Proceedings of the 7th International Conference on Civil Structural and Transportation Engineering (ICCSTE'22)
Niagara Falls, Canada – June 05-07, 2022
Paper No. 118
DOI: 10.11159/iccste22.118

A Method for Track Buckling Prevention of Continuously Welded Rails

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Abstract - The continuously welded rails (CWR) have become popular, which has resulted in lengthened rail life and reduced expenditures on track maintenance since 1950. CWR track is prone to lateral buckling when the rails are under high compressive forces. Conversely, when high tensile forces dominate under cold temperature conditions, rail defect growth can cause rail fractures, weld failures, and rail joints to pull apart. This paper deals with the development of a thermal stress rail damper. The developed rail damper is introduced as one of the most creative means of protecting a CWR structure against thermal longitudinal forces. It is a collection of rail structural elements which should smartly decouple a rail structure from its connecting longitudinal shaking rail and substructure resting on a longitudinal shaking ground, thus protecting a railroad structure's integrity. This paper presents finite element modeling of the CWR incorporating the proposed rail damper to investigate the change in natural rail temperature (RNT) and Longitudinal thermal stress variations, including the effects of each critical parameter on rail behavior. After an extensive parametric study, a comprehensive technique is proposed to include the interaction of various parameters in the accurate and precise practical determination of the longitudinal thermal force and stress distribution on the CWR. It is concluded that the proposed rail damper significantly reduces the risk of rail buckling during warm days. It will prevent or minimize the consequences of track buckling-related derailments.

***Keywords*:** continuously welded rails, buckling, thermal stress, aluminum damper, natural rail temperature

1. Introduction

Changes in temperature within a material result in the contraction or expansion of the material. Traditional rail systems are installed with a bolted joint between rail segments to accommodate the longitudinal movement caused by changes in temperature. However, many rail systems now use CWR, which forgo the expansion joint to provide a smooth continuous surface. The advent of CWR came about when the manufacturing processes allowed the rail to be produced in longer sections than the standard 50-foot lengths of jointed rails. Without the joint, the rail is confined, and thus large axial forces develop in the long sections of the rail as it expands. Ultimately the rail buckles under the load, leading to deformation of the track and potential derailment of trains [1]. Therefore, a system is needed to reduce such deformations and train derailments.

The study aims to review the current methods and techniques for determining the longitudinal stresses that develop in CWR; to propose a new approach for measuring rail stress that is simple, fast, and non-destructive; and to create a new method for quantifying and managing rail stress to prevent buckling caused by thermal stresses. Current methods for determining rail stress involve measuring the neutral rail temperature but are either inefficient or destructive. The primary way under consideration for managing rail stresses is the installation of various dampers. One piston damper and two mass tuned dampers were considered, with the final design being an aluminum damper with properties of a heat sink. The focus of the damper is to reduce the temperatures experienced by the rail.

2. Aluminium Damper

Herein an initial design for a system intended to reduce thermal stresses in CWR is proposed. The system consists of an aluminum “damper” that is attached directly to the steel of CWR (hereafter referred to as “rail” and “damper”). The damper acts as a heat sink: drawing heat from the rail and releasing it into the surrounding environment. The damper design, as seen in Fig. 1, is based on a combination of a typical rail joint and the Rail Temperature Control System (RTCS) initially developed by Physical Sciences Inc (PSI) [2]. For comparison, a cross-sectional and Profile view of the RTCS device is illustrated in Figure 2. The system consists of a clamped solid aluminum device to provide thermal absorption and solar reflectivity to the rail surface. The proposed damper is mounted directly to the rail surface. The proposed damper is mounted with standard 1-inch bolts spaced every six inches in the center. The damper is placed against the surface of the rail, with fins located on the exposed face to promote natural convection, see Fig.1. The mass of the damper and the geometry of the fins will be based on the required rates of heat transfer. The proposed damper would be cast in aluminum and machine finished.

