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Development and Matching of Double Entry Turbines for the Next Generation of Highly Boosted Gasoline Engines

Abstract

Downsizing in combination with turbocharging represents the main technology trend for meeting climate relevant CO_2 emission standards in gasoline engine applications. Extended levels of downsizing involve increasing degrees of pulse charging. Separation of cylinder blow downs, either with double entry turbines or valve train variability, is key for achieving enhanced rated power and low-end-torque targets in highly boosted four-cylinder engines.

However, double entry turbines feature specific development challenges: The aerodynamic design via 3D CFD calculations presents a difficult task as well as the engine performance modeling and matching process in 1D gas exchange simulations. From a manufacturing standpoint, casting of the turbine housing is complex especially for small displacement applications below 1.6 l due to e. g. thermo-mechanical boundaries.

This paper demonstrates how to design and model double entry turbine performance characteristics within 1D gas exchange simulations, requiring special measured and processed turbine data, which is experimentally assessed on a hot gas test bench using a double burner setup. It is shown how the collective of the described development strategies can be used in assessing the potential of different turbine design concepts. This allows the turbocharger to be designed exactly to specific engine requirements.

1 Introduction

Turbocharging in combination with direct injection and variable valve timing represents the major technology trend in modern gasoline engine concepts for achieving high specific power output with low fuel consumption at the same time. As illustrated in Figure 1 the trade-off between high specific power output on the one hand and compelling low-end-torque performance on the other hand can only be achieved by separating the cylinders exhaust blow down events. These performance requirements can either be accomplished through different technological strategies (e.g. scroll separation in exhaust manifold or turbine; variable exhaust valve event) or by increasing the firing distance (three cylinder engine).

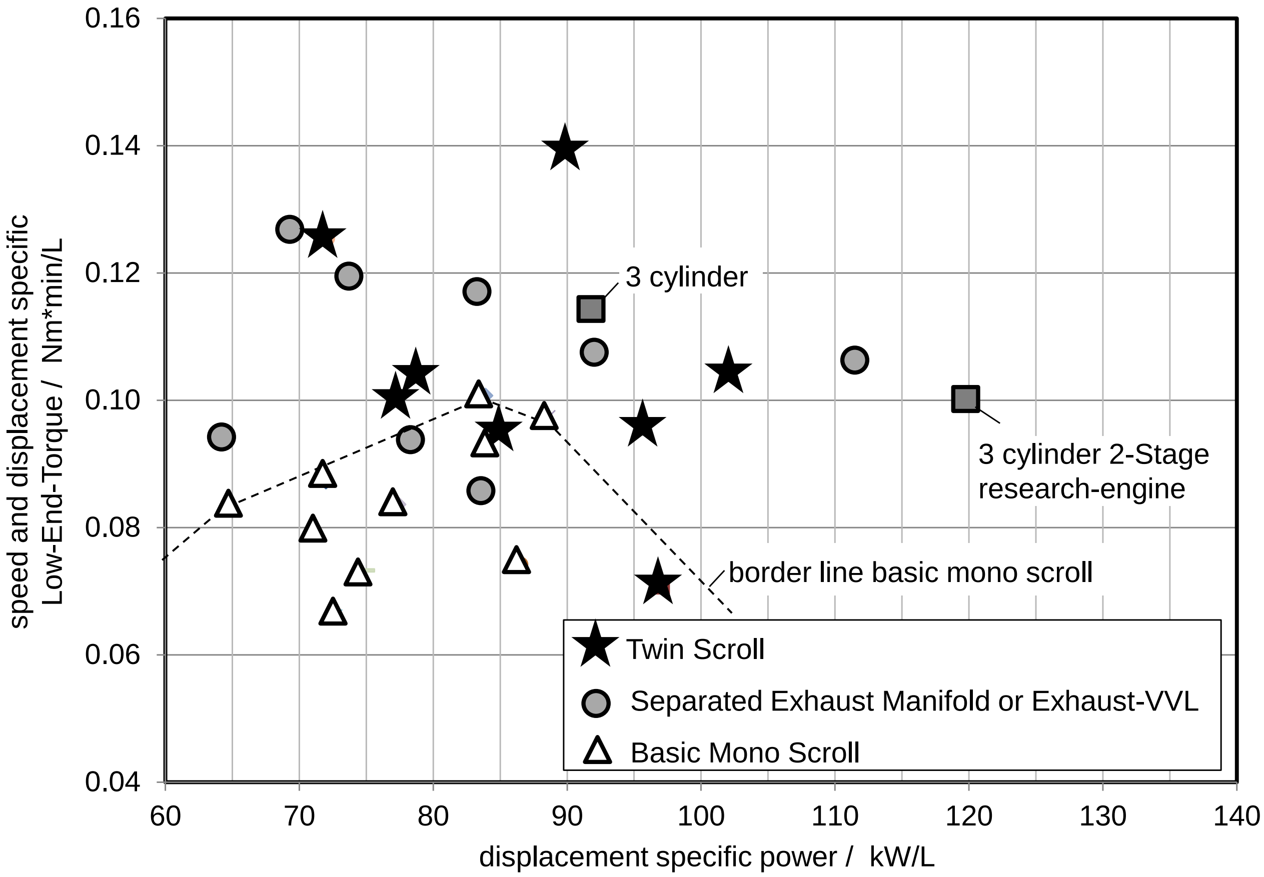

Figure 1: Technology overview of boosted gasoline engines as a function of displacement specific power and displacement and engine speed specific low-end-torque

Double entry turbines feature lower peak efficiencies compared to conventional mono scroll turbines as a result of inherent friction losses on the separation wall of the turbine housing. Figure 2 shows the maximum turbine efficiency observed at steady state mapping on the hot gas test bench versus wheel diameter for a number of turbines of each type. Despite this drawback, double entry turbines are an alternative to complex charging systems on four-cylinder engines and expensive bi-turbo six-cylinder engine concepts (Björnsson et al. [1]) for achieving satisfying low-end-torque combined with high specific power. Double entry turbines in series production today are exclusively designed as twin scroll turbines meaning that the scrolls are divided circumferentially. This paper does only refer to this type of turbine design and the term twin scroll is used synonymously for the entirety of double entry turbines.

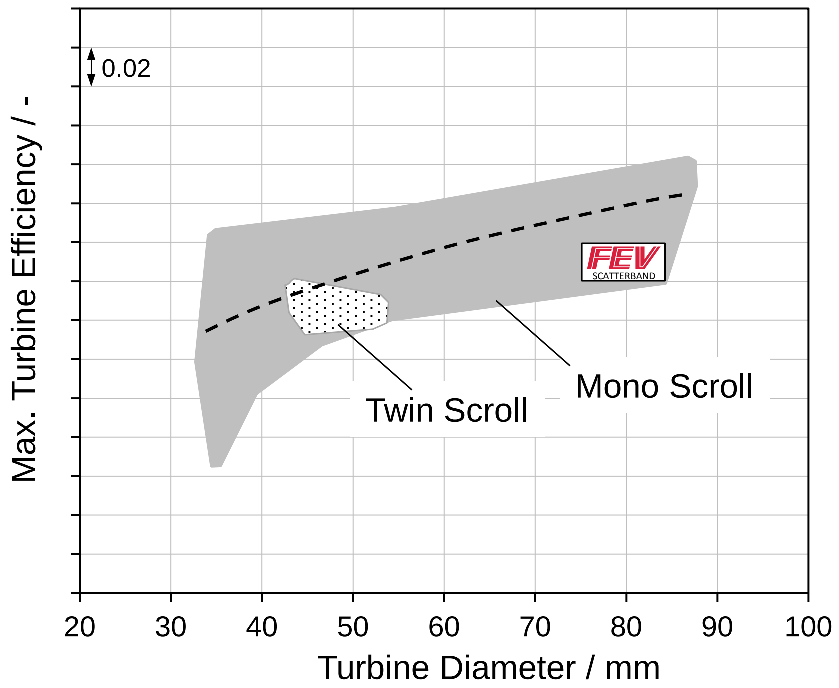

Figure 2: Turbine efficiency vs. turbine diameter for mono and twin scroll turbines.

Björnsson et al. [1] identify the advantages of a double entry turbine comparing to a bi-turbo concept as being:

- Turbine efficiency considering the pulsating nature of the exhaust gas,
- Compressor efficiency as a result of increased compressor size,
- Package and
- System cost.

Additional benefits and drawbacks of double entry turbines compared with conventional mono scroll layouts can be summarized as follows:

- + Possible reduction of residual gas fraction and consequently knock propensity
- + Application of longer intake valve event length with corresponding reduction in part load fuel consumption

- - Complicated casting process and high thermo-mechanical load in the area of the tongue

The layout of a specific double entry turbine for a given target engine poses a complex design task in 1D engine process simulation. The following challenges have to be addressed specifically:

- Measurement data obtained on a hot gas test bench is typically insufficient in order to describe the operating behavior of the turbine on-engine (e.g. flow admission only on one scroll)
- The current modeling approaches do not sufficiently describe the real physical turbine behavior in engine operation even if accurate map data is available

The challenges occurring in the matching and design process of double entry turbines are presented amongst others by Schmalzl [2] and Winkler [3].

The goal of the study presented in this paper is to demonstrate the matching process of a double entry turbine for a 1.6 l direct injection gasoline engine with variable intake and exhaust valve timing. At first, a brief summary about the history and current status in double entry turbine research is given followed by the introduction of characteristic numbers necessary to describe double entry turbine performance characteristics. Next, an experimental mapping method on the FEV hot gas test bench is presented which allows describing the on-engine behavior of double entry turbines. Subsequently, an extended modeling approach based on the work of Brinkert ([4], [5], [6]) is highlighted using the steady state map data in a physical fashion. This extended approach is assessed by gas exchange simulation using the commercial code GT-Power. Additionally, the presented results compare the extended approach with the established methodology recommended in the GT-Power user manual (subsequently referred to as reference method). Within the study special focus is given to the degree of separation between the two scrolls and the influence of a variation of this parameter on engine performance.

2 Background

The subject of double entry turbines has already been present in literature for several decades, focusing especially on the development of modeling strategies. A variety of numerical, analytical and experimental approaches can be found in literature as how to optimize and analyze the turbocharger as a subsystem as well as the interaction of turbomachine and combustion engine:

Engels [7], Pischinger [8] and Wünsche [9] analyze the characteristics of double entry turbines applied to boosted diesel engines using 0D engine process simulation and engine test bench investigations. Müller et al. [10], [11] show the benefits of an asymmetric twin scroll turbine with the use of external exhaust gas recirculation applying engine process as well as CFD calculations. Winkler et al. [12] investigate the asymmetric behavior of a twin scroll turbine by calculating the instantaneous turbine efficiency based on engine data. Copeland et al. [13] analyze the unequal admission characteristic of double entry turbines with 3D CFD in order to determine the occurring flow losses.

Baines et al. [14] perform measurements under equal and unequal admission and thus determine flow velocity and flow angle in close proximity to the wheel of a twin scroll turbine. Fredriksson [15] and Hajilouy [16] present a 1D flow simulation model for double entry radial turbines. Furthermore, the simulation results are verified with test bench data.

Brinkert et al. [5] perform measurements at unequal admission and derive parameters characterizing the flow conditions of the turbine. Additionally, these flow conditions are analyzed using 3D CFD simulation. Winkler [3] and Schmalz [2] investigate the potential of double entry turbines for boosted gasoline engines on an engine test bench. Björnsson et al. [1] show results of 1D process simulation of a boosted gasoline engine in steady state and transient operation. Brinkert [4], [6] presents an extended modeling approach for double entry turbines in engine process simulation. The modeling results of asymmetric as well as symmetric turbines are compared with engine test bench data. Uhlmann and Lückmann [17] introduce a method for measuring unequal admission and cross-flow between the scrolls on the turbocharger hot gas test bench.

3 Characteristic Numbers of Double Entry Turbines

Describing double entry turbines requires additional characteristic numbers complementing the well-known conventional turbocharger performance parameters. These characteristic numbers are introduced and explained in the following chapter.

In order to characterize the flow condition within a double entry turbine Brinkert ([5], [6]) introduces the parameter “Mach Number Ratio” (*sMaR*) which denotes the ratio of the reduced mass flow rates in each scroll. Following this definition the parameter “Mass Flow

Ratio” (*MFR*) is defined in this work as the ratio of the reduced mass flow rate of scroll 1 to the total reduced mass flow rate of the turbine according to Eq. 1:

$$MFR = \frac{\dot{m}_{\mathrm{red,Fl1}}}{\dot{m}_{\mathrm{red,Fl1}} + \dot{m}_{\mathrm{red,Fl2}}} = \frac{1}{1 + \frac{1}{sMaR}} \qquad \text{Eq. 1}$$

The parameter *MFR* offers the major advantage of having a linear correlation with the flow condition (comparative *sMaR* features a hyperbolic dependency). This yields optimal controllability of the parameter during measurement on the hot gas test bench and improved regulation within engine process simulation. With both parameters the flow condition is uniquely assigned to one corresponding value (single, equal, unequal admission and cross-flow). Furthermore, the parameter value occurring at equal admission gives information about the degree of symmetry of the turbine.

Figure 3 illustrates the flow conditions and the corresponding MFR values at equal and single admission for a twin scroll turbine. When the turbine is operating at equal admission, the pressure is equal in both scrolls and the Mass Flow Ratio becomes approximately 0.5. For an ideal flow-symmetric turbine *MFR* is exactly 0.5. For turbine designs with geometric symmetry the *MFR* value at equal admission slightly differs from 0.5 due to the differences of the flow fields approaching the wheel. Usually the inner scroll close to the centre housing features an increased flow capacity. Single admission is characterized by one scroll being blocked. Conclusively, the flow is approaching the turbine wheel solely through the second unblocked scroll. At this flow condition the Mass Flow Ratio takes on values of *MFR* = 1 and *MFR* = 0 respectively.

