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## Piston ring tribology

A literature survey

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## **Abstract**

The tribological considerations in the contacts formed by the piston skirt, piston rings and cylinder liner have attracted much attention over several decades, not least indicated by the large number of articles published on this topic in recent years. Recent studies include modelling, miniaturised experimental work and full-scale engine testing.

This literature survey, covering over 150 references, aims to shed new light on the tribological issues related to the piston assembly. The work is intended as a compact reference volume for internal combustion engines in general, with particular emphasis on diesel engines.

Central topics discussed in this work are the basic functions of the piston and the piston rings, the design and the materials of the components, mechanical and thermal loads on the rings, the contact pressure between ring and liner, the sealing action, blow-by leakage, hot gas erosion damage, exhaust emissions, the lubrication conditions and the influence of combustion products, the coefficient of friction and the friction force, the wear of the sliding surfaces and surface technology for wear reduction.

# Preface

This literature study is part of the project, Tribology of Internal Combustion Engines (in Finnish: Polttomoottorien tribologia). The project, which is running during the years 1999–2002, is part of the Finnish research programme ProMotor. The work was financially supported by the National Technology Agency Tekes, by the Finnish companies Fortum Oil & Gas Oy, Wärtsilä Corporation and Sisu Diesel Oy, by Volvo Technological Development Corporation in Sweden, and by the Technical Research Centre of Finland (VTT).

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Espoo, Finland, on the 3rd of December, 2002

The Authors

# Contents

Abstract.....	3
Preface .....	4
1. Introduction.....	8
2. Piston rings.....	9
2.1 Main functions of piston rings.....	9
2.2 Ring categories .....	10
2.2.1 Compression rings.....	11
2.2.2 Oil control rings .....	12
2.3 Piston ring materials and coatings.....	13
2.4 Sealing ability of piston rings.....	15
2.4.1 Conformability .....	15
2.4.2 Counter surface effects.....	16
2.4.3 Blow-by prevention.....	16
2.4.4 Oil consumption reduction and emission suppression .....	17
3. Piston skirt and ring groove .....	20
3.1 Piston types and geometry .....	20
3.2 Piston skirt materials .....	22
3.3 Piston ring groove.....	24
4. Cylinder liner .....	26
4.1 Cylinder liner materials .....	26
4.2 Plateau honing .....	26
4.3 Macro form deviation of cylinder liners.....	28
5. Piston ring mechanics .....	32
5.1 Piston ring kinematics and kinetics .....	32
5.2 Piston ring forces and moments .....	36
5.3 Ring contact pressure .....	37
5.3.1 Contact pressure from ring compression.....	37
5.3.2 Contact pressure arising from gas pressure.....	38
5.3.3 Additional loads from the deformation of cylinder bore and ring .....	38
5.3.4 Non-uniform pressure distribution due to ring twist.....	38
6. Thermal loads.....	39
6.1 Macro-scale thermal loads.....	39
6.2 Micro-scale thermal loads .....	40

6.2.1	Frictional heating in unlubricated sliding contacts .....	40
6.2.2	Lubricant heating owing to viscous work .....	42
7.	Piston ring and skirt lubrication .....	44
7.1	Oil supply .....	44
7.1.1	Supply mechanisms and oil quantity.....	44
7.1.2	Oil quality.....	46
7.1.3	Contaminations in the oil .....	46
7.1.4	Coke formation, coke ring.....	49
7.2	Lubrication regimes; load, speed, viscosity and counter surface effects .....	49
7.2.1	Oil film thickness calculations .....	51
7.2.2	Oil film thickness measurements .....	54
8.	Friction of piston rings and skirt against cylinder liner .....	58
8.1	Friction of piston rings against cylinder liner.....	58
8.1.1	Ring friction fundamentals.....	58
8.1.2	Friction simulation .....	60
8.1.3	Measured friction forces and coefficients of friction .....	62
8.1.4	Effect of piston ring surface finishing and coating .....	65
8.1.5	Effect of cylinder liner surface finishing and coating.....	66
8.1.6	Effect of cylinder liner out-of-roundness .....	67
8.2	Friction of piston skirt and piston rings against the cylinder liner.....	67
8.2.1	Normal operation conditions.....	68
8.2.2	Scuffing .....	69
8.2.3	Piston seizure .....	70
9.	Wear in the piston-ring-liner system.....	71
9.1	Wear of piston rings .....	71
9.1.1	Running-in wear of piston rings.....	72
9.1.2	Steady-state wear of piston rings .....	72
9.1.3	Wear of ring flank surfaces .....	72
9.1.4	Wear of oil rings.....	73
9.1.5	Simulation of wear of piston rings .....	73
9.2	Wear in the piston ring groove .....	74
9.3	Wear of the piston skirt .....	75
9.3.1	Mild wear of the piston skirt .....	75
9.3.2	Abrasive wear of the piston skirt .....	76
9.4	Wear of the cylinder liner.....	76
10.	Surface technology.....	78
10.1	Coating deposition and surface treatment techniques .....	78
10.2	Surface roughness.....	85

10.2.1 Surface roughness basics.....	85
10.2.2 Topography and texture .....	86
10.2.3 Textured surfaces .....	87
11. Summary .....	88
References .....	90

# 1. Introduction

There are two entirely diverse points of view that make dynamic seals particularly demanding in a tribological sense. Firstly, dynamic seals support the cleanliness of a lubricant and a tribological element to be protected from external contamination, and thus contributes to suppressing wear by three-body abrasion caused by contaminant particles. Simultaneously the seal suppresses the leakage of lubricant from the tribosystem, which is an issue of increasing relevance for environmental protection and cleanliness. Secondly, the counter surfaces of a dynamic seal operate under the same tribological laws as any sliding couples, though with the requirements of low friction and low wear and a long service life. Seals for linear motion are particularly challenging as their counter surface smoothness, direction of sliding, speed and lubrication tend to vary more than in the closed contact forming a seal for a rotating motion.

A common feature of all dynamic seals for linear motion is that they operate against a reciprocating counter surface, for which reason they have to be optimised in terms of sealing ability, low friction and system wear. Wear at the counter surface, caused by reciprocating contact with the piston or the linear bearing, leads to changes in the surface quality of the counter surface, which in turn change the preconditions for the dynamic seal.

Piston rings for current internal combustion engines have to meet all the requirements of a dynamic seal for linear motion that operates under demanding thermal and chemical conditions. In short, the following requirements for piston rings can be identified:

- Low friction, for supporting a high power efficiency rate
- Low wear of the ring, for ensuring a long operational lifetime
- Low wear of the cylinder liner, for retaining the desired surface texture of the liner
- Emission suppression, by limiting the flow of engine oil to the combustion chamber
- Good sealing capability and low blow-by for supporting the power efficiency rate
- Good resistance against mechano-thermal fatigue, chemical attacks and hot erosion
- Reliable operation and cost effectiveness for a significantly long time

The tribological considerations in the contacts formed by the piston skirt, piston rings and cylinder liner, or bore, have attracted much attention over several decades, not least indicated by the large number of articles published on this topic in recent years. Recent studies include modelling, miniaturised experimental work and full-scale engine testing.

This literature survey aims to shed new light on the tribological issues related to the piston assembly. The work is intended as a compact reference volume for internal combustion engines in general, with particular emphasis on literature related to diesel engines.

## 2. Piston rings

In the early steam engines no piston rings were used. The temperatures and the steam pressures were not as high as the corresponding parameters in today's internal combustion engines, and the need for considering thermal expansions and clearances was smaller. Increasing power demands required higher temperatures, which caused stronger heat expansion of the piston material. This made it necessary to use a sealant between the piston and the cylinder liner to allow a decrease in the clearance in cold conditions, i.e. when the clearances were at their maximum. Keeping the clearance between the piston and liner wall at a minimum considerably reduces the combustion gas flow from the combustion chamber past the piston.

The first piston rings used in an engine had the sole task of sealing off the combustion chamber, thus preventing the combustion gases from trailing down into the crankcase. This development increased the effective pressure on the piston. Ramsbottom and Miller were among the pioneers to investigate the behaviour of the piston rings in steam engines. Ramsbottom, in 1854, constructed a single-piece, metallic piston ring. The free diameter of the ring was 10 per cent larger than the diameter of the cylinder bore. When fitted in a groove in a piston, the ring was pressed against the cylinder bore by its own elasticity. Previous piston rings had consisted of multiple pieces and with springs to provide an adequate sealing force against the cylinder bore. Miller, in 1862, introduced a modification to the Ramsbottom ring. This modification consisted of allowing the steam pressure to act on the backside of the ring, hence providing a higher sealing force. This new solution enabled the use of more flexible rings, which conformed better to the cylinder bore (Priest and Taylor, 2000).

In the early days, the ring pack was lubricated solely by splash lubrication; i.e. lubrication by the splashing of the rotating crankshaft into the crankcase oil surface. Subsequently, when the combustion conditions became even more demanding, i.e. with higher temperatures, pressures and piston speeds, oil control rings were introduced. A proper lubricant film on the piston, piston rings and liner wall was required in order to prevent damage. The oil control rings were, and are, especially designed to appropriately distribute the oil on the cylinder liner and to scrape off surplus oil to be returned to the crankcase.

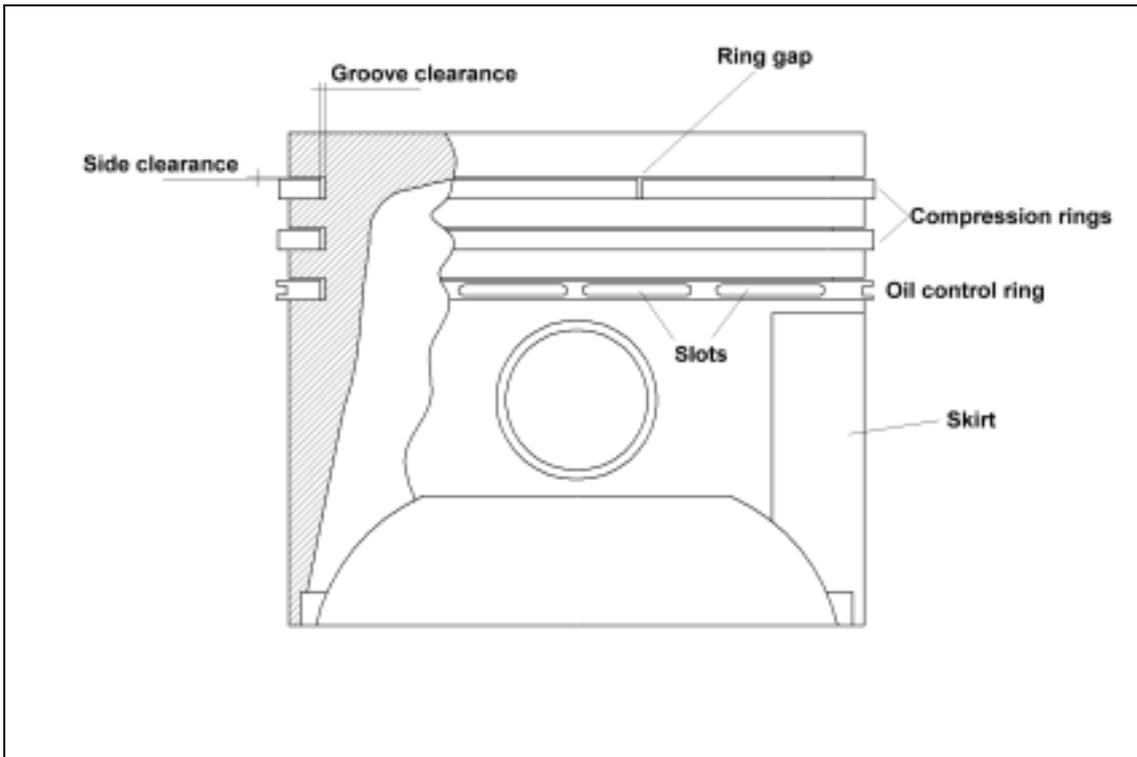
### 2.1 Main functions of piston rings

The functions of a piston ring are to seal off the combustion pressure, to distribute and control the oil, to transfer heat, and to stabilise the piston. The piston is designed for thermal expansion, with a desired gap between the piston surface and liner wall. The

rings and the ring grooves form a labyrinth seal, which relatively well isolates the combustion chamber from the crankcase. The position and design of the ring pack is shown in Fig. 2.1. The ring face conforms to the liner wall and moves in the groove, sealing off the route down to the crankcase. The sealing ability of the ring depends on a number of factors, like ring and liner conformability, pre-tension of the ring, and gas force distribution on the ring faces. Piston rings forces are discussed in greater detail in Section 5.2. Some of the combustion chamber heat energy is transferred through the piston to the piston boundaries, i.e. the piston skirt and rings, from which heat transfers to the liner wall. Furthermore, the piston rings prevent excess lubrication oil from moving into the combustion chamber by scraping the oil from the liner wall during the downstroke. The piston rings support the piston and thus reduce the slapping motion of the piston, especially during cold starts where the clearance is greater than in running conditions. The rings are generally open at one location, at the ring gap, hence easily assembled onto the piston, see Fig. 2.1.

## **2.2 Ring categories**

Piston rings form a ring pack, which usually consists of 2–5 rings, including at least one compression ring. The number of rings in the ring pack depends on the engine type, but usually comprises 2–4 compression rings and 0–3 oil control rings. For example, fast speed four-stroke diesel engines have 2 or 3 compression rings and a single oil control ring. The oil control rings used in diesel engines are two-piece assemblies and spark-ignited engine oil control rings may be three-piece assemblies as well. In addition to the general compression rings and oil control rings there are scraper rings, which have the tasks of both sealing and scraping off the oil from the liner wall. Scraper rings have a beak intended for scraping off the oil, see the Figs. 2.5a and 2.5b.



*Fig. 2.1. Piston and piston rings.*

### **2.2.1 Compression rings**

The compression ring acts as a gas seal between the piston and the liner wall, preventing the combustion gases from trailing down to the crankcase. The rings have a certain pre-tension, i.e. they have a larger free diameter than the cylinder liner, which assists the ring in conforming to the liner. The cylinder gas pressure acts on the back-side of the ring, especially on the top ring, pressing it against the liner. The ring force distribution depends on the face form. With a rectangular face profile the force is higher than with a barrel-shaped face, as the compression pressure is able to act on the face-side of the barrel-shaped ring and thus counteract some of the force owing to ring pre-tension.

Plain compression rings, with a rectangular cross-section, satisfactorily meet the sealing demands of ordinary running conditions and this type of compression ring is the most common one, see Fig. 2.2a. The use of rings with a barrel-shaped face profile (Fig. 2.2b) brings the benefit mentioned above. The ring may have a tapered face profile in order to shorten the running-in period, see Fig. 2.2c. The tapered face profile enables the compression gas pressure to act on the face-side as well and thus relieve the pressure against the liner wall, which reduces the wear rate during running-in. A tapered face profile has a good oil-scraping ability, and the ring can be used as an oil-scraping ring as well as a compression ring.

Bevelled rings can be used as compression rings, see Fig. 2.3. The bevelled profile causes the ring to twist in the ring groove during engine operation. In running conditions the bevelled ring is pressed flat against the liner wall owing to the gas pressure, which causes an additional stress on the ring.

The wedge-type profile or (half) keystone profile is used in order to prevent the ring from seizing in the groove, see Fig. 2.4. High temperature may cause the lubricant in the groove to carbonise. The wedge form makes the ring's axial clearance greater at increasing radial groove clearance. Scraper rings, which are usually used as the second compression rings, can simultaneously be used as oil-scraper rings, see the Figs. 2.5a and 2.5b.

### 2.2.2 Oil control rings

In addition to the task of the compression rings to seal off the combustion chamber from the crankcase, there needs to be some mechanism to distribute the oil evenly onto the liner. The number of oil control rings in a ring pack is one or two. Normally a single oil control ring is sufficient but on occasions a second ring may be required. The appearance of the oil control ring differs from that of the compression ring; see the Figs. 2.5c and 2.5d.

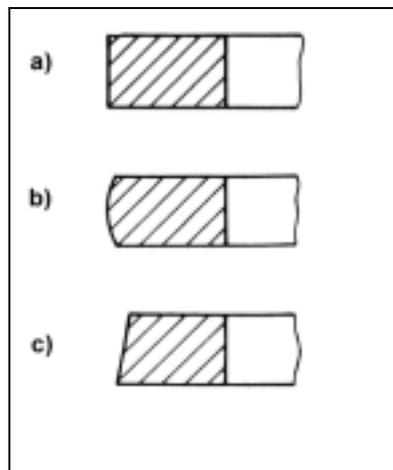


Fig. 2.2. Compression ring cross-section (ISO 6621-1).

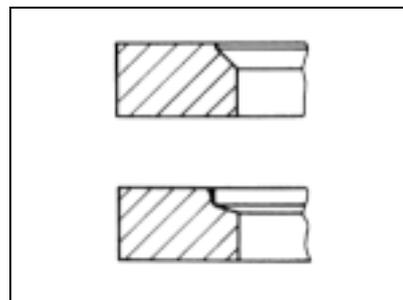
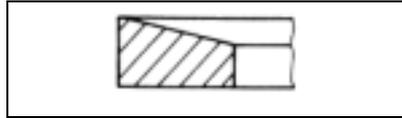
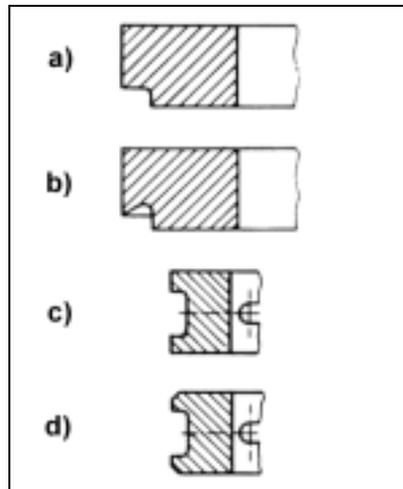


Fig. 2.3. Bevelled ring edge configurations (ISO 6621-1).



*Fig. 2.4. Half keystone ring (ISO 6621-1).*



*Fig. 2.5. Compression and oil control rings (ISO 6621-1).*

The oil control ring is perforated by slots in the peripheral direction, see Fig. 2.1, which provides a way for the excess oil to leave the ring pack area. The scraped oil is collected in the oil control ring groove and transported through the piston back to the crankcase. The scraped oil may run through the possible gap between the liner wall and the piston skirt. With the latter alternative, the oil is forced in front of the oil control ring. The oil control rings may have a coil spring inserted, as the pre-tension of the ring is not sufficient in all instances. The additional force on the oil control rings causes them to have the most extreme lubrication conditions, even though these are the rings that control the oil film. Oil control rings are not always necessary, contrary to the compression rings. Two-stroke spark-ignited engines, for example, have the lubrication oil mixed in the fuel, and therefore need no oil control rings.

### **2.3 Piston ring materials and coatings**

A piston ring material is chosen to meet the demands set by the running conditions. Furthermore, the material should be resistant against damage even in emergency conditions. Elasticity and corrosion resistance of the ring material is required. The ring coating, if applied, needs to work well together with both the ring and the liner materials, as well as with the lubricant. As one task of the rings is to conduct heat to the liner wall, good thermal conductivity is required. Grey cast iron is used as the main material for piston rings (Federal Mogul, 1998). From a tribological point of view, the

grey cast iron is beneficial, as a dry lubrication effect of the graphite phase of the material can occur under conditions of oil starvation. Furthermore, the graphite phase can act as an oil reservoir that supplies oil at dry starts or similar conditions of oil starvation (Glaeser, 1992).

Coatings for rings are widely used. One example of such a coating is chromium, which is used in abrasive and corrosive conditions where running conditions are severe. Hard chrome plating is particularly relevant for the compression ring.

Piston ring surfaces are, in addition to chromium plating, thermally (plasma) sprayed with molybdenum, metal composites, metal-ceramic composites or ceramic composites, as a uniform coating or an inlay coating material (Mollenhauer, 1997).

Experimental work with new powder compositions for thermal spraying has included molybdenum-nickel-chromium alloys, chromium oxide ( $\text{Cr}_2\text{O}_3$ ) with metallic chromium binder, alumina-titania ( $\text{Al}_2\text{O}_3\text{-TiO}_2$ ), tungsten carbide (WC) with metallic cobalt binder,  $\text{MoSi}_2$ , CrC-NiCr (Dufrane, 1989, Radil, 2001). Hard chromium layers can be improved by plasma spraying chromium ceramic on the ring face, thus increasing the thermal load capacity. A dense chromium carbide coating, produced by HVOF coating was found promising for piston ring applications in the work by Rastegar and Richardson (Rastegar and Richardson, 1997).

Thin, hard coatings produced by PVD or CVD include coating compositions like titanium nitride (TiN), chromium nitride (CrN); however coatings of this type are currently used exclusively for small series production for competition engines and selected production engines (Federal Mogul, 1998, Broszeit et al., 1999). Multilayer Ti-TiN coatings have been experimentally deposited onto cast-iron piston rings, and the coating is claimed to be more wear resistant than a chromium plated or phosphated surface, particularly when the number of layers is high (Zhuo et al., 2000).

Haselkorn and Kelley have investigated coatings for use in low-heat rejection engines. They conclude that high carbon iron-molybdenum blend and chrome-silica composite applied by plasma spray, and further chrome nitride applied by low-temperature arc vapour are coatings with properties that meet the demands in low-heat rejection engines (Haselkorn and Kelley, 1992).

Surface coatings/treatments for the entire piston ring surface are based on phosphorus, nitrides, ferro-oxides, copper and tin, as some examples (Federal Mogul, 1998, Mollenhauer, 1997).

The possibility of using ceramic piston rings as a complement to metallic rings in advanced engine applications has been investigated. Miniature tribotests with ceramic materials have included monolithic zirconia, sintered silicon carbide, silicon nitride (Dufrane, 1989), and silicon nitride with a gradient of titanium nitride on the sliding surface (Kustas and Buchholtz, 1996). Unlubricated sliding turned out to be detrimental to the ceramics. Silicon nitride and silicon carbide performed satisfactorily under oil-lubricated sliding conditions, while zirconia suffered from thermal shock cracking.

## **2.4 Sealing ability of piston rings**

Piston rings have to meet various functional demands. The piston ring cylinder liner contact is a dynamic environment. In addition to the problems related to the static sealing capability, the varying shape of the cylinder liner in both the longitudinal and peripheral directions makes the sealing even more difficult in dynamic conditions. The cylinder liner is never ideally cylindrical, see Section 4.2 for details. This requires good conformability of the rings, as described in Section 2.5.1.

### **2.4.1 Conformability**

Conformability means the ability of the piston rings to conform to a deformed cylinder liner. The deformation is caused by thermal and mechanical loading, cylinder head bolt tightening and abrasion. As mentioned above, the cylinder liner has a non-cylindrical axial shape. Piston rings on the other hand are manufactured nearly circular. Deformation of the cylinder liner increases the oil consumption, as the piston rings are not able to fully conform to the shape of the liner. This is the case especially at running-in conditions where the counterparts do not conform to each other. In addition to the cylinder liner deformation experienced by the piston ring under static conditions, the liner cross-section at the location of the piston ring changes shape during the piston motion between the TDC and BDC locations. The ring needs to be flexible to allow rapid changes for obtaining the required shape. Conformability can be improved by increasing the tangential load or by decreasing the momentum of inertia. This, on the other hand, is not always desirable, as higher tangential loads increase the friction. A low momentum of inertia is acquired by decreasing the wall thickness of the ring, which in turn increases the possibility of ring damage.

