Turbocharging solutions for EGR on large diesel engines

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\textbf{Abstract:} Exhaust gas recirculation, EGR, is a well-known method for reducing NO\textsubscript{x} emissions on combustion engines. It can be considered as a state-of-the-art technology for automotive engines, but the introduction for large diesel engines it being investigated. Due to the requirements of large engines as well as the quality of the fuel used, the introduction of EGR is challenging. In this report the possible turbocharging solutions which can enable EGR are described and discussed. The EGR turbocharger and the EGR blower have been identified as the most promising turbocharging devices and realized to the technology demonstrator stage. They have been tested on engines within joint development projects with customers. Some development aspects of these devices are also presented.

\textbf{Key Words:} EGR; Blower; Turbocharger; Turbocharging; NO\textsubscript{x} reduction; Efficiency

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1 Introduction

The limitations on emissions of oxides of nitrogen (NO$_x$) for large engines have so far been very mild, with some local exceptions. This situation should change in the near future. For marine engines the IMO Tier III limitations, characterized by a reduction of about 80% against Tier I for ships operating in coastal ECAs (Emission Controlled Areas), was planned to be in force from 2016. For other engines EPA and Euro IIIb apply with similar limits. Current received wisdom is that the only technologies enabling diesel engines to reach the low emission level required by IMO Tier III are EGR and selective catalytic reduction (SCR). Probably SCR will be preferred for stationary plants, where the space and the logistics required can be afforded better. EGR technology, due to its possibility for on-engine integration, could be preferred for mobile applications, i.e. marine and traction.

A study has been initiated at ABB Turbo Systems with the objective of evaluating the possibilities for adapting the turbocharger system to enable EGR. Most of the known methods from the on-road applications are based on a pressure driven flow, requiring the production of a positive pressure difference from the exhaust to the inlet side of the cylinder. These methods were soon excluded, because they would preclude the possibility of scavenging the combustion volume, which is indispensable on large engines. Among the remaining methods two require a substantial contribution from turbocharging technology: the EGR turbocharger and the EGR blower.

In order to demonstrate the possibilities of these technologies and to be prepared for fulfilling possible market requirements, technology demonstrators have been produced and tested on engines. The thermodynamic results are fully in line with expectations based on simulation studies.

This study handles the components and layouts of the turbocharging system enabling EGR, which represent only a part of the problem. Cooling and cleaning the exhaust gas is a further issue, which must be solved for the successful introduction of the EGR concept. In some cases the results concerning fouling and corrosion have not been fully satisfactory. The path from technology demonstrators to fully industrialized products is still long and requires a lot of development work on the system as well as on its components.
2 Theoretical analysis of the EGR methods

In this report following definitions are used (Figure 1):

- **Internal EGR**: Exhaust gas is retained in the cylinder using appropriate valve timing.
- **HP-EGR** (high pressure or short route): Exhaust gas is recirculated from the exhaust to the inlet manifold.
- **LP-EGR** (low pressure or long route): Exhaust gas is expanded in the turbine and recirculated into the compressor.

![Figure 1. Definition of EGR basic configurations](image)

2.1 Historical background

The first record of ABB Turbo Systems’ interest in the implications of EGR on turbocharging technology is a patent application from 1975 [1], covering introduction of EGR flow into the diffuser of the turbocharger compressor. This idea was aimed at avoiding possible compressor contamination, typical for LP-EGR and to enlarge the available pressure difference for HP-EGR.

Another important pioneering activity was a research project running at the Swiss Federal Institute of Technology (ETHZ) from 1993 to 1998 with participation by ABB Turbo Systems, Wärtsilä New Sulzer Diesel and the Paul Scherrer Institute. In this project an impressive combination of innovative technologies for that time were tested on a 1.5 MW diesel engine (Figure 2) namely:

- High pressure turbocharging with Miller timing and variable turbine geometry (pressure ratio increased from 3.5 to 4.5)
- High pressure cooled EGR using an EGR turbocharger
- Common rail fuel injection
- Selective catalytic reduction

Some results were published in a CIMAC paper [2]. The first experience with the EGR turbocharger is especially interesting. A commercial automotive turbocharger was used. It had not been possible to find a suitable match for the special application, hence a throttle valve
before the turbine was used to reduce the expansion ratio and control the EGR rate.

At less than 20 hours, the durability of the aluminum compressor was very low. This had probably been caused by excessive cooling of the EGR flow, leading to a great deal of water condensation and by the sulfur content of the fuel. Nonetheless interesting findings were gained from that research project.

### 2.2 EGR working principle

Fuel needs a certain amount of oxygen for combustion, which is typically non-homogeneous in a conventional diesel engine. A simple but valid approach to describe it, is to consider that combustion always occurs at near stoichiometric conditions. The temperature reached in the combustion zone is determinant for the formation of NO\textsubscript{x}, excess air is mixed with combustion products only at a later stage.

