Czech Technical University in Prague

Faculty of Mechanical Engineering

Dissertation thesis

New possibilities for gearwheels of automotive gearboxes

Ing. Ondřej Miláček

Doktorský studijní program: Strojní inženýrství

Studijní obor: Dopravní stroje a zařízení

Supervisor

doc. Dr. Ing. Gabriela Achtenová

2021 Prague

Anotace

Tato práce se zabývá náhradou sériového soukolí v automobilní převodovce náhradním soukolím vyrobeného ze slinutých práškových kovů (PM – Powder Metal) a má dvě hlavní části. V rámci první části byla vyrobena a zkoušena PM kola s původní (symetrickou) geometrií a speciální tepelnou úpravou – HIP (Hot Isostatic Pressing). V rámci druhé části byla navržena, vyrobena a zkoušena PM kola s asymetrickým profilem. Tato kola měla speciální povrchovou úpravu, a sice válcování. Aby bylo možné navrhnout ozubení s asymetrickým profilem, musel být pro tento účel vyvinut speciální program. Zkoušky těchto PM soukolí byly provedeny v laboratořích ČVUT v Praze. Jejich výsledky jsou také popsány v této práci.

Abstract

This thesis deals with an application of the gearwheels made of PM (Powder Metal) in automotive gearbox and has two main parts. In the first one there were manufactured and tested PM gearsets with an original (symmetric) geometry and special heat treatment (Hot Isostatic Pressing - HIP). In the second part there were designed, manufactured and tested PM gears with an asymmetric profile. These gears had also special surface treatment – rolling. To be able to design an asymmetric gearing profile a specific software had to be developed. Endurance tests of these PM gearwheels were performed in laboratories of CTU in Prague. Their results are described in this thesis.

Content

List of Symbols

Latin alphabet symbols

symbol	**meaning**	**unit**
A	asymmetry ratio	[-]
a_n	nominal center distance	[mm]
a_w	working center distance	[mm]

b	facewidth	[mm]
c_a	tip -root clearance	[mm]
c_a	profile crowning magnitude	[μm]
c_b	lead crowning magnitude	[μm]
D	ball diameter used for the measurement over balls	[mm]
d	reference diameter	[mm]
d_a	addendum (tip) diameter	[mm]
d_g	balls centers distance (diameter)	[mm]
d_w	working diameter	[mm]
h_{aP0}	addendum height of the rack (tool)	[mm]
E	Young's modulus of elasticity	[MPa]
F	force	[N]
$f_{H\alpha}$	profile slope deviation	[μm]
$f_{H\beta}$	helix slope deviation	[μm]
i	number of teeth to be measured (spaned)	[-]
j_n	normal backlash	[mm]
j_t	transverse backlash	[mm]
$k_{H\ FEM}$	recalculation coefficient for Hertzian stress	[-]
$k_{F\ FEM}$	recalculation coefficient for bending stress	[-]
M	measurement over balls	[mm]
m_n	normal module	[mm]
m_t	transverse module	[mm]
p_{rP0}	tool's protuberance magnitude	[mm]
p_{t0}	transverse pitch at tool's reference plane	[mm]
p_t	transverse pitch at reference diameter	[mm]
p_{tw}	transverse pitch at working diameter	[mm]
r	reference radius	[mm]
r_w	working pitch radius	[mm]
s_{n0} (s_{P0})	tool thickness at its reference plane	[mm]
s_{na}	normal tooth thickness at its addendum	[mm]
s_{ta}	transverse tooth thickness at its addendum	[mm]
s_{tw}	transverse tooth thickness at working diameter	[mm]

s_{t0}	transverse tooth thickness at tool's reference plane	[mm]
x	profile shift coefficient	[-]
z	number of teeth	[-]

Greek Alphabet Symbols

symbol	meaning	unit
α_{gc}	touching angle between the ball and coast flank	[°]
α_{gd}	touching angle between the ball and drive flank	[°]
α_{prP0}	protuberance angle	[°]
α_{KP0}	tip chamfer ramp angle	[°]
$\alpha_n(\alpha_{P0})$	normal profile (pressure) angle	[°]
α_t	transverse profile (pressure) angle	[°]
α_{tw}	working transverse pressure angle	[°]
β	helix angle on reference diameter	[°]
β_w	helix angle on working diameter	[°]
ε_{ball}	angle between tooth axis and ball axis	[°]
ε_{α}	transverse contact ratio	[-]
ε_{β}	axial contact ratio	[-]
ε_{χ}	total contact ratio	[-]
η	efficiency	[-]
ν	Poisson's ratio	[-]
$\nu_{c,d}$	involute's roll angle – coast,drive	[°]
π	Ludolph's Constant	[-]
ρ	osculation radius	[mm]
ρ_{aP0}	addendum radius of the rack (tool)	[mm]
σ	stress	[MPa]
σ_F	bending stress	[MPa]
σ_H	Hertzian stress	[MPa]

Subscripts

symbol	meaning
a	addendum (tip)
coast	relating to the coast side
drive	relating to the drive side
H	Hertzian
F	bending
n	normal
t	transverse
w	relating to working plane(diameter)
0	relating to the tool
1	relating to gear 1 (power input, pinion)
2	relating to gear 2(power output)
Σ	summative

Abbreviations

symbol	meaning
AC	alternating current
CNC	computer numerical control
CTD	constant torque device
CTU	Czech Technical University
CV	constant velocity
DIN	Deutsches Institut für Normung
EDM	electrical discharge machining
EM	electromotor
FEM	finite element method
FFT	fast Fourier transform
FPGA	field programmable gate array
FZG	Forschungsstelle für Zahnräder und Getriebebau
HCR	high contact ratio

HIP	hot isostatic pressing
ICE	internal combustion engine
ISO	International Organization for Standardization
LOM	light optical microscope
MBS	multi body system
PC	personal computer
PGS	planetary gear set
PM	pretensioning mechanism
PM	powder metal
PTU	planetary torque unit
RMS	root mean square

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1 Introduction

Involute gearing has been produced for more than hundred years. Therefore, there have been developed many sophisticated methods and pieces of software for its designing and dimensioning including loading conditions within its lifetime. The description of its geometry is quite demanding and the easiest type of it is the symmetrical one, which is also standardized and commonly used. Till now, the technology (geometry, material, heat treatment, etc.) has been very well fine-tuned. The way to the improvement (e.g. durability lengthening) is either in the geometry optimization or in gearwheels material.

In this thesis is described the current state of the art in the field of gears production. Then, testing methods of gears and improvements of the test bench used within this thesis are described. Furthermore, follows the description of the geometry of the involute gearing with an asymmetric profile and the software for its designing. The method of the strength computation of asymmetric gears is described too. Then the production process, testing and test results of produced gears of the gearset for automotive gearbox manufactured by the PM (Powder Metal) technology is described.

2 Current State of the Art

2.1 Automotive Gearbox Design

The design of automotive gearbox must fulfill many specific requirements. The most important of them is its low weight and for the case of the transversal position also its length. The weight is in automotive industry crucial, because it influences passive resistances of the car and thus also the power demands and final fuel consumption. Furthermore, it must allow shifting. For this reason, the architecture is very specific. There is a requirement to design all parts as light as possible. The shafts are very long and with small diameter, the gearbox casting wall is also very thin, mainly made of aluminum alloy. The consequence of this is that high deformations occur in the mesh and accurate tooth flank modifications are necessary. In addition, very important demands are also the limited build-in space and gearbox noise reduction.

2.2 Gearing Manufacturing Technology

2.2.1 Traditional Gearing Manufacturing Technologies

The gearing manufacturing process can be performed by more methods. The most commonly used method is the milling. This technology is used as the initial phase to get the rough gearing shape. After the milling process the gear is commonly performed the heat treatment and subsequently the final gear's geometry (including tooth flank modifications) is reached by grinding. It is depicted in [3], [4].

. Next possible way how to produce a gear is the **non-chipping** technology. As an example can be mentioned the EDM (Electrical Discharge Machining) technology – wire cutting.

2.2.2 Rolling Technology Description

Rolling technology is widely used e.g. for thread manufacturing. In the branch of drivetrain, it is commonly used for involute splines manufacturing on the shafts in the gearbox or on half axles and flanges. The advantage of this technology compared to standard chipping production methods (milling, turning, grinding, etc.) is in:

- short production time
- better (compact, smooth) surface quality
- improved mechanical properties (tensile strength) thanks to cold forming
- no waste material

Similar situation as while rolling of involute splines arises while gearing rolling. Only the dies profiles are changed accordingly to the desired final shape. An example of the helical gearing rolling is depicted in Figure 2-1.

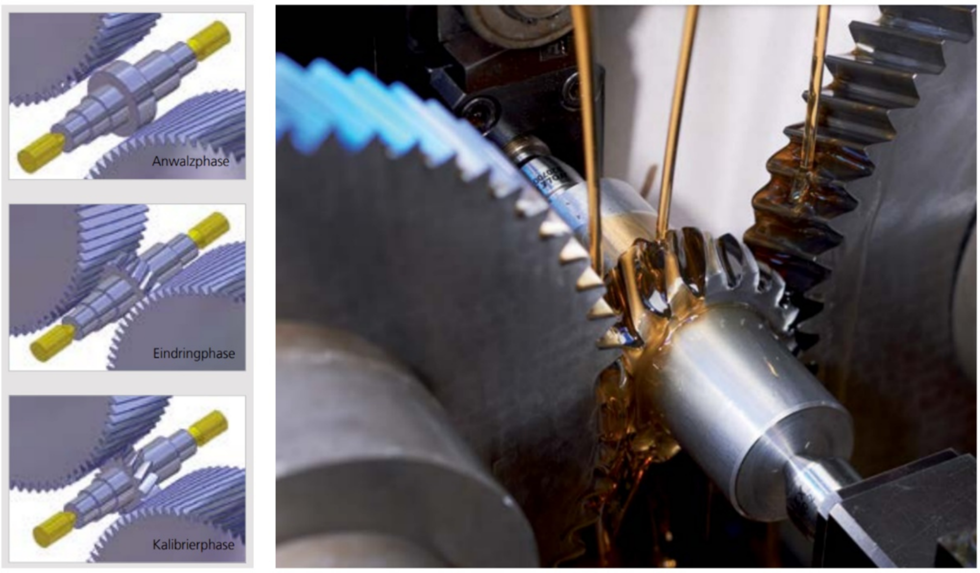

Figure 2-1: Model (left) and real appearance (right) of the helical gearing rolling [7].

2.2.3 Gearing Manufacturing Technology Research Conclusion

In the previous chapters were briefly described possible production technologies used in the gearing field. In automotive (mass production) is standardly used milling and after heat treatment grinding or honing as final operation. The rolling technology is commonly used for involute splines (with small modules) on shafts. It would be interesting to apply this technology also for automotive gearset, especially from the point of view of the base material properties improvement thanks to cold forming. Furthermore, within this thesis there was found a partner, who is worldwide known in the field of the rolling technology. For this reason, the **rolling technology** was chosen as the production technology of investigated gears within this thesis.

2.3 Gears Materials

2.3.1 Standardly Used Materials of Gears

Gears have been produced for many years and thus also many types of materials have been used. The material is always depending on gearset loading conditions. The most common materials in automotive are forged or alloyed steels with a final heat treatment – e.g. carburizing or nitriding – as a hardening treatment. To improve basic material's properties, special coatings can be used, e.g. PVD (Physical Vapor Deposition) technology.

