Contents

Ortlinghaus multi-plate clutches and brakes

Ortlinghaus products are to be found in industrial transmission applications where the ability to transfer and control torque is required in drive outputs. Examples include machine tools, construction machines, marine transmissions, vehicles, heavy machines, gears, textile and paper machines.
The comprehensive range of products offered under the heading "THE TECHNOLOGY OF CONTROLLED TORQUE" offers proven standard and special solutions, which ensure optimum machine and plant efficiency and safety.
Our experienced team of specialist engineers are at your disposal - in particular for new designs - for consultation in selecting the most suitable clutch/brake for your application including determining the size and calculating the required output. In this way you can profit from years of experience gained in connection with large numbers of applications. In order that you can benefit from this consultation service in the most effective manner, we have prepared questionnaires for the individual product groups, with the aid of these you can describe the conditions of your particular application. We recommend that you complete these questionnaires and return them when making enquiries.
The following sections provide an overview of the most important properties of our friction materials, including how to determine the size of unit required, on the lubrication and cooling of clutches and brakes and general hints on installation.

Designations, symbols in formulas and units

Unless stated otherwise, the designations, symbols in formulas and units used in this catalogue are in accordance with VDI guideline 2241 and/or DIN 740 sheet 2.

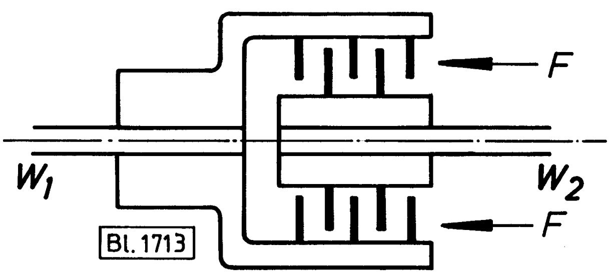

Fig. 1

housing (outer driver) or on a hub (inner driver) in such a way that they cannot rotate freely but can be displaced axially. For the transmission of torque by friction engagement from shaft W_1 to shaft W_2, the set of plates is compressed axially (Fig. 1). The force F required to do this is generated

mechanically,

electromagnetically,

hydraulically or

pneumatically

in accordance with the particular type of clutch.

The original Ortlinghaus Sinus® clutch plate

A particular feature of the Ortlinghaus multi-plate clutch is the use of the Sinus® plate, a component that has been proven over many years.

The special characteristic of this plate is that it is shaped in such a way that a cross-section along any diameter of the plate shows a corrugated or sinusoidal shape and this enables the plate to act like a spring (Fig. 2).

This sinusoidal shape enables the clutch to be engaged smoothly. During the engagement process the area of the frictional surfaces in contact with one another increases continuously and the sinusoidal shaped plates flatten to a plane surface. In the fully engaged state, each Sinus® plate has the shape of a conventional flat plate. The spring action of the Sinus® plates also en sures positive disengagement. Because of the sine-wave contour, only line contact remains in the disengaged position, resulting in minimum drag and idling heat.
It is perhaps worth mentioning that Ortlinghaus world patents in the field of spring-action plates were a real breakthrough with the trademark "Sinus®" becoming a standard term in power transmission technology.

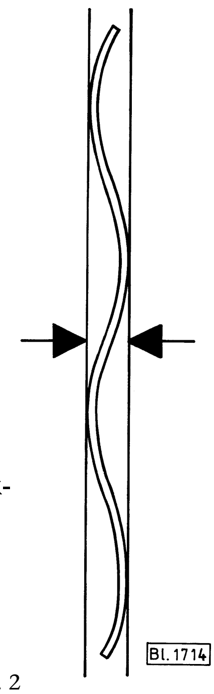

Fig. 2

Plate friction material

Different special friction materials are available for wet and dry-running clutches and brakes. The friction material used represents the most important part of each friction combination, which effectively consist of, in addition, the counter frictional surface and, in the case of wet-running, the oil. The friction combination influences the behavior of the clutch or brake when being engaged and disengaged, the permissible thermal loading, the behavior in terms of wear and thereby the required size of the clutch or brake. Only when these important properties are known can the optimum friction combination for a given application be selected in order to give the desired behavior and service life. In order to provide understanding of the application selection of friction combinations, the following sections will describe the characteristic properties and main areas of use of our different standard friction combinations, namely steel/steel, steel/sintered lining and steel or cast iron/organic friction lining, all of which have proved themselves in use over many years.Should you have special requirements with regard to the dynamic torque, the static torque or the lubricant to be used, please contact us. For such cases, special friction materials such as plates coated with molybdenum are available.

Frictional behavior

The changes in the coefficient of friction during the course of an engagement (or disengagement) process together with the static coefficient of friction μ_0, when torque is being transmitted, depend on a number of factors:

- Combination of materials at the friction surfaces
- Design of friction surfaces, e.g. with grooves or channels
- Surface structure, e.g. sliding finish
- Friction surface pressure
- Sliding speed
- Temperature level and maximum temperature at friction surface
- Dry- or wet-running, e.g. lubrication behavior, direction of cooling oil

The characteristic frictional behavior of our standard friction combinations are represented in figures 3 to 6 (pages 1.05.00 and 1.06.00).

Dry-running clutches and brakes

The friction condition is determined by the laws friction are achieved. The coefficient of static friction μ_0 is in general larger than the coefficient of sliding friction μ. This friction combination will always be subject to wear and for this reason the service life of a clutch or brake is determined by the wear properties of the friction lining and of the counter-friction surface. Since the wear increases at a disproportionately high rate at temperatures above a particular level, design and calculations are based, to a considerable extent, on the thermal behavior.

Wet-running clutches and brakes

Controlled by the properties of the friction materials, the lubrication process in the frictional contact of wet-running clutches and brakes takes place in the area of mixed or boundary friction. The peaks and valleys (the size of these depending on the particular roughness) of the surfaces of the friction plates attempt to make contact with one another. However, they are prevented from coming into contact with one another by a few layers of oil molecules. The binding forces between the oil molecules and the friction surfaces are larger than the shearing forces resulting from the sliding movement. These binding forces are influenced in particular by the interactions between the friction surfaces and the lubricant additives, the effectiveness of which depend on temperature and pressure.

As a result the advantages of wet-running multi-plate clutches and brakes lie in the freedom from wear (following running-in) and in their significantly superior ability to dissipate the heat produced during engagement, this being the result of the cooling action of the oil (internal oiling). In particular when high frequencies of engagement/disengagement are required, a greater amount of frictional work per operation can be permitted than with dry-running. In addition applications with continuous slip are possible allowing the heat to be kept under control even when considerable quantities of heat are being generated.

A further advantage lies in the ability to influence the changes in the coefficient of friction, during each engagement operation, by selection of the material, structure and profile of the friction surfaces in combination with the type and quantity of the oil employed. Of particular technical importance is the torque achieved at the beginning and at the end of the synchronization process. Rapid engagement or the lowest possible dynamic excitation (strength, noise emissions) can be achieved with transmission lines and shaft systems.

Friction combination steel/steel

Through-hardened special plate steel with high resistance to wear is used for this well proven friction combination. This friction combination is only suitable for wet-running. Here the inner plates are given a corrugated shape: the original Ortlinghaus Sinus® plates.

The relationship between the static and dynamic coefficient of friction is:

$$\frac{\mu_0}{\mu} = 1,8 \ldots 2$$

As a result of the above, the engagement behavior of multi-plate clutches with this friction combination, in particular when they are not actuated manually, is characterized by the torque increasing at a rapid rate during the engagement process. This can subject the masses being accelerated to an undesirable jerk at the point of synchronization.

When a clutch is to be engaged dynamically, the pressure with which the friction surfaces are pressed together p_R should not exceed 0.5 N/mm^2 and the sliding speed v_R should not exceed 20 m/s. The large difference between the coefficients of static friction or stiction μ_0 (at v = 0) and the dynamic or sliding friction μ must be taken into account.

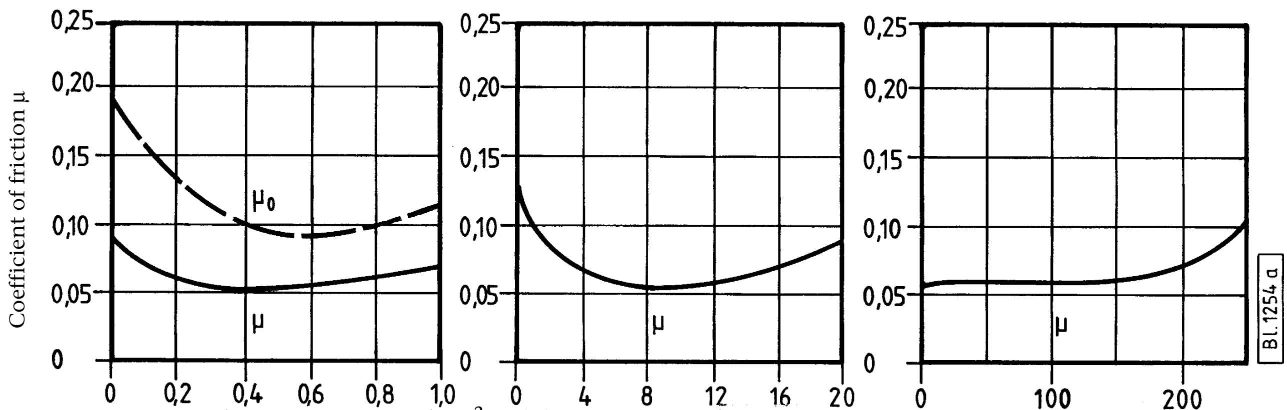

Fig. 3 Friction surface pressure p_R in N/mm^2 Sliding speed v in m/s Plate temperature in °C

Friction combination steel/sintered lining

The continuous developments in powder metallurgy have made possible friction materials suitable for special applications. Increased thermal capacity, consistency of friction coefficient, increased surface pressure and sliding speed, reduced wear can all be achieved with new sinter qualities.

