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RELIABILITY AND SERVICE LIFE OF TRACTOR CLUTCHES

DIPL.-ING. KARL KECK

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RELIABILITY AND SERVICE LIFE OF TRACTOR CLUTCHES

1. Introduction:

Reliability is a product's ability to fulfill its required functions under specified conditions over a certain period of time.

This presentation concerns itself with the **reliability** of a tractor component, specifically the main drive clutch.

Aside from **reliability** we have to take into consideration other competing characteristics affecting the total vehicle. In addition to the omnipresent problem of cost, these characteristics include primarily:

- Operating comfort
- Maintenance of necessary package dimensions
- Universal vehicle use.

The last characteristic, "universal vehicle use" makes it difficult to specify target values for the design because we have insufficient data on clutch stresses and their distribution over the course of time.

However, we can't specify reliability until we know the stress distribution during practical application.

So far as reliability is concerned, this presentation is limited to dry friction clutches, which are used almost exclusively in Europe. I will cite necessary and possible improvements.

On the other hand, the sections pertaining to function and load do not depend on the clutch type. These values also apply generally for the development of other kinds of clutch assemblies in the event that total vehicle design prevents us from achieving the required reliability with a dry friction clutch.

2. Clutch Functions, Failure Modes

The following three functions are required of the clutch (Figure 2); failure of any of these features will result in the clutch having to be repaired:

- **<u>Connecting two shafts</u> mechanically in order to transmit the engine torque to the vehicle**

- **Separating two shafts mechanically in order to permit gear shifting in the transmission**

- **Isolating two shafts in a vibrational sense in order to eliminate transmission noise.**

Reliability is based on the frequency with which the clutch has to repaired and its service life up to that point in time. For example, we can use a Weibull analysis to determine reliability.

Only in some cases can we use objective criteria for evaluating clutch failure or inability to perform its functions. In these cases we can prove the cause of failure by measuring functional charateristics, such as facing wear, breakage or setting of spring components, broken linkage components, etc. In other cases, particularly during the warranty period, clutches are removed for subjective reasons, that is, the user either rejects the functional characteristics from the outset, or he complains that something has changed that can't be documented with clutch measurements.

The following discussion will describe the three cited clutch functions and related types of failure.

2.1 Separating Two Shafts

Mechanical separation and vibrational isolation appear to be of secondary importance to proper clutch design. Generally speaking mechanical separation requires that the slip torque M_R be a clearly defined function of the pedal travel, with the limiting condition $M_R = 0$, which means that given enough clearance, the clutch disc will run free. The mechanical-geometrical relationships involved must not change significantly when affected by temperature, corrosion, contamination and the presence of coolant.

This is hardly the case, as proven by warranty claims on clutches with less than 1000 operating hours. About 50% of clutch repairs can be attributed to this type of problem. People not only complain about incomplete disengagement, but also about the amount of operating force needed for disengagement, and the failure of linkage components.

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As far as vibrational isolation goes, tractor operators actually don't complain much about such problems. This is not to imply that people who drive tractors have less sensitive ears than drivers of passenger cars. The real reason is that the vehicle parameters allow us to provide a solution to the problem using a relatively simple torsion characteristic. Figure 3 shows this kind of 2-stage characteristic curve with a ± 4 degree wind-up angle in the first stage and a total wind-up angle of 18 degrees, which is almost always appropriate for 4 and 6 cylinder engines. The most important factor here seems to be that the first stage can be relatively stiff because of the high moment of inertia of the transmission components. Thus, the first stage eliminates both idle and drive rattle; the latter is fairly typical in tractors if they are driven at excessively low speeds in the higher gears.

Another advantage is that there is usually sufficient package space available in tractor clutches for installing a sturdy damper design. Facing wear and prevalent temperatures dictate a much larger clutch than is required for passenger cars with the same engine torque.

2.2 Connecting Two Shafts

The connection between two shafts ceases to function when the friction facings are completely worn. In dry clutches this wear is an unavoidable result of the friction work performed during the connecting cycle when the shafts are under load and rotate at different speeds.

During the service life of the clutch up to total wear, the resulting moment of the friction must be greater than the engine torque transmitted under all operating conditions--in the case of tractor clutches by a factor of about 1.4. The temperatures involved cause friction coefficient variation of about a factor of 3 and exert a dominating influence on the moment of the friction.

Therefore, two basically different types of failure affect the connecting function. If we plot failure due to total wear according to a Weibull distribution, we get a line that curves to the right for the clutch, as well as for other wear items. As an example Figure 4 shows a sample evaluation of 30 double clutches with 310 mm diameter for tractors in the 50 to 80 kW (70 to

110 hp) range from the Middle European area. All the clutches have been repaired due to complaints of difficult gear shifting. Consequently there is a certain probability that we are dealing with an accidental selection so far as stress is concerned. The curve indicates a minimum service life of about 1000 hours.

The failure curve for "slippage due to facing wear" indicates maximum anticipated clutch reliability. All the other causes for failure overlap with this curve and decrease the overall reliability.

Clutch repair for slippage <u>before</u> the wear reserve has been used up currently account for about 50% of all complaints, almost independent of the clutch model and the absolute number of complaints registered.

However, insufficient torque transmission results in considerably increased friction work and a corresponding increase in wear, making it impossible to determine the actual cause of failure.

Of course, failure as a result of "slippage due to facing wear" would not be particularly important if the curve were mostly or entirely displaced outside the vehicle service life range, which is determined by the design of other components, such as the engine. This is not the case for a tractor with a vehicle service life of about 10,000 hours; however, it could hold true in the future for passenger cars using non-asbestos facings, as shown in Figure 5. In this case we evaluated 100 clutches with 230 mm diameter from a mid-sized vehicle with a turbo-charged diesel engine. All of these clutches were returned because of chatter problems. Consequently in this case there is once more a certain probability that we are dealing with an accidental selection so far as load is concerned. The reliability curve again bends to the right typical of wear parts. The minimum service life amounts to 70,000 kilometers. Since the average service life for a passenger car in this class is about 120,000 kilometers, failure on account of "slippage due to facing wear" only plays a role in 15% of all vehicles.

3. Operating Conditions

Operating conditions and stress on the clutch vary much more for a tractor than they do for a street vehicle. Even an individual tractor has a far broader stress range due to its use with a wide variety of implements, under variable weather conditions and operation by several persons. The stress range for an individual tractor expands considerably if we look at all tractors for a given model. Tractors are sold all over the world today; soil conditions, field size, and environmental conditions, in addition to the type and operating frequency of implements are all different. We have derived the following approximate clutch stresses from our experience and we base our design calculations on these values.

3.1 Engagement Frequency

Approximately 0.1 to 8 per minute. These are mostly start-up cycles.

3.2 Friction Work and Temperatures

Because of a fairly constant ratio of performance to weight--at least in the case of standard tractors--it is meaningful to relate friction work and friction performance to engine performance. Figure 6 shows experience values as well as resulting temperatures for current standard dry friction clutch sizes.

a) Normal continuous load, such as plowing

- Average friction work equivalent to 0.3 to 0.6 % of rated engine performance
- Friction work per kW of engine performance per operating hour: 10,000 to 20,000 Joules
- Ambient temperatures in the bell-housing: 70 to 90° C
- Peak temperature of the friction surfaces: 130 to 150° C.

b) Extreme continuous load, such as front loader application

- Average friction performance equivalent to 2.5 to 5% of rated engine performance

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- Friction work per kW of engine performance per operating hour: 90,000 to 180,000 Joules
- Ambient temperature in the bell-housing: 100 to 160° C
- Peak temperatures of the friction surfaces: 230 to 350° C.

c) Extreme short term load intervals, such as starting up with a high tractive effort

- Slip cycles in a range of about 10 seconds at full engine performance.

In this case it doesn't make any sense to specify average friction performance or friction work per kW of engine performance per operating hour, because this kind of stress would destroy a dry friction clutch within a short time given current cooling conditions. The temperature of the friction surfaces jumps by about 80° C, and between 2000 to 5000 of these cycles will completely deplete the wear reserve.

3.3 Torques Values

We have to count on peak torque values up to 3 times the maximum engine torque.

3.4 Clutch Installation

The clutch is normally installed in a completely closed bell-housing in order to prevent malfunctions caused by contamination. Sliding surfaces and bearings in the release system are even more susceptible in this regard than the clutch itself. The closed bell-housing also prevents outside air from cooling the clutch directly; cooling occurs indirectly by the air in the bell-housing, which is considerably hotter. Since we will also have to deal with closed bell-housings in the future, the following calculations are based on the temperature curve shown in Figure 7. The friction surface temperature is at least 70° C and increases in proportion to the average specific friction performance. Depending upon the clutch design and the size of the bell housing, we will find significant variation, as indicated by the shaded area in the illustration.

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The size of the bell-housing is also limited by the package dimensions of the tractor. In this regard, criteria such as wheel base, frame width, free access to the cab, use of standard production equipment, etc. all play a role. Even in larger tractors in the range from 80 to 120 kW, the bell-housing inner diameter normally limits the clutch size to a maximum diameter of 330 to 360 mm.

3.5 Release System

Because of cost, current release systems operate almost entirely on muscle power. Based on our experience, the pedal load should be limited to maximum 200 N in a new tractor and maximum 300 N during the entire operating life. Maximum pedal travel is from 150 to 200 mm in tractors with a cab, and the release systems have efficiencies between 40 and 60%.

