Energy Consideration

It has been noted that the most common brakes employ friction to transform the braked system's mechanical energy, irreversibly into heat which is then transferred to the surrounding environment -

• Kinetic energy is absorbed during slippage of either a clutch or brake, and this energy appears as heat.

• If the heat generated is faster than it is dissipated, then the temperature rises.

Thorough design of a brake therefore requires a detailed transient thermal analysis of the interplay between heat generated by friction, heat transferred through the lining and the surrounding metalwork to the environment, and the instantaneous temperature of the surface of the drum as well as the lining. For a given size of brake there is a limit to the mechanical power that can be transformed into heat and dissipated without lthe temperatures reaching damaging levels. Temperature of the lining is more critical and the brake size is characterized by lining contact area, A.

The capacity of a clutch or brake is therefore limited by two factors:

1. *The characteristics of the material and,*
2. *The ability of the brake to dissipate heat.*

Heat Generated In Braking

During deceleration, the system is subjected to an essentially constant torque T exerted by the brake, and in the usual situation this constancy implies constant deceleration too.

Application of the work or energy principle to the system enables the torque exerted by the brake and the work done by the brake, U, to be calculated from:-

$U = \Delta E = T\,\Delta\theta$ (2)

Where ΔE is the loss of system total energy which is absorbed by the brake during deceleration, transformed into heat, and eventually dissipated.

The elementary equations of constant rotational deceleration apply, thus when the brake drum is brought to rest from an initial speed ω_o :-

Deceleration = $\omega_o^2/ 2$ (1) $\Delta\theta = \omega_m.\Delta t$; $\omega_m = \omega_o/2 = \Delta\theta/\Delta t$ where ω_m is the mean drum speed over the deceleration period.

The mean rate of power transformation by the brake over the braking period is :-

$P_m = U / \Delta t = T\,\omega_m$ (3) which forms a basis for the selection or the design of the necessary brake dimensions.

The rise in temperature in the lining material is also important as rate of wear is also a function of the temperature. Further for any lining material, the maximum allowable temperature is also another performance criteria.

Temperature Rise

The temperature rise of the brake assembly can be approximated by the classic expression,,

$$\Delta T = \frac{E}{C.m}$$

Where is temperature ΔT is rise in temperature in ^oC,'C' is the specific heat of the brake drum material – (500J/Kg for steel or Cast Iron) and m is the mass (kg) of the brake parts dissipating the heat into the surroundings.

Though the equation appears to be simple, there are so many variables involved that it would be most unlikely that such an analysis would even approximate experimental results.

On the other hand the temperature-rise equations can be used to explain what happens when a clutch or brake is operated frequently. For this reason such analysis are most useful, for repetitive cycling, in pin pointing those design parameters that have the greatest effect on performance.

An object heated to a temperature T_1 cools to an ambient temperature T_a according to the exponential relation

