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OPTIMIZED CLUTCH DESIGN
RELEASE LOAD AND OPERATING COMFORT

DIPL.-ING. PAUL MAUCHER

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Introduction

We have to take the following criteria into consideration when designing clutches (Figure 1):

- satisfactory service life
- reliable transmission of engine torque
- low release system losses
- low release bearing load
- reliable disengagement
- smooth engagement.

In addition to service life and transmission of engine torque, operating comfort is an essential consideration.

Early stages of motor vehicle clutch design used clutch discs without any cushion, which obviously resulted in little engagement comfort. As a result, the clutch lift-off could be lower than it is today.

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It didn't take long for the introduction of clutch discs with cushion deflection to improve modulation. In order to ensure good clutch disengagement, it was necessary to increase the clutch lift-off.

Because of recurrent problems with disengagement and engagement, there has been a tendency to increase clutch lift-off and cushion deflection. As a result, today we often have very high cushion deflection and clutch lift-off, together with the disadvantage of high release bearing load.

High release bearing loads increase friction and elastic losses in the release system and consequently lead to even higher pedal effort and deterioration of engagement performance. This means that measures that were originally intended to improve clutch modulation can have exactly the opposite effect.

The following paper will deal with those criteria which lead to optimum clutch design. Statements regarding heat stress and the clutch safety factor apply specifically to passenger car clutches. The other observations apply in general.

1. Satisfactory Service Life

The primary stress on the clutch occurs during vehicle start-up. Figure 2 shows a schematic representation of a start-up cycle. The friction work generated by the speed differential between the engine and the transmission is converted into heat. The clutch has to be designed to prevent overheating and excessive facing wear. The friction work per unit of facing friction surface--designated as specific heat stress 'a'--may not exceed certain limits.

Based on many years of experience, LuK GmbH uses the following engine speeds to calculate the heat stress:

Start-up on level ground:

$$n_{eng} = \frac{1}{6} \text{ (engine rpm at peak torque)} + 1000 \text{ rpm}$$

and on a 26% grade:

$$n_{eng} = \frac{1}{6} \text{ (engine rpm at peak torque)} + 2000 \text{ rpm}$$

Based on our experience, LuK uses the following limit values for the heat stress:

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- Start-up on level ground:

  26 Nm/cm^2 to max. 32 Nm/cm^2

- Start-up on a 26% grade:

  max. 300 Nm/cm^2

Figure 3 shows the heat stress for a representative group of passenger car clutches for vehicles with between 1 liter and 3 liter engine displacement. The lower part of the graph shows the values for start-up on level ground. The limit values 26 and 32 Nm/cm^2 cited above are shown as broken lines.

It is obvious that in practice the heat stress is not the same for all vehicles. It increases with increasing engine displacement. The average for vehicles with 1 to 1.2 liter engines is about 22.5 Nm/cm^2; for vehicles with 2.6 to 3 liter, it is about 32 Nm/cm^2. The solid line shows the average trend.

The reason for this is that drivers of vehicles with small engines often start up at higher rpms than do drivers in vehicles with larger engines.

They also shift more frequently, exerting correspondingly higher friction work on the clutch. In addition, the wear reserve of small clutches is often reduced.

Therefore clutch design should strive to adhere to the specific heat stress values for start-up on level ground, represented by the solid line. The values represented by the dash-dot line should not be exceeded.

The top graph shows heat stress for start-up on a 26% grade. Most of the values lie below the 300 Nm/cm^2 limit. In this case the average heat stress is nearly constant over the entire range. Only the 3 vehicle groups 1.8 to 2 liter, 2.2 liter and 2.4 liter significantly exceed the limit value. These cases involve diesel vehicles with unfavorable drive line ratios. Grade-ability and the allowable trailer load of these vehicles is limited in comparison to standard designs.

2. Reliable Transmission of Engine Torque

Reliable transmission of engine torque is essential, even under extreme conditions. The criterion for this function is the clutch safety factor, that is, the ratio between the clutch torque capacity and the engine torque. The clutch torque capacity itself is calculated based on the mean friction radius, the clutch clamp load and the coefficient of friction of the clutch facings.

The clutch safety factor for asbestos facings should be min. 1.2, calculated with a coefficient of friction of 0.27. Until now, our goal for non-asbestos facings has been min. 1.3. Based on current test results, we will again be able to calculate with a clutch safety factor of 1.2 for some non-asbestos materials.

In Figure 4 the clutch safety factor is plotted versus the engine displacement. The average value for each engine displacement class is marked with a plus sign. As you can see, the average clutch safety factor is over 1.4. In many cases we could visualize reducing the clutch safety factor, and with it the clamp load, by up to 25%.

3. Low Release System Losses

The release system consists of a series of load transmitting components, such as the lever, bearing, control cables and return springs, all of which contribute more or less significant losses due to friction or elasticity. Some of these losses are very high and have considerable effect on pedal effort and clutch modulation.

At LuK we have measured various mechanical and hydraulic release systems. Figure 5 shows the results. The table in the middle of the slide shows the total loss V_{tot}, divided up into load loss V_F and travel loss V_S. The resulting total loss values for hydraulic release systems amount to up to 40%, and up to 55% for mechanical systems. The graph at the top of the illustration shows the effect on the pedal effort. The broken line shows the theoretical pedal effort curve vs. the calculated pedal travel based on the clutch release bearing load and the lever ratio of the release system without any losses. The solid curve represents a measured pedal effort curve. This example clarifies the effect of losses. The measured pedal effort is about 50% higher and the measured pedal travel about 40% greater than these values would be if there were no losses in the system.

In many cases it is possible to reduce loss significantly with an acceptable cost. If, for instance, we can reduce the total loss in the release system from 50% to 40%, we can cut the pedal effort by 20%. It is often easier to achieve pedal effort reduction in this area than by making changes in the clutch.

4. Cushion Characteristic, Diaphragm Spring Characteristic and Clutch Lift-off

4.1 Relationships within the Clutch

Clutch torque build-up during clutch engagement and release bearing load are dependent on the cushion deflection curve, on the diaphragm spring characteristic, and on the clutch lift-off. Some essential aspects of clutch engagement performance were discussed during the last clutch symposium under the topic "Operating and Engagement Performance of Motor Vehicle Dry Friction Clutches." That presentation studied and illustrated primarily the effect of friction and elasticity in the release system on engagement performance.

Our purpose in this presentation is to show the effects of the cushion and diaphragm spring characteristics as well as the amount of clutch lift-off on the release bearing load and clutch modulation.

Figure 6 illustrates the function of the clutch and the relationships which determine the loads. The clutch is mounted on the flywheel. The disc is clamped between the clutch pressure plate and the flywheel friction surface. The clutch disc facings are cushioned axially by the spring segments. A diaphragm spring supported in the clutch cover presses the pressure plate against the clutch disc. The diaphragm spring fingers enable the clutch to disengage.

There are three primary loads on the clutch: the cushion deflection load, the diaphragm spring load and the release bearing load. The three loads are maintained in constant equilibrium, that is, when the clutch is disengaged, during clutch engagement and when engaged. In each case the (Diaphragm Spring Load F_T minus

Cushion Deflection Load F_B) times Lever Arm a equals Release Bearing Load F_A times Lever Arm b (Equation # 1). Based on this equation, we can derive that the diaphragm spring load minus the cushion deflection load is equal to the release bearing load multiplied by the lever arm ratio, that is the lever ratio of the diaphragm spring in the clutch (Equation # 2).

Consequently the diaphragm spring load minus the cushion deflection load together with the lever ratio of the clutch determines the clutch release bearing load (Equation # 3).

This also means that the cushion deflection load F_B is equal to Diaphragm Spring Load F_T minus Release Bearing Load F_A times Lever Arm b divided by Lever Arm a (Equation # 4). This formula can be used for determining the effective cushion load and thus the clutch torque capacity during the engagement cycle.

Figure 7 shows these relationships based on the clutch characteristic. The broken line represents the diaphragm spring load curve. The cushion deflection characteristic is drawn in as a dash-dot line. The cushion deflection curve must intersect with the diaphragm spring curve at the operating point. In this condition, without release bearing load, the diaphragm spring load has to be equal to the cushion deflection load. The diaphragm spring load minus the cushion deflection load yields the release bearing load multiplied by the clutch lever ratio, shown in the graph as a solid curve.

This is true if we assume that all the other elements in the clutch are rigid. Only then are the relationships between the loads and the respective travel values easy to calculate. In reality the relationships become more complex because of the elastic deformation of various clutch components in conjunction with the non-linear diaphragm spring and cushion deflection characteristics.

4.2 Influence of the Cushion Deflection and Diaphragm Spring Characteristics on Engagement Performance

We conducted clutch measurements with the goal of determining the effects of the diaphragm spring and cushion deflection characteristics on engagement performance or on clutch torque build-up. For this purpose we used clutches with extreme diaphragm spring and cushion deflection characteristics.

The left-hand graph in Figure 8 shows curves for the two clutches we used. Clutch A has a so-called steep characteristic, whereas clutch B exhibits a flat characteristic. Consequently, as can be seen from the graph, the wear capacity of clutch B is lower than that of clutch A.

The graph on the right shows differing cushion deflection curves. Clutch disc 1 demonstrates a strongly progressive curve with a very flat initial slope, while, in contrast, clutch disc 2 is not at all progressive.

The following Figure 9 compares the engagement characteristic and the clutch torque build-up for clutch discs 1 and 2 combined with both clutches A and B. As the graph shows, clutches A and B have absolutely the same engagement characteristic for the same cushion deflection curve, although they represent extremely different diaphragm spring characteristics. In each case the deciding influence is the cushion deflection characteristic.

We have to conclude therefore that the diaphragm spring characteristic--specifically whether it is steep or flat--has practically no influence on clutch torque build-up, that is on clutch engagement performance.

4.3 Release Bearing load Comparison

Figure 10 illustrates the release bearing load curve for the clutch disc with the strongly progressive characteristic curve. It is shown for the clutches A and B used in the previous discussion--one with a very steep and one with a flat diaphragm spring characteristic.

The top graph shows the release bearing load for the new clutch, and the bottom one, the release bearing load after 1.5 mm facing wear. The solid line represents the release bearing load of clutch A with a steep diaphragm spring characteristic. The broken line represents that for clutch B with a flat characteristic.

One can see that the steep diaphragm spring characteristic results in significant advantages for the release bearing load of a new clutch. For the new condition, clutch A with 1100 N release bearing load is significantly better than clutch B with about 1500 N. Once the facing has worn, however, the release bearing load for clutch A, as a result of its steeper diaphragm spring characteristic, increases faster than it does for clutch B. After 1.5 mm facing wear, clutch A still has a small advantage (see bottom graph).

This all speaks well for a relatively steep diaphragm spring characteristic.

4.4 The Influence of the Clutch Lift-off, or the Clutch Lever Ratio, and the Cushion Deflection Characteristic on Operating Comfort

In the preceding section clutches with identical lift-off values were used for better representation of the influence of the cushion deflection and the diaphragm spring characteristic curves on engagement performance and release bearing load.

Based on the torque equation for determining the loads (Figure 6, Equation 3), it was shown that the clutch release bearing load is directly dependent on the clutch lever ratio, hence release bearing load decreases as the lever ratio increases. At the same time the clutch lift-off also decreases.