4. Reliability Goals and Stress Specifications

Depending upon operating conditions, the clutch achieves certain reliability values in performing the functions described here. These values show considerable variation in tractor models currently equipped with LuK clutches, primarily because there is also considerable variation in the design characteristics. The shaded area in Figure 8 shows the overall reliability currently achieved, as a result of the types of failure described above.

- Average service life of 2000 to 6000 operating hours
- Failure rate at 1000 operating hours of 1 to 20%.

In this light, the current situation is more or less satisfactory.

Tractor manufacturers do not have very specific goals for clutch reliability. Nor do we have enough information on the stress distribution. However, since both factors must be specified if development work is going to be meaningful, I would like to pose the following assumptions for discussion:

4.1 Reliability Goal

As noted above, Figure 8 shows the current total clutch reliability range as a shaded area. The clutches on the right side of the area have been rated as excellent, and customers generally reject any cost increases for further improvements in these clutches. Therefore, we suggest that the bold border line on the right be viewed as an adequate goal for clutch reliability in the next generation of tractors. In the process, we are assuming that this total reliability goal will only be achieved if failures due to facing wear do not exceed the dash-dot line, which is defined by the following points:

- Minimum service life of 1000 operating hours
- 80% reliability at 4000 operating hours.

The classification of the two curves, that is the relationship of facing reliability to the total reliability, reflects current LuK experience.

4.2 Stress Specifications

Up until now LuK has based its service life calculations on specific stress values, assigning a specified amount of friction work per each kW of engine performance per operating hour:

- 100,000 Joules per kW per hour as maximum stress, which yields the minimum service life of the clutch.

- 13,000 Joules per kW per hour as minimum stress, which yields the maximum service life of the clutch.

In Figure 9 the maximum stress, assumed as a distribution, is plotted vs. the relative friction performance. It contains portions which reach a friction performance of about 11% of engine performance, accompanied by corresponding high temperatures.

We don't need to specify a distribution for the minimum stress because the relatively low temperature level has no effect on the wear performance of the facings.
These load extremes have been calculated based on field results. In our opinion they still describe the

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extreme conditions for tractor clutch application with sufficient accuracy; therefore we have used them for the following calculations.

The frequency of such extreme conditions within a vehicle population is of critical importance for the slope of the reliability curve.

Past calculations have obviously underestimated the frequency of high stresses. The following discussion is based on the clutch stress distribution shown in Figure 10, which gives us a reliability curve coinciding to current experience. The extreme values of 100,000 and 13,000 Joules per kW per hour are plotted at cumulative frequencies of 0.1 and 99.9 and connected with a straight line. The average load at 50% cumulative frequency, which yields the average service life, is 22,000 Joules per kW per hour. The curve we are primarily interested in runs from the maximum value to this average value and corresponds roughly to a logarithmic standard distribution. For the sample calculation in Figure 11, we used LuK's design limits of 80 Watt/cm^2 with ceramic facings. This yielded a wear reserve of 3 mm. This would apply, for instance, to a

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310 mm diameter clutch for a 75 kW (102 hp) tractor. The calculated result is represented by the solid line on the left side of the graph. The failure rate of just 4% at 1000 operating hours based on the wear alone does not appear to be at all unrealistic. For purposes of comparison the broken line on the right side of the graph shows the goal as cited previously.

5. Required Sizing for the Cited Reliability Goals

5.1 Facings

In addition to the stresses and their resulting temperatures, we need to know the wear performance of the facings in order to calculate the service life. The measured wear rate is plotted for the following facing materials in Figure 12 as a function of the temperature:

- Ceramic facings--solid line
- Organic non-asbestos facings--broken line
- Organic asbestos facings--dash-dot line

Contrary to previous experience, current results from tests with the best non-asbestos organic facings, over the total relevant temperature range, indicate that wear is lower than that of asbestos organic facings. Since asbestos facings are supposed to be taken off the market by the end of this decade, the following discussion is based entirely on the use of either non-asbestos organic or ceramic facings.

5.2 Required Sizing Characteristics
Engine Performance/Friction Surface and Wear Reserve

Figure 13 shows the characteristic sizing values, engine performance per friction surface and wear reserve, which allow us to achieve the required reliability goals under the cited conditions. These values are indicated in lines 1 and 2. In each case, the wear reserve is chosen to ensure a minimum service life of 1000 operating hours. If we require service life of 4000 operating hours with 20% failure rate, we can see, according to line 4, that non-asbestos organic facings exceed this goal by about 10%.

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This results from the flatter reliability curve. Ceramic facings fall short of the goal by about 10%.

As for the cited results, I would like to add:

- This requires a relatively generous clutch design; we should size in the range of 50 to 55 Watt/cm^2 with non-asbestos organic facing material. With ceramic facings, we need to be in the range of 60 to 65 Watt/cm^2 or--if we can achieve the specified wear reserve of 5.6 mm--up to max. 70 Watt/cm^2.

- At the same time a relatively high wear reserve is required. We can only achieve values over 4 to 4.5 mm using a diaphragm spring clutch which is lever-operated or a pull-type version. Values over 6 mm are unrealistic for diaphragm spring clutches.

Figure 14 shows the engine performance ranges covered by the currently prevalent clutch sizes using the cited sizing goals. If in the future we don't have clutch package space in excess of a nominal diameter of 350 mm, it would be a good idea to abandon the use of single-disc dry friction clutches for applications above about 88 kW (120 hp) and resort to using twin plate clutches or wet clutches.

5.3 Pressure Plate Thermal Capacity

At the moment, the performance weight of the pressure plate in LuK tractor clutches is between 5 and 11 kW/kg. In order to avoid overheating and failure as a result of grossly abusive operation, it appears advisable not to exceed values of 6 to 7 kW/kg in the future.

Figure 15 indicates the average pressure plate thickness required, dependent upon the characteristic size value: engine performance per unit of friction surface. If we relate the value 6 kW/kg to the organic facings and 7 kW/kg to the ceramic facings and use the sizing targets 55 and 65 Watt/cm^2 respectively, we arrive at a pressure plate thickness of about 25 mm in both cases.

5.4 Clutch Safety Factor, Pedal Effort

As described earlier, "insufficient torque transmission" accounts for about 50% of current complaints. In spite of our goal for generous sizing, LuK recommends that, we should also continue to design with a nominal clutch safety factor of min. 2.2, based on a friction coefficient of 0.27 for organic and 0.4 for ceramic facings.

We make the following assumptions in order to determine the pedal load based on the clutch safety factor and the resulting clamp load:

- Engines have a rated speed of 2200 rpm and 20% torque rise
- 160 mm pedal travel and 60% total release system efficiency in the tractor
- Mechanical clutch efficiency of 75% using a minimum lift-off of 1.8 mm

If we size at 55 Watt/cm^2 for organic and 65 Watt/cm^2 for ceramic facings, we arrive at the pedal efforts dependent upon the engine performance as shown in

Figure 16. The values on the graph apply for the clutch and linkage system in new condition. Here we should not exceed pedal loads of 200 N because we can count on an increase of about 50% over the operating life. This increase is caused by the decreasing efficiency of the clutch and linkage system, as well as the increasing diaphragm spring load. We experience complaints when the pedal load exceeds 300 N.

The illustration shows that the assumed 60% linkage efficiency is adequate for ceramic facings over the entire performance range, with the exception of the twin plate clutch. For organic facings, on the other hand, we have to introduce improvements in the clutch and the linkage system beyond about 50 kW. Such improvements might include beaded diaphragm spring fingers in the clutch or overcenter pedal assist springs.

5.5 Mass Moment of Inertia of the Clutch Disc

We have found it relatively difficult to eliminate complaints related to poor disengagement and the resulting high shift efforts. In this regard we ought to see positive results from more generous designs because temperature effects, which certainly play a role, will be reduced. However, we shouldn't overlook the fact that large diameters and a high wear reserve result in discs with high moments of inertia. Figure 17 illustrates the levels of inertia we have to expect. We are assuming here that organic friction facings have a density of 2 g/cm^3, and that a third of the friction surface is covered by facings when we use ceramic facings. The graph illustrates the mass moment of inertia we have to expect with both friction materials on discs with and without torsion damper.

6. Summary

We find the reliability of the dry friction tractor clutches currently being used is not entirely adequate. We are suggesting that we meet this problem with more generous designs, simultaneously increasing the facing wear volume and decreasing thermal stress. If we can apply the defined sizing values, we can achieve the desired reliability with regard to facing wear performance and torque transmission. On the other hand, failures due to increased pedal and shift efforts cannot be entirely corrected in the clutch, and sometimes not at all. When this is the case, we have to make improvements in the interactive tractor components as well, such as the release system and the synchronization.

Our cited reliability goal continues to define the dry friction clutch as a wear part which must be changed at least once, in most tractors, during the total service life of 10,000 h. If this is not acceptable, or if the necessary package space can't be made available, then we should use a different kind of clutch. The wet friction clutch is a possibility. It can be designed in such a way that it does not need to be considered a wear part. In this case we have to look at other aspects--in addition to the manufacturing cost--such as the abuse capacity factor and disengagement performance. LuK is of the opinion, however, that we have not entirely exhausted the capabilities of the dry friction clutch.

