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Sequential turbocharging as performance enabler for demanding marine propulsion engines

Turbochargers & Air/Exhaust Management

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ABSTRACT

Large high-speed diesel engines are used for a wide range of applications which include power generation, mining machinery, locomotives, and marine propulsion. The operational profiles and user values of these individual applications are just as diverse - with demanding mining machinery operating at variable speed over an extreme number of load cycles or emergency backup generators running at constant speed and high rated power or demanding marine propulsion units.

This paper discusses how sequential turbocharging can enable powerful marine applications with highly rated torque curves and strong dynamic response. For marine propulsion engines, performance over a wide range of loads and speeds is requested to satisfy all conditions a vessel could meet during its operation, varying from constant speed to quick acceleration requirements in different sea conditions with capabilities of towing or carrying different tonnage. Translated to turbocharging requirements, this means high compressor pressure ratio capabilities corresponding to higher power density together with wide range of volume flow requirements to fit various engine speeds. Inherently, high compressor pressure ratio capabilities come with a trade-off related to a narrower range in terms of volume flow. Parallel sequential turbocharging allows to overcome this trade-off by activating sequentially multiple single TC with high pressure ratio capabilities, thus widening the volume flow range to match the vessel operation requirements. At the same time, the effective turbine flow area changes in the same way and hence maximizes the transient response of the turbocharging system.

By means of engine simulation, this paper highlights the advantages of sequential turbocharging concepts for high-speed diesel engines used in demanding marine propulsion applications compared with standard turbocharging concepts. Additionally, it will address some specific technical challenges associated with this turbocharging concept and discusses how these are addressed with the new multi-purpose turbocharging platform A101-R.

1 INTRODUCTION

Performance and reliability are the chief considerations in high-speed diesel engines employed in performance boats such as: fast ferries, commercial and defense vessels, high-performance fishing boats, specialized marine workboats, and passenger yachts. Vessels like these require a dynamic power capability to be able to transition quickly between low-speed maneuvering and high-speed cruising. This calls on a number of critical engine features to be available simultaneously: high power density in terms of both volume and mass; good transient response; high engine efficiency; and optimal performance over a broad range of operating conditions [1].

To enable both swift vessel acceleration and high continuous speed, high engine power is required over a broad range of engine rotational speeds. In marine applications, power density is particularly critical to enabling high power output without adding excessive weight, where increased weight reduces speed capability and fuel efficiency. In addition, power density enables a reduced engine footprint in the boat hull. Given the limited space on high-performance boats, with priority given to passengers, cargo, and equipment areas, power density helps minimize space required for the engine room. For high-performance vessels, engine brake mean effective pressure (BMEP) above 30 bar can be required.

Marine engines operating along the propeller curve run at different rotational speeds corresponding to various vessel speeds. The propeller law reflects the power required to overcome the hydrodynamic resistance of the propeller and hull at varying speeds [2]. In steady state operation, this operation line also depends on the carried load and environmental conditions (rough seas, strong currents, heavy winds, fouled hull), for which it can be shifted left or right of the ideal fixed pitch propeller curve, depending on whether the factors are facilitating or against the vessel displacement.

During acceleration, the engine moves outside its steady-state operation, causing deviations leading to additional power output for a given engine speed. In terms of turbocharging requirements, low engine speed during acceleration leads to a high pressure ratio from low volume flow, whereas high engine speed during steady state operation results in high volume flows for the turbocharger.

In non-sequential turbocharging systems, the compressor is specified to cover standard operation along the propeller curve, with sufficient surge margin to cover the most loaded steady-state operation as well as acceleration requirements. As high power density is required in marine engines, the compressor must be specified to enable high engine speed at high engine loads. For this, the typical high-pressure-ratio compressor maps are utilized in for single-stage turbocharging. However, that comes with the compromise of a narrow flow range with limited pressure ratio capabilities at low volume flows, and thus limits acceleration capability.

Sequential systems with the cut-in/out of turbocharger (TC) units overcome this compromise by matching the volume flow requirements at different engine speeds to the volume flow capabilities of the high pressure ratio compressor stage. This system improves vessel acceleration through higher torque capabilities from low to high engine speeds, thus fulfilling the requirements for performance vessels [3].

The following sections delve deeper into sequential turbocharging systems, focusing on performance and control optimization through simulation-based analysis. The specific challenges associated with such systems will be derived from the simulation results and translated into turbocharging requirements. Further considerations for the safe and reliable operation of such systems are also discussed.

Finally, the A101-R turbocharger is presented. This turbocharger was specifically designed to offer advanced capabilities tailored to enhance performance and reliability for demanding high-speed diesel applications such as sequential turbocharging.

2 PERFORMANCE AND CONTROL OPTIMIZATION

2.1 Advantages of sequential turbocharging

Engine simulations were performed on a 16-cylinder high-speed Diesel engine (typical bore size of 170 – 190 mm) with a maximum BMEP above 30 bar. This engine is equipped with 4 A131-R turbochargers together with the wastegate to cope with the air requirement at different speeds. The engine operating map is shown in Figure 1.

The sequential turbocharging system with a 4-3-2 TC arrangement—using 4 A131-R TC at high speeds and reducing to 3, then 2 as speed decreases — was later applied to this engine (as shown Figure 16). The selection of the TC switch-over point in the sequential turbocharging system followed a structured approach based on the following criteria:

1. Maintain the minimum relative air-fuel ratio above the smoke limit

2. Ensure the turbocharger speed remains within safe operational limits
3. Optimize switch-over speed to achieve the best fuel consumption performance

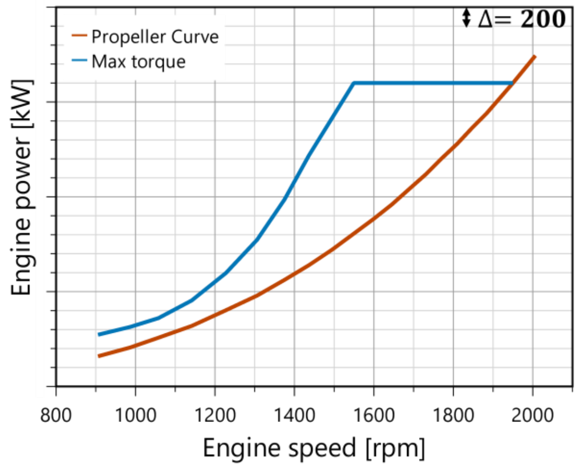


Figure 1. Engine operating map.

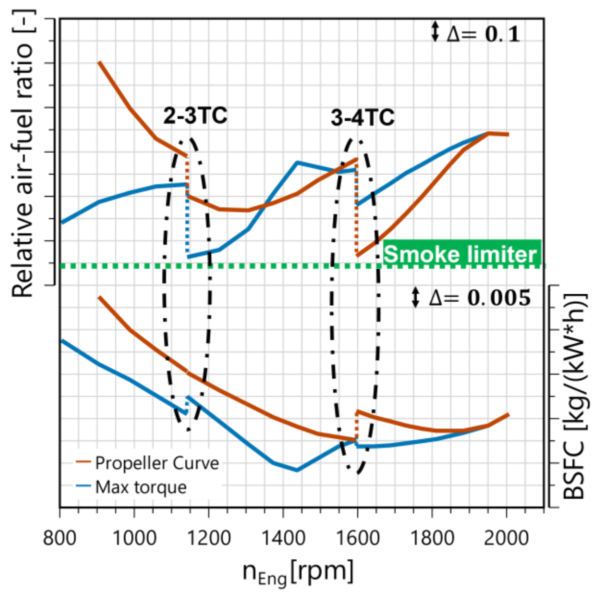


Figure 2. The optimal switch-over points.

It is important to note that the impact of turbocharger cut-out on the combustion process was also considered. Specifically, the shorter combustion duration was evaluated using advanced combustion models integrated into Accelleron's in-house simulation software [4][5]. The maximum torque and propeller curves were considered to determine the optimal switch-over speed following the above-mentioned criteria. The brake specific fuel consumption (BSFC) and relative air-fuel ratio results are shown in Figure 2, indicating optimal switch-over points at approximately 1150 rpm and 1600 rpm, corresponding to 20% and 55% load along the propeller curve, respectively.

This study is divided into two main parts:

1. Comparison of the wastegate and sequential turbocharging approaches, where both the steady-state performance and transient acceleration process will be studied to understand the potential of the sequential turbocharging system.
2. Investigation of the control strategies for the TC switch-over processes, where placement and actuation of the control valves will be analyzed and discussed.

2.1.1 Enhanced low-end torque

Figure 3 compares the operation lines along the maximum engine torque curves for the wastegate and sequential turbocharging approaches. The results show that a further increase in engine torque is not feasible in the wastegate approach without compromising a safe surge margin, especially at low speeds, potentially pushing the operation line into the surge region. With the sequential turbocharging approach, the active compressor experiences a significant increase in volume flow rate due to the turbocharger cut-out, shifting the operation line toward the choke side and farther from the surge line, while staying at comparable turbocharging efficiency levels. This enables an increased engine torque, particularly at partial loads or low speeds.

