Exergy Analysis of a Water Heat Storage Tank

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Abstract: A combined heat and power (CHP) plant generates both electricity and useful heat. A heat storage tank enables a decoupling of electricity and heat delivery. In this study a cylindrical hot water storage tank is considered. Charging, holding time and discharging are numerically simulated applying COMSOL Multiphysics 4.2. The performance of the heat storage is evaluated by an exergy analysis. Exergy is the work potential of a given amount of energy and thus the "valuable fraction" of energy. Performing simulations with three different models, the contribution of heat conduction in the tank wall and heat losses to the environment on the overall exergy loss were determined. Both effects are almost negligible during charging, but dominate during holding time. Additionally, the mass flow rate, the inlet temperature and the inlet tube geometry were varied. The influence on the exergy loss of all three factors is rather small.

Keywords: Heat Storage, Exergy analysis, Combined Heat and Power (CHP) Plant.

1. Introduction

1.1 Combined Heat and Power (CHP) Plant

Thermal power plants convert a part of the supplied heat to work. In conventional power plants the remaining part of the heat is rejected to the environment as waste heat. In a combined heat and power (CHP) plant some or all of this rejected heat is used for heating purposes. The overall efficiency of a CHP plant is higher than the efficiency of a conventional plant in combination with a separate heating device. However, this cogeneration possesses the drawback that, in general, both electricity and useful heat are provided simultaneously. Running the CHP plant during times with a high demand for electricity is only possible when the heat can be released at the same time. This coupling between electricity and heat delivery can be reduced by adding a heat storage tank to the system. In this case it is possible to run the CHP plant in order to produce electricity even if there is no current demand for heat, at least until the heat storage tank is filled.

The present work is an enhancement of a Bachelor Thesis [1] that was conducted in co-operation of the Institute of Technical Thermodynamics / TU Darmstadt and the company IAV GmbH. The water heat storage tank of a natural gas CHP plant is examined. It is a cylindrical container with a volume of 938 l and a height of about 2 m. The steel wall of the tank and the outer polyurethane foam insulation are 2.5 mm and 100 mm thick, respectively. During charging hot water (≈91°C) flows into the tank at the top and cold water (≈40°C) flows out at the bottom. The radii of the feed pipes are 38.1 mm at the top and 76.2 mm at the bottom. The transition between the upper hot water region and the lower cold water region moves downwards during charging. The flow direction is reversed for discharging. In this case hot water flows out of the tank at the top, and cold water flows into the tank at the bottom.

1.2 Exergy

The first law of thermodynamics, a balance of energy, states that energy cannot be created or destroyed. In a thermodynamic process energy can only be converted from one form into another. However, not all forms of energy are equally valuable. The second law of thermodynamics, a balance of entropy, enables a statement about the quality of a certain form of energy. The valuable fraction of the energy in a system, e.g. the internal energy in a heat storage tank, is called "exergy". It is the maximum useful work that can be obtained from the system when it is transferred from its initial state in a reversible process into thermodynamic equilibrium with the environment. All irreversibilities such as friction and heat transfer across a finite temperature difference reduce the amount of work that is actually obtained.

2. Governing Equations

The computations are carried out in a cylindrical coordinate system (r, θ, z), whereas all problems are supposed to be axisymmetric ($u_\theta \equiv 0$, $\partial / \partial\theta \equiv 0$). The Boussinesq approximation is applied in order to consider buoyancy forces. For an isotropic, Newtonian, incompressible fluid hold the following conservation equations.

- mass:

$$\nabla \cdot \mathbf{u} = 0$$

- momentum:

$$\rho_0\left(\frac{\partial \mathbf{u}}{\partial t} + \mathbf{u}\cdot\nabla\mathbf{u}\right) = -\nabla p + \nabla\cdot\left\{\mu\left[(\nabla\mathbf{u}) + (\nabla\mathbf{u})^T\right]\right\} + \rho_0\mathbf{g}\left[1 - \beta(T - T_0)\right]$$

- energy:

$$\rho_0 c\left(\frac{\partial T}{\partial t} + \mathbf{u}\cdot\nabla T\right) = \nabla\cdot(k\nabla T)$$