The question remains as to how the reduction in clutch lift-off affects the rise in clamp load and torque build-up during the engagement cycle, thus influencing clutch operating comfort.

First we will compare two typical examples of push-type clutches with extremely different lift-off values of 1.75 and 1.35 mm at 7 mm release travel.

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In order to obtain a meaningful comparison, we used the same diaphragm spring characteristic for both clutches, as shown in Figure 11.

Figure 12 shows the cushion deflection characteristics with a cushion of 1.0 and 0.7 mm resp. for the clutch discs used.

Figure 7 used characteristic curves to illustrate the relationship between the diaphragm spring load, the release bearing load and the cushion deflection load, which is equal to the clamp load. Only the area to the right of the vertical line passing through the operating point is critical for release bearing load and engagement performance. The following discussion will be limited to this range.

Up until now we have not taken the elasticity of the clutch components into consideration.

Figure 13 shows the characteristic curves without the effect of the cushion deflection, but taking into consideration the elasticity of the load bearing components in the clutch. The release bearing load is shown as a dash-dot line, the clamp load as a broken line, and the pressure plate lift-off as a solid line, all plotted vs. the release travel. The effective pressure plate clamp load--which I will refer to simply as the clamp load--does not drop abruptly to zero at the beginning of the release travel. It decreases to zero only after a certain release travel--a little more than 2 mm in the example shown. Elastic deformations in the clutch are responsible for this. These are: the diaphragm spring finger deflection, the resiliency of the clutch cover and the elasticity of the diaphragm spring supports. All of these act like a linear cushion. Therefore, during the engagement cycle, the clamp load does not increase abruptly, even without any cushion deflection, but rather builds up almost linearly over part of the engagement travel. This portion of the travel varies in direct proportion to the elasticity of the clutch. However, the elasticity of the clutch does not usually ensure smooth engagement performance. Additionally, we need a clutch disc with progressive cushion deflection.

Figure 14 shows the characteristic curves for the clutch with the diaphragm spring characteristic as shown in Figure 11. Here we take into consideration the elasticity of the clutch and the effect of the cushion deflection load. The solid lines represent the individual characteristics for the new clutch, and the broken lines for 1.5 mm facing wear.

While releasing the clutch, the pressure plate releases the load on the clutch facings. In this process, the clamp load, which is always equal to the cushion deflection load, decreases toward zero, beginning at the operating point. As the release travel continues, beginning after about 5 mm, the pressure plate lifts off the clutch facings to provide clearance equal to the pressure plate lift-off minus the cushion travel.

When we compare this example to the previous one without cushion deflection, it is obvious that the clamp load build-up begins much earlier due to the effect of the cushion deflection. At first the load rise is very flat and increases progressively with increasing engagement travel. The clamp load and clearance curves are essentially parallel for new condition and for 1.5 mm facing wear. They are simply displaced somewhat because of increased elastic deformation due to the higher diaphragm spring load and release bearing load as a result of wear. The release bearing load is greater after wear than in new condition because of the steep diaphragm spring characteristic we chose to use.

The relationship between new condition and after 1.5 mm facing wear is similar for both clutch examples. Therefore in the following comparison we have omitted the curves for clamp load and clearance after 1.5 mm facing wear.

Figure 15 compares two push-type clutch assemblies, K1 and K2, with different lift-off values. K1 has a high lift-off and cushion deflection, K2 a relatively low lift-off and a correspondingly low cushion deflection. The curve of the pressure plate clamp load exerted on the clutch disc with respect to the release travel and the clearance are almost identical, although the clutch lift-off for the two clutches is very different.

It is plain to see that given the proper adjustment of the cushion deflection characteristic to the pressure plate lift-off, we can achieve the same engagement characteristic. This is independent of the magnitude of the lift-off, therefore vehicle start-up performance is not dependent upon the amount of lift-off.

Figure 16 compares the release bearing loads for the clutches tested. As might be expected, the push-type clutch K2 with the low lift-off and reduced cushion deflection exhibits an essentially lower release bearing load than clutch K1 with the higher lift-off and cushion deflection.

Theoretically we can expect that the two clutches will exhibit the same engagement performance. To confirm this, we evaluated them in the same vehicle. The pedal effort for K1 was 130 N, and about 100 N for K2. Engagement performance was good in both cases. Clutch K2, with the low lift-off and cushion deflection, tended to be better.

This can be attributed to lower loads and consequently lower friction losses, enabling more sensitive clutch operation.

4.5 Comparing Push-type and Pull-type Clutches

It is generally assumed that a pull-type clutch has a significantly better release bearing load than a push-type clutch. Therefore in Figure 17 we compare a pull-type clutch assembly K3 and the push-type assembly K2 from the previous example. Both clutches have the same lift-off, the same diaphragm spring characteristic, and the same cushion deflection characteristic.

The clamp load build-up for both clutches runs parallel over the first part of the engagement travel, that is, both clutches exhibit equally good engagement performance in this range. Toward the end of the engagement travel, the clamp load of the pull-type clutch rises more steeply. This can have a slight adverse effect on engagement characteristics when the clutch is almost locked up.

Figure 18 compares the release bearing loads of the pull-type clutch K3 and the push-type clutches K1 and K2. The push-type clutch K1 with the high lift-off and cushion deflection has--as expected--the highest release bearing load. As already illustrated in Figure

16, the push-type clutch K2, with its low lift-off and reduced cushion deflection, exhibits a 20% reduction in release bearing load. In comparison to K2, the release bearing load for the pull-type clutch K3 is only about 12% lower.

This proves that the magnitude of the clutch lift-off has no influence on engagement performance, but it does affect the release bearing load, as well as engagement comfort. It also demonstrates that, if a push-type clutch is properly designed, it can achieve almost the same release load level as a pull-type clutch.

5. Summary and Conclusions

Within the framework of this presentation, we have discussed the essential factors involved in clutch design, in particular low release bearing load and engagement comfort.

The important variables for determining clutch dimensions, such as heat stress during start-up and the clutch safety factor, have been analyzed for a large number of passenger cars currently on the market. We have noted the high release system losses in some cases.

Based on calculations and measurements we have analyzed the influence of the diaphragm spring and cushion deflection characteristics as well as the clutch lift-off. We have examined the mutual interaction of these factors, and compared the results using typical examples.

It has been shown that there are still many possibilities for optimizing clutch design.

The following values must be taken into consideration when designing a clutch:

5.1 Satisfactory Service Life

The specific heat stress should not exceed the following limit values (Figure 19):

a) Start-up on level ground:
   Engine displacement limit values:
   22 Nm/cm^2 to 40 Nm/cm^2

b) Start-up on a 26% grade: 300 Nm/cm^2

5.2 Reliable Transmission of Engine Torque

A clutch safety factor of 1.2 is required in order to ensure reliable engine torque transmission. Any significantly higher values are to be avoided in order to keep clutch loads low.

5.3 Low Losses in the Release System

Release systems are frequently subject to high losses. In general LuK has no influence on release system design. The automobile manufacturer can exploit reserves in this area to reduce release bearing load.

5.4 Cushion Deflection Characteristic, Diaphragm Spring Characteristic and Clutch Lift-Off

a) Cushion Deflection Characteristic

Engagement performance is determined primarily by the cushion deflection characteristic. Characteristic curves with strong progressive slopes have proven to be advantageous. The beginning of the cushion deflection curve should be as flat as possible.

b) Diaphragm Spring Characteristic

The diaphragm spring characteristic curve doesn't have any significant effect on engagement performance, but it does influence clutch release bearing load and wear reserve. The diaphragm spring should exhibit a load ratio of about 1 to 0.6 between the peak and valley spring curve loads. This will ensure optimum operating comfort in new condition, limited release bearing load build-up after wear and sufficient wear reserve.

c) Clutch Lift-off

If we reduce the clutch lift-off, we can usually reduce the release bearing load considerably without decreasing engagement comfort. We recommend a lift-off of 1.2 mm. The cushion deflection must then be adapted to the reduced lift-off.

d) Pull-type clutches

With pull-type clutches we can reduce the release bearing load by about 12% in comparison to push-type clutches while using comparable lift-off. This advantage sometimes gets lost due to a more complicated release system.

The advantages of pull-type clutches include:

- simpler design and better cooling of the diaphragm spring
- longer diaphragm spring characteristic curve and as a result, higher wear reserve.

It is necessary to note the disadvantages of the pull-type clutch:

- more complicated release system
- difficult installation and removal from the engine and the transmission
- for disassembly, the transmission must be pulled farther from the engine due to the fact that the release bearing is attached to the clutch.

Consequently, a pull-type clutch only makes sense for engines with extremely high performance and sufficient package space, which means its use is limited to large clutches with high heat stress and the need for high wear reserve.

SATISFACTORY SERVICE LIFE

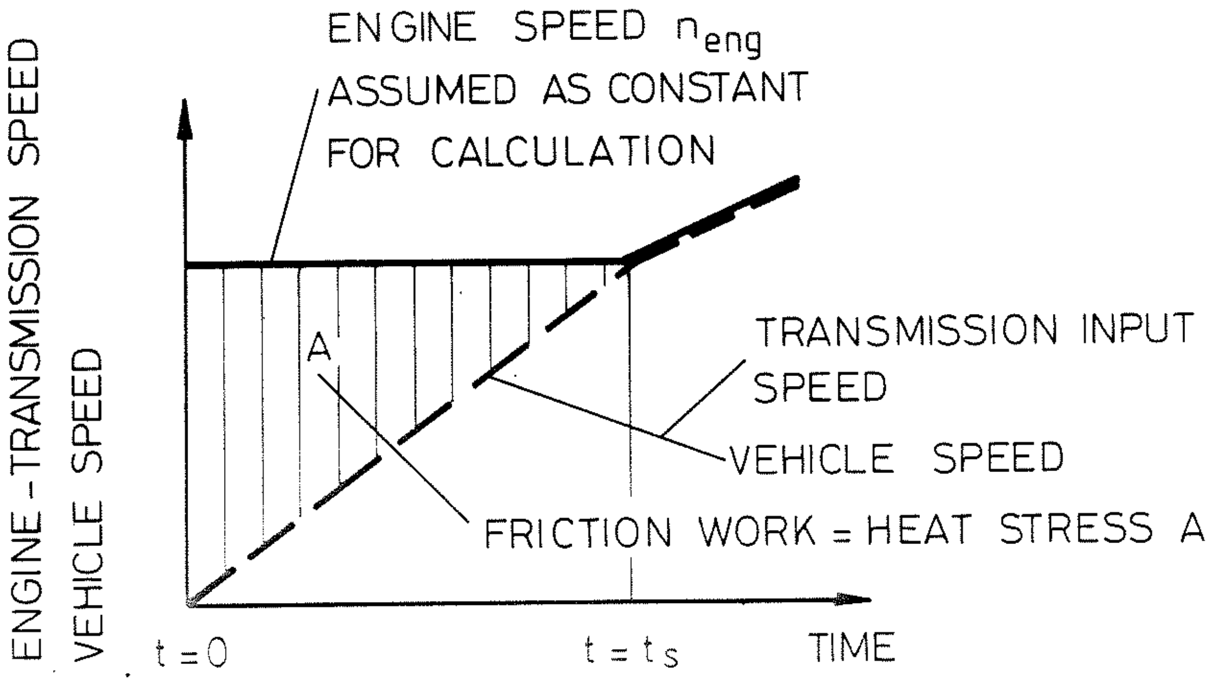
RELIABLE TRANSMISSION OF ENGINE TORQUE

LOW RELEASE SYSTEM LOSSES

LOW RELEASE BEARING LOAD

RELIABLE DISENGAGEMENT

SMOOTH ENGAGEMENT



HEAT STRESS $$A = \int_{t=0}^{t=t_s} M \times \omega_{rel.} \times dt$$

SPECIFIC HEAT STRESS $$a = \frac{A}{F} = \frac{1}{F} \int_{t=0}^{t=t_s} M \times \omega_{rel.} \times dt$$