Nitrogen present in the combustion air acts as inert gas and reduces considerably the flame temperature in comparison to combustion with pure oxygen. Further addition of inert gas, reducing the oxygen concentration in the charge, is a very effective method for further reducing the maximum combustion temperature, and consequently the formation of NO\textsubscript{x}. The simplest way to achieve this effect is to cool down a certain amount of exhaust gas, which is a mixture of inert gases (N\textsubscript{2}, CO\textsubscript{2}, H\textsubscript{2}O) and excess air, and mix it with the charge air. The main advantage of EGR is the ready availability of inert gas in the form of exhaust gas, whereas alternative methods like water addition or nitrogen enrichment require additional media or processes.

It is common practice to measure the amount of EGR by means of the EGR rate, i.e. the amount of exhaust gas referred to the total aspirated flow. Here it is important to note that only the combustion products give a
positive contribution to the desired effect of reducing the temperature in the combustion zone. Excess air can be seen here as a parasitic loss, because it requires energy for recirculation and does not contribute to the EGR target.

Large diesel engines usually run with rather high values for the trapped air-fuel equivalence ratio \( \lambda_v \), with the air-fuel equivalence ratio in the exhaust \( \lambda_{\text{V,tot}} \) additionally increased by scavenging. For this reason, large engines require considerably higher EGR rates to achieve the same level of \( \text{NO}_x \) reduction as small automotive engines. In order to compare different EGR systems, the oxygen concentration in the charge should be used instead of the EGR rate (Figure 3).

A difficult question is how much compressor pressure ratio \( \pi_C \) is needed by an engine with EGR compared to a conventional one. Three cases can be taken as references (Figure 4):

- **Substitution**: The chosen amount of gas replaces a corresponding mass of air, the boost pressure is unchanged, \( \lambda_v \) is reduced
- **Addition**: The amount of fresh air aspirated from the ambient is constant, the EGR flow is added to it. In this case the boost pressure must be considerably increased and \( \lambda_v \) is increased by the recirculated excess air
- **Constant \( \lambda_v \)**. In this intermediate condition the amount of trapped air (fresh plus recirculated) is kept constant. The required boost pressure lies between the two extreme cases, depending on the value of \( \lambda_v \).

There are two reasons why \( \lambda_v \) is kept high on larger engines. The first is the engine thermal load. The excess air has a thermal capacity which reduces the global cycle temperatures and consequently the heat input to the hot engine components. This function can also be produced by the cooled exhaust gas, i.e. the Substitution case above would leave the situation unchanged.

Of course the air-to-fuel ratio is also important for combustion.
Short efficient combustion with minimum particulate emissions is the target. It is well known that EGR tends to give slower combustion with more particulates. This effect can be reduced by means of injection and combustion optimization, but in some cases it could be necessary to increase the level of boost pressure up to the values for constant $\lambda_v$.

2.3 EGR technologies

In the preceding section the influence of EGR on the engine process has been described. From this, it becomes clear that some amount of exhaust gas needs to be cooled and recirculated. The next issue is to modify the turbocharging system to enable the required amount of EGR. Basic research work has given some indications in a very early phase, but when IMO Tier III was announced for marine engines in 2008 it became evident that the time had come to consider the introduction of EGR technology. For this reason, an internal project was started with the target of considering and evaluating the different possibilities and, potentially, to start the necessary development activities.

Based on literature and patent research, as well as on direct information from customers and consultants, ten EGR technology families were identified (Table 3). The evaluation was conducted in various steps. A first screening was made evaluating the thermodynamic potential and feasibility on a conceptual level. In a second step simulation models were established for more detailed analysis. An important element was the use of a model for the reliable prediction of NOx emissions even outside of known operating conditions. For the most interesting technologies the required components were realized based on the simulation results. Finally the systems could be tested on experimental engines. In all cases the test results were good in line with the predictions, no preliminary matching test was necessary.

3 Requirements and technology review

Every device for industrial application on large combustion engines must obviously fulfill several requirements regarding feasibility, technical risk, cost, durability and serviceability. In the case at hand there are additional requirements that apply specifically to the application of EGR on large engines. Some of them apply to all large engine segments:

a) The ability to fulfill the emission limits (e.g. IMO Tier III for marine engines). Since the introduction of EGR has a certain impact on engine design and operation, it is expected that no additional measures are required for emission reduction.

b) Scavenging should be still possible. This issue is discussed in detail in the next section.

