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The Optimised Design of Triple Offset Butterfly Valve

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Abstract - *Nowadays, Triple offset butterfly valves are extensively used to control flow rate at required temperature and pressure. The flow rate can be controlled by a disc which rotates from 0⁰ (closed position of valve) to 90⁰ (open position of valve). Such rotation values depend upon torque value. So torque of triple offset butterfly valve plays an important role in selection of operator (actuator/gear box). In this project reverse engineering was conducted on the already existing dimensions of the triple offset butterfly valve which was used to select the parameter for optimization. Design calculations are carried out for triple offset butterfly valve BFV TCBV 28-150 CLASS (700mm size and 20 bar) based on maximum pressure. The design of triple offset butterfly valve BFV TCBV 28-150 CLASS is optimized in such way that the torque of the valve is minimized. This is achieved by optimizing the dimensions of shaft diameter, seal stack thickness and eccentricity (second offset) of butterfly valve with elliptical cross section. An analytical model and simulation model of main components is been anticipated. The modeling and structural analysis is carried out in Solidworks 2013.The torque value obtained from analytical results is compared with experimental results.*

Key Words: **Shaft diameter. Seal stack thickness, Second offset, Torque optimization**

1. INTRODUCTION

It is quarter turn rotation valve which is used to control flow of fluid through a system. It is another type of a throttling vale. It can be used with different media. It is quickly closed and opened like ball valve. When the disc is rotated quarter turn then it is fully open or closed. Therefore the valve operation time is short. Butterfly valve is an important part of fluid system, design of butterfly requires more calculation for selecting proper application and allow for system requirement. There are some direct and indirect parameters which affects the torque of triple offset butterfly valve. If it is not optimized properly then the torque increases and reduces life of triple offset butterfly valve. This project work will help to find out optimized dimension of triple offset butterfly valve for particular temperature and pressure. Such optimization of 700 mm size TOBV is done by reverse engineering and customer requirement. Generally, design of triple offset butterfly valve is based on circular cross section but this project design and the torque formulas are modified by taking elliptical cross section and define a process of reverse engineering to reduce a torque of TOBV. The prediction and assumption have taken from experience of experts, research paper and AWWA standards (American Water Works Association). This project works deals with analysis to check optimized dimension of individual component whether it is safe or not for manufacturing. The validation of project is done by taking experimental torque reading as well as checking leakages of component and select proper actuator.

Team of Naveen Kumar *et.al* [1] conducted CFD analysis to find out the characteristics of butterfly valve by using Reynolds number. They analyzed effect of disc thickness and disc shape on pressure drop characteristics. They discussed on performance of butterfly valve at low Reynolds number and they also investigated the variation of permanent pressure loss coefficient as well as the dependence of pressure distribution on the valve disc at different angle of butterfly valve with different Reynolds numbers. They showed that pressure loss coefficient is inversely proportional to the valve coefficient (Cv) and opening angle. They conclude that the pressure loss coefficient value is much higher in laminar flow and decreases rapidly with increase in Reynolds number (Re). Farid Vakili-Tahami *et.al*[2], have done analysis of hydrodynamic effect on large size butterfly valve and decide the proper dynamic torque coefficient and find out the dynamic toque of butterfly valve. They used CFD analysis to study hydro mechanical behavior of 1000 mm size of butterfly valve with working pressure of 10 bar. They assumed upstream and downstream pressure is to be constant. They showed that the dynamic torque is maximum between 60 to 80 degree and dynamic torque coefficient is maximum which is 0.5. They used six different shapes of valve disc and they showed that total average pressure on disc is high when disc opening angle is 10 degree. They also discussed the major effect of surface roughness on boundary layer thickness and fluid motion. They showed that torque of component increases when surface roughness increases. A.D. Henderson *et.al*[3], they used CFD to predict hydrodynamic torque at different angle during constant head test. The keywords of this report are Reynolds number and unsteady flow effects. They compared their results with field measurement of full size valve and made numerical model to analyze flow characteristic at higher Reynolds number. They used 3.048 m valve size and a leaf of convex cross section. Their investigation is proposed for hydro-electric power scheme. They proposed a numerical relation of dynamic torque at an angle of 60^0 from opening position of butterfly valve. They also compared velocity profile with power law for verifying flow which is developed by a time it reached the valve. They showed flow characteristics at different angle to find counter and velocity vectors on axial plane downstream from valve axis. Their numerical results showed that downstream are developed by strong vortical flow pattern and that flow becomes unsteady at different range of angle. Aniket R. Thorat *et.al* [5] conducted experiment for design of worm and worm wheel actuator of butterfly valve. They proposed mathematical model and found out the force and strength analysis. They conducted experiment on 700 mm size butterfly valve. They conclude that selected gear box pair has given good performance. K. Jayprakash *et.al* [6] predicted performance factor like hydro-dynamic torque coefficient and flow coefficient by using CFX analysis of K-w turbulence model. They measured both coefficients at an angle 5^0, 10^0 and 15^0. They have done analysis on 1.8 m size of butterfly valve. They made a process for different flow fluid at different angle with different surface roughness of disc of butterfly valve.

