**Duty cycle analysis and thermal simulation for a lightweight disc brake for a regenerative braking system**

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**Abstract**

One of the stated advantages of electric vehicles (EVs), and hybrid vehicles (HVs) is their ability to recuperate braking energy. Regenerative braking (RB) would extend the working range of an EV or HV provided that any extra energy consumption e.g. from increased vehicle mass and system losses did not outweigh the saving from energy recuperation, also reduce duty levels on the brakes themselves, giving advantages including extended brake rotor and friction material life, but more importantly reduced brake mass, minimise brake pad wear. The objective of this paper is to define how much braking energy could be absorbed by a regenerative braking system (RBS) on a passenger car, hence defining the duty envelope of the friction brake. This will enable lighter brakes to be designed and fitted with confidence in a normal passenger car alongside a hybrid electric drive. In this paper, a mathematical analysis (MATLAB) is used to analyse the availability of regenerative braking energy during a single stop braking event. Secondly a computer simulation model based on Advanced Vehicle Simulator (ADVISOR) is used to simulate both single stop and drive cycle braking. Based on both sets of results it is shown how much of the total braking energy could be absorbed by the RBS of an example hybrid car in single stop braking and drive cycle braking. Thermal performance is a key factor which is studied using FEA simulations. Ultimately a design method for lightweight brakes suitable for use on any car-sized hybrid vehicle will be developed. Some results from an experimental lightweight brake disc are shown to illustrate the effects of RBS / friction combination in term of weight reduction.

*Key words:* Hybrid vehicle; regenerative braking; simulation; lightweight; brake disc; design; thermal; temperature.

**Notation**

*A* area of friction interface contact surface on one face of a brake disc (m2)

*Cp* specific heat (J/kg K)

*d* fractional power loss due to vehicle drag

*di* inner diameter of disc (m)

*do* outer diameter of disc (m)

*Di* fatigue damage theory (damage ratio at the *ith* stress level)

*E* Young’s modulus (GPa)

*f1, f2* adhesion utilization at front (1) and rear (2) axle

*FB* brake force (N)

*Fbd* braking force demand (N)

*Freg* regenerative brake force (N)

*g* acceleration due to gravity (m/s2)

*i* transmission gear ratio

*J* deceleration (m/s2)

*k* thermal conductivity (W/m K)

*M* mass of vehicle (kg)

*m* mass of disc (kg)

*ni* number of cycles at the *ith* stress level

*nfi* number of cycles to failure corresponding to the *ith* stress level

*N* motor speed (rev/min)

*PGemax* maximum generation power (kW)

*p* pressure (Pa)

*pf* inlet pressure (Pa)

*pt* threshold pressure (Pa)

*Re* disc effective radius (m)

*Rr* tyre rolling radius (m)

*S* distance (m)

*t* time (s)

*Tb* braking torque (Nm)

*Tb ac* actual front wheel brake torque (Nm)

*Tf ac* friction brake torque (Nm)

*Treg* motor torque (Nm)

*TEMreg*  electric motor regenerative braking torque (Nm)

*vi* final speed (m/s)

*vo* initial speed (m/s)

*X1, X2* proportion of total braking at front, rear axle

*z* rate of braking = J/g

*η* efficiency

*ρ* density (kg/m3)

*µ* dynamic (sliding) friction coefficient between the brake pad and the brake disc

υ Poisson’s ratio

**1. Introduction**

Vehicle brakes are designed to provide adequate deceleration of the vehicle (defined by legislation, manufacturer’s standards, and customer expectations) under all conditions which might be experienced by a driver using the vehicle. When a vehicle is fitted with regenerative braking, kinetic energy that was previously 100% dissipated through the friction brakes (FB) is now partly absorbed by the regenerative braking system (RBS) and partly dissipated by the friction braking, which results in lower duty on the friction brakes. This offers the opportunity to specify smaller and lighter brake system components, e.g. disc, pads, caliper, and actuating system.

