Marine Gas Turbine Infra-Red Signature Suppression: 
Aerothermal Design Considerations

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ABSTRACT:

In recent years it has become evident that the Infrared (IR) Radiation given off by marine gas turbine exhaust systems is highly undesirable for naval vessels and commercial vessels traveling in areas of conflict. As a result, great interest has surfaced in the ways that IR signatures can be reduced. This paper presents an overview of some of the methods that can be used for engine exhaust IR signature suppression (IRSS). The methods considered here involve only ambient air addition for metal and plume cooling.

The present paper describes various IRSS systems and discusses the basic technical criteria for system selection. Basic operating principles are also described. Aerothermal design considerations are discussed and areas requiring special care during the design are highlighted. Because of the confidential nature of the subject, direct quantitative performance comparisons cannot be made.

INTRODUCTION

Infra Red Signature Suppression (IRSS) involves the elimination or reduction of high radiance sources of IR radiation on a system. The present paper concentrates on methods that can be used for engine exhaust IRSS.

Engine exhaust IRSS systems that are used today typically involve film cooling of visible metal surfaces, optical blockage to eliminate direct line of sight of hot metal and dilution plume cooling. Cooling air injection can be accomplished by passive means and/or forced by using fans. Hardware configurations range in complexity from simple film cooled cylindrical ducts to film cooled optically blocked variable area annular ducts. Air for plume cooling can be introduced through a series of film cooling gaps or in one step using eductor type devices. Eductor devices may include single or multiple nozzles, or multi lobed nozzles, with simple mixing tubes, diffusing mixing tubes, or multi ring film cooled diffusing mixing tubes. From a performance standpoint the selection of hardware is based on the level of protection desired, which requires consideration of the threat view angle.

Over the past few years there have been many requests by potential users of IRSS hardware to compare available systems in terms of performance. However, since these systems differ in approach it is sometimes difficult to compare them on an equal footing. This paper is intended to present an overview of some of the systems and to make, wherever possible, comparisons in operation and performance from an aerothermal standpoint. A detailed comparison of other important factors such as cost, and structural considerations will not be made here.

SHIP IR SIGNATURES

Before the various systems are introduced it is appropriate to consider the motivation for installing IRSS equipment. In the paper by Birk and Davis (1988) a simple calculation of IR signatures was performed on a hypothetical ship. Table 1 presents the assumed temperatures and areas used in the analysis.

Based on the above, Birk and Davis (1988) went on to predict the percentage contribution of these sources to the overall IR signature of the ship, relative to a simple background. The results of this calculation are presented in Table 2.

As shown in Table 2 both the plume and the hot exhaust duct surfaces can be significant sources of IR radiation from a ship. The 3-5 µm and the 8-12 µm wavebands were selected

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Table 1: Assumed Conditions for Order of Magnitude Analysis of Different Sources of Radiation on a Ship (from Birk and Davis (1988))

<table>
<thead>
<tr>
<th>Source</th>
<th>Temperature (°C)</th>
<th>Source Effective Area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>hull</td>
<td>20</td>
<td>1500</td>
</tr>
<tr>
<td>plume</td>
<td>400</td>
<td>20</td>
</tr>
<tr>
<td>visible</td>
<td>400</td>
<td>5</td>
</tr>
<tr>
<td>ducting</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2: Estimated Percentage Contributions of Sources to overall Ship Signature (W/sr) Relative to Background at 288 K. From Birk and Davis (1988)

<table>
<thead>
<tr>
<th>Source</th>
<th>3 - 5 µm</th>
<th>8 - 12 µm</th>
</tr>
</thead>
<tbody>
<tr>
<td>hull</td>
<td>1</td>
<td>54</td>
</tr>
<tr>
<td>plume</td>
<td>23</td>
<td>0</td>
</tr>
<tr>
<td>exhaust duct</td>
<td>76</td>
<td>46</td>
</tr>
</tbody>
</table>

because these fall within atmospheric windows through which IR radiation can pass without significant attenuation. These wavebands are also popular for modern IR guided weapons and imaging systems. The calculation of plume radiation has accounted for radiation attenuation by atmospheric CO₂ as described by Birk and Davis (1988). These predicted signatures apply for a side on view of the ship and will vary with view angle and range.

