

CRANFIELD UNIVERSITY

Diana Guiomar SAN BENITO PASTOR

Multidisciplinary Preliminary Design and Integration of the Transmission  
System in a Pusher Geared Contra-Rotating Open Rotor

SCHOOL OF AEROSPACE, TRANSPORT and MANUFACTURING

PhD THESIS

Academic Year: 2016 - 2020

Supervisor: Dr Devaiah NALIANDA  
Associate Supervisor: Dr Vishal SETHI  
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## **ABSTRACT**

One of the main challenges for the aviation industry is to reduce its global environmental footprint. Geared contra-rotating open rotors have the potential to further reduce fuel consumption relative to geared turbofans, but require a more-complex speed-reducing transmission system to drive the propellers. Hitherto, the preliminary design for such transmission systems has been reported independently of the overall engine modelling or has been limited by many pre-determined engine constraints. This has restricted the feasible design space of the engine and the transmission system. Therefore, this research addresses two key questions: Can the preliminary design of the transmission system be done from cycle parameters without imposing additional engine-based constraints? How does the introduction of more rigorous preliminary design processes for the transmission system affect the design space of the open rotor engine?

The transmission system in the contra-rotating open rotor comprises two main components: a power gearbox and a pitch change mechanism. These technology enablers enhance engine performance by decoupling the operation of the propellers from the free power turbine and adjusting the pitch of the blades respectively. The best power gearbox option in a contra-rotating open rotor is a differential planetary gearbox, which enables contra-rotation of the output shafts and a high power transfer with a reduced size. The space envelope of the power gearbox varies with the torque ratio between its output shafts, connected to the propellers. The effect of torque ratio variation on gearbox design has been analysed in this research for equal propeller rotational speeds and different speed ratios between the output shafts. This research shows that potential torque ratios lie between 1.1 and 1.33, with the higher ratios enabling more compact gearboxes having four or five planet gears. However, for a prescribed propeller rotational speed, higher torque ratios would reduce the rotational speed of the low-pressure turbine driving the propellers and potentially reduce its efficiency. Alternatively, increasing propeller rotational speeds would result in thicker radial shafts for the pitch change mechanism connected to the propeller blades. The

presence of these shafts radially traversing the engine's gas path might contribute to the blockage of the exhaust duct.

To address these issues, a preliminary design framework has been developed that combines 0D thermodynamic modelling with flow path sizing and weight estimation and enables assessing the integration effects of the transmission system on engine performance. The engine's optimum performance design space might not be accessible due to mechanical constraints or integration interactions. Relative to a baseline design, the reduction in fuel burnt can be as high as 1% with current technology levels and 3% for future designs with entry into service by 2035. However, the potential performance gains derived from improvements in turbine design might not be achievable when the transmission system is integrated. The integration of the transmission system further reduces potential fuel burn gains to 0.6% relative to a baseline engine design.

A wide range of activities for future work is opened by the methods developed in this research in both performance and mechanical development of the components in the transmission system.

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# NOMENCLATURE

## Abbreviations

|        |                                                                        |
|--------|------------------------------------------------------------------------|
| AF     | Activity Factor [-]                                                    |
| AGBT   | Advanced Gearbox Technology                                            |
| AGMA   | American Gear Manufacturers Association                                |
| APET   | Advanced Propfan Engine Technology                                     |
| ATM    | Aerodynamic Twisting Moment [ $Nm$ ] or [ $lb\ in$ ]                   |
| BPR    | Bypass Ratio [-]                                                       |
| CFD    | Computational Fluid Dynamics                                           |
| CORSIA | Carbon Offsetting and Reduction Scheme for International Aviation      |
| CROR   | Contra-Rotating Open Rotor                                             |
| CRP    | Contra-Rotating Propellers                                             |
| CTM    | Centrifugal Twisting Moment [ $Nm$ ] or [ $lb\ in$ ]                   |
| DEMOS  | Developing advanced Engine Multi-disciplinary Optimization Simulations |
| DPGB   | Differential Planetary Gearbox                                         |
| DREAM  | validation of Radical Engine Architecture systems                      |
| EIS    | Entry Into Service                                                     |
| EOR    | End of Runway                                                          |
| FN     | Net Thrust [ $N$ ]                                                     |
| FP     | Front Propeller                                                        |
| FPR    | Fan Pressure Ratio [-]                                                 |
| FTM    | Frictional Twisting Moment [ $Nm$ ] or [ $lb\ in$ ]                    |
| GE     | General Electric                                                       |
| HCF    | High Cycle Fatigue                                                     |
| HCR    | High-Contact Ratio                                                     |
| HPC    | High-Pressure Compressor                                               |
| HPT    | High-Pressure Turbine                                                  |
| HTR    | Hub-to-Tip Ratio [-]                                                   |
| ICAO   | International Civil Aviation Organisation                              |
| IPC    | Intermediate Pressure Compressor                                       |
| IPT    | Intermediate Pressure Turbine                                          |

|         |                                                               |
|---------|---------------------------------------------------------------|
| KHI     | Kawasaki Heavy Industries                                     |
| LAP     | Large-scale Advanced Prop-fan                                 |
| LCF     | Low Cycle Fatigue                                             |
| LPT     | Low-Pressure Turbine (also referred to as free power turbine) |
| MTBUR   | Mean Time Between Unscheduled Removal [h]                     |
| NA      | Not Available                                                 |
| NASA    | National Aeronautics and Space Administration                 |
| NPR     | Nozzle Pressure Ratio [-]                                     |
| OPR     | Overall Pressure Ratio [-]                                    |
| OREAT   | Open Rotor Engines Advanced Technologies                      |
| PCM     | Pitch Change Mechanism                                        |
| PQA     | Modified non-dimensional power coefficient [-]                |
| PROOSIS | Propulsion Object Oriented Simulation Software                |
| P&W     | Pratt & Whitney                                               |
| RP      | Rear Propeller                                                |
| SAGE    | Sustainable and Green Engine                                  |
| SFC     | Specific Fuel Consumption [ $g/(s\ kN)$ ]                     |
| ToC     | Top of Climb                                                  |
| TQA     | Modified non-dimensional thrust coefficient [-]               |
| TR      | Torque Ratio [-] (also as a roman symbol)                     |
| TRL     | Technology Readiness Level                                    |
| TTM     | Total Twisting Moment [ $Nm$ ] or [ $lb\ in$ ]                |
| UDF     | Unducted Fan                                                  |
| UHBR    | Ultra-High Bypass Ratio                                       |

### **Roman Symbols**

|        |                                                                                                     |
|--------|-----------------------------------------------------------------------------------------------------|
| $A$    | Area [ $m^2$ ]                                                                                      |
| $AN^2$ | Stress parameter [ $RPM^2m^2$ ]                                                                     |
| $CF$   | Contingency factor [-]                                                                              |
| $C_f$  | Surface condition factor for pitting resistance [-]                                                 |
| $C_H$  | Hardness factor [-]                                                                                 |
| $CL$   | Centrifugal load [N]                                                                                |
| $C_p$  | Elastic coefficient [ $(MPa)^{0.5}$ ] or [ $(psi)^{0.5}$ ],<br>Isobaric specific heat [ $J/kg\ K$ ] |

|                 |                                                                     |
|-----------------|---------------------------------------------------------------------|
| $C_P$           | Power coefficient [-]                                               |
| $C_T$           | Thrust coefficient [-]                                              |
| $D$             | Diameter [ $m$ ] or [ $ft$ ]                                        |
| $d$             | Pinion pitch diameter [ $cm$ ] or [ $in$ ]                          |
| $E$             | Modulus of elasticity [ $MPa$ ] or [ $psi$ ]                        |
| $F$             | Gear Face Width [ $cm$ ] or [ $in$ ]                                |
| $F/d$           | Gear aspect ratio [-]                                               |
| $g$             | Gravitational acceleration [ $m/s^2$ ]                              |
| $h$             | Specific enthalpy [ $J/kg$ ]                                        |
| $I$             | Geometry factor for pitting resistance [-]                          |
| $J$             | Geometry factor for bending resistance [-]                          |
| $J$             | Advance ratio [-]                                                   |
| $K_c$           | Pitting resistance constant [ $cm^3$ ] or [ $in^3$ ]                |
| $K_d$           | Total derate factor [-]                                             |
| $K_m$           | Load distribution derate factor [-]                                 |
| $K_o$           | Overload derate factor [-]                                          |
| $K_q$           | Planet interference factor [-]                                      |
| $K_s$           | Size derate factor [-]                                              |
| $K_R$           | Reliability factor [-]                                              |
| $K_t$           | Bending strength constant [ $cm^3$ ] or [ $in^3$ ]                  |
| $K_T$           | Temperature factor [-]                                              |
| $K_v$           | Dynamic derate factor [-]                                           |
| $L$             | Gear design life [h] or Power loading [ $kW/m^2$ ] or [ $HP/ft^2$ ] |
| $L/D$           | Lift-to-drag ratio [-]                                              |
| $\dot{m}$       | Mass flow rate [ $kg/s$ ]                                           |
| $m_{M^{*0.75}}$ | Correction slope for Mach number effects [-]                        |
| $MA$            | Moment arm [ $m$ ] or [ $ft$ ]                                      |
| $M_{h0.75}$     | Helical Mach number at 0.75 blade span [-]                          |
| $M^{*}_{h0.75}$ | Critical helical Mach number at 0.75 blade span [-]                 |
| $M_o$           | Flight Mach Number [-]                                              |
| $m_G$           | Gear Ratio ( $R_P/R_S$ ) [-]                                        |
| $M_n$           | Mach number [-]                                                     |
| $m_p$           | Profile Contact Ratio [-]                                           |

|                   |                                                          |
|-------------------|----------------------------------------------------------|
| $N$               | Number of gear teeth [-]                                 |
| $N_{\text{bld}}$  | Number of Blades [-]                                     |
| $n$               | Rotational speed [ <i>RPM</i> ]                          |
| $n_L$             | Number of cycles [-]                                     |
| $P_t$             | Total pressure [ <i>Pa</i> ]                             |
| $Q$               | Torque [ <i>Nm</i> ] or [ <i>lb in</i> ]                 |
| $q$               | Number of flow paths (number of planet gears) [-]        |
| $PW$              | Power [ <i>W</i> ] or [ <i>hp</i> ]                      |
| $R$               | Gear pitch radius, Radius [ <i>cm</i> ] or [ <i>in</i> ] |
| $R_{\text{DPGB}}$ | Gearbox ratio ( $R_R/R_S$ ) [-]                          |
| $S$               | Stress or strength [ <i>MPa</i> ] or [ <i>psi</i> ]      |
| $SF$              | Safety Factor                                            |
| $S_F$             | Bending safety factor [-]                                |
| $S_H$             | Pitting safety factor [-]                                |
| $SR$              | Speed Ratio ( $SR =  n_S/n_R $ ) [-]                     |
| $T_t$             | Total Temperature [ <i>K</i> ]                           |
| $TR$              | Torque Ratio ( $Q_C/Q_R$ ) [-]                           |
| $U$               | Blade speed [ <i>m/s</i> ] or [ <i>ft/s</i> ]            |
| $V$               | Local axial velocity [ <i>m/s</i> ]                      |
| $V_{ax}$          | Axial flow velocity [ <i>m/s</i> ]                       |
| $V_o$             | Flight speed [ <i>m/s</i> ]                              |
| $W$               | Weight [ <i>kg</i> ] or [ <i>lb</i> ]                    |
| $Y_N$             | Bending resistance factor [-]                            |
| $Z_N$             | Pitting resistance factor [-]                            |

### Greek Symbols

|             |                                               |
|-------------|-----------------------------------------------|
| $\gamma$    | Heat capacity ratio [-]                       |
| $\Delta$    | Variation                                     |
| $\eta$      | Efficiency [-]                                |
| $\Phi$      | Flow coefficient [-]                          |
| $\varphi_n$ | Gear normal pressure angle [°]                |
| $\psi$      | Gear helix angle [°], Loading coefficient [-] |

## Subscripts

|                     |                                |
|---------------------|--------------------------------|
| <i>ac</i>           | allowable contact              |
| <i>at</i>           | allowable bending              |
| <i>c</i>            | contact                        |
| <i>C</i>            | carrier                        |
| <i>DPGB</i>         | differential planetary gearbox |
| <i>exhaust duct</i> | exhaust duct                   |
| <i>FP</i>           | front propeller                |
| <i>h</i>            | hub                            |
| <i>i</i>            | stage number                   |
| <i>in</i>           | inlet                          |
| <i>is</i>           | isentropic                     |
| <i>LPT</i>          | low-pressure turbine           |
| <i>m</i>            | mean                           |
| <i>nozzle</i>       | core exhaust nozzle            |
| <i>out</i>          | outlet                         |
| <i>propeller</i>    | propeller                      |
| <i>P</i>            | planet, pitching (moment)      |
| <i>R</i>            | ring                           |
| <i>ref</i>          | reference                      |
| <i>RP</i>           | rear propeller                 |
| <i>S</i>            | sun                            |
| <i>shaft</i>        | shaft                          |
| <i>t</i>            | bending                        |
| <i>tip</i>          | tip                            |
| <i>UTS</i>          | ultimate tensile strength      |



# 1 Introduction

Innovative solutions are required to assure the sustainable economic growth of the aerospace industry. Emissions of greenhouse gases (mainly CO<sub>2</sub>) and others affecting local air quality (mainly NO<sub>x</sub> and non-volatile particulate matter) must be reduced. Targets set by Flightpath 2050 [1] are driving the aviation industry to achieve ambitious environmental goals including reductions of 75% in CO<sub>2</sub> and 90% in NO<sub>x</sub> per passenger-kilometre, and 65% reduction in noise, by 2050 compared to year 2000 levels. Past incremental technology upgrades in airframe and propulsion system technology have led to large improvements in efficiency. However, achieving further significant reductions in fuel burn consumption with tube-and-wing aircraft configurations and conventional propulsion system technology will be challenging.

The propulsion system represents 80% of improvements in fuel burn in new-generation aircraft entering into service in 2020 relative to 2010 values [2]. All technology concepts for advanced propulsion systems currently considered for integration in large transport aircraft feed from the principle of ultra-high bypass ratio (UHBR) or geared turbofans [3]. Numerous bodies of research have investigated alternative engine architectures optimized for low emissions, noise and weight, by increasing engine thermal and propulsive efficiencies [4]–[10]. One of the engine architectures with high potential for improving propulsive efficiency and fuel consumption is the Contra-Rotating Open Rotor (CROR).

The CROR benefits from low specific thrust and presents the capability to reduce propeller speed and loading to maintain high stage efficiency. The potential performance gains of the CROR relative to single propeller engines and conventional turbofans have been extensively discussed in the literature, from acoustic [11]–[19], aerodynamic [11], [15], [20]–[23] and operability [13], [20] perspectives. Relative to advanced geared turbofans, the characteristics of the CROR are unique to satisfy short- and medium-term goals of performance and emissions, with the potential to obtaining fuel burn benefits around 15% [3].

To date, the CROR still presents some challenges regarding aircraft integration, noise mitigation and mechanical complexity of the propulsion system. Research focus has been predominantly on addressing acoustics and airframe integration challenges and substantially less effort has gone into the mechanical aspects of the CROR. However, the need for advanced performance pitch control and high reduction ratios at high power levels makes the transmission system an essential component in the development of the CROR, namely:

- A power gearbox to decouple the performance of the low-pressure turbine (LPT) from the propellers, and
- A pitch change mechanism (PCM) to adjust and sustain the pitch of the propeller blades for meeting CROR high-performance demands.

Therefore, the integration of the transmission system into the engine model at the preliminary design level becomes a requirement for progress in CROR research and achievement of its potential performance benefits. To that end, there is a need to develop design models of the transmission system components with a similar level of fidelity to the turbomachinery components in the engine. Consequently, this research focusses on the preliminary design and integration of the transmission system in the CROR.

A first step in the research is, therefore, the definition of preliminary design methodologies for the power gearbox and the PCM to comply with performance, space envelope and durability requirements of the CROR application. The end goal is, however, to provide a preliminary design framework for defining the feasible design space of geared CRORs. Figure 1 presents the flow of information in the developed framework.

The first step in any engine model definition is setting the operating requirements: thrust requirements, ambient and flight conditions. Defining top-level characteristics of the engine cycle determines engine performance and enables the calculation of preliminary estimates of the components' size and weight. Therefore, the calculation of engine performance, gas path analysis and weight estimation constitute the CROR engine module.

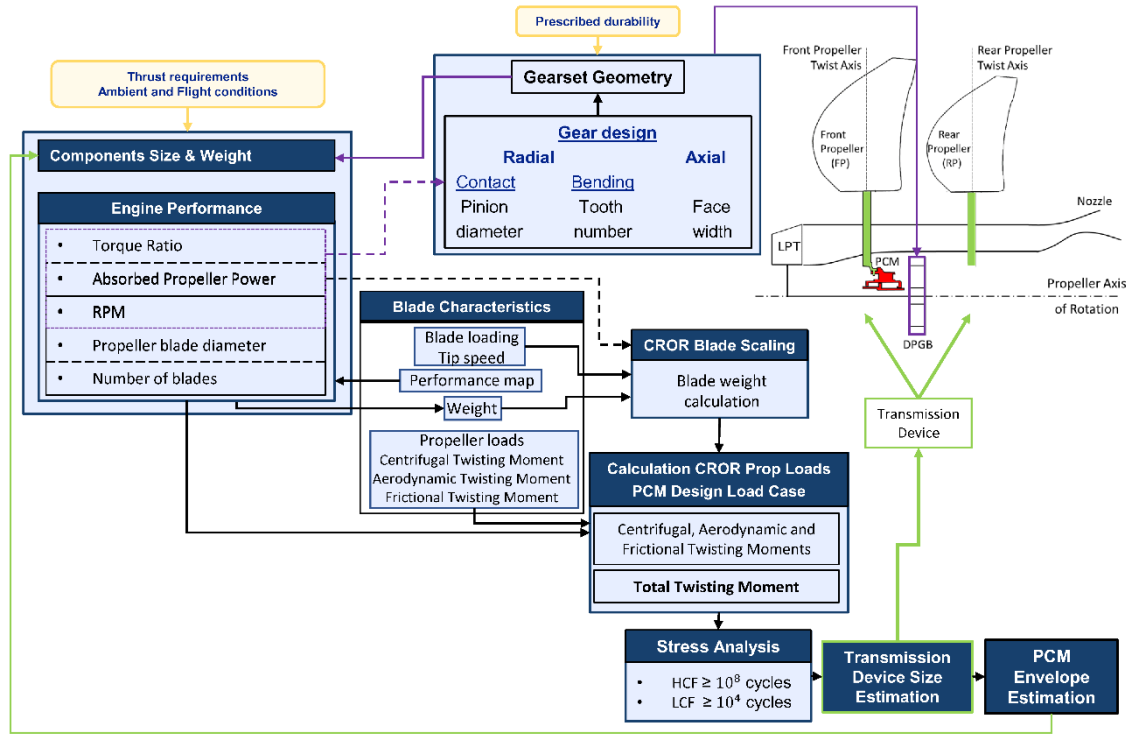


**Figure 1 Overall methodology**

In the CROR, the contra-rotating propellers (CRP) can represent around 95% of the net thrust production. The operation and characteristics of the CRP define the engine's power requirement and the loads generated to achieve a prescribed thrust level. Therefore, propeller blade selection not only affects overall engine performance but also the design of the transmission system. The multidisciplinary CROR design framework developed in this research for transmission system integration requires consistent data for propeller performance and aeromechanical behaviour. However, detailed blade information for advanced propellers is scarce in the literature and is only available in the literature for older blade designs. The use of existing performance and aeromechanical data for older blade designs is suitable for parameter variation trend assessments and design space exploration of the CROR undertaken in this research.

Following the definition of the CROR cycle parameters and CRP characteristics, the first component of the transmission system under evaluation is the power gearbox. In addition to prescribed durability (life), the inputs for its preliminary design are outputs of the engine performance model: torque ratio (TR), propeller

absorbed power and rotational speed of the propellers. In this research, the space envelope of the power gearbox has been defined by the gearset, as explained in chapter 3. Therefore, the engine parameters define the diameter of the gears, tooth number and face width (axial gear span). Gear definition also affects the feasible range of TRs between the output shafts connected to the propellers. The calculated gearbox design information feeds back to the engine module and informs its feasible design space. The integration of the power gearbox in the CROR also enables the identification of feasible locations for the power gearbox in the engine.

The second and last component of the transmission system assessed in this research is the PCM. The preliminary design and sizing of the PCM in this research have been reduced to the design of the transmission device, indicated by the radial shafts in the schematic in Figure 1. The transmission device would be the PCM component with the highest impact on gas path definition for the CROR as it introduces a potential source of blockage to the flow exiting the engine core.

Prior to the preliminary design of the PCM it is necessary to scale the propeller blades and calculate the reaction loads on the component. Engine performance data and design characteristics of a pre-existing blade are used to scale the propeller blades to comply with the requirements of the new CROR application and estimate blade weight. This information is then used to calculate the propeller loads of the current CROR application from the propeller loads of the reference blade: centrifugal, aerodynamic and frictional twisting moments.

The loads withstood by the PCM are defined in terms of high- (HCF) and low-cycle fatigue (LCF). Therefore, the stress analysis to size the PCM is based on a number of cycles to failure. HCF is associated with the application of cyclic stress over a relatively long time (elastic deformation), and LCF would be associated with the number of starts and stops and repeated plastic deformation. Finally, the integration of the PCM into the engine module is done feeding back the component size envelope to the engine module to inform the feasible design space of the CROR.

The present research focusses on a geared pusher CROR. However, the preliminary design methods developed within the research apply to alternative engine configurations.

To achieve the stated aim, the research splits into two main objectives. Within each objective, a set of specific targets were identified and a success criterion was imposed for each of them:

**Objective 1:** *Preliminary design of the power transmission system for a CROR*

**Specific Targets**

**Power Gearbox**

1. Creation of a novel design framework for sizing the power gearbox and identification of feasible torque ratios between the output shafts connected to the propellers for equal propeller rotational speeds and different speed ratios between them.
2. Development of an up-to-date methodology for the design of maximum load capacity gears based on engine requirements of power transmitted, reduction ratio and durability.
3. Validation of the gearbox preliminary design methodology and quantification of the design uncertainties with parametric and sensitivity analyses.

**Pitch Change Mechanism**

4. Development of a preliminary design methodology to calculate the aeromechanic loads generated by the propeller at design point operation to size the PCM.
5. Creation of a methodology for the preliminary mechanical design of a pitch change mechanism based on propeller loads and prescribed life.
6. Sensitivity assessments of the preliminary design methodology for the pitch change mechanism.

## **Success Criteria**

### **Power Gearbox**

1. Delivery of a power gearbox preliminary design module that can be integrated with the preliminary design engine module
2. Delivery of a gear design tool that can be integrated with the engine preliminary design module.
3. Delivery of a design with differences in gear annulus geometry below 1%. Delivery of design trends for the power gearbox using the developed methodology.

### **Pitch Change Mechanism**

4. Delivery of the aeromechanical loads acting on the propellers in accordance with magnitudes presented in the literature.
5. Delivery of a PCM preliminary design tool that can be integrated with the engine module.
6. Delivery of design trends for the PCM integrated with the engine model.

**Objective 2:** *Evaluation of the integration effects of the power transmission system on the feasible design space of the CROR*

## **Specific Targets**

1. Development of an engine performance model representative of the CROR.
2. Flow path sizing and definition of the engine general arrangement.
3. Component and total engine weight estimation.
4. Fuel burn assessments.
5. Integration of the transmission system in the engine model for multi-point engine design assessments and design space exploration.
6. Identification of the feasible design space for future CRORs, to offer a reduced design region that could be further explored with dedicated tools.

### **Success Criteria**

1. Delivery of a verified engine model, which produces performance results consistent with data published in the literature for the CROR.
2. Delivery of the basic geometric characteristics of the CROR gas path, which match data published in the literature.
3. Delivery of weight estimations that match published research data.
4. Delivery of trends verified with literature for the CROR.
5. Quantification of the impact of the transmission system on the CROR in terms of performance, general arrangement, and fuel burn.
6. Delivery of a feasible design space for the CROR.

Chapter 2 presents a literature review that sets the scene of development for low-emission propulsion systems and the integration of transmission systems in a multidisciplinary preliminary design level. Chapter 3 and 4 present the modelling of the main components in the transmission system, namely the power gearbox and the pitch change mechanism. Chapter 5 presents the modelling procedures for the propulsion system and details the baseline CROR configuration. Chapter 6 presents the parametric assessments undertaken on key design parameters of the propulsor. Finally, conclusions and suggestions for future work are presented in chapter 7.



## 2 Literature Review

### 2.1 Development and Growth of the Aviation Industry

One of the main challenges for the aviation industry is to reduce its global environmental footprint while sustaining its current growth rate. The aviation industry has been at the centre of public attention for its environmental impact through noise, pollutant emissions and more recently, carbon dioxide (CO<sub>2</sub><sup>1</sup>) emissions [24]. The aviation sector represents 2% of the global anthropogenic CO<sub>2</sub> emissions [25]. Past trends show that air traffic has been resilient to external shocks and doubled every fifteen years [26], making it the fastest-growing transport sector [24], and passenger numbers could double by 2037 [27]. Narrow-body (single-aisle) aircraft represent close to two-thirds of the commercial aircraft on order and more than half of in-service aircraft in 2015 [28]. Each new generation of aircraft enables a 15% improvement in fuel burn relative to the model it replaces [2]. However, the growth of the aviation industry is matched with a rise in the emissions that it produces.

In this context, the aviation industry committed in 2008 to adopting a set of three ambitious targets to reduce CO<sub>2</sub> emissions from air transport and tackle the challenge of global climate change [29]:

- Target 1:** An average 1.5% annual improvement in fuel efficiency **from 2009 to 2020**
- Target 2:** A cap on net aviation CO<sub>2</sub> emissions **at 2020** levels
- Target 3:** A decrease in aviation net CO<sub>2</sub> emissions **by 2050** to 50% of 2005 levels

The short-term goal (Target 1) has been constantly met and exceeded by industry with an average 2.4% annual improvement since 2009 with incremental improvements in technology [25]. To address the mid-term carbon-neutral growth (Target 2) the International Civil Aviation Organization (ICAO) launched in 2016 the Carbon Offsetting and Reduction Scheme for International Aviation (CORSIA)

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<sup>1</sup> The CO<sub>2</sub> emissions to the atmosphere are of particular concern because of their long residence time

to tackle increases in total CO<sub>2</sub> emissions from international civil aviation above 2020 levels [30]. The long-term goal (Target 3) requires dramatic reductions in fuel burn, which go beyond the capabilities of the evolutionary trend followed by today's technology and require a shift towards revolutionary concepts through improvements in technology (including sustainable alternative fuels), aircraft management and operations, infrastructure and market-based measures [25], [31].

## **2.2 Low-Emission Propulsion Technologies for Sustainable Aviation**

The reduction in fuel consumption is at the core of the development of future engine technologies. Reductions in fuel consumption primarily stem from the increase in overall efficiency and weight reduction of the propulsion system. Improvements in overall efficiency can be achieved by enhancing thermal and propulsive efficiency. For gas turbines with a conventional cycle (simple Brayton cycle), thermal efficiency depends on the overall pressure ratio (OPR) achievable by the compressors, the temperature ratio between the exit of the burner and the engine inlet, component efficiencies and cooling technology. However, significant improvements in thermal efficiency of a simple gas turbine cycle are limited by the mature level of the technology already existing for the driving factors [32], e.g. small compressor blades with minimum losses, cooling technology and materials for the high-pressure turbine.

The increase in propulsive efficiency is the only way to significantly reducing fuel consumption from a medium-term perspective. Improvement in propulsive efficiency implies a reduction in specific thrust of the propulsion system. In the case of turbofans, improving propulsive efficiency requires a reduction in fan pressure ratio (FPR) through the increase of the bypass ratio, which leads to a reduction in fan rotational speed.

The need for compact engines has led the designers to introduce speed-reducing transmission systems to avoid excessively long multistage turbines. Introducing a gearbox in the turbofan to decouple the performance of the fan from the turbine favours a reduction in exit jet velocity (i.e. a reduction in FPR) and an

improvement in propulsive efficiency. However, there is a point, where the reduction in fuel burn from increasing propulsive efficiency is negated by the reduction in transfer efficiency (primarily caused by pressure losses in the bypass duct) [33]. Additionally, the reduction in specific thrust in the case of geared turbofans, as for conventional turbofans, is constrained by nacelle weight and installation drag.

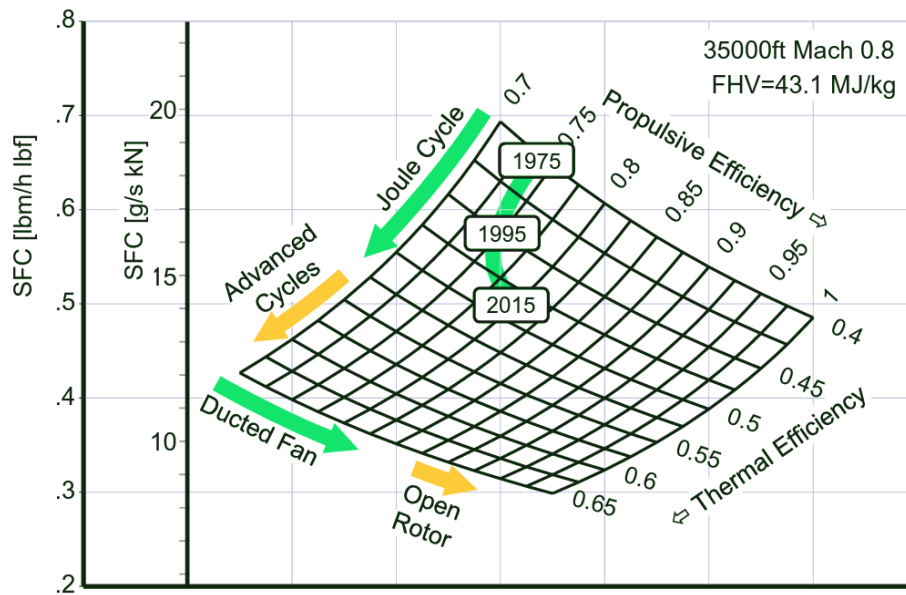


**Figure 2 Turbofan trend with projective open rotor trends [34]**

The requirement for a substantial increase in efficiency has led to a shift in research towards revolutionary configurations that do not follow the evolutionary trend of previous designs (Figure 2). An example of a revolutionary propulsion system with a high potential for propulsive efficiency improvement is the contra-rotating open rotor. This engine technology represents a technological transition between existing propulsion systems and more disruptive technologies. The CROR, therefore, contributes to achieving the mid-term goals and to the technology maturation of key engine components such as power gearboxes and PCMs. Relative to UHBR geared turbofans, the propulsive efficiency in a CROR may increase further with lower associated increases in nacelle drag and transfer efficiency-related losses [33]. The CROR can also further reduce the rotational speed of the propellers and the loading in each row to maintain high stage efficiency. The reduction in specific fuel consumption (SFC) and nacelle drag

compensate for the expected heavier weight of the CROR and can lead to mission fuel burn improvements close to 15% relative to geared turbofans like Pratt & Whitney's GTF [3], [33], [35]. In the context of the CORSIA agreements, described in Ref. [30], the CROR is a strong candidate for achieving medium-term CO<sub>2</sub> emission improvements [35].

The CROR presents a technology readiness level TRL 5 [35], higher than other disruptive technologies for efficiency enhancement with higher potential for achieving the long-term goals. The incorporation of intercooling or reheat processes to the gas turbine cycle are some of the avenues pursued by researchers to achieve dramatic efficiency increases. Another avenue of research is decomposing the propulsion system into several building blocks: sources of thrust, sources of energy, and transmission of thrust to power. Examples of this direction of research would be distributed propulsion or hybrid propulsion systems. The parallel advancement in manufacturing and material technology will also enable a reduction in the weight of the propulsion system and the improvement in operating temperature limits of gas turbines.

## **2.3 The Contra-Rotating Open Rotor**

The CROR is a gas turbine with unducted contra-rotating propellers that are not contained within a casing or nacelle (Figure 3). The CROR concept is derivative of turboprops, and hence is sometimes referred to as "advanced turboprop" or "propfan". Contrary to turboprops, the advance blade design and the presence of a second row of propellers in the CROR allow maintaining high propulsive efficiencies when operating at higher flight speeds (Mach 0.7 to 0.85) and altitude (35,000 to 39,000 ft) [36].

Conventional propellers with straight blades can operate efficiently up to a flight Mach number ( $M_0$ ) of 0.6, beyond which propeller efficiency decreases due to compressibility effects. Research on advanced propellers during the 60s and 70s tackled the reduction of shock losses with the design of highly-loaded, swept-back propeller blades with transonic profiles [36]. Further efficiency improvements can be obtained from the recovery of swirl losses with a second row of propellers, demonstrating high-efficiency operation at high speeds [37].

