ANALYSIS METHOD FOR THE OPTIMAL DESIGN OF HELICOPTER MAIN GEARBOXES WITH A COMBINATION OF STRUCTURAL AND THERMAL INFLUENCE

Barry James
Chief Technical Officer
Romax Technology Ltd.
United Kingdom

Youn Park
Global Head of Aerospace
Romax Technology Ltd.
United Kingdom

George Grant
Engineer
Romax Technology Ltd.
United Kingdom

Keywords: Analysis, Gearbox, Transmission, Design, Structural, Thermal, Damage, Safety

One of the safety critical systems of a rotorcraft is the drivetrain, and within that the transmission. This system has the role of transferring power and rotational speed from the engine to the rotor blades. The transmission in a rotorcraft typically operates in a much more challenging environment than is experienced in other types of vehicle, and the performance and reliability of the transmission is critical and directly related to safety. The stringent weight requirements also impose limits on cooling system design, and the hovering flight requirement can produce quite a challenge thermally as no cooling air flow is coming from the vehicle motion. As such, helicopter transmissions tend to have to tolerate high operating temperatures.

However, the analysis methods established for assessing the transmission under flight loads are typically based on experience. This experience based approach does not lend itself to assessing important trade-offs such as reliability, efficiency and weight early in the design process. Having to rely on testing for confirmation of some aspects of performance means that opportunities for optimising and improving transmission systems are lost and, even if performed, analysis optimisation studies not including the thermal effects may later turn out to not be sufficient. All of this can lead to high expense and time incurred due to issues with the design often only being identified at the testing stage.

It is known that structural loads are a major consideration in the design process as it determines the size and weight of the gearbox and its sub components to ensure robustness. The industry has well established design procedures to address failure modes from structural loads, however, failure modes of coupled structural and thermal influences are not so well understood. It has been observed in recent years that thermal influences can accelerate fatigue damage leading to significant safety issues in helicopters. Romax has developed experience in these areas through collaborative research and industrial projects, giving us good insight into how to couple structural and thermal influences, and we believe that these methods can help design and development engineers in the rotorcraft industry to avoid potential failures in helicopters drivetrains.

1. BACKGROUND

1.1 Introduction

Thermal models are important in gearbox simulation, during its operation a gearbox generates heat at the rolling contact interfaces (gears and bearings) and this heat needs to be removed (typically by the lubricating fluid) to maintain normal operation. Not providing sufficient cooling can result in a variety of different issues ranging from inefficient operation to catastrophic component failure.

Comprehensive computational fluid dynamics (CFD) models with conjugate heat transfer coefficients provide the greatest insight in simulation. However, the problem with these models is that they are required to solve the evolving temperature fields and as a result are extremely computationally expensive, requiring the use of high performance computing (HPC). This limits their practical usage to steady state thermal predictions (still requiring HPC's) where the fluid temperature does not change and only a short transient solve is necessary. Given that access to HPCs is expensive, for most thermal problems alternative, efficient but accurate methods have to be relied upon.

This paper looks at these simpler methods to predict operational temperature profiles within the transmission and it sub components, (shafts, gears and housing), and gear analysis using both structural and thermal loads, in addition to analysis methods for assessing the transmission under structural loads. The combination of structural and thermal analysis methods presented here results in a simulation methodology suitable for assessing important trade-offs and realising the opportunity for optimising designs early in the design process. Prediction of the thermal fields also becomes useful in assessing thermal failure modes of gears such as scuffing, for which better gear bulk temperature data can be provided to improve the resolution of scuffing assessment calculations.

A rotorcraft case study is presented, detailing the types of the concept stage design decisions that can be made through accurate modelling of the thermal profile along with the effects an accurate thermal profile has on fundamental gearbox design parameters.

2. APPROACH & METHODOLOGY

The full study is undertaken across 3 models, a transient 1D thermal model, a steady state 3D thermal model and a coupled thermal and structural 3D Finite Element model.

2.1 1D Thermal Model

Thermal simulations lend themselves well to 1D models due to the simplistic and linear nature of the fundamental equations. While creation of the model structure is simplistic, detailed knowledge is required when assigning material properties, notably the thermal conductivity, k, and heat transfer coefficient, h. Outputs for these model types are typically limited to bulk temperatures. However, this belies the models power. For example, a concept design for a sub system such as lubrication and cooling can be defined including the required mass flow rate, nozzle locations and heat exchanger requirements.

Perhaps more pertinent to the rotorcraft industry is the issue of over temperature. These types of problems occur transiently and it is here that an accurate 1D model can be truly exploited, for example the ability to model the transient behaviour of a gearbox under the loss of lubricant [1]. Significant resources are expended in this area with production units not always meeting the criteria [2]. Undertaking laboratory testing during the design phase is prohibitively costly; in contrast the 1D model allows simulation during the concept stage, combined with the ability to run numerous iterations of the model with minimal computational cost.

2.2 3D Thermal Model

Moving to 3D FE provides great insight into the distribution of temperature within solid structures. This comes at a substantial increase in computational complexity and cost, while typically restricting the simulation to steady state or limited transient thermal predictions. Simulations must now be solved in the time domain because of motion affecting the fluid (e.g. rotating gears).

As the notion of a steady state solution becomes more complex, the cost increases. However computational cost can be saved by using simpler methods to predict approximate steady state answers. This allows the advanced methods to start their iterations closer to the answer. The inverse is also true in that the simpler 1D methods can be made substantially more accurate through calibration of prescribed coefficients based on more advanced CFD methods, and greatly improved in spatial resolution through co-development with FE models.