WET-RUNNING

With this friction combination the coefficient of friction increases from the start of the acceleration process until the point at which the driving and driven parts are rotating at the same speed at a uniform rate depending upon the properties of the oil being used. As a result a flat and uniform acceleration curve together with a smooth start-up of the masses to be accelerated, is achieved. A high friction surface pressure and sliding speed can be selected (p_R up to 4 N/mm^2, v_R up to 40 m/s). This in turn enables the dimensions of clutches and brakes selected to be smaller.

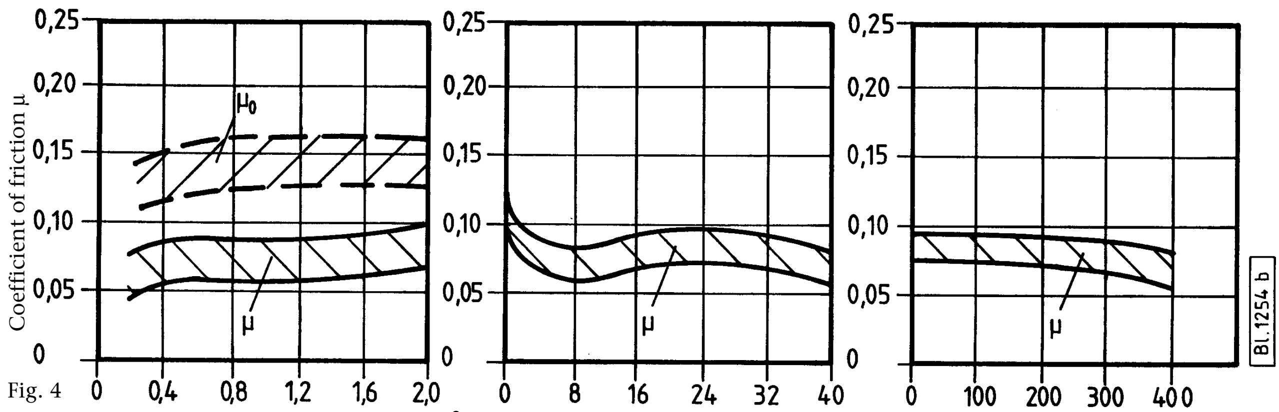

Fig. 4 Friction surface pressure p_R in N/mm^2 Sliding speed v in m/s Plate temperature in °C

DRY-RUNNING

The relationship between the coefficient of static friction and the coefficient of dynamic friction is: $\frac{\mu_0}{\mu} = 1{,}2 \ldots 1{,}3$

The surface pressure and the sliding speed selected must be smaller than with wet-running (p_R up to 2 N/mm^2, v_R up to 25 m/s).

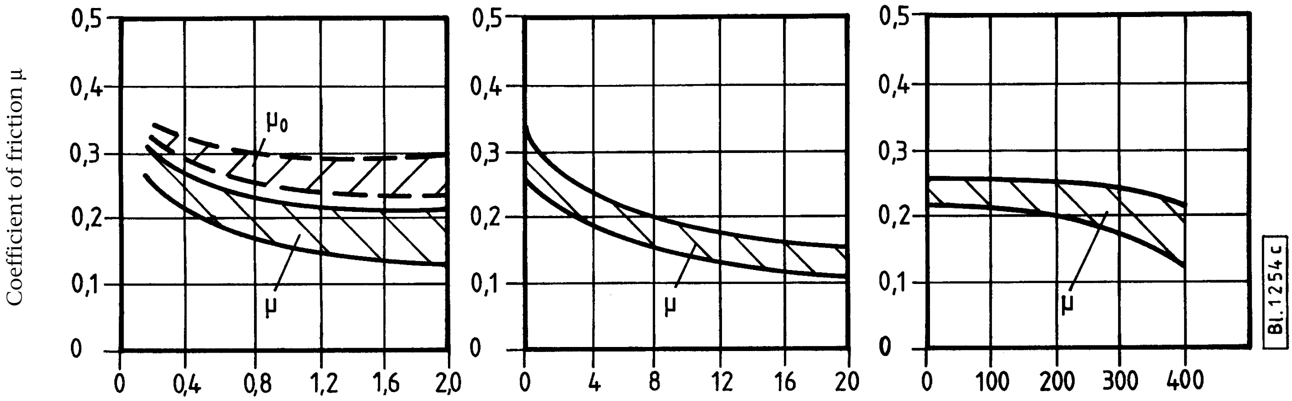

Fig. 5 Friction surface pressure p_R in N/mm^2 Sliding speed v in m/s Plate temperature in °C

Friction combination steel or cast iron/organic friction lining

With this dry-running friction combination, the friction lining is bonded or riveted as segments or as a ring onto the plate in question. The particular advantage of this friction combination lies in the high coefficient of friction and in the favorable ratio of μ_0 to μ, friction surface pressures p_R up to 1 N/mm^2 and sliding speeds v_r up to 20 m/s: $\frac{\mu_0}{\mu} = 1{,}0 \ldots 1{,}3$

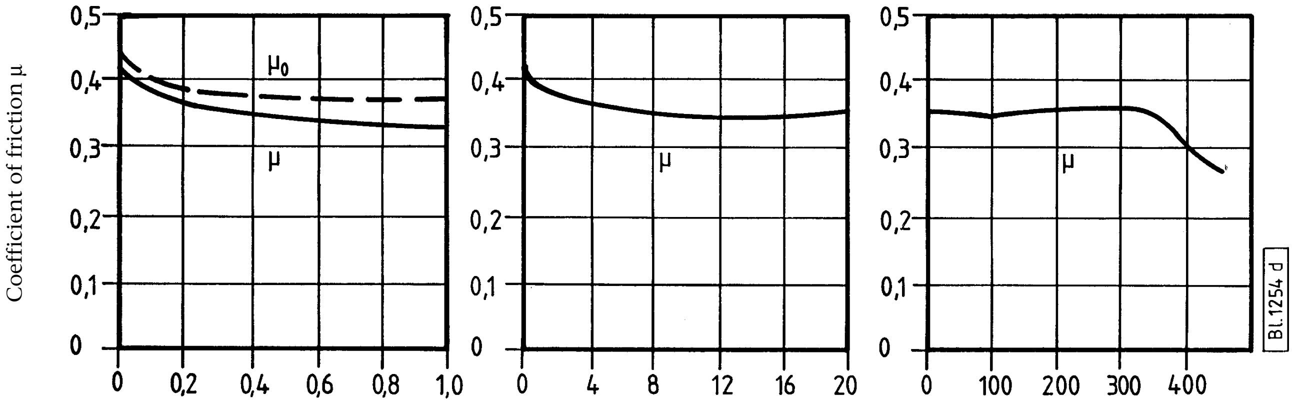

Fig. 6 Friction surface pressure p_R in N/mm^2 Sliding speed v in m/s Plate temperature in °C

Wear characteristics

The wear suffered by plates depends on the work required during engagement (or application), the friction material used and on the composition of the counterplate. It will remain low as long as the temperature resulting from the heat generated during each engagement (or application) does not exceed permissible limits.

Oil as a coolant helps to reduce wear. The oil should pass as close as possible to the friction surface, the most efficient way of doing this is by internal oiling. Special surface designs, e.g. spiral grooves, radial grooves, waffle grooves etc., provide efficient oil ways and reliable dissipation of the heat generated during engagement (or application).

Wet-running clutches and brakes work in general with practically no wear whereas in the case of dry-running units good dissipation of heat can only be achieved, and wear minimized, by constructional measures (such as design, installation and external ventilation).

Friction combination steel/steel

If the oil can keep the local temperature peaks sufficiently low, wear remains minimal, however, if the frictional heat exceeds the permissible specific thermal load (per engagement/application or per hour), wear increases considerably. Seizure and the destruction of the friction surfaces may follow, transition from dynamic friction to static friction.

Friction combination steel/sintered lining
Different qualities of sintered lining are available, these being suitable for wet-running or dry-running in accordance with their particular composition.

With wet-running it is essential that the pores of the sintered material do not become clogged by oil carbon produced when the temperature of the friction surfaces becomes too high. This will cause the coefficient of friction and heat dissipation capability to decrease. The tendency for the temperature of the friction surfaces to increase can be counteracted by using a special profile and ensuring that they receive an adequate supply of fresh oil. It is particularly important that the oil is changed regularly. When these points are observed, extremely low wear can be guaranteed.

Wear with dry-running is higher than with wet-running. The dry running properties of the sinter friction material influence the wear characteristics. Care must be taken that heat is dissipated efficiently.

Friction combination steel or cast iron/organic friction lining
With this dry-running friction combination wear remains low as long as the temperature of the counterplate is not allowed to exceed approx. 150 °C. Above this temperature the slope of the wear curve increases considerably. The critical plate temperature, at which destruction of the friction lining commences, lies at approx. 300 °C.