ENGINE SPEED FOR START-UP ON LEVEL GROUND

$n_{eng.} = \frac{1}{6}$ ENGINE RPM AT PEAK TORQUE + 1000 RPM

ENGINE SPEED FOR START-UP ON A 26% GRADE

$n_{eng.} = \frac{1}{6}$ ENGINE RPM AT PEAK TORQUE + 2000 RPM

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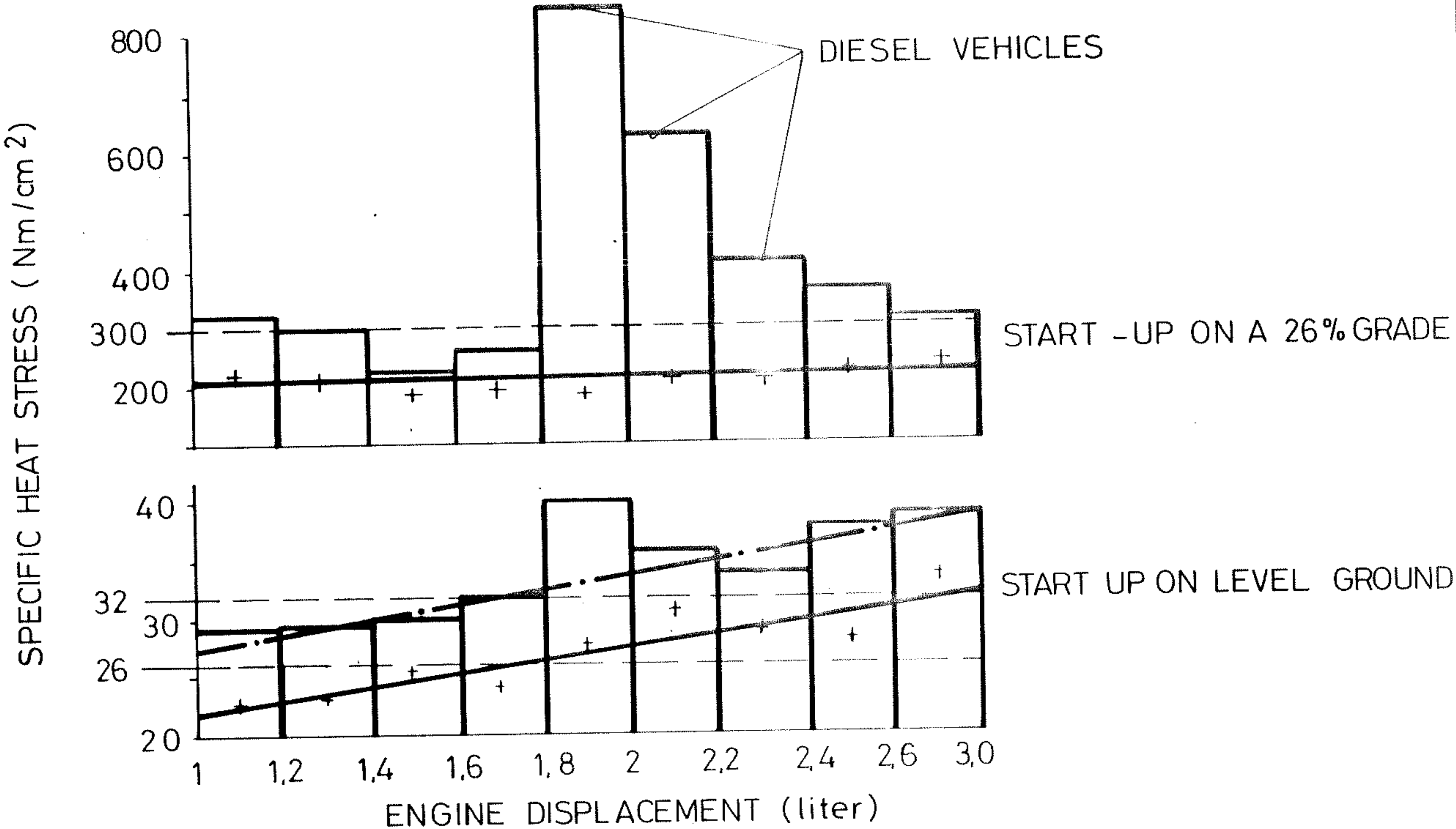

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SPECIFIC HEAT STRESS ON PASSENGER CAR CLUTCHES

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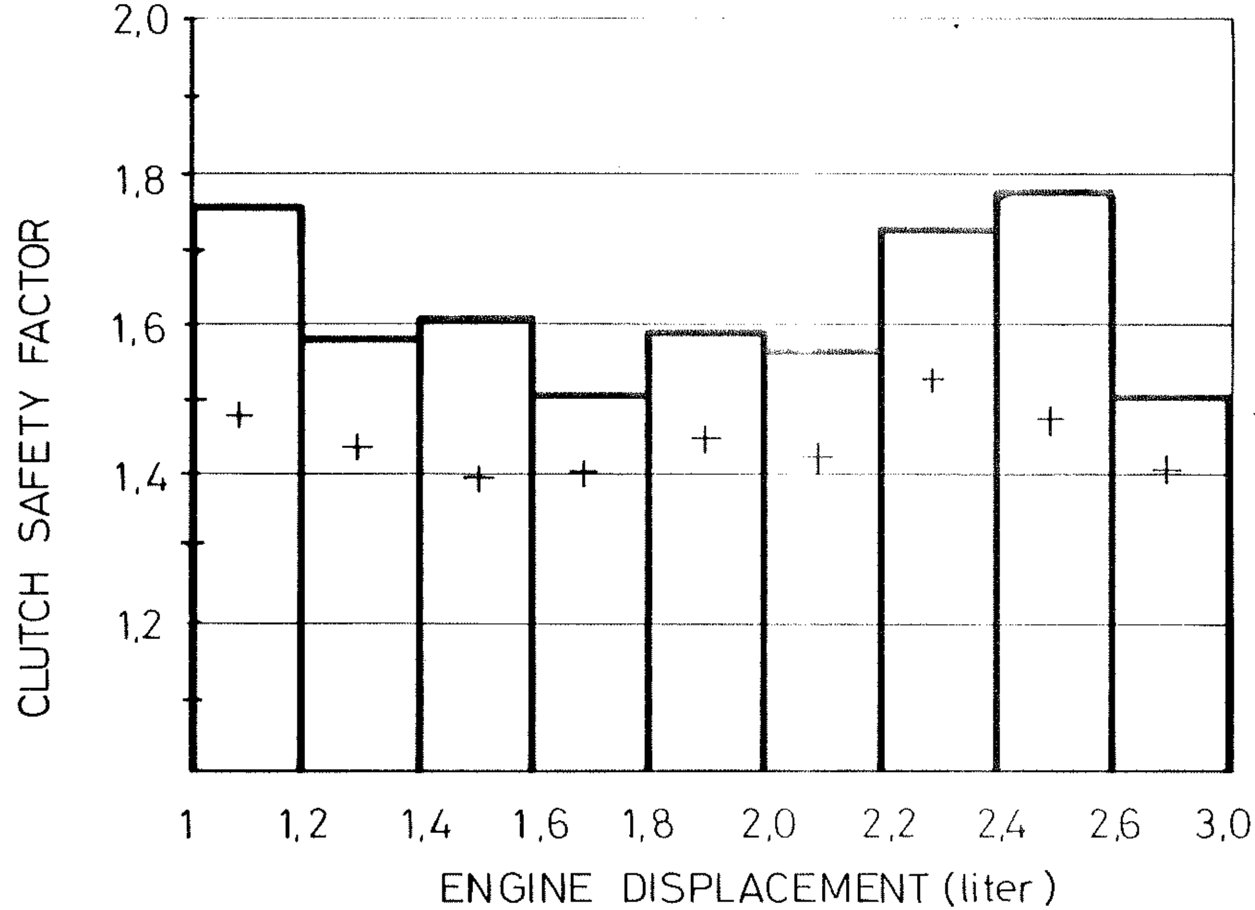

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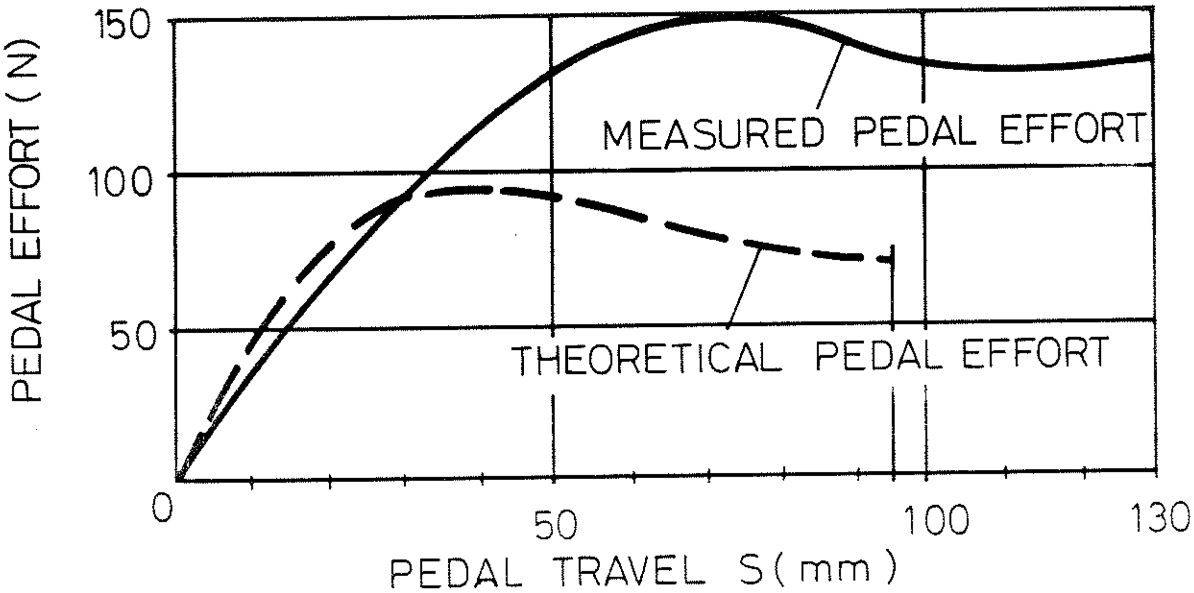
CLUTCH SAFETY FACTOR FOR PASSENGER CAR CLUTCHES