For the exergy of a closed system holds [2]
$W_{ex} = (U - U_0) - T_0(S - S_0) + p_0(V - V_0)$.
If the system is occupied by an incompressible fluid such as water, the last term cancels out and the differences of internal energy and entropy are
$U - U_0 = M_\mathrm{w} c_\mathrm{w}(T - T_0)$ and

$$S - S_0 = M_\mathrm{w} c_\mathrm{w} \ln\left(\frac{T}{T_0}\right).$$

Since the temperature in the heat storage tank is a function of position, an integration over the heat storage tank is necessary in order to compute its exergy content at a certain time:

$$W_{ex} = \iiint_{V_\mathrm{hst}} \rho_\mathrm{w} c_\mathrm{w}\left[(T - T_0) - T_0 \ln\left(\frac{T}{T_0}\right)\right] dV$$

3. Numerical Model

In order to investigate the contribution of thermal conduction in the tank wall and heat losses to the environment, simulations with three different models were carried out:

A: without wall, adiabatic to environment
B: wall included, adiabatic to environment
C: wall and isolation included (see figure 1), heat transfer to environment

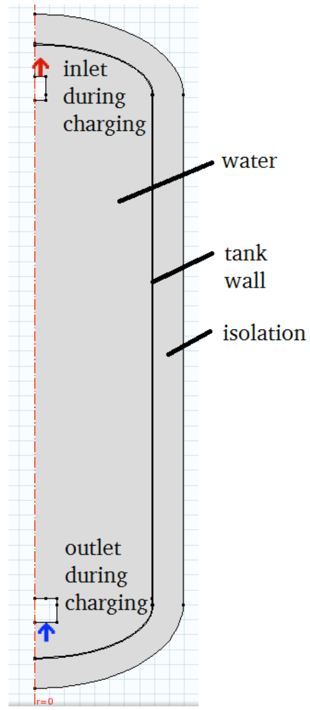

Figure 1. Model C.

A full simulation consists of three parts:

- charging (20 min / 1200 s)
- holding time (up to 10 days)
- discharging (20 min / 1200 s)

Initially, the temperature of the whole system is $T(t = 0) = 313.15\,\mathrm{K}$, and the water is at rest.

The feed pipe at the top of the tank serves as inlet during charging and as outlet during discharging. In both cases the velocity profile of a fully developed laminar flow with a mass flow rate of $\dot{M}_\mathrm{w} = 0.41\,\mathrm{kg/s}$ is prescribed. The inlet charging temperature is $T_\mathrm{in} = 364.17\,\mathrm{K}$. The internal energy difference between the hot water flowing into the tank during charging and the cold water flowing out is 105.1 GJ, whereas the exergy difference is 13.9 GJ. A constant hydrostatic pressure is prescribed at the feed pipe at the bottom of the tank. At the interface between water and inner tank wall holds the no-slip condition. The outer boundary of the model is assumed to be adiabatic for models A and B. In the case of model C, where heat loss to the environment is considered, a heat transfer coefficient $h = 2.45\,\mathrm{W/(m^2K)}$ was estimated applying the Nusselt correlation

$$\mathrm{Nu} = \left(0.825 + \frac{0.387\mathrm{Ra}^{1/6}}{\left[1 + (0.492/\mathrm{Pr})^{9/16}\right]^{8/27}} \right)^2$$

of Churchill and Chu [2].

A structured mesh consisting of quadrilateral elements is used. Performing simulations with meshes of increasing fineness, it as assured that the mesh influence on the results is negligible. A part of a coarse mesh of model C is shown in figure 2. The mesh used for the computation has the same structure but is much finer. It contains about 45000 elements with a quadratic temperature and velocity interpolation and a linear pressure interpolation.

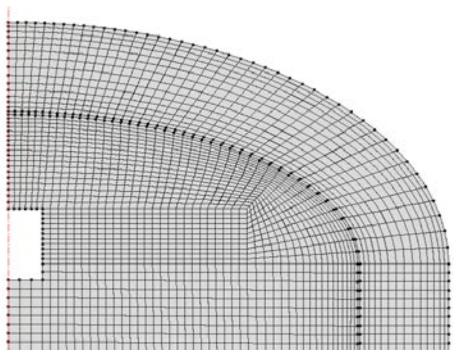

Figure 2. Mesh (detail).

4. Results

4.1 Models A, B and C

Figure 3 shows on the left side the temperature distribution for model C at the end of charging. The water possesses virtually still its initial temperature in the lower half and the higher inlet temperature in the upper half. Between these two regions there is a rather thin layer with a temperature change in axial direction. In this respect the results of models A and B look almost the same. However, in contrast to models A and B, heat transfer to the environment causes also a temperature gradient along the tank wall in the hot water region of model C.

