CONCAVITY ENHANCED HEAT TRANSFER
IN AN INTERNAL COOLING PASSAGE

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ABSTRACT
The present study evaluates an innovative approach for enhancement of surface heat transfer in a channel using concavities, rather than protruding elements. Serving as a vortex generator, a concavity is expected to promote turbulent mixing in the flow bulk and enhance the heat transfer. Using a transient liquid crystal imaging system, local heat transfer distribution on the surface roughened by an staggered array based on two different shapes of concavities, i.e. hemispheric and tear-drop shaped, have been obtained, analyzed and compared. The results reveal that both concavity configurations induce a heat transfer enhancement similar to that of continuous rib turbulators, about 2.5 times their smooth counterparts 10,000 ≤ Re ≤ 50,000. In addition, both concavity arrays reveal remarkably low pressure losses that are nearly one-half the magnitudes incurred with protruding elements. In turbine cooling applications, the concavity approach is particularly attractive in reducing system weight and ease of manufacturing.

NOMENCLATURE

- C: Specific heat; symbol for "hemispheric concavity"
- D: Concavity diameter, 19.1 mm
- E: Concavity depth, 4.8 mm
- f: Friction factor
- H: Height of test channel
- h: local heat transfer coefficient
- k: thermal conductivity of Plexiglas
- Nu: Spanwise-averaged Nusselt number
- Re: Reynolds number based on D
- S: symbol for "smooth wall"
- T: Temperature; symbol for "tear-drop shaped concavity"
- W: Width of test channel
- z: Direction normal to the Plexiglas wall

Greek Symbols

- Δ: Finite Difference
- ρ: density of Plexiglas
- τ: time

Subscript

- D: Concavity diameter
- h: Hydraulic diameter
- i: Initial or at channel inlet
- m: Mixed mean
- o: Smooth wall
- w: Wall

INTRODUCTION
To achieve the performance goals of advanced turbine engines, a turbine will need to operate at sustained turbine firing temperatures of more than 3000 F, while insuring adequate component life. Over the past decades, significant effort has been devoted to developing effective cooling strategy to maintain blade temperature below its metallurgical limit. As a result, modern turbine blades or vanes are equipped with internal cooling passages roughened by protruding ribs arranged in periodic patterns. The ribs, also termed turbulators, promote turbulent mixing in the bulk flow and enhance the heat transfer capability in the cooling channel. A similar enhancement mechanism with short pin-fins, instead of ribs, is also implemented in the trailing section of the blade.

It is a well known fact that heat transfer enhancement is always accompanied with side effects that are detrimental to the engine aerodynamics and cycle efficiency. The surface protrusion
induces excessive pressure loss which elevates the compressor load. The separated flowfield over discretely mounted ribs or pin fins can induce substantial cooling non-uniformity and thermal stresses. Furthermore, advanced cooling channel designs with broken ribs of complex patterns may be difficult to manufacture. Finally, the excessive rib material added to the system may be undesirable when reduction of engine weight-to-thrust ratio is critical.

The potential of heat transfer enhancement using concavity was recently mentioned in a couple of undocumented reports and a paper by Schukin et al (1995). The idea is to implement concavities on a solid surface that can substantially enhance the heat transfer from the surface. Such an enhancement is a result of the so-called "tornado-like" process that bursts from the concavity wall and advances to the bulk of the housed channel. The tornado-like jets inherit self-organized vortex motions and promote turbulent mixing in heat transfer. Ideally a concavity serves as a vortex generator. A review of the limited data available in the Russian reports reveals that the use of concavities generally enhances heat transfer by a factor of 2 to 3 over their smooth-surface counterparts. Nevertheless, one case using the "gradually outlined" concavities, with flows of relatively low Reynolds number, has resulted in an enhancement as high as 4.5 folds. These reports, however, provide no information on concavity shapes, array configurations, or hydrodynamic scales. Such ambiguity hinders any effective follow up research built upon their findings.

The main objective of the present study is to assess the potential of concavity enhancement specifically for cooling of turbine components, such as blades, vanes and combustor liners. Thus the flow conditions here have much higher values in Reynolds number than those suggested in the Russian studies. In terms of geometry, two different concavities were tested: (1) hemispheric cavity (baseline case), and (2) tear-drop shaped cavity. The latter case was pursued following the baseline tests with the goal being to reduce the area of inferior heat transfer inside the hemispheric cavity. The entire study consists of more than sixty test runs which include approximately ten runs for single concavities and fifty runs for concavity arrays in staggered forms. The extent of wake spread around a single concavity provides follow up studies with design guidelines for arrays comprised of multiple concavities.

