Development and application of a predesign tool for aero engine power gearboxes

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ABSTRACT

While gearboxes are enablers for new engine concepts, they introduce additional complexity in aero engine design. The additional weight of power gearboxes competes with potential weight savings in the turbine, while the size of the gearbox poses challenges for the integration in the engine. It is important to balance these effects at an early stage of development. Therefore, a predesign methodology is required to estimate the gearbox mass and size.

In this paper the development, validation and application of a predesign tool for power gearboxes named GtGearbox is presented.

First, the methodology used in this tool is presented. The results are validated and compared with internal as well as published gearbox data. The validation results are evaluated and discussed. GtGearbox is then integrated in MTU's predesign tool MOPEDS. One main application of MOPEDS is the investigation of different propulsion systems on a conceptual design level. How GtGearbox improves typical MOPEDS parameter studies will also be shown.

Keywords: Gearbox; Predesign

NOMENCLATURE

Symbols

AGBT	Advanced Gearbox Technology
BPR	Bypass Ratio
F	Force
FB	Fuel Burn
GTF	Geared Turbofan
GTlab	Gas Turbine Laboratory
K_i	Factor for various additional loads. The index i describes the kind of load. See DIN 3990 [1-3]
LPT	Low Pressure Turbine
M	Torque
MOPEDS	Modular Performance and Engine Design System
OPR	Overall Pressure Ratio
S	Safety Factor
S_o	Sommerfeld Number
Y_i	Factor describing additional effects on the bearable tooth load. The index i describes the kind of effect. See DIN 3990 [1-3]
Z_i	Factor describing additional effects on the tooth contact load. The index i describes the kind of effect. See DIN 3990 [1-3]
b	Width
d	Diameter
i_0	Base Gear Ratio
i	Gear Ratio
k	Compactness
m_n	Normal Module
n	Rotational Speed
z	Number of Teeth
σ	Tension
ψ	Relative Bearing Clearance
η	Dynamic Oil Viscosity
ω	Angular Velocity

Indices

Carrier	Planet Carrier
F	Root
H	Flank
min	Minimum
n	Normal
Planet	Planet
ref	Reference
Ring	Ring
Sun	Sun

1.0 INTRODUCTION

The application of gearboxes to change rotational speeds and transfer power from the low pressure shaft of gas turbines to the propulsors is not new. Most turboprop engines require a reduction in rotational speed due to high diameter differences between the power turbine and the propeller that result in different rotational speed requirements. In recent times, the development of more efficient ultra-high bypass ratio fans makes it difficult to find a good compromise in rotational speeds of turbine and fan. In the PW1000G (GTF™) series, Pratt & Whitney introduced a reduction gearbox to operate fan and turbine at different speeds. In the future a further increase in bypass ratio of ducted engines as well as the development of high power propellers will push the development of new power gearboxes. Furthermore, the introduction of new propulsors like counter rotating fans or open rotor engines would require a power distribution gearbox to avoid multiple or more complex power turbines.

While these gearboxes are enablers for new engine concepts, they introduce additional complexity in aero engine design. The additional weight of such a gearbox competes with potential weight savings in the turbine, while the size of the gearbox poses challenges for the integration in the engine. It is important to balance these effects at an early stage of development. Therefore, a predesign methodology is required to estimate the gearbox mass and size. Such an algorithm will then be an important component of multidisciplinary engine design systems.

The gearbox predesign tool presented in the current paper is intended to consistently extend the existing predesign procedures of DLR and MTU in order to enable a comprehensive consideration of the gearbox in the overall propulsion system simulation.

2.0 TOOL REQUIREMENTS

2.1 Gearbox selection

Ahead of the development of a gearbox predesign tool, a literature study of existing aero engine gearboxes was performed to down select the most relevant gearbox types. In [4] a detailed study on gearbox types was performed by Godston et al. for both single- and counter rotating propulsors. The study differentiated between inline and offset gearboxes and discussed different gearbox designs.

Based on the findings in [4] and supported by the consideration of the most recently implemented gearbox types in aero engines, two gearbox concepts were chosen. For ducted engine concepts as well as for counter rotating applications, a planetary gearbox was found to be the superior solution. For single rotating propellers, a wide range of very different in-service gearbox types was found. The most common gearboxes contain a reduction spur stage and a planetary gear stage.

Both gearbox stage types were selected as initial applications for the gearbox predesign tool (Figure 1). Nevertheless, other gearbox types may be added in the future.

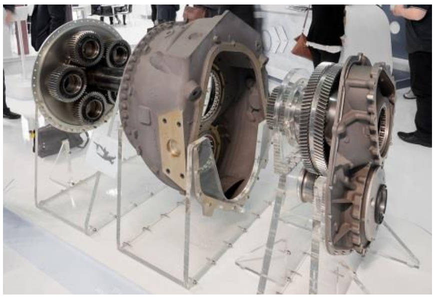


Figure 1: Selected gearbox concepts. Left: planetary gearbox (PW1000G gearbox, ©Pratt & Whitney). Right: combined spur-planetary gearbox (TP400-D6 gearbox, ©Julian Herzog)

2.2 Required capabilities

The detailed design of an aero engine gearbox is a very complex process involving multiple disciplines. Integration issues, heat management, dynamic loads, vibration, maneuver loads, bending loads on the propeller shaft (1P moments) and other effects must be evaluated carefully to ensure a reliable operation.

