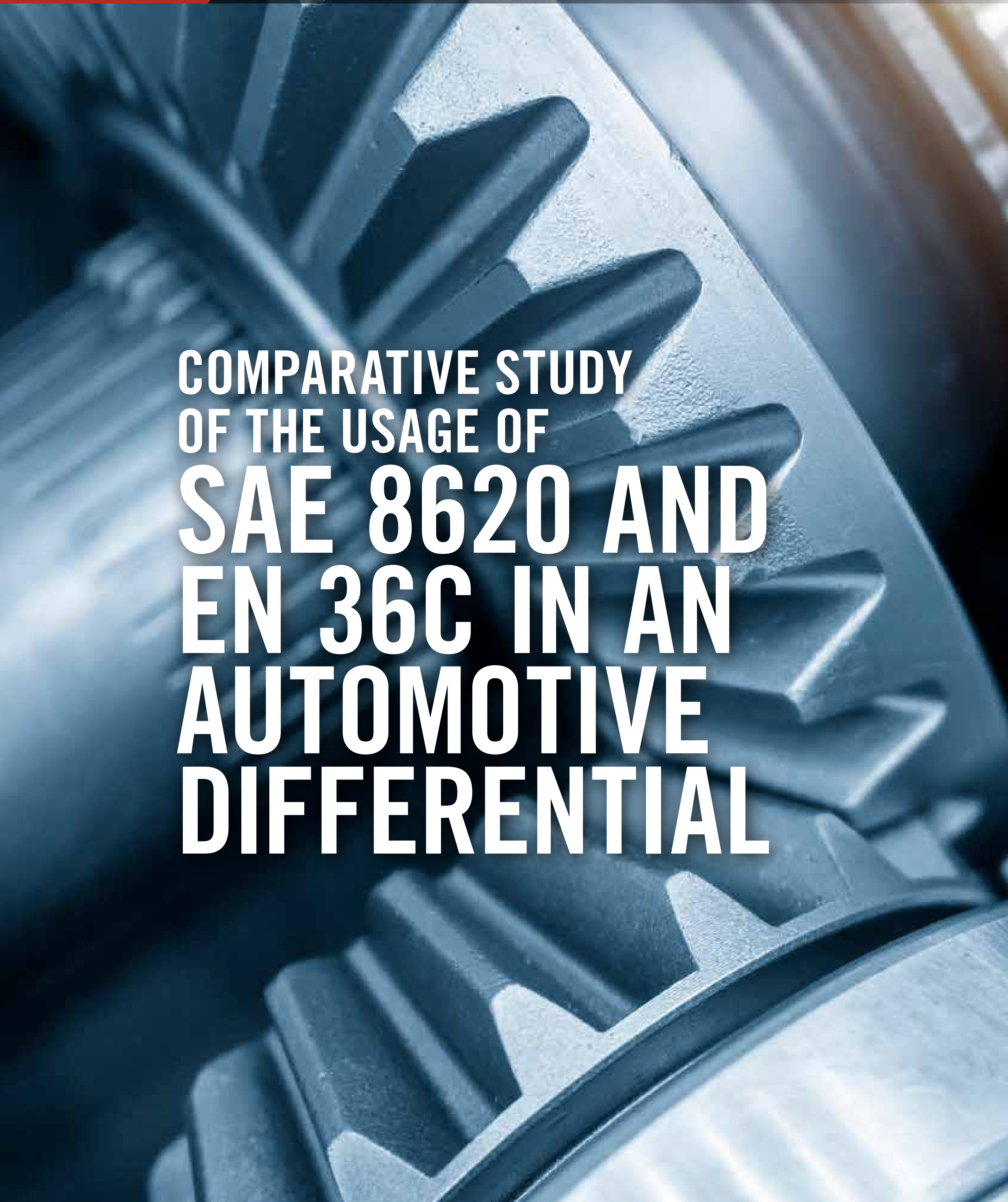

COMPARATIVE STUDY OF THE USAGE OF SAE 8620 AND EN 36C IN AN AUTOMOTIVE DIFFERENTIAL

Using a differential gearbox design, a comparison between two popular case hardening alloy steels shows that EN 36C offers significantly better tool strength, wear resistance, and fatigue properties.