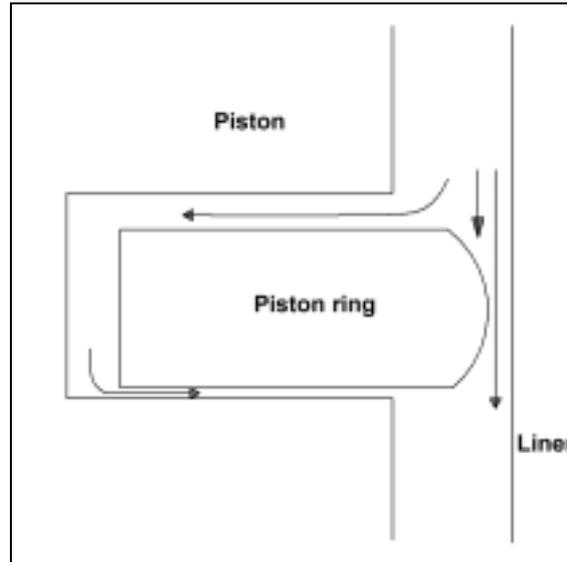
### **2.4.2 Counter surface effects**

The surface roughness of both the cylinder liner and piston ring face affects the lubricating oil film. The oil film thickness is extremely thin where the peaks of the interacting surfaces form contacts. At high oil film pressure conditions the surface asperities deform elastically, reducing the oil film thickness and forming conditions of elastohydrodynamic lubrication.

### **2.4.3 Blow-by prevention**

Blow-by is considered the phenomenon where combustion gases flow from the combustion chamber past the ring pack to the crankcase. The combustion gases flow past the piston ring at various locations: (a) at the piston ring gap, (b) past the front side of the piston ring at starved lubrication conditions, (c) or past the backside of the piston ring when the ring is not in contact with either of the ring-groove walls, see Fig. 2.6. The hot blow-by combustion gases cause the piston and piston rings to overheat. The blow-by disturbs the piston and ring lubrication by affecting the oil film: combustion gases contaminate the lubricant and cause the oil to entrain in them. When the combustion gas reaches the crankcase it pollutes the lubrication oil. Blow-by cannot be totally prevented as long as the rings have gaps and move in their grooves. This means that some blow-by will always have to be allowed. The blow-by affects directly or indirectly the engine power (fuel) efficiency: the blow-by consumes some of the combustion power and increases the friction as a result of less favourable lubrication conditions. The gap between the piston and liner wall is greater on the anti-thrust side of the piston than on the thrust side. This requires that the gap between the back-side of the ring and the ring groove is quite large and thus has a large gas-flow area.

Measurements have shown that the twist of the piston rings affects the amount of blow-by past the ring pack. A negative twist on the second ring can cause instability of the ring, which results in an increase in the blow-by. A positive twist on the second ring can, in turn, cause high land pressure, which may result in radial collapse or axial movement of the ring (Richardson, 1996).



*Fig. 2.6. Combustion gas blow-by past the piston ring.*

#### **2.4.4 Oil consumption reduction and emission suppression**

##### Oil consumption mechanisms

The oil consumption in the piston region has been investigated by means of both computer models and experimental measurements (De Petris et al., 1996, Gulwadi, 2000, Ariga, 1996).

Three different mechanisms of oil consumption occur in the ring-pack system. Oil evaporates from the rings and the cylinder liner into the combustion chamber, oil is thrown off from the ring due to inertia, and gas blowing back towards the combustion chamber entrains oil from the ring pack. In all cases the oil escapes from the cylinder with the exhaust gases.

The evaporated oil trails behind or passes between the top ring and the liner during the downstroke. The amount of oil left behind on the liner is dependent on the scraping effect of the rings (Gulwadi, 2000). De Petris and co-workers point out that reverse blow-by, meaning gas flow from the crankcase to the combustion chamber during the exhaust phase, is one of the most significant means by which the oil is transported into the combustion chamber (De Petris et al., 1996).

During the upstroke, the ring-pack lubrication is dependent of the oil film left on the cylinder liner during the downstroke. The top ring scrapes oil off the liner, and the oil is

accumulated in front of the top ring. This accumulated amount of oil is thrown off the ring at TDC due to inertia (Gulwadi, 2000).

Throughout the engine cycle, oil is present in the ring pack area. This oil becomes entrained in the gases flowing to and from the combustion chamber. This entrained oil comes from accumulated oil at the ring-end gaps and the leading and trailing edges of the rings, from oil on the liner surface, and from oil on the groove surfaces (Gulwadi, 2000).

Since environmental demands are becoming even stricter regarding the exhaust emissions of internal combustion engines, the oil flow into the combustion chamber has to be reduced. As the oil control rings are one of the factors that control the oil flow through the ring pack, their optimisation is important. This optimisation is made by computer modelling, as has been presented by Tian and co-workers. The authors conclude, among other things, that the lubrication between the rails and the liner is greatly affected by the twist of the ring (Tian et al., 1998).

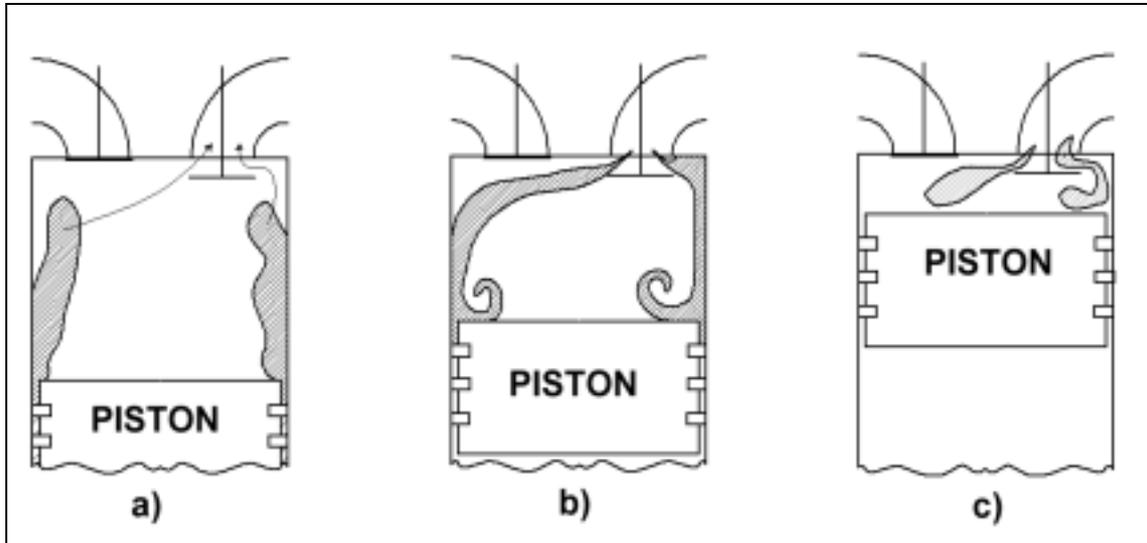
### Emission suppression

Oil consumption reduction actions are related to actions for emission suppression, as the consumed oil, or the oil trailed into the combustion chamber from the ring pack, directly affects the emissions of the engine.

Hydrocarbon emissions are not only caused by lubrication oil that becomes entrained in the combustion gases flowing into and out of the ring-pack area, but also by the fuel-air mixture that flows into the crevices in the ring-pack region and remains there during combustion. During the exhaust phase, these gases flow out of the crevices and mix with the exhaust gas, thus adding to the amount of hydrocarbon in the exhaust gas, see Fig. 2.7. During the blow-down process, unburned hydrocarbons from the ring crevice area move into the combustion chamber as the pressure falls; see Fig. 2.7a. At the beginning of the exhaust stroke, gases, including hydrocarbons, are wiped off from the liner wall and build up into a vortex, see Fig. 2.7b. The vortex builds up even more at the end of the exhaust stroke, see Fig. 2.7c, causing a significant amount of unburned hydrocarbons to leave the combustion chamber.

A method for decreasing the hydrocarbon emissions is to move the top ring higher up on the piston (Lacey and Stockwell, 1999). Thus the area where the air-fuel mixture gases can dwell is decreased. Additionally, the clearance between the ring and the liner wall can be reduced, for reducing the gas volume. Changes of the above type all increase the demands on the lubricant as it is exposed to higher temperatures and higher stresses.

The oil control rings scrape off surplus oil from the liner, which reduces the oil consumption. Appropriate ring designs can control the amount of oil on the liner to be sufficient after the downstroke.



*Fig. 2.7. Gas flow in the combustion chamber and the inter-ring region (After Heywood, 1988).*

### **3. Piston skirt and ring groove**

The main task of the piston is to convert thermal energy into mechanical work. Furthermore, the piston rings seal the combustion chamber from the crankcase and transfers heat to the coolant. The piston skirt acts as a load-carrying surface, which keeps the piston properly aligned within the cylinder bore.

#### **3.1 Piston types and geometry**

There are many piston types developed according to the operating requirements of different engine types. The piston types are commonly categorised by their cooling arrangement, by their primary field of application or by their structure. Well-defined descriptions of different piston types can be found in the Refs. (Röhrle, 1995, Mollenhauer, 1997). Different piston types, regarding the oil supply/cooling mechanism, are discussed in greater detail in Section 7.1.1. Four classifications of piston types, according to their structure, are presented as follows:

1. Uncooled or oil spray-cooled, cast or forged monometal light-alloy pistons for high-speed automotive and small utility vehicle engines.
2. Uncooled or oil spray-cooled cast light-alloy pistons with ring-groove insert for high-speed heavy-duty diesel engines.
3. Single-piece or composite pistons with a cooling gallery for high-speed heavy-duty and medium-speed diesel engines. The right-side of Fig. 3.1 presents a single-piece nodular cast-iron piston, and a two-piece composite piston is shown on the left-side of the Figure.
4. Pistons for two-stroke, low-speed diesel engines.

The most essential areas of the piston are the piston top, the ring belt including the top land, the pin support, and the skirt. The geometry of these areas can vary significantly in compliance with the field of application. The construction of a typical piston and ring assembly is shown in Fig. 2.1 (Section 2.1). Important piston dimensions of four-stroke diesel engines are presented in Fig. 3.2 and Table 3.1.

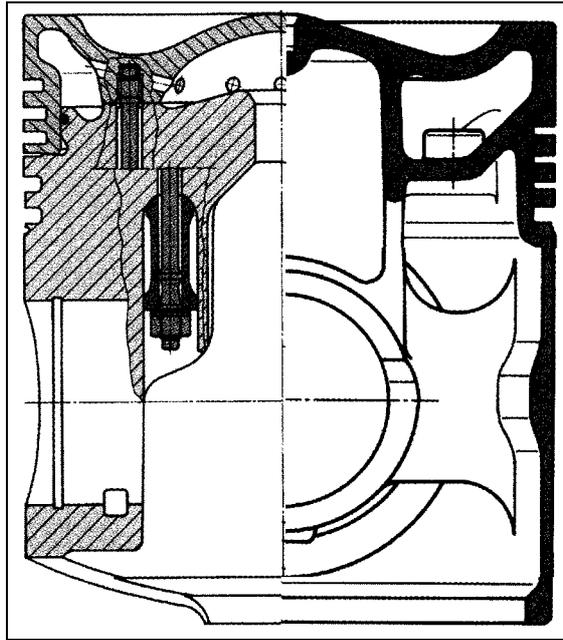
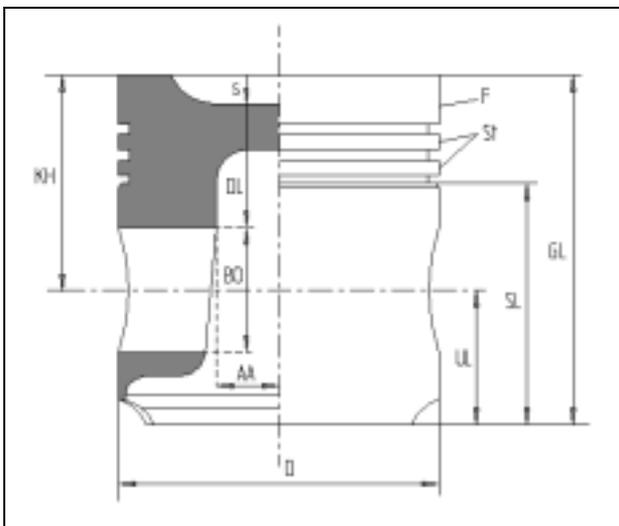


Fig. 3.1. Composite piston with steel top, aluminium lower part and bolting elements (left), single-piece nodular cast-iron piston (right) (After Röhrle, 1995).



Nomenclature:

- F Top land
- s Top deck thickness
- St Ring land
- KH Compression height
- DL Elongation length
- GL Total length
- BO Pin hole and pin diameter
- SL Skirt length
- UL Lower length
- AA Pin boss gap
- D Piston diameter

Fig. 3.2. Piston dimensions (After Röhrle, 1995).

Table 3.1. Main dimensions of light metal pistons (Röhrle, 1995).

	Diesel engines	
	Four-stroke	
Diameter D [mm]	75–180	> 180
Total length GL/D	0.9–1.3	1.1–1.6
Compression height KH/D	0.50–0.80	0.70–1.00
Pin bore diameter BO/D	0.30*–0.40	0.36–0.44
Top land width F/D	0.10–0.20	0.14–0.22
1 <sup>st</sup> ring land St/D**	0.07–0.09	0.07–0.09
Top ring width [mm]	1.5–4.0	3.5–8.0
Skirt length SL/D	0.50–0.90	0.70–1.10
Pin boss spacing AA/D	0.30–0.42	0.28–0.46
Crown thickness s/D	0.10–0.15***	0.13–0.20
Characteristic weight $G_N/D$ [g/cm <sup>3</sup> ]	0.9–1.4	1.1–1.6
* Minimum value for passenger car diesel engines		
** Figures given for diesel engines apply to ring insert pistons		
*** For direct injection engines approx. $0.2 \times$ the combustion cavity diameter		

### 3.2 Piston skirt materials

To achieve low mass forces, high resistance against deformation and fatigue failure, and good sliding properties, the piston materials should fulfil specific requirements, such as:

- low density,
- high strength under temperature variations,
- high heat conductivity,
- good wear characteristics, and
- favourable heat expansion.

Typical piston materials are light alloys, cast iron, nodular cast iron, and alloyed steels. The pistons for high-speed engines are primarily made of aluminium silicon alloys. In addition to aluminium, these alloys contain as constituents 11–13 % silicon and approx. 1 % each of copper, nickel and magnesium. By increasing the silicon content to approx. 18–24 %, the heat expansion and wear can be reduced but the strength falls. The strength of aluminium-silicon materials can be increased by locally casting-in ceramic short fibres made of aluminium oxide, thin single crystalline fibres (whiskers), or

porous metallic parts. In large low-speed two-stroke engines, pistons of high-grade iron-based materials, primarily nodular cast iron with a pearlitic base structure, are still employed today. In high mechanical and thermal load applications the top of the composite piston consists of high-strength materials such as nodular cast iron, or cast or forged steel (40Mn4, 42CrMo4V and X45CrSi9) with high high-temperature strength. The lower parts are typically made of nodular cast iron (Röhrle, 1995).

The trend to increase the power density of the engines sets requirements on enhancing the load-bearing capacity of the pistons. A Ferrocomp®-piston design is presented in a paper by Lipp and Schmidt. The composite piston consists of a forged steel crown and a forged steel skirt. This piston can be used with peak cylinder pressures above 25 MPa and is applicable in a diameter range from 160 to 640 mm. The material used in the forged piston skirt is micro alloyed, high-quality yield-treated steel 38MnVS6. One of the major advantages of forged steel is the higher fatigue strength, especially on the unmachined surface. The elongation at break is more than three times higher and Young's Modulus is approx. 25 % higher compared to cast iron GG-70. The higher density of forged steel ( $7.85 \text{ g/cm}^3$ ) compared to nodular cast iron ( $7.2 \text{ g/cm}^3$ ) is compensated by minimising the draft angle in the areas where the surface stays unmachined and by optimising the structure of the piston assembly (Lipp and Schmidt, 2001).

In a study by Inada, a piston design for high engine output powers is presented. The new features of the design are a nitrided nodular cast-iron piston and a dual cooling nozzle system. The conclusions drawn from the study state that the oxidation-resistance and the thermal fatigue life of a nodular cast-iron piston can be increased by applying the nitriding process, and the piston cooling efficiency can be improved by adopting a dual cooling nozzle system (Inada, 2001).

The surface coatings used for pistons can be divided into the following application fields:

1. Coatings to improve sliding characteristics.
2. Coatings to increase wear resistance.
3. Coatings to improve thermal properties.
4. Coatings to increase the knock resistance.

In terms of tribology, the application fields 1 and 2 are the most interesting ones. Metallic and graphite coatings are used for improving the sliding characteristics of aluminium-silicon piston alloys. Lead and tin coatings are used on aluminium surfaces to achieve favourable running-in properties. The thickness of the metallic coatings on pistons for gasoline engines is typically from 1 to 2  $\mu\text{m}$ . Phenolic resin graphite

coatings, from 10 to 20  $\mu\text{m}$  thick, are used in large pistons and on pistons in automotive gasoline and diesel engines. Thin metallic phosphate layers can improve the adhesion properties of these coatings. To improve the wear resistance of the aluminium pistons in aluminium cylinders, the piston skirt surface can be coated with a wear-resistant iron or chromium layer covered with a thin tin layer for break-in. A hard chromium layer can improve the wear resistance of the ring grooves in steel composite piston crowns (Röhrle, 1995).

Durga and co-workers have studied the influence of the surface characteristics on the frictional behaviour of rubbing surfaces by experiments on baseline and coated piston skirts against different cylinder liner coatings including an atomised spray coating called II-25D (Epoxy+BN+MoS<sub>2</sub>+Graphite). The baseline piston skirt material used in the tests was 318 Aluminium Alloy and the coating applied was the atomised spray coating II-25D. The piston rings were standard production Fe-Mo coated cast-iron rings. The results of the friction rig tests showed that the II-25D coating produces a significant reduction in friction coefficient in the entire piston speed range (boundary friction coefficient 0.07, and cycle average 0.05). The authors mention that, since II-25D is a coating, the low friction behaviour can be expected regardless of the substrate. As a result of single-cylinder engine tests, it is stated that the II-25D coating significantly reduces motoring friction torque relative to the baseline liners at high speeds. With the II-25D coating the power improvement was around 4.5 % (Durga et al., 1998).

### 3.3 Piston ring groove

In a large number of piston designs, the piston ring belt consists of three ring grooves. The piston rings are situated in the grooves between the ring-groove flanges. Since the ring groove and the flanges are part of the piston sealing system, affecting the blow-by of the combustion gases and the oil consumption, the surfaces of the flanges have to be of very high quality. For comparison, the standard side-face finish of a piston ring has a surface quality of  $R_z = 4 \mu\text{m}$  or  $R_a = 0.8 \mu\text{m}$  (ISO 6621-4).

Damaged sealing surfaces lead to increased blow-by and reduced effective combustion pressure. At the same time the increased flow of the hot blow-by gases interferes with the oil film between the sliding surfaces and may cause hot gas damage to the piston rings.

To increase the wear resistance of the ring grooves in the pistons of heavy fuel oil engines, the grooves are typically either induction hardened or chromium plated.

The wear of the ring groove flanks can affect the effective geometry of the ring face against the cylinder liner (Dowson, 1993). In order to improve ring-groove wear

resistance in steel composite piston crowns, a hard chromium layer can be applied. To protect the first ring groove, sometimes also the second, in high-performance diesel engines against wear, so-called ring carriers made of high-alloyed cast iron are cast-in. Ring carriers are preferably made of Niresist, an austenitic cast iron with a thermal expansion coefficient almost equal to that of aluminium (Röhrle, 1995).

## 4. Cylinder liner

Since the piston and the piston rings are moving in the cylinder, the cylinder liner constitutes an important tribological element as a sliding surface against the piston and piston rings.

### 4.1 Cylinder liner materials

The cylinders can be made of cast iron containing phosphorus, manganese, chromium, molybdenum, vanadium and titanium as alloying elements, or steel or aluminium. Cast iron is the most commonly used material in the cylinder liners of larger engines. Material specifications for cast-iron cylinder liners are presented in Table 4.1. Nodular cast-iron cylinders with cermet (ceramic-metal composite) sliding surfaces have been used in some low-speed two-stroke diesel engines. The liner surface can be coated with a hard chromium layer to improve the wear resistance of the cylinder liners (Affenzeller, 1996). Grey cast iron, when used for cylinder liners, is tribologically beneficial, as the graphite phase of the material gives a dry lubrication effect and furthermore acts as an oil reservoir that supplies oil at dry starts or similar conditions of oil starvation (Glaeser, 1992).

*Table 4.1. Typical material composition for cast iron cylinder liners (Affenzeller, 1996).*

	Composition [%]							
	C	Si	Mn	P	Cr	S	Mo	Ni
Standard 45	2.8–3.2	1.7–2.4	0.5–0.8	0.4–0.45	0.25–0.4	< 0.03	-	-
Standard P	2.8–3.2	1.7–2.4	0.5–0.8	0.6–0.8	0.25–0.4	< 0.3	-	-
HE G40	2.6–2.8	1.1–1.6	< 0.8	< 0.08	-	< 0.08	1.0–1.5	1.0–1.5
ASTM 247	3.1–3.4	1.85–2.3	-	< 0.12	< 0.35	< 0.18	0.25	0.50

To improve the wear resistance of an aluminium cylinder, a ceramic particulate phase can be cast-in into the aluminium liner during the manufacturing of the block. Aluminium cylinder blocks can be equipped with a cast-in or pressed-in cast-iron liner.

### 4.2 Plateau honing

The trend in the cylinder bore surface finishing has been directed by demands to reduce oil consumption, increase the durability and increase the resistance against wear and

scuffing. To limit hydrocarbon emissions and particles by reducing the oil consumption, the surface roughness of the cylinder liner should have an  $R_a$  value between 0.25 and 0.4  $\mu\text{m}$  and an  $R_z$  value between 3 and 6  $\mu\text{m}$  (Affenzeller, 1996). This can be achieved with optimally honed bore surfaces. There is a consensus for a surface texture of clean-cut surfaces with smooth, flat plateaux and a regular, consistent arrangement of primary oil-retaining valleys (Lenthall, 1996). The surface finish of the cylinder liner has an influence on the scuffing resistance. The rise in friction, which leads to the scuffing, is associated with the polishing of the liner (Galligan, 1999b).

Honing is applied for finishing the surface of cast-iron cylinder liners. The cutting marks of the honing form a pattern of diagonal valleys on the liner surface. The honing grooves, the volume and the direction of the valleys control the amount of oil available, by keeping the oil on the liner surface and by improving the spreading of the oil. Since the requirements of good sealing properties and optimal lubrication are contrary to each other, the demands on the topography of the cylinder liner are exacting. (Ohlsson, 1996, Lenthall, 1996).

In slow-speed two-stroke diesel engines the liner surface is traditionally finished with a special cutting tool instead of honing in order to support the running in (Lenthall, 1996).

The quality of the honed surface is affected by the bore geometry and diameter tolerances, the surface roughness of the fine bored surface, the machining operations and the number of machining stages, and the material, hardness and type of the honing stones. To obtain the specific surface, the liner is usually machined in four steps:

1. Rough boring – for the basic geometry.
2. Rough honing – for alignment.
3. Fine honing – for desired surface roughness.
4. Plateau-honing – for surface smoothing.

Step 3 removes all the traces of the first two steps. Step 4, the plateau-honing, partly replaces the running-in process of the liner surface, which improves the dimensional tolerance of the cylinder, increases the engine efficiency and decreases oil consumption (Ohlsson, 1996).

Many parameters have been used for characterising the plateau-honed surface:

- 2D-parameters, such as  $R_a$  (mean deviation of the surface roughness),  $R_z$  (mean surface roughness), and  $R_{\text{max}}$ .
- Parameters describing the shape of the surface, such as skewness ( $R_{\text{sk}}$ ) and kurtosis ( $R_{\text{ku}}$ ).