Thermal characteristics
The thermal loading to which a friction clutch or brake can be subjected depends on the following factors:
- Friction work per engagement/application process
- Frequency of engagement/application processes
- Intervals between successive engagement/application processes
- Duration of the clutch engagement or braking process
- Dissipation of heat at clutch or brake

In the case of a clutch, the friction work can be calculated from the masses to be accelerated and the speed difference between the driven and driving machine parts, taking into account any load torque. To keep the energy, which is converted into heat during each engagement process, as low as possible, the total mass of the machine parts to be accelerated must be kept as low as possible. In this connection the most suitable position for the installation of the clutch must be checked.

Thus, for example, in the case of a press, the clutch can be fitted on the eccentric shaft or the intermediate gear. Similar calculations and considerations apply in the case of brakes.

Great significance has to be attached to the **dissipation of heat**. For applications involving a large amount of acceleration work, pneumatically actuated single-plate clutches with large cooling surfaces and cooling ribs are frequently used. The transfer of heat from the clutch to the ambient air is considerably improved by the ventilating effect of the cooling ribs, the effect of these, however, varies with speed.
Where multi-plate clutches are fitted in gearboxes, the frictional heat generated can be dissipated by immersing the set of plates in cooling oil. Here a check must be made that the surface of the gearbox housing is sufficiently large to enable the heat to be dissipated into the ambient air at the required rate. If this should not be the case, an oil cooler will be required.

The following guideline values on coefficients of friction and the thermal limit values to be observed have been taken from **VDI guideline 2241** (see page 1.08.00).

A further **heat characteristic** of a multi-plate clutch, namely the permissible thermal load per hour $q_{permissible}$ in J/mm²/h, is based on the general assumption that the friction work occurs at roughly uniform intervals and an approximately constant level. During the engagement of a clutch, high temperature peaks occur if a high level of energy has to be suddenly transformed into heat. The permissible temperature limit value must be observed in each case.
The guideline value for the **hourly** thermal loads are stated below for a number of friction combinations.

Friction combination steel/steel
The permissible thermal load **per hour** depends to a great extent on the type and quantity of the coolant, the friction surface temperature may not be allowed to exceed 200 to 250 °C.

With splash lubrication:
$q_{perm.} =$ 13 – 17 J/mm²/h

With internal lubrication:
$q_{perm.} =$ 17 – 21 J/mm²/h

		Wet-running				Dry-running		
	Friction combinations	Sintered bronze/ steel	Sintered iron/ steel	Paper/ steel	Steel/ hardened/ steel/ hardened	Sintered bronze/ steel	Organic linings/ cast iron	Steel/ nitrided/ steel/ nitrided
Coefficient of friction	Dynamic coefficient of friction μ	0.05 to 0.1	0.07 to 0.1	0.1 to 0.12	0.05 to 0.08	0.15 to 0.3	0.3 to 0.4	0.3 to 0.4
	Static coefficient of friction μ_0	0.12 to 0.14	0.1 to 0.14	0.08 to 0.1	0.08 to 0.12	0.2 to 0.4	0.3 to 0.5	0.4 to 0.6
	Ratio μ_0 / μ	1.4 to 2	1.2 to 1.5	0.8 to 1	1.4 to 1.6	1.25 to 1.6	1.0 to 1.3	1.2 to 1.5
Technical data guideline values 1)	Max. sliding speed v_R [m/s]	40	20	30	20	25	40	25
	Max. friction surface pressure p_R [N/mm^2]	4	4	2	0.5	2	1	0.5
	q_{AE} [J/mm^2] 2)	1 to 2	0.5 to 1	0.8 to 1.5	0.3 to 0.5	1 to 1.5	2 to 4	0.5 to 1
	$\dot{q}_{Ao}$ [W/mm^2] 3)	1.5 to 2.5	0.7 to 1.2	1 to 2	0.4 to 0.8	1.5 to 2	3 to 6	1 to 2
	Area-related cooling flow $\dot{V}_A$ $\left[\frac{mm^3}{mm^2 \cdot s}\right]$	0.1 to 2	0.1 to 1	0.1 to 2	0.1 to 0.5			
Lubricant	Unalloyed and slightly blended oils	X	X	X	X			
	Oils with additives	–	X	X	X			

1) These guideline values are mutually dependent on one another to a high degree so that the permissible values may be considerably higher or, as the case may be, lower depending on the particular conditions of application.

2) Permissible area-related engagement-work at single engagement operation

3) Permissible area-related friction output (c.f. VDI 2241, page 1, section 3.2.2.)

Friction combination steel / sintered lining

The plates with a sintered lining possess good thermal conductivity and can withstand temperature peaks of up to approx. 500 to 600 °C without there being a risk of surface welding of the plate surfaces or, in the case of clutches running in an oil mist, of increased wear.

Permissible thermal loading per hour:

For dry-running: $q_{perm.} = 20$ J/mm^2/h

For wet-running (with internal oiling):

$q_{perm.} = 150 - 300$ J/mm^2/h

(information documents available on request)

Friction combination steel or cast iron/ organic friction lining

Organic friction linings are suitable for temperatures up to 300 °C. Temperature peaks higher than this can be permitted for a short time but the wear is considerably increased.

Permissible thermal load per hour:

For single-plate clutches with cast iron/organic lining:
$q_{perm.}$ = 100 J/mm^2/h

For multi-plate clutches with steel/organic lining:
$q_{perm.}$ = 15 J/mm^2/h

Use of the different friction combinations

Friction combination steel/steel

This combination is the only one suitable for electromagnetic clutches with flux-type plate packs. It is also used successfully in other kinds of clutches, particularly in applications where the frequency of engagement and the thermal loading are low and in clutches for static (holding) duty with high levels of torque to be transmitted.

Friction combination steel/sintered lining

This friction combination, which is used predominantly wet-running, is employed when high thermal loading as well as high sliding speed and high friction surface pressure are to be expected. Care must be taken that adequate cooling is provided, if possible by means of internal lubrication.

Friction combination steel or cast iron/ organic friction lining

This friction combination, which is used exclusively for dry-running, is employed in applications where the clutch or brake is mounted externally. The high coefficient of friction means that the unit is compact, however, care must be taken that the friction surfaces are kept free of lubricants.

Types of actuation

Selection of the type of actuation method for a particular application depends on

- the control media available on the machine or at the place of installation
- the required engagement (or application) characteristics
- the time required for and precision of engagement/application
- the opportunities for using a remote or programmable control.

Mechanically actuated clutches require the force for engagement/disengagement to be generated outside the clutch. The required manual force, in the region of 100 to 200 N, allows for sensitive engagement/disengagement. If the force for engagement/disengagement is generated pneumatically, hydraulically or magnetically, these clutches can also be operated in an automatic sequence.

In the case of presses, shears and other machine tools, marine propulsion systems, oil drilling equipment, heavy construction machines and also heavy rolling mill drives, compressed air is almost always available or can be easily provided. In such cases pneumatically actuated multi-plate clutches are very often used. With the aid of special valves, it is possible to obtain very short engagement/application times as required, for example, in presses. Precision control valves can be incorporated where large masses must be accelerated slowly and precisely.

In the case of non-stationary drives, e.g. in road vehicles, track-bound vehicles, ships etc., hydraulically actuated multi-plate clutches are an extremely popular solution. Change-over gear systems in vehicle engineering e.g. for Diesel locomotives, crawler tractors, lorries and construction machinery, are being designed more and more with hydraulically actuated multi-plate clutches. Often these are used in conjunction with hydraulic torque converters, in order to reduce the manual effort required by the driver and to increase performance.

In marine engineering, hydraulically actuated multi-plate clutches and hydraulically released, spring-applied brakes are a preferred solution for ships' reversing gears, loading and anchor winches. They offer the advantage of being largely maintenance and wear free. In combination with a hydraulic motor, the hydraulically released, spring-applied brake is becoming more and more popular as a safety feature.

Electromagnetically actuated clutches and brakes have the advantage that they can be controlled from a central point, permitting full automatic control of a machine. They can be used in both dry and wet-running systems. These units can be triggered in conjunction with numerical controls and timed operating cycles. High permissible number of operations per hour can be achieved with good operational accuracy. In the case of construction machinery, winches, mixers, conveyor systems etc., electromagnetic clutches can be operated from the machines' electrical supply system.

Response time and operational accuracy

Provided that a suitable type of actuation and control have been selected, multi-plate clutches can fulfill very stringent requirements in terms of response time and operational accuracy. However, in order to ensure correct operation, the particular characteristics of construction and the different friction material must be taken into account. In general dry-running clutches engage and disengage more precisely than clutches running wet.

Electromagnetic clutches with flux-type plates require, in general, more reaction time (in particular for disengagement) than clutches with solenoid-type actuation, however, an exception to this are clutches with stationary fields. As a result of the air gap in the magnet body and the support plate, the magnetic field breaks down more quickly and the effects of residual magnetism are reduced.

Hydraulically actuated clutches engage and disengage very precisely provided that suitable and correctly designed control elements are used. Oil quality, pipe dimensions and pump capacity also have a considerable influence on the clutch performance.

Dry-running, pneumatically actuated clutches and brakes can withstand very severe conditions as are required, for example, in presses. Even very large clutches can be engaged and disengaged rapidly and precisely provided that there is an adequate quantity of compressed air available and that the recommended dimensions for valves and pipes are maintained.

Selection of clutch and brake size, calculations

Before going into the details of clutch selection, a number of terms should be explained and defined.

- Torque
- Moment of intertia
- Reaction times
- Friction work and thermal load
- Load torque

The formulae and calculation procedures, which follow, are sufficient for most applications. In the case of special application, however, we recommend that the drive data be supplied to us, since it is often the case that extra calculations must be done using empirical values, the discussion of which would be too complex for this publication.

The variables and symbols used are summarized in the following table.

