# INFLUENCE OF THE FLOW CONDITION (LAMINAR / TURBULENT) IN THE FLUID TUBE ON THE COLLECTOR EFFICIENCY FACTOR OF A FIN ABSORBER

Robert Hausner[1], Hubert Fechner[2]

[1] Arbeitsgemeinschaft ERNEUERBARE ENERGIE, A-8200 Gleisdorf, Feldgasse 19, Tel.: ++43 3112 5886, Fax.: ++43 3112 5886 18
[2] Österreichisches Forschungs- und Prüfzentrum Arsenal, A-1030 Wien, Faradaygasse 3, Obj. 219, Tel.: ++43 1 79747 299, Fax.: ++43 1 79747 594

**ABSTRACT.** It is generally accepted that the heat transfer from the fluid tube of a fin absorber to the fluid itself, depends to a great extent on the flow condition. Laminar flow, as compared to turbulent flow, can result in a considerable loss in efficiency which could often be avoided by the appropriate design of the absorber and the absorber arrangement in co-ordination with the actual use. According to the current theory, by optimising the degree of efficiency of a collector, by selecting the appropriate absorber tube, absorber arrangement and the flow rate, the collector output should be improved by several percent, as compared to operation under non-optimal conditions. In general, as an increase in the total friction pressure loss of the collector occurs, limits are set by the practical pump capacity.
This study examines the statement derived from the theory of an improvement in the collector efficiency with turbulent flow and the additional necessary pump power is calculated.

## 1 THEORY AND CONSEQUENCES FOR WORK IN PRACTICE

According to [1] the thermal output of a collector with a fin absorber is directly in proportion to the collector efficiency factor $F'$ and this in turn depends yet again on the heat transfer coefficient fluid – tube $h_{fi}$. The influence of $h_{fi}$ on $F'$ is even greater the higher the heat transfer coefficient absorber plate – ambient $U_L$ .
The heat transfer coefficient fluid – tube $h_{fi}$ depends quite considerably on the flow condition in the tube. $h_{fi}$ can be calculated according to [2], with due consideration to the physical properties of the fluid and the geometry of the tube. $h_{fi}$ –values in the laminar area of about 200 - 500 W/m²K and of about 1000 - 7000 W/m²K in the transition area laminar - turbulent are attained with corresponding change-overs in-between.
Common operating conditions for collectors are in the laminar area (Reynolds' number Re < 2320) respectively in the transition area laminar - turbulent (2320 ≤ Re < 10000). In rarer cases a fully turbulent flow is achieved ($Re \geq 10000$). With normal antifreezes the Reynolds' numbers drop compared to water to about 20 % to 50 % of the values.
A rough calculation for commonly available absorbers shows that the collector efficiency factor $F'$ and, therefore, the collector efficiency $\eta$ can reveal a dependence on the flow condition of several percent (about 3 - 8 % rel.). From this it follows that a collector which was measured in the laminar condition and comes more into the turbulent area when in use, produces better yields than corresponds to the test certificate (figure 1); but the reverse is also possible (e. g.: test with water, operating with antifreeze mixture).