1.1 Problem statement

Weir-BDK is design maker and developer of Butterfly valve. So there is chance to optimize design of 28-150 class TOBV with customer requirement for water media. The existing design torque of butterfly valve of 700 mm size and 150 class type is high and It should be optimized in such way that torque value will get minimum.

1.2 Objective

1. To find out optimized parameter and calculate the torque of triple offset butterfly valve and compare torque values of existing design and optimized dimension design.
2. Structural analysis of optimized dimensions of component of triple offset butterfly valve to check strength of component.
3. For validation, to compare analytical value with experimental value.

2. DESIGN OF TRIPLE OFFSET BUTTERFLY VALVE

2.1 Torque of butterfly valve

The total torque is required to rotate disc from 0 degree to 90 degree. The main objective of triple offset butterfly valve is reduction of torque value. Because the selection of gear box is depends upon the torque value. The total amount of torque is consist of 6 torque which are as follows

Total Torque = $T_s + T_d + T_{bf} + T_p + T_{ecc} + T_h$ (2.1.1)

Where:

T_s = Seating Torque

T_d = Dynamic Torque due to lift effect of the fluid flow on the disc.

T_{bf} = Bearing Friction Torque

T_p = packing torque

T_{ecc} = Eccentricity Torque due to second offset
T_h = Hydrostatic Torque

Seating torque, bearing friction torque and eccentricity torque are primary torque in triple offset butterfly valve. Torque is calculated for open to close potion. Valve is always designed for maximum torque and pressure requirement. The maximum pressure is always present at closed position of TOBV. The differential pressure is directly proportional to a torque, so torque value decreases when the pressure difference between upstream and downstream is decreases. The following torque formula have taken from American Water Works Association (AWWA) and old design torque formulae.

2.2 Design consideration

From different torques, it is clear that seating, eccentricity and bearing friction torque are the effective torques which is acting on valve in between 0^0 to 20^0. The dynamic torque is less effective due to dynamic coefficient (C_t = 0.01)[2]. So it is not considered at the starting but it is effective in between 60^0 to 80^0 (C_t=0.5) but other torque at that angles are small. The dynamic torque is not greater than torque which is at closed position of TOBV. So it is considered that valve can take dynamic torque easily and it is not required to calculate at 75^0 position of triple offset butterfly valve. The hydrostatic torque is function of valve diameter. And its value is small as compared to other torques. This torque can be very significant in valve sizes larger than 36 inch (900m). The iteration is done by modeling software and existing old design.
According to geometry of TOBV valve, torque formulae are modified by taking elliptical cross section which is as shown in table 3.3.1According to given model we have two offset value which is not changed in design calculation. The first offset of butterfly valve = 74.24mm. The third offset of butterfly valve = 25^0

Table 2.2 Modified formulae

According to existing design and AWWA standard	According to elliptical cross section
$T_s = \mu_s \frac{\pi D^2 t_{\theta s} P}{4}$	$T_s = \mu_s \pi a b t_{\theta s} P$
$T_{bf} = \frac{\mu \pi D^2 dP}{8}$	$T_{bf} = \frac{\mu \pi a b d P}{2}$
$T_{ecc} = \frac{U_c \pi D^2 \varepsilon P}{4}$	$T_{ecc} = U_c \pi a b \varepsilon P$
$T_d = U_c C_{t\theta} D^3 P_\theta$	$T_d = U_c C_{t\theta} 8a^3 P_\theta$
$T_h = \frac{\rho \pi g (D U_{c1})^4}{64}\left(1 + \frac{8\varepsilon_2}{D}\right)$	$T_h = \frac{\rho \pi g (2a U_{c1})^4}{64}\left(1 + \frac{8\varepsilon_2}{2a}\right)$
$T_p = \frac{3\pi U_{c1} P_c H_p \mu_p \vartheta d^2}{4}$	$T_p = \frac{3\pi U_{c1} P_c H_p \mu_p \vartheta d^2}{4}$