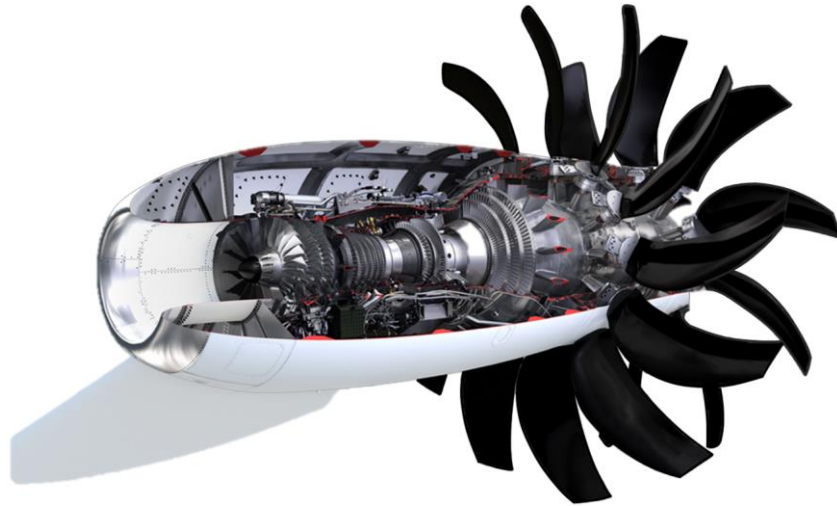


**Figure 3 Contra-rotating open rotor SAGE2 (Safran Aircraft Engines)**

In the 1980s, several flight tests were undertaken to evaluate the fuel efficiency of a CROR at high subsonic flight speeds. The National Aeronautics and Space Administration (NASA) and General Electric (GE) launched the Unducted Fan (UDF) technology demonstrator program, which provided proof of concept for a direct-drive configuration (i.e. without a gearbox), driving the propellers with a contra-rotating power turbine [38]–[40]. In 1987 the full-scale UDF was flight-tested [41]. Pratt & Whitney (P&W) and Allison demonstrated a geared concept with the development of the PW-Allison 578-DX, with a flying demonstrator in 1989 [41]. These pusher CROR configurations never went past flight testing.

Figure 3 shows the outcome of recent developments on the CROR by Safran, which started in 2008 as part of the European research project SAGE (Sustainable And Green Engine) within Clean Sky. The CROR design built on the experience of GE's UDF, opting for a geared configuration with a potential entry into service in 2035 [35]. The direct-drive arrangement would result in high rotational speeds for the propeller blades, which hinder the reduction of acoustic emissions (cabin and community noise). In addition to the acoustic problem, another matter of concern in the development of the CROR was the integration of a power gearbox close to the propellers with blade pitch control [35]. Safran's engine demonstrator used the core section from the Rafale's M88 engine, developed by the company. Based on this existing engine core, the power gearbox was positioned between the core and the propellers [35].

Numerous bodies of research have modelled and simulated the performance of the CROR. Typically, research on CROR performance focusses on proving the potential fuel burn benefits of the CROR and identifying ways to mitigate some of its challenges, e.g. noise and aircraft installation. However, the mechanical complexity of the transmission system and its integration in the CROR, especially in a pusher-configuration, is also a challenge that needs to be addressed early on in the design process.

It is worth noting that the term *transmission system* is often used in literature for the CROR to refer to all components transmitting power from the core to the propellers, e.g. shafts, gears, bearings, etc. However, this research considers the power gearbox and the pitch change mechanism as the main components in the transmission system. Therefore, when the term *transmission system* is used in this document it solely refers to these two components.

### 2.3.1 The Power Gearbox

The power gearbox is an integral part of new engine designs for efficient power operation. In the CROR, it operates as a speed-reducer that transmits power from the high-speed LPT to the low-speed propellers. The main components in the power gearbox of an aero-engine are gears, shafts, and bearings (Figure 4). These components are enclosed in a casing and their configuration strongly relies on the type of application. In addition to the mechanical components, an oil system may lubricate and release the heat generated by the gearbox assembly.

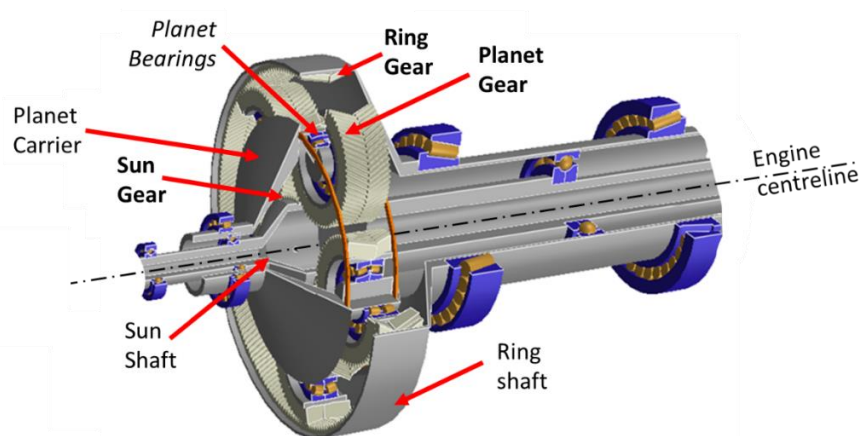


**Figure 4 Power gearbox with a planetary gearset and double-helical gears**

The focus of this research is on size estimation of the transmission system for engine integration in the preliminary design phase. Therefore, only the components dominating the gearbox space envelope have been assessed in detail, i.e. the gears in the gearset.

The use of gear-systems to replace the direct link between the LPT and the fan in aero engines is not new. The introduction of the Lycoming ALF502 and the Garrett TFE731 in the 1970s shows that the concept has a long history in turbofans [42]. However, some challenges arise in the gear design for high-power aero applications regarding efficiency, physical size, weight and mechanical integrity of the transmission system. The incorporation of a power gearbox in large commercial turbofans over 100 kN take-off thrust is relatively new to the aerospace industry [43]. Entering service in 2013, the P&W PG1000G was the first high-thrust turbofan with a gearbox demonstrator rating of 30,000 hp (22.4 MW), three to four times higher than previous geared turbofans like the ALF 502 [42] and more than twice the power of the TP400-D6 turboprop propeller reduction gearbox [44].

Planetary gears, widely utilized in aerospace drives [42], present a compact in-line solution for high-power aero applications. A planetary gear train consists of an assembly of concentric gears, namely the sun, planet and ring gears. The sun and ring gears engage simultaneously by revolving and rotating gears, called planets, which mesh so that their operating circles (i.e. the pitch circles) roll without slip. The sun and ring gears connect to the sun and ring shafts respectively and, the carrier shaft connects the centres of the planet gears and revolves around the centre of the sun gear, which coincides with the engine centreline. The bearings in the gearbox assembly define the location of the shafts and gears, enable the motion between them and absorb the loads generated in operation.

The concentric arrangement of the shafts enables an in-line driving of the equipment, saving space. The multiple load paths in a planetary gear train, determined by the number of planets, transmit the power and favour load distribution resulting in compact transmission designs [45]. This compactness

translates into space envelope and weight savings in the engine. The smaller and stiffer components result in reduced noise and vibration and efficiency increase [45]. Finally, the planetary gearbox only transmits torque, as the resultant radial forces in the input and output shafts cancel out. Consequently, the design of the planetary bearings is simplified.

The contra-rotation of the propellers in the CROR entails a more complex mechanical design of the power gearbox relative to geared turbofans. Contra-rotation of the output shafts can be obtained by differential action of the gears (i.e. all shafts rotate). A differential planetary gearbox (DPGB) can carry more power than a simple planetary configuration by virtue of a higher input rotational speed [46]. Figure 5 shows the DPGB connections in the CROR:

- the sun shaft transmits the power and rotational speed from the low-pressure shaft to the sun gear,
- the carrier shaft links the centres of all the planets to one of the propellers,
- the ring shaft connects the annulus (exterior ring) gear to the other propeller.

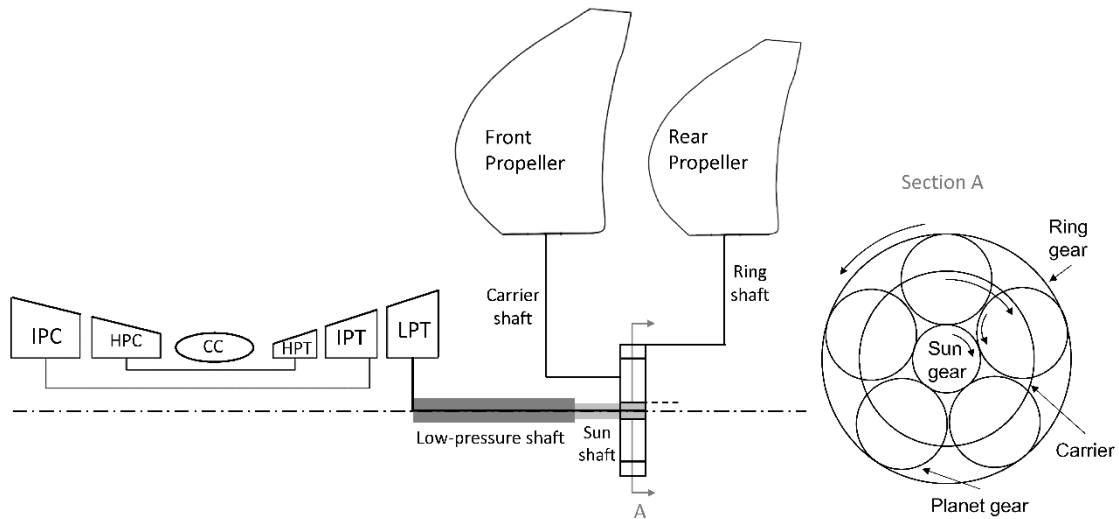


**Figure 5 Sketch of a contra-rotating open rotor engine, with the kinematics and cross-section of the DPGB**

In the early design phases of a new transmission system, none of the gear dimensions is known and the selection of design inputs has a direct impact on

the outcome. The main parameters affecting the size of the power gearbox are the torque requirements, the relative rotational speeds of its shafts (linked to the ratio between gear diameters) and the design durability of the assembly. The relative diameters of the gears in the DPGB determine the gear ratio and the achievable speed reduction between the input and output shafts. The rotational speeds of two out of the three shafts can be independently selected and determine the rotational speed of the third shaft. In the CROR, the propeller rotational speeds define the rotational speed of the LPT for a given ratio of gear diameters. However, the overall radial dimensions of the gearbox can only be calculated if the output (ring) gear diameter or that of the input (sun) gear and the relationship between them is known. The axial dimensions of the gears are also a function of the gear diameter ratio.

The volume of the DPGB constrains the possible locations of the transmission system in the CROR. The most common connection arrangements are sketched in Figure 6, where the DPGB could be located between the LPT and the front propeller, or between propellers.

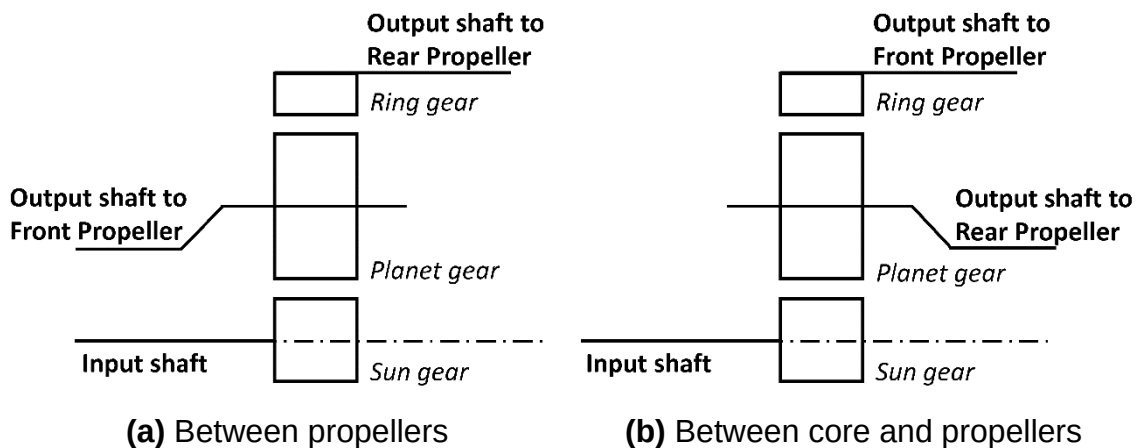


**Figure 6 Location and common shaft connection arrangements for the DPGB**

Even though the power gearbox design for a CROR application has been addressed in past research projects, publications are scarce and few specify the design methodology.

Table 1 provides a summary of the main gearbox design characteristics used in previous projects. The presented data enables the identification of the type of

components to be used in advanced gearboxes and feasible design margins for modern CROR applications.

**Table 1 Design investigations of differential planetary gearboxes for contra-rotating open-rotor engines**

| Research Project           | Design Level                | DPGB Position In Engine              | Power Class [hp]         | Speed Ratio (SR) | Front/ Rear Propeller RPM | Number Of Planets in DPGB | DPGB Design Durability (MTBUR) [h] | Sun Gear Diameter [cm (in)] | Type of gears            | Planet bearings                        |
|----------------------------|-----------------------------|--------------------------------------|--------------------------|------------------|---------------------------|---------------------------|------------------------------------|-----------------------------|--------------------------|----------------------------------------|
| <b>APET [Allison] [47]</b> | Conceptual                  | Between gas generator and propellers | 10,000                   | 8.19:1           | 1329                      | 4                         | 30,000                             | 11.6 (4.6)                  | Single helical           | Tapered roller                         |
| <b>APET [P&amp;W] [48]</b> | Conceptual                  | Between gas generator and propellers | 12,000                   | 8.23:1           | 1233                      | 5                         | 30,000                             | 17.78 (7)                   | High contact ratio (HCR) | Single-row spherical roller            |
| <b>AGBT [P&amp;W] [49]</b> | Conceptual Detailed Testing | Between LPT and front prop           | 12,000                   | 8.32:1           | 1235                      | 5                         | 30,000                             | 18 (7)                      | High contact ratio (HCR) | Single-row spherical roller            |
| <b>AGBT [Allison] [50]</b> | Conceptual Detailed Testing | Between LPT and front prop           | 13,000                   | 8.33:1           | 1140                      | 4                         | 30,000                             | 14.53 (5.72)                | Double helical           | Integral double-row cylindrical roller |
| <b>DREAM [7]</b>           | Conceptual                  | Between propellers                   | 20,000 (estimated value) | 7.06:1           | 860                       | NA                        | NA                                 | NA                          | NA                       | NA                                     |
| <b>KHI [51]–[53]</b>       | Detailed Testing            | Between propellers                   | 20,000                   | NA               | NA                        | 5                         | 50,000                             | NA                          | Double helical           | NA                                     |

Gearbox technologies for a geared CROR were proposed during the 1980s in the Advanced Prop–Fan Engine Technology (APET) programme [47], [48]. The technologies were matured during the Advanced Gearbox Technology (AGBT) programme for high-power transmission and long design life at high shaft rotational speeds [49], [50]. The DPGB presented an unrivalled power-to-weight ratio for the power gearbox in the CROR compared to other architectures as well as higher reliability associated with the smaller part-count and reduced size, which are linked to weight and cost reduction [47]–[50].

The design goals and life requirement of the gearbox were achieved in the AGBT programme by keeping the bearing and gear contact stresses low [50]. Of the bearings in the transmission system, the planet bearings experience the highest

loads and would be custom-designed based on the stringent space-constraints imposed by the planet gears. The diameters of the planet gears are a function of the sun gear diameter and the gear ratio. In the DPGB for a CROR, it has been assumed that the smallest gear (the pinion) is the sun gear, which accumulates the largest number of fatigue cycles [54], and hence determines the dimensions of the gears for the input power, torque and speed reduction ratio range [50]. Therefore, gear dimensions provide a preliminary estimate of the volumetric requirements of the DPGB. There might be benefits in having larger gear diameters, e.g. to favour longer lives for the planet bearings. However, the size of the gearset will already provide a first estimation of the feasible CROR design space, subject to the availability of sufficiently compact and adequate bearing technology. An indication of possible bearing technology options is presented in Table 1. The selection of planet bearing technology is highly dependent on the type of planet gear technology and loading case and should be investigated in future work.

Gear dimensions for the DPGB in the CROR depend on gear technology. The selection of such technology requires the introduction of some basic gear terms, which are schematically represented in Figure 7. A more detailed description of these parameters is provided in section 3.1.1, when addressing gear design.

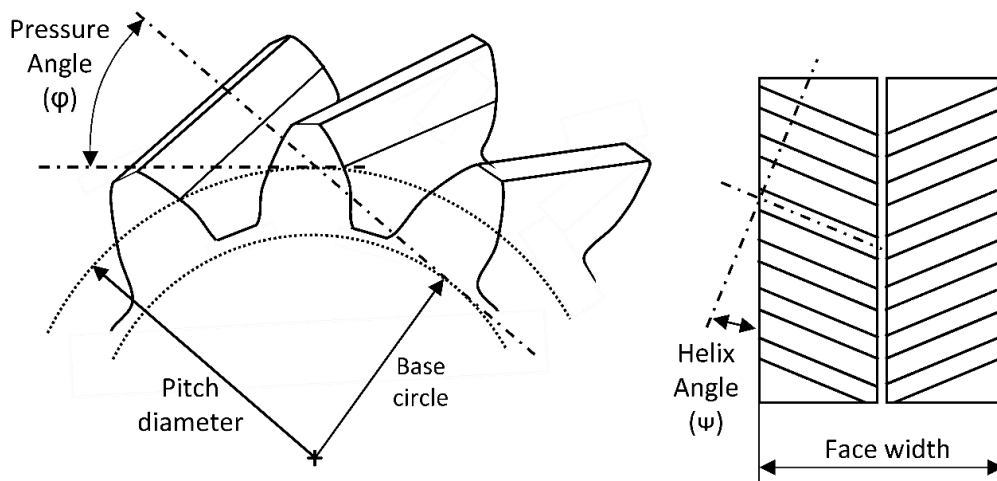


**Figure 7 Helical gear nomenclature**

The APET ([47], [48]) and AGBT ([49], [50]) projects evaluated high-contact ratio (HCR), single-helical and double-helical gears to increase tooth load sharing relative to conventional spur gears (i.e. zero helix angle). Increasing the path of contact reduces tooth unit load and increases load capacity with a reduction in gear vibration.

The increase in contact ratio of HCR gears relative to spur gears (contact ratios of 1.4 to 1.7) derived from slender teeth with low pressure angles. However, studies during AGBT showed that the tooth load reduction of HCR gears did not compensate for the loss of strength with thinner teeth [50]. Single- and double-helical gears increase the tooth path of contact by use of the helix angle. Tests on this type of gears demonstrated proper gear tooth load sharing of helical gearing, low vibration characteristics, and low stress levels [50]. Finally, the results of comparative testing in the AGBT project showed double-helical gears were the best candidate for the DPGB [50].

Double-helical gears have been selected for the DPGB in this research to favour a compact design enabled by their load distribution capability and the cancellation of induced thrust forces from the helical gears. The lower stress levels that double-helical gears present compared to single-helical or spur gears contribute to the durability targets of a high-power CROR gearbox. These targets are defined through the Mean Time Between Unscheduled Removals (MTBUR), which is the standard measure of design life used for gear systems [55].

Another research activity aimed at designing a compact, light and efficient DPGB transmission system for a CROR engine is described in reference [56]. The focus of these assessments was on the design selection of the transmission system and its performance evaluation. The in-line transmission was designed for a 15,000 hp (11.2 MW) power class engine with equal and opposite rotational speed propellers [56], [57]. The study provided a kinematic analysis of the gear system and a representation of the gearbox efficiency characteristics with different input torques and rotational speeds [57].

In the early 2000s, the dramatic rise in fuel prices and the urge for cleaner engine technologies rejuvenated the industry's interest in open rotor technology as one

potential solution to those problems [58]. In project DREAM (validation of Radical Engine Architecture systems) performance and mechanical aspects of a DPGB were incorporated into the engine design setup for a geared CROR [6], [7]. The unavailability of a mechanical preliminary design methodology for the transmission system was highlighted and a methodology was proposed to estimate its dimensions and weight based on the preliminary design of the gearset. The DPGB preliminary design was based on the mechanical strength calculation of the gears to comply with pitting and tooth breakage criteria, which were not specified. The final geometry of the gears in DREAM as per Ref. [7] was obtained iteratively only using standard modules (i.e. ratio of pitch diameter to number of teeth [59]) with the specified design requirements. The width of the gears was also pre-selected. Therefore, the most compact gear design may not have been obtained with the design constraints reported in Ref. [7], limiting the potential impact of the gearbox design on the engine general arrangement. The reasons behind this would be:

- In Ref. [7], as in Ref. [50], the speed ratio is an input to the design, defined as the speed reduction between the input and output shafts. If the front and rear propellers rotate at the same nominal speed, the torque ratio (TR) between the output shafts of the gearbox is also imposed, as shown later in chapter 3 by substituting Eq. (3-7) in Eq. (3-9).
- In Ref. [7], the DPGB speed reduction was based on an estimated upper limit of the permissible stress levels for a three-stage LPT. While this limit must be taken into account in the design process, it should be used as a constraint rather than as an input for the DPGB sizing.
- The research reported in Ref. [7] additionally considered the sun shaft diameter as an input to the design (obtained from the mechanical model of the shaft of the LPT) and set the value for the minimum sun gear diameter. However, as described in Ref. [50], the sun shaft is a quill shaft whose diameter may differ from the LPT shaft and can be adjusted to the gear requirements. This facilitates the DPGB design, as the vibration effects of a long, thin shaft operating at high rotational speeds, i.e. the LPT shaft, do not have to be taken into account for the preliminary volumetric

assessment of the transmission system. If the constraint on the minimum sun gear diameter is removed and the sun pitch diameter is not known, the gearbox dimensions are undefined.

- The study reported in Ref. [7] considered the LPT last-stage hub diameter to be an upper limit for the gearbox outer diameter. However, if the correct area ratio is preserved between the outlet of the LPT and the engine core nozzle, the maximum radial dimensions of the transmission system would not need to fall within this limit.

A schematic representation of the resultant differences in the available design space for the transmission system can be seen in Figure 8, based on the design approaches followed in DREAM, as per Ref. [7], and this research.

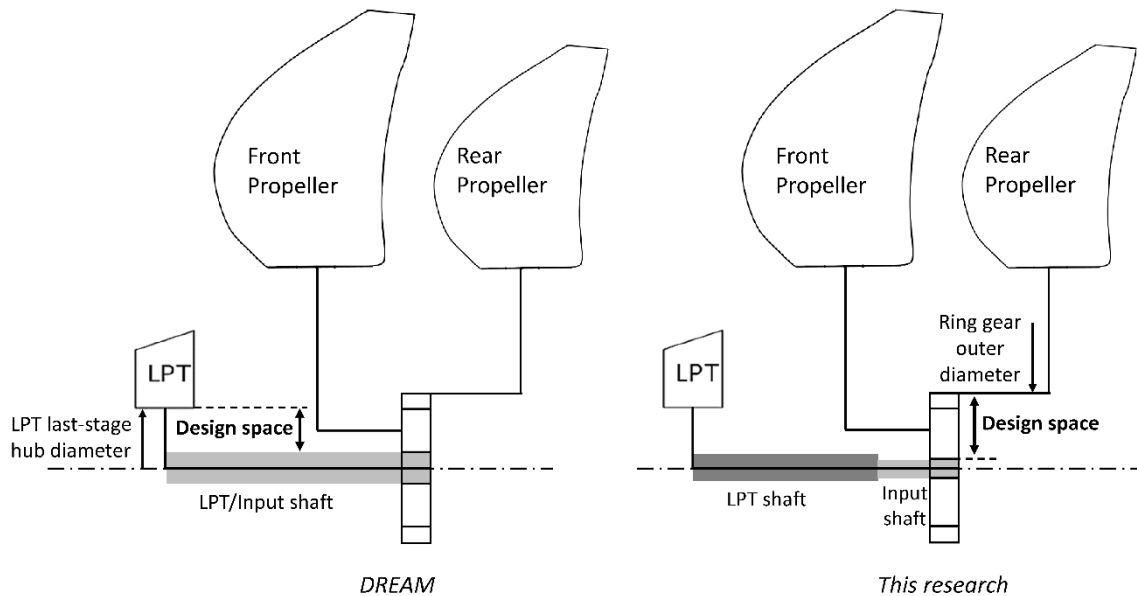


**Figure 8 Increase in design space between DREAM (as per Ref. [7]) and this research favoured by higher constraint flexibility**

The development of a gearbox pre-design tool for aero-engines is reported in Ref. [60]. The gearbox geometry was obtained from engine specifications, with missing data obtained from correlations. The geometry of the gears was calculated iteratively, assessing the stress levels on the gears for a given number of teeth, tooth size, typical pressure and helix angles, and material, to comply with specified safety factors. The planet bearing sizes and shaft dimensions were estimated as a function of the planet gear design and user inputs. A range of

gearbox designs was produced, varying the module and the number of teeth, and the changes in gearbox mass with compactness were evaluated. The tool was validated against internal and published data (NASA's AGBT gearbox [50]), with a maximum geometrical deviation of 3.54% stemming from the differences in the normal module. No information is reported on the modelling of a gearbox for a CROR other than the validation case using information from the AGBT project.

In Japan, Kawasaki Heavy Industries (KHI) investigated a geared CROR engine and the design of its transmission system with a prototype of a DPGB developed to demonstrate TRL appropriate for whole engine development [51]–[53]. Limited information is provided on the design of the DPGB prototype. The reduction ratio and the rotational speeds of the propellers and LPT are not specified and the design process is not described. However, these publications indicate the expected DPGB durability for modern CRORs.

Table 2 shows the areas of research addressed by previous projects on geared CRORs. There is a clear need for an alternative approach to estimate the size of the DPGB in a CROR application, which does not involve convoluted mechanical assessments. The mechanical design of the gearbox is either treated in isolation ([51]–[53]) or is not used to inform the engine model ([7], [49], [50]).

**Table 2 Overview of methodology and scope of DPGB design in this research relative to previous research projects**

| Project         | Torque Ratio ( $TR$ ) selection based on |           | Trade-off between CROR performance and DPGB size |
|-----------------|------------------------------------------|-----------|--------------------------------------------------|
|                 | Performance                              | Mechanics |                                                  |
| AGBT [49], [50] | ✓                                        | ✗         | ✗                                                |
| DREAM [6], [7]  | ✓                                        | ✗         | ✗                                                |
| KHI [51]–[53]   | –                                        | ✓         | ✗                                                |
| This research   | ✓                                        | ✓         | ✓                                                |

### **Gear Design Methodologies for Gearbox Sizing**

Based on the literature review described above, the gearset in a DPGB is assumed to determine the volumetric requirements of the power gearbox, which can comply with the power and durability specifications of the application. The

sizing process of the gears is a function of the pinion pitch diameter and the tooth count [48]. Two commonly-used gear preliminary design methods to estimate the basic geometrical characteristics are the Q-factor method and applying gear manufacturing standards [54]. The Q-factor method is a common way to start the preliminary design of a gearset if previous experience exists for the specific application and details of the method can be found in any mechanical design book (e.g. Ref. [54]).

If no experimental data are available, the estimation of the stresses acting on the gears is one of the main challenges of the preliminary gear design, where several assumptions have to be made to account for the effects of stress concentrations, and gear misalignment or tooth errors [54]. An alternative procedure for calculating the pinion pitch diameter for optimum load capacity is presented in Ref. [61]. The method is based on the classical Hertz equation for surface compressive stress on two curved surfaces [54]. It provides a means of estimating the preliminary gear design for a novel gearset based on allowable stresses when no previous experimental data are available. However, scoring factors and gear misalignment are not considered [48], resulting in optimistic designs.

The second parameter to consider for gear sizing is the tooth-count estimation. A closed-form procedure for designing minimum-weight spur and helical gearsets is presented in Ref. [62]. The pitch diameter is calculated based on surface fatigue following the approach in Ref. [61] and the optimum number of teeth is obtained, simultaneously satisfying the surface fatigue and bending constraints in the pinion.

The understanding of gears has evolved since the formulation of these methods, as reflected in gear manufacturing standards [63]. Therefore, the contribution of this research to gear preliminary design is the updating of the methodologies presented in Refs. [61] and [62] and their incorporation to the design framework of the transmission system in a geared CROR pusher arrangement.

### 2.3.2 The Pitch Change Mechanism

The pitch control mechanism is a key element for novel engine architectures like the UHBR turbofan and the CROR to ensure the correct controllability of the fan and propellers respectively. Engine integration of this mechanism is even more important in the case of the CROR, where the propellers are directly exposed to the freestream without a cowl to redirect the flow. The pitch control mechanism in the CROR enables

- the adjustment of the blade angle to accommodate changes in power to the propellers,
- the conservation of constant propeller rotational speeds,
- the improvement of the speed range in which the propellers can operate and,
- the recovery of most of the swirl energy with the rear propeller.

The pitch angle of the blades in the CROR changes collectively in each propeller but independently between rows. Changes in pitch are required to adjust the thrust of each propeller and compensate for variations in the angle of attack caused by changes in axial flow and air density with altitude.

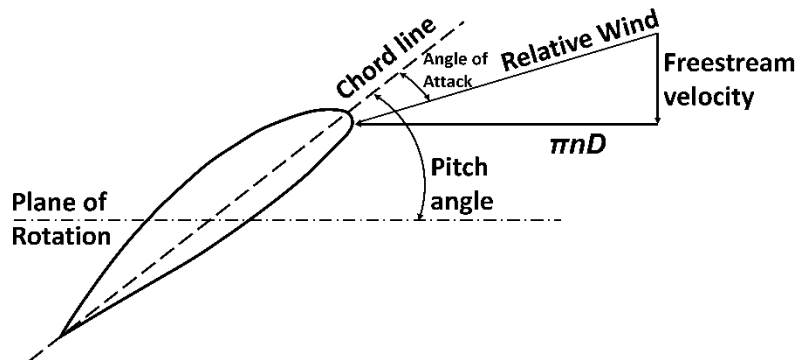


**Figure 9 Pitch angle definition**

Pitch angle is defined by the angle between the plane of rotation and the chord line of the blade, as sketched in Figure 9. Small pitch angles (fine pitch) are common for low-speed take-off. Large pitch angles (coarse pitch) are used at high-speed operation. The feathered position represents a 90° pitch angle, used in case of power failure to minimise drag.

The blade pitch is automatically changed to maintain a constant propeller rotational speed irrespective of the flight condition. All external forces acting on the propeller hub, the blade retention system and the pitch control mechanism have their source in the blade loads and are derived from the inertia and aerodynamic characteristics of the blade [64].

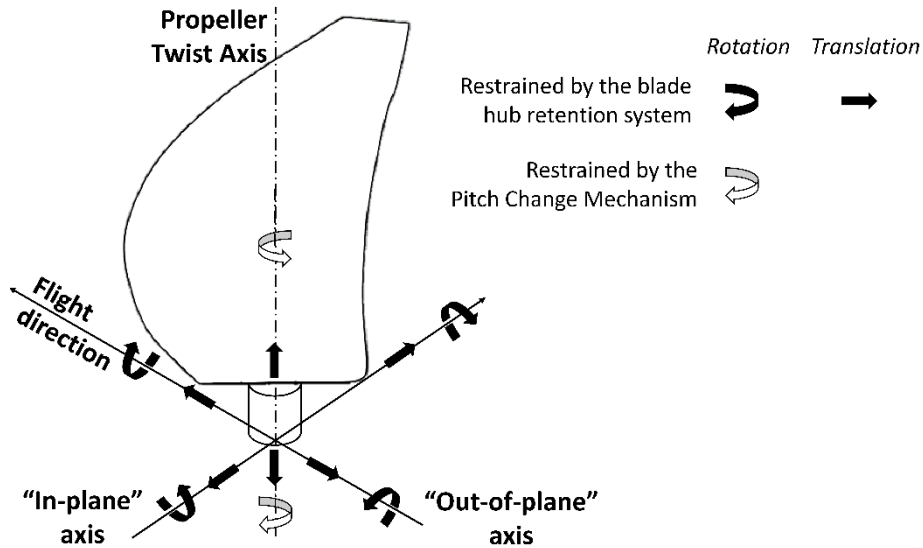


**Figure 10 Schematic representation of the blade-generated forces**

Figure 10 shows the blade-generated forces and moments at the hub. Except for the rotation forces about the twist axis (i.e. the pitch change axis), all the loads are reacted by the blade hub retention system [65]. Figure 11 shows the essential components of the blade hub retention system, which involve the blade shank, blade bearing, blade-hub attachment and the hub barrel [66].

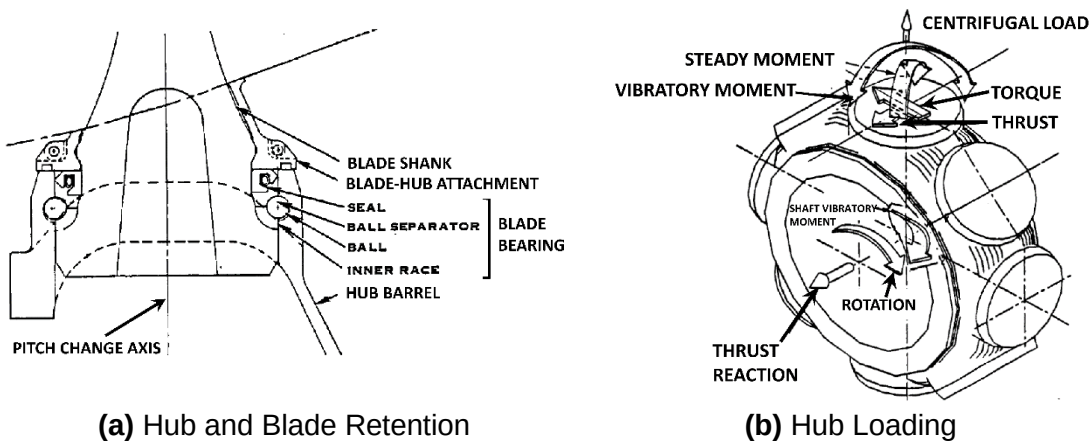


**Figure 11 Main components in the blade retention system (adapted from [64])**

The pitch control mechanism is not part of the blade retention system. It is the mechanism in charge of controlling the pitch angle of the propellers and comprises control and power systems. The control system provides the speed governing, feather function, and hydraulic supply to actuate the power system. The power system traverses the gas path to transmit the pitch change to the propellers and is referred to hereafter as the pitch change mechanism (PCM).