Zuverlässigkeit ist die Fähigkeit eines Produktes, die geforderten Funktionen während einer bestimmten Zeitdauer unter festgelegten Bedingungen zu erfüllen.

Affidabilita è la capacità di un prodotto di assolvere una funzione durante un tempo ben preciso in condizioni prestabilite.

RELIABILITY IS A PRODUCT'S ABILITY TO FULFILL ITS REQUIRED FUNCTIONS UNDER SPECIFIED CONDITIONS OVER A CERTAIN PERIOD OF TIME.

La fiabilité est l'aptitude d'un produit à répondre aux fonctions exigées durant un temps déterminé pour des conditions fixées.

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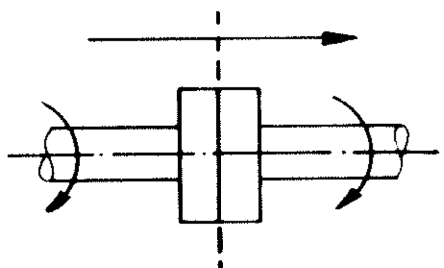
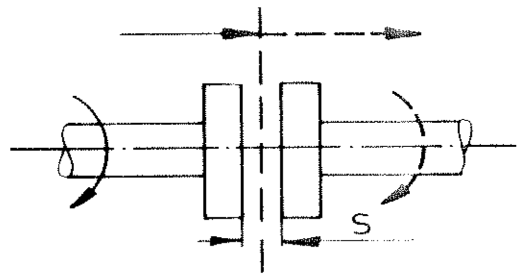
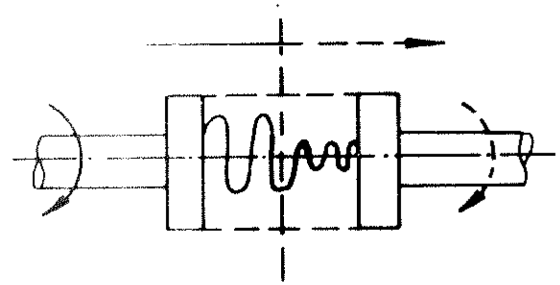
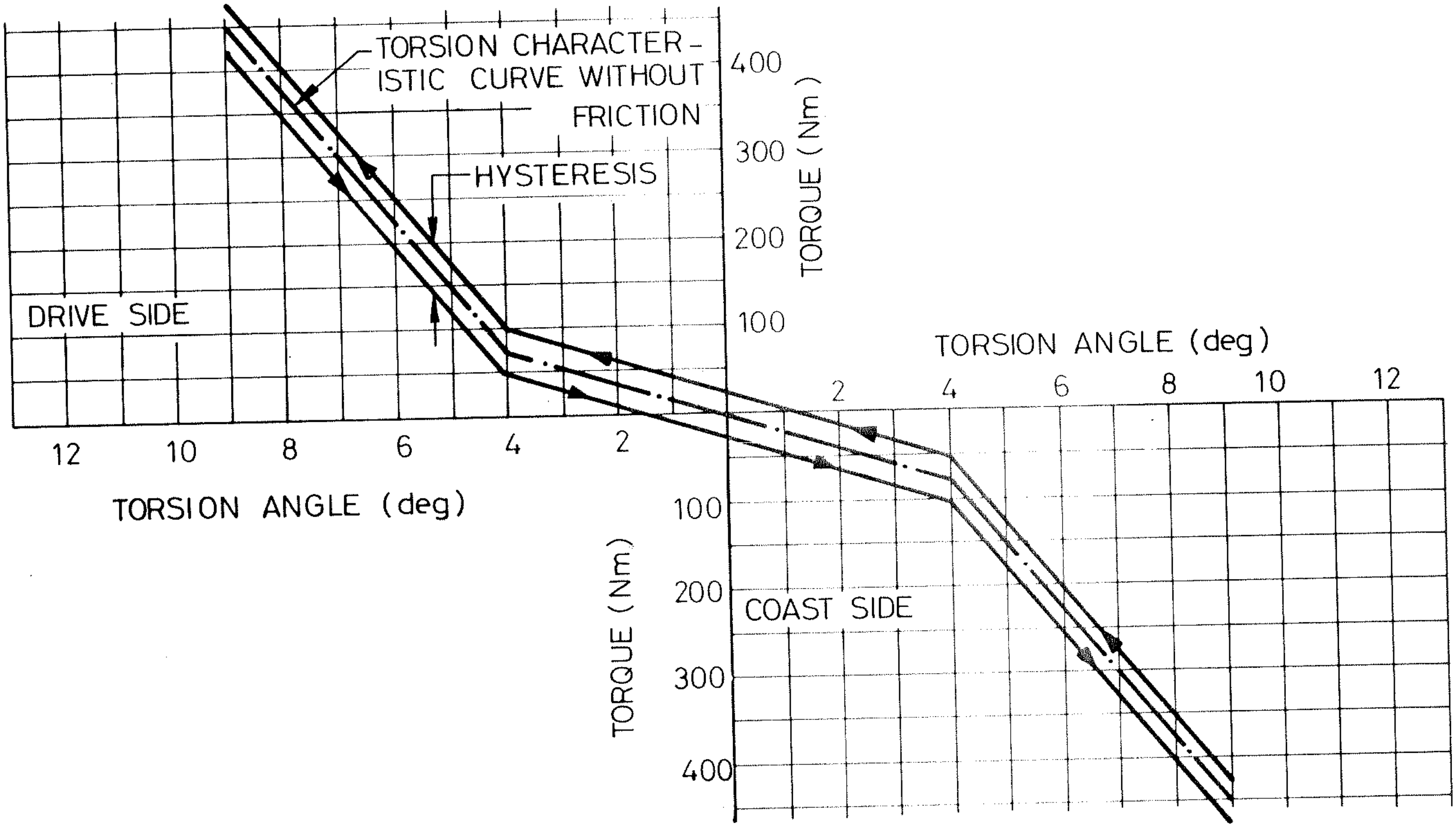
MECHANICAL CONNECTION	MECHANICAL SEPARATION	VIBRATIONAL ISOLATION
 M_1 | M_2	 M_1 | M_2	 ω_1 | ω_2
$M_2 = M_1$ $M_R = K \times M_1$ $> M_1$	$M_2 \stackrel{!}{=} f(s)$ $M_2 \approx 0 (s = s_{max})$	$\dot{\omega}_2 = K \times \dot{\omega}_1$ $< \dot{\omega}_{max.}$

CLUTCH FUNCTIONS



TORSION CHARACTERISTIC CURVE FOR TRACTORS WITH 4 AND 6 CYL. ENGINES

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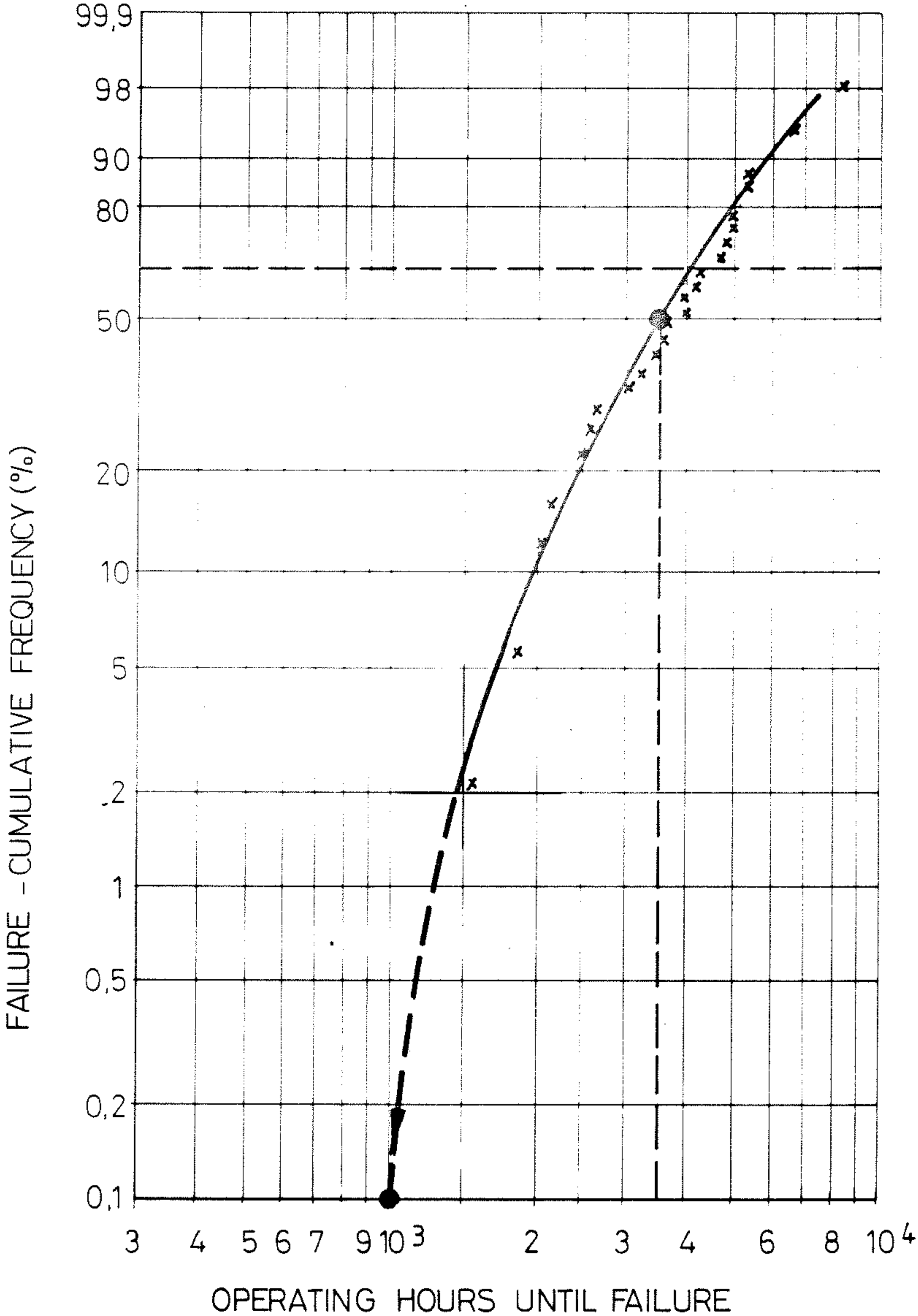