By SAURAV KAMBIL and ANKUR MIGLANI

An automotive differential gearbox is a critical component of the powertrain involved in the transmission of power and its appropriate distribution to the wheels. The gears of a differential are subjected to high stresses over multiple cycles making durability and reliability a significant concern to designers and manufacturers. Case hardened alloy steels such as SAE 8620 and EN 36C are commonly employed in such applications in the gear industry. The latter provides better performance in terms of core strength and wear properties but at a higher cost. This comparative study analyzes the differences in performance offered by SAE 8620 and EN 36C. Two open differential gearboxes are designed for the tipper truck 2516M from Ashok Leyland to simulate a scenario where gear manufacturers consider these materials. For identical pinion dimensions, EN 36C showcases a higher factor of safety by 17 percent, 52 percent, and 20 percent in tooth bending strength, surface wear resistance, and fatigue properties. SAE 8620, on the other hand, shines in the pricing department, offering satisfactory properties at a 16 percent lower cost when considering high volume production.

1 INTRODUCTION

During the execution of a turn, the inner wheels of a vehicle must turn at a slower rate than the outer wheels. Power delivery employing a rigid drive shaft would lock the wheels together, causing slippage and loss of control while cornering [1]. A solution comes with using a particular gear arrangement between the driveshaft and the axle known as the differential. Being found in live-axle housing assemblies of automobiles, trucks, and heavy vehicles, they are responsible for two functions: (i) Power transmission through a 90° bend to the axle and (ii) appropriate power distribution to each wheel.

Figure 1 shows a schematic diagram of a conventional open differential consisting of bevel gears found in most commercial vehicles. Inside its housing, the differential receives power from the drive shaft through the pinion gear — the pinion meshes with the ring gear, which is rigidly attached to the spider gears. The spider gears mesh with the axle shaft gears completing the gear train. The differential assembly moves as a locked unit during straight-line motion, but when executing a turn, the spider gears "walk" around the axle gears allowing for an independent rotation of each wheel.

NOMENCLATURE

P_t= Static tooth load.
M_t = Torque.
d_G= Diameter of gear.
d_P = Diameter of pinion.
v = Pitch line velocity.
a = Helix angle.
BHN = Brinell Hardness Number.
E = Elastic modulus.
s_c = Compressive stress.
s_b = Bending stress.
m = Module.
T_p = No. of teeth on the pinion.
T_g = No. of teeth on the gear.
Y = Tooth form factor.
V = Peripheral speed.
P_D = Dynamic tooth load.
C_s = Service factor.
C_v = Velocity factor.
C = Deformation factor.
W_S = Static tooth load.
GR = Gear ratio.
K = Load stress factor.
e = error between meshing teeth.
S_b = Tooth bending strength.
S_w = Tooth wear strength.
s_m = Midrange stress.
s_a = Amplitude stress.
K_a = Surface condition modification factor.
K_b = Size modification factor.
K_a = Load modification factor.
K_a = Temperature modification factor.
K_a = Reliability factor.
K_a = Miscellaneous effects modification factor.
$S_{f'}$ = Rotary-beam test specimen endurance limit.
S_f = Endurance limit at the critical location of a machine part in the geometry and condition of use.

The differential is a critical component of the automobile powertrain as it is responsible for transmitting power to the wheels. This role involves withstanding significant stresses over multiple cycles during the lifetime of a vehicle. High stresses can cause the gears of a differential to undergo different modes of failure such as tooth bending, plastic flow, pitting, wear, and scoring [2], which in the long run would lead to a significant reduction in power and loss of the vehicle. To increase the life and reliability of the gears, it is vital to choose an appropriate material and incorporate a proper safety factor.

Gears are often made from high-tensile-strength alloy steels. They offer a winning combination of high-tensile strength, high-wear resistance, excellent thermal properties, and the ability to undergo heat-treatment processes that enhance their properties, all at a competitive cost. During manufacturing, the gears can be carburized (i.e., case hardening [3]), which is one of the most critical steps in strengthening gears. This process increases wear resistance significantly while maintaining the ductility of the core, which ensures the capacity to withstand shock load and endure cyclic stresses. SAE 8620 and EN 36C are two case hardening alloy steels commonly used in gears [4-5]. In the carburized form, the former is deployed in the differential of commercial vehicles. While the latter, being a superior grade, is generally used in heavy-duty applications such as high-strength crankshafts, connecting rods, gearing, shafts, and couplings in the automotive and aerospace sectors.

