In-situ Measuring Technique for Heat Capacity Determination of Steam-gas Mixtures at High Enthalpy Geothermal Sites

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ABSTRACT

High enthalpy geothermal sites provide hot steam, which can directly be used in flash power plants for electricity generation. In order to dimension the steam turbine it is essential to know the steam enthalpy expressed in isobaric heat capacity and Joule Thomson coefficient. Geothermal steam is a mixture of both water vapor and non-condensable gases like carbon dioxide, methane and other volatile light hydrocarbons as well as hydrogen sulphide. This gas composition determines the thermodynamic properties of the steam flow. Usually at operating power plant pressure, the gas/steam mixture is far from being an ideal mixture, which makes it difficult to calculate the fluid enthalpy. Here in-situ measurements could help to close the gap of knowledge concerning the influence of the volatiles. For thermal water, heat capacity is already measured by in-situ measuring technique of KIT. In a further step, it is considered to adopt this technique to compressible fluids and to test it on-line at high enthalpy geothermal sites.

1. INTRODUCTION

Direct use of hot steam in steam cycles is performed in many places in the world where geothermal fluid is available at high pressures and temperatures above 200°C. For instance countries like New Zealand, Turkey, Japan, Central America, Russia, Italy as described in Moon and Zarrouk (2012), Iceland, Ólafsson (2014), Indonesia, Pambudi et al. (2015) and Rudiyantoa et al. (2017), Iran, Jalilinasrabady et al. (2012) and Kenya, Cheruiyot (2016) are aware of high enthalpy geothermal resources using dry steam, single flash or double flash power plants. In all cases, the power plant turbine is driven by steam which is already available at the wellhead or which is produced by flashing of hot thermal water. The overall efficiency of the cycle is influenced by many parameters like mass flow, steam temperature and pressure, gas load, ambient conditions and power plant design. The electric power, produced at the turbine can be calculated by means of the steam enthalpies at the turbine's in- and outlet taking account of the turbine efficiency, Strauß (1997). Steam enthalpies depend on pressure and temperature and can be determined by steam tables. In many cases, non-condensable gases are accompanying the steam flow thus enthalpy determination via steam tables is no longer sufficient. In general, enthalpy of a gaseous fluid can be written as:

$$dH = \left.\frac{\partial H}{\partial T}\right|_p \cdot dT + \left.\frac{\partial H}{\partial p}\right|_T \cdot dp \qquad (1)$$

Whereas H indicates the enthalpy of the mixture as function of temperature T and pressure p. The first derivative corresponds to the isobaric heat capacity whereas the second derivative expresses the Joule-Thomson effect. Concerning liquids, the latter can be ignored but for compressible media, pressure dependence of the enthalpy has to be considered. Thus, both terms in equation (1) become important. Within the scope of this work, a measuring technique is presented which allows for both, determination of isobaric heat capacity and Joule-Thomson coefficient as function of temperature and pressure for steam/gas mixtures as being present in high enthalpy geothermal sites. These data allow for performing enthalpy balance concerning power plant components based on site-specific fluid conditions.

2. MEASURING TECHNIQUE

The here presented gas/steam calorimeter works in through-flow conditions and is based on the design of the flow calorimeter for liquids as reported in Schröder et al. (2015).
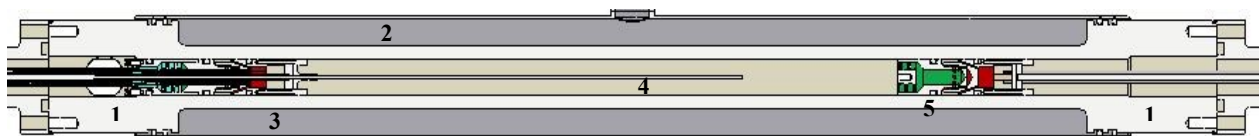

Figure 1: Sectional drawing of the calorimeter and it`s components: 1 PEEK tube, 2 vacuum chamber, 3 static mixer including three thermocouples (inlet probe head), 4 heating rod, 5 static mixture including four thermocouples (outlet probe head). Fluid flow direction is perpendicular to the drawing surface.