Figure 4 compares the temperature distribution of the 3 models after a holding time of 10 days. After charging has ended, the velocity decays fast. Due to buoyancy forces there is a stable stratification with hot water in the upper and cold water in the lower half of the tank. Accordingly, a temperature equalization mainly by conduction takes place.

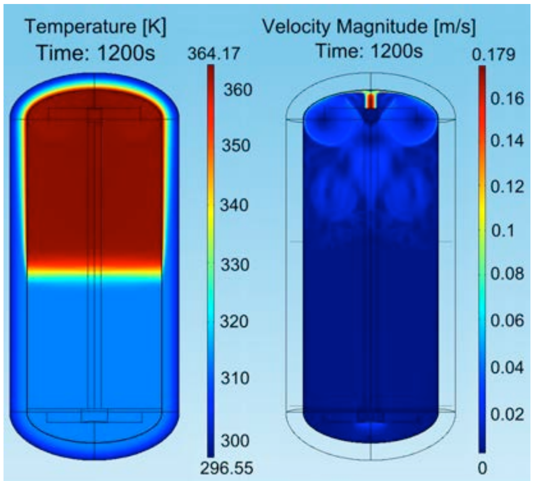

Figure 3. Temperature distribution (left) and velocity field (right) at the end of charging for model C.

The only heat transfer mechanism in model A is thermal conduction in the water. The additional heat conduction in the tank wall, which is included in model B, leads to a lower temperature at the top and a higher temperature at the bottom. Since both models A and B are adiabatic to the environment, the energy content is the same. This holds not for model C, in which the maximum water temperature is much lower after 10 days than in models A and B. Due to heat transfer to the environment the energy content is reduced during holding time.

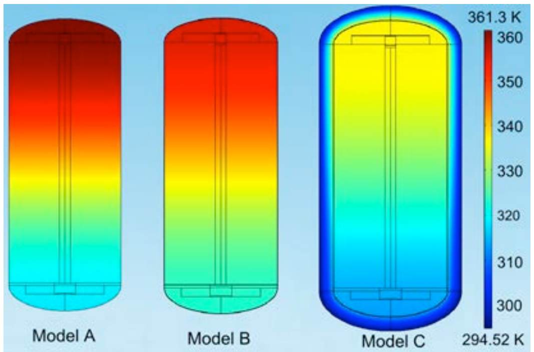

Figure 4. Temperature distribution after 10 days holding time.

Figure 5 shows the exergy loss during holding time in relation to the exergy transferred to the tank by the hot water inlet flow less the exergy of the cold water outlet flow. Thus the initial relative exergy loss values in the diagram correspond to the exergy loss during charging. Mostly because of additional heat conduction in the tank wall this value is higher for models B and C than for model A. The influence of heat conduction in the tank wall is more pronounced during holding time, where the increase of the exergy loss is higher for model B than for model A. An even stronger effect has heat transfer to the environment; the exergy loss of model C exceeds the values of the other two models by a factor of 3-4 and reaches more than 50% after 10 days.

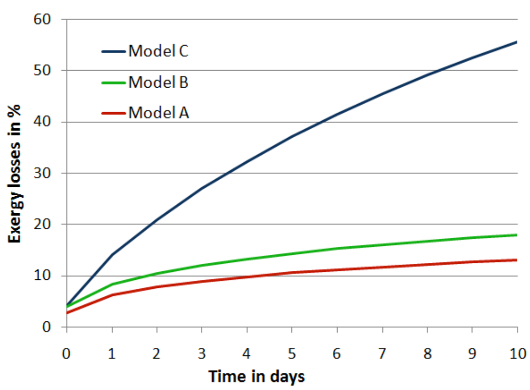

Figure 5. Exergy loss during holding time.

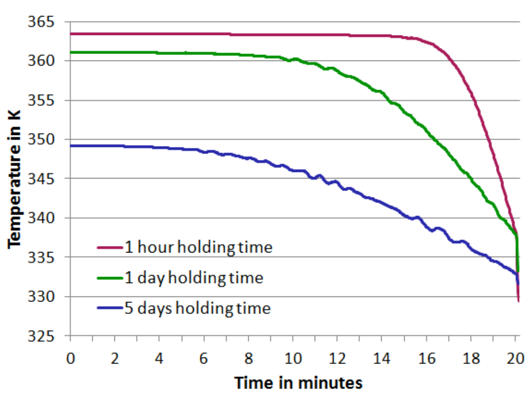

Figure 6. Water outlet temperature after different holding times for model C.