Since both materials offer great strength and durability, they are often used interchangeably as gear materials. When it comes to gear material selection, three primary factors must be considered: strength, durability, and cost. As the importance given to each factor varies with the application, engineers must work to meet the desired performance while keeping the costs to a minimum. Hence, it is essential to analyze the differences in performance offered by SAE 8620 and EN 36C quantitively. This analysis would assist engineers in making a well-informed decision regarding material choice.

This article draws a comparison between two case hardening alloy steels, EN 36C and SAE 8620, used in the gear industry. Two open differential gearboxes are designed using each material for the tipper truck 2516M by Ashok Leyland to simulate a scenario where gear manufacturers consider these materials. The juxtaposition of the steels is carried out in three steps while keeping the power delivery requirement constant. Firstly, the designed differentials are compared in terms of resistance to tooth bending and pitting failure. Secondly, fatigue failure analysis is performed to confirm design longevity. Finally, the weight and cost of manufacturing, often considered the most critical properties, are compared for the designed differentials.

2 MATERIALS AND METHODOLOGY

2.1 MATERIAL PROPERTIES

SAE 8620 is a low-carbon alloy nickel-chromium-molybdenum case hardening steel. This alloy steel is characterized by great external strength and good core toughness. It is generally supplied in the rolled condition with a maximum hardness of 255 BHN, but when carburized, the surface hardness can go up to BHN 680 [4]. Owing to this, SAE 8620 is used extensively by all industry sectors for light- to medium-stressed components requiring high-surface wear resistance with reasonable core strength and impact properties. Its typical uses include gears, camshafts, bearings, fasteners, and piston pins. The technical properties of the steel are shown on an octagon plot in Figure 2. The current price in the Indian market of SAE 8620 rolled steel is listed as well.

EN 36C is a low-carbon nickel-chromium carburizing steel with a high-surface strength while maintaining a ductile core. When surface hardened by a thermo-mechanical process, it offers remarkable toughness along with corrosion resistance, shock resistance, good fracture toughness properties, and is compatible with dynamic load fluctuations. Typical applications include components requiring high toughness and core strength, such as gears, crankshafts, heavy-duty gear shafts in aircraft and trucks. Carburized EN 36C can reach surface hardness values up to BHN 760 [5]. The mechanical properties are illustrated in the octagon plot of Figure 2. The current price in the Indian market of EN 36C rolled steel is listed as well.

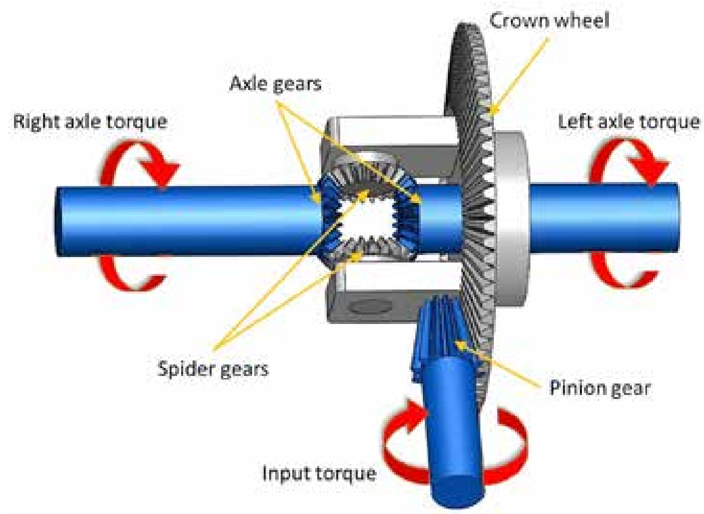

Figure 1: A schematic diagram shows the working of an open differential.

