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The CRISTAL engine: ABC's new medium speed diesel engine, developed to comply with IMO III.

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Abstract: Rudolf Diesel demonstrated his compression ignition engine at the World Fair in Paris in 1900. One year earlier, the first Diesel engine outside of Germany was built under license by the Carels Brothers in Ghent, Belgium. In 1912, this license was brought into the founding of the Anglo Belgian Corporation (ABC). Now ABC is 100 years older and celebrates its centennial jubilee. During this time, the engines have undergone tremendous progress, and are produced for applications all over the world. But, with the increasing focus on emissions and fuel consumption, ABC has taken the next challenge to design and build a completely new engine range with a power output of 650kW/cylinder at 750rpm which is developed to meet the IMO III emission level with engine internal measures. This engine is designed with state of the art components and a unique charging system which has to make it possible to reach the IMO III limits. Furthermore, the engine is developed to work inside and outside the ECA zone's, as well on MDO, HFO and Dual Fuel. The base design of the engine is foreseen to work at different speeds on nominal torque so that the engine has its main applications in as well power generation as marine propulsion. This will make it a multifunctional engine which will set the standard in its category. This paper will describe this new developed engine characteristics and will highlight the new technology that is used to reach the targeted IMO III limit, engine internally. It will include a discussion on the different issues as there are, mechanical design, thermodynamics, emissions, fuel consumption, ... We will also describe the current status of the development and show the available test results.

INTRODUCTION

The customer expectations for engine based systems are clearly set. The engine has to meet the limits set out by the different emission legislations, and with the highly increasing prices of all the fuels, one also demands the lowest possible fuel consumption. But together with these basic requirements it is also expected that an engine is compact so that the cargo space is maximized with a minimal loss for the engine room. Furthermore, one prefers to have an engine with all systems integrated, so that there is no need for separately installed components or after-treatment on board the ship or in the powerhouse.

Then, once everything is running, ease of maintenance and low lifecycle cost is another important topic.

ABC has always listened to the customer needs and puts these demands as a priority for the engine design and service. Of course, requirements as “compactness” and “ease of maintenance” or “simple and robust design” and “implementation of the latest technology” are not so easy to mate. This has been a challenge for years, and certainly with the coming emission legislations, it has not become easier.

ABC succeeded already 100 years in making engines keeping this concept in mind. At the moment we have the current DX and DZ engines in our portfolio which covers a power range from 100kW up to 4000kW.

Now, with the increasing demand for higher engine power and given the above mentioned customer targets, we have set our next target in the development of a complete new engine range that extends our existing engine portfolio upwards in the power range and that complies already now engine internally with the future emission legislation of IMO Tier III. The CRISTAL-engine (Clean Reliable Innovative Sustainable Tow-stage Alternative Line-engine) is born.

ENGINE RANGE

ABC has at the moment an engine range where the DX engine covers the power range from 100kW with a 3 cylinder engine up to 885kW with an 8 cylinder turbocharged engine at 750rpm.

The DZ engine range starts at about 700kW with a 6 cylinder engine, and is available in 6 and 8 cylinder in-line version, as also a V12 and V16 version. With the V16 at 1000rpm, a power output of 4000kW is achieved.

This whole engine range is covered by the necessary emission certificates, namely IMO II, CCNR II, UIC II, EU IIIA and EU IIIB.

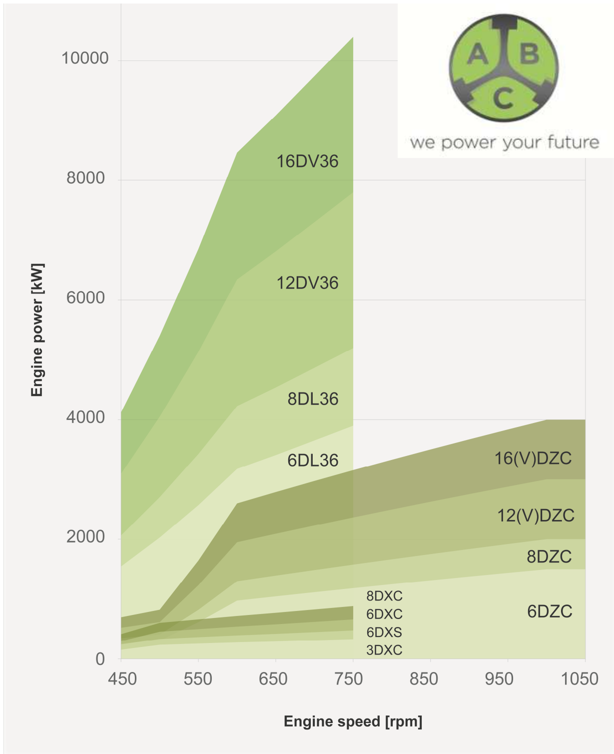

Figure 1 – Engine range of ABC.

Now the newly developed engine range denoted by D36, and with an L or V for in-line or V-engine as also 6,8,12 or 16 depending on the number of cylinders, will have a maximum speed of 750rpm. With an output of 650kW/cylinder it will deliver a power output of 3900kW for the 6 cylinder in-line engine, up to 10400kW for a V16 engine. The complete engine range is presented in figure 1.

THE D36 ENGINE DESIGN

The table below lists the main specifications of the new D36 engine range.

Main engine data of the D36 engine range.	
Cylinder power output	650kW/cyl
Nominal engine speed	750rpm
Number of cylinders	6/8 in-line 12/16 in V

Mean piston speed	10.5 m/s
Bore	365 mm
Stroke	420 mm
Swept volume per cylinder	43.9 dm³
Compression ratio	15.5
Brake mean effective pressure	23.9bar
Nominal peak firing pressure	210bar
Engine design peak firing pressure	240bar
Charging system	2-stage KBB
PTO power	100%
Fuel	MDO,HFO (Dual Fuel)
Main applications	Genset, Propulsion, Stationary

Table 1 – Main data of the D36 engine range.