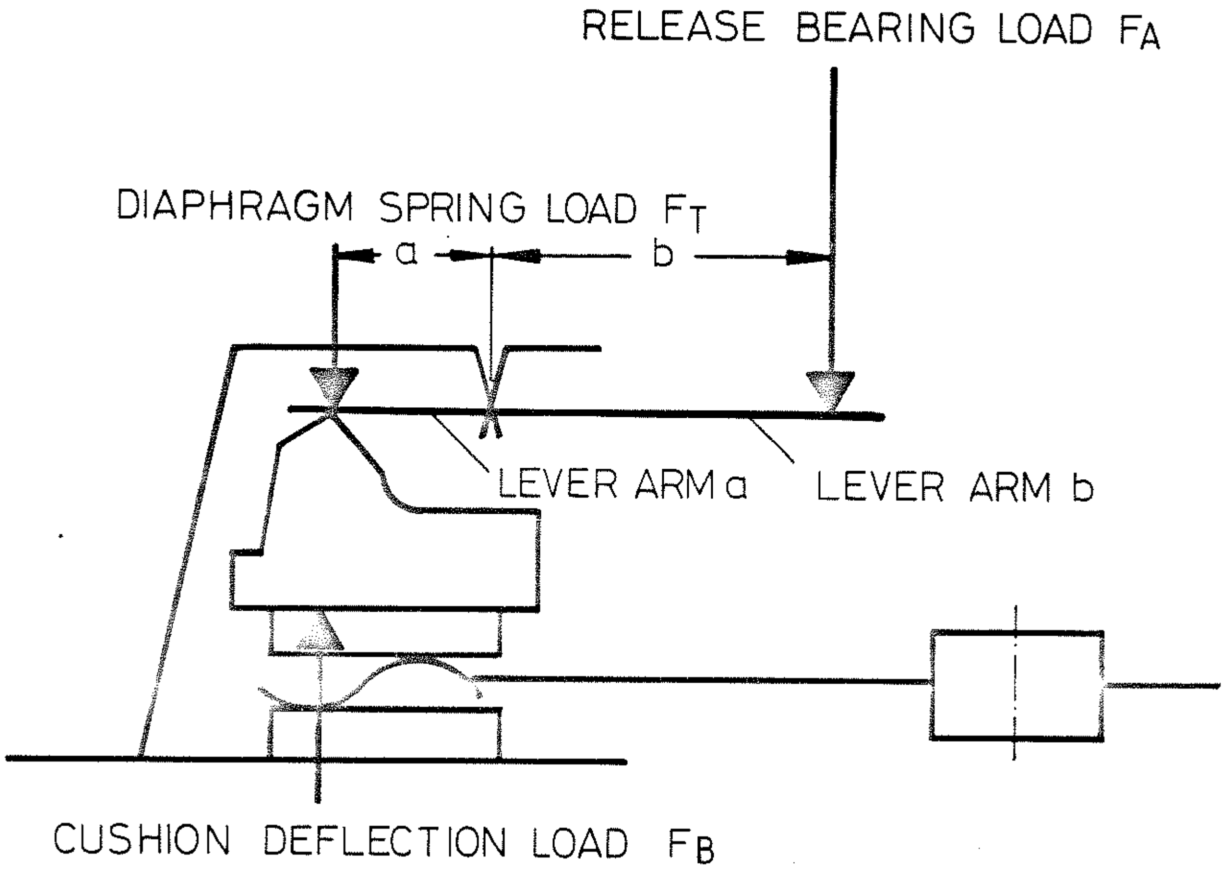
RELEASE SYSTEM	LOSSES		
	$V_{tot.}$ %	V_F %	V_S %
MECHANICAL	32 ÷ 55	20 ÷ 35	15 ÷ 30
HYDRAULIC	28 ÷ 40	15 ÷ 20	15 ÷ 25

$$V_F = \frac{\text{MEASURED PEDAL EFFORT} - \text{THEOR. PEDAL EFFORT}}{\text{MEASURED PEDAL EFFORT}} \times 100\ \%$$

$$V_S = \frac{\text{MEASURED PEDAL TRAVEL} - \text{THEOR. PEDAL TRAVEL}}{\text{MEASURED PEDAL TRAVEL}} \times 100\ \%$$

$$V_{tot.} = 100 - \frac{(100 - V_F)(100 - V_S)}{100}\ \%$$

05 04 86	RELEASE SYSTEM LOSSES	LUK



MOMENT EQUILIBRIUM

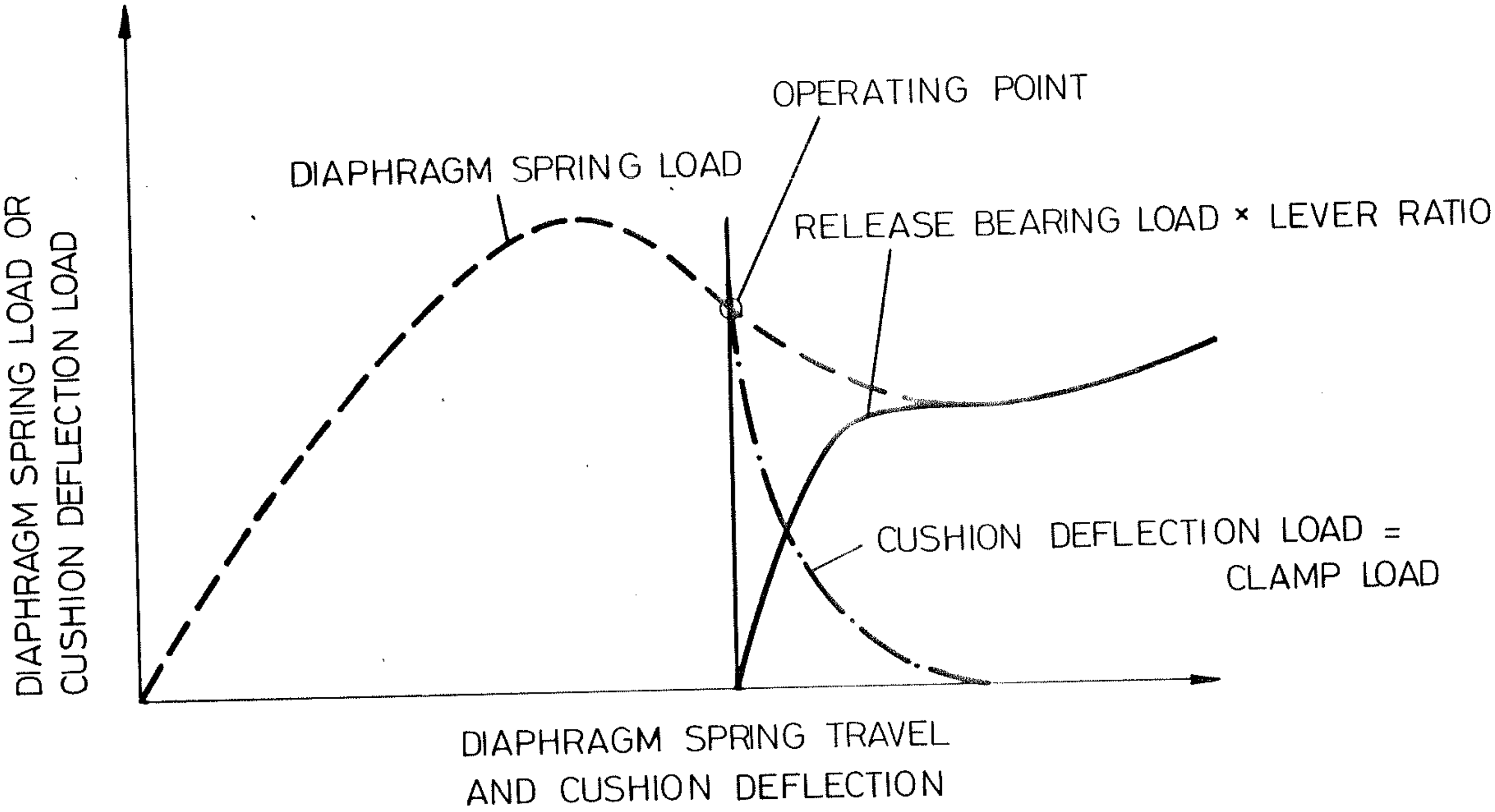
$(F_T - F_B)\,a = F_A \times b$ EQUATION 1

