A NEW GEARBOX GENERATION FOR VERTICAL ROLLER MILLS

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Drive trains of vertical roller mills are usually equipped with heavy duty gearboxes with a horizontal input shaft and a vertical output shaft. Increasing mill sizes bring the existing gearbox concepts to their limits, however. This, and the demand for higher reliability and lower costs, has forced gearbox manufactures to develop new gearbox concepts. In this article, an innovative new gearbox concept that meets all the demands of the latest mill generation is presented.

1 Introduction

Vertical roller mills are widely accepted as the most efficient means for grinding cement raw material and clinker. They are capable of preparing a wide range of feed materials to the required fineness using an energy-efficient process. Although cement raw materials vary considerably in the level to which they can be ground, their drying requirements and abrasion, the roller mill is sufficiently flexible so that it can be adapted to these variations, as well as other specific site requirements. Ever since the roller mill was first introduced to the cement industry, its system capacity has continued to evolve. With the increasing capacity of the mills, the output torque of the gearbox, its main design criterion, had to be increased accordingly (see *Fig. 1*). These increasing mill sizes have brought the currently used gearbox concepts to their limits, and new design concepts have to be found.

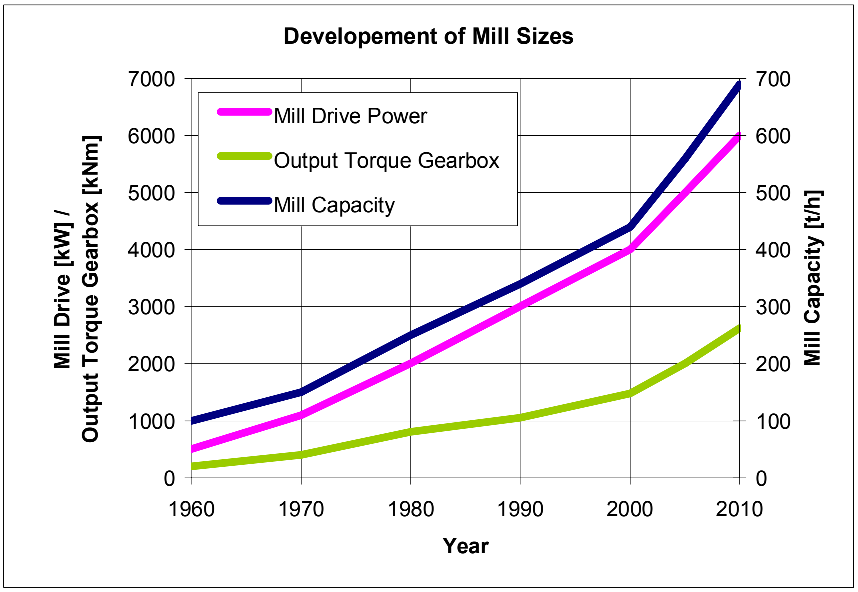


Fig. 1: Development of mill sizes

1.1 Function of the main gearbox in a vertical roller mill

The main gearbox of a vertical roller mill has to perform two major tasks (see also *Fig. 2*):

1. Transmission of the power from the main motor to the grinding table

2. Absorption of the grinding forces and the table weight

The first task is fulfilled by a specific gear arrangement, which reduces the motor speed to the required speed of the grinding table. As only standard motors with a horizontal shaft are used in today's mills, the gearbox needs a horizontal input shaft and a vertical output shaft. The gearbox therefore always includes a bevel stage. The output shaft of the gearbox is designed as a flange that is directly connected to the grinding table.

Absorption of the grinding forces is the second task of the main gearbox. This is achieved by a tilting pad thrust bearing underneath the output flange of the gearbox. The flow of forces from the grinding table to the foundation of the mill goes directly through the gearbox, which must be taken into account while designing the gearbox.