- Functionally characterised 2-dimensional  $R_k$  and 3-dimensional  $S_k$  parameters, such as  $R_{pk}$  and  $S_{pk}$  (Reduced peak height, addresses the running-in properties),  $R_k$  and  $S_k$  (Core roughness depth, addresses the wear and load-carrying capacity), and  $R_{vk}$  and  $S_{vk}$  (Reduced valley depth, addresses the oil volume for lubrication).
- On the probability plotting method based parameters for the control of the manufacturing process, such as  $R_{qpc}$  ( $R_q$  value of the plateau giving a surface roughness value of the plateau-honing step),  $R_{qvc}$  ( $R_q$  value of the fine-honing step) and  $M_{r2c}$  (intersection of the two lines defined by the angular coefficients  $R_{qpc}$  and  $R_{qvc}$ , define the share of plateau versus valley texture in the topography) (Ohlsson, 1996).

The depth of surface deformation created by different machining operations is presented in Table 4.2.

*Table 4.2. Machining operation versus deformation (Lenthall, 1996).*

Machining operation	Depth of deformation
Coarse machining	70–80 $\mu\text{m}$
Fine machining – Ceramic	50–60 $\mu\text{m}$
Base hone – Diamond	25–35 $\mu\text{m}$
Base hone – CBN	10–15 $\mu\text{m}$
Base hone – SiC	10–15 $\mu\text{m}$
Plateau hone –Diamond base	15–20 $\mu\text{m}$
Plateau hone –CBN base	5–10 $\mu\text{m}$
Plateau hone –SiC base	5–10 $\mu\text{m}$

### 4.3 Macro form deviation of cylinder liners

In practice, it is normal that the cylinder liner is not perfectly cylindrical and of nominal bore size along its entire length. The bore distortion causes loss of conformity between the piston rings and cylinder liner. Limited piston ring follow-up performance, in particular caused by bore deformation, causes an increase on the lubricating oil consumption.

The non-circularity of the cylinder bore can be described by a Fourier series (Affenzeller, 1996, Chittenden and Priest, 1993, Ma et al., 1996, 1997a):

$$R(\phi) = \sum_{i=0}^{i=n} (A_i \cos \phi + B_i \sin \phi) \quad (4.1)$$

where:

- R( $\phi$ ) = radial co-ordinate
- $\phi$  = angular co-ordinate
- A<sub>i</sub>, B<sub>i</sub> = amplitude constants
- i = order
- n = highest order distortion to be considered

The co-ordinate system and various Fourier orders of bore distortion are presented in Fig. 4.1.

There are several reasons for the non-circularity of the cylinder bore. The cylinder liners are machined to a level of accuracy specified in terms of tolerances. The allowed difference between maximum and minimum diameter of the cylinder bore may be from 10 to 100 times the thickness of the oil film between piston rings and the liner. Therefore, the deviation from circularity within the manufacturing tolerances is likely to have a significant effect on the performance of the piston assembly. The zero and first-order bore distortions (see Fig. 4.1) are a function of the size and location tolerances of the cylinder bore (Chittenden and Priest, 1993).

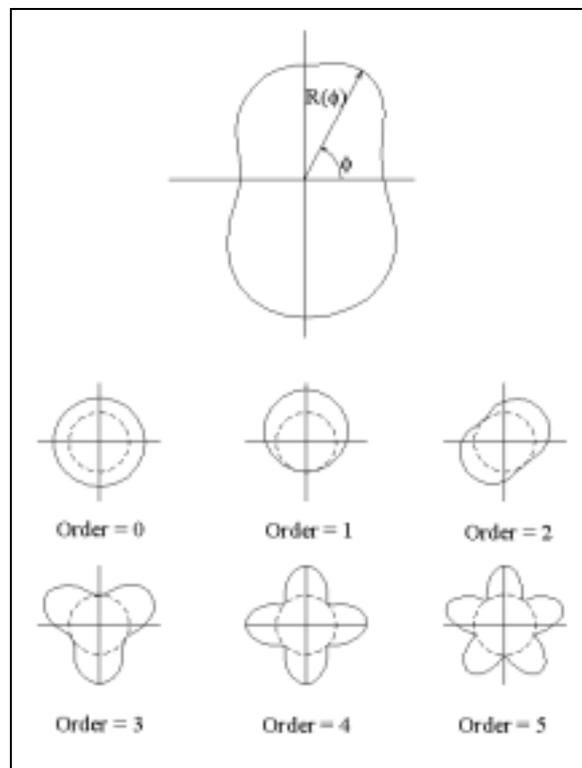


Fig. 4.1. Cylindrical Co-ordinate system and Fourier orders (After Chittenden and Priest, 1993).

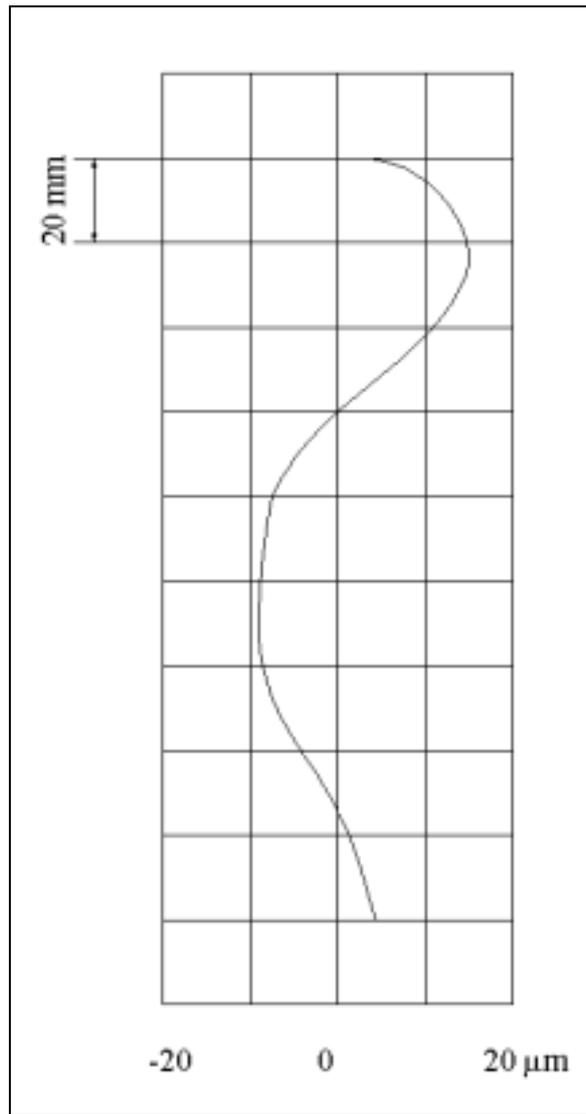
The assembling of the engine can cause deformations into the cylinder liner. An example of this is the tightening of the cylinder head bolts. According to Ref. (Chittenden and Priest, 1993), if a fourth-order distortion (Fig. 4.1) is a major component in a distorted cylinder bore of a particular engine, the tightening of the four cylinder head bolts of this engine is the reason. The clamping of the cylinder liner between the cylinder head and the support at the lower end of the liner in the engine block, results in an outward deformation of the cylinder liner's inner surface (Reipert and Voigt, 2001).

Inadequate cooling or over-cooling of a specific region of the cylinder may cause expansion differences around the circumference of the cylinder and along its length, which leads to distortion of the cylinder bore (Chittenden and Priest, 1993). The magnitude of the thermal expansion is by far larger than the deformation caused by the clamping of the bolts (Reipert and Voigt, 2001).

The gas pressure acts on the cylinder wall in a restricted area. Distortion of the cylinder liner due to the combustion pressure is significant only in highly rated diesel engines with thin-walled wet liners (Chittenden and Priest, 1993).

The maximum deformation by clamping, by thermal expansion and by gas pressure is observed at the upper edge of the cylinder (Reipert and Voigt, 2001). The ring conformability reduces significantly with an increase in the order of bore distortion. Complicated ring geometries with several non-conformed regions arise from a combination of multiple-order bore distortions. (Ma et al., 1997a, Chittenden and Priest, 1993).

In addition to the non-circularity of the cylinder bore, the distortion of the cylinder liner can be axial. An axial plot of cylinder liner distortion is shown in Fig. 4.2.



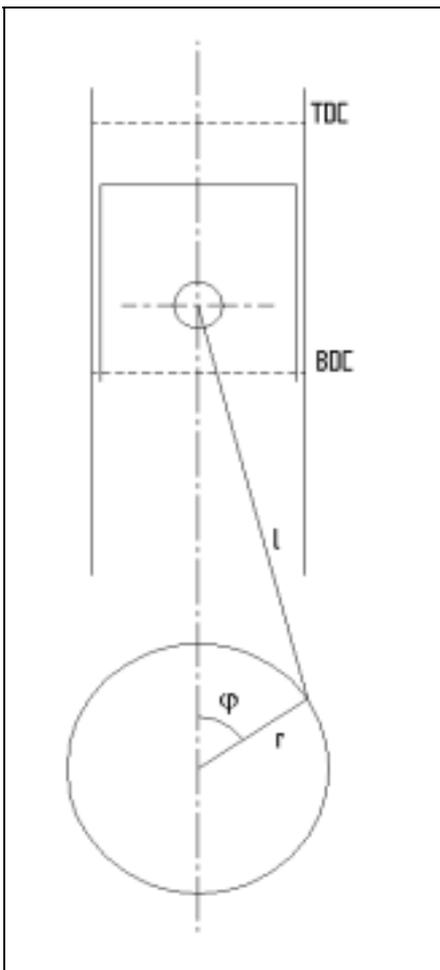
*Fig. 4.2. Axial plot of cylinder liner distortion (After Affenzeller and Gläser, 1996).*

## 5. Piston ring mechanics

### 5.1 Piston ring kinematics and kinetics

One of the major requirements on the ring pack is related to the ring dynamics; radial and axial ring motion and ring twist. Ring motion and ring twist about the ring centre affect the operation of the ring, the oil film formation and the friction between the ring and the liner, the wear of the ring and cylinder liner, and the blow-by across the ring pack.

The primary motion of the piston rings is equal to the reciprocating piston motion. In an analysis of the piston ring lubrication, it is necessary to determine the velocity of the piston ring as a function of the crank angle. The crank mechanics is shown in Fig. 5.1. The instantaneous speed of the reciprocating piston motion can be estimated with decent accuracy with the following formula (Maass and Klier, 1981):



$$v_p \approx r\omega(\sin \varphi + \frac{\lambda}{2} \sin 2\varphi) \quad (5.1)$$

where:

- $v_p$  = instantaneous piston speed
- $r$  = crank radius
- $\omega$  = angular velocity of the crank
- $\varphi$  = crank angle
- $\lambda$  = conrod ratio, =  $r/l$
- $l$  = conrod length

Fig. 5.1. Crank mechanism with piston location parameters.

The exact speed of the piston can be calculated by derivating the equation determining the distance between the piston and the top dead centre. The correlation between piston speed and crank angle is presented in Fig. 5.2. Figure 5.3 presents the piston speed curves for two existing diesel engines as a function of the crank angle.

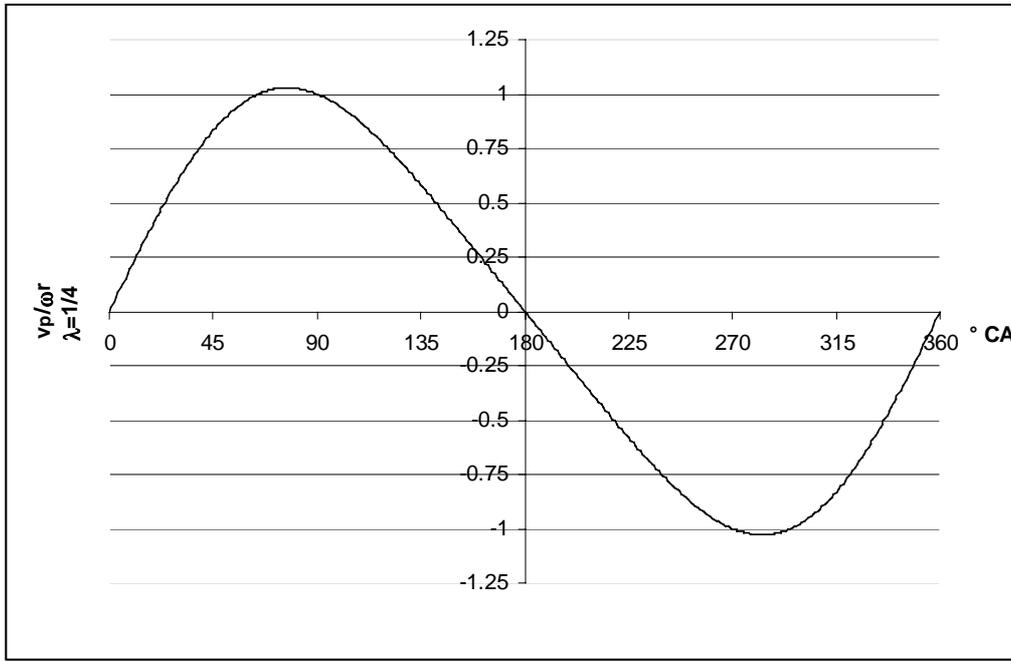


Fig. 5.2. Relative piston speed ( $v_p/\omega r$ -value) plotted against crank angle, when  $\lambda=1/4$ .

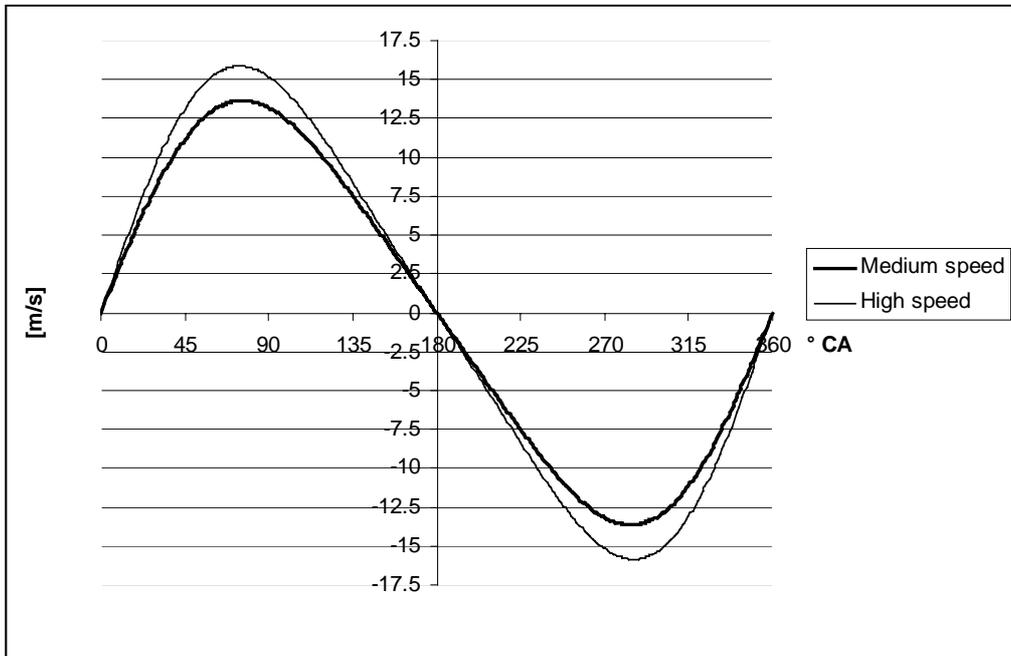


Fig. 5.3. Piston speed curves for a medium-speed and a high-speed diesel engine.

The piston ring acceleration influences the phenomenon of ring lift. The acceleration of the piston can be calculated from the equation of the piston speed by derivating it. The correlation between piston acceleration and crank angle is presented in Fig. 5.4. In Fig. 5.5 the piston acceleration curves for two existing diesel engines as a function of crank angle are presented. The following equation gives an approximate value for the acceleration of the piston (Maass and Klier, 1981):

$$a_p \approx \omega^2 r (\cos \varphi + \lambda \cos 2\varphi), \quad (5.2)$$

where:

$a_p$	= piston acceleration
$r$	= crank radius
$\omega$	= angular velocity of the crank
$\varphi$	= crank angle
$\lambda$	= conrod ratio

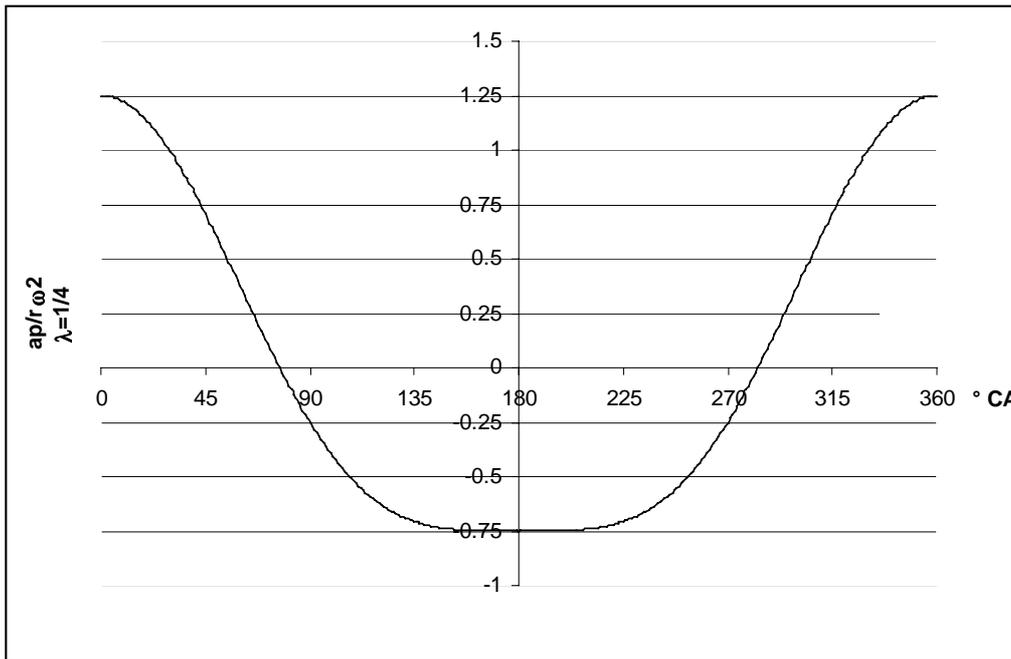


Fig. 5.4. Relative acceleration of the piston ( $a_p/\omega^2 r$ -value) plotted against crank angle, when  $\lambda=1/4$ .

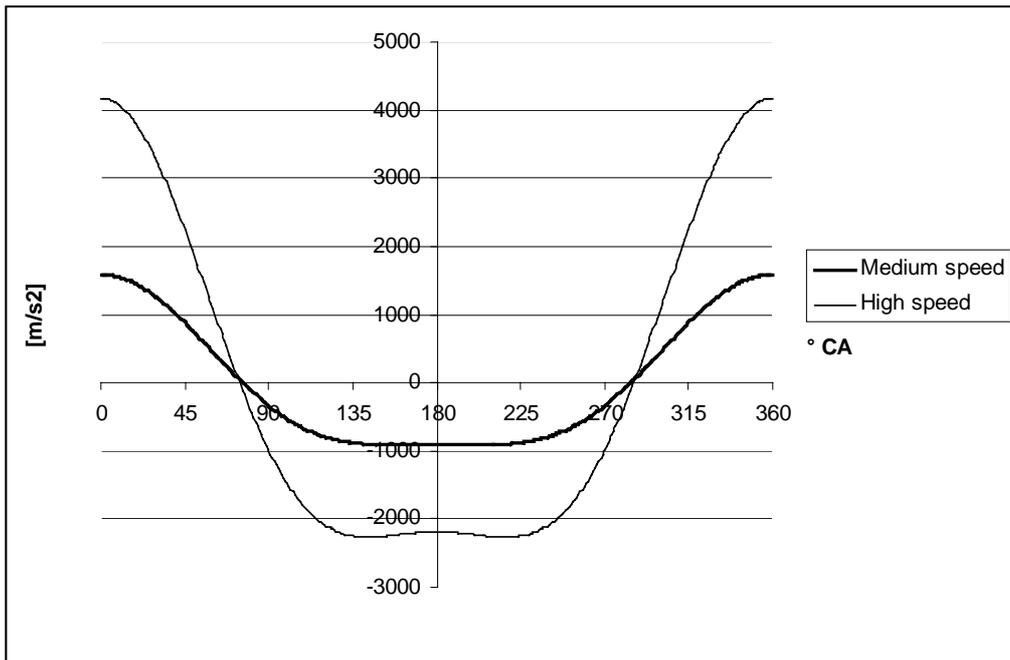


Fig. 5.5. Acceleration curves for medium and high-speed diesel engine pistons.

Apart from the reciprocating motion of the piston, the secondary motion of the piston affects the piston ring operation. The clearance between the piston and cylinder liner allows lateral movements and tilt of the piston according to the forces and moments acting on it. The basic modes of piston motion in the bore are presented in Fig. 5.6. (Haddad and Tjan, 1995, Chittenden and Priest, 1993).

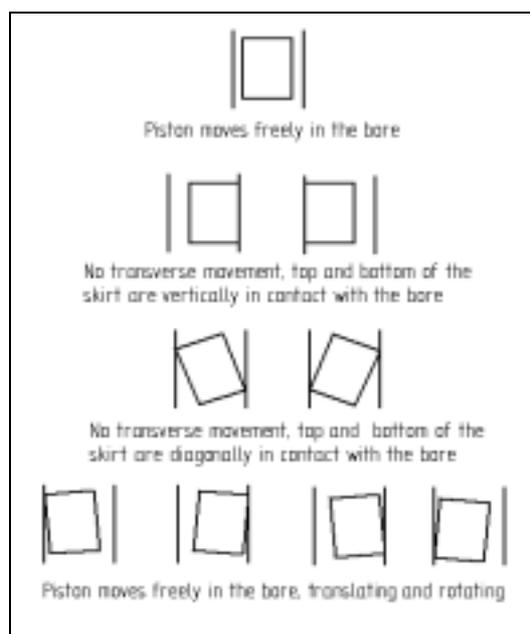


Fig. 5.6. Basic modes of piston motion in the bore (After Haddad and Tjan, 1995).

## 5.2 Piston ring forces and moments

The piston ring secondary motions can be divided into piston ring motion in the transverse direction, piston ring rotation, ring lift, and ring twist. These types of motion result from different loads acting on the ring. Loads of this kind are inertia loads arising from the piston acceleration and deceleration, oil film damping loads, loads owing to the pressure difference across the ring, and friction loads from the sliding contact between the ring and cylinder liner. The forces acting on the ring are presented in Fig. 5.7. (Ejakov et al., 1999).

The gas pressure above, below and behind the ring produces resultant forces on the ring section (Dowson, 1993). The inertia forces acting on the piston rings, as well as those acting on the other reciprocating crank mechanism components, change proportionally to the square of the engine speed (see the Figs. 5.2 and 5.4) (Röhrle, 1995). The side loading of the piston against the cylinder wall is a result of the articulated joint of the connecting rod (Röhrle, 1995). The effect of the clearance between the cylinder liner and the piston on the piston and piston ring motion and to the ring forces is presented in Section 5.1, Fig. 5.6. The shearing of the lubricating film, the sliding friction forces and the contact pressure between the ring and the liner cause normal and tangential forces on the ring face.

The elastic distortion of the piston and liner can affect the effective geometry of the ring face and cylinder liner contact, which causes a non-uniform distribution of the contact pressure between the cylinder liner and the piston ring face and can thus lead to increased blow-by and oil consumption (Dowson, 1993).

