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Dynamic and Meshing Analysis of Planetary Gears Assembly

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ABSTRACT: *A Machine consists of a power source and a power transmission system, which provides control application of the power. Often transmission refers simply to the gear box that uses gears and gears trains to provide speed and torque conversions from a rotating power source to another device The main aim of our project is to focus on the mechanical design and analysis on assembly of gears in Planetarium gear box. Analysis is also conducted by varying the materials for gears, i.e., Titanium Alloy and cast iron. Presently used materials for gears and gear shafts are Cast Iron. In this project to replace the materials with Titanium Alloy for reducing weight of the product decrease deformation and stress factor. Total Model and Static analysis is completely analyzed by considering weight reduction in the gear box. The design is done by using CATIA software. Modeling and assembly is done in CATIA. And following gear design is imported in ANSYS and under static structure analysis, behavior of gears is studied.*

I. INTRODUCTION

Planetary gear trains are one of the main subdivisions of the simple epicyclic gear train family. The epicyclical gear train family in general has a central “sun” gear which meshes with and is surrounded by planet gears. The outermost gear, the ring gear, meshes with each of the planet gears. The planet gears are held to a cage or carrier that fixes the planets in orbit relative to each other. Planetary gear is a widely used industrial product in mid-level precision industry, such as printing lathe, automation assembly, semiconductor equipment and automation system. Planetary gearing could increase torque and reduce load inertia while slowdown the speed. To compare with traditional gearbox, planetary gear has several advantages. One advantage is its unique combination of both compactness and outstanding power transmission efficiencies. A typical efficiency loss in a planetary gearbox arrangement is only 3% per stage. This type of efficiency ensures that a high proportion of the energy being input is transmitted through the gearbox, rather than being wasted on mechanical losses inside the gearbox. Another advantage of the planetary gearbox arrangement is load distribution. Because the load being transmitted is shared between multiple planets, torque capability is greatly increased. Greater load ability, as well as higher torque density is obtained with more planets in the system. The planetary gearbox arrangement also creates greater stability due to the even distribution of mass and increased rotational stiffness.

Based on so many advantages of planetary gear above, we did our 3D model of multiple layers of planetary gear to get the speed reduction. And by using COS- MOSMotion, we achieve the visual movement simulation of it. Also we build the solid model using FDM machine. Finally, we hope our product can be used in industry in the future.

II. PLANETARY GEAR GENERATION

Planetary gear generation

By using Solid Works software to build the 3D model of planetary gear, there are some things to do first, such as keeping a constant velocity ratio between two adjacent gear teeth, generating involute curve and set parameters of spur gear.

2.1 Constant Velocity Ratio

To get a constant velocity ratio, the common normal to the tooth profiles at the point of contact must always pass through a fixed point (the pitch point) on the line of centres. In figure 1, although the two profiles have different velocities at the contact point K, their velocities along N1N2 that is passing through the pitch point P are equal in both magnitude and direction. Otherwise the two tooth profiles would separate from each other. Therefore, we get the velocity ratio, which is equal to the inverse ratio of the diameters of these two tooth profiles. This is called fundamental law of gear-tooth action. For each two mating gearing tooth profiles, they should satisfy the fundamental law to get a constant velocity ratio.

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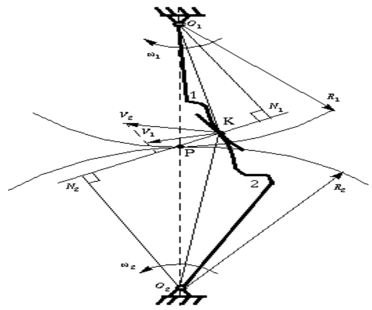


Fig 1. Constant Velocity Ratio

2.2 Generation of Involute Curve

The first step of generation a single tooth of the gear is to generating the involute curve. This involute curve is the path traced by a point on a line as the line rolls without slipping on the circumference of a circle. In Figure 2, let line MN roll in the counterclockwise direction on the circumference of a circle without slipping. When the line has reached the position M'N', its original point of tangent A has reached the position K, having traced the involute curve AK during the motion. As the motion continues, the point A will trace the involute curve AKC.

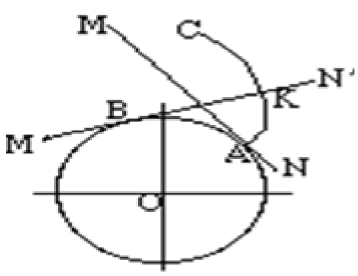


Fig 2 Involute Curve

2.3 Generation of Spur Gear

Usually we use the involute curve to generate a spur gear . The involute has important advantages -- it is easy to manufacture and the center distance between a pair of involute gears can be varied without changing the velocity ratio. Thus, we generate the involute curve to generate a spur gear

2.4 TerminologiesofSpur Gear

To generate a spur gear, some terminologies of the gear should be taken into consideration. The most important parameters in modeling we need to set the planetary gear are numbers of tooth, module, pitch circle diameter, pressure angle, basis circle diameter, addendum and dedendum. Figure 3 shows these terms. And Table 1 lists the standard tooth system for spur gears.