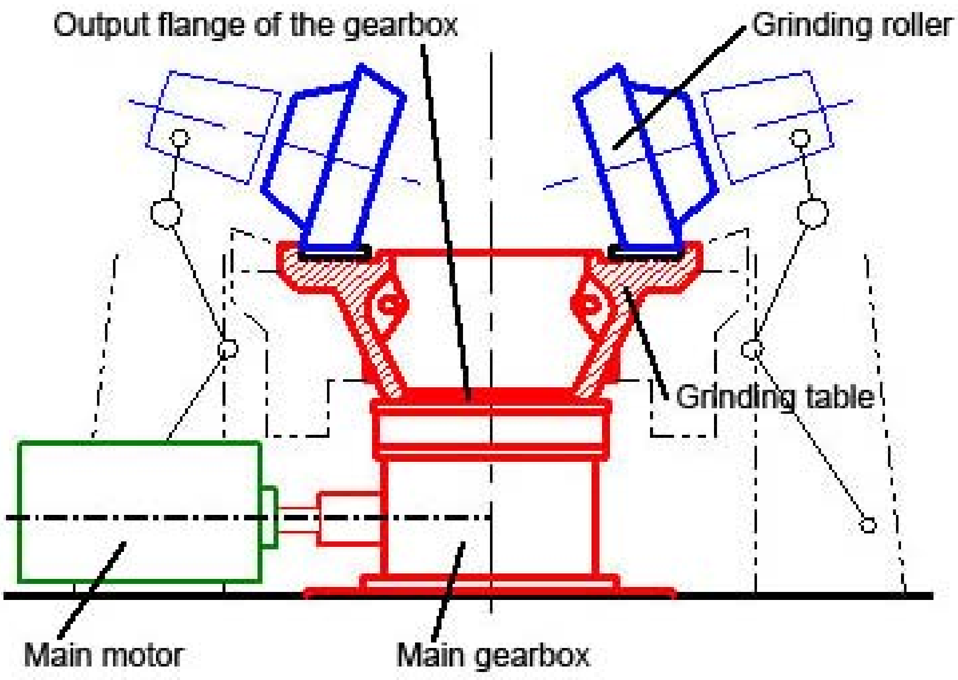


Fig. 2: General setup of gearbox for vertical mill

1.2 Requirements on the gearboxes

The main gearbox is an integral part of the complete mill equipment, and the requirements of the mill manufacturers on the gearbox suppliers are therefore very high.

The following is required:

- High operational reliability
- Very long service life
- Good maintainability
- Favourable price
- High efficiency
- Customer-specific design

Only manufacturers with long-term experience in this business are able to meet these requirements.

2 Today's gearbox technology

Over the years, the gearbox design has changed as a result of the increasing mill sizes and new technologies in gear manufacturing. In the following, we will examine the different gearbox concepts, their applications and their technical limits.

2.1 First Generation

The first gearbox generation for vertical roller mills consisted of two helical gear stages combined with a bevel stage as shown in *Fig. 3*. This concept was used from 50 kW up to 3900 kW. Today, this concept is normally used up to a power of approx. 1000 kW, while larger gearboxes are usually designed with a planetary and a bevel stage (see also *Section 2.2*). The disadvantages of this gearbox design at more than 1000 kW are the very heavy design and the very large gear wheels. The low speed gear wheel of a 3900 kW gearbox has a diameter of approx. 3.7 m and a weight of 23 tons.

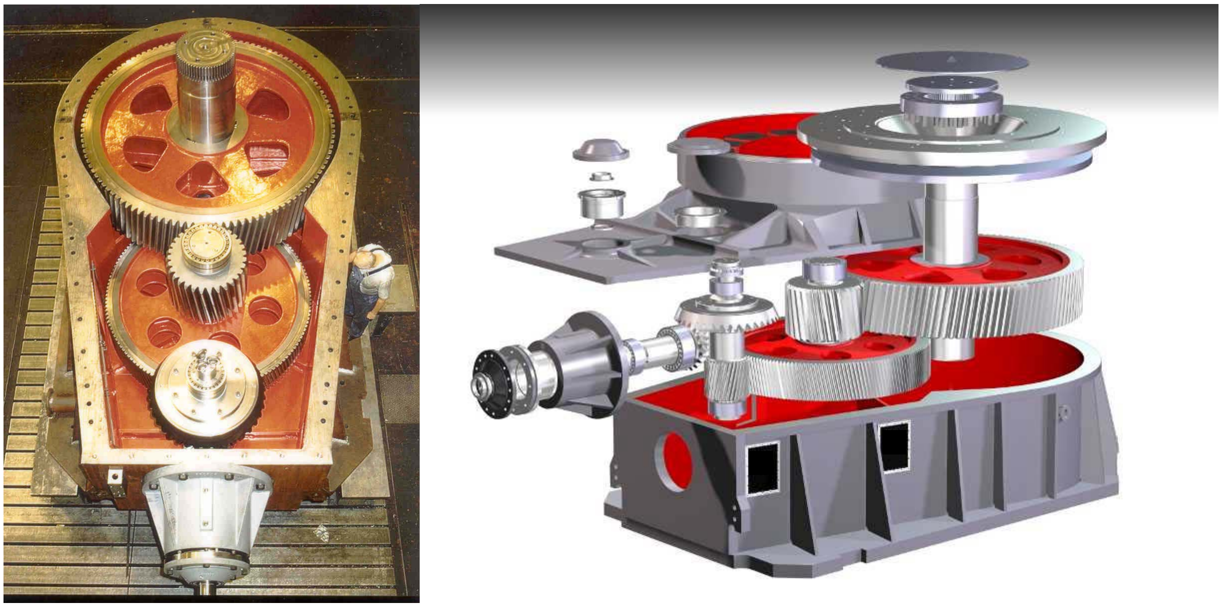


Fig. 3: First generation vertical mill gearboxes

2.2 Today's standard gearbox design

After reaching the limits of the first gearbox generation, the next logical step was to introduce a planetary stage instead of the two helical gear stages. The planetary stage is well known from many different industrial applications, with its great advantage of less material input at the same power. This design reduces the total weight, size and costs significantly in comparison with the two helical gear stage concept. This is a proven concept that has been used by all well-known gearbox manufacturers for almost 40 years. *Fig. 4* shows as an example for this concept a cross-sectional view of a gearbox with a bevel and one planetary stage. The motor power is transmitted via bevel stage and a toothed coupling to the sun pinion of the planetary stage. The planet carrier then transmits the torque to the output flange of the gearbox. This gearbox type has successfully been supplied for powers up to 5600 kW with a ratio up to 46 and bevel wheel diameters up to 2 m, but it became clear, that due to the high demanded gear ratio for this size and the resulting large diameters of the toothed parts the limit of the 2-stage gearbox generation is reached.