04 03 86	EXAMPLE OF TRACTOR CLUTCH RELIABILITY BASED ON FACING WEAR

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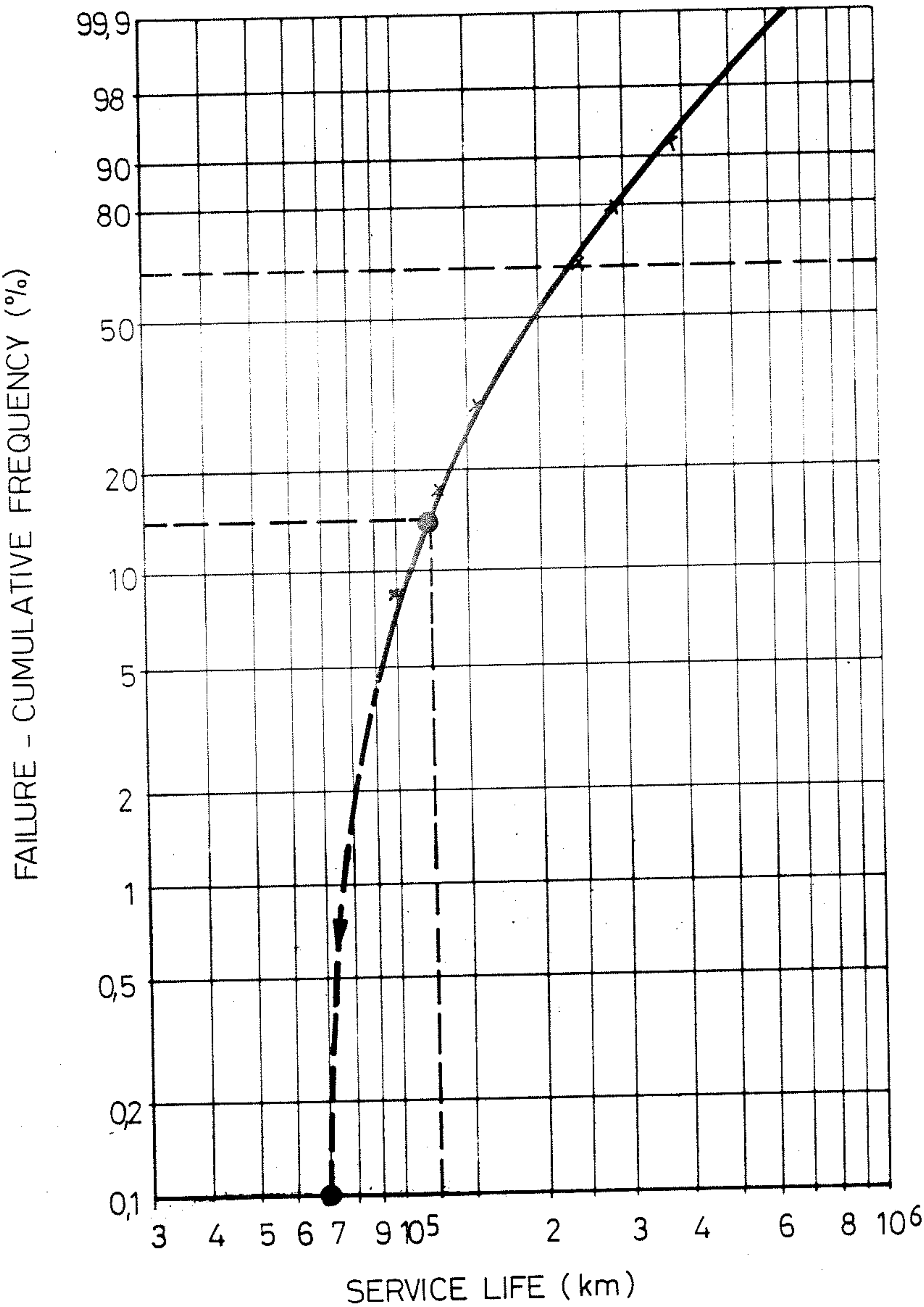

05 03 86	EXAMPLE OF PASSENGER CAR CLUTCH RELIABILITY BASED ON FACING WEAR	LUK

	AVERAGE FRICTION PERFORMANCE / ENGINE PERFORMANCE	FRICTION WORK PER KW PER OPERATING HOUR	AMBIENT TEMPERATURE	TEMPERATURE OF FRICTION SURFACE
NORMAL CONTINUOUS STRESS	0,3 ÷ 0,6 %	10 000 ÷ 20 000 Joule / kWh	70 ÷ 90° C	130 ÷ 150° C
EXTREME CONTINUOUS STRESS	2,5 ÷ 5 %	90 000 ÷ 180 000 Joule / kWh	100 ÷ 160 °C	230 ÷ 350 ° C
EXTREME SHORT - TERM STRESS	SLIP CYCLES UP TO 10 SEC. AT MAX. ENGINE PERFORMANCE			

