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Sensor Cooling by Direct Blowdown of a Coolant

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This article describes results of a proof-of-concept study on cooling of the optical components in a long wavelength infrared (LWIR) sensor using direct blowdown of R23 refrigerant. This technique allows on-demand cooling of optics in a sensor system with an extremely simple and lightweight system. The thermodynamic properties of certain refrigerants such as R23 allow a direct expansion from a high pressure and ambient temperature (300 K) to saturated two-phase conditions at a low pressure without precooling. Refrigerant saturation temperatures between the triple point (118 K) and the normal boiling point (191 K) can be obtained depending upon the level of vacuum available. Experimental results are presented for a 15-cm-diam primary mirror intended for use in a LWIR sensor. The mirror (weighing between 500–750 g) was cooled using direct blowdown of R23 to temperatures of less than 140 K. Cooling times were on the order of 2–3 min. A one-dimensional thermohydraulic model was developed to model the cooldown process. Good correlation was obtained between the model and the experimental results. Further system optimization has been undertaken using the model to minimize system mass for a given aperture sensor optical system and cooling requirement.

Nomenclature

A = passage flow area
A_0 = orifice flow area
c_p = wall specific heat
f = single phase friction factor
g = gravitational acceleration
h = heat transfer coefficient
h_{fg} = latent heat of vaporization
j = superficial velocity
k = orifice discharge coefficient
m = total mass of component
$\dot{m}$ = coolant mass flux
Nu = Nusselt number
P = pressure
Pr = Prandtl number
q = heat flux
Re = Reynolds number
T = temperature
W = passage wetted perimeter
X_{tt} = dimensionless Martinelli parameter
x = quality
z = length coordinate in flow direction
α = void fraction
ρ = density
τ_0 = wall shear stress
ϕ = two-phase frictional multiplier

Subscripts
g = gas phase
l = liquid phase
sat = saturation conditions
w = wall
2ϕ = two phase

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Introduction

LONG wavelength infrared sensors require cooled optics to minimize background noise in the wavelength of interest. Such systems utilize focal planes that are typically cooled to 30 K or less. In general, the optical path (mirrors or lenses) and stray light baffles should be maintained as cold as possible to increase the signal-to-noise ratio (SNR). In practice, the required optics temperature will depend upon the wavelength of interest, the focal plane operating temperature, and the imaging system.

Cooling of the optics differs from cooling of the focal plane due to both the higher allowable temperature and the larger mass. A typical focal plane array may be 10–20 g, and, depending upon required cooldown time, is cooled with either a Joule–Thompson (J–T) open-cycle refrigerator or a mechanical refrigerator. Optical components are typically much larger and thus require much higher cooling power for the same cooldown time. In a previous study,[1] the direct blowdown approach was compared with J–T refrigeration for cooling of LWIR optics. It was found that a direct blowdown approach may result in a lower mass system.

In the direct blowdown cooling scheme, the refrigerant is expanded from a pressurized tank through an orifice and impinges directly on the optical component to be cooled. It then exhausts directly to the ambient, which is at a pressure of 1 atm or below. The lower the exhaust pressure available, the lower the attainable temperature for a given coolant. For space-borne or high-altitude systems, a natural vacuum environment is readily available.

The objectives of this work are to demonstrate rapid cooling of optical components to temperatures of less than 180 K using the direct blowdown technique, and to develop analytical tools for future optimization of the system for minimum mass and cooldown period.

Design Approach, Preliminary Results

In preliminary studies of the direct blowdown cooling process,[1] it was found that rapid cooling occurs during the initial coolant flow period. If coolant flow is maintained, heat transfer will progress from forced convective film boiling to forced convective boiling as the Leidenfrost point is reached. A minimum temperature will be reached that depends on the coolant and the ambient pressure to which the coolant is exhausted. This minimum temperature is typically higher than the saturation temperature corresponding to the exhaust pressure

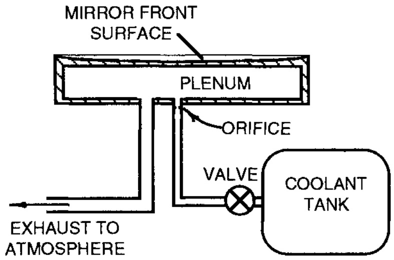

Fig. 1 Direct expansion cooling concept for a mirror.