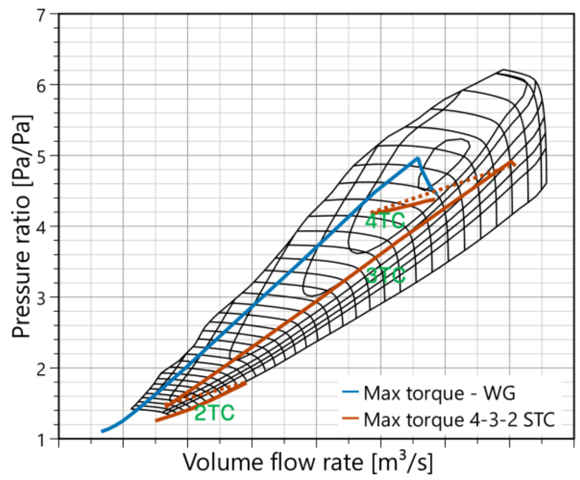


Figure 3. Operation lines of the wastegate and sequential turbocharging approaches along the maximum torque curve.

The potential maximum torque that can be achieved with the sequential turbocharging approach is shown in Figure 4, labeled as the “max torque limit”. It is possible to see that compared with the wastegate approach, sequential turbocharging can significantly extend the engine operating range and enhance low-end torque. Furthermore, as illustrated in Figure 5, the 4TC–3TC–2TC sequential operation eliminates turbocharging limitations on engine performance, enabling the engine to achieve its maximum feasible BMEP across the entire speed range while maintaining a sufficient surge margin. This is demonstrated in Figure 6, which shows the operation line for the maximum feasible engine torque curve on the compressor map for different approaches.

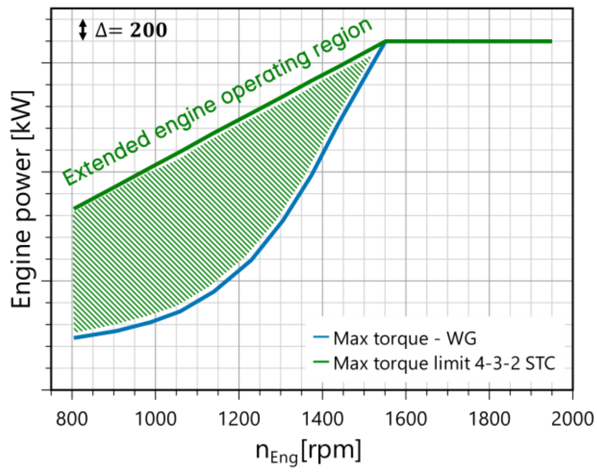


Figure 4. Engine maximum torque limit for wastegate and sequential turbocharging approaches.

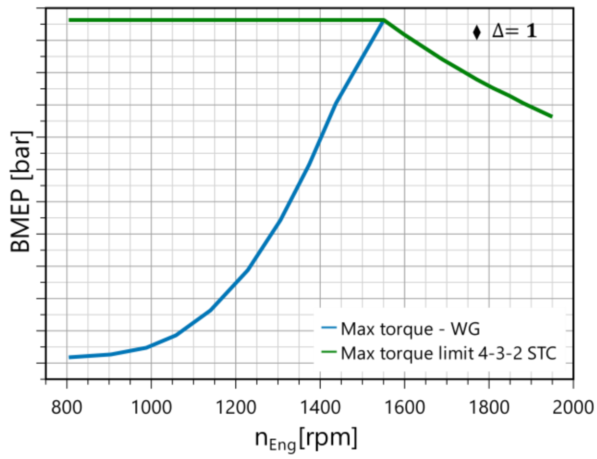


Figure 5. Engine BMEP limit for wastegate and sequential turbocharging approaches.

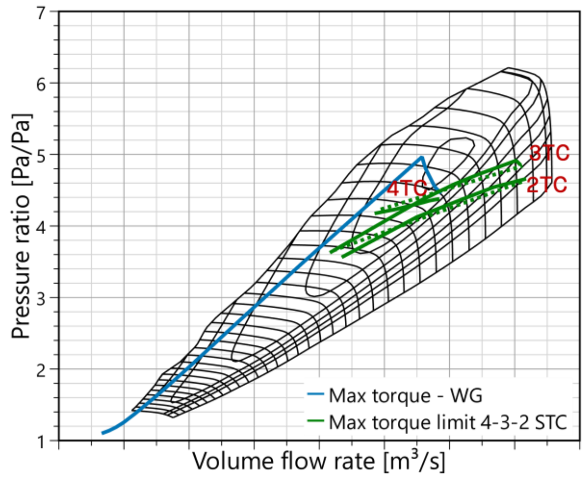


Figure 6. Operation lines of the wastegate and sequential turbocharging approaches along the maximum torque limit.

2.1.2 Improved acceleration capabilities

Transient simulations were conducted to compare the acceleration capabilities of the wastegate and sequential turbocharging approaches. In this work, the acceleration from 5% to 50% engine load was considered, assuming such a process takes place along the engine propeller curve to facilitate the numerical comparison between two turbocharging approaches. In the sequential TC setup, the turbocharger cut-in process was also included, with the additional turbocharger activated at 20% load—identified as the optimal switch point based on previous assessments in Figure 2. The effective turbine area (SEFFT) is defined as the area of an equivalent nozzle with isentropic flow (Ψ denotes the nozzle flow function) and can be related to the widely applied definition of the reduced mass flow ($\dot{\boldsymbol{m}}_{\boldsymbol{red,T}}$) [6], which serves as an indication of turbine power:

$$SEFFT = \dot{m}_{red,T} \cdot \frac{\Psi(PIT)}{\sqrt{R_{exh}}} \tag{1}$$

The equivalent turbine effective area, defined as the sum of the effective areas of all activated turbines and wastegate openings, is shown as a function of relative load along the engine propeller curve in Figure 7 for both approaches. It is possible to see that up to 50% load, the sequential turbocharging approach exhibits a much smaller equivalent turbine effective area, as fewer turbochargers are in operation. This could enhance the acceleration process, which will be discussed further in this section.

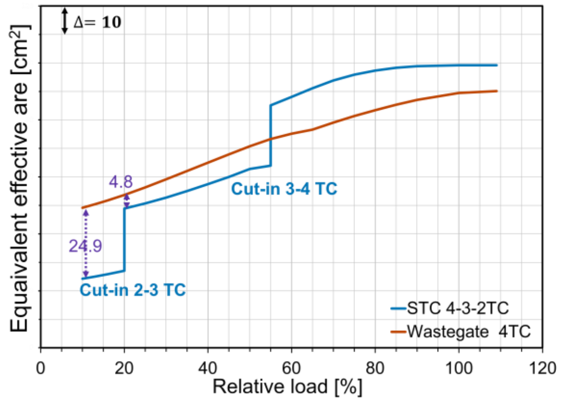


Figure 7. Equivalent turbine effective area variation along the engine propeller curve for sequential turbocharging and wastegate approach.

During the acceleration process, the injected fuel quantity per engine cycle was governed by three key factors:

1. Engine speed control: A PID controller adjusts the fuel mass based on the difference between the current and target engine speeds.
2. Smoke limit control: The relative air-fuel ratio is maintained above the smoke limit (set at 1.5 in this study), which together with the air mass flow rate at the compressor outlet, determines the maximum injected fuel mass per cycle.
3. Peak cylinder pressure control: The peak cylinder pressure remains below the engine's mechanical limit. However, this factor does not influence the current simulations, as the acceleration range from 5% to 50% load ensures the peak cylinder pressure stays well within the safe limit in all scenarios.

The relative air-fuel ratio, injected fuel mass per cycle, intake receiver pressure, and relative engine load during the acceleration process for both sequential turbocharging and wastegate approaches are illustrated in Figure 8-11. The process can be divided into three distinct phases:

- Phase 1 – from the start of acceleration to the start of TC cut-in: For both approaches, acceleration is constrained by smoke limit control, with both operating at the relative air-fuel ratio limit, as illustrated in Figure 8. However, the sequential TC setup could achieve significantly higher intake receiver pressure (Figure 10) due to the smaller turbine effective area, as fewer turbochargers are in operation. This allows for higher amount of fuel injection (Figure 9) and therefore a faster acceleration (Figure 11) compared to the wastegate approach.
- Phase 2 – immediately after TC cut-in: The turbocharger cut-in reduces the intake receiver pressure (pressure ratio) and air mass flow rate at the compressor outlet for two main reasons: (1) a portion of the exhaust enthalpy is diverted to the cut-in turbocharger, which initially does not supply air, and (2) the turbine effective area increases after the cut-in. Such changes in air supply require the injected fuel mass to be adjusted accordingly, as reflected by the drop in fuel mass immediately after the TC cut-in, shown in Figure 9. Furthermore, the sudden increase in turbine effective area could significantly reduce the engine back pressure and therefore improve the engine volumetric efficiency and lower the fraction of trapped exhaust gases (internal exhaust gas recirculation), as illustrated in Figure 12, where the trapped exhaust gas fraction is displayed. Consequently, more fresh air can be entrained in the cylinder, resulting in a slight increase in the relative air-fuel ratio immediately following the turbocharger cut-in (Figure 8).
- Phase 3 – after TC cut-in to the target load/speed: Following the cut-in process, in the sequential TC setup, the system resumes the acceleration process with the additional turbocharger being activated. The turbine effective area remains relatively small compared to the wastegate approach as shown in Figure 7, maintaining higher intake receiver pressure (Figure 10). This allows for a higher amount of fuel injection, resulting in a faster acceleration during this phase as well.

