

## PERFORMANCE EVALUATION OF CHP WITH HEAT STORAGE IN BUILDINGS

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### ABSTRACT

Combined heat and power (CHP) production gains more and more attention. Offices and public buildings often have a large thermal power demand in combination with a fairly large electrical power demand. On the other hand they are seldom occupied by night and in weekends, reducing the actual operational time of the heating system. This in turn brings down the financial benefits of investing in CHP. A second problem is that electrical and thermal demands are often shifted in time. The running time of the engine is again limited this way, as it is often not allowed to deliver electricity to the power grid. A possible solution is using heat storage. This way the CHP-engine can run when the electricity demand is high. In the paper a simulation model of CHP with gas engine and heat storage by means of a hot water vessel is developed. The model is validated through experiments on an engine and a vessel. This model is used to analyze the design, control and performance of cogeneration plants. It is shown that storage is marginal beneficial and the design has to be done with great care.

### INTRODUCTION

About 30 to 40% of the world energy demand is used in buildings in order to provide comfort and building operation [1]. In a building electricity is bought from the public grid and heat is produced with a gas boiler. CHP (Combined Heat and Power), also known as cogeneration, combines these two: generating electricity with a thermal engine and using the heat in the exhaust and cooling system to produce hot water. In large electricity plants heat is usually transferred to the environment while it could be used for heating purposes. That is why a cogeneration plant uses less fuel (primary energy) to produce the same total amount of energy [2]. Cogeneration in buildings can contribute to large energy savings.

However, the use of CHP in buildings is not yet widespread, because of a number of specific problems. The investment is very high due to the expensive gas engine. The unit must run as much as possible in order to obtain a good return on investment

(IRR). Electricity prices are often not high enough during quiet hours, therefore electricity to the grid is less profitable. A cogeneration engine can only produce heat and electricity in a fixed proportion, assuming that no heat is wasted to the atmosphere. This is a problem since daily heat and electricity demand profiles are usually out of phase. The normal control strategy is selling excess electricity to the grid while covering the heat demand. This is however often not profitable. Using heat storage could be a promising technology [3].

There is still a design problem concerning the annual heat demand. This profile can be very different from year to year, which complicates the sizing of the engine and the thermal storage facility. Thus reliable simulations are needed.

A last problem is posed by the integration of the CHP unit in the heating facility, as also mentioned in [3]. A poor control of the boilers can compromise the reliable functioning of the engine. The thermal inertia of the system must be considered; otherwise the boilers will produce too much heat. This raises the temperature of the water in the heating installation and causes the engine to shut down because it does not have sufficient cooling.

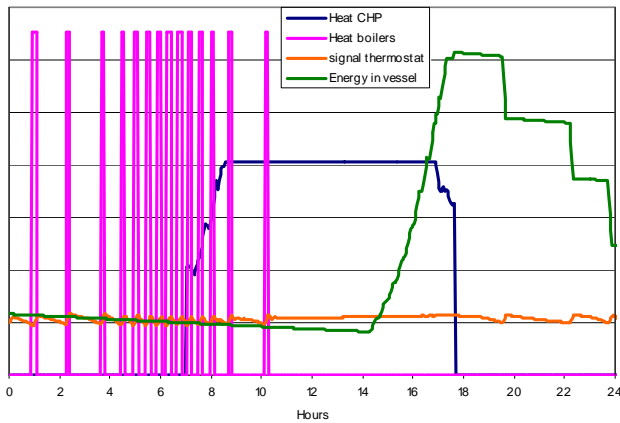
Using heat storage can solve some of the above problems. The shifted profiles pose fewer problems since part of the heat can be stored and released at another time. Compared to a CHP-engine controlled by the thermal demand, the installation is more feasible because of the fact that no electricity is sold to the grid and the engine can follow the electricity profile, which is a more reliable control strategy. Compared to a CHP-engine controlled by the electrical demand and without heat storage, the annual runtime rises because the unit can deliver electricity in periods when the heat demand during the normal hours is less. The excess heat is then used during the quiet hours.

