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Cost-efficient piston ring design for WinGD engines achieved with advanced development methods

Tribology

Markus Weber, WinGD Ltd.

Samuel Affolter, WinGD Ltd.
Richard Mittler, Federal-Mogul Burscheid GmbH
Thomas Stürm, WinGD Ltd.

ABSTRACT

The reliability of marine propulsion engines is key for safe cargo ship operation. Thus, the system of piston rings, cylinder liner and its lubrication must be developed with great care while keeping competitive total cost of ownership in focus.

Piston rings seal the combustion chamber to exploit the outmost of power from the highly pressurized, high-temperature combustion chamber charge. On top of the exposure to high mechanical stress and heat, piston rings undergo extreme accelerations while running with considerable speed on a micrometre-thin oil film.

Gas-tight top piston rings have been introduced as a standard design for piston ring packages of WinGD two-stroke engines in 2008 with great success. This design yields entirely deposit-free ring lands compared to piston ring packages solely based on straight-cut type piston rings. However, to achieve such performance, rigidity and flexibility of gas-tight top piston rings must be balanced carefully, and such piston rings must be manufactured with tolerances of a few hundredths of a millimetre in the area of the ring lock and a running surface roughness in the below-micrometre range.

With steadily increasing specific engine power output and the introduction of X-DF engines having a premixed combustion system, undue thermal deformation of top piston rings was observed under extreme operating conditions. To prepare for even higher requirements of future engine designs, the below mentioned enhancements were introduced, following extensive investigations and using methods like finite element analysis and simulation of piston ring pack dynamics:

- An equalized circumferential contact pressure pattern between piston ring and cylinder liner was achieved by adaptation of piston ring dimensions in general and specifically in the area of the ring lock.
- Reducing the amount of piston rings from three to two yielded a stable pressure distribution in the piston ring package below the top ring during the entire engine cycle as pressure equalization flows between the ring lands are avoided.

The above-mentioned design changes were assessed in the meantime extensively under ship operation conditions. The results prove these optimisations to be effective lowering total cost of ownership by reduced amount of piston rings while keeping component lifetime at the required level and improving system reliability thanks to the achieved simplification on the system and component level.

1 INTRODUCTION

The reliability of marine propulsion engines is key for safe cargo ship operation. Thus, the system of piston, piston rings, cylinder liner and its lubrication must be developed with great care while keeping competitive total cost of ownership in focus.

Piston rings seal the combustion chamber from the piston underside space to make best use of the highly pressurised, high-temperature combustion chamber charge. Beside the exposure to significant mechanical stress and heat, piston rings undergo extreme accelerations while running with considerable speed on a micrometre-thin oil film.

While piston rings are cooled by heat flows to the piston head and through the lubrication oil film to the cylinder liner, their design includes robustness against thermal and mechanical stresses as well as control of gas flow within the piston ring package. Their running surface is an optimised tribological partner to the cylinder liner running surface and the cylinder lubrication oil. Furthermore, piston rings are controlling the stroke-wise distribution of cylinder lubrication oil on the cylinder liner and the mixing of remaining and newly applied lubrication oil. In summary, piston rings, cylinder liner running surface, cylinder lubricant and lubrication system form a tribological system that requires adequate attention for its layout and operation.

Friction induced by the relative motion between the moving piston and the stationary cylinder liner is in case of hydrodynamic state largely determined by the temperature-dependant viscosity of the cylinder lubricant, contact pressure between piston ring and cylinder liner as well as piston speed [1]. Thus, piston ring design matches requirements for piston ring mechanical and thermal stability and the layout of the above-mentioned tribological system.

Piston ring package design, on the other hand, incorporates the layout of a multi-stage flow limiter to minimize losses of the gas from the combustion chamber. Further influencing factors for piston ring package design are deposit control for residues originating from combustion and cylinder lubricant as well as wear control for piston rings, cylinder liner running surface and the piston ring grooves in the piston head. Finally, redundancy for the rare case of piston ring breakage is considered.

For a profound understanding of the function of each component, all components of the tribological system of a modern WinGD engine are presented in chapter 2. In this paper, however, optimisation of piston ring and piston ring package design is in focus without its potential implications on the other components of the entire tribological system.

2 TRIBOLOGY SYSTEM DESIGN OF PISTON RING, CYLINDER LINER AND CYLINDER LUBRICATION SYSTEM

The setup of marine two-stroke diesel engines for direct drive of a propeller is based on a crosshead that is leading the piston rod centrally in the cylinder liner and linking it to the connecting rod, thus transferring the longitudinally acting piston force to the rotating crankshaft. It is worth noting that preceding steam engines of similar cranktrain design were featuring a cylinder lubrication system that dosed the lubricant via quills directly onto the cylinder liner running surface. Contamination of the oil film by steam and condensed water had led to the development of a loss-lubrication principle. A similar concept was later adopted on internal combustion engines to cater for the risk of contamination of the oil film by soot originating from combustion as well as lubricant degradation through thermal stress and oxidation.

Such loss-lubrication system proved to be an even more appropriate solution, when heavy fuel oil with significant sulphur content came into use for such marine engines. Some sulphur contained in the fuel will form sulphuric acid in the combustion chamber, which depletes the corrosion-inhibiting additives of the cylinder lubricant. Sufficient concentration of such additives is thus maintained in the lubricant film by the regular feed of fresh cylinder lubrication oil and by the removal of depleted one.

The mixing of fresh cylinder lubrication oil into the film on the cylinder liner surface and the removal of depleted and contaminated lubricant is one function of the piston ring package. The main function, however, is to form a multi-stage flow limiter to minimize gas loss from combustion chamber to piston underside space (PUS), which is influencing engine efficiency negatively.

