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16. Abstract

A summary is given for each of the thirty technical papers presented at the International Wheelsets Congress held in Tokyo, Japan, 20-23 October 1975. The papers were presented by international experts in rail research and covered a wide range of wheel/rail topics, including dynamic forces, material characteristics, stresses, failure mechanics, fatigue, maintenance and noise.

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Introduction

The International Wheelsets Congress convenes every three years. This was the fifth congress and the first to be held outside of Europe. Twenty-eight papers were presented at the congress with two additional ones being included in the proceedings.

There were sixty-three registered participants from non-Japanese countries and one-hundred thirty-one Japanese participants. The delegates were primarily from operating railways and suppliers.

Papers at previous meetings were primarily confined to wheelsets. However, in recent years the meeting has begun to take on a more general technical content. Discussions on metallurgy, stress measurement and metal behavior showed the increasing awareness of the need to consider the wheel and rail as a system. As might be expected, the theme of this congress in Japan was high speed operation and service life.

In order to produce a report of a useful length, this summary has concentrated on the results presented by each author, thereby eliminating much of the material related to background and technique. Although every effort has been made to accurately interpret the various authors' meanings, it is possible that another reader could understand the material in a different way. Therefore, for complete knowledge of the authors' work, the reader should refer to the original paper. Individual papers have in the past been available from the Engineering Societies Library, 345 East 47th Street, New York, New York, 10017. Page numbers for the English language version are given in parenthesis at the end of each summary. Copies of the extensive discussions of the papers will be available in August 1976, according to the secretariat.

Opening Remarks, H. Shima, Japan

The meeting was opened by Dr. H. Shima, the President of the 5th International Wheelsets Congress. He is also the President of the National Aerospace Development Agency in Japan and President of the Japan Railway Engineers' Association. In his opening remarks he gave a historical review of the Shinkansen or the high-speed line. There are now 268 trains being operated on this line per day, and they carry one-half million passengers daily. His major point was that the building of this line and the success of its operation demonstrates the usefulness of high utilization railway lines in passenger service. There are plans to extend the Shinkansen system.

Several comments were made relative to noise in the high-speed system. The Japanese, in almost all aspects of their society, are very conscious of noise, and the railway is no exception. Dr. Shima stated that the use of elastic material in wheels as a noise reduction technique is not looked upon very favorably in Japan right now because of their instability at high speeds. The initial interest was on preventing noise from entering a car, but now they are concerned about noise radiating to the surroundings. The need for maintaining good track quality absolutely requires the operation of rolling stock having the least detrimental effect upon the track. The need for reducing the axle load in achieving the high speeds was noted. As is exemplified by the design of the Shinkansen system, the Japanese do feel that the individually powered axles are more useful for their high speed system.

Concerning adhesion, Dr. Shima made the statement that the adhesion problem, which certainly does increase at greater speeds, is minimized by every axle being powered. Dispersion of motive power serves the purpose of decreasing the effect of slipping due to speed change or wetting of the rail in rain or snow and makes it possible to obtain higher rates of acceleration and deceleration.

The Japanese use solid wheels with disk brakes mounted on the plate of the wheel. They are interested in decreasing the weight of the wheel itself and, therefore, the unsprung weight of the vehicle. He stated a very great concern for maintaining a uniform wheel profile in service in order that the ride quality be maintained. The use of ultrasonics and other methods of inspection were stressed as a prevention of fatigue failure.

1. <u>The Prototype Electric Railcar Multiple Train for the Nation-wide Shinkansen Network and its Bogie</u>, H. Toyota and K. Kasai, Japan

The development of the new prototype car for use in the expansion of Shinkansen was discussed in this paper. The original system was opened in October 1964 and was extended from Tokyo to Hakata in March 1975. At the present time there are 2,224 electric railcars along this 1,069 kilometer long system. Their current plans are to make a nationwide Shinkansen system which will require a different railcar design. This is because the new cars will be operating in different climates, with different electrical power supplies and with different gradients, longer tunnels, and some shorter spacings between the stations.

Some of the comparative features of the new prototype car are listed in the report. There is a wider range of tare weights of the cars.

For the new cars it is 46-63 metric tons* as compared to 52.5-57.6 metric tons for the existing cars. Considerable amounts of light alloy structure are used in the body of the new cars. In order to decrease the unsprung weight, hollow axles are used. They were solid in the previous cars. Traction motors are larger in these cars and run at a high rpm. The main circuit is thyristor controlled for both powering and braking. These trains operate both in 50 Hz and 60 Hz regions. To accomodate this, they have dual electrical systems with dual frequency capability. Snow melters underneath the car protect the equipment during cold weather. The increased weight of the cars appears to be due to the increased power and the dual frequency requirement and various other improvements over the previous car.

On the development of the bogies they instituted several tests at their technical center on three kinds of bogies from the existing cars, the DT 9010, DT 9011 and the DT 9012. All performed satisfactorily, the only new information found being that there was a greater wheel load fluctuation with the new prototype cars. This was due to the increase in unsprung weight for these cars resulting from the greater power requirements, as mentioned previously. The adoption of the hollow wheel shaft of the DT 9012 bogie considerably reduced the fluctuation of the wheel loads.

In the further development, they had two types of bogies, one having a parallel Cardan drive system with a hollow driving shaft (DT 9013) and the other having a right angle Cardan drive system with a spring brake unit (DT 9014). The DT 9013 was found to be superior.

In order to further reduce the unsprung weight, they decreased

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* U. S. tons = 1.1 x metric tons

the wheel diameter from 1,000 mm on the previously existing cars to 980 mm. Other specifications, such as induction hardening of the axle in order to achieve greater fatigue strength, have been continued. These bogies also incorporate tread cleaners as do the earlier models.

5 references, 1 table, 10 figures, (v I, pp 1, 0-13)

2. <u>Speeds, Axle Loads and Wheel Diameters for Freight Vehicles</u>, C. O. Frederick, United Kingdom

Because of the well-known clearance problems on British railways, there is considerable interest in reducing the diameter of the wheels in order to increase the load capability. However, it is well-known that this has two deleterious effects on track, increasing the axle load, and, because of the smaller diameter wheels, increasing the Hertzian stresses. Therefore, British Railways established a working committee composed of representatives from mechanical and civil engineering departments to determine an actual optimal wheel diameter, considering maintenance cost per mile for a specified type of vehicle on a particular kind of track. The physical limits on the wheel size are clearances within the bogie, axle fit, requirements for transversing crossovers and things of this nature. The minimum wheel diameter currently permissible on British Railways is 689 mm.

The effect of impact of the wheel at a dipped joint is discussed in some detail. Three different types of loads are identified, P_o being the static axle load and P_1 being analogous to a hammer blow on the rail head. The latter principally affects the rail. P_2 is a force of longer duration, which is analogous to the force in the spring of a spring mass system when the spring is in compression. P_2 will affect the ballast and the formation beneath the ballast. All three will play some part in determining the rate of which the track will deteriorate. The author points out that for fatigue considerations in the bolt holes, the shear stresses at the "running on" bolt hole, due to P_1 and P_2, are in opposite directions so that to some extent, the stress range and, hence, the fatigue crack initiation period are dependent on $P_1 + P_2$. The author points out that this conclusion has not been verified. It is stated that P_1 and P_2 type contact stresses will be most important to shelling while still recognizing that the P_o contact stresses will be important since they exist on a more or less continuous basis. The effect of profiles upon the contact stresses is also discussed.

The working group used various expressions for track parameters such as stiffness of the ballast, the track mass and damping, and vehicle parameters such as weight and wheel diameters, etc., to establish a basis for their calculations of the potential damage for the track. In some cases, values of P_1 and P_2 would be within acceptable limits,but a combined value of $P_1 + P_2$ would be outside the limits. They decided that because of the larger potential for bolt hole damage due to the higher level of $P_1 + P_2$, in order to accept this there would have to be a corresponding reduction in damage due to wheel flats. This could be achieved by decreasing the allowable length of the wheel flat. Other parameters, such as the dip angle and the rail head profile, are very important and were used in their calculations. Concerning track stiffness, changes in track stiffness obviously would not affect P_1 contact stress. P_2, however, would be very greatly affected, and, for joints with greater dip, both P_1 and P_2 would be decreased.

The working party also estimated the minimum unsprung mass of wheels and axles that they could assume to be possible under best modern design. With this criteria they calculated track stresses. It was decided that tread braking for these small wheels would not be practical, and disk braking was recommended. The final discussion notes that their results are based on current British Rail practice, and since they do not experience shelling, there is nothing there that they can compare with. However, they do sight two operations on the European continent which are above the limits which they predict for shelling and where shelling, in fact, does occur. In another case where an operation is within the limits that they established, no shelling is experienced.

In their closing discussion, they state that shelling and wear of the wheel could also be lessened by using harder wheel materials. Similarly, the shelling and wear properties of the rail could be bettered by using harder rail steel. Citing data from British Rail, they note that rails generally are harder than the wheels. It is noted, however, that it would not be practical to use a rail harder than wheels in actual operation except on lines seeing only unit-train type service.
4 references, 2 tables, 14 figures. (v I, pp 2, 0-23)

3. Adhesion Coefficients between Wheel and Rail at Very High Speeds and Improvement of Adhesion by Applying Special Composite Sliding Blocks on Wheel Treads for Shinkansen Vehicles, S. Shirai and T. Ohyama, Japan.

The introduction of this paper outlines the importance of adhesion, particularly for operations such as the Shinkansen, which operate at very high speeds in territory where heavy snow and rain occurs quite often. In the original design of the Shinkansen system, adhesion was considered to be an important factor, and a formula was developed which gave adhesion as a function of velocity. The results have shown this formula to be reasonably correct. However, a sliding block which makes contact with the wheels is used to improve the adhesion condition of the wheel rail system. An anti-skid system that reestablishes adhesion, if the adhesion limit is exceeded, is also used.

A test car for measuring wheel slip at various speeds was developed by Japanese National Railways. This was discussed by the authors. As a result of their tests on snowy lines, they show that the adhesion coefficient at the point just before wheel slip occurs was considerably lower than the value expected from their formula. Also they found that upon slip, the sliding speed of the wheel increased very slowly rather than dramatically. In this situation the wheel slip lasted about two kilometers or for about 50 seconds. At the time of the writing of their paper, the Japanese Railways was performing tests at their Railway Technical Research Institute to determine adhesion characteristics at speeds up to 350 km/hr, some of the results of which are contained in this paper.

The range of adhesion coefficients measured in the tests using the test cars ranged from .04 to .07 on wet rail and fell to .02 on snowy rails. At the testing center, adhesion coefficients obtained were between .03 and .13 under wet conditions. It is shown that the distribution of the adhesion coefficient with speed does not agree very well with the formula used for the design of the Shinkansen, the adhesion coefficient measured either with the testing cars, testing train or in the laboratory being generally greater than those predicted by the formula used. Except at the very high speeds, above 200 km/hr, and particularly at 250 km/hr, the result appeared to be very good. It was also shown that the differences between adhesion coefficient and the sliding friction coefficient are very small in case of high speed operation on wet rail. They conclude from a plot of adhesion, running speed and acceleration force, that with several improvements in the circuitry of the system and in the handling of the train, adhesion can be improved to .45 at higher speeds. In this case the maximum train speed may be set at 350 km/hr.

Further tests at the technical center using the rolling machine showed that the relationship of the traction coefficient with the slip ratio was virtually unchanged above a slip ratio of .5 at a rolling speed of approximately 300 km/hr and a maximum contact pressure of 70 kg/mm^2. The adhesion coefficient in the presence of water lubrication was shown to be markedly influenced by surface roughness. For surfaces cleaned with sandpaper, the maximum traction coefficient tended to decrease very little with rolling speed. The decrement was negligible when compared to coefficient fluctuations caused by changes in atmosphere conditions such as temperature and humidity. The minimum adhesion coefficient obtained, even at speeds up to 300 km/hr for this case, was always 0.18. The relationship of slip ratio and the traction force in the region of microslip (i.e. creep) is defined by using Carter's theory. The parameters are the area of the contact surface, the material properties and the friction coefficient. The nondimensional traction force is plotted versus a nondimensional expression for the dry condition, i.e., the slip ratio, at a rolling speed of 300 km/hr and a contact pressure of 40 kg/mm^2. Their experimental data consistently falls below the results predicted by Carter's theory. This discrepancy is still under investigation by JNR.

Experimental work was also performed showing the change in the traction coefficients in the case of spindle oil lubrication. Obviously the traction coefficients were quite low for the low speeds in this case. However, they remained fairly constant with an increase in speed. At speeds above 300 km/hr, the disagreement with the predicted traction coefficient used in the Shinkansen was negligible for a contact pressure of 80 kg/mm^2. For a contact pressure of 55 kg/mm^2, the experimental values were substantially below the design adhesion coefficient for all speeds.