### DYNAMIC SIMULATION IN TRNSYS

Usually mainly static and simplified methods are used to evaluate the feasibility and performance of CHP facilities. A lot of CHP design studies do not take into account the dynamic interaction of the engine with the heating system. Testing a

control strategy becomes thus impossible. In combination with heat storage, this problem is even more significant since the performance of the vessel is very much influenced by this interaction.

A dynamic simulation was built up in TRNSYS [4]. TRNSYS allows designers to study the effects of design changes, control strategies and external influences in a very detailed way (with a resolution of a few minutes or less, as shown in Figure 1).



**Figure 1** : Result of a 24 hr simulation with heat storage

TRNSYS has a modular, graphical approach allowing for quick construction of entire systems out of a few model components, of which mostly HVAC and solar applications. It also allows programming new models or adapting existing ones. There is a special series of models included which simulate the electric and thermal energy demand of a building, using extended climate data. The engine model originally included in TRNSYS was proven to be unsuitable for the study and was adapted using measurements. A model for stratified water vessels was available in TRNSYS but had only been tested for small vessels (less than 1m<sup>3</sup>). To ensure reliable simulation results, a complete test facility was built with a vessel of 2m<sup>3</sup>.

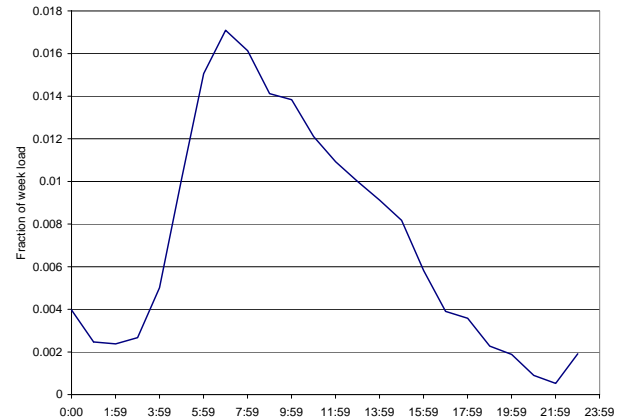
## MODELLING ASUMPTIONS AND VALIDATION

### Thermal and electrical demand modelling

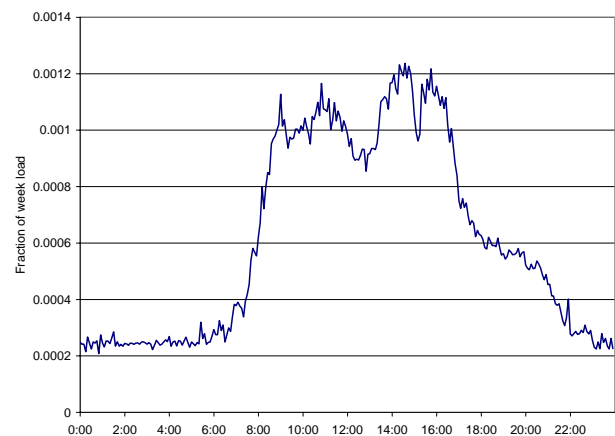
The dimensioning of a CHP starts with the determination of the thermal en electrical demand profile, i.e. the thermal and electrical demand of the building as a function of time. A load duration curve can then be plotted and the capacity of the CHP can be determined. This technique can no longer be used if a storage vessel is used.

In order to do the dynamic simulations both a thermal and an electrical demand profile were constructed out of data obtained from the Provincial school for Industrial Higher education (PIH) [5], in Kortrijk, Belgium. This school has a fairly large building stock and is a nice example for showing the problem of dephased heat and electricity demand. Secondly it is not profitable for the school to sell back electricity to the grid. Figures 2 and 3 give the daily demand profiles for respectively heat and electrical power demand scaled to the daily total load.