This flow calorimeter consists of a PEEK tube (1) with an inner diameter of 16 mm and 735 mm in length. The wall thickness amounts to 6 mm. At each tube end, a probe head (3) and (5) is installed which contains a static mixture and several radially distributed thermocouples. Additionally the entrance probe head (3) is equipped with an electric heating rod (4), 2 mm in diameter and 20 mm in heated length. For insulating the calorimeter, a vacuum chamber (2) encloses the PEEK tube. The fluid flows radially into the tube, passes the inlet probe head while its temperature is recorded. After the fluid has overflowed the electric heating rod,

its temperature is detected by the outlet probe head (5) before it leaves the tube radially. Mass flow of the fluid is detected downstream of the calorimeter by a Coriolis mass flow meter (Coriflow, Wagner Company). In Figure 1 the sectional drawing of the calorimeter is shown, as used for incompressible fluids. Heat capacity of incompressible fluids is determined by following measuring procedure. In- and outlet temperatures are recorded for approx. 20 minutes to ensure temperature uniformity (< 0.05 K) within the calorimeter. Then the electric heater is switched on. Resulting from this, the fluid outlet temperature rises. A stationary state is reached after approx. 15 minutes being characterized by a constant temperature difference between the inlet and the outlet probe head. Finally, the heating rod is switched off and temperature uniformity is awaited before the next heating period with different heating power is started. The specific, isobaric heat capacity is calculated as average of five heating periods according to equation (2).

$$\overline{c_P} = \frac{\overline{\dot{Q}}}{\dot{m} \cdot (\Delta T_{HP} - \Delta T_0 + \Delta T_{in})} \tag{2}$$

$\overline{\dot{Q}}$, $\overline{\dot{m}}$ and $\overline{\Delta T}_{HP}$ being heat flux in [J/s], mass flux in [kg/s] and stationary temperature difference averaged over the heating time at which $\overline{\Delta T}_{HP}$ remains constant. $\overline{\Delta T}_0$ is time averaged over the last 40 seconds of the non-heated period and $\overline{\Delta T}_{in}$ expresses the rise of the inlet temperatures as a result of the small heat impact from the cold end of the heating rod. The in- and outlet temperatures itself are calculated by the arithmetic mean of the three inlet, resp. the four outlet thermocouple signals. Finally, heat capacity is calculated as arithmetic mean of the five $\overline{c_P}$ values indicating the five heating periods. The heating power itself is calculated by time averaged data of DC voltage (U) and currant (I) during the heating periods according to Equation (3).

$$\dot{Q} = 0.96 \cdot U \cdot I \tag{3}$$

The dimensions of this flow calorimeter is suitable for liquid phases as it provides a good heat transfer from the internal electric heating rod to the fluid. Concerning heat capacity determination of steam/gas mixtures the arrangement of the calorimeter is not appropriate due to the fact, that flow velocity will become too low in order to achieve a remarkable heat transfer between the heating rod and the gaseous fluid. This was tested by substituting water flow by airflow. No temperature difference between the inlet and outlet probe head could be detected when the heating rod was switched on. However, it could be realized that the mixing units work well and that no radial temperature distribution is detectable at the probe heads. The vacuum insulation was quite appropriate for liquids and it was assumed to work as well when using gaseous fluids. Against this background, a calorimeter concept, which serves for both liquids and gases was favored and by using additional installations this objective might be reached.

In order to enhance the gas velocity and to achieve a better heat transfer the annulus between the tube wall and the heating rod has to be lowered. Further thermal inertia shouldn't be enhanced much by implementing a lot of material into the calorimeter which has to be heated by the hot gas/steam flow. The design of the additional flow channel is shown in Figure 2. This channel is integrated into the peek tube, which is shown in Figure 1 and fixed to the static mixer of the entrance probe head.

Figure 2: Sectional drawing of additional flow channel which is installed in the PEEK tube (Fig. 1) and which is surrounding the heating rod (3). Arrows mark the flow direction starting from the mixer (1). The position of the thermocouples is indicated by (2).

The new configuration allows for enhancement of the flow velocity and thus ensures an increase in heat transfer from the heating rod to the gas/steam. When running experiments with liquids this installation can be removed easily for using the calorimeter in the configuration shown in Figure 1.

3. RESULTS

3.1 Test measurements

First tests were run in order to check whether heat transfer is sufficient for obtaining a significant temperature difference between the inlet and the outlet thermocouples. For this, experiments with pressurized air were performed. The mass flow meter which is installed downstream of the calorimeter can operate with both liquids and gases. First results are shown in Figure 3 and Figure 4. Mass flux amounted to 12.3 g/min.
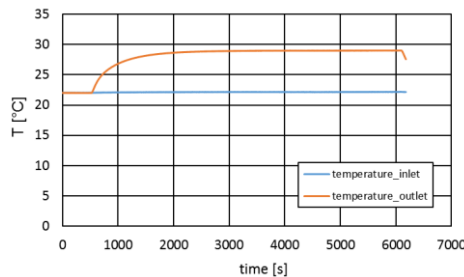