3 DESIGN CALCULATIONS

3.1Seal stack calculation

As per existing geometrical consideration, a force on seal stack is calculated by taking elliptical cross section. Seal stack is consists of graphite and steel which is mention in table. Once force is calculated, it is designed for minimum strength of material i.e. designed for yield strength of graphite material. Calculate minimum thickness by equation

$$A_s = \frac{F}{\tau_{min}} \quad (3.1.1)$$

$$A_s = \pi * 2a_2 * t_s \quad (3.1.2)$$

$$Cos\theta = \frac{t_s}{t_{\theta s}} \quad (3.1.3)$$

Selecting a_2 as minimum major axis of seal stack shown in fig. and it is considered as thickness of rectangular plate. Calculate thickness of seal stack by equation (3.1.2) and minimum thickness $T_{\theta s}$ by equation (3.1.3). Find out the required thickness by multiplying factor of safety (FOS).

After finding the required minimum thickness, a optimization was done by reducing thickness of seal stack existing design and analyze behavior of von misses stress on thickness of seal stack. Now As per model the seal thickness is made up of layer of duplex steel and graphite lamina with different thickness. Existing design have 6 layer of steel and 5 layer of graphite. It was optimized in such way that Cv value was not affected large scale.So there are 4 layers of steel and 3 layers of graphite which are required. So, actual thickness of seal stack was selected 9.2264 mm.

actual minimum thickness	=	8.5090 mm
Required thickness is $T_{\theta s}$	=	9.22645 mm

According to geometry of valve, as thickness of seal stack changes elliptical cross section will also change. According to standards ASME B 16.34 for 700mm size TOBV, the valve diameter of triple offset butterfly valve is 692 mm. the seat diameter of any 700 mm size valve should be below 692 mm.

A minimum cross section of butterfly valve is

$$D_{min} = 0.9 * D_{std} \quad (3.1.4)$$

A minimum diameter = 0.9*692= 622.8 mm

The diameter or major axis should not be less than 622.8 mm. The diameter can be varied between 622.8 and 692 mm. This report optimized an elliptical cross section by reducing thickness of Seal stack and it is done by iteration method of existing model in solidworks 2013. As layer of seal thickness changes then the seat point contact of diameter is changed. According to optimized model of triple offset butterfly valve butterfly valve,
The optimized major axis or seat diameter = 658.9 mm. The previous existing seat diameter= 666 mm
It is between 622.8 and 692 range, then 658.8 values is accepted. This major axis affected the flow but it was in very small scale and fulfilled a customer requirement so it was accepted.

3.2 Shaft calculation

The maximum torque limit that shaft can sustained is 24000 N-m. The material of shaft is mentioned in section 4. A shaft is designed by ASME code design and it is depend upon a application of user. Such shaft diameter is designed for water media. The following equations give optimized diameter of shaft which is as follows

$$T_{eq} = \sqrt{(K_b M)^2 + (K_t T)^2} \quad (3.2.1)$$

$$\tau_t = \frac{16 T_{eq}}{\pi d^3} \quad (3.2.2)$$

According to given equation (3.2.1) and (3.2.2), equivalent torque and required diameter of shaft was calculated respectively. A valve shaft is different from automobile shaft, it is not continuously rotated. So shear strength is calculated by 0.577*yield strength. From optimized seal stack design, use major axis (maximum diameter) and minor axis to calculate force and calculate bending moment by using bending arm (distance). Put values of T_{eq} in equation (3.2.2), found out diameter of shaft. It was 70.8995 mm.

$$M_{eq} = (K_b M + \sqrt{(K_b M)^2 + (K_t T)^2} \quad (3.2.3)$$

$$\sigma_{max} = \frac{32 M_{eq}}{\pi d^3} \quad (3.2.4)$$

Once minimum diameter was calculated, calculate maximum stress for this diameter by equation (3.2.3 and 3.2.4). As minimum diameter is 70.8955 mm and maximum principle stress is 1315.3528 N/mm^2. For this design calculation, for maximum safety, choose diameter in such way that principle stresses should be below 725 MPa. After some iteration and availability of size, 88.9 mm diameter is selected. The max principal stress of 88.9 diameter is 667.19 MPa which is less than yield strength of material then diameter is accepted.

