Proceedings of the Seventh International Conference on
Railway Technology: Research, Development and Maintenance
Edited by: J. Pombo
Civil-Comp Conferences, Volume 15, Paper 9.7
Civil-Comp Press, Edinburgh, United Kingdom, 2026
ISSN: 2753-3239, doi: 10.4203/ccc.15.9.7

Liquid-Cooled Disc Brakes

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Abstract

Railway brakes are designed to absorb a great amount of energy and therefore are heavy non-suspended components. Their mass depends on the cooling technology, even more if they are subject to successive load cycles. The major advantage of liquid cooling is that it significantly reduces the maximum temperature of the brakes and greatly shortens the time needed for them to return to ambient temperature after a heavy braking. The biggest challenge with liquid cooling is circulating the hot and cold water. The pipes can easily pass through the interior of a hollow shaft, but that rotating tube, integrated with the shaft, needs to be connected with another coaxial, non-rotating tube, integrated with the train's structure. Since there are two separate flows, at different temperatures, the more simple arrangement is to dispose the inlet pipe, of fresh water, in one extremity and the outlet pipe, of hot water, in the opposite extremity. The numerical simulations suggest that this novel solution is promising, but it requires of course experimental testing.

Keywords: liquid-cooled brake, disc-brake heating, failure modes, lightweight vehicle, numerical simulation, braking safety

1 Introduction

In normal operation, electrical motors can decelerate a train with regenerative braking, but if they fail or if a stronger deceleration is necessary, disc brakes should be an effective alternative. Can we reduce their mass? This paper addresses the question, with a particular light-train in mind, running at 250 km/h, with two disc brakes in each axle and a total train mass of 6.5 ton per axle.

The research first considered traditional steel-discs (cf. [1,2]), with scarce ventilation, able to perform a single emergency braking, on the assumption that they would rarely be significantly used and, in these occasions, a long cooling time could be accepted, before the train resumes a normal high-speed journey. The research proceeded with highly ventilated discs, to reduce mass and make the brakes available for successive operations [3]. Higher air velocities are important because equation (1) of the appendix (and indirectly equation (2)) shows that the convection coefficient increases almost linearly (with power 4/5) with the Reynolds number and therefore with the air velocity. A number of features was explored, like internal fins acting as blades of a radial fan, an aerodynamic intake to take profit of the train speed, a volute to recover the dynamic pressure, a suction scoop at the exit section of the discharge pipe and the addition a high-conductivity material in the inner layer and the fins. The performance was very good but the complexity too great, in particular because several moving parts were needed. For instance, the fan effect was useful to cool the discs more efficiently, but most of the time would produce an unnecessary resistance if the air flow through the brakes was not reduced to zero. The third research line, that interests the present paper, regards liquid-cooled discs, with an outer heat exchanger and a pump. The performance seems surprisingly good, with the operational temperatures always low and the brakes cooling down very quickly, even with an intense usage.

2 Liquid-cooled brakes

The interest of liquid cooling arises from the fact that its convection coefficient is significantly higher than that of air cooling. Water, or another liquid, could reach the brakes through pipes inside the axle, refresh the discs of the brakes and release that heat in a heat exchanger outside the axle. This arrangement facilitates a more rapid cooling after braking has ended, because water would keep flowing, driven by a pump. In contrast, the highly ventilated solution —using the brake's rotation as a fan, recovering the dynamic pressure in a volute, adding an air intake and an air exhaust— does not contribute to cooling once the train comes to a stop.

In one realization of the liquid-cooling setup, each brake would be made of two parts with grooves hollowed in their inner faces, in a mirror-like design, to form channels when the two discs are closed one against the other, as represented schematically in Figure 1. Fresh liquid would come in through the void of the axle, flow through the channels of the disc brake and be discharged to the outflow pipe inside the axle.