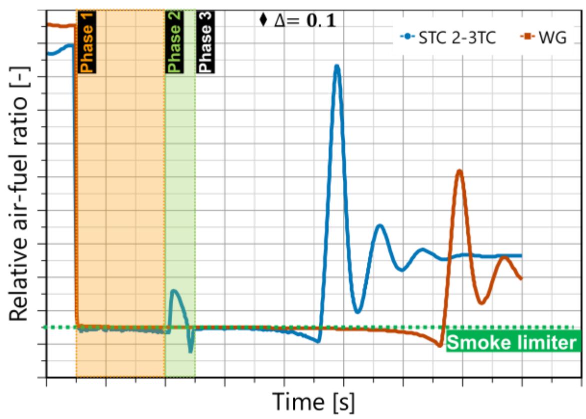


Figure 8. Relative air-fuel ratio during the acceleration process for sequential turbocharging and wastegate approaches.

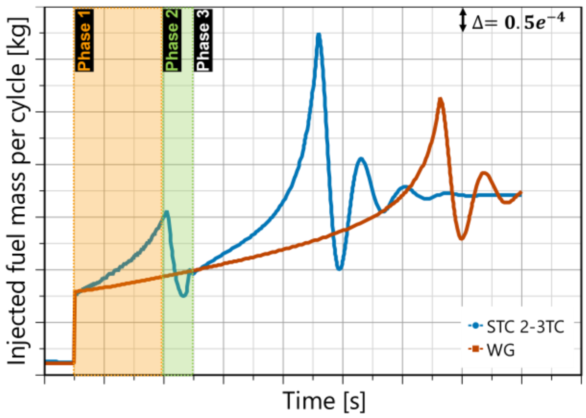


Figure 9. Injected fuel mass per cycle during the acceleration process for sequential turbocharging and wastegate approaches.

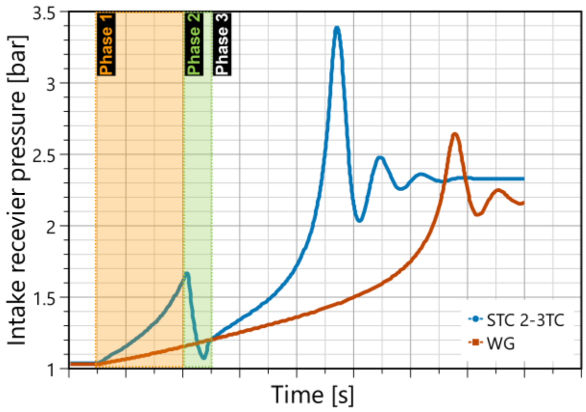


Figure 10. Intake receiver pressure during the acceleration process for sequential turbocharging and wastegate approaches.

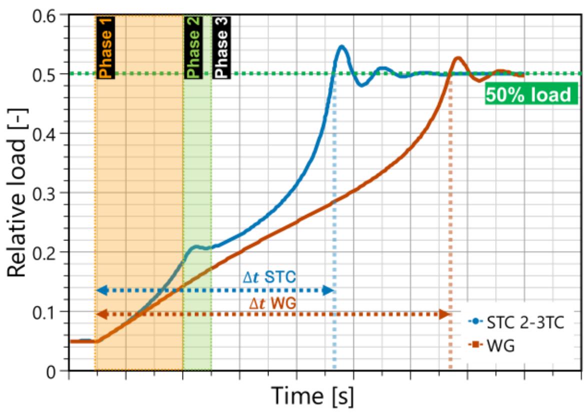


Figure 11. Relative engine load during the acceleration process for sequential turbocharging and wastegate approaches.

Figure 11 shows that with the current sequential TC setup, where the turbocharger cut-in occurs at 20% load (the optimal switch point), the acceleration time from 5% to 50% load is approximately two-thirds of that required by the wastegate approach, highlighting the advantages of sequential turbocharging in improving engine acceleration.

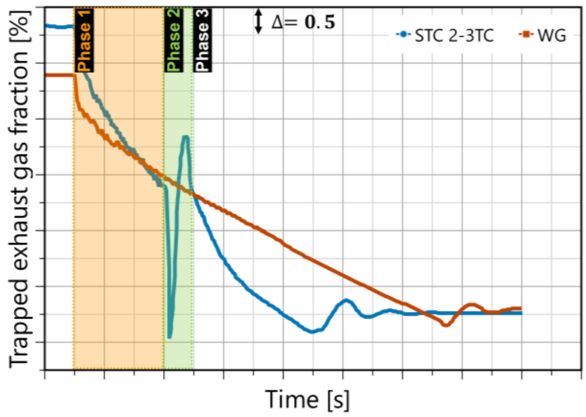


Figure 12. Trapped exhaust gas fraction / internal EGR during the acceleration process for sequential turbocharging and wastegate approaches.

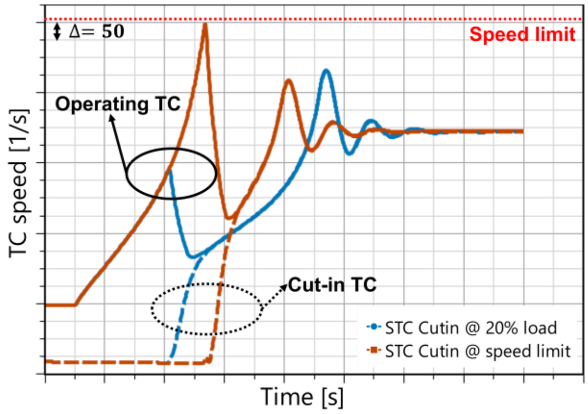


Figure 13. TC speed during the acceleration process for sequential turbocharging with TC cut-in at 20% load and speed limit.

Furthermore, the acceleration time can be further reduced by postponing the TC cut-in event. The latest feasible cut-in point is determined by the speed limit of the operating turbocharger, as shown in Figure 13. The operating TC continues to accelerate until it reaches its speed limit, beyond which a further delay of the cut-in event would risk the turbocharger to overspeed, which must be avoided for safety reasons. A speed margin must be considered to insure the fast-accelerating TC stays under the speed limit during acceleration. A detailed comparison of these two cut-in moments, when the engine reaches 20% load and the operating TC reaches its speed limit, is provided in Figure 14. It can be seen that a delayed TC cut-in results in a higher intake pressure, allowing a higher amount of fuel injection and a faster acceleration process. As shown in Figure 15, postponing the TC cut-in from 20% load to the speed limit can reduce the acceleration time by approximately 15%, leading to a total reduction of about 43% compared to the wastegate approach.

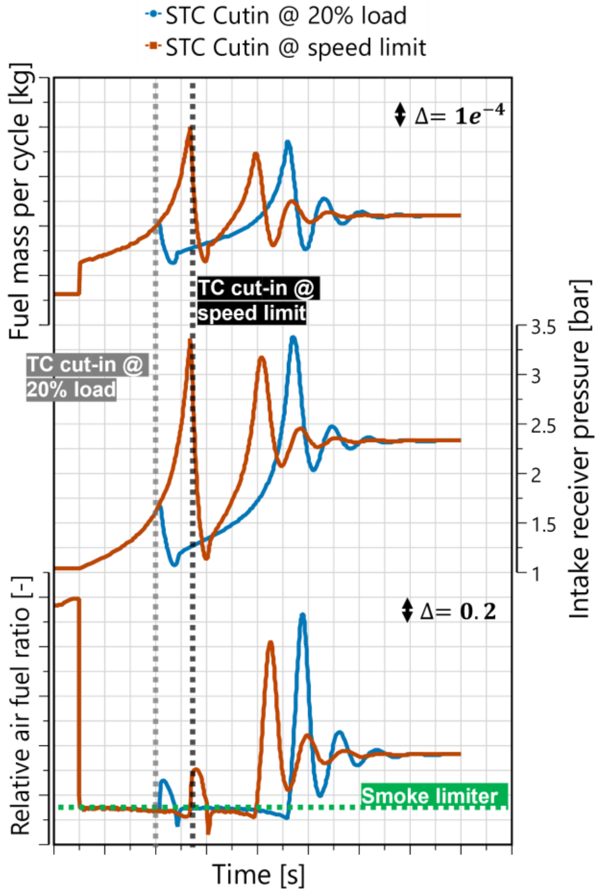


Figure 14. Injected fuel mass per cycle, intake receiver pressure, and the relative air-fuel ratio for the sequential turbocharging with TC cut-in at 20% load and speed limit.

