# HEAT TRANSFER (LOSS OR GAIN) COEFFICIENTS FROM BARE FLAT PLATE SOLAR HEAT COLLECTORS

## Historical Values and Current Understanding

Submitted to:
the Zomeworks Program Office of the
DOUBLE PLAY SYSTEM: SUMMER COOLING AND WINTER HEATING
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(First Edition)
November 19, 2002

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## 1. INTRODUCTION

In solar swimming pool heating with an unglazed collector, as in the "double play" approach of providing summer (night-sky) space cooling and winter (daytime solar) space heating with an unglazed collector, the collector-to-air heat transfer coefficient is very important. There has been much confusion with this number. A significant amount of work has been done to clear up this confusion, and this work has been (to some degree) covered in my annotated bibliography (Ref. 17), but not yet incorporated into any solar energy textbooks. One reason is that humanity has enjoyed cheap fuels for about the last two decades, so that solar energy has not enjoyed much support in those years, and the solar energy textbooks (Refs. 4, 5, 6) have not been updated for about 20 years.

The end of cheap fuels is now in sight (Ref. 18), but for the time being, in the absence of current solar energy textbooks, we must still provide our own data and correlation sources. With this report it is intended to do that for the collector-to-air (external) heat transfer coefficient for the flat plate collector. The report will be a living (electronic) document on the Double-Play website, revised and expanded as needed. If significant errors are found they will not simply be corrected, but they will also be pointed out so that people will be aware of the earlier errors. Since it is expected that many of the people reading and using this material will not be heat transfer specialists, it has been attempted to explain things in detail.

In night cooling for summer cooling operation, the convection heat gains associated with a high heat transfer coefficient can reduce the collector effectiveness when the night sky is very cold and the air is warmer. In day heating for winter heating operation, the convection heat losses associated with a high heat transfer coefficient can reduce the collector effectiveness whenever the sun heats the collector to above the ambient air temperature.

In solar swimming pool heating the convection effects at night and the radiation to the night sky are unimportant, since the collector will be turned off anyway. The forced convection heat transfer loss effects during the hours of collection are however critical, because if these losses are large the collection can be lowered significantly, whenever the sun heats the collector above ambient temperature.

It should be noted that solar swimming pool heaters may well be used for Double Play applications, but a name still has to be invented for this (Triple Play may sound farfetched, and Menage a Trois is not quite right). In mid winter there is no use in heating the pool, and it may be possible to provide space heating with the collector. In summer at night there is no reason why the pool heater could not provide some space cooling. All that is neede is some extra plumbing and control hardware.

Knowledge of the convection heat transfer coefficients on a bare collector is quite important for those reasons. In this report the background on the understanding of these heat transfer coefficients is discussed in some detail, as is the present understanding. It is the intent to include this report in the "Double Play" website, to update it as necessary, and to use it as a vehicle for zeroing in on the most desirable numbers to use for equipment design. It should be noted that external heat transfer coefficients are anything but static numbers: any change in the wind direction or speed will change them, so that they will really be changing almost from second to second. Because of that, the choice of the "desirable numbers to use for equipment design" will really have to be a policy decision, to choose numbers that are fairly reliable yet conservative. Having a comprehensive report will make that choice easier.

## 2. THE PHYSICS OF A FORCED CONVECTION HEAT TRANSFER COEFFICIENT

When air (or water, or some other fluid) flows around a body, the flow involves two regions:

a. Very close to the body, the fluid viscosity causes the fluid to slow down, and a generally fairly thin boundary layer develops in which the fluid is stationary right next to the body, and in which the velocity increases gradually to match with the free-stream velocity a very short distance away. The behavior of the fluid in this boundary layer is controlled by viscous effects, like in the flow of molasses.

b. Outside of this very thin boundary layer, the fluid behaves as if it had no viscosity. The flow is determined primarily by the shape of the body, and the inertial effects in the fluid as it moves around the body. After the fluid has flown around the body, one normally has at least a thin wake containing the fluid that was in the boundary layers. Often however one can have an irregular and large wake behind the body. This is normally due to boundary layer separation effects on bodies that are not "streamlined" enough.