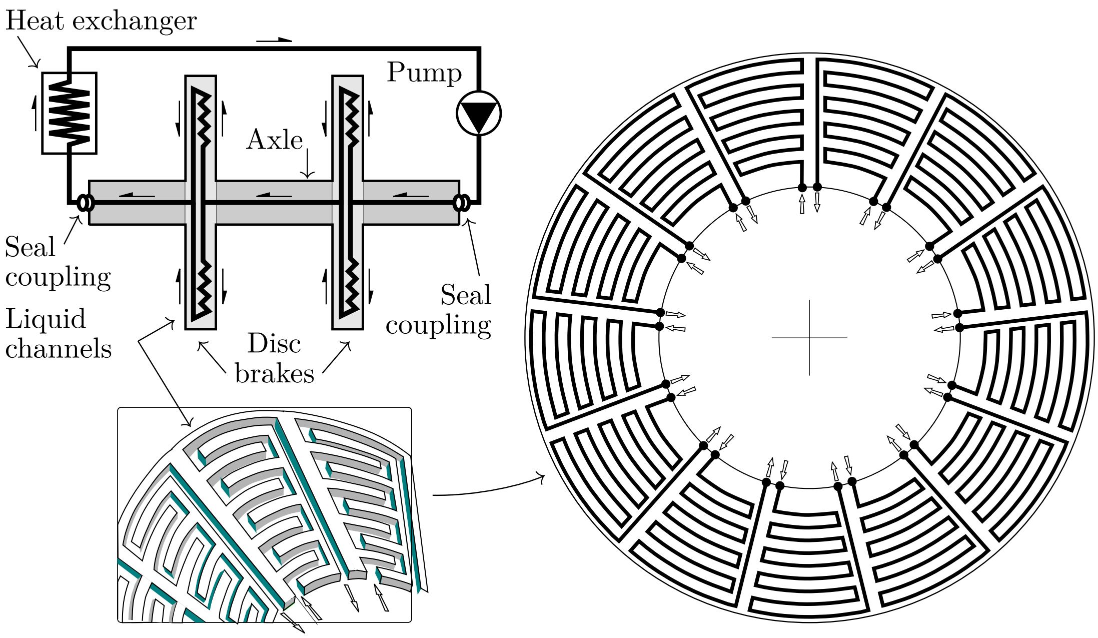

Figure 1: Scheme of liquid channels through the central part of the brake disc. In the lower-left corner, an image of a half-disc.

The biggest challenge is circulating the cold and hot liquid into and out of the axle. The flow inside a hollow axle is easy to arrange, but the connections to the outside, must likely on the shaft's extremities, would connect a rotating tube, integrated with the shaft, to another coaxial, non-rotating tube, integrated with the train's structure. Since there are two separate flows, at different temperatures, the more simple arrangement is to dispose the inlet pipe, of fresh water, in one extremity and the outlet pipe, of hot water, in the opposite extremity.

The reference material of the discs is carbon steel AISI 1010, whose maximum working temperature is $T_{\max} = 600$ °C, in order to avoid a significant degradation of the mechanical strength (the melting temperature is 1267 °C). Other properties are: thermal conductivity $k = 63.9$ W/(mK), specific heat $c = 434$ J/(kgK) and thermal diffusivity $\alpha = 1.88 \times 10^{-5}$ m^2/s. It is worthwhile noting that the maximum temperature of the pads may be even more restrictive requiring heavier discs. As a preliminary approach, the liquid considered in the simulations is fresh water.

Let us assume $n_{ch} = 13$ channels per disc, as in Figure 1, with a $a_{ch} = 6 \times 6$ mm^2 cross-sectional area, a $\ell_{ch} = 1.4$ m length in every one of the 13 channels where water flows at 2 m/s. The total mass flowrate of water is $\dot{m} = 0.94$ kg/s per disc, roughly 2 kg/s per axle. The channels' Reynolds number based on the hydraulic diameter ensures that the flow is turbulent:

$$Re_D = \frac{v\,D_h}{\nu} = 1.2 \times 10^4,$$

where ν is the water kinematic viscosity and v the average velocity in the channels. The water properties are taken from [8].

The convection coefficient in the inner channels can be estimated from the Dittus–Bölter and the Gnielinski correlations for the Nusselt Number, valid for fully developed turbulent flow in non-circular pipes

$$Nu_D = 0.023\, Re_D^{4/5} Pr^n = 75.8 \tag{1}$$

wiht $n = 0.4$ because the surface is hotter than the fluid. Correlation (1) is valid in the range $0.6 < Pr < 160$, $Re_D > 10^4$, $L/D_H > 10$ and applies to hydraulically smooth surfaces. Properties must be evaluated at the fluid temperature. The convection coefficient for this Nu_D is $h = 8 \times 10^3$ W/(m^2 K). The Gnielinski correlation is more accurate in some cases:

$$Nu_D = \frac{(f/8)\,(Re_D - 1000)\,Pr}{1 + 12.7\,(f/8)^{1/2}\,(Pr^{2/3} - 1)}. \tag{2}$$

Properties must be evaluated at the fluid temperature. The correlation is valid in the range $0.5 < Pr < 2000$, $3 \times 10^3 < Re_D < 5 \times 10^6$. It applies primarily to hydraulically smooth tubes, for rough tubes the heat transfer coefficient increases with wall roughness. As a first approximation, the same correlation can be used with friction factors taking into account the surface roughness, which will be assumed $\varepsilon \simeq 46\ \mu$m. The friction factor f may be obtained iteratively from the Colebrook correlation using the Haaland correlation as starting approximation. The result is similar to (1).

The overall heat transfer area of the channels is $A_{ch} = 0.43$ m^2 per disc. If the total thickness of one disc is $\varepsilon_b = 10$ mm, and the radii of the disc are $R_1 = 0.21$ m and $R_2 = 0.35$ m, the mass of an hollowed steel-disc is $M_b = 14.2$ kg (that of a solid disc with the same external dimensions would be $M_b' = 19.3$ kg).

