USE OF THE INTEGRATED FLEXPIN BEARING FOR IMPROVING THE PERFORMANCE OF EPICYCLICAL GEAR SYSTEMS

Gerald Fox and Eric Jallat
Abstract

Epicyclical gear systems have typically been equipped with straddle-mounted planetary idlers having pins supported on the input and output sides of the carrier. Torsional wind-up of the carrier, position accuracy of the pins, machining tolerances of the planetary gear system components and bearings clearances can all contribute to a poor load sharing among the planetary idlers as well as misaligned gear contacts in the deflected state. Use of the double-cantilevered flexible pin concept to achieve better load sharing and gear contact patterns among a multiplicity of planetary idlers, has been used to improve reliability in advanced gear drives for many years. The consequence of this practice is to build a compliant epicyclical system that improves power density in the gear length direction because the probability of achieving a properly centered gear contact is increased. The Integrated Flexpin Bearing, the subject of this paper, is capable of achieving additional power density in the gear diameter direction through integration of bearing components, gearing and shafting. This paper presents one designer’s approach to optimizing an Integrated Flexpin Bearing to improve the reliability of an epicyclical gearbox.
Introduction - The Design Challenge Of An Epicyclical Gearing System

An epicyclical gearing system is particularly well suited for achieving a high-reduction ratio in a relatively small, power dense package. A typical straddle mounted planetary idler is shown in Fig. 1.

This blend of features has made it a popular choice for the designers and, consequently, it has been incorporated into countless types of equipment, including automobile transmissions, off-highway equipment final drives, wind turbine gearboxes and cement mill crusher drives, to name a few.

As with any type of power transmission system, the engineer is faced with many analytical challenges during the design phase to ensure that a highly reliable power train is achieved. In the case of an epicyclical gearing system, this challenge is made particularly difficult due to the complex interaction of revolving and rotating components as they transmit power.

In the traditional epicyclical gearing system, where the distance between planetary gear centerlines is specified by the design to be within a fixed range, it is widely recognized that the load sharing is not equal among the planetary gear meshes. Similarly, the stress distribution at each mesh point contains variability. Load sharing and stress distribution at each mesh point are heavily influenced by the global design configuration, backlash tolerance, component design tolerances, manufacturing accuracy, component deflection and thermal distortion. See Fig. 2 which shows in exaggerated form that contact is made at the mesh points of planet a-1 before any contact is made at the mesh points of the other planets. In a rigid system, this condition imposes unbalanced loading among the planetary idlers.

The traditional approach to accounting for this variability is to apply safety factors during the design phase. Factors accounting for load distribution among planets, the torsional wind up of the planetary carrier assembly, the contact pressure exerted on the gear faces and many more are published by AGMA and other sources and are used frequently in today’s design process with good success. Additionally, very advanced computer programs linking finite element analysis with gear design algorithms and with the safety factors just mentioned are commercially available and are used by most gear design centers in conjunction with their own proprietary design practices.
But, for just about all equipment types, economics dictate the need for increased power density and improved reliability. A common approach is to attempt to build in more planets, thereby reducing forces and stresses at each mesh point. But, as planets are added, so is uncertainty about just how much power each planet is transmitting. Therefore, the safety factors (for example, K gamma) are applied to maintain conservatism of design that defeats some of the effort to achieve more power density.

Alternatively, designers have applied a number of novel designs with various levels of success to build epicyclical gearing systems that help distribute loading among planets more evenly, thereby increasing power density. In general, such improvements employ components in the gear train that are elastically compliant and are intended to compensate for clearance variations without imparting any negative operating characteristics. Some include:

1. Flexible ring gears have been applied, but the effectiveness of this approach is not universal because the radial deflections of the ring gear are not enough to compensate for the clearance (backlash) variations present at the various mesh points.

2. Floating ring gear system (used in some off-highway applications)

3. Floating sun gear

4. Floating planet carrier

5. Double helical gear with floating members

6. Floating planetary gear, also called flexible pin or abbreviated to flexpin. [1]

The remainder of this paper will focus around advancement of number 6 in the form of a new product called the Integrated Flexpin Bearing (to be referred to from this point on as the IFB).

**What is the Integrated Flexpin Bearing?**

The flexible pin design mentioned on line 8 above was created in the mid-1960s by British inventor Ray Hicks. It is based on achieving an equal load distribution among planets by anchoring them onto a planetary carrier in a torsionally compliant fashion. Instead of fixing the angular positions of the planet gears, the flexible pins were designed so that they could deflect independently in a circumferential direction, which ultimately helps equalize the force distribution among the planets while transmitting torque at various levels. This feature will be referred to henceforth in this paper as torsional compliancy.