The introduction of a gearbox has a strong influence on other engine components. Thus it is vital to gain knowledge about the gearbox design at a very early stage of development. Due to the high number of iterations and the low depth of information, the requirements for a preliminary design tool differ from those of the detailed design. Swift computation times are needed to facilitate its integration into highly iterative processes. Furthermore the numerical stability of the design tool must allow for broad parameter variations. Therefore it is necessary to introduce simplifications in the design process. The method must have sufficient accuracy to deliver correct trends in geometry and mass after calibration. Since the method is to be used in preliminary design environments, appropriate interfaces are to be provided for integration.

3.0 METHODOLOGY

Aero engine gearboxes consist of different components. Besides the gears, various shafts, bearings and a gearbox casing are part of the gearbox assembly. The assembly strongly depends on the type of gearbox and varies a lot on existing applications. Besides of the actual gearbox, an oil system for its cooling is required.

The implemented design process can be divided into three major phases: In the first phase, an initial gearbox geometry is derived from given specifications. Missing data is estimated using correlations. In the second phase, a strength calculation is carried out for the resulting geometry. In the third phase, the results are evaluated and a variation and optimization of the geometry is performed.

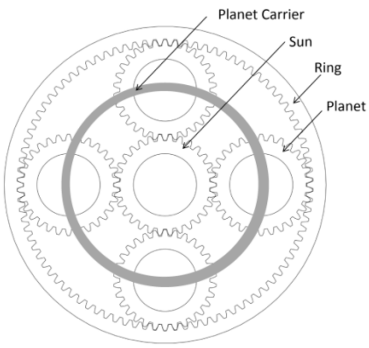


Figure 2: Planetary gearbox stage ($i_0 = -3$)

3.1 Initial Gearbox Geometry

Numerous fundamental geometrical relations between the gears can be derived from the gear ratio. For a spur stage, the gear ratio trivially describes the teeth ratio between the gears. For a planetary stage, the geometric relationships can be expressed as a function of the base gear ratio i_0 which can be described by the Willis Equation (Eq. 1) [5] as a function of the rotational speeds n of all shafts.

$$n_{sun} - i_0 * n_{ring} - (1 - i_0) * n_{Carrier} = 0 \quad (1)$$

The base gear ratio directly relates to the tooth ratio between ring and sun gear (Eq. 2). Assuming no profile shift, the diameter ratios are fixed as well as the ratio between planet and sun teeth number[1]. Neglecting losses, the torque ratio between carrier and ring is known as well.

$$i_0 = \frac{-|z_{Ring}|}{z_{Sun}} \approx -\frac{|d_{Ring}|}{d_{Sun}} \approx \frac{1}{\frac{M_{Carrier}}{M_{Ring}} + 1} \quad (2)$$

For a given number of teeth for the sun gear and the tooth size (described by the normal module m_n) all gear diameters and teeth numbers can be determined. The number of ring teeth must be rounded. To achieve integer number of planet teeth, the difference between sun and ring teeth must be even[2]. In order to ensure a uniform distribution of the planets, the sum of the teeth of ring and sun wheel must be divisible by the number of planets. This condition might be ignored for a grounded planet carrier, as no unbalance is to be expected. These rules often cause that the resulting gear ratio differs from the initial assumption.

The maximum number of planets is always used. This number is determined depending on a user-defined minimum planet distance. If no specification is given by the user, the double helical teeth are assumed to have a pressure angle of 20° and a helix angle of 30° as proposed in [5].

After the design of the planet gears, estimates on bearing sizes and masses are made. Typically different kinds of bearings are used. Journal bearings are suitable for gearboxes without centrifugal and axial forces, e.g. for star configurations. If centrifugal forces, axial forces or possible bending is involved, ball bearings, roller bearings or spherical roller bearings are used.

For multistage gearboxes a connection shaft transfers power between both stages. Within a planetary stage, the planet carrier collects the torque from all planets and transfers it to a propeller shaft (or to the casing if grounded), while a ring shaft connects the ring gear. The initial geometries are based on the gear dimensions and user inputs.

[1] In DIN 3990, diameters and number of teeth for internal gears are defined to be negative. To avoid confusion, absolute values are used here.

[2] This condition could be altered by applying profile shift to the planet. For simplification purposes, GtGearbox assumes no profile shift.

The casing, simplified as a hollow cylinder, is sized dependent on the other gearbox components. The wall thickness and the material are to be specified by the user. Integration issues that will influence the casing design are neglected in this methodology.

3.2 Load Calculation

The occurring loads must be below the bearable loads. Several well established methods to calculate the loads for given gears are available in literature, such as ISO 6336 [6], DIN 3990 [1-3] and AGMA 2001 [7]. In this model, the methods from DIN 3990 are used.