Regarding the test for improving the fundamental adhesion with the special composition block, the authors note that the values that they obtained may not correspond to results obtained in the field. They were attempting to present changes in the adhesion coefficient caused by the block. The effect of the cleaning block in the case of wet rail is to make the metal to metal contact areas as large as possible in order to prevent the effect of elasto-hydrodynamic lubrication at higher speeds. Existing composition cleaning blocks operate at a constant force of 50 kg upon the application of the train brakes. In addition to the obvious cleaning action of the block, during rainfall fine hard particles from the block stick to the tread surface. The function of these materials is to somewhat tear the water film and constitute a number of bridges between the wheel and the rail. The adhesion force is increased by the shear strength of the particles.

By taking data showing the frequency of occurrence of rainfall on Shinkansen and the simultaneous occurrence of wheel damage or wheel flats as the result of slip, they found that there was a very definite correlation between rainfall and wheel damage. Using the cleaning blocks should substantially decrease the number of wheel flats in this situation. This was found to be true in actual service. An additional benefit of the composition blocks is less wheel noise as a result of the smoother running surface.

2 references, 20 figures, (v I, pp 3, 0-13)

4. Steel Wheel Noise and Its Reduction, N. Oda, A. Nishio and S. Nishimura, Japan

In comparing the interior and exterior noise of a rail vehicle, it is initially noted that the exterior noise is not necessarily reduced by a reduction of the interior noise. The wheel rail noise is observed to be predominant in high speed operation although the track structure can significantly affect the peak noise levels. Other work on wheel rail noise is discussed. The purpose of this paper is to show the results of tests on a rolling wheel-testing machine in order to study the rolling contact noise and the possibility of noise reduction by using a resilient wheel is discussed.

Two kinds of slip noise can be generated by the use of the wheel-testing machine. Slip can be generated by skewing the axis of the wheel-set in relation to the axis of the rail wheels, and tangential slip can be generated by giving one of the rail wheels a slight braking torque and the other a driving torque so that the two rail wheels are traveling at different velocities. In the latter case, the axis of the wheelset and the axis of the rail wheels are maintained parallel.

The wheel used on Shinkansen is a straight plate type, 905 mm in diameter. Disk brakes attach to the plate.

The natural vibration and noise characteristics of the wheel were obtained before performing the tests by shaking it with a small electro-dynamic shaker. Clear resonances were observed for transverse excitation at frequencies of 169 and 263, 386 and 986 and 3,637 Hz. Frequencies of 2023 and 3615 Hz were seen for radial excitation. The lowest, or the 169 Hz vibration, was produced by the coupled wheel and axle system acting as a beam with two masses, the masses being the wheels. Higher order modes of transverse vibration were also produced. These would be somewhat similar to hoop modes except that they were also vibrating out of the plane of the wheel. The 263 Hz vibration was also produced by the coupled wheel-axle set rather than by vibrations of the wheel itself. Only the 386, 986 and 3637 Hz modes were produced by out-of-plane vibrations of the wheel itself. Of the radial vibrations at the higher frequencies, the 2023 Hz motion is primarily radial. The 3615 Hz vibration also produces a vibration out of the plane of the wheel, which intensifies the noise radiation in this case.

For the rolling noise test, the wheelset was placed upon the rail wheels without a skew angle, and the rail wheels were maintained at the same velocity so that tangential slip was reduced to a minimum. Peaks were found to occur at 50, 100, 150 and 200 km/hr. The low frequency always predominated at low speed and peaked at around 250 Hz. This sound level was high and relatively constant throughout the test speed range. This noise peak was found to be coincident with the transverse natural vibration, as discussed previously. Components in the range of 1000 to 3000 Hz increased with speed, and a peak appeared at about 2000 Hz at a speed of 200 km/hr. It was concluded that this 2000 Hz noise component resulted from the wheel vibration in the radial, fundamental mode.

Tests show that changing the tangential slip of the wheelset did not appreciably affect the noise level at high speeds. When the braking torque was increased to the point that the creep limit was exceeded, a torsional vibration of the wheelset was produced and noise was generated. [It should be noted here that in the arrangement for transverse slip, the creep force is pulling outward].* The experiments on transverse slip were conducted at a speed of 25 km/hr on the wheel-testing machine. The skew angle between axes of the wheelset and the wheel-testing machine was increased with very small increments. At very small skew angles, only the 250 Hz noise component showed any increase. At a increased skew angle, the 1000 Hz component appeared,and the sound pressure level increased considerably. This test is consistent with the phenomenon of wheel screech produced on a curve.

Tests on resilient wheels were made using only the SAB resilient wheel. In shaker test this wheel also showed primary vibrations at the 140 and 850 Hz locations. However, the pressure levels were lower in comparison to the steel wheel. On the rolling noise tests of the resilient wheel, the results show levels 5-7 db lower than for the steel wheel.

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* Reviewer's Note

These tests were conducted at speeds of 50, 100, 150 and 200 km/hr, and peaks were found at 350, 1000, and 2000 Hz. Their levels were low when compared to the steel wheel,resulting in a lower overall noise level for the resilient wheel. The damping component in this wheel acts to reduce the amount of unsprung weight. These tests indicate that this reduction in unsprung weight and the wheel damping itself are effective in reducing wheel noise.

2 references, 2 tables, 14 figures, (v I, pp 4, 0-11)

5. <u>Dynamic Wheel Loads of Rolling Stock at Very High Speeds</u>, H. Toyota, K. Kasai, S. Tanaka, M. Kobashi, and K. Terada, Japan

This paper discusses the analytical and experimental techniques used to predict the dynamic wheel loads of rolling stock for the new 250 km/hr operations on the Shinkansen. It was determined that the dynamic wheel loads at 250 km/hr should not exceed those currently existing at 200 km/hr.

A single-degree-of-freedom analytical model for the wheel load fluctuations was developed by the authors,taking into consideration the stiffnesses and masses of the system. In the model, they considered the form of the rail at an impact point to be given by an equation where the path of the wheel entering a dipped area is a function of the running speed, a trough gradient (i.e., profile) and the static force on the wheel rim. The resulting characteristic equation considered the wheel load fluctuation, the unsprung mass per wheel, the track equivalent mass, the track equivalent spring constant, and the upward displacement of the unsprung mass as it entered the dipped area.

Initially they considered the track spring constant to be linear. In order to check their model, they used measured and calculated data for an existing car and a 951 type test car. In this case, the existing car had a lower unsprung weight than the 951 type test car. This resulted in a coefficient of wheel load fluctuations (CWLF) ratio of 1.67 for the 951 type car as compared to the existing car. The ratio calculated with their model was 1.22 which they considered to be in great disagreement. Later, a nonlinear track spring constant was proposed which yielded very good agreement for a large number of speeds.

They used this model to make projections for wheel load fluctuations for higher speeds. It was shown that both relatively soft and hard locations on the track could result in comparatively high wheel load fluctuations. The authors point out that this is of great interest because of the increased likelihood of track breakage, rail damage and damage to concrete sleepers with the higher force. As a result of the authors' work, the DT 9012 type bogie with a parallel Cardan system was modified slightly so that its unsprung weight was 1.15 tons. This truck, now called the DT 9013, showed that the wheel load fluctuations were rather great at speeds in the neighborhood of 220 km over relatively poor track, but they were very acceptable at speeds up to 250 km/hr over good track.

4 references, 1 table, 12 figures, (v I, pp 5, 0-10)

6. <u>On the Influence of Resilient Wheels on Adhesion, Dynamic Track-Wheel Forces and Running Gear Environment at High Speeds</u>, Bo G. Cavell, Sweden

The effect of resilient wheels on the adhesion, dynamic track forces and running gear environment at very high speeds is discussed in this paper. An analytical model having 5 degrees of freedom was used. The car body, the bogie frame, the axle, wheel rim and rail, and the track are included. Experimental data obtained from tests on British Railways is also presented. The analytical model is shown to correctly predict force levels resulting from track bed response. These forces, it is noted, can, in fact, cause a worsening of the top rail irregularities which in turn can give higher impact and response forces causing further deterioration of the ballast. The sleepers are shown to cause significant cyclic input to the wheels. It is shown both theoretically and practically that a general reduction in vertical peak wheelset accelerations of some 45% can be achieved with the use of resilient wheels. In the case of random input, it can be theoretically shown that there is a close correlation between the vertical wheelset acceleration and the dynamic increment of contact force of the wheel and rail. This was confirmed on the test of British Rail at a speed of 160 km/hr. A power spectral density plot, up to 100 Hz, at the speed of 160 km/hr, showed several distinguishable frequencies. The first, at 5 Hz, corresponds to the natural frequency bouncing of the bogie spring mass. Ten (10) Hz corresponds to the rotational frequency of the wheel. At 20 Hz, a resonant phenomena in the locomotive equipped with resilient wheels is due mainly to the mass of the wheelset and the traction motor between the axle box springs and the resilience in the resilient wheel. A similar resonance occurs in the case of the solid wheel at about 35 Hz. The rms values for the resilient wheel, however, yield an acceleration of 1.7 m/s^2 as compared to 2.8 m/s^2 for the solid wheel. Further, at 58 Hz the cyclic input of the sleepers is shown. Again, the resilient wheel demonstrates a significant reduction in the peak accelerations.

Where the previous discussion showed that improved acceleration levels acting upon the wheel and axle mounted equipment would result from the use of resilient wheels, it is also shown that an improved acceleration environment in the vehicle itself also occurs. In the case of the wheel-sets' mounted equipment, the improved environment of the components would give better fatigue endurance. The introduction of resilient wheels greatly increases the number of equations of motion and creep required to completely describe the response of the system. Tests on British Rail, however, have shown that this increased complexity does not result in any severe problems of lateral stability owing to the introduction of resilence in the wheels. Considering the effects of the resilient wheels on adhesion, it is suggested that the decrease in fluctuation of wheel forces at the adhesion point, as a result of the use of resilient wheels, will decrease the probability of slip occurring because of low tractive forces during impact.

Considering service life, a condition on Swedish State Railways is described where wheel failures had occurred in an iron ore train equipped with conventional steel tyred wheels when operated over frozen ballast. The use of the resilient wheels reduced the impact forces, reduced the occurrence of wheel slip and resulted in less damage to the wheels owing to the stiff track. Conventional wheels required retruing on the average of every 40,000 to 100,000 km, and, in some cases, as often as every 20,000 km. New tyres were required every 300,000 km,and the locomotive overhaul period was fixed at 450,000 km. In the case of the resilient wheels on this same locomotive, the tyre truing period was extended to 200,000 km,and the locomotive shopping period was increased to 750,000 km.

This benefit was over and above the benefit achieved by the general lower noise levels in the locomotive itself.

6 references, 2 tables, 19 figures, (v I, pp 6, 0-12)

7. Factors Influencing the Frictional and Wear Behavior of the Wheel/Rail System, H. Krause, and J. Scholten, West Germany**

There are some characteristics of railway operations which are affected primarily by design while there are others affected primarily by the material properties of the wheel and rail system. Factors which are considered to be of primary importance in an investigation of frictional and wear behavior are, for example, the axle load, wheel diameter, wheel and rail profile, traveling speed, slip, slip speed, the atmospheric humidity and the mechanical and chemical characteristics of the material. Other factors such as the wheelset design, the control system, the roadbed specifications, the number of the idling axles, temperatures and the manufacturing processes for the materials are considered to be of secondary importance in friction and wear behavior.

Tests were performed using a Bugarcic rolling-friction test machine to investigate the effect of each of these primary parameters upon the frictional and wear behavior of a rail system. In the initial test series, the wheel steel was M-80, and the rail steel was VA for the coefficient of friction, wear, hardness and roughness tests. VA is a wear resistant steel of UIC quality A with a strength of 900 N/mm^2, and M-80 is

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** This paper appeared in the original Proceedings in German. However, an English translation is now available. More detail is given in the review for this paper than for the others because of the original German text.

a common driving wheel steel with a strength of 800 N/mm^2. For texture and internal stress studies, both the wheel and rail steels were Ck 45 (C .44, Mn .66, S .029, V .02, Si .22, P .002, Cr .15)*.