The time history of the outlet temperature at the top during discharging after different holding times in figure 6 corresponds to the exergy loss curve for model C in figure 5. The initial outlet temperature decreases with increasing holding time. The outlet temperature remains almost constant for a certain period of time. This period is longest when only 1 hour has passed after charging and shortest after a holding time of 5 days.

Figure 7 shows the temperature distribution of model C at the end of discharging after one hour holding time. Since the outlet pipe is a little bit below the top of the tank, there is some water left in the tank which has a higher temperature than the outflow temperature at the end of discharging in figure 6.

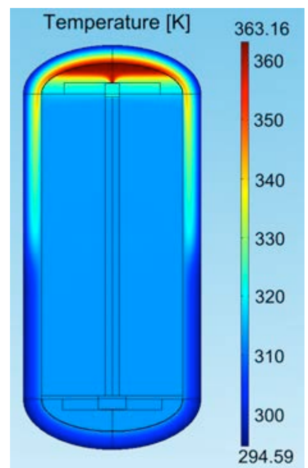

Figure 7. Temperature distribution at the end of discharging (model C).

The results show clearly that heat conduction in the tank wall and particularly heat transfer to the environment cannot be neglected. While models A and B are too optimistic, model C corresponds probably better to the actual processes in the heat storage tank.

In addition, these simulations are a good example for the advantages of an exergy analysis compared to a mere energy balance. Since both models A and B are adiabatic to the environment, their energy content is the same and stays constant during holding time. However, because of additional conduction in the tank wall, the temperature equalization advances faster in model B. Although the energy content after a certain holding time is the same for both models, the energy stored in model A is more “valuable”. This is expressed by the higher exergy value of model A.

4.2 Parameter Variations

Reducing the inlet mass flow rate leads to a smaller inlet velocity and thus reduces the mixing of hot and cold water during charging. The results show that the exergy of the storage tank after charging increases slightly with decreasing mass flow rate. However, the effect is marginal, because the nominal value of the mass flow rate leads already to a fairly good temperature stratification after charging.

The inlet temperature was varied by ±5 K for model A. There is nearly no influence on the exergy loss during charging. With increasing inlet temperature the exergy loss increases during holding time. An inlet temperature change of 5 K causes an exergy loss change of less than 1 percentage point. The influence of the inlet temperature on the exergy loss is probably larger for model C, because also the heat loss to the environment increases with increasing inlet temperature.

The water inlet geometry influences the inlet velocity. Since the exergy loss caused by mixing during charging is small in comparison to the overall exergy loss during holding time, the possible improvement via a modification of the inlet geometry is small.

5. Conclusions

An exergy analysis is an appropriate method in order to evaluate the performance of a water heat storage tank. Exergy losses are mainly caused by:

- mixing during charging
- heat conduction in the tank wall
- heat conduction in water
- heat losses to the environment

Mixing is not dominating and only marginal improvements seem to be possible. Heat conduction in the tank wall could be reduced by an additional inside insulation. However, the effect of conduction in the tank wall is smaller than the effect of conduction in water, whereas the latter can hardly be avoided. This makes such a technically challenging additional insulation questionable.

Heat loss to the environment seems to be the most important reason for exergy loss. Depending on the envisaged holding time, a better insulation could improve the overall performance of the heat storage tank.

6. Nomenclature

Variable	Unit	
c	J / (kg K)	specific heat capacity
$\mathbf{g}$	m/s^2	acceleration of gravity vector
h	W/(m^2 K)	heat transfer coefficient
k	W / (m K)	thermal conductivity
M	kg	mass
$\dot{M}$	kg/s	mass flow rate
p	Pa	pressure
Pr	-	Prandtl number
r	m	radial coordinate
Ra	-	Rayleigh number
S	J/K	entropy
T	K	temperature
t	s	time
$\mathbf{u}$	m / s	velocity vector
u	m / s	velocity
U	J	internal energy
V	m^3	volume
W_{ex}	J	exergy
z	m	axial coordinate
β	1/K	thermal expansion coefficient
μ	kg / (m s)	dynamic viscosity
θ	rad	angular coordinate
ρ	kg / m^3	density

Indices	
0	environment state
hst	heat storage tank
in	inlet
out	outlet
r, θ, z	component in r-/θ-/z-direction
w	water

6. References

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