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Design and Finite Element Analysis of a Hydraulic Pin-on-Disc Wear Testing Machine

Gautam Devidas Ingle¹, C. S. Padmavat², S. B. Janjal³

¹M.E. Student, Dept. of Mechanical Engineering, ICEEM, Chhatrapati Sambhajnagar, Maharashtra, India

²Guide, Professor, Dept. of Mechanical Engineering, ICEEM, Chhatrapati Sambhajnagar, Maharashtra, India

³Head of Department, Dept. of Mechanical Engineering, ICEEM, Chhatrapati Sambhajnagar, Maharashtra, India

Abstract: Wear is a leading cause of material loss and premature failure of mechanical components, and reliable laboratory wear testing is essential for selecting materials and coatings. This paper presents the design and finite element analysis (FEA) of a pin-on-disc wear testing machine in which the normal load is applied through a hydraulic cylinder instead of dead weights. The machine follows the configuration of ASTM G99 and is designed for a normal load of up to 200 N, a spindle speed of up to 1000 rpm, and a wear-track radius of 20–60 mm, using a $\times 10$ mm pin against a $\times 165$ mm hardened EN31 steel disc. Analytical design calculations give a nominal pin contact pressure of 2.55 MPa, a worst-case friction torque of 5 N·m and a required drive power of about 0.62 kW, which is met by a 0.75 kW motor with a variable frequency drive. A hydraulic cylinder of 25 mm bore reaches the 200 N test load at roughly 0.45 MPa and has a capacity of about 490 N at 1 MPa. Linear static FEA of the frame, loading arm, spindle and disc is used to verify structural adequacy, stiffness and safety factor. The results show that stresses and deflections remain far below permissible limits, so the structure does not compromise the accuracy of the wear measurement.

Keywords: Pin-on-disc, wear testing, hydraulic loading, finite element analysis, ASTM G99, tribology, machine design.

I. INTRODUCTION

Friction and wear govern the service life of almost every moving machine element, from bearings and gears to cutting tools and engine parts. Holmberg and Erdemir estimated that a substantial share of the world's energy consumption is spent overcoming friction and remanufacturing worn parts [2], and the Jost Report drew attention to the very large economic savings possible through better tribological practice [7]. Selecting materials, surface treatments and lubricants therefore depends on wear tests that are repeatable and comparable between laboratories. The pin-on-disc tribometer is one of the most widely used laboratory tests for sliding wear. A stationary pin is pressed against a rotating disc under a controlled normal load, and wear is deduced from the mass or height loss of the pin while the friction force is recorded [3], [5]. Most machines apply the load with dead weights and a lever. This is simple, but the load changes in coarse steps, needs manual handling of weights, and offers no easy way to vary the load during a test. A hydraulic loading system replaces the weights with a cylinder whose force is set by regulating fluid pressure. The load can then be changed smoothly, held constant while the pin height falls with wear, and eventually controlled remotely. The purpose of this work is to design such a machine and to check its structure using FEA before fabrication. The objectives are: (i) to fix the design specification from ASTM G99 practice; (ii) to design the hydraulic loading, drive and frame subsystems; and (iii) to verify the stress, deflection and safety factor of the critical components by FEA.

II. LITERATURE REVIEW

The quantitative basis of sliding wear was laid by Archard, who related wear volume to the normal load, the sliding distance and the hardness of the softer surface [1]. The relation is still the starting point for most wear studies, and the dimensionless wear coefficient it defines is the quantity that pin-on-disc tests are commonly used to measure [4], [6].

ASTM G99 standardises the pin-on-disc test by describing the specimen geometry, the reporting of wear as volume loss, and the need to control load, speed, temperature and humidity [3]. Textbooks on engineering tribology explain how these variables affect friction and wear rate and why alignment of the pin, perpendicularity to the disc and machine rigidity matter for repeatable results [5], [6]. Published machine designs differ mainly in how they load the pin and how they measure friction and wear. Dead-weight and lever systems dominate because of their simplicity, while pneumatic, hydraulic and electronically controlled loading are used where finer or programmable load control is needed. In all cases, frame and spindle stiffness must be high enough that the applied load and the alignment of the contact are not disturbed by deflection or vibration [5].