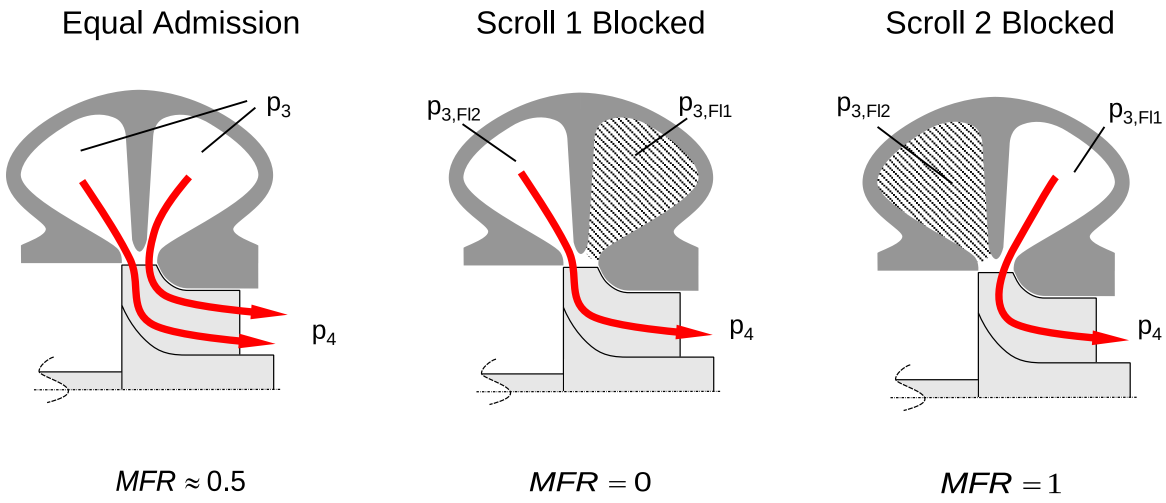

Figure 3: Flow conditions in a twin scroll turbine at equal and single admission

The flow conditions present at equal and single admission can be considered as a special form of mass flow distribution. All remaining flow conditions occurring in between the borderline cases of single admission are referred to as unequal admission or cross-flow respectively and are illustrated in Figure 4.

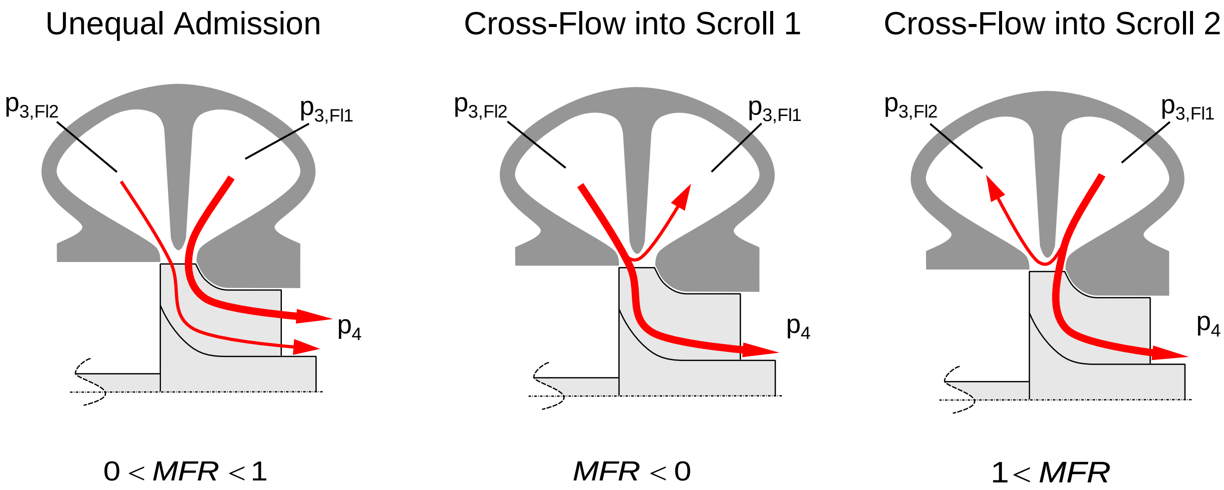

Figure 4: Flow conditions in a twin scroll turbine at unequal admission and cross-flow

The Mass Flow Ratio takes on values between zero and one during unequal admission. If the scroll pressure ratio exceeds the boundaries of the single admission at a given turbine pressure ratio (*MFR* < 0 or *MFR* > 1), interaction occurs between the two scrolls. This means that a fraction of the mass flow going through the scroll with higher pressure level does not reach the turbine wheel, but recirculates into the second scroll in front of the wheel. This phenomenon is commonly described as back-flow, within this paper it is referred to as cross-flow.

The reduced speed represents another important parameter of double entry turbines analog to mono scroll turbines. The reduced speed for double entry turbines is calculated using the enthalpy-averaged temperature at turbine wheel inlet, based on knowledge published by Brinkert [6] and obtained within the FVV project „Extended Turbine Mapping“ [17]. Figure 5 shows the required measurement quantities for calculating the enthalpy-averaged temperature.

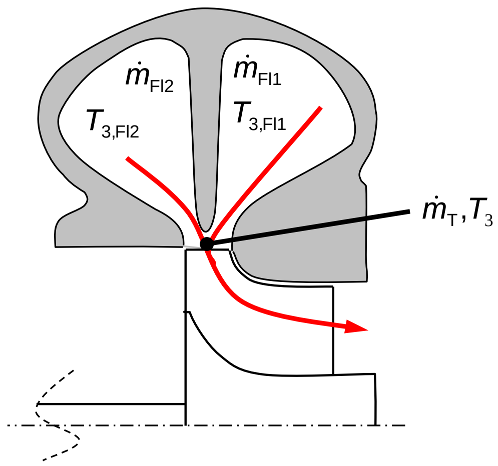

Figure 5: Definition of turbine wheel inlet temperature for calculation of the reduced turbine speed

Using an energy balance at the mixing point of both flows at turbine wheel inlet, the mixing temperature is determined as follows. The error resulting from the assumption of a constant specific heat capacity can be neglected:

$$\Delta\dot{H} = 0$$

$$\Leftrightarrow \dot{m}_{Fl1} h_{3,Fl1.tot} + \dot{m}_{Fl2} h_{3,Fl2,tot} = \dot{m}_T h_{3,tot}$$

$$\Leftrightarrow \dot{m}_{Fl1} c_{p3,Fl1} \left(\overline{T}_{3,tot} - T_{3,Fl1.tot}\right) + \dot{m}_{Fl2} c_{p3,Fl2} \left(\overline{T}_{3.tot} - T_{3,Fl2,tot}\right) = 0$$

$$\Leftrightarrow \overline{T}_{3,tot} = \frac{\dot{m}_{Fl1} c_{p3,Fl1} T_{3,Fl1,tot} + \dot{m}_{Fl2} c_{p3,Fl2} T_{3,Fl2,tot}}{\dot{m}_{Fl1} c_{p3,Fl1} + \dot{m}_{Fl2} c_{p3,Fl2}} \quad \text{Eq. 2}$$

This temperature is then used to calculate the reduced speed of a double entry turbine. Hence, only one unique value of reduced speed exists even in the case of different inlet temperatures in both scrolls.

$$n_{\text{red,T}} = \frac{n_{\text{TC}}}{\sqrt{\overline{T}_{3,\text{tot}}}} \qquad \text{Eq. 3}$$

Supplemental to the turbine pressure ratios and reduced mass flow rates of each individual scroll according to Eq. 4 and Eq. 5, the scroll pressure ratio is introduced by Eq. 6.

$$\Pi_{\text{Fl1,T}} = \frac{p_{3,\text{Fl1,tot}}}{p_{4,\text{st}}} \qquad \Pi_{\text{Fl2,T}} = \frac{p_{3,\text{Fl2,tot}}}{p_{4,\text{st}}} \qquad \text{Eq. 4}$$

$$\dot{m}_{\text{red,FL1}} = \dot{m}_{FL1}\frac{\sqrt{T_{3.\text{FL1}}}}{p_{3,\text{Fl1,tot}}} \qquad \dot{m}_{\text{red,FL2}} = \dot{m}_{\text{FL2}}\frac{\sqrt{T_{3.\text{FL2}}}}{p_{3,\text{Fl2,tot}}} \qquad \text{Eq. 5}$$

$$\Pi_{\text{Scroll}} = \frac{p_{3,\text{Fl1,tot}}}{p_{3,\text{Fl2,tot}}} \qquad \text{Eq. 6}$$

Next, it is necessary to define an average turbine inlet pressure in order to compare different double entry turbines regarding the efficiency and separation characteristics of the scrolls. In the work published by Wünsche et al. [9] an arithmetic mean value of both scrolls is used for this purpose. Alternatively, the pressure ratio of an equivalent mono scroll turbine featuring equal flow capacity and efficiency than the double entry turbine can be used. This pressure ratio is derived from Eq. 7 as follows:

$$\eta_{\text{Mono Scroll}} = \eta_{\text{Twin Scroll}}$$

$$\Leftrightarrow (\dot{m}_{\text{Fl1}} + \dot{m}_{\text{Fl2}})\left(h_{3,\text{tot}} - h_{4,\text{is,st}}\right) = \dot{m}_{\text{Fl1}}\left(h_{3,\text{FL1,tot}} - h_{4,\text{Fl1,is,st}}\right) + \dot{m}_{\text{Fl2}}\left(h_{3,\text{FL2,tot}} - h_{4,\text{Fl2,is,st}}\right)$$

$$\Leftrightarrow \Pi_{\text{Mono Scroll}} = \left(1 - \frac{\dot{m}_{\text{Fl1}}\left(h_{3,\text{FL1,tot}} - h_{4,\text{Fl1,is,st}}\right) + \dot{m}_{\text{Fl2}}\left(h_{3,\text{FL2,tot}} - h_{4,\text{Fl2,is,st}}\right)}{(\dot{m}_{\text{Fl1}} + \dot{m}_{\text{Fl2}})\overline{T}_{3,\text{tot}}\overline{c}_{\text{p}}}\right)^{\frac{-\overline{\kappa}}{\overline{\kappa}-1}} \qquad \text{Eq. 7}$$

Using the enthalpy-averaged turbine inlet temperature from Eq. 2 leads to the corresponding turbine pressure ratio of the mono scroll turbine. For the boundary conditions applied within the following investigations this pressure ratio equals the mass flow averaged pressure ratio according to Eq. 8 with sufficient accuracy.

$$\Pi_{\text{ma,T}} = \frac{\dfrac{\left|\dot{m}_{\text{red,FL1}}\right| p_{3,\text{Fl1,tot}} + \left|\dot{m}_{\text{red,FL2}}\right| p_{3,\text{Fl2,tot}}}{\left|\dot{m}_{\text{red,FL1}}\right| + \left|\dot{m}_{\text{red,FL2}}\right|}}{p_{4,\text{st}}} \qquad \text{Eq. 8}$$

The knowledge of the scroll pressure ratio defined above together with the average turbine pressure ratio is important in determining the flow condition of the turbine within engine process simulations. Figure 6 shows the interaction map of a twin scroll turbine with the individual flow conditions mentioned above.

In this interaction map the turbine pressure ratio is plotted against the scroll pressure ratio thus allowing to divide the operating map in areas of equal flow conditions. If the pressures before and after the scrolls as well as the reduced turbine speed are known, the flow condition can be determined definitely.

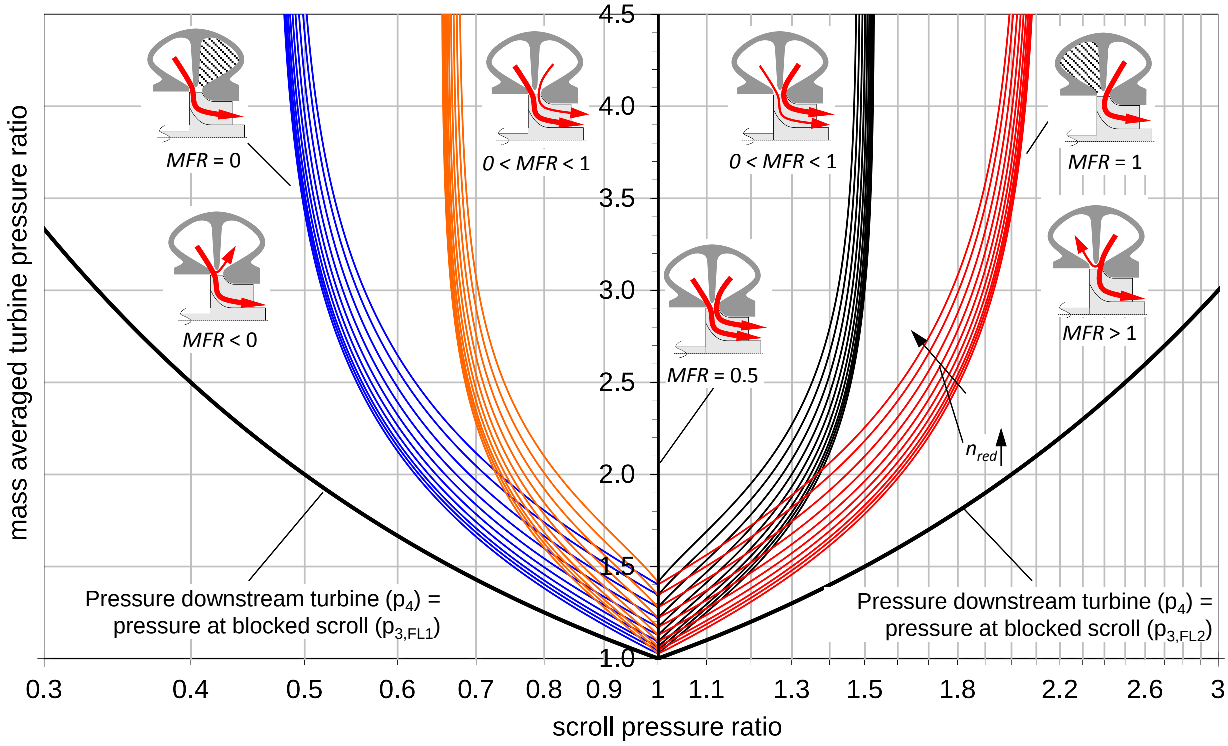

Figure 6: Schematic representation of the flow interaction map

The previously defined pressure ratios can now be used to calculate the isentropic temperatures at turbine outlet according to:

$$T_{4,Fl1,is,st} = T_{3,Fl1,st}\left(\frac{1}{\Pi_{FL1,st}}\right)^{\frac{\kappa_{FL1}-1}{\kappa_{FL1}}} \quad T_{4,Fl2,is,st} = T_{3,Fl2,st}\left(\frac{1}{\Pi_{FL2,st}}\right)^{\frac{\kappa_{FL2}-1}{\kappa_{FL2}}} \qquad \text{Eq. 9}$$