$F_T - F_B = F_A \times \frac{b}{a}$ EQUATION 2

$F_A = (F_T - F_B)\frac{a}{b}$ EQUATION 3

$F_B = F_T - F_A \times \frac{b}{a}$ EQUATION 4

06 04 86	MOMENT EQUILIBRIUM IN THE CLUTCH	LUK



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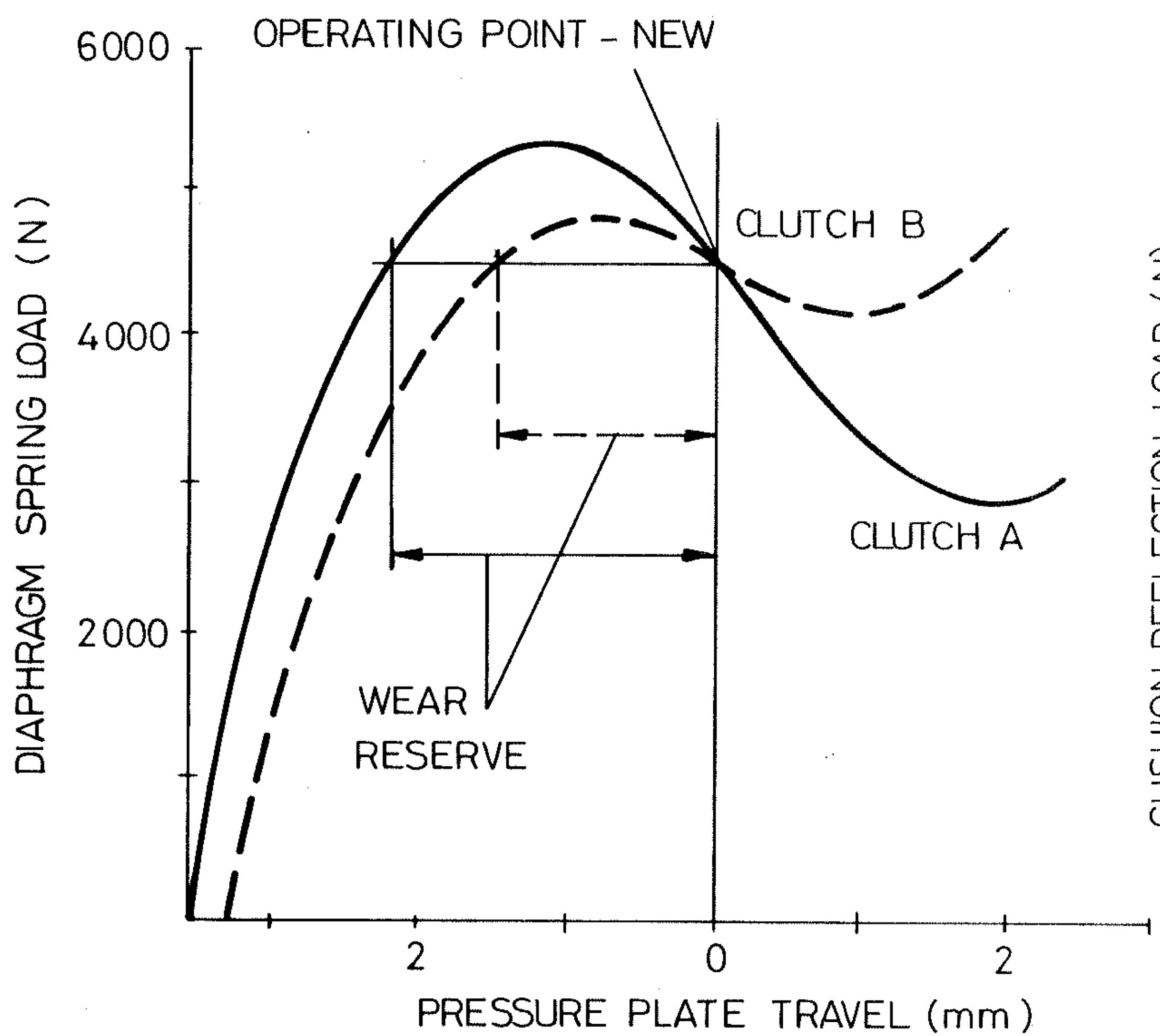
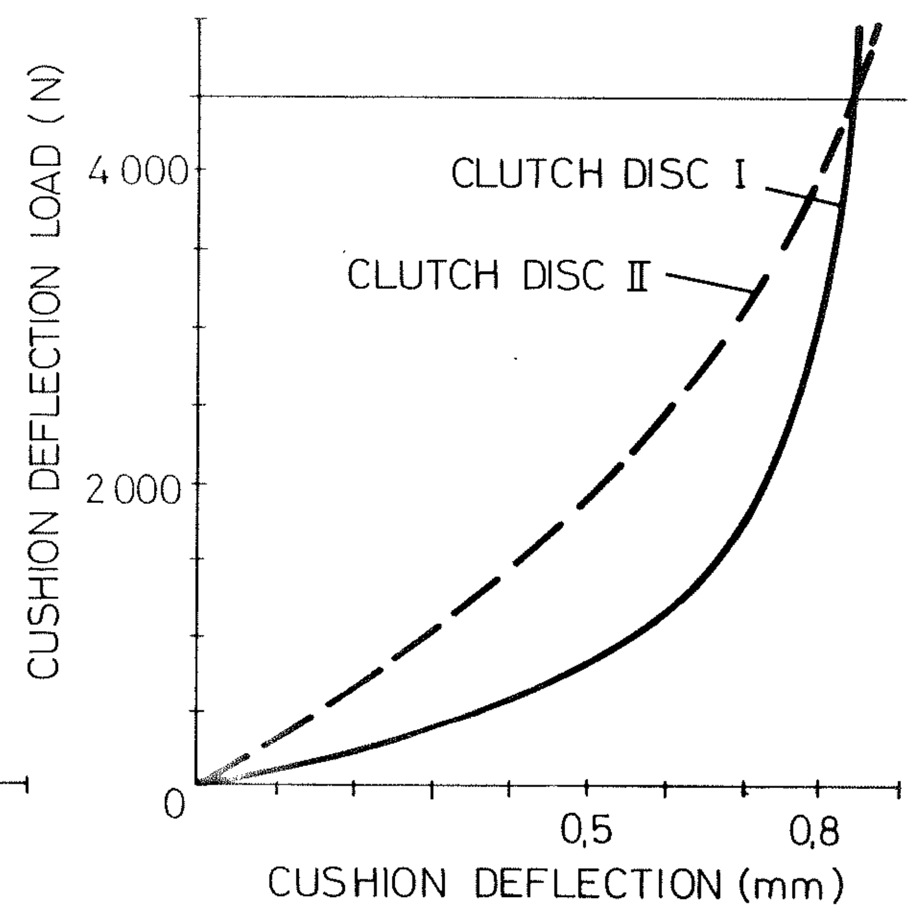
DETERMINING THE RELEASE BEARING LOAD BASED ON THE DIAPHRAGM SPRING AND CUSHION DEFLECTION CHARACTERISTIC CURVES





08 04 86 | DIAPHRAGM SPRING CURVE | CUSHION DEFLECTION CURVE

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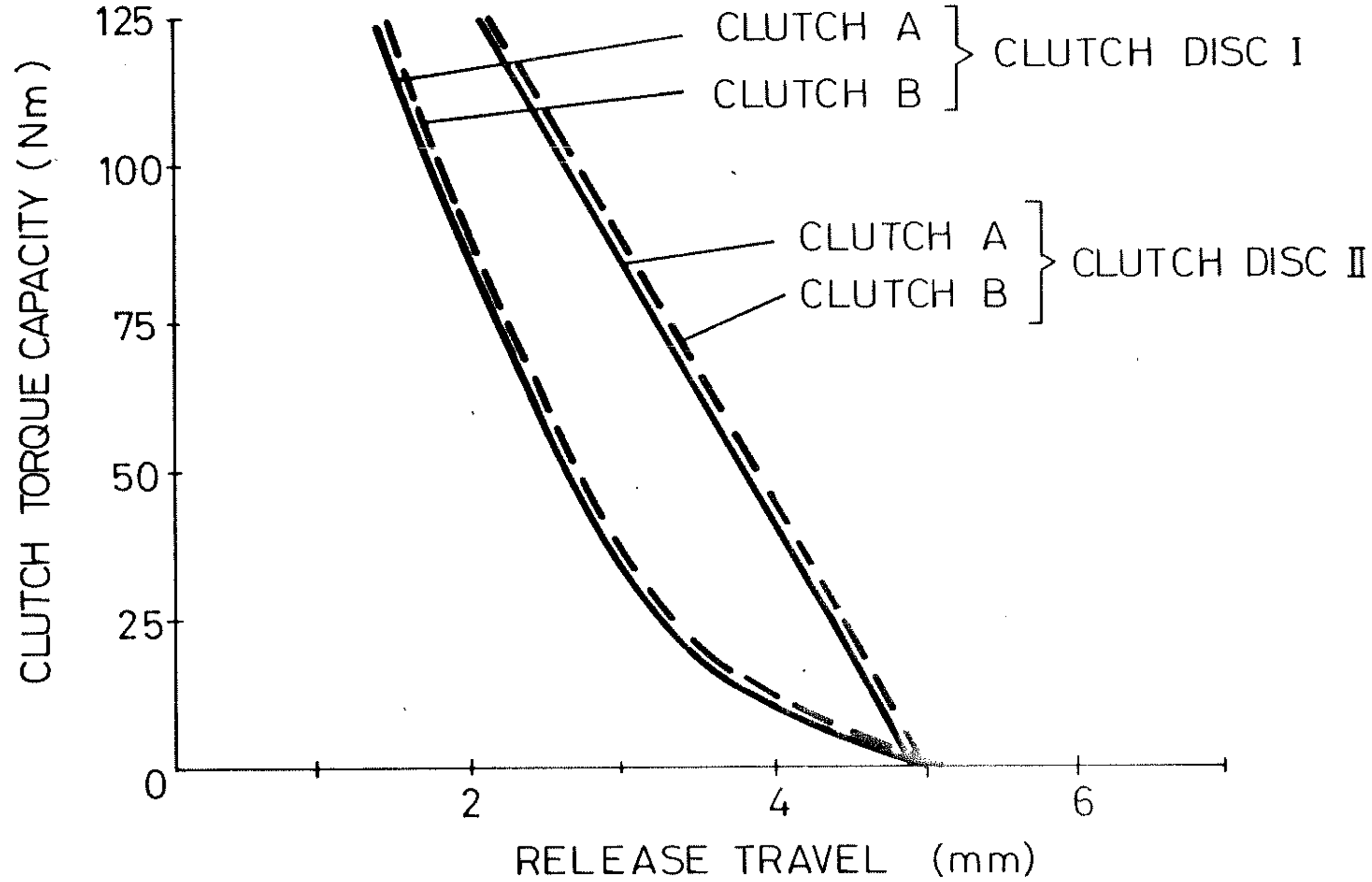
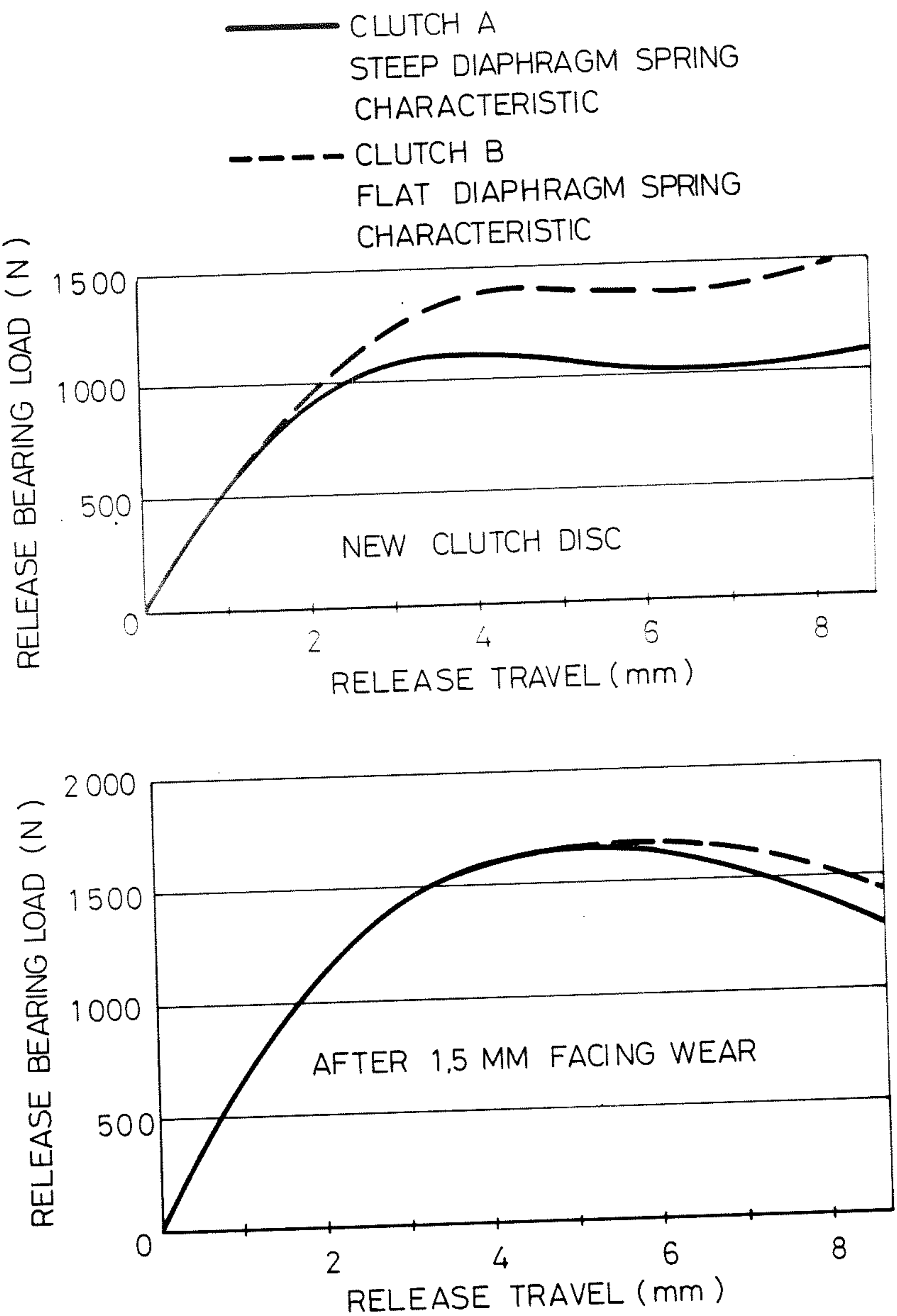