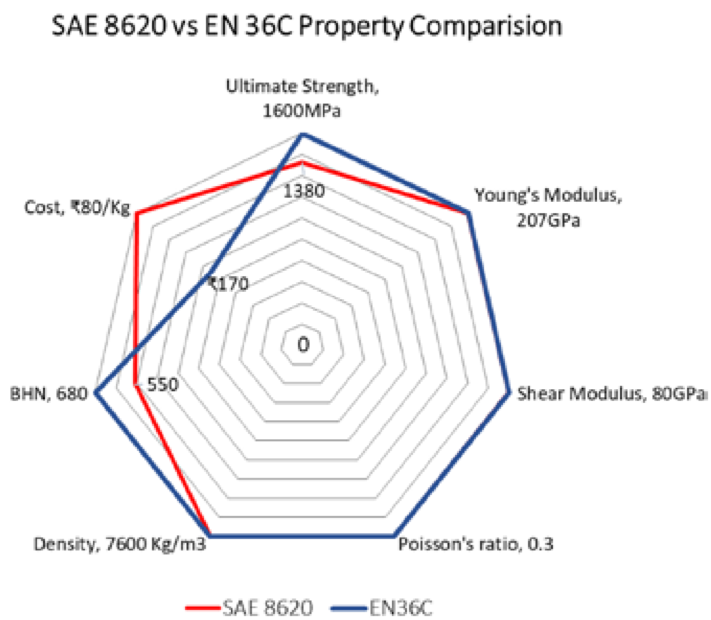

Figure 2: A chart comparing the properties of SAE 8620 and EN 36C.

2.2 DESIGN CONSIDERATIONS

The primary design consideration for this differential is that the engine outputs maximum torque at 2,400 RPM, and the maximum power at the given RPM is 162 BHP. The vehicle under consideration here is a heavy transportation truck built for optimal highway driving conditions, and hence an open differential was considered for the design. The manufacturer's data sheet defines the final drive ratio as 6.25 when standard tires provided by the manufacturer are used. The above data is obtained from the seller's datasheet [6]. Other design assumptions include:

- 20° full depth involute teeth.

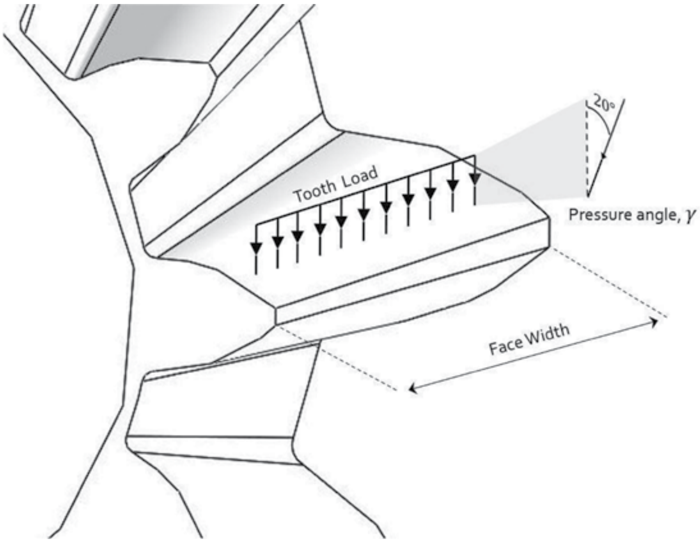

Figure 3: A schematic diagram of the dynamics of a bevel gear tooth.

- **Number of teeth on pinion:** 12.
- **Number of teeth on the crown wheel:** 75.
- **Number of teeth on spider gear:** 20.
- **Number of teeth on axles shaft gear:** 20.
- **Number of spider gears:** 2.
- **Efficiency in power transmission:** 100%.
- The effect of axial forces on the differential is ignored.

2.3 GEAR DESIGN EQUATIONS

The gears are designed around two criteria: (i) tooth bending strength and (ii) surface pitting resistance. The bending strength is estimated using the Lewis equation for bevel gears [7]. In this method, the gear tooth is modeled as a simple cantilever beam taking the tangential load (P_t) at the tip, causing bending stress at the base of the tooth.

Failure due to pitting occurs when the contact stresses between two meshing teeth exceed the surface endurance strength of the material. It can be identified by darker patches or shallow indentations on the toothed surfaces. These irregularities cause friction, resulting in elevated levels of heat being generated. To avoid pitting failure, the effective load on the gear tooth must be less than the wear strength of the material. The wear strength is calculated from Buckingham's equation of wear strength [8] based on Hertz's theory [9] of contact stresses. In this method, the contact stresses are evaluated by modeling the gears as two cylinders pressed together.

The static load on the gear teeth is determined using the following equations:

$$P_t = 2\frac{M_t}{d_P}$$ **Equation 1**

$$M_t = \frac{60 \times 10^6 (KW)}{2\pi n_p}$$ **Equation 2**

where M_t is the torque transmitted between the transmission rod and the pinion gear. In addition to the static load, there exists the dynamic load. To account for this in the early stages, we approximate this using an appropriate velocity (C_v) and the service (C_s) factor. In the later stages, the dynamic load is precisely calculated using Buckingham's equation.

$$P_{eff} = \frac{C_s P_t}{C_v}$$ **Equation 3**

The velocity factor for well-machined teeth is given by:

$$C_v = \frac{5.6}{5.6 + \sqrt{v}}$$ **Equation 4**

The service factor is taken as 1.5 for medium shock gears.

