N88-11159

HEAT TRANSFER IN THE TIP REGION OF A ROTOR BLADE SIMULATOR*

Metzger, D.E., Chyu, M.K., and Moon, H.K.

Mechanical and Aerospace Engineering Department
Arizona State University

INTRODUCTION

In gas turbine engines, the tips of axial turbine blades rotate in close proximity to a stationary peripheral outer ring or shroud. Differential thermal expansion of the rotating turbine wheel and blades and the stationary outer ring causes variations in the size of the clearance gap at the blade tip. For an aircraft engine, significant variations in clearance occur at different operating conditions such as takeoff, cruise, etc. [1]; and, even with active clearance control, the gap can never be eliminated entirely.

Pressure differences between the pressure and suction sides of a blade drives a flow through the clearance gap. This flow is generally detrimental to engine performance for two primary reasons. First, the leak reduces the turbine stage efficiency, and for this reason engine designers are continually concerned with reducing the clearance gap as much as possible.

A second detrimental effect of the clearance gap flow involves the convective heat transfer between the gap flow and the blade tip. The surface area at the blade tip in contact with the hot working gas represents an individual thermal loading on the blade which, together with heat transfer to the suction and pressure side surface area, must be removed by the blade internal cooling flows. Such cooling flows impose a thermodynamic penalty on engine performance, and in this general sense the blade tip heat transfer acts to degrade engine performance.

In addition, heat transfer rates at the blade tips have been observed to be very high; and there are difficulties and uncertainties involved with cooling of the tip region. The result is that turbine blade tips have traditionally been a region very vulnerable to structural damage. This damage can have a severe effect on engine performance. Loss of material from the blade tip increases the clearance gap, increases the flow and heat transfer across the tip, and in general exacerbates the problems with resulting additional structural damage to the tip, etc. Blade tip heat transfer is therefore quite important to the issue of engine durability.

The apparent flow complexities near the tip region, the region's relatively small size, and difficulties encountered in isolating it for measurement, have in the past hindered development of a thorough understanding of convection within the leakage gap. The objective of the

*Work done under NASA Grant NAG3-623.

PRECEDING PAGE BLANK NOT FILMED

current study is to acquire, through experimental and numerical modeling, an improved understanding of the nature of flow and heat transfer in the blade tip region.

BACKGROUND

Figures 1 and 2 depict qualitatively the gas path flowfields in the vicinity of the blade tip and clearance gap on the pressure and suction sides of the blade, respectively [2]. Clearance gaps in modern axial turbines are typically less than one percent of the blade height.

Near the pressure side of the gap, the flow is sink-like and mainstream gas is turned into the small gap. Strong secondary flows can be expected to be present in the gas path as a whole, and this can have the effect of bringing the hottest portion of the mainstream pressure side flow to the vicinity of the clearance gap, especially at the downstream portions of the blade. As the flow emerges from the suction side of the gap, it rolls into a vortex [3] as it meets the oncoming wall flow (Figure 2).

As mentioned in the preceding section, very little information exists regarding the flow and heat transfer within the clearance gaps themselves. Several studies, eg. [4], have addressed the leakage problem in terms of flow alone, in the absence of heat transfer, stimulated largely by its importance in axial compressor performance. A more recent study [5,6], used open channel flows in a water table (with flow leaking under fences or airfoil shapes) to simulate clearance gap leakage in turbine stages; but the heat transfer aspects of the problem were not addressed.

Limited heat transfer information has recently been obtained with short duration testing [7,8] which indicates that heat transfer rates as high as those present at the airfoil leading edge exist in the clearance gap. These measurements are limited in scope and as yet provide little in the way of understanding of the basic heat transfer mechanisms involved and of the variations in heat transfer associated with changes of parameters. In [7], however, results were obtained at several values of the ratio of tip surface temperature to gas temperature, and indicate that this ratio is not a dominant parameter for blade tip heat transfer. This is a common observation in turbulent forced convection which supports the choice of near unity ratios in the present study.

