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HEAT EXCHANGE IN COOLED FLOW AREA OF THE TURBINE,(U)
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HEAT EXCHANGE IN COOLED FLOW AREA OF TURBINE

by

S. Z. Kopelev, S. V. Gurov,
M. V. Avilova-Shul'gina

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О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

*ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ë in Russian, transliterate as yë or ë. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

GREEK ALPHABET

Alpha	Α	α	*α*	Nu	Ν	ν	
Beta	Β	β		Xi	Ξ	ξ	
Gamma	Γ	γ		Omicron	Ο	ο	
Delta	Δ	δ		Pi	Π	π	
Epsilon	Ε	ε	*ϵ*	Rho	Ρ	ρ	*ϱ*
Zeta	Ζ	ζ		Sigma	Σ	σ	*ς*
Eta	Η	η		Tau	Τ	τ	
Theta	Θ	θ	*ϑ*	Upsilon	Τ	υ	
Iota	Ι	ι		Phi	Φ	φ	ϕ
Kappa	Κ	ϰ κ	*ϰ*	Chi	Χ	χ	
Lambda	Λ	λ		Psi	Ψ	ψ	
Mu	Μ	μ		Omega	Ω	ω	

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}

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rot	curl
lg	log

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HEAT EXCHANGE IN COOLED FLOW AREA OF TURBINE

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(Moscow)

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Are presented the results of the theoretical and experimental study of heat exchange in flow areas of the high-temperature cooled turbines GTD [ГТД - gas-turbine engine]. Are obtained criterial relationship/ratios for the generalization of the experimental data on the intensity of cooling of rotor blades under conditions of rotation.

It is shown, that the data on heat exchange, obtained at static installations, can be common for work conditions of blades under turbines. are given the experimental data on the warm-up of cooled blade of one of widespread diagrams - hollow with internal deflector illustration 5, Table 1. References 9. page 105-111.

In turbine the heat exchange on the external and internal surfaces of blades occurs in centrifugal-force field. In connection with this arises the question concerning the competence/validity of the propagation of the data, obtained at static installations, to work conditions of blades under turbines, and on the criteria, which characterize the effect of centrifugal-force field on heat exchange in turbine stage. Let us examine it in connection with the blades of two mostd widely use diagrams - with the transverse and longitudinal (relative to pen) location of cooling channels [1]. From system of equations, which describe the process of heat exchange during the nonseparable nonisothermal flow the airfoil/profile of blade of the flow about of the compressible gas, the presence of centrifugal-force field it is considered only by equation of motion

$$\nu\left[\frac{\partial \mathbf{W}}{\partial \tau} + W(\operatorname{grad} \mathbf{W})\right] = \mathbf{T} - \operatorname{grad} \mathbf{p} + \mu \nabla^2 \mathbf{W}. \qquad (1)$$

Here **T** = ρj is the mass force, referred to unit volume, j is the acceleration, which determines this force. The effect of the mass forces and forces, which create constrained motion, is characterized by difference T - grad p.

For the rotating lattice under value W let us understand the relative speed of flow, which is equivalent to passage to the coordinate system, connected with lattice. In this case to the forces, which act in rigid lattice, will be addded an additional two: centrifugal from movable centripetal acceleration j_{a} and Coriolis from Coriolis acceleration j_K.

Force T can be decomposed on three components: tangential (circular/neighboring) T_u, radial T_r and axial T_a.

In turbine the axial accelerations in the vane channels of rotor blades are comparatively small; therefore axial force T_a can be excluded from examination. Bearing in mind that the tangential component T_u is composed of the force T_t, which appears in rigid lattice during the flow of gas in curvilinear vane channel, and inertial Coriolis force T_K, it is possible to write

$$T = T_t\sqrt{(1 + T_K / T_t)^2 + (T_r / T_t)^2}. \qquad (2)$$

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In turn,, the effect of force field (in our case centripetal and Coriolis) on the character cf boundary-layer flcw it can be revealed cnly when on the different sections of vane channel the amount of the mass forces is different [by 2]. After multiplying ratios T_r/T_t and T_k/T_t to simplexes $\Delta T_{r_{\text{макс}}}/T_r$ and $\Delta T_{к_{\text{макс}}}/T_к$, we will obtain

$$\Delta T_{к_{\text{макс}}} / T_t = \text{idem}, \qquad \Delta T_{r_{\text{макс}}} / T_t = \text{idem}. \qquad (3$$

In the vane channel of the cooled revclving gate in radial direction in particle act the mass centrifugal forces $T_ч$ translational acceleration and the lifts $T_п$, caused by heterogeneous density.