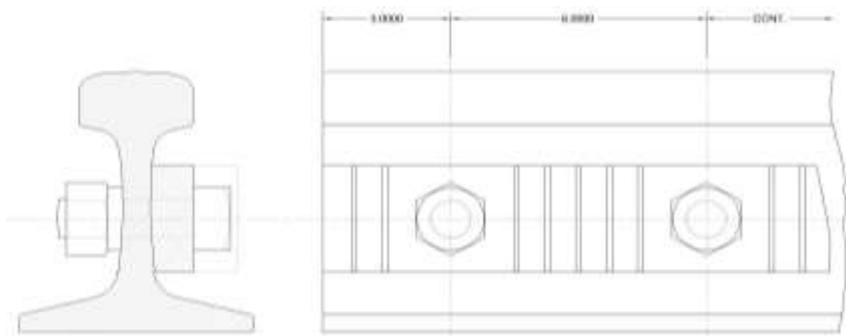

Fig. 1. Damper design.

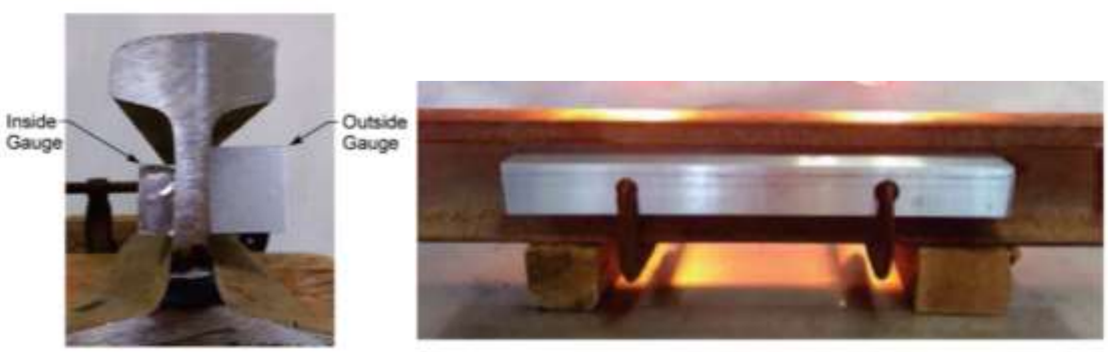

Fig. 2. Cross-sectional and profile view of RTCS device [2].

2.1 Codes and Constraints

After an extensive check and having conducted interviews with advisors from the Public Service Commission trained by the Federal Rail Association, it has been determined that there are no applicable codes that will limit the connection of a device to a rail. The intended device has been designed with a connection modeled after those of other rail devices to adopt best practices in terms of connections to the rail. The device must not exceed the depth of the base of the rail to ensure that rail maintenance machines are not impeded by it, and the connection must not reduce the integrity of the rail. The device's connection to the rail will be further discussed in the forthcoming sections.

3. Numerical Modeling

Initial calculations were performed regarding the change in temperature of both the rail and damper. The analyses are performed as a loop, beginning with the initial energy in the form of heat-induced upon the rail, the heat exchange between the rail and damper, and finally, the cooling of the damper by convection. This process is repeated using the temperatures of both components found in the previous loop and finding the resultant temperatures of the circles of induced heat, heat transfer, and cooling by convection once again. The calculations performed and critical parameters for heat transfer between the rail and damper and cooling of the damper are as follows.

