Engineering Notes

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Optimum Drag Balance for Boundary-Layer Suction

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Nomenclature

c = plate chord
D = drag force
L = power lost due to friction in suction system
l = length over which suction is applied
$\dot{m}$ = suction mass flow rate
p_s = pressure of sucked fluid at entry to control volume
p_0 = pressure of sucked fluid at exit from control volume
Re_{x_t} = Reynolds number based on transition position
Re_{x-x_0} = Reynolds number based on streamwise distance from transition
s = plate span
T = thrust due to expelled fluid
U_∞ = mean flow speed
v_s = suction velocity
v_0 = velocity of expelled fluid
W = power consumed
$W_{\text{momentum loss}}$ = power consumed by momentum loss
W_{pump} = power consumed by suction pump
W_t = total power consumed
W_{thrust} = power produced by thrust of expelled fluid
$W_{\text{wake drag}}$ = power consumed in overcoming wake drag
x = streamwise distance from plate leading edge
x_t = streamwise position of transition
x_0 = streamwise position of initial growth of turbulent boundary layer
η_p = efficiency of propulsive system
η_s = efficiency of suction pump
θ = momentum thickness
θ_c = momentum thickness at plate trailing edge
ν = kinematic viscosity of air
ρ = density of air
τ_w = shear stress

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Introduction

AMONG the methods investigated for the reduction of skin friction drag of commercial aircraft, surface suction is now considered as the most promising technique.[1,2] While real flight tests have been carried out in the U.S. and Europe, questions still remain concerning the optimization and distribution of suction.[3] In this brief Note, which is based on the early work of Head[4] and Edwards,[5] expressions for the various components of the power balance are derived analytically. They are then used with the simple example of a flat plate fitted with suction. Using the e^n method, an estimate of the skin friction drag of the plate is obtained and compared to the other components of the power balance. It is clearly demonstrated that, for a given mean flow speed and disturbance environment (n factor), an optimal transition position exists that results in the minimum power consumed in overcoming drag.

Power Balance for a Flat Plate Fitted with Suction

The overall power balance for a flat plate fitted with suction can be decomposed into the sum of four terms. The first term corresponds to the wake drag that is calculated from the momentum thickness at the trailing edge of the plate. The second term corresponds to the momentum loss at the surface of the suction panel that results from the withdrawal of fluid from the main stream. The third term is equivalent to the pump power required to suck fluid through the porous surface, raise its pressure to ambient, and compress it so that it can be expelled into the main flow at a prespecified velocity. The fourth component is due to the additional thrust produced by the fluid expelled into the main flow. The overall power consumed can then be written in the following form:

$$W_T = W_{\text{wake drag}} + W_{\text{momentum loss}} - W_{\text{thrust}} + W_{\text{pump}} \qquad (1)$$

The objective of boundary-layer suction is therefore to decrease the total power consumed by having the least drag and maximum thrust for the least pump power and momentum loss.

Wake Drag and Momentum Loss

The first two terms of Eq. (1) can be evaluated together by considering the momentum integral equation applied to a flat plate boundary layer with no pressure gradient.[6] For a flat plate the drag is obtained by integration of the shear stress. This gives

$$D = s\int_{x=0}^{c} \tau_w \, \mathrm{d}x = s\int_{x=0}^{c}\left[\rho\frac{\partial}{\partial x}(U_\infty^2\theta) + \rho v_s U_\infty\right]\mathrm{d}x \qquad (2)$$

Assuming one suction panel applying a uniform suction over a length l, Eq. (2) can be rewritten as

$$D = \rho s U_\infty^2\theta_c + \rho s l v_s U_\infty = \rho s U_\infty^2\theta_c + \dot{m}U_\infty \qquad (3)$$