The main market for this engine is foreseen to be in propulsion and in both marine and stationary generator applications. With this market target, the nominal engine speed was set at 750rpm and a power output of 10MW should be possible with 1 engine unit. To continue our history of building robust and reliable engines, we have set a moderate piston speed of 10.5m/s and a bmep (brake mean effective pressure) of not more than 24bar. With these boundaries, the bore and stroke are defined and as a next target we have set the design firing pressure for the engine at 240 bar. Although the nominal firing pressure will be around 210bar, we still have set the design firing pressure at 240bar so to have an engine with sufficient margin inside.

IMO Tier III

The IMO Tier III emission level is one of the most important emission limits that are set out for these kind of engines, so this has been chosen as a next design target.

The IMO legislation has changed during the years, where the base limit was set in the year 2000 and a next reduction by about 20% was implemented from 2011 on. Now the stage three limit implements a reduction of about 80% and will come into force from 2016 on in the ECA (emission controlled areas), which can be seen in the enclosed figure 2.

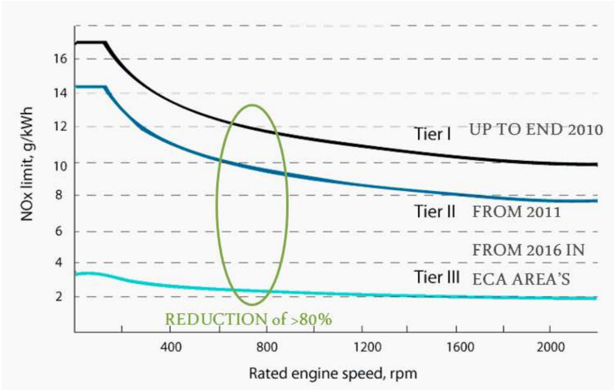

Figure 2 – IMO emission limits.

But not to forget, the IMO III limit is applicable in the ECA's, which implies that the IMO II limit is still applicable outside these zones. One also has to keep in mind that these ECA's do not only limit the emissions, but also the sulfur content of the fuel. This is an additional difficulty as the engine design must be switchable to work in both areas and has to be able to switch from fuel as well, for example between HFO and MDO.

Thermodynamical and combustion design

To reach this low emission limits of the IMO III limit, we have worked on the core issue that produces these emissions: the combustion.

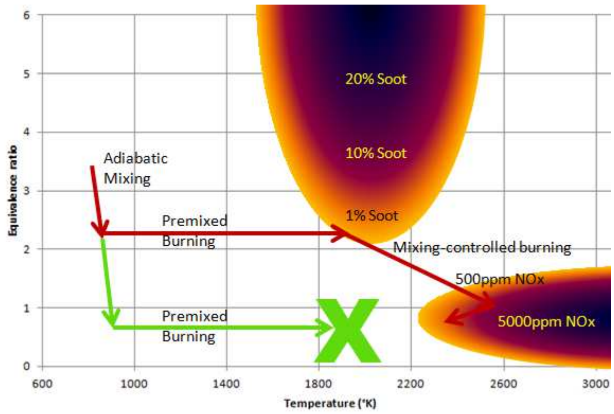

Figure 3 – Emission formation during combustion.

In Figure 3 one can see that during a normal combustion of a diesel engine the combustion process has different stages, as there are adiabatic mixing, premixed burning and also mixing-controlled burning. These stages are coming into the zone where NOx and Soot is formed when we are looking to figure 3. Now, there is another zone possible for the combustion, where there is "no" formation of NOx and Soot, this is the X-marked spot in figure 3. This X-mark has been the target in the design of the combustion process of our D36 engine range.

To reach this mentioned X-mark, we have set up a series of techniques and systems as this is not possible with a single system. A combination of systems are necessary to obtain the low emissions and stay within the IMO Tier III limits:

- Miller timing and VVT (Variable Valve Timing) system
- Two-stage charging system
- Integrated EGR system, the Eureka concept
- Advanced high pressure common rail injection and combustion system

Miller timing and VVT system.

As one of the systems to obtain a lower combustion temperature without influencing the soot and fuel consumption we have set up a Miller timing with a variable valve timing. The Miller timing closes the inlet valve before the piston reaches the bottom dead center so that the air inside the cylinder can expand and, in this way, cools down. The compression phase now starts from a lower temperature which evidently leads to a lower maximum combustion temperature and lower emission formation. This is a first step towards our X-mark. This measure has also the negative result that there is less air mass in the cylinder, with the same charging pressure, which results in a lower lambda value, what could lead to higher temperatures and a higher soot creation, which is in contradiction with the desired result. The solution for this issue is an increase of the charging pressure which is discussed later in this paper.

We have chosen for a closure of the inlet valve at 500°. This gives us the expected results at nominal load, but at part load and start up, we have a too low lambda value in the cylinder to maintain a good combustion. At the low load points, we can't increase the charging pressure as the turbocharger is almost not working. For this reason a VVT system has been chosen so that we can shift to a more normal inlet valve timing for low loads and to the Miller timing for the higher loads.

The mechanical design of the Miller system is very simple, robust and reliable. It is a fully mechanical system which gives us the possibility to shift between at least 2 different positions. A 3D image of the system can be seen in figure 4.