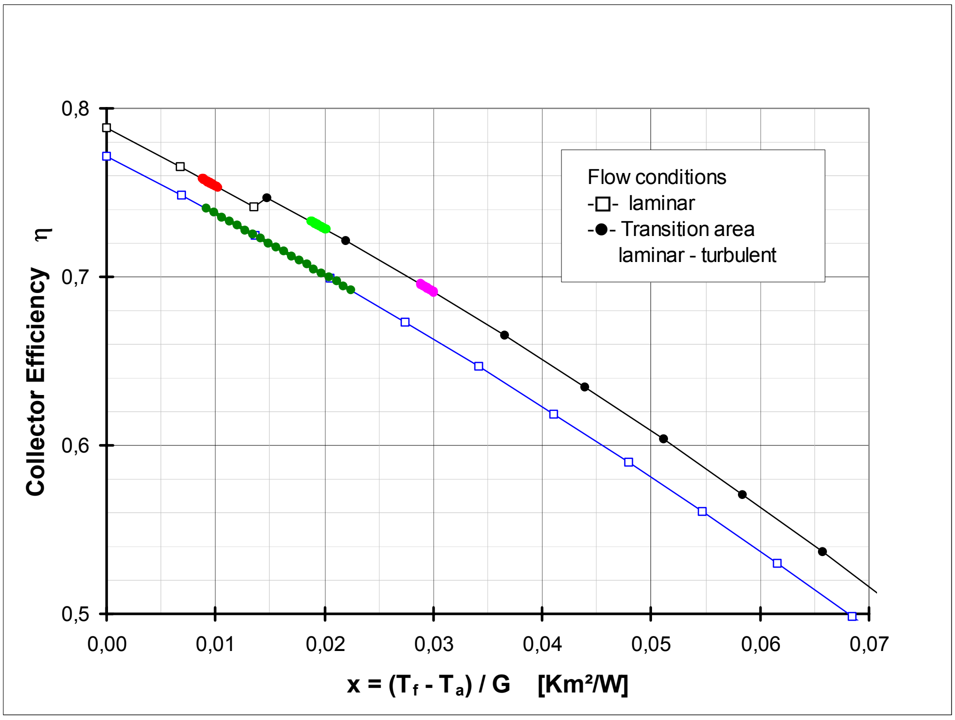


**Figure 1**: Simulation of improvement of collector characteristic line via the transition of laminar to turbulent flow.

**Lower curve**: Example of a collector measured in the extreme laminar area with a harp arrangement. The example drawn from this of a collector operating condition (thick line, entrance to collector $x$ = 0.009 up to collector exit x = 0.022) has the following mean characteristic values : $Re$ = 240, $h_{fi}$ = 218 W/m²K, $F'$ = 0.923.

**Upper curve**: Characteristic line calculated from the lower curve by simulation of the same collector for the **ten-fold** mass flow rate with the same fluid. The flow is still laminar up to $x$ = 0.014, above this the flow is from the transition area laminar-turbulent. The three collector operating conditions shown here on this line (thick lines with $x$ = 0.01, 0.02 and 0.03) have the following characteristic values: $Re$ = 2020, 2540 and 3130, $h_{fi}$ = 404, 933 and 1157 W/m²K, $F'$ = 0.944, 0.957 and 0.958.

For the examination of the collector this can also mean that as a result of the pronounced temperature dependence of the viscosity of the fluid in the lower temperature range, laminar, and in the upper temperature range more favourable flow conditions, set in from the transition area laminar-turbulent. A characteristic line like this should, therefore, demonstrate an increase in the transition area, which is not absolutely obvious in practical testing conditions due to the dispersion of the measured values and the large distance between the measuring points in the regression analytical evaluation.

## 2 SIMULATION AND MENSURATION PROOF

To make quantitative statements, a simulation programme was developed with which it is possible to theoretically calculate the collector characteristic line on the basis of the geometric and physical circumstances of the collector [1, 2]. In this respect the geometry of the absorber (fin, tube), its coating, the collector covering and insulation and the flow condition of the fluid are in particular taken into detailed consideration. An absorber strip is simulated in sections via its length. As an alternative, a measured characteristic line can be used as an input and be converted to other operating conditions. Unequal flow distributions on paralleled absorber tubes are calculated iteratively with due consideration to the resistance coefficients of the tube branch pieces dependent

on the flow distribution and to the properties of the fluid dependent on the temperature and concentration.
Within the framework of collector tests at the Austrian Research and Test Centre Arsenal series of measurements were performed on two suitable collectors to ascertain the dependence of the collector efficiency on the flow condition given a constant collector average temperature and a varied mass flow rate. Figures 2 and 3 compare the measured results with the simulation.
The simulation clearly predicts increases in the collector efficiency. For the area depicted in figure 2 (non-selective coating) 4.9 % (8.4 % rel.) was calculated. Since in the lower laminar area the measured collector is operated over a very wide temperature range, the curvature of the characteristic line is noticeable here as an additional influence. A calculation shows that in the case in point only 0.65 % are accounted for by the curvature of the collector characteristic line clearly making itself noticeable in the lower laminar area, the remaining 4.25 % can be put down to the effect of different flow conditions. The increase is particularly sharp in the area between about 120 and 150 kg/h of throughput amount (transition area).
The measurement also shows a clear increase (3.8 % - 6.5 % rel. – over the entire range) in which the transition area is also recognisable. One explanation for the lower value compared to the simulation – at least in part – can be that areas of disturbance stemming from the manufacturing process (unclean cutting edges, soldering points) cause additional swirling in the area of the inlet-T-pieces of the absorber tubes which have a greater effect in the laminar area.