2.1 Piston ring gap type

In modern WinGD engines, piston rings with two types of ring gaps are applied as shown in Figure 1 and Figure 2.

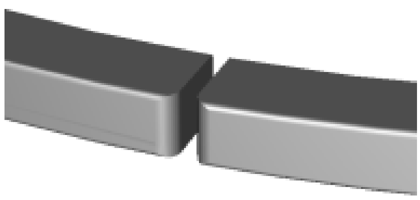

Figure 1: Straight-cut (SC) ring gap.

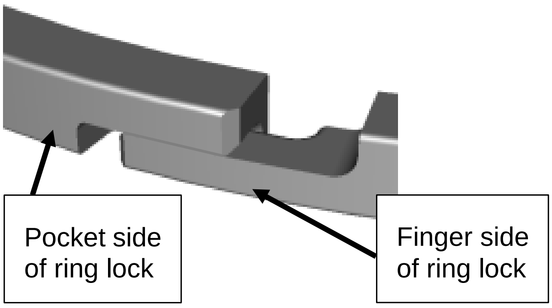

Figure 2: Gas-tight (GT) ring gap.

Gas-tight (GT) piston rings feature a significantly smaller orifice for gas flow than straight-cut (SC) piston rings, because the ends of the ring lock are overlapping each other with only small clearances in between. Therefore, a piston ring package with a GT piston ring releases significantly less gas from the combustion chamber to the PUS compared with a piston ring package containing solely SC piston rings. A further benefit of the GT gap type is the avoidance of transfer of solid residues from the combustion chamber to the ring lands. This prevents piston rings from being blocked in their groove by accumulations of such residues. To achieve this function, the clearance in between pocket and finger part of the GT ring lock needs to be controlled in a narrow range. This requires more complex manufacturing compared to a SC piston ring with an accordingly higher piston ring price.

2.2 Piston ring running surface

While early piston ring designs had a flat running surface in new condition, measurements of running surface profiles of piston rings with many operating hours indicated that such rings get rounded by wear in engine operation. Consequentially, to reduce wear and prolong component lifetime, the piston ring running surface is nowadays shaped accordingly, which is implemented by the radius shown in Figure 3.

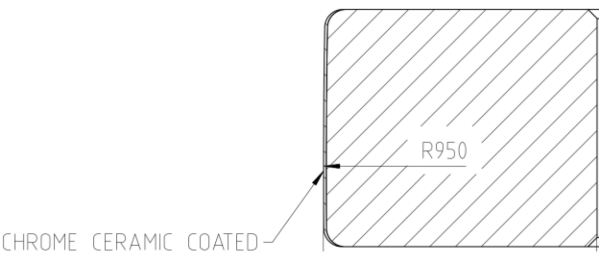

Figure 3: Cross-section of a modern top piston ring for a 920 mm bore WinGD engine with Chrome-Ceramic running surface coating and cylindrically shaped running surface for minimal wear during the running-in period.

Initial piston ring designs did not feature coatings of their running surface. Later and due to increasing power density of engines, so-called running-in coatings were applied to avoid adhesive wear during the running-in period of a new piston ring. The running surface of piston rings of modern WinGD engines is coated with a Chrome-Ceramic (CC) layer as shown in Figure 3. The thickness of the CC coating is adapted for each ring in a given piston ring package to match the lifetime of the piston ring grooves in a piston head.

2.3 Cylinder liner material and running surface

Both piston rings and cylinder liner are components made of cast iron for reasons of manufacturability and cost. The combination of CC-coated piston ring and grey cast cylinder liner running surface is quite ideal tribology-wise as running partners with dissimilar materials are avoiding adhesive wear under high contact pressure. This is the case for the very stable and in case of a scratch fast recovering Chromium oxide layer of the piston ring coating and the iron based grains of the grey cast iron as the atomic bond structure of the two materials does not match for a stable compound [1]. Ground CC coating has a smooth (e.g. $Ra<0.8$ μm) and hard surface (e.g. HV 0.1>800) and the grey cast running surface of the cylinder liner has much lower hardness (e.g. HBS10/3000/30=200). Therefore, a finish of the cylinder liner running surface with low roughness (e.g. $Ra<1$ μm) is preferable to avoid a long running-in procedure with considerable material loss [1]. WinGD cylinder liners feature a criss-cross pattern honed running surface, which is state of the art for automotive and heavy-duty applications since decades.

2.4 Anti-polishing ring

The anti-polishing ring (APR), located in the cylinder liner near top dead centre (TDC) of the top piston ring scrapes away undesirable deposits from the piston top land. The inner diameter of the APR is chosen such that deposits building up on the top land of the piston head cannot interact with the cylinder liner running surface, thus avoiding cylinder lubricant being swept away from the cylinder liner running surface by such deposits.

2.5 Cylinder lubrication system

In parallel to the development of piston ring packages from 2005 as explained in chapter 3.1, the “Pulse Jet” cylinder lubrication system was developed and tested thoroughly, before its introduction in 2012 for all new engine developments and engine updates [2].

In comparison with prior cylinder lubrication systems, “Pulse Jet” aims at a homogeneous circumferential oil distribution, which is based on the finding that piston rings distribute the lubricant well in stroke-wise direction, but less efficiently in circumferential direction [3]. This target is achieved by the tangential injection of the lubricant out of an engine-bore-dependent number of quills into the similar-directed swirl of the cylinder charge by feeding the pump-pressurised cylinder lubrication oil into distribution nozzles as shown in Figure 4.