The power that must be supplied to the propulsion system to obtain the required thrust and mean flow speed must also take into account the efficiency coefficient of the propulsion system. The power associated with the drag is thus given by the expression

$$W = \frac{U_\infty D}{\eta_p} = \frac{\rho}{\eta_p} sU_\infty^3\theta_c + \frac{\dot{m}}{\eta_p} U_\infty^2 \tag{4}$$

where

$$W_{\text{wake drag}} = (\rho/\eta_p)sU_\infty^3\theta_c \tag{5}$$

$$W_{\text{momentum loss}} = (\dot{m}/\eta_p)U_\infty^2 \tag{6}$$

Thrust and Pump Power

The sucked fluid when expelled produces an additional thrust given by the expression

$$T = \dot{m}v_0 \tag{7}$$

and the corresponding power produced is thus given by

$$W_{\text{thrust}} = (\dot{m}v_0/\eta_p)U_\infty \tag{8}$$

We wish to determine the power required to suck air through the porous surface, raise its pressure to ambient, and expel it into the main flow. We therefore define a control volume between the inlet of the perforated plate and the outlet of the suction system. For the given problem, the air is assumed to behave like a perfect gas with constant temperature and density, and height variations through the system are assumed small. The internal and potential energies are therefore constant. Considering also that no heat is supplied to the system, the first law of thermodynamics applied to the control volume yields[7,8]

$$W = \dot{m}[\tfrac{1}{2}(v_0^2 - v_s^2) + (p_0 - p_s)/\rho] + L \tag{9}$$

The additional term L corresponds to the energy losses due to friction and changes of cross section through the system. Accounting for the efficiency coefficient of the pump[9] then leads to

$$W_{\text{pump}} = (\dot{m}/\eta_s)\{\tfrac{1}{2}(v_0^2 - v_s^2) + [(p_0 - p_s)/\rho]\} + (L/\eta_s) \tag{10}$$

The losses due to the blockage of the porous surface are by far the most significant since the open area of perforated sheets usually used in boundary-layer suction is about 0.4%. Measurements of the pressure drop according to the suction flow rate have been performed by Bieler and Preist.[10] These give a model for the prediction of the pressure drop through the porous surface. Knowing the mass flow rate and the corresponding pressure drop, the losses can be calculated according to the formula

$$L = (\dot{m}/\rho)\Delta p \tag{11}$$

Substituting Eq. (11) into Eq. (10) leads to the final expression for the power necessary to drive the suction. It is given by

$$W_{\text{pump}} = (\dot{m}/\eta_s)\{\tfrac{1}{2}(v_0^2 - v_s^2) + [(p_0 - p_s)/\rho]\} + (\dot{m}/\eta_s\rho)\Delta p \tag{12}$$

The prespecified velocity at which the sucked fluid is expelled into the main flow must now be chosen in order to minimize the overall power. As proposed by Head,[4] the optimum velocity can be determined by calculating the derivative of the total power with respect to v_0 and equating this derivative to zero. Using Eqs. (1), (5), (6), (8), and (12), this is written as

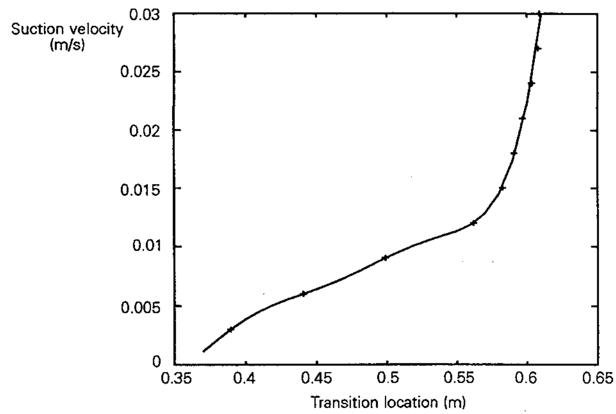

Fig. 1 Relation between transition location and suction flow rate; numerical results after Hackenberg et al.[12]

$$\frac{\partial W_T}{\partial v_0} = -\frac{\dot{m}U_\infty}{\eta_p} + \frac{\dot{m}v_0}{\eta_s} = 0 \tag{13}$$