**EXPERIMENTAL APPARATUS AND PROCEDURES**

The heat transfer experiments performed in the present study utilizes a thermographic imaging system based on thermochromic liquid crystals (TLC). While there are several different approaches to measure the local heat transfer coefficient using the TLC technique, the present study employs the so-called transient method (Vedula and Metzger, 1991; Yu and Chyu, 1995). A brief introduction of the transient method is given below:

Consider an isothermal test surface suddenly exposed to heated flow with a mixed mean temperature of $T_w$. The temporal response of the surface temperature, $T_s(t)$, can be modeled by the one-dimensional, transient heat conduction over a semi-infinite solid beneath the surface. Assuming the entire domain has a uniform initial temperature $T_i$, the temperature profile, $T(x, t)$ is governed by the following equation:

$$k \frac{\partial^2 T}{\partial x^2} = \rho C_p \frac{\partial T}{\partial t}$$  \hspace{1cm} (1)

$$- k \frac{\partial T}{\partial x} \bigg|_{x=0} = h(T_w - T_i)$$  \hspace{1cm} (2)

**Figure 1. Test Section and Liquid Crystal Imaging System**

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The solution of the above equation can be expressed as:

\[
\frac{T_m - T_i}{T_i - T_m} = 1 - \exp\left(-\frac{h^2 \alpha T}{k^2}\right) \text{erfc}\left(\frac{h \sqrt{\alpha T}}{k}\right)
\]

During an experiment, the magnitudes of \(T_m\) and \(T_i\) are usually measured by thermocouples. The value of \(T_i\) is equal to the ambient air temperature prior to the test. Since the values of \(\alpha\) and \(k\), both the material properties of the test section (Plexiglas), are readily available from the manufacturer's specs, the convective heat transfer coefficient, \(h\), can be derived from eq. (5), provided that \(T_m(x)\) is known. In the present study, the determination of \(T_m(x)\) used a custom developed liquid crystal imaging software based on Pentium PC. The imaging system records the time-span required to reach a prescribed temperature and then transforms the time-span to the local heat transfer coefficient over the test surface. Prior to actual tests, the intensity of light reflected from a liquid crystals surface is calibrated first against temperature. The present imaging system traces the time of each pixel within the field-of-view to reach the maximum intensity which is near the green color. According to the calibration, such an intensity based approach appears to be dependent very little on the viewing angle.

In an actual experiment, a perfect step change for the applied fluid temperature is not possible, and the flow temperature usually rises with time throughout the entire test. This complication is accounted for by modifying Eq (5) and use of superposition principle and Duhamel's theorem. The actual gradual change of the flow temperature at the inlet of test section is obtained by using a series of steps from a spline curve fit of the temperature data recorded. The solution is then becomes:

\[
T - T_i = \sum_{i=1}^{N} U(t - \tau_i) \Delta T_m
\]

where

\[
U(t - \tau_i) = 1 - \exp\left[\frac{h^2 \alpha (t - \tau_i)}{k^2}\right] \text{erfc}\left(\frac{h \sqrt{\alpha (t - \tau_i)}}{k}\right)
\]

During a test, a separate temperature recording system is used to record the thermocouple reading at the inlet of the test section. The local heat transfer coefficient can be inferred based on Eqs. (6) and (7).

Another complication lies on the determination of local bulk mean temperature, \(T_m(x)\), which is the sensible reference temperature for defining the local heat transfer coefficient. However, \(T_m\) cannot be readily measured during the process of a TLC transient experiment. Instead, it has to be determined after the test by either approximation or applying the principle of energy balance over the test channel. Chyu et al. (1997) have recently compared several different approaches in converting the heat transfer coefficient based conveniently on the inlet temperature, \(T_i\), to that based on \(T_m\). The present study uses one of the most effective techniques, the so-called “invariant heat flux” method, which is based on local energy balance with time marching. The values of \(h\) differ approximately 10% between their basis on \(T_i\) and \(T_m\). Such a difference can be substantially widened when the test section becomes longer.