Time-temperature relation

$$T_i - T_a = (T_1 - T_a)e^{-(AU/WC)t}$$

Where T1 = instantaneous temperature at time t, ˚C

A= heat transfer area, m^2

U= Heat Transfer coefficient, $W/(m^2.s.\ ^\circ C)$

T1 = Initial temperature, ˚C

Ta = Ambient temperature, ˚C

C - Specific heat

t - time of operation, s

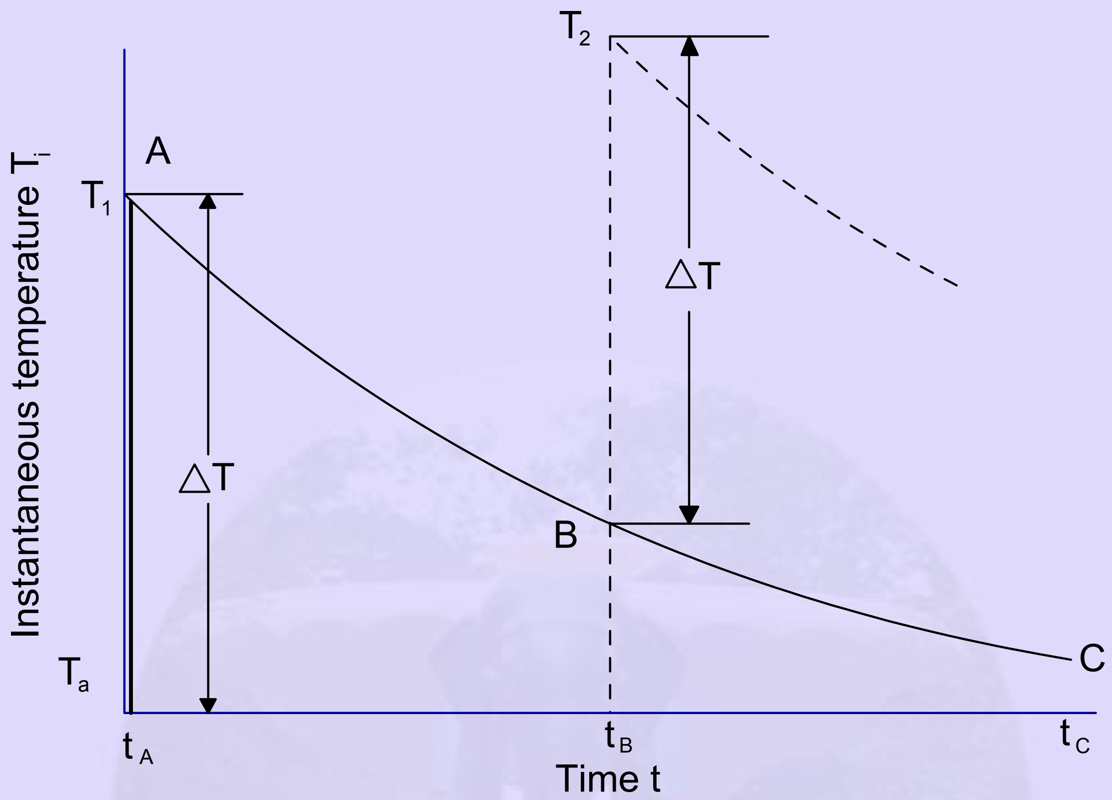

Figure 3.1.16

Figure shows an application of Eq. (a). At time t_A a clutching or braking operation causes the temperature to rise to T_1 at A. Though the rise occurs in a finite time interval, it is assumed to occur instantaneously. The temperature then drops along the decay line ABC unless interrupted by another braking operation. If a second operation occurs at time t_B, the temperature will rise along the dashed line to T_2 and then begin an exponential drop as before. About 5 -10 % of the heat generated at the sliding interface of a friction brake must be transferred through the lining to the surrounding environment without allowing the lining to reach excessive temperatures, since high temperatures lead to hot spots and distortion, to fade (the fall-off in friction coefficient) or, worse, to degradation and charring of the lining which often incorporates organic constituents

In order to determine the brake dimensions the energy need to be absorbed during critical braking conditions is to be estimated.

Energy to be Absorbed

If t is the time of brake application and ω_m the mean or average angular velocity then the energy to be absorbed in braking E

$E = T.\ \omega_m .t = Ek + Ep + Ei$

where E_k is the kinetic energy of the rotating system

E_p is the potential energy of the moving system

E_i is the inertial energy of the system

Energy to be absorbed

$$E_k = \frac{1}{2}\frac{\omega}{g}$$
$$= \frac{1}{2}mv^2$$
$$= \frac{1}{2}\frac{\omega}{g}\left(v_2^2 - v_1^2\right)$$

$$E_p = mgh = \omega h$$
$$E_i = \frac{1}{2}I\omega^2$$
$$= \frac{1}{2}I\left(\omega_2^2 - \omega_1^2\right)$$

Frictional Material

A brake or clutch friction material should have the following characteristics to a degree, which is dependent upon the severity of the service.

- A high and uniform coefficient of friction.
- Imperviousness to environmental conditions, such as moisture.
- The ability to withstand high temperatures together with good thermal conductivity.
- Good resiliency.
- High resistance to wear, scoring, and galling.

Linings

The choice of lining material for a given application is based upon criteria such as the expected coefficient of friction; fade resistance, wear resistance, ease of attachment, rigidity or formability, cost, abrasive tendencies on drum, etc. The lining is sacrificial - it is worn away. The necessary thickness of the lining is therefore dictated by the volume of material lost - this in turn is the product of the total energy dissipated by the lining throughout its life, and the specific wear rate R_w (volume sacrificed per unit energy dissipated) which is a material property and strongly temperature dependent. The characteristics of a typical moulded asbestos lining material is illustrated in the figure below. The coefficient of friction, which may be taken as 0.39 for design purposes, is not much affected by pressure or by velocity - which should not exceed 18 m/s. The maximum allowable temperature is 400°C. However at this temperature the wear is very high. From a lower wear or higher life point, the maximum temperature should not exceed about 200 ^oC

