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POWER SPEED REDUCTION UNITS FOR GENERAL AVIATION PART 5: HOUSING/CASING OPTIMIZED DESIGN FOR PROPELLER-DRIVEN AIRCRAFTS AND HELICOPTERS

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ABSTRACT

The purpose of this paper is to focus on the design of casings for aircrafts and helicopters PSRU (Power Speed Reduction Unit). This paper introduces a rigorous and practical design procedure for gearboxes. The work starts from the experience of the Authors in Formula 1 and Aircraft gearboxes. For certification, safety and durability reasons, aircraft and helicopter gearboxes did not have the same development rate of the Formula 1 counterparts. A brief history of Formula 1 PRSU/gearboxes forms the first part of this paper. This part includes also an introduction to material and manufacturing technologies. Then the modal analysis of the gearbox is discussed, along with the influence of tolerances and operating temperatures. Then cooling is briefly introduced. The gear train is focus of the PSRU. Proper gear meshing in any load and environmental condition is the main requirement of the PSRU. Unfortunately gears and transmissions are the source of many forcing time-varying forces that act on the housing. This forces not only vary with tolerances, temperatures and loads, but also with wear. Therefore, a comparison of the natural frequency of the housing, the torsional critical speed of rotor system and the flexural critical speeds of each of the shafts with the exciting frequency clearly may be used to qualify the gearbox housing. A finite element modelling of the gearbox housing can be carried out to obtain its natural frequency, stress distribution and forced response. Unfortunately, the excitation frequencies vary with tolerances and operating conditions. Furthermore, in aircraft PRSUs, it is common practice to vary the transmission ratio (and the gears) in the same housing. Therefore, the housing should dampen a fairly large number of exciting frequencies. This result is obtained by curved surfaces, ribbing and double walling. This approach also reduces the noise produced by the transmission. In fact, noise radiated by a gearbox is directly related to the vibratory level of its housing. Therefore, an additional aim of this study is to analyze the transfer mechanisms between the static transmission error of a gear pair and the dynamic responses of gear and housing of a gearbox. Aerospace and Formula 1 transmissions have many similarities, with Aerospace engineers working on both sides and importing solution. The great advantage of Formula 1 gearboxes was (until the unlucky Regulations of 2010) that it was extremely easy to make experiments. This is due to the fact that all Formula 1 cars are prototypes with test pilots on board. Therefore, this paper will take advantage of the knowledge achieved in Formula 1 to transfer these data to aerospace PSRU and transmissions [1-2].

Keywords: PSRU, aircraft, helicopter, housing, gear drive, transmission.

INTRODUCTION

In PSRU (Power Speed Reduction Unit) drive system, gears reduce/increase speed, change the direction of drive, and split/combine torque paths. A gearbox housing has certain functional requirements based on which it was designed through the years. Historically the gear design is a typical inside out procedure. Gear transmission is designed along with the bearings, then the housing is tailored to this assembly. Fixtures, appendixes, accessories, oil tank, cooling, lubrication are also considered. Aerodynamics may also require a few adaptations. Therefore, the gear housing first design is based on functional requirements. The modal analysis is then performed. The primary optimization objectives are the natural frequencies of the configuration. It is necessary that they lie sufficiently away from the exciting frequencies that arise from the gearbox; the exciting frequencies are mainly the lowest torsional and flexural of the rotating shafts and the gear meshing frequencies. In this analysis analytical methodology used for bearing dynamic simulation and load transfer is fundamental. However, the first aim of the housing is obviously to house the internals and provide positive locations for the bearings and shafts which are capable of maintaining the shaft centers within prescribed tolerances when running under full load. In fact, large contact surfaces between teeth are essential with the high tangential loads and their resultant separating forces. The casings must also be stiff enough to contain distortion. In fact, joint faces have fixing centers capable of coping with the loads involved without oil leaks occurring.

