Experimental evaluation of the fatigue life of hydraulic gate cast iron rollers

Abdul Nabi Lashari | Ioannis Polyzois | Dimos Polyzois

Department of Civil Engineering, University of Manitoba, Winnipeg, Manitoba, Canada

Correspondence
Dimos Polyzois, Department of Civil Engineering, University of Manitoba, Winnipeg, Manitoba, Canada. Email: dimos.polyzois@umanitoba.ca

Funding information
Manitoba Hydro Research and Development Department, G158; University of Manitoba Structural Engineering Laboratory

Abstract
Fixed-wheel gates with wheels mounted on both sides that roll on roller path plates are used in both emergency intake gates and spillway gates in generating stations. Environmental corrosion along with high wheel loads cause differences in the profile of the roller path surface. Combined with the relatively high torsional stiffness of the gate end girders, a condition of wheel load redistribution occurs where some wheels are relieved of load while others are loaded beyond their maximum design values. Failure of one wheel could jeopardize the overall operation of the gate. Furthermore, frequent operation of these gates result in changes in the stress profile in both wheels and roller paths that, potentially, could lead to failure. Currently, design guidelines for gate wheels and roller paths do not consider the fatigue life of these elements. Research-based evidence is provided to support changes to these guidelines.

KEYWORDS
contact fatigue, contact stresses, fatigue of rollers, hydraulic gates, roller path

1 | INTRODUCTION

Hydroelectric generating stations have two types of gates: emergency intake gates and spillway gates. These two types of gates have very distinct purpose although they provide the same basic function of holding back high volumes of water. The purpose of the emergency intake gate is to cut off the flow of water to the turbine during maintenance procedures or in case of malfunction. Spillway gates, on the other hand, are in place to regulate the elevation of the body of water both upstream and downstream. The type of gates used by Manitoba generating stations is fixed-wheel gates. The only exceptions to this are the intake gates at the Great Falls, Manitoba, Canada, which have a roller train system.

Fixed-wheel gates have been extensively used in many water-resource development projects all over the world. Many types of gates have been invented and have become unpopular, but fixed-wheel gates have remained as one of the most widely used type of gates with many applications. The term fixed-wheel gate applies to a rectangular gate with wheels mounted on the gate, as contrasted with an earlier type using roller chains independent of the gate leaf.1 The hydrostatic load is transferred through a skin plate, onto a structural system of diaphragms, horizontal girders, and vertical end girders that are supported on wheels.2 The water thrust on the gate is transferred by the wheels to the roller paths in the gate slots, fastened to track bases embedded in concrete of the structure, and the wheels rotate on the track as the gate is operated. The advantage of providing wheels is that the frictional forces to be surmounted during gate operation are...
relatively of much smaller magnitude as compared to sliding friction in slide gates and enables the gate to be self-closing under gravity without a push force from the hoist.\(^1\)

Consequently, the wheel is a critical component of the gate assembly. Environmental corrosion and high wheel loads cause differences in the profile of the roller path surface. Combined with the relatively high torsional stiffness of the gate end girders, a condition of wheel load redistribution occurs where some wheels are relieved of load and other wheels are loaded beyond the maximum values for which they have been designed. These loads can be as much as two to three times larger as the original design loadings. Failure of one wheel could jeopardize the overall operation of the gate.\(^2\)

While the design of various gate structural components is carried out based on established national standards, the design of gate rollers involves the use of an empirical formula, based on Brinell hardness, to obtain the initial roller diameter and the tread width.\(^3\) Tread surface Hertzian contact stresses and subsurface shear stresses are computed using methods developed by Thomas and Hoersch.\(^4\) Although the Noonan and Strange\(^3\) formula was based on tests involving small-diameter cylindrical forged steel rollers, it has been subsequently adopted for the design of large-diameter crowned wrought-steel wheels, some in excess of 760 mm in diameter. The applicability of this formula to crowned wheels is questionable. Furthermore, this formula provides no information on the fatigue life of rollers or the relationship between the safe working loads and ultimate load capacity of the wheels, thereby making the safe wheel capacity unknown.\(^5\)

The objective of the research program presented in this article is to examine the performance of cast iron wheels under cyclic loading and assess their contact fatigue behavior through a series of full-scale tests. The high cost associated with full-scale experimental research on the fatigue performance of structural elements under rolling contact has led researchers to rely more on theoretical models than experiment-based evidence to develop appropriate failure criteria.\(^6\)–\(^9\) This article presents experimental results that can be used as a basis for comparison with results derived from theoretical models.