The equation for dynamic load given by Earle Buckingham for bevel gears is as follows:

$$P_d = \frac{21v(Ceb + P_t)}{21v + \sqrt{(Ceb + P_t)}}$$ **Equation 5**

C = deformation factor (N/mm^2).
e = error between meshing teeth.
The effective load on the gear tooth is given by:

$$P_{eff} = C_s P_t + P_d$$ **Equation 6**

To avoid failure of the gear tooth due to bending:
$S_b \geq P_{eff}$ × factor of safety
To avoid failure of the gear tooth due to pitting:
$S_w \geq P_{eff}$ × factor of safety

The beam strength of bevel gears is the maximum value of tangential force at the large end of the gear that the tooth can transmit without undergoing a bending failure. It is given by:

$$S_b = mb\sigma_b Y[1 - {}^{b}/_{A_\circ}]$$ **Equation 7**

where [1 - b/A°] is the bevel factor, b is the face width, and A° is the cone distance.

Y is the tooth form factor and given by:

$$Y = \pi(0.154 - {}^{0.912}/_{T_{eq}})$$ **Equation 8**

The wear strength of bevel gears indicates the maximum value of tangential force at the large end that the tooth can transmit without pitting failure. It is given by:

$$S_w = \frac{bQD_pK}{\cos\gamma} \qquad \text{where, } \tan\gamma = \frac{z_p}{z_g}$$ **Equation 9**

The ratio factor Q is given by:

$$Q = \frac{2z_g'}{z_g' + z_p'} \qquad \text{where, } z' = \frac{z}{\cos\gamma}$$ **Equation 10**

Material constant K is given by:

$$K = \frac{\sigma_c^2 \sin\varphi \cos\varphi \, [\frac{1}{E_p} + \frac{1}{E_g}]}{1.4}$$ **Equation 11**

Where E is the Young's moduli of pinion and gear.

The factor of safety against tooth bending $F \cdot S_b$ is defined as the ratio of the beam strength and the effective tooth load:

$$F.S_b = \frac{S_b}{P_{eff}}$$ **Equation 12**

The factor of safety against pitting failure $F \cdot S_w$ is defined as the ratio of the wear strength and the effective tooth load:

$$F.S_w = \frac{S_w}{P_{eff}}$$ **Equation 13**

2.4 CALCULATION OF GEAR DIMENSIONS

Material 1 – SAE 8620
Material 2 – EN 36C

The results of the design calculation show the two differentials have similar gear modules. In other words, the gears are dimensionally identical. This has two important implications: (i) the differentials weigh the same due to similar densities of the materials, and (ii) identical sizes imply similar space taken up by the differential at the vehicle's undercarriage.

More importantly, the difference in factors of safety obtained from the geometrically identical differentials highlights the differences in performance offered by each material. EN 36C displays significantly greater safety factors, especially wear resistance than SAE 8620, as shown in Tables 1 and 2. This is credited to EN 36C's higher surface hardness — BHN 680 compared to BHN 550 of SAE 8620. EN 36C also displays higher safety factors in tooth bending as it maintains a tough core while maintaining high surface hardness.

3 FATIGUE ANALYSIS

We have completed the design of our gears based on the maximum torque delivered by the engine. However, we must recognize these components undergo numerous stress cycles over their lifetimes. Under cyclic stresses, fatigue is the primary cause of component failure when operating under the yield strength of the material and must be considered in the gear design. The study of fatigue failure is not an exact and absolute science, and it is often difficult to obtain precise results. Predicting fatigue failure involves approximation and reliance on statistical data provided by fatigue strength experiments.

Over a 10-year life of the vehicle, the gear teeth may undergo over 10^9 stress cycles. To avoid fatigue failure, we require an idea of the stresses operating in the gears and the fatigue strength of the steels. To precisely evaluate the stresses, we turn to FEA software. The differential is modeled in SolidWorks 2020, and its components and assembly are shown in Figure 1. The stresses are then analyzed by running a static study of the individual gear models in Ansys Workbench 2020R2. Appropriate fixtures and tooth load are applied to the gear models that accurately represent the dynamics of the differential during vehicle operation, as shown in Figure 4.