Finally, as the complete gearbox design is available, with the stressing checks of the individual components finalized, a manufacturing technique for each part should be detailed. An important part of this process involves the decision on the material to be used for each component. Aerospace and Formula 1 transmission have many similarities, with Aerospace engineers working on both sides and importing solution. The great advantage of Formula 1 gearboxes was (until the unlucky Regulations of 2010) that it was possible to make experiments. This is due to the fact that all Formula 1 cars are experimental prototypes with test pilots on board. Therefore, this paper will take advantage of the knowledge achieved in Formula

1 to transfer these data to aerospace PSRU and transmissions [3-6].

Manufacturing technology

Formula one and aerospace transmissions are the most advanced in the world, with an unparallel combination of performance, weight and power density. A key part of the PSRU is the casing, as it forms not only a housing for the gearbox internals but also acts as an integral part of the vehicle structure. Sand cast or forged magnesium alloys have long been a favored material for gearbox casings, thanks to its mechanical properties. In fact, magnesium alloys are about 35% lighter than aluminum alloys, and certain alloys can be heat-treated to UTS (Ultimate Tensile Strength) values up to 300 MPa, making them attractive because of their high strength to weight ratio. Unfortunately, the stiffness of magnesium is generally only about 60% of aluminum alloys. Therefore, casings being switched from aluminum to magnesium will need larger cross-sections and section moduli to achieve the same stiffness as the aluminum part, but can still achieve a weight saving of about 25%. Of course this fact is true for sand cast parts. A comparison of sand cast magnesium housings and die cast aluminum ones shows that the latter are slightly lighter. Therefore, in recent years the emergence of new casting techniques for aluminum has toppled magnesium from its position as the material of choice. In the past, the major factor governing the weight of aluminum gearbox casings was the casting wall thickness limited to about 6 mm. Traditional sand casting methods did not allow thin wall sections to be cast reliably, thus limiting weight savings irrespective of the actual component strength. However, vacuum assisted and investment-rapid-tooling (lost-wax) casting methods that allow for very thin wall sections are now available. Therefore, engineers can now design casings that are marginally lighter than a magnesium counterpart, while still retaining the same strength characteristics. As a result of these new casting methods, the benefits of aluminum alloy casings now outweigh those of a magnesium unit with comparable production costs. As a point of note, a manufacturer of an aluminum alloy gearbox used formula one racing (year 1996) quotes the weight of the unit as being "in the region of 28kg against the 30kg of the same magnesium alloy unit". The development of new concepts for gearbox housings started on 1993. In that period sand cast magnesium case were premium choices in Formula 1. The longitudinal gearbox, with the gears behind the final drive, had problems of porosity and variable wall thickness, due to problems of positioning the cores accurately during the casting process. Therefore, the structurally optimum wall thickness was accordingly increased. The situation was worsened by the magnesium alloy well-known reduction in stiffness above 100°C. This technology resulted in a gearbox case mass slightly below 29 kg. A new housing was manufactured with TIG welded 2.5mm 300M steel plates. These new gearbox, with were CNC-milled down to 1.2mm wall thickness and ribs, proved to have approximately the same final mass of the magnesium alloy case. The advantage of the steel solution was a larger stiffness and a greater strength. The manufacturing of the welded gearbox was extremely critical and time-consuming with accurately timing welding steps to avoid distortion and oil leaking. The passage to titanium alloy welded gearbox (1995) obtained a reduction in mass of around 40% (17kg).

It was not until the Minardi team began experimenting with a cast titanium unit in 2000 that a viable option for titanium alloy casing construction was found. The use of a rapid tooling casting process, with the extensive use of rapid prototyped “wax” patterns (created using a SLS Selective Laser Sintering process), meant that not only did the production cost of the transmission fall considerably, but the entire design could be optimized to a greater degree. This is due to the complex patterns that could be manufactured through SLS, allowing for internal structures that were not possible with traditional casting methods. Process time is slightly longer however, and cannot compete with traditional casting rates. The cast titanium housing was around 20 kg with considerably improved stiffness and reliability when compared with the titanium alloy welded solution. The titanium alloy increased hot stiffness, also meant better gear reliability and greater efficiency. As the gears do not displace significantly under load, they can be ground to truer profiles for greater efficiency and durability. With a completely different approach, in 1995, Ferrari manufactured a titanium gear case bolted to a CFRP bell-housing/oil tank, onto which the suspension units were mounted and onto which acted suspension loads.