Curiously enough, it is only because of the effects of viscosity, and because of the existence of the boundary layers, that planes are able to fly at all. If air had no viscosity, wings would not be able to generate lift, and bees and birds could never get airborne. The boundary layer can however create problems. If an airplane tries to fly with its wings pointed too high, the boundary layer separates from the top surface of the wings and causes the airflow to separate also, the wings "stall" and lose lift, one gets a large and irregular wake behind the wings, and the airplane stops flying and instead drops down. Many have been killed because their airplane stalled when flying too close to the ground.

The further along the body in the flow direction, the thicker the boundary layer tends to become. Increasing the velocity makes the layer thinner. If the velocity is high enough or the flow length is long enough, the boundary layer flow can no longer be laminar, but instead changes to turbulent flow. Turbulent flow can most easily be seen in cigarette smoke. Ask a nicotine addict to hold a lighted cigarette in quiet air. The smoke will rise in a very orderly and laminar way for about one foot, and then suddenly switch into a turbulent and chaotic flow. Who said that cigarettes are good for nothing?

The boundary layer behaves like a layer of stagnant air that insulates the body and provides a resistance to heat transfer. That is where the heat transfer coefficient comes from in such "forced convection heat transfer" situations: the thicker the boundary layer, the higher the conduction length and hence the higher the resistance to heat transfer, and the lower the heat transfer loss (or gain) coefficient.

The German scientist Ludwig Prandtl was the first to develop equations and correlations on the behavior of the fluid in the boundary layer, about one hundred years ago. Since then a whole engineering field has developed in boundary layer theory, which has proved to be essential in the airplane industry, in heat transfer, in the design of golf balls, etc. Before Prandtl, many scientists maintained that flight was against the laws of physics, despite the fact that birds and bees clearly paid no attention to those laws. The first serious textbook in the field of boundary layer theory was the one of Schlichting (Ref. 11).

There are several dimensionless numbers one often encounters in forced convection heat transfer equations or correlations. Dimensionless numbers are truly dimensionless, a pure number that is the same whether you use English or SI (metric) units to determine them. These contain all of the fluid properties that control the physics of the situation, and also involve one characteristic dimension (e.g. tube diameter, size of the plate or wing, height of smoke column, etc.), the flow velocity, and the heat transfer coefficient. These numbers are:

**a. The Prandtl number Pr:**

Pr = (Specific Heat x Viscosity)/ Thermal Conductivity

The numerator of the Prandtl number gives an indication of how fast the velocity boundary layer thickens, and the denominator gives an indication how fast the temperature or thermal boundary layer thickens, as the fluid flows along the body. The ratio of these two quantities, as given by the Prandtl number, allows one to determine the relative thicknesses of the thermal and velocity boundary layers. This clearly affects the heat transfer in a given flow situation.

**b. The Reynolds number Re:**

Re = (Density x Velocity x Dimension)/Viscosity

This gives an indication of the ratio of the inertial forces over the viscous forces in the flow. For small values of the Reynolds number, the flow is laminar, like the flow of molasses. When the Reynolds number is large enough, the flow becomes turbulent and chaotic. In the case of the cigarette smoke experiment mentioned earlier, in which the flow changed from laminar to turbulent flow at roughly one foot above the smoking tip of the cigarette, this indicates that the Reynolds number reaches the laminar-to-turbulent-flow-transition-value when the "Dimension" (i.e. the height of the smoke column) reaches one foot.

**c. The Nusselt number Nu:**

Nu = (Heat Transfer Coefficient x Dimension)/Therm.Conductivity

This number allows one to calculate the heat transfer coefficient. Some more explanatory comments to be added....