A note on the cooling of the external surface of the discs. The relative air velocity over the external face of the discs is difficult to estimate because of the rotation of the disc. In the open air, without rotation, the relative air velocity would be the train's velocity in the ground reference frame. In the open air, with rotation, the air velocity would range between 0 and $2\,v_{\text{train}}$. As a first estimate, the reference Reynolds number is $Re_L = v_{\text{train}}\, L/\nu$, where ν is the air kinematic viscosity and the reference length $L \simeq D_2$ is the disc's diameter. According to the Chilton-Colburn correlation, the Nusselt number of turbulent flow over isothermal external surfaces is:

$$Nu_L = 0.037\, Re_L^{4/5} Pr^{1/3}, \qquad \text{where } Nu_L = \frac{h\,L}{k}. \tag{3}$$

The air conductivity is k and Pr the Prandtl number, the air properties are taken from [8]. The correlation is valid in the range $0.6 < Pr < 60$ and $Re_L > 10^8$. Properties must be evaluated at the film temperature. The whole flow is considered turbulent due to the vibration of the surfaces, the sharp edges and the rotation that trigger turbulence since the leading edge[1]. Correlation (3) provides only a first approximation, probably with errors of the order of 25% according to [6], section 7.2.6.

[1]For uniform heat flux, instead of uniform surface temperature, the Nusselt number would be approximately 4% larger; the model of uniform temperature is a closer approximation.

Note on radiative cooling of the external faces of the discs. Radiation plays a minor role in the cooling of the discs. The radiative power of a surface is $\dot{W}_{rad} = A_s\, \varepsilon_s \sigma T_s^4$, where A_s is the area of the surface, ε_s its emissivity, T_s its absolute temperature and $\sigma = 5.67 \times 10^{-8}$ W/(m^2 K) the Stefan-Boltzmann constant.

The total area of the two external faces is $A_s \simeq 0.25$ m^2 and its emissivity[2] is of the order of $\varepsilon_s \simeq 0.1$. When the surface is at 500 °C and the surrounding is at 25 °C, the radiative power of the external faces is only 498 W. For this reason, radiation is neglected in this paper.

2.1 Normal operation

In normal operation, the amount of water in the circuit is not relevant, but matters for one of the failure modes modelled in subsection 2.3; in that scenario, the circuit of each disc will contain at least 4 kg of water.

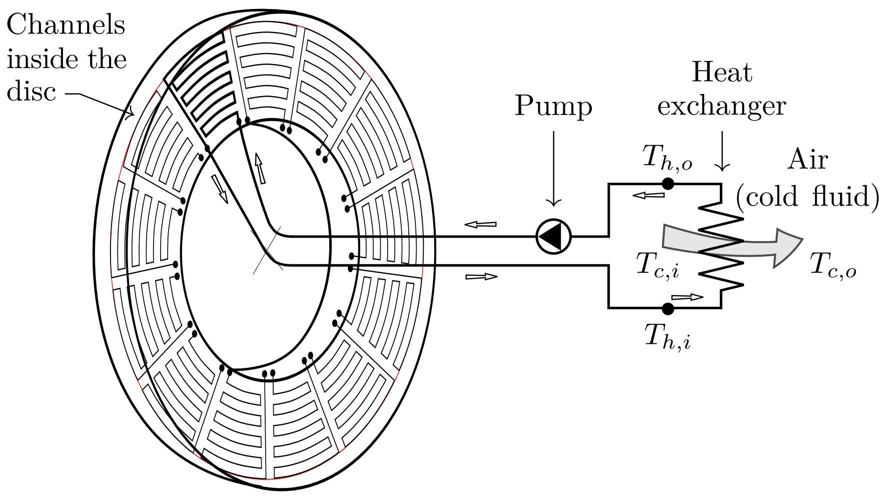

Figure 2: Nomenclature of the liquid cooled brake, with pump and heat exchanger. All channels are supposed to be similarly connected to the water cooling circuit, with all flowrates approximately balanced.

Heat exchanger. In normal operation, the hot water is recirculated by a pump and cooled in a crossflow heat exchanger, similar to current automotive intercoolers. The water enters the heat exchanger at $T_{h,i}$ and exits at $T_{h,o}$; the air enters at $T_{c,i}$ and exits at $T_{c,o}$, see Figure 2; T_s is the temperature of the steel surface of the heat exchanger in contact with the water. The design mass flowrate of water is $\dot{m}_{\mathrm{w}} = 1$ kg/s and so its heat capacity rate is $C_h = \dot{m}\, c_{p\mathrm{w}} \simeq 4184$ W/K; the air flowrate may be about 4

[2]It is difficult to know beforehand the accurate emissivity of the steel surfaces of the discs. In these notes, we assume that the surface is polished and therefore the emissivity is low. Up to about 1000 K, for steel and lightly oxidised steel, $\varepsilon_s = 0.05 \sim 0.25$; for iron oxide, $\varepsilon_s = 0.35 \sim 0.60$; for stainless steel oxidised at high temperature, $\varepsilon_s = 0.6 \sim 0.7$. See Fig. 3 of [9].