Another recent finding is that the sink-like character of the flow entering the clearance gap results in a thin entering boundary layer [9]. Thus pressure side boundary layer fluid is only a small percentage of the total leakage flow; and, in effect, the blade tip is exposed to the full mainstream gas temperature, as indicated schematically in Fig. 3. Also the leakage flow, viewed in a coordinate system fixed to the blade, appears identical to a conventional duct entrance flow, but with one moving wall. The present experimental approach takes this point of view.

Another recent finding exploited in the present approach is that the results of both [5] and [9] indicate that leakage through the gap is basically an inviscid, pressure-driven flow whose magnitude can be calculated from knowledge of the airfoil loading distribution alone. Normal clearance gap heights are, in effect, small enough that the flows through the gap are uncoupled from the details of the flowfields on either side, and to render it essentially two-dimensional. Fig. 4 shows typical resulting trajectories of the leakage flow across the clearance gap.

Almost all of the very limited published work dealing with clearance gap flows involves consideration only of plain flat blade tips. Various tip geometrical treatments (Fig.5) are of interest for their either demonstrated or potential reduction in leakage flow and/or tip heat transfer. There is an apparent total lack of information on heat transfer associated with these various tip treatments. Fig. 5(e), the grooved tip, appears to be the most common treatment in actual use, and is the focus of the present program.

Flow over the grooved tip can be categorized as flow over a rectangular cavity; and both flow and convection heat transfer in this basic geometry have been the subjects of extensive investigation for several decades, eg. [10-13]. In all cases the cavities studied have been installed in wind tunnel walls where the cavity is open to a usually well developed, zero pressure gradient flow over an otherwise smooth surface. The grooved tip differs from this situation by virtue of the confined nature of the geometry as well as by the proximity of the moving shroud. It is, in effect, a completely enclosed rectangular volume both supplied and relieved from narrow short slots at the top side corners. The degree of similarity between the heat transfer characteristics of the grooved tip and those of previous cavity studies is unclear at present.

The present project will model both the confined and moving wall aspects of the grooved blade tip. Meanwhile a preliminary study has been conducted to investigate the effect on heat transfer of confined narrow slot openings to cavities, in the absence of an adjacent moving wall [14]. There is some justification for anticipating that such stationary results may be representative of the blade tip problem as well. For example in [9], for the case of a flat ungrooved blade tip, it was found the relative motion of the shroud has a negligible effect on the blade tip heat transfer for clearance gaps with sizes usually encountered in practice.

Fig. 6 shows typical results from [14] for a confined stationary cavity in terms of local Nusselt numbers at locations on both the rib tops (1 and 2 upstream, 11 and 12 downstream) and on the cavity floor (3 through 10). This particular case is for a shallow cavity with depth to streamwise width ratio, D/W, of 0.1 for several values of clearance to width ratio. The solid symbols are values measured with the cavity present. For comparison, measured values on a corresponding flat surface (D=0) are also shown as open symbols.

The distribution of heat transfer shown in Fig. 6 is typical. Very high heat transfer rates are present on the top surface of the upstream rib, and these have the same general magnitude and character as those measured without the cavity. Nusselt numbers drop by a factor of from 3 to 5 on the adjacent cavity floor, and then tend to rise from front to rear on the floor. These cavity floor values are found to be in quite good agreement with previous unconfined cavity results [12] for these shallow cavities; but as the relative cavity depth is increased the confined cavity exhibits higher floor heat transfer rates than does the corresponding unconfined cavity.

Heat transfer on the top of the downstream rib is back up to its level on the top of the upstream rib. In this case, however, the highest value of Nu is on the upstream end, rather than on the downstream end as is the case for the top of the upstream rib. This behavior supports the notion that the source of the downstream gap flow is largely fluid moving downstream adjacent to the shroud, partially impinging on and flowing down the downstream rib toward the cavity floor. Thus there would be little or no separation at the downstream gap entrance as is the case in the upstream gap.