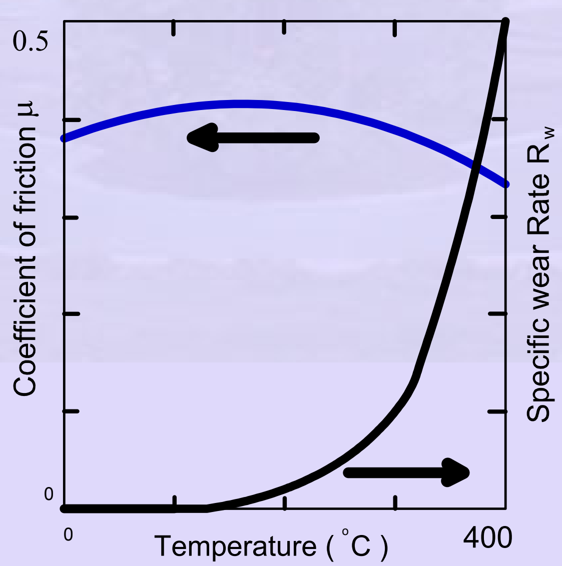

Figure 3.1.17

Linings traditionally were made from asbestos fibers bound in an organic matrix, however the health risks posed by asbestos have led to the decline of its use. Non-asbestos linings generally consist of three components - metal fibers for strength, modifiers to improve heat conduction, and a phenolic matrix to bind everything together.

Brake Design Section

The braked system is first examined to find out the required brake capacity that is the torque and average power developed over the braking period. - The brake is then either selected from a commercially available range or designed from scratch ff a drum brake has to be designed for a particular system (rather than chosen from an available range) then the salient brake dimensions may be estimated from the necessary lining area, A, together with a drum diameter- to-lining width ratio somewhere between 3:1 and 10:1, and an angular extent of 100 °C say for each of the two shoes.

Worked out Example 1

An improved lining material is being tried on an existing passenger car drum brake shown in Figure. Quality tests on the material indicated permissible pressure of 1.0 MPa and friction co-efficient of 0.32. Determine what maximum actuating force can be applied for a lining width of 40 mm and the corresponding braking torque that could be developed. While cruising on level road at 100 kmph, if it is to decelerated at 0.5g and brought to rest, how much energy is absorbed and what is the expected stopping distance?

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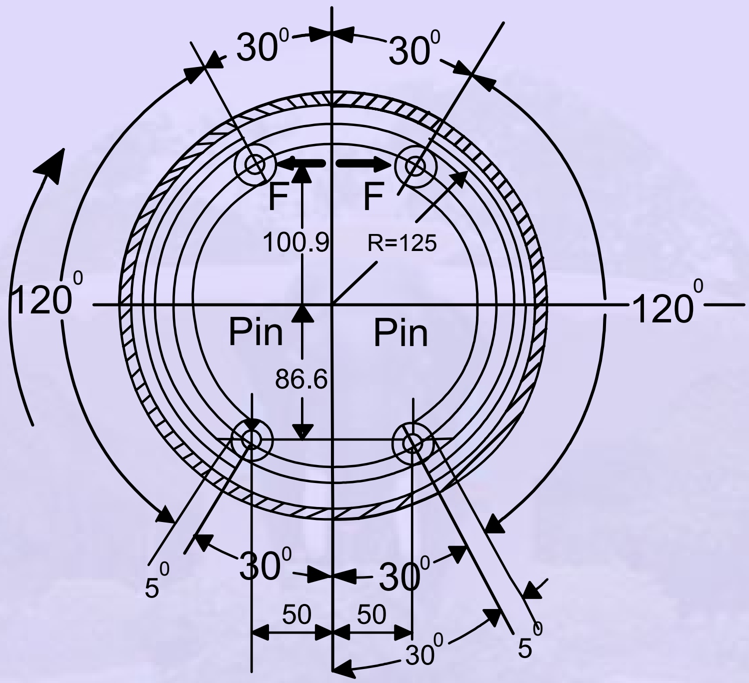