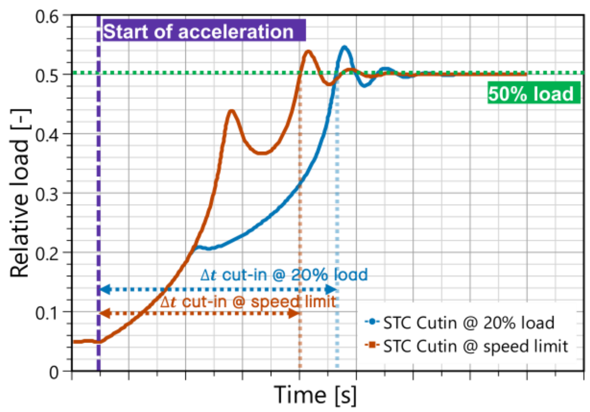
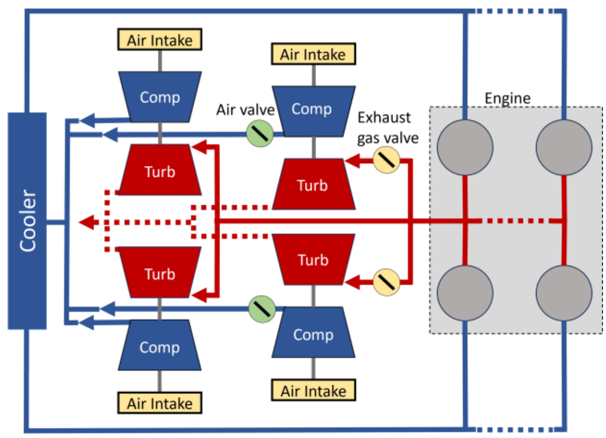


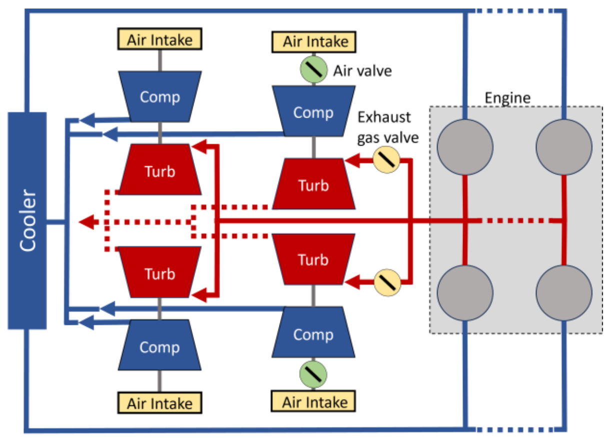
Figure 15. Relative load during the acceleration process for the sequential turbocharging with TC cut-in at 20% load and speed limit.

2.2 Control strategies for TC switch over

Two valves are needed to control the TC switch-over process: one located on the turbine side, referred to as the exhaust gas valve, and another one on the compressor side, referred to as the air valve. In this work, the exhaust gas valve is placed at the turbine inlet, while two potential air valve placements—either compressor inlet or outlet are compared. Note that placing the exhaust gas valve at the turbine outlet is also an option, but it presents certain risks: during cut-out operation, the turbine wheel back wall is exposed to high temperatures and pressures, leading to high-temperature blow-by flow. Such hot gas can induce oil coking on a still shaft, particularly when the engine operates at maximum torque over a long time period. For that reason, this option is not further studied in this paper.



a) Valves at compressor outlet and turbine inlet



b) Valves at compressor inlet and turbine inlet

Figure 16. Schematic of two configurations.

The schematic representations of these control configurations are illustrated in Figure 16. The impact of these two configurations on engine performance will be evaluated in this section. Additionally, different time intervals between the actuation of the exhaust gas valve and the air valve will be analyzed to gain a better understanding of the cut-in and cut-out processes and to find the optimal control strategy for the TC switch-over. The cut-in and cut-out simulation was conducted at 55% engine load along the propeller curve, representing the optimal 3-4 TC switch-over point. For simplicity, it is assumed that the valve areas change linearly during opening and closing, with equal durations for both air and exhaust gas valve transitions. The same air valve specifications are used in both configurations, where the air valve is positioned at the compressor inlet or outlet. A constant injected fuel mass was maintained across all engine cycles throughout the switch process.

2.2.1 TC cut-out process

Figure 17 analyzes the impact of different air valve placements on intake receiver pressure, and relative air-fuel ratio during the TC cut-out process. The air and exhaust gas valves are assumed to close simultaneously, with the closing duration indicated in the figure. The results show that, under such valve timing control, the location of the air valve has no significant influence on engine performance during the TC cut-out process. The slightly lower intake receiver pressure and relative air-fuel ratio observed when the air valve is placed at the compressor inlet can be attributed to increased pressure loss across the valve in this configuration due to the same valve area being applied in both cases.

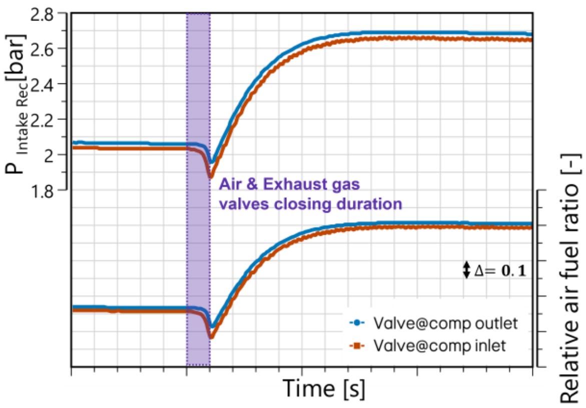


Figure 17. Intake receiver pressure and relative air-fuel ratio during the TC cut-out process.

Figure 18 presents the temperature and pressure changes within the section between the air valve and the cut-out compressor for both configurations, which represents also the thermodynamic state inside the cut-out compressor after the air valve closes. To be more specific, in the configuration where the air valve is positioned at the compressor outlet, the compressor inlet is exposed to ambient air, while the outlet is sealed by the air valve. As a result, the pressure and temperature between the compressor outlet and the air valve gradually decrease to match ambient conditions, eventually filling the compressor with ambient air, as depicted in Figure 18. Conversely, in the configuration of the air valve at the compressor inlet, the cut-out compressor remains connected between the air valve and the operating compressor, which supplies high-temperature and high-pressure air. It can be seen in Figure 18 that after closing the control valves, the volume between the compressor inlet and the air valve gradually fills with hot, pressurized gas from the active compressors. Such a difference in the cut-out compressor's final thermodynamic state influences the cut-in process behavior when the turbocharger is reactivated, which will be discussed in the next section.

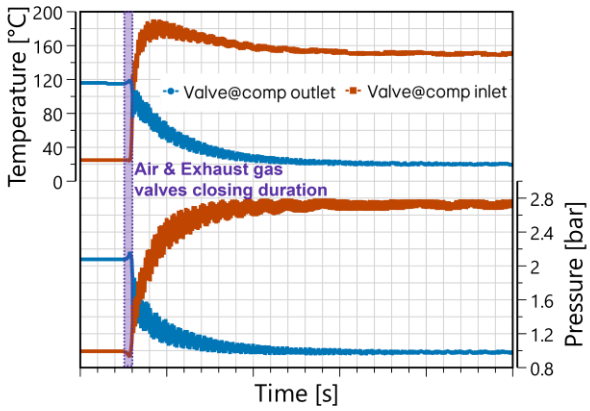


Figure 18. Temperature and pressure between the air valve and the cut-out TC.

Figure 19 and Figure 20 depict the operating and cut-out turbocharger speeds and the engine relative air-fuel ratio during the cut-out process for various time intervals between turbine and compressor deactivation. The closing duration of the exhaust gas valve is also indicated. The time interval, denoted as Δt, represents the actuation time difference between the exhaust gas and air valves where a negative Δt (Δt < 0) indicates that the air valve closes before the exhaust gas valve. Similar trends can be observed across both configurations:

- Δt < 0: when the air valve closes before the exhaust gas valve, exhaust gas continues to flow to the turbine without a corresponding air supply. This results in a lower relative air-fuel ratio during the cut-out process, potentially degrading engine performance and increasing soot emissions. Besides, since the air valve is already closed, the compressor side offers significantly less resistance. Meanwhile, the exhaust energy supplied to the turbine remains nearly unchanged, causing an increase in turbocharger speed. This speed increase is not critical in this operating condition, as the turbocharger speed is relatively low. However, it could become critical at a higher load where the TC could accelerate very rapidly and quickly reach overspeed.