In 1998 Arrows manufactured a complete CFRP gearbox with bonded titanium-alloy transverse bulkheads to carry bearings. The CFRP main case weighed about 9kg. A welcome advantage of the CFRP gearbox was that the oil ran cooler - 100 °C instead of

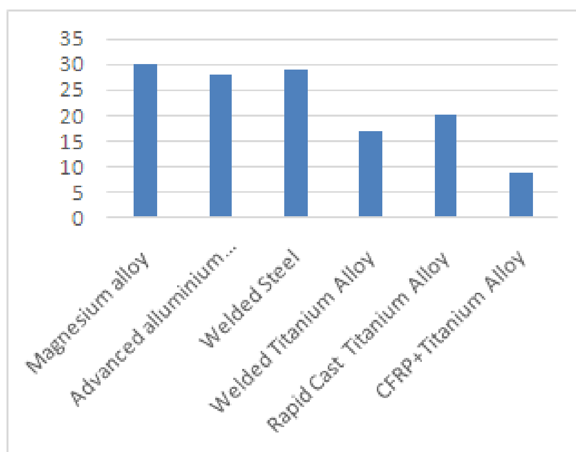

Figure-1. Mass of F1 gearbox with different manufacturing technologies.

the more usual 130 °C. This is due to the insulating properties of the carbon/bismalamide casing that reduced the heat transfer from the extremely hot rear part of the Formula one car to the inside of the housing.

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Figure-1 summarized the gearbox mass with the different manufacturing technology [7-9].

Casing design

The WWII German School used 3D curved surfaces for castings. The DB 605 engine is a good example of this approach (Figure-2).

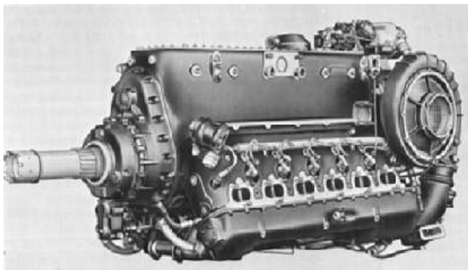

Figure-2. 3D surfaces (WWII DB 605).

The curved surfaces are a good way to compromise between manufacturing simplicity and good overall stiffness and damping. Another classical approach is external ribbing. This approach has two variants: large ribs (Figure-3) or small ribs (Figure-4). The small rib approach often combines the large ribs with the curved surfaces. The double walling is typical of liquid cooled integral crankcases but is also used in gearboxes to reduce noise emissions and to use two casings of different material bolted together (Figure-5).

Usually, the housing is tailored on the transmission with a minimal gap (usually 5mm) to reduce air pumping. A breathing system and the cooling arrangement are added. Mounting points and accessories are included and the resulting shape is reinforced with ribs or double walls to obtain the necessary stiffness and natural frequencies.

In more details, the design of housing continues with the following stages: design of a single/multiple stages speed reduction transmission; determination of the forcing (exciting) frequencies; computation of the critical shaft speeds of the shafts; static analysis of the housing; determination of the natural frequency of the configuration to make sure that the exciting frequencies are sufficiently away from the housing natural frequencies. Finally, a simulation of the manufacturing process is performed and the mould is designed [10-13].

Figure-3. Housing with large ribbing.

Lubrication methods

The preliminary design and layout for the gearbox internals includes the internal gear pack with the location bearings and shafts. At this point the remaining vital internal part of the gearbox is the lubrication system. This system should always start from the beginning of the design, as this one area is traditionally the cause of many problems both at the design stages and when the gearbox is in use. A lubrication system should be effective and efficient at the high speeds and heavy gear tooth loadings that can be experienced at take-off. The lubrication system can be designed in varying ways.

