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Disc Brake Noise Reduction Through Metallurgical Control of Rotor Resonances

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ABSTRACT

The mechanical properties of a gray cast iron disc brake rotor are directly influenced by the amount and morphology of the graphite present throughout the rotor. Two of these properties, the modulus of elasticity and the damping capacity, can have a significant effect on the propensity for the disc brake rotor to produce noise. The noise propensity of a disc brake is in a large part determined by the relationship between the rotor resonances and the resonances of the other brake components such as the pads. In this paper, we are concerned only with the effect that modulus of elasticity has on disc brake noise through its influence on rotor resonances.

The amount and morphology of the graphite in gray cast iron is determined by the carbon content and silicon content of the iron. The carbon and silicon content are measured by one parameter called the carbon equivalent. For gray cast iron the relationship between carbon equivalent and modulus of elasticity is almost linear for the grades used in disc brake rotors. This relationship allows the modulus of elasticity and, in turn, the rotor resonances and resulting brake noise to be influenced by the carbon equivalent of the rotor.

A case study showing the effectiveness of controlling rotor resonances through carbon equivalent to reduce brake noise is presented. The subsequent effect of foundry process control on brake noise propensity is also evaluated.

INTRODUCTION

Gray cast iron has been traditionally and remains presently the choice for disk brake rotors (ref. 1) The modulus of elasticity and the damping capacity are two of the mechanical properties of gray cast iron that can have a sigr ficant effect on the propensity for the disk brake to produce noise. The modulus of elasticity and damping capacity of gray cast iron are properties that are directly influenced by the amount and morphology of graphite present in the iron. The amount and morphology of graphite is to a significant degree determined by the carbon and silicon content of the iron. The carbon and silicon content are measured by on parameter called the carbon equivalent. Therefore, the modulus of elasticity and damping capacity, both of which influence the noise propensity of gray cast iron rotors can be controlled by one metallurgical parameter, the carbon equivalent.

CARBON EQUIVALENT, MODULUS OF ELASTICITY AND ROTOR RESONANCES

The carbon equivalent (CE) of gray cast iron in % is equal to the percent carbon plus one third of the percent silicon (ref.3). This is indicated in the following formula:

$$CE = \%Carbon + (\%Silicon)/3 \qquad (1)$$

The carbon equivalent is a quantity that is commonly reported in the chemical analysis of gray cast iron. In references (4) and (8), the percent phosphorus is included with the percent silicon.

Modulus of Elasticity

The modulus of elasticity of gray cast iron is affected by its chemical composition and microstructure. It is not a constant. This is a unique characteristic that is not found in other ferrous materials such as steel. This characteristic is discussed in references (2), (5), (6), (7) and (8).

The carbon and silicon content as measured by the carbon equivalent of gray iron has a significant effect on modulus of elasticity (E). This characteristic is shown in Figure 1 below. The relationship between carbon equivalent and modulus of elasticity is almost linear for the range of values shown in Figure 1. It can be approximated by the following equation over the range $3.3 \leq CE \leq 4.8$ percent.

$$E\ (\mathrm{GPa}) = 310.3 - 54.1CE \qquad (2)$$

The data from which equation (2) is derived is from reference (8), Table 6.3.4. It is a linear regression fit with a correlation coefficient of 0.982. It is also presented in reference (9). This result is qualitatively supported by data from references (4), (5), and (6).

Rotor Resonances

The theoretical resonances of a disk brake rotor are directly proportional to the square root of the modulus of elasticity E, a function (A) of the geometry and boundary conditions only, and a function (B) of the density and Poissions ratio. This is shown by equation (3).

$$f_i = (E)^{1/2}\ A(h, \lambda_i, a) / B(\mu, \nu) \qquad (3)$$

Where:

$$A(h, \lambda_i, a) = (h/2\pi)(\lambda_i/a)^2 \qquad (4)$$

and:

$$B(\mu, \nu) = (12\mu(1-\nu^2))^{1/2} \qquad (5)$$

Combine equations (3), (4), and (5) to get:

$$f_i = (E)^{1/2}((h/2\pi)(\lambda_i/a)^2)/(12\mu(1-\nu^2))^{1/2} \qquad (6)$$

This is the equation for the resonances of a free-free circular/annular plate (ref. 8).