The sizing is based on the thermal heat demand of the building and secondly on the electrical profile. The heat profile however is dependent of the ambient temperature, which is not the case for process heat in industrial applications, where the dimensioning can be more easily done with measurements. For buildings, each year is different and using the measurements of one year can lead to an oversized or undersized engine. Therefore a correction is needed according to degree days in a standard year. For new buildings, standard profiles can be used, but to be more correct, a simulation can be done.



**Figure 2**: Example of the daily heat demand



**Figure 3**: Example of the daily electrical demand

The thermal load profile was normalised using monthly data from the gas consumption during the years 2000 to 2004. From the measurements of the average day temperatures and the calculation of the degree days per month, a correlation was constructed between the monthly degree days and the heat consumption. This linear correlation is then used to produce a standard year according to degree days per month of the test reference year for Belgium [6]. As a comparison the year 2000 and the test reference year are compared in Figure 4. These data are linearly extrapolated to data per week. The week profile is then used to determine the hourly profile of the building (Figure 2) in combination with imposing a typical profile for a day.

The size of the CHP gas engine was determined by converting the averaged yearly electrical demand profile for the

year 2000 to 2004 to the load duration curve. A capacity of 288 kWe was chosen giving the maximal power production. If the installation is sized on the heat, a larger capacity could be installed. Electricity has to be put on the grid in that case and the economics are not good.

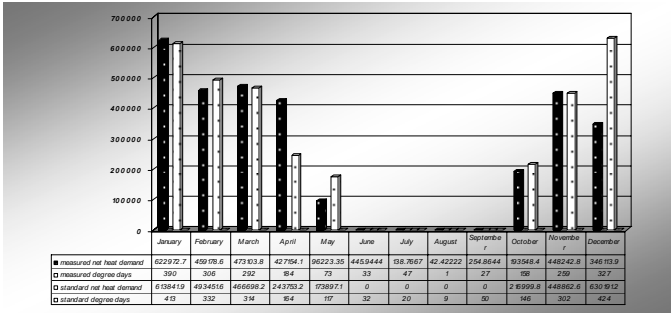


Figure 4: Yearly thermal demand of the PIH

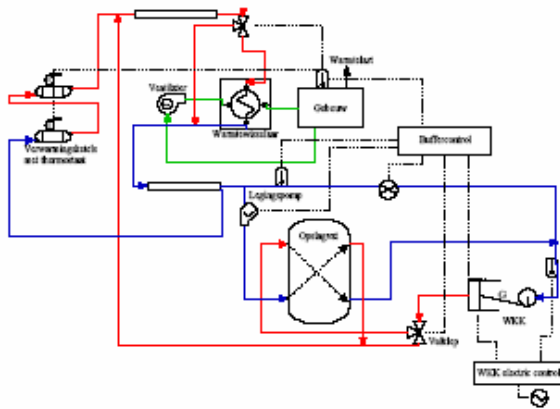


Figure 5: CHP unit and storage vessel in the building heating system

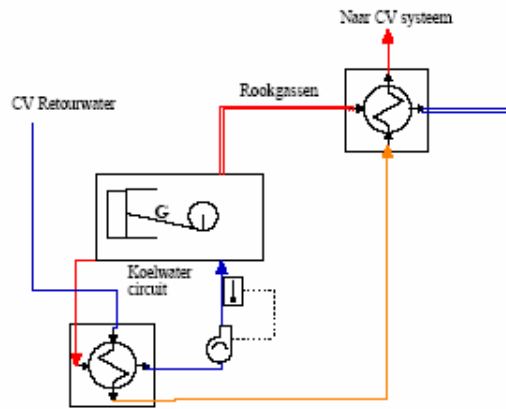


Figure 6: Engine with exhaust gas and cooling water coolers  
Gas Engine Model

As most building integrated CHP units use a reciprocating gas engine a TRNSYS model was designed and programmed for this gas engine. The model can be used for different types of engines and different capacities. Of course, different parameters are set for each engine. The model was verified on a CHP-unit of 288 kWe at PIH.