Three levels of model fidelity are defined below:

2.2.1 Prescribed Heat Transfer Coefficients

This is a 3D FE model with user prescribed values for heat transfer coefficients (HTC), as a result it is reliant on the knowledge of the user if accurate results are to be obtained. Simulations are limited to short duration transient and steady state. Fundamentally this is an enhanced version of the 1D model, as such similar steady state results should be expected if the 1D model is appropriate.

2.2.2 CFD Solved Heat Transfer Coefficients in a Constant Temperature Fluid

The 3D FE with prescribed HTCs method can be greatly improved upon by solving the HTCs. This is done through a combined CFD model with conjugate heat transfer. In these simulations, a 3D CFD model solves alongside an FE model. The conjugate heat transfer model solves the movement of the fluid, and the resulting convection HTCs to the solid (which is modelled in 3D FE).

In applications such as external air cooling, the input temperature (external air) does not change over the simulation, so a long transient solve is not required. These simulations are computationally complex and limited to steady state solutions, where the input fluid temperatures are known.

2.2.3 CFD Solved Heat Transfer Coefficients in a Transient Temperature Fluid

Where the fluid temperature changes over time (e.g. coolant) the HTC is no longer constant. The model is required to be solved over the evolving temperatures to capture effects of the changing HTC.

This type of simulation is generally limited to use in steady state predictions once thermal equilibrium has been reached and is computationally very expensive, potentially requiring the use of an HPC.

2.3 3D Coupled Structural & Thermal Model in RomaxDESIGNER

*Romax***_DESIGNER_** is a virtual design and testing platform that can be used for the development of rotorcraft drivetrains. In this case study the gearbox comprises of 3 types of components, shafts (including gears), housings and bearings.

Shafts can be modelled using either beam elements or 3D FE models (either internally or externally meshed). Gearbox housings are typically complex geometrical shapes and cannot be described using standard analytical models. Instead these are imported as externally meshed 3D FE data files and the reduced stiffness matrix computed and extracted. Bearings are modelled using either internally meshed 3D FE or as non-linear stiffness bearings that take into account the applied load, internal geometry, clearance, and resulting load distribution among the rolling elements.

Thermal profiles from solved 3D finite element models of the components can be imported into *Romax***_DESIGNER_** (Fig. 1), thus enabling the coupled structural and thermal analysis.

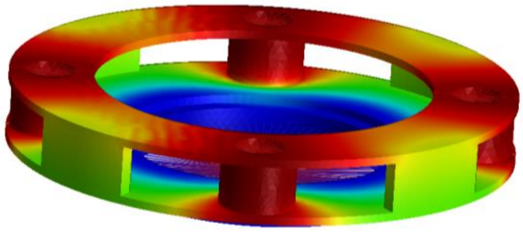


Fig. 1 *- Planet Carrier with a temperature Profile*

2.4 Thermal Failure Modes

There are 2 predominant failure modes linked to heat within gearboxes, a brief description of each is given below and key variables defined for comparison within the case study.

2.4.1 Gear Scuffing

Gear scuffing occurs when the lubricant film thickness starts to break down. Typically this is because the local gear temperature is too high due to high coolant feed temperatures or excessive contact forces or sliding, all of which reduce the lubricant's viscosity and thus reduce the film thickness. A decreased film thickness leads to increased asperity tip contact which in turn increases the friction and subsequently the local temperature.

Fig. 2 *- Gear Scuffing*

The cycle continues until the temperature and contact forces result in mechanical bonding of the asperity tips which are then immediately sheared; a rapid build-up of material on one surface and pitting on the other ensues. This increase in roughness further hinders the oil film contact resulting in greater friction and heat generation leading to rapid gear failure.

*Romax**DESIGNER*** includes within its gear scuffing reporting scheme the safety factor for both the flash method (ISO/TR 13989-1) and the integral method (ISO/TR 13989-2) for assessing gear scuffing. The flash method is the instantaneous temperature at the gear mesh contact surface and the integral method is the time averaged temperature value.

2.4.2 Thermal Loads Degrading Performance

Thermal expansion, in particular that of dissimilar materials (e.g. housings and shafts) can result in the undesirable loading of components and reduction in efficiency. For example, gear mesh misalignment, $F_{\beta x}$, would most likely be affected by thermal expansion, resulting in transmission error predictions made without thermal affects being inaccurate. This is particularly relevant in the aerospace industry where power density is high and components and sub systems such as cooling are designed with minimal weight as a key criterion.

The following variables are defined to demonstrate the effects of thermal loads. Gear mesh misalignment, $F_{\beta x}$, is a measure of the largest gap between two points on the face width, once two gears have been brought into contact.

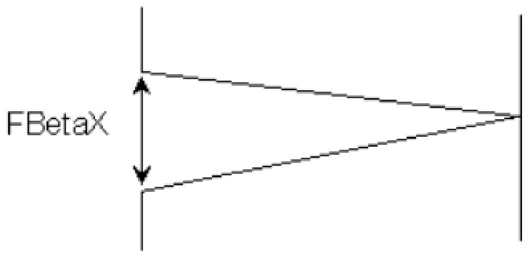


Fig. 3 *- Gear Mesh Misalignment*

The contact face-load factor, $K_{H\beta}$, is the ratio of the peak load and average load over a meshing gear face, its minimum value is 1 which would indicate the entire face is uniformly loaded. The value is used in the gear rating standard ISO 6336-2:2006 Calculation of load capacity of spur and helical gears.

Bearing dynamic equivalent load is the equivalent load applied to a rolling element bearing. It is fundamental to calculating bearing life as defined in ISO 281:2007 Rolling bearings – Dynamic load ratings and rating life.

Bearing misalignment is the angular difference between the inner (typically shaft mounted) and outer (typically housing mounted) bearing race. This value is considered for the more complex bearing rating calculations including both the ISO and DIN technical supplements to ISO 281:2007.