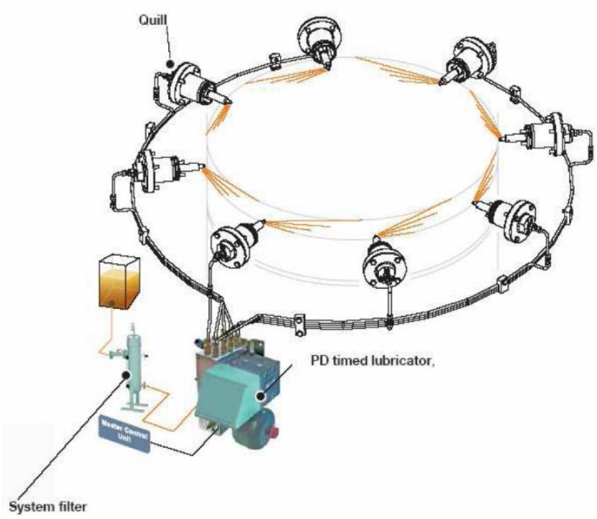

Figure 4: Overview of WinGD Pulse Jet cylinder lubrication system for one cylinder unit.

Each distribution nozzles of the “Pulse Jet” cylinder lubrication system features an engine-bore-dependent number of drillings of variable angle and diameter to achieve a homogeneous lubricant distribution onto the area in between two quills as indicated in Figure 5. Angle and diameter of each drilling are optimised by an accordingly developed software [3].

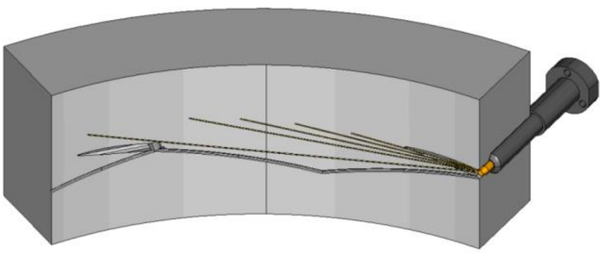

Figure 5: Detail of lubricant distribution of one quill of the Pulse Jet cylinder lubrication system.

3 PISTON RING PACKAGE DEVELOPMENT APPROACH

The design of the piston ring package is a process that is driven by engine design aspects like power density and cost as well as external influences like engine operation patterns, legislation as well as availability of materials and suitable manufacturing processes. Before WinGD’s piston ring package development approach is demonstrated in detail, the development steps of the past three decades are summarised in the following chapter.

3.1 Chronology of serially applied piston ring package designs

Before the introduction of a four-piston-ring-package design in 2002, Sulzer marine two-stroke diesel engine pistons carried five, usually identical, uncoated SC piston rings to seal the combustion chamber from the PUS. Such multi-piston-ring arrangements function as flow limiter with an asymptotic decrease in pressure drop over each piston ring starting from the top ring towards the PUS. Later, thermally sprayed coatings of various kinds were applied on the piston ring running surface, either to improve the performance of running-in at the engine builder or to prolong the lifetime of the component in operation.

Over time, piston ring package design of WinGD marine two-stroke engine developed as follows (year of introduction; engine type; piston ring package design; development target):

- 1996; RTA96C; 5 SC rings with running-in plasma coating.
- 2002; RT-flex96C-B: 4 SC rings, whereas the top piston ring was equipped with lifetime CC coating and 3 SC rings were equipped with running-in plasma coating; cost reduction and piston ring lifetime prolongation
- 2005; RT-flex96C-B: 4 SC rings, all rings with lifetime CC coating; cylinder liner lifetime prolongation
- 2008; RT-flex82C and RT-flex82T: GT top ring and two SC lower rings, all rings with lifetime CC coating; deposit reduction in piston ring package and piston head lifetime prolongation
- 2018-2019 all 520, 620, 720 mm bore engines: GT top ring and one SC lower ring, all rings with lifetime CC coating; reduction of total cost of ownership.

To demonstrate WinGD’s current development approach for piston ring packages, the following paragraphs highlight the major steps in development and testing, before a new piston ring package design is released for serial application.

3.2 Piston ring package release process

WinGD’s process for the definition of a new piston ring package design before it will be introduced on serially produced engines involves the following steps:

- Optimisation potential of the current reference piston ring package is identified
- Potential of tribological or economical benefit is given for the new design in comparison with the reference case
- Dynamic piston ring simulation results for the new design are found without abnormalities and with expected improvements, for current and future levels of maximum firing pressure and mean effective pressure
- Successful 200-hour-trial of the new design on a test engine
- Successful 18'000-hour-test on several units in harsh vessel operation for the target bore size and possibly other suitable bore sizes

In the following paragraphs, the detailed steps for the development of a new design, in this case of a two-ring-package based on the reference of a three-ring-package are demonstrated.

3.3 New piston ring package design layout

For the layout of a piston ring package, a draft of the desired design and data for all interfaces of the piston rings to other combustion chamber components is required, as follows:

- Target piston ring package design, i.e. number and type of piston rings for the new package
- Initial piston ring main dimensions
- Initial GT top piston ring gap dimensions
- Initial piston ring running surface shape
- Wear rates of applied piston ring coating
- Wear rate of piston ring lower flank
- Wear rate of piston ring groove's lower flank
- Maximum allowed piston ring material temperature
- Cylinder liner running surface temperature (average and maximum) for rated engine power and typical operating engine power
- Piston ring groove temperature for rated engine power and typical operating engine power

The draft layout of a new piston ring package design is based on data of existing designs for which many years of operation experience have been accumulated. Using the above-mentioned information, a model for dynamic piston ring simulation is derived and used for subsequent parameter optimization and thorough sensitivity analysis.