Variable, symbol		**Name**	**Unit**	**Designation**
Force	F	Newton	N	$1\ N = 1\ kg \cdot 1\ m/s^2$
Torque	M		Nm	
Mass	m		kg	
Moment of inertia	J		kgm^2	
Work Heat quantity	W Q	Joule	J	1 J = 1 Nm = 1 Ws
Temperature	T	Kelvin Celsius	K °C	1 K = 1 °C
Speed Angular velocity	n ω		min^{-1} rad/s (s^{-1})	$\omega = \frac{\pi \cdot n}{30}$

The different torques used in the calculations

- M_{dyn} = engagement torque (catalogue torque)
- M_a = acceleration torque (deceleration torque)
- M_L = load torque
- M_r = idling or drag torque
- M_{stat} = static or transmitted torque

Definition of engagement torque M_{dyn}

The engagement torque (dynamic torque) M_{dyn} is the effective torque acting on the shaft while the clutch or brake is slipping. M_{dyn} is the torque quoted in the catalogue for a clutch or brake and is the effective torque during acceleration or deceleration up to the point of synchronization between the driving and driven sides.

Definition of acceleration torque M_a

The acceleration torque accelerates the given masses from speed n_1 to n_2 within a given time.

$$M_a = M_{dyn} - M_L \quad \text{in Nm}$$

$$M_a = \frac{J \cdot (\omega_2 - \omega_1)}{t} \quad \text{in Nm}$$

$$M_a = \frac{J \cdot (n_2 - n_1)}{9.56 \cdot t} \quad \text{in Nm}$$

J = moment of inertia in kgm^2
t = acceleration time in s
$n_1\ (\omega_1)$ = speed before acceleration in min^{-1} (s^{-1})
$n_2\ (\omega_2)$ = speed after acceleration in min^{-1} (s^{-1})

Definition of load torque M_L

The load torque is the torque which acts on the output side of the clutch as the result of the load. It is calculated in essence from the force acting directly on the load side and the associated lever arm (Fig. 7).

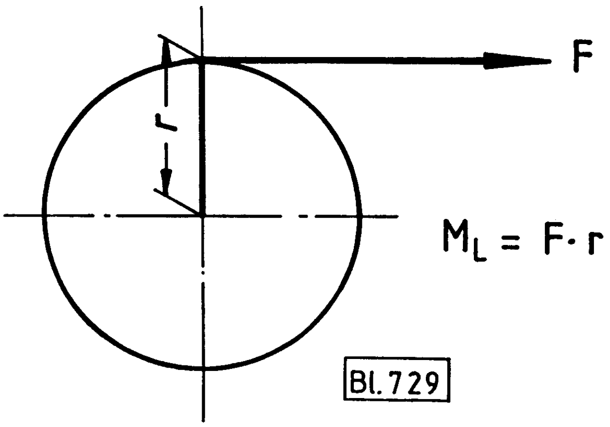

Fig.7

Definition of residual torque M_r

The residual or idling torque is the torque which is still transmitted by the fully disengaged clutch, the value stated being the maximum steady-state value at normal operating temperature.

Definition of static torque M_{stat}

The transmitted (or static) torque M_{stat} is the torque the clutch, when engaged, or the brake, when applied, can be loaded without slip taking place.

Ratio between static and dynamic torque

When determining the size of clutch required, the difference between static and dynamic torques must be considered.

The ratios of dynamic to static torque for the friction combinations below are as follows:

Steel/steel	1.8 to 2
Steel/organic friction lining	1 to 1.3
Steel/sintered lining	1.3 to 1.5

Dynamic moment of inertia

The moment of inertia is defined as the sum of all products resulting from the particles of mass dm and the square of their distances r from the rotational axis.

$$J = \int r^2 \cdot dm$$

The moment of inertia of a rotating body can be described as

$$J = i^2 \cdot m \text{ in kgm}^2$$

If one considers the total mass of the body to be at a distance i (radius of gyration) from the rotational axis.

Referred moments of inertia, in existing gear trains, to the clutch shaft

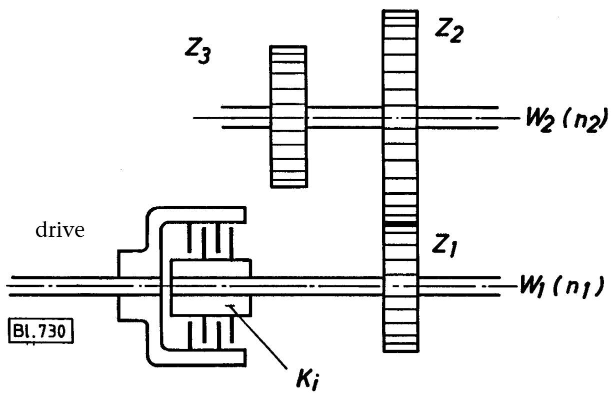

Fig. 8

In the two-shaft system, shown in Fig. 8, the clutch on shaft W_1 has to accelerate the following masses, the moments of inertia of which have to be individually calculated. The sum of all the moments of inertia on W_1 and W_2 are

$$J_1 = J_{Ki} + J_{W1} + J_{Z1} \quad \text{or, respectively}$$

$$J_2 = J_{W2} + J_{Z2} + J_{Z3}$$

The moment of inertia J_2 is reduced to clutch the shaft W_1 by multiplying the inertia by the square of the speed ratio.

$$J_{2\ red\ W1} = J_2 \cdot \left(\frac{n_2}{n_1}\right)^2$$

The total J to be accelerated by the clutch on W_1 is obtained by:

$$J_{totW1} = J_1 + J_{2\ red\ W1} \quad \text{in kgm}^2$$

Converting the effect of a mass moving in a straight line to the moment of inertia J of the clutch shaft

For this, the following formulae apply:

$J = m \cdot \dfrac{v^2}{\omega^2}$ in kgm^2 $\qquad$ $J = 91 \cdot m \cdot \dfrac{v^2}{n^2}$ in kgm^2

m = mass in kg of the body moving in a straight line
v = velocity in m/s of the mass moving in a straight line
n, (ω) = rotational speed of the clutch shaft in min^{-1}, (s^{-1})

Moments of inertia

Moments of inertia of solid steel cylindrical bodies (ρ = 7850 kg/m^3) for a cylindrical height h = 10 mm.