Herein the Hertzian stress on the tooth flank is described as

$$\sigma_H = Z_B * \sigma_{H0} * \sqrt{K_A * K_\gamma * K_V * K_{H\beta} * K_{H\alpha}} \quad (3)$$

The comparison between allowed and maximum stress is described by the safety factor against pitting S_H:

$$S_H = \frac{\sigma_{H,lim}}{\sigma_H} * Z_N * Z_x * Z_L * Z_V * Z_R * Z_W > S_{H,min} \quad (4)$$

The tooth root load is described as

$$\sigma_F = K_A * K_V * K_{F\alpha} * K_{F\beta} * \frac{F_t}{b * m_n} * Y_{FS} * Y_\epsilon * Y_\beta \quad (5)$$

Again the safety factor against tooth root fracture is described by:

$$S_F = \frac{\sigma_{FE}}{\sigma_F * Y_{\delta T}} Y_A * Y_T * Y_N * Y_\delta * Y_x * Y_R > S_{F,min} \quad (6)$$

The load factors Z_i, Y_i and K_i in eq. 3-6 describe different influences on the bearable and occurring loads. The K_i factors are coefficients for additional loads such as dynamic effects, misalignments and others. The factors Z_i describe effects on the tooth contact (i.e. by material pairing or roughness) while Y_i describe various effects on the bearable tooth load (i.e. by additional notches). The description of all individual load factors can be found in [1-3]. In DIN 3990 several methods to calculate the individual load factors Z_i, Y_i and K_i are described with varying complexity. Here, the methods of type “B” are used as they provide most exact results without requiring data which is usually not available during predesign.

In DIN 3990 the gear rims are not taken into account for the calculation of the tooth root load capacity. As the rim of the ring gear is very thin compared to the diameter a more detailed approach is selected by applying a methodology described in VDI 2737 [8]. Here, the ring rim elasticity is considered when calculating the safety factor for the tooth load. In existing gearboxes supporting structures may have a strong influence on the ring rim elasticity and thus on the safety factor according to VDI 2737.

For a given number of teeth, normal module, typical helix and pressure angles, material, and surface roughness, the required tooth width to comply with the required safety factors is calculated iteratively. The required ring rim thickness is determined afterwards.

The feasibility of journal bearings is verified by calculating the Sommerfeld number S_o as defined in [9] (Eq. 7).

$$S_o = \frac{\frac{F}{d * b} * \psi^2}{\eta * \omega} \quad (7)$$

Too high ($S_o > 10$) or too low Sommerfeld numbers ($S_o < 1$) indicate infeasible designs. The calculation of the Sommerfeld number requires a temperature dependent oil model to estimate the dynamic viscosity η of the oil. Herefore, the methodology of Ubbelohde & Walther [10] was implemented.

The mass and geometry of ball or roller bearings is derived by using simple correlations from [11]. An implementation of a Kriging surrogate model of a rolling bearing catalogue is currently in development and will be used to improve the feasibility check and the mass estimates.

For cylindrical shafts the wall thickness is determined using the Van Mises comparative stress and a given safety factor. Other wall thicknesses are given by the user or estimated as a function of torque.

3.3 Gearbox Design Process

For all combinations of normal module and sun or pinion number of teeth, the predesign process minimizes the gear width and the ring rim thickness in order to match the required safety factors. The number of teeth and the normal module is then varied. The values for the normal module are taken from the preferred ranges defined in [12].

As a result, multiple different designs are created that are able to fulfill all requirements but vary strongly in geometry and mass. Exemplary results are shown in Figure 3.

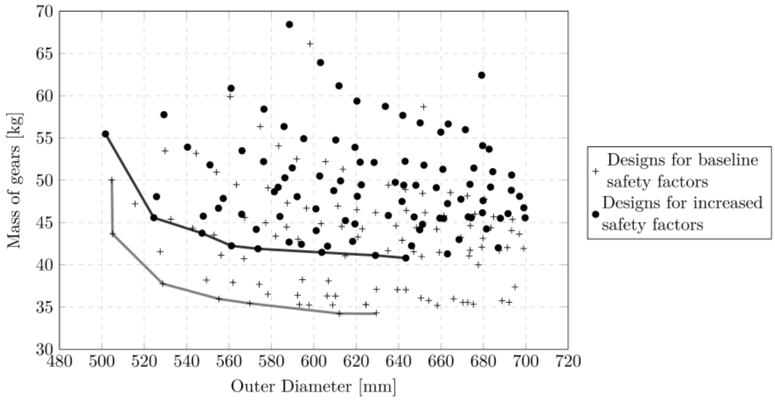


Figure 3: Exemplary Pareto fronts from variation of normal module and number of teeth (from [13], edited)

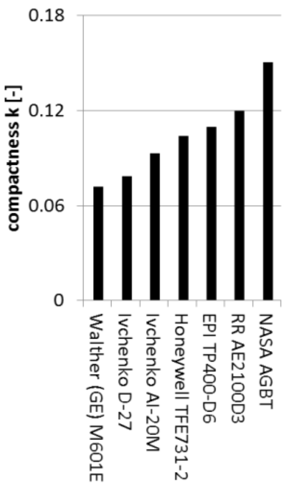


Figure 4: Compactness k of various existing planetary gearbox stages with roller bearings (from [14])