Rolling friction showed an initial increase to a maximum coefficient of friction (μ = 0.43) after which a uniform and slightly lower level (μ = 0.35) was reached,which remained throughout the tests. The total traveling distance of the test was 35 km. Wear showed a linear increase with the distance traveled, the wear of the rail being greater than the wear of the wheel, which was a characteristic of the material pairing. An immediate increase tangential to the vertical axis occurs in the hardness of both the rail and the wheel material after which the values asymtotically approach ultimates. The ultimate wheel material hardness is lower than that of the rail material. Similarly, the roughness also increases immediately and then assumes an almost level asymtotic approach to an ultimate value. In this case, however, the wheel material has a greater index of roughness than does the rail material. The roughness index is greater for the rougher material. A texture parameter is defined as a function of the reflection of a monochromatic beam at the initial condition and at the ultimate. This parameter was seen to increase initially and reach an ultimate texture at about 17 km, after which the texture parameter decreased. This indicates a crystal reorientation during the cold working of the material. The residual stress also increased immediately and attained a final value near to 60 to 90% of the yield point of the material.

In the full tests which followed, several of the previously

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* Reviewer's comment

mentioned primary factors were combined into one variable parameter in order to minimize the number of experiments. For example, axle load, wheel diameter, wheel profile, and rail profile were considered together to be affecting the surface pressure, which could be varied by changing the applied normal force. Therefore, wear of both the wheel material and braking material was investigated as a result of increased normal force. The wear of both materials was seen to increase significantly with increased normal force. Similarly, the traveling speed, slip speed and sliding speed are related. Therefore, one test was performed to investigate these parameters. The wear was seen to increase in both materials with increased slip. Also, the coefficient of friction increased with increased slip. Increased speed was shown to cause a slight increase in the wear of the rail and a very slight increase in the wear of the wheel. The result was a combined total increase of the wear of the entire system. The coefficient of friction, at a slip of 0.3%, was seen to decrease with increased speed, and a similar decrease in the coefficient or friction at a slip of 0.7% was also seen for increased speed. The coefficient of friction at both speeds was greater in the case of 0.7% slip as compared to 0.3% slip. The humidity content was shown to affect both the wear and the coefficient of friction. The wear of both the rail steel and wheel steel decreased with increasing humidity content, the rail steel wear always being higher than that of the wheel steel. The coefficient of friction also showed a tendency to decrease at a fixed percentage of slip with an increase of humidity content.

In comparing these results with those commonly obtained in railway practice, it was noted that the increase in wear with normal force is well-known. However, the increase in the coefficient of friction with normal force, as their data showed, was not as expected. This was felt to be a result of the particular material pairing. Other data indicated that for several material pairings, a maximum occurred at a particular contact pressure after which the coefficient of friction decreased. The increase in the coefficient of friction with slip and the increase in wear at higher speeds was also expected. Although no comparable values were found showing the relationship of wear and atmospheric humidity, the change in coefficient of friction is in accordance with practical experience.

In order to investigate the combined effect of the various wheel and rail materials on wear, tests were conducted where either the rail material or the wheel material would be changed, with the other held constant. The rail materials used were SN rail steel; with a strength of 700 N/mm^2; VA wear resistant rail steel; and B2, a special steel of a chrome manganese base having a minimum strength of 1,100 N/mm^2. With the same wheel material, M-80, this test showed that the ranking, in descending order, of rail wear amounts was VA, SN and B2. It was not expected that the SN steel would show better wear characteristics than the VA steel. Again with the same wheel material, the ranking of rail steels causing the greatest to the least amounts of wheel wear was B2, VA and SN. For tests using VA rail steel and wheel materials BV1, which has a minimum tensile strength of 600 N/mm^2; M-80; and VBV, which is a heat treated BV2 steel having a strength of 900 N/mm^2; the wear of the rail was greatest for the M-80 wheel steel, lesser for the VBV wheel steel and least by a significant amount for the BV1. In this test, wheel wear showed a different order with the VBV material showing the greatest wear, followed in order by M-80 and BV1. It was noted that in this case the lower strength steels appear to give better wear characteristics than observed in actual practice. This may be because, in actual practice, these steels fail as a result of the higher stress levels rather than because of wear failure.

In the study of the effect of the various material properties, it was found that the higher strength wheel steel, when paired with wear-resistant rail steel, resulted in more wear for both the wheel and rail when compared to identical but non-heat-treated steel having a lower tensile strength. It was also showed that steels having different chemical composition, while at the same time having uniform tensile strength, do not exhibit the same resistance to wear under rolling stresses. Also, the use of higher strength pearlitic rail steel, B2, when paired with uniform wheel material, reduced the rail wear to less than one-third of that when paired with materials of equal strength. The wheel wear, on the other hand, increases. It was again found that the use of a higher strength wheel steel, with the same rail steel, clearly increases the wear of the wheel and the rail. But, if rail steel of the same high tensile strength is used, the rail wear is again reduced. If a steel with a bainitic grain is placed in the pairing with the pearlitic wheel steel, the result can be even further improved.

The frictional behavior tests showed insignificant changes in friction for all steel variations except for those having different chemical composition and uniform tensile strength. In the latter case, the frictional behavior somewhat improved for wheels having martensitic and bainitic secondary grain, as compared to pearlitic steels.

With the goal in mind of decreasing the unsprung weight, the use of titanium materials for wheel treads was investigated. Compared to normal steel pairings of M-80 and VA, the rail wear was reduced by a factor of 8 to 10 by the use of the titanium wheel material. The wheel wear remained almost the same. Coefficients of friction at a slip of 0.3% were less than those of the steel pairings, while at a slip of 0.7%, the deviations were minimal. It was not felt that this indicated a disadvantage for titanium. It was further noted that at higher slip rates, i.e., above 0.6%, the coefficient of friction of the titanium/VA steel pair exceeds that of the M-80/VA steel pair.

In summary, it was felt that they have shown that the frictional and wear behavior must be considered to be properties of the system and not properties of the component materials alone. If one quantity in a rolling friction system is changed, the effect on the whole system must be considered. Moreover, it was shown that in a frictional system the strength properties of the materials serve more to accommodate load stresses than to attain certain frictional and wear behavior. Wear of the components can be equally divided between each partner or can be concentrated in just one of the components.

In a private communication with this reviewer, the authors stated that in their texture experiments on rollers of cubic-centered iron material, the crystalographic axis was proved to be parallel to the direction of rolling /313/. The preferred crystalographic plane /135/ was shown to be parallel to the rolling plane. They also stated that while they proved that cold tempering of the metal surface of the rollers did take place due to friction, it was impossible to make a quantitative statement concerning the percentage of cold work during this process.

32 references, 2 tables, 25 figures, (v I, pp 7, 25-48)

8. <u>In-Service Behavior of Railroad Wheels, Metallurgical and Physical Hypothesis Relating of Various Defects</u>, J. Leclerc, C. Camut, B. Dupuis and J. Sebbah, France.

Initially described in this paper are the phenomena of stresses, railway operations, failures, etc. for wheels, the theory of elasticity, fracture mechanics, stress intensity factor, crack initiation and general fatigue theory. The phenomenon of cracking is considered to occur in three stages. The first is a stage of very slow propagation, the second a stage of slow propagation and the third a stage of accelerating propagation leading to brittle fracture. Discussion on the mechanics of wheel shelling under load is also included. Some discussion, with theoretical development included, is also given on the following topics: skidding, the origin of thermal defects, thermal shocks with and without transformation, the heating and quenching of the tread surface during thermal shock, the structural evolution during braking and deformations caused by phase changes in the wheel. Some emphasis is given to the formation of martensite and the martensite temperature as a function of alloying. Several formulas derived from other authors are listed and discussed.

In summary, the authors conclude that increased hardness of wheels enhances the wear characteristics and increases the resistance to shelling. Increased thermal conductivity increases the resistance to thermal shock without transformations. The quenchability of the material decreases the resistance to skidding. An increase in the martensitic temperature increases the resistance to thermal shock with transformation. Increased carbon content increases the resistance to wear and shelling and decreases the resistance to skidding and thermal shock with transformations. Increased manganese content increases the resistance to wear and shelling and decreases the resistance to skidding and thermal shock with transformation. Other alloys, in general, increase the resistances to wear and shelling and decrease the resistance to skidding and thermal shock with and without transformations. It is suggested that quenching and tempering are beneficial effects in all cases except for skidding.

16 references, 2 tables, 7 figures, (v I, pp 8, 17-39)

9. The Development of an Austenitic Manganese Steel Wheel, E. Jones, G. Burbeck and J. B. Lee, United Kingdom.

The authors suggest that the development of an austenitic steel which does not undergo structural transformations upon heating and cooling might provide superior resistance to thermal cracking and spalling in wheels. In order to investigate this possibility, British Railways manufactured a small number of 14% manganese steel wheels, the material being similar to that currently used for rail points and crossings. In general there were several difficulties associated with the manufacture and machining of these wheels. Moreover, in a falling weight test it was shown that the hub of the wheel was displaced from the rim considerably more than was normal. Since the primary emphasis of this paper is on the crack propagation characteristics, it should be pointed out that during brake dynamometer testing, although numerous small transverse cracks were seen to develop on the tread of the wheel, they would be subsequently rubbed out. After 290 dynamometer stops at 97 km/hr, the cracks became numerous, but they still remained only two millimeters in length. The wheel showed very poor wear characteristics although it did not tend to propagate thermal cracks to any great extent. It was observed that if the wheel was in actual service, the rolling contact with the rail would, in effect, work-harden the surface, and this could increase the wear characteristics of the wheel.

Tests were also performed on the dynamometer machine with stops from 145 km/hr. Considerable heat build-up and red banding on the wheel occurred during braking applications. The hot spots which appeared at 97 km/hr were shown to be much larger than those that appeared at 145 km/hr. After 100 stops, the largest cracks present were still found to be only 2 mm long. The rate of material wear, however, was considerably greater in the 145 km/hr tests. In summary, the thermal damage shown by the wheel after brake testing was confined to small cracks at the root of the flange, small cracks toward the front face of the wheel, hot spots from the heating which resulted in elongated spots on the tread of the wheel, and considerable material removal by wear.

It was clearly felt that there was some accomplishment in being able to produce these wheels in view of the difficulty that they had. Metallurgical examinations and mechanical testing have shown that the steel has properties equivalent to British Rail Class B and UIC steels but with greatly enhanced toughness. Static loading tests indicate that the wheel behaved in a manner similar to that of the normal carbon steel wheel of the same shape. Therefore, it was felt that there was no doubt as to the safety of these wheels in service from this aspect.

3 tables, 16 figures, (v I, pp 9, 0-21)

10. <u>Development of A Quenched and Tempered Solid Wheel and Its Evaluation by Means of Drag and Stop Tests</u>, P. Brozzo, L. Callegari, L. Brazzoduro, R. De Martini and C. Bianchi, Italy.

The effect and desired effect of fracture toughness, residual stresses, thermal cracking, wear, hardness and wheel design on the behavior of the wheels in service, as discussed in this paper, is a continuation of work reported at previous International Wheelsets Congresses. For these tests, twenty 100 kg heats of induction melted and "poured under vacumn" steel have been made with two carbon levels, .25 and .35%. Other elements ranged as follows: Si 0.4 - 1.0%, Mn 0.3 - 1.0%, Cr 1.0 - 2.0%, Mo 0.3 - .6%, Ni 0 - 3%, V 0 - .18%, B 0 - .003%. The small ingots were hot-forged into bars 60 x 80 mm in cross section, and they were quenched and tempered aiming at a proof stress of 900 to 1,000 N/mm^2 for the vanadium steels and 700 - 800 N/mm^2 for the vanadium free steels. They were subjected to conventional tensile tests as well as impact and hardenability tests. In addition, they were subjected to tests to determine K_{1D} and COD at $20^{\circ}C$ and to thermal cracking tests. All of the samples met the physical specifications, and, therefore, any decision as to the choice of the material must be based on the thermal cracking tests.

The thermal cracking of the various samples was investigated using a test rig similar to that used in the study of steels for cold rolling work rolls. Also, a formula for calculating the thermal cracking index C was presented. This is a function of the length and the depth of a single crack, the number of observed cracks, the area of contact between the sample and the wheel, and the number of observations. A lower value for the thermal cracking index indicates minimum susceptibility to thermal cracking. Preliminary tests showed a VT-1, fine grained, aluminum refined steel to exhibit the minimum thermal cracking index. Further, only the steel composition characterized by low carbon content, i.e., up to about .2%; high silicon, about 1%; chromium not to exceed 1.3%; manganese up to about 1%; nickle and boron absent; vanadium up to 0.18%; molybdenum up to .5%; exhibited 0 value for cracking index. Only vanadium free steels were considered for further work. As a result of the thermal cracking studies the following basic chemical composition was assumed as to be appropriate: carbon .2 to .24%, manganese .9 to 1.0%, silicon .8 to 1.0%, chromium 1.0 to 1.2% and molybdenum .3 to .4%.