09 04 86	CLUTCH TORQUE COMPARISON	LUK



10 04 86	RELEASE BEARING LOAD COMPARISON BETWEEN CLUTCH A AND CLUTCH B

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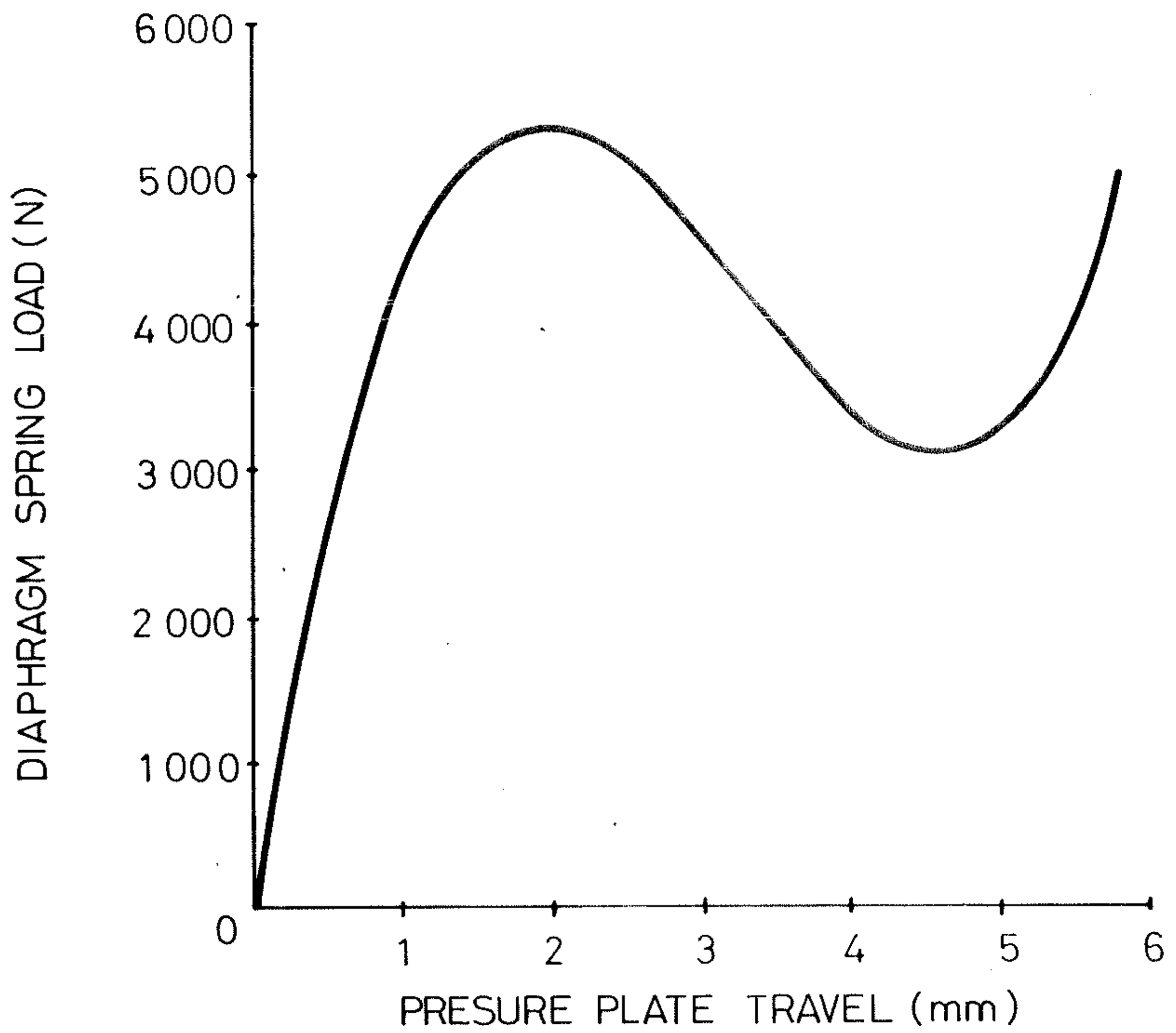
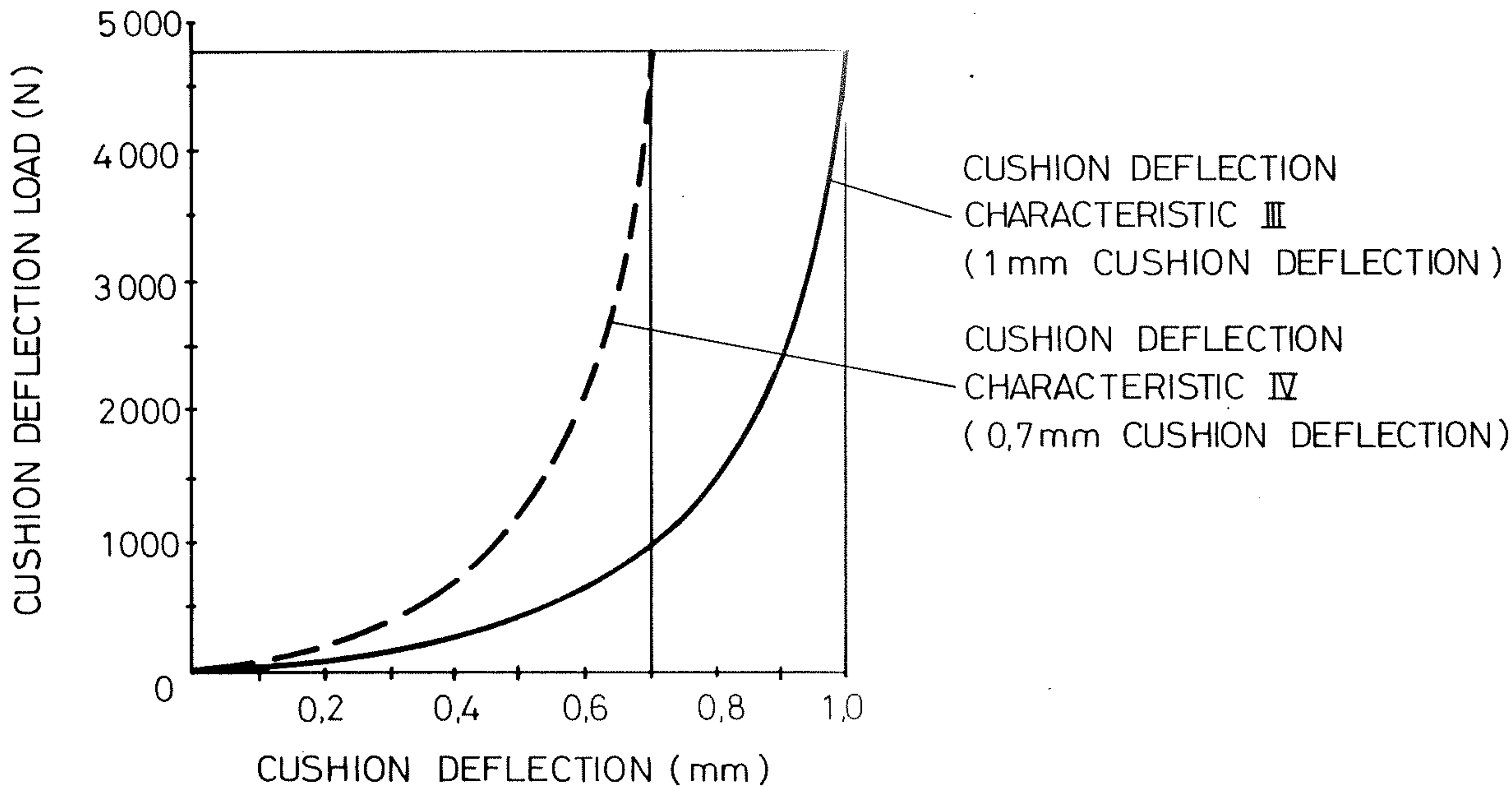
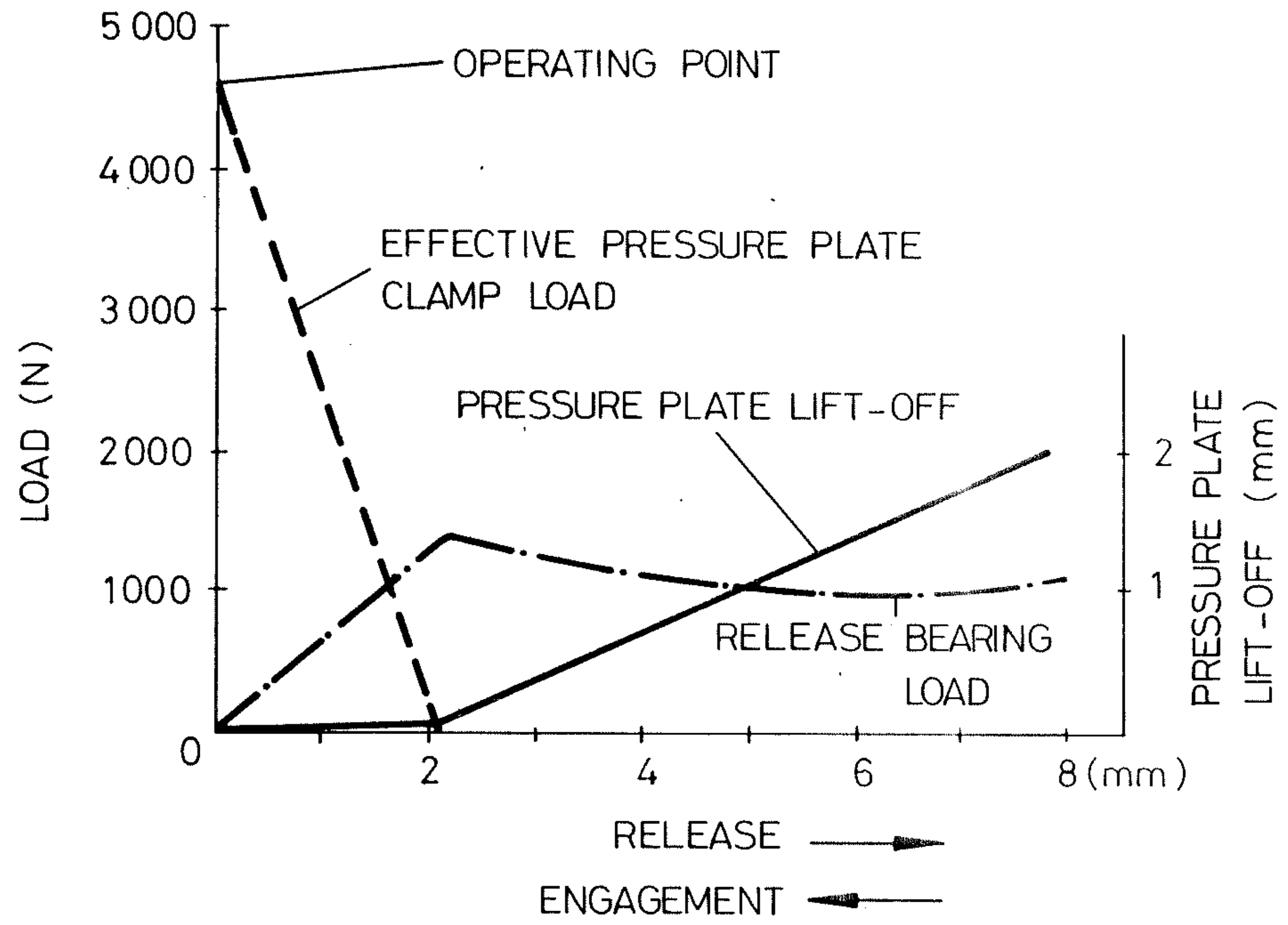