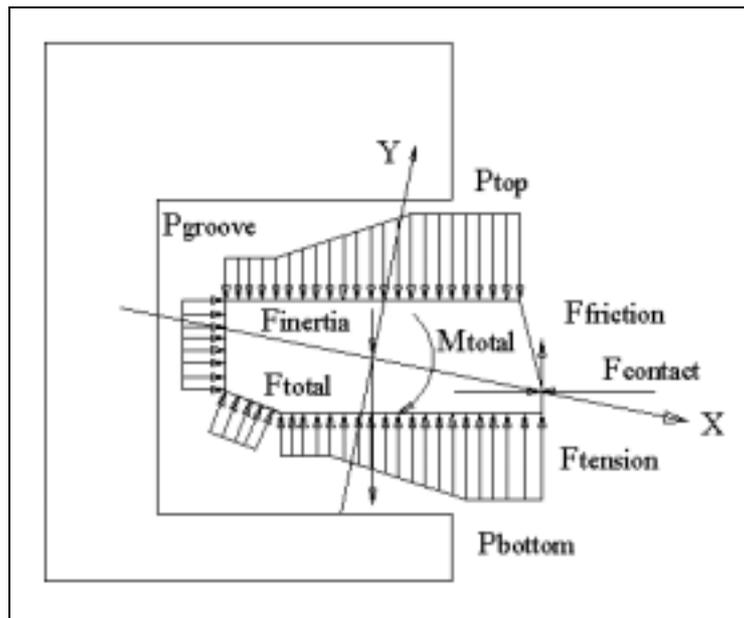


Fig. 5.7. Forces acting on the piston ring (After Ejakov et al., 1999).

The piston pin is often offset from the piston centreline. This arrangement is applied in order to avoid piston-generated noise or to reduce the thermal load on the ring grooves (Röhrle, 1995). Haddad and Tjan have used a computer program to investigate the influence of the offset of a piston pin, centre of gravity, and crank offset from the bore centreline, on the mechanical efficiency and engine noise. The results presented predict that, generally, the kinetic energy loss decreases when the piston pin offset is set to the thrust side of the piston and the mechanical efficiency increases when the piston pin offset is set to the minor thrust side of the piston. In the conclusions of their work, the authors state that the piston pin offset is the most sensitive parameter producing considerable variations in kinetic energy loss and mechanical efficiency. Furthermore, the kinetic energy loss can be reduced, and the mechanical efficiency can be increased by setting the piston pin offset to the thrust side of the piston centre (Haddad and Tjan, 1995). Chittenden and Priest have presented the same kind of results. According to their predictions, the contact situation will be worse and the friction losses will increase if the piston pin offset is positioned towards the minor thrust side of the piston (Chittenden and Priest, 1993).

## **5.3 Ring contact pressure**

### **5.3.1 Contact pressure from ring compression**

The tangential force of the piston rings depends on the piston ring type and the class of nominal contact pressure. For example, the theoretical contact pressure used in the calculation of the tangential forces of rectangular and half keystone rings made of steel is approximately  $0.19 \text{ N/mm}^2$ . The spring force of the compression rings is lower than that of the oil ring (Dowson, 1993). In the case of coil-spring-loaded oil control rings, the pressure is approximately  $1 \text{ N/mm}^2$ . The values of nominal contact pressures and specific tangential forces for various piston rings are tabulated in the standards ISO 6621...6626 (ISO 6621-4).

According to work by Dowson, the loading caused by the elastic spring force on rings with gas pressure acting on the rear of the ring is typically  $10^4$ – $10^5 \text{ Pa}$ . This pressure presents only about 1 % or less of the peak gas pressure (Dowson, 1993).

The contact pressures arising from ring compression are defined for undamaged piston rings. The situation changes significantly in the case of a ring breakage.

### **5.3.2 Contact pressure arising from gas pressure**

The gas pressure behind the first compression ring varies according to the cylinder pressure. The gas pressure behind the second compression ring is already significantly lower than the pressure behind the first compression ring. The gas pressure behind the oil ring stays through the whole work cycle almost equal to the pressure on the crank chamber. With a gas pressure acting on the piston ring, the contact pressure and thus the conformity is greatly increased. On the other hand, the gas pressures are of significance only for a small proportion of the engine cycle (Chittenden and Priest, 1993).

### **5.3.3 Additional loads from the deformation of cylinder bore and ring**

While the ring and/or cylinder bore deformation leads to non-uniform contact pressures, the local pressures can momentarily rise to considerably higher values than the mean contact pressure. In a region of larger distortion, the ring fails to conform to the bore. In order to support the ring loading, the ring must generate greater hydrodynamic pressures in the area where the bore is closest to the ring (Ma et al., 1997a). The cylinder bore deformation is discussed in more detail in Section 4.2.

The contact and surface pressures are affected on a micro-scale of the honing, wear and the surface quality of the cylinder bore together with the surface quality of the piston ring.

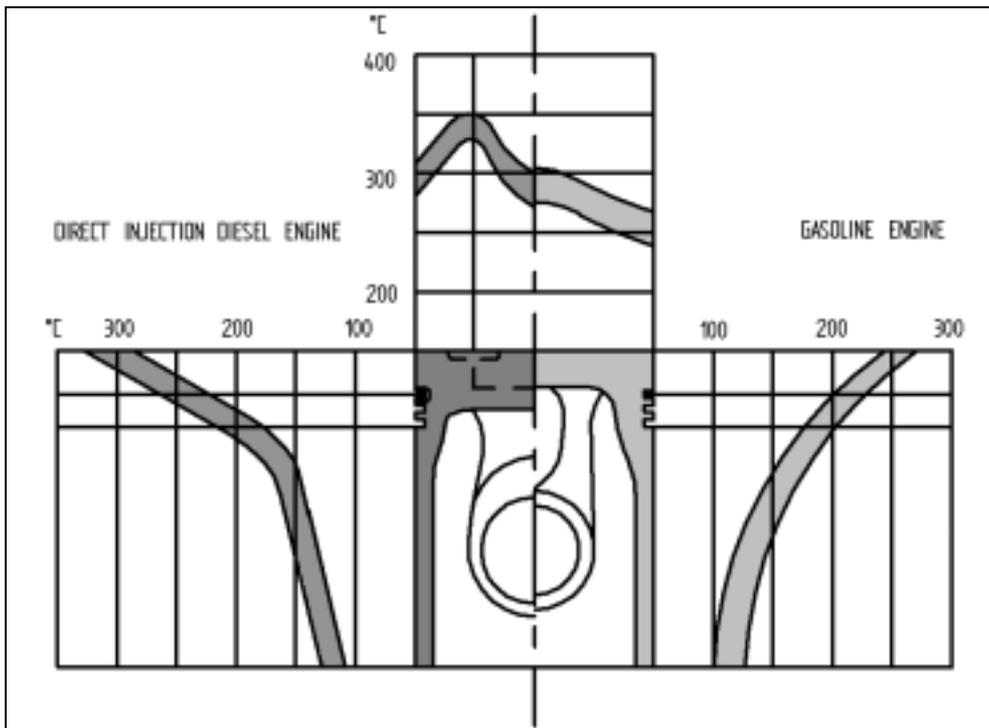
### **5.3.4 Non-uniform pressure distribution due to ring twist**

The ring twist affects the access of the gas pressure flow behind and between the piston rings, which causes non-uniform contact pressures and reduced conformity. In ring twisting, different sections of the face surface of the ring alternate to form the ring/liner contact, and this leads to a non-uniform contact pressure. The ring twist can be affected by the ring face profile. Tian has studied dynamic behaviour of piston rings. He states that using a symmetrical barrel face, the groove's downward tilt angle that gives uniform average contact pressure along the radial direction is less than in using an offset parallel face (Tian, 2002).

## 6. Thermal loads

### 6.1 Macro-scale thermal loads

The ring belt area, including the top land and the first piston ring, is directly thermally exposed to the combustion gases and thus subjected to transient temperature variations. Figure 6.1 presents a schematic overview of the operation temperatures of diesel and gasoline automotive engines at full load (Röhrle, 1995).



*Fig. 6.1. Schematic overview of operation temperatures of automotive engines at full load (After Röhrle, 1995).*

The transient peak temperatures of the gas in the combustion chamber during the combustion in a diesel engine can rise to approximately 2 500°C (Saad et al., 1991). High heat flux causes problems such as thermal stress and deterioration of the lubricating oil film (Liu and Reitz, 1998). The mechanical loads are superimposed on thermal stresses. The high combustion temperature gradients on the piston top cause the hardness of the piston alloy to drop (Röhrle, 1995), and the yield strength and shear strength of the material decrease as the temperature increases.

The temperature of the piston top land is generally higher than that of the piston rings. In the top land area the lubricant is subject to the highest loads with respect to its thermal and oxidative stability. The characteristics of the lubricant in response to these conditions influence on the extent of the carbon deposit formation. Thereby, the

temperatures indirectly contribute to wear, bore polishing and scuffing (Saad et al., 1991).

The local temperature gradients between the combustion gases and the surfaces increase the heat transfer, which primarily takes place by convection, while heat transfer by radiation is low. The deposit formation results in an increase in the radiation absorption. From 40 to 70 % of the total heat flow into the top of the piston is transferred by the piston ring belt area and the cylinder wall surface into the coolant (Röhrle, 1995).

In addition to the above-mentioned features, the heat expansion and heat-related distortion of the piston top land, piston rings and cylinder liner cause potential loss of conformity between the piston rings and cylinder liner.

## 6.2 Micro-scale thermal loads

The temperature on a sliding surface is in most cases higher than the bulk temperature owing to the frictional work that takes place at the sliding surface. According to the first law of thermodynamics, which expresses the principle of conservation of energy, heat and work are two mutually convertible forms of energy. Hence an increase in temperature is a natural consequence of the frictional work. It is worth noticing, however, that thermal energy cannot be completely converted into mechanical work, as is stated in the second law of thermodynamics (Matthews, 2000). Frictional heating appears as the heating of fluids due to viscous flow between surfaces in relative motion, and as the heating of solid surfaces in sliding or rolling contact.

### 6.2.1 Frictional heating in unlubricated sliding contacts

The rise in temperature of a sliding surface due to frictional heat generation depends on the frictional power ( $P_\mu$ ) and the interaction of several factors, such as the real area of the sliding contact, the specific heat of the material, the thermal conductivity of the material, the temperature and volume of the surrounding material, and the cooling provided by a fluid lubricant. The frictional power as such is defined as

$$P_\mu = F_\mu \times v = F_N \times \mu \times v \quad (6.1)$$

where  $F_\mu$  is the friction force,  $v$  the sliding or rolling velocity,  $F_N$  the normal force, and  $\mu$  the coefficient of friction.

The effect of the frictional work on the temperature of the sliding surfaces can be divided into two parts, depending on the depth of influence and the surface area proportion involved:

Firstly, an increase occurs in the bulk surface temperature of the material, at a depth of a few ten of  $\mu\text{m}$  below the surface. The bulk surface temperature can be approximated as being uniform over the entire sliding surface. The bulk surface temperature increases with an increase in load and velocity (Kong and Ashby, 1991).

Secondly, the frictional work causes local or flash temperatures at the surface asperities where the sliding contacts actually take place. Owing to the minor volumes of material involved in the sliding contacts, the flash temperatures are significantly higher than the bulk temperature and the bulk surface temperature (Kong and Ashby, 1991). According to early work by Bowden and Tabor, the flash temperatures, or hot spots, may rise to the melting temperature of one of the two materials that has the lower melting temperature (Bowden and Tabor, 1971). The flash temperature increases with an increase in the sliding velocity, while it is less sensitive to the load applied (Kong and Ashby, 1991).

The above principles for the temperature distribution and levels in a sliding contact have been used for developing the T-maps software, which is intended for the analysis of simple sliding contacts (Ashby et al. 1990, and subsequent versions), see Fig. 6.2. The original T-maps software deals with unlubricated contacts, however the principle can be adopted on boundary lubricated or starved sliding contacts.

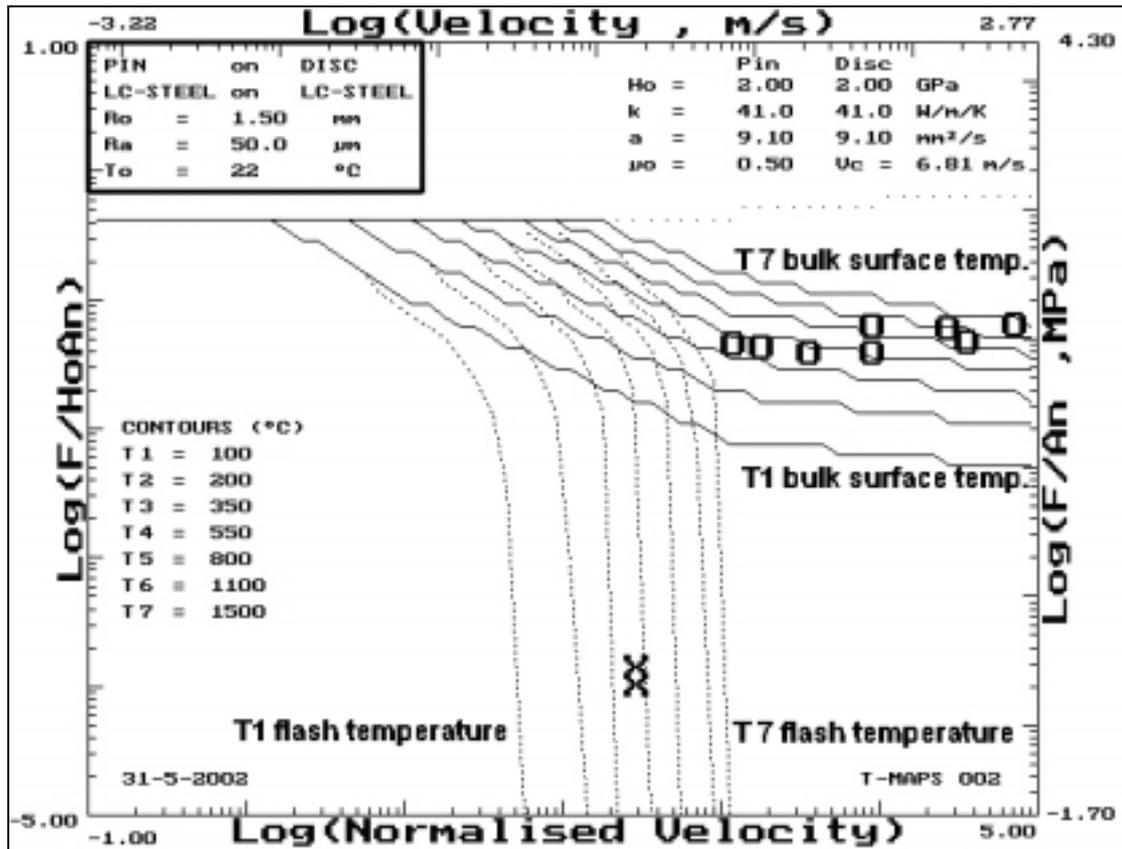


Fig. 6.2. Example of a map of calculated flash temperatures and bulk surface temperatures versus load (vertical axis) and sliding velocity (horizontal axis) for low-carbon steel sliding against itself without lubrication. The graph illustrates how the flash temperatures (temperature contours T1...T7 branching downwards from the centre of the figure) increase rapidly with increases in the sliding velocity, while the corresponding bulk surface temperatures (temperature contours T1...T7 branching to the right from the centre of the figure) require both a higher velocity and a higher load. Graph produced using T-maps 2.0 software, which is described in the Reference (Ashby et al., 1990).

### 6.2.2 Lubricant heating owing to viscous work

The temperature rise in a fluid lubricant owing to viscous work depends on the interaction of several factors like the specific heat of the lubricant and the flow rate of the fluid. The viscous work depends on the dynamic viscosity and the velocity gradient between the stationary and moving surfaces, for a sliding contact as follows:

$$F_{\eta}/A = \eta \times v/h \quad (6.2)$$

where  $F_\eta$  is the viscous force opposing the motion,  $A$  is the surface area of the sliding contact,  $\eta$  the dynamic viscosity of the fluid,  $v$  the sliding velocity and  $h$  the oil film thickness. The frictional power owing to the viscous loss is obtained as the product of the viscous force  $F_\eta$  and the sliding velocity  $v$ . Variations in the viscosity owing to temperature and pressure effects make the frictional power expression more complex. The temperature increase owing to viscous work can normally be ignored, as it is significantly smaller than the effect of frictional work in an unlubricated or boundary lubricated sliding contact.

## 7. Piston ring and skirt lubrication

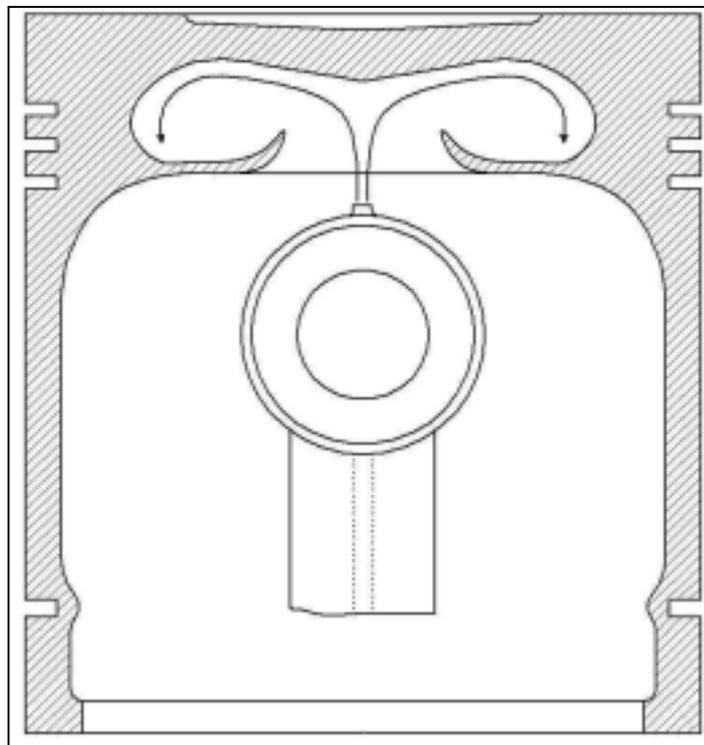
### 7.1 Oil supply

#### 7.1.1 Supply mechanisms and oil quantity

##### Piston types

Piston lubrication is in most cases connected to the cooling of the piston, although not all pistons require cooling. Pistons are divided into groups as follows: pistons with no cooling or oil-spray cooling, oil-cooled pistons with a cooling system, and crosshead engine pistons.

The first group, or pistons with no cooling or oil-spray cooling, is mostly used in high-speed engines. The bottom of the piston is, if required, cooled with an oil spray. The second group, or oil-cooled pistons with cooling system, includes pistons with cooling channels and drillings. The oil is usually supplied to the piston through the connecting rod, from the main bearings. Figure 7.1 displays a so-called shaker piston, where the upper bearing of the connecting rod has a hole from which oil spurts up into the piston. The oil is dispersed to the opposite sides of the piston's inner space as the connecting rod oscillates during the cycle.



*Fig. 7.1. "Shaker"-piston.*

## Oil supply

Oil is needed at the piston ring and liner interface to provide hydrodynamic or mixed lubrication in order to reduce friction and prevent seizure. In addition to the lubricating aspect, the oil acts as a heat carrier that transports heat from the piston and the ring-liner interface. The oil is further needed in the ring groove for preventing the ring from sticking to the groove. Oil is supplied to the piston and piston rings from the crankcase, directly or indirectly. The oil supply method usually depends on the size of the engine and on the required amount of oil. Smaller, high-speed engines use splash lubrication, as the amount of oil supplied this way is usually sufficient. Larger engines that need a higher amount of oil need to have the oil supplied up into the piston. One solution for oil supply to the piston is that the oil is fed from the main bearing or bearings to the crankshaft and further on to the connecting rod and the piston.

## Oil transport mechanisms

Oil transport mechanisms have been studied by a two-dimensional laser-induced fluorescence technique. The system enables studies on the oil distribution on piston surfaces and between the rings and the liner. Oil accumulation on the crown land has been investigated by Thirouard and co-workers. The accumulation of oil was most probably caused by the top-ring up-scraping. Two oil flow mechanisms on the second land were observed: (a) oil flow by inertia in the axial direction and (b) oil being dragged by the gas flow in the circumferential direction. The top ring up-scraping of oil was observed at engine speeds above 1 600 r/min. Owing to increased gas pressure, the piston was found to tilt towards the thrust side. The tilting piston and the ring twist caused the ring upper corner to start scraping oil from the liner wall, as there was no hydrodynamic pressure supporting the radial forces. The up-scraped oil was transported to the crown land. A second mechanism of oil transport to the crown land was also observed; oil flowed to the crown land whenever oil was present on the second land (Thirouard et al., 1998).

Thirouard and co-workers further observed that there were three possible mechanisms of oil transport to the second land: (1) The oil can be scraped to the second land in the top ring down-scraping (downstroke) or second ring up-scraping (up-stroke). With a tapered profile on the second ring, the up-scraping is impossible. (2) The oil can flow through the two upper ring grooves. Oil flows into the ring groove and is pumped out of it as a result of radial ring movement in the groove. (3) The oil can be carried through the gaps in the ring pack by gases, which flows either towards or from the combustion chamber (Thirouard et al., 1998).

### 7.1.2 Oil quality

Oil degrades due to age and contamination. The additives in the base oil fall apart and combustion particles, such as soot and wear particles, contaminate the oil. In the ring-pack area, this occurs especially in the ring and land regions. The degradation is particularly caused by high temperature and blow-by gases.

#### Friction modifiers

A lubricant consists of base oil and additives. The additives vary depending on the operational environment for which the oil is designed. The most common additives are boundary friction reducers, viscosity index modifiers, and anti-wear additives such as ZDDP.

Molybdenum dialkylthiocarbamate (MoDTC) is a base oil additive that reduces the boundary friction in a surface contact. Engine oils with and without friction-reducing additives have been investigated by Glidewell and Korcek, who conclude that friction in fully flooded conditions with MoDTC is clearly lower than with non-friction-modified oils. In starvation conditions the friction with the MoDTC-modified oil may decrease to become equal to that of non-friction modified oils. With age, the friction-reducing effect of MoDTC seems to degrade (Glidewell and Korcek, 1998).

### 7.1.3 Contaminations in the oil

#### Oil exchange interval aspects

The reason for engine oil changes is that the oil is degraded in terms of viscosity and oxidation, and that solid or liquid contaminations become mixed or dissolved into the oil. The presence of contaminations in engine oil is generally undesired, as solid contamination particles potentially cause abrasive wear and liquid contaminations may cause corrosive attacks, tribochemical wear and viscosity changes. Undesired polishing of cylinder liners in diesel engines during operation, commonly called bore polishing, may occur if corrosive species and small abrasive particles are present in the lubricating oil (Godfrey, 2000).

The contaminants in the engine oil build up over time. This disadvantage is normally maintained by replacing the contaminated oil with a new batch. Recent investigations by Bijwe and co-workers have determined the necessity for crankcase oil drainage at set intervals (Bijwe et al., 2000). The oil change intervals can be based on running hours, particularly in the case of smaller engines, or on the condition of the oil, as determined

by oil sample analyses or by the response of a particle sensor (Chambers et al., 1988, Hunt, 1993) in the oil circuit or tank. When establishing an oil analysis programme, the location and time for the sampling of used engine oil should be carefully considered, as this may have a strong influence on how representative the sample is in regard to the condition of the engine. In wet sump engines a special sampling point should be introduced between the oil pump and the oil filter. In dry sump engines, the return line can be used for sampling, provided the pressure is sufficient for allowing sampling, or a vacuum pump can be employed to assist the sampling (Fitch and Troyer, 2000). In this context, particular notice should be made to the fact that the oil at the piston rings is far more contaminated than the oil in the engine sump or oil tank (Fox et al., 1997, Dattoo and Fox, 2002).