A CHP unit is mostly sized on heat demand, but the control of a CHP is often done following the electrical demand, especially if electricity is not sold back to the grid. Therefore, the control module of the model is based on electrical demand. Since the electrical reaction of the engine is very quick (ms), there is no model for transient electrical behaviour.

As a second factor the heat demand is taken into account. If there is no possibility of putting the heat in either the building or the heat buffer, and if therefore the temperature of the cooling water (TCWI) in the engine is too high even after the reduction of the engine capacity, the engine will shut down. The engine is placed parallel with the heat boilers as shown in Figure 5. The vessel is only being filled when the CHP is running, so that the heat from the boilers does not pass the vessel.

The necessary input parameters of the engine are:

- Fuel use in relation to the electricity production (without transients) (i.e. electrical efficiency).
- Heat rejected by the engine at different sources (exhaust gases, engine block and oil cooling) in function of the electrical capacity and the transient time constant to go from one operating point ( $Q_m$ ) to another  $Q_{new}$  as given in eq. (1). The cooling of the turbo is not modelled separately because in a lot of cases, this heat is not used, because of the low temperatures.

$$\tau \frac{dQ_m}{dt} = Q_{new} - Q_m \quad (1)$$

The time constant for the engine at PIH is 8 minutes. If however the engine is starting up from a cold state or shutting down, then this time constant is about 1 hour, except for the exhaust gases where this factor is 5 minutes. If the engine is shut down, the mass flow rate for cooling is set to 0 kg/s and the output temperature of the water is coming down to the outside temperature according to the transient behaviour of the engine. This is necessary to include warm start up of the engine.

The cooling water outlet temperature is set to 90°C. The cooling water flow rate is calculated with

$$m_{cw} = \frac{Q_{cw}}{c_{pw}(T_{out} - T_{in})} \quad (2).$$

The cooling water inlet temperature is given as input from the water circuit. The heat release  $Q_{cw}$  is calculated for each time step with equation (1) with the correct time constant.

The gas flow rate of the exhaust gasses ( $m_{ex}$ ) is passed on by the model in relation to the power output. If the heat contained in the exhaust gasses is  $Q_{ex}$  (assuming 0°C as the reference temperature) the exhaust gas temperature is given by :

$$T_{ex} = \frac{Q_{ex}}{m_{ex} c_{pg,ex}} \quad (3)$$

Again, the heat release  $Q_{ex}$  is calculated for each time step with equation (1) with the correct time constant.

The gas engine is integrated into the water circuit using two heat exchangers, an exhaust gas cooler and cooling water

cooler, as shown in Figure 6. These heat exchangers are standard components in TRNSYS.

In Figure 7 the results are given for the simulation and they are compared to measured data. The simulation results are very close to the measurements.