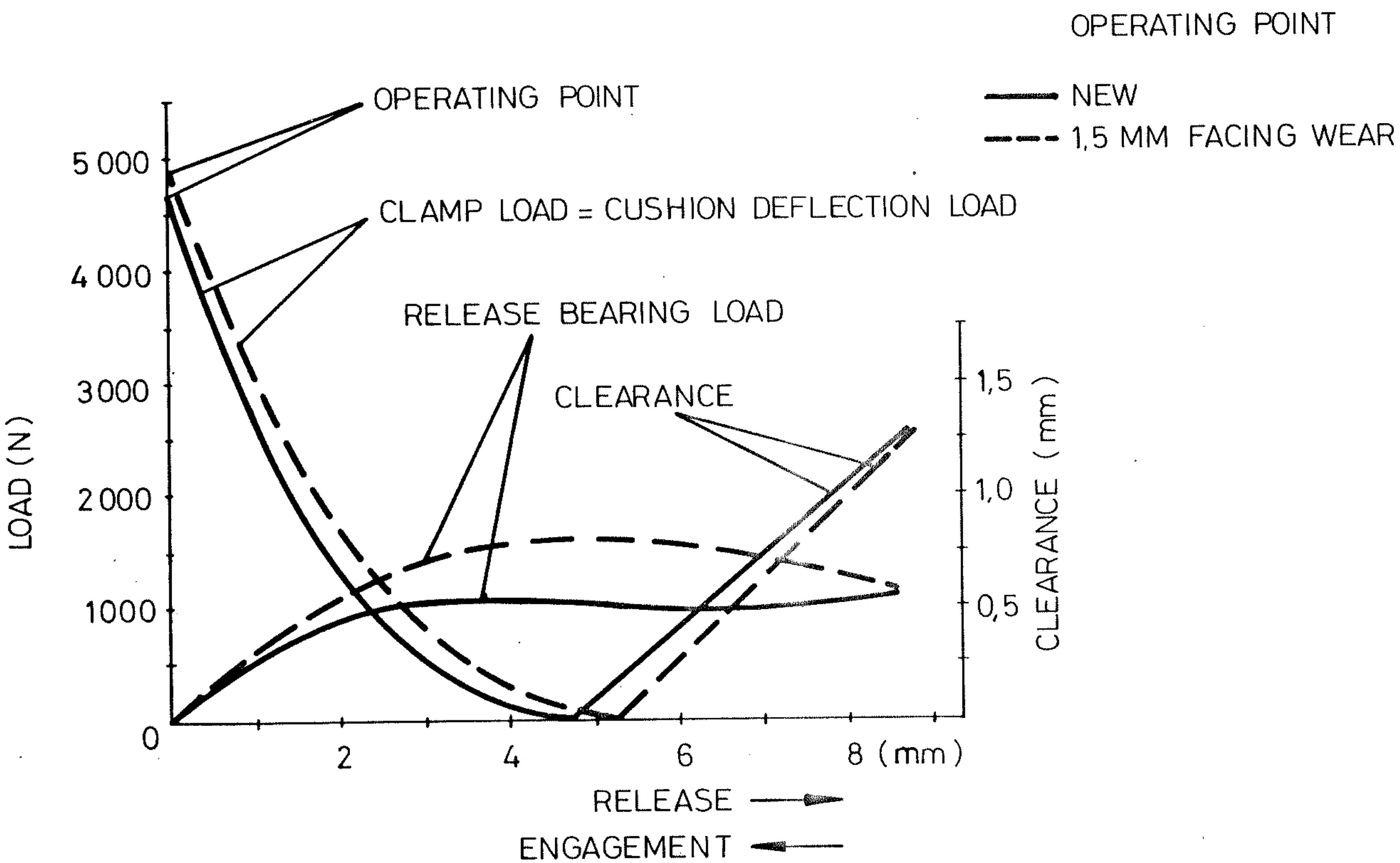
11 04 86	CLUTCH DIAPHRAGM SPRING CHARACTERISTIC



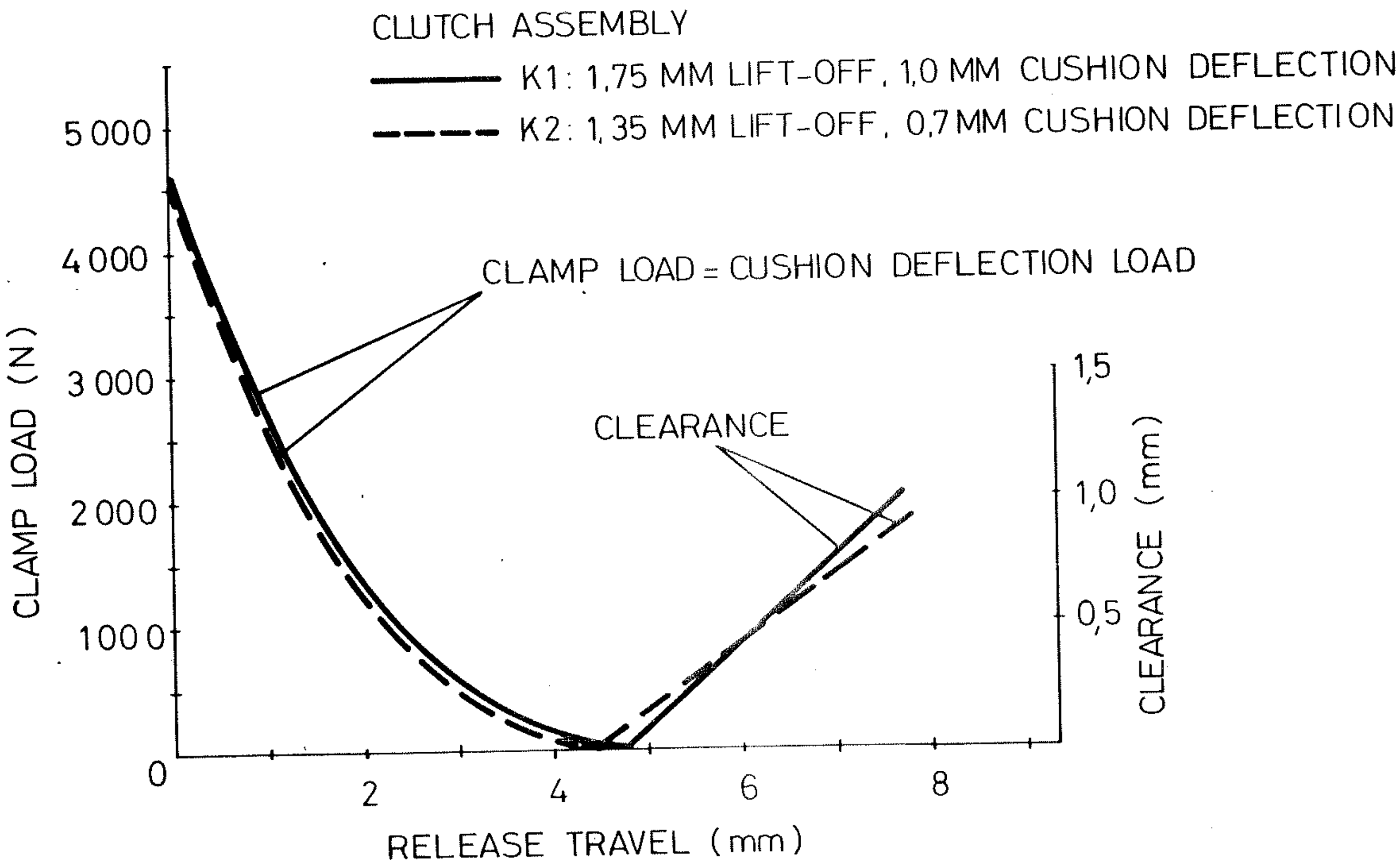
12 04 86	CUSHION DEFLECTION CHARACTERISTICS



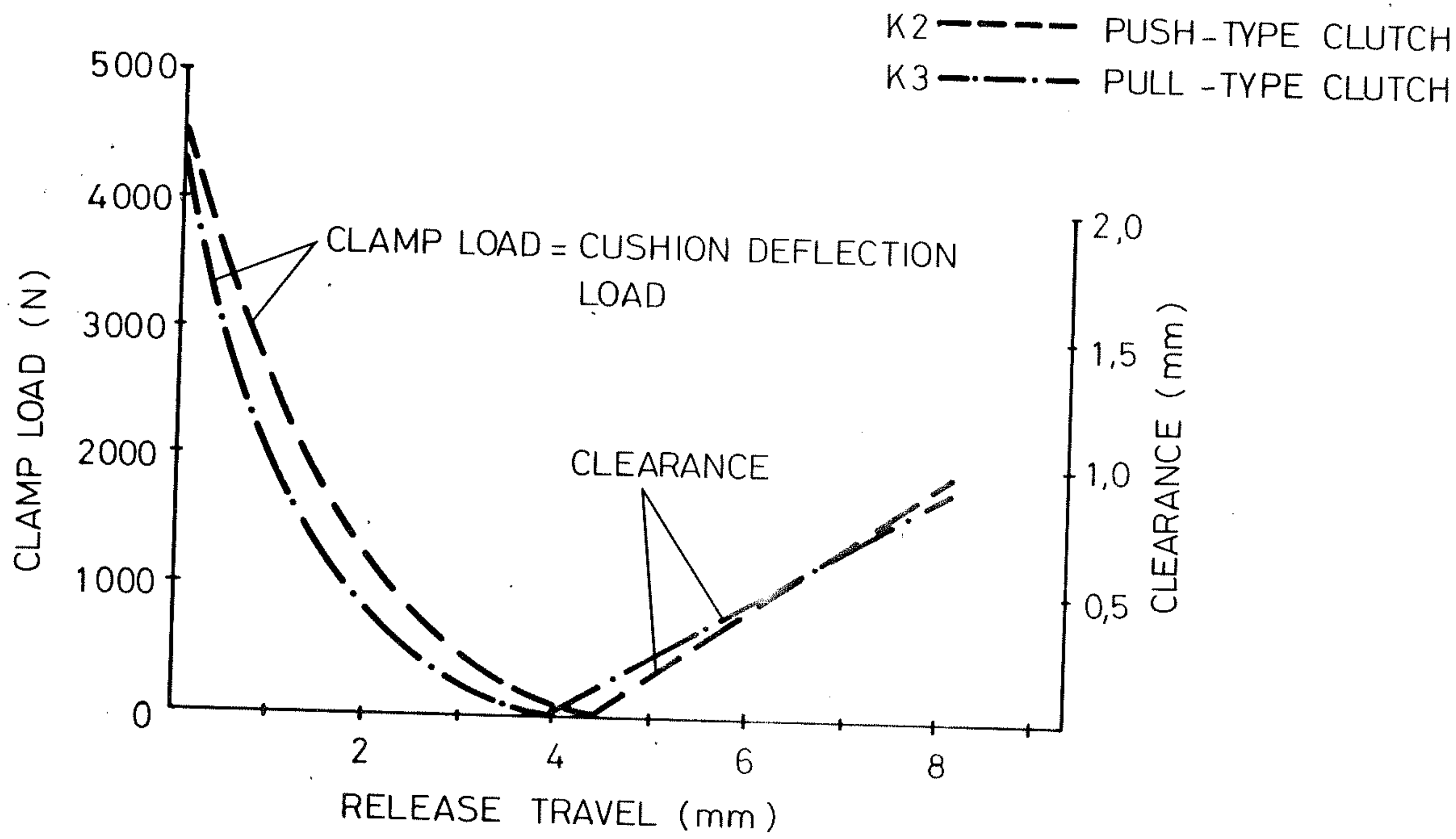
13 04 86	CLUTCH CHARACTERISTIC CURVES WITHOUT THE INFLUENCE OF THE CUSHION DEFLECTION	LUK



