# Design and Development of a Microfluidic Gear Pump for Unmanned Aerial Vehicle (UAV)

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**Abstract - In today's ever growing and expanding field of sciences, the microfluidics finds its way into numerous applications viz. optics, mechatronics, biotech, medical sciences, etc. Research work in this field is now focusing towards using optimum methods of manufacturing and developing better technologies towards micro engineering. In this paper we will be focusing on the design aspect of a micro-gear and perform required analysis of micro-gears. We are also going to study in detail various design parameters and pump parameters of designing micro-gear pump. The material going to be used in this pump for the gears is PLA. We intend to design the microfluidic gear pump using modelling (Solidworks) software. The target application of our pump is the fuel pumps for UAV (Unmanned Aerial Vehicles).**

**Keywords: UAV, Micro fluidic gear Pump, PLA**

## 1. INTRODUCTION

The design focused at providing the defense sector or the military sector and the aviation industry with an indigenous pump that delivers about $5.8333\times10^{-6}$ $m^3$/sec (35 g/min) water. Pump is a device that provides mechanical energy to be imparted to a fluid and manifests in pressure energy increase. Gear pumps have been the main choice of fuel system designers due to low operating cost, long life and high performance.

The conventional or traditional centrifugal pumps required to be primed initially under service conditions except only when the suction has a positive head. Situations may arise in actual practice that the suction to a pump has negative head. The gear pump is beneficial here as it is self-priming and is not constrained by the nature of suction head. There are cases in which a fixed or certain quantity of fluid is required per unit time or we can say per revolution of a pump. In water treatment plant for example, the quantity of chemical dosage is closely monitored and the ability of the dosing microfluidic pump to provide a specified amount of chemical or fluid per pump revolution is crucial. Chemical plants too require microfluidic pumps that can deliver a fixed volume of fluid per unit revolution. Gear pumps are part of the group of positive displacement pumps which have remarkable characteristics of fixed volume discharge per unit revolution of the pump.

The compactness and miniaturization of the gear pump makes it one of the few hot engineering equipment topics for study which ideally any developing country with even a little interest in technology transfer world can start with.

## 2. LITERATURE SURVEY

2.1 Review *DESIGN AND PROCESS PLANNING OF MICRO GEAR MANUFACTURING* Suraj Kumar Sharma1,Prashant Mohan Trivedi2, Sourabh Gupta

1M.Tech. Mechatronics Department, Vellore Institute of Technology, Vellore, Tamil Nadu, India 2M.Tech. Electronics & Communication Department, SRM University, Chennai, India 3B.E. Mechanical Engineering Department, B.I.S.T, Bhopal, Madhya Pradesh, India (2014)in this paper there is brief information about basic introduction of simple spur gear pump.in this paper there is information regarding gear train design ,modelling (3D),scaling ,different manufacturing processes used for micro gears and part handling system.

**2.2** *Design, Fabrication, and Characterization of a Continuous Flow Micro-pump System (June 2016)*

*Ala'aldeen T. Al-Halhouli* in this paper present the fabrication, design and properties of a continuous flow micro-pump system used as in lab-on-chip systems for biotechnological screening. It includes a parallel membrane micro-pump with passive check valves and an extra fluidic capacitor. Keeping in mind the criteria of easy inerrability into a microfluidic platform, the pneumatic actuation principle is most appropriate. Pneumatic micro actuators includes high actuation speed, large stroke and they are free from high electric voltages, for example, on piezoelectric actuators. Also, they are easy to fabricate.

**2.3** *Design Analysis and Testing of a Gear Pump*

*E.A.P. Egbe (May 2013)*

Gear pump has an easy working mechanism consisting two meshing helical or spur gears – the driver and the idler. There are two main groups of gear pumps: the external and the internal one type. Firstly it uses two external spur gears while then it uses one external spur gear and one internal spur gear. The distribution of gears on the suction side provides a partial vacuum which causes liquid to flow in and fluid fills suction (inlet) part. The fluid is carried to the discharge (outlet) part between the revolving gear teeth and fixed casing. The meshing of gears creates an increase in pressure which causes the liquid out from the discharge (outlet) port. One of the ports can become the discharge depending on the direction of revolving gear pair. Top clearances and tight side among the gears and casing avoids the fluid leakages.

