refrigerant migrates to a position in the circuit that depends on the temperature (heat pipe effect) but also on the pipe elevation (gravity effect). The condenser is typically placed above the tank that is above the subcooler and the pump in the preferred refrigeration system embodiment. A certain amount of liquid is expected to stagnate mainly in parts located at the bottom of the installation. The other components are filled with vapour in equilibrium with the liquid. Figure 2 represents parts that are considered liquid or vapour initially in a simple Rankine cycle system studied in this paper. When components are placed in a different way, the initial liquid and vapour position might migrate and those initial conditions have to be changed. The tank, the pump and the subcooler are filled with liquid whereas the piping going from the pump to the condenser is in vapour state. The only way to change the amount of refrigerant in the cycle is thus to increase the level in the tank by changing the initial fluid density (liquid level) or the tank size.

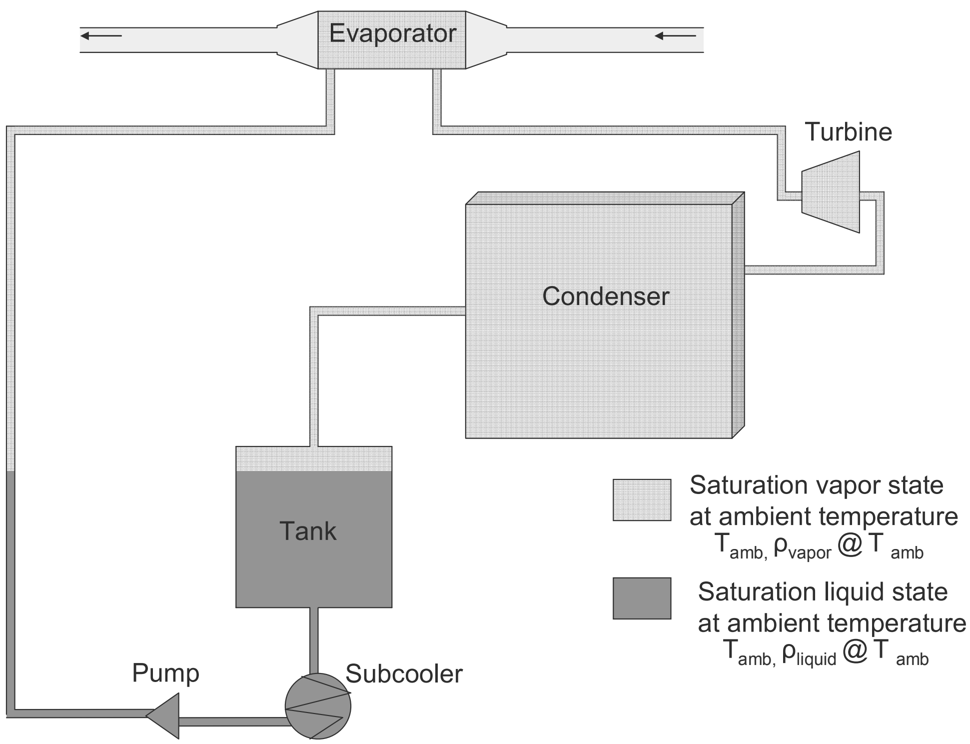


Figure 2 : illustration of initial conditions used in the simulation.

2.2. Starting phase

As the convergence of the model to the steady state is done like in real conditions (transient model), a strategy has been adopted to start the cycle in order to reach a steady state point. This first strategy does not intend to be the final strategy to be adapted on a real system as it requires a deeper control study. However, it gives already a first understanding of parameters influencing the starting process of the cycle. The following paragraph describes the modelling of what could be a starting phase of the specific Rankine cycle architecture shown in fig. 2.

Exhaust gases are considered to go directly through the evaporator a few seconds after the start of the simulation. The effect is the overshoot on the evaporating temperature as shown in fig.3. At the start, the pump speed is progressively increased. This speed ramp turns out to be important to start without having divergences due to sharp transitions. The slope of the pump speed has a high impact on the overshoot inlet turbine temperature shown in fig. 3. The working fluid vaporizes after 10 seconds. An open loop logical control is integrated in the simulator taking the superheating at the turbine inlet as an input. As soon as the superheating becomes more than 10 K, the expander line orifice starts to open. The evaporative pressure starts to increase due to the orifice restriction at the turbine inlet.

This example is a possible starting operating phase for the simulated Rankine cycle. It is important to always keep the temperature under the maximum thermodynamic properties of the fluid simulated (existing in the simulation software) in order to stay in a physical state. In the simulation done, the supercritical state was not considered. As a result, this state has to be avoided to keep fast convergence calculations as supercritical properties were not calculated by GT-Power. It is also important to have a strategy of expander by-pass during the starting phase. If not, a risk exists in admitting liquid in the expander.

When using a volumetric expansion machine, the rotational speed becomes not adapted for the liquid-vapour mixture. If this expander speed is conserved, the mass flow rate imposed by the expander is too important. An accumulation of mass occurs in the condenser increasing the pressure and causing to non physical states convergence.

Using the by-pass properly simplifies the starting phase of the simulation and avoids reaching those non physical states.

The expander by-pass can be useless if the fluid is directly superheated. In that case the initialization is less easy as the pressure together with the superheating has to be managed in the same time. For preliminary studies (Rankine system design, first control strategies), the use of the by-pass has been preferred to separate initialization into phases described in this paragraph.