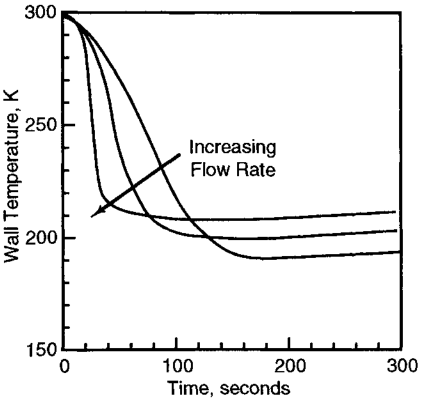

Fig. 2 Typical cooling behavior-direct blowdown.

due to flow losses at the outlet. Higher flow rates result in higher heat transfer rates and shorter cooldown times. However, the minimum attainable temperature is increased due to the greater outlet pressure drop. Typical behavior for the basic system of Fig. 1 are shown in Fig. 2 as a function of flow rate. For a particular desired operating temperature, there exists an optimal flow rate that will minimize coolant mass requirements.

Coolants with critical points above 300 K will provide the most cooling potential, since they will expand to the lowest quality. Coolants that are supercritical at 300 K may also be expanded to two-phase mixtures, but at higher quality. For temperatures less than 150 K, the use of krypton (normal boiling point 119.8 K) as a coolant has been investigated. It was found that a large coolant mass is required due to the high blowdown quality. The lower boiling fluids such as krypton and argon have a high triple-point pressure, which may cause freezing problems. An alternate technique[1] to reduce the minimum temperature while still using a higher boiling fluid is to reduce the coolant flow and to exhaust to a vacuum environment.

If small capillary features exist within the cooling channels, liquid will be bound there by capillary forces, and further flash evaporation cooling will occur following coolant flow termination. This is illustrated by example results in Fig. 3 for a proof-of-concept (POC) secondary mirror with a nickel fiber metal wick inside the coolant plenum. The amount of cooling effect depends upon the quantity of retained liquid and the volume of the wick. The fraction of the wick volume that retains liquid depends on the balance of inertia and capillary forces, since liquid that wets the surface may be either absorbed or entrained in the two-phase flow. An estimate of flow velocities enables an estimate of the pore size required to inhibit entrainment of liquid. For the flow rates and cooling channels of this study, this pore size has been calculated to be between 10–30 μ. A wicking material should be selected that simultaneously minimizes pore size and thermal mass while maximizing pore volume.

Various wicking materials have been evaluated experimentally using a small mock-up mirror test article. The net cooling effect following coolant flow termination was calculated based

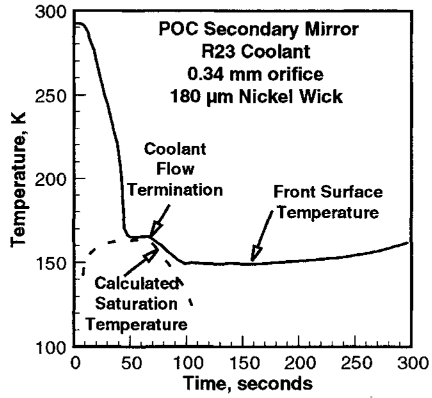

Fig. 3 Observed cooldown behavior with wick material in plenum.

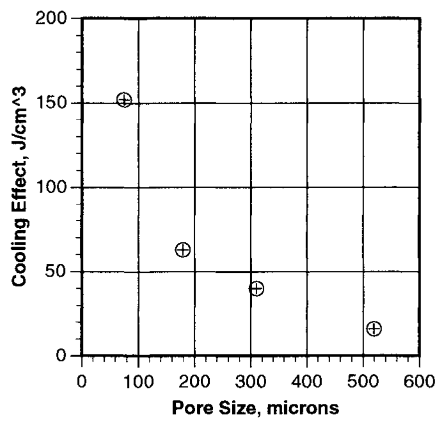

Fig. 4 Observed cooling effect per unit wick volume vs pore size.