AUTOMOTIVE DOUBLE SHOE BRAKE

Figure 3.1.18

Analysis based on leading shoe

$P_a = 1$ MPa $f = 0.32$ $a = 187.5$ mm

$b = 40$ mm $\theta_{max} = 90^0$ $d = 100^2 + 86.1^2$

$= 99.99 \approx 100$mm

$\theta_1 = 5^0$ $\theta_2 = 120^0$ $r = 125$ mm

$$M_n = \frac{p_a brd}{\sin\theta_a}\left[\frac{\theta_2 - \theta_1}{2} - \frac{1}{4}(\sin 2\theta_2 - \sin 2\theta_1)\right]$$

$$= \frac{10^6 * 40 * 10^{-3} * 125 * 10^{-3} * 0.1}{1}\left[\frac{115}{2} * \frac{\pi}{180} - \frac{1}{4}\left(\sin 240° - \sin 10°\right)\right]$$

$$= 40*125*0.1\left[1.003 - \frac{1}{4}\left(-1.03\right)\right]$$

$$M_n = 631.459 \text{ N.m}$$

$$M_f = \frac{f.b.rpm}{\sin\theta_a}\left[r(\cos\theta_1 - \cos\theta_2) - \frac{d}{2}\left(\sin^2\theta_2 - \sin^2\theta_1\right)\right]$$

$$= 0.32 * 40 * 10^{-3} * 125 * 10^{-3} * 10^6\left[0.125(\cos 5° - \cos 120°) - 0.04(\sin^2 120° - \sin^2 5°)\right]$$

$$M_f = 224.85 \text{ N - m}$$

F*a= $M_N - M_f$

$$F = \frac{M_n - M_f}{a} = \frac{631.459 - 224.85}{0.187} = \underbrace{2174.3N}_{\text{Max. actutating force}}$$

$$T = fbr \frac{p}{\sin\alpha}\left(\cos\theta - \cos\theta\right))\left[1 + \frac{F}{M + M}\right]$$

$$T = 0.32 * 40 * 10 * (0.125) * 10 \,(\text{Cos}5 - \text{Cos}120)\left[1 + \frac{406.609}{856.36}\right]$$

$$T = 441.329 \text{ N-m}$$

Running at 100 kmph =100*5/8

= 27.7 m/s

U= 27.7 m/s

Deceleration =0.5*9.8=4.9

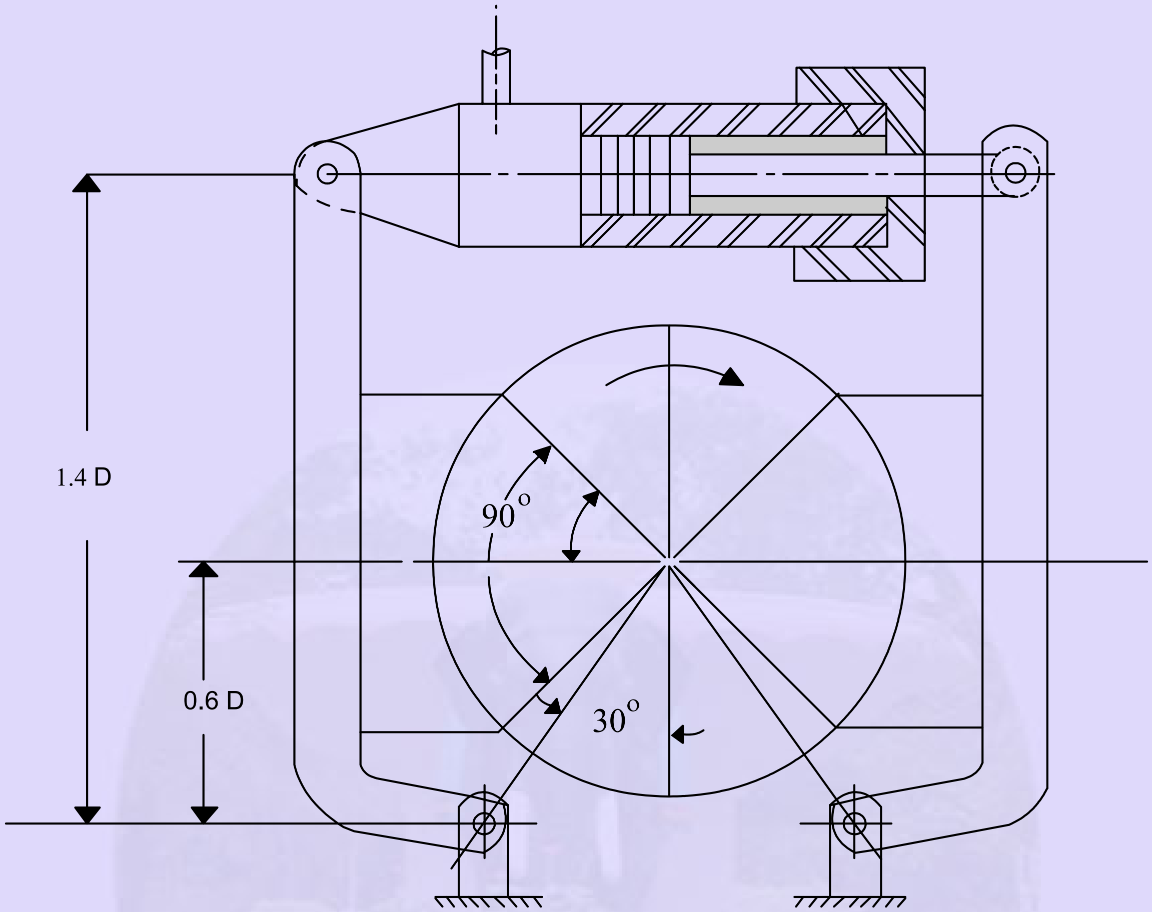
$$V^2 - U^2 = 2aS$$
$$0 - (27.7)^2 = 2*(-4.9)*S$$
$$S = \frac{27.7^2}{2*4.9} = 78.29 \text{ m}$$