3. CASE STUDY

The gearbox used for this case study is shown as part of the drivetrain in Fig. 4 and isolated in Fig.5. The gearbox was chosen as it contains a range of components encompassing a typical rotorcraft gearbox.

Power is input through a 3-way bevel gear mesh and transferred to the rotor via a compound planetary system containing 4 and 8 planets in the 1st and 2nd stage respectively. Power is transferred to the tail rotor shaft via a cylindrical gear mesh and then through a second 3-way bevel gear mesh. The model also includes various ancillary outputs which increase the power load on the system and subsequently the heat inputs in the models.

Fig. 4 *- Full helicopter drivetrain*

The case study simulations are split into two elements, a conservative mission profile to establish correlation between the models, followed by an aggressive mission profile to demonstrate design choices.

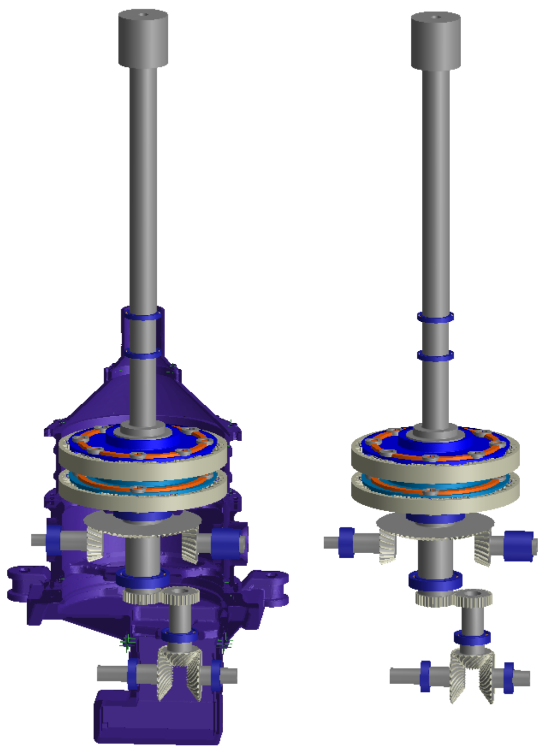


Fig. 5 *- Case Study Gearbox modelled in RomaxDESIGNER*

All simulations were run to steady state conditions in both the 1D and 3D models. Temperature profiles were imported into *Romax**DESIGNER*** and the variation in the previously defined variables investigated.

3.1 3D Model Resolution

Romax has constructed a database of heat transfer coefficients and wetted areas for different operating conditions. This has been obtained from a range of sources including academic papers and internal simulations ranging from basic particle physics to complex CFD run in transient temperature fields for meshing gears.

The accuracy of these variables allows the use of the simplest 3D FE model as defined in Section 2.2. This enables 3D simulations to be conducted on standard desktop computers with short duration runtimes. In addition, it further enhances the 1D model data through the use of accurate coefficients.

3.2 Correlation of 1D and 3D Model Results

The 1D model produces outputs in the form of Fig. 6, a line trace of temperature against time for lump masses. In the case of Fig. 6 these traces pertain to the housing temperatures. Naturally, individual data points can be extracted.

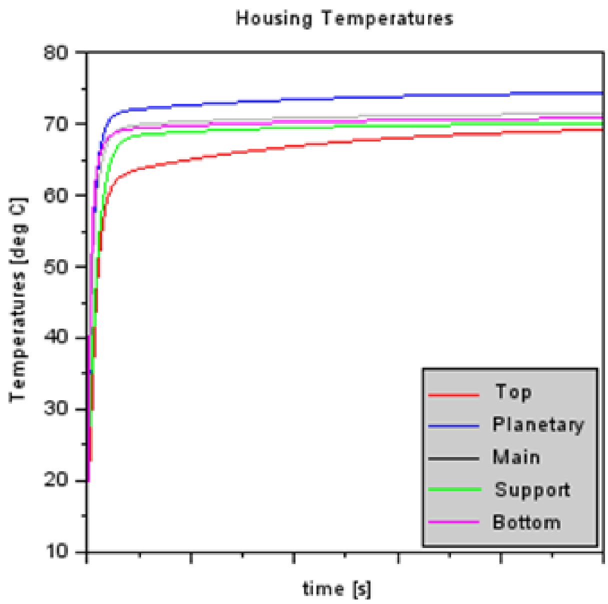


Fig. 6 *- Example 1D Results, Housings*

The 3D FE model produces outputs in the form of temperature profiles over a 3D model as seen in Fig. 7. This produces a visual representation of the temperature profile for the component and allows the identification of local hotspots and a better understanding of temperature throughout the system. Housings were specifically chosen for these 2 examples as they provide the greatest likelihood for discrepancies between the 1D and 3D models due to their size and complexity with respect to heat transfer.

A comparison of 1D and 3D results is provided both visually (Fig. 9) and numerically (Table 1). 1D lump mass temperature values provide an accurate representation for smaller components such as the Tail rotor take off shaft, web and teeth (Fig. 9b). However, as the scale and complexity of the geometry increases, the lump mass value should be interpreted as a mean average value.

The complexity and quantity of models applied to each component does not appear to reduce accuracy. Housings include numerous models defined for conduction and convection, while gears and shafts include multi-layered convection schemes for the various fluids and complex schemes for the conduction pathways.