In this report you will also find the "J-factor" or Colburn factor used. The J-factor is simply a combination of Nu, Pr, and Re that often seems to simplify the correlations. Do not worry about the J-factor, when you run into it. Simply take it apart with algebra, to get first at the Nusselt number and then at the heat transfer coefficient that you are really interested in.

The above may sound confusing, but these dimensionless numbers are very useful. Typically one has a heat transfer geometry of interest, surrounded by a fluid (say air), one determines the fluid properties (from fluid property data tables) and one chooses a characteristic "Dimension" of the geometry that makes sense for both the Reynolds number and the Nusselt number, one runs tests with a series of Reynolds numbers, one determines the heat transfer coefficients by heating or cooling the geometry and measuring the temperature difference between the surface and the fluid, and one calculates the Nusselt numbers. One now has a correlation of Nusselt number in terms of Reynolds number for one Prandtl number (that of air). For most gases the Prandtl number is more or less the same, so that is hard to change. That is however a clue to the usefulness of this approach: the correlation that was obtained can be used with equal confidence for air, argon, helium, carbon monoxide, and nitrogen, since the Prandtl number for all of these gases is nearly the same.

If Jurges (see below) had designed and performed his experiments carefully in such a way as to get a correlation of Nusselt number in terms of Reynolds number for the air Prandtl number, and had used a horizontal as well as a vertical inclination of the plate, the results would have been much more useful for solar energy collectors.

The above discussion clearly does not enable one to predict heat transfer coefficients, but it should help to make the following material more intelligible. It should be noted that one can often predict heat transfer coefficients quite accurately, using the book of Schlichting (Ref. 11), or other boundary layer theory references. There are also many articles one can find in which an author has documented experimental or analytical correlations on specific heat transfer situations, as is clear in Section 4.2 below, and in the list of references. In the Noam Lior discussion in Section 4.2, it seems clear that the whole field of convection heat transfer coefficients hinges on the Reynolds and Prandtl and Nusselt numbers.

## 3 HISTORICAL VALUES

In 1924, Jurges published a study (Ref. 1) on the heat transfer loss coefficient he measured for a 50 cm by 50 cm plate with sharp edges, positioned vertically and parallel to the wind, and heated with steam (presumably at 100 C). He did not attempt to obtain a non-dimensional equation or correlation, but simply gave a dimensional equations to cover two different velocity ranges, as follows:

h = a + b*V**n,

where h is in Btu/(hr sq ft deg F), and V is in feet per second. In this expression, as in many programming languages, "*" means that b and V are multiplied, and "**" means exponentiation, or "raised to the power." Thus V is raised to the power n.

The following table is taken directly from Table 9.8 of the McAdams (Ref. 2) discussion on the Jurges results.

**FACTORS IN THE ABOVE EQUATION**

| | Wind Below 16 Ft/sec | | | Wind from 16 to 100 ft/sec | | |
|---|---|---|---|---|---|---|
| | a | b | n | a | b | n |
| Smooth Surface | 0.99 | 0.21 | 1.0 | 0 | 0.50 | 0.78 |
| Rough Surface | 1.09 | 0.23 | 1.0 | 0 | 0.53 | 0.78 |

It can be appreciated that most collectors are much larger than 5o cm by 50 cm, that they are generally fastened to a roof, and that they do not start with a sharp edge. The boundary layers on a collector can hence be expected to be much thicker than those on the plate of Jurges, and because of that the heat transfer coefficients of Jurges should be much too high. The air-slowing effects of the surrounding roof should make the situation even worse. The rest of this report confirms all of that.

In 1933 a heat transfer text was published by William McAdams, an MIT Professor in Chemical Engineering renowned as an extremely careful scholar and researcher, and subsequently this book went through a number of editions (Ref. 2). In this book the results of Jurges were published to describe surfaces placed parallel to the wind, mentioning the plate size and the vertical position, but not the steam heating used by Jurges.