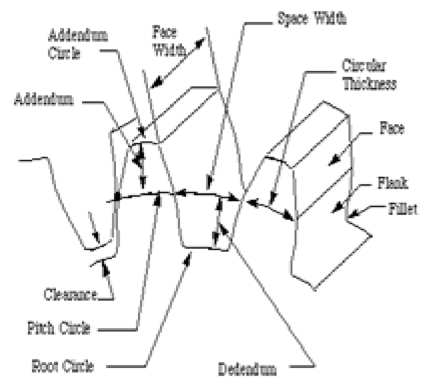
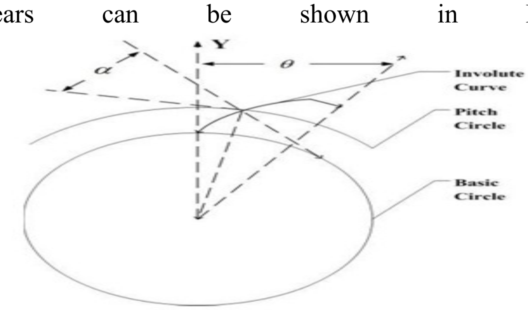


Fig 3. Terminologies Of Planetary Gear Box

circle and the dedendum circle to cut the two involute curves, and add the curve between them, extrude them to the certain width, then we have one tooth. Use the symmetry to arrange the tooth to the entire circle, and we have done with the gear model. The process of generating gears can be shown in Figure



4.

Fig 4: Generating Gears

2.5 Gears Generated

Based on the above method and parameter, we generated our planetary gear system, which contains one ring gear, two sun gears and four planet gears. The following figures show the generated gears

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- Ring gear

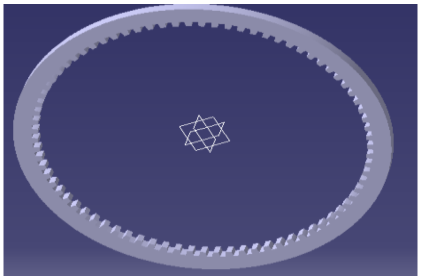

Fig 5: Ring gear

- Sungear

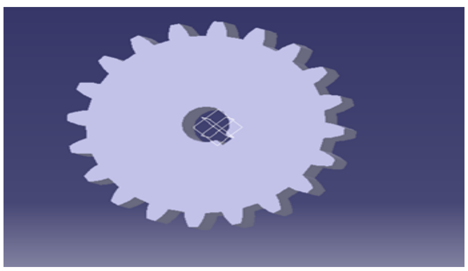

Fig 6: Sun gear

- Planet gear

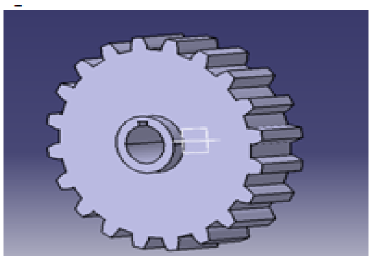

Fig 7: Planet gear

- **High Torque Density -- Compact Design**

An important requirement for automation applications is high torque capability in a compact and light package. This high torque density requirement (a high torque/volume or torque/weight ratio) is important for automation applications with changing high dynamic loads in order to avoid additional system inertia. Depending upon the number of planets, planetary systems distribute the transferred torque through multiple gear mesh points. This means a planetary gear with say three planets can transfer three times the torque of a similar sized fixed--axis "standard" spur gear system.

- **Fixed axis "standard"gearPlanetary" epicyclical "gear system**

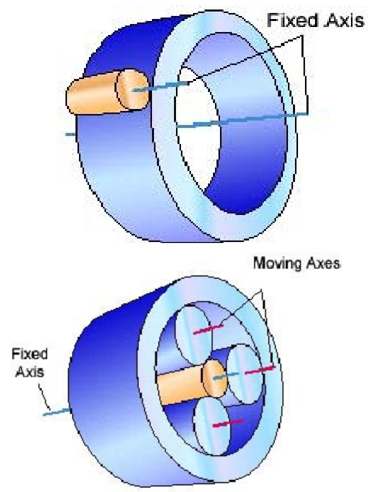

Fig 8 : Fixed axis "standard"gearPlanetary" epicyclical "gear system

- **otational Stiffness/Elasticity**

High rotational stiffness, or minimal elastic windup, is important for applications with elevated positioning accuracy and repeatability requirements; especially under fluctuating loading conditions. The load distribution unto multiple gear mesh points means the load is supported byContacts (whereN=number of planet gears) increasing the tensional stiffness of the gearbox by factor N. This means it considerably lowers the lost motion compared to a similar size standard gearbox; and this is what is desired.

2.6 "Run To Failure" Test

2.6.1 **Test Rig**

The test rig for Run To Failure (RTF) testing is a three-stage gearbox, and is a scaled down arrangement of the one used to drive a surge bin feed conveyor in Syncrude's oil sands mining field operations. The first stage is a bevel gearbox, the second stage is a planetary gearbox, and the third stage is another planetary gearbox. The input shaft is transferred from the bevel gearbox into the other two stages, where speed is reduced and the torque is increased. The gearbox reduction ratio for the first stage is 4, the second stage, is 6.429, and the third is 5.263. This gives a total reduction ratio of 135.338.the below figure shows layout of the stages

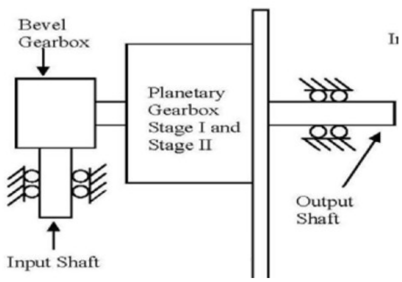

Fig 9 : Test Rig

Stage 1: Bevel Gearbox

This stage consists of a drive shaft gear and bevel gear. The gearbox is operated with an input speed of 1200rpm, fed in from the input shaft into the first stage.