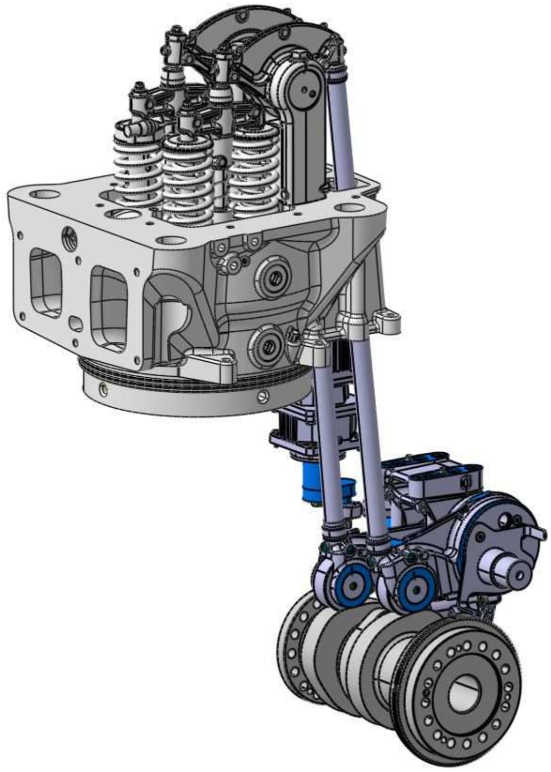

Figure 4 – VVT set up.

Two-stage charging system.

To compensate the negative effects of the Miller timing and to obtain the target lambda of two while introducing additionally EGR, a very efficient charging system was necessary.

With a single-stage charging system, one quickly reaches the limits of the compressor outlet temperature and this sets no margin, as it also limits the possibilities for a future increase of the charging system.

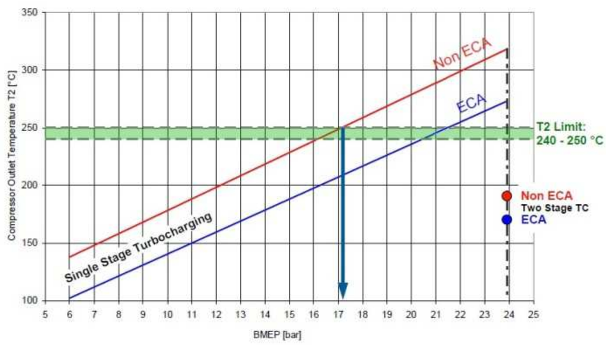

Figure 5 – Comparison between single and two-stage charging system.

In figure 5, one can see that a single-stage charging system would limit the engine to a bmep of 17bar when we are looking to the working principle in the non ECA zone. If this same engine would work within the ECA zone, then the bmep would be able to increase up to around 21bar. Again we want to have a system with some margin which has led us to the choice of a two-stage charging system.

Turbochargers of this type are not readily available on the market and the latest developments were also towards higher compression ratios of 5 and more with single-stage turbochargers. This implies that these turbochargers are optimized for these high pressure ratios, where we now need the best efficiency for the low pressure turbocharger with a compression ratio of 2.6 and the high pressure turbocharger with a compression ratio of 2.3. This leads to a total designed compression ratio of 6 which is extremely high for a single turbocharger, but which is simply feasible for two turbochargers in series. The highest compression ratio is logically chosen for the low pressure turbocharger so that we can optimize the intercooling principle which makes it possible to reach total charging efficiencies of 70% and more.

The practical set up of the turbochargers has been chosen in this way that they are installed both on the engine and that they are as close as possible to have a compact design with low load losses in as well the exhaust as the air system. After an intensive investigation of this matter, we have chosen to combine an axial low pressure turbocharger together with a radial high pressure turbocharger. This gives the advantage that both turbochargers can be built together very short on the exhaust side which gives the desired compact design as well as the ease for isolation and shielding so that the temperatures are kept below the demanded 220°C from the SOLAS (Safety of Live at Sea) regulation. Additionally, we want to stay away from the soot zone out of figure 3, so we need to foresee the needed air as well in stationary as also in dynamical behavior of the engine. By choosing an axial design for the low pressure turbocharger, which is also the largest and normally has the biggest inertia, this one will accelerate much faster compared to a radial one. Then, the smaller high pressure turbocharger is of the radial type, but as this one is much smaller it will not be the limiting factor. If we additionally compare the two-stage design with the single-stage it is also clear that two smaller chargers reach much faster the needed air pressure. This can be seen in the figure 6, where at a certain time the opacity level from a single-stage charging system goes up to 4 times the level obtained with a two-stage charging system.

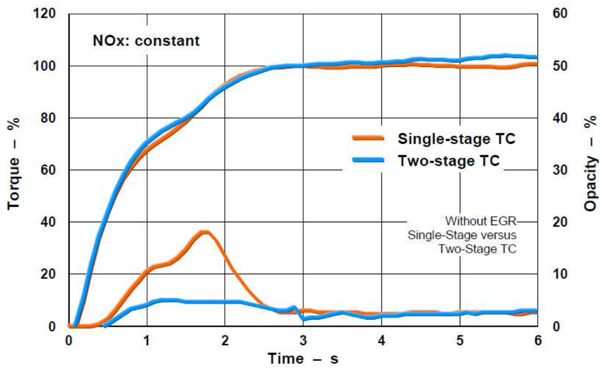

Figure 6 – Dynamic comparison of a single-stage charging system with a two-stage charging system.

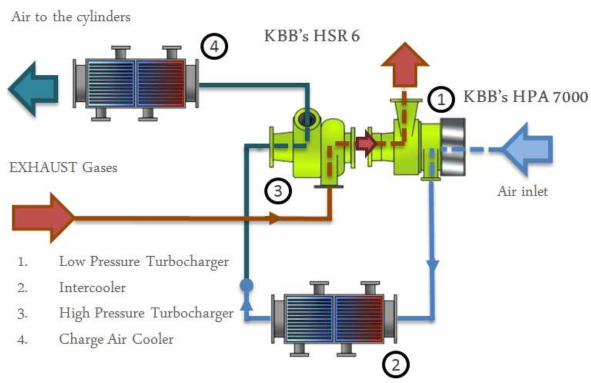

Figure 7 – Schematic set up of the two-stage charging system from the DL36 engine.

Both turbochargers have been newly designed especially for this two-stage application. The low pressure turbocharger is the HPA 7000 from KBB, whereas the high pressure turbocharger is the new HSR 6 from KBB.