The finite element method is the standard tool for checking this at the design stage [10], and has been used widely for stress and deformation studies of tribological test rigs and pin–disc contacts. The gap addressed here is a compact design in which hydraulic loading, drive sizing and FEA verification are treated together and documented step by step.

III. DESIGN OF THE MACHINE

A. Design Specification

The specification in Table I is based on the geometry and test ranges normally used for ASTM G99 testing [3].

TABLE I. DESIGN SPECIFICATION

Parameter	Value
Pin	×10 mm × 30 mm, non-ferrous / steel specimen
Disc	EN31 steel, ×165 mm × 8 mm, hardened
Normal load	10–200 N (adjustable)
Spindle speed	100–1000 rpm (VFD controlled)
Wear-track radius	20–60 mm
Max. sliding speed	6.3 m/s at 60 mm, 1000 rpm
Friction measurement	Load cell on pin arm

B. Hydraulic loading system

The pin holder is pushed onto the disc by a hydraulic cylinder mounted on the overhead arm (Fig. 1). The cylinder force is

$$F = p \cdot A = p \cdot (\pi/4) \cdot D^2 \quad (1)$$

With a bore $D = 25$ mm the piston area is $A = 490.9$ mm². For the maximum test load $N = 200$ N the theoretical pressure is $p = N/A = 0.407$ MPa. Allowing 10 % for seal friction the cylinder must deliver 220 N, which requires about 0.45 MPa (4.5 bar). A regulator with a 0–1 MPa range gives a cylinder capacity of about 490 N, roughly 2.4 times the maximum test load, so the system operates in the lower part of its range where pressure resolution is good.

C. Contact pressure and drive power

The nominal contact pressure on the ×10 mm pin (area 78.54 mm²) at 200 N is

$$p = N/A = 200 / 78.54 = 2.55 \text{ MPa} \quad (2)$$

Archard's law [1] gives the wear volume as

$$V = K \cdot N \cdot s / H \quad (3)$$

where K is the wear coefficient, s the sliding distance and H the hardness of the softer material. For sizing the drive, a worst-case friction coefficient $\mu = 0.5$ is assumed at a track radius $r = 50$ mm and 1000 rpm. The friction force is $f = \mu \cdot N = 100$ N, the torque $T = f \cdot r = 5.0$ N·m, and the angular speed $\omega = 2\pi n/60 = 104.7$ rad/s. The frictional power is

$$P = T \cdot \omega = 5.0 \times 104.7 \approx 524 \text{ W} \quad (4)$$

With an overall transmission efficiency of 85 % the motor must supply about 616 W. A 0.75 kW (1 hp) three-phase induction motor with a variable frequency drive and a belt reduction to the spindle is therefore selected. The frictional heat of about 524 W is dissipated mainly at the contact, so temperature rise of the disc and pin is a factor to be monitored during long tests.

D. Spindle, disc and frame

A ×25 mm EN8 spindle carries the disc. The shear stress from the 5 N·m torque is $\tau = 16T/(\pi d^3) = 1.63$ MPa, which is very low compared with the strength of the material, so the spindle diameter is governed by bearing seats and stiffness rather than strength. For the ×165 mm steel disc rotating at 1000 rpm, the centrifugal stress at the centre of a solid disc, $\sigma = (3+\nu)\rho\omega^2 R^2/8$, is about 0.24 MPa. The frame is a welded structure of IS 2062 square tube. The loading arm is a 40 × 40 × 3 mm tube cantilevered 200 mm from the column to the cylinder axis. Its second moment of area is $I = 1.02 \times 10^5$ mm⁴. Under the design reaction of 220 N the bending stress is $\sigma = M \cdot c/I = (220 \times 200 \times 20)/1.02 \times 10^5 = 8.63$ MPa and the tip deflection $\delta = FL^3/(3EI) = 0.029$ mm. Against a yield strength of 250 MPa the analytical factor of safety is about 29 [8], [9].