Except for the oil oxidation products and the external contaminants (presented below) that limit the useful life of an engine oil, viscosity changes and additive depletion (Glidewell and Korcek, 1998) give reasons for oil changes. The MoDTC/ZnDTP additive system, which is powerful for reducing friction, can be consumed by oxidation, because the same additives are known to contribute to the antioxidant properties of the lubricating oil (Korcek et al., 2001). A recent addition to the analyses for evaluating the remaining useful life of an engine oil is found in the RULER™ (Remaining Useful Life Evaluation Routine) voltammetric analyses, by which the remaining concentrations of zincdialkylthiophosphate (ZDDP) and phenol/aminic antioxidants can be easily monitored (Jefferies and Ameye, 1998).

### Solid contamination particles

The detrimental effect of solid particles in the engine oil is particularly obvious with softer materials, like piston skirts and journal bearings, and in highly loaded contacts like the cam-follower contact. Crankcase oil that is contaminated by abrasive particles leads to wear of the piston skirt below the piston rings, which is indicated by a matt appearance of the surface below the rings, particularly on the thrust sides of the piston skirt. Piston rings do suffer from abrasive wear, if particles of sufficient size are present in relevant concentrations in the engine oil. Bore polishing is another undesired effect of small abrasive particles in the lubricating oil (Godfrey, 2000, Jiang and Wang, 1998), while larger particles can cause scratches in the bore.

The origin of the contamination particles may be soot from combustion, ingested dust in the shape of silica dust and similar minerals, and wear particles consisting of ferrous, copper, lead, chromium, aluminium, tin and nickel alloys (Macián et al., 2001). During the running-in period of an engine, there is a higher probability for larger wear particles than during the stationary operation of the engine (Kaisheng et al., 1998). The use of oil filters with an appropriate nominal retention rate is an effective tool for suppressing the

particle concentration in engine oil (Jiang and Wang, 1998, Jones and Eleftherakis, 1995). Exhaust gas re-circulation without soot filters has been found to increase the level of carbon particles in the lubricating oil.

The main categories of solid particles in crankcase oils are the carbon, or combustion, particles and the metallic, or wear, particles. The carbon particles can be determined by means of Fourier transform infra-red spectroscopy (FTIR), a method that is suitable for the determination of the presence of various organic compounds including reaction products (McClelland and Jones, 2001). The content of metallic particles can be quantitatively determined by atomic absorption spectrometry (ICP-AES), for example (Hunt, 1993). According to experiments by Truhan and co-workers, soot, as an abrasive medium, metallic wear particles and oxidation particles cause less abrasive wear than external silica dust particles of higher concentration (Truhan et al., 1995).

Absolute limits for the types, size groups and concentration of contaminant particles in lubricant oils cannot be established for engines in general, as the limits are individual for each specific type of engine. As a rule of thumb, in all lubricated systems the particle size should remain well below the oil film thickness in any lubricated mechanism, and this very well applies to engines. Oil filters in practical applications can have nominal retention rates in the order of 15...20  $\mu\text{m}$ .

#### Liquid or dissolved contaminations and oil oxidation products

Liquid contamination of oil can occur in a variety of ways - the fuel residues, combustion products, condensed water or lubricant oxidation products as a few examples. A feature in common to all liquid contamination of lubricating oils is that they either affect the viscosity of the oil, or/and cause corrosion of the lubricated surfaces. The effect of the acidic contaminants can be diminished by neutralisation, or an overbased oil can be used in applications where acidic combustion products can be prospected. Except for water, most liquid or dissolved contaminants are difficult to mechanically separate from lubricating oil.

Carboxylic acid and other reaction products resulting from oil oxidation can be identified and quantified by means of Fourier transform infra-red spectroscopy (FTIR). The total acid number (TAN) or the corresponding total base number (TBN), which represents the total effect of the acidic (and base) species that are present in the oil, can be determined with the ASTM D663, D664 and D974 methods (Dong et al., 2000). The water content can be determined by, for instance the Karl Fischer method (DIN 51 777).

The minute oil volume that is entrapped in the ring pack of a piston, is more contaminated than the bulk oil volume of the engine. This is a natural consequence of

the presence of fuel, water vapour and other combustion products in the combustion chamber of the engine. The situation has been made more severe by the tightened exhaust emission legislation, which calls for less oil at the cylinder liner and a longer residence time for the oil in the ring pack. The poor quality of the oil in the inter-ring region should be taken into consideration when assessing the tribological conditions under which the piston rings operate. (Fox, 1996, Moritani and Nozawa, 1999, Gamble et al., 2001).

Work by Fox and co-workers has experimentally verified that the composition of the inter-ring oil is strongly different from the oil in circulation. Their findings show that, in particular during start-up and warming-up of the engine, the lubricant is subjected to substantial dilution by fuel and condensed water, which results in a destabilisation of the lubricant into several liquid and solid phases, reduction in the base number, reduction in anti-oxidancy, changes in the viscosity properties, and improper lubrication of the ring pack (Fox, 1996, Fox et al., 1997, Dattoo and Fox, 2002).

#### **7.1.4 Coke formation, coke ring**

The formation of carbon deposits on the piston top land can lead to the build-up of a sliding shoe, which forms a contact with the cylinder liner thereby causing bore polishing. When further evolved, the bore polishing leads to a local liner wear and increased lubricating oil consumption.

In some engine applications the risk of bore polishing owing to the formation of carbon deposits is prevented by providing the upper part of the liner with an anti-polishing ring. The anti-polishing ring limits the thickness of the carbon deposits on the piston top land and thus restrains the contact between the liner and the piston top land (Amoser, 2001, Geist and Barrow, 2001). Experiences of using anti-polishing rings in large-bore diesel engines are presented in the Reference (Demmerle et al., 2001).

## **7.2 Lubrication regimes; load, speed, viscosity and counter surface effects**

The ring-pack area experiences different kinds of lubrication regimes owing to local speed, load and surface roughness variations, and to variations in the oil supply. The desired lubrication conditions are the fully flooded ones, as the wear of the surface is negligible in this regime. This is, however, nearly impossible to achieve with today's engine power demands. High combustion pressures and piston speeds exceed the optimal ones in terms of lubrication. Therefore, mostly mixed and boundary lubrication

occurs in the ring-pack area. Oil transport to the ring in mixed lubrication is more or less insufficient. For example, a high combustion pressure may disrupt the oil film. The different lubrication regimes are briefly described below:

- Fully flooded ring lubrication: The oil film covers the whole ring surface area. The load is carried solely by the oil film. These conditions typically occur when the piston ring sliding velocity is high and a low pressure acts on the back-side of the ring, as are the conditions during piston mid-stroke.
- Partially flooded ring lubrication: Only a part of the ring is lubricated with oil. In this area the load is carried by the oil film, while the rest of the load is carried by the surface asperities and gas forces. This occurs in the vicinity of the dead centres, where the speed is lower and/or the pressure on the backside of the ring is high.
- Starved ring lubrication: Oil availability on the liner is at its minimum. The oil is forced away from the ring surface area owing to insufficient oil supply and/or a strong gas pressure gradient over the ring. Increasing speed increases the oil film thickness to certain level, after which the speed is too high for the oil film to withstand.

#### Viscosity effects

The effect of the oil viscosity on the frictional behaviour of piston rings has been investigated by Durga and co-workers (Durga et al., 1998). The oil viscosity affects friction values under conditions of pure hydrodynamic lubrication when the rings are fully flooded. Higher friction values occur at higher viscosity. Suggestions are made that a slight increase in friction, which is observed at mid-stroke of the piston motion, could be partially caused by high-speed shear. The rings experience a very high contact pressure at mid-stroke, which could lead to oil starvation and thus friction increase.

#### Surface roughness and surface pattern effects

Surface roughness and textures have a considerable effect on the ring-pack friction. Sui and Ariga have investigated the influence of surface patterns on oil film thickness and ring friction. The oil film thickness of the top ring is hardly at all influenced by a change in the surface pattern and therefore the change in the surface friction is negligible. The second ring and the oil control rings, on the other hand, show differences in friction depending on the direction of the surface texture. A longitudinal pattern gives the highest friction and a transverse groove pattern gives the lowest friction. The influence of the surface pattern, or texture, on the friction occurs under conditions of mixed and boundary lubrication, where contact between sliding surfaces occurs. Sui and Ariga

conclude that, in situations where the surface roughness cannot be reduced to an optimal level, roughness orientation optimisation should be considered (Sui and Ariga, 1993).

Honing the cylinder liner with crosshatches has proved to increase the hydrodynamic action on the ring pack lubrication (Michail and Barber, 1995). Hu and co-workers investigated surface roughness and oil flow factors. They modelled the ring deflection and the contact load. As a result, they showed that the contact pattern and the distribution of the oil film thickness between the cylinder liner and the piston ring are not exactly symmetric. Oil film thickness fluctuations are caused by improper ring design, static distortion of the cylinder liner and dynamic load on the piston ring (Hu et al., 1994).

### **7.2.1 Oil film thickness calculations**

There are many theoretical models of piston ring lubrication. In common for almost all of these models is that they are based on the Reynolds equation. The Reynolds equation includes parameters of the geometry, viscosity, pressure and surface velocities. The equation can be solved for pressure distribution, load capacity, friction force and oil flow.

Oil film thickness calculations consider various types of lubricant modes. The modes are pure hydrodynamic, mixed and pure boundary lubrication; additionally elastohydrodynamic lubrication is on occasion considered as an extension of pure hydrodynamic lubrication. Different lubrication models for computer simulation have been developed accordingly.

Full hydrodynamic lubrication occurs when there is no surface asperity contact. This is possible in the mid-stroke area where the relative surface velocity is at its highest. Full hydrodynamic lubrication requires that the ring area is flooded, i.e. there is always enough of lubrication oil between the ring and the liner wall in order to prevent surface contact. Lubrication conditions at mid-stroke are normally hydrodynamic, as long as there is a sufficient amount of oil available at the leading edge of the ring. Near the dead centres, however, mixed lubrication occurs. Experimental investigations by Han and Lee have shown that the ring face is not fully lubricated, i.e. the ring is partially or totally starved. Hence, the Reynolds boundary condition cannot be used in this case. The authors have developed a new model where the inlet region is in a starved condition and the outlet region has an open-end assumption. These assumptions cause the effective width of the ring face to be 20–30 % of the whole ring width. Still, using this model, the ring face works under flooded conditions at the vicinity of the dead centres (Han and Lee, 1998). Ma and co-workers have compared two different oil availability

models, namely a fully flooded model and a flow-continuity model. The fully flooded model comprises a model, where the piston rings are considered to have a sufficient amount of oil, while the flow-continuity model comprises a model, where the oil film thickness of the preceding ring is considered as the available oil film thickness for the trailing ring. The authors conclude that only approximately 10–40 % of the ring face is covered by an oil film. Therefore a flow-continuity model should be used, rather than a fully flooded model (Ma et al., 1997a).

When the relative surface velocity decreases, the oil film separating the surfaces decreases in thickness. Asperity contact will occur if the oil film becomes thin enough. The oil film boundary value for the mixed lubrication model can be determined in many ways but it always depends on the roughness of the surfaces involved.

Boundary lubrication occurs when the surface contact is continuous. The oil film thickness has decreased to such a low value that the oil film only provides lubrication between the asperities but the load is carried by the surface peaks and not by the oil film.

In hydrodynamic lubrication, where the pressure is high, the surfaces start to elastically deform according to the pressure. During this elastohydrodynamic lubrication regime, the surfaces approach each other more than the clearance would allow, and the elasticity of the surfaces interacts with the lubricant or under the pressure of the lubricant. Elastohydrodynamic lubrication of piston rings occurs in all internal combustion engines during the highly loaded expansion stroke.

Ring-pack simulation models include a variety of features that an actual ring/liner contact comprises. Some features are excluded from certain models, as they are not considered important for the purpose for which the model has been developed. Other features are too complicated to implement, and others make the required processor time very long. Almost every hydrodynamic lubrication model iterates the oil film pressure in order to establish an equilibrium of the pressure value. The ring/liner wall is usually modelled in two dimensions, thus considering the contact to be uniform throughout the circumference. This is not the case in actual situations, as the piston experiences different normal forces on the thrust and the anti-thrust sides. The ring-groove clearances are, for example, different on the thrust and anti-thrust sides because the piston is forced against the thrust side of the liner. For piston tilting, see Fig. 5.6.

Considering the two-dimensional approach, any circumferential variations are omitted by this approach. A three-dimensional approach, on the other hand, consumes a considerable amount of computational processing time, in comparison with a two-dimensional approach. An oil film model should take into consideration factors such as

oil availability in the ring pack, lubricant properties, radial tension of the ring, gas pressures, the running face profiles of the ring, engine speed, liner and land temperatures and surface texture (Tian et al., 1996). Oil film thickness calculations and measurements by Richardson and Borman have shown that the oil film thickness of the oil control ring differs significantly from a theoretical value at the beginning of the downstroke. The measured oil film thickness is greater than the calculated. The reason is presumed to be additional oil transported from the piston skirt, and piston slap-motion. These factors prevent the oil ring from following the cylinder liner and thus causes an increase in the oil film thickness (Richardson and Borman, 1992). Oil film thickness at the oil control ring increases when the ring tangential tension is reduced. The film thickness decreases when the ring width is reduced (Seki et al., 2000).

Models differ by the way in which the oil supply to the ring/liner contact is modelled. The oil film in front of the ring can be considered as being of constant thickness. A model, in which the oil film thickness trailing the previous ring is considered as the input oil film thickness for the following ring is more realistic. However, not even this approach fully corresponds to reality, as the oil film is under additional influence of the land environment between the rings.

Akalin and Newaz have developed an axi-symmetric, hydrodynamic, mixed lubrication model in order to simulate the piston ring and cylinder liner frictional contact. Their simulation results show that the hydrodynamic lubrication regime occurs during the main part of the stroke. The friction coefficient does, however, show an increase at the top and bottom dead centres, as the mixed lubrication regime is dominant in this part of the stroke. The authors have compared the results obtained by simulation to those obtained in a test bench and found that the friction results correlate well (Akalin and Newaz, 2001). The results show that the temperature, surface roughness, and running speed are the most important parameters, as they affect the lubrication regime the most. The effect of the normal load on the friction coefficient during mixed lubrication was low.

In oil film thickness calculation certain assumptions must be made, in order to keep the model size within reasonable limits. The lubricant fluid is usually considered Newtonian, while more sophisticated lubrication models include the shear-thinning characteristics of the lubricant (Tian et al., 1996). Some lubrication models assume viscosity to be constant between the ring face and liner wall. Richardson and Borman have added a varying viscosity model in their lubrication model (Richardson and Borman, 1992).

The hydrodynamic pressure is solved from the lubrication equations, but the pressure under boundary conditions still needs to be specified. The oil pressure in front of and

trailing the ring is assumed to be equal to the gas pressure on the respective side of the ring. On the front side of the ring, the remaining oil on the liner either passes under the ring or accumulates in front of the ring. If the accumulated amount of oil decreases enough, the ring becomes starved. This means that the oil film supporting the ring has less load support than a flooded ring. The ring's front edge will no longer have an oil film between it and the liner wall. There are several ways of attacking this boundary condition in the lubrication models (Richardson and Borman, 1992).

#### Oil film simulation boundary conditions

Various boundary conditions in the lubrication simulation are used. Some models are criticised for causing inaccurate results owing to what has been assumed. The Sommerfeld condition allows both positive and negative pressure values. The Half-Sommerfeld condition sets all negative pressure values to zero, and this is designated as the cavitation zone. Furthermore, mass-conserving algorithms are used in order to take into account the effect of cavitation zones on the oil availability.

#### Bore distortion in lubrication models

Bore distortion plays a significant role in the conformability of the piston ring. The bore distortion directly affects the piston/cylinder liner blow-by, which in turn is critical for oil consumption, emissions and a sufficient lubricant supply. Therefore, it is essential to include bore distortion in a computer model. Loenne and Ziemba have suggested that the bore distortion could be described by terms of a Fourier series (Loenne, 1972, Loenne and Ziemba, 1988). This approach of bore distortion has also been used by Ma and co-workers, who assume that the distortion only occurs in the circumferential direction and not in the axial direction of the cylinder. They additionally point out that bore distortion in some way may be desirable because it can reduce the friction loss of the ring pack (Ma et al., 1997b). At the top dead centre, the combustion pressure, and by this the pressure acting on the back of the top ring, is so high that even larger bore distortions are corrected by the top ring conformability. This suggests that bore distortion might not lead to excessive blow-by. Friction simulation is discussed in greater detail in Section 8.1.2.

### **7.2.2 Oil film thickness measurements**

The lubricating oil film on each ring of the piston is closely related to the oil consumption, piston friction loss and seizure. It is practical to study the influence of individual parameters, such as the engine speed, viscosity of the oil, and the profile of the piston ring, on the lubricating oil film by using test rigs and motored engine tests.

However, in order to understand the correlations between the engine operating conditions, oil film thickness and further the physical phenomenon occurring in the lubricant-surface interaction, the oil film thickness of a particular engine needs to be measured from a firing engine. (Takiguchi et al., 2000).

Works by several authors, using test rigs and motored engines, show that a thicker oil film can be reached by increasing the engine speed or the oil viscosity, or by the decrease of the load (cylinder pressure) or temperature. However, results that do not completely follow the trends expected from the theory have been published. The degree of the influence of these factors is different and their interaction in a firing engine has an important impact on the lubricating oil film thickness. (Richardson and Borman, 1992, Dearlove and Cheng, 1995, Shenghua et al., 1996, Harigaya et al., 2000).

Lubricating oil film thickness under piston rings measured by several authors in test rigs and in motored or fired test engines are tabulated in the study by Grice and co-workers. The minimum oil film thickness presented in the table varies from 0  $\mu\text{m}$  to 12  $\mu\text{m}$  and the maximum film thickness between 2.5  $\mu\text{m}$  and 24  $\mu\text{m}$  (Grice et al., 1990).

The oil film thickness can be measured by measuring the distance between the piston rings and the cylinder with common contactless sensors like capacitive sensors, inductive sensors, eddy current sensors, or sensors based on the Hall-principle (Josef and Merker, 1998). The laser-induced fluorescence technique (LIF) has taken over the oil film thickness measurements (Frølund and Schramm, 1997). The oil film thickness measurements from a firing engine are very difficult and challenging, with high demands and limitations on the tested parameters and on the engine operation conditions. Oil film thickness measuring techniques are presented in the References (Moore, 1993, 1995, 1998, Eilts and Wachtmeister, 1993, Barrow et al., 1995, Frølund and Schramm, 1997, Josef and Merker, 1998, Nakayama et al., 1998).

Arcoumanis and co-authors have constructed a reciprocating test rig for lubrication studies with a normal force that varies alongside the applied load, the position of the piston ring sample during the stroke and the crankshaft speed. The oil film thickness measurement results show minimum values in the order of 1–2  $\mu\text{m}$  just after the TDC and BDC locations and maximum values in the order of 5–11  $\mu\text{m}$  during the mid-stroke of the piston motion (Arcoumanis et al., 1995).

A direct-injection, turbo-charged, six-cylinder, Scania DSC9 engine with a bore diameter of 115 mm and a stroke of 136 mm was used in the fired engine tests carried out by Mattsson (Mattsson, 1995). The results showed that the oil film thickness between the cylinder liner and a second piston ring, at the top reversal position of the ring, on the thrust side of the piston was from 4.5 to 5  $\mu\text{m}$  when the engine was cranked

slowly. The corresponding film thickness at 1 000 rpm and 1 500 rpm and no load was not higher than 0.5  $\mu\text{m}$ . In the conclusion of the study the author states among other things that:

- The oil film thickness increases almost linearly with speed during the compression stroke.
- The idle running conditions increase the oil film thickness for the top ring during the expansion stroke.
- The trends of a computer modelled top ring oil film thickness and the measurements showed good agreement when the speed and load changed, but the calculated values were about 3 to 6 times thicker than the measured values over the speed and load range (Mattsson, 1995).

In a previous study by Takiguchi and co-workers, the oil film thickness was measured on both the thrust and anti-thrust sides of the piston at a location of 33 mm from the top reversal point of the ring. A single-cylinder, four-stroke, naturally aspirated, indirect-injection diesel engine with a bore diameter of 72 mm and a stroke of 72 mm was used as the test engine. The results showed the following trends of the oil film thickness (Takiguchi et al., 1998):

- The tendency that the oil film thickness generally increased as the engine speed increased and/or the engine load decreased, is often reversed. It was found that the variation in the oil film thickness was greater than the variation in engine speed or the load. The oil film thickness varied markedly from stroke to stroke, and between the thrust and anti-thrust sides.
- The oil film thickness of the piston ring increased markedly in the expansion and intake strokes on the thrust side, and in the intake stroke on the anti-thrust side. This was due to the large amount of lubricating oil supplied from the piston skirt to the oil ring around the compression TDC and exhaust TDC, and on the anti-thrust side around the exhaust TDC.
- The ring oil film thickness in the piston's upward strokes was thinner than in the preceding strokes and the oil film thickness tended to become similar among individual rings. This was a result of the scarce oil supply to each ring around the BDC.
- The piston ring oil film thickness was affected to a higher degree by the amount of lubricating oil supplied than by the engine speed or load, and this concerns both the

thrust and anti-thrust sides. The amount of lubricating oil supply varied markedly with the piston's slap motion.

Later works by Takiguchi and co-workers and Seki and co-authors repeat the above listed trends and emphasise the influence of the piston slap on the ring's oil film thickness (Takiguchi et al., 2000, Seki et al., 2000). Further the authors state that:

- A probable cause for the increase in the oil film thickness of the top and second ring on both thrust and anti-thrust sides during the expansion stroke is the restrained access of the gas and thus lower pressure working on the backside of the ring than the gas pressure working on the ring's sliding surface (Takiguchi et al., 2000).
- The effect of the tangential tension of the oil control ring and the width of the ring on the oil film thickness appears mainly around the top dead centre. The oil film thickness increases as the tangential tension is reduced, which significantly affects the of oil film thickness on the scraper and compression rings. (Seki et al., 2000).

The importance of the oil film thickness measurements in fired engines is not only in the numerical values of the oil film thickness. Additionally, it is fundamental to increase the knowledge about the actual phenomena that occur in the piston assembly and influence the piston ring and cylinder liner contact and thus the piston ring lubrication.

## 8. Friction of piston rings and skirt against cylinder liner

This section covers various aspects of the presence of a coefficient of friction between piston ring and cylinder liner on the one hand, and piston skirt and cylinder liner on the other hand. The issue of friction is highly relevant, as the brake thermal efficiency ( $\eta_e$ ) of an internal combustion engine can be expressed as

$$\eta_e = \eta_i \cdot \eta_m = \eta_t \cdot \eta_r \cdot \eta_m \quad (8.1)$$

where  $\eta_i$  is the indicated efficiency,  $\eta_m$  the mechanical efficiency,  $\eta_t$  the cycle efficiency and  $\eta_r$  the relative efficiency (Maass, 1979). As the frictional work losses form a significant proportion of the total mechanical losses, the frictional losses can be regarded as highly important for the brake thermal efficiency of an engine.

Expressed in terms of the cylinder pressure of the engine, the frictional losses (friction pressure,  $p_f$ ) can be expressed as the difference between the indicated mean effective pressure ( $p_i$ ) and the brake mean effective pressure ( $p_e$ ) of the engine, as follows:

$$p_i - p_f = p_e \quad (8.2)$$

### 8.1 Friction of piston rings against cylinder liner

#### 8.1.1 Ring friction fundamentals

The sliding contact between a piston ring and a cylinder liner hosts a variety of different friction mechanisms during one working cycle of the engine. Owing to the variations in load, speed and counter surface effects, the lubrication conditions in a ring/liner contact are strongly transient, which is reflected by variations in the friction, and wear, behaviour.