Consequently, for the similarity of the prccesses of heat exchange it is necessary

$$(\Delta T_{r_{\text{макс}}} / T_t)_{ч} = \text{idem}, \ (\Delta T_{r_{\text{макс}}} / T_t)_{п} = \text{idem}, \ \Delta T_{к_{\text{макс}}} / T_t = \text{idem}. \qquad (4$$

The maximum centrifugal acceleration in revolving gate occurs on periphery, minimum - in root section. The maximum difference in the centrifugal forces can be written (see Fig. 1):

$$(\Delta T_{r_{\text{макс}}})_{\text{п}} = \gamma h F_{\text{м. к}} \left(\frac{u_{\text{п}}^2}{r_{\text{п}}} - \frac{u_{\text{к}}^2}{r_{\text{к}}} \right). \quad (5$$

After designating $2r_{\text{ср}} / h = \chi$, we will obtain

$$(\Delta T_{r_{\text{макс}}})_{\text{п}} = \gamma F_{\text{м. к}} u_{\text{ср}}^2 / \chi^2. \quad (6$$

the force of periphery, which acts on blade,

$$T_t = \frac{G_r}{Z}(W_{1u} - W_{2u}), \quad (7$$

here G_r - the gas flow through lattice, z - the number of blades.

From the equation of the continuity

$$G_{\text{г}} = \gamma W_a t h z. \quad (8)$$

By taking into account (6) - (8), it is possible to write

$$\left(\frac{\Delta T_{r_{\text{макс}}}}{T_t}\right)_{\text{ц}} = \frac{u_{\text{ср}}^2}{\chi^2}\,\frac{F_{\text{м.к}}}{(W_{1u} - W_{2u})\,W_a t h} \quad (9)$$

Since the cross-sectional area of **vane channel** (without taking into account of real profile thickness) $F_{\text{м.к}} = \pi d_{\text{ср}} b / z$, **after** conversions we will obtain

$$K_{\text{ц}} = \left(\frac{\Delta T_{r_{\text{макс}}}}{T_t}\right)_{\text{ц}} = \frac{1}{\mu}\,\frac{u_{\text{ср}}}{W_a}\,\frac{b}{h}\,\frac{1}{\chi^2}. \quad (10)$$

Here $\mu = L_u / u_{cp}^2$ is a factor of the load (L_u - the work, given up by the blade **rim** of elementary step/stage, referred to 1 kg of working medium/propellant).

The kinematics of the step/stage of turbine usually is characterized four by parameters - **by dimensionless** work (coefficient μ), by the dimensionless axial **velocity** W_a / u_{cp}, **by the** degree of reaction μ and by the relation of axial **velocities** W_{1a} / W_{2a}. However, with acceptable for a practice accuracy by its it is possible to assign only two parameters - μ **and** W_a / u_{cp}. Consequently, the first two factors in formula (10) reflect **the kinematic** similarity of step/stages. The latter fact was not considered by some researchers [3, 4].