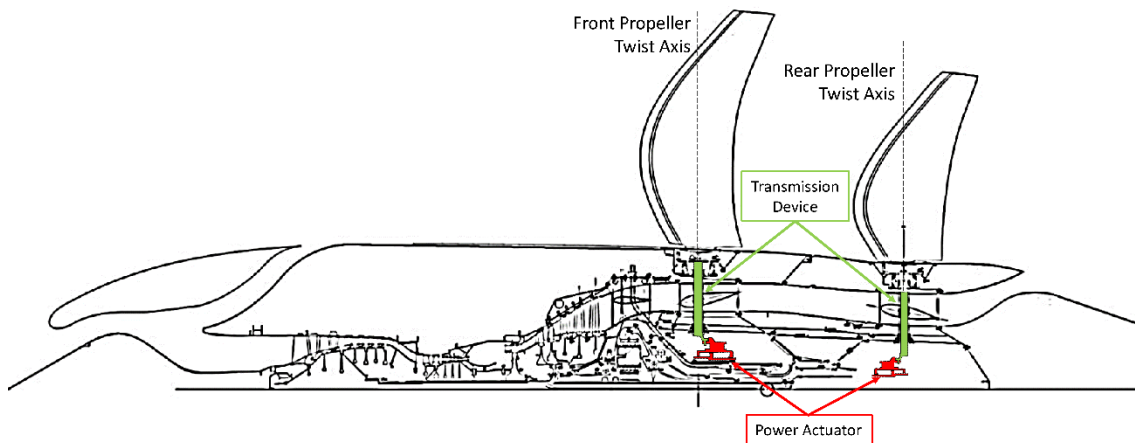


**Figure 12 Schematic general arrangement for a CROR (Safran Aircraft Engines) with main components in the pitch change mechanism (adapted from Deliverable 1.1 of European Commission Grant No. 633436 as cited by Ref. [67])**

The PCM comprises a power actuator and a transmission device (Figure 12).

**The PCM power actuator:** Component/s coaxial with the axis of rotation of the propeller. It comprises a power supply and a prime mover connected to the transmission device. A stationary or rotary power-transfer device might be present between the prime mover and the transmission device.

Figure 13 supports the explanation of the working principle of the power actuator, and the transfer of motion through its subcomponents. The movement of the power actuator in a PCM initiates through the change in pressure in an oil chamber. The axial movement of a linear jack, coaxial to the centre-line of the engine (X-X), is transferred to a load transfer bearing. The outer ring of this bearing thereby pivots a lever arm, which is connected to radial control shafts. The pivoting of these shafts about the radial Z-Z axis (i.e. the pitch change axis)

is linked to the movement of the propeller blades, turning the root support of each blade in the rotary ring that carries them.

The separation of the power actuator part of the PCM into an internal and an external independent portion facilitates the mounting and maintenance of the device. The separation also facilitates the prevention of oil leakages since the internal section can be kept in place inside the hub of the propellers when maintenance is needed in the external part of the mechanism.

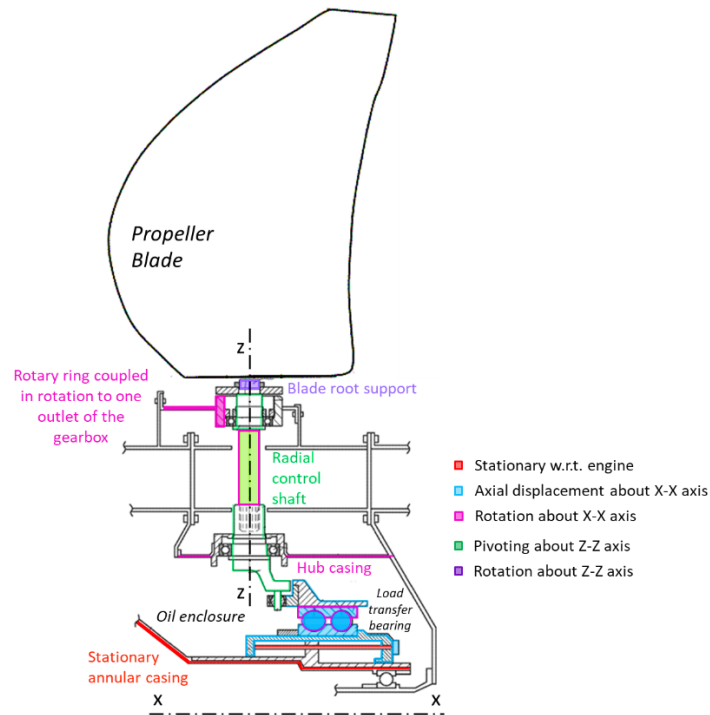


**Figure 13 Schematic representation of the PCM (adapted from Ref. [68])**

**The PCM transmission device:** Component positioned radially to the axis of rotation of the propellers. It connects the mobile part of the power actuator to the base (shank/ root) of the blades. The transmission device transmits the torque generated by a propeller blade about the retention radial axis and translates the axial motion of the actuator to blade rotation about the same axis.

The component assuming the pivot connection between the hub and the blade and the attachment of the blade root is not part of the PCM. Since the PCM is intended to be located in the rear part of the CROR engine, a heat-insulation element is present between the blade insert and the pivot. The presence of a heat

insulation element preserves the pivot from the heat transmitted by the radial shaft.

### **Early assessments of PCM Design for the CROR**

In the 1980s NASA investigated various PCM technologies for single- and contra-rotation arrangements within the APET [48] programme. The key requirement for these mechanisms was their compatibility with the in-line power gearbox (i.e. the DPGB) design in terms of connections and space envelope.

The selection of appropriate pitch control concepts was the result of trade-off studies conducted before APET by Hamilton Standard based on consideration of the power and control systems. Figure 14 shows the concepts evaluated, where possible technology configurations were defined through the combination of subcomponents from bottom to top. Transmission devices, prime movers and power supplies for the pitch control mechanism were evaluated and weighed with a power matrix using the following parameters in order of importance: “safety, reliability, maintainability, weight, performance (accuracy of blade angle control), acquisition cost, impact on gearbox, technical risk, envelope, heat generation (efficiency)” [48].

Within each group, the following technologies were deemed inappropriate for the CROR [48]:

- **Transmission device:** Arrangements with cams and bevel gears shown on the far left of the matrix incurred in elevated weight penalties associated with the large dimensions required for the specific application. The coupling of a ball nut or ball screw with a spring no-back pitch lock led to unsatisfactory blade angle control.
- **Power transfer:** Slip rings incurred high maintenance costs. Transformers and generators driven at propeller rotational speed are heavy. Oil transfer bearings presented low reliability and maintainability for the large diameters required by in-line gearbox installations [48]. However, improvements in this technology are apparent from its reconsideration for modern advanced PCM arrangements. Interest in oil transfer bearings and

their improvement is apparent from recent PCM research [69]. Thrust bearings that transmit pitch-change and pitch-lock loads across the rotating interface rated low on reliability, maintainability, and weight.

Finally, the seven configurations highlighted in Figure 14 indicate the technology combinations deemed suitable in APET for the pitch control mechanism in the CROR application [48]. Out of these configurations, the relative simplicity of the hydraulic piston concept out-rated other configurations, indicated by yellow boxes in Figure 14. The rotary hydraulic concept (green boxes) rated higher than the linear hydraulic concept (blue boxes) in terms of weight and adaptability to contra-rotating propellers [48]. However, both concepts were considered viable for future studies.

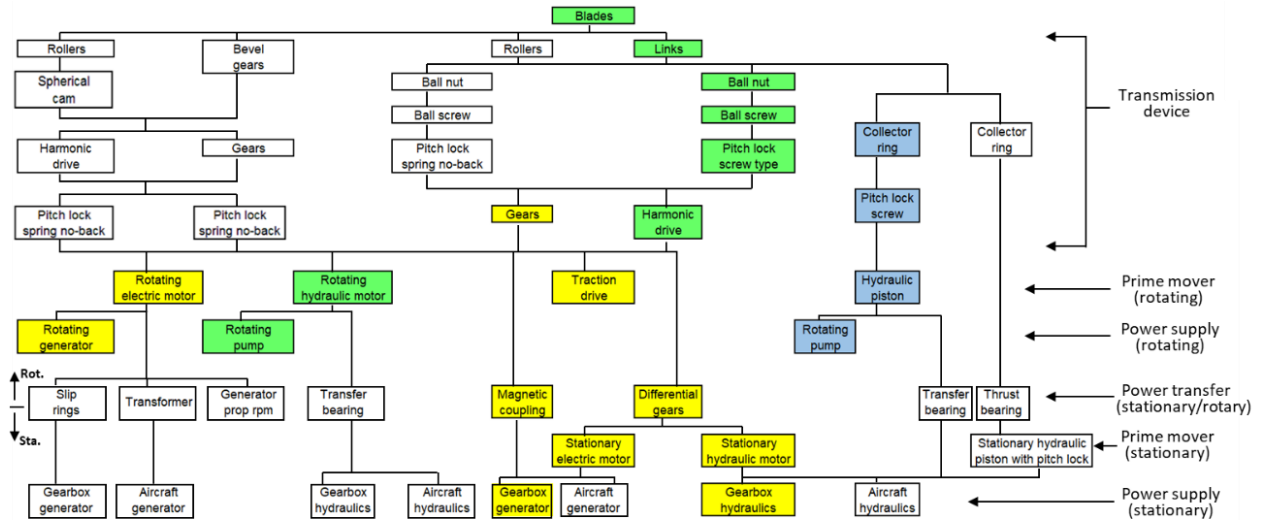


**Figure 14 PCM power system matrix (adapted from Ref. [48])**

Lessons learnt from the APET project (Ref. [48]) have been used to inform the PCM selection in the current research.

### More Recent Assessments of PCM Design for the CROR

More recent projects on Open Rotor Engines Advanced Technologies (OREAT [70] and OREAT II [69]) followed a similar approach to the APET project [48] for the selection of potential concepts. A PCM technology was preliminary designed and developed (up to TRL 5) for a CROR with entry into service (EIS) 2030. Contrary to the approach adopted in APET [48], the PCM in OREAT avoided the

complexity added by pitch-lock mechanisms by making the blade counterweights baseline. These are required to prevent control failures under over-speed or high-drag conditions.

The OREAT consortium identified that there were no western-type approved contra-rotating propeller systems and aimed at developing a simple, sturdy and reliable PCM. The PCM studies in Ref. [69] opted for a linear arrangement for the CROR, relying on mature standard components. The CROR incorporated a PCM on each propeller that allowed for independent control of each rotor stage. The OREAT consortium opted for a single, co-axial actuator driving a collective yoke with individual link rods, a crank lever and a torqueshaft connecting to the root of each blade [70]. The transmission of the loads goes from the blade hub, through the PCM transmission device, to the power actuator.

The research that was undertaken in APET [48] and more recently in OREAT [69], [70] concluded that the relative simplicity of an oil-driven hydraulic piston actuator out-rated any other configuration.

## **2.4 CROR Multi-Disciplinary Preliminary Design**

It has already been made apparent in section 2.3 that the CROR presents a high potential to reduce aircraft fuel burn through an increase in propulsive efficiency. However, there are challenges associated with this engine architecture (e.g. noise, installation, integration of a complex transmission system) that need to be addressed at engine and aircraft level. The solutions proposed to overcome these challenges affect various aspects of engine and aircraft design, performance and operation. Therefore, any radical design changes have to be incorporated as part of the engine's preliminary design phase, where the interactions between components and disciplines can be assessed and improved.

This research fills an existing gap with the development of preliminary design tools to model the transmission system at the same level as the other components in the engine model and simulate their impact on the feasible design space of the CROR.

The high number of unknowns associated with the preliminary design of an engine would make the individual evaluation of each design parameter very time-consuming. In the case of a novel propulsion system architecture, such as the CROR, an even higher level of uncertainty exists due to the presence of additional components (e.g. the power gearbox or the pitch change mechanism). The existence of potentially conflicting objectives would turn the optimisation of some of these parameters into the worsening of others. Consequently, a multi-disciplinary parametric analysis of the engine at mission level is required to assist in the design process and elucidate specific technical solutions. The outcome of this type of design approach would provide insight into the impact of key preliminary design parameters on the overall design.

A design space exploration or a parametric study are particularly interesting ways to identify trends of the design parameters and optimum design regions that could be further explored in later phases of the design process. The tools used in this research to carry out multidisciplinary parametric assessments are based on physics, empirical correlations or simplified design methods. They also include non-dimensional characteristics of components (0-D) and require relatively simple geometrical definitions (normally 1-D or 2-D). The simplified nature of these methods enables establishing trade-offs between engine design variables (e.g. rotational speeds and torque ratios) and overall mission performance (e.g. mission fuel burn). This approach aids in the development process when compromises between design parameters are required as a consequence of performance, operational or regulatory constraints. Absolute values for the design parameters would be obtained at later stages of design, when the technical feasibility of a particular design can also be assessed with specific tools. The nature and complexity of the tools may vary according to the stage of the project, knowledge of the particular technology, required level of detail and available computational power.

The multi-disciplinary design assessments described above are carried out within a simulation platform or framework<sup>3</sup>. A key aspect of multi-disciplinary design frameworks is their modular structure. The term modularity in the context of the multi-disciplinary design indicates that each discipline can be constructed and modified independently from a library of interchangeable modules within the framework. The integrated disciplines are developed in separate modules and accessed through an adapted user interface, benefiting from the extensibility and ease of maintainability of the overall framework.

Tools with such simulation and modelling capabilities are available in the industry, for example, Pratt & Whitney's PMDO (Preliminary Multi-Disciplinary Design Optimization) [71] and MTU's MOPEDS (Modular Performance and Engine Design System) [72]. In the past decade, the number of multi-disciplinary aero-engine simulation frameworks available in the public domain has experienced exponential growth. Various simulation frameworks have been developed by the research community and are available in the public domain for this type of assessments: TERA (Techno-economic Environmental Risk Analysis) [73], EDS (Environmental Design Space) [74] and GTlab (Gas Turbine Laboratory) [75].

## **2.5 Contributions from the Research**

The present study focusses on the modelling and integration of the transmission system in a pusher geared CROR. As research progresses in CRORs, a more complete assessment of the transmission system is vital to fully understand the practical design constraints and the feasible design space of the architecture. Previous research has primarily focused on the performance and design of the turbomachinery components in the CROR. Therefore, there is a gap in research to develop tools and more refined methods for the simulation and design of the power gearbox and pitch change mechanism.

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<sup>3</sup> The word framework in this research is defined as a computer software designed to couple multi-disciplinary analyses and to facilitate the transfer of information between input, execution and output phases of those analyses.

The design of the transmission system must comply with performance, space envelope and durability requirements of the engine application. This leads to the first research question: *Can the preliminary design of the transmission system be done solely from cycle parameters without imposing additional engine-based constraints?* In the preliminary design phase, engine constraints are guesses to the design. Therefore, they may unnecessarily limit the feasible design space of the CROR.

Past research has tackled the design of the power gearbox independently of the CROR engine modelling ([49]–[53]), or has been limited by many pre-determined engine constraints [7], [76]. This indicated the need for a simple preliminary design approach that does not involve complicated mechanical assessments and can be integrated with engine preliminary design studies. In regards to the PCM for a CROR, published research is scarce and has tackled the design problem independently of the engine modelling ([47], [48], [69], [70]). Therefore, the first aim of this research can be summarised as follows:

TO DEVELOP A TRANSMISSION SYSTEM PRELIMINARY DESIGN  
METHODOLOGY BASED ON CYCLE PARAMETERS TO ASSESS,  
COMPARE AND SELECT SUITABLE CONCEPTS FOR THE CONTRA-  
ROTATING OPEN ROTOR

The reason for this aim is twofold: speed up the design process and take into account aspects of the transmission system preliminary design that past literature had not considered. The former is achieved through the development of a methodology for space envelope calculation that does not require an iterative procedure. The latter refers to the removal of engine-prescribed constraints subsequently increasing design flexibility for the transmission system and exhaust duct, i.e. flexibility in area ratio evolution from LPT exit to core exhaust nozzle.

The logical question that follows is: *How does the introduction of more rigorous preliminary design processes for the main components in the transmission system influence the design space of the engine?* Based on the literature reviewed in this research, it is the author's opinion that there is a clear gap in the

literature regarding the integration of the transmission system in the CROR. The modelling and the interference in the exhaust flow and engine gas path of the radial shafts connecting the PCM to the propellers in a pusher CROR have not been tackled by previous research. Therefore, the second research aim can be expressed as follows:

TO DEFINE THE IMPACT OF THE TRANSMISSION SYSTEM ON THE  
FEASIBLE DESIGN SPACE OF THE OPEN ROTOR ENGINE

### 3 Preliminary Mechanical Design Methodology for the Power Gearbox in a Contra-Rotating Open Rotor

This research focusses on integrating and assessing the impact of the transmission system on the performance and gas path of the CROR. Therefore, a key aspect of modelling the transmission system is the development of a power gearbox preliminary design model that can be integrated with the engine model. To that end, a design framework (Figure 15) was required to model and simulate the performance and mechanical requirements of the power gearbox compliant with the CROR requirements. This framework may then be used to performing parametric engine assessments through its integration in the preliminary design CROR model, where the mechanical constraints imposed by the power gearbox determine the feasible design space.

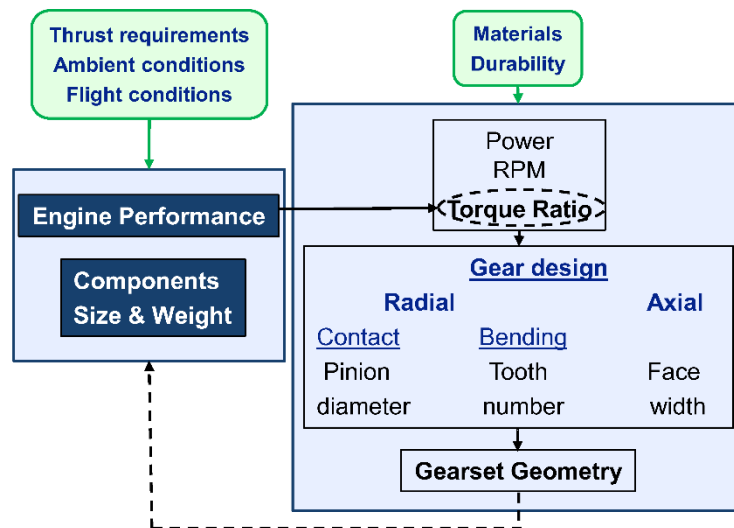


**Figure 15 Developed preliminary design framework for the power gearbox**

This research proposes that a preliminary estimate of the power gearbox envelope (i.e. the volumetric requirements) be obtained from the dimensions of the gearset. Therefore, an up-to-date methodology is proposed for the design of maximum load capacity gears for the CROR application based on the power transmitted, durability and space envelope requirements. Parametric and sensitivity analyses have been undertaken to assess the impact of the design assumptions on gearbox dimensions. The developed framework also enables the identification of feasible torque ratios between the output shafts of the speed-

reducer and the CRP driven by them. The impact of the torque ratio on the size of the gearbox has been analysed for equal propeller rotational speeds and different speed ratios between the output shafts.

The literature reviewed in section 2.3.1 showed that a differential planetary gearbox provided a compact and light solution for the CROR. The relative diameters of the gears in the DPGB define the achievable reduction ratio between the input and output shafts. The basic equations defining the operation of the DPGB are presented in Eqs. (3-1) to (3-5). The power ( $PW$ ) balance between the input and output shafts in the DPGB shown in Eq. (3-1) can also be expressed as a function of the torque ( $Q$ ), and rotational speed ( $n$ ) of the sun ( $S$ ), ring ( $R$ ) and carrier ( $C$ ) shafts and the mechanical efficiency of the gearset ( $\eta_{DPGB}$ ), as per Eq. (3-2).

$$\eta_{DPGB} PW_S + PW_C + PW_R = 0 \quad (3-1)$$

$$\eta_{DPGB} n_S Q_S = n_R Q_R + n_C Q_C \quad (3-2)$$

When the carrier rotates coaxially with the input (sun) shaft, the planets not only revolve about their axes but also orbit and rotate about the centre of the DPGB assembly, as seen from the kinematics of the system shown in Figure 16.

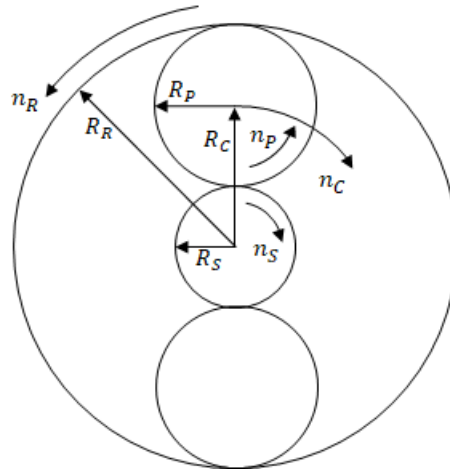


**Figure 16 Kinematics of the differential planetary gearbox**

Assuming the same force is experienced by the teeth in all the gears in the gearbox, Eq. (3-2) can be written in terms of the gear pitch radii ( $R$ ) as shown in Eq. (3-3).

$$\eta_{DPGB} n_S R_S = n_R R_R + n_C (R_S + R_R) \quad (3-3)$$

Therefore, the torque on the carrier ( $Q_C$ ) and ring ( $Q_R$ ) shafts is expressed, in Eq. (3-4) and Eq. (3-5) respectively, as a function of the input torque ( $Q_S$ ), the mechanical efficiency of the gearset ( $\eta_{DPGB}$ ) and the gear ratio ( $m_G$ ), i.e. the ratio of gear diameters as defined later by Eq. (3-13) in section 3.1.1.3.

$$Q_C = \eta_{DPGB} Q_S 2 (1 + m_G) \quad (3-4)$$

$$Q_R = \eta_{DPGB} Q_S (1 + 2 m_G) \quad (3-5)$$

In this research, the torque split or torque ratio is defined as the ratio of torque in the carrier, connected to the front propeller, to torque in the ring shaft, connected to the rear propeller. Therefore, the DPGB and the CROR share a common preliminary design variable that is the torque ratio between the contra-rotating propellers, i.e. between the output shafts of the speed-reducer.

The propellers are the main source of thrust in the CROR. Therefore, the torque ratio between them defines the amount of power required by each propeller row to comply with the engine's thrust requirements. The power into the front propeller ( $PW_{FP}$ ) relative to the rear propeller ( $PW_{RP}$ ) is proportional to the torque ratio (TR) and the ratio of rotational speeds in the front and rear propellers ( $n_{FP}$  and  $n_{RP}$  respectively), as shown in Eq. (3-6).

$$\frac{PW_{FP}}{PW_{RP}} = TR \frac{n_{FP}}{n_{RP}} \quad (3-6)$$

In the absence of gear geometry information for the DPGB, the engine performance inputs can then be used to get an estimate of the gearset design. The torque ratio between the output shafts of the DPGB is directly linked to the ratio between the diameters of the gears (the gear ratio). The characteristics of the DPGB gearset impose a torque ratio rather than a speed ratio between the

input and output shafts and an unequal torque split between output shafts. The reduction ratio in the DPGB is, therefore, expressed by means of the speed ratio ( $SR$ ) defined in Eq. (3-7) as the ratio between the nominal rotational speed of the sun and ring shafts.

$$SR = \left| \frac{n_S}{n_R} \right| \quad (3-7)$$

If both propellers rotate at the same nominal speed ( $|n_C| = |n_R|$ ), Eq. (3-8) represents the relationship between  $TR$  and  $SR$ .

$$TR = \frac{SR + 1}{SR - 1} \quad (3-8)$$

However, the definition of  $TR$  presented in Eq. (3-8) does not contemplate the possibility of having different rotational speeds for the propellers in the CROR ( $|n_C| \neq |n_R|$ ). Therefore, Eq. (3-9) presents a more generic expression of  $TR$ .

$$TR = \frac{SR + 1}{SR - \left| \frac{n_C}{n_R} \right|} \quad (3-9)$$

The DPGB in this research is designed for a prescribed life (durability) of 30,000h based on the information gathered from the literature and summarized in Table 1. The design goals for the power gearbox correspond to the necessary design objectives that would make the geared CROR competitive with current turbofan engines, primarily in terms of reliability and maintainability, efficiency, and low system weight. These designs goals pose a challenge in the gear design for high-power aero applications [42] due to the physical size of the driving gear system, the efficiency of the gear system and the reliability and time on wing (i.e. the durability). The high power demands of the power gearbox in modern aero engines are also challenging in terms of transmission system integration within the engine.

### 3.1 Preliminary Design Methodology for the Differential Planetary Gearset

This study presents a new framework to calculate the space envelope of the transmission system suitable for a CROR application. There are two key elements of the proposed framework to size the DPGB: the development of the preliminary gear sizing methodology and the torque ratio selection between DPGB output shafts, i.e. between CROR propellers. In turn, the gear sizing methodology has two main steps:

1. Sizing of the pinion pitch diameter based on the torque, material selection, gear ratio, total derate factor, contact stress, normal pressure angle, helix angle and profile contact ratio of the gears.
2. Calculation of the gear number of teeth for compliance with torque ratio requirements based on pitting resistance and bending strength calculations.

The required inputs for the preliminary design of the gearbox are the transmitted-power from the LPT, the prescribed design life of the gearset, the rotational speed of two out of the three rotating shafts and an initial value for TR. The gearset is designed at end-of-runway (EOR) take-off condition (sea level,  $M_0 = 0.25$ , ISA+15 K), where torques are maximum.

#### 3.1.1 Sizing of the pinion pitch diameter

The minimum pitch diameter for the pinion gear is a function of the applied load and the allowable surface compressive stress and can be obtained for maximum load capacity from the classical Hertz equation [77]. Therefore, the pitch diameter for the pinion helical gear in a planetary gearset can be calculated from Eq. (3-10) [61].

$$d \geq \left[ \frac{0.7 \frac{Q}{q} E (m_G + 1) \cos^2 \Psi K_d}{S_c^2 \left( \frac{F}{d} \right) m_G \sin \varphi_n \cos \varphi_n m_p} \right]^{\frac{1}{3}} \quad (3-10)$$

$Q$  is the torque on the pinion gear,  $q$  is the number of power paths (i.e. the number of planets),  $E$  is the modulus of elasticity,  $m_G$  is the gear ratio,  $\psi$  is the helix angle,  $K_d$  is the total derate factor [77],  $S_c$  is the contact stress experienced by the pinion gear,  $(F/d)$  is the gear aspect ratio,  $\varphi_n$  is the normal pressure angle and  $m_p$  is the profile contact ratio. The constant 0.7 derives from the elastic coefficient ( $C_p$ ) relating the material properties of mating gears, assuming both are made of the same material and have a Poisson ratio of 0.3 as for most steels [63]. A further explanation of these terms is provided in sections 3.1.1.1 to 3.1.1.10.

The substitution of each of these parameters in Eq. (3-10) determines the minimum diameter of the sun gear (the pinion) capable of complying with the power transmission and durability requirements of the application. Subsequently, the radial and axial dimensions of all the gears in the gearset, i.e. planet gears and external gear, can be calculated. The pitch radii of the planet gears equal the product of the gear ratio ( $m_G$ ) and sun radius ( $R_S$ ). For symmetric gears, the radius of the external gear or ring gear ( $R_R$ ) results from the addition of the sun gear radius and the planet radii ( $R_P$ ) as shown in Eq. (3-11).

$$R_R = R_S + 2R_P \quad (3-11)$$

Finally, the face width ( $F$ ) of the gearset is approximated from the product of the resultant pinion pitch diameter ( $d$ ) and the gear aspect ratio ( $F/d$ ) assumed appropriate for the aerospace sector (refer to section 3.1.1.7).

This process represents a first step to assessing whether a transmission system is suitable for a specific CROR application. The following sections (3.1.1.1 to 3.1.1.10) explain the parameters defining the pinion pitch diameter ( $d$ ) in Eq. (3-10) and present the selection process for the appropriate values.

#### **3.1.1.1 Torque ( $Q$ )**

The propeller loads of the CROR (thrust, weight and pitch couples) are absorbed by the gearbox housing and the carrier and ring shafts, substantially isolating the gearset to ensure the desired design durability [50]. Consequently, the torque transmitted from the LPT to the sun gear is the only load considered for the preliminary mechanical design of the pinion gear.

### 3.1.1.2 Material Selection

The type of material determines the behaviour of the gear under loading. It has a direct impact on the allowable stress of the gears and ultimately on their dimensions. Aerospace gears used in primary drive systems, such as the power gearbox in a CROR, are manufactured from heat-treated steel alloy materials and are surface-hardened to ensure resistance to pitting and to provide tooth-bending strength [78]. In this research, a generic carburized and hardened Grade 3 (premium-quality) steel is assumed to be representative of high-grade aerospace gears [79]. The choice of this type of carbon steel leads to maximum allowable contact and bending stress values of 275,000 psi (1896 MPa) and 75,000 psi (517 MPa) respectively [63] and an elastic constant ( $E$ ) of 30 Mpsi (207 GPa) [80].

### 3.1.1.3 Gear ratio ( $m_G$ )

The gear ratio  $m_G$  is defined as the ratio between the larger and the smaller number of teeth ( $N$ ) in a pair of gears, proportional to the ratio of their pitch-circle radii ( $R$ ), which is never less than 1.0 [59]. It is a function of the pinion pitch circle diameter ( $d$ ) and the centre distance at which the gear pair operates. Therefore, the gear ratio,  $m_G$ , for the CROR application can be defined as the ratio of the planet ( $P$ ) to sun ( $S$ ) pitch-circle radii. Following this definition, the gear ratio can also be expressed as a function of the external gear radius ( $R_R$ ) through the gearbox ratio ( $R_{DPGB}$ ) in Eq. (3-12).

$$R_{DPGB} = \frac{R_R}{R_S} \quad (3-12)$$

However, in the preliminary design of the DPGB for the CROR, there might not be enough information regarding the dimensions of the gears. In the absence of this information, the gear ratio  $m_G$  can be expressed as a function of an engine-related parameter such as the torque ratio between the propellers ( $TR$ ) as presented in Eq. (3-13).

$$m_G = \frac{N_P}{N_S} = \frac{R_P}{R_S} = \frac{R_{DPGB} - 1}{2} = \frac{TR - 2}{2(1 - TR)} \quad (3-13)$$

The  $TR$  between the output shafts is defined in Eq. (3-14) as the nominal ratio between the torque in the carrier ( $Q_C$ ) and ring ( $Q_R$ ) shafts, connected to the front ( $FP$ ) and rear ( $RP$ ) propellers respectively. This definition implies that the  $TR$  is always greater than one, and the propeller connected to the carrier shaft always experiences the highest torque.

$$TR = \left| \frac{Q_C}{Q_R} \right| = \left| \frac{Q_{FP}}{Q_{RP}} \right| = \frac{2(1 + m_G)}{1 + 2m_G} = \frac{R_{DPGB} + 1}{R_{DPGB}} \quad (3-14)$$

Due to the geometrical relations of the gears in a DPGB, an even torque split between the two propellers is not possible. An initial value of 1.33 is selected for  $TR$ , which corresponds to an  $m_G$  of 1.01 as per Eq. (3-13), as in previous DPGB assessments for open rotors [50]. Higher  $TR$  figures are possible, but are not considered here as the planet gears rather than the sun gear may then become the life-limiting gear component. The value of  $m_G$  is linked to the torque ratio between the output shafts as shown in Figure 17.

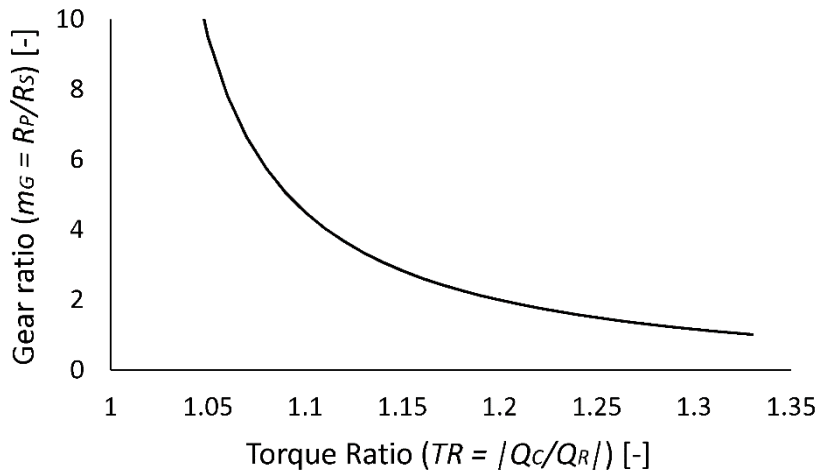


**Figure 17 Gear ratio ( $m_G$ ) against torque ratio ( $TR$ ) for the DPGB**

#### **3.1.1.4 Number of Planets ( $q$ )**

The number of planets in the DPGB is a function of the gear ratio ( $m_G$ ) and the number of teeth in the meshing gears [81]. In the absence of information on the number of teeth, a factor  $K_q$  is introduced in Eq. (3-15) to calculate the number of planets and prevent possible meshing interference between them.

The author has conducted a sensitivity analysis of  $K_q$ . A  $K_q$  value of 0.96 results in a big number of small teeth and a value of 0.92 leads to a small number of larger teeth. Therefore, a mean value between the two, i.e. 0.94, has been assumed for the CROR application. The maximum number of planets in the DPGB will be the next lower integer from Eq. (3-15), which can also be expressed as a function of  $TR$ .

$$q = \frac{K_q \pi}{\text{asin}\left(\frac{m_G}{1 + m_G}\right)} = \frac{K_q \pi}{\text{asin}\left(\frac{2 - TR}{TR}\right)} \quad (3-15)$$

### 3.1.1.5 Total derate factor ( $K_d$ )

The total derate factor [77] accounts for the design imperfections in the gear design and is defined in Eq. (3-16). There are different sources of derate, which can be grouped into the overload, dynamic, size, and load distribution factors ( $K_o$ ,  $K_v$ ,  $K_s$ , and  $K_m$  respectively). These factors are commonly selected based on the analysis of actual operating conditions and the aerospace industry will have established proprietary values based on experience. Therefore, the values proposed in this research are estimates of factors that could be used for gears in a modern CROR application.

$$K_d = K_o K_v K_s K_m \quad (3-16)$$

The selected values are for a highly loaded, high-speed aero gear application and are defined as follows:

- The overload factor ( $K_o$ ) considers the effect of all externally applied loads above the nominal tangential load [63]. The power source to the DPGB in this study is a gas turbine with a free power turbine running at constant-speed (at take-off, i.e. design point) and the driven elements are contra-rotating propellers, which are assumed to cause only a light shock to the operating conditions, leading to a value of  $K_o = 1.25$  [82].
- The dynamic factor ( $K_v$ ) makes allowance for internally generated gear tooth loads, which are induced by non-conjugate meshing action of the gear teeth, with values ranging between 1.09 and 1.15 for higher to lower

accuracy grades, i.e. tolerance levels [83]. This study assumes maximum precision of the gear design, hence a value of  $K_v = 1.09$ .