- Δt >> 0: when the air valve closes significantly later than the exhaust gas valve, the turbocharger does not have sufficient exhaust enthalpy to maintain its rotation and pressure ratio. The high-pressure air from the active turbochargers can then flow backward through the cut-out compressor, potentially causing reverse flow or even reverse rotation, which can damage the bearing system of the turbocharger. As shown in Figure 19, a sharp drop in turbocharger speed can be observed when Δt is very high.

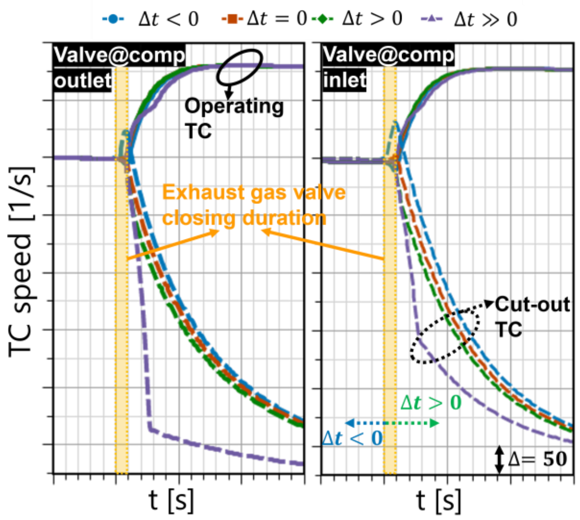


Figure 19. Operating and cut-out turbocharger speeds during the cut-out process for different time intervals between turbine and compressor deactivation.

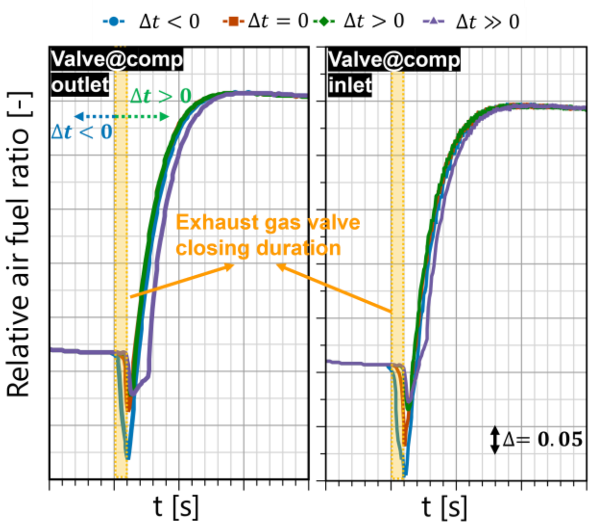


Figure 20. Engine relative air-fuel ratio during the cut-out process for different time intervals between turbine and compressor deactivation.

This suggests that closing the air valve earlier than the exhaust valve (Δt < 0) should be avoided to prevent an extremely low relative air-fuel ratio and potentially critical overspeed due to the loss of compressor load. Besides, selecting an appropriate absolute value of Δt considering the actual valve movements when Δt > 0 is crucial to avoid significant reverse flow or even reverse rotation.

2.2.2 TC cut-in process

Figure 21 compares the intake receiver pressure and engine relative air-fuel ratio for two different air valve placement configurations during the TC cut-in process. The exhaust gas and air valve opening durations are also illustrated, showing that the air valve opens slightly after the exhaust gas valve to prevent reverse rotation (such aspects will be further discussed in Figure 25 and Figure 26). It can be observed that, with the current time control interval, the minimum relative air-fuel ratio during the cut-in process remains nearly identical for both configurations.

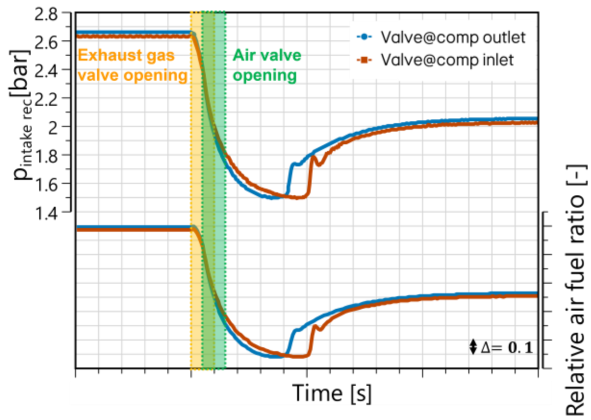


Figure 21. Intake receiver pressure and relative air-fuel ratio for different air valve placements during the TC cut-in process.

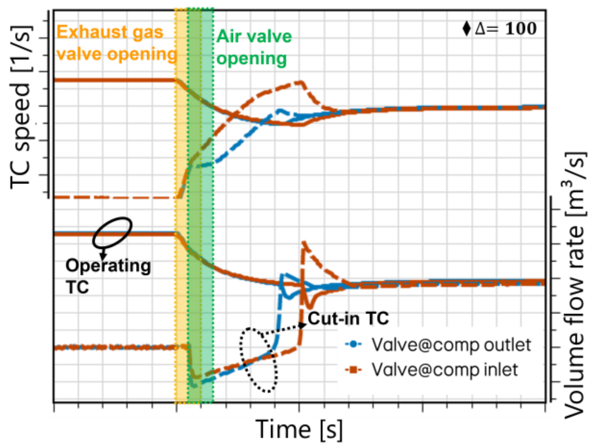


Figure 22. TC speed and compressor volume flow rate of the operating and cut-in TC for different air valve placements.

Figure 22 presents the TC speed and compressor volume flow rate for both the operating TC and the cut-in TC during the transition process. The results indicate that placing the air valve at the compressor inlet significantly accelerates the TC, resulting in a faster speed increase and a larger overshoot in cut-in TC speed. This rapid acceleration generates high-pressure air, which reduces the volume flow rate in the other operating TCs and increases the risk of compressor surge. This risk is highlighted in Figure 23, which shows the operating line of the active TC during the cut-in process for both configurations.

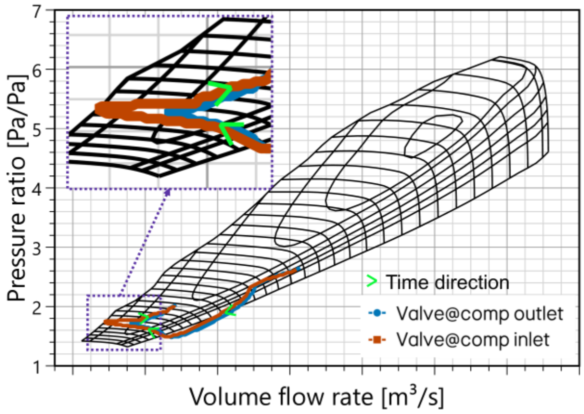


Figure 23. Operation lines in operating TC during TC cut-in process for different air valve placements.

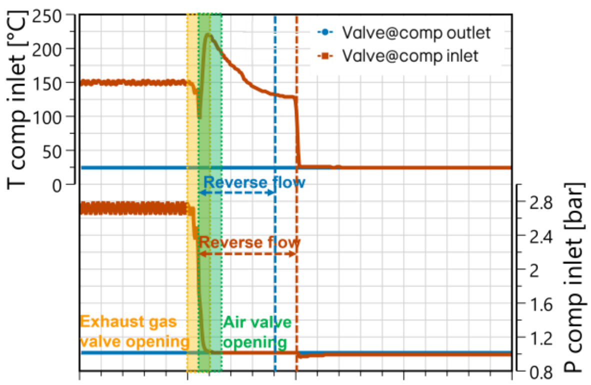


Figure 24. Temperature and pressure at compressor inlet for different air valve placements.

The faster acceleration of the cut-in TC, when the air valve is placed at the compressor inlet, can be attributed to the initial thermodynamic state inside the compressor before activation. As shown in Figure 24, which displays the compressor inlet temperature and pressure during the cut-in process, the cut-in compressor is filled with high-pressure, high-temperature gas before activation when the air valve is positioned at the compressor inlet. In contrast, when the air valve is placed at the compressor outlet, the compressor is filled with ambient air before activation. Once the air valve opens, the following behaviors are observed:

- Air valve at the compressor inlet: The compressor inlet pressure initially drops to ambient levels, while the temperature rises due to reverse flow. This results in a very low gas density at the compressor inlet, reducing the torque required to compress the air. Consequently, with the same power supply from the turbine side, the TC spins much faster.

- Air valve at the compressor outlet: The compressor inlet remains connected to ambient air, with both temperature and pressure assumed to be at ambient conditions. In this case, the gas density at the compressor inlet is much higher than when the air valve is placed at the inlet, requiring more torque to compress the air and resulting in slower TC acceleration.