3.4 Dynamic piston ring simulation tool PRiME3D®

Since more than a decade, WinGD is utilising Tenneco's PRiME3D® for the investigation of the behaviour of existing and newly designed piston ring packages. The simulation tool is applicable for all types of internal combustion engines and incorporates years of intensive research to transform the physical boundary conditions of engine kinematics and ring design [4]. From 2008 until today, the physical methodology was expanded to the conditions prevailing in large two-stroke marine propulsion engines and validated on one of WinGD's test engines within the HERCULES-B project of the European Union [5].

To reach such an understanding, it was necessary to solve the physics for compressible fluid flow in the clearances of the piston and closed gap of the piston ring under the conditions of a subsonic gas flow. The calculation of gas flow and its pressure gradients is fundamental for predicting the behaviour of piston rings in an internal combustion engine. As a result, the influence of piston ring design on the tribological system of piston ring, cylinder liner and the physical (not chemical) properties of the cylinder lubricant film is understood by the output of the tool, and thus, engine components can be optimised accordingly.

In a first step, the variation of elastic energies characterised by differential equations in conjunction with the boundary and equilibrium conditions in large two-stroke marine propulsion engines was considered. Then, additional
for state-of-the-art designs like GT top piston rings and temperature-optimised piston ring shapes was integrated.

Thus, PRiME3D® yields thorough understanding of the energy model and system behaviour of piston ring dynamics in large two-stroke marine propulsion engines. Design and functionality of a two-stroke piston ring are strongly influenced by piston speed and combustion chamber pressure as well as cylinder lubrication oil feed. With PRiME3D® it is possible to understand the fundamentals of gas flow, oil transport and ring behaviour and their effects on combustion gas loss and oil distribution without the need of expensive engine tests.

The simulation of piston ring dynamics consists of three major elements, which influence each other: gas flow, hydrodynamics and ring motion. Small variations regarding a single component (e.g., manufacturing tolerances) can change the system behaviour significantly. Therefore, sensitivity analysis is crucial for the applicability of the simulation results to real engine operation.

PRiME3D® as tool for engine developers provides technical core values for piston ring tribology and further information that is used for component optimization within the WinGD piston ring package design process as follows:

- Axial motion and twist of all piston rings in their grooves
- Gas pressure and massflow in the piston ring package
- Oil film thickness below the piston rings.

3.5 Reference three-piston-ring-package

The three-piston-ring-package was introduced as standard design for serial application from 2008 on, starting with engines of 820 mm bore size. This decision was taken after extensive field tests were carried out on RTA96C and RT-flex96C-B engines.

Figure 6: Piston head with three piston rings of a WinGD 9X82 engine after 55'001 operating hours on heavy fuel oil without overhaul (by 01.09.2024).

The piston shown in Figure 6 exceeds the expected lifetime of 36'000 operating hours by far, however, actual piston head and piston ring lifetime may vary depending on vessel-specific operating conditions and fuel type in use.

Deposit build-up is visible on the topland due to combustion residues originating from heavy fuel oil operation. The thickness of these deposits is controlled by an APR to avoid bore polishing. Traces of the scraping action of the APR can be recognized from the vertical stripes on the black deposit just above the top piston ring (see Figure 6). Despite ample availability of combustion residues on the piston top land, no deposit build-up exists in the lower ring lands thanks to the sealing effect of the GT top ring.

The strict control of deposit build-up on ring lands and in piston ring grooves is the pre-requisite for the free movement of all piston rings in axial, radial and tangential direction for the entire lifetime of the piston head. Free movement of piston rings is required to avoid hard contact between piston rings and cylinder liner as the clearance between piston skirt and cylinder liner allows the piston to move freely in all directions.

The two lower piston rings function as an emergency gas sealing in case that the top ring would break or lose tension in engine operation.

Since the introduction of the three-piston-ring-package in 2008 as indicated in paragraph 3.1, 1505 engines with 9608 three-ring-pistons similar to the one shown in Figure 6 have accumulated approximately 62 million operating hours without major difficulties.

3.6 Results of dynamic piston ring simulation for three-piston-ring-package

Results of dynamic piston ring simulation for a three-piston-ring-package are shown in Figure 7.

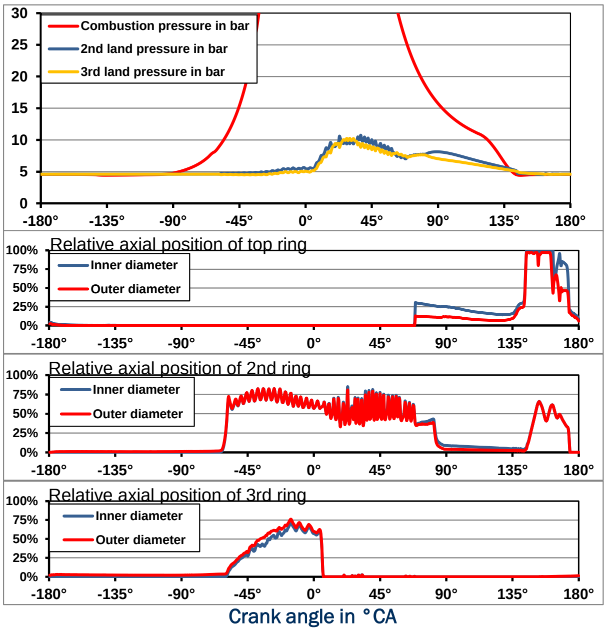

Figure 7: PRiME3D® simulation results for a three-piston-ring-package regarding gas pressure as well as piston ring axial motion and twist.