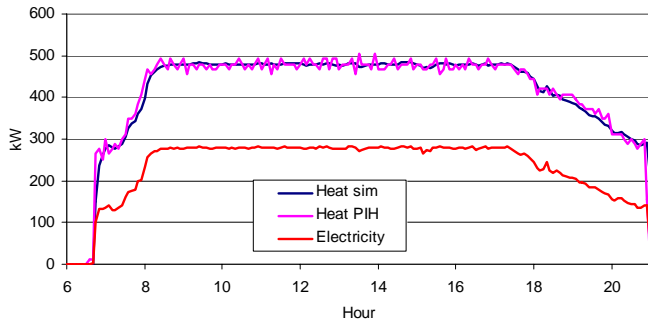


Figure 7: Engine model validation

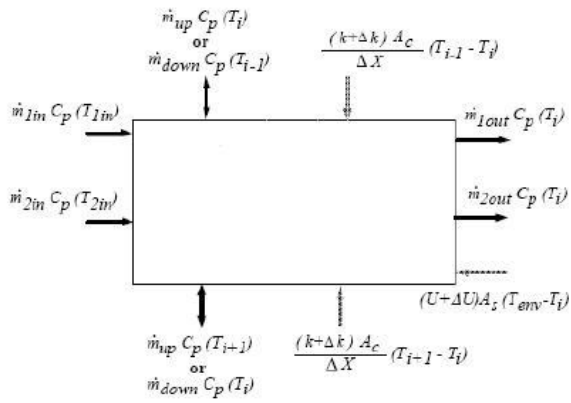


Figure 8: Energy balance of a thermal layer

### Storage vessel model

A lot of thermal storage methods are available but, in practice, only the storage of hot water in vessels is interesting for the current application. The vessel is typically working with a pressure of 3 bar and a temperature of maximum 90°C. The vessels use the effect of stratification. This means that hot water is extracted at the top and cold water is supplied at the bottom, generating temperature layers since hot water has lower density than cold water. This provides maximum water temperature to the heating system.

The model is based on the mass flow rate, the temperature of incoming and outgoing water and the position of the incoming water. The vessel is divided into thermal layers with each an initial temperature. This allows for taking into account the effects of the thermal stratification. For each thermal layer, an energy balance (eq. (4)) is written, taking into account energy losses through temperature drop, conduction to connecting layers, mass flows to connecting layers and heat losses to the environment. In Figure 8 the control volume is given for each thermal layer in the vessel. The loss coefficient to the environment can be determined analytical, but is best determined by experimental measurements.

$$M_i c_p \frac{dT_i}{dt} = \frac{\lambda A}{\Delta x_{i+1 \rightarrow i}} (T_{i+1} - T_i) + \frac{\lambda A}{\Delta x_{i \rightarrow i-1}} (T_i - T_{i-1}) + U_i A_{wwi} (T_{env} - T_i) + \dot{m}_{down} c_p (T_{i-1} - T_i) - \dot{m}_{up} c_p (T_i + T_{i+1}) + \dot{m}_{1out} c_p T_{1in} + \dot{m}_{2in} c_p T_{2in} - \dot{m}_{2out} c_p T_{2out} \quad (4)$$

The model is verified with experimental data for a vessel with a capacity of 2000l, a height of 2.1m and a diameter of 1.1 m. The expansion vessel is 300l. With a rise of 50°C in average temperature the volume expands with 3.61%, which is captured by the expansion vessel. The experimental setting is given in figure 9.