The ring friction is determined by the ring load, the surface properties and the lubrication conditions as determined by the sliding velocity and the viscosity and availability of the oil. The ring load comprises the ring pre-tension and the gas forces acting on the back-side of the ring. Experiments by Takiguchi and co-workers with two-ring and three-ring pistons have shown that the number of rings influences the frictional behaviour of the ring pack, but that the total tension of the piston rings in the ring pack finally determines the friction losses (Takiguchi et al., 1996).

In terms of the commonly used Stribeck diagram, the lubrication conditions in a ring/liner contact experiences strong and rapid movements on the horizontal axis of the diagram (Taylor, 1998). As a short summary of the friction mechanisms active in a ring/liner contact, the friction mechanism active in the vicinity of the dead centres of the piston movement is a combination of boundary or mixed lubrication with an additional lubricant film squeeze effect at the dead centres, while the friction mechanism active in the mid-stroke of the piston motion is hydrodynamic lubrication (Wakuri et al., 1995, Arcoumanis et al., 1997, Durga et al., 1998, Coy, 1998). The maximum friction force, which occurs under conditions of mixed lubrication in the vicinity of the TDC, has been found to decrease with increasing oil viscosity, while the friction pressure  $p_f$ , which is strongly affected by the hydrodynamic lubrication conditions between the TDC and BDC locations, has been found to increase with an increase in the viscosity of the lubricating oil (Hamatake et al., 2001).

The oil formulation strongly affects the formation of boundary layers on the mating surfaces. Anti-wear additives like ZDDP, friction modifiers like molybdenum dialkylthiocarbamate (MoDTC) and Ca-additives that may form wear-resistant layers of  $\text{CaCO}_3$  strongly control the boundary friction conditions. Recent studies by several authors have shown that, in particular, the organomolybdenum compounds MoDTC and molybdenum dithiophosphate (MoDTP) strongly reduce the coefficient of friction under boundary lubricated sliding conditions (Glidewell and Korcek, 1998, Tung and Tseregounis, 2000, Saini et al., 2001, Hamatake et al., 2001, Korcek et al., 2001). Work by Zhang and co-workers has shown that organomolybdenum-sulphur compounds decompose to form lubricious  $\text{MoS}_2$  on metal surfaces at high temperatures (Zhang et al., 2001).

The availability of oil, in terms of either poorly lubricated or fully flooded conditions, will determine the precise friction mechanism at the dead centres and the effect of the elastohydrodynamic lubrication between the dead centres of the piston motion. In addition to the oil supply, the momentary friction mechanism depends on the load, speed, actual lubricant viscosity, and the geometry of the sliding contact.

Under conditions of lubricating oil starvation, grey cast iron provides certain reduction in the friction forces, by the lubrication effect of the graphite phase and by the oil reservoir provided by the graphite phase of the material (Glaeser, 1992).

The frictional behaviour of the piston, piston ring and cylinder liner can be expressed in several different ways, depending on the purpose of the tribological analysis. The most detailed analyses comprise friction curves, in which the (i) coefficient of friction, or (ii) the friction force is plotted with respect to either (iii) the piston stroke, or (iv) the angular position of the crankshaft. Friction curves of any of the above-mentioned types

serve for tribological analyses of the lubrication conditions, and the probability for wear, in the ring/liner or piston/liner contact. Weighted average friction measurement results, in terms of (v) the coefficient of friction, or (vi) the friction force, for the mid-stroke region of the piston movement can be useful for the assessment of variations in the lubrication conditions following from different tribological parameter combinations. Weighted average friction measurement results, in terms of (vii) the coefficient of friction or (viii) the friction force, for the entire working cycle contain less information for use in lubrication analyses but they are more useful for the assessment of the frictional power losses of the engine. For enabling calculations of the frictional power loss on the basis of a friction coefficient curve for a piston/liner pair, the normal force between the piston assembly and the cylinder liner needs to be known in detail. Weighted average friction force measurement results can furthermore be expressed as (ix) the friction pressure,  $p_f$ , which is equal to the difference between the indicated mean effective pressure and the brake mean effective pressure (see formula (8.2) above).

### 8.1.2 Friction simulation

A strict relationship prevails between oil film thickness and friction. Wakuri and co-workers note that since perfectly hydrodynamic lubrication in the ring pack cannot be ensured, the theoretical estimation of the friction should always include a mixed lubrication model with asperity contacts (Wakuri et al., 1995). Nevertheless, not all computer models include friction calculations. The friction models that have been included in the computer simulation vary essentially depending on what the author of the model considers as significant.

Computer simulation of ring performance generally includes mixed lubrication models. The interaction between the ring and liner wall becomes more and more important as the oil film thickness decreases. The oil film thickness is, when minimising the friction losses, decreased in order to make the hydrodynamic friction loss as small as possible. It is evident that surface contact will occur at thin oil films. Surface contact in turn increases the total friction losses; thus small friction loss levels require compromises between hydrodynamic and sliding contact friction. Surface characteristics of piston ring face and liner wall surfaces are of great importance for the conditions of mixed lubrication, where surface asperity contact occurs but is not dominant. An oil film exists between the two interacting surfaces, but it is not able to completely separate them.

The oil viscosity undergoes decreases when the temperature increases, and the viscosity decreases when the shear stress increases. The phenomenon is called shear thinning and is usually taken into account in computer simulation. Various models for shear thinning behaviour have been developed. The simulation model works as a hydrodynamic model

until the oil film thickness decreases to a certain value, after which the mixed lubrication model is included in the simulation. This critical value of the oil film is determined from the roughness of the interacting surfaces.

The top ring experiences a higher frictional power owing to surface contact rather than hydrodynamic action. The combustion gas acts on the back of the ring and presses it against the cylinder liner. The sliding frictional power at the top ring surface is increased at increasing load. Owing to the sealing capability of the first compression ring, the gas pressure on the back-side of the other piston ring is so low, that the frictional power is not affected by load variations (Reipert and Voigt, 2001).

Sui and Ariga have investigated the effects of the ring surface topography on the ring/liner interface friction. They developed a ring-pack friction model, which is based on the mixed lubrication concept. The simulation results were verified in a moving liner test bench. The lubrication model was extended for studies on the ring-pack friction in firing engine conditions. Results obtained from these tests indicate that up to 9 % reduction in the friction loss is possible by changing the surface pattern. The friction reduction results in an increase in the oil film thickness, which also reduces the friction near the dead centres. Furthermore, the oil ring friction seems to be most sensitive to surface roughness variations (Sui and Ariga, 1993).

Studies of two-ring pistons in spark-ignited engines have been made by Takiguchi and co-workers in order to clarify their characteristic oil film thickness, ring friction and oil consumption. The authors found out that though two-ring pistons have greater blow-by than three-ring pistons, the oil film thickness for two-ring pistons in turn becomes thinner, which leads to a very small increase in oil consumption, compared to three-ring pistons. Furthermore, the authors conclude that the piston ring force can be reduced regardless of the number of piston rings, by reducing the tension of the piston rings (Takiguchi et al., 1996).

The influence of laser-textured cylinder bore and piston ring surfaces on the friction has been studied by Ronen and co-workers. The computational models used are an accurate model that takes into account the inertia and squeeze film effects, and an approximate model that ignores these two factors. When using a computer model where only the inertia effects are ignored, there is no saving in computer time. It is shown that instantaneous clearance and friction results obtained from the approximate model may differ considerably (Ronen et al., 2001).

A model for the prediction of the engine friction has been presented by Taylor. The results include simulations for fully warmed-up conditions and cold-start conditions, where the total engine friction is investigated. According to the results, the total engine

friction immediately after a cold-start is four to five times higher than at fully warmed-up conditions (Taylor, 1997).

### **8.1.3 Measured friction forces and coefficients of friction**

Piston rings are often - and must often be - tested in motored or fired engine bench tests. Miniaturised or downscaled tests for the evaluation of the tribological properties of the ring/liner/lubricant system are less expensive than the engine tests, and different phenomena can be investigated more easily without interactions from other phenomena acting in parallel. Examples of different ways in which the friction measurement results can be expressed are mentioned in the last paragraph of Section 8.1.1.

Experimental investigations by different authors on the tribological properties of the piston ring / cylinder liner have employed a piston ring or a ring segment sliding under reciprocation against a part of a cylinder liner or a complete floating cylinder liner. The simplest test set-ups are based on commercially available oscillating wear test rigs (DIN 51 834) and pin-on-disc test equipment for general tribological test purposes, while more advanced studies have employed more application-oriented test equipment or motored or fired test engines. A crank or eccentric mechanism, or an electromagnetic actuator has been employed for achieving a horizontal or vertical reciprocating motion of a piston ring or cylinder liner specimen in the test rigs. The friction force has been measured either from the ring or from the liner. In some works, the friction graphs comprise characteristic peaks at the start and stop locations, while other authors present friction graphs that lack any peaks of significance at the start and stop locations. This difference may more strongly reflect differences in the response of the friction force measurement arrangements than in the tribological phenomena studied. (Patterson et al., 1993, Noorman et al., 2000).

Arcoumanis and co-authors have constructed a reciprocating test rig for lubrication studies with a normal force that varies alongside the applied load, the position of the piston ring sample during the stroke and the crankshaft speed. The friction curves measured with the installation show peak values just after the TDC and BDC positions of the piston ring, higher friction forces with higher velocities and higher friction forces but lower coefficients of friction with higher loads (Arcoumanis et al., 1995).

Dearlove and Cheng have measured the coefficient of friction of a chromium plated piston ring with a barrel profile oscillating against a polished and honed cast iron cylinder liner sample under lubrication with five different oils at 30°C temperature, and they report average coefficients of friction of approximately  $\mu = 0.07$  at mid-downstroke, at 200 rpm engine speed and 40...80 N normal force. At an engine speed of

400 rpm, the corresponding coefficient of friction was approximately  $\mu = 0.06$ , and at 600 rpm the value was  $\mu = 0.03$ . The authors present indications of mixed and hydrodynamic lubrication. The normal load applied on the tests was quite low, which may explain the strong reduction in the coefficient of friction when increasing the engine speed from 200 to 600 rpm (Dearlove and Cheng, 1995). A minor although noticeable increase in the friction coefficient at mid-stroke, owing to increased hydrodynamic shear loss with increasing speed, has been presented by Sui and Ariga, who have compared the friction force of a ring pack at 650 and 1 500 rpm (Sui and Ariga, 1993). The above results presented by Dearlove and Cheng seem to represent lubrication conditions corresponding to the left from the minimum value of the Stribeck curve, while the above results presented by Sui and Ariga seem to represent lubrication conditions corresponding to the right from the minimum value of the same curve.

Due to a lower oil viscosity, a lower coefficient of friction was recorded by Sui and Ariga at 93°C temperature than at 30°C temperature (Sui and Ariga, 1993), an observation which is in agreement with work by Hamatake and co-workers, who present lower coefficients of friction as associated with oils of lower viscosity (Hamatake et al., 2001).

Glidewell and Korcek have presented results obtained in reciprocating tests with a molybdenum-coated cast-iron piston ring sliding against a cast-iron cylinder liner sample. In the measurements, which were carried out with different oils at 100°C temperature, weighted average coefficients of friction in the range of  $\mu = 0.10...0.11$  were obtained with oils not containing a friction modifier, while coefficients of friction of  $\mu = 0.04...0.05$  were obtained with oils containing a molybdenum dialkylthiocarbamate (MoDTC) friction modifier. The results by Glidewell and Korcek clearly show the necessity to consider the complete formulation of a lubricating oil when analysing the results of piston ring friction measurements (Glidewell and Korcek, 1998).

In tests with pieces of chromium-plated cast-iron piston rings sliding in an oscillating mode against samples of honed cast-iron cylinder liners, lubricated with a 1 % solution of stearic acid in white oil, Galligan and co-workers have measured average coefficients of friction of approximately  $\mu = 0.095$  at 100°C temperature and approximately  $\mu = 0.125$  at 202° temperature, while sliding at the latter temperature gave rise to scuffing. When the oil was replaced by a fully formulated 15W50 motor oil, the average coefficient of friction was approximately  $\mu = 0.11$  within the same temperature interval, while sliding at approximately 248°C gave rise to scuffing (Galligan et al., 1999a, 1999b).

Friction graphs from tests with pieces of piston rings sliding in an oscillating mode against samples of cylinder liners that comprise clearly higher coefficients of friction in the vicinity of the TDC and BDC locations, or U-shaped (mostly  $\cup$ -shaped) friction graphs (Fig. 8.1), have been presented by Akalin and Newaz, Andersson, Glidewell and Korcek, Dearlove and Cheng, and Durga and co-workers. However, the authors have not very much focussed on the analysis of different lubrication regimes within the U-shaped friction graphs (Akalin and Newaz, 1998, Andersson, 2002, Glidewell and Korcek, 1998, Dearlove and Cheng, 1995, Durga et al., 1998). Durga and co-workers have reported peak values at the TDC and BDC locations in the range of  $\mu = 0.10\text{...}0.15$  and mid-stroke values in the range of  $\mu = 0.05\text{...}0.10$ , the respective values depending on the actual lubricant, surface quality and surface material (Durga et al., 1998). For a chromium-plated ring against a cast-iron cylinder liner lubricated with SAE 5W30 motor oil at room temperature, Akalin and Newaz show peak values of  $\mu = 0.12\text{...}0.15$  for the TDC and BDC positions, and lower values down to  $\mu = 0.02\text{...}0.03$  in the mid-stroke regions (Akalin and Newaz, 1998).

Tests with piston ring and cylinder liner samples performed by Tung and Tseregounis using a reciprocating rig gave average coefficients of friction of approximately  $\mu = 0.08$  with low additive oils, and values down to  $\mu = 0.03$  for oils with high Mo concentrations and high Mo/S or Mo/Zn ratios (Tung and Tseregounis, 2000).

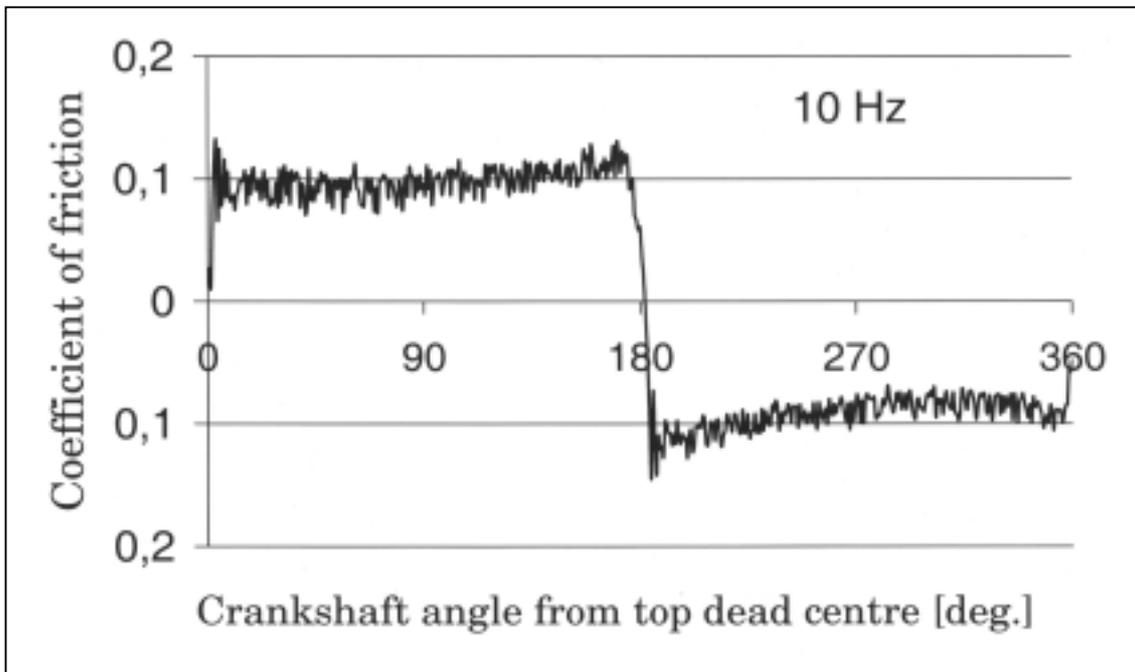


Fig. 8.1. Example of a  $\cup$ -shaped friction graph. The curve corresponds to 360 degrees of crankshaft rotation, for 10 Hz oscillating frequency of the piston ring (after Andersson, 2002).

A conclusion of the above piston ring friction measurements is that the friction curve obviously has a U-shape, with peak values in the range of  $\mu = 0.10\text{...}0.15$ , mid-stroke values in the order of  $\mu = 0.02\text{...}0.10$  and weighted average values of  $\mu = 0.04\text{...}0.11$ .

Another approach for increasing the understanding of piston ring friction has been presented by Akimoto and co-authors, who have investigated a phenomenon of half-sticking of the top piston ring by using a visual access device in the shape of a sapphire glass window in the cylinder liner. The phenomenon of half-sticking partly locks the top ring in the groove for a short period of time. The half-sticking, which arises from the presence of an excess of combustion chamber deposits in the ring groove was found to be more severe with rings with a high angle of torsion and low actual side clearance (Akimoto et al., 2000). Work by Urabe and co-workers, and Hamatake and co-workers, on the effect of exhaust gas re-circulation (EGR) has shown an increase in the wear of the piston rings and the coefficient of friction when using EGR, especially without soot filtering (Urabe et al., 1998, Hamatake et al., 2001). The increase in friction and wear is assumed to occur from excessive carbon deposits in the ring groove following the unfiltered exhaust gas re-circulation (Urabe et al., 1998).

#### **8.1.4 Effect of piston ring surface finishing and coating**

As for any tribological surface, the surface finish is of great importance for the lubrication conditions and the frictional behaviour. For decades now, piston rings have been coated, both for suppressing wear and for reducing friction. Hard chromium coatings with oil-retaining porosity or channels on piston rings are good examples of a traditional beneficial piston ring coating.

The influence of the surface finish of the piston assembly on the total frictional losses of an engine has been studied by engine tests by Wakuri and co-authors, who have shown that the frictional losses increase when run-in pistons, piston rings and cylinder liners are replaced by new components that need to be run-in (Wakuri et al., 1995). Ma and co-workers have shown that the coefficient of friction is strongly reduced during the running-in stage of the engine operation (Ma et al., 1998), and Priest and Taylor have shown the same effect in terms of fuel consumption (Priest and Taylor, 2000). The benefits of reducing the surface roughness, in particular for reducing the friction and wear under mixed lubrication conditions, are supported by earlier analyses performed by Sui and Ariga (Sui and Ariga, 1993).

Investigations by Glaeser and Gaydos, on ceramic coatings in a reciprocating test rig with a flat-on-flat geometry have show that a chromia ( $\text{Cr}_2\text{O}_3$ ) coating applied on both sliding surfaces and lubricated with a polyalphaolefine oil can give a coefficient of

friction in the range of  $\mu = 0.05 \dots 0.08$  at  $260^\circ\text{C}$  temperature. Their study was related to the development of the adiabatic diesel engine (Glaeser and Gaydos, 1993).

### **8.1.5 Effect of cylinder liner surface finishing and coating**

A plateau-honed cylinder liner surface profile, which consists of a fairly flat base surface with a network of deep scars, has been found appropriate for the lubrication of the piston assembly. The oil-retaining volume of the honed cylinder liner surface is of substantial relevance for the tribological performance of the system. For some decades, surface roughness parameters like the  $R_{sk}$  (profile skewness),  $R_{vk}$  (reduced trough depth) and  $R_a$  (arithmetic average) have been used as measures on the oil-retaining capability of the groove pattern that has been produced onto the liner surface by honing.

Durga and co-workers have investigated the effect of different surface roughness values of a cast iron cylinder liner on the coefficient of friction, and report mid-stroke values in the order of  $\mu = 0.12$  with an  $R_a$  value of  $0.3 \mu\text{m}$  and  $\mu = 0.08$  with an  $R_a$  value of  $0.07 \mu\text{m}$  with a 5W30 oil and engine speeds of 100 and 450 rpm. Plasma-sprayed coatings, even with a fairly coarse surface finish ( $R_a 0.3 \mu\text{m}$ ), gave lower coefficients of friction than the honed cast-iron cylinder liners under identical test conditions (Durga et al., 1998).

According to the work by Galligan and co-authors, the coefficient of friction in the beginning of an oscillating test is lower ( $\mu = 0.1$  against 0.13) with a highly polished cylinder liner than with a liner with a standard surface finish. However, after a certain sliding distance the coefficient of friction is at the same level irrespective of the difference in initial surface quality (Galligan et al., 1999a).

As described below, in the sections concerning scuffing, the surface quality of the cylinder liner largely determines the scuffing resistance of the cylinder and piston assembly combination, from which point of view too smooth a surface liner may be unfavourable.

From the above investigations on the influence of the surface quality of the cylinder liner on the engine friction it can be concluded that an optimum surface quality can be established, which considers the aspects of both friction reduction and scuffing suppression. Coatings on cylinder liner surfaces may offer further benefits.

### **8.1.6 Effect of cylinder liner out-of-roundness**

Deviations from cylindricity of cylinder liners cause local variations in the contact pressure between the piston rings and the cylinder. The wear of the cylinder liner and the likelihood of bore polishing and piston ring scuffing (Munro, 1976) are likely to be pronounced on areas subjected to higher contact pressure. In the case of severe roundness errors, areas of particularly low contact pressure between ring and liner are subjected to increased risk of combustion gas blow-by, particularly with stiff piston rings and high crankshaft speeds.

## **8.2 Friction of piston skirt and piston rings against the cylinder liner**

The friction between the piston skirt and the cylinder liner is controlled by the diameter clearance, the piston design and the tilting action of the piston, the piston skirt design, the surface roughness and the preconditions for lubrication (Röhrle, 1995).

Friction measurements by different authors have been carried out with piston rings in firing engines, by using the floating cylinder liner or movable bore technique. In the measurement arrangements, the cylinder liner is allowed to move axially for the minute distance that is necessary for enabling force measurements from the liner. As the corresponding normal force on a particular piston ring is known only to certain degree, a concise friction coefficient curve cannot normally be established. The friction force readings are, however, highly useful, as they express the axial loads on the rings, the frictional losses of the engine and the friction force variations in actual engines. Alternatively to employing engine tests using the floating liner technique, the piston assembly friction has been studied by motored engine tests.

Clarke, Sherrington and Smith have reported on the development of a floating liner technique and friction graphs recorded using their equipment with loading by compressed air. Friction force peaks recorded at the TDC and BDC locations of the piston motion were about twice as high as the uniform friction force at mid-stroke at an engine speed of 100 rpm. At 500 rpm engine speed the force peaks were not visible, probably owing to the mass inertia of the floating liner (Clarke et al., 1989, 1990).

Wakuri and co-workers report friction force measurement results with peak values of 570, 420 and 380 N in firing engine tests with SAE 30 oil at 70°C temperature at 500, 700 and 1000 rpm engine speed. The corresponding coefficient of friction was  $\mu = 0.08$  at the intermediate engine speed. With a SAE50 oil the friction force peak value was 300 N and with a SAE10 oil 560 N at 70°C temperature and 700 rpm engine speed. A

higher gas pressure peak (7 MPa) gave a higher friction force peak value (560 N) than did a lower (5 MPa) gas pressure peak value (310 N). These findings by Wakuri and co-workers indicate that the oil viscosity supports the formation of the oil film between piston and cylinder, that a higher velocity improves the conditions for hydrodynamic lubrication, and that a higher cylinder pressure gives higher frictional losses in the piston and cylinder contact, particularly in the ring/liner contact (Wakuri et al., 1995).