### 2 | CURRENT DESIGN PRACTICE

The current design practice by Manitoba Hydro\(^10\) calls for crowned wheels is to be designed in accordance with the criteria outlined by Skinner\(^11\) in his ASCE Paper No. 3000 “Fixed Wheel Gates for Penstock Intakes.” These criteria were developed based on work conducted by Noonan and Strange\(^3\) who devised an experimental procedure to study the relationship between the load on cylindrical steel rollers and deformation or permanent set of the rollers. As a result of their work, a method for evaluating the critical stress causing permanent set in steel rollers was developed. The critical stress, \(\sigma_{cr}\), was expressed in terms of the Brinell hardness number (BHN) by Skinner, as follows\(^11\):

\[
\sigma_{cr} = 0.17 \cdot \text{BHN} - 151.7 \text{ (MPa per millimeter diameter per millimeter width).}
\]  

The critical stress is then equated to the projected area of the cylinder (product of the cylinder diameter and tread length). Knowing the critical stress, for a given diameter of a roller, the required tread width may be computed. The experiments by Noonan and Strange\(^3\) were based on the assumptions that testing of plates to find the load at which they will become permanently deformed by hardened steel rollers is analogous to the testing of metal by means of a Brinell hardness machine and follows from conclusions reached by Wilson.\(^12\) The testing involved solid steel rollers ranging in size from 38 to 254 mm in diameter with a height-to-width aspect ratio ranging from 0.25 to 0.5. For design purposes, the authors recommended that a safety factor of 2 is adequate since failure in the rollers is local. The empirical equation 1 was presented by Skinner\(^11\) originated from work by Noonan and Strange\(^3\) on behalf of the U.S. Bureau of Reclamation.

There are, however, important limitations to the work of Noonan and Strange.\(^3\) Their experimental work involved only solid cylindrical rollers and roller plates made of stainless steels and commercial grades of steel and bronze. It is, therefore, questionable whether the results of this study are applicable to crowned cast iron and carbon steel wheels. In addition, the diameter of typical vertical lift gate wheels used in hydroelectric-generating stations ranges from 685.5 to 838 mm with aspect ratios of 20 to 25. These wheels are considerably larger than those tested by Noonan and Strange\(^3\) and have relatively thin webs. The authors clearly state that the valid range of applicability of Equation 1 is for rollers less than 254 mm in diameter. Skinner’s work\(^11\) was an attempt to validate the work by Noonan and Strange\(^3\) for large diameter wheels. His work dealt with gate wheels fabricated from A57 wrought iron. Skinner\(^11\) also reported that the stress in the tread is the governing factor in gate wheel design and also recommended that a safety factor of 2 be applied to Equation 1 for a wheel over load condition and a safety factor of 3 be applied on the critical stress for normal wheel loads.\(^13\) While the empirical expression given by Equation 1 was used to size a roller, the design against failure was based...
on the maximum shear stress theory,\textsuperscript{4} which states that the maximum shear stress, $V_u$, developed when two bodies are in direct contact is one-third the maximum compressive stress, $C_u$, at the point of contact,\textsuperscript{14} that is,

$$V_u = \frac{C_u}{3}. \quad (2)$$

The shear resistance of the wheels, however, varies with the type of material used and must be determined experimentally. A simple approach would be to relate shear strength to hardness since hardness can be easily obtained. However, most of the information available involves the relationship between the tensile strength of steel and hardness.\textsuperscript{5} Lieson and Jurinal\textsuperscript{15} developed the following relationship between the ultimate tensile strength, $T_u$, and the BHN for plain carbon steel:

$$T_u = 3.45 \cdot \text{BHN (MPa)}. \quad (3)$$

The applicability of Equation 3 was limited to a range between 200 and 350 BHN, with greater variation in the ultimate tensile strength exhibited for high BHNs. Lieson and Jurinal\textsuperscript{15} also presented the following relationship between the ultimate tensile strength, $T_u$, and the BHN for cast iron:

$$T_u = 2.02 \cdot \text{BHN} - 202.5 \text{ (MPa)}. \quad (4)$$

Equation 4 was limited to materials whose BHN ranged from 150 to 300. A number of other relationships between tensile strength and BHN have also been developed for cast iron.\textsuperscript{16} There is no direct relationship between the ultimate shear strength, $V_u$, and hardness. Rather, the relationships between the shear and tensile strength and between the tensile strength and hardness have been used to derive a relationship between shear strength and hardness.\textsuperscript{5} For low carbon steel, the shear to tensile strength ratio is\textsuperscript{17}:

$$\frac{V_u}{T_u} = 0.7. \quad (5)$$

Combining Equations 3 and 5, the following relationship between the ultimate shear strength and BHN was obtained:

$$V_u = 0.7(3.45 \cdot \text{BHN}) = 2.4 \cdot \text{BHN} \leq 612 \text{ (MPa)}. \quad (6)$$