However there is a question of what happens if for some reason, the RBS is not able to carry any braking duty (e.g. no remaining energy storage capacity-battery, or system failure) and the vehicle is required to meet expected performance standards on the friction braking alone. In the event of RBS failure, the legislation (No. 13H)(Regulation, 2008) states that all electric or hybrid vehicles shall be capable of providing indication of brake failure and a warning signal must be provided to the driver when this occurs. In this case the friction brakes must be able to decelerate the car safely whatever its speed and load on any up or down specification of gradient in the allowable stopping distance equivalent to 55% g (5.4 m/s2).

A ‘downsized’ friction brake system may still be able to provide expected performance standards of braking in the absence of regenerative braking, but would not be able to do so for any extended period of usage. So, if a lightweight braking system were to be fitted to a hybrid vehicle with regenerative braking, its failure would not compromise vehicle safety but would limit the operational life of the friction brakes, meaning that (a) the RBS should be repaired as soon as possible, and (b) the friction brake system should always be maintained or replaced after regenerative braking failure.

The question then is; “how far can the downsizing of the friction braking go?” There is a need to identify and justify what size and weight saving can be made in a vehicle’s friction braking system while maintaining a safe level of performance (within a specified operational envelope) in the event of regenerative braking failure. An example of how this might be handled in practice could be to actively limit the vehicle driving speed in the event of regenerative brake system failure by the engine management control system, in a similar way that “limp-home” mode is invoked where the On Board Diagnostics (OBD) has identified an engine fault which could affect emissions.

The purpose of this paper is to present an analysis of how weight (and hence cost) reduction in a friction braking system might be reliably achieved to deliver a specified level of braking duty capability for safe operation in a vehicle with regenerative braking. Braking energy flows in hybrid vehicles have been simulated to investigate the relationship between the available braking energy from the front wheels of the car and the total braking energy in a typical urban driving cycle. Results from other studies have showed that 50% - 60% of braking energy can be recovered by regenerative braking in urban driving [1]. Regenerative braking can recover about 45% of total kinetic energy for a city bus [2].

Weng-yong [3] designed a system to distribute braking into regenerative braking torque and mechanical friction torque. This was intended to give maximum use of kinetic energy recovery, and could be applied to vehicles with Emulated Engine Compression Braking (EECB). The algorithm was based on regenerative torque optimization to maximise the actual regenerative power, reduce the thermal load and increase the life-span of the front brake discs.

Peng [4] designed a combined braking control strategy based on a new method of HV braking torque distribution in which the hydraulic braking system worked together with the regenerative braking system to meet the requirements of vehicle longitudinal braking performance, and to maximise regenerated energy for a parallel HV. Hydraulic braking torque could be adjusted by a logic threshold strategy, and a fuzzy logic control strategy was used to adjust the regenerative braking torque. The proposed braking control strategy was demonstrated by simulation using a low adhesion coefficient road (below 0.3) for emergency braking.

**2. Regenerative Braking**

For the work presented here, a test car has been used to investigate duty levels and braking performance. The vehicle data are summarised in Table 1; it has two front wheels with disc brakes and two rear wheels with drum brakes. Legislation requires that the car can decelerate to rest at a minimum of 6.43 m/s2 or 0.66 g from speeds up to 100 km/h (vehicles of category M1 - cars) although manufacturers’ own specifications often far exceed this. For a Type 0 test on this particular car, with the engine disconnected (as defined in Regulation 13H, Annex 4), the total kinetic energy to be dissipated by each front brake is 231 kJ. The axle brake torque for each front brake is 1154 Nm providing a brake force (*FB*) of 3845 N. The vehicle stops from 100 km/h in 4.3 seconds and develops an initial braking power of 107 kW.