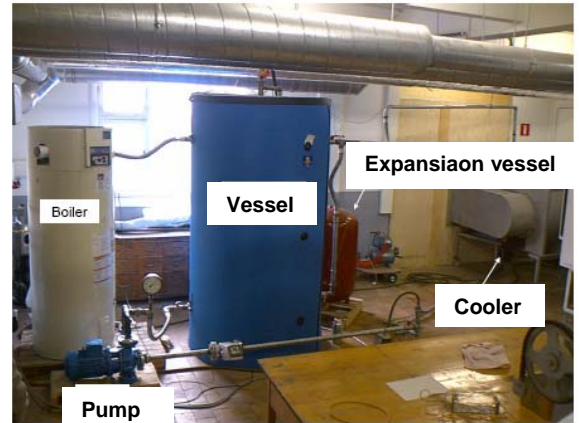
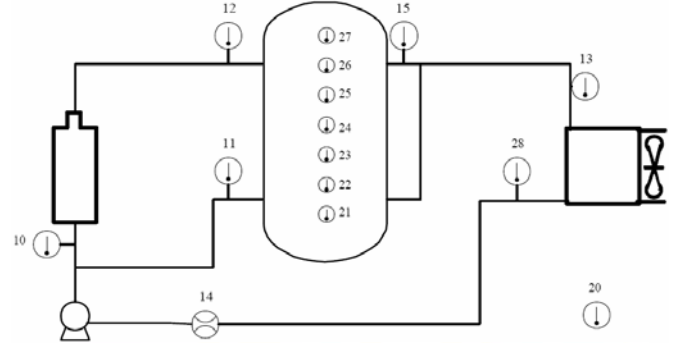
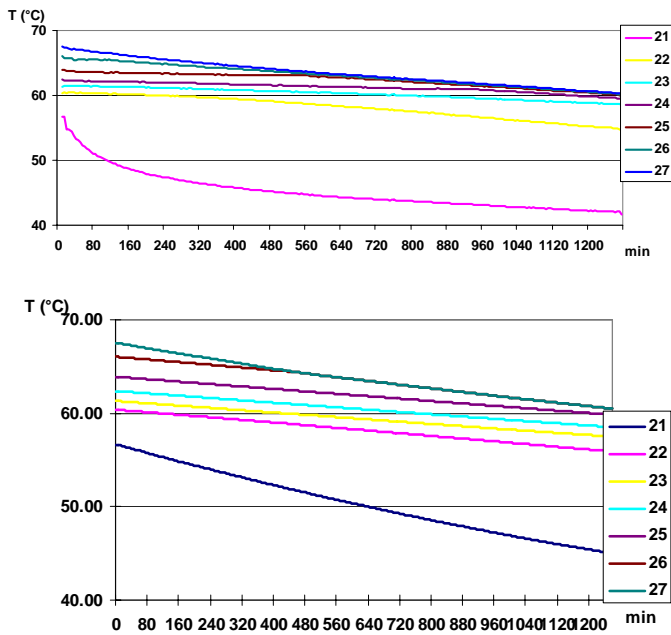


Figure 9: Experimental setup for the vessel

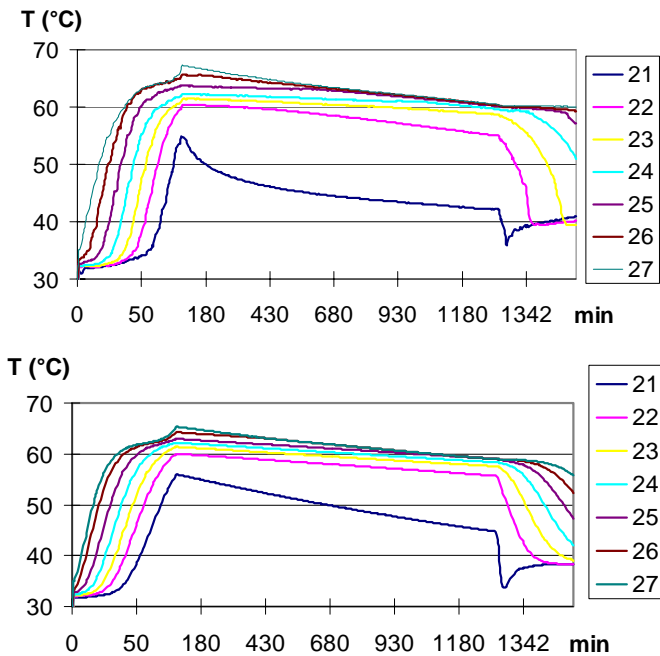
The loss coefficient to the environment is determined experimental by doing a static measurement. The buffer is filled with hot water until the average temperature is 50°C. The temperature is measured in 7 places (TC21-TC27) to measure 7 thermal layers. All valves are then closed and the temperatures are measured in the layers every 10 minutes (Figure 10 TOP) during 12 to 24 hours. The bottom of the vessel is not insulated, causing the lower temperature to go down more quickly. It is assumed that the loss coefficient of the lowest layer is twice as high as of the other layers. By using the energy balance of every thermal layer, a loss coefficient for this buffer is determined from the measurements to be 1.7 W/(m<sup>2</sup>K).