Most low power (<200HP@6500rpm) manual automotive transmissions and aircraft PRSUs rely on splash lubrication. The problem with splash lubrication is that it is speed dependent. There are centrifugal effects, hydrodynamic effects and effects from the gears working as pumps. As the gearbox is run through its revolution range, the oil jets will switch over and move around. This is particularly critical for roller and ball bearings with small projections that would intercept the main oil jet. The main problem of this gearbox is the energy dissipated by dipping and the necessity to keep the oil level under strict tolerances. For this reason a tight visual-and-automatic control of oil level is strictly necessary. A proper shaped oil tank that allows significant oil consumption without altering significantly the oil level is also mandatory. Another approach is to install a simple recirculating system within a wet sump gearbox. This system has no external fittings, except for the filler plug, the oil breather and the drain. The most common system incorporates an external oil cooler and oil collector tank. Therefore the gearbox has a dry sump, a scavenge and a pressure oil pipe, an oil breather and a drain plug. The oil filler is installed onto the oil tank, which could also incorporate the oil breather/pressurizer.

The lubricating oil is circulated around the system by means of a positive displacement pump that operates with reasonable efficiency. The level of this efficiency is extremely dependent upon the design of the pump inlet and outlet. Therefore, it is essential to provide a smooth-flowing system along with the pump body.

In a simple recirculating system, the pump retrieves the oil from a sump in the bottom of the gearbox casing. The sump should be designed so that it collects the oil as it drains from the internal running gear to the bottom of the casing. It is also essential that the rotating gears in the gearbox do not dip into the collected oil, as this would create heat in the oil with the relevant reduction in efficiency. The oil from the sump is scavenged through a gallery, cast and machined in the gearbox casing. At some point in this gallery, between the sump and the oil pump, a filter should be included. This is a cylindrical frame covered with fine mesh gauze, with a rod-type magnet in the center of the tube. The oil from the pump is delivered through cast and machined galleries in the gearbox casings. The method usually adopted to lubricate the bearings is an oil jet machined into one of the drilled oil galleries. Oil jets are also used to lubricate and cool the drive gears. Research has shown that, oil sprayed onto the

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rotating gears before the point of mesh, creates wear, heat and lowers efficiency. The gear tooth is worn by the wedge of oil sprayed into the mesh point as the gears rotate, having a hydraulic effect on the tooth surfaces in the meshing zone. The continual spray action as the gears rotate erodes the tooth surfaces. This phenomenon can be easily be confused with the initiation of pitting and can ultimately lead to tooth failure. Wedge spray erosion generates heat, with a resultant loss of efficiency. On the contrary, as the lubricating oil is sprayed onto the rotating gears after the point of mesh, tests showed that a film of oil is retained on the gear tooth face which is thick enough to prevent metal-to-metal contact. The majority of the oil is used as a coolant to remove the heat created by the gear meshing under load. A quick removal of heat reduces the amount of wear on the tooth surfaces reduced with no erosion effect. The system that includes the oil tank and radiator works in a similar way except that it is more complex. In this case the oil pump has two stages, the first stage is a scavenge pump, generally 2 or 3 times larger than the second stage, which scavenges the mixture oil and air from the gearbox sump, through the filter to an collector, from where it is piped into the radiator and then into the oil tank. The second stage of the pump draws the oil from the oil reservoir into the oil gallery system via another external fitting, the gallery system being identical to that of the recirculating system. The choice of the lubrication system depends mainly on cooling requirements.

Vibrations

The main source vibratory and acoustical behavior of gearboxes is the Static Transmission Error under load (STE). STE comes from the meshing frequency due to variable elastic deflections of gear teeth meshing under load, teeth-gears manufacturing errors and shaft misalignments. Under operating conditions, STE generates dynamic forces and moments transmitted through bearings to the housing.