These temperatures are the basis for determining the isentropic power in each scroll:

$$P_{is,Fl1} = \dot{m}_{Fl1}\left(h_{3,FL1,tot} - h_{4,Fl1,is,st}\right) = \dot{m}_{Fl1}c_{p,3,Fl1}\left(T_{3,FL1,tot} - T_{4,Fl1,is,st}\right) \qquad \text{Eq. 10}$$

$$P_{is,Fl2} = \dot{m}_{Fl2}\left(h_{3,FL2,tot} - h_{4,Fl2,is,st}\right) = \dot{m}_{Fl2}c_{p,3,Fl2}\left(T_{3,FL2,tot} - T_{4,Fl2,is,st}\right)$$

The efficiency of the double entry turbine can now be calculated by dividing the consumed compressor power by the sum of the isentropic powers of both scrolls:

$$\eta_{\mathrm{is,T}}\eta_{\mathrm{mech,TC}} = \frac{P_{\mathrm{C}}}{P_{\mathrm{is,Fl1}} + P_{\mathrm{is,Fl2}}} \qquad \text{Eq. 11}$$

As a new parameter for characterizing double entry turbines the “Scroll Separation Level” (*SSL*) is introduced. The parameter is deduced from the mass flow difference between equal and single admission. Specifically, *SSL* is defined as the ratio of the actual mass flow difference to that occurring at ideal separation.

$$SSL = \frac{a}{b} = \frac{\dot{m}_{\mathrm{red,\,Equal\ Admission}} - \dfrac{\dot{m}_{\mathrm{red,\,Single\ Admission,Fl1}} + \dot{m}_{\mathrm{red,\,Single\ Admission,Fl2}}}{2}}{\dot{m}_{\mathrm{red,\,Equal\ Admission}} - \dfrac{\dot{m}_{\mathrm{red,\,Equal\ Admission}}}{2}} = \left(2 - \frac{\dot{m}_{\mathrm{red,\,Single\ Admission,Fl1}} + \dot{m}_{\mathrm{red,\,Single\ Admission,Fl2}}}{\dot{m}_{\mathrm{red,\,Equal\ Admission}}}\right) \qquad \text{Eq. 12}$$

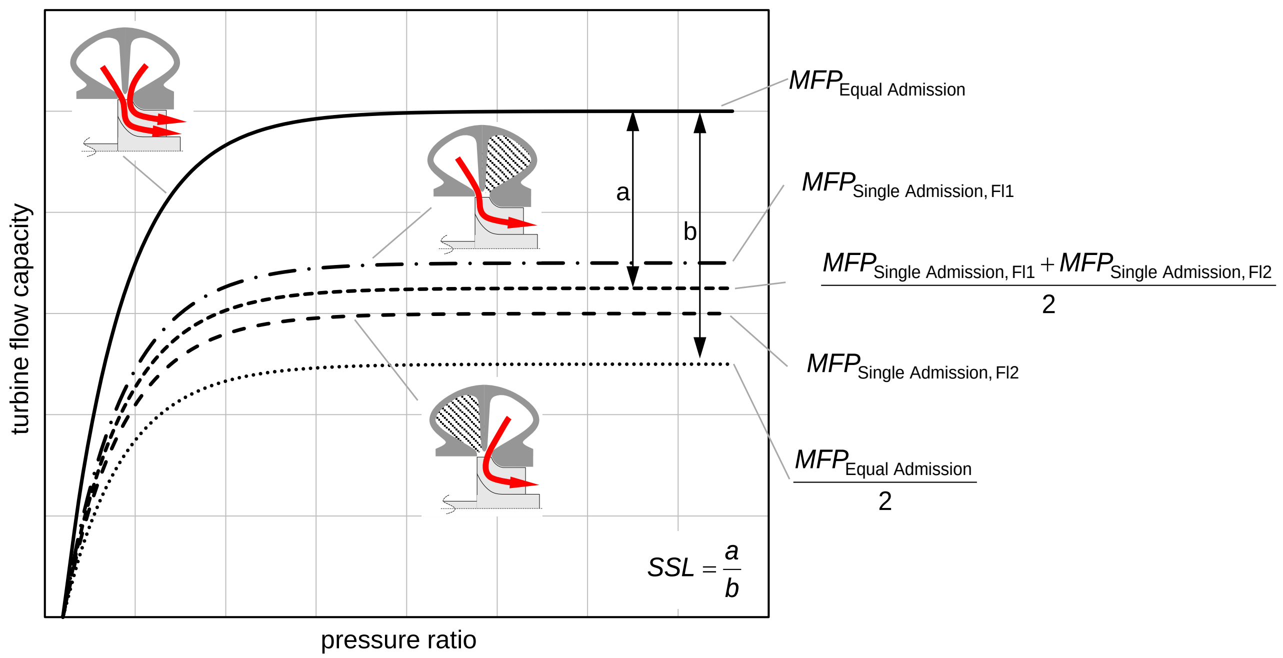

Figure 7: Definition of scroll separation level for double entry turbines

Ideal separation where there is no interaction between the scrolls is characterized by the equal admission flow capacity being equal to the sum of the single admission flow capacities of scroll 1 and scroll 2. This theoretical case denotes the denominator in the definition of *SSL* in Eq. 12. In order to obtain one unique value for *SSL* in the case of turbines not featuring flow-symmetry, the arithmetic mean value is used for the single admission flow capacity in the numerator of Eq. 12. This condition occurs for turbine designs that have scrolls with geometrical symmetry, but diverging single admission flow capacities as was already described in chapter 4. Thus defined, the separation level can take on value in the range from 0 to 100 %. The borderline case of *SSL* = 0 % corresponds to a mono scroll turbine (flow capacity at single admission equivalent to equal admission). Hence, a value of *SSL* = 100 % reflects a double entry turbine with ideal separation.

4 Experimental Methodology

In this section the experimental setup for measuring the performance characteristics of double entry turbines is depicted. The measurements are performed on a hot gas test bench and completely cover the operating range of the turbine relevant for engine operation.

The three special flow conditions emerging at equal and single admission (see Figure 3) can be measured on any turbocharger hot gas test bench in principle allowing to quantify the thermodynamic performance in terms of flow capacity and efficiency in these operating modes. In this work, turbine operation occurring in between single and equal admission is additionally examined and, for the first time, operating points where cross-flow between the two scrolls occurs are investigated. For this type of measurement an extension of the conventional hot gas test bench setup is required which is subsequently elaborated in more detail.

The method for mapping double entry turbines on the hot gas test bench applied in this work is based on a setup with double burner operation which was developed within the FVV research project "Extended Turbine Mapping" [17]. This type of measurement can only be conducted to a limited extent on a conventional turbocharger hot gas test bench equipped with only one single combustion chamber. Here, the hot gas mass flow has to be divided in order to feed both turbine scrolls. Varying the Mass Flow Ratio *MFR* inevitably affects the temperature distribution in the inlet of scroll 1 and scroll 2. Hence, only one of the two inlet temperatures can be kept constant, the respective second temperature varies from operating point to operating point.

FEV's hot gas test bench offers the unique feature of two combustion chambers which can be operated at the same time. Using the double burner setup, one burner is connected to one scroll of the turbine which allows controlling pressure as well as temperature in each scroll separately and independently.

Besides obtaining the equal admission (*MFR* = 0.5) and single admission (*MFR* = 0 or 1) maps, it is also possible to measure maps with intermediate flow conditions (0 < *MFR* < 0.5 or 0.5 < *MFR* < 1) with this setup. Figure 8 schematically highlights the experimental setup that was developed.

In order to measure maps which characterize the flow condition where interference between the scrolls in form of cross-flow occurs (*MFR* < 0 or *MFR* > 1), a setup similar to single admission is used.

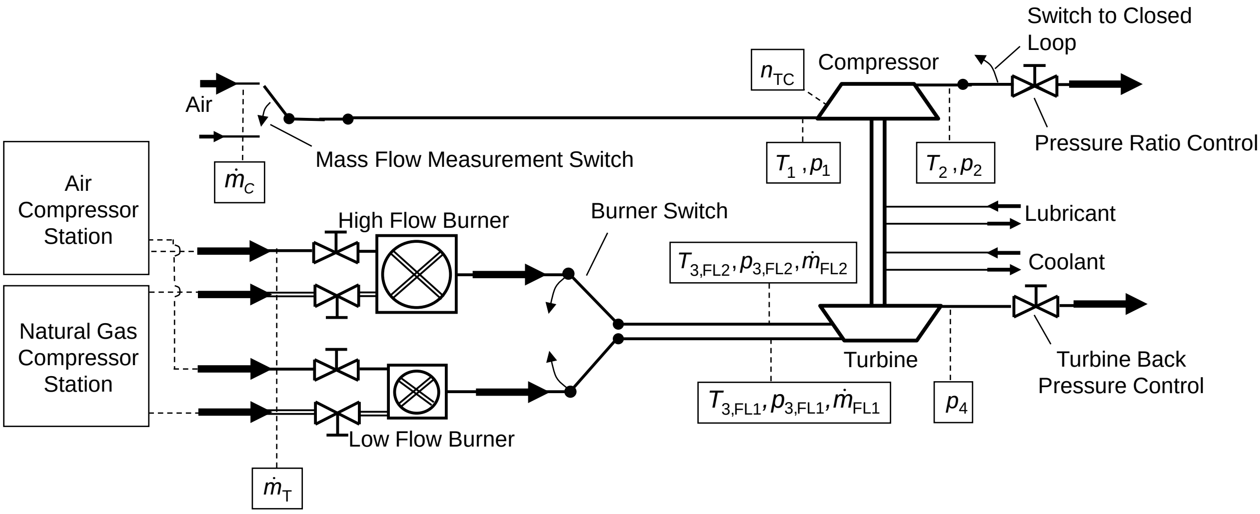

Figure 8: Schematic of test bench setup for measuring double entry turbines

The only difference lies in the fact that the second scroll is being opened to ambient, rather than being physically blocked. In detail the second scroll is connected to a mass flow meter with an intermediate water-air heat exchanger and back pressure control valve. The heat exchanger is required for reducing the gas temperature and therefore protecting the measurement equipment. The back pressure valve is used to regulate the ratio of the mass flow going through the turbine wheel to the fraction of mass flow which is cross-flowing from one scroll to the other. This setup is schematically shown in Figure 9.

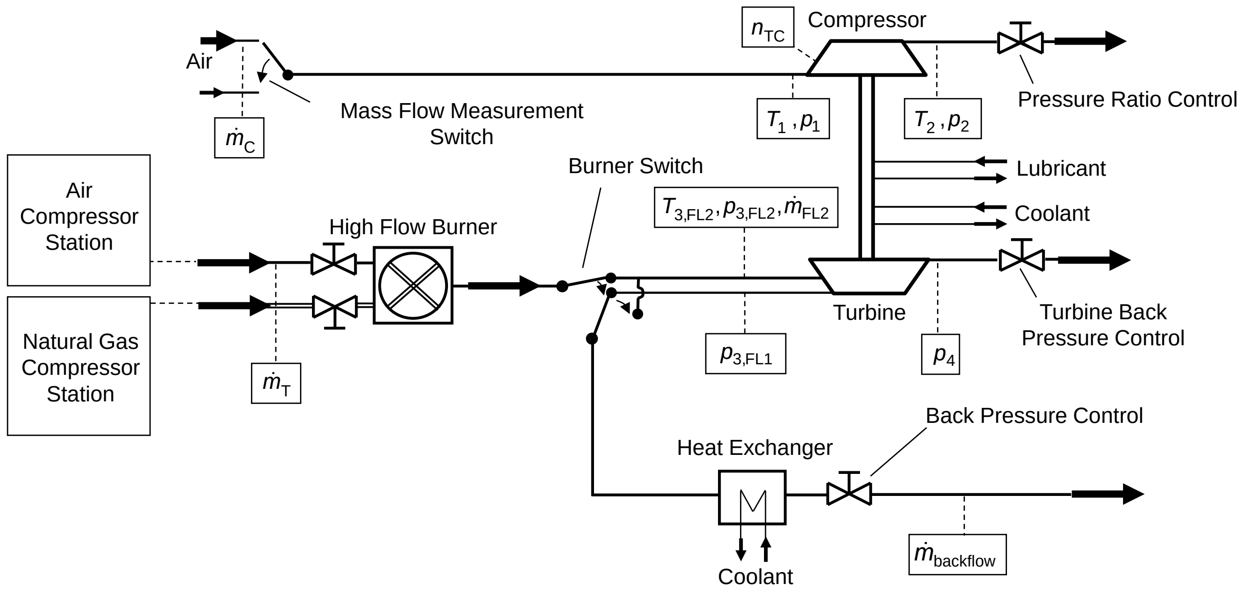

Figure 9: Schematic of test bench setup for measuring scroll interaction

5 Method of Investigation

Within the following section the modeling approach for double entry turbines employed within engine process simulation is explained. Furthermore, the utilized engine model is described in conjunction with the boundary conditions applied within the conducted simulation study.