Torsional compliancy was achieved by applying a double-cantilever beam design that is illustrated in Fig. 3. Simply stated, when the two tangential forces are applied to the flexpin gear, the angular deflection caused by the bending of the pin cantilevered from a carrier wall can be offset in the opposite direction by the angular deflection caused by bending of the sleeve cantilevered from the other end of the pin. If sections of the pin and sleeve are carefully designed with that goal in mind, the deflection at each gear contact will follow a circumferential translation meaning the axis of the gear contact will not tip side to side from angular positioning inaccuracy nor lead from torsional wind up of the carrier.

**FIGURE 3 – DEFLECTION CHARACTERISTIC OF THE FLEXIBLE PIN**
Flexible pins have been designed into various types of equipment and the designs have typically included assembly of separable components including gears, pins, mounting sleeves, backing plates, capscrews and various type of rolling element bearing races and bushings. See Fig. 4 for a typical mounting arrangement.

FIGURE 4 – TYPICAL FLEXIBLE PIN DESIGN

Such a design achieves the objective of creating a torsionally compliant system as illustrated in Fig. 5. Additionally, since gears are less prone to be tipped off axis because the single sided planetary carrier can no longer wind up, it can be argued that gear contacts have a much higher probability of remaining centered at all meshes. It follows then that the flexible pin permits the designer to specify narrower gears and still avoid stress concentrations at the edges of the face. Power density is therefore improved in the axial direction.

FIGURE 5 – COMPLIANT PLANET CARRIER SYSTEM USING FLEXIBLE PINS

A substantial improvement can be realized if one takes advantage of modern bearing technology and advances the entire design to the next level, which is full integration of the gear with the outer races of the bearings, and full integration of the sleeve with the inner races of the bearings. This advancement is the Integrated Flex-pin Bearing (IFB) and is illustrated in Fig. 6. This novel approach to design and construction of the flexible pin arrangement provides increased opportunity to add power density to an epicyclical gear drive in the axial and radial directions. The beam strength of the sleeve and gear are both increased from the integration of the bearing components allowing for downsizing especially in the radial direction.

FIGURE 6 – CROSS-SECTIONAL VIEW OF THE IFB

Other features of the IFB include:

1. Elimination of sub-assembling the flexible pin arrangement
2. Elimination of many components for weight savings and cost reduction
3. There are no tight-fitted outer races that can still process inside the gear creating wear, debris and extra bearing clearance
(4) Unitized construction permits elimination of the bearing retainer and insertion of more rolling elements, adding to bearing capacity and bearing life.

(5) Very precise control of the mounted bearing clearance range at the bearing factory for maximizing bearing life, gear positioning and consistency of deflection between adjacent IFB’s. Note: A combination of press fitting and welding the stationary adjustment rib at its required position on the inner race sleeve has been developed and thoroughly tested to ensure adequate holding force and durability at loads significantly in excess of the maximum applied loads.

(6) Higher assurance of equally distributed loading to each bearing row

**Design of the Integrated Flexpin Bearing**

There are many factors influencing the global deflection characteristics of the IFB including:

(1) Applied loading

(2) Gear profile and section

(3) Number, size and location of rolling elements

(4) Internal bearing clearance

(5) Sleeve section

(6) Length and diameter of the of each cantilever

(7) Pin length and diameter

(8) Pin groove geometry

(9) Planetary carrier deflection

To specify an IFB design, the authors’ company has developed an advanced design model that combines a parametric 2D and 3D finite element analysis coupled with a Fourier series representation of the varying conditions of load distribution on the raceways. This allows computational time to be significantly compressed. Using this proprietary modeling technique (also accounts for deflection of the carrier), the radial deflection of the IFB and gear misalignment can now be thoroughly assessed and accurately predicted.

A typical IFB design is illustrated in Fig. 7. U.S. and International patents have been applied for.

**FIGURE 7 – INTEGRATED FLEXPIN BEARING ASSEMBLY**

Minimum pin section (A) and gear section (B) dictate to a large degree the overall size of the IFB. See previous Fig. 6.