## 3. PROJECT CONCEPT

### 3.1Schematic

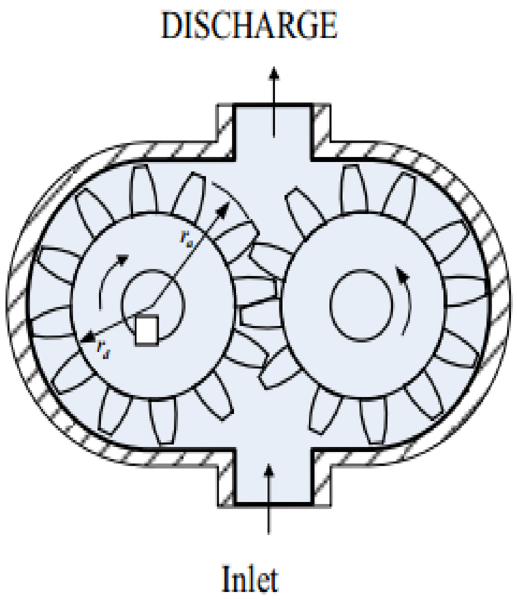


Fig 3.1 Cross section of a gear pump

### 3.2 Gear pump

A microfluidic gear pump displaces the fluid at an appropriate pressure by using the meshing of gear pair. They are one of the most common types of pumps for hydraulic fluid power applications. Gear pumps are also widely used in chemical installations to pump fluid with a certain viscosity. The gear pumps are of two types; external gear pump and internal gear pump. Microfluidic Gear pumps are also called as positive displacement pumps (fixed displacement pump), meaning they displace a constant quantity of fluid in each revolution.

### 3.3 Theory of operation

As fluid enters, the moving gear pair which are in mesh separates and fluid goes to the area in between the teeth and travel further. The fluid is travelling by use of gears form suction to discharge side of the pump, where the rotation of gear which are in mesh displaces fluid at discharge side. The clearance between gear tooth and casing body is too small, and ideally it is taken as 10 µm. The small clearances, along with high speed of rotation, effectively control the fluid from leakage through wall of casing. The reliable rigid design of gears which are in mesh and houses permits very high pressure of fluid. There are many types of gears which are used in a gear pump, they are as follows: helical gear pair, herringbone gear pair, lobe shaped gears (rotors) and mechanical design of gear pair which allow the stacking of micro fluidic gear pumps.

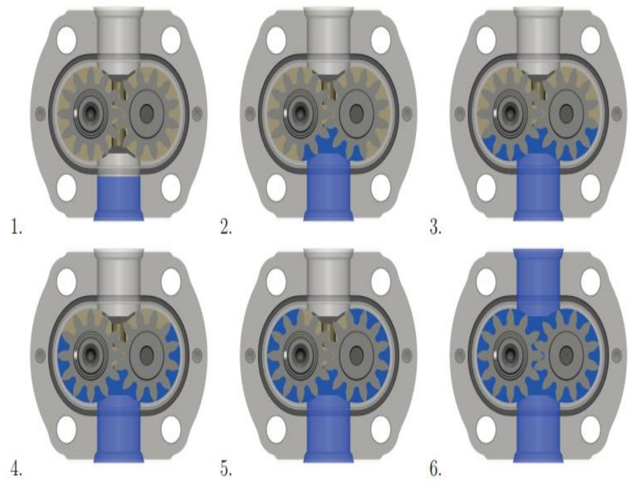


Figure 3.2 Theory of Operation of Gear Pump

## 4. PROBLEM STATEMENT

Design of Microfluidic Gear Pump for required flow rate $5\times10^{-6}m^3/s$ using 18000 rpm motor by appropriate material using appropriate process considering cost and time in manufacturing process.

## 5. TARGET APPLICATION

UAV stands for 'Unmanned Aerial Vehicle'. It is an unmanned aircraft that can be remotely controlled by a pilot on the ground or it can fly autonomously based on pre-programmed maps or more complicated dynamic automation setup. An UAV is currently employed for a number of operations, even for reconnaissance and aerial attack. A UAV is known to be capable of being remotely controlled, sustainable at a fixed altitude of flight powered by a jet engine or combustion engine. Although a cruise missile can be assumed to be UAV, it is treated separately because the vehicle acts as the weapon.