5.1 Modeling of Double Entry Turbines

5.1.1 Reference Model of the Double Entry Turbine

In order to allow for a comparison of the extended modeling approach introduced later on with an established method, a reference model has to be defined. In this case, the modeling approach recommended and described by Gamma Technologies in the GT-Power user manual [18] is chosen for this purpose.

The reference method is based on the knowledge of the turbine behavior at equal admission; both scrolls are treated as completely separated turbines. The flow capacity of the individual scrolls is then scaled by:

$$\dot{m}_{red,T,SingleAdmission} = \frac{\dot{m}_{red,T,EqualAdmission}}{2} \qquad \text{Eq. 13}$$

The efficiency map remains unchanged. The interaction between the two scrolls is commonly accounted for by integrating an orifice. The cross sectional area of the orifice is usually calibrated operating point dependent to match measured engine data. In case no measurement data is available to validate the model, a common approach for determining the cross flow area is to apply the following correlation:

$$d_{orifice} = \sqrt{d_{tongue}^2 - d_{impeller}^2} \qquad \text{Eq. 14}$$

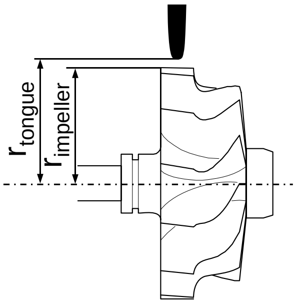

Figure 10 presents the reference modeling approach for double entry turbines in GT-Power by means of the required input parameters. The distribution of the turbine maps is done automatically by the software. Cross-flow has to be modeled manually by the user by adjusting the orifice object.

The benefits of such a modeling strategy can be summarized as follows:

- Only measured map data of equal admission is required as input which is usually provided by the turbocharger manufacturer
- No complex and time-consuming post-processing of the map data is necessary before usage in an engine process simulation software

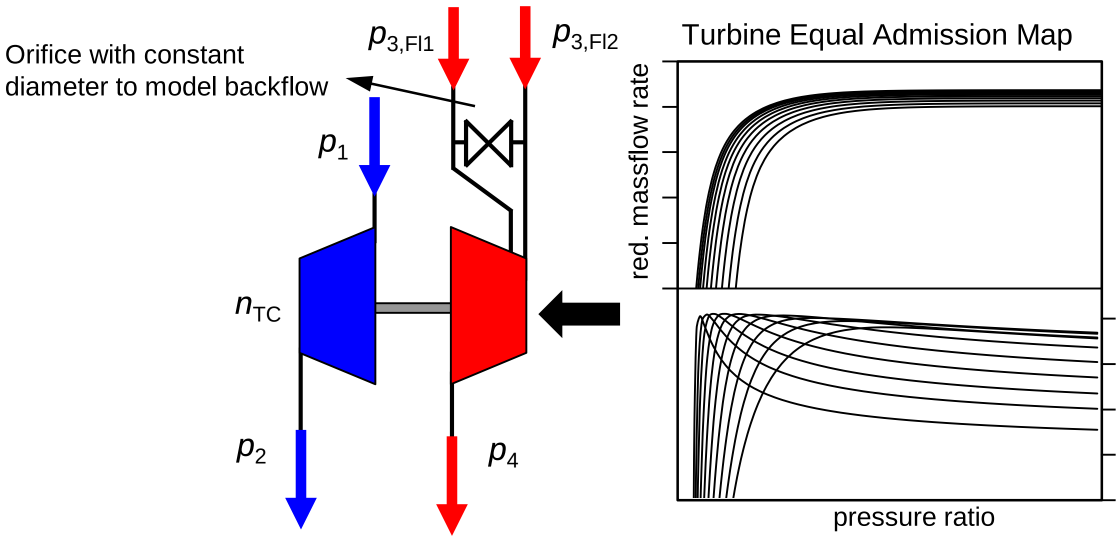

Figure 10: Reference modeling approach for twin scroll turbines [18])

One drawback of the reference modeling approach arises from the assumption that the flow capacity of each scroll equals half the flow capacity of the equal admission map and that the efficiency map is directly taken from the equal admission conditions. Figure 11 shows actual twin scroll turbine measurement data proving that these assumptions made in the reference approach do not reflect the real physical behavior of the turbine.

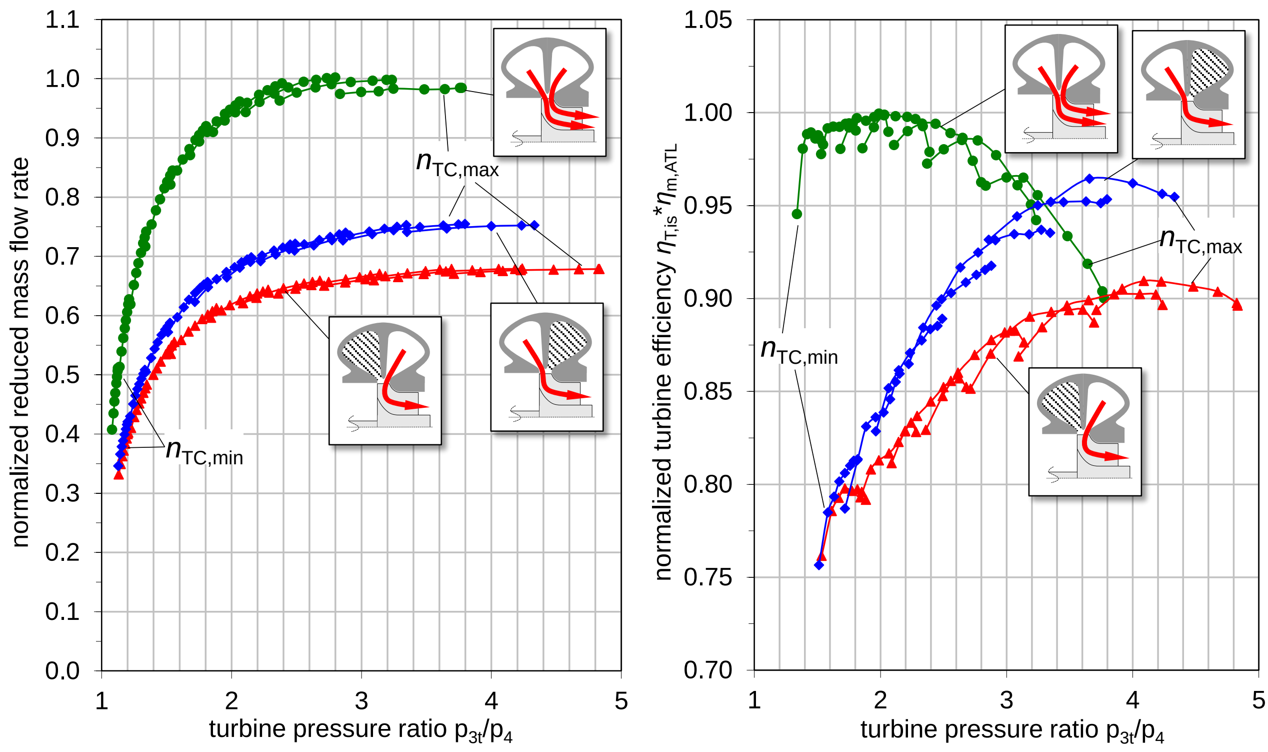

Figure 11: Flow capacity (left) and turbine efficiency (right) in equal and unequal admission

Plotting the mass flow characteristic versus pressure ratio it is observed that the sum of both single scrolls (red and blue) exceeds the equal admission value by 20 % at Π_T = 3. In turn, this means that the flow capacity calculated with the reference approach following Eq. 13 is dramatically underestimated. This typically results in the choice of an oversized turbine within the matching process. The impact of the degree of separation between the scrolls on the flow capacity of the individual scrolls is explained in more detail in chapter 0. The fairly different efficiency characteristics at equal and single admission constitute another factor of uncertainty in the simulation. The location of maximum efficiency is shifted towards higher turbine pressure ratios: At equal admission, peak efficiency occurs at a pressure ratio of approximately Π_T = 2; at single admission this value accounts for approximately Π_T = 4. As a result of this effect the turbine efficiency at single admission is up to 15 % lower compared to equal admission at low pressure ratios. At high pressure ratios this trend is reversed and the single admission efficiency excels the efficiency level obtained at equal admission by up to 5 % points.

The uncertainties resulting from modeling the cross-flow area using a 1D orifice combined with the differences in the mass flow and efficiency maps depending on the flow condition (*MFR*) make the predictive matching of the turbine a difficult task. A further disadvantage of the reference approach arises from the fact that the degree of separation is not considered and thus cannot be varied. The turbine model assumes complete separation (*SSL* = 1) between the scrolls. The only way to adjust the degree of separation is by varying the diameter of the orifice simulating cross-flow between the scrolls. Figure 12 illustrates the impact of the cross-flow diameter on flow capacity (left) and separation level (right) in GT-Power.

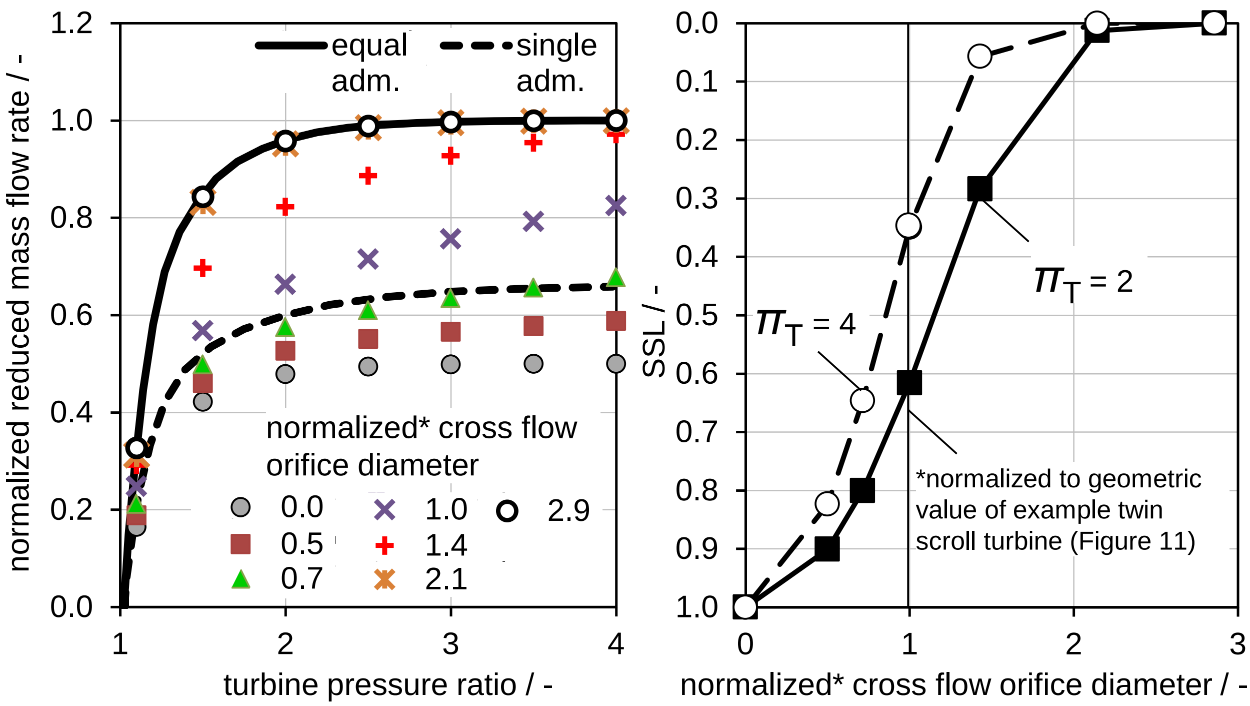

Figure 12 Flow capacity (left) and Scroll Separation Level (right) as a function of the cross-flow diameter in GT-POWER

The actual flow capacity is normalized to the maximum flow capacity at equal admission. The actual cross-flow diameter is normalized to the diameter derived from the geometrical correlation. As expected, the flow capacity increases whereas the separation level decreases when enlarging the cross-flow diameter. In case the cross-flow diameter exceeds the geometrical value by more than 40 % the model behaves like a mono scroll turbine (*SSL* = 0). Using the geometrically determined cross-flow diameter results in the flow capacity being too high and the separation level being too low compared to the measurement. A reduction in diameter of approximately 30 % leads to best compliance with the measured separation level (not with the efficiency characteristic though!). One noticeable fact is the strong dependency of flow capacity and separation level from the turbine pressure ratio. This implies that with the reference method the correct prediction of double entry turbine behavior is only possible in exceptional cases even when the separation level is known from measurements.

5.1.2 Extended Modeling of the Double Entry Turbine

The doubly entry turbine modeling method introduced by Brinkert ([4], [6]) is extended by the implementation of cross-flow between the scrolls and the representation of the performance characteristics as a function of the Mass Flow Ratio *MFR*. The model is based on the performance maps that cover the operating range of the turbine relevant for engine operation. For this, the comprehensive measurement database gathered within the FVV research project “Extended Turbine Mapping” is used. These maps obtained from hot gas test bench measurements are physically extended over turbine pressure ratio, so as to cover the whole operating range.

The model consists of three individual turbine objects: Two turbine objects act on a common shaft together with the compressor and represent the flow characteristic of one scroll. Multiple performance maps for different values of *MFR* are stored in each of these turbine objects. The modeling strategy is similar to that applied for VGT turbines, where in this case the VGT rack position is substituted by the *MFR* value. As opposed to VGT turbines the rack position is not externally controlled, but results from the states (pressures and mass flows) in the individual scrolls during the engine cycle.

