On the Determination and Combination of Loss Coefficients for Compressible Fluid Flows[1]

H. S. Hillbrath[2]

I would like to compliment the author on a valuable and useful paper. Similar techniques have been used for the determination of loss coefficients in the testing of Saturn S-IC components and the Fanno loss factor has been found to be the best method of data correlation. However, I know of no other adequate treatment generally available which substantiates the validity of this method for cases other than that of a straight pipe.

The author discusses and gives data for the case in which the nozzle in Fig. 2(*a*) or 2(*b*) is choked; however, he does not point out the greatly increased control of the test condition which this affords relative to the unchoked case. Note that for this case the specimen outlet Mach number is determined solely by the nozzle area ratio, and that the Reynolds number is a direct function of nozzle inlet pressure, and is only slightly affected by temperature. If, as is often the case, it is desired to have an approximately uniform distribution of test points over a region of Mach and Reynolds numbers, this method can result in great reductions of test time. The objection may be raised that the nozzle discharge coefficients are not accurately known at this condition; however, this uncertainty is generally less than 1.0 percent which is usually not significant.

An interesting point which the author does not discuss, but that seems to warrant experimental investigation, is the effect of the isentropic exponent and whether results obtained with air, for example, are valid for other gases such as steam or helium.

It may be helpful to those not familiar with the author's previous work to point out that equation (1) is equivalent to the relation:

$$4f\,\frac{L_{\max}}{D} = \frac{1 - M^2}{\gamma M^2} + \frac{\gamma + 1}{2\gamma}\ln\frac{(\gamma + 1)M^2}{2\left(1 + \dfrac{\gamma - 1}{2}M^2\right)}$$

where M is the Mach number.[3]

Author's Closure

Mr. Hillbrath's remarks are appreciated. It is interesting to note that others also find the Fanno-type loss coefficients useful in correlating data. The use of nozzles operating in the choked regime for controlling test conditions is well known. However, an additional unchoked nozzle should be used to meter the flow, to avoid the unnecessary uncertainties which would be introduced by the undetermined choked nozzle discharge coefficients. The question of model testing and similitude is one which must be considered carefully. Professor Kline's book[4] presents a recent approach to this problem. The equation given in the discussion concerning $4fL_{\max}/D$ represents the loss coefficient from a given plane of interest to the critical state, while equation (1) of the paper represents the loss coefficient from a given plane of interest to *any other* plane of interest, directly. Thus, while these two equations are related, they are clearly not identical. Again, the author would like to thank Mr. Hillbrath for his useful comments.

[1] The discussion to the paper by R. P. Benedict, published in the January issue of the JOURNAL OF ENGINEERING FOR POWER, TRANS. ASME, Series A, vol. 88, 1966, p. 67, was received at ASME Headquarters after the paper was published.

[2] The Boeing Company, Launch Systems Branch, New Orleans, La. Assoc. Mem. ASME.

[3] A. H. Shapiro, *The Dynamics and Thermodynamics of Compressible Fluid Flow*, The Ronald Press, New York, N. Y., vol. 1, 1953, p. 167.

[4] S. J. Kline, *Similitude and Approximation Theory*, McGraw-Hill Book Company, Inc., New York, N. Y., 1965.

Cycles for Supplying Steam to Desalting Evaporators of Dual-Purpose Power-Generation Plants[1]

E. D. Howe[2]

This paper presents an interesting collection of possible cycles for combination plants. The figures given in Table 1 would be more significant if the "Water to Power Ratio" involved both commodities on similar bases. It is noted that both commodities may be regarded as given in rates of production, water in gallons per day, and power in MW hr per hr. By dividing the numbers in the last column of Table 1 by 24,000, the result will be in terms of gallons of water per kwhr of electrical energy. Since the demands for water and power are usually expressed in gallons per capita per day and kwhr per day, respectively, the quotient would be gallons of water per kwhr. In urban U.S.A. the demand ratio of these two quantities is about 25 to 30 gallons of water per kwhr. On this basis, the figures in Table 1 vary from a minimum of 1.3 for Cycle F-6 to a maximum of 18 for Cycle F-2. Thus all the cycles proposed would tend to increase electrical capacity at a rate greater than water capacity. This suggests that consideration should be given to the utilization of off-peak power for water production and possibly to distillation systems in which steam heat can be used at two different pressures—a low pressure during periods of peak electrical demand and a higher pressure during off-peak periods. The objective would be to keep the water production essentially constant while the electrical output was changed from full load to a lesser load, say, half-load. If cycles could be developed to accomplish this result, a significant saving in equipment and cost of water could result.