The objective then of engine exhaust IRSS is to reduce or eliminate the exhaust duct and plume signatures for the desired range of view angles. It can be shown that for metal cooling this is possible by cooling the visible metal to near ambient conditions. However, because of the selective radiating characteristics of the plume it is not necessary to cool the plume to the same degree as the metal to get a similar reduction in the plume signature. The following section describes a number of ways in which the metal and plume signatures can be reduced.

IR SIGNATURE SUPPRESSION HARDWARE

Birk and Davis (1988) described two systems for IRSS developed and applied in Canada. These were the DRES Ball and Eductor Diffuser (E/D) as shown in Figures 1 and 2. These systems rely on film cooling for metal surface cooling and turbulent mixing of entrained air for plume dilution cooling.

Figures 3 and 4 show how these devices replace the upper sections of a ships uptakes within a typical ships funnel. These two devices are by no means the only way to reduce IR signatures.

The problem of IRSS can be broken down into three steps: 1) cooling the top most visible part of the exhaust metal to provide limited view angle protection, 2) addition of a large quantity of cooling air to cool the exhaust gases, 3) provide additional metal cooling and optical blockage to increase the view angle protection.

Figure 1: Sketch of DRES Ball Device

Figure 2: Sketch of Eductor/Diffuser Device

FILM COOLING METAL FOR LIMITED VIEW ANGLE PROTECTION

To provide a limited amount of view angle protection against IR guided threats it is necessary to cool the visible portions of the exhaust uptake. This can be done in a number
of ways including convective and fin cooling from the unheated side of the uptake, or by film cooling on the heated side of the uptake. Film cooling is usually used because it is capable of cooling the metal to near ambient temperatures. For convective or fin cooling schemes this degree of cooling requires large convective coefficients on the coolant side and high thermal conductivity materials.

In film cooling air is injected along the metal surface on the hot side of the ducting. The film cooling layer isolates the metal from the hot exhaust gases and thereby eliminates the convective heat transfer between the metal and the hot gases. The film cooling does not eliminate thermal radiation from the hot gases to the metal but this is usually a small effect.

Figures 5 and 6 show two examples of film cooled devices that can be used for cooling the upper two diameters of the exhaust duct. Figure 5 shows the self entraining film cooled diffuser (SED) and Figure 6 shows a blown film cooled cylinder (FCC).

The SED device is basically the same as the diffuser section of the DRES Ball and Eductor/Diffuser devices as shown in Figures 1 and 2. Other devices such as the USN BLISS cap (boundary layer induced stack suppressor) vary slightly from that shown, but work on a similar principle. The FCC is similar to the UK Cheese Grater used for IRSS.

Figures 7 shows qualitatively the pressure distribution in the SED and FCC devices. The important feature to note is that in the SED the gas flow diffusion results in the pressure rising from below atmospheric pressure at the inlet to atmospheric pressure at the exit, whereas in the FCC the pressure is dropping. This means that the SED can be a passive device (i.e. no fans are required) because air will be naturally drawn in through the various film cooling gaps. The FCC requires fans for film cooling. When the FCC fans are shut down there will be hot gases escaping through the film cooling slots into the funnel spaces. It would be highly undesirable from an IR signature standpoint to have escaping gases heat funnel spaces and increase funnel IR emissions.

The SED devices typically involve a number of concentric rings which overlap to form the film cooling slots. The diffusion angle plays an important role in the ability of the diffuser to provide a significant pressure recovery. As in solid wall diffusers, large diffusion angles result in diffuser stall. With large diffusion angles the flow entering the diffuser will separate from the wall and the pressure will be nearly constant through the diffuser. Thus the pressure to drive air through the film cooling gaps will be reduced. Diffusion half angles less than 5 degrees are recommended.

