HEURISTIC MODEL OF A FRICTION AND WEAR TRIBOMETER

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ABSTRACT

A heuristic ideation technique is applied to evolve a bench, table-type pin - on - disc tribometer. For the same test load and specimen, eighteen (18) sliding speeds (ranging from 0.14 to 9.75m/s) could be simulated primarily under dry conditions but adaptable to boundry and mixed lubrication modes. The maximum end load is 995N on each pin, that is 1990N on test specimen. The friction is measured with aid of a tensometric beam load cell while wear is by weight loss assessment.

The optimisation criteria include: simplicity of operation, maintainability, laboratory ergonometrics, availability of materials,ease of manufacture, reliability and reproducibility of test results, low weight and cost, small dimensions of machine, etc.

1.0 INTRODUCTION

PREAMBLES

"Life is a friction phenomenon". Virtually every activity in this physical octave is a friction phenomenon perhaps due to the density of the octave . The sun rays do not reach the earth's surface without resistance to its motion. Earthquakes are attributed to a sudden adjustment of long built-up stresses (or internal friction) to an intolerable point.The interraction between land and sea characterised by unstable interface; and posing some problems to coastal civilisation is not without the friction drama. The bibical antedulivian civilisations and the buried continent of Atlantis are popular victims of this drama. International and Intranational friction has to rise and fall of many nations like Babylon, Persia Roman empires and more recently Soviet Union.

External and internal friction in every individual not excluding the toddling child give rise to growth, wear and tear. It is therefore not surprising that friction initially improves the perfomance of engineering machines during the running-in period and decreases same during catastrophic wear period. Industrial tribological practices try to postpone the advent of the latter period. It is the estimation of the actual resistance to relative motion between two bodies and the associated wear which forms the goal of the heuristic tribometer.

1.1 FRICTION AND WEAR EVALUATION

Friction can be measured by means of the repose angle (the inclined plane), friction torque (four-ball tester), friction force (Plint dry tester), etc or by means of load cells on a tensometric beam (figure 11)

Similarly, wear can be measured severally as in figure 1. Wear measurement by height variation (Schipper and Odi-Owei, 1992) or wear scar variation (Ossia and Odi-Owei,1998) is quantitative. Wear evaluation by the debris analysis is more qualitative than quantitative. Other qualitative methods include: Spectroscopy (Ball, 1997), and Vibration monitoring (Ossia and Odi-Owei, 1996), Barwell (1979) indicated that weight loss measurement is the most sensitive indicator of metal removal.

The heuristic model of tribometer adopts the weight loss approach for conditions where:

(a) the subject is a sub-assembly of a machine;
(b) some elements like bearings, gears, etc are tested;
(c) material samples are tested to select the optimal material.

1.2 MACHINERY SURVEY

Tribological contacts could be surface , line or point contact (figure 2). Point and line contacts are necessary for extreme pressure and antiwear conditions, while, surface contacts are for conditions similar to friction pairs like brakes, clutches, couplings, bushings, etc. Choice of contact is also governed by the desired relative motion at the mating pairs interface. Gears for instance exhibit combined rolling and sliding; hence the use of disc-on-disc test specimens as in the rolling disc tribometer (Illiuc, 1980) for the gear lubricant testing.

Several tribo-testers can be systematized based on certain criteria as in table 1 for description purposes. From the table, the Plint tester for instance is of the surface contact system of tribo testers based on the contact type criterion.

2.0 OPTIMAL IDEATION

2.1 IDENTIFICATION AND JUSTIFICATION OF NEED

To avert the ill consequences of friction and wear, it is necessary to quantify the actual friction and wear involved in a particular situation. Such measurements cannot be done without tribometers. It is therefore evident that apart from fostering tribological research, tribometers would be good tools for the optimal selection of engineering materials, improving machine design, greater machinery safety and reliability, reducing machine failure and economic loss, product quality control and assurance.

From table 1 existing machines are designed and built to test specific tribo-properties and are not without their limitations. Most often it is difficult to compare the results from different tribometers; because the systems and environments of the tests are lagely variant. It therefore become necessary to evolve a particular tribometer to test our own materials, systems and environment.

2.2 DESIRABLE TRIBOMETER FEATURES

- Capable of measuring friction and wear under dry sliding conditions, though adaptable for mixed lubrication modes;
- Capable of tribological measurements under different sliding speeds and loads for the same specimen. Easy means of speed variation is another important feature;
- Uniform contact pressure at the interface and stable contact area;
- Capable of estimating or measuring the average interface temperature;
- Capable of producing reliable and reproducible results;
- Economically cheap and good ease of realisation
- Good simplicity of operation and maintenance
- High ease of mounting and dismounting of both specimen and counter specimen requiring no skills,
- High ease of being manufactured.