Very interesting technology seems to be the PM (Powder Metal) technology. Its positives are very remarkable in the mass production. Furthermore, next advantage of this technology is especially in the product's homogeneity and isotropy. The detailed description of this technology branch follows.

2.3.2 Powder Metal (PM) Materials Description

"While a crude form of iron powder metallurgy existed in Egypt as early as 3000 B.C., and the ancient Incas made jewelry and other artifacts from precious metal powders, mass manufacturing of P/M products did not begin until the mid- or late-nineteenth century. At this time, powder metallurgy was used to produce copper coins and medallions, platinum ingots, lead printing type, and tungsten wires, the primary material for light bulb filaments. By the 1920s the tips of tungsten carbide cutting tools and nonferrous bushings were being produced. Self-lubricating bearings and metallic filters were other early products." [8].

Generally, the powder metal technology is processed in three basic steps: powder preparation (mixing), die compaction and sintering. In following chapters these operations will be briefly described.

The process starts with the used powder mixing, when a base powder is mixed with alloying elements to reach desired material properties. Next step is the kneading together with the polymers and a wax to assure the shape stability after the pressing or molding. The main pressing operation "die compaction" follows. After the compaction the shape of the pressed part is fixed only by additional material and is in so called "green state" and the pressing product is usually called "**green part**". The strength of this product is very low and is used only for the manipulation. It can be broken only by hand! This compaction process is schematically depicted in Figure 2-2.

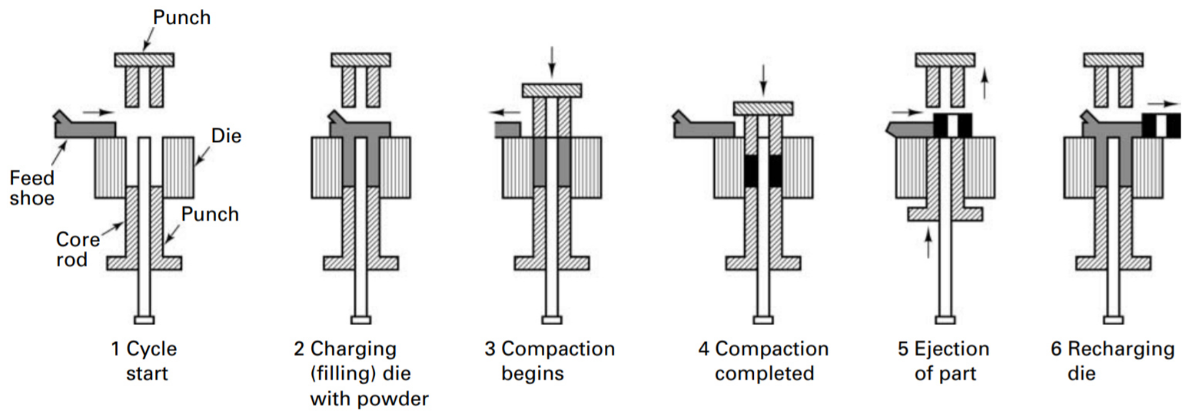

Figure 2-2: Typical compaction sequence showing the functions of the feed shoe, die, core rod, and upper and lower punches. Loose powder is shaded; compacted powder is solid black [9].

Next operations are performed in a sintering furnace with controlled atmosphere, to remove the oxygen and prevent the oxidation of the metal parts in the green part. Very often some inert gasses are used, e.g. argon. At the beginning in the low temperature phase the green part is slightly preheated to allow the debinding operation, when almost all additional binding material is removed (melted). Subsequently the high temperature sintering follows. In this phase the powder particles, which are still only touching each other, are sintered. This means that these powder particles are firmly connected and the product gains ultimate strength. During this operation the volume of the sintered part is reduced. This fact must be taken into account while the designing of the pressing die geometry. The schematic structure of the PM material in certain stages of the sintering process in the furnace is depicted in Figure 2-3.

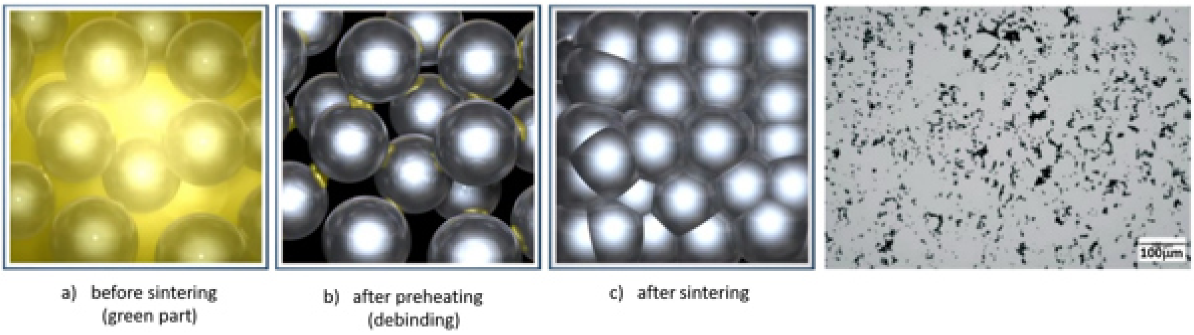

Figure 2-3: Schematic structure of the PM material in certain stages of the sintering process (left) [10], typical structure (porosity) in a metallographic cut of PM part [11].

The PM technology can be very productive, especially in the mass production of parts with more complicated design, which could be standardly produced only by CNC milling from a full material. Next very important advantage of this technology in comparison to wrought steel is the green part's homogeneity and isotropy.

On the other hand, very important disadvantage is unfortunately a natural property of all PM products. It is the porosity. A typical example of the material structure is depicted in Figure 2-3 (right).

If the part made by PM technology is loaded by tensile stress (e.g. while bending), the surface porosity can help to a crack initiation. This issue can be eliminated by additional technology treatment (densification), which will be described in following chapters.

2.3.3 Hot Isostatic Pressing (HIP) Technology Description

The biggest problem of the PM technology is the natural porosity. One of possible ways to this issue elimination, is the usage of the HIP technology.

HIP technology is an additional process which improves product's mechanical properties. It reduces the porosity and simultaneously increases the part density. It is a combination of application of high temperature and pressure. This is standardly done in high pressure containment vessel. The gas inside the vessel is an inert one, to avoid chemical interaction with densified parts. After closing and filling the vessel with the gas it is heated. This causes also increasing of the gas pressure, which is applied to parts from all directions. For this reason, this procedure is called "**Hot Isostatic Pressing**".

In PM technology branch the standard process is the PM part compaction and afterwards its sintering in the furnace. The HIP process is only an additional option to improve its mechanical properties. An example of this technology application is depicted in Figure 2-4.

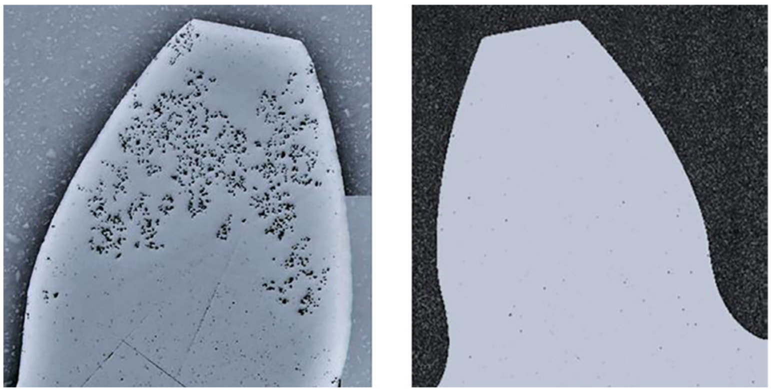

Figure 2-4: Illustration of HIP:ing with still open porosity (left) and closed porosity (right) [12].

Besides density increase HIP process also increases material homogeneity, reduces internal voids and additional internal stresses caused by technology, e.g. casting, forging, welding, machining etc.. One of many descriptions of this process is described in [12].

2.3.4 Application of PM gears in automotive gearbox

The PM technology has already been applicated in automotive gearbox. The paper [34] describes the application of PM gears made by pressing, sintering and surface densification using transverse rolling from the point of view of the geometry quality. "It is shown that PM gears can be manufactured well within the DIN "quality 8" tolerance fields" [34].

Next application of PM gears in automotive gearbox is described in [35]. In this case a gearset of a passenger car **Smart Fortwo** was by the company Höganäs AB redesigned to the PM variant to maintain the original reliability, tested and evaluated after 200 000 km of real driving. In this case the HIP technology wasn`t applied.

2.3.5 Gearing Material Research Conclusion

In the previous paragraphs were very briefly described common materials used in the gearing production. Very innovative and interesting seem to be especially the PM material. For this reason, the **PM material** was chosen as an investigated gears material within this thesis.

2.4 Gearing Geometry

2.4.1 Gearing with an Involute Profile

The most commonly used involute profile for general purposes is the profile "1,25/0,38/1,0 ISO 53:1998 Profil A" with a normal pressure angle of 20°. By using of this profile there are commonly alternating one and two teeth pairs in the mesh, which causes the abrupt stiffness change along the line of action. This situation, where the transverse contact ratio $\varepsilon_\alpha \in (1,2)$ is for the spur gearing depicted in Figure 2-5 (left). To eliminate this issue, the gearing with **HCR (High Contact Ratio)** profile can be used. The aim of this profile is to reach the integer value of $\varepsilon_\alpha \in N$ (2, 3, 4 …). To preserve original transverse pitch, the value of normal pressure angle α_n must be reduced. An example of such gearing is depicted in Figure 2-5 (right), where 2 teeth pairs are permanent in the mesh.

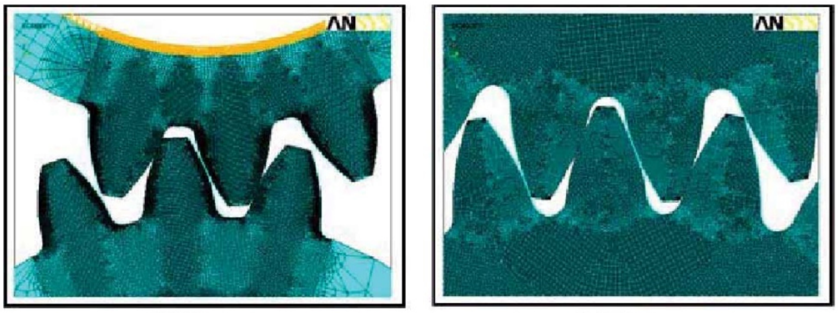

Figure 2-5: Typical appearance of standard NCR (left) and HCR (right) gearing [33].

In automotive industry (mass production) this profile is commonly used, because it is worth to buy a very expensive manufacturing tool with this special profile despite its high price.

Except involute gearing with symmetric non-standard profile, the next possibility is a usage of the involute gearing with an **asymmetric profile**. This means, that there are different pressure angles on both sides, Figure 2-6. This profile can be used for contact pressure reduction if loading conditions are significantly different on both tooth sides, e.g. in helicopter rotors drivetrains.