$$E = T.\omega_{av}.t$$
$$= 441.329\left(\frac{1}{2}\right)\left(\frac{27.7}{0.125}\right).\left(\frac{78.24}{27.7}\right)$$
$$= 138206$$
$$= 138.2\text{KJ}$$

Worked out example 2

A spring set, hydraulically released double shoe drum brake, schematically shown at Fig 2 is to be designed to have a torque capacity of 600 N.m under almost continuous duty when the brake drum is rotating at 400 rpm in either direction. Assume that the brake lining is to be molded asbestos having a friction coefficient of 0.3 and permissible pressure of 0.8 MPa. The width of the brake shoe is to be third of drum diameter and the remaining proportion's are as shown in figure.

$$a = 1.4D$$
$$b = \frac{1}{3}D$$
$$d = \frac{0.6D}{\cos 30^0} = 0.693D$$



Double block brake

Figure 3.1.19

Determine the required brake drum diameter, width of the lining and the spring force required to be set.

$$a = 1.4D$$

$$b = \frac{1}{3}D$$

$$d = \frac{0.6D}{\cos 30^0} = 0.693D$$

$$= 0.8*10^6*\frac{1}{3}D*\frac{D}{2}*0.6028D.\left[\frac{\pi}{4}-\frac{\sin(210°)-\sin 30°}{4}\right]$$

$$M_N = 92373.3D^3*1.035$$

$$M_J = \frac{P_a brf}{1}\int_{\theta_1}^{\theta_1}(r - a\cos\theta)\sin\theta d\theta$$

$$= P_a brf\left[r(\cos\theta_1 - \cos\theta_2) - \frac{d}{2}(\sin^2\theta_2 - \sin^2\theta_1)\right]$$

$$= 0.8*10^6*\frac{D}{3}*\frac{D}{2}*0.3\left[\frac{D}{2}(1.225) - \frac{0.6928}{2}(0.866)\right]$$

$$= 12500D^3$$

P_a 'for the trailing shoe

$$P_a' = P_a\left(\frac{M_N - M_F}{M_N + M_F}\right)$$

$$= P_a\left(\frac{95606.36 - 12500}{95606.36 + 12500}\right)$$

$$= 0.7687P_a$$

Torque due to trailing and leading shoe=

$$\tau = \int fdN.r$$

$$= \int f.r\frac{P_a\sin\theta}{\sin\theta_a}.rd\theta.b$$

$$= \frac{fbr^2}{\sin\theta_a}\left[P_a\int_{\theta_1}^{\theta_2}\sin\theta d\theta + P_a'\int_{\theta_1}^{\theta_2}\sin\theta d\theta\right]$$

$$= \frac{fbr^2}{\sin\theta_a}.(P_a + P_a').(\cos\theta_1 - \cos\theta_2)$$

$$= \frac{0.3*\frac{D}{3}*\frac{D^2}{4}}{1}\left[1.7687*0.8*10^6\right](\cos 15° - \cos 1.5°)$$

$$= 43324\ D^3$$

$$T_{total} = 600\ Nm$$

Therefore $\boxed{D = 240.14mm}$

Actuating force due to spring

$$F = \frac{M_N - M_F}{1.4 * D} = \frac{83106 * 0.240^2}{1.4} = 3423.22N$$

Actuating force =3423.22 N

$$\text{Living width} = \frac{D}{3}$$
$$= 80.04 \text{ mm}$$