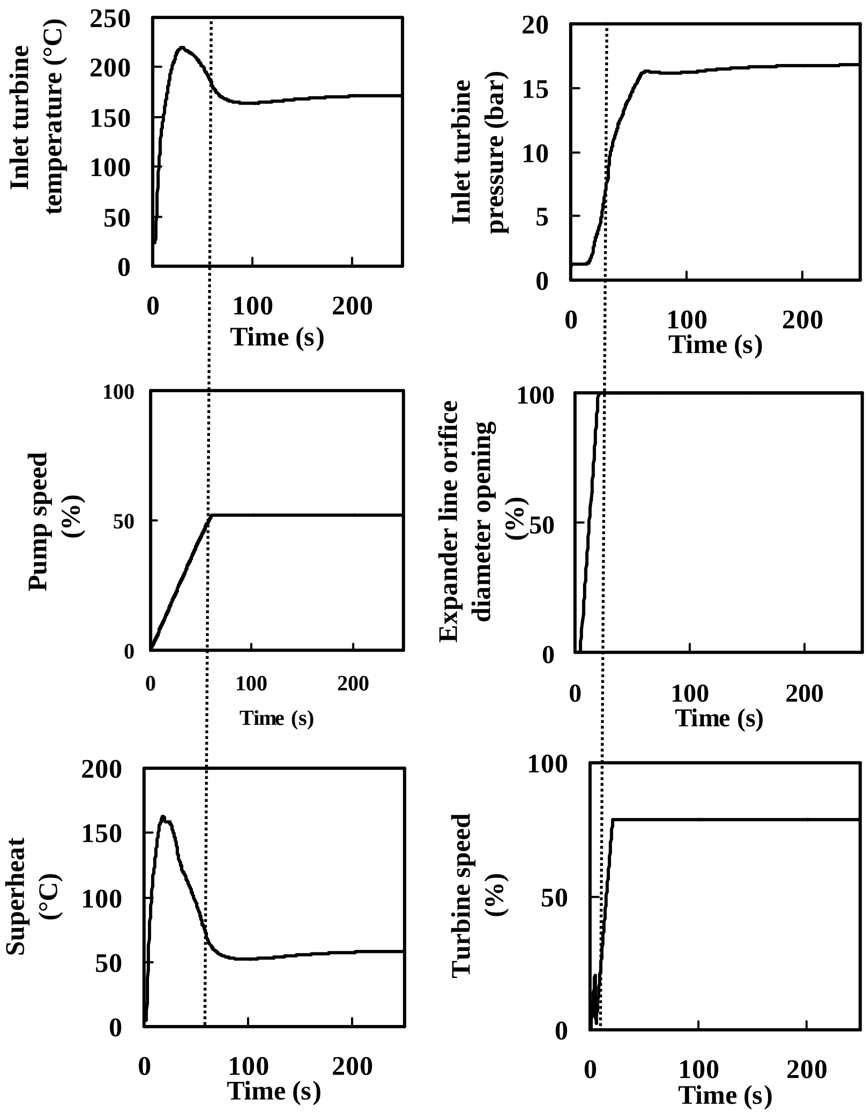


Figure 3 : starting phase of the Rankine cycle under a typical simulation.

3. Studying examples

2.1. Main control variables for a Rankine cycle system on a truck

The main variables to control the Rankine cycle strongly depend on the environment and the application. In case of organic Rankine cycle (ORC) stationary applications, the amount of heat rejected by the cycle can be easily controlled by managing fans on the cooling system side. In the truck application, it is shown that it is not more possible due to limited cooling capacities **[6]**. Consequently, the evaporator by-pass (needed to manage the amount of heat coming into the Rankine cycle), the expander by-pass and the pump speed are only considered here as the main drivers to control the Rankine system on a truck. The expander speed is not considered controllable as it is imposed by the load applied to its shaft:

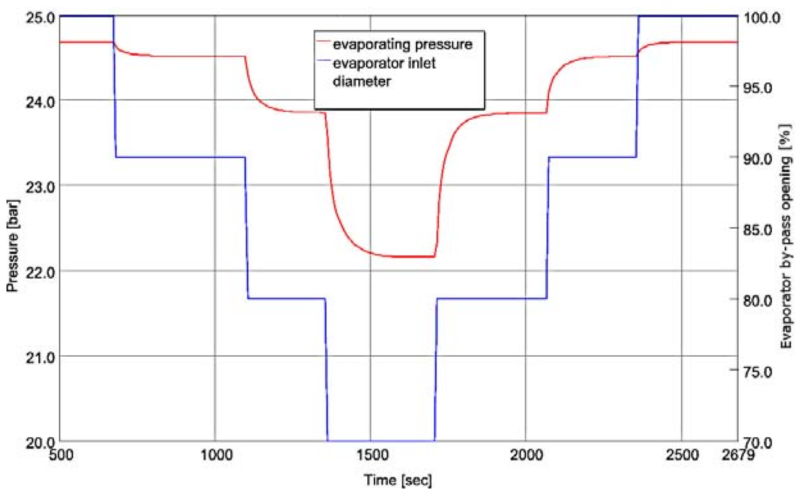
- when considering a mechanical connection to the engine, the speed is imposed by the rotational speed of the engine. Most of the time a truck runs at a fix engine rotational speed. As a result, it is expected to have a roughly fixed rotational speed.
- Considering an electrical generator connected to the Rankine system to produce on board electricity, it is again difficult to control the speed as electrical consumption varies a lot, applying a variation of load and therefore in speed to the expansion machine.