The optimum gearbox cannot be derived directly as different aspects must be considered. In Figure 3 it becomes obvious that the gearbox with the smallest diameter is considerably heavier than a slightly larger one due to its increased width. A Pareto front exists between these parameters in this example. Other aspects as gearbox width or inner diameter (to ensure a connection to a shaft) create other dependencies that must be taken into account as well. The form of the Pareto fronts may vary strongly, depending on the gearbox design (i.e. type of bearings, size of planet carrier). In particular a Pareto front between gearbox width and diameter will often occur as both parameters influence the bearable loads of a gearbox.

The manual selection of the preferred member is not suitable during automated predesign processes. Therefore, constraints for important geometric values can be set by the user to restrict the design space to certain requirements.

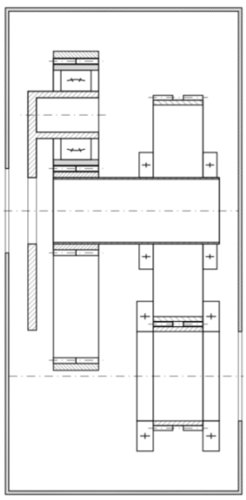


Figure 5: Combined spur-planetary Gearbox from GtGearbox

If the installation space available for the gearbox is known, maximum diameters and gearbox width can be defined within GtGearbox. Depending on the type of study, a limitation of absolute geometry values is often not suitable as intended size variations will occur and geometric limits must be constantly adapted. Therefore in [15] the compactness k, which describes the ratio of the gear rim width and the outer diameter of the ring gear is introduced as an additional parameter. An analysis of the compactness of existing planetary stages with roller bearings is shown in Figure 4. Existing planetary stages with journal bearings are more compact. Due to very poor data availability in the public domain no reliable statistics can be shown for this.

After the definition of the geometric constraints, GtGearbox will derive the lightest gearbox that fulfills all requirements. The resulting gearbox design is visualized as a 2D plot in axial and radial direction. An exemplary result for a spur-star gearbox in axial direction can be found in Figure 5.

4.0 VALIDATION

To validate GtGearbox internal as well as published gearbox data is used. The validation results are evaluated and discussed in the following.

4.1 NASA Advanced Gearbox Technology (AGBT)

In the 1980s an advanced 13,000 horsepower differential planetary gearbox for a future propfan propulsion system was developed by the National Aeronautics and Space Administration (NASA) in cooperation with the Allison Gas Turbine Division. In [16] the geometry and the load case of the gearbox are described in detail. As the AGBT gearbox was designed for rig tests, an overall comparison of its mass may be misleading. Nevertheless, the published gearbox component geometries are highly suitable for validation purposes. The cross section of the gearbox can be found in Figure 6. The gearbox consists of four planets with double helical gearing with a helix angle of $\beta = 26°$ and double row cylindrical roller bearings. The normal module of the gears is $m_n = 3.63\ mm$.

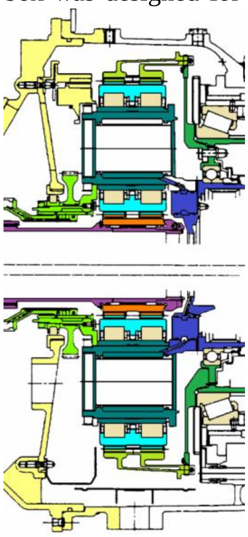


Figure 6: NASA AGBT gearbox from [16], edited

4.1.1 Recalculation

In a first step the known geometry is used to validate the load calculation of GtGearbox. Therefore the transmitted power, the shaft speeds, the materials and all available geometries are set as an input for GtGearbox and the safety factors based on DIN 3990 and VDI 2737 are calculated (Table 1).

Table 1: Calculated safety factors against pitting S_H and against tooth root fracture S_F for given geometry of the NASA AGBT gearbox

Gears in mesh		S_H	S_F
sun-planet	sun	1.356	1.926
	planet	1.356	1.340
planet-ring	planet	2.704	1.334
	ring	2.704	1.633

The results match the recommended minimum safety factors from literature ([9]) very well. Therefore, the load calculation implemented in GtGearbox is well suited to determine and evaluate the occurring loads for this gearbox.

4.1.2 Predesign

To validate the design process of GtGearbox, an approach to redesign this gearbox is initiated. Herefore, the same load case is used. Assumptions on the material of the gears, the helix angle, the wall thickness of the planet carrier and the type of the planet bearing are made. Additionally the previously determined safety factors as well as the ratio of rim thickness and normal module for sun gear and planet gear, the ratio of the gap in the center of the teeth and the normal module are adopted.

In the gearbox predesign process of GtGearbox the number of teeth of the sun gear z_{sun} and the normal module m_n are varied. Initially, no geometric constrains are defined. The GtGearbox calculation results in 283 gearbox designs which differ strongly in mass, width and height. In Figure 7, all resulting gearbox designs are shown as a function of their compactness.