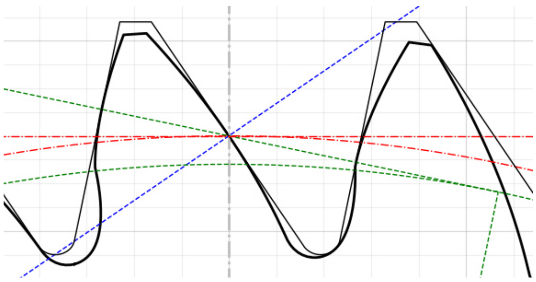

Figure 2-6: Appearance of the involute gearing with an asymmetric profile.

Involute gearing with an asymmetric profile is also described in [26], [27], [28], [29], [30], [31] and [32].

2.4.2 Gearing with Non-Involute Profiles

2.4.2.1 Cycloidal Gearing

Next type of gearing can be marked as gears with non-involute profiles. The first type is the **cycloidal gearing**. Its characteristic S-shaped teeth are created by rolling the roll circle along the pitch line. There are more possible geometries, so called **epicycloidal** and **hypocycloidal** gearing, when the roll circle is not being rolled along a pitch line but along the pitch circle. Its design is described in [13].

2.4.2.2 Wildhaber-Novikov Gearing

Next type of gearing with non-involute profile is the Wildhaber - Novikov gearing. Its characteristic property is the **circular profile** in the transverse plane. The positive property of Wildhaber-Novikov gearing lays in the large contact area, and thus very low contact pressure. It is also very positive from the point of view of lubrication. For both these reasons the load - carrying capacity is several times higher in comparison to involute gearing. The disadvantage of this type of gearing lays in strictly given value of the helix angle - depending on the facewidth - to assure constant ratio.

There are still several gearing types, e.g. "**Extended version of Wildhaber-Novikov gearing**", where the tooth height is increased and the tooth flank consists of a convex circle at the addendum and a concave circle at the dedendum. This allows to finetune the gearing properties.

Next improvement to this gearing invented prof. Nagata in 1981, who improved the gearing geometry directly on the manufacturing tool. The new profile is called **Wildhaber-Novikov-Nagata gearing** and has circular addendum and a dedendum that consists of two involutes that blend at mid-dedendum.

2.4.2.3 Convoloid Gearing

Convoloid gearing has very similar design as Wildhaber-Novikov-Nagata gearing. It was introduced in 2011 by Bernard Berlinger and John Colbourne. This type of gears is commonly used in wind turbines transmissions for its advantage in lowering contact stresses.

“The addendum has a convex shape, while the dedendum is concave. The transition zone at the pitch point seems to be “S-shaped” rather than the straight section of Nagata’s development. The authors report that the tooth profiles are computer calculated as a point cloud for each application case individually.” This gearing and its characteristic properties are described in [13] and [14].

2.4.3 Gearing Geometry Research Conclusion

In the previous chapters there were described the most important types of used gearing geometry. For this thesis it is very important to reach real data from the measurement. Therefore, the finally chosen geometry variant had to fulfill two requirements; it must be non-standard and furthermore manufacturable, to be able to produce it and test it in the laboratory. The compromise of these two conditions was the **helical involute gearing with an asymmetric profile**.

2.5 Demands on New Gear Design

In previous chapters were described geometrical, material and technological aspects of the current gears production. From these descriptions arose demands on new gear solution. The designed gearset should have the potential of:

- endurance strength increase
- production costs reduction
- maintaining original gearset size (original gearbox housing)
- weight reduction

To enable the fulfilment of these above stated demands following methods (ways) can be used:

- new material
- new technology
- new geometry
- combination of above-mentioned ways

3 Thesis Targets

In previous chapter there was described the current state of the art in the field of gears from the point of view of the manufacturing technology, material and geometry.

The intention of this thesis is to apply new technologies on PM material in the field of gears dedicated for **automotive transmissions** and to determine their influence on gearset's endurance. It has two main parts.

In the first part the topic is the application of the HIP technology on gearing with standard symmetric geometry and chip manufacturing. The innovative is in this case not the HIP technology itself, but the field of usage of these HIP:ed PM gears.

The second part deals with the design of asymmetric gear profile to reduce contact stresses between tooth flanks. Furthermore, the gearwheels were manufactured with help of the rolling technology.

In order to be able to design gears with an asymmetric profile, it was necessary to create the appropriate software, because at the time we started this work, no commercial software was available. For the strength analysis of the gearing with an asymmetric profile FEM method was used.

Precise targets definition of this thesis was influenced also by real production possibilities, which were performed by cooperating companies, namely the **Höganäs AB** from Sweden as a specialist in PM (Powder Metal) branch and German **Profiroll Technologies GmbH**, a specialist in branch of the rolling technology. Next very important role played a car producer, who provided us all necessary support during the gears manufacturing, measuring and testing.

Almost all endurance test results of gears made of new materials (or using new technologies) were performed on special test rigs very. In this case the target is to shorten the test, thus the torque is increased appropriately. This means, that gearboxes, shafts and bearings are very robust, to have the gearing as the weakest part in the whole testing circuit. Tested gearing is commonly a spur gearing, the lubrication is realized using pipes, which deliver the oil directly in the gear mesh, or at least the oil circulation is used not to exceed allowed temperature.

In the automotive gearbox the situation is very different from above mentioned loading conditions. The weight is very important, therefore the gearbox cast is made of the aluminum alloy with thin walls, long thin shafts lead to high displacements due to bending and the oil can be cooled only using fans. Local loading conditions are unique for each speed stage in the gearbox.

From all these limiting conditions emerged targets of this thesis to be fulfilled. These targets are stated in following points:

1. **Determine endurance potential (strength) of gears with original design made from PM material with special technology (HIP) based on test results performed directly in automotive gearbox**

2. **Determine endurance potential (strength) based on test results of gears with asymmetric profile made from PM material with special surface treatment (rolling) based on test results performed directly in automotive gearbox**

To be able to accomplish the above stated targets, these special gears must be designed, produced and tested. For this reason, this thesis continues with the detailed description of gears testing methods in the Chapter 4.

4 Automotive Gearbox Endurance Testing

The resistance against all possible gearing failures is the most important parameter, and thus must be tested. These methods are described for example in the standard **ISO 6336-5:2003** in its Chapter 4.1. Their short description follows.

4.1 Allowable stress numbers determination - Method A

"The allowable stress numbers for contact and bending are derived from endurance tests of gears having dimensions closely similar to those of the gears to be rated, under test conditions which are closely similar to the intended operating conditions." [15] Using this method is the best way how to determine the gearing endurance, because here the loading conditions are very close to the real operating regime. For testing of the gearing in this way two basic principles can be used, the "Open" or "Closed (Back-to-back)" Test Bench. In case of the "**Open Test Bench**" power is created in the driving dynamometer, flows through the gearbox and finally it is wasted in the braking dynamometer.

The second way how to test the gearbox according to the above mentioned method "A" is to use the **"Closed (Back-to-Back) Test Bench"**. In comparison to opened test bench, this type of test bench has big advantage in its power demands, because of usage the power circulation. Its basic principle and power flow diagram are depicted in Figure 4-1. The closed test bench consists of two gearboxes (gearsets), one of them is so called "tested" one. The second is the "technological" one and serves only for the closing of the circuit. The essential condition for correct function is that both these gearsets must have same gear ratio.

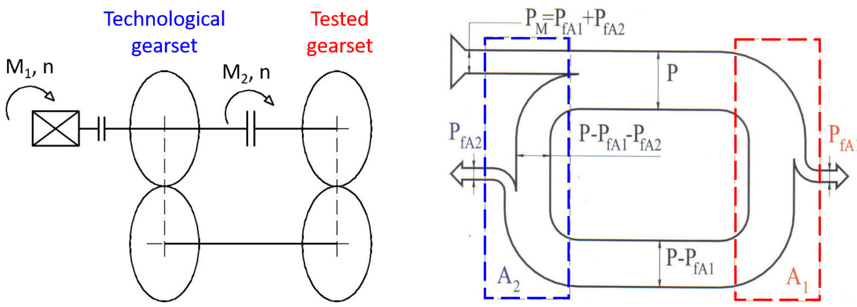

Figure 4-1: Scheme (left) and the power flow diagram (right) of the closed (back-to-back) test bench for gearset testing [16].

The input shaft is divided into two parts by a friction clutch. Torque M_2 is used for loading, torque M_1 to overrule only the passive resistances in the circuit. In the Figure 4-1 is in the power flow diagram visible the amount of power, which can be reused in the circuit due to the circulation.

4.2 Next Methods for Allowable Stress Numbers Determination

Next used test method is the "**Method B**". Tests performed by this method use a type of loading, which does not exactly correspond to real gearing loading in a gearset. The typical example of is the **pulsator testing**. Next testing methods are not worth mentioning in this short thesis version.

4.3 Test Method Choosing within this Thesis

In previous chapters were described methods, which can be used for gearing testing. From their description it is clear, that the way how to get best results, is within usage of the method A (4.1), which enables to apply testing conditions very close to the real ones. To get the most representative test results, **testing according to the method A (4.1) was chosen within this thesis**.

5 Used Test Bench Description

This test bench is placed in newly built laboratories of the CTU in Prague in ***Roztoky Science and Technology Park (VTP Roztoky)***. Present appearance of this test bench is depicted in Figure 5-1.

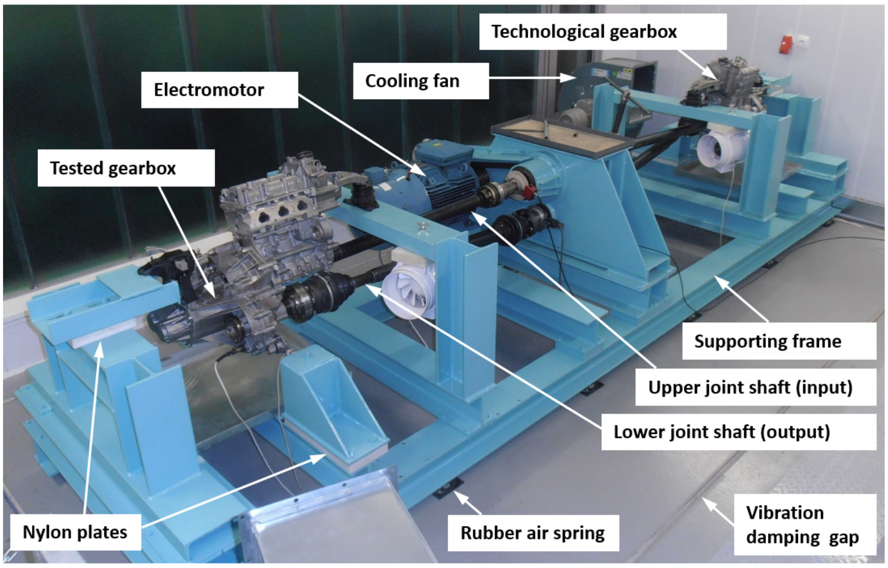

Figure 5-1: Back-to-back test rig in the CTU laboratory in Roztoky Science and Technology Park

It is a back-to-back test stand designed directly for testing of the automotive gearboxes **MQ200**. The specialty of this stand is the way of mounting of gearboxes on the test rig. Both gearboxes are installed in the same manner as they are functioning in the car. The gearbox is screwed together with engine cast. This assembly is then hung up to the test-stand frame on two silent-blocks, firstly on the engine side and secondly on the gearbox side. Finally, a reaction strut is mounted to the lower part of the gearbox. Because the gearsets are tested directly in the original gearbox housing, same conditions in the gear mesh are ensured, e.g. shaft and bearing deformations, lubrication, etc. This fact is very positive from the point of view of test results credibility.