3.1 Determining the heat input for the system

A control rail is considered first to determine the heat (Q) induced upon the system by increasing ambient temperature. This control rail does not have the damper connected to it and is allowed to experience temperature changes based on time and temperature data. Assuming that the temperature of the control rail gradually increases to 130^{O}F and then remains constant for some time, the required heat to cause this temperature increase can be calculated:

$$Q = mc\Delta T \tag{1}$$

Where Q is the heat (which is a measurement of energy) that is induced upon the rail to change the given mass (m) of steel with a set specific heat (c) to change the temperature of the rail in each incremental amount (ΔT). The Q is calculated using ΔT as the data that changes over time. The heat-induced upon the control rail and the rail with the device is calculated using a linear increase in ambient temperature until the control rail reaches 130^{O}F. Using heat as the input rather than temperature is an important distinction: heat considers the material's specific heat, which is a constant parameter that determines the resulting temperature change of the steel. By deciding on this change in heat over time, a set of data is obtained that can be used to calculate temperature change in a rail with the damper affixed when the exact difference in heat (energy) is induced upon it. Calculations for determining the heat input using the temperature change in the control rail can be found in the reference [3].

3.2 Determining the respective temperatures of the rail and damper caused by heat input

Given the calculated heat inputs at each time step, the temperature of the rail with the damper attached can be calculated using a loop for each time step. The initial temperature can be obtained by subtracting the assumed starting rail temperature from a calculated ΔT using the equation with steel's mass and specific heat. Additionally, the temperature of the damper caused by the same induced heat can now be calculated using the same heat equation manipulated to solve for ΔT:

$$\Delta T = \frac{Q}{mc} \tag{2}$$

Specific heat mass (m) and specific heat (c) of the aluminum damper are used. Once the initial temperatures of the steel and aluminum are obtained (by subtracting the previous temperature from ΔT), the cooling of the damper by convection can be considered.

Cooling of the Damper:

$$T(t) = T_{ambient} + [T_0 - T_{ambient}]e^{-kt} \tag{3}$$
$$k = \frac{\alpha A}{cm}$$

$T_{ambient}$ is the temperature of the air surrounding the damper,
T_0 is the temperature of the damper before cooling occurs, calculated in the previous step
t is the time passed (in seconds)
c is the specific heat of each respective material
k is a thermal conductivity constant
α is a heat transfer coefficient given as:

$$\alpha = \alpha_{convection} + \alpha_{conduction} + \alpha_{radiation} \tag{4}$$

The driving factor behind the cooling of the rail is convection based on the geometry of the fins. The surface area is balanced with structural integrity and the need for air circulation. As airflow cannot be depended upon in real-world applications, airflow will have to be generated by the shape of the damper, where heat is focused primarily at the lower portion of the fins, heating the air at the bottom, and forcing the heated air to rise and replenish with cooler air. Conduction between the atmosphere and damper and radiation from the damper itself will be negligible compared to the rate of convection. Thus, a heat transfer coefficient (α) only considering convection will be used to calculate the thermal conductivity constant, *k*, which also considers the mass and surface area of the damper. The heat transfer coefficient α used is the lower limit for aluminum convection. Using the lower limit for this constant, the subsequent cooling rate can be assumed to be a minimum. Given more ideal (though less predictable) real-world convection circumstances, this cooling rate can only be increased. A new temperature for the damper is found in performing this cooling calculation. It can be used to determine the heat exchange that results in the initial temperatures of the system for the next time step.

3.3 Heat Exchange Between the Rail and Damper

Using the concept of conservation of energy, it can be assumed that (under ideal conditions) the net energy is equal to zero:

$$Q_{Steel} + Q_{Aluminum} = 0 \tag{5}$$

Given this assumption and the additional assumption that the final temperature of each material is equal for each time step, we can calculate the final temperature (T_f) for the system at each time step:

$$m_S c_S (T_f - T_{iS}) + m_A c_S (T_f - T_{iA}) = 0 \tag{6}$$

$$T_f = \frac{m_S c_S T_{iS} + m_A c_A T_{iA}}{m_S c_S + m_A c_A} \tag{7}$$

Q is energy in the form of heat, *m* is the mass of each respective material, *c* is the specific heat of each material, and *T* is the initial and final temperatures. The initial heat of the steel rail and aluminum was calculated in the previous step. For the conduction occurring between the rail and the damper, the mass drives the design, as the mass must be sufficient to draw heat away from the rail at the most significant rate possible. As the damper will cover most of the surface area of the rail, it is assumed that most of the heat will be focused on the head of the rail, and thus the majority of the damper mass should be placed as close to the railhead as possible.