In the bevel gear stage the very large wheel sizes become a problem during heat treatment due to the significant amount of distortion in the hardening process. The large diameters of the wheel also increase the tilting moments as a result of the tooth forces and also increase the displacement of the engaged tooth zone due to an angular misalignment of the wheel shaft, which influences the contact pattern of the toothing.

In addition, it is more difficult to achieve a high pitting resistance due to small radii of curvature when design engineers are forced to get small pinions as a consequence of high ratios in this stage. In the planetary stage the annulus and to a certain degree the planets became larger and heavier. Such parts are difficult to purchase and to manufacture, handling and availability become a critical issue, machine capacities are often limited and accuracies decrease with the increasing size.

Besides all the technical reasons a two-stage gear solution is not an economical optimum beyond a certain size. All the above clearly shows the demand for a new gear concept.

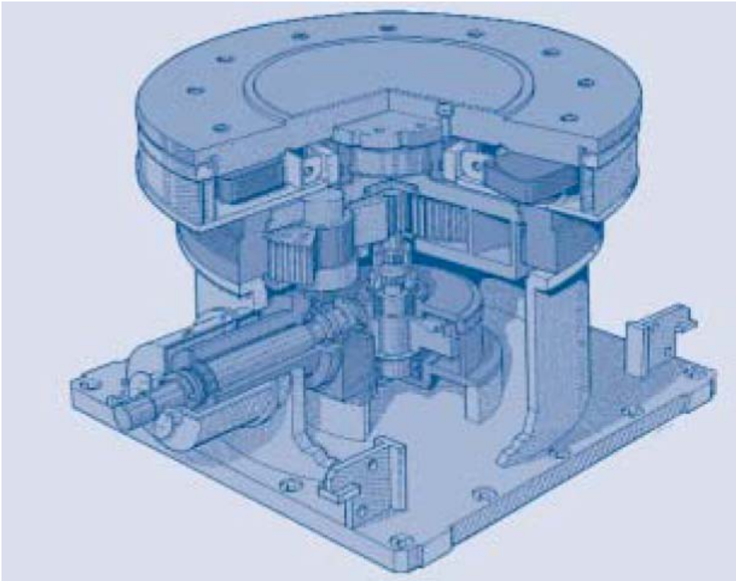


Fig. 4: Example of two stage (bevel/planetary) gearbox

2.3 Gearbox concepts for the latest mill generation

Gearbox manufacturers are looking for new concepts that are suitable for gear ratios greater than 45 at a rated power of up to 7,000 kW. Two different solutions for a 3-stage gearbox are on the market today. A innovative new concept is going to be presented in the following sections.

3 Development of a new 3-stage gearbox concept

3.1 Requirements on a new gearbox generation

Before the start of the development process itself, a market research project was carried out, and the latest developments were discussed in detail with the mill manufacturers. These analyses and requirement for cost reductions formed the boundary conditions for the design of the new gearbox.

From the customer's point of view, the following requirements were of primary interest:

- High availability of the plant
- Low maintenance costs
- Short installation times
- Customer-specific design
- High efficiency

In addition to the customer requirements, the following additional aspects have been taken into consideration for the design:

- Weight reduction
- Standardisation of the components
- Reduction transport cost
- Reduction of order processing time

These criteria formed the foundation based on which the development of a new gearbox concept was started.

3.2 Different concepts

The most important aspects of a gearbox concept are the arrangement of the gear wheels, the casing and the axial bearing.

3.2.1 Arrangement of the gear wheels

From a large number of possible arrangements, three technically feasible alternatives emerged as the most promising ones as depicted in *Fig. 5*.

