Research Inventy: International Journal Of Engineering And Science
Vol.3, Issue 2 (May 2013), PP 01-07
Issn(e): 2278-4721, Issn(p):2319-6483, Www.Researchinventy.Com

Design Analysis and Testing of a Gear Pump

E.A.P. Egbe
(Mechanical Engineering Department,
Federal University of Technology Minna, Nigeria)

Abstract - *Nigeria depends heavily on importation of goods and machines. A shift from this trend requires the development of locally available technology. The design analysis of a gear pump that aimed at delivering 4.0913x10-4m3/s (24.55litres/min) of oil was carried out in this work. Available technology was utilized in the design and fabrication of the external gear pump. The design considered relevant theories and principles which affect the performance of a pump. The parts of the pump were produced locally from available materials. The performance of the pump was characterized and the test results showed a volumetric efficiency of 81.47 per cent at a maximum delivery of 20litres/minute. The discharge dropped with increase in pressure head at a rate of -0.344Litres/m.*

Keywords - *External gear, Involutes, Hydraulic, Pressure, Discharge.*

I. INTRODUCTION

The design aimed at providing the down-stream sector of the petroleum industry and the small scale industry with an indigenous pump that could deliver about 4.09123x10-4m3/s (24.55litre/minute) of hydraulic oil. Pump is a device that enables mechanical energy to be imparted to a fluid and manifests in pressure energy increase. Pumps have wide applications in science and engineering including, public water supply, irrigation, up-steam/down-stream petroleum sector, auto-mobile, haulage equipment and chemical dosage. Gear pump is the main choice of fuel system designers due to long life, low maintenance cost and high performance [1], [2].

The conventional centrifugal pumps must be primed first under service condition except when the suction has a positive head. Situations arise in practice in which the suction to a pump has negative head. The gear pump is self priming and not constrained by the type of suction head.

There are situations in which a fixed quantity of fluid is required per unit time or per revolution of the pump. In water treatment for example, the amount of chemical dosage is closely regulated and the ability of the dosing pump to supply a specified amount of chemical per pump revolution is critical. Chemical plants equally require pumps that can deliver a fixed volume of fluid per unit revolution. Gear pumps belong to the group of positive displacement pumps which are characterized with fixed volume discharge per unit revolution of the pump.

The compactness of the gear pump makes it one of the few engineering equipment which any developing country with kin interest in technology transfer should start with. Effective technology transfer can only be realized when optimum application of available technology is combined with focus on products with maximum local content.

II. LITERATURE REVIEW

Gear pump has a simple mechanism consisting two meshing spur or helical gears – the driver and the idler. There are two major classes of gear pumps: the external type and the internal type. The former uses two external spur gears while the latter uses one external spur gear and one internal spur gear.

The separation of gears on the suction side creates a partial vacuum which causes liquid to flow in and fill the suction side. The liquid is carried to the discharge side between the rotating gear teeth and the fixed casing. The meshing of the gears generates an increase in pressure which forces the liquid out through the discharge line. In principle, either of the ports can become the discharge depending on the direction of rotation. Tight side and top clearances between the gears and housing prevent the fluid from leaking backwards. The amount of fluid pumped in one revolution depends on the amount of fluid that can be trapped within the gear teeth space. Thus the discharge per unit revolution is a function of the size of the gears and the number of teeth [3].

External gear pumps with spur gears are classified as low capacity pumps. When capacity higher than 912 litres per minute is required helical and herringbone gears are employed [4]. This work is limited to the former.
The parameters of cutting tools such as pressure angle, diametral pitch, tooth addendum and dedendum have been standardized and these standards form the foundation for gear design [5], [6]. The involute profile remains the basic geometrical form for modern gearing. The 20^o pressure angle standard involute profile has a diametral pitch, circular pitch, addendum, dedendum, and minimum clearance given respectively by:

