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Design and Construction of a Laboratory-Scale Belt Friction Testing Apparatus

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***Abstract*—*This electronic document presents the design, fabrication, and experimental evaluation of a laboratory-scale belt friction testing apparatus tailored for educational and research applications in mechanical power transmission. The limited availability of cost-effective belt friction test rigs in engineering laboratories presents a significant educational barrier. An experimental quantitative methodology was implemented to compare the friction performance of V-belts and flat belts across rotational speed variations ranging from 100 rpm to 144 rpm. The custom-built apparatus integrates a 0.5 HP AC motor, 1:20 worm gearbox, frequency inverter, pulleys, pillow block bearings, and calibrated tension measurement scales. Frictional coefficients were derived via the classical Euler-Eytelwein equation using empirical tight-side (F1) and slack-side (F2) tension forces. Experimental findings demonstrated that the apparatus operates with high precision and reliability. The average coefficient of friction was 0.2286 for the V-belt and 0.2185 for the flat belt. The wedging action within the V-pulley groove yielded higher friction capabilities and reduced slip propensity relative to flat belt contact. Escalating shaft speed from 100 rpm to 144 rpm increased linear belt velocity from 1.727 m/s to 2.480 m/s, directly affecting transmitted torque and power capacity. The developed apparatus proves to be an effective, low-cost platform for experimental teaching and empirical research.*

***Keywords*—** *Belt friction testing apparatus: coefficient of friction: Euler-Eytelwein equation: flat belt: power transmission: V-belt.*

I. INTRODUCTION

Power transmission systems constitute fundamental mechanical assemblies responsible for transferring energy from prime movers to driven machinery. Among mechanical drive configurations, belt drive systems are widely utilized owing to their structural simplicity, low initial cost, ease of maintenance, and capacity to absorb operational shock loads [1]. These systems find extensive application across industrial production lines, agricultural equipment, automotive power units, and educational engineering laboratories.

The operational efficiency of a belt drive relies fundamentally upon the interfacial friction generated between the flexible belt element and the pulley rim. Frictional force determines maximum power capacity, torque transfer efficiency, and kinematic fidelity. However, operational phenomena such as belt slip, tension decay, and localized surface wear frequently degrade system performance [2]. These behaviors are dictated by belt geometry, initial tensioning, operational angular velocity, pulley wrap angle, and surface friction characteristics.

V-belts and flat belts represent two major structural categories utilized in mechanical power transmission [3]. V-belts utilize a trapezoidal cross-section that creates a wedging mechanism inside matching pulley grooves, augmenting effective normal forces and frictional grip. Conversely, flat belts rely exclusively on direct surface contact, providing high efficiency under high-speed, light-load regimes but exhibiting greater slip vulnerability under fluctuating loads. Quantitative comparative analysis of these belt geometries under standardized laboratory conditions is essential for validating theoretical mechanics models.

Despite the ubiquitous presence of belt drives in industrial machinery, undergraduate mechanical engineering laboratories frequently lack accessible, dedicated test equipment for empirical belt friction analysis [4]. Commercial test rigs are often prohibitively expensive and constructed as closed systems, limiting student interaction with individual components. Consequently, key concepts such as belt tension dynamics, friction coefficient calculation via Euler-Eytelwein formulation, slip ratio evaluation, and torque capacity are predominantly taught through theoretical lectures without direct laboratory validation [5].

To bridge this pedagogical and practical gap, this study details the design, construction, and calibration of a laboratory-scale belt friction testing apparatus. The primary objective is to evaluate the frictional performance and power transmission metrics of V-belts and flat belts across controlled rotational speed variations, providing an adaptable, open-architecture test rig for academic research and laboratory instruction.