This final system temperature for each time step is then compared to the temperature of the control rail at the same time step to obtain a temperature differential. Calculations for the temperature differential over time using the base damper design can be found in [3]. Although these calculations are derived from proven theory, the simultaneous nature of the processes makes the timeframe of the techniques challenging to calculate. It will be essential to perform experiments and use finite element modeling to validate the calculations. Successful calculations will be indicated by experiments and models that yield proportional cooling results, though the time scale may be skewed. Table 1 lists the constants used throughout the numerical modeling process.

Table 1. Constants are used for numerical modeling.

Constant	**Value**
Specific Heat of Aluminum c_A	0.89 J/g-K
Specific Heat of Steel c_S	0.51 J/g-K
Heat Transfer Coefficient α	0.001

4. Parametric Study

4.1 Base Damper Dimensions

Temperature differential calculations were first performed considering a solid aluminum block's base damper design. Once these were obtained, a parametric study was conducted by incrementally changing the dimensions of the damper. Because mass will affect the heat transfer capabilities of the damper and assembly coupled with the surface area of the damper affects its cooling capabilities, determining the ideal ratio of mass to surface area will be the priority. It was determined that to maximize the heat transfer between the rail and damper, the height of the damper should be maximized to rest along with the entire web of the rail with a height of 4 inches and a 1-inch depth. Once a satisfactory temperature differential resulted from these changes to dimensions, the heat sinking properties of the damper became the focus.

4.2 Fin Width

Maximizing surface area while minimizing mass was the goal of creating an appropriate number of fins with heat sinking properties. The thickness of the fins was established at 1/8”, as this was the thinnest feasible fin that could maintain the necessary durability for handling, installation, and maintenance of the damper. Thicker widths increase the surface area of each fin but limit the number of fins that can fit per linear foot and thus determine the maximum surface area, as shown in Figure 3.

Fin widths of 1/8” were also the most effective at producing the maximum temperature differential between the rail with the device and the control rail. These changes in temperature differential as it varied by fin width can be observed in Figure 4. The spacing between the fins was set as at least twice the fin width to provide proper air circulation for natural convection.

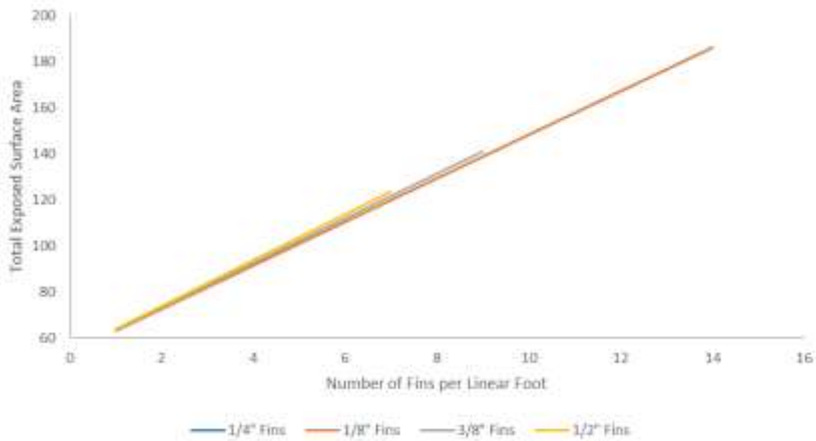

Fig. 3: Fin width to the surface area.