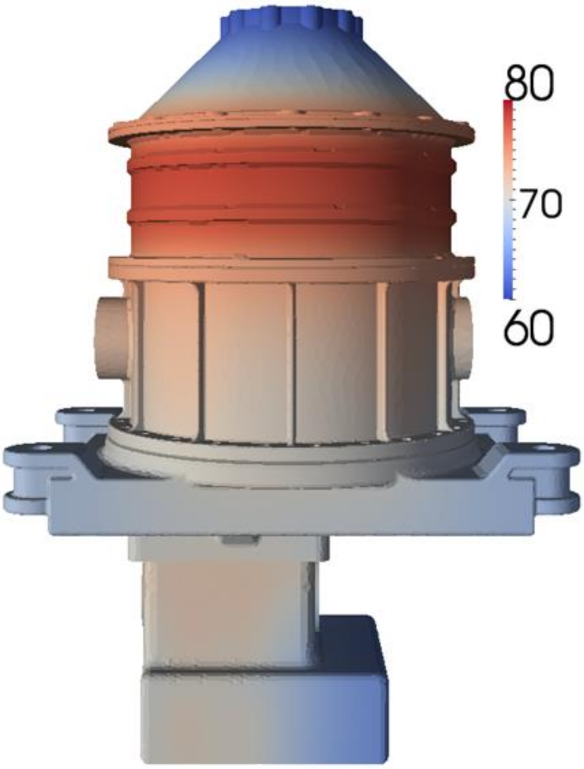


Fig. 7 *- Example 3D Results, Housings*

The correlation between the 1D and 3D thermal models is good, confirming that both the 3D profile is suitable for use in the next design stage and that the 1D model is suitable for transient simulations.

Table 1 *- A comparison of 1D and 3D thermal values*

Component	Temperature [°C]	
	1D	**3D**
Main Housing	71.5	68-75
Tail Rotor Drive Shaft		
Shaft	89	88.5-89
Web	89.7	90
Teeth	89.7	90
Stage 1 Planetary Carrier	94	87-94
Offset Gear Shaft		
Shaft	86.7	85-90
Bevel Web	83.2	83
Bevel Teeth	83.2	83
Cylindrical Web	91.3	91
Cylindrical Teeth	91.3	92

3.3 Coupled Thermal-Static Analysis

Temperature profiles for the simulated conservative mission profile were imported into FE components within *Romax**DESIGNER*** (see Fig. 8 and the corresponding Fig. 9d). An analysis of the gearbox with respect to the identified variables has been undertaken and the results provided to demonstrate the effects of including a temperature profile.

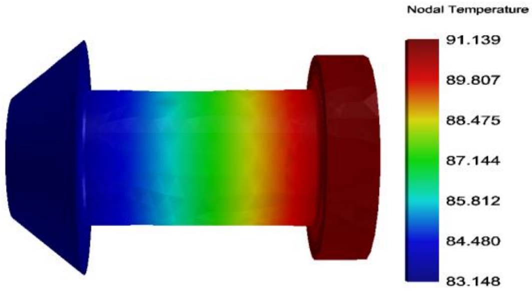
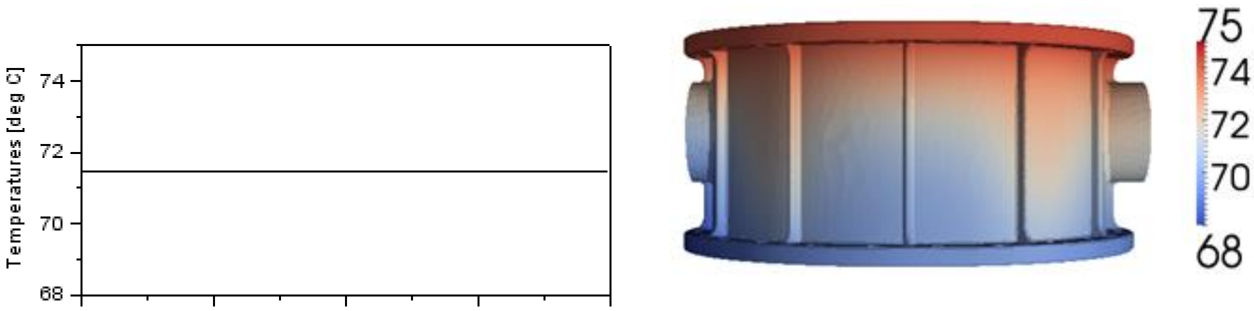


Fig. 8 *- Temperature Profiles in Romax**DESIGNER***

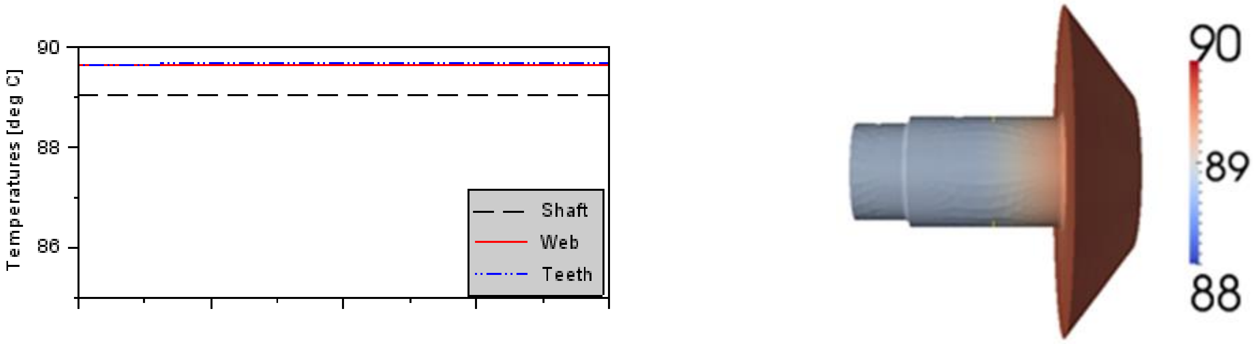
The mesh misalignment of the bevel gear sets with and without a temperature profile are shown in Fig. 10. The average radial (ΔE) change is 61.26µm (40.2%) while the average axial (ΔX) change is 90.46µm (601% or 88.4% with the 2 low value ΔX_P data points removed which skew the results). These are both significant changes, with three of the values even changing sign.