Four sets of trial wheels were manufactured with chemical composition within this range. These wheels were subjected to both drag braking and the stop braking type tests. After the drag tests, the wheels were subjected to residual stress measurements. The results of the tests showed that for the normalized carbon steel wheel exhibiting original compressive stresses in the rim of about 120 N/mm^2, these almost vanished in the test. In the quenched and tempered alloy steel wheel, these original compressive stresses were substantially retained, since only a decrease from 350 N/mm^2 to 200-300 N/mm^2 was observed. The authors conclude, therefore, that the higher the proof stress in the wheel, the greater the tendency to retain the initial compressive state of stress in the rim.

Thermal cracking tests conducted under three conditions of stop tests, namely, those for standard braking procedure for passenger cars, and emergency stopping procedure for passenger cars, and electric rail car braking procedures, yielded somewhat inconclusive results in that the thermal cracking observed was inconsistent. The authors attributed this to problems caused by the uneven contact of the brake shoe to the wheel which may be the result of no smoothing or wearing effect because of the absence of a rail. A marked difference in the wear tests was exhibited by the two grades of steel at the lower speed level. The quenched and tempered steel exhibited a much better behavior. At the upper speed level, however, the two grades behaved almost the same as far as wear is concerned.

In conclusion, the authors feel that their tests confirmed the good quality of the VT1 wheel as far as thermal cracking resistance is concerned. On the other hand, they emphasize that further improvement must be made in the fracture toughness of the VT1 wheel. Their criteria for fracture toughness is that the wheel material must be capable of tolerating a flaw of a depth equal to the full rim thickness without leading to brittle, catastrophic failure.

8 references, 8 tables, 7 figures, (v I, pp 10, 0-13)

11. High-Toughness Steel for Railroad Solid Wheels, T. Kigawa, R. Isomura, Y. Tanaka and K. Tokimasa, Japan

High carbon steel (0.60 to 0.75% carbon, STY 80) with high wear resistance and strength has been used in Japan as a wheel material, of which resistance to wear, shelling, brittle type rim fracture and so on has been studied in Japan since 1930.

Recently it is required to develop high-toughness wheel materials with as high strength as STY 80. Especially, in service at a cold climate, high fracture toughness is needed to avoid the occurrence of brittle type rim fracture from thermal cracks on the tread of wheels.

One of the authors studied the residual stress in the rim of

STY 80 solid wheels due to ontread drag braking and their effect on the brittle type rim fracture. According to the authors and coworker's results, the brittle type rim fracture in a railroad solid wheel can be explained using fracture mechanics theory and, as discussed by Davies at the Sheffield Conference, can be intimately related to the following three factors,

(a) the size of a crack on the tread, c

(b) the level of a residual tensile stress developed in the rim, σ_R

(c) the fracture toughness of a wheel material, K_{IC}.

The following relationship is probably established between these factors

$$\eta \sigma_R \sqrt{\pi c} = K_{IC}$$

where η is a constant associated with the shape of a crack and the distribution of the residual stress in the rim.

Thus, it is the most effective procedure for preventing the wheel fracture to raise the fracture toughness of the wheel material.

The present study is associated with the development of high-toughness wheel material which has higher fracture toughness than STY 80. It was found that 0.21% V steel exhibits good properties in V-notched Charpy impact test and ASTM WOL test. And the results of the dynamometer test and the service test conducted at Sumitomo Metal Industries and Japanese National Railways revealed that the solid wheels made of 0.21% V steel have better properties than those of STY 80, in particular resistance to brittle type rim fracture.

(Authors' abstract)

15 references, 7 tables, 7 figures, (v I, pp 11, 0-10)

12. <u>Residual Stresses in Straight Webbed Monoblocked Wheels</u>, A. Babb, B. Sweeney, and G. Birkbeck, United Kingdom

This paper was a progress report on an investigation by the authors of the combined effect of dynamic stresses in wheels arising from the surface loading which are superimposed upon the residual stresses as manufactured in the new wheel. The primary emphasis is on the latter. The residual stresses which arise from manufacturing are more difficult to determine than the dynamic stresses arising from service. Most of the wheels have been subjected to some form of tempering either after oil-quenching or rim-spraying with water. A subsidiary investigation was therefore carried out in order to determine the effects of tempering upon the residual stresses. Laboratory experiments were conducted to determine the stress relief obtained on bars of tyre steel mechanically stressed to represent residual stresses and then tempered at different temperatures. The pattern of residual stresses in a wheel can be looked upon as a result of internal stresses and surface stresses, all of which must balance against each other. In one wheel of this test, chordal grooves of varying depths were cut in the wheel rim and the residual stresses measured at the bottom of the grooves by trepanning. A second wheel was turned down in successive stages to represent reprofiling and the rim residual stresses measured at each stage. It is planned to measure stresses in new wheels and then follow the service of these wheels by remeasuring each time the wheel is reprofiled.

In the investigation of surface stresses in two rim-sprayed and tempered wheels, wheels numbered 1 and 2, the stresses were measured at nine positions along each of three radial lines spaced at 120 degrees around a 836 mm diameter straight webbed Freight Liner type wheel machined, rim-sprayed and tempered at 610°C. The tread surface was found to be in compression, the values varying from -79 to -52 N/mm^2. Radial stresses at all positions on the web were tensile, ranging up to 73 N/mm^2. The bicycle wheel effect was predominate in all of the types of wheels examined other than those oil-quenched and tempered. The bicycle wheel pattern is such that the tension in the spokes of the wheel produce compression in the rim. With the rim-sprayed wheels this effect was even accentuated. Rim-spraying produced residual stresses in the rim about equal to the .2% proof stress. In so-called self-tempered wheels, the reduced temperature gradient, between the rim and the boss, produces somewhat lower stresses. Subsequent furnace tempering of the fully rim-sprayed wheels reduced the overall stress index by a factor of five. In oil-quenched and tempered wheels the rapid quenching of the whole wheel caused the inside versus the outside mechanisms to predominate, so that after tempering, these wheels tended to show compressive residual stresses over all surfaces. It is pointed out that one of the supposed advantages of rim-sprayed wheels is that the compressive residual stresses in the tread surface resist fatigue cracks from shelling or thermal cracks in the tread. It may also be that the higher compressive residual stresses within the body of the rim would also resist crack propagation. These stresses are obtained, however, at the expense of allowing tensile radial stresses in the web.

In the investigation of tread braked wheels, straight webbed wheels with no web offset were used. The stresses from radial mechanical loads are low compared to the large mechanically induced stresses arising from lateral or flange loads. In addition to the mechanical stresses, there are stresses arising from thermal loading and, in this case, the conditions for a full-stop brake and a drag brake are considered. The thermal stress acts essentially as a change in mean stress. For conditions as proposed by previous British Rail investigators, the results show drag braking stresses of +450 N/mm^2 radial at the flange side of the hub/web transition and +400 N/mm^2 radial at the equivalent outer position. The respective results for stop braking are +320 N/mm^2 and +300 N/mm^2 radial.

A fatigue analysis of the wheels showed that with the exceptions of the rim-sprayed but not tempered wheels and the rim-sprayed and self-tempered wheels, all others would have adequate fatigue life under full-stop brake conditions. The greatest margin of safety exists in oil-quenched and tempered wheels closely followed by rim-sprayed and tempered wheels. Under drag braking conditions the rim-sprayed and self-tempered wheels are again inadequate while the normalized wheels are marginally acceptable. Again the oil-quenched and tempered wheels show the best performance, closely followed by rim-sprayed and tempered wheels. The thermal stresses shown at the tread are all in circumferential compression and, when combined with the residual circumferential compressive stresses, can result in a compressive yield in some portion of the tread area of the rim upon cooling. This will result in tensile circumferential stresses in the rim which will tend to reduce the radial stresses in the hub/web area giving an increased margin between the stress range in service and the limiting stress range for failure of the wheel. It was established that the critical crack size for failure was largest in the oil-quenched and tempered wheels and in the wheels rim-sprayed and tempered at 610°C (approximately 4 mm under drag braking and 6 mm for stop braking).

Disk braked wheels were also investigated. The analysis showed, taking into account the reduction in limiting stress range due to the inner ring of bolt holes for the disk brake mount, that oil-quenched and tempered wheels offer the greatest safety margin. As in the tread braking case, they are closely followed by the rim-sprayed and tempered wheels.

The effect of a repeatedly applied rolling load on the wheel tread will be to produce a residual compressive stress in the rolling zone. In general, however, this stress pattern will occupy a small volume of the rim material and will be accomodated largely by local redistribution of stress in the tread area. Its effect on the hub/web transition stresses will be small and,therefore,can be neglected in the fatigue analysis detailed in this paper.

4 references, 2 tables, 14 figures, (v I, pp 12, 0-26)

13. Wheels and Braking, A. Revillon and A. Leluan, France

Explanations for some of the damage caused in wheels are presented in this paper. Of primary concern are the changes in shape due to modifications in residual stress at the surface and in the body of the wheel, localized faults due to metal transformations and brittle fractures caused by overshooting the impact resistance of the steel.

Initially the paper discusses methods for determining the residual force both in the body and on the surface of new and used wheels. For destructive methods, using a rectangular beam for a model, they discuss the relationship of elastic internal stresses and nonelastic surface stresses in demonstrating, by the use of cutting and planing methods, the determination of the residual stress. The analogy is then made as to how this is done for wheels. They state that the superposition of these two methods, i.e., cutting and planing, enables the calculation, with good accuracy, of both surface and near surface stresses with error margin of 5%. In the body there is an error margin of 20%. X-ray diffraction is used as the nondestructive method, and again the authors give a theoretical discussion of the use of X-ray diffraction for residual stress measurement.

In this discussion, σ_1 is the radial stress, σ_2 the circumferential stress and σ_3 is the stress parallel to the wheels' axis. The residual stresses in new monoblock wheels were found to be as follows. Under the tread σ_1 is nil; σ_2 has a maximum compression of 10 hbar on the surface, gently decreasing to 0 at about 50 mm from the tread. σ_3 is very small. Under the lateral faces of the rim and in the rim itself, σ_1 is very small; σ_2 has a maximum compression of 10 hbar, again gently decreasing to reach an average of -5 hbar at mid-thickness of the rim, and σ_3 is again very small. In the plate, σ_1 and σ_2, measured in the thickness sense at various distances from the axis of the wheel, rarely are greater than 10 hbar in tension. The sign, of course, varies according to the manner of cooling and the location along the plate.

An investigation was undertaken to determine the influence of the tread upon the distribution of residual forces. For this, four wheels of the same metal heat from the same type electric rail car equipment were examined. They were moderately under load considering both the braking and the load carried. The average value of the load was 500 daN per wheel of 860 mm diameter when new. The wheel conditions before testing were as follows: No. 1, new; No. 2, 85,000 km of service; No. 3, 200,000 km of service; No. 4, also 200,000 km of service and from the same axle as No. 3 but profiled just before measuring. The results showed that under the tread, σ_1 remained essentially nil throughout the service, σ_2 remained compressive at the surface with a maximum value considerably higher than for new wheels (-30 hbar instead of -10 hbar). Beneath the surface and toward the center of the rim, σ_2 remained small and in tension but appeared to move toward the center of the wheel. σ_3 was highly compressive near the tread surface, being on the order of -30 to -40 hbar. Beneath the surface, the wheel covering 85,000 km showed approximately 20 hbar tension. With increased service to 200,000 km, the tension below the surface decreased to less than 10. Further, by reprofiling the other 200,000 km wheel the level of tension was reduced slightly more. There was essentially no tensile stress at the center of the rim of the reprofiled 200,000 km wheel. The gradient of these under-tread forces on wheels 2, 3 and 4 is greater than that for the new wheel. In the rest of the rim and in the plate, the stresses remain essentially the same, within measurement error. The only effect of the Hertzian forces was the creation of a crescent shaped section under the tread of approximately 60 to 70 mm of length and 20 to 30 mm in thickness.

In order to investigate the influence of intense braking on the distribution of residual stresses in wheels, these stresses were measured in two straight, thick plate wheels, one new and the other having undergone a series of brake tests on a testbed without any rail loading. The test wheel was given 342 stop brakings from speeds of 100, 120 and 200 km/hr. Only slight buckling of the wheel occurred, even with very intense brake forces. In the rim, however, the tensile stresses reached +40 hbar on the surface, changing from -10 hbar as found in the new wheel. In the plate, the tensile forces σ_1 and σ_2 became compressive.