The role of UAV in military operations is growing at mindboggling rates. Tactical and theater level UAVs, alone had flown over 100,000 hours in response to Operation ENDURING FREEDOM (OEF) and Operation IRAQI FREEDOM (OIF) in 2005. Unprecedented advances in tech are allowing more and more ability to be placed on even

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smaller airframes which is causing a large growth in the number of SUAS being deployed on the battleground. Such is the newness of the SUAS in combat that no formal DOD wide recording procedures have been formed to keep track of SUAS flight hours. As the capabilities increase for all types of UAV, nations carry on to subsidize their study and development leading to further improvements enabling them to perform a variety of missions. UAV no longer perform just intelligence, reconnaissance and surveillance missions, although this is still their predominant type. Their roles have expanded to fields containing electronic attack (EA), strikes, suppression and or destroying air defense of the enemy (SEAD/DEAD), network node or communication relays, combat search and rescue (CSAR), and derivatives of such themes. The cost of such UAVs vary from a few thousand to a few million dollars, and the aircrafts used in these systems vary in size from a Micro Air Vehicle (MAV) (weighing less than a pound) to an aircraft weighing over 40,000 pounds.


Figure 5.1. Pictorial of an UAV

### 5.2 The problem statement- Fuel Pump for UAV

Fuel supply is one of the biggest causes of engine related drawbacks in UAVs powered by IC engines. Almost all drones, carbureted or fuel injected, have suffered from the consequences of an insufficient fuel delivery (so many unknowingly yet fail). While simple in concept, a vast majority of iterations seen over the years do not meet one or more of the basic requirements for smooth and reliable operation.

Mostly, the problem is related to an insufficient fuel delivery (pump /pickup /header) system & in most instances the root cause of the problem is the fuel pump itself. Most low volume fuel pumps in the market were not designed to be self-priming with a relatively high compression ratio. The stringent requirement of a UAV fuel pump is the continuously pumping of air until the main pickup or clunk is re-immersed in fuel without vapor locking, regardless of the application (carbureted or fuel injected). If the fuel pump doesn't satisfy this requirement one must ensure that, in every phase of the flight & in every possible orientation encountered, the fuel suction or clunk will never take in any air (must be completely immersed in fuel). This is anything but possible to ensure in the UAS application, thus your pump has to be self-priming. Failure in understanding and meeting such a simple requirement always has & continues to cause the loss of countless units & expensive payloads.


Figure 5.2 'MOSCAT' fuel pump

## 6. DESIGN CALCULATION OF GEAR PUMP

### 6.1 DESIGN OF GEARS

Input parameter/Required parameters:

1. Flow Rate : $5\times10^{-6}m^3$/s
2. Required rpm: 18000
3. No. of teeth : 24
4. Face width : 0.003m=3mm

Geometrical design:

$$Pd = \frac{n}{D} \tag{1}$$

$$Pc = \frac{\pi D}{n} = \frac{\pi}{Pd} \tag{2}$$

$$a = \frac{1}{Pd} = \frac{Pc}{\pi} \tag{3}$$

Dedendum

$$d = \frac{1.25}{Pd} \tag{4}$$

Clearance:

$$c = \frac{0.25}{Pd} \tag{5}$$

Where n is number of teeth on gear, D is the circular pitch diameter. Mott stated that the recommended working depth be $2/P_d$. Therefore the following deductions which can be made from the aforementioned conditions are:

The outside diameter of gear is:

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$$D + 2a = \frac{D+2}{Pd} = \frac{D+2}{Pd} = \frac{D(n+2)}{n} \qquad (6)$$