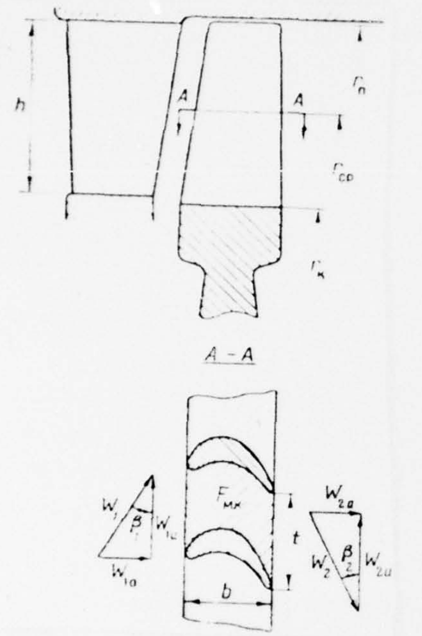

Fig. 1.

Fig. 1. Schematic of the elementary step/stage of turbine.

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Relative speed W_a/u_{cp} is connected with flow angles at entrance and exit from lattice by the relationship/ratio

$$\frac{W_a}{u_{cp}} = \frac{\mu}{\operatorname{ctg}\beta_1 + \operatorname{ctg}\beta_2}. \quad (11)$$

Then

$$K_u = \frac{1}{\mu^2}\frac{\operatorname{ctg}\beta_1 + \operatorname{ctg}\beta_2}{\chi^2\bar{h}}. \quad (12)$$

Here $\bar{h} = h/b$ is strain of blade. Hence it follows that for the concrete/specific/actual step/stage

$$K_u = f(\mu). \quad (13)$$

the maximum amount of Coriolis forces occurs of the root of blade and on periphery, since they on module/modulus are equal to $2\gamma\omega W_r$ (here ω - the angular rate of rotation of lattice, W_r - the speed of the secondary flows). The relationship/ratio of the velocity of the secondary flows and mean-expenditure over section velocity is changed insignificantly [5]. After accepting it as constant, we will obtain that in the lattices impellers the effect of Coriolis forces on heat exchange with an accuracy to constant is characterized by that expression for a criterion $K_Ц$, that also the effect of centrifugal forces.

In the intensely cooled lattices impellers the maximum nonisothermicity in flow $\Delta T_r = T_r - T_{cr}$, here T_r, T_{cr} - are temperatures of flow and wall of blade. Therefore

$$\mathrm{Gr}_r = 2\frac{\Delta T_r}{T_r}\frac{1}{\mu^2\chi\bar{h}} \qquad (14)$$

or, for a concrete/specific/actual step/stage,

$$\overset{\text{п.}}{\mathrm{Gr}_r} = f(\Delta T_r / T_r \mu^2). \qquad (15)$$

If we consider the radial cooling channel in blade as pipe of constant cross-section, then equation of motion can be written in the form

$$dp + \gamma W dW = -\zeta \frac{\gamma W^2}{2} \frac{dh}{D} + \gamma \frac{u^2}{r} dh, \qquad (16)$$

where ζ is a coefficient of friction, D - the hydraulic diameter of channel, and the action of centrifugal forces is considered by positive term $\gamma u^2 dh/r$.

After accepting as characteristic linear size the height/altitude of channel h and after dividing both parts of the equation of motion to velocity head at output/yield from channels $\gamma W_{\text{в}}^2$, we will obtain

$$K_{\text{н}}^{*} = \frac{u_{\text{ср}}^2}{W_{\text{в}}^2 \chi^2}. \quad (17)$$

According to the equation of the continuity

$$W_{\text{в}} = G_{\text{в}} / \gamma F_{\text{в}}. \quad (18)$$

Since $G_{\text{в}} = G_{\text{г}} \bar{G}_{\text{в}}$, keeping in mind (8), it is possible to write

$$K_{\text{н}}^{*} = K_{\text{н}}^{\text{г}} \frac{\bar{h}}{\bar{C}_{\text{в}}^2 (F_{\text{г}}/F_{\text{в}} z)}. \quad (19)$$

In blades with the transverse (relative to pen) location of cooling channels their value δ, as a rule, is constant or is changed insignificantly; therefore

$$T_t^* = (\bar{G}_a G_r / z)(W_{1u}^* - W_{2u}^*) = \bar{G}_a \gamma_a (W_{1u}^* - W_{2u}^*) W_a th. \quad (20)$$