The stress study reveals that the maximum von mises stress is 236 MPa occurring in the pinion gear teeth, as shown in Figure 5. This is not surprising as more work is done per tooth as compared to the crown gear it is mated with. How does this cyclic stress compare to the fatigue strength of SAE 8620 and EN 36C? We first require the fatigue strength of the two steels. The fatigue strength is closely related to the material's toughness as a higher toughness reduces the opportunity for cracks to develop and grow. Several techniques for estimating the fatigue strength of steels have been developed based on experimental behavior. Rossel and Fatemi's method [10] correlates the fatigue strength with material hardness from technical data of 69 steels. The correlation provides a least-squares fit over the data with a correlation coefficient R^2 of 0.91. The correlation coefficient is a statistical measure of the strength of the relationship between the movement of two variables. The range of the values lies between -1 and +1. A value close to +1.0 indicates a strong positive correlation. The relation is given by:

$$S_{f'} = 1.34 \times \text{BHN} \quad \text{Equation 14}$$

This gives us a fatigue strength of 737 MPa for SAE 8620 and 911MPa for EN 36C. The Marin factors [11] are employed to account for the effects of surface condition, size, loading, temperature, and miscellaneous items to the theoretical fatigue strength $S_{f'}$.

$S_f = K_a K_b K_c K_d K_e K_f S_{f'}$

K_a = Surface condition modification factor.

K_b = Size modification factor.

K_a = Load modification factor.

K_a = Temperature modification factor.

K_a = Reliability factor.

K_a = Miscellaneous effects modification factor.

$S_{f'}$ = Rotary-beam test specimen endurance limit.

S_f = Endurance limit at the critical location of a machine part in the geometry and condition of use.

Applying the Marin equation to our design case, we get:

SAE 8620 S_f = 353 MPa

EN 36C S_f = 430 MPa

The next step in evaluating design safety is to evaluate the effective cyclic stress on the gear teeth. The fatigue strength data is experimentally derived for fully reversed stresses, but the stress cycles in our design range from 0 to a maximum value of 236 MPa. We shall use the midrange stresses σ_m and amplitude stresses σ_a to fully define the stress cycle in our design.

$$S_{f'} = 1.34 \times \text{BHN} \quad \text{Equation 15}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{236-0}{2} = 118 \text{ MPa} \quad \text{Equation 16}$$

Once we have the midrange and stress amplitudes, we employ the modified Goodman line to calculate the factor of safety in terms of fatigue.

The modified Goodman line is defined by the equation:

$$\frac{\sigma_a}{S_f} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \quad \text{Equation 17}$$

Here, n is the factor of safety against fatigue failure. The modified Goodman line gives us a factor of safety against fatigue of 2.38 for SAE 8620 and 2.87 for EN 36C.

SAE 8620 Parameters	Pinion Gear	Crown Gear	Spider Gear	Axle Gear
Teeth	12	75	20	20
Module	6	6	5	5
Pressure angle (deg)	20	20	20	20
Helix angle (deg)	0	0	0	0
Contact ratio	1	1	1	1
Diameter (mm)	72	450	100	100
Face Width (mm)	95.4	95.4	21.2	21.2
Factor of Safety bending	1.26	1.26	2.15	2.15
Factor of Safety pitting	1.84	1.84	2.41	2.41

Table 1: SAE 8620 differential parameters.

EN 36C Parameters	Pinion Gear	Crown Gear	Spider Gear	Axle Gear
Teeth	12	75	20	20
Module	6	6	5	5
Pressure angle (deg)	20	20	20	20
Helix angle (deg)	0	0	0	0
Contact ratio	1	1	1	1
Diameter (mm)	72	450	100	100
Face Width (mm)	95.4	95.4	21.2	21.2
Factor of Safety bending	1.47	1.47	2.50	2.50
Factor of Safety pitting	2.81	2.81	3.69	3.69

Table 2: EN 36C differential parameters.

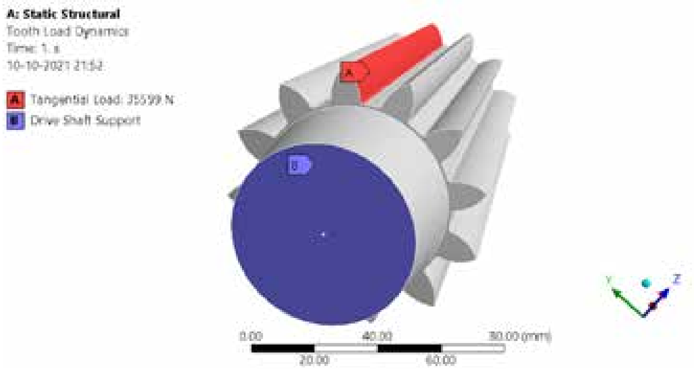

Figure 4: Fixtures and loads operating on the pinion gear.