Whole depth:

$$a + d = \frac{2.25}{Pd} = \frac{2.25 \times D}{n} \qquad (7)$$

The level of noise in gears is, as we know, a function of the gear geometry, the clearance and the precision. The equations expressed here in (1) to (7) were adopted in designing the geometries and tolerances of all the required components of the gear pump. The discharge as specified is: 5.8333×$10^{-7}m^3$/sec = 35 g/min. At the initial stage of gathering relevant and required information on available technology and research for this work it was understood that the available 20 degree involute gear cutter is limited to a minimum of about 12 number of teeth on a gear. We choose 24 numbers of teeth for our gear. The volume of fluid displaced per revolution (here denoted by $D_p$) is equal to volume of fluid which is trapped within the space of gear teeth and pump housing (Figure 1). The trapped volume can be given by:

$$Dp = \frac{\pi \times (ra^2 - rd^2) \times b}{2} \qquad (8)$$

Where $r_a$ and $r_d$ are the addendum and dedendum radii respectively and b is gear face width. The geometry of the gears in Figure indicates that the addendum radius is given by:

$$ra = \frac{D}{2} + a = \frac{D \times (2+n)}{(2n)} \qquad (9)$$

Similarly the dedendum radius is given by:

$$rd = \frac{D}{2} - d = \frac{D \times (n-2.5)}{(2n)} \qquad (10)$$

Substituting expressions for $r_a$ and $r_d$ into Equation 9 resulted in:

$$Dp = \frac{\pi(b \times D^2)(9n - 2.35)}{(8n^2)} \qquad (11)$$

The constraint on our available 20 degree involute cutter suggests that 'n' must be greater than or equal to 12 and the value n = 24 was used. The speed of the motor was selected as 18000rpm and pump discharge as 5×$10^{-6}m^3$/sec. The face width 'b' and pitch circle diameter 'D' remained unknown parameters in (11). However 'b' must be however greater than 8/$p_d$ but at same time be less than 16/$p_d$. Substituting from (1) yielded:

$$\frac{8D}{n} < b < \frac{16D}{n}$$

For pitch circle diameter (D):

60×Q/RPM=πb $D^2$[qz-2.35]/8×$Z^2$

By putting parameters:

60× 5×$10^{-6}$/18000=π [0.003$D^2$] [q24-2.35]/8×$24^2$

PCD=mm

=0.064329094488 inch

Displacement ($D_p$) =Q/ (rpm)

=5×$10^{-6}$×60/ (18000)

$D_p$ = 16.66666667×$10^{-9}m^3$/rev

Center distance (C) = PCD = 6.175805327×$10^{-3}$mm

Outside diameter ($D_o$) = N+2/$p_d$

= 24+2/3.886132857×10+33$m^{-1}$

= 6.690455771×$10^3$mm

Where $p_d$ = dimetral pitch=3.886132857×$10^{+3}m^{-1}$

| DESIGN PARAMETERS OF GEARS | MAGNITUDE |
|---|---|
| No. of teeth (n) | 24 |
| Pitch circle diameter (D) | 6.175805327×$10^{-3}$m |
| Center distance (C) | 6.175805327×$10^{-3}$m |
| Outside diameter($D_o$) | 6.690455771×$10^{-3}$m |
| Dimetral pitch($p_d$ ) | 3.886132857×$10^{+3}m^{-1}$ |
| Addendum (a)<br>Addendum diameter | 0.257325222×$10^{-3}$m<br>6.690455771×$10^{-3}$m |
| Dedendum (d)<br>Dedendum diameter | 0.321656522×$10^{-3}$m<br>5.532492273×$10^{-3}$m |
| pressure angle (Ø ) | 20 ° |
| Circular pitch ($p_c$) | .808411026×$10^{-3}$m |
| thickness (t) | 0.44205513×$10^{-3}$m |
| face width (b) | 0.003m |

**Stress analysis and material selection:**

The dimensions of our gear which were obtained on the lines of expected discharge were not altered under consideration of stress. Stress analysis simply made it possible to select optimum materials under the actual operating conditions.

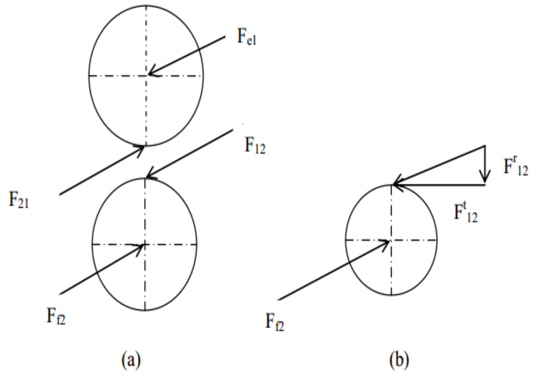


Figure 6.1: Forces acting on the gears and component of forces.

