A Generalized One Dimensional Computer Code for Turbomachinery Cooling Passage Flow Calculations

Ganesh N. Kumar
Sverdrup Technology, Inc.
NASA Lewis Research Center Group
Cleveland, Ohio

Richard J. Roelke
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio

and

Peter L. Meitner
Propulsion Directorate
U.S. Army Aviation Research and Technology Activity—AVSCOM
Lewis Research Center
Cleveland, Ohio

N89-22862
Unclas
0211725
G3/34

(NASA-TM-102079) A GENERALIZED
ONE-DIMENSIONAL COMPUTER CODE FCB
TURBOMACHINERY COOLING PASSAGE FLCW
CALCULATIONS (NASA. Lewis Research Center)
14 p CSCL 20D

NASA

US ARMY
AVIATION
SYSTEMS COMMAND
AVIATION R&T ACTIVITY

A GENERALIZED ONE DIMENSIONAL COMPUTER CODE FOR TURBOMACHINERY COOLING PASSAGE FLOW CALCULATIONS

Ganesh N. Kumar*
Sverdrup Technology, Inc.
NASA Lewis Research Center Group
Cleveland, Ohio 44135

Richard J. Roelke
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

and

Peter L. Meitner
Propulsion Directorate
U.S. Army Aviation Research and Technology Activity – AVSCOM
Lewis Research Center
Cleveland, Ohio 44135

ABSTRACT

A generalized one dimensional computer code for analyzing the flow and heat transfer in the turbomachinery cooling passages has been developed. This code is capable of handling rotating cooling passages with turbulators (inline, staggered, or inclined), 180 degree turns (both sharp and gradual turns, with and without turbulators), pin fins, finned passages, by-pass flows, tip cap impingement flows, and flow branching. The code is an extension of a one-dimensional code developed by P. Meitner. In the subject code, correlations for both heat transfer coefficient and pressure loss computations have been developed to model each of the above mentioned type of coolant passages. The code has the capability of independently computing the friction factor and heat transfer coefficient on each side of a rectangular passage. Either the mass flow at inlet to the channel or the exit plane pressure can be specified. For a specified inlet total temperature, inlet total pressure and exit static pressure, the code computes the flow rates through the main branch and the sub branches (if any), flow through tip cap for impingement cooling (if any), in addition to computing the coolant pressure, temperature and heat transfer coefficient distribution in each coolant flow branch. Predictions from the subject code for both non-rotating and rotating passages agree well with experimental data. The code was used to analyze the coolant passage of a research cooled radial rotor.

INTRODUCTION

Increasing the turbine inlet temperature is a very lucrative approach to improving the thermal efficiency and specific thrust of gas turbine engines. However, this practice is limited by the availability of materials which can withstand these high temperatures. Current materials cannot be exposed to temperatures above 1250 K without a dramatic decrease in life [1]. Blades must be cooled if significantly higher turbine inlet temperatures are desired. As the cooling scheme becomes more complex, a better understanding of the internal pressure and heat transfer coefficient distributions is necessary to be able to predict the hot section metal temperatures accurately. Typical cooling concepts of modern turbine blading are shown in figure 1. Figure 1(a) shows the cooling scheme for an axial rotor, while figure 1(b) shows a hypothetical cooled radial rotor. The shown cooling passages may have heat transfer enhancement devices in the form of turbulators, pin fins, finned passages, leading edge tip cap impingement flows, etc. The main coolant flow may also be split into sub branches as shown in figure 1(a) and 1(c). It should be noted that to obtain a 2-D distribution of heat transfer coefficient in the plane of the blade surface will really require a 3-D analysis. Because of the complex nature of the flow field in cooling passages, it is extremely difficult to develop a generalized 3-D flow analysis code capable of modeling different cooling passage configurations and obtaining detailed distribution of heat transfer coefficients and pressure drops. Such information is required in the detailed thermal and structural analysis of the rotor and blading [2]. Hence it was felt that development of a generalized 1-D code which can model various types of cooling passages including flow branching should be useful in the design and analysis of cooled turbomachinery. Based on the literature review, the authors conclude that such a generalized 1-D code is not available in the open literature.

The code reported in this paper is an extension of the code developed by Meitner [3], and is applicable to any combination of cooling passage configurations mentioned above. Specifically, the reported code differs from the code of reference [3] in that it can calculate and balance flows in sub-branches, and can thus compute flows in passages shown in figure

1(a) and 1(c) whereas the code of reference [3] will not be capable of handling these passages. Also, the code reported in this paper can make separate calculations for the leading(suction) surface, trailing(pressure) surface, and side walls of a rotating channel of rectangular cross section. This is very important for capturing the experimentally observed behavior in rotating environment. In addition, a more detailed modeling of each type of cooling passage above has been incorporated in the present code compared to the code of Meitner. The code can predict the mass flow rates, internal pressure and temperature distribution, heat transfer coefficient and heat flux distribution at the walls along the passage. In a limited sense, it can predict the heat transfer coefficient distribution along each side of rectangular passage. It integrates the 1-D compressible momentum and energy equations along a defined flow path and will account for area changes, mass addition/subtraction, Coriolis acceleration, centrifugal force effects, bouyancy force effects, friction and heat addition. Flow can be bled off for tip cap impingement cooling and a flow 'bypass' can be specified in which the coolant flow is taken off at one point in the flow channel and re-introduced at a point further downstream in the same channel. As mentioned above, other types of passages that can be modeled include plain passages, rib-roughened passages (staggered and inline turbulators at various angles of attack), different pin fin arrangements (with or without lateral flow ejection, as shown in figure 1(a)), finned passages, 180 degree turns (both sharp and gradual) and flow branching.

Based on the most recent experimental data available, correlations have been obtained to model the various types of rotating cooling passages. The user has the option of specifying the temperature and pressure at inlet to the main branch and either the main flow rate or the exit pressures from the sub branches. The present version of the code cannot accomodate supersonic flow anywhere in the cooling passage. The flow can at best be choked at exit from one or more of the branches. However, it is felt that this is not a serious restriction for modeling turbomachinery cooling passages.
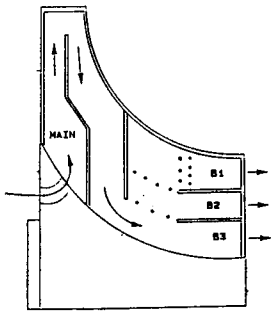
Comparisons have been made with a wide variety of non-rotating and rotating passage experimental data representing various types and combinations of cooling passages. It is shown that the predictions compare well with the experimental data.
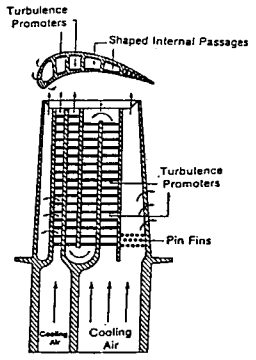
(a): Axial Turbine [9]
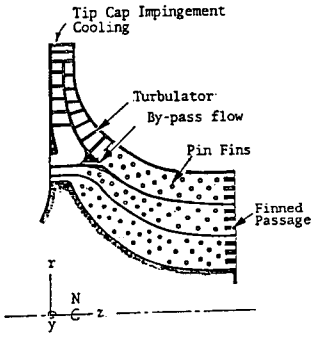
(b): Radial Turbine

(c): Radial tubine with Flow branching

Figure 1. Typical Cooling Concepts of Modern Gas Turbines

2

FLOW PATH MODELING AND COMPUTATION PROCEDURE

The flow path is defined in cartesian co-ordinates z,r and y as shown in figure 1(b). The flow path geometry is described at nodes and the type of passage between each set of nodes is specified using different parameters in the code input.