Figure 3: Mean inlet and mean outlet temperatures inside of the calorimeter when the heating rod is switched on and off.
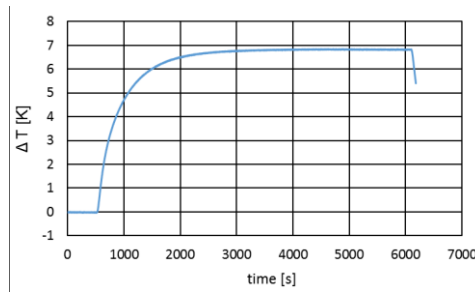
Figure 4: Temperature difference during the heating period.

Figures 3 and 4 indicate that heat transfer from the heating rod to the air is good and that a significant temperature difference can be detected. Prior to the turn on of the electric heating temperature equality was reached between in inlet and outlet.

3.2 Calibration methods

Because of long and narrow flow paths within the calorimeter, pressure drop was quite high and varied between 0.05 MPa and 0.14 MPa when mass flux was adjusted between 5.5 g/min and 17.3 g/min. On the basis of this finding the effect of pressure reduction within the calorimeter has to be taken into account because the compressibility of the media has an impact on the energy balance which is expressed by the Joule-Thomson effect. For this Equation (2) cannot be used is this form because pressure effect on the energy balance is eliminated. Instead of Equation (2) energy balance for compressible fluids has to be formulated which results in Equation (4)

$$\Delta \dot{Q} = \dot{m} \cdot c_P \cdot (\Delta T - \mu_{JT} \cdot \Delta p) \tag{4}$$

whereas μ_{JT} is the Joule Thomson coefficient which indicates cooling ($\mu_{JT} < 0$) or heating effects ($\mu_{JT} > 0$) in real gases during expansion. The real gas effect plays an important role in gas liquefaction processes and cannot be neglected here due to the high pressure-drop in the calorimeter. This makes calibration of the flow calorimeter necessary.

Two ways of calibration methods come into account:

1. Temperature difference as function of pressure drop is recorded at turned off heating rod. Depending on the gas temperature and pressure, Joule Thomson coefficient can be calculated for the desired temperature / pressure span.

2. Heating rod is turned on and heating power is adjusted just as much as is necessary to eliminate the temperature difference. Heat flux is then plotted against the pressure-drop whereas the gradient indicates the unknown constant $(-\dot{m} \cdot c_P \cdot \mu_{JT})$. This procedure has to be made as well for different gas temperatures and pressures.

Both calibration methods require perfect insulation of the calorimeter tube. Further, heat resistance of the calorimeter material has to be considered when determining the respective time interval for calibration because the material has to be in thermal equilibrium with the gas. Both difficulties are challenging and at this point of time these problems are not solved satisfactorily in order to get reliable results. Further test measurements are necessary for the determination of the respective time interval in which the Joule Thomson effect can be identified without the influence of heat loss and thermal inertia. Preliminary results are shown in Figure 5. The measured data are corrected taking the Joule Thomson coefficient from literature, Lemmon et al. (2010) into account at respective gas temperature and pressure.
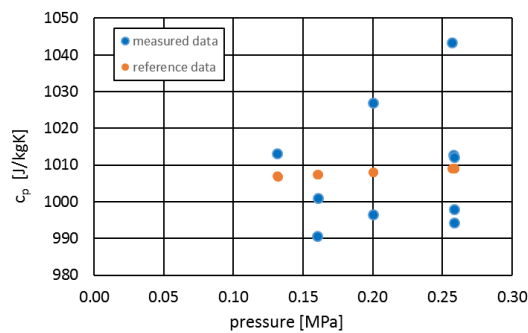

Figure 5: Isobaric heat capacity of air at 22°C for different entrance pressures. Reference Data is taken from Lemmon et al. (2010).

4. CONCLUSION

The instrument for measuring isobaric heat capacity of liquids under through-flow conditions is modified in order to be suitable for steam/gases requirements. First test measurements show that a significant temperature difference between in- and outlet of the calorimeter can be achieved which indicates that the heat transfer from heating rod to the gas is sufficient. Due to the compressibility of gaseous fluids, enthalpy change as a result of pressure drop has to be taken into account. In order to figure out the Joule Thomson coefficient some efforts are still necessary for eliminating heat losses and reducing thermal inertia of the system. This will allow a better calibration of the calorimeter for enhancing the accuracy of heat capacity determination.

Schröder et al.

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