2.3 CONCEPTUALISATION

Based on the need and the desired features, ideation techniques are utilised to obtain the overall tribometer lay-out (figure 3) and the broad structural chart (figure 4)

2.4 SOLUTION OPTIMISATION

2.4.1 POWER SOURCE ASSESSMENT

A synchronous electric motor is chosen because its speed would be constant and independent of the load (Huges, 1980); unlike asynchronous motors. Of course, electric motors are easily available, relatively cheap and have smaller sizes compared to other sources of power.

2.4.2 TRANSMISSION ASSESSMENT

Based on ease of manufacture, maintainability, availability, simplicity of design, space conservation, adaptability or flexibility, and simplicity of operation criteria a belt and pulley mechanical transmission is optimal as in figure 5.

Specifically, various transmission drives like chain and sprockets, gears, couplings, clutches, belts and pulleys are analysed using the criteria: ease of manufacture, maintainability, availablility, simplicity of design, space conservation, adaptability or flexibility and simplicity of operation each weighted as 10, 10, 3, 3, 4, 3, and 5 respectively to evolve the pie chart in figure 5. infact, a vee-belt provides greater traction.

2.4.3 TRIBOLOGICAL CONTACT ASSESSMENT

Analysing the three types of contacts using ease of evaluating wear volume, simplicity of realisation, initial contact area and contact pressure stability during test criteria; following the procedure explained in section 2.4.2 a surface tribological contact is the optimal solution (figure 6).

2.4.4 SURFACE CONTACT ASSESSMENT

Optimisation criteria here include: geometric limitation of the test specimen to the duration of test, similarity of test system to common practical systems, ease of test specimen preparation, adaptability to measure other wear modes, ease of realisability evaluation of sliding distance. As shown in figure 7, based on the above criteria and following the procedure explained in section 2.4.2 a pin-on-disc (face-loaded-Sc4) contact is the optimal solution.

2.4.5 WEAR TRACK ANALYSIS OF THE PIN-ON-DISC TRIBO-CONTACT

Two wear tracks can be simulated with the chosen contact, namely, (a) circular track, and (b) spiral track. These are considered under the following criteria:

(a) Sliding distance (S):

For the circular track; $S = \frac{2\pi nrt}{3600}$ ....(1)

For the spiral (Archimedes') track, according to Odi-Owei and Nwachukwu (1990);

$$S = \frac{1}{a}\left[\left(\frac{a\theta + b}{2}\right)\sqrt{(a\theta + b)^2 + a^2} + \frac{a^2}{2}\ln\left\{(a\theta + b) + \sqrt{(a\theta + b)^2 + a^2}\right\}\right] \quad \ldots\ldots (2)$$

Equation-1 is simple and less complex to evalute under experimental conditions than equation-2. The expression in equation-2 is not only complex and cumber some to handle but also contains a multi-value function - the natural logarithmic term (Landzinski, 1987 and Spiegel, 1983)

(b) Sliding speed and friction coefficient (μ)

These will be constant for the circular wear track when the spindle speed is constant; but for the spiral track every contact asperity will experience a different sliding speed and hence friction coefficient even with constant spindle speed; because of the constantly changing radius. For most engineering materials kinetic friction is at least an empirical function of sliding speed (Freeman, 1962).

Remark(s): A circular wear track will be simulated with the chosen pin-on-disc tribo-contact.

2.4.6 LOADING ASSESSMENT

With vibration sensitivity, cost of procurement, sensitivity of loading device, availability of device, simplicity, and stability of loading as optimisation criteria; following the procedure explained in section 2.4.2 a hydraulic loading system is analysed to be the optimal solution (figure 8).

2.5 OPTIMAL CONCEPTUAL MODEL

The optimal solution resulting from the above analysis is summarized as the narrow structural chart shown in figure 9 and the schematic diagram of the tribometer modelled as in figures 10 and 11.

3.0 PARTICULAR DESIGNING

Design principles of the optimal state of load, optimal material, and optimal relationship between interconnected quantities are applied to give optimal goemetrical, dynamic and material matching which evolved into the heuristic model of the friction and wear tribometer. The slider sub-assembly (figure 12) gives the tribometer its three circular wear tracks for every spindle speed; other sub-assemblies include: body sub-assembly, drive sub-assembly, and hydraulic loader sub-assembly.

The tribometer is modelled to have six different transmission paths (pulley driven) to give 18 sliding speeds, end loads and allowable contact pressure as in table-2. Tribometer specifications include:

Electric motor	-	690 rpm, 1.5KW, synchronous;
Electric supply	-	380V, 50Hz;
Minimum sliding speed	-	0.14 [m/s];
Maximum sliding speed	-	9.75[m/s];
Size of disc specimen	-	ϕ150 x 15 [mm];
Size of pin counter specimen	-	ϕ8 x 35 [mm];
Minimum wear track diameter	-	60 [mm];
Maximum wear track diameter	-	120 [mm];

Maximum permissible end load on each pin - 995 [N].