An investigation of the residual forces in wheels taken out of service after receiving excessively intense brakings showed that in the body of the wheel, except for the zone affected by the tread, the stress distribution is similar to that of a wheel braked very intensely on the test bed. The absolute value of the maximum stresses is lower, however. The authors postulate that this may be caused by in-service brakings possibly being slightly less intense than those seen on the bench, because of the superimposed compressive force induced by the tread in the zone affected by this tread. In recapitulating these tests results, the authors state that the condition of the residual stress in wheels in service is a function of the initial distribution of stresses in new wheels, the tread, which increases the compression in the part of the tyre affected by a tread, and also the braking. The chemical composition for these wheels was according to C48 TS, C .51%, Mn 0.72%, S .032% and P .015%.

The authors were interested in explaining the origin and the development of heat cracking. The origin appears to be wholly metallurgical in nature, and the development seems to be tied to mechanical and heat stress and tread wear. In the case of wheels removed from service, an examination of heat cracked treads did not permit any explanation of the phenomena. This was, they felt, because of the work hardening of the tread, the wear caused by adhesion and braking, and deterioration caused by atmospheric corrosion. Laboratory machines, therefore, were used for crack origin studies. Cracks no deeper than 0.02 mm were caused as a result of braking. These appeared at the spots heated by the brake shoes. The origin, the authors feel, is explained by the appearance of martensite. On the matter of the development of flaws after their origin, the authors note that in wheels showing wear on the tread, it was often found the shallow flaws would disappear before growing.

An explanation of evolution of the flaws in the tread of the wheel is given. Two main flaw location categories are listed. Category a is when faults begin to appear in the zone affected by the tread (the middle of the tread). Two principle layers are distinguishable at this location. Layer 1 is a superficial layer 3 to 4 mm thick in which, as a result of the cooling pattern, the residual stresses alternate between tensile and compression during braking. This alternation of stresses from tension to compression allows the development of fatigue cracks which will not be eliminated in service. A second layer, 15 mm thick, is such that the residual forces for surface treated wheels are in compression when new and are increased by tread compression. The authors feel that since only in the case of exceptional braking do the momentary tensile forces occur, and since the Hertzian stresses are practically negligible, the probability of radial development of a flaw is very low. The differences between the superficial layer and the 15 mm thick layer explain the development of heat cracks in the middle of the tread. More specifically, a flaw beginning in the superficial zone will progress till reaching the boundary of the superficial zone and the thicker layer underneath. Conversely, the Hertzian forces, developed at each wheel revolution, cause flaw evolution parallel to the tread by fatigue. This forms a flaking. The authors also note that flaws caused by inclusions, situated at the interface of the two zones, are also developed. Category b faults are those which occur outside of zone a. These originate mostly from shock or abnormal brake shoe placement, and are a potential source of fatigue only when the wheel has been heavily braked.

1 table, 25 figures, (v I, pp 13, 24-44)

14 <u>Track Gage Stability Investigation on Wheelsets with Blocked Braked Solid Wheels</u>, E. Raquet and G. Tacke, West Germany

It was initially noted that the axial deformation of block braked solid wheels depends essentially on the shape of the wheel disk. The present paper was aimed at an investigation of this correlation and clarification of the mechanisms attending to the temperature rise of the wheels due to braking and leading to deformation. A computational approach was used for both temperature calculations and stress and strain analysis. Further, tests on a test bed were conducted in order to verify the calculations.

Three wheel types were used. The first solid wheel had an S-shaped cambered disk. The second solid wheel had an undulated disk but no camber. The third solid wheel had a straight disk and no camber. The chemical composition of the wheels was according to specifications BV2, namely C 0.53 to 0.58%, Si 0.3 to 0.5%, Mn 0.55 to 0.7%, P max 0.045%, S max 0.045% and Cr 0.15%. For computation of the thermal stresses and strains of the wheels, it was first necessary to determine the temperature distribution. This was obtained using a "TESI" computer program with suitable data from train operations. A comparison of the temperatures predicted in the calculations with those obtained during braking tests on the test bed showed good agreement. Next, it was necessary to use the "ANTRAS" computer program to calculate the stresses. It was noted that the stress calculations are based on completely elastic behavior of the material, and the stresses calculated thus apply only within the range of Hooke's law. Wheel number 1 showed, by far, the greatest axial deformation, i.e., the narrowing of the gage and, according to the deformation, high bending stresses at the transition from the hub to the wheel disk and in the outer portion of the disk. The second wheel showed, from the computational analysis, only a very small amount of axial deformation,while in the wheel disk, high and pronounced stress maxima occurred owing to the undulation. The third wheel also showed a small amount of axial deformation, and its wheel disk exhibited a well balanced stress pattern. The circumferential stresses in the wheel tread were of about the same magnitude in all three cases and exceeded the yield point of the material considerably.

Apart from the axial deformation of the wheel disk, the expansion of the hub upon heating of the rim due to braking is an important criteria for assessing the usability of a wheel. Owing to considerable axial deflection in the direction of gage narrowing, the first wheel shows appreciable expansion of the hub on the tread side, through which the shrink fit with an oversize of 1.3o/oo is weaker by about 30%. This also promotes the tendency of the hub and shaft contact surfaces to rust. The results of braking tests confirmed the computational stress analysis.

In conclusion, the authors state that during a temperature rise of the rim due to braking, cumulative thermal stresses in compression are superimposed on the compressive residual stresses. This causes the yield point of the material to be exceeded at elevated temperature. The theoretical and experimental investigations showed that the residual stresses of the wheels in new condition and after exposure to braking must be known if the mechanism attending the axial deformation of the wheels is to be clarified.

8 references, 1 table, 10 figures, (v I, pp 14, 10-18)

15 Solid Wheels:Requirements and Compromises, S. Wiedemann, East Germany

The world wide introduction of solid wheels caused much optimism concerning the elimination of all sources of faults inherent to the wheel. However, because of the action of the brake shoe on a solid wheel, other operational related failures have occurred. These are primarily the formation of cracks on the wheel tread and the plastic deformation of the wheel. By specifying a carbon content of less than 0.55%, it has been felt that the risk of thermal cracking has been diminished. The plastic deformation of solid wheels has been regarded as a wheel design problem. These approaches have had partial success in eliminating wheel failures. The author proposes to address in depth, however, the two causes of wheel failure as they might affect future wheel design.

For consideration, the author chose a quenched and tempered steel having chemical composition C .50%, Mn .78%, Si .35%, P less than .05% and S less than .055%. Data is also furnished showing the effect of temperature on the strength characteristics of steel, the modulus of elasticity and the coefficient of linear thermal expansion.

Considering the solid wheel modeled as a thick walled hollow cylinder, the tangential stresses are calculated and compared to the apparent yielding point as a function of the temperature difference. For temperatures greater than 400°C, linear compressions in the wheel rim must be expected. Also, it is shown through computations that for temperatures greater than 500°C, a working material defect beneath the wheel tread resulting from Hertzian stresses may be expected. This will be noticed as pittings. It is postulated that, in spite of the simplicity of these two examples, the order of magnitude of surface temperature which should not be exceeded in order to assure the fatigue strength of solid wheels is demonstrated. Considering deformations, the author notes that using the conventional heating and wheel deformation theory, plastic deformations are to be expected for wheel rim temperatures at the running tread exceeding 500°C.

Disk plates without a camber are more favorable with regard to lateral displacements than those having a camber. In the case of a steep disk plate, however, the press fit between the hub and wheelset shaft is impaired when the running tread is heated. The heating of the running tread causes a temporary enlargement of the hub boring. The wear and corrosion behavior as a result of various hardnesses in the wheel and the rail, are discussed. The similarity between the wheel/rail system and a roller bearing is noted.

In summary, the author states that the next development stage for wheels should concentrate on the conditions of the wheel heated by the brake shoe. Braking devices and oscillation conditions between the wheel and the rail should be included in a complex manner. Furthermore, the full range of possibilities in the field of materials engineering has obviously not been exhausted. The risks which are inherently involved in solid wheels and shoe brakes can only be eliminated by the application of disk brakes.

16 references, 2 tables, 19 photographs, (v I, pp 15, 9-28)

16. The Effects of Wheel Design and Service Environment on The State of Stress in 28 in. (0.712 meter) Diameter Freight Car Wheels, G. Novak, L. Greenfield and D. Stone, USA

The introduction of low level flat cars in the United States and the requirement in certain areas for greater overhead clearances required the introduction of a 28 in. diameter freight car wheel. Service experience with the B-28 wheel has shown a susceptibility to plate cracking which is not related to the magnitude of the Hertzian stresses. As a result, a modified B-28 wheel has been introduced. This is the D-28 wheel having a thicker plate and larger hub and rim fillet radii. The CB-28 wheel has not experienced plate failures. The purpose of this paper was to present the results of computer analysis of stress levels experienced in the three 28 in. wheel designs.

For input data, the authors assumed a vertical rail load of 204,608 N, a lateral flange load of 88,964 N, a thermal load equivalent to that imposed by a emergency stop from 112.6 km/hr and a tractive load induced by the wheel rail and wheel brake shoe interfaces, each being 3,202 N. Further data included a compressive load of 27,578 N, caused by the normal force of the brake shoe against the wheel tread and an inertial force produced by the rotation of the wheel mass. Octahedral shear stresses were calculated using methods previously developed by the authors. It was concluded that a combination of all of the six loads previously outlined was required to produce the most critical stress state.

The maximum octahedral shear stresses produced in the B-28 wheel by this load application is 16,000 psi (114.45 MPa) and occurs after approximately 45 degrees of wheel rotation. A maximum octahedral stress intensity of about 13,500 psi (93.08 MPa), near the back rim fillet, also occurs after 45° of wheel rotation. The peak stresses in the CB-28 wheel near the front hub fillet is 2,400 psi (16.54 MPa) in the wheel rail plane and increases to a maximum stress of 5,600 psi (38.61 MPa) at 180°. The maximum octahedral stress near the back rim fillet is approximately 8,000 psi (55.16 MPa) in the wheel rail load plane. This stress,however,is diminished during 180° of rotation to a minimum of 3,000 psi (20.68 MPa).

Analysis of the D-28 design showed some improvement in the level of stress when compared to the B-28 design. The maximum stress of 15,200 psi (105.49 MPa) near the front hub fillet occurs after approximately 30° of wheel rotation. A maximum octahedral shear stress of 5,100 psi (35.16 MPa), near the back rim fillet, occurs after 45° of wheel rotation.

The cyclical stress patterns developed in all of the designs were investigated under the load combination described in the introduction. The stresses near the front hub fillet for the B-28 design were found to be a maximum of 16,600 psi (114.45 MPa) and a minimum of 13,500 psi (93.08 MPa). Further, the stresses in the corresponding location in the CB-28 design did not exceed 5,600 psi (38.6 MPa). The stress levels in the rim plate region showed the same effect only at a slightly reduced level. The B-28 design had a maximum stress of 13,300 psi (91.7 MPa) with a minimum stress of 5,900 psi (40.68 MPa). The CB-28 wheel also showed a cyclical variation in the stress intensity during rotation but again at a much lower level, 8,400 psi (57.9 MPa) maximum, and 2700 psi (18.62 MPa) minimum. The peak stress in the D-28 design near the front hub fillet was 14,800 psi (102.04 MPa) and increased to a maximum of 15,300 psi (105.49 MPa) after approximately 30° of wheel rotation. The maximum octahedral shear stress near the back rim fillet was 5,100 psi (35.16 MPa) occurring at approximately 45° of wheel rotation.

It can be concluded that the magnitude of the plate stresses near the hub produced by the addition of on-tread braking are considerably greater in the B-28 and D-28 wheels than in the CB-28 wheel. However, the plate stresses near the rim are considerably lower in the D-28 and CB-28 than in the B-28 wheel.

4 references, 13 figures, (v I, pp 16, 0-13)

17. Evaluation of Various Designs of Wheels in Heavy Load, High-Speed and Severe Braking Service, G. Novak, and B. Eck, USA

A method of analyzing wheel contours, under any combination of thermal and mechanical loads that exist in a service environment, has been presented. Two 36-inch wheel designs have been investigated under several service load combinations with the following results.

The application of a vertical rail load generated a maximum octahedral shear stress in the rim section of a wheel approximately 45° from the wheel-rail load plane. This stress increased significantly when the vertical load was shifted from the tape line position to a position 3/4 of an inch from the rim face. An increase in the rim thickness produced a corresponding reduction of the stress level generated in the rim region by this loading condition. By introducing the effects of wheel velocity in conjunction with the rolling loads, the maximum octahedral plate stress level was reduced approximately 15%.

When emergency braking conditions were combined with the rolling loads, the rim stresses were increased significantly, but the plate stresses developed in the wheel were of approximately the same magnitude as those produced by only rolling loads. However, the magnitude of these plate stresses was increased considerably when thermal loads generated by a 15 minute drag braking cycle were coupled with the rolling and tractive loads. Although stresses generated under these service load combinations produce maximum plate stresses, the magnitude of the stresses in the rim should not be disregarded. The temperatures in the rim can be of sufficient magnitude to produce a reduction in the mechanical properties of the wheel material in this region.