Some additional requirements are special for marine engines:
c) Availability: Reliability is in general a strong requirement on large engines, on a ship the main engine must always be available for safety reasons. Solutions where the engine can run in case of an EGR failure are thus mandatory.
d) Switchability. The IMO regulations say that an engine must fulfill the Tier III limits within the Emission Controlled Areas (ECAs) and the Tier II limits when sailing on the open sea. Since EGR gives a certain fuel consumption penalty and is not necessary for Tier II, it is an obvious step to envisage operation without EGR outside the ECAs.

In the following these requirements will be considered for the evaluations of the different technologies.

### 3.1 Internal EGR

Simulations have been performed to evaluate the potential of internal EGR. In Figure 5 it can be seen that internal uncooled EGR has a lower effect on NO\textsubscript{x} emissions due to the higher cycle temperatures. It can also be noted that specific fuel consumption and exhaust gas temperatures increase, but the slope is moderate up to an equivalent EGR rate of about 20\% and then becomes very steep, leading to unacceptable values. It is evident that internal EGR does not fulfill the emissions limits as stated in a). Simulation results could be validated on a test engine equipped with the valve control management VCM. There are ideas for effecting some cooling of internal EGR, e.g. with direct water injection, but this would require large amounts of water and has yet unknown consequences on engine durability.

### 3.2 Scavenging

On automotive engines turbocharging efficiencies are low and valve overlap is absent. The exhaust gas pressure is mostly higher than the pressure on the inlet side of the cylinder, then it is very simple to use this pressure difference to drive high pressure EGR.
Large engines are conceived for scavenging, which is incompatible with the pressure driven HP-EGR concept. Scavenging is an absolute must for 2-stroke engines but is also very important for large 4-stroke diesel engines for the following reasons:

- Scavenging allows more air to be introduced into the cylinder and consequently increases the power density of the engine for a given boost pressure level
- Scavenging allows a reduction in exhaust gas temperature by dilution, which is a must on engines burning heavy fuel in order to avoid deposits on the turbine
- Scavenging has a very important effect on engine thermal load, because by substitution of the residual gas, the cycle temperature is considerably lowered.

The latter effect must not be confused with a cooling effect of the scavenging air. Heat transfer from the cylinder walls to the scavenging air is marginal; the temperatures of the critical engine components are mainly reduced because scavenging reduces the heat input by removing the hot gas.

Reducing the exhaust gas temperature has always a detrimental effect on the gas exchange work. On the other side the scavenging process is not linear. The additional benefit on the engine operation of additional scavenging is exponentially decreasing with increasing scavenging flow. It is then important to limit scavenging flow.

These effects are summarized in Table 1 in the simulation results for a medium-speed engine with and without scavenging. Boost pressure and air-to-fuel ratio are kept constant, therefore engine power must be reduced.

<table>
<thead>
<tr>
<th>Δp engine [bar]</th>
<th>Ref.</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overlap std</td>
<td>std</td>
<td>reduced</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Power [%]</td>
<td>100</td>
<td>82</td>
<td>90</td>
<td>84</td>
</tr>
<tr>
<td>Δ Spec. fuel cons. [%]</td>
<td>-</td>
<td>+5.7</td>
<td>+4.8</td>
<td>+6.7</td>
</tr>
<tr>
<td>Exhaust gas temp. [°C]</td>
<td>520</td>
<td>576</td>
<td>574</td>
<td>579</td>
</tr>
<tr>
<td>Δt Exhaust valve [°C]</td>
<td>-</td>
<td>+43</td>
<td>+30</td>
<td>+48</td>
</tr>
</tbody>
</table>

All these considerations support the scavenging requirement b), which leads to the exclusion of all pressure driven technologies for application on large engines.

### 3.3 Special solutions to drive EGR

Several ideas have been generated aiming at enabling HP-EGR even in the presence of a positive pressure difference over the cylinder. A first option is to make use of the pressure waves on the gas side. Simulations have shown that on an engine with pulse turbocharging, even with ideal valves without losses, a maximum EGR rate of 5% can be achieved.
High amplitude pressure waves are generated in a low volume exhaust manifold discharging into the turbine. As soon as mass is discharged in another direction (EGR) the wave amplitudes are strongly reduced. The reed valves used in the automotive field can only improve pressure driven EGR, but will not act significantly against a pressure difference in the opposite direction. A second option is to take EGR into a zone with high air velocity and low static pressure. This zone could be a dedicated venturi or the throat section of the compressor diffuser. An evaluation has shown that these solutions are not feasible. Unlike a carburetor, where a small mass has to be mixed to the main flow, the EGR flow should be brought to a considerably higher pressure purely by means of exchange of momentum with the air. The process cannot work efficiently and would considerably reduce turbocharging efficiency.

Other ideas involve additional exhaust or inlet valves with unconventional timing to enable EGR. Simulations have shown that the impact on the engine cycle and consequently on efficiency is very large.