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The forces acting on the two of our gears under loading conditions are shown in Figure 6.1. The useful transmitted load, as we know, which is involved in transmission of power is the tangential component of force exerted by gear 1 on gear 2 and given by:

$$W_t = F_t$$

The standard Lewis stress equation was modified to account for dynamic factor, geometry factor, and stress concentration factor and the resulting load became,

$$W_t = \frac{(K_y \times b \times Y \times \sigma)}{(K_f \times Pd)} \qquad (12)$$

Where $K_y$ = dynamic factor, Y= geometry factor, σ = stress, b =face width, $P_d$ = dimetral pitch and $K_f$ = concentration factor. It is indicated that $K_f$ for $20^0$ involute gear is expressed by,

$$K_f = 0.18 + \left[\left(\frac{t}{r_f}\right)^{1.5} \times \left(\frac{t}{l}\right)^{1.45}\right] \qquad (13)$$

Here standard gear root fillet radius $r_f$ is $0.3/P_d$ = $7.719756659\times10^5$m, l is working depth = $2/P_d$= $5.146504439\times10^{-4}$m and t is tooth thickness = Pc/2 = πD/2n = $4.0425513\times10^{-4}$m. Similarly $K_y$ can be expressed as,

$$K_y = \frac{3}{(3+v)} \qquad (14)$$

Where v is pitch line velocity in m/s. The design was thus considered to be satisfactory when the load computed from (17) was equal or greater than the dynamic load on our gear. The pump was assumed to withstand theoretically a maximum discharge pressure of 44 Psi = 3 bar. Recalling equation (8), the torque applied on our gear through the shaft was calculated to be,

$$T = \frac{Dp(P1-P2)}{2\pi} \qquad (15)$$

Efficiency of transmission of torque from a motor to a pump was assumed to be as 70%. Therefore the motor torque can be stated as $T_m$ = T /7. However the load on any tooth in a gear, W is, as we know, a function of torque transmitted and it can given by:

$$W_r = \frac{W_d}{2} = T \qquad (16)$$

Applying the maximum possible torque and known pitch circle diameter yielded,

$$W_t = \frac{T \times 2}{D}$$

The dynamic load on the gear is given by,

$$W_d = \frac{W_t \times (3+v)}{3} \qquad (17)$$

Where v is the pitch line velocity

Substituting and Calculating:

$K_f$=0.18+ [($4.04205513\times10^{-4}$/$0.719756659\times10^{-5}$) ^0.15] × [($4.04205513\times10^{-4}$/ $5.146504439\times10^{-4}$0.45]

= 1.329847485

$K_y$ = 3/ (3 + 5.820559394) where v=Pitch line velocity

=5.820559394z m/sec

= 0.340114483

Therefore from (15) T = 5.411268065× $10^{-3}$ Nm

Therefore = Tact = 7.73038295× $10^{-3}$ Nm

From (16),

$W_r$ = $T_m$=$T_{act}$ =7.73038295× $10^{-3}$ Nm

$W_d$= 7.360584122 N

Substituting into (17) yielded a dynamic load value of $1.138122695\times10^{-3}$N

The following available materials were considered for use:

1) ABS (ACRYLONITRILE BUTADINE STYRENE)
2) POLYAMIDE NYLON- 6
3) PLA (POLYLACTIDE)

The load transmission capacity,$W_t$ for each material as computed from Equation 17 as shown for ABS. The value of geometry factor for 20 degree involute gear with 24 teeth is 0.337. All known and calculated values were substituted into (17) to give the following,

Therefore from (12) for ABS,

$W_t$= 2.503441265 N

Figure 6.1.2 Selection of material for gear

Though PLA demonstrated the least safety margin, followed by ABS and NYLON 12PA, the results actually indicate that all available materials here could be used.