<u>Integrated EGR system, the Eureka concept.</u>

As shortly mentioned before, the usage of EGR is also incorporated to meet the targeted combustion concept. By using EGR, we will use about 20% of exhaust gases in the air inlet system so that it will help to reduce the temperatures during the combustion process. The potential of EGR has already been demonstrated on a DZ-based test engine [1].

Due to the high charging efficiency the pressure in the outlet collector is lower than the one in the inlet collector. This makes that the exhaust gases will not simply flow in that direction, but this flow will have to be forced in that direction. Again different concepts have been considered, one could for example work with a kind of EGR "pump", but we have chosen to optimally use the two-stage charging concept as this gives us the perfect possibility to integrate the EGR system.

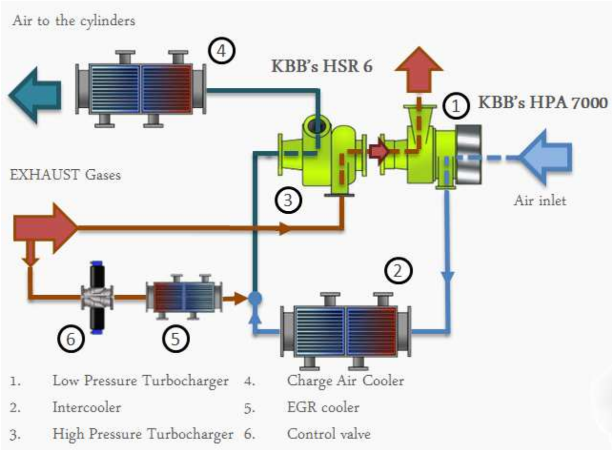

Figure 8 – Schematic set up of two-stage charging system and EGR implementation.

A part of the exhaust gases coming from the exhaust collector are deviated and cooled in a specially designed exhaust cooler which has a high temperature part and a low temperature part. After the low temperature part, a condensate trap is installed so that the cooled exhaust gases are dry. These gases are then entered just before the air inlet of the second and high pressure turbocharger, which makes that the exhaust gases are directly mixed in an ideal way. They again increase in temperature by the compression so that eventual residual moisture in the exhaust gas is directly evaporated again. Then this mixture of air and exhaust gas is fed into a second air cooler which cools it down before it is led into the air collector. We call this our "Eureka" system and the schematic set up is shown in figure 8, where the integration on the engine is shown in figure 9.

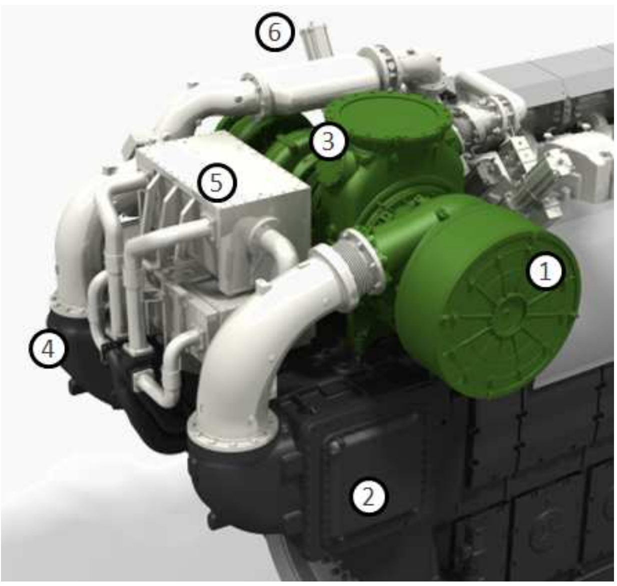

Figure 9 – Set up of the Eureka concept on the engine.

Advanced high pressure common rail injection system.

For the injection system of this engine, we have chosen a common rail injection system from Heinzmann that is able to run on as well diesel oil, MDO, IFO and HFO. The HFO will be a logical choice outside the ECA area, where the MDO with a low sulfur content will be mainly used in the ECA. A rail pressure of 1800bar is set, together with an accumulator set up of the system to have the lowest possible pressure drop during the injection.

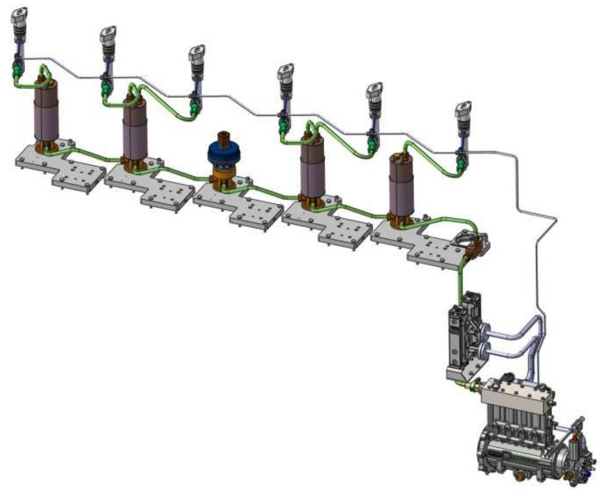

Figure 10 – Injection set up with accumulators and 1 central pump.

The combustion itself

To have an optimal combustion, we have set up the ideal mixture in the inlet collector, as described in the above paragraphs. We have made the required mixture of air and exhaust gas and delivered it to the collector at the cooled temperature of about 50°C.