The FCC devices usually consist of rolled sheet metal which has been perforated with a series of overlapping slots as shown in Figure 6. These slots are created by cutting and bending the metal to form a scoop like slot. The overlapping pattern is used to ensure uniform film cooling of the metal surface. Finite length slots are used for structural

Figure 3: Typical DRES Ball Installation

Figure 4: Typical Eductor/Diffuser Installation
Film cooling can be introduced in a number of ways as described by Goldstein (1971). Possible slot geometries include continuous slots, finite length overlapping slots, or uniformly distributed round or other shaped reasons, rather than continuous slots like in the SED device.

Film cooling can be tangential (i.e. parallel to the surface) or can be injected at an angle α to the surface. The overlapping concentric rings of the SED device result in the film being injected tangentially. The type of slots used in the FCC device typically result in non-tangential film injection.

For film cooling the duct metal the slot size and the slot streamwise spacing can be determined from available empirical relations for film cooling. Mukherjee (1976) proposed the following three-stage correlation for tangential air injection assuming 1) a fully developed turbulent boundary layer, 2) an ideal slot geometry and 3) no pressure gradient.

\[
\eta = \begin{cases} 
1 & \text{for } B < 1 \\
1 + 0.525 B^{0.47} \frac{Pr}{\gamma} & \text{for } 1 < B < 4 \\
1 + 0.329 B^{0.80} \frac{Pr}{\gamma} & \text{for } B > 4 
\end{cases}
\]

where,
\[
\eta \text{ = film cooling effectiveness} \\
m \text{ = main stream} \\
c \text{ = coolant stream} \\
ad \text{ = adiabatic wall} \\
T \text{ = temperature}
\]
For non tangential air injection, the film cooling is not as effective and therefore a correction to the above relations is necessary. Mukherjee (1976) presented the following correction term for non tangential film injection.

\[ \beta = 1 + 0.0015 \frac{R_{ec} \mu_c / \mu_s}{P_r} \sin \alpha \] (4)

where, \( \alpha \) = injection angle
\( = 0 \) for tangential

Other correction terms exist for such film degrading effects as injection slot lip thickness which generates a wake that degrades the cooling film. For a discussion of this the reader is directed to Mukherjee (1976).

To use the above correlations the designer must choose a desired metal surface temperature based in IR signature requirements. With the exhaust gas and ambient air temperatures as known quantities, it is then possible to determine a desired film cooling effectiveness. Space and fabrication considerations usually dictate a desired number of slot locations and this determines the surface length to be cooled. Estimates of gas velocities must then be made based on gas flow rates and areas, and predicted pressure distributions in the device. Then by using the above equations it is possible to estimate gap sizes. Hot flow model tests are usually performed to verify the design.

The selection of a metal cooling scheme depends on a number of factors including size and weight of the device, operational simplicity, engine performance penalties and cost. Both the SED and the FCC provide effective metal cooling with very little back pressure (approximately 250 Pa (1 inch of water)). Both the SED and the FCC require cooling air intakes with appropriate operational considerations such as moisture separation. For the same level of protection (i.e. view angle) the SED will tend to be larger and heavier than the FCC. However, when the volume and weight of fans and ducting for the FCC is included, along with the necessary precautions required for when the fans are shut down, the SED may be the superior choice. A true cost comparison between the SED and the FCC can only be performed when all components and their associated engineering costs are considered in the analysis (i.e. fans and ducting, and any other special precautions).

The film cooling gas layers provided in either the SED or the FCC devices mix with the primary gas stream and result in some peripheral cooling of the exhaust plume. However, if the cooling air flow rate is designed for metal cooling, the flow rates are not sufficient to provide any significant plume cooling with respect to the plume IR signature. For plume cooling some other means of air injection is required.

COOLING AIR INJECTION FOR PLUME COOLING

A large quantity of ambient air is required to provide a significant reduction in the average exhaust plume temperature. Such quantities of air are usually introduced using an eductor type device as shown in Figure 2, or the air can be introduced using a large number of film cooling slots such as in the DRES Ball of Figure 1. The DRES Ball will be discussed further in a later section. Another way of course is to use large fans but this option will not be discussed here.

Eductors typically consist of a suction chamber or plenum, a driving nozzle, a mixing tube and a diffuser as shown in Figure 8. For ship applications the suction chamber is usually some space within the funnel and air reaches this space through funnel mounted louvers.