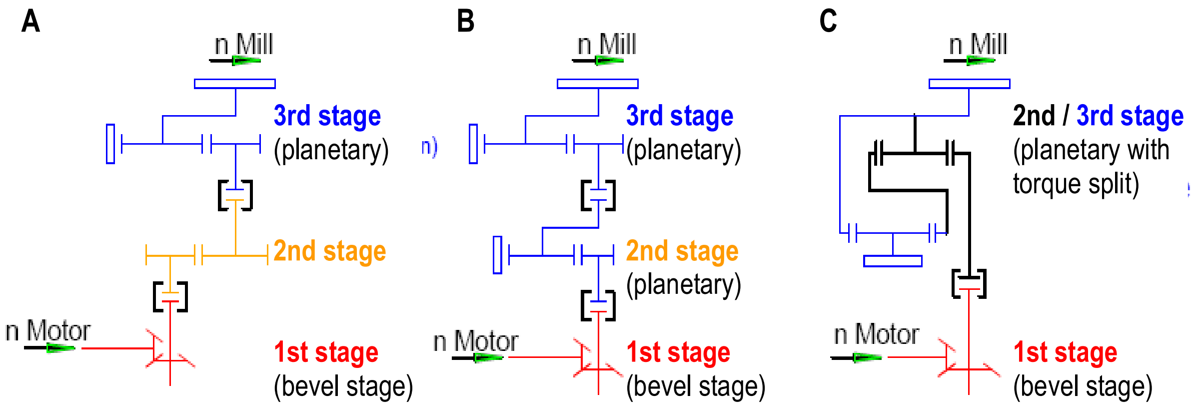
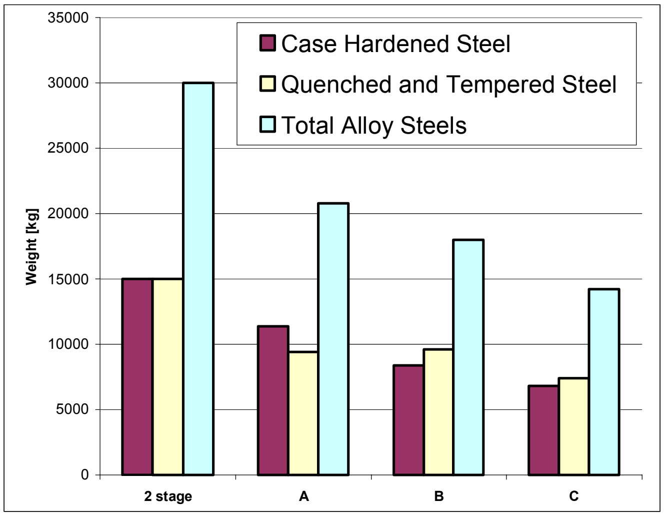


Fig. 5: Gear wheel arrangement alternatives

Tab. 1 shows the optimum and maximum possible gear ratio ranges for the different concepts. It is obvious that the PV concept is the most appropriate design for the required ratios.

		Bevel Stage	Single Helical	Single Helical	Planetary	Planetary	Planetary w/torque split	Ratio range
1st generation	optimal	2 - 3.5	3 - 5	3 - 5				18 - 125
	max.	- 5.5	- 6	- 6				- 200
Today's standard concept	optimal	2 – 3.5			5 - 9			10 - 31
	max.	2 - 5.5			3 - 11			- 60
New concepts: (refer to *Fig. 5*)								
A (bevel/single helical/planetary)	optimal	2 - 3.5	3 - 5		5 - 9			30 - 160
	max.	- 5.5	- 6		3 - 11			- 350
B (bevel/planetary/ planetary)	optimal	2 - 3.5			5 - 9	5 - 6.5		50 - 205
	max.	2 - 5.5			3 - 11	3 - 6.5		18 - 350
C (bevel/double planetary with torque split)	**optimal**	**2 - 3.5**					**12 - 22**	**24 - 77**
	max.	- 5.5					10 - 30	20 - 105
Demanded gear ratio range for different mill types between 2500 kW and 7000 kW:								**30 - 65**

Tab.1: Comparison of gear ratios



Tab. 2: Comparison of steel weights in different gearbox designs

In addition to the gear ratio, the weight of the required case hardened and quenched and tempered steel is also a good measure for comparing the gearboxes with regard to manufacturing costs. *Tab. 2* shows that the C concept is the most cost-effective option.

In the next step, draft drawings were designed for the A and the C concept (see *Fig. 6*), in order to examine the two concepts in more detail with regard to the cost and manufacturing aspects. The B concept was rejected at an early stage, since a low overall gearbox height was not compatible with a flexible design of the housing diameter.

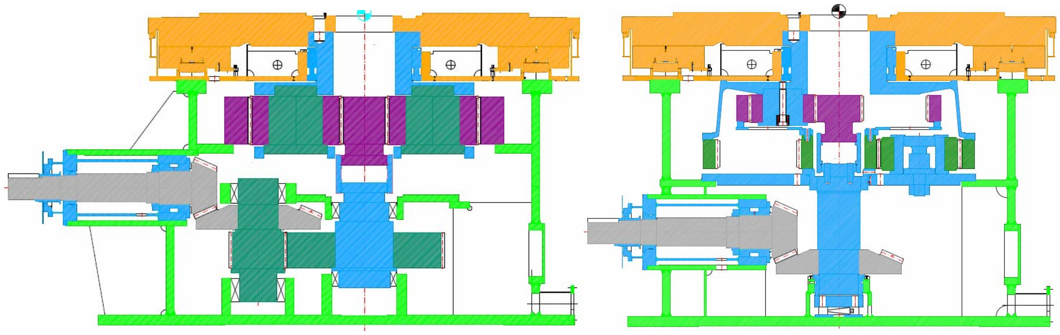
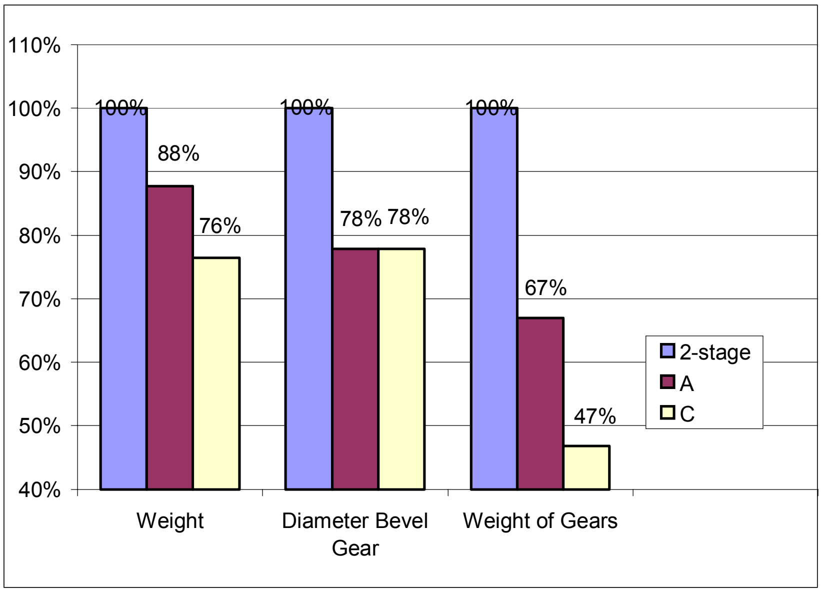