High dynamic mesh loads may correspond to the excitation of some critical (natural) housing modes amplifying the high potential energy stored by the meshing stiffness. Also critical shaft speeds can be excited by the time-average meshing stiffness and bearings stiffnesses. Finally, a dispersion of critical speeds takes place due to the variability of STE, resulting from geometry and assembly errors authorized by designers (tolerances) and operating conditions.

In this context, the Taguchi's method is valuable to estimate in a very simple way the statistical moments of a function of multiple random variables of known probability density. In this case, statistical moments are estimated from Gauss-quadrature numerical integration.

Figure-4. Micro-ribbing on the casting of an automotive engine.

The Taguchi's methods allow calculating the response function for the samples that are necessary to take into account the eventual nonlinearity of the response function. The number of samples is equal to the product of the number of levels chosen for each factor. The basic assumption is that the intercorrelation between variables is low. Usually 3 levels for each factor are needed to treat the three retained random parameters. Therefore, the principal advantage of this method is the ease of its numerical implementation and its short computing time.

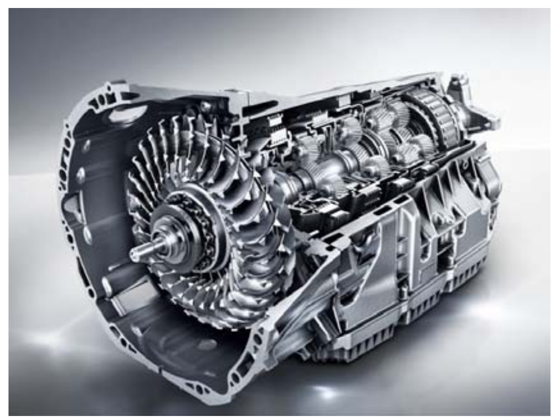

Figure-5. Double walled automotive transmission.

Probability density function cannot be calculated by the modified Taguchi's method. However, from the estimation of the influence of the uncertain parameters computed samples is achieved by using variance analysis. The STE is obtained by a set of successive rotational positions of the driving wheel. The ideal tooth contact lines contained in the action plane are discretized in a some number of slices. The STE and the meshing stiffness are periodic functions in relation with the cyclic variation of the number of in contact teeth pair. The evaluation of these functions is time-consuming and requires many calculation steps. Therefore, the final equation has the general form (1).

$$M\ddot{X} + C\dot{X} + \left(K + k(t)D\right)X = E \tag{1}$$

In this equation, X is the vibratory response of the meshed housing nodes, M and K are the mass and stiffness matrices provided by the finite element method with the addition of the bearings stiffness matrices, k(t) is the periodic meshing stiffness, D is a coupling matrix of the two toothed wheels and E is the generalized force

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vector which results from the STE. Matrix C represents damping which is usually very small and can be neglected in most cases. The vibratory response is then computed with standard numerical time integration. Generally, K and D are non-linear due to bearings. In fact the dynamic properties of any rolling bearing varies during operation, especially the stiffness which mostly depends on rotating speed and loads applied. Therefore, the rolling bearing varying stiffness is modelled by bearing dynamic matrixes. For example, the radial stiffness of single-row ball bearing with radial contact and without radial clearance has the form of equation (2).

$$K_r = 1.19 z^{\frac{2}{3}} D^{\frac{1}{3}} F^{\frac{1}{3}} \tag{2}$$

Where K_r is the radial stiffness [N/mm], D is ball diameter [mm], F is the external radial load [N] and z is the number of rolling elements. Equation (2) is very simplified since it does not include clearance and rotational speed. For oblique contact ball and roller bearing stiffness depends mainly of rotational speed and play (wear). Play depends also on assembly and assembly tolerances. Therefore, even with the Taguchi method vibration calculations are complicated. Even natural modes analysis (3) that comes from the eigenvalues of equation (1) is time consuming and should be repeated for various speeds, and load conditions.

$$\left(-M\omega^2 + \left(K + \bar{k}D\right)\right)X = 0 \tag{3}$$

Therefore, the design method practically used is very simplified. In this simplified methods the designer checks that forcing frequencies are well away from housing natural ones.