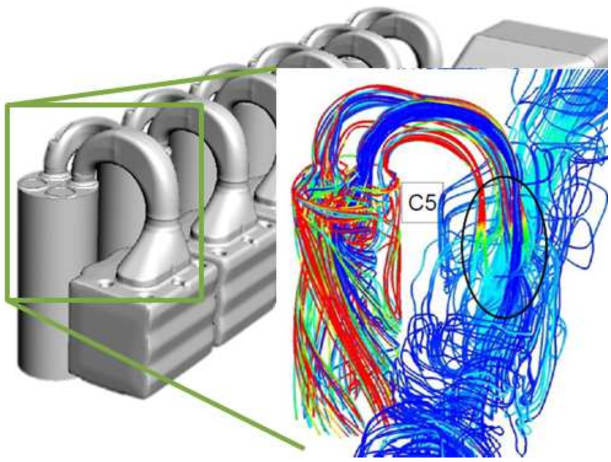

Figure 11 – Velocity simulation in the inlet system

Now we have to fill the 6 cylinders as equally as possible and therefore we have simulated the velocity inside this collector and inlet system to the cylinder, see figure 11. When comparing the flow pattern of the different cylinders, we have seen that the flow is quite similar between them. In this way we can expect that every cylinder is supplied with the same mixture and the same mass of air and exhaust gas mixture.

As a following step, the behavior of the gases inside this cylinder has been evaluated. The swirl level and the velocity inside the cylinder has also been simulated and can be seen in figure 12.

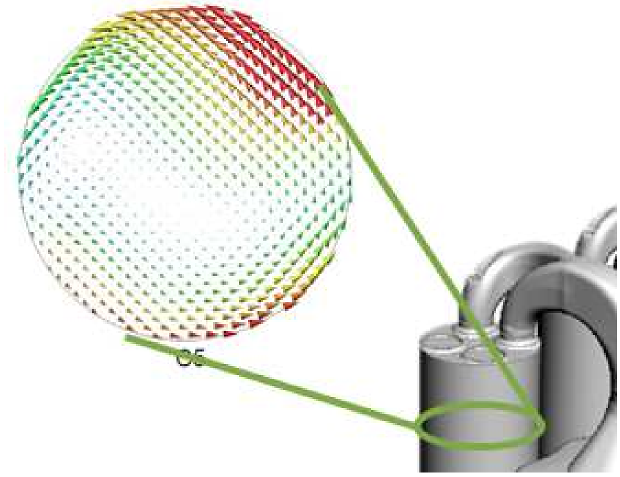

Figure 12 – Velocity distribution inside the cylinder.

With this information and the chosen injectors, we have also put those two together to see how they interact. In figure 13 one can see the different stages of the injection and although every spray looks symmetrical at first sight, especially the lower spray tends more to the right then to the left. This

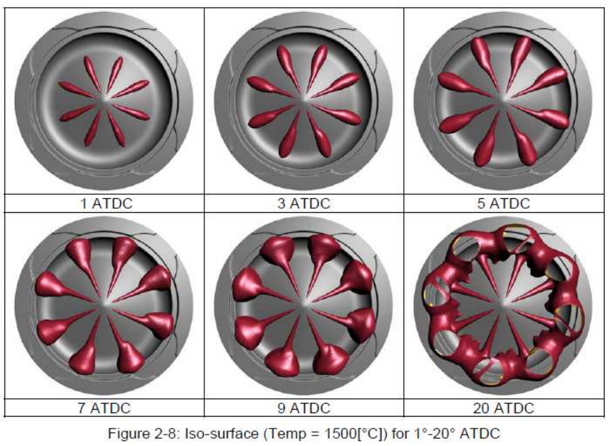

Figure 13 – Interaction between injection and velocity inside the cylinder

is of course logical when we are looking back to the above mentioned velocity distribution inside the cylinder. These simulations are confirming us that the spray patterns are not overlapping so that there is sufficient space between the sprays and that the air can interact with the fuel to make a good mixture and a proper combustion.

As a next step, we now want to check one single spray and look how we can work on that to have the best evaporation and combustion. As this is a very hard issue to simulate, we have chosen to check this by means of measuring and looking at the evaporation and combustion of one single spray.

Based on a long tradition of cooperation with Ghent University, we have already started years ago with the design of a combustion chamber so that we could have a look into the “heart” of the engine, see [2]. Together, we have designed and installed a combustion chamber which can handle up to 300bar pressure and which is equipped with an ABC injection system so that one spray of the injector can be visualized into the smallest detail. This combustion chamber is called the GUCCI (Ghent University Combustion Chamber I) and can be seen in figure 14.

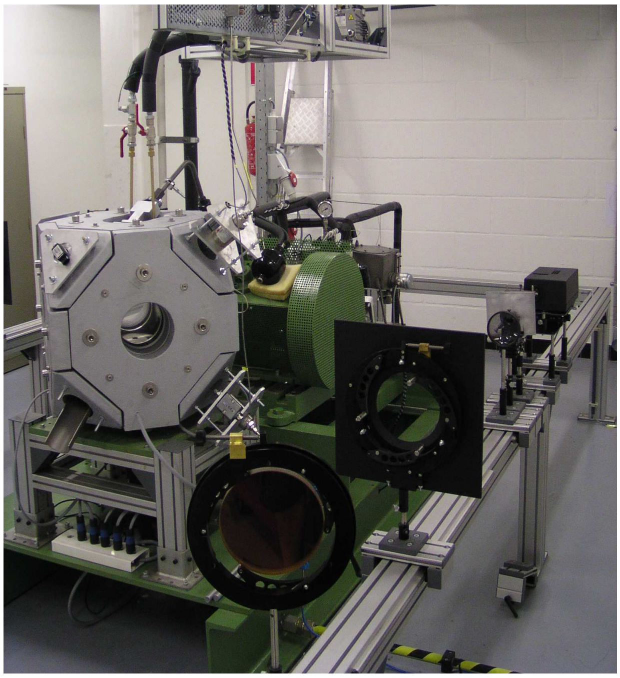

Figure 14 – The GUCCI.

With this tool we have made images of the injection pattern of one spray where we have the ability to distinguish the evaporated zone from the fluid zone, or to look at the region where the soot is formed. An example of only one injection is shown in figure 15.