In summary, the preliminary study without relative shroud motion does indicate that the presence of a transverse cavity lowers heat transfer rates in a narrow gap entrance flow; but the effects of the adjacent moving shroud have not yet been determined. These effects will be investigated with the apparatus (now nearing completion) and planned computations described in the next section.

RESEARCH PLAN

In order to achieve the desired resolution of local transfer coefficients and to avoid experimental uncertainties inherent in the direct measurement of local heat transfer, the experiment for the moving-shroud situation utilizes the analogy between heat and mass transfer [15], and local mass transfer coefficients are inferred from sublimation rates on naphthalene surfaces [16]. The thermal counterpart of a naphthalene mass transfer surface is the convective heat transfer from a wall with uniform wall temperature. One of the most important advantages of using such a mass transfer system is the capability of obtaining this well-defined boundary condition which is practically difficult to establish via direct thermal methods.

Figs. 7 and 8 display schematic views of the recently completed test section. The shaded area in these figures, representing a cavity-like grooved tip, is the mass transfer-active surface cast with a thin layer of naphthalene, approximately 1.5 mm in thickness. The naphthalene sublimes as it is exposed to air flow. To ease fabrication and measurement, the test cavity is comprised of three separate aluminum flat plates, each representing the upstream wall, the cavity floor and the downstream wall, respectively. Naphthalene is cast on designated areas of these plates before the cavity is assembled. The cavity aspect ratio can be varied by either adjusting the mounting screws or changing the complete set of aluminum plates.

The moving-shroud is modeled by a flat, seamless, Neoprene belt (Belting Industry, #BIC 500) driven by a speed-adjustable, 3/4 HP, D.C. motor (Dayton Inc.). In conjunction with the motor speed controller, additional pulleys and speed-conversion sheaves are installed to provide various belt speeds as desired. The cavity can be rotated in different orientation relative to the belt moving direction. During a test run, as shown in Fig. 8, the laboratory compressed air supply is first introduced to a plenum adjacent to the test cavity, then flows over the cavity, and subsequently discharges to the surrounding atmosphere. The entire test assembly including plenum and cavity is situated above and contacts the moving belt, with the cavity opening facing downward. Between the test assembly and the moving belt, there are several teflon pads mounted on the contacting surface of the test assembly to reduce dynamic friction and to prevent air leakage. An additional teflon plate is placed underneath and against the belt which effectively eliminates belt vibration as it moves over the test section. Preliminary tests at full anticipated belt speeds indicate that this design is very satisfactory.

To determine the local mass transfer rates, naphthalene surface profiles are measured both before and after a test run, and point by point subtraction of the two profiles gives the sublimed depth of naphthalene which is used in calculating the local mass transfer coefficient. Prior studies [17,18] indicate that the surface profile measurement procedure is tedious and time consuming; it may takes more than an hour to measure approximately two hundred data points, and even longer for data reduction. Meanwhile, human error and the long measurement time may lead to significant inaccuracies. To eliminate these errors, increase precision, and reduce the time required for data acquisition, an automated, microcomputer controlled measurement system has been developed. The system provides vast improvement with regard to data accuracy as well as operational convenience. Fig. 9 shows a block diagram giving a schematic view of the entire arrangement of the measurement system. The system consists of a depth gauge along with a transducer amplifier, a digital multimeter, a two-axis, stepper-motor driven positioner, a stepper-motor controller, and a microcomputer.

The depth gauge used to measure the naphthalene surface profile is a linear variable differential transformer (LVDT, Schaevitz Engineering) capable of measuring depth change with a resolution of approximately 2.5 x 10^{-6} mm (one microinch). This resolution is two-to-three orders of magnitude smaller than the operating range of measured naphthalene sublimation. The gauge head, featuring a very light load, provides virtually traction-free operation. The LVDT is connected to an electronically compatible transducer amplifier (Schaevitz Engineering, ATA101) which supplies excitation and converts the AC signal output of the gauge to a DC voltage which can be calibrated to provide a signal of 1 volt per 2.5 x 10^{-3} mm (0.001 inch). The output of the amplifier is measured with a digital voltmeter (Fluke 8840A, $5^1/_2$ digits) and recorded on the floppy disk of the microcomputer for further data reduction.