For specified entrance conditions and a specified (or guessed) flow rate, the flow solution is obtained by marching along the defined flow path from node to node. The pressure and temperature at each node is calculated from the values at the previous node by:

$$p_n = p_{n-1} + (dp/dx)_n * dx_n \tag{1}$$

$$T_n = T_{n-1} + (dT/dx)_n * dx_n \tag{2}$$

where $(dp/dx)_n$ and $(dT/dx)_n$ are given by the solution of momentum and energy equations as [3]:

$$\left(\frac{dp}{dx}\right)_n = \frac{1}{C_2}\left[\left\{\left(C_1 \frac{3}{2}\frac{\dot{m}}{C_p}\left(\frac{RT}{pA}\right)^2 + \frac{T}{\dot{m}}\right.\right.\right.$$

$$\left.-\left(\frac{C_{p,inj}T_{inj}}{\dot{m}C_p} + \frac{V_{inj}^2}{2\dot{m}C_p}\right)\right) + \frac{V_{inj,x}}{A}$$

$$\left.-\frac{2(\dot{m}/A)RT}{pA}\right\}\left(\frac{d\dot{m}}{dx}\right)_n + \left\{\frac{(\dot{m}/A)^2 RT}{pA} - \frac{C_1}{C_p A}\right.$$

$$\left.\left(\frac{\dot{m}RT}{pA}\right)^2\right\}\left(\frac{dA}{dx}\right)_n - \frac{4fRT(\dot{m}/A)^2}{2pD_h}$$

$$+\frac{N^2 rp}{RT}\frac{dr}{dx} - \frac{4h_c A C_1(T_w - T_{aw})}{\dot{m}C_p D_h}\right] \tag{3}$$

and

$$\left(\frac{dT}{dx}\right)_n = \frac{1}{C_T}\left\{-\left[\frac{3\dot{m}}{2C_p}\left(\frac{RT}{pA}\right)^2 + \frac{T}{\dot{m}}\right.\right.$$

$$\left.\left.-\left(\frac{C_{p,inj}T_{inj}}{C_p\dot{m}} + \frac{V_{inj}^2}{2\dot{m}C_p}\right)\right]\left(\frac{d\dot{m}}{dx}\right)_n\right.$$

$$+\left[\frac{1}{C_p A}\left(\frac{\dot{m}RT}{pA}\right)^2\right]\left(\frac{dA}{dx}\right)_n$$

$$+\left[\frac{1}{pC_p}\left(\frac{\dot{m}RT}{pA}\right)^2\right]\left(\frac{dp}{dx}\right)_n$$

$$+\frac{4h_c A(T_w - T_{aw})}{\dot{m}C_p D_n}\right\} \tag{4}$$

where

$$(d\dot{m}/dx)_n = (\dot{m}_n - \dot{m}_{n-1})/(l_n) \tag{5}$$

$$(dA/dx)_n = (A_n - A_{n-1})/(l_n) \tag{6}$$

$$(dr/dx)_n = (r_n - r_{n-1})/(l_n) \tag{7}$$

The constants C1, C2 and CT are defined as follows [3]:

$$C_1 = \frac{(\dot{m}/A)^2 R}{p\left[1 + \frac{(\dot{m}/A)^2 R^2 T}{p^2 C_p}\right]} \tag{8}$$

$$C_2 = 1 - \frac{(\dot{m}/A)^2 RT}{p^2} + \frac{C_1}{pC_p}\left[\frac{(\dot{m}/A)RT^2}{p}\right]^2 \tag{9}$$

$$C_T = 1 + \frac{(\dot{m}/A)^2 R^2 T}{p^2 C_p} \tag{10}$$

Iteration takes place in each marching step until a convergence is acheived. After marching through all the nodes, the calculated branch or sub branch exit pressures are compared with the given exit pressures (if specified). If necessary, a correction is made to the inlet flow rate and the marching procedure repeated until the calculated and specified exit pressure values agree within a prescribed tolerance.

For flows where the local coolant passage Mach number becomes high (> 0.9), pressure changes in the flow direction become highly non-linear. A provision has been made in the code to automatically add additional nodes between the original nodes and repeat the iteration process mentioned above until the Mach number values at each of the original nodes do not change between successive increases in the number of nodes.

In the present paper, for the different types of cooling passsages mentioned above, both friction factor and heat transfer coefficients are modeled with

3

correlations developed from experimental data available in the literature or using the correlations available in the literature (with modifications). For validating the correlations developed, computations using these correlations have been compared with experimental data from an entirely different source than those used in the development of correlations.

The correlations developed are generally of the form:

$$Nu \quad or \quad f = a(ND1)^b(ND2)^c(ND3)^d... \quad (11)$$

where Nu is the Nusselt number, f is the friction factor, ND1, ND2, etc are relevant non-dimensional quantities like Reynolds number (Re), Rotational Rayleigh number (Ra), Rossby number (Ro), Prandtl number (Pr), rib height (or roughness height) to hydraulic diameter ratio (e/D_h), the rib pitch-to-height ratio (P/e), channel aspect ratio (W/H), pin-fin spanwise spacing to diameter ratio (S/d), pin-fin streamwise spacing to diameter ratio (X/d), pin height to pin diameter ratio (H/d), etc. The correlations were obtained from the experimental data using the SAS [4] multiple correlation and regression analysis procedure. The correlation equations are not shown in the paper due to space limitations. Additional details of the correlation equations and the code will be presented in a future NASA report.

In the present version of the 1-D code, correlations have been included to model any combination of the following types of passages:

a) Plain passages (Smooth and Rough): Based on the magnitude of relative roughness parameter (e/D_h), the passages are classified as smooth or rough and the corresponding correlation equations are used.