The third turbine object connects both turbine scrolls in order to simulate cross-flow based on measured data (see Figure 13). The idea of modeling the cross-flow behavior with an additional turbine object goes back to discussions within the FVV research project “Extended Turbine Mapping” of Lückmann and Brinkert [19]. The maps in this turbine object allow regulating the amount of cross-flow as a function of the flow condition. In the region where $0 < MFR < 1$ both scrolls are completely separated and there is no mass flow through the third turbine object. In case *MFR* is exceeding this value range, the corresponding maps are chosen and the cross-flow mass flow is calculated depending on the scroll pressure ratio.

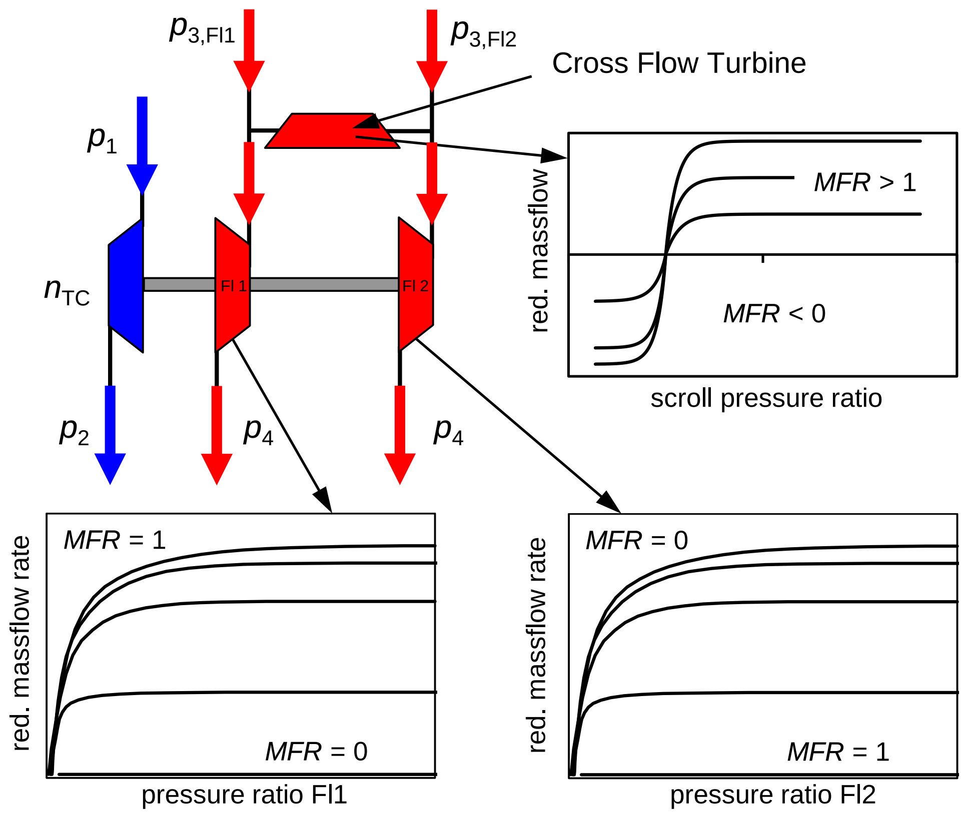

Figure 13: Extended Modeling approach for the double entry turbine in GT-Power

Different approaches for regulating the maps have been investigated. The method based on investigations by Brinkert [19] appears to be most stable numerically especially in conjunction with the here introduced cross-flow modeling approach. For this reason it is applied in the current study. The flow condition is determined depending on the reduced turbine speed and the pressure ratio in each scroll and the corresponding maps are chosen in the turbine object. Calculating the reduced turbine speed in the engine process simulation is done analogous to the measurement process with the enthalpy averaged temperature (see Eq. 2)

The temperature which is used to derive the reduced turbine speed is determined according to Eq. 2 based on the crank angle resolved mass flow rates, specific heat capacities and temperatures in the individual scrolls. The third turbine object replaces the orifice simulating interaction between the scrolls. Contrary to the orifice approach measured physical maps are stored as a function of the *MFR* value.

With this method it is possible, for the first time, to simulate the interaction of the scrolls in a physical way based upon measured test bench data. This allows different hardware versions ranging from low to complete separation to be simulated and evaluated.

5.2 Engine Model

The base model for the conducted simulation study constitutes a 1.6 l gasoline engine with mono scroll turbine. The performance targets as well as the key parameters of the investigated engine are summarized in Table 1.

Table 1: Specification of investigated engine

Displacement	1.596	l
Bore	79	mm
Injection system	DI	-
Exhaust gas legislation	EU5	
Max. torque	240 @ 1600 1/min	Nm
Max. power	132 @ 5700 1/min	kW
Cam phaser intake/exhaust	60 / 60	° CA
Turbine volute	Mono Scroll	-
Water cooling of turbocharger	yes	-
Boost pressure control	Wastegate	-

5.3 Simulation Boundary Conditions

Based on the engine specified in 5.2 an engine model has been built and calibrated in GT-Power. This model was then extended in two versions with the double entry turbine modeling approach described above.

In a next step, different versions of double entry turbines have been created. The turbine characteristics are described by using generic performance maps. These generic maps are derived by extrapolating measured data considering physical correlations. The motivation for using generic maps arises from the possibility to not only scale turbine size (flow capacity), but also the level of separation between the scrolls. This especially incorporates the implications of the *SSL* parameter on the efficiency characteristic of the turbine.

The following boundary conditions have been applied for the simulation study:

- Turbine mass flow and efficiency maps at single admission are identical for both scrolls, basically meaning a symmetric turbine has been used
- The maximum permissible turbine inlet temperature is set to 950 °C and is independent of the tongue length
- Turbine inlet temperature is controlled via the combustion air/fuel ratio
- Valve timing and valve lift profiles are optimized individually for each investigated version
- Valve lift is scaled depending on event length under the assumption of constant valve acceleration
- Rated power of the engine is kept constant
- Gas exchange losses at rated power are calibrated to equal the base engine value for all double entry turbine versions
- Residual gas fraction within the cylinder does not exceed the base engine value after valve timing optimization.
- Combustion is kept constant in terms of burn duration and center of combustion (location of 50 % burned mass fraction)

The impact of different turbocharger design parameters on the resulting performance characteristics is physically evaluated by using the FEV turbocharger database. This empirical data comprises measurements of more than 150 turbochargers from 14 different turbocharger suppliers (Uhlmann et al. [20]). With this approach it is possible to determine characteristic performance parameters (e.g. efficiency, flow capacity, friction losses, scroll separation level (*SSL*)) depending on design parameters like wheel diameter, A/R or length of the separating wall between the scrolls. Furthermore, this procedure ensures the correct scaling of the turbine for the subsequent matching process.

Figure 14 shows the impact of the turbines Scroll Separation Level (*SSL*) on peak efficiency at equal admission. This description corresponds to a vertical slice in Figure 2 for a given turbine wheel diameter.

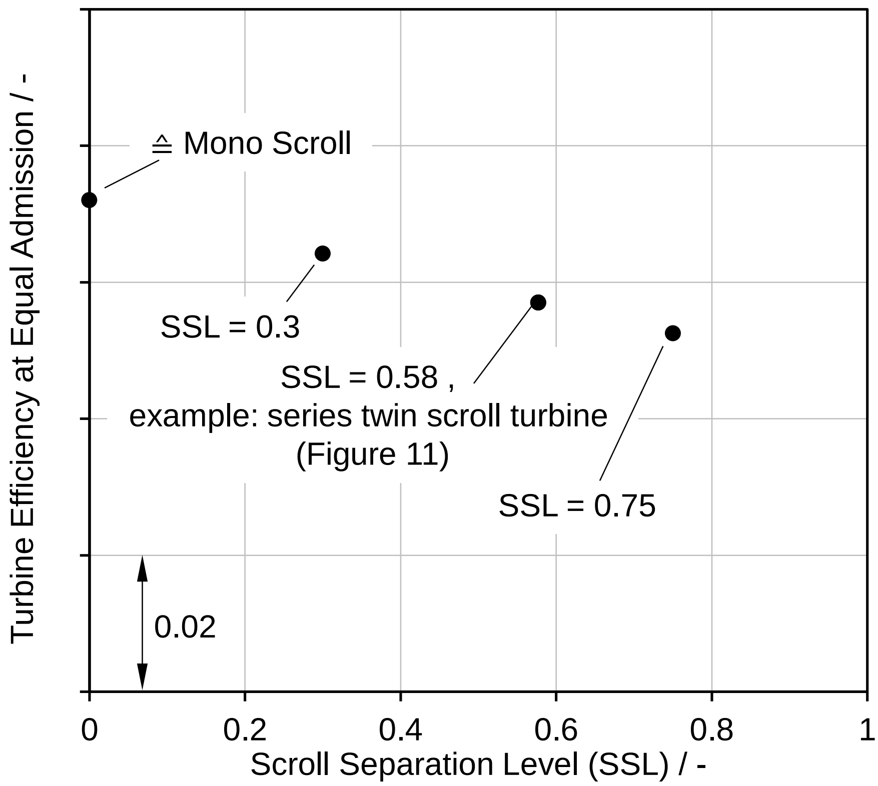

Figure 14: Maximum turbine efficiency vs. scroll separation level for twin scroll turbines

Turbine peak efficiency deteriorates with increasing degree of separation (equivalent to increasing *SSL*) as a result of the flow losses occurring at the separation wall between the scrolls. An *SSL* value of approximately 0.58 corresponds to the twin scroll turbine depicted in Figure 11. Here, the distance between tongue (separation wall) and turbine wheel is reduced to the minimum extend permitted by manufacturing complexity and thermo-mechanical boundaries. The ratio of single to equal admission efficiency as a function of the turbines separation level is shown exemplarily in Figure 15 for two different turbine pressure ratios. The observed trends are based on data of a number of twin scroll turbines measured on FEV’s hot gas test bench.

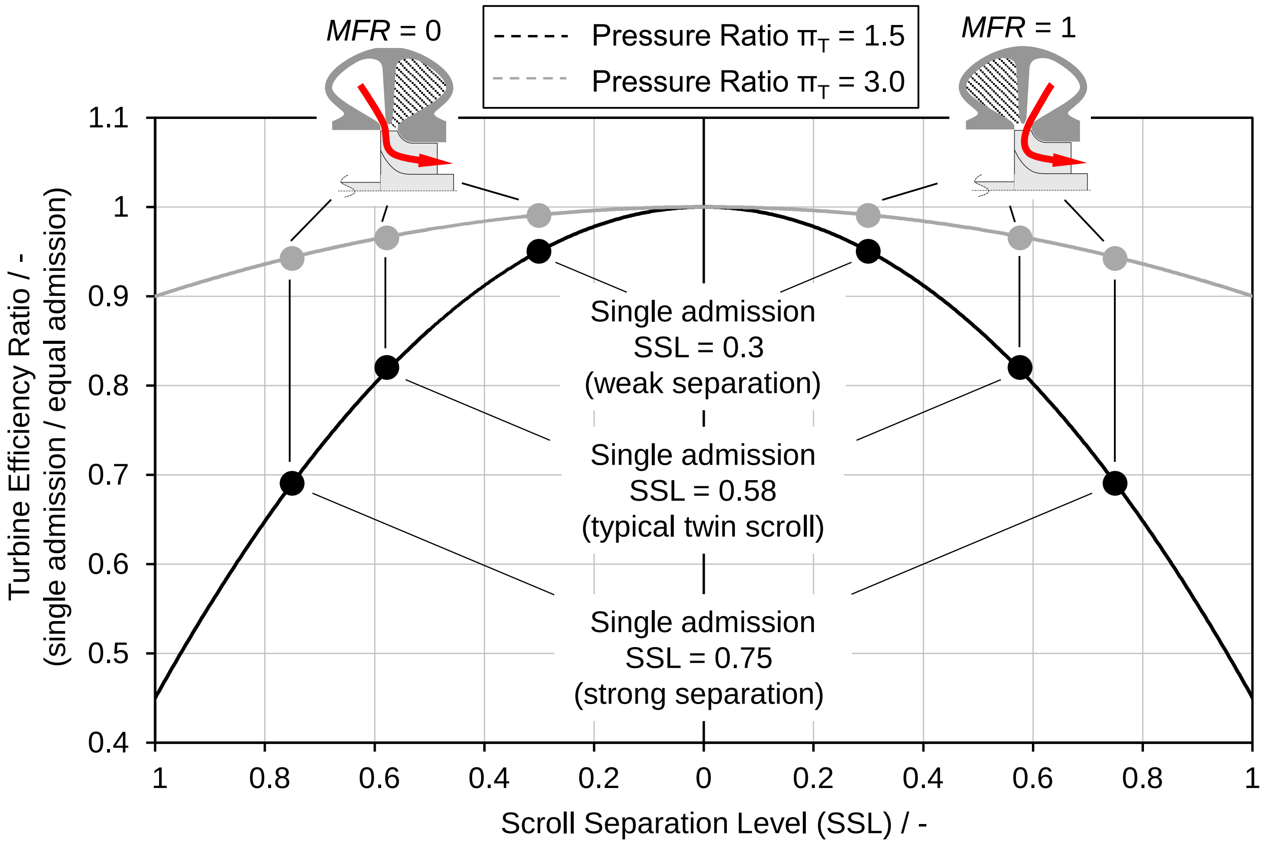

Figure 15: Turbine efficiency for twin scroll turbines at single admission vs. scroll separation level at constant turbine pressure ratio of 1.5 and 3.0, normalized to efficiency at equal admission

The efficiency at equal admission (*MFR* = 0 or 1) decreases with increasing separation at a pressure ratio of 1.5. Looking at a pressure ratio of 3.0 this trend is considerably alleviated and can actually reverse at even higher pressure ratios (upwards opened parabola).