Other critical inputs include:

(1) Load spectrum

(2) Speed spectrum

(3) Minimum required bearing life

(4) Gear dimensions or tooth volume available and gear ratio

(5) Lubrication condition (temperature, viscosity, surface finishes)

(6) IFB component material

(7) Mechanical characteristics of the planetary carrier

(8) Minimum radial displacement required under the nominal load

(9) Maximum permissible gear misalignment
FEA modeling yields outputs such as those illustrated in Fig. 8 and Fig. 9. These allow for close study of load distribution, stress concentration, and the global IFB deflection characteristics that ultimately influence the gear and bearing lives.

Parametric FEA modeling facilitates iteration so that design optimization of the IFB may be achieved in a reasonably short period of time. Assuming all input is available, the modeling and analysis may be achieved in approximately one day’s work. Numerous dimensional variables are investigated to ascertain influence on the IFB deflection characteristics. They are all interrelated and will impact deflection to varying degrees among different gearbox designs. Two charts are offered in Figures 10 and 11 showing the influences of groove diameter, pin diameter, groove length and pin length on gear misalignment.

Similar graphs could be shown for many other variables such as groove location, gear section, sleeve diameter, bearing internal geometry, and bearing setting. All these and more are explored during the optimization process.

Properly designed, the desired radial displacement of the IFB results in circumferential translation of the gear contact centerline with nil misalignment. It should remain somewhat proportional to the torque applied to the system. Analytical results are graphed in Fig. 12. Static testing conducted subsequently indicates a modest departure from perfect linearity, especially at lighter torque, because of the complexity in modeling the tight fitted connection between the flexible pin and the carrier wall. Nev-
ertheless, the error is small enough to be within normally required design accuracy.

**FIGURE 12 – PERCENT LOADING VS. RADIAL DISPLACEMENT OF THE IFB**

**Epicyclical Gearing System Design**

Perhaps if there is a negative aspect of adapting an IFB design, it is that pins are cantilevered from a single-sided carrier. It follows that compared with pins supported at both ends in a typical planetary carrier design, the available shear area per flexible pin is divided by 2. Additionally, the bending moment increases because of the cantilevered design. It follows, therefore, that when using the IFB, more planets must often be added in order to maintain acceptable stress and deflection levels.

Adding more planets has positive and negative effects. On the positive side, adding planets should reduce the load per planet. On the negative side, adding planets on the same carrier pitch diameter could necessitate reducing the planet gear diameter so that they do not interfere with one another, therefore reducing the maximum gear ratio possible when using lesser pins and larger diameter planet gears. This is a valid observation, but it can be compensated for in many instances by shifting part of the required ratio to other stages of the gear train.

Another argument on the negative side that may be refuted is that adding flexpins will worsen load distribution on a percentage basis (K gamma increases or worsens). This is often the case using the rigid, fixed distance straddle carrier design; however, the torsional compliancy afforded by the IFB design in large part compensates for this. Test results in Fig. 13 show how the load-sharing factor K gamma is reduced using flexible pins [2][3]. For example, using 7 pins, K gamma published by AGMA equals 1.47 while K gamma using the flexpin equals 1.20.

**FIGURE 13 – LOAD SHARING FACTOR COMPARISON**

Another objection commonly raised is that by using more planets, weight is increased. This does not have to be the case. If the system using the IFB is designed and manufactured properly, narrower gears may be employed because gear misalignment will be minimized, keeping the Hertzian contact well centered on the gear face and away from the edges. In standard terms, KH beta is improved and easier to predict.

Epicyclical gearing systems can be designed in a number of single- and double-stage configurations, depending on space available and the ratio desired. One arrangement that has been applied to wind turbine gearboxes is shown in Fig. 14. Its ratio (N2/N1) happened to be 12.2/1 [4].

**FIGURE 14 – TYPICAL EPICYCLICAL GEARING SYSTEM INCORPORATING THE IFB**
Preloaded Tapered Roller Bearing – Preferred Choice for the IFB

The preloaded tapered roller bearing imparts superior performance characteristics to an IFB design, and is commonly regarded as the preferred bearing choice. It can be shown analytically that a preloaded tapered roller bearing will offer advantages over other bearing types possessing larger amounts of radial clearance, especially when the IFB is required to operate with misaligned gear contacts.