Figure 1 shows a schematic of the entire test system. Air flow, introduced from an in-house compressor, was heated by an adjustable heater to the desired flow temperature before it enters the test section. A CCD color camcorder was positioned upright above the test section. The lighting condition was carefully adjusted so adequate illumination on the test surface, while...
avoiding the undesired light reflection, could be achieved. The video output from the camcorder was recorded in a VCR during the experiment. Later, via a frame grabber, the video recorded in the VCR is digitized and transmitted to the hard disk of a computer for further imaging processing and data reduction. The temperature of the mainstream air flow was measured by a K-type thermocouple located near the inlet section. The test began by heating the compressed air while diverting it away from the test section through a three-way ball valve. When the air flow diverted reached steady state in both temperature and flow rate, the three-way valve was suddenly opened and directed the heated air into the test section. The image and temperature recording started simultaneously.

Figure 2 reveals key geometric variables for the concavity arrays and the flow channel. The test section, made of 12.7 mm thick (1/2") Plexiglas plates, is a rectangular channel, 76.2 mm (3") wide and 304.8 mm (12") long. Since a typical test is completed within a minute, the thickness of the test plate is sufficient to ensure the validity of the one-dimensional conduction model for evaluating the local heat transfer coefficient. During a test run, either the bottom plate or both the top and bottom plates are roughened with concavities and coated with a liquid crystals layer. There are fourteen rows of concavities arranged in a staggered fashion. A concavity of hemispheric profile was fabricated using a 19.1 mm (3/4") diameter ball-mill oriented normal to the surface. The depth of the drilling (E) is 4.8 mm (3/16"), which is one-quarter the concavity diameter (D). The tear-drop shaped concavity was made by drilling a hemispheric concavity first, followed by a horizontal milling process with a 20-degree inclination. If the "center" of a concavity is defined at the point where the indentation is the deepest, the center-to-center distances, or pitches, among any adjacent concavities form an isosceles triangle for both arrays. The pitch size is fixed at 19.1 mm (3/4") along both longitudinal and transverse directions, which is the same as the concavity diameter (D) and approximately 2.3 times the diameter of the actual circular opening of a hemispheric concavity (8.24 mm). This 2.3 ratio is so chosen that heat exchanger arrays, e.g. pin-fins and tube bundles, generally optimizes their heat transfer enhancement, if arranged in such a fashion. However, no proven evidence in the literature implies if this is the case for concavities.

To provide a better contrast for the liquid crystal display, the backside of each test plate was painted black. Since the system is expected to be dominated by the vortex bursts, the channel height, or more specifically, the ratio of characteristic lengths between the concavity and the housed channel, is deemed to be one of the most important variables. The test channel was so designed to permit three different channel heights, i.e. 6.35 mm (0.25"), 19.1 mm (0.75") and 39.1 mm (1.5"). The corresponding value of channel-height to concavity-depth ratio (E/H) is 1.33, 4, and 8, respectively. Further, the corresponding channel width-to-height ratio (W/H) is 12, 4 and 2, respectively. The cases studied were comprised of twenty-one separate tests of different surface combinations and Reynolds numbers for a given channel height. Hence there are sixty-three runs in total. The Reynolds number based on the concavity diameter (ReD) ranges from 10,000 to 30,000. The corresponding Reynolds number based on the channel hydraulic diameter and bulk mean velocity (Reim) varies between 10,000 to 52,000. The maximum uncertainties of the heat transfer and friction data present in this study, estimated by the method of Kline and McClintock (1953), are less than 7%.

RESULTS AND DISCUSSION

To ease results presentation, a special nomenclature is defined as follows. The letters S, C and T represent smooth surface, hemispheric cavity, and tear-drop shaped cavity, respectively. A test case is represented by a symbol that combines these three letters; e.g. S(S/C), C(S/C), T(T/T), etc. The first letter signifies the surface on which the liquid crystals heat transfer measurement was made. The two subsequent letters included in a parenthesis and separated by a slash sign describes the surface conditions in the test channel. The symbol S(S/C) represents that the test channel has a smooth upper wall and a lower wall roughened by hemispheric cavities, and the data is for the smooth wall. On the other hand, the symbol C(C/S) means that the data is taken on the hemispheric-concavity-roughened wall in the same channel. Further, T(T/T) implies that the data is taken on one of the channel surfaces which are roughened by the tear-drop shaped concavities.