The complementary relationship between thermally and mechanically induced octahedral shear stresses is seen as an essential element in a cumulative damage or fracture theory capable of predicting wheel failure and thereby establishing a rational basis for determining wheel condemning limits.

[Author's Abstract]

13 references, 17 figures, (v I, pp 17, 0-15)

18. Strength Design of Car-Axle Based on Service Load and Fatigue Strength,

S. Tanaka, K. Hatsuno and H. Nakamura, Japan

The authors demonstrate the use of fatigue theory in developing the endurance life for axles on Japanese National Railways. Using measured data taken from operations on several JNR lines, the authors show magnitudes and frequencies of occurrence for stress levels in axles. Laboratory tests were performed on both full sized wheelsets and small 25 and 50 mm samples. The authors demonstrated the earlier occurrence of crack initiation in their samples, using programmed fatigue cycles.

16 references, 2 tables, 21 figures, (v II, pp 18, 0-12)

19. Influence of Sub-Critical Heat Treatment on Ultrasonic Attenuation and Fatigue Limit of Railway Axles, R. De Martini, L. Brazzoduro and G. Stevanin, Italy

The acoustical transparencies of axles are becoming very important for two reasons. First, the manufacturing uniformity can be determined by measuring the acoustical transparency of the axle, and secondly, the detection of fatigue cracks in service is made easier with axles showing a good acoustical transparency. The authors have studied the effect of a subcritical thermal treatment which should improve the axles ultrasonic testability. At the same time they were interested in being sure that no adverse influence on the mechanical and metallurgical characteristics of the steel was caused. The chemical composition of the material tested was C 0.2 to 0.25%, Mn 0.75 to 0.95%, Si 0.2 to 0.3%, S less than 0.035%, P less than 0.035%, and Al 0.02 to 0.04%. The mechanical properties were as follows: UTS 50 - 60 kg/mm^2, YP greater than .5 X UTS, E_5 greater than 22%, KCU greater than 5 kg/cm^2. All axles were normalized to 870-890°C in order to obtain a ferritic-pearlitic structure with a secondary grain size of 5-6 ASTM. After normalization, the axles were tempered at a temperature of about 600°C. The axle specimens subjected to subcritical treatment were cooled in water after tempering, where those without subcritical treatment were air cooled.

The ultrasonic investigation was made with conventional equipment having an excitation frequency of 2.25 MHz. The measure of attenuation was made by obtaining an initial instrumentation calibration. Attenuation in axles to be inspected was then determined by counting the number of back echoes appearing on the screen. They chose this method rather than an expression in dB because they felt this was simpler.

The results of this test showed that the material subjected to SCT showed a considerably lower attenuation for longitudinal wave propagation in comparison to the untreated material. Further, the SCT had the effect of preventing, or at least seriously hindering, transverse wave propagation. The tests performed at higher frequencies, namely 5, 10 and 15 MHz, showed the same effect for SCT but with decreasing magnitude so that at 15 MHz the effect completely disappeared. The authors postulated that since a metallographic examination showed that the subcritical treatment led to no structural change, a change in the sub-micro constituents must explain the polarization or selectivity for the longitudinal and transverse waves. Magnetic permeability tests, conducted on the samples in order to attempt to confirm the hypothesis just given, directed the authors toward an investigation of the residual stress in the material.

An investigation of the residual stresses on both the axles and the cylindrical test pieces indicated that the material subjected to SCT showed residual compressive stresses on its surface both in the longitudinal and tangential direction. These values were near to the yield point of the material, and the values of the tangential stresses were systematically near to 20% higher than the longitudinal stresses. All untreated test pieces showed practically no residual stresses. Since the outer layer was in compression, therefore, the inner core must be in residual tension. An electron microscope examination of the samples showed that for those with subcritical treatment SCT, dislocation density was of an order of magnitude greater than those without subcritical treatment. Tests on the mechanical properties of the pieces cut from axles with and without SCT were also carried out. These showed that with SCT the yield point was lower than without SCT. However, the ultimate tensile strength UTS was greater for the subcritically treated sample. The result is a different stress and strain curve for the two samples and a possible influence on the material toughness of the strain hardening due to the subcritical heat treatment. Impact tests run from -80°C to +20°C showed essentially no difference for the two types of material. Fatigue tests showed a definite increase in the fatigue limit of the pieces subjected to subcritical treatment. The reason for this, the authors postulated, was the increase in dislocation density in the ferrite because of the plastic deformations caused by the subcritical treatment and the presence in the outer zone of the axle of residual compressive stresses. By comparing two types of fatigue tests, the authors state that the effect of the residual stresses on the fatigue limit is about twice that of the structural component due only to the higher dislocation density. It was also shown that the subcritical treatment had a similar effect upon axles having a higher carbon content.

13 tables, 13 figures, (v II, pp 19, 0-12)

20. <u>Evaluation on the Means for the Improvement in Fatigue Strength of Press-Fitted Axles</u>, K. Nishioka, K. Hirakawa, H. Komatsu, and S. Sugawara, Japan

It is well known that a press-fitted member reduces the fatigue strength of an axle from 1/2 to 1/3 of the potential strength available without fitted members. Several methods commonly used to improve the fatigue strength of press - fitted assemblies are listed. The objective of the paper was to offer a preferable axle design method by summarizing and evaluating the available means of improving fatigue resistance of axles. The fatigue limit based on crack initiation as well as that based on complete failure is discussed.

The effect of the axle design upon the fatigue strength can be summarized as follows: 1) the raised wheel seat axle design could increase both of the fatigue strengths, σ_{w1} and σ_{w2}, about twice and 1.5 times, respectively, in comparison with the straight axle design;* 2) the relief groove design has the same effect as the raised wheel seat design, although larger values for the diameter ratio (D/d)** and for the radius of curvature should be taken as an optimum groove shape in this case; 3) change in press-fit shape sometimes has a certain preferable influence on the fatigue strength, although this is somewhat limited; 4) in fatigue of grooved axle design, the size effect is nearly the same as that in straight axle assembly and 5) in the case of torsional fatigue, the raised wheel seat design has little advantage.

The effect of the residual stress of the axle is inevitably

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* σ_{w1} - fatigue limit based on crack initiation

σ_{w2} - fatigue limit for complete failure

** D - axle diameter

d - diameter at the groove

combined with the effect of surface hardness, since the treatment to introduce the residual stresses also hardens the surface. The following are known from these results: 1) the fatigue strength σ_{w2} could be raised by two or three times by compressive residual stresses, regardless of the methods used to introduce the residual stresses; 2) when the surface hardness is relatively high, such as is the case in induction hardening or tufftriding, the fatigue strength based on micro-crack initiation, σ_{w1}, has a tendency to increase very slightly with the increase in compressive residual stresses; 3) the fatigue strength of σ_{w1} could not be raised by the compressive residual stresses when the surface is not hardened by techniques such as subcritical quenching and cold rolling; 4) the fatigue strength σ_{w1mag} takes a value near to σ_{w2} when the surface is highly hardened.* This means that a micro-crack of nearly 0.01 mm in depth does not easily propagate to the magna-cracks depth of 0.10 mm when the axle surface is highly hardened.

17 references, 1 table, 16 figures, (v II, pp 20, 0-11)

21. Research into the Best Usage of S.N.C.F. Axles, A. Revillon and A. Lelaun, France

The authors observe that since many changes have occurred in the design and in the operation of railway systems since the current design method for axles on SNCF was begun in 1940, there is reason to review the design process. The objectives of this research work are to check that

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* σ_{w1mag} - determined by minimum crack detectable by wet magnetic particle method ($\approx$ 0.1 mm deep).

the design method remains valid under the modern working conditions of the rolling stock, and to examine the extent to which it is feasible to reduce the non-suspended mass of the axle. Investigations were carried out in order to determine a more precise approach to defining the conditions necessary in fixing components to wheelsets, to investigate the possibilities offered by some special steels, to improve the fatigue resistance by surface treatment and to study the effects of different anti-corrosion protections.

For the test program, about 100 test pieces were made. These were installed on rotary traction compression machines and rotary bending machines. The test pieces used were smooth tapered aeronautical type test shapes and several bar shapes with various combinations of diameters, grooves, press fits, etc. The testing was conducted over a period which the authors felt would correspond to the life span of wheelsets. The tests yielded a value for fatigue strength for EDR steel. A reduction of about 50% in the fatigue limit resulted from a wheel fixed onto a smooth axle. Also, for wheels fixed onto an axle with a wheel seat of greater diameter than the axle, the fatigue limit can be optimized greater than that for a smooth bar, depending on an optimization of the diameters. When the wheel and an adjoining collar are both pressed onto the same axle, the authors conclude that the diameters of the two seats should be at least 1.12 times that of the body. Also, a shallow groove of approximately 5% of the seat diameter should separate the two seats, and the ratio of the two diameters should be on the order of 1.12 for optimization. For a wheel and collar fitted onto the same axle with the seats close together but not touching, the ratio between the diameters of each seat and the body should be at least equal to 1.12. Further, the length of the wheel boss should be slightly longer (2 to 3 mm) than the length of the wheel seat, allowing a protrusion at each end. The connection radius should be chosen in order to avoid large increases in force, and machining should be carefully done. In this design, if cracks do occur, they will occur in an area away from the joint and thus be more easily detected.

The authors investigated the possibilities offered by using two special steels, namely 25 CD4 and 30 NC11. These are steels having tensile and yield stress values higher than for EDR steels. An increase in the fatigue limit over EDR steel was shown, but they also experienced an increase in the susceptibility to notch cracking when compared to EDR steel. Other tests being performed, but not yet reported, are on the effect of surface treatment and an investigation of the effect of anticorrosion protection.

7 tables, 9 figures, (v II, pp 21, 16-29)

22. <u>The Application of Journal Roller Bearings with Cylindrical Roller Bearings or Tapered Roller Bearings on Short Axle Journals for the Use under Highest Speeds</u>, W. Voelkening and H. Heuberger, West Germany

In the UIC, a standard short axle journal for freight car wheelsets has an axle load of up to 22 tons.* The axle journals have a diameter of 130 mm and a total length of 191 mm. The axles for traveling railbound vehicles can be used for a load up to 25 tons. The problem of fitting the short roller bearings to the longer AAR journal has arisen, and the authors

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* Metric Tons, U. S. Tons = 1.1 x Metric Tons

have investigated the effect upon the bending stress in the axle journals of the shortening of the longer AAR journals. The results show that for the same bending stress in the long axle journals, a 25% higher load can be allowed. The need for very good quality seals and high precision in the fitting of the journals for high speed service is noted. Tests comparing an elastic journal roller bearing with one fixed rigid to the bogie show that forces approximately 1/3 smaller occur in the case of the elastic journal roller bearing. Additionally, tests run at speeds up to 200 km/hr have shown reductions in peak forces across the bearings by about 15% for the elastic bearing. When passing rail irregularities, these forces were reduced by 40%. These axial-elastic journal roller bearings have seen much service in several countries.

6 references, 10 figures, (v II, pp 22, 11-20)

23. <u>Dynamic Behavior of High-Speed, Rolling-Stock Rolling Bearings</u>, K. Kakuta, T. Sawamoto and T. Igarashi, Japan

The high reliability of rolling bearings is of paramount importance for the safe and reliable operation of rapid transit trains. The bearing reliability should be underlined by an accurate and fully detailed grasp of the true behavior of rolling bearings and their environment with first-hand observations and actual measurements. Based on the data showing axle box vibration and the dynamic behavior of bearing elements, collected from measurements of wheel-axle rolling bearings on high-speed transit trains running at high speeds, this paper is intended to clear and improve the design criterions of rolling bearings for use on high-speed transit train wheel-axles. Cylindrical roller bearings and ball bearings of the types commonly used in Japan will be discussed.