**Table 1** Test car technical data

|  |  |
| --- | --- |
| Front brakes | Ventilated disc |
| Rear brakes | Drum |
| Gross Vehicle Mass, *M* (kg) | 1495 |
| Disc surface outer diameter, *Do* (m) | 0.258 |
| Disc surface inner diameter, *Di* (m) | 0.146 |
| Tyre rolling radius, *Rr* (m) | 0.3 |

*2.1. Regenerative force distribution (RFD)*

The braking force of a hybrid vehicle is provided by friction brakes and the regenerative braking system. During the braking phase, the RB causes the wheels to apply torque to the motor / generator which absorbs power and slows down the vehicle. The friction braking will be activated when higher deceleration is required to provide additional stopping power. LaPlante [5] found that a HV can generate between 14% and 48% of extra braking power by using regenerative braking in the Federal Urban Driving Schedule (FUNDS), and 53% of extra braking power in Japan’s 10-15 mode (combination of five driving cycles for Japanese Driving Cycles). The regenerative braking and friction brake give the total braking force for a HV as illustrated schematically in Fig. 1.

Braking force, *T* (N)

RB

FB

Deceleration, J (m/s2)

**Fig. 1** Regenerative and friction brake force contributions

*2.2. The front wheel brake force*

The front wheel brake force depends upon the relationship between the brake pedal position and the master cylinder pressure. The total friction braking force is essentially proportional to the master cylinder pressure, but the deceleration of the vehicle must first be considered for the distribution between front and rear wheels.

*2.3. Regenerative braking torque (RBT)*

In an emergency situation the regenerative braking from an electric motor/generator is unlikely to be able to supply sufficient braking torque for the required deceleration, and has to be operated together with the friction brakes to provide the required braking power. The available regenerative brake force, *Freg* applied to the two front wheels can be written

*Freg =* (1)

Maximum generation power *PGemax* depends on the size of the electric motor/generator, and may be further limited by the rate at which energy can be transferred to the battery. For an electric motor it is safe to assume that the maximum generated power is equal to the nominal drive power as this means that the motor/generator and electric storage system will be operating within safe current limits. Using the symbol *N* for the motor speed in rev/min, the electric motor regenerative braking torque is calculated as

 (2)

The condition for *N* ≤ 1500 is applied as this reflects that the full power capacity of the motor can be used above 1500 rpm, with an approximately linear rise in power below this level (equal to a constant motor torque).

Fig. 2 shows that the torque is higher for a 50 kW motor/generator compared to smaller e.g. 20 kW and 10 kW motors. This shows that more powerful electric motors allow more regenerative braking torque than lower power motors. The Toyota Prius (ICONIC 2004-2009) model uses a 30 kW motor/generator in its regenerative braking system; the next generation (2010) has a 60 kW motor [6]. Regenerative braking torque also depends on battery storage [7].

 **Fig. 2** Regenerative braking torque, *Treg*comparisons

If the required braking torque, *Tb* is smaller than the available motor torque, *Treg,* the front wheels could theoretically achieve 100% regenerative braking (*Treg* > *Tb*) with purely regenerative front wheel braking. In practice power is limited by the design of the hybrid vehicle powertrain in terms of safe current limits and energy transfer rates (power). Under emergency braking the required vehicle deceleration is higher; the friction braking must work together with the regenerative braking (*Treg* < *Tb*). The distribution of brake forces between the front and rear wheels must be designed to achieve vehicle stability (e.g. high efficiency without premature rear wheel lock). The actual front wheel brake torque, *Tb ac* for a vehicle fitted with regenerative braking at the front wheels is calculated from

*Tb ac = Tf ac + TEMreg* (3)

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**Fig. 3** Regenerative braking map for a range of motor/generator sizes

The contribution that an electric hybrid system can give varies over the speed range. Theoretical values in Fig. 3 show the different between the torque generated by a 60 kW electric motor/generator compared with smaller power motors, especially at lower speed. This enables the operating point of the motor/generator to be specified to maximise regenerative power recuperation during braking.

*2.5 Electric motor / generators for a parallel hybrid braking system*

The configuration of a parallel hybrid vehicle’s braking system is similar to a conventional braking system which uses a hydraulic or pneumatic actuator to deliver braking force [1]. This configuration, as shown in Fig. 4, has all the major components of conventional brakes with the addition of regenerative braking from an electric motor / generator at the front axle.