Authors' Closure

The authors wish to thank Professor Howe for his fine comments and suggestions. His suggestions should be considered for certain dual-purpose plant application where considerable flexibility is required between water and electrical production.

However, the added capital expenditure incurred for the variable pressure operation of the water plant should be carefully evaluated.

[1] The discussion of this paper by P. Leung and R. E. Moore, published in the January issue of the JOURNAL OF ENGINEERING FOR POWER, TRANS. ASME, Series A, vol. 88, 1966, p. 22, was received at ASME Headquarters after the paper was published.

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Flow Losses in Abrupt Enlargements and Contractions[1]

J. R. Turner[2]

The discusser would like to call the attention of the authors to the article by Hall and Orme[3] in which the writers have described the compressible flow through a sudden enlargement by a straightforward analysis. By combining continuity, momen-

[1] The discussion to this paper by R. P. Benedict, N. A. Carlucci, and S. D. Swetz, published in the January issue of the JOURNAL OF ENGINEERING FOR POWER, TRANS. ASME, Series A, vol. 88, 1966, p. 73, was received at ASME Headquarters after the paper was published.

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[3] W. B. Hall and E. M. Orme, "Flow of a Compressible Fluid Through a Sudden Enlargement in Pipe," *Proc. of Inst. of Mech. Eng.*, vol. 169, no. 49, 1955.

tum, and the "steady-flow energy" (first law) equation, they found the solution[4]

$$(M_2^*)^2 + \left\{\frac{1}{\phi}\frac{\gamma-1}{\gamma+1}M_1^* - \frac{2\gamma}{\gamma+1}M_1^* - \frac{1}{\phi M_1^*}\right\}M_2^* + 1 = 0$$

$$\frac{p_2}{p_1} = \phi\frac{M_1}{M_2}\sqrt{\left\{\frac{(\gamma-1)M_1^2+2}{(\gamma+1)M_2^2+2}\right\}}$$

where M^* and M are related, by definition,

$$M^2 = \frac{2(M^*)^2}{(\gamma+1)-(\gamma-1)(M^*)^2}$$

In a discussion of the paper,[3] Unterberg rearranged this solution to express the total pressure ratio

$$\frac{p_{t2}}{p_{t1}} = \phi\frac{M_1}{M_2}\left[\frac{(\gamma-1)M_2^2+2}{(\gamma-1)M_1^2+2}\right]^{\frac{1}{2(\gamma-1)}}$$

and from this equation and the Mach number relationships shown above provided working charts, for $\gamma = 1.4$, from which the total pressure ratio may be found if the inlet Mach number M_1 (or the pressure ratio p_1/p_{t1}) and the area ratio ϕ are known.

Hall and Orme verified their analytical results with air tests over a range of area ratios from $0.0292 \leq \phi \leq 0.255$ and inlet Mach number $0 \leq M_1 \leq 1.0$.

Hall and Orme seem to have provided the analysis which the present authors are seeking when they say "we can find no compressible counterparts to the constant-density, abrupt area-change solutions given previously." Further, Hall and Orme's equations show the total pressure ratio to be a function of the fluid, through its dependence on the ratio of specific heats; there is no single quantity which can characterize the flow loss across a given configuration independent of the fluid involved, for compressible flow.

D. G. Wilson[5]

Reliable data on contraction and enlargement losses in compressible flow are needed, and the authors will be fulfilling a service if their data fill this need. The introductory statement to the problem contains some unusual approaches, and the usefulness of this work would be much increased if the authors could clarify the following points.