on the additional cooling and the thermal mass of the samples. Results are shown in Fig. 4 for the cooling effect per unit wick volume vs the wick pore size. Wick materials were copper, nickel, or aluminum. It can be seen that as the size of the pore decreases, the wick material provides greater cooling effect. Estimates of the liquid retention show that on the order of 50% of the total pore volume is filled with liquid at coolant shutoff. For the larger pore materials, very little liquid retention occurs. Another feature was also observed. For the smaller pore materials a lower minimum temperature is reached, indicating lower outlet pressure drop during the coolant flow period. Observed minimum temperatures (with coolant flow) matched closely the computed saturation temperature based on the measured exhaust plenum pressure for the smaller pore materials. The decreased outlet pressure drop is attributed to the increased wetting and subsequent higher quality in the outlet flow.

Primary Mirror Design/Test Results

Of the potential optical elements in an LWIR sensor, the primary mirror will have the largest dimensions and, hence, poses the greatest challenge for the cooling system. A 15-cm-diam POC primary mirror was designed to demonstrate the capability of our cooling technique. The design approach was to maximize internal heat exchange area and wick volume while maintaining open flow passages to the exhaust plenum. A copper feltmetal wick material was used in the mirror cavity with integral heat transfer passages. This structure was built from strips of wick material and bonded in place using small dots of epoxy. The manifold structure on the back side of the mirror provided a single exhaust port concentric to the coolant inlet. This reduces the complexity of system integration.

The POC mirror is shown in Fig. 5. The entire shell is constructed of aluminum. The aluminum parts weigh 375 g. The copper wick material weighed an additional 370 g. The

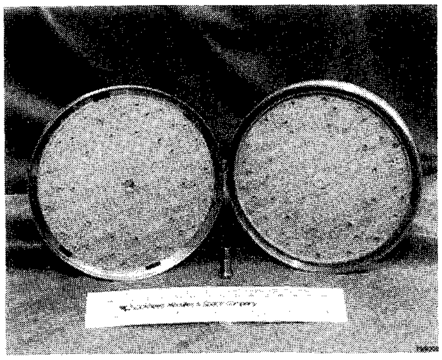

Fig. 5 POC primary mirror.

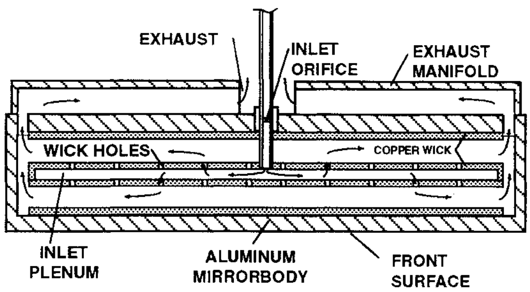

Fig. 6 POC primary mirror flow schematic.

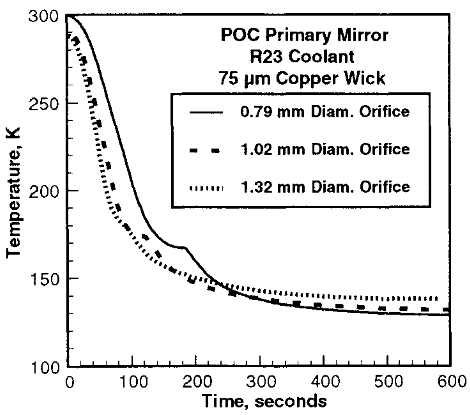

Fig. 7 POC mirror cooldown vs orifice diameter.

two halves are bonded together at the circumference. Coolant enters the inner chamber as shown in Fig. 6 and flows through holes in the wick into the wick passages. It then flows outward to the gutter area and through slots into the manifold that is bonded to the back half of the mirror. It flows radially inward to the outlet and exhausts directly to the ambient. Provision can be made to duct the exhaust to another location for system integration.

The POC primary mirror was tested for three different flow rates by using supply orifices of 0.79, 1.04, and 1.27 mm in diameter. The results are shown in Fig. 7. The large mass of the mirror results in cooldown periods of the order of 3 min. A significant amount of liquid retention is obtained as indicated by the additional cooling effect following coolant flow termination.

