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RESEARCH

Estimation and Optimization of Worm Drive Efficiency using Taguchi and Symmetric Quasi-D Optimal Methods

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A B S T R A C T

In this study, the Taguchi and symmetric quasi-D optimal methods were applied to comprehensively evaluate the influence of a wide range of factors affecting worm drive efficiency: oil kinematic viscosity υ, friction coefficient f, rotational speed n, speed ratio u, output torque T_2, module m, and diameter factor q, thereby choosing an appropriate method when studying the efficiency of mechanical drives and also choosing an optimal parameter domain to ensure maximum efficiency when designing, manufacturing, and operating the worm drive. Quadratic regression equations were obtained for the most influential factors, and the results showed a high goodness-of-fit and accuracy. Experimental models were developed to evaluate and compare the effects of the factors on efficiency. Overall, the regression equations can be used to predict the efficiency of worm drives and to select the parameters that improve the efficiency of worm drives or help develop new worm drive products. The worm testing system can be used to test the efficiency of worm drives or develop new worm drive products.

1. INTRODUCTION

Worm drives have gears with nonintersecting and perpendicular axes. Owing to their self-locking ability, worm drives are commonly used for lifting machines such as cranes and winches, but this advantage is coupled with a high sliding velocity between the meshing gear teeth, which causes higher friction loss than other gear reducers. Another disadvantage is the low efficiency (50%–93%) of worm drives; therefore, they are often used in small and medium power ranges (<60kW). Increasing the speed of gears in worm drives creates conditions favorable for forming a lubricating layer in the meshing gears, which reduces friction.

The efficiency of worm drives is crucial to their design. Previous calculations can assist engineers in comparing different solutions and selecting the best one for a kinematic scheme. It is essential to comprehensively study the influencing factors and determine the appropriate efficiency during the design process. Additionally, determining the combined efficiency of worm and gear drives in two-stage and multistage reducers has important scientific and practical applications.

Various researchers have studied the worm drive efficiency and influencing factors. Magyar and Sauer [1] created formulas to calculate the power loss due to the influencing factors and used a test bench to compare measurements and simulations. They demonstrated a general agreement between measured and simulated efficiencies, which indicated that their simulation is suitable for analyzing each loss component and optimizing the worm drive efficiency. Mautner et al. [2] used a testing kit to evaluate the effect of the oil viscosity and type on the efficiency of a large worm drive with a center distance of 315 mm. The measured efficiency was higher for high-viscosity synthetic oil (ISO VG 460) than for low-viscosity synthetic oil (ISO VG 220). Siebert [3] and Muminović et al. [4] emphasized the advantages of using synthetic oil instead of mineral oil to improve efficiency.

Several studies have proposed and experimentally proven correlations between the influencing factors and efficiency in case of worm drives. The MEGT laboratory constructed a test bench to study the worm drive efficiency, and their results showed that the efficiency is improved via good lubrication, which increases the input rotational speed and reduces power losses [3]. The power losses from gear meshing are most affected by changes in speed because increasing the speed decreases the friction coefficient of the engagement, which reduces power losses. The bearing losses and no-load power losses are also reduced, but their effect on the total loss is negligible [5]. A common practice is limiting worm drives to a rotational speed of <2500 rpm because high speeds remove oil from their surfaces [6]. The optimal rotational speed for worm drive efficiency is 1500–3000 rpm for all types of lubricants.

The relationship between the worm lead angle γ, diameter factor q, and thread number z_1 is given by tan $\gamma = z_1/q$. To improve the worm drive efficiency, γ can be increased by increasing z_1. Increasing γ must also consider limits on the thread trimming, module, and size to balance the parameters. Increasing γ reduces the friction coefficient, which improves transmission efficiency. However, in practice, γ is rarely greater than 20°–25° because the correspondng increase in efficiency is negligible; moreover, increasing z_1 increases the complexity of thread fabrication [5] and decreases the transmission ratio. The worm drive efficiency is the highest if the diameter factor q is low and module m is high. The reduction in q is limited by the bending strength of the worm shaft [7]. The basic series of q values is 8, 10, 12.5, 16, 20, and 25. The supplementary series of q values is 7.1, 9, 11.2, 14, and 18, and the values of 7, 11, and 12 are also used in some cases. Lower q values are used to design high-speed worm drives to avoid high peripheral velocities. Higher q values are used for worm drives with high-speed ratios or a large distance between bearings to ensure sufficient rigidity. The ratio q/z_2 is usually set to 0.22–0.4, where z_2 is the number of gear teeth.

The effect of the gear ratio u on the transmission efficiency should also be considered. Decreasing u increases the efficiency. Excessively large u can result in self-locking transmission (efficiency η < 50%) [8]. A gear ratio u = 10–80 is considered normal for worm drives. The acceptable gear ratios u for worm drives are standardized and should not deviate by >4%. To limit the range of tools required to cut wheels, the elements of worm gears are standardized. Thread numbers z_1 = 1, 2, and 4 correspond to speed ratios u of 30–80, 16–30, and 8–16, respectively. In some cases, z_1 can be 3 or 6. The standard may be disregarded if separate wheels are cut using a fly cutter, or when a worm gear has a strictly defined design and dimensions. The number of teeth z_2 should not be <22–26; otherwise, the contact surface will noticeably diminish.

Mautner et al. [8] showed that increasing the output shaft torque T_2 improves transmission efficiency. Load-dependent losses include efficiency losses of worm drives and bearings, but the bearing effect results in a nearly constant loss ratio. Other power loss components include idling, sealing, and oil churning, although these are independent of the load.

The friction coefficient f' affects the power loss due to meshing, which is the main cause of transmission loss under a load. Increasing the friction coefficient increases the meshing loss, which reduces the transmission efficiency. The friction coefficient is affected by many factors, including the sliding velocity, the surface roughness of the worm gear, the material of the worm, and the amount of lubricant oil. The most important influencing factors are the sliding velocity v_s and surface roughness R_a. The friction coefficient f' can be approximated [10] $f' = 0.048/v_s^{0.36}$ (steel-bronze) or $f' = 0.06/v_s^{0.36}$ (steel-cast iron). Increasing the sliding velocity v_s decreases the friction coefficient f' because the lubrication mode of the mesh pair changes from semi-wet friction to wet friction.

The materials of the worm and worm gear are important factors for power loss. In almost all cases, the worm is made of a material harder than that of the worm gear. The worm wheel should be made of bronze to ensure sufficient hardness. According to DIN 3996 [14], the copper alloyCuSn12Ni affords a lower power loss than the gray iron GJS 400. This is demonstrated by reduced power loss during the mating process (i.e., power loss coefficient) [9]. Furthermore, the materials have a noticeable effect on the friction coefficient f'. Losses are minimized in case of case-hardened, finely polished steel worms and gears with a tin–phosphor bronze rim.

For drives with cast iron gears and steel worms, the friction angle can be within 3.5°–6°. Smaller values should be considered when the sliding velocity exceeds 1–2 m/s. For gears with a bronze wheel rim and steel worm, the friction coefficient (f') data were collected from experiments on worm drives operating with antifriction bearings.

Worm drive lubrication is a complicated issue that affects transmission efficiency. AGMA 6034-B92 [12] and DIN 3996 [14] did not consider the influence of the lubricant type and surface roughness on worm efficiency. The lubricant influences the bearing efficiency primarily by reducing power losses, which include churning losses and friction losses in lubrication regimes. For the same input rotational speed n_1 and output torque T_2, increasing the kinematic viscosity of the lubricant improves the efficiency of worm drives [2].

Losses can also increase by increasing the temperature, which is affected by heat dissipation and power loss inside the gearbox. Heat dissipation is handled in relation to the surface area of the gearbox, the speed of the gears, and the method by which it is cooled. The gear speed and oil stirring of the lubricant considerably contribute to heat dissipation, particularly in case of large gears running at medium or high speeds. Air cooling or other means have been shown to improve efficiency for a given allowable temperature increase [11]. The dynamic oil viscosity decreases with increasing temperature.