For other heights h, multiply the values in the table by $\dfrac{h}{10}$

D	Moment of intertia **J = $kgcm^2$**					1 $kgcm^2$ = 0,0001 kgm^2				
mm	0	1	2	3	4	5	6	7	8	9
30	0,062	0,071	0,081	0,091	0,103	0,116	0,129	0,144	0,161	0,178
40	0,197	0,218	0,24	0,263	0,289	0,316	0,345	0,376	0,409	0,444
50	0,482	0,521	0,563	0,608	0,655	0,705	0,758	0,814	0,872	0,934
60	0,999	1,067	1,139	1,214	1,293	1,376	1,462	1,553	1,648	1,747
70	1,85	1,958	2,071	2,189	2,311	2,438	2,571	2,709	2,853	3,002
80	3,157	3,317	3,484	3,657	3,837	4,023	4,216	4,415	4,622	4,835
90	5,056	5,285	5,521	5,765	6,017	6,277	6,546	6,823	7,108	7,403
100	7,707	8,02	8,342	8,674	9,016	9,368	9,73	10,102	10,485	10,879
110	11,283	11,699	12,127	12,566	13,016	13,479	13,954	14,442	14,942	15,455
120	15,981	16,52	17,073	17,64	18,22	18,815	19,425	20,049	20,688	21,342
130	22,011	22,696	23,397	24,114	24,848	25,598	26,365	27,149	27,95	28,769
140	29,606	30,461	31,335	32,227	33,137	34,068	35,017	35,986	36,976	37,985
150	39,015	40,066	41,138	42,231	43,346	44,483	45,642	46,824	48,028	49,256
160	50,507	51,781	53,08	54,403	55,75	57,122	58,52	59,943	61,391	62,866
170	64,367	65,895	67,45	69,033	70,643	72,281	73,947	75,642	77,366	79,119
180	80,902	82,715	84,558	86,432	88,337	90,273	92,24	94,24	96,272	98,337
190	100,43	102,56	104,73	106,93	109,16	111,43	113,73	116,07	118,44	120,86
200	123,3	125,79	128,31	130,87	133,47	136,1	138,78	141,49	144,25	147,04
210	149,88	152,75	155,67	158,63	161,63	164,67	167,75	170,88	174,05	177,27
220	180,53	183,83	187,18	190,58	194,02	197,51	201,05	204,63	208,26	211,93
230	215,66	219,44	223,26	227,14	231,06	235,04	239,06	243,14	247,27	251,45
240	255,69	259,97	264,32	268,71	273,16	277,67	282,23	286,85	291,52	296,25
250	301,04	305,88	310,79	315,75	320,77	325,85	331,0	336,2	341,46	346,79
250	301,04	305,88	310,79	315,75	320,77	325,85	331,0	336,2	341,46	346,79
260	352,17	357,62	363,14	368,71	374,35	380,06	385,83	391,66	397,56	403,53
270	409,56	415,66	421,83	428,07	434,38	440,75	447,2	453,72	460,3	466,96
280	473,69	480,5	487,37	494,32	501,35	508,45	515,62	522,87	530,2	537,6
290	545,08	552,63	560,27	567,98	575,78	583,65	591,61	599,64	607,76	615,96
300	624,24	632,6	641,05	649,59	658,2	666,91	675,7	684,57	693,54	702,59
310	711,73	720,96	730,27	739,68	749,18	758,77	768,45	778,22	788,09	798,05
320	808,1	818,25	828,5	838,84	849,27	859,81	870,44	881,17	892,0	902,92
330	913,95	925,08	936,31	947,64	959,08	970,61	982,26	994,0	1005,8	1017,8
340	1029,8	1042,0	1054,3	1066,7	1079,2	1091,8	1104,5	1117,3	1130,2	1143,3
350	1156,4	1169,7	1183,1	1196,6	1210,2	1224,0	1237,8	1251,8	1265,9	1280,1
360	1294,4	1308,8	1323,4	1338,1	1352,9	1367,8	1382,9	1398,0	1413,3	1428,8
370	1444,3	1460,0	1475,8	1491,7	1507,8	1524,0	1540,3	1556,8	1573,3	1590,1
380	1606,9	1623,9	1641,0	1658,3	1675,6	1693,2	1710,8	1728,6	1746,6	1764,6
390	1782,9	1801,2	1819,7	1838,3	1857,1	1876,1	1895,1	1914,3	1933,7	1953,2
400	1972,9	1992,7	2012,6	2032,7	2053,0	2073,4	2093,9	2114,6	2135,5	2156,5
410	2177,7	2199,0	2220,5	2242,1	2263,9	2285,9	2308,0	2330,3	2352,7	2375,3
420	2398,0	2421,0	2444,1	2467,3	2490,7	2514,3	2538,0	2562,0	2586,0	2610,3
430	2634,7	2659,3	2684,1	2709,0	2734,1	2759,4	2784,9	2810,5	2836,3	2862,3
440	2888,5	2914,9	2941,4	2968,1	2995,0	3022,1	3049,3	3076,8	3104,4	3132,2
450	3160,2	3188,4	3216,7	3245,3	3274,1	3303,0	3332,1	3361,5	3391,0	3420,7
460	3450,6	3480,7	3511,0	3541,5	3572,2	3603,1	3634,2	3665,5	3697,0	3728,7
470	3760,6	3792,7	3825,0	3857,5	3890,2	3923,2	3956,3	3989,7	4023,2	4057,0
480	4091,0	4125,2	4159,6	4194,2	4229,1	4264,1	4299,4	4334,9	4370,6	4406,6
490	4442,7	4479,1	4515,7	4552,5	4589,6	4626,9	4664,4	4702,1	4740,0	4778,2
500	4816,7	4855,3	4894,2	4933,3	4972,6	5012,2	5052,0	5092,1	5132,4	5172,9
510	5213,7	5254,7	5296,0	5337,5	5379,2	5421,2	5463,4	5505,9	5548,6	5591,6
520	5634,8	5678,3	5722,0	5766,0	5810,2	5854,7	5899,4	5944,4	5989,7	6035,2
530	6080,9	6126,9	6173,2	6219,8	6266,6	6313,7	6361,0	6408,6	6456,5	6504,6
540	6553,0	6601,7	6650,6	6699,9	6749,4	6799,1	6849,2	6899,5	6950,1	7000,9
550	7052,1	7103,5	7155,2	7207,2	7259,5	7312,0	7364,9	7418,0	7471,4	7525,1
560	7579,1	7633,4	7688,0	7742,8	7798,0	7853,5	7909,2	7965,2	8021,6	8078,2
570	8135,2	8192,4	8249,9	8307,8	8365,9	8424,4	8483,1	8542,2	8601,6	8661,3
580	8721,3	8781,6	8842,2	8903,1	8964,3	9025,9	9087,8	9150,0	9212,5	9275,3
590	9338,5	9401,9	9465,7	9529,9	9594,3	9659,1	9724,2	9789,6	9855,4	9921,4
600	9987,9	10054	10121	10189	10256	10325	10393	10463	10531	10600
610	10670	10740	10811	10882	10953	11024	11096	11168	11241	11314
620	11387	11461	11535	11609	11684	11759	11834	11910	11986	12063
630	12140	12217	12295	12373	12451	12530	12609	12688	12768	12849
640	12929	13010	13092	13173	13256	13338	13421	13504	13588	13672
650	13756	13841	13927	14012	14098	14185	14272	14359	14446	14534
660	14623	14712	14801	14890	14981	15071	15162	15253	15345	15437
670	15529	15622	15716	15809	15904	15998	16093	16189	16285	16381
680	16478	16575	16672	16770	16869	16968	17067	17167	17267	17367
690	17468	17570	17672	17774	17877	17980	18084	18188	18293	18398
700	18503	18609	18716	18823	18930	19038	19146	19255	19364	19473
710	19584	19694	19805	19917	20029	20141	20254	20367	20481	20596
720	20710	20826	20942	21058	21175	21292	21409	21528	21646	21766
730	21885	22005	22126	22247	22369	22491	22614	22737	22860	22985
740	23109	23234	23360	23486	23613	23740	23868	23996	24125	24254
750	24384	24514	24645	24777	24908	25041	25174	25307	25441	25576
760	25711	25846	25983	26119	26256	26394	26532	26671	26811	26951
770	27091	27232	27374	27516	27658	27802	27945	28090	28235	28380
780	28526	28673	28820	28967	29116	29264	29414	29564	29714	29866
790	30017	30170	30322	30476	30630	30784	30940	31095	31252	31409
800	31566	31724	31883	32042	32202	32363	32524	32686	32848	33011
810	33174	33339	33503	33669	33835	34001	34168	34336	34505	34674
820	34843	35014	35184	35356	35528	35701	35874	36048	36223	36398
830	36574	36751	36928	37106	37284	37464	37643	37824	38005	38187
840	38369	38552	38736	38920	39105	39291	39477	39664	39852	40040
840	38369	38552	38736	38920	39105	39291	39477	39664	39852	40040
850	40229	40419	40609	40800	40992	41184	41377	41571	41765	41960
860	42156	42352	42549	42747	42946	43145	43345	43545	43747	43948
870	44151	44354	44559	44763	44969	45175	45382	45589	45798	46007
880	46216	46427	46638	46850	47062	47276	47490	47705	47920	48136
890	48353	48571	48789	49008	49228	49449	49670	49892	50115	50339
900	50563	50788	51014	51241	51468	51696	51925	52155	52385	52616
910	52848	53081	53314	53549	53784	54019	54256	54493	54731	54970
920	55210	55450	55692	55934	56176	56420	56664	56909	57155	57402
930	57650	57898	58147	58397	58648	58900	59152	59405	59659	59914
940	60170	60426	60683	60941	61200	61460	61721	61982	62244	62507
950	62771	63036	63301	63568	63835	64103	64372	64642	64912	65184
960	65456	65729	66003	66278	66554	66831	67108	67386	67666	67946
970	68227	68508	68791	69075	69359	69644	69930	70217	70505	70794
980	71084	71374	71666	71958	72252	72546	72841	73137	73434	73731
990	74030	74330	74630	74931	75234	75537	75841	76146	76452	76759
1000	77067									

Reaction times for friction clutches closed by actuation

See Fig. 9 (frictional engagement is produced by the application of the actuation force).

Response delay t_{11} is the period between the starting of the actuation and the time at which the torque starts to rise (inherent clutch time).

Rise time t_{12} is the period from the time the tor que starts to rise until the nominal dynamic torque M_{dyn} has been reached.

Engagement time t_1 is the sum of the response delay and rise times $t_1 = t_{11} + t_{12}$.

Slipping time t_3 is the period during which the friction faces of a clutch move relative to one another under the contact pressure.

Reaction times for friction clutches opened by actuation

See Fig. 9 (frictional engagement is interrupted by the application of the actuation force; frictional engagement is produced by, for example, spring pressure)

Response delay t_{21} is the period between the discontinuation of the actuation force and the time at which the torque starts to fall in relation to M_{stat}

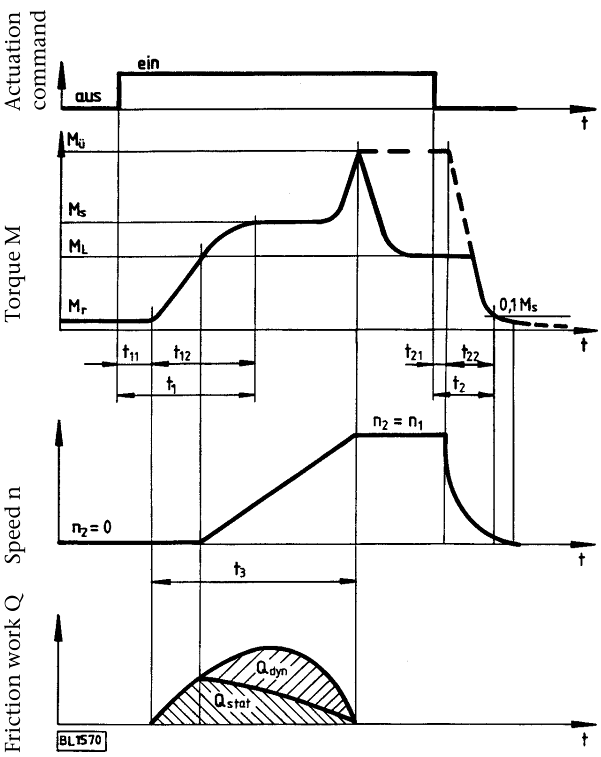

Fig. 9 aus = off, ein = on

Fall time t_{22} is the period from the time the torque starts to fall until it has fallen to 10 % of the engagement torque M_{dyn}

Disengagement time t_2 is the sum of the response delay and the fall time $t_2 = t_{21} + t_{22}$.

