Experimental determination of turbocharger heat flows for validation of a power separation approach

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Abstract. This work presents a new method to measure the heat flow from a turbocharger to the ambient air. The experimental setup, performed on a hot gas test bench, includes an insulated box with two openings, encapsulating the turbocharger. These openings enable a controlled air flow through the box, which is measured at the inlet duct of the box. With temperature measurements at the inlet and outlet of the box, the quantitative heat flow can be calculated. With this information, an approach to estimate those heat flows from standard turbocharger test bench measurements can be validated and refined. The performed measurements at different turbine inlet temperatures and turbine mass flows show a good conformity with expected heat flows. Due to convective and radiative effects, the heat flows increase with both turbine mass flow and turbine inlet temperature. However, the setup with the insulated box has an effect on the overall thermal conditions, since the convection around the turbocharger decreases if the specimen is not as exposed as in common test bench setups.

1. Introduction

The precise prediction of turbocharged engine performance is a central element of automotive powertrain design. Modelling of turbochargers in process simulation usually relies on maps of stationary operating points, which are measured on hot gas test benches with standardized conditions. But the thermodynamic description of the turbocharger behavior is sensitive to the thermal conditions, which distort the isentropic efficiencies significantly on the turbine side. This is caused by heat flows between turbine and compressor gases, oil, cooling liquid, and the surrounding air, see Figure 1, which cannot be prevented by the duct insulation commonly used at test benches. At low rotor speed and pressure ratios, this can result in calculated isentropic turbine efficiencies greater than one, which is a violation of the first law of thermodynamics. The process of expansion within the turbine at low rotor speed is shown in Figure 3. The temperature under diabatic conditions $T_{4,dia}$ is the one measured at the test bench, but the temperature, which would be obtained under adiabatic conditions $T_{4,adi}$ is the one of interest and the relevant value to calculate the correct isentropic turbine efficiency. Hence, the turbine efficiency is usually defined as the ratio of compressor power and isentropic turbine power [1] to avoid the temperature measurement at the turbine outlet. Although thermal distortions are mostly averted this way, turbine efficiency and the mechanical efficiency are inseparably merged in one coefficient, which has a negative influence on the results calculated with map based turbocharger models in process simulation. Turbocharger rotation speed and turbine outflow
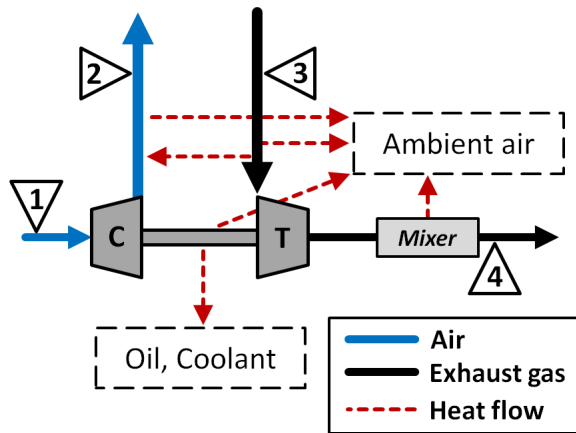
Figure 1. Scheme of turbocharger test bench and occurring heat flows within the measurement plane boundaries.
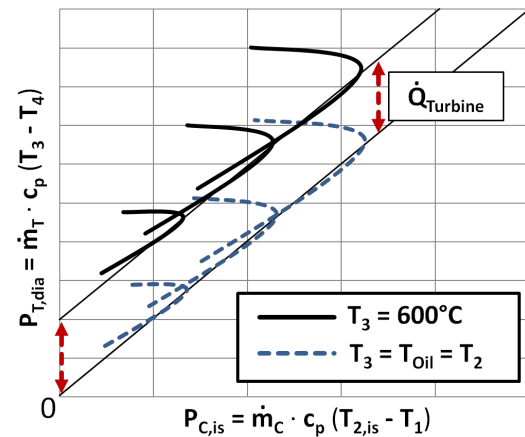

Figure 2. Turbine power curves under hot and adiabatic conditions (schematic).

temperature prediction are particularly difficult with conventional maps, if the turbine inlet temperature is different from the 600°C as prescribed in the standard test bench code.