3.3 Disc and body calculation

As Seal stack diameter changed, so diameter and thickness of body and disc also changed in design model. This project proposed required optimum thickness according to new design model. Calculate thickness of body and disc and check whether it is safe or fail as per new design by simulation. According to old design, a thickness of body was 24 and it was designed for 31 bar and optimized by following equation (3.3.1)

$$t_{shell} = \frac{PR}{(SE)-(0.6P)} \quad (3.3.1)$$

Thickness of as per ASME B16.34, it should be 16 mm. So it was found to be 20 mm with allowable corrosion allowance. Disc thickness designed for 20 bar on the basis of circumferential stress and optimized dimension was found to be 48.187 mm and it validated by simulation whether it is safe or not. It was calculated by equation (3.3.2)

$$t_{disc} = \sqrt{\frac{1.24P}{\tau C}} \quad (3.3.2)$$

3.4 Second offset

The second offset of butterfly valve is generally based on experience and experiment. Ii is also obtain by analyzing the previous different sizes of valve and set the value of that offset according to prediction. In the previous design of TOBV, they have taken approximate value on the basis of experience but it is very high because it is directly proportional to eccentricity torque and it affects the eccentricity torque and increases the deflection of disc. So it should be low as possible as. The approximate value of eccentricity = 0.05* shaft diameter

4 TORQUE CALCULATIONS

The difference between existing design parameter optimized and optimized parameter shown in table

4.1 According to existing design

Table 4.1 Torques of existing design

According to Existing design and AWWA standard	Torque value (N-m)
$T_s = \mu_s \frac{\pi D^2 t_{\theta s} P}{4}$	2611.715
$T_{bf} = \frac{\mu \pi D^2 dP}{8}$	8413.089
$T_{ecc} = \frac{U_c \pi D^2 \varepsilon P}{4}$	4491.858
$T_d = U_c C_{t\theta} D^3 P_\theta$	6.203574
$T_h = \frac{\rho \pi g (DU_{c1})^4}{64}\left(1 + \frac{8\varepsilon_2}{D}\right)$	101.728
$T_p = \frac{3\pi U_{c1} P_c H_p \mu_p \vartheta d^2}{4}$	132.9688
$\mathbf{T_s + T_d + T_{bf} + T_p + T_e + T_h}$	**15757.56**

4.2 According to Optimized Parameter

The existing design in report of triple offset butterfly valve is based on elliptical cross section. The optimizing parameter of triple offset butterfly valve is as follows

Table 4.2 Previous and optimized parameter of component of TOBV

Parameter	Old (mm)	New (mm)
seal thickness (t)	14.28	9.2264
diameter of shaft (d)	92	88.9
Eccentricity(ε)	6.14	4.75
valve diameter	666	658.8

Seating, bearing, eccentricity torque formulae are modified and other are taken from American Water Works Association (AWWA).

The calculations for the torque of triple offset butterfly valve based on new optimized design are as follows

Table 4.3 Torque of Modified formulae

According to elliptical cross section	Torque value (N-m)
$T_s = \mu_s \pi a b t_{\theta s} P$	1610.0545
$T_{bf} = \dfrac{\mu \pi a b d P}{2}$	7756.755
$T_{ecc} = U_c \pi a b \varepsilon P$	3315.5983
$T_d = U_c C_{t\theta} 8 a^3 P_\theta$	6.0045445
$T_h = \dfrac{\rho \pi g (2aU_{c1})^4}{64}\left(1 + \dfrac{8\varepsilon_2}{2a}\right)$	95.941791
$T_p = \dfrac{3\pi U_{c1} P_c H_p \mu_p \vartheta d^2}{4}$	124.15885
$T_s + T_d + T_{bf} + T_p + T_e + T_h$	**12908.513**

The theoretical optimized torque
= previous design torque – optimized design torque
= 15757.56 – 12908.513 N-m
= 2849.047 N-m
The above values reduce that much torque by optimizing a dimension of components of triple offset butterfly valve.