*2.6 Hydraulic braking torque*

Calculations of the regenerative braking torque assuming that the friction brakes have a fixed ratio (*X1= 0.8* and *X2= 0.2*) braking force distribution on the front and rear wheels have been made for different sizes (power) of motor / generators to investigate how they affect regenerative braking in a passenger car. The calculation was programmed using MATLAB, see Fig. 5.

Fig. 7 shows the torque available from regenerative braking at low speed (15 km/h) for a range of motor powers (using gear ratios to maintain the motor at maximum power generation) as shown in Table 2. The torque is also expressed as a percentage of required front wheel torque. Where this torque exceeds 100% the regenerative braking must reduce braking torque by either a lower gearing, or by limiting current flow from the motor / generator. When the available regenerative torque is below 100%, the friction brake must be operated to supply the difference. Figures 8 and 9 show the corresponding results for 30 km/h and 60 km/h. The regenerative system is seen to provide less of the torque, and thus is able to regenerate less of the braking power from higher speeds.



Brake booster

Brake fluid

reservoir

 Mechanical

 Electrical

 **MODULATOR**

**ECU**

WHEEL SPEED SENSORS

Motor/generator and controller

Energy storage

o

o

o

o

HV ECU

Brake pedal

Master cylinder

Position sensor

Pressure

sensor

Brake caliper

Brake rotor

Axle

**Fig. 4** Parallel hybrid brake system

**YES**

**NO**

Calculate brake force demand, *Fbd*

*Fbd = FB+Freg*

*pf* = 0

Calculatepressure, *pf*

*pf = (p-pt)-pr*

Calculate *Tb* from

*Tbd = Tb + Treg*

*Tb = Treg*

Regenerative torque for *Treg*

Brake force, *FB*

*FB = 4\*(p-pt)\*A\*μ\*Re/Rr*

Regenerative torque motor, *TEMreg*

*TEMreg* = Power\*60/2\*π\**N*

**Driver pedal input**

Regenerative torque, *Treg*

*Treg = i\*TEMreg\*η*

*η* = 0.9

Regenerative force, *Fr*

*Fr = Treg/Rr*

***FB > Freg***

Both Regenerative

+

Friction braking

Regenerative only

**Fig. 5** Flowchart of regenerative braking used in calculation

**Table 2** Gear ratios for peak power used in simulations

|  |  |  |  |
| --- | --- | --- | --- |
| Gear ratio | 15 km/h, 0.15g | 30 km/h, 0.15g | 60 km/h, 0.4g |
| Lower | 1.53 | 0.76 | 0.38 |
| Higher | 7.64 | 3.82 | 1.91 |

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**Fig. 6** Friction braking energy recuperation potential for different motor/generator size (single stop braking for vehicle speed 15 km/h at 0.15 g)



**Fig. 7** Friction braking energy recuperation potential for different motor/generator size (single stop braking for vehicle speed 30 km/h at 0.15 g)



**Fig. 8** Friction braking energy recuperation potential for different motor/generator size (single stop braking for vehicle speed 60 km/h at 0.4 g

From the graphs above it can be seen that at 15 km/h the available regenerative braking torque produced by higher power motors (60 kW) must be limited because the regenerative torque is higher than the demanded braking torque, which would result in more deceleration than required. It could also result in the adhesion utilisation limit being exceeded at the front or rear wheels (*f1*, *f2*) leading to wheel lock. The fraction that the regenerative braking torque can provide varies depending on the vehicle speed and motor power. In this study the results shown in Fig. 6, 7 and 8 indicate that a 40 kW electric motor could recover up to about 45% of the total braking energy in single stop braking from 60km/h depending on the system efficiency.

**3. Braking energy in urban driving**

Driving in an urban area or in heavy traffic generates high energy dissipation from frequent braking. The test car (Table 1) was modelled to demonstrate the energies involved, and their variation with vehicle speed and deceleration during typical urban driving cycles. In the model the car was equipped with a 30 kW electric motor / generator similar to the Toyota Prius. The software package ADVISOR was used to perform calculations for a model hybrid electric vehicle; ADVISOR provides a convenient programming environment to quantify fuel economy, performance and emissions for advanced vehicle modelling [8]. The urban driving cycles that are used in this study were ECE, NEDC, INDIA URBAN, UDDS and FTP. Table 3 shows the results obtained.