Methods to correct the thermodynamic performance of turbine and compressor have been developed in previous works, for example by Lüddecke [2] et. al. However, the described method requires significantly increased experimental effort, which is hardly reasonable for non-research users. A potentially more accessible method to separate the aerodynamic power of turbine and compressor as well as the bearing friction losses from the occurring heat flows has been developed by Baar [3] et al. This approach can be conducted with any turbocharger test bench data, only a reliable method to measure the turbine outflow temperature is mandatory. At the chair of powertrain technologies, this is done with a static flow mixer placed in the turbine outlet pipe between the turbine and the respective measurement plane to create a uniform temperature distribution across the turbine outlet duct. The turbine outlet temperature is required to calculate the diabatic turbine power, which can be correlated with the isentropic compressor power as shown in Figure 2. Previous works from Savic [4] et al. have shown, that the interpolated line through the respective compressor power maxima of each speed line intersects the turbine power axis at a value, which corresponds to the total amount of heat transferred from the exhaust gas to other turbocharger parts and the ambient air. This theory is supported by turbocharger test bench data, which has been recorded with close to adiabatic conditions. The turbine inlet temperature and the mean oil temperature have been matched with the compressor outlet temperature for those measurements to minimize the heat losses, especially on the turbine side. As expected, the line interpolated through the isentropic compressor power maxima intersects the y-axis close to zero since there are no significant heat losses, which is schematically shown in Figure 2. The 'power-based approach' has been developed on the assumption, that the isentropic compressor power and the bearing friction are not dependent on the turbine inlet temperature. The extra turbine power apparently needed under hot conditions for the same compressor power output, can be traced back solely to the temperature T_4, which is distorted by heat losses. Hence, the diabatic turbine power measured under hot conditions can be corrected by shifting the data downwards to match the adiabatic measurements and obtain the pure aerodynamic turbine power using the equation

$$P_{T,adi} = \dot{m}_T \cdot c_p \cdot (T_3 - T_4) - \dot{Q}_{Turbine} \qquad (1)$$

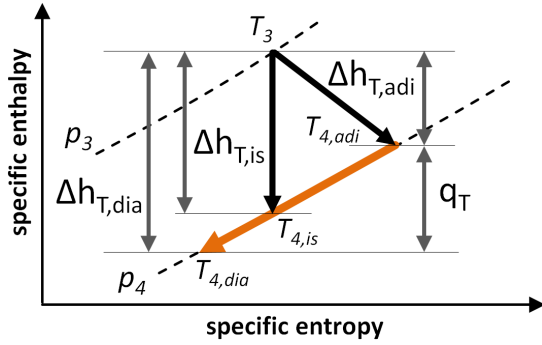where $\dot{m}_T$ is the turbine mass flow, c_p is the specific heat capacity of exhaust gas and the

Figure 3. Schematic thermodynamics of turbine expansion process (total to static).
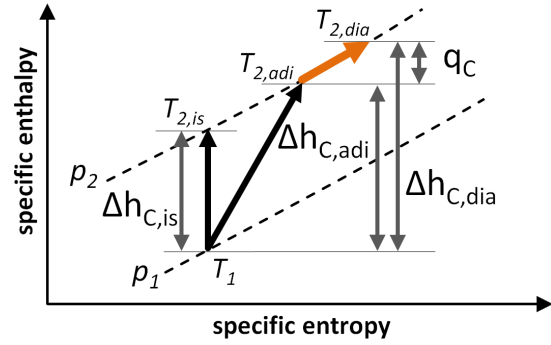

Figure 4. Schematic thermodynamics of compression process (total to total).

temperature at the turbine inlet T_3 is the total temperature. The measurements under adiabatic conditions are not mandatory to use the power-based approach. However, for validation purposes, several adiabatic measurements on differently sized turbochargers have been conducted prior to this investigation. The results showed a similar correlation between the diabatic and adiabatic turbine power curves as displayed in Figure 2.