Given the impeccable reputation of McAdams, and given the fact that there were no other values and that the budgets were tight, the Jurges values were widely accepted in the solar energy field, first by Hottel (Ref. 3), and then in virtually all of the solar energy texts in the 1970s and 1980s (Refs. 4, 5, 6). Only the numbers below a 16 feet/second wind speed were mentioned (as shown in the Noam Lior material given below). Values even somewhat larger than the Jurges values were recommended by ASHRAE to describe the heat transfer loss coefficients on the outside of buildings (Ref. 7).

The first technically serious paper about the flat plate collector is the paper of Hottel and Woertz (Ref. 3) in 1942. In this paper a start was made on the formulation of what has since become known as the "Hottel-Whillier model" of the flat plate collector, in honor of Prof. Hoyt C. Hottel, and of Austin Whillier, who did his ScD thesis for Hottel (Ref. 8), and presented the results later at one of the first meetings of ISES (Ref. 9). Before the 1942 Hottel and Woertz paper the value of the external heat transfer loss coefficient was quite unimportant, because nobody would have known what to do with this coefficient anyway.

### 3.1 Sometimes the Errors in this Number are Unimportant

For many people, and for a long time, having the wrong values for the external heat transfer coefficient was not that harmful. Very few people used bare collectors, and most collectors had one or even two layers of glazing. In such glazed collectors, the outside heat transfer coefficient only represents one of a number of heat transfer resistances in series. The effect of an error in this number is reduced because of the other resistances. It might indeed even be cancelled out by the researcher: if in test number one the results are off because an excessively high external heat transfer coefficient is used, the researcher might make an adjustment to the values of the internal heat transfer coefficient, so that in test number two the right answer is obtained because now several of the input numbers are wrong, canceling each other.

ASHRAE accepted and recommended even larger numbers (Ref. 7) than the ones of Jurges, but there was really little concern. In most cases the main resistance to the heat transfer affecting the comfort in buildings was in the wall and not on the outside surface, and for ASHRAE the main purpose of the calculations was to make sure that the heating and cooling equipment specified was large enough.

### 3.2 In Solar Swimming Pool Heating or in a Double Play Application the Numbers Can Make a Big Difference

In the late 1960s and early 1970s I was designing and building a solar swimming pool heater for my house in Pasadena, and writing the associated do-it-yourself manual for the Copper Development Association or CDA (Ref. 10). At the time, I had been working and teaching in heat transfer for over 10 years. The Jurges heat transfer coefficients seemed much too large to me. If indeed the heat transfer coefficients were that large an unglazed swimming pool heater would make no sense. It seemed easy enough to use the book of Schlichting (Ref. 11) to determine the heat transfer coefficients one would get for a horizontal collector of reasonable dimensions, not for one of 50 cm by 50 cm. The results were roughly half as large as the numbers of Jurges, and these were the ones I published in my manual (Ref. 10, Figure 3) and used in my calculations. With these numbers my unglazed design seemed a reasonable proposition, and the performance seemed to agree fairly well with expectations, after the pool heater was turned on in March 1973.

The swimming pool heater is still operating properly, nearly 30 years later. It still provides over five months of pool temperatures above 80 F (instead of only two months before the pool got solar heating), and during the swimming season the pool temperature sometimes gets as high as 92 F. The pool heater got coverage in the June 1974 issue of Sunset magazine, and has been the subject of a number of publications (Refs. 12-16). Between the CDA and ASES, about 100,000 copies of the manual were distributed. It is impossible to say how many solar swimming pool heaters have been built based on the manual. At a solar energy conference in Mexico however I met with several architects and contractors who use the manual routinely to build such heaters for clients, and I was assured that in and around Cuernavaca (an elegant bedroom and resort community just south of Mexico City) there are many dozens of such heaters.

I am currently preparing an updated edition of the do-it-yourself solar swimming pool heating manual, which should be published in late 2003 or early 2004.