In order to decrease the cooldown period and the coolant mass required, two additional design modifications were implemented. The first modification to the mirror design was removal of approximately 50% of the wick mass. The resulting decrease in the mass of copper wick was 180 g. The unit was retested and the results are shown in Fig. 8. The second modification was a reduction in the front surface wall thickness from 3.2 to 1.9 mm (0.125 to 0.075 in.). This resulted in a

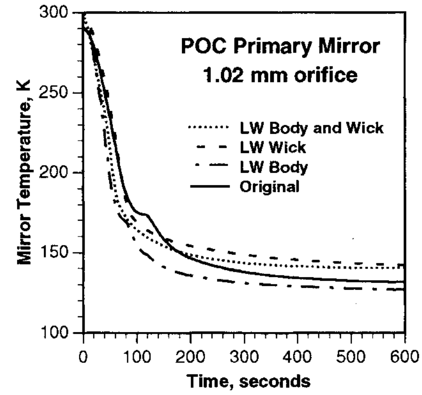

Fig. 8 POC mirror cooldown for modified geometries.

decrease in the aluminum mass of 90 g. The reduced mass POC mirror was retested and the results are shown in Fig. 8. A decreased cooldown period was observed, as expected.

Correlation of Test Results

A simplified model of the direct blowdown cooling process was also developed. Temperature gradients across the mirror are assumed negligible, allowing use of a lumped mass thermal model. The thermal model is solved using forward differencing in time. Two-phase heat transfer and pressure drops are treated using a one-dimensional flow model for the internal cooling passages, along with simple relationships for pressure drops due to singularities, bends, etc. During each time iteration of the thermal model, the flow model is integrated to give current values of the mean coolant saturation temperature and the mean overall heat transfer coefficient. Temperature dependence of the fluid properties is accounted for using curve-fit relationships.

Thermal Model

The temperature change of the solid wall is given by

$$\frac{\mathrm{d}T}{\mathrm{d}t} = \frac{hA}{mC_p}(T_{\mathrm{sat}} - T) \tag{1}$$

where it has been assumed that the wall temperature and heat transfer coefficient are uniform. In practice, one can expect temperature differences of the order of 30°C during the peak cooling period, but these gradients are still small compared to $T_{\mathrm{sat}} - T_w$. Following coolant flow termination, the flash evaporation is treated using an energy balance in conjunction with empirically determined heat transfer coefficients to determine temperature response.

Flow Model

A one-dimensional flow model is used to compute the pressure drop and heat transfer coefficients within the coolant plenum. Pressure drop in the primary flow passages is calculated using a standard separated flow model. Singularities such as bends, contractions, and expansions are calculated using simple pressure-drop correlations.

Assuming steady flow, constant flow area, and constant wall shear stress on the wall periphery; while neglecting the pressure drop in the radially outward flow from the inlet plenum, the one-dimensional conservation equations for the primary flow passages are given by

continuity

$$\frac{\mathrm{d}(\dot{m}A)}{\mathrm{d}z} = 0 \tag{2}$$

momentum

$$-\frac{dP}{dz} = \frac{\tau_0}{A} + \dot{m}\frac{d}{dz}\left[\frac{(1-x)^2}{\rho_l(1-\alpha)} + \frac{x^2}{\rho_g\alpha}\right] + \rho_{2\phi}g\sin\theta \quad (3)$$

where

$$\rho_{2\phi} = (1-\alpha)\rho_l + \alpha\rho_g \quad (4)$$

For the current study, the last term in Eq. (3) is zero, since the passage is tested in a horizontal position. If kinetic and potential energy terms are neglected and internal heat generation is negligible, the energy conservation equation reduces to

energy

$$\frac{dx}{dz} = \frac{qW}{\dot{m}Ah_{fg}} \quad (5)$$

The frictional pressure drop in Eq. (3) must be empirically determined. A two-phase multiplier is used to relate the frictional term to that which would occur if the vapor phase alone flows through the passage:

$$\phi_g^2 = \frac{\left(\frac{dP}{dz}\right)_{2\phi}}{\left(\frac{dP}{dz}\right)_g} \quad (6)$$

where the single-phase gas pressure drop is given by

$$\left(\frac{dP}{dz}\right)_g = f\frac{\rho j_g^2}{2} \quad (7)$$

The two-phase multiplier is given by[2]