times greater, for the heat capacity rates of the cold fluid and the hot fluid to be similar, $C_h = C_c$. The heat exchanger has an effectiveness value[3], commonly found in current crossflow heat exchangers [6], say $\varepsilon = 0.7$. The water outlet temperature in steady state is

$$T_{h,o} \;=\; T_{h,i} - \varepsilon\,\big(T_{h,i} - T_{c,i}\big). \tag{4}$$

Flow through the discs' channels. Assuming the steel surface in contact with the water is at uniform temperature, the meaningful average temperature-difference is the logarithmic-mean temperature difference (in contrast to an arithmetic-mean temperature difference), due to the exponential nature of the water-temperature variation. The log-mean temperature difference [6] is

$$\Delta T_{\log} \;=\; \frac{\Delta T_o - \Delta T_i}{\ln\big(\Delta T_o/\Delta T_i\big)}, \qquad \text{with } \Delta T_o = T_s - T_{h,o} \text{ and } \Delta T_i = T_s - T_{h,i}. \tag{5}$$

The convection heat transfer rate is

$$\dot{w}_{\text{conv}} \;=\; hA_{\text{wet}}\Delta T_{\log}, \tag{6}$$

where h is the average convection coefficient and A_{wet} the wet area of all the channels. The power absorbed by the water flowing through the brake channels is

$$\dot{w}_{\text{water}} \;=\; \dot{m}_{\text{w}}\, c_{p\text{w}}\big(T_{h,i} - T_{h,o}\big). \tag{7}$$

Remark that $T_{h,o}$ denotes the water outlet temperature of the heat exchanger and thus is the channels' inlet temperature; likewise, $T_{h,i}$ is the water inlet temperature at the heat exchanger, i.e., the channels' outlet temperature. Refer to Figure 2. A steady-state (or quasi steady-state) energy balance equates $\dot{w}_{\text{conv}}$ and $\dot{w}_{\text{water}}$, (6) and (7). Isolating $T_{h,i}$ and $\Delta T_{\log}$:

$$T_{h,i} \;=\; T_s \;-\; \frac{T_s - T_{h,o}}{\exp\left(\dfrac{hA_{\text{wet}}}{\dot{m}_{\text{w}}c_{p\text{w}}}\right)} \tag{8}$$

and

$$\Delta T_{\log} \;=\; \big(T_s - T_{h,o}\big)\, \frac{1 - \dfrac{1}{\exp\left[\big(hA_{\text{wet}}\big)/\big(\dot{m}_{\text{w}}c_{p\text{w}}\big)\right]}}{\big(hA_{\text{wet}}\big)/\big(\dot{m}_{\text{w}}c_{p\text{w}}\big)} \tag{9}$$

Time integration. The blue lines at the right-hand side of Figure 3 display the temperature evolution of the hot face of the brake (solid line) and the outlet temperature of the cooling water (dashed line). The hot face of the brake reaches the maximum temperature, 203 °C, at 6 s of elapsed time since braking starts and then that surface cools down to about 41 °C when the train stops.

[3] The effectiveness ε is the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate, $q_{\max} = C_{\min}(T_{h,i} - T_{c,i})$, where $C_{\min}$ is the minimum heat capacity rate and $T_{h,i}$ and $T_{c,i}$ are the hot inlet and cold inlet temperatures.