Stage 2: Planetary Gearbox I

This stage planetary gear set with three planet gears, one sun gear, and one ring gear. The sun gear is driven while the ring gear is held fixed as the planets rotate around the sun. The input isprovided from Stage 1, and the output is fed into Stage 3.

Stage 3: Planetary Gearbox II

As in Stage 2, this gearbox stage is also comprised of a planetary gear set. Similarly, the sun gear is driven while the ring gear is held fixed as the planets rotate around the sun. However, unlikeStage 2, the planetary gear set has four planet gears. The input for this stage is provided fromStage 2. The original input speed of 1200 rpm at Stage 1, after undergoing reduction, results inan output speed of 8.867 rpm.For Run To Failure data collection, five accelerometer sensors were used to capture the vibration data [4]. Sensors (A1, A2, A3, A4, and A5) were placed on the housing of the gearbox, asillustratedon the schematic in Figure 2-2. Figure

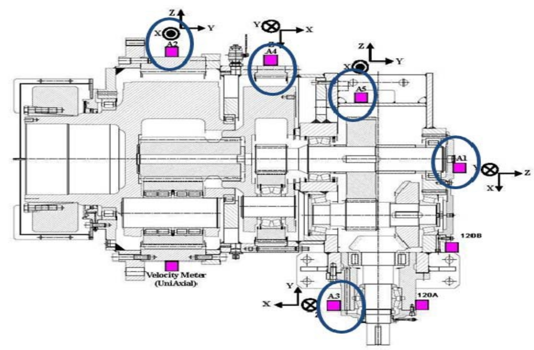

Fig10: Sensor Orientation from Schematic

Figure 10 shows the placement of the accelerometer sensors is located on the housing of thehousing of the actual test rig. The five accelerometers from the schematic are renamed to correspond with the sensor sensitivity type, stage, and mounting orientation

Input-h is placed on the housing of Stage 1, the bevel gearbox. Planetary1-v and HS Planetary1-v are placed on the housing of Stage 2, Planetary Gearbox I. Planetary2-v and HS Planetary2-v are placed on the housing of Stage 3, Planetary Gearbox II.

“HS” denotes high sensitivity and corresponds with the sensor specification of 5000mV/g.Otherwise, it is 100mV/g. “v” and “h” denote vertical and horizontal mounting respectively

During testing, the data was collected from all five sensors simultaneously while the gearbox was initially set to operate to the point of failure. In the actual test however, the gearbox was run 9 to a condition that was run just past a "sufficiently poor" state but before failure. The "sufficiently poor" state is what a field engineer, through visual inspection, would determine to be in a deteriorated state that would pose a large enough risk of failure , and would result in the gearbox being taken out of commission and replaced. In contrast, failure is defined as where the machine is completely inoperable.

2.7 Evolution of the Gearbox Condition

We are most interested in Stage 3, Planetary Gearbox II, as it is this stage that bears the most load and

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would likely fail first. As such, the data from collected from accelerometer planetary 2- v was selected for analysis. The gearbox was put into 500 hours of operation, made of 13 runs. Each run consisted of a continuous operation followed by an inspection. Five minutes (300s) of vibration data was collected once every hour. In inspection, the test rig is completely disassembled. The planetary gearbox is reduced to their individual gears. Each gear has been previously marked so that it can be insured that the gears are put back into their correct positions on reassembly. Figure 10 shows a planet gear from stage 3 that has been marked "P2".

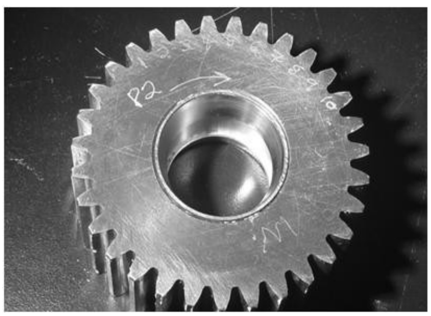

Fig11: Evolution of the Gearbox Condition

For each gear, a picture was taken of the gear teeth and side profiles for documentation [6] [7]. inspection for this research project is one that is much more thorough than one that would actually occur in practice on the field. Figures 2-5 and 2-6 shows a gear tooth from a planet gear after run 4 and run 12 respectively.

2.8 Model Identification

The purpose of this chapter is to identify an appropriate model for the RTF vibration data corresponding to the situation when the planetary gearbox is in a healthy state. Statistical time series models are used to fit data corresponding to the healthy, and in control, state of the system. These selected models are used in the development of fault detection schemes, which are presented in the next chapter, In the previous chapter, we looked at applying data pre-processing techniques to the original signal. In particular, we used time synchronous averaging to produce a TSA signal that exhibited discernable increases of energy with each run; evident in all files with the exception of file 9 which showed behaviour that was uncharacteristic of a gearbox in its deterioration state. (AR) and mixed (ARMA) statistical time series models are considered in this chapter. Autoregressive integrated moving average (ARIMA) model is also considered for nonstationary series. Note that the former are actually subclasses of the ARIMA model. Furthermore, no exogenous input was considered as the data was obtained with all other inputvariables held constant (speed and load).