Forcing frequencies determination

The vibration analysis of housing is based on the fact that only very specific elements in the rotating parts of any machine will produce forces in the casings that will cause vibration at specific frequencies. One of the most important of the forcing frequencies is the rotational velocity of the shafts, and it arises from the fact that any rotor will always have a certain amount of residual imbalance. This imparts a radial centripetal force on the bearings, causing the structure to vibrate at the fundamental frequency and to its harmonics; usually the second and fourth orders are significant. Another forcing frequency comes from the “bearing tones”, which are characteristic of each bearing geometry, are forces generated by defects in the races and rolling elements of the bearing itself. Roller and ball bearings inner race rotating and our race stationary the natural frequencies can be calculated with equations (4), (5), (6) and (7).

$$FTF = \frac{S}{2}\left(1 - \frac{B_d}{P_d}\cos\varphi\right) \tag{4}$$

$$BPFI = \frac{zS}{2}\left(1 + \frac{B_d}{P_d}\cos\varphi\right) \tag{5}$$

$$BPFO = \frac{z}{2}\left(1 - \frac{B_d}{P_d}\cos\varphi\right) \tag{6}$$

$$BSF = \frac{P_d S}{2B_d}\left(1 - \left(\frac{B_d}{P_d}\right)^2 \cos^2\varphi\right) \tag{7}$$

Where S is the difference of rotational speed between inner and outer ring (RPS). FTF is the fundamental train frequency. BPFI is the fundamental frequency roll frequency pass of the inner ring. BPFO is the fundamental frequency roll frequency pass of the outer ring. BSF is rolling element spin frequency. B_d is the rolling element nominal diameter. P_d is the pitch diameter and φ is the true contact angle for oblique contact bearing. φ depends on velocity, preload and load. All the calculated frequencies [Hz] have a tolerance depending on manufacturing quality. Usually gears are AGMA quality 12 and bearings have tolerances between H/h5 and H/h6.

Housing natural frequency evaluation

It is then check that the natural frequencies of the housing are away from the forcing ones. In this case errors are common in the FEA (Finite Element Analysis) evaluation. Usually FEA overestimates frequencies. The normal FEA calculation errors [%] for an aluminum alloy die casting are summarized in Table-1.

Table-1. FEA aluminum alloy die casting errors.

Calculated frequency %	100
True frequency %	92
Standard deviation	5%

Table-1 shows that normally true frequencies lie within 97= (92+5) % and 87=(92-5)% of the calculated values.

This is due to the material model, the calculation method and the manufacturing technology. It is necessary to avoid that natural housing frequencies lie close to the forcing one. It is also good rule that the gearbox casing do not have important natural modes in the most important frequency range for the human ear [250-2000Hz]. Since noise amplitude depends on the vibrating surface extension, ribs should contain this dimension. That is a reason for micro-ribbing. This technique is often applied on surfaces that proved to be critical from acoustic measurements.

Static evaluation and bearing misalignments

Figure-6 shows a atypical arrangement of a large PRSU with titanium alloy micro-ribbed casings. Large

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PRSUs are to be avoided at all costs since dimensions increase displacements with a cubic law. The same happens for the mass. In this case the requirement of propeller shaft position was mandatory.

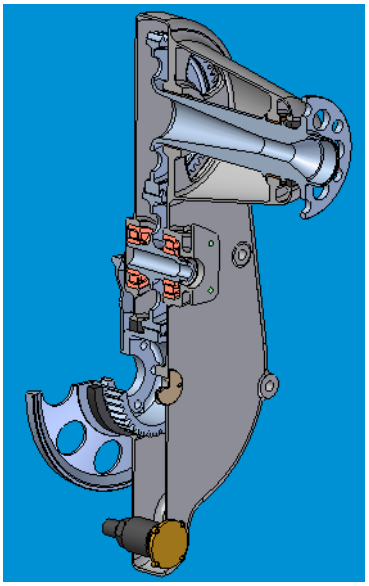

Figure-6. PRSU example.