Figure 25 and Figure 26 illustrate the operating and cut-in TC speeds and the engine relative air-fuel ratio during the cut-in process for various time intervals between turbine and compressor activation. The exhaust gas valve opening duration is also indicated. The time interval, denoted as Δt, represents the actuation time difference between the exhaust gas and air valves. It is important to emphasize that activating the air valve before the exhaust gas valve, or having a very small interval between their actuations, is not recommended. This is due to the fact that high-pressure air from the operating TCs can cause reverse flow in the cut-in compressor, which may result in reverse rotation if there is not sufficient power from the turbine side.

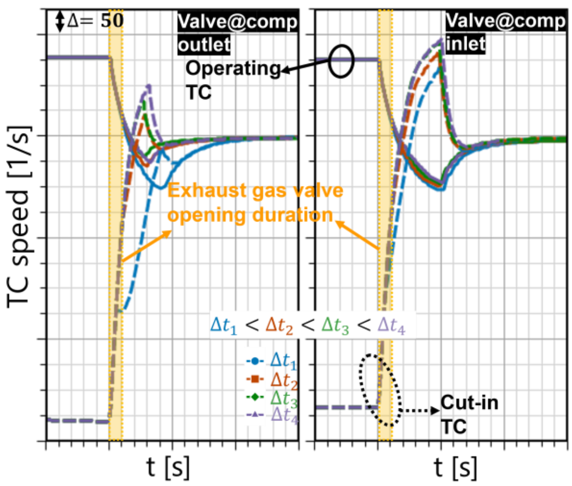


Figure 25. Operating and cut-in turbocharger speeds during the cut-in process for different time intervals between turbine and compressor activation.

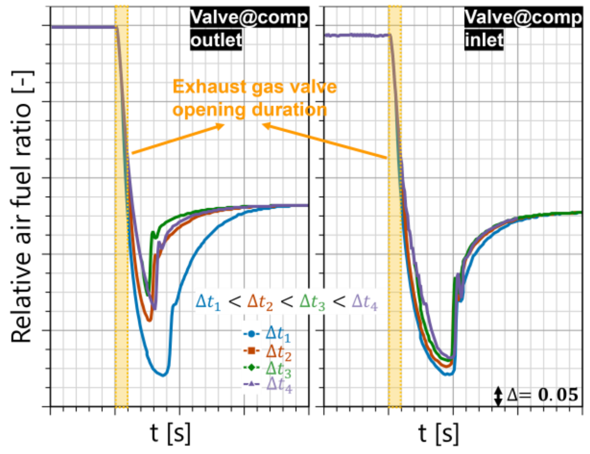


Figure 26. Engine relative air-fuel ratio during the cut-in process for different time intervals between turbine and compressor activation.

When the air valve is placed at the compressor outlet, two key effects of increasing the time delay between valve actuation can be observed:

- A longer time delay can facilitate a faster pressure build-up in the cut-in TC due to less resistance from the compressor, thereby increasing the minimum relative air-fuel ratio during the cut-in process. As shown in Figure 26, a moderate increase in the time interval can improve the air supply to the engine and enhance its performance.
- However, an excessively long time delay is also not recommended. As shown in Figure 25, a prolonged interval leads to a much faster acceleration of the cut-in TC, which can result in a higher pressure ratio, reducing the volume flow rate of the other operating TCs and therefore lowering the engine relative air-fuel ratio as well as increasing the risk of surge in the active units. Moreover, if an excessive time delay is used, it is recommended to monitor the cut-in TC speed when a high enthalpy level is available at the turbine inlet.

When the air valve is placed at the compressor inlet, the turbocharger speed and engine relative air-fuel ratio are less affected by the time interval between valve actuations. In this configuration, the cut-in TC tends to accelerate more rapidly and consistently overshoots the speed of the operating TC. However, this rapid acceleration introduces a risk of surging in the operating TC and may also cause overspeed of the cut-in TC, particularly if the cut-in takes place at higher loads where the stable TC speed is elevated. It is worth noting that placing the air valve at the compressor outlet generally delivers better engine acceleration compared to placing it at the compressor inlet due to the lower speed overshoot of cut-in TC unit, provided that the time interval between the two-valve actuations is optimized.

3 SPECIFIC CHALLENGES

As shown in the previous section, the sequential system comes with changing boundary conditions on the turbocharger unit depending on the valve layout, TC activation state, and switchover process. These constraints are to be taken into account while designing a turbocharging solution able to operate reliably during steady state and cut-in/out transition. A sequential turbocharging system offers superior performance compared to standard turbocharging configurations; however, this advantage comes with increased system complexity, necessitating greater development and validation efforts. To fully capitalize on the benefits of this complexity, a tailored turbocharging solution must be selected. The following sections address these requirements and discuss about the criteria to be considered when designing the valves to be placed on the engine.

3.1 Application challenges translated to turbocharging requirements

As emphasized in the performance study, sequential systems require modern high pressure TC systems to enable BMEP levels beyond 30 bar in combination with high torque over a wide range of engine speeds. Thanks to the cut-in/out of TC units, the volume flow spread is narrowed down and high pressure ratio compressor stage with high solidity vaned diffusor design can be used. Such compressors yield highest single stage pressure ratio with high efficiency. A compromise between high compressor pressure ratio capabilities and map width must be reached, ensuring that the sequential operation line fits within the compressor map boundaries.

To enable vessel acceleration, the turbocharger needs to accelerate as fast as possible along the engine operation line and during the cut-in process to provide enough air necessary for the combustion. Most of these requirements must be addressed in the turbine stage design. A part load optimized turbine efficiency coupled with a turbine flow characteristic with low effective areas at low expansion ratios, reduced inertia, and increased flow capacity permitting the highest turbine effective areas for the smallest turbine diameters are the key parameters to enable fast TC shaft acceleration. As shown in the performance study, the sequential turbocharging system allows reduction of the total turbine effective area of the turbocharging system thus enabling a faster acceleration of the activated TCs.

Sequential turbocharging system come with some challenges on the turbocharger as it involves a substantial number of load changes and TC cut-in/out thus cyclically heating up and cooling down the turbocharger materials. The higher the frequency and the larger the temperature amplitudes of the cycles, the greater the thermomechanical fatigue (TMF), eventually resulting in cracks. With more cycles comes more fatigue cycles, initiators of cracks in the hot parts. Typically, turbine casing and nozzle ring are the most exposed parts. A robust TC design with the hot parts optimized for TMF is required.

To achieve high compressor pressure ratio (PIC) for ensuring the required power density, the TC operating speed must be at the highest level. This necessitates optimizing the rotating parts, compressor, and turbine wheels, for high-cycle fatigue (HCF), low-cycle fatigue (LCF), creep, and maximum material temperature.

The fast acceleration requirements, number of starts and stops, and wide operation line characteristics of the cut-in/out process are to be considered when designing a robust bearing system. When the rotor is stopped, the shaft has a metal-to-metal contact with the bearings as it is “sunk” to the radial bearing bottom. During engine start, a slight mixed friction causes some slight wear to the bearings, consequently a higher start number exposes the radial bearings to more wear cycles. A special attention must be given to the axial thrust bearing when the engine breathing line is operating far from the standard propeller line which is typically the case for sequential turbocharging. The thrust load depends on the operation point in the compressor map and in some cases, it can even reverse from the main to the auxiliary thrust bearing. Consequently, a strong axial bearing system is required to operate reliably in such systems.

Often used in mission-critical applications where reliability and uptime are paramount, durability and serviceability are critical for turbochargers used in sequential systems. They must withstand demanding operating conditions while offering predictable maintenance intervals and minimal downtime for maximal cost efficiency over the life cycle.

Finally, a robust shaft sealing is required for sequential operation to minimize potential oil leakages from the bearing casing into the compressor and turbine casings. Especially during the transition to cut-out operation and turbocharger “idling”, where the pressure in the bearing casings can be higher due to elevated engine crankcase pressure than the one at the compressor and turbine back wall.

All these requirements were addressed while designing the new A101-R turbocharging platform, more details on its dedicated features will be given in the Section 3.3.

3.2 Valve layout and design

3.2.1 Effect of valve location on TC

The effect of valve location on the switchover process in terms of thermodynamic performance was studied in the section 2.2. This section will shift focus to its impact on TC components emphasizing required design features and ideal system layout.

Starting with the valve located at the compressor inlet, it can in some cases enable a simplified design as the valve would be located between the air filter and the compressor inlet with a limited number of interfering parts in the surrounding. However, several challenges need to be considered:

- As the valve is located at the compressor inlet, the compressor wheel of the deactivated TC is exposed to hot charge air from the other TCs that are in operation as shown in Figure 18. This hot air can reach a temperature at which aluminium material starts to weaken. In case the turbocharger is activated and rapidly accelerated to high speed, a burst of the overheated compressor wheel cannot be excluded. In that arrangement, a more robust and more costly compressor wheel material such as titanium could be chosen. Alternatively, a design incorporating a dedicated charge air cooler for each TC would enable lowering the backflow temperature. In this configuration, the compressor outlet of each turbocharger would be directly connected downstream of its respective charge air cooler, ensuring that the charge air flowing back to the cut-out compressor wheel is effectively cooled. As for this particular case, special attention must be given to the risk of condensation as it is critical to compressor wheel erosion.