Fig. 6: Draft drawings of the A and C gearbox concepts

Once the costs of the two remaining variants had been determined more accurately, it became clear that the C concept was the one with the lowest overall weight and the lowest costs, despite its higher number of components. The disadvantage of a larger number of components is more than compensated by a reduction of the size of these components. Smaller parts allow for cost savings in procurement, transport and for higher manufacturing quality especially in the upper range of sizes. Some of the key dimensions of the two gearbox concepts are shown in *Tab. 3*. A 2-stage gearbox as described in *Section 2.2* is used as reference.

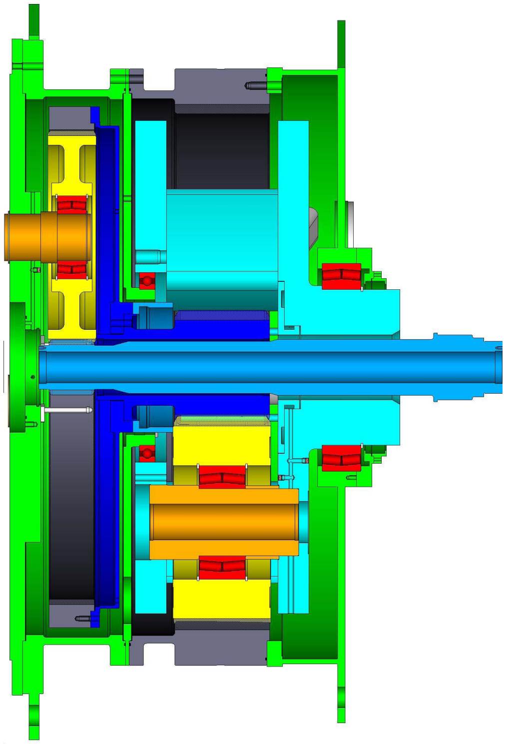


Tab. 3: Comparison of key dimensions

After this evaluation it was decided to proceed with the C concept and the actual development work on the gearbox concept was started.

3.3 The C concept to become the new 3-stage gearbox

The C gearbox concept has over many years successfully been used in gearboxes for marine, turbo, wind power and mill applications. Among others, it has been applied in central drive roller mill gearboxes (see *Fig. 7*) and, in recent years, in the main drive of wind turbines (see *Fig. 8*).



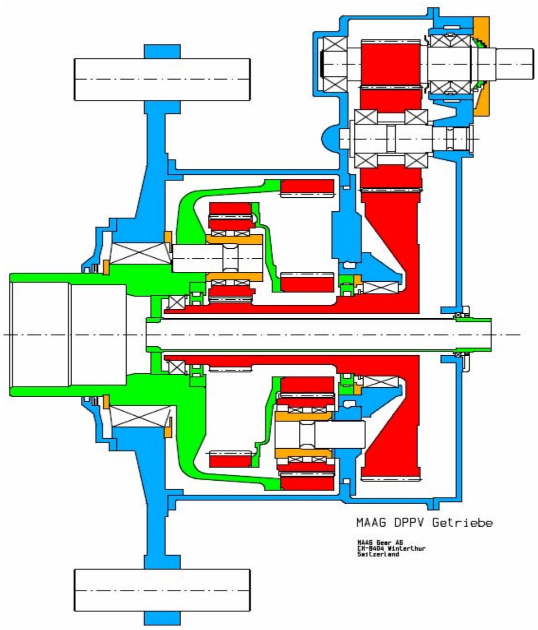


Fig. 7: C concept implemented in central drive roller mill gearbox

Fig. 8: C concept implemented in wind turbine gearbox

Fig. 9 shows the implementation of the C concept in a vertical mill gear schematically. The sun pinion of the 1st stage is driven by the motor via bevel gear stage and distributes the power to 3 planets. Here, the torque is split and while 100% of the torque is directed through the planet bolts and the planet carrier of stage 1 to the output shaft, another portion of the torque is guided through the 2nd planetary stage to the output shaft.