### 3.4 EGR pumps

Taking into account the preceding considerations, the conclusion is that some kind of EGR pump is definitely required. The function of the EGR pump can be represented by different elements of the system:

- The main compressor in case of LP EGR
- Some of the engine pistons in case of donor cylinders
- A dedicated EGR blower
- A dedicated EGR turbocharger

#### 3.4.1 LP-EGR

LP-EGR is the simplest method to implement EGR without a large impact on the layout of the engine. For this reason it has also been used as a first step in engine tests, to verify engine response to EGR [11]. The advantages of this system are:

- No need for an additional EGR pump
- Freedom in the layout of the EGR path
- Very good switchability

These advantages are to be weighed against following demerits:

- Less favorable thermodynamic results than with other systems
- Space requirements for the EGR path due to the lower density
- Reliability: Exhaust gas is mixed with air and goes through the main compressor, which works at higher temperature and with a higher risk of damages from corrosion/erosion.

The latter point is considered the most critical. LP-EGR appears not to be feasible on large engines burning low quality fuels and using advanced turbocharging systems. An additional issue is that a failure in the EGR...
path involving the main compressor would also impair engine operation even without EGR.

### 3.4.2 Donor cylinder

According to requirement b), all methods that make scavenging impossible should be excluded, but the donor cylinder concept is worth some additional consideration, because the pump function is given to a limited numbers of cylinders, i.e. scavenging is possible for some cylinders. Different configurations are possible and some of them have been evaluated by means of simulations in comparison with a reference case with an EGR blower. The case study has been conducted for a model of a small medium-speed engine with six cylinders and a mean effective pressure of 23 bar. The results are given in Table 2 in term of exhaust gas temperature $t_{TI}$, which may be taken as an indicator of the amount of scavenging and thermal load, as well as specific fuel consumption. All cases are matched to the same level of NO$_x$ emissions.

**Table 2: Simulation results with donor cylinder compared to EGR blower.**

<table>
<thead>
<tr>
<th>N. of donor cyl.</th>
<th>0/6</th>
<th>6/6</th>
<th>3/6</th>
<th>3/6</th>
<th>2/6</th>
<th>2/6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power std cyl. [%]</td>
<td>100</td>
<td>-</td>
<td>100</td>
<td>110</td>
<td>100</td>
<td>107</td>
</tr>
<tr>
<td>Power donor cyl [%]</td>
<td>-</td>
<td>100</td>
<td>100</td>
<td>90</td>
<td>100</td>
<td>85</td>
</tr>
<tr>
<td>$t_{TI}$ non donor [°C]</td>
<td>560</td>
<td>-</td>
<td>589</td>
<td>612</td>
<td>583</td>
<td>605</td>
</tr>
<tr>
<td>$t_{TI}$ donor [°C]</td>
<td>-</td>
<td>680</td>
<td>691</td>
<td>612</td>
<td>751</td>
<td>605</td>
</tr>
<tr>
<td>Δ Spec. fuel cons. [%]</td>
<td>-</td>
<td>+6.4</td>
<td>+1.2</td>
<td>+1.2</td>
<td>+2.5</td>
<td>+0.7</td>
</tr>
</tbody>
</table>

The table shows that, generally, the donor cylinders operate without scavenging and consequently with higher thermal loading, while the others operate at a higher mechanical load because they need to compensate the reduced output of the donor cylinders. The penalties could be balanced, in the best case, by a simpler engine concept without external EGR pump, but control complexity and long term reliability are open question.

### 3.4.3 EGR Turbocharger

A turbocharger is a turbo-machine which provides compressed air for the combustion and takes the power from the expansion of the exhaust gas. An EGR turbocharger does not correspond to this definition. It is properly called an EGR blower driven by an exhaust gas power turbine, which takes additional power from the exhaust gas. Due to the different conditions of mass flow and pressure ratio it would be convenient in energetic term to match the compressor and power turbine independently and transmit the power flexibly by suitable means. Of course, a direct connection is preferred for practical reasons, but it is challenging due to a serious mismatch between compressor and turbine. The mismatch and the low compressor pressure ratio are the reasons why high efficiency cannot be expected from an EGR turbocharger. According
to the application, values in the range of 30 to 60% are realistic. The effective efficiency is further reduced because a throttle valve at the EGR turbine intake is required to control the EGR rate. The control margin for this throttle valve must be rather high especially for applications with variable engine speed. With fixed geometry and without throttling, the EGR rate would be approximately constant over engine load but variable with engine speed.

For the engine the gas flow used to drive the turbine represents an energy loss (wastegate), hence a reduction in turbocharging efficiency cannot be avoided. The turbocharging efficiency change is influenced by the EGR turbocharger efficiency, since it defines the required exhaust gas flow for a given EGR rate. This influence is obvious but it is very small; for a 1% change in turbocharging efficiency the efficiency of the EGR turbocharger must be changed by roughly 10%.