The two-axis positioner, produced by LinTech Inc., is used to locate the different measurement locations on a test surface. Each axis of the positioner is driven by a stepper-motor (Superior Electric, MO-63), and has the same 457.2 mm (18 inch) travel range. With the motor controller (Allen-Bradley, Fastrak 4000) connected, the stepper-motor

C-3

rotates 200 steps per revolution and, combined with the pitch of lead screw, is capable of transforming each step into a linear increment of one thousands of an inch in each axis.

To ensure that the system has proper operation throughout the measurement process and to optimize the rate of data acquisition, a microcomputer (Zenith 150, IBM-PC compatible) along with a program written in BASIC language (Microsoft, GWBASIC) and IEEE-488 (National Instrument, GPIB) interfacing commands serves as a controller to the movements of stepper-motors as well as the flow of data. The bounds of measurement domain and distance of each step increment can be specified in the computer program. The computer then controls the number of steps of a stepper-motor to change location on the test surface by these specified increments during measurements. Through the IEEE-488 interface, as previously mentioned, the computer also provides space for data storage and executes data reduction procedure.

Successful execution of a local measurement in subliming mass transfer is critically dependent on precise positioning and accurate thickness change readings. Moreover, the successful measurement in which many important parameters are present and must be studied in detail requires rapid data acquisition. The computer-controlled, automated measurement system developed for the present study fulfills all of these requirements. Area-averaged results can also be obtained from the local data via numerical integration if desired. At present, the hardware has been successfully installed and tested, and the software development is nearly complete.

Although the present study primarily emphasizes an experimental approach, a numerical computation simulating the experimental study is also planned. A finite-difference program, frameworked by the SIMPLER algorithm [19], is employed for solving the prime variables, the velocity components and pressure of the Navier-Stokes equations. For turbulence modeling, a two-equation, k-ε model with modifications for low-Reynolds-numbers will be used [20]. The entire computer program has been successfully developed, and is available in an IBM-3081 mainframe computer at Arizona State University. Simulations will be carried out with actual geometric and flow parameters from the experiments as inputs to the program.

SUMMARY

The present study of heat transfer in the tip region of a rotor blade simulator is now in its initial stages. The objective is to acquire, through both experimental and computational approaches, improved understanding of the nature of the flow and convection heat transfer in the blade tip region. Such information should enable designers to make more accurate predictions of performance and durability, and should support the future development of improved blade tip cooling schemes.

REFERENCES

1. Hennecke, D.K., "Heat Transfer Problems in Aero-Engines," Heat and Mass Transfer in Rotating Machinery, Hemisphere, Washington D.C.,1984, pp. 353-379.

2. Metzger, D.E. and Mayle, R.E., "Heat Transfer - Gas Turbine Engines," Mechanical Engineering, Vol. 105, No. 6, 1983, pp.44-52.

3. Allen, H.W., and Kofskey, M.G., "Visualization Study of Secondary Flows in Turbine Rotor Tip Regions," NACA TN 3519, 1955.

4. Lakshminarayana, B., "Methods for Predicting the Tip Clearance Effects in Axial Flow Turbomachinery," Journal of Basic Engineering, Trans. ASME, Vol. 92, 1970, pp. 467-482.

5. Booth, T.C., Dodge, P.R., and Hepworth, H.K., "Rotor-Tip Leakage: Part I - Basic Methodology," Journal of Engr. for Power, Trans. ASME, Vol. 104, 1982, pp.154-161.

6. Wadia, A.R. and Booth, T.C., "Rotor-Tip Leakage: Part II - Design Optimization Through Viscous Analysis and Experiment," Journal of Engr. for Power, Trans. ASME, Vol. 104, 1982, pp. 162-169.

7. Dunn. M.G., Rae, W.J., and Holt, J.L., "Measurement and Analysis of Heat Flux Data in a Turbine Stage: Parts I and II, ASME Papers No. 83-GT-121 and 83-GT-122, 1983.