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STRESS ON TRACTOR CLUTCH

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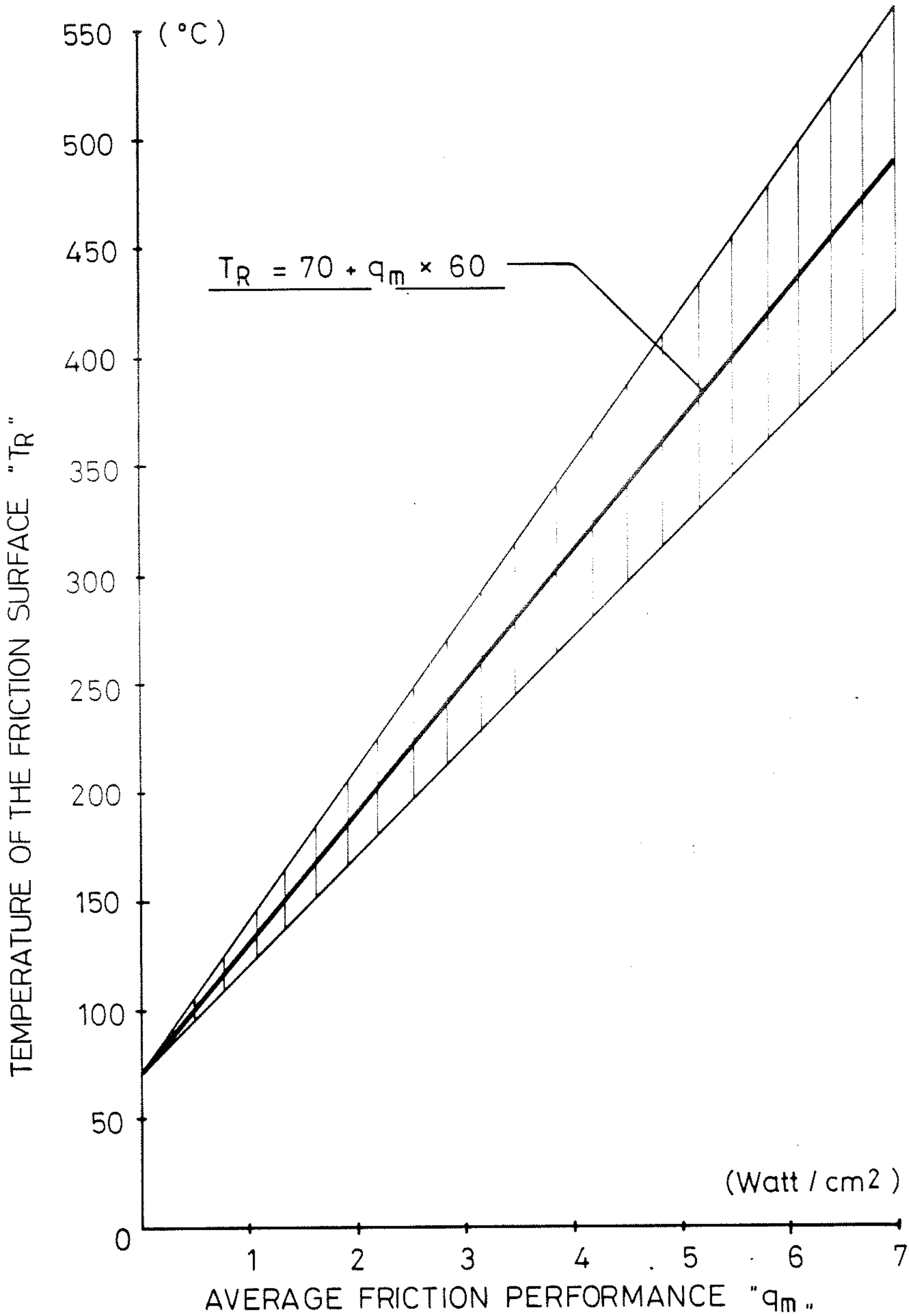
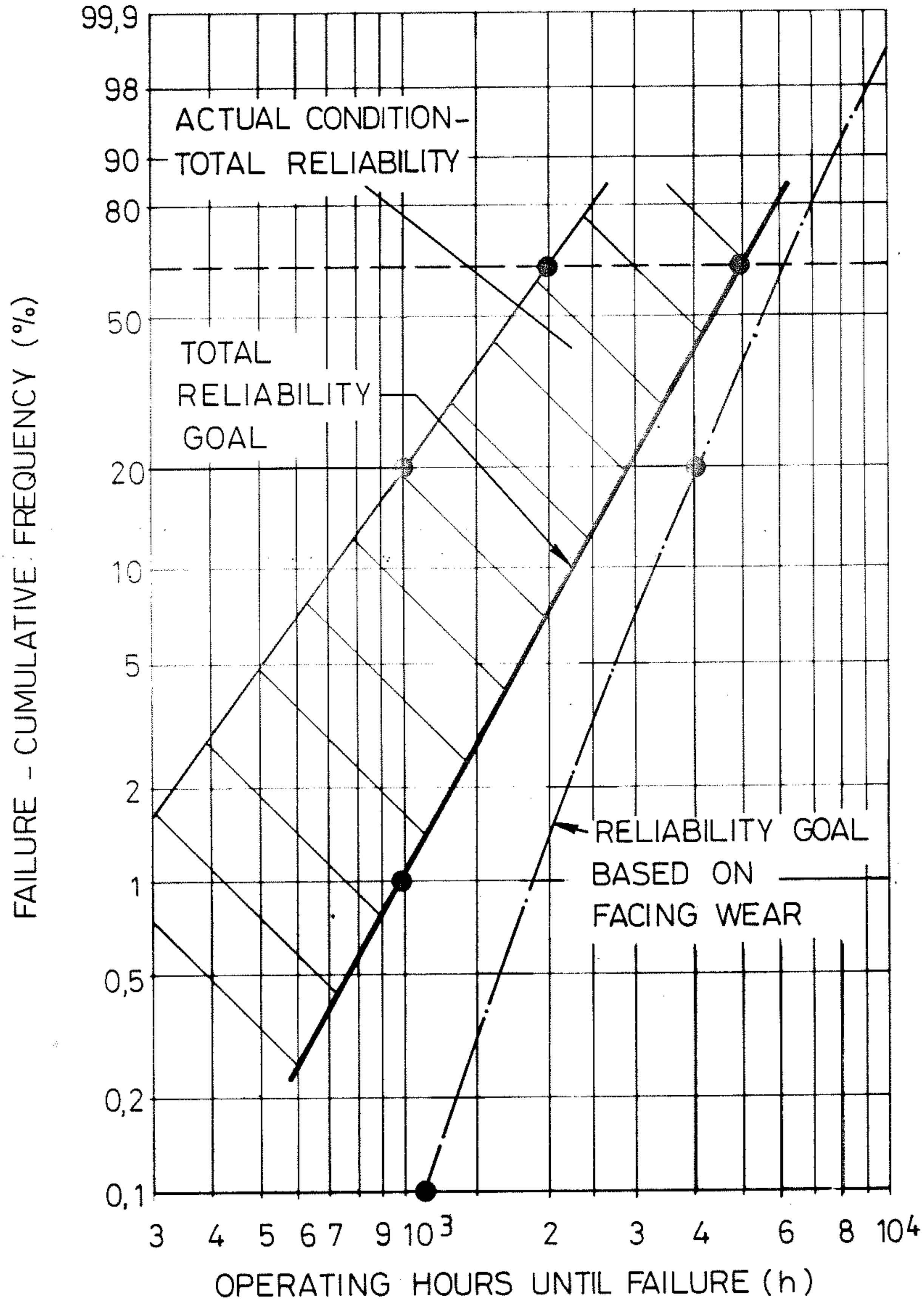
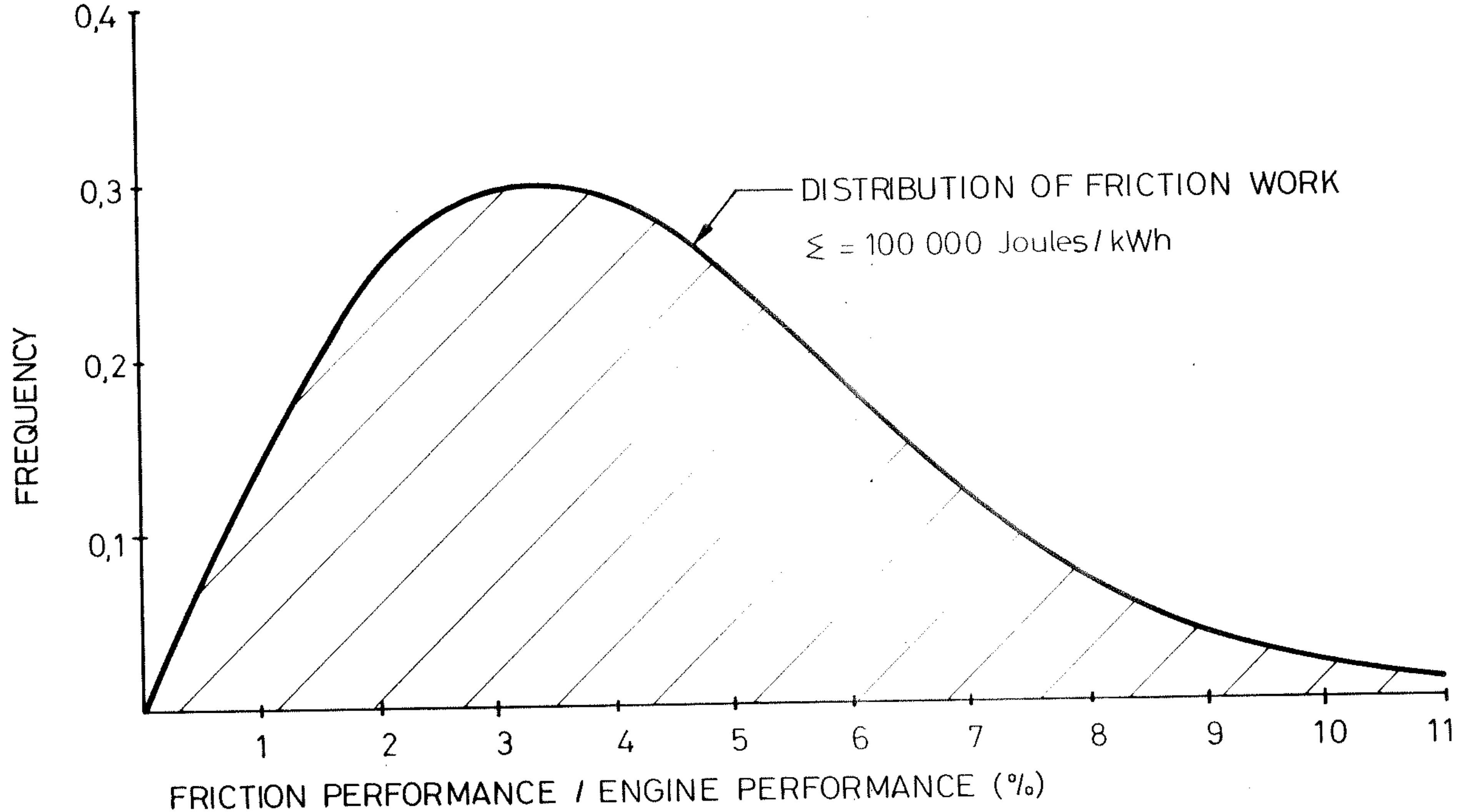