Several standards are available for calculating worm drive efficiency, but they have many limitations. For example, AGMA6034-B92 [12], ISO/TR 14521 [13], and DIN 3996 [14] do not consider the influence of the lubricant type and surface roughness. AGMA6034-B92 [12] considers the rotational speed to be substantially important when calculating the efficiency, whereas ISO/TR 14521[13] and DIN 3996 weight [14] consider it to be substantially less important.

Miladinović et al. [28] applied the Taguchi–Grey method to study worm drive efficiency. They focused on investigating the effect of influencing factors on the output power and efficiency of a worm gear reducer. The standard orthogonal matrix L27 was chosen, and the following factors were considered: the input rotational speed n, the viscosity of the lubricant, and the current intensity. Chothani and Maniya [16] optimized the churning power loss effect for a worm gear by using a response surface methodology based on the rotatable central composite design (CCD). The CCD design was used for three variables and five levels of factors with six axial (star) points, six replicates at the central points, and eight factorial (core) points. The controlling variables were the input rotational speed, lubricant oil volume, and lubricant temperature, which were chosen based on pilot experiments and literature reviews. Chothani and Maniya [17] used the Taguchi–Grey relational analysis to determine the influence of the lubricant type and the volume and rotational speed of the worm on multiple responses of a worm gear (i.e., input torque and lubricant heating time). They designed the test rig to directly measure the input torque and lubricant heating time of a worm drive. Their experimental results showed that the input torque and lubricant heating time can be improved.

Previous studies have considered only the influence of an individual factor or a group of related factors on transmission efficiency. Therefore, a comprehensive evaluation of the factors influencing worm drive efficiency is essential. The aforementioned studies did not use the design of experiments to study the factors influencing worm drive efficiency, and they did not replace the complex formulas with quadratic polynomials for easy practical application. In this study, the Taguchi method was applied as a screening design to identify influencing factors and response surface methods with three levels for each factor to obtain quadratic polynomial regression equations, which were applied to optimize the worm drive efficiency. The results of this study can be used by designers to accurately determine the efficiency of worm drives in the design stage and to select parameter values to optimize the efficiency.

2. EFFICIENCY CALCULATION

2.1 Worm drives efficiency losses

The output shaft power P_2 of worm drives is always less than the input shaft power P_1 owing to power losses during operation, mainly caused by the geometry, friction, and material wear. The efficiency of worm drives is determined by the ratio of the output shaft power to the input shaft power and depends on the loss efficiency components (Figure 1):

$$\eta = 1 - (\eta_\text{z} + \eta_\text{b} + \eta_\text{s} + \eta_0 + \eta_\text{x}), \tag{1}$$

where η denotes the total efficiency, η_z denotes the gear meshing efficiency loss, η_b denotes the load-dependent efficiency loss of the bearings, η_s denotes the sealing efficiency loss, η_0 denotes the efficiency loss during idling, and η_x denotes other losses.

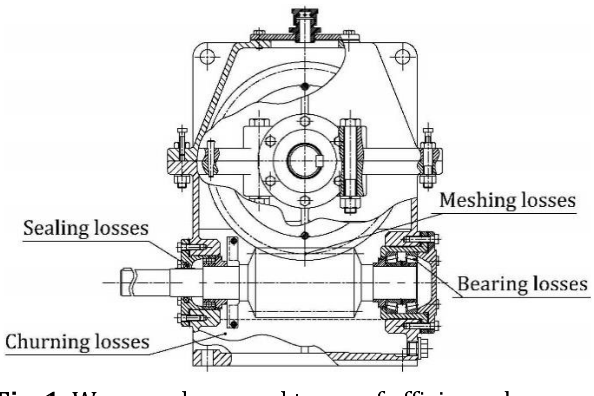


Fig. 1. Worm reducer and types of efficiency losses.

2.2 Load-dependent efficiency losses

Figure 2 shows that forces F_{t1} and F_{t2} are tangent forces acting on the worm and worm gear. It is advantageous to expand the cylinder d_1 (pitch diameter of the worm) in the plane and represent the thread of the worm as an inclined plane and the worm gear as a slider [10]; α denotes the pressure angle, and p_z denotes the pitch of the worm.

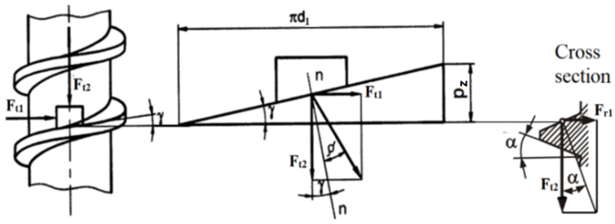


Fig. 2. Forces on the worm gear during engagement [10].

The worm drive meshing efficiency η_zm is determined using a relationship derived for the worm, which is also valid for the worm drive in general:

$$\eta_{zm} = \frac{\tan\gamma}{\tan(\gamma + \rho')} \tag{2}$$

where γ = tan (z_1/q) denotes the worm lead angle, z_1 denotes the thread number (starts) of the worm, and q denotes the diameter factor; the friction angle ρ' = arctan f', where $f' = f_\text{zm}/\cos\alpha$ denotes the corresponding coefficient of friction, α denotes the pressure angle (Figure 2), and f_zm is the average friction coefficient for the sliding surface between the worm and worm gear.

The efficiency loss due to friction during engagement can be determined from the worm lead angle and friction coefficient of the worm pair [1, 10, 18, 19]:

$$\eta_\text{z} = 1 - \eta_\text{zm} = 1 - \frac{\tan\gamma}{\tan(\gamma + \rho')} \tag{3}$$

The efficiency loss η_z increases with the lead angle up to $\gamma = 45° - \rho'/2$. Further increasing the lead angle can cause the efficiency loss η_z to start increasing.

The friction angle ρ' and the corresponding coefficient of friction f' decrease sharply with an increase in the sliding velocity v_s. This can be attributed to the conditions associated with oil wedge formation in the gears. Lower values for

ρ' and f' are suitable for a hardened, grounded, and buffed worm operating under heavily lubricated conditions.

The bearing loss efficiency η_b is determined as follows:

$$\eta_b = \frac{P_{VL}}{P_1} \tag{4}$$

P_{VL} is the power lost by the bearing (W) and can be determined by [9]:

$$P_{VL} = M \cdot n_i \cdot \frac{\pi}{30} \cdot 10^{-3} \tag{5}$$

where n_i is the rotational speed of the shaft (rpm) and M is the total frictional moment given by [20]:

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{6}$$

where M_{rr} and M_{sl} are frictional moments of rolling and sliding, respectively, and M_{seal} and M_{drag} are the seal and drag losses (N·mm), respectively. M_{rr} is calculated as follows:

$$M_{rr} = G_{rr}(\upsilon \cdot n)^{0.6} \tag{7}$$

where $G_{rr} = 1.03 \cdot 10^{-6} {d_m}^{1.83} {F_a}^{0.54}$ and $G_{rr} = R_1 {d_m}^{2.38}(F_r + R_2 Y F_a)^{0.31}$ for thrust ball bearings and tapered roller bearings, respectively; R_1 and R_2 are geometric constants for M_{rr} [20]; $d_m = 0.5(d + D)$ is the mean bearing diameter (mm); υ is the kinematic viscosity of the lubricant at the operating temperature (mm²/s) (the base oil viscosity for grease lubrication); F_r is the radial load (N); and F_a is the axial load (N).

Moment M_{sl} is calculated as follows:

$$M_{sl} = G_{sl} \cdot f_{sl,} \tag{8}$$

where G_{sl} is a variable dependent on the bearing type, mean bearing diameter d_m, radial load F_r, and axial load F_a. $G_{sl} = 1.6 \cdot 10^{-2} {d_m}^{0.05} F_a^{\frac{4}{3}} \cdot 0.05$ and $G_{sl} = S_1 {d_m}^{0.82} \cdot (F_r + S_2 Y F_a) \cdot 0.002$ for thrust ball bearings and tapered roller bearings, respectively; S_1 and S_2 are geometric constants for moment M_{sl} of tapered roller bearings [20].