The power-based approach can also be applied on the diabatic compressor power to separate it from any heat flow disturbances, which are generally small compared to the turbine side. Previous investigations have shown, that the compression process is mainly influenced by added heat from the exhaust gas side [5], which is schematically shown in Figure 4, and almost constant across the compressor operation range. Higher temperatures at the compressor outlet cause an underestimation of the isentropic compressor efficiency, hence the compressor power can be corrected using

$$P_{C,adi} = \dot{m}_C \cdot c_p \cdot (T_2 - T_1) - \dot{Q}_{Compressor} \tag{2}$$

where $\dot{m}_C$ is the compressor mass flow, c_p is the specific heat capacity of air and T_1 and T_2 are total temperatures. The heat flow $\dot{Q}_{Compressor}$ is obtained analogously by plotting the diabatic compressor power correlated to the isentropic compressor power and shifting the data by the occurring offset. The corrected values for turbine and compressor power can now be used to calculate corrected efficiencies and the friction power as shown in the Equations 3 and 4. Using corrected maps and an external input for the bearing friction in process simulation makes the model independent of the overall thermal conditions. However, turbine and compressor performance are still characterized thermodynamically, so the heat flows between the components and the surrounding air have to be added as a sub model to receive the best results. This is done with a network of thermal nodes with connections of convective, conductive and radiative heat transfer to the respective parts.

$$\eta_{T,is} = \frac{P_{T,adi}}{P_{T,is}} \tag{3}$$

$$P_{Friction} = P_{T,adi} - P_{C,adi} \tag{4}$$

While the total heat losses of the turbocharger can be estimated with this approach, it cannot be distinguished, how the individual heat flows are allocated between the components and ambient air. Investigations with conjugate heat transfer (CHT) simulations have shown, that the latter is a relatively large part of the total heat flow [5]. The heat flow box has been developed to determine the exact amount of heat transferred from the turbocharger to the surrounding air during stationary turbocharger operation. This information will help to clarify, if the offset

between the turbine power charts in Figure 2 directly correlates to the external heat flows measured. The dependency of external heat flow on turbine inlet temperature, turbine mass flow and compressor load are also subjects of the investigation.

2. Experimental Setup

The test bench, which the turbocharger and heat flow box have been investigated on, complies with international test bench standard codes such as the norm SAE J1826. Compressor and turbine flow are separate, the turbine side is supplied with air by a screw compressor and mixed with diesel fuel in a combustion chamber. The compressor load is controlled by two valves arranged in parallel, a complete scheme of the test bench can be found in the appendix Figure A1. The object of investigation is a turbocharger for a 2.0-liter engine of series production with a variable geometry turbine, which is fully open for all tests. It is not water-cooled and the oil is conditioned at 3 bars relative and 90°C. As an addition, the turbocharger parts have been equipped with a total of 21 thermocouples to record the outer surface temperature distribution on the housing parts of turbine, bearings and compressor. The 1 mm thermocouple tips have been glued or welded (depending on the expected temperature) into 1.5 mm drill holes in the housing. This is not mandatory for the heat flow box measurement as a standalone system, but gives an impression of the convective conditions between turbocharger and air. Housing wall temperatures have also been recorded during the reference measurement, which was a standard 600°C turbine inlet temperature map with the turbocharger exposed in the air-conditioned test cell and insulated ducts within the measurement plane boundaries.