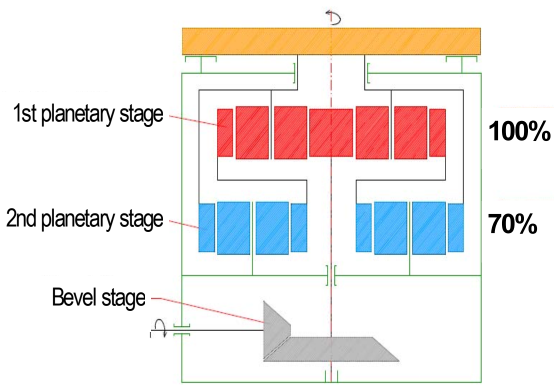


Fig 9: C concept implemented in the new gearbox

The power split is calculated according to *Equation 1*. It is obvious that power distribution between 1st and 2nd planetary stage only depends on the ratio of the number of teeth, and, thus, is geometrically defined. In the new gearbox, the number of teeth has been selected in a way that the 2nd planetary stage takes approx. 70% of the load transmitted. The great advantage of this gear arrangement now is that the 2^{nd} stage can be designed for only 70% of the output torque and therefore with dimensions that are considerably smaller than those of a conventional planetary stage transmitting 100% of the load. The torque split stage closes the gap between a single-stage and a double stage planetary gearbox in an ideal way: the single-stage planetary gearbox with a gear ratio of 10 max. and the two-stage planetary gearbox, which is only suitable for a gear ratio of approx. 25. The torque split stage represents the most compact gearbox solution in a gear ratio range of 12 - 25.

$$P_{II} = P\left(1 - \frac{\frac{z_{3I}}{z_{1I}} + 1}{i_{ges}}\right)$$

Equation 1: Load distribution between 1st and 2nd planetary stage

3.4 Flexpin

The C concept is furthermore characterised by a lower weight and a more compact design than in the concepts commonly used nowadays. In order to achieve a balanced load distribution to the five planets of the 2^{nd} stage, an additional flexible element is used, the so-called “Flexpin”. The Flexpin was originally developed in the 1960ies, and has been successfully used in hundreds of gearboxes.

In most planetary gearboxes, the sun pinion is used as adjustable element in order to ensure a good load distribution between the planets, i.e. to compensate the manufacturing tolerances. This is, however, an insufficient compensation, since no movement and thus no compensation is possible between planet and ring gear, which are both rigidly mounted (see *Fig. 10*). Furthermore, the sun pinion compensation only works for planetary stages with 3 planets. If more planets are used, this set-up yields a statically indeterminate system that is unable to guarantee complete compensation and balanced distribution of the load.

In the Flexpin solution the sun pinion and the ring gear are rigid while the planets are mounted to the planet carrier using flexible bolts - the Flexpins. This way, each planet can individually adapt itself in radial direction between the two central gears (see *Fig. 10*). In addition to the radial direction, movement is also possible in the circumferential direction, which leads to an optimal load distribution between the individual planets. The Flexpin has been designed in a way that the movement of the planets is always parallel to the planet carrier. So, tilting is not possible, and the planet always carries the load over its full width. The effectiveness of this kind of load compensation and the excellent width-distribution of the load has been confirmed on the test bed and in the field.

In addition to good load compensation, however, the “Flexpin” also contributes to minimising the load peaks. High load peaks that arise from the very dynamic milling process are significantly reduced by the additional flexibility. This is particularly favourable for the sensitive bevel gear stage.

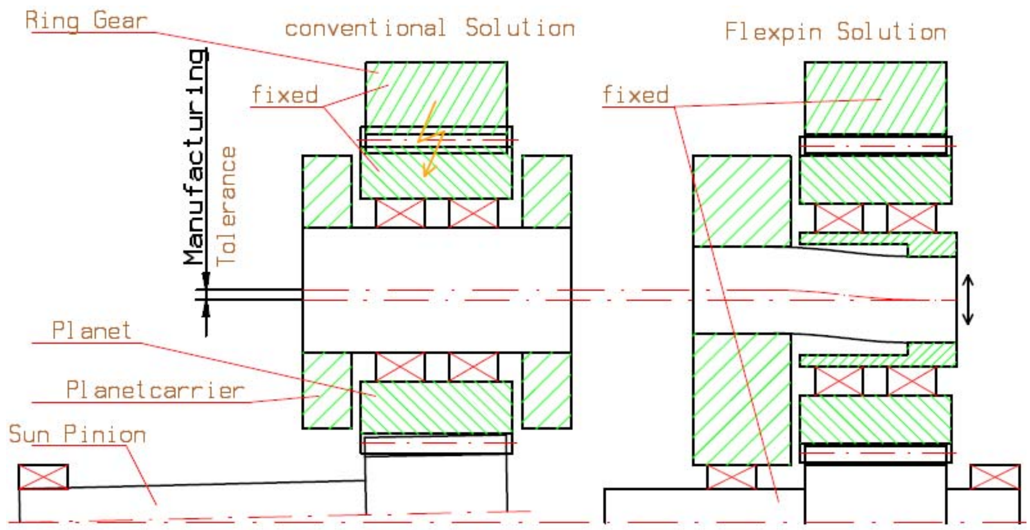


Fig. 10: Conventional and Flexpin planet mounting system

3.5 Standardisation, Modular system

A further very important requirement on the new development of the gearbox was a standardisation of the components and complete modules that would be as comprehensive as possible. The primary reason for standardisation requirement was a reduction of the costs, which could be achieved through the following:

- Reduction of the variety of parts, and thereby the storage and parts administration costs
- High production quantities leading to better purchasing conditions
- Better planning of the logistics and the stock holdings
- Reduction of order specific design costs
- Faster drawing up of quotations.