The sliding friction coefficient f_{sl} can be set according to the type of lubrication: 0.05 for mineral oils, 0.04 for synthetic oils, and 0.1 for transmission fluids. Moreover, f_{sl} can be set to 0.02 and 0.002 for cylindrical and tapered roller bearings, respectively. R_1, R_2, S_1, and S_2 are geometric constants for the sliding frictional moment of tapered roller bearings [20]. Moments M_{seal} and M_{drag} can be ignored because their losses are negligible.

2.3 Nonload-dependent efficiency losses

Nonload-dependent efficiency losses include those due to sealing η_s, idling η_0, and other losses. Sealing efficiency loss η_s can be determined as follows:

$$\eta_s = \frac{P_{VD}}{P_1} \tag{9}$$

where P_{VD} is the sealing power loss (W). According to [13], P_{VD} can be calculated as follows:

$$P_{VD} = 7.69 \cdot 10^{-6} d_i^2 \cdot n_i \tag{10}$$

where d_i is the internal diameter of the seal (mm) and n_i is the rotational speed (rpm).

Idling efficiency loss η_0 can be determined as follows:

$$\eta_0 = \frac{P_{V0}}{P_1} \tag{11}$$

P_{V0} is the no-load power loss (W) and can be calculated by [13]:

$$P_{V0} = 10^{-7} a \cdot \left(\frac{n_1}{60}\right)^{4/3} \cdot \left(\frac{\upsilon_{40}}{1.83} + 90\right) \tag{12}$$

where a is the center distance (mm).

Other efficiency losses η_x represent other factors affecting the overall efficiency of the worm drive, such as oil stirring and temperature. The power loss due to oil stirring in worm drives has not been accurately determined and verified. For a worm under a worm gear with a small volume of lubricant, the worm is completely immersed in the lubricating oil during operation and moves at high speed. The resulting power loss due to oil stirring is significant. When the worm is on top, the worm gear is immersed in oil and rotates at a slow speed; therefore, the power loss due to oil stirring is considerably small. Thus, the power loss due to oil stirring is often neglected, but it can be calculated as follows [21]:

$$P_{stirr} = \rho_{oil} \cdot \omega^3 \cdot b_2 \cdot {r_2}^4 \cdot C_m \tag{13}$$

where ω is the angular velocity (rad/s), ρ_{oil} is the density of the lubricating oil (kg/m³), b_2 is the face width of the gear (mm), and r_2 is the pitch circle radius of the gear (mm). C_m is a dimensionless churning torque coefficient [21], which is expressed as follows:

$$C_\mathrm{m} = 17.08\left(\frac{h}{a}\right)^{-0.13} \cdot \left(\frac{V}{a^3}\right)^{-0.28} \cdot R_\mathrm{e}{}^{-0.9} \cdot F_\mathrm{R}{}^{-0.38} \cdot u^{-0.08} \quad (14)$$

where h is the height of the oil level from the gear base (mm), V is the total volume of the lubricant (m^3), F_R is the Froude number, and R_e is the Reynolds number.

F_R is the ratio between the forces of inertia and gravity (g = 9.81 m/s^2):

$$F_\mathrm{R} = \frac{\omega^2 r_2}{g} \quad (15)$$

Reynolds number R_e is expressed as follows:

$$R_\mathrm{e} = \frac{\omega r_2 b_2}{V} \quad (16)$$

The efficiency loss due to oil stirring is expressed as follows:

$$\eta_\mathrm{stirr} = \frac{P_\mathrm{stirr}}{P_1} \quad (17)$$

where P_{stirr} is the stirring power loss (W). This study assumed that $\eta_x = \eta_{stirr}$. Moreover, the aforementioned equations are used to calculate the worm drive efficiency.

From the aforementioned formulas, it can be seen that the rotational speed n, speed ratio u, output torque T_2, module m, oil kinematic viscosity υ, friction coefficient f, and diameter factor q have an impact on the efficiency of the worm drive.

3. DESIGN OF EXPERIMENTS

In this study, different response surface methods have been conducted to replace the formulas in Section 2 with quadratic regression equations. Thus, working and geometrical parameters could be chosen to improve the overall efficiency of worm drives. This progress consists of two stages: screening design and response surface methods.

In Stage 1, a screening design experiment was performed to identify the most influential factors. Many screening design methods are available, such as balanced design, Plackett–Burman design, random balance, and expert opinion. The Taguchi method is used for screening design in this study. The experimental results were applied to the regression analysis and analysis of variance (ANOVA) to select the most influential factors affecting the output parameters.

In Stage 2, the selected influencing factors were used with response surface methods to develop quadratic regression equations for analyzing and finding domains with optimal efficiency. This required performing experiments so that each factor varied by at least three levels.

In practice, describing the optimum region of a response surface using a linear regression equation is not possible because this area has a substantial curvature. The optimum region can be described by higher-order polynomials, such as the following second-order equation:

$$y = b_0 + \sum_{i=1}^{k} b_i x_i + \sum_{i=1}^{k} b_{ii} x_i^2 + \sum_{\substack{i\neq j\\ i,j=1}}^{k} b_{ij} x_i\, x_j \quad (18)$$

Five influencing factors were considered: the speed ratio u, diameter factor q, output torque T_2, rotational speed n, and friction coefficient f. The output was the worm drive efficiency η. The total number of coefficients of the regression equation was given by $p = 1 + 2\cdot 5 + 4\cdot 5/2 = 21$.

Symmetric compositional plans such as the face-centered central composite design (FCCCD), Box–Wilson plan, and Box–Hunter plan consist of a core and star points. The core is a full factorial experiment (FFE) (type 2^k) or fractional factorial experiment (FRFE) (type 2^{k-p}). The star points have a shoulder α, where an FCCCD plan with three-level factors has α = ±1. The experimental accuracy improves as the number of factors is increased. The advantage of this approach is the compositionality and symmetry [22,23]. Compositional planning is not always convenient. It is often more convenient and economical to plan and implement a large series of experiments at once than several small series in succession, particularly in cases where the response surface in the factor space being studied is nonlinear. In many cases, the use of a noncompositional plan is associated with higher efficiency, such as when the response surface area is known to have nonlinear characteristics [24,25]. The Box–Behnken design and Pesochinsky's symmetric quasi-D optimal methods of experiments can be used to model the response surface and is not based on FFE or FRFE. This design requires three levels for each factor. The Box–Behnken design makes it possible to study the effects of various factors in sequence if the other factors are maintained constant. The authors [26] previously used the Box–Behnken response surface methodology to study the influence of different factors on the efficiency of helical gears. In this study, Pesochinsky's symmetric quasi-D optimal methods were used to compare the calculated and analyzed results with the Box–Behnken and FCCCD methods. Table 1 presents the number of experiments N and the block of the symmetric quasi-D-optimal, Box–Behnken, and FCCCD methods. The coefficients of a regression equation can be determined according to the following matrix:

$$B = (X^T X)^{-1}(X^T Y), \tag{19}$$

where X is the design matrix, X^T is the transpose of X, and Y is the output matrix.

The R-square (R^2) value measures the goodness-of-fit of regression models and can be calculated as follows:

$$R^2 = 1 - ESS/TSS \tag{20}$$

ESS is the explained sum of squares, and TSS is the total sum of squares:

$$ESS = \sum_{\mathrm{i}=1}^{n} (\hat{y}_{\mathrm{i}} - \bar{y})^2 \ and \ \ TTS = \sum_{\mathrm{i}=1}^{n} (y_{\mathrm{i}} - \bar{y})^2 \tag{21}$$

Thus, quadratic regression equations were obtained using the quasi-D optimal method. The obtained regression equation was compared with those obtained using the FCCCD and Box–Behnken design methods based on the R^2 values.

Table 1. Number of experiments N and blocks for the FCCCD, Box–Bẹhnken, and quasi-D optimal methods with five factors.