The rise of the gearbox mass with decreasing compactness can be related to an increasing mass of the planet bearings. For a compactness > 0.16, the gearbox mass gradient vanishes and gearbox mass becomes almost independent of the compactness.

As described in chapter 3, the lightest design is selected by default.

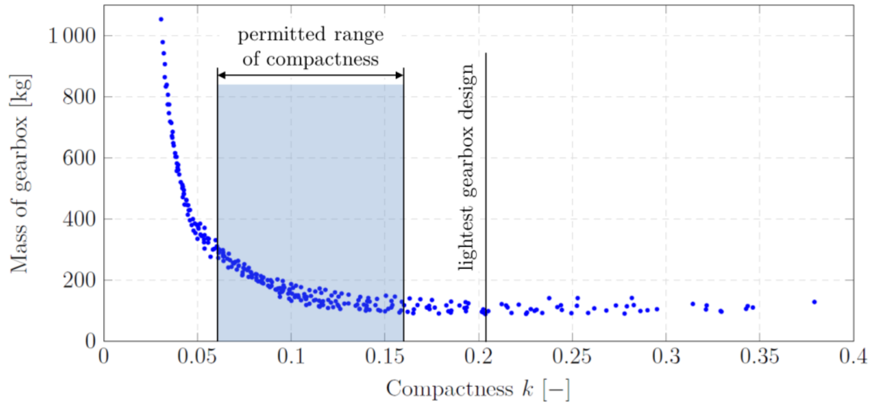


Figure 7: Mass of different gearbox designs dependent on their compactness k

The lightest gearbox has a compactness $k = 0.2036$, which differs from the compactness of the real NASA AGBT gearbox ($k_{AGBT} = 0.15$). It has a smaller diameter but a higher width than the reference. A limitation of the maximum gearbox width or of the compactness would set the best member closer to the reference gearbox. A limitation of the compactness according to Figure 4 ($0.06 \leq k \leq 0.16$) improves the conformity with the reference geometry significantly.

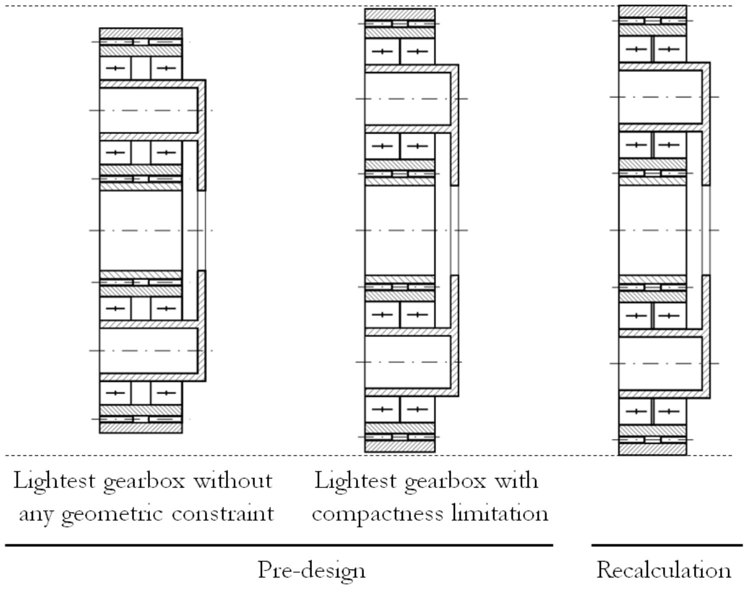


Figure 8: Comparison of predesign results and the recalculation of the NASA AGBT gearbox

Figure 8 shows the different results of the lightest gearbox design in comparison to the recalculated NASA AGBT gearbox.

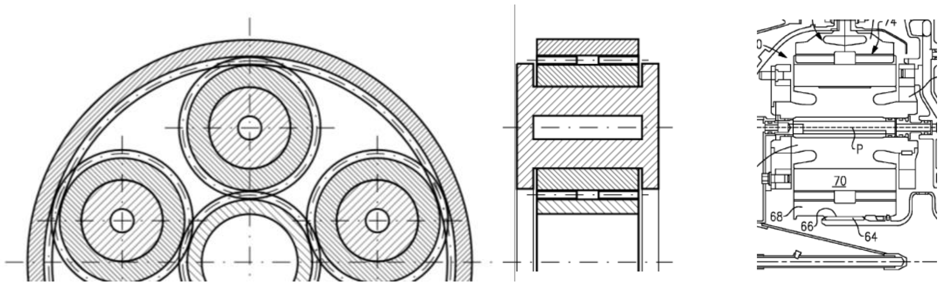
Table 2: NASA AGBT gearbox geometry and relative deviation of the lightest gearbox design within the permissible compactness range

Parameter		Reference (AGBT)	Relative deviation of GtGearbox
Normal module	mm	3.63	− 3.54 %
Number of sun gear teeth	−	36.00	2.78 %
Base gear ratio	−	3.67	− 0.49 %
Number of planets	−	4.00	0.00 %
Sun gear rim width	mm	86.00	3.44 %
Sun gear pitch circle diameter	mm	145.30	−0.84 %
Ring gear rim thickness	mm	15.06	− 2.65 %
Ring gear outer diameter	mm	563.20	0.10 %

In Table 2 the relative deviations of the lightest gearbox design within the permissible compactness range in relation to the NASA AGBT gearbox are shown. The maximum geometrical deviation of 3.54 % is mostly caused by the normal module of the AGBT gearbox, which is not within the regarded preferred range of DIN780 [12].