Computational fluid dynamics (CFD) has been used prior to the construction to optimize the shape of the box. The air stream has to surround the whole turbocharger without creating any recirculation. The box consists of two angular-shaped funnels with a cuboid in the middle, which contains the turbocharger and its peripheral as seen in Figure 5. Natural convection, caused by the hot turbocharger parts, creates a uniform air flow through the box from bottom to top. The amount of air passing through the box is measured by a vortex flowmeter integrated into the inlet pipe. The respective temperatures at the inlet and outlet are measured with five thermocouples each, evenly distributed across the ducts. The rate of heat flow from the
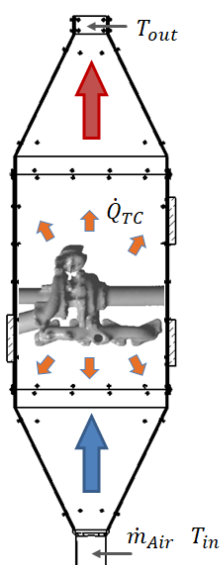

Figure 5. Complete experimental setup, concept and implementation.

Figure 6. Setup of turbocharger inside the box, view from above.

turbocharger to ambient air can be obtained using the equation

$$\dot{Q}_{TC} = \dot{m}_{HFB} \cdot c_p \cdot (T_{out} - T_{in}) \tag{5}$$

with the measured values as displayed in Figure 5. With a measurement accuracy of 1.5°C of the thermocouples and an accuracy of 1.5% of the vortex flowmeter, the mean accumulated uncertainty of the calculated heat flow rate is ±6.5%, which is sufficient to prove the feasibility of this novel measurement concept.

Each wall of the box consists of two layers of aluminum plating with an insulation layer in between, to create a system which is close to adiabatic. The three layers are assembled with threaded rivets and screws and built into an aluminum framework. The temperature on the outer surface of the box is measured at ten evenly distributed points, to estimate the amount of heat leaking through the system. The box has an orifice for each of the four gas ducts leading to and away from turbine and compressor, two orifices for the oil system and one for electric wiring, the whole assembly is showcased in Figure 5. The insulation on the ducts has been added according to the reference measurement without the box to recreate the thermal conditions. The same has been done with the insulation on the duct parts inside the box as seen in Figure 6, which is a view on the turbocharger from the top without the lid on. The smoothed aluminum walls on the inner side of the box serve as a reflector for the heat radiated mainly from the turbine parts. All connected edges of the box are sealed with high-temperature silicone.

3. Results and Discussion

Measurement results are shown for three different turbine inlet temperatures of 400, 600 and 800 degrees Celsius. The operating points have been distributed across the standard compressor map with three points on four different speed lines reaching from choke line to surge line. The additional measurements include the air flow through the box and the temperature difference between the inlet and the outlet of the box. The temperature difference is plotted in Figure 7. It is visible, that the heat flow to ambient air seems to rise significantly with the turbine mass flow. This complies with the turbine housing temperatures, which can be seen in Figure 8 for a turbine inlet temperature of 600 degrees. The reason for that increase is a larger convective heat flow from exhaust gas to housing at higher pressure and velocity. The gradients of the
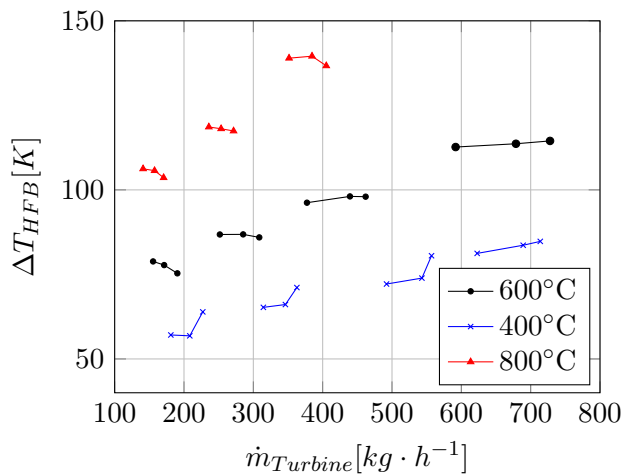