**Table 3** Percentage of braking energy in different driving cycles

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
|  | Max. speed (km/h) | Ave. speed(km/h) | Max.decel(m/s2) | Travellingdistance (km) | Total tractionenergy(kJ) | Total brakingenergy(kJ) | Ratio of braking energy to traction energy (%) |
| ECE | 50.00 | 18.26 | 0.82 | 0.99 | 151.5 | 90.6 | 59.8 |
| NEDC | 120.00 | 33.21 | 1.39 | 10.93 | 2571.2 | 694.6 | 27.0 |
| INDIA URBAN | 62.56 | 23.41 | 2.10 | 17.49 | 2721.7 | 1640.3 | 60.3 |
| UDDS | 91.25 | 31.51 | 1.48 | 11.99 | 2276.1 | 1110.6 | 48.8 |
| FTP | 91.25 | 25.81 | 1.48 | 17.77 | 3577.7 | 1599.5 | 44.7 |

The minimum braking energy dissipation is in the New European Driving Cycle (27%) and the maximum braking energy dissipation comes from heavy traffic cycles such as ECE and India Urban where it has reached up to 60%. The percentage of total braking energy that can be utilised by regenerative braking at a front wheel is lower, and is given in Table 4. This shows that the available regenerative braking energy ranges from 28% to 43% on each wheel.

**Table 4** Available regenerative braking on two front wheels

|  |  |  |  |
| --- | --- | --- | --- |
|  | Total brakingenergy(kJ) | Total RBS energy on front wheels (kJ) | Available RBS on front wheels to total braking energy (%)(both front wheels) |
| ECE | 90.6 | 27.6 | 61.0 |
| NEDC | 694.6 | 301.5 | 86.8 |
| INDIA URBAN | 1640.3 | 461.2 | 56.2 |
| UDDS | 1110.6 | 336.3 | 60.6 |
| FTP | 1599.5 | 501.5 | 62.8 |

**4. Discussion**

The calculations above provide a basis for designing a HV braking system to recuperate maximum braking energy from the front wheels. The power capacities of the electric motor / generator are usually not sufficient to handle the large braking power when braking from high speeds, or at high deceleration. The electric motor/generator can provide up to its maximum braking torque and the friction braking can provide the remaining braking force demand. It has been shown that when the vehicle deceleration is less than 0.4 g at low speeds, the electric motor itself can provide all the required brake torque and no conventional braking is needed. However, when the required braking deceleration is higher than 0.4 g the required braking torque for the front wheels is greater than the electric motor / generator can provide. In this case the conventional brake has to apply additional braking force to provide the remaining force. From the calculations and simulation results, the use of regenerative braking in hybrid cars allows the braking energy dissipated by friction braking at the front wheels to be reduced between 30% and 45%. This shows that using regenerative braking in passenger cars gives a lower duty requirement to the friction brake, and the use of a lightweight brake is possible.

**5. Lightweight brake design concepts**

The conventional design for a front brake disc for a passenger car is a ventilated disc made from cast iron. Automotive manufacturers could fit new hybrid car models with lightweight components to increase their efficiency and performance, for example replacing conventional cast iron brake rotors with a thinner solid disc of appropriate material. This would reduce vehicle mass, help reduce fuel consumption and thus meet vehicle legislation in terms of vehicle emissions of CO2, HC, and NOx [9]. FEA has been used to estimate the disc temperature during vehicle braking [10, 11, 12, 13], and the results presented next investigate a design for a lightweight brake disc for the front axle of a hybrid car.