The maximum elastic shear stress is based on Skinner\textsuperscript{11} who established a maximum BHN for wrought steel of 255. According to the current design procedure for cast iron wheels, the maximum shear stress due to the applied loads, computed from Equation 2, must be less than or equal to the ultimate shear stress given by Equation 6. This, however, is the maximum elastic shearing stress suggested by Skinner\textsuperscript{11} for wrought steel wheels. Polyzois and Muzyczka\textsuperscript{18} established the following expression for cast iron:

$$V_u = 2.82 \cdot \text{BHN} - 282 \leq 564 \text{ (MPa)}. \quad (7)$$

3 | CONTACT FATIGUE

Contact fatigue is a surface-pitting-type failure commonly found in ball or roller bearings.\textsuperscript{19,20} This type of failure can also be found in gears, cams, valves, rails, and gear couplings. Contact fatigue has been identified in both ferrous and non-ferrous metals and in ceramics and cermets. Contact fatigue differs from classical structural bending or torsional fatigue in that it results from a contact or Hertzian stress state. The contact geometry and the motion of the rolling elements produce an alternating subsurface shear stress. Subsurface plastic strain builds up with increasing cycles until a crack is generated. The crack then propagates until a pit is formed. Once surface pitting is initiated, the bearing becomes noisy and rough running. If allowed to continue, fracture of the rolling element and catastrophic failure occurs. Rolling contact components have a fatigue life (number of cycles to develop a noticeable fatigue spall). Nevertheless, unlike structural fatigue, contact fatigue has no endurance limit. If one compares the fatigue lives of cyclic torsion with rolling contact, the
latter are seven orders of magnitude greater.\textsuperscript{21} Contact fatigue produces a surface damage that is unique and well recognized. Familiar examples are found in fatigue of ball and roller bearings. Even though this spall is small, it would grow in size until roller fracture would occur, as bearing operation continues.\textsuperscript{19}

Sciammarella \textit{et al}\textsuperscript{22} conducted an experimental evaluation of rolling contact fatigue (RCF) in railroad wheels and reported that cracks due to contact fatigue lead to a damaging phenomenon called shelling. This phenomenon has actually been observed in a number of spillway gates operated by Manitoba Hydro,\textsuperscript{13} as shown in Figure 1. Zhang and Ren\textsuperscript{23} also reported similar failures.

Sciammarella \textit{et al}\textsuperscript{22} acknowledge that the fatigue contact problem is very complex due to a large number of variables involved. These authors emphasize that experimental verification is of paramount importance in the development of numerical models.

Jalalahmadi and Sadeghi\textsuperscript{7} developed a numerical model by incorporating a damage mechanics modeling approach into a Voronoi finite-element method. According to these authors, their model considers microcrack initiation, coalescence, and propagation stages. Computer modeling of process of accumulation of RCF damage in railway wheels was also the subject of an investigation by Sakalo \textit{et al}.\textsuperscript{24}

Cracks initiated and growing under rolling-sliding contact were also the subject of a study by Daves \textit{et al}.\textsuperscript{6} These authors used a two-dimensional finite-element model to evaluate the influence of contact pressure and slip on the crack driving force, crack growth rate and growth direction.

Haidemenopoulos \textit{et al}\textsuperscript{25} investigated RCF on three subway rail sections, approximately 1.5 m long over a 2-year period of service. Based on operational data provided by the subway operator, it was estimated that over the 2-year operation period, the rails had been subjected to 430,000 loading cycles (pass of a single wheel from a given rail location). The operational axle load (normal load) was determined to be 117.6 kN. The wheel material was steel ER 92. The authors conducted metallurgical analysis of RCF crack initiation and propagation, including geometrical characteristics of RCF cracks—length, depth from surface, angle of propagation, and spacing between cracks. They observed that RCF damage is characterized by an array of nearly parallel cracks, which initiate at the surface and propagate at an average angle of 22°-27° with the rail surface. In several cases, subsurface crack initiation was documented. Zhao \textit{et al}\textsuperscript{26} also observed that local RCF has occasionally been observed to propagate 2.5 to 8.5 mm deep along shallow angles to the surface on wheels of high-speed railway wheels. Huang \textit{et al}\textsuperscript{27} reported that fatigue cracks grow along a larger angle to depth on the rail rollers than on the wheel rollers due to the difference of materials.

Languel \textit{et al}\textsuperscript{28} proposed a numerical approach based on a steady-state algorithm to predict the RCF crack initiation in railway wheels in practical conditions. The effect of sliding on fatigue life was also studied and the results showed that the case of rolling with sliding is more severe than without sliding.