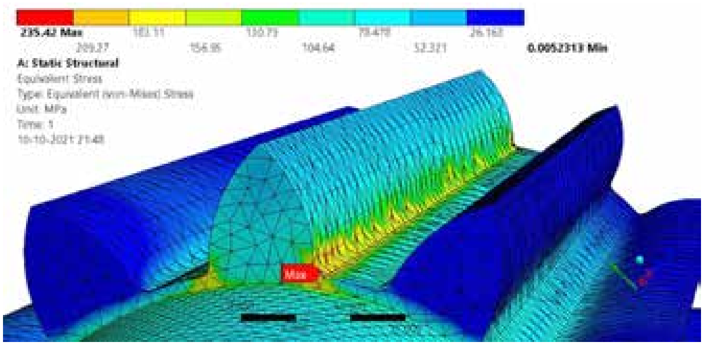

Figure 5: Equivalent (Von Mises) stress distribution of the pinion gear.

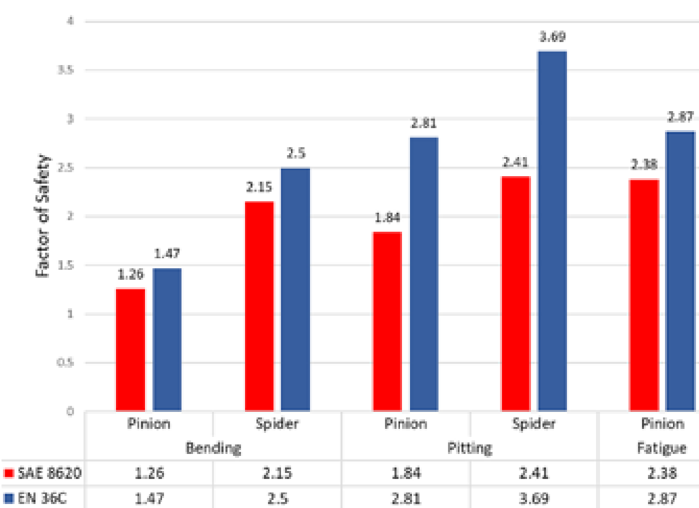

Figure 6: Comparison of factor of safety of SAE 8620 and EN 36C.

Description	SAE 8620	EN 36C
Raw material (rolled steel)	₹80/Kg	₹170/Kg
10 sets (prototyping)	133,000 INR per set	155,000 INR per set
10,000 sets (high volume)	29,900 INR per set	34,900 INR per set
Tooling cost (high volume)	6,500,000 INR	6,500,000 INR
Total Cost (high volume)	305,500,000 INR	355,500,000 INR

Table 3: Cost comparison.

4 COST ANALYSIS

The final stage of our comparative analysis is to estimate the cost of production of the differential gearboxes for SAE 8620 and EN 36C. The total cost of the differential gearbox for both materials is calculated for prototyping (10 units) and high-volume production (10,000 units). The data is presented in Table 3.

5 CONCLUSION

This article draws a comparison between popular two case hardening alloy steels, EN 36C and SAE 8620, used in gears. A differential gearbox was designed for the Ashok Leyland 2516 for two materials, SAE 8620 and EN 36C. Appropriate gear-design equations were used to design the differential, and these components were analyzed in terms of stresses and fatigue. The results show that EN 36C offers significantly better tooth strength, wear resistance, and fatigue properties. For identical pinion dimensions, EN 36C showcases an increase in factor of safety of 17 percent, 52 percent, and 20 percent in these respective properties. SAE 8620, on the other hand, shines in the pricing department, offering satisfactory properties at a 16 percent lower cost when considering high-volume production. This study provides a quantitative difference in the performance of the materials, which can assist engineers in making a well-informed decision regarding material choice. Specific to the vehicle considered in our study, Ashok Leyland 2516 is positioned as an entry-level tipper truck and operates in a highly competitive market. Hence, SAE 8620 is the preferred choice of material as it offers a balance between cost and performance. When the performance requirements justify the added cost, EN 36C is a strong candidate.

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