8. Epstein, A.H., Guenette, G.R., Norton, R.J.G., and Cao Yuzhang, "Time Resolved Measurements of a Turbine Rotor Stationary Tip Casing Pressure and Heat Transfer Field," AIAA Paper No. 85-1220, 1985.

9. Mayle, R.E., and Metzger, D.E., "Heat Transfer at the Tip of and Unshrouded Turbine Blade," Proceedings of the Seventh International Heat Transfer Conference, Vol. 3, 1982, pp. 87-92.

10. Roshko, A., "Some Measurements of Flow in a Rectangular Cutout," NACA TN No. 3488, 1955.

11. Chapman, D.R., "A Theoretical Analysis of Heat Transfer in Regions of Separated Flow," NACA TN No. 3792, 1956.

12. Yamamoto, H., Seki, N. and Fukusako, S., "Forced Convection Heat Transfer on Bottom Surface of a Cavity," Journal of Heat Transfer, Trans. ASME, Vol.101, 1979, pp. 475-479.

13. Aung,W. "An Interferometric Investigation of Separated Forced Convection in Laminar Flow Past Cavities," Journal of Heat Transfer, Trans. ASME, Vol.105, 1983, pp.505-512.

14. Metzger, D.E., and Bunker, R.S., "Cavity Heat Transfer on a Transverse Grooved Wall in a Narrow Flow Channel," ASME Paper No. 85-HT-57, 1985.

15. Eckert, E.R.G., "Analogies to Heat Transfer Processes," Measurements in Heat Transfer, ed. by Eckert, E.R.G. and Goldstein, R.J., Hemisphere Publication, Washington D.C., 1976.

16. Goldstein, R.J., Chyu, M.K. and Hain, R.C., "Measurement of Local Mass Transfer on a Surface in the Region of the Base of a Protruding Cylinder with a Computer-Controlled Data Acquisition System," International Journal of Heat and Mass Transfer, Vol. 28, 1985, pp. 977-985.

17. Goldstein, R.J. and Taylor, J., "Mass Transfer in the Neighborhood of Jets Entering a Crossflow," Journal of Heat Transfer, Vol. 104, 1982, pp. 715-721.

18. Goldstein, R.J. and Karni, J., "The Effect of a Wall Boundary on Local Mass Transfer for a Cylinder in Cross-Flow," Journal of Heat Transfer, Vol. 106, 1984, pp. 260-267.

19. Partankar, S.V., Numerical Heat Transfer and Fluid Flow, Hemisphere Publication, Washington D.C., 1980.

20. Jones, W.P. and Launder, B.E., "The Prediction of Laminarization with a Two-Equation Model of Turbulence," International Journal of Heat and Mass Transfer, Vol. 15, 1972, pp. 301-313.