After implementing the initialization shown in the previous section, it is possible to identify time responses and gains output obtained as a result of a step input. Figures 4, 5 and 6 show the impacts on a Rankine cycle using a simple volumetric expander model.

The evaporator by-pass closing or opening influences both the temperature and the evaporative pressure in the cycle. For the system designed, the time response seems in the order of 100 seconds for the evaporative pressure (fig.4a) and the corresponding temperature (fig. 4b) when the evaporator by-pass is closed (no waste heat gases go through the evaporator).

The expander by-pass has a relatively low impact on the inlet expander temperature (fig. 5b). Small peaks in temperature are observed but remain low (less than 10 K) despite important variation of by-pass orifice diameter. It has however a radical impact on the evaporative pressure (fig.5a) in the cycle. The evolution is very fast and important compared to the evaporator by-pass impact. The time response follows the step imposed (less than 10 seconds after).

The pump speed also impacts both the inlet turbine temperature and the evaporative pressure (fig. 6a and 6b) within 2 bar for the given step variation. However, the evolution in inlet turbine temperature is faster compared to the evaporator closing (20 to 30 seconds). When comparing the impact on the superheating, it can be shown that the pump imposes the superheating (fig.6c). When analysing the transients observed it can be concluded that the evaporator by-pass time response is slow compared to the pump speed control and the expander by-pass control. The pump speed variation and the expander speed by-pass are two actuators to be changed first while the evaporator by-pass opening or closing occurs. Furthermore, what is shown is that the pressure can be managed by means of the expander by-pass. The temperature can not be directly managed as it is affected by all the actuators. However, the superheating strongly depends on the pump speed and the evaporator thermal power.



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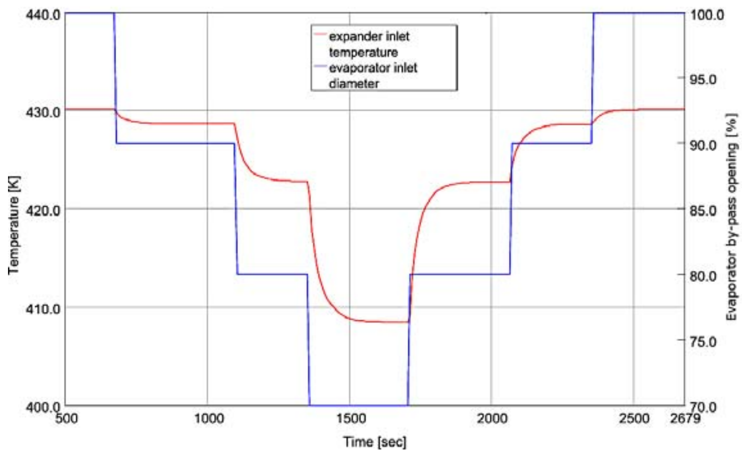

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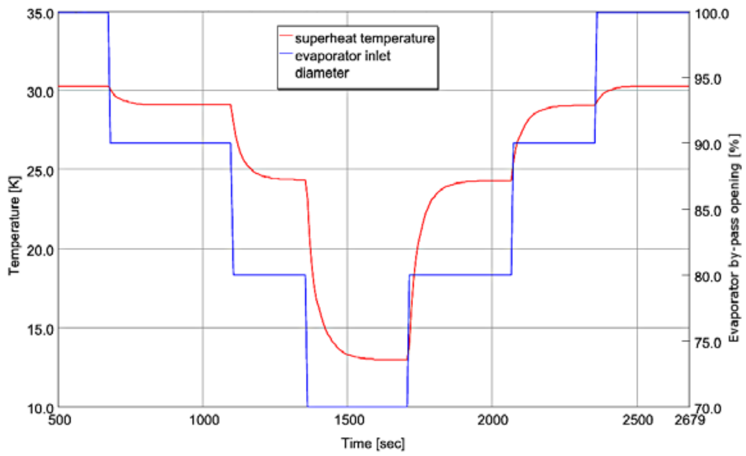
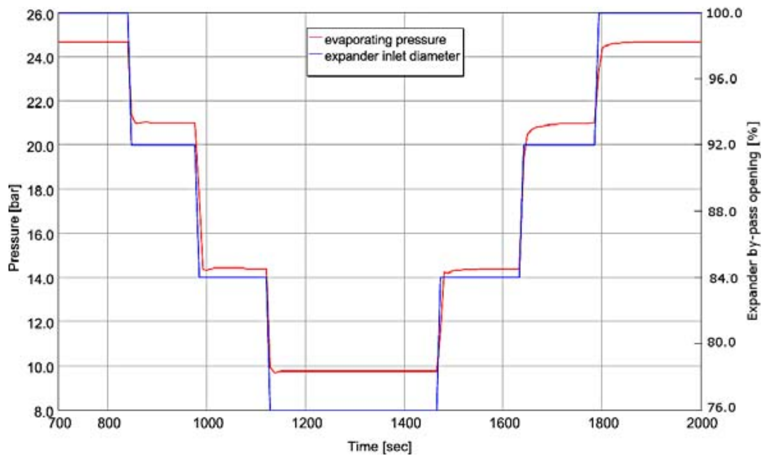


(c)

Figure 4 : Impact of the evaporator by-pass opening on the evaporative pressure, the inlet turbine temperature and the superheating.