It should be made clear whether the head loss referred to in the first equation is a loss of total head or a loss of static head. It seems to express the loss of total head across a sudden enlargement when there is no recovery of static pressure; is this the case?

In the second equation, the first term is stated in the nomenclature to be the "external heat transfer, volumetric flow rate." It would seem that this term should, in fact, be the external heat transfer per unit mass of fluid. Likewise the work term should be expressed for unit mass of fluid. There are then three energy terms representing the change in enthalpy, but the last two terms have the dimensions of head.

The third equation introduces a so-called "frictional head, force" (which would seem to be a contradiction in terms) into the energy equation. This equation is not significant unless there are some very specific and restrictive statements about the friction. For friction to appear in the energy equation, it must be in the form of frictional energy, and it must be a frictional energy supplied from outside the system and dissipated within it.

The fourth equation continues this treatment, while in the fifth equation a head loss is equated to a frictional force and then to an energy added to a head.

It is difficult to use the data when they are based on an approach of this type. I should greatly appreciate having these points clarified.

[4] Notation is same as authors', except M is Mach number and ϕ is ratio of areas A_1/A_2.

[5] Technical Director, Northern Research and Engineering Corporation, Cambridge, Mass.

Authors' Closure

We would like to thank Mr. Turner for calling attention to the Hall and Orme reference which is very much to the point. It had escaped our attention (and incidentally, that of our other references). The first equation of this discussion is a valuable contribution to our study, representing as it does a specific adiabatic relation between conditions at planes 1 and 2 of our Fig. 1. However, the total pressure ratio of Unterberg should be recognized as nothing more than a form of continuity which applies equally well for the sudden enlargement as for any other flow passage. It follows directly from equations (35) and (36) of the paper, and holds for any adiabatic process.

Hall and Orme's solution is everywhere within 1 percent of our solution, while their data match our solution as well as they match their own. In addition, while it is certainly true that loss appears to exhibit a fluid dependence through γ, we find this effect to be on the order of 0.2 percent (for γ varying from 1.1 to 1.6, for M_1 varying from 0 to 0.8, and for β varying from 0 to 1). This insignificant fluid effect would be well hidden by experimental uncertainties. Thus, the first conclusion of our paper is inescapable: namely, the total pressure loss across a given abrupt enlargement, for the same inlet conditions (R_1 or M_1), is essentially the same for all fluids. This means that the simplistic constant density relations can be used to predict the total pressure drop across a sudden enlargement regardless of the fluid involved.

Incidentally, there are several serious typographical errors in the discussion equations. The second equation should read

$$\frac{p_2}{p_1} = \phi\frac{M_1}{M_2}\sqrt{\left\{\frac{(\gamma-1)M_1^2+2}{(\gamma-1)M_2^2+2}\right\}}$$

while the fourth equation should read

$$\frac{p_{t2}}{p_{t1}} = \phi\frac{M_1}{M_2}\left[\frac{(\gamma-1)M_2^2+2}{(\gamma-1)M_1^2+2}\right]^{\frac{(\gamma+1)}{2(\gamma-1)}}$$

In reply to Mr. Wilson: The head loss of equation (1) has clearly to do with the total pressure and, of course, there is static pressure recovery, as seen by referring to equations (5) and (16). It is common practice, endorsed by the ASA, to allow one symbol to stand for several variables. Thus, the comma between external heat transfer and volumetric flow rate signifies that Q stands for either depending on the context. All quantities in the energy equation (2) are on a per pound basis, i.e., ft lb/lb = head. Similar comments apply to the "frictional head, force" question, i.e., F is either frictional head or force depending on use. The authors are sorry to have caused this inconvenience.

Formation of SO_3 in a Noncatalytic Combustor[1]

A. B. Hedley[2]

I find this paper very interesting and most heartening as it

[1] By Richard E. Barrett, John D. Hummell, and William T. Reid, published in the JOURNAL OF ENGINEERING FOR POWER, TRANS. ASME, Series A, vol. 88, 1966, pp. 165–172.

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