The relation between the efficiencies of the EGR turbocharger and the turbocharging system is valid in general, but the consequences are different for 2- and 4-stroke engines.

The 2-stroke engine requires EGR-rates between 30 and 40%, the penalty on turbocharging efficiency is very high, comparable to that in power turbine applications (Figure 6). Since 2-stroke engines require very high turbocharging efficiency, the application of EGR turbochargers has not been considered for that kind of engine.

For applications on 4-stroke engines with single stage turbocharging the situation is better, because the required EGR rate is lower and the 4-stroke engine is less sensitive to turbocharging efficiency variations.

### 3.4.4 EGR Blower

The EGR blower works as an external EGR pump, like the turbocharger, but the gas turbine is replaced by an electric drive. This simplifies the system layout, but the challenge is that electrical machines have a much lower power density than gas turbines. In order to keep installation dimensions in an acceptable range, high-speed machines must be considered, and these are an emerging technology still having rather high costs. An alternative solution with conventional motor and gear-box is not considered competitive for reasons of weight, space requirement and cost.

The main advantages of the EGR blower vis-à-vis the EGR turbocharger are its higher flexibility, since the machine can be controlled independently of engine speed, and higher turbocharging efficiency due to the elimination of the gas flow needed to drive the gas turbine. On the negative side are the cost and complexity of the electrical drive as well as the limited
power. An electric motor has a design power with limited overload possibilities, whereas the gas turbine power is only limited by the energy available in the system.

The EGR blower is the preferred solution for 2-stroke marine engines, because its positive effect on turbocharging efficiency predominates against the potentials for lower cost and better engine efficiency given by the EGR turbocharger [3, 10].

### 3.5 Evaluations

The interactions of the turbocharging system with external EGR pumps and LP-EGR can be studied by means of simple energy balance tools, when the boundaries with the engine are kept constant. Some results are shown in Figure 7. It can be seen that the largest negative impact on $\Delta p$ over the cylinders is given by LP-EGR. This is due to the fact that a pressure difference is needed to drive the gas between the turbine outlet and the compressor inlet and that the mixed temperature at the compressor inlet is higher than ambient. Both effects have a direct impact on the efficiency of the main turbocharging system. An EGR blower introduces some energy into the system, therefore the $\Delta p$ can be slightly increased. The EGR turbocharger gives a small $\Delta p$ reduction. In the right diagram results from the combined effects of the gas exchange and the drive power required for the EGR blower are presented.

In general, the efficiency losses grow with the pressure ratio. The EGR blower shows a different behavior: The efficiency loss is higher than with LP-EGR at low pressure ratio and lower than EGR turbocharger at a very high pressure ratio. The EGR turbocharger shows a different behavior: The efficiency loss is higher than with LP-EGR at low pressure ratio and lower than EGR turbocharger at a very high pressure ratio.

**Figure 7. Efficiency analysis of the EGR pumps**

- **Application field**: 1-stage turbocharging
- **Boundary conditions**: - 20% EGR rate at 100°C - $\lambda_v$ = constant
- **Efficiencies**: Turbocharger 65%, blower 70%, EGR turbocharger 60%
- **$\Delta p$ for LP EGR**: 100 mbar
- **bem = 22 bar**, air consumption 7 kg/kWh
high pressure ratio.
The review of the considered EGR technologies is summarized and evaluated in Table 3. High pressure EGR with blower or turbocharger as the EGR pump provide the best performance. LP-EGR or donor cylinders may be considered as simpler-to-implement alternatives with some performance penalties.

Table 3: Evaluation of the considered EGR methods.

<table>
<thead>
<tr>
<th>EGR-Method</th>
<th>NOx-reduction</th>
<th>Scavenging</th>
<th>Impact on engine design</th>
<th>Impact on TC design</th>
<th>Impact on efficiency</th>
<th>Cost</th>
<th>Overall assessment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>LP</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>•</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>HP with Blower</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>○</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>HP with TC</td>
<td>●</td>
<td>●</td>
<td>•</td>
<td>•</td>
<td>•</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>Donor cylinders</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>○</td>
<td>•</td>
<td>●</td>
</tr>
<tr>
<td>Momentum</td>
<td>○</td>
<td>•</td>
<td>•</td>
<td>•</td>
<td>•</td>
<td>●</td>
<td>○</td>
</tr>
<tr>
<td>Back pressure</td>
<td>●</td>
<td>●</td>
<td>•</td>
<td>●</td>
<td>•</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>Pressure peaks</td>
<td>○</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>•</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>Add. Exhaust valve</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>•</td>
<td>●</td>
<td>●</td>
</tr>
<tr>
<td>Add. Inlet valve</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
<td>●</td>
</tr>
</tbody>
</table>

4 2-stage turbocharging

Most of the considerations so far are of a general nature and apply to systems with 1- and 2-stage turbocharging. There are a few points to be discussed specifically regarding 2-stage turbocharging systems.