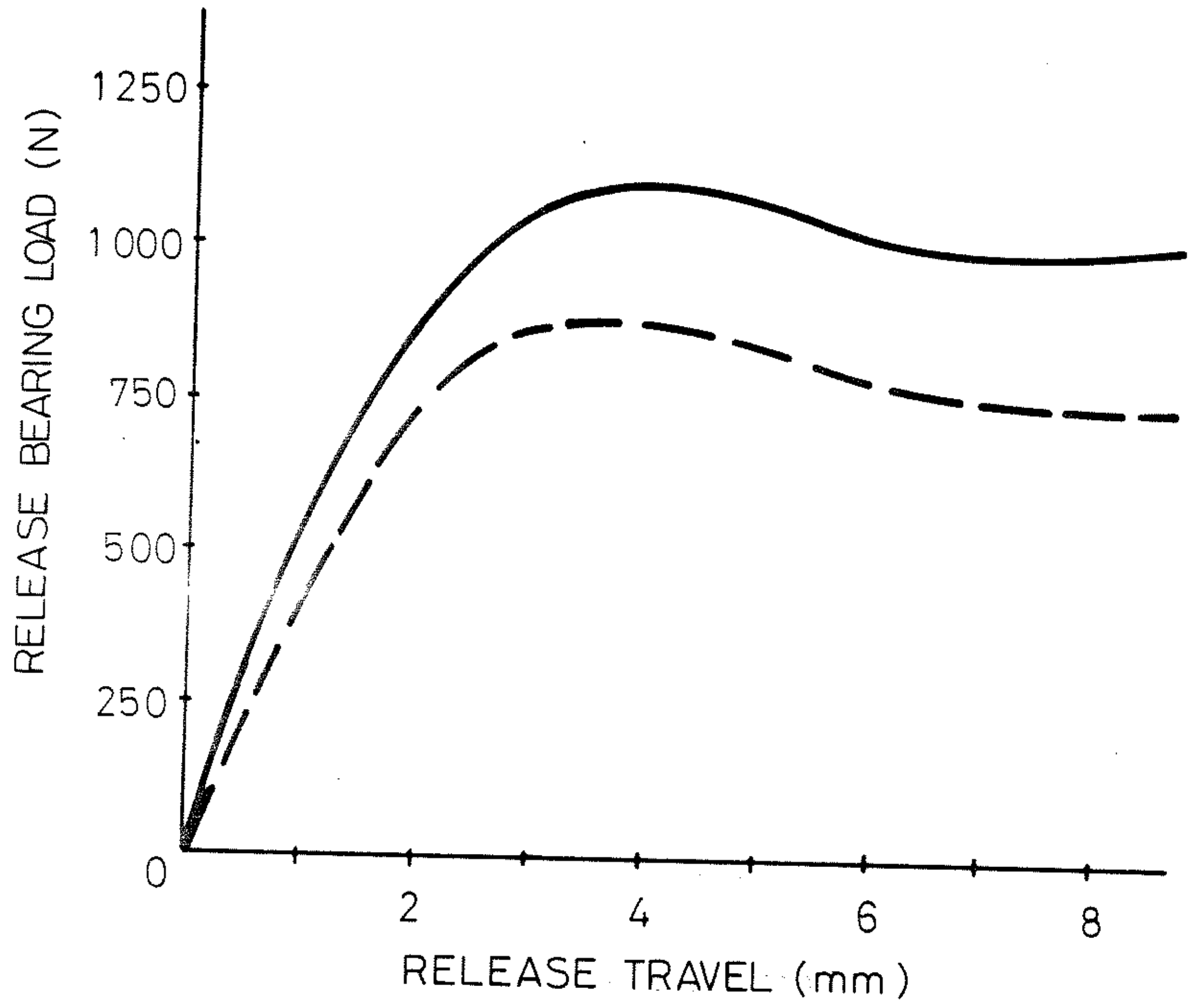
14 04 86	CLAMP LOAD, RELEASE BEARING LOAD AND CLEARANCE FOR CLUTCH K1 (WITH CUSHION DEFLECTION CHARACTERISTIC III)	LUK



15 04 86	CLAMP LOAD AND CLEARANCE FOR K1 AND K2	LUK



17 04 86	CLAMP LOAD AND CLEARANCE FOR PUSH-TYPE CLUTCH K2 AND PULL-TYPE CLUTCH K3

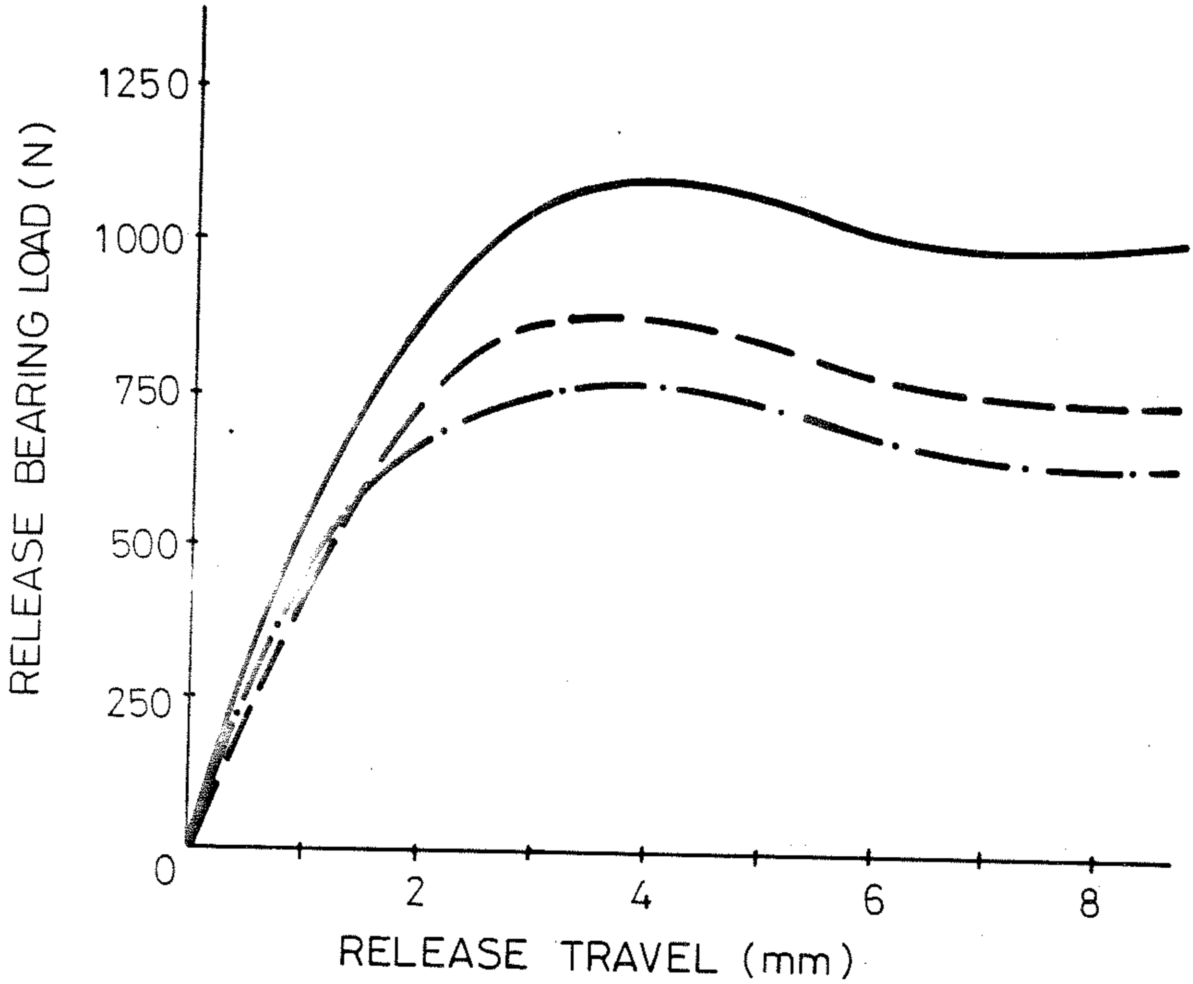


CLUTCH ASSEMBLY

K1 : PUSH - TYPE CLUTCH , 1,75 MM LIFT-OFF

K2 : PUSH - TYPE CLUTCH, 1,35 MM LIFT-OFF

16 04 86	COMPARISON OF RELEASE BEARING LOAD FOR CLUTCHES K1 AND K2	LUK



CLUTCH ASSEMBLY

K1 : PUSH-TYPE CLUTCH, 1,75 MM LIFT-OFF

K2 : PUSH-TYPE CLUTCH, 1,35 MM LIFT-OFF

K3 : PULL-TYPE CLUTCH, 1,35 MM LIFT-OFF

18 04 86	RELEASE BEARING LOAD COMPARISON FOR K1, K2, K3

LUK

- SPECIFIC HEAT STRESS:

  START-UP ON LEVEL GROUND:

  LIMIT DEPENDENT ON ENGINE DISPLACEMENT:
  22 Nm / CM^2 - 40 Nm / CM^2

  START-UP ON 26% GRADE:

  MAX. 300 Nm / CM^2

- CLUTCH SAFETY FACTOR: MIN. 1,2

- SLOPE OF DIAPHRAGM SPRING CURVE: $F_{MAX.} : F_{MIN.} = 1 : 0{,}6$

- CLUTCH LIFT-OFF: ABOUT 1,2 MM

19 04 86	RECOMMENDED DESIGN GOALS	LUK