Sandström\textsuperscript{29} used a probabilistic analysis approach to examine subsurface RCF damage of railway wheels. The results showed that the probability of fatigue damage is very much dependent on the rail configuration and train speed.

Faccoli \textit{et al}\textsuperscript{30} carried out an experimental investigation to study and compare the response to cyclic loading of the high-performance railway wheel steels. The main damage phenomenon for both steels in the dry rolling-sliding condition was surface-initiated RCF due to ratcheting. Monitoring the crack initiation behavior of subsurface RCF of railway wheels was the subject of an investigation by Zeng \textit{et al}\textsuperscript{31} and Peixoto and de Castro.\textsuperscript{32}
The state of stress produced by rolling contact is concentrated in a small volume of material and produces intense plastic strain. The strain accumulates as the same volume is stressed with each rolling cycle until a crack is initiated and forms a spall. In the real world of contact fatigue, the mechanisms involved can be quite complex. Most models assume a condition of ideal geometric surfaces and little input by heat generation, environmental conditions, and in-homogeneities of material. Hertz stress analysis assumes a circular, elliptical, or line contact surface area between curved surfaces (depending on the geometry of the contacts) and a parabolic pressure distribution with the maximum pressure at the center of the contact.19

4 | EXPERIMENTAL PROGRAM

The experimental program presented here involved laboratory testing of a grey cast iron roller R₁, and a roller path plate, P₁, under cyclic loading. The roller, shown in Figures 2 and 3, was 838 mm in diameter with an 89-mm flange thickness and a crown radius of 914 mm. Roller R₁ was made of gray cast iron and was obtained from the Kelsey Hydro Generating Station located on Nelson River, Manitoba. It was never placed into service. There were three 63.5-mm-diameter holes spaced at 120° apart to facilitate handling. The roller path P₁ consisted of a rectangular steel plate measuring 381 mm × 178 mm with a thickness of 51 mm. Hardness measurements were taken in both the roller and the roller path plate prior to testing. The hardness measurements indicated that the roller hardness profile for Roller R₁ varied from 391 BHN at the rolling surface to 219 BHN at 38 mm below the rolling surface. The roller path plate P₁ was AISI 1050 Medium Carbon Steel (no heat treatment) and had a hardness of 291 BHN.

A unique test setup was designed and constructed for this special fatigue type testing, as shown in Figures 4 and 5. The roller was placed in this steel fixture horizontally. A 12.7-mm-thick and 305-mm-diameter circular Teflon sheet was placed beneath the roller in order to avoid friction between the roller and the steel fixture during cyclic testing. A 152-mm-diameter and 495-mm-long solid steel shaft was inserted through the steel fixture and the 152-mm-diameter hole...
of the roller. The shaft was completely locked into place and the roller could rotate freely. Before inserting the shaft into the roller, lubricating grease was applied to the inside roller hole and around the shaft in order to minimize the friction between the roller and the shaft during cyclic testing. After the roller was placed into the test fixture, the whole steel fixture was brought into contact with the roller path plate (Figure 5).

Four 2.5-m-long high strength bars were used to apply the service compressive radial load on the roller. The nominal thread diameter of these bars was 25.4 mm and their ultimate strength was 567 kN. Each bar was instrumented at its mid-length with a single-element unidirectional strain gauge and was calibrated using a 267-kN capacity testing machine in order to monitor the radial load applied on the rollers.

The service compressive radial load on the roller was applied by compressing the whole steel fixture against the roller path plate by pushing the test fixture through four high strength rods, which ran through the strong concrete wall and steel fixture. The high strength rods were already calibrated in order to monitor the strain values during cyclic testing. Four hydraulic jacks were used to pull the test fixture. To ensure that all four rods were equally stressed, two spreader steel beams were used. All jacks were pumped simultaneously and a uniform and constant static tensile load was induced in each high strength bar. The test fixture was fixed to the concrete floor after applying the required service radial load on the roller.

The high strength rods were already calibrated in order to monitor the strain values during cyclic testing. A radial compressive load of 838 kN was applied to Roller R₁ by tensioning the four high strength rods using hydraulic jacks. The
allowable radial compressive load for a roller similar to that tested was determined to be 734 kN for serviceability and 1050 kN for strength criteria. The objective of applying radial load during cyclic testing was to check the serviceability criteria and not strength. The average applied load prior to testing was recorded as 838 kN and varied during testing from 799 to 850 kN. The applied load of 838 kN was determined to be 38% of the capacity of the roller, based on the average of three ultimate tests of similar rollers (2197 kN) conducted by Muzyczka, providing a factor of safety of, approximately, 2.5. The test fixture was fixed to the concrete floor after applying the required service radial load on the roller.