Friction Factor: For laminar flow in circular, rectangular or square ducts, based on the data available in Kays and Crawford [5] in the form of tables and figures, correlations have been obtained to model both smooth and rough non-rotating passages and for both fully developed and entrance regions of flow. For turbulent flow and in the transition region in ducts, correlations listed in Welty, Wicks and Wilson [6] have been used for both smooth and rough passages

Heat Transfer Coefficient: For laminar flow, correlations are developed based on the data of Kays and Crawford [5] for the fully developed as well as the entrance regions of flows in both circular and non-circular ducts. For fully developed turbulent flows, correlations given by Kreith and Bohn [7] and/or those in reference [5] have been included. Entrance effects are modeled based on Kays and Crawford [5].

b) Turbulated Passages: Correlations have been incorporated to model the heat transfer coefficient and pressure drop distributions for rectangular or circular passages with transverse rib rougheners (both inline and staggered arrangements and at any given angle of attack). Correlations have been included for both the turbulated walls and the smooth side walls based

on the work of Han, et al. [8,9] for in line rib arrangement and Taslim and Spring [10] for staggered arrangement. Correlations include the effects of non dimensional parameters like rib height to hydraulic diameter ratio (e/D_h), rib pitch-to-height ratio (P/e), channel aspect ratio (W/H), Reynolds number (Re) and rib angle of attack ratio ($\alpha/90$). Based on Boyle's data [11], entrance effects have also been incorporated in the correlations.

c) Pin Fins: Based on the work of Metzger, et al. [12,13,14,15], correlations are available to model pressure drop and heat transfer coefficients (array averaged) for different staggered pin fin configurations. Nusselt number (Nu) correlations which include the effects of streamwise pin spacing are given by Metzger [12]. The local row resolved heat transfer coefficient correction has been correlated based on the data of Metzger [12] as well as the corrections to Nu to include effects of channel convergence. Nusselt number correlations are also available based on the work of VanFossen [16]. It should be noted that in VanFossen correlations, the Nusselt numbers are based on hydraulic diameter DV which depends on the open volume in the pin fin bank and the total heat transfer surface area whereas the Metzger correlations are based on pin diameter. In a recent review paper on staggered array pin fin heat tansfer, Armstrong and Winstanley [17] recommend the use of Metzger or VanFossen correlations for short pin fins (H/d <= 3) and the correlation of Faulkner [18] for long pin fins (H/d > 3). Friction factor correlations for both short and long pin fins are based on the work of Metzger [13].

The effect of channel convergence on friction factor is incorporated based on the work of Brown [19]. For pin fin channels with lateral ejection, the correlations developed by Lau et al. [20] are used.

d) 180 Degree Turns: Based on the experimental data of Metzger, et al. [21,22,23,24], correlations were developed for heat transfer coefficient and pressure loss in smooth and turbulated sharp 180 degree turns. For gradual 180 degree turns, the pressure loss coefficients were based on the equations from reference [25] and the correlations for the heat transfer coefficient variation was based on the experimental data of Baughn, et al. [26].

e) Finned Passages: For cooling passage channels with fins parallel to the flow direction (usually near the trailing edges of blades, figure 1(b)),an effective heat transfer coefficient is obtained [3] which can be applied to the corresponding plain passage surface area without fins.

f) Flow Branching: Pressure loss due to flow branching has been modeled based on reference [25].

g) Leading Edge Tip Cap Impingement Cooling: Modeling of the tip cap impingement cooling is based on the development by Meitner [3] who utilized the data of Chupp et al. [27]. It includes the effects of rotation on pressure changes across the tip cap impingement chamber and the temperature increase due to impingement in the chamber. The model also computes the mass flow through the impingement holes.

Corrections have been applied to the Meitner model based on the work of Epstein et al. [28] which include the effects of rotation on heat transfer in a impingement cooled turbine blade. Modeling of by-pass flows (see figure 1 (b)) is also based on Meitner's development [3].

h) Rotational Effects of Plain Passages: Effects on both pressure drop and heat transfer coefficient due to Coriolis forces, centrifugal forces, and centripetal buoyancy have been included for ducts (circular, square or rectangular) rotating in orthogonal and/or parallel modes (with respect to the axis of rotation) and for both laminar and turbulent flows.

For laminar flow in plain passages with smooth or rough walls, and with their axes normal to the axis of rotation, the pressure loss correlations are based on the work of Mori and Nakayama [29] and heat transfer correlations are based on the work of Morris [30] and for moderate temperature differences on the analytical work of [29]. For turbulent flows, the heat transfer correlations are based on the work of Clifford [31] or Morris and Ayhan [32,33] with corrections for the leading and trailing surfaces based on Harsagama and Morris [34]. The user also has the choice of basing it on the theoretical work of Mori, Fukada and Nakayama [35]. The pressure loss correlations are obtained from the work of Ito and Nanbu [36] or from reference [35].

For laminar flow in plain passages rotating about a parallel axis, the pressure drop correlations are based on the rotating friction factor given by

$$f_{rot} = f_{st} + f_{ex} \qquad (12)$$

$$f_{ex} = 0.503 \, Ro^{1.06} \qquad (13)$$

where f_{ex} is "the excess friction factor" suggested by Morris [36]. The heat transfer correlations are based on the work of Morris and Dias [38] and for moderate temperature differences on the theoretical work of Mori and Nakayama [39]. For turbulent flows, the heat transfer correlations are based on the work of Morris and Dias [40] and Woods and Morris [41] and for moderate temperature differences on the theoretical work of Nakayama [42]. The pressure drop correlations are based on [43]. For the entrance region, the heat transfer correlations are based on work of Morris and Woods [43].

It has been shown from the experimental data of Hajek et al. [44], and Guidez [45] that the heat transfer characteristics due to rotation are radically different for the leading and trailing surfaces and also for the centrifugal (radially outward) and centripetal (radially inward) passages. These features have been captured in the correlations used in the present code.

i) Rotating Channels with Heat Transfer Enhancement Devices:

The combined effects of rotation and heat transfer enhancement devices (like ribs, pin fins etc.) are obtained by first computing the heat transfer coefficient and friction factor (pressure drop) distribution due to enhancement devices and then computing the effects of rotation as a ratio of a) Nu(rotating)/Nu(non-rotating) and b) f(rotating)/f(non-rotating) based on the experimental correlations mentioned above. The validity of the correlations used is shown in the following section.

It should be noted that the user always has the option of providing the pressure loss coefficient, friction factor, and/or heat transfer coefficient for any given interval in the form of data to be curve fitted, equations, or a set of given values.

COMPARISON WITH EXPERIMENTAL DATA

In the following paragraphs, the predictons from the code are compared with a wide variety of experimental data for various types of heat transfer enhancement devices. As mentioned earlier, the predictions from the code are compared with the data from the sources which are different from those used in the correlation. Both non-rotating and rotating passage data with and without various heat transfer enhancement devices are compared to give a clear picture of the range of application of the code.

a) Non-Rotating Passage Data:

For a straight duct with 90 degree turbulators, the code predictions are compared with the experimental data of Taslim and Spring [46,47] in figures 2 and 3. Figure 2 compares the mean Nusselt number variation with Reynolds number (Re) for a fixed rib height to channel hydraulic diameter ratio (e/D_h, also known as blockage factor), rib pitch to height ratio (P/e) and channel aspect ratio (W/H). The definition of these terms are clarified in the inserts in figures 2 and 3. Figure 3 compares the Nusselt number enhancement, expressed as (Nu/Nu_{st}), with a variation in the blockage factor e/D_h for fixed values of P/e, W/H, and Re. Predictions are in excellent agreement with the experimental data.