5.1 Measurement Devices and Equipment

During the measurement within this thesis, the test bench was continuously improved. It has two pretensioning devices (torque units). For constant torque regime the worm gearset with friction clutch is used, for dynamic testing (variable torque) the planetary torque unit (PTU) can be used.

To be able to correctly performe the endurance test, some important values must be measured, namely the torque and speed of the input and output shaft, oil temperatures and reactional forces in the struts. For the fault detection the vibrations measurement system was used.

For assuring correct function of the PTU, an additional reducing gearbox had to be used. Furthermore, also the oil cooling system had to be designed and created for the PTU. Finally, the two shifting robots were designed to allow unmanned function of this test bench. Of course, for the controlling of this test bench special programming Labview had to be created.

Thanks to performed tests were also experimentally determined all power losses in the testing circuit. This whole power analysis of the closed circuit was presented in Detroit in [39].

6 Asymmetric Gearing

Within this thesis, involute gearing with an asymmetric profile had to be designed, manufactured and tested. First step to perform it was to design theoretical profile of this gearing. For this purpose, there was a need of a special software.

This “project” started in the spring of 2016. At that time, it still was not possible to use any commercial software for this purpose. The module in KissSoft for the gearing with an asymmetric profile (BETA Version) has been published only in the March of 2018, based on [26]. For this reason, special software for designing of the involute gearing with asymmetric profile had to be developed. The initial step was to describe the geometry of the asymmetric involute gearing.

6.1 Nomenclature and Usage of Asymmetric Gearing

Initially, the side (flank) nomenclature should be defined. When driving a car, the power is transmitted at the “drive” flank; with Internal combustion engine (ICE) braking, power is transmitted from the wheels to the engine via the “coast” flank of the teeth. In the case of a passenger car, load conditions at both tooth flanks can be very different. This important difference is caused by ICE turbocharging (torque increase). This means that one tooth flank is used much more often and with higher loading torque than the opposite one. These values of cycles and equivalent torques were computed from the loading spectrum according to the methodic from [20].

From the point of view of gearing durability, a symmetric profile in the case of passenger cars is not ideal. One side is always either overloaded or has excessive durability, particularly from the perspective of Hertzian (contact) stresses, which affect pitting formation. To optimize the durability at both tooth sides according to loading conditions, it is useful to reduce the contact stress on the drive side. This reduction can be achieved using an asymmetric gearing profile. This means that the most important parameter of the rack profile - angle α_n (α_{P0}) - is not the same for both sides.

6.2 Geometry of Asymmetric Involute Gearing

When designing the macrogeometry of a **symmetric** gearset, all input parameters must be determined. The result is then the sum of required profile shift coefficients x_Σ, which is calculated using Formula (6-1). The value of x_Σ is calculated with the condition of no backlash.

$$x_\Sigma = x_1 + x_2 = \frac{z_1 + z_2}{2 \cdot tg\ \alpha_n} \cdot (inv\ \alpha_{tw} - inv\ \alpha_t) \tag{6-1}$$

In the case of symmetric profile (rack), the tooth thickness is simply divided into two identical halves. In the case of asymmetric gearing, the situation is different. Figure 6-1 shows the situation in the transverse plane during manufacturing, i.e. the mesh of the tool (rack) with a gearwheel. All values with the symbol ”0” are related to a tool. A basic parameter is the thickness at tool’s reference plane s_{t0}, which equals half of the transverse pitch p_{t0}, as in the case of a symmetric version.

$$s_{t0} = \frac{p_{t0}}{2} = \frac{\pi \cdot m_t}{2} \tag{6-2}$$

This tool thickness is a sum of partial tool thicknesses at both sides (drive and coast)

$$s_{t0} = s_{t0\ drive} + s_{t0\ coast} \tag{6-3}$$

Furthermore, the asymmetry ratio "A" depends on the transverse profile angles and can be defined as the ratio of both these thicknesses

$$A = \frac{s_{t0\ drive}}{s_{t0\ coast}} = \frac{\tan \alpha_{t\ drive}}{\tan \alpha_{t\ coast}} \tag{6-4}$$

For symmetric gearing this value is $A = 1$. For asymmetric gearing there is standardly higher value of the profile angle α_t at the drive side. For this reason, also the thickness on the drive side is higher than on the coast side and thus the asymmetry ratio is standardly $A > 1$.

$$s_{t0\ drive} = s_{t0} \cdot \frac{A}{1+A} \quad , \quad s_{t0\ coast} = s_{t0} \cdot \frac{1}{1+A} \tag{6-5}$$

Regarding the tooth profile, the rack shift must be considered. If the profile shift coefficient is equal to zero, the tooth thickness s_t at the reference diameter *d* is the same as the thickness of the tool (gap) s_{t0}. If the rack is in a general position, i.e. shifted about the distance of *"x.m$_n$"* as depicted in Figure 6-1, tooth thicknesses at reference diameter on both sides are still divided in the same ratio *A* and their values are

$$s_{t\ drive} = s_{t0\ drive} + x \cdot m_n \cdot \tan \alpha_{t\ drive} = s_{t0\ drive} + x \cdot m_t \cdot \tan\ \alpha_{n\ drive} \tag{6-6}$$

$$s_{t\ coast} = s_{t0\ coast} + x \cdot m_n \cdot \tan \alpha_{t\ coast} = s_{t0\ coast} + x \cdot m_t \cdot \tan\ \alpha_{n\ coast} \tag{6-7}$$

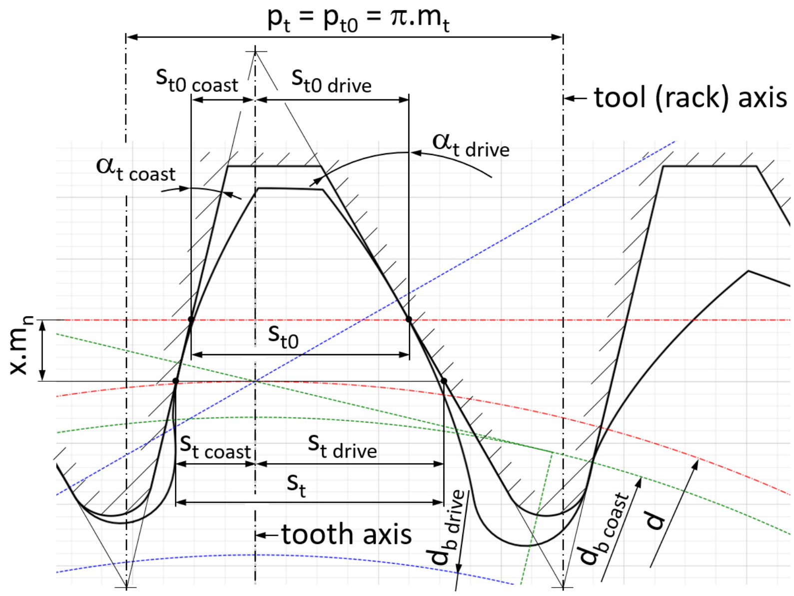

Figure 6-1: Situation in transverse plane while manufacturing of asymmetric gearing

To be able to derive a formula for a needed sum of both profile shift coefficients for the case of asymmetric gearing without a backlash, the basic condition of correct mesh must be used, Formula (6-8). The meaning is that the sum of tooth thicknesses of mating gears at their working pitch diameters must remain the transverse pitch.

$$p_{tw} = s_{tw1} + s_{tw2} \tag{6-8}$$

From this basic formula can be after performing more mathematical steps derived the final result, which is the sum of profile shift coefficients $x_1 + x_2$ of both gearwheels for the condition of no backlash, Formula (6-9).

$$x_\Sigma = x_1 + x_2 = \frac{(z_1 + z_2) \cdot (inv\alpha_{tw\,drive} - inv\alpha_{t\,drive} + inv\alpha_{tw\,coast} - inv\alpha_{t\,coast})}{2 \cdot (\tan\alpha_{n\,drive} + \tan\alpha_{n\,coast})} \tag{6-9}$$

As it was mentioned, value of x_Σ resulting from the Formula (6-9) is derived for the condition of theoretical mesh with no backlash between tooth flanks. Similarity with the Formula **Fehler! Verweisquelle konnte nicht gefunden werden.** is obvious at the first glance. The only difference is, that appropriate expressions are used separately for each tooth side, instead of one value identical on both sides as for symmetric profile. One of these two values $x_{1,2}$ must be set and the second one is then calculated from x_Σ.

6.3 Gearset Depiction including the Backlash

For assuring of correct gearset function some minimal backlash is needed between tooth flanks while meshing. To be able to determine and depict this circumferential transverse backlash j_t, the attention must be focused again on working pitch diameters $d_{w\,1,2}$. Fundamental condition for meshing without a backlash expressed by the Formula (6-8) then changes to the Formula (6-10), where circumferential backlash j_t is already considered.

$$p_{tw} = s_{tw1} + s_{tw2} + j_t \quad , \qquad j_t = p_{tw} - s_{tw1} - s_{tw2} \tag{6-10}$$

From Formula (6-10) it is clear, that the sum of both tooth thicknesses $s_{tw\,1,2}$ must be smaller than for the case without clearance. These thicknesses are influenced by the values of profile shift coefficients. This means that for this case, including the backlash, the Formula (6-9) cannot be used. When considering a real case including the backlash, both values of $x_{1,2}$ must be entered and resulting transverse backlash j_t can be then calculated according to the Formula (6-10).

More important than circumferential transverse backlash j_t is the normal one j_n. In the case of asymmetric gearing, the same formula is used as for a symmetric profile, but relevant angles differ at both sides, Formula (6-11).

$$j_{n\,drive} = j_t \cdot \cos\alpha_{tw\,drive} \cdot \cos\beta_w \quad , \quad j_{n\,coast} = j_t \cdot \cos\alpha_{tw\,coast} \cdot \cos\beta_w \tag{6-11}$$

While meshing of drive flanks, the backlash between coast flanks appears (and conversely), Figure 6-2. Normally the drive side is the more important one and has a higher value of mesh angle than the coast one. For this reason, backlash at the coast side is standardly higher too.

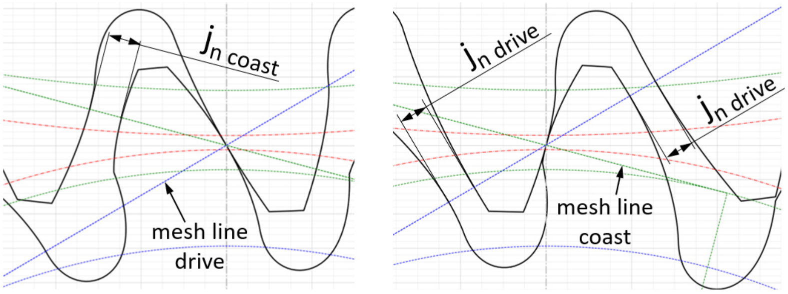

Figure 6-2: Normal backlash j_n between coast/drive flanks while drive/coast flanks mesh (left/right).