Evaluation of the data presented in Figure 7 allows drawing the following conclusion:

- small differential pressure over the second ring (uppermost graph showing combustion and second and third ring land pressure) leads to an undefined position of the second ring in its groove from 60 °CA before TDC and 80 °CA after TDC (two graphs in the middle marked with “2nd Ring at gap” and “2nd Ring at back”)

The above findings demonstrate optimisation potential in economical and technical aspects for the three-piston-ring-package as follows:

- Simulation results indicate that the pressure distribution within the three-ring package with a GT top ring leads to the situation that the second ring has no defined position on the upper or lower flank of its ring groove, but “floats” in between these two positions for nearly half the engine cycle due to small and fluctuating differential pressure over the second ring according Figure 7.

- A piston with two piston rings offers lower first cost for engine production in comparison with a three-piston-ring-package. Cost for engine overhaul is reduced as less spare piston rings need to be purchased and less piston ring grooves need to be refurbished, once the piston head has reached the end of its lifetime because of piston ring groove wear.

3.7 Development of two-piston-ring-package for WinGD engines

Results of dynamic piston ring simulation for a two-piston-ring-package are shown in Figure 8.

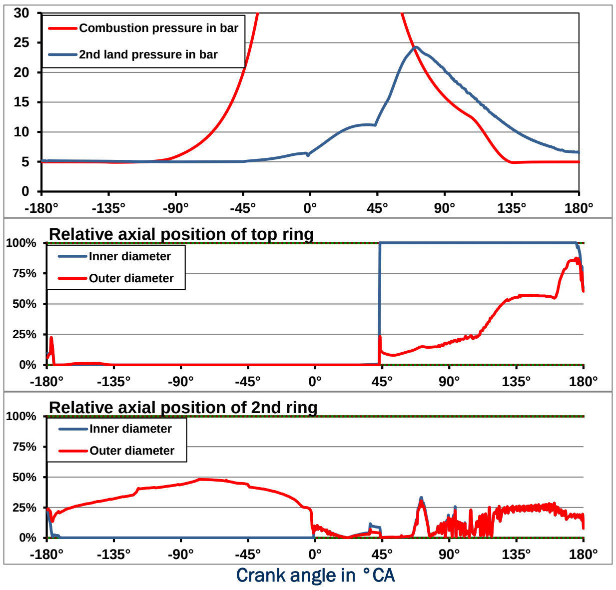

Figure 8: PRiME3D® simulation results for a two-piston-ring-package regarding gas pressure as well as piston ring axial motion and twist.

Evaluation of the data presented in Figure 8 indicates the following:

Top piston ring (please refer to the two graphs marked with “top ring” of Figure 8):

- [-180..-90] °CA: during the later scavenging phase (after BDC) of the engine cycle, the top piston ring is forced to the bottom flank of its groove by the acceleration force induced by the piston movement

- [-90..45] °CA: during compression and early expansion phase of the engine cycle, the top piston ring is forced to the bottom flank of its groove by the pressure force induced by the combustion chamber pressure despite the counteracting acceleration force induced by the piston movement

- 45 °CA: during the early expansion phase of the engine cycle, pressure builds up in the second ring land due to the high combustion chamber pressure and the non-negligible gas loss through the top ring lock in this part of the cycle. As, however, combustion chamber pressure decreases rapidly due to the on-going expansion, the continuously reducing differential pressure over the top ring and the still acting acceleration force induced by the piston movement lead to a distinct change of the contact line between the top ring and its groove from the bottom flank to the top flank at the inner diameter (green line, ID=inner diameter)

- [45..180] °CA: the negative differential pressure over the top ring and the counteracting acceleration force induced by the piston movement lead to the change of the contact line from the bottom flank to the top flank at the outer diameter of the top ring. This process takes the remainder of the expansion cycle, which means that the top piston ring is moving very smoothly from the bottom flank to the top flank without sudden impact (red line, OD=outer diameter). Such smooth motion of the top ring in its groove is a key enabler for extending the lifetime of both piston ring and piston head

- during the early scavenging phase of the engine cycle near BDC, the top ring returns to the bottom flank of its groove. Here, both pressure and acceleration forces are low, resulting in a smooth movement without major impact on piston ring and the groove flanks with positive impact on the lifetime of piston ring and piston head

Lower piston ring (please refer to the two graphs marked with “2nd ring” of Figure 8):

- [-180..0] °CA: during the second part of the scavenging phase (after BDC) and the compression phase of the engine cycle, the lower piston ring undergoes a twist, which is due to the asymmetrical barrel shape of its running surface and the low pressure-force acting on it

- [-45..0] °CA: during the second part of the compression phase, the acceleration force induced by the piston movement exceeds the gradually rising pressure force acting on the lower ring, thus forcing it to change the contact line from the bottom flank to the top flank of its groove. The ring twist, however, continues to exist until TDC, as no adequate force or moment exists to press the ring with its upper flank against the top flank of its groove

- [0..110] °CA: during the expansion phase of the engine cycle, the rising differential pressure over the ring forces the lower ring to change the contact line from the top flank to the bottom flank of its groove, where it stays, until the exhaust valve opens and the pressure in the combustion chamber reduces down to the pressure in the exhaust gas receiver of the engine. The differential pressure over the lower ring during this flank shift is low, thus ensuring a smooth transition of the ring. Such smooth motion of the top ring in its groove has the above-mentioned positive effect on the lifetime of piston ring and piston head

- [110..180] °CA: in this part of the scavenging phase of the engine cycle until BDC, the second ring land pressure is exceeding the combustion chamber pressure due to the continuous expansion in the combustion chamber. Simultaneously, gas is flowing from the second ring land to the PUS, resulting in a not clearly defined pressure difference over the lower ring. According to the simulation, the motion of the lower ring in this phase is mainly controlled by the acceleration force induced by the piston movement and the friction force acting on the running surface of the ring. This yields a smooth lift-off of the ring from the lower ring groove flank. The low differential pressure and the other small forces and moments acting on the lower ring ensure a smooth movement of the ring in its groove with the according positive impact on the lifetime of piston ring and piston head.