In figures 4 and 5, the code predictions for a two-pass serpentine passage with a 180 degree turn, for both smooth and turbulated walls, are compared with experimental measurements. The calculated Nusselt number variation along the passage is compared with the data of Chandra [48] in figure 4 and with the data of Boyle [11] in figure 5. Agreement with data of [48] is very good for both smooth and turbulated passages. The reason for the reduction in the Nusselt number in the 180 degree turn region ($X/D_h = 12$ to 14) for the turbulated case is that there are no turbulators in the turn region. In the case of Boyle's data, the 180 degree turn region ($X/D_h = 14$ to 16) has turbulators.

Predictions agree very well with Boyle's data [11] at higher values of Re where as at low values of Re ($< = 20000$), the agreement is good except in the region
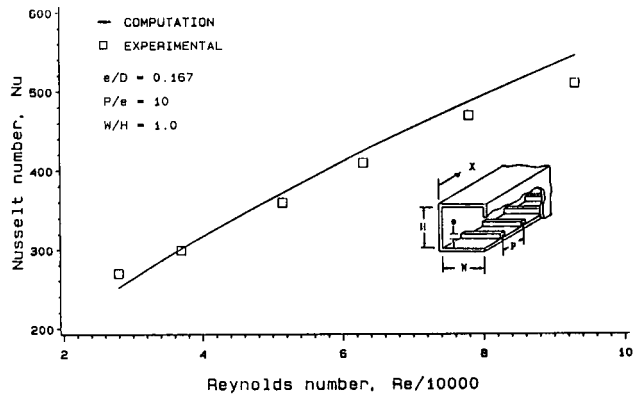
Figure 2. Comparison of Nusselt number variation with Reynolds number with the turbulated passage data of Taslim and Spring [46,47]
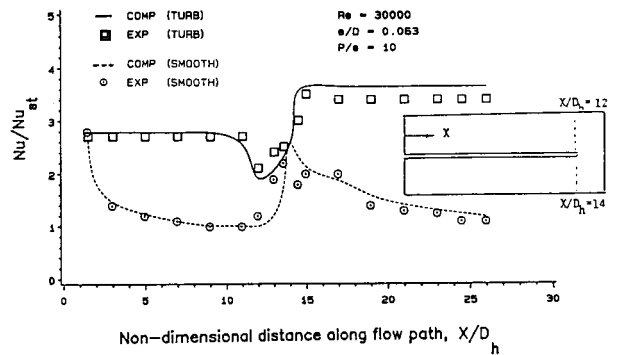
Figure 4. Comparison of variation of Nusselt number with X/D_h with the two-pass rib roughened passage (at 90° with a sharp 180 deg. turn) data of Chandra [48].
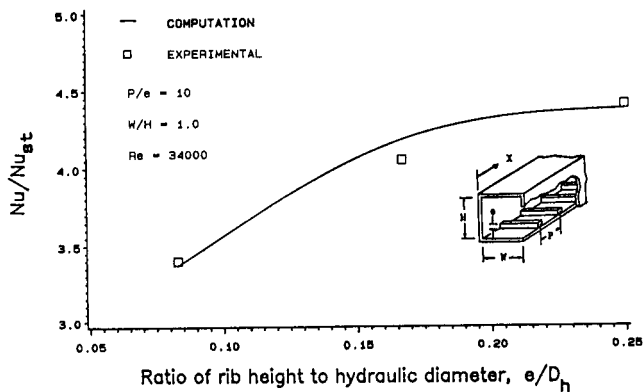
Figure 3. Comparison of Nusselt number variation with blockage factor (e/D_h) with the tubulated passage data of Taslim and Spring [46,47])
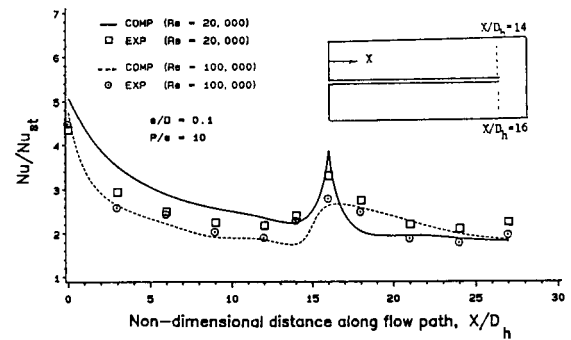
Figure 5. Comparison of variation of Nusselt number with X/D_h with the two-pass turbulated passage data of Boyle [11].

immediately after the 180 degree turn where the computed Nu drops sharply compared to the experimental data. The maximum error is of the order of 20 to 30% in this region. The reason for this behavior is not known at present.

For passages with pin fins, the code predictions are compared with the experimental data of Lau et al. [49] and Peng [50] in figures 6 and 7. Figure 6 compares the predicted Nu variation with row number with the data of Lau [49] and figure 7 compares the predicted variation of friction factor (f) and Colburn factor (j) with Reynolds number with the data of Peng [50]. Both the figures show that the predictions are in excellent agreement with the experimental data.
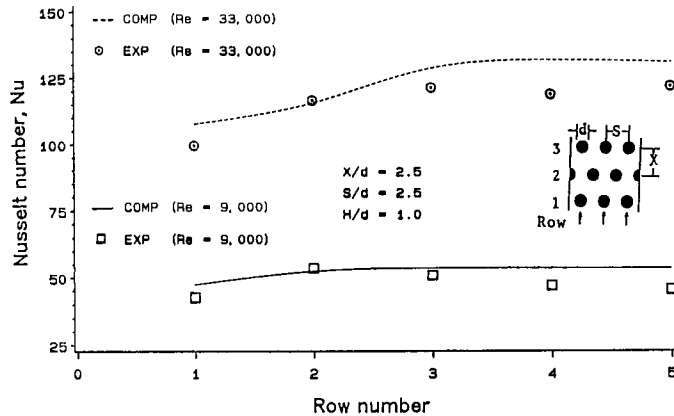
6

Figure 6. Comparison of variation of Nusselt number with row number with the pin fin data of Lau [49].