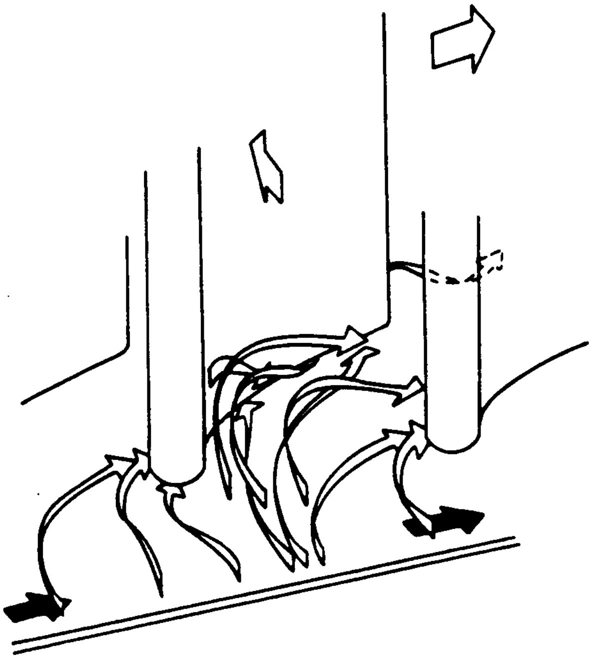

Fig. 1 Gap Vicinity Flow on Pressure Side

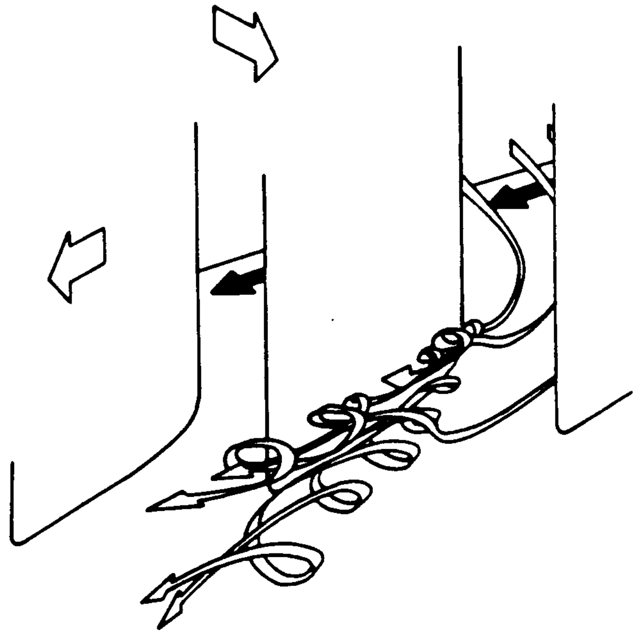

Fig. 2 Gap Vicinity Flow on Suction Side

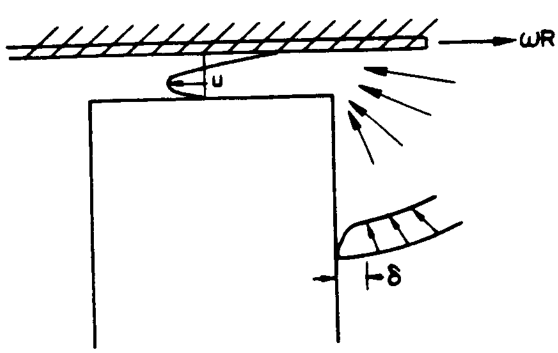

Fig. 3 Sink-like Character of Flow into the Clearance Gap