Method	Block	Group					Total
		Factors with 0	Factors with ±1	FFE and FRFE	Replicates	Numbers in blocks	
FCCCD	1	-	x_1, x_2, x_3, x_4, x_5	2^5	1	32	45
	2	x_2, x_3, x_4, x_5	x_1	2^1	1	2	
	3	x_1, x_3, x_4, x_5	x_2	2^1	1	2	
	4	x_1, x_2, x_4, x_5	x_3	2^1	1	2	
	5	x_1, x_2, x_3, x_5	x_4	2^1	1	2	
	6	x_1, x_2, x_3, x_4	x_5	2^1	1	2	
	Center	x_1, x_2, x_3, x_4, x_5	-	-	3	3	
Box–Behnken	1	x_3, x_4, x_5	x_1, x_2	2^2	1	4	46
	2	x_2, x_4, x_5	x_1, x_3	2^2	1	4	
	3	x_2, x_3, x_5	x_1, x_4	2^2	1	4	
	4	x_2, x_3, x_4	x_1, x_5	2^2	1	4	
	5	x_1, x_4, x_5	x_2, x_3	2^2	1	4	
	6	x_1, x_3, x_5	x_2, x_4	2^2	1	4	
	7	x_1, x_3, x_4	x_2, x_5	2^2	1	4	
	8	x_1, x_2, x_5	x_3, x_4	2^2	1	4	
	9	x_1, x_2, x_4	x_3, x_5	2^2	1	4	
	10	x_1, x_2, x_3	x_4, x_5	2^2	1	4	
	Center	x_1, x_2, x_3, x_4, x_5	-	-	6	6	
Quasi-D optimal	1	-	x_1, x_2, x_3, x_4, x_5	2^{5-2}, $1 = x_1x_3x_4$ $1 = x_2x_3x_5$	1	8	50
	2	x_1	x_2, x_3, x_4, x_5	2^{4-1}, $1 = x_3x_4x_5$	1	8	
	3	x_2	x_1, x_3, x_4, x_5	2^{4-1}, $1 = -x_3x_4x_5$	1	8	
	4	x_3	x_1, x_2, x_4, x_5	2^{4-1}, $1 = -x_1x_2x_4x_5$	1	8	
	5	x_4	x_1, x_2, x_3, x_5	2^{4-1}, $1 = -x_2x_3x_5$	1	8	
	6	x_5	x_1, x_2, x_3, x_4	2^{4-1}, $1 = -x_1x_3x_4$	1	8	
	Center	x_1, x_2, x_3, x_4, x_5				2	

4. RESULTS

4.1 Screening design based on the Taguchi method

In Stage 1, screening design was performed based on the Taguchi method L27 with N = 27 experiments to select the factors affecting efficiency the most. Table 2 presents the levels and values used for the seven factors considered in this study.

To achieve results in the screening design and response surface methods for obtaining regression equations, spreadsheets in Excel are prepared according to the calculation sequence presented in Section 2. The Minitab software was used to perform ANOVA and ANOM; the results are shown in Table 3.

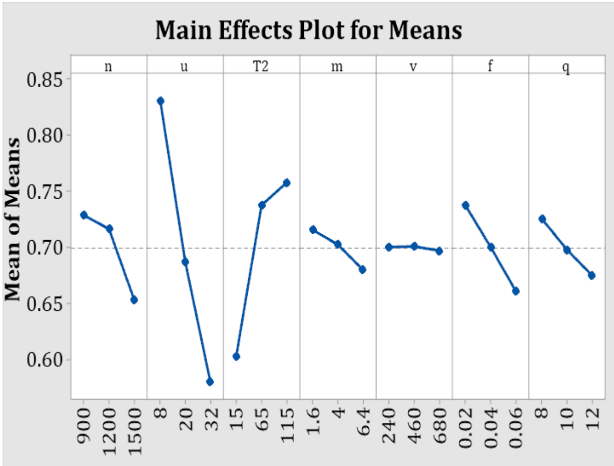


Fig. 3. Main effects plot for means.

The results in Table 3 and Figure 3 indicate that the five most influential factors on the efficiency are the speed ratio u (38.6903%), output shaft torque T_2 (23.9599%), friction coefficient f (11.8490%), rotational speed n (11.6795%), and diameter factor q (7.7658%). This result is satisfactory because the number of threads z_1 is chosen and the lead angle is calculated based on the value of the gear speed ratio u and the diameter factor q. The calculated lead angle and the coefficient of friction f strongly affected efficiency. The aforementioned five factors were used in the next step, design of experiments, to obtain regression equations with different response surface methods.

The highest value of the corresponding efficiency was obtained using a set of parameters n1u1T3m1v2f1q1, which means n = 900 rpm; u = 8; T_2 = 115 N·m; m = 1.6 mm; v = 460 mm^2/s; f = 0.02; and q = 8. Further, the efficiency value predicted by the Taguchi method is 0.969076, and the calculated result obtained using formulas (Section 2) is 0.932233. The error in estimating the efficiency according to the Taguchi method is considerably large. Therefore, another approach was used to investigate the influence of factors on efficiency and to simplify the calculation procedure. This approach was based on quadratic regression equations.

4.2 Efficiency estimation and optimization

In this section, the quasi-D-optimal method was used to study the efficiency of transmission systems and find the efficiency dependence on geometrical and working parameters in terms of quadratic regression equations. From these equations, it is pointed out that the parameter domain to ensure maximum efficiency when designing, manufacturing, and operating the worm drive. Table 4 shows the levels and ranges of the values of the selected influencing factors. The quasi-D-optimal method is firstly used for this step.

Table 2. Levels and values of seven factors considered in the Taguchi method.

N	Factors	Unit	Symbol	Levels			Intervals
				1	2	3	
1	Rotational speed	rpm	n	900	1200	1500	600
2	Speed ratio	-	u	8	20	32	24
3	Output torque	N·m	T_2	15	65	115	100
4	Module	mm	m	1.6	4	6.4	4.8
5	Oil kinematic viscosity	mm^2/s	v	240	460	680	440
6	Friction coefficient	-	f	0.02	0.04	0.06	0.01
7	Diameter factor	-	q	8	10	12	4

Table 3. ANOVA and ANOM results.

Level	***n***	***u***	***T_2***	***m***	***v***	***f***	***q***
1	0.7287	0.8311	0.6024	0.7155	0.7002	0.7374	0.7253
2	0.7163	0.6868	0.7375	0.7024	0.7007	0.7000	0.6977
3	0.6529	0.5800	0.7579	0.6799	0.6969	0.6605	0.6749
Delta	0.0758	0.2511	0.1555	0.0355	0.0038	0.0769	0.0504
%	11.6795	38.6903	23.9599	5.4700	0.5855	11.8490	7.7658
Rank	4	1	2	6	7	3	5

Table 4. Levels and value of factors for quasi-D-optimal method.

N	**Factors**	**Unit**	**Symbol**	**Code**	**Levels**			**Intervals**
					-1	0	1	
1	Speed ratio	-	*u*	x_1	8	20	32	24
2	Output torque	N·m	T_2	x_2	15	65	115	100
3	Friction coefficient	-	*f*	x_3	0.02	0.04	0.06	0.01
4	Rotational speed	rpm	*n*	x_4	900	1200	1500	600
5	Diameter factor	-	*q*	x_5	8	10	12	4

The design matrix of of the quasi-D optimal method is shown in Table 5. Based on the results obtained through the experiment conducted to obtain quadratic regression equations, spreadsheets in Excel are also prepared according to the calculation sequence presented in Section 2. Minitab software is used to process the experimental results; Table 6 presents the efficiency calculation results obtained using ANOVA for the coefficients of the regression equation.

Table 5. Design matrix and results of the quasi-D optimal method.