4.0 CONCLUSION AND RECOMMENDATION

- The heuristic method, though common in seeking solutions to mathematical formulation can be applied to design problems connected with researching for optimal solution; giving sound optimisation criteria.
- Within the limits of the desired tribometer features, ideation method, optimisation techniques and criteria, design assumptions, calculations and drawings; it is concluded that the need has been achieved.
- It is highly necessary that tribological investigations and practices be diversified into other areas instead of the present confine to engineering. It is expected that these would converge at, unveil or evolve some universal laws - say a universal law of friction.

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NOTATIONS

a	-	Arbitrary constant
b_s	-	Minimum wear track radius
n	-	Angular speed of the tribometer spindle [rev. per minute]
r	-	Wear track radius
t	-	Experimental test duration
F_N	-	Normal force on test pin
S	-	Sliding distance
V	-	Sliding speed
μ	-	Coefficient of friction
0	-	Angle of revolution of test specimen
ω	-	Angular speed of the tribometer spindle [rad. per second]

Table-1: Criteria of systematization of some existing tribometers

S/No	Criterion / Machine	Type of Contact			Type of Motion		Nature of Contact		Contact Area		Test Pressure		Loading Method			Wear Evaluation Method						Environment of test			Geometry of Mating Pairs							
		Surface	Line	Point	Sliding	Rolling	Dry	Lubricated	Stable	Unstable	Stable	Unstable	Gravity	Spring	Hydraulic	Direct height Variation	Artificial height Variation	Wear scar Variation	Radio-Isotopy	Weight loss	Ferrography	Controlled gas	Controlled pressure	Atmosphere	Pin - Disc	Disc - disc	Bush - flat surface	Crossed - cylinder	Ball - ball	Cylinder - cylinder	Cylinder - flat surface	Ball - flat surface
01	Plint tester (Model 97)	*			*		*		*		*				*		*							*	*							
02	Plint -4- ball			*	*			*		*		*			*			*						*	*				*			
03	Timken		*		*			*		*		*	*	*				*						*	*			*				
04	Bartel		*		*			*		*		*	*					*						*	*					*		
05	Amsler		*		*	*		*		*		*	*	*				*						*	*	*						
06	Illiuc		*		*			*		*		*	*					*				*					*					
07	NEL crossed cylinder			*	*	*		*		*		*	*					*						*				*				
08	Almen		*		*			*		*		*			*			*						*							*	
09	Vacuum pin on Disc			*	*		*			*		*	*					*					*		*							
10	Iwanowski	*	-		*		*	*	*	*	*	*		*		*									*	*							
11	Wolf - Neale			*	*		*			*		*	*							*					*	*							
12	Bethune - Waterhouse			*	*		*			*		*	*							*		*										*
13	Falex		*		*			*		*		*	*	*				*						*							*	

Table - 2: Load - Speed Characteristics of the Model

Transmission Path *	a→c ; e→h			a→c ; f→i			a→c ; g→j			b→d ; e→h			b→d ; f→i			b→d ; g→j		
Nominal Spindle Speed [rpm]	1552.50			690.00			258.75			258.75			115.00			43.13		
Wear Track No	3	2	1	3	2	1	3	2	1	3	2	1	3	2	1	3	2	1
Wear Track Radius [mm]	60	45	30	60	45	30	60	45	30	60	45	30	60	45	30	60	45	30
Nominal Sliding Speed [m/s]	9.75	7.32	4.88	4.34	3.25	2.17	1.63	1.22	0.81	1.63	1.22	0.81	0.72	0.54	0.36	0.27	0.20	0.14
Maximum Allowable End Load On Each Pin [N]	30.75	41.01	61.51	69.20	92.25	138.38	184.50	245.90	369.00	184.50	245.90	369.00	415.51	553.51	831.02	995.00	995.00	995.00
Corresponding Maximum Contact Pressure [MPa]	0.62	0.82	1.23	1.38	1.85	2.77	3.69	4.92	7.38	3.69	4.92	7.38	8.31	11.1	16.6	19.90	19.90	19.90
Corresponding Maximum Allowable Gauge Pressure [MPa]	0.031	0.042	0.063	0.070	0.094	0.141	0.188	0.250	0.376	0.188	0.250	0.376	0.423	0.564	0.846	1.013	1.013	1.013

* Numbers describing transmission path refer to the pulley numbers used in the schematic diagram in figure 10