Friction work and thermal load

Type of loading

During the engagement of a clutch, friction work is carried out which generates heat. This heat must be absorbed by the friction surfaces or dissipated without the rated thermal capacity of the clutch or friction combination being exceeded. A calcula-tion, to confirm this, is essential for most cases of application.

The total amount of heat Q_s produced by a clutch engagement operation is the result of the load torque and the acceleration (or deceleration) torque applied for the slip time i.e. Q_{stat} and Q_{dyn} (Fig. 9).

Influence of the load torque on the thermal load

Since the load torque M_L acts continuously, the acceleration torque available ($M_a = M_{dyn} - M_L$) must be large enough to enable acceleration to be carried out within a reasonable time to avoid excessive thermal load. As illustrated in Fig. 10, a ratio of M_{dyn}/M_L of less than 2 will cause a very rapid increase in the thermal load Q_s since Q_{dyn} is constant for a given clutch.

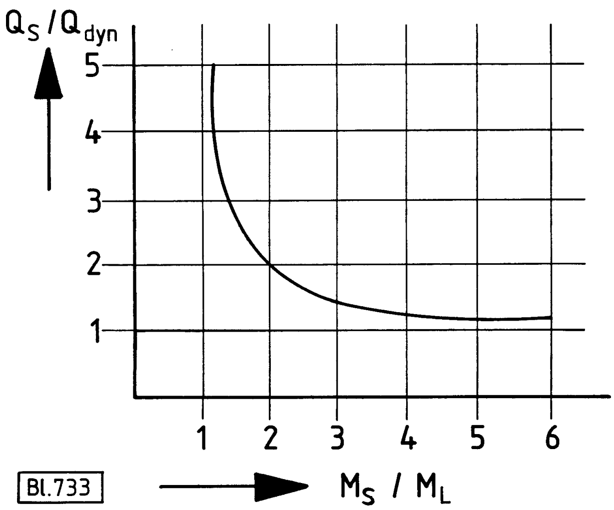

Fig. 10 Mü = Mstat Ms = Mdyn

Calculation of the thermal load

The heat generated by individual or repeated clutch engagement operations (or brake applications) can be calculated with the aid of the following formula:

$$Q_S = \frac{J \cdot (\omega_2 \pm \omega_1)^2}{2} \cdot \frac{M_{dyn}}{M_{dyn} \pm M_L} \quad \text{in J per engagement or application}$$

$$Q_S = \frac{J \cdot (n_2 \pm n_1)^2}{182{,}4 \cdot 10^3} \cdot \frac{M_{dyn}}{M_{dyn} \pm M_L} \quad \text{in kJ per engagement or application}$$

and

$$Q_h = Q_S \cdot S_h \quad \text{in J/h}$$

J = moment of inertia of all parts to be accelerated or decelerated in kgm^2

n_1, (ω_1) = speed of the output shaft before the acceleration process or after the deceleration process in min^{-1} (s^{-1})

n_2, (ω_2) = speed of the output shaft after the acceleration process or before the deceleration process in min^{-1} (s^{-1})

S_h = number of engagements/applications per hour

$n_2 + n_1$ speed difference between the internal and external clutch plates

$\frac{M_{dyn}}{M_{dyn} - M_L}$ = load factor if the effect of M_{dyn} is diminished by M_L

$\frac{M_{dyn}}{M_{dyn} + M_L}$ = load factor if the effect of M_{dyn} is enhanced by M_L

In the case of pure acceleration of masses, from stationary, the energy which is absorbed as heat by the clutch is the same as the energy transmitted into the masses.

If the change of speed is carried out in stages (e.g. in power-shift gearboxes), the thermal loading on each clutch is reduced with the number of stages. The most severe thermal loading occurs when the total acceleration or braking process is carried out by just one single clutch.

Thermal characteristic values

A clutch or brake can only absorb/dissipate a particular amount of heat, which is generated by the friction work, without overheating or excessive wear taking place. The permissible amount of heat and hence the permissible amount of the friction work varies with the friction material and the heat transfer characteristics of the clutch or brake. The limiting situation is determined by the maximum amount of heat that can be absorbed/ dissipated either per clutch **engagement operation** (or brake application) or **per hour** depending on the particular application.

The characteristic values q_{Aperm} in J/mm^2, which relate to specific friction pairs, are available on request. Typical, permissible values for q are given in the section "Thermal behavior".

q_A or q_{AE} in J/mm^2 is the work per unit area for one engagement/application operation.

q_{Ao} in W/mm^2 is the friction work per unit area which occurs at the start of the engagement/ application process, i.e. at the highest relative speeds.

q_{Ah} in $J/mm^2/h$ is the friction work per unit area and **hour** in repeated engagement/application operations carried out at approximately uniform intervals **of time**.

Selection of the correct clutch size

Clutch size is determined by two factors:

- Max. torque to be transmitted
- Max. engagement work

Calculation of required torque capacity

The nominal torque of the prime mover can be calculated with the following formulas:

$$M = \frac{P}{\omega} \quad \text{in Nm}$$

P = nominal power rating of prime mover in W

ω = angular velocity in s^{-1}

or

$$M = \frac{9550 \cdot P}{n} \quad \text{in Nm}$$

P = nominal power rating of prime mover in kW

n = speed of prime mover in min^{-1}

In addition to establishing the nominal torque to be transmitted, it is necessary to consider the torsional characteristics of the prime mover and the driven machine. Internal combustion engines, reciprocating pumps and reciprocating compressors rotate with a high degree of non-uniformity and consequently larger clutches should be selected. It is usually difficult to establish the peak transient torque, therefore, it is common practice to apply a safety factor K selected from the table below.

Minimum safety factors

Prime mover / **Type of application**	Electric motors steam and gas turbines multi-cylinder combustion engines	2-cylinder combustion engines	Single cylinder combustion engines
	Safety factor K		
Generators, chain conveyors, centrifugal conpressors, sand blasting blowers, textile machines, conveyor systems, fans and centrifugal pumps	1.5	2	2.5
Elevators, bucket conveyors, rotary kilns, wire winders, crane travel and trolley drives, winches, agitators, shears, machine tools, washing machines, looms, brick extruders	2	2.5	3
Excavators, drilling rigs, briquetting presses, mine ventilators, rubber rolling mills, hoisting drives, pug mills, reciprocating pumps, tumblers, joggers, combination mills	2.5	3	3.5
Reciprocating compressors, frame saws, wet-presses, paper mangles, roller conveyors, drying rolls, roller mills, cement mills, centrifuges	3	3.5	4

Required Torque

$M_{nec.} = K \cdot M$ in Nm

At start-up, or if subjected to overload, squirrel cage motors will develop two to three times their nominal torque for brief periods. In order to prevent excessive slip in such cases, the torque capacity of the clutch selected should be relatively higher than the nominal torque of the motor. As a rule, clutch selection should be based on the engagement torque M_{dyn} which is always lower than the transmitted torque M_{stat}. Note that if there is a load torque, it should never be more than 30 - 50 % of the engagement torque in order to allow the driven parts to be accelerated effectively. For details on this see also Fig. 10.

Slipping time
If the available acceleration torque $M_a = M_{dyn} - M_L$ is known, the acceleration time or slipping time t_3 can be calculated:

$$t_3 = \frac{J \cdot (\omega_2 - \omega_1)}{M_{dyn} - M_L} \text{ in s or } t_3 = \frac{J \cdot (n_2 - n_1)}{9{,}56\,(M_{dyn} - M_L)} \text{ in s}$$

J = moment of inertia in kgm^2
M_{dyn} = engagement torque in Nm
M_L = load torque in Nm

Note that the engagement times t_1 for the particular clutch type must be added to give the total time (Fig. 9).

Calculations for clutches and brakes for crank drives

In applications such as presses and guillotines where kinetic energy is stored in a flywheel, the required clutch torque must be calculated from the required torque on the driven side. If the braking time is of critical importance, the required braking torque is determined from the permissible braking angle.

Important: Load torques of the driven machine must be taken into account, together with the masses, during calculations.

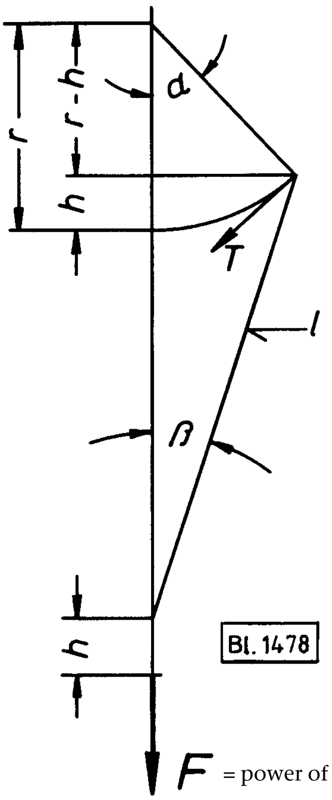

Fig. 11 F = power of press

Transmittable torque

$$M_{stat\text{-}crank} = r \cdot \frac{F \cdot \sin(\alpha + \beta)}{\cos \beta} \quad \text{in Nm}$$

The following diagram (Fig. 12) shows the values for sin a, provided that the crank radius r and the throw of the press h are known, using the following formula.