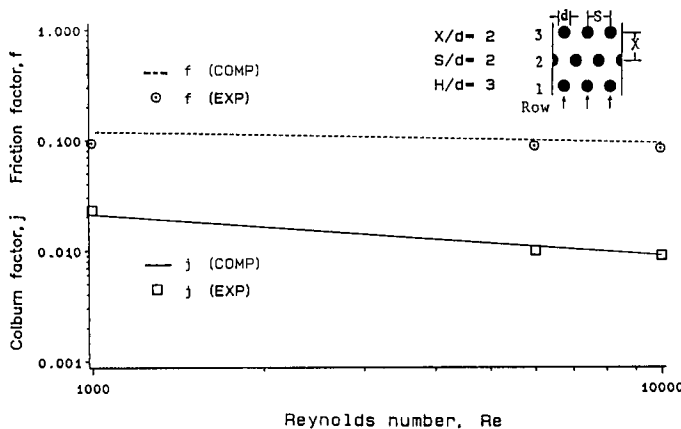Figure 7 Comparison of variation of friction factor and Colburn factor with Reynolds number with the pin fin data of Peng [50].

b) Rotating Passage Data: Extensive comparison has been made with the rotating passage data (both smooth and turbulated serpentine passages) of Hajek, et al. [44]. A schematic of the serpentine passage of Hajek is shown in figure 8. As shown in figure 8, four typical zones have been chosen for comparison with code predictions to illustrate the validity of the correlations developed for rotating channels. Zones 1 and 2 are in the radially outward (centrifugal) section and zones 3 and 4 are in the radially inward (centripetal) section of the passage. As explained earlier, separate correlations are used to predict the effects of rotation on the leading surface (suction surface), trailing surface (pressure surface) and the two side walls of the coolant passage. This enables the code to compute the rotational effects more accurately.
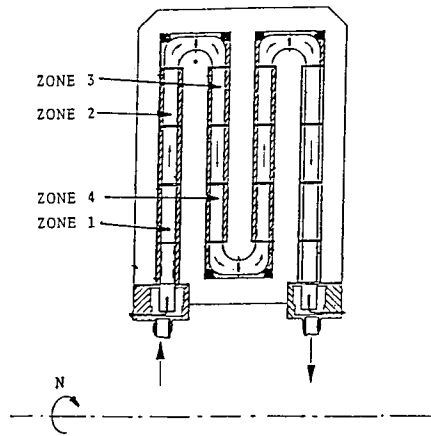

Figure 8. Schematic of the rotating serpentine passage of Hajek [44] showing the four zones compared with the code predictions.

Figure 9(a)-(d) compare the Nusselt number variation with Rossby number (Ro) for the above mentioned four zones along the passage. As seen in figures 9(a) and 9(b), the predictions clearly show the experimentally observed trend for the centrifugal sections zones 1 and 2, that as the Rossby number increases, due to increasing rotational speed, Nusselt number increases on the trailing (pressure) surface and decreases on the leading (suction) surface. This trend is observed for both smooth and turbulated passages. This is due to the increased secondary flow effects at higher values of Ro. Figures 9(c) and 9(d) compare Nusselt number variation with Ro for the centripetal channel. These figures also show that the trend of Nu variation with Ro is in general opposite to that of the centrifugal section. Comparison of pressure variations along the passage with the data of Hajek [44] is made in figure 10 for the turbulated passages. All the above figures clearly show that the code predictions agree well with the data of Hajek [44].
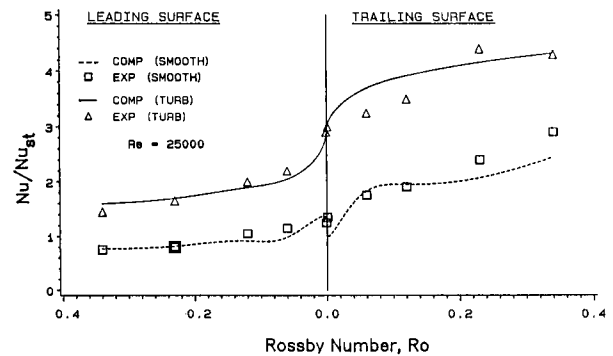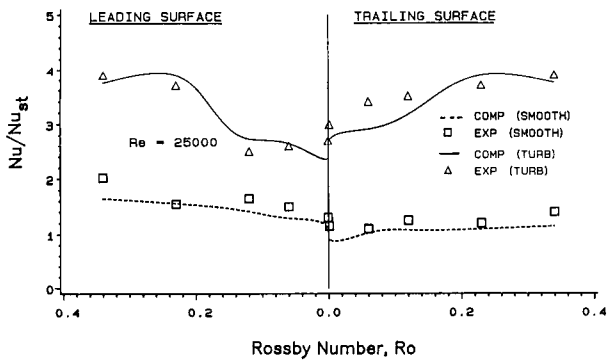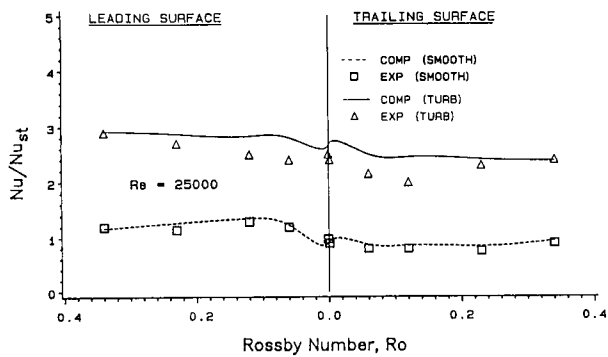

(a): Zone 1.

Figure 9. Comparison of Nusselt number variation with Rossby Number for the smooth and 45 deg. turbulated serpentine rotating passage data of Hajek [44].
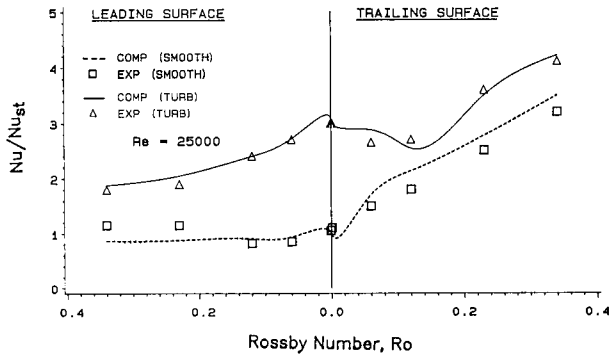
(b): Zone 2.

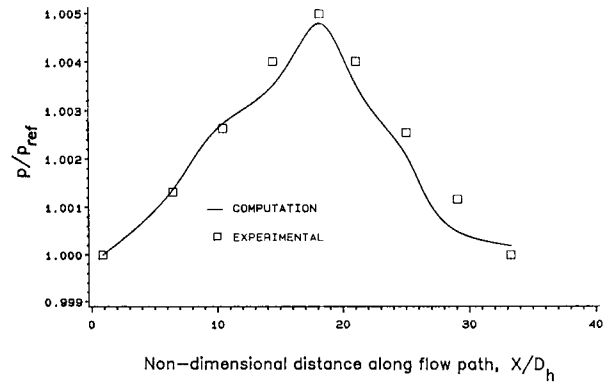(c): Zone 3.

(d): Zone 4.
Figure 9. Concluded

Figure 10. Comparison of pressue distribution along the passage with the turbulated rotating serpentine passage data of Hajek [44].