The optimum velocity of the expelled fluid is, therefore,

$$v_0 = \frac{\eta_s}{\eta_p} U_\infty \tag{14}$$

which is of the same order of magnitude as the mean flow speed. In most applications of boundary-layer suction, v_s is less than about 2% of the mean flow speed, v_s^2 is therefore negligible compared to v_0^2 in Eq. (12). Using Eq. (14), the expression for the pump power now simplifies to

$$W_{\text{pump}} = \frac{\dot{m}}{\eta_s}\left(\frac{1}{2}\frac{\eta_s^2 U_\infty^2}{\eta p^2} + \frac{p_0 - p_s}{\rho}\right) + \frac{\dot{m}\Delta p}{\eta_s\rho} \tag{15}$$

Substituting Eq. (14) into Eq. (8) also yields

$$W_{\text{thrust}} = (\dot{m}\eta_s/\eta_p^2)U_\infty^2 \tag{16}$$

Estimation of the Position of Transition

In order to estimate the total power, the four contributions specified by Eqs. (5), (6), (15), and (16) need to be calculated. Assuming a known mean flow speed and suction rate, the only difficulty in the calculation of the total power is in fact due to the wake drag, which requires the knowledge of the momentum thickness at the trailing edge of the plate. For the flat plate configuration where the boundary layer is partly laminar and partly turbulent, the momentum thickness at the plate trailing edge can be estimated using a Blasius profile type growth in the laminar region and a one-seventh power growth in the turbulent region. According to the analysis presented by Young,[11] the momentum thickness in the turbulent boundary layer at the trailing edge of a flat plate is given by the expression

$$\theta_c = 0.037(c - x_0)Re_{x-x_0}^{-1/5} \tag{17}$$

This is related to the transition position by the expression

$$x_0 = x_T - 36.9x_T R_{x_T}^{-3/8} \tag{18}$$

Substituting Eq. (17) into the expression for the power because of the wake drag [Eq. (5)] leads finally to

$$W_{\text{wake drag}} = 0.037(\rho/\eta_p)sU_\infty^3(c - x_0)Re_{x-x_0}^{-1/5} \tag{19}$$

The only requirement for the estimation of the wake drag is therefore the transition location from which x_0 can be deduced by using Eq. (18). To relate the transition region with the suction flow rate and the mean flow condition, the computer code developed by Hackenberg et al.[12,13] was used together with the e^n method. The plate used in this simulation was chosen to be 1 m long × 0.5 m wide with a panel allowing suction from 0.1 to 0.2 m from the leading edge of the plate. The perforated surface assumed for the suction panel has the characteristics of that studied by Bieler and Preist.[10] Figure 1 shows the results of the computation, for a mean flow speed of 50 ms^{-1} and an amplification factor of 7. This calculation clearly illustrates the significant delay of the transition onset that can be achieved with relatively low suction flow rates. Figure 1 also shows that beyond a certain transition location (in this case $x = 0.58$ m) no significant effect is achieved even for extremely high suction velocities.

Effect of Suction on the Power Balance

To illustrate the evolution of the overall power balance derived previously, the transition prediction code was applied to three mean flow speeds of 30, 40, and 50 ms^{-1}, with amplification factors $n = 5$, 6, and 7, respectively. As far as the characteristics of the suction system are concerned, the two efficiency coefficients are assumed equal to 0.9, the static pressures at the entry and exit of the suction system are also assumed to be equal, and the air is assumed to be at normal conditions, i.e., $\rho = 1.19$ kg m^{-3}, $\nu = 1.53 \times 10^{-5}$. The four contributors to the overall power and their sum according to Eq. (1) are plotted in Fig. 2 vs transition location, for $U_\infty = 50$ ms^{-1} and $n = 7$. From this figure, it can first be concluded that the main contributors to the overall power are those associated with the wake drag and with the pump. It must, however, be emphasized that even if the powers associated with the thrust and momentum loss are equal, and therefore, cancel each other, they are not exactly negligible if compared individually to the other two. It can also be noticed that the powers due to the pump, the momentum loss and the thrust, vary nonlinearly with the increase in transition delay, whereas the wake drag decreases almost linearly with increasing transition position. The curve corresponding to the evolution of the total drag shows the advantage in using a small amount of suction, the severe penalty in using suction rates that are too high, and the existence of an optimum transition location $x = 0.6$ m.