The cyclic load was applied to the rolling path plate through a hydraulic actuator. The automatic cyclic movement of the hydraulic actuator piston rod was controlled by electronic relays through two magnetic sensors installed on the actuator. An hour meter was also installed on the pump to monitor the running time on the motor and a digital meter was installed on the pump to monitor the number of cycles for the hydraulic actuator to deliver the rolling force.

As shown in Figure 5, the rolling path plate was attached to an 863 mm × 609 mm × 76 mm steel plate (Plate 1), which was free to roll on a series of solid round bars that were attached to a second plate (Plate 2). Plate 2 was permanently attached to the 609-mm-thick strong concrete wall of the Structural Engineering Laboratory.

Roller R1 was instrumented with 10 strain gauges. Nine of these were installed on the rim of the roller, as shown in Figure 6. These were installed on the flat surface of the rim, shown in Figure 7, approximately, 20 mm from the contact surface with the roller path and 10 mm apart, measured circumferentially. Strain Gauge 9 was installed in the rim of the wheel, as shown in Figure 7. All strain gauges were aligned in the radial direction. Strain Gauge 14 was installed on the web, approximately, 88 mm from the contact surface with the roller path, as shown in Figure 6. Zeroing of the strain gauges was carried out before the load was applied to the roller. All strain gauges indicated initial strains when the cyclic loading began. The strain gauges on the rods were used to monitor the applied load.

Due to friction between the roller and the shaft during a cyclic testing, heat was generated at the interface of the roller bushing and the shaft and this heat was transmitted throughout the roller. In order to monitor the temperature of the roller surface, which could affect the strain gauges, a thermocouple was installed at the rim surface of the roller where strain gauges were installed. The thermocouple was wired to a digital thermometer, and daily temperature readings were recorded and corrections were applied to the strain gauge values, according to the manufacturer’s recommendations. A cooling fan was also installed to lower surface temperature.

Roller R1 was tested to a million cycles. The test was continuous and uninterrupted. The roller was rolled back and forth on Plate P1 for a total circumferential distance of 75 mm or 37.5 mm from either side of the central strain gauge 9 on Roller R1 and central location on Plate P1. The frequency of the cyclic testing for Roller R1 was set as 3 seconds per cycle (0.33 Hz). The test duration was not based on the life expectancy of the roller, but rather it was purposely chosen to be open-ended in order to develop a database for serviceability based on load deformation of both roller and roller path. Thus, one cycle consisted of a total travel of 150 mm with a speed of 50 mm/second.
TABLE 1  Material properties of Roller \( R_1 \) and Roller Path Plate \( P_1 \)

| Specimen | Specification | Carbon content, % | Hardness (BHN) | Elastic modulus GPa (ksi) | Poisson’s ratio | Yield strength MPa (ksi) | Ultimate tensile strength MPa (ksi) |
|----------|---------------|------------------|----------------|--------------------------|----------------|-------------------------|-----------------------------------|
| \( R_1 \) | Gray cast iron ASTMA48 Class 20A | 3.69 | 219-391 | 103.4\(^a\) (15 000) | 0.27\(^a\) | 196\(^a\) (28.4) | 265\(^a\) (38.4) |
| \( P_1 \) | AISI 1050 Medium Carbon Steel (no heat treatment) | 0.49 | 291 | 207\(^b\) (30 000) | 0.28\(^b\) | 413.7\(^b\) (60) | 724\(^b\) (105) |

\(^a\)References 13, 16, 33, 34.  
\(^b\)References 33, 35, 36, 37, 38, 39, 40.

In order to observe the extent of damage under cyclic testing, the contact areas of rollers and plates were scanned after testing using the electron microscope (scanning electron microscope [SEM]) at the Materials Testing Laboratory in Mechanical Engineering Department, University of Manitoba. Samples were extracted from both tested and nontested areas of the roller and the plate. Initially, large pieces of chunk were cut from the rollers and plates using abrasive water-jet cutting technology at the MGI Canada Inc., Selkirk, Manitoba. After that, small samples in exact dimensions were cut using a lathe machine at the Selkirk Machine Works (1982) Ltd., Selkirk, Manitoba.

Two samples were extracted from each tested contact area of the roller and plate; one for scanning the contact-surface and the other for scanning the inside surface perpendicular to the contact surface in order to observe the extent and depth of damage. All samples were 20 mm in thickness. Control samples extracted from the nontested areas of rollers and plates were 20 mm × 15 mm in dimension. Those samples that were extracted from the tested contact areas of the specimens were 20 mm × 15 mm and 30 mm × 10 mm.