These variables are defined in the Definitions section.

Effect of Carbon Equivalent on Rotor Resonances

Differentiating equation (3) and applying some algebra, it can be shown that the variation of rotor resonant frequency (Δf) in percent is equal to one half the variation in modulus of elasticity (ΔE) in percent. This is shown by the following equations:

$$\Delta f = \Delta E / 2 \qquad (7)$$

where Δf is the change in frequency ∂f_i divided by the nominal frequency, f_i:

$$\Delta f = (\partial f_i / f_i) \qquad (8)$$

and similarly:

$$\Delta E = (\partial E / E) \qquad (9)$$

Applying a similar process to equation (2) for E as a function of carbon equivalent CE and taking the absolute value, get the following relationship for the variation in modulus of elasticity, E:

$$\Delta E = \Delta CE / ((5.78/CE)-1) \qquad (10)$$

where ΔCE is the variation in carbon equivalent in percent and is defined as the change in carbon equivalent, ∂CE divided by the nominal value of the CE:

$$\Delta CE = (\partial CE / CE) \qquad (11)$$

Combining equations (10), (11) and (7), get the variation in resonant frequency in percent as a function of the nominal carbon equivalent in percent and the variation in carbon equivalent in percent:

$$\Delta f = \Delta CE / (2.0((5.74/CE)-1)) \qquad (12)$$

The range of carbon equivalent where this relationship is valid is $3.3 \leq CE \leq 4.8$ percent. From equation (12), the variation in resonant frequency in percent will fall within the following range:

$$0.676\ \Delta CE \leq \Delta f \leq 2.55\ \Delta CE \qquad (13)$$

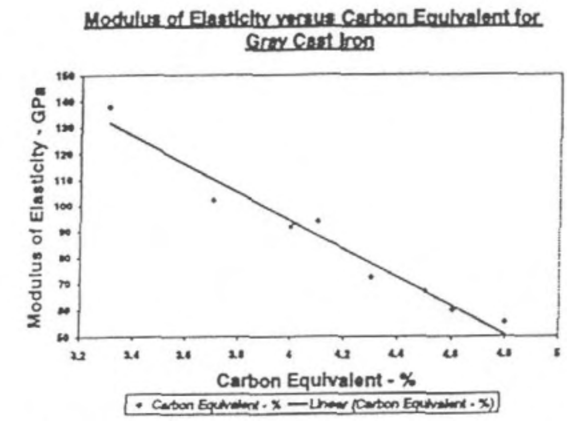

Figure 1 Modulus of Elasticity versus Carbon Equivalent for Gray Cast Iron Data Sources: Ref. (8)

CASE STUDY

The effect of carbon equivalent on the propensity for a disk brake to produce noise is analyzed in the following vehicle case study. Two geometrically identical rotors were produced from two different grades of gray cast iron. The chemical analysis of these rotors is shown in Table 1 below.

Table 1

Rotor Material Characteristics

Rotor No.	Grade No.	Chemical Analysis - %			
		C	Si	"CE" %	"E" GPa
1	G3000	3.201	2.460	4.021	92.76
2	G3500	3.254	1.846	3.869	100.99

The microstructure of both rotors was similar. Rotor number 1 had a hardness of BHN 217. The microstructure consisted of flake graphite in a pearlitic matrix in both the tophat section and rotor section. The graphite flakes in the tophat section were type B and C and size 5 – 5. The graphite flakes in the rotor section were type B and C and size 3 – 4 according to the American Foundrymen's Association Chart (ref,10).

Rotor number 2 had a hardness of BHN 207. The microstructure consisted of flake graphite in a pearlitic matrix in both the tophat section and rotor section. The graphite flakes in the tophat section were type B and C and size 5 – 5. The graphite flakes in the rotor section were type C and E and size 4 – 5 according to the American Foundrymen's Association Chart (ref,10).