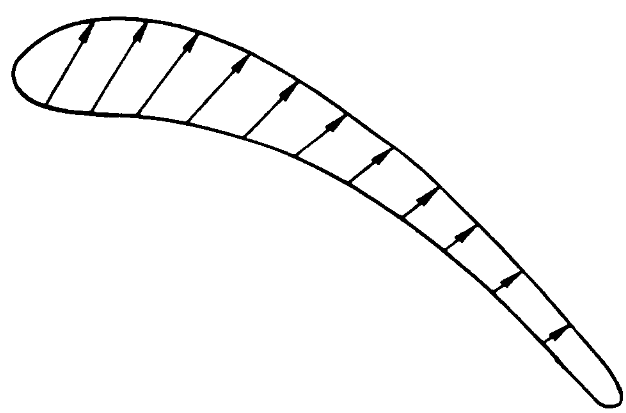

Fig. 4 Typical Clearance Gap Leak Flow Trajectories

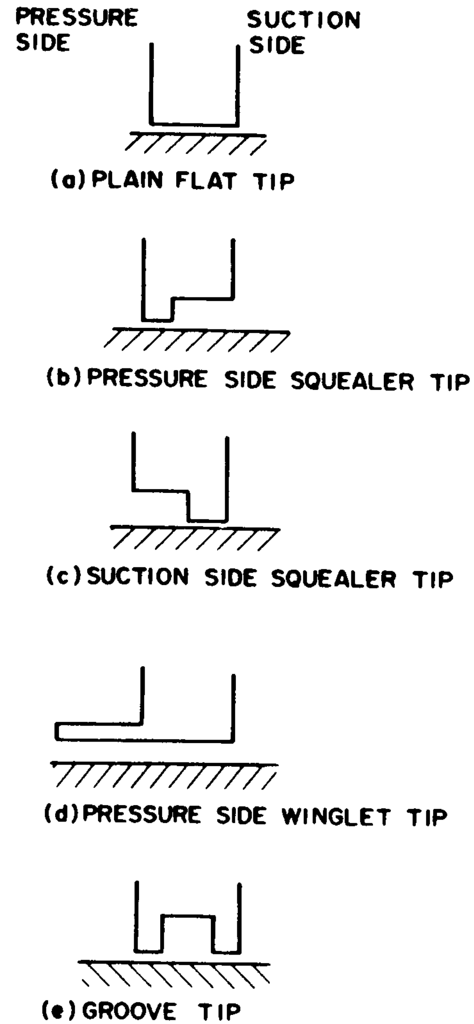

Fig. 5 Variation on Tip Geometry

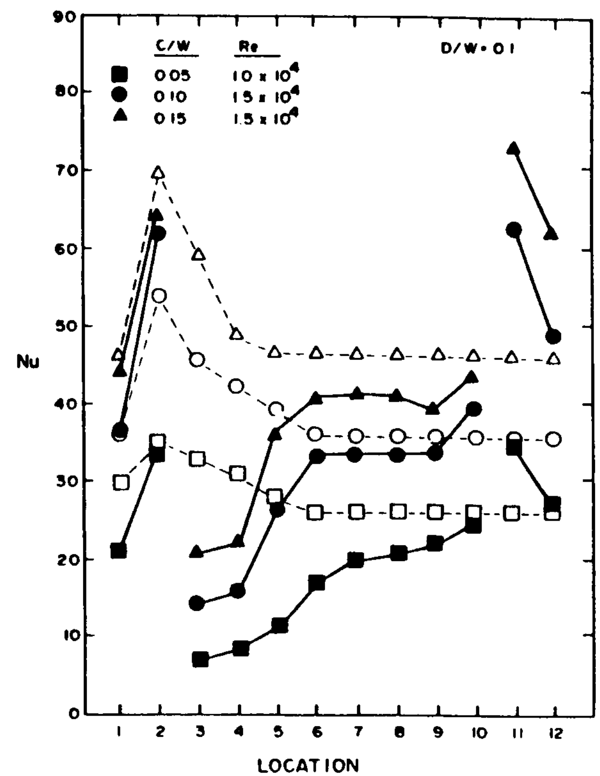

Fig. 6 Local Nu Distribution for Confined Cavities