The primary gas stream (in this case the exhaust gases) is accelerated through the primary nozzle and is vented into the mixing tube. The resulting jet entrains air from the suction chamber. The two fluids mix in the

\[ \text{Figure 8: Sketch of an Ejector Pump Showing Basic Parts and Typical Pressure Distribution} \]
mixing tube. The combined flow is then decelerated in the diffuser to recover kinetic energy from the stream. The diffuser need not be present if it is not desired to reduce the kinetic energy in the combined flow. Figure 8 also shows the pressure distribution in a typical eductor system.

The performance of an eductor system used for IRSS is measured by 1) its ability to pump cooling air, 2) the uniformity of the temperature profile at the exit of the device (i.e., the degree of mixing between the primary gas stream and the secondary cooling air stream), and 3) the required back pressure applied on the engines to obtain a given pumping rate. One other important performance item is the resulting jet noise created by the device. For a further discussion of noise effects the reader is directed to Birk and Davis (1988).

The importance in the ability of the eductor to pump cooling air with minimum back pressure is obvious. The requirement for the exhaust gases to be uniformly cooled at the exit of the device is due to IR signature considerations. A plume with a uniformly cooled profile will have a lower IR signature that a plume with a centre peaked temperature profile even if the average plume temperatures are the same.

Extensive research has been conducted to establish design data for eductors, otherwise known as jet pumps or ejectors. Typically for IRSS hardware applications the gas flows are incompressible (i.e., Mach numbers less than 0.3) and highly turbulent (Red greater than 200000). For high Reynolds number liquid/liquid jet pumps Blevins (1984) presented the following recommendations based on a literature review of jet pumps.

1) The entrance to the mixing tube should be a smooth convergent channel transition which results in a secondary flow which is as parallel to the primary stream as possible while at the same time minimizing friction losses.

2) The primary nozzle exit and the mixing tube entrance should be such that 0.8 < 1/d < 2 where l is the standoff distance between the nozzle exit and the mixing tube entrance and d is the primary nozzle diameter.

3) The optimum mixing tube L/D is 7 but depends somewhat on Reynolds number and entrance conditions. A smoothly convergent divergent mixing tube does not give better performance than a cylindrical tube with conical inlet and diffuser.

4) The optimal diffuser half angle is approximately 2.5 degrees.

As noted above these recommendations apply for highly turbulent liquid/liquid jet pumps. However since eductors are also highly turbulent devices the above recommendations should apply.

Figure 9 shows typical performance curves for water jet pumps from Silvester and Mueller (1968). The curves shown in Figure 9 apply for a design which follows the recommendations for optimal jet pump design as indicated in the sketch shown in the figure. Because these curves were based on liquid pumps their application is limited by cavitation considerations. In gas applications the curves are limited by compressibility effects. Primary jet Mach numbers for ship applications are typically limited to values no larger than 0.3 for back pressure reasons.

It must also be noted that these curves are for isothermal streams and therefore some correction is required for primary and secondary stream temperature differences. Pucci and Ellin (1977) used a pumping coefficient which accounts for temperature differences between the primary and secondary flows. This pumping coefficient is defined as follows.

\[
pumping\ coefficient = \frac{W_s}{W_p} T^{0.44} \tag{5}
\]

where,

\[
W_s = \frac{W_s}{W_p} \tag{6}
\]

\[
s = secondary\ or\ cooling\ stream\ 
\]

\[
p = primary\ or\ hot\ gas\ stream\ 
\]

\[
W = mass\ flow\ rate\ 
\]

\[
T = gas\ absolute\ temperature\ 
\]

For ship applications the total pressure rise through the eductor (H_t - H_0, where d is discharge, s is suction and H is pressure head) is equal to the atmospheric pressure minus the plenum pressure. This pressure drop should be small for ship applications since the secondary air is made available at near atmospheric pressure and therefore the operating point of the eductor is near the so-called zero loss point where the performance curve meets the horizontal axis. This is illustrated in Figure 9.