The contact face load factors, $K_{H\beta}$, for the planets in the 1st planetary stage are provided in Fig. 11. The average change is 0.54 (37.6%) and if this is restricted to just the sun and planet mesh this increases to 0.79 (55.9%). Given the slope of the contact stress-cycles curve for limited life in ISO 6336 for case carburised gears, a change of $K_{H\beta}$, and hence stress, of 37.6% changes the predicted life of the gear by a factor of 8 and a change of 55.9% changes the predicted life by a factor of no less than 19. It is clear that such influences must not be ignored.

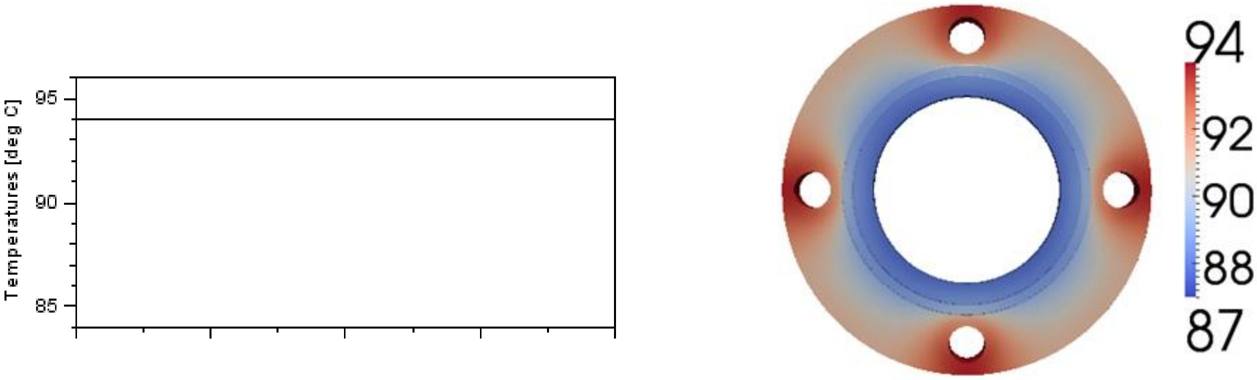
Flash scuffing safety factors are displayed in Fig. 12 for the 1st Planetary stage (noting that this is a simulation of a conservative mission profile). As with $K_{H\beta}$, the mesh between the planets and sun have adverse changes with the scuffing factor reducing by ~45% in 2 cases. The simulation still passes, however, the corresponding values for the 1st iteration of the aggressive mission profile reduce these safety factors below 1 in the case of planet 1 and to <1.1 for planet 3.



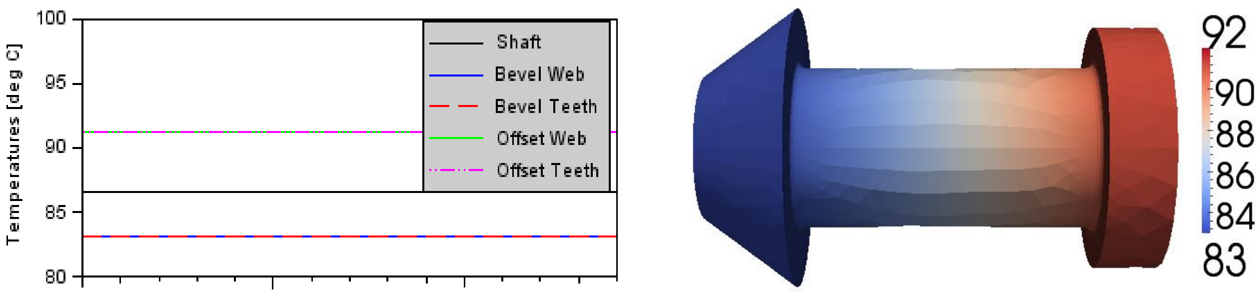
a. *Main Housing*



b. *Tail Rotor Take Off Shaft*



c. *Stage 1 Planetary Carrier*



d. *Offset Gear Shaft*

Fig. 9 *– A comparison of 1D and 3D temperatures on various gearbox components*

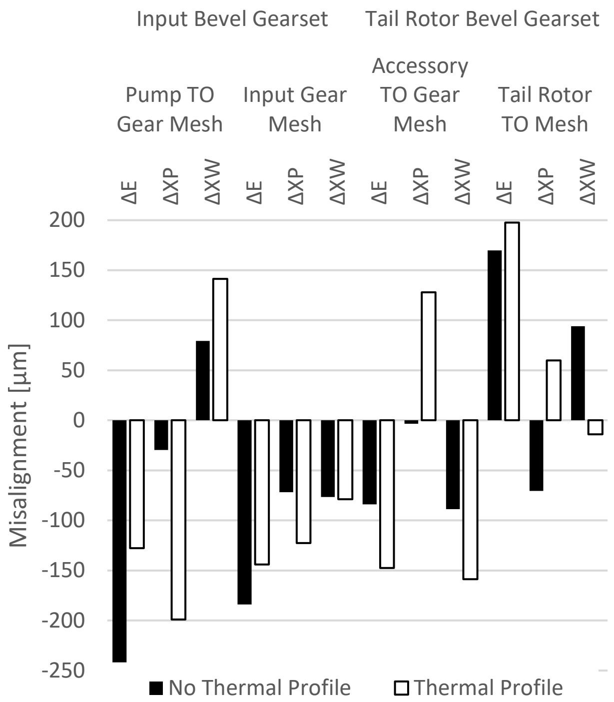


***Fig. 10** – A comparison of $F_{\beta X}$ with and without a temperature profile for the two 3-way bevel gearsets. ΔE is the radial separation and ΔX is the axial separation of the pinion and wheel.*