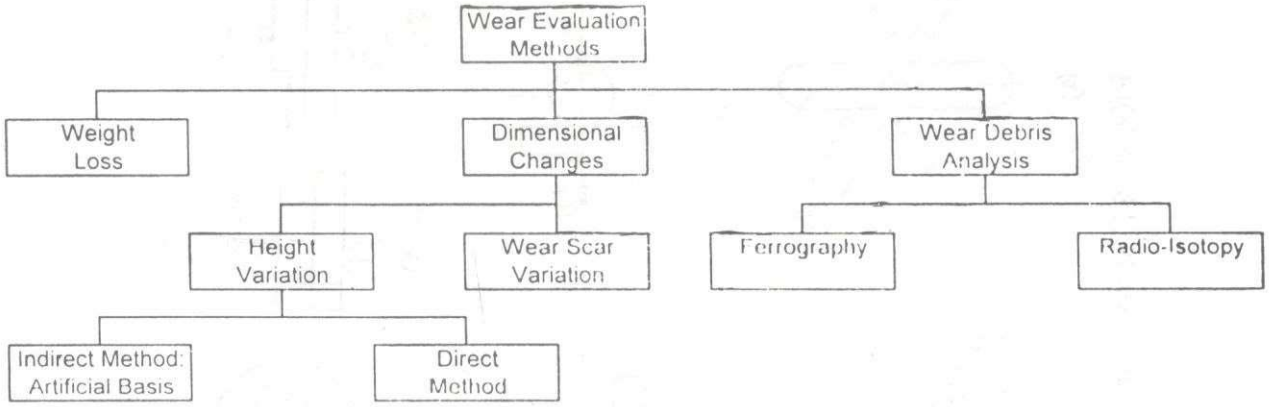


Fig. 1 Wear Evaluation Methods

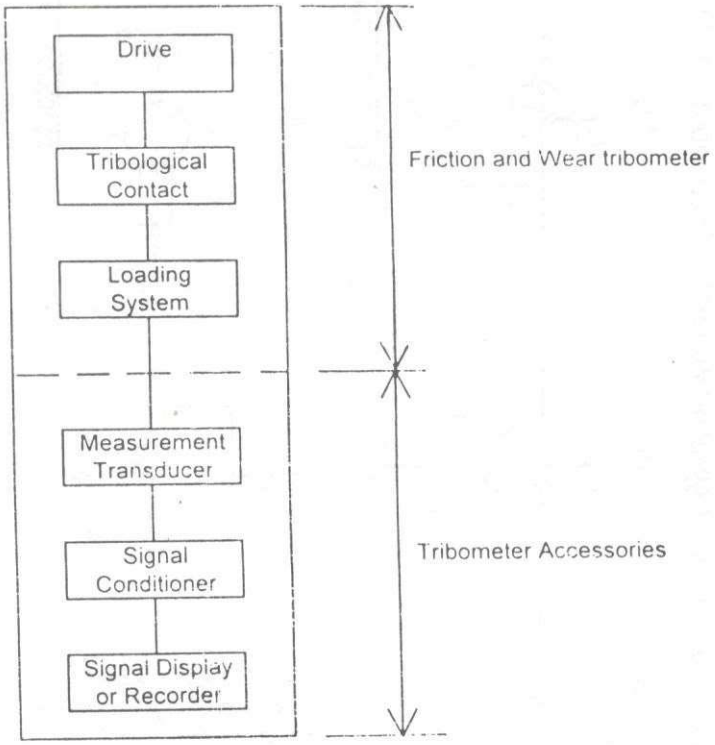


Fig. 3 Overall Tribometer Layout