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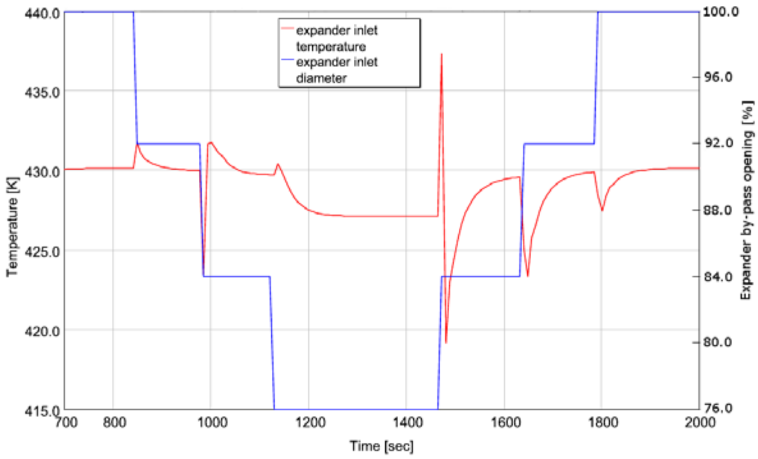

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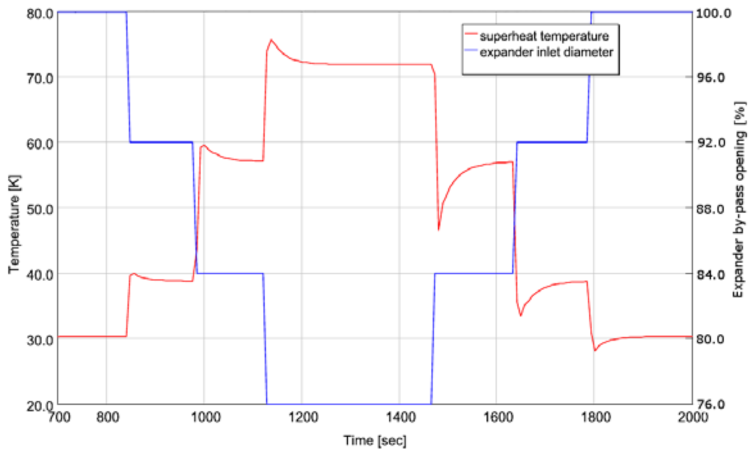
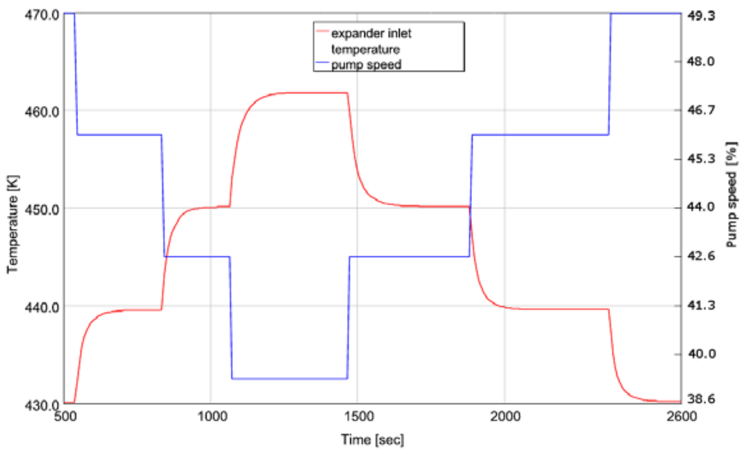


(c)

Figure 5 : impact of the expander by-pass opening on the evaporative pressure, the inlet turbine temperature and the superheating.



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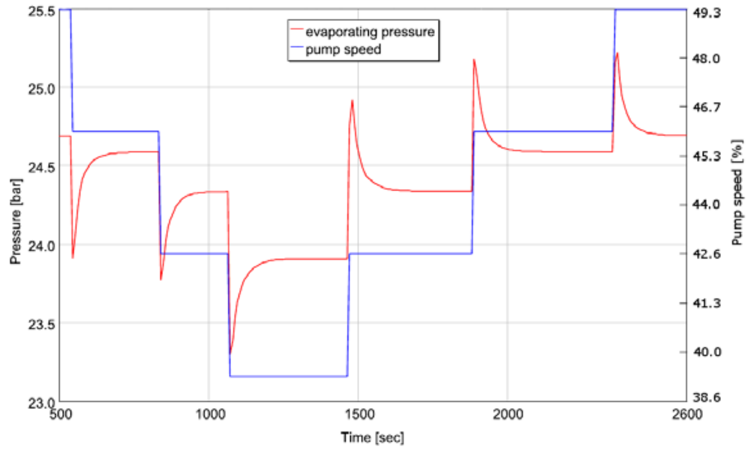

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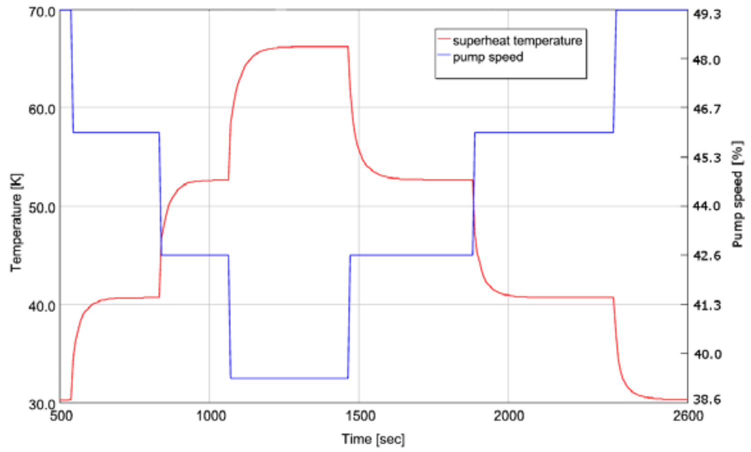


(c)

Figure 6 : impact of the pump speed on the evaporative pressure, the inlet turbine temperature and the superheating.