$P_d = n/D$ 1

$P_c = \pi D/n = \pi/P_d$ 2

$a = 1/P_d = P_c/\pi$ 3

dedendum = $d = 1.25/P_d$ 4

clearance = $c = 0.25/P_d$ 5

where n = number of teeth, D = circular pitch diameter.
Mott [5] indicated that the recommended working depth is $2/P_d$. The following deductions can be made from the foregoing conditions:

. The outside diameter of gear = $D + 2a = D + 2/P_d = D(n+2)/n$ 6

. Whole depth = $a+d = 2.25/Pd = 2.25D/n$ 7

These conditions which have been tested by the American Gear Manufacturers Association (AGMA) were used without further prove.
The torque required to drive an ideal pump at constant speed is given by:

$$T = \frac{D_p}{2\pi}(P_1 - P_2) \text{ Nm} \qquad 8$$

where Dp is the displacement of the pump in cubic metre per revolution of the driving gear. In practice there are torque losses due to viscous and dry friction. There are also fluid losses due to leakage and compressibility of the fluid. The performance and life expectancy of a pump depend on the properties of the liquid being pumped.

III. DESIGN ANALYSIS

The geometry of the relevant components determines the flow rate of the pump. Thus the first stage in this work involved a geometrical design to determine the dimensions of all components to satisfy the target discharge, followed by stress analysis to determine the most appropriate available material.

3.1 Geometrical design

The level of noise in gears is a function of the gear geometry, the clearance and the precision. The relationships expressed in (1) to (7) were adopted in designing the geometries of all the components of the pump. The discharge specification was $4.0913 \times 10^{-4} m^3/s$ or 24.55litres/min **-** and the available hydraulic test rig has an inlet port of 32mm diameter. At the preliminary stage of gathering relevant information on available technology for this work it was found out that the available 20^o involutes gear cutter was limited to a minim of 12 number gear teeth.

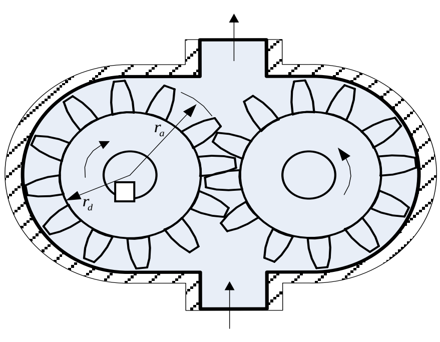


Figure 1: Cross section of the gear pump.

The volume of fluid displaced per revolution (denoted by D_p) is equal to volume of fluid trapped within the space of gear teeth and housing (Figure 1). The trapped volume is given by:

$$D_p = \frac{\pi (r_a^2 - r_d^2) b}{2} \quad 9$$

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 1 indicates that the addendum radius is given by:

$$r_a = \frac{D}{2} + a = \frac{D(2+n)}{2n} \quad 10$$

Similarly the dedendum radius is given by:

$$r_d = \frac{D}{2} - d = \frac{D(n-2.5)}{2n} \quad 11$$

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

$$D_p = \frac{\pi [bD^2](9n - 2.35)}{8n^2} \quad 12$$

The constraint on available 20^o involute cutter suggests that n must be greater than or equal to 12 and the minimum possible value n=12 was used. The speed of the motor was specified as 1400rpm and pump discharge as $4.082x10^{-4}m^3/s$. The face width – b - and pitch circle diameter – D - remains unknown parameters in (12). However two constraints were known on the face width. The face width must be longer than the diameter of discharge port. This implied that b > 32mm. Design considerations indicates that face width should be greater than 8/Pd but less than 16/Pd [5]. Substituting from (1) yielded:

$$\frac{8D}{n} < b < \frac{16D}{n} \quad 13$$

The upper limit is a critical value that must be avoided in order to prevent failure resulting from dynamic forces due to misalignment and bending of the gear under load. Moreover the noise level increases with face width. The first constraint was therefore used for initial design and the second constraint was used for cross checking. Let b=38mm for initial design. Equation 12 becomes,

$$\frac{60 \times 4.082 (10^{-4})}{1400} = \frac{\pi [bD^2](9n - 2.35)}{8n^2} = \frac{\pi [0.038 D^2](9 \times 12 - 2.35)}{8 \times 12^2} \quad 14$$