Hamatake and co-workers have studied the friction force of the piston ( $\varnothing$  105 mm) of a diesel engine with a floating liner, and they report friction peak forces of 400 N immediately after the moment of combustion, when the cylinder pressure was about 6 MPa, the crankshaft speed 1 000 rpm, and the temperature of the SAE 30 oil used was 70°C. At the same engine speed, the friction pressure  $p_f$  was about 40 kPa at 0...25 % of full load. The maximum friction force was found to decrease with increasing oil viscosity, while the friction pressure  $p_f$  was found to increase with an increase in the viscosity of the lubricating oil (Hamatake et al., 2001).

Work by Golloch and co-authors on a  $\varnothing$  128 mm diesel piston assembly has shown maximum friction forces up to 2400 N for a mean cylinder pressure of 20 bar, a maximum cylinder pressure of 170 bar and a crankshaft speed of 800 rpm (Golloch et al., 2002). Similar work by Urabe and co-workers on a  $\varnothing$  108 mm diesel piston assembly has shown maximum friction forces up to 320 N for a maximum cylinder pressure of 60 bar and a crankshaft speed of 1 200 rpm (Urabe et al., 1998). The difference in the above findings may arise from different cylinder pressures and cylinder diameters, and from the response of the two floating liner systems to the oscillating frequencies applied.

In a review on lubrication models for engines, Coy presents peak friction forces in the order of 450 N and the corresponding friction mean effective pressures in the order of 42 kPa from engine tests with 10W50 oils at 70°C temperature, 1 000 rpm engine speed and one-quarter full load (Coy, 1998).

Durga and co-workers have performed motored engine tests, in which cylinder liner coatings were found favourable. A smoother (Ra 0.07  $\mu$ m) cast-iron cylinder liner surface gave rise to lower frictional losses than a rougher liner surface (Ra 0.48  $\mu$ m) in motoring torque tests up to 6 000 rpm, but not above this speed (Durga et al., 1998).

### **8.2.1 Normal operation conditions**

Under normal operating conditions the lubrication conditions for the piston skirt and cylinder liner surfaces resemble the lubrication conditions of the ring pack, although with lower contact pressures, hence more favourable lubrication conditions, at the piston skirt.

## 8.2.2 Scuffing

Piston ring scuffing is a randomly occurring phenomenon, which consists of local micro welding, or material adhesion, between a piston ring and a cylinder liner. The scuffing phenomenon was a larger problem a few decades ago (Aue, 1976) than today. However, there is risk for an increase in scuffing problems when the oil film thickness in the ring/liner contact is reduced for environmental protection reasons (Aue, 1976, Lacey and Stockwell, 1999).

On the piston ring and cylinder liner surfaces evidence of scuffing may be found in the shape of wear scars indicating, e.g., plastic deformation, abrasive ploughing and the adhesive transfer of work hardened cast iron to a chromium-plated piston ring, and a "white layer" that indicates that the temperature has locally exceeded 750°C (Lacey and Stockwell, 1999). Metallurgical investigations by Shuster and co-workers on initial scuffing failures have shown the presence of minor iron-based particles on the face surfaces of Mo- and Cr-coated piston rings, and the presence of martensitic transformation on the cylinder liner surface (Shuster et al., 1999). One of several locations, where the risk of scuffing is large is between TDC and mid-stroke where the product  $F \times v$  reaches a maximum (Willn and Brett, 1976).

The scuffing phenomenon is normally preceded by conditions of locally starved lubrication, and the occurrence of flash temperatures, or "hot spots", in the ring/liner contact. The local temperature increase can, in turn, cause the formation of local thermal expansion of the cylinder wall material towards the bore ("thermal bump"), which instantly leads to an increase in the contact pressure in the ring/liner contact. The lubrication conditions may be poor either owing to a large surface roughness (low  $\lambda$  value) or to cylinder bore polishing (lowered  $R_{vk}$  value) that has erased the honing marks and, consequently, the oil reservoirs from the surface. The cylinder liner bore polishing can be a consequence of two- and three-body abrasive wear and plastic material flow at the ring and liner surfaces (Galligan et al., 1999a, 1999b).

Experimental investigations by Galligan and co-workers have shown that an increase in operating temperature, an increase in load and an increase in the oscillating frequency of the test samples shorten the time before scuffing occurs, while a good availability of lubricating oil increases the time before scuffing occurs (Galligan et al., 1999a, 1999b).

Excessive deposits of carbon on the top land of the piston, i.e. above the top piston ring, may contribute to the scuffing, by wiping off the oil from certain locations of the cylinder liner. This problem may be overcome by the use of a steel ring of slightly smaller diameter than the cylinder bore diameter at the top end of the cylinder liner, with the purpose of scraping off excessive carbon deposits from the piston before the

deposits form a hard layer (Amoser, 2001). Carbon deposits in the piston ring grooves and the ring sticking owing to the deposits increase the probability of scuffing (Munro, 1976).

From this point of view, it can be concluded that piston ring scuffing is likely to occur if the surface roughness is too high or too low. Poor running-in properties of piston rings, owing to surface layers that are too wear resistant, may increase the risk of scuffing. Additional protection against scuffing is achieved by engine oil additives, which promote the formation of an effective boundary lubrication film and thus suppress the scuffing phenomenon. The effect of the reaction layers in the ring/liner contact is obvious in recent studies with different oil formulations, where large differences in time-to-scuff under otherwise identical sliding conditions have been shown (Galligan et al., 1999a, 1999b). However, for each reaction layer system formed from surface-reactive additives there exists a maximum temperature, above which the reaction layer breaks down and the probability of scuffing increases dramatically. Another feature of particular interest is the volatility of the oil at the cylinder wall, which controls the evaporation of the oil or part of its constituents at high temperature, and thus controls the availability of the oil in the ring/liner contact (Lacey and Stockwell, 1999).

Durga and co-workers have reported from LS-9 Scuff Resistance Tests that scuffing occurred at a coefficient of friction of  $\mu = 0.11$  or higher, while at a coefficient of friction of  $\mu = 0.09\text{--}0.10$  scuffing did not occur (Durga et al., 1998). Recent work on the detection of piston ring scuffing has involved the use of acoustic emission detectors, by which the scuffing damage is subdivided into scuffing origin, irreversible scuffing and severe scuffing (Shuster et al., 2000).

For similar reasons as for piston rings, piston skirt scuffing may become a problem, particularly with aluminium pistons in aluminium cylinder liners. Since the introduction of aluminium cylinders in automobile engines three decades ago, iron plating of the piston skirt has been the prime solution against piston scuffing. Recently, Wang and Tung have presented the results of a scuffing resistance study on various candidate coatings for aluminium piston skirts in aluminium cylinder liners (Wang and Tung, 1999).

### **8.2.3 Piston seizure**

Piston seizure is a phenomenon that has not attained much attraction in terms of research. The seizure comprises jamming of the piston in the cylinder bore owing to strong adhesion between the mating materials. In practice, piston seizure is the result of tribological and/or thermal overloading of the piston ring pack and skirt, and can be preceded by piston scuffing. Alternatively, and particularly in the case of new engines, piston seizure can be the result of insufficient piston/liner clearance.

## 9. Wear in the piston-ring-liner system

### 9.1 Wear of piston rings

It is commonly assumed that the wear of piston rings proceeds according to a mild mechanism of mild two-body abrasive wear against the cylinder liner, expressed by the formulae presented by Archard, Archard and Hirst, Preston, Rabinowicz or Holm, while in reality the wear process is significantly more complicated (Gupta, 2001, Kauzlarich and Williams, 2001). The wear of piston rings and cylinder liners can be accelerated by three-body abrasive wear caused by minor abrasive particles in the lubricating oil. The contaminant particles causing the three-body abrasive wear can originate from the oil sump or from the combustion chamber.

In addition to the two-body and three-body abrasive wear, the overall wear rate can be tribochemically accelerated by aggressive components in the lubricant that have been entrapped in the ring zone. Aggressive combustion products are formed in particular when highly sulphuric fuels are used. Concerning most tribological applications, literature on the influence of the tribochemical wear on the overall wear of piston rings is only available to a rather limited extent. Experiences of chromium plated piston rings show that they offer good protection against wear caused by acidic combustion products (Federal-Mogul, 1998).

Under conditions of poor lubrication, strong adhesive forces between the piston rings and cylinder liner may occur, leading to piston ring scuffing that comprises high friction forces and the formation of severe wear scars on the piston, ring and cylinder surfaces.

As presented by Coy in his qualitative wear transition model, conditions of hydrodynamic lubrication at the mid-stroke region of the piston motion give rise to full film lubrication ( $\lambda > 5$ ) and zero wear, while sliding under less favourable conditions in the vicinity of the dead centres of the piston motion cause mixed lubrication ( $\lambda = 1...5$ ) and wear inversely proportional to the oil film thickness (Coy, 1998).

For low wear rates, the wear volume of piston rings can be determined by comparison of surface roughness profiles or cross section profiles before and after the tests (Shuster et al., 1999). Alternatively, the wear can be estimated from changes in relevant surface roughness parameters representing certain proportions of the piston ring face surface area (Sherrington and Mercer, 2000). For high wear rates, the wear volume can be determined from macro geometrical changes or mass loss.

In addition to sliding wear, surface degradation of piston rings can take place due to blow-by of hot gases from the combustion chamber, where the temperature of the

combustion gas is in the excess of 2 000°C. The blow-by can cause local melting or hot gas erosion damages, or burn scars, on the rings. In engines where ring deterioration owing to blow-by is likely to occur, the use of molybdenum or similar heat-resistant coatings is essential (Brauers and Neuhäuser, 1989).

### **9.1.1 Running-in wear of piston rings**

The most intensive wear of the piston ring pack and the cylinder liner normally occurs in the running-in, or break-in, stage of the engine, during which the most predominant surface profile peaks are worn off by the counter surface, and the surfaces eventually obtain improved conformity (Priest and Taylor, 2000). The abrasive wear of the piston rings decreases when the cylinder liner surface irregularities become smaller, and the wear of the cylinder liner decreases when the piston rings become smoother by running-in. Consequently, under favourable conditions this self-stabilising process leads to a decrease in the wear rate of both the piston rings and the cylinder liner (Hu et al., 1991, Ma et al., 1998).

According to experimental work presented by Henein and co-workers, the wear rate of a cylinder liner is approximately 12 times higher during the first hour of operation than during the subsequent two hours (Henein et al., 1998). Wear tests with neutron bombarded compression rings in fired engine tests including a gamma ray spectrometer have shown that the piston ring wear rate during the start-up period was up to 45 times the steady-state wear rate, and that approximately 84 % of the ring wear occurred during the first approximately 22 minutes of operation (Perrin et al., 1995).

### **9.1.2 Steady-state wear of piston rings**

In steady-state conditions, subsequently to the running-in stage of the piston rings and cylinder liner, the wear rates of the respective counter bodies is a fraction of the level experienced during the running-in. Unless thermomechanical fatigue fracturing of the piston rings or their surface layers occurs, the low level of the wear rates can continue until the components are taken out of use.

### **9.1.3 Wear of ring flank surfaces**

Owing to the relative motion between the ring flank surfaces and the grooves in the pistons, and the dynamic axial loads on the rings, wear occurs in the ring/groove contact. This phenomenon is further described under Section 9.2, in conjunction with

details on the wear of the ring grooves. The ring flank wear can be reduced by the use of coatings on the ring flank surfaces.

#### 9.1.4 Wear of oil rings

The tribology of the oil rings is not analysed separately in greater detail in this work, mainly owing to the lack of specific information available in the literature. The oil rings work under an oil film thickness that is thinner than that of the compression rings on average, owing to the high spring forces that continuously act on the oil ring, and for this reason the wear of the oil rings cannot be neglected. Some secondary wear is known to occur as the result of interactions between the different parts of compound oil rings (Federal-Mogul, 1998). Hard carbon particles from the combustion process have been observed to cause wear on the spring expanders of three-piece oil rings. Surface nitriding provides a means of preventing the wear of the oil rings (Esser, 2002).

#### 9.1.5 Simulation of wear of piston rings

The wear rate can be mathematically defined using the Archard wear equation:

$$V = kWx_s \quad (9.1)$$

Where  $k$  is the wear factor,  $W$  the normal load,  $x_s$  the sliding distance and  $V$  the worn volume. The wear factor,  $k$  is in turn a function of the surface properties, the oil and operating conditions. The wear factor has to be determined by empirical methods, for example, from bench tests.

Wear of the piston rings and the cylinder liner is perhaps the most difficult phenomenon to implement in a calculation model. Wear parameters most certainly require empirical data. The time required for simulation increases when wear models are included in the simulation software. In terms of the simulation, wear comprises less understood phenomena than friction or lubrication. Even though wear might be considered a minor factor in calculation models, it should be remembered that the wear of a piston ring alters the ring profile. Therefore, wear is a phenomenon to be included in a realistic model. Priest and Taylor have investigated piston ring wear modelling. They point out that piston ring designs with emphasis on wear resistance may be non-optimal considering the lubricational and frictional properties (Priest and Taylor, 2000).

Priest and Taylor have examined the correlation between predicted and measured wear results. They have modelled the top ring of a Caterpillar 1Y73 engine and compared the results over 120 hours of engine running. The results suggest that the wear was not equal at all locations along the circumference of the ring. Ring twist is also an important

factor to take into consideration in wear modelling, and the authors conclude that especially the top ring profile wear is considerable during the 120-hour test. Owing to the wear, the surface roughness of the rings and cylinder wall is significantly reduced during the test (Priest and Taylor, 2000).

Wear is minimal at full hydrodynamic or elastohydrodynamic (EHD) lubrication conditions, as nearly no asperity contact occurs in these conditions. At the hydrodynamic or elastohydrodynamic lubrication regimes, it is not necessary to consider surface contacts in the simulation. If the lubrication conditions change into mixed lubrication, where there is asperity contact, the wear model becomes more important, as has been shown by Coy (Coy, 1998). This, in turn, leads to variations in the ring friction. Consequently, the ring profile has to be maintained in order to keep the ring friction at a desired level. Sui and co-workers suggest that the best way to maintain the ring profile is to consider the surface topography effects at an early stage of the ring-pack design (Sui and Ariga, 1993).

With a transition wear model, Coy has investigated the wear of the top ring related to the distance from the TDC. According to this model, the wear rate is at its maximum at the TDC and decreases as the piston moves downwards. When the piston approaches the BDC, the wear rate increases again. The higher wear rate areas correspond to the areas where mixed and possibly even boundary lubrication occurs (Coy, 1998).

Piston ring profile wear has been investigated by Priest and co-workers. The lubrication model used did not determine the cyclic variations in the torsional twist angle of the ring. This feature is, however, needed for the wear model. Therefore, a ring twist pattern was assumed according to a predicted twist angle. Piston ring profile wear and lubricant degradation during a 120-hour cycle was included in the wear model. The results show good agreement between the predicted and experimental results (Priest et al., 1999).

Commercial simulation software is limited in terms of the simulation of the wear rate, as they require a wear coefficient that reflects the wear of both mating surfaces. Ring profile change owing to surface wear is ignored, the primary reason being an extensive processing time and challenging implementation of the calculation results.

## **9.2 Wear in the piston ring groove**

Wear of the parallel surfaces in piston ring grooves, commonly called ring-groove wear, occurs mainly in the top ring groove. The main reason for the wear is the combined effect of gas forces and radial motion of the ring, and the wear process is accelerated by poor lubrication and a high temperature. The reasons for the radial motion of the ring

are the cylinder distortion, the secondary movement of the piston and piston tilt allowed by the piston/cylinder clearance. Mass forces, friction forces, axial ring movement and ring rotation increase the ring groove wear. Instationary gas pressure and gas blow-by may cause radial vibrations in the ring, which accelerates the ring groove wear at the ring-groove contact areas. (Röhrle, 1995, Affenzeller and Gläser, 1996, Federal Mogul, 1998, Mollenhauer, 1997).

As the result of the wear, the lower surface of the ring groove becomes rough and rounded towards its edge, and the upper surface becomes rough (Röhrle, 1995, Affenzeller and Gläser, 1996). In addition to the deformation of the groove arising from wear, the width of the groove increases, and the side clearance between ring and groove increases.

Alternatively to wear, the ring-groove tribosystem may suffer from ring welding, as a consequence of overheating of the top ring or partial seizure of the piston owing to poor lubrication (Willcock, 1996).

In certain diesel engines for passenger cars and all advanced diesel engines for commercial vehicles, ring groove wear in aluminium pistons is prevented by a cast-in ring carrier made of austenitic Niresist cast steel or similar wear-resistant material (Mollenhauer, 1997, Röhrle, 1995).

## **9.3 Wear of the piston skirt**

### **9.3.1 Mild wear of the piston skirt**

As part of the running-in of the piston assembly, mild wear of the thrust and anti-thrust sides of the piston skirt are a normal consequence. In continuous operating conditions, the wear of the piston skirt is normally insignificant, as the hydrodynamic lubrication conditions are disturbed only by the reversal of the motion at the TDC and BDC locations. Wear of the piston skirt normally remains low, despite the hardness reduction of the piston material at the operating temperature. For improving the break-in properties of the skirt, and for providing conditions of hydrodynamic lubrication and low wear, the piston skirt and the cylinder liner must fulfil certain requirements of surface smoothness. Surface roughness values (parameter not indicated) in the order of 1.5...3  $\mu\text{m}$  are typical of the piston skirt (Röhrle, 1995). Soft coatings on the piston skirt can be applied for supporting a mild wear process during the running-in of the piston/cylinder pair.

### **9.3.2 Abrasive wear of the piston skirt**

When abrasive particles (soot, dust or metallic wear particles) enter the sliding interface between the piston skirt and the cylinder liner, severe abrasive wear scars may form on the piston skirt surface. The wear is normally accompanied by similar abrasive wear on the cylinder liner surface. Piston skirt wear is in certain cases suppressed by a harder coating on the skirt.

## **9.4 Wear of the cylinder liner**

Wear of the cylinder liner is caused to a great extent by the action of the piston rings. Practical observations and theoretical analyses correlate well in terms of the strongest wear of the cylinder liners taking place in the vicinity of the top reversal point of the top piston ring, where the thermal, chemical, erosive and abrasive conditions are the severest. A high sulphur content of the fuel can increase the proportion of tribochemical wear of the cylinder liner dramatically, particular at low cylinder surface temperatures. High wear of the cylinder liner is furthermore associated with the top reversal point of the second piston ring, and to a less extent with the bottom reversal points of the piston rings. Carbon deposits above the ring pack on the piston may significantly increase the cylinder liner wear in the TDC region (Affenzeller and Gläser, 1996).

Cylinder bore polishing, which can be subdivided into light, medium and heavy polishing, is the first occurrence of wear in a cylinder liner. A light degree of bore polishing increases the oil consumption. When the bore polishing has evolved to a stage of heavy polishing, and most of the oil-retaining honing pattern has been erased, the risk of lubrication starvation and scuffing is obvious (Dong et al., 1995).

The thermal loads cause lubricant degradation by ageing and partial evaporation. The chemical loads comprise dilution by fuel, acidic combustion products and water vapour from the combustion process. The erosive loads comprise the mechanical effect of the flushing by hot gases along the upper parts of the cylinder liner surface, and the removal of oil from the liner surface. The wear of the cylinder liner is additionally accelerated by solid carbon particles from the combustion process, and possibly by dust from the intake air that can contribute by causing abrasive wear. Wear of cylinder liners occurs as well in the mid-stroke region of the piston ring motion. The wear of the cylinder liner is higher on the anti-thrust side than on the thrust side of the liner, owing to the distribution of the thrust forces during the different cycles of the engine. (Dong et al., 1995, Priest and Taylor, 2000).

Break-in wear tests have shown that the wear of a cylinder liner is predominant at the beginning of the break-in process (Ma et al., 1998), with wear rates during the first hour of operation that are approximately 12 times those during the second or third hour of operation, and that the strongest wear occurs close to the top ring reversal point (Henein et al., 1998). According to tests by Ma and co-workers, the cylinder liner wear reaches a steady-state after 3.5 hours (Ma et al., 1998).

Terheki and co-authors have experimentally, albeit under dry running conditions, determined the wear mechanisms responsible for the degradation of cast-iron cylinder liners. The investigation showed that truncation (plastic deformation of surface profile asperities), adhesion, delamination and ploughing (abrasive two-body wear) can be present in the wear of the cylinder liners (Terheci et al., 1995). Owing to the tests being performed without lubrication, the findings are primarily representative of strongly starved sliding conditions.

For low wear rates, the wear volume of cylinder liners can be determined by comparison of surface roughness profiles or cross-section profiles before and after the tests (Ohlsson, 1996, Priest and Taylor, 2000). Alternatively, the wear can be estimated from changes in relevant surface roughness parameters or the bearing area curve (the Abbot curve, see Chapter 10) representing certain proportions of the piston ring face surface area (Pawlus, 1997, Sherrington and Mercer, 2000, Kumar et al., 2000).

Means to suppress the wear of the cylinder liner are presented in Chapters 4 and 10.

# 10. Surface technology

## 10.1 Coating deposition and surface treatment techniques

Functional coatings and surface treatments offer several possibilities to improve the sliding properties of metal surfaces. In internal combustion engines, coatings and surface treatments are commonly applied on one or several components of the piston, piston ring and cylinder liner system.

The range of coatings and surface treatments currently used in engine components in the reciprocating system covers a variety of different coating compositions and deposition techniques. Surface coating and treatment techniques are continuously developing and new groups of coatings are introduced on the market, and even more methods are being developed and experimentally evaluated. Two different groups of coating application techniques are currently being commercially used for the deposition of wear-resistant coatings on piston rings, namely thermal spraying and galvanic coating. A third, widely used surface treatment method for piston rings is surface nitriding, which is a combined hardening and coating process for producing a graded iron nitride structure on the ring surface region. In addition to this, soft coatings for friction reduction are applied. New types of coatings are being developed and it can be forecasted that they will eventually be applied in production engines.

Coated ring faces may be susceptible to edge flaking, hence precautions like edge radiusing (before or after the coating deposition), and the use of inlays or semi-inlays covering only part of the ring face is used (ISO 6621-4).

### Thermal spraying

Plasma spraying and other types of thermally sprayed coatings are mainly applied on piston rings for large-bore diesel engines. The coating is built up from a powder, which is heated and bombarded onto the piston ring surface. The temperature of the powder, and the temperature increase owing to thermal energy dissipated at the collision between the powder particles and the piston ring surface, cause the particle to partly melt and adhere to the piston ring surface. Higher energy processes include the plasma spraying, the detonation gun spraying and the high-velocity oxy-fuel (HVOF) process (Holmberg and Matthews, 1994). For thermally sprayed coatings, most often cemented carbide powders with a metallic binder phase, or single phase or composite ceramic powders are used. The exploded wire spray method is in use for the deposition of Fe-Mo coatings (Durga et al., 1998).

Molybdenum coated piston rings, with molybdenum coating on the full width of the face or as an inlay covering part of the ring face width, are used by many diesel engine manufacturers (ISO 6621-4). Coating by molybdenum is especially useful for preventing hot gas erosion deterioration of the piston ring face surfaces, which may occur due to blow-by of combustion gases from the combustion chamber (Affenzeller and Gläser, 1996, Brauers and Neuhäuser, 1989). Molybdenum is a tribomaterial used particularly in high-temperature applications (Glaeser, 1992). The resistance of the molybdenum coating against adhesive and cohesive failures, which can be determined by the standard test ASTM D633, is controlled by the coating deposition process parameters (Babu et al., 1996).