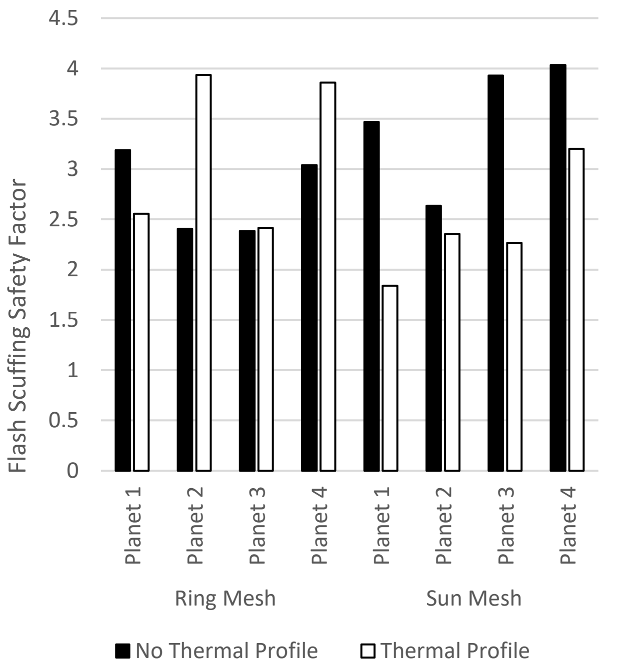


***Fig. 12** - A comparison of the Flash Scuffing Safety factor with and without temperature profiles for the 1st Planetary Stage*

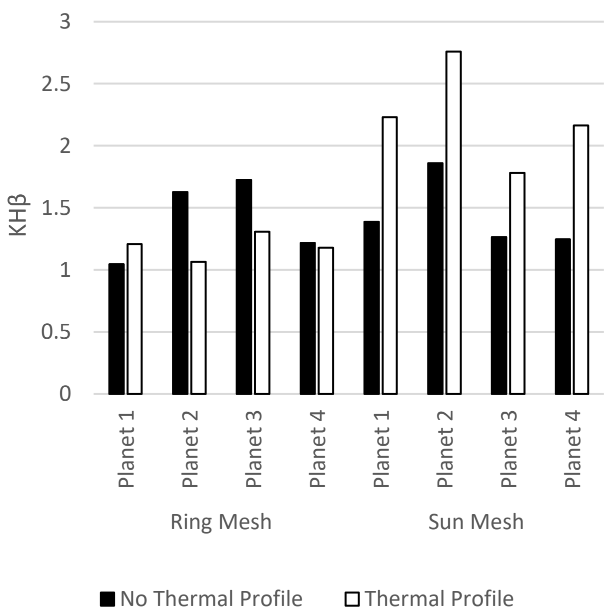


***Fig. 11** – A comparison of $K_{H\beta}$ with and without temperature profiles for the 1st Planetary Stage*

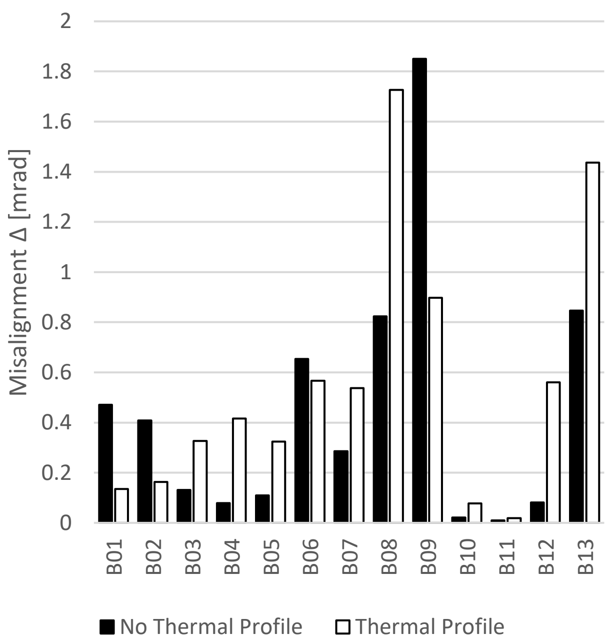


***Fig. 13** - A comparison of bearing misalignment with and without a temperature profile*

A comparison of bearing misalignments is displayed in Fig. 13. It should be noted that no optimisation of bearing selection has been undertaken for this case study. To put these values into context, bearing capacity is based upon a general assumption that misalignment will not exceed ~ 0.5mrad [3] [4], where this is exceeded it has an adverse effect on the life. The average change is 0.36mrad, with a maximum positive change of 0.9mrad, almost double the threshold value. The percentage change in the ISO equivalent loads are displayed in Fig. 14, the average absolute change is 49.5%.

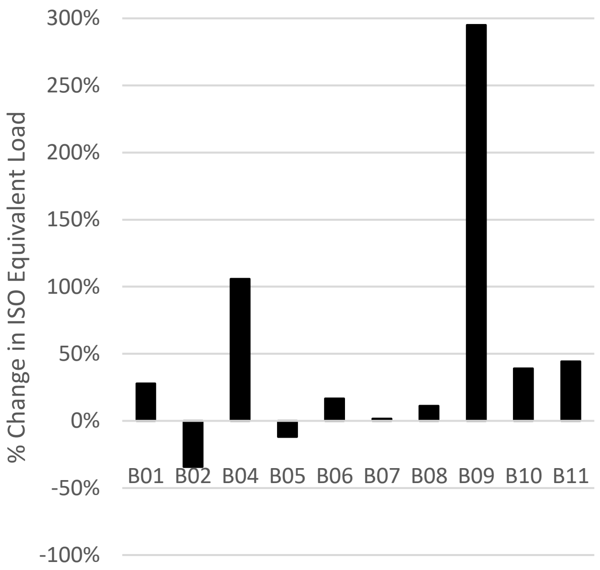


Fig. 14 *- Percentage change in the ISO equivalent load (with and without a temperature profile) for the gearbox bearings, note B03, B12 and B13 are excluded.*

Bearings B03, B12 and B13 have been removed from **Fig. 14** as they would render the graph illegible. Their values are displayed in Table 2.