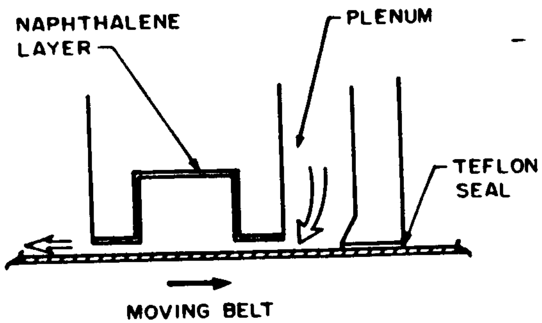

Fig. 7 Schematic View of Test Section

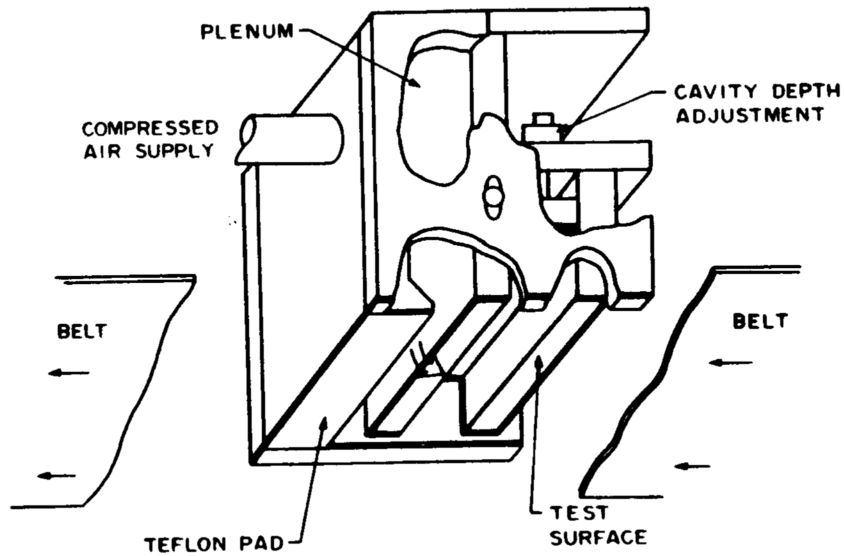

Fig. 8 Schematic View of Test Assembly

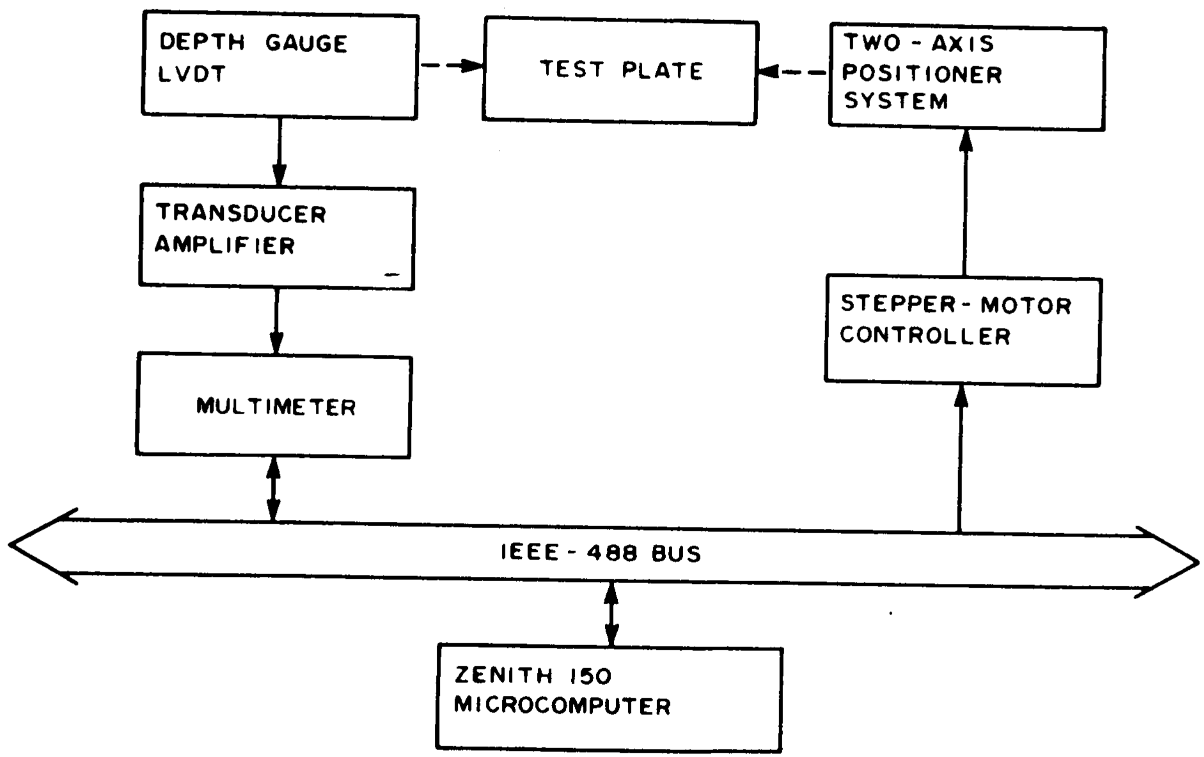

Fig. 9 Local Mass Transfer Surface Measurement System