The geometry of asymmetric gearing and created software appearance were in detail described in [41] and [44].

6.4 Software Description

This software was developed in the Python programming software. The KissSoft program served as an example of its appearance. In the final version the tool profile includes standard technological issues (properties) - the tip chamfer and the protuberance. In addition, the situation while manufacturing is possible to be depicted – the meshing of the gear with the tool (rack).

Each gearwheel has its own manufacturing tool. The only common parameters of these two tools are angles α_{P0} at both sides to assure correct mesh. All other parameters can be different, e.g. addendum and dedendum heights. In total, four separate profiles of tooth flanks can be defined. All these parameters can be set in a separate pop-up window after clicking the button "Modify", which can be seen in Figure 6-4. Its appearance is depicted in Figure 6-3.

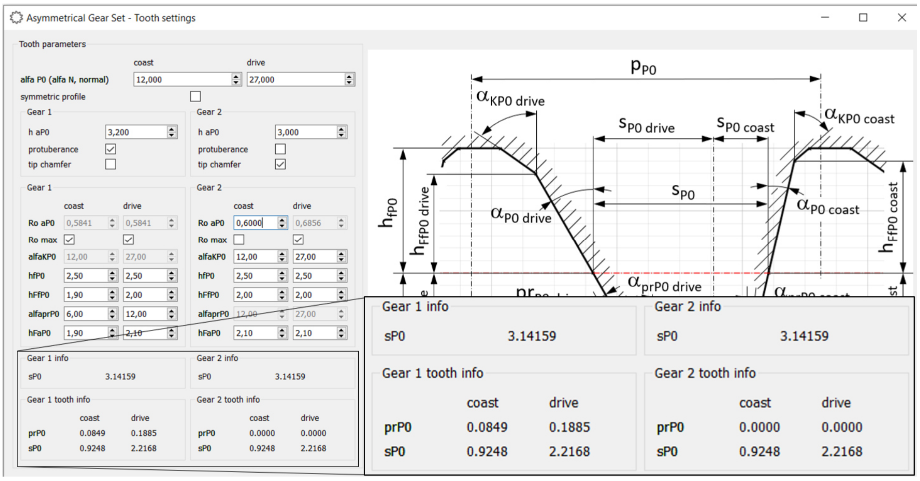

Figure 6-3: Appearance of the window for setting of a tool (rack) parameters with asymmetric profile. This window pops-up after clicking the button "Modify".

In Figure 6-4 can be also seen that the checkbox "no backlash" is checked. For this reason, the value of x_{Σ} was calculated and depicted in the tab „Gear set info" and furthermore, it is possible to set only the value of x_1 and the value of x_2 is computed automatically without a possibility to be changed.

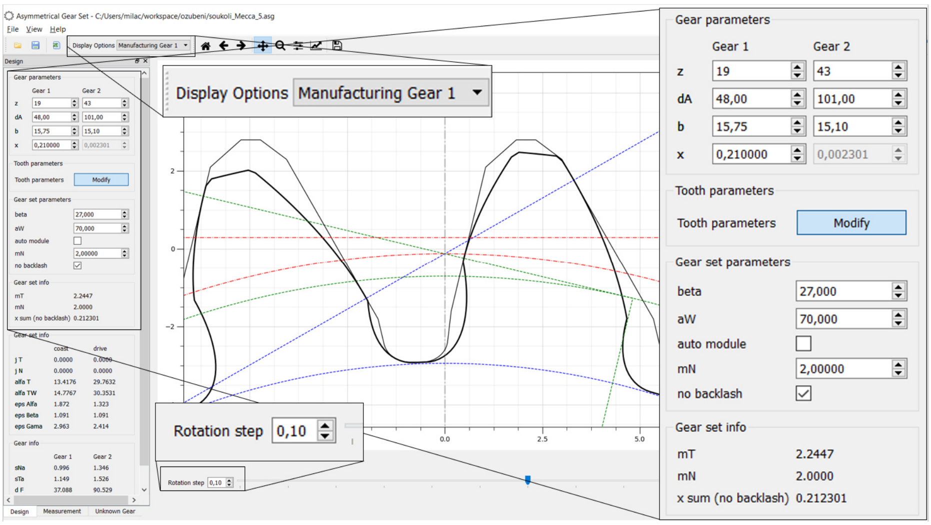

Figure 6-4: Appearance of the program– display option "Manufacturing gear 1".

All used formulas for completing the geometry calculations were found in [25].

Today, it is possible to use it only for external, spur and helical, symmetric and asymmetric design. All necessary parameters of the gearset can be set on the left side. In the left bottom corner are three tabs: “Design”, “Measurement” and “Unknown gear”, Figure 6-6. In the tab “Design”, basic input parameters of the gearset can be entered.

Thanks to this program, it is also possible to calculate the value of the measurement over teeth (span measurement) and over balls. This can be seen in the tab ”Measurement” in the bottom-left corner. In the case of symmetric teeth both these variants are possible. In the case of asymmetric gearing, only the measurement over balls is possible because there is no common tangent between opposite involute flanks. Graphical depiction of the situation with embedded ball with the diameter “D” is depicted in Figure 6-5 and is meaningful only for spur gearing when the depicted ball really touches both flanks. For helical gearing, the calculated value of the ball centers diameter d_g is correct, but because of the helix angle, the inserted ball does not touch the depicted transverse gear profile.

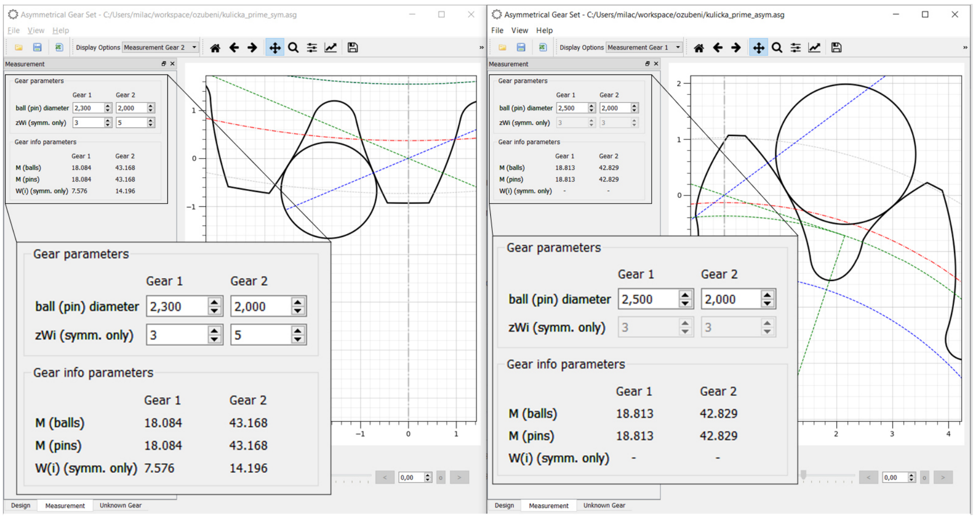

Figure 6-5: Appearance of the program tab „Measurement“ for spur gearing with an symmetric (left) and asymmetric (right) profile. Embedded ball touches the transverse tooth profile.

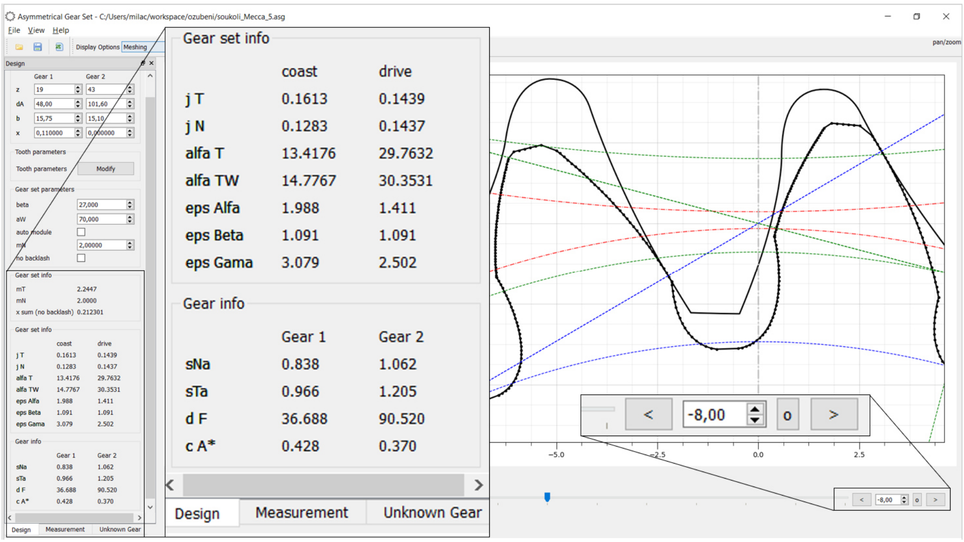

Figure 6-6: Appearance of the program including tip chamfer and protuberance – display option “Meshing”. Gear 1 is depicted using interpolating spline with highlighted equidistant points.

For gearset optimizing, all necessary values are computed and depicted in the part “Gear set info” e.g. normal and tangential backlash (j_n), transverse contact ratio (ε_α), axial contact ratio (ε_β), total contact ratio (ε_χ), minimum tip thickness without chamfering (s_{na}) and relative minimum tip - root clearance (c_a*), Figure 6-6. It is possible to change the angular position of the gearset, so the whole “passing through the mesh” of a gear pair or the gearwheel – rack meshing, can be observed. It can be done with arrows in the bottom-right corner, Figure 6-6. To be able to set a precise gear mesh position, it is possible to set the magnitude of rotational step (bottom-left corner of Figure 6-4) and absolute angular position of the gearset (bottom-right corner of the Figure 6-6).

6.5 Reliability Computation Method for Asymmetric Gearing

Analytically, the Hertzian stress at a single tooth contact can be computed using a standard Formula (6-12), where both osculation radii of touching bodies are known. In the case of a helical gearset, these bodies are cylinders with radii $\rho_{1,2}$.

$$\sigma_H = \sqrt{\frac{F_n}{\pi \cdot b} \cdot \frac{E}{2 \cdot (1-\nu^2)} \cdot \left(\frac{1}{\rho_1} + \frac{1}{\rho_2}\right)} \tag{6-12}$$

For involute gearing, the values of these osculation radii depend on its geometry. This case is depicted in Figure 6-7, which shows that these values are given, based on the involute geometry, as

$$\sigma_H = \rho_{1,2} = r_{w\,1,2} \cdot \sin \alpha_{tw} \tag{6-13}$$

This means that the contact stress can be reduced while maintaining the gearset’s main dimensions only by increasing the working transverse pressure angle a_{tw}. It is also clear that these osculation radii are much larger at the drive side (blue) than at the coast side (green).

Unlike for symmetric gearing, at the time when this thesis was created, no standardized method for calculation of the strength of involute gearing with an asymmetric profile existed. For this reason, FEM simulation must be firstly used for symmetric gearing. These results must be compared with the stress obtained from a standardized calculation. In this calculation are included all meshing and loading conditions using necessary coefficients (e.g. software KissSoft).