**6. Lightweight brake disc design**

One area in which lightweight discs are well developed is the motorcycle. Although motorcycles are much lighter (in term of gross weight vehicle mass) than cars, the design duty level of a motorcycle front disc brake is surprisingly high, largely because of the high speed performance required. The rotor is designed to withstand possible emergency braking from a high speed of 200 km/h and could reach a total kinetic energy of 231.8 kJ for a motorcycle weight of 300 kg including the rider. Itoh discussed the early design of motorcycle brake discs and the likely development of lightweight brake discs in the future. A motorcycle disc brake was developed in 1969 by Tokico using a one piece stainless steel disc with an aluminium caliper. The trend of disc brake development has generally concentrated on weight reduction and pad material improvement. Current disc brakes for motorcycles mostly use stainless steel with an aluminium alloy casting for the rotor and caliper body respectively. Progress has been made on a lightweight disc brake using a carbon composite rotor and a magnesium forging for the caliper body [14]. It was found that the performance of brakes using advanced lightweight materials can be very competitive, but are too expensive for road use. Stainless steel has been adopted for motorcycle brake discs apparently for mainly cosmetic reason; many researchers e.g. Boniardi [15] investigated the lifespan of stainless steel brake discs and found that small cracks can occur after a few thousand miles of use, usually located near to the fixing holes on the flange. The cracks were found to be caused by thermal cyclic strain during brake action. Boniardi used two types of brake discs made from martensitic stainless steel. Each disc had a different chemical composition (type A and type B discs); these discs were then assessed against AISI 410 standards. The results show that the life of a brake disc depends upon the position of the ventilation hole in the disc, the shape of the spokes and the material properties at high temperature. The cracks that were found had possibly developed from excessive tempering of martensite at the high working temperatures. The type A disc, which contained greater amounts of vanadium and molybdenum, was preferred because it was more resistant to high temperature.

A prediction method was proposed by Yuasa [16] for crack initiation in motorcycle brake discs under extreme braking conditions. The tests were conducted at a constant braking torque using one-piece type brake discs made of SUS410DB. These brake discs had several ventilation holes to dissipate heat and to refresh the pad surface from extreme high temperature. Temperature distributions were measured using thermocouples at the locations where the disc temperature was expected to be highest. Strain gauges were located at fixing holes to measure changes where cracks were expected to initiate. A method to predict the fatigue life of a disc was proposed by Ichikawa [17] using an S-N curve to find the structural damage when a material is subject to cyclic loading. From the equation of damage (Miner’s law) he calculated damage at each strain level, *Di = ni / nfi*. The results agree reasonably well with the experimental life. This type of analysis would be necessary if lightweight brake discs were to be designed in this form because crack propagation leading to failure may be the limiting life parameter.

**7. Finite element models**

3-D finite element models of two brake discs were developed using the ABAQUS / CAE 6.8 software package. The two types of disc modelled were a ventilated disc and a solid disc. Pads and piston assemblies were modelled using 8-node coupled temperature and displacement elements in a cylindrical coordinate system. The ventilated disc had a total of 2175 elements with 3128 nodes, and the solid disc had a total of 1188 elements with 1728 nodes (Fig. 10). The contact surface frictional behaviour was simulated with a wheel rotational speed of 74 rad/s (the average maximum speed based on single stop braking) with an initial disc temperature of 20°C. Frictional heat was generated by pressing the pads against the disc with a uniform pressure of 6 MPa on the piston side of the discs.

|  |  |
| --- | --- |
| ∅ 258 mm∅ 258 mm(a) | (b) |
| **Fig. 10** (a) FE model of standard  ventilated disc. Total mass is  5.71 kg | (b) FE model of lightweight solid disc with hub adapter. Total mass is 3.75 kg |

*7.1. Calculation of braking temperatures during single stop braking*

The geometries of the lightweight solid disc and the standard ventilated disc are given in Table 5, and the properties of the materials for the two discs are given in Table 6. FEA simulation of single stop braking was used to determine the effect of the vehicle mass on front brake temperatures in terms of the local temperatures and stress. Thermal conduction and convective heat transfer were the two modes of heat transfer considered. A convection heat transfer coefficient of 100 W/m2 K was assumed over all exposed surfaces and radiative heat transfer was assumed negligible. This is a realistic approximation as radiative transfer only becomes significant at higher temperatures than those involved here [18]. Both discs had a heat flux applied at the interface with the pad, this interface moved as different parts of the disc came into contact with the pads. Reference [9] gives an equation describing average heat flux for single stop braking.