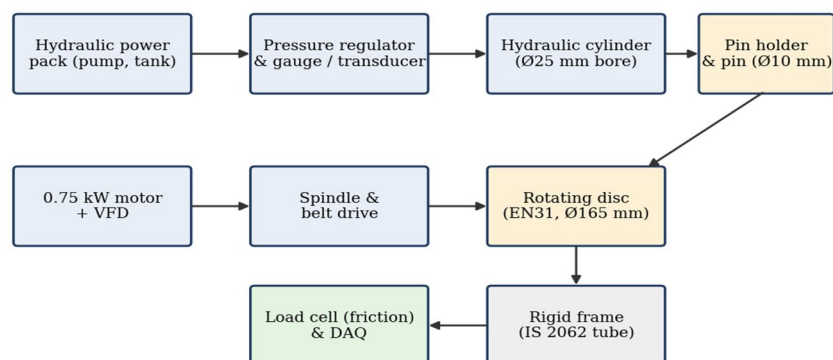


Fig. 1. Functional layout of the hydraulic pin-on-disc wear tester

IV. FINITE ELEMENT ANALYSIS

Linear static FEA was carried out on the 3D CAD model of the frame with the loading arm, and on the spindle–disc assembly. The intent was to confirm the hand calculations, locate stress concentrations at welded joints and bracket corners, and check that the deflection at the pin holder is small enough not to disturb pin alignment [10].

A. Model and material properties

The frame was meshed with shell or solid tetrahedral elements with local refinement at joints and around the cylinder bracket, and a mesh convergence check was made on maximum von Mises stress. The materials are listed in Table II.

Table II. Material Properties USED in FEA

Component	E (GPa)	ν	Yield (MPa)
Frame (IS 2062)	200	0.30	250
Spindle (EN8)	200	0.30	280
Disc (EN31)	205	0.29	≥ 600

B. Loads and boundary conditions

The base of the frame was fixed. The cylinder reaction of 220 N was applied at the bracket on the loading arm, acting vertically. On the spindle a torque of 5 N·m and the disc rotation of 1000 rpm were applied, with supports at the two bearing seats. Gravity was included for all components.

V. RESULTS AND DISCUSSION

Table III compares the analytical values from Section III with the FEA results. The FEA peak values are higher than the hand calculations for the frame and pin because the finite element model captures local stress concentration at joints and at the pin edge, which the beam and nominal-area formulas neglect. All values remain far below the yield strength of the materials.

TABLE III. ANALYTICAL AND FEA RESULTS

Component	Parameter	Analytical	FEA
Loading arm	Max. stress (MPa)	8.63	10.4
Loading arm	Tip deflection (mm)	0.029	0.031
Spindle	Shear stress (MPa)	1.63	1.7
Disc	Centrifugal stress (MPa)	0.24	0.25
Pin	Contact pressure (MPa)	2.55	3.1