- The size factor ( $K_s$ ) accounts for the unevenness in material properties [63]. For aerospace helical gears, the gear size is not considered to have a detrimental effect, hence  $K_s$  has a value of unity [78].
- The load distribution factor ( $K_m$ ) modifies the rating equations to reflect the non-uniform distribution of the load along the lines of contact [63]. Following the calculation procedure in Ref. [63], a value of  $K_m = 1.4$  is used in this study.

The resultant total derate factor can be calculated substituting the different factors in Eq. (3-16). The selected values apply to any novel aero-engine gear design with similar technology and application characteristics. Therefore, a resultant value of  $K_d = 1.9$  is used to calculate the pinion pitch diameter from Eq. (3-10).

#### 3.1.1.6 Contact stress ( $S_c$ )

If the size of the pinion gear were known, the contact stress ( $S_c$ ) would be calculated using the information on the material properties, the geometry of the gear, and the transmitted torque [63]. In the absence of the geometry,  $S_c$  is assumed to equal a derated value of the allowable stress ( $S_{ac}$ ) of the material as per Eq. (3-17). The derate factors for contact stress account for uncertainties in the design and are commonly referred to as “safety factors”. However, some of these uncertainty factors have been removed from the unknown area of the pitting safety factor ( $S_H$ ) and are considered to be predictable within the design method [63].

$$S_c \leq \frac{S_{ac} Z_N C_H}{S_H K_T K_R} \quad (3-17)$$

The hardness factor ( $C_H$ ), temperature factor ( $K_T$ ) and reliability factor ( $K_R$ ) have a value of unity for the application under analysis (refer to Ref. [63] for further details). The pitting resistance factor ( $Z_N$ ) adjusts the values of  $S_{ac}$  for gear design lives other than  $10^7$  cycles and accounts for the relationship between the stress and the number of cycles (S-N characteristics) for the gear material. In high-

speed helical gears, the value of  $Z_N$  should not exceed 0.68 [83] when the design life exceeds  $10^{10}$  cycles as would be the case in modern CRORs.

$$Z_N = 2.466 n_L^{-0.056} \leq 0.68 \quad (3-18)$$

In Eq. (3-18), the number of cycles ( $n_L$ ) is a function of the gear design life ( $L$ ) in hours, the rotational speed of the pinion ( $n_S$ ) in RPM and the number of contact paths ( $q$ ), equivalent to the number of planets in the DPGB. It is apparent from the expression in Eq. (3-19) that the higher  $n_S$  and the higher the design life requirements, the bigger the number of cycles experienced by the gear.

$$n_L = 60Ln_Sq \quad (3-19)$$

#### 3.1.1.7 Gear aspect ratio ( $F/d$ )

The gear aspect ratio ( $F/d$ ) represents the relationship between face width ( $F$ ) and pitch diameter ( $d$ ) of the pinion gear. It is an indicator of gear sensitivity to misalignment [84]. In the absence of experience, AGMA proposes Eqs. (3-20) and (3-21) as an initial approximation of  $F/d$  for single- and double-helical external gears respectively in industrial gearsets [84]. The width of the groove or gap between the helical gears is not included in the face width that makes contact with a mating gear, so the axial length of the double-helical gears will be greater than  $F$ . Figure 18 shows gear size estimations obtained with these equations.

$$\frac{F}{d} = \frac{m_G}{m_G + 1} \quad (3-20)$$

$$\frac{F}{d} = \frac{2m_G}{m_G + 1} \quad (3-21)$$

The approximation in Eq. (3-21) for double-helical gears favours gearbox designs with face widths larger than the pitch diameter of the pinion gear. Wider gears would offer higher resistance to pitting fatigue and result in smaller overall radial dimensions for the DPGB for the same gear ratio ( $m_G$ ). However, the estimated gear aspect ratios in Eqs. (3-20) and (3-21) will be conservative for most applications [84]. One such application is the aerospace industry, where gear standards are intended as guidelines [54].

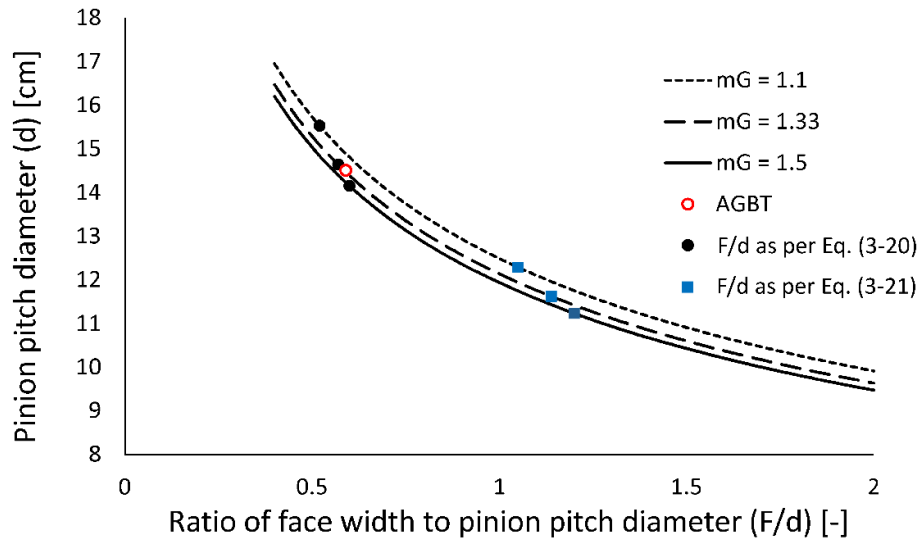


**Figure 18 Variation of pinion pitch diameter ( $d$ ) with gear aspect ratio ( $F/d$ ). Marked  $F/d$  values for the AGBT gearbox [50] and estimates with Eqs. (3-20) and (3-21) [84]**

The trend followed by the aerospace industry favours narrower face widths, with values of  $F/d$  around 0.6. Examples of this trend are the power gearbox in the PW1100G geared turbofan (approximately 0.62 [85], [86]) or the DPGB in NASA's AGBT project indicated in Figure 18 (approximately 0.6 [50]). One reason behind this choice for aero gas turbines stems from the need for compact engine designs. The length and space between components need to be minimised because longer engines result in weight, performance and installation penalties. Finally, higher gear quality and the more stringent design practices in the aerospace sector enable better load distribution in the gears [54]. Therefore, Eq. (3-20) offers a better correlation of  $F/d$  with gear ratio ( $m_G$ ) for double-helical aero gears than Eq. (3-21).

### 3.1.1.8 Normal pressure angle ( $\varphi_n$ )

The pressure angle in the normal plane of a helical tooth is referred to as  $\varphi_n$  [59]. The normal pressure angle represents the angle at which the load is transmitted from one tooth to another one in mesh. High values of  $\varphi_n$  can provide higher load capacity, but also worse smoothness and noisier gear meshing [50]. For high-speed helical gears, the value of  $\varphi_n$  should not be greater than  $25^\circ$  [83]. For this study, a value of  $22.5^\circ$ , as proposed in Ref. [50], is selected. The combination of  $\varphi_n$  and helix angle of the gears has an impact on the profile contact ratio ( $m_p$ ), as

shown in Figure 19, which will also affect the size of the pinion gear ( $d$ ) according to Eq. (3-10).

### 3.1.1.9 Helix Angle ( $\psi$ )

The helix angle represents the tooth tilt in the lengthwise direction of the gear [54]. High helix angles increase the load sharing capacity of the gear and must be of the same degree at each mesh. The selection of the helix angle is a compromise between the strength of the teeth and smooth operation of the gears. Double-helical gears cancel out the thrust loads and allow higher helix angles than single-helical gears. For advanced high-speed helical gear units, their helix angles typically range from  $30^\circ$  to a maximum of  $45^\circ$  [54]. A high helix angle,  $\psi = 45^\circ$  was selected in this study to achieve a compact DPGB design for the CROR application.

### 3.1.1.10 Profile contact ratio ( $m_p$ )

The profile contact ratio represents the number of teeth in contact at each mesh cycle in the plane tangent to the tooth surface. It has a big impact on gear design as it determines the teeth excitation, hence vibration level and noise [77]. A higher number of teeth in contact reduces the load carried by each pair of teeth and reduces the dynamic forces during engagement and disengagement. The relationship between the profile contact ratio and the normal pressure angle ( $\phi_n$ ) on the teeth is presented in Figure 19 for different helix ( $\psi$ ) values.

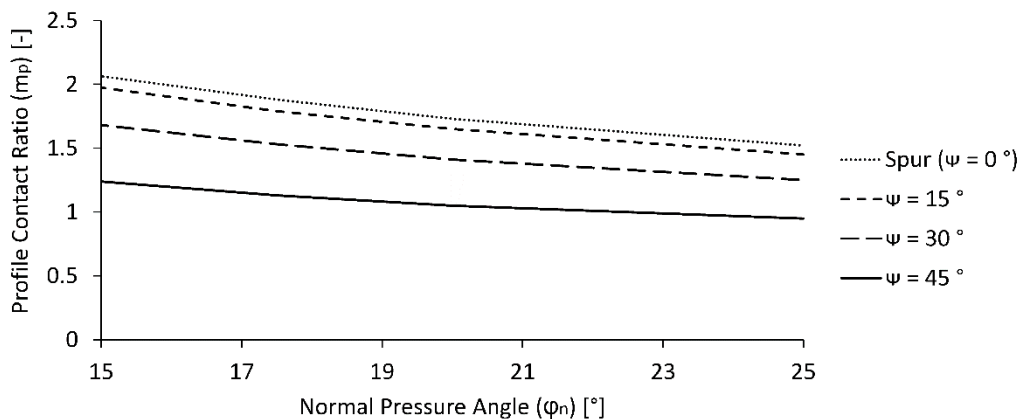


**Figure 19 Normal pressure angle ( $\phi_n$ ) against contact ratio ( $m_p$ ) for different helix angles ( $\psi$ ) (adapted from Table 4.5 in Ref. [54])**

### 3.1.2 Calculation of the gear tooth number for compliance with torque ratio requirements

The calculation of the number of teeth in a gearset must simultaneously comply with tooth resistance to contact and bending stress requirements. It is common practice to use standards such as AGMA 2001-D04 [63] to rate gear teeth, i.e. determine the allowable transmitted power for pitting resistance and bending strength. For each gear mesh, the power rating expressions in Ref. [63] can be re-arranged to define constants for pitting resistance ( $K_c$ ), in Eq. (3-22), and bending strength ( $K_t$ ), in Eq. (3-23) [84]. These constants are proportional to the product of face width ( $F$ ) and pitch diameter ( $d$ ) squared, which is a simplified representation of the solid gear volume and is indicative of the gear weight [87].

$$K_c = F d^2 = \frac{Q}{q} \frac{1}{I} K_d C_f \left( \frac{C_p}{S_c} \right)^2 \quad (3-22)$$

$$K_t = \frac{F d^2}{N} = \frac{Q}{q} \frac{1}{J} K_d \frac{1}{S_t} \quad (3-23)$$

$N$  is the number of gear teeth,  $Q$  is the torque on the gear,  $q$  is the number of load paths (number of planets in the DPGB),  $K_d$  is the total derate factor defined in Eq. (3-16),  $C_f$  is the surface condition factor for pitting resistance, which equals unity assuming appropriate surface conditions [63],  $C_p$  is the elastic coefficient [63] (related to material properties of mating gears),  $S_c$  is the contact stress defined in Eq. (3-17), and  $S_t$  is the bending strength defined in Eq. (3-25).  $I$  and  $J$  are the geometry factors for pitting and bending resistance respectively. For external helical gears, such as the sun gear in the DPGB, Ref. [84] indicates a value of 0.5 for  $J$  and the approximation of  $I$  as

$$I = \frac{1 + 0.00682 \varphi_n}{4.0584} \frac{m_G}{m_G + 1} \quad (3-24)$$

where  $\varphi_n$  is the normal pressure angle in degrees and  $m_G$  is the gear ratio, defined in Eq. (3-13). If the size of the pinion gear were known, the bending stress ( $S_t$ ) would be calculated using the information on the material properties, the geometry of the gear and the transmitted torque [63]. In the absence of the

geometry,  $S_t$  is assumed to equal a derated value of the allowable stress ( $S_{at}$ ) of the material. Some of the uncertainty factors (i.e. derate factors) have been removed from the unknown area of the bending safety factor ( $S_F$ ) and are considered to be predictable within the design method [63].

$$S_t \leq \frac{S_{at} Y_N}{S_F K_T K_R} \quad (3-25)$$

The temperature factor ( $K_T$ ) and the reliability factor ( $K_R$ ) have a value of unity for the application under analysis [63]. The bending resistance factor ( $Y_N$ ) adjusts the values of  $S_{at}$  for gear design lives other than  $10^7$  cycles and accounts for the relationship between the stress and the number of cycles (S-N characteristics) for the gear material. In high-speed helical gears, the value of  $Y_N$ , defined in Eq. (3-26), should not exceed 0.8 [83] when the design life exceeds  $10^{10}$  cycles, where  $n_L$  is the number of cycles as defined in Eq. (3-19).

$$Y_N = 1.6831 n_L^{-0.0323} \leq 0.8 \quad (3-26)$$

The method proposed by Errichello in Ref. [62], used in this research, defines the optimum number of teeth in the pinion gear ( $N_S$ ), for optimum load capacity, as the ratio between the constants for pitting resistance ( $K_c$ ) defined in Eq. (3-22) and bending strength ( $K_t$ ), defined in Eq. (3-23), resulting in Eq. (3-27).

$$N_S = \frac{K_c}{K_t} = \frac{J}{I} \left( \frac{C_p}{S_c} \right)^2 S_t \quad (3-27)$$

The method proposed in Ref. [62] dates back to 1989 and was included in 1992 in the AGMA 901-A92 information sheet for gear design [84]. In April 2020, AGMA reaffirmed the use of this method to preliminary design helical gear pairs in the revision of the information sheet. Therefore, the method proposed in Ref. [62] has been deemed appropriate for the preliminary design of the transmission system of a CROR with EIS 2035.

The expression in Eq. (3-27) estimates the optimum number of gear teeth based on load capacity limiting factors. The effects of scuffing (a form of wear), surface (pitting) and bending fatigue directly relate to the number of teeth in the pinion

and limit gear load capacity. The magnitude of these effects is a function of gear geometry, application requirements and material properties. Tooth undercut also sets a lower limit for the number of teeth.

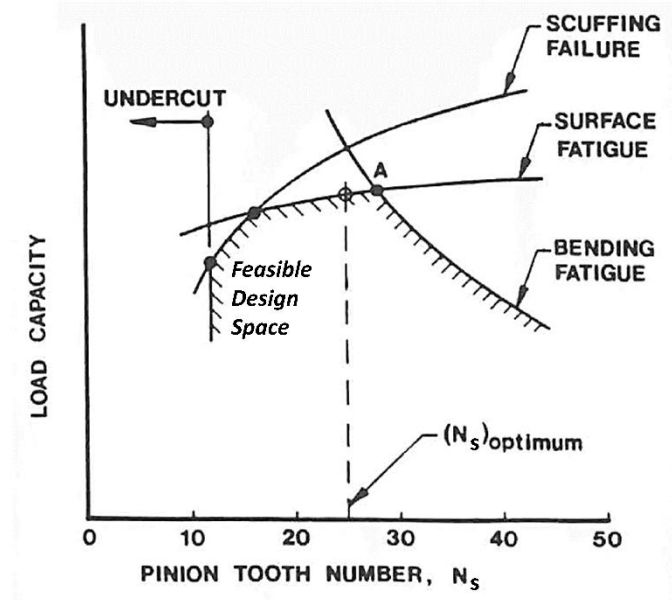


**Figure 20 Schematic representation of the feasible number of teeth in the pinion gear (adapted from Ref. [62])**

Figure 20 shows a schematic representation of the feasible gear tooth numbers defined by the combination of physical factors. Point A represents the point of maximum load capacity, where pitting resistance and bending strength are balanced. Pitting progress is relatively slow, as seen by the almost flat line in Figure 20, and surface fatigue failure progressively manifests through increased gear vibration and noise. In contrast, bending fatigue crack propagation rapidly progresses across the base of a tooth, providing little to no warning before breakage. Therefore, the optimum number of teeth defined by Eq. (3-27) would be located to the left of point A, allowing for a safety margin against bending fatigue. [62]

There is a potential issue in service, mainly relating to tooth bending stress, if the same pairs of teeth keep meeting too regularly. This is often avoided by choosing the tooth count of one of the gears as a prime number. The remaining gears in the set require tooth numbers that are not multiples of the chosen prime to avoid

pairs of teeth on different wheels continually meshing with each other. This would not be a problem if every tooth were perfect and equally spaced to its neighbours, but in practice no tooth is perfect, so the aim of the designer should be to reduce the potential for uneven tooth wear. However, this practice restricts designs to a more limited range of discrete torque ratios and, as in the case of the AGBT [50], is not always applied. Small numbers of teeth should be avoided to minimise undercutting of the teeth on external gears, and large numbers should be avoided because smaller teeth are more sensitive to tolerances.

The number of teeth in the planet gears can be calculated by rearranging Eq. (3-13), and from Eq. (3-28) for the ring gears.

$$N_R = N_S (2m_G + 1) \quad (3-28)$$

At this point, the gear design can be determined based on the calculated pinion pitch diameter ( $d$ ) and its tooth number ( $N_S$ ).

### **3.2 Validation of the Preliminary Design Methodology of the Gearset in the Power Gearbox**

The predictive capability of the proposed methodology has been validated against the geometrical data provided in Ref. [50] for the AGBT project. The double-helical gears in Ref. [50] presented a pressure angle ( $\phi_n$ ) of  $22.5^\circ$ , a helix angle ( $\psi$ ) of  $26^\circ$ , a profile contact ratio ( $m_p$ ) of 1.31, and an approximate gear aspect ratio ( $F/d$ ) of 0.6. The power requirements, speed ratio ( $SR$ ) and rotational speed of the propellers are the inputs to the kinematic model. The substitution of  $SR$  in Eq. (3-8) allows the calculation of  $TR$  and subsequently the gear ratio ( $m_G$ ) from Eq. (3-13). The material selected in Ref. [50] for the pinion (sun) gear was CBS 600, whose properties have been approximated in this study as a Grade 2 case-hardened, carburized steel alloy. The choice of this type of carbon steel led to values of maximum allowable contact and bending stress values of 225,000 psi (1,551 MPa) and 65,000 psi (448 MPa) respectively [63] and an elastic constant ( $E$ ) of 30 Mpsi (207 GPa). The derate factor ( $K_d$ ) is calculated using the values presented in section 3.1.1.5. Finally, the pitting safety factor ( $S_H$ ) is calculated to match the pinion gear diameter ( $d$ ) of the gearset in Ref. [50].

Subsequently, the bending safety factor ( $S_F$ ) is calculated to match the number of teeth in the pinion gear ( $N_S$ ). The outcome of these calculations is a 0.6 % difference in the calculated pinion pitch diameter compared to the AGBT pinion gear as per Ref. [50]. In addition to the pinion pitch diameter, the parameters used for validation of the results are the gear tooth number, face width and ring pitch diameter. Based on these results, the upgraded methodology proposed in this research for the sizing of the DPGB is deemed to be validated and appropriate for the CROR application using a gearbox arrangement similar to the one in Ref. [50].

### **3.3 Parametric Analyses for the Preliminary Design Methodology of the Gearset in the Power Gearbox**

Sensitivity analyses of the parameters affecting the DPGB preliminary design have been performed to overcome the limitation of having only a single validation case and to assess the impact of the assumptions on the design. These assessments have been performed on the DPGB for the baseline CROR defined in sections 5.1 and 5.2, with a gearbox design power requirement at take-off of 20,000 hp (14.9 MW), and both propellers rotating at the same nominal speed (similar to Ref. [7]) unless otherwise specified. The size variation of the gearset has been evaluated for the different parameters defining the size of the pinion pitch diameter ( $d$ ) in Eq. (3-10), and a description of their effect on the preliminary gearbox design is provided later.

#### **3.3.1 Gear design life and number of planets**

The design life (durability) selected for the DPGB, together with the number of planets and the rotational speed of the input shaft, directly affects the number of load cycles that the pinion gear experiences, as expressed in Eq. (3-19). Longer times between unscheduled removals (i.e. higher gear design lives) lead to bulkier gearbox arrangements, as shown in Figure 21.

The design life of transmission systems reported in the literature for open rotor applications ranges from 30,000 h to 50,000 h for older and more modern designs respectively. A prescribed life of 30,000 h has been adopted, as indicated in

Figure 21. It is also apparent that the presence of a higher number of planets results in more compact gearbox designs. For the gear design life of 30,000 h, a DPGB with five planets is 7 % smaller in pitch diameter than one with four planets and 16 % smaller than one with three planets.

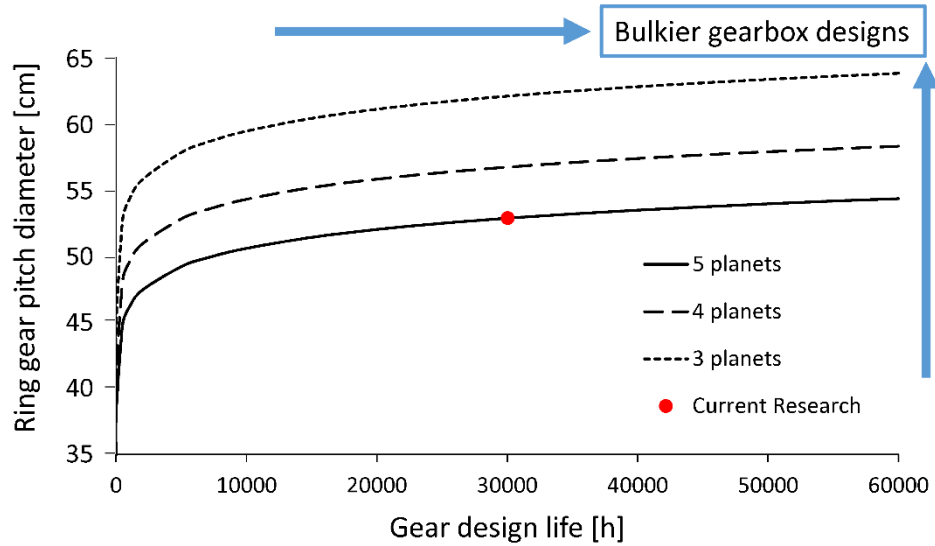


**Figure 21 Change in pinion (sun) gear diameter with a prescribed gear design life**

### 3.3.2 Torque and Material Selection

The transmitted power, proportional to the transmitted torque ( $Q$ ), and the selected gear material also have a direct impact on the size of the gearbox. The torque in the sun shaft is an output of the engine performance calculations. It is apparent from Eq. (3-10) that higher torque values lead to larger gear designs. Additionally, assuming all gears are made of the same steel, the use of materials with a higher surface hardness and modulus of elasticity ( $E$ ), i.e. stiffer materials, results in higher allowable contact stresses on the gears. Consequently, more compact designs can be obtained if advanced steel materials (e.g. Grade 3) are used in future aero applications.

### 3.3.3 Normal pressure angle ( $\phi_n$ ), helix angle ( $\psi$ ) and profile contact ratio ( $m_p$ )

The variation of the normal pressure angle ( $\phi_n$ ), helix angle ( $\psi$ ), and profile contact ratio ( $m_p$ ) of the gears needs to be assessed in combination. The profile contact ratio reduces with the increase of the gear tooth normal pressure angle

and helix angle (refer to Figure 19). The impact of this variation on gear diameter can be seen in Figure 22, where it is apparent that higher values of  $\varphi_n$  and  $\psi$  favour more compact designs.

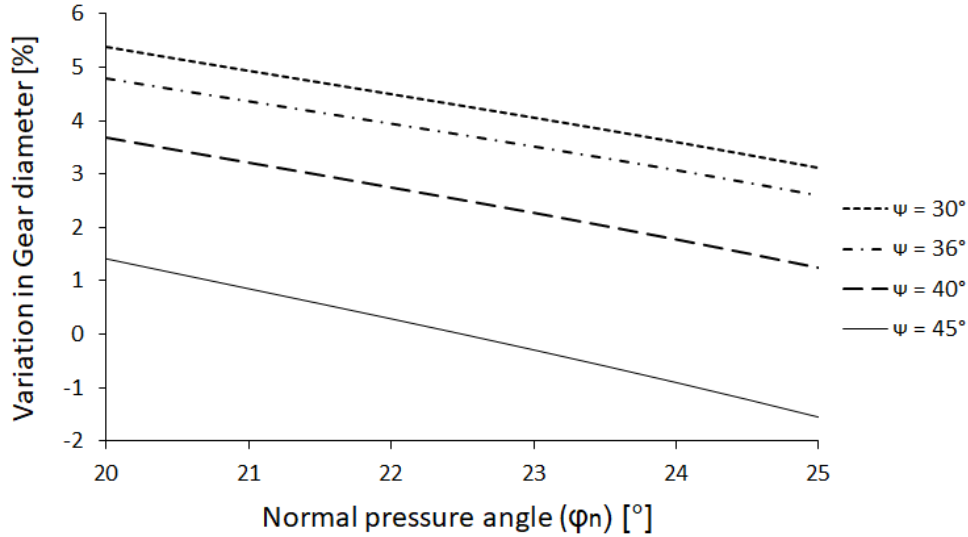


**Figure 22 Variation of the pinion (sun) pitch diameter with normal pressure angle ( $\varphi_n$ ) and helix angle ( $\psi$ )**

A normal pressure angle of  $22.5^\circ$  and a helix angle of  $45^\circ$  have been selected for the DPGB preliminary design in this research to achieve a compact design. However, high helix angles will impose a high degree of twist on the ring section, which presents a slender gear for its diameter. This could be countered with an axially offset ring outer mounting and increasing the stiffness of the ring gear, but this will affect the system's ability to minimise external-load-induced imposed structural distortion. Higher helix angles will also result in higher heat generation stemming from the more highly loaded sliding contact for a given pressure angle. The presented methodology, therefore, provides a means of assessing the impact of variations in normal pressure angle, helix angle and profile contact ratio, on the preliminary gearbox design, but further refinements in the selection of these parameters would require the use of more specific tools.

### 3.3.4 Gear ratio ( $m_G$ ) and torque ratio ( $TR$ )

The gear ratio ( $m_G$ ) of the gearset determines the power or torque split between the gears, as well as the speed reduction that can be achieved with the

transmission system. The size variation of the DPGB has been assessed as a function of these parameters for two main cases: (1) output shafts with equal rotational speeds ( $|n_C| = |n_R|$ ), and (2) output shafts with different rotational speeds ( $|n_C| \neq |n_R|$ ). An evaluation of the effects of  $TR$  on the gearbox preliminary design for the case of unequal rotational speeds of the propellers has not been found in the open literature. Therefore, the presented results could be informative for future performance and noise studies, where the torque ratio design objectives from the engine and the transmission system may be conflicting.

**Output shafts with equal rotational speeds ( $|n_C| = |n_R|$ )** The torque ratio between the output shafts cannot be higher than 1.33 if the sun gear is the pinion as deduced from Eq. (3-13). The characteristics of the DPGB also preclude an equal torque split between the output shafts, so  $TR$  must be higher than unity.

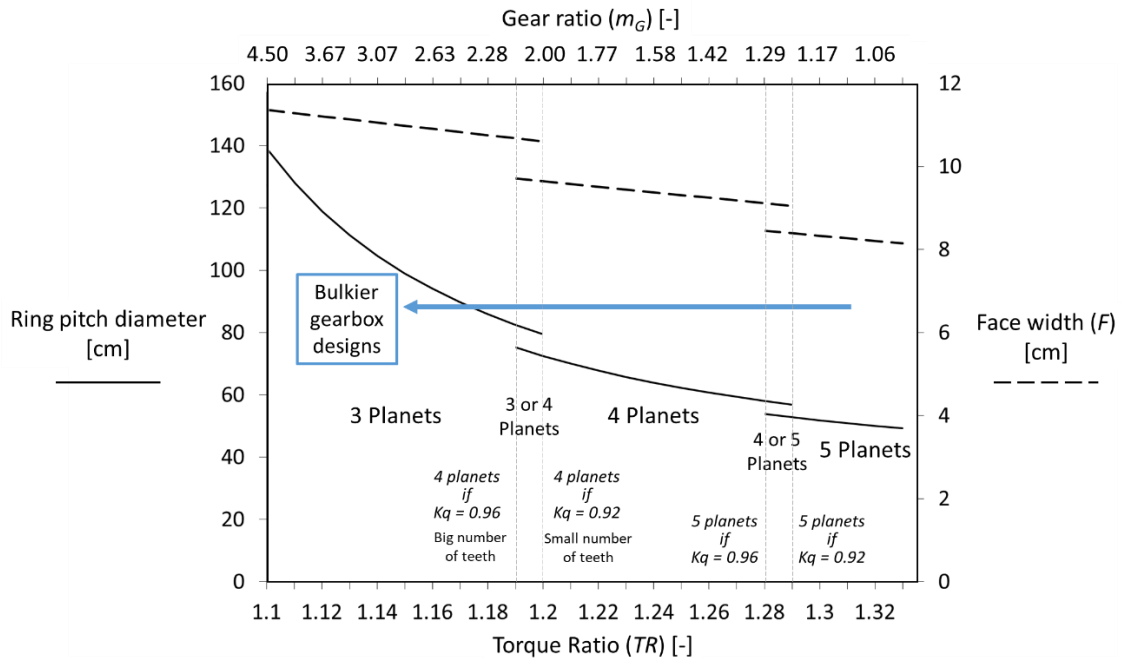


**Figure 23 Variation of ring pitch diameter and face width with torque ratio**

Figure 23 shows the variation of radial (ring pitch diameter) and axial (face width) dimensions of the DPGB with gear ratio ( $m_G$ ) and  $TR$  between the output shafts connected to the propellers. The radial dimensions of the DPGB, considerably higher than the axial ones, may determine the possible locations of the gear train in the CROR. Higher torque ratios result in more compact designs, where the sun is the pinion in the gearset up to a value of  $TR = 1.33$ . For higher values of  $TR$ ,

the planets in the DPGB would be the smallest gears and the methodology proposed in this paper should be applied to them instead.

From the perspective of the transmission system pre-design, a practical lower limit for the torque ratio could only be obtained if a more detailed analysis of its components would be performed with more advanced tools. The speed reduction capability of the gearbox increases with decreasing values of  $TR$ , increasing the rotational speed of the LPT shaft if the rotational speed of the propellers is kept constant. However, in the CROR, the centripetal stresses generated in the LPT would also impact the mechanical feasibility of the propulsor. A possible upper limit in the speed ratio ( $SR$ ) of 10 has been proposed in Ref. [88] for the DPGB in a CROR application “for questions of mechanical feasibility”. This corresponds to a minimum  $TR$  of 1.22 as per Eq. (3-8), i.e. a maximum gear ratio ( $m_G$ ) of 1.75 and a gearbox ratio ( $R_{DPGB}$ ) of 4.5.

**Output shafts with different rotational speeds ( $|n_c| \neq |n_r|$ )** If the carrier shaft is connected to the front propeller and rotates faster than the ring shaft, a given  $SR$  can be achieved at a higher  $TR$  resulting in more compact gearbox designs (Figure 24). Higher torque ratios correspond to lower gear ratios ( $m_G$ ) and a reducing difference between the diameters of the sun and planet gears as  $m_G$  approaches unity. The resultant smaller planet gears would experience higher pitch-line and orbital velocities with higher tooth bending stress and higher centripetal loads imposed on the planet bearings, which could be detrimental to their life.

Figure 24 shows that the higher the  $SR$ , the smaller the torque difference between the propellers. A specified gear ratio ( $m_G$ ) can only be obtained with one ratio of output rotational speeds for each gearbox  $SR$ . The  $SR$  of 10, can be achieved with four or five planets (refer to Figure 23) when the rotational speed of the carrier shaft will be higher than or equal to the ring shaft, i.e. for  $m_G = 1.015$ -1.75, as shown in Figure 24. It is also apparent from Figure 24 that an  $SR$  close to 7 would be the minimum speed reduction attainable with the DPGB if both output shafts rotate at the same speed and the sun gear is preserved as the pinion gear, i.e.  $m_G > 1$ .