II. THEORITICAL FRAMEWORK AND RESEARCH MODEL

The fundamental mechanics governing flexible power transmission elements on cylindrical pulleys are modeled using the Euler-Eytelwein (Capstan) relationship [9]. For a flexible flat belt wrapped around a pulley with contact angle θ (in radians) and static coefficient of friction μ, the ratio between tight-side tension (F_1) and slack-side tension (F_2) under limiting friction conditions is defined by (1):

$$\frac{F_1}{F_2} = e^{\mu\theta} \quad (1)$$

Taking the natural logarithm of both sides yields the direct expression for the coefficient of friction given in (2):

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$$\mu = \frac{\ln(F_1/F_2)}{\theta} = \frac{2.303\log_{10}(F_1/F_2)}{\theta} \quad (2)$$

For V-belt configurations, the effective coefficient of friction (μ_{eff}) is enhanced due to the groove groove angle β (typically 34°–40°), as expressed in (3):

$$\mu_{\text{eff}} = \frac{\mu}{\sin(\beta/2)} \quad (3)$$

Linear belt speed (v) in meters per second (m/s) is calculated from pulley pitch diameter (d) in meters and shaft rotational speed (N) in rpm using (4):

$$v = \frac{\pi \cdot d \cdot N}{60} \quad (4)$$

Net power transmitted (P) in watts (W) and operational torque (T) in Newton-meters (N·m) are derived from force differentials and linear velocity according to (5) and (6):

$$P = (F_1 - F_2) \cdot v \quad (5)$$

$$T = \frac{60 \cdot P}{2 \cdot \pi \cdot N} = (F_1 - F_2) \cdot \left(\frac{d}{2}\right) \quad (6)$$

When accounting for centrifugal force at higher linear speeds, high-speed tension adjustments modify the slack-side tension expression as formulated in (7):

$$F_2 = \frac{P}{e^{\mu\theta} - 1} + \left(\frac{q}{g}\right) \cdot v^2 \quad (7)$$

where q represents mass per unit length of belt (kg/m) and g is gravitational acceleration (9.81 m/s^2) [11].

III. Research Methodology

This study employed a quantitative experimental design focused on hardware construction and empirical data collection. Tests were conducted over a four-month operational period in Bekasi Regency, West Java, Indonesia.

A. Hardware Apparatus Construction

The constructed test rig consists of a structural steel frame supporting a 0.5 HP AC induction motor driven by a variable frequency drive (VFD) speed controller. Motor output is coupled to a 1:20 ratio worm gearbox to deliver elevated torque at operational shaft speeds (100–150 rpm). Key mechanical elements include:

- Precision-machined steel shafting mounted on heavy-duty pillow block bearings.
- Interchangeable test pulleys accommodating FM-type V-belts and Habasit flat belts.
- Force measuring instrumentation comprising calibrated spring scales for tight-side tension (F_1) and hanging scales for slack-side tension (F_2).
- Non-contact digital optical tachometer for real-time rotational speed (rpm) verification.

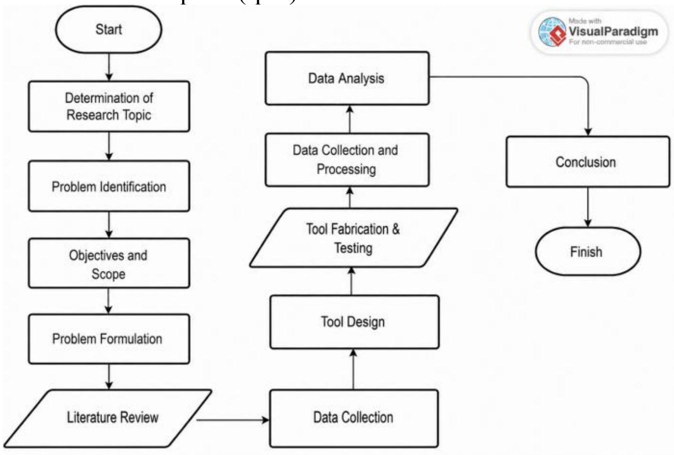

Figure 1. Schematic diagram of the laboratory-scale belt friction testing apparatus

B. Experimental Procedure

Testing was executed across standardized rotational speed intervals (100, 110, 120, 130, and 144 rpm) for both V-belt and flat belt samples under a constant contact wrap angle (θ = 90° = 1.5708 rad). Tension forces (F_1 and F_2) were logged upon reaching steady-state velocity. Derived parameters including velocity, friction coefficient, torque, and power were computed using equations (1)–(7).