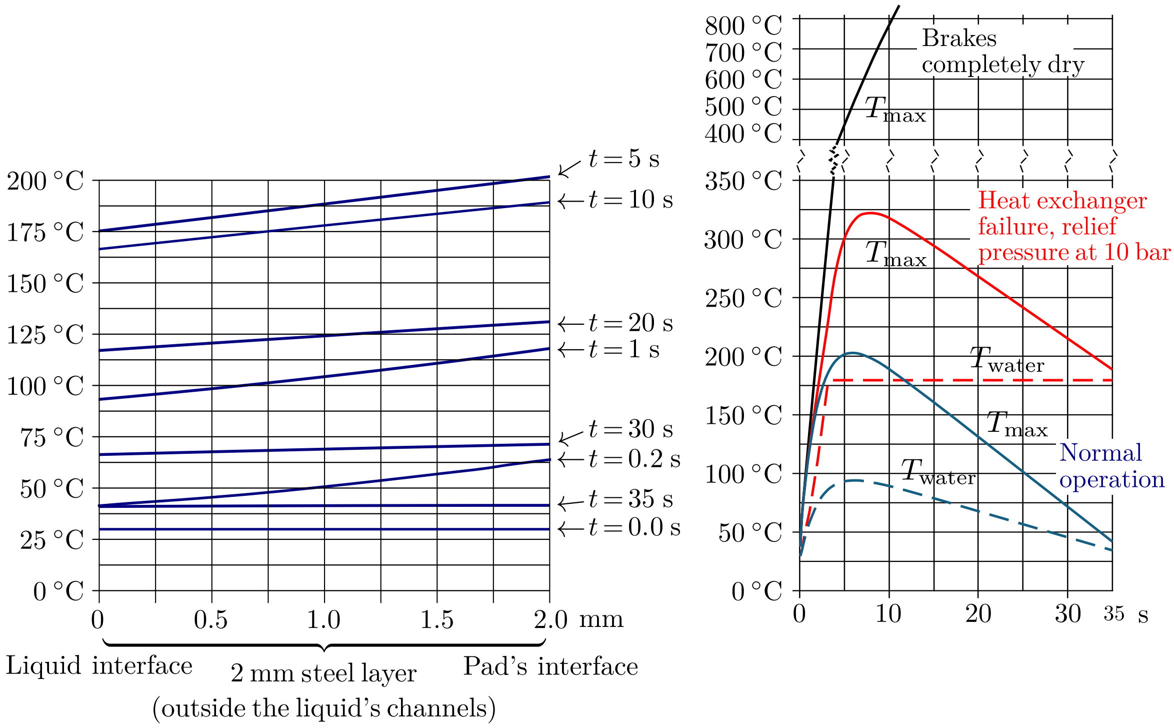

Figure 3: Left: temperature distribution in the 2 mm steel layer outside the channels of liquid in normal operation. Right: time evolution of the maximum disc-temperature and water temperature in normal operation (blue); with water recirculating but the heat exchanger completely unable to cool it (red); and a water leakage that leaves the brakes dry (black). Brake-only 2 m/s^2 deceleration from 250 km/h to complete stop.

The maximum temperature of the water is not high and the brake ends up at almost atmospheric temperature (the simulations considered an ambient temperature of 30 °C). The peak temperature of the water is 94 °C, also reached after 6 s from beginning. Although the temperature of the water stream remains below the boiling temperature at atmospheric pressure, the excess temperature (surface temperature above the saturation temperature of the water) may reach 100 °C, giving rise to vapour bubbles on the steel surface, which grow, detach from the surface and subsequently condense in the liquid. Most of the water stream would remain in liquid state at atmospheric pressure and if the system admits some overpressure, its temperature could exceed 100 °C without boiling, the temperature allowance depending on the overpressure. For instance, the saturation temperature at 5 bar is 156 °C.

Boiling heat transfer. The high heat transfer coefficients associated with boiling make it attractive, specially in the peak moments when thermal performance is most required. The centrifugal force of the rotating brake significantly increases buoyancy (about 700 times gravity at 250 km/h) and probably enhances the boiling dynamic [6]. Rohsenow correlation for nucleate boiling (*ibid.*, equation 10.5) relates the convection coefficient with the square root of gravity, thus the brake rotation would increase convection up to $\sqrt{700} \simeq 25$ times. Kutateladze and Zuber (*ibid.*, equation 10.6) state that the critical heat flux of nucleate boiling depends on the $\frac{1}{4}$ power of gravity, i.e., the critical heat power for a train at 250 km/h would be about 5 times the heat power without rotation[4]. Typically, nucleate boiling exists for excess temperatures up to 30 °C, but this threshold is probably lower under the mentioned centrifugal acceleration. In nucleate boiling, first, isolated bubbles form at nucleation sites; then, as the excess temperature increases, more nucleation sites become active and more bubbles are formed, which interfere and eventually coalesce. In these stages, the heat convection coefficient increases up to several hundred times the value without boiling, till the critical excess temperature is reached (which is about 30 °C under standard gravity, see namely chapter 10 of [6]).

Centrifugal acceleration, water pressure, flow balance between the several channels in parallel and the peculiar geometry of the channels make it difficult to anticipate the thermal consequence of boiling. On the other hand, since boiling would probably improve the brake's thermal behaviour, the present simulations, that did not address it, provide conservative estimates of the brakes' performance.

The left-hand side plot of Figure 3, representing the temperature distribution along the 2 mm thick external layer of the discs, between the channels region and the discs' outer surface, at several elapsed times after the start of braking, shows that the temperature of the solid part is roughly homogenous. These numerical simulations neglected the cooling flux of the discs' outer surface to the air and the fraction of braking power absorbed by the pads, because these two contributions are less important and not accounting for them introduces an error on the safe side.

Water recirculation by means of a pump. In the conditions assigned above, the Reynolds number of the flow in the channels is 1.2×10^4. The friction coefficient of the channels is $f \simeq 0.03$, the pressure loss coefficient of $90°$ sharp-angled elbows at this Reynolds number [7] is $K = 1.3$, each channel exhibits 18 such elbows in series, the total pressure-loss in each loop, neglecting the interference between successive elbows, is

$$\Delta p \; = \; \left(f \, \frac{\ell_{ch}}{D_{ch}} + 18K \right) \frac{1}{2} \rho v_2^2 \; = \; 6.1 \times 10^4 \; Pa.$$

The other pressure losses of the circuit, mainly due to the heat exchanger, may amount to 5×10^3 Pa. The volume flowrate assigned above for the two brakes of an axle is 2×10^{-3} m^3/s. The pumping power is 165 W, assuming an hydraulic efficiency of 80%. Rounding the elbows can reduce the pressure loss and the pumping power, although the pumping power required is not significant, specially because the pumps need only run for a relatively short time.

[4] [6], section 10.4.2, also remarks that the critical heat flux depends strongly on pressure, mainly through the pressure dependence of surface tension and the heat of vaporization. Cichelli and Bonilla (ref. [13] of [6]) have experimentally demonstrated that the peak flux increases with pressure up to one-third of the critical pressure, i.e., up to 74 bar for water (the critical pressure of water is 221 bar).