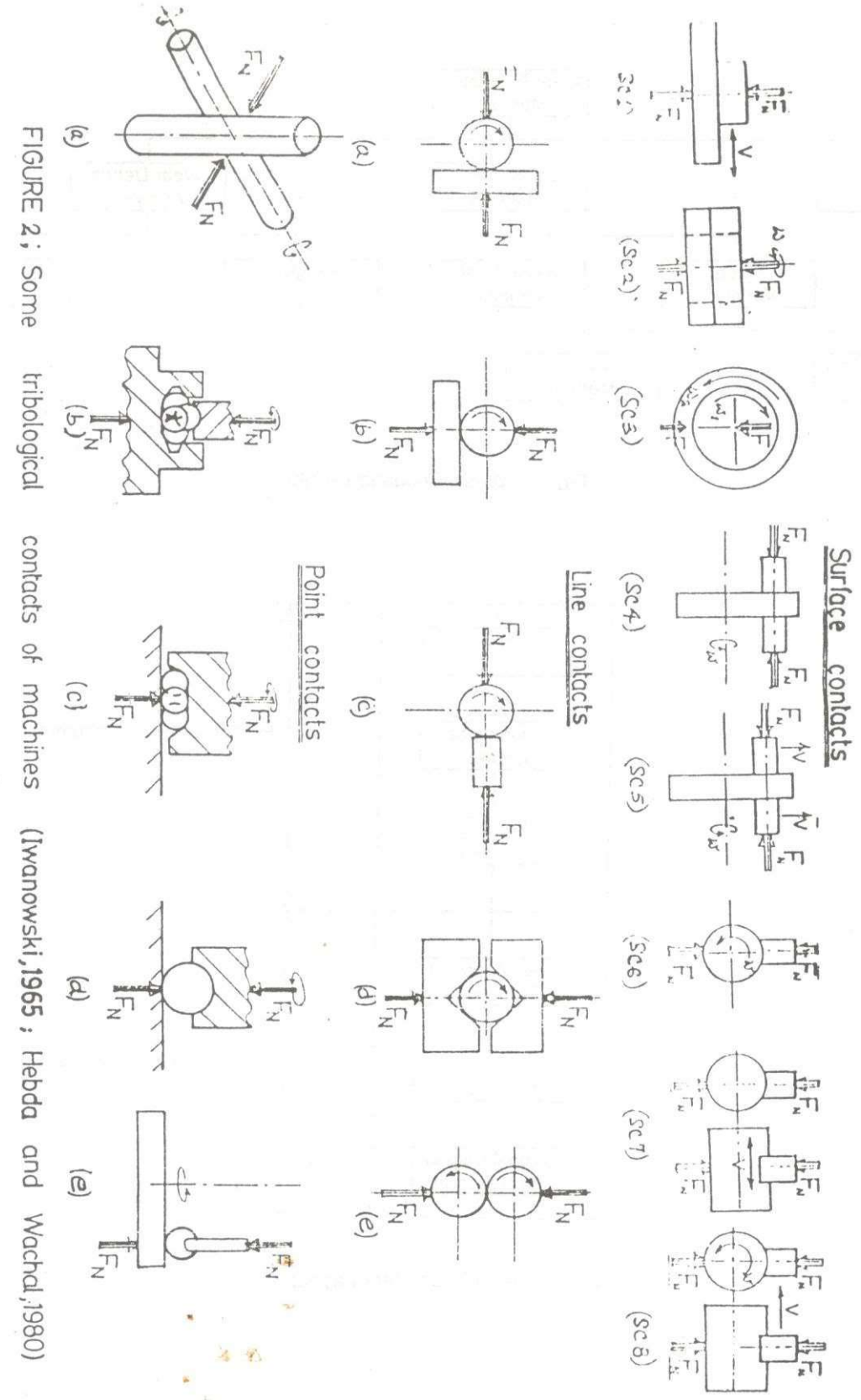


FIGURE 2; Some tribological contacts of machines (Iwanowski,1965 ; Hebda and Wachal,1980)