Bearing in mind that

$$W_{1u}^* = W_{1u} \frac{W_a^*}{W_a}, \quad W_{2u}^* = W_{2u} \frac{W_a^*}{W_a}, \quad \frac{W_a^*}{W_a} = \bar{G}_a \frac{t}{\delta},$$

we will obtain

$$T_t^* = \bar{G}_a (t / \delta) \gamma (W_{1u} - W_{2u}) W_a th. \quad (21)$$

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Maximum difference in the centrifugal forces

$$(\Delta T_{r\,\text{макс}})_{\text{ц}} = \gamma F_{\text{м.к}} u_{\text{ср}}^2/\chi^2 \qquad (22)$$

and

$$F_{\text{м.к}} = \pi d_{\text{ср}} b\delta / tz. \qquad (23)$$

By utilizing expressions (20) - (23), it is possible to write

$$\left(\frac{\Delta T_{r\,\text{макс}}}{T_t}\right)_{\text{ц}} = \frac{1}{\mu \bar{G}_{\text{в}}}\,\frac{u}{W_a}\,\frac{(\delta/t)^2}{\chi^2 \bar{h}},$$

or

$$K_{\text{ц}}'' = K_{\text{ц}}'(\delta/t)^2/\bar{G}_{\text{в}}^2. \qquad (24)$$

Thus, for each concrete/specific/actual step/stage the effect of centrifugal forces on heat exchange is characterized by three parameters: 1) by the factor of the load of step/stage μ, 2) by relation $\Delta T_r / T_r$ **and** 3) by the relative air flcw rate for cooling $\bar{G}_B$.

Experimental research of blades with the transverse location of cooling channels was conducted on static packages and on three turbine stages. Their size/dimensions differed among themselves more than 2.5 times. The construction of stand of static packages, the procedure of experiments and processing the experimental data are given in [6]. The fundamental geometric characteristics of the tested lattices are given in table.

The design concept of rotors is given in [7]. Before tests were obtained the discharge characteristics of channels in trailing edges, deflectors and blades in collection with deflectors. The stability of characteristics was checked both in the process of tests and after their completion.

The thermal condition of blades was estimated at the dimensionless coefficient $\theta = (t_r - t_л) / (t_r - t_в)$. which are the virtually single-valued function of the ratio of heat-transfer coefficients from air and gas. The temperature of the stagnant (in relative motion) flow of gas on blades t_r was determined from measurement by the special thermocouples, established/installed on blades, and but to the measurements of the temperature field of gas after turbine. Both methods ensured completely satisfactory agreement. The temperature of blades was measured by the thermccouples, established/installed into four points on airfoil/profile (entering edge, back, the concave part of the airfoil/prcfile, trailing edge) and in three sections by height (root, average, peripheral). Each measurement conducted simultaneously on three blades. The transmission of thermocurrents to recorder (electronic potentiometer EPP-09) was realized through the mercury slip ring. The temperature of air coolant T_B', pressure p_B' and its expenditure/consumption were measured directly at the entrance into rotor [7].

(1) Геометрические параметры	(2) Номера решеток			
	1	2	3	4
(3) Хорда l, мм	78	49	60	41
(4) Шаг t, мм	47	31	43	32
(5) Ширина b, мм	74	45	57	41
(6) Угол входа β_{1k}	53°	55°	56°	38°
(7) Угол выхода β_{2k}	37°	31°30′	42°	32°30′
(8) Диаметр входной кромки D, мм	4,8	4,0	4,7	5,8
(9) Диаметр выходной кромки d, мм	2,8	2,4	2,7	1,7
(10) Относительная высота %	5,0	9,0	5,5	5,3
	(11) Лопатки с переменным по высоте профилем			(12) Лопатки постоянного профиля

Key: (1). Geometric parameters. (2). Numbers of lattices. (3). Chord l mm. (4). Space t, mm. (5). Width b, mm. (6). Angle of entry (7). Angle of departure (8). Diameter of entering edge D, mm. (9). Diameter of trailing edge d, mm. (10). Relative height/altitude (11). Blades with alternating/variable by height airfoil/profile. (12). Blades of constant airfoil/profile.