The **NYLON 12PA** is used here because of its cost and the fact that it was readily available.

| MATERIAL | TENSILE STRENGTH Mpa | $W_d$ N | $W_t$ N($10^{-6}$) | SAFETY MARGIN ($W_d/W_t$) |
|---|---|---|---|---|
| ABS (ACRYLONITRILE BUTADINE STYRINE) | 44.60 | 7.3605841 22 | 2967506.3 58 | 403161.819 |
| POLYAMIDE NYLONE-12PA | 46 | 7.3605841 22 | 3060656.7 32 | 415817.105 3 |
| PLA (POLYLACTIDE) | 57.8 | 7.3605841 22 | 3845781.7 83 | 522483.23 |

### 6.2 DESIGN OF SHAFT

The shaft which carries a spur gear must resist shear force that's because of applied torque and that is due to the bending load. The bending load was assumed to be acting

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through the midpoint of the shaft. The bending force was acting on the shaft is 7.73038295×$10^{-3}$N (from calculations).The bending moment diagram shown in Fig. 6.2 represents that the reaction $R_A$ and reaction $R_B$are given as follows

$$R_A = R_B = \frac{F_t}{2} == 3.865191475 \times 10^{-3}\, N$$

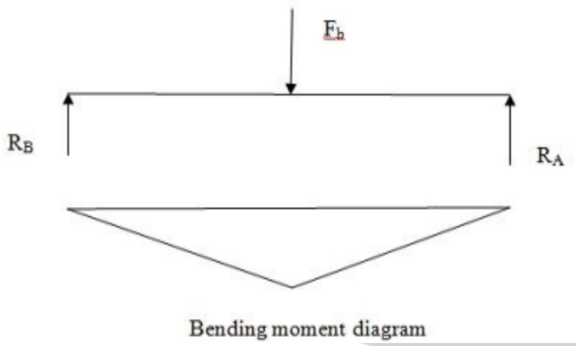



Figure 6.2 Bending forces and the bending moment diagram.

The maximum bending moment as given in the diagram is,

$$M_{max} = RA\, x\, 18\, x\, 10^{-4} = 5.81711317x\, 10^{-6}Nm$$

The bending stress:

$$\sigma_x = \frac{32Mmax}{d^3} \quad (18)$$

While stress due to torsion:

$$\tau = \frac{16T}{d^3} \quad (19)$$

The maximum shear stress on the shaft is given by:

$$\tau_{max} = (16/\, d^3)x\, \sqrt{}\, (M^2 + T^2) \quad (20)$$

By implementing the maximum allowable shear stress theory (theory of failure), the shaft diameter was calculated as:

$$d^3 = \frac{[16\, x\, 8\, \sqrt{}(M^2 + T^2\,)]}{\times} \quad (21)$$

Substitution of yield stress values of available materials into (21) produced the diameter values given in the table

Figure 5.2.1 Selection of material for shaft

The **shaft diameter of 1.918mm of ABS material** was selected due to cost and ease of manufacturing.

| MATERIAL | $\sigma_t$(Mpa) | d(mm) |
|---|---|---|
| ABS (ACRYLONITRILE BUTADINE STYRINE) | 44.60 | 1.918560314 |
| PLA POLYLACTIDE | 57.8 | 1.759723619 |

## 6.3 DESIGN OF PUMP HOUSING

Gears and housings are the most complex components of a gear pump. On contrary, the shafts are the simplest components- easy to design & produce. The ongoing led to sacrificial design of the shaft with respect to the housing. The maximum permissible torque in the PLA shaft was calculated from equation (20). That is,

$$Ts_{max} = \frac{\pi x\, d^3 x}{16} = \frac{\pi x\, d^3 x\, \sigma_{ty}}{2\, x\, 16}$$