Water pressure and thermal expansion allowance. The overall pressure loss requires the circuit to be pressurized to prevent cavitation and early boiling inception in the hot parts. In the calculations presented in this text, the expansion of the water and the metal parts were not taken into consideration. The linear thermal expansion coefficient for common inox steel is about $\alpha = 1.5 \times 10^{-5}$ /°C. A ΔT temperature increase expands the linear dimensions of the steel parts by a factor $(1 + \alpha\Delta T)$ and their inner volume by a factor $(1 + \alpha\Delta T)^3 \simeq 1.003$ for $\Delta T = 70$ °C. From 30 °C to 100 °C, the specific volume of liquid water [8] changes from 0.1004 m^3/kg to 0.1044 m^3/kg, a 1.040 increase. This means that, if there is no provision to allow a greater expansion of the solid parts, heating naturally increases the water pressure, even more if some solid parts of the circuit heat with delay relative to the water. So, the point is not only the initial cold-stage overpressure, but also the allowance of internal volume increase in order to bound that overpressure when the water is hot.

Mechanical design of the disc. The two halves of the disc can be connected using interlocking pins and screwed together at the periphery. If needed, a reinforcing collar can be press-fitted to the outer perimeter using a heat-shrink process. Figure 4 shows a schematic representation of this arrangement.

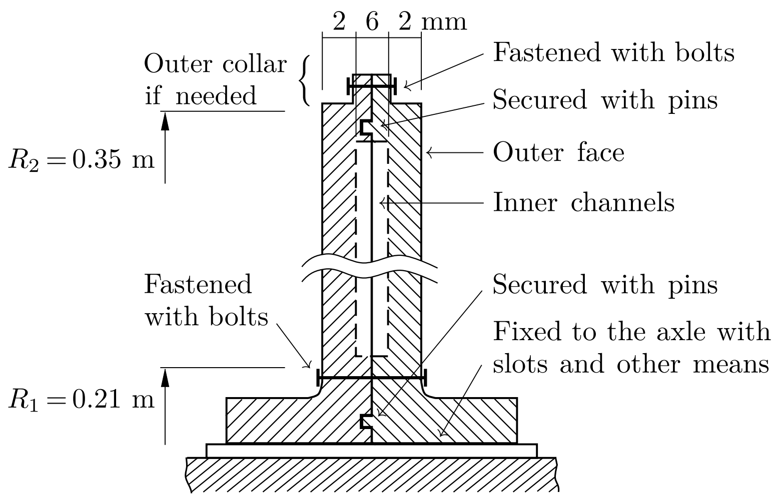

Figure 4: Two halves of the disc connected with interlocking pins and screwed together.

During braking, the pins and screws are subjected to shear; the clamping force is supported by the contact surfaces in the channels' region. The clamping force is equal or almost equal on both sides, so that the bending moment is expected to be negligible.

Mechanical design of the coupling between the disc-channels and the outer heat exchanger. The liquid reaches the disc brakes flowing axially through the axle, the connection between that pipe and the outer heat exchanger is best made through the axle's extremities. The difference in rotational velocity of the two coaxial pipes is

25 rps (1500 rpm) when the train runs at 250 km/h. To avoid having two opposing flows at each end of the axle, all the inlet cold flow feeding the two brakes can enter through one end and all the exhaust hot flow can exit through the opposing end. If we limit the local water velocity at the couplings to 3 m/s, the internal diameter of the couplings is 30 mm and the difference in velocity of the two pipes is 2.3 m/s.

The industry offers a wide range of sealing technologies. Radial lip seals are the most common type of rotary seals; mechanical seals can be made from a variety of materials including ceramic, carbon and metal and are typically used in high-pressure and high degree of sealing applications; the packing used in gland seals can be made of various materials such as graphite, PTFE, aramid fibre and carbon.

Figure 5 represents one possible embodiment of the coupling between the rotating pipe, inside the axle, and the non-rotating part of the pipe. The non-rotating part of

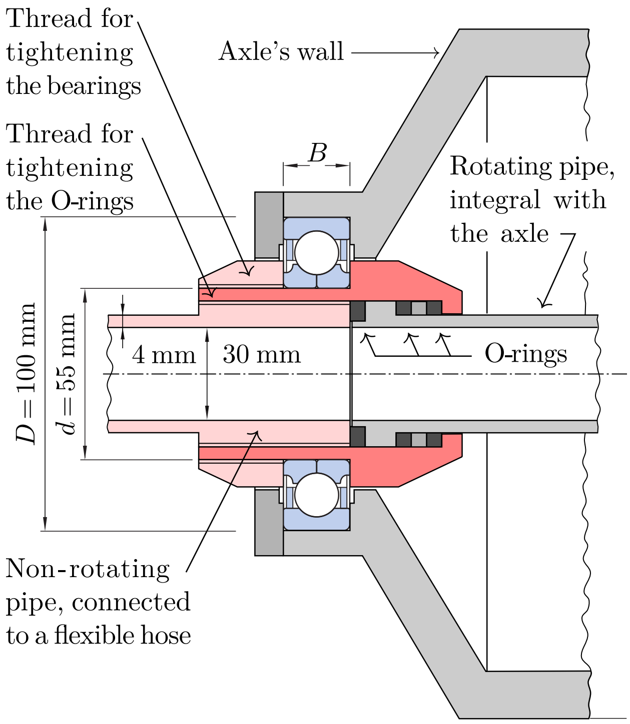

Figure 5: One embodiment of the coupling between the rotating and the fixed pipes.