4.2 Other Aero Engine Planetary Gearboxes

In addition to the NASA AGBT gearbox three planetary gearboxes in star configuration with fixed planet carriers of modern aero engines were assessed with GtGearbox, henceforth referred to as GB I, GB II and GB III. Each of them consists of five planets with double helical gearing and journal bearings. The modelling of the planet carrier was derived from published illustrations of modern aero engine gearboxes in [17].



(a) (b)

Figure 9: Generic modelling of the planet carrier in GtGearbox (a) based on published illustrations of modern aero engine gearboxes (b) [17]

In a first step, similar to the approach in chapter 4.1.1, all safety factors of GB I, GB II and GB III are calculated based on available internal geometric data. Again a good conformity with literature was observed.

By applying the methodology described in chapter 4.1.2, the design of the three gearboxes is then derived by GtGearbox for given load cases and safety factors.

Table 3: Relative deviation of the lightest gearbox design within the permitted compactness range in relation to the respective reference gearbox

	GB I	GB II	GB III
Normal module	− 1.84 %	− 1.62 %	9.86 %
Number of sun gear teeth	2.27 %	3.13 %	−9.38 %
Base gear ratio	0.55 %	0.06 %	0.21 %
Number of planets	0.00 %	0.00 %	0.00 %
Sun gear rim width	0.50 %	−1.06 %	3.31 %
Sun gear pitch circle diameter	0.35 %	1.50 %	−0.47 %
Sun gear rim thickness	− 1.57 %	−1.35 %	9.92 %
Ring gear rim thickness	0.04 %	−0.28 %	− 3.73 %
Ring gear outer diameter	0.63 %	1.00 %	−0.31 %

A compactness limitation close to the reference design ($0.95 \cdot k_{ref} \leq k \leq 1.05 \cdot k_{ref}$) was applied. The relative deviations of the lightest gearbox design in comparison to its respective reference gearbox are shown in Table 3.

The geometries of GB I and GB II are reproduced with only minimal deviations. Although the lightest gearbox design of GB III has small deviations regarding the outer dimensions (gear width, ring diameter), there are slightly larger deviations regarding the normal module and the number of sun gear teeth. The greater normal module results in an increased sun gear rim thickness which reduces the inner diameter of the sun gear. An additional limitation of the minimum inner sun diameter would improve this result. The mass deviation between the calculated and reference designs was found to be below 10%.

5.0 APPLICATION

5.1 GtGearbox in the Overall Predesign Context

The main characteristics of an engine design are already fixed in the conceptual design phase. The major task during this phase is therefore to find the optimum engine cycle for a specific set of boundary conditions. Besides financial advantages engine designs are increasingly influenced by environmental factors. The fulfilment of goals such as defined within ACARE by the European Communities will have a high impact on future engine developments. Therefore it is essential not only to account for thermodynamics but also for disciplines such as weight, noise, emissions and costs already within the conceptual design phase [18]. The main objective of the conceptual design phase is to identify these designs which are advantageous for the specific engine requirements. Another objective of the conceptual design phase is the refinement of the design selected to maximize the overall system benefit.

Parameter studies help to identify these engine designs. During these studies parameters such as the overall pressure ratio (OPR) and the bypass ratio (BPR) are varied over a specified range. Every calculation point of the study describes a single engine design. As the geometry of the engine is affected by the variation of the OPR and BPR the weight changes as well. This in turn has an impact on parameters such as the aircraft mission fuel burn. Fuel burn is one of the main parameters of interest during the conceptual design phase [19].

To be able to perform studies like the ones described above multi-disciplinary engine design tools have been created worldwide.

DLR has started to develop a preliminary design process for innovative engine concepts. This highly iterative multi-fidelity process involves several experts and simulation tools from different disciplines [20]. The collaborative process architecture is based on a central data model approach to meet the challenges of data management and data exchange. The process itself is managed and monitored via the GTlab framework [20, 21]. Based on its modular structure, the framework enables the subsequent extension of data structures so that new engine components and disciplines can be added.