2.9 Original Signal

In this section, we will proceed to illustrate the modelling of the OS signal. The OS signal is the raw data collected from the gearbox. Though it was first assumed that the data was stationary 30 and that AR and ARMA models could be applied, it was later found that this data is nonstationary. As such, modelling of the differenced data was attempted.Akaike information criterion (AIC) was used as the first approach for model identification for autoregressive and mixed models to identify the initial guess of the model orders. The residuals of these models were tested for normality using Kolmogorov-Smirnov test and were rejected. Since the model order was selected by AIC using a graphical approach, there may be an error due to the manner in which the minimum value of AIC was selected. For each type of model, a numerical approach was used to observe the effect of higher orders; to see whether a suitable model exists.

2.10 Application of Control Charts for Early Fault Detection

In Chapter 4, we were able to identify and fit a model for the TSA signal. We found that AR(300) was a better model than AR(100), based on having higher p-values from statistical tests for independence and normality.

Prototypical Gearbox Design

A prototypical gearbox design was developed for use as both an analytical and test model. Every component in the prototypical gearbox design was carefully reviewed to optimize weight, facilitate the use of high capacity materials, and to ensure that manufacturability is maintained. The design was also easily reconfigurable

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with a variety of materials and finishes which facilitated its usage in further developing the proposed technologies. For the purposes of this study, the gearbox was designed to use a ring gear with five planetary gear stages with a ~1500:1 overall gear ratio, 48.3-mm (1.9-in) housing diameter (excluding flanges), and a 78.7 mm (3.1 in) overall length (excluding shaft extensions). Each stage uses optimized

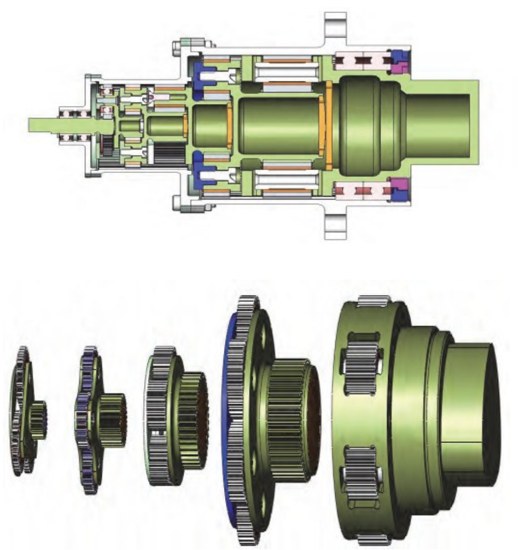

Fig. 11 - Prototypical gearbox design

Prototypical Gearbox Rated Capacity

The prototypical gearbox design was analysed per ANSI/AGMA 2001-C95. Allowable bending stress versus cycles and allowable contact stress versus cycles was extrapolated from the AGMA standard based on the material hardness used. Stress factors were selected to match typical space planetary gearbox applications as much as possible. Based upon the analysis, the following capacities were predicted (assuming a 1.5 Reliability Factor for 99.99% reliability):

- 8.6 N-m (342 in-lb) operating torque against the endurance limit (1E7 Cycles)
- 6.2 N-m (497 in-lb) operating torque at 3.33 RPM for 205.1 hr (Pitting Life = Bending Life)
- 9.5 N-m (792 in-lb) for short term operation of 4 hr or less
- 37.5 N-m (1217 in-lb) operating torque for momentary operation (bushing dynamic limit, ≤5 minutes rated gear operation at this load)
- 205 N-m (1812 in-lb) static torque limit for gears (2.0 Safety Factor)

- **est Evaluation Parameters**

Bushing Materials

For applications in which rolling element bearings are not suitable due to either capacity or packaging constraints, heritage planetary gearbox designs have traditionally utilized oil-impregnated SAE 841 bronze bushings. SAE 841 bushings have a Pressure-Velocity (PV) rating of 1.75 MPA-m/s (50 kpsi-ft/min) with a peak intermittent pressure of 28 MPa (4 kpsi). Often the bushing will be the limiting factor in the dynamic and/or static rating of the gear stage. An alternate material under consideration is Toughmet 3AT by Materion Brush, a spinodal/copper/nickel/tin alloy. Toughmet 3AT has a published PV rating of 4.6 to 9.0 MPa-m/s (132,000 to 260,000 psi-ft/min) depending upon the surface finish of mating parts. Maximum low-speed pressure was not provided and the testing that generated the PV ratings for the Toughmet 3AT material was performed at speeds of 1.5-2.0 m/s (300-400 ft/min) with additional lubricant added during the test. Space planetary gearboxes do not allow any lubrication to be added during life and often operate at speeds at or below 0.5 m/s (100 ft/min). It was decided to test both SAE 841 bronze and Toughmet 3AT bushings without providing additional lubrication and over a speed range closer to the typical space planetary applications. The purpose of the test was to determine if published PV ratings are valid and [3] also to show a direct comparison between the two materials

Gear and Journal Surface Treatments[5]

The literature for Toughmet AT recommended Metalife Industries MLP as a surface treatment that could provide

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lower friction, higher PV ratings, and longer life in lubricant starved conditions. Metalife MLP is a thin dense chrome coating with a proprietary polymer compound added. Oerlikon recommended adding their Balinit C coating to gear teeth and mating surfaces of bushings to extend life and capability. Balinit C is an amorphous carbon-tungsten carbide coating (WC/C) with a high surface hardness and a low coefficient of friction that claims higher bearing load capacity, lower sliding wear, improved scuffing resistance, and reduced pitting particularly in applications with boundary lubrication conditions such as slow moving gears in contact.