Correlation of Carbon Equivalent with Resonant *Frequency Shift*

The values for "E" in Table 1 were calculated from Equation (2) and the measured carbon equivalent "CE." Using these values for E and Equation (3), it can be shown that rotor no. 2 should have resonant frequencies that are 4.2% higher than rotor no. 1 or visa versa. The results of these calculations are shown in Table 2 below. In simple terms, the predicted resonance of rotor 2 is the measured resonance of rotor 1 multiplied by the square root of the ratio of the modulus of elasticity for each rotor or visa versa.

TABLE 2
Comparison of Measured vs Predicted Resonant Frequencies for Rotors 1 & 2

Measured Resonant Frequencies - Hz		Predicted Resonant Frequencies - Hz	
Rotor 1	Rotor 2	Rotor 1	Rotor 2
2300	2450	2347	2397
3400	3550	3401	3543
4700	4900	4694	4897
6250	6500	6227	6513
7950	8300	7951	8284
9900	10250	9820	10316
12000	12400	11879	12504
14200	14700	14083	14796
16550	17150	16430	17245

STATISTICAL CORRELATION

Statistically there is a good correlation between the measured and predicted values for the rotor resonances in Table 2.

- The correlation coefficient from a linear regression analysis of the measured versus predicted rotor resonances is 0.99998.
- A Chi-Squared Test of the predicted versus measured resonances supports the hypothesis that the shift in rotor resonance can be predicted by carbon equivalent.

These results support the case for using carbon equivalent as one of the parameters to control when it is desirable to control rotor resonances.

Resonance Shift versus Noise Propensity

In the remaining discussion, it will be shown that this shift in rotor resonance can reduce the propensity for the disk brake to produce noise. The differences between the measured resonances for rotors 1 and 2 and the brake pad resonances is calculated in Table 3. This data is also graphically presented in figure 2, below. For rotors 1 and 2, this is difference 1 and 2, respectively. The average of these differences for rotors 1 and 2 is 360 Hz and 100 Hz respectively.

When the proximity or difference between disc brake rotor and pad resonances is less than approximately 200 Hz, experience has shown that the propensity for noise generally increases. This happens because of intra-modal excitation and modal locking between the pad and rotor. Based on this, rotor 2 should show a higher propensity for noise than rotor 1.

TABLE 3
Comparison of Resonances for Rotors 1 & 2 and Current Production Pad Assembly

Measured Resonant Frequencies - Hz				
Rotor 1	Difference 1	Rotor 2	Difference 2	Pad Assy.
2300	250	2450	100	2550
3400	300	3550	150	3700
4700	-----	4900	-----	-----
6250	400	6500	150	6650
7950	-----	8300	-----	-----
9900	400	10250	50	10300
12000	-----	12400	-----	-----
14200	450	14700	50	14650
Average	360	Average	100	

VEHICLE TESTS

To verify the above hypothesis, both rotors were subjected to the Bosch Braking System (BBS) WI-561 1000 Stop Cold Chamber Evaluation. This test involves approximately 50 cold stops from 0 degrees F alternating with 200 miles of Simulated Los Angeles City Traffic (SLACT) at the Bosch Automotive Proving Grounds (BAPG).

In order to reduce vehicle specific effects, these tests were performed twice using two vehicles. Initially, new

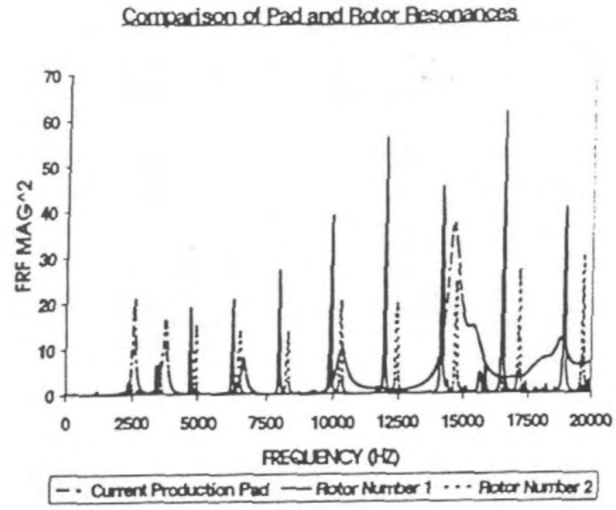

Figure 2 Spectrum of pad and rotor resonances

rotors and pads were installed on each vehicle - two of the no. 1 rotors on one vehicle and two of no. 2 rotors on the other vehicle. Then, upon completion of the first 1000 stop test, new rotors and pads were installed on the opposite vehicles and the 1000 stop test repeated.

