FORCED-AIR DIESEL LOCOMOTIVE COOLING: PREDICTION OF NOISE AND ENERGY CONSUMPTION UNDER REALISTIC OPERATIONAL CONDITIONS

Sebastian Knirsch§, Dietmar Mandt§, Uwe Mauch§
Konrad Bamberger#, Thomas Carolus#

§Voith Turbo GmbH & Co. KG
89522 Heidenheim, Germany
sebastian.knirsch@voith.com

#Institut für Fluid und Thermodynamik
University of Siegen 57068 Siegen, Germany
thomas.carolus@uni-siegen.de

ABSTRACT

An important subsystem in most surface transport vehicles is the forced-air cooling module. Under specific operational conditions of the vehicle the cooling system is the major noise source and the component with the largest consumption of energy. A comprehensive time domain simulation model was developed for simulation of the cooling module in a Diesel locomotive under realistic operational conditions. It includes the components that produce waste heat such as the engine, the turbo transmission, the brake, etc. and the cooling module with its fans. Given the operation of the locomotive e.g. in terms of speed vs. time along a track and its load, data from experimental full scale tests agree well with predictions from the time domain model. The onset of cooling fan operation is predicted well, with it their instantaneous energy consumption and sound radiation. Three optimized cooling unit assemblies for the new locomotive Voith Gravita 15L had been developed and pre-assessed utilizing the model and eventually tested in the locomotive under realistic operational conditions. A new thermodynamically advanced cooling unit with aerodynamically and acoustically optimized fans was found superior by approx. 2 dB (A) less sound power radiation and some 30% less energy consumption as compared to the benchmark. It is anticipated that those advantages are even more distinct as the ambient temperature decreases.

The work is part of the European FP7 transport research project ECOQUEST.

Keywords: Fan, noise, locomotive cooling, time domain model, system optimization

NOMENCLATURE

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<td>average</td>
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<td>cooling system</td>
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<td>div</td>
<td>spherical divergence</td>
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<td>equivalent</td>
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Abbreviations

AMESim Advanced Modeling Environment for Performing Simulations of Engineering Systems
BHE Balancing heat exchanger
ECOQUEST Efficient Cooling Systems for Quieter Surface Transport
HT high temperature
LT low temperature

Introduction

Most components of a locomotive power train produce waste heat that has to be transferred into the free atmosphere, even under extreme temperature conditions due to an unfavorable climate. Hence, an important subsystem, as in nearly all surface transport vehicles, is the cooling module. Typically the ram pressure acting on a locomotive cannot be used so that any cooling system is of forced-air type. The fans providing the cooling airflow consume considerable energy while radiating undesired sound. Under specific operational conditions they are the major noise sources with a dominating demand of power among all subsystems - a typical situation is engine cooling upon arriving at a station.

In contrast to non-mobile cooling units the specifications for mobile units are severe with respect to low weight and compactness, energy consumption and last but not least overall noise emission. In the train industry the extensive testing of prototypes as carried out in the automotive industry is prohibitive because of small numbers of specimens, high cost and timing. However, not meeting the standards at the end of a long development cycle may lead to an economic disaster. Moreover, the classical design cycle is still sequential: The capacity of the heat exchanger is specified by the vehicle manufacturer (OEM). Manufacturers of cooling modules then struggle to comply with the OEM's targeted overall vehicle performance. This may be illustrated by a cooling system for Diesel-hydraulic locomotives which is the object of this contribution. The Diesel engine in traction mode and the hydrodynamics transmission in braking mode are the major heat sources and require most cooling capacity. Many studies have been carried out in the past to analyze and improve the cooling fans (see e.g. [1], [2], [3], [4], [5]). However, the overall energy consumption and noise emission is not necessarily determined by the isolated component properties as measured under steady-state laboratory conditions. On the contrary, modern cooling systems are based on complex coolant networks with different temperature levels in different hydraulic circuits and sophisticated controllers. The interaction of all components in the complete system under realistic conditions is complex and transient. Hence, an important issue for train manufacturers and suppliers is the accurate prediction of the energetic and acoustic performance of a new vehicle at the earliest possible stage of the design process.

A model based prediction tool would allow optimizing components and avoiding sub-optimal thermal and acoustic performance and cost of the complete vehicle. To the knowledge of the authors such a tool is not available so far and hence is an essential aim of the current ECOQUEST project (Efficient Cooling Systems for Quieter Surface Transport) within the European 7th framework program. Within this project special attention is given to the effect of locomotive loading, realistic drive cycle and realistic environmental conditions. Compiling a tool that takes all this into account requires a thorough system analysis and a multi-domain modeling effort.