N^0	Code factors						Uncode factors					Result		
	x_0	x_1	x_2	x_3	x_4	x_5	*u*	T_2	*f*	*n*	*q*	η	$\hat{\eta}$	RESI
1	+1	−1	−1	−1	+1	+1	8	15	0.02	1500	12	0.80295	0.787288	0.01566
2	+1	+1	−1	−1	−1	+1	32	15	0.02	900	12	0.61483	0.592649	0.02218
3	+1	−1	+1	−1	+1	−1	8	115	0.02	1500	8	0.90621	0.909551	−0.00334
4	+1	+1	+1	−1	−1	−1	32	115	0.02	900	8	0.80053	0.801734	−0.0012
5	+1	−1	−1	+1	−1	−1	8	15	0.06	900	8	0.75258	0.745574	0.00701
6	+1	+1	−1	+1	+1	−1	32	15	0.06	1500	8	0.42636	0.409707	0.01665
7	+1	−1	+1	+1	−1	+1	8	115	0.06	900	12	0.78722	0.785875	0.00135
8	+1	+1	+1	+1	+1	+1	32	115	0.06	1500	12	0.49928	0.499663	−0.00038
9	+1	0	−1	−1	−1	+1	20	15	0.02	900	12	0.64846	0.690888	−0.04243
10	+1	0	+1	−1	−1	+1	20	115	0.02	900	12	0.84971	0.808596	0.04111
11	+1	0	−1	+1	−1	−1	20	15	0.06	900	8	0.5479	0.584624	−0.03672
12	+1	0	+1	+1	−1	−1	20	115	0.06	900	8	0.74975	0.705079	0.04467
13	+1	0	−1	−1	+1	−1	20	15	0.02	1500	8	0.62243	0.682411	−0.05998
14	+1	0	+1	−1	+1	−1	20	115	0.02	1500	8	0.87491	0.833358	0.04155
15	+1	0	−1	+1	+1	+1	20	15	0.06	1500	12	0.43134	0.479542	−0.0482
16	+1	0	+1	+1	+1	+1	20	115	0.06	1500	12	0.67328	0.627762	0.04552
17	+1	−1	0	−1	−1	−1	8	65	0.02	900	8	0.91986	0.918316	0.00154
18	+1	+1	0	−1	−1	−1	32	65	0.02	900	8	0.76247	0.773048	−0.01058
19	+1	−1	0	+1	−1	+1	8	65	0.06	900	12	0.7924	0.790622	0.00178
20	+1	+1	0	+1	−1	+1	32	65	0.06	900	12	0.47356	0.481651	−0.00809
21	+1	−1	0	−1	+1	+1	8	65	0.02	1500	12	0.89805	0.892930	0.00512
22	+1	+1	0	−1	+1	+1	32	65	0.02	1500	12	0.6842	0.689334	−0.00513
23	+1	−1	0	+1	+1	−1	8	65	0.06	1500	8	0.82269	0.817917	0.00477
24	+1	+1	0	+1	+1	−1	32	65	0.06	1500	8	0.5449	0.548791	−0.00389
25	+1	−1	−1	0	−1	+1	8	15	0.04	900	12	0.76897	0.760746	0.00822
26	+1	+1	−1	0	−1	−1	32	15	0.04	900	8	0.56108	0.552516	0.00856

27	+1	−1	+1	0	−1	−1	8	115	0.04	900	8	0.86752	0.875542	−0.00802
28	+1	+1	+1	0	−1	+1	32	115	0.04	900	12	0.60812	0.626796	−0.01868
29	+1	−1	−1	0	+1	−1	8	15	0.04	1500	8	0.77098	0.751883	0.0191
30	+1	+1	−1	0	+1	+1	32	15	0.04	1500	12	0.43996	0.436595	0.00337
31	+1	−1	+1	0	+1	+1	8	115	0.04	1500	12	0.83175	0.845713	−0.01396
32	+1	+1	+1	0	+1	−1	32	115	0.04	1500	8	0.67795	0.691018	−0.01307
33	+1	−1	−1	−1	0	−1	8	15	0.02	1200	8	0.83014	0.818464	0.01168
34	+1	+1	−1	−1	0	−1	32	15	0.02	1200	8	0.65102	0.636505	0.01451
35	+1	−1	+1	−1	0	+1	8	115	0.02	1200	12	0.89497	0.913919	−0.01895
36	+1	+1	+1	−1	0	+1	32	115	0.02	1200	12	0.73744	0.747013	−0.00957
37	+1	−1	−1	+1	0	+1	8	15	0.06	1200	12	0.70288	0.694790	0.00809
38	+1	+1	−1	+1	0	+1	32	15	0.06	1200	12	0.36213	0.349129	0.013
39	+1	−1	+1	+1	0	−1	8	115	0.06	1200	8	0.81954	0.841723	−0.02218
40	+1	+1	+1	+1	0	−1	32	115	0.06	1200	8	0.59823	0.609288	−0.01106
41	+1	−1	−1	−1	−1	0	8	15	0.02	900	10	0.83605	0.832236	0.00381
42	+1	+1	−1	−1	+1	0	32	15	0.02	1500	10	0.60312	0.587280	0.01584
43	+1	−1	+1	−1	−1	0	8	115	0.02	900	10	0.9073	0.919243	−0.01194
44	+1	+1	+1	−1	+1	0	32	115	0.02	1500	10	0.75904	0.768928	−0.00989
45	+1	−1	−1	+1	+1	0	8	15	0.06	1500	10	0.70957	0.699700	0.00987
46	+1	+1	−1	+1	−1	0	32	15	0.06	900	10	0.4256	0.415825	0.00978
47	+1	−1	+1	+1	+1	0	8	115	0.06	1500	10	0.79762	0.817219	−0.0196
48	+1	+1	+1	+1	−1	0	32	115	0.06	900	10	0.55463	0.566980	−0.01235
49	+1	0	0	0	0	0	20	65	0.04	1200	10	0.75171	0.744470	0.00724
50	+1	0	0	0	0	0	20	65	0.04	1200	10	0.75171	0.744470	0.00724

Table 6. ANOVA results for the coefficients of the regression equation.

Term	Coef	SE Coef	T-Value	P-Value	VIF
Constant	0.7445	0.0179	41.53	0.000	0.00
u	−0.11587	0.00437	−26.49	0.000	1.00
T_2	0.06717	0.00437	15.36	0.000	1.00
f	−0.07831	0.00437	−17.90	0.000	1.00
n	−0.01130	0.00437	−2.58	0.015	1.00
q	−0.02514	0.00437	−5.75	0.000	1.00
u^2	0.01497	0.00978	1.53	0.137	1.00
T_2^2	−0.04757	0.00978	−4.86	0.000	1.00
f^2	−0.00110	0.00978	−0.11	0.912	1.00
n^2	−0.00985	0.00978	−1.01	0.322	1.00
q^2	−0.00942	0.00978	−0.96	0.343	1.00
$u{\cdot}T_2$	0.01603	0.00489	3.28	0.003	1.00
$u{\cdot}f$	−0.02865	0.00489	−5.86	0.000	1.00
$u{\cdot}n$	−0.00231	0.00489	−0.47	0.640	1.00
$u{\cdot}q$	−0.01227	0.00489	−2.51	0.018	1.00
$T_2{\cdot}f$	0.00000	0.00489	0.00	1.000	1.00
$T_2{\cdot}n$	0.00763	0.00489	1.56	0.130	1.00
$T_2{\cdot}q$	−0.00068	0.00489	−0.14	0.890	1.00
$f{\cdot}n$	0.00030	0.00489	0.06	0.951	1.00
$f{\cdot}q$	−0.00947	0.00489	−1.94	0.063	1.00
$n{\cdot}q$	0.00103	0.00489	0.21	0.835	1.00

Variance inflation factors (VIFs) for all coeficients are equal to 1; those variables are independent and not correlated. The regression equation in uncoded units is expressed as follows:

$$\eta = 0.527 - 0.00489\,u + 0.002741\,T_2 + 1.00\,f + 0.000186\,n + 0.0526\,q + 0.000104\,u^2 - 0.000019\,T_2^2 - 2.7\,f^2 - 0.00236\,q^2 + 0.000027\,u{\cdot}T_2 - 0.1194\,u{\cdot}f - 0.000001\,u{\cdot}n - 0.000511\,u{\cdot}q + 0.000001\,T_2{\cdot}n + 0.000051\,f{\cdot}n - 0.237\,f{\cdot}q + 0.000002\,n{\cdot}q$$

The response optimization option in Minitab was used to determine the highest efficiency η = 0.9387, which was obtained using the following values for the selected influencing factors: q = 9.5758, u = 8, T_2 = 90.7576 N·m, f = 0.02, and n = 1112.1212 rpm. The optimal set of influencing factors is shown in Figure 4.