The combination of a double row of cylindrical roller bearings and a row of ball bearings is the type commonly used on the New Tokaido Line. Double row cylindrical roller bearings with flanges are commonly used on high-speed container wagons of Japan National Railways. For these investigations the acceleration was measured in a frequency range up to 20 KHz in order to meet the specifications of MIL and IEC standards. At 100 km/hr, the axle box fitted with a double row cylindrical roller bearing on a MINDEN drive bogie, when passing over joints of 25 meter long rails, produced a maximum acceleration of approximately 26 g. Similarly, when the axle box was fitted on an axle with a wheel of 840 mm diameter and a flat was ground on the tread measuring 55 mm by 55 mm square, the acceleration at 90 km/hr was approximately 20 g. By plotting the two cases for several running speeds, the authors show that the acceleration for both the rail joint and the tyre flat are very similar in magnitude. They note, then, that the magnitude of acceleration during normal running of a smooth wheel on a rail without joints was one-half to one-fourth of the magnitude produced by the rail joint. The axle box acceleration is known to increase in proportion to the weight of the car capacity and can reach a value of over 100 g at railway switch points or other singular parts. A frequency analysis of the acceleration pulse, obtained when passing over a rail joint, and compared to the shape obtained during normal running, showed that the shapes were very much similar. The primary increase is noted in the increase in the low frequency component, under 500 Hz, for a car running over a rail joint. The acceleration pulse due to a tyre flat indicated four peaks. The frequency did not change, in spite of a change in speed. These peaks were at 250 Hz, 630-800 Hz, 1.6-2 KHz and 3.15-4 KHz, which are assumed to correspond to natural vibrations of the wheel and the bogie. When the running speed was increased, the acceleration component at peaks other than 250 Hz also increased. It was also shown that as the acceleration of the axle box increased there was a pronounced increase in the stress of the retainer.

The two bearing rows were strain gaged in order to investigate the distribution of load within one journal bearing. The investigation showed some cases of even distribution and other cases of radial loads varying by as much as 68%. The bore of the axle box actually deformed into an oval shape in service, and the bearing outer ring was influenced by this deformation.

The cylindrical roller bearings used on the New Tokaido Line are designed to have a dmn value less than 300 $(10)^3$, even at maximum operating speed of 210 km/hr. The maximum ratio of the radial load to the basic load rating is approximately 0.12. The strength of the retainer was investigated by strain gaging, and it was found that for a square flat on the wheel measuring 55 mm by 55 mm, as the speed increased, the retainer stress actually decreased from approximately 1.5 kg/mm^2 at 20 km/hr to slightly less than 1 kg/mm^2 at 100 km/hr. For a rail joint, however, the retainer stress increased from .5 kg/mm^2 to approximately 1.5 kg/mm^2 when passing over rail joints at the 20 km/hr and 100 km/hr speeds, respectively. The force acting on the retainer pocket of the cylindrical roller bearing increased significantly with speed for both the rail joint and the tyre flat cases. An investigation of motion of the rolling element showed that the revolving slip of a set of cylindrical rollers or retainer was as little as 1%.

The authors state that after approximately 2 million km of service on the New Tokaido Line, bearings that are inspected seldom show any apparent damage.

6 references, 13 figures, (v II, pp 23, 0-10)

24. On the Fatigue Life and Accuracy of Axle-Bearing Unit Type Roller Bearings, A. Adachi, T. Miki, and Y. Ohmori, Japan

The ABU (Axle-Bearing Unit) is a rotating end of cap type tapered journal roller bearing which has been widely adopted by railroads the world over. Its excellent performance has been noted. The present paper represents an effort to further study the bearing life using large sophisticated test machines. The bearing studied was a Koyo ABU tapered journal roller bearing, class F, having an I. D. of 157.162 mm and O. D. of 252.412 mm. The cone and cup widths were 177.8 mm and 184.15 mm, respectively. Eight bearings were used in the test.

Two bearings were mounted in the test machine, one on each axle of the wheel set. The radial load and axial thrust loads were 22,700 kg and 5,000 kg, respectively. The test was driven at 1,000 revolutions per minute, and the duration was 2 to 5 times the calculated life, or 496 to 1,278.7 hours. During the tests, temperature rise and bearing vibration were continuously monitored. The bearing was disassembled after the test for complete examination, and the grease was thoroughly checked.

At the end of the test it was noted that 5 of the 8 bearings showed normal fatigue spalling. Those which showed spalling had been subjected to an average of 3.2 times the calculated life, with a minimum 2.03 times and a maximum of 5.35 times. All of the spalling occurred on the inboard cone raceways where the thrust load was applied. The temperature rose initially to a peak around 117°C in 10 to 20 hours and was followed by a gradual cooling until it stablized at 100 to 150 hours. The average peak temperature rise was 86.1°C while the average stablized temperature was 69.1°C. The row under thrust load showed a higher temperature than the row under no thrust load. No temperature abnormalities were noted. The vibration monitoring showed an inital vibration of 1.5 to 2 g on every sample bearing. When spalling occurred however, the vibration showed an increase to 4-4.5 g or about 2 times the initial reading. An analysis of the bearings after the test showed that the initial starting torque of the bearing before the test had decreased from 289.8 kg-cm to a range of 69.3 to 81.9 kg-cm. All bearings were free from grease leakage with one slight exception. Changes in bearing dimension due to operations during the test were found to be within tolerances. It was noted that the appearance of the three bearings which had not spalled was found to be excellent at the end of the test. The grease condition at the end of the test on all bearings was found to be satisfactory and the condition of the seals was found to be normal.

8 tables, 4 figures, (v II, pp. 24, 0-10)

25. The Technology Problems of Heat-Refining of the Solid Railway Wheel Rims from the Producer's Standpoint, K. Mitura, O. Žarybncikŷ, Czechoslovakia

It was noted that since the new UIC specifications contain requirements for heat-treating the rim of railway wheels, it is well to investigate the mechanical properties and structure of the solid wheels

in both the as-normalized state and the heat-treated state. Initially, a discussion is given on the various relationships of metallurgical and mechanical properties of steels. Before beginning studies on full size wheels, laboratory investigations were conducted on the effect of cooling rate and tempering on the structure and the hardness. This was done using Jominy hardenability tests. On the basis of the laboratory tests, the rims of solid 920 mm dia. wheels were subjected to heat-treatment testing under pilot plant conditions where one side of the rim was cooled. For this, three separate heats were used. The carbon content was 0.43%, 0.5%, and 0.56%, respectively. The percentage of the other constituents was also slightly increased in the heat with higher carbon content. Heat-treatment was conducted for four periods of 2, 2.5, 3 and 3.5 minutes. The mechanical properties of the plate and of the rim after water spraying and after spraying and tempering were investigated. Also, the variations in structure after water spraying and after tempering following water spraying were investigated. Similar investigations were made of the properties resulting from pilot plant trials with three sided cool rim. The deformation of the wheel upon water spraying was also discussed.
13 references, 9 tables, 32 figures (v II, pp 25, 0-41)

26. <u>A Technical, Economic Rational for Railroad Wheel Challenges</u>,
R. Beetle, USA

The emergence of energy and material considerations into wheel manufacturing will further demonstrate the increased advantages of cast steel wheels. A discussion of AAR specifications for wheels was given, and the evolution of current wheel design was briefly discussed.

The energy considerations in the manufacture of both cast steel and wrought steel wheels are given,as are the capital investment requirements for the two types of wheel plants. The author noted advantages of nondestructive inspection in guaranteeing a uniform wheel product. Also, work by others on fracture toughness has shown that the flaw size which becomes critical should be identical for comparable wrought and cast steel wheels. The importance of controlling the braking application on wheels and the advantages of using composition type brake shoes was discussed. 14 references, 4 tables, 5 figures, (v II, pp 26, 0-14)

27. X-Ray Detection of Fatigue Damage in Railroad Wheel Shafts, H. Ohuchida, M. Nagao, and E. Kuboki, Japan

X-ray studies were conducted in order to investigate 1) the size effect of notched specimens upon fatigue, 2) the surface effect on fatigue, 3) a probability problem and 4) an influence of machine finishing conditions upon fatigue damage detection. Also, the rapid evaluation of S-N curves was demonstrated. For this investigation the long specimens were 10 mm in diameter and 300 mm long, the surface area being nearly equal to that of the large specimens (60 mm in diameter and 60 mm long) and the small specimens (10 mm in diameter and 20 mm long).

The results of notched specimen fatigue experiments on the size effect suggests the necessity of using additional structural considerations in the mechanics models. The X-ray study revealed that micro-yieldings at the notch root surface are the same at the fatigue limit whether or not the specimens are of the same shape or size. Moreover, the changes in X-ray broadening on the specimen surface during the fatigue process were found to distribute near to the same line for all kinds of specimens.

A study of the surface area effect upon fatigue showed that the fatigue limit for the large electro-polished and for annealed specimens was 17 kg/mm^2 and 19 kg/mm^2, respectively. The values for annealed and machined small specimens were 20 kg/mm^2 and 21 kg/mm^2, respectively. The fatigue limits of the long specimens were very close to those of the large annealed specimens. Thus the area effect was found to be very important, since the static tensile strengths were the same in the different specimen sizes. The micro-yielding on an arbitrarily selected test area on the long specimens was distributed all over the specimen surface, and its value was mainly a function of fatigue life. The probability study of fatigue life with large wheel shaft specimens at the same stress amplitude (24.0 kg/mm^2) show that X-ray observation can detect those specimens in a group that are too weak to withstand a service load. Further, it was shown that the change in residual stress of large wheel shaft specimens is dependent upon the fatigue life rather on the stress amplitude. Fatigue life scattering was proved to be caused by the scattering of cyclic micro-yielding. The study of the effect of machine finishing conditions on X-ray fatigue detection showed that while the initial residual stress of the lightly machined specimens was nearly equal to that of the normally machined one (i.e., -30 to -40 kg/mm^2), it decreased monotonously under cyclic stress, and the decreasing rate of residual stress was nearly the same as that of normally machined specimens. The change of residual stress, which was considered to be a function of the average plastic strain, represents fatigue damage. A method for the rapid evaluation of S-N curves was sucessfully demonstrated.

10 references, 2 tables, 9 figures, (v II, pp 27, 0-11)

28. <u>Periodic Inspection and Maintenance of the Wheelset</u>, A. Barajas, Mexico

The sole objective of this presentation was to inform those countries in attendance at the Fifth International Wheelsets Congress of the techniques used by American railroads in relation to the maintenance of wheels and axles. These techniques are based on procedures set forth by the Association of American Railroads and by the experience of the member railways. The author describes the general techniques for both visual and nondestructive inspection of axles and wheels. Both magnetic and ultrasonic flaw detection techniques are mentioned. The most common failures to be detected are described, and established inspection intervals and techniques are discussed.

41 figures, (v II, pp 20, 0-45)

29. <u>Axle Maintenance of High-Speed and Very High Speed S.N.C.F. Vehicles</u>, J. Quessart, France

The new railroad to begin operation in 1980 or 1981, running 420 km from Paris to the southeast of France, has caused equipment engineers to investigate the problems of maintenance of the multiple unit trains. The speed will be 260 km/hr. Previous experience on the S.N.C.F. will be used to make projections for this new very high-speed service. Also the domain of very high-speed service is being investigated through the use of two experimental machines. The TGV 001, in service since April 1972, has seen 17,000 km of service at speeds over 260 km/hr and 1,160 km of service since 1974, 2,300 of which were at speeds over 260 km/hr.

For the current high-speed equipment, visual inspections are carried out at fixed intervals by specialized teams at the vehicles home base. Ultrasonics are used only to check for faults in the body of the tyre of monobloc locomotive wheels. The interval between examinations is one year, equivalent to a running distance of about 350,000 to 400,000 km. The inspections are intentionally simple. More complete inspections for locomotives are carried out every one million km. This operation involves complete magnetoscopic examinations of the axle, fixing new wheels onto re-machined wheel seats, replacing axle boxes after checking and changing the axle box grease. The coaches are inspected every 270,000 km, primarily for comfort reasons. The wheel life span for the locomotives is 1 million km, where for the coaches it is about 800,000 km. Reprofiling of the wheels on vertical lathes is performed only for the locomotives. Using a copying lathe, the whole reprofiling operation takes 30 to 40 minutes, where a small amount of metal is removed in one pass. Forty-five minutes to one hour is required in order to eliminate faults on the tread, where two or three passes are needed. Heat detectors are mounted on each axle box on the CC 6500 locomotives.

The experience of the railroad networks show that non-alloy UIC steels give very good results in service. Axle protection from corrosion and impact of small objects at high speeds is important. In the repair of axles, any cracked part is scraped. Welding, metalization, electrolytic deposits, etc., are not permitted because they can lower the fatigue limit. The wheel seats are machined each time the wheel is replaced. Removal under oil pressure is used to avoid damage. Diameter limits are strictly defined. Any groove marks are eliminated. Where possible, crack detection is done by magnetoscopy since this method is more reliable and sensitive than ultrasonics.