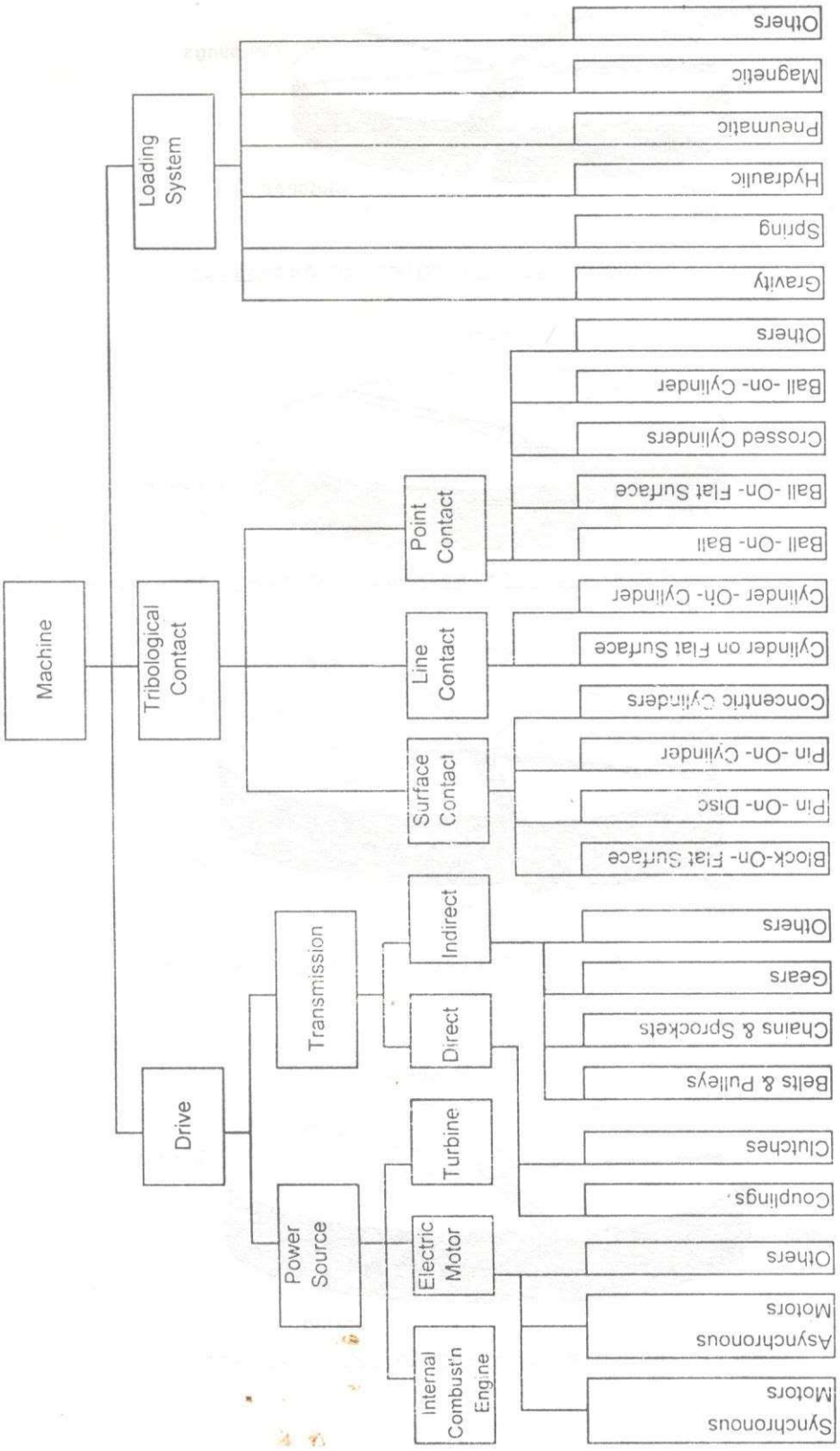


Figure 4, Broad structural chart

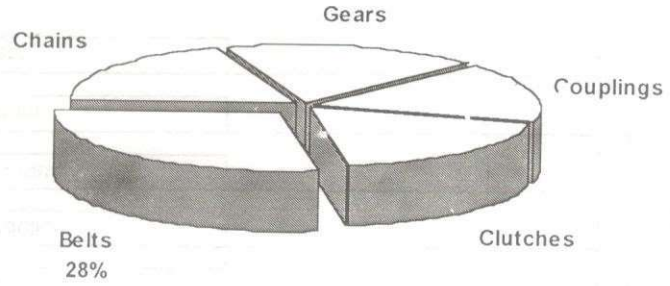


Figure 5; Transmission Assessment

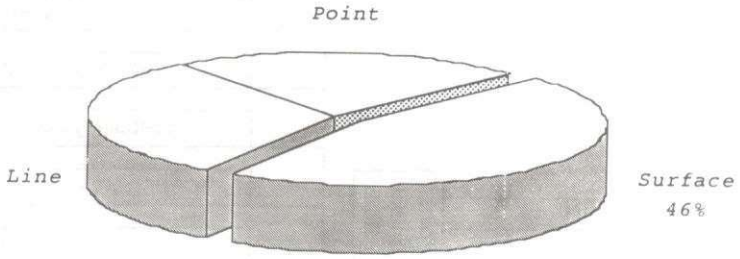


Figure 6 ; Tribological Contact Assessment

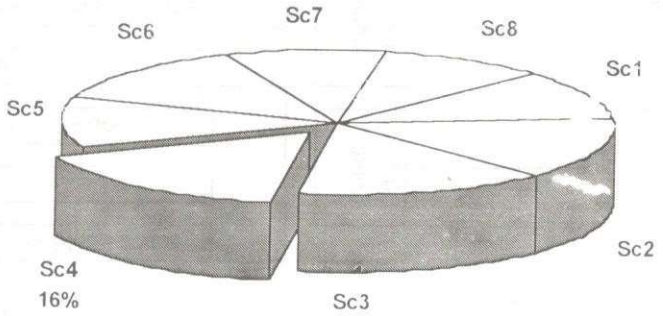