Key findings of the above-mentioned simulation results are:

- The lower ring experiences a significant pressure difference between top and bottom flank after TDC. As this force competes with the acceleration force due to the piston movement, there is movement of the lower ring in its groove, however, no sudden flank change takes place, which would cause additional wear of the ring groove flanks

- In a ring twist situation, the top piston ring allows more gas leakage, because gas will flow not only through the clearances of the overlapping ring lock, but also in between the lower flank of the ring and the groove flank. Such stronger leakage creates a differential pressure over the second ring resulting in the second ring being forced to the bottom flank of its groove. In this situation, top and second ring form a two-stage flow limiter to control the gas leakage from the combustion chamber to the PUS. The same effect applies, if the top piston ring would break or loose tension

- A two-piston-ring-package with a GT top ring and a SC lower ring seals the combustion chamber reliably over the entire engine cycle and with suitable redundancy for the case of a top ring failure with the lowest possible effort of piston ring material and piston ring groove machining. Furthermore, cost for piston overhaul is minimal and therefore, this arrangement is the most sustainable solution for combustion chamber sealing at present.

3.8 Reliability of two-piston-ring-packages in operation

Meanwhile, the above described two-piston-ring-package was introduced as a serial solution for the engine types X52, X52DF, X62-B, X62-S2.0 and X72-B. Some of the first WinGD engines equipped with two-piston-ring-package are now in operation for more than five years.

Figure 9 shows the condition of a two-ring-piston after more than four years of operation on natural gas without overhaul. Like the results of the three-piston-ring-package, no deposit build-up exists in the lower ring lands thanks to the sealing effect of the of the GT top piston ring.

In the meantime, totally 117 installations with 727 two-ring pistons like the one shown in Figure 9 have accumulated over two million operating hours without major difficulties. Therefore, this ring-package design as well as the approach used in its development can be considered proven.

Figure 9: Part of piston head with two piston rings of a WinGD 7X52DF engine after 22’700 operating hours seen through inlet port (by 19.11.2024)

As indicated above, the lower piston ring assists the top ring in gas sealing in regular operation, but also in case, if the top ring would break or lose tension. Such fatal top ring damages are very rare in operation. Nevertheless, the related WinGD guideline recommends switching off fuel injection of the affected unit until the next opportunity for a piston ring exchange, e.g. the next port stay of the vessel is at hand.

3.9 Improvement potential for dynamic piston-ring-package simulation

As shown above, dynamic piston-ring-package simulation has proven to be a valuable tool in the course of engine component development and design. Refinement of the established simulation models will improve predictive quality and thus reliability of future piston-ring-package designs. Such improvement requires a higher degree of validation, which is foreseen by measuring specific data in test setups, such as:

- Pressure and temperature of piston top land and ring lands to calculate gas flow through the piston-ring-package and heat transfer in between the piston head, piston rings and cylinder liner

- Lube oil film thickness as function of stroke to assure the proper tribological function of piston rings and cylinder liner

- Axial, radial and tangential piston ring motion to quantify the influence of gas, friction and acceleration forces acting on the piston rings.

4 PISTON RING DEVELOPMENT APPROACH

Large marine propulsion engines encounter variable operating conditions within their lifespan:

- Shop test (duration: about 20 hours): running-in procedure with new components, challenging load-up cycles and overload operation during performance demonstration of the engine

- Sea trial (duration: some days): highly variable engine load for demonstration of safe engine operation under extraordinary situations, e.g., vessel speed and endurance trials, crash-stop manoeuvre and operation with units cut off

- Vessel operation (duration depending on application: 20-40 years): prolonged operation with varying loads under constantly changing ambient conditions, variable trim and heeling situations, use of fuels of changing quality with related influence on firing pressure and deposit formation in the combustion chamber

Engine design must incorporate proper robustness to be able to cope with the above-mentioned and further operating condition set-ups, which requires appropriate testing before the serial release of an engine type.

In 2019 during the prototype testing phase of the development of the X92DF (Dual-Fuel) engine, top piston rings were repeatedly damaged under certain operating conditions. Root-cause-analysis concluded that the ring lock area suffered from excessive contact pressure between piston ring and cylinder liner as shown in Figure 10. The related disruption of the lubricant film on the liner running surface led to scuffing between piston ring coating and cylinder liner material, which destroyed both running surfaces.

This result triggered the optimization of this top ring design that had a satisfactory track record on several hundred units in the X92 (diesel) engine since its introduction in 2015. Details of this optimization process are explained below.

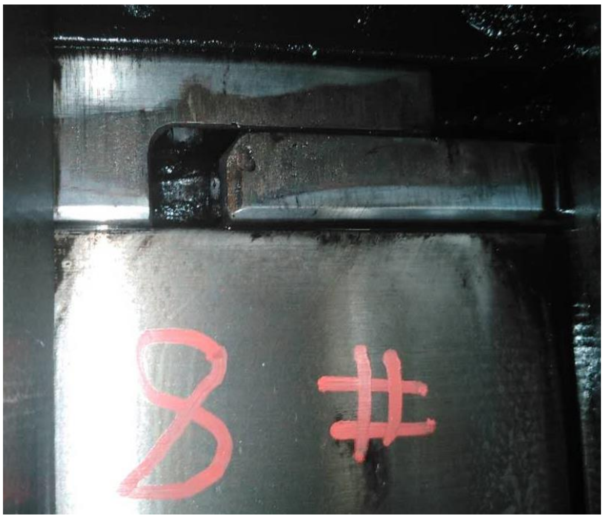

Figure 10: Damaged gas-tight top piston ring lock after workshop operation

4.1 Details of root-cause-analysis

The investigation of engine performance data and engine operation profile during prototype testing yielded the following results, after the above-mentioned damage was detected repeatedly:

- The damage was detected in all cases after operation in gas mode (premixed combustion)
- The damage appeared only above a certain engine load threshold
- The damage started from the top piston ring
- The extent of damage varied strongly from unit to unit of the affected engine.