The axles of the two SNCF experimental very high-speed locomotives are generally of the same specification as those of other stock. The status of rim bands of monobloc wheels is more strict at this time, and investigations are being carried out to show if this is necessary. Also, a special high-speed axle box grease is used. Investigations are being carried out to show if this is necessary. Also,investigations to find a wear resistant steel for wheels are being conducted. Wheel profile is a stability factor in very high-speed operations and it is hoped that the progression towards the allowed limits will be as slow as possible in the hope of limiting the frequency of profile repairs. The inspection rules for axles at very high-speed service will be essentially the same as those for high-speed locomotives. The only change will be a systematic machining of the wheel profile at intervals of about 3 months (100,000 to 120,000 km). The reprofiling installations will treat the two axles from the same bogie simultaneously,which should permit the reprofiling of 26 axles of a train set in a maximum of 8 hours. In addition, before each reprofiling, an automatic ultrasonic examination of the rim band of monobloc wheels is envisioned.

7 figures, (v II, pp 29, 9-20)

30. <u>On the New Maintenance Control System for the Wheel and Axle Assemblies of Rolling Stocks</u>, K. Ishii, Japan

Two objectives are discussed in the paper, namely to outline the efforts used for assuring that the wheel and axle assemblies used on the New Tokaido Line do not fail in service, and to discuss the shop efficency in handling this maintenance. For the Japan National Railways high-speed stock, the quantity of wheelsets maintained is quite large.

In the span of 1970 to 1971, the inspection of all kinds of rolling stock on JNR was significantly changed. Data obtained on the reliability of the equipment combined with newly developed materials led to the decision to increase the time between inspection intervals for the rolling stock. Also, at the intermediate inspections, the inspection was simplified and limited to the running device only. In the paper, some of the history of the inspection systems for the New Tokaido Line is discussed along with the history of the other electric passenger cars on the previously existing lines.

The maintenance program for the New Tokaido Line consists of general overhauls with regular inspections at frequent intervals and two bogie inspections between general overhauls. The regular inspection was maintained at every 30,000 kilometers (30 days) under the new program. The bogie inspection interval was increased from every 240,000 km (8 months) to 300,000 km (10 months). Similarly, the period for general overhaul was increased from 720,000 km (24 months) to 900,000 km (30 months).

For the bogie, the regular inspection interval consists of inspection of the axles in place under the car. The bogie inspection consists of dismounting of the wheelset from the bogie, "a little dismount" of the wheels, inspection of the axles and turning of the wheel tread.

At the general overhaul, the wheelset is dismounted from the bogie, the wheels are dismounted from the axle, the axles and gears are inspected and the wheel tread is turned. It is also noted that the wheel tread is reprofiled as the occasion calls,which the author suggests is once every two or three regular inspections.

The axles are made of JIS standard carbon steel, commonly used for machine structures. To increase the total fatigue strength of the axles, high frequency induction hardening is applied throughout the length of the axle, with the exception of a small area in the middle of the wheel seat and the ends of the journal. The wheels are high carbon steel of a solid type which are manufactured by hot rolling and rim quenching. Disk brakes are mounted on both sides of the wheel plate. Mounting and dismounting of the wheels to the axles are aided by using an oil injection technique. To make the mounting and the dismounting process easier, an oil injection hole and groove is fitted into the wheels.

As the bogie enters the maintenance flow pattern, it is first disassembled. After washing, the axle is ultrasonically inspected with both normal and angle beam probes. The next step is the dimensional inspection of the wheels, the data being stored in the computer. Next, through using the oil injection method, the wheels are dismounted from the axles. It must be noted that at this point the wheels are stored in a vertical storage rack according to their measured dimension. No further mating of the two wheels and the particular axle from which they came is made. The axle and gear are now inspected by magnetic particle method. Following an inspection of the gear box and bearings, the outside diameter of the wheel seat is measured and stored in the computer. Now the axle is mated up with any two wheels chosen by the computer to have internal diameters giving a suitable interference fit and rolling diameters within specified limits. (It appears that no machining of either the axles or the wheel bore is performed at this time).* The measurement of the wheel bore and diameter and the axle diameter is fully automatic. After the wheels are pressed on the axle, a holding force check at 100 metric tons is made to ensure a proper fit. As wheels are removed from service and new wheels are mounted on used axles, a boring machine with numerical control automatically matches the wheel bore to the axle dimension.

The main factor causing the replacement of axles is the occurrence of micro-fatigue cracks on the wheel seats. These are usually about 50 to 150 microns in depth and are detected by magnetic particle inspection.

As a result of investigation made by JNR, it has been shown that the main contributor to the lowering of fretting fatigue strength of the axle is the concentrated stress induced by the frictional force due to fretting. The stress concentration was shown to increase with both the contact pressure and the coefficient of friction. The occurrence of cracks is mainly influenced by the contact pressure and the cyclic stresses acting on the fretted area. The reason why the non-propagating cracks occur is the fact that the influenced region of the fretting fatigue is limited to a very thin surface layer. Also, it was shown that the stress level for crack initiation decreases linearly with the contact pressure, while on the other hand, the fracture stress tends to a definite value after an initial decrease with contact pressure. Tests showed that the

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* Reviewer's Comment

larger the compressive residual stresses in the axle and the deeper the hardened layer, as a result of induction hardening, the larger the fracture limit. The distribution of residual stress and the depth of the hardened layer were graphically shown.

2 tables, 9 figures, (v II, pp 30, 0-22)

High-Speed Operations of Japan National Railways

The high-speed network connecting Tokyo to Osaka was opened October 1, 1964. This was a completely new right-of-way with no grade crossings and having no service other than the Shinkansen high-speed trains. In 1965, the first year of full operation, approximately 21 million passengers were carried with 40 trainsets. There were 110 scheduled trains on each day. In 1974, based on estimated data, there were 133 million passengers carried.

In March of 1975, the high-speed service was extended from Okayama to Hakata. With the completion of this new line, the total Shinkansen length was increased to 1069 km (664 miles). The number of trainsets has been increased to 133, and there are now 258 daily scheduled trains on both weekdays and Sundays and holidays. The maximum operating speed is 210 km/hr (130.5 mi/hr) with a scheduled speed of 162.8 km/hr (101.2 mi/hr) for the Tokyo to Osaka Line and 166.6 km/hr (103.5 mi/hr) on the Osaka to Okayama Line.

The Shinkansen network is treated by Japan National Railways as a completely separate operating unit so that costs and other operating data are easily identified. According to data furnished by Japan National Railways, the Shinkansen has consistently shown greater income than expenses. In fact, in 1973, with an expense of 135,000 M yen ($450 M),* the revenue was 290,300 M yen ($968 M), yielding a positive balance of 155,300 M yen ($517.9 M).

The distance from Tokyo to Osaka is 515.4 kilometers (320.3 mi)

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* at 299.83 yen/dollar

which is traversed in a time of three hours and ten minutes for nonstop service. The construction of the Tokyo to Osaka line took five and one-half years, and the cost was 380,000 million yen ($1.28B). High-speed trains making scheduled stops also operate between the two points.

Tour of Hamamatsu Work Shop

At the conclusion of the Wheelsets Congress, a tour was conducted of the Shinkansen repair facility located at Hamamatsu, between Tokyo and Osaka. The facility occupies 410,000 square meters of land with a building area of 105,000 square meters. There are 181 office workers and 1,941 shop workers. All of the 2,128 Shinkansen cars (133 sets) are assigned to Hamamatsu. Also assigned for maintenance here are 725 other electric railcars and four wrecking cranes.

The facility was modern, clean and well organized. At the time of general inspection, the trainsets are completely disassembled. The trucks are removed, all electrical equipment is removed and the interior of the car is completely gutted. A complete inspection of the car body is made, and, at each of the stations where the various pieces of equipment are taken, a full inspection of the equipment is made. When the unit is reassembled it is in effect a new trainset. The shop superintendent stated that he could assure a 90% availability of trainsets with this maintenance program. The details given in the last paper (No. 30) of the congress are for the Hamamatsu workshop and will not be repeated again.

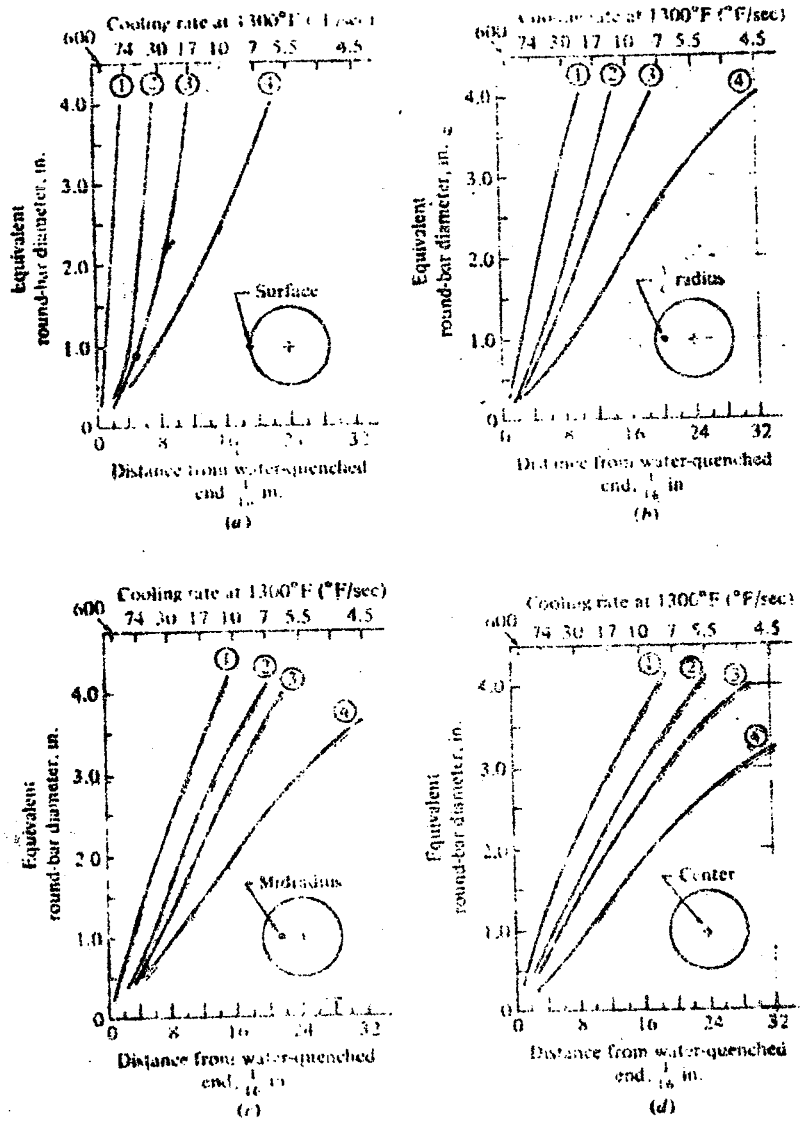

Fig. 8.33 *Relationships between cooling [illegible] in round bars and in Jominy locations. 1 = still water, 2 = mildly agitated oil, 3 = still oil, [illegible] molten salt.*

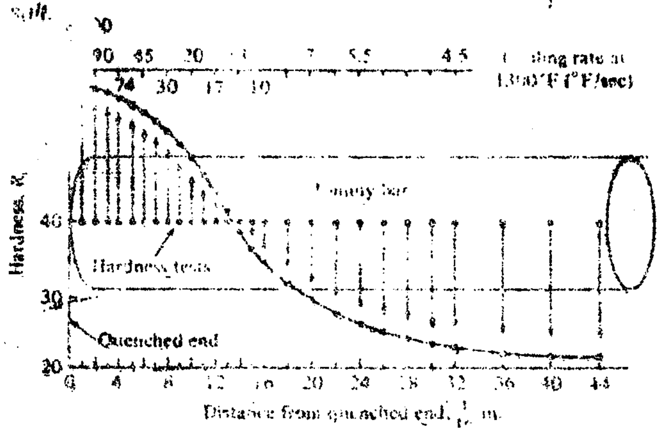

Fig. 8.32 *Typical hardness distribution in Jominy bars.*

(A. G. Guy, "Elements of Physical Metallurgy," 2d ed., Addison-Wesley Publishing Company, Inc., Reading, Mass., [illegible] 444.)

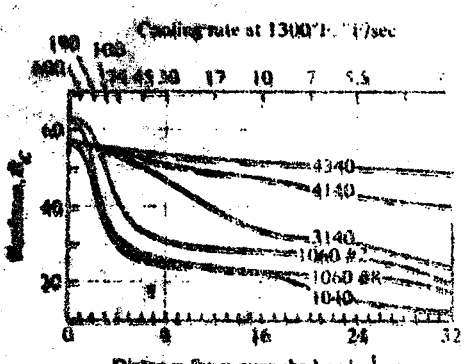

Fig. 8.34 *Hardenability curves for several [illegible] and austenite grain [illegible]*

(L. H. Van Vlack, [illegible])