Figure 7; Surface Contact Assessment

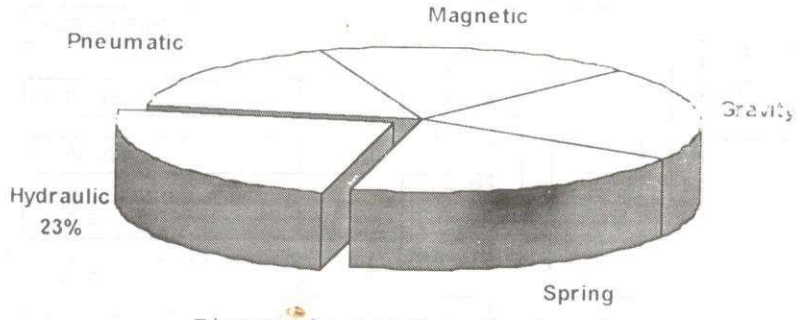


Figure 8; Loading System Assessment

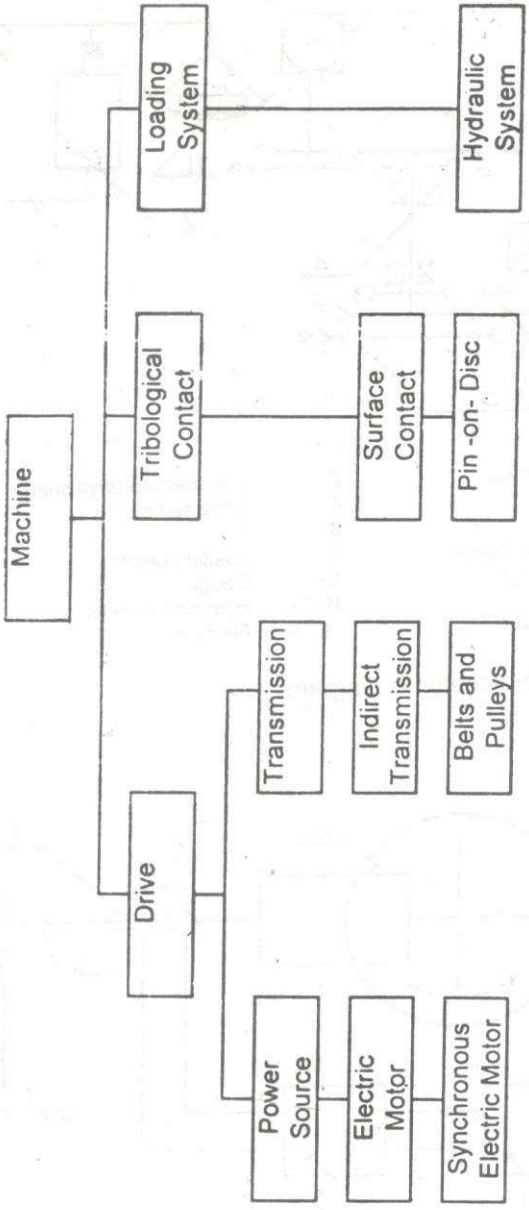
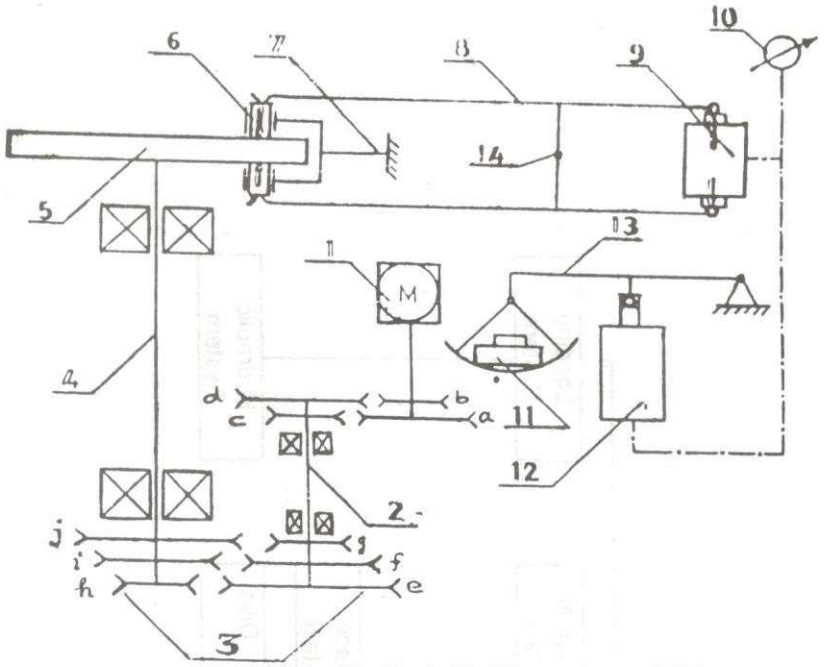
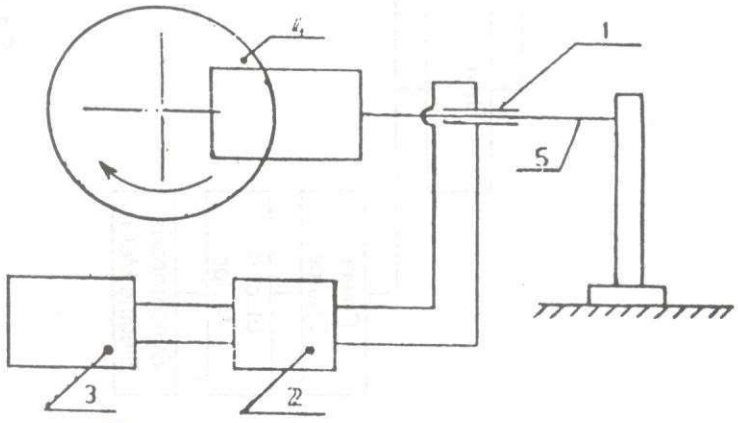