The rotors and pads were hand selected such that their resonances were similar. Resonances were verified with a plus or minus 25 Hz measurement error and the pads were all from the same batch. The results of these tests are presented in Table 4, below.

TABLE 4
Vehicle Test Summary – BBS WI-561 1000 Stop Cold Chamber Evaluation

	Rotor 1	Rotor 2
Total Number of Stops	12982	13877
Noisy Stops:		
End of Stop Crunch	17	46
Moan / Howl	9	18
Low Frequency Squeal (< 5kHz)	0	65
High Frequency Squeal (> 5kHz)	3	0
Total Number of Noisy Stops	29	129
Percent of Noisy Stops	0.22%	0.93%
Total Test Miles	3765	3885
Number of Noisy Stops per 1000 Miles	8	33
Number of Noisy Stops per 1000 stops	2	9

From Table 4, rotor no. 2 has almost 4 times the number of noisy stops per 1000 stops and 4 times the number of noisy stops per 1000 miles. It exhibited more end of stop crunch, more moan / howls, and more low frequency squeals than rotor number 1. This supports the original hypothesis above that rotor 2 should show a much higher propensity for noise than rotor 1.

The total number of stops in Table 4 also includes all o the SLACT noise evaluation stops. There were no noises observed during the SLACT part of the test for either vehicle. Although the total number of stops was different for each rotor, both rotors saw only 1000 cold chamber evaluations. The differences occurred becaus of weather and scheduling problems during the SLACT sections.

Damping effects

As mentioned initially, the carbon equivalent also influences the damping. Based on the carbon equivalen for rotors 1 and 2, the damping would be .15% and .13% respectively (ref. 4). This slightly higher damping value for rotor 1 over rotor 2 could have also contributed to its reduced propensity for noise. However, modal damping measurements showed the rotor number 2 to have an average modal damping of 0.25% and rotor number 1 to have an average modal damping of 0.18%. Since rotor number 1 was quieter than rotor number 2, the damping appears not to be a significant effect in this case.

PROCESS CONTROL CONSIDERATIONS

At this point, it has been shown that the shift in resonant frequencies of geometrically identical gray cast iron rotors can be predicted by the difference in carbon equivalent of the material and that this factor can be used to control the noise. The implication of this factor is that an uncontrolled variability in the iron making process could, through the relationship between carbon equivalent and resonance shift result in increased brake noise. If the variation in carbon equivalent from foundry process control could be estimated, equation (12) would provide a method of evaluating this effect. To determine this variation, a batch of 5 similar rotors from 3 different foundries were subjected to chemical analysis. The results of this analysis are shown in Table 5. ΔCE is the variation in carbon equivalent in percent and is assumed to be equal to the normalized sample standard deviation. This is defined as the magnitude of the sample standard deviation of the carbon equivalent in percent divided by the sample average value of the CE in percent (see eq. (11)). As shown at the bottom of Table 5, the average sample standard deviation ΔCE for all three foundries is 1.17%. The resulting variation in rotor resonances, Δf from equation (12) would be 1.27%.

This data is typical of foundry process control and falls within the guidelines presented in SAE J431 (ref.3) for Automotive Gray Iron Castings. Table A1 in this reference shows typical carbon equivalent values to vary from 3.9% to 4.2% for grade G3000. If it is assumed that this spread in CE values covers 6 sigma (6 standard deviations), then the resulting value for ΔCE, calculated from equation (11) is 1.28%. This is greater than the sample standard deviation of 1.17% for the three foundries shown in Table 5. This means that these foundries are on the average controlling carbon equivalent within acceptable limits.