$$\sin \alpha = \sqrt{1 - \left(\frac{r - h}{r}\right)^2}$$

$$\sin \beta = \frac{r \cdot \sin \alpha}{l}$$

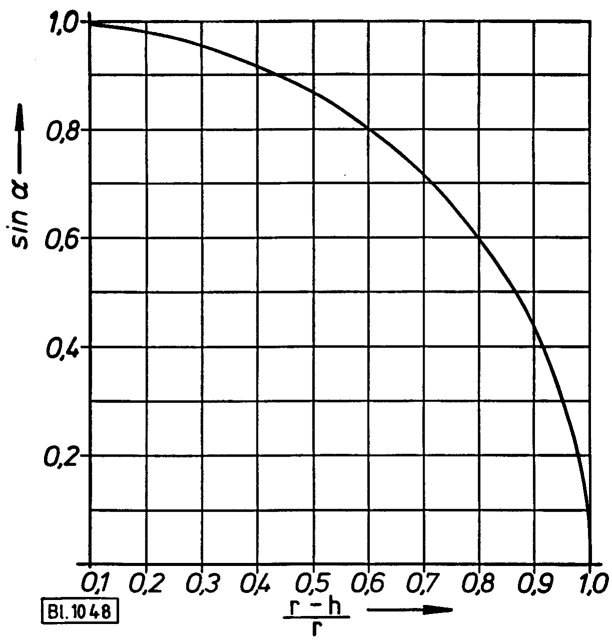

Fig. 12

Braking process

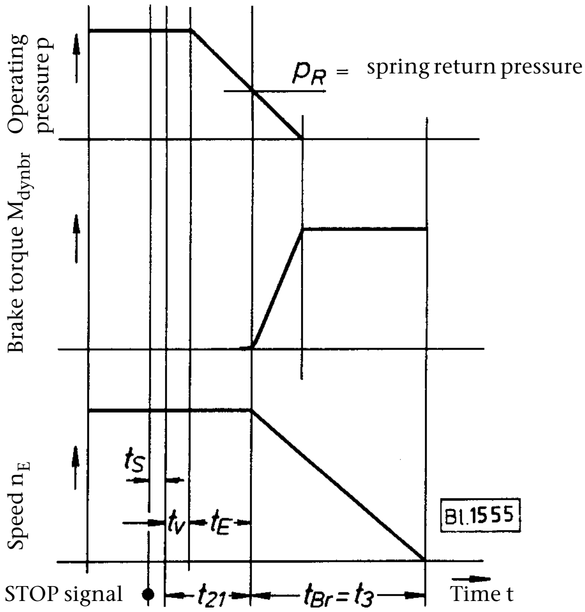

Fig. 13

Significance of the symbols in the formulas:

F	= power of press in N
r	= crank radius in m
l	= length of connecting rod in m
h	= throw of the press in m
t_s	= electrical time element of the contactor in seconds
t_v	= electrical reaction time of the contactor in seconds
t_E	= discharge time for the cylinder in seconds
t_{21}	= $t_v + t_E$ = disengagement delay in seconds
t_{Br}	= t_3 = mechanical braking time in seconds
$n_E(\omega_E)$	= speed of the eccentric shaft in min^{-1} (s^{-1})
$n_K(\omega_K)$	= speed of the clutch shaft in min^{-1} (s^{-1})
α	= crank angle, working angle before bottom dead centre in degrees or radians
β	= connecting rod angle before bottom dead centre in degrees or radians
γ	= braking angle in degrees or radians
M_{statK}	= static clutch torque in Nm
M_{dynBr}	= braking torque in Nm
k	= correction factor which takes into account the non-linearity of the braking torque
k	~ 1.2 – 1.3
ΣJ	= moment of inertia in kgm^2 of all moving parts + clutch and brake

Fig. 13 shows the actuation pressure, torque and speed from the moment the brake engagement signal is given until the masses to be braked are at a standstill. The mechanical braking time t_{Br} or t_3 can be calculated as follows:

$$t_3 = k \cdot \frac{\Sigma J \cdot \omega_K}{M_{dyn}} \text{ in s} \quad \text{or} \quad t_3 = k \cdot \frac{\Sigma J \cdot n_K}{9{,}56 \cdot M_{dyn}} \text{ in s}$$

The braking angle is calculated using the following formulas:

$$\gamma = \omega_E \cdot (t_s + t_{21}) + \frac{\omega_E}{2} \cdot t_3 \quad \text{in radians or}$$

$$\gamma = 6 \cdot n_E \, (t_s + t_{21}) + 3 \cdot n_E \cdot t_3 \quad \text{in degrees}$$

Required thermal capacity

The heat generated by the engagement process must be dissipated by the clutch without the critical temperature being exceeded. The thermal load per engagement and per hour can be calculated as described on page 1.14.00. Clutch ratings (thermal characteristic values) for the different clutch types are available on request and our technical staff will be pleased to assist where further information is required.

Single engagement

Heat transfer during the engagement cycle is negligible and the total amount of heat generated must be absorbed by the components directly involved in the friction process. Consequently, the permissible thermal load depends mainly on the type of friction lining and on the lubrication arrangement. The basic characteristics of the different friction lining materials are given on page 1.07.00.

Repeated engagements

If engagements are repeated over an extended period of time at approximately equal time intervals, the heat generated will be conducted to the outer surfaces of the clutch and dissipated by ventilation or cooling oil. After a certain running time a steady-state temperature will establish itself in the clutch or brake.
Adequate cooling is very important and, if necessary, forced ventilation or internal lubrication should be employed.

Slipping at constant speed

Under certain conditions, e.g. with safety clutches, the clutch will slip for a particular time with full torque and constant speed. The amount of heat generated can be calculated by the following formula:

$$Q = M_{dyn} \cdot \omega \cdot t \quad \text{in J or} \quad Q = \frac{M_{dyn} \cdot n \cdot t}{9548} \quad \text{in kJ}$$

M_{dyn}	=	engagement torque in Nm
ω	=	angular velocity in s^{-1}
n	=	speed differential in min^{-1}
t	=	slipping time in s

Attention should be paid to the fact that the permissible slipping time is relatively short for most applications.

Lubrication and cooling of clutches and brakes

Oil-lubricated multi-plate clutches are normally installed in gearbox housings. Cooling oil can thus be supplied in the form of splash oil, or by means of immersion (up to $^1/_{10}$ of the diameter) or as internal oil through the shaft.

For applications with high thermal loading or high idling speeds, internal lubrication, the most intensive form of cooling, is recommended. The required cooling flow in relation to the frictional surface is as follows:

$$0.5 \text{ to } 2 \text{ mm}^3/(\text{mm}^2 \cdot \text{s})$$

In this way dry friction and increased idling heat are prevented. Furthermore thermal load absorption is increased through the uniform heat distribution and improved cooling. The oil flow rate should be adapted to the particular operating conditions.

A compromise must normally be made on the question of oil type when a multi-plate clutch runs in a gearbox. Lubrication oils, as used for highly stressed gear wheels and bearings, are not always suitable for multi-plate clutches.

In general oils to be used with multi-plate clutches should fulfill the following requirements:

- high heat and aging resistance
- neutral behavior with regard to copper and steel
  at operating temperatures
- no oil carbon deposits
- good thermal conductivity and cooling effect
- low foaming, in particular with hydraulically actuated multi-plate clutches
- viscosity
  (see recommendation chart on page 1.19.00)

Surface design of plates

The interaction of various plate surface patterns such as, for example, spiral grooves, radial slots and waffle patterns, with the special properties of the various oils, makes it possible to solve most application problems. Torque build-up and thus clutch engagement time and thermal capacity can be modified by the appropriate choice of plate surface pattern, cooling oil type and cooling oil flow rate.

Bearing lubrication for dry-running multi-plate clutches and brakes

In the case of housings with roller bearings, seals must be fitted to prevent the bearing grease getting on to the friction surfaces.
It is recommended that the bearings are packed with grease on assembly and that the facility to carry out re-greasing is dispensed with.

Oil recommendations for multi-plate clutches and brakes

	Application			
	Mechanically and hydraulically actuated multi-plate clutches of average speed $v^{1)} \sim 5$ to 12 m/s		Electromagnetically actuated multi-plate clutches, mechanically and hydraulically actuated multi-plate clutches with higher speeds $v^{1)} > 12$ m/s	
	In Germany	**Abroad**	**In Germany**	**Abroad**
ARAL	Kosmol TL 68 64 mm^2/s	Kosmol TL 68 64 mm^2/s	Kosmol TL 46 44 mm^2/s	Oel CMS 22 mm^2/s
BP	Energol HL 46 46 mm^2/s	Energol THB 46 46 mm^2/s	Energol HL 32 32 mm^2/s	Energol THB 32 32 mm^2/s
CASTROL	HYSPIN VG 46 46 mm^2/s	PERFECTO T 46 46 mm^2/s	HYSPIN VG 32 32 mm^2/s	PERFECTO T 32 32 mm^2/s
CHEVRON	GST Oil 46 46 mm^2/s	GST Oil 46 46 mm^2/s	GST Oil 32 32 mm^2/s	GST Oil 32 32 mm^2/s
DEA	Astron HL 46 46 mm^2/s	Eterna LTD 46 46 mm^2/s	Astron HL 32 32 mm^2/s	Eterna LTD 32 32 mm^2/s
ELF	POLYTELIS 46 46 mm^2/s	POLYTELIS 46 46 mm^2/s	POLYTELIS 32 31 mm^2/s	POLYTELIS 32 31 mm^2/s
ESSO	TERESSO 68 (previously 52) 64 mm^2/s	ESSTIC 68 (previously 50) 64 mm^2/s	TERESSO 32 (previously 43) 30 mm^2/s	ESSTIC 32 (previously 42) 34 mm^2/s
FINA	CIRKAN 68 ISO 62 mm^2/s	BAKOLA 68 64 mm^2/s	CIRKAN 38 F 39 mm^2/s	CIRKAN 38 F 42 mm^2/s
FUCHS	RENOLIN MR 15 49,6 mm^2/s	RENOLIN MR 15 49,6 mm^2/s	RENOLIN MR 10 34 mm^2/s	RENOLIN MR 10 34 mm^2/s
MOBIL OIL	D.T.E. Oil Medium 43,4 mm^2/s	D.T.E. Oil Medium 43,4 mm^2/s	D.T.E. Oil Light 29,6 mm^2/s	D.T.E. Oil Light 29,6 mm^2/s
OIL BY SHELL	Morlina Oil 46 46 mm^2/s	Morlina Oil 68 68 mm^2/s	Morlina Oil 46 46 mm^2/s	Morlina Oil 46 46mm^2/s
TEXACO	Rando Oil C 65 mm^2/s	Regal Oil R&O 68 63 mm^2/s	Rando Oil B 43 mm^2/s	Regal Oil R&O 46 42 mm^2/s

1) v = peripheral speed at the outer diameter of the clutch or brake

Viscosity at 40° C

1 $mm^2/s \sim 1$ cSt.