Solving (14) for D yielded, D = 0.039974m; say 40mm. Substituting this value into Equation 13 in order to verify complaint to the second constraint yields,

8x40/12=26.666< 38 =b < 16x40/12=53.333 15

This implied that the two constraints were satisfied and the final geometry of the gears became, b= 38mm, D=40mm, n=12, P_d= 7.62/in, r_a = 23.333mm and r_d = 15.8333mm.

Moreover the thickness of a tooth is half the size of circular pitch and substituting into (2) produced a value of 5.235mm.

3.2 Stress analysis and material selection.

The dimensions of the gears which were obtained on basis of expected discharge were not altered under consideration of stress. Stress analysis made it possible to select suitable materials under the operating conditions.

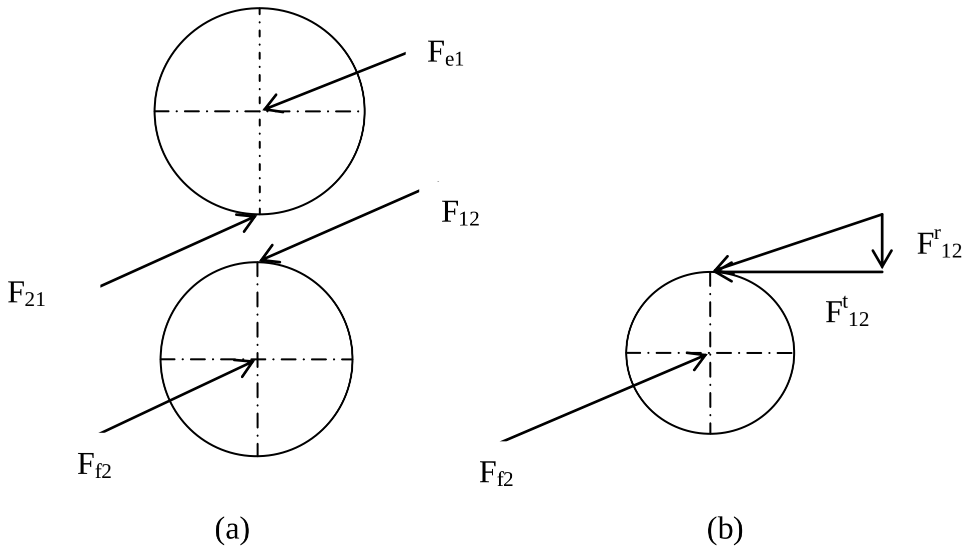


Figure 2: Forces acting on the gears and component of forces.

The forces acting on the two gears under load are shown in Figure 2. The useful transmitted load that is involved in transmission of power is the tangential component of force exerted by gear 1 on gear 2 and given by:

$$Wt = F^{t}_{12} \qquad 16$$

The standard Lewis stress equation was modified to account for dynamic factor, geometry factor, and stress concentration factor [7], [5] and the resulting load became,

$$Wt = \frac{k_v bY\sigma}{P_d K_f} \qquad 17$$

where k_v = dynamic factor, b =face width, Y= geometry factor, σ = stress, P_d = diametral pitch and K_f = concentration factor. It is indicated that K_f for 20^o involute gear [7] is expressed by,

$$K_f = 0.18 + \left(\frac{t}{r_f}\right)^{0.15}\left(\frac{t}{l}\right)^{0.45} \qquad 18$$

where standard gear root fillet radius is 0.3/Pd = 1mm, l= working depth = 2/Pd= 6.666mm and t=tooth thickness = Pc/2 = πD/2n = 5.236mm.
Similarly kv is expressed by,

$$Kv = \frac{3}{(3+v)} \qquad 19$$

where v is the pitch line velocity in m/s.