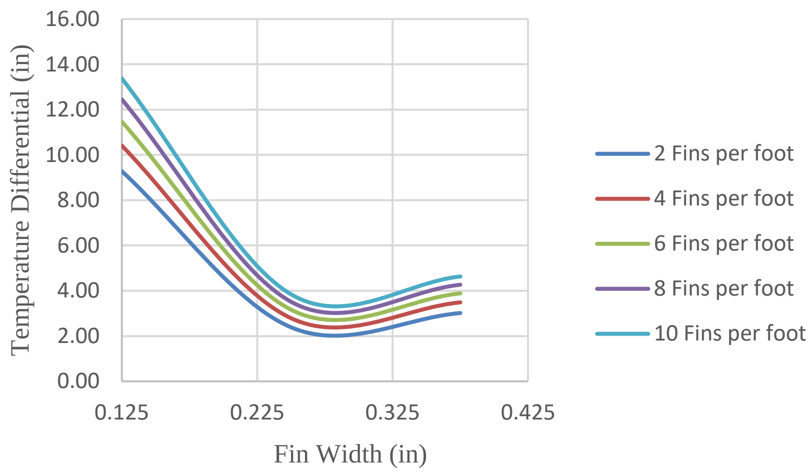

Fig. 4: The variance of temperature differential with fin width.

4.2 Fin Depth

Once the optimum fin width of 1/8" was studied, the study focused on choosing the best depth of the fins. It was discovered that as the depth of the fins increased, the temperature differential first decreased before bottoming out and then slightly rising again. Because the thermodynamic principles involved use exponential equations that rely on the ratio between surface area and mass, the temperature differential plots for fin width and fin depth follow their respective curves that are not linear. Figure 5 depicts the changes in this differential as the fin depth changes. It was determined that a depth of 3/4" provided the optimum surface area to the mass ratio when coupled with a depth of 1/8", resulting in the best temperature differential. Thus, a depth of ¾" is to be used.

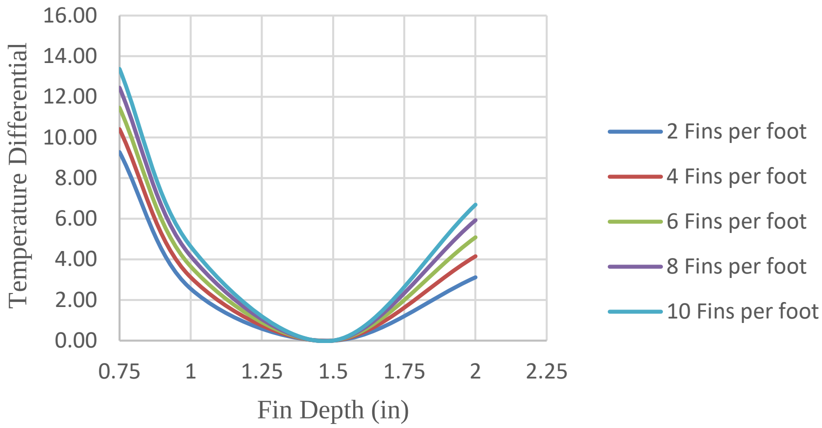

Fig. 5: The variance of temperature differential with fin depth.

4.3 Optimum Fins per Foot of Length

The final parameter studied was the number of fins per foot of damper length. The design of the damper is to be one foot long, but since it may be modified for locational needs, the fins are designed as a number per foot of length. Figure 6 demonstrates the increase in temperature differentials as the number of fins per foot increases.