The investigation of the damaged top piston rings yielded furthermore the following results:

- The extent of damage was most significant in the ring lock area, whereas the rest of the ring circumference was generally not affected
- The contact pattern on the running surface near the ring lock indicated a local, radial deformation of the ring base material with mechanical or thermal origin. The remaining circumference of the piston ring did not show unusual contact pattern
- A test with a pin installed in tangential direction on the pocket side of the ring lock confirmed that the ring lock was not closing entirely in operation. WinGD GT top piston rings are laid out for safe operation below the temperature limit of disintegration of the cylinder lubricant. An entirely closed ring lock would have indicated that the piston ring base material had reached this limit, which was not the case. Thus, the reason for the detected deformation at the ring lock was not an overheating of the entire top ring base material, but a local effect in the ring lock area

As the cycle-to-cycle variation in gas operation of X-DF engines is larger than in diesel operation, the variable appearance of the damage from unit to unit and the load-dependence of the damage in gas operation was well explained. The top ring was mainly affected, because it is directly in contact with the combustion gas and because it bears 90-95% of the pressure difference between combustion chamber and the PUS due to its design (refer to chapter 2.1).

It was, however, not evident from the root-cause-analysis results, if the damage was caused by mainly mechanical or mainly thermal deformation. Therefore, both enhanced robustness against mechanical and thermal deformation was targeted for the improved design of top piston ring of X92DF.

4.2 Ring lock design optimization

4.2.1 Mechanical robustness

The cross-section of the original piston ring in the ring lock area is shown in Figure 11:

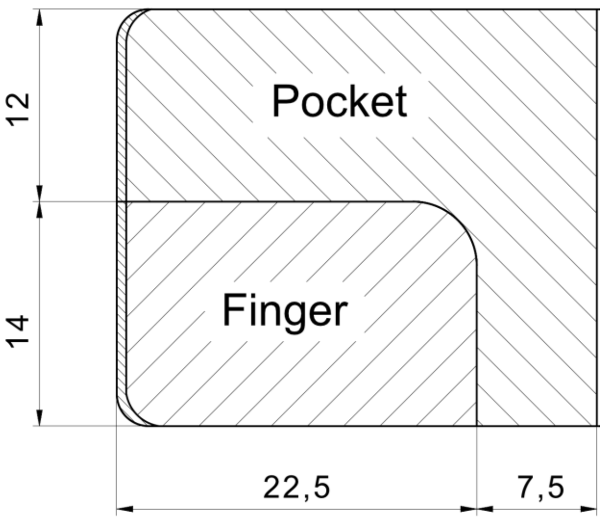

Figure 11: Cross-section of the original piston ring design in the ring lock area and its main dimensions

To improve mechanical robustness of the given design, the following measures were applied:

- The height of the “Finger” part of the ring lock was enlarged to achieve a nearly quadratic cross-section to optimize stiffness in axial and radial direction and thus to reduce the radial bending by potentially acting thermal stresses

- By the above, the “Pocket” part of the ring lock changed shape towards a symmetrical L-profile, which features optimal stiffness in both radial and axial direction

- The above measure increases the vertical area of the “Pocket” part along the separation line towards the “Finger” part. Such enlarged area increases the counterforce against the combustion pressure acting on the backside of the “Pocket” part. For this purpose we assume that there is a clearance between “Pocket” and “Finger” part for combustion gas to enter between the two parts. The width of this clearance is depending on the manufacturing of the piston ring and may vary within 0.05-0.15 mm.

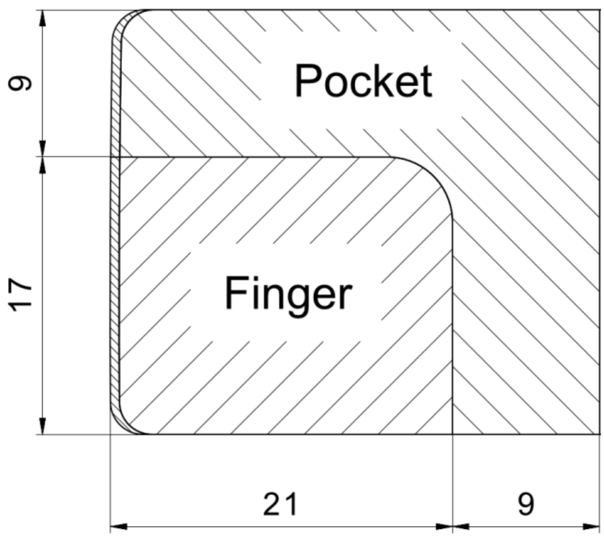

Figure 12: Cross-section of the optimised piston ring design in the ring lock area and its main dimensions

4.2.2 Thermal robustness

To optimize thermal robustness, the heat transfer from combustion gas to both “Pocket” and “Finger” parts needs to be minimised, which was achieved by the following measures:

- The clearance between “Pocket” and “Finger” part was reduced from [0.05..0.15] to [0..0.1] to reduce the massflow of combustion gas that is potentially heating the ring lock area

- The deburring of the edges of the coating at the radial end of the clearance between “Pocket” and “Finger” part towards the running surface was restricted to be carried out with minimal removal of material. This measure also yields a reduction of the massflow of combustion gas that is potentially heating the ring lock area.