2.1. Temperature limitation

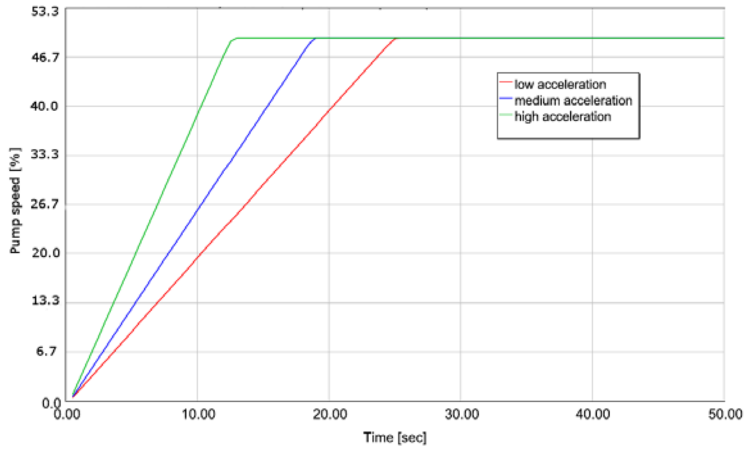
The Rankine cycle starting phase can lead to exceed the maximum working fluid (WF) temperature allowed. This limitation can be imposed by the temperature stability of the fluid **[21]**. That is the case for instance with the HFC-245fa taken as an example in simulations presented in this paper. The temperature limitation can also be imposed by the lubricant, for instance in a scroll machine. The latter types of machines have been shown to be a promising technology in expansion mode **[22]**. In that case, most lubricant oils are characterized by maximum temperatures in the range of 150 to 170 °C. Finally, the generator, if present in the system, can impose a limitation on the temperature. When considering a hermetic type expansion machine, the generator is directly in contact with the fluid. In that case, the fluid temperature is directly limited by the maximum generator temperature allowed. For opened machines, the generator is linked by the expander through the shaft. The WF is not in direct contact to the generator. The latter case also means a temperature limitation (but less critical) as a part of the heat from the working fluid goes to the generator by radiation and conduction through the shaft.

It has been shown in **[6]** that the cooling power on the truck was limited in particular when recovering on exhaust gases. The cooling system is currently designed to reject the heat for the engine coolant only. This amount of rejected power on the engine coolant is nearly the same than the one on exhaust gases. This means that the heat rejection of a Rankine cycle system recovering on exhaust gases would nearly double the amount of heat rejection on the truck. As it seems difficult with today's cooling architecture, it is expected to have Rankine cycle engagement and disengagement with the cooling capacity available for a pure exhaust gases heat recovery. The Rankine system is expected to start as fast as possible to recover a given amount of power as soon as the cooling power is available on the truck.

As illustrated in fig.3, when both the pump speed increase and the expander by-pass line opening are done in the same time a temperature overshoot exists. This overshoot comes from the fact that when started, the vapour in the evaporator is directly superheated and that the initial mass flow rate imposed by the pump during first starting seconds is low compared to the gas side temperature and mass flow rate. This overshoot can be diminished mainly by managing the heat coming into the cycle (evaporator by-pass) and the slope of the pump speed increase. The effect of the slope of the pump speed increase is shown in fig. 7a showing different slopes used as well as the resulting inlet turbine temperatures (fig. 7b). The low acceleration shows a physical behaviour (no superheating oscillations). A higher acceleration combined to the simultaneous increase in pressure done by the expander line opening (conversely expander by-pass closing) leads to a decrease in superheating. A strategy is applied to close the expander line when the superheating becomes negative. This strategy is applied by logical tests done on the working fluid pipes in the simulation. The superheating is sensed upstream the turbine. When this superheating is greater than 10 K, the expander by-pass line opens with a given ramp. When the superheating is less than 10 K, the expander by-pass line closes progressively (5 seconds for a full opening or closing is used) keeping the same ramp than for the opening. It turns out that when the superheating is decreased suddenly by a rapid increase in pressure, the expander by-pass line start a closing phase (the expander is by-passed). This closing phase decreases the evaporative pressure increasing the superheating. The superheating becomes positive and the expander by-pass line starts to increase another time. Oscillations appear in the cycle due to this expander line closing and opening strategy. Finally, the superheating levels off to the steady state value. This explains why the temperature oscillates in the medium for the high acceleration pump.

Solutions found in the simulation are: the management of the expander speed (volumetric expander used in this example), a lower expander by-pass opening speed and the use of a limited orifice diameter (the valve opens at the same speed but not fully). It can be shown in fig.8 that all those measures have an effect as they limit the increase in pressure and therefore the decrease in superheating. However, the first one is generally technically more difficult to implement due to difficulties to control the expander speed. Using a smaller final by-pass expansion diameter leads to a lower evaporative pressure value as part of the fluid is by-passed. This solution is not preferred as it leads to a low evaporative pressure and less power recovered at the start of the cycle. The adequate solution is therefore to use a slower expander opening.

This simple case illustrates that a necessary minimum amount of time will be necessary to start the Rankine system.