5 SIMULATION MODEL

5.1 Material selection

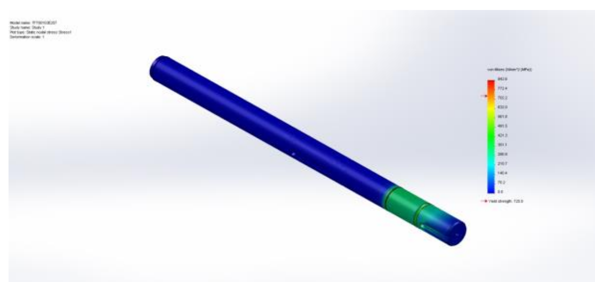
Component name	Material
Shaft	17-4PH H1150-D
Seal stack	AISI 316 Stainless Steel Sheet (SS)+C (Graphite)
Disc	CF8M
Body	Cast Carbon Steel

5.2 Shaft

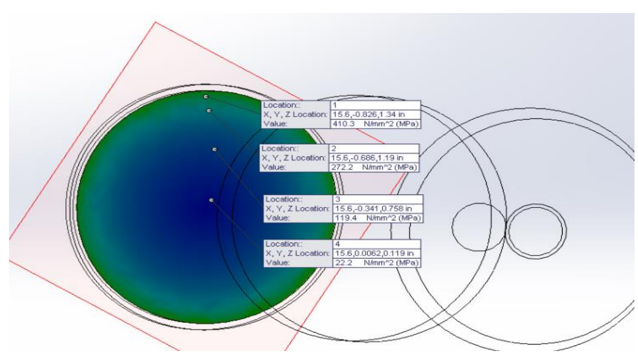
The maximum and minimum von misses stress of shaft are 842.632 N/mm² (MPa) and 3.85064e-017 N/mm² (MPa) respectively. As the shaft is designed for 24000 N-m then it can easily take 13853 N-m. so we will get below N/mm² stress value. It is shown in fig (5.2.1). As the 0.18% volume of shaft stress crosses yield stresses but it is very small. It happened due to stress concentration factor at the groove which is shown in fig.(5.2.2). So, we calculated average stress of that portion and it was 206.025 MPa which was less than the yield strength of steel 17-4 PH (H11 50 -D) (725MPa). It is shown in table 5.2. Shaft was accepted.

Table 5.2 Average stress of shaft

Stress points	Stress	Avg. stress (MPa)	Yield stress (MPa)	Remark
1	410.3			
2	272.2	206.025	725	safe
3	119.4			
4	22.2			



(a)



(b)

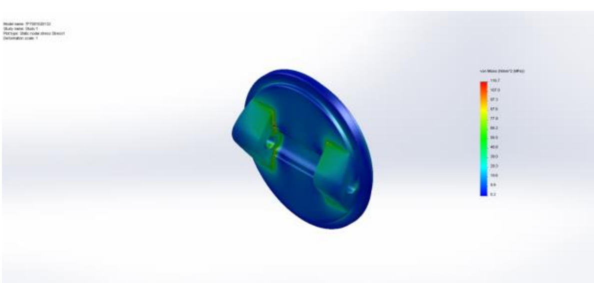
Figure5.2.1 (a) Von misses stress analysis of shaft (b) Average stress of shaft at delicate region

Fig. 5.2.2: 0.18% yielding of shaft

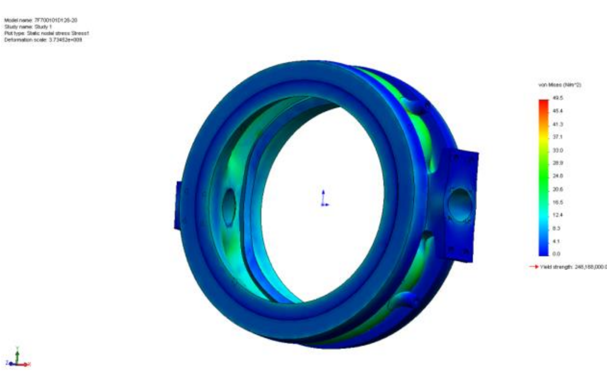
5.3 Seal stack, Body and Disc

The von misses stress analysis of seal stack disc and body is as shown in figure 5.2.3.

The simulation software is used to check stresses induced in components and it was compared with yield stresses.