Table 2 *- ISO equivalent loads of the 3 exceptional results*

Bearing	ISO Equivalent Load [N] Without Temperature Profile	ISO Equivalent Load [N] With Temperature Profile
B03	0.25	4602.3
B12	141	19859.1
B13	2299.6	40545.3

These bearings were previously lightly loaded. However, with the inclusion of a temperature profile the load used in the L_{10m} life value has been significantly increased. In the case of B13, which serves the tail rotor take off shaft the L_{10m} life value has decreased from to 925820 hours to 3.32 hours as a result of the increased misalignment and loading. As previously stated the bearing design has not been optimised for this case study; however the scale of decrease does aid to demonstrate the requirement for a temperature profile.

3.4 Concept Design Decisions from the 1D & 3D model

The primary purpose for the development of the 1D and 3D thermal models was to create a coupled solution for thermal and structural analysis of a gearbox. However, during development additional uses pertaining to thermal modelling became apparent. These capabilities are demonstrated through the second aggressive mission profile simulation. The aggressive mission profile should be considered as running to steady state at a power level beyond the existing design capabilities and could for example be used to investigate the feasibility of uprating the gearbox. The first iteration of results (no changes in design) are displayed in Fig. 15.

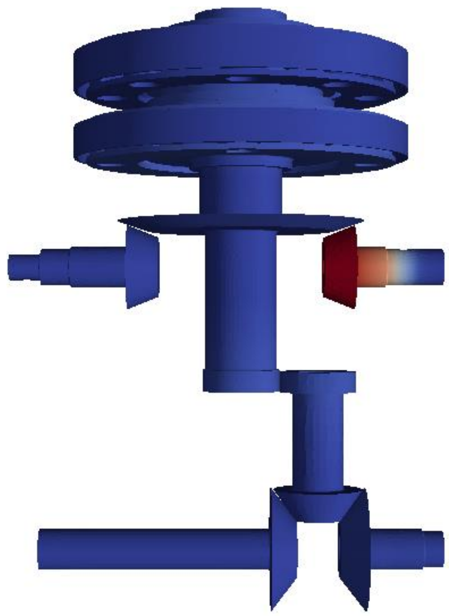


Fig. 15 *- Aggressive Mission Profile – Iteration 'a'*

Running at this increased power a majority of the gearbox is at a normal operating temperature with the exception of the input pinion, evident by the hotspot. Alterations to the design are required to reduce this temperature to an acceptable level, these iterations are recorded in Table 3.

Table 3 - *Details of the input shaft design iterations*

Iteration	Design Changes
a	No Design changes
b	Iteration a + Improved surface finish of the input bevel gear
c	Iteration b + Additional cooling to the web at both the toe and heel of the input bevel gear (Fig. 16).
d	Iteration c + Further improvement to the surface finish of the gear (e.g. super finishing)
e	Iteration d + Internal cooling of the input shaft bore

Normalised results from the 1D model are displayed in Fig. 17, as would be expected the lump mass temperatures decrease through the design iterations. Two aspects of the component are modified, its surface finish and its cooling regime. An improvement in surface finish improves the gear's efficiency, while an inclusion of cooling to the web and internally are simulated through an increased convection area. As the limits of surface finishes are reached (e.g. super finishing) and unused cooling area diminishes, the temperature curve starts to flatten out. At this stage an investigation of the 3D results is useful to provide a better interpretation of the situation.

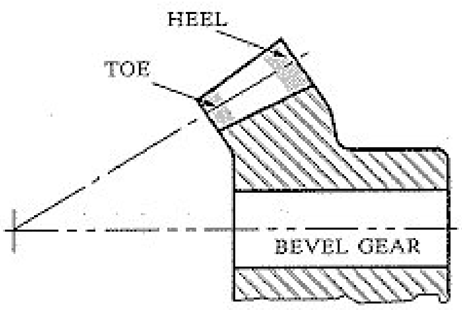


Fig. 16 *- Bevel gear terminology*

The 3D results are shown in Fig. 18. As would be expected an improvement in surface finish retains a similar temperature profile with reduced peak values, while the increased cooling areas are visible in Fig. 18c and e through a change in the temperature profile to the face and web of the gear. By iteration 'e' the temperature has been significantly decreased, if it were still in excess of the ideal operating temperature then additional local input variables such as those listed in Table 4 could be considered.

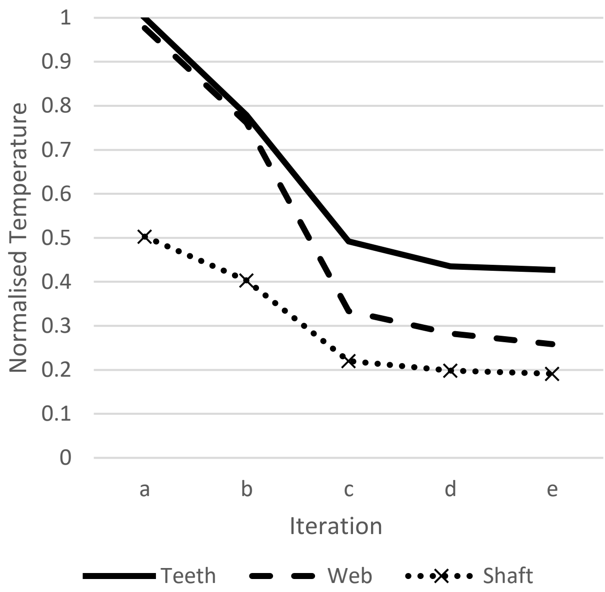


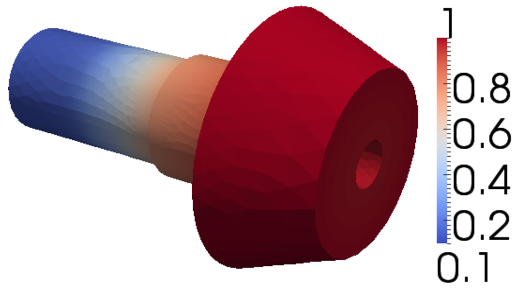
Fig. 17 *- Input shaft relative temperatures throughout the design iterations*