= [π (5.510$^{-4}$) x 57.8x]$10^{-6}$ / 32

=0.03092154Nm

The maximum pressure associated with the torque of 9.440952254 x$10^{-4}$Nm was calculated with Equation (8). Therefore the maximum pressure in cylindrical housing was computed from (8) as,

$$Ts_{max} = 0.03092154Nm$$
$$= \frac{D_p \times (P)}{2}$$

Therefore,

P = 11.65714383Mpa

Solving for P yielded a pressure of 11.65714383Mpa. This pressure can cause failure of the shaft & only a marginal difference would ensure a sacrificial failure of the shaft before failure of the housing. A safety margin of 1.1 is adopted in the design. This means an internal pressure of 11.65714383Mpa ×1.1 = 12.82285822MPa & the two stressed components in very thin pressure vessels are, axial & circumferential stresses. The circumferential stress is very critical & important in cylindrical pressure containers and is given by,

$$\sigma_y \frac{P}{t} \quad (22)$$

Where P= internal pressure = 12.82285822Mpa r = internal radius of cylinder = $r_a$+0.5c = 3.365222788×$10^{-3}$and t= wall thickness. Using PLA with a yield strength of 57.8 MPa and a design factor of 1.5, yielded the thickness as,

t = 12.82285822×$10^6$× 3.365222788×$10^{-3}$ / 57.8

= 9.67530046×$10^{-4}$m

## 7. STATIC ANALYSIS SPUR GEAR DESIGN BY USING ANSYS WORKBENCH

### 7.1STATIC ANALYSIS

Static analysis is used to determine response of gear pair to study loads applied on them without changing time that is unchanged time .Response of gear is in terms of stress, strain and displacement

### 7.2PROCEDURE OF STATIC ANALYSIS:

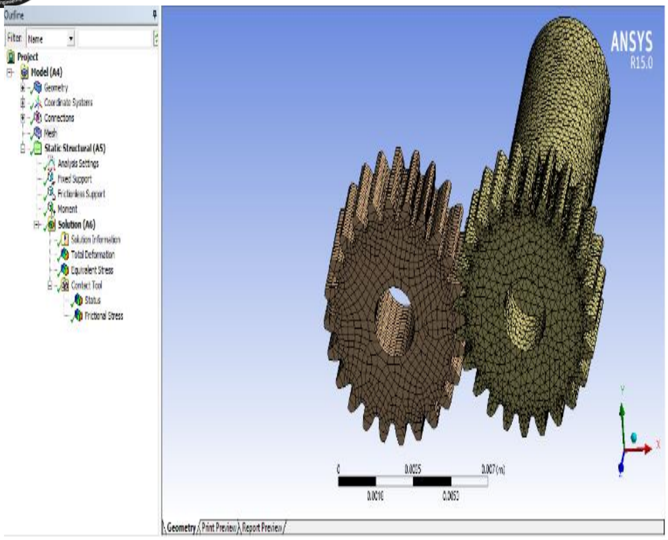
First of all, we have prepared assembly in SOLIDWORKS for spur gear and save this part as IGES for exporting into ANSYS workbench Environment .Import IGES mode in ANSYS workbench simulation module.

### 7.2.1GEAR PAIR

1. The gear pair modeled in SOLIDWORKS and later imported in ANSYS 15 for the static analysis.

**1. Meshing**

Fine meshing is done to get accurate results of contact stress figure shows finite element model of spur gear>

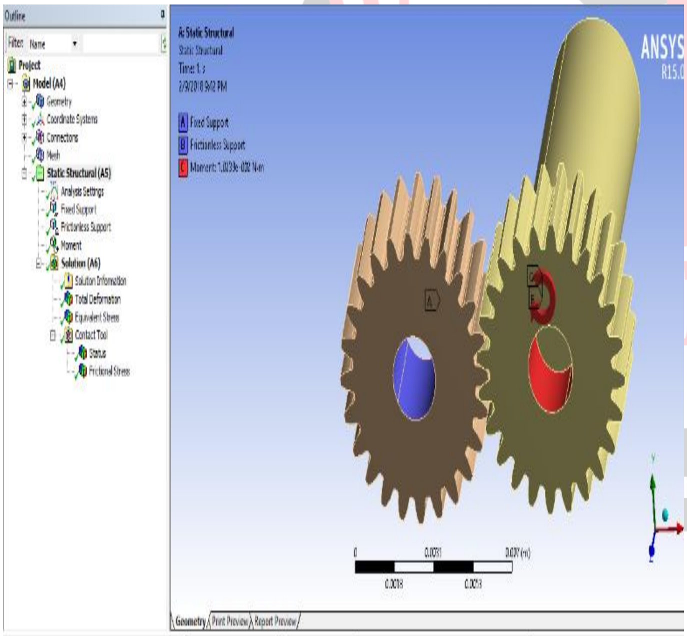



## 2. Boundary conditions

*Boundary conditions are mentioned below:*

- The fixed support is given at the inner rim of the pinion
- Frictionless support is given at rim of the gear and the torque is applied at the gear.
- Moment was applied on gear.