Fig. 9 Optimal Conceptual Design



1 - Electric motor,
2 - Intermediate drive shaft,
3 - Pulleys;
4 - Drive spindle,
5 - Disc,
6 - Pin,
7 - Tensometric beam
8 - Loading beam,
9 - Working cylinder,
10 - Gauge,
11 - Scale pan,
12 - Hydraulic cylinder,
13 - Loading lever,
14 - Fulcrum

Fig.10, Tribometer schematic diagram



1 - Load cell with electrical resistance strain gauge,
2 - Signal conditioner,
3 - Display or recorder,
4 - Roating disc,
5 - Tensometric beam

Fig. 11 Friction measurement instrumentation

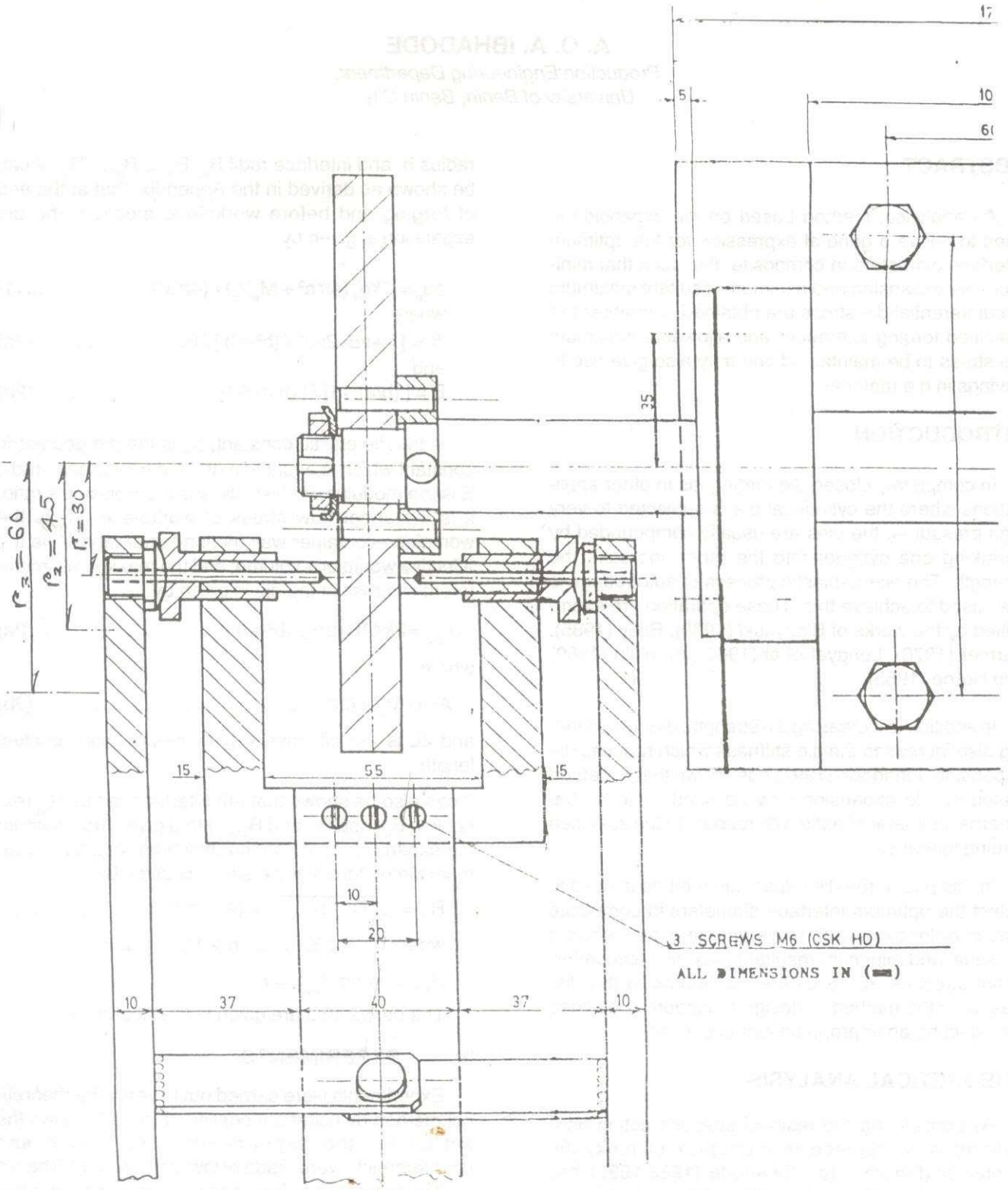


Fig. 12 TRIBOMETER SLIDER SUB-ASSEMBLY