4.2.3 Finite Element modelling of original and optimised ring lock design

Before prototype piston rings were manufactured for testing, both the original and the adapted design were modelled using the Finite Element (FE) method. The 3D-models of both piston ring designs including a section of the corresponding cylinder liner in full scale featured 370’000 elements and took into account the below boundary conditions, resembling a situation of the piston ring in the engine cycle, shortly after having passed TDC under maximum firing pressure:

- 200 bar combustion chamber pressure

- 100 bar pressure in the clearance between Pocket and Finger part of the piston ring

- Zero bar pressure below piston ring

- Uniformly distributed radial contact pressure between piston ring and cylinder liner running surface (emulation of piston ring pre-tension)

- 150 °C piston ring running surface temperature

- 200 °C piston ring inner diameter surface temperature

- 150 °C cylinder liner running surface temperature

The contact between piston ring and cylinder liner as well as the contact against a flat plate located in axial direction below the piston ring is assumed to be frictionless. This simplification is justified for a static model, where the relative velocity of the components towards each other is irrelevant. Based on the above, the piston ring models are free to deform like a piston ring located in its groove of the piston head.

As the FE simulation of the optimised piston ring design yielded significantly reduced contact pressure in the relevant areas of its running surface compared to the original design, prototype piston rings were manufactured to assess the performance of the optimised design in a test engine. Results are presented in the following paragraph.

4.2.4 Testing of optimised piston ring design and release for serial application

A 12X92DF engine was equipped with piston rings of the optimised design to be run on a regular shop test including overload diesel operation and rated load gas operation. Figure 13 shows the ring lock of the top piston ring of the optimised design after shop test.

The optimised piston ring design performed well on all units in both diesel and gas operation. The overall condition is acceptable and according to the expectation given by the Finite Element model results. The level of contact pressure on both parts of the ring lock is reduced so far that on none of the test units scuffing between piston ring and liner occurred. Based on this result, further thorough testing on production engines and on engines in vessels was carried out to expose the new piston ring design to the full range of potentially arising operation conditions as laid out at the beginning of chapter 4.

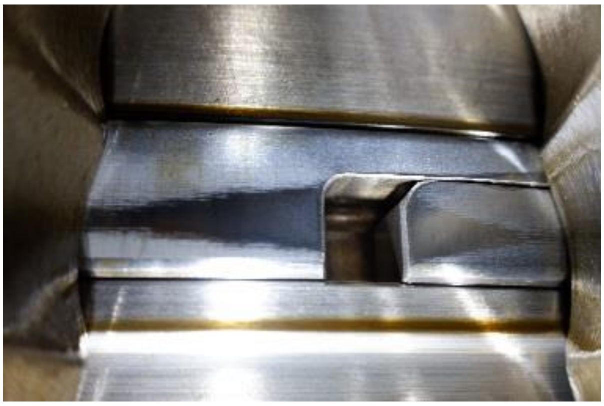

Figure 13: Piston ring lock of the optimised design seen through the inlet port of the according cylinder liner after the shop test of a 12X92DF engine. In the centre of the picture, the pin to measure the extent of ring lock closure during operation is visible.

Based on the highly satisfactory results of this testing campaign, the optimised top piston ring design was assessed as fit for purpose and has been released for serial application on all X92 and X92DF engines by mid 2020. Simultaneously with the development of the new top piston ring designs above, engine tuning parameters of the X92DF engine type were optimised such that the thermal and mechanical load acting on the top ring was reduced, thus contributing further to reliable operation of this engine type in service. Since the delivery of the first vessel with X92DF engine in 2020, more than 300'000 operation hours have been accumulated on 16 engines with 176 units without major difficulties [6].

5 CONCLUSIONS AND OUTLOOK

The methods for the development of both new piston ring and piston ring package designs as presented in chapters 3 and 4 have proven to be successful.

During the development of the two-piston-ring-package as described in chapters 3.7f, optimisation potential has been identified, which is the refinement of capabilities of dynamic piston ring simulation yielding more precise calculation of time-dependent massflow of combustion gas through the ring pack and more detailed information on the heat transfer between the gas in the ring land volumes and both piston head and piston rings.

Piston ring and piston head development is partially a continuous design improvement process and partially triggered by factors like more stringent emission legislation or the availability of renewable fuels. WinGD's current engine portfolio enables ship owners using various fuel types of fossil and renewable nature. Retrofit solutions will empower existing engines to use "Future Fuels" as well. The demand for such engines and retrofit solutions will trigger the development of adapted piston ring designs to cope with the given conditions for these applications.

Furthermore, the continuous increase of mean effective pressure in large marine propulsion engines as well as more stringent fuel efficiency requirements to reduce the CO_2 footprint of marine cargo transport will increase maximum firing pressure and therefore combustion chamber gas temperature of these engines. This will require even more robust piston ring designs than applied today. To achieve such, the development methods described in this paper consisting of a combination of the evaluation of extensive field-testing measurement results, validated simulation methods and "traditional" engineering including engine tests will be applied in a refined approach in the coming years.

6 DEFINITIONS, ACRONYMS, ABBREVIATIONS

APR: Anti-polishing ring

BDC: Bottom dead centre

CC: Chrome-ceramic

GT: Gas-tight

FE: Finite element

PRiME3D®: Dynamic piston ring simulation tool of Tenneco Inc.

PUS: Piston underside space

SC: Straight-cut

TDC: Top dead centre

WinGD: WinGD Ltd.

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9 CONTACT

For inquires regarding this paper, please contact markus.weber@wingd.com