The correlations shown in Figure 14 and Figure 15 are included in the generic maps which serve as input to the engine process simulation within GT-Power. The generic maps are based on FEV's database which comprises more than 150 turbochargers from 14 manufacturers. Using generic maps enables to design a virtual turbocharger as shown by Uhlmann [20]. Turbocharger performance parameters like efficiency, location of peak efficiency within the map or separation level of the turbine can be adjusted within the boundaries of the scatter data.

Different separation levels of the turbine at single admission can also be illustrated in the interaction map as shown in Figure 15.

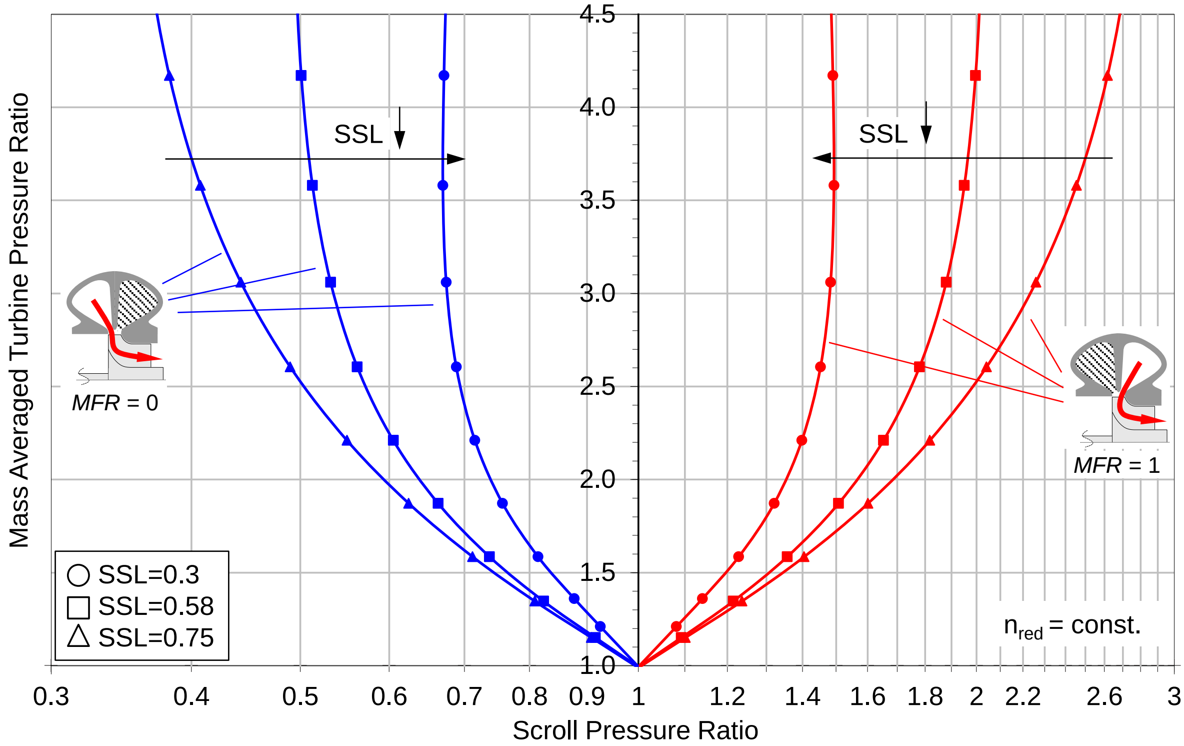

Figure 16: Variation of scroll separation level in the interaction map

SSL = 0.58 equals separation of the scrolls until shortly before the wheel of the series production twin scroll turbine shown in Figure 11. This turbine design features an intermediate spread in the interaction map as expected. The additional version with strong separation (*SSL* = 0.75) exhibits a high spread in the interaction map; accordingly the version with weak separation (*SSL* = 0.3) offers a low spread.

6 Results

In the following chapter simulation results of the full load investigations are presented for six different turbine configurations:

- Base model with mono scroll turbine
- Twin scroll turbine modeled according to reference method
- Extended twin scroll model with *SSL* = 0.3
- Extended twin scroll model with *SSL* = 0.58
- Extended twin scroll model with *SSL* = 0.75
- Extended twin scroll model with *SSL* = 0.75, but efficiency map of *SSL* = 0.58

The size of the different turbines used for the comparison is scaled in such a way that the gas exchange losses at rated power remain constant. Considering additional boundary conditions like a constant turbine inlet pressure is as well possible, but has to be investigated in a future study. The optimization of valve timings, valve lift profiles and turbine size is performed by means of Design of Experiments (DoE). The matching and optimization process is described in chapter 6.1. After that, the simulated effects on engine performance in the low-end-torque region are presented in 6.2. Finally, the full load simulation results are examined in detail.

6.1 Turbine Matching

Matching the correct turbine size depends on the given target engine and the applied boundary conditions which for this study were chosen as constant gas exchange losses at rated power and reduced residual gas fraction compared to the base layout. Considering optimum calibration of valve timing, event length and turbine flow capacity it is possible to achieve the rated power target with equivalent gas exchange losses for all turbine versions. The residual gas fraction is reduced for all double entry turbines compared to the base concept with mono scroll turbine (Figure 17).

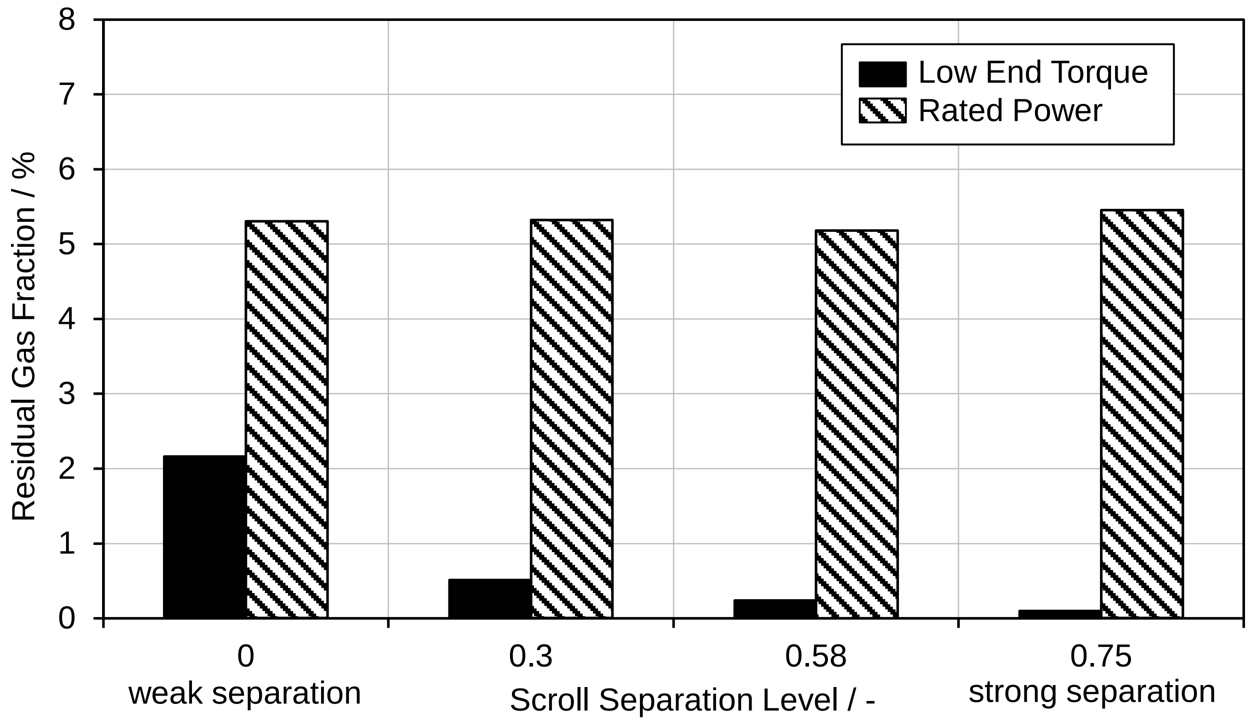

Figure 17: Residual gas fraction at rated power and low-end-torque (LET)

Thus, any negative effects on the engines knock behavior are eliminated. Figure 18 shows the result of the matching process in terms of the flow capacity at equal admission.

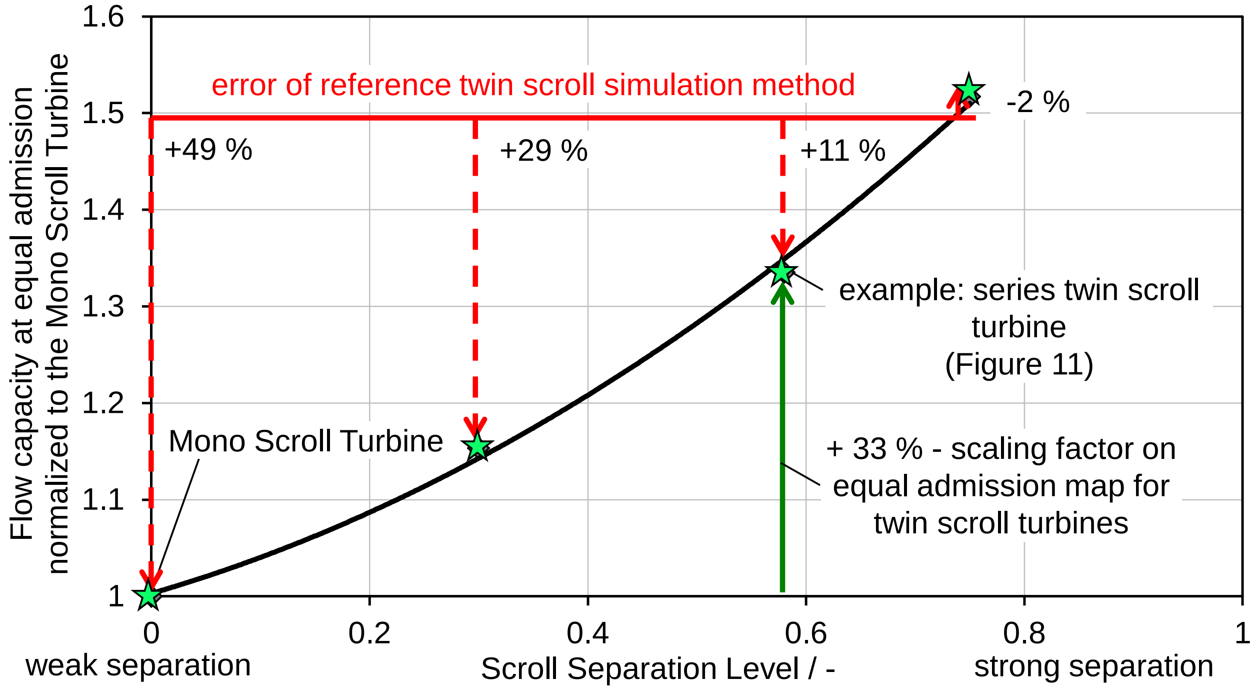

Figure 18: Normalized flow capacity at equal admission against scroll separation level

The results based on the presented extended modeling approach suggest that stronger separation requires the turbine size to be increased. Hence, the flow capacity at equal admission of a state-of-the-art twin scroll turbine (*SSL* = 0.58) has to be approximately 33 % higher compared to an equivalent mono scroll turbine.

The reference modeling approach suggested for implementation in GT-Power does not take into account the level of separation between the scrolls. Instead complete separation is assumed with cross-flow being accounted for by an orifice according to Eq. 14. Without a sufficient measurement database for validating the orifice diameter this approach results in an overestimation of the separation level and hence of the required size of the twin scroll turbine. In this particular example the reference model yields a twin scroll turbine which is matched approximately 11 % to large in size (considering the assumed geometry of the series production twin scroll turbine from Figure 11).

In case of a weaker separation (*SSL* = 0.3) the error in turbine size matching increases to approximately 29 %. In the theoretical borderline case of matching a mono scroll turbine (*SSL* = 0) with the double entry reference model the error amounts to 49 %.

6.2 Low-end-torque Performance

Following the flow capacity layout of the twin scroll turbine, the engine simulation results in the low-end-torque region are presented. Figure 19 shows the estimated low-end-torque speeds of the reference model compared to the extended approach presented before for a typical twin scroll application with separation level of *SSL* = 0.58.

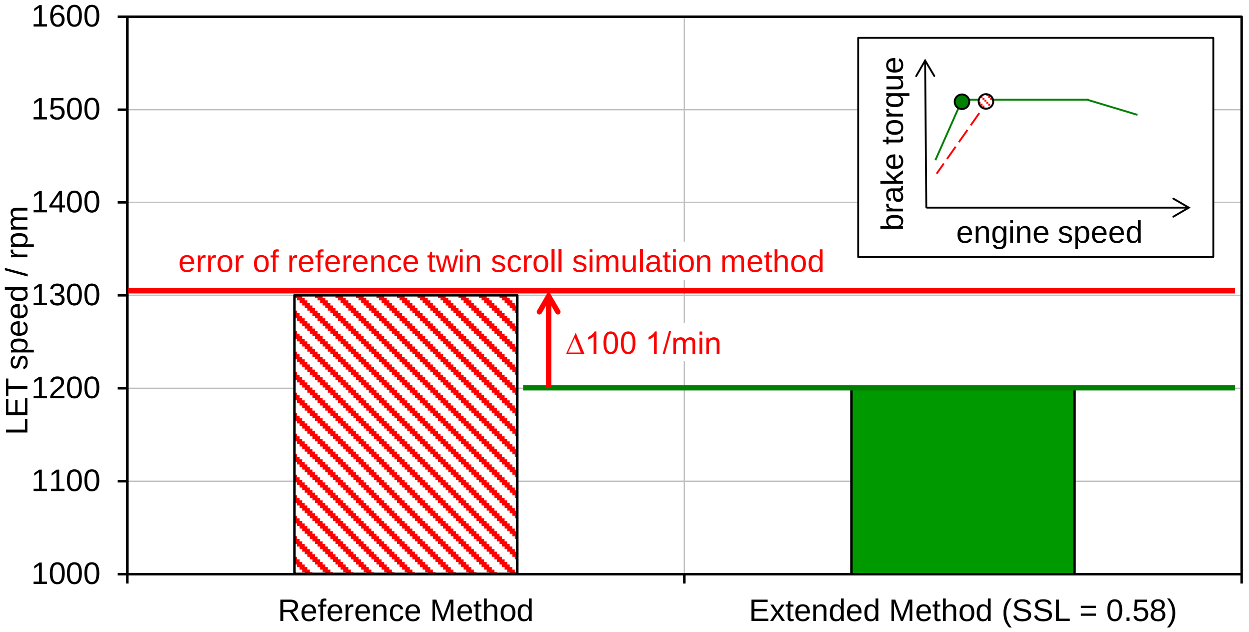

Figure 19: Engine speed at low-end-torque for reference and extended model (at a Scroll Separation Level of *SSL* = 0.58)

The deviation in predicted low-end-torque speed occurring between the different modeling approaches amounts to approximately 100 1/min. This result is in agreement with engine test bench investigations published by Winkler [3] and Schmalzl [2] who observed similar phenomenona (twin scroll turbine layout to large; shift of low-end-torque to higher speeds). This deviation is remarkable considering the fact that the difference in low-end-torque speed arising between mono scroll and the optimum double entry version (*SSL* = 0.44) accounts for only 320 1/min (Figure 19).