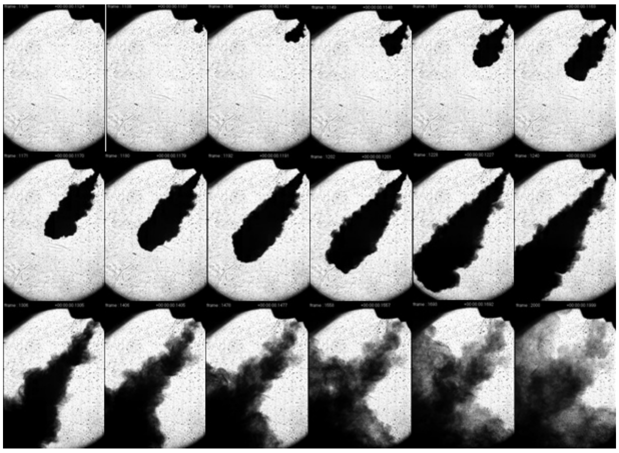

Figure 15: Injection visualized in the GUCCI.

Out of these experiments we have gathered additional information which helps us in defining the best injection pressure, injection angle and injection hole.

Crankcase

The crankcase is cast in nodular cast iron GJS 400 and uses cross bolts to the main bearing cap to ensure a stiff engine block design. The block is of

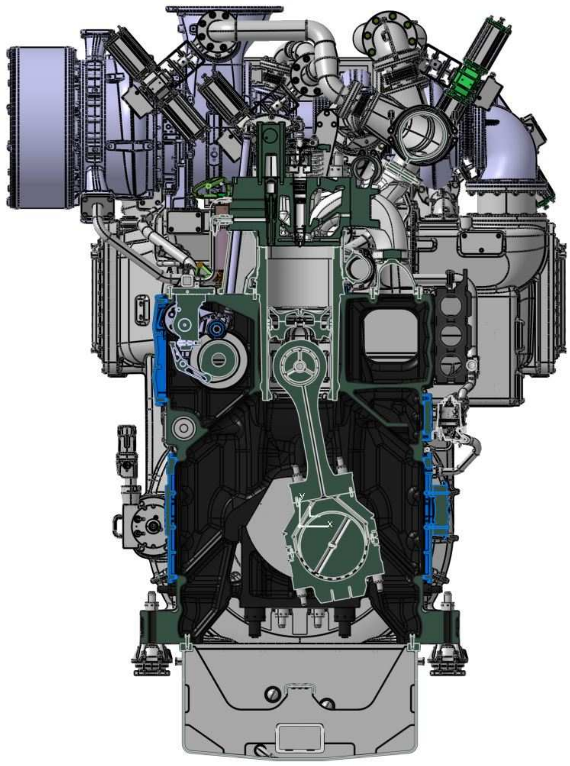

Figure 16 – Cross section of a 6DL36 engine

the dry type and a lot of systems have been integrated in the block design, see figure 16. The camshaft bearing holes are machined directly into the block so that they are perfectly positioned and give a very rigid support to the camshaft. The oil collector is also integrated in the block and the necessary oil supply lines are drilled directly in the block. On the other side of the camshaft, we have an integrated air collector with a “waved” outer shape so that the sound production of this wall is minimized. Below the air collector, a special gallery is located to collect the gases from the oil sump so that they can stabilize and so that the oil can already be separated as much as possible from the gases.

The crankcase also has two supporting beams so that the engine can be installed with the use of elastic mountings.

Conrod

The conrod is of the marine type and can be split in three parts. This part is one of the highest dynamically loaded parts of the engine and therefore an intensive study has been executed.

Starting with a forged part out of 42CrMo4-V the conrod is fully machined so that the surface roughness is fully removed. For this part, a stationary strength calculation is made, but the HCF (High Cycle Fatigue) calculation is of course of the utmost importance. To set up this calculation model, we started with the base material. Based on specific tests of the 42CrMo4-V material, a Haigh diagram is made. As it is standard for these kind of limit values, they represent a 50% survival probability (Gauss distribution). Then additional factors have to be introduced so that we reach a survival probability of more than 99.99% when taking additional influences into account as:

- Surface roughness
- Statistical influence
- Stress gradient and rearrangement
- Mesh quality
- Load dispersion

This leads then to a second and limiting Haigh diagram for the material in this application.

The conrod is then to be meshed and simulated over the complete engine cycle, so that as well the gas exchange, as also the combustion and expansion phase have to be calculated. During this calculation the highest stress level is shown and mentioned on the 3D part. In figure 17 we see the point where the highest stress level is to be expected and the stress level in this point is indicated on the diagram during the four cycles (blue dots in a triangle shape).

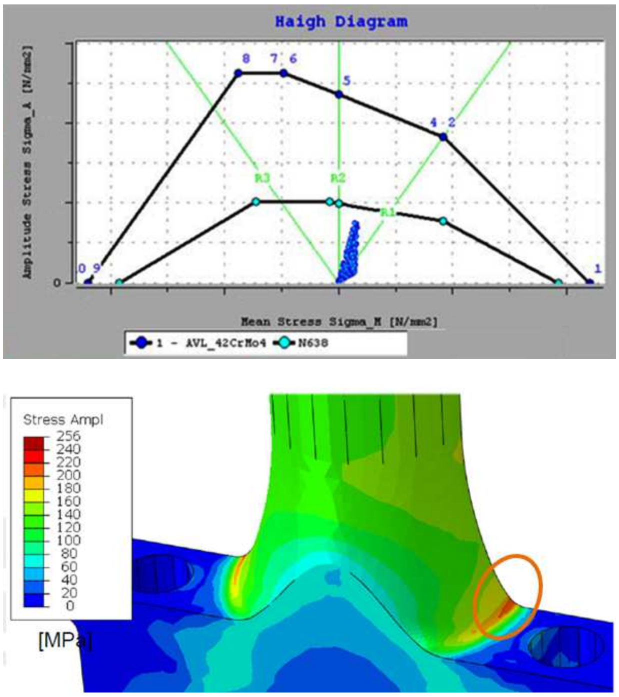

Figure 17 – Haigh diagram and indication of highest loaded point of the HCF calculation on a conrod

Cylinder head

The cylinder head is one of the most complex parts in the engine, as it has to resist the combustion pressure of 240bar, but it also has to integrate the water cooling channels, the compressed air connections to the starting valve, the oil drillings to lubricate the rocker lever, the injector with the needed fuel and cooling connections and then also the in and outlet channel with the necessary valves. Furthermore, the part must resist the high temperatures of every combustion which is a high cycle phenomena, whereas the starting, stopping and load reducing of the engine leads to a possible low cycle fatigue effect.