the pipe (left-hand side of Figure 5) is integral with a inner ring of a bearing. The O-rings represented in the figure can be made of high-performance carbon-filled PTFE, graphite, or tungsten carbide. The part that embraces them and is directly in contact with the inner ring of the bearing can be screwed onto the non-rotating pipe to facilitate adjustment of the contact pressure of the O-rings. The axial distance between the O-rings represented in the figure has the purpose of ensuring a co-axial alignment of the two pipes, maintaining a watertight seal while allowing continuous rotation. A flexible hose can be used on the outer side of the assembly, the non-rotating one (left-hand side of Figure 5), to avoid side loadings and torques on the coupling. A quick search of commercially available solutions confirms that standard rotary unions can handle speeds up to 1500 rpm and pressures ranging from 6 to 64 bar. Let us assume for the pipes an allowable stress $\sigma_{\max} = 3 \times 10^7$ Pa. An overpressure of 30 atmospheres requires a 1.5 mm thickness. A bending moment $M_f = 100$ Nm requires a 4 mm thickness, as represented in Figure 5.

2.2 Mechanical design of the pads with liquid-cooled discs

The pads have similar dimensions to common brake pads, the disc-pad friction coefficient is 0.4, a conservative value according to the literature, and the pad contact-area stated in that table is $0.125 \times 0.2 = 0.025$ m^2.

Representative properties of sintered metal used in brakes operating dry are listed in Table 1. For a -2 m/s^2 deceleration, the braking force at the wheels divided by the number of pads is 3250 N, the braking force at the pad is 5223 N, the clamping force, with the friction coefficients of Table 1 is $11.6 \sim 34.8$ kN. For the contact pressures of that table, the contact area of the pads ranges from 0.0056 m^2 to 0.0338 m^2 and, given the average 0.28 m radius and assuming a 0.14 m development in the radial direction, the arc length of the pads should be between 0.04 m and 0.86 m. This last value, a worst case scenario —with the smallest friction coefficient and smallest contact pressure—, corresponds to an angular development of about 180°. For a less uncommon friction coefficient of 0.25, a contact area of 0.0203 m^2 is enough, even with the smallest contact pressure, and the arc length of the pads would be limited to 0.14 m, which gives an angular development of 30°.

Table 1: Representative properties of sintered metal, dry, [4], Tab. 18.1

Friction coefficient, rubbing against smooth cast iron or steel:	0.15 – 0.45
Maximum contact pressure:	1030 – 2070 kPa
Maximum bulk temperature:	232 – 677 °C

Table 2: Properties of two sintered cermet (ceramic-metal) brake pad materials [5]

	K	ρ	c_p	$\overline{\mu}$ (*) 400–500 kPa	1470 kPa
FMC-11	35 W/(m K)	4700 kg/m^3	478.9 J/(kg K)	0.54	0.42
MCV-50	30.78 W/(m K)	5300 kg/m^3	511.6 J/(kg K)	0.28	0.22

FMC-11: 64% Fe, 15% Cu, 3% SiO_2, 6% $BaSO_4$, 3% asbestos, 9% graphite
MCV-50: 64% Fe, 10% Cu, 5% B_4C, 5% SiC, 5% $FeSO_4$, 3% asbestos
(*) Average friction coefficients in the temperature range from 0 to 800 °C.

According to [5], the cermet materials of Table 2 can withstand temperatures such as 450 °C in medium and 750 °C in heavy braking mode.

With the cermet materials of Table 2, the heat capacity of the pads is negligible, compared to the energy released during a brakes-only operation from 250 km/h to rest. The overall energy dissipated by air-convection around the pads is also negligible during heavy braking. The cooling provided by conduction to the discs and the whole calliper is the leading contribution. Thus, assuming, as in the previous simulations, that *all* the braking power was dissipated at the discs, we get an upper bound of the thermal load effectively transferred to the discs. The importance of cooling the discs is highlighted by the fact that the discs are the main means of cooling the pads. If the maximum discs' temperature is similar to that of the red or blue lines of Figure 3, the temperature of the pads will not reach 350 °C, well below the 450 °C minimum safe temperature of cermet materials.

Another way of looking at the feasibility of the system is to evaluate the product of the clamping pressure by the rubbing velocity. The maximum rubbing velocity is 44 m/s (for a train at 250 km/h); for a clamping pressure of $11.6 \sim 20.9$ kPa, the pressure times rubbing velocity is $510 \sim 920$ kPa.m/s, while the typical values used in industrial shoe brakes with poor heat dissipation is 1000 kPa.m/s for continuous operation and 2100 kPa.m/s for occasional operation ([4], Table 18.3).

2.3 Exam of several failure modes

This subsection examines the consequences of two failure modes of liquid-cooled brakes. Of course, after one of these severe incidents, the train would be sent to the repair shop.