The loss coefficient is being implemented in the model. A value for the conductivity of water is set on 0,64 W/mK and the ambient temperature is measured. The maximum deviation of the simulation to the experimental measurements (Figure 10) is 0.3°C, except for the lowest thermal layer. The decline is different, but the end result is again correct.



**Figure 10:** Temperature drop in the vessel due to heat loss (TOP : measurement BOTTOM : simulation)

The dynamic reaction of the vessel is also verified. Water is added to the vessel at 70°C with a 50kW gas boiler. For cooling of the vessel, a cooling capacity of 15 kW is present with an air cooler, cooling the water of about 70°C to about 50°C. In Figure 11 complete cycle is given. The first 90 minutes the vessel is heated. During about 21 hours, the heat is kept in the buffer before heat is extracted form the vessel after about 1350 minutes. The temperature dip is a consequence of the cold water in the pipelines. The same cycle is simulated as given in Figure 14 BOTTOM. A good agreement is reached.



**Figure 11:** Temperature drop in the vessel due to heat loss (TOP : measurement BOTTOM : simulation)

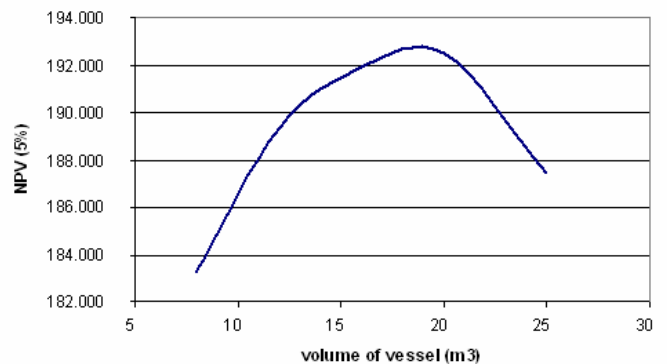
## PERFORMANCE EVALUATION AND ECONOMICS

### Sizing of the installation

As a case study the installation of the PIH was chosen. The engine size was set to 288 kWe. For this engine a suitable vessel had to be selected. For the sizing of the vessel, not only the volume matters, but also the ratio between the diameter of the cylinder vessel, which determines the loss over the surface of the vessel and the height that determines the stratification. Between these effects, there is an optimum, depending on the isolation materials of the vessel. The optimum is for a ratio of ½. With this ratio the surface of the casing is equal to the surface of the upper and lower circle and the total surface of the vessel is minimal.

The capacity of the storage vessel has to be determined with a simulation, as the vessel capacity is influenced by the capacity of the CHP-installation itself. The sizing of a CHP is done by the thermal load duration curve. If however electricity has to be put on the grid in this situation, a smaller engine is installed. This smaller engine will have a lower thermal capacity and can have more running hours, if of course the electrical profile allows this. The installation of a buffer means that an engine can have more running hours, certainly if the thermal capacity is lower than the optimum capacity. Of course there can be some variation in possible capacities, depending on the load duration curves. Simulations are in order here, to make a good fit of an engine and a buffer capacity. As a target for optimisation the minimal total costs can be set.

In figure 2, a simulation is done for the engine at PIH, working with different buffer volumes. The time step for the dynamic simulation is 2.5 minutes, a compromise between detailed simulation and the calculating time. A larger buffer means more running hours, but also a larger investment and more thermal losses. From Figure 12 the optimal NPV (Net Present Value) is found for a vessel of 20 m³.



**Figure 12 :** NPV (5%) for different buffer volumes

### Energy performance

The PIH average energy use profiles as derived in the previous sections was used to analyse the energy performance of the CHP with and without vessel and a conventional heating system. A reference electrical efficiency of the electricity from the net is taken on 40%. The boiler efficiency is 90% for separated heat production. A large saving in primary energy for the CHP in comparison with non-CHP is shown in Figure 13.

Installing a buffer with the CHP gives similar primary energy use as without a buffer vessel. Only during spring and autumn, when the heat demand of the building is lower, an energy use reduction is seen. The main reason for this is the longer running hours of the gas engine with a buffer in these periods.