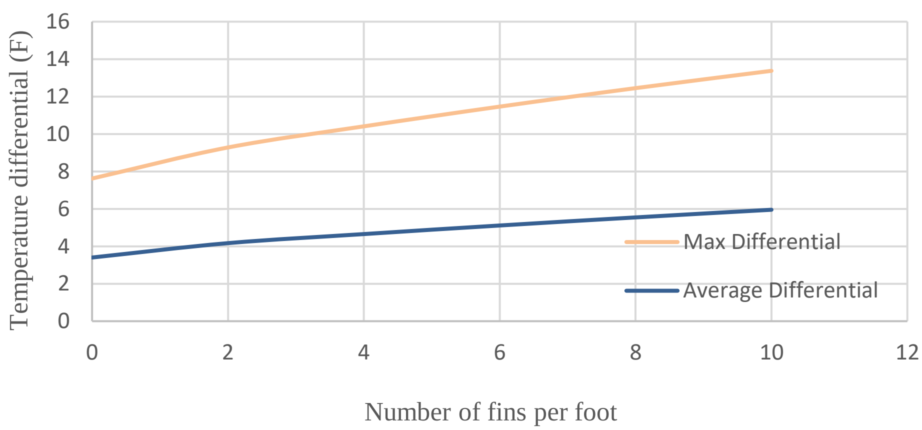

Fig. 6: The variance of temperature differential with an increase in ins per Foot.

The maximum number of fins studied was 10 per foot. This limit was set with the intention of adequate cooling of the fins. As convection drives the cooling process of the damper, it is essential that fins are not too closely spaced and insulating rather than cooling the surface along with the damper. Because the most excellent maximum and average temperature differentials were observed with a maximum of 10 fins per foot, this was chosen for the design.

4.4 Results of Parametric Study and Final Design

The goal of the parametric study is to decide upon a final design with dimensions and a standard temperature differential. This temperature differential was then compared to the temperature differential of the base damper design to infer an expected percent increase in the differential between the base design and that with more heat sinking properties. Table 2 lists the percent increases of the maximum temperature differentials and the average temperature differentials for each fin increase.

Table 2. Percent increase of temperature differentials over base damper design.

Number of Fins per ft	Percent Increase of Max Differential	Percent Increase of Average Differential
2	22%	22%
4	37%	37%
6	50%	50%
8	63%	63%
10	75%	75%

The final design of the damper includes ten 1/8”-wide ¾”-deep fins per foot of length and has a temperature differential increase of 75% over that of the damper design with no fins. A cross-sectional and profile view of the damper can be seen in Figure 7, and the finished dimensions for the damper are listed in Table 3. A cross-sectional view of the damper with dimensions is depicted in Figure 8. Calculations performed for the parametric study and the final damper design can be found in [3].

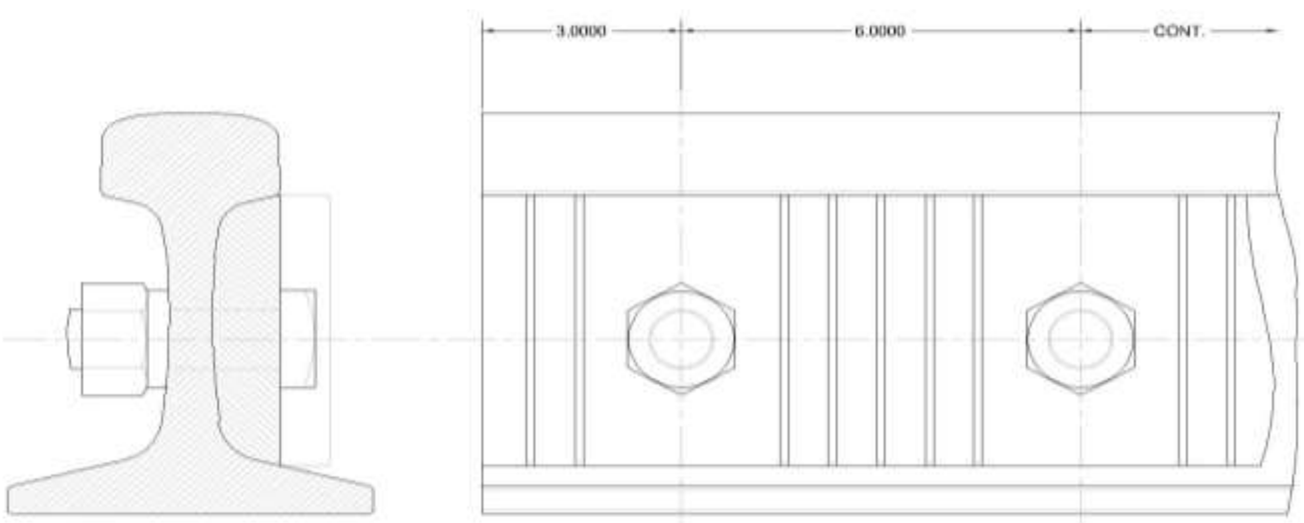

Fig. 7: Drawing of final damper design.