07 03 86	FRICTION SURFACE TEMP. VS. THE AVE. FRICTION PERFORMANCE WITH CLOSED BELL - HOUSING	LUK



08 03 86	RELIABILITY GOALS	LuK



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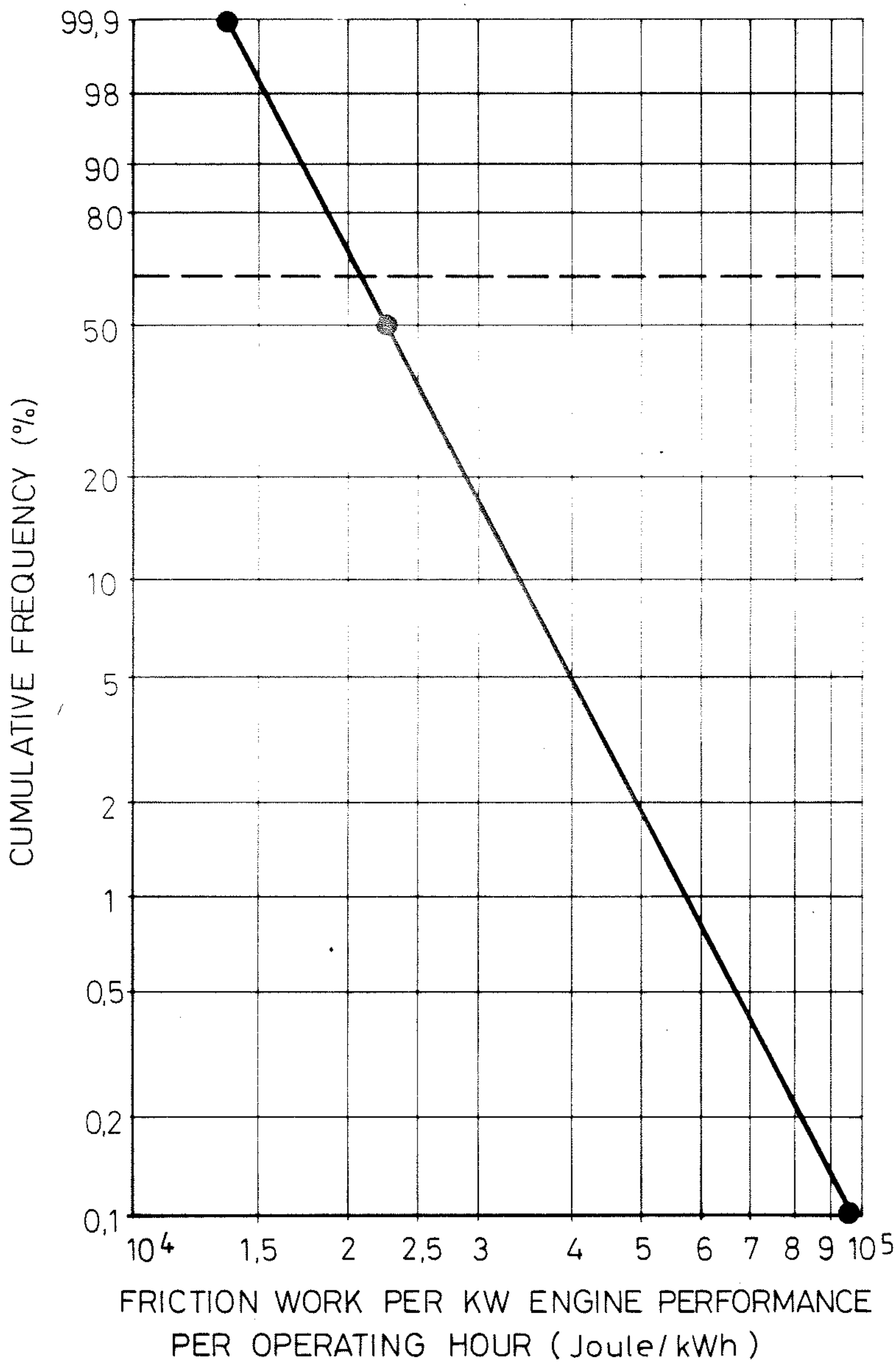
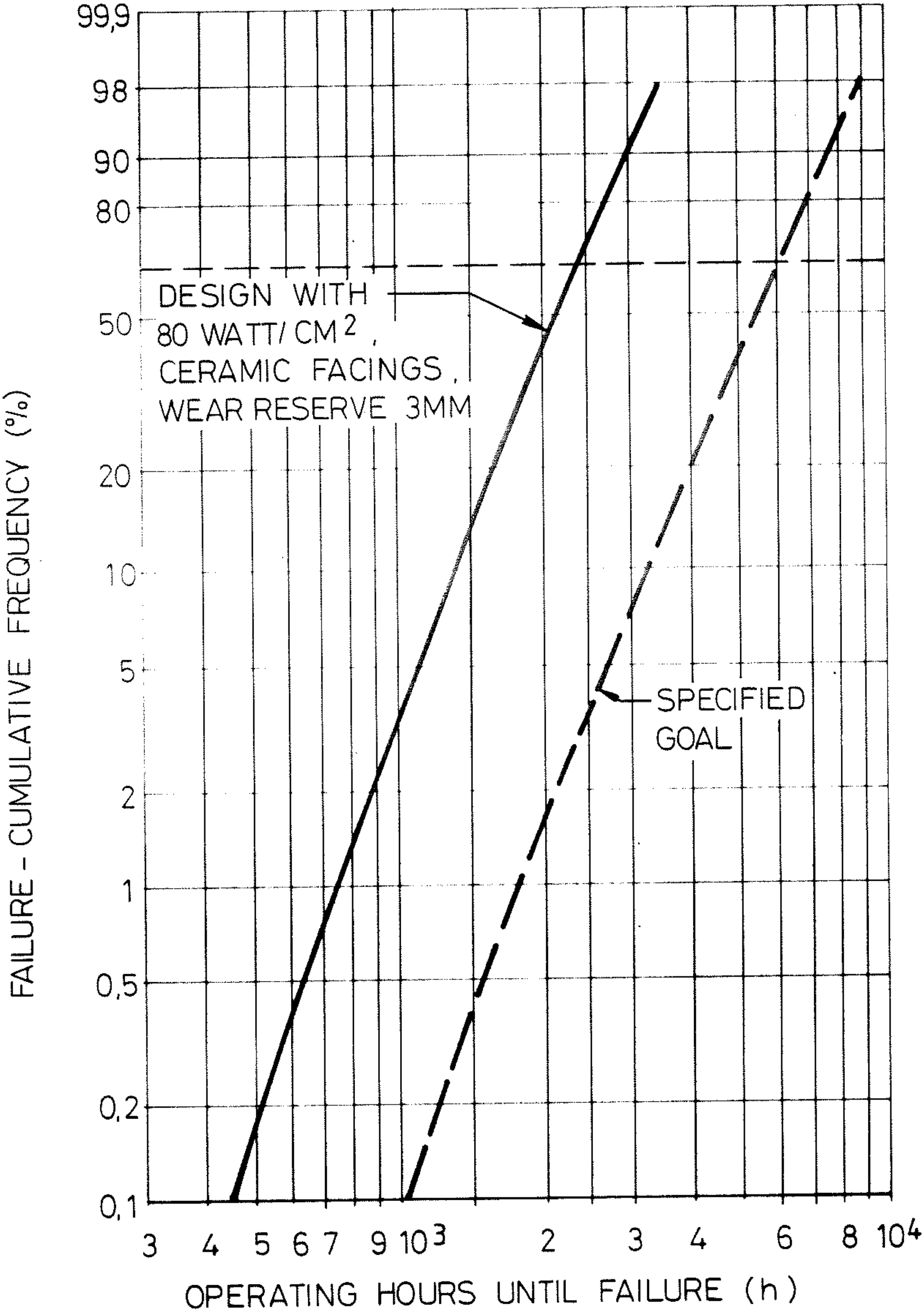
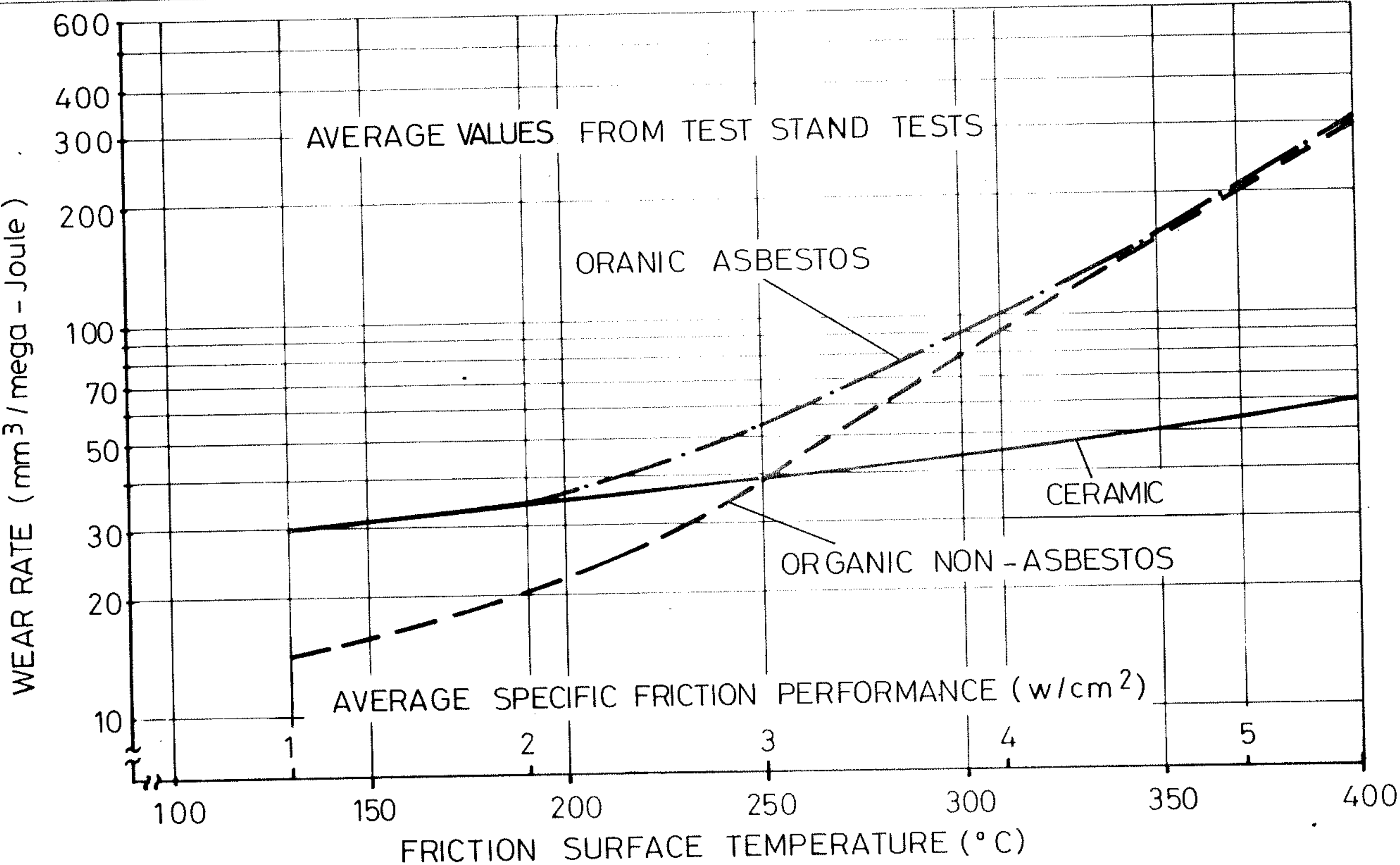
ASSUMED STRESS DISTRIBUTION FOR MINIMUM SERVICE LIFE