Comparison of Nusselt number variation with the centrifugal passage data of Guidez [45] is made in figure 11 and with the data of Iskakov and Trushin [51] figure 12. These figures show that the code predictions are in good agreement with the experimental data in both cases with a maximum error of about 20%. The trends shown in these figures are the same as the centrifugal passage data of Hajek [44].
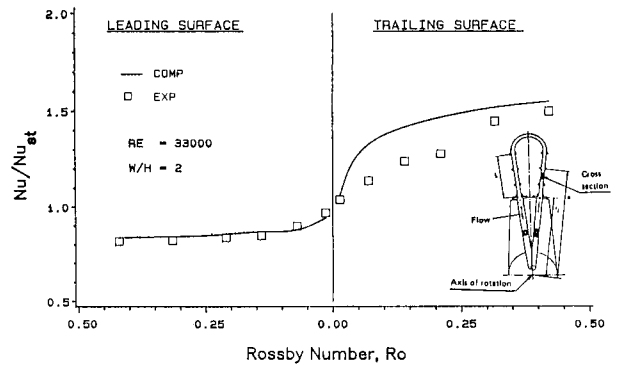

Figure 11.Comparison of Nusselt number variation with Rossby Number with the data of Guidez [45]. at $X/D_h = 7.4$.
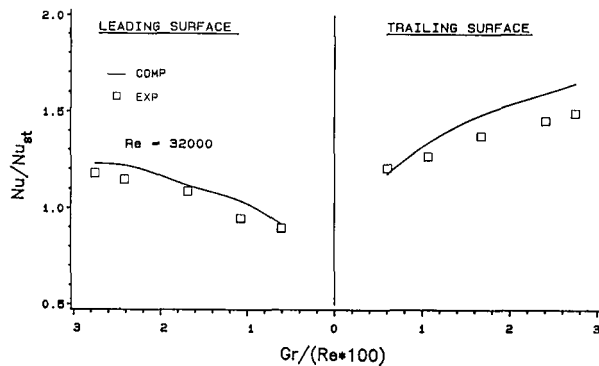
8

Flow path location

Figure 12. Comparison of Nusselt number with the parameter Gr/Re with the data of Iskakov and Trushin [51] at $X/D_h = 7$.

Figure 13. Pressure distribution along the coolant passage of a NASA research radial turbine.

COOLED RADIAL TURBINE ROTOR ANALYSIS

The coolant flow of a cooled NASA research radial turbine rotor, which will be tested at the Lewis Research Center, has been analyzed using the code. The cooling passage consists of a main passage splitting into three branches and includes pin fins and finned passages. The schematic of the cooling passage is shown in figure 1(c). For a given inlet total temperature and pressure and exit pressures from the branches, the present code was used to predict the distribution of pressure along the entire passage and also the mass flow through the main and sub branches. The results are shown in figure 13 and table 1 respectively. These predictions will be compared with the experimental results as they become available.

Table 1. Mass flow computations for the coolant passage of a NASA research radial turbine

Branch	$\dot{m}/\dot{m}_{ref}$ Code Computation
Main	6.734
B1	2.399
B2	1.632
B3	2.703

$$\dot{m}_{ref} = 0.001 kg/s$$

CONCLUDING REMARKS

A generalized one dimensional flow code applicable to compressible flow through complex rotating turbine cooling passages with various heat transfer enhancement devices has been developed. Even though this is a 1-D code, the effects of rotation on heat transfer is taken into account in a quasi 2-D fashion, i.e. a separate correlation is available for each side of a rectangular passage. The code has been validated using a wide variety of both non rotating and rotating passage data. The authors feel that the code will be useful to the turbine designer in evaluating the effects of various cooling passage designs on the overall performance of the turbine. For more accurate computations, it may be necessary to include a localized 3-D analysis where flow reversals and flow stagnation may cause the 1-D code predictions at these locations to be in error by a wide margin.

NOMENCLATURE

A	area
C_p	specific heat at constant pressure
D	diameter
D_h	hydraulic diameter defined by $D_h = 4A/P$
d	pin fin diameter
e	surface roughness; or rib height
f	fanning friction factor defined by $f = \delta p D_h / (2l V^2 \rho)$
Gr	rotational Grashof number defined by $Gr = N^2 r (T_w - T_{aw}) D_h^3 \beta / \nu^2$
H	flow channel height
h	enthalpy
h_o	coolant heat transfer coefficient
j	Colburn factor defined by $j = h_c Pr^{2/3} / (\rho V C_p)$
k	thermal conductivity
l	interval length
$\dot{m}$	coolant mass flow rate at inlet to the channel

N   rotational speed
Nu Nusselt number; $Nu = h_c/(D_h k)$
Nu_{st} Nusselt number for a stationary plain passage
P   perimeter; or pitch of turbulators
Pr Prandtl number; $Pr = Cp\mu/k$
p   pressure
p_{ref} reference pressure
R   gas constant
Ra Rotational Rayleigh number; defined by
$$Ra = GrPr$$
$$= rN^2\beta(T_w - T_{aw})D_h^3 Pr/(\mu/\rho)^2$$
Re Reynolds number; defined by
$$Re = \rho V D_h/\mu$$
Ro Rossby number; defined by $Ro = ND_h/V$
r   radius; co-ordinate direction used in the code
S   pin fin spacing in spanwise (transverse) direction
T   temperature
T_b bulk temperature of the fluid
T_w wall temperature
T_{aw} adiabatic wall temperature
V   fluid velocity
W flow channel width
W/H channel aspect ratio (ribs (if any) on side W)
x   distance along the flow path
X   pin fin spacing in streamwise direction or streamwise distance from entrance to channel
y   co-ordinate direction used in the code
z   co-ordinate direction used in the code
β volume expansion coefficient
γ ratio of specific heats
δ a change in a quantitiy
μ absolute viscosity
ρ density

SUBSCRIPTS
  aw adiabatic wall
  ex excess factor due to rotation
  inj injection
  n   interval number
  st stationary case
  ref reference value
  rot rotating case
  x   component in the direction of main stream flow

SUPERSCRIPTS
  ( )' total conditions

REFERENCES

[1] Rodi, W. and Scheuerer, G. "Calculation of Heat Transfer to Convection Cooled Gas Turbine Blades," Transactions of the ASME, Vol. 107, July 1985, pp 620-627.

[2] Kumar, G.N. and DeAnna, R.G. "Development of a Thermal and Structural Analysis Procedure for Cooled Radial Turbines," ASME paper no. 88-GT-18, presented at the 33rd International Gas Turbine and Aeroengine Congress, Amsterdam, The Netherlands, June 6-9, 1988.

[3] Meitner, P.L. "Computer Code for Predicting Coolant Flow and Heat Transfer in Turbomachinery," to be published as AVSCOM TR No. 89-C-008, 1989

[4] SAS User's Guide, Version 5.16, 1987, SAS Institue Inc., Box 8000, Cary, NC 27512.

[5] Kays, W.M., and Crawford, M.E., "Convective Heat and Mass Transfer," McGraw-Hill Book Company, New York, Second Edition, 1980.

[6] Welty, J.R., Wicks, C.E., and Wilson, R.E., "Fundamentals of Momentum, Heat, and Mass Transfer," Second Edition, John Wiley & Sons, 1979.

[7] Kreith, F., and Bohn, M.S., "Principles of Heat Transfer," Fourth Edition, Harper & Row Publishers, New York, 1986.