Methodology

Multi-domain modeling and optimization strategy

The cooling unit as investigated in this study includes the hydraulic network, the fan, the hydrostatic drive of the fan and associated network and the controller. Because of the complexity, the complete system is modeled with the aid of a commercial multi-domain simulation tool [6], here LMS Imagine.Lab Amesim [7]. This tool allows transient modeling of all components and their interaction (see e.g. [8], [9]).

So called object diagrams of components are generalized block diagrams and allow modeling the signal flow as well as the energy flow between different components. They are used to graphically model complex systems. Object diagrams can be used for modeling of mechatronic systems, made of mechanical systems, electrical circuits, drive drains, control blocks, hydraulic systems, etc. In general the energy flow is defined by a product of the variable effort $a$ which corresponds to the potential, and flow $b$ which defines the energy transport by carrier $q$:

$$P = \sum_{i=1}^{n} a_i b_i = \sum_{i=1}^{n} E_i(q_i, ..., q_n) \frac{\partial q_i}{\partial t}$$  (1)

[10]. The energy fluxes at an element total 0, i.e.

$$\sum_{i=1}^{n} E_i = 0$$  (2)

Here we couple the object diagrams for the Diesel engine, the turbo transmission and the cooling unit.

The fans are described in terms of their measured steady-state pressure-rise, power consumption and sound power vs. flow rate characteristics. Within this study it is convenient to employ scaling laws. Hence, we express pressure rise and shaft power vs. flow rate characteristics in terms of the usual non-dimensional coefficients [11]:

$$hyd$$
$$i$$
$$ini$$
$$mech$$
$$mic$$
$$ref$$
$$sca$$
$$ter$$
$$tip$$

Blade tip

number of energy flow
initial
mechanical
microphone
reflection
scattering
terrain
\[ \varphi = \frac{\dot{V}}{4 \pi^2 D_{np}^3 n} \]  
\[ \psi = \frac{\Delta p}{\frac{1}{2} \pi^2 D_{np}^2 n^2 \rho} \]  
\[ \lambda = \frac{P_s}{8 \pi^4 D_{np}^5 n^2 \rho} \]

The efficiency is then:
\[ \eta = \frac{\psi}{\varphi} \]

At any time instance the moments acting on the fan shaft are in equilibrium. We take into account the torque of the motor, the torque required by the fans due to their aerodynamic loading and the inertia of the rotating parts:
\[ \tau_{\text{shaft}} + \tau_{\text{fan}} + I \dot{\omega} = 0 \]

Additional transient aerodynamic affects are neglected.

The acoustic scaling law as proposed by Madison [12] links the overall acoustic power of a fan to tip speed and rotor tip diameter as:
\[ P_{ac} \sim u_{tip}^2 D_{tip} \]

With that we define a specific sound power level:
\[ L_{w,spec} = L_w - 10 \log \left( \frac{u_{tip}^2 D_{tip} \rho M a^{(a-3)}}{P_0} \right) \]

Here the acoustic reference power is \( P_0 = 10^{-12} \) W, and the empirical Mach number exponent \( a \) is set to 5.

To estimate the spectral shape of the sound power at different rotational speeds the spectra are scaled frequency-wise via the Strouhal number:
\[ S_f = \frac{f}{BPF} \]

with BPF being the blade passing frequency [13]. Fig. 1 and Fig. 2 show examples of a fan sound spectrum as scaled up for larger rotor size and rotational speed.

Fig. 3 shows the subsystems “hydrostatic fan drive”, “fan” and “heat exchanger” that form the complete system.

Bond graph theory [14] is utilized to model the system in explicit graphical form in terms of energy flow. The bond graph language allows to link sub-systems with various physical properties in a generalized way through power interactions. In different physical domains the interpretations of power is a combination of “effort” (e.g. torque, pressure, temperature, voltage) and “flow” (e.g. angular velocity, volume change rate, entropy change rate, current). Therefore, coupled systems in different energy domains can be coupled by generalized variables. For the cooling system the hydrostatic, mechanical and aerodynamic systems are interacting where the form of energy varies. By portraying the system in terms of power bonds the system is compiled with bond graph standard elements as in Fig. 4. (e.g. basic 1-port elements: resistor R, inertia I and sources SF; basic 2-port elements: transformers TF and gyrators GY; basic 3-port junction elements: junction 0 and 1).
flow in different components of the fan drive. The mechanical shaft (Fig. 5a) with eq. (7) is a function of both torques and the inertia. The boundary conditions and the communication with other modules are based on the energy flow, a torque-velocity product. With the input of mechanical and aerodynamic power the fan block (Fig. 5b), based on the dimensionless variables, computes the aerodynamic characteristics. Because all component performance curves are non-linear, the solution requires iterations by finding a common shaft speed of hydrostatic drive and fan where torque from the hydrostatic drive is in equilibrium with the aerodynamic torque. With the quadratic resistance (Fig. 5c) of the heat exchangers the operating point on the fan’s characteristics is found. Analog to the Bond graph method the subsystems “communicate” by power interaction with a combination of effort and flow value. Similarly the subsystems:

- water circuits
- water pumps
- heat exchanger (water side)
- controller

are modeled with these multi-domain technique [15]. Note that during transient simulation not only the overall performance of the cooling unit but also all interfacing variables such as fan speed, torques and energy consumption, sound radiated, etc. are available.

![Fig. 3: Multi domain subsystem of the fan drive](image)

![Fig. 4: Linearized Bond graph plot of the fan drive](image)

![Fig. 5: Energy flow chart of: a) Inertia of the fan, b) Aerodynamic fan performance, c) heat exchanger characteristic](image)

The complete model of the cooling unit described is also used for optimization. “Energy consumption” and “emitted sound power” are the target functions. The transient simulation yields exactly the two target functions which have to been minimized. The difficulty that both target functions vary with time during the operating cycle, is overcome by defining time-averaged values as target functions, here the equivalent sound power level [16]:

$$L_{W, eq} = 10 \log \left( \frac{1}{T} \int_0^T 10^{\frac{1}{10} L_{W}(t)} \, dt \right)$$  \hspace{1cm} (11)$$

and the mean hydrostatic output power:

$$P_{CS, av} = \frac{1}{T} \int_0^T P_{CS}(t) \, dt$$  \hspace{1cm} (12)$$

The design space contains:

- principal system layout (thermal coupling/decoupling of the water/oil circuits, coupling/decoupling of water pump from drive train etc.)
- control strategy (type of controller, integration of ambient temperature sensor etc.)
- size and number of heat exchangers / fans and further geometrical details of the cooling unit
- operating cycle
- ambient conditions

Often the model is implemented in a more comprehensive model of the locomotive that operating in a given environment [15]. This includes:

- the Diesel engine
- the turbo transmission
- the transmission set (cardan shaft, axle gear, wheel etc.)
- the transient vehicle body (vehicle, boogie)
- other acoustic sources as in [17] and [18].

Furthermore, the locomotive interacts mutually with other components such as the track, the environment and the driver (Fig. 6). Most components of the locomotive’s powertrain produce heat that has to be transferred into the environment at a huge range of ambient conditions.
Railway noise is caused by a combination of a number of complex sources and - in general - consists of:

- Rolling noise
- Impact noise
- Traction noise (engine, compressor, fan, other)
- Deceleration noise braking (breaking, squeal)
- Curve squeal
- Aerodynamic vehicle noise

At low vehicle speed traction noise dominates the noise level. At higher speed range the rolling noise becomes more relevant. For high speed vehicles the aerodynamic noise is the most important sound source.

An extensive data base on railway-relevant acoustic sources and their modeling in terms of rather simple correlations is given in [17] and [18]. On top of traction noise as obtained by the detailed multi-domain simulation and described above we take all other noise source into account but will not discus those in the present contribution. Aerodynamic vehicle noise is neglected completely because of the low speed of the locomotive considered.

Scattering and reflection of propagated sound into free atmosphere is taken into account, assuming no significant refraction e.g. from terrain and vehicle body. The model is based on the geometric ray ansatz of diffraction as e.g. proposed in [19] for railway noise propagation.

**Full scale testing**

Besides modeling, full scale tests serve assessing the simulation method and subsequently the impact of new designs. In contrast to standard stand still or passing locomotive acoustic tests, in this study microphones (Bruel&Kjaer, type 4190) were placed on the moving locomotive in the vicinity of the fans and the cooling system (Fig. 8). In order to reduce transmission of structural-borne sound, the microphone mountings were embedded in insulating rubbery material.