Figure 3 reveals the distribution of local heat transfer coefficient (αL) for different W/H ratios.
Figures 3 and 4 give the sample results of heat transfer on the surface of concavities for the cases of the narrowest channel height ($H = 0.25$). Figure 3 gives a two-dimensional, gray-level plots of the magnitude of heat transfer coefficient ($h$) on the concavity roughened wall for $Re = 15,000$. The region with brighter contrast implies a higher heat transfer coefficient than the darker region. The domain of the plots spans nearly four periods starting near the end of the third period, ensuring that the data present lie primarily in the fully developed regime. Figures 4(a) and 4(b) show the distributions of streamwise-resolved and spanwise-averaged heat transfer coefficient normalized by their smooth channel counterparts, i.e., $Nu_{h}/Nu_{o}$. This ratio is an indication of the extent of heat transfer enhancement by the concavity technique. The values of $Nu_{o}$ are the area averaged Nusselt numbers over a corresponding smooth duct, i.e., the S/S case. They compared reasonably well, within 10%, with the conventional Dittus-Boelter correlation for turbulent channel flow. Figure 5 exhibits the overall area-averaged, heat transfer enhancement. Note that, compared to a smooth surface, the effective area for heat transfer increases about 17% for the hemispheric concavity and 22% for the tear-shaped concavity. The following discussion is based collectively on the trends revealed from these figures.

Knowledge of the local surface heat transfer coefficient ($h$) or Nusselt number ($Nu$) is critical to the improvement of concavity cooling technology. As displayed in Figure 3, the liquid crystal imaging technique has indeed revealed detailed insight of the surface heat transfer distribution. The local heat transfer coefficient varies significantly and periodically over the entire surface, including the concavity. Compared the brightness level, the lowest magnitude of $h$ or $Nu$ always exists in the upstream portion of the hemispheric concavity, and the highest $h$ occurs on...
The flat surface immediately downstream of the concavity. Within a concavity, the value of \( h \) increases with the streamwise coordinate toward downstream. To a great extent this characteristic is similar to that prevailing in the rectangular cavity.

The current design of tear-drop shaped concavity is an attempt to minimize the area of low heat transfer coefficient in the hemispheric concavity. The data shown in Figure 3(b) mark characteristics distinctly different from those of the hemispheric concavity. While the heat transfer coefficient in the upstream portion of the tear-drop concavity is reasonably improved, one most significant improvement due to such a concavity re-shaping is the substantial increase in the generally low heat transfer coefficient inside a concavity. Compared to the hemispheric case, the local minimum \( h \), now with a much higher value, locates near the center of the concavity, rather than upstream part of the concavity. As evidenced in Figure 3(c), the magnitude of the local minimum heat transfer coefficient over the entire test domain, which always locates inside a concavity, increases by almost 50%. This leads to an important observation that the range of Nusselt number variation is substantially smaller, and it may offer more uniform cooling than its the hemispheric counterpart.

As a contrast to the substantial difference in minimum heat transfer, the local maximum heat transfer coefficients are comparable in both magnitudes and streamwise location. Similar to the hemispheric case, the flat region between adjacent tear-drop concavities remains the area of the highest heat transfer. To some extend, however, the upstream extension due to tear-drop shaping, may have caused a slight area reduction in the region that would otherwise inherit higher heat transfer due to wake shedding directly behind a hemispheric concavity. This finding suggests that further heat transfer enhancement may be possible with improved concavity designs.

To illustrate the effects of the opposite-wall conditions on the surface heat transfer, Figure 4 shows the spanwise-resolved \( Nu/Nu_0 \) on both concavity-roughened and smooth walls for different surface combinations. Although the sample results shown here are for \( W/H = 4 \) and Re = 23,000, they represent the general trends exhibited in all the cases studied. As expected, the heat transfer from a concavity roughened surface is generally higher with both channel walls roughened (C/C or T/T) than with only one wall roughened (C/S or T/S). However, such an effect is more pronounced with the tear-drop shaped concavity than with the hemispheric concavity. The values of \( Nu/Nu_0 \) for the hemispheric case are virtually the same, near unity, with a smooth or a roughened opposite wall. On the other hand, the magnitude of \( Nu/Nu_0 \) is higher for about 10 to 20% when both walls are roughened by the tear-drop shaped concavities. Hence the tear-drop shape appears to be more sensitive to the geometric nature of the opposite wall. This also implies that more active mixing in the flow bulk may be more significant for the tear-drop shape.