By comparing both these results, recalculation coefficients for contact stresses and for root bending stresses at each gearwheel, are defined using Formula (6-14).

$$k_{H\;FEM} = \frac{\sigma_{H\;KissSoft}}{\sigma_{H\;FEM}} \quad , \qquad k_{F\;FEM_{1,2}} = \frac{\sigma_{F\;KissSoft_{1,2}}}{\sigma_{F\;\;FEM_{drive\;side_{1,2}}}} \tag{6-14}$$

These coefficients are then used for backward recalculation of FEM results of asymmetric gearing to be comparable with symmetric gearing results.

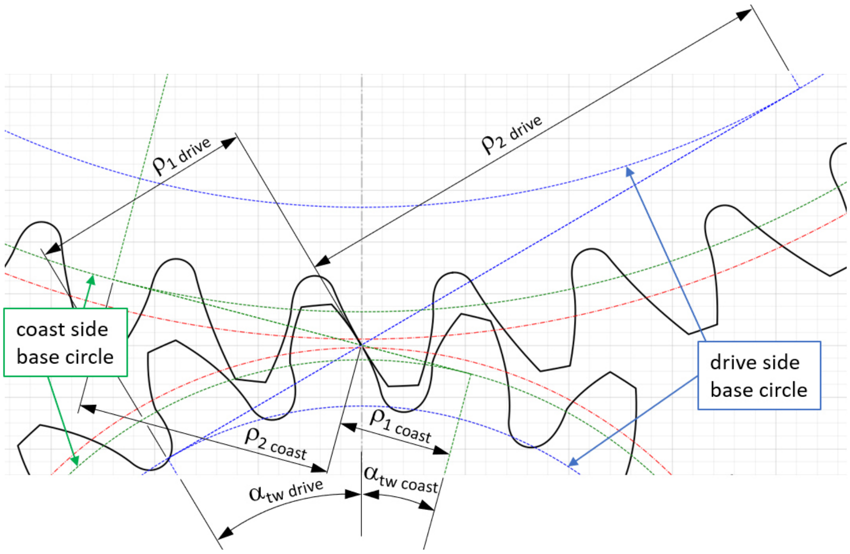

Figure 6-7: Osculation radii of involute gearset for single tooth contact in the mesh point.

The Figure 6-8 depicts an example of gearset FEM analysis. It's a comparison of symmetric and asymmetric variants of a gearset. The decrease in the contact pressure by the same load (torque) for the asymmetric variant is approx. 9.3%. Text with values of maximum stresses (von Mises [GPa]) are enlarged to be readable.

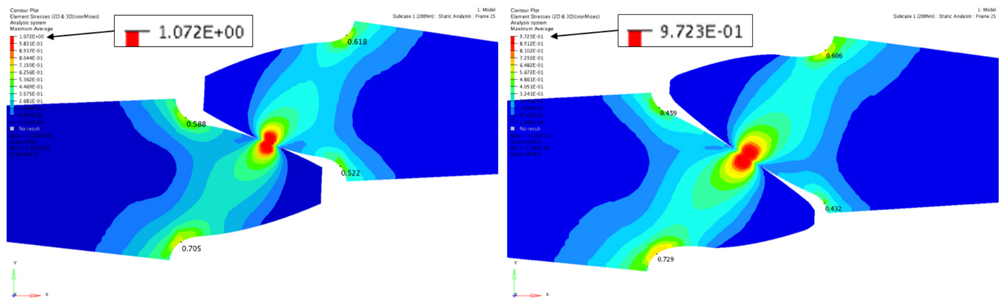

Figure 6-8: Example of FEM analysis of symmetric and asymmetric gearing, both profiles were designed using developed program.

On the other hand, asymmetric profile (higher pressure angle) also has one negative consequence - the increase in radial forces, which cause higher loading at bearings. The method of strength calculation for gears with asymmetric profile was published in [42].

6.6 Selection of Asymmetric Gearing Profile Parameters

According to the methodic used in [25] with the condition of the same safety against pitting at both sides using Life factor Z_{NT} can be determined the asymmetry ratio. This means that the drive angle is chosen, and the coast angle is according to this methodology calculated. This method leads to large undercut, so the coast angle had to be increased. In the next step three gearset's geometries were designed and one of them was selected to be produced (a_{nD}/a_{nC}= 29°/11°).

Very important for the final variant choice was the stress comparison. All profile variants were generated in newly created software (including original symmetric one).

The transverse gear profile was simply pulled into 3D for 1 mm as a spur gearing. The loading torque applied at this thin slice was then divided by the width of the gearset's thinner gear. An example of this computation is depicted in Figure 6-9 for contact stress (left) and root bending stresses (right).

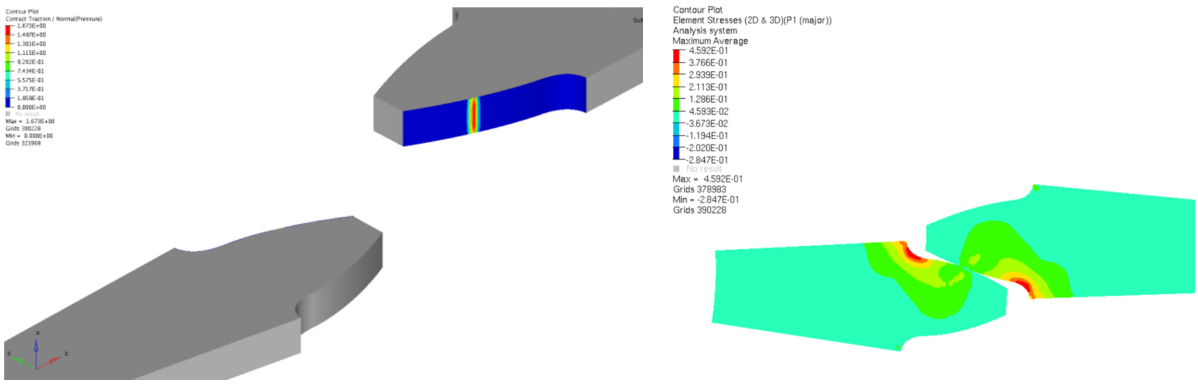

Figure 6-9: FEM analysis of the original symmetric gearing - contact (left), root bending (right)

The symmetric gearset was also computed in KissSoft. Obtained stress values from this computation include complete geometry (helix angle) and all standard stress and durability requisites, i.e., material properties and all necessary coefficients describing the geometric and loading conditions in the mesh according to the standard DIN 3990. By comparing both these results (KissSoft, FEM), the recalculation coefficients can be defined for contact and for root bending stress at each gearwheel according to Formulas (6-14). All these stress values are stated in Table 6-1.

Gearing Strength [MPa]	KissSoft (DIN 3990)		FEM (Optistruct)		k_{FEM}	
	gear1 z=48	gear2 z=31	gear1 z=48	gear2 z=31	gear1 z=48	gear2 z=31
Sigma H	1131		1401		**0,807452**	
Sigma F	608	578	443,1	459,2	**1,371948**	**1,258123**

Table 6-1: Comparison of bending and contact stresses at original (symmetric) gearing of 6th speed.

Using these coefficients were recalculated stresses obtained from the FEM analysis of all asymmetric variants. By usage of an asymmetric profile the significant decrease of the contact and bending stresses can be achieved.

From the point of view of stresses, the variant A would be the best choice. On contrary, from the point of view of the manufacturing (rolling) technology, the variant without undercut would be the best choice. Final variant was a compromise between these two conditions, the variant **B (29/11°)**.

7 Tested Gear Selection

Within this thesis, two gearsets should be tested. The choice was limited by their geometry and usage. Firstly - both the gearwheels had to be dismountable from the shafts, and secondly, they had to be well accessible – to be able to make visual inspection during testing. For these reasons, the 3rd and 6th speeds were selected. Both these gearsets are depicted in Figure 7-1.

At both gearsets another PM technology was used. In case of the 3rd speed the original symmetric geometry was maintained, the surface densification was performed by HIP technology by the company Höganäs AB. In case of the 6th speed, HIP technology was applied too (with symmetric profile). Furthermore, next test was performed - the gearing geometry was changed to asymmetric one and the surface densification was performed by rolling technology by the company Profiroll Technologies GmbH.

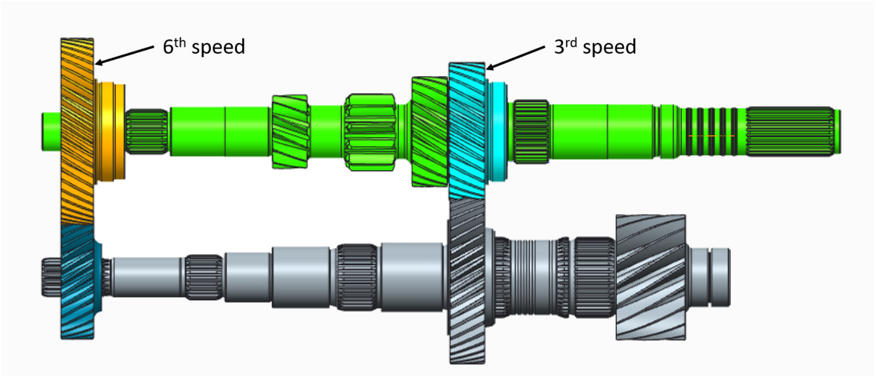

Figure 7-1: Two selected gearsets to be replaced with the gearsets made from the PM material instead of standard wrought steel.

7.1 HIP Project Description

This whole project was created in cooperation with the Swedish company Höganäs AB, which is one of the leaders in the PM technology field. The target was to perform dynamic test of PM gears directly in the car. For this reason, our next partner is a car manufacturer who had provided necessary support during all performed endurance tests. Because such "in car" driving test is very expensive, we have decided to start with a constant load level test to verify the material potential. Next dynamic tests should follow. To be as close as possible to real condition in automotive gearbox, special back-to-back test rig, which is located in laboratories of CTU in Prague, was an ideal device for such testing. This test was performed on two gearsets for each speed stage (3rd and 6th).

7.1.1 PM Material Determination

The exact PM material and its heat treatment was determined by the company Höganäs AB. For all PM gearwheels in this investigation the material was **Astaloy® 85Mo** with additional application of **HIP technology**, which was performed by the company Bodycote in Sweden.

7.1.2 HIP:ed Gears Manufacturing Process

In general, in the mass production PM parts are compacted in a final shape and afterwards they are sintered. Because of low number of the gears only simple PM pucks were compacted, sintered, HIP:ed and roughly turned . Subsequently these pucks were machined to the final shape, heat treated, ground and the synchrorings were welded. Finally, we had for each speed stage 6 gearsets ready for testing.

7.1.3 HIP:ed Gears Test Results – 3rd Speed

Before each endurance test of PM HIP:ed gearset the contact pattern test was performed to depict real load distribution in the gearing's mesh. The pitting was observed optically directly in the gearbox after certain number of testing hours.

As a starting point it was necessary to have some reference test results. For this reason, there were performed identical endurance tests with serial gearing for both speed stages. In case of the 3rd speed three gearsets were tested. Afterwards two PM HIP:ed gearset tests followed. Test results (number of hours until pitting detection) for the 3rd speed are graphically depicted using a bar diagram in Figure 7-2.