Capacity Validation

AGMA analysis guidelines are typically conservative and do not always directly correlate to lightweight planetary gearboxes for space applications. Additionally, the parameters used in AGMA gear calculations were developed from testing larger gears, different environments, different materials, different lubricants, and different load conditions than typically experienced in Space applications. When testing is traditionally performed on a program, gearing is only tested to the loads and life specified for the application, and not tested up to or beyond the calculated rated loads and life of the design. Development testing was needed to validate the calculated allowable loads and life for the various base gear materials used and ensures that the calculations are not over stating the capability of the gearboxes

The Gearbox design

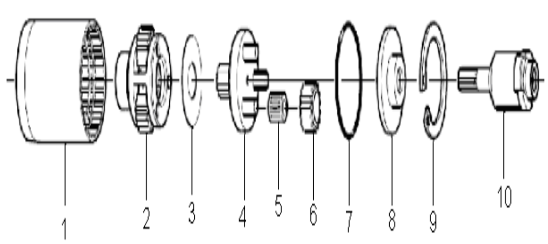

Fig 12: Gearbox design

The gearbox focused on, and used for model validation in this report is a 2 stage planetary gearbox made by Atlas Copco. The exploded view shows the components of the gearbox that will be explained in this chapter.

1. gear rim that functions as ring wheel

2. the planetary holder stage 2 complete assembly with planetary wheels

3. thin distance washer between stage 1 and 2

4. planetary holder stage 1 also acting sun wheel stage 2

5. needle bearing

6. planetary wheel stage 1

7. o-ring

8. thick washer or endplate stage 1

9. circlip to keep the thick washer in place

10. ingoing shaft from electric motor with sun wheel

The gearbox is made from hardened steel and has a gear ratio of 35 times in order to convert a high rotational speed into a large torque.

The power comes in from the electric motor to shaft (10) which is the sun wheel of the first stage. The sun wheel drives the planetary wheels (6) witch rotates in the gear rim causing a lower speed on the planetary holder (4) stage 1. The power is transferred to the planetary wheels stage 2 (2) through the sun wheel stage 2 (4) that has the same rotational speed as the planetary holder stage 1. The outgoing shaft is the planetary holder (2) stage 2 which has 35 times lower rotational speed than the electric motor.

The thin washer (3) is used as a distance between the planetary holders stage 1 and 2. It has a smooth surface since the planetary holders has different speeds.

III Literature Reviews

Few literature reviews on the design and analysis of this gear system were studied and following conclusion were made out of them:

- Planetary gear system is compact, light weight and have higher torque density than the conventional gear system.
- Studies were mainly based on single stage planetary gear system
- The extensive research in the field of planetary gear design and analysis have already been done
- In single stage differential planetary gear high reductions ratios are possible but it will also work for low torque applications

Dr. Alexander Kapelevichin his study presented few research gaps, for high reduction ratios if less than 3 stages wereselected would result in bulky and large gear system. It was also found that stresses getting induced due to the manufacturin g errors and assembly variations brought major damage to the whole planetary gear system .

Design of a 3-Stage Planetary Gearbox

Considering the power input to the gear system and all the other relevant parameters, the number of stage was rounded to be three with the first stage being that of helical gears followed by two planetary gear stages. Since the power input was high to go with higher reduction ratio required, to opt for planetary gear system was the best decision considering the compactness planetary gearbox will provide over all other gear systems.

According to the Handbook of Gear from G.M. Maitra, Design parameters that defined a planetary gear systems were at first , the number of planets each planetary gear stage would carry, the number of teeth on every gear, module for gears, pitch circle diameter the torque transmitted in each gear stage. It was found that number of planets for each planetary stage should be kept 4 and the logic behind that decision that since each planet would thus be at 90˚ to each other, so the radial forces will be opposite to each other and thus there wouldn't be any case of unbalancing as centrifugal forces will cancel out each other. After rounding on number of planets, all the other gear parameters are calculated .

As Robert G. Parker presents in his research that In many industries, it is of highest priority to maximize the power density and improve load sharing among the planets and for that it is imperative to have thin ring rings and what this does is, it leads to elastic deflection of the ring gear. There is also a possibility that sun and planets might also get deformed elastically but the ring gear is especially susceptible because it lacks the backing of any additional constraint from the bearings.

G.G.Antony presents that an important requirement for automation applications is high torque capability in a compact and light package. High Torque density is required for automotive applications because with change in high dynamic loads in order

to avoid additional system inertia and depending upon the selected nu mber of planets, planetary gearboxes distribute the torque to be transmitted through multiple gear mesh points. This means planetary gears with four planets can transfer four times the torque of a similar sized conventional gear system.

Tobias Schulze puts it in his research

Christopher G. Cooley found out currently there aren't any experimental results for planetary gears operating at high speeds but few experimental research do exist for planetary gear system. In his paper he has taken cases where only measurements of the housing were taken and no information on the motions of the individual gear bodies were to be determined. He experimentally investigate the sun and ring gear motions of a production planetary gear using inductance vibrometers and also perform modal experiments on stationary planetary gears. It measures individual gear vibrations by instrumenting each gear with accelerometers .