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Since during air flow from the place of measurement to the section of blade in question its temperature increases because of the work of centrifugal forces, the **temperature** t_B during the definition of coefficient θ was computed as **sum** $t_B = t_B' + (u^2/2c_pJ)$, here u is the peripheral speed in the section in question, c_p – is heat capacity of the air, J - termal equivalent of work.

The estimation of leakages according to labyrinth seals, clearances in the circuit of supply and the scarf joint of blades with disk conducted according to datum, on obtained by the preliminary **blasting of rotors** [7]. The broad band of a change in the parameters μ, $\Delta T_r/T_r$ and $\bar{G}_B$ was achieved by a change in the load of turbine, number of revolutions (55-100o/o), of the temperature of the gas before turbine (650-1490°K), the temperatures of air coolant

(350-800°K), and the expenditure/consumption of the air, take/selected for cooling (0.8-3.3o/o).

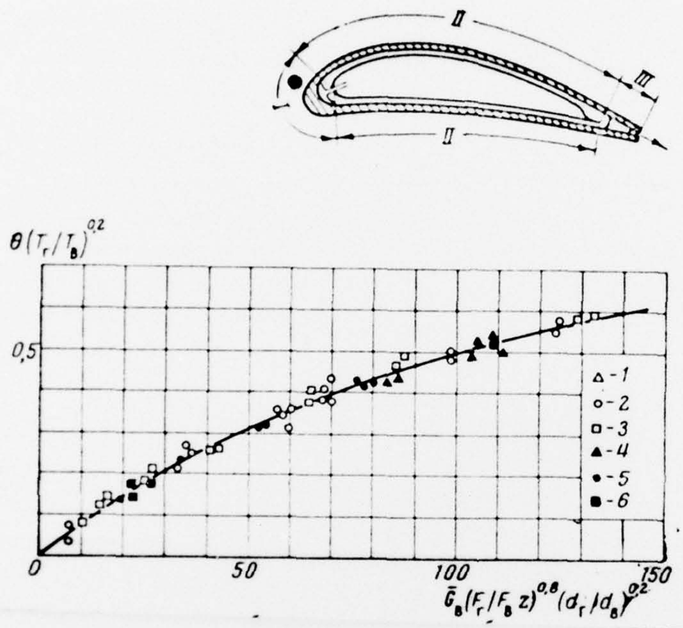

Fig. 2.

Fig. 2. Effect of centrifugal-force field on heat exchange in the cooling channels: 1.2.3 - lattice 1.2.3 in statics, 4.5.6 - in rotation.

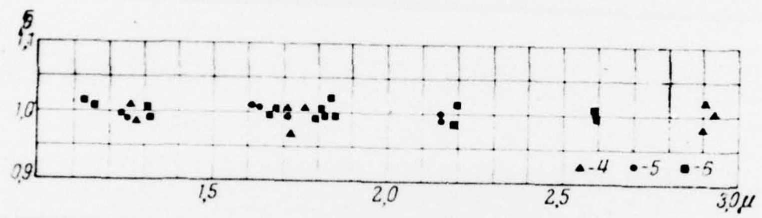

Fig. 3. The effect of the loading of turbine on heat emission in vane channels for a trailing edge (see designations to Fig. 2), $\bar{\theta}$ is a ratio of the intensities of cooling in rotation and in statics.

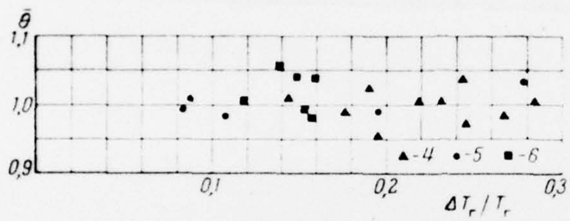

Fig. 4. The effect of cooling lattice on heat emission in vane channels for a trailing edge (see designations to Fig 2).

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During processing the experimental data the mode/conditions of tests were grouped in such a way as to determine the effect of each of these parameters individually.