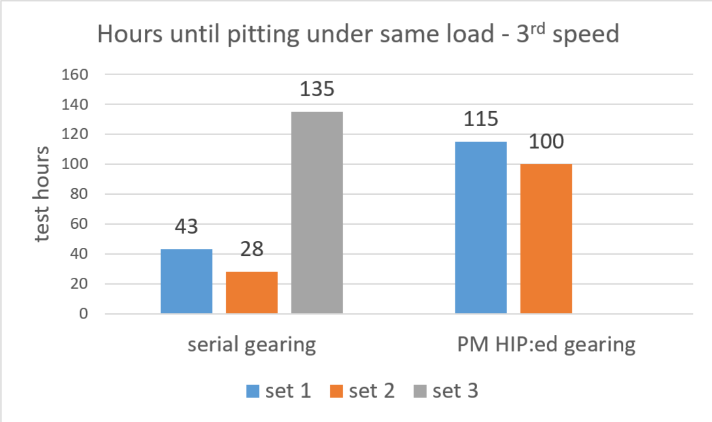

Figure 7-2: Test results of the 3rd speed - serial and PM HIP:ed gearsets – hours until pitting.

The appearance of pits on gearwheel's flanks is depicted in Figure 7-3. It was observed directly in the gearbox using a videoscope, so the quality is quite poor.

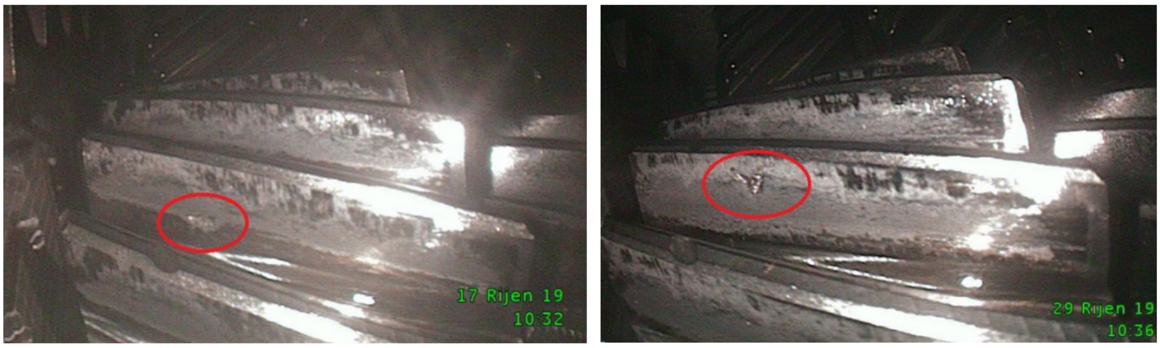

Figure 7-3: Pitting on PM HIP:ed gearwheel (3rd speed pinion) – gearset 1 - after 115 hours (left), gearset 2 – after 100 hours (right)

7.1.4 HIP:ed Gears Test Results – 6th Speed

As in the case of 3rd speed – comparative test with serial gearing was performed. In this case pitting occurred 6 times. Results of these tests are graphically depicted using a bar diagram in Figure 7-4. Afterwards, two tests with PM HIP:ed gearsets were performed.

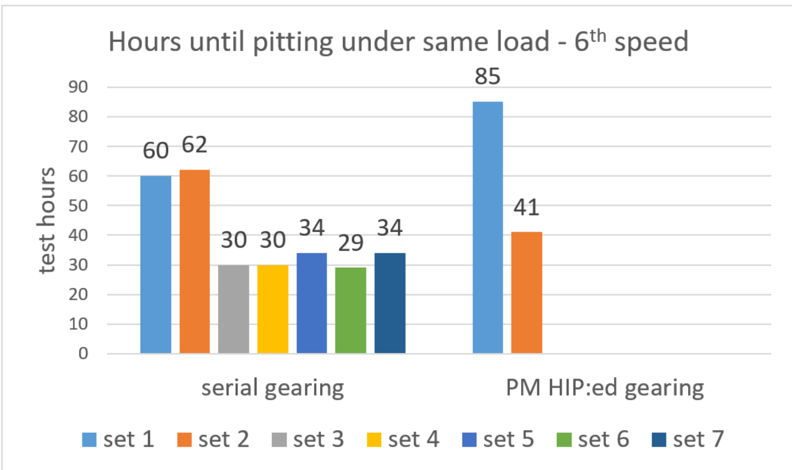

Figure 7-4: Test results of the 6th speed - serial and PM HIP:ed gearsets – hours until pitting

The appearance of pits on gearwheel's flanks is depicted in Figure 7-5. It was observed directly in the gearbox using a videoscope, so the quality is quite poor.

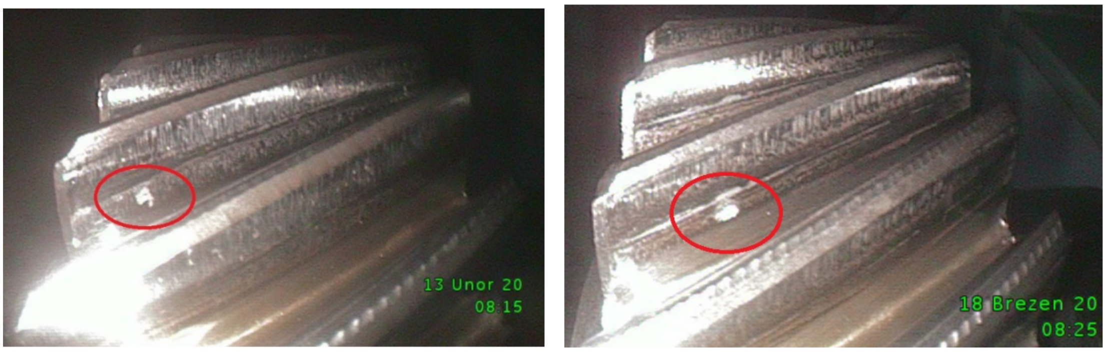

Figure 7-5: Pitting on PM HIP:ed gearwheel (6th speed) – gearset 1 - after 85 hours (left), gearset 2 – after 41 hours (right).

7.1.5 PM HIP:ed Gears Test Conclusion

From tests results it is clear, that in case of the 3rd speed the resistance against pitting was even higher by 56 %, in case of the 6th speed it was by 58 % higher in comparison with serial gearing. This holds true if the values of arithmetic means of test results (hours until pitting) are compared. Of course, these results are based only on low number of performed tests. To increase the probability, the number of performed tests would have to be increased accordingly. Results of both symmetric HIP:ed gears testing were published in [43].

By describing the results of performed tests of symmetric PM HIP:ed gears was accomplished thesis target number 1.

7.2 Asymmetric PM Gearwheels Project Description

The second part of this thesis deals with the rolling technology. For this reason, the cooperation with a specialist in this branch was necessary. For testing of asymmetric gears were chosen again 3rd and 6th speed. Finally, only the 6th speed was produced by the German company **Profiroll Technologies GmbH**. The aim was to produce a **PM gearset with an asymmetric profile** and densify its surface using the **rolling technology**. This gearset should be then tested in the laboratory at CTU in Prague to compare its strength with original and HIP:ed PM gears.

7.2.1 Rolled Asymmetric PM Gearset Production Process

The manufacturing process of these gears was very close to the one for HIP:ed gears. Blanks were compacted from a material **Astaloy® 85Mo**, sintered, rolled and heat treated. Real situation during the rolling process is depicted in Figure 7-6 (left).

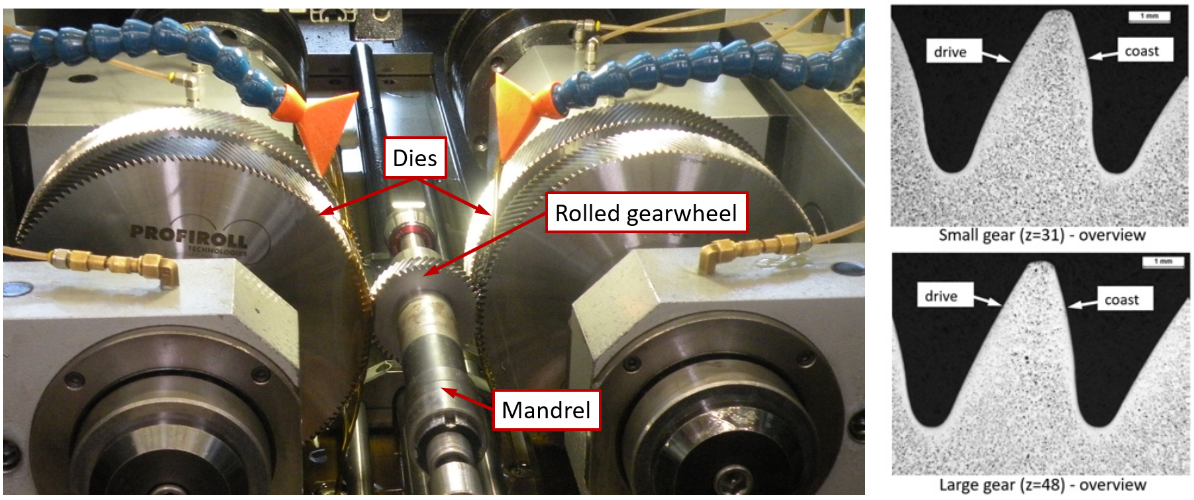

Figure 7-6: Gearwheel (z=48) during the rolling operation in the company Profiroll Technologies (left), structure analysis result from LOM of both rolled gears

7.2.2 Material Structure Analysis of Rolled PM Gears

In Figure 7-6 (right) is depicted the material structure of asymmetric rolled PM gears. The thin densified layer can be seen at the first glance. The density profile measurement on the large gear (z=48) showed that on the drive flank the densification depth with 98% full density was 50 µm, on the coast flank and the root it was 150 µm, Figure 7-7. On the small gear (z=31) on the drive flank the densification depth with 98% full density was 50 µm, on the coast flank and the root it was 150 µm.

Required radial force during rolling was adjusted according to performed measurement of the densification depth. The target was to reach the relative density of 98% into the depth of 0,05 mm. The case depth in the large gear (z=48) was 0.6 mm in both flanks and 0.4 mm in both roots.

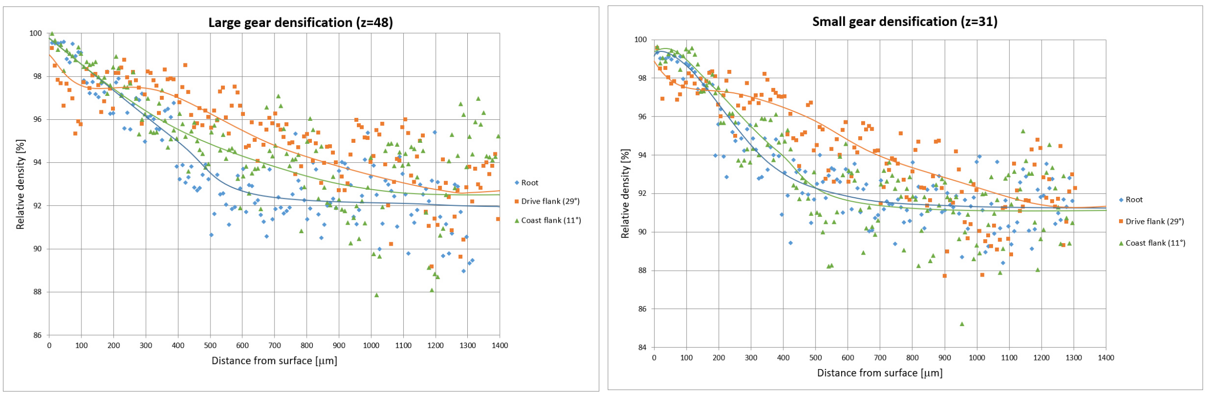

Figure 7-7: Density profiles of both flanks and the root on the large (left) and small (right) gear.