In performing the usual bearing calculation, a common assumption is that the gear contact remains centered. In fact, some percentage of units from a population of a specific epicyclical gearing design will operate with off-centered gear contacts. Off-centered gear contacts can be created by a number of variables including:

1. Inadequate lead compensation for varying amounts of torsional wind up of a 2-sided carrier
2. An improperly designed flexpin deflection characteristic
3. Misaligned carrier pin holes
4. Component wear
5. Excess bearing clearance
6. Improperly designed or machined gear profiles

When gear contacts are off-center, the pressure centroid of the gear contact is shifted side to side, creating an uneven load distribution on the bearings. Compounding this is that when one or both gear contacts shift, the separating forces are no longer directly opposed, and an overturning moment is created. This new distribution of loading now affects not only the bearing life calculation but also the angle of gear misalignment.

Figure 15 has been generated using proprietary software developed by the authors’ company. It illustrates, for one set of gearing, how stress along the gear face is shifted at varying degrees of misalignment. It also shows how the centroid shifts off-center as well.

In performing the usual bearing calculation, a common assumption is that the gear contact remains centered. In fact, some percentage of units from a population of a specific epicyclical gearing design will operate with off-centered gear contacts. Off-centered gear contacts can be created by a number of variables including:

1. Inadequate lead compensation for varying amounts of torsional wind up of a 2-sided carrier
2. An improperly designed flexpin deflection characteristic
3. Misaligned carrier pin holes
4. Component wear
5. Excess bearing clearance
6. Improperly designed or machined gear profiles

When gear contacts are off-center, the pressure centroid of the gear contact is shifted side to side, creating an uneven load distribution on the bearings. Compounding this is that when one or both gear contacts shift, the separating forces are no longer directly opposed, and an overturning moment is created. This new distribution of loading now affects not only the bearing life calculation but also the angle of gear misalignment.

Figure 15 has been generated using proprietary software developed by the authors’ company. It illustrates,
As stated above, gear contacts are oftentimes not perfectly centered, as shown in Fig. 17A. Figure 17B is a plot showing comparison again of the various bearing arrangements when the centroid of the gear contact is shifted 60% to one side or the other. Under this condition, both bearings translate radially and because loading is shifted more onto one of the bearing rows, a misalignment angle is introduced in the “Y” direction. The preloaded tapered roller bearing arrangement begins without clearance so that the resulting displacements are minimized, thereby reducing the misalignment angle. The lightly preloaded tapered roller bearing arrangement shows slight improvement compared with a cylindrical roller bearing having normal mounted clearance, and substantial improvement compared with one set with higher clearance.

The two gear contacts of a planetary idler may also be offset on opposite sides. In other words, the gear contact between planet and ring gear may be offset left of the theoretical centerline to varying degrees, while the gear contact between planet and sun gear may be offset to the right by varying degrees. When this condition occurs, the separating forces will no longer oppose and will induce an overturning moment. Figure 18A depicts this scenario when the mesh centroid between the planet and ring gear is 60% of half the face width in one direction, while the mesh centroid between the planet and sun gear is the same distance in the opposite direction. Because the tangential forces in our example are offset equally and remain in the same direction, radial deflection in the “y” direction remains unaffected. However, the moment loading created by the offset separating forces creates radial displace-
ment, tilting and gear face misalignment in the “ZZ” plane of the paper. In Fig. 18B it is shown that the lightly pre-loaded tapered roller bearing reduces these movements and misalignments.

**FIGURE 18A – PLANET WITH OVERTURNING MOMENT**

**FIGURE 18B – RADIAL DISPLACEMENT WITH OVERTURNING MOMENT**

Unlike the flexible pins shown in figure 4, the IFB is a unitized bearing package that has the unique feature of having its internal clearance pre-set at the bearing factory to very exacting specifications. This ensures that the mounted clearance variation among IFB assemblies will be miniscule, which means that the IFB deflection characteristics will be extremely consistent within a large population of product. Planet-to-planet backlash variation will be minimized. Obviously, this will have an equal impact on backlash variation within a population of gear drives, as well.

**Timken Static Testing**

Static testing was designed and conducted at the authors’ company to confirm the accuracy of the analytical models. Figure 19 shows the test bed on which all experiments were conducted. An IFB was cantilevered from a massive and rigidly supported carrier wall, keeping in mind that it was necessary to be able to discern between the deflection of the IFB and deflection of the carrier wall as data was collected and evaluated. This was accomplished quite successfully so that values derived for comparison are accurate.

**FIGURE 19 – STATIC DEFLECTION TEST STAND**

Displacements were measured at numerous critical locations on the IFB and carrier wall with LVDT probes, as well as dial indicators. Strain gauges were installed on the pin fillet radius and between the planet carrier and pin to confirm our FEA predictions.