In a recent overview, Harrison describes atmospheric plasma-sprayed coatings for cylinder surfaces in series production aluminium engine blocks; good results have been obtained with Fe-C alloys that partly oxidize during the deposition, forming lubricious iron oxides like FeO and Fe<sub>3</sub>O<sub>4</sub>, and with non-corroding, sulphur resistant iron-based alloys containing chromium and molybdenum (Harrison, 2002). For the same kind of applications, promising results have been obtained with thermally sprayed titania (TiO<sub>2</sub> or TiO<sub>x</sub>) coatings on aluminium substrates (Buchmann and Gadow, 2001). An alternative for improving the tribological performance of aluminium block engines is to use coated pistons (see below).

For adiabatic diesel engines, Gaydos and co-workers have successfully studied the use of a chromia (Cr<sub>2</sub>O<sub>3</sub>) cylinder liner coating against a tungsten diselenide (WSe<sub>2</sub>) piston ring coating (Glaeser and Gaydos, 1993). For similar applications, i.e. low-heat rejection diesel engines, Haselkorn and Kelley have studied the tribological performance of ring and liner specimens coated with plasma-sprayed high-carbon-iron-molybdenum and plasma-sprayed chromia-silica coatings, and found them of potential use on cylinder liner or piston skirt sliding surfaces (Haselkorn and Kelley, 1992).

HVOF-sprayed chromium carbide coating is suggested by Shuster and co-workers as an efficient protection against scuffing and wear (Shuster et al., 1999). Work by Durga and co-authors show a reduction in the coefficient of friction when applying porous plasma-sprayed FFS (Stainless steel + Ni-BN) or M-1P (Fe-FeO-C) coatings, when comparing surfaces with equal surface roughness, and even lower coefficients of friction with a porous and honed surface (Durga et al., 1998). Experimental investigations by Ahn and co-workers on plasma-sprayed zirconia thermal barrier coatings against chromium-plated steel in unlubricated reciprocating motion at 200°C specimen temperature have shown that the wear resistance of the coatings was better than that of cast iron in the same kind of tests, partly due to wear protection obtained by a tribofilm formed from wear debris (Ahn et al., 1997). Engine tests with different advanced chromium carbide (Cr<sub>2</sub>C<sub>3</sub>) based sprayed coatings have shown the potential of this group of coatings,

particularly when molybdenum is included in the coating composition (Rastegar and Richardson, 1997). Full-scale engine testing of aluminium bronze coated cast-iron piston rings against  $\text{Cr}_3\text{C}_2$  based coatings on cylinder liners, and  $\text{Cr}_2\text{O}_3$ -based coatings on cast-iron rings and cylinder liners, have given promising results in terms of component wear and lubricant sensitivity (Miyake et al., 2001).

Plasma-sprayed or otherwise thermally sprayed layers on the face surface of a piston ring can be applied as a uniform (with minor deviations) layer or as an inlay in a groove, which means that the effective width of the coated part is less than the actual width of the ring face (Federal Mogul, 1998, Mollenhauer, 1997, ISO 6621-4).

Thermally sprayed ceramic or ceramic-metal composites need to be ground, honed, lapped or polished before the surfaces are taken into use, and the surface finishing needs to be optimised individually for each coated system (Radil, 2001).

### Electrochemical coating

Galvanic, or electrochemical, coatings are based on a donor anode, an electrolyte and an electric power source. In the coating process, metal ions from the anode move through the electrolyte to the substrate at a rate that is defined by the electric current density (Holmberg and Matthews, 1994). The most widely used galvanic coating for the wear protection of piston rings is hard chromium plating (Affenzeller and Gläser, 1996).

In electrochemical chromium plating, chromic acid ( $\text{H}_2\text{CrO}_4$ ), or the aqueous solution of chromium trioxide ( $\text{CrO}_3$ ), acts as the chromium donor, whereas an inert anode is used (Newby, 2000). In electrochemical plating, the components to be coated act as cathodes, and locations for hydrogen gas formation. Precautions in the deposition process or post treatment may be necessary for avoiding hydrogen embrittlement of the coated surface material.

Since the 1940's hard chromium coatings have often been produced as porous. Originally the porous chromium coating was developed for piston rings and aluminium cylinder bores of aircraft and diesel engines. Different mechanical and chemical techniques are applied for obtaining the porous topography. As the as-coated hard chromium surface is rather coarse, it normally requires honing, lapping, grinding or polishing, followed by cleaning before being taken into use (Newby, 2000). In practice, all hard chromium-plated piston rings are lapped or polished after the coating deposition, and during this post-treatment the ring can simultaneously be given its barrel or other profile shape. A feature typical of hard chromium coatings on piston rings is the network of minute cracks that covers the surface and provides pockets for lubricating oil. From an environmental protection point of view, the use of chromium plating on

piston rings is less desirable than the use of non-chromium-plated steel rings (Affenzeller and Gläser, 1996).

Chromium-plated piston rings have proven their durability in long-term use. Severe wear of Cr-plated piston rings may, however, occur owing to the lack of oil retention volume on the cylinder liner surface, martensitic transformation on the cylinder liner surface, or thermomechanical fatigue of the coating (Shuster et al., 1999). The abrasive wear resistance of a hard chromed piston ring can be improved by adding hard ceramic particles to the coating during the deposition of the coating (Federal Mogul, 1998). Hard chromium coatings are occasionally used for improving the wear resistance of ring grooves in steel composite piston crowns (Röhrle, 1995).

Galvanic (electrochemical) coating materials furthermore include copper (Cu) and tin (Sn) as friction-reducing coatings for piston ring face surfaces (Federal Mogul, 1998).

Improvements in the wear and scuffing resistance of aluminium pistons in overeutectic aluminium cylinder liners is obtained by ferritic iron plating of the piston skirt (Affenzeller and Gläser, 1996). Scuffing experiments by Wang and Tung with various coatings on piston skirts in aluminium cylinder liners have shown that a selectively plated Ni-W coating and suspended ceramic particulate plated Ni-P-BN can provide the same tribological benefits as an iron coating (Wang and Tung, 1999).

### Nitriding and carbonitriding

Surface nitriding or carbonitriding are processes for producing a hardened surface consisting of an iron nitride, or a harder iron carbo nitride, at the outermost region of the piston ring material, and a diffusion zone between the nitride layer and the substrate material. During the surface treatment, the steel component is exposed to a nitrogen-containing atmosphere, a molten salt bath or a plasma at high temperature (450...610°C depending on process), which causes diffusion of nitrogen (and carbon) into the steel, and reactions between the iron, nitrogen and carbon atoms. The process is terminated by cooling at a rate that gives the desired hardness and toughness properties. In molten salt bath processes, cyanides and cyanates are normally used. Gas nitriding is carried out in ammonia gas, with the addition of carbon dioxide or carbon monoxide gas for carbonitriding. Plasma nitriding uses a mixture of oxygen and nitrogen gas as the nitrogen donor, and the process is activated by electrical discharges in an insulated chamber. The reaction kinetics, the depth of diffusion of the nitrogen atoms into the steel structure and the hardness depth are controlled by the process, temperature and time. Gas nitriding requires the longest processing time, 20...100 hours, while molten salt nitriding gives the shortest time, 10 minutes to 4 hours. The thickness of the nitride layer and the thickness of the diffusion zone depend on the process and the substrate

material. The surface hardness obtained by the nitriding process and the hardness gradient in the surface region depend on the material, with 1 300 HV as an average typical of various steel and cast-iron piston ring materials (Brauers and Neuhäuser, 1989, Federal Mogul, 1998).

The wear of carbonitrided piston rings made from a specific steel has been found almost equal in high-temperature tests, irrespective of the nitriding process applied. The response of different piston ring materials to carbonitriding is obvious from the differences in their wear rates in engine tests, where carbonitrided chromium-alloyed steels were superior to carbonitrided cast iron or low-alloy steels in terms of wear resistance. In most cases, traditional chromium-plated piston rings turned out to be equally or more wear-resistant than carbonitrided piston rings in engine tests. For smaller internal combustion engines operating under lower loads, for which chromium-plated or molybdenum spray-coated rings are not required, carbonitrided rings can be used, as has been the case in certain production engines. In engine tests, the system wear of the ring flank and the ring groove were equal or larger than in the case of untreated cast iron rings. Corrosive attacks on the carbonitrided rings were obvious, and their role in the total wear of the rings may be significant, particularly in engines that are frequently used for shorter duration (Brauers and Neuhäuser, 1989).

Low-carbon steel surfaces that are aluminised before being nitrided reveal a higher hardness than nitrided plain low-carbon steel, partly due to the formation of aluminium nitride (AlN) in the surface structure (Bindumadhavan et al., 2000).

The nitriding process is subject to development efforts, as for instance oxygen-sulphur nitriding of grey cast iron has been recently studied (Baranowska, 1998).

#### Ferro-oxidising

Ferro-oxidising can be carried out on all surfaces of an otherwise uncoated piston ring (ISO 6621-4).

#### Graphite coatings

Graphite coatings are deposited on aluminium and cast iron piston skirt surfaces as a resin that contains pigments. A method called Grafal<sup>®</sup> is based on a phenolic single-stage resin with fine colloidal graphite. The adhesion of the graphite coating can be improved by applying a metallic phosphate layer onto the surface prior to the application of the coating (Röhrle, 1995, Mollenhauer, 1997). An atomised spray coating consisting of epoxy resin, boron nitride, molybdenum disulphide and graphite on the piston skirts and cylinder liners has been studied by Durga and co-workers, who

have found that the amount of boron nitride must be kept below a certain limit in order to suppress cylinder liner wear (Durga et al., 1998).

### Ion exchange reactions

Thin layers of lead (Pb) and tin (Sn) are produced onto as-machined piston skirt surfaces by ion-exchange reactions, by applying Pb and Sn salts onto the surfaces for a certain reaction time. The thin Pb and Sn layers are soft and improve the running properties at the piston skirt and cylinder liner contact, and they are commercially widely used (Röhrle, 1995).

### Phosphate coatings

A phosphate (zinc phosphate or manganese phosphate) coating can be formed onto piston rings by chemical reaction with phosphate crystals. The phosphate layer is softer than the piston ring material and supports effective running-in, and the phosphate treatment can reduce the formation of wear scars (Federal Mogul, 1998, ISO 6621-4).

### Anodising

Anodising is the name given to for a controlled electrochemical process for obtaining a dense aluminium oxide grading on the surface of an aluminium component. Thicker layers are obtained by a hard anodising process. Anodising has been studied with reference to its use on piston skirts against aluminium cylinder liners, but turned out to be a less successful solution (Wang and Tung, 1999). Hard anodising is occasionally applied on piston tops for thermal protection reasons (Röhrle, 1995).

### Ion implantation

A procedure to expose a metal surface to accelerated metal ions, like nitrogen or titanium, which diffuse into a certain depth of the material, is called ion implantation. Ion implantation increases the hardness of metals without significantly affecting the microgeometry of the surface, which has been proven successful in several applications. Studies by Qui et al. have, however, shown by pin-on-disc and reciprocating tests that nitrogen ion implantation alone is insufficient for providing improvements in the tribological performance of aluminium piston materials (Qiu et al., 1996).

### Vacuum methods

The novel surface coating techniques that have potential for piston rings can be found among the group of hard diamond-like carbon coatings, titanium nitride and chromium

nitride coatings and other types of PVD and CVD coatings produced under partial vacuum.

The diamond-like hard carbon coatings (DLC) have shown beneficial friction and wear properties in tribological investigations (Ronkainen, 2001). By applying different coating deposition parameters, the amount of, for instance,  $sp^2$  (graphite) and  $sp^3$  (diamond) bonds and hydrogen in the DLC structure can be controlled, which strongly affects the tribological properties of the coating. This group of coatings offers potential for use as coatings on piston rings. Scuffing experiments by Wang and Tung with various coatings on piston skirts for aluminium cylinder liners have shown that a PVD DLC coating with a silicon interlayer can provide lower friction but causes more wear than current iron coatings (Wang and Tung, 1999). Miniature tests with DLC-coated piston ring samples in an oscillating rig have shown that this group of coatings can offer wear and friction reductions, particularly when the DLC coating is doped with appropriate metal ions (Arps et al., 1996). The tribological benefits of applying a metal-containing a-C:HMe hard carbon coating on pistons for hydraulic cylinders, for reducing the wear and the friction, have been shown by Vetter and Nevoigt (Vetter and Nevoigt, 1999). The implementation of the latter results into internal combustion engines will require that the results of complementary investigations are taken into consideration.

The benefits of multi-layered titanium - titanium nitride (Ti/TiN) compound coatings in comparison with electroplated Cr coatings and phosphated ring surfaces have been demonstrated by Zhuo and co-workers by model experiments with piston ring segments and cylinder liner segments, which all revealed less wear when coated with the Ti/TiN coatings (Zhuo et al., 2000).

Chromium nitride ( $CrN$  and  $Cr_2N$ ) coatings on piston rings have been studied in model wear tests by Broszeit, Friedrich and co-workers, who have found that these coatings reduce the wear to about one-tenth of the wear of electroplated chromium, while the coefficient of friction was similar for the  $Cr_xN$  and the Cr coatings. The necessity to re-design the entire tribosystem with reference to the thin hard coatings is pointed out by the authors (Friedrich et al., 1997, Broszeit et al., 1999). Haselkorn and Kelley have studied the tribological performance of ring and liner specimens coated with low-temperature arc vapour deposited CrN coatings and found the coatings of potential use on cylinder liner or piston skirt sliding surfaces (Haselkorn and Kelley, 1992).

## 10.2 Surface roughness

Deviations from absolute mathematical geometry can be expressed as form deviations, surface waviness and surface roughness. The different aspects refer to the magnitude of the wavelength of the deviations from absolute form, with form errors representing the longest wavelength and surface roughness the shortest wavelengths of the deviation (Ohlsson, 1996). Each of the above parameters has an influence on the tribological performance of the piston/cylinder couple. Form errors, surface waviness and surface roughness of pistons and cylinder liners (Lenhof and Zwein, 2002) can be metrologically determined from roundness and surface roughness measurements. For piston rings, which are elastic, the primary need is to determine the surface roughness.

The surface roughness of piston rings, piston skirts and cylinder liners has been taken into particular consideration in the development of surface roughness measurement techniques and the presentation of measurement results as surface roughness parameters.

### 10.2.1 Surface roughness basics

Traditionally, the surface roughness measurements have been carried out along lines, resulting in 2-dimensional surface roughness parameters. More developed equipment nowadays offers an option to perform surface roughness measurements on a surface, for obtaining 3-dimensional surface roughness parameters. For distinguishing between 2-dimensional and 3-dimensional surface roughness parameters, the former ones are denominated  $R_{\text{subscript}}$  and the latter ones  $S_{\text{subscript}}$ .

Surface roughness parameters for common use are the arithmetic average surface roughness ( $R_a$  or  $S_a$ ) and the profile skewness ( $R_{sk}$  or  $S_{sk}$ ). For piston, ring and cylinder liner surfaces, certain surface texture height parameters were developed some decades ago, and new parameters are continuously being taken into use. The surface texture height parameters, which are parameter presentations of the information in bearing area curve (the Abbot curve), are statistically representative values with the following definitions (compare with Fig. 10.1) (Ohlsson, 1996, Shuster et al., 1999):

$R_{pk}$  or  $S_{pk}$ , reduced peak height, or the surface profile peaks that are erased by running-in.

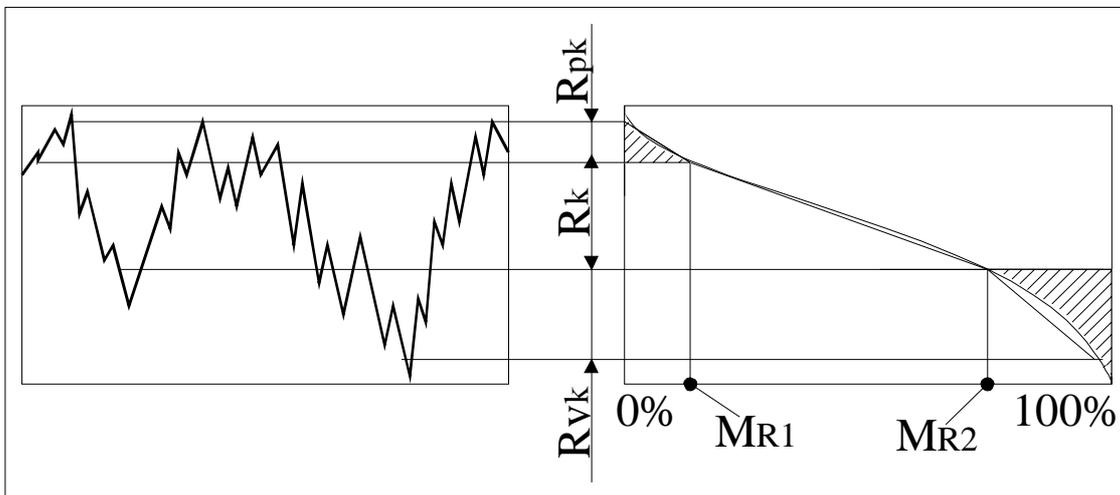
$R_k$  or  $S_k$ , core roughness depth, or the long-term running surface that determines the life time of the component.

$Rvk$  or  $Svk$ , reduced trough depth, or the oil retaining capability of the groove pattern of a honed surface or similar.

MR1 (or  $S_{r1}$ ) and MR2 (or  $S_{r2}$ ), the material components (in %) determined for the line of intersection coinciding with the upper and lower limits of the roughness core profile.

$Rvk/Rk$ , this ratio expresses the plateauess of the cylinder liner texture; a plateau surface has a  $Rvk/Rk$  ratio of about three.

The surface roughness can be determined by a variety of methods and equipment. To some extent each method and type of equipment gives its characteristic features to the surface roughness parameters, which means that the results are only partly inter-comparable (Ohlsson, 1996).



*Fig. 10.1. Graphical representation of the surface texture height parameters for a 2-dimensional measurement.*

### 10.2.2 Topography and texture

The magnitude of the surface roughness, which can be expressed by means of surface roughness parameters, as well as the texture, and which indicates how the surface irregularities are repeated on the surface, is of fundamental importance to the tribological performance of, for instance, a piston ring and cylinder liner contact. The mechanically textured cylinder liner pattern, the so-called plateau-honed cylinder, is a good example of improved tribological behaviour by surface texturing. For the same reason, chromium-plated piston rings can be lapped subsequently to the deposition of the coating. The texture formed by a network of oil channels on certain type of chromium-plated piston rings is similarly beneficial as the honing pattern on a cylinder

liner (Federal-Mogul, 1998). Recent experiments by Galligan and co-workers have again shown the role of an optimal cross-honed texture with oil retention grooves on the cylinder liner for the prevention of piston ring scuffing (Galligan et al., 1999a, 1999b).

### **10.2.3 Textured surfaces**

Post-processing of coated (or uncoated) piston ring surfaces has recently been developed by the introduction of the laser texturing technique, which allows the production of any pattern of cavities onto a piston ring or other sliding surface (Ronen et al., 2001, Steinhoff et al., 2001).

## 11. Summary

The piston of an internal combustion engine is the first component of the mechanism converting the chemical energy of the fuel into mechanical work. The heat produced in the combustion of the fuel, a large part of which is conducted to the cylinder liner surface by the piston ring pack, reduces the hardness and wear resistance of the piston and ring materials and causes oxidation and evaporation of the oil on the upper cylinder walls. Water vapour, acidic combustion products, carbon deposits and particles originating from the combustion process contribute to the wear of the piston, rings and cylinder liner. When the combustion products and wear particles from the piston ring area are mixed into the lubricating oil on the piston rings, an oil composition with poor lubricating properties is formed on the piston rings.

The piston acts as a seal between the combustion chamber and the crankcase, and is consequently subjected to gas pressure differences with strong variations. The gas pressure gradient across the piston is utilised for increasing the contact pressure between the first, and partly the second, compression ring against the cylinder liner surface, by allowing the cylinder pressure to act on the back-side of the rings. The spring loads of the rings are responsible for the rest of the contact pressure. The oil ring is normally loaded by a stronger spring force and operates under thinner oil films than the compression rings on average. Limited blow-by leakage of combustion gas from the combustion chamber through the piston ring pack to the crankcase is allowed, for ring clearance reasons and for maintaining the gas loading of the rings. The blow-by leakage may, however, disturb the oil film formation in the ring/liner contact, and under unfavourable conditions the blow-by may rise to a level at which a hot gas erosion damage occurs on the piston ring.

Lubrication conditions at ring/liner contacts strongly depend on the oil quality, the oil transport, interaction of nearby rings, surface materials, surface quality and texture in terms of honing, sliding velocity and transients, ring radial force and transients, and the contact geometry and its variations due to ring twist, axial and radial motion of the ring and cylinder bore deformation. For advanced computer simulation of the lubrication conditions of the ring/liner contact, most of the above parameters can nowadays be taken into consideration, with the interactions of the complete piston ring pack and oil starvation issues as recent and highly relevant features. Measurements of the oil thickness in the ring/liner contact provide a valuable tool for the tribological analyses of the ring-pack lubrication conditions. The thinnest oil films occur in the vicinity of the dead centres, in particular the TDC. As oil entrains into the exhaust gases and thus increases the emissions, the tightening exhaust emission legislation causes a general pressure to reduce the oil film thickness in the ring/liner contact.

The coefficient of friction and the friction force between the respective components of the piston assembly against the cylinder liner reflect the energy consumption of the oscillating components, indicate the probability for wear of the sliding surfaces, and reflect the lubrication conditions at the respective position of the piston. In the vicinity of the reversal points of the piston motion, where the sliding velocity is at its lowest, the coefficient of friction reaches its maximum values, particularly at TDC locations where the combustion pressure rises to its maximum level. Owing to the above preconditions, the ring and liner contact operates under conditions of boundary or mixed lubrication in the vicinity of the dead centres and hydrodynamic lubrication at the mid-stroke regions of the piston motion. Soft coatings, with the mission of reducing the friction under conditions of boundary lubrication, are applied onto the piston skirts and occasionally onto rings for reducing friction during the running-in stage.

After an initial phase of running-in, the wear of the sliding surfaces of a piston assembly and a cylinder liner sets to a steady level of mild wear. The wear mechanism responsible for the running-in wear is a two-body abrasion, as the result of surface asperity interactions in the sliding interface. Most of the wear takes place in the vicinity of the top dead centre and the bottom dead centre of the piston motion, as indicated by locations of intensive cylinder liner wear. Under misfortunate conditions, for example as the result of cylinder bore polishing that has erased the honing pattern from the cylinder liner, adhesive wear between ring and cylinder liner may occur and result in piston ring scuffing. Severely contaminated lubricating oil has the potential to cause three-body abrasive wear of the sliding counter surfaces. Hard piston ring coatings, like chromium plating, thermally sprayed molybdenum or surface nitriding, are currently applied for protecting the piston rings against abrasive wear, damage by combustion gas erosion and fatigue damages.

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Abstract The tribological considerations in the contacts formed by the piston skirt, piston rings and cylinder liner have attracted much attention over several decades, not least indicated by the large number of articles published on this topic in recent years. Recent studies include modelling, miniaturised experimental work and full-scale engine testing.  This literature survey, covering over 150 references, aims to shed new light on the tribological issues related to the piston assembly. The work is intended as a compact reference volume for internal combustion engines in general, with particular emphasis on diesel engines.  Central topics discussed in this work are the basic functions of the piston and the piston rings, the design and the materials of the components, mechanical and thermal loads on the rings, the contact pressure between ring and liner, the sealing action, blow-by leakage, hot gas erosion damage, exhaust emissions, the lubrication conditions and the influence of combustion products, the coefficient of friction and the friction force, the wear of the sliding surfaces and surface technology for wear reduction.			
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