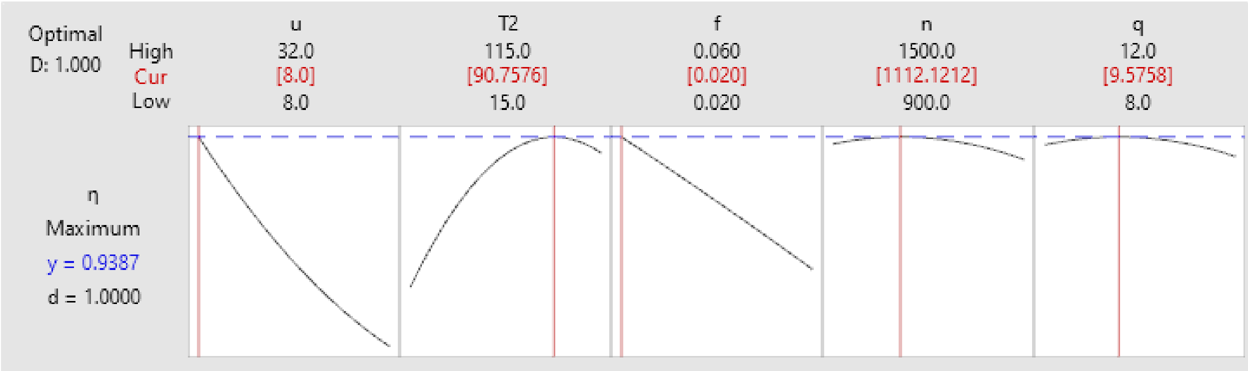


Fig. 4. Optimal set of influencing factors.

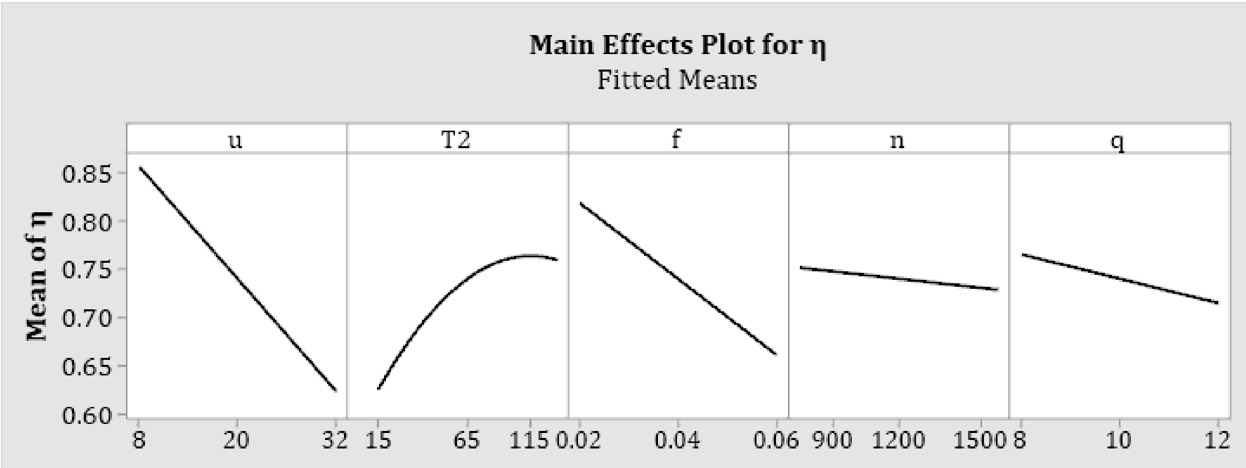


Fig. 5. Influence of factors on efficiency.

ANOVA for the regression equation of the quasi-D optimal method yielded the following results:

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	20	1.05881	0.05294	69.19	0.000

In case of the quasi-D optimal method, the residual sum of squares (RSS) is 0.0276609 and R-square is 97.95% and goodness-of-fit measure yielded good results in the model summary. Figure 5 shows the influence of each factor on the efficiency. Figures 6 and 7 depict the response surfaces, which show the dependence of efficiency on the selected influencing factors.

From the obtained regression equation, a set of working parameters and geometrical parameters can be chosen so that the efficiency reaches the highest value or is within the desired value range. Figure 8 shows the gear ratio u and torque T_2 selected to ensure that the efficiency was in the range of 0.70–0.85.

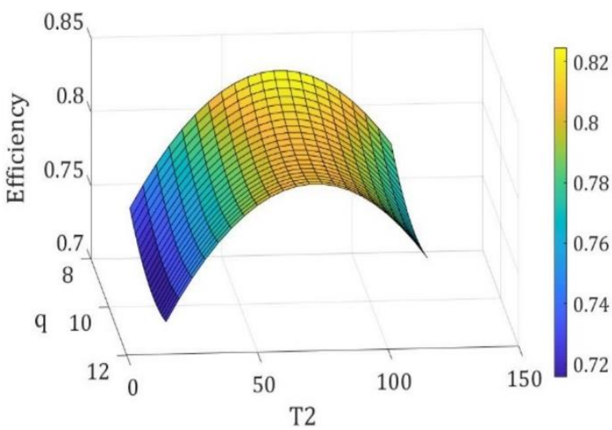


Fig. 6. Dependence of efficiency on the diameter factor q and torque T_2.

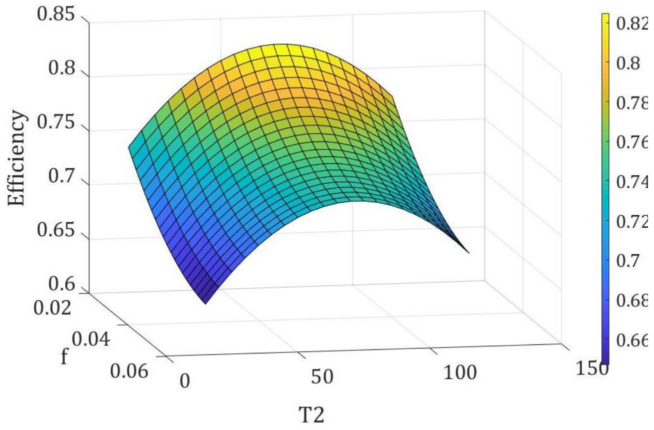


Fig. 7. Dependence of efficiency on the gear ratio u and friction coefficient f.

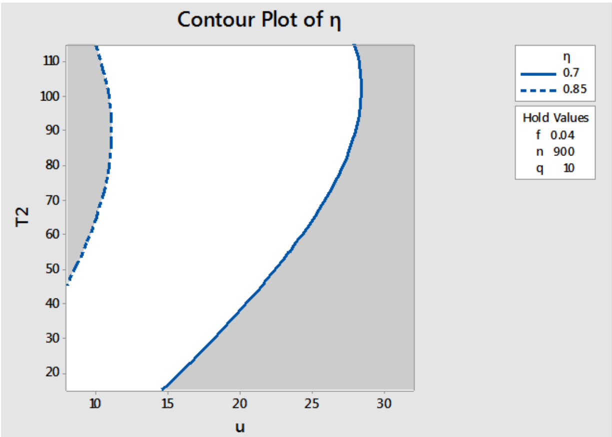


Fig. 8. Domain of reasonable parameters: speed ratio u and torque T_2.

The residual sum of squares (RSS) measures the level of variance in the error term of a regression model. A smaller RSS indicates a better fit between the model and data. Figures 9 and 10 compare the calculation results using a regression equation (quasi-D optimal method) and the formulas in Section 2.

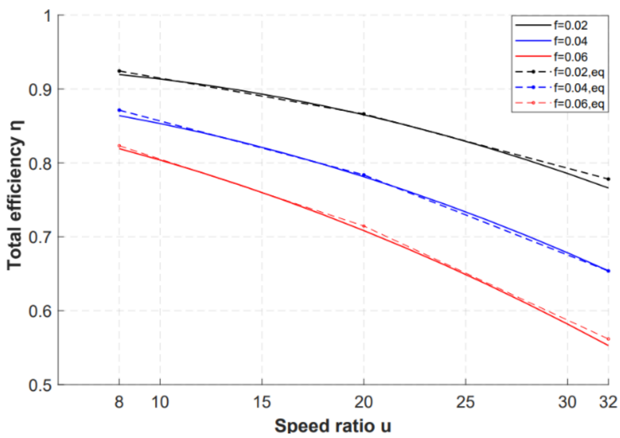


Fig. 9. Efficiency comparison based on different gear ratios u and friction coefficients f.