4.1 EGR Turbocharger

Since pressure levels and turbocharging efficiency are high in the case of 2-stage turbocharging, the engine Δp is increased and the EGR compressor works at a higher pressure ratio. The turbine driving it can work in parallel to the high pressure (HP) turbine, i.e. with a lower expansion ratio compared to a single stage system. The consequence is that the mismatching problem is reduced and higher efficiencies can be achieved.

4.2 Semi short route

The existence of easily accessible points between the two turbocharging stages leads to the idea of connecting the exhaust manifold with the duct between the com-
pressor stages. In this way the function of the EGR pump is assumed by the HP compressor (Figure 8).

Efficiency is lower because the EGR gas must first be throttled from exhaust manifold pressure to the air pressure level between the compressors, then cooled and compressed to the boost pressure by the HP compressor. Being that the compressor mass flow rate is larger than that of the turbine a certain mismatch is also present.

With this method an EGR mixture is compressed by a main compressor as in the case of LP-EGR, but the situation concerning fouling, corrosion and erosion might be less challenging since the HP compressor operates at a low pressure ratio. This system has already been run on research engines using high quality fuel with very low sulfur content [4].

### 4.3 Donor cylinder on semi-short route

A further idea is the combination of the preceding layout with a donor cylinder (Figure 9). In this solution the donor cylinders work with a larger $\Delta p$ than the other cylinders, and cylinder scavenging is even improved. Also, throttle losses are eliminated, and the reduced back pressure for the donor cylinders can be converted into piston work. The simulation results of the two semi-short route systems are compared with an EGR turbocharger and the classic donor cylinder in Table 4. It can be seen that the donor cylinder on the semi-short route provides very good results, comparable with those of the EGR turbocharger. The donor cylinder itself makes 6% more power with same fuelling as the other cylinders thanks to the improved gas exchange work. The flow through the throttle valve between the exhaust manifolds changes direction compared to the classic donor case. Consequently only one donor cylinder out of 6 is required, because additional EGR gas can be taken from the exhaust manifold of the remaining 5 cylinders.

![Figure 9. Semi-short route + donor cylinder layout (Patent pending)](image)

Table 4. Simulation results with 2-stage turbocharging

<table>
<thead>
<tr>
<th></th>
<th>EGR-TC</th>
<th>Semi-short</th>
<th>Donor semi-short</th>
<th>Donor classic</th>
</tr>
</thead>
<tbody>
<tr>
<td>N. of donor cyl.</td>
<td>0/6</td>
<td>0/6</td>
<td>1/6</td>
<td>2/6</td>
</tr>
<tr>
<td>Power std cyl. [%]</td>
<td>100</td>
<td>100</td>
<td>99</td>
<td>106</td>
</tr>
<tr>
<td>Power donor cyl [%]</td>
<td>-</td>
<td>-</td>
<td>106</td>
<td>88</td>
</tr>
<tr>
<td>$t_{TI}$ non donor [°C]</td>
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<td>550</td>
<td>545</td>
<td>530</td>
</tr>
<tr>
<td>$t_{TI}$ donor [°C]</td>
<td>-</td>
<td>-</td>
<td>510</td>
<td>580</td>
</tr>
<tr>
<td>$\Delta$ Spec. fuel cons. [%]</td>
<td>-</td>
<td>+1.4</td>
<td>+0.1</td>
<td>+2.2</td>
</tr>
</tbody>
</table>
5 EGR turbocharger

5.1 Requirements

Taking into account the energy balance of a turbocharger the following relationship can be established for the tip speed ratio of the turbine:

\[ \nu = \frac{\dot{u}_T}{c_0} = \frac{D_T}{D_C} \sqrt{\frac{\dot{m}_T \cdot \eta_{Vol} \cdot \eta_{ST} \cdot \eta_{Mech}}{2 \cdot \mu_{ac}}} \]

\( \eta_{Vol} \) ... Compressor volumetric efficiency
\( \eta_{ST} \) ... Isentropic turbine efficiency
\( \eta_{Mech} \) ... Mechanical efficiency
\( \mu_{ac} \) ... Specific compressor work \( (\Delta h/\dot{u}_c^2) \)

For a typical turbocharger the diameter ratio is in the range \( D_T/D_C = 0.9 \) to 1 and the mass flow ratio \( \dot{m}_T/\dot{m}_C \) is close to 1. Considering typical values for the efficiencies and the specific compressor work the tip speed ratio can be calculated. The result is usually very close to a value of 0.7, which gives the best turbine efficiency. In the case of the EGR turbocharger the mass flow ratio is around 0.25, leading to a tip speed ratio below 0.3 and turbine efficiency below 50%. In this case, reducing the specific compressor work can help, but the only possibility for substantially increasing the tip speed ratio is to increase the diameter ratio. Unfortunately this leads to unusual requirements in term of component design: the compressor has a high mass flow rate with a small diameter, i.e. a very high specific flow capacity; the turbine must have a large diameter and a very low flow capacity; extremely small turbine trims are required. Calculated turbine efficiency for a representative case in dependence of the diameter ratio is shown in Figure 10.