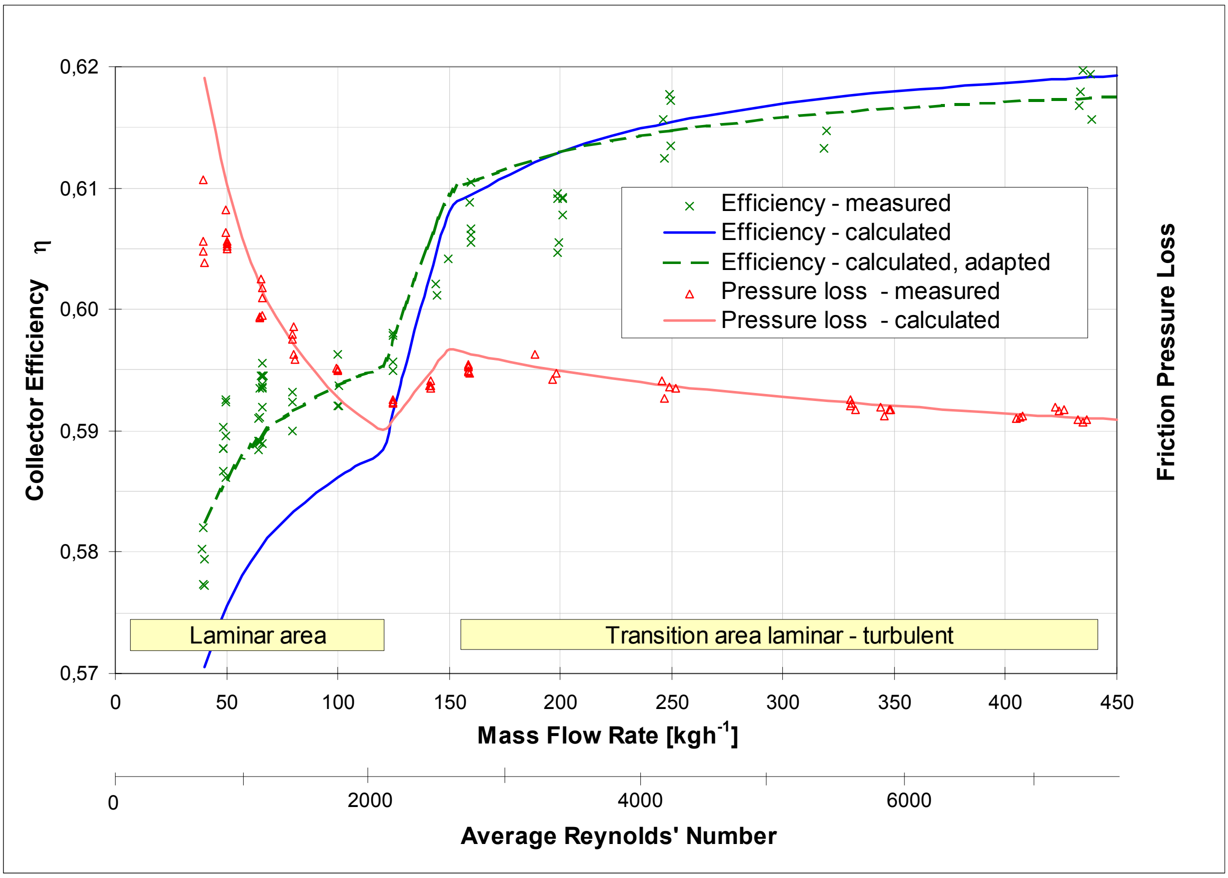


**Figure 2:** Comparison of simulated dependence of collector efficiency on the mass flow rate of a non-selective coated fin collector with a harp arrangement to measured values for the medium water, an average fluid temperature $T_f$ = 32 °C, solar radiation $G$ = 909 W/m² and an ambient temperature $T_a$ = 20 °C. The approximate recorded Reynolds' numbers are average values across the entire collector arrangement.