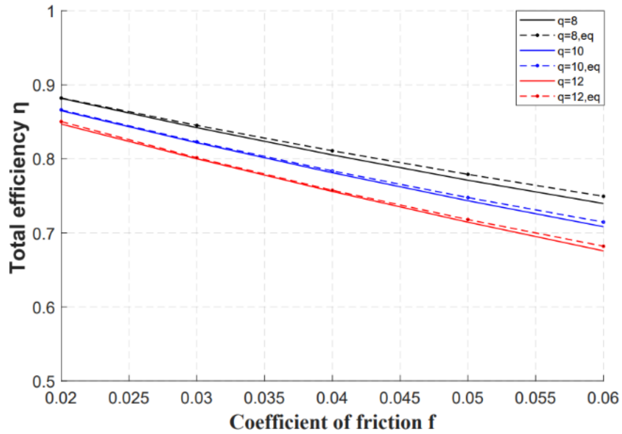


Fig. 10. Efficiency comparison based on different diameter factors q and friction coefficients f.

The regression equations of the quasi-D optimal method were used to determine the dependence of the efficiency on the selected influencing factors: the gear ratio u, diameter factor q, friction coefficient f, and output torque T_2. These were used to build efficiency dependency graphs, which can be used to easily look up and select parameter values to obtain the appropriate efficiency during the design process.

To compare the quasi-D optimal method with other response surface methods, similarly proceeded with the FCCCD and Box–Behnken methods using the numbers of experiments N and the design matrix shown in Table 7. The results of the different methods were compared with the following set of values: u = 8, q = 8, n = 900 rpm, T_2 = 90 N·m, and f = 0.02. The highest efficiency η was obtained for the quasi-D optimal method η_{max} = 0.9285, and the equations in Section 2 yielded a calculated efficiency of η_{cal} = 0.9273; thus, the error was only 0.12%. The FCCCD method and Box–Behnken method resulted in errors of 2.05% and 1.1%, respectively. The quasi-D optimal method resulted in the smallest error. Therefore, the research results according to the quasi-D optimal method is used to calculate the worm transmission efficiency and study the efficiency of other transmissions.

Based on RSS and R^2, the quasi-D optimal method is the most reasonable because it has the highest aforementioned indexes, followed by the FCCCD method and Box–Behnken methods. Moreover, the comparision between the worm transmission efficiency calculated using the regression equations of response surface methods in Table 7, with those reported by previous studies [4,8,27], in which the same input parameters were used, is shown in Table 8. Efficiency values obtained using the regression equation based on the quasi-D optimal method and efficiency values reported by previous studies have negligible deviations (0.59%, 0.68%, and 2.67%) at specific values of the factors.

Table 7. Comparison of results.

	quasi-D optimal	**FCCCD**	**Box–Behnken**
Number of experiments N	50	45	46
Regression equations	$\eta = 0.527$ $-\ 0.00489\ u$ $+\ 0.002741\ T_2$ $+\ 1.00\ f$ $+\ 0.000186\ n$ $+\ 0.0526\ q$ $+\ 0.000104\ u^2$ $-\ 0.000019\ T_2^2$ $-\ 2.7\ f^2$ $-\ 0.00236\ q^2$ $+\ 0.000027\ u{\cdot}T_2$ $-\ 0.1194\ u{\cdot}f$ $-\ 0.000001\ u{\cdot}n$ $-\ 0.000511\ u{\cdot}q$ $+\ 0.000001\ T_2{\cdot}n$ $+\ 0.000051\ f{\cdot}n$ $-\ 0.237\ f{\cdot}q$ $+\ 0.000002\ n{\cdot}q$	$\eta = 1.347$ $+\ 0.00326\ u$ $+\ 0.003825\ T_2 - 3.296\ f$ $-\ 0.000346\ n - 0.0532\ q$ $-\ 0.000097\ u^2$ $-\ 0.000028\ T_2^2 + 43.9\ f^2$ $+\ 0.00290\ q^2$ $+\ 0.000026\ u{\cdot}T_2$ $-\ 0.11738\ u{\cdot}f$ $-\ 0.000001\ u{\cdot}n$ $-\ 0.000510\ u{\cdot}q$ $-\ 0.1807\ f{\cdot}q$	$\eta = 0.602$ $+\ 0.00482\ u$ $+\ 0.00539\ T_2$ $+\ 0.12\ f$ $-\ 0.000003\ n$ $+\ 0.0238\ q$ $-\ 0.000112\ u^2 - 0.000030\ T_2^2$ $+\ 8.4\ f^2$ $-\ 0.00084\ q^2$ $+\ 0.000017\ u{\cdot}T_2 - 0.1202\ u{\cdot}f$ $-\ 0.000001\ u{\cdot}n - 0.000568\ u{\cdot}q$ $-\ 0.000001\ T_2{\cdot}q$ $-\ 0.222\ f{\cdot}q$
The residual sum of squares (*RSS*) and R^2	$S = 0.0276609$ $R^2 = 97.95\%$	$S = 0.0288219$ $R^2 = 97.87\%$	$S = 0.0290721$ $R^2 = 97.58\%$
Maximum Efficiency (by Regression equations) $u = 8$; $T_2 = 90$N·m; $f = 0.02$; $n = 900$rpm; $q = 8$	$\eta = 0.9285$ SE = 0.0017557	$\eta = 0.9478$ SE = 0.0052502	$\eta = 0.9383$ SE = 0.0285432
Maximum Efficiency (by theory formulas)	0.9273	0.9273	0.9273
Error	0.12%	2.05%	1.1%

Table 8. Comparison of efficiencies obtained using the regression equation of this study with those reported by other experimental studies.

N	**Factors–Parameters**	**Units**	**Symbol**	**References**			
				[8]	[27]	[4]	[4]
1	Speed ratio	-	u	20.5	7.5	18	18
2	Diameter factor	-	q	10	12	10	10
3	Output torque	N·m	T_2	100	50	200	300
4	Friction coefficient	-	f	0.04	0.04	0.04	0.04
5	Center distance	mm	a	100	65	90	90
6	Oil kinematic viscosity at 40°C,	mm²/s	υ_{40}		220	100	100
7	Efficiency value when experimenting in the reference papers			0.73	0.84	0.80	0.76
8	Efficiency values according to the regression equation obtained by the quasi-D optimal method			0.7500	0.8450	0.7946	0.7510
9	Error between (7) and (8)			2.67%	0.59%	0.68%	1.18%

5. EXPERIMENT TO VERIFY THE RESULTS

To verify the accuracy of the theory, an experimental model was built to measure the efficiency of the worm drive. A kinematic diagram and model are shown in Figure 11 with the principle of direct measurement of the input and output torque of the worm drive using torque sensors.

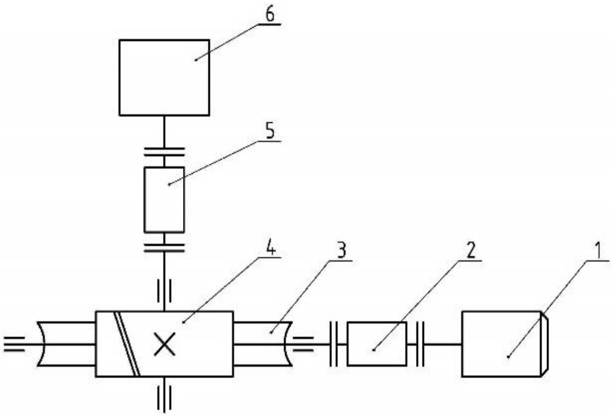


Fig. 11. Kinematic diagram and efficiency measurement kit, 1. Input motor; 2. Torque sensor 1; 3. Worm gear; 4. Worm; 5. Torque sensor; 6. Brake.