After the completion of SEM testing, all control samples were sent to Arrow Laboratory, Inc. in Wichita, Kansas, US, for the determination of carbon content. Based on available information of surface pore structure, BHN, and carbon contents, specimens were identified and material properties were specified through literature search. Mechanical properties are listed in Table 1.
FIGURE 8  Strain vs number of cycles in Roller R1

5 | RESULTS AND DISCUSSION

As the roller moved back and forth, all microstrains varied significantly. The strain readings for gauges 5 to 13 are shown in Figure 8. Maximum tensile strains were observed in all gauges installed on the rim surface of the roller when these gauges were farthest away from the contact point, and minimum tensile strains were recorded when the gauges were either in contact with the roller path plate or very close to the contact point.

The magnitude of the strain on a particular point on a roller varied, depending on the relative position of the point with respect to the point in direct contact with the roller path plate. As shown in Figures 9 and 10, the strain recorded by gauge 9 during the first cycle of testing varied from 842 με (in tension), when the gauge was in direct alignment with the point in contact with the roller path, to 1340 με (in tension), when the gauge was the farthest away from the contact point.

Before testing the roller under cyclic loading, the roller and roller path were aligned so that the central gauge #9 coincided exactly with the center of the rolling surface of roller path plate. However, during cyclic testing of the roller, it was noted that the roller had started slipping gradually with respect to roller path Plate after about 300 000 cycles and it was not rolling symmetrically, as was set to do in the beginning of the test. At the end of a million cycles, the roller had slipped a maximum distance of 44.5 mm. This is the reason that gauges 5 and 6 were in contact with the roller path plate at all times, whereas gauges 12 and 13 were far away from the contact area during each cyclic movement. For this reason, there was significant change in the strain curves for gauges 5, 6, and 7 as compared to that of gauges 12 and 13.
A static compressive radial load was applied to the roller in an FE ANSYS model. In the laboratory testing program, a constant radial compressive load along with a rolling load was applied to roller and roller path plate during cyclic operation of the roller. Before initiating each cyclic test, the radial compressive load was applied gradually and slowly by tensioning the four high strength rods using hydraulic jacks. The values of the radial compressive load, after releasing the jack pressure and just prior to starting the cyclic testing, were used in the FEA. The strain values just prior to starting of the cyclic testing were retrieved to compare with the FE results. In the following discussion, the strain data from the laboratory testing program when the roller and roller path plate were stationary are compared to the FEA results at two locations, strain gauges 9 and 14.

A tensile microstrain of $+999.6$ was recorded by Strain Gauge 9 under the static radial load of 838 kN. A tensile microstrain of $+947.6$ was recorded at the same location by the FEA. The difference between the laboratory test result and FEA result is 52 microstrains (5.2%). The fact that this strain is tensile is attributed to the bulging of the rim under the high applied load.

At the end of the cyclic testing, the rolling contact surface was examined visually and indentation measurements were taken. Plate P1 exhibited a maximum indentation of 1.48 mm. The tested contact area of the roller and the roller path is shown in Figures 11 and 12. Despite the very small distance of travel of 75 mm, traveled by the roller during each cycle, the contact area went through severe deformation and distortion. Both major and minor visual cracks were observed. Cracks were vertical, horizontal, diagonal, as well as longitudinal.

In order to observe the extent of damage that took place under cyclic testing, tested contact areas were scanned using an SEM. Figures 13 and 14 show the tested and nontested surface areas of the roller using the SEM. Large cracks are very
clearly seen in Figure 14. Figures 15 and 16 show the control and tested surface areas of the rolling path plate. Several large cracks are evident here, as well.

In the majority of the gauges installed on the rim surface of Roller R₁, it was found that with the increase in the number of cycles, the strain dropped from a higher tensile strain to a lower tensile strain or even to compressive strain. This may be attributed to the formation of microcracks after repeated cyclic loading and the material might have lost its stiffness in the vicinity of these microcracks. The literature review reveals that with the increase in the size of microcracks, the local stress within the vicinity of microcracks decreases. The cracks that are generated by contact fatigue, however, do not lead to overall wheel failure, but rather are precursor to localized damage, often referred to as shelling.²²

The recorded strains are much smaller than the ultimate strains under static loading reported by Muzyczka¹³ indicating that while the cast iron roller was designed with a safety factor of almost 2.5, the presence of cyclic loading deteriorated its structural performance. Due to the unpredictability nature of the fatigue life of such rollers, it is recommended that their use be curtailed in the future.
6 | CONCLUSIONS