$$\phi_g = X_{tt}/(1-\alpha) \quad (8)$$

A relationship for relating the void fraction to the dimensionless Martinelli and pipe inclination parameters can be derived by summing the separated flow steady momentum equations for each phase, neglecting body forces. This equation is[2]

$$X_{tt}^2 = (1-\alpha)^2\frac{1+75(1-\alpha)}{\alpha^{2.5}} \quad (9)$$

where the dimensionless Martinelli parameter is given by

$$X_{tt}^2 = \frac{\left(\frac{dP}{dz}\right)_l}{\left(\frac{dP}{dz}\right)_g} \quad (10)$$

and the Wallis annular flow model for interfacial shear [$=1+75(1-\alpha)$] has been inserted. The above equations, together with the flow passage geometry and the fluid properties, allow integration of the pressure drop along the primary flow passage for a known total mass flow rate and inlet quality. Determination of the absolute pressure distribution requires additional relationships for the pressure drops due to singularities and a boundary condition. The pressure drop for two-phase flow through holes in the wick is related to the single-phase orifice pressure drops, assuming the entire flow was liquid or vapor:

$$\Delta P_{2\phi} = \Delta P_v[1.26\sqrt{(\Delta P_l/\Delta P_v)} + 1]^2 \quad (11)$$

The single-phase pressure drops are calculated from

$$\Delta P_l = (1/2\rho_l)(\dot{m}/KA_0)^2 \quad (12)$$

$$\Delta P_g = (1/2\rho_g)(\dot{m}/KA_0)^2 \quad (13)$$

where the discharge coefficient K is taken as 0.61 for liquid or 0.84 for gas.

The ratio of two-phase heat transfer to that assuming single-phase liquid flowing alone is determined based on the Martinelli parameter[3]:

$$Nu_{2\phi} = 3.5Nu_l(1/X_{tt})^{0.5} \quad (14)$$

The single-phase gas heat transfer is given by the Dittus–Boelter relation:

$$Nu_g = 0.021Re_g^{0.8}Pr^{0.6} \quad (15)$$

The initial heat transfer coefficient corresponds to the single-phase Nu_g. This value is increased linearly to the two-phase value based on the difference between the wall temperature and the saturation temperature. The heat transfer coefficient equals the two-phase value when this temperature difference decreases to the Leidenfrost point, about 10°C for the conditions of this study.[1]

Following coolant shutoff, the exhaust pressure decays quickly to less than 1 torr (depending upon the ambient pressure) and the retained liquid evaporates at a temperature approach-

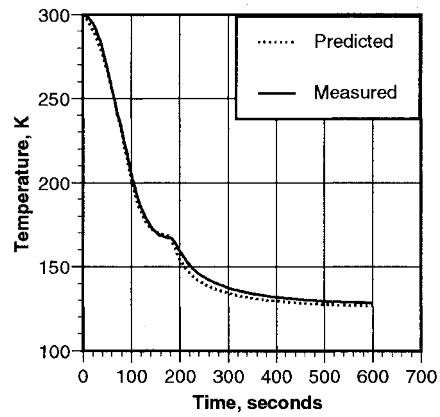

Fig. 9 Predicted vs measured results for 0.79-mm-diam orifice.

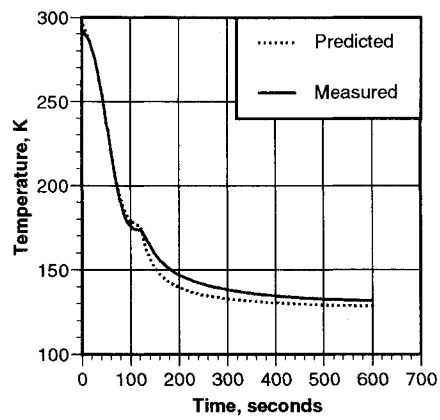

Fig. 10 Predicted vs measured results for 1.02-mm-diam orifice.

ing the triple point of the coolant. In the experimental system, a large vacuum pump was used to simulate an ambient space environment. Heat transfer rates during this flash evaporation period were computed empirically. Upon coolant flow termination, the fluid temperature in the model is immediately reset to the triple point. The heat transfer coefficient is reset to the empirical value and the retained liquid mass (a model input) is decreased according to the energy balance and the latent heat of vaporization. The heat transfer coefficient is assumed to decrease linearly from its initial value to zero based on the computed remaining liquid mass. This assumption was made for convenience to account for the reduction in evaporating surface area and the recession of the evaporating front away from the wall. It allows for a crude comparison of cooling systems with differing wick volumes.