The same curves can be obtained for different mean flow speeds and amplification factors. Figure 3 shows the behavior of the total power balance for five different flows: 1) $U_\infty = 50$ ms^{-1}, $n = 5$; 2) $U_\infty = 50$ ms^{-1}, $n = 6$; 3) $U_\infty = 50$ ms^{-1}, $n = 7$; 4) $U_\infty = 40$ ms^{-1}, $n = 5$; and 5) $U_\infty = 30$ ms^{-1}, $n = 5$. These curves show the totally different power balance that is achieved depending on the speed and the amplification factor that is related to the mean flow turbulence level. The power balances represented by curves 1, 2, and 3 were computed assuming the same mean flow speed of 50 ms^{-1} and different amplification factors of 5, 6, and 7, respectively. The increase of the amplification factor that is correlated to a decrease of the turbulence level allows an optimum transition location further downstream with lower power penalties. Assuming a fixed turbulence level and a variation of the mean flow speed as shown by curves 1, 4, and 5 leads to a similar conclusion: decreasing the mean flow speed allows the further postponement of the transition region with a more advantageous power balance.

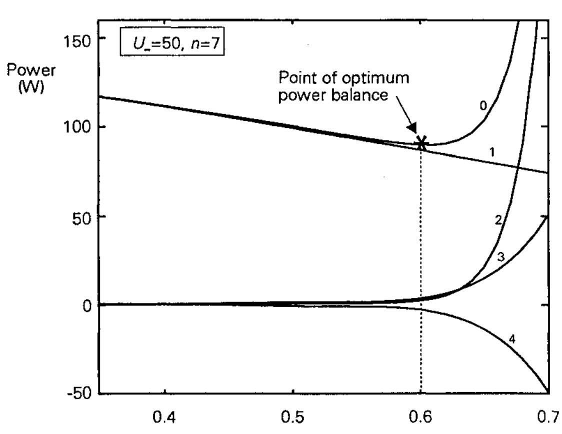

Fig. 2 Detailed power balance for the flat plate with suction. Power related to, 0, total drag; 1, wake drag; 2, pump; 3, momentum loss; and 4, thrust.

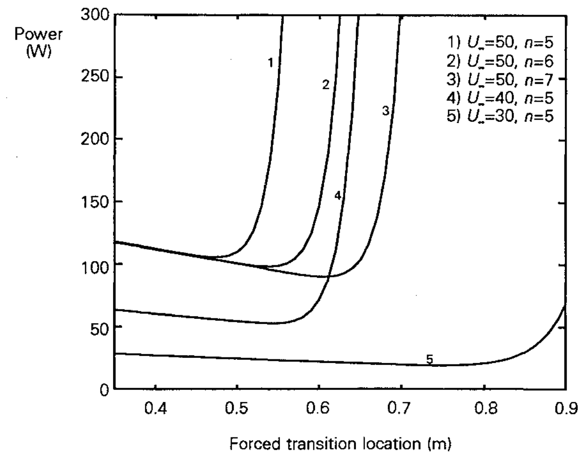

Fig. 3 Total power balance for various mean flow speeds and amplification factors.

Conclusions

Even though the derivation of the overall power balance presented in this work requires several approximations, it demonstrates that suction not only reduces the wake drag, but more importantly, provides an aerodynamically more efficient system. Up to an optimum amount of suction that should not be superseded, it has been shown that the power penalties induced by the suction system are much lower than the power equivalent to the reduction in wake drag. It has also been shown, however, that the optimal position of transition associated with the minimum power consumption depends not only on the mean flow speed, but also on the disturbance environment that is characterized by the amplification factor.