10 03 86	ASSUMED STRESS DISTRIBUTION FOR A TRACTOR POPULATION	LUK



11 03 86	RELIABILITY BASED ON FACING WEAR - SAMPLE CALCULATION



12 03 86	WEAR RATE VS. FACING MATERIAL AND TEMPERATURE

FACING MATERIAL		ORGANIC, NON-ASBESTOS			CERAMIC			
SIZING (w/cm^2)		50	55	(60)	60	65	70	1
WEAR RESERVE (mm)		3,3	4,7	(6,4)	4,5	5	5,6	2
SERVICE LIFE (h) AT CUM. FAILURE FREQ.	0 %	1000	1000	(1000)	1000	1000	1000	3
	10 %	3 300	3 400	(3 500)	3 000	3000	3000	4
	20 %	4 200	4 400	(4 600)	3 700	3 700	3 700	5
	62,5 %	6 820	7 380	(7 900)	5 400	5 400	5 500	6

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REQUIRED CHARACTERISTIC SIZING VALUES FOR THE SPECIFIED RELIABILITY GOAL

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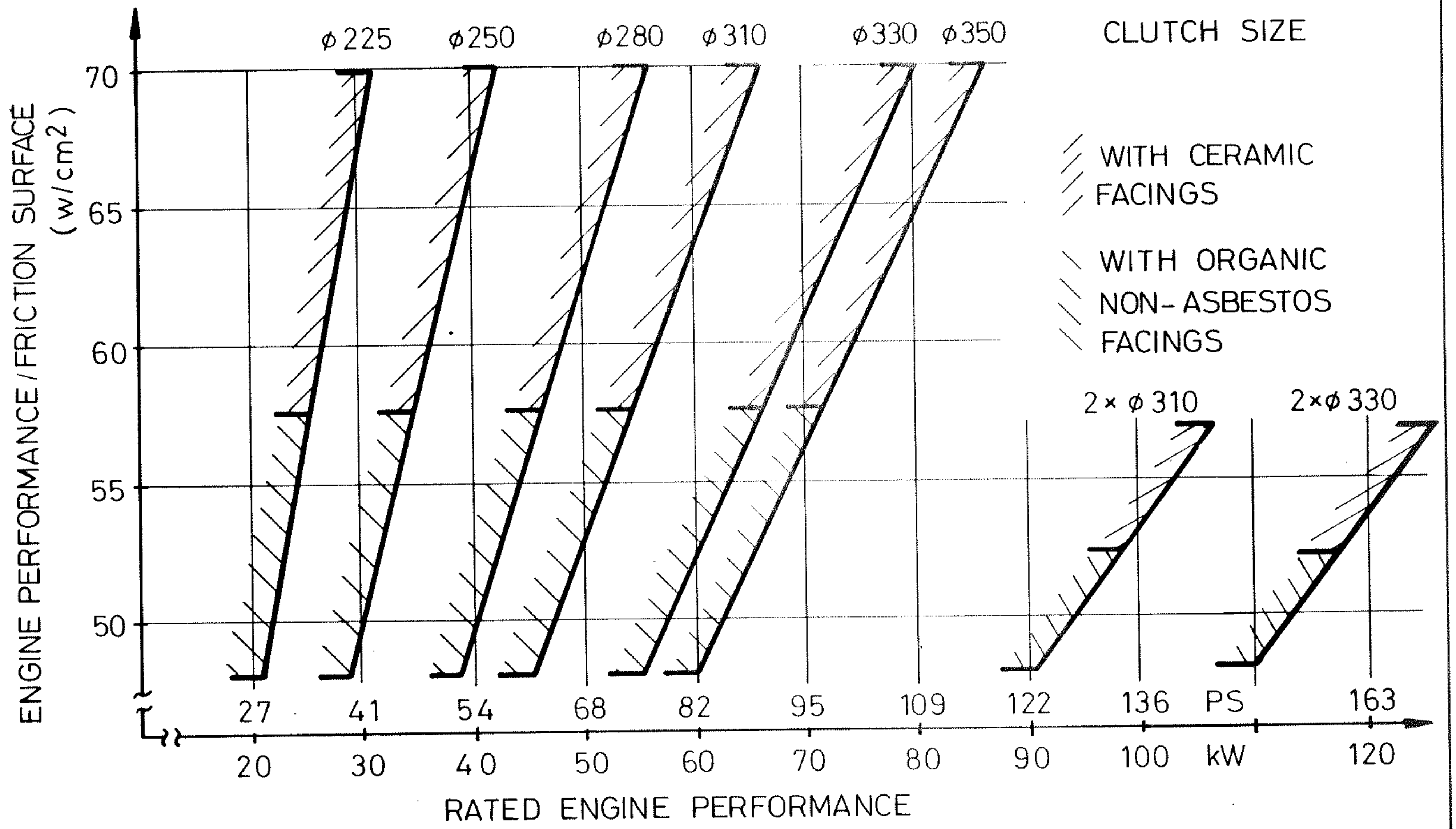
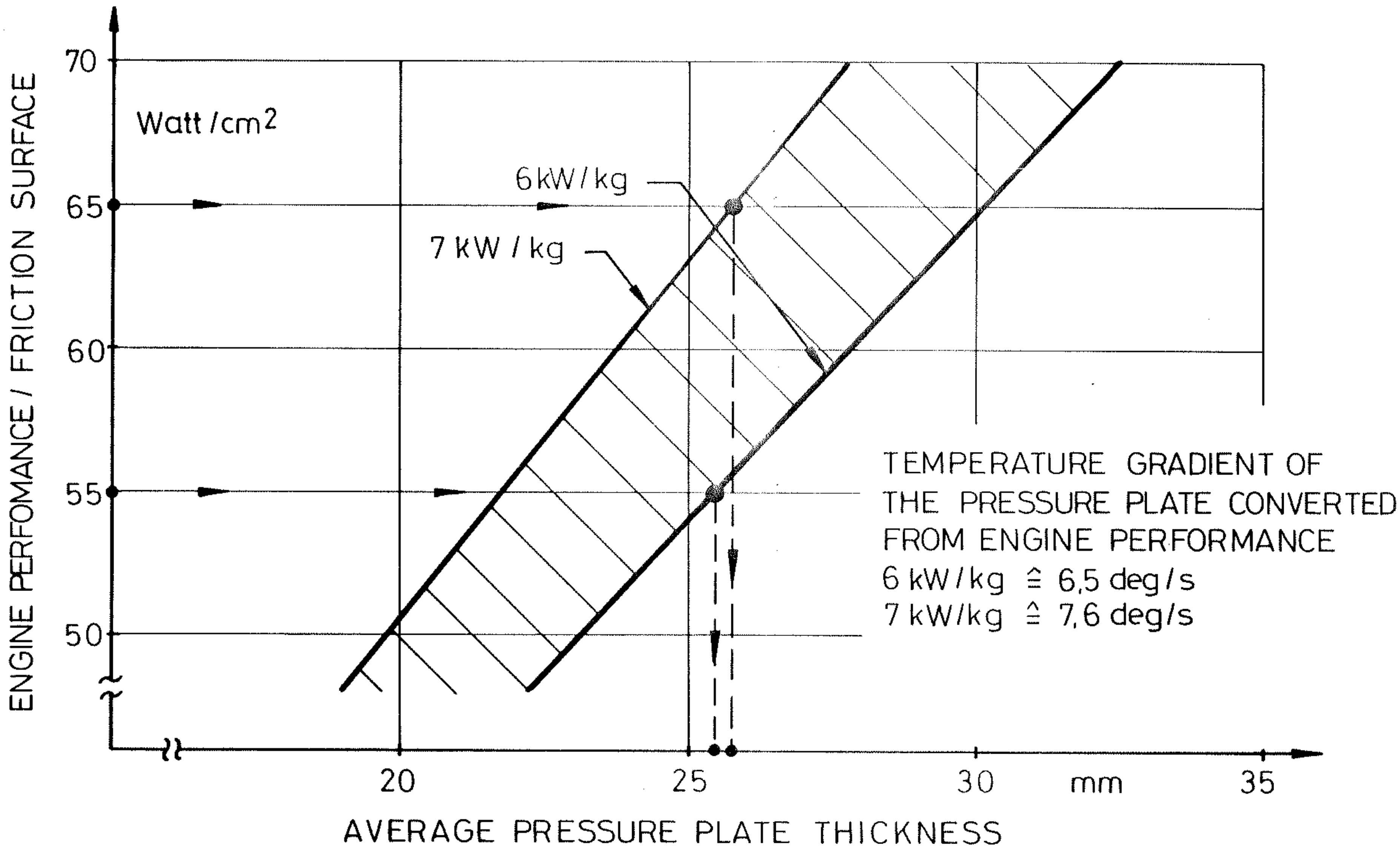
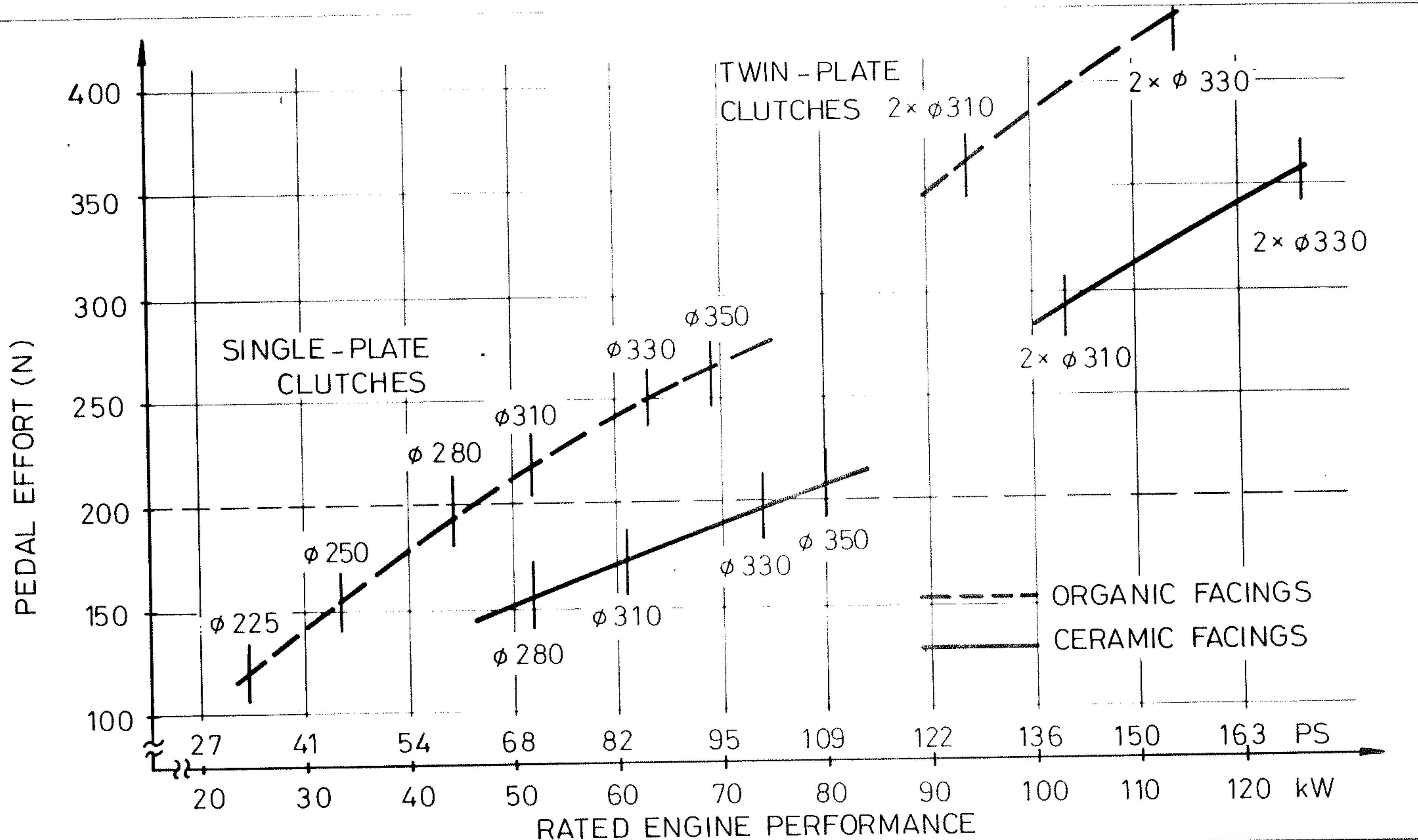