In a Double Play application the heat transfer coefficient numbers are also critical. If the heat transfer coefficients are too high, the collector will not be as effective in using solar inputs for winter heating, or sky cooling for summer cooling. It may be advisable to install windbreaks to reduce these heat transfer coefficients and to improve the collector performance, and if the windbreaks have reflective surfaces (for solar concentration and for a better viewfactor to the night sky) it might be even better. If wrong values are used for the heat transfer coefficient, the performance predictions will be wrong. One might end up with an installation where none is warranted, with the wrong size of installation, or with none at all in a location in which one should have been installed.

## 4. CURRENT UNDERSTANDING

### 4.1 Formal Changes in the Numbers on the Heat Transfer Coefficients

The "Energy Crisis" of the 1970s unleashed a wave of new R&D in solar energy, and it was almost inevitable that the Jurges numbers would be revised. Professor Noam Lior tells the story best. Prof. Noam Lior of the University of Pennsylvania wrote Chapter 4 in Volume 5 of the MIT series on Solar Heat Technologies, Fundamentals and Applications (Ref. 17), in which he gives an overview in Section 4.6 that is included verbatim and in full below.

**4.2 Noam Lior Section 4.6 of Reference 17, verbatim and in full: "Heat Transfer from the Collector Exterior"**

Unless the collector is well shielded from the wind, the convective thermal resistance between its exterior and the ambient is dominated by forced convection due to wind flow over the collector (otherwise, the resistance is associated with natural convection). It is a quaint fact that the only way to calculate the heat transfer coefficient h due to forced convection over flat plates till fairly recently was by using the dimensional empirical correlation developed by Jurges in 1924 (Ref. 1, see McAdams, Ref. 2) for flow parallel to a plate:

$$h = 5.7 + 3.8\ V,$$

where V is the wind velocity, with all units in SI. The lack of
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(((((FdW Note: Notice this is in SI and not in English units, and that it refers exclusively to the low speed part of the Jurges results.)))))
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chracteristic length and the independence from properties and inclinations limit the validity of this correlation severely. The Russian literature shows the use of dimensionless correlations in the laminar flow regime, which also takes angles of attack and yaw into account [Azevov et al (Refs. 19, 20), Azevov and Vakhidov Ref. 21)].

Before development of improved predictive equations for convective wind effects is attempted, it must be realized that (1) wind varies with time in speed, direction, and turbulence, and its mean velocity changes with height, (2) wind arriving at the collector is affected by the topography upstream of the collector and surrounding it [see Kind et al (Ref. 22), Kind and Kitaljevich (Ref. 23), Lee (Ref. 24)], (3) sharp differences in wind speed were found even on the face of the collector [Oliphant (Ref. 25)], (4) free stream turbulence of the wind, in part generated by upstream and surrounding obstacles, has an important effect on heat transfer and can explain the difference between wind tunnel results with low turbulence and results obtained in the natural environment, where the free stream turbulence may be 20% (based on the local velocity) and the convective heat transfer coefficient twofold higher [Test et al (Ref. 26), Francey and Papaioannou (Ref. 27), Lee (Ref. 24)], and (5) the shape of the leading edge of the collector affects convection at its surface downstream through such phenomena as separation, reattachment, and redevelopment [see Ota and Kon (Ref. 28), Test and Lessman (Ref. 29)].

Sparrow and coworkers [Sparrow and Tien (Ref. 30), Sparrow et al (Refs. 31, 32)] conducted a series of experiments in a windtunnel, to develop correlations for convective heat transfer, using the naphtalene sublimation technique as analog to heat transfer. Square and rectangular plates of about 2 to 5 inches size were placed at different angles of attack and yaw, and both windward and leeward plate configurations were investigated for Reynolds numbers from 20,000 to 100,000 (laminar flow). They developed a correlation for windward orientation:

$$J = 0.86\ Re^{-0.5}, \qquad (47)$$

where J is the Colburn factor:

$$J = Nu/(Re * Pr),$$

where

$$Re = [Density * Velocity * (4A/C)]/Viscosity,$$

and where

$$Nu = [h * (4A/C)]/Conductivity$$

where A is the area of the plate and C is the length of its perimeter.