Figure 10 shows the zero loss points from Figure 9 along with a number of performance points for model eductors from various sources. The figure also includes points for eductors with SED devices and one point for a no-fan DRES Ball. For the eductors with SED devices and for the DRES Ball the area ratio is defined as follows.

\[
A^* = \frac{A_m}{A_p} = area\ ratio \tag{7}
\]

where,

\[
m = mixing\ tube\ 
\]

\[
p = primary\ 
\]

\[
A = area\ 
\]

for SED devices,

\[
A^* = \frac{A_m + \sum A_i}{A_p} \tag{8}
\]

where,

\[
A_i = areas\ of\ individual\ gaps\ in\ SED\ 
\]
for DRES Ball,
\[ A^* = \frac{(A_1 + \sum A_i)}{A_1} \]  \tag{9}

where,
\[ A_1 = \text{flow channel area at first outer duct gap} \]

\[ A_i = \text{areas of individual gaps in outer duct, centre body and diffuser} \]

As shown in Figure 10 the performance of model ship eductor systems fall short of the optimum as described by the above mentioned design recommendations. For example, it is almost always impossible to have mixing tube L/D ratios above 3 because of space limitations. Also, Bonnington and King (1976) state that eductor performance drops rapidly for Reynolds numbers below 100,000 and this is possible for ship eductor systems.

It appears from Figure 10 that some eductor systems are better than others. However, care must be taken in judging these various systems based on this data alone. The differences in performance are due to the effects of Reynolds and Mach numbers, nozzle shaping, and mixing tube L/D. As a general rule, eductor performance degrades as the system geometry diverges from the optimum as described above.

Pucci and coworkers have studied the eductor specifically for application in ships for stack gas cooling. They have come up with following design recommendations.

1) the optimum standoff between the nozzle exit and the mixing tube is \(0.5\) \(D_n\) (where \(D_n\) is the mixing tube diameter)

2) based on a study of nozzles consisting of 3, 4 and 5 circular subnozzles it was determined that increasing the number of nozzles increases pumping and mixing. However the benefit from going from 4 to 5 nozzles was small.

The above recommendations apply for eductors with mixing tube L/D ratios between 2 and 3 and nozzle area to mixing tube area ratios \(A_n/A_p\) between 2 and 3. These devices did not have diffusers at the end of the mixing tube.

The recommended standoff distance \(0.5\) \(D_n\) agrees well with the liquid jet pump recommendations considering the geometries of the devices studied by Pucci and his coworkers.

Because of the constraint on mixing tube L/D and its adverse effect on the pumping ability of the eductor, nozzle design has been extensively studied. Generally speaking, by increasing the perimeter of the nozzles thereby increasing the interface area between the primary and secondary flows, it is possible to enhance mixing thereby partially offsetting the effect of a short mixing tube.

To enhance mixing between the primary and cooling stream air systems designers have used various nozzle shapes. As already mentioned Pucci and Ellin (1977) have studied nozzles consisting of 3, 4 or 5 circular nozzles in a regular cluster as shown in Figure 11, and found that beyond 4 nozzles the improvement in mixing and pumping was small. Toulmay (1984) has studied multi lobe nozzles and other shaped nozzles, also shown in Figure 11, and found that for short mixing tubes the multi lobed nozzle was superior over the other designs. Following the lead of Pucci, our experience has been limited to 4 nozzle configurations or 4 lobed nozzles as shown in Figure 12. We found little difference in the
pumping ability and the exit temperature and velocity profiles from the different designs. However, we did not study these aspects as carefully as Pucci and therefore these aspects warrant some further analysis.