Figure-6 shows a PSRU with a 3-shafts arrangement. In this solution the intermediate shaft holds the propeller governor pump. The lubrication is forced and uses the pump of the piston engine. An oil filter and a collector is positioned at the bottom of the housing. Oil is taken from the engine sump and the engine pump directs a jet onto the upper gear mesh at exit station.

The spur gear solution avoids axial loads. Therefore the engine pinion is supported by the crankshaft journal bearings. The spur gear also compensates the axial displacement of the crankshaft.

The casing is arranged in two halves with a vertical separation (not shown in Figure-6). In order to save weight the propeller shaft has journal bearings. For this reason a maximum static displacement (misalignment) of 4μm is allowed between journal bearing centers at full load. This result is obtained through a titanium alloy casting. Microgrooves are present in the structure but not modeled in this CAD model that is used for FEA. In the FEA model the microgrooves are taken into account by modifying the material data. Walls are 3 mm thick (overall) with ribs of 3mm. The static simulation results are depicted in Figure-7.

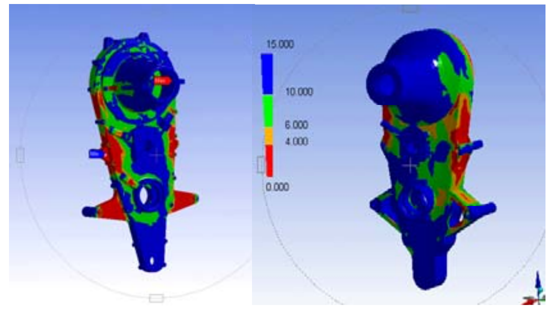

Figure-7. Static SF for a PSRU.

Figure-7 shows the typical problem that large parts of the housing are unloaded (blue color) with SF (Safety Factor) larger than 15. Unfortunately these parts are resonating, so thickness cannot be reduced for dynamic (vibrations) and manufacturing reasons. Misalignment on the propeller shaft bearings are controlled by small displacements on the bearing supports.

CONCLUSIONS

This paper introduces a practical design procedure for gearboxes. For certification, safety and durability reasons, aircraft and helicopter gearboxes did not have the same development rate of the Formula 1 counterparts. Aerospace and Formula 1 transmissions have many similarities, with Aerospace engineers working on both sides and importing solution. The great advantage of Formula 1 gearboxes was (until the unlucky Regulations of 2010) that it was extremely easy to make experiments. This is due to the fact that all Formula 1 cars are prototypes with test pilots on board. Therefore, this paper will take advantage of the knowledge achieved in Formula 1 to transfer these data to aerospace PSRU and transmissions. Therefore, a brief history of Formula 1 PRSU/gearboxes forms the first part of this paper. This part includes also an introduction to material and manufacturing technologies. Then cooling is briefly introduced. Unfortunately gears and transmissions are the source of many forcing time-varying forces that act on the housing. This forces not only vary with tolerances, temperatures and loads, but also with wear. Therefore, a comparison of the natural frequency of the housing and the forcing components of the transmission is the most feasible approach. A finite element modelling of the gearbox housing can be carried out to obtain its natural frequency. FEA (Finite Element Analysis) errors and the variation of the excitation frequencies with tolerances and operating conditions are also discussed. Furthermore, in aircraft PRSUs, it is common practice to vary the transmission ratio (and the gears) in the same housing. Therefore, the housing should dampen a fairly large number of exciting frequencies. This result is obtained by curved surfaces, ribbing and double walling. This approach also reduces the noise produced by the transmission. In fact, noise radiated by a gearbox is directly related to the vibratory level of its housing. Therefore, an additional aim of this approach is to dampen the frequencies in the most sensible part of the spectrum

(250-2000Hz). Finally, a simulation example is briefly introduced.

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