[8] Han, J.C., and Park, J.S., "Developing Heat Transfer in Rectangular Channels with Rib Turbulators," International Journal of Heat and Mass Transfer, Vol. 31, No. 1, pp 183-195, 1988.

[9] Han, J.C., Park, J.S., and Ibrahim, M.Y., "Measurement of Heat Transfer and Pressure Drop in Rectangular Channels with Turbulence Promoters," NASA Contractor Report 4015 (or AVSCOM Technical Report 86-C-25), pp 1-197, 1986.

[10] Taslim, M.E., and Spring, S.D., "Experimental Heat Transfer and Friction Factors in Turbulated Cooling Passages of Different Aspect Ratios, Where Turbulators are Staggered," Paper No. AIAA-88-3014, AIAA/ASME/SAE/ASEE 24th Joint Propulsion Conference, July 11-13, Boton, Massachsetts.

[11] Boyle, R.J., "Heat Transfer in Serpentine Passages with Turbulence Promoters," ASME Paper 84-HT-24, presented at the 22nd National Heat Transfer Conference, Niagara Falls, NY , August 5-8, 1984.

[12] Metzger, D.E., Shepard, W.B., and Haley, S.W., "Row Resolved Heat Trasnfer Variations in Pin

Fin Arrays Including Effects of Non-Uniform Arrays and Flow Convergence," ASME Paper No. 86-GT-132, presented at the 31st International Gas Turbine Conference and Exhibit, Dusseldorf, West Germany, June 8-12, 1986.

[13] Metzger, D.E., Fan, Z.X., and Shepard, W.B., "Pressure Loss and Heat Transfer Through Rows of Short Pin Fins," Heat Transfer 1982, Vol. 3, U. Grigull et al. eds., Hemisphere Publishing Company, Washington, 1982, pp 137-142.

[14] Metzger, D.E., and Haley, S.W., "Heat Transfer Experiments and Flow Visualization for Arrays of Short Pin Fins," ASME Paper No. 82-GT-138, April 1982.

[15] Metzger, D.E., Berry, R.A., and Bronson, J.P., "Developing Heat Transfer in Rectangular Ducts with Staggered Arrays of Short Pin Fins," Trans. of ASME, Journal of Heat transfer, Vol. 104, Nov. 1982, pp 700-706.

[16] VanFossen, G.J., "Heat Transfer Coefficients for Staggered Arrays of Short Pin Fins," ASME Journal of Engineering for Power, Vol. 104, pp 268-274, 1982.

[17] Armstrong, J., and Winstanley, D., "A Review of Staggered Array Pin Fin Heat Transfer for Turbine Cooling Applications," Trans. of ASME, Journal of Turbomachinery, Vol. 110, Journal of Turbomachinery, January 1988, pp 94-103.

[18] Faulkner, F.E., "Analytical Investigation of Chord Size and Cooling Methods on Turbine Blade Requirements," NASA CR-120882, 1971, pp-189-194.

[19] Brown, A., Mandjicas, B., and Burgraf, F., "Blade Trailing Edge Heat Transfer," ASME Paper No. 80-GT-45.

[20] Lau, S.C., Han, J.C., aqnd Kim, Y.S., "Turbulent Heat Transfer and Friction in Pin Fin Channels With Lateral Flow Ejection," Trans. of ASME, Journal of Heat Transfer, Vol 111, No. 1, February 1989, pp 51-58.

[21] Metzger, D.E., and Sahm, M.K., "Heat Transfer around Sharp 180-Degree Turns in Smooth Rectangular Channels," Trans. of ASME, Journal of Heat Transfer, Vol. 108, August 1986, pp 500-506.

[22] Metzger, D.E., Plevich, C.W., and Fan, C.S., "Pressure Loss through Sharp 180-Degree Turns in Smooth Rectangular Channels," Trans. of ASME, Journal of Gas Turbines and Power, Vol. 106, July 1984, pp 677-681.

[23] Metzger, D.E., Fan, C.S., and Plevich, C.W., "Effects of Transverse Rib Roughness on Heat Transfer and Pressure Losses in Rectangular Ducts with Sharp 180-Degree Turns," Paper No. AIAA-88-0166, presented at the AIAA 26th Aerospace Sciences Meeting, Reno, Nevada, January 11-14, 1988.

[24] Fan, C.S., and Metzger, D.E., "Effects of Channel Aspect Ratio on Heat Transfer in Rectangular Passage Sharp 180-Degree Turns," ASME Paper No. 87-GT-113, presented at the ASME 32nd International Gas Turbine Conference and Exhibition, May 31-June 3, 1987, Anaheim, CA.

[25] Belvins, R.D., "Advance d Fluid Dynamics Handbook," Mcmillan Publishing Co, 1986, pp 56-70.

[26] Baughn, J.W., Iacovides, H., Jackson, D.C., and Launder, B.E., "Local Heat Transfer Measurements in Turbulent Flow Around a 180-deg Pipe Bend," Trans. of ASME, Journal of Heat Transfer, Vol. 109, February 1987, pp 43-48.

[27] Chupp, R.E., Helms, H.E., McFadden, P.W, and Brown, T.R., "Evaluation of Internal Heat Transfer Coefficients for Impingement Cooled Turbine Airfoils," Journal of Aircraft, Vol. 6, 1969, pp 203-208.

[28] Epstein, A.H., Kerrebrock, J.L., Koo, J.J., and Preiser, U.Z., "Rotational Effects on Impingement Cooling," Heat Transfer and Fluid Flow in Rotating Machinery, Wen-Jei Yang, Editor, Hemisphere Publishing Company, 1987, pp 86-101.

[29] Mori, y., and Nakayama, W., "Convective Heat Transfer in Rotating Radial Circular Pipes (1st Report, Laminar Region," Int. Journal of Heat and Mass Transfer, Vol. 11, 1968, pp 1027-1040.

[30] Morris, W., "Heat Transfer and Fluid Flow in Rotating Coolant Channels," Research Studies Press-Wiley, 1981.

[31] Clifford, R.J. "Rotating Heat Transfer Investigations on a Multi-Pass Cooling Geometry", paper presented at the Conference on Heat Transfer and Cooling in Gas Turbines, Bergen, Norway, May 6-10, 1985; AGARD-CP-390.

[32] Morris, W.D., and Ayhan, T. "Observations on the Influence of Rotation on Heat Transfer in the Coolant Channels of Gas Turbine Rotor Blades", Proc. I. Mech Engrs., Vol 193., 1979, pp 303-311.

[33] Morris, W.D., and Ayhan, T. "An Experimental Study of Turbulent Heat Transfer in a Tube which Rotates about an Orthogoanal Axis", Proc. XIV ICHMT Symposium on Heat and Mass Transfer in Rotating Machinery, Dubrovnik, Yugoslovia, August 30- September 3, 1982.