The simulated curve presented in figure 2 was further adapted with an adaptation factor to the measured data. The adaptation factor was defined as a multiplicative correction value, which is used on the theoretical efficiency changes. It was determined at a value of 0.72 so that the measured points are sufficiently depicted by the adapted theoretical curve. This means that the measured changes in the efficiency, as a result of variations in the mass flow rate, only equal about 72 % of the theoretical changes.
Moreover, the measured and calculated loss in pressure of the entire collector arrangement is depicted (total loss in pressure including the collection tubes and T-pieces) divided by the quadrat of the mass flow rate. This clearly shows the transition from a laminar to a turbulent flow. One recognises the excellent agreement of theoretical and measured values in the transition area and the good agreement in the upper laminar area and in the subsequent transition area.
The efficiency of the selectively coated collector (figure 3) reveals a lower dependence on the flow condition, as was to be expected. The increase in the efficiency with an increase in the mass flow rate is also obvious here. Details are lost here in the dispersion of the measured values.
For this collector we renounced the determination of an adaptation factor due to the relatively larger measured value dispersions compared to the overall change in the efficiency.

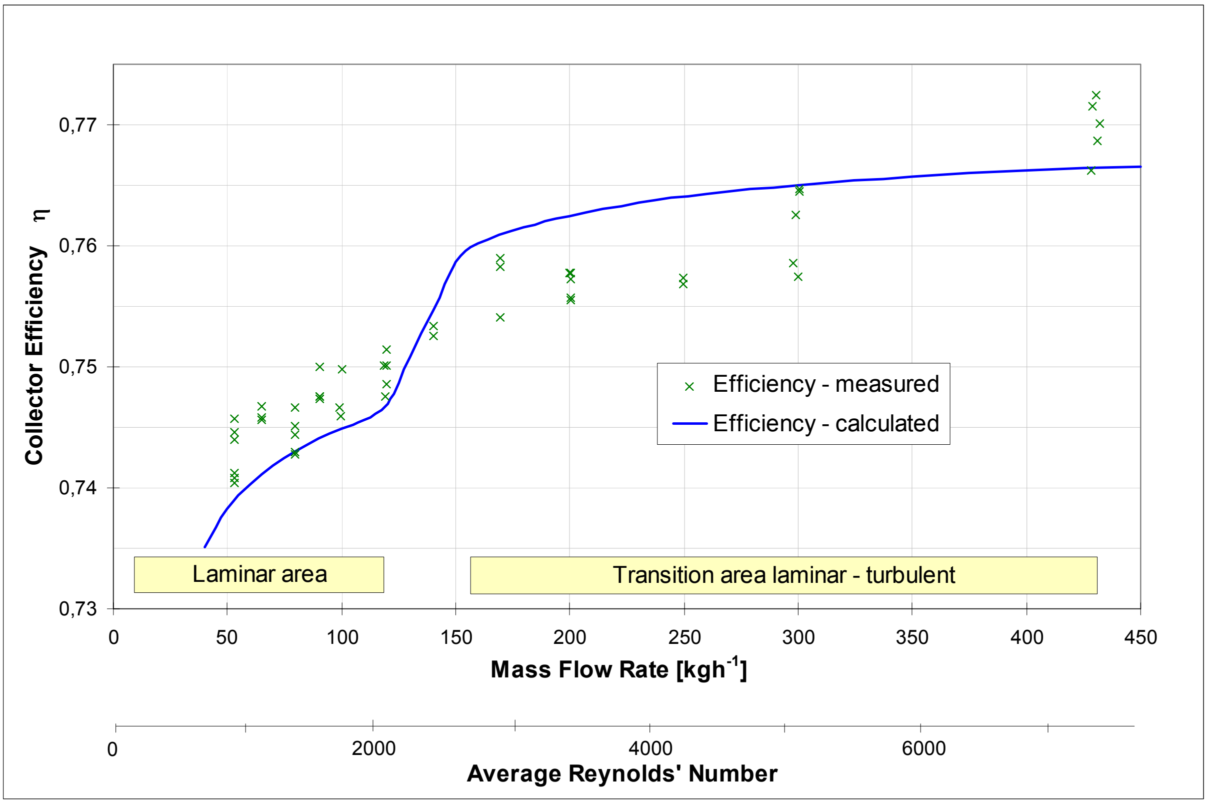


**Figure 3:** Comparison of simulated dependence of collector efficiency on the mass flow rate of a selectively coated fin collector with a harp arrangement to measured values for the medium water, an average fluid temperature $T_f = 32$ °C, solar radiation $G = 879$ W/m² and an ambient temperature $T_a = 20$ °C.