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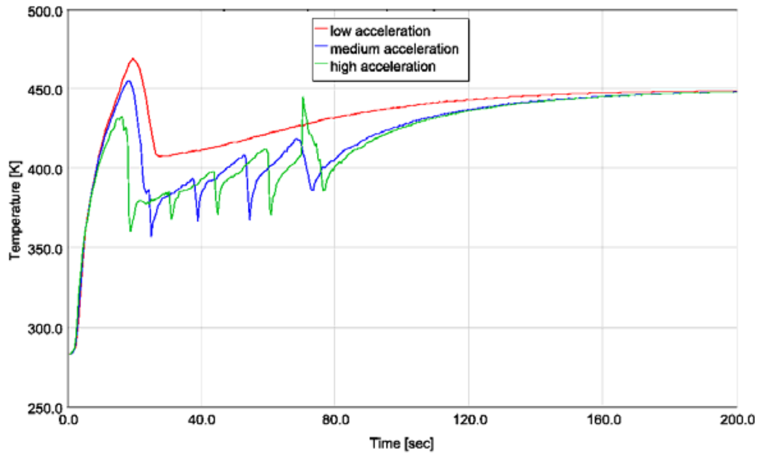


(b)

Figure 7 : Evolution of the inlet expander temperature with the simulation time for three pump speed acceleration. Low acceleration is preferred to avoid oscillations. (a) – imposed pump acceleration ramps. (b) – evaporator outlet temperature.

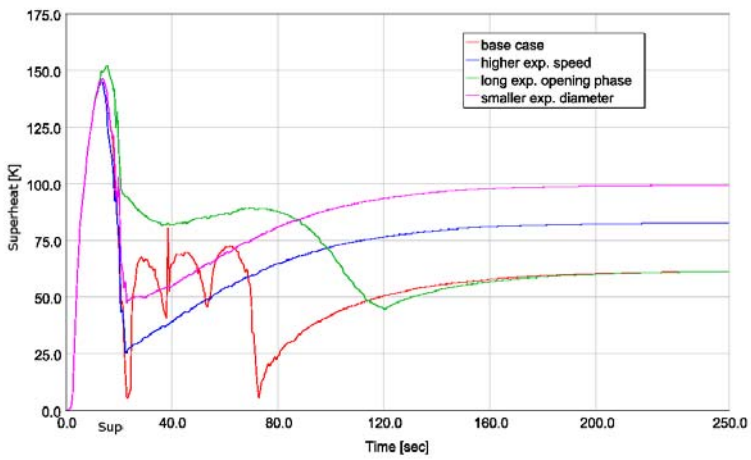


Figure 8 : influence of a higher expander speed, a longer expander opening phase and a smaller expander diameter with the high initial acceleration pump speed.

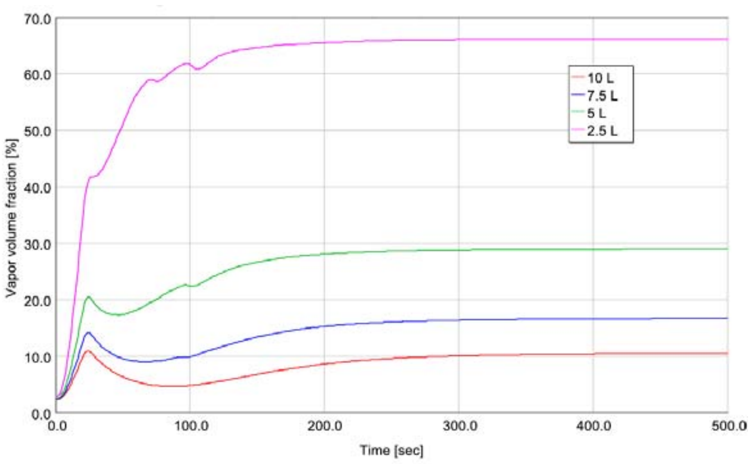
To conclude, the Rankine cycle system should start as fast as possible. However, it has a time response that should be respected. A too large amount of heat initially fed into the cycle leads to an overshoot in evaporator outlet temperature. As a result, the mass flow rate imposed by the pump should increase as fast as possible to limit the maximum temperature (generally limited either by the refrigerant properties in the software or physically on a real system by component temperature tolerances mentioned previously). This action has to be performed with a slower expander by-pass opening (or expander speed if controllable) to avoid that the superheating becomes null implying liquid droplets going into the turbine. It can also lead to oscillations if a strategy consisting in closing the bypass line at a given superheating is used.

2.1. Tank sizing

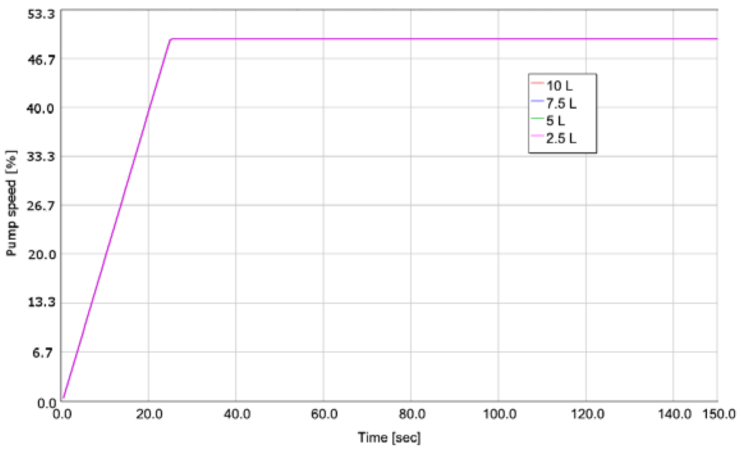
The tank is expected to be temporarily emptied at the start of the cycle due to a sharp increase in the pump speed. The worst scenario is shown in this paragraph. The Rankine cycle starts when the recovered gases conditions are relatively high in temperature and mass flow rates. The evaporator by-pass is fully opened. Figure 9 shows the vapour volume fraction (VVF) of the tank defined in (15).