In the small gear (z=31) the teeth were through hardened, while the case depth in the coast root was 0.5 mm and in the drive root it was 0.3 mm. In Figure 7-8 is depicted the asymmetric gearset after all necessary production operations. In this state the gearset was ready to be tested.

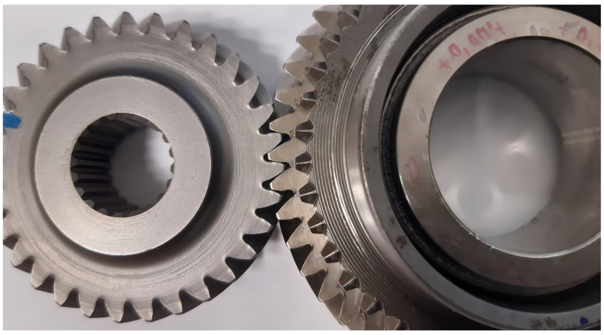

Figure 7-8: Asymmetric rolled PM gearset of the 6^{th} speed ready for testing

7.2.3 Weight Comparison of both Gear Variants

As it was mentioned, the basic material of these gears is **Astaloy® 85Mo**. Its natural property is the porosity, which logically leads to the density reduction (**7,25 g/cm³**) and thus also the weight reduction. This fact has positive influence on passive resistances and fuel consumption of the vehicle.

For the smaller gear (z=31) the weight saving is **6,4 %** (whole volume), for the pinion (z=48) it is only **3 %** because of usage of original synchroring made of steel.

7.3 Asymmetric Gears Test Results

The aim of these gears testing was to find out the potential of the rolling technology application on PM gears. For this reason, no additional flank surface treatment was performed after rolling. These gears were tested under same loading conditions as both HIP:ed gears. Surprisingly, this test ended after 1 hour and 35 minutes by a tooth breakage, Figure 7-9.

Due to this unexpected test result, the contact pattern test was performed once more. This time also the coast flank was investigated to verify whether the backlash is sufficient. By free rotating (torque = 0 Nm) there was found some imprint (trace) on the coast side of the small gear (z=31). This means that the backlash between flanks is not sufficient and therefore interference occurs.

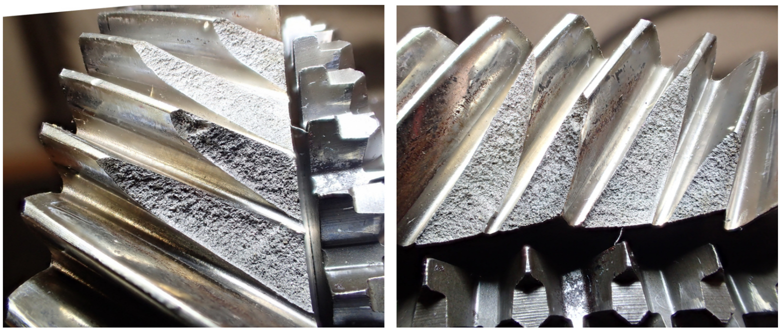

Figure 7-9: Test result of the first asymmetric rolled gearset – tooth breakage after 1h and 43min

This issue was solved by additional grinding of the coast flank to assure sufficient backlash. Because of very thin densified layer, grinding magnitude was divided to both gears into the depth of 0,05 mm. This increased the backlash about 0,1 mm. The grinding wasn't performed correctly, unevenly around the gear's circumference. Similar situation arose unfortunately also on the drive flank, which shouldn't be ground at all. The test was afterwards performed. It stopped after only 1,5 hours due to vibrations increase. There was detected a crack – still before tooth breakage. In detail is the crack depicted in Figure 7-10 (left). The crack structure was investigated by Höganäs. The result is that this was a fatigue crack, because directly in the corner there is a small area with very smooth surface, Figure 7-10 (right).

Regarding the material structure of this gear, the densification was checked optically using LOM. It was observed that the material porosity is very different around the gear’s circumference. The crack was caused by the combination of local overloading and too high porosity. Furthermore, next cracks were also found on the small gear (z=31) at least on 6 teeth, Figure 7-11.

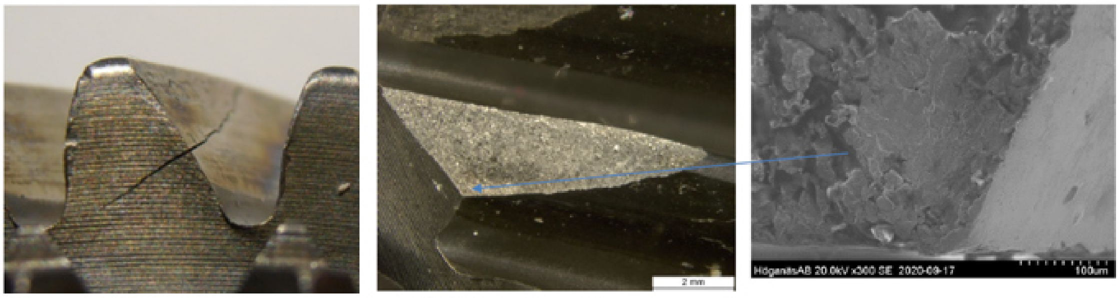

Figure 7-10: Tooth breakage of asymmetric rolled PM gear with ground coast flanks after 1,5 hours.

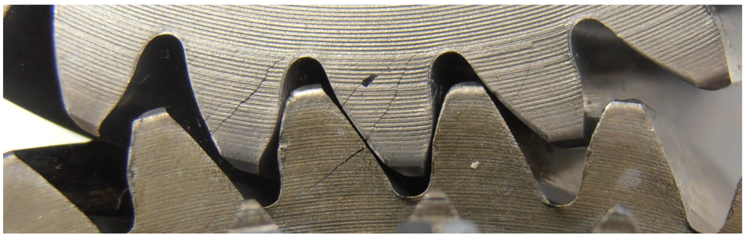

Figure 7-11: Meshing of PM rolled gears with asymmetric profile, both gears have already cracks.

All these tooth fractures, which occurred in less than two testing hours on both tested asymmetric gearsets are showing, that the surface geometry is crucial for the mesh quality and thus for the gear strength, even with sufficient backlash.

For this reason, it was necessary to grind our asymmetric gears to reach comparable tooth flank quality as HIP:ed gears. In the Czech Republic it was not possible, the only company which would be able to grind these asymmetric gears is the German, company Kapp Niles. Unfortunately, due to COVID 19 crisis there were no financial resources to perform this grinding operation.

Until these asymmetric PM rolled gears will not be ground to have comparable mesh quality as serially produced gears, it would be meaningless to test (destroy) them, because the test result wouldn’t describe the material and technology potential, which is the aim of this thesis.

For this reason, thesis target number 2 cannot be accomplished.

7.4 Tests Results Comparison of PM vs Serial Gears

In the laboratories of the CTU in Prague have been performed endurance tests of gearing with either special heat or surface treatment for many years. Within these tests were also tested serial gearsets. In the Table 7-1 is shown a comparison between PM HIP:ed and serial gears with same loading conditions, which were performed in labs of CTU in Roztoky Science and Technology Park.

	3rd speed		6th speed	
	serial gears	HIP:ed gears	serial gears	HIP:ed gears
Hours until pitting - mean value	68,7	107,5	56,6	63
ratio	**1,57**		**1,11**	

Table 7-1: Strength comparison between PM HIP:ed and serial gears of the 3rd and 6th speed.

For the case of the 3rd speed the strength increase of HIP:ed versus serial gears is 57%. For the case of the 6th speed the strength increase was "only" 11 %.

8 Conclusion

In this thesis are presented new progressive possibilities in the field of gearing from the point of view of the material, geometry and technology. The PM technology seems to be very interesting from the point of view of the production, because it can replace expensive rough milling and turning operations. Only the final tooth flank grinding must be maintained. For the case of the HIP:ed PM gears the tests were performed successfully. Unfortunately, for the case of rolled asymmetric PM gears could not be performed without final grinding. To be able to design the gearing with asymmetric involute profile, the software for this purpose was developed. Despite some natural disadvantages, PM material seems to be very perspective, especially for the mass production, which is for the automotive typical.

8.1 Thesis Targets Accomplishment

Beside standard thesis targets from the Chapter 3, which are described below, also the weight reduction should be mentioned in this chapter. The weight reduction potential of asymmetric rolled PM gears due to natural porosity was described in the Chapter 7.2.3. Exact values of the weight reduction were 3 % for the pinion and 6,4 % for the gear.

The description of thesis targets accomplishment follows.

1. ***Determine endurance potential (strength) of gears with original design made from PM material with special technology (HIP) based on test results performed directly in automotive gearbox.***

   These gears were produced and tested. Results of these tests are clearly depicted in the Figure 7-2 and Figure 7-4. The increase of the resistance against pitting of PM HIP:ed gears was for the 3rd speed by **56 %**, for the 6th speed by **57 %**.

   For the 6th speed can be the number of performed tests, used for the result comparison, increased by some other tests. The consequence of it is the reduction of the strength increase from previous 57% to "only" **11 %**, see Table 7-1.
2. ***Determine endurance potential (strength) based on test results of gears with asymmetric profile made from PM material with special surface treatment (rolling) based on test results performed directly in automotive gearbox***

   These gears were designed and produced too. Unfortunately, the surface microgeometry wasn't accurate enough to be comparable with symmetric serial geometry. This issue could be fixed by additional grinding, but unfortunately, due to the COVID19 crisis, needed financial support wasn't found. For this reason, this target couldn't be fulfilled.

8.2 Future Steps

Within this thesis were designed and produced PM gears, which were also tested in the automotive gearbox using a test bench described in the Chapter 5. All these tests were performed with constant torque and speed (rpm), because they were considered as the initial ones, to determine the PM material potential. In future steps, these gears should be tested also in dynamic mode, i.e. the load spectrum should be applied. This can be performed using an open test bench at CTU in Prague.

In next step these PM gears should be loaded by an ICE for assuring correct input torque signal (course) and the final step should be the testing directly in the car on the testing circuit.

Next very interesting opportunity seems to be in the noise reduction potential of the PM material thanks to its natural porosity. But such tests cannot be performed directly in the automotive gearbox, because there are too many disturbing aspects (bearings, more gearsets, etc.), so the noise difference between standard and PM gearset is not measurable.

As an outlook to next work all these topics are summarized in next steps:

- finetune the design of the PTU to enable dynamic tests on the back-to-back test rig
- perform the grinding of **asymmetric rolled** gears and test them
- perform dynamic tests of the **symmetric HIP:ed** and **asymmetric rolled** gears on the open test stand
- perform dynamic tests of the **symmetric HIP:ed** and **asymmetric rolled** with the ICE
- perform dynamic tests of the **symmetric HIP:ed** and **asymmetric rolled** gears directly in the car on the testing circuit
- determine the noise reduction potential of PM gears

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