Returning to the Case Study for a moment, it is possible that both rotor 1 and 2 could have both been produced by Foundry number 3. The ΔCE calculated for rotors 1 and 2 is 1.31%. This is less than the ΔCE of 1.39% for Foundry 3 as shown in Table 5. This means that rotors 1 and 2 could have been produced during the normal operation of Foundry 3. Assuming that the pad properties are stable, this is a possible explanation for the some of the unexplained random brake noise problems that crop up in previously quiet production systems.

If it is accepted that when the proximity or difference between disc brake rotor and pad resonances is less than approximately 200 Hz, the propensity for noise generally increases, then the above level of foundry process control is probably not adequate. To show this, use the data from Table 5 for Foundry 1. This foundry had the lowest variation. Assume first that a ΔCE of 0.99% and the corresponding Δf of 1.14% are reasonable estimates of sigma and can be used to estimate a desired rotor to pad modal separation for reduced noise propensity. Assume also that it is desirable to have plus or minus 3 sigma (99.7%) of the rotor population and plus or minus 3 sigma (99.7%) of the pad population have resonances that are never within 200 Hz of each other. Finally, assume that plus or minus three sigma (3Δf = 3.42%) encompasses the resonant frequency variation of 99.7% of the rotor population and 99.7% of the pad population. The assumption that the pad population has the same variability as the rotor population has no basis in fact and is used only for example calculations. The actual data from the pad population should be used for realistic calculations. However, based on these assumptions, it can be calculated that, for example at 2 kHz, the desired rotor to pad modal separation is 349 Hz, and at 20 kHz, the desired rotor to pad modal separation is 1516 Hz. The magnitude of these modal separations would be difficult to achieve without major changes to pad and rotor geometry.

TABLE 5

Summary of Carbon Equivalent Data and Resonant Frequency Variation from Rotor Chemical Analysis from Three Foundries

Foundry Number	Sample Average CE - %	Sample Standard Deviation (ΔCE) -%	Δf from Eq. (12) %
1	4.027	0.99	1.14
2	3.846	1.13	1.12
3	3.986	1.39	1.54
	Averages:	1.17%	1.27%

However, significant improvements in foundry process control are technically feasible. Using the steel industry as a guide (ref.11), the carbon content in steel can typically be controlled to a standard deviation 0.01% carbon. If it is assumed that carbon equivalent can be controlled to the same degree, then for rotor number 1, ΔCE would be 0.25%. Follow the same method as described in the above paragraph. For 99.7% of the pad and rotor population to never have less than 200 Hz modal separation, the modes should be separated by 440Hz at 14200 Hz. This compares favorably with the 450 Hz separation for rotor number 1 shown in Table 3. This was the quiet rotor. It should be remembered that all of the above calculations are based on the assumption that pad variability is the same as rotor variability. This being the case, the application of the 200Hz minimum modal separation limit is technically feasible with improvements in pad and rotor process control.

CONCLUSIONS

- The shift in resonant frequencies of geometrically identical gray cast iron rotors numbers 1 and 2 can be predicted by the difference in carbon equivalent of the iron. This means that these resonant frequencies can be significantly shifted by varying the carbon equivalent of the iron.
- This shift in resonant frequencies can be used to reduce the propensity for the disc brake to produce noise.
- The relationship between carbon equivalent and rotor resonant frequency can be used to set targets for process control to reduce the noise propensity of a disc brake.

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Definitions, Acronyms, Abbreviations

CE = Carbon equivalent

E = Modulus of Elasticity

c = Damping Capacity

f_{ij} = the resonant frequency of the i,j mode

$A(h, \lambda_{ij}, a) = (h/2\pi)(\lambda_{ij}/a)^2$

$B(\mu,\nu) = (12\mu(1-\nu^2))^{1/2}$

a = the outside radius of the plate

h = the thickness of the plate

μ = the mass density of the plate

ν = Poisson's ratio

λ_{ij} = the dimensionless frequency parameter generally dependent on the

geometry and boundary conditions of the plate

i = number of nodal diameters

j = number of nodal circles

∂ = partial derivative