For applications on 4-stroke engines with single stage turbocharging an EGR turbocharger with diameter ratio around 1.4 is considered feasible.

The opportunity for a function test with an EGR turbocharger came on a medium-speed engine with a 2-stage turbocharging system. As mentioned above, the mismatching problem is less critical with 2-stage turbocharging: the mass flow ratio is around 0.5 and a diameter ratio of around 1.25 is sufficient to reach a good turbine efficiency.

![Figure 10. EGR turbocharger: turbine efficiency vs. diameter ratio.](image)
5.2 Design

A technology demonstrator had been derived from a serial TPS44F turbocharger. The turbine is a serial one, except that the trim has been chosen one position lower than the usual range. The compressor stage, on the other hand is a dedicated design for the application. The configuration giving the largest available specific volume flow at low pressure ratio has been chosen and the diameter has been reduced by about 30%. Since the operating temperature is in the range of 150 to 200°C and the medium may be aggressive, the compressor wheel has been milled from a turbine steel.

Additional attention has been paid to the sealing on the backside of the compressor. Design optimization and sealing air have been used to avoid any contamination of the bearing zone with exhaust gas. The rotor and the turbocharger can be seen in Figure 11.

![EGR turbocharger](image)

**Figure 11.** EGR turbocharger.

5.3 Consideration regarding switchability

Since the application was intended for a marine engine, switchability was a requirement. The system has been matched for operation with EGR. Thus the turbochargers are too small for operation without EGR. Without any variability this would cause overpressure and run beyond the speed limit and into the choke limit.

The simplest solution has been chosen as a control device [5]: a bypass for the HP turbine to be opened in operation without EGR (Figure 12). By opening the bypass the speed of the HP turbocharger is reduced and the operating point of the LP compressor moves back from the choke line. Naturally the turbine bypass rep-
resents a certain energy loss for the turbocharging system. A better solution would be the extended use of variable valve timing, as enabled by the ABB Turbo Systems product VCM, Valve Control Management [6]. The simulation results with the two control options are shown in Figure 13 for full engine load. The evolution of the operating point by switching EGR off is shown in the compressor maps as well as in a trade-off diagram showing specific fuel consumption and NO$_x$ emissions. Operation without EGR and with turbine bypassing represents a compromise to allow running the engine in an acceptable manner without advanced variability. An advanced variable valve timing system like VCM allows optimization of the Miller effect and valve overlap matched to the given turbocharging system. Of course the VCM system would also offer the potential to optimize the part load operation of the engine in both operating modes.

![Diagram](image)

Figure 13. Operating points in EGR off mode.

### 5.4 Experience from engine tests

The thermodynamic results from the engine tests have been reported at the last conference [7]. The predictions based on simulations have been substantially validated and the EGR turbocharger performed well during all the phases of the project, with efficiencies around 60%. The experience can be considered fully positive from the thermodynamic point of view. A major issue during the tests was fouling in the EGR path. This is due to the fuel used: marine distillate (MDO). Typical fuel for marine applications is heavy fuel oil (HFO) with sulfur content in the region of 3% = 30000 ppm. For this kind of fuel positive results have been achieved by introducing a wet scrubber in the EGR path to remove the oxides of sulfur (SO$_x$) [10]. MDO is considered a low sulfur fuel with 1000 ppm, which is still much higher than the concentrations found in the automotive field, i.e. below 10 ppm. The EGR turbocharger was inspected in ABB Turbo Systems facilities after about 300 running hours (Figure 14).
The heavy fouling that can be seen in the left hand picture is tolerable, since the compressor has proven to be very robust in term of its performance stability. However, the corrosive attacks that can be seen after cleaning show that the concept is not yet mature. There is margin for improving the corrosion resistance of the materials employed, but the EGR gas cleaning must be improved [9].

6 EGR blower

As mentioned above, the EGR blower solution has been adopted for the first EGR applications on large 2-stroke engines. The preliminary evaluations have shown that it could also be an alternative solution for small 4-stroke engines, especially when the requirements concerning variable engine speed cannot be fulfilled by an EGR turbocharger. To verify this idea it was decided to produce a technology demonstrator of an electrically driven blower for application on small 4-stroke engines. A major issue is here to achieve a power density in the electric drive which is in the same range as an EGR turbocharger, i.e. to apply a high-speed electric motor.