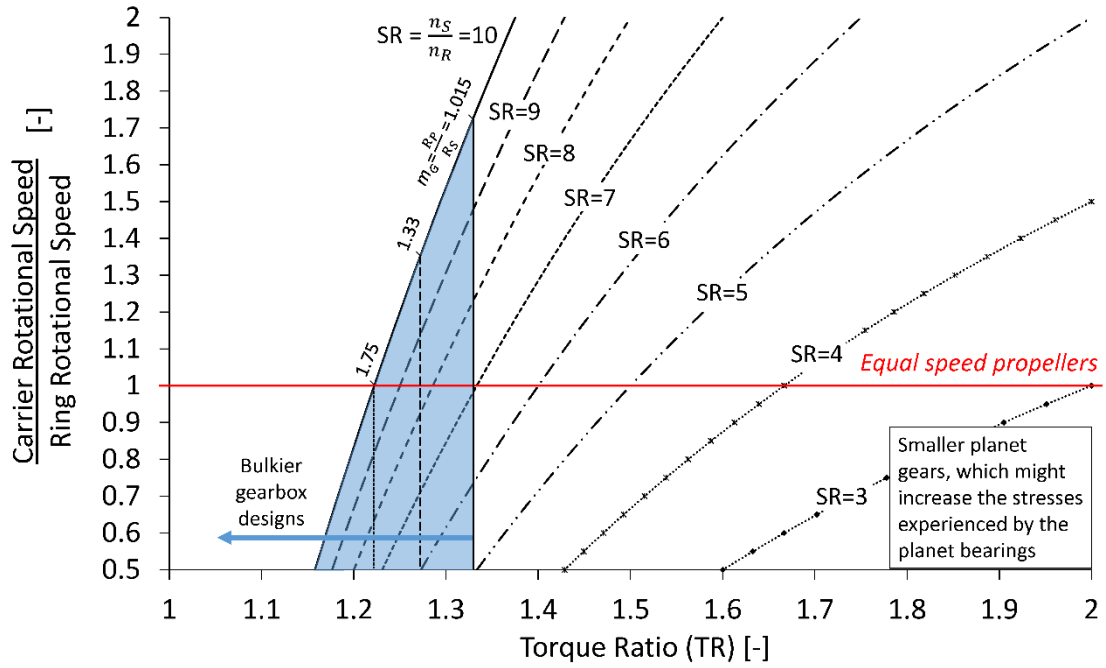


**Figure 24** Variation of  $TR$  and gear ratio ( $m_G$ ) with  $SR$  and rotational speed ratio between the output shafts for the DPGB in a CROR, with highlighted feasible design area for the sun as the pinion gear

## **4 Preliminary Mechanical Design Methodology for the Propeller Pitch Change Mechanism in a Contra-Rotating Open Rotor**

The primary focus of this research is the preliminary design and integration of the transmission system in the CROR. As the result of the literature review presented in section 2.3.2, the pitch change mechanism investigated in this research for the CROR sizing and performance assessments is based on the most recent studies of the OREAT project presented in Ref. [69]. The PCM traverses the gas path to transmit the pitch change to the propellers and comprises

- the power actuator, coaxial with the engine's axis of rotation, which starts the pitch change movements, and
- the transmission device, connecting the mobile part of the power actuator to the pivots of the blades, which is arranged radially to the engine's axis of rotation.

The interference with the gas path is a potential source of blockage to the flow exiting the engine core. The design and envelope of the PCM potentially modify the requirements of cross-sectional area in the exhaust duct. Therefore, modelling and simulation of the pitch control system in this research focusses on the PCM component with the highest impact on CROR performance and gas path definition, i.e. the transmission device. At the time of the study, there is no information available in the literature on predicting the size of the PCM for a modern CROR application.

A preliminary design methodology is developed in this research to provide a preliminary estimate of the PCM envelope. This may then be used to assess the potential blockage that the PCM would cause on the exhaust gas path of the engine, which will be proportional to the number of blades in the propeller. Figure 25 presents a flowchart of the PCM's preliminary design methodology based on the design requirements imposed by the CROR application. The developed methodology is a first step in the assessment of PCM integration in the CROR.

The coupling of the PCM with the CROR performance model described in chapter 5 enables the undertaking of parametric assessment, as discussed in chapter 6.

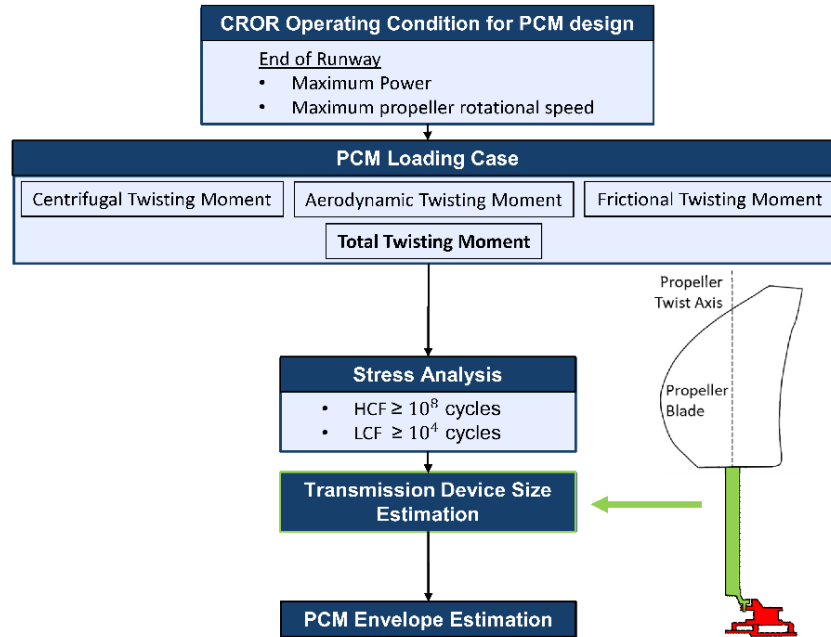


**Figure 25 PCM design methodology**

The loads acting on the PCM determine its envelope (volumetric size) and design. Therefore, the design of the PCM primarily depends on the operation of the propellers. The propeller loads define the actuation force required from the PCM to change or preserve a pitch setting. The PCM reacts the rotation forces about the twist axis. Changes to the operation or the design of the propellers will influence the loads acting on the PCM by changing the magnitude of the reacted loads, and hence its design.

#### 4.1 Design Point Loading Case

The blade and PCM transmission device are assumed a matched set, and hence the loads produced at the blade shank are directly transmitted to the PCM. The operation of the CROR and the aeromechanical characteristics of the propeller blades define the design point loading case for PCM sizing:

- The operating condition requiring blade angle control under the highest propeller loads determines the design point for the PCM. The design point

for the PCM corresponds to take-off/ EOR operating conditions, where the centrifugal loads are maximum [89].

- The aeromechanical definition of the blade determines the inertial and aerodynamic forces generated under specific operating conditions and defines the magnitude of the loads transmitted to and reacted by the PCM.

It is difficult to predict the aeromechanic behaviour of the propeller blades and the loads they generate without detailed information on the blade design and operation. The methodology developed in this research estimates propeller aeromechanical behaviour based on the scaling of load data from an existing blade design, as opposed to a more complex computational fluid dynamics (CFD) analysis. Availability in the literature of consistent performance and aeromechanical information for the CROR propeller blades is, therefore, a requirement. The selected blade design would be representative of the technology level of modern propellers. A complete explanation of the design, structural analysis and aerodynamic modelling of advanced propeller blades for the CROR application can be found in Ref. [90].

Testing and development of new generations of blades for the CROR began in Europe in the early 2000s within the Clean Sky programme and in the U.S. with the partnership between GE, NASA and the Federal Aviation Administration (FAA) [58]. The main difference of these blades relative to designs from the 1980s was the simultaneous optimization for acoustic and aerodynamic requirements with CFD/CAA (computational aeroacoustics) methods. Reference [58] provides an overview of historical and contemporary efforts for CROR blade designs at NASA. A similar review of European CROR research can be found in Ref. [91].

However, comprehensive information relative to the aeromechanic behaviour and performance of the blade designs from these projects is rarely available in the literature. The only advanced propeller blade for which this information is available is the SR-7, developed within NASA's Large-scale Advanced Prop-fan (LAP) project [65]. The SR-7 is a high-sweep blade with a 2.74 m diameter. However, the SR-7 blade design could be scaled to be representative of propellers with tip diameters between 3.6 m and 4.9 m [90]. Even though Refs.

[65] and [90] date back to the 1980s, the lack of availability in the literature of a consistent set of data for newer blade designs makes the SR-7 blade the best candidate for the current research.

Table 3 presents the propeller design parameters of the baseline SR-7 blade and the propellers used for the CROR modelling in the current research. The higher propeller diameters in modern CRORs require higher power levels to produce a prescribed thrust. The SR-7 power loading for a blade diameter of 4.27 m is 595 kW/m<sup>2</sup> at take-off and 257 kW/m<sup>2</sup> at cruise [90]. These values are consistent with what would be expected for a state-of-the-art blade with EIS in 2030 [92].

**Table 3 Propeller design parameters**

|                               |              | <b>SR-7 [93]</b> | <b>CROR<br/>Front Prop</b> | <b>CROR<br/>Rear Prop</b> |
|-------------------------------|--------------|------------------|----------------------------|---------------------------|
| <b>Tip diameter</b>           | <b>[m]</b>   | 2.74             | 4.26                       | 3.83                      |
| <b>Hub-to-tip ratio (HTR)</b> | <b>[-]</b>   | 0.24             | 0.37                       | 0.34                      |
| <b>Rotational Speed</b>       | <b>[RPM]</b> | 1698             | 1100                       | 1100                      |
| <b>Tip speed</b>              | <b>[m/s]</b> | 244              | 245                        | 221                       |
| <b>Number of blades</b>       | <b>[-]</b>   | 8                | 12                         | 9                         |
| <b>Blade weight [65]</b>      | <b>[kg]</b>  | 19               | -                          | -                         |

The rotational speed of the propellers is limited by the blade tip speed that would yield optimum aeroacoustics. The slower rotation of the propellers consequently limits the rotational speed of the LPT. The combination of these factors results in larger LPT designs, which need to accommodate the DPGB. Consequently, modern contra-rotating propellers present higher hub-to-tip ratios (HTR) than older designs. The difference in HTR between the SR-7 blade and the CROR propellers prevents the direct scaling of the blade geometry. However, to a first-order approximation, the aerodynamic and mechanic load information available for the SR-7 blade can be scaled for the propellers with a higher HTR to simulate their aeromechanical behaviour.

Stuermer estimated the aerodynamic behaviour of a modern propeller blade (i.e. tip diameter of 4.27 m and HTR of 0.35) with geometric characteristics similar to the SR-7 through RANS (Reynolds-Averaged Navier–Stokes) computations [94]. The outcome of the performance design assessments was a close match of the design lift coefficient of the modelled propeller with the SR-7 and the production

of similar trends for other aero characteristics, i.e. chord-to-diameter ratio, thickness-to-diameter ratio and twist distribution. To a first-order approximation, the distribution of lift and thrust produced by each blade section is proportional to the lift coefficient. Therefore, the aerodynamic behaviour and the aerodynamic loads generated in the CROR propeller blades in this research are assumed proportional to those of the SR-7 blade.

The scaling of the centrifugal forces depends on the external and internal blade design. The methodology used for scaling the centrifugal loads in this research is a two-step approach. Firstly, the difference in HTR of the CROR blades relative to the SR-7 is addressed. The newer propellers present higher hub-to-tip ratios and higher blade diameters. The contribution of the blade hub region to the overall generation of centrifugal loads is relatively small [95]. Therefore, to a first-order approximation, the contribution of the blade hub to the overall blade centrifugal loads is neglected and the geometry of the CROR blades is assumed to be proportional to the SR-7 design. Based on this assumption, the final step is the direct scaling of the centrifugal loads.

In addition to the forces generated by the propeller blades, the PCM must also overcome blade retention friction. Therefore, the total twisting moment (TTM) reacted by the PCM consists of a centrifugal (CTM), aerodynamic (ATM) and frictional twisting moment (FTM). The characteristics of these moments are discussed in sections 4.1.1, 4.1.2 and 4.1.3 respectively.

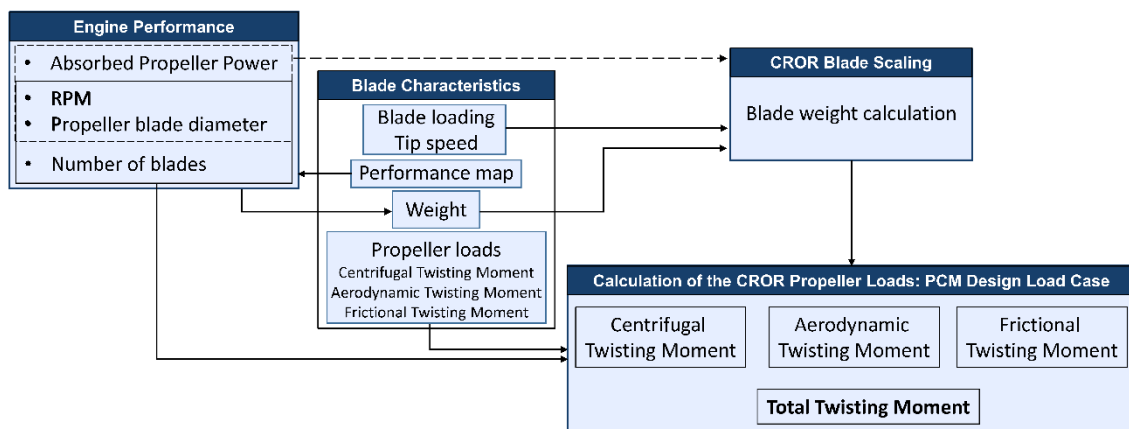


**Figure 26 Flowchart of the PCM design load case defined by propeller loads**

The methodology developed in this research to calculate the propeller loads reacted by the PCM is presented in Figure 26. The development of such a methodology is a preliminary step to estimating the impact of PCM integration on CROR architecture.

#### 4.1.1 Centrifugal Loads and Centrifugal Twisting Moment (CTM)

The centrifugal twisting moment (CTM) is the moment produced by the centrifugal forces acting on the blade. These forces develop due to the operation of the propeller in a radial centrifugal force field. To visualise the effect of the centrifugal load, Figure 27 shows the blade cross-section divided into two centres of mass relative to the propeller twist axis, each of them generating a centrifugal force vector (with radial and in-plane components). Only the in-plane components, in the plane of rotation, have moment arms at a distance “a” and “b” from the stacking axis (coincidental with the pitch axis) and generate a centrifugal twisting moment acting on the propeller twist axis.

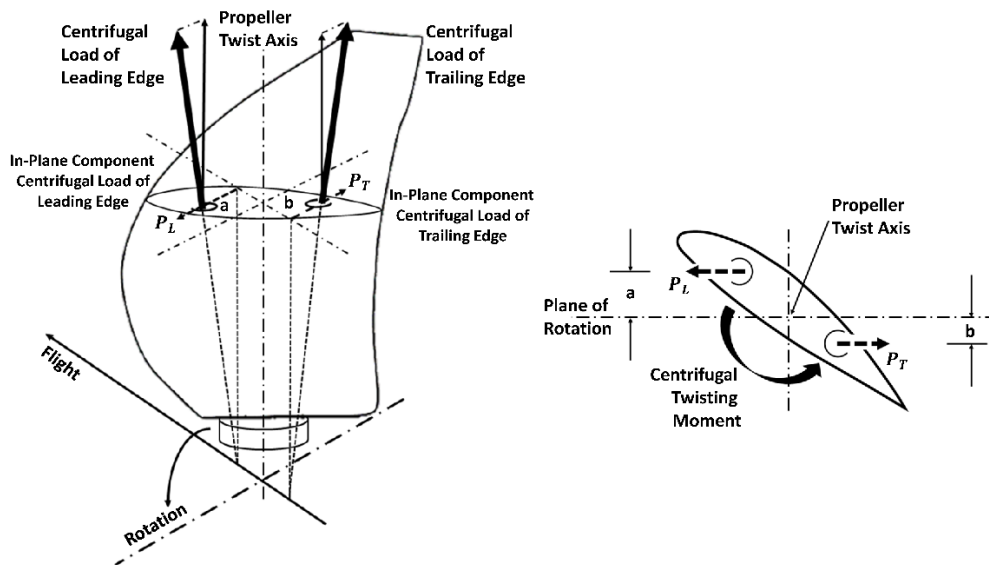


**Figure 27 Development of the centrifugal twisting moment in a propeller blade**

The integration of the centrifugal twisting moment produced by each of the blade segments from hub to tip leads to a net twisting moment about the stacking axis at the blade root. The resultant centrifugal twisting moment is proportional to the square of the propeller rotational speed. A positive CTM tends to align the blade parallel to the plane of rotation, acting towards a flat pitch.

The scaling of the centrifugal twisting moment of the reference propeller blade to the CROR in this research depends on scaling factors for the centrifugal load and the moment arm. The scaling of the moment arm must be proportional to the blade chord. The chord evolution is a function of the section radius. If each blade section is proportional to the previous one, the ratio of the distance at which the centrifugal loads are generated to the blade chord is preserved. Therefore, Eq. (4-1) shows the relationship between the reference and scaled CTMs.

$$CTM = \frac{N_{bld}}{N_{bld,ref}} \frac{W}{W_{ref}} \left( \frac{D}{D_{ref}} \right)^2 \left( \frac{n}{n_{ref}} \right)^2 CTM_{ref} \quad (4-1)$$

The number of blades ( $N_{bld}$ ), blade weight ( $W$ ), blade tip diameter ( $D$ ), and the rotational speed of the propeller ( $n$ ) are used to scale the CTM of the reference SR-7 blade to the propeller design in this research.

#### 4.1.2 Aerodynamic Twisting Moment (ATM)

The aerodynamic loads on the blade develop due to the movement of the propeller through the air. Therefore, the calculation of the blade aerodynamic loads depends on the flight condition. The aerodynamic forces of the SR-7 blade were calculated using lifting line theory [65]. Each portion of the blade produces a lift and a drag force as the propeller rotates through the air and the total aerodynamic loads are the result of integrating the outcomes of each blade section. The results are primarily dependent on the velocity vector, the angle of attack, the chord and the lift coefficient of each blade element.

The centre of pressure, where the aerodynamic forces act, is at a distance of a quarter chord from the leading edge of the blade. Consequently, these loads can be resolved at the blade shank (Figure 28), and at the blade retention to

- a force and a moment acting in the direction of flight, i.e. the thrust and the pitching moment ( $M_p$ ),
- a force and moment acting against the direction of rotation. i.e. the torque and the yawing moment ( $M_y$ ), and
- a moment about the pitch change axis, i.e. the aerodynamic twisting moment (ATM) [65].

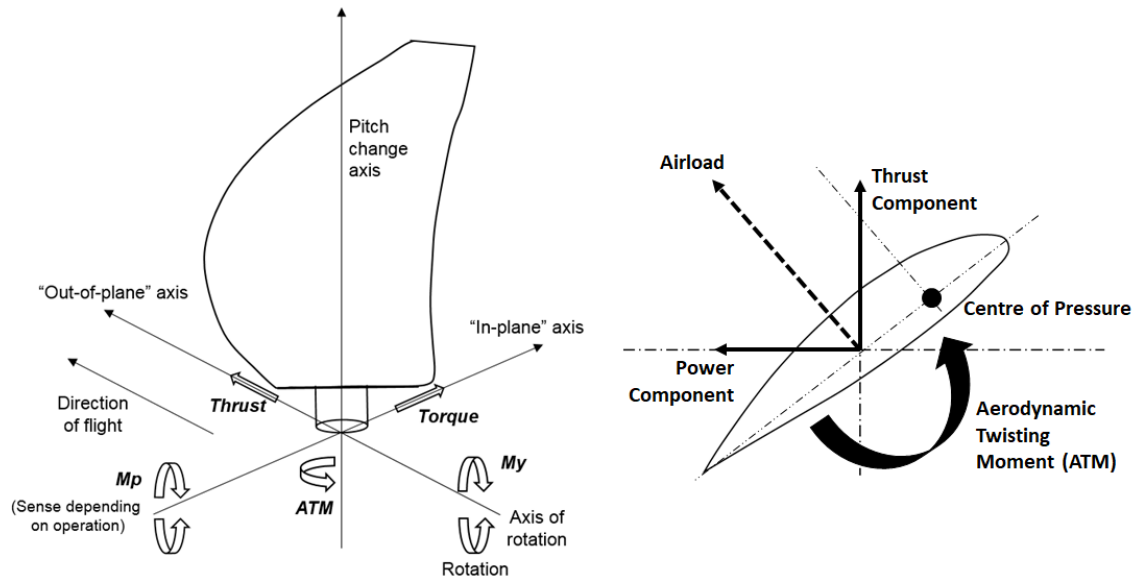


**Figure 28 Air loads at the blade shank**

The ATM is the only force acting on the PCM assembly, and hence the only one further explored in this research. The ATM is the moment produced by the summation of unbalanced aerodynamic forces acting on the pitch change axis of the blade [95]. Consequently, a moment is produced about the pitch axis, which has to be reacted by the PCM to maintain the blade angle setting [90]. The ATM is a function of the power absorbed by the propellers, pitch angle, rotational speed, air density and effective airspeed [65]. The ATM acts either toward low or high pitch depending on the flight condition [47].

The ATM for the SR-7 blade is less than 15% of the CTM, either adding or subtracting from it [95]. The CROR propeller blades in the current research are of the SR-7 type. For this reason, a conservative approach is adopted, taking the ATM to be equal to 15% of the CTM, as shown in Eq. (4-2). However, the ATM for other blade designs would require calculation through lifting-line theory.

$$ATM = \pm 0.15 CTM \quad (4-2)$$

#### **4.1.3 Frictional Twisting Moment (FTM)**

The frictional twisting moment is the resultant of the blade retention friction and opposes the change in blade angle [89]. The FTM always adds to the load that has to be reacted by the PCM actuator and is the same for all blade angles.

However, the contribution of FTM to TTM is relatively small, i.e. around 4% of TTM [95]. FTM is a function of blade centrifugal load ( $CL$ ), moment arm ( $MA$ ) and coefficient of friction ( $\mu$ , taken as 0.005 based on previous experience) [95]:

$$FTM = CL \frac{MA}{2} \mu \quad (4-3)$$

The moment arm for FTM calculations represents the maximum distance between the blade pitch axis and the point of actuation of the activation force in the PCM system. Therefore,  $MA$  would be defined by

- the radius of the radial shaft connecting the PCM to the propellers, and
- the distance from the shaft to the point of activation force in the PCM.

In the preliminary design phase,  $MA$  is unknown. Initially,  $MA$  will be considered equal to the value proposed for LAP's SR-7 blade, i.e. 7.62 cm (3 in) [95]. After obtaining the first estimation of PCM radial shaft dimensions (section 4.3),  $MA$  scaling could be recalibrated. However, the dominant factor in FTM scaling is the centrifugal load and  $MA$  has a relatively minor effect.

Finally, FTM scales similarly to CTM:

$$FTM = \frac{N_{bld}}{N_{bld,ref}} \frac{W}{W_{ref}} \frac{D}{D_{ref}} \left( \frac{n}{n_{ref}} \right)^2 \frac{MA}{MA_{ref}} FTM_{ref} \quad (4-4)$$

#### 4.1.4 Total Twisting Moment (TTM)

The total twisting moment (TTM) is the total moment reacted by the PCM resulting from blade centrifugal and aerodynamic twisting moments and the friction twisting moment produced by the pitch-change action. The uncertainties related to the aircraft and the blade configuration are addressed by applying a 20% margin to all calculated blade loads, i.e. application of a contingency factor (CF) of 1.2 [95]. The TTM can therefore be calculated using Eq. (4-5).

$$TTM = CF \ CTM + CF \ ATM + FTM \quad (4-5)$$

## 4.2 Stress Analysis: Definition of the Loading Case

The PCM is subjected to the effects of cyclic loading caused by the operation of the propeller blades. Fatigue is the primary failure mode defining the design of the PCM [95]. The components in the PCM have to withstand two main loading cases: high-cycle fatigue (HCF) and low-cycle fatigue (LCF). Therefore, the envelope of the PCM is estimated for survival under these two conditions.

**Effects of High Cycle Fatigue:** HCF is associated with the application of cyclic stress over a relatively long time. The HCF acting on the PCM is the result of the combined effect of the steady load imposed by the TTM and the cyclic load of the ATM. Subsequently, the PCM envelope can be estimated by imposing a minimum number of cycles that the component would need to survive without failing. The number of cycles assumed for the safe operation of the PCM under HCF during LAP [95] was  $10^8$  cycles, which is also adopted in this research. The parameter  $MA$  in Eqs. (4-6) and (4-7) is the moment arm for actuation of the reaction force.

$$HCF = \frac{TTM}{MA} \pm \frac{ATM}{MA} > 10^8 \text{ cycles} \quad (4-6)$$

**Effects of Low Cycle Fatigue:** The number of starts and stops is the second loading case that needs to be considered in the sizing of the PCM actuator. For the calculation of the LCF, half of the TTM is applied as a steady load and half as an alternating load. The number of cycles assumed for the safe operation of the PCM under LCF is  $10^4$  cycles [95].

$$LCF = \frac{TTM/2}{MA} \pm \frac{TTM/2}{MA} > 10^4 \text{ cycles} \quad (4-7)$$

## 4.3 PCM Sizing: Design Envelope of the Transmission Device

The transmission device in the PCM provides the link between the blade and the pitch change actuator. It is sized to withstand the blade/activator loads imposed on it. The design of the transmission device can be reduced to the design of the load transfer shaft, and hence follow the guidelines of shaft design [96]. The calculation of the shaft cross-section provides a preliminary estimate of the

blockage in the CROR gas path due to the presence in the exhaust duct of the PCM transmission device connected to the contra-rotating propellers.

The loading state defined by the HCF and LCF (section 4.2) is used for sizing the radial shafts in the transmission device. The PCM is assumed to operate in a similar environment to the DPGB, thus similar materials will be used: a generic carburized and hardened Grade 3 (premium-quality) steel. The shafts in the transmission device are solid shafts designed for infinite life. The most common failure criteria for shafts is the modified Goodman criteria (refer to any mechanics book, e.g. Ref. [97]). Therefore, to a first-order approximation, the outer shaft diameter ( $D_{shaft}$ ) can be obtained from Eq. (4-8), where TTM is the steady torsional loading, the ultimate tensile strength ( $S_{UTS}$ ) is a material property and a safety factor ( $SF_{shaft}$ ) of 1.5 is commonly used in aerospace applications when no pre-existing experience exists [96].

$$D_{shaft} = \left[ \frac{16 SF_{shaft}}{\pi} \sqrt{3} \left( \frac{TTM}{S_{UTS}} \right) \right]^{\frac{1}{3}} \quad (4-8)$$

## 5 Modelling of the Contra-Rotating Open Rotor

This research focusses on the integration of the power-transmission components in the CROR and the assessment of their effects on engine performance and gas path design. In the performance-modelling phase of research, an engine model was required to simulate the performance of the CROR. This chapter details the overall specification of the conceptual engine model. The simulation tool used in this research for modelling the CROR is PROOSIS [98].

The engine configuration evaluated is a geared CROR pusher arrangement. Two spools form the gas generator, which refers to the compressors, combustor and turbines providing energy to the propulsor. The propulsor comprises a free power turbine (or low-pressure turbine), driving contra-rotating propellers through a transmission system, which enables the free turbine and the propellers to run at their optimum rotational speeds and efficiencies.

The propellers in the CROR are the primary source of thrust in the propulsion system and determine the power transferred through the transmission system from the LPT. In practice, the design rotational speed of the propellers is limited by compressibility losses and noise emissions from the high-subsonic helical Mach numbers at the blade tips. Therefore, the rotational speed of each propeller is an engine control variable, and hence a design parameter of the engine model. The rotational speed of the propellers will also define the design of the components in the transmission system: the DPGB and the PCM.

The CROR comprises two mechanically independent sections: the gas generator and the propulsor. Section 5.1 describes the baseline definition for the gas generator in the CROR and the selection of engine design parameters. Section 5.2 details the baseline definition of the propulsor and the development of a performance model for the LPT. The modelling of the power gearbox and the pitch change mechanism is detailed in chapters 3 and 4 respectively.

The performance parameter for the gas generator and propulsor shown in Table 4 and Table 5 respectively define the ‘baseline’ configuration for the propulsion system against which alternative CROR configurations are compared in this

research (chapter 6). Design of the baseline engine architecture was based on previous research on geared CRORs, which was used to verify the engine model, namely Ref. [99]. A multi-point design of the engine architecture was simulated at top-of-climb (design point for turbomachinery) and, take-off/ EOR (design point for the DPGB and PCM). The control parameter for these points was the engine prescribed thrust. Matching of engine thrust requirements for a flight condition (i.e. altitude,  $M_0$  and ISA deviation) defines the thermodynamic cycle.

Section 5.3 details the generation of the preliminary general arrangement for the CROR; Section 5.4 details the weight estimation of the components in the CROR and; Section 5.5 details the modelling methodology followed to simulate the integration of the CROR with an aircraft and to calculate its mission fuel burn for a specific mission.

## **5.1 Baseline definition of the gas generator in the CROR**

Design of the engine performance at a particular operating point requires the selection of various parameters such as net thrust, combustor outlet temperature, component technology level (e.g. efficiencies and pressure ratios) and fuel flow rate (or turbine entry temperature, TET). The design point of the turbomachinery components in the CROR is Top-of-Climb (ToC), i.e. altitude of 10,668 m,  $M_0$  0.75 and ISA deviation +10 K. Table 4 presents the design parameters of the gas generator at design point for a prescribed ToC thrust of 24.02 kN.

Increasing TET and overall pressure ratio (OPR) increases thermal efficiency and reduces SFC. Conventionally, the maximum allowable delivery temperature of the high-pressure compressor (HPC) constrains the optimal OPR. A limit for modern engines of around 970 K is proposed in Ref. [100]. However, in the case of the CROR, the small size of the core may replace the HPC temperature constraint with a practical minimum blade height requirement to avoid significant efficiency loss from tip clearance effects [33]. An HPC last stage blade height limit of 10 mm is proposed in Ref. [100] and used as a design constraint in this research. The selection of a single stage for the high- and intermediate-pressure turbines (HPT and IPT respectively) sets the maximum achievable pressure ratio in the HPC and the intermediate-pressure compressor (IPC) respectively.

**Table 4 Design parameters of the CROR gas generator at Top-of-Climb (10,668 m,  $M_0$  0.75, ISA+10 K, FN = 24.02 kN) ([7])**

| Component | Parameter [units]              | Value            |
|-----------|--------------------------------|------------------|
| IPC       | Pressure ratio [-]             | 7.0              |
|           | Number of stages [-]           | 6                |
|           | Isentropic efficiency [-]      | 0.87             |
|           | IPC-HPC duct pressure loss [%] | 0.75             |
| HPC       | Pressure ratio [-]             | 6.5 (OPR = 45.2) |
|           | Number of stages [-]           | 6                |
|           | Isentropic efficiency [-]      | 0.87             |
| Burner    | Exit temperature (TET) [K]     | 1725             |
|           | Pressure loss [%]              | 4                |
| HPT       | Number of stages [-]           | 1                |
|           | Isentropic efficiency [-]      | 0.9              |
| IPT       | Number of stages [-]           | 1                |
|           | Isentropic efficiency [-]      | 0.9              |
|           | IPT-LPT duct pressure loss [%] | 0.75             |

The materials and design technology constrain the maximum value of TET. The prescribed isentropic efficiencies of the turbomachinery in the gas generator reflect the technology level of the components. The burner has an estimated efficiency of 99% with an annular design. Typical design values are used for duct and burner pressure losses [101].

## 5.2 Baseline definition of the propulsor in the CROR

The propulsor in the CROR comprises the low-pressure turbine, the transmission system, contra-rotating propellers and the core exit nozzle. Details of the components in the transmission system have been provided in chapter 3 for the DPGB and chapter 4 for the PCM. Table 5 summarises the choice of design parameters for the propulsor.

The gearbox assessments presented in section 3.3.4 demonstrated that 1.29 is the minimum torque ratio that would result in the most compact gearbox configuration (i.e. five-planet DPGB) for a CROR, as shown in Figure 23. The smaller the TR, the bigger the speed-reduction between the input and output shafts, which is beneficial for LPT design as it allows the turbine to rotate faster

and closer to its optimum operating point. The DPGB enables high mechanical efficiency [50].

**Table 5 Design parameters of the CROR propulsor at Top-of-Climb (10,668 m,  $M_0$  0.75, ISA+10 K, FN = 24.02 kN)**

| Component                                                        | Parameter [units]                                              | Value      |
|------------------------------------------------------------------|----------------------------------------------------------------|------------|
| LPT                                                              | Number of stages [-]                                           | 4          |
|                                                                  | Isentropic efficiency [-]                                      | 0.94       |
| DPGB                                                             | Mechanical efficiency at EOR* [-]                              | 0.99       |
|                                                                  | Torque ratio ( $TR = Q_{\text{carrier}}/Q_{\text{ring}}$ ) [-] | 1.29       |
| CRP                                                              | Rotational speed of the propellers [RPM]                       | $\pm 1100$ |
|                                                                  | Propeller efficiency [-]                                       | 0.89       |
|                                                                  | Net efficiency [-]                                             | 0.86       |
|                                                                  | Blade geometry [-]                                             | SR-7       |
|                                                                  | Hub diameter [m]                                               | 1.6        |
|                                                                  | Front propeller diameter [m] & blade number [-]                | 4.26, 12   |
|                                                                  | Rear propeller diameter [m] & blade number [-]                 | 3.83, 9    |
|                                                                  | Propeller spacing [m]                                          | 1.06       |
| Nozzle                                                           | Pressure ratio [-]                                             | 1.3        |
|                                                                  | Pressure loss [%]                                              | 2          |
| *EOR condition: sea level, $M_0 = 0.25$ , ISA+15 K, FN = 97.9 kN |                                                                |            |

The blade geometry selected for CROR simulations in this research is the SR-7 propeller defined in Ref. [102]. The combination of blade diameter and maximum tip helical Mach number define the rotational speed of the propellers at the design point. The efficiency of each propeller is assumed constant for any design point assessment in this research. The target value of 0.86 at top-of-climb reflects the expected net efficiency of a modern contra-rotating propeller [103]. The net efficiency of the CRP characterises the aerodynamic performance of the CROR and captures

- the effects of wake and shock losses on propeller (adiabatic) efficiency through the compressibility correction, and
- the effects of the induced losses from exit swirl and kinetic energy loss associated with tip vortices and wakes through the induced velocities.