Table 3. Dimensions of final damper design with ten fins.

Component	Dimension (in)	
Damper Block		
Overall Height (*h*)	4.3112	in
Overall Depth (*d*)	0.8800	in
Length (*l*)	12.000	in
Cross Section Area	3.3454	in^2
Damper Fins		
Depth (d_f)	0.7500	in
Height (h_f)	4.3112	in
Width (w_f)	0.1250	in
Fillet Radius	0.0625	in
No. of Fins	10	
Exposed Surface Area	109.88	in^2
Total Volume	44.184	in^3
Mass	4.4184	lb-m

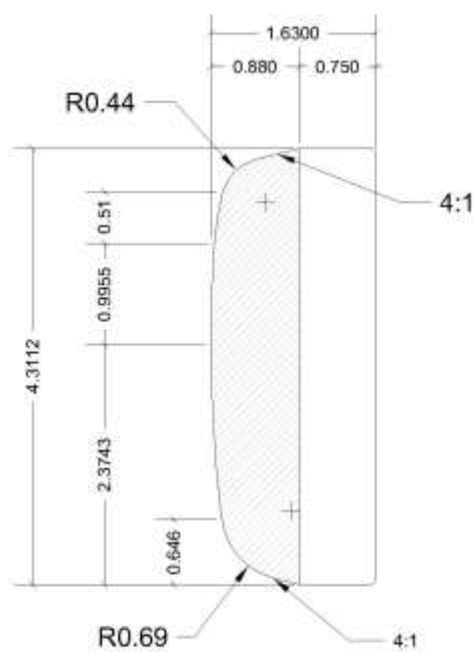

Fig. 8: Cross-section of final damper design with dimensions.

5. Fabrication, Installation

As designed, it is recommended that the damper be installed between ties (about every three feet) in locations where summer temperatures are moderate to high. Spacing may be reduced in average temperatures. Additionally, dampers should be installed in any climate at critical buckling locations along the track, such as the bottom of hills, around curves, and the entrance and exits of bridges. As designed, the damper may be installed in place by a railroad technician by bolting the device into a drilled hole in the steel. It is also possible to pre-install the device along the rail during fabrication before the rail's installation, which would reduce the installation cost. The design calls for Grade 1 aluminum, but further studies may analyze the suitability of recycled aluminum, which would reduce its price per unit.

6. Future Work Recommendations

The authors recommend implementing the damper design described herein. As designed, rail temperatures during the day's heat may be managed, resulting in reduced heat accumulation of heat from day to day during warmer months. Minimizing heat fluctuations during the day allows the rail to reach a lower temperature during the night. In addition to reducing the likelihood of buckling due to thermal stress, managing extreme temperatures in the rail may preserve its physical qualities and its neutral temperature range.