$$VVF = \frac{V_{\tan k,vap}}{V_{\tan k,vap} + V_{\tan k,liq}} \qquad (15)$$

Equation 15 shows that the VVF is an image of the amount of liquid in the reservoir. A sharp peak increase in the VVF is shown and is linked to the increase in the pump speed during first seconds as shown in fig. 9 comparing different tank sizes used. For these simulations, the pipe elevation is considered as being null. Gravity effect is not considered. The pump speed ramping up is done during the first 25 seconds of the simulation leading to an increase in VVF. Then, once the conditions are stable, a certain amount of time is necessary to reach steady state conditions in the tank. As a first step, different tank size comprised between 100 seconds and 200 seconds are investigated. The tank density is initially the same meaning the for a tank size increase the initial amount of refrigerant introduced raises. The smaller tank reaches the steady state conditions more rapidly. Contrary to what could be expected, there is more scattering in the values of the VVF (related to the different tank sizes) at the final steady state condition than during the initial condition due to the pump liquid suction. Figure 9 also shows that the smallest tank would be sufficient for the Rankine cycle simulated and given initial conditions (a 2 liter tank is further investigated in this paragraph). The only impact between the different tank sizes is the tank liquid level that is reached once the steady state condition is observed. The larger the tank is, the higher the VVF is and the lower the tank liquid level is. This can be explained by the fact that as the initial amount of refrigerant increase only in the tank, in steady state, the fluid is stoked in then stoked in the tank.



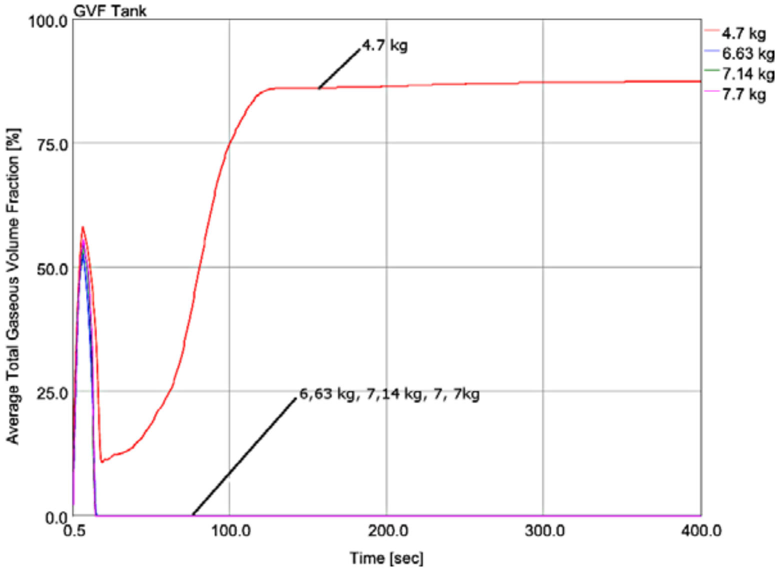
(a)



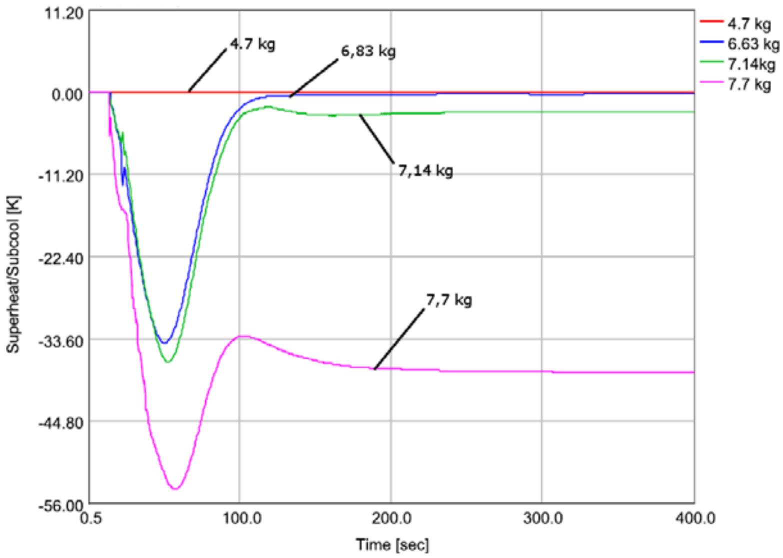
(b)

Figure 9 : evolution of the vapor volume fraction within the reservoir against time for 4 different tank sizes.