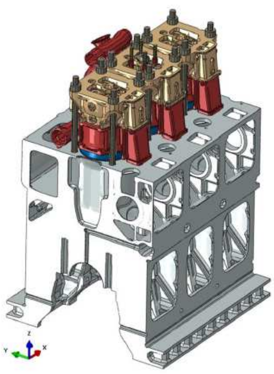

Figure 18 – Model used for the cylinder head calculation.

Not to take any risk on this component, we have executed a series of calculations to ensure the correct design of this part. As a first one we have checked the temperature distribution in the cylinder head and we see that this stays well below a 400°C temperature limit which we have set, see figure 19.

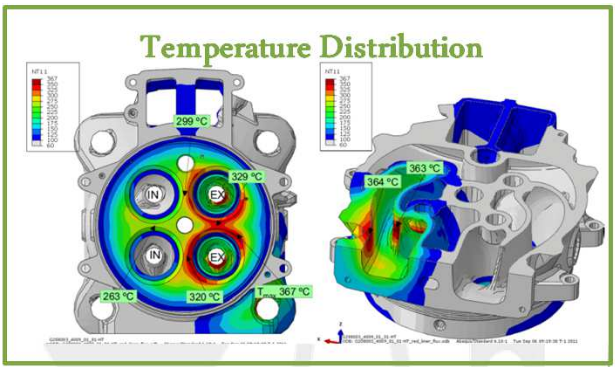

Figure 19 – Temperature distribution on the cylinder head.

Secondly the heat transfer coefficient has been calculated and evaluated. It is to be expected that this one is higher on the side of the exhaust valves compared to the inlet valves, but still one can see in figure 20 that the heat distribution is quite evenly distributed.

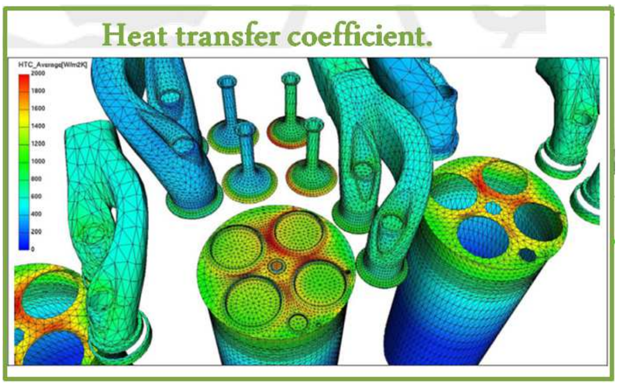

Figure 20 – Heat transfer coefficient

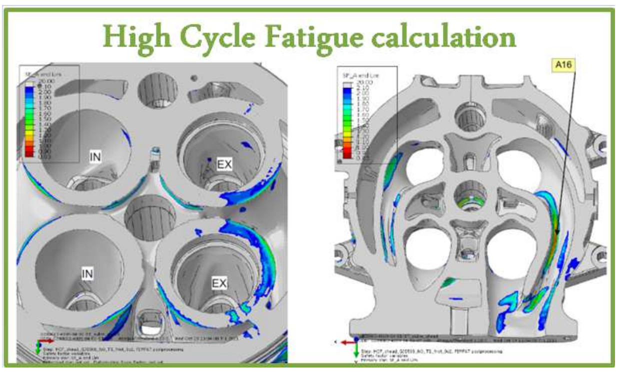

Figure 21 – High cycle fatigue calculation in the cylinder head.

As described above, the cylinder head is influenced by a high cycle phenomena where the combustion pressure has the largest influence. This HCF factor is simulated in a similar way as for the conrod and is shown in figure 21.

In the cylinder head itself, two inlet ports and two outlet ports are designed to manage the gas exchange. To maintain an equal distribution between both ports a flow simulation has been executed. In figure 22 one can see the flow velocity during the exhaust stroke which is symmetrical for both ports.

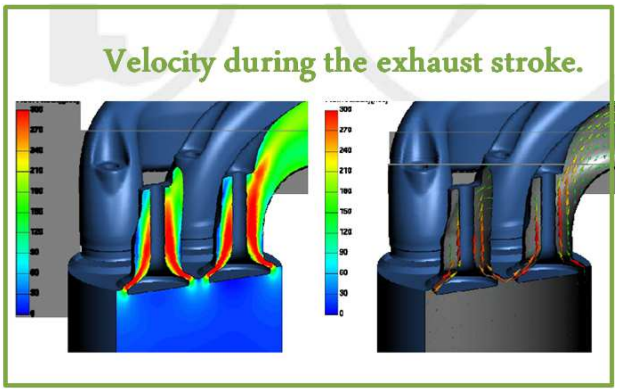

Figure 22 – Velocity in the exhaust path during the exhaust stroke.

The low cycle fatigue has also been evaluated, but here together with the liner and the engine block to see the result of the complete structure. Mainly the so called “liner flange” has been modeled into detail as this part become’s a high assembly load and is heavily submitted to the low cycle phenomena, where temperature is the main reason behind it. The points with the highest levels have been searched and can be seen in figure 23.