Nozzle shaping affects more than pumping and mixing, it can also affect the applied back pressure. Ship applications usually require that eductor back pressures be in the range between 2000 and 5000 Pag (8 to 20 inches of water) depending on various conditions. In eductor devices where there is a large standoff distance between the nozzle exit and the mixing tube entrance the applied back pressure is almost entirely due to the dynamic pressure of the nozzle flow (i.e., the dump loss). Therefore a streamlined nozzle design will result in a lower back pressure for the same area ratio than a crudely shaped nozzle. Figure 13 shows measured loss coefficients $K$ for the various nozzle designs shown in Figure 12. In this case (i.e., low Mach numbers) $K$ is defined as follows.

$$K = \frac{\Delta P}{\rho \frac{U^2}{2}} = \frac{A_u^2}{A_n^2} + 1 \quad (10)$$

where,

- $\Delta P$ = pressure drop through nozzle
- $A_u$ = uptake flow area
- $A_n$ = nozzle flow area
- $\rho$ = fluid density
- $U$ = uptake flow velocity

It should be pointed out that the nozzle designs have $K$ factors that vary with area ratio. This is due to the fact that exact geometrical similarity is not always maintained when the area ratio changes. Reynolds and Mach number effects may also be present.

High loss coefficients can result in unnecessary increases in back pressure and therefore care must be taken when selecting nozzle designs. Model tests are highly recommended to verify nozzle back pressures.

To summarize then, an eductor system for plume cooling should be designed with the optimum ejector in mind. Ship space constraints will usually result in shorter mixing tubes than recommended and this will lead to a requirement for multiple nozzles or multi lobed nozzles.

PLUME AND METAL COOLING

For TRSS hardware designs where plume cooling and some limited metal cooling is desired it is necessary to use both the eductor and an SED or FCC device or a device...
and therefore bends are undesirable. In addition it is more difficult to uniformly film cool a bend because of pressure gradients. Birk and Westran (1987) studied a concept IR suppressor for a gas turbine powered battle tank which included an eductor like device with a bend. They found that flow separation in the bend resulted in degraded eductor performance and also loss of film cooling in the bent section.

The DRES Ball device shown in Figure 1 provides full view angle protection with the use of a film cooled optical block centre body. As can be seen from the Figure, the DRES Ball device does not use an eductor for plume cooling air entrainment. Because of the large number of cooling air gaps used to cool all of the visible surfaces, it is not necessary to use an eductor. The film cooling air is sufficient when mixed with the primary gas stream to cool it. However, one can argue that the DRES Ball is an annular cascaded eductor. The cooling air addition is brought about by accelerating the primary stream in the annular duct thereby reducing its static pressure sufficiently to entrain cooling air through the various film cooling gaps. Figure 14 shows the typical pressure distribution in the device along the outer duct and centre body surfaces. The pressure shown is for a fan assisted design. Passive devices differ in that static pressures in the flow channel are below atmospheric at all the cooling air gap locations. The pressures on the two surfaces are different because of stream line curvature in the device.

As mentioned earlier, Toulmay (1984) has shown that cooling air addition through a series of gaps is slightly less efficient than through a single gap such as used in an eductor. However the penalty appears to be small based on the fact that the DRES Ball device gives similar plume temperatures and back pressures as the eductor based devices. In fact there are other benefits that result from the DRES Ball design such as reduced noise levels as discussed by Birk and Davis (1988). The noise level is reduced because the flow is more contained by the structure of the DRES Ball.

Because air is introduced at the centre of the gas stream by the centre body the resulting exit temperature and velocity profile is more uniform then for an eductor with a short mixing tube. Figure 15 shows typical exit profiles for an eductor device with a SED and the no fan DRES Ball. As can be seen from the figure the peak temperature in the DRES Ball is lower than in the eductor with an SED.

As with the eductor devices the back pressure caused by the DRES Ball is an important consideration. Typical back pressures for DRES Ball type devices range between 2000 and 5000 Pps (8 to 20 inches of water) depending on the level of cooling and the space requirements that must be satisfied. To ensure minimum back pressure it is necessary to ensure that the diffuser in the DRES Ball is not stalled. It is also important to ensure that the flow does not separate from the downstream side of the centre body since
The level of protection required will then dictate the level of metal (ie. view angle) and plume cooling. The performance of a system will ultimately be determined by design constraints such as available space.

Except for the film cooled cylinder all the designs presented can be passive, that is they entrain cooling air naturally. In all cases some back pressure penalty must be paid for pumping the necessary cooling air for metal and plume cooling. However with careful design back pressure can be kept to a minimum.

REFERENCES


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