IV. Results and Discussion

A. Kinematic Velocity Analysis

The linear velocity of the belt was evaluated across specified shaft speed settings. Experimental data confirmed a strict linear relationship between input angular speed and linear velocity, as summarized in Table I.

TABLE 1. Belt Velocity Experimental Data

Measured at pitch diameter d = 0.329 m

Shaft Speed (rpm)	Angular Velocity (rad/s)	Measured Belt Velocity (m/s)
100	10.47	1.727
110	11.52	1.900
120	12.57	2.072
130	13.61	2.245
144	15.08	2.480

Increasing shaft rotational speed from 100 rpm to 144 rpm produced an increase in linear belt velocity from 1.727 m/s to 2.480 m/s (an overall expansion of 43.6%). This consistent kinetic scaling confirms structural alignment with standard pitch diameter equations and validates motor controller stability.

B. Comparative Friction Coefficient Analysis

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The primary experimental metric was the effective friction coefficient calculated via tension force differentials. Table II summarizes the comparative mean friction coefficients recorded for V-belt and flat belt specimens.

TABLE 2. Summary of Friction Coefficient Results
Evaluated at wrap angle θ = 1.5708 rad

Belt Specimen Type	Mean Tight Tension F1 (N)	Mean Slack Tension F2 (N)	Mean Friction Coefficient (μ)
FM-Type V-Belt	20.00	10.80	0.2286
Habasit Flat Belt	10.90	10.50	0.2185

The experimental results show that the V-belt achieved a higher average coefficient of friction ($\mu = 0.2286$) compared to the flat belt ($\mu = 0.2185$). This 4.62% increase in effective friction is directly attributable to the trapezoidal profile of the V-belt, which wedges into the pulley groove under tension. This geometric wedging amplifies normal contact stress on the groove sidewalls without requiring higher baseline tension from the operator.

For illustrative calculation, Trial 1 flat belt data with F_1 = 19 N, F_2 = 15 N, and $\theta = 1.5708$ rad yielded:

$\mu = 2.303 \cdot \log_{10}(19 / 15) / 1.5708 = 2.303 \cdot (0.1027) / 1.5708 = 0.1504$

Under corresponding drive conditions, the V-belt sustained significantly higher tension differentials ($F_1 - F_2 = 9.2$ N) compared to the flat belt ($F_1 - F_2 = 0.4$ N), demonstrating superior power throughput prior to the onset of macro-slip.

C. Torque and Transmitted Power Evaluation

Using equations (5) and (6), transmitted mechanical power was calculated for both belt configurations. At an initial test speed of $v = 0.77$ m/s, the V-belt transferred 7.084 W of power with an effective torque of 0.509 N·m, whereas the flat belt delivered 0.308 W under equivalent tensioning settings.

Centrifugal force corrections evaluated via equation (7) for high-velocity conditions ($q = 0.126$ kg/m) yielded modified slack-side tension values of F2 = 294.28 N for the V-belt and F2 = 439.54 N for the flat belt under maximum speed conditions. These findings confirm that centrifugal effects remain negligible at low operational speeds (< 3 m/s) but must be incorporated when scaling the test rig for high-speed industrial simulations.

V. CONCLUSION

A laboratory-scale belt friction testing apparatus was successfully designed, constructed, and experimentally validated. The experimental findings lead to the following conclusions:

1. The developed apparatus operates reliably across controlled speed ranges (100–144 rpm), providing accurate measurement of rotational velocity, belt tension, and contact parameters.
2. Linear belt speed scales predictably from 1.727 m/s to 2.480 m/s across the tested rpm range, adhering to theoretical pitch kinematics.
3. The V-belt configuration demonstrated superior friction capability ($\mu = 0.2286$) compared to the flat belt ($\mu = 0.2185$) due to pulley groove wedging action, enabling higher torque capacity and lower slip propensity.
4. The constructed test rig provides a low-cost, robust educational platform for engineering laboratories to demonstrate belt friction dynamics, tension relationships, and power transmission principles.

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