[34] Harsagama, S.P. and Morris, W.D. "The Influence and Rotation on the Heat Transfer Charecteristics of Circular, Triangular, and Square Sectioned Coolant Passages oof Gas Turbine Rotor Baldes" Trans. of ASME, Journal of Turbomachinery, Vol 110, January 1988, pp 44-49.

[35] Mori, Y., Fukuda, T., and Nakayama, W., "Convective Heat Transfer in a Rotating Radial Circular Pipe (2nd Report)," International Journal of Heat and Mass Transfer, Vol. 14, pp 1807-1824, 1971.

[36] Ito, H., and Nanbu, K., "Flow in Rotating Straight Pipes of Circular Cross Section," ASME Trans., Journal of Basic Eng., Vol. 93, No. 3, 1971, pp 383-397.

[37] Morris, W.D., "A Pressure Transmission System for Flow Resistance Measurements in a Rotating Tube," Journal of Phys. E: Sci. Instrum., vol 14., pp 208-214., 1981.

[38] Morris, W.D., and Dias, F.M., "Laminar Heat Transfer in Square Sectioned Ducts which Rotate in Parallel Mode," Power Ind. Res., Vol. 1, 1981.

[39] Mori, y., and Nakayama, W., "Forced Convective Heat Transfer in a Straight Pipe Rotating Around a Parallel Axis (1st Report, Laminar Region)," Int. Journal of Heat and Mass Transfer, Vol. 10, 1967, pp 1179-1194.

[40] Morris, W.D., and Dias, F.M., "Turbulent Heat Transfer in a Square Sectioned Tube", Journal of Mechanical Engineering Science, Vol. 22, no. 2, 1980, pp 95-101.

[41] Woods,J.L., and Morris, W.D., "A Study of Heat Transfer in a Rotating Tube", ASME Journal of Heat Transfer, Vol 102, Nov. 1980 pp 612-616.

[42] Nakayama, W., "Forced Convective Heat Transfer in a Straight Pipe Rotating Around a Parallel Axis (2nd Report, Turbulent Region)," Int. Journal of Heat and Mass Transfer, Vol. 11, 1968, pp 1185-1201.

[43] Morris, W.D., and Woods, J.L., "Heat Transfer in the Entrance Region of Tubes that Rotate about a Parallel Axis", Journal of Mech Engr Science, Vol. 20, No. 6, 1978, pp 319-325.

[44] Hajek, T.J., Wagner, J.H., and Johnson, B.V., "Coolant Passage Heat Transfer with Rotation," Turbine Engine Hot Section Technology 1987, NASA CP-2444, October 1987, pp 211-223.

[45] Guidez, J. "Study of the Convective Heat Transfer in Rotating Coolant Channel," ASME paper no. 88-GT-35, presented at the 33rd International Gas Turbine and Aeroengine Congress, Amsterdam, The Netherlands, June 6-9, 1988.

[46] Taslim, M.E., and Spring, S.D., "Friction Factors and Heat Transfer Coefficients in Turbulated Cooling Passages of Different Aspect Rations, Part I: Experimental Results," Paper No. AIAA-87-2009, presented at the AIAA/ASME/SAE/ASEE 23rd Joint Propulsion Conference, June 29 - July 2, 1987, San Diego, California.

[47] Taslim, M.E., and Spring, S.D., "An Experimental Investigation of Heat Transfer Coefficients and Friction factors in Passages of Different Aspect Ratios Roughened with 450 Turbulators," Proceedings of the 1988 ASME National Heat Transfer Conference, Houston, Texas, July 24-27, 1988, Vol. 1, pp 661-668.

[48] Chandra, P.R., Han, J.C., and Lau, S.C., "Effect of Rib Angle on Local Heat/Mass Transfer Distribution in a Two-Pass Rib-Roughened Channel," ASME paper no. 87-GT-94 presented at the 32nd International Gas Turbine Conference and Exhibition, May 31 - June 4, 1987, Anaheim, California.

[49] Lau, S.C., Kim, Y.S., and Han, J.C., "Effects of Fin Configuration and Entrance Length on Local Endwall Heat/Mass Transfer in a Pin Fin Channel," ASME paper No. 85-WA/HT-62, presented at the ASME Winter Annual Meeting, Nov 17-21, 1985, Miami Beach, Florida.

[50] Peng, Y., "Heat Transfer and Friction Loss Charecteristics of Pin Fin Cooling Arrangements," Trans. of ASME, Journal of Engineering for Gas Turbines and Power, Vol. 246, January 1984, pp 246-251.

[51] Iskakov, K.M., and Trushin, V.A., "The Effect of Rotation on Heat Transfer in the Radial Cooling Channels of Turbine Blades," Thermal Engineering, Vol. 32, No. 2, 1985, pp 93-96.

Report Documentation Page

1. Report No. NASA TM-102079 AVSCOM TR 89-C-013	2. Government Accession No.	3. Recipient's Catalog No.

4. Title and Subtitle		
A Generalized One Dimensional Computer Code for Turbomachinery Cooling Passage Flow Calculations		5. Report Date
		6. Performing Organization Code

7. Author(s)	8. Performing Organization Report No.
Ganesh N. Kumar, Richard J. Roelke, and Peter L. Meitner	E-4839

9. Performing Organization Name and Address	10. Work Unit No. 535-05-10 1L161102AH45
NASA Lewis Research Center Cleveland, Ohio 44135-3191 and Propulsion Directorate U.S. Army Aviation Research and Technology Activity—AVSCOM Cleveland, Ohio 44135-3127	11. Contract or Grant No.
	13. Type of Report and Period Covered

12. Sponsoring Agency Name and Address	Technical Memorandum
National Aeronautics and Space Administration Washington, D.C. 20546-0001 and U.S. Army Aviation Systems Command St. Louis, Mo. 63120-1798	14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

A generalized one dimensional computer code for analyzing the flow and heat transfer in the turbomachinery cooling passages has been developed. This code is capable of handling rotating cooling passages with turbulators (inline, staggered, or inclined), 180 degree turns (both sharp and gradual turns, with and without turbulators), pin fins, finned passages, by-pass flows, tip cap impingement flows, and flow branching. The code is an extension of a one-dimensional code developed by P. Meitner. In the subject code, correlations for both heat transfer coefficient and pressure loss computations have been developed to model each of the above mentioned type of coolant passages. The code has the capability of independently computing the friction factor and heat transfer coefficient on each side of a rectangular passage. Either the mass flow at inlet to the channel or the exit plane pressure can be specified. For a specified inlet total temperature, inlet total pressure and exit static pressure, the code computes the flow rates through the main branch and the subbranches (if any), flow through tip cap for impingement cooling (if any), in addition to computing the coolant pressure, temperature and heat transfer coefficient distribution in each coolant flow branch. Predictions from the subject code for both nonrotating and rotating passages agree well with experimental data. The code was used to analyze the coolant passage of a research cooled radial rotor.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Turbine cooling Cooling passage flows Heat transfer	Unclassified—Unlimited Subject Category 34

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No of pages	22. Price*
Unclassified	Unclassified	14	A03