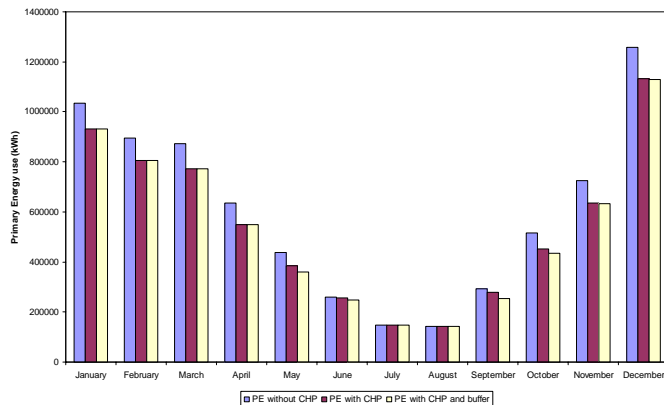


Figure 13 : Primary Energy use for different scenarios

		CHP	CHP+buffer
Capacity	kWe	288	288
electrical efficiency CHP	--	31.69%	31.61%
thermal efficiency CHP	--	56.70%	56.05%
eq. Full load hours	h/year	2,401	2,697
Buffer	m <sup>3</sup>		20
starts/stops		261	261
buffer loss			0.69%
Investment	€	270,345	297,916
O&M	€/y	14,605	15,858
fuel cost	€/y	57,237	64,436
Avoided cost boiler	€/y	37,187	41,384
electrical gains	€/y	73,092	82,092
Subsidaries	€/y	17,458	18,871
Profits	€/y	55,895	62,053
Payback Time	year	5.96	5.93
IRR	--	10.92%	11.07%
NPV	€	80,999	91,644

Table 1 : Economics in a Reference year

Comparison of the economics of CHP with and without buffer is done in Table 1. In Table 2 economic results are given for a cold (1996) and a hotter year (2002).

The results from Table 1 show that the profits for a CHP with buffer are slightly higher than for a simple CHP. This is mainly due to the higher number of running hours. The extra investments of the buffer vessel are compensated for, resulting in a higher IRR and NPV for the CHP with buffer. The difference though is limited.

In a cold year the relative gain becomes bigger. This is mainly caused by the further growth of number of running hours. In a warm year the profit of installing a buffer vessel is even lost. Using a buffer is thus only marginally profitable and profits are strongly influenced by heat demand.

		1996		2002	
		CHP	CHP+buffer	CHP	CHP+buffer
eq. Full load hours	hr/y	2,589	2,784	2,294	2,647
Payback Time	year	5.58	5.74	6.21	6.04
IRR	--	12.62%	11.88%	9.91%	10.59%
NPV	€	105,666	104,537	66,709	84,028

Table 2 : Comparison between cold and hot year

## CONCLUSION

For a new CHP-installation, or when retrofitting an existing CHP, a buffer can certainly be an advantage for those installations who are electrically controlled and where the electrical profile is mostly shifted from the thermal profile.

Looking at primary energy use CHP with buffer gives extra possibilities for energy use reduction, due to the extra running hours during low heat demand periods. This causes an extra amount of running hours of about 12%.

From the economic point of view installing a buffer gives a small profit of about 11% on a yearly basis. Climatic conditions though will influence the profit significantly, where colder years are more profitable.

## NOMENCLATURE

A	[m <sup>2</sup> ]	Area
C <sub>p</sub>	[J/KgK]	Heat capacity
m <sub>0</sub>	[kg/s]	Mass flow rate
M	[kg]	Mass
λ	[W/mK]	Thermal conductivity
Q	[W]	Heat
T	[s]	Time
τ	[°C]	Temperature
τ	[-]	Time constant
U	[W/m <sup>2</sup> K]	Convection coefficient
X	[m]	Distance

## ACKNOWLEDGEMENT

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