Fixed-wheel gates with wheels mounted on both sides that roll on roller path plates are used in both emergency intake gates and spillway gates in hydrogenerating stations. Frequent operation of these gates results in changes in the stress profile in both wheels and roller paths that, potentially, could lead to failure. Unfortunately, current design criteria for gate wheels and roller paths do not consider the fatigue life of these elements. In this article, results from an experimental program involving full-scale rollers are presented, which show that the presence of cyclic loading deteriorated the structural performance of the cast iron roller. Due to the unpredictability nature of the fatigue life of such rollers, it is recommended that their use be curtailed in the future. The results also showed that the medium carbon steel (without heat-treatment) plates performed very poorly and must not be used as roller path plates. Future work may explore the use of heat-treated stainless steel or high carbon steel roller path plates with heat treatment as an alternative material for roller paths. The results from this study showed that sliding occurred between the roller and the roller path, a phenomenon that, as Langue et al.\textsuperscript{28} point out, would be more severe on the fatigue life of rolling wheels than without sliding. Future work, whether experimental and/or theoretical, should address the effect of sliding on rolling fatigue.

This article is unique in that it provides experimental evidence for the fatigue life of hydraulic gate rollers. While numerous theoretical studies have been published in this area, the high cost associated with testing of full-scale rollers has made access to experimental evidence difficult. The data presented in this article will make it easier for those researchers developing theoretical models to verify their models. While the rollers used in this experimental program were designed for emergency intake gates and spillway gates, the results have a broader application, more specifically as they apply to railway wheels.

ACKNOWLEDGEMENTS

This study was made possible by funds provided by the Manitoba Hydro Research and Development Department (Grant Number G158) and the support by the University of Manitoba Structural Engineering Laboratory technical staff. The specimens used in the experimental investigation were donated by Manitoba Hydro.

CONFLICT OF INTEREST

The authors have no conflict of interest relevant to this article.
NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| BHN    | Brinell hardness number |
| $C_u$  | maximum compressive stress |
| MPa    | megapascal |
| $P_1$  | roller path specimen #1 |
| $R_1$  | cast iron specimen #1 |
| SEM    | scanning electron microscope |
| $T_u$  | maximum tensile stress |
| $V_u$  | Maximum shear stress |
| $\mu e$ | microstrain |
| $\sigma_{cr}$ | critical stress |

ORCID

Dimos Polyzois https://orcid.org/0000-0003-1000-872X

REFERENCES

1. Sagar BTA. Fixed wheel gates-uses and limitations. Paper presented at: Proceedings of the ASCE National Conference on Hydraulics Engineering. 1989:1162-1168.
2. Polyzois D, Muzyczka WJ, Garroni JD. Finite element analysis of cast iron spillway gate wheels. Comput Struct. 1995;55(4):741-748.
3. Noonan NG, Strange WH. Tests on rollers. Technical Memorandum No. 399. U.S. Bureau of Reclamation; 1934.
4. Thomas HR, Hoersch VA. Stresses due to the pressure of one solid upon another. Engineering Experiment Station Bulletin University of Illinois; July 1930; 212:14-22.
5. Polyzois D, Muzyczka WJ. Behaviour of cast-iron spillway gate wheels. J Mater Civ Eng. 1994;6(4):495-512.
6. Daves W, Kráčalkl M, Scheriau S. Analysis of crack growth under rolling-sliding contact. Int J Fatigue. 2019;121:63-72.
7. Jalalahmadi B, Sadeghi F. A Voronoi FE fatigue damage model for life scatter in rolling contacts. J Tribology. 2010;132:1-14.
8. Sadeghi F, Jalalahmadi B, Slack TS, Raje N, Arakere NK. A review of rolling contact fatigue. J Tribology. 2009;131:1-15.
9. Dang Van K. Duality, inverse problems and nonlinear problems in solid mechanics modelling of damage induced by contacts between solids. C R Mecanique. 2008;336:91-101.
10. Manitoba Hydro. Design requirements for gate rollers. Specification No. 1819, Lime Stone G. S. Spillway Gates, B8. 1986.
11. Skinner SJ. Fixed Wheels for Penstock Intakes. Transactions of the American Society of Civil Engineers, Paper No. 3000. New York, NY: American Society of Civil Engineers; 1957:740-771.
12. Wilson WM. Tests on the bearing value of large rollers. Bulletin of the Engineering Experiment Station No. 162. Urbana, IL: University of Illinois; 1927.
13. Muzyczka WJ. Ultimate load capacity of cast iron wheels for vertical lift fixed-wheels gates (MSc thesis). Winnipeg, Manitoba: University of Manitoba, Department of Civil Engineering; 1992.
14. Roark RJ. Roark's Formulas for Stress and Strain. 6th ed. New York, NY: McGraw Hill; 1989.
15. Lieson C, Jurinal RC. Handbook of Stress and Strength-Design and Material Application. Collier-MacMillan: London, England; 1963.
16. Angus HT. Cast Iron: Physical and Engineering Properties. 2nd ed. London, England: Butterworth; 1976.
17. Davis HE, Troxell GE, Hauck GFW. The Testing of Engineering Materials. Vol 1982. 4th ed. New York, NY: McGraw-Hill; 1982.
18. Polyzois D, Muzyczka WJ. Ultimate load capacity of cast iron wheels for vertical lift fixed-wheel gates. Paper presented at: Proceedings of the CSCE Annual Conference; June 8-11, 1993:317-326.
19. Lampman SR, ed. ASM handbook. Fatigue and Fracture. Vol 19. Materials Park, OH: ASM International; 1996.
20. Harris T. The Endurance of Modern Rolling Bearings, AGMA Paper 269.01. Rolling Bearing Analysis. New York, NY: John Wiley; 1964.
21. Bhargava V, Hahn GT, Rubin CA. Rolling contact deformation and microstructural changes in high strength bearing steel. Wear. 1989;133:69.
22. Sciammarella CA, Chen RJS, Gallo P, Berto F, Lamberti L. Experimental evaluation of rolling contact fatigue in railroad wheels. Int J Fatigue. 2016;91:158-170.
23. Zhang G, Ren R. Study on typical failure forms and causes of high-speed railway wheels. Eng Fail Anal. 2019;105:1287-1295.
24. Sakalo V, Sakalo A, Tomashhevskiy S, Kerentcev D. Computer modelling of process of accumulation of rolling contact fatigue damage in railway wheels. Int J Fatigue. 2018;111:7-15.
25. Haidemenopoulos GN, Sarafoglou PI, Christopoulos P, Zervaki AD. Rolling contact fatigue cracking in rails subjected to in-service loading. Fatigue Fract Eng Mater Struct. 2016;39(9):1161-1172.
26. Zhao X, An B, Zhao X, Wen Z, Jin X. Local rolling contact fatigue and indentations on high-speed railway wheels: observations and numerical simulations. Int J Fatigue. 2017;103:5-16.
27. Huang YB, Shi LB, Zhao XJ, Cai ZB, Liu QY, Wang WJ. On the formation and damage mechanism of rolling contact fatigue surface cracks of wheel/rail under the dry condition. Wear. 2018;400-401:62-73.
28. Langueh AMG, Brunel J-F, Charkaluk E, Dufrénoy P, Tritsch J-B, Demilly F. Effects of sliding on rolling contact fatigue of railway wheels. *Fatigue Fract Eng Mater Struct*. 2013;36:515-525.