- When located too close to the compressor inlet, the valve can generate non-uniform flow before the wheel, precursor of blade vibration leading to potential HCF issues.

- A deactivated TC will have a high pressure at the compressor wheel backwall. On one hand, the positive pressure ratio across the sealing helps to maintain oil tightness. On the other hand, blow-by from the compressor casing to the bearing casing will increase, potentially elevating the bearing casing pressure. In the case where the valve on turbine side is located at the inlet, the turbine casing will have a pressure around ambient and potentially below the bearing casing pressure thus increasing the risk of TC leakage on turbine side. An effective sealing system between the compressor and bearing casings is required.

A valve located at the compressor outlet comes with the advantage of a lower material temperature in cut-out operation with no risk of flow disturbances at the compressor inlet permitting the usage of aluminium as cost-effective wheel material. Oil leakage from the bearing casing to the compressor casing is a challenge as the pressure at the compressor wheel backwall is around ambient and the bearing casing pressure can be high enough to create a positive pressure gradient favoring leakages. A robust and effective sealing design must be ensured between the compressor and bearing casings. Sealing air system is a concept used in axial turbines to improve the sealing on turbine side and is provided from the compressor's back wall. During cut-out operation, the pressure in the compressor casing will reduce to ambient and the sealing air should be provided outside of the TC, typically from the intake receiver. In that case, a bearing casing adaption is required.

Two types of valves, passive and active, can be used on the air side. Passive valves are usually selected for their simplicity enabling cost-effectiveness and reliability. They rely on natural flow dynamics and pressure difference in the system to operate, however, they come with certain challenges: a proper design must be ensured to correctly respond to pressure changes. As they are dependent on the fluid dynamic, they do not provide the fine control required for high-performance engines. As highlighted in the section 2.2, a precise switching timing must be set to ensure optimum performance with the highest minimum air-to-fuel ratio (λCMIN) while avoiding reverse rotation, overspeed, and surge. On the other hand, active valves allow fine controlling these parameters, guaranteeing optimal effectiveness during switch over process.

On the hot side, the valve can be considered as the TC actuator as it activates the turbine thus initiating the TC rotation powering the compressor.

One of the main challenges of a valve located at the turbine inlet is the oil leakage from the bearing to the turbine casing. When the TC is cut-out, the pressure at the turbine wheel back wall reduces around ambient pressure, and the bearing casing pressure can be high enough to cause leakages. As for the compressor side, a robust oil tightness system must be considered in the TC design, together with a good management of the engine crankcase pressure indirectly driving the TC bearing casing pressure.

3.2.2 General valve design criteria

When designing a sequential turbocharging system, Accelleron recommends using active valves at the compressor outlet and turbine inlet to maximize TC reliability and engine performance.

With this setup, particular attention must be given to the oil supply design and blow-by gases system to avoid oil leakage on the lubricated stationary turbochargers. The engine crankcase pressure driving the TC bearing casing pressure, as well as the TC oil supply must be properly managed to reduce negative pressure ratio across shaft seals and avoid the TC being oversupplied with oil, potentially flooding the TC bearing housing. Furthermore, under pressure in the air and gas pipes should also be avoided to prevent oil leakage. A common air or gas piping between running and cut-out TCs can generate a suction effect from the running TCs flow. This can reduce the static pressure at the compressor and turbine wheel backwalls, to a level where oil leakage from the bearing casing may occur, even if the system is designed with an effective oil supply and blow-by gas management system. For environmental considerations, it may be necessary to recirculate ventilation gases from the engine's crankcase and/or a vent tank back through the engine. To achieve this, the ventilation gases can be directed upstream of the compressor inlet. In a sequential system, the feeding of ventilation gases must be provided upstream of a non-switchable turbocharger that is in permanent operation, to allow a permanent suction of the vent gas by the rotating compressor.

Aside from the requirements for ensuring reliable operation of the turbocharger unit, the valves mounting should account for installation constraints and cost optimization but also for ensuring their optimal performance and reliability during the engine lifetime.

The valves must be positioned in an area with proper heat dissipation or thermal shielding to withstand high exhaust temperatures, which often exceed 650°C in high-speed marine Diesel engines. Optimal thermal management must be ensured to guarantee control and avoid degradation or failure leading to engine performance degradation.

The design of the valves must be optimized to reduce pressure losses leading to decreased turbocharging performance and compromising the engine power output critical for these marine applications.

The space-constrained environment of these engine installations requires a compact and accessible design to minimize the footprint of the turbocharging system while allowing maintenance access.

Composed of a moving part on a shaft, the valve is sensitive to vibrations that could lead to mechanical wear or misalignment over time, reducing its effectiveness. Vibrations and pulsations being common in high-speed Diesel engines, the valve must be mounted in a position, where the excitation does not harm the mechanical integrity of the flap and especially the actuator.

3.3 Safety

In addition to the specific design requirements to ensure a reliable sequential turbocharging system, it is crucial that the engine builder addresses certain safety aspects: Proper valve control, combined with continuous turbocharger speed monitoring, is essential to prevent TC overspeed conditions that could result in rotor burst. This risk is particularly high when a switchable TC is cut-in late during power ramp up or cut-out early during ramp down. Especially when the air valve is positioned at the compressor inlet a rotor speed overshoot is more likely to occur if the cut-in is not properly managed.

Furthermore, a valve located at the compressor inlet can lead to compressor wheel overheating of a cut-out TC if the active TCs are operated at high pressure ratio. The resulting high material temperatures could compromise the mechanical integrity and increase the likelihood of failure upon activation of the cut-out TC.

Another critical aspect is oil tightness, particularly of the cut-out units, where leakage would represent a fire hazard. To mitigate this risk, engine builders must ensure an effective crankcase ventilation and optimize the piping system to prevent the running TCs from drawing a suction effect on the cut-out units.

4 A101-R: VERSATILE TC PLATFORM ENABLING SEQUENTIAL SYSTEMS

4.1 Product Concept: Application Packages and Core Unit

The A101-R turbocharger was first introduced at CIMAC 2023 as a versatile platform designed for large high-speed Diesel engines with the particularity of being utilized across a broad spectrum of applications, each with distinct needs and end-user value. This section focuses on the specific benefits and features of the new TC platform when used in sequential turbocharging systems.

The A101-R family’s requirements are extensive due to the broad range of applications covered by large HS Diesel engines and can be summarized as follows: a robust, reliable, and cost-effective turbocharger platform featuring two frame sizes, designed to accommodate 12 to 20V-cylinder engine configurations with 2 and 4 TC arrangements. It must support two-stage configurations and include optimized components to meet the diverse performance demands across various applications (high pressure ratio for single or low-pressure compressor stages, moderate pressure ratio with large volume flow variability for high-pressure compressor stage, and part-load optimized turbine stage with high LCF robustness). Platform durability must be aligned with engine-specific maintenance schedules. Additionally, it must feature an optimized design capable of handling deep thermal cycles and include a robust shaft sealing system for enhanced reliability. Furthermore, its design must be compact and modular, facilitating service-friendly packaging.

The modular approach used by the engine builders to design large HS Diesel engines must be taken into consideration when designing a turbocharging solution to cope with the requirements of a multi-purpose platform, allowing both:

- A standardized design enabling compactness and standardized interfaces for optimized on-engine packaging, with mechanical capabilities regarding robustness and durability for cyclical operation whilst avoiding oversizing for less demanding applications.

- Versatile characteristics to fit the applications in terms of performance parameters (pressure ratio, volume flow, optimum efficiency along the engine operating line) and enabling advanced turbocharging systems such as sequential systems as well as 2-stage arrangements.

A101-R platform aims to feature the right level of standardization balancing a common design basis and dedicated design features. As for the multi-purpose engine design, the versatile TC platform must match all application criteria translated to the product requirements listed above, while avoiding a product having fundamentally different variants in order to benefit from design and feature commonalities enabling lower development and validation efforts, commonalities on supply chain as well as manufacturing processes.

When sorted in terms of size, turbocharger requirements, and customer value drivers, the HS Diesel engine market is composed of three main subsegments: "EPG" (Electrical Power Generation), "Marine" & "Oil & Gas" and "Off-road". A dedicated application package was developed to answer the specific challenges of each segmentation. The foundation is a standardized Core Unit incorporating the features matching the common requirements. More details about the Core Unit and the Application Packages are outlined in Paper No. 47, 2023 CIMAC Congress [7].