Th.Costopoulos in his paper presented a novel design for asymmetric gear teeth aiming at the minimization of the fillet bending stress. In his paper he put forward the following design concepts , firstly the gear teeth are made as thick as possible at the root fillet area and consequently the mating teeth are made as thin as possible at the tip. Secondly the sharp and pointed teeth are avoided in order to retain the good working properties of the standard 20 involute. He also talked regarding its insensitivity to center distance errors and its standard rated pitting and scoring resistance, the working portion of the driving side of the gear is involute. It was also suggested that the root fillet of the working side be replaced by a circular fillet and not the standard trochoidal generated from the circular tip of the generating hobbing tool and the idea behind that the circular root fillet been introduced was to increase the bending resistance of the gear teeth compared with the conventional trochoidal one.

IV. CATIA

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4.1 Introduction

CATIA (Computer Aided Three-dimensional Interactive Application) is a multi-platform CAD/CAM/CAE commercial software suite developed by the French company Dassault systems. Written in the C++ programming language, CATIA is the cornerstone of the Dassault systems product lifecycle management software suite.

CATIA competes in the high-end CAD/CAM/CAE market with Creo Elements/Pro and NX (Unigraphics).

4.2 History

CATIA (Computer Aided Three-Dimensional Interactive Application) started as an in-house development in 1977 by French aircraft manufacturer Avions Marcel Dassault, at that time customer of the CAD/CAM/CAE software to develop Dassault's Mirage fighter jet. It was later adopted in the aerospace, automotive, shipbuilding, and other industries.

Initially named CATI (Conception AssisteTridimensionnelle Interactive – French for Interactive Aided Three-dimensional Design), it was renamed CATIA in 1981 when Dassault created a subsidiary to develop and sell the software and signed a non-exclusive distribution agreement with IBM.

- In 1984, the Boeing Company chose CATIA V3 as its main 3D CAD tool, becoming its largest customer.
- In 1988, CATIA V3 was ported from mainframe computer to UNIX.
- In 1990, General Dynamic Electric Boat Corp chose CATIA as its main 3D CAD tool to design the U.S. Navy's Virginia class Submarine. Also, Boeing was selling its CADAM CAD system worldwide through the channel of IBM since 1978.
- In 1992, CADAM was purchased from IBM, and the next year CATIA CADAM V4 was published.
- In 1996, it was ported from one to four Unix operating systems, including IBM AIX, Silicon Graphics IRIX, Sun Microsystems Sun-OS, and Hewlett-Packard HP-UX.
- In 1998, V5 was released and was an entirely rewritten version of CATIA with support for UNIX, Windows-NT and Windows-XP (since 2001).
- In 2008, Dassault released CATIA V6. While the server can run on Microsoft Windows, LINUX or AIX, client support for any operating system other than Microsoft Windows was dropped.
- In November 2010, Dassault launched CATIA V6R2011x, the latest release of its PLM2.0 platform, while continuing to support and improve its CATIA V5 software.

In June 2011, Dassault launched V6 R2012

Developer(s)	Dassault Systems
Stable release	V6R2011x / November 23, 2010
Operating system	Unix / Windows
Type	CAD software
License	Proprietary
Website	WWW.3ds.com

Table 4.1 : Details of CATIA

4.3 Scope

Commonly referred to as a 3D Product Lifecycle Management software suite, CATIA supports multiple stages of product development (CAX), including conceptualization, design (CAD), manufacturing (CAM), and engineering (CAE). CATIA facilitates collaborative engineering across disciplines, including surfacing & shape design, mechanical engineering, and equipment and systems engineering.

CATIA provides a suite of surfacing, reverse engineering, and visualization solutions to create, modify, and validate complex innovative shapes, from

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subdivision, styling, and Class A surfaces to mechanical functional surfaces.

CATIA enables the creation of 3D parts, from 3D sketches, sheet metal, composites, molded, forged or tooling parts up to the definition of mechanical assemblies. It provides tools to complete product definition, including functional tolerances as well as kinematics definition.

CATIA facilitates the design of electronic, electrical, and distributed systems such as fluid and HVAC systems, all the way to the production of documentation for manufacturing.

Starting To Catia:

To start CATIA there may be icon on the desktop or you may have to look in the Start menu at the bottom of left of the screen Windows taskbar. The program takes a while to load, so be patient. The start-up is complete when your screen looks like the following figure, which is a default CATIA screen.

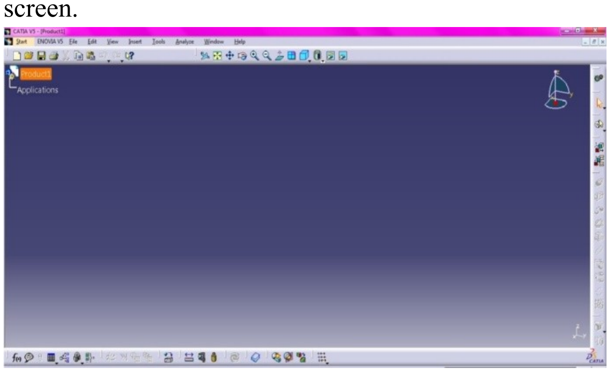

Fig14: CATIA Window Page

Now click on the start button at the top of the toolbar it shows different modules as shown in below figure for modeling select 'Mechanical Design' in that again select it shows options as shown below select 'Part Design'.