The additional baseline design parameters for the CRP are based on the optimisation results of previous research by Bellocq on a similar CROR with similar operating requirements [7], [99]:

- A diameter of 4.26 m (14 ft) for the front propeller is representative of modern CROR configurations [99]. Aircraft integration limits the diameter of the propellers, where the length of the pylon determines their maximum diameter.
- The rear propeller is clipped to avoid impingement of the front-blade tip vortex at design conditions, i.e. 10% clipping is common for the rear propeller [99].
- The number of blades in the front and rear propellers was selected to achieve approximately equal power loading per blade in both propellers at the design point [99].
- The nacelle hub diameter is defined by aerodynamic constraints as 1.6 m [99]. The hub diameter of the propellers is limited by the exit tip diameter of the LPT, which is designed to accommodate the integration of the power gearbox. Propeller hub-to-tip ratios between 0.35 and 0.4 are typical for modern CROR designs [104].
- The spacing between propellers is 0.25 times the tip diameter of the front propeller [92].
- A nozzle baseline pressure ratio of 1.3 maximises LPT power extraction. The presence of contra-rotating struts in the exhaust duct would result in relatively high pressure losses.

### **5.2.1 Modelling of the Low-Pressure Turbine in the CROR**

The low-pressure turbine in the CROR is a free turbine and the driver of the contra-rotating propellers. The calculation of the LPT design point isentropic efficiency is intended to capture the effects of key engine preliminary design parameters on CROR performance. The methodology proposed allows an insight into how changes in the propulsor design parameters affect the power generation of the turbine, and hence the power supply to the propellers. This would provide a preliminary estimate of the space available for the power gearbox.

Equation (5-1) provides a simple definition of the power generated by the LPT.

$$PW_{LPT} = \dot{m} C_p (Tt_{in} - Tt_{out}) \quad (5-1)$$

Therefore, the power required from the LPT to achieve a prescribed thrust for the CROR is a function of:

1. Gas generator thermodynamic performance: Defines the inlet conditions of the LPT, namely gas conditions (e.g. isobaric specific heat,  $C_p$ ), mass flow rate ( $\dot{m}$ ), total pressure and temperature ( $Tt$ ).
2. LPT efficiency: The calculation of the LPT isentropic efficiency is an iterative process (explained later in detail).
3. Nozzle pressure ratio (NPR): Defines the pressure drop across the LPT in the preliminary design of the CROR. In this research, NPR is a preliminary design parameter for the CROR. However, in practice, the change in NPR would be the result of changing the LPT design, e.g. annulus geometry, detailed gas path design or rotational speed.
4. Target value of contra-rotating propeller net efficiency.
5. Rotational speed of the front and rear propellers ( $n_{FP}$  and  $n_{RP}$  respectively) and torque ratio ( $TR$ ): These are inputs to the performance model. The combination of these parameters determines the rotational speed of the low-pressure shaft ( $n_{LPT}$ ), as shown in Eqs. (5-2) and (5-3).

$$n_{LPT} = n_{FP} 2 (1 + m_G) + |n_{RP}|(1 + 2m_G) \quad (5-2)$$

$$n_{LPT} = n_{FP} \frac{TR}{TR - 1} + |n_{RP}| \frac{1}{TR - 1} = \frac{1}{TR - 1} (n_{FP} TR + |n_{RP}|) \quad (5-3)$$

The efficiency of the LPT was calculated based on the following preliminary design constraints and assumptions:

*Preliminary design constraints for the LPT*

- The hade angle is limited to 40° and 20° for the tip and hub lines respectively to prevent excessive losses (estimated values from discussion with experts).

- The LPT blade centrifugal stresses are proportional to the maximum rotational speed and the annulus area [34]. Therefore, the maximum stress in the LPT is assessed with the  $AN^2$  parameter. The  $AN^2$  structural parameter is not a hard limit on the design but a good indicator at the preliminary design phase of the stresses at the blade hub of the LPT exit. A lower bound is set around  $32 \cdot 10^6 \text{ RPM}^2\text{m}^2$  to be representative of current technology levels [33]. An upper bound of  $50 \cdot 10^6 \text{ RPM}^2\text{m}^2$  is selected to reflect improvements in turbine design and materials [34].
- The external diameter of the LPT is constrained by the CRP module.

*Preliminary design assumptions for the LPT*

- The inlet hub-to-tip ratio and axial Mach number are inputs to the design.
- Linear mean diameter distribution from inlet to exit.
- Every stage is designed for constant mean diameter and inlet axial velocity.
- A constant loading coefficient ( $\psi$ ) defines the work (power extraction) of each LPT stage.
- Calculations based on constant  $C_p$  and  $\gamma$  (implications in Ref. [101]).
- The number of LPT stages is an input to the CROR preliminary design model. It results from an iterative calculation (described below) to ensure high efficiency for the baseline engine design with a minimum number of LPT stages (target efficiency value of 0.94 from Ref. [105]). The baseline number of stages is preserved for further design point assessments and is not a degree of freedom to the model. This simplified approach enables the assessment of design-point performance variation relative to the baseline with focus on the integration effects of the transmission system on the engine.

Based on the LPT design assumptions presented above, a lower number of stages would increase stage loading. The associated approach of the LPT annulus to the engine's centreline and reduction in LPT exit area would exacerbate the integration problem of the transmission system. The

magnitude assessment of these effects and the potential performance trade-offs is suggested as future work.

- A 12.5% reduction in the LPT hub line from inlet to outlet has been assumed. This percentage reduction results from a sensitivity analysis on annulus line evolution for all design points evaluated in chapter 6. The outcome of the analysis showed that a minimum 12.5% reduction in the LPT hub line was required to comply with hade angle constraints, especially at the tip-line. High hade angles at the tip line could be associated with losses in performance.

The LPT design in the CROR has to accommodate the DPGB integration downstream. A constant hub design would set the hub line of the turbine at its furthest position from the engine centreline, freeing the most space for the integration of the power gearbox. However, the resultant turbine design would be too aggressive and would not comply with hade angle constraints. Therefore, the LPT annulus design in this research is constrained by the prescribed hade angles at the hub and tip lines.

The first step in the preliminary design of the LPT, required to estimate the isentropic efficiency of the component, is to define its flow-path shape. The flow annulus area ( $A$ ) is calculated at the inlet of the LPT from the known cycle parameters (i.e. mass flow rate, total temperature, and total pressure) for a prescribed axial Mach number ( $M_n$ ). Assuming constant axial velocity, the flow annulus area is also calculated at the LPT outlet. The inlet hub-to-tip ratio ( $HTR$ ) enables the calculation of the LPT hub, mean and tip diameters ( $D_h$ ,  $D_m$ ,  $D_{tip}$  respectively) from appropriate rearrangement of Eqs. (5-4) to (5-6) knowing two out of the five unknowns involved.

$$A = \frac{\pi}{4} (D_{tip}^2 - D_h^2) \quad (5-4)$$

$$HTR = \frac{D_h}{D_{tip}} \quad (5-5)$$

$$D_m = \frac{D_h + D_{tip}}{2} \quad (5-6)$$

The selection of the HTR and Mn at the inlet of the LPT affect its annular geometry, and hence its isentropic efficiency. A baseline value of 0.44 is selected for the LPT inlet Mn ( $Mn_{in,LPT}$ ) and 0.9 for the inlet HTR ( $HTR_{in,LPT}$ ) to achieve a high baseline isentropic efficiency, as proposed by Hendricks [105] for a similar CROR architecture. A sensitivity analysis was performed on the assumed values to determine their impact on engine performance. Figure 29 shows that a 1% change in  $HTR_{in,LPT}$  translates into a 0.5% change in LPT isentropic efficiency ( $\eta_{is,LPT}$ ). However, a 2.5% change in  $Mn_{in,LPT}$  would only result in a 0.25% change in  $\eta_{is,LPT}$ . Therefore, the analysis shows the dominant effect of  $HTR_{in,LPT}$  relative to  $Mn_{in,LPT}$  on  $\eta_{is,LPT}$ . The refinement of these parameters is a source of potential improvement in  $\eta_{is,LPT}$  and should be addressed in further research.

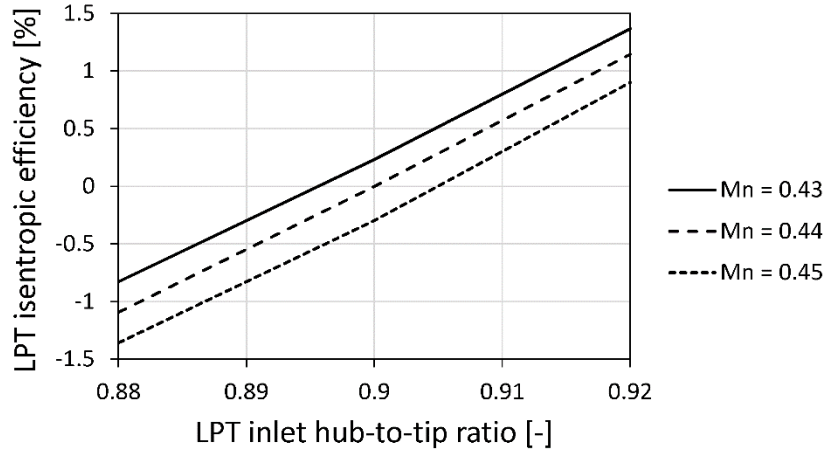


**Figure 29 Variation of the LPT isentropic efficiency for the baseline CROR configuration with LPT inlet Mach number and hub-to-tip ratio**

Assuming a mean diameter distribution in the LPT determines the exit radii and the flow annulus shape. The blade speed can be calculated from the rotational speed of the shaft knowing the mean diameter at every stage. At the same time the rotational speed of the LPT shaft is defined by imposed aerodynamic (e.g., propeller helical tip Mach number) and structural criteria (e.g., turbine blade stress parameter  $AN^2$ ). The blade speed ( $U$ ) is calculated at the blade mean radius ( $R_m$ ) from Eq. (5-7) and is constant throughout the respective stage ( $i$ ).

$$U_i = 2\pi n_{LPT} R_{m,i} \quad (5-7)$$

To calculate the LPT isentropic efficiency the values of flow coefficient ( $\Phi_i$ ), stage loading ( $\Psi_i$ ) and stage reaction have to be calculated or imposed. The LPT total enthalpy drop ( $\Delta h_{LPT}$ ) is iteratively calculated from the cycle parameters at the outlet of the gas generator and the prescribed nozzle pressure ratio. Equation (5-8) defines  $\Delta h_{LPT}$  as the ratio between the isentropic power generated and LPT inlet mass flow rate.

$$\Delta h_{LPT} = \frac{PW_{LPT,is}}{\dot{m}_{LPT,in}} \quad (5-8)$$

A constant stage-loading coefficient ( $\Psi_i$ ) is calculated solving a system of  $N_{stg}+1$  equations defined by the expressions in Eq. (5-9) and (5-10), where  $N_{stg}$  represents the number of stages in the LPT.

$$\Psi_i = \frac{\Delta h_i}{U_i^2} = const \quad (5-9)$$

$$\Delta h_{LPT} = \sum_1^{N_{stg}} \Delta h_i \quad (5-10)$$

The stage enthalpy drop ( $\Delta h_i$ ) and stage blade speed ( $U_i$ ) differ for each LPT stage. However, the stage loading coefficient will be constant for all of them.

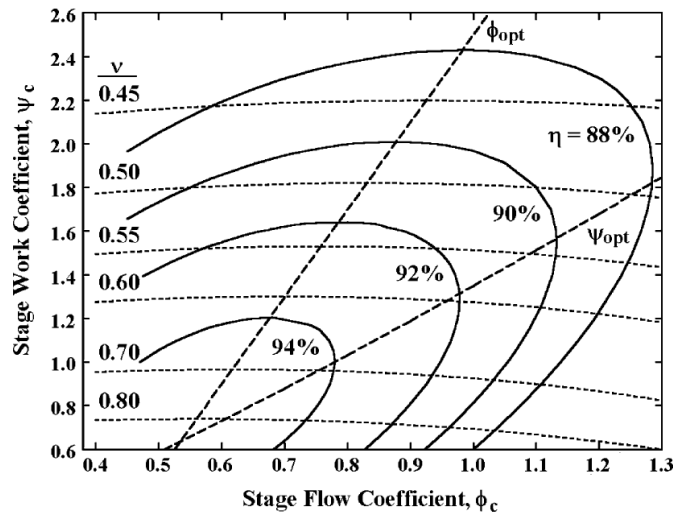


**Figure 30 Smith chart [106]**

The isentropic efficiency of each stage ( $\eta_{is,i}$ ) in the LPT is obtained from a Smith chart, which assumes a reaction ratio of 0.5 (Figure 30). The engine performance

model calculates the mean axial flow velocity ( $V_{ax}$ ), required to read the Smith chart from the LPT inlet conditions and cross-sectional area. A technology correction is applied to the calculated efficiency through a technology factor (TF) of 1.02, representative of EIS 2035 requirements [99].

Once the LPT-stage isentropic efficiency is calculated, the outlet  $Tt$  is calculated

$$Tt_{out,i} = Tt_{in,i} - \frac{\Psi_i U_i^2}{C_p} \quad (5-11)$$

The total pressure at the exit of the stage is obtained with Eq. (5-12) from the calculated LPT-stage isentropic efficiency and the stage total temperatures.

$$Pt_{out,i} = Pt_{in,i} \left( 1 - \frac{\left( 1 - \frac{Tt_{out,i}}{Tt_{in,i}} \right)^{\frac{\gamma}{\gamma-1}}}{\eta_{is,i}} \right) \quad (5-12)$$

For the stage calculation,

$$Tt_{in,i+1} = Tt_{out,i} \quad (5-13)$$

$$Pt_{in,i+1} = Pt_{out,i} \quad (5-14)$$

Finally, Eq. (5-15) defines the isentropic efficiency of the LPT.

$$\eta_{is,LPT} = \frac{1 - \frac{Tt_{LPTout}}{Tt_{LPTin}}}{1 - \left( \frac{Pt_{LPTout}}{Pt_{LPTin}} \right)^{\frac{\gamma}{\gamma-1}}} \quad (5-15)$$

The gas properties adopted for efficiency calculations are 1135 J/kg K for  $C_p$  and 1.338 for  $\gamma$ .

The calculation process of the isentropic efficiency was verified matching the calculated total exit pressure of the LPT to the value obtained with the prescribed NPR at the design point operating condition.

### 5.2.2 Modelling of the Contra-Rotating Propellers

The contra-rotating propeller model used in this research is intended to provide a preliminary estimate of the performance of the propellers. The model allows an insight into how changes to the propulsor design will influence the overall performance of the CROR. The performance of the CRP is estimated using the model developed by Bellocq, presented in Refs. [6], [7] and [99]. The model calculates the thrust produced by the propellers, the induced flow field in which they operate and the effect of compressibility losses. The performance of the CRP is simulated using a 0-D performance model of the CRP suitable for trade-off assessments and preliminary design studies (TRL 1-2). The model enables the independent design and definition of the operating condition of each propeller to simulate their combined operation and interactions between them.

The operating condition of a propeller is defined in terms of shaft power, propeller rotational speed, inflow velocity, ambient conditions (pressure altitude and ambient temperature) and inflow angle. Propeller characteristics define the performance of each row of propellers. For an isolated propeller, propeller maps often present the performance in a non-dimensional dimensional form to capture the relationship between

- the independent operating parameters, i.e. the free stream Mach number, advance ratio ( $J$ ) and blade pitch angle, and
- the output power coefficient ( $C_P$ ), the thrust coefficient ( $C_T$ ) and the propeller efficiency ( $\eta_{propeller}$ ).

The advance ratio is the ratio of flight speed ( $V_0$ ) to blade tip speed ( $U_{tip}$ ). Alternatively, it is commonly expressed as the ratio between flight speed and the product of propeller rotational speed ( $n$ ) in rotations per second and the diameter of the propeller ( $D$ ).

$$J = \frac{\pi V_0}{U_{tip}} = \frac{V_0}{n D} \quad (5-16)$$

The power coefficient is a function of the power going to the propeller shaft ( $PW$ ), air density ( $\rho$ ), rotational speed of the propeller and propeller diameter. It can also

be expressed as a function of the advance ratio and disk power loading ( $L$ ), which represents the amount of kinetic energy transmitted to the air as it passes through the propeller disk and is analogous to FPR in turbobans [36].

$$L = \frac{PW}{D^2} \quad (5-17)$$

$$C_P = \frac{PW}{\rho n^3 D^5} = L \frac{J^3}{V_0^3 \rho} \quad (5-18)$$

$$C_T = \frac{FN}{\rho n^2 D^4} \quad (5-19)$$

The expression for advance ratio, power coefficient and thrust coefficient ( $C_T$ ) presented in Eqs. (5-16) - (5-19) represent the operation of the propeller under free airstream conditions. These parameters are high-level characteristics of the performance of the propeller and contain implicit pitch angle information.

The efficiency of the individual propellers is often presented in the propeller performance maps as a function of the advance ratio ( $J$ ) and the coefficient of power ( $C_P$ ). The propeller efficiency in Eq. (5-20) captures how much of the shaft power is turned into useful thrust. Therefore, this means that the swirl losses, profile/turbomachinery losses and propulsive efficiency are included in the number [107].

$$\eta_{propeller} = \frac{V_0 FN}{PW} = J \frac{C_T}{C_P} \quad (5-20)$$

Therefore, the thrust produced by a propeller can be expressed as

$$FN = \frac{PW \eta_{propeller}}{V_0} \quad (5-21)$$

The map (the characteristics) of a propeller as the one in Figure 31 is representative of the operation of a single-rotating propeller with a given design. Therefore, the performance characteristics would not be valid for a propeller with a different design, i.e. different hub or nacelle shapes, hub-to-tip ratios or flight Mach numbers [99]. However, these limitations can be overcome, as presented in the following paragraphs.

The shape of the propeller hub or engine nacelle in a CROR pusher configuration has an effect on the flow field across the propeller. The assessment of nacelle shape effects on propeller performance would require detailed design and is outside of the scope of the current research.

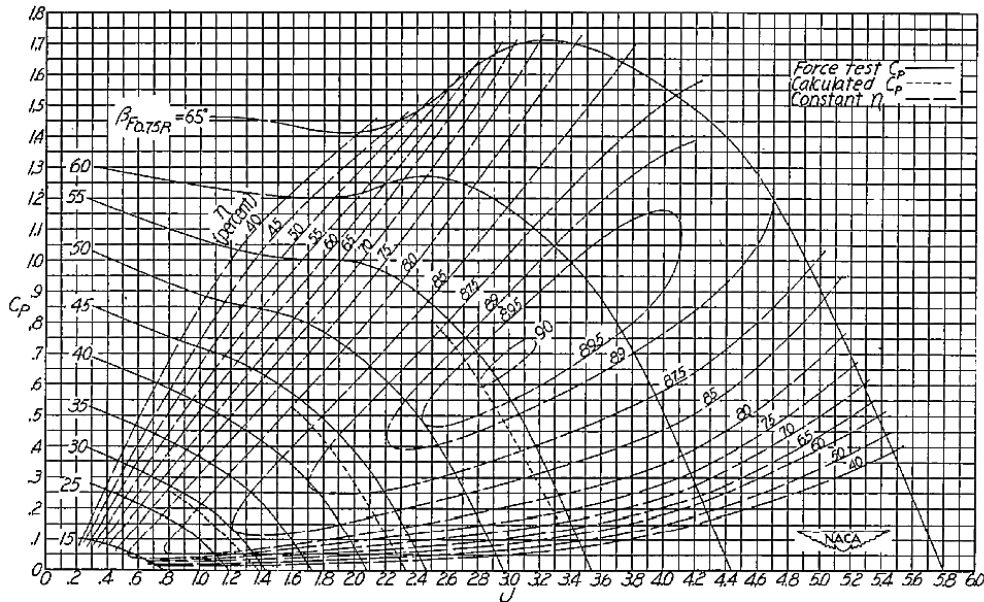


**Figure 31 Propeller performance map [108]**

If the hub-to-tip ratio of the propeller differs from the design used to generate the performance characteristics, the blade would present a different distribution of forces (i.e. a different incidence angle to each blade section). The power required by the propeller and the thrust produced would differ operating at the same advance ratio and propeller pitch.

To a first-order approximation, accounting for the effects of varying HTR can be achieved through modification of the terms used to define propeller performance. If the HTR of the propeller changes from the design point, the blade area would no longer be proportional to the propeller diameter. Therefore, the blade area can be considered proportional to the disc area defined in Eq. (5-22). Consequently, the modified non-dimensional power and thrust coefficients (i.e. PQA and TQA respectively) would define the propeller performance at a given flight condition using Eqs. (5-23) to (5-25).

$$A_{disk} = \frac{\pi}{4} D^2 \left( 1 - \left( \frac{D_h}{D} \right)^2 \right) \quad (5-22)$$

$$FN = \rho n^2 D^2 A_{disk} TQA \quad (5-23)$$

$$PW_{mech} = \rho n^3 D^3 A_{disk} PQA \quad (5-24)$$

$$\eta_{propeller} = \frac{V_0 FN}{PW} = J \frac{TQA}{PQA} \quad (5-25)$$

The net efficiency of the CRP is defined in Eq. (5-26), where  $V$  represents the axial velocity of each propeller operating in the flow field of the other propeller.

$$\eta_{net,CRP} = \frac{V FN}{PW} \Big|_{FP} + \frac{V FN}{PW} \Big|_{RP} \quad (5-26)$$

### 5.2.2.1 Compressibility effects

The maps used for the contra-rotating propellers are a function of the freestream Mach number. The rigorous calculation of the performance of the contra-rotating propellers requires the use of multiple performance characteristics for the various Mach numbers. The use of such a library of maps would be limited to the existence of a wind tunnel database for a given propeller design. Alternatively, propeller maps could be generated for a given geometry following the modelling method proposed in Ref. [109].

The compressibility effects in this research are calculated following the simplified method proposed by Bellocq [6] based on the conclusions of the work by Mitchell [110] for advanced propellers:

- The propeller characteristics are represented with a propeller map produced at low speeds, where the helical Mach number at a three-quarter span ( $M_{h0.75}$ ) is lower than the critical helical Mach number at the same blade span ( $M^*_{h0.75}$ ).
- The efficiency of each propeller ( $\eta_{propeller}$ ) is obtained from the propeller map from the corresponding values of power coefficient ( $C_p$ ) and advance ratio ( $J$ ).

- The efficiency value obtained from the low-speed propeller map is corrected with the following functions:

$$M_{h0.75} < M_{h0.75}^*; \quad \eta_{propeller} = \eta_{propeller \text{ at low speed}} \quad (5-27)$$

$$M_{h0.75} > M_{h0.75}^*; \quad \eta_{propeller} = \eta_{propeller \text{ at low speed}} - (M_{h0.75} - M_{h0.75}^*) m_{M_{h0.75}^*}$$

The critical helical Mach number and the rate of decrease in propeller efficiency beyond that value ( $m_{M_{h0.75}^*}$ ) are inputs to the model. The helical Mach number at a three-quarter blade-span ( $M_{h0.75}$ ) is calculated with Eq. (5-28).

$$M_{h0.75} = M_0 \sqrt{1 + \left( \frac{0.75\pi}{J} \right)^2} \quad (5-28)$$

The design helical tip Mach number for operation of the propeller blades in this research (SR-7 blades) is between 1.0 and 1.2 [110]. The blade designs for the SR-7 and modern propeller blades present high-swept at the tip, delaying compressibility effects. Therefore, a critical Mach number of 1.1 has been assumed for the modelling of the propeller blades. The slope for efficiency decrease after the critical Mach number is also assumed from [110].

#### 5.2.2.2 Propeller interaction effects

The performance estimation of a contra-rotating propeller requires the incorporation of propeller interactions to the operating point of the isolated performance characteristics. The present study predicts the aerodynamic performance of the propellers following the method proposed by Bellocq for a given blade geometry. The extensive set of equations defining the interaction effects are available in Ref. [99] and are not transcribed in this thesis. The preliminary design method is based on Blade Element Momentum Theory and represents an alternative to the more computationally expensive lifting line methods used for CROR systems. The proposed methodology is a low-order method, which assumes mutually induced velocities to be proportional to the self-induced velocities of the propellers. The method accounts for the slipstream

contraction effect of the front rotor in the velocity field of the rear propeller and inter-rotor spacing. Performance simulations in Ref. [99] validated the prediction accuracy of the overall power and efficiency quantities for partial fields of propellers using SR-7 blades with differences below 1% away from stall regions.

The calculation of the mutually induced velocities in Bellocq's work for the CRP is based on a low loading assumption [99]. However, modern blade designs would experience higher loading, and hence the speeds involved in old blade designs would not be of the same order of magnitude. Therefore, the CRP modelling approach proposed in Ref. [99] presents some limitations for its application on modern blade propellers.

Dubosc proposed an alternative method for the calculation of mutually induced velocities [92]. The method was validated with EIS 2030 state-of-the-art blades developed for the SAGE2 CROR within the European Clean Sky research project. Dubosc's work followed a similar CRP design approach to Bellocq's and close similarities exist between their research [92]. However, Dubosc's method presents a higher level of complexity and some of the coefficients used in the method are blade dependent. In terms of propeller interaction effects, common points in both studies are the calculation of mutually induced velocities based on self-induced velocities, and the calculation of self-induced velocities from one-dimensional theories. In turn, the main points of discrepancy between both methods refer to the calculation of compressibility effects and mutually induced velocities.

According to Dubosc, the approach in Ref. [92] for the calculation of induced velocities would result in total power differences between 0.5% and 3% and differences in thrust production between 0.8% and 4% relative to Bellocq's approach in Ref. [99]. Given the comparative nature of the assessments undertaken in the current research for design space exploration, the simplified approach proposed in Ref. [99] is used as it enables trend definitions to identify feasible CROR configurations.

### 5.3 CROR Design Point Sizing: Gas Path Generation

The characteristics of the components in the CROR require multi-point design calculations for estimating the dimensions of the gas path configuration. The thermodynamic cycle of the engine is sized at a design point to then calculate its steady-state performance at off-design conditions. The gas path generation methodology was verified against the CROR layout produced by Hendricks [105].

Propeller design and size estimation in this research is based on previous research on geared CRORs [99]. A discussion of the parameter selection and implications is provided in section 5.2.

The design point for the DPGB and the PCM is EOR take-off condition (sea level,  $M_0 = 0.25$ , ISA+15 K). The torque requirements to size these components are defined by the baseline rotational speed of the propellers and the power required for achieving a prescribed thrust requirement of 97.9 kN per engine at EOR. The details of the DPGB sizing are presented in chapter 3. The details of the PCM sizing are presented in chapter 4.

The basic dimensions for the turbomachinery components are obtained from performance data and pertinent design criteria. The baseline cycle parameters have been defined in section 5.1 for the gas generator and section 5.2 for the LPT. The design point for the turbomachinery components in the CROR is ToC (altitude of 10,668 m,  $M_0$  0.75 and ISA deviation +10 K). Therefore, the core and LPT are sized to meet the prescribed thrust requirements of 24.02 kN per engine at ToC.

Turbomachinery sizing in this research is primarily intended to provide preliminary estimates of engine dimensions to define the space available for the transmission system. Flow-path definition also enables weight estimation for the turbomachinery components (section 5.4.1). Conventionally, flow-path estimates for turbomachinery in preliminary design assessments are produced from the flow-annulus stage-wise dimensions assuming axial gap distributions and blade aspect ratios. However, a simplified approach is adopted in this research, where turbomachinery is sized based on component gross size (i.e. outer or mean

diameter) and major physical or aerothermodynamic variables related to the cycle parameters from correlations provided in Ref. [111]. The lower level of detail of the method in Ref. [111] provided estimates of turbomachinery annuli evolution adequate for the research scope.

Sagerser, Lieblein and Krebs produced simplified equations from correlations of cruise and lift engine data of VTOL aircraft, which are also applicable to “cruise” engines [111]. Regarding compressor sizing, the correlations in Ref. [111] provide a simplified method based on number of stages, inlet and outlet hub to tip ratios. In this research, the calculation was done on a stage-by-stage basis to accommodate changes in HTR through the compressor. Sizing the compressor by stages splits the pressure ratio in the component. This modification to the approach in Ref. [111] enables compressor length estimation for high pressure ratio modern propulsion engines, as verified with the CROR gas path in Ref. [105]. Regarding turbine sizing, estimates of aspect ratio in turbine stators and rotors, axial chord and spacing provided in Ref. [111] give a preliminary indication of current turbine dimensions [112]. Combustor length is defined by the length-to-height ratio of the combustor dome, and characteristics for modern combustors are provided in Ref. [113]. The geometry of a duct is defined by the flow characteristics and dimensions of the components connected to it [114]. The assembly of the resulting component annuli and length estimates generates the engine’s flow-path.

## **5.4 CROR Weight calculation**

The weight estimation methods used in this research provide a preliminary estimate of the engine weight. A modular approach to estimating the weight of the components allows an insight into how changes to the design parameters of the CROR will influence the overall weight of the engine. The component weight estimates have been compared against the estimates by Hendricks [105] and Larsson [33] for similar CROR configurations.

### 5.4.1 Turbomachinery

The weight of the gas generator and the LPT are estimated using correlations based on physical principles proposed by Sagerser, Lieblein and Krebs in Ref. [111] as updated by Jackson in Ref. [115]. The equations estimate the weight and dimensions of each component separately primarily based on the geometrical characteristics of the components. The correlations in Ref. [111] also account for the weight of the nacelle, structure and accessories. The method in Ref. [111] is sound for engine preliminary design estimates but requires a pressure level term as technology factor for the core components to account for high-pressure ratios [115].

The suitability of the method in Ref. [111] for turbofan engines has been assessed by Jackson, who recommended revising the method for propulsion engines with high pressure ratio cores [115]. However, the simplicity of the method in Ref. [111] makes it unique for weight and size estimates for preliminary design assessments and has been used in past research to estimate the weight of ultra-high bypass turbofan [116] and geared CROR engines [117].

### 5.4.2 Propeller Assembly

The weight of the propeller assembly has been estimated in this research using weight relationships generated for the SR-7 propellers based on experimental data [118]. The relationships were produced for maximum power delivered to the propellers and maximum tip speed ( $U_{tip}$ ) at which the rotor normally operated, i.e. power loading ( $PW/D^2$ ) of 70 shp/ft<sup>2</sup> and 800 ft/s for the SR-7. The reference weight value for the SR-7 propeller assembly is a function of its diameter and the number of blades ( $N_{bld}$ ). The weight of the contra-rotating propeller assembly obtained using this method includes the weight of blades, retention, pitch control and anti-icing systems.

From the reference value, the weight of the propeller assembly can be obtained at different operating conditions through the use of tip speed and power loading corrections. Therefore, the weight of each row of propellers in the CROR is calculated from Eq. (5-29).

$$W_{FP \text{ or } RP} = W_{SR-7_{800,70}} \frac{N_{bld,FP \text{ or } RP}}{N_{bld,SR-7}} \left( \frac{U_{tip,FP \text{ or } RP}}{800} \cdot \frac{L_{FP \text{ or } RP}}{70} \right)^{0.3} \quad (5-29)$$

### 5.4.3 Power Gearbox

The weight of the power gearbox has been calculated from the correlation proposed by Hendricks [105] for power gearboxes in CRORs (Figure 32). This simplified approach does not consider the effect of the change in the number of planets with torque ratio. Therefore this will be reflected in the smoothness of the weight and fuel burn results of the parametric analyses in chapter 6.

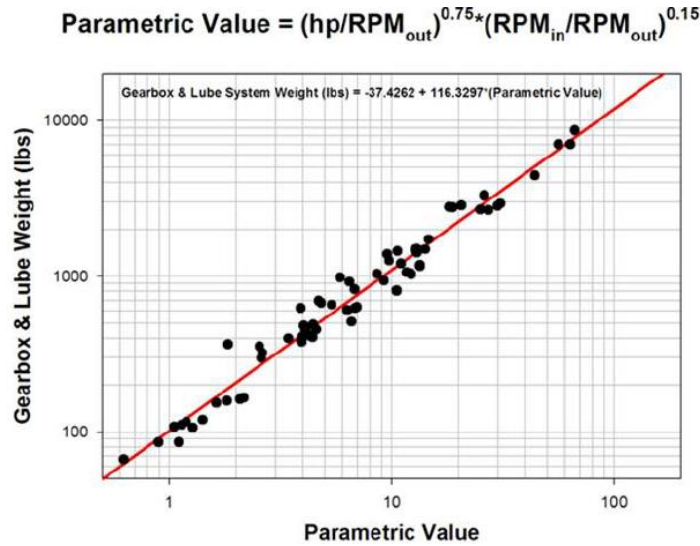


**Figure 32 Weight correlation for the gearbox and lubrication system [105]**

## 5.5 Baseline Aircraft and Mission Definition

This research used a simplified approach for the estimation of mission fuel burn based on the Breguet range equation in Eq. (5-30). This equation assumes an aircraft in level-flight cruising at a constant flight-speed ( $V_0$ ), where the engine thrust entirely balances the drag force. The fuel flow is approximated by the rate at which the aircraft is losing mass over the cruise period ( $W_{\text{initial}}$  relative to  $W_{\text{final}}$  under gravitational acceleration  $g$ ) assuming constant lift-to-drag ratio ( $L/D$ ) and engine overall efficiency (or SFC). The derivation of the Breguet range expression in Eq. (5-30) can be found in any aircraft performance manual, e.g. Ref. [119]. The discrepancy in calculated fuel burn from using the simplified approach

proposed by the Breguet range equation relative to higher-fidelity aircraft performance tools is in the order of 2.4% for turbofans [120]. A higher discrepancy might be expected for CRORs, which are most efficient during climb. However, this level of fidelity is assumed to be appropriate to undertake the preliminary design assessments in this research.

$$Range = \frac{V_0(L/D)}{g SFC} \ln \left( \frac{W_{initial}}{W_{final}} \right) \quad (5-30)$$

The single-aisle aircraft with the CROR mounted in the rear would fall into the same class as the Airbus A320 and Boeing 737 [121]. The engine assessments require top-level characteristics of a reference CROR aircraft and mission to assess its fuel consumption:

- Design PAX: 162, namely a maximum payload of 14,696 kg [121]
- Fuel on board: 12,315 kg [121]
- Aircraft take-off with full payload and fuel load
- Base airframe weight: 37,161 kg [121]. The weight of the CROR is added to the base airframe weight to produce the aircraft's operating empty weight
- Mission range: 3250 nmi from lift-off to touch down [121]
- Design speed at ToC and cruise conditions: 0.75  $M_0$
- Pusher CROR rear-mounted in the aircraft fuselage, with a lift-to-drag ratio (L/D) of 16.33 [122]
- The aircraft and engine technologies correspond to EIS 2035

The evaluation of mission fuel burn combines the assessment of engine performance and general arrangement through SFC and engine weight respectively for a fixed mission and aircraft configuration. This may then be used to undertake parametric assessments on the engine to identify optimum areas of the design space.