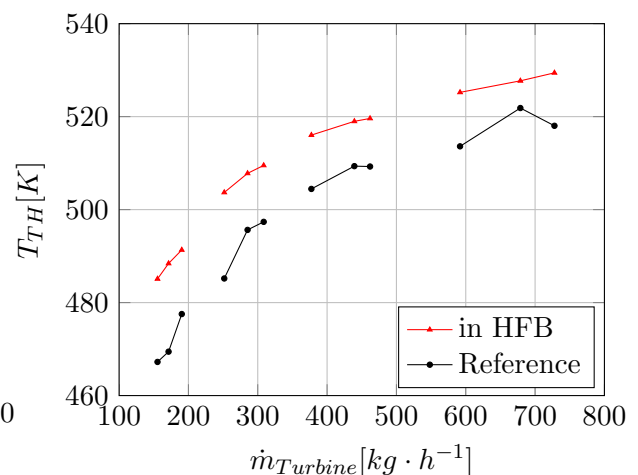**Figure 7.** Temperature difference related to turbine total mass flow for different turbine inlet temperatures.

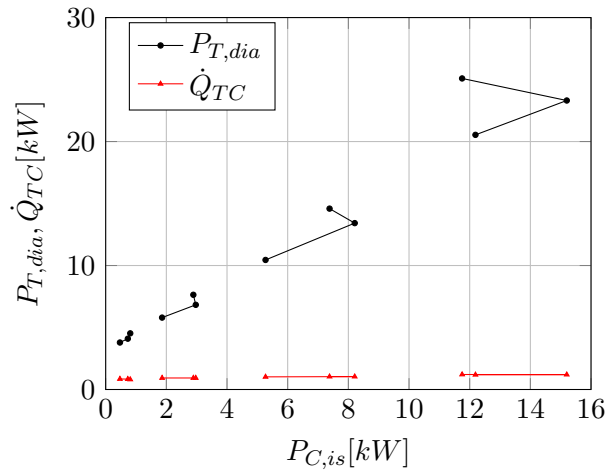**Figure 8.** Turbine housing temperature T_{TH}, $T_3 = 600°C$.

Figure 9. Diabatic turbine power compared to the measured heat flow rate, $T_3 = 600°C$.
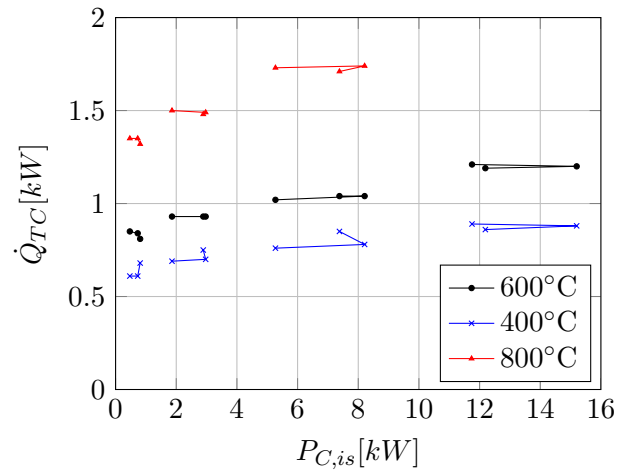
Figure 10. Heat flow to ambient air at different turbine inlet temperatures.

temperature difference plots in Figure 7 also become steeper with higher overall temperatures. This is mainly caused by the radiated heat, which increases proportional to the fourth power of the turbine housing temperature according to the Stefan-Boltzmann law [7]. The radiated heat is mostly reflected at the aluminum walls and causes the whole turbocharger to heat up further compared to the reference measurement without the box. This discrepancy can be seen in Figure 8, where the mean turbine housing temperature is plotted in relation to the turbine mass flow. There is an almost constant offset between the measurement with a heat flow box setup and the one without of about 10 kelvin. It is assumed, that the majority of the radiated heat is converted into convective heat, which then can be measured with the thermocouples at the outlet of the box.