Figure 5 gives the overall heat transfer enhancement vs. Reynolds number for the present designs. The sample results in Figures 5(a) and 6(b) are for \( W/H = 4 \) and 12, respectively. The overall values of enhancement on the concavity roughened wall with either hemispheric or tear-drop shape are somewhat comparable, around 2.2 to 2.7, though the tear-drop shape being consistently higher. Having included the effect of increased area (17% for hemispheric and 22% for tear-drop), this level of enhancement is comparable to most of the rib turbulators but is lower than some of the complex broken ribs (Han et al., 1995). However, it is not unlikely that such levels of enhancement can be substantially increased with further innovative improvements.

All the trends discussed above appear to be insensitive to the channel height, or \( W/H \) ratio, and the Reynolds number within the present test range. However, the effect of vortex burst from the concavity wall was visibly evident with the narrowest channel (\( W/H = 12 \)), as the liquid crystals changed their colors in a pattern according to the array configuration on the opposite smooth wall during the actual transient test. Equally interesting is that this effect produces virtually no quantitative influence on the heat transfer on that surface. The overall implication is that, although vortex generation is directed to enhance the bulk mixing in a channel, its effect prevailing in the near wall region remains to be most dominating, at least from the heat transfer standpoint. It is possible that the most effective vortex bursting may only predominate in a specific range of the characteristic length ratio between the concavity and the housed channel. Nevertheless, this geometric factor is expected to be dependent on the flow conditions, such as the value of Reynolds number.

As an important criterion to evaluate any enhancement technology, Figure 6 shows the pressure characteristics for the concavity arrays as compared to their smooth channel counterparts. For a given Re, the case with both channel walls roughened by the tear-drop shaped concavities (T/T) inherits the greatest pressure loss, followed by C/C, T/S, and C/S. Consistent with the typical trend, the index of pressure penalty (\( f/f_0 \)) increases with the Reynolds number. Most notable are the values of \( f/f_0 \) ranging between 1.5 to 5, that collectively represent a much lower pressure penalty than those of virtually all other blade cooling techniques based on protruding elements, such as rib turbulators and pin fins. The corresponding index of pressure penalty for rib turbulators in a square channel with a less than...
10% blockage ratio is around 6 to 10. The difference is by nearly a factor of 2. Figure 7 shows the relationship between the heat transfer enhancement (\(\frac{\text{Nu}}{\text{Nu}_0}\)) vs. pressure penalty (\(\frac{f}{f_0}\)). Also included in the plot for comparison are the corresponding data for several existing enhancement configurations (Han et al., 1995).

**SUMMARY AND CONCLUDING REMARKS**

As an exploring research, the present study examines the potential of using concavities, rather than protruding elements, for augmenting surface heat transfer with forced convection. The application is directed toward internal cooling of turbine airfoils. A concavity, serving as a vortex generator, is expected to promote the turbulent mixing in the flow bulk and, accordingly, enhance the heat transfer from all the participating walls. Using a custom developed, automated liquid crystal imaging system operated in a transient mode, the local heat transfer distribution on the surface roughened by an staggered array based on two different shapes of concavities, i.e. hemispheric and tear-drop shaped, have been obtained, analyzed and compared. Although the tear-drop shaped concavity consistently induces higher heat transfer than the hemispheric concavity, they result in similar levels of overall enhancement, about 2.5 times their smooth counterparts for 10,000 \(\leq \text{Re} \leq 50,000\). This enhancement is comparable to most of the rib turbulators, but is lower than some of the complex broken-rib configurations. In addition to such respectable heat transfer enhancement, both concavity arrays reveal remarkably low pressure penalty that is superior to any existing enhancement technology based on protruding elements. The improvement is nearly by a factor of two in terms of the magnitude of relative pressure drop coefficient. In light of the fact that this study marks only the beginning of a systematic research on an innovative cooling technology, further advances in both practical designs as well as fundamental understanding toward the subject are expected. Besides cooling enhancement, the concavity approach is particularly attractive in reducing system weight and ease of manufacturing.

**REFERENCES**