Although the implementation of this device would serve to mitigate thermal stresses as designed, it is further recommended that the benefits of the design could only be increased should it be combined with a device like the one it was inspired by the RTCS used in a previous Department of Transportation study [2]. However, the RTCS device is not a solid block of aluminum but a rectangular aluminum channel filled with a classified heat effusive material that undergoes a phase change when heated, therefore "soaking" substantial amounts of energy acquired from heat transfer. Because of the design and material held within the RTCS, the damper described here cannot surpass its cooling abilities. However, the results modeled by numerical modeling suggest that in creating a damper that performs as a heat sink with fins, the cooling of the rail can be increased by up to 75% over the cooling anticipated by a damper without fins. The RTCS device exhibited a maximum temperature differential of about 20^{O}F. Therefore, it is inferred that adding fins as designed would increase the temperature differential of a device similar to the RTCS by approximately 15^{O}F. Figure 9 illustrates the temperature differential of the RTCS, the proposed damper with fins, and finally, a combination of the two, where the differential of the RTCS is increased by 75% [3]. A reduction of this magnitude would mean that the rail temperature would reduce from about 130^{O}F to about 95^{o} F, which is the ambient temperature set for the numerical model. A combined device would significantly reduce thermal stresses and deformations as it would reach an efficient cooling rate within hours. This reduction in the thermal stresses would prevent buckling and minimize train derailments, thus achieving the goal of this study.

7. Conclusion

This paper proposes a thermal damper design to mitigate the buildup of excessive axial forces that can develop in continuously welded rails (CWR). Previous research into methods of measuring and dissipating such forces and governing codes and constraints was briefly covered, culminating in the final design of an aluminum heat sink. Each aspect of the thermal path was explored and used to influence the physical properties of the damper. A mathematical model was developed and used to perform a parametric study on the properties such as fin dimensions and spacing. The final design maximizes the contact area with the rail web with 10 1/8" fins and provides a 75% greater temperature differential than the base design. Future work includes developing a finite element model of the rail and damper system to corroborate the results and allow quick design alteration. Furthermore, an experiment was designed that closely mimics previous work done in the thermal dampening of rails to provide a credible framework for analyzing the efficacy of the damper and serve as a check for finite element modeling. Finally, recommendations for a further improved damper are presented, seeking to maximize the damper's cooling.

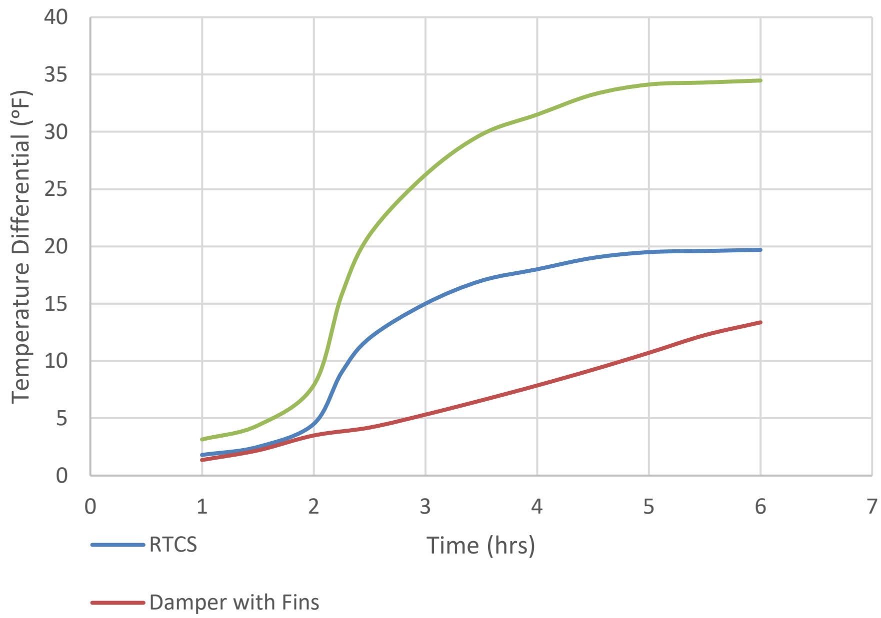

Fig.9: Temperature differentials of RTCS and proposed devices.

Acknowledgments

The authors would like to thank Konrad Crockford, Director of the Compliance Division, and Bradley Fuchs, Railroad Track Safety Inspector from the North Dakota Department of Transportation (NDDOT), for their invaluable discussions and support throughout this study.

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