## 6 CROR Results and Design Space Exploration

The focus of this research is to integrate the transmission system into the engine preliminary design to identify the feasible design space for the CROR that results in optimum fuel burn. However, the combination of transmission system modelling (chapters 3 and 4) and engine performance modelling (chapter 5) brings into perspective the complexity of the problem. In the CROR, the performance of the propulsor dominates the performance of the engine. Therefore, this research evaluates the influence of the propulsor preliminary design parameters with the highest impact on the design space of a geared pusher CROR.

Common metrics used to assess the quality of engine designs are SFC and weight due to their direct effect on mission fuel consumption. The propulsor design parameters that dominate these metrics are

- Core nozzle pressure ratio (NPR),
- Rotational speed of the front ( $n_{FP}$ ) and rear propellers ( $n_{RP}$ ) and,
- Torque ratio (TR) between the propeller shafts.

In the evaluation of these parameters, only one has been varied at a time unless otherwise specified, with the rest of the CROR design parameters fixed at the design point. The results are expressed as percentage changes relative to the baseline engine configuration presented in chapter 5.

However, the design space exploration of the CROR would give an incomplete view of the problem if only SFC and weight were factored in. Mechanical design constraints and the effect of the transmission system on the gas path also play a key role in determining the feasible design space:

1. The allowable stress levels in the LPT are defined by the  $AN^2$  parameter. A lower bound of  $32 \cdot 10^6 \text{ m}^2\text{RPM}^2$  represents the  $AN^2$  parameter for the current level of technology. An upper bound of  $50 \cdot 10^6 \text{ m}^2\text{RPM}^2$  is the  $AN^2$  parameter selected to represent LPT stress levels by 2035.
2. The minimum cross-sectional area of the exhaust duct ( $A_{\text{exhaust duct}}$ ) must be larger than the exit nozzle area ( $A_{\text{nozzle}}$ ). This condition is presented in

Eq. (6-1). Having an exhaust duct with a cross-sectional area smaller than the exit nozzle area will accelerate the flow further than what is already accelerated by the contra-rotating propellers and increase the losses.

$$A_{exhaust\ duct} > A_{nozzle} \quad (6-1)$$

This research has assumed that the area of the exhaust duct can be reduced from the exit of the LPT to the end of the propellers to accommodate the power gearbox (the DPGB) and compensate for the blockage generated in the exhaust gas path by the presence of the pitch-transmission radial shafts (Figure 33). To evaluate the integration of the transmission system, it has also been assumed that the tip diameter of the exhaust duct is defined by the LPT outlet tip diameter.

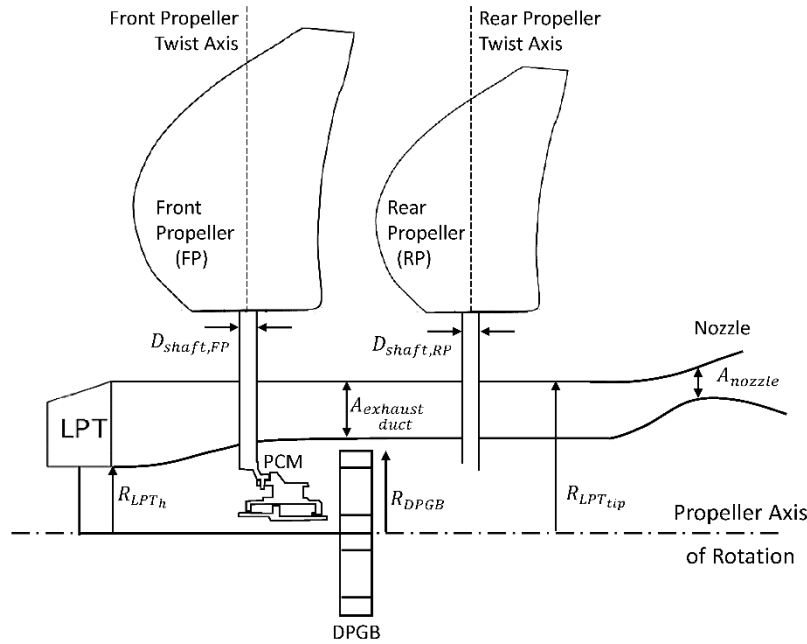


**Figure 33 Schematic of the propulsor area design constraints for the CROR**

Therefore, the integration of the transmission system in the gas path is assessed through the reduction in the cross-sectional area of the exhaust duct, which is defined by changes in its hub line. If the duct is not straight from the outlet of the LPT to the end of the CRP assembly, the outer diameter of the DPGB ( $D_{DPGB}$ ) defines the narrowest section in the duct:

$$A_{exhaust\ duct} = \frac{\pi}{4} (D_{tip,LPTout}^2 - D_{DPGB}^2) \quad (6-2)$$

However, the effective cross-sectional area of the exhaust duct further reduces due to the passing of the PCM shafts through the hot gas exhaust. In this case, the hub diameter ( $D_h$ ) of the exhaust duct is defined by whichever is biggest,  $D_{DPGB}$  or the outlet LPT hub radius ( $D_{h,LPTout}$ ):

$$A_{exhaust\ duct} = \frac{\pi}{4} (D_{tip,LPTout}^2 - D_h^2) - N_{bld} D_{shaft} \big|_{FP/RP} (D_{tip,LPTout} - D_h) \quad (6-3)$$

Finally, an additional condition has been imposed on the CROR design regarding the location of the power gearbox in the propulsor. If  $D_{DPGB}$  is bigger or equal to  $D_{h,LPTout}$ , the gearbox must be located between the propellers. The reason for this condition is to ensure engine compactness, as it directly affects engine weight and fuel burn.

$$\text{If } D_{DPGB} \geq D_{h,LPTout}, \quad DPGB \text{ located between propellers} \quad (6-4)$$

When the condition in Eq. (6-4) is not fulfilled, the gearbox could be located between the LPT and the propellers.

## 6.1 Nozzle Pressure Ratio Assessments

The nozzle pressure ratio represents the ratio between the pressure at the inlet of the core exit nozzle and the ambient pressure. In a CROR, the NPR defines the enthalpy drop across the free power turbine (the LPT) and the power supply to the propellers. In practice, changes in NPR would be the result of changes in the LPT exit area.

Lower values of NPR enable a higher enthalpy drop in the LPT and hence a higher power supply to the CRP. Lower values of NPR also result in higher input torques to the DPGB and an increase of  $D_{DPGB}$ , which may exceed  $D_{h,LPTout}$ . Therefore, the hub line of the exhaust duct would need to rise to accommodate the power gearbox, and hence  $A_{exhaust\ duct}$  reduces.

Figure 34 shows that  $A_{exhaust\ duct}$  would be smaller than  $A_{nozzle}$  at values of NPR below 1.25, rendering the CROR design infeasible, as indicated by the meshed area. The crossing of the PCM radial shafts through the gas path further reduces the effective  $A_{exhaust\ duct}$  and limits the design space to a NPR close to 1.3.

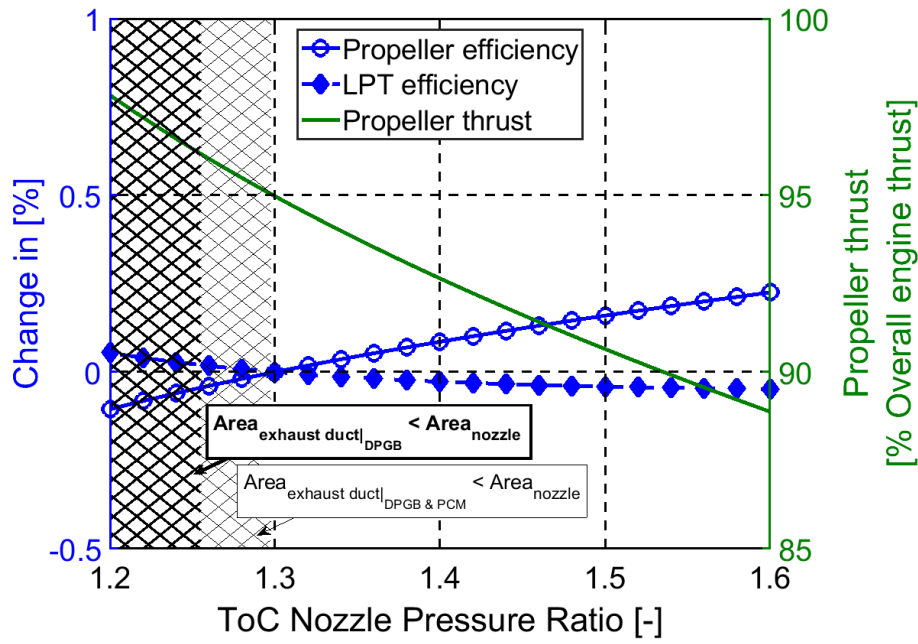


**Figure 34 Variation of net propeller efficiency, LPT isentropic efficiency and percentage of propeller thrust with ToC NPR**

Figure 34 presents the change with ToC NPR of the LPT isentropic efficiency ( $\eta_{is,LPT}$ ), net propeller efficiency ( $\eta_{NET,CRP}$ ), and change in propeller thrust as a percentage of the overall engine thrust. Even at the higher values of NPR the propellers produce almost 90% of the engine's net thrust. The reduction in power going to the propellers with increasing NRP improves the net efficiency of the CRP assembly. The less loaded propellers generate lower induced velocities, reducing the detrimental effect that they have on propeller efficiency. Finally, changes in LPT isentropic efficiency with the range of NPR presented in Figure 34 are relatively small (around 0.1%). The changes in LPT enthalpy drop with NPR are counteracted by changes in stage mean diameter and axial velocity, and consequently, result in small changes in stage loading and flow coefficient.

The baseline number of LPT stages has been calculated to achieve a target  $\eta_{is,LPT}$  of 0.94 for the reference engine (NPR = 1.3, TR = 1.29 and  $\pm 1100$  RPM propeller rotational speed). At the design point, the LPT stage count could be changed with enthalpy drop ( $\Delta h_{LPT}$ ), here measured through changes in NPR, to reduce engine weight. Higher values of NPR lead to lower enthalpy drops in the LPT, which could then provide high efficiency levels with a reduced number of stages.

However, the load in each LPT stage would increase and the target  $\eta_{is,LPT}$  cannot be achieved. For a constant LPT rotational speed, the LPT annulus design would need to be pushed away from the engine centreline to increase blade speed and increase efficiency. However, the small variation in LPT efficiency in Figure 34 for the baseline stage count indicates that the target efficiency could not be achieved with a lower number of stages for the current LPT design, as defined by its performance characteristics.

Figure 35 presents the change in thermal, propulsive and overall efficiencies and SFC with NPR at ToC. Increasing NPR increases the core jet velocity, resulting in an improvement in thermal efficiency. The reduction in LPT power production at higher values of NPR results in a smaller thrust production from the propellers and a worsening of the CROR propulsive efficiency. The overall efficiency reaches an optimum value at an NPR of 1.29, but 1.3 is the closest feasible value with the mechanical constraints assumed. The evolution of SFC mirrors the evolution of the overall efficiency.

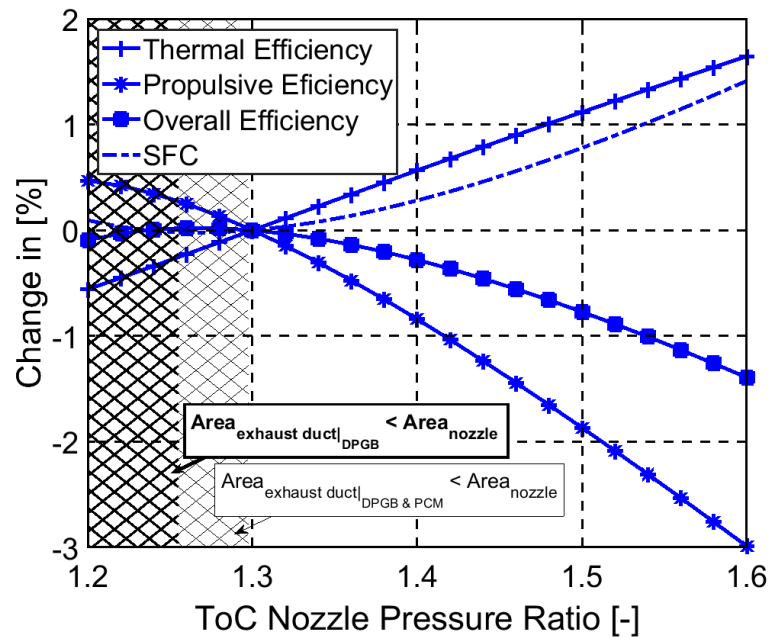


**Figure 35 Variation in ToC SFC, thermal, propulsive and overall engine efficiencies with ToC NPR**

Figure 36 presents the change in cruise SFC, weight and fuel burn for a prescribed mission range. Variation in mission fuel burn with NPR exceeds the

variation in SFC due to the contribution of weight change. As the power extracted from the LPT reduces, and hence the power supplied to the propellers reduces, the weight of the LPT and the CRP reduce. The weight reduction brings the optimum ToC NPR for fuel burn (1.37) to higher values than the optimum SFC.

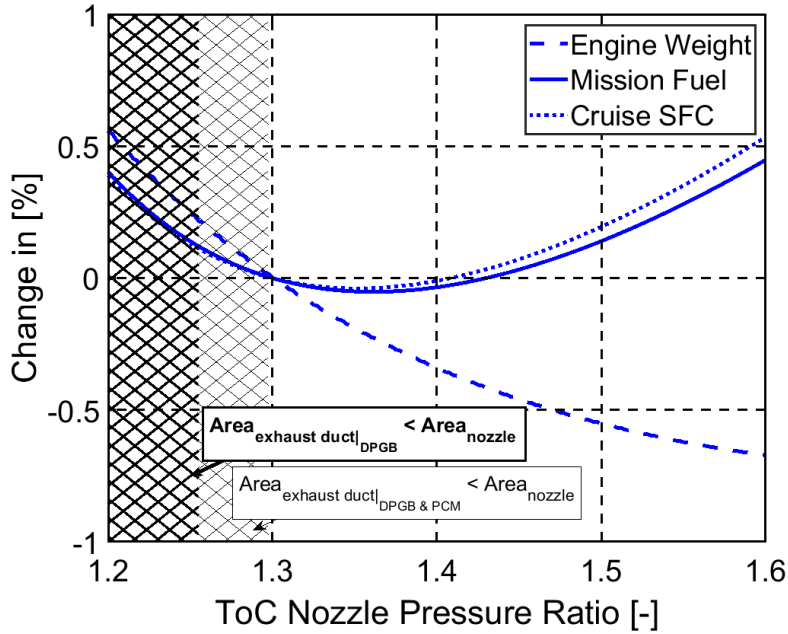


Figure 36 Change in cruise SFC, CROR weight and fuel burn with ToC NPR

## 6.2 Torque Ratio Assessments

The torque ratio defines the reduction ratio between the low-pressure shaft and the propellers. In this research, TR has been defined in Eq. (3-14) as the ratio between the torque in the carrier and ring shafts of the DPGB, which are connected to the front and rear propellers respectively. Therefore, by definition, lower values of TR lead to higher reduction ratios between the low-pressure shaft connected to the LPT and the propellers, improving the efficiency of each of these components. However, lower values of TR result in bulkier gearbox arrangements, as seen from Figure 23 in section 3.3.4. These trade-offs have been assessed through the integration of the transmission system in the CROR model. The mechanical constraints (the LPT stress levels) and the reduction in  $A_{\text{exhaust duct}}$  define the infeasible design space of the CROR, which is meshed out in the performance plots and coloured lines are used to indicate each condition.

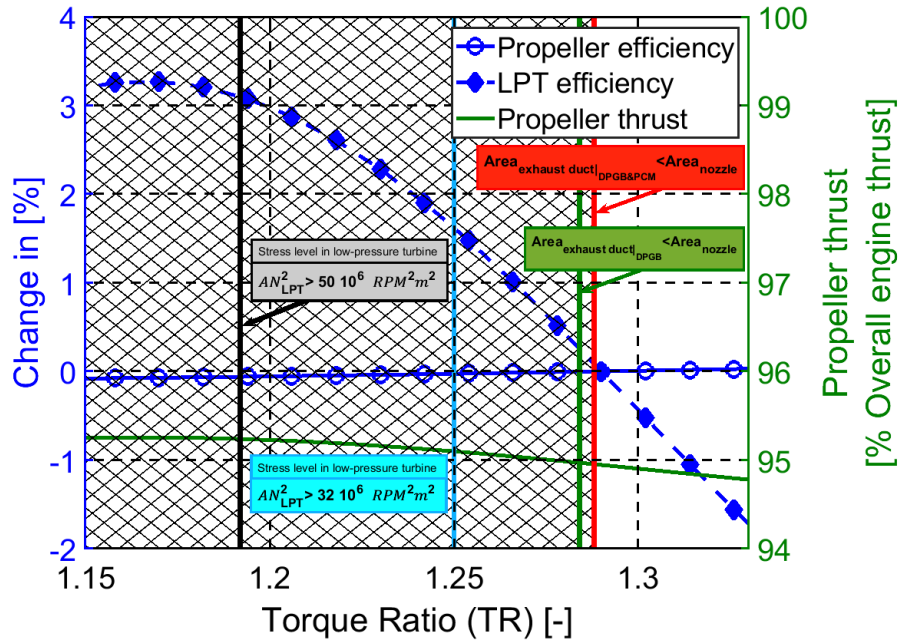


**Figure 37 Change in net propeller efficiency, LPT isentropic efficiency and percentage of propeller thrust with torque ratio (TR) at ToC**

Figure 37 shows the change in net propeller efficiency, LPT isentropic efficiency and the percentage of overall thrust produced by the propellers with TR at ToC (i.e. at turbomachinery design point). For a fixed value of ToC NPR, the LPT power generation, and hence the power supply to the propellers, remains almost constant (variation around 0.5%). Therefore,  $\eta_{NET,CRP}$  remains almost constant too. The use of a more elaborate propeller model (as proposed in Ref. [99]) including the calculation of propeller efficiency with changes in propeller loading would yield the same results (differences below 0.05%).

The reduction in thrust production from the propellers at higher TR is the result of changes in the induced velocities. The change in TR in this assessment defines the changes in power split between the front and rear propellers because their rotational speeds are kept constant, as seen from Eq. (3-6). Therefore, reducing TR proportionally reduces the power going to the front propeller. This accentuates the effect of the rear propeller induced velocities on the front propeller reducing the overall thrust production of the propellers.

Higher LPT rotational speeds with lower TR increase the isentropic efficiency of the LPT with a fixed number of stages up to a point of maximum efficiency. A

lower number of turbine stages could be used in this region to reduce engine weight while ensuring good turbine efficiency. Lower values of TR also result in a reduction in the LPT exit area, which may be beneficial from a stress perspective. However, the contribution of the increase in rotational speed dominates the stress levels at the hub of the LPT exit blade, estimated in this research with the  $AN^2$  parameter. The light-blue line in Figure 37 indicates the minimum TR (1.25) achievable before the blade stress parameter ( $AN^2$ ) exceeds  $32 \cdot 10^6 \text{ RPM}^2\text{m}^2$ . Improvement in LPT design and materials may improve the stress tolerance of the LPT, and reduce the minimum TR achievable with an upper bound of  $50 \cdot 10^6 \text{ RPM}^2\text{m}^2$  as marked with a black line.

Figure 38 shows that the lower and upper bounds of the  $AN^2$  parameter represent a potential reduction in ToC SFC of 1% and 3% respectively relative to the baseline TR value of 1.29. However, reducing TR increases the space envelope of the DPGB. At values of TR below 1.27 the design becomes infeasible, and below 1.28 with the integration of the PCM, as shown in Figure 37 and Figure 38 by the green and red lines respectively.

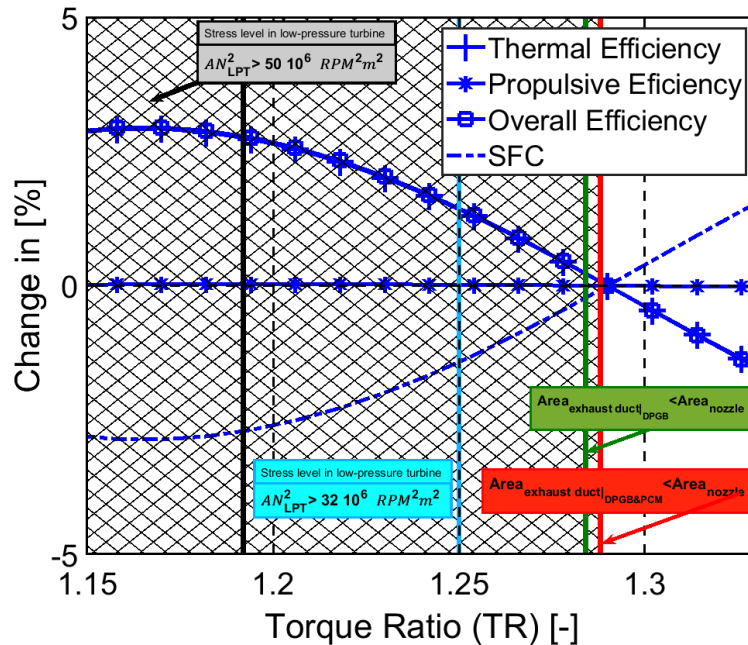


**Figure 38** Variation in ToC SFC, thermal, propulsive and overall efficiency with TR

Another possible way of reducing the stress levels in the LPT (i.e. reducing  $AN^2$ ) and increasing the feasible design space region is to increase the ToC NPR

beyond the baseline value of 1.3. Increasing NPR would increase the gas density at the exit of the LPT and reduce its exit area, resulting in a reduction of the  $AN^2$  parameter if the rotational speed of the low-pressure shaft is preserved. The trade-off between TR and NPR is discussed in section 6.3.

Like the changes in thrust production from the propellers, the propulsive efficiency of the CROR remains almost constant with TR. The improvement in LPT isentropic efficiency with lower TR translates into an increase in thermal efficiency and overall efficiency. The SFC evolution with TR mirrors the behaviour of the overall efficiency. The optimum ToC SFC would be obtained at  $TR = 1.17$ , which is beyond the feasible design space as defined by the mechanical constraints.

Finally, changes in TR present a twofold contribution to fuel burn variation through its effect on LPT and DPGB designs. Variation in rotational speed of the LPT affects its isentropic efficiency and annular geometry, contributing to changes in SFC and engine weight. Secondly, TR defines the space envelope and operation of the DPGB, affecting engine weight and gas path definition.

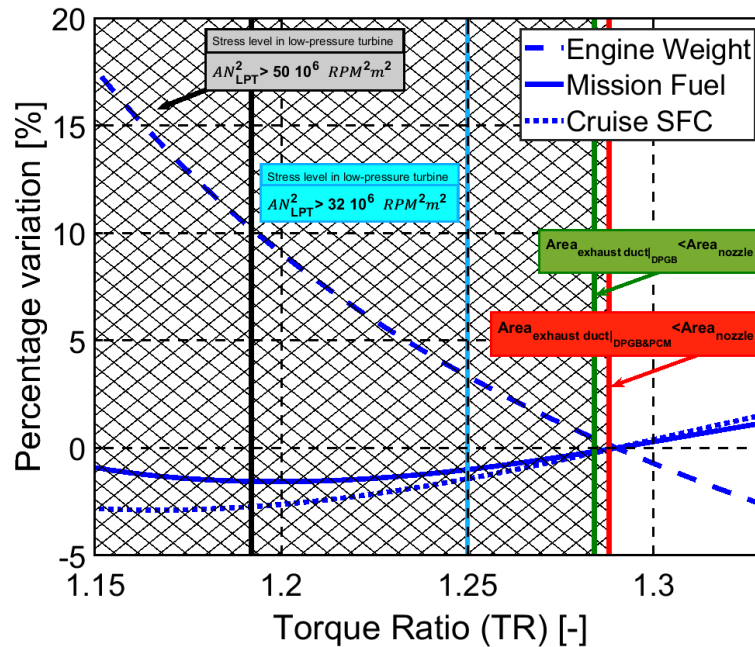


**Figure 39 Change in cruise SFC, CROR weight and fuel burn with TR**

Figure 39 presents the change in cruise SFC, weight and fuel burn for a fixed mission range. The changes in DPGB and CRP weight dominate the variation in

engine weight. The weight of the CRP assembly decreases as more power goes to the front propeller with increasing TR, i.e. as the power loading in the rear propeller decreases. The weight of the rear propeller reduces faster than the weight of the front propeller increases, resulting in a total CRP weight reduction. The smoothness of the weight curve is due to the weight estimation method used for the DPGB. However, abrupt decreases would be expected at the torque ratios where there would be a change in the number of planets as in Figure 23.

### 6.3 Combined NPR and TR assessments

The effects of NPR and TR have been discussed separately in sections 6.1 and 6.2 respectively. The assessments showed that low ToC NPR and low TR at constant propeller rotational speed result in lower ToC SFC. However, increasing NPR at lower values of TR could lead to further performance benefits.

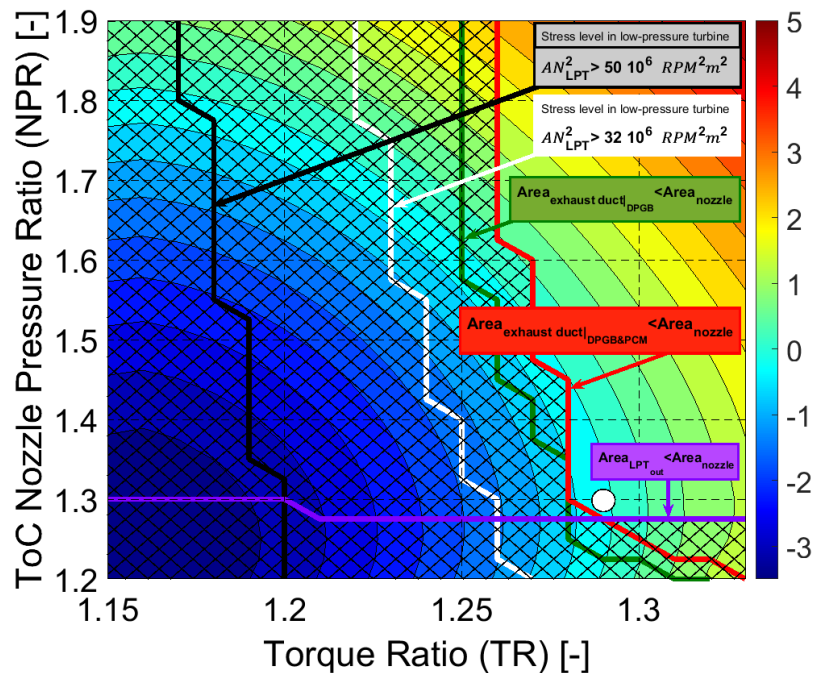


**Figure 40 Percentage change in ToC SFC with TR and ToC NPR relative to the baseline CROR (white dot)**

Figure 40 shows the percentage change in ToC SFC with ToC NPR and TR relative to the baseline engine configuration (1.3 NPR and 1.29 TR), indicated with a white circular marker. A maximum 1.44 % improvement in SFC could be achieved with a TR of 1.25 and an NPR of 1.325 for the  $AN_{LPT}^2$  limit of  $32 \cdot 10^6$

RPM<sup>2</sup>m<sup>2</sup> (white line). The potential improvement in SFC would further increase for higher values of  $AN_{LPT}^2$  with up to a 3% decrease relative to the baseline at 1.19 TR and 1.35 NPR for the upper bound of  $50 \cdot 10^6$  RPM<sup>2</sup>m<sup>2</sup> (black line).

However, the potential improvements in ToC SFC are reduced by the integration of the transmission system in the engine. The other mechanical constraint in the CROR design stems from the changes in  $A_{exhaust\ duct}$ , which must be bigger than  $A_{nozzle}$ . If the hub line of the exhaust duct is defined by the outer diameter of the DPGB (green line in Figure 40), only a 0.6 % improvement in ToC SFC is possible at 1.37 NPR and 1.27 TR. The improvement in ToC SFC further reduces when considering PCM blockage of the exhaust gas path, and a maximum 0.4% reduction can be achieved with 1.3 NPR and 1.28 TR (red line in Figure 40).

The purple line in Figure 40 represents the case where the exhaust hub diameter is defined by the LPT exit hub diameter. At TR values above 1.28, the DPGB could be located under the LPT, provided sufficient space is available for shaft connections and the lubrication system.

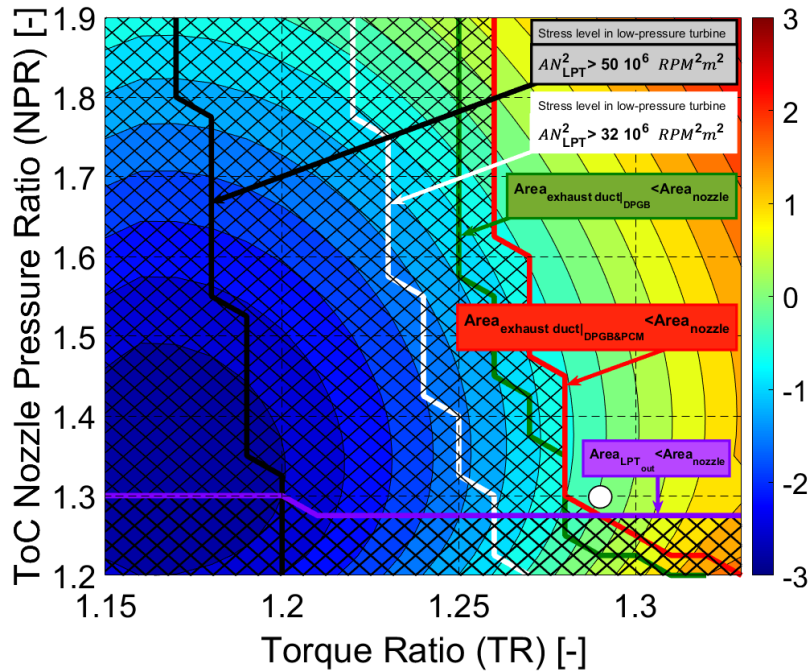


**Figure 41** Percentage change in fuel burnt with TR and ToC NPR relative to the baseline CROR (white dot)

Figure 41 shows the variation in fuel burnt with ToC NPR and TR. The changes in fuel consumption would follow similar trends to the ToC SFC plot in Figure 40. However, the percentage improvement in fuel burnt is relatively lower due to the effect of engine weight variation. The stronger contribution of TR to CROR weight variation relative to the effect of ToC NPR is apparent from Figure 39 and Figure 36 respectively.

The accuracy of the results depends on the design assumptions adopted for modelling the CROR. The focus of research has been on the integration of the transmission system and its impact on the propulsor. Therefore, sensitivity analyses in this study address key design parameters of the components in the propulsor, where the transmission system is located. Section 3.3 presents extensive parametric and sensitivity analyses for the DPGB. In section 5.2.1 the sensitivity of LPT inlet parameters was assessed at component level through variations in  $HTR_{in,LPT}$  and  $Mn_{in,LPT}$ .

The combined effects of transmission system integration and LPT design modifications would provide a gauge for assessing the accuracy of the multidisciplinary design framework developed in this research for trend assessments. Figure 42 shows the variation in ToC SFC with TR and NPR for different values of  $HTR_{in,LPT}$  and  $Mn_{in,LPT}$  relative to the baseline CROR, which is represented by a white circle in the central image. Increasing  $HTR_{in,LPT}$  by 1% reduces SFC by 0.5% relative to the baseline. An additional 2.5% variation in  $Mn_{in,LPT}$  further reduces SFC by 0.3%. Therefore, the trend accuracy of the model is  $\pm 0.8\%$ .

Lower values of  $Mn_{in,LPT}$  result in bigger LPT exit areas. The cycle conditions at the inlet of the LPT remain constant. Therefore, preserving a constant  $HTR_{in,LPT}$  pushes the LPT annulus design away from the engine centreline, freeing more space for the integration of the DPGB. Additionally, reducing  $Mn_{in,LPT}$  would increase blade mean diameter, and hence blade speed, increasing turbine efficiency.

Alternatively, increasing  $HTR_{in,LPT}$  at a constant  $Mn_{in,LPT}$  pushes the LPT annulus design away from the engine centreline. The tip diameter of the exhaust duct is

defined by the exit tip diameter of the LPT. Therefore, pushing the LPT annulus outwards frees more space for the DPGB.