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Jump Phenomena in Y-Shaped Intake Ducts

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Nomenclature

A = duct area
f, g = functions
M = Mach number
m = mass flow rate
P, p = total and static pressure, respectively
R = gas constant
S = sections of duct
T = total temperature
γ = ratio of specific heats
μ = mass flow factor

Subscripts
C = compressor face
E = entry station
l, u = lower, upper branch of solution
M = mixing station
1, 2, 3 = duct section 1, 2, 3

Introduction

THE air intake of a fighter aircraft must meet the engine mass flow demand over a range of aircraft speeds and attitudes[1] with high total pressure recovery and low distortion. Y-shaped ducts are a popular choice for air intakes in single-engined fighter aircraft. The intakes are normally side-mounted and the two limbs of the duct merge inside the fuselage into one and feed the engine. Y-shaped ducts are normally expected to operate in a steady, symmetric manner. In this case, the engine mass flow demand is met by the two limbs by inducing equal mass flows, each being half of what the engine requires.

Steady, asymmetric operation where the two limbs induce unequal mass flows, though not immediately obvious, can never be ruled out. Martin and Holzhauser[2] reported such an operation as early as 1950. The flow in this case, even if smooth in the individual ducts, can be expected to be highly distorted on mixing. The available duct length within the fuselage is very likely to be insufficient to smooth out the distortion before the flow reaches the engine face. This Note proposes a simple flow model that explains the phenomenon that causes transition from symmetric to asymmetric operation.

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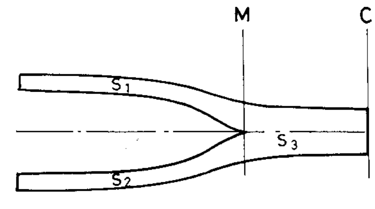

Fig. 1 Y-shaped duct, sections S_1, S_2, and S_3, and stations E, M, and C.

Flow Model

The duct is made up of three sections (Fig. 1). The first two sections, S_1 and S_2, are the two symmetric limbs up to the station where they merge. Section S_3 is the region where the two flows mix and does not play a major role in the analysis that follows. Stations E, M, and C (Fig. 1) are the entry, the merging plane, and the compressor face, respectively. Each duct has a performance characteristic in the form of total pressure ratio across it against mass flow through it. Sections S_1 and S_2 can have an asymmetric operation if, for a given total entry pressure P_E, the mixing plane static pressure p_M is the same for both sections.

For S_1 (or S_2) one can estimate p_M/P_E for any given mass flow rate m_1 (or m_2) as follows: The total pressure ratio is known from the duct characteristic while the total temperature remains unchanged throughout the duct:

$$P_M/P_E = f(m), \qquad T_M = T_E \tag{1}$$

For a given P_E and T_E, and for A_M known from the duct geometry, one can use Eq. (1) to calculate the mass flow factor $\mu = m\sqrt{T_M}/(A_M P_M)$. Mach number M_M is solved for using the compressible flow relation:

$$\sqrt{\gamma/R}M_M = \mu\{1 + [(\gamma - 1)/2]M_M^2\}^{(\gamma+1)/2(\gamma-1)} \tag{2}$$

Thus, we can get

$$\begin{aligned} p_M/P_E &= (p_M/P_M)(P_M/P_E) \\ &= \{1 + [(\gamma - 1)/2]M_M^2\}^{-\gamma/(\gamma-1)}(P_M/P_E) = g(m) \end{aligned} \tag{3}$$

Consider an engine mass flow rate demand of m. This can be met by m_1 and m_2 through S_1 and S_2 as follows:

$$m = m_1 + m_2, \qquad p_{M_1}/P_E = g_1(m_1), \qquad p_{M_2}/P_E = g_2(m_2) \tag{4}$$

The flows m_1 and m_2 will have to satisfy the compatibility condition that $p_{M_1} = p_{M_2}$, which means the static pressure at the mixing station is the same for both limbs. This leads to the following condition, which is a nonlinear equation in one variable m_1 and a parameter m:

$$g_1(m_1) - g_2(m - m_1) = 0 \tag{5}$$