On Fig. 2 are shown the results in order tc determine the effect of each of these parameters individually.

On Fig. 2 are shown the results of the generalization of the experimental data, μ = 1.6 = const, $\Delta T_r / T_r$ = 0.26 = const. Generalization is made for the section of trailing edge, analogous dependences are obtained also for an entering edge and concave - convex parts of the airfoil/profile.

On Fig. 3 is shown the effect of the loading of the step/stage of turbine on heat emission from gas to blade on section III ($\Delta T_r / T_r$ = 0.15 = const, $\bar{G}_B$ = 1.8o/o = of const), while on Fig 4 - density in nonisothermal flow (μ = 1.75; $\bar{G}_B$ = 3.3o/o). As is evident, in the range of a change of the parameters μ, $\Delta T_r / T_r$ and $\bar{G}_B$ in question the rate of heat exchange under the static conditions and during rotation (with different number of revolutions) remains virtually constant/invariable. Analogous results are obtained for the sections of blade I, II.

The experimental study of the thermal condition of blades with the longitudinal (relative to pen) location of cooling channels is undertaken in [8] (see table, lattice 4).

The temperature of gas on blades during tests was maintained by constant/invariable and equal to 1440°K, air -310°K, expenditure/consumption -1.8o/o. From Fig. 5 it is evident that the coefficient θ also virtually is not changed in all investigated velocity range (6500-12500 r/min: μ = 1.0-1.5. Even a very substantial change in the angle of the inleakage (from -20 to +7°) of the flow of gas to rotor blades (Fig. 5) does not affect their temperature.

It is necessary to keep in mind that the level of flow turbulence in the static installations, where the preheating of air is realized in combustion chambers GTD, is so great that upon transition to revolving gates the effect on it of the velocity of the rotation impeller is not exhibited [9].

Conclusions. 1. In the investigated range of a change of the parameters μ, $\Delta T_r/T_r$ and $\bar{G}_0$ characteristic for turbines of GTD, the intensity of heat exchange in air-cooled blades, measured during tests at static installations and in turbine stages under conditions of rotation, remains constant/invariable.

2. The experimental data, obtained at static installations, can be common for work conditions of blades under turbines, which substantially simplifies the procedure of experiment and the construction of test benches.

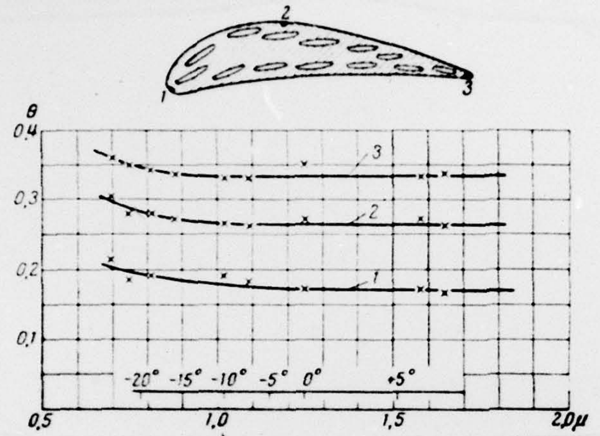

Fig 5

Fig. 5. Effect of the loading of turbine on the dimensionless temperature of blade with the longitudinal (relative to pen) location of cooling channels.

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C591	FSTC	5
C619	MIA REDSTONE	1
D008	NISC	1
H300	USAICE (USAREUR)	1
P005	ERDA	1
P055	CIA/CRS/ADD/SD	1
NAVORDSTA (50L)		1
NAVWPNSCEN (Code 121)		1
NASA/KSI		1
[redacted]		[redacted]
AFIT/LD		1

ORGANIZATION		MICROFICHE
E053	AF/INAKA	1
E017	AF/RDXTR-W	1
E404	AEDC	1
E408	AFWL	1
E410	ADTC	1
E413	ESD	2
	FTD	
	CCN	1
	ETID	3
	NIA/PHS	1
	NICD	5