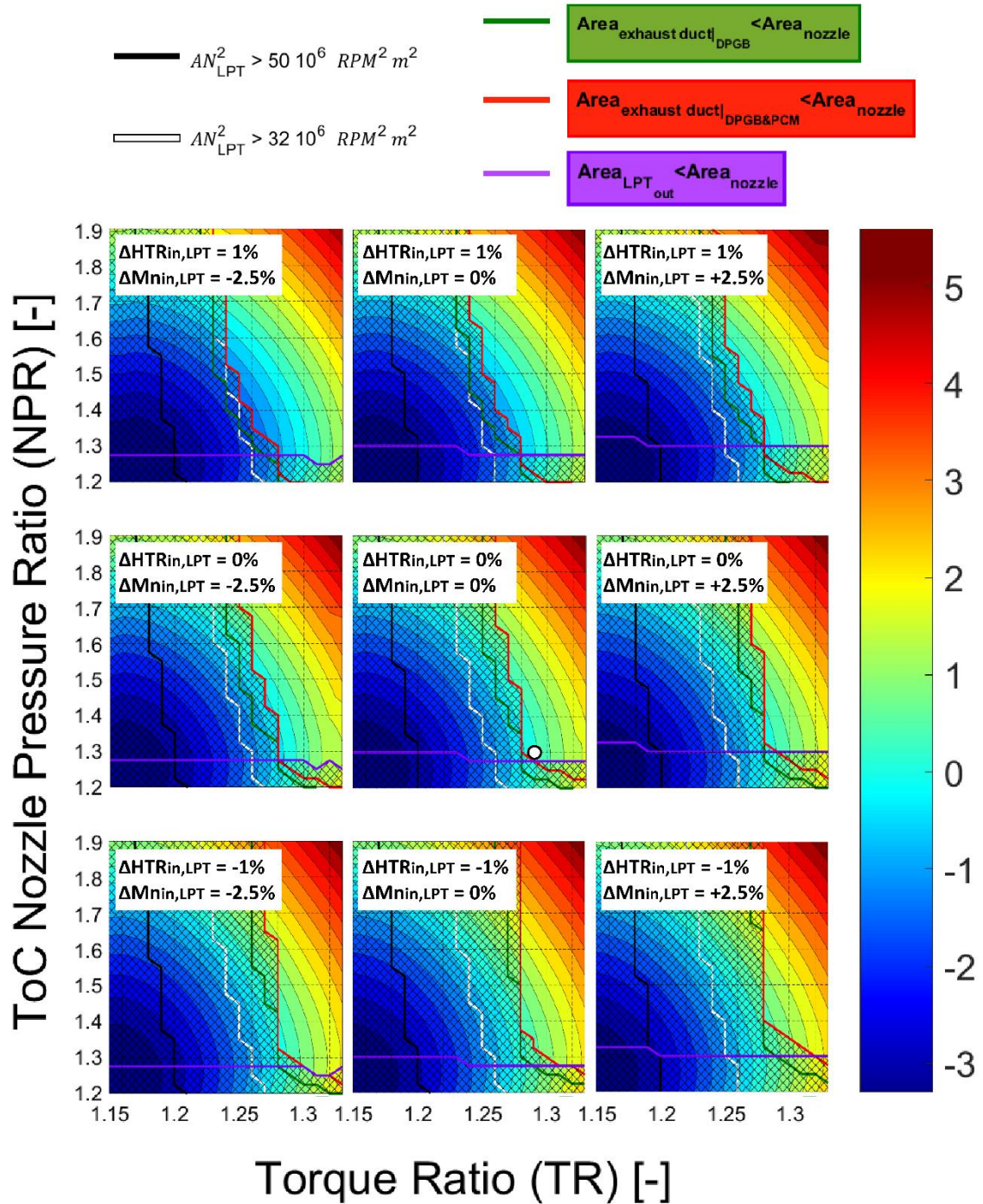


Figure 42 Sensitivity assessment of the effect of  $\text{HTR}_{\text{in,LPT}}$  and  $\text{Mn}_{\text{in,LPT}}$  variation on ToC SFC with NPR and TR relative to the baseline CROR (white dot)

Therefore, it can be concluded that higher  $HTR_{in,LPT}$  and lower  $Mn_{in,LPT}$  would provide optimum performance in terms of SFC.

## 6.4 Propeller Rotational Speed Assessments

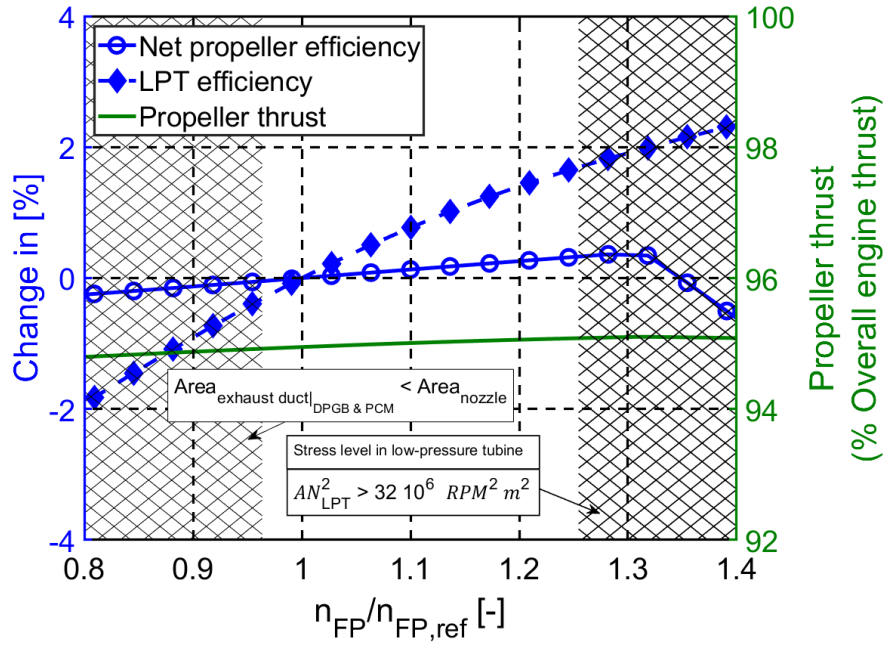
The effect on engine performance of propeller rotational speed has been assessed in two ways: varying the individual rotational speed of each row of propellers and, varying both propellers simultaneously.

Changing the rotational speed of the propellers changes the rotational speed of the LPT as shown by Eq. (5-3). These assessments keep the engine baseline values of ToC NPR and TR. Therefore, increasing the rotational speed of either propeller, while preserving the rotation rate of the other one constant, increases the rotational speed of the LPT. For a fixed number of LPT stages, the increase in rotational speed of either propeller also increases the isentropic efficiency and the power generated by the LPT, and hence the power going to the CRP.

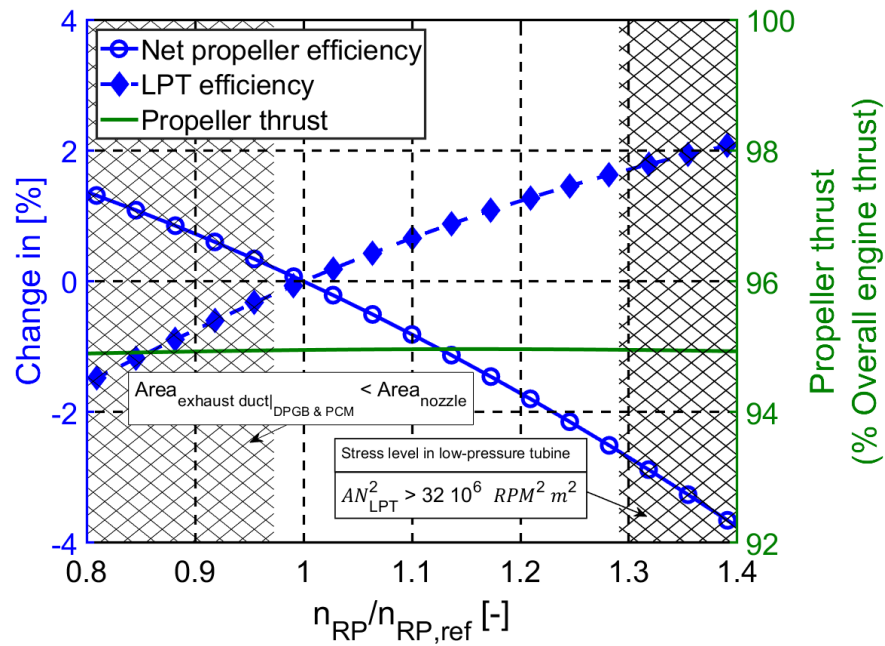
The change in rotational speed of either propeller also affects the power split between the propellers, as shown in Eq. (3-6). The power into the front propeller relative to the rear propeller changes proportionally to the ratio of rotational speeds in the front and rear propellers ( $n_{FP}$  and  $n_{RP}$  respectively) and TR. The evaluated range of rotational speeds is defined comparatively to the baseline reference value of  $\pm 1100 \text{ RPM}$  defined in section 5.2, namely  $n_{FP,ref}$  and  $n_{RP,ref}$ .

### 6.4.1 Variation of the Rotational Speed of the Individual Propellers

Figure 43 shows the change in net propeller efficiency, LPT isentropic efficiency and percentage of total thrust produced by the CRP with changes in the rotational speed of the individual propellers. Increasing the rotational speed of the front propeller increases the share of power going to the front propeller (Figure 43a). Therefore, the thrust produced by the front propeller increases and  $\eta_{NET,CRP}$  improves up to the point where compressibility effects dominate. At the same time, less power going to the rear propeller reduces the detrimental effect of the induced velocities on the front propeller. Therefore, the increase in power produced by the LPT translates into an increase in thrust produced by the propellers.



(a) Front propeller



(b) Rear propeller

**Figure 43 Change in net propeller efficiency, LPT isentropic efficiency and thrust produced by the propellers with propeller rotational speed**

Increasing the rotational speed of the rear propeller (Figure 43b) levels the share of power going to each propeller. However, for the baseline value of 1.29 TR, more power always goes to the front than to the rear propeller. Therefore, the increase in thrust produced by the rear propeller does not compensate for the

reduction in front propeller thrust resulting in a small reduction in the total thrust produced by the propellers and in net propeller efficiency.

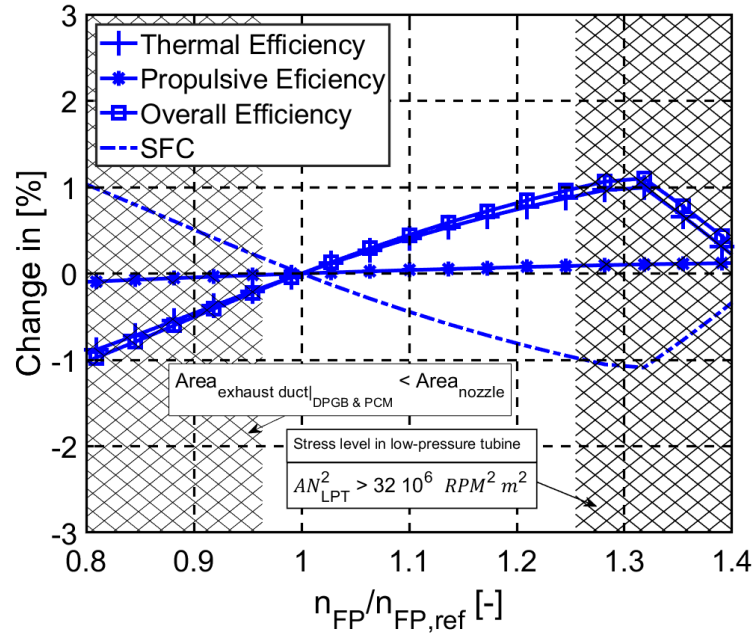
The difference in slope of the net propeller efficiency in Figure 43a and Figure 43b is caused by the contribution of the induced velocities to the rotational speed of the rear propeller shaft. When  $n_{FP}$  increases in the assessments in Figure 43a, the rotational speed of the rear propeller slightly increases too, contributing to the thrust production of the propellers. This effect is stronger than the induced velocities that the rear propeller generates for the front propeller. However, when  $n_{RP}$  increases, the share of LPT power going to the front propeller reduces more significantly due to the bigger increase in the rotational speed of the rear propeller relative to its baseline value.

Increasing the rotational speed of the front or rear propeller relative to their reference value improves the isentropic efficiency of the LPT by a similar percentage. The increase in rotational speed and power generated (enthalpy drop) of the LPT, lead to a reduction in turbine exit area. The result is a reduction in work and flow coefficients and an increase in LPT isentropic efficiency.

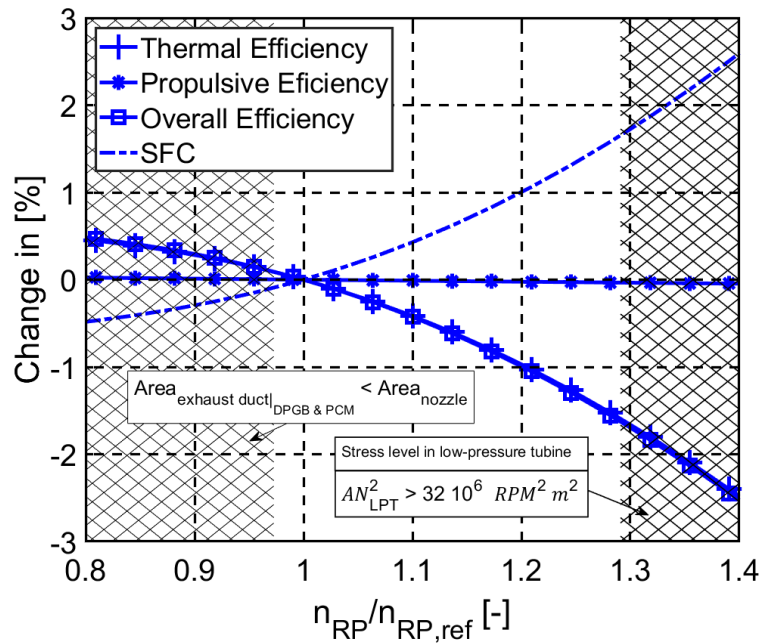
The design space shown in both cases presented in Figure 43 is reduced by the effect of stress levels in the turbine (measured with the  $AN^2$  parameter) and the reduction in cross-sectional area of the exhaust duct. The stress level increases with LPT rotational speed with increases in rotational speed of either propeller. This effect is indicated with the hashed area on the right-hand side of the images in Figure 43, where the stress level in the LPT corresponds to current technology levels of the  $AN^2$  parameter. Advances in turbine design and materials will improve the stress tolerance level and a bigger area of the design space will then become feasible. However, the rotational speed of the propellers would also be limited by acoustic constraints.

The left-hand side of the design space in Figure 43 will be limited by the integration of the transmission system in the CROR. The hashed region on the left-hand side of Figure 43 reflects the reduction in the cross-sectional area of the exhaust duct caused by the combined effect of the power gearbox and the blockage effects of the PCM shafts going through the gas path. As the rotational

speed of either propeller reduces, the torque of the DPGB input shaft increases, and hence the outer diameter of the power gearbox increases.

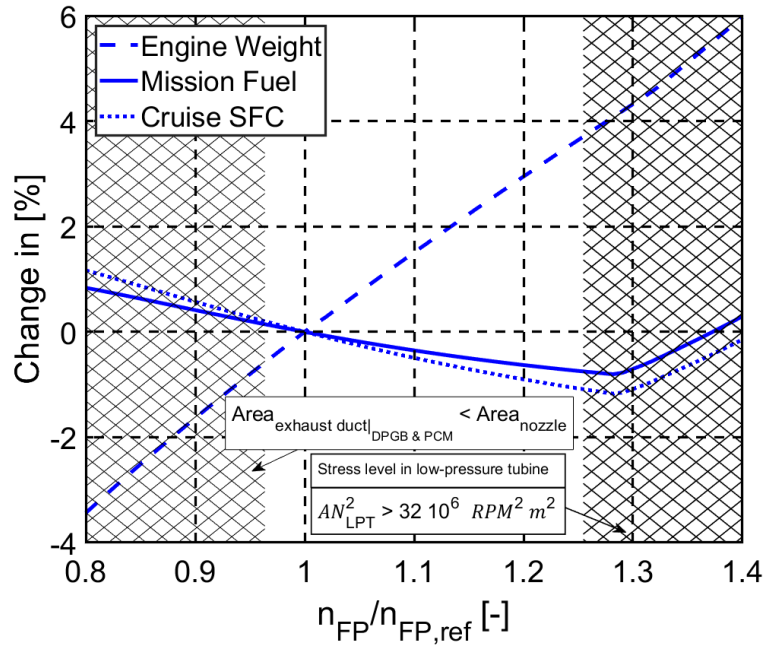


(a) Front propeller

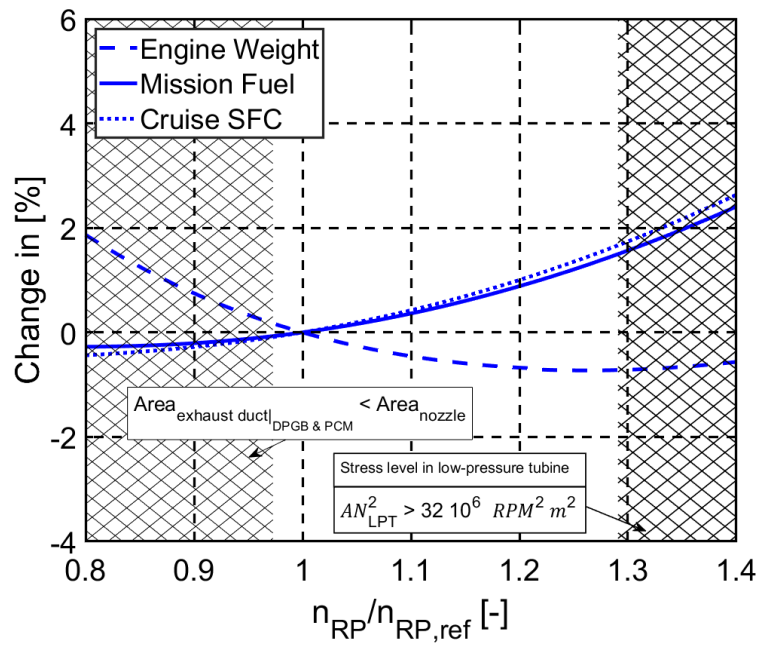


(b) Rear propeller

**Figure 44 Variation in ToC SFC, thermal, propulsive and overall engine efficiencies with propeller rotational speed**



(a) Front propeller



(b) Rear propeller

**Figure 45 Change in cruise SFC, CROR weight and fuel burn with propeller rotational speed**

The power generated by the LPT and transmitted to the propellers barely changes with  $n_{FP}$  or  $n_{RP}$ , as indicated by the small change in the percentage thrust produced by the propellers (Figure 43). Therefore, propulsive efficiency barely changes with propeller rotational speed. Changes in thermal efficiency dominate

the changes in overall efficiency. A minimum in ToC SFC is obtained at higher rotational speeds of the front propeller (Figure 44a). However, the opposite trend is observed in Figure 44b from increasing the rotational speed of the rear propeller. The same opposing trends are also apparent in the fuel burn changes presented in Figure 45. The changes in SFC and fuel burn with changes in rotational speed of each propeller are dominated by the effect of the induced velocities on the performance based on the modelling assumptions.

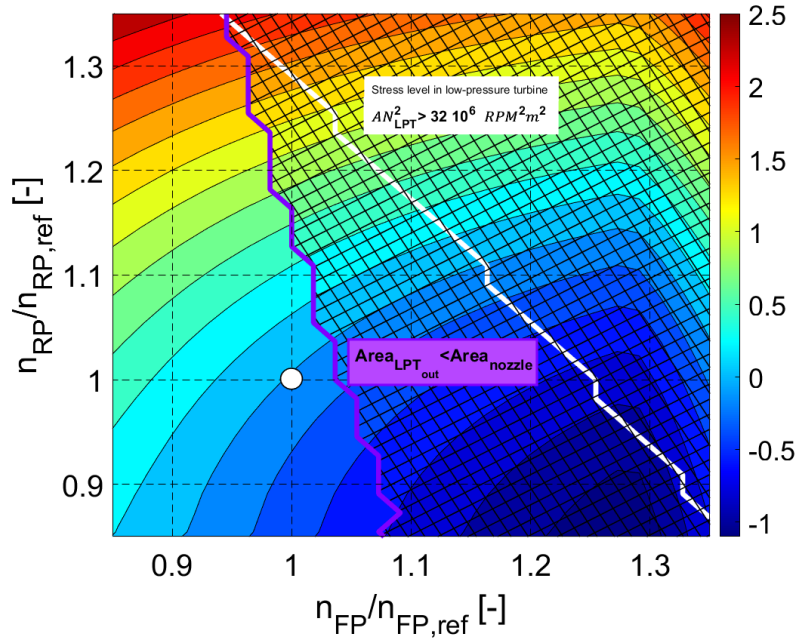
#### **6.4.2 Simultaneous variation of the rotational speed in propellers**

The effects of changes in propeller rotational speed on performance have been discussed separately in section 6.4.1. These assessments showed that high rotational speeds of the front propeller ( $n_{FP}$ ) and low rotational speeds of the rear propeller ( $n_{RP}$ ) lead to lower ToC SFC. Figure 46 shows the changes in ToC SFC with propeller rotational speed relative to the baseline (white marker) to assess the contribution of each propeller and define the feasible design space for the CROR.

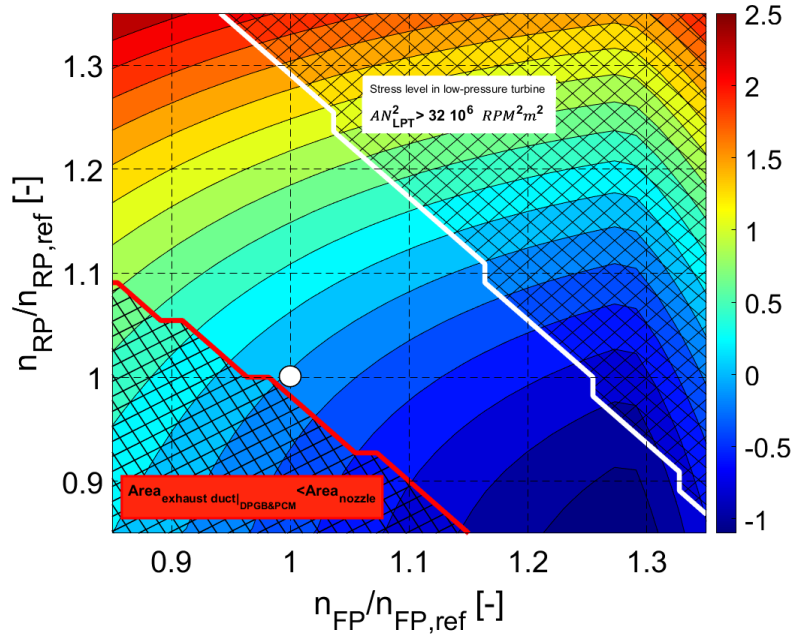
The changes in nozzle exit area are closely related to changes in mass flow through the core, which is the primary factor defining the shape of the variation in the ToC SFC curves in Figure 46. The same shape applies to the exit area variation of the LPT. However, for the LPT the exit area is also influenced by the efficiency of the turbine, which improves at higher rotational speeds. The changes in the exit nozzle area and LPT exit area are around 6% between the maximum and minimum points in Figure 46.

However, the blockage of the exhaust gas path caused by the PCM shafts increases at higher rotational speeds of the propellers shafts. The propeller-generated loads increase with the square of the rotational speed, requiring thicker radial shafts to transmit the changes in pitch. This effect is shown by the purple line in Figure 46a, where the feasible design area reduces significantly and the design region leading to optimum ToC SFC becomes unfeasible. The shape of the purple line is defined by the uneven contribution to LPT rotational speed of the front and rear propeller rotational speeds as shown in Eq. (5-3). Finally,

increases in the stress levels of the LPT are to be expected at higher rotational speeds as indicated by the white line.



(a) Restrictive design space



(b) New design space

**Figure 46 Percentage change in ToC SFC with propeller rotational speed**

Figure 46b indicates the infeasible design area due to the integration of the DPGB in the CROR. The rotational speed of the LPT dominates the changes in torque

entering the DPGB. This stems from the small variation in LPT power with propeller rotational speed. Therefore, at high propeller rotational speeds, the torque to the DPGB would be minimum.

Consequently, there might be potential benefits from modifying the LPT design for the high propeller rotational speed cases. One option would be to reduce the number of LPT stages while preserving high efficiency levels, which would reduce engine weight and favour more compact CROR designs. If the baseline number of stages is preserved, the LPT annulus could be brought closer to the engine centreline. This might allow the reduction of the propeller hubs, increasing the working area of the propellers and providing potential performance benefits. However, the operation of the propellers at high speeds would also be limited by acoustics. These aspects should be investigated as part of future work.

## 7 Conclusions and Future Work

**Overview:** Research focus has been on the preliminary design and integration of the transmission system in the CROR for the identification of its feasible design space. The transmission system in the CROR must comply with performance, space envelope and durability requirements. However, data on engine parameters are unavailable at the preliminary design phase. Any imposed engine constraints are guesses to the design, which may unnecessarily limit the feasible design space of the CROR. Past research has either tackled the design of the transmission system independently of the CROR engine modelling ([49]–[53]) or has been limited by many pre-determined engine constraints [7], [76]. Therefore, the first research question answered in this thesis is whether the preliminary design of the transmission system can be done solely from cycle parameters without imposing additional engine-based constraints.

This research provides a framework to calculate the space envelope of a power gearbox (a DPGB) for the CROR. The framework developed enables the identification of feasible regions for the preliminary design of the transmission system to produce compact gearbox arrangements and high speed-reduction ratios between the input and output shafts. Sensitivity analyses have been performed on key parameters affecting the size estimation of the transmission system, to evaluate the impact of assumptions on the design (refer to section 3.3).

The integration of the transmission system in the CROR then becomes a primary factor for the identification of feasible engine designs. It is, therefore, a major consideration that must be taken into account in the development of novel technology. This leads to the second research challenge. It is necessary to quantify the impact that incorporating the preliminary design of the transmission system into the engine model has on the feasible design space of the CROR. In other words, how does the integration of the transmission system affect the design space of the CROR?

This research provides a parametric study on key propulsor design parameters: nozzle pressure ratio, torque ratio and propeller rotational speeds. After the feasible design region is estimated based on the variation of these cycle parameters, the engine performance assessments need to be combined with the integration of the transmission system. The integration of the up-to-date gearbox sizing methodology into the engine preliminary design framework is used to inform the design-space exploration assessments. In such evaluations, the mechanical constraints of the transmission system and the LPT are considered in the engine preliminary design. This may then be used to carry out a full-blown optimisation.

**Transmission system** This research has shown that the axial span of the gearset in the DPGB is an order of magnitude smaller than the radial envelope. The gearset face width estimated in this research is smaller than 0.15 m for all design cases assessed, which represents less than 15% of the estimated spacing requirement between propellers (around 1 m in this research). This outcome should be considered when deciding the location of the DPGB in more detailed CROR assessments.

Additionally, the framework presented in this research enables the identification of the feasible range of torque ratios between propeller shafts that comply with the mechanical integrity of the transmission system. Higher torque ratios between the output shafts, i.e. lower gear ratios ( $m_G$ ), result in more compact gearbox designs. However, higher torque ratios for equal rotational speed in the propellers result in lower rotational speed for the LPT. This study shows that potential torque ratios lie between 1.1 and 1.33, with the higher ratios enabling more compact gearboxes having four or five planet gears. This trade-off has also been evaluated at the engine level.

Future work for power gearboxes offers a wide range of possibilities that might be worth taking forward in future research. Recommendations for further work are proposed below:

- **Gearbox weight estimation** This research has shown how the number of planets in the CROR power gearbox affects its size envelope, and hence its

weight. Existing approaches for DPGB weight estimation are based on correlations for pre-existing gearbox designs and do not reflect the change in the number of planets for different operating conditions. These approaches might be appropriate for design trend assessments, but not for exact weight calculations. Additionally, the effect on the DPGB weight of having different rotational speeds in the output shafts connected to the propellers is not captured by these trends. Therefore, the development of accurate gearbox weight estimates for future DPGBs is required for the design of complete CROR propulsion systems. A possible way forward might be the development of a simple weight estimation tool for the DPGB based on gear solid volume (as opposed to total system weight). The development of such a tool could be extrapolated from the minimum-weight equations proposed in Ref. [87] for simpler epicyclic gearbox designs.

- Further preliminary design methodologies for gearbox components** The methodology developed in this research for the power gearbox preliminary design assumes the gearset to be the envelope-defining component in the gearbox assembly. The simplicity of the approach enables the integration of the gearbox in the CROR model to undertake preliminary design trend estimations. However, increasing the accuracy of further CROR assessments would require the development of similar methodologies for the preliminary design of the bearings, shafts and casing to account for all components in the gearbox assembly and the interaction between them. The preliminary design of the sun and ring shafts can be estimated from the expressions in Ref. [96]. The design of the carrier shaft is more complex than that of the sun and ring shafts. The carrier not only needs to resist the estimated torque and pitching moment loads from the front propeller, but also the loads resulting from the misalignment of the planet gears in the DPGB assembly. For the AGBT project, discussed in Ref. [50], the sizing of the carrier was achieved with a 3-D finite element model (FEM) of the entire gearbox assembly. The model was created combining the geometry and load information of all the other assemblies interacting with the carrier, i.e. load distribution from the planet bearings (from the bearing analysis), position of planet gears and operation

of the assembly. Therefore, further gearbox preliminary designs would require 3-D modelling of the assembly to assess the interactions between components. The following section addressing further work on transmission systems hints at a way forward for bearing design.

- **Bearing selection and design** Once the gears and shafts in the gearbox have been defined, the next step would be to determine the type of bearings and their position on the gearbox. Bearings support axial and radial loads, which define the adequate type of bearing for each location. The future of bearing technology includes magnetic (long term alternative) and hybrid (ceramic, short/medium term alternative) bearings, which have the higher speed and diameter capabilities required by advanced gas turbines [123]. Regarding the selection of planet bearing technology, it is highly dependent on the type of planet gear technology and operating loads. The most common theory used on bearing-design computer codes by bearing manufacturers, the International Organization of Standardization (ISO), and the American National Standards Institute (ANSI)/Anti-Friction Bearing Manufacturers Association (AFBMA) is Lundberg-Palmgren theory, which is based on Hertz contact stress theory [124]. Following Lundberg and Palmgren's approach, the preliminary design of the planet bearings can be done in isolation based on a finite life and reliability at an acceptable risk [125]. However, the final design would strongly rely on the interaction with the planet gears and the load distribution during operation. The development of planet bearings for the stringent CROR operating conditions would require, therefore, a higher level of design. This aspect would require the calculation of the gear-bearing interaction loads and the assessment of the stresses generated, which may affect the design envelope of the DPGB assembly.
- **Gearbox engine location** This research assessed the location of the DPGB in the CROR based on the interaction between the power gearbox and the exhaust duct, positioning it either between the LPT and the CRP or between propellers. Positioning the DPGB between propellers could potentially result in shorter engine configurations. However, the need for an oil system for lubrication and cooling of the DPGB might introduce a lack of flexibility on the

spacing between the front and aft rotors, which is a key parameter to propellers efficiency and noise pattern. An example of this choice is the SAGE CROR architecture conceived by Rolls-Royce. Alternatively, positioning the DPGB between the LPT and the front propeller would enable a direct oil network connection between the gearbox and the gas generator. An example of this choice is the SAGE2 CROR developed by Safran.

These trade-offs indicate the need for further examination of the optimum location of the gearbox in the CROR. Additionally, the space envelope available for the DPGB underneath the gas path would be highly dependent on the LPT design, as shown by the sensitivity assessments undertaken in this research. Therefore, further research should evaluate the preliminary design of the propulsor as a whole to better understand the interactions between components and achieve more accurate engine layout estimations.

The preliminary design and modelling of the PCM in this research was done to a lower level of detail than the DPGB. However, valuable insights into the integration effects of this component on engine performance have been obtained. The areas of research for the pitch control mechanism that need further exploration are numerous, from the sizing of the PCM actuator to the integration of the control system into the engine preliminary design assessments or the development of weight and size correlations for quick integration into engine optimization assessments.

**Design Space Exploration** The consideration of mechanical constraints on the gas path of the CROR suggests that fuel burn savings as high as 3% could be achieved relative to the baseline engine going to advance turbine designs and using advanced materials. An example of a modification to turbine design and materials for blade root stress reduction is the use of highly-tapered hollow blades from low-density materials, with a subsequent weight reduction [34]. However, the difference in pitch to chord ratio from the blade root to tip might pose a challenge for highly efficient work extraction. This should be addressed as part of further LPT design in the context of the CROR application.

The margin for those potential fuel burn savings reduces when the transmission system is integrated into the CROR and its integration is considered in the design of the exhaust gas path. The potential fuel burn gains then reduce to 1%. The outcome obtained reinforces the claim of this research that the feasible design space of the CROR must be defined bearing in mind the design of the transmission system during the preliminary design.

The accuracy of the design framework depends on the modelling accuracy of the individual components in the CROR. The performance accuracy of the CROR model primarily depends on the modelling of the contra-rotating propellers, which dominate the thermodynamic cycle calculations. The modelling approach adopted in this research, presented in chapter 5, would result in prediction accuracies of the propeller overall power and efficiency quantities with differences below 1%. The mechanical assumptions in the modelling of the CROR would also add uncertainty to the results. In the absence of experimental data or detailed simulation information for similar CROR designs, the sensitivity assessments in section 6.3 showed  $\pm 0.8$  % variation in the results, which is indicative of the accuracy of the tool.

This shows how necessary a further examination of the propulsor system is to obtaining maximum performance benefits with the CROR. An optimisation of the propulsor is suggested for future work. The main points that this optimisation would have to address are the design of optimum efficiency free power turbines compatible with the integration of the transmission system; a detailed analysis of the flow conditions in the exhaust duct to identify the sources of loss and the potential energy recovery with the contra-rotating propellers; the refinement of propeller performance estimation methods; the incorporation of a variable area guide vane to optimise engine performance.

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