29. Sandström J. Subsurface rolling contact fatigue damage of railway wheels—a probabilistic analysis. *Int J Fatigue*. 2012;37:146-152.

30. Faccoli M, Petrogalli C, Lancini M, Ghidini A, Mazzù M. Rolling contact fatigue and wear behavior of high-performance railway wheel steels under various rolling-sliding contact conditions. *J Mater Eng Perform*. 2017;26(7):3271-3284.

31. Zeng D, Xu T, Wang J, et al. Investigation of the crack initiation of subsurface rolling contact fatigue in railway wheels. *Int J Fatigue*. 2020;130:1-10.

32. Peixoto DFC, de Castro PMST. Fatigue crack growth of a railway wheel. *Eng Fail Anal*. 2017;82:420-434.

33. Smith WF. *Structure and Properties of Engineering Alloys*. 2nd ed. New York, NY: McGraw-Hill; 1993.

34. Bauccio M, ed. *ASM Metals Reference Book*. Materials Park, OH: ASM International; 1993.

35. Davis JR, ed. *ASM Specialty Handbook: Carbon and Alloy Steels*. Materials Park, OH: ASM International Handbook Committee; 1996.

36. Davis JR, ed. *Metals Handbook*. 2nd ed. Materials Park, OH: ASM International Handbook Committee; 1999.

37. Gale WF, Totemeier TC, eds. *Smithells Metals Reference Book*. 8th ed. Burlington, MA: Elsevier Butterworth-Heinemann; 2004.

38. Harvey PD. *Engineering Properties of Steel*. Metals Park, OH: ASM; 1982.

39. Shackelford JF, Alexander W, eds. *The CRC Materials Science and Engineering Handbook*. 3rd ed. Boca Raton, FL: CRC Press; 2000.

40. Steiner R. *ASM Handbook, Volume 1, Properties and Selection*. Materials Park, OH: The ASM International; 1996.

**How to cite this article:** Lashari AN, Polyzois I, Polyzois D. Experimental evaluation of the fatigue life of hydraulic gate cast iron rollers. *Engineering Reports*. 2020;2:e12117. [https://doi.org/10.1002/eng2.12117](https://doi.org/10.1002/eng2.12117)