**Table 5** Brake disc comparison

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
|  | Thickness (mm) | Disc diameter (mm) | Effective radius (mm) | Friction | Piston diameter (mm) |
| Standard ventilated disc | 22 | 258 | 101 | 0.4 | 53.8 |
| Solid lightweight disc | 7 | 258 | 101 | 0.4 | 53.8 |

**Table 6** Material properties used in the FE models for solid disc (steel) and ventilated disc (cast iron)

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
|  | Thermal conductivity, *k* (W/m K) | Specificheat, *Cp* (J/kg K) | Mass density, *ρ* (kg/m3) | Young’s modulus, *E* (GPa) | Poisson’s ratio, υ |
| Cast iron | 43 | 500 | 7200 | 116 | 0.25 |
| Stainless steel | 25 | 460 | 7800 | 200 | 0.30 |

Temperatures during single stop braking were predicted for both discs using FEA simulations for vehicle masses of 1000 kg, 1500 kg and 2000 kg. Fig. 11 shows the predicted temperature profiles at a point on the rubbing surface of each disc without any regenerative braking (100% duty level on both discs). Fig. 12 shows the corresponding results where 30% of the braking energy has been absorbed by regenerative braking. A peak is seen at every revolution on all curves as the measurement point moves past the friction pad. In both cases the ventilated discs remain cooler than the solid discs, by approximately 50°C. The high speed stop results are shown in Fig. 13; the vehicle equipped with regenerative braking has a solid disc but has a much reduced peak temperature compared with the solid disc without regenerative braking. This suggests that a lightweight brake could be used in a hybrid car even though the brake disc mass is reduced (from approximately 5.25 kg to 3.75 kg). The temperature of a solid disc with regenerative braking could possibly be further reduced by additional design improvements or by providing an extra cooling system for the disc surface.

**Fig. 11** Temperature profiles of ventilated and solid discs without regenerative braking (100% braking duty level for both)

**Fig. 12** Temperature profiles of ventilated disc (100% duty level) and solid disc with regenerative braking (70% duty level)

**Fig. 13** Comparison of peak temperatures on ventilated and solid discs, and the solid disc with regenerative braking for single stop braking

**8. Summary, conclusion and further work**

An analysis method has been presented which calculates the effect of regenerative braking and friction braking on a small car. The regenerative braking energy during single stop braking was analysed using ADVISOR, which was also used to analyse potential regenerative braking energy in selected urban cycles e.g. ECE, NEDC, INDIA URBAN, UDDS and FTP. FEA thermal analysis of lightweight brake discs has predicted the temperature performance of a lightweight brake disc fitted to a medium-sized car. Comparisons were made without any other modifications and showed that regenerative braking has a significant effect upon peak disc temperature during single stop braking. A solid brake disc was shown to give very similar results to a ventilated brake disc where regenerative braking accounts for 30% of the total braking energy. In view of the brake duty advantages that regenerative braking offers, a prototype lightweight brake could effectively be designed for use on a hybrid car with RB.

It can be concluded that the combination of friction braking and regenerative braking can reduce the duty level on the front friction braking to the extent that a lightweight brake disc could be designed and used effectively to provide the required performance levels. Based on the results, the total braking energy in one stop braking from 15 km/h could be recuperated by a 30 kW motor / generator. In urban cycles, and between 30% and 45% could be recovered for a medium size (1500 kg) hybrid car.

This study has quantified the potential for the use of lightweight brake discs for friction braking in conjunction with regenerative braking using standard motor / generators within drive cycles. A lightweight friction brake could be designed for lower duty but this needs further analysis and experimental verification. Thermal stress analysis will be performed using FEA and then verification and validation will be made using the test car on a rolling road facility at the University of Bradford.

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