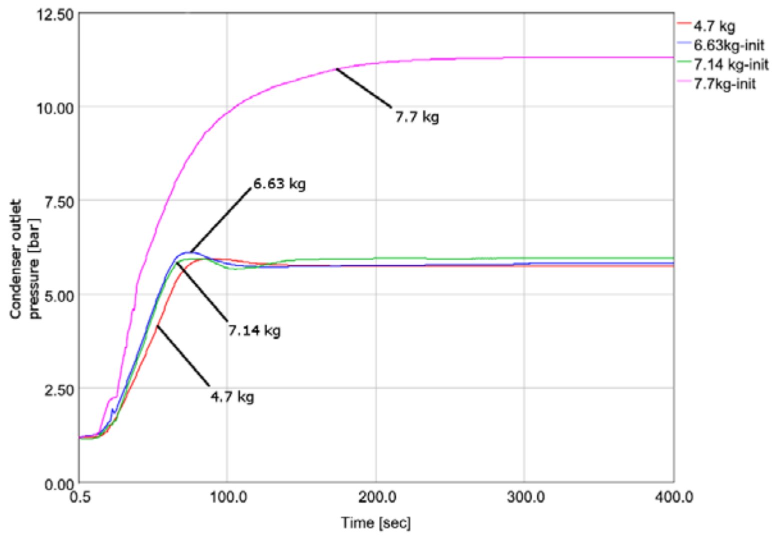
The same reasoning can be done keeping the tank size fixed and changing the charge of fluid initially introduced (the condenser becomes filled in liquid) for a 2 liter tank. It is possible to find the adequate fluid charge. This effect is illustrated in figure 10 showing the VVF (fig. 10a) as well as the condenser subcooling (fig. 10b) for this tank.



(a)



(b)



(c)

Figure 10 : (a) vapor (gaseous) volume fraction within the tank, (b) tank liquid subcooling and (c) condenser outlet pressure (c).

It shows that the starting phase in the simulation has a first tank level decrease and then increase. Two steady state tank conditions are encountered depending on the amount of fluid initially introduced. The tank is finally either partially filled with liquid or full of liquid. Once full of liquid, the tank fluid can subcool (cases at 6,63, 7,14 and 4.70 kg).

When the charge is too high, a certain amount of liquid is stored in the condenser increasing the subcooling and resulting in a condensing pressure increase (shown in figure 10c). For 7.7 kg introduced, the pressure increases a lot to around 11 bar. This effect can be explained by a lower amount of heat transfer surface available for the phase change in the condenser. The only way to condense is to increase the temperature difference between the working fluid and the cooling fluid and that is why the condensing pressure raises. As a consequence, the condensing pressure increase directly affects the expansion machine performance.

When the charge is not sufficient so that during transient states the tank VVF reaches 100 % a simulation stopper is introduced. If not, this has the effect to bring a mixture of vapour and liquid to the tank. Divergences are observed in the model and the mass balance in the Rankine loop becomes not correct that is why lower refrigerant charges are not shown.

As a conclusion, results presented have shown that the tank VVF was of high importance and needed to be monitored during simulation runs in order to keep a good convergence of the model and stay at the optimum cycle performance. This tank VVF depends on the ratio between the volume of the installation and the refrigerant charge introduced. A slight increase in VVF is expected (a decrease in tank level) when the pump speed changes occur. It is, however, more influenced by the amount of fluid needed to fill the system once the optimized steady state operating point is reached.

3. Conclusion

This paper showed the importance of transient simulations of ORC systems. Such simulations yield in a better understanding of the cycle behaviour in critical situations (starting phase), which can help defining appropriate component design and control.

Several component models are presented. They are based on the component library available in GT-Power. The initial conditions as well as the strategy to initialize the cycle are shown as very important to reach the given steady state requested. It is however strongly correlated to the considered conditions in temperature and pressure and also the initial charge of refrigerant used.

The transient simulations pointed out the key components and inputs to simply control the system in order to reach the convergence (pump speed ramp, expander by-pass ramp). Such simple considerations can then be used to a deeper control system or on other Rankine cycle system. As an example, it is possible to link the Simulink software to this GT-Power model in order to develop a detailed control algorithm integrating the initialization, the transient phases between operating points and the shutdown of the Rankine cycle. Predictive strategies are also possible.

It was shown that a compromise does exist between the time necessary to reach the steady state regime and the limitations on the maximum inlet turbine temperature. Pump speed, evaporator by-pass and expander by-pass are found to be the three control variables in the system presented for the truck application.

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Nomenclature

Abbreviations

WF Working fluid.
WHR Waste heat recovery.
SIL Software in the loop.

A Cross –sectional flow area, m².
A_s Heat transfer surface, m².
BSR Blade speed ratio.
c_p Specific heat, J/kg/K.
D Displacement, m^3.
H Total enthalpy, J.
h Specific enthalpy, J/kg.
k Convective heat transfer coefficient, W/m²/K.
$\dot{m}$ Mass flow rate, kg/s.
N Rotational speed, rpm.
Nu Nusselt number.
p Pressure, Pa.
Pr Prandtl number.
Re Reynolds number.
T Temperature, K.
U Blade speed, m/s.
u Fluid speed, m/s.
V Volume, m^3.
$\dot{V}$ Volumetric flow rate, m^3/s.
VVF Vapor volume fraction.

Greek symbols

μ Dynamic viscosity, Pa.s.
η Efficiency.
ρ Dendity, kg/m^3.
χ_f Friction coefficient.
χ_p Pressure loss coefficient.
δ Diameter, m.
τ Pressure ratio.
γ Adiabatic coefficient of the WF considered.

Subscripts and superscripts

exp	Expander (turbine).
fit	Fitted against data.
gas	Relative to the waste heat gas side.
in	Relative to the inlet.
is	Isentropic.
liq	Liquid.
norm	Normalized.
out	Relative to the outlet.
peak	Peak efficiency.
turb	Turbine.
vap	Vapor.
vol	Volumetric.
wf	Relative to the working fluid side.

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