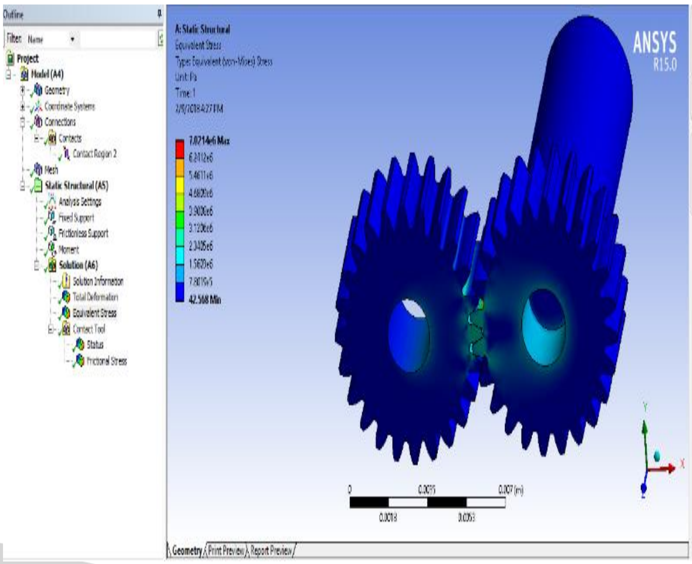
Torque applied on model allows tangential rotation but restricts radial translation of model. . Moment of 0.010238594N-m is applied as a driving torque.



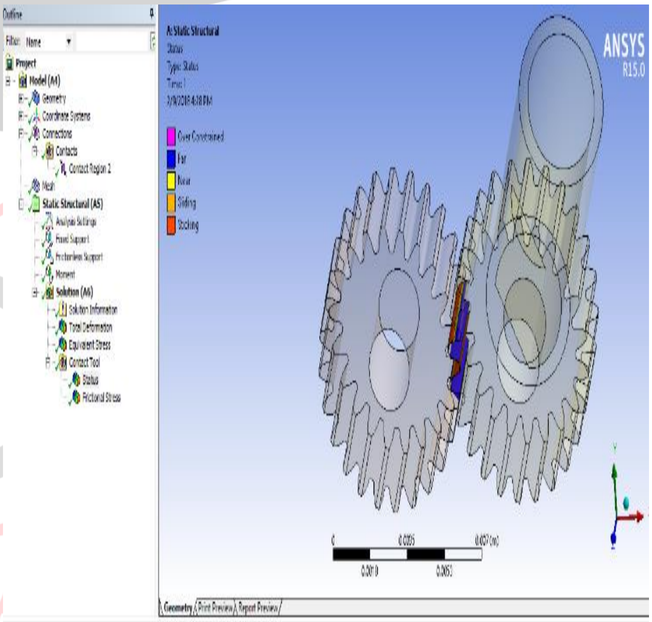

## 3. RESULTS AND DISCUSSION

### Von-mises (Maximum Equivalent Stress)

The below figure indicates contact stress on tooth meshing area made up of Nylon12PA .the maximum and minimum von-mises stress values are 7.0241N/$mm^2$and 4.2568e-5N/$mm^2$ which is lower than maximum allowable stress 46N/$mm^2$ from this result we can conclude that design is safe and can be manufactured .




### Contact tool




## 8. CONCLUSION

The design and analysis of an external microfluidic gear pump was successfully carried out in this work. This work showed a good prospect for the design and subsequent application of microfluidic gear pump which will serve as a platform for technological advancements and research in this field.

Although the fluid under study was, for ease and prototype studies, assumed to be water future research can use actual fuel for testing and simulate the actual conditions viz. temperature, pressure and other parameters for better results on the pump. The above designed pump worked sufficiently under standard atmospheric conditions and room temperature with water as a fluid.

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