MTU started the development of its preliminary multi-disciplinary engine design tool MOPEDS (Modular Performance and Engine Design System) in the early 1990s. The main tasks of this tool is to simulate the behaviour of existing engines, e.g. within competitive studies or to conduct parameter studies aiming to find the optimum engine cycle for future propulsion systems considering all interdisciplinary dependencies. The prototype of the tool was described by Schaber [22] in 2000. The basic technical structure of MOPEDS was already presented by Schaber et al. [23] and Jeschke et al. [18] in 2002. MTU has constantly improved and further developed the program system [19, 24]. Today MOPEDS is one of the main tools in MTU’s advanced programs department [25]. Given the rising importance of gears it seems only natural to also integrate a more in-depth analysis and design tool for gears. For that task GtGearbox was chosen. MOPEDS automatically provides the input for the GtGearbox calculation consistent with the engine cycle, starts GtGearbox and retrieves the results, especially feasible gear ratio, gearbox mass and installation space.

5.2 Application Study

To demonstrate the capabilities of GtGearbox in the context of a propulsion system predesign tool a typical parameter study varying the bypass ratio BPR and the overall pressure ratio OPR was performed in MOPEDS for a generic geared turbofan design. For $OPR = const.$ and increasing BPR the gear ratio is rising due to lower required corrected tip speeds of the fan which as consequence require lower fan shaft speeds. For $BPR = const.$ and increasing OPR the required gear ratio in this study is rising due to an increase in the IPC total pressure ratio, higher required LPT power and consequently a higher required LPT speed assuming constant aerodynamic LPT loading and stage number [15]. The resulting required gear ratios i for each combination of OPR and BPR are depicted in Figure 10.

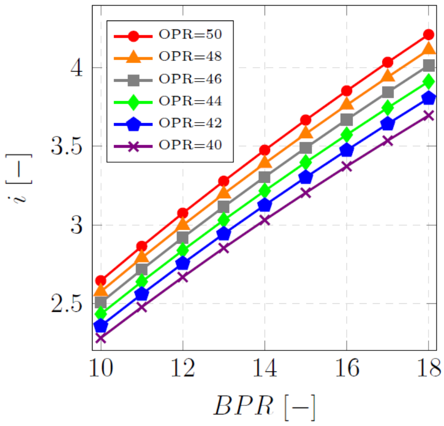


Figure 10: Required gear ratio i depending on BPR and OPR

The required gear ratio may not be achieved, as integer number of teeth within the gearbox is required. A deviation in the magnitude of 0.5% - 1% may occur, depending on the number of teeth. A pre-investigation showed that the impact of adjusting the fan speed to reach feasible gear ratios on the cycle is minor, Figure 11. Consequently an adjustment of spool speeds during the parameter study to account for feasible gear ratios did not need to be considered.

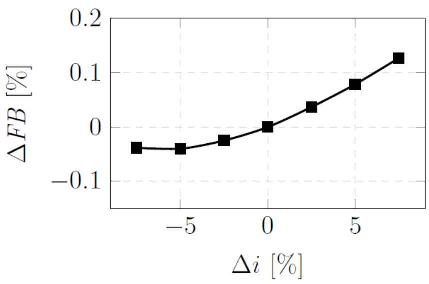


Figure 11: Impact on fuel burn due to a variation of fan shaft speed

However, in each point of the parameter study GtGearbox is used to determine the mass and the installation space of the required gearbox. Due to the large range of resulting gear ratios, different gearbox types are examined. For low gear ratios $i < 3.5$ a planetary gearbox in star configuration with the fan connected to the ring is considered ($i = |i_0|$). For higher gear ratios $i > 3.3$ a gearbox in planetary configuration (rotating planet carrier, $i = |i_0| + 1$) is investigated. The concepts differ in their mechanical integration as well as in their effect on the other engine components due to the different rotational directions. Furthermore, the gearbox design itself differs a lot. The assumptions for the star configuration are adopted from the validation case in chapter 4.2, while the assumptions for the planetary configuration are based on the geometry of the first stage of a modern turboprop gearbox. While the star configuration is assumed to have journal bearings, the centrifugal forces created by the rotating planet carrier make it difficult to use journal bearings for the planetary configuration. Therefore spherical roller bearings are assumed for this variant. Furthermore, an adapted planet carrier design is assumed.

In Figure 12 the resulting gearbox mass for each design is depicted as a function of the gear ratio i. The results are calibrated to a given reference design (i=3.4, star configuration). Apart from the GtGearbox results, the calibrated outcome of gearbox mass correlations from literature [26, 27] and a simple correlation ($mass = f(torque)$) are shown.

The evaluated methodologies show consistent results, the observed deviations are within $\pm 10\%$ for the regarded parameter range. The uncalibrated mass derived by GtGearbox show a very good agreement with the correlation method according to [26].

The GtGearbox results show an offset and a different gradient between the two fundamentally different gearbox configurations, since both configurations differ in their mechanical design. The other correlations to assess the gearbox mass are unable to take this effect into account. The results of GtGearbox show that for the given assumptions the mass of the planetary configuration is slightly lower but has a steeper gradient.

However, the results depend on the assumptions made for each configuration (i.e. rim thicknesses, or mechanical parts). To compare both concepts in general, a more detailed study would be required.