14 03 86	APPLICATION LIMITS FOR VARIOUS CLUTCH SIZES FOR THE SPECIFIED RELIABILITY GOALS



15 03 86	PRESSURE PLATE THICKNESS VS. CHARACTERISTIC SIZING VALUES	LUK



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PEDAL EFFORT VS. ENGINE PERFORMANCE AND FACING MATERIAL

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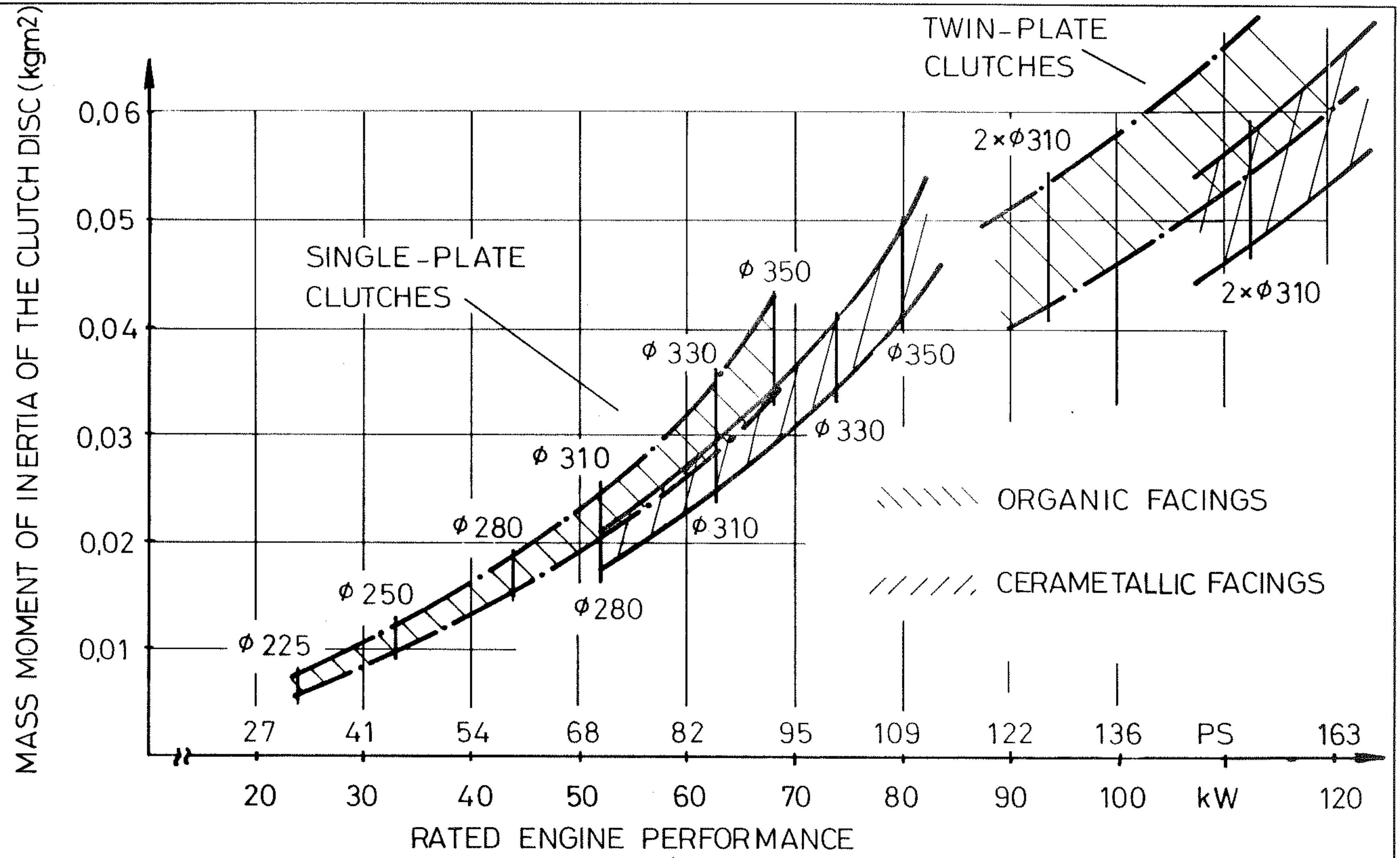

17 03 86	INERTIA OF THE CLUTCH DISC VS. ENGINE PERFORMANCE AND FACING MATERIAL	LUK