Figure 20 shows comparisons between measured and calculated results. The percentage of error is within reasonable accuracy for design purposes verifying that the design tool may now be used with confidence.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Measured / calculated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear misalignment</td>
<td>+22 %</td>
</tr>
<tr>
<td>Radial deflection</td>
<td>+5 %</td>
</tr>
<tr>
<td>Stress level</td>
<td>+10 %</td>
</tr>
<tr>
<td>Linear behaviour</td>
<td>+15 %</td>
</tr>
<tr>
<td>KHb</td>
<td>+6% (see note below)</td>
</tr>
</tbody>
</table>

**FIGURE 20 – STATIC TEST RESULTS**
Note: Improvement value considers planet gear and planet carrier deflection only. Ring and sun gear profiles are assumed to have 0 error. KHb values to be confirmed with additional dynamic testing.

Dynamic Testing

Dynamic testing was performed in conjunction with Maag AG on a newly developed gearbox for wind turbines. This gearbox was designed in response to the Wind Turbine Industry’s expressed need for a gearbox that possesses sufficient compliancy to withstand a highly fluctuating duty cycle, common in turbulent wind conditions.

Among other objectives during the dynamic testing, assessing the gear contact patterns became of vital importance, especially since the Maag gearbox contains seven IFBs on the high-torque input carrier and five more on the lower-torque carrier.

FIGURE 21 shows the test setup. Two gearboxes were connected in-line between their low-speed, high-torque shafts. Torque was applied at varying levels into the gearboxes with opposing hydraulic cylinders connected to the gearbox mounting pins. Rotation was provided by an electric motor connected to one of the sun gear output shafts. The power necessary for the electric motor to drive the test apparatus was equivalent to the total power losses.

Various parameters were measured. Figure 22 shows test results for load sharing among planets (K gamma factor). Two important observations are immediately apparent. First, while the usual published values of K gamma are constant, K gamma using the IFB decreases as load increases. Stated another way, load distribution among planets improves as load increases when using the IFB.

Secondly, and most importantly, K gamma is substantially less than normally published values. In the case shown here, at 100% loading, K gamma is reduced 27%. Stated another way, the design load for each planet can be reduced by the same amount, creating an opportunity for downsizing or improving reliability.

FIGURE 22 – LOAD SHARING AMONG PLANETS

IFB Design Guidelines for Wind Turbine Gearboxes

(1) Spur gears are much preferred over helical gearing. Even with shallow helix angles, the overturning moments produced by helical gearing thrust forces will not permit an IFB to be designed with optimal compliance.

(2) Various epicyclical gearing system configurations are possible.

(3) Gear faces and rolling elements can be coated with special surface treatments to improve life and efficiency.
(4) Max desirable IFB raceway contact stress for the weighted operating torque condition should be less than 1450 Mpa.

(5) Required bearing life should be 175,000 hours L-10 using advanced bearing analysis tools.

(6) Gear wall section height at the center of the gear (between the two bearing rows) shall be 3 times the gear module.

(7) The radial deflection (circumferential translation) under the nominal loading condition should be at least equal or greater than the nominal gear backlash.

(8) For wind turbine gearbox applications, the IFB must maintain design integrity at loading 3 times the maximum operating load and 2 times the extreme load condition.

Conclusion

Use of the Integrated Flexpin Bearing is a natural progression in the field of epicyclical gearing system design. It affords the designer an effective approach to improving load distribution and reliability.

Perhaps equally important, new analytical methods and tools have now been developed to ensure that the complex workings of the IFB can be determined with sufficient accuracy and within a reasonable amount of time for use in the design phase. Results of prototype testing should therefore be more predictable, reducing overall design cycle time.

These tools have been validated with testing reported by the authors’ company and Maag AG. As more designs are created, more knowledge will be accumulated and tools will continue to be refined.

Acknowledgments

Urs V. Giger, Gear Consultant, Zurich, Switzerland

Maag AG associates for their extensive and enthusiastic cooperation throughout this development project.

Raymond J. Hicks (Gearing Consultant and retired owner of Compact Orbital Gears) for inventing the flexible pin design and for his expert guidance.

The Timken Company’s associates whose collaborative efforts have made the development of the IFB possible.

References


Bearings • Steel •
Precision Components • Lubrication •
Seals • Remanufacture and Repair •
Industrial Services

www.timken.com

Timken® is the registered trademark of
The Timken Company.

© 2008 The Timken Company
Printed in U.S.A.
03-05 Order No. 5867