The increase of temperature difference in Figure 7 also substantiates the assumption, that the power offset in Figure 2 is in fact not constant. The total heat loss, hence the distance between diabatic and adiabatic/corrected turbine power, should increase with higher mass flows. On the contrary, the operation point of the compressor does not seem to have a significant impact on the temperature difference across the box. Although the compressor housing temperature increases near the surge line, the turbine mass flow and thus the turbine housing temperature decreases analogously, if the rotor speed is constant. It can be concluded, that the power correction is of a constant value within one speed line using the power-based approach.

However, the quantitative amount of the total heat flow cannot be measured by the heat flow box alone. Although the heat flow to ambient air is a large part of it, it does not include the heat flows of the exhaust gas to the compressor air and the oil circuit. The measured rate of heat flow to ambient air at a turbine inlet temperature of 600°C can be seen in Figure 9. With a volume flow of 33 $m^3 \cdot h^{-1}$ the calculated heat flow rate of 0.8 to 1.2 kW is a lot smaller than the diabatic turbine power output. Also, it is only part of the total heat loss which has been estimated with the power separation approach at 3 kW. Apart from heat leakage through the heat flow box walls, there is also an unknown amount of heat transferred from the exhaust gas to the compressor gas and the oil circuit, which is difficult to determine in experiments. The same applies for the heat flows measured and displayed in Figure 10. The total heat flows needed to correct the turbine power according to Equation 1 is larger than the values determined. However, the influence of turbine mass flow and turbine inlet temperature is also visible in the calculated heat flows.

To get a better understanding, how much heat leaked through the insulated walls of the box, measurements of the outer wall temperature have been conducted. The temperature distribution has been calculated based on the measurements of ten points spread evenly across the box. At the orifices for the exhaust gas ducts and close to the exhaust manifold in particular, see Figure 6, there are hotspots on the outer surface of up to 100°C. The average temperature of the total surface area is about 60°C. Depending on the convective conditions in the test cell, the heat flow from the outer surface of the box to the air is estimated at 0.5 to 0.8 watts. For further experiments, the insulation of the box needs improvement to achieve a better confidence for the turbocharger heat flow values.

4. Conclusion

A new method to directly measure the transferred heat from a turbocharger to the ambient air has been introduced. The concept of a box containing a turbocharger and a controlled air flow has successfully been implemented in an experimental setup. The change of enthalpy from the inlet to the outlet of the box has been measured by recording the respective temperatures and the air flow. Although the measured temperature difference between the inlet and outlet has matched the expected behavior at various boundary conditions, the exact determination of the heat flow has been affected by insufficient insulation of the box. However, valuable data has been collected to substantiate the power-based approach. The increase of heat flow with rising turbine mass flows and turbine inlet temperatures has proven to be an important factor, if the heat losses are estimated with the power-based approach, which can be quantified with more accurate measurements in the future. It has also been established, that the compressor operating point has only a minor effect on the heat flow to ambient air. Furthermore it has been noticed, that the thermal conditions vary with the setup of the box. By changing the convective and radiative conditions around the turbocharger, the housing temperatures changed significantly. A way to convert the radiated heat into convection rather than reflecting it should be considered for further investigations.