A reduction in separation level can be realized by reducing the tongue length (increased distance between tongue tip and turbine wheel) (Björnsson et al. [1]).

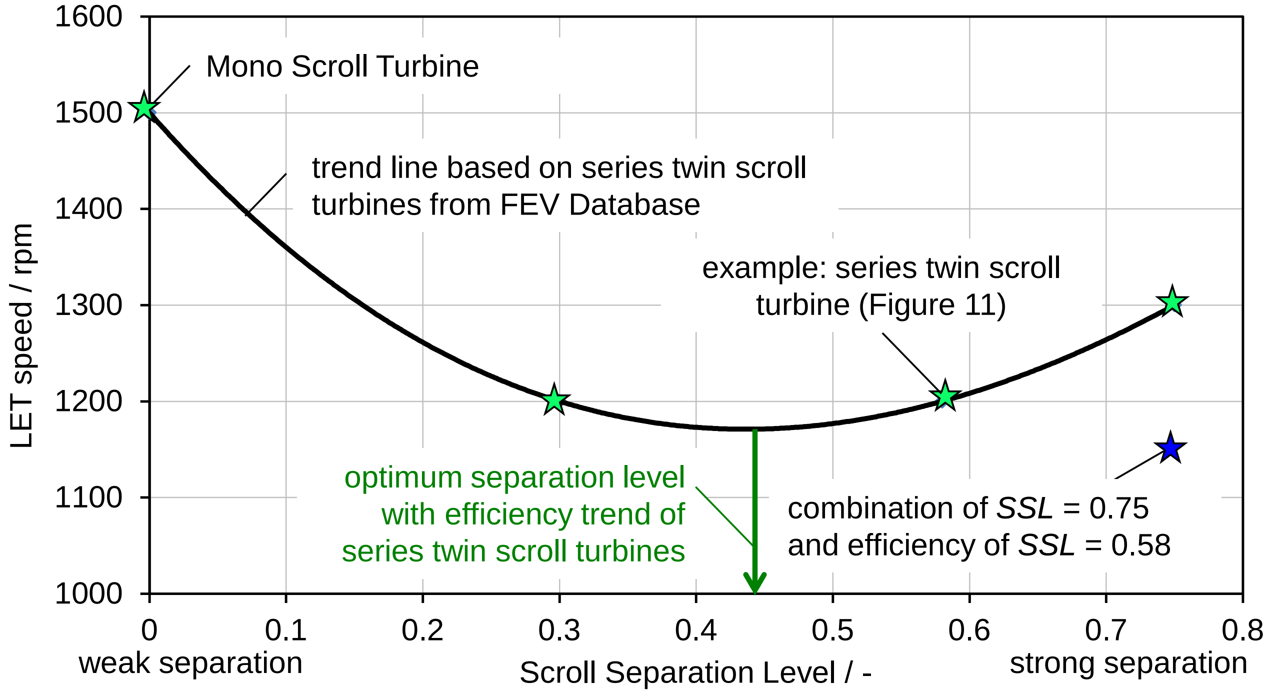

Figure 20: Engine speed at low-end-torque for varying Scroll Separation Levels with extended modeling approach

This conclusion may be surprising, because in previous publications i.e. by Björnsson et al. [1] and Engels [7] an increased level of separation has been suggested. However, it is to be kept in mind that in this study for the first time the turbine behavior has been modeled physically depending on the separation level. The performance maps of the turbine are derived from a number hot gas measurements of series production twin scroll turbines. The trends in the efficiency characteristic depicted in Figure 13 and Figure 14 are derived from that database of measurements. In order to assess the impact of the assumed efficiency characteristic a further turbine version was generated and investigated. This turbine features the flow capacity map of the highest separation level version of *SSL* = 0.75, combined with the efficiency map of the medium *SSL* = 0.58 variant. The results show that increasing the separation level from 0.58 to 0.75 while maintaining the efficiency level leads to an improvement in low-end-torque which could already be achieved at 1150 1/min. This result leads to the conclusion that the development of future double entry turbine concepts should aim at high separation levels without sacrificing efficiency.

6.3 Analysis of Full Load Performance

Figure 21 shows the interaction map of the twin scroll turbine with *SSL* = 0.58. The corresponding single admission maps (*MFR* = 0 and 1) are plotted schematically as boundary curves. Additionally, the turbine operation during one engine cycle at low-end-torque speed as well as rated power is included. This diagram shows that the pressure ratio level and correspondingly also the average pressure ratio is increasing with increasing engine speed. However, the turbine encounters all individual flow conditions (cross-flow, single admission, unequal admission and equal admission) within one engine cycle independent of engine speed.

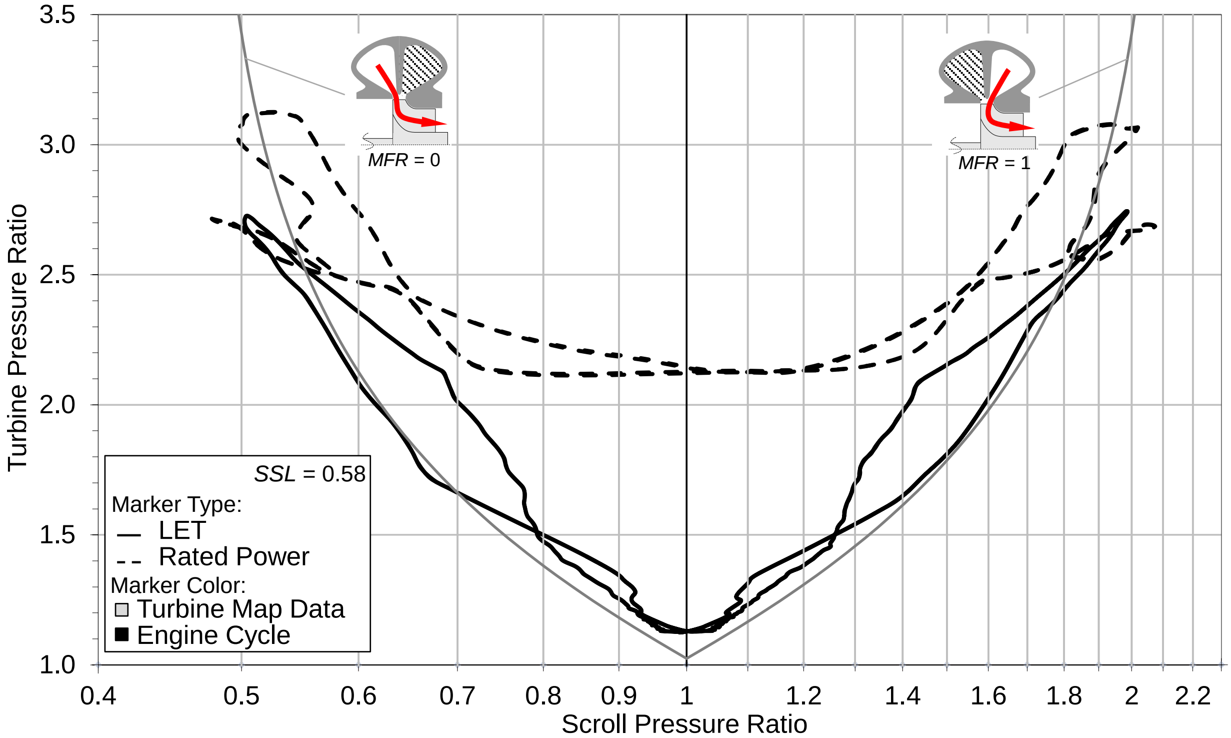

Figure 21: Interaction map including the engine cycle at low-end-torque and the single admission boundary curve for varying separation levels

In chapter 0 it was shown that turbine peak efficiency as well as the ratio of single admission to equal admission efficiency deteriorates with increasing level of separation (see Figure 14 and Figure 15). This observation raises the question why the optimum separation level was determined to be at a value of *SSL* = 0.44 despite the drawbacks in peak efficiency and the apparent operating point shift into areas of lower efficiency. In order to explain this phenomenon the *MFR*-histogram of one engine cycle is plotted as a function of scroll separation level at rated power and low-end-torque in Figure 22.

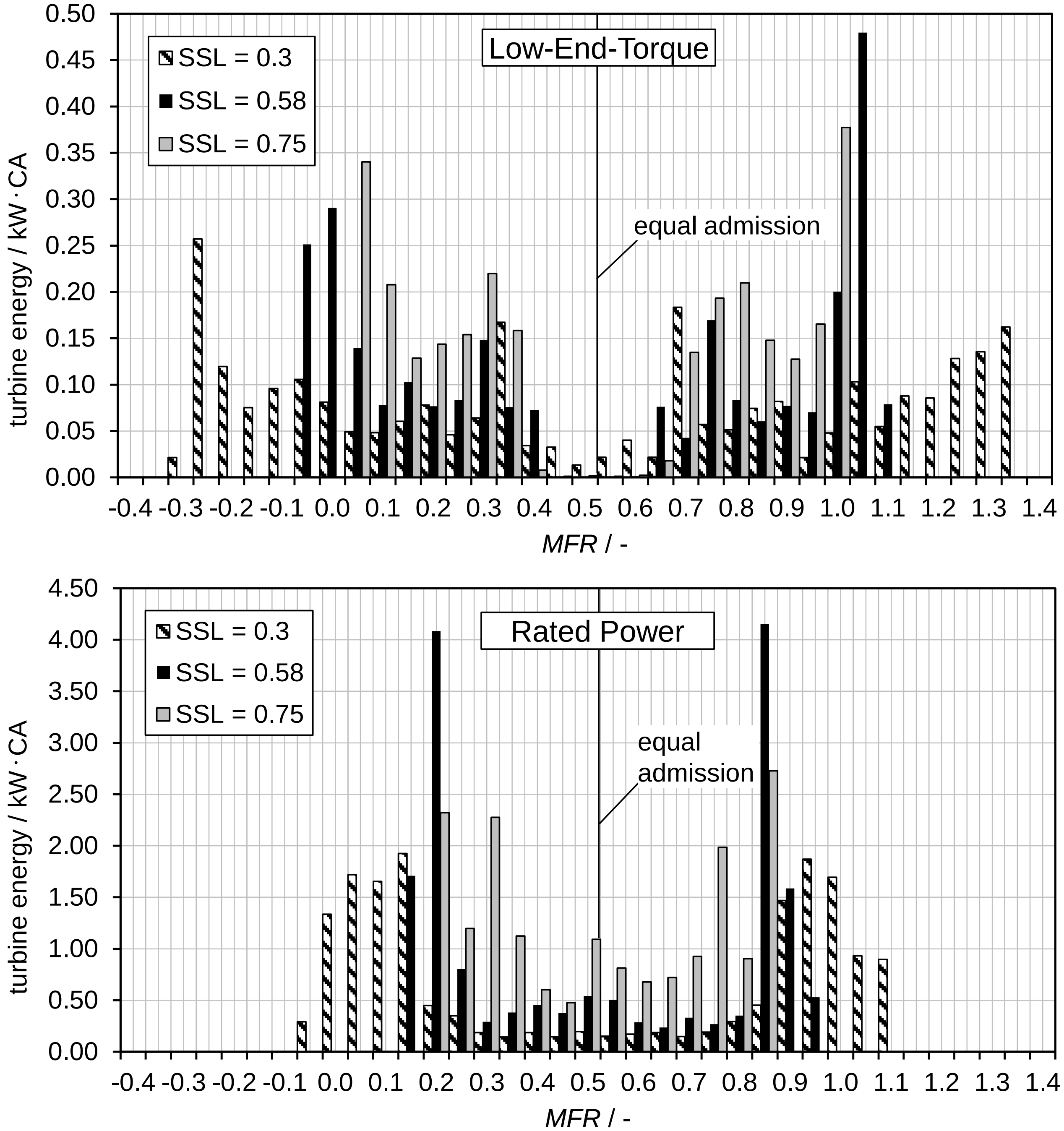

Figure 22: Histogram of *MFR* during an engine cycle as a function of the Scroll Separation Level (*SSL*)

With increasing separation level the scatter of flow conditions (*MFR*-values) occurring in one engine cycle is reduced independent of the engine operation point. Due to this turbine operation is shifted into areas of higher efficiency. A similar phenomenon can be observed when analyzing different engine operating speeds. The flow conditions (*MFR*-values) at rated power are shifted towards equal admission (*MFR* = 0.5) in comparison to the low-end-torque speed.