Table 4 *– Potential input variables and their implementation*

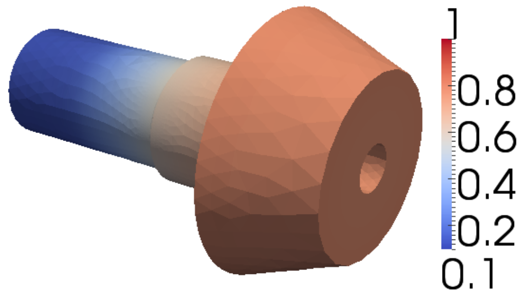
Design Variables	Implementation
Surface Finishes	Increased gear efficiency
Additional Spray Cooling Nozzles	Increased convection areas
Shrouds	Reduction of windage losses
Increased Cooling Flow	An increase of mass flow rate
Materials	Different thermal properties
Component Sizes	Increased mass and 1D dimensions

In addition to these localised design considerations the model can be expanded to global gearbox systems including:

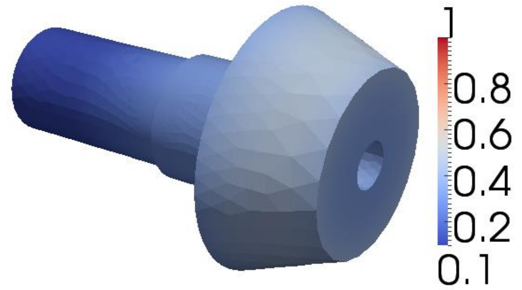
- sizing of pumps and oil cooling
- identification and sizing of auxiliary coolant reservoirs
- split torque pathways



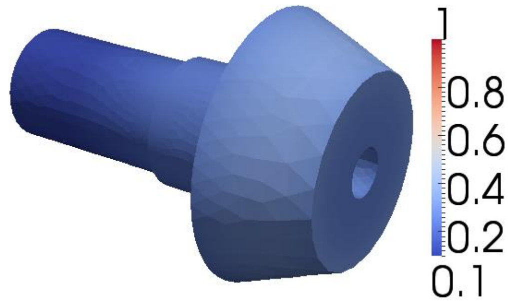
a.



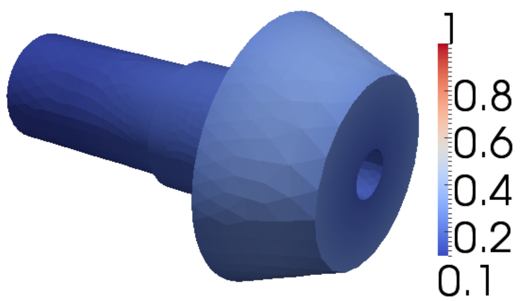
b.



c.



d.



e.

Fig. 18 *– 3D results of the input shaft design iterations*

And further expanded to simulate missions and test profiles including

- CS29.927 Loss of lubricant test
- Investigating uprating the gearbox

This is all possible at an early stage in the design process.

4. CONCLUSION

Rotorcraft transmissions operate in a challenging environment where both the thermal and structural effects need to be considered from the outset. The limitations of the current CFD packages inhibit this coupled analysis due to their complexity and runtimes. This paper presents two tools with runtimes measured in minutes and hours as opposed to weeks and months:

1. A 3D structural-thermal steady state model demonstrating the requirement for this coupled analysis.
2. A 1D steady state and transient thermal model that provides powerful information at concept stage.

The accuracy of the 1D model is dependent on the input variables, in particular the heat transfer coefficient. When correct, the model becomes a powerful tool with the ability to undertake both steady state and transient simulations which can be used to investigate both local and global design considerations at the concept stage.

In particular, the transient and steady state aspects concerning the coolant system are likely to be of interest. Output values have been shown to match those of 3D FE for smaller components. When the size of the component or the temperature gradient increase, the 1D value is comparable to a mean average value.

The 3D model provides an improved visual understanding of the temperature profiles. However its fundamental purpose is to provide this profile for a structural-thermal analysis program such as *Romax**DESIGNER***.

The importance of a coupled model has been demonstrated with key variables directly linked to component life analysed both with and without a temperature gradient. The magnitude of change for the variables within the case study was typically between 40-90%. The implications on a design optimised without considering the thermal effects is clear, with the average gear life reducing by a factor of 8. And further emphasised by the tail rotor take off bearing, where the L_{10m} rating dropped from a value nearly two orders of magnitude above a typical rotorcraft transmission fatigue life down to 3 hours.

5. Bibliography

[1] EASA, "CS-29 Large Rotorcraft," EASA, 2016.

[2] D. Mba, S. Place, H. Rashid and R. Keong, "Helicopter Main Gearbox Loss of Oil Performance," European Aviation Safety Agency, 2011.

[3] The Timken Company, Timken Engineering Manual, 2011.

[4] SKF, "Misalignment," SKF, [Online]. Available: http://www.skf.com/uk/products/bearings-units-housings/ball-bearings/principles/bearing-basics/basic-selection-criteria/misalignment/index.html. [Accessed 29 June 2017].