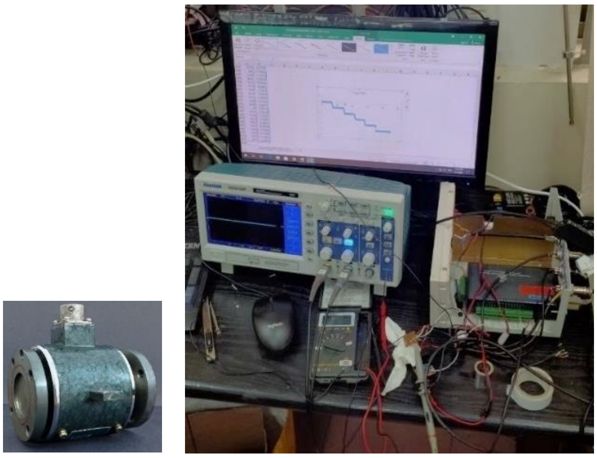
The motor torque transmitted to the reducer is determined using the torque sensor, similar to the output torque. To create load for the experimental process, a brake system using a disk brake was installed.

After determining the input and output torques on the sensor and gear ratio parameter, the following formula is applied to calculate efficiency of the system:

$$\eta = \frac{P_{\text{out}}}{P_{\text{in}}} = \frac{T_{\text{out}}}{u \cdot T_{\text{in}}} \tag{22}$$

Experimental results according to this method provide the efficiency outcome of the worm in the operating state. Herein, power losses include sliding friction upon contact between the worm and worm gear and power loss in other elements, such as rolling bearing and oil stirring. The testing kit parameters are as follows: gear ratio u = 10, number of revolutions n = 960 rpm, torque T_2 = 19.5823 Nm, coefficient of friction f = 0.035, and diameter factor q = 8.

To measure the output torque T_{out} and input T_{in}, two dynamic state torque sensors from SOHGOH KEISO Co. Ltd is installed on the system. It is necessary to perform sensor calibration (Figure 12) before starting the measurement to obtain the correct zero reference point and also before measuring ranges corresponding to the torque levels because the signal taken from the sensor is voltage (V).

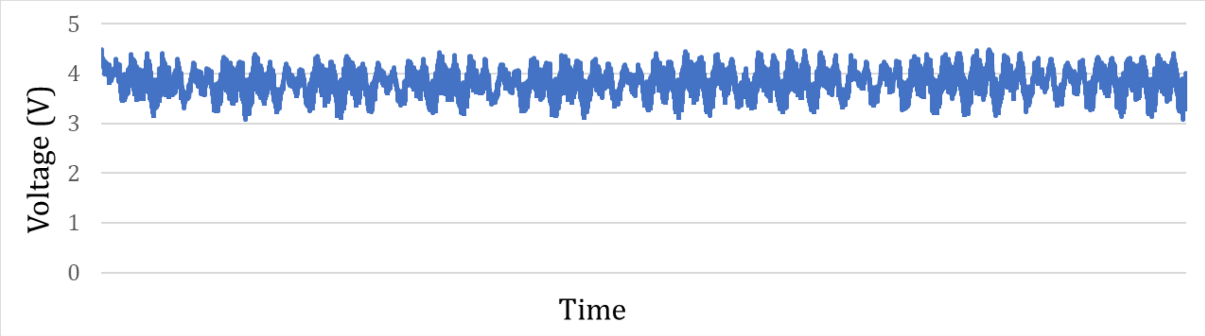


(a) (b)

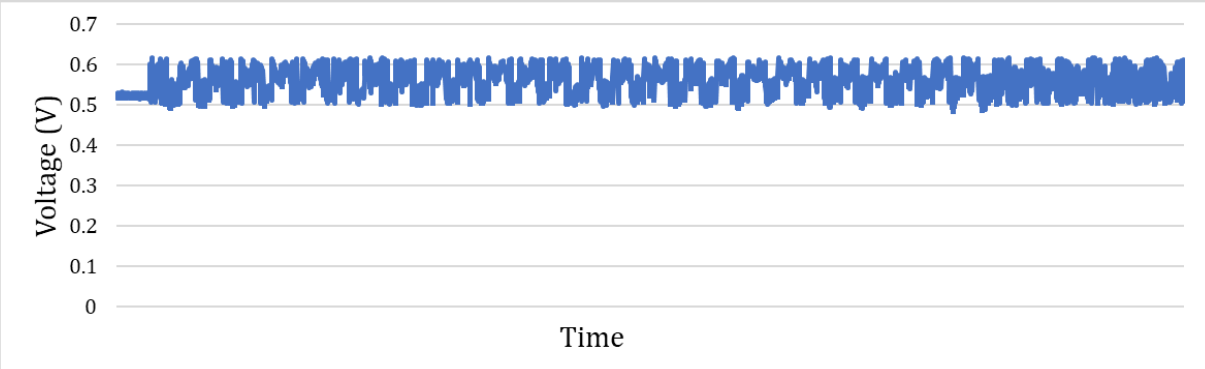
Fig. 12. Sensor (a) and signal readers (b).

The experiment thrice is performed at the output sensor position of the reducer. The first measurement was conducted in 26 seconds, the second in 30 seconds, and the third in 26 seconds. With a sampling frequency of 200 Hz, the voltage was considered only when the machine was running stably (Figure 13a).

Similarly, the experiment thrice is performed at the input sensor position of the reducer. The first measurement was conducted in 20 seconds, the second in 22 seconds, and the third in 20 seconds (Figure 13b).



(a)



(b)

Fig. 13. Wave diagram at the first measurement when the sensor is at the output (a) and input (b).

Measurement results are presented in Table 9. The comparison shows that when measured using the experiment kit, efficiencies are approximately 1.28% and 1.05%, different from efficiency values determined using the regression equations by the FCCCD (η = 0.7892) and quasi-D optimal (η = 0.7869) methods.

Table 9. Test results.

Shafts	Parameters	Measurement results			Average results	Standard deviation	Efficiency
		1	2	3			
Input	Average voltage (V)	0.5375	0.5608	0.5767	0.5583	0.0197	0.7764
	Momen T (N·m)	2.4788	2.5020	2.5861	2.5223	0.0565	
Output	Average voltage (V)	3.8262	3.7710	3.7752	3.7908	0.0307	
	Momen T (N·m)	19.7694	19.4774	21.001	19.5823	0.8090	

Errors in the regression equation are caused by different reasons, such as errors in the assembly process that lead to non-coaxiality, vibration when performing measurements using the sensor, and calculation errors. The theoretical efficiency is close to that stated in the manufacturer's catalog; therefore, the theory of factors affecting efficiency and the efficiency calculation procedure in Section 2 are considered reliable. The deviation of efficiency measured using the experiment kit and from the value determined using the regression equation according to the quasi-D optimal method is only 1.05%.

6. CONCLUSION

The aforementioned results obtained using the Taguchi method showed that the five most influential factors affecting efficiency are speed ratio u (38.6903%), output shaft torque T_2 (23.9599%), friction coefficient f (11.8490%), rotational speed n (11.6795%), and diameter factor q (7.7658%). These factors were used with response surface methods to obtain quadratic regression equations. The regression equations yielded a maximum efficiency of 0.9285 with an error of only 0.12% compared with the original formulas. Different response surface methods with three levels were considered. All obtained regression equations showed a high goodness-of-fit (R^2 > 97.5%) with low standard deviation. The best results were obtained in case of symmetric quasi-D-optimal method (R^2 = 97.95%), followed by the FCCCD method. The obtained quadratic regression equations can be used to precisely predict worm drive efficiency and determine optimal transmission efficiency during a design process, thereby helping designers to select suitable parameter values for improving overall efficiency. Furthermore, the study completed the evaluation of worm transmission efficiency for operating parameters to ensure the highest efficiency, thereby contributing to energy consumption saving and increasing life service.

An experimental kit was built to evaluate worm drive efficiency in special value ranges. The differences were 1.05% and 1.28% between the experimental results and results obtained using the quadratic regression equations according to the symmetric quasi-D-optimal and FCCCD methods, respectively. Thus, the results of this study have a significant reference value and potential for further improvement. In addition to the flexibility in the profile frame, the test bench was applicable to measuring worm drives of different capacities and sizes.

Overall, the symmetric quasi-D-optimal method can be used when studying the efficiency of mechanical drives. Although the results of this study had errors when compared with the theory, they demonstrated a considerable reference value and potential for further design improvement to implement newly developed products.

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