Sequential turbocharging, being part of the "Marine" subsegment, comes with specific challenges listed earlier in the paper on top of the standard marine requirements such as: durability and serviceability to achieve cost efficiency throughout the life cycle; up- and downstream options, advanced shaft sealing variant, and water-cooled bearing housing, enabling enhanced reliability and long overhaul intervals under demanding operating conditions; and special turbocharging arrangements, namely sequential systems.

4.2 Component characteristics and dedicated features matching Sequential Turbocharging requirements

A101-R family incorporates state-of-the-art and tailored components to ensure efficacious and reliable operation for marine applications, including sequential systems.

The newly designed turbine stage TV49, was optimized to enhance LCF capability by reducing the local stress maxima, enabling higher rotational speeds compared to the existing A100 turbines, for heavy-duty cyclical applications. The maximum operating speed for high power density off-highway applications was increased by 11%, allowing for a substantial 30% increase in achievable compressor pressure ratio. This supports sequential operation with cut-in/out events, enabling greater power density – a key criterion for high-performance vessels utilizing a parallel sequential turbocharging setup.

Highlighted in the performance study, the transient capability is a critical factor for fast acceleration while minimizing emissions (high $\lambda CMIN$ during acceleration and cut-in process). Once more compared to the existing A100 turbines, TV49 stage features a 12% increase in flow capacity, driven by a 27% improvement in specific flow capacity ($SEFFT/DT^2$). Simultaneously, the mean turbine diameter was reduced by 5.9%, resulting in lower mass and moment of inertia, which enhances the transient response at a comparable absolute flow capacity. Aside from the enhanced specific flow capacity, the turbine was part load optimized with improved efficiency and a steeper mass flow parameter curve at mid turbine pressure ratio, also favoring faster acceleration by supplying the compressor with more power at part load. Endowed with an extensive portfolio of matching material, including four different trims and multiple nozzle rings, the turbine's performance characteristic can be tailor-made to meet the specific needs of the engine, optimizing performance from part load to full load, while keeping the efficiency of the various possible matching at a high level.

The A101-R range is equipped with five different compressor versions, four of which are dedicated to high pressure ratio capabilities with various flow capacities. They match the wide range of possible flows with the new turbine stage TV49, covering engine power ratings from about 900 kW to around 2000 kW per turbocharger. These compressor stages are based on high solidity vaned diffusor design, aluminum impellers, and backwall cooling enabling pressure ratios above 5.8 together with high efficiency fitting the engine requirements for high power marine applications with high BMEP beyond 30 bar. This particular design usually results in a narrow map compared to stages without vaned diffusors. However, the compressor maps of the A101-R are wide enough to support sequential turbocharging applications. A wide selection of available trims and diffusor configurations, tailored for each compressor version, allows for precise placement of the operating line position within the compressor map boundaries. This reduces compromise imposed at switching points caused by limitations such as the surge line, speed limit, or lower-efficiency zones.

Field experience showed some challenges with radial TPS or A100 turbochargers on high-cycle applications such as railways, off-highway trucks, or marine applications such as tugboats. The hot turbine casing and cold bearing housing, being in contact, experience some relative movements during deep thermal cycles, such that the surface contact of the joint area between the two casings wears. The turbine casing expands and shrinks following the gas inlet temperature whereas the bearing housing, being cooled, has a lower and more stable material temperature, thus having less relative dimensional change during the thermal cycles.

Turbochargers installed in marine vessels can experience many thermal cycles depending on the operating conditions, such as varying speeds, loads, and environmental factors. Sequential turbocharging systems control turbocharger activation to optimize engine efficiency and performance. Frequent cut-in/out events can result in repeated heating and cooling cycles, contributing to thermal stress on turbocharger components.

In order to maintain the casing positioning and gas tightness with significant relative movement between the casings during deep thermal cycles throughout the full turbocharger lifetime, the A101-R was designed with an optimized turbine-to-bearing casing joint connection together with a reinforced clamping system. These functions were extensively validated on test rig as well as on field engines operated with many thermal cycles.

Sequential systems lead to a specific operation line, far from the typical breathing line of conventional propeller curve or constant speed operation line. This can lead to axial forces increasing strongly and must be accounted for when designing the axial thrust bearing system. Developed to work in a 2-stage setup where the high-pressure TC can reach high compressor outlet pressure while operating far from the standard breathing line, the bearing system can also handle axial thrust forces that can be encountered with sequential turbocharging systems. For faster acceleration, an option with a ball-bearing cartridge can be selected and fits in place of the standard bearing cartridge with hydrodynamic radial and axial bearings.

Finally, the shaft sealing design was enhanced to improve oil and gas tightness. For sequential application, oil leakage into the compressor and turbine casings of the non-activated turbochargers can occur, especially during the cut-out switching process where a negative pressure difference between the compressor backwall and bearing casing can arise. The new TC generation features a further improved shaft sealing system compared to the previous products. The oil spray direction onto the hot casing walls was optimized to improve cooling while preventing the oil from reaching the sealing areas. Additionally, a second piston ring on the compressor side sealing can be fitted to reduce further blow-by gas and improve oil tightness to the minimum, a key feature on sequential systems.

In summary, the new A101-R range can meet the requirements to operate with sequential turbocharging systems as it embodies all the necessary features required for reliable operation on such challenging systems. Its advanced performance and breadth of matching material guarantee optimum operation with minimum compromise.

4.3 Product validation and Field experience

The product range was validated through a complete qualification program including simulations, testbed measurements, and field validation. This program covers a complete range of items such as performance, blade vibration, bearing and lubrication systems, shaft sealing, containment, thermo-mechanical fatigue, vibration, noise, and speed limit.

In addition to the internal qualification process, Accelleron performs field validation with its new products to further confirm the correct operation of its turbocharger units on end-user applications. Inspections were performed after 3'000 and 5'000 running hours on various installations. First inspections have been performed on a rail application, the most demanding for a turbocharger, showing good results.

The product has been launched successfully; commercial deliveries have started for numerous engine builders worldwide. These units are being utilized across a wide range of applications, including marine, rail, and electrical power generation, for various fuel types.

4.4 Serviceability

Accelleron turbochargers are designed to offer predictable maintenance intervals and minimal downtime, ensuring reliable operation. With a cost-efficient design, they reduce total ownership costs over their lifecycle while delivering robust performance. Their simplified structure enhances ease of maintenance, allowing for quicker and more efficient servicing. Additionally, the global availability of spare parts and service via the 100+ Accelleron service stations ensure operational continuity by enabling rapid repairs and minimizing disruptions. This combination of durability, serviceability, and cost-effectiveness makes these turbochargers an indispensable solution for marine applications.

5 CONCLUSIONS

In this paper, simulation results analyzing the advantages of sequential turbocharging and optimization of the valve switchover process were presented. Then, the turbocharger requirements and the valve layout to enable reliable sequential systems were discussed. Finally, the new A101-R turbocharger family was highlighted as a tailored turbocharging solution for marine high-speed diesel engines with sequential turbocharging.

The simulation results showed that the sequential system can significantly enhance low-end torque capabilities and engine acceleration, while maintaining high engine speed. In these categories, the sequential system outperforms a conventional waste-gate approach. A valve located at the compressor outlet and turbine inlet paired with a fine control of the valve actuation timing is recommended, in order to maximize turbocharger reliability and engine performance.

To maximize the potential of the overall system, the turbocharger must feature compressor with high pressure ratio capability, while allowing for a wide flow range that is typical of sequential operation, and be paired with a turbine characterized by high part load performance, to enable fast acceleration. Robust sealing and bearing systems should be implemented to cope with the varying boundary conditions during cut-in/out operations.

The new A101-R turbocharger family was developed with dedicated components and tailored features enabling utmost performance and reliability of a sequential turbocharging system.

Beyond marine applications, A101-R is a multi-purpose turbocharging platform dedicated for large high-speed diesel engines and covering a wide range of applications – from power generation to large off-highway vehicles. The A101-R's modular design facilitates mounting on standardized engine platforms. Additional features on the shaft and turbine side, designed to ensure reliable operation throughout the turbocharger lifetime, were also validated in a complete qualification process that included field testing. Commercial deliveries are ongoing, and the turbochargers are being used across the globe for a large range of applications. The global network of Accelleron turbocharger service reinforces the availability and the uptime of Accelleron turbochargers operating on demanding applications.

6 DEFINITIONS, ACRONYMS, ABBREVIATIONS

BMEP: Break Mean Effective Pressure

DT: Mean Turbine Diameter

HS: High-Speed

LCF: Low-Cycle Fatigue

PIC: Compressor pressure ratio total/total

R_{exh}: Specific gas constant for exhaust gas

SEFFT: Turbine effective area

TC: Turbocharger

TMF: Thermomechanical Fatigue

WG: Waste-Gate

λCMIN: Minimum air-to-fuel ratio

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