They found practically no effect of angles of attack or yaw. With the collector leeward (wind blowing at its back), they found that for Reynolds numbers below 60,000 windward-face plates exhibit heat transfer coefficients about 10% higher than leeward-face ones, but this was reversed as Re exceeded 60,000: at Re = 100,000 the windward-face coefficient became 15% lower than the leeward-face one. They also determined that adding coplanar plates at the edges of the collector moves the highly convecting edge zones to these passive edge plates, and the heat loss can be reduced by up to about 10%.

The above correlation, as well as experimental results obtained by Kind et al (Ref. 22) and Kind and Kitaljevich (Ref. 23) obtained in highly turbulent flows generated in a wind tunnel, give heat transfer coefficients that may be as much as four-times lower than the Jurges correlation. Onur and Hewitt (Ref. 33) made convective heat transfer experiments with 6 inch models under a free jet and obtained results about 10% lower than those of Sparrow and coworkers. Kind and Kitaljevich (Ref. 23) also found that heat transfer coefficients for solar collectors mounted at an angle on a flat horizontal roof are 50% higher than those for collectors mounted flush with an inclined roof.

Truncellito et al (Ref. 34) obtained numerical solutions for turbulent forced convection over a plate with an angle of attack for Reynolds numbers up to 300,000. They found that the Nusselt number increases slightly with the angle of attack and that the J-factor is the same for Re approximately equal to 30,000 as that predicted by equation (47), but is increasingly larger as Re increases. Correlations of Nu as a function of Re, Pr, and the angle of attack were provided.

Lior and Segal (Ref. 35) have done experimental studies in a wind tunnel to determine convective heat transfer coefficients on a solar collector array composed of three parallel rows, all facing the wind, with variable spacing and inclination, and with a Reynolds number between 48,000 and 850,000. For the upstream plate the Colburn factor was found to be slightly higher than that predicted by equation (47). It was up to about 40% higher for the second plate, due to effects of the wake generated by the plate upstream, but only up to about 30% higher for the third plate, due to the flow pattern. It was a weak function of inclination and spacing for the two downstream plates.

Forced convection heat transfer information for external flow around cylindrical collectors can be found in the book by Zukauskas (Ref. 36) which deals exclusively with heat transfer from cylinders exposed to external flow.

**PROGRESS SUMMARY** Although important progress was made during the last decade in the understanding of forced convection over inclined plates for Reynolds numbers below 100,000, and in advancing beyond the limitations of the Jurges correlation, we have only begun to understand and to try to predict heat transfer due to wind in the natural environment and for realistic collector/surround geometries. The extension of the past work to larger Reynolds numbers, attainment of better agreement between results of different investigators, and the accounting for real geometries and the natural environment are needed.

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FdW Note. This is the end of Noam Lior's marvelous Section on external heat transfer coefficients in the MIT volume on collectors (Ref. 17, published in 1990). Some more recent material will be discussed in Section 4.3 (below), and recommended equations and numbers (that will probably change for a while before things settle down) will be presented in Section 4.4. There is a lot of work to do.

### 4.3 More Recent Results

Prof. Noam Lior will send me some more recent technical papers, that have come out since the publication of Ref. 17, and those will be discussed in this section. Any assistance from members of the Double Play group will also be appreciated.

### 4.4 Current Equations and Correlations

This section, still to be written, will involve the current equations and correlations that should be used instead of the equation of Jurges (Ref. 1), as obtained from the literature references mentioned in Sections 4.2 and 4.3 above.

### 4.5 Properties of Air, and Sample Calculations

This section, still to be written, will have detailed values on the properties of air (in SI and in English units), and will show detailed sample calculations using some of the equations and correlations of Section 4.4.

## 5. RADIATION COUPLING TO THE SKY, TO THE ATMOSPHERE, AND TO CLOUDS

This is still to be figured out and to be written about......