Figure 23 illustrates the optimized exhaust valve event length for all investigated turbine versions. The event length is normalized to the value of the mono scroll base case for easier interpretation. Increasing *SSL* allows for longer event lengths to be run compared to the base layout while keeping the residual gas fraction constant (rated power) or even slightly reduced (low-end-torque). The longer event lengths of the double entry turbines therefore yield additional potential for fuel consumption reduction in part load operation. The benefits of a prolonged exhaust valve event for boosted gasoline engines in part load and NEDC cycle is shown by Budack [21] amongst others.

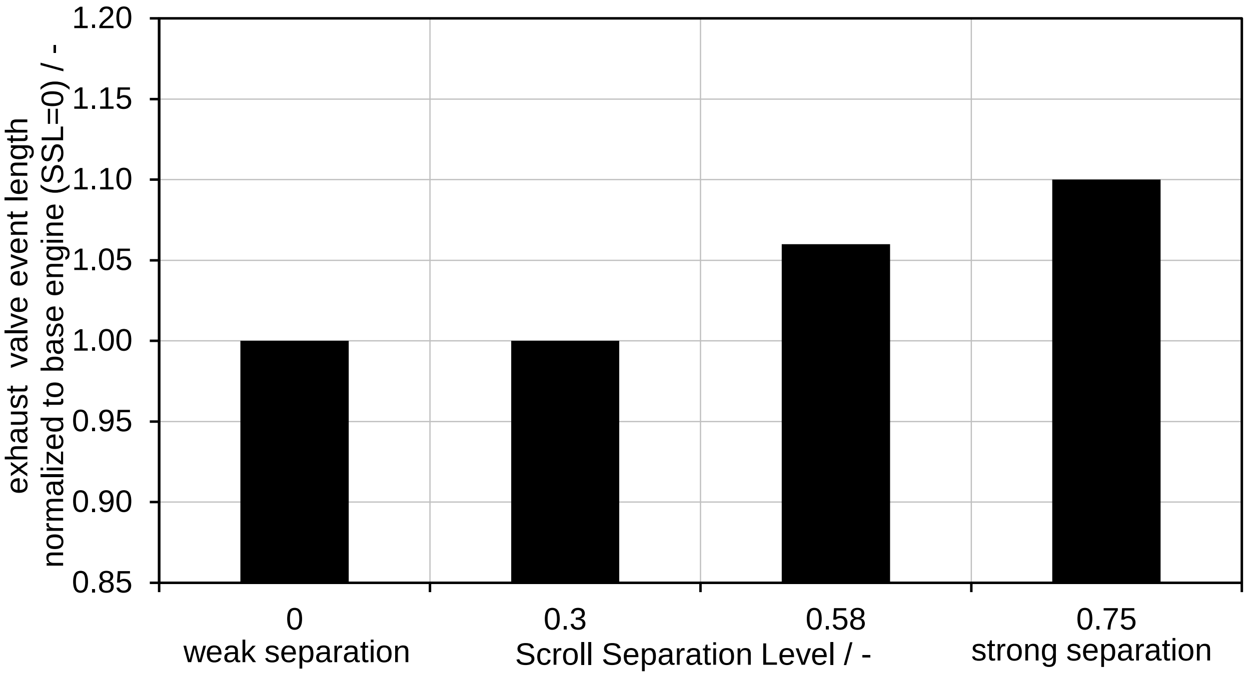

Figure 23: Exhaust valve length normalized to base engine with mono scroll turbine (*SSL* = 0)

An interesting side effect can be observed regarding the simulated thermocouple and gas temperatures at turbine inlet (Figure 24). The thermocouple temperature was used as the reference for the air-fuel-ratio controller in order to not exceed the maximum turbine inlet temperature. Hence, the thermocouple temperature was constant in all simulated cases. The corresponding gas temperature, however, is approximately 80 K lower for the double entry versions compared to the mono scroll version. The gas temperature can normally not be measured directly on the engine test bench. Instead the measured thermocouple temperature is taken for determining the thermo-mechanical limit regarding the maximum permissible turbine inlet temperature.

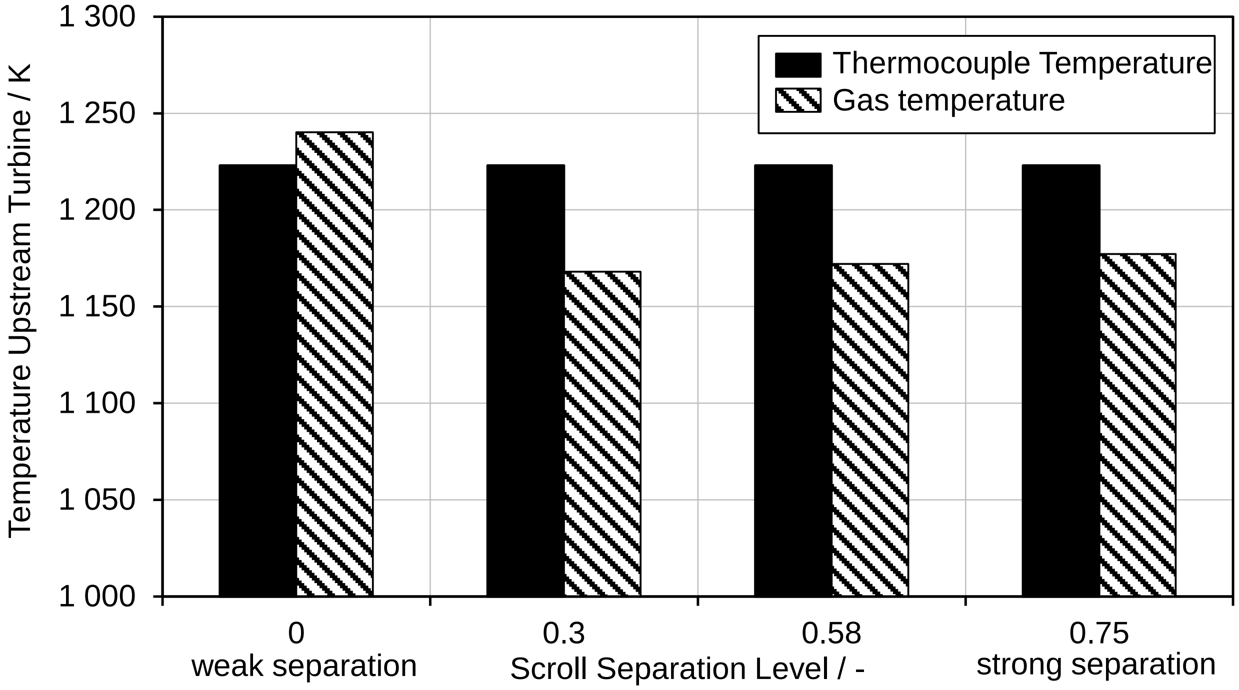

Figure 24: Comparison of simulated thermocouple und gas temperature at turbine inlet as a function of scroll separation level.

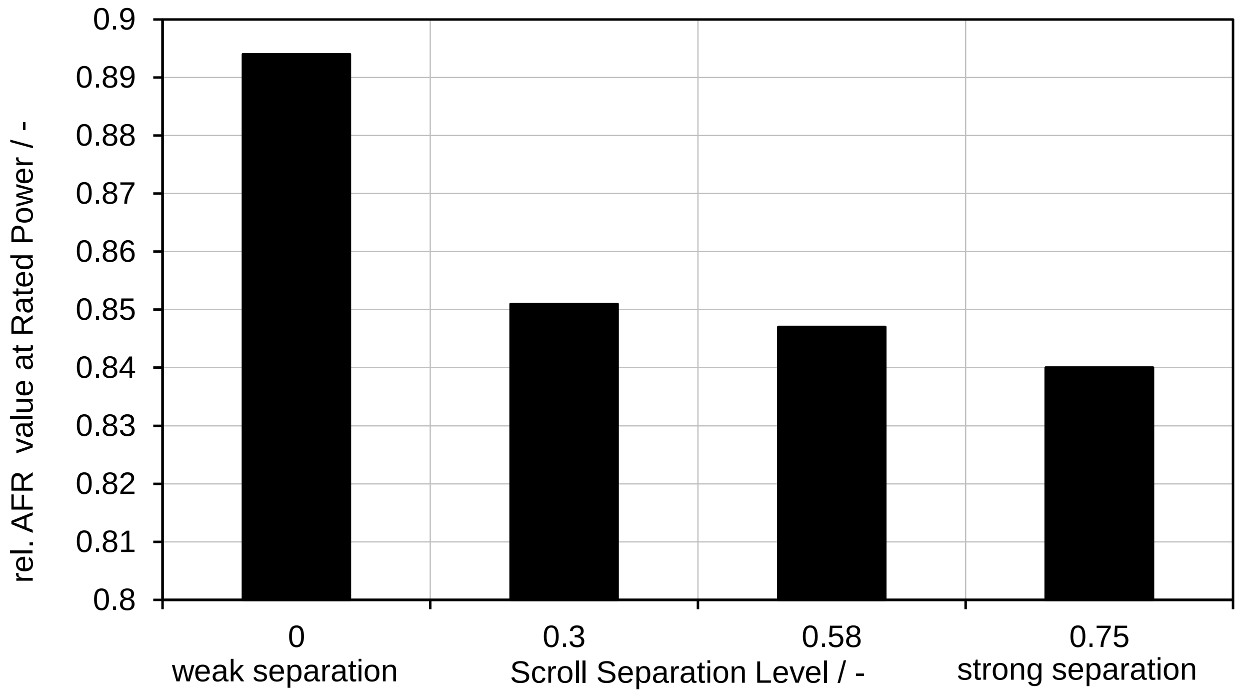

Figure 25: Relative AFR at rated power as a function of the Scroll Separation Level

In the present study this deviation between gas and thermocouple temperature leads to an increased fuel enrichment requirement at rated power of approximately 9 % points (Figure 25). This translates into a fuel consumption penalty of about 16 g/kWh at rated power. If the gas temperature is controlled to be constant at turbine inlet the fuel consumption difference reduces to about 7 g/kWh. These results demonstrate the need for further investigations on the impact of the exhaust gases flow characteristic (e.g. amplitude and frequency of the blow down pulses) on the measured temperature at turbine inlet. Significant fuel consumption potential can be employed by determining the “correct” exhaust gas temperature and required enrichment.

Following key facts can be summarized: The presented extended approach for modeling double entry turbine behavior allows the numerical analysis of turbines with different separation levels. Depending on the applied boundary conditions (degree of boosting; exhaust manifold volume; valve train variability) the optimum turbine (flow capacity; separation level) can be matched for a given engine application.

7 Summary and Outlook

The layout of double entry turbines in 1D engine process simulation is complex and requires physical modeling of the complete system consisting of engine and turbocharger in order to ensure a successful matching of the subsystems.

In this work a method for measuring the performance of double entry turbines on FEV’s hot gas test bench was presented. This procedure allows replicating the turbines flow conditions occurring in pulsating engine operation with steady state conditions present on the hot gas test bench. The method is based on using a double burner setup (one burner supplying one turbine scroll) and an additional parameter introduced in order to describe the flow condition (*MFR* = Mass Flow Ratio).

A novel extended approach for integrating the measurement data from the hot gas test bench into 1D engine process simulation was presented. Compared with the existing approach this allows to physically describe the behavior of double entry turbines in engine relevant operation depending on the level of separation between the scrolls. Applying this method enables a precise layout and matching of the turbine depending on the separation for a certain target engine. Accordingly, the required number of turbocharger prototypes and experimental effort on the engine test bench can be considerably reduced.

The deviation between simple approach and extended FEV modeling accounts for 11 to 29 % in flow capacity and approximately 100 1/min in achieved low-end-torque speed (considering the boundary conditions applied in the present study). Using the simple approach almost invariably results in a too large turbine matching.

For the investigated engine concept (1.6 l four cylinder with direct injection and variable intake and exhaust valve timing) and defined boundary conditions an optimum separation level between the scrolls of *SSL* = 0.44 has been determined. This value is well below the separation level of the series production twin scroll turbine presented in this paper (*SSL* = 0.58) and may design wise be achieved by increasing the distance between tongue and wheel. The improvement in low-end-torque speed compared to the base mono scroll concept amounts to 300 1/min at reduced residual gas fraction in the cylinder. This result is especially remarkable considering that the base engine achieves its low-end-torque already at 1500 1/min. Increasing the separation level to *SSL* = 0.75 while maintaining the efficiency characteristic of *SSL* = 0.58 shows the potential to reduce the low-end-torque speed even further to 1150 1/min. This outcome implies that turbine development should tend towards concepts with high separation level combined with favorable efficiency characteristics.

8 Acknowledgements

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9 Abbreviations and Symbols

9.1 Abbreviations:

AFR	Air-Fuel-Ratio
A/R	Area/Radius Ratio
BMEP	Brake Mean Effective Pressure

BSFC	Brake Specific Fuel Consumption
CFD	Computational Fluid Dynamics
FL	Flow (Scroll)
LET	Low-End-Torque
MFR	Mass Flow Ratio
PMEP	Pumping Mean Effective Pressure
SSL	Scroll Separation Level
sMaR	Scroll Mach Number Ratio
VTG	Variable Turbine Geometry
VVL	Variable Valve Lift
VVT	Variable Valve Timing

9.2 Variables:

c_p	Specific Heat Capacity at Constant Pressure
d	Diameter
h	Enthalpy
$\dot{m}$	Mass Flow Rate
n	Speed
p	Pressure
r	Radius
T	Temperature
$\bar{T}$	Enthalpy Averaged Temperature
Π	Pressure Ratio
η	Efficiency

9.3 Indices

1	Compressor Inlet
2	Compressor Outlet
3	Turbine Inlet
4	Turbine Outlet
C	Compressor
Is	Isentropic
T	Turbine
TC	Turbocharger

ma	Mass Averaged
tot	Total
st	Static
FL1	Flow 1 (Scroll 1)
FL2	Flow 2 (Scroll 2)

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