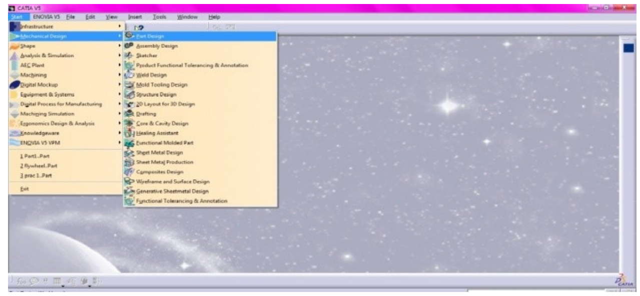

Fig15 : Selecting the module**.**

4.8 Design And Analysis Of Planetarium Gear Box In Catia:

After selecting the Part Design module the screen is as shown in below figure 16. In the screen there will be three planes XY, YZ, and ZX Planes. The XY plane represents Top or Bottom view, the YZ plane represent front or back view and ZX plane represent Right side or Left Side view. In that three planes select ZX-plane and select sketcher your screen looks like as shown in the figure 17.

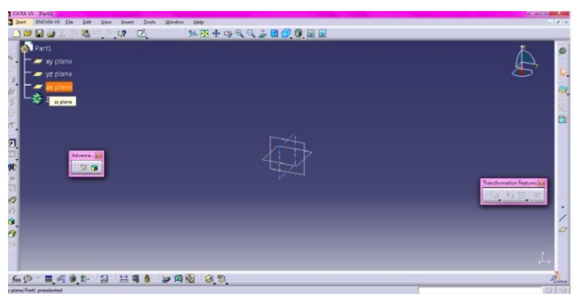

Fig 16: Selecting the plane

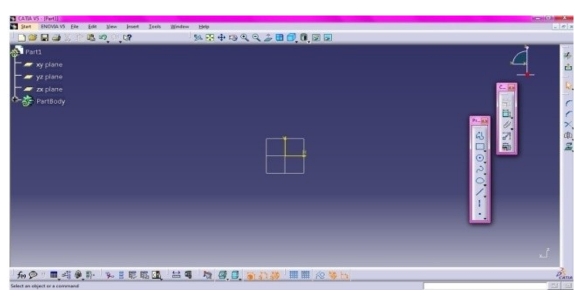

Fig 17: Select Sketcher

By using Circle Command We Can draw the circle by the diameter Of planet gear box is 116mm

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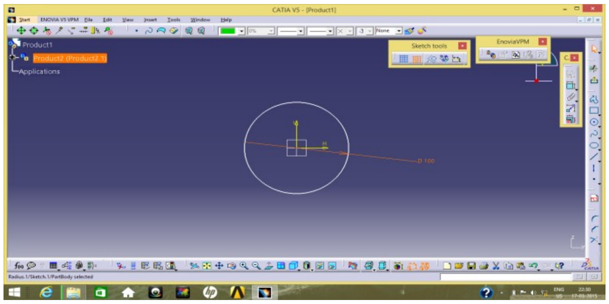

Fig 18: Circle Diameter

After we can take the reference lines for to draw the involute teeth shape of teeth as shown in below figure.

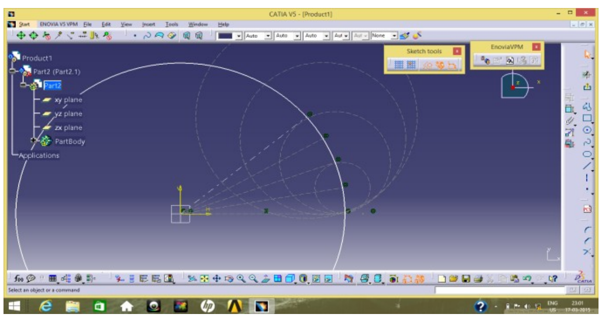

Fig 19: Draw The Involute Teeth Shape Of Teeth

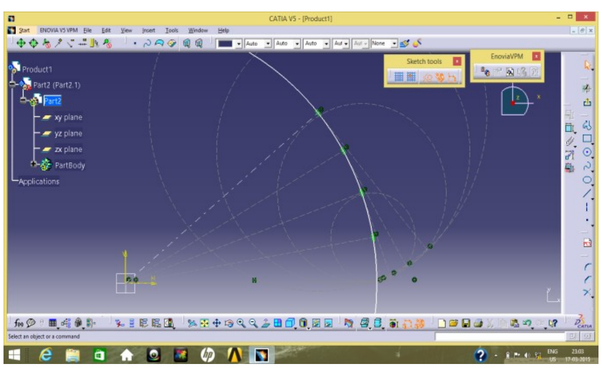

Fig 20: Reference lines

ANSYS

5.1 History

The company was founded in 1970 by Dr. John A Swanson Analysis systems, Inc. SASI. Its primary purpose was to develop and market finite element analysis software for structural physics that could simulate static (stationary), dynamic (moving) and heat transfer (thermal) problems. SASI developed its business in parallel with the growth in computer technology and engineering needs. The company grew by 10 to 20 percent year, and in 1994 it was sold to TA Associates. The new owners took SASI's leading software called ANSYS as their flagship product and designated ANSYS, Include as the new company name.

5.2 Introduction:

The ANSYS program is a computer program for a finite element analysis and design. The ANSYS program can also be used to calculate the optimal design for given operating conditions using the design optimization feature.

ANSYS is commercial finite-element analysis software with the capability to analyze a wide range of different problems. ANSYS runs under a variety of environments, including IRIX, Solaris, and Windows NT. Like any finite-element software, ANSYS solves governing differential equations by breaking the problem into small elements. The governing equations of elasticity, fluid flow, heat transfer, and electro-magnetism can all be solved by the Finite element method in ANSYS. ANSYS can solve transient problems as well as nonlinear problems. This document will focus on the basics of ANSYS using primarily structural examples.

A total of six windows are opened when you start ANSYS.