The design was considered satisfactory when the load computed from (17) was equal or greater than the dynamic load on the gear.
The pump was expected to withstand a maximum discharge pressure of 10.2 $x10^5$Pa. Recalling (8), the torque applied on gear through the shaft was found to be,

$$T = \frac{D_p}{2\pi}(P_1 - P_2) = \frac{4.082\ x10^{-4}}{2\pi}\frac{60}{1400}(10.2\,x10^{5}) = 2.84\text{Nm}$$

Efficiency of torque transmission from motor to pump was taken as 70%. Therefore the motor torque, T_m = 2.84/.7 = 4.057Nm. However the load on a tooth, W is a function of torque transmitted and it is given by:

$$Wr = W\frac{D}{2} = T \qquad 20$$

Applying the maximum possible torque of 4.057Nm and known pitch circle diameter yielded,

$$W_t = \frac{Tx\,2}{D} = 4.057\frac{2}{0.04} = 205\text{N}$$

The dynamic load on the gear is given by,

$$W_d = \frac{(3 + v)}{3}[W_t] \qquad 21$$

where v= pitch line velocity = $\frac{2\pi N}{60}\frac{D}{2} = \frac{2\pi 1400}{60}\frac{0.04}{2}$ = 2.932m/s

Substituting into (21) yielded a dynamic load value of 405.3533N

The following available materials were considered for use:

1) 0.2% C hardened steel.
2) 0.4%C hot-rolled steel
3) AISI 1020 cold rolled steel
4) Aluminium wrought (2024-T4).

The load transmission capacity, W_t, for each material was computed from Equation 17 as shown for .2%C hardened steel. The value of geometry factor for 20^o involute gear with 12 teeth is 0.264 [7]. All known values were substituted into (17) to give,

$$W_t = \frac{k_v bY\sigma}{P_d K_f} = \frac{0.506 \ x 0.038 \ x.264 \ x 426 \ x 10^6}{0.03 \ x 1.33} = 54.32479MN$$

The safety margin was obtained by dividing the load bearing capacity of the material - 54.32479MN - by the dynamic load and it was found to be 134018 for .2%C hardened steel. Similar calculations were carried out for the other materials and Table 1 presents the values.

Table 1: Load bearing capacity of available materials

Material	σ_t (N/m^2	W_d	W_t	Safety margin
0.2% C hardened steel.	$427x10^6$	405.3533	54324700	*134918*
0.4%C hot-rolled steel	$365x10^6$	405.3533	46435300	*114557*
AISI 1020 cold rolled steel	$414x10^6$	405.3533	52670300	*129936*
Aluminium wrought (2024-T4).	$331x10^6$	405.3533	42110770	*103886*

Though aluminium demonstrated the least safety margin, followed by 0.4%C hot-rolled steel, the results generally indicate that all available materials could be used.

The 0.4%C hot-rolled steel was used on the basis of cost and the fact that it was available in annealed state. This implied that the material was not subjected to surface treatment before machining.

3.2.2 Shaft design

The shaft must be capable of resisting shear forces due to applied torque and that due to bending load. The bending force was considered acting through the centre of the shaft.

The bending load was earlier found to be 205N. The loading configuration shown in Figure 3 indicate that the reactions R_A and R_B are given by,

$R_A = R_B = F_b/2 = 205/2 = 102.5N$

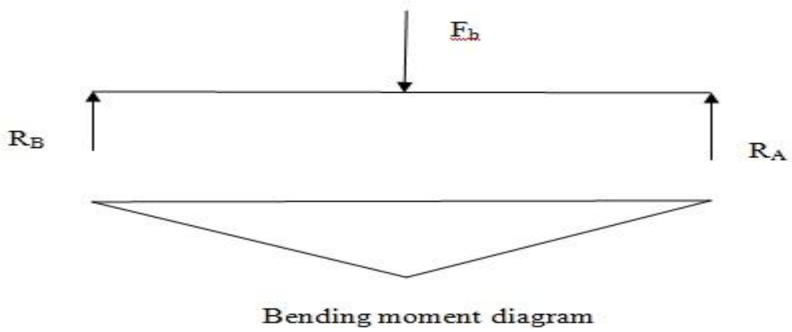


Figure 3: Bending forces and the bending moment diagram.