Total water leakage. After a complete water leakage before a heavy braking from 250 km/h to still, the steel discs reach the melting temperature (1267 °C) after 20 s, and, even before 20 s, the steel would lose its mechanical strength and the brakes would become unserviceable. The black line on the right-side plot of Figure 3 displays the temperature evolution of the hot faces of the disc during the initial 10 s. To prevent the discs from failing before the train stops, their mass should be doubled ($M_b = 32.2$ kg for a bulk design temperature of 600 °C); without that mass increase, the discs heat to 800 °C in about 10 s, while the train is still running at 180 km/h. Even with a doubled mass, the discs might be metallurgically injured and require replacement.

The complete drainage may occur in one brake or even in the two brakes of the same axle, but it seems unlikely that the issue would happen simultaneously on all axles. In practice, all the disc-brakes would operate during the first 10 s and, after that, only some brakes would remain operational. If half the brakes are disabled and their axles lose completely any braking capability, from then on, the deceleration would be reduced to one half, say 1 m/s^2. Instead of taking 35 s and 1225 m from 250 km/h to still, the train would take 60 s and 1850 m to stop. Of course, this scenario did not consider the existence of regenerative braking, which could compensate the failure of the brakes.

Heat exchanger fully clogged on the air side. If a complete failure of the heat exchanger occurs —suppose the heat exchanger gets fully clogged on the air side—

and the water keeps circulating, but without cooling, the temperature of the whole brake and that of the water will rise to a dangerous level. The water would reach a very high pressure, vaporize and eventually cause the system to burst.

In such an emergency, a relief valve in the water circuit could limit the pressure, allow some vapour to form and bound the rise of the water temperature.

Keeping the pressure below 10 bar ensures that the water temperature does not exceed 180 °C as long as some water is still liquid. From 30 °C to 100 °C at atmospheric pressure, the liquid water absorbs $\Delta h_{f_1} = 4.184 \cdot 70 = 293$ kJ/kg; from 100 °C to 180 °C along the saturation line, the water absorbs more $\Delta h_{f_2} = 763 - 417 = 346$ kJ/kg; and the complete vaporization at 180 °C absorbs more $\Delta h_{fg} = 2015$ kJ/kg, i. e., a total

$$\Delta h_{f_1} + \Delta h_{f_2} + \Delta h_{fg} = 2654\,\mathrm{kJ/kg}.$$

At cruise speed, the kinetic energy per brake is 7963 kJ, so about 3 kg of water per brake would absorb all that heat.

The red lines of the right-hand side plot of Figure 3 represent the temperature of the discs' hot face (solid line) and that of the outlet water (dashed line) when water keeps flowing in the pipes, the heat exchanger does not cool it and the water pressure is allowed to build up only to 10 bar. In this scenario, the water exiting the brakes reaches the boiling temperature in about 3 seconds; from then on, the relief valve lets vapour exit the system, keeping the water pressure and temperature constant. For a constant braking effort, the power dissipated by the disc-pads friction decreases as the train speed is reduced; so, as cooling efficiency is high, the disc reaches a peak temperature of 322 °C in 7 s elapsed time and has already cooled down to 190 °C when the train finally stops.

In the preferred embodiment, the relief valve is placed in a position such that its outflow is mostly saturated vapour, because this reduces the mass of water lost to the ambient (keeps more water in circulation inside the system), insures that the pump does not cavitate and the brakes receive saturated liquid water instead of vapour. The numerical simulation presented here assume that, even when vaporization occurs, the disc channels are mainly filled with liquid, which is favoured by natural stratification due to the centrifugal force. Indeed, the centrifugal force is greater than 9.8 m/s^2 for train speeds higher than 9.6 km/h (at this train speed, at a 0.28 m radius the discs turn at 1.66 m/s).

The red lines of Figure 3 correspond to the relief valve opening at 10 bar. Although the system loses a significant amount of water, the peak temperature of the steel is only 322 °C, ensuring that the brake functions properly until the end of the braking process.

Once the train has come to a complete stop, the relief valve or other device should let air enter the system, to prevent the internal pressure from dropping significantly below the atmospheric pressure when the water cools, specially if some vapour, previously formed, condenses in the system. Since the heat-exchanger failure is a serious malfunction that makes it impossible for the train to continue its commercial service, all it takes is for a small plate to collapse inward by effect of the drop in internal pressure, to protect the remaining parts.

3 Conclusion. Cooling alternatives for the disc brakes

Three major options were mentioned. A simple and cheap solid disc, not meant to endure successive heavy braking, relying mainly on its thermal inertia; a sophisticated air-ventilated disc with several additional features to improve its performance and reduce its mass; and liquid-cooled discs, with an external heat exchanger. This last one was the topic of the present paper.

The configuration with the best performance is the liquid-cooled disc, because it is lighter and absorbs a greater amount of energy, keeping a lower temperature, but it is also a complex option, with sensitive components. It seems a good lead for future research, but still needs practical testing before it can be put confidently into service. In our opinion, the key feature is the connection between the rotating pipe inside the hollow axle and the non-rotating pipe, integrated with the train structure.

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