If the demand for gearboxes of this kind is considered with regard to the individual sizes, it must be stated that it will continue to involve individual or small series manufacture in the future. In addition to the low quantities involved, there is also a list of other reasons that speak out against a high level of standardisation, such as the different gear ratios, table loads, table diameters and customer wishes. In view of this wide range of requirements, the introduction of a comprehensive standardisation would appear to be a difficult task. It is at exactly this point that the C concept is able to take full advantage of its strengths with regard to compactness. An overview of the standardisation is shown in *Fig. 11*. All the components of the torque split stage, the bevel gear stage (with the exception of the bevel gear and pinion) and all the bearings have been standardised for each size of gearbox. This covers all the forged parts, which require the most procurement time. The required variation in the overall gear ratio is provided in the bevel gear stage. This is necessary, as the gear ratio varies from project to project depending on the type of mill. The axial bearing with its drive flange can be mentioned as a further module. Here, a limited number of sizes are available in order to cover the different table diameters and loads. The axial bearing segments themselves are standardised, and can be used for all gearbox sizes. The last important element is the casing, which is custom manufactured. In this way, with a single component, it is possible to offer the customer the greatest possible freedom with regard to diameter, overall height and axis height (see also *Section 3.7*).

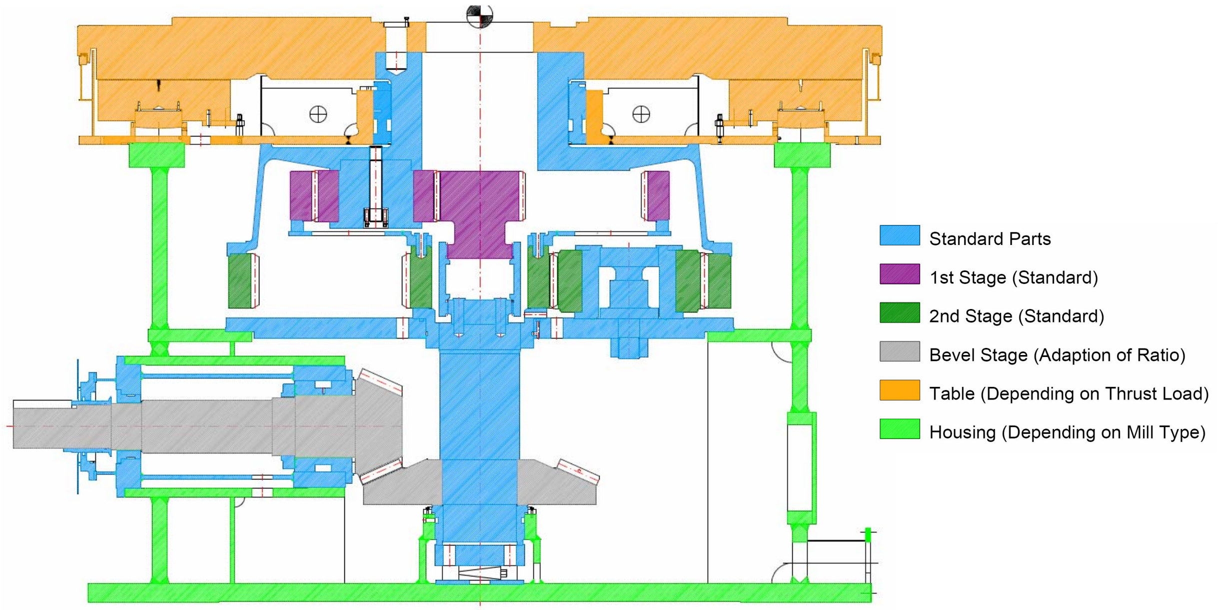


Fig. 11: Cross-section of the new 3-stage gearbox

This standardisation concept provides an optimal combination of the requirements of Production, Purchasing and Logistics on the one hand, and the greatest possible customer benefit on the other.

3.6 Sliding bearings in the bevel gear stage

The bearings of the bevel gear stage have been newly developed. Although a sliding bearing has been used in the planetary stage for many years, the bevel gear stage has so far been fitted with roller bearings. Different variants of sliding and roller bearings have been examined from both a technical and economic point of view regarding their applicability in the bevel gear stage. Since the bearings have a considerable influence on the load pattern characteristics of the bevel gears, the available options have been examined in detail. The roller bearings on the one hand have the well-known disadvantages of a limited life cycle and a low damping factor. The sliding bearing on the other hand need more radial clearance and have a higher power loos due to friction. Bearing clearance is known to be relevant for the load pattern characteristics of the bevel gears and the power loss is minimised by chosing the appropriate clearance for the sliding bearing and the oil feed. At the same time, the roller bearings are not held in stock by the manufacturers for very large bearing dimensions. This causes very long lead times, and the gearbox manufacturer is forced to maintain his own stock of bearings. In case of emergency, sliding bearings can be repaired on-site by field service engineers in a short time, so that downtime of the system can be minimised. All the above said led to the decision to use sliding bearings for the complete gearbox which, largely due to a theoretically unlimited life cycle of the bearings, makes a considerable contribution to the operational readiness of the gearbox.