1. Utility Menu (top) – contains functions that are available for throughout the ANSYS session , such as file controls, selections, graphic controls and parameters. You also exit the ANSYS program from the file pull down menu.

2 .Main Menu (bottom left) – contains the primary ANSYS functions, organized by the pre-processor, solution, general, postprocessor, design optimizer.

3. Toolbar (Middle Right) – contains push buttons that execute commonly used ANSYS commands. More push buttons can be added.

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4. Input window (middle left) – shows program prompt messages and allow you to type in commands directly.

5. Graphic window (bottom right) – a window where graphics are shown and graphical picking are made.

6. Output window (not shown here) – shows text output from the program, such as listing of data etc. It is usually positioned behind the other window and can be put to the front when necessary.

The steps in any finite element analysis can be divided in three phases:

1. Preprocessing – define the model such as mesh, loads, and boundary conditions
2. Solution – assembling and solving the system of equation.
3. Post processing – extracting relevant result from the solution.

5.2.1 Preprocessing Steps

-
  pecify job name and title.
- Set preferences.
- Define element types and options.
- Define real elements.
- Define material properties.
- Define the model starting with two rectangles.
- Change plot controls and replot.
- Change working plane (WP) to polar and create first circle.
- Move the WP and create second circle.
- Add areas (rectangles and circles).
- Create line fillet.
- Create area fillet.
- Add remaining areas together.
- Create first bolt hole.
- Move WP and create second bolt hole.
- Subtract the holes from the bracket.
- Mesh the area.

5.2.2 Solution Steps

- Apply displacement constraint.
- Apply pressure load.
- Solve
- **Begin the work in ANSYS WORKBENCH**

  Start the ANSSY 14.5 WORKBENCH from the desktop by double click on it. The start-up screen will look like this.

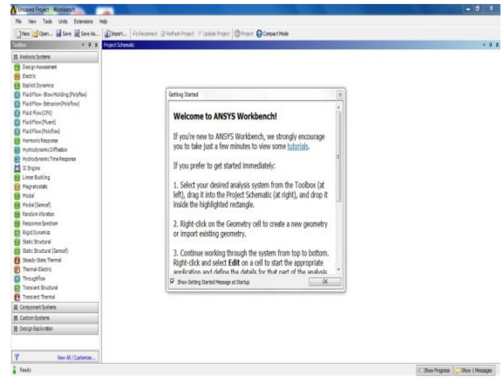
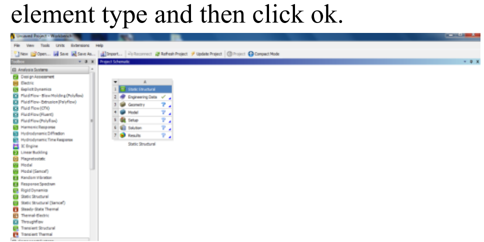


Fig 5.1 Start-Up Screen Of Ansys Workbench

After selecting Structural define type of element in Preprocessor, select Add/edit, a dialogue box namely Element type appears, click on button ADD and select Solid 10 node 187 in library of element type and then click ok.



-
- **Fig 5.2** workbench project mode
- Import the geometry model from the computer which is already done in catia by giving right click on the geometry. The below figure shows the following.

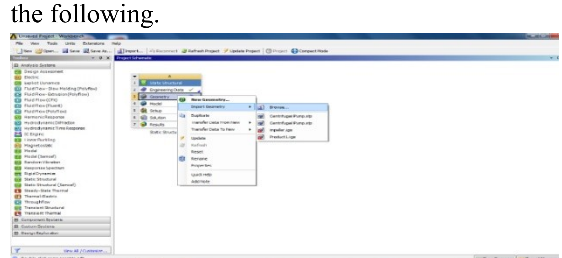


Fig.5.3. Importing model to Ansys Work Bench

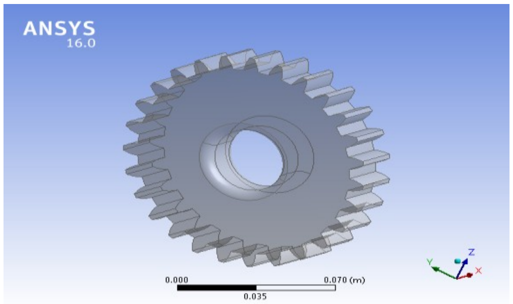
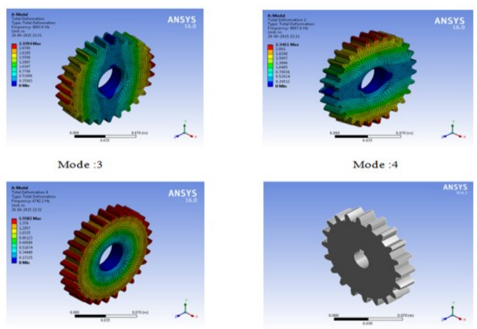


Fig.5.4. Importing the sun gear



Fi

g.5.5. Meshing of sun gear

Mode :1 Mode :2

V. CONCLUSION

Designing Planet gear box is one of the difficult task, major part of our project is done in CATIA v5, which includes designing spur gears with involute teeth with required dimensions and assembly of messing gears

Dynamic analysis is done by using ANSYS on planet and sun gear using different material, static analysis is done on meshing assembly of sun and planet gears using different material

Comparing to both material,titanium alloy has both good static and dynamic analysis features

The cost is comparing to the both materials titanium alloy is high and its strengths are very good properties

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