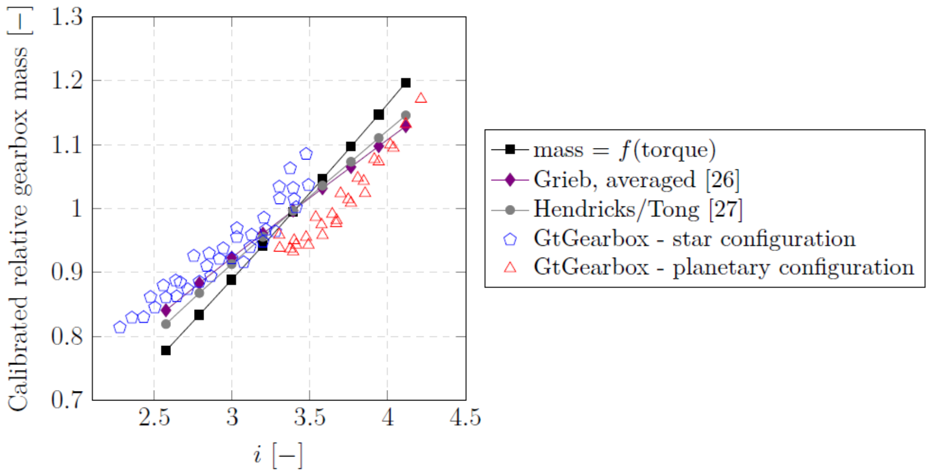


Figure 12: Calibrated gearbox mass as a function of gear ratio for both gearbox configurations. Results from GtGearbox and from various mass correlation methods.

As part of the total engine weight the gearbox mass calculated in GtGearbox affects the fuel burn (Figure 13, left). For each combination of OPR and BPR the resulting GA of the engine with its gearbox can be plotted. This allows an assessment of the required and available installation space, leading to geometric constraints for the gearbox and other engine components (Figure 13, right).

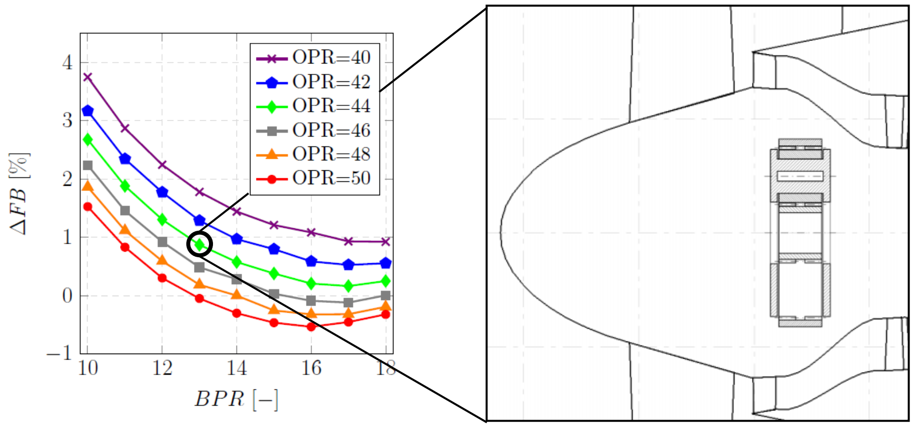


Figure 13: Delta fuel burn dependent on OPR and BPR (left) and illustration of the gearbox design within the general arrangement of the engine for $OPR = 44$ and $BPR = 13$ (right)

6.0 CONCLUSIONS AND OUTLOOK

In this paper the development of a predesign tool for aero engine power gearboxes is presented. It is based on existing load capacity methods and is tailored to the requirements of the preliminary engine design. The tool GtGearbox is capable of designing combined multistage (spur + planetary) gearboxes for propeller engines as well as planetary gearboxes for ducted fans or counter rotating propulsors. A single objective optimization of the gearbox mass is not sufficient as Pareto fronts between important design values (i.e. mass, width, diameter) may exist. Therefore geometric constraints are introduced to limit the design space of the gearbox.

The validation of the tool GtGearbox was performed for two different existing use cases. Data derived from the advanced gearbox technology programme of NASA (AGBT) [16] was used to check the calculated safety factors for the various load capacities. The results match the recommendations for gearbox designs very well. For given safety factors the optimized gearbox design was compared to the geometry of NASA's AGBT. Without any geometric constraints, the resulting gearbox was found slightly smaller in diameter but larger in width than the reference. By limiting the width to diameter ratio of the gearbox the results improve and only minor differences to the reference are found.

As a second validation case, three gearbox variants of a modern aero engine were designed with GtGearbox and compared to the real geometries. The GtGearbox results differ by less than 5% for two of the three variants in the main geometric sizes. For the third variant, deviations of less than 10% can be observed. The calculated gearbox masses deviate from the reference by less than 10% for all gears.

The application study demonstrated that the adjustment of spool speeds to feasible gearbox designs does not significantly improve the accuracy of cycle studies as this effect is found to be small.

The application of the described tool leads to an improvement in the level of detail of the gearbox mass assessment. The calculated gearbox dimensions allow a more accurate evaluation of gearbox integration effects.

Future plans include the extension of the tool to other gearbox designs and additional gearbox components. These include an oil system model as well as a more detailed bearing model to enhance the feasibility, mass and geometry estimations.

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