## 6. SITE-SPECIFIC EFFECTS

Heat transfer experiments to determine the value of the heat transfer loss coefficient can only be useful if the wind is clean (i.e. fairly free of gusts and of gross turbulence) and carefully measured, or if the gusts and the turbulence is properly quantified.

In a collector on a home one has a different situation. The home may be surrounded by trees or other homes. These might blanket the collector so that it has an extra low wind velocity environment, or they might channel the wind so that the collector almost seems to be in a wind tunnel. In addition one does not know the wind velocity locally, unless one installs an anemometer to measure the wind velocity locally. Otherwise one knows the wind velocity measured on a high (say 10 meter) meteorological tower at some local airport or on top of some high building. This is what is reported in the weather reports, and what gets incorporated into the "TMY" (Typical Meteorological Year) weather data tapes or discs for the nearest documented TMY site.

The shape of the collector is also a concern. A collector is not a sharp edged plate sitting in a free airstream. It is instead a rectangular box connected somewhere to a roof on a house. When the wind finally gets to the collector it has already been partly slowed down, and the collector starts off with a boundary layer that was created before the wind ever got to the collector.

Finally, everybody knows that the wind is not steady, but generally comes in gusts, and also has frequent changes in the precise direction. The heat transfer coefficient must be changing accordingly, virtually on a minute by minute or even second by second basis.

In "Double-Play Applications" one will not be able to expect very precise estimates of these variable and transient effects on a site-by-site basis. We must come up with some reasonable "rule-of-thumb" numbers or guidelines.

## 7. DESIRABLE MEASUREMENTS TO BE PERFORMED ON THE DOUBLE PLAY PROGRAM

Despite the fact that much progress has clearly been made since the days of Jurges, the uncertainty in heat transfer coefficients is still probably one of the more troubling uncertainties in defining the equipment performance in the Double Play approach. It is likely that at some point in the work on the Double Play program it will be desirable to perform some careful experiments.

These could involve getting some idea of the effect of the surrounding obstacles, such as trees, buildings, etc.

They could involve the use of artificial obstacles or windshields in the immediate vicinity to the collector, to reduce the heat transfer coefficients so as to maximize the heating influence of the sun and/or the cooling influence of the night sky.

They could involve carefully built and instrumented simulated collectors on which one small patch was designed and controlled (say with heaters) so as to determine the heat transfer coefficient in that area.

They could involve a simulated collector in which the whole front surface involved a massive copper plate, so as to have enough plate conduction to get a uniform temperature. By using heaters and thermocouples on this plate one could then get average heat transfer coefficients for the whole plate.

They could involve some measurements on radiation effects. One can make some measurements on the radiation to the night sky, to the atmosphere, and to clouds by making such a test surface (large or small) in shiny and black versions, and by measuring the difference. Some measurements (or estimates) of this radiation effect must be made, for the air temperature and the sky temperature are often very far apart, and crude efforts to lump their effects are bound to end in failure.

They could involve some more tests to investigate the effect of such things as roof (i.e. collector) angle, and wind from several angles.

All of these experiments are ones in which one is trying to get some number to within perhaps 20%. This may look crude and easy, but it is not. It will require very careful test design and construction, and very careful testing to get that close. It can be a very costly experience to do such a test, and then to find out later that the results are useless because some important factor was overlooked.

It should be noted that the wind direction is not constant, and that the factor influencing the heat transfer coefficient will hence change from day to day and even from minute to minute. Since one cannot possibly hope to be able to keep up with such a variable effect, the most one can hope for is for some approximate numbers that on the average will give the right answers. This will require some modeling, and this could simply involve getting someone to program a custom TRNSYS subroutine.

The final numbers should not be grossly in error, as were the numbers of Jurges (Ref. 1), that were around 4 times too high. It will however be best if the numbers are slightly too high. That way one would be able to sell and install systems with confidence, and the client might be pleasantly surprised, and not disappointed.

## 8. CONCLUSIONS AND RECOMMENDATIONS

Still to be written.

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