The maximum bending moment as shown on the diagram is,

Mmax. = RA x 3.6 x 10^{-2} = 3.69Nm.

The bending stress: σx = $\frac{32M_{max.}}{d^3}$ 22

While stress due to torsion: $\tau = \frac{16T}{\pi d^3}$ 23

The maximum shear stress on the shaft is given by:

$$\tau_{max.} = \frac{16}{\pi d^3}\sqrt{(M^2 + T^2)} \quad 24$$

Variation in pressure from suction to discharge suggested that shaft would be subjected to fatigue load and a factor of safety of four was used to account for fatigue. By applying the maximum shear stress theory of failure, the shaft diameter was calculated from (24) as:

$$d^3 = \frac{16\,x8}{\pi\sigma_{ty}}\sqrt{(M^2 + T^2)} \quad 25$$

Substitution of yield stress values of available materials into (25) produced the diameter values shown in Table 2.

Table 2: Determination of shaft diameter

Material	σ_t (MN/m^2	d (mm)
Mild steel	217	10.12
Structural steel	248	9.7
0.2% C hardened steel.	427	8.1
0.4%C hot-rolled steel	365	8.5
AISI 1020 cold rolled steel	414	9.8
Wrought iron	207	10.3

Since the shaft was to operate in oil medium, corrosion was not considered and thus mild steel with a standard diameter of 12mm was used.

3.2.3 Design of pump housing

The gears and housing are the most complex components of the pump. On the contrary the shafts are the simplest component and easy to produce. The foregoing led to a sacrificial design of the shaft with reference to the housing. The maximum possible torque in the mild steel shaft was computed from (23). That is,

$$\text{Tsmax} = \frac{\pi d^3 \tau}{16} = \frac{\pi d^3 \sigma_{ty}}{2x16} = \frac{\pi(0.012)^3 217\ x10^6}{2x16} = 36.81\text{Nm}$$

The maximum pressure associated with the torque of 36.81 was calculated with Equation 8. Therefore the maximum pressure in cylindrical housing was computed from (8) as,

$$\text{Tsmax} = 36.81 = \frac{D_p}{2\pi}(P_1 - P_2) = \frac{4.082\ x10^{-4}}{2\pi}\frac{60}{1400}(\Delta P)$$

Solving for ΔP yielded a pressure of 13.22055MPa. This pressure could just cause failure of the shaft and only a marginal difference would ensure a sacrificial failure of the shaft before the housing. A margin of 1.1 was adopted in this design. This implies an internal pressure of 13.22055x1.1 = 14.54MPas and the two stress components in thin pressure vessels are, circumferential and axial stresses. The circumferential stress is more critical in cylindrical pressure vessels and it is given by,

$$\sigma_y = \frac{\Pr}{t} \quad 26$$

where P= internal pressure = 14.54MPa, r = internal radius of cylinder = r_a+0.5c = 47.49mm and t= wall thickness (hatched in Figure 1). Using steel (UNS-G10180 –HR) with a yield strength of 220.63MPa and a design factor of 1.5, (26) yielded the thickness as,