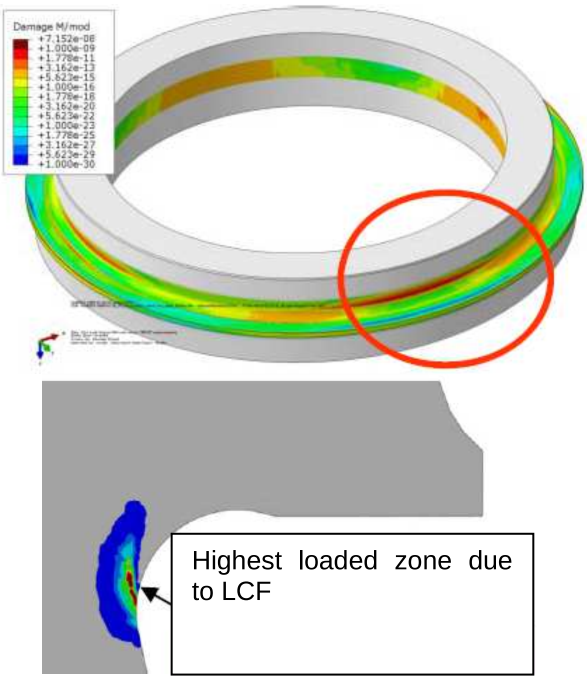

Figure 23 – LCF result on liner flange

The simulation and calculation tools have been fully used in the design phase, but also in the next step of the casting, simulation with magma soft has been executed by the foundry Focast to obtain a high quality casted part. One can see in figure 24 that the number of risers and their place has been changed to optimize the cast ability of the part.

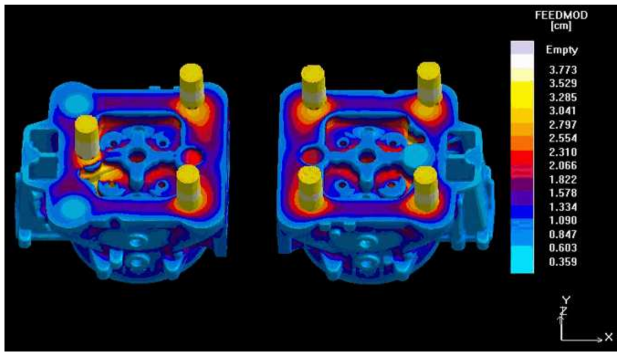

Figure 24 – Different variants of risers have been evaluated with Magma soft.

Serviceability

As one of the other main targets of the engine design, we wanted to have an engine which is easy to service. As a part of this, we worked on a concept where a complete cylinder unit can be exchanged in one single movement. In figure 25 we

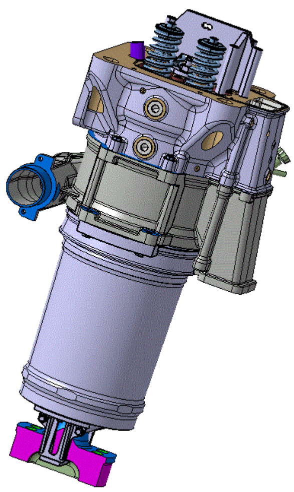

Figure 25 – Cylinder unit

show the unit which can be exchanged. Starting from below, a special clamping tool is used to attach the upper part of the conrod with the liner. With a second tool, the liner is attached to the water jacket, which is in its turn bolted to the cylinder head on which a lifting tool is mounted. With this last tool, the complete unit can be exchanged relatively fast by a new unit.

Outer dimensions

As we have learned from our customers, the outside dimensions of an engine are of the utmost importance as to minimize the engine room so that the cargo room can be maximized. In this way the engine length is critical and has been closely followed during the design phase of this engine.

Starting from the flywheel, we have first a room in the engine block where the distribution gearwheels are integrated and then we have the different cylinders, as is to expect in a normal engine design. With a 100% power take off at the front end side of the engine, we wanted to design the engine in this way that a very short connection was possible between the crankshaft end and the elastic coupling. To obtain this, we have chosen to integrate the pumps into the front end box, rather than attaching them to the box. This has several advantages: the accessibility of the torsion vibration damper is easier, the engine is more compact and both the pumps and the piping towards the pumps is integrated and gives an excellent look. This complete design makes the 6DL36 engine so compact that it is about 0.5m shorter than other engines with the same power output.

Figure 26 – The prototype 6DL36 engine.

CONCLUSIONS

With the new D36 engine range, ABC extends its power range so that it can meet the customer demand for a higher power output. The engine has been designed to meet the future IMO III emission limits with state of the art technology and a unique “Eureka” charging configuration with an integrated EGR design. The results of long term investigations into the injection and combustion process have been implemented so that the emission reduction is being executed in the heart of the engine: the combustion process itself.

The complete mechanical design has been checked and evaluated by the latest techniques, but also the sound castability has been evaluated by these methods.

Although high tech systems needed to be integrated on the engine, a simple design with easy maintenance has been kept as design target. We have always seen the engine as one compact part, with the emphasis on the short length of the engine, without the need to install additional emission after-treatment system next to the engine.

The D36 engine range is now in production and the first prototype 6DL36 engine and can be seen on the figure 26.

NOMENCLATURE

bmep	brake mean effective pressure
CRISTAL	Clean Reliable Innovative Sustainable Tow-stage Alternative Line
ECA	Emission Controlled Areas
EGR	Exhaust Gas Recirculation
GUCCI	Ghent University Combustion Chamber I
HCF	High Cycle Fatigue
HFO	Heavy Fuel Oil
IFO	Intermediate Fuel Oil
IMO	International Maritime Organization
LCF	Low Cycle Fatigue
MDO	Marine Diesel Oil
PTO	Power Take Off

SOLAS Safety of Live at Sea

VVT Variable Valve Timing

Lambda air to fuel equivalence ratio or air excess ratio

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