Nomenclature

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c_p	specific heat capacity	Subscripts:	
h	specific enthalpy	1	upstream compressor
$\dot{m}$	mass flow	2	downstream compressor
P	power	3	upstream turbine
p	pressure	4	downstream turbine
$\dot{Q}$	rate of heat flow	adi	adiabatic
q	specific heat flow	C	compressor
T	temperature	dia	diabatic
η	efficiency	HFB	heat flow box
		is	isentropic
		T	turbine
		TC	turbocharger
		TH	turbine housing

Appendix
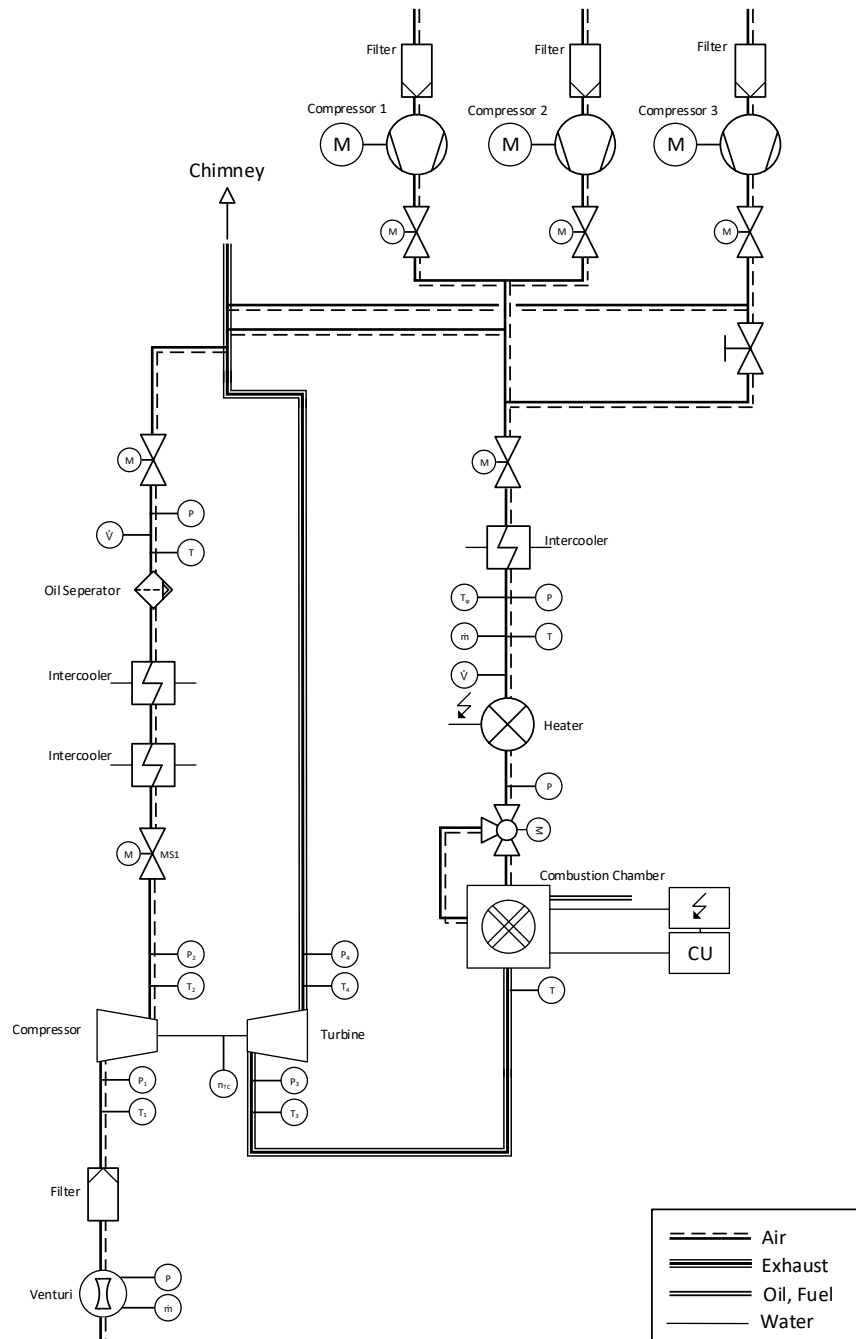
Figure A1. Scheme of hot gas test bench at Technische Universität Berlin [4].

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