Prior to the locomotive measurements, the effect of the headwind was investigated in a low noise and low turbulence wind tunnel of the University Siegen. The acoustic signal was captured continuously during the measurements, but split into intervals of 1 s in the data postprocessing. In each interval, the acoustic power spectral density was computed by the Matlab function pwelch(p,window,noverlap,nfft,fs) where the input was selected as follows:

- p: signal to be analyzed
  - here: acoustic pressure from measurements
- fs: sampling rate
  - here: fs = 25600
- window: window vector
  - here: Hann window with window size = fs
- noverlap: overlapping of windows
  - here: no overlapping (noverlap = 0)
- nfft: number of FFT points
  - here: equal to window length

This input selection leads to a frequency resolution \( \Delta f = 1 \) Hz.

The locomotive, towing 475 tons of freight cars, is operated on a test track from Lüneburg to Amelinghausen (Germany). With approx. 23 km distance and 17% up- and downhill gradient the track is such that all relevant subsystems including the cooling module are temporarily activated.

Following cooling module assemblies were investigated during the test campaign:

- two benchmark fans mounted in a standard cooling module
- two optimized fans mounted in a standard cooling module
- benchmark fans mounted in an advanced cooling module
- optimized fans in the advanced cooling module

All cooling modules were designed for the same maximal cooling power. The most advanced design with optimized thermo-hydraulic circuit is presented in Fig. 12.
The optimized fan, Fig. 9, is as described by Bamberger and Carolus [22]. While the benchmark fan was designed using analytical methods, the new fan was designed by steady-state CFD simulations embedded in an optimization algorithm. The system dependent installation effects were taken into account by using the actual flow profile as inlet boundary condition. NACA airfoil sections were used for the blade design and the spanwise distribution of camber and position of maximum camber were the optimization parameters. The target function was total-to-total fan efficiency meaning that losses due to secondary flows should be minimized. Although it turned out that optimization yielded an increase of efficiency by only roughly one percentage point, it was anticipated that the impact on noise is more significant because even small and local flow separations can heavily increase the unsteady blade forces which are the acoustic sources. Most of the energy saving originated from the implementation of optimal guide vanes. The benchmark fan unit has a diffuser downstream of the impeller, but only structural vanes without aerodynamic function. In contrast, the optimized fan unit is equipped with guide vanes that were integrated into the diffuser. Guide vane inlet and outlet angles as well as the diffuser opening angles at hub and shroud were also optimized by CFD simulations and an optimization algorithm. This method resulted in much better performance as compared to analytically designed guide vanes or diffusers. Besides the aerodynamic improvement this measure is also assumed to reduce the equivalent sound power level because it leads to lower fan speeds required to obtain a specified air flow rate. This reduces the sound emission in partial load of the cooling system. In addition, the maximum cooling capacity is increased as there is a higher flow rate at the maximum fan speed. The diameter of the benchmark fan was selected as large as possible to minimize exit losses. However, this leads to little space for the inflow nozzle which is quite small and needs to be truncated at the sides. Reduction of fan diameter enabled a slightly bigger and non-truncated inflow nozzle. The main benefit is the avoidance of areas with strong backflow which occur at the benchmark fan inflow. A smoother inflow profile is assumed to reduce the leading edge noise which generally results from impingement of turbulent structures with the blade leading edge. Since a reduction of fan diameter increases the kinetic energy of the jet leaving the fan and thus the exit losses, this measure must be considered a compromise between aerodynamic and acoustic performance. In state-of-the-art designs, the axial length of the fan unit is kept to a minimum. This is not only to reduce cost but also to avoid obstruction of the heat exchangers. Prior to the fan optimization a CFD based parameter study was carried out which revealed that neither fan diameter nor the axial length has a significant impact on losses in the heat exchangers. Consequently, it is concluded that the axial length may be increased such that there is more space between the impeller and the guide vanes which reduces the rotor-stator interaction noise. Moreover, additional axial length was required to allow for more efficient deceleration of the flow velocity in the guide vanes and diffuser.

RESULTS

All results reported here refer to a relatively short section of the full run of the locomotive. However, it is chosen such it contains acceleration, applying the brakes, and phases with constant speed and coasting. The target speed vs. time serves as an input into the simulation model (Fig. 10, top). Moreover, the engineer was asked to follow a similar course of target speed vs. time throughout each run during full scale testing. Nevertheless, the true vehicle speed in the tests deviates somewhat from the target.