High-speed electric motors are an emerging technology. Examples are offered for short operating time applications as e-boosters for automotive applications, but industrial machines for continuous operation are rare and very expensive. For the present application an innovative technology under development at ABB was chosen, using an induction motor. The diesel engine used in the tests has a power output around 1000 kW. The de-
sign power for the electric motor has been set at 20 kW, i.e. 2% of engine power. The volume flow for the EGR blower is about 1/3 of the compressor stage of the EGR turbocharger described in section 5. Scaling down the stage would lead to a much smaller diameter and a speed of about 85000 rpm, which would be challenging in term of feasibility and efficiency. It has been decided to maintain the same diameter for the EGR blower and to trim the compressor wheel to reduce the volume flow by a factor of 3. Consequently, a new air casing was designed, since the serial part would have been too large for the application. The design point for the electric motor was finally fixed to 20 kW at 50000 rpm. The machine is shown in Figure 15. Major challenges on a high-speed electric motor are its bearings and cooling system. Since the machine was meant as technology demonstrator for function tests on an experimental engine, it had to run for a limited number of hours. As a result the bearings and cooling system were procured with minimum effort using pragmatic solutions.

6.1 Thermodynamic evaluations

The thermodynamic potential with the EGR blower has been compared by means of simulation with that of the EGR turbocharger and of the donor cylinder concept. The engine model is a small medium-speed 4-stroke for application on a locomotive and operating on a line which is very close to a propeller curve. The engine has one single stage turbocharger.

The diagram in Figure 16 shows the resulting curves for exhaust gas temperature, specific fuel consumption and air-to-fuel ratio. Following considerations can be made for the different technologies:

**EGR blower**

The specific fuel consumption for the engine alone is the best. Taking the electric drive power consumption into account it is slightly higher than the turbocharger in the full load region only. Regarding thermal load the best values are achieved, with the exception of the lowest point where EGR is deactivated.

**EGR turbocharger**

It gives the best specific fuel consumption at full load and very low
load, due to the smaller turbine. The thermal load is higher than with the blower, but still may be considered acceptable. Critical is the point at low load and reduced speed, where the air-to-fuel ratio is very low.

*Donor cylinder*

The optimized configuration (see Section 3.4.2) gives the worst values regarding fuel consumption and thermal load among the solutions considered for this engine. The thermal load is considered unacceptable for a highly loaded medium-speed engine designed with cylinder scavenging. The comparison is made keeping the boundary conditions constant for all options. Further optimization is possible considering a variability for the main turbocharger [8] as well as changing the EGR rate curve over engine load.

### 6.1 Experience from engine tests

The thermodynamic performance of the EGR blower was good during the whole test period (350 running hours). A very low sulfur fuel with 8 ppm was used and in the short operating time no significant fouling or corrosion was observed. Under some operating conditions the electric motor power was not sufficient because a higher EGR rate was required to fulfill the goals. The bearing performance showed sensibility to the operation stress, as these components are highly loaded. The preliminary design used in the test required a very careful operation.

### 7 Summary and Outlook

EGR is a technology with the potential to enable large diesel engines to fulfill the future regulations on NOx emissions. A wide range of EGR technologies has been analyzed with focus on their application for large diesel engines. The results of the study is that EGR turbocharger and EGR blower are the most suitable technologies for large engines. An EGR turbocharger technology demonstrator has been applied on a marine medium-speed engine with 2-stage turbocharging. The thermodynamics of the concept were validated successfully. The operation of EGR with MDO as a fuel needs further development to keep fouling and corrosion under control. A technology demonstrator for an EGR blower has been applied to a small medium-speed engine for traction or marine applications with low sulfur fuel. In this case, also good thermodynamic results were achieved, but since the degree of innovation is higher, more development effort is needed for product industrialization. It can be said that both EGR blower and turbocharger fulfill the necessary thermodynamic requirements and the differences are considered small.
For a decision other criteria like cost, technical risk, packaging/mounting and time to market may have a higher weighting.

The specific scope of the demonstration, reduction of NO\textsubscript{x} emissions below legislation limits at acceptable fuel consumption and thermal load could be achieved. The experiences accumulated in the development and testing of the two concepts are very valuable.

No further development is currently scheduled for the two machines so far. The reason is the uncertainty of the market. The application of IMO Tier III in 2016 is under discussion and most market players have not yet communicated a clear strategy about the best technology to be applied in every segment.

Should the market demand a substantial volume of EGR pumps, ABB Turbo Systems has demonstrated its ability to be a strong partner for joint development activities.

**References**