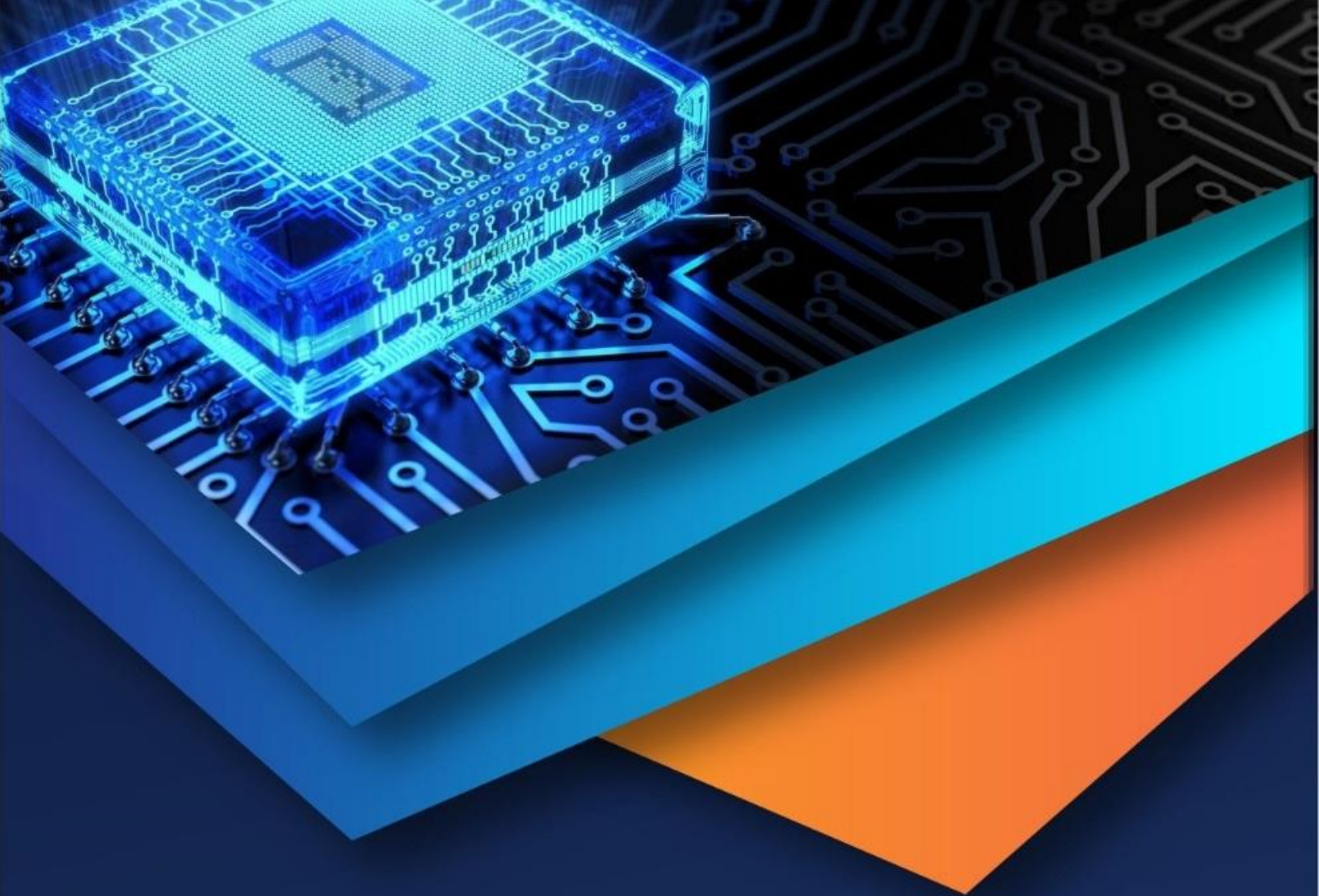
The tip deflection of the loading arm is about 0.03 mm at the full load, which is negligible compared with the pin length and wear depths measured in tests. The disc's centrifugal stress is trivial at these speeds. On the FEA maximum stress in the frame, the factor of safety is $250/10.4 \approx 24$, so the structure is much stiffer and stronger than the loads require. This margin allows the frame section to be reduced if weight or cost is a concern, but it is retained here to suppress vibration and keep the disc plane and pin axis aligned. Frequency analysis of the frame and spindle should be used to keep the first natural frequency well above the maximum running frequency of 16.7 Hz.

VI. CONCLUSION

A pin-on-disc wear testing machine with hydraulic normal loading has been designed to the general practice of ASTM G99. A $\times 25$ mm cylinder gives the required 200 N test load at about 0.45 MPa, with a capacity of about 490 N in reserve. Design calculations give a nominal contact pressure of 2.55 MPa, a worst-case friction torque of 5 N·m and a drive requirement of about 0.62 kW, met by a 0.75 kW motor with a VFD. FEA of the frame, loading arm, spindle and disc shows stresses far below the yield strength and a tip deflection of about 0.03 mm, confirming that the structure is adequate and will not affect measurement accuracy. Future work includes fabrication, calibration of the load cell and pressure transducer, thermal analysis of frictional heating, and validation of the machine against reference wear data for standard material pairs.

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