## 3 PUMP POWER

The improvement in heat transfer as a result of the transition to turbulent flow increases the loss in pressure. The increase in pump power connected to this was calculated as an example. Absorber strips were arranged differently for two collector fields (selective and non-selective) (figure 4), and the friction pressure loss and thermal collector performance for an operating condition which occurs frequently was calculated with due consideration to the factor for adaptation ascertained (see figure

2) for the overall collector circuit (collector, tubes, fittings, heat exchanger). The necessary electrical pump power was calculated on the basis of the hydraulic pump power via the pump efficiency [3].

The internal arrangement of the 36 absorber strips was performed in different variants beginning with complete paralleling to a complete serial connection. In the first variant (36 parallel) the hydraulic pump performance required for the entire collector circuit equals 1.5 W. For this about 54 W are to be used electrically [3]. Here the thermal collector output equals 4270 W for the selective type respectively 3370 W for the non-selective type.

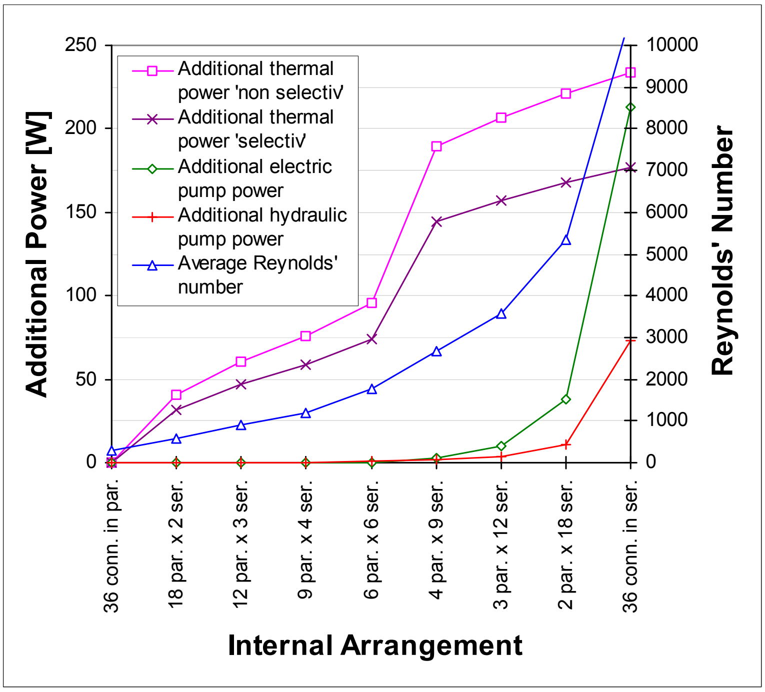


**Figure 4:** Simulation results for the gain in thermal power and the additional pump power needed for an exemplary collector field (9 m² strip absorber, selectively respectively not selectively coated) for a frequently occurring operating condition. Fluid: 35 % Propylenglycol, spec. mass flow rate 50 kg/m²h, average fluid temperature 50 °C, ambient temperature 20 °C, solar irradiation 800 W/m².

In this special example the optimum arrangement (in relation to the selected operating condition) is with 3 strips parallel and 12 such groups in series since here the additional gain in thermal collector power compared to the neighbouring arrangement variant (13 W with selective respectively 17 W with non-selective) still exceeds the additional pump power required (7 W). By including the primary energy requirement to make the electrical energy ready, variant 4 parallel x 9 serial would be the best but would, however, lead to construction problems with this collector type.

In this example the gain in thermal performance of the optimum variant (3 parallel x 12 serial) compared to the purely parallel arrangement (which is frequently found in practice) equals 157 W (3.7 % rel.) in the selective, respectively 207 W (6.3 % rel.) in the non-selective type and another 10 W electrical power (19 % rel.) are to be used for this in addition.

Even better conditions can be expected for the future with small pumps with enhanced efficiency rates which are currently being developed [3].

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