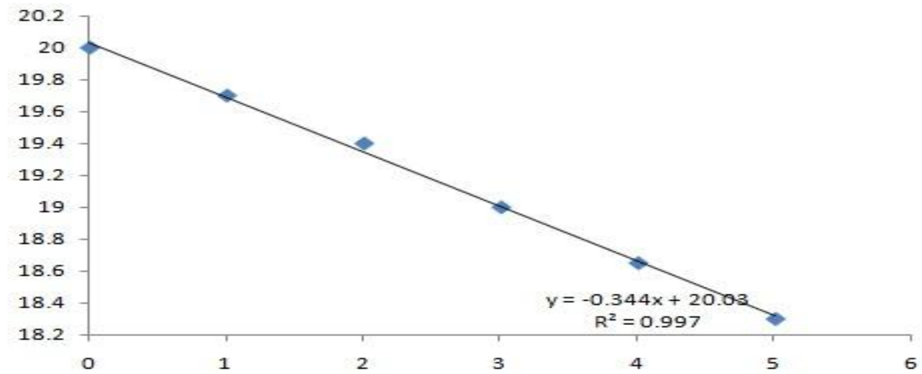
$$t= \frac{Pr}{\sigma_y} = \frac{14.54 x 10^6 x 0.04749}{220.63 x 10^6} = 4.69x10^{-3}m;\ \text{say 5mm.}$$

IV. PUMP TESTING

The components of the external gear pump were assembled and the pump was coupled to an electric motor (0.64kw, 1400rpm). The suction port was connected to the oil tank of a laboratory hydraulic rig and the discharge to the inlet of test rig. The oil discharge (in litres/minute) of the pump was measured at different pressure heads.

4.1 Test results and discussion

Figure 4 shows the variation in discharge with pressure head in metres. The gradual rate of drop in discharge with increase in head was **-**.344 per metre. The drop in discharge with increase in pressure head was due to increase in losses and pumps generally have this characteristic. The designed flow rate was 24.55litres/minute. The maximum discharge at zero head was 20litres/minute (Figure 4). The volumetric efficiency of a pump is the ratio of the actual flow rate to the theoretical (or designed) flow rate. The test result indicated a maximum volumetric efficiency of 81.47 per cent. This value is very high for a prototype. The theoretical discharge assumed perfect geometry, absence of slip and friction losses. However the gears, shafts, housing, wear plate, journal bearings and cover plate of this pump were machined manually and perfect geometry cannot be expected from such processes. Moreover flow restriction at the discharge port was not ruled out since the discharge port was smaller than the face width of the gears. Pressure build up on the discharge would normally increase slip losses and thus a drop in volumetric efficiency.



CONCLUSION

The design analysis, fabrication and testing of an external gear pump was successfully carried out in this work. This work indicated a good prospect for the design and fabrication of small machines/equipment which will serve as a spring board for technological transfer and development of our country. The components of this gear pump were fabricated by machining. Further work is required to investigate other processing routes for mass production of external gear pumps in Nigeria.

REFERENCES

[1] Manring, N.D. and Kasaragadda, S.B. The Theoretical Flow Ripple of an External Gear Pump, *Journal of Dynamic Systems, Measurement, and Control, Transactions of the ASME, vol. 125*, 2003, 396-404.

[2] Ragunathan, C., Manoharan, C. Dynamic Analysis of Hydrodynamic Gear Pump Performance Using Design of Experiments and Operational Parameters, *IOSR Journal of Mechanical and Civil Engineering, 1, (6)*, 2012, 17-23.

[3] Majundar, S.R. *Oil Hydraulic Systems, Principles and Maintenance* (Tata McGraw-Hill Publishing Corp. Ltd, New Delhi, 2001).

[4] Holland, F.A., and Chapman F.S., *Pumping of liquids* (Reinhold Publishing Corp., New York. 1966).

[5] Mott, R.L. *Machine Elements in Mechanical Design*, (Macmillan Publishing Comp. New York, 1992) .

[6] Kapelevich, A. and McNamara, T. Direct Gear Design for Optimal Gear Performance, *SME Gear Processing and Manufacturing Clinic/AGMA's EXPO' 03,* Columbus, OH, 2003.

[7] Shigley, J.E. Mechanical Engineering Design, 3rd Edition, McGraw-Hill Kogakusha Ltd, Tokyo, 1977.