The above data is not standard and should be checked on a case-to-case basis.
The listing of the oils should not be interpreted as an evaluation.
Oil types of other manufacturers on request.

Installation instructions and tolerances

General installation instructions for Ortlinghaus clutches and brakes

In addition to the operation and installation descriptions given in the individual sections of this catalogue, attention should be paid to a few general rules which apply to the designing of drive systems incorporating multi-plate clutches.

The basic design of the clutches means that the two clutch halves must be aligned precisely and that the bearings must be arranged appropriately. In splitshaft applications, the bearings must be situated as close to the clutch as possible. If a centering bearing is necessary, it must be given adequate lubrication, particularly when idling.

In order to avoid additional heating and/ orde-struction of the clutch or brake, the components of the clutch and brake and the shafts must be located axially in such a way that the distances between the different parts are as specified in the drawings. Shouldered shafts used in conjunction with locking rings or shaft nuts are the best means of location. Should grub screws be used, they must be secured.

The maximum rotational speed of the clutch is determined both by its size and by the way in which it is incorporated into the overall design. In most cases the velocity at the mean effective radius of the friction area of the plates should not exceed 20 m/s. Higher speeds may be permissible under particular conditions.

Please note that all types of clutch are subject to wear. Regular inspections, adjustments and renewal of friction linings will be necessary with many types of clutch in order to ensure high operational efficiency and a long service life. To this end it is advisable that sufficient access for inspection and/or removal of the clutch is provided.

Our experienced engineering team is always willing to advise and assist in selecting the most suitable Ortlinghaus clutch or brake for your application.

Recommended tolerances and basic types of drive housings for Ortlinghaus clutches and brakes

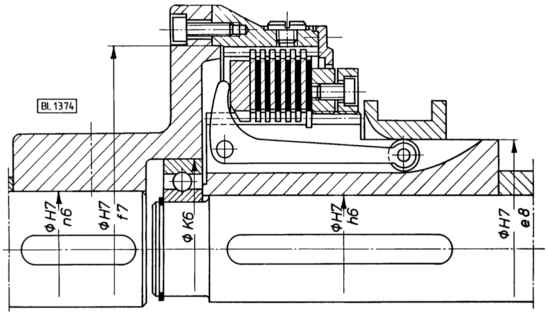

Fig. 14:
Mechanically actuated clutch with shoulder housing with flange hub; centering ball bearing

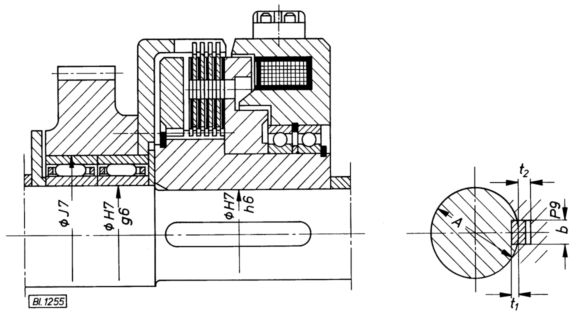

Fig. 15:
Stationary field electromagnetic clutch with cup housing; needle bearing

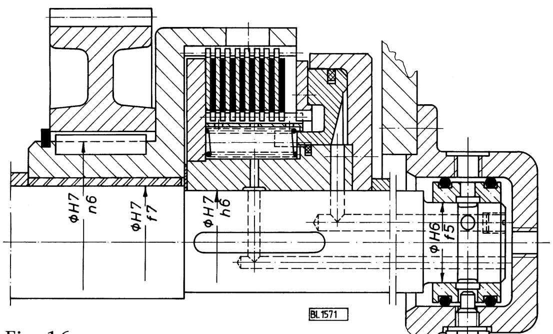

Fig. 16:
Hydraulically actuated clutch with hub housing; plain bearing

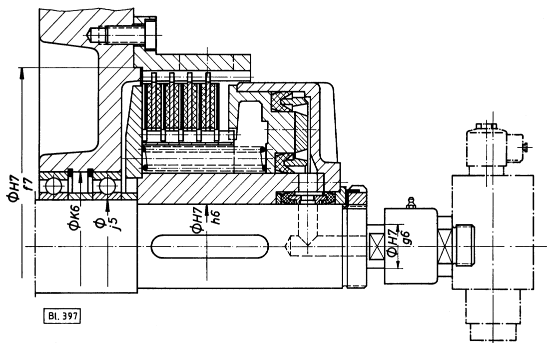

Fig. 17:
Pneumatically actuated clutch with flange housing; rolling bearing

Recommended tolerances, bores and keyways

When designing or installing Ortlinghaus clutches and brakes please select the bores and keyways listed below, this is to simplify storage and shorten delivery times. Account should be taken of our preferred sizes of bores as given in the clutch and brake dimension lists.

Standard tolerance H7 for bores to DIN 7154.

Hub keyway tolerance P9 for keys and keyways to DIN 6885 (tolerance JS9 on request).

Key	Shaft diameter A	Shaft keway b x t_1	Hub keway b x t_2
2 x 2	over 6 to 8	2 x 1.2 + 0.1	2 x 1.0 + 0.1
3 x 3	over 8 to 10	3 x 1.8 + 0.1	3 x 1.4 + 0.1
4 x 4	over 10 to 12	4 x 2.5 + 0.1	4 x 1.8 + 0.1
5 x 3	over 12 to 17	5 x 1.9 + 0.1	5 x 1.2 + 0.1
5 x 5	over 12 to 17	5 x 3.0 + 0.1	5 x 2.3 + 0.1
6 x 4	over 17 to 22	6 x 2.5 + 0.1	6 x 1.6 + 0.1
6 x 6	over 17 to 22	6 x 3.5 + 0.1	6 x 2.8 + 0.1
8 x 5	over 22 to 30	8 x 3.1 + 0.2	8 x 2.0 + 0.1
8 x 7	over 22 to 30	8 x 4.0 + 0.2	8 x 3.3 + 0.2
10 x 6	over 30 to 38	10 x 3.7 + 0.2	10 x 2.4 + 0.1
10 x 8	over 30 to 38	10 x 5.0 + 0.2	10 x 3.3 + 0.2
12 x 6	over 38 to 44	12 x 3.9 + 0.2	12 x 2.2 + 0.1
12 x 8	over 38 to 44	12 x 5.0 + 0.2	12 x 3.3 + 0.2
14 x 6	over 44 to 50	14 x 4.0 + 0.2	14 x 2.1 + 0.1
14 x 9	over 44 to 50	14 x 5.5 + 0.2	14 x 3.8 + 0.2
16 x 7	over 50 to 58	16 x 4.7 + 0.2	16 x 2.4 + 0.1
16 x 10	over 50 to 58	16 x 6.0 + 0.2	16 x 4.3 + 0.2
18 x 7	over 58 to 65	18 x 4.8 + 0.2	18 x 2.3 + 0.1
18 x 11	over 58 to 65	18 x 7.0 + 0.2	18 x 4.4 + 0.2
20 x 8	over 65 to 75	20 x 5.4 + 0.2	20 x 2.7 + 0.1
20 x 12	over 65 to 75	20 x 7.5 + 0.2	20 x 4.9 + 0.2
22 x 9	over 75 to 85	22 x 6.0 + 0.2	22 x 3.1 + 0.2
22 x 14	over 75 to 85	22 x 9.0 + 0.2	22 x 5.4 + 0.2
25 x 9	over 85 to 95	25 x 6.2 + 0.2	25 x 2.9 + 0.2
25 x 14	over 85 to 95	25 x 9.0 + 0.2	25 x 5.4 + 0.2
28 x 10	over 95 to 110	28 x 6.9 + 0.2	28 x 3.2 + 0.2
28 x 16	over 95 to 110	28 x 10.0 + 0.2	28 x 6.4 + 0.2
32 x 11	over 110 to 130	32 x 7.6 + 0.2	32 x 3.5 + 0.2
32 x 18	over 110 to 130	32 x 11.0 + 0.2	32 x 7.4 + 0.2
36 x 12	over 130 to 150	36 x 8.3 + 0.2	36 x 3.8 + 0.2
36 x 20	over 130 to 150	36 x 12.0 + 0.3	36 x 8.4 + 0.3
40 x 22	over 150 to 170	40 x 13.0 + 0.3	40 x 9.4 + 0.3
45 x 25	over 170 to 200	45 x 15.0 + 0.3	45 x 10.4 + 0.3
50 x 28	over 200 to 230	50 x 17.0 + 0.3	50 x 11.4 + 0.3