In addition to the sliding bearings, a system for adjusting the bevel gear stage has been developed. The mechanism allows the field service engineer to set or re-adjust the tooth pattern and the gear backlash both during assembly and on-site. In addition, the pinion including all bearings can simply be removed from the casing without the need tof remove the gearbox from the mill.

3.7 Casing

Another important aspect of the development work is the gear casing. It is not only accommodating the gears and their forces, but also transmitting the milling forces to the foundation. The milling forces caused by the milling rollers pressing against the milling table are first transmitted from the table to the axial bearing segments through the flange of the driven shaft. The segments are supported by the casing which is guiding the forces to the mill foundation (see *Fig. 12*). The ideal structure for this casing would be a closed cylinder. It is, however, necessary to provide an opening in the casing for the horizontal drive shaft. This opening must be as small as possible in order to keep the stresses in the casing at minimal level. In the A concept design the opening needs to be bigger due to the bevel gear stage shifted towards the input shaft (see *Fig. 12*). The resulting stresses in the casing are increased, and additional measures must be taken, such as supports, wall reinforcement and ribbing, in order to reduce these stresses to a tolerable level.

The opening in the C concept design is small enough that the thickness of the casing wall can be uniform over the whole cylinder. The wall thickness is also designed so that additional ribbing is not required. This reduces the danger of fatigue cracks in highly stressed welding seams, and considerably simplifies design and manufacturing.

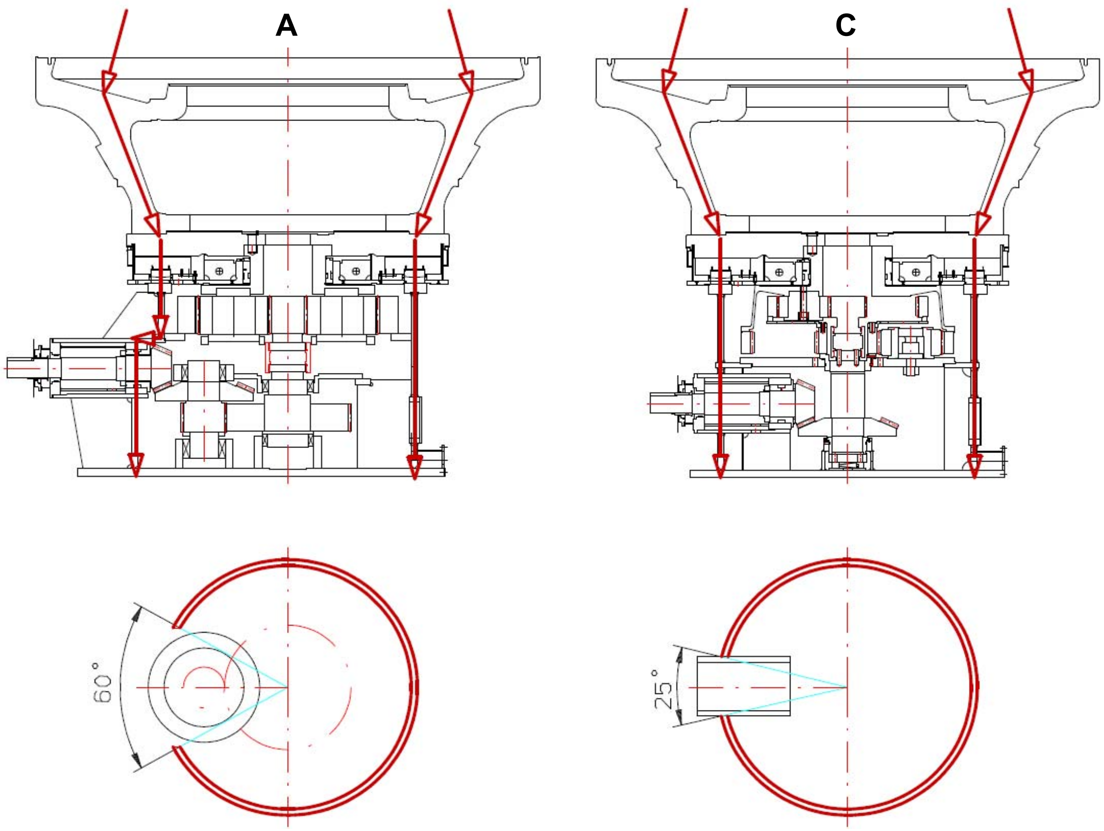


Fig. 12: Flow of forces through the gear casing

4 Conclusion

With the development of the third generation of vertical roller mill gearboxes, the C concept solution provides an innovative alternative to the solutions available on the market. The gearboxes are characterised by technical innovations that are based on tested and proven components. The new gearbox design is state-of-the-art in the industry and provides an optimum in efficiency and technical reliability for the customers and the gearbox manufacturer.