(a)



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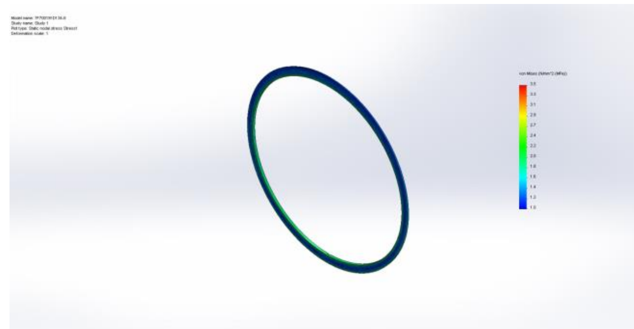

(c)

Fig 5.2.3 Von misses stress analysis of (a) Disc (b) body and (c) seal stack

No	component name	von misses stress(MPa)	Yield stress (MPa)	Remark
1	Seal stack	3.5	69	Safe
2	Disc	116.738	487.7	Safe
3	Body	49.5	248.168	Safe

As the von misses stress of all optimized components is less than yield stresses then all are safe for manufacturing.

6 EXPERIMENTAL RESULTS

A standard pressure value for triple offset butterfly valve is shown in table (6.1.1)

Table 6.1.1 Testing parameter

No	Testing	Testing Pressure (Kg/cm²)
1	Hydro shell testing	31
2	Hydro seat (seal) testing	22
3	Air seat (Seal) testing	6

6.1 Experimental reading

The torque of triple offset butterfly valve was measured by torque range with addition of mechanical advantage.
Total torque of triple offset butterfly valve
= Mechanical advantage * torque range value
=13853 N-m

Table 6.1.2 Experimental reading

No	Testing	Reading	Criteria	Remark
1	Hydro shell testing	No leakage		Safe
2	Hydro seat testing	0.4 ml	<6 ml	Safe
3	Air seat testing	166 droplet	<441 droplet in 2 min	Safe
4	Torque	13853 Nm		


(a)

(b)

Figure 6.1.1 (a) Testing setup (b) Hydro shell testing



(a) (b)

(c)

Figure 6.1.2 (a) Hydro seat testing (b) 0.4 ml leakage (c) air seat testing

7 RESULTS AND DISCUSSION

Shaft was designed for water media with 24000 N-m, but it takes only 13853 N-m, but gear box selected for worst condition and it was calculated by equation
Total torque for gear selection = valve torque * Service factor * FOS

From given data and company standards, Service factor (S.F) for simple on off condition and Media factor (M.F) for gas, dirty unfiltered water was selected on the basis of application.

So total torque for gear selection = 22857.45 N-m. So according to worm gear selection standards, the PAC-08SGC81worm gear box can be proposed.

The optimized dimension of different component is as follow in table (7.3.1) and the results are shown in table (7.3.2).

Table 7.3.1 Modified parameter

Parameter	Old (mm)	New (mm)
Seal thickness (t)	14.28	9.2264
Diameter of shaft (d)	92	88.9
Eccentricity(ε)	6.14	4.75
valve diameter or major axis	666	658.8
body thickness	24	20

Torque Reading

Table 7.3.2 torque comparisons

No	Torque name	Torque (N-m)
1	Theoretical Torque of existing design	15757.56
2	Theoretical Torque of Optimized design	12908.513
3	Reduction in Torque	2849.047
4	Experimental optimized torque	13853
5	error between experimental torque and optimized torque	6.82%

8. CONCLUSION

In this present project, work on optimization of torque of a Triple offset butterfly valve (BFV TCBV 28-150 CLASS) is done. This is done by employing reverse engineering on the dimensions of the shaft; seal stack thickness and second offset value of a triple offset butterfly valve. At first the torque was calculated based on old design parameters and parameters for optimization were identified. Such parameters were optimized by changing dimension of component with analytical calculation. Torque formulae were modified on the basis of elliptical cross section. It was analyzed by solidworks analysis and found that the von misses stresses is less than yield strength of material so optimized components were safe for manufactured. Experimental reading proved that the torque of BFV TCBV 28-150 CLASS is minimized by optimizing the seal stack thickness, shaft diameter and 2nd offset value. The error between theoretical torque value and experimental reading is 6.81% which is acceptable. This report proposed the PAC-08SGC81worm gear box.

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