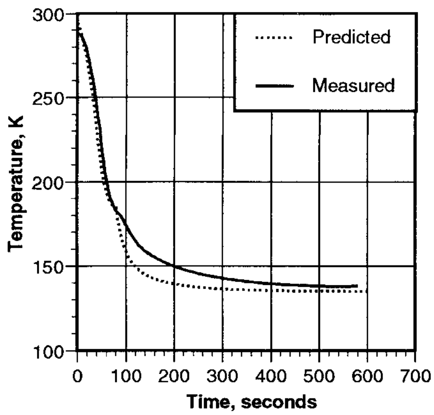

Fig. 11 Predicted vs measured results for 1.32-mm-diam orifice.

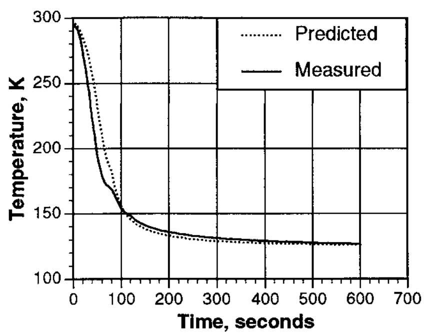

Fig. 12 Predicted vs measured results for modified POC mirror; lighter body.

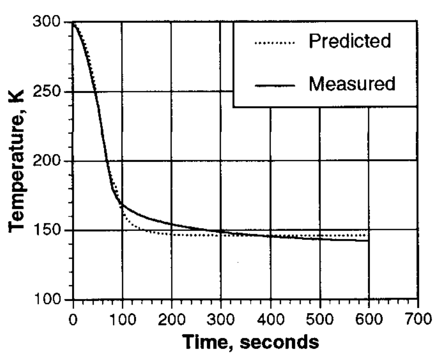

Fig. 13 Predicted vs measured results for modified POC mirror; lighter wick material.

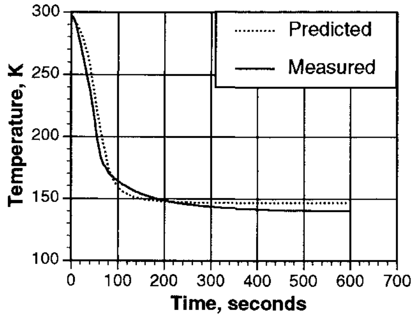

Fig. 14 Predicted vs measured results for modified POC mirror; lighter body and wick material.

Comparison with Test Results

The model was used to compute the temperature history for conditions of the tests. The measured average mass flow rate and exhaust pressure (outside the mirror) were used as boundary conditions. An inlet quality of 62% was used based on an isenthalpic blowdown from 300 K to low pressure. The analytical and experimental results are compared in Figs. 9–11 for the three different R23 flow rates tested. Results for the modified mirror geometries with lesser wick and wall masses are shown in Figs. 12–14. In both cases, the agreement was good.

Conclusions

An approach has been developed to rapidly cool relatively massive sensor optical elements to near-cryogenic temperatures using a direct blowdown technique. This approach results in a simple system, consisting only of a coolant tank, a valve, the associated plumbing, and an orifice, in addition to the heat exchanger that is integral to the component to be cooled.

The approach has been verified using a 15-cm aluminum POC primary mirror. High heat removal rates (approximately 500 W) were demonstrated using R23 coolant. Cooling of a 500–750-g aluminum and copper mass to less than 160 K took less than 2 min. Minimum temperatures of between 135–140 K were obtained, and temperatures less than 150 K were maintained for several minutes following coolant shutoff.

Transient temperature response has been correlated using a one-dimensional analytical model that accounts for two-phase pressure drop and heat transfer. Further optimization of the design using the model will allow reductions in system mass for a given cooling requirement.

Acknowledgment

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