System simulation and optimization

Fig. 10 shows the vehicle speed vs. time and the associated engine speed and power. It is interesting to compare the actual vehicle speed with the target speed (maximum permissible vehicle speed at this section) which clearly reveals the effect of inertia taken into account in the model. Fig. 11 picks the temperatures at two monitoring points and the fan speed. For this system, obviously during acceleration or breaking mode the coolant temperatures are increasing which in turn is compensated by an increase of fan speed.
Utilizing this model and applying an optimization strategy, a more advanced cooling module was designed. A key feature of this layout is a balancing heat exchanger between high and low temperature (HT, LT, respectively) water circuit, (Fig. 12). In the benchmark module the forced air flow rate is determined exclusively by the circuit with the highest thermal load. In the advanced system the peak load at HT is shifted into the LT circuit which delays the onset of the fans and/or decreases fan speed. This has the potential of a reduction of fan power consumption and noise emission. Moreover, a modified control strategy was applied.

Validation of simulation

In Fig. 13 the instantaneous overall sound power as predicted from the model and measured full scale on the locomotive is compared. Given some unavoidable uncertainty of the ambient conditions, the agreement is more than satisfactory. The very short peaks in the measured sound pressure level up to 110 dB(A) are the horn signals actuated by the engineer during the run on the test track. Fig. 14 shows the corresponding vehicle performance. Even the spectral contents of the sound emitted is in rather good agreement, Figs. 15 and 16.
Fig. 15: Sound pressure Campbell diagram during measured operation cycle

Fig. 16: Sound pressure Campbell diagram during simulated operation cycle

Comparison of cooling module assemblies

Table 1 show the predicted performance improvements by the most advanced module assembly (optimized fans in the advanced cooling module) as compared to the benchmark system (two benchmark fans mounted in a standard cooling module). The advanced cooling system requires 33% less external energy and radiates 2.2 dB (A) less of equivalent sound power. The improvements become even more distinct as the ambient temperature decreases (e.g. during winter).

Table 2 contains data from the full scale test. The ambient temperature was approx. 20° C and nearly constant throughout the test. The full scale test target speed of the train was somewhat modified as compared to the original target to foster activation of forced-air cooling. Also, the course of speed vs. time (run 1 and 2) for two independent runs varied slightly because of the engineer’s manual operation. Therefore the characteristic energy and noise values are different for both runs depending on the utilization of the cooling unit for driving cycles with varying traction efforts. Nevertheless, these tests confirm nicely the numerical predictions: the optimized fan with advanced cooling module achieves a reduction of equivalent sound pressure by approx. 2 dB (A). (The instantaneous maximum sound pressure is even reduced by some 5 dB (A).) The quotient of hydrostatic power (applied to the fans shaft) to cooling power improves from 1.02% or 1.23% to 0.76%.

Tab. 1: Comparison of the worst and best cooling module assembly at different ambient temperature (data from simulation)

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<th>Reduction of equivalent sound [dB]</th>
<th>Energy saving [%]</th>
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<td>-0.1</td>
<td>2.4</td>
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<td>+30</td>
<td>-1.7</td>
<td>23.4</td>
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<td>+20</td>
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<td>33.0</td>
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<td>+10</td>
<td>-2.0</td>
<td>34.8</td>
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<td>- 20</td>
<td>-4.7</td>
<td>60.7</td>
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Tab. 2: Comparison of cooling module assemblies (experimental data from full scale tests, ambient temperature approx. 20° C)

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<th>Standard fans, advanced cooling module</th>
<th>Optimized fans, standard cooling module</th>
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<td>99.2</td>
<td>99.0</td>
<td>98.0</td>
<td>97.5</td>
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<td>2</td>
<td>98.3</td>
<td>97.5</td>
<td>97.3</td>
<td>96.5</td>
</tr>
<tr>
<td>1</td>
<td>115.2</td>
<td>109.4</td>
<td>111.6</td>
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<tr>
<td>2</td>
<td>113.2</td>
<td>109.6</td>
<td>110.8</td>
<td>108.7</td>
</tr>
<tr>
<td>$P_{\text{tot}}/P_{\text{CS}} \times 100$ [%]</td>
<td>1</td>
<td>1.02</td>
<td>0.96</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>1.23</td>
<td>0.89</td>
<td>1.05</td>
<td>0.76</td>
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</table>

CONCLUSIONS

Energy consumption and noise radiation from large forced-air cooling units for Diesel locomotive under realistic operation and load were investigated. For that a complex simulation model was developed and validated successfully by full scale tests on the track. The new simulation tool allowed the development and pre-assessment of three optimized cooling unit assemblies. As an outcome, a new, thermodynamically advanced cooling unit with aerodynamically and acoustically optimized fans was found superior by approx. 2 dB (A) less sound pressure radiation and some 30% less energy.
consumption as compared to the benchmark. It is anticipated that those advantages are even more distinct as the ambient temperature decreases.

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